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Preface to the Twelfth Edition

The Gas Processors Suppliers Association is an organization of companies with specialized knowledge of the supply and service needs of the gas processing and related industries. A major service to them is embodied in the Engineering Data Book, which was first published in 1935. Over 150,000 copies of the Ninth, Tenth, and Eleventh Editions of the book were distributed for use throughout the world by engineers, operating personnel, and students.

The Twelfth Edition of the Engineering Data Book, available in two versions — FPS and SI — is an attempt to assemble, in a single compilation, basic design information together with data and procedures that can be used by field and plant engineers to determine operating and design parameters. It is also intended as an aid to design engineers who, in spite of increasing availability of computer routines and other sophisticated design methods, require a general reference work as a guide to accepted engineering practice for estimating, feasibility studies, preliminary design, and for making on-site operating decisions.

The loose-leaf format of the Data Book permits periodic updating to meet the changing technology of the process industries.

GPSA recognizes that the maintenance of the Data Book is a continuing task. Users' comments and suggestions are welcome. Any such comments should be made in writing to:

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A Brief History of the Engineering Data Book and Sponsoring Organizations

The GPSA Engineering Data Book was first published in 1935 as a booklet containing much advertising and a little technical information. In subsequent editions, technical information was expanded and the Data Book gradually became the accepted engineering reference work for the gas processing industry. In addition, the Data Book has found wide acceptance in the petroleum refining, gas transmission, and petrochemical industries.

The Gas Processors Suppliers Association (GPSA) was organized in 1928 as the Natural Gasoline Supply Men's Association (NGSMA). Its principal purpose was as a service organization to the parent Natural Gasoline Association of America (NGAA).

Both organizations underwent name changes in subsequent years in response to changing industry conditions. In 1961, the organizations became known as the Natural Gas Processors Association (NGPA) and the Natural Gas Processors Suppliers Association (NGPSA). In 1974 the names changed to the current Gas Processors Association (GPA) and Gas Processors Suppliers Association (GPSA).

Users of the manual should note that numerous references throughout the book may refer to publications of these organizations by the names in effect at the time of the publication.

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SECTION 1

General Information

This section includes a brief description of the technical activities of the Gas Processors Association (GPA) and the Gas Processors Suppliers Association (GPSA). It includes a compilation of current GPA standards, publications, and Research Reports, a glossary of terms used in the gas processing industry and several tables of common conversion factors and equivalents. At the end of the Section is a listing of other technical associations and societies that publish codes, standards and other information pertinent to the natural gas processing industry.

GPA TECHNICAL ACTIVITIES

The Gas Processors Association is an international organization with a membership of about 100 companies, each of which is engaged in one or more functions of the natural gas and gas processing industry, including: natural gas production, gathering and processing; production of natural gas liquids (NGLs); volume transport or further processing of natural gas or gas liquids; or other commerce in natural gas or gas liquids. Its principal functions include the advancement of gas processing technology through development and promulgation of international standards for NGL product specifications, test methods, measurement, and handling. In addition, GPA sponsors basic research in thermodynamic and transport properties of natural gases, gas liquids, synthetic gases, and related process stream components. The dissemination of GPA technology is accomplished through publication of industry standards, research reports, and computer programs. This technology is developed in response to industry needs by a technical committee organization with the following structure and responsibilities:

Technical Committee — The parent body responsible for assessing the technological needs of the gas processing industry, establishing priorities, assigning problems to appropriate working sections, and reviewing results and recommendations of sections as needed.

The seven working sections of the Technical Committee are:

Section A, Facilities Design and Optimization — Maintains a continuing study of all phases of gas processing design (process, mechanical, electrical, instrumentation) of broad industry interest; this responsibility does not include analytical methods nor basic data on product properties, but does include onstream process and plant control instrumentation.

Section B, Analysis — Studies and develops analytical procedures and methods to improve accuracy and reliability of test methods and data to meet the needs of the gas processing industry. All of the industry's sampling procedures, specification test methods, and analytical standards have issued from this section.

Section C, Specifications — Maintains existing product specifications, and generates new and revised specifications in response to changing industry and market needs. All U.S. and many international specifications for LP-gas and other NGL products are based on the work of this section.

Section F, Technical Data Development — Monitors the status, availability, and reliability of thermodynamic and physical property data needed for design and operation of natural gas and gas processing facilities. This section identifies industry needs for pertinent data, develops research projects to provide needed data, and supervises research to meet those needs. The section also evaluates new correlations and computer models for calculation and application of thermodynamic and physical properties applicable to natural gas processing and transport.

Primary responsibility for initiation and supervision of thermodynamic and physical properties research is vested in two steering committees comprised of highly qualified and experienced specialists.

Section H, Product Measurement and Handling — Monitors measurement data and correlations for both natural gas and natural gas liquids, and reviews handling procedures for gas liquids, including underground storage technology. The section initiates improvements in standards for accurate measurement and safe handling of both natural gas and gas liquids.

Section L, Computer Technology and Data Distribution — This Section is focused on two primary areas: 1) the continued development of the GPA Home Page on the Internet and; 2) the pursuit of data in support of the association.

Section M, Operations and Maintenance — This Section was formed to assist member companies in addressing and resolving their operations and maintenance issues, and to be a forum where such issues on gas gathering and processing can be discussed. It is anticipated that the results of this committee can supplement the programs of the GPA Regional Meetings.

Section N, LNG Committee — This is a new committee that is being established with the following goals related to the LNG industry:

1. Identify and collect relevant standards and reports
2. Recommend and steward technical and research activities
3. Develop documents: best practices, guidelines, specifications, etc.
4. Interface with associations with overlapping functions
5. Represent membership to agencies, government, etc.
6. Improve public awareness of LNG safety and security
7. Provide forums for discussing LNG issues and technologies

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FIG. 1-1

Typical Components of Industry Streams

	Components											
	CO_2	H_2S	N_2	C_1	C_2	C_3	iC_4	nC_4	iC_5	nC_5	C_6	C_{7+}
Inert Gas	•		•									
Acid Gas	•	•										
LNG			•	•	•	•	•	•				
Natural Gas	•	•	•	•	•	•	•	•	•	•	•	•
LPG					•	•	•	•				
Natural Gasoline						•	•	•	•	•	•	•
NGL					•	•	•	•	•	•	•	•
Condensate (Stabilized)							•	•	•	•	•	•

GPSA TECHNICAL ACTIVITIES

The Gas Processors Suppliers Association is an affiliated organization of approximately 350 companies that cater to the supply and service needs of the natural gas producing and processing industry. The principal technical function of GPSA is to publish GPA and other technical information through the GPSA Engineering Data Book. In addition, technical experts from GPSA companies participate in GPA technical committee activities described above, and also provide valuable support and counsel in directing thermodynamic research and technical data development for the advancement of the industry's technology.

Editorial Review Board — The Editorial Review Board for the Engineering Data Book is the single GPSA technical committee. The Board is composed of recognized experts from both GPA and GPSA member companies, and is responsible for the continuing review and revision of the Engineering Data Book to assure that it meets the needs of the natural gas processing industry with the latest data and technology.

Definitions of Words and Terms Used in the Gas Processing Industry

absorber
A tower or column that provides contact between natural gas being processed and a liquid solvent.

absorption
The operation in which one or more components in the gas phase are transferred to (absorbed into) a liquid solvent.

absorption factor
A factor which is an indication of the tendency for a given gas phase component to be transferred to the liquid solvent. It is generally expressed as A = L/KV where L and V are the molar flows of liquid and vapor, and K is the average value of the vapor-liquid equilibrium constant for the component of concern.

absorption oil
A hydrocarbon liquid used to absorb and recover components from the natural gas being processed.

acid gas
The hydrogen sulfide and/or carbon dioxide contained in, or extracted from, gas or other streams.

adiabatic expansion
The expansion of a gas, vapor, or liquid stream from a higher pressure to a lower pressure in which there is no heat transfer between the gas, vapor, or liquid and the surroundings.

adsorbent
A solid substance used to remove components from natural gas being processed.

adsorption
The process by which gaseous components are adsorbed on solids because of their molecular attraction to the solid surface.

amine (alkanolamine)
Any of several liquid compounds containing amino nitrogen generally used in water solution to remove, by reversible chemical reaction, hydrogen sulfide and/or carbon dioxide from gas and liquid hydrocarbon streams.

API Gravity
An arbitrary scale expressing the relative density of liquid petroleum products. The scale is calibrated in degrees API, calculated by the following formula:

$$\text{Deg API} = \left[\frac{141.5}{\gamma\ (60°\text{F}/60°\text{F})}\right] - 131.5$$

γ = relative density

associated gas
Gaseous hydrocarbons occuring as a free-gas phase under original oil-reservoir conditions of temperature and pressure.

atmospheric pressure
The pressure exerted on the earth by the earth's atmosphere. A pressure of 760 mm of mercury, 29.92 inches of mercury, or 14.696 psia is used as a standard for some measurements. State regulatory bodies have set other standards for use in measuring the legal volume of gas. Atmospheric pressure may also refer to the absolute ambient pressure at any given location.

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barrel
A common English-unit mesure of liquid volume which, in the petroleum industry, equals 42 U.S. liquid gallons for petroleum or natural gas liquid products measured at 60°F and equilibrium vapor pressure. One barrel equals 0.159 cubic meters, or 6.29 barrels per cubic meter (See Fig. 1-2).

blanket gas
A gas phase maintained in a vessel containing liquid to protect the liquid against air contamination, to reduce the hazard of explosion, or to maintain pressure of the liquid. The source of the gas is external to the vessel.

blow case
A small vessel in which liquid is accumulated and then forced from the vessel by applying gas or air pressure above the liquid level.

blowdown
The act of emptying or depressuring a vessel. This may also refer to discarded material, such as blowdown water from a boiler or cooling tower.

boilaway test
Sometimes used to describe the GPA weathering test for LP-gas. Refer to definition for "weathering test".

bottoms
The liquid or residual matter which is withdrawn from the bottom of a fractionator or other vessel during processing or while in storage.

B-P mix
A liquefied hydrocarbon product composed chiefly of butanes and propane. If it originates in a refinery, it may also contain butylenes and propylene. More specifically, it conforms to the GPA specifications for commercial B-P mixes as described in GPA Standard 2140.

breathing
The movement of vapor in or out of an atmospheric pressure storage tank because of a change of level of the stored liquid, a change in the temperature of the vapor space above the liquid, or a change of atmospheric pressure.

bs&w (basic sediment and water)
Waste that collects in the bottom of vessels and tanks containing petroleum or petroleum products.

bubble point
The temperature at a specified pressure at which the first stable vapor forms above a liquid.

butane, commercial
A liquefied hydrocarbon consisting predominately of butane and/or butylene and which conforms to the GPA specification for commercial butane defined in GPA Standard 2140.

butane, normal
In commercial transactions, a product meeting the GPA specifications for commercial butane and, in addition, containing a minimum of 95 liquid volume percent normal butane. Chemically, normal butane is an aliphatic compound of the paraffin series having the chemical formula C_4H_{10} and having all of its carbon atoms joined in a straight chain.

calorimeter
An apparatus which is used to determine the heating value of a combustible material.

carbonyl sulfide
A chemical compound of the aldehyde group containing a carbonyl group and sulfur (COS). Sometimes a contaminant in natural gas and NGL. It may need to be removed in order to meet sulfur specifications.

casinghead gas
Unprocessed natural gas produced from a reservoir containing oil. It contains heavier hydrocarbon vapors and is usually produced under low pressure from a casing head on the well.

charcoal test
A test standardized by the Gas Processors Association and the American Gas Association for determining the natural gasoline content of a given natural gas. The gasoline is adsorbed from the gas on activated charcoal and then recovered by distillation. The test is prescribed in Testing Code 101-43, a joint publication of AGA and GPA.

chromatography
A technique for separating a mixture into individual components by repeated adsorption and desorption on a confined solid bed. It is used for analysis of natural gas and NGL.

Claus Process
A process to convert hydrogen sulfide into elemental sulfur by selective oxidation.

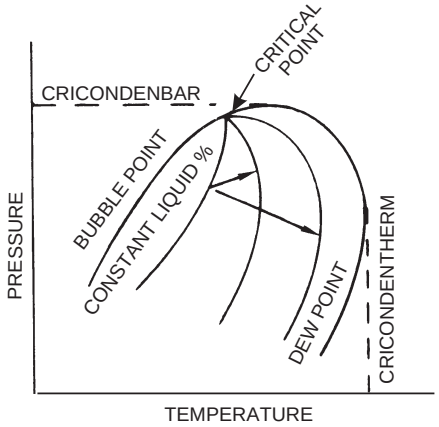
compressibility factor
A factor, usually expressed as "Z," which gives the ratio of the actual volume of gas at a given temperature and pressure to the volume of gas when calculated by the ideal gas law.

compression ratio
The ratio of the absolute discharge pressure from a compressor to the absolute intake pressure. Also applies to one cylinder of a reciprocating compressor and one or more stages of a rotating compressor.

condensate
The liquid formed by the condensation of a vapor or gas; specifically, the hydrocarbon liquid separated from natural gas because of changes in temperature and pressure when the gas from the reservoir was delivered to the surface separators. In a steam system it may be water that is condensed and returned to the boilers.

convergence pressure
The pressure at a given temperature for a hydrocarbon system of fixed composition at which the vapor-liquid equilibrium K-values of the various components in the system become, or tend to become, unity. The convergence pressure is used to adjust vapor-liquid equilibrium K-values to the particular system under consideration. (See TP-22)

copper strip test
A test using a small strip of pure copper to determine qualitatively the hydrogen sulfide corrosivity of a product. Refer to GPA LP-gas copper strip test (Copper Strip Method), ASTM D-1838 test procedure.



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cricondenbar
The highest pressure at which liquid and vapor phases can exist at equilibrium in a multicomponent system.

cricondentherm
The highest temperature at which liquid and vapor phases can exist at equilibrium in a multicomponent system.

critical density
The density of a substance at its critical temperature and critical pressure.

critical pressure
The vapor pressure of a substance at its critical temperature.

critical temperature
For a pure component, the maximum temperature at which the component can exist as a liquid.

cryogenic plant
A gas processing plant which is capable of producing natural gas liquid products, including ethane, at very low operating temperatures, usually below minus 50°F.

cubic meter
A unit of volume measurement commonly used in international commerce for petroleum, petroleum products and natural gas. One cubic meter measured at 60°F = 264.172 U.S. gallons = 6.29 barrels = 35.315 cubic feet measured at 60°F.

deaerator
An item of equipment used for removing air or other non-condensible gases from a process stream or from steam condensate or boiler feed water.

debutanizer
A fractionator designed to separate butane (and more volatile components if present) from a hydrocarbon mixture.

dehydration
The act or process of removing water from gases or liquids.

demethanized product
A product from which essentially all methane and lighter materials have been removed.

demethanizer
A fractionator designed to separate methane (and more volatile components if present) from a hydrocarbon mixture.

depropanizer
A fractionator designed to separate propane (and more volatile components if present) from a hydrocarbon mixture.

desiccant
A substance used in a dehydrator to remove water and moisture. Also a material used to remove moisture from the air.

desulfurization
A process by which sulfur and sulfur compounds are removed from gases or liquid hydrocarbon mixtures.

dew point
The temperature at any given pressure, or the pressure at any given temperature, at which liquid initially condenses from a gas or vapor. It is specifically applied to the temperature at which water vapor starts to condense from a gas mixture (water dew point), or at which hydrocarbons start to condense (hydrocarbon dew point).

distillation
The process of separating materials by successively heating to vaporize a portion and then cooling to liquefy a part of the vapor. Materials to be separated must differ in boiling point and/or relative volatility.

doctor test
A qualitative method for detecting hydrogen sulfide and mercaptans in NGL. The test distinguishes between "sour" and "sweet" products.

dry gas
(1) Gas whose water content has been reduced by a dehydration process. (2) Gas containing little or no hydrocarbons commercially recoverable as liquid product. Gas in this second definition preferably should be called lean gas.

end point
The maximum temperature observed on the thermometer during an ASTM distillation test.

EP-mix (ethane-propane mix)
A product consisting of a mixture of essentially ethane and propane.

expansion turbine
A device which converts part of the energy content of a gas or liquid stream into mechanical work by expanding the gas or liquid through a turbine from which work is extracted.

extraction
The process of transferring one or more components from one liquid phase to another by virtue of different solubility in the two liquids. It is also used to indicate removal of one or more constituents from a stream.

field separator
A vessel in the oil or gas field for separating gas, hydrocarbon liquid, and water from each other.

flash point
The lowest temperature at which vapors from a hydrocarbon liquid will ignite. See ASTM D-56.

fractionation
See definition of "distillation." Generally used to describe separation of a mixture of hydrocarbons into individual products based on difference in boiling point and/or relative volatility.

freeze valve
A specially constructed and calibrated valve designed and used solely for determining the water content in propane product. See ASTM D-2713.

gas constant (R)
The constant multiplier in the Ideal Gas Law. Numerically, R=PV/T, if V is the volume of one mole of an ideal gas at temperature T and pressure P.

gas hydrate
Refer to definition of "hydrate".

gas injection
The injection of natural gas into a reservoir to maintain or increase the reservoir pressure or reduce the rate of decline of the reservoir pressure.

gas lift
A method for bringing crude oil or water to the surface by injecting gas into the producing well bore.

gas-oil ratio (GOR)
The ratio of gas to liquid hydrocarbon produced from a well. This may be expressed as standard cubic feet of gas per barrel of stock tank liquid.

gas processing
The separation of constituents from natural gas for the purpose of making salable products and also for treating the residue gas to meet required specifications.

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gas processing plant
A plant which processes natural gas for recovery of natural gas liquids and sometimes other substances such as sulfur.

gas-well gas
The gas produced or separated at surface conditions from the full well stream produced from a gas reservoir.

gas-well liquids
The liquid separated at surface conditions from the full well stream produced from a gas reservoir.

gathering system
The network of pipelines which carry gas from the wells to the processing plant or other separation equipment.

gpm/GPM
(1) gpm (gallons per minute): The term used to describe the rate of flowing fluid in gallons per minute. (2) GPM — Preferably Gal/Mcf (gallons per thousand cubic feet): This term refers to the content in natural gas of components which are recoverable or recovered as liquid products.

heat medium (heating medium)
A material, whether flowing or static, used to transport heat from a primary source such as combustion of fuel to another material. Heating oil, steam, and an eutectic salt mixture are examples of heat mediums.

heating value (heat of combustion)
The amount of heat obtained by the complete combustion of a unit quantity of material. The gross, or higher, heating value is the amount of heat obtained when the water produced in the combustion is condensed. The net, or lower, heating value is the amount of heat obtained when the water produced in the combustion is not condensed.

heavy ends
The portion of a hydrocarbon mixture having the highest boiling point. Usually hexanes or heptanes and all heavier hydrocarbons are the heavy ends in a natural gas stream.

hexanes plus (or heptanes plus)
The portion of a hydrocarbon fluid mixture or the last component of a hydrocarbon analysis which contains the hexanes (or heptanes) and all hydrocarbons heavier than the hexanes (or heptanes).

hydrate
A solid material resulting from the combination of a hydrocarbon with water under pressure.

immiscible
Liquids that will not mix nor blend to give homogeneity are said to be immiscible.

ideal gas (also called "perfect" gas)
A gas that obeys the ideal gas law expressed as PV=RT, see Fig. 1-4.

inerts
Elements or compounds not acted upon chemically by the surrounding environment. Nitrogen and helium are examples of inert constituents of natural gases.

isobutane
In commercial transactions, a product meeting the GPA specification for commercial butane and, in addition, containing a minimum of 95 liquid volume percent isobutane. Chemically, a hydrocarbon of the paraffin series with the formula C_4H_{10} and having its carbon atoms branched.

jacket water
Water which fills, or is circulated through, a casing which partially or wholly surrounds a vessel or machine element in order to remove, add, or distribute heat in order to control the temperature within the vessel or element.

Joule-Thomson effect
The change in gas temperature which occurs when the gas is expanded at constant enthalpy from a higher pressure to a lower pressure. The effect for most gases at normal pressure, except hydrogen and helium, is a cooling of the gas.

lead acetate test
A method for detecting the presence of hydrogen sulfide by discoloration of paper which has been moistened with lead acetate solution. See ASTM D-2420.

lean gas
(1) The residue gas remaining after recovery of natural gas liquids in a gas processing plant. (2) Unprocessed gas containing little or no recoverable natural gas liquids.

lean oil
Absorption oil as purchased or recovered by the plant, or oil from which the absorbed constituents have been removed.

lift gas
Gas used in a gas lift operation.

light ends
The low-boiling, easily evaporated components of a hydrocarbon liquid mixture.

light hydrocarbons
The low molecular weight hydrocarbons such as methane, ethane, propane and butanes.

LNG (liquefied natural gas)
The light hydrocarbon portion of natural gas, predominately methane, which has been liquefied.

loading rack
A structural and piping installation alongside a railroad track or roadway used for the purpose of filling railroad tank cars or transport trucks.

LPG (liquefied petroleum gas)
Refer to definition of "LP-gas".

LP-gas (liquefied petroleum gas)
Predominately propane or butane, either separately or in mixtures, which is maintained in a liquid state under pressure within the confining vessel.

LRG (liquefied refinery gas)
Liquid propane or butane produced by a crude oil refinery. It may differ from LP-gas in that propylene and butylene may be present.

LTX (low temperature extraction unit)
A unit which uses the cooling of a constant enthalpy expansion to increase liquid recovery from streams produced from high pressure gas condensate reservoirs. Also called LTS (low temperature separation) unit.

Mcf
An abbreviation for one thousand cubic feet of gas.

MMcf
An abbreviation for one million cubic feet of gas.

mercaptan
Any of a homologous series of compounds of the general formula RSH. All mercaptans possess a foul odor.

miscible flood
A method of secondary recovery of fluids from a reservoir by injection of fluids that are miscible with the reservoir fluids.

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natural gas
Gaseous form of petroleum. Consisting predominately of mixtures of hydrocarbon gases. The most common component is methane.

natural gasoline
A mixture of hydrocarbons, mostly pentanes and heavier, extracted from natural gas, which meets vapor pressure, end point, and other specifications for natural gasoline as adopted by the GPA. See GPA Standard 3132.

natural gas processing plant
Term used for gas processing plant, natural gasoline plant, gasoline plant, etc.

NGL (natural gas liquids)
Natural gas liquids are those hydrocarbons liquefied at the surface in field facilities or in gas processing plants. Natural gas liquids include ethane, propane, butanes, and natural gasoline.

odorant
An odoriferous compound added to natural or LP-gas to impart a distinctive odor for detection of fugitive vapors. Ethyl mercaptan is the most widely used odorant for LP-gas, while tertiary butyl mercaptan, usually mixed with small amounts of other compounds, is the predominant odorant for natural gas.

oil-well gas
Gas that is produced from an oil well.

on-stream factor
The percentage of time a unit is on-stream.

operating factor
The percentage of time a unit is performing the function for which it was designed.

outage
The vapor volume in a liquid vessel left for liquid expansion. Sometimes referred to as ullage.

packaged unit
A shop-assembled group of equipment and accessories which needs only foundations, inlet and outlet piping, and utility connections to make an operating unit.

packed column
A fractionation or absorption column filled with packing designed to give the required contact between the rising vapors and the descending liquid.

peak shaving
The use of non-conventional fuels to supplement the normal supply of pipeline gas during periods of extremely high demand.

pentane-plus
A hydrocarbon mixture consisting of isopentane (C_5H_{12}) and heavier components with higher boiling points.

pigging
A procedure for forcing a device through a pipeline for cleaning purposes, separating products, or inspecting the line.

pipeline gas
Gas which meets a transmission company's minimum specifications.

propane
A normally gaseous paraffinic compound (C_3H_8). The term includes all products covered by GPA specifications for commercial and HD-5 propane. See GPA Standard 2140.

propane, commercial
A liquefied hydrocarbon product consisting predominately of propane and/or propylene and which conforms to the GPA specification for commercial propane as defined in GPA Standard 2140.

propane HD-5
A special grade of propane consisting predominately of propane and which conforms to the GPA specification for HD-5 propane as defined in GPA Standard 2140.

raw gas
Unprocessed gas, or the inlet gas to a gas processing plant.

raw mix liquids
A mixture of natural gas liquids prior to fractionation. Also called "raw make".

real gas
A gas that does not obey the ideal gas law. Instead its behavior is expressed as PV = zRT where z is the compressibility factor and $z \neq 1.0$. For an ideal gas z = 1.0, see Fig. 1-4.

recovery
That percent or fraction of a given component in the plant feed which is recovered as plant product.

recycle
Return of part of a process stream to a point upstream from where it was removed to enhance recovery or control.

reflux
In fractionation, the portion of condensed overhead returned to the column to enhance achievable purity of the overhead product.

reflux ratio
A way of giving a relative measurement to the volume of reflux. Usually referred either to the feed or overhead product.

relative density (See specific gravity)

relief system
The system for safely relieving excess pressure to avoid exceeding equipment design pressure.

residue
The material which remains after a separation process. (1) Residue gas is that gas remaining after the recovery of liquid products. (2) Residue may also be the heaviest liquid or solid remaining after distillation or reclaiming process.

retrograde condensation (vaporization)
Condensation or vaporization that is the reverse of expected behavior. Condensation caused by a decrease in pressure or an increase in temperature. Vaporization caused by an increase in pressure or a decrease in temperature.

rich gas
Gas feed to a gas processing plant for liquid recovery.

rich oil
The oil leaving the bottom of an absorber. It is the lean oil plus the absorbed constituents.

RVP (Reid Vapor Pressure)
The vapor pressure of a material measured by the Reid Method and apparatus as detailed in ASTM Test Procedure D-323.

s & w (See bs&w)

saturated compounds
Hydrocarbon compounds having no unsaturated carbon valence bonds. Natural gas and natural gas liquids are mixtures of saturated compounds.

saturated liquid
Liquid which is at its boiling point or is in equilibrium with a vapor phase in its containing vessel.

saturated vapor
Vapor at its dew point.

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scf (standard cubic feet)
Gas volume at standard temperature and pressure.

shrinkage
The reduction in volume of a gas stream by removal of some of its constituents such as for recovered products, fuel, or losses.

SNG (Synthetic or Substitute Natural Gas)
The gas product resulting from the gasification of coal and/or gas liquids or heavier hydrocarbons.

solution gas
Gas which originates from the liquid phase in an oil reservoir.

sour
Liquids and gases are said to be "sour" if they contain hydrogen sulfide, carbon dioxide, and/or mercaptans above a specified level. It also is used to refer to the feed stream to a sweetening unit.

sour gas
Gas containing undesirable quantities of hydrogen sulfide, mercaptans, and/or carbon dioxide. It also is used to refer to the feed stream to a sweetening unit.

specific gravity
The ratio of the mass of a given volume of a substance to that of another equal volume of another substance used as standard. Unless otherwise stated, air is used as the standard for gases and water for liquids, with the volumes measured at 60°F and standard atmospheric pressure.

splitter
A name applied to fractionators, particularly those separating isomers (e.g., butane splitter refers to a tower producing most of the isobutane in the feed as overhead and most of the normal butane in the feed as bottoms).

sponge absorbent
An absorbent for recovering vapors of a lighter absorbent that is used in the main absorption process of a gas processing plant.

stabilized condensate
Condensate that has been stabilized to a definite vapor pressure in a fractionation system.

stabilizer
A fractionation column designed to reduce the vapor pressure of a liquid stream.

stage separation system
A system of separators where the liquid portion of the well effluent is separated from formation gas and flash vapors.

still
The column where the absorbed product is recovered from the lean absorption oil. In plants using a low molecular weight absorption oil, the still is designed as a fractionation column. In plants using a high molecular weight absorption oil, the still may use steam or other fluids as stripping medium. Also used to refer to regenerators in amine treating and glycol dehydration systems.

strapping
A term applied to the process of calibrating liquid storage capacity of storage tanks in increments of depth.

stream day
A continuous 24 hour period of plant operation.

stripper
A column wherein absorbed constituents are stripped from the absorption oil. The term is applicable to columns using a stripping medium, such as steam or gas.

stripping factor
An expression used to describe the degree of stripping. Mathematically, it is KV/L, the reciprocal of the absorption factor.

stripping medium
As stated under "stripper", the medium may be steam, gas, or other material that will increase the driving force for stripping.

sulfur
A yellow, non-metallic chemical element. In its elemental state, it exists in both crystalline and amorphous forms. In many gas streams, sulfur may be found as volatile sulfur compounds, such as hydrogen sulfide, sulfur oxides, mercaptans, and carbonyl sulfide. Reduction of the concentration of these gaseous sulfur compounds is often necessary for corrosion control and possibly for health and safety reasons.

sulfur dioxide (SO_2)
A heavy, colorless, suffocating gas that is chemically an oxide of sulfur. Conversion of the gaseous sulfur oxides to sulfur is necessary for corrosion control, for health and safety reasons, and for complying with governmental standards.

sweet
Gas containing essentially no objectionable sulfur compounds. Also, treated gas leaving a sweetening unit.

sweet gas
Gas which has no more than the maximum sulfur and/or CO_2 content defined by (1) the specifications for the sales gas from a plant; (2) the definition by a legal body. Also, the treated gas leaving a sweetening unit.

temperature correction factor
A factor for correcting volume at a given temperature to that at a specific reference temperature. Reference temperature most commonly used in the petroleum industry is 60°F.

therm
A unit of gross heating value equivalent to 100,000 Btu.

tonne
A unit of mass measurement, commonly used in international petroleum commerce; an expression for the metric ton, or 1000 kilograms.

trayed column
A vessel wherein gas and liquid, or two essentially immiscible liquids, are contacted, usually counter-currently on trays. Also refer to packed column.

turboexpander
Refer to definition of "expansion turbine."

ullage (See outage)

unsaturated compounds
Hydrocarbon compounds having one or more unsaturated valence bonds, i.e., ethylene, propylene. These compounds are not found in natural gas streams or gas liquids because of their relatively high chemical reactivity. Unsaturates are produced by a thermal cracking or chemical reaction and can be found in synthetic gas (SNG) or light refinery gases (LRG).

vapor pressure (true vapor pressure)
The pressure exerted by the equilibrium vapor of a liquid when confined in a closed previously evacuated tank or test apparatus.

vapor pressure gasoline
A descriptive phrase for natural gasoline meeting a specified vapor pressure.

vapor pressure, GPA
Vapor pressure as specified by GPA procedures.

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vapor recovery
Equipment or process for the recovery of desired components from stock tank vapors or vapors from some other source.

volatile sulfur
An obsolete term referring to sulfur compounds that will vaporize readily (See sulfur).

weathering
The evaporation of liquid caused by exposing it to the conditions of atmospheric temperature and pressure. Partial evaporation of liquid by use of heat may also be called weathering.

weathering test
A GPA test for LP-gas for the determination of heavy components in a sample by evaporation under specified conditions.

weight in air
Weight compared to a standard with no correction for air buoyancy.

wellhead
The assembly of fittings, valves, and controls located at the surface and connected to the flow lines, tubing, and casing of the well so as to control the flow from the reservoir.

wet gas
(1) A gas containing water, or a gas which has not been dehydrated. (2) A term synonymous with rich gas. Refer to definition of "rich gas".

Wobbe number
A number proportional to the heat input to a burner at constant pressure. In British practice, it is the gross heating value of a gas divided by the square root of its gravity. Widely used in Europe, together with a measured or calculated flame speed, to determine interchangeability of fuel gases.

Conversion Factors

In these tables, factors for conversion, including conversions to the International System of Units (SI), are based on ASTM Standard for Metric Practice, E380-91. The latest edition of this publication should be studied for more detail on the SI system, including definitions and symbols.

In calculating derived factors in the tables that follow, exact conversions were used, when available, rather than the 7-digit round-offs listed in ASTM E380 conversion tables. Derived factors given below are rounded to the same number of significant digits as the source factors.

In any conversion of fundamental measurement units, some confusion may result due to redefinition of units used in earlier tables. For example, in 1959 a small refinement was made in the definition of the yard, which changed its length from 3600/3937 meter (or 1 inch = 25.4000508 mm) to 0.9144 m exactly (or 1 inch = 25.4 mm exactly). The tables below are based on the new definition, but one should be aware that where U.S. land measurements are concerned, the old relationship applies. Refer to ASTM E380-91, note 13, for more detail.

Energy Units Conversion

Confusion may arise in the definition of units for heat or energy. In the tables below, the Btu (IT) and calorie (IT) are used. These are the heat units recommended by the International Conference on the Properties of Steam, as defined:

1 Btu (IT) = 1055.055 852 62 joule (exactly)
1 Calorie (IT) = 4.186 800 joule (exactly)

For information only, other definitions that may be used elsewhere:

1 Btu (Mean) = 1055.87 joule
1 Btu (39°F) = 1059.67 joule
1 Btu (60°F) = 1054.68 joule
1 Btu (Thermochemical) = 1054.350 joule
1 calorie (Mean) = 4.190 02 joule

FIG. 1-2

Conversion Factor Tables

Velocity
(Length/unit of time)

ft/sec	ft/min	Miles/hr (U.S. Statute)	m/sec	m/min	km/hr
1	60	0.6818182	0.3048	18.288	1.09728
0.01666667	1	0.01136364	5.08×10^{-3}	0.3048	0.018288
1.466667	88	1	0.44704	26.8224	1.609344
3.280840	196.8504	2.236936	1	60	3.6
0.05468066	3.280840	0.03728227	0.016667	1	0.06
0.9113444	54.68066	0.6213712	0.2777778	16.66667	1

Energy

Ft-lbf	Kg-meter	Btu (IT)	Kilo-calorie (IT)	Hp-hr	Kilowatt-hr	joule (J)
1	0.1382550	1.285068×10^{-3}	3.238316×10^{-4}	5.050505×10^{-7}	3.766161×10^{-7}	1.355818
7.233014	1	9.294911×10^{-3}	2.342278×10^{-3}	3.653037×10^{-6}	2.724070×10^{-6}	9.806650
778.1692	107.5858	1	0.2519958	3.930148×10^{-4}	2.930711×10^{-4}	1055.056
3088.025	426.9348	3.968321	1	1.559609×10^{-3}	1.163×10^{-3}	4186.8
1980000	273744.8	2544.434	641.1865	1	0.7456999	2684520.
2655224	367097.8	3412.142	859.8452	1.341022	1	3600000.
0.7375621	0.1019716	9.478171×10^{-4}	2.388459×10^{-4}	3.725061×10^{-7}	2.777778×10^{-7}	1

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FIG. 1-2 (Cont'd)

Conversion Factor Tables

Length

Inches	Feet	Yards	Miles (U.S. Statute)	Millimeters	Meters
1	0.08333333	0.02777778	1.578283×10^{-5}	25.4	0.0254
12	1	0.3333333	1.893939×10^{-4}	304.8	0.3048
36	3	1	5.681818×10^{-4}	914.4	0.9144
63360.	5280	1760	1	1609344	1609.344
0.03937008	3.280840×10^{-3}	1.093613×10^{-3}	6.213712×10^{-7}	1	0.001
39.37008	3.280840	1.093613	6.213712×10^{-4}	1000	1

Area

Sq inches	Sq feet	Sq yards	Acres	Sq miles (U.S. Statute)	Sq meters
1	6.944444×10^{-3}	7.716049×10^{-4}	1.594225×10^{-7}	2.490977×10^{-10}	6.4516×10^{-4}
144	1	0.1111111	2.295684×10^{-5}	3.587006×10^{-8}	9.290304×10^{-2}
1296	9	1	2.066116×10^{-4}	3.228306×10^{-7}	0.8361274
6272640.	43560.	4840.	1	0.0015625	4046.856
4014489600	27878400	3097600.	640	1	2589988.
1550.0031	10.76391	1.195990	2.471054×10^{-4}	3.861022×10^{-7}	1

Capacity–volume

Cu inches	Cu feet	Cu yards	Liters	Cu meters	U.S. gallons	Imp. gallons	Barrels (42 U.S. gal)
1	5.787037×10^{-4}	2.143347×10^{-5}	0.01638706	1.638706×10^{-5}	4.329004×10^{-3}	3.604649×10^{-3}	1.030715×10^{-4}
1728	1	0.03703704	28.31685	0.02831685	7.480520	6.228833	0.1781076
46656	27	1	764.5549	0.7645549	201.9740	168.1784	4.808905
61.02374	0.03531467	1.307951×10^{-3}	1	0.001	0.2641720	0.2199692	6.289810×10^{-3}
61023.74	35.31467	1.307951	1000	1	264.1720	219.9692	6.289810
231.0000	0.1336806	4.951132×10^{-3}	3.785412	0.003785412	1	0.8326739	2.380952×10^{-2}
277.4196	0.1605437	5.946064×10^{-3}	4.546092	0.004546092	1.200950	1	0.02859406
9702.001	5.614584	0.2079475	158.9873	0.1589873	42	34.97230	1

Mass

Ounces	Pounds	Short tons	Long tons	Kilograms	Metric tons
1	0.0625	3.125×10^{-5}	2.790179×10^{-5}	0.02834952	2.834950×10^{-5}
16	1	5×10^{-4}	4.464286×10^{-4}	0.4535924	4.535924×10^{-4}
32000	2000	1	0.8928571	907.1847	0.9071847
35840	2240	1.12	1	1016.047	1.016047
35.27396	2.204623	1.102311×10^{-3}	9.842065×10^{-4}	1	0.001
35273.96	2204.623	1.102311	0.9842065	1000	1

Weight per unit of area

Lb/sq ft	Lb/sq in	kg/sq cm	kg/sq m	Short tons/sq ft	Long tons/sq ft	kg/sq mm
1	0.006944444	4.882428×10^{-4}	4.882428	0.0005	4.464286×10^{-4}	4.882428×10^{-6}
144	1	0.07030695	703.0695	0.072	0.06428571	7.030695×10^{-4}
2048.161	14.22334	1	10000	1.024081	0.9143578	0.01
0.2048161	0.001422334	0.0001	1	1.024081×10^{-4}	9.143578×10^{-5}	0.000001
2000	13.88889	0.9764855	9764.855	1	0.8928571	0.009764855
2240	15.55556	1.093664	10936.64	1.12	1	0.01093664
204816.1	1422.334	100	1 000 000	102.4081	91.43578	1

Weight per unit of area, pressure

kgf/cm^2	kPa	lbf/in^2	Mm mercury (0°C)	in. mercury (32°F)	in. water (39.2°F)	atmospheres (standard)	Millibars
1	98.06650	14.22334	735.561	28.9591	393.712	0.9678411	980.6650
0.01019716	1	0.1450377	7.50064	0.295301	4.01474	0.009869233	10
0.07030695	6.894757	1	51.7151	2.03603	27.6807	0.06804596	68.94757
0.00135951	0.133322	0.0193367	1	0.0393701	0.535253	0.00131579	1.33322
0.0345315	3.38638	0.491153	25.4	1	13.5954	0.0334210	33.8638
0.00253993	0.249082	0.0361263	1.86827	0.0735541	1	0.00245825	2.49082
1.033227	101.3250	14.69595	760.002	29.9213	406.794	1	1013.250
0.001019716	0.1	0.01450377	0.750064	0.0295301	0.401474	9.869233×10^{-4}	1

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FIG. 1-3

A.P.I. and Baumé Gravity Tables and Weight Factors

A.P.I. gravity	Baumé gravity	Specific gravity	Lb/ U.S. gal	U.S. gal/ lb
0	10.247	1.0760	8.962	0.1116
1	9.223	1.0679	8.895	0.1124
2	8.198	1.0599	8.828	0.1133
3	7.173	1.0520	8.762	0.1141
4	6.148	1.0443	8.698	0.1150
5	5.124	1.0366	8.634	0.1158
6	4.099	1.0291	8.571	0.1167
7	3.074	1.0217	8.509	0.1175
8	2.049	1.0143	8.448	0.1184
9	1.025	1.0071	8.388	0.1192
10	10.00	1.0000	8.328	0.1201
11	10.99	0.9930	8.270	0.1209
12	11.98	0.9861	8.212	0.1218
13	12.97	0.9792	8.155	0.1226
14	13.96	0.9725	8.099	0.1235
15	14.95	0.9659	8.044	0.1243
16	15.94	0.9593	7.989	0.1252
17	16.93	0.9529	7.935	0.1260
18	17.92	0.9465	7.882	0.1269
19	18.90	0.9402	7.830	0.1277
20	19.89	0.9340	7.778	0.1286
21	20.88	0.9279	7.727	0.1294
22	21.87	0.9218	7.676	0.1303
23	22.86	0.9159	7.627	0.1311
24	23.85	0.9100	7.578	0.1320
25	24.84	0.9042	7.529	0.1328
26	25.83	0.8984	7.481	0.1337
27	26.82	0.8927	7.434	0.1345
28	27.81	0.8871	7.387	0.1354
29	28.80	0.8816	7.341	0.1362
30	29.79	0.8762	7.296	0.1371
31	30.78	0.8708	7.251	0.1379
32	31.77	0.8654	7.206	0.1388
33	32.76	0.8602	7.163	0.1396
34	33.75	0.8550	7.119	0.1405
35	34.73	0.8498	7.076	0.1413
36	35.72	0.8448	7.034	0.1422
37	36.71	0.8398	6.993	0.1430
38	37.70	0.8348	6.951	0.1439
39	38.69	0.8299	6.910	0.1447
40	39.68	0.8251	6.870	0.1456
41	40.67	0.8203	6.830	0.1464
42	41.66	0.8155	6.790	0.1473
43	42.65	0.8109	6.752	0.1481
44	43.64	0.8063	6.713	0.1490
45	44.63	0.8017	6.675	0.1498
46	45.62	0.7972	6.637	0.1507
47	50.61	0.7927	6.600	0.1515
48	50.60	0.7883	6.563	0.1524
49	50.59	0.7839	6.526	0.1532
50	50.58	0.7796	6.490	0.1541
51	50.57	0.7753	6.455	0.1549
52	51.55	0.7711	6.420	0.1558
53	52.54	0.7669	6.385	0.1566
54	53.53	0.7628	6.350	0.1575
55	54.52	0.7587	6.316	0.1583
56	55.51	0.7547	6.283	0.1592
57	56.50	0.7507	6.249	0.1600
58	57.49	0.7467	6.216	0.1609
59	58.48	0.7428	6.184	0.1617
60	59.47	0.7389	6.151	0.1626
61	60.46	0.7351	6.119	0.1634
62	61.45	0.7313	6.087	0.1643
63	62.44	0.7275	6.056	0.1651
64	63.43	0.7238	6.025	0.1660
65	64.42	0.7201	5.994	0.1668
66	65.41	0.7165	5.964	0.1677
67	66.40	0.7128	5.934	0.1685
68	67.39	0.7093	5.904	0.1694
69	68.37	0.7057	5.874	0.1702
70	69.36	0.7022	5.845	0.1711
71	70.35	0.6988	5.817	0.1719
72	71.34	0.6953	5.788	0.1728
73	72.33	0.6919	5.759	0.1736
74	73.32	0.6886	5.731	0.1745
75	74.31	0.6852	5.703	0.1753
76	75.30	0.6819	5.676	0.1762
77	76.29	0.6787	5.649	0.1770
78	77.28	0.6754	5.622	0.1779
79	78.27	0.6722	5.595	0.1787
80	79.26	0.6690	5.568	0.1796
81	80.25	0.6659	5.542	0.1804
82	81.24	0.6628	5.516	0.1813
83	82.23	0.6597	5.491	0.1821
84	83.22	0.6566	5.465	0.1830
85	84.20	0.6536	5.440	0.1838
86	85.19	0.6506	5.415	0.1847
87	86.18	0.6476	5.390	0.1855
88	87.17	0.6446	5.365	0.1864
89	88.16	0.6417	5.341	0.1872
90	89.15	0.6388	5.316	0.1881
91	90.14	0.6360	5.293	0.1889
92	91.13	0.6331	5.269	0.1898
93	92.12	0.6303	5.246	0.1906
94	93.11	0.6275	5.222	0.1915
95	94.10	0.6247	5.199	0.1924
96	95.09	0.6220	5.176	0.1932
97	96.08	0.6193	5.154	0.1940
98	97.07	0.6166	5.131	0.1949
99	98.06	0.6139	5.109	0.1957
100	99.05	0.6112	5.086	0.1966

The relation of Degrees Baumé or A.P.I. to Specific Gravity is expressed by the following formulas:

For liquids lighter than water:

$$\text{Degrees Baumé} = \frac{140}{G} - 130, \qquad G = \frac{140}{130 + \text{Degrees Baumé}}$$

$$\text{Degrees A.P.I.} = \frac{141.5}{G} - 131.5, \qquad G = \frac{141.5}{131.5 + \text{Degrees A.P.I.}}$$

For liquids heavier than water:

$$\text{Degrees Baumé} = 145 - \frac{145}{G}, \qquad G = \frac{145}{145 + \text{Degrees Baumé}}$$

G = Specific Gravity = ratio of the weight of a given volume of oil at 60°F to the weight of the same volume of water at 60°F.

The above tables are based on the weight of 1 U.S. gallon (3.785 liters) of oil with a volume of 231 cubic inches (3785 cubic centimeters) at 60°F (15.56°C) in air at 760mm of pressure and 50% humidity. Assumed weight of 1 U.S. gallon of water at 60°F in air is 8.32828 pounds (3.77764 kg).

To determine the resulting gravity by mixing oils of different gravities:

$$D = \frac{md_1 + nd_2}{m + n}$$

D = Density or Specific Gravity of mixture
m = Volume proportion of oil of d_1 density
n = Volume proportion of oil of d_2 density
d_1 = Specific Gravity of m oil
d_2 = Specific Gravity of n oil

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FIG. 1-4

Values of the Gas Constant R in PV = nRT

Basis of units listed below is 22.4140 liters at 0°C and 1 atm for the volume of 1 g mole. All other values calculated from conversion factors listed in tables.

n	Temperature	Pressure	Volume	R	n	Temperature	Energy	R
gm mol	K	atm	liter	0.082 057 477	gm mol	K	calorie	1.985 9
gm mol	K	atm	cm^3	82.057	gm mol	K	joule	8.314 5
gm mol	K	mm Hg	liter	62.364				
gm mol	K	bar	liter	0.083 145	lb mol	°R	Btu	1.985 9
gm mol	K	kg/cm^2	liter	0.084 784	lb mol	°R	hp-hr	0.000 780 48
gm mol	K	kPa	m^3	0.008 314 5	lb mol	°R	Kw-hr	0.000 582 00
lb mol	°R	atm	ft^3	0.730 24	lb mol	°R	ft-lb	1 545.3
lb mol	°R	in.Hg	ft^3	21.850				
lb mol	°R	mm Hg	ft^3	554.98	k mol	K	joule	8 314.5
lb mol	°R	lb/in^2	ft^3	10.732				
lb mol	°R	lb/ft^2	ft^3	1 545.3				
lb mol	K	atm	ft^3	1.3144				
lb mol	K	mm Hg	ft^3	998.97				
k mol	K	kPa	m^3	8.3145				
k mol	K	bar	m^3	0.083 145				

FIG. 1-5

Commercial Base Pressure Conversion Factors

(Factors to Convert to Other Base Pressures)

Given base pressure (See notes 1&2)	13.9	14.65	101.325 kPa @ 15°C	760 mm Hg or 14.696	14.696 @ 59°F	14.7	14.73	14.73 Sat.	30" Hg	14.9	15.025	16.4
13.9	1.0000	0.9488	0.9440	0.9458	0.9440	0.9456	0.9437	0.9603	0.9434	0.9329	0.9251	0.8476
14.65	1.0540	1.0000	0.9950	0.9969	0.9950	0.9966	0.9946	1.0122	0.9943	0.9832	0.9750	0.8933
101.325 kPa @ 15°C	1.0593	1.0050	1.0000	1.0019	1.0000	1.0016	0.9996	1.0173	0.9993	0.9882	0.9800	0.8978
14.696 or 760 mm Hg	1.0573	1.0031	0.9981	1.0000	0.9981	0.9997	0.9977	1.0153	0.9974	0.9863	0.9781	0.8960
14.696 @ 59°F	1.0593	1.0050	1.0000	1.0019	1.0000	1.0016	0.9996	1.0173	0.9993	0.9882	0.9800	0.8978
14.7	1.0576	1.0034	0.9984	1.0003	0.9984	1.0000	0.9980	1.0156	0.9976	0.9866	0.9784	0.8963
14.73	1.0597	1.0055	1.0004	1.0023	1.0004	1.0020	1.0000	1.0177	0.9997	0.9886	0.9804	0.8982
14.73 Sat.	1.0413	0.9880	0.9830	0.9849	0.9830	0.9846	0.9826	1.0000	0.9823	0.9714	0.9633	0.8826
30" Hg	1.0601	1.0058	1.0007	1.0026	1.0007	1.0024	1.0003	1.0180	1.0000	0.9889	0.9807	0.8984
14.9	1.0719	1.0171	1.0119	1.0139	1.0119	1.0136	1.0115	1.0294	1.0112	1.0000	0.9917	0.9085
15.025	1.0809	1.0256	1.0204	1.0224	1.0204	1.0221	1.0200	1.0381	1.0197	1.0084	1.0000	0.9162
16.4	1.1795	1.1195	1.1138	1.1159	1.1138	1.1156	1.1134	1.1331	1.1130	1.1007	1.0915	1.0000

FIG. 1-6

Pressure Equivalents

psia	in.Hg @ 32°F	mm Hg @ 0°C	kPa
1	2.03603	51.7151	6.8948
0.491153	1	25.400	3.38638
0.019337	0.3937	1	0.1333
0.14504	0.2953	7.5006	1
13.9	28.3008	718.8399	95.83772
14.65	29.82784	757.62621	101.00882
14.696	29.9215	760.0051	101.32598
14.6959	29.9213	760.00	101.3250
14.7	29.9296	760.21197	101.3536
14.73	29.9907	761.7634	101.5604
14.73456	30.00	761.999	101.5918
14.9	30.3368	770.55499	102.73252
15.025	30.59135	777.01937	103.5944
16.4	33.39532	848.12764	113.0747

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FIG. 1-7

Viscosity Relationships

$$\text{Kinematic viscosity (centistokes)} = \frac{\text{absolute viscosity (centipoises)}}{\text{density (g/cm}^3\text{)*}}$$

$$\text{ft}^2/\text{sec} = (\text{centistokes})(1.07639 \times 10^{-5})$$

$$\text{centistokes} = (\text{ft}^2/\text{sec})(92903.4)$$

* Usually same as specific gravity.

APPROXIMATE VISCOSITY CONVERSIONS

ft^2/sec (50–100 SSU)	=	$[(\text{SSU})(2.433 \times 10^{-6})] - (0.00210/\text{SSU})$
ft^2/sec (100–350 SSU)	=	$[(\text{SSU})(2.368 \times 10^{-6})] - (0.00145/\text{SSU})$
ft^2/sec (over 350 SSU)	=	$[\text{SSU (at } 100°\text{F)} (3.2310 \times 10^{-6})]$
centistokes (50–100 SSU)	=	[(SSU) (0.226)] – (205.3/SSU)
centistokes (100–350 SSU)	=	[(SSU) (0.220)] – (147.7/SSU)
centistokes (over 350 SSU)	=	[SSU (at 100°F or 37.8°C) (0.21576)]
centistokes (over 350 SSU)	=	[SSU (at 210°F or 98.9°C) (0.21426)]
centistokes (over 500 SSU)	=	[SSU (at 122°F or 50°C) (2.120)]
centistokes (over 300 Redwood #1)	=	[Redwood #1 (Standard) (0.255)]
centistokes (over 50 Redwood #2)	=	[Redwood #2 (Admiralty) (2.3392)]
centistokes (over 18 Engler)	=	(Engler) (7.389)
centistokes (over 20 Stormer)	=	(Stormer) (2.802)
centistokes (over 1.0 Demler #10)	=	(Demler #10) (31.506)
centistokes (over 1.3 Demler #1)	=	(Demler #1) (3.151)
centistokes (over 14 Parlin #20)	=	(Parlin Cup #20) (61.652)
centistokes (over 230 Ford #4)	=	(Ford Cup #4) (3.753)
centistokes	=	6200 Barbey

VISCOSITY – UNIT CONVERSIONS

KINEMATIC VISCOSITY

MULTIPLY	BY	TO OBTAIN
ft^2/sec	92903.04	centistokes
ft^2/sec	0.092903	sq meters/sec
sq meters/sec	10.7639	ft^2/sec
sq meters/sec	1 000 000.0	centistokes
centistokes	0.000 001	sq meters/sec
centistokes	0.000 010 763 9	ft^2/sec

ABSOLUTE OR DYNAMIC VISCOSITY

lbf-sec/ft^2	47880.26	centipoises
lbf-sec/ft^2	47.8803	Pascal-sec
centipoises	0.000 102	kg-sec/sq meter
centipoises	0.000 020 885 4	lbf-sec/sq ft*
centipoises	0.001	Pascal-sec
Pascal-sec	0.020 885 4	lbf-sec/sq ft
Pascal-sec	1000	centipoises

* Sometimes absolute viscosity is given in terms of pounds mass. In this case — (centipoises)(0.000672) = lbm/ft . sec.

ABSOLUTE TO KINEMATIC VISCOSITY

centipoises	1/density (g/cm^3)	centistokes
centipoises	0.000 671 97/density (lb/ft^3)	ft^2/sec
lbf-sec/ft^2	32.174/density (lb/ft^3)	ft^2/sec
kg-sec/m^2	9.80665/density (kg/m^3)	sq meters/sec
Pascal-sec	1000/density (g/cm^3)	centistokes

KINEMATIC TO ABSOLUTE VISCOSITY

centistokes	density (g/cm^3)	centipoises
sq meters/sec	(0.10197)[density (kg/m^3)]	kg-sec/sq meter
ft^2/sec	(0.03108) [density (lb/ft^3)]	lbf-sec/ft^2
ft^2/sec	(1488.16) [density (lb/ft^3)]	centipoises
centistokes	(0.001) [density (g/cm^3)]	Pascal-sec
sq meters/sec	(1000) [density (g/cm^3)]	Pascal-sec

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1 calorie (15°C) = 4.185 80 joule
1 calorie (20°C) = 4.181 90 joule
1 calorie (Thermochemical) = 4.184 000 joule

The fundamental relationship between the Btu and the calorie:

$$\frac{\text{gram–pound relationship}}{\text{Fahrenheit–Celsius scale relationship}}$$

or: $$\text{Btu} \times \frac{453.592\ 29}{1.8} = \text{calorie (IT, mean, or other)}$$

1 therm = 100,000 Btu = 105.5056 x 10^6 J
= 105,505.6 kJ = 1.055 056 x 10^5 kJ.
(Btu denotes British Thermal Units)
(ref: Physical Properties of Natural Gases, Gas Unie, 1988 p. 23)

1 decatherm = 1,000,000 Btu = 1 MMBtu = 10^6 Btu

1 thermie = 1 Mcal (15°C) = 4.1858 x 10^6 J

1 terajoule = 1 TJ = 1 x 10^{12} Joule

Gas Volume Relationships

Gas volume are commonly referred to in "standard" or "normal" units.

Standard conditions commonly refers to gas volumes measured at:

15°C and 101.3250 kPa

60°F and 14.696 psia

These P & T conditions are often indicated as Standard Temperature and Pressure, abbreviated to STP. However standard conditions can refer to other combinations of pressure and temperature as might be agreed between gas buyer and seller. There is no internationally accepted standard for STP. In 1980 GPA adopted 15°C, 101.3250 kPa (abs) as standard conditions for SI units. Thus standard molar volumes are:

23.645 std m^3/kmol at 15°C (288.15 K), 101.3250 kPa. GPA SI standard conditions.

379.49 std ft^3/lb mol at 60°F (519.67°R), 14.696 psia.

Other standard conditions sometimes used are:

P	T
101.3250 kPa	20°C
100.0 kPa	15°C
14.73 psia	60°F
14.40 psia (99.28 kPa) and others	60°F

Normal conditions refers to gas volumes measured at 0°C and 101.3250 kPa (760 mm Hg), often abbreviated to NTP (Normal Temperature and Pressure). The normal molar volume is:

22.414 Nm^3/kmol at 0°C (273.15 K), 101.3250 kPa

Example 1-1 — Consider 100 kg of 16 mole weight gas. This is equivalent to 220.5 pounds and 6.25 kmol (13.78 lb mols). The volume occupied by this mass can be found by using the relationships presented above:

147.8 std m^3 (15°C, 101.3250 kPa)

140.1 Nm^3 (0°C, 101.3250 kPa)

5,229 std ft^3 (scf) (60°F, 14.696 psia)

Thus the following volume conversions individually represent the same mass (kg, kmol or lb, lb mol) of gas assuming either constant z or ideal gas behavior.

1 Nm^3 = 37.33 scf. Same mass on either side of this equation.

1 std m^3 = 35.38 scf. Again, same mass on either side of this equation, but 1 std m^3 (15°C) contains slightly smaller mass than in 1 Nm^3 (0°C). Gas is less dense at 15°C than at 0°C.

Other Useful Relationships

Physical conversion of ft^3 and m^3 is given in Fig. 1-2.

Absolute zero: 0 K (–273.15°C), 0°R (–459.67°F).

1 Thermochemical unit x 0.999 331 2 = IT Unit (Btu or other)

1 Thermochemical cal/gm x 1.8 x 0.999 331 2 = IT Btu/lb

Entropy, 1 Btu/(lb • °R) = 4.186 8 kJ/(kg • K)

Enthalpy, 1 Btu/lb = 2.326 kJ/kg

7000 Grains = 1 lb_m

1 Grain = 64.798 91 mg

1 Grain/100 scf = 22.888 352 mg/m^3

1 Grain/U.S. gallon = 17.118 06 g/m^3

1 Hectare = 10 000 m^2 = 100 are = 2.471 acres

°C = 5/9 (°F – 32)

°F = 9/5 (°C) + 32

K = °C + 273.15 = 5/9 °R

°R = °F + 459.67 = 1.8 K

1 newton of force = 1 kg • m/s^2 = 1 N

1 pascal pressure = 1 N/m^2 = 1 Pa

Notes:

1. Pressure is stated as psia except where specifically indicated otherwise.
2. Base temperature is assumed to be 60°F except where specifically indicated otherwise.
3. Saturated water vapor pressure at 60°F equals 0.2561 psi (International Critical Tables).
4. 60°F = 15.6°C 15°C = 59°F
5. 30" Hg @ 32°F is normally assumed equivalent to 14.73 psi commercially.
6. To convert heating value or gas price from one base pressure to another, the reciprocals of the above conversion factors apply.
7. kPa represents kPa (abs) unless indicated as kPa (ga) for gauge pressure.
8. kJ/(kmol • K) = kJ/(kmol • °C)

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FIG. 1-8

Approximate Atomic Weights

Name	Symbol	Atomic Number	Approximate Atomic Weight
Aluminum	Al	13	27.0
Antimony	Sb	51	121.8
Argon	Ar	18	39.9
Arsenic	As	33	74.9
Barium	Ba	56	137.3
Boron	B	5	10.8
Bromine	Br	35	79.9
Cadmium	Cd	48	112.4
Calcium	Ca	20	40.1
Carbon	C	6	12.0
Chlorine	Cl	17	35.5
Chromium	Cr	24	52.0
Cobalt	Co	27	58.9
Copper	Cu	29	63.5
Fluorine	F	9	19.0
Gold	Au	79	197.0
Helium	He	2	4.0
Hydrogen	H	1	1.0
Iodine	I	53	126.9
Iron	Fe	26	55.8
Lead	Pb	82	207.2
Lithium	Li	3	6.9
Magnesium	Mg	12	24.3
Manganese	Mn	25	54.9
Mercury	Hg	80	200.6
Molybdenum	Mo	42	95.9
Neon	Ne	10	20.2
Nickel	Ni	28	58.7
Nitrogen	N	7	14.0
Oxygen	O	8	16.0
Phosphorus	P	15	31.0
Platinum	Pt	78	195.1
Potassium	K	19	39.1
Radon	Rn	86	222.0
Selenium	Se	34	79.0
Silicon	Si	14	28.1
Silver	Ag	47	107.9
Sodium	Na	11	23.0
Strontium	Sr	38	87.6
Sulfur	S	16	32.1
Tin	Sn	50	118.7
Titanium	Ti	22	47.9
Tungsten	W	74	183.9
Vanadium	V	23	50.9
Zinc	Zn	30	65.4

Note: Precise values of atomic weights vary. For example 1959 values were based on oxygen = 16.000, whereas 1961 values use the isotope C^{12} = 12.000 as the basis. The above values are rounded to one decimal place for use in less precise calculations.

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FIG. 1-9

Gaseous Composition of Air

Gas	Symbol	Molecular Weight	Volume %
Basic constituents			
Nitrogen	N_2	28.013	78.084 ± 0.004 %
Oxygen	O_2	32.0	20.946 ± 0.002 %
Argon	Ar	39.948	0.934 ± 0.001 %
Trace gases			
Neon	Ne	20.183	18.12 ± 0.04 ppm
Helium	He	4.003	5.239 ± 0.05 ppm
Krypton	Kr	83.80	1.14 ± 0.01 ppm
Xenon	Xe	131.30	0.087 ± 0.001 ppm
Hydrogen	H_2	2.016	0.5 ± 0.01 ppm
Impurities			
Water	H_2O	18.015	0.1 – 2.8 %
Carbon dioxide	CO_2	44.011	300 ± 30 ppm
Methane	CH_4	16.043	1.5 to 2.5 ppm
Carbon monoxide	CO	28.010	0.061 to 1.1 ppm
Sulphur dioxide	SO_2	64.06	1.1 ppm
Nitrous oxide	N_2O	44.012	0.5 ppm
Ozone	O_3	47.998	0.011 to 0.11
Nitrogen dioxide	NO_2	46.005	0.005 to 0.02 ppm
Radon	Rn	222	Trace
Nitric oxide	NO	30.006	Trace

Openshaw, D. and Cain, S., "Ultra-pure Cryogenic Nitrogen Generator", TCE, The Chemical Engineer, The Institution of Chemical Engineers, Rugby, England, UK., November 2002, p. 30.

FIG. 1-10

Greek Alphabet

α A=Alpha	ι I=Iota	ρ P=Rho
β B=Beta	κ K=Kappa	σ Σ=Sigma
γ Γ=Gamma	λ Λ-Lambda	τ T=Tau
δ Δ=Delta	μM=Mu	υ Y=Upsilon
ε E=Epsilon	ν N=Nu	ϕ Φ=Phi
ζ Z=Zeta	ξ Ξ=Xi	χ X=Chi
η H=Eta	ο O=Omicron	ψ Ψ=Psi
θ Θ=Theta	π Π=Pi	ω Ω=Omega

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GPA Publications*

TECHNICAL STANDARDS MANUAL — A one-volume loose-leaf manual that contains the official test methods and specifications of the GPA. Included are GPA Standards 101-43, 2140, 2142, 2145, 2165, 2166, 2172, 2174, 2177, 2186, 2187, 2188, 2194, 2261, 2265, 2286, 2377, 3132, 8173, 8182, 8186, 8195, RB-181, RB-194 and ASTM Test Procedures D86, D130, D156, D216, D287, D323, D1070, D1265, D1267, D1657, D1837, D1838, D2158, D2163, D2713, D2784. Subject tabbed for ready reference.

PLANT OPERATIONS TEST MANUAL — A one-volume loose leaf manual containing the significant tests commonly used in the operation of a plant. This convenient compilation contains test methods which originated with ASTM, API, PEA, GPA, U.S. Bureau of Mines, Betz Laboratories, Girdler Corp. and Fluor Corporation. In addition, it contains all of the information in the "Technical Standards Manual."

CORROSION DETECTION REPORT — A practical and convenient field guide to detecting, locating, and measuring common corrosion problems in gas processing plants and related equipment.

PLANT PROJECT AND DESIGN CHECK LIST — Prepared by Technical Section A to serve as a guide in planning and preparing job and equipment specifications using categories normally required for a processing plant.

ENERGY CONSERVATION CHECK LIST — Prepared by Technical Section A, and patterned after the earlier "Plant Project and Design Check List", it was developed to serve as a guide check list for energy conservation within various units of equipment.

SAFETY INSPECTION CHECK LIST — Prepared by the Safety Committee to show the plant and design engineer items of importance to check from the viewpoint of safety in design and operation. It consists of approximately 60 pages covering 15 areas in the gas processing plant.

GUIDE FOR PERFORMANCE TESTING OF PLANT EQUIPMENT — Prepared by Section A as a guide for checking the performance of various items of equipment within a plant. Over 200 pages divided into five major sections: A. Plant Acceptance and Performance Tests; B. Towers and Vessels; C. Engines and Turbines; D. Compressors, Pumps and Blowers; and E. Heating and Cooling.

BIBLIOGRAPHY ON HYDROCARBONS — A comprehensive bibliography of important articles on hydrocarbons and associated compounds. Over 100 pages with 1300 abstracts, cloth bound, 9 x 12 inches. The text material covers the period from 1946-1960 inclusive. It contains a wealth of timesaving references for petroleum engineers, chemists, researchers, and students. J. A. Muckleroy.

NORTH AMERICAN STORAGE CAPACITY FOR LIGHT HYDROCARBONS AND U.S. LP-GAS IMPORT TERMINALS — A biennial report compiling the storage facilities for light hydrocarbons in the U.S. and the terminal facilities in the U.S. capable of receiving imported LP-gas. Excel Diskette available.

SALES OF NATURAL GAS LIQUIDS AND LIQUEFIED REFINERY GASES — A joint publication of API, GPA, PERC, and NPGA summarizes annual survey data on gas liquids sales — by product, by major market uses, and by state.

NGL SUPPLY/DEMAND/INVENTORY DATA — A compilation of historical gas plant and refinery production of gas liquids production, by month and by product. Data are extended to include total monthly supply, demand and inventories of individual products. Also available on LOTUS 123 diskette. Current monthly summaries of these data are available on request.

LP-GAS ODORIZATION SYMPOSIA PROCEEDINGS — Collection of papers presented at two symposia on LP-gas Odorization Technology in 1989 and 1990. Co-sponsored by the National Propane Gas Association and the Propane Gas Association of Canada, these proceedings are a compilation of information on LP-gas odorants, odorization practices and equipment, and original research into the behavior of odorants in the LP-gas fuel system.

REPORT OF INVESTIGATIONS–ODORIZATION OF LP-GAS — A summary of research findings and studies conducted during 1986-1990 by a joint task force of representatives from GPA, the National Propane Gas Association and the Propane Gas Association of Canada. Includes task force recommendations for effective odorization of LP-gas.

ANNUAL CONVENTION PROCEEDINGS — Collection of all technical papers presented in the technical forums and general sessions of the GPA annual conventions.

Standards and Bulletins

Specifications

GPA Standard 2140 — Liquefied Petroleum Gas Specifications and Test Methods.

GPA Standard 3132 — Natural Gasoline Specifications and Test Methods.

Analytical Methods

AGA-GPA CODE 101 — Standard Compression and Charcoal Tests for Determining the Natural Gasoline Content of Natural Gas.

GPA 2100 — Tentative Method for the Qualitative Determination of COS in Propane.

GPA Standard 2165 — Standard for Analysis of Natural Gas Liquid Mixtures by Gas Chromatography.

GPA Standard 2177 — Analysis of Demethanized Hydrocarbon Liquid Mixtures Containing Nitrogen and Carbon Dioxide by Gas Chromatography.

GPA Standard 2186 — Tentative Method for the Extended Analysis of Hydrocarbon Liquid Mixtures Containing Ni-

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trogen and Carbon Dioxide by Temperature Programmed Gas Chromatography.

GPA Standard 2187 — Tentative Method for the Determination of Ammonia in Liquid Propane.

GPA Standard 2l88 — Tentative Method for the Determination of Ethyl Mercaptan in LP-gas Using Length of Stain Tubes.

GPA Standard 2194 — Tentative Low Pressure Field Method for Determining Ethyl Mercaptan Odorant in LP-gas Using Length of Stain Tubes.

GPA Standard 2198 — Selection, Preparation, Validation, Care, and Storage of Natural Gas and Natural Gas Liquid Reference Standard Blends.

GPA Standard 2261 — Analysis for Natural Gas and Similar Gaseous Mixtures by Gas Chromatography.

GPA Standard 2265 — GPA Standard for Determination of Hydrogen Sulfide and Mercaptan Sulfur in Natural Gas (Cadmium Sulfate-Iodometric Titration Method).

GPA Standard 2286 — Tentative Method of Extended Analysis for Natural Gas and Similar Gaseous Mixtures by Temperature Programmed Gas Chromatography.

GPA Standard 2290 — Tentative Method for the Determination of Carbonyl Sulfide (COS) in Unodorized Liquid Propane.

GPA Standard 2377 — Test for Hydrogen Sulfide and Carbon Dioxide in Natural Gas Using Length of Stain Tubes.

Measurement Standards

GPA Standard 2142 — Standard Factors for Volume Correction and Specific Gravity Conversion of Liquefied Petroleum Gases.

GPA Standard 2145 — Physical Constants for Paraffin Hydrocarbons and Other Components of Natural Gas. Data are given in both English and SI Units.

GPA Standard 2172 — Calculation of Gross Heating Value, Relative Density and Compressibility of Natural Gas Mixtures from Compositional Analysis.

GPA Standard 8173 — Method for Converting Mass of Natural Gas Liquids and Vapors to Equivalent Liquid Volumes. Data are given in both English and SI Units.

GPA Standard 8182 — Tentative Standard for the Mass Measurement of Natural Gas Liquids.

GPA Standard 8186 — Measurement of Liquid Hydrocarbon by Truck Scales.

GPA Standard 8195 — Tentative Standard for Converting Net Vapor Space Volumes to Equivalent Liquid Volumes.

GPA Reference Bulletin 181 — Heating Value as a Basis for Custody Transfer of Natural Gas. A reference to provide authoritative interpretation of accepted procedures for determining heating values.

GPA Reference Bulletin 194 — Tentative NGL Loading Practices.

Sampling Methods

GPA Standard 2166 — Obtaining Natural Gas Samples for Analysis by Gas Chromatography.

GPA Standard 2174 — Method for Obtaining Liquid Hydrocarbon Samples Using A Floating Piston Cylinder.

Miscellaneous Standards

GPA Publication 1167 — GPA Glossary-Definition of Words and Terms Used in the Gas Processing Industry.

** GPA Committees periodically update or revise GPA publications. The last two digits of the year in which the publication is revised are appended to the publication number, e.g., GPA 2145-02.*

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GPA Research Reports

Results of most of the GPA sponsored research projects since 1971 have been published as numbered Research Reports. In some cases, individual company sponsored data have been published as Technical Publications.

RR-1 *Bibliography of Data Sources for Low-Temperature Vapor-Liquid Equilibria in Non-Absorber Oil Systems* — R. D. Gunn and S. V. Mainkar, University of Texas, Austin, Texas. Project 692-A.

RR-2 *K-Values in Highly Aromatic and Highly Naphthenic Real Oil Absorber Systems* - Grant Wilson and Sherman T. Barton, P-V-T, Inc., Houston, Texas. Project 691.

RR-3 *Enthalpy and Entropy of Non-Polar Liquids at Low Temperatures* - K. C. Chao and R. A. Greenkorn, Purdue University, Lafayette, Indiana.

RR-4 *Evaluation of Eight Enthalpy Correlations* - Kenneth E. Starling, David W. Johnson and Philip C. Colver, University of Oklahoma, Norman, Oklahoma. Project 68l.

RR-5 *Experimental Measurements of Vapor-Liquid Equilibrium Data for the Ethane-Carbon Dioxide and Nitrogen-n-Pentane Binary Systems* - Kurata-Swift Consulting Engineers, Lawrence, Kansas. Project 692-B.

RR-6 *Enthalpies of Methane-C_7 Systems* - Bert Eakin, Grant M. Wilson and Will E. DeVaney, P-V-T, Inc., Houston, Texas. Project 661.

RR-7 *The Equilibrium Phase Properties of the Binary Systems: Nitrogen-Hydrogen Sulfide, Isobutane-Hydrogen Sulfide, Isobutane-Carbon Dioxide and Isobutane-Ethane* - D. B. Robinson and G. J. Besserer, University of Alberta, Edmonton, Alberta.

RR-8 *1971-1972 Enthalpy Correlation Evaluation Study* - Kenneth E. Starling, University of Oklahoma, Norman, Oklahoma. Project 713.

RR-9 *Enthalpies of Hydrogen Sulfide-Methane-Ethane Systems* - Bert Eakin and Will E. DeVaney, P-V-T, Inc., Houston, Texas. Project 722.

RR-10 *Solubility of Solid Carbon Dioxide in Pure Light Hydrocarbons and Mixtures of Light Hydrocarbons* - Fred Kurata, Center for Research, Inc., Lawrence, Kansas.

RR-11 *GPA Experimental Enthalpy Values Referred to Two Base Levels* - John M. Lenoir and Gene A. Cochran, University of Southern California, Los Angeles, California. Project 733.

RR-12 *Enthalpy and Phase Boundary Measurements on Carbon Dioxide and Mixtures of Carbon Dioxide with Methane, Ethane and Hydrogen Sulfide* - Grant M. Wilson and James M. Peterson, Brigham Young University, Provo, Utah. Project 731.

RR-13 *Prediction of Absorber Oil K-Values and Enthalpies* - John H. Erbar, Oklahoma State University, Stillwater, Oklahoma. Project 714.

RR-14 *Solubility of Heavier Hydrocarbons in Liquid Methane* - Fred Kurata, Center For Research, Inc., Lawrence, Kansas. Project 738-A.

RR-15 *The Equilibrium Phase Properties of Selected Binary Systems at Low Temperature: Nitrogen-Hydrogen Sulfide, Ethane-Hydrogen Sulfide and n-Butane-Carbon Dioxide* - D. B. Robinson and H. Kalra, University of Alberta, Edmonton, Alberta. Project 738.

RR-16 *Solubility Limits of Heavy Hydrocarbons in NGL and LNG Mixtures* - Will DeVaney, Bert Eakin and James M. Berryman, P-V-T, Inc., Houston, Texas. Project 735-A.

RR-17 *Smoothed Experimental Enthalpy Data for Three Methane-Ethane Binaries and a Methane-Ethane-Propane Ternary* - John E. Powers, Andre W. Furtado, Ravi Kant and Adriana Kwan, University of Michigan, Ann Arbor, Michigan. Project 723.

RR-18 *The Equilibrium Phase Properties of Selected Binary Systems: n-Heptane-Hydrogen Sulfide, n-Heptane-Carbon Dioxide and i-Butane-Nitrogen* - D. B. Robinson and H. Kalra, University of Alberta, Edmonton, Alberta. Project 745.

RR-19 *Vapor Phase Data for the Binary Systems of Methane with n-Butane, n-Pentane, n-Hexane and n-Heptane* - Roger J. J. Chen, Patsy S. Chappelear and Riki Kobayashi, Rice University, Houston, Texas. Project 692-B.

RR-20 *K-Values for the Methane-n-Butane, Methane-n-Pentane and Methane-n-Hexane Systems* - Doug Elliot, Y. N. Lin, T. C. Chu, Patsy S. Chappelear and Riki Kobayashi, Rice University, Houston, Texas. Project 692-B.

RR-21 *Dew-Point Values for the Methane-Carbon Dioxide System* - S. C. Hwang, Ho-Mu Lin, Patsy S. Chappelear and Riki Kobayashi, Rice University, Houston, Texas. Project 739.

RR-22 *Solubility of Hydrocarbons in Cryogenic LNG and NGL Mixtures* - James P. Kohn and Kraemer D. Luks, University of Notre Dame, Notre Dame, Indiana. Project 735.

RR-23 *Measurement of Ethane and Propane Recovery and Total Fraction Condensed for Simulated Natural Gas Mixtures* - Grant M. Wilson, John R. Cunningham, B. Steve Lofgren and Veldon E. Messick, Thermochemical Institute, Brigham Young University, Provo, Utah. Project 737.

RR-24 *Enthalpy and Phase Boundary Measurements on Mixtures of Nitrogen with Methane, Carbon Dioxide and Hydrogen Sulfide* - Grant M. Wilson, John R. Cunningham and Paul F. Nielsen, Brigham Young University, Provo, Utah. Project 741.

RR-25 *The Vapor-Liquid Equilibrium of the CH_4-CO_2 System at Low Temperatures* - S. C. Mraw, S. C. Hwang and Riki Kobayashi, Rice University, Houston, Texas. Project 739.

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RR-26 *Statistical Thermodynamics of Solutions in Natural Gas and Petroleum Refining* - M. D. Donohue and J. M. Prausnitz, University of California, Berkeley, California. Project 721.

RR-27 *Solubility of Hydrocarbons in Cryogenic LNG and NGL Mixtures* - James P. Kohn and Kraemer D. Luks, University of Notre Dame, Notre Dame, Indiana. Project 735-A.

RR-28 *The Characterization of the Heptanes and Heavier Fractions for the GPA Peng-Robinson Programs* - D. B. Robinson and D. Y. Peng, University of Alberta, Edmonton, Alberta. Project 756.

RR-29 *The Equilibrium Phase Compositions of Selected Aromatic and Naphthenic Binary Systems: Toluene-Carbon Dioxide, Toluene-Hydrogen Sulfide, Methylcyclohexane-Carbon Dioxide and Methylcyclohexane-Hydrogen Sulfide* - D. B. Robinson and H-J Ng, University of Alberta, Edmonton, Alberta. Project 755, 755-A and 757-B.

RR-30 *High Temperature V-L-E Measurements for Substitute Gas Components* - Will DeVaney, James M. Berryman, Pen-Li Kao and Bert Eakin, P-V-T, Inc., Houston, Texas. Project 757.

RR-31 *The Equilibrium Phase Properties of a Synthetic Sour Gas Mixture and a Simulated Natural Gas Mixture* - D. B. Robinson, H. Kalra and Hans Rempis, University of Alberta, Edmonton, Alberta. Project 737-A & 737-B.

RR-32 *Vapor-Liquid Compositions of Propane-Butane Mixtures in Cold Weather Field Tests* - Thomas H. May and Dorab N. Baria, University of North Dakota, Grand Forks, North Dakota. Project 789-A.

RR-33 *Solubility of Hydrocarbons in Cryogenic LNG and NGL Mixtures* - James P. Kohn and Kraemer D. Luks, University of Notre Dame, Notre Dame, Indiana. Project 735.

RR-34 *Sour Water Equilibria: Ammonia Volatility Down to ppm; Effect of Electrolytes on Ammonia Volatility; pH vs. Composition* - Grant Wilson, Richard S. Owens and Marshall W. Roe, Wilco Research, Provo, Utah. Project 758-A.

RR-35 *Computer Simulations of Vapor-Liquid Compositions of Propane-Butane Mixtures in Cold Weather Field Tests* - Thomas H. May and Dorab N. Baria, University of North Dakota, Grand Forks, North Dakota. Project 789-B.

RR-36 *Literature Survey for Synthetic Gas Components--Thermodynamic Properties* - P. Pendergraft, M. Marston, M. Gonzales, V. Rice and J. Erbar, Oklahoma State University, Stillwater, Oklahoma. Project 746.

RR-37 *Enthalpy Measurements on Synthetic Gas Systems: Hydrogen-Methane, Hydrogen-Carbon Monoxide* - James M. Berryman, Will E. DeVaney, Bert E. Eakin and Nancy L. Bailey, P-V-T, Inc., Houston, Texas. Project 742.

RR-38 *A Preliminary Version of the PHC Equation of State Computerized for Engineering Calculations* - E. C. Hohmann, California State Polytechnic University, Pomona, California. Project 771.

RR-39 *The Equilibrium Phase Properties of Selected m-Xylene and Mesitylene Binary Systems: CH_2-m-Xylene, CO_2-m-Xylene, CH_4-Mesitylene* - D. B. Robinson, H-J Ng and S-S Huang, University of Alberta, Edmonton, Alberta. Project 755-B.

RR-40 *Measurement of Ethane and Propane Recovery and Total Fraction Condensed in the Bubble Point Region of Two Simulated Natural Gas Mixtures* - John R. Cunningham, John L. Oscarson and Mark L. Jenson, Brigham Young University, Provo, Utah. Project 737.

RR-40A *Phase Equilibria of a High Nitrogen Content Synthetic Natural Gas* - James P. Kohn and Robert C. Merrill, University of Notre Dame, Notre Dame, Indiana. Project 795.

RR-4l *Vapor-Liquid Equilibrium Data on Water-Substitute Gas Components: $N2$-H_2O, H_2-H_2O, CO-H_2O and H_2-CO-H_2O and H_2S-H_2O* - Paul C. Gillespie and Grant Wilson, Wilco Research Co., Provo, Utah. Project 758-B.

RR-42 *Predicting Synthetic Gas and Natural Gas Thermodynamic Properties Using a Modified Soave Redlich Kwong Equation of State* - J. H. Erbar, A. K. Jagota, S. Muthswamy and M. Moshfeghian, Oklahoma State University, Stillwater, Oklahoma. Project 752.

RR-43 *The Equilibrium Phase Properties of Selected m-Xylene and Mesitylene Binary Systems: m-Xylene-Mesitylene-H_2S, Mesitylene-CO_2* - D. B. Robinson and S-S Huang, University of Alberta, Edmonton, Alberta. Project 755-B.

RR-44 *Vapor-Liquid and Liquid-Liquid Equilibria in the Methane-Toluene System and Relation of Liquid-Liquid Equilibrium Behavior at Low Temperatures to Vapor-Liquid Equilibria Behavior at High Temperatures and Elevated Pressures* - Yeuh-Neu Lin, Shuen-Cheng Swang and Riki Kobayashi, Rice University, Houston, Scott W. Hopke, Exxon Production Company, Houston, Texas. Project 757.

RR-45 *The Water Content and Correlation of the Water Content of Methane in Equilibrium with Hydrates, (I); and the Water Content of a High Carbon Dioxide Simulated Prudhoe Bay Gas in Equilibrium with Hydrates, (II)* - Keichi Aoyagi, Kyoo Y. Song and Riki Kobayashi, Rice University, Houston, Texas, E. Dendy Sloan and P. B. Dharmawardhana, Dept. of Chemical Engineering, Colorado School of Mines, Golden, Colorado. Project 775.

RR-46 *Vapor Liquid Equilibrium Study of the H_2-CH_4 System at Low Temperatures and Elevated Pressures* - Joint research report for GPA and Gas Research Institute. Jane Huey Hong and Riki Kobayashi, Rice University, Houston, Texas. Project 757.

RR-47 *Behavior of CH_4-CO_2-H_2S Mixtures at Sub-Ambient Temperatures* - Joint research report for GPA and Canadian Gas Processors Association. D. B. Robinson, H-J. Ng and A. D. Leu, University of Alberta, Edmonton, Alberta. Project 738-A.

RR-48 *Vapor-Liquid and Liquid-Liquid Equilibria: Water-Methane; Water-Carbon Dioxide; Water-Hydrogen Sulfide; Water-nPentane; Water-Methane-*

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nPentane - Paul Gillespie and Grant Wilson, Wiltec Research Company, Provo, Utah. Project 758-B.

RR-49 *Liquid-Liquid-Vapor Equilibria in Cryogenic LNG Mixtures* - James P. Kohn, University of Notre Dame, Notre Dame, Indiana and Kraemer Luks, University of Tulsa, Tulsa, Oklahoma. Project 795.

RR-49A *Liquid-Liquid-Vapor Equilibria in Cryogenic LNG Mixtures (raw data)*. James P. Kohn, University of Notre Dame, Notre Dame, Indiana and Kraemer D. Luks, University of Tulsa, Tulsa, Oklahoma. Project 795.

RR-50 *Measurement and Interpretation of the Water Content of a Methane-Propane (5.31 mol %) Mixture in the Gaseous State in Equilibrium with Hydrate* - Hyoo Y. Song and Riki Kobayashi, Rice University, Houston, Texas. Project 775.

RR-5l *The Equilibrium Phase Properties of Selected Naphthenic Binary Systems: Methylcyclohexane-Nitrogen, Ethylcyclohexane-Hydrogen Sulfide and n-Propylcyclohexane-Hydrogen Sulfide* - D. B. Robinson, S-S Huang and A. D. Leu, University of Alberta, Edmonton, Alberta. Project 755-B.

RR-52 *Vapor-Liquid Equilibria for Sour Water Systems with Inert Gases Present* - Jonathan L. Owens, John R. Cunningham and Grant Wilson, Wiltec Research Co., Provo, Utah. Project 805.

RR-53 *Experimental Densities and Enthalpies for Water-Natural Gas Systems* - John J. Scheloske, Kenneth R. Hall, Philip T. Eubank and James C. Holste, Texas A & M University, College Station, Texas. Project 772.

RR-53A *Thermophysical Properties Tables for Wet, Sweet and Sour Natural Gases: Data Supplement to RR-53* - John J. Scheloske, Kenneth R. Hall, Philip T. Eubank and James C. Holste, Texas A & M University, College Station, Texas. Project 772.

RR-54 *The Equilibrium Phase Behavior of Several Solute Gases in the Solvent Phenanthrene* - Robert L. Robinson, Jr., Philip J. Carlberg, John J. Heidman and Yick-Kwan Chen, Oklahoma State University, Stillwater, Oklahoma. Project 757-A.

RR-55 *Phase Equilibrium Studies for Methane/Synthesis Gas Separation: The Hydrogen-Carbon Monoxide-Methane System* - Joint research report for GPA and Gas Research Institute. Jane Huey Hong and Riki Kobayashi, Rice University, Houston, Texas. Project 757.

RR-56 *Measurement of Total Fraction Condensed and Phase Boundary for a Simulated Natural Gas* - Joint research report for GPA and Gas Research Institute. John L. Oscarson and Bert Saxey, Brigham Young University, Provo, Utah. Project 737.

RR-57 *The Phase Behavior of Two Mixtures of Methane, Carbon Dioxide, Hydrogen Sulfide and Water* - D. B. Robinson, S-S Huang, A. D. Leu and H-J Ng, University of Alberta, Edmonton, Alberta. Project 758-A.

RR-58 *The Equilibrium Phase Properties of Selected Naphthenic Binary Systems: Ethylcyclohexane-Carbon Dioxide, Ethylcyclohexane-Nitrogen and Ethylcyclohexane-Methane* - D. B. Robinson, C-J Chen and H-J Ng, University of Alberta, Edmonton, Alberta. Project 755-B.

RR-59 *Vapor-Liquid Equilibrium Measurements on the Systems N_2-Toluene, N_2-m-Xylene, and N_2-Mesitylene* - S. Laugier, D. Legret, J. Desteve, D. Richon and H. Renon, A.R.M.I.N.E.S., Paris, France. Project 755-C.

RR-60 *Liquid-Liquid-Vapor Equilibria in Cryogenic LNG Mixtures: Phase II* - James P. Kohn and Robert C. Merrill, University of Notre Dame, Notre Dame, Indiana and Kraemer D. Luks, University of Tulsa, Tulsa, Oklahoma. Project 795.

RR-6l *An Evaluation of the GPSA Engineering Data Book Volume Correction Factor Table for Light Ends* - David B. Manley, University of Missouri, Rolla, Missouri. Project 819.

RR-62 *Water-Hydrocarbon Liquid-Liquid-Vapor Equilibrium Measurements to 530 degrees F* - Joint research report for GPA and American Petroleum Institute. C. Jeffrey Brady, John R. Cunningham and Grant Wilson, Wiltec Research Co., Provo, Utah.

RR-63 *Experimental Enthalpies for Pure Toluene and Pure Methylcyclohexane* - Luis E. Cediel, Philip T. Eubank, James C. Holste and Kenneth R. Hall, Texas A & M University, College Station, Texas. Project 792-82.

RR-64C *GPA Data Bank of Selected Enthalpy and Equilibria Values*- Thomas E. Daubert, Pennsylvania State University, University Park, Pennsylvania. Projects 806 & 822.

RR-65 *Vapor-Liquid Equilibria for Sour Water Systems at High Temperatures* - Jonathan L. Owens, John R. Cunningham and Grant M. Wilson, Wiltec Research Co., Provo, Utah. Project 805-82.

RR-66 *Equilibrium Phase Composition and Hydrating Conditions in Systems Containing Methanol, Light Hydrocarbons, Carbon Dioxide and Hydrogen Sulfide* - Joint research report for GPA and Canadian Gas Processors Association. H-J. Ng and D. B. Robinson, D. B. Robinson & Assoc., Ltd., and University of Alberta, Edmonton, Alberta. Project 825-82.

RR-67 *Liquid-Liquid-Vapor Equilibria in Cryogenic LNG Mixtures: Phase III-Nitrogen Rich Systems* - James P. Kohn and Robert C. Merrill, Jr., University of Notre Dame, Notre Dame, Indiana and Kraemer D. Luks, University of Tulsa, Tulsa, Oklahoma. Project 795-82.

RR-68 *Excess Enthalpy Experimental Data-Binary Systems: Water + n-Pentane, Water + n-Hexane, Water + n-Heptane, Water + n-Octane* - C. J. Wormald, C. N. Colling, N. M. Lancaster and A. J. Sellers, University of Bristol, Bristol, England. Project 773-A-79.

RR-69 *Evaluation of GPA*SIM Computer Program with GPA Data Bank of Selected Enthalpy Values* - Thomas E. Daubert, Pennsylvania State University, University Park, Pennsylvania. Project 822-82.

RR-70 *Phase Equilibrium Studies for Processing of Gas From CO_2 EOR Projects* - Project supported jointly by the Gas Processors Association and special industry contributions. Jane Huey Hong and Riki Ko-

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bayashi, Rice University, Houston, Texas. Project 826-82.

RR-71 *Water Content of NGL in Presence of Hydrates* - Kevin A. Sparks and E. Dendy Sloan, Colorado School of Mines, Golden, Colorado. Project 775-B-82.

RR-72 *Measurement of Ethane and Propane Recovery and Total Fraction Condensed for a Simulated Natural Gas in the Retrograde Region* - John L. Oscarson, Brigham Young University, Provo, Utah. Project 8l5-81.

RR-73 *Heats of Mixing of Steam with N_2, CO_2, H_2, CH_4, and CO at High Temperatures and Pressures Using a New High Temperature Calorimeter* - Grant M. Wilson and C. Jeff Brady, Wiltec Research Co., Provo, Utah. Project 773-79 & 80.

RR-74 *The Influence of Methanol on Hydrate Formation at Low Temperatures* - H-J. Ng and D. B. Robinson, D. B. Robinson & Assoc., Ltd., Edmonton, Alberta. Project 825-83.

RR-75 *Vapor-Liquid Equilibria Measurements on the Systems N_2-n-Propylcyclohexane, CO_2-n-Propylcyclohexane, CH_4-n-Propylcyclohexane, CH_4-n-Propylbenzene and CO_2-n-Propylbenzene* - S. Laugier, P. Alali, A. Valtz, A. Chareton, F. Fontalba, D. Richon and H. Renon, A.R.M.I.N.E.S., Paris, France. Project 755-C-82.

RR-76 *Phase Equilibrium Studies for Processing of Gas from CO_2 EOR Projects-Phase II* - Project supported jointly by the Gas Processors Association and special industry contributions. Jane Huey Hong and Riki Kobayashi, Rice University, Houston, Texas. Project 826-83.

RR-77 *Evaluation of GPA*SIM Computer Program with GPA Data Bank of Selected V-L-E Data* - Thomas E. Daubert, Pennsylvania State University, University Park, Pennsylvania. Project 806-83.

RR-78 *Sulfur Compounds and Water V-L-E and Mutual Solubility MESH-H_2O; ETSH-H_2O; CS_2-H_2O; and COS-H_2O* - Paul C. Gillespie and Grant M. Wilson, Wiltec Research Co., Provo, Utah. Project 758-80.

RR-79 *Liquid-Liquid-Vapor Equilibria in Cryogenic LNG Mixtures (Phase IV - Nitrogen-Rich Systems)* - James P. Kohn and Feliciano M. Llave, University of Notre Dame, Notre Dame, Indiana and Kraemer D. Luks, University of Tulsa, Tulsa, Oklahoma. Project 795-83.

RR-80 *The Water Content of CO_2-Rich Fluids in Equilibrium with Liquid Water or Hydrate* - Kyoo Y. Song and Riki Kobayashi, Rice University, Houston, Texas. Project 775-83.

RR-81 *Evaluation of Peng-Robinson Computer Program with GPA Data Bank of Selected Enthalpy Values* - Thomas E. Daubert, Pennsylvania State University, University Park, Pennsylvania. Project 822-83.

RR-82 *Vapor-Liquid Equilibria Measurements on the Systems Ethane-Toluene, Ethane-n-Propylbenzene, Ethane-Metaxylene, Ethane-Mesitylene, Ethane-Methylcyclohexane* - S. Laugier, A. Valtz, A. Chareton, D. Richon and H. Renon, A.R.M.I.N.E.S., Paris, France. Project 755-C-83.

RR-83 *Excess Enthalpy Experimental Data Binary Systems: Water-Carbon Monoxide, Water-Carbon Dioxide* - C. J. Wormald, N. M. Lancaster, A. J. Sellars, University of Bristol, Bristol, England. Project 773-A-81.

RR-84 *Evaluation of Equi-Phase (Peng-Robinson) Computer Program with GPA Data Bank of Selected V-L-E Data* - Thomas E. Daubert, Pennsylvania State University, University Park, Pennsylvania. Project 806-84.

RR-85 *Enthalpies of Solutions of CO_2 in Aqueous Diglycolamine Solutions* - Scott P. Christensen, James J. Christensen and Reed M. Izatt, Brigham Young University, Provo, Utah. Project 821-84.

RR-86 *Properties of CO_2-Rich Mixtures* - Joint research report for GPA and Gas Research Institute. K. R. Hall, J. C. Holste, P. T. Eubank and K. N. Marsh, Texas A & M University, College Station, Texas. Project 842-84.

RR-87 *Hydrate Formation and Equilibrium Phase Compositions in the Presence of Methanol: Selected Systems Containing Hydrogen Sulfide, Carbon Dioxide, Ethane or Methane* - H-J. Ng, C-J. Chen and D. B. Robinson, D. B. Robinson & Assoc., Ltd., Edmonton, Alberta. Project 825-84.

RR-88 *Vapor-Liquid Equilibria Measurements on the Systems Ethane-n-Propylcyclohexane, Propane-Cyclohexane, Propane-n-Propylcyclohexane, Propane-n-Propylbenzene and Propane-Mesitylene* - S. Laugier, A. Valtz, A. Chareton, D. Richon and H. Renon, A.R.M.I.N.E.S., Paris, France. Project 755-C-84.

RR-89 *Vapor-Liquid Equilibrium and Condensing Curves for a Typical Gas Condensate* - H-J Ng and D. B. Robinson, D. B. Robinson & Assoc., Ltd., Edmonton, Alberta. Project 815-82 & 83.

RR-90 *Vapor-Liquid Equilibrium Measurements on the Ammonia-Water System from 313K to 589K* - Joint research report for GPA and AIChE-DIPPR. C. Gillespie, W. Vincent Wilding and Grant M. Wilson, Wiltec Research Co., Provo, Utah. Project 758-B-81.

RR-91 *Liquid-Liquid-Vapor Equilibria in Cryogenic LNG Mixtures - Phase V* -James P. Kohn and F. M. Llave, University of Notre Dame, Notre Dame, Indiana and Kraemer D. Luks, University of Tulsa, Tulsa, Oklahoma. Project 795-85.

RR-92 *The Effect of Ethylene Glycol or Methanol on Hydrate Formation in Systems Containing Ethane, Propane, Carbon Dioxide, Hydrogen Sulfide or a Typical Gas Condensate* - H-J Ng, C. J. Chen and D. B. Robinson, D. B. Robinson & Assoc., Ltd., Edmonton, Alberta. Project 825-85.

RR-93 *Gas Solubilities and Vapor-Liquid Equilibrium Measurements of H_2, CO_2 and NH_3 in Water-Phenol Mixtures from 110 degrees F to 550 degrees F* - Paul C. Gillespie, Jonathan L. Owens and Grant M. Wilson, Wiltec Research Co., Provo, Utah. Project 758-B-81 & 82.

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RR-94 *Phase Equilibrium Studies for Processing of Gas from CO_2 EOR Projects (Phase III): A. The Effect of Toluene as an Extractive Agent; B. The Effect of Methane on V-L-E Behavior in CO_2-Rich Gas n-Pentane Quasi-binary Mixtures* - Project supported jointly by GPA and special industry contributions. Jane Huey Hong and Riki Kobayashi, Rice University, Houston, Texas. Project 826-84 & 85.

RR-95 *Application of a High Pressure, Multiproperty Apparatus for the Measurement of Thermodynamic and Transport Properties to Measure the Phase and Volumetric Properties of a Supercritical Extractive System: The Carbon Dioxide-n-Hexadecane System* - Joint research report for Gas Processors Association and Gas Research Institute. T. Charoensombut-amon and Riki Kobayashi, Rice University, Houston, Texas. Project 826-85.

RR-96 *Vapor-Liquid Equilibrium and Condensing Curves in the Vicinity of the Critical Point for a Typical Gas Condensate* - H-J. Ng, C. J. Chen and D. B. Robinson, D. B. Robinson & Assoc., Ltd., Edmonton, Alberta. Project 815-A-84.

RR-97 *Excess Enthalpy Experimental Data Binary Systems: Water + Ethylene, Water + Ethane, Water + Propane, Water + n-Butane - Ternary System: Water + Methane + Carbon Dioxide* - C. J. Wormald and N. M. Lancaster, University of Bristol, Bristol, England. Project 773-B-82.

RR-98 *Glycol Vaporization Losses in Super-Critical CO_2* - Y. Adachi, Patrick Malone, Toshikuni Yonemoto and Riki Kobayashi, Rice University, Houston, Texas. Project 856-85.

RR-99 *The Water Content of CO_2-Rich Fluids in Equilibrium with Liquid Water and/or Hydrates* - Kyoo Y. Song and Riki Kobayashi, Rice University, Houston, Texas. Project 775-85.

RR-100 *Water Content of NGL in Presence of Hydrates* - Mark S. Bourrie and E. Dendy Sloan, Colorado School of Mines, Golden, Colorado. Project 775-A-83 & 84.

RR-101 *Vapor-Liquid Equilibria Measurements on the Systems Methane-Methylcyclohexane, Nitrogen-n-Propylbenzene, Hydrogen Sulfide-n-Propylbenzene, Propane-Toluene, Propane-m-Xylene and Propane-Methylcyclohexane* - A. Chareton, A. Valtz, C. Lafeuil, S. Laugier, D. Richon and H. Renon, A.R.M.I.N.E.S., Paris, France. Project 755-C-85.

RR-102 *Enthalpies of Solution of CO_2 in Aqueous Methyldiethanolamine Solutions* - Keith E. Merkley, James J. Christensen and Reed M. Izatt, Brigham Young University, Provo, Utah. Project 821.

RR-103 *Enthalpies and Phase Boundary Measurements: Equal Molar Mixtures of n-Pentane with Carbon Dioxide and Hydrogen Sulfide* - John R. Cunningham, Brigham Young University, Provo, Utah. Project 763.

RR-104 *Equilibrium Solubility of Carbon Dioxide or Hydrogen Sulfide in Aqueous Solutions of Monoethanolamine, Diglycolamine, Diethanolamine and Methyldiethanolamine* - R. N. Maddox, A. H. Bhairi, James R. Diers and P. A. Thomas, Oklahoma State University, Stillwater, Oklahoma. Project 841.

RR-105 *Vapor-Liquid Equilibrium and Condensing Curves for a Gas Condensate Containing Nitrogen* - H-J Ng, H. Schroeder and D. B. Robinson, D. B. Robinson & Assoc., Ltd., Edmonton, Alberta. Project 815-85 & 86.

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OTHER SOURCES OF INFORMATION

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AGA **American Gas Association**
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AIChE **American Institute of Chemical Engineers**
3 Park Ave.
New York, NY 10016
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AISC **American Institute of Steel Construction**
1 East Walker Dr., Ste 3100
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AISI **American Iron and Steel Institute**
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ANSI **American National Standards Institute**
25 West 43rd St., 4th Floor
New York, NY 10036
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API **American Petroleum Institute**
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Washington, D.C. 20005
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www.api.org

ASHRAE **American Society of Heating, Refrigerating & Air-Conditioning Engineers**
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www.ashre.org

ASME **American Society of Mechanical Engineers**
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ASTM **American Society for Testing and Materials**
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CTI **Cooling Tower Institute**
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www.cti.org

GTI **Gas Technology Institute**
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ISA **Instrument Society of America**
67 Alexander Drive
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(919) 549-8411
www.isa.org

ISO **International Organization for Standardization**
Case postale 56
CH-211 Geneva, Switzerland
Phone: 4122 749 0111
Fax: 4122 733 3430

NACE **National Association of Corrosion Engineers**
P.O. Box 201009
Houston, TX 77216-1009
(281) 492-0535
www.nace.org

NFPA **National Fire Protection Association**
1 Batterymarch Park
Quincy, MA 02269
(617) 770-3000
www.nfpa.org

NOAA **National Oceanic and Atmospheric Administration**
National Climatic Center
151 Patton Ave., Rm 120
Asheville, NC 28801-5001
Phone: (828) 271-4800
Fax: (828) 271-4876
www.ncdc.info@noaa.gov

NPGA **National Propane Gas Association**
1150 17th St. NW, Ste. 310
Washington, DC 20036
(202) 466-7200
www.npga.org

SPE **Society of Petroleum Engineers**
Box 833836
Richardson, TX 75083
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Tech Street
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TEMA **Tubular Exchanger Manufacturers Association**
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SECTION 2

Product Specifications

This Section contains the following specifications for natural gas liquid products:

Fig. 2-1: GPA specifications for commercial propane, commercial butane, commercial butane-propane mixtures, and Propane HD-5.[1]

Fig. 2-2: GPA specifications for natural gasoline.[2]

These are "official" industry standards, representing a broad industry consensus for minimum quality products. Producers, purchasers, or pipeline companies may adopt variations of these specifications.

The gas plant designer and operator, as well as purchasers, will also be concerned with specifications for other plant products, including residue gas, raw mix streams, ethane, propane, ethane-propane mixes, normal butane, iso-butane, and plant condensate. Although there are no "official" industry specifications for normal butane, common commercial transactions for normal butane stipulate that the product shall meet all specifications for commercial butane and, in addition, be composed of a minimum of 95 volume percent normal butane. Common commercial specifications for iso-butane stipulate that the product contain a minimum of 95 volume percent iso-butane, and also meet all specifications for commercial butane.

Likewise, there are no industry standard specifications for ethane or ethane-propane (EP) mixes. However, GPA Technical Section C on product specifications has provided a summary of typical quality criteria in industry use as shown in Fig. 2-3.

Quality specifications for natural gas have historically been individually negotiated and prescribed in contracts between purchasers or pipeline companies and the producer or processor. Specification parameters for pipeline quality natural gas may include heating value, composition, contaminants, water content, and hydrocarbon dew point. Specification limits for these parameters may vary widely depending on the pipeline system, climatological conditions, end use, and other factors. Example pipeline quality gas specification parameters are shown in Fig. 2-4.

FIG. 2-1

GPA Liquefied Petroleum Gas Specifications
(This Table Extracted From GPA Standard 2140-97)

Product Characteristics	Product Designation				
	Commercial Propane	Commercial Butane	Commercial B-P Mixtures	Propane HD-5	Test Methods
Composition	Predominantly propane and/or propylene.	Predominantly butanes and/or butylenes.	Predominantly mixtures of butanes and/or butylenes with propane and/or propylene.	not less than 90 liquid volume percent propane; not more than 5 liquid volume percent propylene.	ASTM D-2163-91
Vapor pressure at 100°F, psig, max. at 37.8°C, kPa (ga), max.	208 1434	70 483	208 1434	208 1434	ASTM D-1267-95
Volatile residue: temperature at 95% evaporation, °F, max. or °C, max. butane and heavier, liquid volume percent max. pentane and heavier, liquid volume percent max.	 –37 –38.3 2.5 —	 36 2.2 — 2.0	 36 2.2 — 2.0	 –37 –38.3 2.5 —	 ASTM D-1837-94 ASTM D-2163-91 ASTM D-2163-91
Residual matter: residue on evaporation of 100 ml, max. oil stain observation	 0.05ml pass (1)	 — —	 — —	 0.05 ml pass (1)	 ASTM D-2158-92 ASTM D-2158-92
Corrosion, copper strip, max.	No. 1	No. 1	No. 1	No. 1	ASTM D-1838-91 (Note A)
Total sulfur, ppmw	185	140	140	123	ASTM D-2784-92
Moisture content	pass	—	—	pass	GPA Propane Dryness Test (Cobalt Bromide) or D-2713-91
Free water content	—	none	none	—	—
(1) An acceptable product shall not yield a persistent oil ring when 0.3 ml of solvent residue mixture is added to a filter paper in 0.1 increments and examined in daylight after 2 minutes as described in ASTM D-2158.					
NOTE A: "This method may not accurately determine the corrosivity of the liquefied petroleum gas if the sample contains corrosion inhibitors or other chemicals which diminish the corrosivity of the sample to the copper strip. Therefore, the addition of such compounds for the sole purpose of biasing the test is prohibited."					

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FIG. 2-2

GPA Natural Gasoline Specifications and Test Methods

Scope: These specifications state the required properties of Natural Gasoline. Natural gasoline is a mixture of liquid hydrocarbons extracted from natural gas, composed principally of pentanes and heavier hydrocarbons, although varying amounts of butanes may be included, depending on the commercial grade.

Natural gasoline is defined further for commercial purposes by the following:

Product Characteristic	Specification	Test Method
Reid Vapor Pressure	10-34 pounds	ASTM D-323
Percentage evaporated at 140°F	25-85	ASTM D-216
Percentage evaporated at 275°F	Not less than 90	ASTM D-216
End Point	Not more than 375°F	ASTM D-216
Corrosion	Not more than classification 1	ASTM D-130 (modified)
Color	Not less than plus 25 (Saybolt)	ASTM D-156
Reactive Sulfur	Negative, "sweet"	GPA 1138

In addition to the above general specifications, natural gasoline shall be divided into 24 possible grades on the basis of Reid vapor pressure and percentage evaporated at 140°F. Each grade shall have a range in vapor pressure of four pounds, and a range in the percentage evaporated at 140°F of 15%. The maximum Reid vapor pressure of the various grades shall be 14, 18, 22, 26, 30, and 34 pounds, respectively. The minimum percentage evaporated at 140°F shall be 25, 40, 55, and 70, respectively. Each grade shall be designated by its maximum vapor pressure and its minimum percentage evaporated at 140°F, as shown in the following table:

Grades of Natural Gasoline
Percentage Evaporated at 140°F

Reid Vapor Pressure, psi	25%	40%	55%	70%
34	Grade 34-25	Grade 34-40	Grade 34-55	Grade 34-70
30	Grade 30-25	Grade 30-40	Grade 30-55	Grade 30-70
26	Grade 26-25	Grade 26-40	Grade 26-55	Grade 26-70
22	Grade 22-25	Grade 22-40	Grade 22-55	Grade 22-70
18	Grade 18-25	Grade 18-40	Grade 18-55	Grade 18-70
14	Grade 14-25	Grade 14-40	Grade 14-55	Grade 14-70
10				

LP-GAS SPECIFICATION PARAMETERS

LP-gas specifications of GPA Standard 2140, shown in Fig. 2-1, are the industry standards in the United States. International specifications, adopted in ISO 9162, are shown in Fig. 2-5.[14]

In many cases, specification parameters for LP-gas are based on simple "pass-fail" test methods that can be performed quickly and easily by field personnel. These specifications and test methods are intended to assure products that can be safely handled in transport systems, and that will perform adequately and safely in their end-use markets. Unfortunately, many of these tests tell the design engineer or plant operator little about product composition or quantitative limits. The following discussion is intended to provide an indication of product composition and quantitative limits imposed by these industry specifications.

FIG. 2-3

Representative Quality Criteria for Ethane Streams

	Weight Percent		
	Low	High	Common
High-Ethane Raw Streams			
Methane and Lighter	1.0	5.0	1.0
Ethane, Propane, Isobutane & Heavier	Remainder	Remainder	Remainder
Contaminant Maximums:			
Hydrogen Sulfide	No. 1 Strip	50 ppmw	No. 1 Strip
Carbon Dioxide	100 ppmw	3,500 ppmw	500 ppmw
Sulfur	5 ppmw	200 ppmw	200 ppmw
Oxygen	300 ppmw	–	–
Moisture	75 ppmw	No free	No free
Ethane-Propane Mixes			
Methane and Lighter	0.6	1.0	0.6
Ethane	20	80	50
Propane	20	80	50
Isobutane & Heavier	0.2	4.5	4.5
Contaminant Maximums:			
Hydrogen Sulfide	No. 1 Strip	No. 1 Strip	No. 1 Strip
Carbon Dioxide	500 ppmw	3,000 ppmw	500 ppmw
Sulfur	5 ppmw	143 ppmw	100 ppmw
Oxygen	500 ppmw	1,000 ppmw	1,000 ppmw
Moisture	10 ppmw	No free	50 ppmw
High-Purity Ethane			
Methane and Lighter	1.5	2.5	2.5
Ethane	90.0	96.0	90.0
Propane	6.0	15.0	6.0
Isobutane & Heavier	0.5	3.0	2.0
Contaminant Maximums:			
Hydrogen Sulfide	6 ppmw	10 ppmw	10 ppmw
Carbon Dioxide	10 ppmw	5,000 ppmw	10 ppmw
Sulfur	5 ppmw	70 ppmw	50 ppmw
Oxygen	5 ppmw	5 ppmw	5 ppmw
Moisture	13 ppmw	No free	76 ppmw

Vapor Pressure

Vapor pressure is a critical specification that must be observed for safe and efficient utilization of propane, butane, and butane/propane mixtures in domestic and commercial installations, and to comply with various regulations governing transport vessels and cylinders.[3]

The GPA vapor pressure specification limit for propane meets the requirements of U.S. Department of Transportation regulations by effectively limiting the ethane content of commercial propane and propane HD-5 to a maximum of approximately 7 volume percent. Any appreciable quantity of propylene, permitted in commercial propane only, would necessarily reduce the amount of permissible ethane due to the higher vapor pressure of propylene relative to that of propane. Likewise, variations in the butane content of propane, limited to 2.5 volume percent, will impact the amount of ethane permitted by the vapor pressure specification.

Moisture Content

Moisture in propane must be controlled to very low concentrations to avoid hydrate formation in pipelines and freezing in tanks, regulating equipment, and other equipment in the distribution system.

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FIG. 2-4

Example Pipeline Quality Natural Gas
Adapted from AGA Report No. 4A, May 2001

	Minimum	Maximum
Major and Minor Components, mol%		
Methane	75	–
Ethane	–	10
Propane	–	5
Butanes	–	2
Pentanes and heavier	–	0.5
Nitrogen and other inerts	–	3
Carbon Dioxide	–	2-3
Total Diluent Gases	–	4-5
Trace Components		
Hydrogen sulfide	–	0.25-0.3 gr/100 scf
Mercaptan sulfur	–	–
Total sulfur	–	5-20 gr/100 scf
Water vapor	–	4.0-7.0 lb/mmcf
Oxygen	–	1.0%
Other Characteristics		
Heating value, Btu/scf-gross saturated	950	1150
Liquids: Free of liquid water and hydrocarbons at delivery temperature and pressure.		
Solids: Free of particulates in amounts deleterious to transmission and utilization equipment. (3-15 microns, max.)		

Although a properly designed and operated dehydration system produces very dry propane, moisture can and does enter the transportation and distribution system at many points, such as storage tanks, loading racks, and transport vessels.

There are two recognized methods for determining acceptable levels of moisture in propane products: the GPA Cobalt Bromide Test, and the Valve Freeze method (ASTM D-2713). Both are "pass-fail" tests that provide qualitative determinations of commercially "dry" propane, but neither method yields quantitative measures of moisture in the product.

The Cobalt Bromide test is based on the work of Hachmuth[4], which determined acceptable levels of moisture in commercial equipment, and correlated these levels with results of the test procedure. The test is based on observation of color changes of cobalt bromide salt caused by the humidity of the gas or vapor surrounding it. In practice, the cobalt bromide is supported on white cotton wadding and exposed to a stream of propane vapors chilled to 32°F. The color of the cobalt bromide changes from green to lavender at about 30% relative humidity, indicating "wet" propane. Propane-water system data[7,8,11] indicate that the water content of saturated propane vapors at 32°F is approximately 530 ppmw. The water content of saturated propane liquid is approximately 35 ppmw at 32°F. At 30% saturation at 32°F, commercially "dry" propane as measured by the Cobalt Bromide test will be about 159 ppmw in the vapors and about 10 ppmw in the liquid. Based on these specification limits at 32°F, Fig. 2-6 gives maximum allowable water content of liquid propane at other system temperatures.

The valve freeze method was developed to detect excessive moisture in liquid propane, and is preferred by some over the Cobalt Bromide test. The test device is a specially constructed and calibrated orifice valve designed to simulate expansion of propane through a pressure regulator. A liquid sample of the product to be tested is passed through the valve at a preset flow rate. The time required for the valve to freeze and interrupt flow due to moisture in the product determines whether or not the product is commercially "dry." Test data reveals that a freeze-off time of more than 60 seconds indicates less than 30 ppmw moisture in the liquid product. The method is not applicable to propane products containing anti-freeze agents such as methanol. It is also affected by the temperature of the liquid sample.

A third method, the Bureau of Mines dew point tester, is a simple field test still used by some, but is not recommended because its accuracy is dependent on many poorly controlled variables, such as temperature and pressure of the system. This method was originally developed by the U.S. Bureau of Mines and is still used as a field method to determine moisture content in natural gas systems.

Butane specifications stipulate "no free water." Since butane cannot be used in vapor withdrawal systems at temperatures below its boiling point, water content is not detrimental for most butane uses.

Sulfur Content

Certain compounds of sulfur must be controlled to extremely low levels in LP-gases to prevent corrosion in distribution facilities. The standard test for corrosivity is the Copper Corrosion Test (ASTM D-1838). In this test, a polished copper strip is immersed in the product sample for one hour at 100°F. The test strip is then rated by comparison with the four color classifications of ASTM Copper Corrosion Standards:

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FIG. 2-5

Specifications for Liquefied Petroleum Gases
This table extracted from ISO 9162-1989

Characteristics	Method of Test	Commercial Propane ISO-F-LP	Commercial Butane ISO-F-LB
Gauge vapor pressure at 104°F, psig, max.	ISO 4256[2] or ISO 8973	225[1]	75[1]
Volatility			
C_2 hydrocarbons		Report value[3]	
C_4 hydrocarbons, mole %, max.	ISO 7941	7,5[1]	
C_5 hydrocarbons, mole %, max.	ISO 7941	0,2[6]	2,5[6]
Unsaturated hydrocarbons, mole %	ISO 7941	Report value[4]	Report Value[4]
Dienes, mole %, max.	ISO 7941	0,5[6]	0,5[6]
Residual matter	[5]	[5]	[5]
Corrosiveness to copper, max.	ISO 6251	1	1
Sulfur, mg/kg, max.	[7]	50[6]	50[6]
Hydrogen sulfide	ISO 8819	Pass	Pass
Free water content		None[8]	None[8]

1) Certain national standards and/or regulations may prescribe other limits.

2) In case of dispute about the vapor pressure, ISO 4256 shall prevail.

3) Physical limitations of transport and storage facilities shall be considered for refrigerated transport; a common limit is 2 mole % maximum.

4) Certain national standards may prescribe a limitation on unsaturated hydrocarbons.

5) It has not been possible to take a decision on a value for inclusion in this specification because there is no suitable ISO test method and no satisfactory correlation between the test results obtained using methods developed in different countries. Limits will be specified when a suitable test method has been adopted. Until a suitable test method has been adopted, the user of this specification is referred to the introductory discussion and proposed test method discussed in ISO 9162-1989. Report the value for the sample concerned, and the method used to determine it, as additional information.

6) Certain national standards may prescribe other limits.

7) An ISO method specific to liquefied petroleum gas is being developed. Until such an ISO method is available, ISO 4260 is recommended as the referee method. Other methods specified in national standards, and having a valid precision statement, are acceptable for quality control purposes.

8) The presence of water shall be determined by visual inspection. For shipments of refrigerated liquefied petroleum gases at their atmospheric boiling point, it is essential that the water content is below the saturation level at this temperature as determined by a method agreed between buyer and seller.

No.1 Slight tarnish (light to dark orange)

No.2 Moderate tarnish (red, lavender, brassy gold)

No.3 Dark tarnish (magenta, red, green)

No.4 Corrosion (black, dark gray, brown)

Specifications for LP-gas products stipulate that the product must not exceed a No.1 strip, indicating product that is not corrosive to copper or brass fittings in the distribution system.

FIG. 2-6

Maximum Water Content of Dry Commercial Liquid Propane*

Temp., °F	H_2O, ppmw	% Saturated
100	21	8
80	18	11
60	15	16
40	11	24
32	10	30

* These numbers are approximate

LP-gas from some sources may contain trace amounts of carbonyl sulfide (COS), which alone is non-corrosive. However, COS can hydrolyze in the presence of water, either liquid or vapor, to corrosive hydrogen sulfide (H_2S). The copper strip corrosion test requires that the test cylinder be wetted with approximately 1 ml of water prior to introduction of the sample, which is intended to hydrolyze the COS into H_2S and ensure detection of corrosive product. If COS is suspected, additional product testing (See GPA Standard 2290) and treatment is recommended to prevent its introduction into the distribution system.

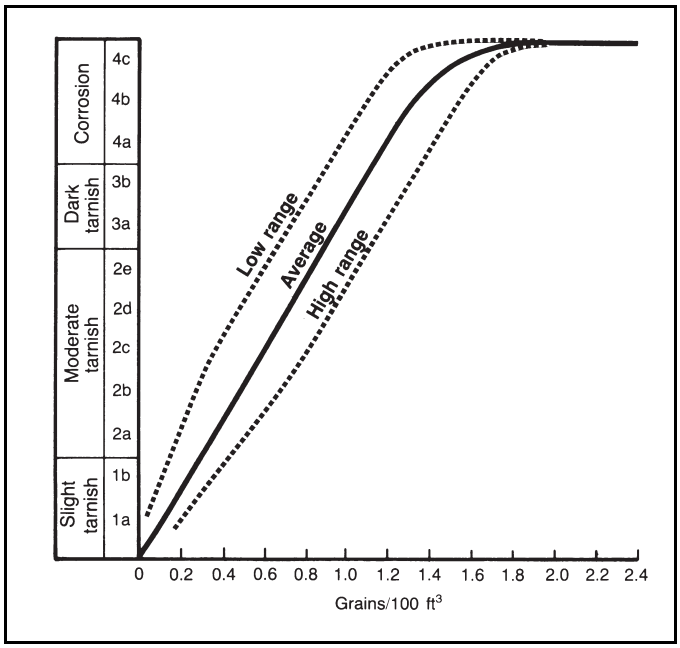
Sulfur chemistry is extremely complex, and little scientific work has been done to correlate copper strip test results, either quantitatively or by species, with the multitude of sulfur compounds that may be encountered in natural gas liquid products. However, specific test data by Pyburn et al[6] indicate that hydrogen sulfide concentrations of 0.55 ppmw or more in propane almost certainly will produce a failure of the copper strip test; as little as 0.3 ppmw might fail, depending on interpretation of the test. These data also suggest that the threshold limit of elemental sulfur is approximately 5-10 ppmw, although concentrations as low as 2 ppmw may cause failure of the copper strip test.

Additional data suggest that elemental sulfur in the presence of hydrogen sulfide results in a synergistic effect that will cause failure of the copper strip test with trace concentrations

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FIG. 2-7

Concentration H_2S vs. Copper Strip Produced



of either contaminant. Specifically, concentrations of 0.5 ppmw elemental sulfur, together with 0.28 ppmw hydrogen sulfide in propane were observed to fail the corrosion test. The threshold value of hydrogen sulfide may be as low as 0.12 ppmw in the presence of 0.5 ppmw elemental sulfur.

Fig. 2-7 is based on field test data on corrosivity limits in high ethane content mixtures. The curves are approximate and should not be used for design purposes. Perry[10] states that a No.1 copper strip is generally thought to represent hydrogen sulfide concentrations below 1-2 ppmw. As noted above, other research[6] indicates a synergistic corrosion effect of extremely small amounts of H_2S if free sulfur is present in the stream.

From other observations, Pyburn et al[6] report the following additional conclusions about corrosive elements in LP-gas that may affect the copper corrosion test:

1. H_2S and elemental sulfur are the corrosive agents.
2. H_2S and elemental sulfur act synergistically on the copper corrosion test.
3. Polysulfides may contribute to copper corrosion.
4. Mercaptans in the presence of elemental sulfur enhance corrosion.
5. Mercaptans in the presence of hydrogen sulfide inhibit corrosion.
6. To be safe, all sulfur compounds should be reduced to the lowest possible level.
7. *No hydrogen sulfide should be present.*
8. If any sulfur compounds are present, the product should be dry.
9. Mixing of propane or other products from two sources, both of which passed the copper corrosion test, may result in a failure of the mixture to pass the test.
10. Changes in sulfur species may occur during transit under some conditions.

GPA specifications permit a maximum of 123 ppmw total sulfur in Propane HD-5; 185 ppmw in commercial propane; and 140 ppmw in butanes. It should be noted, however, that these same specifications require that the product not exceed the No.1 copper strip. If these two specifications are in conflict, the more stringent will govern.

To convert from parts per million by weight (ppmw) of H_2S to grains per 100 ft^3, measured at 60°F and 14.7 psia, use the following formula for any hydrocarbon:

$$\text{grains}/100\ ft^3 = (\text{ppmw}_{H2S})(MW_{gas})(0.001845) \qquad \textbf{Eq 2-1}$$

Volatile Residue

The volatile residue specification, coupled with the vapor pressure limit, assures essentially single-component products for propane and butane, and essentially two-component products for butane-propane mixtures. The predominant volatile residue in propane is butane, limited to 2.5 volume percent; the predominant volatile residue in butane is pentane or heavier hydrocarbons, limited to 2 volume percent. These compositional limits assure that 95 volume percent of the product will be vaporized at temperatures slightly above the normal boiling point of the product, which may be confirmed by a field distillation test.

Non-volatile Residue

The non-volatile residue specification imposes a rigid limit on extraneous contaminants boiling above 100°F, which may result in fouling of vaporizers used in liquid withdrawal systems. The principal sources of such contaminants are typically compressor oils, valve lubricants, plasticizers from rubber hoses, corrosion inhibitors, or other heavy petroleum products from pumps, pipelines, or storage vessels that may be used in multiple services. The specification limit is 0.05 ml per 100 ml of test sample, or 500 ppmv. The common residues in the gasoline boiling range usually present little problem in utilization systems. However, very heavy residues can be present, and must be identified and eliminated from the system.

Non-Specification Contaminants

Although the range of possible contaminants in LP-gas is very great, serious problems are rare, but can occur. The most serious contaminants encountered are ammonia (See GPA Standard 2187), fluorides, and organic fluorides. NFPA 58[12] stipulates that LP-gas "shall not contain ammonia," because of its propensity to induce stress corrosion cracking in copper and brass fittings commonly used in propane utilization systems. Fluorides and organic fluorides are possible contaminants in propane produced from refinery operations, notably HF alkylation processes, which dictates that production from such units should be thoroughly and continuously monitored to prevent contamination of the distribution system.

Natural gas liquids destined for petrochemical feedstocks may also be subject to additional contractual specification limits, depending on special down-stream processing requirements. Some of these requirements may include limits on dienes, isomers, alcohols, halides and other trace contaminants.

Odorization

Regulatory codes applicable to the distribution of natural gas and LP gas require these gases to be odorized such that the gas is readily detectable by a distinct odor, at one-fifth of the lower limit of flammability of the gas in air.[12] An excellent reference covering

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- Odorization regulations
- Chemical and physical properties of odorants
- Odorization system design considerations
- Odorization equipment and stations
- Monitoring methods
- Safety and environmental
- Odorant stability and olfactory response

has been published by the American Gas Association.[13]

Iso 13734 – "Natural Gas – Organic Sulfur Compounds used as Odorants – Requirements and Test Methods" will be another reference when it is published.

REFERENCES

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11. Kahre, L.C., Phillips Petroleum Company Project Report DA-2, 1972.
12. National Fire Protection Association (NFPA) No. 58, "Standard for the Storage and Handling of Liquefied Petroleum Gases."
13. AGA Odorization Manual, Dec. 2000.
14. ISO 9162 (1989), "Specifications for Liquefied Petroleum Gases," International Standards Organization.

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SECTION 3

Measurement

The information presented in this section provides sufficient information for determining flow quantities with a reasonable degree of accuracy, but not necessarily to the accuracy desired for custody transfer. Agreement of acceptable accuracy for custody transfer should be between the parties involved, and supplemental information and procedures may be required, such as the API Manual of Petroleum Measurement Standards.

FIG. 3-1

Nomenclature

- C = Pitot tube flow coefficient
- C' = the product of multiplying all orifice correction factors
- CNT = volume indicated by the number of pulses or counts
- C_{pl} = liquid pressure correction factor. Correction for the change in volume resulting from application of pressure. Proportional to the liquid compressibility factor, which depends upon both relative density and temperature.
- C_{ps} = correction factor for effect of pressure on steel. See API Manual of Petroleum Measurement Standards, Chapter 12, Section 2
- C_g = gravity correction factor for orifice well tester to change from a gas specific gravity of 0.6
- C_{tl} = liquid temperature correction factor. Proportional to the thermal coefficient which varies with density and temperature
- C_{ts} = correction factor for effect of temperature on steel
- C_u = velocity of sound in the gas non-flowing condition.
- d = orifice diameter, in.
- D = internal pipe diameter of orifice meter run or prover section, in.
- DL = Minimum downstream meter tube length, in.
- D_p = the difference between the flowing pressure and the equilibrium vapor pressure of the liquid.
- D_u = diameter of the meter bore.
- ρ = flowing fluid density, lb/ft^3
- e = orifice edge thickness, in.
- E = orifice plate thickness, in.
- E_m = modulus of elasticity for steel $[(30)(10^6)]$ psi
- F = liquid compressibility factor
- F_a = orifice thermal expansion factor. Corrects for the metallic expansion or contraction of the orifice plate. Generally ignored between 0° and 120°F
- F_g = specific gravity factor applied to change from a specific gravity of 1.0 (air) to the specific gravity of the flowing gas
- F_{gt} = gravity-temperature factor for liquids
- F_c = orifice calculation factor
- F_n = numeric conversion factor
- F_{na} = units conversion factor for pitot tubes
- F_{pb} = pressure base factor applied to change the base pressure from 14.73 psia
- F_{pm} = pressure factor applied to meter volumes to correct to standard pressure. See API Manual of Petroleum Measurement Standards, Chapter 12, Section 2
- F_{pv} = supercompressibility factor required to correct for deviation from the ideal gas laws = $\sqrt{1/Z}$
- F_s = steam factor
- F_{sl} = orifice slope factor
- F_{tb} = temperature base factor. To change the temperature base from 60°F to another desired base
- F_{tf} = flowing temperature factor to change from the assumed flowing temperature of 60°F to the actual flowing temperature
- F_{tm} = temperature correction factor applied to displacement meter volumes to correct to standard temperature. See API Manual of Petroleum Measurement Standards, Chapter 12, Section 2
- G, G_1 = specific gravity at 60°F
- G_f = specific gravity at flowing temperature
- H = pressure, inches of mercury
- h_m = differential pressure measured across the orifice plate in inches of mercury at 60°F
- h_w = differential pressure measured across the orifice plate in inches of water at 60°F
- $\sqrt{h_w P_f}$ = pressure extension. The square root of the differential pressure times the square root of the absolute static pressure
- k = ratio of the specific heat at constant pressure to the specific heat at constant volume
- K = a numerical constant. Pulses generated per unit volume through a turbine, positive displacement, coriolis or ultrasonic meter
- Key = $F_n (F_c + F_{sl})$ = orifice factor
- L = distance between upstream and downstream transducer.
- LTB = Length of tube bundle, in flow conditioner, in. (See Fig. 3-3)
- MF = meter factor, a number obtained by dividing the actual volume of liquid passed through the meter during proving by the volume registered by the meter
- P = pressure, psia

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FIG. 3-1 (Cont'd)

P_b = base pressure, psia
Pf_1, Pf_2 = static pressure at either the upstream(1) or downstream(2) pressure tap, psia
P_p = operating or observed pressure, psig.
P_s = pressure at which the base volume of a meter prover was determined, usually 0 psig.
P_1/P_2 = pressure ratio
Q = gas flow rate, cu ft/day
Q_h = rate of flow, std. cu ft/hr or gal./hr
R_h = maximum differential range, in. of water
R_p = maximum pressure range of pressure spring, psi
S = square of supercompressibility
SEP = Separation distance between elbows, in. (See Fig. 3-9)
SPU = Signal Processing Unit, the electronic microprocessor system of the multi-path ultrasonic meter.
t1 = transit time from upstream transducer to downstream transducer.
t2 = transit time from downstream transducer to upstream transducer.
T_b = base temperature, °F
T_f = flowing temperature, °F
T_{meas} = reference temperature of the orifice plate bore, °F
UL = Minimum upstream meter tube length, in. (See Fig. 3-9)
UL1 = See Fig. 3-9
UL2 = See Fig. 3-9
UM = ultrasonic meter for measuring gas flow rates
V = velocity of flowing gas.
VOS = velocity of sound
X = axial distance separating transducers
Y = expansion factor to compensate for the change in density as the fluid passes through an orifice
Y_{CR} = critical flow constant
Z = compressibility factor
β = ratio of the orifice or throat diameter to the internal diameter of the meter run, dimensionless

The American Petroleum Institute Manual of Petroleum Measurement Standards is the source of a large portion of the information presented in this Section. Copies of the complete API Manual of Petroleum Measurement Standards or specific chapters and sections can be ordered from:

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5	Metering
6	Metering Assemblies
7	Temperature Determination
8	Sampling
9	Density Determination
10	Sediment and Water
11	Physical Properties Data
12	Calculation of Petroleum Quantities
13	Statistical Aspects of Measuring and Sampling
14	Natural Gas Fluids Measurement
15	Guidelines for use of SI units
16	Petroleum Measurement by Weight
17	Marine Measurement
18	Custody Transfer
19	Evaportion Loss Measurement
20	Allocation Measurement of Oil and Gas

Chapter 14 is of particular interest to Gas Processors because it applies specifically to the measurement of gas and liquefied gas products. Chapter 14 is further divided as follows:

SECTION	API TITLE
14.1	Sampling of Natural Gas
14.2	Compressibility Factors of Natural Gas
14.3	Concentric, Square-Edged Orifice Meters (AGA Report No. 3)
14.4	Converting Mass of Natural Gas Liquids and Vapors to Equivalent Liquid Volumes (GPA 8173)
14.5	Calculation of Gross Heating Value, Specific Gravity, and Compressibility of Natural Gas Mixtures from Compositional Analysis (GPA 2172)
14.6	Installing and Proving Density Meters
14.7	Mass Measurement of Natural Gas Liquids (GPA-8182)
14.8	Liquefied Petroleum Gas Measurement

FLOW CALCULATION GUIDE

The Flow Calculation Guide (Fig. 3-2) is the suggested starting point for any fluid measuring determination. This chart identifies the necessary data to determine flows of gas, liquid, and steam using mechanical meters or differential pressure devices. The equations to determine either volumetric or mass quantities are shown in Fig. 3-4.

GAS MEASUREMENT

Orifice-Meter Measurement

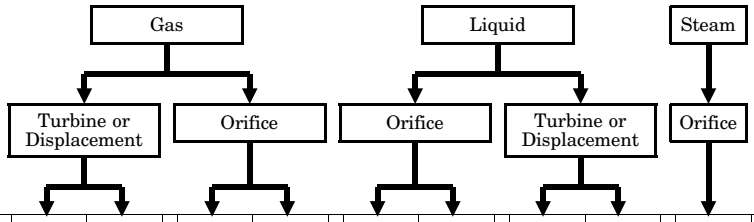
The most commonly used differential measurement device, the orifice meter, is widely accepted for use in measuring volumes of liquids or vapors.

The procedures presented in this chapter for calculating flow by use of an orifice meter are designed to provide *approximate* solutions using a hand calculator or equivalent, and do not include the rigorous, iterative solution procedures required when using the Reader-Harris/Gallagher flow equation recommended for accurate, custody transfer calculations with computing equipment. The procedures for applying the Reader-Harris/Gallagher equation can be found in Chapter 14.3 of the API Manual of Petroleum Measurement Standards.

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FIG. 3-2

Flow Calculation Guide



Factors	Units	Mass	Vol	Mass	Vol	Mass	Vol	Mass	Vol	Mass	Fig. #
Units of Measurement	⇒	lb/hr	scf/hr	lb/hr	scf/hr	lb/hr	gph	lb/hr	gph	lb/hr	
Square Root of Differential	$\sqrt{h_w}$	—	—	•	•	•	•	—	—	•	—
Square Root of Static Pressure	$\sqrt{P_f}$	—	—	—	•	—	—	—	—	—	—
Orifice Factor (KEY)	Fn • $(F_{c+}F_{sl})$	—	—	•	•	•	•	—	—	•	3-13
Pressure Base Factor	F_{pb}	—	•	—	•	—	—	—	—	—	3-4
Flowing Temperature Factor	F_{tf}	—	—	—	•	—	—	—	—	—	3-4
Temperature Factor	F_{tm}	—	•	—	—	—	—	—	—	—	3-1
Temp Correction Factor (Liquids)	C_{tl}	—	—	—	—	—	—	—	•	—	3-1
Pressure Correction Factor (Liquids)	C_{pl}	—	—	—	—	—	—	—	•	—	3-4
Pressure Factor	F_{pm}	—	•	—	—	—	—	—	—	—	3-1
Supercompressibility Factor	F_{pv}	—	—	—	•	—	—	—	—	—	23-4
Square of Supercompressibility Factor	S	—	•	—	—	—	—	—	—	—	3-1
Density	ρ	•	—	—	—	—	—	•	—	—	3-1
Square Root of Density	$\sqrt{\rho}$	—	—	•	—	•	—	—	—	•	—
Specific Gravity Factor — Gas	F_g	—	—	—	•	—	—	—	—	—	3-4
Gravity-Temperature Factor — Liquid	F_{gt}	—	—	—	—	—	•	—	—	—	3-4
Meter Factor	MF	•	•	—	—	—	—	•	•	—	3-1
Count (Volume)	CNT	•	•	—	—	—	—	•	•	—	3-1
Constant				1.0618		1.0618					
Steam Factor	F_s									•	3-27/28
Expansion Factor	Y	—	—	•	•	—	—	—	—	•	3-1
Temperature Base Factor	F_{tb}	—	•	—	•	—	—	—	—	—	3-4
Orifice Thermal Expansion Factor	F_a	—	—	•	•	•	•	—	—	•	3-4

Notes:

1. This guide is intended for use in obtaining approximate flows when used in conjunction with data contained in this section as referenced in the far right hand column.
2. To obtain flow, substitute areas containing dots with known numbers and multiply top to bottom.
3. The number of factors used may vary depending on method of calculation in specific application, content of flowing stream, and individual contractual agreements.
4. Factors appearing in shaded areas are not generally necessary for calculating approximate flows.
5. The factors F_{pv}, S, C_{tl}, C_{pl}, Y, and F_a must be obtained for the specific substance being measured.

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The orifice meter consists of static pressure and differential-pressure recording gauges connected to an orifice flange or orifice fitting. The orifice meter tube (meter run) consists of upstream and downstream sections of pipe for which size and tolerance have been determined through calculation and which conform to specifications set forth in API Chapter 14.3 (AGA Report No. 3).

The orifice plate is held perpendicular to flow by flanges or a fitting. Bore, circumference, edge sharpness, and other tolerances must meet specifications as set forth in API Chapter 14.3 (AGA Report No. 3).

Orifice Flanges (Fig. 3-5a) — Orifice flanges require that the line be shut down and depressured in order to inspect or change the orifice plate. The flange bolts are loosened and removed. The flanges are spread by use of "jack" bolts, and the plate is removed.

When orifice flanges are used, the pressure tap hole placement may be determined by measuring from the face of the flange to the pressure tap hole center. Allowable variations are shown in Figure (3-7).

Single Chamber Orifice Fitting (Fig. 3-5b) — This fitting also requires that the line be shut down and depressured in order to inspect or change the orifice plate. However, this fitting does not require breaking apart the flanges. Instead, the bolts are loosened on the cover plate and the cover plate removed. The orifice plate holder and orifice plate are then removed from the fitting. These fittings provide precise alignment of the orifice plate.

Dual-Chamber Orifice Fitting (Fig. 3-5c) — This fitting allows the removal and inspection of an orifice plate while the line remains under pressure. It allows the orifice plate holder and orifice plate to be raised into the upper cavity of the fitting by the use of a crank handle. A valve is then closed to separate the upper cavity from the lower cavity of the fitting. The upper cavity is then depressured, the top cover plate removed, and the orifice plate cranked out.

Orifice Plates — The minimum, maximum, and recommended thicknesses of orifice plates for various pipe sizes are given in Figure 3-6. Also shown in this figure are maximum allowable differential pressures for stainless steel plates of the recommended thickness at a maximum temperature of 150 degrees F.

The thickness of the orifice plate at the orifice edge (e) shall meet the following:

The minimum thickness is defined by $e \geq 0.01d$ or $e > 0.005$ in. whichever is greater.

The maximum thickness is defined by $e \geq 0.02d$ or $\leq 0.125d$ whichever is smaller

If the thickness of the orifice plate must be greater than permitted by these limitations, the downstream edge shall be cut away (beveled or recessed) at an angle of 45° ± 15° or less to the face of the plate, leaving the thickness of the orifice edge within these requirements. All orifice plates which are beveled should have the square-edge side (i.e., the side opposite the beveling) stamped "inlet" or the beveled side stamped "outlet".

The upstream face of the orifice plate shall be flat and perpendicular to the axis of the meter tube, when in position between the orifice flanges or in the orifice fitting. Any plate that does not depart from flatness along any diameter by more than 0.010 inch per inch of the dam height, (D-d)/2, shall be considered flat. See Figure 3-8.

The upstream edge of the orifice shall be square and sharp so that it will not show a beam of light when checked with an orifice edge gauge, or alternately will not reflect a beam of light when viewed without magnification. The orifice shall not have a burred or feathered edge. It shall be maintained in this condition at all times. Moreover, the orifice plate shall be kept clean at all times and free from accumulation of dirt, ice, and other extraneous material. Orifice plates with small nicks in the edge can be expected to increase the flow measurement uncertainty.

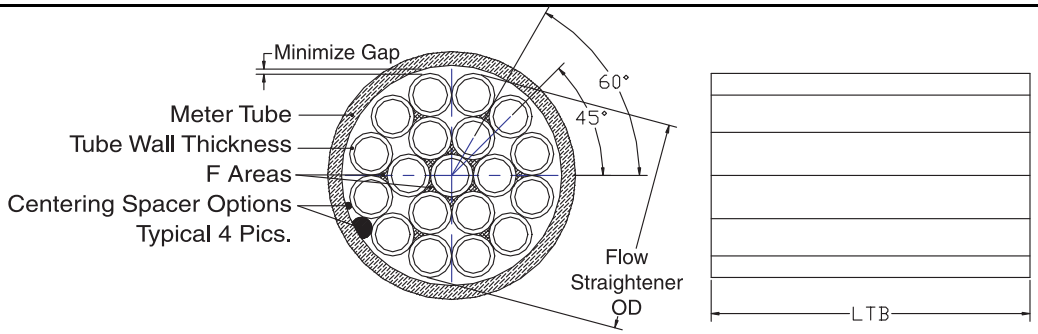
Meter Tubes — The term "meter tube" shall mean the straight upstream pipe ahead of the orifice fitting of the same internal diameter as the orifice fitting (length UL on Fig. 3-9) and the similar downstream pipe (length DL on Fig. 3-9) following the orifice.

The sections of pipe to which the orifice flanges are attached or the sections adjacent to the orifice flange or fitting shall comply with API Chapter 14.3 (AGA Report No. 3).

See Figures 3-10, 3-11, and 3-12 for proper meter tube lengths.

Flow Conditioners (Fig. 3-3) — The purpose of flow conditioners is to eliminate swirls and cross currents set up by the pipe, fittings and valves upstream of the meter tube. Please refer to API Chapter 14.3 (AGA Report No. 3) for detailed specifications for flow conditioners.

FIG. 3-3
1998 Uniform Concentric 19-Tube Bundle Flow Straightener



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FIG. 3-4

Flow Calculation Guide Equations

GAS
- TURBINE OR DISPLACEMENT
  - MASS — lb/hr = ρ • MF • CNT — Eq. 3-1
  - VOLUME — scf/hr = $F_{pb} \cdot F_{tm} \cdot F_{pm} \cdot S \cdot MF \cdot CNT \cdot F_{tb}$ — Eq. 3-2
- ORIFICE
  - MASS — lb/hr = $\sqrt{h_w} \cdot F_n\ (F_c + F_{sl}) \cdot \sqrt{\rho} \cdot 1.0618 \cdot F_a \cdot Y$ — Eq. 3-3
  - VOLUME — scf/hr = $\sqrt{h_w} \cdot \sqrt{P_f} \cdot F_n\ (F_c + F_{sl}) \cdot F_{pb} \cdot F_{tf} \cdot F_{pv} \cdot F_g \cdot Y \cdot F_{tb} \cdot F_a$ — Eq. 3-4

LIQUID
- TURBINE OR DISPLACEMENT
  - MASS — lb/hr = ρ • MF • CNT — Eq. 3-1
  - VOLUME — gph = $C_{pl} \cdot C_{pl} \cdot MF \cdot CNT$ — Eq. 3-5
- ORIFICE
  - MASS — lb/hr = $\sqrt{h_w} \cdot F_n\ (F_c + F_{sl}) \cdot \sqrt{\rho} \cdot 1.0618 \cdot F_a$ — Eq. 3-6
  - VOLUME — gph = $\sqrt{h_w} \cdot F_n\ (F_c + F_{sl}) \cdot F_{gt} \cdot F_a$ — Eq. 3-7

STEAM
- ORIFICE
  - MASS — lb/hr = $\sqrt{h_w} \cdot F_n\ (F_c + F_{sl}) \cdot F_g \cdot Y \cdot F_a$ — Eq. 3-8

Other Equations

$$Q = C' \sqrt{h_w \cdot P_f}$$

$$C' = F_n\ (F_c + F_{sl}) \cdot F_{pb} \cdot F_{tf} \cdot F_{pv} \cdot F_g \cdot Y \cdot F_{tb} \cdot F_a$$

$$F_a = 1 + [0.0000185\ (T_f - T_{meas})] \text{ for 304/316 Stainless Steel}$$

$$F_{pb} = \frac{14.73}{P_b} \qquad C_{pl} = \frac{1}{1 - (F \cdot D_p)}$$

$$F_{tb} = \frac{460 + T_b}{520} \qquad F_n\ (F_c + F_{sl}) = \text{Key} = \text{orifice factor (Fig. 3-13)}$$

$$F_{tf} = \sqrt{\frac{520}{460 + T_f}}$$

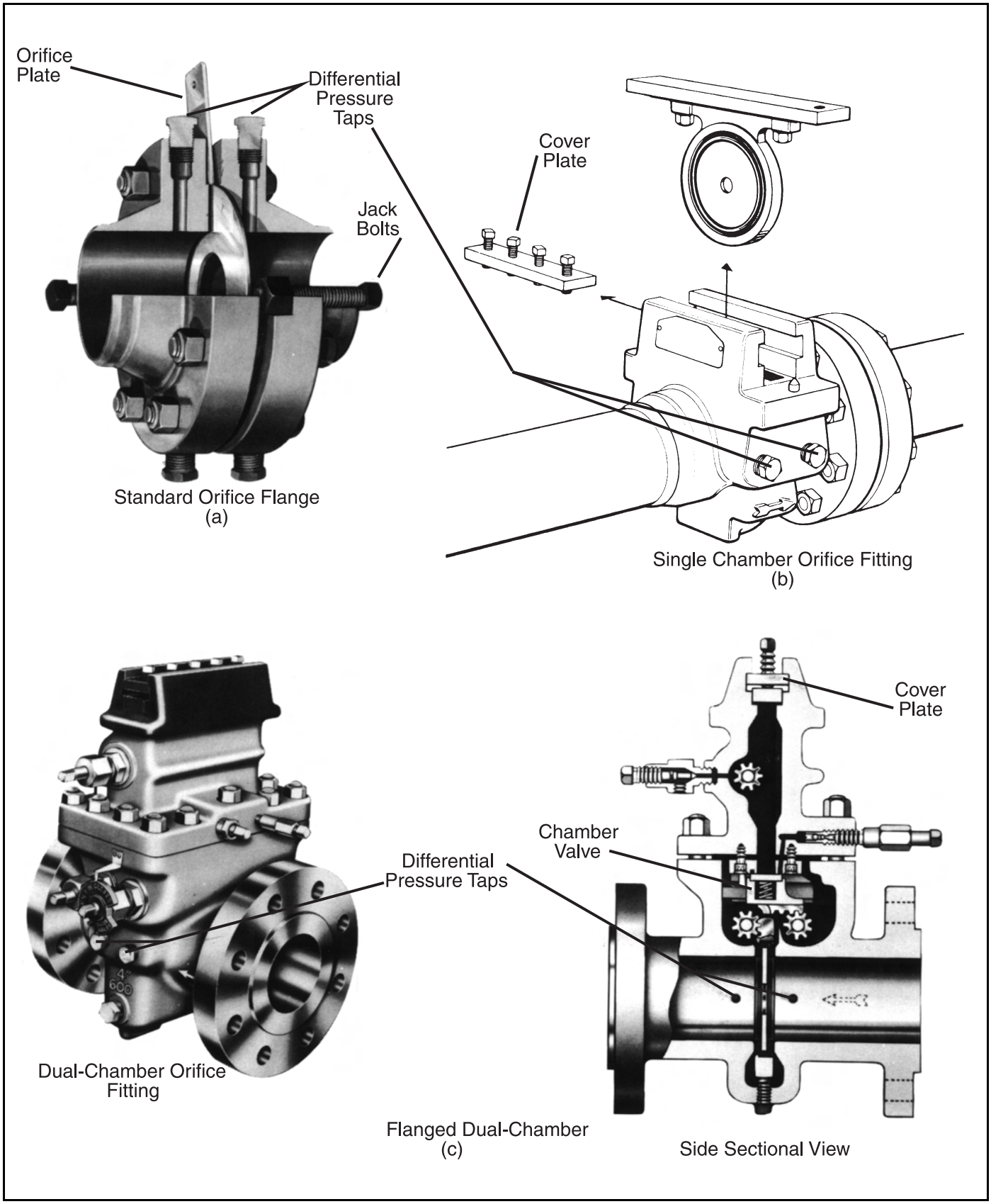
$$F_g = \sqrt{\frac{1.0000}{G}} \qquad F_{gt} = \frac{1.0057}{\sqrt{G_1}} \sqrt{\frac{G_f}{G_1}}$$

(See Fig. 23-12 and 23-13 for G_f/G_1 conversions.)

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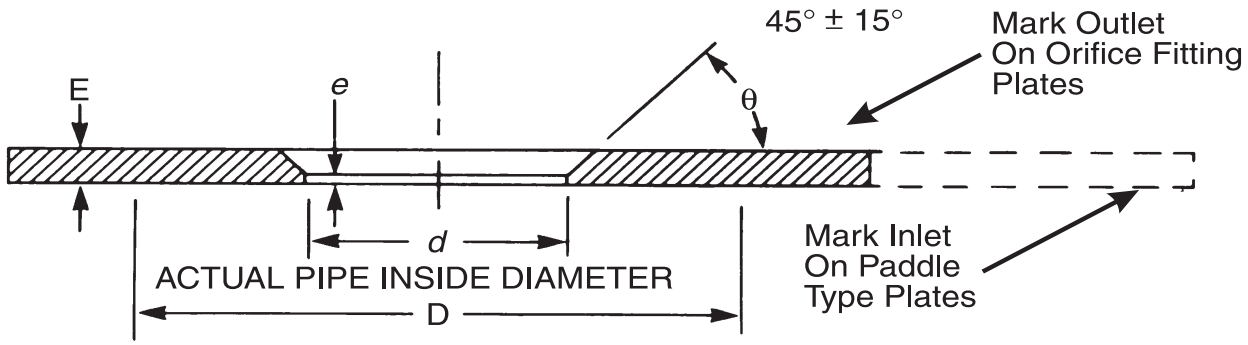
FIG. 3-5

Orifice Plate Holders



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FIG. 3-6
Orifice Plate Thickness and Maximum Allowable Differential Pressure Based on the Structural Limit



Nominal Pipe Size (NPS) (inches)	Published Inside Pipe Diameter (inches)	Orifice Plate Thickness E (inches)			Maximum Allowable ΔP(In. H_2O) Orifice Fitting	Maximum Allowable ΔP(In. H_2O) Orifice Flanges
		Minimum	Maximum	Recommended		
2	1.687	0.115	0.130	0.125	1000	1000
	1.939	0.115	0.130	0.125	1000	1000
	2.067	0.115	0.130	0.125	1000	1000
3	2.300	0.115	0.130	0.125	1000	1000
	2.624	0.115	0.130	0.125	1000	1000
	2.900	0.115	0.130	0.125	1000	1000
	3.068	0.115	0.130	0.125	1000	1000
4	3.152	0.115	0.130	0.125	1000	1000
	3.438	0.115	0.130	0.125	1000	1000
	3.826	0.115	0.130	0.125	1000	1000
	4.026	0.115	0.130	0.125	1000	1000
6	4.897	0.115	0.163	0.125	345	1000
	5.187	0.115	0.163	0.125	345	1000
	5.761	0.115	0.192	0.125	345	1000
	6.065	0.115	0.192	0.125	345	1000
8	7.625	0.115	0.254	0.250	1000	1000
	7.981	0.115	0.319	0.250	1000	1000
	8.071	0.115	0.319	0.250	1000	1000
10	9.562	0.115	0.319	0.250	570	1000
	10.020	0.115	0.319	0.250	570	1000
	10.136	0.115	0.319	0.250	570	1000
12	11.374	0.175	0.379	0.250	285	1000
	11.938	0.175	0.398	0.250	285	1000
	12.090	0.175	0.398	0.250	285	1000
16	14.688	0.175	0.490	0.375	465	1000
	15.000	0.175	0.500	0.375	465	1000

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FIG. 3-6 (Cont'd.)

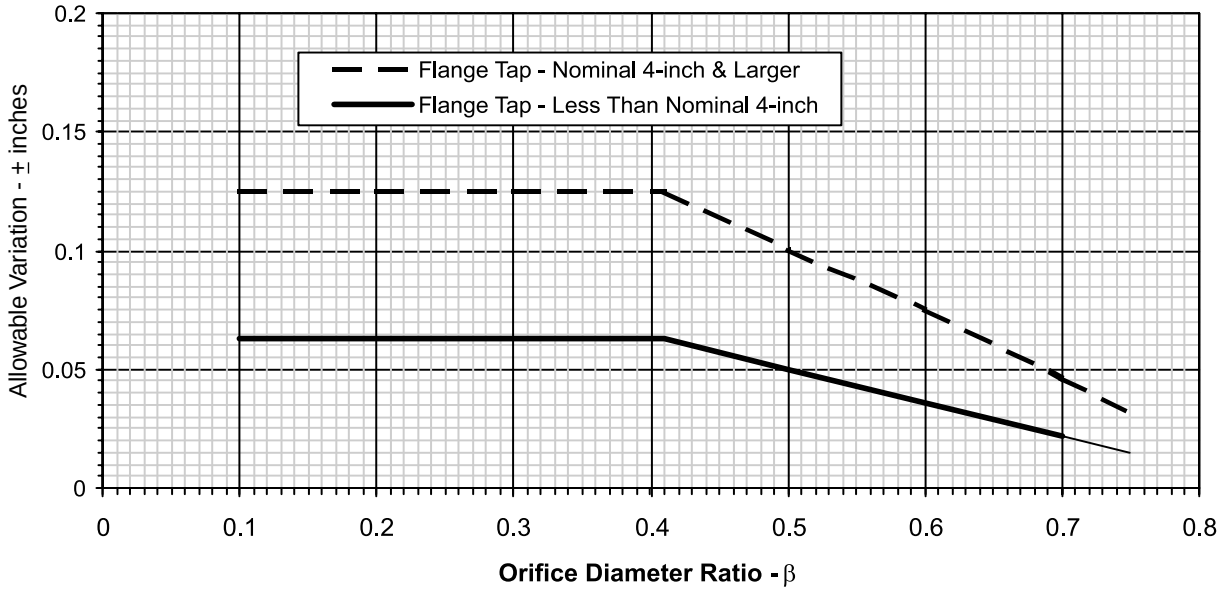
Orifice Plate Thickness and Maximum Allowable Differential Pressure Based on the Structural Limit

Nominal Pipe Size (NPS) (inches)	Published Inside Pipe Diameter (inches)	Orifice Plate Thickness E (inches)			Maximum Allowable ΔP(In. H_2O) Orifice Fitting	Maximum Allowable ΔP(In. H_2O) Orifice Flanges
		Minimum	Maximum	Recommended		
	15.025	0.175	0.500	0.375	465	1000
20	18.812	0.240	0.505	0.375	235	1000
	19.000	0.240	0.505	0.375	235	1000
	19.250	0.240	0.505	0.375	235	1000
24	22.624	0.240	0.505	0.500	360	1000
	23.000	0.240	0.562	0.500	360	1000
	23.250	0.240	0.562	0.500	360	1000
30	28.750	0.370	0.562	0.500	180	1000
	29.000	0.370	0.578	0.500	180	1000
	29.250	0.370	0.578	0.500	180	1000

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FIG. 3-7

Allowable Variations in Pressure Tap Hole Location

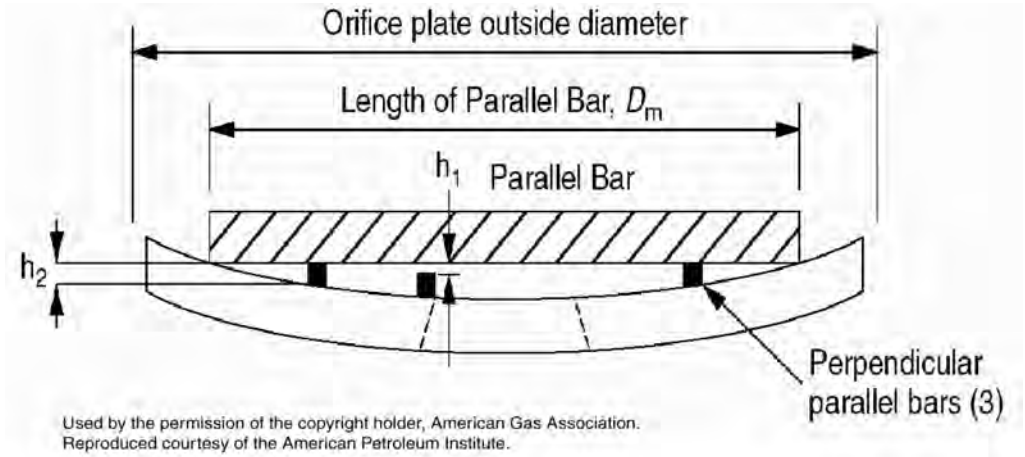


Tolerance specified for β ratio of 0.75 should be used in the design of new installations.
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FIG. 3-8

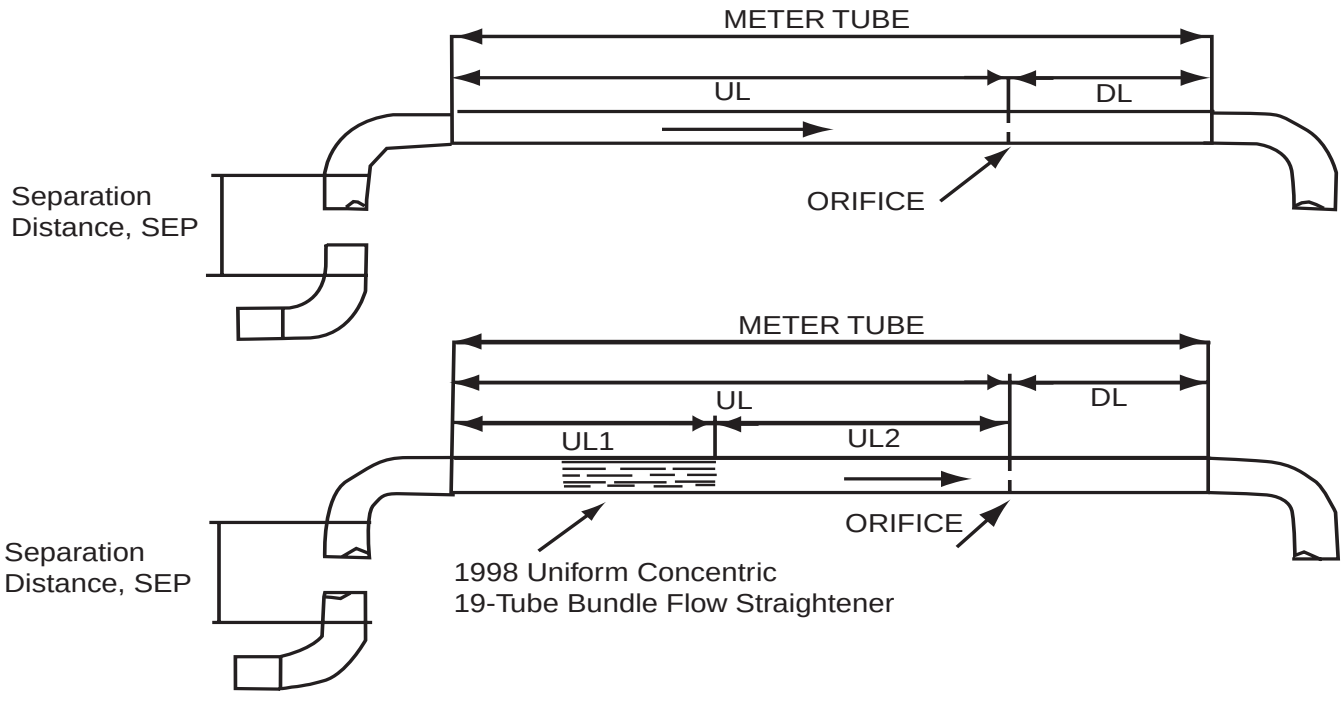
Alternative Method for Determination of Orifice Plate Departure from Flatness (Departure from Flatness = h_2 - h_1)



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FIG. 3-9

Orifice Meter Tube Layout for Flanged or Welded Limit



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FIG. 3-10
Orifice Meter Installation Requirements without a Flow Conditioner

Minimum straight unobstructed meter tube length from the upstream and downstream side of the office plate (in multiples of published internal pipe diameter, D_1)

Diameter ratio β	a. Single 90° elbow. b. Two 90° elbows in the same plane with SEP > $30D_i$. c. Two 90° elbows in perpendicular planes with SEP > $15D_i$.	Two 90° elbows in the same plane "SEP" configuration spacer SEP ≤ $10D_i$.	Two 90° elbows in the same plane "SEP" configuration $10D_i$ < SEP ≤ $30D_i$.	*Two 90° elbows in perpendicular planes, SEP < $5D_i$.*	Two 90° elbows in perpendicular planes, $5D_i$ ≤ SEP ≤ $15D_i$.	Single 90° Tee used as an elbow but not as a header element	a. Single 45° elbow. b. Two 45° elbows in the same "SEP" configuration SEP ≥ $22D_i$.	Gate valve at least 50% open	Concentric reducer	Any other configuration (catch all category)*	Downstream meter tube length
	UL	UL	UL	UL	UL	UL	UL	UL	UL	UL	DL
	6	10	10	50	19	9	30	17	6	70	2.8
0.30	11	10	12	50	32	9	30	19	6	108	3.0
0.40	16	10	13	50	44	9	30	21	6	145	3.2
0.50	30	30	18	95	44	19	30	25	7	145	3.5
0.60	44	44	30	95	44	29	30	30	9	145	3.9
0.67	44	44	44	95	44	36	44	35	11	145	4.2
0.75	44	44	44	95	44	44	44	44	13	145	4.5
Recommended length for maximum range β ≤ 0.75	44	44	44	95	44	44	44	44	13	145	4.5

UL – Minimum meter tube length upstream of the orifice plate in internal pipe diameter, D_i (See Figure 3-9).Straight length shall be measured from the downstream end of the curved portion of the nearest (or only) elbow or of the tee or the downstream end of the conical portion of reducer or expander.

DL – Minimum downstream meter tube length in internal pipe diameters, D_i (See Figure 3-9).

SEP – Separation distance between piping elements in internal pipe diameter, D_i, measured from the downstream end of the curved portion of the upstream elbow to the upstream end of the curved portion of the downstream elbow.

Note : The tolerance on specified lengths for UL and DL is ± $0.25D_i$.

* – This installation exhibits strong effect of Reynolds number and pipe roughness on the recommended length due to rate of decay of swirl. The present recommendations have been developed for high Reynolds numbers and smooth pipes to capture the worst case.

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FIG. 3-11

Orifice Meter Installation Requirements With 1998 Uniform Concentric 19-Tube Bundle Flow Straightener for Meter Tube Upstream Length of $17D_i \leq UL < 29D_i$.

Diameter Ratio, β	Single 90° elbow $R/D_i = 1.5$	Two 90° elbows out of plane $SEP \leq 2D_i$ $R/D_i = 1.5$	Single 90° Tee used as an elbow but not as a header element	Partially closed valves (at least 50% open)	High swirl combined with single 90° Tee	Any fitting (catch all category)	Downstream meter tube length
	UL2	UL2	**UL2**	**UL2**	**UL2**	**UL2**	**DL**
0.10	5-14.5	5-14.5	5-14.5	5-11	5-13	5-11.5	2.8
0.20	5-14.5	5-14.5	5-14.5	5-11	5-13	5-11.5	2.8
0.30	5-14.5	5-14.5	5-14.5	5-11	5-13	5-11.5	3.0
0.40	5-14.5	5-14.5	5-14.5	5-11	5-13	5-11.5	3.2
0.50	11.5-14.5	9.5-14.5	11-13	**	11-13	***	3.5
0.60	12-13	13.5-14.5	*	Not allowed	*	Not allowed	3.9
0.67	13	13-14.5	Not allowed	Not allowed	Not allowed	Not allowed	4.2
0.75	14	Not allowed	Not allowed	Not allowed	Not allowed	Not allowed	4.5
Recommended tube bundle location for max range of β	13 $\beta \leq 0.67$	**13.5-14.5** $\beta \leq 0.67$	13 $\beta \leq 0.54$	**9.5** $\beta \leq 0.47$	**13** $\beta \leq 0.54$	9.5 $\beta \leq 0.46$	4.5

NOTES:

Lengths shown under the UL2 column are the dimensions shown in Figure 3-9 expressed as the number of published internal pipe diameters (D_i) between the downstream end of the 1998 Uniform Concentric 19-Tube Bundle Flow Straightener and the upstream surface of the orifice plate.

* – $13D_i$ allowed for up to $\beta = 0.54$

** – $9.5D_i$ allowed for up to $\beta = 0.47$

*** – $9.5D_i$ allowed for up to $\beta = 0.46$

SEP – Separation distance between elbows measured as defined in Figure 3-10.

UL1 = UL - UL2

Note : The tolerance on specified lengths for UL, UL2 and DL is $0.25D_i$.

Not allowed means that it is not possible to find an acceptable location for the 1998 Uniform Concentric 19-Tube Bundle Flow Straightener downstream of the particular fitting for all values of UL

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FIG. 3-12
Orifice Meter Installation Requirements With 1998 Uniform Concentric 19-Tube Bundle Flow Straightener for Meter Tube Upstream Length of UL ≥ 29D_i.

Diameter Ratio, β	Single 90° elbow R/D_i = 1.5	Two 90° elbows out of plane SEP ≤ 2D_i R/D_i = 1.5	Single 90° Tee used as an elbow but not as a header element	Partially closed valves (at least 50% open)	High swirl combined with single 90° Tee	Any fitting (catch all category)	Downstream meter tube length
	UL2	UL2	UL2	UL2	UL2	UL2	DL
0.10	5-25	5-25	5-25	5-13	5-23	5-13	2.8
0.20	5-25	5-25	5-25	5-13	5-23	5-13	2.8
0.30	5-25	5-25	5-25	5-13	5-23	5-13	3.0
0.40	5-25	5-25	5-25	5-13	5-23	5-13	3.2
0.50	11.5-25	9-25	9-23	7.5-15	9-19.5	11.5-14.5	3.5
0.60	12-25	9-25	11-16	10-17	11-16	12-16	3.9
0.67	13-16.5	10-16	11-13	10-13	11-13	13	4.2
0.75	14-16.5	12-12.5	12-14	11-12.5	14	Not allowed	4.5
Recommended tube bundle location for max range of β	14-16.5 β ≤ 0.75	**12-12.5** β ≤ 0.75	**12-13** β ≤ 0.75	**11-12.5** β ≤ 0.75	**13** β ≤ 0.75	**13** β ≤ 0.67	4.5

NOTES:

Lengths shown under the UL2 column are the dimensions shown in Figure 3-9 and as defined in Figure 3-11.

SEP – Separation distance between elbows measured as defined in Figure 3-10.

UL1 = UL - UL2

Note: The tolerance on specified lengths for UL, UL2 and DL is ± 0.25D_i.

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Gas Orifice Calculations

Orifice Sizing — A simple calculation is often needed to properly size an orifice plate for new or changing flow rates through existing meter tubes. The procedure uses an existing or assumed flow quantity, a desired differential pressure at a specific static pressure, an estimated flowing temperature, and a determined or assumed specific gravity. The Key orifice coefficient is calculated from the gas flow equation. This calculated value is then compared to Fig. 3-13, and the next larger size is usually selected.

Example 3-1 — Size an orifice plate in gas service.

Given Data:

Line Size, D	=	4.026 in.
Flange Taps		
Specific Gravity	=	0.700
Flowing Temperature	=	100°F
Flowing Pressure	=	75 psia
Flow Rate	=	14,200 cu ft/hr (14.73 psia, 60°F)
Desired Differential	=	50 in. of water

To determine the approximate orifice size required, the corresponding Key_g (natural gas) is calculated using appropriate terms of Eq 3-4;

$$Q_h = Key_g \cdot F_{tf} \cdot F_g \cdot \sqrt{h_w \cdot P_f} \qquad \text{from Eq 3-4}$$

Rearranging,

$$Key_g = \frac{Q_h}{F_{tf} \cdot F_g \cdot \sqrt{h_w \cdot P_f}} \qquad \textbf{Eq 3-11}$$

$$Key_g = \frac{14{,}200}{0.9636\ (1.1952)\ \sqrt{50 \cdot 75}}$$

$$Key_g = 201.342$$

Referring to Key_g (Fig. 3-13) for a 4.026 inch line with flange taps, access the Key_g value which approximates the calculated number. A 1.000 in. orifice size would be selected which has a Key value of 201. More precise calculations would include other corrections. For more precise custody transfer calculations, please refer to API Chapter 14.3 (AGA Report No. 3).

Orifice Flow Rate — The following example illustrates a calculation of flow rate through an orifice.

Example 3-2 — Calculate an approximate flow rate for the orifice using appropriate terms from Eq 3-4.

Given Data:

Line Size, D	=	6.065 in.
Orifice Size, d	=	3.500 in.
Flange Taps		
Flowing Temperature	=	70°F
Flowing Pressure	=	90 psia
Differential	=	60 in. of water
Specific Gravity	=	0.750

From Fig. 3-3,

$$Q_h = Key_g \cdot F_{tf} \cdot F_g \cdot \sqrt{h_w \cdot P_f} \qquad \text{from Eq 3-4}$$

$$Q_h = 2646\ (0.9905)\ (1.1547)\ \sqrt{60 \cdot 90}$$

$$Q_h = 222{,}387 \text{ cu ft/hr @ } 14.73 \text{ psia and } 60°F$$

More precise calculations would include other corrections.

For more precise custody transfer calculations, please refer to API Chapter 14.3 (AGA Report No. 3).

Orifice Well Test

Often it is necessary to determine an approximate flow quantity from a well head or field separator vent to the atmosphere for test purposes. The use of a "well head tester" has been a common practice since the early days of the oil and gas industry. See Figure 3-14. An orifice is installed between a pair of flanges, at the outlet of a pipe nipple which is at least eight pipe diameters long. The square edge of the orifice faces the flow. The diameter of the pipe nipple should not be greater than the preceding fittings. The pressure connection may be made in the upstream flange or at any point in the pipe nipple within three diameters from the orifice. The pressure differential across the orifice is the difference between the upstream pressure and atmospheric pressure.

An approximate flow rate may be calculated from:

$$Q = 16{,}330\ (1 + \beta^4)\ (d^2)\ \sqrt{H\ (29.32 + 0.3H)} \cdot F_{tf} \cdot C_g \qquad \textbf{Eq 3-12}$$

For conditions other than 60°F (flowing) and G of 0.6, correction factors must be applied.

$$F_{tf} = \sqrt{\frac{520}{460 + T_f}} \qquad \textbf{Eq 3-13}$$

$$C_g = \sqrt{\frac{0.60}{G}} \qquad \textbf{Eq 3-14}$$

Example 3-3 — Calculate the daily gas flow through a 1-inch orifice in a nominal 3-inch pipe. The gas gravity is 0.70, the flowing temperature is 60°F, and the pressure upstream of the orifice is 5 inches Hg. The published ID of a 3-inch pipe is 3.068 in.

$$Q = 16{,}330\left[1 + \left(\frac{1.0}{3.068}\right)^4\right](1.0)^2\ \sqrt{5\ [29.32 + (0.3)\ (5)]}$$

$$\cdot\ 1.000 \cdot 0.9258 = 190{,}000 \text{ std cu ft per day}$$

Ultrasonic Measurement

This section gives a short overview of ultrasonic meters. If meter design and custody quality calculations are required, please refer to American Gas Association Report #9. These meters are designed for measurement of single-phase fluid only.

An ultrasonic meter (UM) is a fluid velocity-sensing device. (See Figure 3-15) The flowing gas velocity is determined by the transit times of high frequency pulses between two matched transducers. One is designated as upstream and one as downstream due to the position in the meter and the direction of flow. These transducers attach into the pipe wall but do not protrude into the gas stream, thus creating a zero pressure drop. There are simple, single path meters that consist of one pair of transducers and multi-path meters with three or more pairs of transducers. Each pair of transducers measures the transit time of each sound pulse transmitted from the up-

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FIG. 3-13

Plate Sizing and Approximate Flowrate, Natural Gas, Natural Gas Liquids and Steam (For Flange Taps)

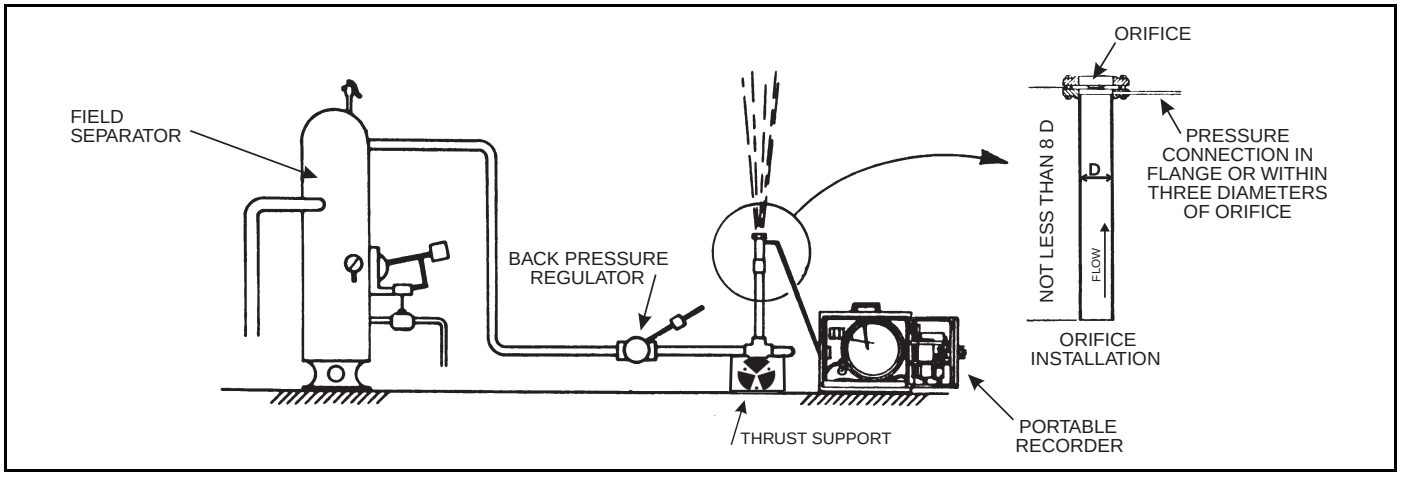
Plate ID	KEY								Plate ID
	Nominal Meter Tube ID (Inches)								
Inches	2	3	4	6	8	10	12	16	Inches
0.25	13								0.25
0.50	50	50							0.50
0.75	115	113	113						0.75
1.00	210	202	201	200					1.00
1.25	347	320	315	313	313				1.25
1.50	557	471	457	452	451	451			1.50
1.75		663	630	617	615	614			1.75
2.00		911	836	809	804	802	801		2.00
2.25			1084	1029	1019	1016	1015		2.25
2.50			1382	1280	1262	1256	1254	1252	2.50
2.75			1746	1563	1532	1522	1518	1516	2.75
3.00				1882	1830	1815	1809	1805	3.00
3.25				2241	2159	2135	2126	2120	3.25
3.50				2646	2519	2483	2469	2460	3.50
3.75				3104	2914	2860	2839	2827	3.75
4.00				3627	3345	3267	3237	3219	4.00
4.25				4227	3816	3705	3664	3639	4.25
4.50					4330	4176	4119	4085	4.50
4.75					4894	4682	4604	4558	4.75
5.00					5513	5226	5120	5059	5.00
5.25					6195	5809	5669	5588	5.25
5.50					6949	6434	6251	6146	5.50
5.75						7107	6869	6734	5.75
6.00						7830	7524	7351	6.00
6.25						8609	8220	8001	6.25
6.50						9451	8957	8682	6.50
6.75						10362	9739	9397	6.75
7.00						11351	10570	10147	7.00
7.25							11453	10933	7.25
7.50							12393	11756	7.50
7.75							13394	12620	7.75
8.00							14463	13524	8.00
8.25							15606	14473	8.25
8.50								15468	8.50
8.75								16511	8.75
9.00								17607	9.00
9.25								18758	9.25
9.50								19968	9.50
9.75								21242	9.75
10.00								22584	10.00
10.25								23999	10.25
10.50								25495	10.50

Definitions and Equations:	**Calculations:**		
	Gas	**Liquid**	**Steam**
D = Nominal diameter of Meter Tube in inches.	8	8	8
Q_h = Desired Flow in Cubic Feet per Hour at Base Conditions.	620000		
Q_{mh} = Desired Flow in Pounds per Hour		140000	1700000
h_w = Desired Differential in Inches of Water.	50	50	50
P_f = Flowing Pressure in psia.	370		250
ρ_f = Flowing Density in Pounds per Cubic Foot.		31.2276	
G = Gas Specific Gravity	0.570		
$KEY_g = (h_w \bullet P_f / G)^{0.5}$	180.156		
$KEY_l = 1.06 \bullet (h_w \bullet \rho_f)^{0.5}$		41.885	
$KEY_s = 4.47 \bullet (h_w \bullet P_f)^{0.5}$			499.761
Table Entry = Desired Flow Divided by Key.	3441	3342	3402
Select Plate Size From Table	4.00	4.00	4.00
Table Entry For Selected Plate Size	3345	3345	3345
Approximate Flow = Table Value (For Selected Plate Size) x KEY	602621	140106	1671701

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FIG. 3-14

Typical Test Set-Up for Measuring Gas from a Separator Vent



stream transducer to the downstream transducer with the flow (t1), and from the downstream to the upstream transducers against the flow (t2). The transit time for a signal traveling with the gas flow is less than travel time against the gas flow. The difference in these transit times relates to the gas velocity along that specific path. Manufacturers intentionally produce a time delay between pulses to insure no interference occurs between the series of pulses. Various calculations and methodologies are then used to calculate the average gas velocity and flow rate at line conditions.

A single path meter monitors only one path's mean velocity at one elevation in the gas flow. Since most gas flow is not fully symmetrical, the use of a single path UM would have inaccuracies dependent on the flow velocity profile. Single path meters are generally used for operational balancing and flare measurement and **are generally not accepted for custody transfer measurement.**

A multi-path UM continuously monitors three or more mean velocities at different elevations in the gas stream of the metered area. The averages of these mean velocities are used to calculate the gas flow rate. Meter designs of various meter manufactures are able to minimize the effect of non-symmetrical flow profiles on the overall meter accuracy.

It is recommended that UM's with three or more paths be used for custody measurement (based on available data).

Volume calculations

Once the actual cubic feet (acf) is calculated by the SPU or flow computer, AGA Report #7 calculations are used to calculate standard cubic feet (Eq. 3.2).

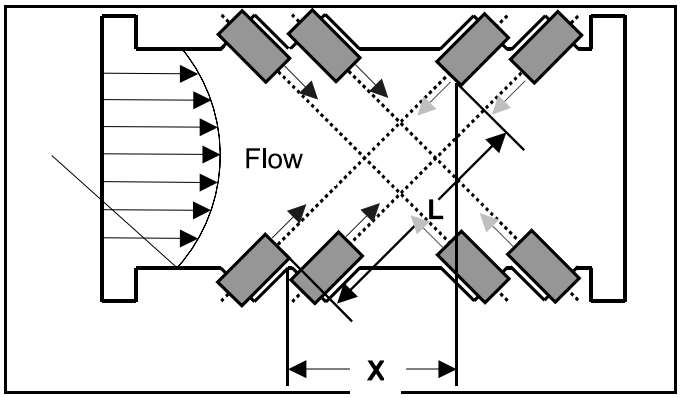
UM Meter Tube and Approach Piping Considerations

The upstream and downstream lengths of the meter tube should be designed in accordance with manufacturer's requirements and AGA Report #9. The approach piping configuration upstream of the meter run can induce asymmetric flow profile distortions that may exist up to 100 pipe diameters downstream of the disturbance. It is a good practice to always design the approach piping to eliminate close-coupled 90º angles out of plane, headers and high pressure drops near the inlet to any UM. These are the common contributors to swirl. Tests conducted at the Southwest Research Institute under the sponsorship of GTI indicate that the failure to use a flow conditioner could produce significantly variable results. If the meter is intended for custody transfer, a flow calibration using a properly certified lab should be performed. The calibration must be performed using the actual UM and its related flow conditioning equipment and meter run assembly.

Although it's always a good practice to install flow conditioners, it is especially important if the approach piping lengths are limited and flow disturbances will be present upstream of the run. There are various manufacturers and designs of flow conditioners. Always consult the manufacturers of both the flow conditioner and the meter and build the spool pieces immediately upstream and downstream of the conditioner to meet the manufactures' and AGA Report #9 specifications. API Chapter 14.3 flow conditioner requirements may be used as a reference for designing and verifying flow conditioning systems and ensuring they effective.

The spool piece immediately downstream of the meter should be built to a length to accommodate the desired number of monitoring points downstream of the meter. The first of these points should be a minimum of 2 pipe diameters downstream of the UM. These monitoring points normally consist of a sample probe, thermal well, test well, etc. The sample probe shall be located in the first monitoring point. The ther-

FIG. 3-15
Ultrasonic Flow Meter



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mal well must be located between 2 and 5 pipe diameters downstream of the UM. The pressure tap can be located on the meter or up to 5 pipe diameters downstream of the meter. (See AGA report #9.)

If the meter is used for bi-directional flow measurement, upstream piping configuration should be used on both ends of the meter.

If the measured gas stream contains contaminates such as, but not limited to, dirt, paraffin, condensate, water, rust, compressor oil, etc., a separator or filter may be considered for installation immediately upstream of the measurement facility.

If a control valve or pressure regulating device is installed up or downstream of the UM, consideration should be taken to reduce or eliminate the amount of ultrasonic noise created by these devices under conditions that could effect measurement accuracy. Ultrasonic noises created by these devices may distort or overpower the ultrasonic signal of the meter, creating a complete loss of signal or a diminished accuracy level. If these noisy conditions exist, consult the meter manufacture for possible solutions. Proper piping system design can isolate the UM from ultrasonic noise sources to some extent. For example, since ultrasonic noise does not change direction readily, installing tees in the line with one branch capped and the other open to flow will significantly reduce the transmission of ultrasonic noise downstream. When such a design is used, be sure to consider the potential for erosion if the flowing velocity impacting the tee is high. Also, be sure the flow conditioning system is able to eliminate any swirl or severe turbulence generated by such piping, before the flow reaches the UM.

Calibration of the UM

The initial calibration of the UM is normally the dry calibration. Dry calibration insures that the proper calibration parameters were entered into the meter electronics and all components are operating correctly. Dry calibration entails blinding both ends of the meter and filling the meter body with a known gas (normally nitrogen). Using a known gas, the VOS is calculated and compared to the measured VOS from the meter. The transit times from upstream and downstream transducers will be the same, since the meter is in a no flow condition. The accurate measurement of the VOS in the test within certain limits can generally guarantee a given accuracy.

Dry calibrations usually insure the minimum meter accuracy of +/- 1.0%. If accuracy levels better than +/- 1.0% are desired, the meter should be flow calibrated at a flow lab. The flow calibration should be performed at a lab that has the flowing capacity to verify the meter up to 100% of the meter's flowing capacity. Six to ten velocities representative of the meter's range should be verified in the calibration process. The average of these verifications may be used for a composite meter factor if the meter is linear within the stated tolerances over the expected normal operating range of the meter, or an algorithm can be written to correct the metered volume that correspond to the adjustments required over the calibrated flow range.

If the meter is intended for custody transfer measurement, a flow calibration should be performed. This flow calibration shall meet or exceed the requirements stated in AGA Report #9.

Calibration of the pressure and temperature sensing devices is critical and should be calibrated to meet the requirements of API MPMS Chapter 21 Section 2.

UM Meter Maintenance and Verification

If a meter is flow calibrated, log files should be collected at various flow rates during the flow calibration. The manufactures should supply software that can be used for the collection of these logs. These logs can be used as a "baseline" to compare with future data. This is extremely valuable as a diagnostic tool and can be useful in determining if a meter is beginning to experience electronic problems, transducer problems, or pipe wall contamination.

Another method of meter verification is the VOS check. If the gas composition is known, the VOS at operating conditions can be calculated using third party software. This calculated VOS can be compared to the measured VOS at operating conditions to verify the meter's accuracy.

UM manufacturers currently offer software packages that help establish the meter's performance and enable the user to verify the meter's performance throughout the meter's history. Training on these software packages and procedures should be sought out from the meter manufacturer.

LIQUID MEASUREMENT

Volumetric Measurement Meters (Orifice Meters)

Liquid volume measurement by an orifice meter can be determined by following the guidelines established in API Chapter 14.8. As with gas measurement, the primary element should consist of an orifice plate, the orifice holder with it's associated tap holes to sense the differential and static pressure, and the upstream and downstream piping "meter tube". The differential and static pressure readings are sensed at the flange taps by a secondary element sensor or transducer. The temperature of the fluid should also be recorded by the temperature sensor or transducer. Note that the meter is the tertiary device that records the output of the sensors/transducers.

The Reader-Harris, Gallagher equation used with orifice meters produces discharge coefficients accurate within +/- 0.5%. Measurement using orifice meters must include this uncertainty, as well as the uncertainty in the metering equipment, unless the metering system is proven against a traceable standard (see API Chapter 4), similar to the way turbine meters and PD meters are typically proven. Then the overall system uncertainty may be reduced to +/- 0.25%.

Some fluid physical properties also need to be known. Examples may include density, viscosity, and compressibility to accurately determine volume using the AGA Report #3 method. For systems performing custody transfer mass measurement for light hydrocarbons such as ethane, ethylene, E/P Mix, high ethane raw mix NGLs, etc., the flowing density of the stream should be measured with a density meter. Then the mass of the delivery may be determined by multiplying the volume at flowing conditions from the meter/ELM, times the density of the flowing stream from the density meter. Details of this method can be found in API Chapters 14.4, 14.6, 14.7, 14.8 and 21.2.

A calculation routine to calculate *approximate* volumes of liquids by orifice meter is included in the Excel spreadsheet

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located on the diskette on the inside cover of this book or on the CD version of the book.

Example 3-4 — Calculate an approximate orifice size for the given flow rate and line size.

Line Size, D	=	3.068 in.
Flange Taps		
Specific Gravity at 60°F	=	0.690
Flowing Temperature	=	40°F
Flow Rate	=	3400 gal. per hr
Desired Differential	=	50 in. of water

To determine the approximate orifice size required, the corresponding Key_l is calculated using appropriate terms of Eq 3-7.

$$Q_h = Key_l \cdot F_{gt} \cdot \sqrt{h_w}$$

or $$Key_l = \frac{Q_h}{F_{gt} \cdot \sqrt{h_w}}$$ From Eq 3-7

$$= \frac{3400}{1.2193 \cdot \sqrt{50}} = 394.35$$

Referring to the Key values (Fig. 3-13) for a 3.068 inch line with flange taps, access the value listed which approximates the calculated Key_l. A 1.5 inch orifice diameter would be selected, which has a 471 Key_l value.

Orifice Flow Rate — The liquid flow rate through an orifice is calculated using the appropriate equation in Fig. 3-4. The initial calculation can be completed using only the Key_l and the F_{gt} correction factors to solve for Q_h since those factors are most significant.

Example 3-5 — Calculate a liquid flowrate for the given orifice setting.

Line Size, D	=	8.071 in.
Orifice Size, d	=	4.000 in.
Flange Taps		
Specific Gravity at 60°F	=	0.630
Flowing Temperature	=	80°F
Differential	=	36 in. of water

To determine the flowrate from Fig. 3-4,

$$Q_h = Key_l \cdot F_{gt} \cdot \sqrt{hw}$$ From Eq 3-7

The value of Key_l from Fig. 3-13 is 3345 for an 8.071 in. line with a 4.0 in. orifice. The value of F_{gt} is calculated from the Fig. 3-4 equation.

Therefore,

$$Q_h = 3345 \cdot 1.2560 \cdot \sqrt{36} = 25{,}208\ gal/hr$$

For more precise calculations, refer to Chapter 14.8 of the API Manual of Petroleum Measurement Standards.

Turbine Meters

Turbine meters are velocity-sensing devices. The direction of flow through the meter is parallel to a turbine's rotary axis and the speed of rotation of the rotor is proportional to the rate of flow.

The turbine meter normally consists of one moving part; an impeller held in place by high pressure, low drag bearings. A magnetic transducer mounted in the meter body is used to count revolutions as the flow passes. The pulses from the transducer are determined for a known volume passing through the meter to develop a factor in pulses per gallon, or other desired unit volume. Turbine meter components are shown in Fig. 3-16. Expected accuracies of plus or minus 0.25% can be attained by certain turbine meters where proper stream conditions are maintained and the meter is properly installed and proven.

Doing mass measurement with turbine meters is often preferred where conditions in temperature, pressure, intermolecular adhesion and solution mixing present difficulty in converting volumes from flowing conditions to standard conditions, such as with ethane, natural gas liquids (NGL), or ethane-propane mixes. To do this properly an online densitometer needs to be used. Refer to GPA 8182 or API Chapter 14, Section 7 (14.7) for further details on mass measurement for NGLs.

Positive Displacement Meters

Displacement meters take a physically enclosed volume of fluid and move it from upstream to downstream of the metering point. The sum of these operations is an indication of the amount of liquid, which is moved over a period of time.

An expected accuracy of 0.25% for a positive displacement (PD) meter can be attained when it is properly installed and proven. Application is normally limited to those fluids that exhibit some lubricating properties because of the multiple moving parts of a positive displacement meter. Typical applications are butane and heavier products since ethane and propane have minimal lubricating properties. Fig. 3-17 shows some internal details of a positive displacement meter. PD meters may perform mass or volumetric measurement, depending on their configuration and companion equipment.

Turbine and Positive Displacement Meter Selection and Performance

Turbine and positive displacement meter installations should include the following considerations:

- Application to proper flow ranges
- Upstream strainers to protect meter internals from foreign material
- Pulsation and vibration
- Proper upstream flow conditioning
- Significant rate changes
- Changes in flow temperature, pressure, and density
- Back pressure (2 times DP across meter plus 1.25 times equilibrium vapor pressure is minimum recommended)
- Connections to prove the meter
- Verification that C_{tl} and C_{pl} will not be applied when the meters are performing mass measurement, except during provings.

The normally acceptable performance of a turbine or positive displacement meter will result in a change in the pulse count of less than 0.05% between meter prover runs, and less than 0.25% between provings. If the factor changes more than 0.25% between provings:

- meter maintenance may be required
- a total flow adjustment must be made

If the factor changes more than 0.5% between provings:

- the turbine must be pulled and inspected for damage or wear
- a total flow adjustment must be made

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- the turbine must be proven again following inspection

More details about turbine and positive displacement meter installations, operation, and proving are available in Chapters 4, 5, 6, and 12 of the API Manual of Petroleum Measurement Standards.

Mass Measurement Meters

Mass measurement of a flowing fluid is advantageous where the physical properties of the fluid are not well defined or available. Mass measurement is especially important in measuring streams containing ethane and methane because of substantial solution mixing effects. Mass measurement is accomplished by multiplying the volume of the fluid at flowing conditions, over a defined period of time, by the density of the fluid at flowing conditions during that same time. This procedure eliminates the need for the correction factors (C_{tl} and C_{pl}) for the metered volume. The total stream mass can be converted into pure components by using a weight analysis of the fluid. Refer to GPA 8182 or API Chapter 14, Section 7 (14.7) for futher details on mass measurement for NGLs.

An example of converting stream mass to pure components is illustrated in Fig. 3-18. Be sure to use the most current version of GPA 2145 for conversion factors.

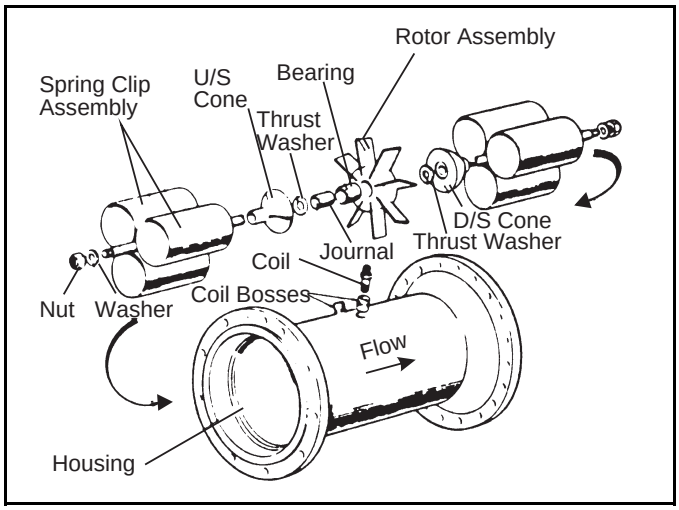
Several more different techniques and processes have been developed to directly measure the mass of a flowing fluid. The devices utilize the principle that angular momentum of a mass is directly proportional to the mass velocity. The resistance of a mass to change direction is measured by different types of devices using combinations of sensitive mechanical and electrical sensors and transmitters that can result in a variety of electronic signals. Mass flow meter installations may not require upstream and downstream piping usually associated with other types of measurement. Proving mass flow meters may involve a complicated arrangement of flow and density measuring equipment, or access to an alternate proving station, or use of a master mass meter comparison.

Coriolis Meters

The Coriolis meter is a mass-measuring device. It consists of a sensor, a transmitter and peripheral devices to provide monitoring, alarm, and/or control functions.

FIG. 3-16

Typical Turbine Meter Components



The sensor consists of two flow tube(s), the *drive* coil and magnet, two *pick-off* coils and magnets and the RTD. During operation, process fluid entering the sensor is split, half passing through each flow tube. The drive coil is energized causing the tubes to oscillate up and down in opposition to one another.

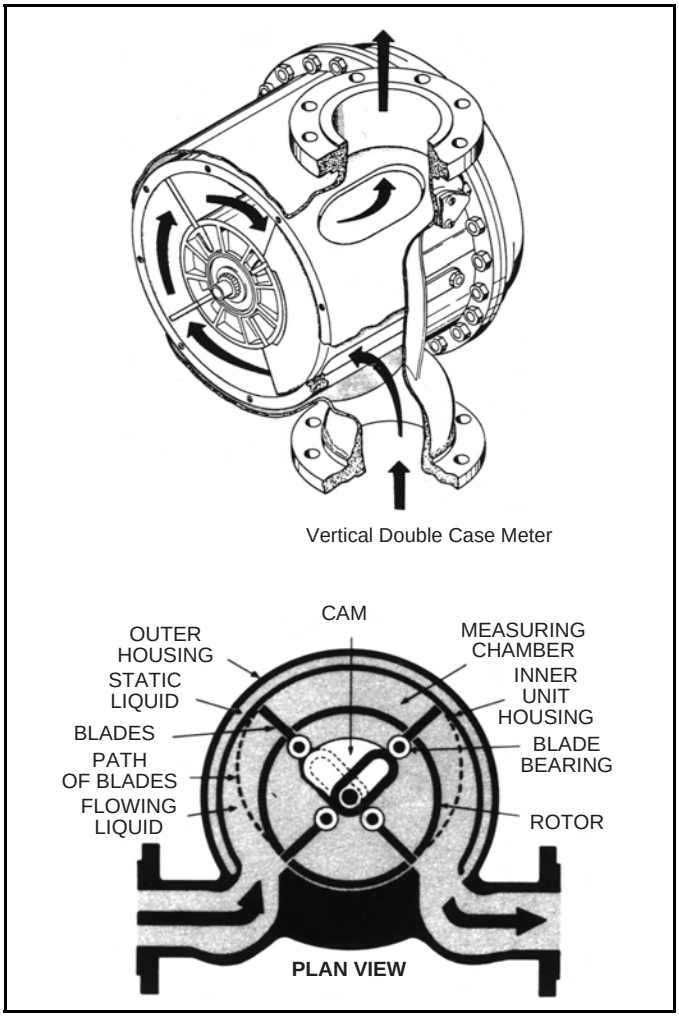
The *pick-off* coils are mounted on one tube while the magnets are mounted on the other. Each coil moves through the uniform magnetic field of the adjacent magnet as the two tubes move. The voltage generated from each pickoff coil creates a sine wave representing the motion of one tube relative to the other. When there is no flow, the two sine waves produced are in phase. When there is flow, the induced Coriolis force causes the tubes to twist, resulting in two out-of-phase sine waves. The time difference in the sine waves is directly proportional to the mass flow rate through the tubes (this may only be true at a fixed pressure).

The density of the fluid is calculated from the frequency of oscillation of the tubes.

The transmitter provides three actions. First, it sends a pulsed current to the sensor drive coil causing the flow tubes to vibrate. Second, it processes the sensor input signals, performs calculations, and produces various outputs to peripheral devices. Most commonly, the output of the meter is a pulsed

FIG. 3-17

Example Positive Displacement Meter



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output. Third, it allows communication with an operator or control system. Figure 3-19 shows the components of a Coriolis meter.

Detail description of how a Coriolis meter operates can be found in appendix A of the API Coriolis Liquid Measurement Draft Standards.

When configuring the meter, users should provide some means to block in the flow so the zero flow condition can be verified. Zero verification of the meter is required from time to time as part of the normal operating procedures. Zeroing is necessary when the zero offset has shifted outside the defined limits. Since the meter should be proven after each zero, unnecessary zeroing should be avoided to minimize potential errors associated with meter factor reproducibility.

The Coriolis meter should be proven under conditions as close to normal operating conditions as practical. The result of a meter proving will be a new or reaffirmed meter factor (MF). This meter factor may be entered in accessory equipment, the Coriolis transmitter, or applied manually to the quantity indicated. The preferred method is to input the meter factor into the accessory equipment due to its audit trail capabilities. A Coriolis meter is normally set up with calibration factors from the manufacturer. These factors, although adjustable, should not be changed. Figure 3-20 shows a typical schematic of a Coriolis meter installation. For more information on Coriolis meter, please refer to the API draft standard, Measurement of Single-phase Intermediate and Finished Hydrocarbon Fluids by Coriolis Meters.

The Coriolis meter has gained popularity in recent years as it presents a number of advantages over other types of meters. A Coriolis meter has an accuracy range of (+/-0.1%) and acceptable repeatability. It provides multi-variable measurement in one device: mass flow rate, volumetric flow rate, density and temperature. It is very tolerant of the changes in the fluid quality and flow rate. It may also be used as a bi-directional meter. Ease of installation and low maintenance are other bonuses as there are no special mounting, no flow conditioning, no straight pipe run requirements and no moving parts.

Like all other types of meters, the Coriolis meter has its own down side. There is a significant pressure drop across the meter making it unsuitable for an existing operation where additional pressure drop cannot be tolerated.

Density Measurement

Densitometer — A densitometer measures density (mass/unit volume) of a fluid. In a vibrating densitometer (Fig. 3-21), a drive coil forces a tube to vibrate at some natural frequency, which is a function of the tube's mass per unit length.

As a fluid flows through the tube, the resonant frequency varies with the density of the fluid. A buoyant force densitometer (Fig. 3-22) operates on the principle that buoyant forces acting on a floating body are a function of the fluid surrounding the body. A float surrounded by fluid is attached to a beam suspended on a pivot. As the density of the fluid changes, the buoyant force provides an angular displacement around the pivot point, which is proportional to the density change. The displacement is picked up and transmitted electronically. A densitometer installed to record flowing density should be installed close to the pipeline and all connections and parts insulated. A densitometer in flowing service can be proven with an auxiliary pycnometer. Refer to Chapter 14.6 of the API Manual of Petroleum Measurement Standards for more details on installation and calibration of density meters. Note that 14.6 will become part of 9.4 in the future.

Meter Proving

The purpose of proving a turbine or PD meter is to establish a factor which, when multiplied by the meter pulses and divided by the nominal K-factor, will give an accurate, corrected metered volume. Compensation for meter factors can be accomplished external to the meter through either mathematical or electronic means. The meter factor may be defined as a number, which corrects meter registration to actual volume. A meter factor (MF) is applicable to only one set of operating conditions. To establish a meter factor, the meter must be compared to a known prover volume. By applying the necessary temperature and pressure factors to the prover volume and by dividing by the change in the meter reading, a meter factor is established.

FIG. 3-18

Example Calculation Converting Stream Mass to Component Gallons

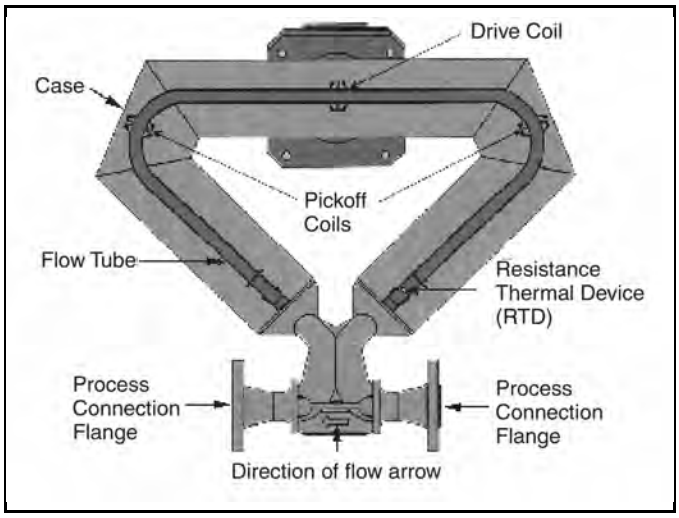
Total Pounds of Stream = Metered Gallons (Meter Factor) (Recorded Actual Flowing Density lb/gal.) (Density Factor)
= 65,994 (1.004) (4.227) (1.002)
= 280,633 Pounds

Component	Analysis Mol %	Molecular Weight*	lb/mol of Mixture	Weight %	Pounds per Component	lb/gal. (wt. in vacuum)*	Component Gallons
Carbon Dioxide	4.40	44.010	1.936	4.34	12,180	6.8534	1,777
Methane	1.10	16.042	.176	.39	1,094	2.5	438
Ethane	41.57	30.069	12.500	28.00	78,577	2.9716	26,443
Propane	28.31	44.096	12.484	27.97	78,493	4.2301	18,556
Isobutane	8.01	58.122	4.656	10.43	29,270	4.6934	6,236
Normal Butane	5.93	58.122	3.447	7.72	21,665	4.8696	4,449
Pentanes Plus	10.68	88.394	9.440	21.15	59,354	5.548	10,698
Totals	100.00			100.00	280,633		68,597

*Use values from GPA 2145 – Latest Edition.

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FIG. 3-19
Components of a Coriolis Meter



$$MF = \frac{\text{Volume of liquid in the prover corrected to standard conditions}}{\text{Change in meter reading corrected to standard conditions}}$$ **Eq. 3-15**

Standard conditions in liquid measurement are 60°F and the equilibrium vapor pressure of the liquid at 60°F (or atmospheric pressure for liquids with a vapor pressure of less than 0 psig).

Meter Proving Systems — Meter proving systems should include the following considerations:

- All proving vessels should be calibrated by qualified personnel using test measures certified by the U.S. Bureau of Standards and be issued a "Certificate of Calibration."
- All meter proving systems should be equipped with a thermometer and pressure gauge located near the meter.
- Proving vessels shall have pressure gauges and thermometers on the inlet and outlet. Gauges and thermometers should be checked periodically against a deadweight tester and certified thermometer to verify accuracy.
- All valves in the proving systems, which affect the accuracy of the proving, should provide for observation of valve integrity, e.g., double block and bleed valves.
- Proving conditions should approximate operating conditions.

Additional information concerning meter proving can be found in Chapter 4 of the API Manual of Petroleum Measurement Standards.

Meter Proving Reports — A meter proving report is the record of a meter proving and must be completed to calculate the net volume being measured by the meter being proven. A meter proving report may contain considerable information but the following information is usually considered the minimum necessary to identify the proving and allow calculation of net volumes: proving report number, date and time, meter owner, company of person responsible for proving fluid used, fluid specific gravity, fluid vapor pressure (at operating temperature), meter size, totalizer pulses, prover size, pulses per unit volume, uncorrected prover volume, temperature of fluid at meter and prover, pressure of fluid at meter and prover, and operator's name and signature. If the proving is being done on a meter used for custody transfer, it is customary for a representative of each custodial company to witness the proving and sign the report if it is satisfactory. The parties involved in the proving should agree on what constitutes an acceptable proving, and if it is not acceptable, what must be done to correct the problem.

Example 3-6 — The following information was taken from a meter proving report. Calculate a meter factor with which to correct the routine meter recorded volumes.

FIG. 3-20
Typical Installation of a Liquid Coriolis Meter

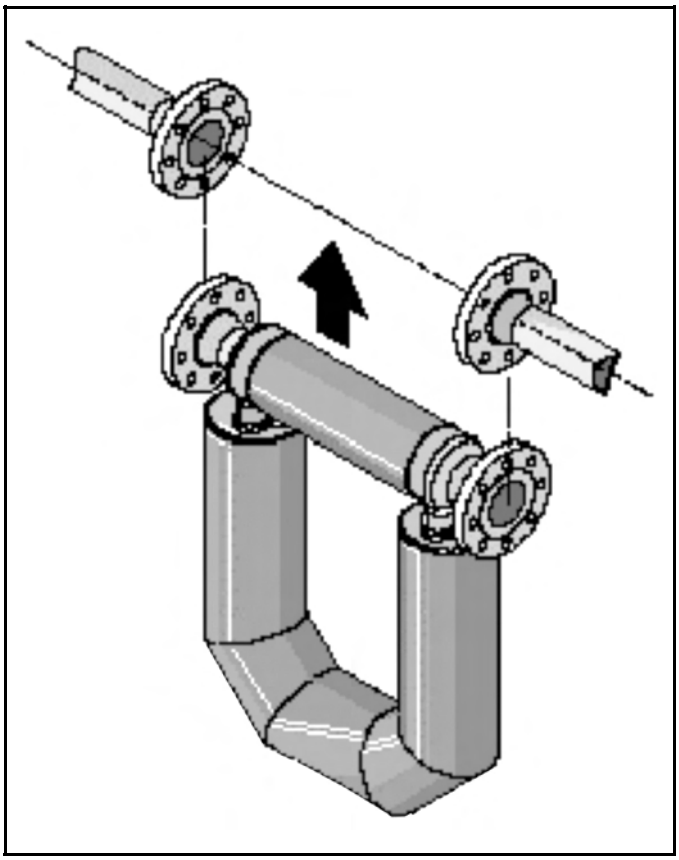
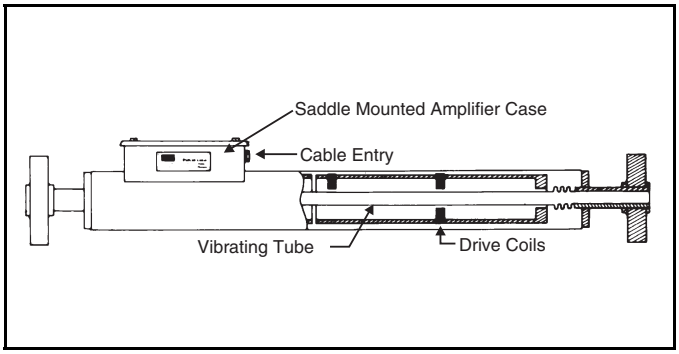


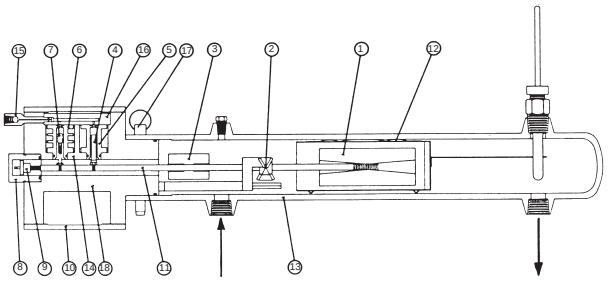
FIG. 3-21
Vibrating Tube Densitometer



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FIG. 3-22

Buoyant Force Densitometer



1 Float
2 Pivot
3 Counterweight
4 Restoring Magnet
5 Restoring Coil
6 Sensing Coil
7 Sensing Slug
8 Calibration Plug
9 Zero Calibration Weight
10 Knurled Sleeve
11 Balance Beam
12 Float Shield
13 Barrel Chamber
14 Terminal Strip
15 Vent
16 Coil Cover
17 Closure Pin
*18 Plug-in Circuit Board

* Not Shown

Given Data:
Product = Propane-Butane Mix
Specific Gravity = 0.544
Totalizer Pulses Per Barrel = 13188
Uncorrected Prover Volume = 2.0734 bbl
Vapor Pressure = 115 psig
Prover Size = 12 in. pipe with 0.375 in. wall
Vertical Double Case Meter

Proving Results:

Run No.	Temperature, °F		Pressure, psig[2]		Pulse Count/Round Trip
	Prover	Meter	Prover	Meter	
1	76.6	76.0	385	395	28629
2	76.8	76.8	385	395	28626
3	76.8	76.0	385	395	28635
4	77.6	77.0	385	395	28634
5	77.0	77.2	385	395	28633
6	77.0	76.6	385	395	28631
Average	77.0	76.6	385	395	28631.3
Average (rounded)	77.0	76.5[1]	385	395	28631[3]

NOTES:

1. Average temperatures are rounded to the nearest 0.5°F.

2. Pressures are read to the nearest scale division.

3. Pulse count is rounded to the nearest count.

The volume of a steel prover must first be determined then the characteristics of the fluid are used to correct the fluid to standard conditions using the corrected prover volume.

Prover Corrections:

1. Uncorrected prover volume = 2.0734 bbl
2. The correction for steel expansion at 77.0°F, C_{ts} (Fig. 3-23) = 1.0003
3. The correction for the pressure effect on the steel prover, C_{ps} (Fig. 3-24) = 1.0004
4. The correction for the effect of temperature on the prover fluid, C_{tl} (Reference No. 4) = 0.9780
5. The correction for the effect of pressure on the prover fluid, C_{pl} (Reference No. 2) = 1.0078
6. Combined prover corrections
   = $C_{ts} \bullet C_{ps} \bullet C_{tl} \bullet C_{pl}$ = 0.9863
7. Corrected prover volume
   = 0.9863 x 2.0734 bbl = 2.0450 bbl

Metered fluid corrections:

1. Uncorrected metered volume $= \dfrac{\text{Total meter pulses recorded}}{\text{manufacturer's pulses per barrel}}$

   $= \dfrac{28631}{13188} = 2.1710$ bbl

2. The correction for the effect of temperature on the metered volume, C_{tl} (Reference No. 4) = 0.9789
3. The correction for the effect of pressure on the metered volume, C_{pl} (Reference No. 2) = 1.0080
4. Corrected metered volume = $C_{tl} \bullet C_{pl} \bullet 2.1710$
   = 2.1421 bbl
5. Meter factor $= \dfrac{\text{corrected prover volume}}{\text{corrected metered volume}} = \dfrac{2.0450}{2.1421} = 0.9547$

Fig. 3-25 is an example meter proving report. The uncorrected prover volume is that volume of the prover tube between the switches, Fig. 3-26. Several meters can utilize one prover if piping manifolds exist. If more than one product is proven by the same prover, precautions must be taken to prevent contamination. Double block-and-bleed valves are commonly used to isolate specification products. Additional information on meter proving can be found in Chapter 4 of the API Manual of Petroleum Measurement Standards.

STEAM MEASUREMENT

The steam flow equation (Fig. 3-4, Eq. 3-8) can be used to calculate steam flow rate by applying a steam factor, F_s, found in Fig. 3-27 for saturated steam and Fig. 3-28 for superheated steam. The F_s factor makes corrections for volume and density at the pressure and temperature conditions of the steam.

This orifice meter volume calculation is also included in Excel spreadsheet located on the diskette on the inside cover of this book or on the CD version of the book.

MISCELLANEOUS MEASUREMENT DEVICES

Pitot — A pitot tube is a cylindrical probe installed in a flowing fluid which senses the impact pressure created by the velocity of a flowing fluid. The probe senses the impact pressure by a hole in the upstream face of the tube, and the static pressure on the side or downstream face of the tube. The use of a pitot tube is severely limited by its sensitivity to velocity profile.

The multiport averaging pitot (Fig. 3-29) attempts to overcome the velocity profile sensitivity by providing several ports to sense the varying impact pressures across the diameter of the pipe. The pressures are averaged and the differential pressure between this average and the static pressure detected on the downstream side of the probe, or from the line itself, is used to determine flow.

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FIG. 3-23

Temperature Correction Factors for Mild Steel, C_{ts}

C_{ts} for mild steel having a cubical coefficient of expansion of 1.86 x 10^{-5} per °F

Observed Temperature, °F	C_{ts} Value	Observed Temperature, °F	C_{ts} Value
(–7.2) - (–1.9)	0.9988	73.5 - 78.8	1.0003
(–1.8) - 3.5	0.9989	78.9 - 84.1	1.0004
3.6 - 8.9	0.9990	84.2 - 89.5	1.0005
9.0 - 14.3	0.9991	89.6 - 94.9	1.0006
14.4 - 19.6	0.9992	95.0 - 100.3	1.0007
19.7 - 25.0	0.9993	100.4 - 105.6	1.0008
25.1 - 30.4	0.9994	105.7 - 111.0	1.0009
30.5 - 35.8	0.9995	111.1 - 116.4	1.0010
35.9 - 41.1	0.9996	116.5 - 121.8	1.0011
41.2 - 46.5	0.9997	121.9 - 127.2	1.0012
46.6 - 51.9	0.9998	127.3 - 132.5	1.0013
52.0 - 57.3	0.9999	132.6 - 137.9	1.0014
57.4 - 62.6	1.0000	138.0 - 143.3	1.0015
62.7 - 68.0	1.0001	143.4 - 148.7	1.0016
68.1 - 73.4	1.0002	148.8 - 154.0	1.0017

Note: This table is suitable for application in meter proving; in prover calibration use the formulas. For the formula used to derive the tabulated values and to calculate values, see Section 12 of the API Manual.

The flow formula for the pitot tube is very similar to that for an orifice. F_{na} is a units conversion factor and C is a flow coefficient for this type installation.

$$F_b = (F_{na})(C) \bullet D^2 \qquad \textbf{Eq 3-19}$$

Installation requirements include precise alignment of the tube and upstream flow conditioning which provides a symmetrical velocity profile. Use of the averaging pitot in a pipe with a diameter or wall thickness other than design will cause erroneous results.

NATURAL GAS SAMPLING

The purpose of natural gas sampling is to secure a representative sample of the flowing gas stream for a specific period of time. Naturally, the more often the sampling system samples the flowing stream, the more likely it is to be truly representative of a stream with varying composition.

Sampling systems consist of numerous components and must include some key elements including a sample probe, any necessary connecting tubing, sample containers or sample valves and appropriate heat-tracing and insulation. For on-line gas chromatographs (g.c.s) or on-stream analyzers, the sampling system ends at the injection valve on the inlet of the g.c.

The sample may be collected on a spot, composite or continuous basis.

Spot sampling simply means that a technician manually collects a sample directly from the stream at scheduled intervals or as needed, but usually on a monthly or longer regular basis.

Composite samples are usually collected on a weekly or monthly basis. Composite sampling systems should grab small samples on a flow proportional basis, then inject them into the composite sampling cylinder. There are composite sampling systems that work on a simple time cycle (time proportional sampling), but they are not recommended, especially if they continue to sample even when flow has stopped. If a time proportional system is already in service, it must be equipped with a flow switch or similar device to ensure that when flow stops, sampling will stop.

Continuous systems provide a steady flow of sample through a sample loop that passes near a composite sampler or on-stream analyzer. In the case of an on-line chromatograph, the injection valve of the g.c. is able to admit and distribute a sample from the flowing loop that is representative of the flow in the main line. Sample rate flow loops must be carefully sized and generally should operate at velocities around 5 ft/sec., but this may vary if the sample loop is exceptionally long (over 100 feet).

The sample point is usually located downstream of the meter run and must be remote from severe flow disturbances such as control valves and orifice plates by at least five nominal pipe diameters. For flowing streams that are not near their hydrocarbon dew point, the probe should be positioned either upstream or downstream of the meter tube, and at least 5 pipe diameters downstream of any flow disturbing elements, such as elbows, swirl generators, headers, valves and tees. If the sample source is at or near its hydrocarbon dewpoint, some research has indicated that the probe should be located at least 8 pipe diameters downstream of any flow disturbance, including an orifice meter. The sample point must not be installed within the upstream or downstream engineered sections of the meter tube, since the fitting and probe could produce disturbances in the flow profile going through the meter in the run. The probe installed in the sample point extends into the center 1/3 of the internal diameter of the meter run to ensure no heavy materials or contaminants migrating along the pipe walls are allowed to contaminate the sample. Note that for large diameter pipelines, the probe never needs to be longer than 10 inches. The probe is equipped with an outlet valve to allow the system to be shut in when no sampling is being performed or to perform maintenance on downstream equipment in composite or continuous sampling systems.

The tubing connecting the sample probe to the downstream sample system(s) should be internally clean, as short as practical (usually 6 to 24 inches maximum) and made of either nylon or stainless steel. Stainless steel is actually preferred due to its strength and flexibility and resistance to melting and/or sharp edges, but nylon is not porous and when used safely, can also give good analytical results. Teflon, carbon steel, plastic tubing, Tygon tubing and many other materials do not perform well.

Care must be taken to insure there are no leaks in the sampling system. Typically, if a leak occurs, smaller molecules tend to escape preferentially and create a bias in analytical results. If the leak is large, there may be enough cooling to produce condensation in the sample system and cause the samples to be very non-representative.

Note that whenever the sample line is operating in ambient temperatures below the flowing temperature of the stream, the line may need to be heat-traced and insulated. If the ambient temperature is lower than the dewpoint temperature of the flowing stream, heat tracing and insulation are required. Be sure that the heat tracing is properly and safely done, using electrically limited tracing meeting appropriate electrical

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FIG. 3-24

Pressure Correction Factors for Steel, C_{ps}

(All measurements are in pounds per square inch gauge.)

Prover Dimensions										
Factor C_{ps}	6-in. Pipe 0.25-in. Wall	6-in. Pipe 0.280-in. Wall	8-in. Pipe 0.322-in. Wall	8-in. Pipe 0.375-in. Wall	10-in. Pipe 0.365-in. Wall	10-in. Pipe 0.375-in. Wall	12-in. Pipe 0.375-in. Wall	14-in. Pipe 0.312-in. Wall	14-in.Pipe 0.375-in. Wall	Factor C_{ps}
1.0000	0–61	0–69	0–60	0–71	0–54	0–56	0–46	0–34	0–42	1.0000
1.0001	62–183	70–207	61–181	72–214	55–163	57–168	47–140	35–104	43–127	1.0001
1.0002	184–306	208–346	182–302	215–357	164–273	169–281	141–234	105–174	128–212	1.0002
1.0003	307–428	347–484	303–423	358–499	274–382	282–393	235–328	175–244	213–297	1.0003
1.0004	429–551	485–623	424–544	500–642	383–491	394–506	329–421	245–314	298–382	1.0004
1.0005	552–673	624–761	545–665	643–785	492–601	507–618	422–515	315–384	383–466	1.0005
1.0006	674–795	762–900	666–786	786–928	602–701	619–731	516–609	385–454	467–551	1.0006
1.0007	796–918	901–1038	787–907	929–1071	711–819	732–843	610–703	455–524	552–636	1.0007
1.0008	919–1040		908–1028		820–928	844–956	704–796	525–594	637–721	1.0008
1.0009					929–1038	957–1068	797–890	595–664	722–806	1.0009
1.0010							891–984	665–734	807–891	1.0010
1.0011							985–1078	735–804	892–976	1.0011
1.0012								805–874	977–1061	1.0012
1.0013								875–944		1.0013
1.0014								945–1014		1.0014
1.0015										1.0015
1.0016										1.0016
1.0017										1.0017
1.0018										1.0018
1.0019										1.0019
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1.0024										1.0024

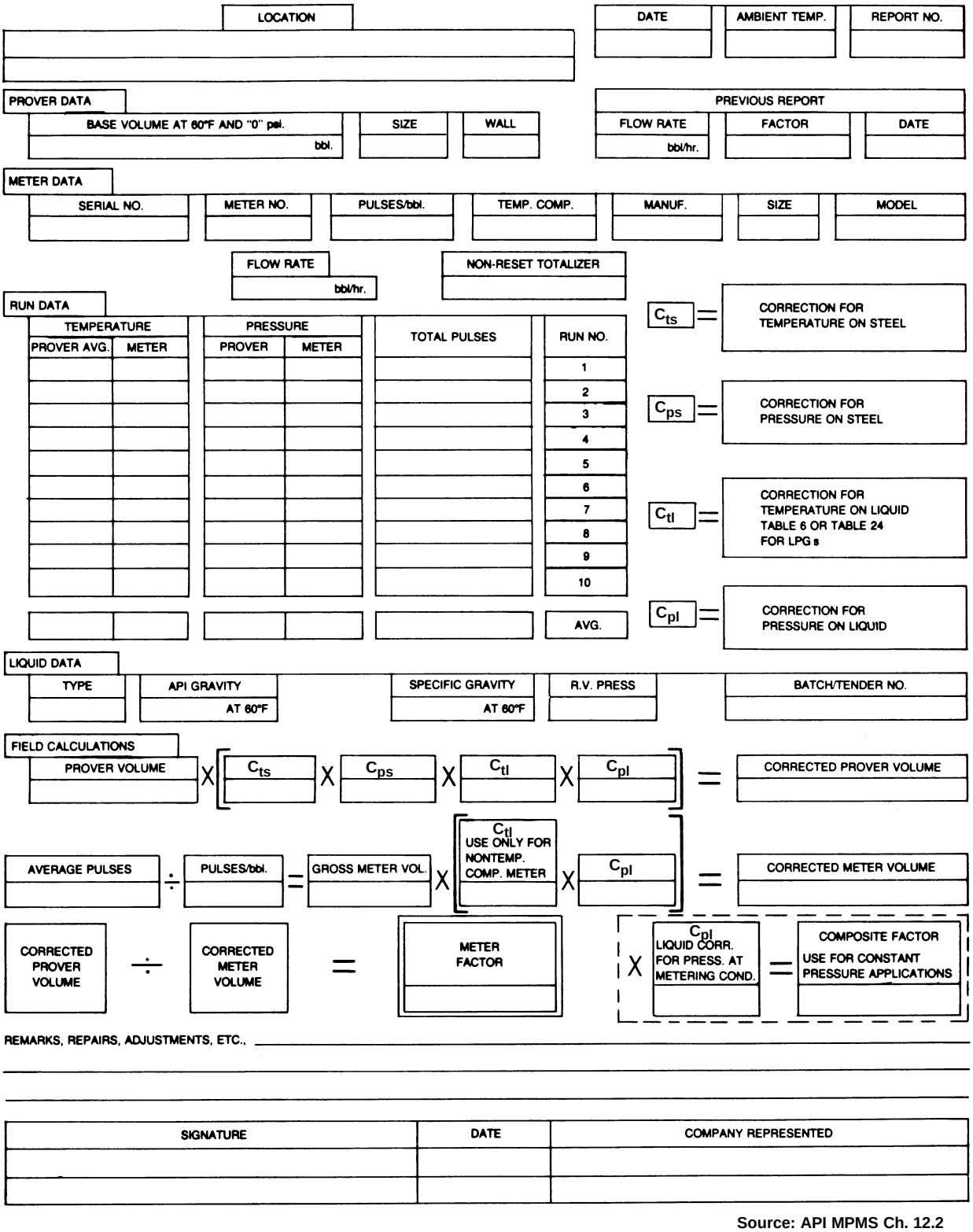
Prover Dimensions									
Factor C_{ps}	16-in. Pipe 0.375-in. Wall	18-in. Pipe 0.375-in. Wall	20-in. Pipe 0.375-in. Wall	24-in. Pipe 0.375-in. Wall	26-in. Pipe 0.375-in. Wall	26-in. Pipe 0.500-in. Wall	30-in. Pipe 0.500-in. Wall	30-in. Pipe 0.500-in. Wall	Factor C_{ps}
1.0000	0–36	0–32	0–29	0–24	0–22	0–30	0–25	0–21	1.0000
1.0001	37–110	33–97	30–87	25–72	23–66	31–89	26–77	22–64	1.0001
1.0002	111–184	98–163	88–146	73–120	67–111	90–150	78–129	65–107	1.0002
1.0003	185–258	164–228	147–204	121–169	112–155	151–209	130–181	108–149	1.0003
1.0004	259–331	229–293	205–262	170–217	156–200	210–270	182–232	150–192	1.0004
1.0005	332–405	294–358	263–321	218–266	201–245	271–329	233–284	193–235	1.0005
1.0006	406–479	359–423	322–379	267–314	246–289	330–390	285–336	236–278	1.0006
1.0007	480–553	424–489	380–438	315–362	290–334	391–449	337–387	279–321	1.0007
1.0008	554–627	490–554	439–496	363–411	335–378	450–510	388–439	322–364	1.0008
1.0009	628–700	555–619	497–555	412–459	379–423	511–569	440–491	365–407	1.0009
1.0010	701–774	620–684	556–613	460–508	424–467	570–630	492–543	408–450	1.0010
1.0011	775–848	685–749	614–672	509–556	468–512	631–689	544–594	451–492	1.0011
1.0012	849–922	750–815	673–730	557–604	513–556	690–750	595–646	493–535	1.0012
1.0013	923–995	816–880	731–788	605–653	557–601	751–809	647–698	536–578	1.0013
1.0014	996–1069	881–945	789–847	654–701	602–646	810–870	699–750	579–621	1.0014
1.0015		946–1010	848–905	702–749	647–690	871–929	751–801	622–664	1.0015
1.0016			906–964	750–798	691–735	930–990	802–853	665–707	1.0016
1.0017			965–1022	799–846	736–779	991–1049	854–905	708–749	1.0017
1.0018				847–895	780–824		906–956	750–792	1.0018
1.0019				896–943	825–868		957–1008	793–835	1.0019
1.0020				944–991	869–913			836–878	1.0020
1.0021				992–1040	914–957			879–921	1.0021
1.0022					958–1022			922–964	1.0022
1.0023								965–1007	1.0023
1.0024									1.0024

Notes: 1. This table is based on the following equation: $C_{ps} = 1 + \dfrac{P_p - P_s D}{E_m}$

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FIG. 3-25

Meter Proving Report

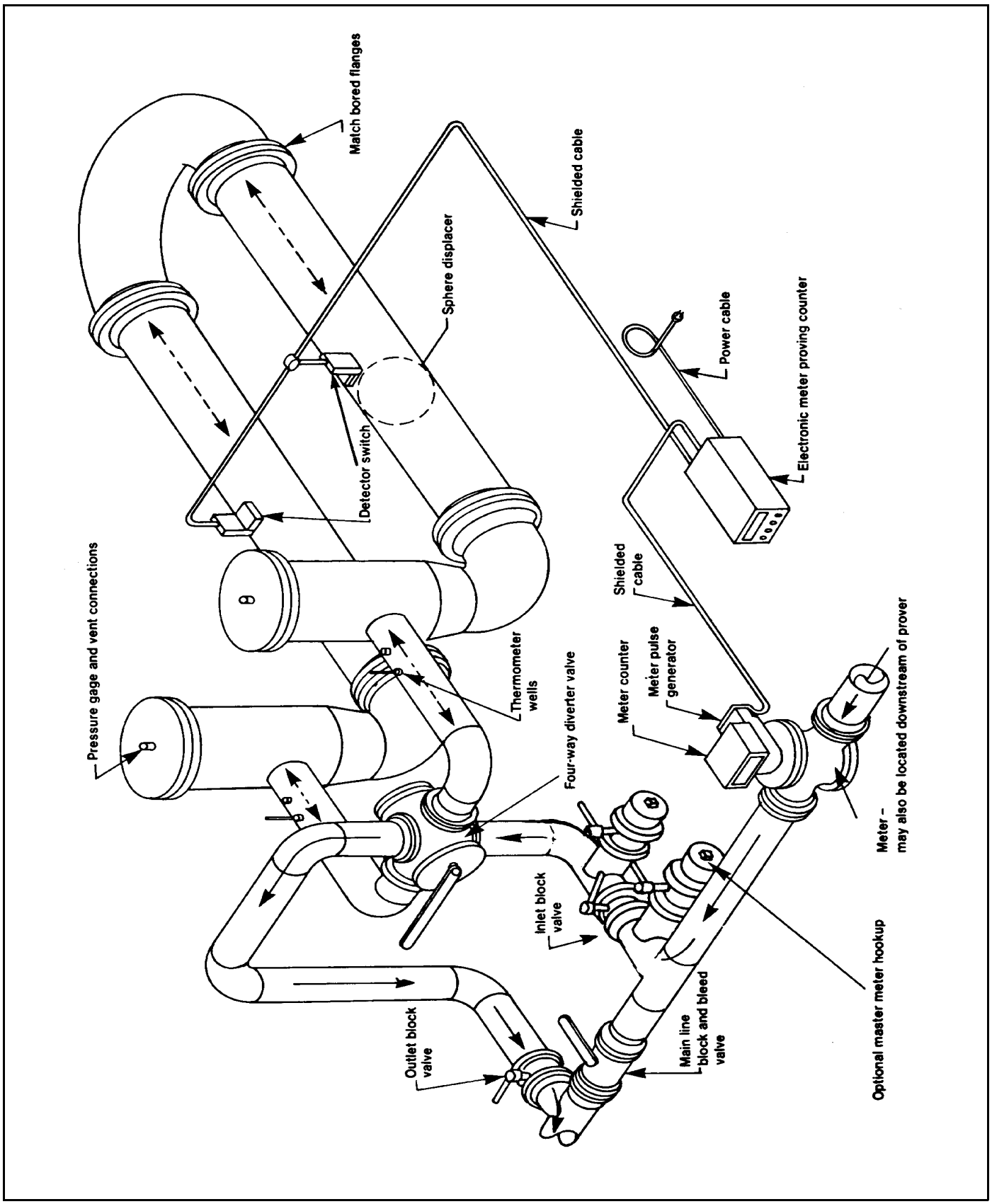


Source: API MPMS Ch. 12.2

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FIG. 3-26

Example Bidirectional Pipe Prover



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FIG. 3-27

Steam Coefficient Factors, Saturated Steam

$$F_s = \frac{1.0618}{\sqrt{P_f v}}, \text{ For } \textit{Bellows–Type} \text{ Meters}$$

Pressure, psig	Steam Quality, Per Cent										
	100	99	98	97	96	95	94	93	92	91	90
1	.0534	.0537	.0539	.0542	.0545	.0548	.0551	.0554	.0557	.0560	.0563
5	.0530	.0533	.0535	.0538	.0541	.0544	.0547	.0550	.0553	.0556	.0559
10	.0526	.0529	.0531	.0534	.0537	.0540	.0543	.0546	.0549	.0552	.0555
15	.0523	.0526	.0528	.0531	.0534	.0537	.0539	.0542	.0545	.0548	.0551
20	.0520	.0523	.0526	.0528	.0531	.0534	.0537	.0540	.0543	.0546	.0549
25	.0518	.0521	.0524	.0526	.0529	.0532	.0535	.0537	.0540	.0543	.0546
50	.0511	.0513	.0516	.0518	.0521	.0524	.0527	.0529	.0532	.0535	.0538
75	.0506	.0508	.0511	.0514	.0516	.0519	.0522	.0525	.0527	.0530	.0533
100	.0503	.0505	.0508	.0510	.0513	.0516	.0518	.0521	.0524	.0527	.0530
125	.0500	.0503	.0505	.0508	.0510	.0513	.0516	.0519	.0521	.0524	.0527
150	.0498	.0501	.0503	.0506	.0509	.0511	.0514	.0517	.0519	.0522	.0525
175	.0497	.0499	.0502	.0504	.0507	.0510	.0512	.0515	.0518	.0521	.0524
200	.0496	.0498	.0501	.0503	.0506	.0509	.0511	.0514	.0517	.0520	.0522
250	.0494	.0497	.0499	.0502	.0504	.0507	.0510	.0512	.0515	.0518	.0521
300	.0493	.0496	.0498	.0501	.0503	.0506	.0509	.0511	.0514	.0517	.0520
350	.0493	.0495	.0498	.0500	.0503	.0505	.0508	.0511	.0513	.0516	.0519
400	.0493	.0495	.0498	.0500	.0503	.0505	.0508	.0511	.0513	.0516	.0519
450	.0493	.0495	.0498	.0500	.0503	.0505	.0508	.0511	.0513	.0516	.0519
500	.0493	.0496	.0498	.0501	.0503	.0506	.0508	.0511	.0514	.0516	.0519
550	.0494	.0496	.0499	.0501	.0504	.0506	.0509	.0511	.0514	.0517	.0520
600	.0494	.0497	.0499	.0502	.0504	.0507	.0509	.0512	.0515	.0517	.0520

FIG. 3-28

Steam Coefficient Factors, Superheated Steam

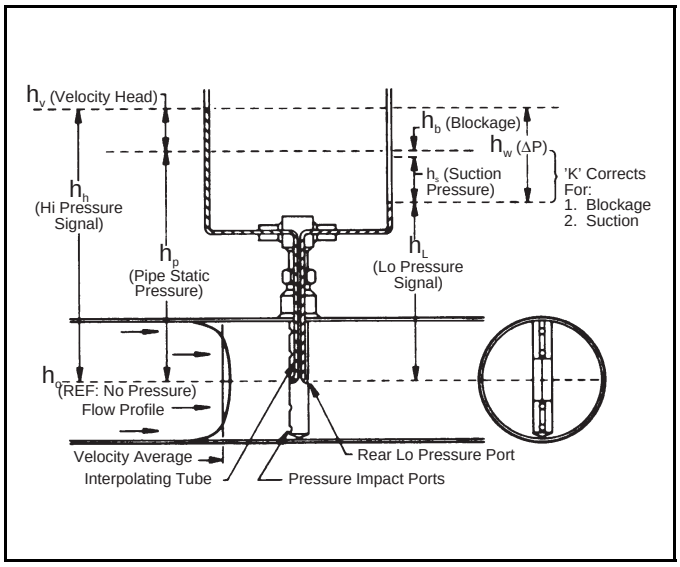
$$F_s = \frac{1.0618}{\sqrt{P_f v}}, \text{ For } \textit{Bellows–Type} \text{ Meters}$$

Pressure, psig	Total Steam Temperture, °F										
	300	350	400	450	500	550	600	700	800	1000	1200
25	.0505	.0488	.0472	.0459	.0446	.0435	.0424	.0405	.0388	.0360	.0338
50	.0510	.0491	.0475	.0460	.0447	.0436	.0425	.0405	.0389	.0361	.0338
75		.0494	.0477	.0462	.0449	.0437	.0426	.0406	.0389	.0361	.0338
100		.0498	.0479	.0464	.0450	.0438	.0426	.0407	.0389	.0361	.0338
125		.0501	.0482	.0466	.0451	.0439	.0427	.0407	.0390	.0361	.0338
150			.0485	.0468	.0453	.0440	.0428	.0408	.0390	.0361	.0338
175			.0488	.0470	.0454	.0441	0429	.0408	.0391	.0362	.0339
200			.0491	.0472	.0456	.0442	.0430	.0409	.0391	.0362	.0339
250				.0476	.0459	.0444	.0432	.0410	.0392	.0362	.0339
300				.0481	.0462	.0447	.0434	.0411	.0393	.0363	.0339
350				.0486	.0466	.0449	.0436	.0412	.0393	.0363	.0339
400				.0492	.0470	.0452	.0438	.0414	.0394	.0364	.0340
450					.0474	.0455	.0440	.0415	.0395	.0364	.0340
500					.0478	.0458	.0442	.0416	.0396	.0364	.0340
600					.0488	.0465	.0446	.0419	.0398	.0365	.0341
700						.0472	0451	.0422	.0400	.0366	.0341
800						.0480	.0457	.0425	.0401	.0367	.0341
900						.0489	.0463	.0428	.0403	.0368	.0342
1000						.0500	.0469	.0431	.0405	.0369	.0342
1100							.0477	.0435	.0407	.0370	.0343
1200							.0485	.0438	.0409	.0371	.0343
1300							.0494	.0442	.0412	.0372	.0344
1400							.0505	.0446	.0414	.0372	.0344
1500							.0518	.0450	.0416	.0373	.0345

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FIG. 3-29

Basic Operating Principle—Multiple Averaging Pitot



codes for the area classification (typically Class I Group D Division I or II).

Realize that the dewpoint of a gas stream is absolutely critical to accurate sampling. If any component in the sampling system causes the temperature of even a portion of the gas stream being sampled to cool to or below the hydrocarbon dewpoint, the sample will be depleted of heavy components and can no longer be truly representative of the stream. Note that the Btu content in this situation is not always too low. If the sample system continues to condense heavy components for an extended amount of time, accumulations may reach the point that liquid droplets enter the sample and actually cause the indicted Btu content and calculated relative density (specific gravity) to be too high.

The sample cylinders used in spot sampling should be stainless steel, single cavity cylinders. Single cavity cylinders are recommended, due to the difficulty of fully cleaning piston cylinders between use. Residue that may remain in the piston cylinders and their seals may produce incorrect analyses. The cylinders should be equipped with standard design sample valves that are screw open or closed (not 1/4 turn ball valves) and have a flow passage of approximately 1/8 inch diameter.

It may be convenient to use piston (constant pressure) cylinders in composite sampling systems, since you can easily see that the system is working or not working as the level indicator moves. If constant pressure/piston cylinders are used and oil or grease contamination is present in the system, they must be disassembled between use, carefully cleaned and then the seal rings must be replaced if the cylinder is expected to provide representative samples.

Note that if sampling is being performed to determine the levels of volatile or reactive contaminants, such as H_2S, the cylinder may need to be lined with an epoxy/phenolic lining. Even then, particularly reactive materials, such as H_2S or ethyl mercaptan are likely to be lost prior to analysis unless the sample is collected on-site and analyzed immediately. Even a few minutes delay can reduce detectable levels of reactive materials. Shipping a sample to a remote lab and delaying analysis beyond a couple of hours will essentially ensure that the indicated levels of the reactive/volatile material will be too low or perhaps not detectable at all.

The two spot sampling methods that are most recommended are the fill and empty method and the helium pop method. The displacement methods also performed reasonably well during the recent API research studies.

The fill and empty method requires that the cylinder be equipped with a “pigtail” following the sample cylinder outlet valve. While leaving the sample cylinder inlet and outlet valves open, the probe outlet valve and the valve at the end of the pigtail are cycled to alternately fill and empty the sample cylinder. The pigtail ensures that the heat of compression created when the sample cylinder is filled more than offsets the Joule-Thomson cooling produced when the sample cylinder is depressurized. It does this by insuring the maximum pressure drop while depressurizing the sample cylinder is far removed from cylinder itself, at the end of the pigtail. The ability of the fill and empty procedure to actually elevate the temperature of the sampling cylinder above the flowing temperature of the stream being sampled during many operating conditions makes this method the most desirable when the ambient temperature is at or near the hydrocarbon dewpoint of the stream. The pigtail should be approximately 1/4 inch tubing and be at least 36 inches in length, although it may be coiled to make the apparatus easier to handle. The coils should not touch one another, otherwise the heat loss at the end of the pigtail may be transferred quickly across the coils to the sample cylinder. There should be another sample valve, similar to the sample cylinder valves on the outlet of the pigtail. The flow passage through this valve must not be larger than the passage through the cylinder valves. Refer to API Chapter 14.1 for the detailed procedures for performing the fill and empty method and to either API Chapter 14.1 or GPA 2166 for the number of fill and empty purge cycles required at various line pressures.

See Figures 3-30 and 3-31 for two typical fill and empty method configurations:

The helium pop method has the advantage of being a very simple and quick process, but API research showed the method to have a slight negative bias (where the indicated Btu is slightly below the actual Btu). The sample cylinder will be the standard type, properly cleaned, with a 5 psig blanket of dry helium gas in the cylinder. The cylinder is connected to the sample probe outlet valve via a short connection that includes a vent valve. The vent valve will allow a gentle purge of the volume between the probe and the sample cylinder inlet valve prior to sampling. Once the inlet system is purged, close the purge valve, then open the sample cylinder inlet valve. After a moment to allow the cylinder to equalize with line pressure, close the cylinder inlet valve – the sample has been collected. See Figure 3-32 for a typical helium pop method configuration.

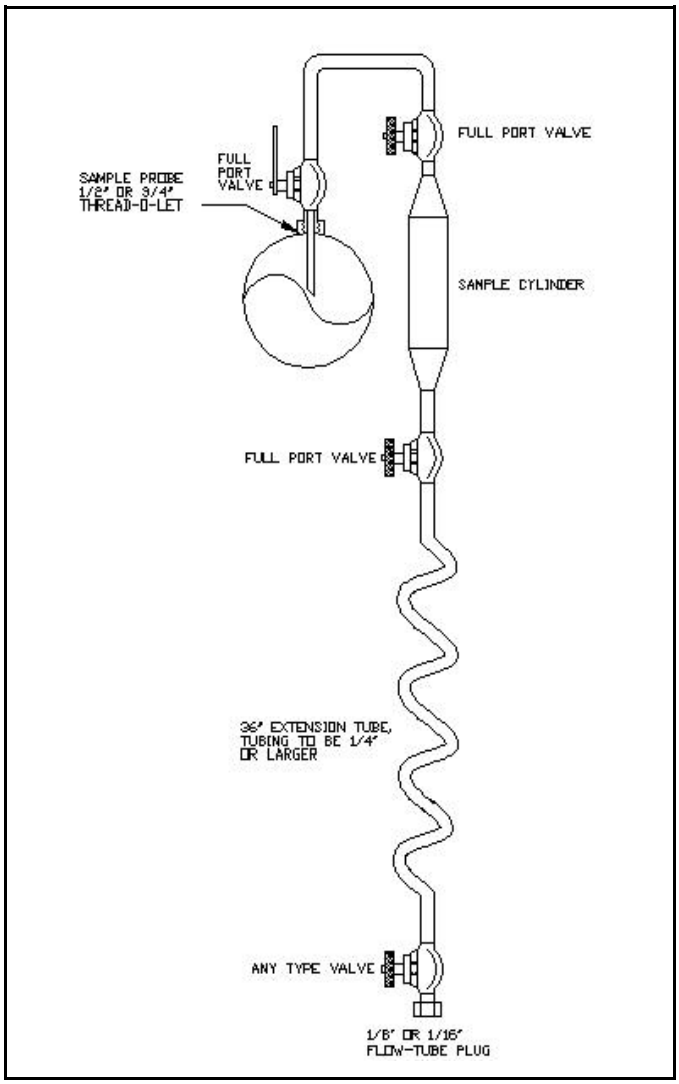
There are several other spot sampling methods detailed in the GPA 2166 standard and referenced in the API Chapter 14.1 standard. For relatively rich, unprocessed gas streams, the fill and empty, helium pop or the displacement methods should be used. If the gas is *continuously* lean and dry and sampled at conditions well above its hydrocarbon and water dewpoint, any method in the GPA standard is acceptable, even the continuous purge method. Note that the residue gas streams on the outlet of gas plants do not fit this criteria. Although much of the time the gas is lean and dry, during plant upsets the residue will contain much heavier components not stripped out in the process and may be essentially like the rich,

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unprocessed gas coming into the front end of the plant for processing. Once the residue gas metering and sampling systems are contaminated by rich, unprocessed gas, liquid contaminates from process carry-overs, or other liquid heavy components, it may take days for the sample system to collect uncontaminated samples unless the system is well-designed and the proper sampling methods are utilized.

Once a spot or composite sample has been collected, the cylinder should be removed, its valves checked to ensure they are securely closed, then caps with Teflon tape on the threads should be installed to ensure no leakage occurs during handling or transport of the cylinders. Any time the cylinder is transported, ensure that it is properly restrained or packaged and that it is properly labeled. The vehicle used to transport the cylinder must have the completed paperwork readily available and completed fully and accurately. Most companies require a sample label or tag be attached to the cylinder that shows where the sample was collected, how it was collected, who collected it and when it was collected. See the API and GPA standards for details.

FIG. 3-30
Fill and Empty Sampling Method (Alternate Configuration)

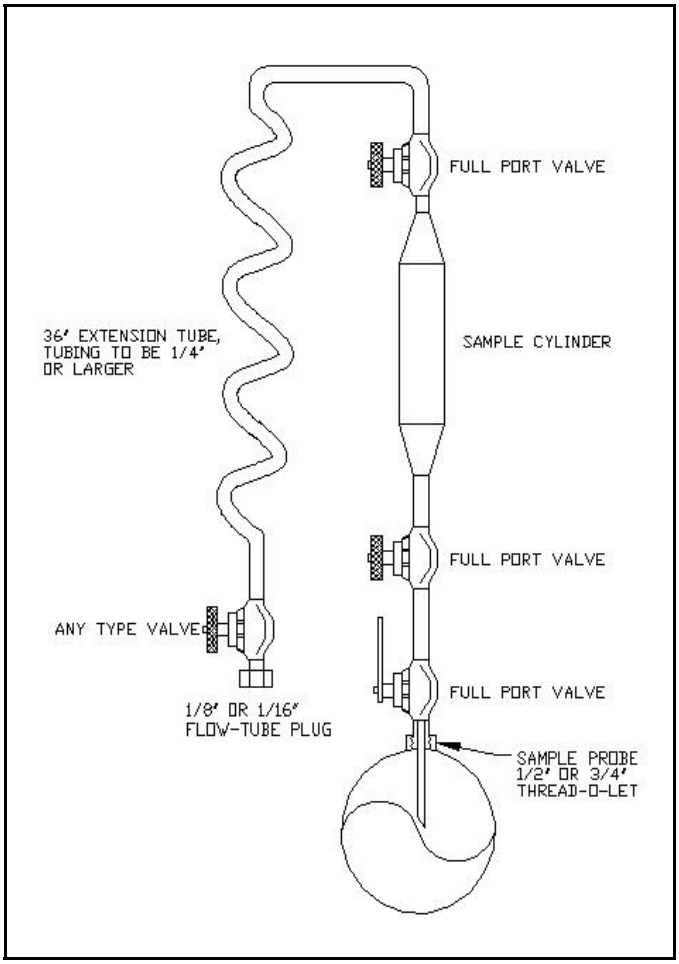


NATURAL GAS LIQUID SAMPLING

Utilizing Floating Piston Cylinders

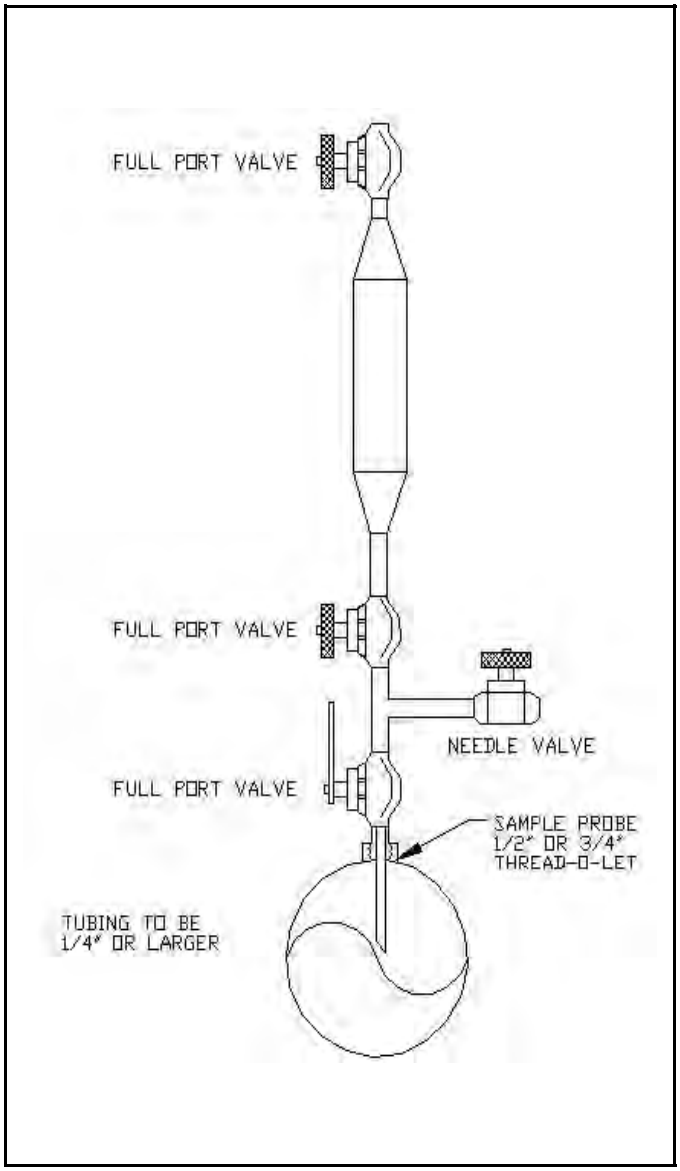
Liquid sampling requires special precautions to accumulate and transfer representative samples. Pressure in the sample cylinder and/or accumulator must be maintained at 1.5 times the product vapor pressure. Maximum product vapor pressure should be determined using the highest ambient temperature or flowing temperature (whichever is highest) to determine the minimum pre-charge pressure. A method to break up stratification must be provided prior to transfer of the sample to another container and laboratory analysis. Maintaining the appropriate pressure and mixing the sample can be satisfied by using floating piston sample cylinders with mixers (Fig. 3-33). A "rattle ball" or agitator may be used in place of the mixing rod shown. The floating piston cylinders are precharged on one end with an inert gas at a pressure 1.5 times above product vapor pressure. This prevents sample vaporization, which could result in erroneous analysis. This design also provides a compressible inert gas cushion to allow for thermal expansion of the liquid. A pressure relief valve is needed, but should it discharge, the integrity of the sample will be lost. Liquid sample cylinders shall not be filled over 80 percent full.

FIG. 3-31
Fill and Empty Sampling Method (Preferred Configuration)



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FIG. 3-32
Helium Pop Sampling Method



Samples are acquired through a sample probe inserted into the center third of the flowing stream. The probe should be mounted in the top or side of the line. Continuous samplers should use a continuous flowing sample loop (speed loop) or a probe mounted sample pump to ensure the most current sample is always added to the sample container. Speed loops must have a driving device such as an orifice, differential pump, available pressure drop, or "scoop" probes. The driving device should be sized to provide a complete exchange of liquid in the sample loop once per minute. The sample pump must be set to gather flow proportional samples to ensure a true representative sample is obtained. If flowing pressures are higher than the sample accumulator pre-charge pressure, then the sample pump must prevent "free-flowing" of product into the sample container. See Fig. 3-34 for an example continuous sampling application. Speed loop lines may require insulation when cold ambient temperatures have a significant effect on viscosity. The product in the sample container must be thoroughly mixed before being transferred to a transport cylinder. Details and alternative methods for obtaining liquid samples are found in GPA 2174.

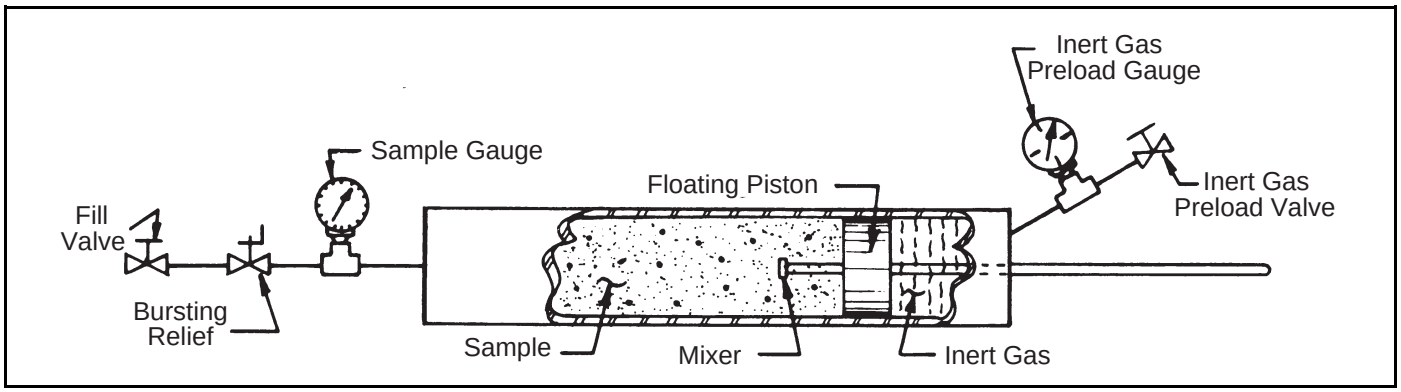
NATURAL GAS AND NATURAL GAS LIQUID CHROMATOGRAPHY

Introduction

To properly install a chromatograph system, the following must be considered:

- Product Stream
- Analytical Components of Interest
- The Analytical Method
- Carrier Gas
- Sample Preparation
- Introducing the Sample into the Chromatograph
- Column Selection
- Detectors
- Detector Signal Integration and Data Handling
- Safety Considerations (ventilation, area classification, etc.)
- Purchasing the System
- Installation
- Calibration and Verification
- Training and Follow-up

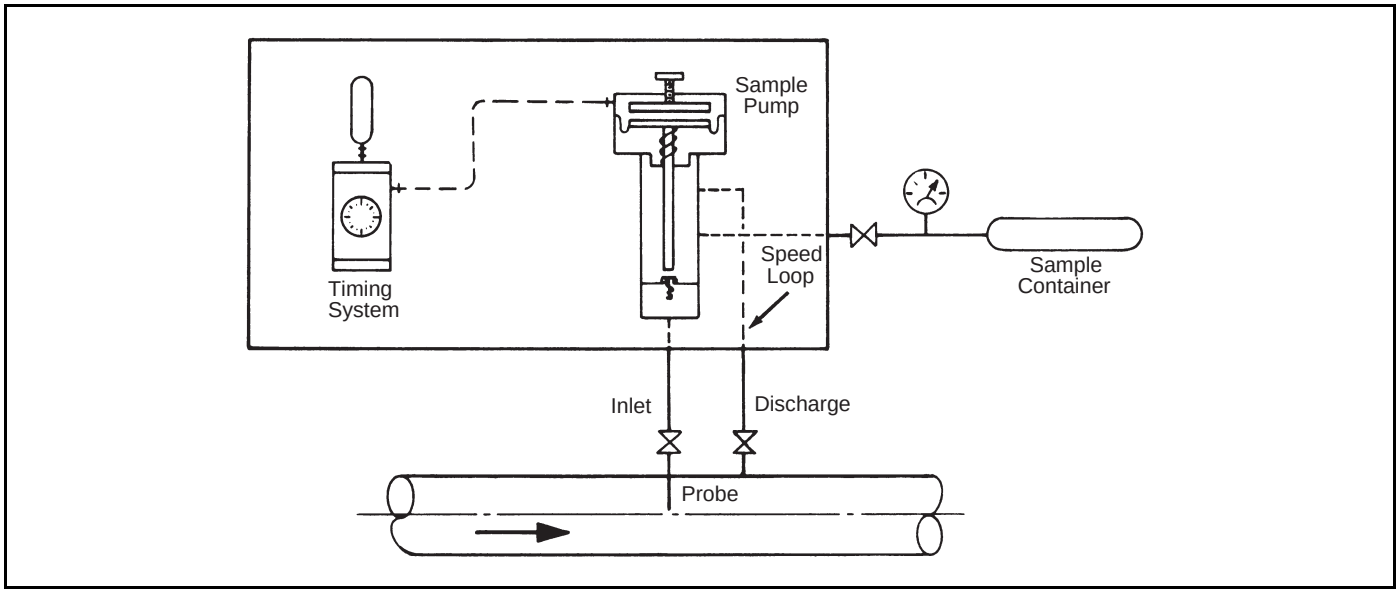
FIG. 3-33
Liquid Sample Container



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FIG. 3-34

Continuous Sampler



- Maintenance and Trouble Shooting

The following GPA Technical Standards offer guidance in Natural Gas and Natural Gas Liquids Chromatographic Analysis:

- GPA 2165: Method for Analysis of Natural Gas Liquid Mixtures by Gas Chromatography
- GPA 2177: Method for the Analysis of Demethanized Hydrocarbon Liquid Mixtures Containing Nitrogen and Carbon Dioxide by Gas Chromatography
- GPA 2186: Tentative Method for the Extended Analysis of Demethanized Hydrocarbon Liquid Mixtures Containing Nitrogen and Carbon Dioxide by Temperature Programmed Gas Chromatography
- GPA 2261: Method for Analysis of Natural Gas and Similar Gaseous Mixtures by Gas Chromatography
- GPA 2286: Tentative Method for the Extended Analysis of Natural Gas and Similar Gaseous Mixtures by Temperature Programmed Gas Chromatography
- GPA 2199: The Determination of Specific Sulfur Compounds by Capillary Gas Chromatography and Sulfur Chemiluminescence Detection
- GPA 2198: Selection, Preparation, Validation, Care and Storage of Natural Gas and Natural Gas Liquids Reference Standard Blends

The Product Stream and Components of Interest

The first step in selecting a chromatograph system is to identify the product stream and the components of interest. This vital step frequently gets skipped or receives too little attention.

"Typical" samples must be analyzed for every identifiable and significant component. One spot sample will usually not be enough.

Consideration should be made of all possible plant configurations and operating conditions that may cause compositions to change. Even the seasons of the year may need to be considered when selecting samples to develop the typical composition. For new installations where samples are not yet available, industry experts in similar applications should be consulted.

The components of interest may not always include everything in the sample. Many on-line analyzers are designed to monitor a single component or the ratio of two components. Sometimes minor components are included in peaks along with larger fractions. Speed of analysis is often important in the on-line application and discarding or ignoring components that are not of interest can help shorten analysis time. For product accounting purposes, a detailed laboratory analysis of the hexanes-plus components may require extended analysis time. Quite often, it is necessary to make choices between the speed of analysis and the detail or the accuracy of analysis.

The Analytical Method

Natural gas samples are typically heated to 140 ºF to completely vaporize the sample in the cylinder. Samples withdrawn from this 100% vapor sample represent the composition of the entire cylinder and withdrawal of the sample will not compromise the sample left in the cylinder. Of course, the sample line from the cylinder to the chromatograph sample valve and the sample valve itself must be heated to prevent condensation of heavy (low volatility) components in the sample.

The analytical method includes sample introduction, component separation in the column, detection and component quantification. There are many approaches to selecting the method. Personal experience and the complexity of the analysis determine these approaches.

The simplest approach is to select an industry standard (refer to GPA standards listed in the introduction) or to choose a vendor-supplied method where all the development and verification work has already been done. "Fine tuning" an analytical method can be very time consuming, so if a vendor has an off-the-shelf application that matches the need, it may be wise to choose his system.

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Colleagues in the industry may have already developed a method that performs the required analysis. Usually, they are willing to share their experience.

Since method development is time consuming, developing one's own method should be considered as a last resort. One would choose to develop his own if the required method does not exist or if there are special requirements of the method such as increased speed or improved selectivity. Complete method development includes sample preparation and introduction, component separation, detector selection, peak integration and data handling.

Carrier Gas

Carrier gas "carries" the sample through the chromatograph system from the sample introduction system to the detector. It is typically Ultra High Purity Helium. Hydrogen or other gases may be selected for special applications.

Sample Preparation and Introduction

The sample preparation and introduction portion of the chromatograph system is arguably the most critical. A chromatograph cannot produce good results without a good sample.

The sample must be withdrawn from the pipe either as 100% liquid or 100% vapor. For Natural Gas samples, a vaporizer regulator at the sample point adds heat to the sample and reduces the pressure of the sample to ensure that it is completely vaporized. The transfer lines to the chromatograph for a vapor sample loop must also be heated to prevent condensation of heavy components.

Sample preparation and introduction for on-line chromatograph systems presents more of a challenge than for laboratory instruments. Of course, the most ideal on-line chromatograph would sit on top of the pipe and directly draw the sample from the system. But the most ideal location for the chromatograph may be some distance from the sample point.

Liquid sample speed loops transport the liquid sample from the sample point to the chromatograph for sampling and back to a lower pressure location in the product piping system. Just as in laboratory chromatograph systems, filtration and moisture removal are important. The system must clean and/or dehydrate the stream without changing the composition of the sample to be analyzed.

Tubing from the sample point to the on-line chromatograph must be small in diameter to ensure high velocities: the analytical results must be as close to real-time as possible. A sample that has been traveling in a large diameter loop for any length of time would only produce historical results.

In laboratory systems, it is important that the sample in the sample cylinder is either completely in the liquid phase or completely in the vapor phase. In a two-phase state, the vapor portion will have a higher concentration of the more volatile (light ends) components than the sample and the liquid phase will have a higher concentration of the less volatile (heavy ends) components. A sample withdrawn from the top or from the bottom of the sample cylinder will not truly represent the composition of the entire cylinder. Sample withdrawal from either phase will irrevocably change the composition of the sample left in the cylinder.

Natural convection of a heated, vaporized sample will keep the sample mixed and homogenous. Samples that are completely liquid must be mechanically mixed to ensure a homogenous sample for analysis.

The introduction of a natural gas sample into the chromatograph includes drawing a vacuum on the sample tubing to assist in vaporizing and removing residual components from previous samples. The sample loop is then purged (typically at (or below) atmospheric pressure) through the sample loop before injection.

It is necessary to maintain pressure well above bubble point pressure on the liquid sample during sample withdrawal. This can be achieved by using an appropriate displacement medium or by using a floating piston/positive displacement sample cylinder. The transfer lines from the sample cylinder to the liquid sample valve and the liquid sample valve itself must remain cool to prevent vaporization of the lighter (more volatile) components.

A block valve/purge control valve system is installed downstream of the liquid sample valve to maintain pressure on the liquid sample until the sample is injected. This system is also used to purge the sample through the sample valve.

It may be necessary to install filters and moisture traps in the sample lines that lead to the gas or liquid sample valves. Of course, the filters and traps must not alter the composition of the sample.

Since most of the troubleshooting problems in a chromatographic system are associated with sample preparation and sample introduction, it is very important to design or select this portion of the system with care.

Column Selection

The column has been called the heart of the chromatographic system. It separates the components of interest from the rest of the sample. Column selection, installation and setup can be as much an art as a science. The first approach should be to review industry standards for an appropriate method. Others in the industry may already be performing the desired analysis and be willing to share information. Vendors may have already developed an appropriate method and sell the entire system as a bundled package.

If an already existing method cannot be found, columns can be selected from vendor catalogs. These catalogs usually include chromatogram examples of column performance. Columns may separate components based on boiling points or by chemical polarity. Columns may be packed (for larger volume samples) or capillary (for more efficient separation).

Optimum column temperatures and carrier gas flow must be determined. Increasing flow rates generally reduces individual component retention time and reduces sharpness of resolution. Increasing temperature has the same effect as increasing flow.

Sometimes combinations of columns can be used to achieve the best results. Special valving can selectively direct sample flow to different columns at different times in the analytical process. Valving and trap columns can be used to direct some components to the detector and other components to be vented to the atmosphere.

Creative temperature programming of the column oven can create optimum separation conditions at appropriate times during the sample analysis. Components with the low-boiling-

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point temperatures elute from the column first and are usually separated from each other at lower column temperatures. Higher boiling point components come off the column rather slowly and increased temperatures help to speed the process. An oven temperature program that starts off rather cool and heats up as the analysis is in progress may help the column separate both the low and high boiling point components effectively.

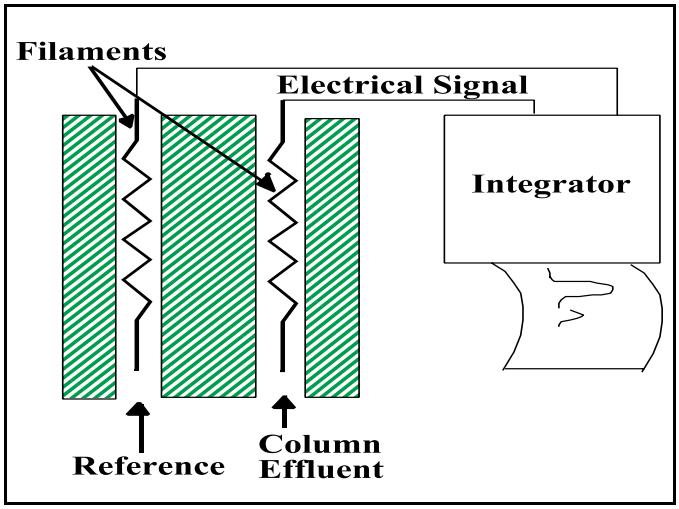
Detectors

The thermal conductivity detector is considered a universal detector. It will detect any component that has a thermal conductivity different from the carrier gas. Helium is the most frequently used carrier gas. It has greater thermal conductivity than any other component except hydrogen.

The thermal conductivity detector is a heated wire filament or thermistor bead. See Figure 3-35. Its electrical conductivity is partially determined by its temperature. As product components elute from the end of the column, they cause the detector temperature to increase as compared to the temperature of a filament with pure carrier gas (reference gas) flowing. This changes the electrical conductivity and the changed electrical signal is electronically interpreted by the integrator or computer system to produce a chromatogram peak whose area is proportional to the composition of the component.

Unfortunately, the thermal conductivity detector is not extremely sensitive. It may not be the detector of choice for trace components.

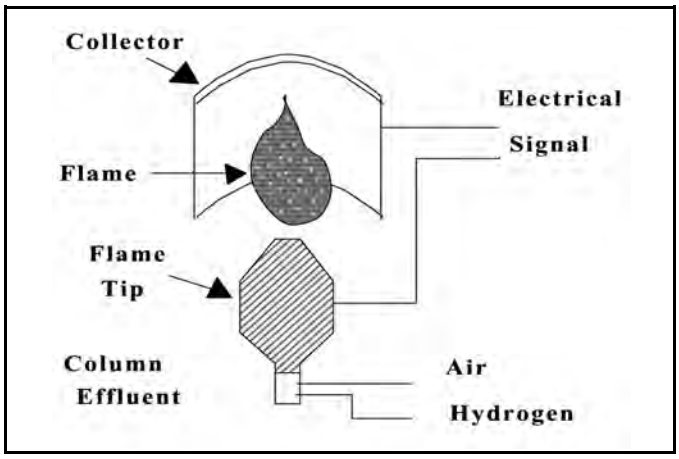
FIG. 3-35
Thermal Conductivity Detector



The flame ionization detector (Figure 3-36) is more sensitive than the thermal conductivity detector for hydrocarbons and many other materials. This detector is not sensitive to all the components found in natural gases and natural gas liquids. (Carbon Dioxide and Nitrogen, for example). The presence of the flame may be objectionable in some environments and must be fully considered.

The effluent from the column flows through a hydrogen-air flame. The flame jet is electrically charged. As the hydrocarbons are broken up in the flame, ionized particles are formed. These particles then flow across an electrically charged collector that develops an electrical signal proportional to the concentration of the component.

FIG. 3-36
Flame Ionization Detector



There are many specialized detectors that selectively detect only certain components such as sulfur or halogens.

Integration and Data Handling

Integration and data handling can be very simple or extremely complicated. An on-line analyzer may produce a simple chart recorder trace to indicate the concentration of a component of interest. A laboratory chromatograph might have a simple integrator to print out a calibrated report form.

Integrated peak information can be imported into other PC-based programs to do post-run calculations. These results can be carried over a laboratory information management system to provide data throughout the laboratory or the company. On-line chromatographs can provide a broad spectrum of alarm and control functions.

Calculation and data handling packages can be purchased as bundled packages hired out to contract programmers or developed in house. It is important to establish communication protocol between individual parts of the chromatographic system before purchasing equipment. It would be very embarrassing to have a chromatograph and an integrator that could not communicate with each other.

Safety Considerations

Consult the electrical and ventilation codes prior to purchasing the system. The location of the chromatograph system may dictate electrical class/division requirements. Hazardous areas or sample components may dictate special ventilation requirements. Hazardous substance detectors, alarms and shutdowns may be required.

Purchasing the System

When selecting a vendor, several things should be considered. Personal experience with a vendor may heavily influence the decision. Company-wide ownership of certain brands of equipment might tip future purchases in favor of that equipment due to spare parts inventory and local expertise considerations.

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Local vendor support should be considered. Are spare parts readily available? Is there a resident vendor technician? Is training provided?

Before selecting a vendor, it is important to list all criteria. The criteria should then be prioritized. Sometimes this process makes the vendor choice obvious. Visiting other facilities with systems in operation may help in the selection.

Consider different approaches to purchasing. Some vendors will place their equipment on approval. Sometimes it makes more sense to rent or lease the equipment. Check for sale or trade-in options.

Installation

Proper site preparation can dramatically simplify installation of the equipment. Pay careful attention to manufacturer provided physical dimensions; few things are as embarrassing as to discover that a piece of equipment will not fit. Room to perform maintenance functions and for expansion should be planned.

Verify electrical power. Many laboratory chromatographs require a 20-amp receptacle. Power supplies to laboratory equipment should have isolated grounding.

Close attention should be paid to the vendor-specified quality of bottled gasses. All gas and sample lines should be thoroughly cleaned before installation of the chromatograph system.

Communications equipment should be verified. Has the proper wiring been run? Are the appropriate connectors attached?

During the actual setup process, it is important to document everything. Fine-tuning of a chromatograph system can be time consuming. Each change and its results should be noted. Performance evaluation and calibration information should be retained. Careful record keeping and documentation of initial parameters can be invaluable in future troubleshooting.

Calibration and Verification

Calibration should be performed in accordance with the referenced industry analytical methods. Note that calibration standards should be gravimetrically prepared (weighed components).

Verification of the calibration may include any or all of the following:

- Plot of Molecular weight of components against calibration response factors as outlined in Appendix B of GPA 2198.
- Running the calibration as an unknown on a chromatograph that is calibrated to a verified standard.
- Comparing results from the newly calibrated standard with the results of another laboratory. See appendix on laboratory certification in API 14.1 and GPA 2177 for comparison guidelines.

Training and Follow Up

The importance of training cannot be overemphasized. It is irresponsible to install a piece of equipment and leave without ensuring that the operators and technicians have sufficient knowledge to use and maintain the system. Frequent follow up helps to "sell" the system to field personnel and to ensure that the newly installed system is performing up to expectations.

Maintenance and Troubleshooting

Appropriate maintenance schedules should be established. Visual checks of system performance should be done frequently. Peak shapes and retention times can tell a great deal about the condition of the system. Accuracy checks should be done on a frequent basis.

Carrier gas flows and pressures should be checked at least daily. The carrier gas must be Ultra High Purity. Sample conditioning systems need to be checked. Filter change schedules should be determined based on the history of the system. Sample systems for on-line chromatographs require frequent attention.

Chromatographic troubleshooting has been the topic of several books. It would be impossible to do justice to the subject in this paper. But a few basic techniques can be discussed.

The first order of troubleshooting is to ask, "What is different?" Sometimes clearly identifying the symptoms that are different from the norm will make the solution obvious. Carefully compare the "normal" conditions documented at system startup to existing conditions.

A second question to ask is, "What have I changed?" Has a new piece of equipment been added to the power supply source? Have carrier gas bottles just been changed?

Many subsystems work together to produce the final chromatographic report: sample conditioning and injection; column condition, flow rates, temperatures; detector performance; integrator hardware and software functions; secondary calculation, alarm and control functions.

The best approach to these subsystems is to divide and conquer. As an example, carrier gas flow can be diverted directly to the detector in order to isolate possible column problems from the system. By selectively isolating each subsystem, the chromatograph troubleshooting process can be greatly simplified.

Summary

Selection, installation and operation of a chromatograph system need not be a difficult task if careful attention is paid to the details outlined in this paper. Prior planning of sample systems, column selection, detectors and data handling equipment can ensure a successful installation.

Note: The CD located on the inside cover of this book contains volume calculation spreadsheets for your use.

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2. Manual of Petroleum Measurement Standards, American Petroleum Institute, Washington, D.C.
3. GPA Standard 2145, Table of Physical Constants for Hydrocarbons and Other Components of Interest to the Natural Gas Industry, Gas Processors Association, Tulsa, OK.
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SECTION 4

Instrumentation

Instrumentation in a gas processing plant is usually comprised of a system of pneumatic, hydraulic, and electronic devices for measurement and control of all the process variables (pressure, flow, temperature, etc.) which are pertinent to the operation of the plant. In addition, computers are normally included in the instrumentation system to handle functions such as data gathering and transmission, bulk data storage, display, alarms, logging, computations, and control. Since the advent of integrated circuit electronics, specifically the microprocessor, many types of instruments are becoming more intelligent or "computerized."

The basic purposes of this section are to provide:

- A ready reference of definitions and symbols associated with gas plant instrumentation.
- Guidelines and design information for good process measurement, signal transmission, signal indication, and control response.
- A reference of terminology which pertains to the instrumentation of gas plants and related facilities.

FIG. 4-1

Nomenclature

Controller Symbol	Description	Output Expression
P	Proportional	$CO = K_p (PV - SP) + MO$
I	Integral (Reset)	$CO = K_i \int (PV - SP)\, dt$
D	Derivative (Rate)	$CO = K_d[d(PV - SP)/dt]$

C_v = valve flow coefficient
CO = controller output
d = valve inlet diameter
D = internal diameter of the pipe
F_d = valve style modifier
F_F = liquid critical pressure ratio factor, dimensionless
F_k = ratio of specific heats factor, dimensionless
F_L = liquid pressure recovery factor of a valve without attached fittings, dimensionless
F_p = piping geometry factor, dimensionless
F_R = Reynolds number factor, dimensionless
G_f = liquid specific gravity at upstream conditions, ratio of density of liquid at flowing temperature to density of water at 60° F, dimensionless
G_g = gas specific gravity (ratio of density of flowing gas to density of air with both at standard conditions, which is equal to the ratio of the molecular weight of gas to the molecular weight of air), dimensionless
k = ratio of specific heats, dimensionless
K = gain
K_c = cavitation index, dimensionless
K_d = derivative gain constant
K_i = integral mode gain constant
K_n = normalization constant
K_p = proportional mode gain constant
K_u = ultimate sensitivity
M = molecular weight, atomic mass units
MO = manual-mode controller output
$N_1, N_2 ... N_x$ = numerical constants for units of measurement used
P_c = absolute thermodynamic critical pressure
P_u = ultimate time period
P_v = vapor pressure of liquid at valve inlet temperature, psia
P_1 = upstream absolute static pressure, measured two nominal pipe diameters upstream of valve-fitting assembly, psia
P_2 = downstream absolute static pressure, measured six nominal pipe diameters downstream of valve-fitting assembly, psia
ΔP = pressure differential, $P_1 - P_2$
PB = proportional band setting
PB_u = ultimate proportional band setting
PV = process variable measurement
(PV – SP) = error signal
q = volumetric flow rate
Q_g = gas or vapor flow rate, lb/hr or cu ft/hr
Q_L = liquid flow rate, gpm
SP = setpoint
T = absolute temperature of gas at inlet, °R
T_c = time constant
T_d = derivative mode time constant
T_i = integral mode time constant
T_t = absolute upstream temperature (in degrees R)
w = weight or mass flow rate
X = ratio of pressure drop to absolute inlet pressure ($\Delta P/P_1$), dimensionless
X_c = pressure drop ratio for the subject valve at critical flow, with $F_k = 1.0$, dimensionless

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FIG. 4-1 (Cont'd)

Nomenclature

Y = expansion factor, ratio of flow coefficient for a gas to that for a liquid at the same Reynolds number, dimensionless

Z = compressibility factor, dimensionless

γ_1 = specific weight, upstream conditions

ξ = damping factor

A/D: Analog-to-digital.

Actuator: A device which accepts the output from a control system and moves a final control element (usually a valve) to change a process condition. See also "Final Control Element."

Adaptive Control: Method of control whereby tuning (response) of the control system is varied with the process conditions, unlike other control where tuning is manual and remains constant.

Algorithm: Mathematical representation of the action performed by a controller such as proportional, integral, derivative, or combinations of those modes.

Alphanumeric: A character set that contains both letters and digits and perhaps other characters such as punctuation marks.

Analog Computer: A computing device comprised of functional modules such as amplifiers, multipliers, dividers, etc., interconnected in such a way as to facilitate the solution of a set of mathematical expressions or to implement some control strategy. The input to and the output from an analog computer are continuous signals as contrasted with a digital computer which updates an output every scan.

Attenuation: An increase or decrease in signal magnitude between two points or between two frequencies.

Baud Rate: The number of bits or discrete pieces of information transmitted per second.

Bit: Abbreviation for "binary digit." A single character in a binary number, represented by zero (0) or one (1).

Byte: The number of adjacent binary digits operated upon as a unit.

Cascade Control: Controllers arranged such that the output of one controller manipulates the setpoint input of a second controller instead of manipulating a process variable directly.

Control Action, Derivative (Rate): Control action in which the controller output is proportional to the rate of change of the input.

Control Action, Integral (Reset): Control action in which the controller output is proportional to the time integral of the error signal.

Control Action, Proportional: Control action in which the controller output has a linear relationship to the error signal.

Controller: A device which receives a measurement of the process variable, compares that measurement with a setpoint representing the desired control point, and adjusts its output based on the selected control algorithm to minimize the error between the measurement and the setpoint. If an increase in the measured process variable above the setpoint causes an increase in the magnitude of the controller output, the controller is said to be "direct acting." If a process variable increase above the setpoint causes a decrease in the magnitude of the controller output, the controller is "reverse acting."

Data Base: A collection of values for process variables, setpoints, scaling factors, control parameters, limits, constants, identifiers, etc. for access by the application programs in a computer-based control system.

Data Highway: A high-speed serial or parallel data path which connects several units of a distributed control or data collection system.

DDC: **Direct Digital Control**. A control technique in which a digital computer is used as the sole controller and its output is used to set the final control element. This is in contrast to supervisory control.

Dead Band: The range through which an input may vary without changing the output. In a mechanical instrument such as a meter movement or strip-chart recorder, the dead band is caused by friction and slack or "play" in the readout mechanism. In a controller, dead band is a zone around the setpoint in which the measurement may vary without initiating a compensating controller response.

Dead Time: The interval of time lag between the initiation of a controller output or stimulus and the start of the resulting observable process response.

Dedicated Control: Using one controller to control one process variable.

Derivative Control: A mode of control using an algorithm which anticipates when a process variable will reach its desired control point by sensing its rate of change. This allows a control change to take place before the process variable overshoots the desired control point. See also "Control Action, Derivative (Rate)."

Derivative Time: The time difference by which the output of a proportional-derivative (PD) controller leads the controller input when the input changes linearly with time.

Digital Computer: An electronic machine for performing calculations on discrete quantities of data. Usually includes bulk storage devices such as disks, tape units, etc., in addition to internal memory. Also includes devices for printing and/or displaying output data.

Distributed Control System: Any control system in which the degradation or failure of any single element will affect only the control loop, or related loops, in which it operates.

EPROM (Erasable Programmable Read-Only Memory): A memory device with information placed into it during manufacture that cannot be altered by the computer. It can only be erased and reprogrammed with special equipment.

Error Signal: The signal resulting from the difference between the setpoint reference signal and the process variable feedback signal in a controller.

Feedback Control: A type of control whereby the controller receives a feedback signal representing the condition of the controlled process variable, compares it to the setpoint, and adjusts the controller output accordingly.

Feedforward Control: A type of control which takes corrective action based on disturbances before the process variable is upset.

Final Control Element: That component of a control system (such as a valve) which directly changes the manipulated variable.

Gain: The ratio of change in output divided by the change in input that caused it. Both input and output must be in the same units; hence gain is a dimensionless number.

Hierarchy: The ranking or precedence of the elements in a supervisory system. For example, a lower ranking element such as a local controller affects only one variable while a

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higher ranking element such as a computer might affect many variables.

Hysteresis: Difference between upscale and downscale output in instrument response when subjected to the same input approached from opposite directions.

Integral (Reset) Control: A control algorithm which attempts to eliminate the offset (caused by proportional control) between the measurement and setpoint of the controlled process variable. See also "Control Action, Integral (Reset)."

Integral (Reset) Time: The proportionality constant in the equation relating the controller output to the error for integral control $CO = K_i \int (PV - SP)\, dt$.

Where: $K_i = K_p/T_i$. K_p is the integral gain of the controller. T_i is the time required to produce a change in controller output equal to the change in error input.

Integral Windup/Windown: Also called "controller windup/windown" or "reset windup." Saturation of the controller output at its maximum positive or negative value due to an error signal existing for an excessive period of time. Can be caused by the controller being left on "automatic" when the measurement transmitter is out of service.

Intrinsically Safe: Refers to equipment or wiring which is incapable of releasing sufficient electrical or thermal energy under either abnormal or normal conditions to cause ignition of a specific hazardous atmospheric mixture in its most easily ignited concentration.

I/P Transducer: (Current-to-pneumatic). A device which converts an electrical current signal to a proportional pneumatic signal for the purpose of interfacing electronic and pneumatic parts of a control system. A typical I/P transducer might convert a 4-20 mA signal from an electronic controller to a 3-15 psig signal to actuate a pneumatic valve.

I/O Devices: Input/output devices used to enter data into and receive data from a computer or control system. Examples are analog and digital input and output devices for handling process measurements and conditions as well as "business" type devices such as terminals, printers, plotters, etc.

Noise: In process instrumentation, an unwanted component of a signal or variable. Noise may be expressed in units of the output or in percent of output span.

Offset: The steady-state deviation of the controlled variable from the set-point, usually caused by a disturbance or a load change in a system employing a proportional-only controller such as a level controller. Offset will eventually be reduced to zero by the integral action in a PI or PID controller.

P Controller: A controller which produces proportional control action only.

Parallel Data: Data transmission where all data bits of a data word are processed at once.

PD Controller: A controller which produces proportional plus derivative (rate) control action.

PI Controller: A controller which produces proportional plus integral (reset) control action.

PID Controller: A controller which produces proportional plus integral (reset) plus derivative (rate) control action.

PROM (Programmable Read-Only Memory): A device with information placed into it during manufacture that cannot be altered by the computer. It can, however, be reprogrammed using special equipment.

Proportional Band: The change in the controller error signal required to produce a full range change in output due to proportional control action. It is the reciprocal of gain expressed as a percentage: PB(%) = 100/k.

Proportional Control: A mode of control using an algorithm which causes the output of a controller to change in a linear fashion as the error signal (process variable – setpoint difference) changes. See also "Control Action, Proportional."

RAM (Random Access Memory): Memory which contains no pre-programmed information but is loaded and/or altered by the computer system. It is of a "volatile" nature in that all the contents are lost when electrical power is removed. RAM memories are usually provided with battery backup power systems, making it "non volatile."

Ramp: An increase or decrease of a variable at a constant rate of change with respect to time.

Reset Rate: The inverse of integral time; usually expressed as "repeats per minute."

ROM (Read-Only Memory): Memory with information placed into it during manufacture that cannot be altered.

Serial Communications: Sending bits of information in succession along a single circuit (pair of wires).

Setpoint: The desired value at which a process variable is to be controlled.

Software: A set of programs and associated data tables which causes the hardware components of a computer system to perform the desired tasks.

Split-Ranging: Action in which two or more final control elements are actuated by a single controller output. For example, in a heating circuit, 0-50% of the controller output operates a primary heat source and the 50-100% portion of the controller output operates a secondary heat source.

Steady-State: The condition when all process properties are constant with time, transient responses having died out.

Supervisory Control: A method of computer control whereby a computer or master station provides setpoints to individual controllers which independently perform the actual control algorithms.

System Control Diagram: A diagram used to define the process functionality to achieve the overall operating and control philosophy.

Telemetry: A technique which permits a measured quantity to be transmitted and interpreted at a distance from the measuring location. Form, or types of telemetry include analog, digital, frequency, and pulse.

Transmitter: A device that converts a process measurement (pressure, flow, level, temperature, etc.) into an electrical or pneumatic signal suitable for use by an indicating or control system.

Word, Computer: A group of bits treated as a unit and capable of being stored in one computer location. Some common word lengths are 8 bits, 16 bits, and 32 bits.

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GENERAL INSTRUMENTATION CONSIDERATIONS

Type Selection

Often the type selection of an instrument is pre-determined by whatever is available, or what will be compatible with the rest of a system. There are cases, however, where the choice to install pneumatic or electronic instrumentation must be made by comparing the features of each type. Fig. 4-3 lists some of the attributes of each type to aid in this comparison.

Identification

An instrument may perform a single function such as a temperature indicator (TI), or a combination of functions such as a flow recording controller (FRC). Fig. 4-2 covers the common symbols on process and mechanical flow sheets, also called Piping and Instrument Diagrams (P&IDs). Table 1 in Fig. 4-2 shows the accepted Instrument Society of America (ISA) letter designations and their meanings when used in instrument identifications. Special identification requirements may be encountered in certain applications, e.g., offshore requirements of API-RP 14-C. Instrument Society of America publication ISA-S5.1, "Instrumentation Symbols and Identification," should be referred to for more detailed information.

PNEUMATIC POWER SUPPLIES

The pneumatic power supply is more commonly known as the instrument air system. The main considerations of an instrument air system are:

1. Adequate Capacity: The minimum capacity of the system should be the sum of the individual requirements of each air-consuming instrument in the system, plus a supplemental volume for purges, leaks, additions, etc. If accurate consumption figures are not available, an estimated consumption volume of 0.5 cubic foot per minute for each air-consuming device is usually adequate. The air storage tank should have sufficient capacity to maintain this rate for about five minutes or such time as is considered adequate to perform an emergency shut-down of the plant or to switch over to a backup air system. Also the air storage tank capacity should be large enough to prevent excessive cycling of the compressor.
2. Filtering and Regulation: Instrument air systems are normally designed for pressures up to 125 psig and should be protected by relief valves. Instrument air should be free from all contamination such as oil, water, and any hazardous or corrosive gases. Non-lubricated compressors should be used if possible. Where lubricated compressors are used, an oil removal separator is required. The presence of oil may cause instrument contamination and possibly create a combustible mixture. After being compressed, instrument air must be cooled to remove the major portion of the contained water. A final drying system must be used to reduce the water dewpoint to at least 10°F below the ambient temperature at line pressure. An afterfilter may be required to remove particulate carryover from the dehydrators.
3. Proper Distribution: The air distribution system should be free of any "pockets" where liquid could accumulate. If this is not possible, drain valves should be installed. All supply lines should connect to the top of the air manifold or "header." Instrument air filter-regulators should be provided at each air-consuming device to reduce the line pressure to the supply pressure recommended by the instrument manufacturer. This also provides one more stage of protection from contaminants. Instrument Society of America Standards ISA-S7.3 and ISA-S7.4 are references for additional information.
4. Non-Air Systems: Natural gas has been used instead of instrument air in some remote installations where compressed air was not available. This practice should be avoided if at all possible due to safety and pollution problems and the additional filtering and clean-up of the gas which must be done to protect the instruments. The user must be cognizant of all applicable regulations when considering the use of any combustible gas in instrumentation service. Some small-scale systems have used bottled nitrogen for instrument gas. This is quite acceptable, but non-bleed type instruments should be used to keep the consumption to a minimum.
5. Hydraulic Powered Devices: Hydraulic actuators are sometimes used on valves or rams where very high thrusts (up to 50,000 pounds force) are required for operation. Due to the problems of transmitting very high pressure signals, a local pump powered by an electric motor is often used to form what is commonly known as an "electro-hydraulic actuator."

ELECTRONIC POWER SUPPLIES

Installation and interconnection wiring requirements are regulated by the National Electrical Code (NEC) Article 500 (Hazardous Locations) and Article 725 (Remote Control & Signal Circuits). Special attention should be given to Article 725. The requirements pertaining to physical protection of wiring, isolation and spacing of conductors depending upon class, and minimum wire sizes are often overlooked in an instrumentation installation.

Power Outages and Interruptions

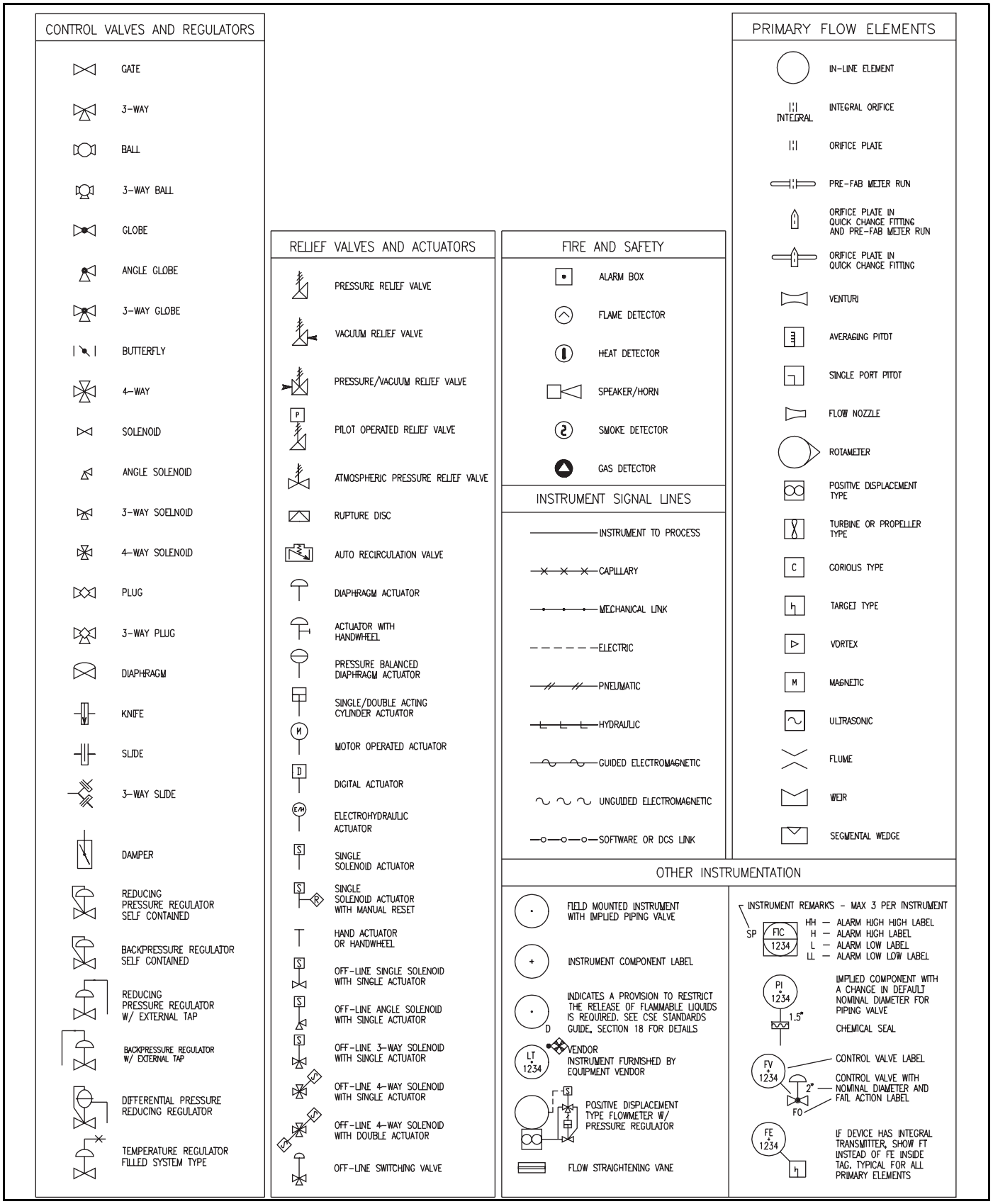
It is usually the responsibility of the consumer, not the electric utility company, to provide protection for connected electronic equipment against upsets such as voltage spikes caused by lightning, high or low voltage surges, etc. The frequency of power outages and average time for service to be restored should be determined to assist in the design of electronic power supply protection and battery backup systems. The power company should be able to provide data about their reclosure gear (equipment which attempts to restore service after a current surge has tripped the substation or sectionalizing breakers). Also, a record of power outages in the local substation area and storm frequency charts will be very useful. A typical reclosure operation description is shown in Fig. 4-4.

Example 4-1 — If a plant can tolerate loss of power to its electronic equipment for six seconds, and an average of 50 power outages per year is expected, then, according to Fig. 4-4, 84% of those outages will be restored on the first reclosure attempt, and the remaining 16% or approximately eight power outages per year can be expected to disrupt plant operations. Note: Storm frequency charts are often available from manufacturers of surge arresting devices. These charts may be used in case power outage records are not available from the power company. Climatic data for a particular area may be obtained from the National Oceanic and Atmospheric Administration (NOAA) in Asheville, North Carolina.

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FIG. 4-2

Instrumentation Symbols



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FIG. 4-2 (Cont'd)

Instrumentation Symbols

INSTRUMENT OR FUNCTION SYMBOLS


FIELD MOUNTED INSTRUMENT


INSTRUMENT MOUNTED ON MAIN CONTROL ROOM PANEL


INSTRUMENT MOUNTED BEHIND MAIN CONTROL ROOM PANEL


INSTRUMENT FUNCTION MOUNTED ON LOCAL PANEL


INSTRUMENT FUNCTION MOUNTED BEHIND LOCAL PANEL


INSTRUMENT SHOWN ON ANOTHER DRAWING


FIELD MOUNTED LIGHT


LIGHT MOUNTED ON MAIN CONTROL ROOM PANEL


LIGHT MOUNTED ON LOCAL PANEL


LOCAL DCS SHARED CONTROL FUNCTION


PRIMARY DCS SHARED DISPLAY SHARED CONTROL FUNCTION ACCESSIBLE TO OPERATOR


PRIMARY DCS SHARED DISPLAY SHARED CONTROL FUNCTION INACCESSIBLE TO OPERATOR


AUXILIARY DCS SHARED DISPLAY SHARED CONTROL FUNCTION ACCESSIBLE TO OPERATOR


AUXILIARY DCS SHARED DISPLAY SHARED CONTROL FUNCTION INACCESSIBLE TO OPERATOR


HARD WIRED INTERLOCK


LOCAL PLC SHARED CONTROL FUNCTION


PRIMARY PLC SHARED DISPLAY SHARED CONTROL FUNCTION ACCESSIBLE TO OPERATOR


PRIMARY PLC SHARED DISPLAY SHARED CONTROL FUNCTION INACCESSIBLE TO OPERATOR


PRIMARY COMPUTER SHARED DISPLAY AUXILIARY PLC SHARED DISPLAY SHARED CONTROL FUNCTION ACCESSIBLE TO OPERATOR


PRIMARY COMPUTER SHARED DISPLAY AUXILIARY PLC SHARED DISPLAY SHARED CONTROL FUNCTION INACCESSIBLE TO OPERATOR


LOCAL COMPUTER SHARED CONTROL FUNCTION


SHARED CONTROL FUNCTION ACCESSIBLE TO OPERATOR


SHARED CONTROL FUNCTION INACCESSIBLE TO OPERATOR


AUXILIARY COMPUTER SHARED DISPLAYED SHARED CONTROL FUNCTION ACCESSIBLE TO OPERATOR


AUXILIARY COMPUTER SHARED DISPLAYED SHARED CONTROL FUNCTION INACCESSIBLE TO OPERATOR

INSTRUMENT SYMBOLOGY DEFINITION

MEANINGS OF IDENTIFICATION LETTERS

THIS TABLE APPLIES ONLY TO THE FUNCTIONAL IDENTIFICATION OF INSTRUMENTS

	FIRST-LETTER		SUCCEEDING-LETTERS		
	MEASURED OR INITIATING VARIABLE	MODIFIER	READOUT OR PASSIVE FUNCTION	OUTPUT FUNCTION	MODIFIER
A	ANALYSIS		ALARM		
B	BURNER, COMBUSTION		USER'S CHOICE	USER'S CHOICE	USER'S CHOICE
C	USER'S CHOICE			CONTROL	
D	USER'S CHOICE	DIFFERENTIAL			
E	VOLTAGE		SENSOR (PRIMARY ELEMENT)		
F	FLOW RATE	RATIO (FRACTION)			
G	USER'S CHOICE		GLASS, VIEWING DEVICE		
H	HAND				HIGH
I	CURRENT (ELECTRICAL)		INDICATE		
J	POWER	SCAN			
K	TIME, TIME SCHEDULE	TIME RATE OF CHANGE		CONTROL STATION	
L	LEVEL		LIGHT		LOW
M	USER'S CHOICE	MOMENTARY			MIDDLE, INTERMEDIATE
N	USER'S CHOICE		USER'S CHOICE	USER'S CHOICE	USER'S CHOICE
O	USER'S CHOICE		ORIFICE, OPEN		
P	PRESSURE, VACUUM		POINT (TEST) CONNECTION		
Q	QUANTITY	INTEGRATE, TOTALIZE			
R	RADIATION		RECORD		
S	SPEED, FREQUENCY	SAFETY		SWITCH	
T	TEMPERATURE			TRANSMIT	
U	MULTIVARIABLE		MULITIFUNCTION	MULTIFUNCTION	MULTIFUNCTION
V	VIBRATION, MECHANICAL ANALYSIS			VALVE, DAMPER, LOUVER	
W	WEIGHT, FORCE		WELL		
X	UNCLASSIFIED	X AXIS	UNCLASSIFIED	UNCLASSIFIED	UNCLASSIFIED
Y	EVENT, STATE OR PRESENCE	Y AXIS		RELAY, COMPUTE, CONVERT	
Z	POSITION, DIMENSION	Z AXIS		DRIVER, ACTUATOR UNCLASSIFIED FINAL CONTROL ELEMENT	

NOTE: FOR FURTHER EXPLANATORY NOTES WITH REGARD TO ABOVE TABLE, SEE ANSI/ISA-S5.1 STANDARD ENTITLED "INSTRUMENTATION SYMBOLS AND IDENTIFICATION"

FUNCTION LABELS

Σ SUMMING FUNCTION	$\sqrt[n]{}$ ROOT EXTRACTION FUNCTION	$-K$ REVERSE PROPORTIONAL FUNCTION
Σ/n AVERAGING FUNCTION	X^n EXPONENTIAL FUNCTION	I/P CURRENT TO PNEUMATIC
Δ DIFFERENCE FUNCTION	$f(\omega)$ UNSPECIFIED FUNCTION	P/I PNEUMATIC TO CURRENT
K PROPORTIONAL FUNCTION	$f(t)$ TIME FUNCTION	$+$
$\int$ INTEGRAL FUNCTION	$>$ HIGH SELECTING FUNCTION	$-$ BIAS FUNCTION
d/dt DERIVATIVE FUNCTION	$<$ LOW SELECTING FUNCTION	$\pm$
X MULTIPLYING FUNCTION	$\succ$ HIGH LIMITING FUNCTION	
$\div$ DIVIDING FUNCTION	$\prec$ LOW LIMITING FUNCTION	

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FIG. 4-3

Instrument Type Features

Pneumatic	Electronic
Advantages	
1. Intrinsically safe, no electrical circuits. 2. Compatible with valves. 3. Reliable during power outage for short period of time, dependent on size of air surge vessel.	1. Greater accuracy. 2. More compatible with computers. 3. Fast signal transit time. 4. No signal integrity loss if current loop is used and signal is segregated from A.C. current.
Disadvantages	
1. Subject to air system contaminants. 2. Subject to air leaks. 3. Mechanical parts may fail due to dirt, sand, water, etc. 4. Signal boosters often needed on transmission lines of over 300 feet. 5. Subject to freezing with moisture present. 6. Control speed is limited to velocity of sound.	1. Contacts subject to corrosion. 2. Must be air purged, explosion proof, or intrinsically safe to be used in hazardous areas. 3. Subject to electrical interference. 4. More difficult to provide for positive fail-safe operation. 5. Requires consideration of installation details to minimize points 1, 2, 3, and 4.

FIG. 4-4

Typical Reclosure Gear Operation for Power Outages of Commercial Utilities

Reclosure Attempt	Time	% Successful
First Reclosure	0.1 sec or less	84
Second Reclosure	15-45 sec	10
Third Reclosure	120 sec	1.5
Manual Intervention	–	4.5

Power Supply Specifications

Three manufacturer specifications which should be carefully noted are: (1) regulation, (2) ripple, and (3) short-circuit protection. Regulation is an indication of how well power supply output voltage remains constant as the electrical load is removed and re-connected. Good regulation implies no interaction between connected devices on the same power supply. Ripple is the amount of AC variation on the DC output with a constant load on the power supply. This is especially critical when the outputs of transmitters are connected to analog-to-digital (A/D) converters in a computer or microprocessor based installation. For example, if the A/D precision allows resolution to the nearest 10 millivolts, power supply ripple should be less than 1/3 of this or 3.3 millivolts, unless the noise can be rejected by the converter. Short-circuit protection is a means by which the power supply current is limited at a safe maximum in case the output is accidentally shorted-out at some point. All power supplies should include short circuit protection to prevent serious damage. Power supplies should always have the common side of the output separate from chassis ground to permit the common to be grounded at a single point to a "high quality" instrument ground.

Uninterruptible Power Supplies

Uninterruptible Power Supplies (UPS), often referred to as battery back-up systems, should be sized to span the third reclosure time if power supply levels must be maintained to minimize erratic plant behavior. However, if the instrument air compressor is driven by an electric motor, it usually is not beneficial to maintain battery power past the time the air supply is depleted. All instrument needs required for an orderly shutdown should be considered. Back-up power is most economically provided by "floating" the batteries across the output of the DC power supplies. The circuits should be designed to prevent over-charging or under-charging of the batteries as well as to prevent damage to the regulator circuits when the AC input is disrupted. If the AC input power must be backed up, batteries are used to feed an inverter which transforms DC power into AC power of the proper voltage and frequency. Static switches are available to automatically switch the power supply input from the normal AC line to the inverter when AC line power is lost. Manufacturer recommendations for environmental requirements must be observed to assure reliability of electronic power supplies, static switches, etc. Regular maintenance of battery systems is mandatory since batteries have a limited life-span compared to other electronic components and terminal corrosion may cause problems. Standby generators may be required in some installations to permit instrument operations to continue beyond the time limit of the battery system. Additional information on standby power systems is included in Section 16.

SENSING DEVICES

Some of the more common types of sensing devices for the measurement of process variables are described as follows:

Pressure Sensors

Manometer (Fig. 4-5) — Two different pressures are applied to two separate openings in a transparent vessel containing a liquid. The difference in the heights of the liquid is used as a measure of the differential pressure. This difference should be corrected for temperature and gravity of the liquid in the manometer (usually either water or mercury). Pressures are often expressed in units such as "inches of water" or "millimeters of mercury."

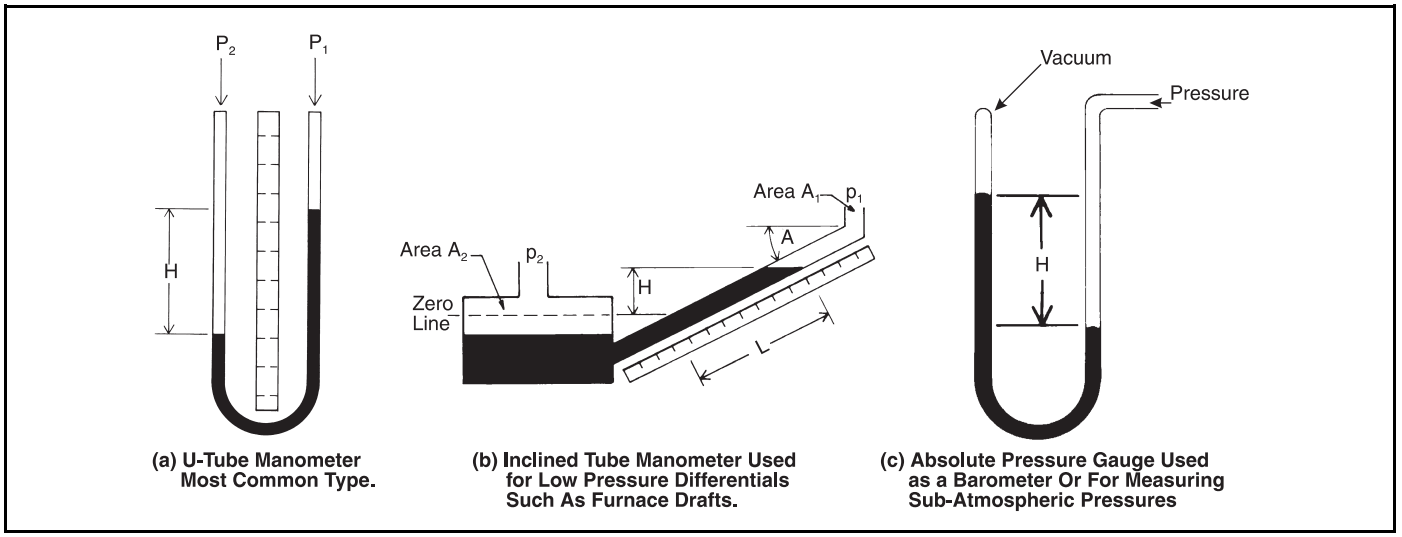
Bourdon tubes (Fig. 4-6) — A Bourdon tube is a metallic coil constructed from a metal tube having the desired elastic quality and corrosion resistance. The tendency of the tube to straighten under pressure causes a mechanical linkage to move a pointer or initiate pneumatic or electronic transmission of the measured pressure. Dampners should be used where pulsation is a problem. Condensate traps should be used upstream of the device in steam service. The pressure indicated is "gauge" pressure which is relative to that of the surroundings. Bourdon gauges are also available as "compound" types which indicate vacuum as well as positive pressure.

Bellows (Fig. 4-7) — A tubular device with pleated sections somewhat like an accordion. It is flexible along its axis and lengthens or shortens according to the applied pressure. The bellows is usually used in low pressure or vacuum service but types are available for use with high pressures (up to several thousand psi). Typical diameters range from 1/2" to 12". They are often used in force-balance type transmitters and other applications where small displacements are required. Like the Bourdon tube, it indicates pressures as "gauge" or relative to its surroundings.

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FIG. 4-5

Types of Manometers



(a) U-Tube Manometer Most Common Type.

(b) Inclined Tube Manometer Used for Low Pressure Differentials Such As Furnace Drafts.

(c) Absolute Pressure Gauge Used as a Barometer Or For Measuring Sub-Atmospheric Pressures

FIG. 4-6

Types of Bourdon Tubes

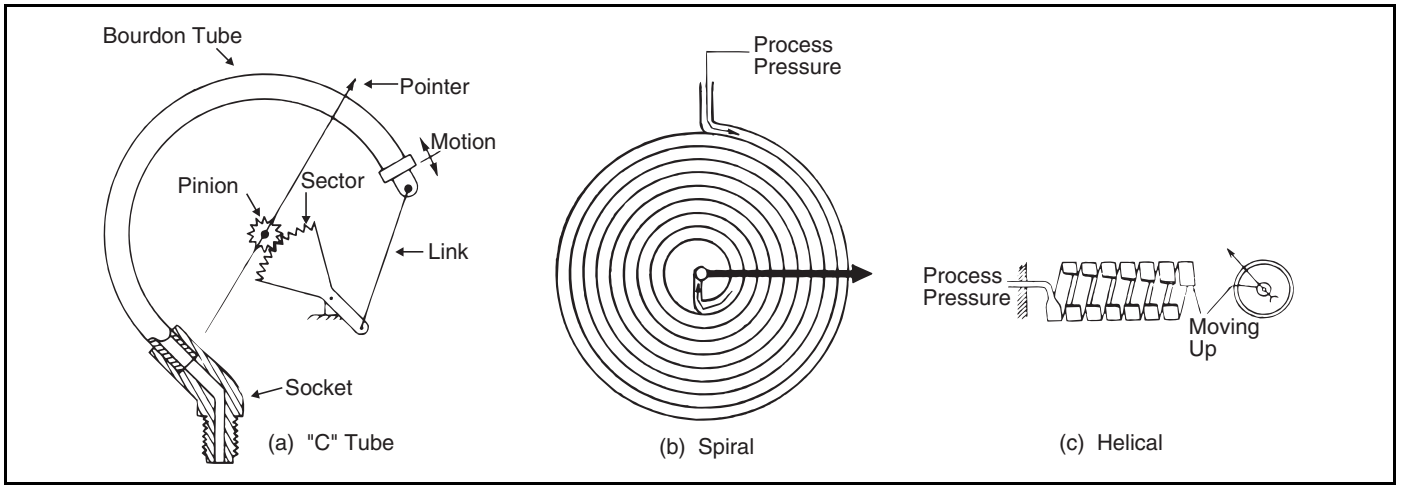
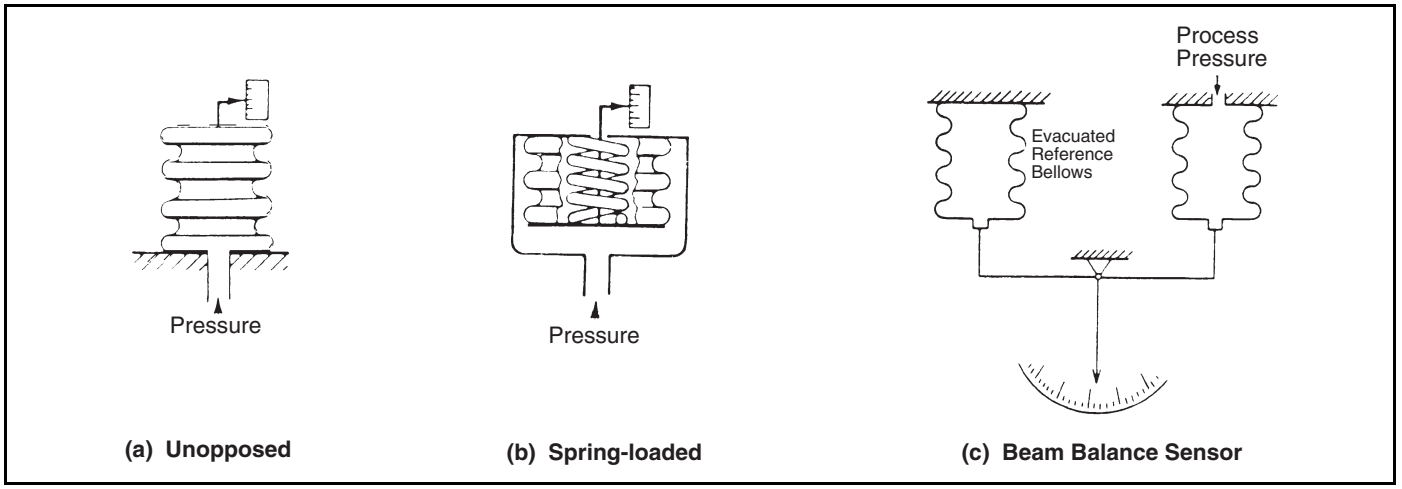


FIG. 4-7

Types of Bellows



(a) Unopposed

(b) Spring-loaded

(c) Beam Balance Sensor

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Diaphragm (Fig. 4-8) — A flat or curved seal with a link attached to an indicator or transmission device. A diaphragm may have its own deflection properties such as with a metallic type or it may be attached to a spring or other elastic member such as with non-metallic diaphragms.

Electrical Pressure Transducers

The primary sensing element of many electrical pressure transducers usually takes the form of a Bourdon tube, bellows, or diaphragm to generate a movement which is transmitted to a strain gauge. A strain gauge is a device using resistance wire connected in a Wheatstone bridge configuration to generate an electrical signal proportional to the movement and hence proportional to the process variable being measured. Other types of electrical pressure transducers use properties of inductance, capacitance, or magnetic coupling to convert a pressure measurement to an electrical signal.

Level Sensors

Gauge glass (Fig. 4-9) — This is the most commonly used visual process-level device. Gauge glasses are generally classified as either transparent or reflex types. A transparent gauge glass consists of either a glass tube or an arrangement of flat glass plates in some type of holder. Since the process fluid level is viewed directly, the transparent gauge glass is normally used with opaque fluids. The reflex type has reflecting prisms to aid in viewing transparent fluids. Caution should be observed when handling and installing these and/or any tempered glass instrument. Scratches or chips can reduce the strength of the glass and cause safety problems.

Chain and tape float gauges (Fig. 4-10) — Used in large, unpressurized storage tanks where the entire full-to-empty range must be measured.

Lever and shaft float gauges (Fig. 4-11) — Used on either unpressurized or pressurized vessels where only a small range of level must be measured. The range of measurement is determined by the length of the float arm, but usually is between a few inches and a few feet.

Displacer level measuring device (Fig. 4-12) — One of the most frequently used level measuring devices is the torque tube displacer. It is attached to the free end of a torque tube which has elastic properties that permit it to twist as the displacer tries to float. This slight turning of the free end of the torque tube is connected to an indicator or transmitter. Torque tube displacement gauges are normally limited to level spans of ten feet.

Head-pressure level gauges (Fig. 4-13) — The true level of a liquid can be determined by dividing the measured hydrostatic head by the density of the liquid. This method requires a knowledge of the densities of all phases of the liquid. Some of these methods are: pressure gauge, bubble tube, and differential pressure measurement. The bubbler (Fig. 4-13a) is used at vacuum and low pressures and is especially good for services such as molten sulfur and dirty liquids. In "boiling-liquid" service (Fig. 4-13b), a condensate trap must be used on the vapor leg. The level of trapped condensate in the vapor leg will usually be different than the vessel liquid level, requiring compensation of the transmitter.

Electrical type level gauges and switches (Fig. 4-14) — Two common types of level gauges are the float-magnetic gauge configuration and the conductive type shown in Fig. 4-14. Slight tension on the tape reel permits the follower magnet to track the float at the liquid level in the device in Fig. 4-14a. The position of the reel represents the level and is either connected to an indicating device or a transmitter. The device shown in Fig. 4-14b illustrates the use of a conductive fluid for high and low level alarm indication.

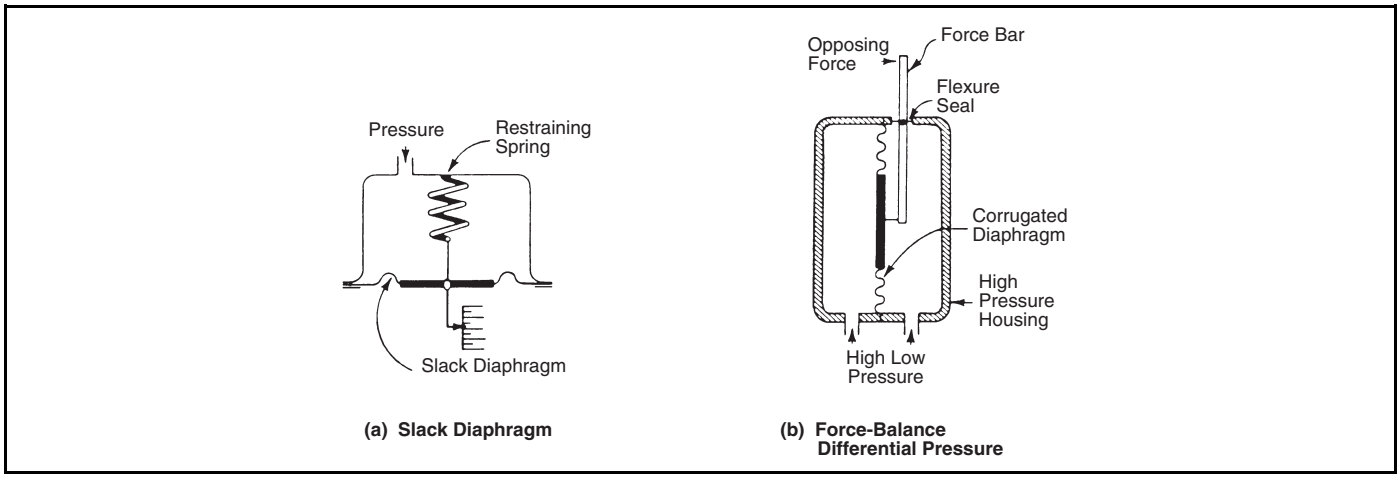
Capacitance probes (not illustrated) — A continuous method of level measurement based on electrical properties. This method uses an electrode placed inside a vessel (or in a protective shell inside the vessel). The capacitance between the electrode and the wall of the vessel or shell varies as the dielectric constant varies. The dielectric in this case is the fluid, hence the capacitance varies in proportion to the liquid level. This capacitance is then measured, and converted to a level measurement to be indicated or transmitted.

Other methods — Some of the other level detection methods use principles of:

1. Absorption of nuclear radiation as in a gamma gauge.
2. Absorption of ultrasonic waves.
3. Viscosity of the liquid.
4. Thermal conductivity and expansion.

FIG. 4-8

Diaphragm Pressure Elements



(a) Slack Diaphragm

(b) Force-Balance Differential Pressure

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FIG. 4-9

Flat Glass Gauge Glasses

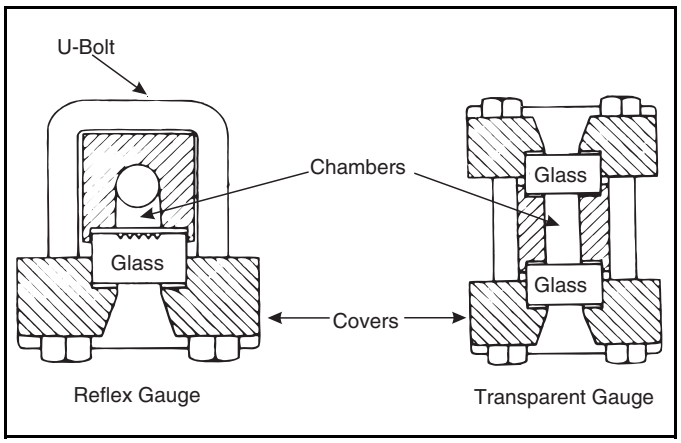
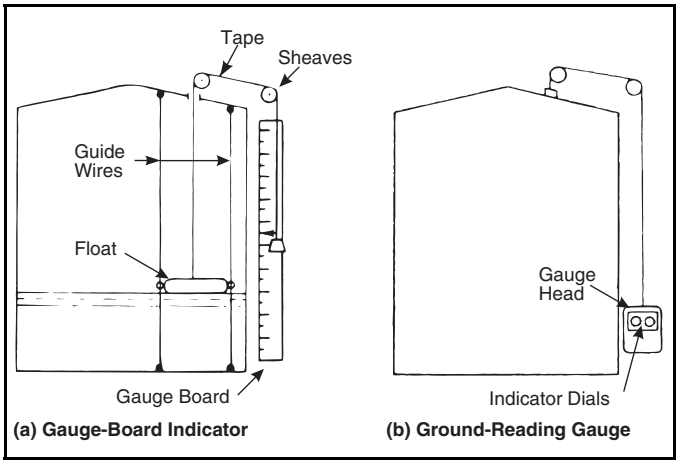


FIG. 4-10

Chain and Tape Float Gauge



Temperature Sensors

Thermocouples — An ordinary thermocouple consists of two different kinds of wires (dissimilar metals) joined together at one end to form the measuring or "hot" junction. Where the free ends are connected to the measuring instrument, a reference or "cold" junction is formed. The millivolt readings measured by the instrument represent the difference in the temperatures of the two junctions and can be converted to temperature by various methods using conversion data from thermocouple tables. The reference temperatures normally used to generate thermocouple tables are 32°F and 70°F. Fig. 4-15 shows some of the common thermocouple types, their usable temperature ranges, and the materials of construction.

Thermocouples used for process measurements are usually protected by a thermowell. The mass of the thermowell should be kept to a minimum in the interest of faster response. The thermocouple must be in thermal contact with the thermowell. This is accomplished by the use of a thermally conductive lubricant or physical contact between the thermocouple and the well. In many measurement and control applications, electrical grounding of the thermocouple at the measurement point must be avoided.

FIG. 4-11

Lever and Shaft Float Gauge

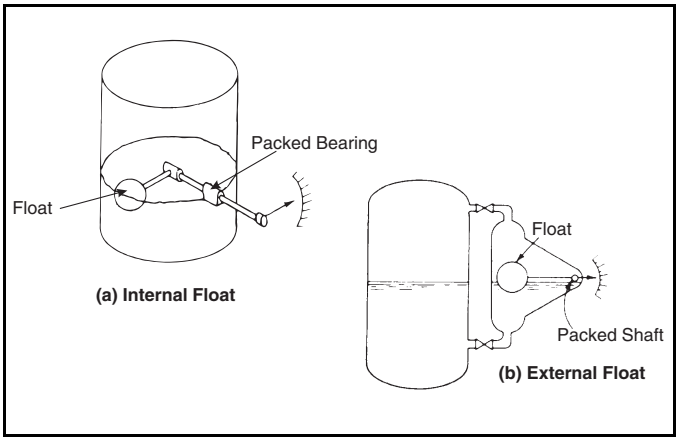


FIG. 4-12

Displacer Level Measuring Device

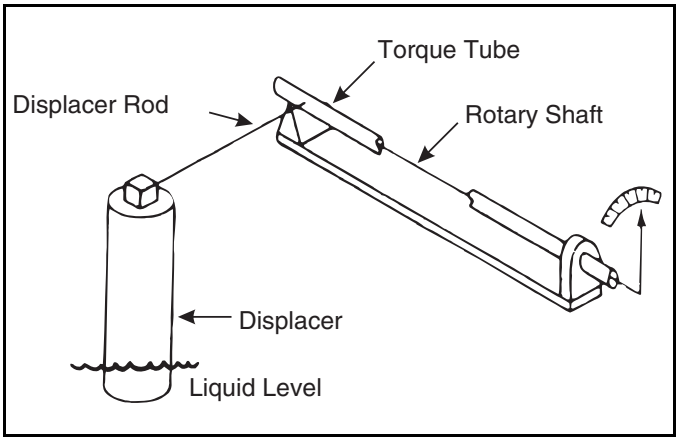
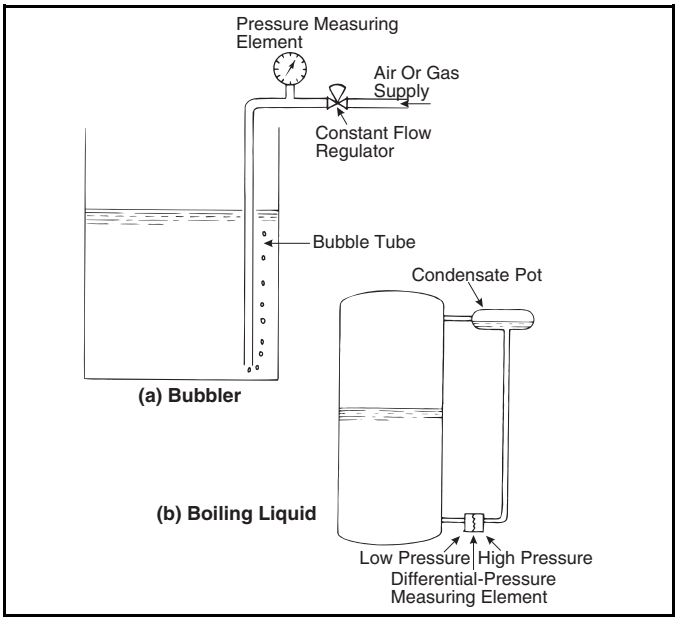


FIG. 4-13

Head Pressure Level Gauges



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In applications other than a laboratory environment, maintaining a reference junction temperature is not practical. Therefore, the reference junction is usually at or a part of the measuring instrument where the reference temperature can be measured and compensated for the measurement circuits. Various series arrangements of thermocouples may be made to obtain differential temperatures or temperature averages.

Qualified personnel may check indicating or recording temperature devices measuring thermocouple potentials using portable equipment compatible with the thermocouple and with compensating circuitry identical to the primary device. The use of incompatible equipment could result in erroneous results, especially in low temperature applications. At low temperatures, extreme care must be taken to eliminate sources of moisture in thermocouple installations. Common properties for different types of thermocouples are given in Fig. 4-15. Conversion tables for converting millivolts to temperatures can be found in NBS Circular #561, or obtained from thermocouple suppliers for common types.

Resistance thermometers — These are often called RTD's for "Resistance Temperature Devices." Since the resistance of metals changes as the temperature changes, a resistance thermometer can be constructed using this principle. The metals that fit a near linear resistance temperature relationship requirement best are platinum, copper, and nickel. An accurate resistance measuring device utilizing a Wheatstone bridge is calibrated in units of temperature rather than resistance. RTD's are used in applications where faster responses and greater accuracies are required than may be obtained with thermocouples. Also RTD's have a fairly high electrical output which is suitable for direct connection to indicators, controllers, recorders, etc. The use of RTD's may also be more economical in some installations since the extension wires may be of copper rather than the more expensive thermocouple extension wire. A reference temperature source is not required for calibration. A special class of resistance thermometer is the thermistor device. It is low in cost, has fast response, and is very stable, but is limited to use at temperatures below 600°F.

Filled-system thermometers — These are simple, reliable, low cost devices. A bulb is attached to a capillary tube which is connected to a measuring element (bellows, Bourdon tube, etc.) in an indicating or transmitting device. The system is filled with a liquid or gas which changes in volume or pressure as the temperature of the bulb changes. The length of the capillary run is normally less than 250 feet.

Glass stem thermometers — These devices are normally used in the office, laboratory, or other non-process areas. Breakage is a problem; accuracy is from 0.1 to 2.0 degrees depending upon the range.

Bimetallic thermometers — The sensing element consists of two metals with different coefficients of expansion bonded together and attached to an indicator. These are inexpensive, but not very accurate and are normally used in on-off temperature thermostats where precise control is not required, or in process applications where relative changes are to be monitored. They should be calibrated at or near the normal operating point of the temperature being monitored.

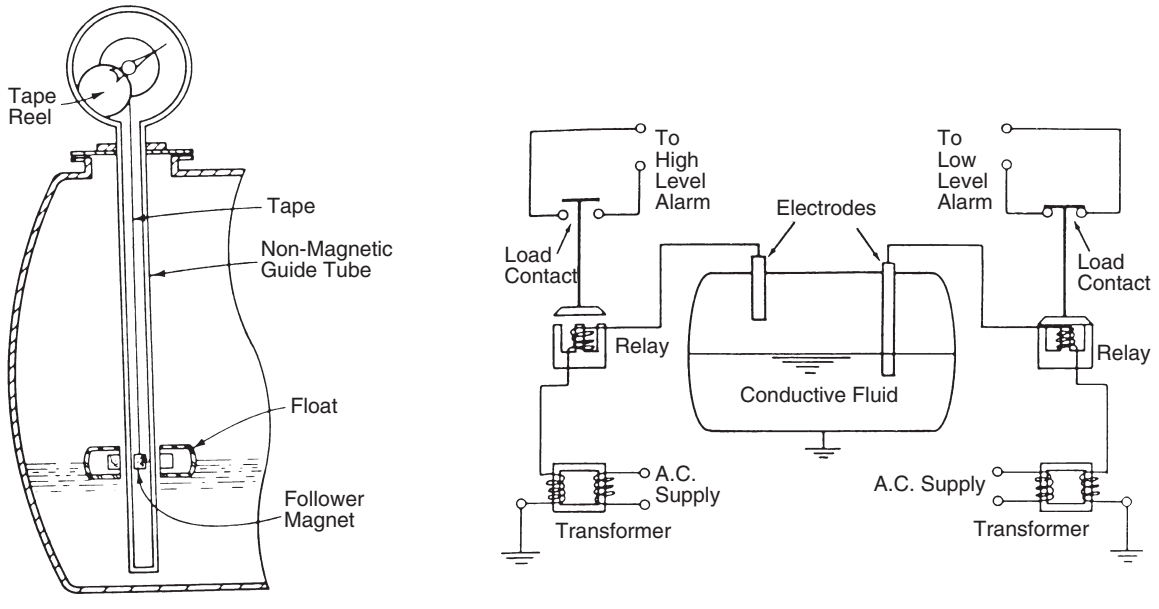
Flow Sensors

Variable head flow meters — Flow meters in this class detect a pressure difference across a flow element specially designed to create that pressure difference. The most common flow element is the orifice plate, but other elements also in use are flow nozzles, venturi tubes, pitot tubes, averaging pitot tubes, target plates, and pipe elbows. Refer to Section 3 for information relating to orifices and orifice plates.

Variable area flow meters (Fig. 4-16) — This type includes the familiar rotameter. The differential pressure across the device is held constant, and the area through which

FIG. 4-14

Electrical Level Gauges/Switches



(a) Magnetic Gauge

(b) Conductivity Level Switch

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FIG. 4-15

Properties of Thermocouples

Thermocouple Wires

Limits of Error of Thermocouples for Standard Wire Sizes

Type	Temperature Range, °F	Limits of Error	
		Standard	Special
Chromel/Alumel (K)	32 to + 530 + 530 to + 2300	± 4°F ± ¾%	± 2°F ± ⅜%
Iron/Constantan (J)	32 to +530 +530 to +1400	± 4°F ± ¾%	± 2°F ± ⅜%
Copper/Constantan (T)	–300 to –75 –150 to –75 –75 to + 200 +200 to + 700	— ± 2% ± 1½% ± ¾%	± 1% ± 1% ± ¾% ± ⅜%
Platinum/Platinum - 10% Rhodium (S) Platinum/Platinum - 13% Rhodium (R)	32 to +1000 +1000 to +2700	± 5°F ± ½%	— —
Chromel/Constantan (E)	32 to +600 600 to ±1600	± 3°F ± ½%	— —

the fluid passes changes due to the movement of the float up and down the tapered tube. These are usually limited to use with relatively small flows where visual indication is sufficient.

Turbine meters — These use a small permanent magnet mounted on the meter tube to create a magnetic field. A small turbine is mounted inside the tube and turns with a speed proportional to the flow rate. As each vane of the turbine passes through the magnetic field the magnetic flux is disturbed which induces a pulse in a pickup coil mounted on the outside of the meter. The pulse rate is proportional to the flow rate. Pulses are then counted and converted to standard flow units.

Positive displacement meters — Positive displacement meters and metering pumps measure discrete quantities of the flowing fluid. The rotating element is mechanically coupled to a transmitter or counter which integrates or totals the counts to provide an indication in units of gallons, liters, cubic feet, etc. Some common types are: rotating vane, bi-rotor, rotating paddle, oscillating piston, and oval gear meters. They are used for custody transfer devices such as gas meters or gasoline pumps.

Electromagnetic Flowmeter

If an electrical conductor is moved in a magnetic field, an electrical voltage is introduced in the conductor which is perpendicular to both the direction of motion and the magnet field and whose magnitude is proportional to the magnetic field strength and the velocity of the movement. The characterization of the laws of induction also applies to the movement of a conductive fluid in a pipe through a magnetic field and is the basis for the electrostatic flowmeter.

Ultrasonic Flow Meters

Ultrasonic flow measurement is based on sending and receiving acoustic signals through the flow. The difference in transit time between transducers, built in at opposite sides of the pipe gives signals that can be transferred to flow.

A sound wave travels faster with the flow than one propagated against the flow. The difference in transit times is proportional to the medium's mean flow velocity.

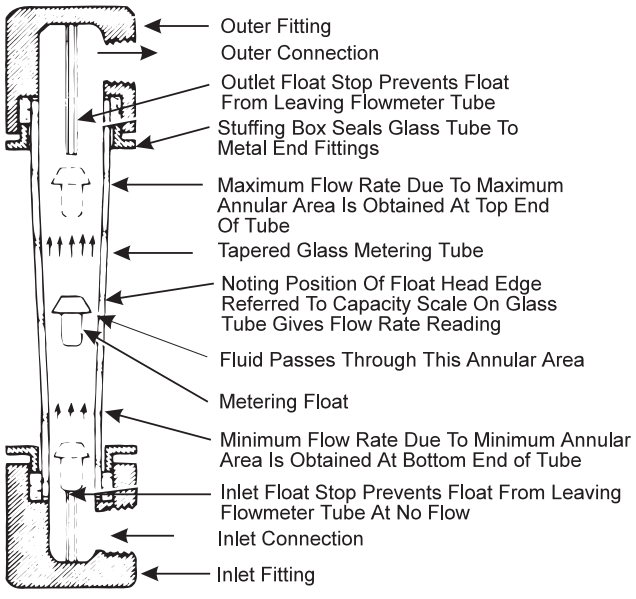
By installing more than one pair of transducers, a larger range of the flow profiles across the metering section can be covered and thereby increase the accuracy of the meter.

Other flowmeters — Some other flowmeter types occasionally encountered are:

- Doppler effect or ultrasonic flowmeters
- Vortex shedding flowmeters
- Laser velocimeters

FIG. 4-16

Rotameter



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- Thermal meters
- Nuclear Magnetic Resonance meters
- Gas ionization meters
- Cross-correlation devices
- Mass flowmeters

All flow meters should be calibrated using the fluid being measured, or, if a different fluid is used for calibration, the properties of the calibrating fluid must be related to the fluid of measurement.

SIGNAL TRANSMITTERS

Pneumatic Transmitters

A pneumatic transmitter is a device that senses some process variable and translates the measured value into an air pressure which is transmitted to various receiver devices for indication, recording, alarm, and control. The signal range of 3-15 psig is the accepted industry standard; however, other ranges may be encountered. This signal is proportional to the range of measurement of the process variable. For example, 3-15 psig can represent 0-100 psi, 500-1000 gpm, –50 to +50°F, etc.

The prime function of a transmitter is to reproduce the low-energy measurement signal with sufficient energy that it may be transmitted over an appreciable distance or used as a power source to a control device. The low-energy measurement signal is that position or movement associated with the action of the process variable on the sensing element (bellows, diaphragm, Bourdon tube, etc.). Pneumatic transmitters operate in a manner similar to proportional controllers.

Electronic Transmitters

Electronic transmitters perform the same function as pneumatic transmitters: a low energy process-related signal is converted into a higher energy signal suitable to connect to other instruments in the system. The output signal of most electronic transmitters is a 4-20 mA, 10-50 mA, or 1-5 Vdc signal. Other ranges often encountered are: 0-10 Vdc, 2-10 Vdc, and 0.25-1.25 Vdc. Electronic transmitters are also classified as force balance or motion balance types.

Connection Methods — Of great concern to the instrument engineer is the method by which electronic transmitters are connected in the instrumentation system. The "two-wire," "three-wire," and "four-wire" classifications are often used to describe the method of connection.

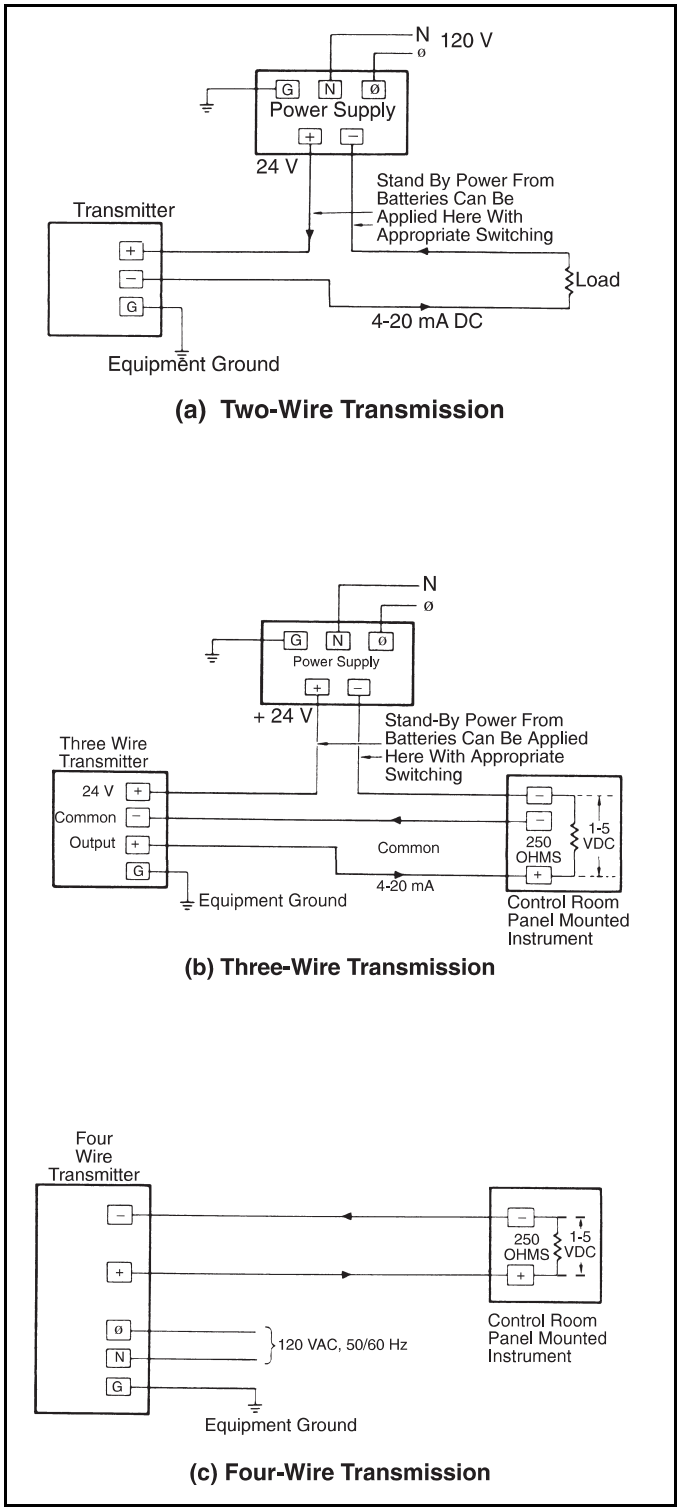
Two-wire transmitters (Fig. 4-17a) — These are the simplest and most economical and should be used wherever load conditions will permit. In a two-wire system the only source of power to the transmitter is from the signal loop. Referring to Fig. 4-18a, the 4 mA "zero-end" current is sufficient to drive the internal circuitry of the transmitter and the current from 4 to 20 mAs represents the range of the measured process variable. The power supply and the instruments are usually mounted in the control room.

Three-wire transmitters (Fig. 4-17b) — Some transmitters require more power than the signal loop (4-20 ma, etc.) can supply to support their internal circuitry. A DC common wire is run from the instrument to the transmitter. This permits the transmitter to draw whatever power it needs from the power supply and produce the desired signal current at the transmitter output.

Four-wire transmitters (Fig. 4-17c) — Some transmitters have their own internal power supply and require no connection to the DC power supply. A 120 vac source is connected directly to the transmitter and its output signal loop is connected only to the receiving instrument. These are often used where an instrument is "added on" to an existing instru-

FIG. 4-17

Connection Methods



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mentation installation to avoid adding to the load of the DC supplies. The disadvantage is the need for AC power at the instrument site.

Signal Converters

Signal converters are used either to achieve compatibility between different types of instruments or for isolation purposes. Some common forms of signal converters are:

Pneumatic-to-electronic (P/I) — These are electronic pressure transmitters designed for 3-15 psig input range and the desired output range (4-20 ma, etc.).

Electronic-to-pneumatic (I/P) — I/P converters are pneumatic transmitters with an electro-magnetic device connected to a nozzle-baffle arrangement which generates a pneumatic output signal which is proportional to the input signal.

Isolators — These are usually electronic current-to-current or voltage-to-voltage converters which provide electrical isolation to eliminate unwanted ground loop currents or common mode voltages.

Electric signal converters — These fit the same category as I/Ps and P/Is in that they change the signal from one range to another. Examples are 4-20 mA to 0-10 vdc, 1-5 Vdc to 10-50 mA, etc.

Frequency converters — Frequency to DC converters typically receive pulse inputs from turbines or positive displacement flowmeters and provide a proportional 4-20 ma, 10-50 mA or voltage output. Voltage output converters are often referred to as F/V (frequency-to-voltage) converters or transmitters. V/F (voltage-to-frequency) converters are often used to interface standard "current-loop" type instrumentation to control devices requiring frequency or pulse-train setpoint inputs. These are commonly used in speed indicators for high speed centrifugal equipment.

RECORDERS AND INDICATORS

Recorders

A recorder is a device used to plot the value of one or more measured variables, generally against time, but in some cases against another associated variable or variables. Recorders are often classified in the following ways:

1. According to use, i.e., whether the recorder is an integral part of the measuring/controlling system or is a general purpose type such as would be used in a laboratory or with a chromatograph.
2. According to method used to drive the pen(s). This refers to whether the pen is directly connected to the sensing element or to some type of pen positioning mechanism activated by the measuring signal.
3. According to chart type. This primarily refers to whether the recorder is of the circular or strip chart type and whether the time-axis drive is powered by a mechanical spring, electrical motor, or pneumatic drive.
4. Analog or Digital. Analog recorders are the more familiar strip chart and circular types. Digital recorders include such things as strip printers, data loggers, electronic totalizers, and computer-related devices such as data terminals and printers.

Indicators

An indicator is any device which presents a visual display of a measured quantity such as temperature, pressure, humidity, voltage, etc. Indicators are included in an instrumentation system either as independent devices (denoted as TI, PI, FI, etc.) or as a part of a controlling device (TIC, PIC, etc.). Indicators may be classified in the following groups:

Mechanical type — In these indicators the measured quantity causes the movement of a pointer along a graduated scale. This movement is due to the action of the measured quantity on a diaphragm, bellows, electromagnetic coil, or other sensing device which is mechanically linked to the pointer. This includes pressure gauges, filled tube dial thermometers, voltage and current meters, level gauges, etc.

Electronic analog type — These are analog indicators with no moving parts. A signal from the sensing device activates an optical display attached to graduated scale. A common type uses a bank of 200 tiny gas filled tubes which are illuminated additively in proportion to the magnitude of the process signal. This results in resolution of 0.5% of full scale.

Digital type — Digital indicators include an analog-to-digital converter which changes the electrical process signal to binary format which is then displayed in numerical form. Typical displays consist of light emitting diodes (LED's), liquid crystal displays (LCD's), gas filled tubes, etc.

CONTROL CONCEPTS

Control Loops

A control circuit is commonly referred to as a "loop." A control loop may be classified as either "open" or "closed" depending upon whether the control adjustments are manual settings (open loop) or automatically determined by some type of feedback controller (closed-loop).

Open loop (Fig. 4-18a) — In an open-loop control system, an operator makes a manual adjustment to a device (valve) which controls the flow of a manipulated variable (steam) to attempt to achieve some set-point (desired temperature) value of a controlled variable (hot water). However, this adjustment is only valid for the conditions under which the operator made the adjustment. Any disturbance such as a change in inlet water temperature, steam temperature, heat loss to the surroundings, or throughput will cause the outlet temperature to change.

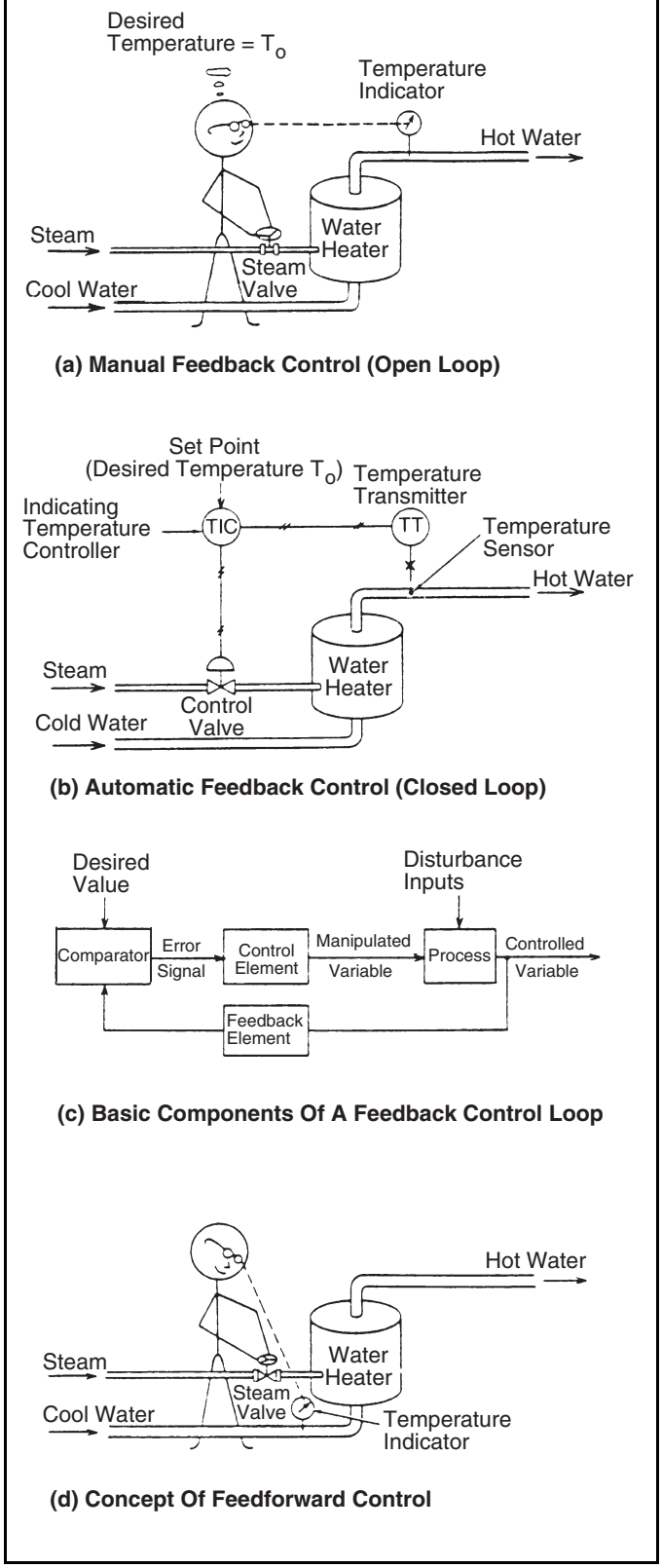
Closed loop (Fig. 4-18b) — If appropriate measuring and controlling elements are added to the system, the loop is closed by the inclusion of an automatic feedback controller. The controller detects any difference between the set-point and measurement signals (error signal) and produces an output signal to drive the valve in the proper direction to adjust the heat input to cause the measurement to reach the set-point value.

Feedback control (Fig. 4-18c) — The basic components of a feedback control loop are shown in block diagram form in the figure. The "comparator" actually represents the entire controller and any associated signal converters. The "control element" is the valve, the "feedback element" is the transmitter, and the "process" is the mixing of the steam and cold water inside the water heater.

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FIG. 4-18

Control Concepts



Feedforward control (Fig. 4-18d) — Feed forward control (often called "Predictive Control") is actually a form of open-loop control. An input variable (cold water temperature) is monitored and the manipulated variable (steam flow) is adjusted accordingly to compensate for changes in the input variable. Feedforward control is almost always used in conjunction with feedback control to overcome the effects of some expected disturbance.

CONTROL MODES AND CONTROLLERS

Basic forms of control action or "modes" used in most process control are: two-position or "on-off" control, proportional control, integral or "reset" control, and derivative or "rate" control. The latter three modes are often used in various combinations with each other.

Two-Position (on-off) Controllers

The simplest form of control action is "on-off" control, in which the controller output either energizes or de-energizes some two-state device such as a relay or an open-shut type valve. The two-position controller is used extensively in home heating and cooling systems, refrigerators, hot water tanks, air compressors, and other applications where the cost of more precise control is not justified. Most two-position controllers are reverse-acting, i.e., when the measured variable is above the set-point, the controller turns the manipulated variable OFF, and when the measured variable is below the setpoint, the controller turns the manipulated variable ON. A "deadband" or differential gap exists around the zero error condition to minimize cycling. This is often implemented as a pair of control points: one where the controller will "kick-on" and the other where the controller will "kick-off" as opposed to a single setpoint.

Proportional, Integral, and Derivative Control Modes

When the cycling nature of "on-off" control cannot be tolerated, a controller using some combination of the proportional, integral, and derivative modes is normally used. The normally used combinations of these modes are: (P) Proportional only; (PI) Proportional plus integral; (PD) Proportional plus derivative; and (PID) Proportional plus integral plus derivative. Integral mode is rarely used alone since it adds lag to the system without the benefits of proportional mode. Derivative mode is never used alone since its output is zero except when the error signal (PV – SP) is changing. Fig. 4-19 shows the response of each mode to illustrated error signal.

Direct and Reverse Acting Controllers

The error signal term (PV – SP) indicates "reverse" controller action (the controller output decreases when the measurement increases). Most controllers may be switched from one type of action to the other, and the choice depends upon the action of the final control element and the direction that the measured controlled variable responds to the manipulated variable.

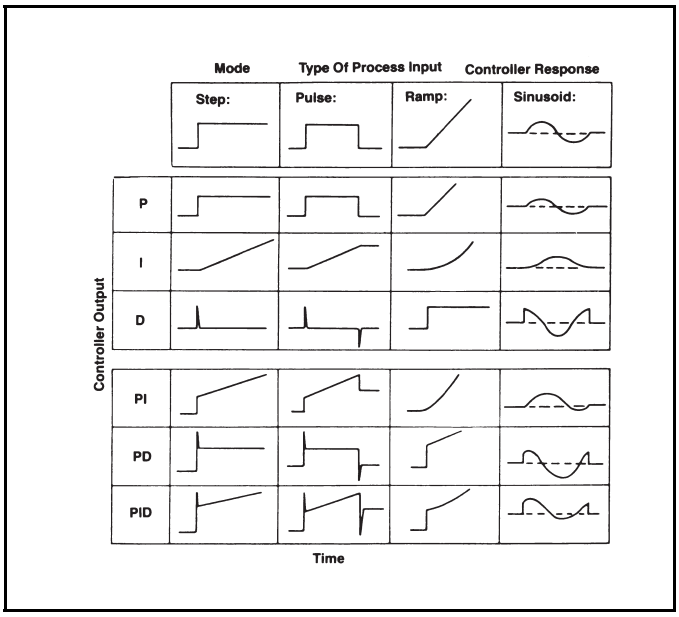
Proportional Mode (P)

The "proportional-only" mode is the simplest of the three modes of control. It is characterized by a continuous linear relationship between the error signal (PV – SP) and the controller output.

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FIG. 4-19

Responses of Proportional, Integral, and Derivative Control Modes to Various Process Inputs and Disturbances



The proportional controller is simple, inexpensive, and does not introduce any additional time lags into the control system. Its chief disadvantage is its inability to cope with load changes.

In the expression:

$$CO = K_p (PV - SP) + MO \qquad \textbf{Eq 4-1}$$

the term MO represents the controller output while on "manual" (just before the controller is switched to "automatic"). Thus MO is the controller output required to maintain the process variable at the setpoint value with the present load on the system. The terms PV and SP in the equation are always in the same units, but not in the same units as CO and MO. The gain constant K_p can include a conversion factor to make the units compatible.

Proportional gain is often expressed in terms of Proportional Band, where:

$$\text{Proportional Band (PB \%)} = 100/\text{Gain} \qquad \textbf{Eq 4-2}$$

The proportional band represents the percentage of full range change of the error signal (PV – SP) that will cause full range change in the output of the controller.

Offset

"Offset" is the deviation of the controlled variable from the setpoint expressed in process units. Offset will occur whenever process load conditions have changed or a setpoint change has been made such that the original manual output, MO, of the controller is not sufficient to maintain the controlled process variable at the setpoint, SP. Offset may be minimized by using high controller gains (narrow proportional band); however, high gain may be impractical or may cause instability in some processes. Some controllers are equipped with a manual reset control which can be used to "zero-out" the offset while in the automatic mode. Controllers not so equipped are often switched back to "manual," adjusted for zero offset, then switched back to "automatic." Since this manual resetting would need to be done after every upset, integral mode is often added to give automatic reset action.

Proportional Plus Integral Mode (PI)

The "PI controller" is the most widely used of the various controller types (70-90% of the controllers in a typical plant). Combining the expressions for proportional and integral modes gives the expression for the PI controller:

$$CO = K_p (PV - SP) + K_i \int (PV - SP)\, dt + MO \qquad \textbf{Eq 4-3}$$

Because of the relationship between the controller gain and the process gain, the integral mode gain K_i can be expressed as K_p/T_i, where T_i is the integral mode time constant. This gives the more familiar form of the PI controller equation:

$$CO = K_p [(PV - SP) + 1/T_i \int (PV - SP)\, dt] + MO \qquad \textbf{Eq 4-4}$$

The integral term will cause the valve (or other final control element) to move at a speed proportional to the amount of the offset until all offset is removed. The integral time constant T_i is also known as reset time, and since it represents the time for the integral action to "repeat" the amount of the proportional action, the reciprocal of T_i (called "reset rate") is often given in units of "repeats per minute."

Proportional Plus Derivative Mode (PD)

Feedback control has a deficiency in that no corrective action is taken until some error signal exists. Problems are often encountered in the control of processes with large time constants and/or dead times. In controlling these slow processes, corrective action needs to be taken as soon as possible or the time to recover will be too long. Derivative action adds to the controller output based on the rate of change of the error signal (PV – SP). This is also called "rate" or "preact" control action, and control using derivative (rate) action is known as "anticipatory" control. Combining the proportional and derivative terms gives the expression for PD control action:

$$CO = K_p (PV - SP) + K_d [d(PV - SP)/dt] + MO \qquad \textbf{Eq 4-5}$$

The derivative gain constant K_d may be expressed as $K_d = K_p T_d$. The derivative time constant T_d represents the time required for the contribution from the derivative term to equal the contribution from the proportional term with a steadily changing or "ramp" error signal. Derivative is, by its nature, sensitive to signal noise and can cause excessive wear on the control valve. Sufficient signal filtering should be done to reduce the noise when derivative action is used in the controller.

Proportional Plus Integral Plus Derivative Mode (PID)

Many controllers include circuitry for proportional, integral, and derivative modes which permits the user to select the desired control action. Combining terms as before gives the expression for PID control action:

$$CO = K_p (PV - SP) + K_i \int (PV - SP)\, dt + K_d [d(PV - SP)/dt] + MO \qquad \textbf{Eq 4-6}$$

As with the PI controller, the integral gain may be expressed as $K_p/(T_i)$ and the derivative gain may be expressed as $K_p/(T_d)$. This gives the more familiar form:

$$CO = K_p [(PV - SP) + 1/T_i \int (PV - SP)\, dt + T_d [d(PV - SP)/dt]] + MO \qquad \textbf{Eq 4-7}$$

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An alternate form used in many digital computer systems uses a normalization constant which "lumps" the units conversion factors in K_n:

$$CO = K_n [Kp(PV - SP) - 1/T_i \int (PV - SP)\, dt + T_d [d(PV - SP)/dt]] + MO \quad \textbf{Eq 4-8}$$

Fig. 4-20 compares the advantages and disadvantages of the various control modes for most desirable control.

FIG. 4-20

Control Mode Comparisons

Mode	Advantages	Disadvantages
On-Off	Simple, inexpensive	Constant cycling
Proportional	Does not add lag	Almost always has offset
Integral	Eliminates offset	Adds time lag to system
Derivative	Speeds up response	Responds to noise

Controller Tuning

Controller tuning has been referred to as the most important, least understood, and most poorly practiced aspect of process control. It is not an exact science and considerable "trial and error" must be used to achieve an acceptable combination of the tuning parameters for a particular process. "Good" control is a matter of definition and depends upon such factors as individual preference, process disturbances and interactions, product specifications, etc. Fig. 4-21a shows typical controller responses for different damping factors (ξ) to a step-change in load or setpoint. The damping factor may be defined as the ratio of actual damping to critical damping. The reader is referred to Perry's *Chemical Engineers' Handbook*, Section 22 for a more complete definition of damping factors. The "over-damped" responses ($\{\xi > 1\}$) cause no overshoot. The "under-damped" responses ($\xi < 1$) cause considerable oscillation about the setpoint, but the controlled variable eventually "lines-out" on the setpoint. The "critically-damped" response ($\xi = 1$) would seem ideal but often the time required to reach the setpoint is excessive. The quarter-decay response allows overshoot but each "swing" above and below the set-point is one-fourth that of the previous swing. This is the response that is usually considered acceptable for most processes. See Fig. 4-21b for an example of quarter-decay response.

Techniques for tuning controllers generally fall into one of two classes. First, there are some methods based on parameters determined from the closed-loop response of the system. Second, some methods are based on parameters determined from the open-loop response curve commonly known as the process reaction curve. The popular Ziegler-Nichols method falls into the second category.

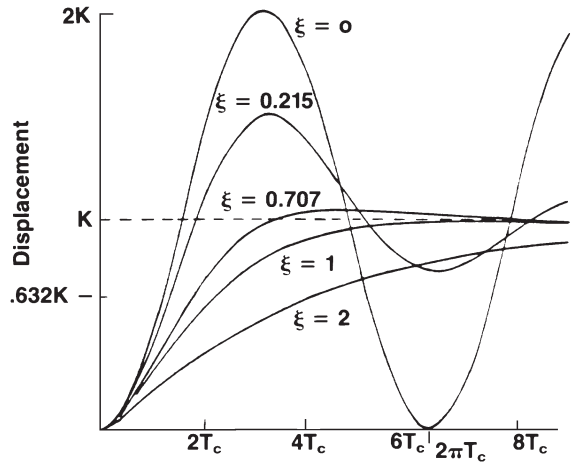
Ziegler-Nichols Method[1]

This method was first reported by Ziegler and Nichols in 1942 and is also known as the "ultimate" method. The method determines the ultimate or maximum controller gain (proportional action only) for which the system is stable. The ultimate gain and ultimate period (time for one cycle at ultimate gain) are then used to calculate initial controller settings for quarter-decay response. The following steps may be used to determine ultimate gain and period:

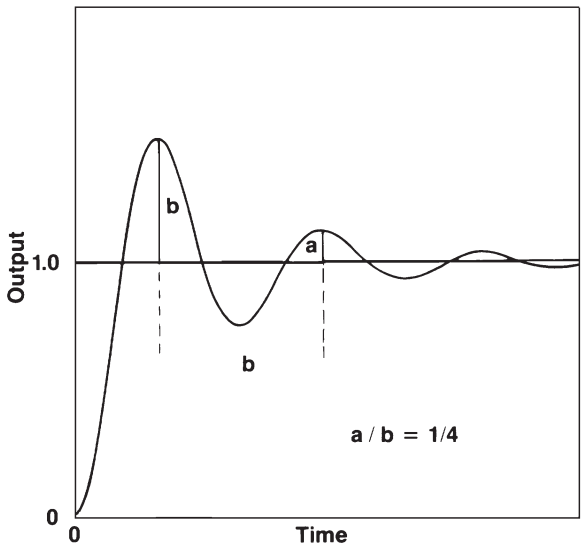
1. Switch the controller to automatic.

FIG. 4-21

Typical Response Curves



(a) Response Of Oscillatory Or Second-Order System For Various Values Of Damping Factor, ξ



(b) Response Curve For 1/4 Decay Ratio

2. Tune out all integral (reset) and derivative (rate) action. (set T_i = max., set T_d = zero)
3. Set the proportional gain K_p at some arbitrary value, impose an upset on the process, and observe the response. (Moving the setpoint briefly and returning will create an upset.)
4. If the response is stable (curve C in Fig. 4-22), increase the gain and repeat the upset.
5. If the response is unstable (curve A in Fig. 4-22), decrease the gain and repeat the upset.
6. When the response exhibits sustained cycling (curve B in Fig. 4-22), note the controller gain setting and the ultimate period (P_u).

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FIG. 4-22

Typical Responses Obtained When Determining Ultimate Gain and Ultimate Period

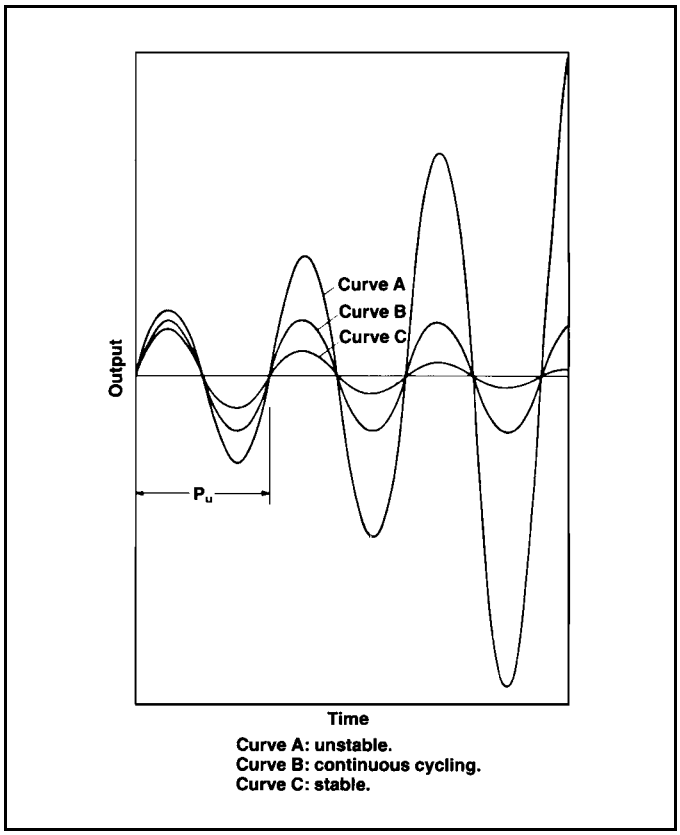


Fig. 4-23 gives relative controller gain, integral time, and derivative time for the various control mode combinations for quarter-decay response as related to ultimate controller gain setting, K_u, and ultimate period P_u. Gain settings are also shown in units of proportional band, PB.

Fig. 4-24 shows some typical settings for various types of process controllers.

Example 4-2 — An example using the Ziegler-Nichols method is given below:

For a certain temperature control system, the ultimate sensitivity K_u was found to be 0.4 psi per deg F, and the ultimate period P_u was found to be two minutes. A three mode PID controller is required.

Using Fig. 4-24:

Proportional gain K_p:

$$K_p = 0.6\ K_u = 0.6\ (0.4\ psi/°F) = 0.24\ psi/°F$$

Integral time constant T_i:

$$T_i = P_u/2,\ T_i = 2/2 = 1.0\ minute$$

Derivative time constant T_d:

$$T_d = P_u/8,\ T_d = 2/8 = 0.25\ minutes$$

Control Mode Considerations

The process control engineer has the responsibility for matching the many and variable characteristics of the process to be controlled with the most effective control hardware available.[6] Fig. 4-25 provides guidelines for choosing the mode of control for various types of applications based upon the process reaction rate and size and speed of load changes.

Special considerations should be made in applying a "split-range" controller. A common example is a column temperature controller on a cryogenic demethanizer. In this system the first half (0-50%) of the controller output actuates the "free" heat exchange with the incoming feed, and the second half (50-100%) of the controller output actuates the supplemental heat from the hot oil system. Adaptive gain control may be required since the heating value of the hot oil is much greater than that of the gas used in the heat exchange.

FIG. 4-23

Ziegler-Nichols Settings for 1/4 Decay Response[1]

Mode	K_p	or	PB(%)	T_i	T_d
(P)	0.5 K_u		2(PB_u)	max.	zero
(PI)	0.45 K_u		2.2(PB_u)	P_u/1.2	zero
(PD)	0.6 K_u		1.65(PB_u)	max.	P_u/8.0
(PID)	0.6 K_u		1.65(PB_u)	P_u/2.0	P_u/8.0

FIG. 4-24

Typical Controller Settings

Process	Gain	PB(%)	Integral		Derivative
			T_i (sec)	min/repeat	T_d (sec)
Flow	0.6-0.8	167-125	3.0-1.8	0.05-0.03	0.0
Pressure	5.0	20.0	120-60	2.0-1.0	0.0
Temp.	1.0-2.0	100-50	120-30	2.0-0.5	6.0-12

EMBEDDED ADVANCED CONTROL

Embedded advanced control will usually give an improved plant performance over that achievable with traditional techniques. By introducing Embedded Advanced Control, a high level of reliability and security is provided to maximize control system uptime. Since embedded advanced control tools have direct access to controller I/O, they may access process measurements and actuators with no communication jitter or delay. This allows use of these tools on the fastest processes.

CONTROL VALVES

Selecting the proper control valve for each application involves many factors. The valve body design, actuator style, and plug characteristic are critical items for selection. Proper valve sizing is necessary for accurate, efficient, economical process control. In areas where personnel will be affected, noise prediction and control becomes a significant factor.

Engineering application guidelines, nomographs, and equations presented in the following pages may be used to determine the correct control valve configuration, size and flow characteristics, and to predict noise levels for most applications. The material presented here may also be used to evaluate the performance of valves installed in existing plants.

The equations given in this section are used to calculate the flow coefficient (C_v or C_g) required for a valve to pass the re-

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FIG. 4-25

Control Mode vs. Application

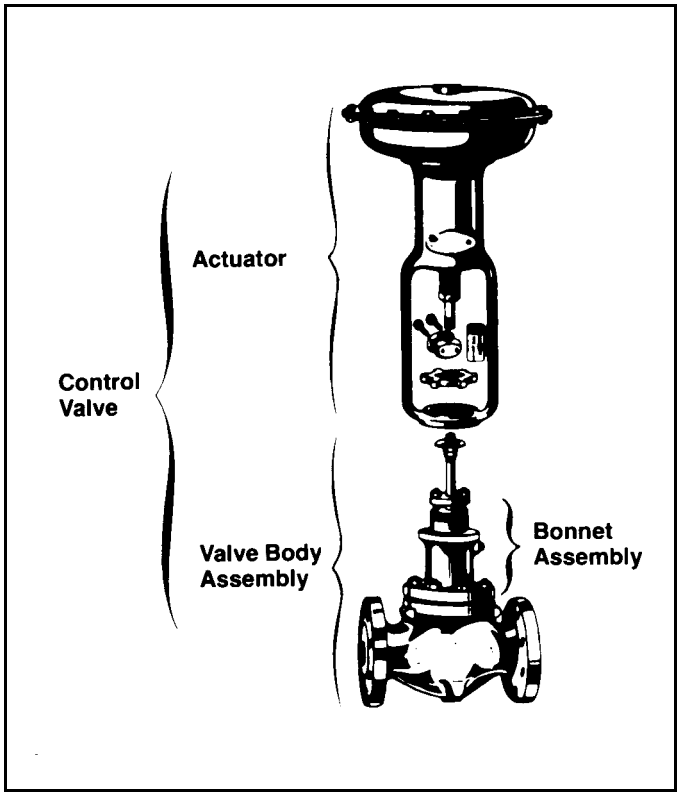
Control Mode	Process Reaction Rate	Load Changes		Applications
		Size	Speed	
On-off: Two-position with differential gap	Slow	Any	Any	Large-capacity temperature and level installations. Storage tanks, hot-water supply tanks, room heating, compressor suction scrubbers.
Floating, Single-speed with adjustable neutral zone	Fast	Any	Small	Processes with small dead time. Industrial furnaces and air conditioning.
Proportional	Slow to Moderate	Small	Moderate	Pressure, temperature, and level where offset is not objectionable. Kettle reboiler level, drying-oven temperature, pressure-reducing stations.
Proportional-plus-derivative (rate)	Moderate	Small	Any	Where increased stability with minimum offset and lack of reset wind-up is required. Compressor discharge pressure.
Proportional-plus-integral (reset)	Any	Large	Slow to Moderate	Most applications, including flow. Not suitable for batch operations unless overpeaking is allowed.
Proportional-plus-integral-plus-derivative	Any	Large	Fast	Batch control; processes with sudden upsets; temperature control.

quired flow. Most valve manufacturers publish flow coefficients for each valve style and size.

A brief description of the two major components of a control valve, the valve body and the actuator, is presented in Fig. 4-26.

FIG. 4-26

Relationship of Major Components

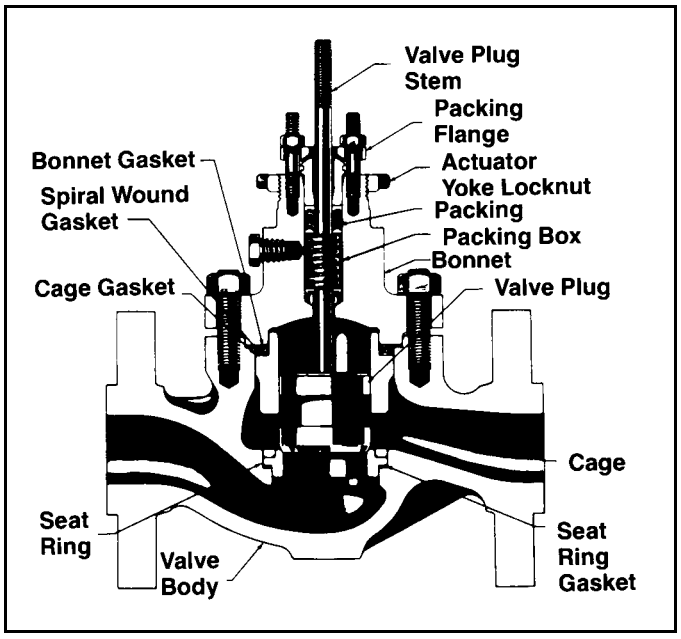


Control-Valve Bodies

The control-valve body (see Fig. 4-27) regulates the rate of fluid flow as the position of the valve plug is changed by force from the actuator. Therefore, the valve body must permit actuator thrust transmission, resist chemical and physical effects of the process, and provide the appropriate end connections to mate with the adjacent piping. It must do all of this without external leakage. Most valve body designs are of the globe style, but other configurations such as ball and butterfly styles are available. Final selection depends upon detailed review of the engineering application.

FIG. 4-27

Push-Down-to-Close Valve Body Assembly



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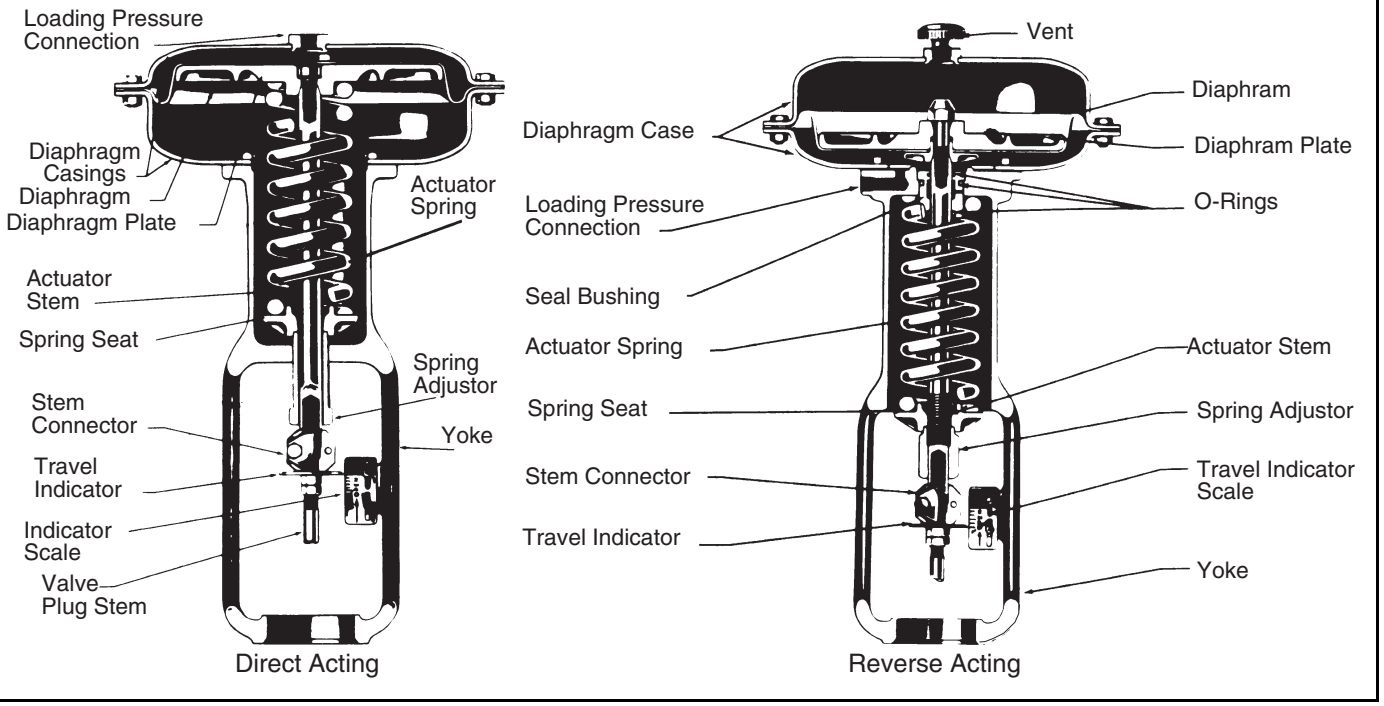
Control-Valve Actuators

Pneumatically operated control-valve actuators are the most popular type in use, but electric, hydraulic, and manual actuators are also widely used. The spring-and-diaphragm pneumatic actuator (see Fig. 4-28) is commonly specified, due to its dependability and its simplicity of design. Pneumatically operated piston actuators provide integral positioner capability and high stem-force output for demanding service conditions, such as high differential pressure or long valve stem travel distance.

The linear flow-characteristic curve shows that the flow rate is directly proportional to the valve travel. This proportional relationship produces a characteristic with a constant slope so that with constant pressure drop (ΔP), the valve gain will be the same at all flows. (Valve gain is the ratio of an incremental change in flow rate to an incremental change in valve plug position. Gain is a function of valve size and configuration, system operating conditions, and valve plug characteristic.) The linear-valve plug is commonly specified for liquid level control and for certain flow control applications requiring constant gain.

FIG. 4-28

Typical Spring-and-Diaphragm Actuator Assemblies



Discussion of Flow Characteristics and Valve Selection

The flow characteristic of a control valve is the relationship between the flow rate through the valve and the valve travel as the travel is varied from 0 to 100%. "Inherent flow characteristic" refers to the characteristic observed during flow with a constant pressure drop across the valve. "Installed flow characteristic" refers to the characteristic obtained in service when the pressure drop varies with flow and other changes in the system.

Fig. 4-29 illustrates typical flow-characteristic curves. The quick-opening flow characteristic provides for maximum change in flow rate at low valve travel with a fairly linear relationship. Additional increases in valve travel give sharply reduced changes in flow rate. When the valve plug nears the wide open position, the change in flow rate approaches zero. In a control valve, the quick-opening valve plug is used primarily for on-off service; however, it is also suitable for many applications where a linear valve plug would normally be specified.

In the equal-percentage flow characteristic, equal increments of valve travel produce equal percentage changes in the existing flow. The change in flow rate is always proportional to the flow rate just before the change in position is made for a valve plug, disc, or ball position. When the valve plug, disc, or ball is near its seat and the flow is small, the change in flow rate will be small; with a large flow, the change in flow rate will be large. Valves with an equal-percentage flow characteristic are generally used for pressure control applications. They are also used for other applications where a large percentage of the total system pressure drop is normally absorbed by the system itself, with only a relatively small percentage by the control valve. Valves with an equal-percentage characteristic should also be considered where highly varying pressure drop conditions can be expected.

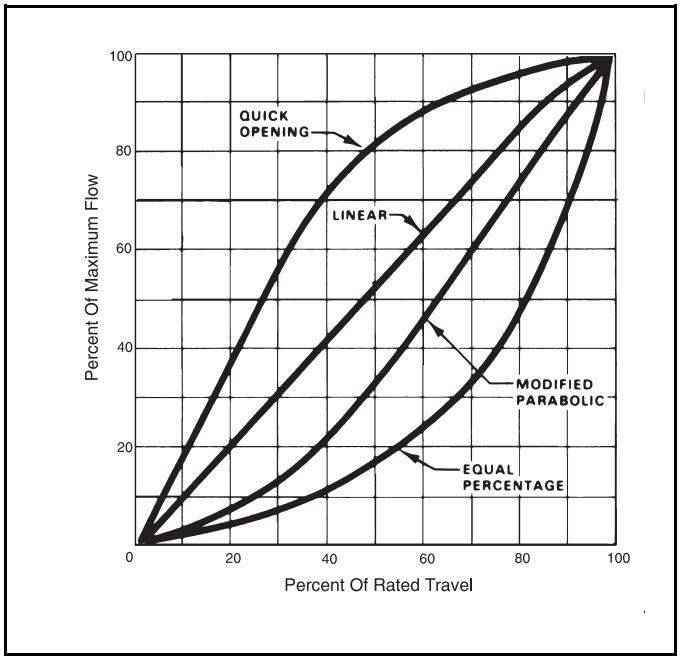
The modified parabolic-flow characteristic curve falls between the linear and the equal-percentage curve.

Note: Where detailed process knowledge is lacking, as a rule of thumb, use equal-percentage characteristics at 70% opening for the valve sizing.

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FIG. 4-29

Example Flow Characteristic Curves



FUNDAMENTALS OF CONTROL VALVE SIZING AND NOISE PREDICTION

Gas Service

Critical Pressure Drop — Critical flow limitation is a significant problem with sizing valves for gaseous service. Critical flow is a choked flow condition caused by increasing gas velocity at the vena contracta. The vena contracta is the point of minimum cross-sectional area of the flow stream which occurs just downstream of the actual physical restriction. When the velocity at the vena contracta reaches sonic velocity, additional increases in pressure drop, ΔP, (by reducing downstream pressure) produce no increase in flow.

In the ISA sizing procedure critical flow limitations are addressed by calculating Y, the expansion factor, for utilization within the actual sizing equation.

$$Y = 1 - \frac{X}{3F_k X_c} \qquad \textbf{Eq 4-9}$$

where,

$$F_k = \frac{k}{1.4} \qquad \textbf{Eq 4-10}$$

Critical pressure drop, and thus critical flow, is realized when $X \geq F_k X_\tau$. Therefore, since the flow can't exceed that produced at the critical pressure drop the value of Y in the following sizing equations should never be less than 0.67.

$$Y = 1 - \frac{X}{3F_k X_c} = 1 - \frac{1}{3} = 0.67 \qquad \textbf{Eq 4-11}$$

Likewise the value of X in the equations should never exceed $F_k X_c$.

Sizing Calculation Procedure – The compressible fluid sizing equations (see Fig. 4-30) can be used to determine the flow of gas or vapor through any style of valve. Absolute units of temperature and pressure must be used in the equation. Most commonly the equations are used to calculate the required C_v and thus valve size for a given set of service conditions. The equations can likewise be rearranged to calculate the flow or pressure drop for a given valve and set of service conditions. The steps are:

1. Select the appropriate sizing equation based on the stated inlet conditions and units of measurement. The limitations on Y and X as discussed above must be observed in all the sizing equations.
2. Calculate an initial, approximate required C_v based on an assumed Rated Pressure Drop Ratio Factor, X_c Initial assumptions for the value of X_c can be based on the general style of valve. See Fig. 4-32.
3. From the valve manufacturer's sizing data select a specific valve type and size such that the listed C_v is equal to or greater than the calculated C_v. The X_c associated

FIG. 4-30

Valve Sizing Equations
(Use Fig. 4-31 for value of Numerical Constant, N)

Flow Basis and Units	Equation
Mass Flow with Specific Weight, γ_1	$w = N_6 F_p C_v Y \sqrt{X P_1 \gamma_1}$
Volumetric Flow with Specific Gravity, G_g	$q = N_7 F_p C_v P_1 Y \sqrt{\frac{X}{G_g T Z}}$
Mass Flow with Molecular Weight, M	$w = N_8 F_p C_v P_1 Y \sqrt{\frac{XM}{TZ}}$
Volumetric Flow with Molecular Weight, M	$q = N_9 F_p C_v P_1 Y \sqrt{\frac{X}{MTZ}}$

FIG. 4-31

Numerical Constants for Gas and Vapor Flow Equations

Constant		Units Used in Equations					
	N	w	q*	p, Δp	γ_1	τ	d,D
N_5	0.00241	–	–	–	–	–	mm
	1000	–	–	–	–	–	in
N_6	2.73	kg/h	–	kPa	kg/m^3	–	–
	27.3	kg/h	–	bar	kg/m^3	–	–
	63.3	lb/h	–	psia	lb/ft^3	–	–
N_7	4.17	–	m^3/h	kPa	–	K	–
	417	–	m^3/h	bar	–	K	–
	1360	–	scfh	psia	–	°R	–
N_8	0.948	kg/h	–	kPa	–	K	–
	94.8	kg/h	–	bar		K	–
	19.3	lb/h	–	psia	–	°R	–
N_9	22.5	–	m^3/h	kPa	–	K	–
	2250	–	m^3/h	bar	–	K	–
	7320	–	scfh	psia	–	°R	–

*q is in cubic feet per hour measured at 14.73 psia and 60°F, or cubic meters per hour measured at 101.3 kPa and 15.6°C.

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with the listed Cv should then be used in the chosen sizing equation to calculate a revised, required C_v. This iteration process continues until the calculated C_v and equals the manufactuer's listed C_v.

4. For a new valve selection a valve size is typically chosen such that the maximum, calculated C_v is close to 75% to 85% of valve travel. This allows for process variability while maintaining flow capability. The minimum, calculated C_v should typically occur at or about 10% of valve travel.
5. F_p is the Piping Geometry Factor. It corrects the sizing equations for the effects of fittings such as reducers and expanders that are attached to the valve body ends. F_p values can be determined via test or calculated per the ANSI/ISA S75.01 standard. If the valve has no such fittings attached, e.g., the nominal value size and nominal pipe size are the same, then F_p = 1.0. Refer to the full standard for the F_p calculations in cases where fittings do exist.

Other valve configurations, such as ball and butterfly valves, can be sized in a similar manner using the unique X_c and C_v values derived by the manufacturers.

FIG. 4-32

Typical C_V, X_C and F_L Values for Valves*

Valve Style	Body Size, Inches	Flow Characteristic					
		Equal Percentage			Linear		
		C_v	X_c	F_L	C_v	X_c	F_L
Globe	1	8	0.74	0.88	17	0.61	0.84
	1-1/2	17	0.69	0.84	30	0.70	.082
	2	25	0.70	0.85	62	0.68	0.77
	2-1/2	49	0.66	0.84	84	0.71	0.81
	3	66	0.66	0.82	118	0.70	0.82
	4	125	0.67	0.82	181	0.74	0.82
	6	239	0.74	0.85	367	0.78	0.84
	8	268	0.60	0.85	526	0.74	0.87
Ball	1	16	0.53	0.86	–	–	–
	2	59	0.53	0.81	–	–	–
	3	120	0.50	0.80	–	–	–
	4	195	0.52	0.80	–	–	–
	6	340	0.52	0.80	–	–	–
	8	518	0.54	0.82	–	–	–
	10	1000	0.47	0.80	–	–	–
	12	1530	0.49	0.78	–	–	–
Butterfly	2	60	0.37	0.69	–	–	–
	3	111	0.40	0.69	–	–	–
	4	238	0.40	0.69	–	–	–
	6	635	.040	0.69	–	–	–
	8	1020	0.40	0.69	–	–	–
	10	1430	0.40	0.69	–	–	–
	12	2220	0.40	0.69	–	–	–
	14	2840	0.40	0.69	–	–	–
	16	3870	0.40	0.69	–	–	–

*At approximately 70% of valve travel. Maximum valve capacity may be estimated using the values given in this figure in conjunction with Fig. 4-29. For a more detailed analysis of capacity capabilities of a given valve at other percentages of travel, consult the valve manufacturer's data.

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Aerodynamic Noise Prediction — Aerodynamic noise, the most common type of control valve noise, is the result of Reynolds stresses and shear forces that are the results of turbulent flow. Noise from turbulent flow is more common in valves handling compressible gases than in those controlling liquids.

The valve manufacturer should provide noise predictions or furnish adequate data to calculate expected noise levels since noise characteristics vary greatly with the type and model of valve being considered or in use. Most control valve manufacturers have valve sizing software available that include the valve sizing and noise prediction routines.

Liquid Service

The procedure used to size control valves for liquid service should consider the possibility of cavitation and flashing since they can limit the capacity and produce physical damage to the valve. In order to understand the problems more thoroughly, a brief discussion of the cavitation and flashing process is presented below.

Cavitation — In a control valve, the fluid stream is accelerated as it flows through the restricted area of the orifice, reaching maximum velocity at the vena contracta. Simultaneously, as the velocity increases, an interchange of energy between the velocity and pressure heads forces a reduction in the pressure.

If the velocity increases sufficiently, the pressure at the vena contracta will be reduced to the vapor pressure of the liquid. At this point, vapor cavities or bubbles, the first stage in cavitation, appear in the fluid stream. Downstream from the vena contracta, the fluid stream undergoes a deceleration process resulting in a reversal of the energy interchange which raises the pressure above the liquid vapor pressure.

The vapor cavities, or bubbles, cannot exist at the increased pressure and are forced to collapse or implode. These implosions are the final stage in the cavitation process. They potentially produce noise, vibration, physical damage, and other performance problems. In order to avoid cavitation completely, the pressures at all points within the valve must remain above the vapor pressure of the liquid. Cavitation can occur as the result of changes in the mean pressures through the valve, but also from localized changes due to flow separations and other local disturbances that are not indicated by examining just the mean inlet, vena contracta, and outlet pressures. Determining when a problem-causing level of cavitation is present represents a considerable challenge. The reader is referred to ISA RP75.23, "Considerations for Evaluating Control Valve Cavitation." This recommended practice provides more information on the cavitation process as well as suggesting a common terminology and methodology for making safe valve selections in cavitating applications. That recommended practice establishes the definition of a cavitation index, K_c, as follows:

$$K_c = \frac{P_1 - P_v}{P_1 - P_2} \qquad \textbf{Eq 4-12}$$

The evaluation of K_c at any given set of service conditions can then be compared to the manufacturer's valve operating limit. As discussed in the recommended practice, the selection of the appropriate operating limit for a given situation is dependent on the service conditions but should also consider other influences such as duty cycle, location, desired life, and past experience. All of these point to the need to consult the valve manufacturer when selecting a valve for cavitation control.

Flashing — The first stages of cavitation and flashing are identical; that is, vapor forms as the vena contracta pressure is reduced to the vapor pressure of the liquid. In the second stage of the flashing process, a portion of the vapor formed at the vena contracta remains in the vapor state because the downstream pressure is equal to or less than the vapor pressure of the liquid.

After the first vapor cavities are formed, the increase in flow rate will no longer be proportional to an increase in the square root of the body differential pressure. When sufficient vapor has been formed, the flow will become completely choked. As long as the inlet pressure (P_1) remains constant, an increase in pressure drop (ΔP) will not cause flow to increase.

Sizing Information — The following section is based on ISA-S75.01, "Flow Equations for Sizing Control Valves." The reader is referred to that standard for more complete discussion of these equations and methods. As that standard points out, these equations are not intended for situations involving mixed-phase fluids, dense slurries, dry solids, or non-Newtonion liquids. In these cases the valve manufacturer should be consulted for sizing assistance.

The ISA methodology recognizes the impact of service conditions that will cause the liquid to vaporize at some point between the inlet and outlet of the valve. This vaporization results in either cavitation or flashing, causing a breakdown in the normal relationship between C_v and $\sqrt{\Delta P}$ and ultimately a limit to the flow through the valve regardless of an increasing pressure drop caused by decreasing P_2. The recognition of this comes in the form of a separate sizing equation for each regime, nonvaporizing and vaporizing. Each must be solved and then the larger calculated C_v chosen as the required value.

This discussion of liquid sizing will be further restricted to:

1. Turbulent flow streams: There are usually flow streams that are not either high viscosity or low velocity. The majority of process plant control valves do operate in the turbulent regime, however if the Reynolds number for a process is less than 4000 the reader is referred to the ISA standard where a non-turbulent flow correction method can be found.
2. Valve installed without fittings attached to the valve ends: When fittings are present there are, as with the previous gas sizing discussion, necessary modifications to the sizing equations to accommodate the additional disturbance to flow. This discussion will be limited to the case where there are no fittings attached, therefore the valve size and pipe size are the same, $F_p = 1.0$. Refer to the full ISA standard for the proper methods if fittings are present.

Sizing Calculation Procedure —

1. Select the appropriate sizing equations based on the stated inlet conditions and units of measurement from Fig. 4-35.
2. Calculate the C_v required using the equation for nonvaporizing flow.
3. Calculate the C_v using the equation for vaporizing flow. An initial assumed value of F_1 can be taken from Fig. 4-32 or the manufacturer's literature. F_F the liquid critical pressure ratio factor, can be found from Fig. 4-33 based on the critical pressure and inlet vapor pressure for subject liquid. Fig. 4-34 lists critical pressures for some common fluids. The user must at this point iterate

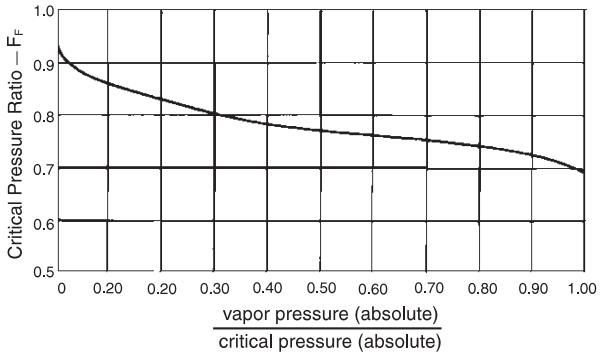
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through this calculation accounting for the variation in F_1 and valve-rated C_v due to valve style, size, trim, flow direction, etc.

4. Select the higher of the two calculated C_v's as the required C_v.
5. From the valve manufacturer's sizing data, select a specific valve type and size such that the listed C_v is equal to or greater than the calculated C_v.
6. See the previous section on Cavitation and consult the manufacturer's data for appropriate valve cavitation operating limits.

FIG. 4-33

Critical Pressure Ratios for All Liquids, F_F



Use this curve for all liquids. Determine the ratio by dividing the pressure which may be obtained from Figure 4-34. From the ratio thus calculated the critical pressure ratio, F_F, may be read from this curve.

FIG. 4-34

Critical Pressure of Various Liquids

psia			
Ammonia	1636	Isobutane	529
Argon	706	Isobutylene	580
n-Butane	551	Methane	668
Carbon Dioxide	1071	Nitrogen	493
Carbon Monoxide	508	Nitrous Oxide	1048
Chlorine	1118	Oxygen	737
Dowtherm A	465	Phosgene	823
Ethane	708	Propane	616
Ethylene	731	Propylene	667
Fluorine	809	Refrigerant 11	635
Helium	33	Refrigerant 12	597

FIG. 4-35

Liquid Valve Sizing Equations

Use Fig. 4-36 for value of Numerical Constants, N

Flow Basis and Units	Equation
Nonvaporizing Mass Flow with Specific Weight, γ_1	$w = N_6 F_p C_v \sqrt{(P_1 - P_2)\gamma_1}$
Nonvaporizing Volumetric Flow with Specific Gravity, G_f	$q = N_1 F_p C_v \sqrt{\frac{P_1 - P_2}{G_f}}$
Vaporizing Mass Flow with Specific Weight, γ_1	$w = N_6 F_L C_v \sqrt{(P_1 - F_F P_v)\gamma_1}$
Vaporizing Volumetric Flow with Specific Gravity, G_f	$q = N_1 F_L C_v \sqrt{\frac{P_1 - F_F P_v}{G_f}}$

FIG. 4-36

Numerical Constants for Liquid Flow Equations

Constant		Units Used in Equations					
N		w	q	p, Δp	d, D	γ_1	v
N1	0.0865	–	m^3/h	kPa	–	–	–
	0.865	–	m^3/h	bar	–	–	–
	1.00	–	gpm	psia	–	–	–
N_6	2.73	kg/h	–	kPa	–	kg/m^3	–
	27.3	kg/h	–	bar	–	kg/m^3	–
	63.3	lb/h	–	psia	–	lb/ft^3	–

INSTALLATION, TROUBLESHOOTING, AND CALIBRATION

Installation and Troubleshooting

Control system troubleshooting logically falls into two categories: (1) the repair of control systems that previously functioned well, and (2) the successful modification of poorly commissioned systems that have never worked properly due to improper application, poor design, faulty hardware, or improper operating procedures (Fig. 4-37). Different techniques are employed for each category.

Failed Systems

- Control system malfunctions normally are reported by the process operator. A discussion with the operator should yield some clues as to the source of the problem, since he has probably been observing it for several hours, or days.
- The next step is to use the "process of elimination" to localize the problem. If replacement of an element with a known good one causes the problem to disappear, this is usually conclusive! Often this simple approach of parts changing will save time by avoiding a detailed system analysis. However, if the situation permits, the "bad" part should be temporarily re-installed to verify a "hard" failure rather than a "hung-up" condition which is often reset by the procedure of substitution.

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- A "detailed system analysis" may be required if a control system has a number of interactive or serially dependent components and especially if more than one component is faulty. The process-of-elimination tests may have shown some conflicting results in this case. A complete control system diagram should be used to help isolate possible problem areas, separate cascade loops, etc. This step usually requires the services of the control engineer or someone familiar with all the control loop components and their functions. Caution should be taken to assure that control system response is observed over a sufficient length of time to detect problems in slow changing processes. Strip chart recorders are very useful in this analysis. Conversely, sequential event recorders may be needed to diagnose intermittent problems which occur only for very brief periods at irregular intervals. Recorders are available which can resolve events to within one millisecond for use in troubleshooting the fastest control loops.

In a well designed control loop, the process is the slowest responding element. Thus the rate of change of a disturbance as it initially propagates around the loop should point to the origin of the problem. In some cases, where multiple symptoms are present, answering the question: "What is common to all symptoms?" will locate the problem. It is often necessary to make observations over a period of time plus taking parallel measurements to completely document the problem and point to the source of the trouble.

Poorly Commissioned Systems

Some control systems may not have been properly commissioned or were not adequately designed. In this case it is necessary to determine how the system is meant to function before proper control can be implemented.

- The first step, after determining how a particular control loop is meant to function, is to put the final control element (valve, usually) on manual control and adjust it for the desired response of the controlled variable. If manual adjustment of the manipulated variable does not cause some response in, or controlling action on, the controlled variable then the control strategy is not suitable and some redesign is in order. However, some control situations such as compressor surge control are too fast for human reaction times and must be verified by other means. Many troublesome control loops are mistakenly declared to be faulty because it is not always realized that there is a trade-off between speed of response and loop stability. Also, there is an inherent interaction between certain control loops. An example is the interaction between temperature and pressure control loops in a distillation column.

FIG. 4-37

Common Measurement Problems

Variable	Symptom	Problem Source	Solution
Pressure	Zero shift, air leaks in signal lines.	Excessive vibration from positive displacement equipment.	Use independent transmitter mtg., flexible process connection lines. Use liquid filled gauge.
	Variable energy consumption under temperature control.	Change in atmospheric pressure.	Use absolute pressure transmitter.
	Unpredictable transmitter output.	Wet instrument air.	Mount local dryer, use regulator with sump, slope air line away from transmitter.
	Permanent zero shift.	Overpressure.	Install pressure snubber for spikes.
Level (diff. press.)	Transmitter does not agree with level.	Liquid gravity change.	Gravity compensate measurement, or recalibrate.
	Zero shift, high level indicated.	Water in process absorbed by glycol seal liquid.	Use transmitter with integral remote seals.
	Zero shift, low level indicated.	Condensable gas above liquid.	Heat trace vapor leg. Mount transmitter above connections and slope vapor line away from transmitter.
	Noisy measurements, high level indicated	Liquid boils at ambient temperature.	Insulate liquid leg.
Flow	Low mass flow indicated.	Liquid droplets in gas.	Install demister upstream; heat gas upstream of sensor.
	Mass flow error.	Static pressure change in gas.	Add pressure recording pen.
	Transmitter zero shift.	Free water in fluid.	Mount transmitter above taps.
	Measurement is high.	Pulsation in flow.	Add process pulsation dampner.
	Measurement error.	Non-standard pipe runs.	Estimate limits of error.
Temperature	Measurement shift.	Ambient temperature change.	Increase immersion length. Insulate surface.
	Measurement not representative of process.	Fast changing process temperature.	Use quick response or low thermal time constant device.
	Indicator reading varies second to second.	Electrical power wires near thermocouple extension wires.	Use shielded, twisted pair thermocouple extension wire, and/or install in conduit.

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- If the process variable is controllable with the loop in the manual mode, note which measurement must be observed to make the decision to open or close the valve; then determine whether the automatic control system makes adjustments based upon the same variables. For example, it may be determined that a single variable controller is insufficient, if the controlled variable deviates too much due to disturbances, feed composition changes, etc. In this case it may be necessary to change the control scheme to include override control arrangements or two or more controllers in cascade to achieve the desired automatic control.

Poor Performance

Two major sources of poor performance in control loops are: (1) excessive time delay between the actuation of the control element (valve) and the resulting change in the measurement of the controlled variable, and (2) a variation in loop gain due to a change in process conditions.

- Excessive time delay can result from process lag and dead-time and from instrument dead-time (such as the cycle time of a chromatograph) and should be compensated for in the control system. To achieve stability in a system with excessive time delay, the controller gain must be low and reset (integral) time must be long. Recovery from process upsets will be slow. When dead-time is present, loop performance will deteriorate proportionally to the square of the dead-time.
- Variable loop gain is a common problem in cascade control loops where the secondary controller is a flow controller without square root extraction on the measurement signal. Since an orifice plate differential pressure signal is proportional to the square of the flow, the setpoint to the secondary controller is also a "flow-squared" signal. This setpoint signal is provided by the output of the primary controller in a cascade configuration. Therefore, the primary loop gain increases rapidly as flow is reduced; if that controller was tuned at high flows it will become unstable at low flows.

Calibration

Calibration of measurement transducers is a vital part of instrument maintenance and should be performed on a regular basis.

Pressure transmitters — Moderate to high pressure units are usually calibrated with a "dead weight tester." This is a hydraulic device in which weights are added to one side of a hydraulic circuit to generate a known pressure which is applied to the input of the transducer under calibration. If the transducer is to be calibrated in units of absolute pressure, a barometer should be used to measure atmospheric pressure and an adjustment of the added weights made. The dead weight tester is normally a test bench device. Low pressure units are calibrated with a pneumatic calibrator which supplies a precise air pressure. Since they are easily adjustable, 3-15 psig devices are usually calibrated by this method. Pneumatic calibrators are also test bench devices. Field or process area checks are often made with a hand-held bulb pump and an accurate pressure gauge.

Differential pressure transmitters — Differential pressure level transmitters are usually calibrated with a mercury manometer or a low level pneumatic calibrator. If remote seals are used, the seal paddle relative elevation should be the same during calibration as when the transmitter is installed. Zero elevation and suppression can be done with a bench calibrator, but if the transmitter has a zero shift with changes in static pressure, this shift must be removed after the transmitter is installed.

Calibration of differential pressure transmitters for flow measurements is done using the same equipment as for levels. An additional consideration must be made for those electronic transmitters which have a square root extractor as an integral part of the transmitter. In this case, the transmitter output is the square root of the input differential pressure. Fig. 4-38 shows the square root input-output relationship.

FIG. 4-38

Square Root Input/Output Relationship

Input		Output	
% of span	Signal level	% of span	Roots
0	3 psi /4 mA	0.0	0
9		30.0	3
25	6 psi /8 mA	50.0	5
50	9 psi /12 mA	70.7	7.07
75	12 psi /16 mA	86.6	8.66
100	15 psi /20 mA	100.0	10

Temperature transmitters — Temperature transmitters are by far the most difficult to calibrate because of the difficulty in generating precise controlled temperatures. Calibration of pneumatic filled-bulb type temperature transmitters is normally a lengthy test bench procedure. Temperature calibrators for generating process temperatures are of three varieties: (1) electrically heated direct air, (2) hot oil bath, and (3) a gas fluidized bed using a thermally conductive powder. All these require a considerable amount of time (20-40 minutes avg.) to reach a steady state temperature. Electronic temperature transmitters which use resistance temperature devices (RTD) have an advantage in that the sensing element calibration needs only a single point check and the transmitter can be calibrated electronically using the known intrinsic properties of the sensor. Field checking of these transmitters is practical.

Thermocouple transmitter calibration requires an accurate millivolt source to simulate the thermocouple signal at the upper and lower limits of the temperature range of the transmitter being calibrated. However, since all thermocouple tables show millivolt readings based on some reference temperature, usually 32°F, some sort of reference temperature compensation must be done. Historically, a reference junction in an ice bath was used to provide ice point compensation. Also a reference junction at ambient temperature was often used in conjunction with a thermometer to read the ambient temperature and provide a reference millivoltage. Modern devices make temperature transmitter calibration much simpler because of built-in reference compensation. Some digital devices merely require the entering of the desired temperature and they provide the correct millivoltage to simulate a thermocouple with automatic reference compensation. Also, compact electronic modules are available for ice point compensation and thermocouple linearization. Some electronic calibrators supply the millivolt signal as well as power for two-wire transmitters and indicators. The millivolt source is set to the values

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which correspond to the upper and lower temperature limits of the range of the transmitter and the transmitter is adjusted so that it produces the desired zero-scale (3 psi, 4 ma, etc.) and full-scale (15 psi, 20 ma, etc.) outputs. When field checking electronic devices, all safety codes must be observed. Calibrators which are battery powered and intrinsically safe are recommended.

COMPUTER SYSTEMS

Analog Computers

A process control analog computer is composed of electronic modules such as amplifiers, summers, controllers, multipliers, dividers, square and square root devices, lead and lag modules, limiters, and special dead-time devices. These modules are based around the integrated circuit operational amplifier and are interconnected to implement the desired control strategy. The process control analog computer is very reliable, low in cost, and easy to use, but has been made obsolete to some degree by the more versatile digital computer.

Digital Computers

Digital computers are attractive for many applications in the process control field because of their speed, accuracy, flexibility, display and logging capabilities, and ability to perform complex calculations and store and transmit vast amounts of data.

Digital computers come in almost any size or shape, ranging from tiny single board special purpose devices to large data processing computers.

Programmable logic controllers (PLC) — A special purpose class of microcomputers designed to implement a series of sequential functions such as ladder network diagrams. They are best suited to batch type operations or machine control. Some examples are: dehydrator control, compressor loading control, engine start-up sequencers, and product blenders. Some PLC's provide for data logging and/or display of information on CRT terminals.

Microcomputers — Microcomputers usually include a higher level software system and more versatile input/output hardware than PLC's. They are often used as the process control computer in small to medium sized installations such as gas plants.

Minicomputers — The minicomputer incorporates a more complex, higher speed arithmetic logic unit as the central processor. Minicomputers accommodate large amounts of memory and may include a high speed disk unit for data and program storage as well as input/output ports to interface with a variety of peripheral equipment. Minicomputers are normally used as the process control computer in larger installations such as refineries, chemical plants, etc.

Process input/output equipment — In addition to the array of printers, CRT terminals, disk units, etc., normally found with a computer system, there is a process input/output system. This is a set of electronic modules, usually a "card-cage" type subassembly, used to interface the process signals to the computer. Most process I/O signals fall into one of the following groups:

- Analog (flows, pressures, levels, temperatures, etc.)
- Digital (on-off status sense or actuation)
- Pulse (tachometers, counters, etc.)
- Serial (coded data)

DIGITAL FIRST-LEVEL CONTROL SYSTEMS

First-level controllers are those which actually manipulate the valves or other final control devices to maintain the process variable at a desired setpoint.

Individual controllers — Controllers located either on the control room panel board or in the process area. These operate independently or may receive a setpoint from another controller or computer system in a cascade arrangement.

Direct digital controllers (DDC) — DDC controllers exist as algorithms in the software of a digital computer, and, through the appropriate transducers, continually sample the respective process measurements, compare them with their corresponding setpoints, and manipulate the appropriate valve or other final control device. Reliability of the computer is essential in a DDC system.

Distributed control systems (DCS) — In a DCS installation the controllers and measurement circuits are modularized in small groups (e.g. eight controllers per module) for greater security against failure. The controllers exist in a combination of hardware and software and may be part of a control scheme programmed in a master DCS system. The controllers may also receive setpoints from a separate supervisory computer system. DCS systems support a variety of communication methods, such as a high speed "data highway" serial data transmission concept which can interface to many different computers. Any digital first-level control system must be backed up by a battery powered uninterruptible power supply (UPS) to prevent loss of control of the process during AC power line interruptions.

Fieldbus — Fieldbus technology is the basis of the next generation of process control. It is an all digital, serial, two-way communication system that interconnects devices in the field such as sensors, actuators and controllers. Fieldbus is a Local Area Network (LAN) for instruments, with built-in capability to distribute a control application across the network.

It has the ability to distribute control among intelligent field devices in the plant and digitally communicate that informtion at high speed.

With full use of field intelligence, process management is no longer just process control. It is now also asset management: gathering and using a wealth of new information from intelligent transmitters, valves and analyzers. It includes configuring, calibrating, monitoring, performing diagnostics, and maintaining records from anywhere in the plant — while the process is running.

The fieldbus can be used for control applications such as temperature, level and flow control. Devices can be powered directly from the fieldbus and operate on wiring that was previously used for 4-20mA devices. The fieldbus can also support intrinsically safe (I.S.) fieldbuses with bus powered devices. An I.S. barrier is placed between the power supply in the safe area and I.S. device in the hazardous area.

ANALYTICAL INSTRUMENTS

Cyclic Analyzers

Many analytical instruments are cyclic, or sampled data devices, such as the chromatograph. These automatically take a sample of the process stream, analyze it, and transmit the re-

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sults to the desired device. Since most analysis cycles take from one to 20 minutes to complete, considerable "dead-time" is introduced into a control loop using this type of measurement. Dead-time compensation should be included in the control scheme for proper control response. Process stream sampling is an item of vital concern in good chromatography or with any analysis technique. Some points of consideration are:

- Sample probe: Must be located at a point in the process where the material to be analyzed is in the desired phase (vapor or liquid) at sufficient pressure and flow.
- Sample lines: Should be kept as short as possible for minimum transport time. May need to be heat-traced to keep vapor samples from condensing. Sample flow must be great enough to completely flush the line between analysis cycles.
- Sample filters: Used to keep any particulate matter out of the analyzer. Knock-out pots or other devices may be required to remove liquid condensate.
- Sample pumps: Required for certain low pressure processes. Vacuum aspirators may be used for low sample flows.

Other cyclic or batch type instruments determine such quantities as vapor pressure, dew point, end point, flash point, Btu content, and sulfur content.

Continuous Analyzers

Other analytical instruments perform continuous analyses of a process stream for a specific parameter such as oxygen content, gas density, etc.

Some continuous analyzers pump or flow a sample through a detector cell. Others place the detector element directly in the process stream. The methods of detection vary widely and may include the use of nuclear radiation, optics, vibrating reeds, flotation cells, catalytic "burners," and other methods.

Fig. 4-39 shows a typical block diagram for a process chromatograph installation. Fig. 4-40 identifies several types of continuous process analytical instruments and a type of detection used by each.

FIG. 4-39

Typical Process Chromatograph System

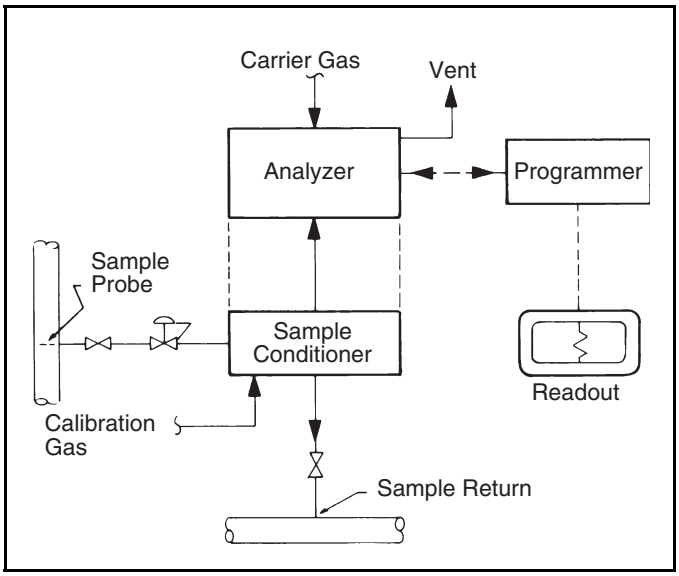


FIG. 4-40

Continuous Analysis Instruments

Analyzer	Detector (Sensing Method)
Liquid Density	Resonant Mass
Liquid Viscosity	Viscosimeter
Liquid Level	Ultrasonic, Gamma Ray
Gas Density	Torque Measurement
Sulfur in Oil	X-Ray Attenuation
pH	Electrolytic
Oxygen	Paramagnetic, Coulometric
Trace Moisture	Electrolytic Cell
Nitrous Oxides	Chemiluminescence
CO, CO_2, SO_2	Infra-red Absorption
H_2S, SO_2	Ultra-violet
Light Hydrocarbons	Chromatograph
Btu (Heating Value)	Calorimeter, Chromatograph

System Control Diagram

During plant development the process engineers specify the process through the development of the P&IDs (Piping & Instrument Diagram). Throughout this effort, the process engineers depict the total plant behavior. However, the P&IDs provide limited facilities for documentation of the overall functionality and operational aspects of the plant.

It's the control system engineer's task to design the control system to fulfill the process functionality required to achieve product specifications as well as the requirements imposed by the overall operating & control philosophy and manning levels.

The operator's understanding of the operational efficiency of the plant is a difficult task without proper documentation of the overall control and monitoring functions available. Often, operational problems within the different systems cannot be identified until the system is in operation, leading to major modifications in late project phases. In order to get a common understanding of control system functionality between all involved parties at an early project stage, System Control Diagrams have been introduced.[2]

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NOTES:

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SECTION 5

Relief Systems

This section is concerned with the design and operation of pressure relieving systems for gas processing plants. The principal elements of pressure relief systems are the individual pressure relief devices, the flare piping system, the flare separator drum, and the flare — including igniters, tips, sealing devices, purge and steam injection for smokeless burning. Application of relief devices must comply with appropriate ASME Vessel Codes. Design of relief systems must also comply with applicable state and federal codes and laws as well as the requirements of the insurance covering the plant or installation. State and federal regulations not only cover safety but also environmental considerations such as air and water pollution and noise abatement. This section presents a convenient summary of relief system information obtained from API and other sources, abridged and modified for this data book. Final design work should be consistent with the full scope of API, ASME, and other code and specification requirements.

FIG. 5-1

Nomenclature

- a = sonic velocity, ft/sec
- A = required discharge area of the valve, sq in. Use valve with the next larger standard orifice size/area
- A_B = bellows area, sq in.
- A' = discharge area of the valve, sq in., for valve with next standard size larger than required discharge area
- Ac = total outside surface area of container, sq ft
- A_D = disk area, sq in.
- A_N = nozzle seat area, sq in.
- A_P = piston area, sq in.
- A_w = total wetted surface area of vessel, sq ft
- A_3 = vessel area exposed to fire, sq ft
- B = liquid expansion coefficient, 1/°F, at relieving temperature [or (Vol/Vol)/°F]
- C = drag coefficient
- C_p = specific heat at constant pressure, BTU/(lb • °F)
- C_v = specific heat at constant volume, BTU/(lb • °F)
- C_1 = coefficient determined by the ratio of specific heats of the gas or vapor at standard conditions
- d = flare tip diameter, inches
- D = particle diameter, ft
- f = correction factor based on the ratio of specific heats
- F = factor due to insulation (see Fig. 5-17)
- F' = relief valve factor, dimensionless
- F_2 = coefficient for subcritical flow (Fig. 5-12)
- F_S = spring force, pounds
- gpm = flow rate, gallons per minute at flowing temperature and pressure
- g = acceleration due to gravity, 32.2 ft/sec^2
- g_c = gravitational constant, 32.2 (ft • lbm)/(lbf • sec^2)
- G = specific gravity of gas referred to air = 1.00 at 60°F and 14.696 psia; or, if liquid, the specific gravity of liquid at flowing temperature referred to water = 1.00 at 60°F
- h_{L1} = enthalpy of saturated liquid at upstream pressure, Btu/lb
- h_{L2} = enthalpy of saturated liquid at downstream pressure, Btu/lb
- h_{G2} = enthalpy of vapor at downstream pressure, Btu/lb
- H = height of vapor space of vessel, ft
- H_l = latent heat of the liquid exposed to fire, Btu/lb
- H_S = flare stack height, ft
- I = radiation intensity at point X, Btu/(hr • ft^2)
- k = specific heat ratio, Cp/Cv (see Section 13)
- K_b = capacity correction factor due to back pressure
- K_c = combination correction for rupture disk = 0.9
  = 1.0 no rupture disk installed
- K_d = coefficient of discharge, obtainable from the valve manufacturer
- K_n = correction factor for Napier steam equation
- K_{sh} = correction factor due to the amount of superheat in the stream
- K_v = capacity correction factor due to viscosity
- K_w = capacity correction factor due to back pressure (Fig. 5-14)
- L = drum length, ft
- L/D = length to diameter ratio of pipe
- L_f = length of flame, ft
- M = Mach number at pipe outlet
- MW = molecular weight of gas or vapor
- MABP = maximum allowable back pressure, psig
- NHV = net heating value of flare gas, Btu/lb
- P = set pressure, psig
- P_{CF} = critical-flow pressure, psia
- P_n = normal operating gas pressure, psia
- P_1 = upstream relieving pressure, psia. This is the set pressure plus the allowable overpressure plus the atmospheric pressure
- P_2 = downstream pressure at the valve outlet, psia
- P_b = back pressure, psig
- ΔP = pressure drop, psi
- ΔP_w = pressure drop, in. of water
- Q = heat input, Btu/hr

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FIG. 5-1 (Cont'd)

Nomenclature

Q_a = minimum required capacity expressed as cfm of air at standard conditions
Q_r = heat released, Btu/hr
Q_v = flow through valve, scfm
r = ratio of downstream pressure to upstream pressure, P_2 / P_1
R = distance from flame center to point X, ft
Re = Reynolds number (dimensionless)
R_o = universal gas constant (10.73) $\frac{\text{psia} \cdot \text{ft}^3}{\text{lb mole} \cdot {}^\circ\text{R}}$
S = specific heat, Btu/(lb • °F)
t = temperature, °F
T = absolute temperature of the inlet vapor, °R
T_n = normal operating gas temperature, °R
T_1 = gas temperature, °R, at the upstream pressure
T_w = vessel wall temperature, °R
U_d = maximum allowable vapor velocity for vertical vessel, ft/sec
V = gas velocity, ft/sec
V_{ex} = exit velocity, ft/sec
V_w = wind velocity, ft/sec (= 1.47 V'_w)
V'_w = wind velocity, mph
W = flow, lb/hr
W_{hc} = hydrocarbon flow, lb/hr
W_{stm} = steam flow, lb/hr
W_f = flare gas flow rate, lb/hr
W_r = vapor rate to be relieved by the relief valve, lb/hr
x_i = weight fraction of component i in total stream
X = distance from the base of the stack to another point at the same elevation, ft
X_c = see Fig. 5-21
Y_c = see Fig. 5-21
Z = compressibility factor at flowing conditions

Greek

Δ = prefix, indicates finite increment
ε = fraction of heat radiated
ρ_L = density of liquid, lb/cu ft
ρ_v = density of vapor, lb/cu ft
θ = angle of flare flame from vertical, degrees
μ = viscosity at flowing temperature, centipoise

Nomenclature for this section is covered in Fig. 5-1. Pressure relief valve is a generic term applied to relief valves, safety valves, or safety relief valves. Definition by type of relief valve is covered in the relief device description. Relief valve characteristics related to pressure vessel requirements are illustrated in Fig. 5-2.

RELIEF DEVICE DESIGN

Pressure relief valves or other relieving devices are used to protect piping and equipment against excessive over-pressure. Proper selection, use, location, and maintenance of relief devices are essential to protect personnel and equipment as well as to comply with codes and laws.

Determination of the maximum relief required may be difficult. Loads for complex systems are determined by conservative assumptions and detailed analysis. By general assumption, two unrelated emergency conditions caused by unrelated equipment failures or operator error will not occur simultaneously (no double jeopardy). The sequence of events must be considered. The development of relief loads requires the engineer to be familiar with overall process design, including the type of pump drives used, cooling water source, spares provided, plant layout, instrumentation, and emergency shutdown philosophy.

This section suggests methods to calculate relief capacity for most emergency conditions, including fire. A common reference for determining individual relieving rates is contained in Section 3 of API RP 521.[1] The design of the proper relieving device must take into consideration all of the following upset conditions for the individual equipment item if such upset can occur. Each upset condition must be carefully evaluated to determine the "worst case" condition which will dictate the relieving device capacity.

Blocked Discharge

The outlet of almost any vessel, pump, compressor, fired heater, or other equipment item can be blocked by mechanical failure or human error. In this case, the relief load is usually the maximum flow which the pump, compressor, or other flow source produces at relief conditions.

Fire Exposure

Fire is one of the least predictable events which may occur in a gas processing facility, but is a condition that may create the greatest relieving requirements. If fire can occur on a plant-wide basis, this condition may dictate the sizing of the entire relief system; however, since equipment may be dispersed geographically, the effect of fire exposure on the relief system may be limited to a specific plot area. Vapor generation will be higher in any area which contains a large number of uninsulated vessels. Various empirical equations have been developed to determine relief loads from vessels exposed to fire. Formula selection varies with the system and fluid considered. Fire conditions may overpressure vapor-filled, liquid-filled, or mixed-phase systems.

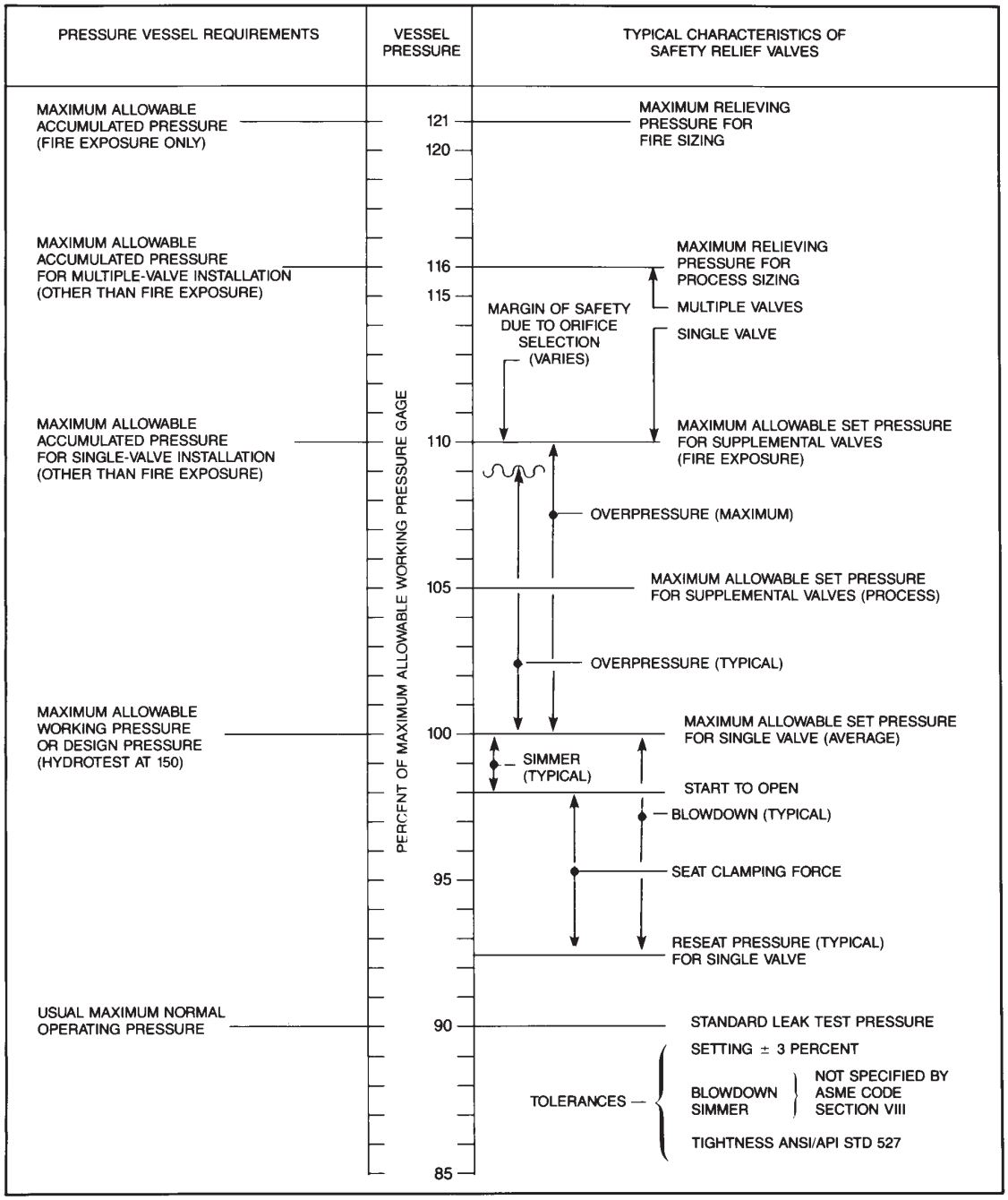
Tube Rupture

When a large difference exists between the design pressure of the shell and tube sides of an exchanger (usually a ratio of 1.5 to 1 or greater), provisions are required for relieving the low pressure side. Normally, for design, only one tube is considered to rupture. Relief volume for one tube rupture can be calculated using appropriate sizing equations in this section. When a cool media contacts a hot stream, the effects of flashing should be considered. Also the possibility of a transient overpressure caused by the sudden release of vapor into an all-liquid system should be considered.

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FIG. 5-2

Characteristics of Safety Relief Valves for Vessel Protection[1]



Notes:

1. The operating pressure may be any lower pressure required.
2. The set pressure and all other values related to it may be moved downward if the operating pressure permits.
3. This figure conforms with the requirements of the *ASME Boiler and Pressure Vessel Code*, Section VIII.
4. The pressure conditions shown are for safety relief valves installed on a pressure vessel (vapor phase).

Courtesy American Petroleum Institute

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Control Valve Failure

The failure positions of instruments and control valves must be carefully evaluated. In practice, the control valve may not fail in the desired position. A valve may stick in the wrong position, or a control loop may fail. Relief protection for these factors must be provided. Relief valve sizing requirements for these conditions should be based on flow coefficients (manufacturer data) and pressure differentials for the specific control valves and the facility involved.

Thermal Expansion

If isolation of a process line on the cold side of an exchanger can result in excess pressure due to heat input from the warm side, then the line or cold side of the exchanger should be protected by a relief valve.

If any equipment item or line can be isolated while full of liquid, a relief valve should be provided for thermal expansion of the contained liquid. Low process temperatures, solar radiation, or changes in atmospheric temperature can necessitate thermal protection. Flashing across the relief valve needs to be considered.

Utility Failure

Loss of cooling water may occur on an area-wide or plant-wide basis. Affected are fractionating columns and other equipment utilizing water cooling. Cooling water failure is often the governing case in sizing flare systems.

Electric power failure, similar to cooling water failure, may occur on an area-wide or plant-wide basis and may have a variety of effects. Since electric pump and air cooler fan drives are often employed in process units, a power failure may cause the immediate loss of reflux to fractionators. Motor driven compressors will also shut down. Power failures may result in major relief loads.

Instrument air system failure, whether related to electric power failure or not, must be considered in sizing of the flare system since pneumatic control loops will be interrupted. Also control valves will assume the position as specified on "loss of air" and the resulting effect on the flare system must be considered.

SPECIAL RELIEF SYSTEM CONSIDERATIONS

Equipment

The following equipment considerations should be followed for relief system design.

Fired Heaters — If there is a possibility that the process side of a fired heater may be blocked in, then a relief valve should be provided to protect the heater.

Pumps — Relief valves are required on the discharge of each positive displacement pump. Normally, these reliefs are piped back to the process upstream of the pump rather than to the flare system. Isolation valves around the relief valves may not be required if the pump itself can be isolated for maintenance.

Vessels and Tanks — Vessels or tanks which are subject to atmospheric "breathing" due to cooling of gas or liquid contents are normally protected by "breather" valves or vacuum relief valves. Vessels in higher pressure service are normally designed for full vacuum if upset conditions can cause a vacuum situation — otherwise vacuum relief valves must be provided to meet code requirements.

Compressors — Each positive displacement compressor must have a relief valve on the discharge of each stage upstream of the block and check valves in order to protect the compressor. These reliefs are normally piped to a closed system.

Low Temperature Flaring

When low temperature streams are relieved, the flare system piping and equipment exposed to cryogenic temperature may require stainless steel or other acceptable alloys.

The system should be designed for the coldest process stream to be relieved plus the lower temperature effect of the expanding fluid (Joule-Thomson effect). Materials selection should be made according to applicable code recommendations.

RELIEVING DEVICES

Several pressure relief devices are certified and approved under Section VIII of the ASME Boiler and Pressure Vessel Code[2] covering unfired pressure vessels. They include spring loaded direct-acting relief valves, pilot operated relief valves, and rupture disks. When the governing code is ANSI B31.3[3] or ANSI B31.8[4], other types of pressure relieving devices such as monitoring regulators, series regulators, weight-loaded relief valves, liquid seals, etc. are permitted. The discussion below is limited to ASME, Section VIII, devices. The devices must be compatible with the service and the overall design of the system. See ASME, Section I, for fired boiler relieving criteria.

Conventional Relief Valves

In a conventional relief valve, the inlet pressure to the valve is directly opposed by a spring. Tension on the spring is set to keep the valve shut at normal operating pressure but allow the valve to open when the pressure reaches relieving conditions. This is a differential pressure valve. Most conventional safety-relief valves available to the petroleum industry have disks which have a greater area, A_D, than the nozzle seat area, A_N. The effect of back pressure on such valves is illustrated in Fig. 5-3. If the bonnet is vented to atmospheric pressure, the back pressure acts with the vessel pressure so as to overcome the spring force, F_S, thus making the relieving pressure less than when set with atmospheric pressure on the outlet. However, if the spring bonnet is vented to the valve discharge rather than to the atmosphere, the back pressure acts with the spring pressure so as to increase the opening pressure. If the back pressure were constant, it could be taken into account in adjusting the set pressure. In operation the back pressure is not constant when a number of valves discharge into a manifold.

A cut-away of a conventional relief valve is shown in Fig. 5-3. Materials of construction for relief valves vary by service.

Balanced Relief Valves

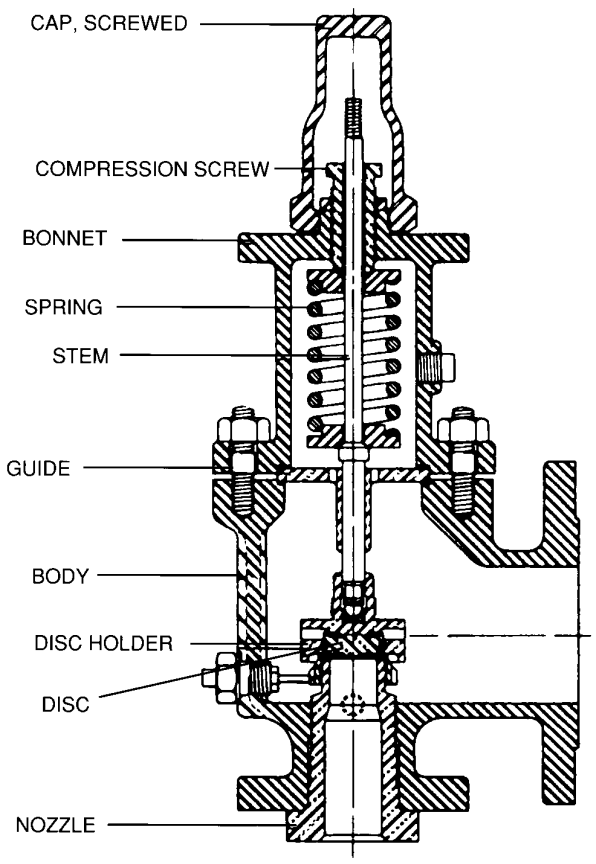
Balanced safety-relief valves incorporate means for minimizing the effect of back pressure on the performance characteristics — opening pressure, closing pressure, lift, and relieving capacity.

These valves are of two types, the piston type and the bellows type, as shown diagrammatically in Fig. 5-4. In the piston type, of which several variations are manufactured, the guide is vented so that the back pressure on opposing faces of the valve disk cancels itself; the top face of the piston, which has

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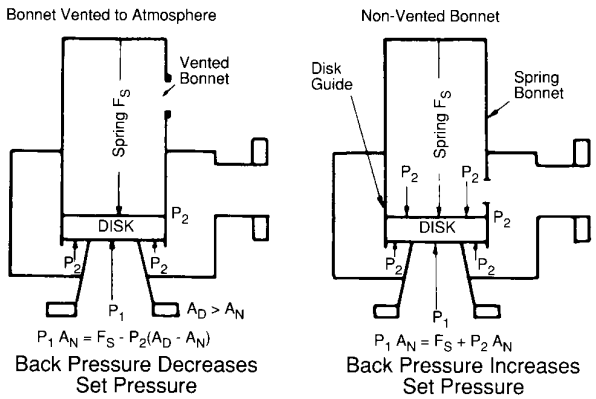
FIG. 5-3

Conventional Safety-Relief Valve



Valve Cross Section

Courtesy Lonergan Company



Effect of Back Pressure on Set Pressure of Conventional Safety-Relief Valves[6]

Courtesy American Petroleum Institute

the same area, A_P, as the nozzle seat area, A_N, is subjected to atmospheric pressure by venting the bonnet. The bonnet-vented gases from balanced piston-type valves should be disposed of with a minimum restriction and in a safe manner.

In the bellows type of balanced valve, the effective bellows area, A_B, is the same as the nozzle seat area, A_N, and, by attachment to the valve body, excludes the back pressure from acting on the top side of that area of the disk. The disk area extending beyond the bellows and seat area cancel, so that there are no unbalanced forces under any downstream pressure. The bellows covers the disk guide so as to exclude the working fluid from the bonnet. To provide for a possible bellows failure or leak, the bonnet must be vented separately from the discharge. The balanced safety-relief valve makes higher pressures in the relief discharge manifolds possible. Both balanced-type valves shown in Fig. 5-4 should have bonnet vents large enough to assure no appreciable back pressure during design flow conditions. If the valve is in a location in which atmospheric venting (usually not a large amount) presents a hazard, the vent should be piped to a safe location independent of the valve discharge system. The user should obtain performance data on the specific type of valve being considered.

A cross section drawing of a balanced (bellows) relief valve is shown in Fig. 5-4.

Pilot Operated Relief Valves

A pilot operated relief valve consists of two principal parts, a main valve and a pilot. The valve utilizes a piston instead of a shaft. Inlet pressure is directed to the top of the main valve piston. More area is exposed to pressure on the top of the piston than on the bottom so pressure, instead of a spring, holds the main valve closed. At the set pressure, the pilot opens, reducing the pressure on top of the piston thereby allowing the main valve to open fully. For some applications, pilot-operated relief valves are available in minimum friction, light-weight diaphragm construction (in place of heavy pistons). Pilot operated valves can allow backflow if downstream pressure exceeds set points. Backflow prevention is required on valves connected to common relief headers. A check valve, split piston type valve, or backflow preventer in the pilot line can be used. A typical pilot operated relief valve is shown in Fig. 5-5. This style valve should be considered for applications involving high back pressure, high operating pressure, or where premium seat tightness is desired.

Resilient Seat Relief Valves

With the use of metal-to-metal seat conventional or balanced type relief valves where the operating pressure is close to the set pressure, some leakage can be expected through the seats of the valve (refer to API Standard 527)[5]. Resilient seat relief valves with either an O-ring seat seal or plastic seats can provide seat integrities which exceed API Standard 527 (Fig. 5-6); however, there are limitations of temperature and material compatibility when using these valves, and manufacturer guidelines should be consulted. Although such valves provide near zero leakage until seat damage occurs, the resilient seats may erode rapidly once leakage begins.

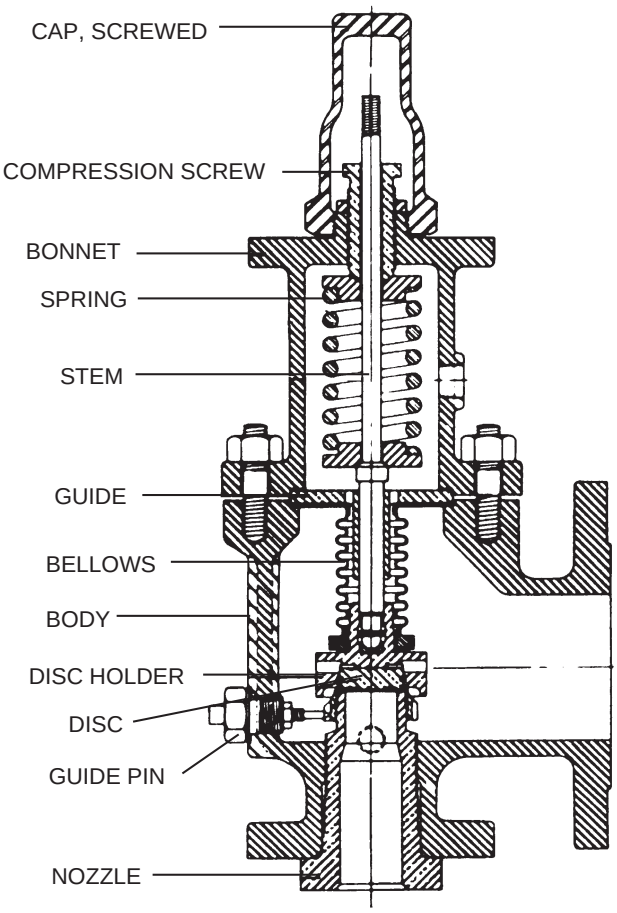
Rupture Disk

A rupture disk consists of a thin diaphragm held between flanges. The disk is designed to rupture and relieve pressure within tolerances established by ASME Code. Rupture disks can be used in gas processing plants, upstream of relief valves, to reduce minor leakage and valve deterioration. In these in-

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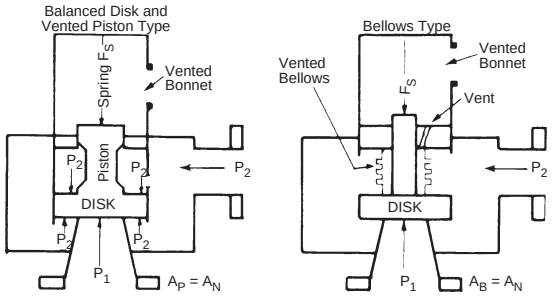
FIG. 5-4

Balanced Safety-Relief Valve



Bellows Valve Cross Section

Courtesy Lonergan Company



Note: Back Pressure has very little effect on set pressure

Effect Of Back Pressure on Set Pressure of Balanced Safety-Relief Valves[6]

Courtesy American Petroleum Institute

stallations, the pressure in the cavity between the rupture disk and the relief valve should be monitored to detect a ruptured disk. In some applications a rupture disk with a higher pressure rating is installed parallel to a relief valve. A rupture disk is subject to fatigue failure due to operating pressure cycles. To establish recommended replacement intervals, consult rupture disk suppliers.

Rupture disks should be used as the primary relieving device only if using a pressure relief valve is not practical. Some examples of such situations are:

(a) Rapid rates of pressure rise. A pressure relief valve system does not react fast enough or cannot be made large enough to prevent overpressure, e.g., an exchanger ruptured tube case or a runaway reaction in a vessel.

(b) Large relieving area required. Because of extremely high flow rates and/or low relieving pressure, providing the required relieving area with a pressure relief valve system is not practical.

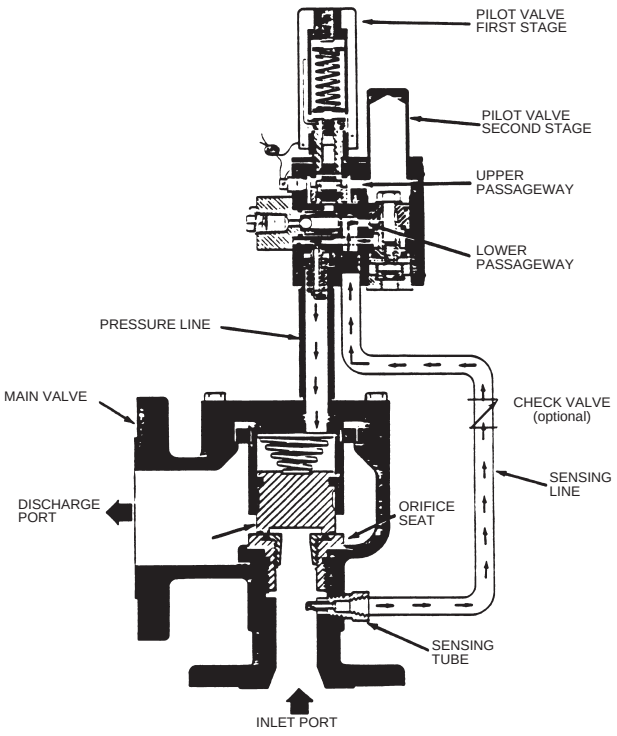
(c) A pressure relief valve system is susceptible to being plugged, and thus inoperable, during service.

SIZING OF RELIEF DEVICES

After the required relief capacity of a relief valve has been determined, the minimum orifice area required must be calculated. Industry standard for orifice designation, orifice area, valve dimensions, valve body sizes, and pressure ratings are available. The standard orifices available — by letter designation, orifice area, and valve body size — are shown in Fig. 5-7.

FIG. 5-5

Pilot Operated Relief Valve

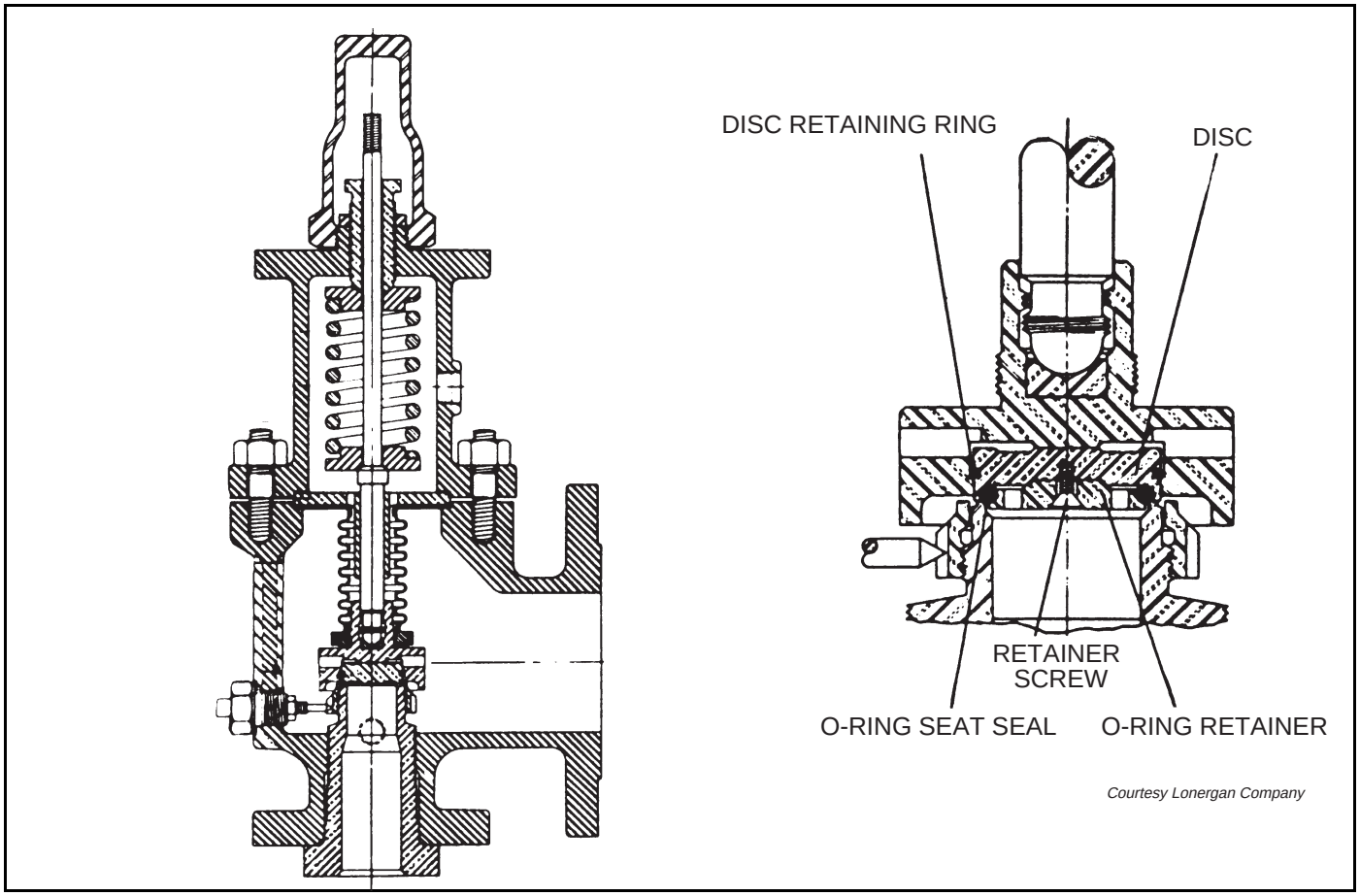


Courtesy GPE Controls

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FIG. 5-6

O-Ring Seals
Conventional and Bellow Valves



In addition to the standard sizes, many relief valves are manufactured with orifice areas smaller than "D", and some pilot-operated relief valves contain orifice areas larger than "T." Manufacturers should be contacted for information on non-standard sizes.

The set pressure and the overpressure allowed must be within the limits permitted by the applicable codes. System analysis must include downstream piping. For instance, consider the use of a relief valve made for a vessel with a maximum allowable working pressure of 100 psig. The relief valve set pressure is 100 psig, and the maximum allowable overpressure is 10% (10 psig). The vessel pressure when relieving must be limited to 110 psig (100 psig set pressure plus 10 psi maximum overpressure). Pressure buildup downstream of the relief valve should never cause the vessel pressure to exceed the maximum allowable overpressure.

Sizing for Gas or Vapor Relief

The rate of flow through a relief valve nozzle is dependent on the absolute upstream pressure (as indicated in Eq 5-1, Eq 5-2, and Eq 5-3) and is independent of the downstream pressure as long as the downstream pressure is less than the critical-flow pressure.[6] However, when the downstream pressure is increased above the critical flow pressure, the flow through the relief valve is materially reduced (e.g., when the downstream pressure equals the upstream pressure, there is no flow).

The critical-flow pressure, P_{CF}, may be estimated by the perfect gas relationship shown in Eq 5-5.

As a rule of thumb if the downstream pressure at the relief valve is greater than one-half of the valve inlet pressure (both pressures in absolute units), then the relief valve nozzle will experience subcritical flow.

Critical Flow — Safety valves in gas or vapor service may be sized by use of one of these equations:

$$A = \frac{W\sqrt{(T_1)(Z)}}{(C_1)(K_d)(P_1)(K_b)(K_c)\sqrt{MW}} \qquad \textbf{Eq 5-1}$$

$$A = \frac{Q_v\sqrt{(T_1)(MW)(Z)}}{(6.32)(C_1)(K_d)(P_1)(K_b)(K_c)} \qquad \textbf{Eq 5-2}$$

$$C_1 = 520\sqrt{k\left(\frac{2}{k+1}\right)^{\frac{k+1}{k-1}}} \qquad \textbf{Eq 5-3}$$

C_1 and K_b can be obtained from Figs. 5-8, 5-9, 5-10, and 5-11. For final design, K should be obtained from the valve

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FIG. 5-7

Relief Valve Designations

Standard Orifice Designation	Orifice Area cm^2	Orifice Area $(in.^2)$	1 × 2	1.5 × 2	1.5 × 2.5	1.5 × 3	2 × 3	2.5 × 4	3 × 4	4 × 6	6 × 8	6 × 10	8 × 10
D	0.710	0.110	•	•	•								
E	1.265	0.196	•	•	•								
F	1.981	0.307	•	•	•								
G	3.245	0.503			•	•	•						
H	5.065	0.785				•	•						
J	8.303	1.287					•	•	•				
K	11.858	1.838							•				
L	18.406	2.853							•	•			
M	23.226	3.60								•			
N	28.000	4.34								•			
P	41.161	6.38								•			
Q	71.290	11.05									•		
R	103.226	16.0									•	•	
T	167.742	26.0											•
		in.	1 × 2	1.5 × 2	1.5 × 2.5	1.5 × 3	2 × 3	2.5 × 4	3 × 4	4 × 6	6 × 8	6 × 10	8 × 10
		mm	25 × 50	38 × 50	38 × 62	38 × 75	50 × 75	38 × 100	75 × 100	100 × 150	150 × 200	150 × 250	200 × 250
			Valve Body Size (Inlet Diameter × Outlet Diameter)										

manufacturer. A value for K of 0.975 may be used for preliminary sizing.

Subcritical Flow — For downstream pressures, P_2, in excess of the critical-flow pressure, P_{CF}, the relief valve orifice area can be calculated from

$$A = \frac{W\sqrt{Z\,T_1}}{(735)\,(F_2)\,(K)\sqrt{MW\,(P_1)\,(P_1 - P_2)}} \qquad \textbf{Eq 5-4}$$

F_2 is taken from Fig. 5-12.

$$P_{CF} = P_1\left(\frac{2}{k+1}\right)^{\frac{k}{k-1}} \qquad \textbf{Eq 5-5}$$

Balanced pressure relief valves should be sized using Eq. 5-1 or Eq. 5-2 and the back pressure correction factor supplied by the valve manufacturer.

Sizing for Steam Relief

Safety-relief valves in steam service are sized by a modification of Napier's steam flow formula. Valve manufacturers can supply saturated steam capacity tables. A correction factor, K_{sh}, must be applied for safety valves in superheated steam service.

For safety-relief valves in steam service, the required area may be estimated from the following equations from the ASME Code:

(For ASME Section VIII)

$$A = \frac{W}{(51.5)\,(P_1)\,(K_{sh})\,(K_d)\,(K_c)\,(K_n)\,(K_b)} \qquad \textbf{Eq 5-6}$$

$$K_n = \frac{0.1906\,P_1 - 1000}{0.2292\,P_1 - 1061} \qquad \textbf{Eq 5-7}$$

where $P_1 > 1500$ psia and $< = 3200$ psia.

See Fig. 5-13 for superheat correction factors. For saturated steam at any pressure, $K_{sh} = 1.0$.

Sizing for Liquid Relief

Turbulent Flow — Conventional and balanced bellows relief valves in liquid service may be sized by use of Eq 5-8. Pilot-operated relief valves should be used in liquid service

FIG. 5-8

Values of Coefficient C_1[6]

k	C_1
0.4	216.9274
0.5	238.8252
0.6	257.7858
0.7	274.5192
0.8	289.494
0.9	303.0392
1.0	315.37*
1.1	326.7473
1.2	337.2362
1.3	346.9764
1.4	356.0604
1.5	364.5641
1.6	372.5513
1.7	380.0755
1.8	387.1823
1.9	393.9112
2.0	400.2962
2.1	406.3669
2.2	412.1494

*Interpolated values, since C_1 becomes indeterminate as k approaches 1.00

Note: Calculated from Eq. 5-3.

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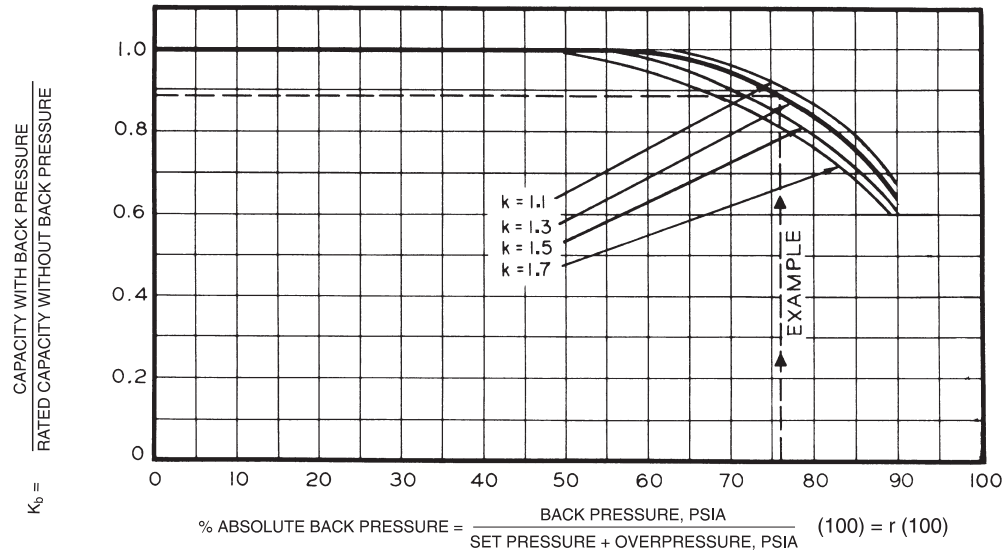
FIG. 5-9

Values of C_1 for Gases

	Mol mass	k	C_1		Mol mass	k	C_1
Acetylene	26	1.28	345	Hydrochloric acid	36.5	1.40	356
Air	29	1.40	356	Hydrogen	2	1.40	356
Ammonia	17	1.33	351	Hydrogen sulfide	34	1.32	348
Argon	40	1.66	377	Iso-butane	58	1.11	328
Benzene	78	1.10	327	Methane	16	1.30	346
Carbon disulfide	76	1.21	338	Methyl alcohol	32	1.20	337
Carbon dioxide	44	1.28	345	Methyl chloride	50.5	1.20	337
Carbon monoxide	28	1.40	356	N-butane	58	1.11	328
Chlorine	71	1.36	352	Natural gas	19	1.27	345
Cyclohexane	84	1.08	324	Nitrogen	28	1.40	356
Ethane	30	1.22	339	Oxygen	32	1.40	356
Ethylene	28	1.20	337	Pentane	72	1.09	325
Helium	4	1.66	377	Propane	44	1.14	331
Hexane	86	1.08	324	Sulfur dioxide	64	1.26	342

FIG. 5-10

Constant Back Pressure Sizing Factor, K_b, for Conventional Safety-Relief Valves (Vapors and Gases Only)[6]



Example:

	Set pressure (MAWP)	= 100 pounds per square inch gauge
	Superimposed back pressure	= 70 pounds per square inch gauge
	Spring set	= 30 pounds per square inch
	Built-up back pressure	= 10 pounds per square inch
	Percent absolute back pressure	$= \left(\frac{(70 + 10 + 14.7)}{(100 + 10 + 14.7)}\right)(100) = 76$ percent
	Follow dotted line — K_b	= 0.89 (from curve)
	Capacity with back pressure	= (0.89)(rated capacity without back pressure)

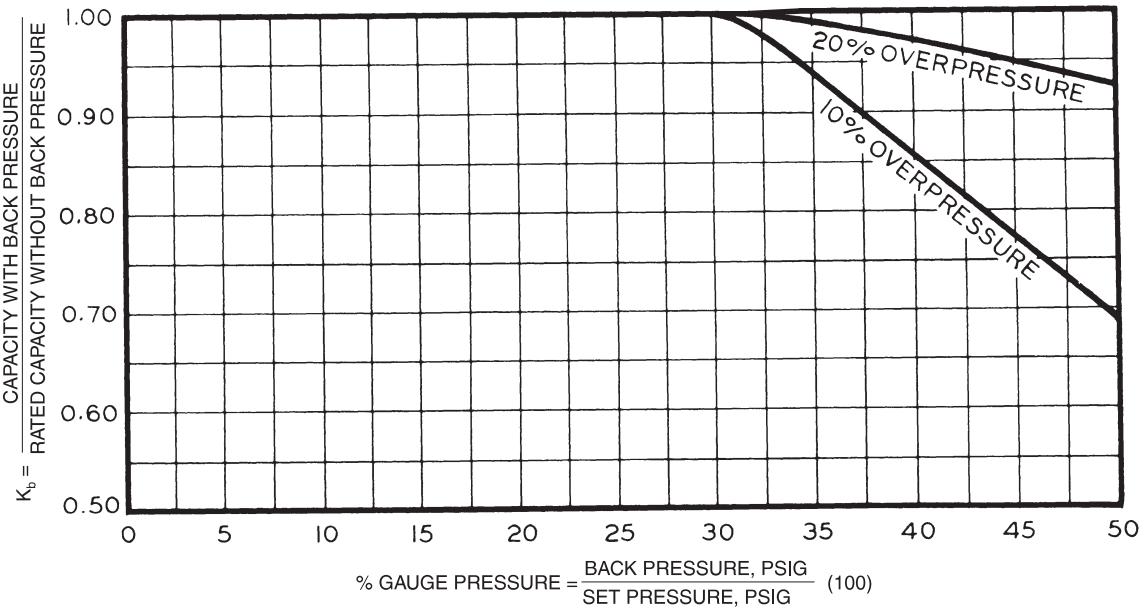
Note: This chart is typical and suitable for use only when the make of valve or the actual critical-flow pressure point for the vapor or gas is unknown; otherwise, the valve manufacturer should be consulted for specific data.

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FIG. 5-11

Variable or Constant Back-Pressure Sizing Factor, K_b, for Balanced Bellows Safety-Relief Valves (Vapors and Gases)[6]



Note: The above curves represent a compromise of the values recommended by a number of relief valve manufacturers and may be used when the make of valve or the actual critical-flow pressure point for the vapor or gas is unknown. When the make is known, the manufacturer should be consulted for the correction factor.

These curves are for set pressures of 50 pounds per square inch gauge and above. They are limited to back pressure below critical-flow pressure for a given set pressure. For subcritical-flow back pressures below 50 pounds per square inch gauge, the manufacturer must be consulted for the values of K_b.

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only when the manufacturer has approved the specific application.

$$A = \frac{(gpm)\sqrt{G}}{(38)(K_d)(K_c)(K_w)(K_v)\sqrt{(P_1 - P_b)}} \qquad \textbf{Eq 5-8}$$

Laminar Flow — For liquid flow with Reynolds numbers less than 4,000, the valve should be sized first with $K_v = 1$ in order to obtain a preliminary required discharge area, A. From manufacturer standard orifice sizes, the next larger orifice size, A′, should be used in determining the Reynolds number, Re, from either of these relationships:

$$Re = \frac{(gpm)(2{,}800)(G)}{\mu\sqrt{A'}} \qquad \textbf{Eq 5-9}$$

$$Re = \frac{(12{,}700)(gpm)}{\mu_s\sqrt{A'}} \qquad \textbf{Eq 5-10}$$

After the Reynolds number is determined, the factor K_v is obtained from Fig. 5-15. Divide the preliminary area (A′) by K_v to obtain an area corrected for viscosity. If the corrected area exceeds the standard orifice area chosen, repeat the procedure using the next larger standard orifice.

Sizing for Thermal Relief

The following may be used to approximate relieving rates of liquids expanded by thermal forces where no vapor is generated at relief valve setting and maximum temperature.[6] These calculations assume the liquid is non-compressible.

$$gpm = \frac{(B)(Q)}{500\,(G)\,(S)} \qquad \textbf{Eq 5-11}$$

Coefficients for this equation may be found in Fig. 23-2.

Typical values of the liquid expansion coefficient, B, at 60°F are:

API Gravity	Specific Gravity	B, 1/°F
Water	1.000	0.0001
3 - 34.9	1.052 - 0.850	0.0004
35 - 50.9	0.850 - 0.775	0.0005
51 - 63.9	0.775 - 0.724	0.0006
64 - 78.9	0.724 - 0.672	0.0007
79 - 88.9	0.672 - 0.642	0.0008
89 - 93.9	0.642 - 0.628	0.00085
94 - 100	0.628 - 0.611	0.0009
n-Butane	0.584	0.0011
Isobutane	0.563	0.0012
Propane	0.507	0.0016

For atmospheric conditions, such as solar radiation, the surface area of the item or line in question should be calculated. Solar radiation should be determined for the geographic area [typically 250-330 Btu/(hr • sq ft)] and applied to the surface area to approximate Q (Btu/hr).

When the flow rate is calculated, the necessary area for relief may be found from the turbulent liquid flow equations.

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Sizing for Mixed Phase Relief

When a safety relief valve must relieve a liquid and gas, it may be sized by:

- Determining the rate of gas and the rate of liquid that must be relieved.
- Calculating the orifice area required to relieve the gas as previously outlined.
- Calculating the orifice area required to relieve the liquid as previously outlined.
- Summing total areas calculated for liquid and vapor to obtain the total required orifice area.

When the material to be relieved through a safety valve is a flashing liquid, it is difficult to know what percentage of the flashing actually occurs in the valve in order to estimate the total required orifice area using the above four steps. A conservative approach is to assume an isenthalpic expansion and that all the vapor is formed in the valve. Then, using enthalpy values, calculate the amount of vapor formed using Eq 5-12. Once the amounts of vapor and liquid assumed present in the valve are determined, the total required orifice area can be calculated.

$$\text{Weight Percent Flashing} = \left[\frac{h_{L1} - h_{L2}}{h_{G2} - h_{L2}}\right](100) \quad \textbf{Eq 5-12}$$

The above sizing method for mixed flows and flashing liquids assumes sonic velocity does not exist. For sizing near sonic conditions where maximum flow may occur, use the method of Fauske[7] for sizing. API 520 Part 1 Appendix D presents a more rigorous method for two phase liquid vapor relief.

Sizing for Fire

The method of calculating the relief rate for fire sizing may be obtained from API RP 521[1], API Standard 2510[8], NFPA 58[9], and possibly other local codes or standards. Each of these references approach the problem in a slightly different manner.

Many systems requiring fire relief will contain liquids and/or liquids in equilibrium with vapor. Fire relief capacity in this situation is equal to the amount of vaporized liquid generated from the heat energy released from the fire and absorbed by the liquid containing vessel. The most difficult part of this procedure is the determination of heat absorbed.

NFPA 58[9] applies to LP-gas (propane and butane) systems. It presents the relief capacity requirements for vessels in terms of the rate of discharge of air, Q_a, and area of the vessel according to the equation:

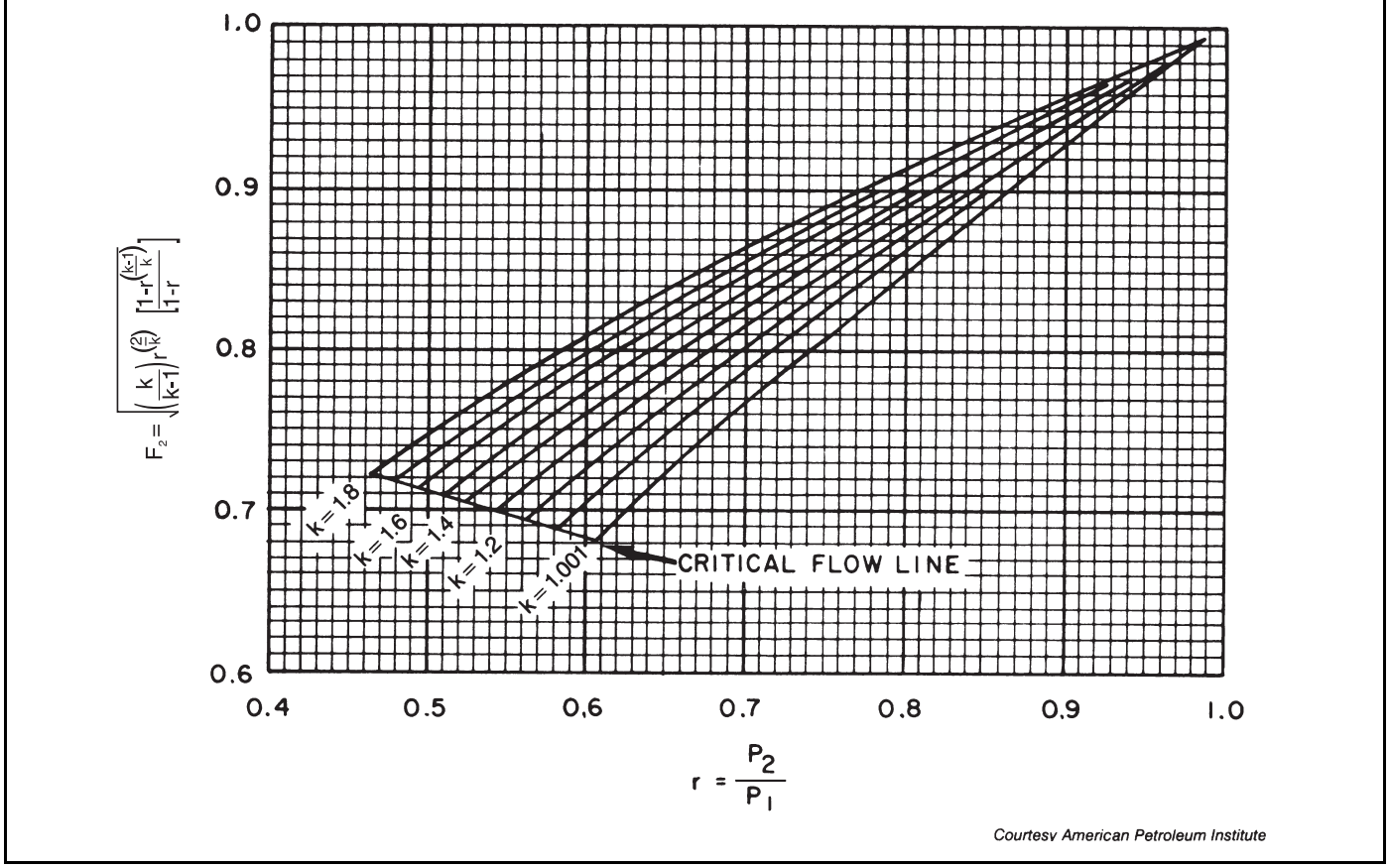
$$Q_a = (53.632)\,(A_c)^{0.82} \quad \textbf{Eq 5-13}$$

This relief capacity is at 120 percent of maximum allowable working pressure (20% overpressure is permissible for fire only).

After determination of Q_a, the required relief valve nozzle size is calculated using Eq 5-2 with the appropriate properties of air.

FIG. 5-12

Values of F_2 for Subcritical Flow[6]



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API RP 521, applies to refineries and process plants. It expresses relief requirements in terms of heat input from the fire to the vessel where adequate drainage and fire fighting equipment exist.

$$Q = (21{,}000)\,(F)\,(A_w)^{0.82} \qquad \textbf{Eq 5-14}$$

The F factor in equation 5-14 is determined from Fig. 5-16. A_w in equation 5-14 is the total wetted surface, in square feet. Wetted surface is the surface wetted by liquid when the tank is filled to the maximum operating level. It includes at least that portion of a tank within a height of 25 feet above grade. In the case of spheres and spheroids, the term applies to that portion of the tank up to the elevation of its maximum horizontal diameter or a height of 25 feet, whichever is greater. Grade usually refers to ground grade but may be any level at which a sizable area of exposed flammable liquid is present.

The amount of vapor generated is calculated from the latent heat of the material at the relieving pressure of the valve. For fire relief only, this may be calculated at 121% of maximum allowable working pressure. All other conditions must be calculated at 110% of maximum allowable working pressure for single relief devices. When the latent heat is determined, required relieving capacity may be found by:

$$W = Q / H_l \qquad \textbf{Eq 5-15}$$

The value W is used to size the relief valve orifice using Eq 5-1 or Eq 5-4.

The latent heat of pure and some mixed paraffin hydrocarbon materials may be estimated using Fig. 12 of API RP 520[6]. Latent heat data may also be obtained by performing flash calculations. Mixed hydrocarbons will boil over a temperature range depending on the liquid composition; therefore, consideration must be given to the condition which will cause the largest vapor generation requirements due to the heat input of a fire. Precise unsteady state calculations are seldom necessary.

The latent heat will approach a minimum value near the critical. When no better information is available, a conservative minimum value of 50 Btu/lb is typically used.

For vessels containing only vapor, API RP 520 has recommended the following equation for determining required relief area based on fire[2]:

$$A = \frac{(F')\,(A_3)}{\sqrt{P_1}} \qquad \textbf{Eq 5-16}$$

F′ may be read from Fig. 5-17.

When a vessel is subjected to fire temperatures, the resulting metal temperature may greatly reduce the pressure rating of the vessel. Design for this situation should consider an

FIG. 5-13

Superheat Correction Factors for Safety Valves in Steam Service[6]

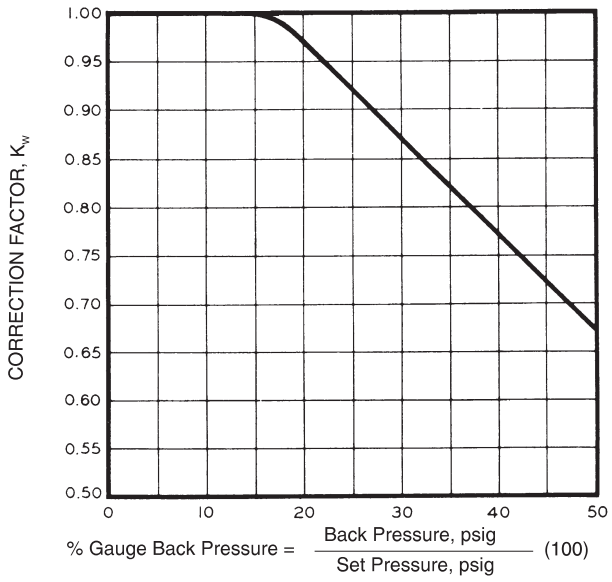
Set Pressure psig	Total Temperature Superheated Steam, °F									
	300	400	500	600	700	800	900	1000	1100	1200
	Correction Factor, K_{sh}									
15	1	0.98	0.93	0.88	0.84	0.8	0.77	0.74	0.72	0.7
20	1	0.98	0.93	0.88	0.84	0.8	0.77	0.74	0.72	0.7
40	1	0.99	0.93	0.88	0.84	0.81	0.77	0.74	0.72	0.7
60	1	0.99	0.93	0.88	0.84	0.81	0.77	0.75	0.72	0.7
80	1	0.99	0.94	0.88	0.84	0.81	0.77	0.75	0.72	0.7
100	1	0.99	0.94	0.89	0.84	0.81	0.77	0.75	0.72	0.7
120	1	0.99	0.94	0.89	0.84	0.81	0.78	0.75	0.72	0.7
140	1	0.99	0.94	0.89	0.85	0.81	0.78	0.75	0.72	0.7
160	1	0.99	0.94	0.89	0.85	0.81	0.78	0.75	0.72	0.7
180	1	0.99	0.94	0.89	0.85	0.81	0.78	0.75	0.72	0.7
200	1	0.99	0.95	0.89	0.85	0.81	0.78	0.75	0.72	0.7
220	1	0.99	0.95	0.89	0.85	0.81	0.78	0.75	0.72	0.7
240		1	0.95	0.9	0.85	0.81	0.78	0.75	0.72	0.7
260		1	0.95	0.9	0.85	0.81	0.78	0.75	0.72	0.7
280		1	0.96	0.9	0.85	0.81	0.78	0.75	0.72	0.7
300		1	0.96	0.9	0.85	0.81	0.78	0.75	0.72	0.7
350		1	0.96	0.9	0.86	0.82	0.78	0.75	0.72	0.7
400		1	0.96	0.91	0.86	0.82	0.78	0.75	0.72	0.7
500		1	0.96	0.92	0.86	0.82	0.78	0.75	0.73	0.7
600		1	0.97	0.92	0.87	0.82	0.79	0.75	0.73	0.7
800			1	0.95	0.88	0.83	0.79	0.76	0.73	0.7
1000			1	0.96	0.89	0.84	0.78	0.76	0.73	0.71
1250			1	0.97	0.91	0.85	0.8	0.77	0.74	0.71
1500				1	0.93	0.86	0.81	0.77	0.74	0.71
1750				1	0.94	0.86	0.81	0.77	0.73	0.7
2000				1	0.95	0.86	0.8	0.76	0.72	0.69
2500				1	0.95	0.85	0.78	0.73	0.69	0.66
3000					1	0.82	0.74	0.69	0.65	0.62

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FIG. 5-14

Variable or Constant Back-Pressure Sizing Factor K_W on Balanced Bellows Safety-Relief Valves (Liquids Only)[6]



Note: The above curve represents a compromise of the values recommended by a number of relief-valve manufacturers. This curve may be used when the make of the valve is not known. When the make is known, the manufacturer should be consulted for the correction factor.

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emergency depressuring system or a water spray system to keep metal temperatures cooler. A discussion on vapor depressuring may be found in API RP 521, paragraph 3.19. For additional discussion on temperatures and flowrates due to depressurization and fires refer to Reference 15.

RELIEF VALVE INSTALLATION

Relief valve installation requires careful consideration of inlet piping, pressure sensing lines (where used), and startup procedures. Poor installation may render the safety relief valve inoperable or severely restrict the valve's relieving capacity. Either condition compromises the safety of the facility. Many relief valve installations have block valves before and after the relief valve for in-service testing or removal; however, these block valves must be sealed or locked opened.

Inlet Piping

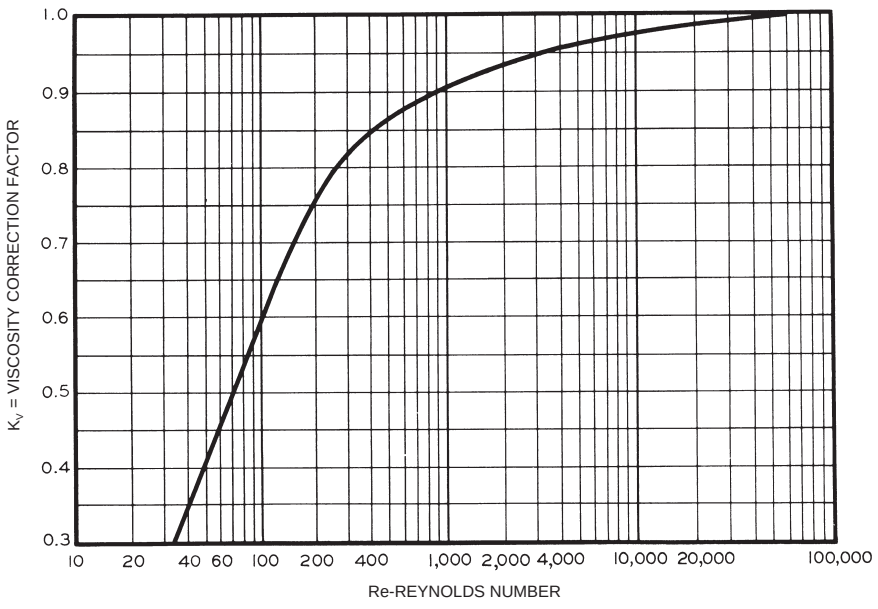
The proper design of inlet piping to safety relief valves is extremely important. Relief valves should not be installed at physically convenient locations unless inlet pressure losses are given careful consideration. The ideal location is the direct connection to protected equipment to minimize inlet losses. API RP 521, Part II recommends a maximum inlet pressure loss to a relief valve of three percent of set pressure. This is also an ASME Code requirement. This pressure loss shall be the total of the inlet loss, line loss, and the block valve loss (if used). The loss should be calculated using the maximum rated flow through the safety relief valve.

Discharge Piping

An adequate discharge pipe size for direct spring operated valves is critical. Improperly sized, it may cause valve failure. Pressure losses may occur in discharge headers causing excessive back pressure and excessive back pressure may cause the relief valve to close. When the valve closes, the back pressure in the discharge header decreases,

FIG. 5-15

Capacity Correction Factor Due to Viscosity



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FIG. 5-16

Environmental Factors[8]

Environment	F Factor
Bare metal vessel	1.0
Insulation thickness: [a,b] (inches)	
1	0.30
2	0.15
4	0.075
6	0.05
8	0.037
10	0.03
12 or more[c]	0.025
Concrete thickness, in inches	Double above F factor for equal thickness.
Water-application facilities[d]	1.0
Depressuring and emptying facilities[e]	1.0
Underground storage	0.0
Earth-covered storage above grade	0.03

Notes:

[a] To take credit for reduced heat input, the insulation shall resist dislodgment by fire-hose stream, shall be noncombustible, and shall not decompose at temperatures up to 1,000 degrees Fahrenheit. If insulation does not meet these criteria, the environmental F factor for a bare vessel should be used.

[b] These factors are based on an arbitrary thermal conductivity value of 4 British thermal units per hour per square foot per degree Fahrenheit per inch of insulation and a temperature differential of 1,600 degrees Fahrenheit when using a heat input value of 21,000 British thermal units per hour per square foot in accordance with conditions assumed in API RP 520. Where these conditions do not exist, engineering judgment should be exercised either in selecting a higher F factor or in providing other means of protecting tanks from fire exposure.

[c] The insulation credit is arbitrarily limited to the F factor shown for 12 inches of insulation even though greater thickness may be used. A further credit, if taken, would result in a relieving device size which would be impractically small but may be used if warranted by design considerations.

[d] Water films covering the metal surfaces can, under ideal conditions, absorb substantially all incident radiation. However, the reliability of effective water application is dependent upon many factors. Freezing weather, high winds, clogging of system, unreliability of water supply, and tank surface conditions are a few factors which may prevent adequate or uniform water coverage. Because of these uncertainties, the use of an environment F factor other than 1.0 for water spray is generally not used.

[e] Depressuring devices may be used, but no credit for their use should be allowed in sizing pressure relief valves for fire exposure.

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and the valve re-opens. Rapid cycling can occur as this process is repeated. In addition, set pressure of other conventional valves connected to the same discharge system may be adversely affected by such excessive back pressure. See Fig. 5-18.

Lift and set pressures of pilot operated relief valves with the pilot vented to the atmosphere are not affected by back pressure; however, if the discharge pressure can exceed the inlet pressure (e.g., tanks storing low vapor pressure material), a back-flow preventer (vacuum block) must be used.

Balanced relief valve set pressure will not be as significantly affected by back pressure as direct spring operated valves. Balanced relief valves will suffer reduced lift as back pressure increases. This is shown in application of the K_b factor for balanced valves (Fig. 5-11).

The valve relieving capacity of any relief valve is restricted by back pressure if the discharge pressure exceeds the critical-flow pressure.

Discharge piping must be at least the same size as the valve outlet but is generally larger in diameter than the valve outlet size to limit back pressure.

A useful article on the sensitivity of relief valves to proper inlet and outlet piping design has been published by Van Boskirk.[13]

Reactive Force

On high pressure valves, the reactive forces during relief are substantial and external bracing may be required. See equations in API RP 520 for computing this force.

Rapid Cycling

Rapid cycling can occur when the pressure at the valve inlet decreases at the start of relief valve flow because of excessive pressure loss in the piping to the valve. Under these conditions, the valve will cycle at a rapid rate which is referred to as "chattering." The valve responds to the pressure at its inlet. If the pressure decreases during flow below the valve reseat point, the valve will close; however, as soon as the flow stops, the inlet pipe pressure loss becomes zero and the pressure at the valve inlet rises to tank pressure once again. If the vessel pressure is still equal to or greater than the relief valve set pressure, the valve will open and close again. The inlet pressure loss should be limited to 3%.

An oversized relief valve may also chatter since the valve may quickly relieve enough contained fluid to allow the vessel pressure to momentarily fall back to below set pressure only to rapidly increase again. Rapid cycling reduces capacity and is destructive to the valve seat in addition to subjecting all the moving parts in the valve to excessive wear. Excessive back pressure can also cause rapid cycling as discussed above.

Resonant Chatter

Resonant chatter can occur with safety valves when the inlet piping produces excessive pressure losses at the valve inlet and the natural acoustical frequency of the inlet piping approaches the natural mechanical frequency of the valve's basic moving parts. The higher the set pressure, the larger the valve size, or the greater the inlet pipe pressure loss, the more likely resonant chatter will occur. Resonant chatter is uncontrollable; that is, once started it cannot be stopped unless the pressure is removed from the valve inlet. In actual application, however, the valve can self-destruct before a shutdown can take place because of the very large magnitude of the impact forces involved.

Seat Leakage of Relief Valves

Seat leakage is specified for conventional direct spring operated metal-to-metal seated valves by API RP 527. The important factor in understanding the allowable seat leak is that it is stated at 90% of set point. Therefore, unless special seat lapping is specified or soft seat designs used, a valve operating with a 10% differential between operating and set pressures may be expected to leak.

There are no industry standards for soft seated valves. Generally leakage is not expected if the valve operates for one minute at 90% of set pressure and, in some cases, no leakage

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at 95%, or even higher. Longer time intervals could promote leakage. Application of soft seated valves is limited by operating temperatures. Manufacturer guidelines should be consulted.

RELIEF SYSTEM PIPING DESIGN

Grouping of Systems

It is not unusual to relieve non-flammable and non-toxic materials, such as air, steam, carbon dioxide, and water directly to the atmosphere. To protect personnel, precautions need to be taken in venting hot, cold, or high pressure materials.

In certain cases, hydrocarbon relief vapor streams may be discharged to the atmosphere. However, the decision to discharge hydrocarbons or other flammable or hazardous vapors to the atmosphere must ensure that disposal can be accomplished without creating a hazard, such as formation of flammable mixtures at grade level or on elevated structures, exposure of personnel to toxic or suffocating atmosphere, ignition of relief streams at the point of emission, excessive noise levels, corrosion, or air pollution.

Depending on plot plan, the range of equipment design pressures, desirability of isolating certain streams, etc., it may prove desirable to provide two or more flare systems, such as separation of high pressure and low pressure headers. Other factors influencing this decision include stream temperatures, compositions, quantities, and economics.

Load Determination

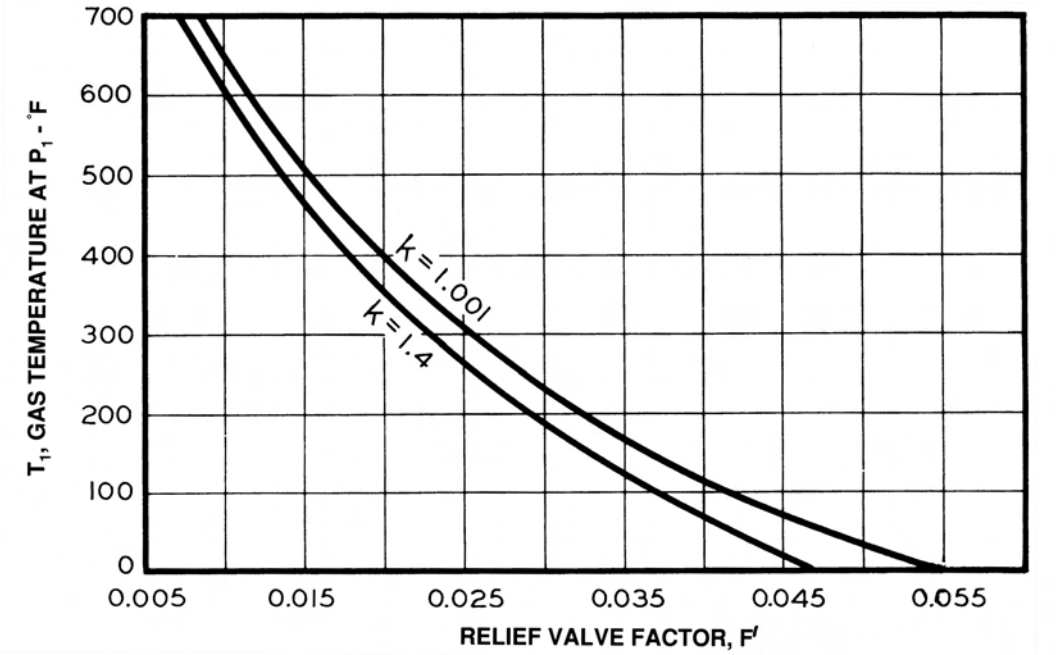
The maximum relief load anticipated for each relief system may be determined by tabulating the relief loads discharging to each header system at the conditions expected to exist in the system. For further discussion, see Section 5.2 of API RP 521.

Back Pressure Consideration

After load determination, it is necessary to decide on the location of the flares, and size of the headers and flare lines. Location and height of the flares must consider radiation heat and emissions. This will involve fixing the maximum back pressure for the system and choosing between conventional, pilot operated, or balanced safety valves for the various relief stations. Factors which tend to justify balanced valves are a potential for high flare header back pressure and economic line sizing criteria.

FIG. 5-17

Relief-Valve Factors for Noninsulated Vessels in Gas Service Exposed to Open Fires[6]



Notes:

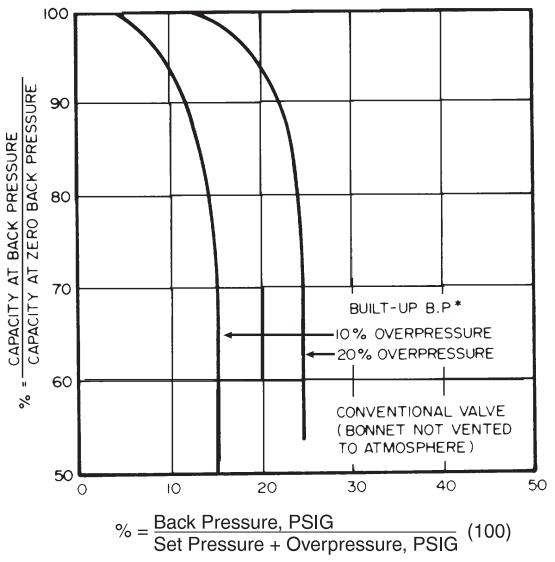
1. Fig. 5-9 gives k values for some gases; for others, the values can be determined from the properties of gases as presented in any acceptable reference work.
2. These curves are for vessels of carbon steel.
3. These curves conform to the relationship $F' = \left(\frac{0.1406}{C_1 K_d}\right)\left(\frac{\Delta T^{1.25}}{T_1^{0.6506}}\right)$
4. The curves are drawn using 1,100 degrees Fahrenheit as the vessel wall temperature. This value is a recommended maximum temperature for the usual carbon steel plate materials whose physical properties at temperatures in excess of 1,100 degrees Fahrenheit show signs of undesirable tendencies. Where vessels are fabricated from alloy materials, the value for Tw should be changed to a more proper recommended maximum.

   It is recommended that the minimum value of F′ = 0.01 (when it is unknown use 0.045).

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FIG. 5-18

Typical Effects of Variable Back Pressure on Capacity of Conventional Safety-Relief Valves[6]



Sizing Methods

Line sizing for flare headers and relief lines requires the use of compressible flow equations. Computer programs are available for use in sizing flare headers. A calculation method is outlined below:

1. Start at the flare tip, where the outlet pressure is atmospheric, use design flows and work toward the individual relief valves (pressure drop across the tip will vary with make — check with the manufacturer). Typical tip drop is 55" water (see Eq 5-24). Flare tip velocity can vary from Mach 0.2 for open pipe flares to Mach 0.5 or greater for engineered flare tips and wind shields. Pressure drop through seals must also be included.
2. Establish equivalent pipe lengths between points in the system and establish losses through fittings, expansion, and contraction losses.
3. Limit the maximum allowed velocity at any part of the flare system to Mach 0.7 (average velocity of Mach 0.4 is recommended).
4. Estimate properties of gases in the headers from the following mixture relationships (i indicates the ith component).

$$MW = \Sigma W_i / \Sigma (W / MW)_i \qquad \textbf{Eq 5-17}$$

$$T = \Sigma W_i T_i / \Sigma W_i \qquad \textbf{Eq 5-18}$$

$$\mu = \Sigma x_i \mu_i (MW)_i^{0.5} / \Sigma x_i (MW)_i^{0.5} \qquad \textbf{Eq 5-19}$$

5. Calculate the inlet pressure for each section of the line by adding the calculated pressure drop for that section to the known outlet pressure.
6. Calculate sections of pipe individually using the inlet pressure of a calculated section as the outlet pressure for the new section.
7. Continue calculations, working towards the relief valve.
8. Check calculated back pressure at the relief valve against the maximum allowable back pressure (MABP). The calculated back pressure should be less than the MABP.
9. Limit the MABP to about 40% of the set pressure for balanced-bellows relief valves (consult valve manufacturer), and 10% of the set pressure for conventional relief valves.
10. Adjust line size of headers until the calculated back pressure does not exceed the MABP for each valve in the system.

The method outlined above employs sizing equations which assume isothermal flow in the flare header. This is adequate for most uses; however, if the actual flow condition differs greatly from isothermal, the use of more complex equations and methods is required to predict pressure and temperature profiles for the headers.

The choice of piping material other than carbon steel may be dictated by temperatures and pressures in some parts of the flare system. Flare systems relieving fluids that produce cryogenic temperatures may require special metallurgy.

KNOCKOUT DRUMS

Gas streams from reliefs are frequently at or near their dew point, where condensation may occur. A knockout drum is usually provided near the flare base, and serves to recover liquid hydrocarbons, prevent liquid slugs, and remove large (300-600 micron diameter) liquid particles. The knockout drum reduces hazards caused by burning liquid that could escape from the flare stack. All flare lines should be sloped toward the knockout drum to permit condensed liquid to drain into the drum for removal. Liquid traps in flare lines should be avoided. If liquid traps are unavoidable, a method for liquid removal should be provided. The locating of the flare knockout drum also needs to take into account radiation effect from the burning flare.

A seal may be provided in the flare system between the knockout drum and the flaretip to prevent flashbacks. For added safety, a small amount of fuel gas may be used to continuously purge the system and eliminate the entrance of air (oxygen).

Sizing

The diameter of a vertical knockout drum can be calculated from the maximum allowable vapor velocity using the following equation:

$$U_d = (1.15) \sqrt{\frac{(g)\,(D)\,(\rho_L - \rho_v)}{(\rho_v)\,(C)}} \qquad \textbf{Eq 5-20}$$

This equation may be used to calculate the allowable velocity based on a particle size of 300 to 600 microns (0.001 to 0.002 ft). The value of C in Eq 5-20 is determined using Fig. 5-19. API RP 521 is a common reference for designing flare knockout drums.

FLARE SYSTEMS

Sizing and safety in designing a flare facility involve consideration of thermal radiation, explosion hazards, liquid carryover, noise, and ground level concentration of toxic material.

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Types

Flares can be classified as:

1. Pipe Flares — Vertical or horizontal pipes with external ignition pilot.
2. Smokeless Flares — Vertical, single, or multiple burners designed to properly mix adequate oxygen from the air with relieved vapors for complete combustion.
3. Endothermic Flares — Elevated incinerators for low heat content streams.

Pipe Flares — This type of flare is used for hydrocarbon vapor streams such as methane, hydrogen, and carbon monoxide which produce limited smoke when burned. Heavier hydrocarbons may be burned but with smoking. Velocity for "open-pipe" flares should be limited to Mach 0.4 to minimize flame lift-off.

Smokeless Flares — Smokeless flares are used for smokeless disposal of hydrocarbon streams. These flares use a variety of methods such as steam, high pressure fuel gas, water spray, an air blower, or high velocity vortex action to mix air with the gas.

Fired or Endothermic Flares — These flares are a form of elevated incinerator for use on low heat waste streams, such as sulfur plant tail gas. When the heat content of the waste stream is below 115 Btu/scf, a fired flare with a high energy assist gas may be required for complete combustion.

Thermal Radiation

Thermal radiation is a prime concern in flare design and location. Thermal radiation calculations must be done to avoid dangerous exposure to personnel, equipment, and the surrounding area (trees, grass). The following calculation procedure is a convenient means to find the height of the flare stack and the intensity of radiation at different locations. For flares with a smokeless capacity exceeding approximately 10 percent, methods which account for gas composition, mixing, and smoke as presented in API 521 or as summarized in reference 14 should be used. Flare vendors should be consulted for large flowrates or when smokeless requirements exceed approximately 10 percent. Calculations given by API 521 or other methods may yield different answers.

Spherical Radiation Intensity Formula:

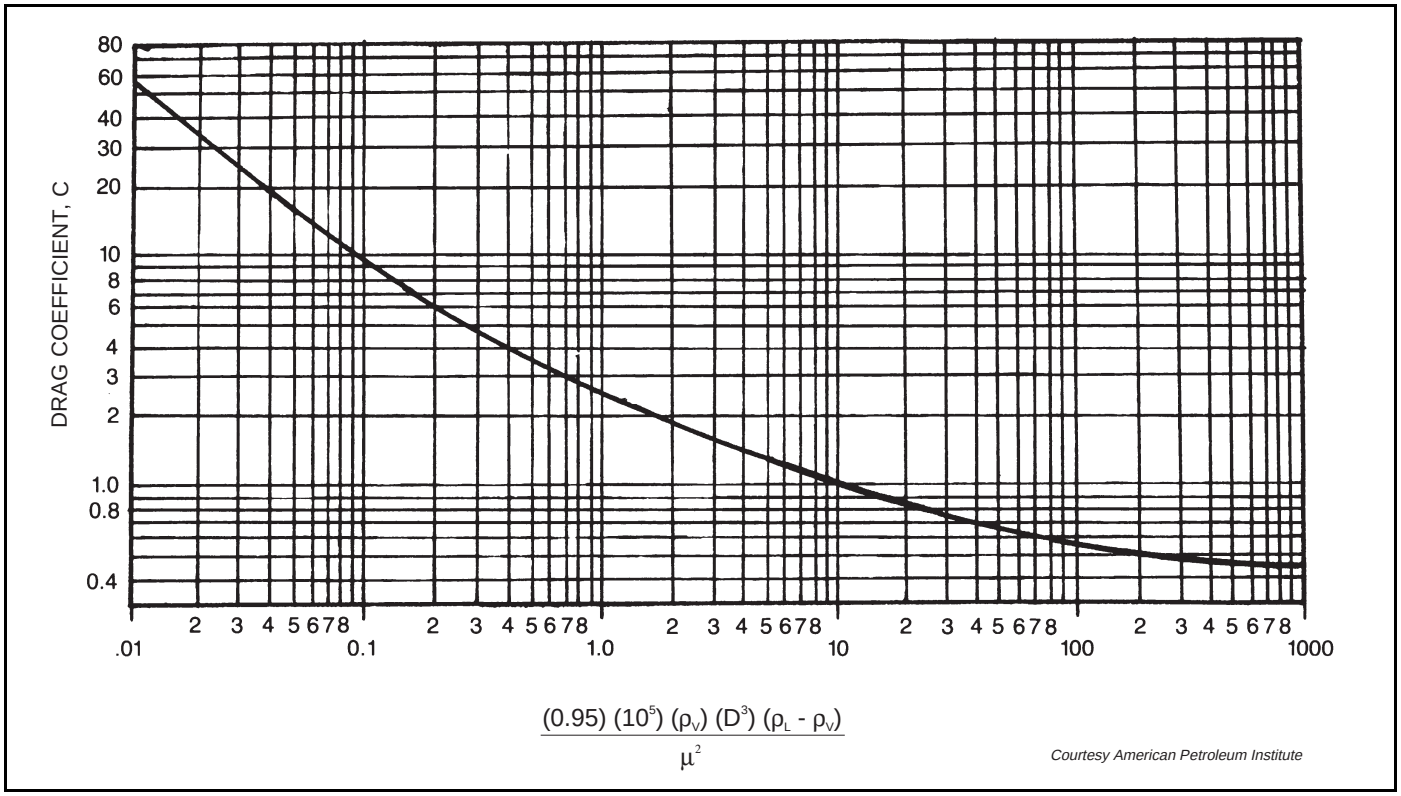
$$I = \frac{(W_f)\,(NHV)\,(\varepsilon)}{4\,\pi\,(R^2)} \qquad \textbf{Eq 5-21}$$

This equation has been found to be accurate for distances as close to the flame as one flame length.

Equation 5-21 is valid so long as the proper value of fraction of heat radiated, ε, is inserted. Classically, ε has been considered a fuel property alone. Brzustowski et al.[10] experimentally observed a dependence of ε on jet exit velocity. Other authors have presented models that consider the carbon particle concentration in the flame. The fraction of heat radiated is a function of many variables including gas composition, tip diameter, flare burner design, flowrate and velocity, flame temperature, air-fuel mixing, and steam or air injection; therefore a flare supplier should be consulted to determine the specific values for a given application. A list of vendor recommended

FIG. 5-19

Determination of Drag Coefficient[1]



Courtesy American Petroleum Institute

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FIG. 5-20

Fraction of Heat Radiated Values for Flared Gases

Carbon Monoxide	0.075
Hydrogen	0.075
Hydrogen Sulfide	0.070
Ammonia	0.070
Methane	0.10
Propane	0.11
Butane	0.12
Ethylene	0.12
Propylene	0.13

The maximum value of ε for any gas is 0.13.

fraction of heat radiated values for the most frequently flared gases is shown in Fig. 5-20.

To calculate the intensity of radiation at different locations, it is necessary to determine the length of the flame and its angle in relation to the stack (see Fig. 5-21). A convenient expression to estimate length of flame, L_f, is shown below, based on information from equipment suppliers.

$$L_f = (10)\,(d)\sqrt{\frac{\Delta P_w}{55}} \qquad \textbf{Eq 5-22}$$

or from API RP 521,

$$L_f = 3.94\,[(Q_r)\,(10^{-6})]^{0.474} \qquad \textbf{Eq 5-23}$$

For conventional (open pipe) flares, an estimate of total flare pressure drop is 1.5 velocity heads based on nominal flare tip diameter. The pressure drop equivalent to 1 velocity head is given by:

$$\Delta P_w = \frac{(27.7)\,\rho\,V^2}{(2\,g_c)\,(144)} = \frac{\rho\,V^2}{344.8} \qquad \textbf{Eq 5-24}$$

ΔP_w is the pressure drop at the tip in inches of water. After determining tip diameter, d, using Eq 5-25, and the maximum required relieving capacity, flame length for conditions other than maximum flow can be calculated using Eq 5-22 and Eq 5-24.

Common practice is to use tip velocities of up to Mach 0.5 for short term emergency flows and Mach 0.2 for maximum continuous flowing.

$$d = \left(\sqrt{\frac{1.702 \cdot 10^{-5} \cdot W}{P_2 \cdot M} \cdot \left(\frac{Z \cdot T}{k \cdot MW}\right)^{0.5}}\right) \cdot 12 \qquad \textbf{Eq 5-25}$$

Sonic velocity of a gas is given by:

$$a = 223\sqrt{k\,\frac{T}{MW}} \qquad \textbf{Eq 5-26}$$

The center of the flame is assumed to be located at a distance equal to 1/3 the length of the flame from the tip.

The angle of the flame results from the vectorial addition of the velocity of the wind and the gas exit velocity.

$$\theta = \tan^{-1}\left(\frac{V_w}{V_{ex}}\right) \qquad \textbf{Eq 5-27}$$

$$V_{ex} = 550\sqrt{\frac{\Delta P}{55}} \qquad \textbf{Eq 5-28}$$

Note: API gives a greater lean angle

The coordinates of the flame center with respect to the tip are:

$$X_c = (L_f / 3)\,(\sin\theta) \qquad \textbf{Eq 5-29}$$

$$Y_c = (L_f / 3)\,(\cos\theta) \qquad \textbf{Eq 5-30}$$

The distance from any point on the ground level to the center of the flame is:

$$R = \sqrt{(X - X_c)^2 + (H_s + Y_c)^2} \qquad \textbf{Eq 5-31}$$

Equations 5-21 and 5-31 allow radiation to be calculated at any location.

The stack height results from considering the worst position vertically below the center of the flame for a given condition of gas flow and wind velocities (see Fig. 5-21).

$$R^2 = (H_s + Y_c)^2 \qquad \textbf{Eq 5-32}$$

$$R = (H_s + Y_c) \qquad \textbf{Eq 5-33}$$

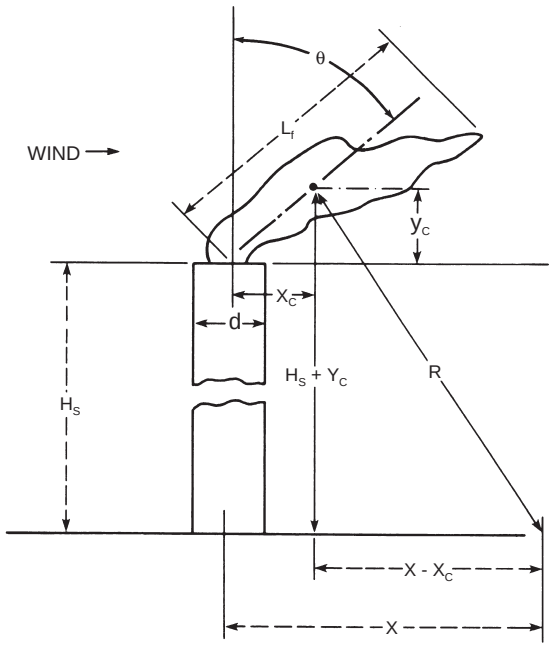
$$H_s = (R - Y_c) \qquad \textbf{Eq 5-34}$$

$$H_s = R - [(L_f / 3)\,(\cos\theta)] \qquad \textbf{Eq 5-35}$$

This method assumes that for different wind velocities the length of the flame remains constant. In reality this is not true.

FIG. 5-21

Dimensional References for Sizing a Flare Stack



Courtesy American Petroleum Institute

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When the wind blows at more than 60 miles per hour, the flame tends to shorten. For practical design, this effect is neglected.

The radiation levels commonly used for designs are: (These do not include solar heat.)

For equipment protection	5000 Btu/(hr • sq ft)
Personnel, few seconds escape only	3000 Btu/(hr • sq ft)
Personnel, one minute exposure	2000 Btu/(hr • sq ft)
Personnel, several minutes short exposure	1500 Btu/(hr • sq ft)
Personnel, continuous exposure	500 Btu/(hr • sq ft)

Equipment protection should be evaluated on a case by case basis, as various pieces of equipment have different protection needs.

Solar radiation adds to the calculated flame radiation and is dependent upon specific atmospheric conditions and site location. A typical design range is 250 to 330 Btu/(hr • sqft).

The cooling effect of fluids flowing inside piping, along with wind cooling, can be used to reduce thermal radiation effects.

Smokeless Operation

Most smokeless flares utilize outside motive forces to produce efficient gas/air mixing and turbulence from the momentum transferred by the high velocities of the external motive jet streams (steam, fuel, gas, etc.). The assist medium mass requirements are low for steam and fuel gas because of their high velocity relative to the flare gas. Typical values for steam or fuel gas are from 0.20 to 0.50 pounds of assist gas per pound of hydrocarbon flow.

Eq 5-35 predicts steam use for a given hydrocarbon molecular weight gas to be burned in a smokeless flare.

$$W_{stm} = W_{hc}\left[0.68 - \left(\frac{10.8}{MW}\right)\right] \qquad \textbf{Eq 5-36}$$

The water spray and air blower methods provide necessary mixing with low velocities and greater mass flow rates. For water spray flares, typical water rates are 1 to 5 pounds of water per pound of hydrocarbon. These rates are highly dependent upon the method of water injection and the degree of atomization of the water stream. Wind also has a significant effect on water spray flares and may greatly reduce their effectiveness. The blower assisted flare uses approximately 3 to 7 pounds of air per pound of hydrocarbon to produce smokeless operation. Forced draft from a blower assists combustion and air/gas turbulence, assisting smokeless operation. Tip velocities up to Mach 0.5 are common.

Pilots and Ignition

Reliable pilot operation under all wind and weather conditions is essential. Flaring operations are for the most part intermittent and non-scheduled. The flare must be instantly available for full emergency duty to prevent any possibility of a hazardous or environmentally offensive discharge to the atmosphere. Wind-shields and flame-retention devices may be used to ensure continuous piloting under the most adverse conditions. Most pilots are designed to operate at wind velocities of 100 miles per hour and higher.

Most flares employ a remote pilot ignition system.

Seals

An effective seal is required to prevent air from entering the flare system and forming an explosive mixture prior to the designed ignition point. Refer to API RP 521 for recommended seal methods.

Location and Regulations

Flare design must comply with local, state, and federal regulations regarding pollution, noise, and location. Permits are usually required prior to construction.

APPLICABLE CODES, STANDARDS, AND RECOMMENDED PRACTICES

The designers of relief systems should be familiar with the following documents related to pressure relief valves in process plants and natural-gas systems.

ASME Codes — ASME Boiler and Pressure Vessel Code Section I — Paragraphs PG-67 through PG-73, and Section VIII — Paragraphs UG-125 through UG-136 and appendices M and 11.

In the "scope", certain vessels are excluded from ASME Code requirements, including all vessels under 15 psig operating pressure.

ANSI Codes — ANSI B31.1 Power Piping.

ANSI B31.3 — Chemical Plant and Petroleum Refining Piping

ANSI B31.8 — Gas Transmission & Distribution Systems.

API Publications — API RP 520 Part I, Design — This API design manual is widely used for fire sizing of relief valves on both liquid and gas filled vessels: (a) liquid vessels — Section 4, 5, and 6; (b) gas-filled vessels — Appendix C.3. This RP covers only vessels above 15 psig.

API RP 520 Part II, Installation — This includes: (a) recommended piping practices, (b) calculation formulae for reactive forces, and (c) precautions on preinstallation handling and dirt.

API RP 521 Guide for Pressure - Relieving and Depressuring Systems, September, 1982, 2nd Edition — An excellent document on everything from causes of overpressure through flare stacks.

API Guide for Inspection of Refinery Equipment — Chapter XVI Pressure Relieving Devices — Gives (a) guide for inspection of record keeping, and (b) frequency of inspection, paragraph 1602.03.

API Std. 526 — Flanged Steel Safety Relief Valves for Use in Petroleum Refineries — Gives industry standards as to dimensions, pressure-temperature ratings, maximum set pressures, body materials.

API Std. 527 — Commercial Seat Tightness of Safety-Relief Valves with Metal-to-Metal Seats — Permissible leakage of conventional and bellows valves and testing procedure.

API Std. 528 — Standard for Safety Relief Valve Nameplate Nomenclature — Standard covering information that should go on the nameplate of a safety relief valve.

API Std. 620 — Recommended Rules for Design and Construction of Large Welded-Low-Pressure Storage Tanks — Covers tanks at less than 15 psig. Section 6 of this document gives recommendations on relief valve types and sizing.

API Std. 2000 — Venting Atmospheric and Low-pressure Storage Tanks (Non-Refrigerated and Refrigerated) — Covers

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tanks at less than 15 psig. Capacity requirement calculations for both pressure and vacuum. Also fire-sizing method for low pressure tanks.

API Std. 2508 — Design and Construction of Ethane and Ethylene Installations at Marine and Pipeline Terminals, Natural Gas Processing Plants, Refineries, Petrochemical Plants, and Tank Farms — Covers the design, construction, and location of refrigerated (including autorefrigerated) liquefied ethane and ethylene installations, which may be associated with one or more of the following: railroad, truck, pipeline stations, or marine loading or unloading racks or docks.

API Std. 2510 — Design and Construction of LPG Installations at Marine and Pipeline Terminals, Natural-Gas-Processing Plants, Refineries, Petrochemical Plants, and Tank Farms.

API Bulletin 2521 — Use of Pressure Vacuum Vent Valves for Atmospheric Pressure Tanks to Reduce Evaporation Loss.

NFPA Publications — NFPA 58 and 30 — These cover storage and handling of liquefied petroleum gases. Section B.10 covers safety devices installation and testing.

NFPA 59 — Covers LP-gas, Utility Plants

NFPA 59A — Covers LN Gas, Storage and Handling

OSHA Publications — OSHA Title 29, Part 1910 - Part 1910 includes handling, storage, and safety requirements for LPG and ammonia.

CGA (Compressed Gas Association) Publications — Series of standards covering transportation, handling, and storage of compressed gases including:

Pamphlet S-1.2 Safety Relief Device Standards

Part 2: Cargo and portable tanks for compressed gases.

Pamphlet S-1.3 Safety Relief Service Standards

Part 3: Compressed Gas Storage Containers.

REFERENCES

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2. ASME Boiler and Pressure Vessel Code, Section I and Section VIII, American Society of Mechanical Engineers (ASME), New York, NY.
3. ANSI B31.3 — Chemical Plant and Petroleum Refinery Piping, The American Society of Mechanical Engineers (ASME), 345 East 47th Street, New York, NY 10017.
4. ANSI B31.8 — Gas Transmission and Distribution Piping, The American Society of Mechanical Engineers (ASME), 345 East 47th Street, New York, NY 10017.
5. API Standard 527 — Commercial Seat Tightness of Safety Relief Valves with Metal-to-Metal Seats, American Petroleum Institute, 1220 L Street, NW, Washington, DC 20005.
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5. Straitz III, J. F., "Flaring for Safety and Environmental Protection", Drilling-DCW, November 1977.
6. Straitz III, J. F., "Make the Flare Protect the Environment", Hydrocarbon Processing, October 1977.
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SECTION 6

Storage

This section provides general guidelines that will aid in the selection of the proper type of storage to be used in a particular application. Fig. 6-2 will assist with the selection. Various codes, standards, and recommended practices should be used

FIG. 6-1

Nomenclature

A′, B′, C′, D′	=	coefficients used in Fig. 6-14
A	=	surface area, ft^2
b	=	ellipse minor radius, ft
B_{max}	=	vapor pressure of liquid at maximum surface temperature, psia
B_{min}	=	vapor pressure of liquid at minimum surface temperature, psia
D	=	cylinder diameter, ft
f(Zc)	=	cylinder partial volume factor, dimensionless
f(Ze)	=	head partial volume factor, dimensionless
H	=	correction factor for horizontal surfaces
H_n	=	depth of liquid in cylinder, ft
H_p	=	height of liquid, ft
H_T	=	height, ft
k	=	thermal conductivity, Btu/[(hr • sq ft • °F)/in.]
K	=	equilibrium constant, y/x, dimensionless
K_l	=	head coefficient, dimensionless
L	=	length, ft
MW_i	=	molecular weight of component i, lb/lb mole
n_g	=	number of moles of vapor
n_i	=	number of moles of component i
P	=	absolute pressure, psia
P_a	=	atmospheric pressure, psia
P_c	=	critical pressure, psia
P_R	=	reduced pressure, dimensionless
Q	=	heat flow, Btu/sq ft • hr
R	=	gas constant, 10.73 psia • ft^3/(R • lb mole)
R_l	=	cylinder radius, ft
R_i	=	thermal resistance of insulation (X/k), (hr • sq ft • °F)/Btu
T	=	temperature, °R
T_a	=	ambient air temperature, °F
T_c	=	critical temperature, °R or °F
T_f	=	temperature drop through surface air film, °F
T_h	=	hot face temperature, °F
T_i	=	temperature drop through insulation, °F
T_m	=	mean temperature of insulation, °F
T_{max}	=	maximum average temperature, °F
T_{min}	=	minimum average temperature, °F
T_R	=	reduced temperature, dimensionless
T_s	=	outside surface temperature, °F
V	=	volume, ft^3
W	=	width, ft
x_i	=	mole fraction of component i in the liquid phase
X	=	insulation thickness, in.
y_i	=	mole fraction of component i in the vapor phase
Z	=	compressibility factor

Greek

α	=	radians
Δ	=	absolute internal tank pressure at which vacuum vent opens, psia
π	=	3.14159...
ϕ	=	required storage pressure, psia
Σ	=	summation
DP	=	design pressure is the pressure at which the most severe condition of coincident pressure and temperature expected during normal operation is reached. For this condition, the maximum difference in pressure between the inside and outside of a vessel or between any two chambers of a combination unit shall be considered. (ASME Code for Unfired Pressure Vessels, Section VIII)
MAWP	=	maximum allowable working pressure shall be defined as the maximum positive gauge pressure permissible at the top of a tank when in operation, which is the basis for the pressure setting of the safety-relieving devices on the tank. It is synonymous with the nominal pressure rating for the tank as referred to in API Standards 620 and 650.
OP	=	operating pressure is the pressure at which a vessel normally operates. It shall not exceed the maximum allowable working pressure of the vessel. A suitable margin should be allowed between the pressure normally existing in the gas or vapor space and the pressure at which the relief valves are set, so as to allow for the increases in pressure caused by variations in the temperature or gravity of the liquid contents of the tank and other factors affecting the pressure in the gas or vapor space. (API Standard 620)
RVP	=	Reid Vapor Pressure is a vapor pressure for liquid products as determined by ASTM test procedure D-323. The Reid vapor pressure is defined as pounds per sq in. at 100°F. The RVP is always less than the true vapor pressure at 100°F.
TVP	=	true vapor pressure is the pressure at which the gas and liquid in a closed container are in equilibrium at a given temperature.

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FIG. 6-2

Storage

	Atmospheric Pressure†‡	0 to *2.5 psig†‡	2.5 to 15 psig‡	Above 15 psig§	Underground
Crude Oils	X	X	X	–	X
Condensate	X	X	X	X	X
Oils	X	X	–	–	X
Natural Gasoline	X	X	X	–	X
Butanes	–	X°	X°	X	X
Propane	–	X°	X°	X	X
Raw NGLs	–	X°	X°	X	X
Ethane	–	X°	X°	X	X
Petrochemicals	–	X°	X°	X	X
Natural Gas	–	–	–	X	X
LNG	–	X°	X°	X	–
Treating Agents	X	X	–	–	–
Dehydration Fluids	X	X	–	–	–
Specialty Chemicals	X	X	X	–	–
Solid Materials	X	–	–	–	–
Water	X	–	–	–	–

* Some materials may require a slight positive pressure to exclude air, oxygen, and/or water, and conserve valuable/toxic vapors.

† API Standard 650 governs

‡ API Standard 620 governs

§ ASME Unfired Pressure Vessel Code, Section VIII governs

° Refrigerated only

Note: Vacuum conditions may exist and must be considered in tank design. Examples: low ambient temperatures or evacuating without relieving.

to supplement the material provided. Manufacturers should be consulted for specific design information pertaining to a particular type of storage.

STORAGE CLASSIFICATION

Above Ground

Atmospheric — Atmospheric pressure tanks are designed and equipped for storage of contents at atmospheric pressure. This category usually employs tanks of vertical cylindrical configuration that range in size from small shop welded to large field erected tanks. Bolted tanks, and occasionally rectangular welded tanks, are also used for atmospheric storage service.

Low Pressure (0 to 2.5 psig) — Low pressure tanks are normally used in applications for storage of intermediates and products that require an internal gas pressure from close to atmospheric up to a gas pressure of 2.5 psig. The shape is generally cylindrical with flat or dished bottoms and sloped or domed roofs. Low pressure storage tanks are usually of welded design. However, bolted tanks are often used for operating pressures near atmospheric. Many refrigerated storage tanks operate at approximately 0.5 psig.

Medium Pressure (2.5 to 15 psig) — Medium pressure tanks are normally used for the storage of higher volatility intermediates and products that cannot be stored in low pressure tanks. The shape may be cylindrical with flat or dished bottoms and sloped or domed roofs. Medium pressure tanks are usually of welded design. Welded spheres may also be used, particularly for pressures at or near 15 psig.

High Pressure (Above 15 psig) — High pressure tanks are generally used for storage of refined products or fractionated components at pressure above 15 psig. Tanks are of welded design and may be of cylindrical or spherical configuration.

Underground

Gas processing industry liquids may be stored in underground, conventionally mined or solution mined caverns. No known standard procedures are available for this type storage; however, there are many publications and books covering the subject in detail.

WORKING PRESSURES

A design working pressure can be determined to prevent breathing, and thereby save standing storage losses. However, this should not be used in lieu of any environmental regulatory requirements regarding the design of storage tanks. The environmental regulatory requirements for the specific location should be consulted prior to the design of storage facilities. Generally there are regulatory requirements specifying the type of storage tank to be used, based on the storage tank

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capacity and the vapor pressure of the product being stored. In addition there are usually specific design requirements, for example in the type of seals to be used in a floating roof tank.

The working pressure required to prevent breathing losses depends upon the vapor pressure of the product, the temperature variations of the liquid surface and the vapor space, and the setting of the vacuum vent.

$$\phi = B_{max} + (\Delta - B_{min}) \frac{(T_{max} + 460)}{(T_{min} + 460)} - P_a \qquad \textbf{Eq 6-1}$$

The above relation holds only when B_{min} is less than Δ; that is, when the minimum vapor pressure is so low that air is admitted into the vapor space through the vacuum vent. When B_{min} is equal to or greater than Δ, the required storage pressure is,

$$\phi = B_{max} - P_a \qquad \textbf{Eq 6-2}$$

Under this condition air is kept out of the vapor space.

Fig. 6-3 is presented as a general guide to storage pressures for gasolines of various volatilities in uninsulated tanks. These data for plotting the curves were computed from Eqs 6-1 and 6-2 using the following assumptions:

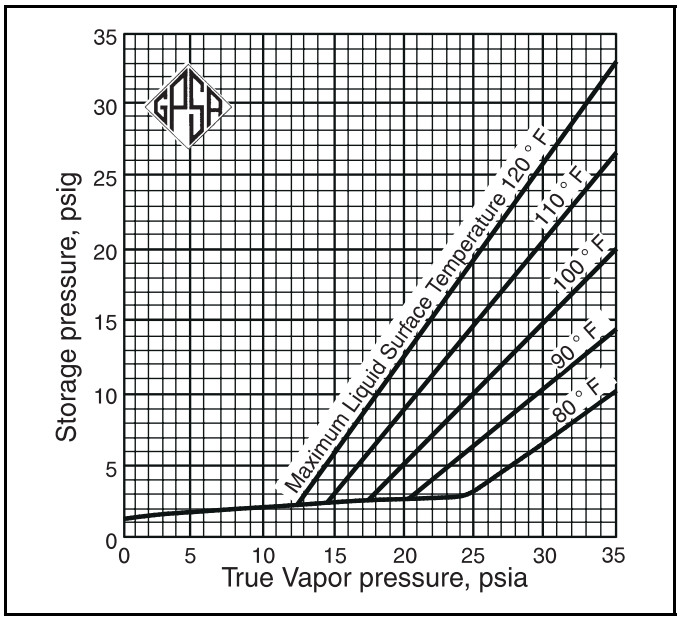
- Minimum liquid surface temperature is 10°F less than the maximum liquid surface temperature.
- Maximum vapor space temperature is 40°F greater than the maximum liquid surface temperature.
- Minimum vapor space temperature is 15°F less than the maximum liquid surface temperature.
- Stable ambient conditions (ambient temp. 100°F).

These temperature variations are far greater than would be experienced from normal night to day changes. Therefore, the lower, nearly horizontal line, which shows a required storage pressure of 2.5 psig for the less volatile gasolines is conservative and allows a wide operating margin.

Maximum liquid surface temperatures vary from 85 to 115°F. Sufficient accuracy will generally result from the assumption that it is 10°F higher than the maximum temperature of the body of the liquid in a tank at that location.

FIG. 6-3

Storage Pressure vs. True Vapor Pressure



Example 6-1 — To illustrate the use of Fig. 6-3, suppose a 24 psia true vapor pressure (TVP) natural gasoline is to be stored where the liquid surface temperature may reach a maximum of 100°F. A vertical line extended upward from the 24 psia mark at the bottom of the chart intersects the 100°F line at 9.3 psig. The design pressure of the tank should be a minimum of 10.23 psig (9.3 psig + 10%).

Fig. 6-4 can be used as follows:

- As quick reference to determine true vapor pressures of typical LPGs, natural gasolines, and motor fuel components at various temperatures.
- To estimate the operating pressure of a storage tank necessary to maintain the stored fluid in a liquid state at various temperatures.
- For converting from true vapor pressure to Reid Vapor Pressure (RVP).
- For simple evaluation of refrigerated storage versus ambient temperature storage for LPGs.

Example 6-2 — Determine the TVP of a 12 psi RVP gasoline. In addition, estimate the design pressure of a tank needed to store this same 12 RVP gasoline at a maximum temperature of 120°F. Using Fig. 6-4, a vertical line is extended upwards from the 100°F mark (100°F is used as the reference point for determining RVP) at the bottom of the chart to the intersection of the 12 psi RVP line, read true vapor pressure of 13.2 psia. A vertical line is also extended from the 120°F mark to intersect the 12 RVP gasoline line. Now going horizontal, the true vapor pressure axis is crossed at approximately 18.1 psia. The storage tank should therefore be designed to operate at 18.1 psia (3.4 psig) or above. The design pressure of the tank should be a minimum of 10% above the operating gauge pressure or approximately 18.5 psia.

Example 6-3 — Evaluate the options of refrigerated storage versus ambient temperature storage for normal butane. From Fig. 6-4 the vertical line is extended up from the 100°F (assumed maximum) mark to intersect the normal butane line at approximately 51.5 psia (36.8 psig). The working pressure of the tank should be 36.8 psig plus a 10% safety factor, or 55.2 psia. This same product could be stored in an atmospheric pressure tank if the product is chilled to 32°F. This temperature is determined by following the normal butane line down until it intersects the 14.7 psia horizontal vapor pressure line. Reading down to the bottom scale indicates the storage temperature at 32°F. The pressurized tank would require more investment due to the higher working pressure of 55.2 psia (40.5 psig) and the thicker shell requirement. The refrigerated tank would require less investment for the tank itself, but an additional investment would be necessary for insulation and for refrigeration equipment which requires additional operating expenses. The economics of each type of storage system can be evaluated to determine which will be the most attractive.

The graphical method of converting from RVP to TVP is an approximation and is generally more accurate for lighter components. Crude oils with very low RVPs could vary significantly from this graphical approach. This is due to the fact that during the Reid test the highest vapor pressure materials tend to evaporate leaving a residue which has a lower vapor pressure than the original sample. Equation 6-3 was developed by A. Kremser in 1930 to relate the two vapor pressures at 100°F.

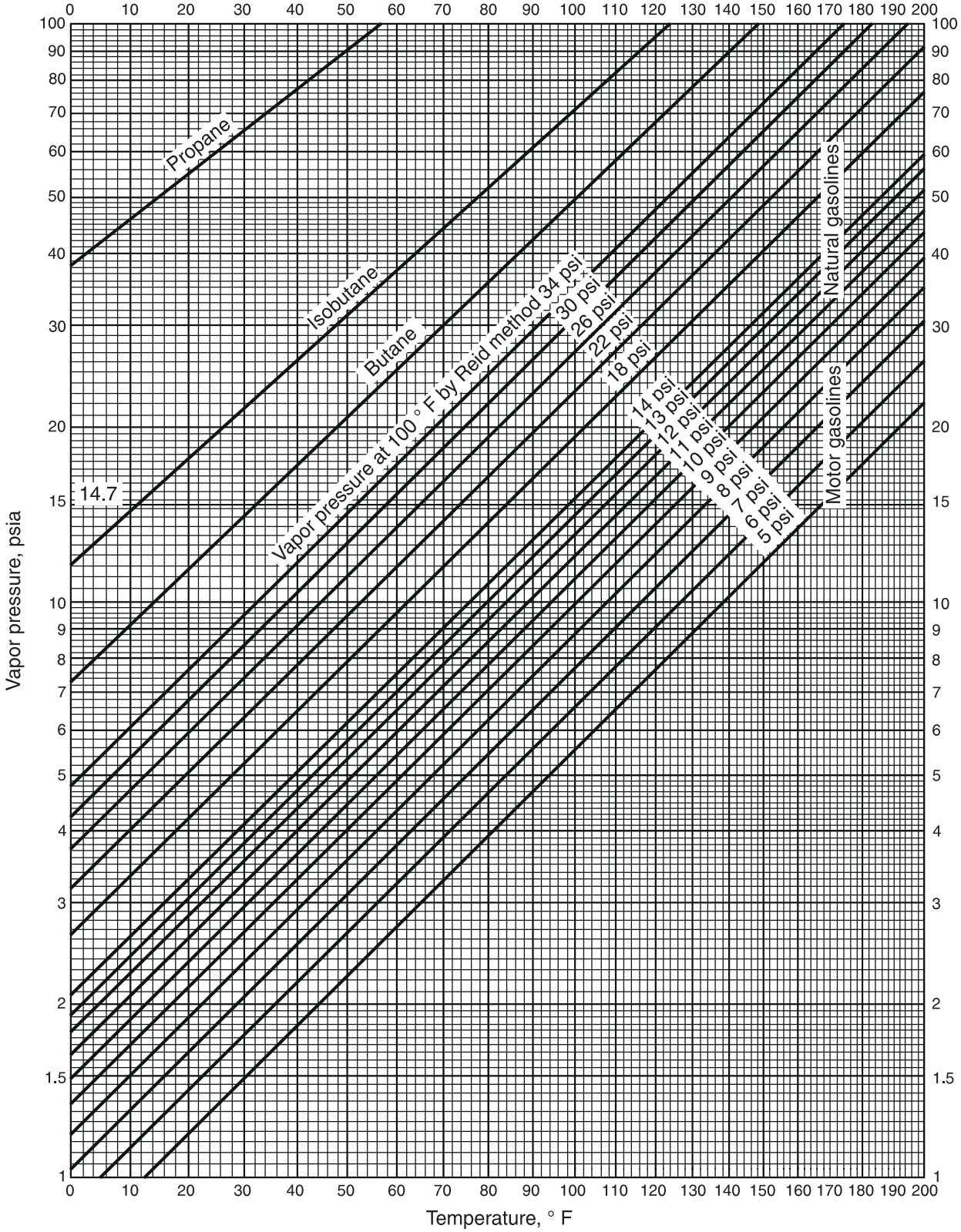
$$TVP = (1.07)(RVP) + 0.6 \qquad \textbf{Eq 6-3}$$

Using this formula for the 12 psi RVP gasoline example would calculate a 13.4 psia TVP versus the 13.2 determined

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FIG. 6-4

True Vapor Pressures vs. Temperatures for Typical LPG, Motor, and Natural Gasolines



CAUTION: This chart cannot be used to determine crude oil vapor pressure. Refer to API Technical Data Book Fig. 5B1.2

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graphically. The RVP is less than the true vapor pressure at 100°F. Published data indicate the ratio of true vapor pressure to Reid vapor pressure may vary significantly, depending on the exact composition of the stored liquids. Ratios from 1.03 to 1.60 have been verified by test data.[1,2] Before entering the final design phase of any storage project, test data should be gathered on the fluid to be stored.

TYPES OF STORAGE

Above Ground

For operating pressures above 15 psig, design and fabrication are governed by the ASME Code, Section VIII.

Spheres — Spherical shaped storage tanks are generally used for storing products at pressures above 5 psig.

Spheroids — A spheroidal tank is essentially spherical in shape except that it is somewhat flattened. Hemispheroidal tanks have cylindrical shells with curved roofs and bottoms. Noded spheroidal tanks are generally used in the larger sizes and have internal ties and supports to keep shell stresses low. These tanks are generally used for storing products above 5 psig.

Horizontal Cylindrical Tanks — The working pressure of these tanks can be from 15 psig to 1000 psig, or greater. These tanks often have hemispherical heads.

Fixed Roof — Fixed roofs are permanently attached to the tank shell. Welded tanks of 500 barrel capacity and larger may be provided with a frangible roof (designed for safety release of the welded deck to shell joint in the event excess internal pressure occurs), in which case the design pressure shall not exceed the equivalent pressure of the dead weight of the roof, including rafters, if external.

Floating Roof — Storage tanks may be furnished with floating roofs (Fig. 6-6) whereby the tank roof floats upon the stored contents. This type of tank is primarily used for storage near atmospheric pressure. Floating roofs are designed to move vertically within the tank shell in order to provide a constant minimum void between the surface of the stored product and the roof. Floating roofs are designed to provide a constant seal between the periphery of the floating roof and the tank shell. They can be fabricated in a type that is exposed to the weather or a type that is under a fixed roof. Internal floating roof tanks with an external fixed roof are used in areas of heavy snowfalls since accumulations of snow or water on the floating roof affect the operating buoyancy. These can be installed in existing tanks as well as new tanks.

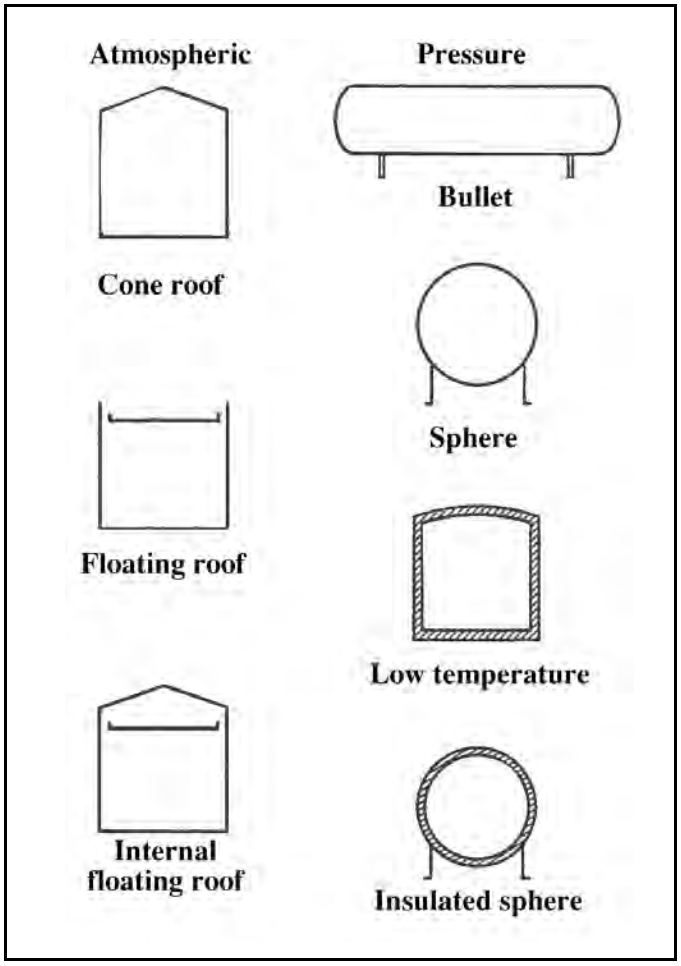
Both floating roofs and internal floating roofs are utilized to reduce vapor losses and aid in conservation of stored fluids.

Environmental rules for new equipment restrict cone-roof tanks without vapor-recovery facilities to materials having a true vapor pressure at the tank temperature of less than 1.5 psia and floating-roof tanks to materials of less than 11.1 psia.

Above 11.1 psia, a pressure vessel or vapor recovery scheme is mandatory. These rules should be considered minimum requiremens. Toxic or odoriferous materials will need better emission control than a floating-roof tank can provide. Local conditions sometimes call for such strict hydrocarbon emission limits that the use of floating-roof tanks comes under question. See Fig. 6-5 for common storage vessels.

FIG. 6-5

Seven Common Storage-Vessels For Low- and High-Pressure Services



Bolted — Bolted tanks are designed and furnished as segmental elements which are assembled on location to provide complete vertical, cylindrical, above ground, closed and open top steel storage tanks. Standard API bolted tanks are available in nominal capacities of 100 to 10,000 barrels, designed for approximately atmospheric internal pressures. Bolted tanks offer the advantage of being easily transported to desired locations and erected by hand. To meet changing requirements for capacity of storage, bolted tanks can be easily dismantled and re-erected at new locations.

Specialty — Pipe Storage (Fig. 6-7) — Pipe that is used specifically for storing and handling liquid petroleum components or liquid anhydrous ammonia must be designed and constructed in accordance with any applicable codes.

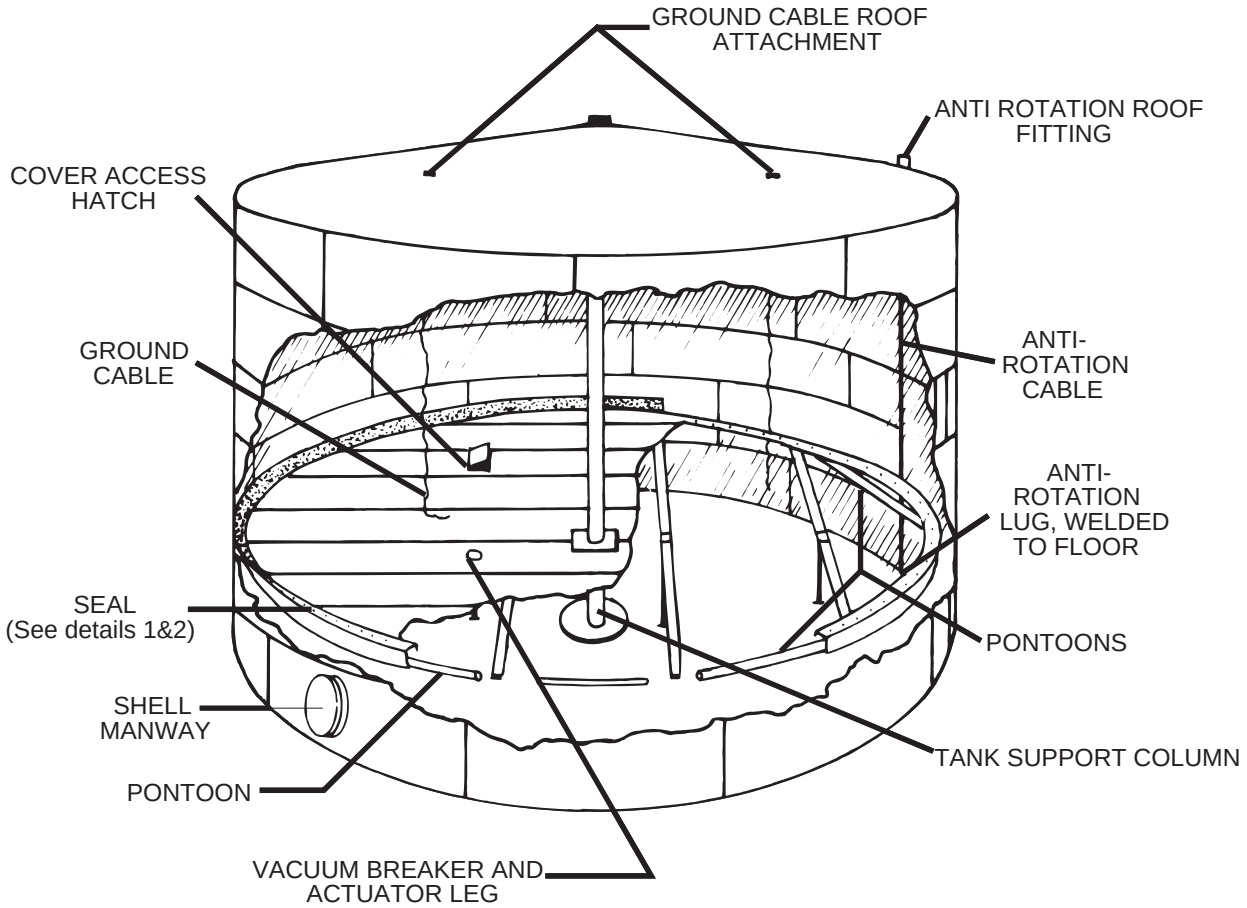
Flat-Sided Tanks — Although cylindrical shaped tanks may be structurally best for tank construction, rectangular tanks occasionally are preferred. When space is limited, such as offshore, requirements favor flat-sided tank construction because several cells of flat-sided tanks can be easily fabricated and arranged in less space than other types of tanks. Flat-sided or rectangular tanks are normally used for atmospheric type storage.[1]

Lined Ponds[2] — Ponds are used for disposal, evaporation, or storage of liquids. Environmental considerations may pre-

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FIG. 6-6

Typical Arrangement of Internal Floating Roof Tank

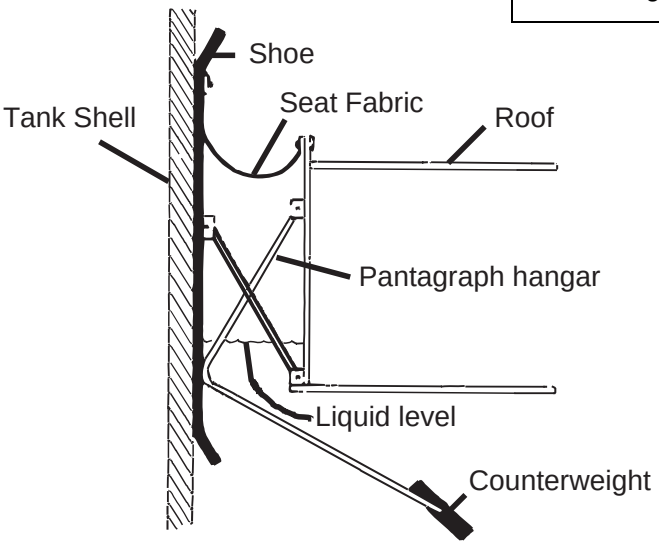


(Also applicable to external floating roof tanks if dome roof is removed)

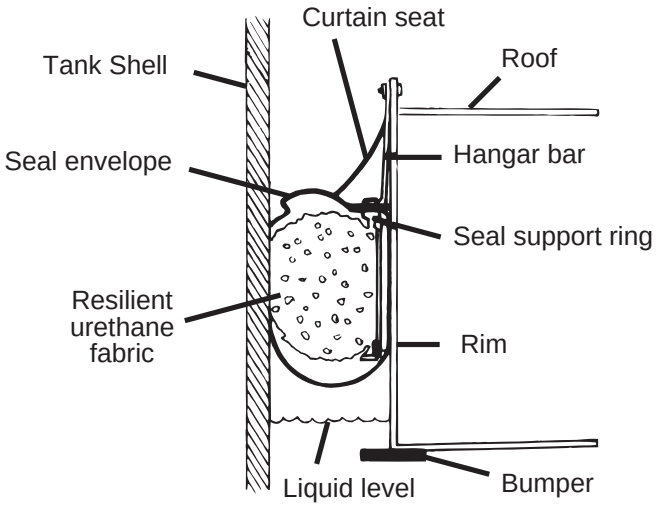
Courtesy of CE Natco

Example Types of Seals for Floating Roof Tanks

Detail 1 - Shoe seal



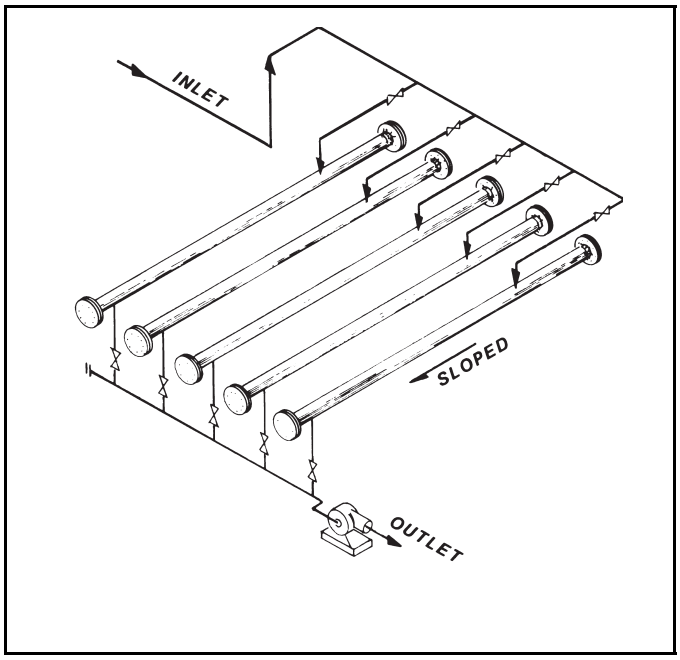
Detail 2 - Tube Seal



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FIG. 6-7

Pipe Storage



clude the use of lined ponds for the storage of more volatile or toxic fluids. Linings are used to prevent storage liquid losses, seepage into the ground, and possible ground water contamination. Clay, wood, concrete, asphalt, and metal linings have been used for many years. More recently, a class of impervious lining materials has been developed that utilize flexible synthetic membranes. Commonly used lining materials are polyvinyl chloride, natural rubber, butyl rubber, and Hypalon®. Polyethylene, nylons, and neoprenes are used to a lesser extent.

Some of the most important qualities of a suitable liner are:

- High tensile strength and flexibility.
- Good weatherability.
- Immunity to bacterial and fungus attack.
- Specific gravity greater than 1.0.
- Resistance to ultraviolet-light attack.
- Absence of all imperfections and physical defects.
- Easily repaired.

Leak detection sometimes must be built into the pond system, especially where toxic wastes or pollutants are to be stored. Types of leak-detection systems that are commonly used are underbed (French) drainage system, ground resistivity measurement, and monitor wells, and any combination thereof.

Pit Storage — Pit storage is similar to pond storage but is only used on an emergency basis. The use of this type of storage is limited by local, state, and federal regulations.

Underground

Underground storage is most advantageous when large volumes are to be stored. Underground storage is especially advantageous for high vapor pressure products.

Types of underground storage are: (1) caverns constructed in salt by solution mining or conventional mining, (2) caverns constructed in nonporous rock by conventional mining, and (3) caverns developed by conversion of depleted coal, limestone, or salt mines to storage.

Solution Mined Caverns — The cavern is constructed by drilling a well or wells into the salt and circulating low salinity water over the salt interval to dissolve the salt as brine.

The cavern may be operated by brine displacement of product, pumpout methods, vapor displacement, or as in the case of gas, by product expansion (see Figs. 6-8, 6-9, and 6-10).

Most solution mined caverns are operated using the brine displacement technique (Fig. 6-8). A suspended displacement string of casing is installed near the bottom of the cavern and product is injected into the annulus between the product casing (casing cemented at cavern roof) and the displacement casing, forcing brine up the displacement casing. The procedure is reversed for product recovery. In this type of operation, a brine storage reservoir is usually provided. Detail 1 of Fig. 6-8 provides the typical piping for the wellhead of an underground storage well.

Some solution mined caverns are operated “dry” by installing a pump at cavern depth either within the cavern or in a well connected to the cavern. Both submersible electric driven pumps and line shaft pumps (deep well vertical turbine pumps) are used for this purpose (see Fig. 6-9).

Conventional Mined Caverns — Conventional mined caverns can be constructed any place a nonporous rock is available at adequate depth to withstand product pressures. An engineer or geologist experienced in underground storage should evaluate any specific site for the feasibility of constructing underground storage. Most product caverns are constructed in shale, limestone, dolomite, or granite. This type cavern is operated “dry” (product recovered by pumping).

Refrigerated Storage

The decision to use refrigerated storage in lieu of pressurized storage is generally a function of the volume of the liquid to be stored, the fill rate, the physical and thermodynamic properties of the liquid to be stored, and the capital investment and operating expenses of each type of system.

The parameters involved in selecting the optimum refrigerated storage facility are:

- Quantity and quality of product to be stored.
- Fill rate, temperature, and pressure of incoming stream.
- Shipping conditions for the product.
- Composition of the product.
- Cooling media (air, water, etc.) available.
- Availability and cost of utilities.
- Load bearing value of soil.

The proper choice of storage and the proper integration of the storage facility with the refrigeration facilities are important to overall economy in the initial investment and operating costs. Fig. 6-11 provides some general guidelines to use when selecting a storage system for propane.

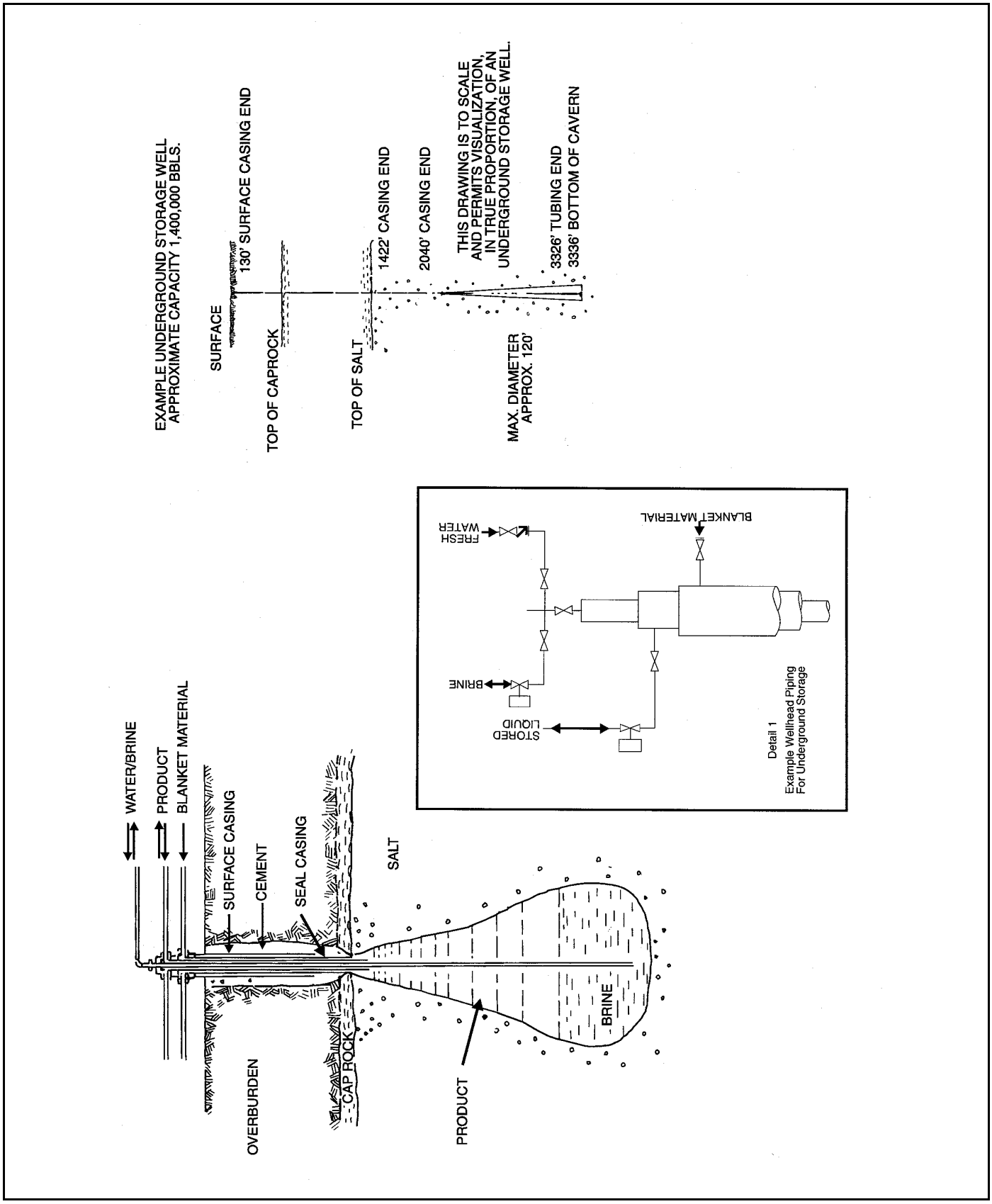
When using refrigerated storage, the liquid to be stored is normally chilled to its bubble point temperature at atmospheric pressure. Refrigerated storage tanks normally operate at an internal pressure between 0.5 and 2.0 psig.

In some cases, pressurized-refrigerated storage is attractive. In this type of refrigerated storage, the product to be stored is chilled to a temperature that allows it to be stored at

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FIG. 6-8

Brine Displacement Cavern Operation (Solution Mined Cavern)



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FIG. 6-9

Pump-Out Cavern Operation (Fracture Connected Solution Mined Cavern in Bedded Salt)

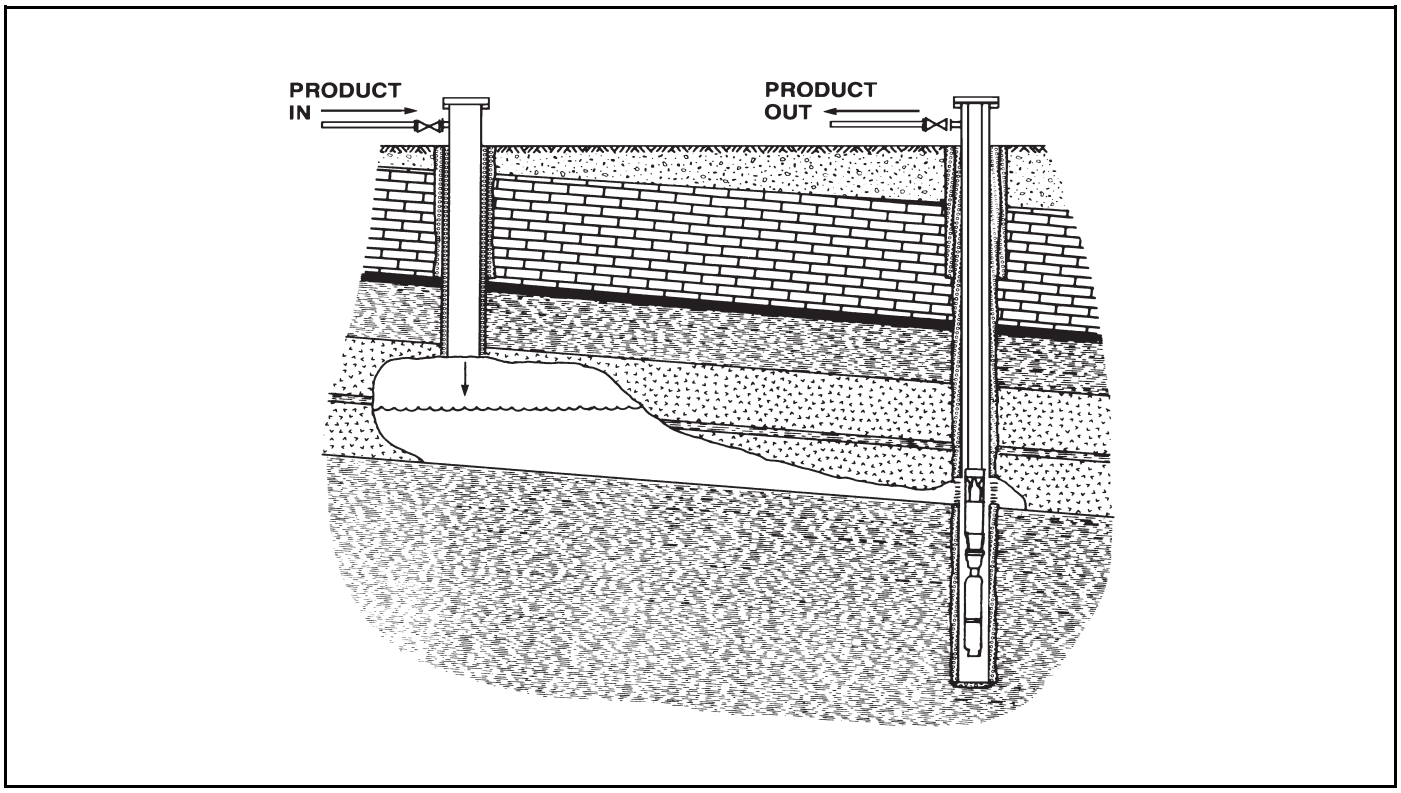
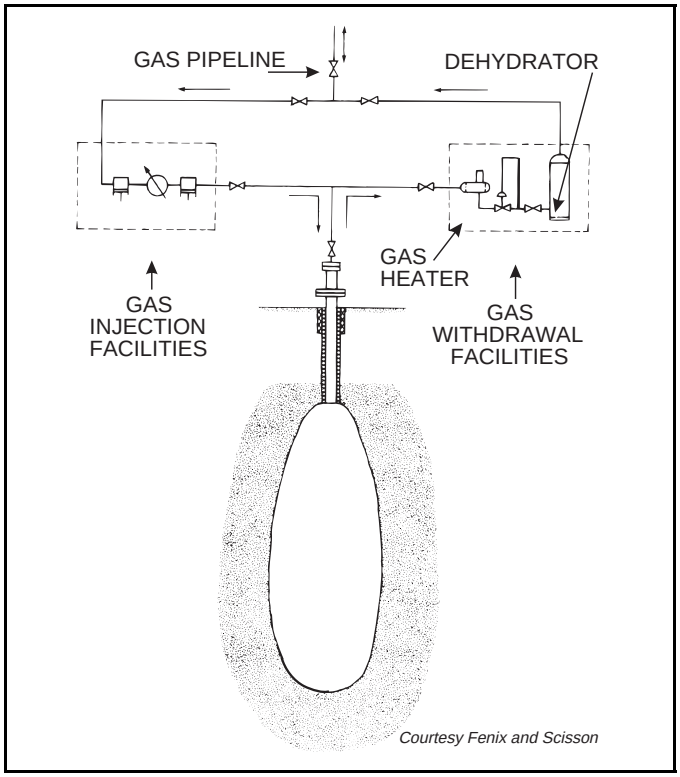


FIG. 6-10

Compression/Expansion Cavern Operation (Solution Mined Cavern)



a pressure somewhere between atmospheric pressure and its vapor pressure at ambient temperature.

Refrigeration requirements normally include the following basic functions:

- Cooling the fill stream to storage temperature.
- Reliquefying product vaporized by heat leak into the system.
- Liquefying vapors displaced by the incoming liquid.

Other factors which should be considered are:

- Pump energy requirements
- Barometric pressure variations
- Product compositions
- Non-condensables
- Solar radiation effects
- Superheated products

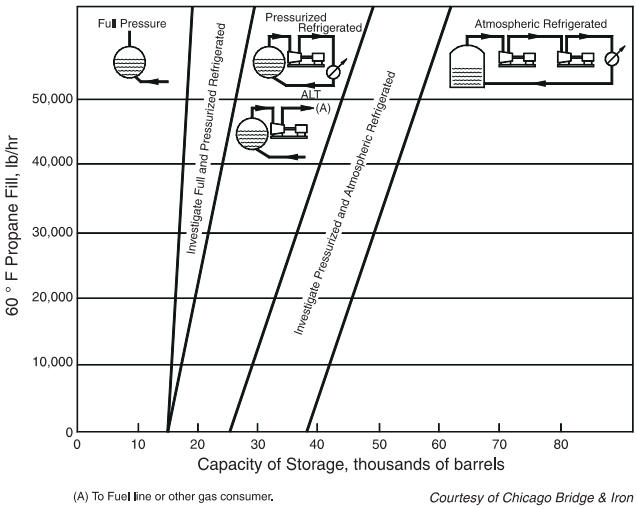
Refer to Section 14 of this Data Book for information on refrigeration. Tables R.2.2, R.2.3, and R.2.4 of API 620, Appendix R, should be consulted for specific service temperatures and impact requirements of materials used as primary and secondary components in refrigerated storage tanks. Refrigerated facilities require specialized insulation systems, which are described later in this Section.

Foundations for the various types of low temperature storage vessels are designed much the same as foundations for ordinary spheres and pressure cylinders. One caution must be noted. Most low temperature liquids are lighter than water and the vessels are designed to store this lighter liquid. Therefore, it is a common practice to design foundations for the total weight of contained product and to water test the vessel at 1.25 times the product weight.

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FIG. 6-11

General Guidelines for the Economic Storage of Pure Pro-



Flat bottom vessel foundations in low temperature service present an additional problem. The container is a heat-sink and, if no provision is made to supply heat, a large quantity of soil eventually will reach temperatures below the freezing point of water. Moisture in the sub-soil will freeze and some "heaving" could occur. A heat source consisting of electrical resistance heating cable or pipe coils with a warm circulating liquid is generally installed below the outer tank bottom to maintain the soil temperature above 32°F. Foundations for low temperature vessels must also be designed to minimize differential settling.

Liquids at low temperatures can be stored in frozen earth caverns at essentially atmospheric or very low pressures. An excavated hole (usually lined) is capped by an insulated metal dome and refrigerated to maintain impervious "walls of ice." Vapors from the liquid are continuously recompressed and condensed.

DEFINITIONS

The following definitions of tank containment are from BS7777. Extracts from BS 7777 Part 1: 1993 reproduced with permission from BSI under license number 2002NC0111. BSI publications can be obtained from BSI Customer Services, 389 Chiswick High Road, London W4 4AL (Tel +44 (0) 20 8996 9001).

Single Containment Tank

Either a single tank or a tank comprising an inner tank and an outer container designed and constructed so that only the inner tank is required to meet the low temperature ductility requirements for storage of the product.

The outer container (if any) of a single containment storage tank is primarily for the retention and protection of insulation and to constrain the vapor purge gas pressure, but is not designed to contain refrigerated liquid in the event of leakage from the inner tank.

A single containment tank is normally surrounded by a low bund wall (Fig. 6-12) to contain any leakage.

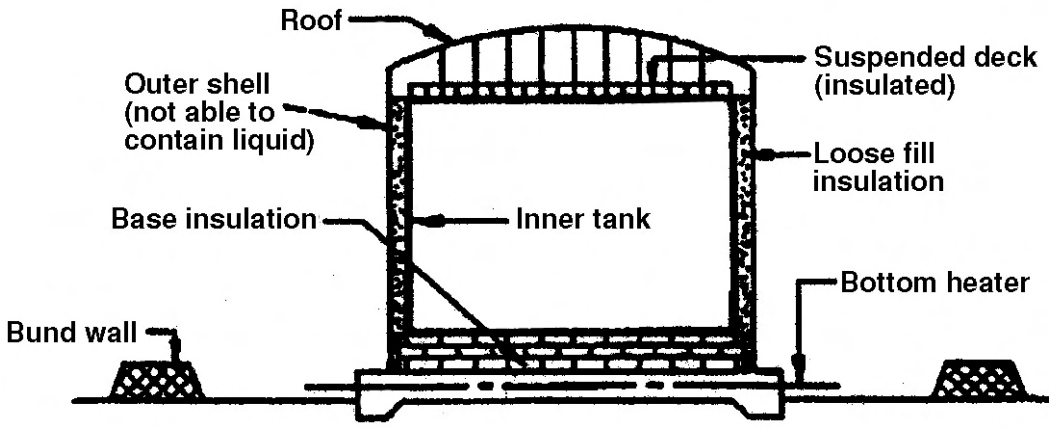
Double Containment Tank

A double tank designed and constructed so that both the inner tank and the outer tank are capable of independently containing the refrigerated liquid stored. To minimize the pool of escaping liquid, the outer tank or wall is located at a distance not exceeding 6 m from the inner tank.

The inner tank contains the refrigerated liquid under normal operating conditions. The outer tank or wall is intended to contain the refrigerated liquid product leakage from the

FIG. 6-12

Single Containment Tank

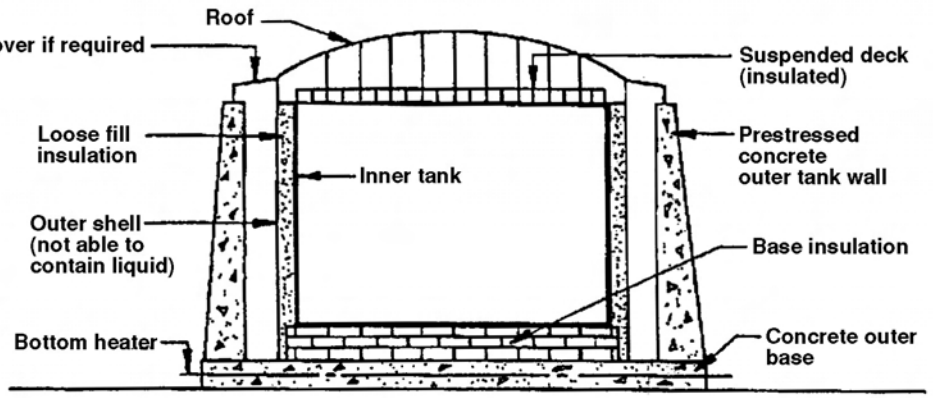


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FIG. 6-13

Double Containment Tank



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inner tank, but it is not intended to contain any vapor resulting from product leakage from the inner tank (Fig. 6-13).

Full Containment Tank

A double tank designed and constructed so that both the inner tank and the outer tank are capable of independently containing the refrigerated liquid stored. The outer tank or wall should be 3 feet to 6 feet distant from the inner tank.

The inner tank contains the refrigerated liquid under normal operating conditions. The outer roof is supported by the outer tank. The outer tank is intended to be capable both of containing the refrigerated liquid and of controlled venting of the vapor resulting from product leakage after a credible event (Fig. 6-14).

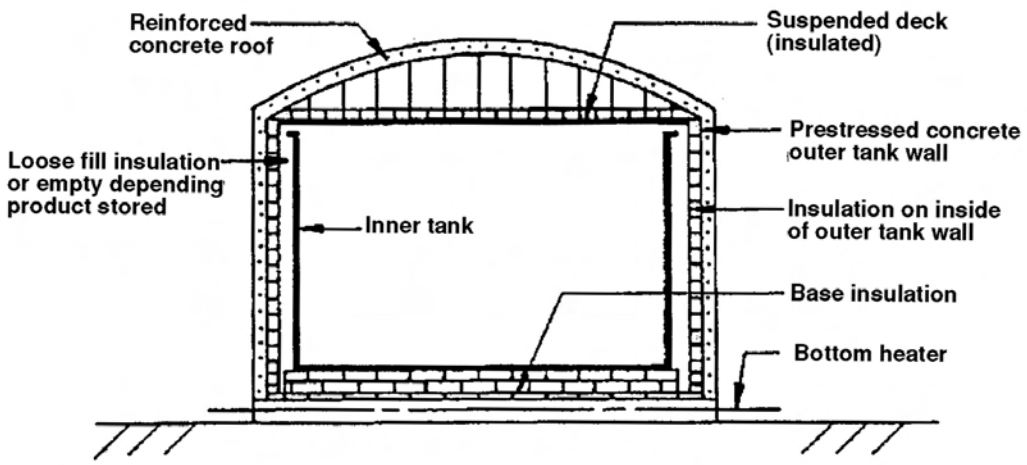
MATERIALS OF CONSTRUCTION

Vessel/Tank Materials

Metallic — Shop welded, field welded, and bolted storage tanks are customarily fabricated from mild quality carbon steel. Most common for welded tanks are A-36 structural steel and A-283 grade "C" structural quality carbon steel. Sheet gauge steels for bolted tanks are of commercial quality having a minimum tensile strength of 52,000 psi. A-612, A-515, and A-516 mild quality low carbon steels are used for fabricating the higher pressure storage products such as spheres and "bullets." Various API and ASME Codes (listed in the References) to which the storage tank is fabricated, set forth the welding

FIG. 6-14

Full Containment Tank



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procedures, inspection procedures, testing requirements, and material selection. Some storage applications or service conditions (low temperature storage) require storage tanks to be fabricated from metals such as low alloy stainless steel, aluminum, or other specialty materials.

Non-Metallic — Older non-metallic tanks were customarily constructed from wood. Plastic materials have now replaced wood. These materials have the advantage of being non-corroding, durable, low cost, and lightweight. Plastic materials used in the construction are polyvinyl chloride, polyethylene, polypropylene, and fiberglass-reinforced polyesters. The fiberglass-reinforced polyester (FRP) tanks are available in the larger sizes and are the most common. FRP tanks are suitable for outdoor as well as indoor applications. FRP tanks with special reinforced shells are designed for underground storage service. Above ground tanks are primarily vertical, with or without top heads.

Non-metallic tanks constructed of unreinforced plastics such as polyvinyl chloride or polyethylene materials are available in sizes up to about 6 ft in diameter by 11 ft high (2400 gallons). Horizontal underground FRP tanks will hold up to 12,000 gallons. Above ground vertical FRP tanks can store from 12,000 to 24,000 gallons, depending upon the shell construction.

The temperature limits of plastic tanks are 40°F to 150°F. Color must be added to the outer liner for protection against ultraviolet radiation. The inner liner must be selected for compatibility with the chemical or product stored. Protection from mechanical abuse such as impact loads is a necessity. Good planning dictates that plastic storage should not be located next to flammable storage tanks. All closed plastic tanks should be equipped with pressure relief devices.

Protective Coatings

Internal — Use of internal coatings is primarily to protect the inside surface of the tank against corrosion while also protecting the stored contents from contamination.

Consideration must always be given to such factors as the type of product being stored, type of coating available, type of surface to be coated, surface preparation, compatibility of coatings, and number of coats required to obtain maximum protection.

Many types of internal coatings are available. Due to the unlimited types and applications, only a few will be described as follows:

Coal Tar — Among the oldest and most reliable coatings. Extremely low permeability; protects surface by the mechanical exclusion of moisture and air; extremely water resistant; good resistance to weak mineral acids, alkalis, salts, brine solutions, and other aggressive chemicals.

Epoxy Resin Coatings — Excellent adhesion, toughness, abrasion resistance, flexibility, durability, and good chemical and moisture resistance. Typical applications include linings for sour crude tanks, floating roof tanks, solvent storage tanks, drilling mud tanks, sour water, treated water, and pipelines.

Rubber Lining — Used as internal lining for storage tanks which are subjected to severe service such as elevated temperatures or for protection from extremely corrosive contents, such as concentrated chlorides and various acids such as chromic, sulfuric, hydrochloric, and phosphoric.

Galvanized — Galvanizing (zinc coating) is highly resistant to most types of corrosion. Bolted steel tanks are ideally suited for galvanizing since all component parts are galvanized by the hot-dip process after fabrication but before erection. Galvanized bolted tanks are recommended where the oil produced contains sulfur compounds and/or is associated with hydrogen sulfide gas. Galvanizing is also effective against corrosion in seacoast areas where atmospheric salt conditions accelerate corrosion problems.

External — The basic requirements for external coatings are appearance and weather protection.

Numerous types of external coatings are available, ranging from basic one-coat primers to primers with one or more topcoats. Environmental conditions usually dictate the extent of coating applied. Offshore and coastal installations require more extensive coatings as compared to inland locations.

Insulation

Types — The four basic types of thermal insulating material are: fibrous, cellular, granular, and reflective. These materials differ in many characteristics. Refer to Fig. 6-15 for a description of these materials and typical conductivity values and principal properties of common industrial insulations.

Uses — Principal uses of insulation are for personnel protection, process temperature control, prevention of condensation, and conservation of energy.

Personnel Protection — Personnel protection is accomplished by the application of insulation of proper thickness where the surface temperature should be limited to approximately 150°F or as specified by applicable codes or company standards.

Process Temperature Control — Insulation thickness is specified in this case to help control the temperature of the process fluid. Electrical, steam, hot process fluid, hot oil, glycol-water tracing is used to add heat to the process line to balance the heat loss. The insulation thickness must be matched to the energy input to achieve the desired result. Freeze protection is another use for insulation. This includes fluids which have higher viscosities or freeze points.

Condensation — Insulation thickness must be sufficient to keep the outside surface of the insulation above the dew point of the surrounding air. Moisture condensation on a cool surface in contact with warmer humid air must be prevented because of the deterioration of the insulation. In addition to the required thickness of insulation, a vapor tight membrane must be properly applied to the insulation. As a rule, insulation thickness for condensation control is much greater than the thickness required for conservation of energy.

Conservation of Energy — High fuel costs increase the need for more insulation. A rule of thumb for estimating the thickness of insulation is to apply the thickness that produces a heat loss of 3 to 5 percent or less from the surface. Specific insulating materials and thicknesses for any large application should be determined with the assistance of the manufacturer. Fig. 6-15 contains 3 graphs which permit the rapid estimation of the thickness of thermal insulation required to give a desired heat flow or surface temperature when the hot face and ambient temperature are known. The method is based on elementary heat transfer theory and reliable experimental data. The following examples illustrate the use of these graphs.

Example 6-4 — A rectangular duct is operating at 450°F. The duct is finished with a silicone coated fabric. The ambient tem-

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FIG. 6-15

Constants for Determining Thermal Conductivity and Unit Heat-Transfer Rate for Some Common Insulating Materials

$$k = A' + B'T + C'T^2 + D'T^3 \quad \text{where } T = °F$$

Insulation	A′	B′	C′	D′	Temperature range, °F
Calcium silicate ASTM C533-80 Class 1	0.3504	5.196×10^{-4}			100-700
White fiberglass blankets with binder					
density = 3.0 lb/ft^3	0.2037	6.161×10^{-6}	1.403×10^{-6}	-5.0×10^{-10}	50-800
density = 6.0 lb/ft^3	0.2125	-2.325×10^{-4}	1.797×10^{-6}	-7.97×10^{-10}	50-900
Rigid fiberglass sheet					
ASTM C-547-77 Class 1	0.2391	9.192×10^{-4}	6.942×10^{-10}		11-121
ASTM C-547-77 Class 2	0.2782	1.226×10^{-3}			37-204
ASTM C-612-77 Class 1	0.2537	3.051×10^{-4}	1.950×10^{-6}		0-250
ASTM C-612-77 Class 3	0.2631	2.301×10^{-4}	1.614×10^{-6}		0-275
density = 4.0 lb/ft^3	0.2113	3.857×10^{-4}	1.20×10^{-6}		0-300
density = 6.0 lb/ft^3	0.1997	2.557×10^{-4}	9.048×10^{-7}		0-300
Cellular glass foam					
ASTM C-552-79 Class 1	0.3488	5.038×10^{-4}	1.144×10^{-7}	7.172×10^{-10}	–300-500
Mineral wool					
Basaltic rock blanket, 9 lb/ft^3	0.2109	3.382×10^{-4}	5.495×10^{-7}		0-800
Basaltic rock blanket, 12 lb/ft^3	0.2798	9.508×10^{-5}	6.478×10^{-7}		0-800
Metallic slag block, 6 lb/ft^3	0.1076	5.714×10^{-4}	3.124×10^{-7}		0-600
Metallic slag block, 18 lb/ft^3	0.3190	8.870×10^{-5}	2.174×10^{-7}		0-1200
Mineral-wool-based-cement	0.4245	6.293×10^{-4}	-1.638×10^{-7}	3.533×10^{-10}	0-950
Preformed expanded perlite					
ASTM C-610-74	0.3843	3.0×10^{-4}	2.2381×10^{-7}		50-750
Expanded perlite-based cement	0.6912	5.435×10^{-4}			50-650
Expanded polystyrene block					
ASTM C-578-69 GR2	0.1711	2.760×10^{-4}	1.796×10^{-6}	-3.997×10^{-9}	–58-110
Polyurethane, 2.2 lb/ft^3					
aged 720 days at 77°F and 50% relative humidity	0.1662	-4.094×10^{-4}	-5.273×10^{-6}	2.534×10^{-8}	–58-32
(85% closed cell)	0.1516	-3.370×10^{-4}	7.153×10^{-6}	-2.858×10^{-8}	32-122
new polyurethane	0.1271	-2.490×10^{-4}	-7.962×10^{-7}	4.717×10^{-8}	–58-32
(95% closed cell)	9.72×10^{-2}	7.813×10^{-4}	-7.152×10^{-6}	2.858×10^{-8}	32-122
Exfoliated vermiculite (insulating cement)					
Aislagreen*	0.480	6.0×10^{-4}			0-1200
ASTM C-196-77	0.8474	5.071×10^{-4}			0-1200

*Trademark of Cia. Mexicana de Refractarios A.P. Green S.A. (Mexico City)

perature is 80°F. It is desired to maintain a surface temperature of 130°F. What thickness of cellular glass foam is required? What is the heat loss?

Solution Steps using Fig. 6-16

$T_h = 450$ $\quad \Delta T_i = 450 - 130 = 320$
$T_s = 130$ $\quad \Delta T_f = 130 - 80 = 50$
$T_a = 80$ $\quad T_m = (450 + 130)/2 = 290$

In Fig. 6-16 at ΔT_f of 50, project vertically to curve A, then horizontally to the left to a heat loss (Q) of 98 Btu/(hr • sq ft).

Project horizontally to the right along the 98 Btu/hr • sq ft Q line to the point in Fig. 6-16b corresponding to a temperature drop through the insulation (ΔT_i) of 320, then vertically downward to an insulation resistance (R_i) of 3.3.

From Fig. 6-15, ASTM C-552-79 Class 1 cellular glass foam has a k at $T_m = 290°F$:

$$k = 0.3488 + (5.038)(10^{-4})(290) + (1.144)(10^{-7})(290)^2 + (7.172)(10^{-10})(290)^3$$

$$k = 0.52$$

Multiply required insulation resistance R_i by k to obtain required thickness (X).

$$X = (3.3)(0.52) = 1.7 \text{ inches}$$

Example 6-5 — In Example 6-4, if the heat loss [98 Btu/ (hr • sq ft)] is specified instead of a surface temperature of 130°F, the following procedure is used.

Project a line horizontally on Fig. 6-16a from a heat loss of 98 Btu/(hr • sq ft) to curve A, then vertically downward to a ΔT_f of 50. Surface temperature = 80 + 50 = 130°F. The rest of the solution remains the same.

Example 6-6 — Assume the same conditions as Example 6-4 except that the surface to be insulated is a 4" O.D. duct.

After determining the required thickness of 1.7 inches for a flat surface, go to Fig. 6-16c. Project horizontally from 1.7" for a flat surface to the line representing a 4" O.D. duct then vertically to an actual thickness of 1.35". The heat loss of 98 Btu/hr • sq ft of outside insulation surface remains the same.*

The heat loss per linear foot of outside duct surface (including insulation) is:

$$\frac{\pi \text{ OD}}{12}(Q) = \left(\frac{6.7 \cdot \pi}{12}\right)98 = 171 \text{ Btu/hr / linear ft}$$

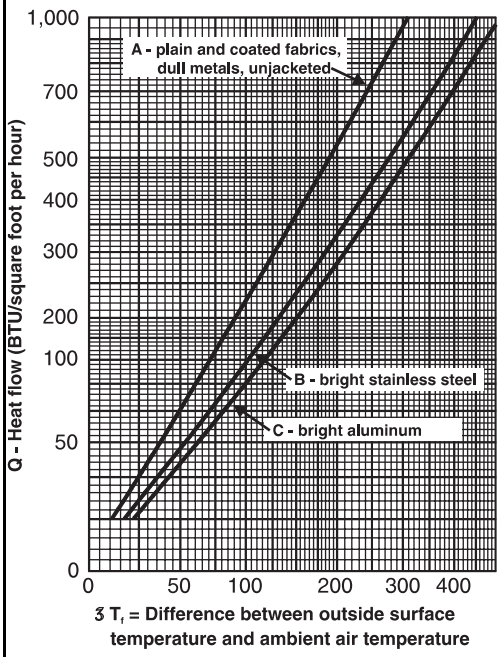
*The insulation surface temperature on tubing and ducts in the horizontal position is generally higher than in the vertical position for the same heat flow. To correct for the horizontal

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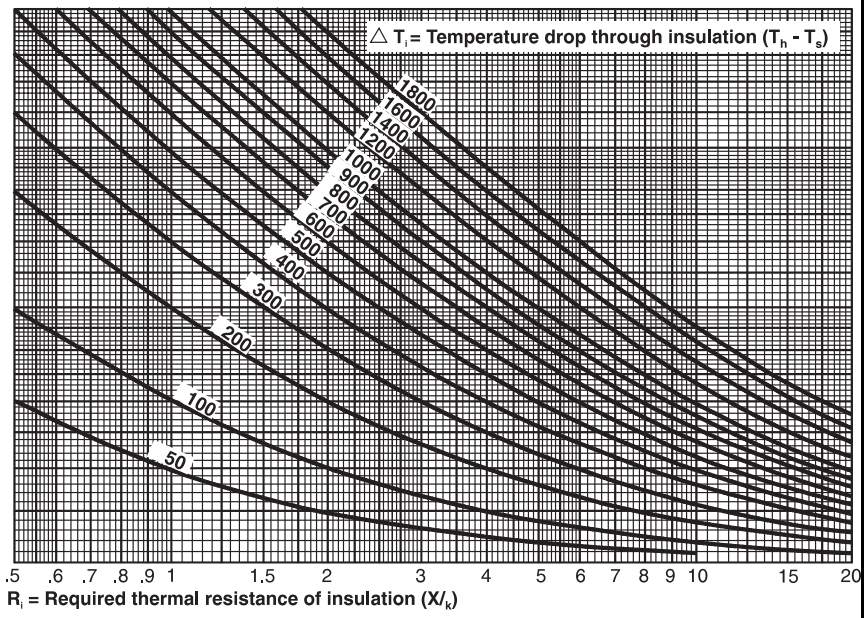
FIG. 6-16

Heat Flow Through Insulation

6-16a

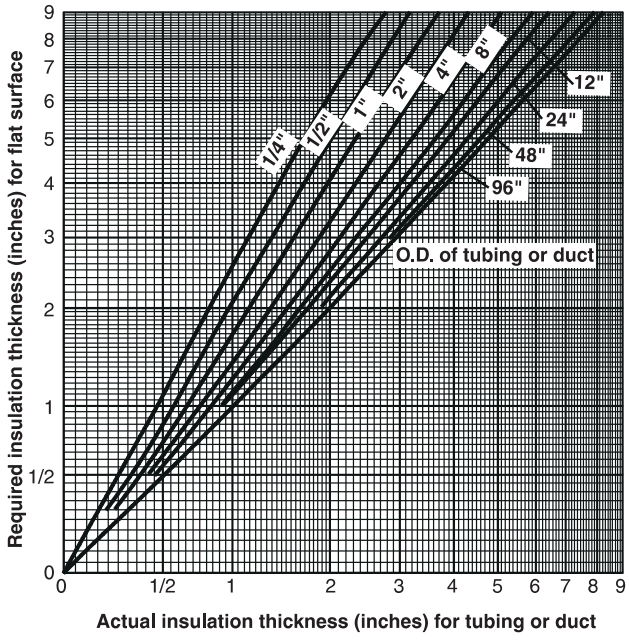


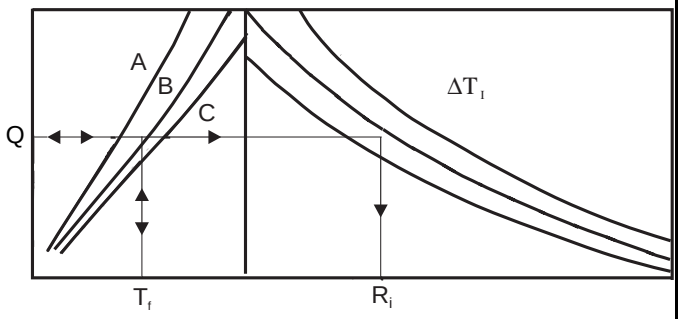
6-16b

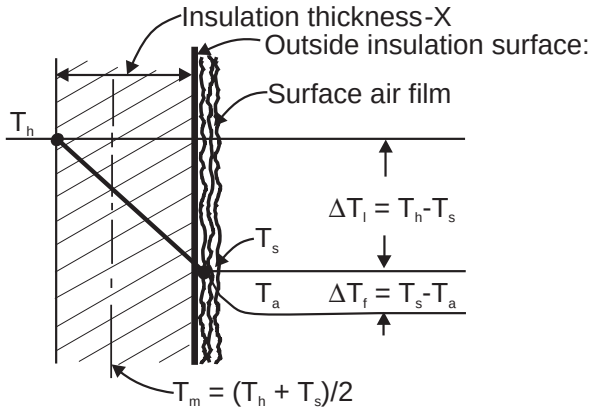


How to Use Figs. 6-16 a & b

6-16c







Courtesy John Mansville

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position, multiply the ΔT_f for flat surfaces obtained from Fig. 6-15a by the following factors (H):

Q (Btu/hr • sq ft)	10-99	100-199	200-299	300 and up
H	1.35	1.2	1.10	15.0

Example 6-7 — A furnace is operating at 1100°F. The outside surface is stainless steel. The ambient temperature is 75°F. It is desired to limit the heat loss to 150 Btu/hr • sq ft.

What thickness of mineral wool and cellular glass foam is required? What is the surface temperature?

Solution Steps $T_h = 1100$ $T_a = 75$
$Q = 150$ Btu/hr • sq ft

From Fig. 6-16a at Q of 150 Btu/hr • sq ft, project horizontally to curve B, then vertically to $\Delta T_f = 105$. T_s is (105 + 75) = 180. In this case a combination of mineral wool (metallic slag block, 18 lb/ft^3) on the hot face backed by cellular glass foam (ASTM C 552-79 Class 1) is to be used. From Fig. 6-15, the temperature limit of cellular glass foam is 500°F. The inner face temperature between the two materials should be close to, but not above, this limit.

ΔT_i (mineral wool) (1100 – 500) = 600

T_m (mineral wool) (1100 + 500)/2 = 800

k (from Fig. 6-15) of mineral wool at 800°F (T_m) = 0.53

ΔT_i(cellular glass foam) (500 – 180) = 320

T_m(cellular glass foam) (500 + 180)/2 = 340

k (from Fig. 6-15) of cellular glass foam at 340°F (T_m)=0.56

Using Fig. 6-16b, project horizontally along the 150 Q line to a ΔT_i (mineral wool) of 600°F, then vertically to an insulation resistance of 4.0. Thickness of mineral wool required (4.0 x .53) = 2.12 or 2.5 inches.

Similarly, project horizontally along the 150 Q line to a ΔT_i (cellular glass foam) of 320°F then vertically to an insulation resistance of 2.2. Thickness of cellular glass foam (2.2)(0.56) = 1.23 or 1.5 inches.

In the case of multiple layer construction Fig. 6-16c should not be used to convert to a circular cross section.

Refrigerated Tank Insulation Systems — Low temperature insulation is required for both spherical and flat bottomed cylindrical refrigerated tanks. Two types of insulation systems are commonly used for low temperature service — single wall and double wall.

In the single wall system, the vessel wall is designed to withstand the design service conditions of the liquid to be stored. The outer surface of this wall is then covered with a suitable insulating material such as rigid polyurethane foam. An aluminum jacket is then installed to provide protection against the elements and physical damage. It is extremely important that the insulation be sealed with a good vapor barrier to minimize air leakage and thereby reduce the quantity of water that may migrate into the insulation. Such moisture migration can ultimately damage the insulation.

The welded steel plate outer shell of a double wall system provides containment and vapor protection for the insulation material, generally perlite. The outer wall also provides protection against fires at temperatures up to 600°F. Double wall tanks are considered in storing products at temperatures below –28°F. This system minimizes heat leak which generally means lower operating and maintenance costs. As an added safety feature, the outer wall is completely sealed and therefore permits the insulation space to be continually purged with an appropriate inert gas, which keeps the insulation isolated from outside humid air. Figs. 6-15 and 6-17 provide a range of typical thermal conductivities for various types of insulating and tank shell materials.

APPURTENANCES

Storage tanks can be provided with any number of appurtenances, depending on the appropriate design codes and the requirements of the user. A tank may be fitted with mixers, heaters, relief/vacuum breaking devices, platforms and ladders, gauging devices, manways, and a variety of other connections which include manways, sumps, inlet and outlet nozzles, temperature gauges, pressure gauges, vents, and blowdowns.

SITE PREPARATION AND INSTALLATION

Dikes

Dikes are often required to contain the volume of a certain portion of the tanks enclosed depending on the tank contents. Dikes are used to protect surrounding property from tank spills or fires. In general, the net volume of the enclosed diked area should be the volume of the largest tank enclosed (single failure concept). The dike walls may be earth, steel, concrete, or solid masonry that are designed to be water tight with a full hydrostatic head behind it. Local codes and specifications may govern construction. If more than one tank is within the diked area, curbs or preferably drainage channels should be provided to subdivide the area in order to protect the adjacent tanks from possible spills.

Many codes, standards, and specifications regulate the location, design, and installation of storage tanks depending on their end use. Selecting the proper specification and providing adequate fire protection for the installation may allow lower insurance rates over the life of the installation. A partial list of applicable codes, standards, and specifications can be found at the end of this section.

Grounding

Metallic storage tanks used to store flammable liquids should be grounded to minimize the possibilities of an explosion or fire due to lightning or static electricity.

CATHODIC PROTECTION

Cathodic protection can be applied to control corrosion that is electrochemical in nature where direct current is discharged from the surface area of a metal (the anodic area) through an electrolyte. Cathodic protection reduces corrosion of a metal surface by using a direct current from an external source to oppose the discharge of metal immersed in a conducting medium or electrolyte such as soil, water, etc.

PRODUCT RECOVERY

Vapor Losses

Vapors emitted from the vents and/or relief valves of a storage tank are generated in two ways:

- Vapors that are forced out of the tank during filling operations.

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FIG. 6-17

Summary of Specifications for Low-Temperature and Cryogenic Steels[1] [2]

ASTM/AISI Spec & Grade	Lowest Usual Service Temp.[5]	Thermal Conductivity, Btu/[(hr · sq ft · °F)/in] at noted mean temperature, °F					Example Applications	
		+200	+68	−50	−150	−320	Uses	Liquids Stored
Carbon Steels								
A333 Grades 1 & 6	−50	354	349				Welded pressure vessels and storage tanks, when weight and strength are not critical. Refrigeration and transport equipment.	Butane, Isobutane, Sulfur Dioxide, Refrigerant 12, Ammonia, Refrigerant 22, Propane, Propylene
A334 Grades 1 & 6	−50	354	349					
A442 Grades 55 & 60	−50	354	349					
A516 Grades 55, 60, 65, & 70[3]	−50	354	349					
A537[4]	−50	354	349					
Alloy Steels								
A517 Grade F	−50		260				Highly stressed pressure vessels. Tank trucks for handling LP gases.	LP Gases
A203 Grades A&B - 2¼% Ni	−75	288	267	248	234	146	Tanks, vessels, and piping for liquid propane.	Propane
A333 Grade 7 - 2¼% Ni	−75	288	267	248	234	146		
A334 Grade 7 - 2¼% Ni	−75	288	267	248	234	146		
Alloy Steels								
A203 Grades D&E - 3½% Ni	−150	270	253	226	214	122	Land-based storage of liquid propane, carbon dioxide, acetylene, ethane and ethylene.	Propane, Carbon Dioxide, Acetylene, Ethane, Ethylene
A333 Grade 3 - 3½% Ni	−150	270	253	226	214	122		
A334 Grade 3 - 3½% Ni	−150	270	253	226	214	122		
Stainless Steels								
AISI - 300 Series (Type 301)	−150	113	103	94	86	56.4		
Alloy Steels								
A333 Grade 8 - 9% Ni	−320	209	189	180	156	91.3	Large tonnage oxygen-producing equipment. Transportation and storage of methane, oxygen, nitrogen, and argon.	Ethylene, Methane, Oxygen, Carbon Monoxide, Nitrogen, LNG, Argon
A334 Grade 8 - 9% Ni	−320	209	189	180	156	91.3		
A353 Grade 8 - 9% Ni	−320	209	189	180	156	91.3		
ASME Code Case 1308 - 9% Ni								
Stainless Steels								
AISI - 300 Series (Type 302)	−320	113	101	91	81	56.4		
Stainless Steels								
AISI - 300 Series (Type 304)	−452	113	117.6	91	90.0	56.4	In petrochemical, nuclear, missile, and other areas where purity of product is essential. Handling liquid hydrogen rocket fuel.	Hydrogen, Helium

(1) Exact composition selected for a given application is dependent upon type of product (e.g., plates, tubular products, etc.) and/or section size (i.e., thickness). For more information, consult API Standard 620 or the tank manufacturer.

(2) Values are maximum unless otherwise specified or a range is given.

(3) To ASTM A300 specifications.

(4) With modifications.

(5) The Charpy V-notch impact test may be used to qualify a particular grade of steel for a design temperature that is lower than is normally permitted. API Standard 620 and ASME Boiler & Pressure Vessel Code, Section VIII, Division I, should be consulted for specifics pertaining to the Charpy V-notch tests.

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- Vapors that are generated by vaporization of the liquid stored in the tank.

A vapor recovery system should be sized to handle the total vapor from these two sources.

Displacement Losses — Vapors that are forced out of the tank are generally called displacement losses. A storage tank is generally not pumped completely dry when emptied. The vapor above the remaining liquid in the tank will expand to fill the void space at the vapor pressure of the liquid stored in the tank at storage temperature. As the tank is filled, the vapors are compressed into a smaller void space until the set pressure on the vent/relief system is reached. There are also some filling losses that are associated with the expansion of the liquid into the tank. Fig. 6-18 provides a graphical approach to estimating the filling losses as a percentage of the liquid being pumped into the tank.

Vaporization Losses — This type of loss is characterized as the vapors generated by heat gain through the shell, bottom, and roof. The total heat input is the algebraic sum of the radiant, conductive, and convective heat transfer. This type of loss is especially prevalent where light hydrocarbon liquids are stored in full pressure or refrigerated storage. This is less prevalent but still quite common in crude oil and finished product storage tanks. These vapors may be recovered by the use of a vapor recovery system.

To calculate vaporization in tanks, sum up the effects of radiant, conductive, and convective heat inputs to the tank. Approximate vapor losses in lb/hr can then be calculated by dividing the total heat input by the latent heat of vaporization of the product at the fluid temperature.

Liquid Equivalents of Tank Vapors — The following procedure may be followed to calculate the liquid equivalent of vapor volumes above stored LP-gas liquids:

General Approach

Data Required:

1. Liquid product composition in mole % or mole fraction.
2. Temperature and pressure of the product from which the liquid sample was obtained.
3. Vapor-liquid equilibrium K values at an assumed 1,000 psia convergence pressure (see Section 25).

Calculation Procedure:

1. With the liquid product composition, calculate the bubble point pressures of the product at assumed temperatures: i.e., 60°F, 80°F. From the bubble point calculations, a vapor pressure chart can be made for this specific product composition.
2. From the bubble point calculation in (1), the product vapor composition can be obtained: i.e.,

$$\Sigma(y_i) = \Sigma(K_i x_i) = 1.0 \qquad \textbf{Eq 6-4}$$

3. Calculate the compressibility factor for the vapor by either (a) or (b).
   a. Compressibility factor charts, Section 23. Pseudocritical and pseudoreduced temperatures and pressures must be calculated to obtain a compressibility factor.
   b. Equations of state.
4. Calculate the total number of moles of vapor for volume V, by using the modified ideal-gas equation:

$$PV = n_g ZRT,\ n_g = PV/ZRT = \text{total moles vapor} \qquad \textbf{Eq 6-5}$$

5. Calculate the gallons of liquid equivalent in the vapor phase by multiplying the total number of moles of vapor by the mole fraction of each component by the gal./mole factors for that component from Fig. 23-2.

$$\Sigma[n_g(y_i)\ (\text{gal./mole})_i] = 60°\text{F gallons in vapor phase} \qquad \textbf{Eq 6-6}$$

Example 6-8 — Determine three points of data used to plot Fig. 6-18.

1. Calculate composition of vapor at the three data points.

	Liquid C_3 Composition	Bubble-point pressures					
		0°F, 42 psia		60°F, 114 psia		120°F, 255 psia	
	x	K	y	K	y	K	y
C_2	0.03	4.35	0.1305	3.15	0.0945	2.55	0.0765
C_3	0.95	0.909	0.8633	0.945	0.8975	0.962	0.9136
iC_4	0.02	0.309	0.0062	0.398	0.0080	0.493	0.0099
	1.00		1.0000		1.0000		1.0000

2. Determine compressibility factor at the three points.

Vapor			
Average MW, $\Sigma(y_i MW_i)$	42.353	42.884	43.163
Pseudo T_c, °R	651	655	658
Pseudo P_c, psia	628	624	622
T_R	0.707	0.794	0.881
P_R	0.067	0.183	0.410
Z (Section 23)	0.913	0.855	0.730

3. Calculate moles of vapor per 1000 gal. of vapor.

$$n_g = \frac{PV}{ZRT} \text{ and } n_i = (n_g y_i)$$

$$V = \frac{1{,}000}{7.48} = 133.7 \text{ cu ft}$$

$+n_i$, moles	C_2	0.1626	0.3019	0.5741
	C_3	1.0757	2.8673	6.8556
	iC_4	0.0077	0.0256	0.0743
	$n_g = \Sigma n_i$	1.2460	3.1948	7.5040

4. Calculate liquid equivalent gallons (60°F) per 1000 gallons vapor.

	gal./mole			
C_2	10.126	1.646	3.057	5.813
C_3	10.433	11.223	29.915	71.524
iC_4	12.386	0.095	0.317	0.920
Liquid equivalent, gal.		12.964	33.289	78.257

Suggested Simplified Approach

By using a typical product analysis, calculations can be made as outlined above, and from these calculations (see example 6-8) vapor pressure and gallon equivalent charts can be drawn as shown in Fig. 6-19. A convenient unit of vapor space volume should be used, such as 1,000 gal.

Vapor Recovery Systems

Vapor recovery systems are generally used to prevent pollution of the environment and to recover valuable product. Two basic types of vapor recovery systems may be encountered. One is designed to gather toxic wastes that would pollute the atmosphere but are not valuable enough to warrant full recovery. In this type system, the vapors are generally gathered and incinerated. If incineration will not meet government disposal standards, the vapors are generally compressed and condensed into a liquid and sent to a liquid disposal system.

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The vapor recovery systems that are typically used with refrigerated storage tanks are generally integrated with the product refrigeration systems. In these types of systems, the vapors are generally compressed, condensed, and put back into the tank with the fill stream.

Vapor recovery systems on atmospheric pressure, ambient temperature storage tanks do not normally require a refrigeration system to condense the vapors. They are generally compressed through one stage of compression, condensed in either an air cooled or water cooled exchanger, and then put back into the tank. Fig. 6-20 provides the flow schematic of this system.

FIG. 6-18

Filling Losses from Storage Containers

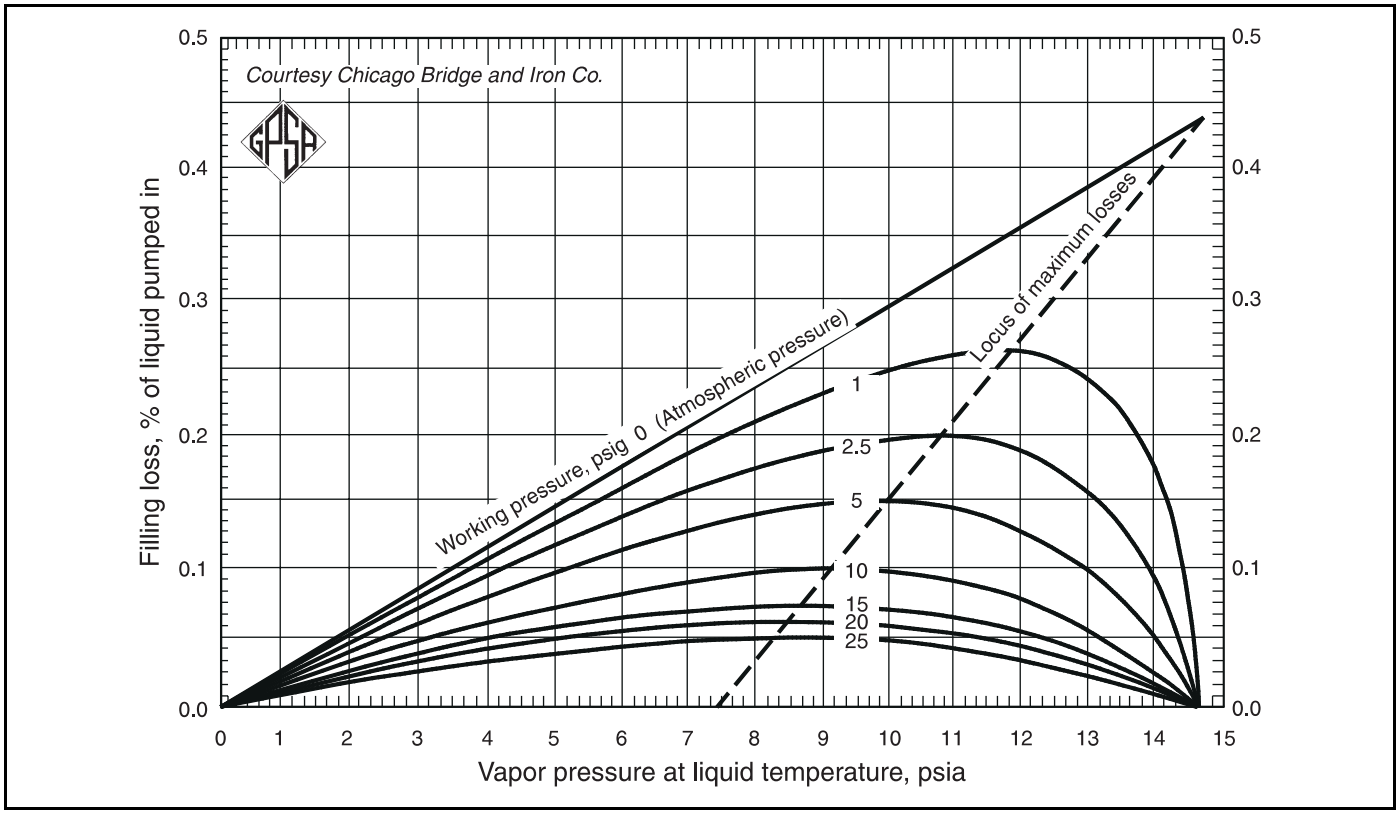


FIG. 6-19

Liquid Equivalent of Tank Vapor

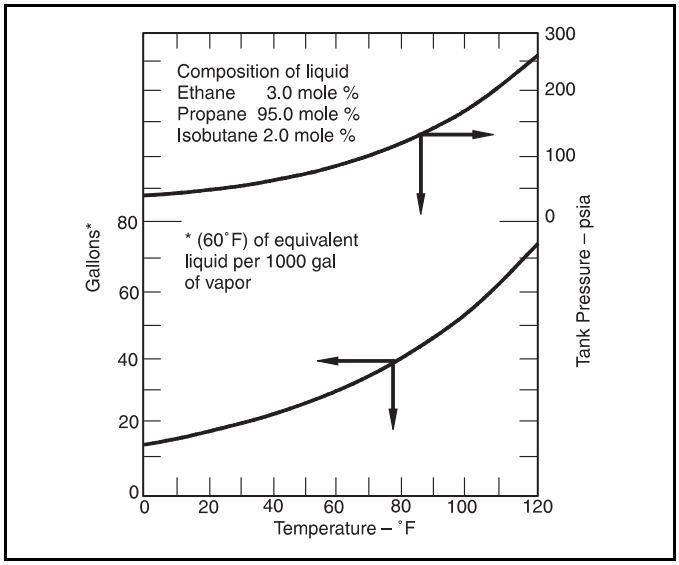
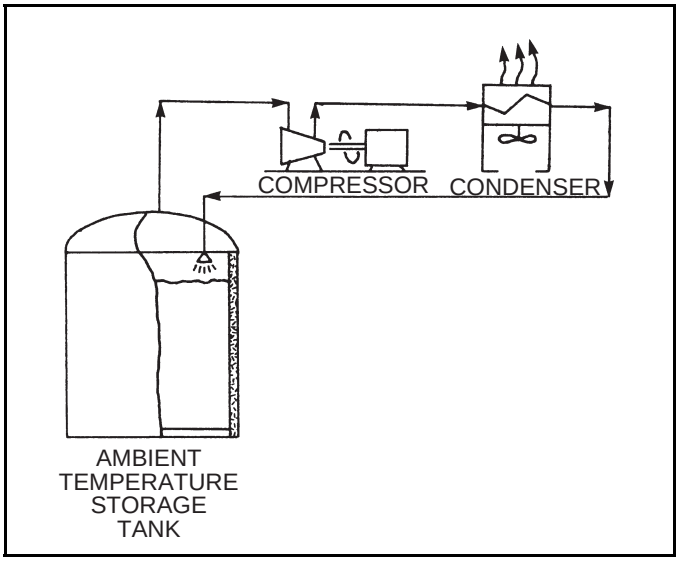


FIG. 6-20

Ambient Temperature Vapor Recovery Cycle



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LIQUID STORAGE

Desirability of Large Units

The Hortonsphere vessel permits the storage of a large volume in one unit with only one set of pipe connections and fittings. Batteries of shop built-up cylindrical tanks have been used to provide large volumes of pressure storage. This practice necessitates the use of multiple pipe connections and the duplication of tank fittings, vents and foundations. A battery of cylindrical tanks will generally occupy about four times more ground space than the same volume of storage in a Hortonsphere vessel. This factor is an important consideration in many locations where land values are high and space is at a premium.

The Hortonsphere vessel has less surface area for a given capacity than a container of any other shape. It is also true that the larger it is, the less its surface area per unit of volume. For these reasons, the liquid stored in a Hortonsphere vessel of large capacity changes temperature more slowly than in small vessels. Since the required operating pressure is a function of the temperature, the internal pressure in a large Hortonsphere vessel for liquid storage is less likely to exceed the setting of the relief valve during short periods of extremely hot weather. A large Hortonsphere vessel is, therefore, more efficient in preventing loss of vapors from a given volatile liquid than a smaller one designed for the same working pressure.

The larger units of storage are also more desirable because the cost per unit of capacity is less. Having less surface area, they provide a structure that is more economical to paint and maintain. The cost of insulation, when required, is also lower per barrel of capacity.

Capacities

Hortonsphere vessels for liquid storage are commonly built in the English and Metric capacities shown in Fig. 6-21. Intermediate or larger sizes and pressures can be supplied if desired.

FIG. 6-21

Hortonsphere Vessels for Liquid Storage

Nominal Capacity (Barrels)	Diameter (Ft-In)	ASME VIII Division 1 Pressure (psi)	ASME VIII Division 2 Pressure (psi)	Actual Volume (Ft^3)	Inside Surface Area (Ft^2)
1,000	22-3	380	532	5,768	1,555
1,500	25-6	327	456	8,682	2,043
2,000	28-0	306	432	11,494	2,463
2,500	30-3	282	399	14,494	2,875
3,000	32-0	266	377	17,157	3,217
4,000	35-3	240	341	22,934	3,904
5,000	38-0	222	315	28,731	4,536
6,000	40-6	207	295	34,783	5,153
7,500	43-6	191	273	43,099	5,945
10,000	48-0	171	246	57,906	7,238
12,000	51-0	160	230	69,456	8,171
15,000	54-9	147	213	85,931	9,417
20,000	60-6	131	190	115,948	11,499
25,000	65-0	120	175	143,793	13,273
30,000	69-0	111	163	172,007	14,957
40,000	76-0	98	145	229,847	18,146
50,000	81-10	89	132	286,939	21,038

Gaging Table

A theoretical gaging table can be furnished with each Hortonsphere vessel for liquid service. This table is accurate because each installation conforms closely with the specified dimensions. The table shows the liquid volume for every inch of depth.

Accessories

The Hortonsphere vessel is furnished with a standard set of accessories including a stairway, handrail at the top, and top and bottom manholes. Nozzles are furnished as specified. One or more dependable pressure relief valves must be provided and, in most cases, vacuum relief valves are recommended for spheres designed for low internal pressure.

PARTIAL VOLUMES IN STORAGE TANKS

The volume or size of a storage tank is determined by the configuration of the tank that is used (horizontal or vertical cylinder, sphere, rectangle). Each configuration uses different formulas for determining the total and partial volumes. Figs. 6-22 through 6-28 can be used to determine total and partial volumes in most common storage tanks.

STANDARDS AND CODES

ANSI A12.1
: Safety Requirements for Floor and Wall Openings, Railings, and Toeboards.

ANSI A14.1
: Requirements for Fixed Industrial Stairs.

ANSI A14.3
: Safety Code for Fixed Ladders.

ANSI A11.197
: Measurement and Calibration of Upright Cylindrical Tanks, Method for (ASTM D 1220-65, API 2550).

ANSI Z11.198
: Measurement and Calibration of Horizontal Tanks, Method for (ASTM D 1410-65, API 2551).

ANSI A11.1988
: Measurement and Calibration of Spheres and Spheroids, Method for (ASTM D 1408-65, API 2552).

ANSI Z11.202
: Liquid Calibration of Tanks (ASTM D 1406-65 API 2555).

ANSI/ASME B31.4
: Liquid Petroleum Transportation Piping System.

API Recommended Practices for Leached Underground Storage prepared by API Committee 510.

API Specification 12 B
: Specifications for Bolted Tanks for Storage of Production Liquids.

API Specification 12 D
: Specifications for Field Welded Tanks for Storage of Production Liquids.

API Specification 12 F
: Specifications for Shop Welded Tanks for Storage of Production Liquids.

API Standard 620
: Recommended Rules for Design and Construction of Large, Welded Low-Pressure Storage Tanks.

API Standard 650
: Welded Steel Tanks for Crude Storage.

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FIG. 6-22

Circumference, Area, and Volume of Circles and Cylinders

Diam. ft	Circumference		Area of Circle		Volume of cylinder/foot of height			Diam., ft
	Feet	Meters	sq ft	sq meters	U.S. gal.	Imperial gal.	U.S. bbls (42 gal.)	
1	3.14	0.9576	0.785	.0730	5.9	4.9	0.140	1
2	6.28	1.9151	3.142	.2919	23.5	19.6	0.560	2
3	9.42	2.8727	7.069	.6567	52.9	44.0	1.259	3
4	12.57	3.8302	12.566	1.1675	94.0	78.3	2.238	4
5	15.71	4.7878	19.635	1.8241	146.9	122.3	3.497	5
6	18.85	5.7454	28.274	2.6268	211.5	176.1	5.04	6
7	21.99	6.7029	38.485	3.5753	287.9	239.7	6.85	7
8	25.13	7.6605	50.266	4.6698	376.0	313.1	8.95	8
9	28.27	8.6180	63.617	5.9102	475.9	396.3	11.33	9
10	31.42	9.5756	78.540	7.2966	587.5	489.2	13.99	10
11	34.56	10.5332	95.033	8.8289	710.9	591.9	16.93	11
12	37.70	11.4907	113.097	10.5071	846.0	704.5	20.14	12
13	40.84	12.4483	132.732	12.3312	992.9	826.8	23.64	13
14	43.98	13.4059	153.938	14.3013	1,151.5	958.9	27.42	14
15	47.12	14.3634	176.715	16.4173	1,321.9	1,100.7	31.47	15
16	50.27	15.3210	201.062	18.6792	1,504.0	1,252.4	35.81	16
17	53.41	16.2785	226.980	21.0871	1,697.9	1,413.8	40.43	17
18	56.55	17.2361	254.469	23.6409	1,903.6	1,585.1	45.32	18
19	59.69	18.1937	283.529	26.3407	2,120.9	1,766.1	50.50	19
20	62.83	19.1512	314.159	29.1863	2,350.1	1,956.9	55.95	20
22	69.12	21.0663	380.133	35.3154	2,843.6	2,367.8	67.70	22
24	75.40	22.9815	452.389	42.0283	3,384.1	2,817.9	80.57	24
26	81.68	24.8966	530.929	49.3249	3,971.6	3,307.1	94.56	26
28	87.97	26.8117	615.752	57.2052	4,606.1	3,835.4	109.67	28
30	94.25	28.7268	706.858	65.6692	5,287.7	4,402.9	125.90	30
32	100.53	30.6420	804.248	74.7170	6,016.2	5,009.6	143.24	32
34	106.81	32.5571	907.920	84.3485	6,791.7	5,655.3	161.71	34
36	113.10	34.4722	1,017.88	94.5637	7,614.2	6,340.2	181.29	36
38	119.38	36.3873	1,134.11	105.3626	8,483.8	7,064.3	201.99	38
40	125.66	38.3024	1,256.64	116.7453	9,400.3	7,827.4	223.82	40
42	131.95	40.2176	1,385.44	128.7117	10,363.8	8,629.7	246.76	42
44	138.23	42.1327	1,520.53	141.2618	11,374.4	9,471.2	270.82	44
46	144.51	44.0478	1,661.90	154.3956	12,431.9	10,351.8	296.00	46
48	150.80	45.9629	1,809.56	168.1132	13,536.4	11,271.5	322.30	48
50	157.08	47.8781	1,963.50	182.4145	14,688.0	12,230.4	349.71	50
55	172.79	52.6659	2,375.83	220.7215	17,772.4	14,798.7	423.15	55
60	188.50	57.4537	2,827.43	262.6769	21,150.7	17,611.7	503.59	60
65	204.20	62.2415	3,318.31	308.2805	24,822.7	20,669.3	591.02	65
70	219.91	67.0293	3,848.45	357.5324	28,788.4	23,971.5	685.44	70
75	235.62	71.8171	4,417.86	410.4326	33,047.9	27,518.3	786.86	75
80	251.33	76.6049	5,026.55	466.9811	37,601.2	31,309.7	895.27	80
85	267.04	81.3927	5,674.50	527.1779	42,448.2	35,345.8	1,010.67	85
90	282.74	86.1805	6,361.73	591.0230	47,589.0	39,626.4	1,133.07	90
95	298.45	90.9683	7,088.22	658.5163	53,023.5	44,151.6	1,262.47	95
100	314.16	95.7561	7,853.98	729.6580	58,751.9	48,921.5	1,398.85	100
110	345.58	105.3317	9,503.32	882.8862	71,089.7	59,195.0	1,692.61	110
120	376.99	114.9073	11,309.73	1,050.7075	84,602.7	70,446.9	2,014.35	120
130	408.41	124.4829	13,273.23	1,233.1220	99,290.6	82,677.3	2,364.06	130
140	439.82	134.0585	15,393.80	1,430.1297	115,153.6	95,886.1	2,741.75	140
150	471.24	143.6342	17,671.46	1,641.7305	132,191.7	110,073.3	3,147.42	150
160	502.65	153.2098	20,106.19	1,867.9245	150,404.7	125,239.0	3,581.07	160
170	534.07	162.7854	22,698.00	2,108.7116	169,792.8	141,383.1	4,042.69	170
180	565.49	172.3610	25,446.90	2,364.0919	190,356.0	158,505.6	4,532.29	180
190	596.90	181.9366	28,352.87	2,634.0654	212,094.2	176,606.5	5,049.86	190
200	628.32	191.5122	31,415.93	2,918.6320	235,007.4	195,685.9	5,595.42	200

NOTES:

1. If diameters are assumed as meters, values in the columns "Circumference Feet" and "Area of Circle Square Feet" will represent circumference in meters and area of circle in square meters respectively.
2. If diameters are assumed as meters, values in column "Area of Circle Square Feet" will represent volume of cylinder in cubic meters per vertical meter of height.

Formula to determine capacity per foot of vertical height of cylinder.

D = Diameter in Feet

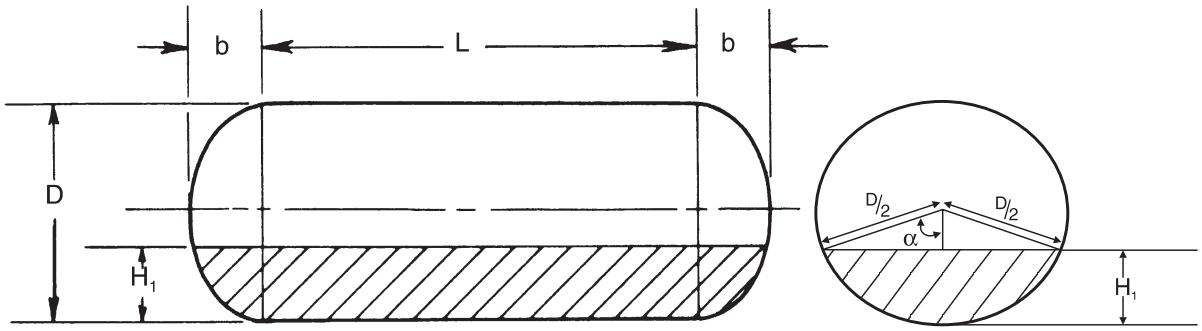
$0.1398854\ D^2$ = Barrels of 42 U.S. Gallons per vertical foot.
$5.875185\ D^2$ = U.S. Gallons per vertical foot.
$4.892148\ D^2$ = Imperial Gallons per vertical foot.
$0.022240\ D^2$ = Cubic Meters per vertical foot.
$0.785398\ D^2$ (D in meters) = Cubic meters per vertical meter.

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Fig. 6-23

Partial Volume in Horizontal and Vertical Storage Tanks with Ellipsoidal or Hemispherical Heads

HORIZONTAL CYLINDRICAL TANKS



Total volume = volume in 2 heads + volume in cylinder

$$= 1/6\ \pi\ K_1\ D^3 + 1/4\ \pi\ D^2\ L$$

$$\alpha = 2 \times \text{Atan}\left(\frac{H_1}{\sqrt{\left(2 \times H_1 \times \frac{D}{2}\right) - H_1^{\,2}}}\right)$$

where α is in radians

$K_1 = 2\ b/D$ $\quad Ze = H_1/D$ $\quad Zc = H_1/D$

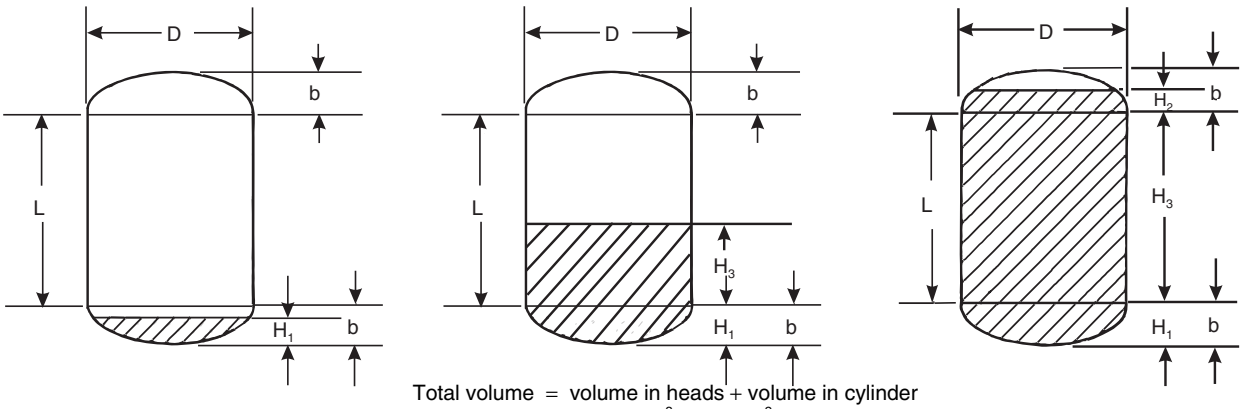
Partial volume $= 1/6\ \pi\ K_1\ D^3 \times [f(Ze)] + 1/4\ \pi\ D^2\ L \times [f(Zc)]$

f(Zc) = Horizontal cylinder coefficient (see Fig. 6–24) or $f(Zc) = \left(\frac{\alpha - \sin(\alpha) \times \cos(\alpha)}{\pi}\right)$

f(Ze) = Ellipsoidal coefficient (see Fig. 6–25) or $f(Ze) = -\left(\frac{H_1}{D}\right)^2 \times \left(-3 + \frac{2H_1}{D}\right)$

For elliptical 2:1 heads, $b = 1/4\ D$, $K_1 = \frac{1}{2}$

VERTICAL CYLINDRICAL TANKS



Total volume = volume in heads + volume in cylinder

$$= 1/6\ \pi\ K_1\ D^3 + 1/4\ \pi\ D^2\ L$$

Partial volume $= 1/6\ \pi\ K_1\ D^3 \times [f(Ze)] + 1/4\ \pi\ D^2\ H_3$

$K_1 = 2\ b/D$

$Ze = (H_1 + H_2)/K_1\ D$

f(Ze) = Ellipsoidal coefficient (see Fig. 6–25) or $f(Ze) = -\left(\frac{H_1 + H_2}{2b}\right)^2 \times \left(-3 + \left(\frac{H_1 + H_2}{b}\right)\right)$

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FIG. 6-24

Coefficients for Partial Volumes of Horizontal Cylinders, f(Zc)

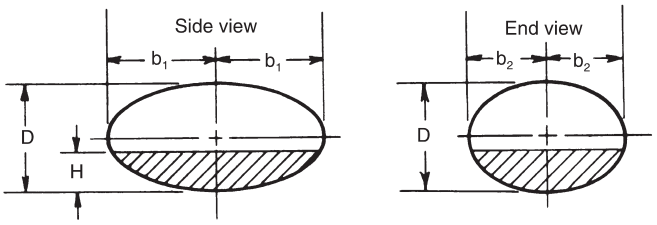
Zc	0	1	2	3	4	5	6	7	8	9
.00	.000000	.000053	.000151	.000279	.000429	.000600	.000788	.000992	.001212	.001445
.02	.004773	.005134	.005503	.005881	.006267	.006660	.007061	.007470	.007886	.008310
.04	.013417	.013919	.014427	.014940	.015459	.015985	.016515	.017052	.017593	.018141
.06	.024496	.025103	.025715	.026331	.026952	.027578	.028208	.028842	.029481	.030124
.08	.037478	.038171	.038867	.039569	.040273	.040981	.041694	.042410	.043129	.043852
.10	.052044	.052810	.053579	.054351	.055126	.055905	.056688	.057474	.058262	.059054
.12	.067972	.068802	.069633	.070469	.071307	.072147	.072991	.073836	.074686	.075539
.14	.085094	.085979	.086866	.087756	.088650	.089545	.090443	.091343	.092246	.093153
.16	.103275	.104211	.105147	.106087	.107029	.107973	.108920	.109869	.110820	.111773
.18	.122403	.123382	.124364	.125347	.126333	.127321	.128310	.129302	.130296	.131292
.20	.142378	.143398	.144419	.145443	.146468	.147494	.148524	.149554	.150587	.151622
.22	.163120	.164176	.165233	.166292	.167353	.168416	.169480	.170546	.171613	.172682
.24	.184550	.185639	.186729	.187820	.188912	.190007	.191102	.192200	.193299	.194400
.26	.206600	.207718	.208837	.209957	.211079	.212202	.213326	.214453	.215580	.216708
.28	.229209	.230352	.231498	.232644	.233791	.234941	.236091	.237242	.238395	.239548
.30	.252315	.253483	.254652	.255822	.256992	.258165	.259338	.260512	.261687	.262863
.32	.275869	.277058	.278247	.279437	.280627	.281820	.283013	.284207	.285401	.286598
.34	.299814	.301021	.302228	.303438	.304646	.305857	.307068	.308280	.309492	.310705
.36	.324104	.325326	.326550	.327774	.328999	.330225	.331451	.332678	.333905	.335134
.38	.348690	.349926	.351164	.352402	.353640	.354879	.356119	.357359	.358599	.359840
.40	.373530	.374778	.376026	.377275	.378524	.379774	.381024	.382274	.383526	.384778
.42	.398577	.399834	.401092	.402350	.403608	.404866	.406125	.407384	.408645	.409904
.44	.423788	.425052	.426316	.427582	.428846	.430112	.431378	.432645	.433911	.435178
.46	.449125	.450394	.451663	.452932	.454201	.455472	.456741	.458012	.459283	.460554
.48	.474541	.475814	.477086	.478358	.479631	.480903	.482176	.483449	.484722	.485995
.50	.500000	.501274	.502548	.503821	.505094	.506367	.507640	.508913	.510186	.511458
.52	.525459	.526731	.528003	.529275	.530547	.531818	.533090	.534362	.535633	.536904
.54	.550875	.552143	.553413	.554682	.555950	.557218	.558486	.559754	.561021	.562288
.56	.576212	.577475	.578739	.580002	.581264	.582527	.583789	.585051	.586313	.587574
.58	.601423	.602680	.603937	.605192	.606447	.607702	.608956	.610210	.611463	.612717
.60	.626470	.627718	.628964	.630210	.631455	.632700	.633944	.635189	.636432	.637675
.62	.651310	.652545	.653780	.655015	.656249	.657481	.658714	.659946	.661177	.662407
.64	.675896	.677119	.678340	.679561	.680781	.681999	.683217	.684434	.685650	.686866
.66	.700186	.701392	.702597	.703802	.705005	.706207	.707409	.708610	.709809	.711008
.68	.724131	.725318	.726505	.727690	.728874	.730058	.731240	.732422	.733603	.734782
.70	.747685	.748852	.750017	.751181	.752345	.753506	.754667	.755827	.756984	.758141
.72	.770791	.771935	.773076	.774217	.775355	.776493	.777629	.778765	.779898	.781030
.74	.793400	.794517	.795632	.796747	.797859	.798969	.800078	.801186	.802291	.803396
.76	.815450	.816537	.817622	.818706	.819788	.820869	.821947	.823024	.824100	.825175
.78	.836880	.837934	.838987	.840037	.841085	.842133	.843178	.844221	.845263	.846303
.80	.857622	.858639	.859655	.860668	.861680	.862690	.863698	.864704	.865708	.866709
.82	.877597	.878575	.879550	.880523	.881494	.882462	.883428	.884393	.885354	.886314
.84	.896725	.897657	.898586	.899514	.900440	.901362	.902283	.903201	.904116	.905029
.86	.914906	.915788	.916668	.917544	.918419	.919291	.920159	.921025	.921888	.922749
.88	.932028	.932853	.933677	.934497	.935313	.936128	.936938	.937747	.938551	.939352
.90	.947956	.948717	.949476	.950232	.950983	.951732	.952477	.953218	.953957	.954690
.92	.962522	.963211	.963896	.964577	.965253	.965927	.966595	.967260	.967919	.968576
.94	.975504	.976106	.976704	.977297	.977885	.978467	.979045	.979618	.980187	.980750
.96	.986583	.987080	.987568	.988053	.988530	.989001	.989466	.989924	.990375	.990821
.98	.995227	.995579	.995923	.996257	.996581	.996896	.997200	.997493	.997777	.998048
1.00	1.000000									

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FIG. 6-25

Table of Coefficients and Formulas for Determining Partial Volumes in Ellipsoids and Spheres

GENERAL ELLIPSOID



Total volume of ellipsoid = $1/6\ \pi\ K_1K_2\ D^3$

Partial Volume = $1/6\ \pi\ K_1K_2\ D^3\ [f(Ze)]$

$K_1 = 2b_1/D \quad K_2 = 2b_2/D$

$Ze = H/D$ f(Ze) = ellipsoidal coefficient (see below) or $= -\left(\frac{H_1}{D}\right)^2 \times \left(-3 + \frac{2H_1}{D}\right)$

Coefficients for Partial Volumes of Ellipsoids or Spheres, f(Ze)										
Ze	0	1	2	3	4	5	6	7	8	9
.00	.000000	.000003	.000012	.000027	.000048	.000075	.000108	.000146	.000191	.000242
.02	.001184	.001304	.001431	.001563	.001700	.001844	.001993	.002148	.002308	.002474
.04	.004672	.004905	.005144	.005388	.005638	.005893	.006153	.006419	.006691	.006968
.06	.010368	.010709	.011055	.011407	.011764	.012126	.012493	.012865	.013243	.013626
.08	.018176	.018620	.019069	.019523	.019983	.020447	.020916	.021390	.021869	.022353
.10	.028000	.028542	.029090	.029642	.030198	.030760	.031326	.031897	.032473	.033053
.12	.039744	.040380	.041020	.041665	.042315	.042969	.043627	.044290	.044958	.045630
.14	.053312	.054037	.054765	.055499	.056236	.056978	.057724	.058474	.059228	.059987
.16	.068608	.069416	.070229	.071046	.071866	.072691	.073519	.074352	.075189	.076029
.18	.085536	.086424	.087315	.088210	.089109	.090012	.090918	.091829	.092743	.093660
.20	.104000	.104962	.105927	.106896	.107869	.108845	.109824	.110808	.111794	.112784
.22	.123904	.124935	.125970	.127008	.128049	.129094	.130142	.131193	.132247	.133305
.24	.145152	.146248	.147347	.148449	.149554	.150663	.151774	.152889	.154006	.155127
.26	.167648	.168804	.169963	.171124	.172289	.173456	.174626	.175799	.176974	.178153
.28	.191296	.192507	.193720	.194937	.196155	.197377	.198601	.199827	.201056	.202288
.30	.216000	.217261	.218526	.219792	.221060	.222331	.223604	.224879	.226157	.227437
.32	.241664	.242971	.244280	.245590	.246904	.248219	.249536	.250855	.252177	.253500
.34	.268192	.269539	.270889	.272240	.273593	.274948	.276305	.277663	.279024	.280386
.36	.295488	.296871	.298256	.299643	.301031	.302421	.303812	.305205	.306600	.307996
.38	.323456	.324870	.326286	.327703	.329122	.330542	.331963	.333386	.334810	.336235
.40	.352000	.353441	.354882	.356325	.357769	.359215	.360661	.362109	.363557	.365007
.42	.381024	.382486	.383949	.385413	.386878	.388344	.389810	.391278	.392746	.394216
.44	.410432	.411911	.413390	.414870	.416351	.417833	.419315	.420798	.422281	.423765
.46	.440128	.441619	.443110	.444601	.446093	.447586	.449079	.450572	.452066	.453560
.48	.470016	.471514	.473012	.474510	.476008	.477507	.479005	.480504	.482003	.483593
.50	.500000	.501500	.503000	.504500	.506000	.507500	.509000	.510499	.511999	.513499
.52	.529984	.531481	.532979	.534476	.535972	.537469	.538965	.540461	.541956	.543451
.54	.559872	.561362	.562852	.564341	.565830	.567318	.568805	.570292	.571779	.573265
.56	.589568	.591046	.592523	.594000	.595476	.596951	.598425	.599898	.601371	.602843
.58	.618976	.620437	.621897	.623356	.624815	.626272	.627728	.629183	.630637	.632090
.60	.648000	.649439	.650878	.652315	.653750	.655185	.656618	.658050	.659481	.660910
.62	.676544	.677957	.679368	.680778	.682187	.683594	.684999	.686403	.687806	.689207
.64	.704512	.705894	.707273	.708652	.710028	.711403	.712776	.714147	.715516	.716884
.66	.731808	.733153	.734497	.735839	.737178	.738516	.739851	.741185	.742517	.743846
.68	.758336	.759641	.760943	.762243	.763541	.764837	.766130	.767422	.768711	.769997
.70	.784000	.785259	.786515	.787769	.789021	.790270	.791516	.792761	.794002	.795241
.72	.808704	.809912	.811118	.812321	.813521	.814719	.815914	.817106	.818295	.819482
.74	.832352	.833505	.834655	.835802	.836946	.838088	.839226	.840362	.841494	.842624
.76	.854848	.855941	.857031	.858117	.859201	.860281	.861358	.862432	.863502	.864570
.78	.876096	.877124	.878148	.879170	.880187	.881202	.882213	.883220	.884224	.885225
.80	.896000	.896958	.897913	.898864	.899811	.900755	.901695	.902631	.903564	.904493
.82	.914464	.915348	.916228	.917103	.917976	.918844	.919708	.920568	.921425	.922277
.84	.931392	.932196	.932997	.933793	.934585	.935373	.936157	.936936	.937712	.938483
.86	.946688	.947408	.948124	.948836	.949543	.950246	.950944	.951638	.952328	.953013
.90	.972000	.972538	.973070	.973598	.974121	.974640	.975153	.975662	.976165	.976664
.92	.981824	.982263	.982697	.983126	.983550	.983969	.984382	.984791	.985194	.985593
.94	.989632	.989968	.990298	.990623	.990943	.991258	.991567	.991871	.992169	.992462
.96	.995328	.995556	.995778	.995994	.996205	.996411	.996611	.996805	.996994	.997177
.98	.998816	.998931	.999040	.999143	.999240	.999332	.999417	.999497	.999571	.999640
1.00	1.000000									

Note: Coefficients apply for the volume of 2 ellipsoidal or hemispherical heads not the volume for 1 head.

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FIG. 6-26

Approximate Surface and Volume of Spheres

Diameter in Ft.	Surface of Sphere in Sq. Ft.	Volume of Sphere			Diameter in Ft.	Surface of Sphere in Sq. Ft.	Volume of Sphere		
		Cu. Ft.	U.S. Gals.	U.S. Bbls.			Cu. Ft.	U.S. Gals.	U.S. Bbls.
1	3.14	0.52	3.92	.09	61	11,690	118,847	889,037	21,168
2	12.57	4.19	31.33	.75	62	12,076	124,788	933,481	22,226
3	28.27	14.14	105.75	2.52	63	12,469	130,924	979,382	23,319
4	50.27	33.51	250.67	5.97	64	12,868	137,258	1,026,764	24,447
5	78.54	65.45	489.60	11.66	65	13,273	143,793	1,075,649	25,611
6	113.10	113.10	846.03	20.14	66	13,685	150,533	1,126,062	26,811
7	153.94	179.59	1,343.46	31.99	67	14,103	157,479	1,178,026	28,048
8	201.06	268.08	2,005.40	47.75	68	14,527	164,636	1,231,565	29,323
9	254.47	381.70	2,855.34	67.98	69	14,957	172,007	1,286,701	30,636
10	314.16	523.60	3,916.79	93.26	70	15,394	179,594	1,343,460	31,987
11	380	697	5,213	124	71	15,837	187,402	1,401,863	33,378
12	452	905	6,768	161	72	16,286	195,432	1,461,935	34,808
13	531	1,150	8,605	205	73	16,742	203,689	1,523,699	36,279
14	616	1,437	10,748	256	74	17,203	212,175	1,587,178	37,790
15	707	1,767	13,219	315	75	17,671	220,893	1,652,397	39,343
16	804	2,145	16,043	382	76	18,146	229,847	1,719,378	40,938
17	908	2,572	19,243	458	77	18,627	239,040	1,788,145	42,575
18	1,018	3,054	22,843	544	78	19,113	248,475	1,858,721	44,255
19	1,134	3,591	26,865	640	79	19,607	258,155	1,931,131	45,979
20	1,257	4,189	31,334	746	80	20,106	268,083	2,005,398	47,748
21	1,385	4,849	36,273	864	81	20,612	278,262	2,081,544	49,561
22	1,521	5,575	41,706	993	82	21,124	288,696	2,159,594	51,419
23	1,662	6,371	47,656	1,135	83	21,642	299,387	2,239,571	53,323
24	1,810	7,238	54,146	1,289	84	22,167	310,339	2,321,498	55,274
25	1,963	8,181	61,200	1,457	85	22,698	321,555	2,405,400	57,271
26	2,124	9,203	68,842	1,639	86	23,235	333,038	2,491,299	59,317
27	2,290	10,306	77,094	1,836	87	23,779	344,792	2,579,219	61,410
28	2,463	11,494	85,981	2,047	88	24,328	356,818	2,669,184	63,552
29	2,642	12,770	95,527	2,274	89	24,885	369,121	2,761,217	65,743
30	2,827	14,137	105,753	2,518	90	25,447	381,704	2,855,341	67,984
31	3,019	15,599	116,685	2,778	91	26,016	394,569	2,951,581	70,276
32	3,217	17,157	128,345	3,056	92	26,590	407,720	3,049,959	72,618
33	3,421	18,817	140,758	3,351	93	27,172	421,161	3,150,499	75,012
34	3,632	20,580	153,946	3,665	94	27,759	434,893	3,253,225	77,458
35	3,848	22,449	167,932	3,998	95	28,353	448,921	3,358,160	79,956
36	4,072	24,429	182,742	4,351	96	28,953	463,247	3,465,327	82,508
37	4,301	26,522	198,397	4,724	97	29,559	477,875	3,574,750	85,113
38	4,536	28,731	214,922	5,117	98	30,172	492,807	3,686,453	87,773
39	4,778	31,059	232,340	5,532	99	30,791	508,048	3,800,459	90,487
40	5,027	33,510	250,675	5,968	100	31,416	523,599	3,916,792	93,257
41	5,281	36,087	269,949	6,427	101	32,047	539,465	4,035,475	96,083
42	5,542	38,792	290,187	6,909	102	32,685	555,647	4,156,531	98,965
43	5,809	41,630	311,412	7,415	103	33,329	572,151	4,279,984	101,904
44	6,082	44,602	333,648	7,944	104	33,979	588,978	4,405,858	104,901
45	6,362	47,713	356,918	8,498	105	34,636	606,131	4,534,176	107,957
46	6,648	50,965	381,245	9,077	106	35,299	623,615	4,664,962	111,071
47	6,940	54,362	406,653	9,682	107	35,968	641,431	4,798,239	114,244
48	7,238	57,906	433,166	10,313	108	36,644	659,584	4,934,030	117,477
49	7,543	61,601	460,807	10,972	109	37,325	678,076	5,072,359	120,771
50	7,854	65,450	489,599	11,657	110	38,013	696,910	5,213,250	124,125
51	8,171	69,456	519,566	12,371	111	38,708	716,090	5,356,726	127,541
52	8,495	73,622	550,732	13,113	112	39,408	735,619	5,502,811	131,019
53	8,825	77,952	583,120	13,884	113	40,115	755,499	5,651,527	134,560
54	9,161	82,448	616,754	14,685	114	40,828	775,735	5,802,900	138,164
55	9,503	87,114	651,656	15,516	115	41,548	796,329	5,956,951	141,832
56	9,852	91,952	687,851	16,377	116	42,273	817,284	6,113,705	145,564
57	10,207	96,967	725,362	17,271	117	43,005	838,603	6,273,185	149,362
58	10,568	102,160	764,213	18,196	118	43,744	860,290	6,435,415	153,224
59	10,936	107,536	804,427	19,153	119	44,488	882,348	6,600,417	157,153
60	11,310	113,097	846,027	20,144	120	45,239	904,779	6,768,217	161,148

Note: If diameters are assumed as meters, values in columns "Surface of Sphere in Square Feet" and "Volume of Sphere — Cubic Feet" will represent Surface of Sphere in Square Meters and Volume of Sphere in Cubic Meters respectively.
Surface area of sphere = $3.141593\ D^2$ Square Feet

Volume of Sphere = $\begin{cases} 0.523599\ D^3 \text{ Cubic Feet} \\ 0.093257\ D^3 \text{ Barrels of 42 U.S. Gallons} \end{cases}$

Number of barrels of 42 U.S. Gallons at any inch in a true sphere = $(3d - 2h)\ h^2 \times 0.000\ 053\ 968\ 1$ where d is diameter of sphere and h is depth of liquid both in inches.

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FIG. 6-27

Partial Volumes of Spheres — U.S. Gallons

Diameter of Tank		Depth of Liquid, ft.															
ft	(in.)	2	4	6	8	10	12	14	16	18	20	25	30	35	40	45	50
2	(24)	31.3	–														
4	(48)	125.3	250.7	–													
6	(72)	219.3	626.7	846	–												
8	(96)	313.3	1003	1692	2005	–											
10	(120)	407.3	1329	2538	3509	3917	–										
12	(144)	501.3	1755	3384	5013	6267	6768	–									
14	(168)	595.4	2131	4230	6518	8617	10152	10748	–								
16	(192)	689.4	2507	5076	8022	10967	13536	15354	16043	–							
18	(216)	783.4	2883	5822	9526	13317	16920	19960	22059	22843	–						
20	(240)	877.4	3259	6768	11030	15667	20305	24566	38075	30457	31334	–					
25	(300)	1112	4199	8883	14790	21542	28765	36081	43116	49492	54832	51200	–				
30	(360)	1347	5139	10998	18550	27417	37225	47597	58156	68528	78336	97920	105753	–			
35	(420)	1582	6079	13113	22310	33293	45685	59112	73197	87564	101836	134639	158630	167932	–		
40	(480)	1817	7019	15228	26070	39168	54146	70627	88237	106599	125338	171359	211506	239903	250676	–	
45	(540)	2052	7959	17344	29830	45053	52606	82143	103278	125635	148838	208079	264383	311874	344677	356918	–
50	(600)	2287	8899	19459	33530	50918	71066	93658	118318	144670	172338	244799	317259	383845	438679	475889	489599

FIG. 6-28

Approximate Capacities (U.S. Gallons) of Rectangular Tanks for Each Foot of Liquid*

Tank Width, ft	Tank Length, ft											
	0.5	1.0	1.5	2.0	2.5	3.0	3.5	4.0	4.5	5.0	5.5	6.0
1	3.74	7.48	11.22	14.96	18.70	22.44	26.18	29.92	33.66	37.40	41.14	44.88
2	7.48	14.96	22.44	29.92	37.40	44.88	52.36	59.84	67.32	74.81	82.29	89.77
3	11.22	22.44	33.66	44.88	56.10	67.32	78.55	89.77	100.99	112.21	123.43	134.65
4	14.96	29.92	44.88	59.84	74.81	89.77	104.73	119.69	134.65	149.61	164.57	179.53
5	18.70	37.40	56.10	74.81	93.51	112.21	130.91	149.61	168.31	187.01	205.71	224.42
6	22.44	44.88	67.32	89.77	112.21	134.65	157.09	179.53	201.97	224.42	246.86	269.30
7	26.18	52.36	78.55	104.73	130.91	157.09	183.27	209.45	235.64	261.82	288.00	314.18
8	29.92	59.84	89.77	119.69	149.61	179.53	209.45	239.38	269.30	299.22	329.14	359.06
Tank Width, ft	**Tank Length, ft**											
	6.5	**7.0**	**7.5**	**8.0**	**8.5**	**9.0**	**9.5**	**10.0**	**10.5**	**11.0**	**11.5**	**12.0**
1	48.62	52.36	56.10	59.84	63.58	67.32	71.06	74.81	78.55	82.29	86.03	89.77
2	97.25	104.73	112.21	119.69	127.17	134.65	142.13	149.61	157.09	164.57	172.05	179.53
3	145.87	157.09	168.31	179.53	190.75	201.97	213.19	224.42	235.64	246.86	258.08	269.30
4	194.49	209.45	224.42	239.38	254.34	269.30	284.26	299.22	314.18	329.14	344.10	359.07
5	243.12	261.82	280.52	299.22	317.92	336.62	355.32	374.03	392.73	411.43	430.13	448.83
6	291.74	314.18	336.62	359.06	381.51	403.95	426.39	448.83	471.27	493.71	516.16	538.60
7	340.36	366.55	392.73	418.91	445.09	471.27	497.45	523.64	549.82	576.00	602.18	628.36
8	388.99	418.91	448.83	478.75	508.68	538.60	568.52	598.44	628.36	658.29	688.21	718.13

1 cu ft = 7.4805 U.S. gal.

Total Volume = $W \cdot L \cdot H_T$ = cu ft

Partial Volume = $W \cdot L \cdot H_p$ = cu ft

* For imperial gallons, divide above capacities by 1.2

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API Standard 2000
Venting Atmospheric and Low-Pressure Storage Tanks.

API RP 12 RI
Recommended Practice for Setting, Connecting, Maintenance and Operation of Lease Tanks.

API RP 50
Recommended Gas Plant Good Operating Practices for Protection of Environment.

API RP 200
Fire Protection for Refineries.

ASME Code for Unfired Pressure Vessels, Section VIII, Division I & II.

AWWA D-100
Welded Tanks

AWWA D-103
Bolted Tanks

BS7777
Flat Bottomed, Vertical Cylindrical Tanks for Low Temperature Service.

Federal Register, Part 1910, Occupational Safety & Health Standards, Subpart D, Walking-Working Surfaces.

GPA North American Storage Capacity for Light Hydrocarbons and U.S. LP-Gas Import Terminals.

IOCC Underground Storage of Liquid Petroleum Hydrocarbons in the United States. (Interstate Oil Compact Commission.)

National Association of Corrosion Engineers.
Item No. 51101 - Electrochemical Techniques for Corrosion.
Item No. 52044 - Coatings and Linings for Immersion Service.

NACE - TPC Publication No. 5
Corrosion Control in Petroleum Production.

National Board of Fire Underwriters (NBFU)

National Fire Protection Association (NFPA)
No. 11, 30, and 20-26

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5. Underwriters Laboratories (UL)
No. 142 Steel Above Ground Tanks
No. 58 Steel Underground Tanks

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SECTION 7

Separation Equipment

PRINCIPLES OF SEPARATION

Three principles used to achieve physical separation of gas and liquids or solids are momentum, gravity settling, and coalescing. Any separator may employ one or more of these principles, but the fluid phases must be "immiscible" and have different densities for separation to occur.

FIG. 7-1

Nomenclature

- A = area, ft^2
- A_p = particle or droplet cross sectional area, ft^2
- C = empirical constant for separator sizing, ft/hr
- C^* = empirical constant for liquid-liquid separators, (bbl • cp)/(ft^2 • day)
- C' = drag coefficient of particle, dimensionless (Fig. 7-3)
- D_i = separator inlet nozzle diameter, in.
- D_p = droplet diameter, ft
- D_v = inside diameter of vessel, ft
- G_m = maximum allowable gas mass-velocity necessary for particles of size D_p to drop or settle out of gas, lb/(hr • ft^2)
- g = acceleration due to gravity, 32.2 ft/sec^2
- H_l = width of liquid interface area, ft
- J = gas momentum, lb/(ft • sec^2)
- K = empirical constant for separator sizing, ft/sec
- K_{CR} = proportionality constant from Fig. 7-5 for use in Eq 7-5, dimensionless
- L = seam to seam length of vessel, ft
- L_l = length of liquid interface area, ft
- M = mass flow, lb/sec
- M_p = mass of droplet or particle, lb
- MW = molecular weight, lb/lb mole
- P = system pressure, psia
- Q = estimated gas flow capacity, MMscfd per ft^2 of filter area
- Q_A = actual gas flow rate, ft^3/sec
- R = gas constant, 10.73 (psia • ft^3)/(°R • lb mole)
- Re = Reynolds number, dimensionless
- S_{hl} = specific gravity of heavy liquid, water = 1.0
- S_{ll} = specific gravity of light liquid, water = 1.0
- T = system temperature, °R
- t = retention time, minutes
- U = volume of settling section, bbl
- V_t = critical or terminal gas velocity necessary for particles of size D_p to drop or settle out of gas, ft/sec
- W = total liquid flow rate, bbl/day
- W_{cl} = flow rate of light condensate liquid, bbl/day
- Z = compressibility factor, dimensionless

Greek:

- ρ_g = gas phase density, lb/ft^3
- ρ_l = liquid phase density, droplet or particle, lb/ft^3
- μ = viscosity of continuous phase, cp

Filter Separators: A filter separator usually has two compartments. The first compartment contains filter-coalescing elements. As the gas flows through the elements, the liquid particles coalesce into larger droplets and when the droplets reach sufficient size, the gas flow causes them to flow out of the filter elements into the center core. The particles are then carried into the second compartment of the vessel (containing a vane-type or knitted wire mesh mist extractor) where the larger droplets are removed. A lower barrel or boot may be used for surge or storage of the removed liquid.

Flash Tank: A vessel used to separate the gas evolved from liquid flashed from a higher pressure to a lower pressure.

Line Drip: Typically used in pipelines with very high gas-to-liquid ratios to remove only free liquid from a gas stream, and not necessarily all the liquid. Line drips provide a place for free liquids to separate and accumulate.

Liquid-Liquid Separators: Two immiscible liquid phases can be separated using the same principles as for gas and liquid separators. Liquid-liquid separators are fundamentally the same as gas-liquid separators except that they must be designed for much lower velocities. Because the difference in density between two liquids is less than between gas and liquid, separation is more difficult.

Scrubber or Knockout: A vessel designed to handle streams with high gas-to-liquid ratios. The liquid is generally entrained as mist in the gas or is free-flowing along the pipe wall. These vessels usually have a small liquid collection section. The terms are often used interchangeably.

Separator: A vessel used to separate a mixed-phase stream into gas and liquid phases that are "relatively" free of each other. Other terms used are scrubbers, knockouts, line-drips, and decanters.

Slug Catcher: A particular separator design able to absorb sustained in-flow of large liquid volumes at irregular intervals. Usually found on gas gathering systems or other two-phase pipeline systems. A slug catcher may be a single large vessel or a manifolded system of pipes.

Three Phase Separator: A vessel used to separate gas and two immiscible liquids of different densities (e.g. gas, water, and oil).

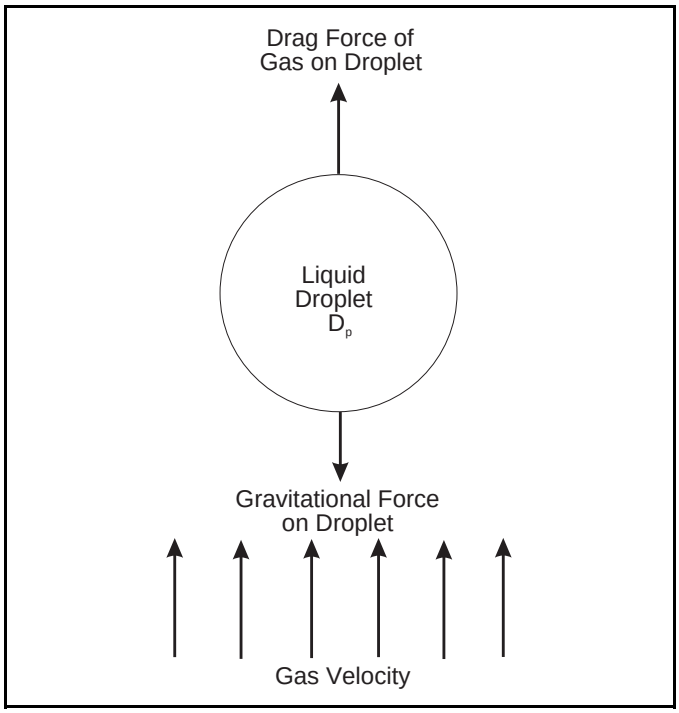
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Gravity Settling

Liquid droplets will settle out of a gas phase if the gravitational force acting on the droplet is greater than the drag force of the gas flowing around the droplet (see Fig. 7-2). These forces can be described mathematically using the terminal or finite-settling velocity calculation, Eq 7-1. The nomenclature for all equations in this section and terminology used are listed in Fig. 7-1.

FIG. 7-2

Forces on Liquid Droplet in Gas Stream



$$V_t = \sqrt{\frac{2\ g\ M_p\ (\rho_l - \rho_g)}{\rho_l\ \rho_g\ A_p\ C'}} = \sqrt{\frac{4\ g\ D_p\ (\rho_l - \rho_g)}{3\ \rho_g\ C'}} \qquad \textbf{Eq 7-1}$$

The drag coefficient has been found to be a function of the shape of the particle and the Reynolds number of the flowing gas. For the purpose of this equation, particle shape is considered to be a solid, rigid sphere. The Reynolds number is defined as:

$$Re = \frac{1{,}488\ D_p\ V_t\ \rho_g}{\mu} \qquad \textbf{Eq 7-2}$$

Fig. 7-3 shows the relationship between drag coefficient and particle Reynolds number for spherical particles.

In this form, a trial and error solution is required since both particle size (D_p) and terminal velocity (V_t) are involved. To avoid trial and error, values of the drag coefficient are presented in Fig. 7-4 as a function of the product of drag coefficient (C') times the Reynolds number squared; this technique eliminates velocity from the expression.[1] The abscissa of Fig. 7-4 is given by:

$$C'\ (Re)^2 = \frac{(0.95)\ (10^8)\ \rho_g\ D_p^{\ 3}\ (\rho_l - \rho_g)}{\mu^2} \qquad \textbf{Eq 7-3}$$

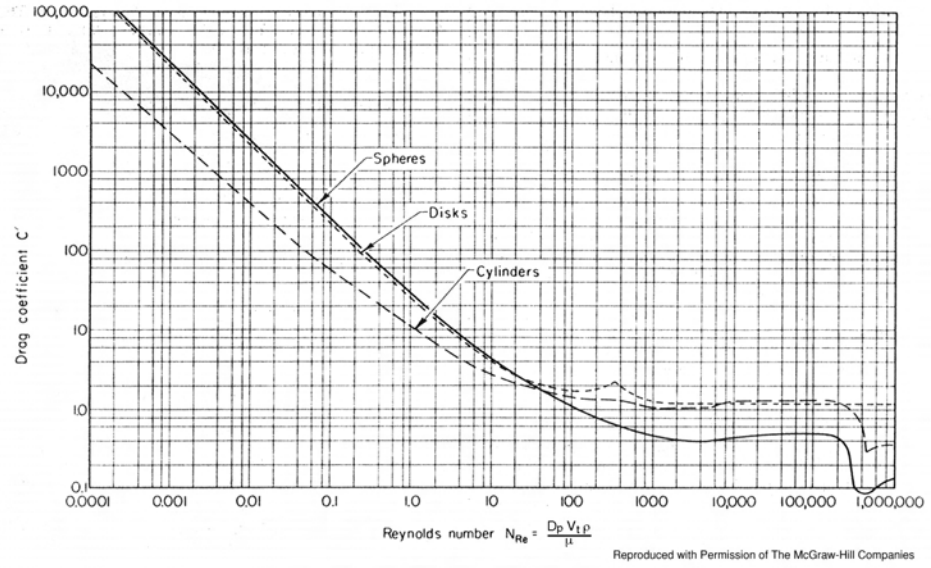
As with other fluid flow phenomena, the gravity settling drag coefficient reaches a limiting value at high Reynolds numbers.

As an alternative to using Eq 7-3 and Fig. 7-4, the following approach is commonly used.

The curve shown in Fig. 7-3 can be simplified into three sections from which curve-fit approximations of the C' vs Re curve can be derived. When these expressions for C' vs Re are substituted into Eq 7-1, three settling laws are obtained as described below.

FIG. 7-3

Drag Coefficient and Reynolds Number for Spherical Particles[1]

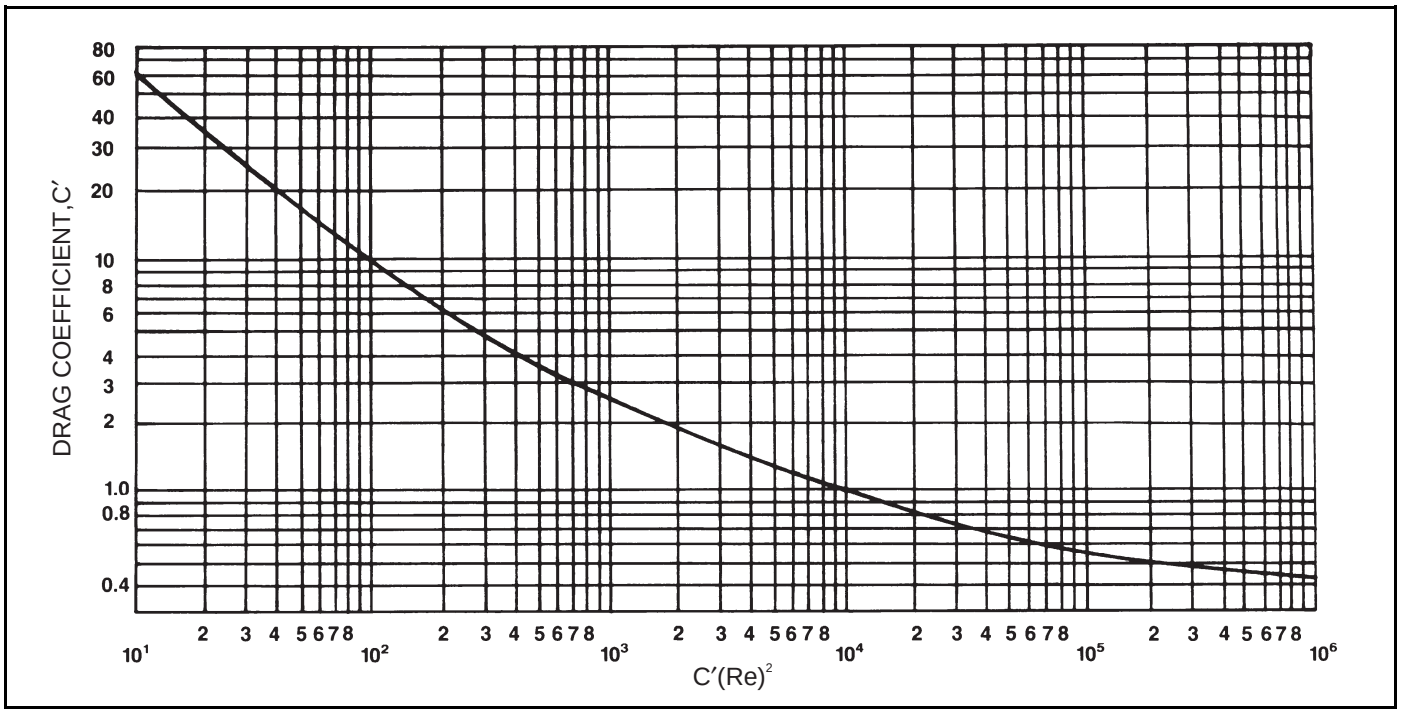


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FIG. 7-4

Drag Coefficient of Rigid Spheres[2]



Stoke's Law

At low Reynolds numbers (less than 2), a linear relationship exists between the drag coefficient and the Reynolds number (corresponding to laminar flow). Stoke's Law applies in this case and Eq 7-1 can be expressed as:

$$V_t = \frac{1{,}488\ g\ D_p^2\ (\rho_l - \rho_g)}{18\ \mu} \qquad \textbf{Eq 7-4}$$

The droplet diameter corresponding to a Reynolds number of 2 can be found using a value of 0.025 for K_{CR} in Eq 7-5.

$$D_p = K_{CR}\left[\frac{\mu^2}{g\ \rho_g\ (\rho_l - \rho_g)}\right]^{0.33} \qquad \textbf{Eq 7-5}$$

A summary of these equations is presented in Fig. 7-5, which also provides general information regarding droplet sizes and collection equipment selection guidelines.

By inspection of the particle Reynolds number equation (Eq 7-2) it can be seen that Stoke's law is typically applicable for small droplet sizes and/or relatively high viscosity liquid phases.

Intermediate Law

For Reynold's numbers between 2 and 500, the Intermediate Law applies, and the terminal settling law can be expressed as:

$$V_t = \frac{3.49\ g^{0.71}\ D_p^{1.14}\ (\rho_l - \rho_g)^{0.71}}{\rho_g^{0.29}\ \mu^{0.43}} \qquad \textbf{Eq 7-6}$$

The droplet diameter corresponding to a Reynolds number of 500 can be found using a value of 0.334 for K_{CR} in Eq 7-5.

The intermediate law is usually valid for many of the gas-liquid and liquid-liquid droplet settling applications encountered in the gas business.

Newton's Law

Newton's Law is applicable for a Reynold's number range of approximately 500 – 200,000, and finds applicability mainly for separation of large droplets or particles from a gas phase, e.g. flare knockout drum sizing. The limiting drag coefficient is approximately 0.44 at Reynolds numbers above about 500. Substituting C′ = 0.44 in Eq 7-1 produces the Newton's Law equation expressed as:

$$V_t = 1.74\sqrt{\frac{g\ D_p\ (\rho_l - \rho_g)}{\rho_g}} \qquad \textbf{Eq 7-7}$$

An upper limit to Newton's Law is where the droplet size is so large that it requires a terminal velocity of such magnitude that excessive turbulence is created. For the Newton's Law region, the upper limit to the Reynolds number is 200,000 and K_{CR} = 18.13.

DROPLET SIZE DISTRIBUTIONS AND ENTRAINMENT LOADINGS

Liquid Drop Sizes in Gas-Liquid Systems

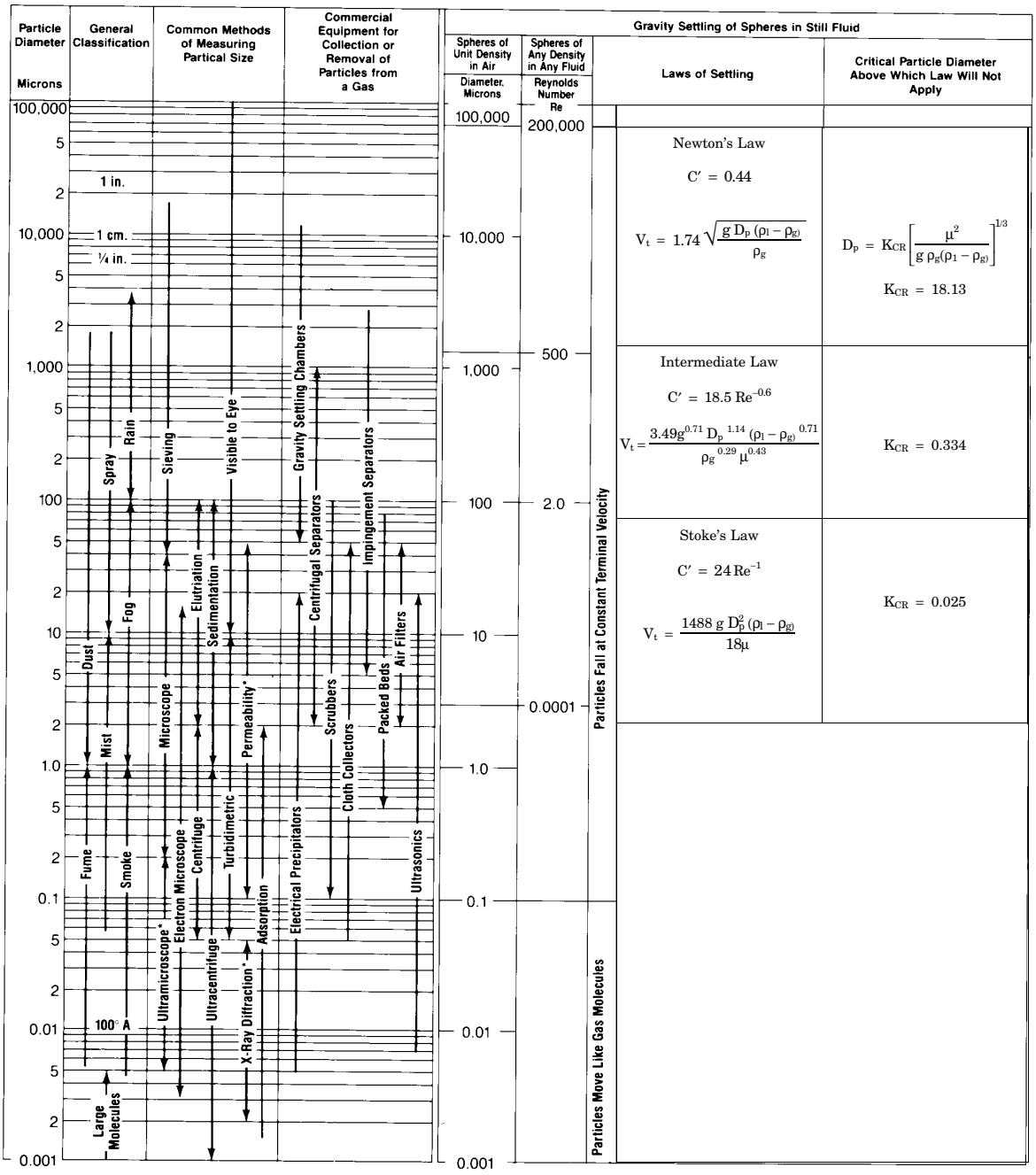
The gravity settling theory above provides valuable insight into the significance of certain physical properties and the physics that together influence the separation of dispersed droplets from a continuous phase, e.g. liquid droplets from a gas or liquid droplets of a specific density from a liquid phase of another density. However, a couple of problems remain:

- Determination of the actual droplet sizes that need to be dealt with.
- Determination of the amount of entrained liquid in droplet form.

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FIG. 7-5

Gravity Settling Laws and Particle Characteristics



Adapted From Ref. 7-3

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These issues apply to both gas-liquid and liquid-liquid separations.

Neither of these are subject to precise calculation and yet equipment separation performance is dependent on both. As a result, the sizing of separation equipment, vapor-liquid and liquid-liquid, is still largely based on empirical methods. While progress is being made in this area, there is some way to go before droplet size distribution and entrainment loading prediction will be fully incorporated into separation equipment sizing procedures. Fig. 7-6 provides an indication of typical drop size distributions for various gas-liquid systems.

TYPES OF SEPARATORS

Separators are usually characterized by orientation as vertical or horizontal. They may be further classified as two-phase (gas-liquid) or three-phase (gas-liquid-liquid). Horizontal separators can be single- or double-barrel and can be equipped with sumps or boots.

Parts of a Separator

Regardless of shape, separation vessels usually contain four major sections plus the necessary controls. These sections are shown for horizontal and vertical vessels in Fig. 7-7. The inlet device (A) is used to reduce the momentum of the inlet flow stream, perform an initial bulk separation of the gas and liquid phases, and enhance gas flow distribution. There are a variety of inlet devices available and these are discussed in more detail in a later section.

The gas gravity separation section (B) is designed to utilize the force of gravity to separate entrained liquid droplets from the gas phase, preconditioning the gas for final polishing by the mist extractor. It consists of a portion of the vessel through which the gas moves at a relatively low velocity with little turbulence. In some horizontal designs, straightening vanes are used to reduce turbulence. The vanes also act as droplet coalescers, which reduces the horizontal length required for droplet removal from the gas stream.

The liquid gravity separation section (C) acts as a receiver for all liquid removed from the gas in the inlet, gas gravity, and mist extraction sections. In two-phase separation applications, the liquid gravity separation section provides residence time

FIG. 7-6

Typical Partical Size Distribution Ranges From Entrainment Caused By Various Mechanisms

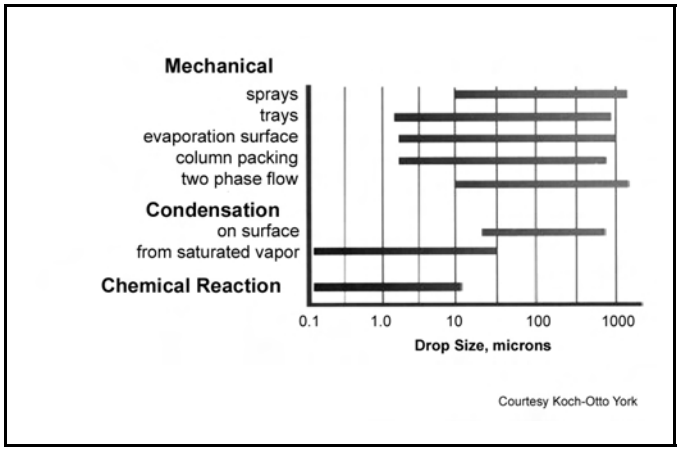
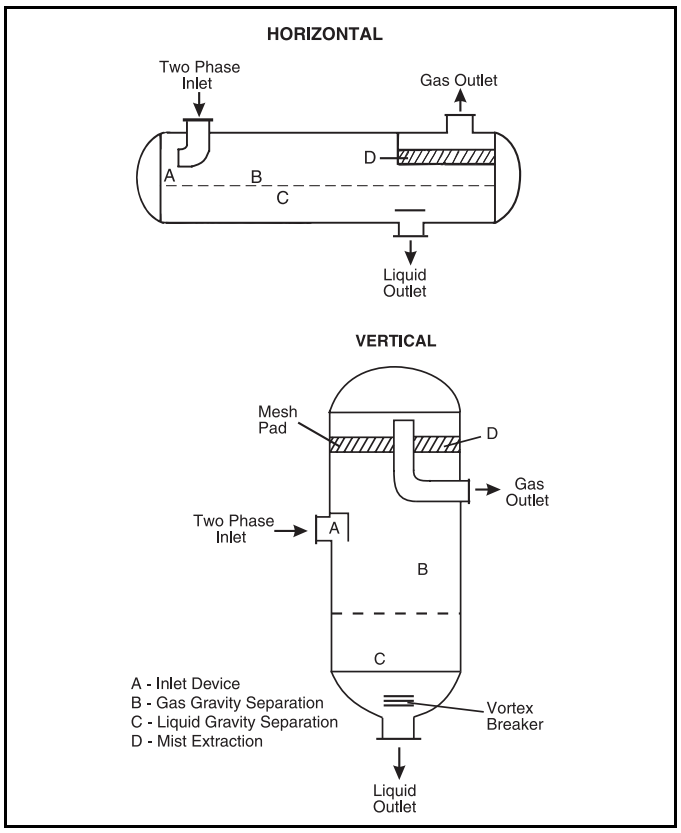


FIG. 7-7

Gas-Liquid Separators



for degassing the liquid. In three-phase separation applications the liquid gravity section also provides residence time to allow for separation of water droplets from a lighter hydrocarbon liquid phase and vice-versa. Depending on the inlet flow characteristics, the liquid section should have a certain amount of surge volume, or slug catching capacity, in order to smooth out the flow passed on to downstream equipment or processes. Efficient degassing may require a horizontal separator while emulsion separation may also require higher temperature, use of electrostatic fields, and/or the addition of a demulsifier. Coalescing packs are sometimes used to promote hydrocarbon liquid – water separation, though they should not be used in applications that are prone to plugging, e.g. wax, sand, etc.

The mist extraction section (D) utilizes a mist extractor that can consist of a knitted wire mesh pad, a series of vanes, or cyclone tubes. This section removes the very small droplets of liquid from the gas by impingement on a surface where they coalesce into larger droplets or liquid films, enabling separation from the gas phase. Quoted liquid carryover from the various types of mist extraction devices are usually in the range of 0.1 - 1 gal/MMscf.

Separator Configurations

Factors to be considered for separator configuration selection include:

- What separation quality is required by downstream equipment and processes?
- How well will extraneous material (e.g. sand, mud, corrosion products) be handled?

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- How much plot space will be required?
- Will the separator be too tall for transport if skidded?
- Is there enough interface surface for 3-phase separation (e.g. gas/hydrocarbon/glycol liquid)?
- Can heating coils or sand jets be incorporated if required?
- How much surface area is available for degassing of separated liquid?
- Must surges in liquid flow be handled without large changes in level?
- Is large liquid retention volume necessary?
- What are the heat retention requirements (e.g. freeze protection)?

Vertical Separators

Vertical separators, shown in Fig. 7-8, are usually selected when the gas-liquid ratio is high or total gas volumes are low. In a vertical separator, the fluids enter the vessel through an inlet device whose primary objectives are to achieve efficient bulk separation of liquid from the gas and to improve flow distribution of both phases through the separator. Liquid removed by the inlet device is directed to the bottom of the vessel. The gas moves upward, usually passing through a mist extractor to remove any small entrained liquid droplets, and then the vapor phase flows out of the vessel. Liquid removed by the mist extractor is coalesced into larger droplets that then fall through the gas to the liquid reservoir in the bottom. The ability to handle liquid slugs is typically obtained by increasing vessel height to accommodate additional surge volume. Level control is normally not highly critical and liquid level can fluctuate several inches without affecting the separation performance or capacity of the vessel. Except for knockout drum applications, mist extractors are normally used to achieve a low liquid content in the separated gas in vessels of reasonable diameter.

Typical vertical separator L/D ratios are normally in the 2–4 range.

As an example of a vertical separator, consider a compressor suction scrubber. In this service the vertical separator:

- Does not need significant liquid retention volume
- A properly designed liquid level control loop responds quickly to any liquid that enters, thus avoiding tripping an alarm or shutdown
- The separator occupies a small amount of plot space

Horizontal Separators

Horizontal separators are most efficient when large volumes of liquid are involved. They are also generally preferred for three-phase separation applications. In a horizontal separator, shown in Fig. 7-9, the liquid that has been separated from the gas moves along the bottom of the vessel to the liquid outlet. The gas and liquid occupy their proportionate shares of the shell cross-section. Increased slug capacity is obtained through shortened retention time and increased liquid level. Fig. 7-9 also illustrates the separation of two liquid phases (glycol and hydrocarbon). The denser glycol settles to the bottom and is withdrawn through the boot. The glycol level is controlled by an interface level control instrument.

Horizontal separators have certain advantages with respect to gravity separation performance in that the liquid droplets or gas bubbles are moving perpendicular to the bulk phase velocity, rather than directly against it as in vertical flow, which makes separation easier.

In a double-barrel separator, the liquids fall through connecting flow pipes into the external liquid reservoir below. Slightly smaller vessels may be possible with the double-barrel horizontal separator, where surge capacity establishes the size of the lower liquid collection chamber.

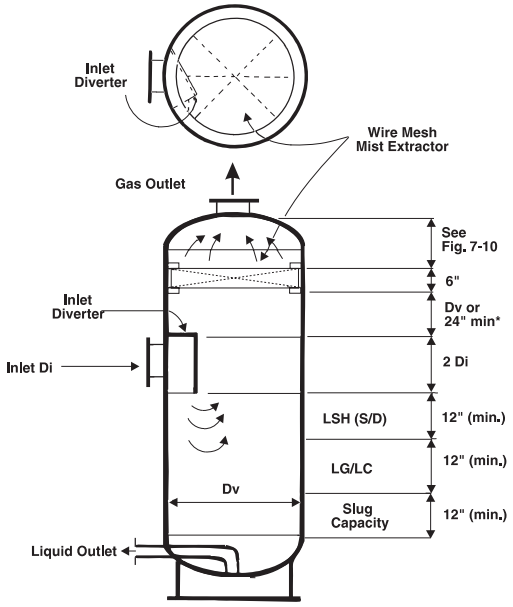
Typical L/D ratios for horizontal separators normally fall in the range of 2.5–5.

As an example of a horizontal separator, consider a rich amine flash tank. In this service:

- There is relatively large liquid surge volume required for longer retention time. This allows more complete release of the dissolved gas and, if necessary, surge volume for the circulating system.
- There is more surface area per liquid volume to aid in more complete degassing.
- The horizontal configuration handles a foaming liquid better than vertical.
- The liquid level responds slowly to changes in liquid inventory, providing steady flow to downstream equipment.

FIG. 7-8

Vertical Separator with Wire Mesh Mist Extractor



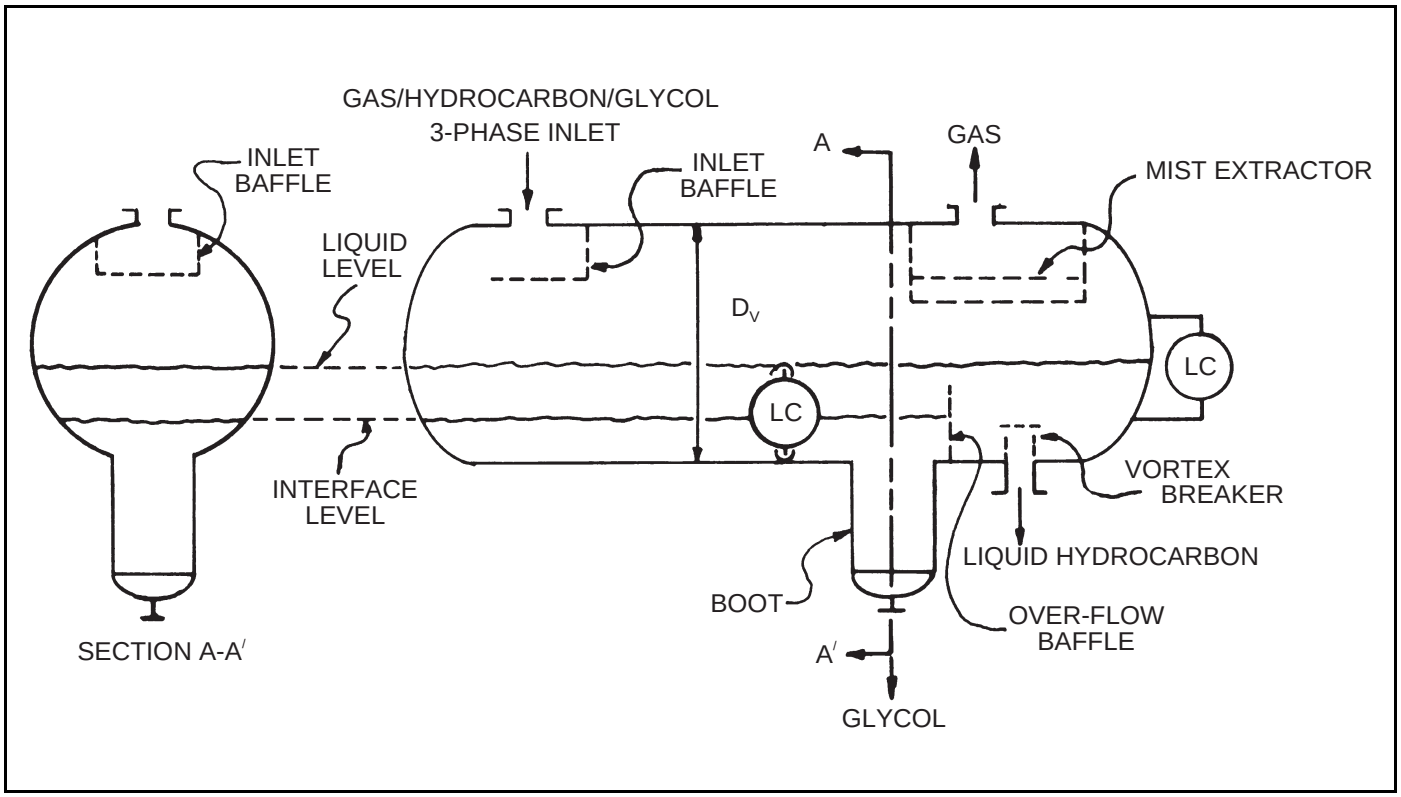
LSH (S/D) Level Switch High (Shutdown)
LG/LC Level Gauge/Level Controller

- Dimensions may be influenced by instrument connection requirements.
- For small diameter separators (≤ 48 inch ID.) with high L/G inlet flow ratios this dimension should be increased by as much as 50%.
- May use syphon type drain to:
  A. reduce vortex possibility
  B. reduce external piping that requires heating (freeze protection)

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FIG. 7-9

Horizontal Three-Phase Separator with Wire Mesh Mist Extractor



VESSEL INTERNALS

Inlet Devices

The importance of the inlet device with respect to separation performance has been identified only relatively recently, mainly through the use of Computational Fluid Dynamics (CFD) modeling. The main functions of the inlet device are:

- Reduce the momentum of the inlet stream and enhance flow distribution of the gas and liquid phases.
- Efficient separation of the bulk liquid phase.
- Prevent droplet shattering and re-entrainment of bulk liquid phase.

There are several different types of separator inlet devices that are commonly used:

- no inlet device
- diverter plate
- half-pipe
- vane-type
- cyclonic

In addition to the inlet device itself, it has been determined that the inlet piping configuration is also important. The vane-type and cyclonic inlet devices generally provide improved separation performance compared to the others. The separator/inlet device manufacturer should be contacted for specific design/performance details.

Mist Extraction Equipment

Mist extractors are used to separate the small liquid droplets from the gas phase that were not removed by the inlet device or gas gravity settling section (main "body") of the separator. These droplets are typically less than 150–500 micron in size and usually much smaller. It is generally not economic to separate these droplets by gravity alone by making the separator larger. The different types of mist extractors use principles other than simple droplet settling by gravity to achieve efficient removal of small droplets.

Wire-Mesh

Wire-mesh mist extractors, or pads, are made by knitting wire, metal or plastic, into tightly packed layers which are then crimped and stacked to achieve the required pad thickness. If removal of very small droplets, i.e. less than 10 micron, is required, much finer fibers may be interwoven with the primary mesh to produce a co-knit pad. Mesh pads remove liquid droplets mainly by impingement of droplets onto the wires and/or co-knit fibers followed by coalescence into droplets large enough to disengage from the bottom of the pad and drop through the rising gas flow into the liquid holding part of the separator. Mesh pads are not recommended for dirty or fouling service as they tend to plug easily. Wire mesh pads are normally installed horizontally with gas flow vertically upwards through the pad. Performance is adversely affected if the pad is tilted more than 30 degrees from the horizontal.[4] Problems have been encountered where liquid flow through the pad to the sump is impaired due to dirt or sludge accumulation, causing a higher liquid level on one side. This provides the serious potential of the pad being dislodged from its mounting brackets, making it useless or forcing parts of it into the outlet pipe. Firmly secure the top and bottom of the pad so that it is not dislodged by high gas flows, such as when a pressure relief valve lifts or during an emergency blowdown situation. Fig-

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ures 7-10 and 7-11 illustrate typical wire mesh installations in vertical and horizontal vessels.

Most installations will use a 6-inch thick pad with 9-12 lb/ft^3 bulk density. Minimum recommended pad thickness is 4 inches[4]. Manufacturers should be contacted for specific designs.

Separation Performance — There are two main aspects to mesh pad separation performance.

- droplet removal efficiency
- gas handling capacity

Droplet removal efficiency is typically given by the manufacturer as a curve showing % removal as a function of droplet size at design flow and a nominal liquid loading. These curves are usually based on tests of an air-water system at atmospheric pressure.

The gas capacity of mesh pads is almost universally specified by a load or sizing factor, K, as utilized in the Souders and Brown[5] equation given below:

$$V_t = K\sqrt{\frac{\rho_l - \rho_g}{\rho_g}} \qquad \textbf{Eq 7-8}$$

The required mist extractor area is obtained from

$$A = \frac{Q_A}{V_t} \qquad \textbf{Eq 7-9}$$

The design K value provides a certain degree of margin before liquid entrainment/carryover becomes excessive. Efficiency and capacity are normally inversely related, i.e. as droplet removal efficiency increases, allowable gas throughput decreases.

Because normal pressure drops across mesh pads are so low (less than 1 inch of water) this is not typically a major area of concern in gas processing operations. Fig. 7-12 provides a summary of performance parameters.

The K capacity factor for mesh pads is often derated for higher pressure operation, Fig. 7-13. All factors being equal, this is normally due to the reduction in surface tension of the liquid phase that occurs with increasing pressure.

Mesh pads normally operate efficiently over a range of 30–110% of the design gas rate.

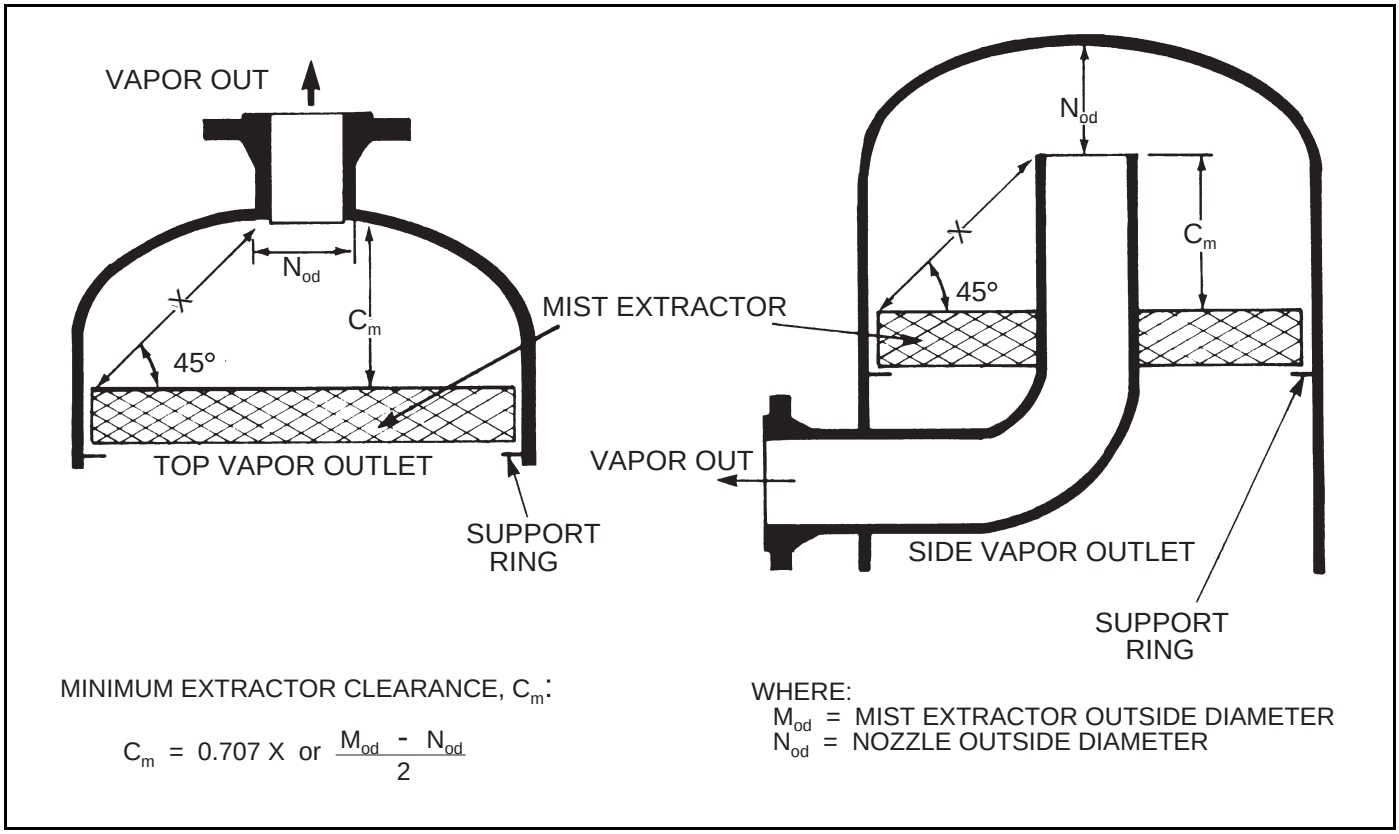
The gas capacity of a wire-mesh pad is typically defined in terms of a K "constant" as given in Fig. 7-12. This is an oversimplification. Among other things, K is also a function of the amount of entrained liquid reaching the mesh pad. As would be expected, K decreases with increasing inlet liquid loading. For typical mesh pad designs, liquid loads greater than 1 gpm/ft^2 are considered high and will require deration of the standard K factor, to prevent excessive entrainment carryover. However, per the earlier discussion on droplet size distributions, it is difficult to predict what the inlet liquid loading reaching the mist extractor will be for a given separator application.

Vane

Vane or chevron-type mist extractors (vane-pack) use relatively closely spaced blades arranged to provide sinusoidal or zig-zag gas flow paths. The changes in gas flow direction com-

FIG. 7-10

Minimum Clearance — Mesh Type Mist Eliminators



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FIG. 7-11

Horizontal Separator with Knitted Wire Mesh Pad Mist Extractor and Lower Liquid Barrel

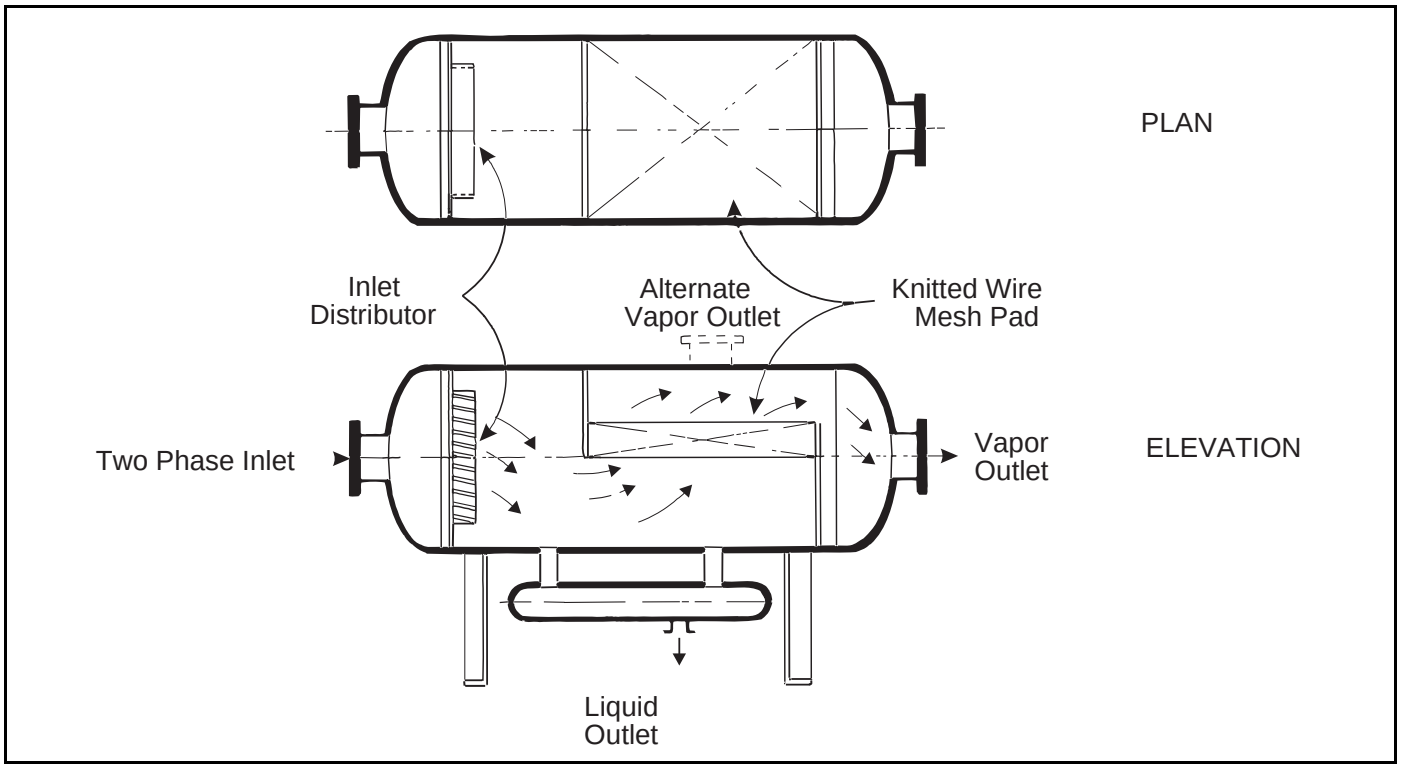


FIG. 7-12

Mesh Pad Separation Performance

Droplet removal efficiency:	99–99.5% removal of 3–10 micron droplets. Higher removal efficiency is for denser, thicker pads and/or smaller wire/co-knit fiber diameter.
Gas capacity, K, ft/sec	0.22–0.39. Generally, the lower capacities correspond to the mesh pad designs with the highest droplet removal efficiencies.

FIG. 7-13

Adjustment of K Factor for Pressure[6]

Pressure, psig	Percent of Design Value
Atmospheric	100
150	90
300	85
600	80
1,150	75

bined with the inertia of the entrained liquid droplets, cause impingement of the droplets onto the plate surface, followed by coalescence and drainage of the liquid to the liquid collection section of the separator. Fig. 7-14 shows a typical vane-type mist extractor. Vane packs may be installed in either horizontal or vertical orientations, though capacity is typically reduced significantly for vertical upflow applications. Recently developed hollow vane designs with interconnected liquid drainage passages are capable of high gas handling capacities in a vertical upflow orientation. Vanes differ from wire mesh pads in that they typically do not drain the separated liquid back through the rising gas stream. Rather, the liquid can be routed into a downcomer that carries the fluid directly to the liquid holding section of the separator. Vane packs are better suited to dirty or fouling service as they are less likely to plug due to their relatively large flow passages. A vertical separator with a typical vane mist extractor is shown in Fig. 7-15.

A number of different vane pack designs are available. Pack thicknesses are generally in the range of 6–12 inches. Vanes are usually arranged in a zig-zag or sinusoidal pattern, with vane spacings of 1–1.5 inches typical. Vane types include no-pocket, single-pocket and double-pocket styles.

Separation Performance— As for mesh pads, the key performance parameters are droplet removal efficiency and gas handling capacity. Eq 7-8 and the load/sizing factor K, can also be utilized for calculating the capacity of vane-type mist extractors. Fig. 7-16 provides a summary of performance parameters.

Vane packs typically have pressure drops in the range of 0.5–3.5 inches of water.

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FIG. 7-14

Cross-Section of Vane Element Mist Extractor Showing Corrugated Plates with Liquid Drainage Traps

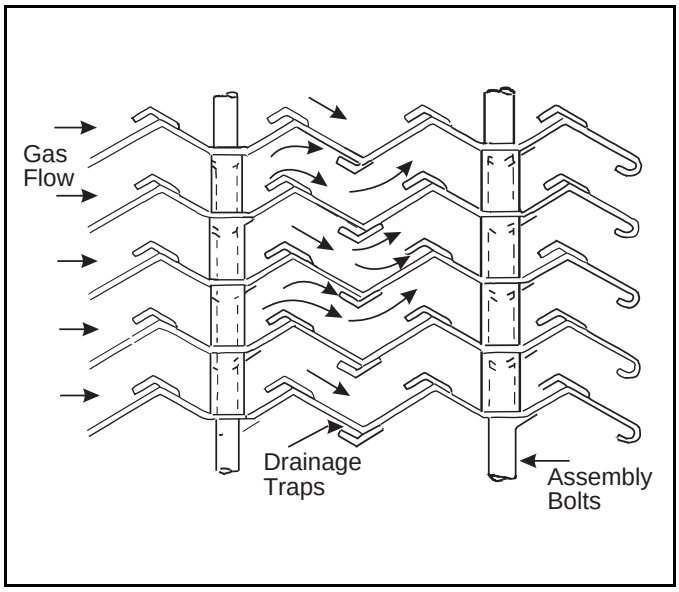
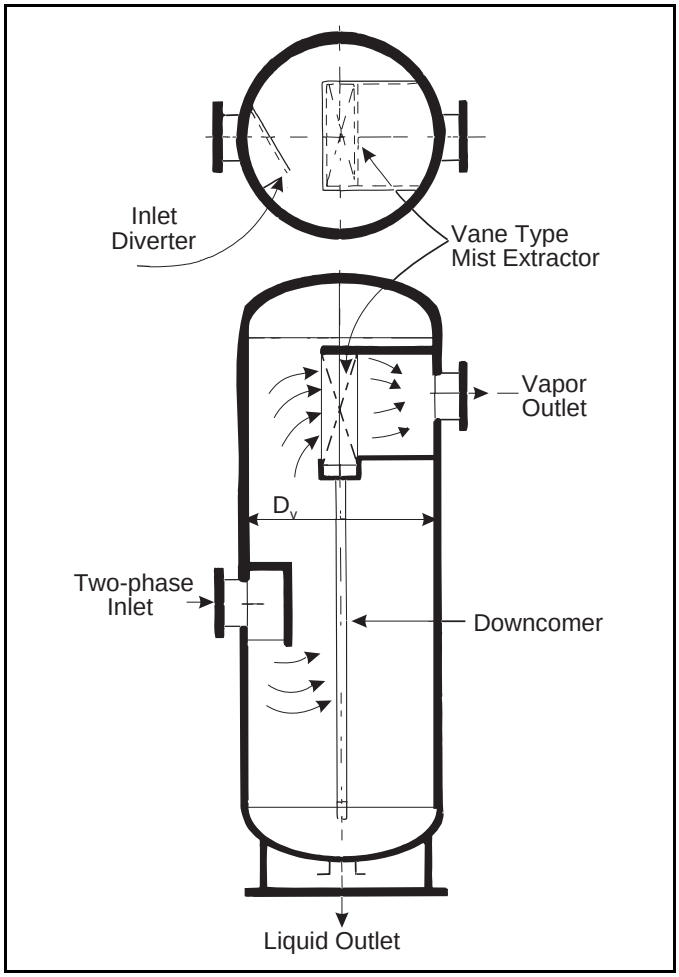


FIG. 7-15

Vertical Separator with Vane-Type Mist Extractor



As for wire mesh extractors, the droplet removal efficiencies quoted above for vane-type units are typically based on tests performed on air-water systems at atmospheric pressure. Testing has shown that for mesh type extractors, the low pressure air-water droplet removal efficiency results correlate reasonably well to higher pressure gas-hydrocarbon liquid systems. Vane packs on the other hand show a drop-off of removal efficiency as pressure increases. This is primarily a result of the decreasing allowable gas velocity with increasing pressure caused mainly by increased gas density. As gas velocity decreases, droplet inertia decreases and the droplets tend to follow the gas streamlines more easily through the vane passages, and exit the vane pack without being captured. Mesh pads also rely on velocity/droplet inertia to remove liquid droplets via impingement but they are less susceptible to capture efficiency reduction than vane packs because mesh pads have far more collection "targets", i.e. wire/fiber filaments.

Turndown is generally more of an issue with vane-packs, with droplet removal efficiency decreasing measurably as velocity decreases from design.

Vane-type mist extractors are also impacted by inlet liquid loading, but generally have considerably more tolerance towards liquids than mesh-pads. The manufacturer should be contacted for specific designs and applications.

FIG. 7-16

Typical Vane Pack Separation Performance

Droplet removal efficiency:	99% removal of droplets greater than 10–40 microns. Higher removal efficiency is for thicker packs, with closer vane spacings and more passes (bends).
Gas capacity, K, ft/sec	Horizontal flow: 0.9–1.0 Vertical up-flow: 0.4–0.5 The higher capacities are generally associated with pocketed vane designs.

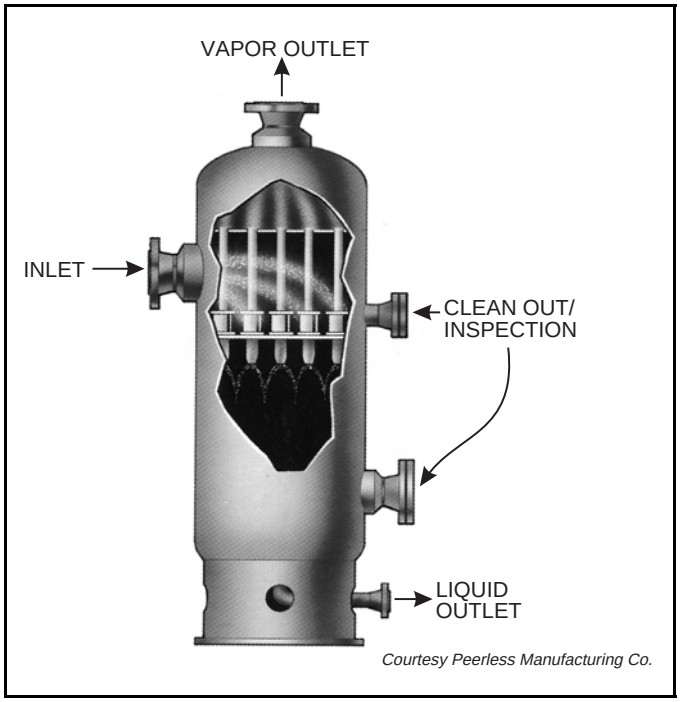
Cyclonic

There are several types of centrifugal separators that serve to separate entrained liquids, and solids if present, from a gas stream. For mist extraction applications, reverse-flow, axial-flow and recycling axial-flow cyclones are typically used in multi-cyclone "bundles." Cyclonic mist extractors use centrifugal force to separate solids and liquid droplets from the gas phase based on density difference. Very high G forces are achieved which allows for efficient removal of small droplet sizes. The main advantage of cyclonic mist extractors is that they provide good removal efficiency at very high gas capacity. This generally allows for the smallest possible vessel diameter for a given gas flow. Cyclonic mist extractors are often used in low liquid load gas scrubbing applications, and for high pressure gas-liquid separation. These devices are proprietary and cannot be readily sized without detailed knowledge of the characteristics of the specific internals. The manufacturer of such devices should be consulted for assistance in sizing these types of separators. A typical centrifugal separator is shown in Fig. 7-17. Disadvantages of centrifugal separators are:

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FIG. 7-17

Vertical Separator with Centrifugal Elements



Courtesy Peerless Manufacturing Co.

- Some designs do not handles slugs well
- Pressure drop tends to be significantly higher than for vane or clean-knitted mesh mist extractors
- They have a relatively narrow operating flow range for highest efficiency

Separation Performance — As stated, the selection and design of de-misting cyclone bundles should be left to the manufacturers. While not a correct representation of the separation physics employed by cyclonic devices, the familiar K factor as used in Eq 7-8 can be used to provide indicative gas capacity for a multicyclone mist extractor. K factors will tend to range from 1 ft/sec for reverse flow multicyclones to 3 ft/sec or higher for the newest recycling axial-flow multicyclones. The liquid handling capacity of a multicyclone bundle is typically somewhat higher than that for vane-type mist extractors. The gas capacity factor, K, is based on the multicyclone bundle cross-sectional area, assuming 2 inch cyclone tubes and typical cyclone–cyclone pitch dimensions and layout arrangement.

GAS-LIQUID SEPARATOR SIZING

Specifying Separators

Separator designers need to know pressure, temperature, flow rates, and physical properties of the streams as well as the degree of separation required. It is also prudent to define if these conditions all occur at the same time or if there are only certain combinations that can exist at any time. If known, the type and amount of liquid should also be given, and whether it is mist, free liquid, or slugs.

For example, a compressor suction scrubber designed for 70-150 MMscfd gas at 400-600 psig and 65-105°F would require a unit sized for the worse conditions, i.e. 150 MMscfd at 400 psig and 105°F. But if the real throughput of the compressor varies from 150 MMscfd at 600 psig, 105°F to 70 MMscfd at 400 psig, 65°F then a smaller separator is acceptable because the high volume only occurs at the high pressure.

An improperly sized separator is one of the leading causes of process and equipment problems. Inlet separation problems upstream of absorption systems (e.g. amine and glycol) can lead to foaming problems, and upstream of adsorption systems (e.g. molecular sieve, activated alumina, and silica gel) can cause fouling, coking, and other damage to the bed. Equipment such as compressors and turbo-expanders tolerate little or no liquid in the inlet gas steam, while pumps and control valves may have significant erosion and/or cavitation when vapors are present due to improper separation. In addition, direct-fired reboilers in amine and glycol service may experience tube failures due to hot spots caused by salt deposits caused by produced water carryover into the feed gas.

Design Approach

There is as much art as there is science to properly design a separator. Three main factors should be considered in separator sizing: 1) vapor capacity, 2) liquid capacity, and 3) operability. The vapor capacity will determine the cross-sectional area necessary for gravitational forces to remove the liquid from the vapor. The liquid capacity is typically set by determining the volume required to provide adequate residence time to “de-gas” the liquid or allow immiscible liquid phases to separate. Operability issues include the separator’s ability to deal with solids if present, unsteady flow/liquid slugs, turndown, etc. Finally, the optimal design will usually result in an aspect ratio that satisfies these requirements in a vessel of reasonable cost. These factors often result in an iterative approach to the calculations.

VAPOR HANDLING

Separators without Mist Extractors

Separators without mist extractors are not frequently utilized. The most common application of a vapor-liquid separator that does not use a mist extractor is a flare knockout drum. Mist extractors are rarely used in flare knockout drums because of the potential for plugging and the serious implications this would have for pressure relief. is typically a horizontal vessel that utilizes gravity as the sole mechanism for separating the liquid and gas phases. Gas and liquid enter through the inlet nozzle and are slowed to a velocity such that the liquid droplets can fall out of the gas phase. The dry gas passes into the outlet nozzle and the liquid is drained from the lower section of the vessel.

To design a separator without a mist extractor, the minimum size diameter droplet to be removed must be set. Typically this diameter is in the range of 300 to 2,000 microns (1 micron = 10^{-4} cm or 0.00003937 inch).

The length of the vessel required can then be calculated by assuming that the time for the gas flow from inlet to outlet is the same as the time for the liquid droplet of size D_p to fall from the top of the vessel to the liquid surface. Eq 7-10 then relates the length of the separator to its diameter as a function of this settling velocity (assuming no liquid retention):

$$L = \frac{4\,Q_A}{\pi\,V_t\,D_v} \qquad \textbf{Eq 7-10}$$

If the separator is to be additionally used for liquid storage, this must also be considered in sizing the vessel.

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Example 7-1—A horizontal gravity separator (without mist extractor) is required to handle 60 MMscfd of 0.75 specific gravity gas (MW = 21.72) at a pressure of 500 psig and a temperature of 100°F. Compressibility is 0.9, viscosity is 0.012 cp, and liquid specific gravity is 0.5. It is desired to remove all entrainment greater than 150 microns in diameter. No liquid surge is required.

Gas density, $\rho_g = \frac{P\,(MW)}{RTZ} = \frac{(514.7)\,(21.72)}{(10.73)\,(560)\,(0.90)}$

$= 2.07\ lb/ft^3$

Liquid density, $\rho_l = 0.5\,(62.4) = 31.2\ lb/ft^3$

Mass flow, $M = \frac{(60)\,(10^6)\,(21.72)}{(379)\,(24)\,(3600)} = 39.8\ lb/sec$

Particle diameter, $D_p = \frac{(150)\,(0.00003937)}{12} = 0.000492\ ft$

From Eq 7-3,

$$C'\,(Re)^2 = \frac{(0.95)\,(10^8)\,\rho_g\,D_p^3\,(\rho_l - \rho_g)}{\mu^2}$$

$$= \frac{(0.95)\,(10^8)\,(2.07)\,(0.000492)^3\,(31.2 - 2.07)}{(0.012)^2}$$

$= 4738$

From Fig. 7-4, Drag coefficient, $C' = 1.40$

Terminal velocity, $V_t = \sqrt{\frac{4\,g\,D_p\,(\rho_l - \rho_g)}{3\,\rho_g\,C'}}$

$$= \sqrt{\frac{4\,(32.2)\,(0.000492)\,(29.13)}{3\,(2.07)\,1.40}}$$

$= \sqrt{0.212} = 0.46\ ft/sec$

Gas flow, $Q_A = \frac{M}{\rho_g} = \frac{39.80}{2.07} = 19.2\ ft^3/sec$

Assume a diameter, $D_v = 3.5\ ft$

Vessel length, $L = \frac{4\,Q_A}{\pi\,V_t\,D_V} = \frac{4\,(19.2)}{\pi\,(0.46)\,(3.5)}$

$= 15.2\ ft$

Other reasonable solutions are as follows:

Diameter, feet	Length, feet
3.5	15.2
4.0	13.3
4.5	11.8
5.0	10.6

Example 7-2—What size vertical separator without a mist extractor is required to meet the conditions used in Example 7-1?

$A = \frac{Q_A}{V_t} = \frac{19.2}{0.46} = 41.7\ ft^2$

$D_v = 7.29\ ft$ minimum

Use a 90-inch ID vertical separator.

Separators with Mist Extractors

Of the four major components of a separator that were discussed in a previous section, the mist extractor has the most impact on separated gas quality with respect to carried over liquid content. The sizing equations and parameters provided in the mist extraction section size the mist extractor itself, not the actual separation vessel. The gas capacities of the various types of mist extractors is generally inversely related to the amount of entrained liquid that the mist extractor is required to remove.

- the amount of liquid in the separator feed gas.
- the inlet flow condition of the feed, i.e. multiphase flow pattern, .
- the type of inlet device used.
- the sizing/dimensions of the gas gravity separation section of the separator.

For lightly liquid loaded separation applications, e.g. less than 10-15 BBL/MMSCF (gas scrubbing application), the mist extractor performance will be controlling and will generally dictate the cross-sectional area requirements of the gas gravity separation section. For a vertical separator, this will determine the vessel diameter. A horizontal vessel would not typically be used in a lightly loaded gas-liquid separation application. Good flow distribution to the mist extractor is still required.

In more heavily liquid loaded separation applications, a vertical or horizontal separator configuration may be chosen depending on the specific conditions. In either case, some "pre-conditioning" of the gas phase, specifically reduction of the gas entrainment loading, may be required ahead of the final mist extraction element.

Vertical Separators

Historically, the gas handling capacity of conventional vertical separators that employ mist extractors has normally been calculated from the Souders and Brown equation, Eq 7-8, using "experience-based" K factors. Typical K values for vertical separators from API 12J[7] are presented in Fig. 7-18.

In qualitative terms, the ranges of K given above may be taken to reflect difficulty of the separation conditions, i.e. from non-ideal/difficult to ideal/easy. As indicated in Fig. 7-18, K is also a function of vessel height. This reflects the fact that a certain minimum distance is required to establish a relatively uniform velocity profile before the gas reaches the mist extractor. Theoretically, it is not simply the vessel height that is important with respect to velocity profile, but the vertical height between the inlet device and the mist extractor. As gas handling capacity is based on an allowable limit for liquid carryover into the separated gas stream, and the final liquid removal element is the mist extractor, the mist extractor has a significant influence on the K value used for separator sizing.

The vertical height of the vessel is also influenced by the liquid handling requirements and general vessel layout criteria as indicated in Fig. 7-8. Typically, the liquid phase will occupy the lower third of the vessel height.

A design that optimizes the inlet feed flow condition and utilizes an efficient inlet device, may provide enough feed gas

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FIG. 7-18

Typical Values of K for Vertical Separators

Height, feet	K, ft/sec
5	0.12 – 0.24
10 or taller	0.18 – 0.35

* assumes vessel is equipped with a wire-mesh mist extractor

pre-conditioning to allow the vessel diameter to be sized equivalent to the mist extractor. However, traditionally the method typically used has been to "oversize" the gas gravity section, i.e. vessel diameter, relative to the mist extractor. This has generally been done in two ways:

- Derate the mist extractor K factor and use this reduced K value in Eq 7-8 to determine the vessel diameter. Guidelines as to determination of the appropriate deration factor are not well defined. A relatively low liquid loading application with steady flow, a low inlet velocity and a good inlet device should require minimal deration of the extractor K factor, i.e. deration factor approximately equal to 1. On the other hand, an application with significant liquid volumes, unsteady flow, high velocity inlet and a simple diverter plate inlet device may require a deration factor of 0.5. Normally, it would be more economic to improve the inlet flow condition/device than to significantly oversize the vessel relative to the mist extractor requirements.
- Select a separable droplet size and size for the vessel diameter using Eq 7-1. A droplet size of 150 microns has been typically specified. This may be overly conservative for vane pack and cyclonic mist extractors, which generally have higher gas and liquid capacities than mesh pads.

Comparison of Eq 7-8 & Eq 7-1, indicates that

$$K = \sqrt{\frac{4_g D_P}{3C'}} \qquad \textbf{Eq 7-11}$$

This shows the approximate equivalence of the empirical K and the more theoretical droplet separation sizing methods. However, the value of K in Eq 7-8 as used in practice depends on other factors besides droplet size, drag coefficient, and liquid entrainment loading, including: type of internals, unsteady flow, surface tension, liquid viscosity, foaming, gas velocity profile uniformity, degree of separation required, etc. Additional duration may be required to account for these factors.

Horizontal Separators

Eq 7-8 can also be used for calculating the gas capacity of horizontal separators. However, some modifications are required to reflect the fact that in the gas gravity separation section of a horizontal separator, the liquid droplets are falling perpendicular to the gas flow rather than in direct opposition as occurs in a vertical separator. This makes it easier to separate droplets in a horizontal vessel. Partially offsetting this advantage is the fact that in a horizontal separator, the liquid gravity separation section is occupying part of the vessel cross section, leaving reduced area for gas flow.

In calculating the gas capacity of horizontal separators, the cross-sectional area of that portion of the vessel occupied by liquid (at maximum level) is subtracted from the total vessel cross-sectional area. Typical horizontal separator designs will have the normal liquid level at the half-full point. Values of K for horizontal separators from API 12J are given in Fig. 7-19.

There is some disagreement as to how K should vary with separator length. The API 12J recommendation is shown in Fig. 7-19. Many separators are greater than 10 feet in length, with some reaching 50 feet or more. The relationship shown in Fig. 7-19 for adjusting for length will give K factors greater than 1 ft/sec for large separators. These higher values of K for large (long) horizontal separators are generally considered to be overly optimistic. In practice, K = 0.5 ft/sec is normally used as an upper limit for horizontal separators equipped with wire-mesh mist extractors. Separators equipped with vane-type or cyclonic mist extractors may utilize higher K values than those for mesh pads.

The same general principles as discussed for vertical separators apply for horizontal separators with mist extractors in high liquid loading applications. For a horizontal separator, mesh pad and cyclonic type mist extractors will normally be installed horizontally with vertical upflow, while vane pack may be installed horizontally, vertically, or sometimes in a vee-pattern. Additionally, in a horizontal separator, the liquid droplets are settling perpendicular to the gas flow which makes separation easier. For these reasons, the approach of derating the mist extractor K to calculate the cross-sectional area of the gas gravity section is not as straightforward as for a vertical vessel. Typically the required cross-sectional area of the gas gravity section of a horizontal separator is sized based on droplet settling theory. The procedure is similar to that discussed previously for separators without mist extractors. For vessels with mesh pad mist extractors a typical droplet size for design is 150 microns. For separators equipped with vane-type or cyclonic mist extractors, a larger drop size may be appropriate, which may allow for a smaller vessel. The vessel manufacturer should be consulted.

FIG. 7-19

Values of K for Horizontal Separators

Length, ft	K, ft/sec
10	0.40 – 0.50
Other	$K_{10}\left(\frac{L}{10}\right)^{0.56}$

* assumes vessel is equipped with a wire-mesh mist extractor

In calculating the gas capacity of horizontal separators, the cross-sectional area of that portion of the vessel occupied by liquid (at maximum level) is subtracted from the total vessel cross-sectional area

LIQUID HANDLING

The design criterion for separator liquid handling capacity is typically based on the following two main considerations:

- Liquid degassing requirements.
- Process control/stability requirements.

Generally, one or the other of these factors will dictate. Liquid capacity is typically specified in terms of residence time, which must be translated into vessel layout requirements for

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dimensioning purposes. Residence time establishes the separator volume required for the liquid as shown in Eq 7-12:

$$U = \frac{W(t)}{1440} \qquad \textbf{Eq 7-12}$$

Typical residence times are shown in Figures 7-20 and 7-21.

Note that except for the Natural gas – condensate application, the residence times specified in Fig. 7-20 are primarily based on process control stability/operability.

These values are primarily intended to reflect liquid degassing requirements. In practice, process control stability and operability requirements will often override the degassing requirements.

Vessel layout recommendations, including liquid handling requirements, are given in Fig. 7-8 and 7-9 for vertical and horizontal separators, respectively.

The retention time requirements given in Figures 7-20 and 7-21 are not specific to vessel orientation. However, the liquid degassing process actually involves the separation of gas bubbles from the liquid phase, which under ideal conditions can be described by the gravity settling equation, Eq 7-1. Similar to liquid droplet settling out of the gas phase, it is easier for a gas bubble to rise perpendicularly through the moving liquid in a horizontal separator than directly against the downflowing liquid in a vertical vessel. Theoretically, for equal liquid residence times, the horizontal separator should be slightly more efficient at degassing. However, this has not typically been an issue in practice. If it is deemed necessary to calculate vessel liquid handling requirements for a degassing constraint according to gravity settling theory, a gas bubble size of 150–200 microns has been suggested by several sources.

FIG. 7-20

Typical Retention Times for Gas/Liquid Separator

Application	Retention Time, minutes
Natural Gas – Condensate separation	2 – 4
Fractionator Feed Tank	10 – 15
Reflux Accumulator	5 – 10
Fractionation Column Sump	2
Amine Flash Tank	5 – 10
Refrigeration Surge Tank	5
Refrigeration Economizer	3
Heat Medium Oil Surge Tank	5 – 10

FIG. 7-21

API 12J Retention Times for Gas-Oil Separators

Oil Gravity	Liquid Retention Time, min
> 35	1
20 – 35	1 to 2
10 – 20	2 to 4

Three-Phase and Liquid-Liquid Separation

The gas handling requirements for three-phase separation are dealt with in a similar manner as discussed for two-phase separation. Traditionally, sizing for liquid-liquid separation has involved specification of liquid residence times.

Fig. 7-22 provides suggested residence times for various liquid-liquid separation applications. These figures generally assume equal residence times for both the light and heavy liquid phases.

While the residence time approach for liquid-liquid separation equipment design has been widely used in industry for years, it does have some limitations.

- the typical approach of assuming equal residence times for both liquid phases may not be optimum, e.g. It is generally much easier to separate oil droplets from water than vice-versa. Settling theory (Eq 7-1) explains this as being due to the lower viscosity of water compared to oil.
- Residence times do not take into account vessel geometry, i.e. 3 minutes residence time in the bottom of a tall, small diameter vertical vessel will not achieve the same separation performance as 3 minutes in a horizontal separator, again according to droplet settling theory.
- The residence time method does not provide any direct indication as to the quality of the separated liquids, e.g. amount of water in the hydrocarbon or the amount of hydrocarbon in the water. Droplet settling theory can not do this either in most cases, but there is some empirical data available which allows for approximate predictions in specific applications.

FIG. 7-22

Typical Retention Times for Liquid-Liquid Separation

Type of Separation	Retention Time, minutes
Hydrocarbon/Water Separators[7]	
Above 35° API hydrocarbon	3 – 5
Below 35° API hydrocarbon	
100°F and above	5 – 10
80°F	10 – 20
60°F	20 – 30
Ethylene Glycol/Hydrocarbon[8] Separators (Cold Separators)	20 – 60
Amine/Hydrocarbon Separators[9]	20 – 30
Coalescer, Hydrocarbon/Water Separators[9]	
100°F and above	5 – 10
80°F	10 – 20
60°F	20 – 30
Caustic/Propane	30 – 45
Caustic/Heavy Gasoline	30 – 90

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Removal of very small droplets may require the use of specialized internals or the application of electrostatic fields to promote coalescence.

Liquid-liquid separation may be divided into two broad categories of operation. The first is defined as "gravity separation," where the two immiscible liquid phases separate within the vessel by the differences in density of the liquids. Sufficient retention time must be provided in the separator to allow for the gravity separation to take place. The second category is defined as "coalescing separation." This is where small particles of one liquid phase must be separated or removed from a large quantity of another liquid phase. Different types of internal construction of separators much be provided for each type of liquid-liquid separators. The following principles of design for liquid-liquid separation apply equally for horizontal or vertical separators. Horizontal vessels have some advantage over verticals for liquid-liquid separation, due to the larger interface area available in the horizontal style, and the shorter distance particles must travel to coalesce.

There are two factors that may prevent two liquid phases from separating due to differences in specific gravity:

- If droplet particles are so small that they may be suspended by Brownian movement. This is defined as a random motion that is greater than directed movement due to gravity for particles less than 0.1 micron in diameter.
- The droplets may carry electric charges due to dissolved ions. These charges can cause the droplets to repel each other rather than coalesce into larger particles and settle by gravity.

Effects due to Brownian movement are usually small and proper chemical treatment will usually neutralize any electric charges. Then settling becomes a function of gravity and viscosity in accordance with Stoke's Law. The settling velocity of spheres through a fluid is directly proportional to the difference in densities of the sphere and the fluid, and inversely proportional to the viscosity of the fluid and the square of the diameter of the sphere (droplet), as noted in Eq 7-3. The liquid-liquid separation capacity of separators may be determined from Equations 7-13 and 7-14, which were derived from Equation 7-3.[9] Values of C* are found in Fig. 7-23.

Vertical vessels:

$$W_{cl} = C^* \left(\frac{S_{hl} - S_{ll}}{\mu} \right) (0.785)\, D_v^2 \qquad \textbf{Eq 7-13}$$

Horizontal vessel:

$$W_{cl} = C^* \left(\frac{S_{hl} - S_{ll}}{\mu} \right) L_l\, H_l \qquad \textbf{Eq 7-14}$$

Since the droplet size of one liquid phase dispersed in another is usually unknown, it is simpler to size liquid-liquid separation based on retention time of the liquid within the separator vessel. For gravity separation of two liquid phases, a large retention or quiet settling section is required in the vessel. Good separation requires sufficient time to obtain an equilibrium condition between the two liquid phases at the temperature and pressure of separation. The liquid capacity of a separator or the settling volume required can be determined from Eq 7-12 using the retention time given in Fig. 7-22.

The following example shows how to size a liquid-liquid separator.

FIG. 7-23

Values of C* Used in Equations 7-13 and 7-14

Emulsion Characteristic	Droplet Diameter, microns	Constant,[10] C*
Free Liquids	200	1,100
Loose Emulsion	150	619
Moderate Emulsion	100	275
Tight Emulsion	60	99

Example 7-3—Determine the size of a vertical separator to handle 600 bpd of 55° API condensate and 50 bpd of produced water. Assume the water particle size is 200 microns. Other operating conditions are as follows:

Operating temperature = 80°F

Operating pressure= 1,000 psig

Water specific gravity = 1.01

Condensate viscosity = 0.55 cp @ 80°F

Condensate specific gravity for 55° API = 0.76

From Eq 7-13, $W_{cl} = C^*[(S_{hl} - S_{ll}) /] \times 0.785 \times D_v^2$

From Fig. 7-23 for free liquids with water particle diameter = 200 microns, C* = 1,100

600 bbl/day = 1,100 x [1.01 – 0.76) / 0.55] x 0.785 x D_v^2

$D_v^2 = 660 / 392.5 = 1.53$ ft^2

$D_v = 1.24$ feet

Using a manufacturer's standard size vessel might result in specifying a 20-inch OD separator.

Using the alternate method of design based on retention time as shown in Equation 7-12 should give:

$V_1 = ql(t) / 1440$

From Fig. 7-22, use 3 minutes retention time

$V_1 = 650 \times 3 / 1440 = 1.35$ bbl

$V_1 = 1.35 \times 42 = 56.7$ gallons

Assuming a 20-inch diameter and 1,480 psig working pressure, a vessel would be made from a 1.031-inch wall seamless pipe which holds 13.1 gal/ft. The small volume held in the bottom head can be discounted in this size vessel. The shell height required for the retention volume required would be:

Shell height = V_1 / vol/ft = 56.7 / 13.1 = 4.3 feet

This would require a 20 inch OD x 10 foot separator to give sufficient surge room above the liquid settling section for any vapor-liquid separation.

Another parameter that should be checked when separating amine or glycol from liquid hydrocarbons is the interface area between the two liquid layers. This area should be sized so the glycol or amine flow across the interface does not exceed approximately 2,000 gal/day/ft^2.

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The above example indicates that a relatively small separator would be required for liquid-liquid separation. It should be remembered that the separator must also be designed for the vapor capacity to be handled. In most cases of high vapor-liquid loadings that are encountered in gas processing equipment design, the vapor capacity required will dictate a much larger vessel than would be required for the liquid load only. The properly designed vessel has to be able to handle both the vapor and liquid loads. Therefore, one or the other will control the size of the vessel used.

FILTER SEPARATORS AND COALESCING FILTERS

General

There are two main types of filtration equipment used in gas-liquid separation service in the gas processing industry: the filter separator or "filter sep" and the coalescing filter. Both equipment types are of proprietary design, and the manufacturer should be contacted for detailed selection and sizing.

Filter Separator

Filter separators are available in horizontal and vertical orientations, with horizontal the most common. Fig. 7-24 shows a horizontal filter separator. This type of separator is often used for solids and liquid removal in relatively low liquid loading applications. A filter separator is a two-stage device. Gas enters the inlet nozzle and passes through the filter section, where solid particles are filtered from the gas stream and liquid particles are coalesced into larger droplets. Any free liquids are also removed in the first section. The coalesced droplets pass through the filter riser tubes and are carried into the second section of the separator, where a final mist extraction element removes these droplets from the gas stream. Flow through the filter elements is from an outside-to-inside direction. A pressure drop of 1-2 psi is normal in a clean filter separator. If solids are present, it will normally be necessary to replace the filter elements at regular intervals. A 10 psi pressure drop criteria is often used for filter changeout. Removal of the filters is achieved via a quick-opening closure.

The second stage of a filter separator contains a mist extraction device. As for a conventional separator this may be a mesh pad, vane pack or multicyclone bundle. The same issues regarding mist extractor selection criteria, sizing, etc. apply as discussed previously. Mesh pads and vane pack are most commonly utilized.

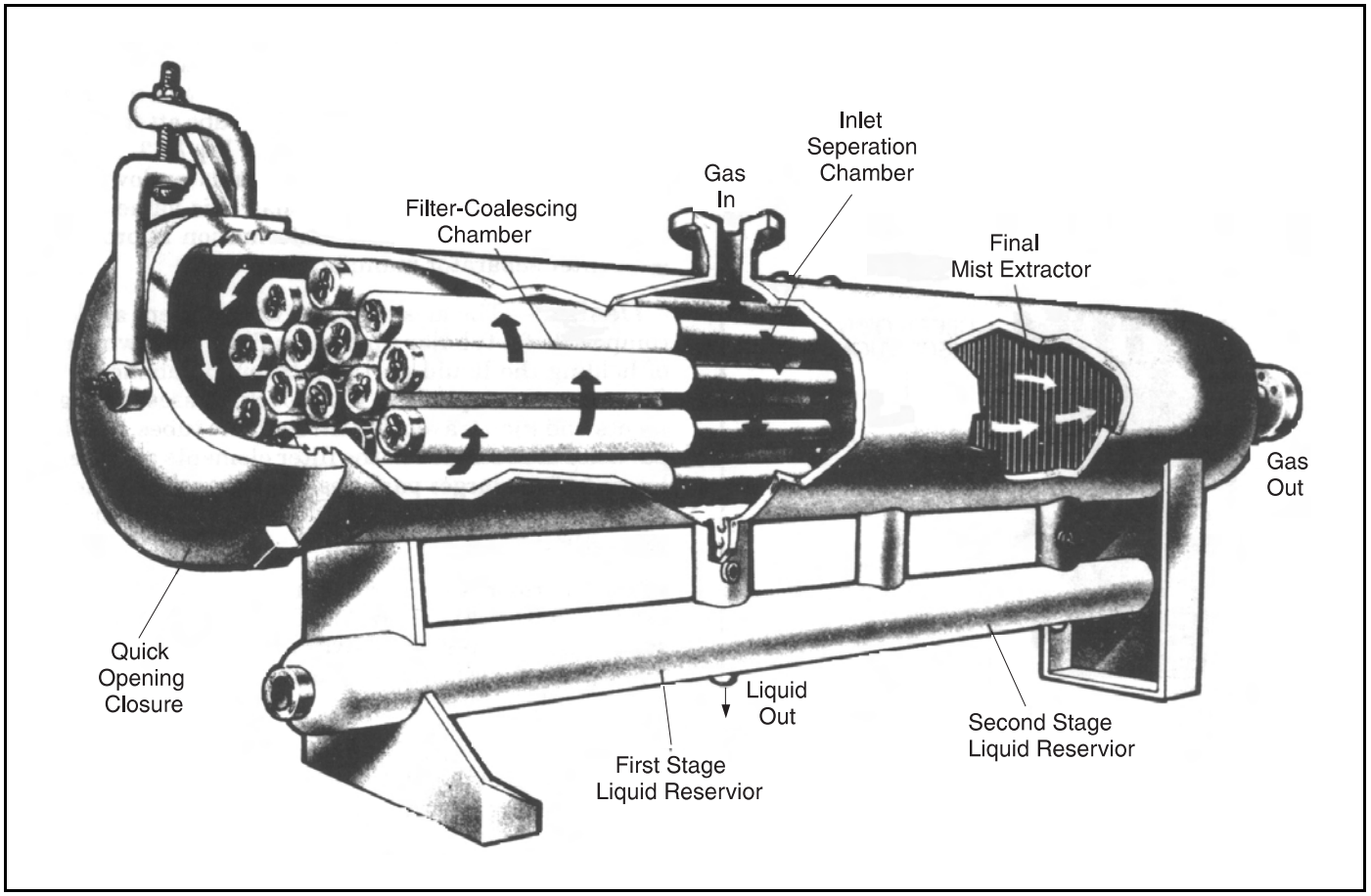
The design of filter separators is proprietary and a manufacturer should be consulted for specific sizing and recommendations.

Coalescing Filter

The coalescing filter is a more recent (early 1980s) piece of separation equipment designed for "gas polishing' service. Fig.

FIG. 7-24

Horizontal Filter-Separator



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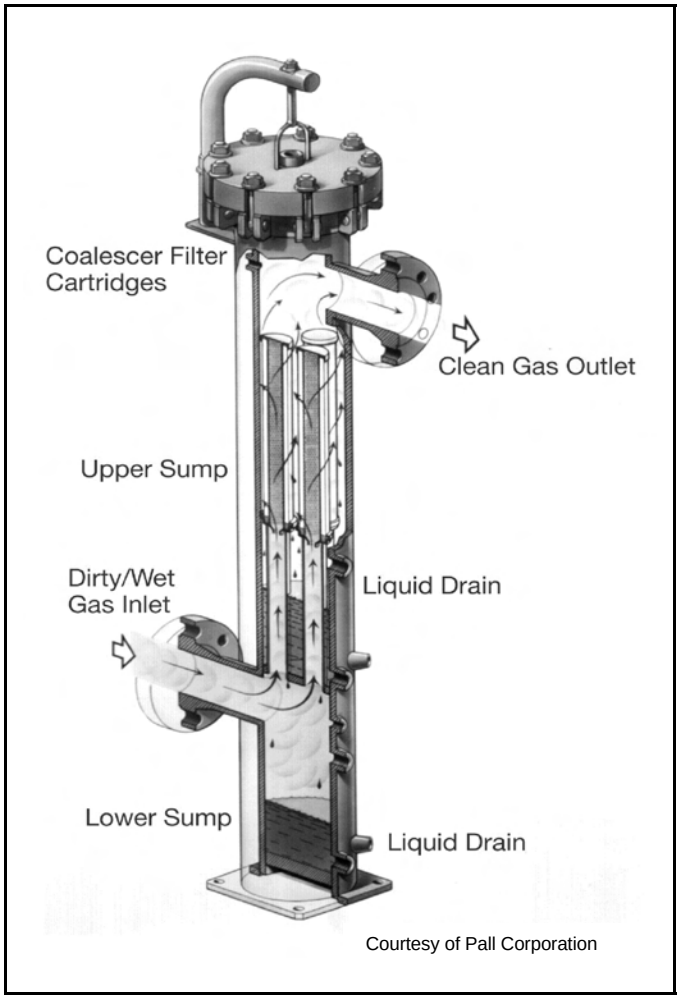
7-25 illustrates a typical coalescing filter. A coalescing filter is typically intended to remove fine liquid aerosols/mist from gas streams where entrained liquid loads are low. A typical filtration rating for coalescing filter elements is 0.3 microns absolute. This means that solid spherical particles larger than 0.3 micron are unable to pass through the filter element. Aerosol liquid coalescing performance is not as easily quantifiable and is subject to several factors. A coalescing filters are normally used to protect equipment/processes that are particularly sensitive to contamination. Two of the most common applications are upstream of mole sieve systems and amine contactors. The unit is typically intended to remove carryover from an upstream conventional separator and/or any liquids that may condense from the gas phase due to temperature or pressure reduction. Coalescing filters can experience short filter element life if the gas contains appreciable amounts of solids, e.g. corrosion products.

PARTICULATE REMOVAL — FILTRATION

Filtration, in the strictest sense, applies only to the separation of solid particles from a fluid by passage through a porous medium. However, in the gas processing industry, filtration commonly refers to the removal of solids and liquids from a gas stream.

FIG. 7-25

Typical Coalescing Filter



The most commonly used pressure filter in the gas processing industry is a cartridge filter. Cartridge filters are constructed of either a self-supporting filter medium or a filter medium attached to a support core. Depending on the application, a number of filter elements are fitted into a filter vessel. Flow is normally from the outside, through the filter element, and out through a common discharge. When pores in the filter medium become blocked, or as the filter cake is developed, the higher differential pressure across the elements will indicate that the filter elements much be cleaned or replaced.

Cartridge filters are commonly used to remove solid contaminants from amines, glycols, and lube oils. Other uses include the filtration of solids and liquids from hydrocarbon vapors and the filtration of solids from air intakes of engines and tubing combustion chambers.

Two other types of pressure filters that also have applications in the gas processing industry include the edge and precoat filters. Edge filters consist of nested metallic discs enclosed in a pressure cylinder that are exposed to liquid flow. The spacing between metal discs determines the solids retention. Some edge filters feature a self-cleaning design in which the discs rotate against stationary cleaning blades. Applications for edge filters include lube oil and diesel fuel filtration as well as treating solvents.

Precoat filters find use in the gas processing industry; however, they are complicated and require considerable attention. Most frequent use is in larger amine plants where frequent replacement of cartridge elements is considerably more expensive than the additional attention required by precoat filters.

The precoat filter consists of a course filter medium over which a coating has been deposited. In many applications, the coating is one of the various grades of diatomaceous earth that is mixed in a slurry and deposited on the filter medium. During operation, additional coating material is often added continuously to the liquid feed. When the pressure drop across the filter reaches a specified maximum, the filter is taken offline and backwashed to remove the spent coating and accumulated solids. Applications for precoat filters include water treatment for water facilities as well as amine filtration to reduce foaming. Typical designs for amine plants use 1-2 gpm flow per square foot of filter surface area. Sizes range upward from 10-20% of the full stream rates.[11]

Filtration Equipment Removal Ratings[12]

The two main methods of specifying removal ratings for filters are 1) nominal rating, and 2) absolute rating. Nominal rating typically means that 90% (or sometimes 95%) by weight of the contaminants above a specified size (e.g. 10 μm) has been removed. The 2% (or potentially 5%) of the contaminant passing through the filter is not defined by the test. Therefore, it is possible to have particle much larger than the nominal size (e.g. 30 μm to 100 μm).

Absolute rating can be defined by one of two standards. The National Fluid Power Association's (NFPA's) standard of absolute rating states that the diameter of the largest hard spherical particle that will pass through a filter under specified test conditions is an indication of the largest opening in the filter. The Beta (β) Rating System determined by the Oklahoma State University, "OSU F-2 Filter Performance Test" determines the ratio of the number of particles of a given size in the influent divided by the number of particles of the same

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given size in the effluent. This results in the following equation for relating the β value to removal efficiency:

$$\% \text{ removal} = (\beta - 1) / \beta \times 100 \qquad \textbf{Eq 7-15}$$

Most "absolute" filters typically have a β of 10,000 (99.99% removal).

Special Applications – Slug Catcher Design[13]

Slug catcher design is a special application of gas-liquid separator design. Performing these calculations is a combination of pipeline multiphase hydraulics and separator sizing. There are two main types of slug catchers, vessel and pipe. Reference 13 provides design details and an example of slug catcher design.

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SECTION 8

Fired Equipment

Fired equipment transfers heat produced by combustion of the fuel to the process stream. In gas processing equipment, the fuel is usually natural gas; however, ethane, propane, or light oils are sometimes used. The process stream varies widely, e.g., natural gas, heavier hydrocarbons, water, glycol, amine solutions, heat transfer oils, and molten salts.

Fired equipment can be classified as:

1. Direct fired heaters where the combustion gases occupy most of the heater volume and heat the process stream contained in pipes arranged in front of refractory walls – the radiant section – and in a bundle in the upper portion – the convection section. Convective heaters are a special application in which there is only a convection section.

FIG. 8-1

Nomenclature

- a = constant, Eq. 8-21
- A = area, sq ft
- AO = dry combustion air mols per dry fuel mols, for stochiometric combustion, mols/mols
- B = parameter defined by Eq 8-21
- c = number of carbon atoms in fuel molecule
- C = constant, Eq 8-4 and Eq 8-8
- C_d = burner spud discharge coefficient
- C_p = specific heat, Btu/(lb • °F)
- CO = carbon monoxide mols in mols of dry flue gases, mols/mols
- CO_2 = carbon dioxide mols in mols of dry flue gases, mols/mols
- CO_{2_o} = carbon dioxide mols in mols of dry flue gases for stochiometric combustion, mols/mols
- d = diameter of pipe or cylinder or fin, in.
- D = diameter of pipe or cylinder, ft
- EA = volume percentage of excess combustion air, %
- f_f = Fanning friction factor
- F = view factor, dimensionless
- Fl_o = mols of dry flue gases per mols of dry fuel for stochiometric combustion, mols/mols
- g = acceleration due to gravity = 32.17 ft/sec^2
- g_c = 32.17 lb mass • ft/(lb force • sec^2)
- G = Ratio of air to fuel, lb/lb; or
  = mass velocity, lb/(sec • sq ft)
- GHI = gross heat input, Btu/hr
- GTE = gross thermal efficiency, Eq 8-17a
- Gr = Grashof number, dimensionless
- h = heat transfer coefficient, Btu/(hr • sq ft • °F)
- H = heat content or enthalpy, Btu/lb; or
  = height of stack, ft; height of fin, in.
- Hav = available draft, inches water column (gauge)
- HHV = higher or gross heating value, Btu/SCF
- hy = number of hydrogen atoms in fuel molecule
- I = average tube radiant heat flux intensity (circumferential), Btu/(hr • sq ft)
- k = thermal conductivity, Btu/[(hr • sq ft • °F)/ft]
- L = length of heat transfer surface, ft; or
  = mean beam length, ft; or
  = characteristic dimension, ft
- LHV = lower or net heating value, Btu/SCF
- m = constant, Eq 8-4 and Eq 8-8
- M = mass flow rate, lb/hr
- MPH = velocity, miles/hr
- n = number of fins per foot, number of tube rows
- ni = number of nitrogen atoms in fuel molecule
- Nu = Nusselt number, dimensionless
- NHI = net heat input, Btu/hr
- NPS = nominal pipe size, in.
- NTE = net thermal efficiency, Eq 8-17b
- o = number of oxygen atoms in fuel molecule
- O_2 = oxygen mols in mols of dry flue gases, mols/mols
- O_{2_o} = oxygen mols in mols of dry flue gases for stochiometric combustion, mols/mols
- P = partial pressure of $CO_2 + H_2O$, atm
- P_B = barometric pressure, in. Hg @ 32°F
- ΔP = pressure difference, psi
- Pg = burner fuel gas pressure, psia
- Pr = Prandtl number, dimensionless
- q_{gs} = gas flow rate at standard conditions, SCF/day
- q_l = liquid flow rate, gals./min
- Q = heat transfer or heat input or heat content (rates), Btu/hr
- r = ratio of flue gases to heat release, lb/MMBtu
- rf = fouling heat flow resistance, hr • sq ft • °F/Btu
- R = fraction of total heat liberation absorbed in radiant section (Fig. 8-20)
- Re = Reynolds number, dimensionless
- s = number of sulfur atoms in fuel molecule
- S = tube spacing, in.
- SCF = standard cubic feet at 14.696 psia and 60°F
- SG = specific gravity
- t = fin thickness, in.
- T = temperature, °R
- ΔT = temperature difference, °F

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FIG. 8-1 (Cont'd)

Nomenclature

U = overall heat transfer coefficient, Btu/(hr • sq ft • °F)
UHT = useful heat transferred or heat duty, Btu/hr
V = velocity, ft/sec
w = weight of air, lb
wt = wall thickness, in.
Δx = distance in direction of heat transfer, ft or in.
Y = expansion factor, dimensionless

Greek

β = volumetric coefficient of thermal expansion, 1/°F
ε_1, ε_2, = emissivities of combustion gases and wall, respectively
μ = viscosity of fluid, cp
ρ = density of fluid, lb/cu ft
σ = Stefan-Boltzmann constant, 0.171 (10^{-8}) Btu/(hr • sq ft • $°R^4$)
π = 3.1416

Subscripts

a = atmospheric air at operating conditions
as = air at standard conditions
B = barometric
b = bulk
c = convective
cs = cross section (projection)
f = fin; fouling; friction; Fanning
g = gas
gs = gas at standard conditions
i = inside, internal
LM = log mean base e
m = middle surface
o = outside, external, overall
p = pipe
r = radiant
s = stack
w = wall
1 = burner operating conditions 1
2 = burner operating conditions 2

2. Firetube heaters where the combustion gases are contained in a firetube that is surrounded by a liquid that fills the heater shell. This liquid may be either the process stream or a heat transfer medium that surrounds the coil bundle containing the process stream.

Fig. 8-2 lists the common applications and general characteristics of these heaters.

FIG. 8-2

Heater Applications and Characteristics

Direct Fired	Firetube
Applications	
Hot oil heater Regeneration gas heaters Amine and stabilizer reboilers	Indirect fired water bath heaters (line heaters) Propane and heavier hydrocarbon vaporizers Hot oil and salt bath heaters Glycol and amine reboilers Low pressure steam generators
Characteristics	
More ancillary equipment and controls Higher thermal efficiency Requires less plot space Forced or natural draft combustion	Heat duty usually less than 10 MMBtu/hr Easily skid mounted Forced or natural draft combustion Less likely to have hot spots or tube rupture

HEAT TRANSFER

Conduction

Fourier's law of conduction gives the rate of heat transfer through substances resulting from vibrations and interactions between adjacent molecules as opposed to overall motion or mixing of the molecules. Conduction always applies to solids and rarely to fluids.

Fundamental equations for steady heat conduction through some common solid shapes, ignoring border conditions, are:

For unidimensional perpendicular heat flow through flat walls, as in heat flow through a square or very large cylindrical tank wall:

$$Q = 12 \cdot k \cdot A \cdot \frac{\Delta T}{wt} \qquad \textbf{Eq 8-1}$$

For heat transfer in cylindrical geometry where the heat transfer is normal to the axis, as in heat flow through a cylindrical vessel or pipe wall:

$$Q = \frac{2 \cdot \pi \cdot L \cdot k \cdot \Delta T}{ln\,(D_o/D_i)} = \frac{2 \cdot \pi \cdot L \cdot k \cdot \Delta T}{ln\,(d_o/d_i)} \qquad \textbf{Eq 8-2a}$$

For radial heat flow through a spherical vessel:

$$Q = \frac{2 \cdot \pi \cdot k \cdot \Delta T}{(1/D_i) + (1/D_o)} = \frac{\pi \cdot k \cdot \Delta T}{(6/d_i) + (6/d_o)} \qquad \textbf{Eq 8-2b}$$

Fig. 8-3 gives the thermal conductivities and densities of commercial refractories and insulation. Similar data for metals are given in Fig. 8-8 and Fig. 9-8.

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FIG. 8-3

Properties of Commercial Refractories and Insulations

Products	Max Service Temp, °F (Note 1)	Thermal Conductivity, Btu/((hr • ft^2 • °F)/in) at Mean Temperature, °F (Note 16)					Density, lb/ft^2	Cold Crush Strength, lb/in^2	Notes
		500	1000	1500	2000	2500			
FIREBRICK									
H-W Karundal XD	3300	24.4	20.4	19.6	19.6	19.7	187	15600	2
H-W UFALA	3000	13.0	12.8	13.1	13.6	14.4	157	7000	
APGreen KX-99	2500	9.7	9.9	10.2	10.7	11.1	143	10700	
APGreen Empire S	2400	8.3	8.8	9.2	9.5	9.9	132	2100	
INSULATING BRICK									
APGreen Greenlite 30	3000	2.6	3.1	3.6	4.0	4.5	61	1175	
Thermal Ceramics K-25	2500	1.1	1.3	1.5	1.7		38	220	
Thermal Ceramics K-23	2300	1.0	1.2	1.5	1.7		31	145	
Thermal Ceramics K-20	2000	0.9	1.1	1.4			29	125	
HEAVY CASTABLE									
APGreen Greencast 94	3400	14.5	12.6	11.3	10.7	10.8	163	5800	2,3,4,5
APGreen Mizzou	3000	7.8	7.7	7.5	7.4	7.4	139	3000	3,11
LIGHT CASTABLE									
APGreen Kast-O-Lite 25	2600	3.6	3.5	3.7	4.0		86	1300	8
APGreen 45-L	2500	2.7	2.6	2.5			71	2500	3,6,7,8
APGreen Castable 22	2200	1.7	1.9	2.2	2.5		53	275	3,8
APGreen Cast Block Mix	1600	0.6	1.0	1.5			22	20	8
GUN & RAM MIXES									
Premier 85 RAM HS	3000	21.7	14.1	12.6	11.9	12.1	175	5000	5,9
H-W Tuff Shot LI	2600	6.2	6.5	6.7	6.8	6.9	119	3450	6,11
APGreen Kast-O-Lite 26-LI GR	2600	3.6	3.7	3.9	4.3		92	2000	
CERAMIC FIBER									
Thermal Ceramics Saffl	2800	0.38	0.7	1.0	1.5	2.3	3		4
Fiberfrax Durablanket 2600	2600	0.51	1.15	2.18	3.56		6		
Fiberfrax Durablanket 2600	2600	0.51	0.98	1.81	2.96		8		
Fiberfrax 550 Paper	2300	0.42	0.75	1.33	2.12		12		
Thermal Cer Cerablanket	2150	0.5	1.0	1.7	2.6		6		
Thermal Cer Cerablanket	2150	0.37	0.8	1.5	2.1		8		
BLOCK & BOARD									
Fiberfrax Duraboard LD	2300	0.56	0.85	1.24	0.73		16	50	
Fiberfrax Duraboard HD	2300	0.56	0.85	1.47	1.61		26	70	
Thermal Ceramics TR-20	2000	0.64	0.77	0.92			25	201	12
USG K-FAC 19	1900	0.51	0.89				18.5	33	10,13
Schuller Thermo 12	1200	0.45	0.64				15	160	10,14
PARTEK Paroc 1212	1200	0.42	0.98				12		10,13
Schuller 1000 SpinGlas	850	0.50					3		10,15

NOTES

1. Maximum Service Temperature listed has no safety factor included.
2. 90-94% Alumina product for extreme temperature or high velocity service.
3. Cast properties listed, gunning product available but properties will be different.
4. Low silica product.
5. For burner blocks and severe service.
6. Can be used as one shot (single layer) lining.
7. High performance medium weight lining.
8. May be used as back-up insulation in 2 layer lining.
9. 85% Alumina ramming plastic for burner blocks, etc.
10. For external insulation only.
11. May be used as hot face lining in dual layer system.
12. Diatomaceous earth base.
13. Mineral wool base.
14. Calcium silicate base.
15. Fiber glass base.
16. Convert to Btu /[(hr • ft^2 • °F) /ft] before using in equations in this section.

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Example 8-1 — Estimate the loss per linear foot through a one inch layer of block insulation covering an 8 in. Sch 40 steam header. Assume:

$T_i = 250°F$

$T_o = 50°F$

$k = 0.0417$ Btu/[(hr • sq ft • °F)/ft.]

(Both ft and in. are usually used in the thermal conductivity units for insulating materials; the following equations use ft.)

Solution Steps

From Eq 8-2a

$$Q = \frac{2 \cdot \pi \cdot k \cdot \Delta T}{ln\,(d_o / d_i)}$$

$$= \frac{2 \cdot \pi\,(0.0417)(250–50)}{ln\,(10.625 / 8.625)}$$

$= 251.2$ Btu/hr per linear ft

Convection

Heat transfer between a solid and an adjacent fluid occurs by movement of the fluid molecules. Hot molecules leave the surface of the solid and are replaced by cold ones. Most of the resistance to this form of heat transfer occurs in a thin film or layer next to the solid surface. This layer exists even if the bulk fluid flow is violently turbulent.

Newton's law of cooling applies to convective heat transfer

$$Q = h \cdot A \cdot \Delta T \qquad \textbf{Eq 8-3}$$

Natural or free convection — occurs when the only force promoting the fluid flow results from temperature differences in the fluid. Under these conditions the heat transfer coefficient is obtained from the Nusselt equation.

$$Nu = C\,(Gr \cdot Pr)^m \qquad \textbf{Eq 8-4}$$

Where

$$Nu = \frac{h \cdot D_o}{k} = \frac{h \cdot d_o}{12 \cdot k} \qquad \textbf{Eq 8-5a}$$

or $$Nu = \frac{h \cdot L}{k} \qquad \textbf{Eq 8-5b}$$

$$Gr = \frac{2.214 \cdot 10^6 \cdot D_o^3 \cdot \rho^2 \cdot g \cdot \beta \cdot \Delta T}{\mu^2}$$

$$= \frac{1281 \cdot d_o^3 \cdot \rho^2 \cdot g \cdot \beta \cdot \Delta T}{\mu^2} \qquad \textbf{Eq 8-6a}$$

or $$Gr = \frac{2.214 \cdot 10^6 \cdot L^3 \cdot \rho^2 \cdot g \cdot \beta \cdot \Delta T}{\mu^2} \qquad \textbf{Eq 8-6b}$$

$$Pr = \frac{2.419 \cdot C_p \cdot \mu}{k} \qquad \textbf{Eq 8-7}$$

The constants C and m depend on the shape and size of the solid surface, the orientation of the surface to the fluid, whether the solid is hotter than the fluid or vice versa, and the magnitude of (Gr • Pr). A brief summary of C and m for some usual situations is given in Fig. 8-4.

Nu, Gr, and Pr are dimensionless when the units indicated in Fig. 8-1 are used in equations 8-4 through 8-7. The physical properties are those of the fluid at the film temperature, which is often assumed to be the average of the solid surface and bulk fluid temperatures. Fluid properties may have to be evaluated at an assumed film temperature, and this assumption then confirmed from the results — see Example 8-4.

FIG. 8-4

Heat Transfer Constants[2] for Eq 8-4 (Natural or Free Convection)

Configuration	D or Y (Y = Gr • Pr)	C	m
Vertical Plates or Cylinders	$Y < 10^4$	1.36	0.20
	$10^4 < Y < 10^9$	0.55	0.25
	$10^9 < Y$	0.13	0.33
Horizontal Plates:			
Facing Up	$10^5 < Y < 2(10^7)$	0.54	0.25
Facing Up	$2(10^7) < Y < 3\,(10^{10})$	0.14	0.33
Facing Down	$3(10^5) < Y < 3(10^{10})$	0.27	0.25
Long Horizontal Cylinders L > D	D < 0.1	0.53	0.25
	0.1 < D < 0.5	0.47	0.25
	0.5 < D	0.11	0.33
Short Horizontal Cylinders L = D < 8 in.	$Y < 10^{-5}$	0.49	0.00
	$10^{-5} < Y < 10^{-3}$	0.71	0.04
	$10^{-3} < Y < 1$	1.09	0.10
	$1 < Y < 10^4$	1.09	0.20
	$10^4 < Y < 10^9$	0.53	0.25
	$10^9 < Y$	0.13	0.33

The coefficient of thermal expansion β for low pressure gas (i.e. ideal gas) equals 1/(T, °R). It is left in this form for use in Gr and is not converted to 1/°F for dimensional consistency.

Example 8-2 — What is the heat transfer coefficient for natural convection around a 3 in. Sch 40 pipe surrounded by water at 190°F? Assume T_o for the pipe is 114°F.

Solution Steps

$T_{film} = (190 + 114)/2 = 152°F\ (612°R)$

From Eq. 8-4, 8-5a, 8-6a and 8-7 and Fig. 8-4:

$$Nu = 0.47\,(Gr \cdot Pr)^{0.25}$$

$$\frac{h \cdot d_o}{12 \cdot k} = 0.47\left[\frac{1281 \cdot d_o^3 \cdot \rho^2 \cdot g \cdot \beta \cdot \Delta T \cdot C_p \cdot 2.419}{\mu \cdot k}\right]^{0.25}$$

$d_o = 3.5$ in.
$\rho = 61.8$ lb/cu ft
$g = 32.17$ ft / sec^2
$\beta = 0.00035°F^{-1}$
$\Delta T = 190 - 114 = 76°F$
$C_p = 1.00$ Btu/(lb • °F)
$\mu = 0.63$ cp
$k = 0.365$ Btu/[(hr • sq ft • °F)/ft]

The properties of water (ρ, β, μ, k) are based on the film temperature.

Substituting:

$$h = \frac{(12)(0.47)(0.365)}{3.5}\left[\frac{1281(3.5)^3(61.8)^2(32.17)(0.00035)(76)(1.0)(2.419)}{(0.63)\,(0.365)}\right]^{0.25}$$

$$= 0.588\,[1.889\,(10^9)]^{0.25}$$

$= 122.6$ Btu/(hr • sq ft • °F)

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Forced convection — occurs when the fluid flow adjacent to a solid is promoted by external force, e.g., pumping, agitation, etc. The result is a substantial increase in the heat transfer rate. The Dittus-Boelter correlation is:

$$Nu = C \cdot Re^{m} \cdot Pr^{0.33} \qquad \textbf{Eq 8-8a}$$

Utilization of a viscosity correction term gives the Sieder-Tate correlation:

$$Nu = C \cdot Re^{m} \cdot Pr^{0.33} (\mu_b/\mu_w)^{0.14} \qquad \textbf{Eq 8-8b}$$

Where

$$Re = \frac{1488 \cdot D \cdot V \cdot \rho}{\mu} = \frac{124 \cdot d \cdot V \cdot \rho}{\mu} \qquad \textbf{Eq 8-9a}$$

$$= \frac{0.5263 \cdot M}{D \cdot \mu} = \frac{6.316 \cdot M}{d \cdot \mu}$$

$$= \frac{4.221 \cdot q_l \cdot \rho}{D \cdot \mu} = \frac{50.66 \cdot q_l \cdot \rho}{d \cdot \mu}$$

$$= \frac{q_{gs} \cdot SG}{598 \cdot D \cdot \mu} = \frac{0.02007 \cdot q_{gs} \cdot SG}{d \cdot \mu}$$

or

$$Re = \frac{1488 \cdot L \cdot V \cdot \rho}{\mu} \qquad \textbf{Eq 8-9b}$$

Equation 8-8b need be used only for high viscosity fluids such as glycol.

As before, Re is dimensionless when the units indicated in Fig. 8-1 are used in equations 8-9a and 8-9b. The constants C and m depend on the configuration and the type of fluid flow — laminar, intermediate, or turbulent — which is characterized by the magnitude of the Reynolds number. Fig. 8-5 lists values for the more common situations.

Example 8-3 — Find the heat transfer coefficient for 10 MMscfd of 0.6 gravity natural gas flowing at 2000 psia in a 3 in. Sch 80 pipe when the pipe wall and gas temperatures are 104 and 72°F, respectively.

Solution Steps

$T_{film} = (104 + 72)/2 = 88°F$

From Eq 8-7, 8-8a and 8-9a and Fig. 8-5:

$$Nu = 0.023 (Re)^{0.8} (Pr)^{0.33}$$

$$\frac{h \cdot d_i}{12 \cdot k} = 0.023 \left(\frac{0.02 \cdot q_{gs} \cdot SG}{d_i \cdot \mu}\right)^{0.8} \left(\frac{2.419 \cdot C_p \cdot \mu}{k}\right)^{0.33}$$

where: $d_i = 2.90$ in.

$\mu = 0.019$ cp

$k = 0.03$ Btu/[(hr • sq ft • °F)/ft]

$C_p = 0.73$ Btu / (lb • °F)

$q_{gs} = 10 (10^6)$ scfd

$$Re = \frac{0.02 \cdot q_{gs} \cdot SG}{d \cdot \mu} =$$

$$= \frac{0.02 (10) (10^6) 0.6}{2.9 (0.019)} = 2.18 (10^{6})$$

$$Pr = \frac{2.419 \cdot Cp \cdot \mu}{k} = \frac{2.419 \cdot 0.73 \cdot 0.019}{0.03}$$

$$= 1.118$$

$$h_i = \frac{(0.023) (12) (0.03)}{2.9} [2.18 (10^6)]^{0.8} (1.118)^{0.33}$$

$= 349.9$ Btu /(hr • sq ft • °F)

Overall Heat Transfer Coefficient

So far, only the individual or local heat transfer coefficients have been considered. As discussed in Section 9 "Shell and Tube Heat Exchangers" the individual heat transfer coefficients are combined into an overall heat transfer coefficient. See Fig. 9-3 for calculation of ΔT_{LM}.

$$Q = U \cdot A \cdot \Delta T_{LM} \qquad \textbf{Eq 8-10}$$

U must be based on some specific area. Considering all the resistances to radial heat transfer through a hollow cylinder whose wall is made of two different materials (i.e. metal pipe and a layer of insulation), the overall heat transfer based on the insulation outside diameter is:

FIG. 8-5

Heat Transfer Constants for Equation 8-8[2,3] Forced Convection

Configuration	Characteristic Length	Re	Pr	C	m
Flat plate parallel to flow	Plate length	$10^3 < Re < 10^5$	> 0.6	0.648	0.50
Cylinder axis perpendicular to flow	Cylinder diameter	$1 < Re < 4$	> 0.6	0.99	0.33
		$4 < Re < 40$	> 0.6	0.91	0.39
		$40 < Re < 4000$	> 0.6	0.68	0.47
		$4(10)^3 < Re < 4(10)^4$	> 0.6	0.193	0.62
		$4(10)^4 < Re$	> 0.6	0.0266	0.81
Inside pipes	Inside pipe diameter	$10^4 < Re$	0.7 < Pr < 700	0.023	0.80
Outside bank of tubes	Staggered outside tube diameter	$2(10)^3 < Re$	> 0.6	0.33	0.60
	In line	$2(10)^3 < Re$	> 0.6	0.26	0.60

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$$U_o = \frac{1}{\frac{1}{h_o} + \frac{D_o}{D_i \cdot h_i} + \frac{D_o \cdot ln\,(D_o/D_m)}{2 \cdot k_o} + \frac{D_o \cdot ln(D_m/D_i)}{2 \cdot k_i} + rf_o + rf_i \cdot \frac{D_o}{D_i}}$$

$$= \frac{1}{\frac{1}{h_o} + \frac{d_o}{d_i \cdot h_i} + \frac{d_o \cdot ln\,(d_o/d_m)}{24 \cdot k_o} + \frac{d_o \cdot ln(d_m/d_i)}{24 \cdot k_i} + rf_o + rf_i \cdot \frac{d_o}{d_i}} \qquad \textbf{Eq 8-11}$$

When there is only one solid layer, delete the fourth term in Equation 8-11 and change the subscript m to i and delete the subscript on k in the third term.

Example 8-4 — Find the overall heat transfer coefficient for a 3 in. Sch 80 pipe submerged in a 190°F water bath. 10 MMscfd of 2000 psia natural gas is to be heated from 60 to 84°F. Refer to Examples 8-2 and 8-3; note that the heat flow through the pipe wall is similar to Example 8-1.

Solution Steps

Use h_i, h_o, and the conduction through the pipe wall to find U_o. Then check the heat fluxes to see that the right film temperatures were used. k = 26.2 Btu/[(hr • sq ft • °F)/ft] for the pipe wall at 109°F (value from Fig. 8-8). Assume clean pipe.

From Eq 8-11

$$U_o = \frac{1}{\left[\frac{1}{122.6} + \left(\frac{3.50}{2.90\,(349.9)}\right) + \frac{3.50 \cdot ln\,(3.50/2.90)}{24\,(26.2)}\right]}$$

$$U_o = 79.0 \text{ Btu/(hr • sq ft • °F)}$$

$$Q = U_o\, A_o\, \Delta T_{LM} = 79.0\left(\pi \frac{3.5}{12}\right)\frac{(190 - 60) - (190 - 84)}{ln\left(\frac{130}{106}\right)}$$

$$= 8516 \text{ Btu/hr per linear ft}$$

To confirm the film temperatures and the validity of the individual heat transfer coefficients, the heat fluxes outside, through, and inside the pipe must be compared with the overall heat flux. Consider one linear foot of pipe. At the average gas temperature of 72°F, calculate the fluxes for one linear foot of pipe.

$$Q_o = h_o\, A_o\, \Delta T_o = (122.6)\left(\pi \frac{3.5}{12}\right)(190 - 114)$$

$$= 8538 \text{ Btu/hr per linear ft}$$

$$Q_p = \frac{2 \cdot \pi \cdot L \cdot k \cdot \Delta T}{ln\,(d_o/d_i)} = \frac{(2)\,(3.14)(1)(26.2)(114 - 104)}{ln\,(3.50\,/\,2.90)}$$

$$= 8749 \text{ Btu/hr per linear ft}$$

$$Q_i = h_i\, A_i\, \Delta T_i = 350.1\left(\pi \frac{2.9}{12}\right)(104 - 72)$$

$$= 8506 \text{ Btu/hr per linear ft}$$

The agreement is close enough for purposes of this example.

The above refers to clean pipes. Fouling occurs with continued use. Sometimes, fouling factors are specified, e.g., 0.001 internally and externally. These are corrections that are added to $1/U_o$. So,

$$1/U_o = 0.01265 + 0.001 + \left(\frac{3.50}{2.90}\right)(0.001)$$

$$U_o = 67.3 \text{ Btu/(hr • sq ft • °F)}$$

Finned Tubes

When combustion gases flow externally and heat a liquid in a pipe, there is a significant disparity between h_i (usually over 150 Btu/[hr • sq ft • °F]) and h_o (usually less than 10 Btu/[hr• sq ft • °F]). To overcome this and make better use of a given length of pipe, the external surface is finned. When the gas flow is normal to the pipe axis, helical fins (typically 0.05 to 0.105 in. thick, 0.5 to 1.5 in. high, and 24 to 72 fins/ft) are used. The result is an increase of up to tenfold in the external area of the pipe.

The total external surface area of a finned pipe and the cross sectional or projected area restricting normal gas flow per linear foot are:

$$A_o = \frac{\pi \cdot d_o}{12}(1 - n \cdot t/12) + \left(\frac{n \cdot \pi}{2}\right)\left(\frac{(d_f^{\,2} - d_o^{\,2})}{144}\right) \qquad \textbf{Eq 8-12}$$

and

$$A_{cs} = \frac{d_o}{12} + \frac{n \cdot t\,(d_f - d_o)}{144} \qquad \textbf{Eq 8-13}$$

The surface area of the fins is not as efficient as the external surface of the pipe because heat absorbed at the fin surface must be conducted to the fin base before it can pass through the pipe wall. Fin efficiencies are given in Fig. 8-6. These efficiencies are applied to the total external area.

Another important consideration is fin tip temperature. This can be obtained from the fin efficiency and Fig. 8-7. Fig. 8-8 gives the thermal conductivities and maximum recommended fin tip temperatures for the more common ferrous construction materials.

Example 8-5 — Calculate the external surface area and fin efficiency for 4 in. Sch 80 pipe with the following finning: 36 fins/ft, 1.25 in. high, 0.105 in. thick. Assume h_o = 3.9 Btu/(hr • sq ft • °F), k_f = 25 Btu/[(hr • sq ft • °F)/ft].

Solution Steps

Abscissa in Fig. 8-6:

$$0.4\, H_f \sqrt{\frac{h_o}{k_f\, t}} = 0.4\,(1.25)\sqrt{\frac{3.9}{(25)\,(0.105)}} = 0.61$$

$$d_f = d_o + 2\, H_f = 4.5 + 2\,(1.25) = 7.0 \text{ in.}$$

$$\frac{d_f}{d_o} = \frac{7.0}{4.5} = 1.56$$

From Fig. 8-6, fin efficiency is 87%.

From Eq 8-12:

$$A_o = \pi\left(\frac{4.5}{12}\right)\left(1 - \frac{(36)\,(0.105)}{12}\right) + \frac{36}{2}\,\pi\left(\frac{7.0^2 - 4.5^2}{144}\right)$$

$$= 12.10 \text{ sq ft per linear ft}$$

Radiation

An estimate of the radiant heat flux between two surfaces is:

$$\frac{Q}{A} = \frac{\sigma \cdot F\,(T_1^{\,4} - T_2^{\,4})}{(1/\varepsilon_1 + 1/\varepsilon_2 - 1)} \qquad \textbf{Eq 8-14}$$

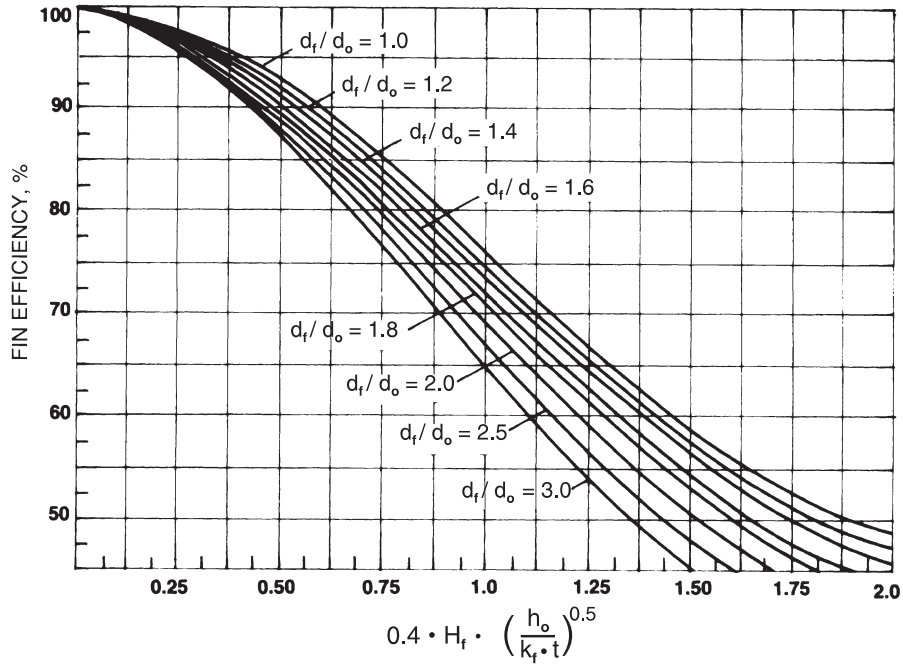
The geometric or view factor, F, is the fraction of the surface area that is exposed to and absorbs radiant heat. The equation for F must be determined from an analysis of the geometry. In using Eq 8-14, it is recommended[6] that F not be less than 0.67. Also, all temperatures are absolute.

Fig. 8-9 gives the emissivities of common construction metals, oxides, refractories, and insulation materials. The emissivity of combustion gases is more complex because it depends on the temperature and the product (P • L). See Figs. 8-10, 8-11, and 8-12.

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FIG. 8-6

Fin Efficiency Chart[4]



Example 8-6 — What is the radiant heat flux to a 3 ft length of a 2 ft ID firetube when the combustion gases inside the tube are at 2800°F and the firetube wall is at 300°F? Assume 20% excess air is used.

Solution Steps

$$F = \frac{\text{curved surface area}}{\text{total surface area}} = \frac{\pi \cdot D \cdot L}{\pi \cdot D \cdot L + 2\,(\pi \cdot D^2/4)}$$

$$= \frac{2\,(3)}{2\,(3) + 2\,(4/4)} = 0.75$$

From Fig. 8-10, $P_{CO_2 + H_2O} = 0.24$

From Fig. 8-11, $L = 2.0$ ft, so $P \cdot L = 0.48$

From Fig. 8-12, $\varepsilon_1 = 0.12$

From Fig. 8-9, $\varepsilon_2 = 0.79$ (steel, oxidized at 1100°F)

Equation 8-14, $$\frac{Q}{A} = \frac{0.173\,(10^{-8})\,0.75\,(3260^4 - 760^4)}{(1/0.12 + 1/0.79 - 1)}$$

$$= 16{,}990 \text{ Btu/(hr} \cdot \text{sq ft)}$$

Note that T is in °R.

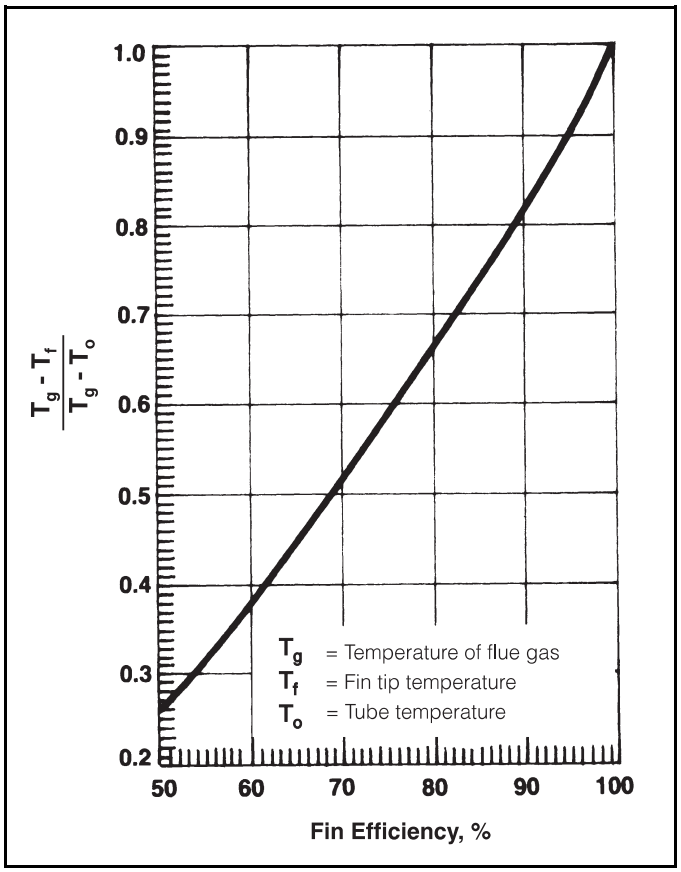
Heat Losses

Heat losses from equipment surfaces occur primarily by radiation and convection. Fig. 8-13 gives the combined heat transfer coefficient, $h_c + h_r$, in terms of the wind velocity and the temperature difference between the surface and the surrounding air.

Example 8-7 — How much heat can be saved per linear foot by covering an 8 in. Sch 40 steam header, carrying 15 psig steam at 250°F, with a 1 in. thick layer of block insulation? Assume ambient conditions are 30°F with a 15 mph wind.

FIG. 8-7

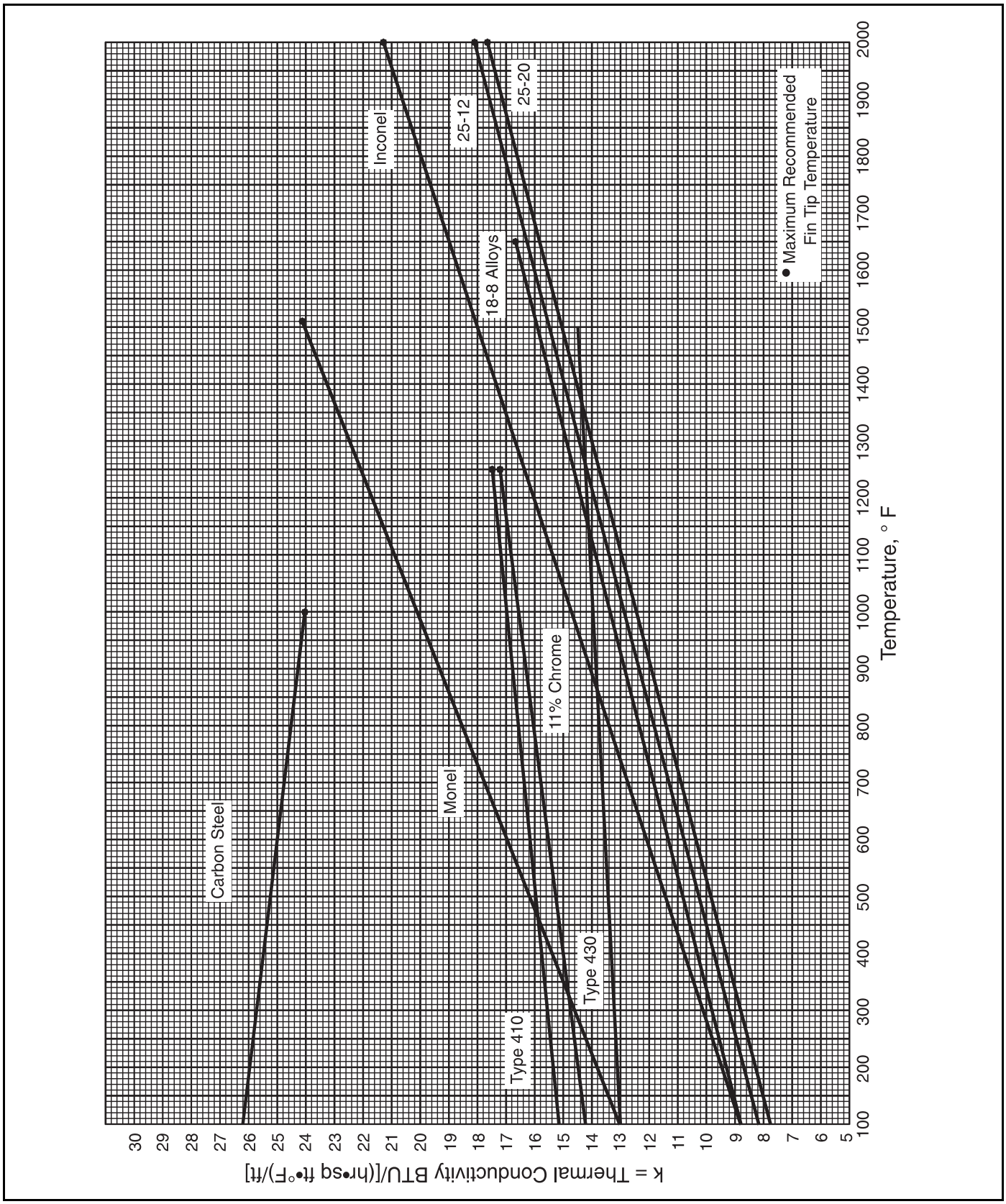
Fin Tip Temperature[5]



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FIG. 8-8

Thermal Conductivity of Ferrous Materials[4]



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FIG. 8-9

Normal Total Emissivity of Various Surfaces[3]

A. Metals and Their Oxides

Surface	T, °F*	Emissivity*
Aluminum		
Highly polished plate, 98.3% pure	440-1070	0.039-0.057
Polished plate	73	0.040
Rough plate	78	0.055
Oxidized at 1110°F	390-1110	0.11-0.19
Aluminum-surfaced roofing	100	0.216
Calorized surfaces, heated at 1110°F:		
Copper	390-1110	0.18-0.19
Steel	390-1110	0.52-0.57
Brass		
Highly polished:		
73.2% Cu, 26.7% Zn	476-674	0.028-0.031
62.4% Cu, 36.8% Zn, 0.4% Pb, 0.3% Al	494-710	0.033-0.037
82.9% Cu, 17.0% Zn	530	0.030
Chromium; (see Nickel Alloys for Ni-Cr steels)	100-1000	0.08-0.26
Copper		
Commercial, emeried, polished, but pits remaining	66	0.030
Commercial, scraped shiny but not mirror-like	72	0.072
Polished	242	0.023
Plate, heated long time, covered with thick oxide layer	77	0.78
Cuprous oxide	1470-2010	0.66-0.54
Iron and steel		
Metallic surfaces (or very thin oxide layer):		
Cast iron, polished	392	0.21
Ground sheet steel	1720-2010	0.55-0.61
Smooth sheet iron	1650-1900	0.55-0.60
Oxidized surfaces:		
Iron plate, pickled, then rusted red	68	0.612
Completely rusted	67	0.685
Rolled sheet steel	70	0.657
Oxidized iron	212	0.736
Cast iron, oxidized at 1100°F	390-1110	0.64-0.78
Steel, oxidized at 1100°F	390-1110	0.79
Iron oxide	930-2190	0.85-0.89
Sheet steel, strong rough oxide layer	75	0.80
Dense shiny oxide layer	75	0.82
Cast plate:		
Smooth	73	0.80
Rough	73	0.82
Cast iron, rough, strongly oxidized	100-480	0.95
Wrought iron, dull oxidized	70-680	0.94
Steel plate, rough	100-700	0.94-0.97
High temperature alloy steels (see Nickel Alloys)		
Monel metal, oxidized at 1110°F	390-1110	0.41-0.46
Nickel		
Electroplated on pickled iron, not polished	68	0.11
Plate, oxidized by heating at 1110°F	390-1110	0.37-0.48
Nickel oxide	1200-2290	0.59-0.86
Nickel alloys		
Chromnickel	125-1894	0.64-0.76
Nickelin (18-32 Ni; 55-68 Cu; 20 Zn), gray oxidized	70	0.262
KA-2S alloy steel – (8% Ni; 18% Cr), light silvery, rough, brown, after heating	420-914	0.44-0.36
After 42 hr. heating at 980°F	420-980	0.62-0.73
NCT-3 alloy (20% Ni; 25% Cr), brown, splotched, oxidized from service	420-980	0.90-0.97
NCT-6 alloy (60% Ni; 12% Cr), smooth, black, firm adhesive oxide coat from service	520-1045	0.89-0.82
Tin – bright tinned iron sheet	76	0.043 & 0.064
Zinc		
Commercial, 99.1% pure, polished	440-620	0.045-0.053
Oxidized by heating at 750°F	750	0.11
Galvanized sheet iron, fairly bright	82	0.228
Galvanized sheet iron, gray oxidized	75	0.276

B. Refractories, Building Materials, Paints, and Miscellaneous

Surface	T, °F*	Emissivity*
Asbestos		
Board	74	0.96
Paper	100-700	0.93-0.945
Brick		
Red, rough, but no gross irregularities	70	0.93
Silica, unglazed, rough	1832	0.80
Silica, glazed, rough	2012	0.85
Grog brick, glazed	2012	0.75
See Refractory Materials below.		
Carbon		
T-carbon (Gebr. Siemens) 0.9% ash (this started with emissivity at 260°F of 0.72, but on heating changed to values given)	260-1160	0.81-0.79
Enamel, white fused, on iron	66	0.897
Glass, smooth	72	0.937
Gypsum, 0.02 in. thick on smooth or blackened plate	70	0.903
Marble, light gray, polished	72	0.931
Oak, planed	70	0.895
Oil layers on polished nickel (lube oil)	68	
Polished surface, alone		0.045
+0.001-in. oil		0.27
+0.002-in. oil		0.46
+0.005-in. oil		0.72
Infinitely thick oil layer		0.82
Oil layers on aluminum foil (linseed oil)		
Al foil	212	0.087†
+1 coat oil	212	0.561
+2 coats oil	212	0.574
Paints, lacquers, varnishes		
Snowhite enamel varnish on rough iron plate	73	0.906
Black shiny lacquer, sprayed on iron	76	0.875
Black shiny shellac on tinned iron sheet	70	0.821
Black matte shellac	170-295	0.91
Black lacquer	100-200	0.80-0.95
Flat black lacquer	100-200	0.96-0.98
White lacquer	100-200	0.80-0.95
Oil paints, sixteen different, all colors	212	0.92-0.96
Aluminum paints and lacquers		
10% Al, 22% lacquer body, on rough or smooth surface	212	0.52
26% Al, 27% lacquer body, on rough or smooth surface	212	0.3
Other Al paints, varying age and Al content	212	0.27-0.67
Al lacquer, varnish binder, on rough plate	70	0.39
Al paint, after heating to 620°F	300-600	0.35
Plaster, rough lime	50-190	0.91
Porcelain, glazed	72	0.924
Quartz, rough, fused	70	0.932
Refractory materials, 40 different	1110-1830	
poor radiators		0.65-0.75
		0.70
good radiators		0.80-0.85
		0.85-0.90
Roofing paper	69	0.91
Rubber		
Hard, glossy plate	74	0.945
Soft, gray, rough (reclaimed)	76	0.859
Water	32-212	0.95-0.963

* When two temperatures and two emissivities are given, they correspond, first to first and second to second, and linear interpolation is permissible.

† Although this value is probably high, it is given for comparison with the data, by the same investigator, to show the effect of oil layers. See Aluminum, part A of this table.

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Solution Steps

Using Fig. 8-13 the heat loss from the bare pipe is:

$$Q = (h_c + h_r)\, A_o\, (T_p - T_a) = (5.85)\,\frac{(\pi\ 8.625)}{12}\,(250 - 30)$$

$$= 2906 \text{ Btu/hr per linear ft}$$

For the insulated pipe, assume the outside surface of the insulation is at 50°F. Then from Fig. 8-13:

$$h_c + h_r = 4.5 \text{ Btu/(hr • sq ft • °F)}$$

$$Q = (4.5)\,\frac{(\pi\ 10.625)}{12}\,(50 - 30) = 250 \text{ Btu/hr per linear ft}$$

This agrees closely enough with the heat flow through the insulation — Example 8-1.

Heat saved = 2906 – 250 = 2656 Btu/hr per linear ft

FIG. 8-10

Partial Pressure of CO_2 Plus H_2O

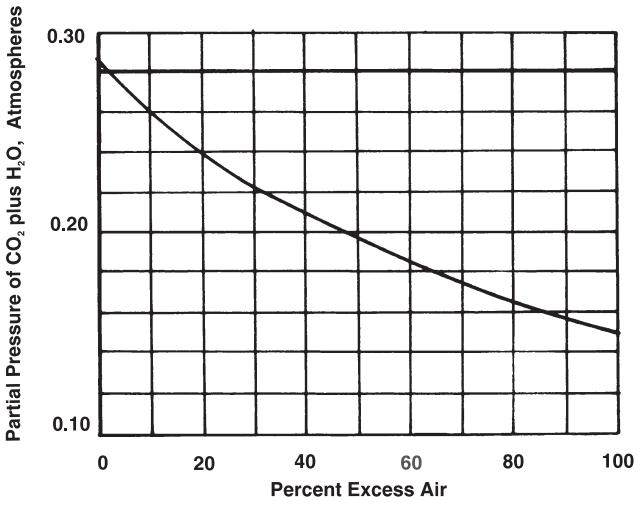


FIG. 8-11

Beam Length for Gas Radiation[8]

Dimension Ratio	Mean Beam Length, L
Rectangular Furnaces, Length-Width-Height, In Any Order	
1-1-1 to 1-1-3 1-2-1 to 1-2-4	(2/3) (Furnace Volume)$^{1/3}$
1-1-4 to 1-1-∞	(1) (Smallest Dimension)
1-2-5 to 1-2-∞	(1.3) (Smallest Dimension)
1-3-3 to 1-∞-∞	(1.8) (Smallest Dimension)
Cylindrical Furnaces, Diameter-Height	
1-1	(2/3) (Diameter)
1-2 to 1-∞	(1) (Diameter)

FIG. 8-12

Gas Emissivity[10]

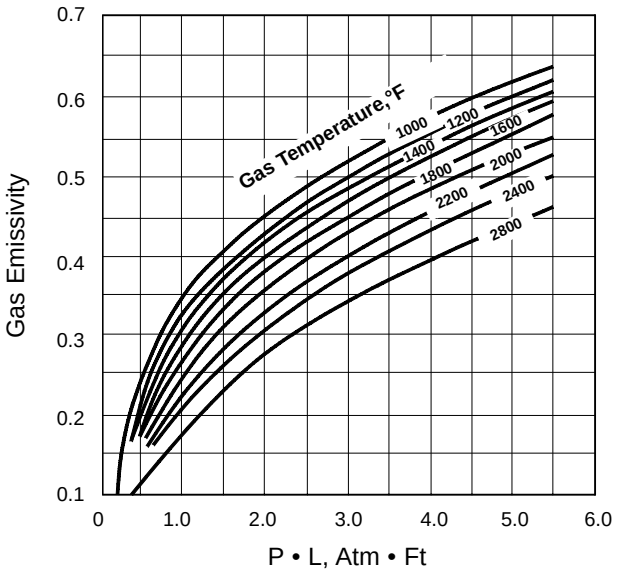
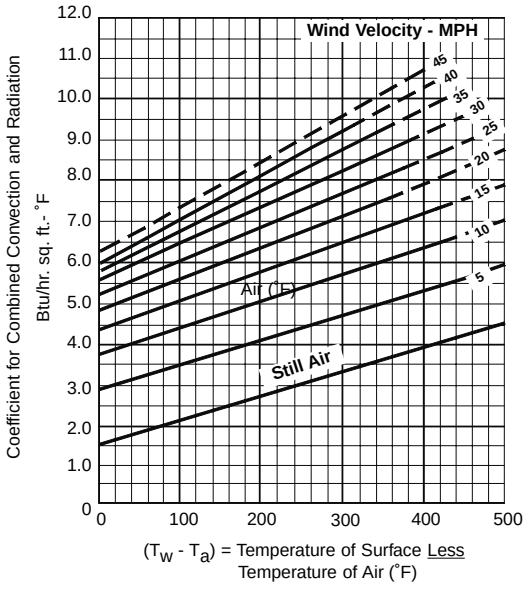


FIG. 8-13

Combination Convection and Radiation Film Coefficients for Air in Contact with Vertical Walls or Surfaces[11]



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COMBUSTION

Combustion is the rapid chemical reaction between oxygen and a combustible material that releases heat and light. Usually the combustible material is a hydrocarbon and ambient air supplies the oxygen. Complete combustion occurs when there is sufficient oxygen to convert all of the carbon to carbon dioxide and all of the hydrogen to water. Incomplete combustion means that there is either unburned or partially reacted fuel, i.e., carbon monoxide, hydrogen, etc.

Methane is the main constituent of natural gas. It reacts with oxygen to form carbon dioxide and water.

$$CH_4 + 2\ O_2 \rightarrow CO_2 + 2\ H_2O$$

This stoichiometry is typical of all hydrocarbons. One atom of carbon requires one molecule of oxygen and four atoms of hydrogen require one molecule of oxygen. The theoretical air is that needed for complete combustion of the carbon and hydrogen, i.e., two molecules of oxygen for one molecule of methane. Excess air is that supplied in addition to what is required. For example, 20% excess air means that the air supplied is 1.2 times the stoichiometric amount.

The following reaction represents the complete combustion in oxygen of an arbitrary carbon-based fuel compound. For combustion in air, add N_2 to both sides of the reaction equation in an amount equal to the oxygen input multiplied by 3.774.

$$C_cH_{hy}O_oS_sN_{ni} + \left(c + \frac{hy}{4} - \frac{o}{2} + s\right)O_2 \rightarrow$$

$$(c)\ CO_2 + \left(\frac{hy}{2}\right)H_2O + (s)\ SO_2 + \left(\frac{ni}{2}\right)N_2$$

Fig. 8-14 shows how the composition of the flue gases depends on the amount of combustion air.

Air Requirements

Fig. 8-15a shows the scf of dry air needed per scf of paraffinic hydrocarbons for complete combustion in terms of the specific gravity of the fuel. In using this figure any inert components in the fuel, e.g., nitrogen, carbon dioxide, etc., must be excluded. Fig. 8-15b shows the mass of humid air required per mass of dry air at 760 mm Hg and percent relative humidity.

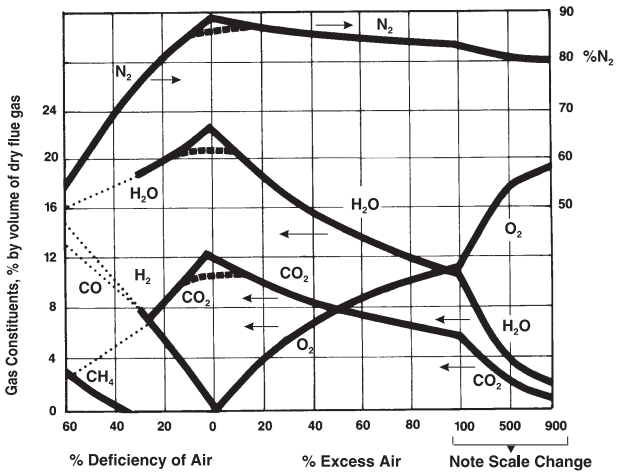
Air is about 20.9% oxygen on a dry basis, hence 4.77 mols (or scf) of air supply 1.0 mol (or scf) of oxygen. Applying this to methane, 9.54 mol (or scf) of air are needed for every mol (or scf) of methane.

The effect of water vapor in the air is relatively small at low and moderate temperatures. Saturated air at 60°F contains 1.75% water. Still this should be considered and 2-3% more air is usually added if exact calculations are not made. The water content in saturated air increases rapidly with temperature; e.g. at 100°F saturated air contains about 6.5% water, and at 115°F it contains about 10%.

Some situations may yield a higher amount of water vapor coming from the combustion air and fuel gas. Consider the complete combustion of 1 mol of water saturated methane at 100°F and 15 psig with 20% excess air with air also water saturated at 100°F. This situation introduces 0.79 mol of water from the air and 0.032 mol of water from the gas. Additionally, 2 mols of water from the methane combustion is added water resulting in approximately 21% water in the flue gas of which 30% is from the air and gas humidity. Also, steam or water addition for NO_x control introduces more water vapor to the flue gas. These situations increase the wet bulb temperature

FIG. 8-14

Effect of Fuel/Air Ratio on Flue Gas Analysis for 1108 Btu/SCF Natural Gas (0.63 Gas Gravity) Containing 83% CH_4 and 16% C_2H_6[1]



NOTE: Values for rich mixtures depend somewhat on combustion chamber design. The average values shown are within ½% of correct for H_2, CO, and CO_2, but may be as much as 2% low for other constituents. Some external heat is usually required for mixtures with less than 70% aeration (dotted lines). Dashed lines show the trends with poor mixing or quenching. Gas constituent percentages are on a dry basis to allow comparison with gas analyzer readings, that measure the gas volumes after water vapor has condensed out of the sample. With the correct amount of air (10.56 ft^3), each cubic foot of this fuel gas produces 2.14 ft^3 H_2O, 1.15 ft^3 CO_2, 8.39 ft^3 N_2; so $\%CO_2 = 100 \times 1.15\ ft^3\ CO_2 \div (1.15 + 8.39)\ ft^3$ dry flue gas = 12.1%.

of the flue gas. Water condensation should be considered in mass and energy balances and excess air calculations. Errors in consideration of water vapor content and air temperature may cancel a 10% excess air calculation, resulting in incomplete fuel combustion. Designs and operations should consider local weather conditions and seasonal changes.

The theoretical air requirement of an arbitrary carbon-based fuel compound, in mols of dry air per mol of fuel, can be calculated with the following equation.

$$AO = 4.77 \cdot \left[c + \frac{hy}{4} - \frac{o}{2} + s\right] \qquad \textbf{Eq 8-15}$$

Analysis of the flue gases provides useful information about the actual excess air and the efficiency of fired equipment. The following equations provide the excess air percentage for a sulfur and oxygen free, carbon-based fuel combustion without soot formation. Analysis must be molar and on a dry basis.

$$EA = \frac{Fl_o}{AO}\left[\frac{CO_{2_o}}{CO_2 + CO}\right] \cdot 100$$

$$= \frac{Fl_o}{AO}\left[\frac{O_{2_o}}{20.9 - O_2 + CO}\right] \cdot 100 \qquad \textbf{Eq 8-16}$$

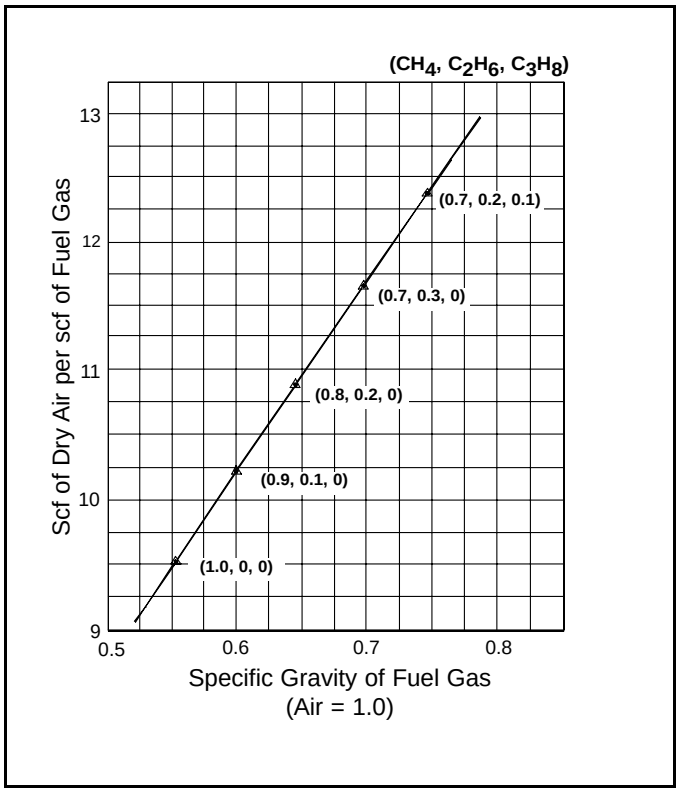
Many forced-draft burners supply a fixed volume of air. Fig. 8-18 shows the effect of ambient temperature and barometric pressure on the amount of air actually delivered.

The air ideal density and ideal relative density changes with the humidity. For the same relative humidity, the hotter the air, the higher the influence of the humidity in the air densities, because of the higher water vapor fraction, as can be seen in Fig. 8-16 and Fig. 8-17. In these graphs the atmosphere

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FIG. 8-15a

Standard Cubic Feet of Dry Air Needed per Standard Cubic Foot of Hydrocarbon for Complete Combustion



pressure is 760 mm Hg; for other conditions use the equation shown in the graph.

Heating Value

The heating value is the amount of ideal heat released by a unit of fuel, mass or standard volume, during complete combustion, at constant pressure, with the stoichiometric amount of dry air. The fuel humidity must be stated, as well as whether the gas volume is real or ideal. This involves a heat balance at standard conditions of temperature and pressure, (usually 60°F [15°C] and 760 mm Hg).

Hydrogen in the fuel burns to water and when the flue gases are cooled to 60°F, the physical state — either vapor or liquid — of this water must be assumed. So the latent heat of vaporization of the water may or may not be considered to be part of the heating value. The result is two definitions for the heating value. The higher or gross heating value, HHV, includes the heat of condensation and the lower or net heating value, LHV, assumes the water remains in the vapor state.

Fig. 23-2 in the "Physical Properties" section gives the net and gross heating values of most pure hydrocarbons. For mixtures calculate the molar, or volume, average.

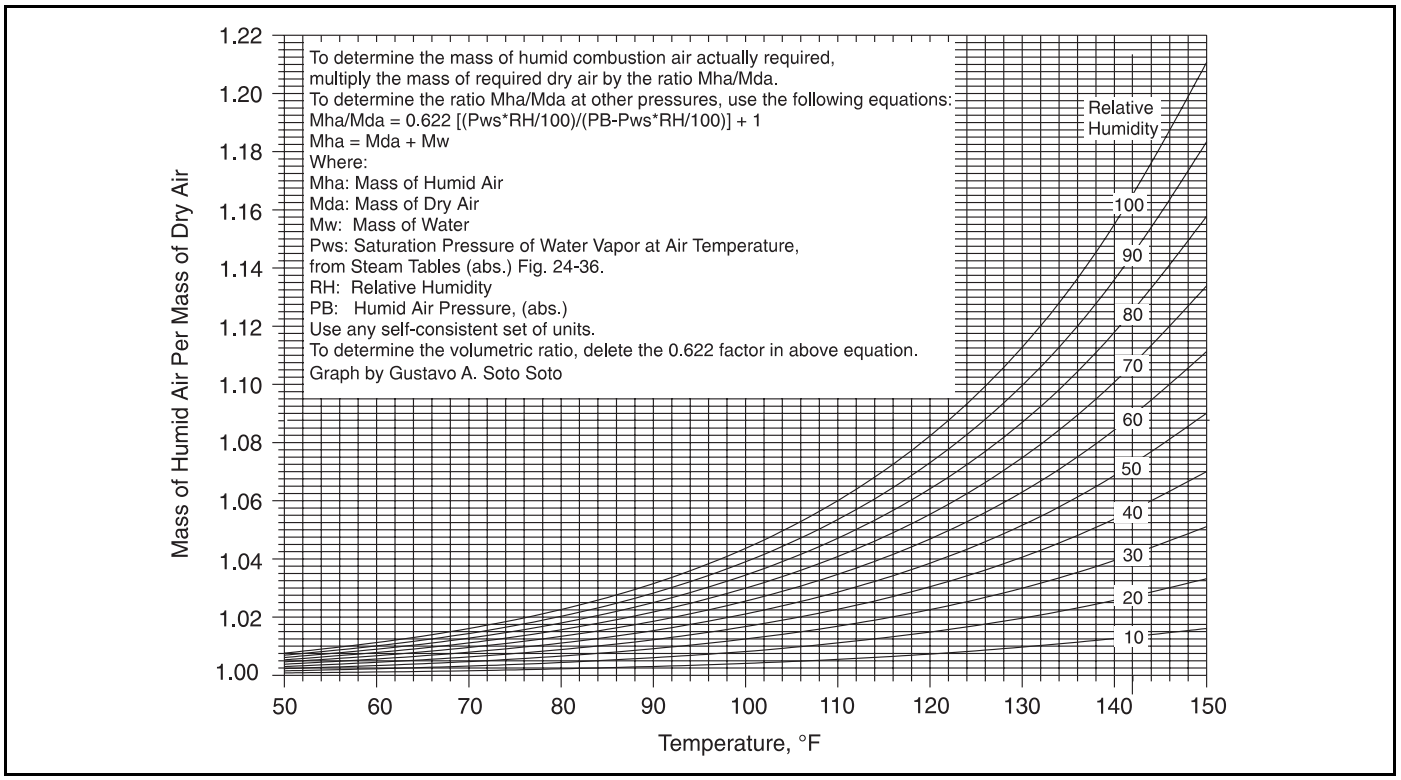
The higher, ideal, dry heating value of sweet natural gas at 60°F and 760 mm Hg may be calculated with the following equation:

$$HHV = 1568.72 \bullet SG - 2524.88 \bullet X_{CO2} - 1658.37 \bullet X_{N2} + 141.05 \qquad \textbf{Eq 8-17}$$

Where HHV is in Btu/scf, SG is the ideal relative density, and X is the fraction of carbon dioxide and nitrogen if present.

FIG. 8-15b

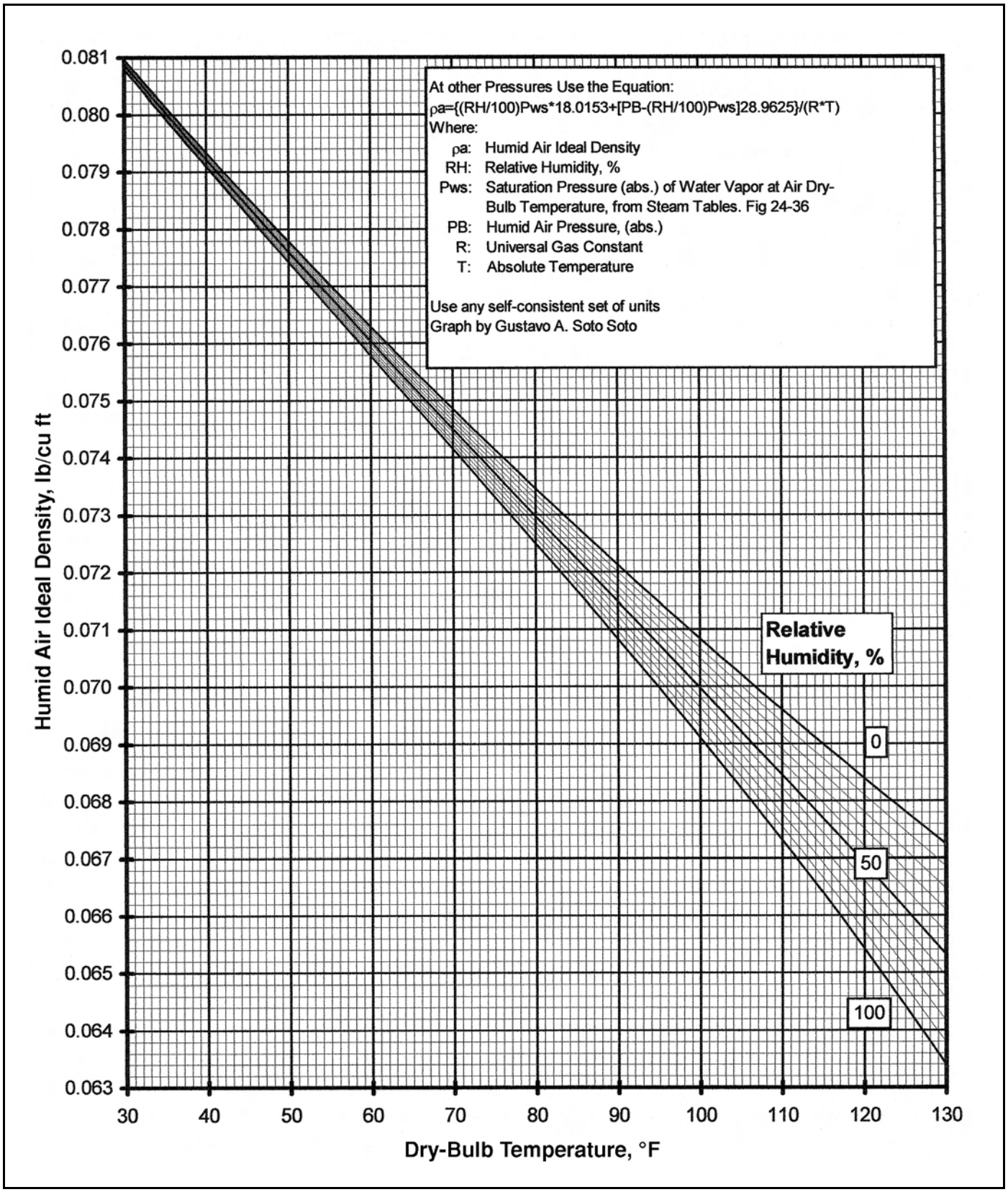
Mass of Humid Air Per Mass of Dry Air at 760 mm Hg and Percent Relative Humidity



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FIG. 8-16

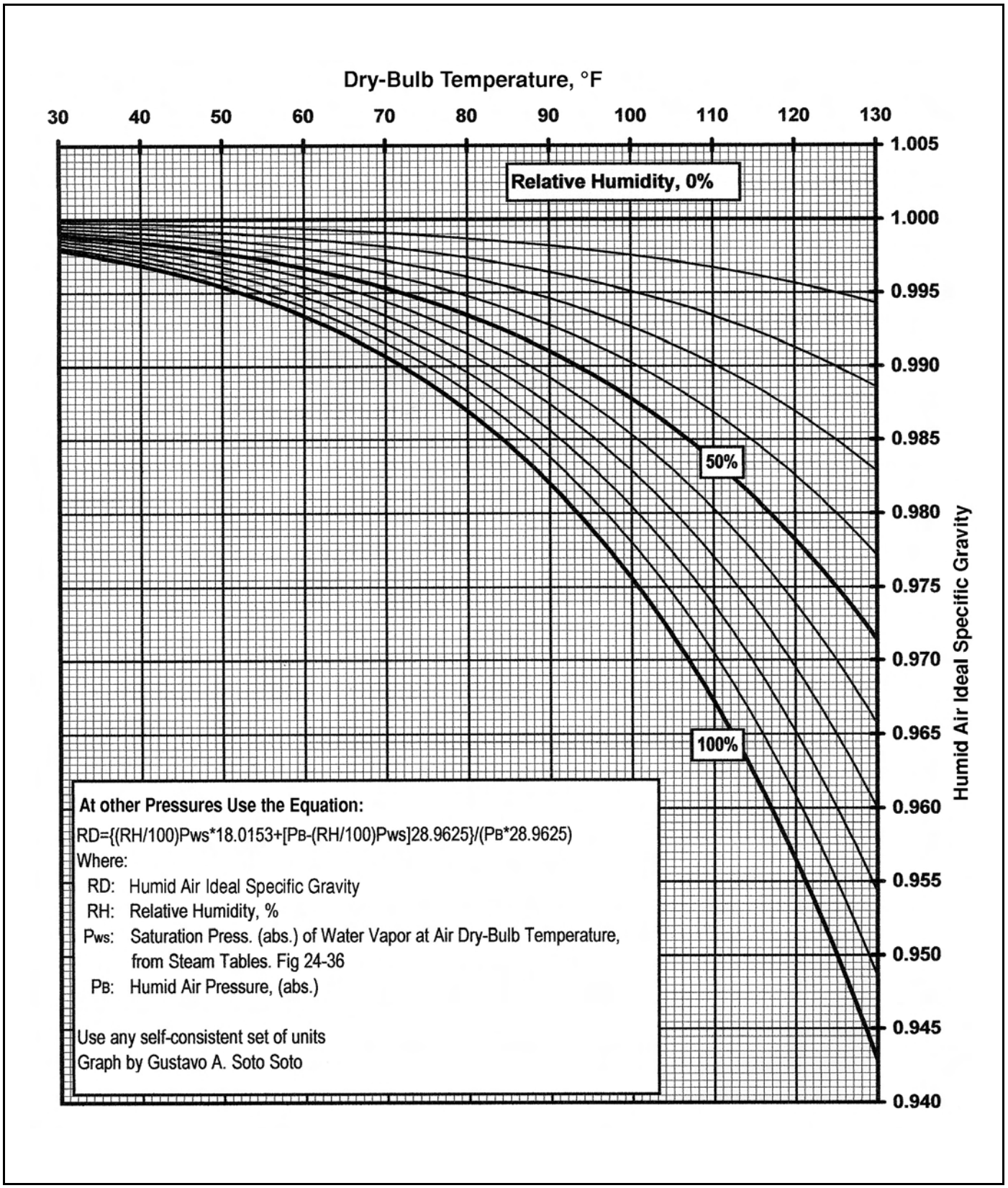
Humid Air Ideal Density
760 mm Hg Air Pressure (Absolute) and Indicated Dry-Bulb Temperature and Relative Humidity



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FIG. 8-17

Humid Air Ideal Specific Gravity
760 mm Hg Air Pressure (Absolute) and Indicated Dry-Bulb Temperature and Relative Humidity



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Thermal Efficiency

The two ways to express the heat released during combustion result in two definitions for the thermal efficiency, gross and net.

$$GTE = \frac{UHT}{GHI} \qquad \textbf{Eq 8-17a}$$

$$NTE = \frac{UHT}{NHI} \qquad \textbf{Eq 8-17b}$$

Therefore NTE exceeds GTE. There is a tendency to use the gross heating value with the net thermal efficiency even though the bases are inconsistent because the numerical values are higher than the corresponding net heating value and gross thermal efficiency.

In the above equations, when GHI and NHI include only the fuel energy input, the efficiency is called Fuel Efficiency. When it also includes the input total enthalpy above the reference temperature of all the incoming streams, the efficiency is called Thermal Efficiency. The total enthalpy must include not only the sensible energy of the combustion air, fuel, and steam, if used, but also the water vapor latent heat. In some high efficiency fired heater and waste-heat-recovery units water from the air, fuel and combustion may condense.

When hot humid fuel and/or air are used, as in the case of using regeneration gas as fuel, after passing the regenerators, the amount of latent heat may be considerable. In the case of using gas turbine exhaust gases in place or in conjunction with fresh air, above the water vapor, due consideration must be given to the CO_2 content in the enthalpy calculations. In the heat balance, care must be taken in defining the thermodynamic volume of control, and the routes of any recirculated or bypassed stream.

Fig. 8-19 shows how the gross thermal efficiency can be determined from the excess air and stack gas temperature. Especially for insulated heaters or furnaces, the combustion efficiency is close to the gross thermal efficiency. The difference is the heat lost through the walls to the surroundings.

The typical enthalpy of the combustion (or stack) gases when natural gas is burned with 20% excess dry air is shown in Fig. 8-20.

Draft

Combustion air is obtained by natural, forced, and induced draft. Natural draft uses the buoyant effect of the hot flue gases in the stack to draw air into the combustion zone. Forced draft is the result of an air blower or fan. Induced draft refers to a blower in the stack.

The draft available (inches of water) from a stack is the natural (or static) draft less the frictional and exhaust gas velocity losses. As defined below, the draft is always negative. The available draft should be sufficient to overcome the head losses in the air inlet registers, convection section tube coils, baffles, damper, and any waste heat recovery devices. Also, it should contain a margin of safety to allow the damper to be in an intermediate position to deal with site conditions such as atmospheric pressure changes, humidity changes, and temperature changes (daily and seasonal variations). Also, the draft should be sufficient to obtain a negative pressure along the entire heater fire side flow path.

In references 35 and 36, information can be found about pressure losses in various devices such as coils, duct transitions, etc. Changes in velocity head should be considered in

FIG. 8-18

Effect of Ambient Temperature and Barometer Pressure on Air Actually Delivered

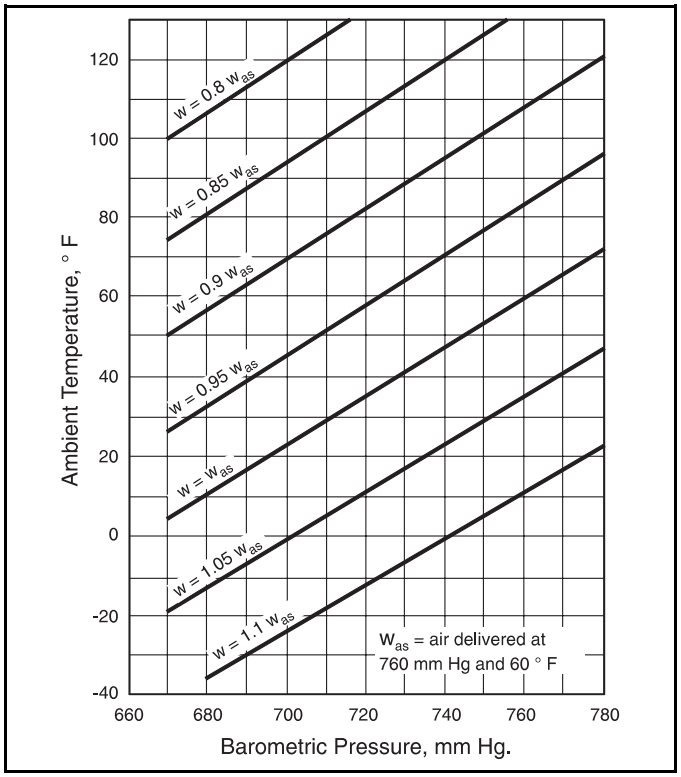
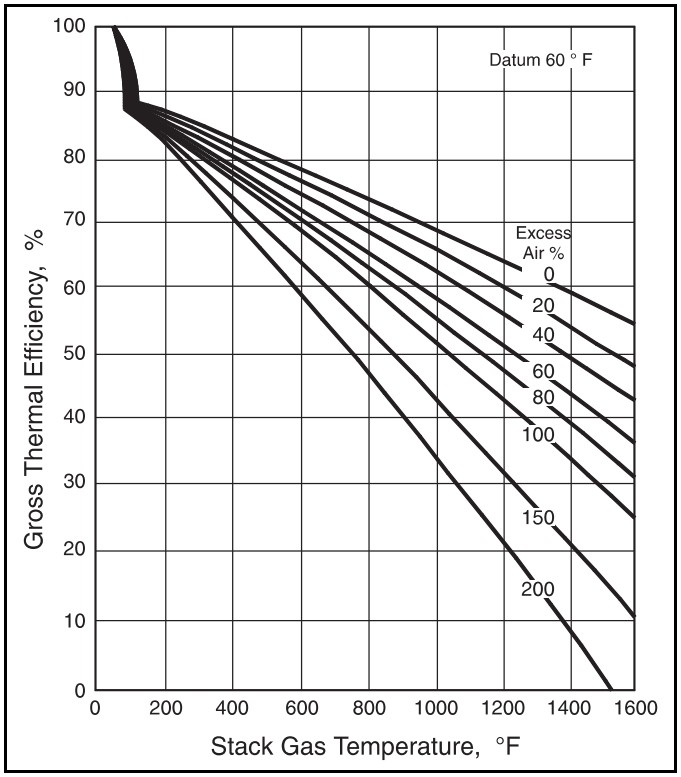


FIG. 8-19

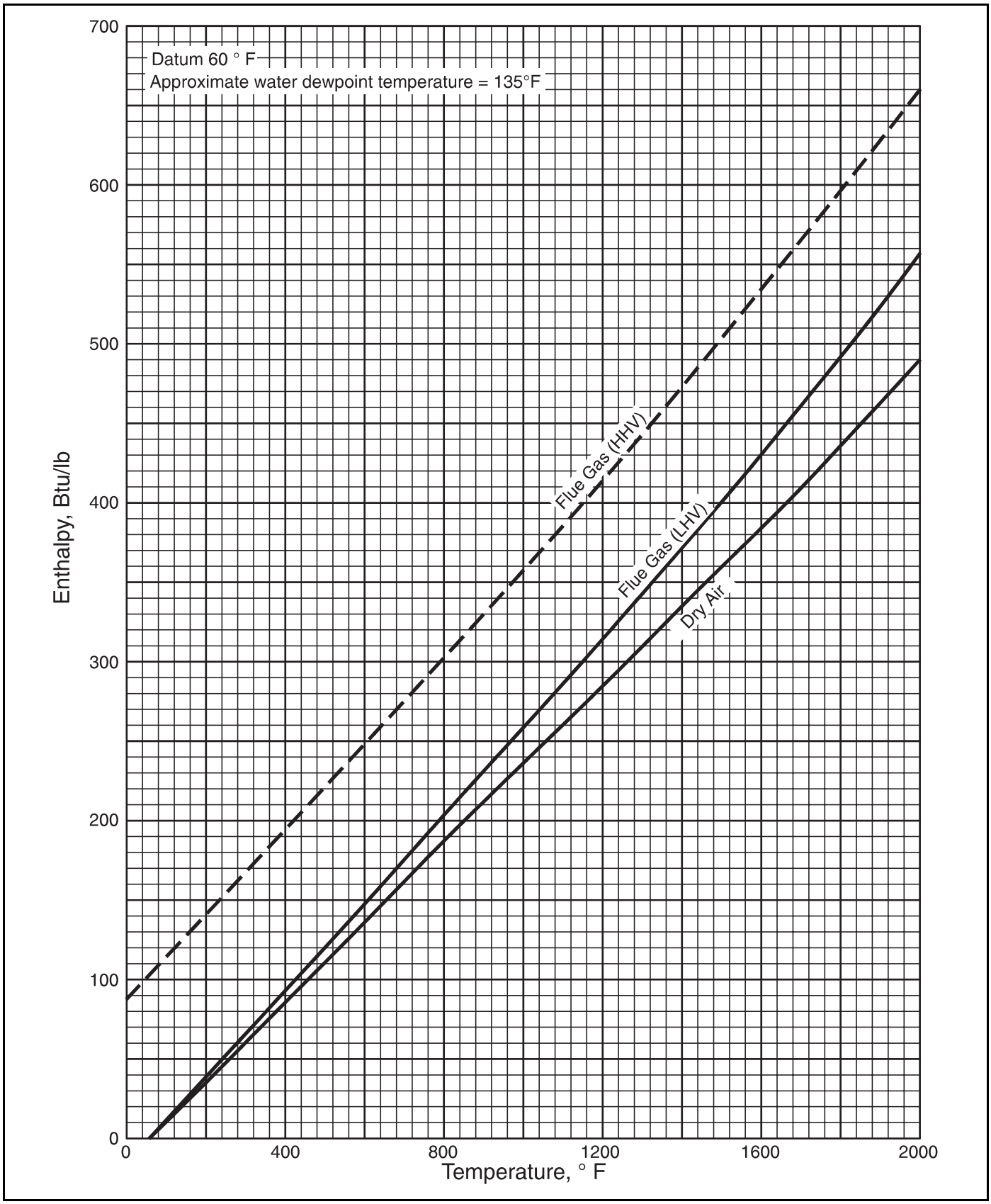
Gross Thermal Efficiency for a Gas with HHV = 1000 Btu/scf



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FIG. 8-20

Typical Enthalpy of Combustion Gases for a Dry Natural Gas Fuel and 20% Excess Dry Air



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head loss calculations because of the low densities of the air and flue gas. The absolute roughness of the inner stack depends on the material, construction, and lining, if any. The absolute roughness may range from 0.001 to 0.030 ft. The equipment manufacturer should be consulted for more precise values of stack roughness and for other factors which may influence total head loss.

Refer to Fig. 17-2 to determine the Fanning friction factor (f_f) using the Reynolds number determined from Eq. 8-9a and the relative roughness (dimensionless).

$$Hav = 0.192 \cdot H_s (\rho_g - \rho_a) + 0.0029 \cdot V^2 \cdot \rho_g \left(4 \cdot f_f \cdot \frac{H_s}{D_s} + 1\right) \quad \textbf{Eq 8-18}$$

Also

$$Hav = 0.2545 \cdot H_s \cdot P_B \cdot \left(\frac{SG_g}{T_g} - \frac{SG_a}{T_a}\right) + + \frac{2.737 \cdot T_g}{P_B \cdot SG_g \cdot D_s^4} \cdot \left(\frac{M}{100{,}000}\right)^2 (4 \cdot f_f \cdot \frac{H_s}{D_s} + 1) \quad \textbf{Eq 8-19}$$

Since ρ_g is less than ρ_a, the first term in Eq 8-18 which is the total available driving force, is always negative. This must be decreased by adding the losses from friction and resistance in the second term.

Burners

Four types of burners are commonly used in direct fired heaters:

- **Inspirating pre-mix burners**. The passage of fuel gas through a venturi pulls in the combustion air. These burners have short dense flames that are not affected by wind gusts.
- **Raw gas burners**. Some of the air required for combustion is pulled in by a venturi. The rest of the air is admitted through a secondary air register. These burners have larger turndown ratios, require lower gas pressures, and are also quieter.
- **Low NO_x burners.** Designed to reduce formation of NO_x below levels generated in conventional burners during normal combustion. The most common are fuel staged and air staged to reduce the flame temperature and excess air.
- **Combination gas and oil burners**. An oil burner is added to the gas spider so that fuel oil can also be used. One-tenth pound of steam per pound of fuel is usually required to atomize the oil.
- **Self-recuperative burners.** A heat exchanger in the burner apparatus heats air using the flue gases.
- **Self-regenerative burners.** There is a pair of heat exchanger storing bodies in the burner apparatus; alternatively one body stores the heat from the flue gases, while the other regenerates with the combustion air receiving heat from the flue gases.
- **Submerged combustion burners.** The flame is located in a submerged chamber where heat is transferred to the water and the flue gases are discharged through a bubbling tube into the water.
- **Radiant Wall burners.** The flame is designed to spread across the burner tile and the furnace wall refractory without any forward projection into the firebox.

Burner air registers are sometimes used to control the excess air. Ten to fifteen percent excess air is a compromise for best thermal efficiency and lowest NO_x levels. Air leakage into the heater around sight door openings, header box gaskets, terminal penetrations, etc. should be minimized.

Unmuffled burners have 100 to 110 dBA noise levels. Requirements for 85 or 90 dBA noise levels, measured three feet from the heater, require noise attenuation plenums and orifice mufflers.

Most burners have a continuous pilot flame that releases up to 50 MBtu/hr. The pilot is lit by hand or with a spark plug. The pilot should be left on when:

- The furnace will not overheat during no-flow conditions.
- The fuel is ethane or heavier gases.
- The furnace is used intermittently, e.g., regeneration gas heaters.
- The pilot heat release does not affect furnace turndown ratio.
- The refractory must remain dry for fast start-up.
- Nuisance shutdowns are unavoidable.

Gas Burner Performance

Several factors influence gas burner performance, such as the gas pressure, temperature, and composition that affect heating value, gas density and combustion characteristics; also, the aeration, partial heat load, local altitude, etc. Some typical problems that can occur as a consequence of incorrect burner selection, or from changing operating conditions, or using non-interchangeable gas include flashback, yellow tipping, flame lifting, sooting, and inadequate heat input.

The change in gross heat input through a gas burner orifice caused by changes in the operating conditions and gas composition can be estimated with the following equation, where the pressure and temperature are absolute. The equation can also be used for the lower heating value. The term in parenthesis is called the Wobbe Index, which is a gas interchangeability parameter.

$$\frac{GHI_2}{GHI_1} = \frac{Y_2 \cdot \sqrt{\frac{Pg_2 \cdot \Delta P_2}{T_{g2}}} \cdot \left(\frac{HHV_2}{\sqrt{SG_2}}\right)}{Y_1 \cdot \sqrt{\frac{Pg_1 \cdot \Delta P_1}{T_{g1}}} \cdot \left(\frac{HHV_1}{\sqrt{SG_1}}\right)} \quad \textbf{Eq 8-20}$$

In Eq 8-20 the temperature and pressure are absolute, and the expansion factor (Y) is a function of the burner nozzle characteristics and the fuel inlet and outlet pressures. For low inlet gas pressure the expansion factor is approximately 1, and for small changes in the fuel gas pressure the expansion factors Y_1 and Y_2 have similar values and may be ignored. In other circumstances consult the burner manufacturer for data.

Flue Gas Condensation

Flue gas water condensation may produce corrosion problems caused by acid gases such as SO_2 when present in the flue gas because of the sulfur content of some fuels, and therefore temperature control of the flue gas above the SO_2 dewpoint should be considered. However, highly efficient fired equipment or waste heat recovery equipment burning sweet fuels and/or with acid resistant duct materials may utilize flue gas water condensation taking advantage of its latent heat.

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Flue Gas Dew Point

Flue gas normally has high vapor water content, coming from the hydrocarbon combustion, the atmospheric air, and, eventually from steam injected in the heater. The flue gas temperature is reduced by the heat transfer along the heater flow path; and may reach the dew point. This depends on fuel composition, air humidity, excess air, steam injection, heater and heat-recovery units efficiency, gas turbulence and boundary layer behavior. Some oxides from the flue gas dissolve in the condensed water and form acids causing a variation in the dew point. When high-quality sweet fuels are used, the water condensation is not a problem if adequate materials are used. When sour fuels are used the acid formation may cause corrosion problems if the flue gas is not above the dew point.

The following equations may be used to estimate the acid dew point, where the dew points are given in K and partial pressures in mm Hg.[41]

For sulfurous acid:

$$\frac{1000}{T_{DP}} = 3.9526 - 0.1863 \bullet ln\,(P_{H2O}) + 0.000867 \bullet ln\,(P_{SO2}) - 0.000913 \bullet ln\,(P_{H2O}) \bullet ln\,(P_{SO2})$$

For sulfuric acid:

$$\frac{1000}{T_{DP}} = 2.276 - 0.0294 \bullet ln\,(P_{H2O}) - 0.0858 \bullet ln\,(P_{H2SO4}) - 0.0062 \bullet ln\,(P_{H2O}) \bullet ln\,(P_{H2SO4})$$

For nitric acid:

$$\frac{1000}{T_{DP}} = 3.6614 - 0.1446 \bullet ln\,(P_{H2O}) - 0.0827 \bullet ln\,(P_{HNO3}) + 0.00756 \bullet ln\,(P_{H2O}) \bullet ln\,(P_{HNO3})$$

In feedwater economizer applications, typical acid dew point temperature, minimum allowable stack temperature and minimum allowable feedwater inlet temperature for some fuels are given in the following table:

Acid Dewpoint Temperature of Various Fuels[42]

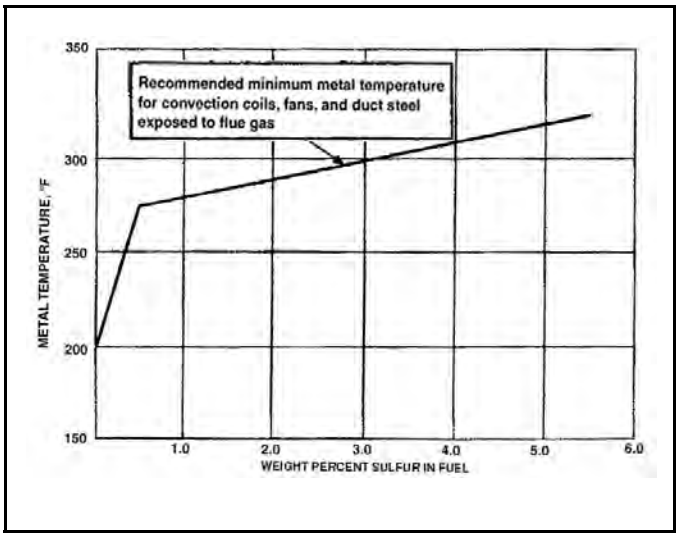
Fuel	Acid Dewpoint Temperature (°F)	Minimum allowable Stack Temperature (°F)	Minimum Allowable Feedwater Inlet Temp. (°F)
Natural Gas	150	250	210
#2 Oil	180	275	210
Low Sulfur Oil	200	300	220

Courtesy of Kewanee Boiler Manufacturing

API Standard 560[23] recommends the minimum metal temperature for the parts in contact with the flue gases, as a function of the sulfur content in the fuel, as shown in the Fig. 8-21.

FIG. 8-21

Recommended Minimum Metal Temperatures[23]



NO_x Control

The main factors influencing NO_x formation are flame temperature, excess air in the flame, time in which combustion gases are at flame temperature, and fuels containing nitrogen compounds.

In natural draft heaters the most common means of changing NO_x emissions is through the use of low NO_x burners. The premix and raw gas burners produce NO_x levels greater than 0.13 lbs NO_x /MMBtu of burner heat release. These are generally used when no NO_x requirement is specified. The upper end of the low NO_x burner design uses a partially staged raw gas burner to achieve levels of 0.08 to 0.12 lbs NO_x /MMBtu. The mid range low NO_x burner design uses staged gas to achieve levels of 0.05 to 0.08 lbs NO_x /MMBtu. For lower NO_x requirements staged gas tips and internal flue gas recirculation are combined to produce NO_x emissions at 0.03 lbs NO_x /MMBtu and below. These numbers are based on clean fuel gas (not oil burning) and are dependent on firebox temperature and excess air.

The current Lowest Achievable Emissions Requirement (LAER) requires NO_x to be less than 9 ppm (at 3% O_2, dry) with future regulations expected to reduce to less than 5 ppm (1 ppm = 0.011 lb NO_x / MMBtu). LAER also requires less than 30 ppm of CO and untralow hydrocarbon emissions.

A staged-air burner is a low NO_x burner in which a portion of the combustion air is injected downstream of the burner block to mix with the combustion products and unburned fuel from the primary combustion zone.

A staged-fuel burner is a low NO_x burner in which a portion of the fuel is mixed with all of the combustion air within the burner block while a second portion of the fuel is injected downstream of the burner block to provide delayed combustion.

The injection of water or steam is also used to reduce the NO_x formation by reduction of the peak flame temperature.

DIRECT FIRED HEATERS

Direct fired heaters vary in size from 0.5 MMBtu/hr small package regeneration gas heaters to 1000 MMBtu/hr steam hydrocarbon reformer heaters. In the gas processing industry, the usual range is 1 to 20 MMBtu/hr.

Types

There are two basic configurations: cylindrical and cabin, see Fig. 8-22. The simplest design is vertical-cylindrical with only radiant tubes. The NTE is about 60% and the stack gas temperature is 1200°F or more. The burner in the floor fires upward. A stainless steel baffle slows the exit flow of the hot

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gases and reradiates heat back to the top part of the tubes. There is a short stack that usually has no damper. The design is low cost and suited for low cost fuel.

Adding a convection section improves the NTE to about 80%. The radiant section may be either cylindrical or cabin, and the coil configuration either helical or serpentine. These heaters cost more than the all-radiant type but they use less fuel for any given duty.

By cooling the combustion gases to about 300°F, the NTE can be increased to over 90%. This requires either a combustion air preheater using exhaust gas or an additional convection section. These units have the highest capital cost and lowest fuel requirement for any given duty.

Design modifications are used when the tube material is expensive. A bridge wall is installed down the center of the cabin. The radiant tubes are placed above the bridge wall so that they are, in effect, double fired.

Cylindrical or Cabin? Vertical or Horizontal Tubes?

Cylindrical heaters have the following advantages:

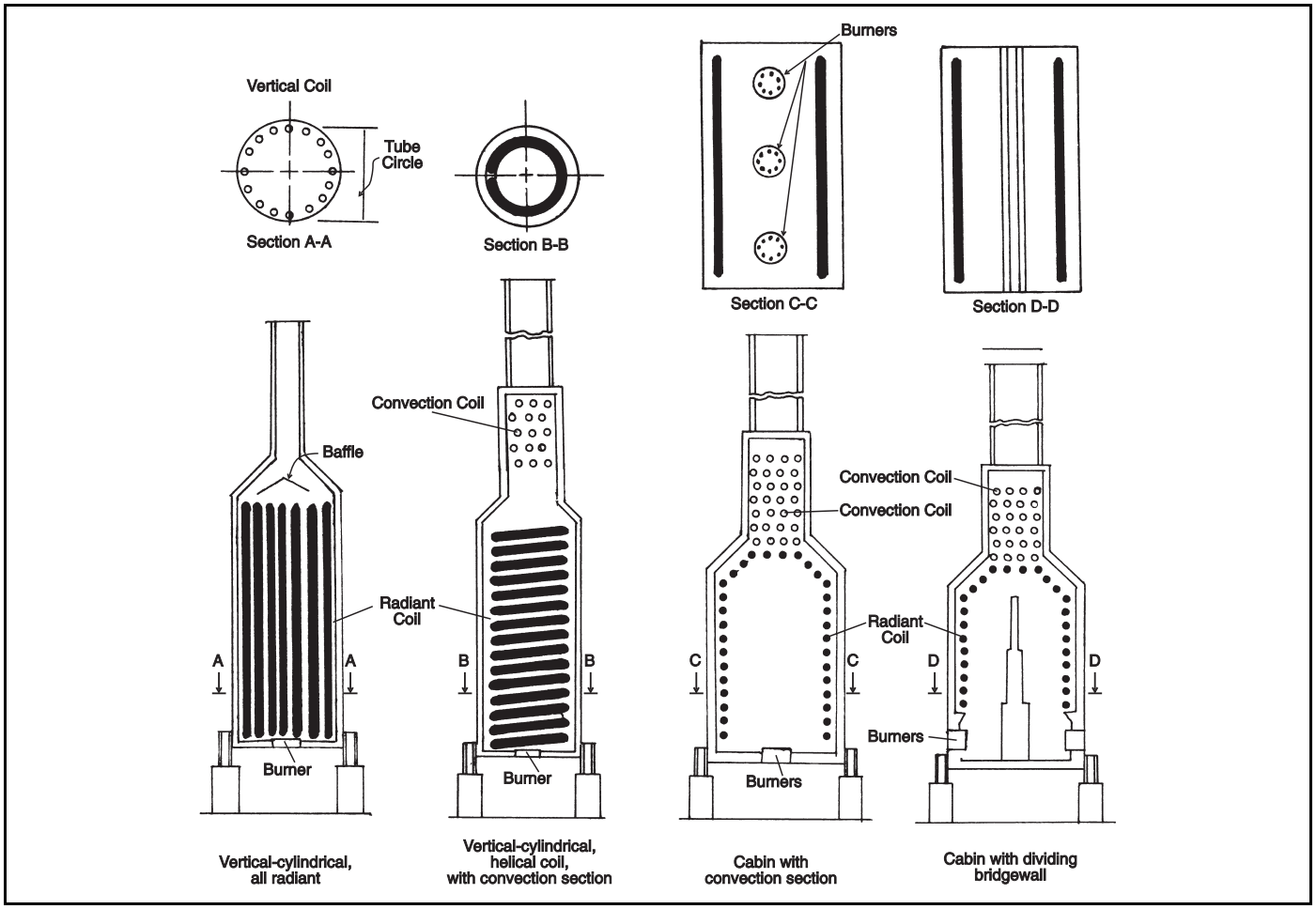
- They require the smallest plot area for a given duty.
- The cost is usually 10% to 15% lower in the larger sizes.
- They can accommodate more parallel passes in the process coil.
- For large duties, a cylindrical heater has a taller firebox and more natural draft at the burner.
- The flue gas velocity is usually higher in the convection section, hence, the flue gas film coefficient is higher.
- Fewer expensive tube supports or guides are required in the convection section.
- The noise plenums or preheated combustion air plenums are smaller.
- Fewer soot blowers are required in the convection section. Soot blowers are not needed for gaseous fuel.
- If coil drainage is a problem, a helical coil may be used when there is only one pass.

Cabin heaters have the following advantages:

- The process coil can always be drained.
- Two-phase flow problems are less severe. (Slug flow can generally be avoided.)
- Cabins can accommodate side-firing or end-firing burners instead of only vertically upward firing. This permits the floor of the heater to be closer to the ground. (Some burner manufacturers prefer to fire liquid fuels horizontally.)

FIG. 8-22

Example Cylindrical and Cabin Direct Fired Heaters



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- A smaller capital investment is required when the duty is less than 10 MMBtu/hr.

Radiant Section

The radiant section or firebox should:

- Obtain complete combustion of the fuel with a reasonable amount of excess air, i.e., 10% to 15%.
- Contain the flame and avoid impingement on the tubes.
- Distribute the radiant heat flux.
- Cool the combustion gases to 1500-1900°F to protect the convection section.

The proportions of the firebox are the key to good performance. Generally the flame length should be 60% of the firebox length and the clearance between the flame and tubes at least 1.5 ft. For small cylindrical heaters, the tube circle should be equal to the length of the firebox. For small cabin heaters, the width, height, and tube length should be equal. For large heaters the height of a cylindrical heater is twice the tube circle, and for cabins a good ratio of width to height to length is 1:2:4. The firebox shell is reinforced steel plate. The insulation behind the tubes is usually 5 in. of 1:2:6 lumnite (cement), haydite (aggregate), vermiculite (insulation) castable. The floor is at least 6 in. of 1:2:6 castable, often with a firebrick surface. The bridge wall is always firebrick.

By far the most common tube material is A-106B carbon steel. The nominal size range is 2 to 8 in. with 3 and 4 in. the most prevalent. Short radius return bends are standard and the tubes are usually 1.5 nominal diameters from the refractory wall. For these arrangements, the maximum heat flux directly facing the flame is 1.9 times the average flux. (With long radius return bends the maximum heat flux is 1.45 times the average.) The flux to the front 60° of the tube is 1.8 times the average and the front half-tube flux is 1.5 times the average. Any flux maldistribution due to tall narrow fireboxes or short flames, usually less than 15%, must be added to this. For double firing, the circumferential maldistribution is reduced from 1.8 to 1.25.

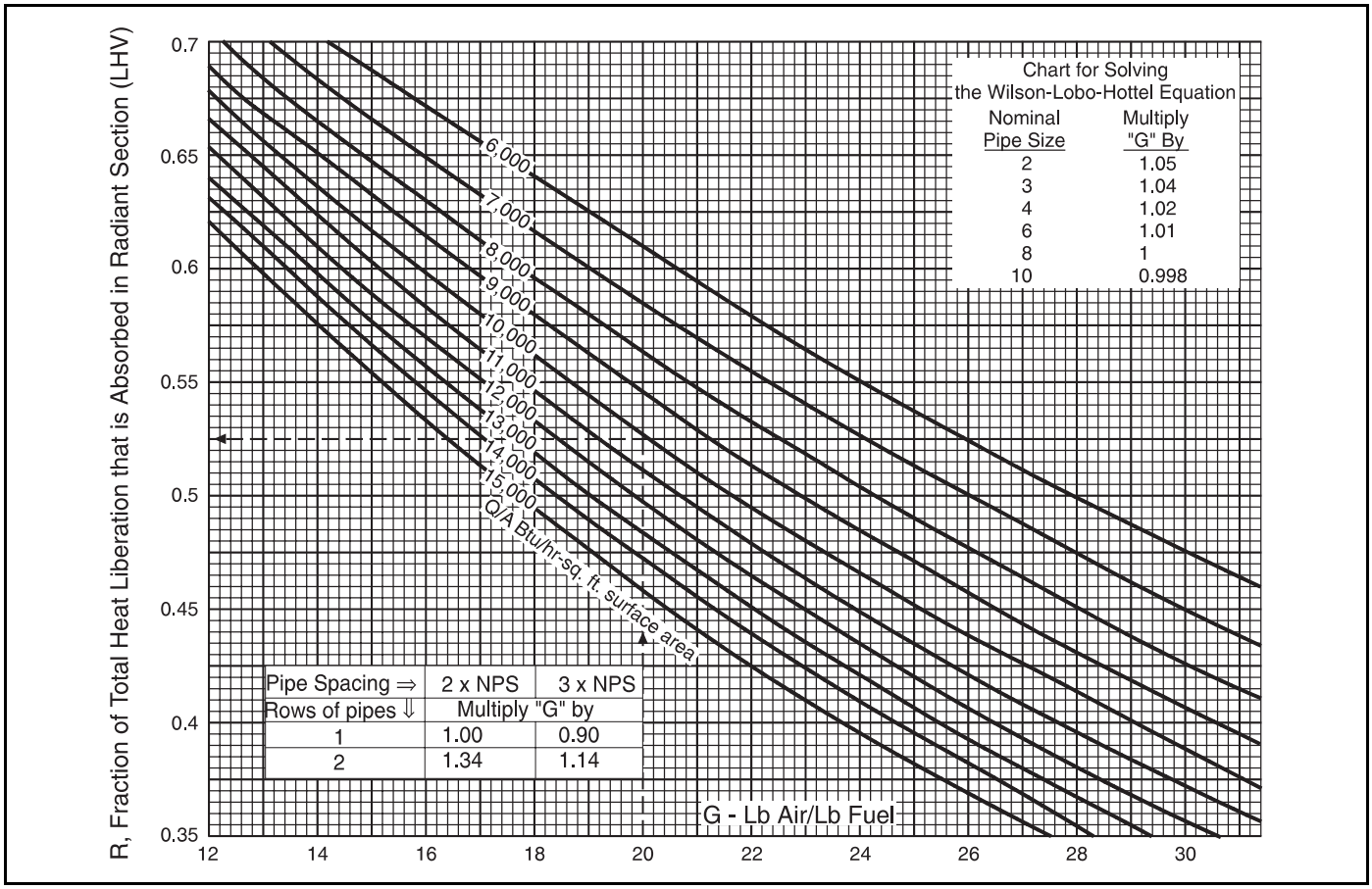
Equations 8-21 and 8-22, as well as Fig. 8-23, may be used to obtain an estimate of the absorbed heat in the radiant section of a fired heater, expressed as a fraction of the total net heat liberation, in terms of the average heat flux to the tubes (circumferential), the arrangement of the tubes, and the air to fuel weight ratio. These equations are solutions of the Wilson-Lobo and Hottel equation.

$$B = \frac{d_o \cdot n \cdot \pi \cdot I \cdot G^2}{S \cdot a} \qquad \textbf{Eq 8-21}$$

$$R = 1 - \frac{\sqrt{(B^2 + 70.56 \cdot 10^6 \cdot B)} - B}{35.28 \cdot 10^6} \qquad \textbf{Eq 8-22}$$

FIG. 8-23

Chart to Estimate the Fraction of Total Sensible Heat Liberation That is Absorbed in the Radiant Section of a Direct Fired Heater



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where “a” is a constant depending on the arrangement of the tubes. The “a” value is:

	Tube spacing	
No. of rows	2 • NPS	3 • NPS
1	0.88	0.73
2	0.99	0.91

The total heat liberation consists of the lower heating value of the fuel and the sensible heat in combustion air, recirculated flue gas, and fuel and atomizing steam, all heat contents referred to a datum of 60°F.

Fig. 8-23 provides an estimate of the fraction of the total heat liberation that occurs in the radiant section in terms of the allowable heat flux to the tubes. The lb air/lb fuel fired is needed and this can be obtained from either Fig. 8-24 if the LHV of the fuel is known, or by stoichiometry.

Fig. 8-23 is for fired heaters with one row of 8 in. pipes, spaced two pipe nominal sizes (NPS). Correction factors for other designs, to be multiplied by G prior to graph readings, are shown in the figure.

Example 8-8 — Estimate the radiant tube area for a 10 MMBtu/hr regeneration gas heater. To avoid overheating the tubes, a radiant flux of 10,000 Btu/(hr • sq ft) is specified. The design calls for 4 in. Sch 80 tubes on an 8 ft tube circle. The fuel is 0.61 gravity gas with LHV of 1000 Btu/scf. Use 20% excess air.

Fuel gas and combustion air are supplied at 60°F. The heater NTE is 80%. The tubes are arranged in one row at 8" spacing.

Solution Steps

r = 970 lb flue gas/MMBtu (Fig. 8-24).

$$r \cdot LHV = 970 \frac{\text{lb flue gas}}{\text{MMBtu}} \cdot 1000 \frac{\text{Btu}}{\text{scf fuel gas}}$$

$$= \frac{970 \text{ lb flue gas}}{1000 \text{ scf fuel gas}}$$

$$\text{Weight of 1000 scf fuel gas} = \frac{(1000)(0.61)(29)}{379.5} = 46.6 \text{ lb}$$

Weight of combustion air = 970 – 46.6 = 923.4 lb

G = 923.4/46.6 = 19.8 lb air/lb fuel

Using Eq. 8-21

$$B = \frac{(4.5)(1)(3.14)(10{,}000)(19.8^2)}{(8)(0.88)} = 7.87 \cdot 10^6$$

Using Eq 8-22

$$R = 1 - \frac{\sqrt{(7.87 \cdot 10^6)^2 + (70.56 \cdot 10^6)(7.87 \cdot 10^6)} - 7.87 \cdot 10^6}{35.28 \cdot 10^6}$$

R = 0.519 (Note: Fig. 8-23 yields R = 0.525, so use the average of R = 0.522.)

$$Q_r = \frac{UHT \cdot R}{NTE} = \frac{(10)(0.522)}{0.8} = 6.525 \text{ MMBtu/hr}$$

$$\text{Heat transfer area} = \frac{Q_r}{I} = \frac{6.525 \cdot 10^6}{10{,}000} = 652.5 \text{ sq ft}$$

The surface area of 4 in. pipe is 1.178 sq ft/linear ft.

Total tube length = 652.5/1.178 = 553.9ft

There are 36 vertical tubes in a cylindrical heater with an 8 ft diameter tube circle when the tubes are 8 in. center to center.

Tube length = 553.9/36 = 15.4 ft

Convection Section

The purpose of the convection section is to transfer as much heat as possible from the combustion gases leaving the radiant section. As always there is the trade-off between capital cost, i.e., adding more tubes, and operating cost, i.e., improved thermal efficiency.

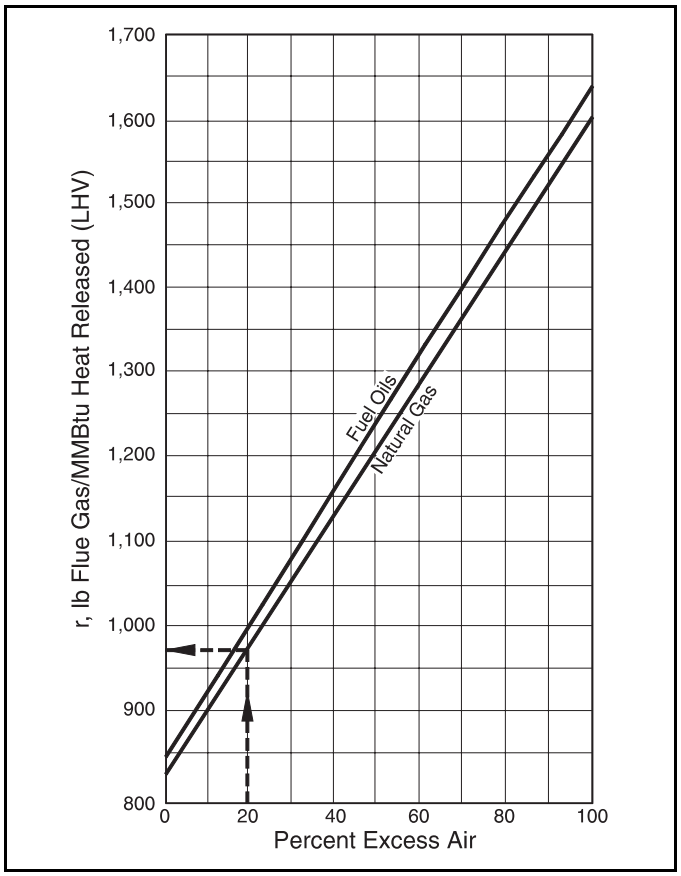
The construction is similar to that for the radiant section, a steel plate shell with internal castable or ceramic fiber insulation. The tubes are staggered, and the space between the sidewall and the tube is filled with “corbels” to prevent the flue gases from bypassing the end tubes.

The first two rows of the convection section are called shock tubes and they “see” the firebox flame. The first row receives the full radiant heat flux and also some convective heat transfer. It has the highest heat transfer flux in the heater and is always bare tubes. The second shock row receives about one-third of the radiant flux as well as convective heat transfer from the flue gas. It is also bare tubes. If long radius return bends are used, the third row will receive radiant heat and it too should be bare tubes.

Helical fins, sometimes serrated to increase turbulence, are used as soon as possible, i.e., when the fin tip temperature is not excessive, e.g., 1000°F for carbon steel, see Fig. 8-8. Typi-

FIG. 8-24

Flue Gas Rates[9]



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cally when natural gas is the fuel, the fins are 1 in. high, 0.06 in. thick and up to 72 fins per linear ft. For oil fired heaters where soot deposition is possible, the fins are 1 in. high, 0.105 in. thick and not more than 36 fins per ft. Often the first finned row has fewer, shorter, and thicker fins to reduce the fin tip temperature. Where ash and soot fouling are expected, a lane is left every four or five rows for soot blowers. These are tubes equipped with nozzles that direct steam against the tubes. Soot blowing is intermittent and is seldom used more than once every shift.

The fins compensate for the low flue gas heat transfer coefficient. Typically, the heat flux in the convection section is 2000–4000 Btu/(hr • sq ft) of finned surface or 12,000–24,000 Btu/ (hr • sq ft) on a bare tube basis.

Cast iron tube supports can be used below 800°F and 25% chrome – 12% nickel is good up to 2000°F. With high vanadium or sodium levels in the fuel oil, 50% chrome – 50% nickel must be used.

The distance between supports for horizontal tubes should be the lesser of 35 outside tube diameters or 20 ft. The distance between supports on vertical tubes should not exceed either 70 tube diameters or 40 ft. Usually the return bends are external to the tube sheets. This prevents flue gases from bypassing the tube fins.

Fig. 8-25 shows approximate external heat transfer coefficients for 3, 4, and 6 in. steel pipe arranged in staggered rows and surrounded by combustion gases.

Example 8-9 — Design the convection section for the 10 MMBtu/hr regeneration gas heater of Example 8-8. The heat loss is assumed to be 2% of the heat release. Use six 4 in. Sch 80 tubes on 8 in. center-to-center spacing with 8 ft effective length in each row. After two rows of bare shock tubes use finned pipe, 36 fins/ft, 1.25 in. high, 0.105 in. thick. Assume pipe wall temperatures of 200 to 470°F across the finned part of the convection section and average values of 480° and 500°F for the two shock rows.

$$= (6)\,(8)\left[\frac{8}{12} - \frac{4.5}{12}\right] = 14.0 \text{ sq ft}$$

Solution Steps

Fig. 8-26 summarizes the design of both the radiant and convection sections. A trial and error solution for assumed temperatures is required. Details follow for the converged solution.

Q_{total} = duty/NTE = 10 (10^6)/0.80 = 12.5 MMBtu/hr

r = 970 lb flue gas/MMBtu (Fig. 8-24)

Flue gases flow rate = 12.5(970) = 12,125 lb/hr

Assume that the setting loss of 2% or 0.25 MMBtu/hr occurs in the radiant section.

The heat content (LHV) rate of the combustion gases leaving radiant section:

$Q_{radiant\ exit}$ = 12.5 – 6.525 – 0.25 = 5.725 MMBtu/hr

The enthalpy (without latent heat) of the exit gas from radiant section:

H = 5.725 (10^6)/12,125 = 472.1 Btu/lb

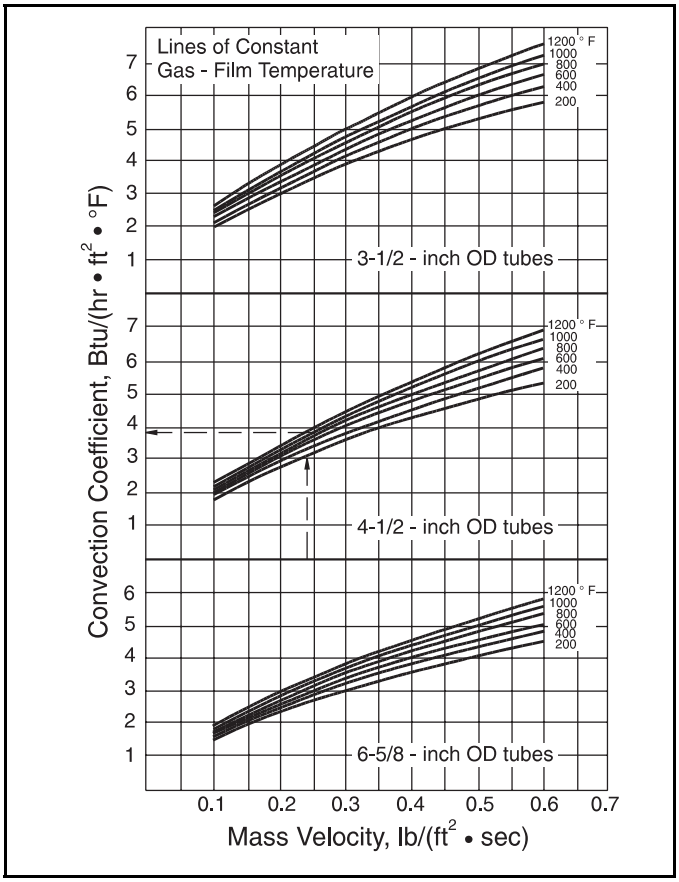
T_g = 1730°F (Fig. 8-20, Flue Gas – LHV)

Convection Section:

Area for gas flow = (no. of tubes) (L) (spacing – D)

FIG. 8-25

Flue Gas Convection-Coefficients for Flow Across Staggered Banks of Bare Tubes[9]



G_g = 12,125/(14.0) (3600) = 0.241 lb/(sec • sqft)

First shock row. Assume the average gas temperature is 1625°F and tube wall temperature is 500°F.

$$T_g \text{ mean} = \frac{500 + 1625}{2} = 1062°F$$

h_o = 3.8 Btu/(hr • sq ft • °F) (Fig. 8-25)

A = 1.178 sq ft per linear ft (Example 8-8)

A_{tubes} = 48 (1.178) = 56.54 sq ft

$Q_c = h_o A \Delta T$ = (3.8) (56.54) (1625 – 500)

= 0.242 MMBtu/hr

I = Q/A = 10,000 Btu/(hr • sq ft) (Example 8-8)

Q_r = (Q/A) (A) = 10,000 (56.54) = 0.565 MMBtu/hr

$Q_c + Q_r$ = (0.242 + 0.565) 10^6 = 0.807 MMBtu/hr

$Q_{exit\ gases}$ = (5.725 – 0.807) 10^6 = 4.918 MMBtu/hr

$H_{exit\ gases}$ = 4.918 (10^6)/12,125 = 405.6 Btu/lb

T_g(exit) = 1520°F (Fig. 8-20, Flue Gas – LHV)

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Second shock row is analogous except that the radiant heat flux is one third of that for the first row, i.e., 3333 Btu/(hr • sq ft)

$Q_r = (3333)(56.54) = 0.188$ MMBtu/hr

With $h_o = 3.7$ Btu/(hr • sq ft • °F)

$Q_c = (3.7)(56.54)(1465 - 480) = 0.206$ MMBtu/hr

$Q_c + Q_r = 0.206 + 0.188 = 0.394$ MMBtu/hr

$Q_{exit\ gases} = (4.918 - 0.394)\ 10^6 = 4.524$ MMBtu/hr

$H_{exit\ gases} = 4.524\ (10^6)/12{,}125 = 373.1$ Btu/lb

T_g(exit) = 1410°F (Fig. 8-20, Flue Gas – LHV)

Finned rows. The combustion gas mass velocity increases because of the decreased cross sectional area of finned pipe. From Eq 8-13.

$$A_{cs} = \frac{4.5}{12} + \frac{36\ (0.105)\ (7.0 - 4.5)}{144}$$

$$= 0.441 \text{ sq ft/linear ft}$$

$$G_g = \frac{12{,}125}{[48\ (0.667 - 0.441)(3600)]}$$

$$= 0.310 \text{ lb/(sec • sq ft)}$$

$Q_f = 10.0 - 6.525 - 0.807 - 0.394 = 2.274$ MMBtu/hr

$Q_{exit\ gases} = 12.50 - 10.00 - 0.25 = 2.25$ MMBtu/hr

H_g(exit) $= 2.25\ (10^6)/12{,}125 = 185.6$ Btu/lb

T_g(exit) = 730°F (Fig. 8-20, Flue Gas – LHV)

Assuming that HHV is 10% more than LHV, the gross heater efficiency is 80% / 1.1 = 72.7%. Note that this agrees closely with Fig. 8-19 for 20% excess air and 730°F.

Pipe and gas temperatures are:

$T_{p1} = 200$°F; $T_{p2} = 470$°F; $T_{p\ av} = 335$°F

$T_{g1} = 1410$°F; $T_{g2} = 730$°F; $T_{g\ av} = 1070$°F

$\Delta T_{LM} = [(1410 - 470) - (730 - 200)]/\ ln\ (940/530) = 715$°F

$T_{film} = (335 + 1070)/2 = 703$°F

$h_o = 4.0$ Btu/(hr • sq ft • °F)(Fig. 8-25)

$Q_f = h_o\ A_o\ \Delta T_{LM}$ (Eq 8-10)

$A_o = 2.274\ (10^6)/(4.0)\ (715) = 795$ sq ft

This is the theoretically required surface area. From Example 8-5, the fin efficiency is 87% and the external surface area of the finned pipe is 12.10 sq ft/linear ft.

$L_{pipe} = 795/[(0.87)\ (12.10)] = 75.5$ ft

With 48 linear ft per row this is 1.6 rows. Use 2 rows.

Stack Draft

The stack draft must overcome the gas friction loss in the convection section, burner, and stack. The stack diameter is often sized for 15 to 20 ft/sec stack gas velocity. The stack is normally bare carbon steel but must be lined if the flue gas temperature exceeds 750°F or if the fuel has high sulfur content. All wall temperatures should be above the dew point of the flue gas.

Typical draft profiles for direct fired natural draft heaters are shown in Fig. 8-27. There are two ways to control the flow of combustion air: stack dampers or combustion air registers. There should be a slight vacuum in a natural draft heater to prevent leakage of the flue gases. There is usually an increase in pressure of 0.01 in. H_2O per ft of firebox height and several inches pressure drop across the convection section.

Example 8-10 — Find the available draft in a 2 ft ID by 20 ft long stack attached to the top of the convection section for the 10 MMBtu/hr regeneration gas heater of Examples 8-8 and 8-9. Assume dry air at 60°F and 14.7 psia.

Solution Steps

Assume $\mu_g = \mu_a$ from Fig. 23-26.

$$Re = \frac{0.5263 \bullet M}{D \bullet \mu} = \frac{(0.5263)(12{,}125)}{(2)(0.024)} = 132{,}946$$

For an absolute roughness of 0.002 feet, the relative roughness is 0.001 then,

$f = 0.005$ (Fig. 17-2)

From Eq 8-18

$$Hav = 0.192 \bullet H_s\ (\rho_g - \rho_a) + 0.0029 \bullet V^2 \bullet \rho_g \left(4 \bullet f_f \bullet \frac{H_s}{D_s} + 1\right)$$

Calculate ρ_g and ρ_a from the ideal gas law.

$$V = \frac{M}{3600 \bullet A \bullet \rho} = \frac{12{,}125}{3600 \bullet \pi \bullet (0.0335)} = 32.0 \text{ ft/sec}$$

$$Hav = 0.192\ (20)\ (0.0335 - 0.0766) + 0.0029\ (32)^2\ 0.0335$$

$$[4(0.005)\ (20/2) + 1]$$

$$= -0.046 \text{ in. } H_2O$$

The flue gas ideal relative density changes with the combustion air humidity, the fuel composition and with the steam addition to the heater. Burning a typical RD=0.6 dry natural gas, with 20% excess air at 760 mm Hg, without addition of steam, the hotter the combustion air, the higher the influence of the humidity on the flue gas relative density, due to the higher water vapor fraction, as can be seen in Fig. 8-28. For other conditions use the equation shown on the graph.

Insulation

Insulation protects the heater shell from the hot combustion gases and usually reduces the heat losses to less than 2% of the heat release. Three common types of insulation are:

LHV Castable Refractory — This is a mixture of lumnite (cement), haydite (aggregate), and vermiculite (insulation) in 1:2:6 proportions. This low cost, concrete-type insulation has a density of 55 lb/cu ft, a low coefficient of expansion, and negligible shrinkage. It is held to the vertical walls with bullhorns — V-shaped steel wire anchors welded to the outer casing. If a high sulfur (more than 1%) fuel is burned, to prevent the sulfur trioxide in the flue gas from attacking the iron in the lumnite, use a low-iron (1%) cement. LHV castable withstands high (150-200 ft/sec) gas face velocities. Rain or snow entering through the stack will not hurt the concrete. Before start-up, cure and dry out the refractory to avoid "spalling" or flaking caused by unequal thermal expansion.

Ceramic Fiber — A sandwich construction of this material in two densities is sometimes used. Because the ceramic fiber is porous, a protective coating is normally applied first to

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FIG. 8-26

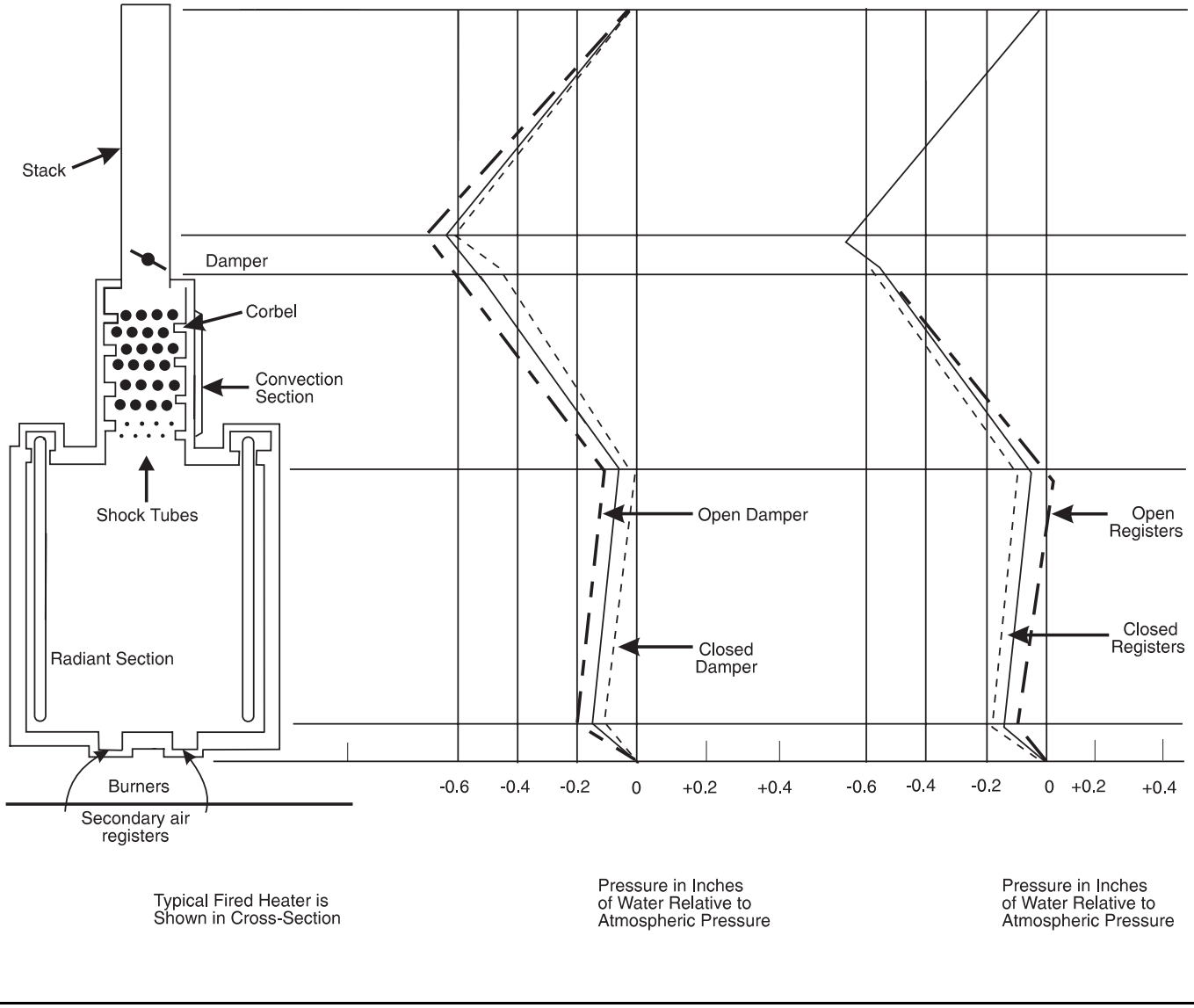
10 MMBtu/hr Regenertion Gas Heater

Section	Gas Heat Content Rate MMBtu/hr (LHV)		Heat Transfer MMBtu/hr	Exit Gas Enthalpy Btu/lb	Exit Gas Temperature °F
	In	Out			
Radiant	12.500	5.725*	6.525**	472.1	1730
Shock Row 1	5.725	4.918	0.807	405.6	1520
Shock Row 2	4.918	4.524	0.394	373.1	1410
Finned	4.524	2.250	2.274	185.6	730

* Heat losses are 0.25 MMBtu/hr
** From Example 8-8

FIG. 8-27

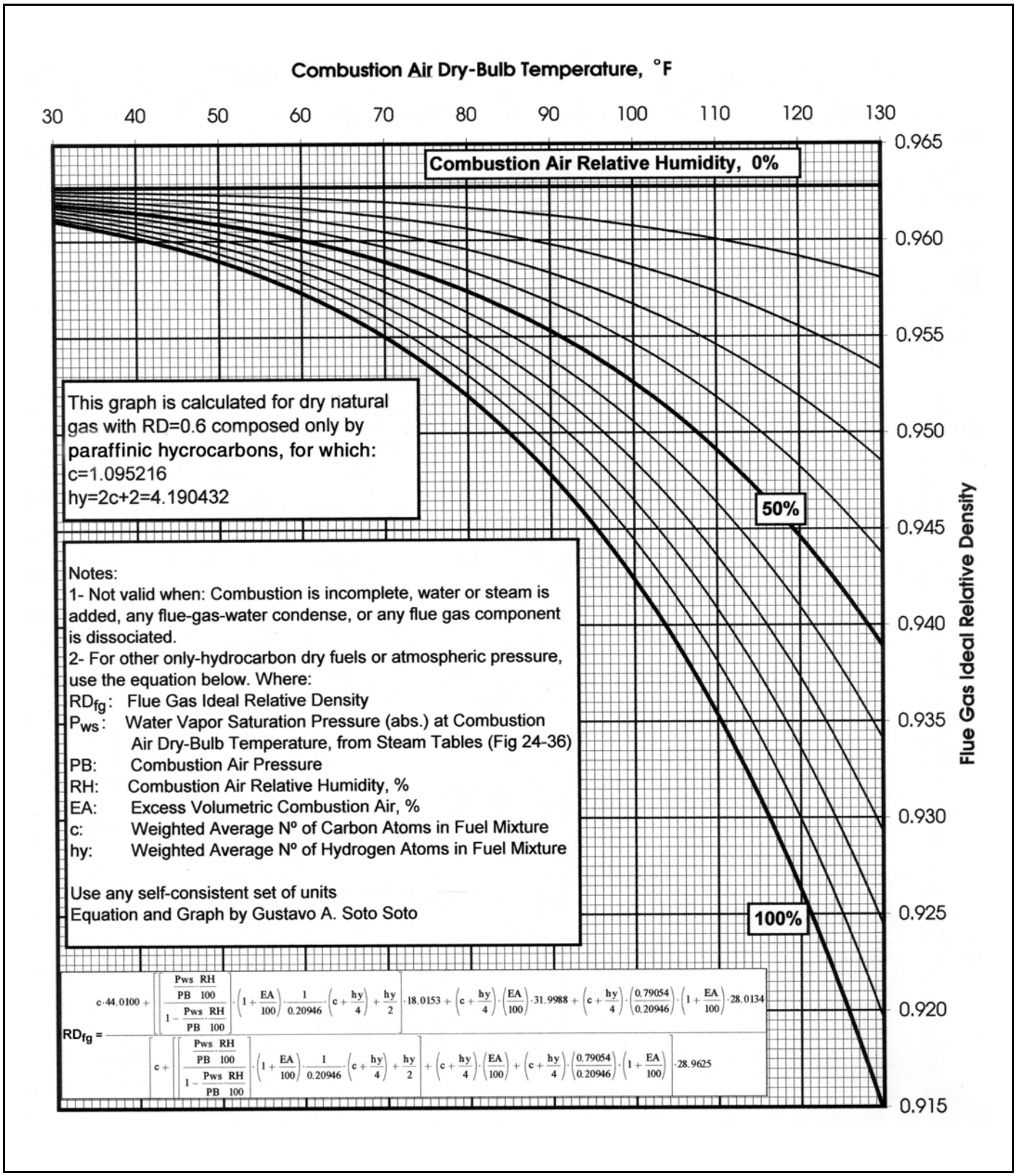
Natural Draft Profiles



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FIG. 8-28

Flue Gas Ideal Relative Density
Fuel 0.6 RD Dry Natural Gas, Burning with 20% Excess Air.
Combustion Air at 760 mm Hg and Indicated Temperature and Relative Humidity



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the steel casing, for protection against sulfur oxides in the flue gas. A 2 or 3 inch layer of 4 lb/cu ft ceramic fiber is impaled over thin stainless steel studs welded to the casing. Over this is placed a one inch layer of 8 lb/cu ft material. Stainless steel washers twisted onto the studs hold the ceramic fiber in place. This light weight material is much lighter than LHV and is suitable for convection sections. Advantages include ease of application in freezing weather, when water in refractory is a problem; and no need to dry out or cure the ceramic fiber when it is first applied. For cyclic operations, less heat is stored in the fiber which reduces warm-up time. Rain or snow entering the heater during shutdown may drench the fiber, causing it to tear away from the studs on a vertical wall. It cannot withstand gas face velocities above 50 ft/sec. If soot blowing or steam lancing of finned tubes is required, LHV castable refractory must be used or the fiber must be covered with an 1/8 in. stainless steel erosion shield. The erosion shield is also needed for gas turbine exhaust heat recovery units.

Insulating Firebricks (Ifb) — These are used for bridgewalls and floors. Ifb are quite dense, ca. 150 lb/cu ft, are shipped loose, and field installed. They must be dried out slowly. After proper installation, firebricks are sturdy and resist deterioration from weather and high gas velocities.

External Insulation — In sulfur plants and in cold climates, external insulation is frequently used to maintain a minimum stack temperature to prevent condensation within the stack.

Other Design Considerations

Film temperature — Temperatures above 500°F will cause many hydrocarbons to decompose and coke layers to deposit in the tube. This increases the film and metal temperatures and leads to tube failures. Actual decomposition temperature is highly dependent upon the fluid characteristics.

Snuffing — If a tube ruptures, fire could break out in the firebox. Connections are needed to admit CO_2 or a steam or water spray to snuff out the fire. The velocity of this vapor should be kept below 80 ft/sec to avoid erosion of the refractory.

Purging — If the pilot flame or electric spark fails, the firebox may contain an explosive mixture of gases. Before attempting to relight the heater, this mixture must be purged.

Sampling — To control the excess air, samples of flue gas are needed from various points of the heater. The amount of excess air at the burner may differ from that in the stack due to leakage of ambient air through terminal holes, box gaskets, cracks in the casing, etc.

Flue gas temperature — To determine the thermal efficiency of the heater, the stack temperature must be monitored.

Process coil thermowells — The firing rate is controlled to maintain the correct process stream outlet temperature.

Draft gauges — These are needed to set the stack damper or burner air registers.

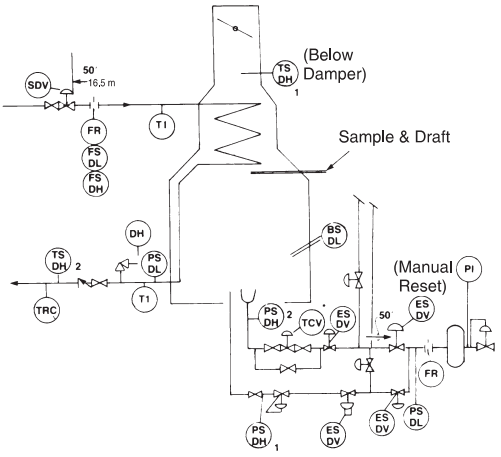
Soot blowers — In liquid fuel systems these can be used intermittently to reduce fouling of the finned tubes.

Controls

Fig. 8-29 shows an example control system for a fired heater. Fig. 8-30 lists the shutdown/alarms as shown with some comments about the proper installation and use of these particular

FIG. 8-29

Example Direct Fired Reboiler



devices. The control system as depicted by these figures should not be considered complete but only representative of the conditions to be carefully considered in designing a control system for fired equipment.

The following are some indications of possible trouble:

- The burner flame is not symmetrical, pulsates or breathes, is unusually long or lazy, lifts off the burner, etc.
- The burner is not aligned and/or the flame is too close to the tubes.
- Lack of negative pressure at the top of the firebox.
- The stack gas is smoky.
- The gas in the firebox appears hazy.
- There are unequal temperatures, more than 10°F difference, on the process pass outlets.
- The stack temperature increases steadily with no change in the process heat duty.
- The fuel gas control valve is wide open.
- The fuel gas composition or pressure varies widely.
- The tubes in the heater are not straight.

Options to Improve the Thermal Efficiency

Option I. Add Convection Surface

Effects:

1. Stack temperature is reduced.
2. Furnace efficiency is increased.
3. Heat release is decreased.
4. Flue gas pressure drop in the convection section is increased.
5. Draft is decreased.
6. Tube side pressure drop is increased.
7. NO_x is reduced.

Things to consider:

1. Increasing stack height.

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FIG. 8-30

Heater Alarm/Shutdown Description

Note: Alarms and shutdowns as shown are not to be considered as meeting any minimum safety requirement but are shown as representative of types used for control systems.

Basic Criterion – The failure of any one device will not allow the heater to be damaged.

Schematic Label	Alarm/Shutdown Description	Regeneration Gas Heater	Hot Oil Heater and Direct Fired Reboiler
TSH-1	High Stack Temperature	See Note 1	See Note 1
TSH-2	High Outlet Temperature	See Note 1	
FSL	Low Mass Flow Through Tubes	See Notes 2 & 4	See Notes 3 & 4
BSL	Flame Failure Detection	See Note 5	See Notes 5 & 6
PSL	Low Fuel Pressure		See Note 6
PSH-1	High Fuel Pressure	See Note 7	See Note 7
PSH-2	High Cabin Pressure	See Note 8	Not applicable, if natural draft

Notes:

1. A direct immersion jacketed thermocouple is preferred because the response is ten times faster than a grounded thermocouple in a well. A filled bulb system is a poor third choice. The high stack gas temperature shutdown should be set approximately 200°F above normal operation.
2. An orifice plate signal should be backed up by a low pressure shutdown to ensure adequate process stream flow under falling pressure conditions.
3. The measurement should be on the heater inlet to avoid errors due to two-phase flow.
4. Differential pressure switches mounted directly across an orifice plate are not satisfactory due to switch hysteresis. An analog differential pressure transmitter with a pressure switch on the output is recommended. The analog signal should be brought to the shutdown panel so that the flow level can be readily compared with the shutdown point.
5. The flame scanner should be aimed at the pilot so that a flameout signal will be generated if the pilot is not large enough to ignite the main burner.
6. If the heater design precludes flame scanners, a low fuel gas pressure shutdown should be installed to prevent unintentional flameout. This shutdown should detect gas pressure at the burner.
7. Either burner pressure or fuel control valve diaphragm pressure may be used. This shutdown should be used whenever large load changes are expected. It prevents the heater from overfiring when the temperature controller drives the fuel valve wide open to increase heat output with insufficient air.
8. This shutdown should block in all lines to the heater because, when activated, a tube has probably ruptured. Gas is probably burning vigorously outside the heater.

2. More weight from added convection tubes.
3. Check structure and foundation to see if added weight can be supported.
4. If not, design adjacent structure to house convection tubes and support stack.
5. Consider increased pumping cost for process stream.
6. If fuel is to be changed, some existing convection tubes may have to be removed to accommodate soot blowers.

Option II. Add Economizer for Waste Heat Recovery

Waste heat options:

Steam generation
Steam superheating
Boiler feedwater heater

Effects:

1. Stack temperature is reduced.
2. Furnace efficiency is increased.
3. Flue gas pressure drop in the convection section is increased.
4. Draft is decreased.
5. No change in process stream operation.
6. No change in NO_x.

Things to consider:

1. Study increased load on structure and foundation as in Option I.
2. Will added boiler capacity lower efficiency of existing boilers?
3. Check possibility of a temperature cross.

Option III. Install Air Preheat System

Effects:

1. Stack temperature is reduced.
2. Natural draft is decreased.
3. Furnace efficiency is increased.
4. Firebox temperature is increased.
5. Heat flux rates are increased.
6. NO_x increases unless burners are changed.

Things to consider:

1. Induced draft and forced draft blowers must be installed.
2. Burners must be replaced.
3. Check if tube supports and refractory will withstand higher temperatures.
4. Plot space near furnace must be available.
5. System is self-contained.
6. Considerably more instrumentation must be installed.

Convection Heaters

Heaters in which all the heat transfer is convective (there is no radiant section) are unique modifications of direct fired heaters. Because all aspects of the operation, e.g., fuel combustion, combustion gas temperature, tube wall temperature, etc., are controlled, these heaters are ideally suited for offshore platforms and other applications that demand a high degree of safety.

Fig. 8-31 is a sketch of a convection heater, with recycle regulating the heat flux. The cylindrical steel shell is internally

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insulated with light castable or ceramic fiber, and the combustion and heat exchange sections are separated by an insulated wall. The fuel gas and all of the combustion air are fed to the short-flame, pre-mix burners. The combustion gases mix rapidly with recycled stack gases to produce the inlet gas to the heat exchange section. This gas at 1200-1500°F flows under the dividing wall and then upward across the finned tubes of the process coil. Part of the cooled gas at 350-500°F leaves as stack gas and the remainder is recycled. The flow in the process coil is downward and counter-current to the hot gases.

Recycling some of the stack gas controls the inlet gas temperature to the heat exchange section, and in turn the maximum film temperature of the process stream. The combined volume of the combustion and recycled gases is much greater than that in most direct fired heaters. This results in higher gas velocities, higher external heat transfer coefficients, and a smaller process coil surface area. The thermal efficiency is high because pre-mix burners operate satisfactorily at 10% excess air and the stack gas temperature can be as low as 100°F above the inlet process stream temperature. The price paid for this is the electricity to run the recirculation fan; this is usually more than offset by the fuel saving.

Convection heater applications include:

- Heating regeneration gas in adsorption processes
- Vaporizing and superheating LNG and LPG
- Preheating or post-heating gas, in pressure reduction
- Heating heat transfer fluids
- Heating sensitive fluids or gases where close bulk-to-film temperatures are critical
- Combined waste-heat recovery with supplementary burners
- Steam Generation

FIRETUBE HEATERS

Firetube heaters range in duty from 60 MBtu/hr glycol reboilers to 12 MMBtu/hr oil or gas pipeline heaters. The design, controls, and operation of firetube heaters vary widely from those used in simple, unattended "wellhead" equipment to those used in complex, well-instrumented, gas plants.

Water Bath Heaters

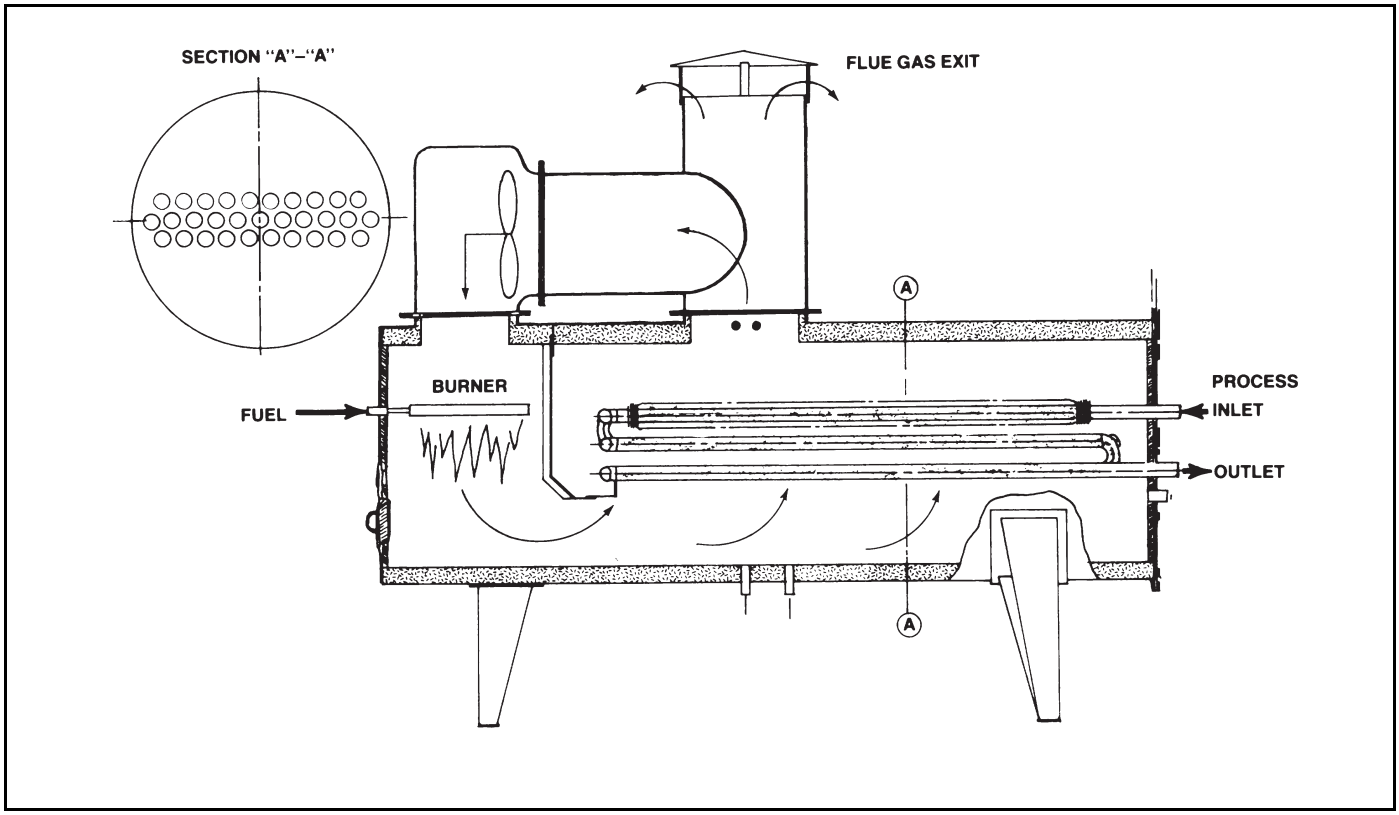
Fig. 8-32 is a sketch of an indirect fired water bath heater. This design is typical of all indirect fired vessels. The firetube is in the bath in the lower half of the vessel and the process coil is in the upper half of the vessel. The heat transfer medium, in this case water, fills the vessel. A fill hatch, drain, wells for thermostats, and a coil to preheat the fuel are standard. Sizes range from 2 ft OD by 5 ft long to 12 ft OD by 30 ft long. Only one firetube is needed below 5 MMBtu/hr.

Some small heaters are uninsulated. More frequently 1 to 2 in. of weatherproof ceramic fiber or light castable insulation is placed around the cylindrical shell. The ends are left bare so as not to impede access to the coil, burner, or stack. As shown in Example 8-7, insulation can reduce heat losses up to 75%.

A long lazy yellow flame increases the fire tube life and increases the radiant flame area.

FIG. 8-31

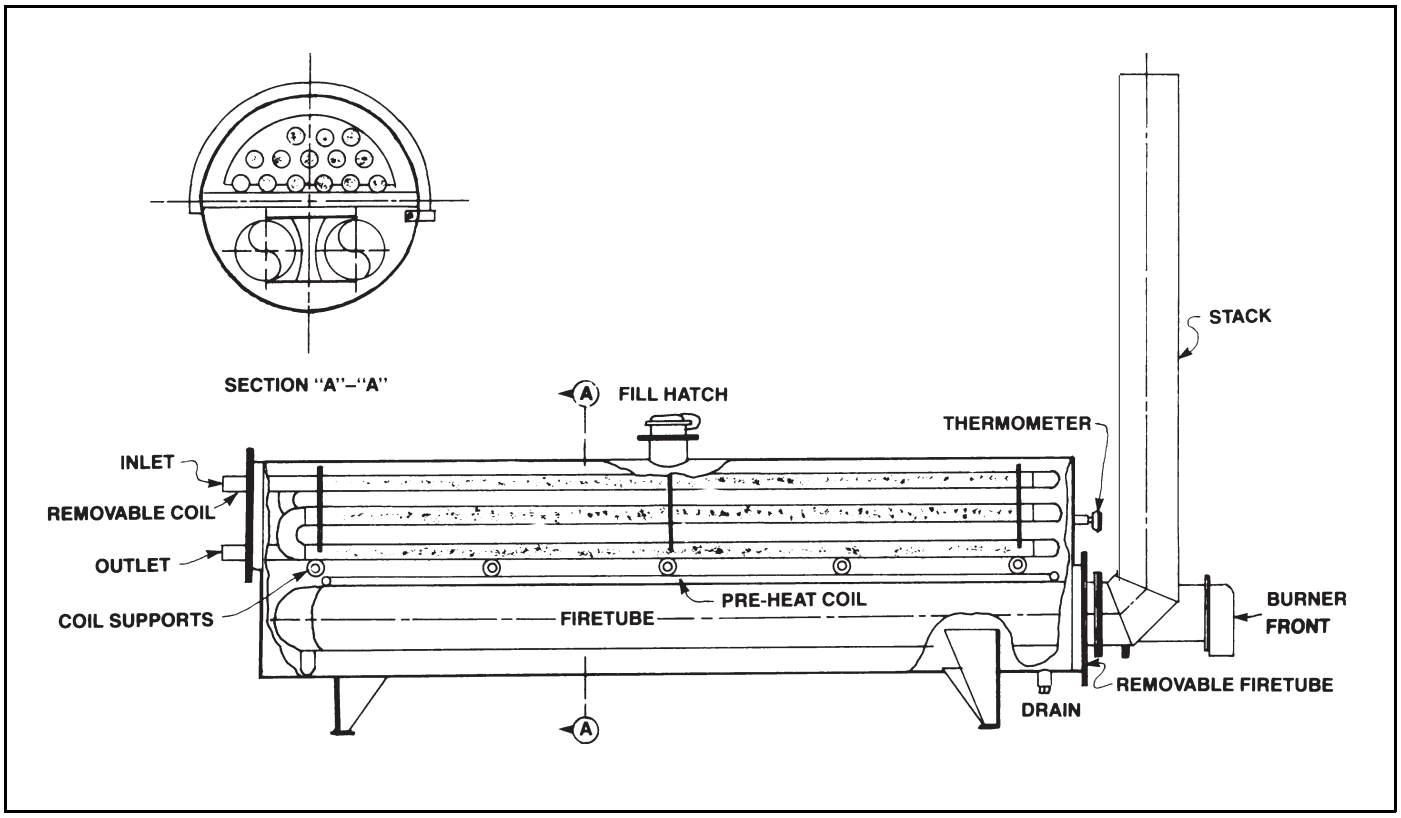
Convection Heater



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FIG. 8-32

Indirect Fired Water Bath Heater



Almost every coil bundle requires a unique design to meet the requirements of heat duty, working pressure, corrosion allowance, sour gas service, NACE MR-01-75, and governing codes, usually ASME Section VII or ANSI B31.3. Use of a P-H diagram will simplify calculations when both sensible and latent heat changes occur in the process stream.

Example 8-11 — What heat duty is required to vaporize 50 gpm of liquid propane at 60°F and 235 psig and superheat the vapor by 20°F?

Solution Steps

Refer to Section 24 "Thermodynamics" and the P-H diagram for propane. At 60°F and 249.7 psia the enthalpy of propane is 287 Btu/lb. The exit enthalpy at 142°F (20°F above the dew point) and 249.7 psia is 462 Btu/lb. The liquid density is 31.7 lb/ cu ft and this corresponds to 4.24 lb/gal.

$$q_l = 50 \text{ gpm}$$

$$M = (50)(4.24)(60) = 12720 \text{ lb/hr}$$

$$UHT = M(H_2 - H_1)$$

$$= (12720)(462 - 287) = 2.23 \text{ MMBtu/hr}$$

Usually long radius return bends connect the passes in the serpentine coil; however, short radius return bends sometimes "fit" the coil bundle into the shell. The heat duty and pressure drop determine the pipe diameter and the number of parallel flow paths and passes in the process coil. Often the internal process stream heat transfer coefficient is much larger than the external water bath coefficient, e.g., Example 8-4. Two parallel flow paths of four passes may be an alternative heat transfer design to one flow path of eight passes. The effect on the process stream pressure drop is significant. Because the pressure drop for turbulent flow is proportional to the 1.83 power of the velocity and the pipe length per pass has been halved, the two flow path pressure drop approaches one seventh of that for one pass.

In many wellhead applications the coil contains a choke that throttles the gas from well pressure to processing or pipeline pressure. This divides the coil into preheat and postheat passes. The rapid expansion is isenthalpic and, if the gas temperature falls too low, hydrates form.

Example 8-12 — Find the optimum distribution between preheat and postheat duty for expanding methane from 3000 psia and 80°F to 1000 psia and 100°F.

Solution Steps

Refer to Fig. 8-33 which is a P-H diagram for CH_4 on which the line AGB for hydrate formation is superimposed. (This is a combination of Figures 20-19 and 24-23.) Consider the following three alternative paths.

Path CDE expands the gas immediately and then heats it. This results in the smallest coil area because the largest log mean temperature difference between the water bath and the methane is available. But the expansion crosses the hydrate line and the gas will freeze.

Path CHE supplies all the heat needed and then expands the gas. This is feasible, but not desirable because the lowest temperature difference requires the largest coil area.

Path CFGE first heats the methane so that the expansion just touches the hydrate line, expands it, and heats to the exit

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temperature. This is the minimum coil area that corresponds to an operable path. So:

Preheat duty: (476 – 448) = 28 Btu/lb CH_4

Postheat duty: (510 – 476) = 34 Btu/lb

In practice, some penetration of the hydrate line is possible; and the balance between "preheat" and "postheat" passes is such that the lowest temperature is 10 to 15°F below the hydrate line.

Freezing of the water bath is a potential problem. If the heater is insulated, a continuous pilot suffices. Several different antifreeze additives have been tried and all have shortcomings:

- Methyl alcohol is volatile and has to be replenished. It is also a fire hazard.
- Calcium chloride and rock salt in concentrations that are effective are very corrosive.
- Glycols are generally accepted as the safest and most trouble-free additive. The decomposition products are acidic; it is recommended that corrosion and rust inhibitors be used concurrently.

Glycols reduce the heat transfer coefficient of the bath significantly. For example, a 50% by weight solution of ethylene glycol reduces the firetube flux by 20% and the external bath heat transfer coefficient for the process coil by 40%. (See Fig. 8-34)

Fig. 8-34 compares the bath properties of fire tube heaters. The advantage of using water as the heat transfer medium is apparent. The relatively low bath temperature results in the lowest stack temperature and the highest firetube efficiency. Note that the firetube efficiency does not account for heat losses. For large well insulated heaters the overall process NTE may exceed 80%. But for small uninsulated heaters with intermittent operation the process NTE may be as low as 60%.

Water bath heater applications include:

- Heating high pressure gas and/or oil in oilfield production
- Heating natural gas at city gate stations from main gas pipelines
- Heating light hydrocarbon liquids to superheated state for alternate or standby fuel sources

Low Pressure Steam Heaters

When a process stream temperature of 160 to 220°F is needed, a 15 psig steam heater can be used to reduce the required size of the tube bundle. Construction, shown in Fig. 8-35, is under ASME Section IV code. Steam outlet and condensate return connections are standard so that the steam may be used in external exchangers if desired. The condensing steam has an external process coil heat transfer coefficient of 800-1200 Btu/(hr • sq ft • °F). It is important to vent all air at start-up. With insulation and controls, the NTE can be as much as 80%, which is close to the efficiency for a firetube.

Low pressure steam heater applications include:

- Low pressure steam generators (without process coil)

FIG. 8-33

Methane Pressure-Enthalpy Diagram

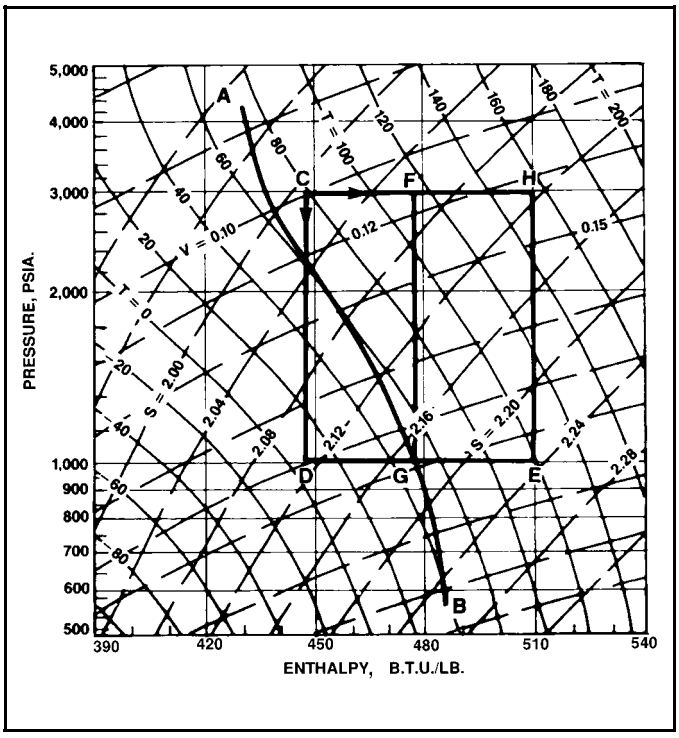


FIG. 8-34

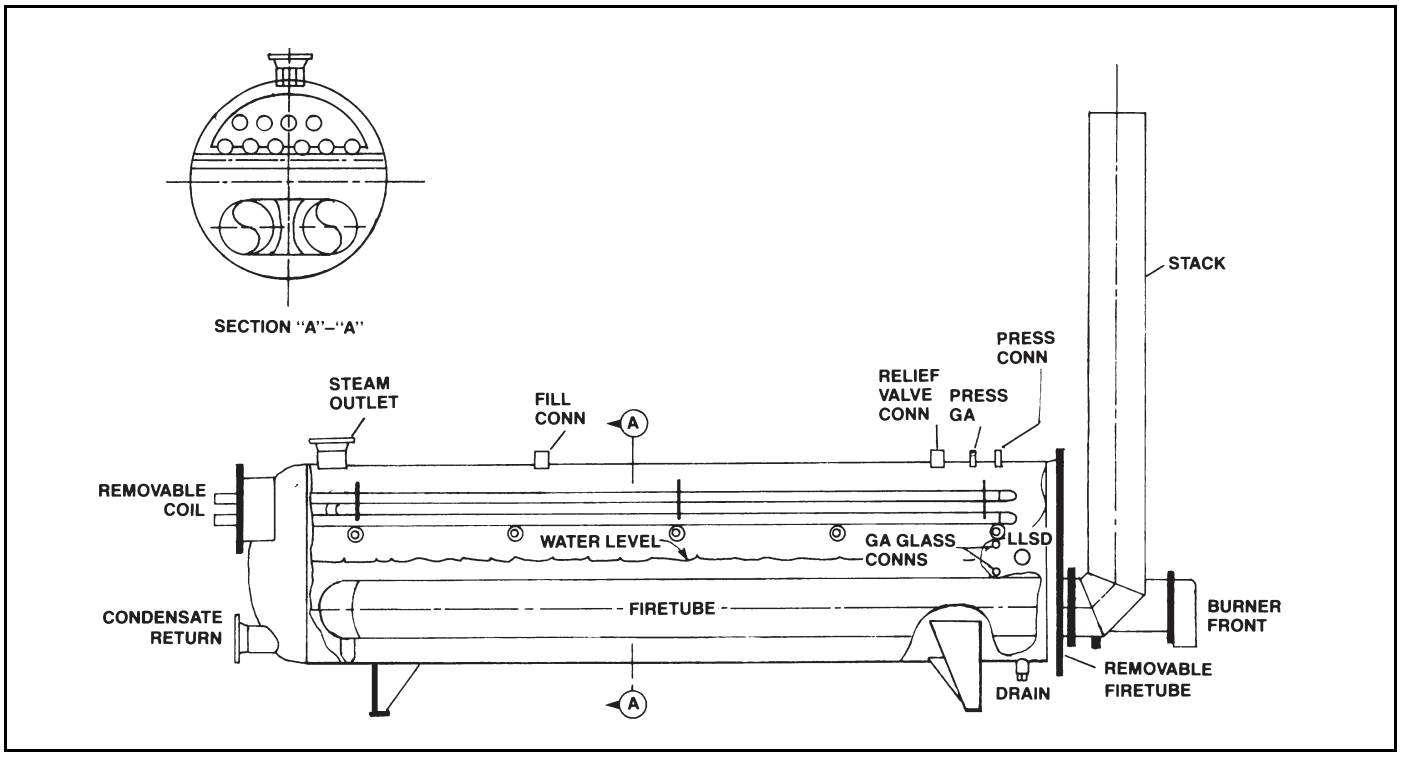
Bath Properties of Fire Tube Heaters

Heater	Bath Temp. °F	Outside Coil Bundle h_o Btu/(hr • sq ft • °F)	Firetube Flux Q/A Btu/(hr • sq ft)	Stack Temp. °F	Firetube Efficiency NTE %
Water Bath	180-195	160	10,000 - 13,000	750 - 900	76-82
50 % Ethylene Glycol	195-205	115	8,000 - 10,000	800 - 900	76-80
Low Pressure Steam	245-250	1000	15,000 - 18,000	800 - 900	76-80
Hot Oil	300-550	40	6,000 - 8,000	900 - 1100	71-76
Molten Salt	400-800	200	15,000 - 18,000	1000 - 1200	68-74
TEG Reboiler	350-400	–	6,000 - 8,000	800	75-80
Amine Reboiler	245-270	–	6,500 - 10,000	900	75-80

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FIG. 8-35

15 psig Steam Bath Heater



- Maintaining flow temperatures for heavy hydrocarbons in storage

Hot Oil Heaters

These heaters furnish a heating bath to 600°F or higher, which is hot enough for dry desiccant or hydrocarbon recovery regeneration gas. Another less severe application is heavier hydrocarbon vaporization prior to injection into a gas pipeline to raise the heating value.

Manufactured heat transfer oils are "blended" for about a 200°F operating range. For example, Fig. 8-36 gives typical heat transfer properties for a 300° to 550°F polyphenyl ether.

The advantages of hot oils are:

- Low vapor pressure at ambient temperature
- Always liquid and easy to handle
- Blended for a specific temperature range
- Higher specific heat than normally occurring hydrocarbons

The disadvantages include:

- Escaping vapors are environmentally undesirable
- Low heat transfer properties, see Fig. 8-34. (The firetube flux is half that of a water bath heater and the external process coil heat transfer coefficient is about one quarter.)
- Usually an ANSI Class 300 flange design is required
- When overheated, the oils will oxidize and coke on the firetube. Also, they can be ignited
- Ethers are expensive
- The ethers are hygroscopic and must be kept dry

Molten Salt Heaters

Molten salt baths operate at 400° to 800°F. They are mixtures of sodium nitrate and the nitrites of sodium and potassium. The advantages include:

- Good heat transfer properties, see Fig. 8-34. The firetube flux is as high as that for a low pressure steam heater. Note the process coil heat transfer coefficient of 200 Btu/(hr • sq ft • °F) is partially due to the small diameter tubes that usually comprise the coil bundle.
- Thermally stable to 1000°F.
- The salt will not ignite.

The disadvantages are:

- More difficult to handle, large lumps must be broken up.
- The salt is hygroscopic. If wet, it decrepitates on melting and is hazardous.

Because of the higher operating temperature there are no flange joints on the shell except for the salt loading hatch, see Fig. 8-37. On large heaters the expansion of the firetube may be sufficient to warrant an expansion joint for the stack. The shell is insulated to protect personnel.

One of the requirements for heating regeneration gas is a very low pressure drop across the coil bundle. To realize this and still obtain good heat transfer, the coil bundle consists of many, small diameter, U-tubes in parallel. These are welded into a tube sheet that is attached to a channel header. To reduce thermal stresses caused by the cold inlet gas, the inlet half of the tube sheet is insulated.

Molten salt heater applications include:

- Regeneration gas service

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- Vaporization of hydrocarbon liquids in stabilization service

Direct Fired Reboilers

Gas treating and dehydration frequently employ direct fired reboilers. More detailed process descriptions are given in Sections 20 and 21. Fig. 8-38 is a sketch of a typical direct fired reboiler. The rich fluid containing the sour gas or water is boiled in the reboiler to remove the sour gas or the water. The lean fluid then is used to treat or dehydrate the process gas stream again. Surge tanks for the lean fluid may be integral with the reboiler as shown, or may be mounted as a separate vessel beneath the reboiler.

Firetubes, Burners, Stacks

Firetubes typically range from 6 to 30 in. ID and from 5 to 30 ft long. Normally the burner flame extends halfway down the first leg. A mitered joint return bend is used to reduce the resistance to flow of the combustion gases.

Example 8-13 — What is the firetube flux when the combustion gases are at 2800°F and the firetube wall is at 300°F. Assume that the fuel is natural gas and the heat release is 4 MMBtu/hr with 20% excess air in a 2 ft ID pipe.

Solution Steps

Refer to example 8-6. The radiant heat flux is 16,990 Btu/(hr • sq ft). The convective heat flux must be added.

r = 970 lb flue gas/MMBtu heat release (Fig. 8-24)

Combustion gas flow is 4 (970) = 3880 lb/hr

From Eq 8-7, 8-8b and 8-9a and Fig. 8-5

$$Nu = 0.023 \cdot Re^{0.8} \cdot Pr^{0.33} \cdot (\mu_b/\mu_w)^{0.14}$$

k = 0.038 Btu/[hr • sq ft • °F/ft] at average T = 1550°F

Cp = 0.248 Btu/(lb • °F)

μ_b = 0.045 cp

μ_w = 0.023 cp

$$\frac{h \cdot 2.0}{0.038} = 0.023\left[\frac{0.5263\ (3880)}{2\ (0.045)}\right]^{0.8}\left[\frac{2.419 \cdot 0.248 \cdot 0.045}{0.038}\right]^{0.33}\left[\frac{0.045}{0.023}\right]^{0.14}$$

$$h = \frac{0.038\ (0.023)}{2.0} \cdot (3052.5)(0.8933)(1.0985)$$

h = 1.31 Btu/(hr • sq ft • °F)

Q/A_c = 1.31(2800 – 300) = 3,275 Btu/ (hr • sq ft)

Total heat flux = 16,990 + 3,275 = 20,265 Btu/(hr•sq ft)

This is a maximum firetube flux and is typical only for water bath or low pressure steam heaters.

Even though firetube failure is rare, it is advisable to prevent movement or flexing with restraining bars. This prevents weakening of the weld joints at the end plate. In addition, when the fuel is oil, include a drain with plug in the bottom of the firetube leg between the end plate and the burner flange. Then the firetube can be drained if any oil accidentally gets past the burner.

Inspirating partial pre-mix burners are used in the vast majority of firetube heaters. The gas is preheated before expansion, flow control, and flow through the burner. While the burner draws the primary air into the firetube, it is the stack draft, usually less than 1 in. H_2O, that overcomes the pressure drop of the combustion gas flow and admits the secondary air. The stack height is 10 to 20 ft.

An alternative to the burner front shown in the equipment sketches is a flame arrestor, the element of which provides many small tortuous paths between rolled sheets of thin corrugated aluminum. Sucking the combustion air through the element is an additional pressure drop for the stack draft to overcome. The term arrestor is somewhat of a misnomer because any fire in the firetube or around the burner is contained rather than extinguished. The passage through the arrestor cools the gas so that external combustion does not occur.

Controls

Fig. 8-39 shows an example control system for an indirect fired heater. Fig. 8-40 lists the shutdown/alarms as shown with some comments about the proper installation and use of these particular devices. The control system as depicted by these figures should not be considered complete but only representative of the conditions to be carefully considered in designing a control system for fired equipment. The controls probably vary more than the design of the heater. For example, a wellhead line heater or glycol dehydrator may have no more than an on-off thermostat for the main burner and a small continuous pilot. A line heater, hydrocarbon vaporizer, or amine reboiler may have all of the controls listed in Fig. 8-40.

Troubleshooting

The following problems can occur with firetube heaters.

- **Bath level loss** can be the result of too high a bath temperature. This is often caused by the temperature controller on the process stream. Fouling of the process coil, internal and/or external, means a hotter bath is needed to accomplish the same heat transfer. The coil should be removed and cleaned.

If fouling of the coils is not the problem, water losses can be reduced with a vapor recovery exchanger mounted on top of the heater shell. It consists of thin tubes that condense the water vapor. Vapor losses can also be reduced by altering the composition of the heat medium or, in drastic cases, by changing the heat medium.

- **Shell side corrosion** is caused by decomposition of the bath. (The decomposition products of amines and glycols are corrosive.) Some decomposition and corrosion is inevitable; however, excessive decomposition is usually due to overheating near the firetube. Corrosion inhibitors are commonly added. There are numerous reasons for overheating the bath: localized ineffective heat transfer caused by fouling, excessive flame impingement, etc. An improper flame can sometimes be modified without system shutdown. Fouling, however, requires removal of the firetube.
- **Inadequate heat transfer** may result from improper flame, under-firing, firetube fouling, coil fouling, poor shell fluid dynamics, too small a firetube or coil, etc. If it is not improper design, then it is most likely fouling or an improper flame. The solution may be a simple burner adjustment to correct the air to fuel mixture.
- **High stack temperature** can be the result of an improper air to fuel mixture. A leak of combustible material from the process side to the firetube is also a cause. It can also be the result of excessive soot deposition in the firetube.
- **Firetube failure** is most commonly caused by localized overheating and subsequent metallurgical failure. These

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FIG. 8-36

Typical Physical Properties of Hot Oil

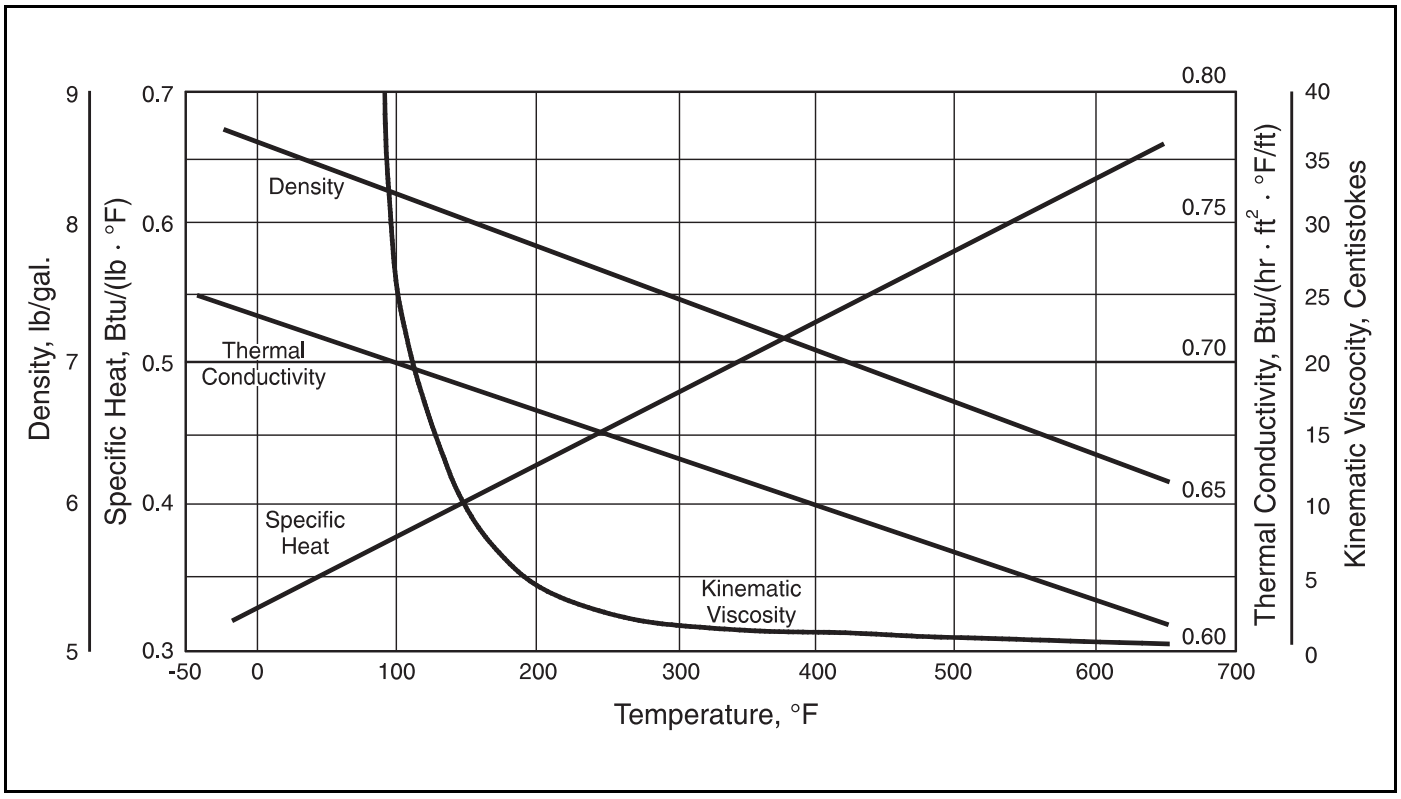
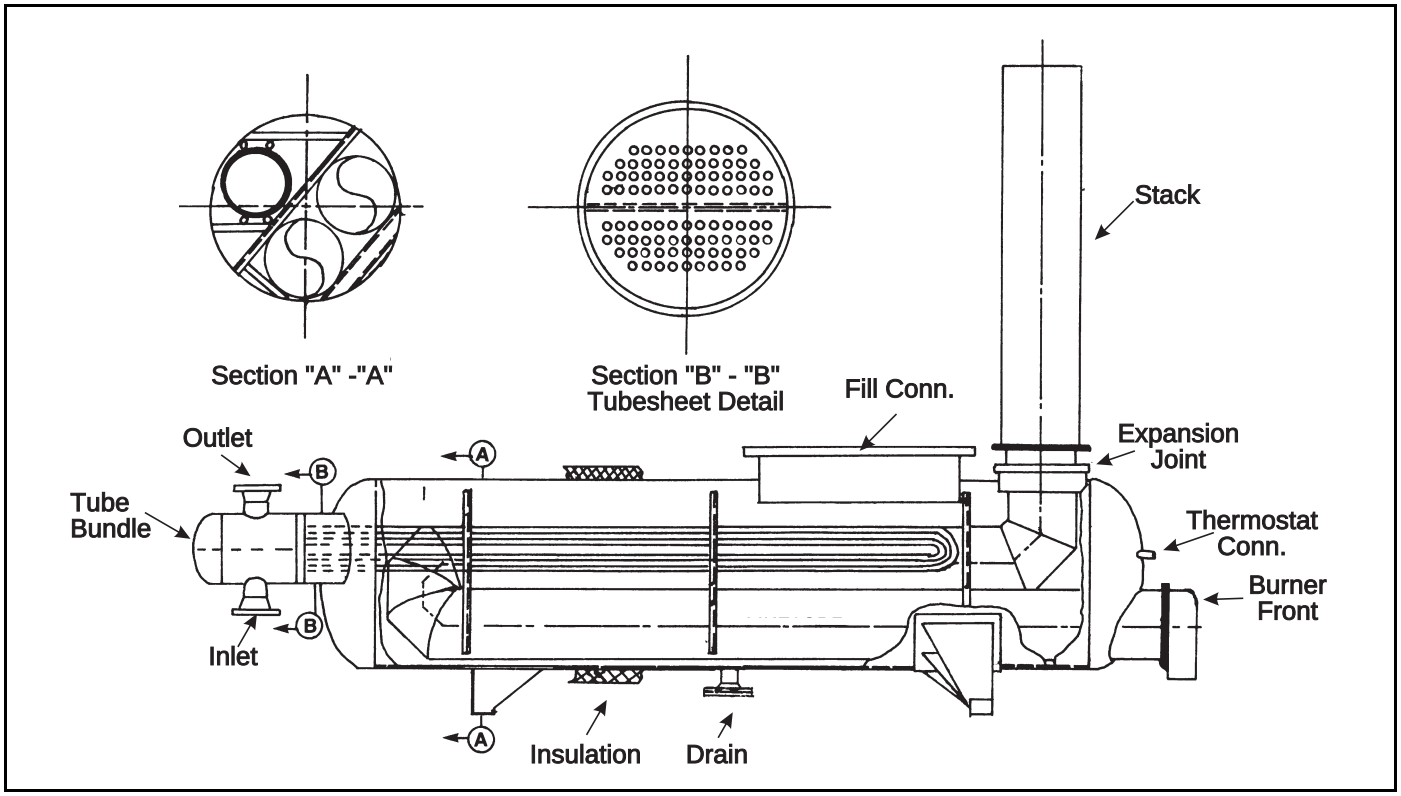


FIG. 8-37

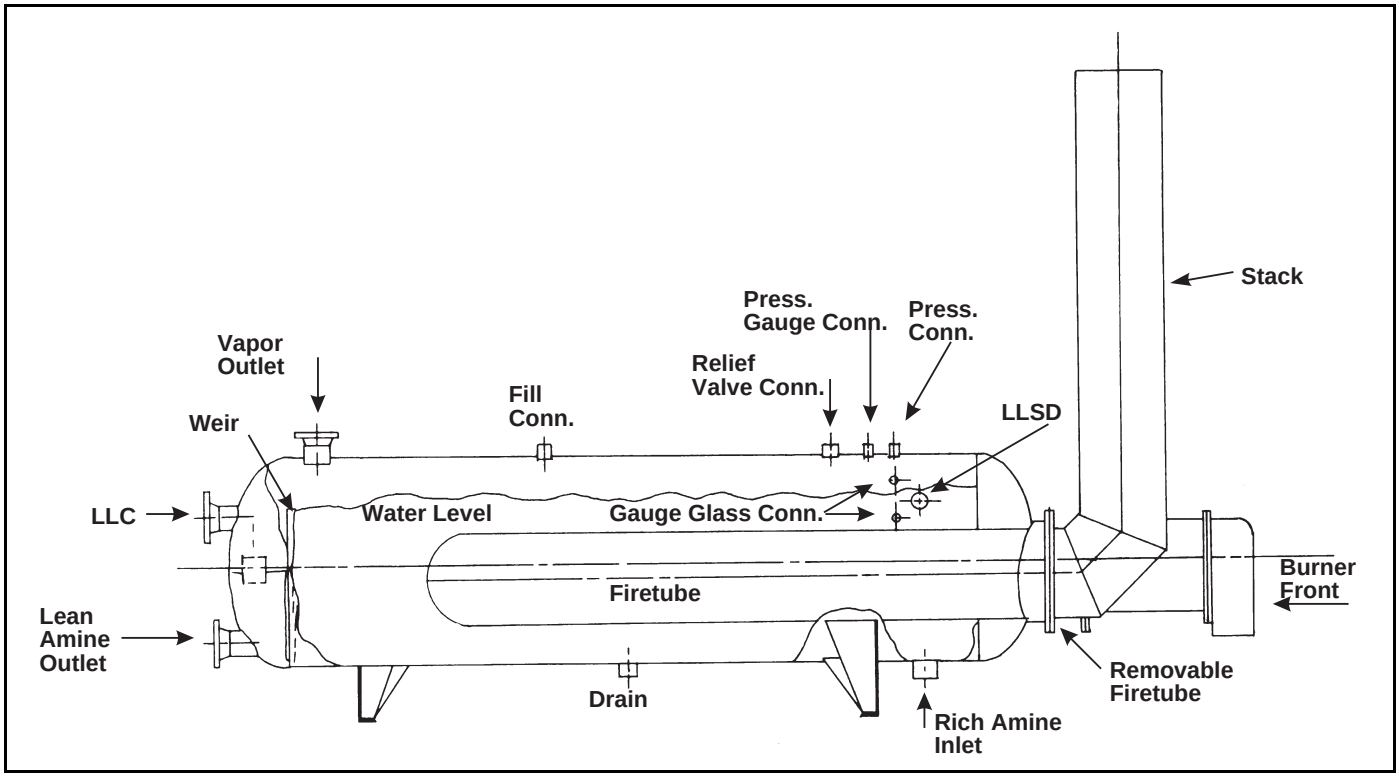
Salt Bath Heater



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FIG. 8-38

Amine Reboiler



"hot spots" are caused by hydrocarbon coking and deposition on the bath side. Firetube corrosion is caused by burning acid gases for fuel. The most damaging corrosion occurs in the burner assembly. There is little that can be done except to change the fuel and this may be impractical. Proper metallurgy is essential when burning acid gases.

- **High or low fuel gas pressure** can have a dramatic effect on the operation of a firetube heater. Burners are typically rated as heat output at a specified fuel pressure. A significantly lower pressure means inadequate heat release. Significantly higher pressure causes over-firing and overheating. The most common causes of a fuel gas pressure problem are the failure of a pressure regulator or an unacceptably low supply pressure.

Improved Thermal Efficiency

Economic incentives have promoted the development of devices to improve the thermal efficiency, i.e., reduce the excess air and the stack temperature.

Control of the flow of air into the firetube or the gas flow in the stack with dampers is sensitive because the relatively weak stack draft is easily influenced by an additional pressure drop. Several designs are available: axially rotating vanes around the burner, a pivoting horseshoe around the burner, a hinged plate over the air inlet duct, a rotating plate in the stack, etc.

Methods to increase firetube heat transfer are shown in Fig. 8-41. An economizer (end view) consisting of longitudinally finned tubes inserted into the return leg of the firetube adds heat transfer surface; a turbulator increases the heat transfer coefficient; and internal fins both add area and increase turbulence. Often additional equipment, e.g., a pump to circulate the bath through the economizer or an actuator to position the damper, is needed.

With good control of the excess air, i.e., 5% to 10% and a stack temperature of 400°F, the NTE approaches 90%. However, the pressure drop across the firetube increases and the stack draft decreases. This means that a forced draft burner may be required. The economics are usually favorable and short payout periods for these modifications are common.

Hot Oil System

A simplified schematic of the major components of a hot oil system is given in Fig. 8-42. The heat transfer medium is pumped through a fired heater to the heat exchangers and returns to the pump suction surge vessel. In some cases a fired heater is replaced by a waste heat source, such as the exhaust stack of a fired turbine. The slip-stream (typically less than 5%) filter is optional but it will help to retain the performance characteristics of the heat medium.

Proper design of the heater is critical for satisfactory operation. The heat transfer fluid must have sufficient velocity, generally 4 to 10 ft/sec, to avoid excessive film temperatures on the heater tubes. Hot spots can lead to tube failure and fluid degradation. Design and capacity of a heater should be limited so that the maximum film temperature does not exceed the maximum recommended operating temperature of the fluid.

The surge vessel is provided with blanket gas and vent connections. Expansion room for the hot oil from ambient to operating conditions must be provided. On small systems the surge tank may be sized to hold all of the heat medium. Two piping arrangements are used for the surge vessel; flow

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FIG. 8-39

Indirect Fired Heater

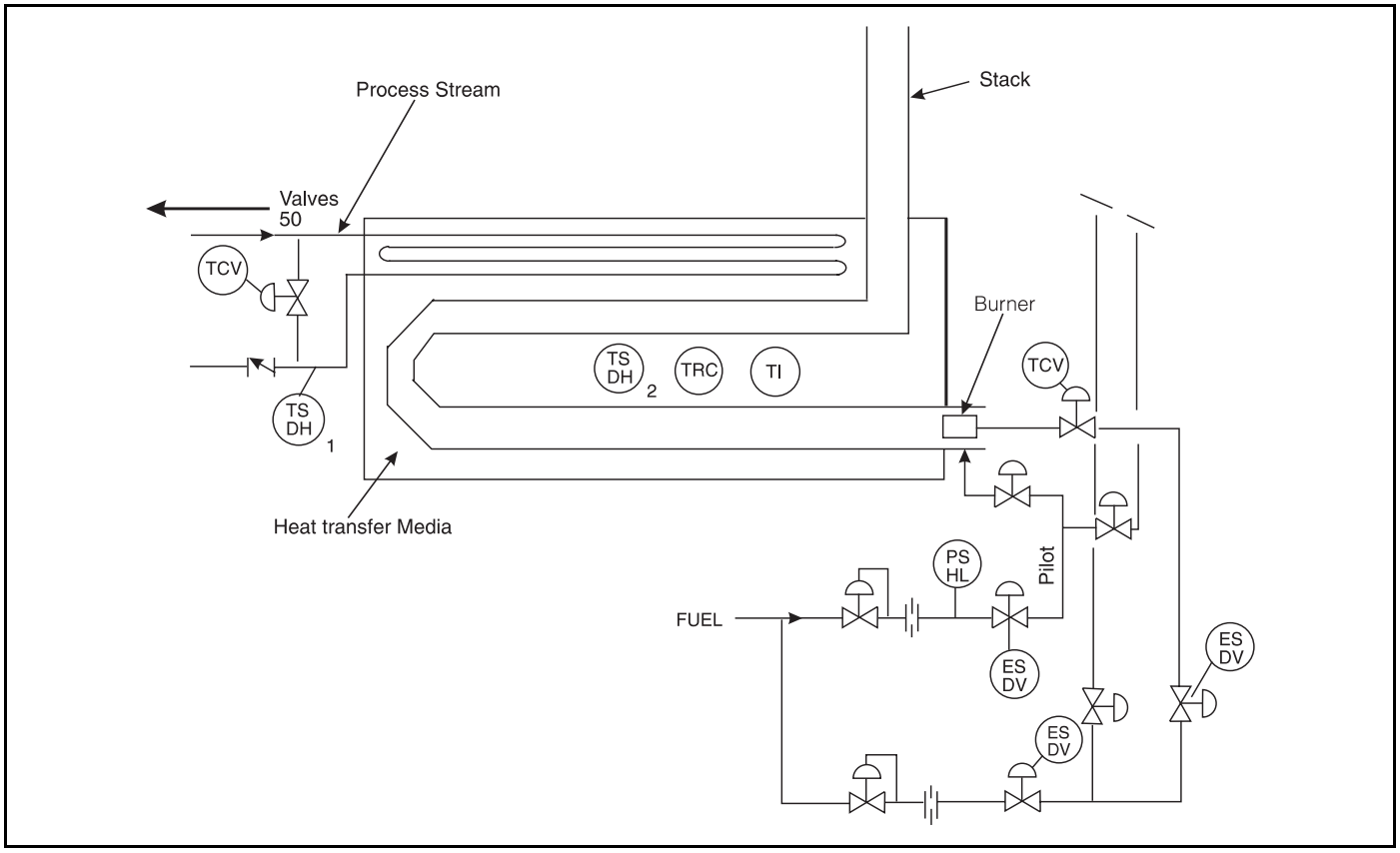


FIG. 8-40

Bath Heater Alarm/Shutdown Description

Note: Alarms and shutdowns as shown are not to be considered as meeting any minimum safety requirement but are shown as representative of types used for control systems.

Schematic Label	Alarm/Shutdown Description	Line Heater	Hydrocarbon Reboiler	Low Pressure Steam Heater	Hot Oil or Salt Heater	Glycol Reboiler	Amine Reboiler
TSDH-2	High Bath Temperature	No Note 1, 2	Yes	No	Yes	Yes	Yes
LSL	Low Bath Level	No Note 2	No	Yes	Yes Note 3	No Note 2, 3	Yes Note 2, 3
PSL	Low Fuel Pressure	Yes Note 4	Yes	Yes	Yes Note 4	Yes	Yes
PSH	High Fuel Pressure	Yes Note 4	Yes	Yes	Yes Note 4	Yes	Yes
BSL	Flame Failure Detection	No Note 2, 5	Yes Note 5	No Note 2	Yes Note 5	No Note 2	No Note 2
PSH	High Vessel Pressure	No	No	Yes Note 6	No	No	Yes Note 6

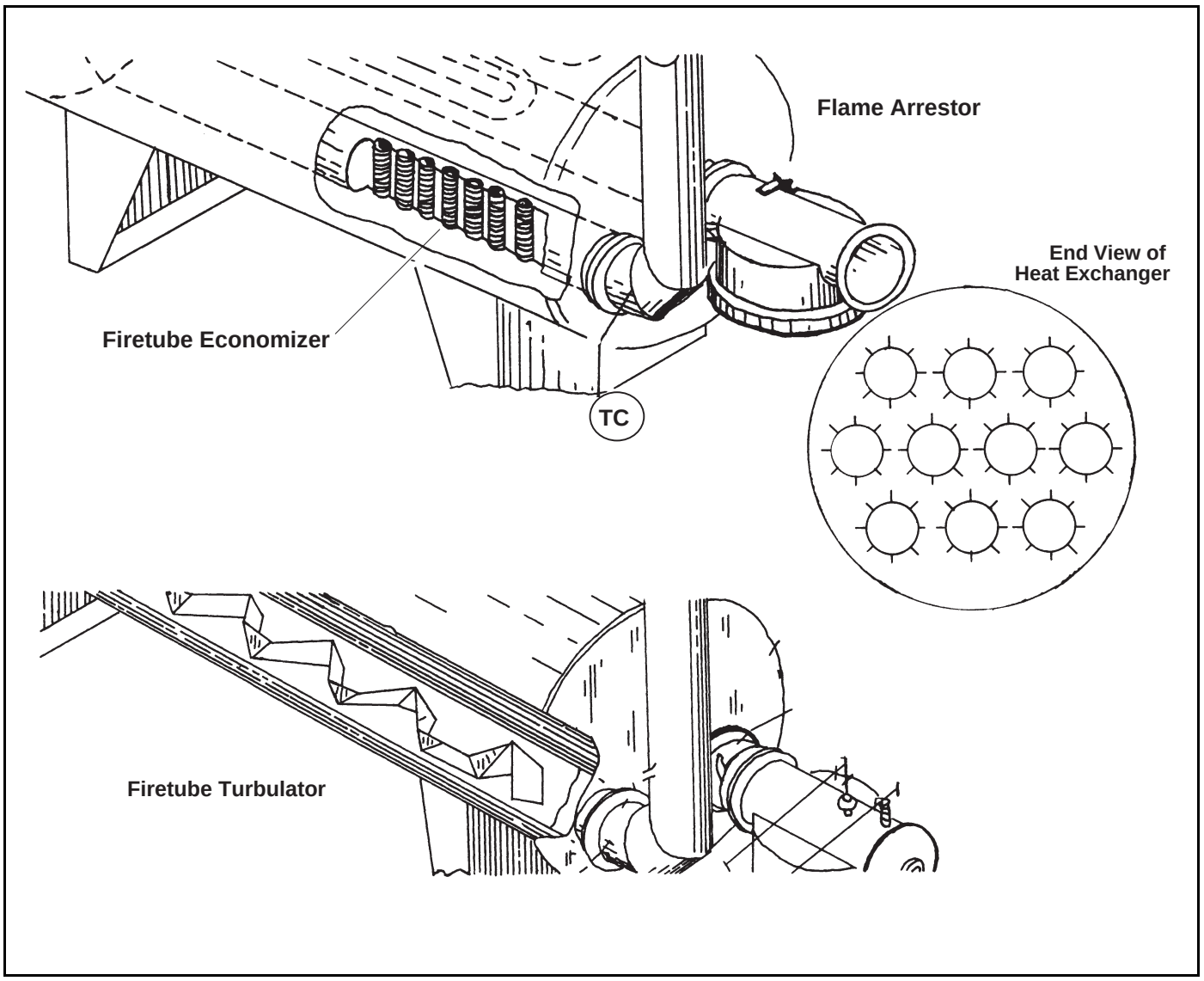
Notes:

1. When the process stream is oil, a high bath temperature shut down precludes the danger of coking.
2. This instrumentation is for heaters located in gas processing sections. Wellhead units have a minimum of controls.
3. Low bath level protects both the firetube and bath when it will coke (hot oils, glycol, amine) or decompose (molten salt).
4. Low/high fuel pressure is always used when the fuel gas is taken from the exit process stream.
5. Optical UV scanners or flame rods should be used because of the speed of response.
6. Code requirements, ASME Section IV or VIII.

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FIG. 8-41

Methods to Increase Firetube Heat Transfer



through the vessel or the vessel as a surge riding on the pump suction line.

Pump head requirements are usually 40 to 80 psi. Frequently the pump is spared and provided with isolating valves for servicing. A slip-stream should pass through the off-line pump to maintain the off-line pump at operating temperature. Selection of pumps should consider the high operating temperature and its effect on seals, packing rings, and gaskets. Slight leakage may occur at startup; the pump gland should not be tightened until the system reaches operating temperature. Suction strainers should be used during startup.

Piping design and installation must include consideration to minimize vapor traps and to relieve expansion and contraction stresses. The number of flanges in a hot oil system should be minimized; ANSI Class 300 lb flanges will aid in minimizing leakage.

During initial startup, the system may contain water, which must be slowly vaporized and removed. The surge vessel is sometimes the high point in the system to aid in this operation. On shutdown, fluid should be kept circulating to dissipate residual refractory heat. Planned maintenance should include analysis of the heat transfer medium, inspection of insulation, and inspection of heater and mechanical equipment.

Consult suppliers of hot oils to obtain design information such as Fig. 8-36 or suggestions such as reference 12.

VAPORIZERS

Vaporization Processes

Vaporization is a process in which a substance that is in liquid state is brought to vapor state. A variety of vaporization processes may be used in the gas industry. The process depends on the liquid composition, pressure and temperature; the required vapor pressure and temperature; the energy source type and temperature; and other aspects, such as,

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whether the liquid is flowing in a steady state process or whether the liquid is in a tank.

Some common vaporization processes in the gas industry are:

- **High pressure refrigerated and non-refrigerated liquids**

  Steady state flowing liquid.

  The liquid pressure is reduced through a pressure control valve or regulator, and the expansion vaporizes the liquid. If the resulting vapor temperature is too low, the fluid is heated. If the required vapor temperature is well below ambient temperature, the heat may come from the atmosphere or from a water source, using a heat exchanger placed downstream of the pressure reducer. If not, the heat may be supplied using a process stream heat exchanger or a heater usually placed upstream of the pressure reducer. If there is no requirement to reduce the liquid pressure, heat shall be supplied using criteria similar to above.

- **High pressure non-refrigerated LPG, propane and butanes**

  Liquid withdrawal from a tank containing the liquid.

  The liquid being extracted from the tank is vaporized in a similar process described above. The reduction of the liquid volume in the tank may be compensated with vapor supplied to the tank. Otherwise, there is vaporization inside the tank, causing some temperature and pressure drop to the liquid and vapor in it. If the heat transfer from the ambient to the tank and its content is such that the temperatures and pressures are not acceptable, additional heat can be supplied to the tank content.

  Vapor withdrawal from a tank containing the liquid.

  The vaporization occurs inside the tank, causing some temperature and pressure drop to the liquid and vapor in it. If the heat transfer from the ambient to the tank and its content is such that the temperatures and pressures are not acceptable, additional heat can be supplied to the tank content, or to the flowing vapor, or both.

- **Refrigerated liquids around atmospheric pressure**

  Unless there is sufficient liquid column to provide the pressure, the liquid shall be pumped to an adequate pressure before the vaporization process. If the required vapor temperature is well below ambient temperature, the heat may come from the ambient, using a heat exchanger placed downstream of the pump. If not, the heat may be supplied using a process stream heat exchanger or a heater.

Vaporization Equipment

The heat for vaporization may come from an ambient source, such as the air or water, or geothermal or sun energy, for the lower temperature applications. Alternatively, it may come from a process stream, steam, hot water, flue gases from a waste heat recovery unit or from a fired heater, and electricity, for higher temperature applications.

- **Stream vaporization**

To vaporize a stream, a heat exchanger or a heater may be used. Fired equipment commonly used are convection and water bath type.

- **Vaporization in the tank**

To vaporize the tank content internal or external equipment may be used to supply the heat. Internal equipment may be a heat exchanger inside the tank, such as a pipe bundle, which circulates a heated transfer fluid at sufficient temperature. External equipment may be a heat exchanger or a heater, circulating some liquid from the tank. For relatively small tanks, external heat may be supplied using fired or electric radiant heaters or electric resistance tracing. Fired equipment commonly used are convection and water bath type.

Submerged Combustion Water Bath Vaporizers

These heaters have one or more burners with the flame located in submerged chamber where heat is transferred to the water and the flue gases are discharged through a bubbling tube into the water. Because of the low bath temperature, most of the heat content of the flue gases is transferred to it, and most of the water vapor in the flue gases condenses giving up the latent heat. These heaters have very low NO_x emission because of the flame chamber is cooled by bath water thermosyphon circulation through a jacket, the cooling of the flue gases and excess air control. The flue gases in contact with the water turn it slightly acidic, because the CO_2 dissolves in it, forming carbonic acid; to avoid sulfur acids formation, the fuel gas must be sweet. These heaters are used for LNG and propane vaporization.

WASTE HEAT RECOVERY

Economical and environmental considerations may lead to the use of waste heat recovery systems. Flue gases from fired equipment, combustion engines and gas turbines are common heat sources for consideration to use waste heat recovery. The recovered heat may be used in the same equipment to increase its thermal efficiency and to supply heat to other equipment. Fired equipment also may combine the heating of several process streams at different temperatures. Typical applications include combustion air and/or fuel preheating in the same equipment. Burners can be self-recuperative and self-regenerative. Some applications include additional heat input by refiring.

Heat transfer systems may have:

- Streams contact, as in bubbling flue gases in water heaters.
- No contact of streams, as in shell and tube heat exchangers.
- An intermediate heat transfer fluid by forced circulation as in pumped oil systems, or by natural circulation as in thermosyphons. The intermediate fluid may have a phase change as in heat pipes.
- A solid heat transfer media, such as in rotary or alternating regenerative devices.

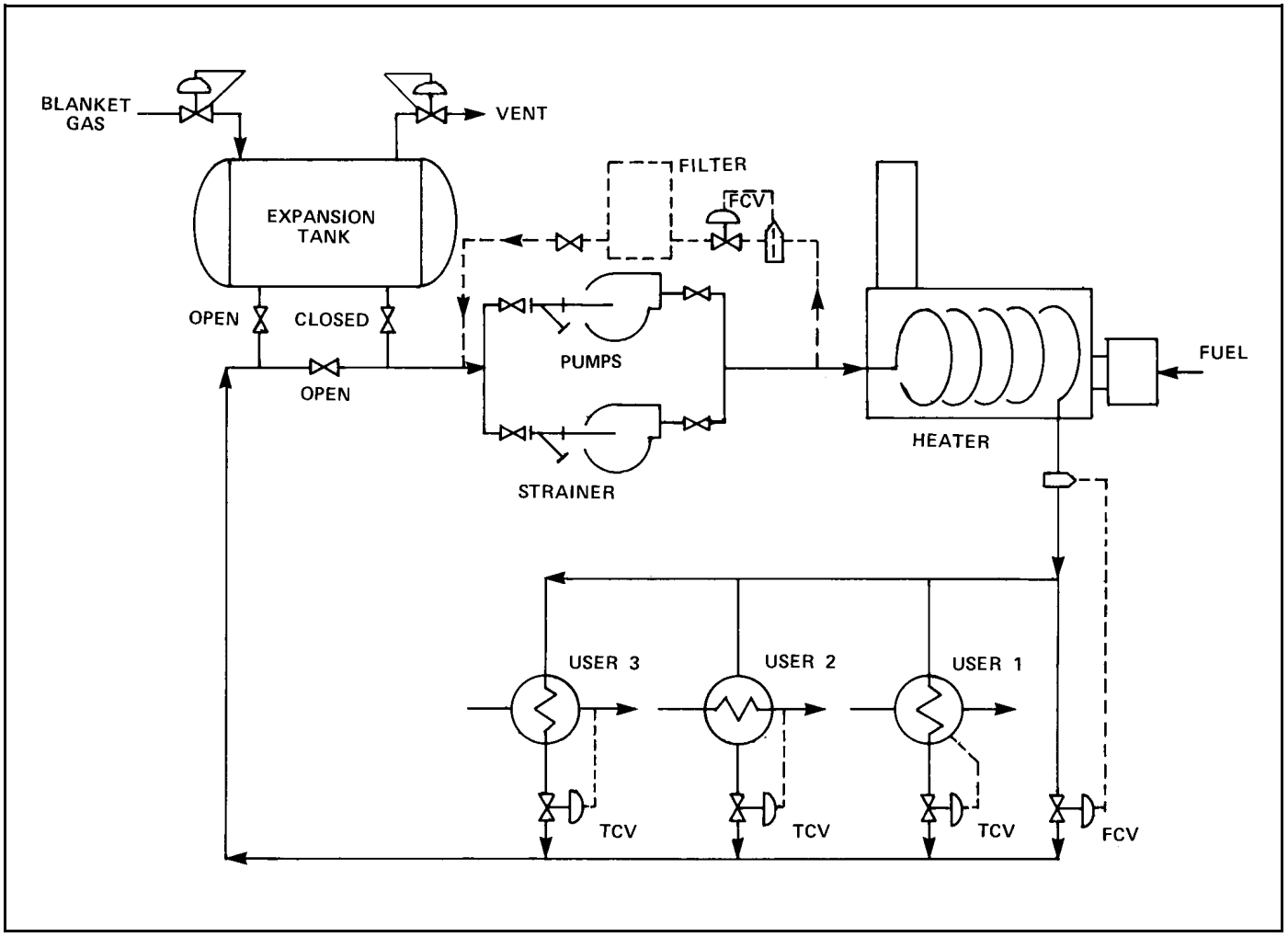
In some special devices, such as combustion engine and turbine exhaust gas silencers, a fluid is circulated for heat recovery.

Applications of waste heat recovery from fired equipment, combustion engine and gas turbine flue gases include process stream heating, water heating, steam generation, liquids and solids regeneration, heat supply to absorption chillers, combined heat and power systems, and mechanical power as in turbochargers. Heat recovery equipment introduces additional head losses in the flue gas path, with the consequence

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FIG. 8-42

Example Hot Oil System



of possible changes in required stack height, fan power, and damper position. In the case of combustion engines and gas turbines, there is a loss of power.

Exhaust gases from equipment with high excess air, such as gas turbines which contain 13 to 17% molar volume of oxygen, temperatures between 850°F and 1050°F and pressures up to 10 inches water column, may be used to supply the oxygen for the combustion of fuels in fired heaters.

OXIDIZERS

In processing plants, many hazardous air pollutant (HAP) gases and some volatile liquids may be recovered or burned in flares. Some compounds can not be recovered because it is not technically or economically feasible, or can not be burned in flares because of their chemical nature or because they are too diluted in a non-combustible substance. Oxidizers, also called incinerators, are equipment for the thermal destruction of HAP. In gas processing plants oxidizers are used to thermally destroy, by oxidization, the HAP that can not be recovered nor burned in flares. In the ideal oxidizer, the wastes are converted to carbon dioxide and water, and when the wastes contain HAPs such as sulfur or halogenated compounds, they are converted to non-hazardous compounds.

The basic thermal oxidizer normally consists of a chamber through which the waste gas or atomized liquid flows with sufficient combustion air; heat is supplied to obtain a high temperature. The equipment is designed to provide sufficient residence time and turbulence for oxidation. Common range of the oxidizing chamber temperature is from 1400 to 2000°F and common residence time ranges from 0.25 to 1 second.

The higher the oxidizing temperature the higher the destruction efficiency, but also the higher the NO_x production. In addition, the incoming wastes may contain nitrogenated compounds. The selected temperature is a compromise between the efficiency and NO_x emission. Excessive NO_x may be produced, in which case a high-temperature reducing chamber may be used.

To remove non-combustible particles and to absorb secondary pollutants from effluent gases, a quench-water tank and/or a water-spraying chamber may be used.

Secondary pollutants may be hydrochloric and/or hydrofluoric and/or SO_x acids. The wastewater may be passed through a filter where particles are retained, and the wastewater is

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neutralized with a caustic salt, to adjust the pH, and recycled. Dynamic separators and/or filters may be used for dry particle separation.

Waste heat may be recovered with a recuperative or a regenerative exchanger to heat the incoming gases, and/or to heat a process stream or produce/reheat steam. Catalytic chambers may be provided, to improve the oxidization reaction and to reduce the energy consumption by reduction of the operating temperature to the range of 350°F to 600°F.

Concentration of the HAP may be used to reduce the oxidizer load or recovery. Concentration of the gaseous stream may be achieved by absorption or adsorption, regeneration or nonregeneration, and condensation. A fired water evaporation concentrator may be used in the case of diluted aqueous waste.

The oxidizer design data provided to the manufacturer, should include the incoming waste gas composition, maximum and minimum flow, pressure and temperature, identification of the source, fuel gas information and site data. The gas composition analysis to be supplied to the manufacturer should include, at least the components given in the Fig. 8-43.

Sulfur containing gases may be burned in a flare or oxidizer or fed to a sulfur recovery unit. The election depends on environmental regulations, sulfur content of the gas, the permitted sulfur emissions, and economical aspects. In the case where sulfur is not recovered, it may be necessary to scrub the combustion gases.

INERT GAS GENERATORS

An oxygen-free gas, containing mainly nitrogen, carbon dioxide and water vapor may be obtained with a type of "inert" gas generator consisting of a combustion chamber where air enters and the oxygen content is totally consumed by the combustion in a controlled burner. The exiting gas may be cooled by preheating the combustion air and fuel, and if needed, cooled further with a gas cooler, and then the condensed water separated. The combustion air may be obtained from a heater, boiler or gas turbine exhaust, and a refiring burner consumes the remaining oxygen. The gas may be dehydrated as needed.

ELECTRIC HEATERS

These are heaters in which the energy source is electricity, and resistances, inducing coils, or radiant elements may be the heating elements. Electric resistances, protected by an electrical-insulated sheath, may be installed internally in a vessel, a tank or a pipe, or externally around or adjacent to them, in which case external insulation may be needed. High temperature electric resistances are also used in incinerators. Also, electric radiant heaters may be used to externally heat a tank or a vessel.

Electric intensity and resistance equations may be found in Section 18.

The heat flux of a resistance, also called watt density, is the rated power per unit heating-surface area of the resistance, and indicates the potential to transmit heat. The higher the watt density, the greater the possibility for excessive sheath temperature, and may cause resistance or sheath damage or excessive temperature of the fluid film in contact with it. So the heating element selection must consider not only the heating element materials and performance, but also the fluid properties and heat transfer relationships, and due consideration must be given to the movement of the fluid around the elements.

In working with highly flammable or explosive fluids, it may not be safe to install the electric element in direct contact with the fluid, in which case an intermediate heat transfer fluid may be used.

There is a wide variety of applications for electric heaters in process plants, such as:

- Circulating fluid heaters, for liquid or gas heating or liquid vaporization.
- Bath heaters, for liquid or gas heating or liquid vaporization.
- Piping heat tracing.
- Equipment lube oil temperature control.
- Refrigerated tank foundation temperature control.
- Electric equipment space heating.
- Incinerators.

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SECTION 9

Heat Exchangers

The process engineer is frequently required to analyze heat exchanger designs, specify heat exchanger performance, and determine the feasibility of using heat exchangers in new services. This section is prepared with these specific operations in mind and is not intended as a design manual.

FUNDAMENTALS OF HEAT TRANSFER

The basic definitions and equations used in heat exchanger calculations are reviewed below:

Heat Balances

For no phase change of the hot fluid:

$$Q = \dot{m}_H \, Cp_H \, (T_{H1} - T_{H2}) \qquad \textbf{Eq 9-1}$$

For isothermal condensing of the hot fluid:

$$Q = \dot{m}_H \, \lambda_H \qquad \textbf{Eq 9-2}$$

For no phase change of the cold fluid:

$$Q = \dot{m}_C \, Cp_C \, (T_{C2} - T_{C1}) \qquad \textbf{Eq 9-3}$$

For isothermal boiling of the cold fluid:

$$Q = \dot{m}_C \, \lambda_C \qquad \textbf{Eq 9-4}$$

Basic Heat Transfer Relations

Q = UA (LMTD) for single-pass designs **Eq 9-5a**

Q = UA (CMTD) for multi-pass designs **Eq 9-5b**

FIG. 9-1

Nomenclature

A = area, sq ft
BP = baffle spacing, in.
C = tube count factor
Cp = specific heat, Btu/(lb • °F)
CMTD = Corrected Log Mean Temperature Difference, °F
D = diameter, in.
F = LMTD correction factor
f = ratio of one value to another
FD = free diameter, in.
G = mass velocity, lb/(sq ft • sec)
GTTD = Greatest Terminal Temperature Difference, °F
h = film coefficient, Btu/(hr • sq ft • °F)
H = height, in.
k = thermal conductivity, Btu/[(hr • sq ft • °F)/ft]
L = length, in.
LMTD = Log Mean Temperature Difference, °F
LTTD = Least Terminal Temperature Difference, °F
$\dot{m}$ = mass flowrate, lb/hr
N = number of exchangers
Np = number of passes
ΔP = pressure drop, lb/sq in.
P = pressure, lb/sq in. absolute
p = temperature efficiency
PHE = plate and frame heat exchanger
Q = heat transferred, Btu/hr
R = heat capacity rate ratio
Re = Reynolds number = (123.9 DG)/μ
RC = tube rows crossed
r = film resistance (hr • sq ft • °F)/Btu
SP = number of baffle spaces
T = temperature, °F
t = temperature, °F
TMTD = True Mean Temperature Difference, °F
U = overall heat transfer coefficient, Btu/(hr • sq ft • °F)
W = width, in.
WTD = weighted temperature difference, °F
X = weight fraction
λ = latent heat, Btu/lb
μ = viscosity, centipoise
ρ = density, lb/cu ft

Subscripts

b = boiling
C = cold fluid
c = condensing
f = fouling
H = hot fluid
i = inside
in = inlet
L = liquid
m = mean value
n = n^{th} value
o = outside
out = outlet
2Φ = two-phase
w = wall
v = vapor
1 = first value
2 = second value

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Shell and Tube Exchangers

For tubular heat exchangers, the heat transfer area generally referred to is the effective outside bare surface area of the tubes, and the overall heat transfer coefficient must also be based on this area.

Effective Temperature Difference

In most instances the local temperature difference between the hot stream and the cold stream will not have a constant value throughout a heat exchanger, and so an effective average value must be used in the rate equation. The appropriate average depends on the configuration of the exchanger. For simple countercurrent and co-current exchangers (Fig. 9-2), the Log Mean Temperature Difference (LMTD) applies.

FIG. 9-2

Countercurrent Flow and Co-current Flow

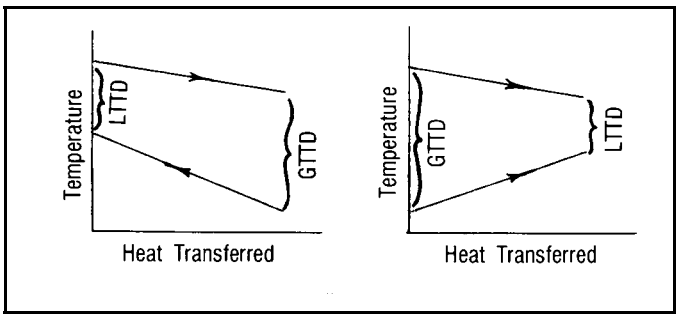


Fig. 9-3 defines LMTD in terms of Greatest Terminal Temperature Difference (GTTD) and Least Terminal Temperature Difference (LTTD), where "terminal" refers to the first or last point of heat exchange in the heat exchanger.

For exchanger configurations with flow passes arranged to be partially countercurrent and partially co-current, it is common practice to calculate the LMTD as though the exchanger were in countercurrent flow, and then to apply a correction factor to obtain the effective temperature difference.

$$\text{CMTD} = (\text{LMTD})\,(\text{F}) = \text{Corrected Mean Temperature Difference} \qquad \textbf{Eq 9-6}$$

The magnitude of the correction factor, F, depends on the exchanger configuration and the stream temperatures. Values of F are shown in Figs. 9-4, 9-5, 9-6, and 9-7 for most common exchanger arrangements. In general if the value obtained for F is less than 0.8, it is a signal that the selected exchanger configuration is not suitable, and that one more closely approaching countercurrent flow should be sought.

Heat Exchange with Non-Linear Behavior

The above Corrected Log Mean Temperature Difference (CMTD) implicitly assumes a linear relation between duty and stream temperature change. Some situations for which this assumption is not applicable include process streams which undergo a very large temperature change so that the physical properties change significantly, multi-component condensing or boiling with non-linear duty vs. temperature curves, and exchangers in which the process stream undergoes both phase change and sensible cooling or heating.

These situations may be handled by dividing the exchanger into zones which may be treated individually with the linear assumption. The overall exchanger performance may be represented in terms of the weighted average performance of the zones in the overall rate equation. The following equations may be taken as the rate equations for the overall exchanger and for the nth zone of the exchanger.

$$Q_{Total} = U_{wtd}\,A_{Total}\,(WTD) \qquad \textbf{Eq 9-7}$$

$$Q_n = U_n\,A_n\,(LMTD)_n \qquad \textbf{Eq 9-8}$$

Then the weighted temperature difference may be defined as:

$$(WTD) = \frac{\Sigma\,[U_nA_n(LMTD)_n]}{\Sigma[U_nA_n]} = \frac{Q_{Total}}{\Sigma[Q_n/(LMTD)_n]} \qquad \textbf{Eq 9-9}$$

And the weighted overall heat transfer coefficient becomes:

$$U_{wtd} = \frac{Q_{Total}}{A_{Total}(WTD)} = \frac{\Sigma[Q_n/(LMTD)_n]}{A_{Total}} \qquad \textbf{Eq 9-10}$$

In multi-component, two-phase (vapor/liquid) flow regimes undergoing heat transfer, the vapor and liquid composition changes that occur are related to the extent of continuous contact of the two phases. If the vapor phase is maintained in contact with the liquid, the total change in enthalpy (or other properties) that accompanies the composition change is termed "integral." If the vapor is continuously removed from contact with the liquid as it is formed, the property changes are termed "differential." An accurate representation of temperature difference and heat transfer in these cases depends on correct consideration of the phase separation that occurs in the heat transfer equipment.

Overall Heat Transfer Coefficient

$$U_o = \frac{1}{\left[\frac{1}{h_o} + \left(\frac{A_o}{A_i}\right)\left(\frac{1}{h_i}\right) + r_w + r_{fo} + \left(\frac{A_o}{A_i}\right)r_{fi}\right]} \qquad \textbf{Eq 9-11}$$

Metal Resistance for Plain Tubes

The metal resistance is calculated by the following equation:

$$r_w = \left(\frac{D_o}{24k_w}\right)\ln\left(\frac{D_o}{D_i}\right) \qquad \textbf{Eq 9-12}$$

Values of the tube metal thermal conductivity are found in Fig. 9-8 for several materials of construction at different metal temperatures.

Fouling Resistances

Fouling resistances depend largely upon the types of fluid being handled, i.e., the amount and type of suspended or dissolved material which may deposit on the tube walls, susceptibility to thermal decomposition, etc., and the velocity and temperature of the streams. Fouling resistance for a particular service is usually selected on the basis of experience with similar streams. Some typical values are given in Fig. 9-9 and in the TEMA Standards.

Film Resistances

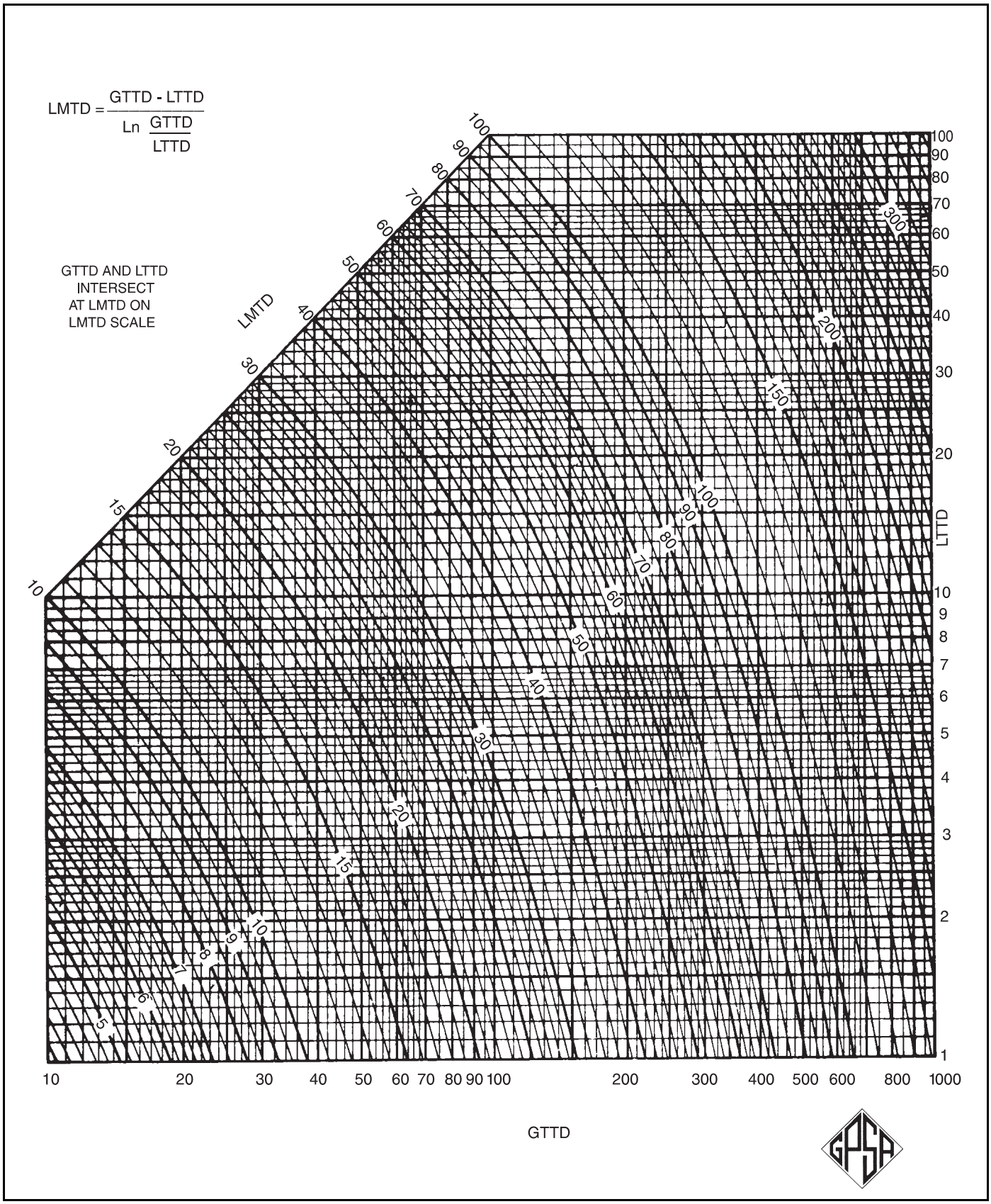
Equations for calculating the film coefficients, h_o, and h_i, for the simpler common geometries, as functions of flow rate and fluid properties, may be found in heat transfer references and in engineering handbooks. Some typical values of film resistances are given in Fig. 9-11. Some common overall heat transfer coefficients are shown in Fig. 9-9.

Film coefficients, film resistances, and overall heat transfer coefficient are related as follows: $h_i = 1/r_i$, $h_o = 1/r_o$, and $U = 1/\Sigma r$ (as in Eq 9-11).

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FIG. 9-3

LMTD Chart



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FIG. 9-4

LMTD Correction Factor (1 shell pass; 2 or more tube passes)

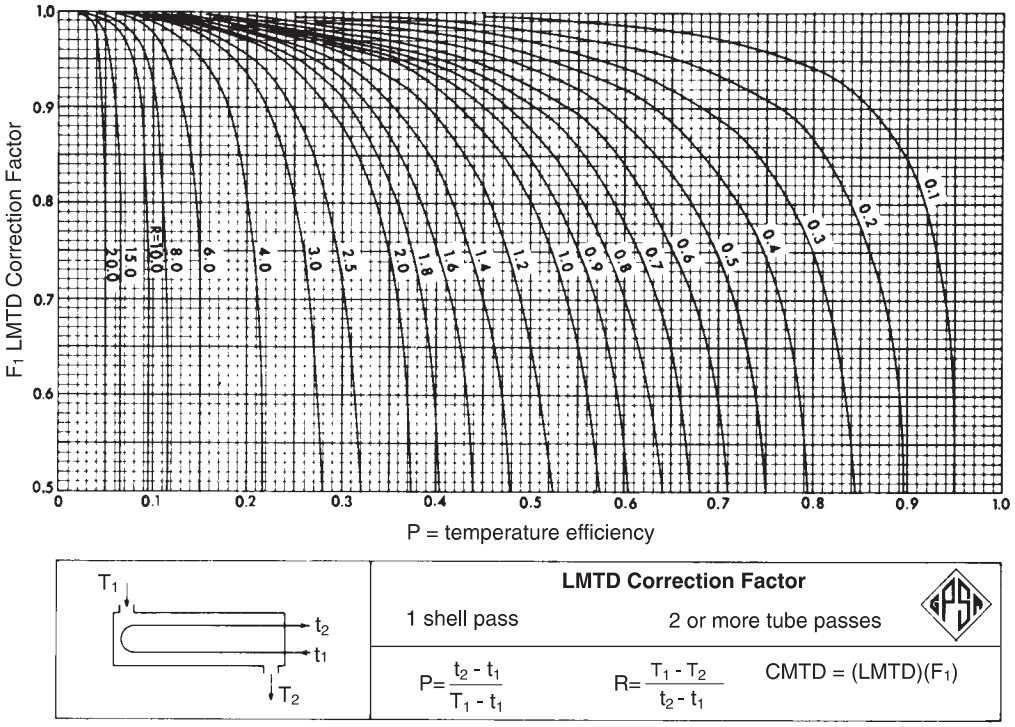
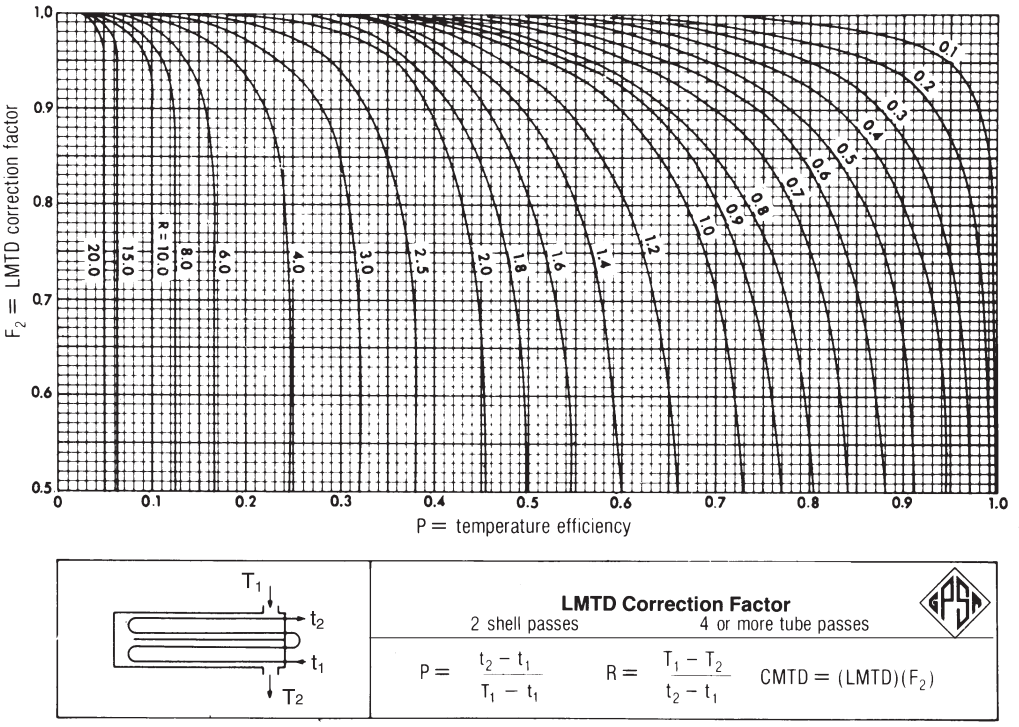


FIG. 9-5

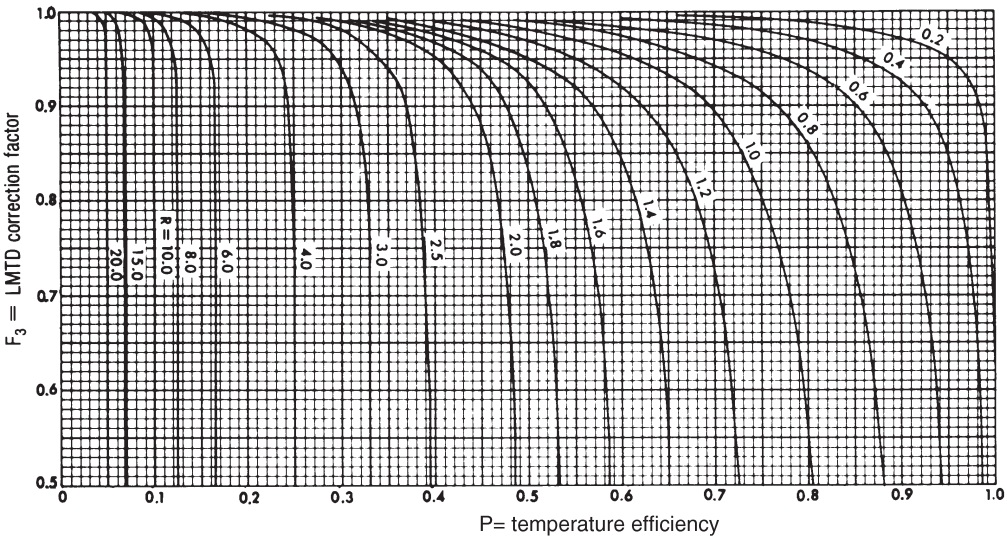
LMTD Correction Factor (2 shell passes; 4 or more tube passes)

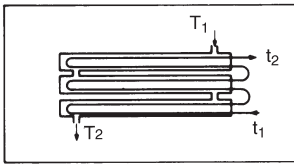


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FIG. 9-6

LMTD Correction Factor (3 shell passes; 6 or more tube passes)





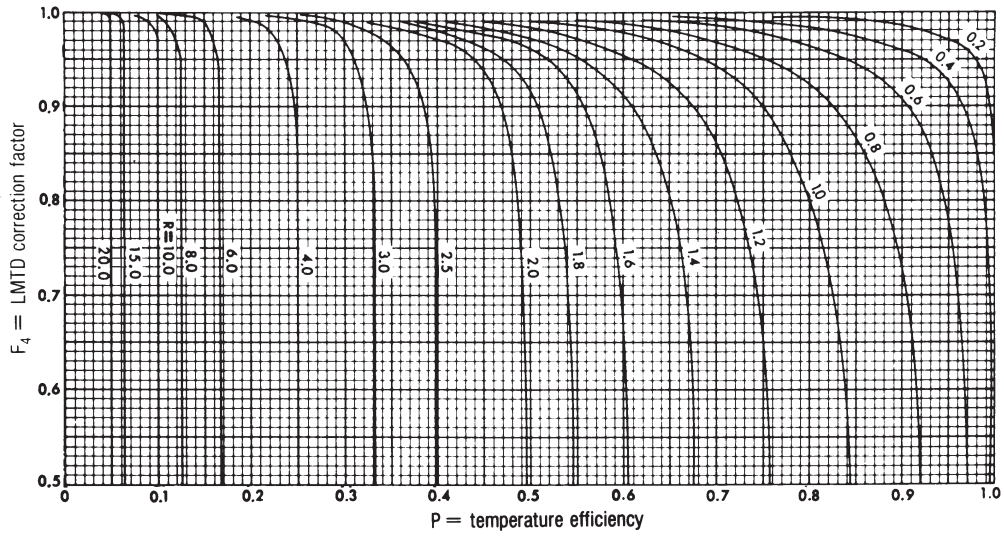
LMTD Correction Factor

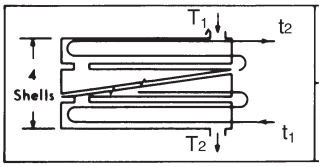
3 shell passes — 6 or more tube passes

$$P = \frac{t_2 - t_1}{T_1 - t_1} \qquad R = \frac{T_1 - T_2}{t_2 - t_1} \qquad CMTD = (LMTD)(F_3)$$

FIG. 9-7

LMTD Correction Factor (4 shell passes; 8 or more tube passes)





LMTD Correction Factor

4 shell passes — 8 or more tube passes

$$P = \frac{t_2 - t_1}{T_1 - t_1} \qquad R = \frac{T_1 - T_2}{t_2 - t_1} \qquad CMTD = (LMTD)(F_4)$$

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FIG. 9-8

Typical* Metal Thermal Conductivities, k_w

Material	Btu/[(hr • sq ft • °F)/ft] Metal Temperature			
	200°F	400°F	600°F	800°F
Aluminum, 3003 Tempered	104	106	–	–
Carbon Steel	29	28	26	24
Carbon Moly (½%) Steel	25	25	24	22
2¼% Cr, 1% Mo Steel	21	22	21	20
13 Cr	16	16	16	16
304 Stainless Steel	9.3	10	11	12
Admiralty	70	79	89	–
Copper	225	224	223	–
90-10 CuNi	30	34	42	49
70-30 CuNi	18	21	25	30
Nickel 200	39	35	33	33
NiFeCrMoCu (Alloy 825)	7.1	8.1	9.1	10
Titanium	12	11.5	11.2	11.2

* Excerpt from TEMA Standards

FIG. 9-9

Typical Heat Transfer Coefficients, U, and Fouling Resistances, r_f

Service and (r_f)	U	Service and (r_f)	U
Water (0.002)/			
100 psi Gas (0.001)	35-40	Rich (0.001)/Lean Oil (0.002)	80-100
300 psi Gas (0.001)	40-50	C_3 Liq/C_3 Liq (0.001)	110-130
700 psi Gas (0.001)	60-70	MEA/MEA (0.002)	120-130
1000 psi Gas (0.001)	80-100	100 psi Gas/500 psi Gas	50-70
Kerosene (0.001)	80-90	1000 psi Gas/1000 psi Gas	60-80
MEA (0.002)	130-150	1000 psi Gas/Cond. C_3 (0.001)	60-80
Air (0.002)	20-25	Steam (0.0005) Reboilers	140-160
Water (0.001)	180-200	Hot Oil (0.002) Reboilers	90-120
Condensing with water (0.002)/		Heat Transfer Fluid (0.001) Reboilers	80-110
C_3 or C_4 (0.001)	125-135		
Naphtha (0.001)	70-80		
Still Overhead (0.001)	70-80		
Amine (0.002)	100-110		

U in Btu/(hr • sq ft • °F)
r_f in (hr • sq ft • °F)/Btu

Performance Evaluation With Sensible Heat Transfer

To predict the performance of a particular exchanger in a new service or to compare different designs for a given service, it is useful to understand the effects of changes in the variables on film resistance to heat transfer and pressure drop. If variables (subscripted "1") are used for a reference basis (as those values given in Fig. 9-11 are intended to be) a proration to a new condition (subscripted "2") can be applied based on ratioing the correlation of the variable at the new condition to the reference condition. For film coefficients and pressure drop determinations, Fig. 9-10 summarizes these ratios for the applicable variables. If tube side film resistance and pressure drop at new conditions involving turbulent flow were desired, the variable arrays would be:

$$\frac{h_i}{h_2} = \frac{r_2}{r_1} = \left(\frac{\mu_2}{\mu_1}\right)^{0.47}\left(\frac{k_1}{k_2}\right)^{0.67}\left(\frac{Cp_1}{Cp_2}\right)^{0.33}\left(\frac{G_1}{G_2}\right)^{0.8}\left(\frac{D_{i2}}{D_{i1}}\right)^{0.2} \quad \textbf{Eq 9-13}$$

and,

$$\frac{\Delta P_2}{\Delta P_1} = \left(\frac{\mu_2}{\mu_1}\right)^{0.2}\left(\frac{G_2}{G_1}\right)^{1.8}\left(\frac{\rho_1}{\rho_2}\right)\left(\frac{D_{i1}}{D_{i2}}\right)^{1.2}\left(\frac{Np_2}{Np_1}\right) \quad \textbf{Eq 9-14}$$

Shell side film resistance and shell side pressure drop have similar arrays. In Fig. 9-10 all the variables that change in a vertical column apply where the flow regime is appropriate.

The stream types and flow regimes shown in Fig. 9-11 are typical for most fluids encountered in gas plants. These base values of film resistance and pressure drop are used with the relationships given in Fig. 9-10 to evaluate an exchanger design or to project the performance of an exchanger in a new service. This can best be understood by following Example 9-1.

Example 9-1 — The heat exchanger specification sheet, Fig. 9-12, shows the heat transfer requirements and the mechanical design configuration for an oil-to-oil exchanger. Evaluate the indicated performance of this design.

Solution Steps

1. Check the heat balance on the data sheet. (See Fig. 9-12)

$$Q_H/Q_C = \frac{[\dot{m}_H Cp_H(T_{H1} - T_{H2})]}{[\dot{m}_C Cp_C(T_{C2} - T_{C1})]}$$

$$= \frac{[475{,}723(0.542)\,(197 - 100)]}{[650{,}860(0.60)\,(124 - 60)]} = 1.0$$

2. Calculate the LMTD.

$$\begin{array}{rcl} 197 & \rightarrow & 100 \\ \underline{124} & \leftarrow & \underline{60} \\ 73 & & 40 \end{array}$$

$$LMTD = \frac{(73 - 40)}{\ln(73/40)} = 54.9°F$$

Since the exchanger is countercurrent flow, the CMTD is the LMTD.

3. Check the required heat transfer coefficient.

$$U = \frac{25{,}000{,}000}{(4525)(54.9)} = 100.6 \text{ Btu/(hr • sq ft • °F)}$$

4. Calculate the tube side pressure drop and resistance to heat transfer with the relationships shown in Fig. 9-10 and the values shown in Fig. 9-11.

The total cross sectional flow area

$$= \frac{[784\,\pi\,(D_i)^2]}{4} = (784)\,(0.3019) = 237 \text{ sq in.}$$

$$G = \frac{(650{,}860)}{[(3600)(237/144)]} = 109.8 \text{ lb/(sq ft • sec)}$$

$$Re = \frac{(123.9)(0.62)(109.8)}{0.21} = 40{,}200$$

Therefore, it is turbulent flow since Re > 2000.

From Fig. 9-10 (see Note †), the ratio of the second to the first resistance is:

$$(r_i)_2 = (f)\,(r_i)_1$$

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FIG. 9-10

Variables in Exchanger Performance

Variable*	Flow Regime	$r_2 = (f) \cdot (r_1)$†		$\Delta P_2 = (f) \cdot (\Delta P_1)$†	
		Shell (f)	Tube (f)	Shell (f)	Tube (f)
Viscosity	Turbulent	$(\mu_2/\mu_1)^{0.27}$	$(\mu_2/\mu_1)^{0.47}$	$(\mu_2/\mu_1)^{0.15}$	$(\mu_2/\mu_1)^{0.2}$
Viscosity – bulk to wall correction	Streamline		$\left[\frac{\mu_1}{\mu_2} \cdot \frac{\mu_{w2}}{\mu_{w1}}\right]^{0.14}$		(μ_2/μ_1)
Thermal conductivity	Turb. or Streamline	$(k_1/k_2)^{0.67}$	$(k_1/k_2)^{0.67}$		
Sp. heat capacity	Turb. or Streamline	$(C_{p1}/C_{p2})^{0.33}$	$(C_{p1}/C_{p2})^{0.33}$		
Mass velocity	Turbulent	$(G_1/G_2)^{0.6}$	$(G_1/G_2)^{0.8}$	$(G_2/G_1)^{1.85}$	$(G_2/G_1)^{1.8}$
(or mass flowrate)	Streamline		$(G_1/G_2)^{0.33}$		(G_2/G_1)
Density	Turb. or Streamline			(ρ_1/ρ_2)	ρ_1/ρ_2
Tube diameter	Turbulent	$(D_{o2}/D_{o1})^{0.4}$	$(D_{i2}/D_{i1})^{0.2}$	$(D_{o1}/D_{o2})^{0.15}$	$(D_{i1}/D_{i2})^{1.2}$
Tube diameter	Streamline		$(D_{i2}/D_{i1})^{0.33}$		$(D_{i1}/D_{i2})^2$
Tube length	Streamline		$(L_2/L_1)^{0.33}$		
Tube passes	Turb. or Streamline				(N_{p2}/N_{p1})
No. baffle spaces	Turb. or Streamline			SP_2/SP_1	
No. tube rows crossed‡				RC_2/RC_1	

* Use consistent units for any one variable in both cases.

† f is the ratio of the new value to the old value for a given variable. The overall f is the product of the individual fs.

‡ Number of rows of tubes exposed to cross flow (as opposed to parallel flow). This number is determined by baffle and bundle geometry

FIG. 9-11

Base Values for Use with Fig. 9-10(1)

Fluid	Flow Regime	Local r	Tubeside (One Pass)						
			k	Cp	ρ	ΔP/ft	μ(2)	G_i	D_i
Water	Turbulent	0.0009	0.358	1.0	62.1	0.068	0.764	265	0.62
HC Oil	Turbulent	0.0038	0.0788	0.5	46.9	0.061	0.726	185	0.495
Methane	Turbulent	0.0058	0.02	0.54	0.27	0.137	0.0113	31.2	0.62
HC Oil	Streamline	0.049	0.0716	0.526	51.3	(4)	(3)	42.3	0.834
			Shellside						
			k	Cp	ρ	ΔP(6)	μ(5)	G_o(7)	D_o
Water	Turbulent	0.0005	0.395	1.007	59.8	0.23	0.282	156.7	0.625
HC Oil	Turbulent	0.0028	0.0762	0.557	46.8	0.25	0.549	132.4	0.625
Methane	Turbulent	0.0038	0.037	0.655	0.23	0.09	0.0182	6.19	0.625

(1) Symbols and units are defined in Fig. 9-1

(2) Bulk average viscosity

(3) 6.62 and Wall viscosity is 27.75

(4) 0.14 psi for a 17 ft tube

(5) Average film viscosity

(6) Crossflow ΔP/baffle space/10 tube rows crossed between centroids of cut openings.

(7) Average crossflow mass velocity (see crossflow area calculation in Fig 9-13)

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Use base values from Fig. 9-11 for $(r_i)_1$ conditions.

$$f = \left(\frac{\mu_2}{\mu_1}\right)^{0.47}\left(\frac{k_1}{k_2}\right)^{0.67}\left(\frac{Cp_1}{Cp_2}\right)^{0.33}\left(\frac{G_1}{G_2}\right)^{0.8}\left(\frac{D_{i2}}{D_{i1}}\right)^{0.2}$$

$$= \left(\frac{0.21}{0.726}\right)^{0.47}\left(\frac{0.0788}{0.078}\right)^{0.67}\left(\frac{0.50}{0.600}\right)^{0.33}\left(\frac{185}{109.8}\right)^{0.8}\left(\frac{0.62}{0.495}\right)^{0.2}$$

$$= 0.840$$

Basis: (Inside Area)

$(r_i)_2 = f(r_i)_1$ and $(r_i)_1 = 0.0038$ from Fig. 9-11

$= (0.840)\,(0.0038)$

$= 0.0032$ (hr • sq ft • °F)/Btu

From Fig. 9-10 (see Note †), the ratio of the second to the first pressure drop is:

$(\Delta P_i)_2 = (f)\,(\Delta P_i)_1$

Use base values from Fig. 9-11 for $(\Delta P_i)_1$ conditions.

$$f = \left(\frac{\mu_2}{\mu_1}\right)^{0.2}\left(\frac{G_2}{G_1}\right)^{1.8}\left(\frac{\rho_1}{\rho_2}\right)\left(\frac{D_{i1}}{D_{i2}}\right)^{1.2}\left(\frac{Np_2}{Np_1}\right)$$

$$= \left(\frac{0.21}{0.726}\right)^{0.2}\left(\frac{109.8}{185}\right)^{1.8}\left(\frac{46.9}{38.3}\right)\left(\frac{0.495}{0.62}\right)^{1.2}\left(\frac{1}{1}\right)$$

$$= 0.285$$

$(\Delta P_i)_2 = (f)\,(\Delta P_i)_1 = (0.285)\,(0.061) = 0.017$ psi/ft

For a 30 ft tube length the total ΔP is 30(0.017) = 0.51 psi

5. Calculate the shell side pressure drop and resistance to heat transfer with the relationships shown in Fig. 9-10, the values shown in Fig. 9-11, and the data shown in Fig. 9-13.

$$G = \frac{(475{,}723)}{(3600)(156.8/144)} = 121.4 \text{ lb/(sq ft • sec)}$$

From Fig. 9-10 (see Note †), the ratio of the new to the old resistance is:

$(r_o)_2 = (f)\,(r_o)_1$

Use base values from Fig. 9-11 for $(r_o)_1$ conditions.

$$f = \left(\frac{\mu_2}{\mu_1}\right)^{0.27}\left(\frac{k_1}{k_2}\right)^{0.67}\left(\frac{Cp_1}{Cp_2}\right)^{0.33}\left(\frac{G_1}{G_2}\right)^{0.6}\left(\frac{D_{o2}}{D_{o1}}\right)^{0.4}$$

$$= \left(\frac{0.34}{0.549}\right)^{0.27}\left(\frac{0.0762}{0.077}\right)^{0.67}\left(\frac{0.577}{0.542}\right)^{0.33}\left(\frac{132.4}{121.4}\right)^{0.6}\left(\frac{0.75}{0.625}\right)^{0.4}$$

$$= 0.998$$

$(r_o)_2 = (f)(r_o)_1 = (0.998)(0.0028) = 0.0028$ (hr • sq ft • °F)/Btu

From Fig. 9-10 (see Note †), the ratio of the new to the old pressure drop is:

$(\Delta P_o)_2 = (f)\,(\Delta P_o)_1$

Use base values from Fig. 9-11 for $(\Delta P_o)_1$ conditions. Obtain tube rows crossed between baffle window centroids from Fig. 9-13.

$RC_2 = 23$ ($RC_1 = 10$ per note on Fig. 9-11)

Obtain the number of crossflow spaces, which is one more than the number of baffles, from Fig. 9-12.

$SP_2 = 19$ [$SP_1 = 1$ per note on Fig. 9-11 since $(\Delta P_o)_1$ is for one baffle space.]

$$f = \left(\frac{\mu_2}{\mu_1}\right)^{0.15}\left(\frac{G_2}{G_1}\right)^{1.85}\left(\frac{\rho_1}{\rho_2}\right)\left(\frac{D_{o1}}{D_{o2}}\right)^{0.15}\left(\frac{SP_2}{SP_1}\right)\left(\frac{RC_2}{RC_1}\right)$$

$$= \left(\frac{0.34}{0.549}\right)^{0.15}\left(\frac{121.4}{132.4}\right)^{1.85}\left(\frac{46.8}{41.2}\right)\left(\frac{0.625}{0.750}\right)^{0.15}\left(\frac{19}{1}\right)\left(\frac{23}{10}\right)$$

$$= 38.3$$

$(\Delta P_o)_2 = (f)\,(\Delta P_o)_1 = (38.3)\,(0.25) = 9.58$ psi

6. Calculate the tube metal resistance.

$D_o = 0.75$ in.

$D_i = 0.62$ in.

$k_w = 30$ Btu/[(hr • sq ft • °F)/ft] , from Fig. 9-8.

$$r_w = \frac{D_o}{24\,k_w}\left[\ln\left(\frac{D_o}{D_i}\right)\right] = \frac{0.75}{(24)(30)}\left[\ln\left(\frac{0.75}{0.62}\right)\right]$$

$$= 0.0002 \text{ (hr • sq ft • °F)/Btu}$$

7. Calculate the overall heat transfer coefficient.

$$\Sigma r = r_i\left(\frac{A_o}{A_i}\right) + r_o + r_w + r_{fo} + r_{fi}\left(\frac{A_o}{A_i}\right)$$

$$= 0.0032\left(\frac{0.1963}{0.1623}\right) + 0.0028 + 0.0002 + 0.0020$$

$$+ 0.001\left(\frac{0.1963}{0.1623}\right) = 0.0101$$

$$U = \frac{1}{\Sigma r} = \frac{1}{0.0101} = 99.0 \text{ Btu/(hr • sq ft • °F)}$$

8. Compare the required heat transfer coefficient calculated in step 3 to the value calculated in step 7. (Available U = 99.0; required U = 100.6) The available value is 1.6% less than the required value, and the calculated pressure drops are less than the pressure drops allowed in Fig. 9-12. Therefore, by these calculations, the unit will perform adequately.

CONDENSERS

The purpose of a condenser is to change a fluid stream from the vapor state to the liquid state by removing the heat of vaporization. The fluid stream may be a pure component or a mixture of components. Condensation may occur on the shell side or the tube side of an exchanger oriented vertically or horizontally.

Condensing the overhead vapors of a distillation column is an example of condensing a mixed vapor stream. A vertical exchanger flanged directly to the top of the column might be used. The condensed liquid drains back into the column countercurrent to the vapor entering the condenser. The major concerns in designing this type exchanger are keeping the vapor velocity sufficiently low to prevent flooding the exchanger and evaluating an appropriate temperature profile at the condensing surface to determine an effective temperature difference. The technical literature addresses criteria for flooding determination[1] and special flow characteristics of falling liquid films. A useful estimate for determining an effective temperature difference can be made by assuming an isothermal condensate film at the saturation temperature of the last condensate formed. If the condensing temperature range exceeds 10°F, consulting a specialist is recommended for a more rigorous calculation procedure.

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FIG. 9-12

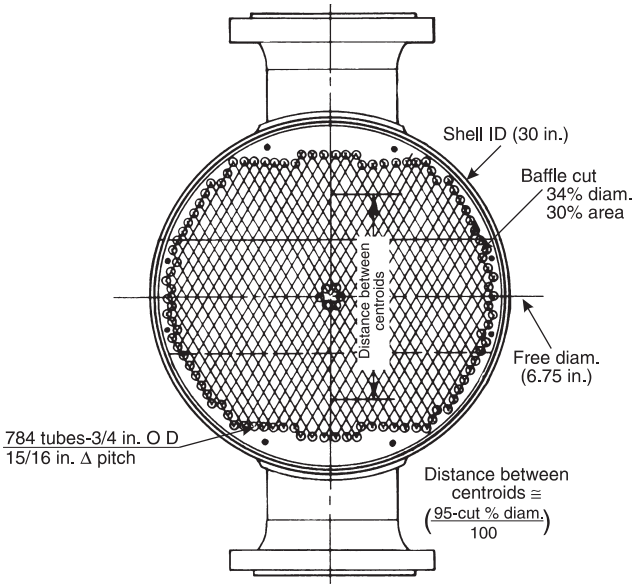
Heat Exchanger Specification Sheet

					Job No.	
Customer					Reference No.	
Address					Proposal No.	
Plant Location					Date	Rev.
Service of Unit	Lean Oil to Rich Oil Exchanger				Item No.	
Size	30-360 Type NEN (Hor/Vert)				Connected In	Parallel Series
Surf./Unit (Gross/Eff.)	4525 Sq Ft:	Shells/Unit	One	Surf/Shell (Gross/Eff.)	4525	SqFt
PERFORMANCE OF ONE UNIT						
Fluid Allocation		Shell Side		Tube Side		
Fluid Name		Lean Oil		Rich Oil		
Fluid Quantity, Total	Lb/Hr	475,723		650,860		
Vapor (In/Out)						
Liquid		475,723	475,723	650,860	650,860	
Steam						
Water						
Noncondensable						
Temperature (In/Out)	°F	197	100	60	124	
Density (#/ft3) AVG.		41.2		38.3		
Viscosity, Liquid (AVG.)		0.34		0.21		
Molecular Weight, Vapor						
Molecular Weight, Noncondensable						
Specific Heat (AVG.)	Btu/Lb °F	0.542		0.600		
Thermal Conductivity	Btu Ft / Hr Sq Ft °F	0.077 (AVG.)		0.078 (AVG.)		
Latent Heat	Btu/Lb @ °F					
Inlet Pressure	Psia	124.7		449.7		
Velocity	Ft/S					
Pressure Drop, Allow./Calc.	PSI	12 / 9.58		10 / 0.51		
Fouling Resistance (Min.)		0.0020		0.0010		
Heat Exchanged	25,000,000	Btu/Hr: MTD (Corrected)	54.9	°F		
Transfer Rate, Service	100.6	Clean	148.4	Btu/Hr Sq Ft °F		
CONSTRUCTION OF ONE SHELL				Sketch (Bundle/Nozzle Orientation)		
	Shell Side	Tube Side				
Design/Test Pressure Psig	185 /Code	500 /Code				
Design Temperature °F	650	650				
No. Passes per Shell	One	One				
Corrosion Allowance In.	0.0625	0.0625				
Connections Size & Rating In	10"	14"				
Out	10"	14"				
Intermediate						
Tube No. 784 OD 0.75 In.; Thk (Min/Avg) 0.065 In.; Length 30 Ft; Pitch 0.9375 In. X				◁ 30 △ 60 □ 90 ◇ 45		
Tube Type SA-214 (welded steel)		Material				
Shell SA-516-70 ID 30 OD In.		Shell Cover		(Integ.) (Remov.)		
Channel or Bonnet (2) SA-516-70 (Int.)		Channel Cover (2) SA-516-70 (Rem.)				
Tubesheet-Stationary (2) SA-516-70		Tubesheet -Floating				
Floating Head Cover		Impingement Protection				
Baffles-Cross SA-283-C Type Segmented		% Cut (Diam/Area) 34 In.		Spacing: c/c 18-5/8		
Baffles-Long		Seal Type				
Supports-Tube SA-283-C U-Bend		Type				
Bypass Seal Arrangement Spacers		Tube-Tubesheet Joint				
Expansion Joint NONE		Type				
ρv^2-Inlet Nozzle 1717	Bundle Entrance 2220	Bundle Exit 1210				
Gaskets-Shell Side		Tube Side				
-Floating Head						
Code Requirements ASME Sec. VIII Div 1 (with Stamp)		TEMA Class "C"				
Weight/Shell 20,600	Filled with Water 29,900	Bundle		Lb.		
Remarks						

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FIG. 9-13

Heat Exchanger Detail Design Results



Leakage – use TEMA tolerances		
tube hole = 576 x 0.75"π x 1/64"	=	21.2 sq in
shell crack = 30"π x 0.175" x 0.60	=	9.9 sq in
		31.1 sq in
Window flow area:		
window area = (0.237) $(30")^2$	=	213.3 sq in
tubes in window = (208) (0.785) $(0.75")^2$	=	91.8 sq in
		121.5 sq in
		leak 31.1 sq in
		152.6 sq in
Cross flow area:		
Free area = 6.75" x 18 5/8"	=	125.7 sq in
Leakage	=	31.1 sq in
Net cross flow area	=	156.8 sq in
BP = baffle pitch = 18 5/8"		
FD = free diameter = 6.75"		
Note: window area ≅ cross flow area		

The condensing of a pure component occurs at a constant temperature equal to the saturation temperature of the incoming vapor stream. Frequently a vapor enters a condenser superheated and must have the sensible heat removed from the vapor before condensation can occur. If the condensing surface temperature is greater than the incoming vapor saturation temperature, the superheat in the vapor is transferred to the cold surface by a sensible heat transfer mechanism ("drywall" condition). If the condensing surface temperature is less than the saturation temperature of the incoming vapor, a condensate film will be formed on the cold surface. The sensible heat is removed from the vapor at the condensate-vapor interface by vaporizing (flashing) condensate so that the heat of vaporization is equal to the sensible heat removed from the vapor. Under this "wet wall" condition, the effective temperature of the vapor is the saturation temperature, and the effective heat transfer mechanism is condensation. The determination of the point in the desuperheating zone of a condenser where "drywall" conditions cease and "wet wall" conditions begin is a trial and error procedure. A method frequently employed to give a safe approximation of the required surface is to use the condensing coefficient and the CMTD based on the vapor saturation temperature to calculate the surface required for both the desuperheating zone and the condensing zone.

The following Example 9-2 will illustrate the use of the heat release curve to calculate the surface required and the LMTD for each zone in a condenser for a pure component application.

Example 9-2 — A propane refrigerant condenser is required to condense the vapor stream using the heat release curve as shown in Fig. 9-14. This stream enters the condenser superheated and leaves the condenser as a subcooled liquid. Assume that a single-tube pass, single-shell pass, counterflow exchanger is used so that LMTD correction factors do not apply. Note that the propane is on the shell side. The overall heat transfer coefficients for each zone are as follows:

Desuperheating: (180°F to 108°F)

$$U_v = 69.9 \text{ Btu/(hr • sq ft • °F)}$$

$$h_v = 111.1 \text{ Btu/(hr • sq ft • °F)}$$

Condensing: (108°F to 108°F)

$$U_c = 140.0 \text{ Btu/(hr • sq ft • °F)}$$

Subcooling: (108°F to 95°F)

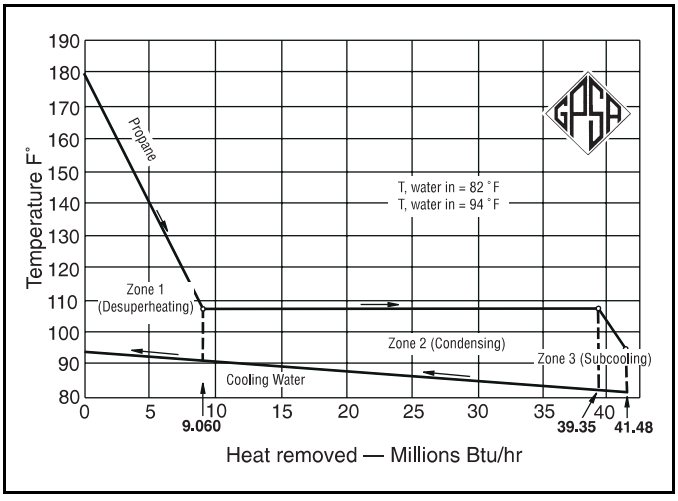
$$U_L = 114.5 \text{ Btu/(hr • sq ft • °F)}$$

Solution Steps

1. Calculate the surface temperature (outside wall) on the vapor side at the refrigerant stream inlet using the following equation:[2]

FIG. 9-14

Propane Condensing Curve



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$$T_{wo} = T_v - [U_v (T_v - T_C)/h_v]$$

2. If the surface temperature calculated in step 1 is greater than the vapor saturation temperature, calculate the amount of desuperheating that will be done by a sensible heat transfer mechanism. If the surface temperature is less than the vapor saturation temperature, assume that the desuperheating duty will be done by a condensing heat transfer mechanism.
3. Obtain the duty for the appropriate temperature ranges from Fig. 9-14.
4. Solve the equation Q = UA (LMTD) for the required surface area in each zone. The sum of these areas is the surface required for the exchanger.

$$T_{wo} = 180 - \left[\frac{69.9\,(180 - 94)}{111.1}\right] = 125.90°F$$

The surface temperature at the vapor inlet is greater than the saturation temperature, therefore, "drywall" desuperheating will take place initially. By trial and error, calculate the duty required when the assumed bulk vapor temperature results in a surface temperature less than the saturation temperature, thereby marking the transition from "drywall" to "wet wall" desuperheating. Assume the vapor bulk temperature is 133°F.

$$Q_v = 9{,}060{,}000 \text{ Btu/hr from Fig. 9-14.}$$

$$Q_1 = \frac{(180 - 133)}{(180 - 108)}\,(9{,}060{,}000)$$

$$= 5{,}910{,}000 \text{ Btu/hr} \quad \text{(Heat removed from vapor between 180°F and 133°F)}$$

$$T_C = 94 - \left[\frac{5{,}910{,}000}{41{,}480{,}000}\,(94 - 82)\right] = 92.3°F$$

(T_C is the water temperature at $Q_v - Q_1$, see Fig. 9-14)

$$T_{wo} = 133 - \left[\frac{69.9}{111.1}\,(133 - 92.3)\right] = 107.4°F$$

Include the remainder of the desuperheating duty in the condensing zone. (Zone 2.)

ZONE 1

$$Q = Q_1 = 5{,}910{,}000 \text{ Btu/hr from above.}$$

$$\begin{array}{rcl} 180 & \rightarrow & 133.0 \\ \underline{94} & \leftarrow & \underline{92.3} \\ 86 & & 40.7 \end{array}$$

$$LMTD = \frac{86 - 40.7}{\ln(86/40.7)} = 60.6°F$$

$$A = \frac{5{,}910{,}000}{(69.9)\,(60.6)} = 1{,}395 \text{ ft}^2$$

ZONE 2

$$Q = Q_v - Q_1 + Q_c \quad (Q_c \text{ from Fig. 9-14})$$

$$= 9{,}060{,}000 - 5{,}910{,}000 + 30{,}300{,}000$$

$$= 33{,}450{,}000 \text{ Btu/hr}$$

$$T_C = 94 - \frac{(5{,}910{,}000 + 33{,}450{,}000)}{41{,}480{,}000}\,(94 - 82) = 82.6°F$$

(NOTE: You do **not** use 133°F as the inlet temperature to this zone.)

$$\begin{array}{rcl} 108.0 & \rightarrow & 108.0 \\ \underline{92.3} & \leftarrow & \underline{82.6} \\ 15.7 & & 25.4 \end{array}$$

$$LMTD = \frac{25.4 - 15.7}{\ln(25.4/15.7)} = 20.2°F$$

$$A = \frac{33{,}450{,}000}{(140.0)\,(20.2)} = 11{,}830 \text{ ft}^2$$

ZONE 3

$$Q = Q_L = 2{,}120{,}000 \text{ Btu/hr from Fig. 9-14}$$

$$\begin{array}{rcl} 108.0 & \rightarrow & 95 \\ \underline{82.6} & \leftarrow & \underline{82} \\ 25.4 & & 13 \end{array}$$

$$LMTD = \frac{25.4 - 13}{\ln(25.4/13)} = 18.5°F$$

$$A = \frac{2{,}120{,}000}{(114.5)\,(18.5)} = 1000 \text{ ft}^2$$

Total Area

$$\Sigma A = 1{,}395 + 11{,}830 + 1{,}000 = 14{,}225 \text{ ft}^2$$

The condensing of a vapor mixture requires additional considerations to those outlined above for pure components. If the mixture condenses over a narrow temperature range, the pure component analysis is applicable. If, on the other hand, the mixture condenses over a wide temperature range, the problem is complicated by the relationship between heat and mass transfer rates. This is also true in the case of condensing vapors in the presence of noncondensables.

The calculation of the condensing coefficient involves physical and thermodynamic properties of the condensing fluid, the two-phase flow regime involved, and the heat exchanger type, geometry, and orientation. The detailed design of such a condenser should be left to a specialist.

REBOILERS AND VAPORIZERS

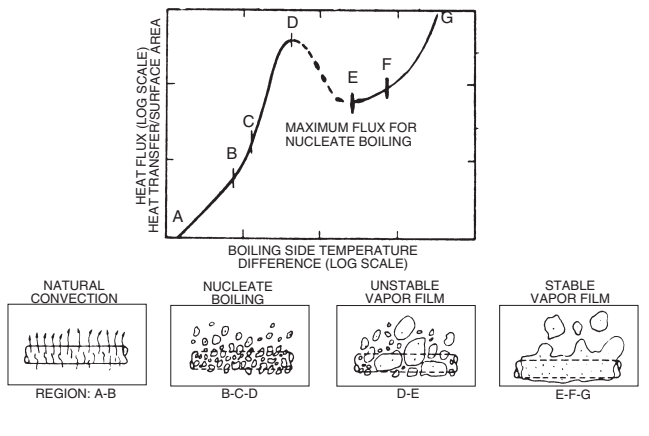
The "Pool Boiling Curve"

Boiling, as applied to reboilers and vaporizers, can be a complicated relationship of heat and mass transfer. The simplest form is pool boiling where hot fluid inside a tube causes vapor generation on the outside surface of the tube from a pool of liquid. The heat exchange capability of a fluid in pool boiling is determined by empirically correlated data and represented as a "Pool Boiling Curve" that is specific for a composition and a pressure. Fig. 9-15 is a typical representation of single tube data. The shape and regions of the curve in Fig. 9-15 are characteristic of pure component fluids and most mixtures dominated by a single component. Mixtures that boil over a wide temperature range or are being vaporized near their critical point may not fit the characteristic shape or regions. With appropriate curve data and heating medium temperature level, a well designed 'pool' reboiler will have heat fluxes in the region analogous to the B-C-D region of Fig. 9-15. Exchangers designed in the E-F-G regions (high surface temperature) may foul faster from thermally activated chemical reactions than would a wet wall design. In the flux inversion region (D-E)

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FIG. 9-15

A Typical Pool Boiling Curve



erratic, if not reversed, control behavior will occur. Fig. 9-16 summarizes some typical heat flux ranges for some common fluids.

Effective Temperature Difference

Boiling, like condensing, may not occur at a constant heat transfer coefficient. The basic definition of Log Mean Temperature Difference may not apply. The effective temperature difference, often called the True Mean Temperature Difference (TMTD), must be determined based on installation and fluid conditions at the reboiler. Elevation of a fluid's bubble point by static head being added to a column's sump pressure means a subcooled liquid must be heated to a bubble point higher than the bottom tray liquid temperature. With countercurrent or co-current flow arrangements, an incrementally evaluated Weighted Temperature Difference (WTD) is appropriate. However, in crossflow and pool boiling, a different analysis must apply. In pool boiling, a temperature rise is not readily predictable along a particular geometric flowpath. For design purposes, the TMTD is often taken as if the pool were isothermal at the vapor outlet temperature.

FIG. 9-16

Typical Overall Boiling Heat Flux Ranges

Heat Medium	Boiling Fluid	Heat Flux Range, Btu/(hr • sq ft)
Hot Oil	C_1-C_2 HC	7,000 - 8,000
	C_3-C_5 HC	6,500 - 14,000
	Rich Oil	2,500 - 3,500
	Amines	3,500 - 5,500
HC Gas	C_1-C_2 HC	2,000 - 4,500
Steam	C_1-C_2 HC	8,000 - 13,000
	C_3-C_5 HC	10,000 - 15,000
	Rich Oil	4,000 - 6,000
	Amines	4,500 - 6,500

Hydraulic Effects

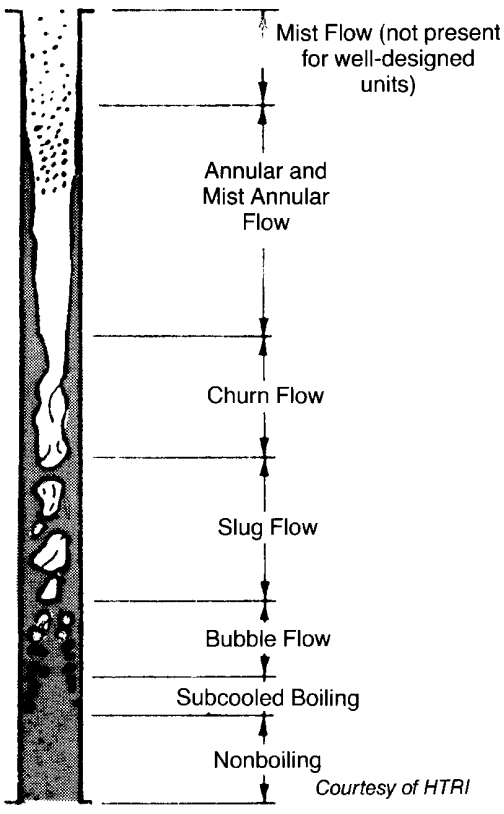
When the geometric flowpath of a boiling fluid is well defined (all boiling except pool boiling), the effects of liquid and vapor velocities are part of a design or operating analysis. Liquid and vapor co-exist in 'regimes' as illustrated in Fig. 9-17. Typically, these regimes progress to termination between Slug Flow and Mist-Annular Flow in a reboiler. In these regimes the heat transfer coefficient has two important contributing parts, convective boiling and nucleate boiling. When liquid is recirculated to a reboiler, the heat transfer coefficient is maximized and such limiting conditions as Mist Flow, Vapor Film Boiling, and two-phase momentum transfer instability may be avoided. The latter form of instability occurs when liquid feed to a reboiler has pulsations (intermittent liquid flow reversal) generated by an instantaneous vapor acceleration pressure drop that temporarily causes the total pressure drop to exceed available static head. The heat medium temperature affects the same limiting conditions and may be the controlling variable when recirculation is not possible. Design and operating analysis requires a study of hydraulics, heat medium temperature, and exchanger geometry for a particular fluid to define valid limitations on a reboiler. Analytical methods are available in the technical literature noted in the bibliography.

Types of Reboilers

Kettle — Kettle reboilers are commonly applied when a wide range of process operations (high turndown capability), large heat exchange surface, or high vapor quality is required. Installations include column bottom reboilers, side reboilers,

FIG. 9-17

Two-Phase Flow Regimes in Vertical Tubes

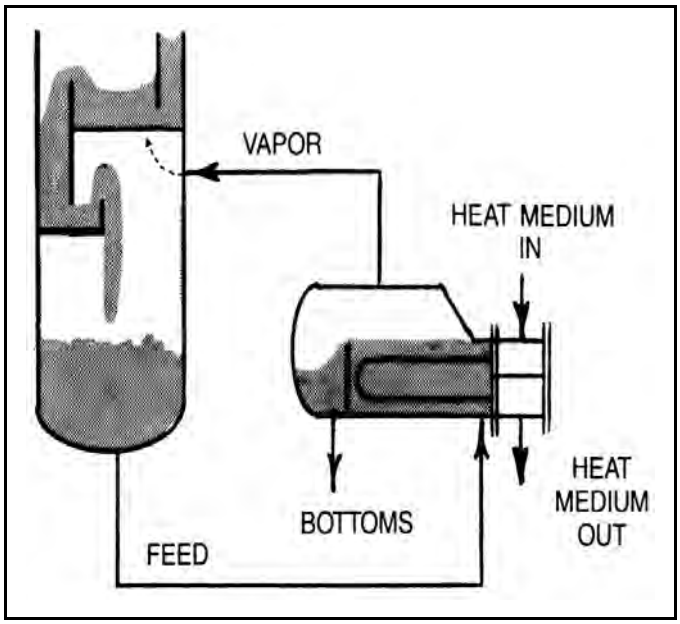


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or vaporizers. Fig. 9-18 shows a typical kettle. Kettles are generally more costly than other reboiler types due to shell size, surge volume size, and uncertainty in the TMTD. Without TMTD or fouling problems, a column-internal (stab-in) reboiler would be suitable if the required surface is relatively small.

FIG. 9-18

Kettle Reboiler on Column Bottoms



Recirculating thermosyphon — Recirculating thermosyphon reboilers are applicable when process operations are consistently near design rates. Typically, these are vertical tube side boiling, like Fig. 9-19, or for large surface requirements, horizontal shell side boiling. Installation requires a fixed static head, such as a partitioned column sump or a head drum, for recirculation. Recirculating thermosyphon reboilers are generally the least costly of reboiler types (other than column-internal type) due to maximized heat transfer, accurate TMTD, and relatively low fouling tendencies (due to higher velocities).

"Once-through" — Once-through reboilers are applicable when the feed is available without the capability for recirculation. These boilers may be called "thermosyphons" when taking a column tray liquid as feed such as shown in Fig. 9-20. Once-through reboilers and vaporizers have the lowest fluid residence time on the hot surface and have a fixed downstream pressure which fixes the inlet pressure to the reboiler (externally fixed head is not required). However, they have the narrowest range of stable hydraulics and heat medium temperatures in the wet wall regions of boiling due to the fixed flowrate. Substantial process judgment and analytical support are required for satisfactory performance. Once-through reboilers can be in either the horizontal or vertical position and have been designed for either shellside or tubeside boiling.

"Pump-through" — Pump-through or pump-around reboilers are applicable when handling viscous liquid or particulate-laden liquid, and when liquid heating by pressure suppressed vaporization are desirable. Any arrangement of shell side or tube side boiling, vertical or horizontal may be used, but Fig. 9-21 is a typical arrangement. Pump-through reboilers may or may not include recirculated liquid, but usually do. Suppressed vaporization operation requires a throttling valve in the outlet line of the reboiler to generate vapor at the downstream fixed pressure.

Type Selection — Reboiler type selection generally follows the guidelines of Fig. 9-22.

FIG. 9-19

Recirculating Thermosyphon Reboiler on Column Bottoms

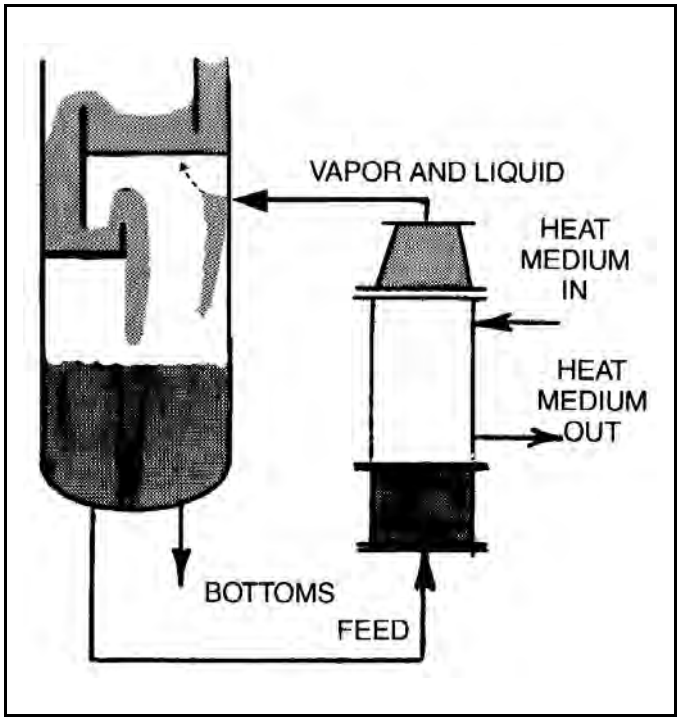
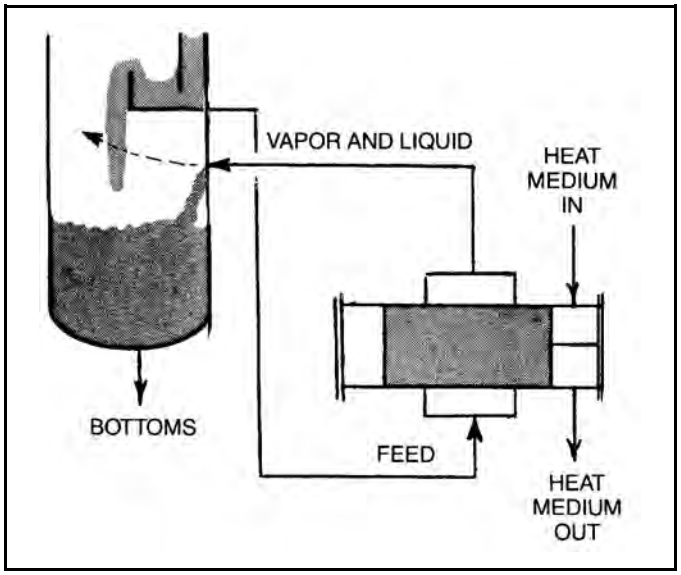


FIG. 9-20

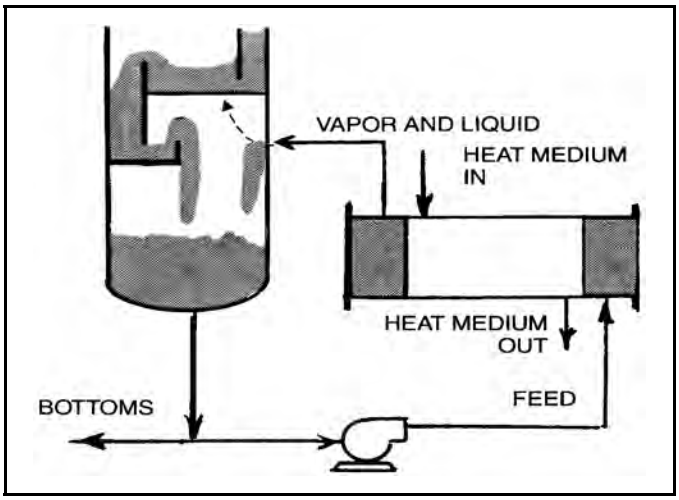
Once-Through Reboiler with Bottom Tray Feed



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FIG. 9-21

Pump Through Reboiler on Column Bottoms



SELECTION OF EXCHANGER COMPONENTS

Industry Standards

Shell and tube heat exchanger technology for gas, chemical, and petroleum processing plants has developed a broad basis of common understanding through the "Standards of Tubular Exchangers Manufacturers Association" (TEMA). These "TEMA Standards" provide nomenclature, dimensional tolerances, manufacturer's and purchaser's responsibilities, general installation and operating guidelines, and specific design and fabrication practices.

The design and fabrication practices of TEMA are in three classifications, called Class "R," "C," or "B." Class "R" is applied to services with severe operating and maintenance characteristics. Class "C" is for the least severe characteristics. Class "B" is for chemical process applications between Classes "R" and "C." All classes are intended to be limited to ASME Code, Section VIII, Div. 1, cylinder wall thicknesses of less than about 2", and stud diameters of less than about 3"; though thicker components can be applied by the design practices specified.

TEMA Standards provide a "Recommended Good Practice" for the designer's consideration in areas outside of the limits of the specified standards. Guidance and references are noted for seismic design, large diameter exchangers, tube vibration, tube-to-tubesheet stress analysis, nozzle loading analysis, and numerous other design-limiting features.

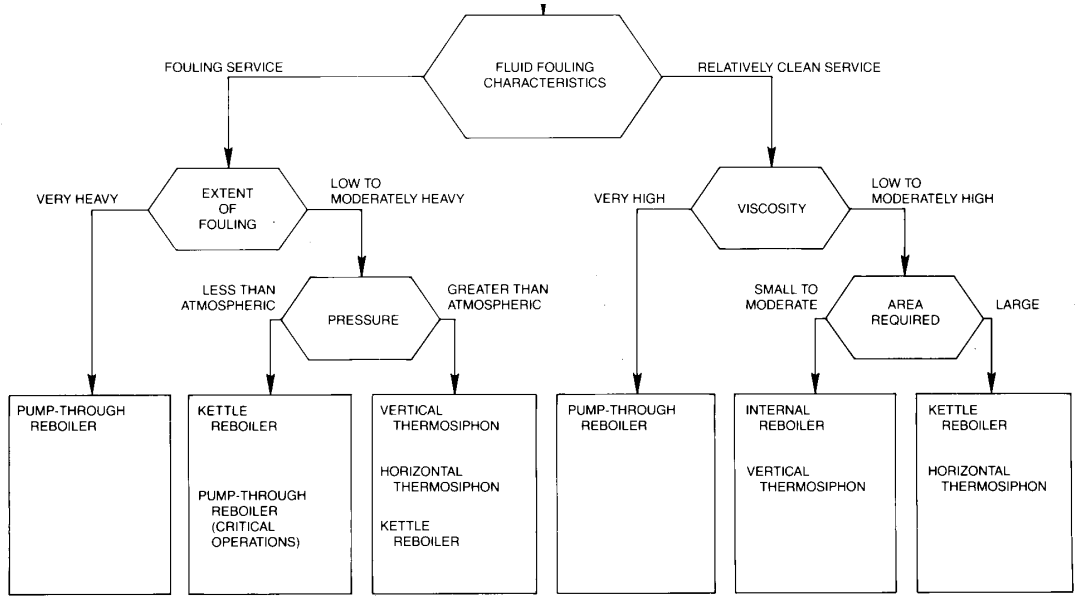
Detailed understanding of shell and tube exchangers for use in the process industry requires an understanding of the TEMA Standards. Other industry standards as may be offered by ASME, API, or ANSI can be applied in a particular situation with or without TEMA Standards. The purchase order and specification sheet for a particular service will normally identify the applicable industry standards.

Nomenclature

Fig. 9-23 summarizes the major shell-and-tube exchanger components other than tubes and baffles. The letters are used for a standard nomenclature in the industry. A three-letter type designation in the order of front head type, shell type, and rear head type is used. For example, an AJS would have a front head that is removable with a removable cover, a shell that is

FIG. 9-22

Reboiler Selection Chart

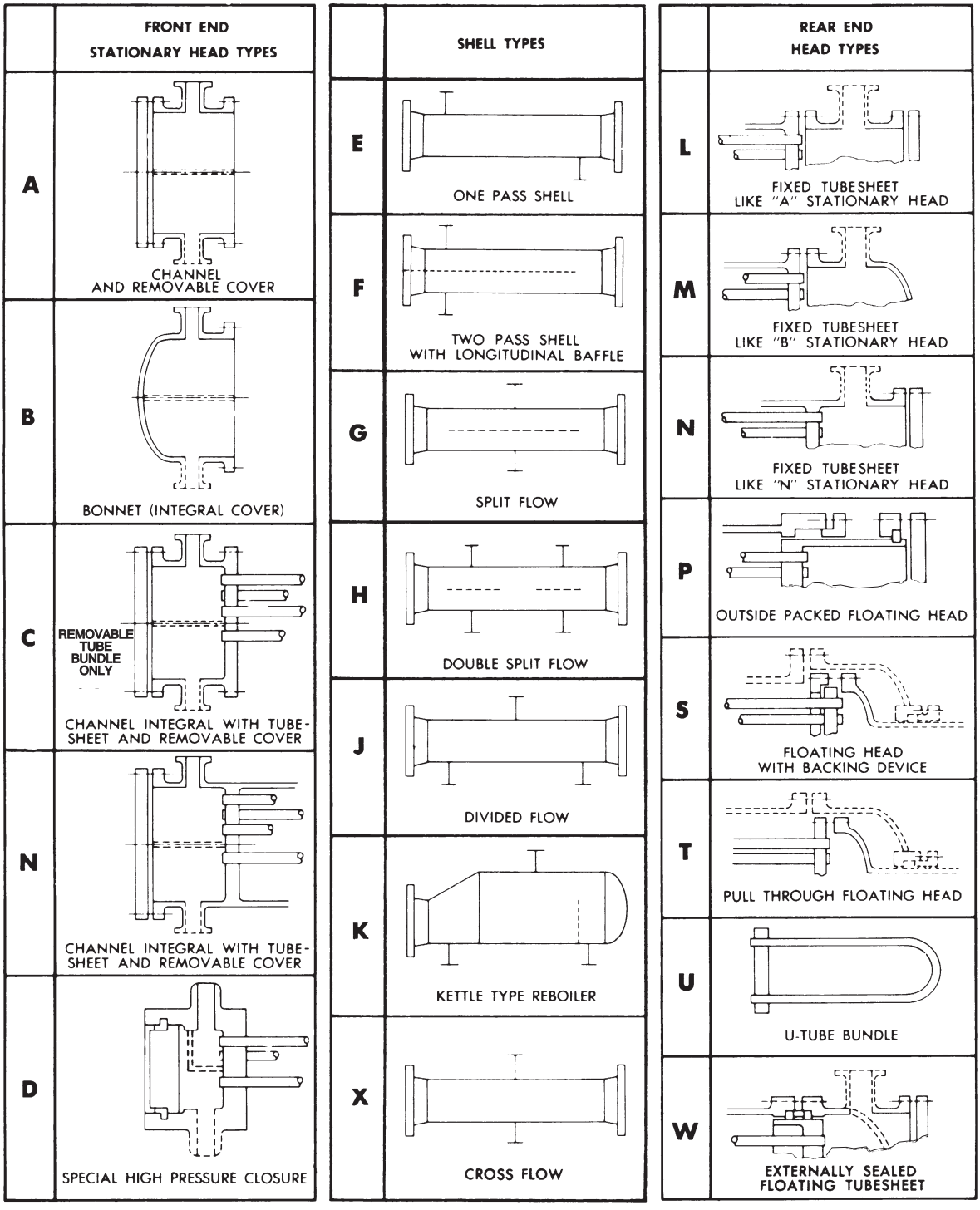


Courtesy HTRI

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FIG. 9-23

Shell and Tube Exchanger Nomenclature



Courtesy of TEMA

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arranged for divided flow, and a rear floating head with a backing device (usually a split-ring). Factors to consider in selecting a shell and tube exchanger type are summarized in Fig. 9-24.

Tube Wall Determination

The required tube wall thickness is determined from the ASME Code, Section VIII, Division 1 for cylinders under internal or external pressure. If U-tubes are used, the thinning of the tube wall in the bends must be considered. A minimum wall tube whose thickness is equal to or greater than the calculated thickness may be used, or an average wall tube whose minimum thickness is equal to or greater than the calculated thickness may be used. It is satisfactory to use an average wall tube that is one BWG heavier than the required minimum wall thickness; however, it is not always possible to substitute a minimum wall tube that is one BWG thinner than a specified average wall thickness tube. If the calculated wall thickness is less than the value recommended by TEMA, the TEMA values are used. Fig. 9-25 summarizes standard tube data.

Shell Size and Tube Count Estimation

The tube count in a given shell diameter varies with the tube diameter, tube spacing and layout (pitch), type of tube bundle, number of tube passes, and the shell side entrance and exit area allowed. After selecting an appropriate tube outside diameter and tube length, the number of tubes required to result in a given heat transfer surface can be calculated using the external square foot/foot data from Fig. 9-25.

Fig. 9-26 is a plot of tube count vs. diameter for four different triangular tube pitches most commonly used in shell and tube exchangers. Entering these curves with the required tube count will give a diameter which can be corrected for the various factors noted to determine the actual shell diameter required.

To correct for square pitch, multiply the shell inside diameter from Fig. 9-26 by 1.075. No correction factor is needed for any other pitch. To allow for entrance or exit areas, multiply shell inside diameter from Fig. 9-26 by 1.02 for each inlet or outlet area to be used. Fig. 9-27 is a table of factors to correct inside shell diameter for pass arrangement.

Fig. 9-28 is a table of adders to correct for type of construction.

Example 9-3 — Determine the shell diameter for 320 tubes, 1" OD spaced on a 1-1/4" square pitch layout, four-pass tubes, in a split ring type floating head shell and tube exchanger, with inlet flow area allowed.

Solution Steps

1. From the top curve of Fig. 9-26 read 25" corresponding to 320 tubes for the given tube spacing and pitch.
2. Correct for square pitch by multiplying by 1.075.
3. Using Fig. 9-27 correct for four pass by multiplying by 1.05.
4. Correct for inlet flow area by multiplying by 1.02.

   Accumulative multiplier is 1.075 x 1.05 x 1.02 = 1.15.

   Partially corrected diameter = 25" x 1.15 = 28.75".
5. From Fig. 9-28, correct for split ring floating head by adding 1" + 28.75 = 29.75".

So use a 30" ID shell for this tube count and configuration.

Enhanced Surface Tubing

Heat exchanger applications in which one of the fluids has a high heat transfer coefficient relative to the other fluid can benefit (either from lower first cost of a new exchanger or increased capacity in an existing unit) by use of specially enhanced tube surfaces on the side with the low coefficient. One commonly used tube is a “low finned” tube which has extruded fins on the outside of the tube and the diameter outside the fins is no greater than the outside diameter of the plain ends

FIG. 9-24

Shell and Tube Exchanger Selection Guide (Cost Increases from Left to Right)

Type of Design	"U" Tube	Fixed Tubesheet	Floating Head Outside Packed	Floating Head Split Backing Ring	Floating Head Pull-Through Bundle
Provision for differential expansion	individual tubes free to expand	expansion joint in shell	floating head	floating head	floating head
Removeable bundle	yes	no	yes	yes	yes
Replacement bundle possible	yes	not practical	yes	yes	yes
Individual tubes replaceable	only those in outside row	yes	yes	yes	yes
Tube interiors cleanable	difficult to do mechanically, can do chemically	yes, mechanically or chemically	yes, mechanically or chemically	yes, mechanically or chemically	yes, mechanically or chemically
Tube exteriors with triangular pitch cleanable	chemically only	chemically only	chemically only	chemically only	chemically only
Tube exteriors with square pitch cleanable	yes, mechanically or chemically	chemically only	yes, mechanically or chemically	yes, mechanically or chemically	yes, mechanically or chemically
Number of tube passes	any practical even number possible	normally no limitations	normally no limitations	normally no limitations	normally no limitations
Internal gaskets eliminated	yes	yes	yes	no	no

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FIG. 9-25

Characteristics of Tubing

Tube O.D. Inches	B.W.G. Gauge	Thickness Inches	Internal Area Sq. Inch	Sq Ft External Surface Per Foot Length	Sq Ft Internal Surface Per Foot Length	Weight Per Foot Length Steel Lbs*	Tube I.D. Inches	Moment of Inertia (Inches^4)	Section Modulus (Inches^3)	Radius of Gyration (Inches)	Constant C**	O.D. / I.D.	Transverse Metal Area Sq. Inch
1/4	22	0.028	0.0296	0.0654	0.0508	0.066	0.194	0.00012	0.00098	0.0791	46	1.289	0.0195
1/4	24	0.022	0.0333	0.0654	0.0539	0.054	0.206	0.00010	0.00083	0.0810	52	1.214	0.0158
1/4	26	0.018	0.0360	0.0654	0.0560	0.045	0.214	0.00009	0.00071	0.0823	56	1.168	0.0131
1/4	27	0.016	0.0373	0.0654	0.0571	0.040	0.218	0.00008	0.00065	0.0829	58	1.147	0.0118
3/8	18	0.049	0.0603	0.0982	0.0725	0.171	0.277	0.00068	0.0036	0.1166	94	1.354	0.0502
3/8	20	0.035	0.0731	0.0982	0.0798	0.127	0.305	0.00055	0.0029	0.1208	114	1.230	0.0374
3/8	22	0.028	0.0799	0.0982	0.0835	0.104	0.319	0.00046	0.0025	0.1231	125	1.176	0.0305
3/8	24	0.022	0.0860	0.0982	0.0867	0.083	0.331	0.00038	0.0020	0.1250	134	1.133	0.0244
1/2	16	0.065	0.1075	0.1309	0.0969	0.302	0.370	0.0021	0.0086	0.1555	168	1.351	0.0888
1/2	18	0.049	0.1269	0.1309	0.1052	0.236	0.402	0.0018	0.0071	0.1604	198	1.244	0.0694
1/2	20	0.035	0.1452	0.1309	0.1126	0.174	0.430	0.0014	0.0056	0.1649	227	1.163	0.0511
1/2	22	0.028	0.1548	0.1309	0.1162	0.141	0.444	0.0012	0.0046	0.1672	241	1.126	0.0415
5/8	12	0.109	0.1301	0.1636	0.1066	0.601	0.407	0.0061	0.0197	0.1865	203	1.536	0.177
5/8	13	0.095	0.1486	0.1636	0.1139	0.538	0.435	0.0057	0.0183	0.1904	232	1.437	0.158
5/8	14	0.083	0.1655	0.1636	0.1202	0.481	0.459	0.0053	0.0170	0.1939	258	1.362	0.141
5/8	15	0.072	0.1817	0.1636	0.1259	0.426	0.481	0.0049	0.0156	0.1972	283	1.299	0.125
5/8	16	0.065	0.1924	0.1636	0.1296	0.389	0.495	0.0045	0.0145	0.1993	300	1.263	0.114
5/8	17	0.058	0.2035	0.1636	0.1333	0.352	0.509	0.0042	0.0134	0.2015	317	1.228	0.103
5/8	18	0.049	0.2181	0.1636	0.1380	0.302	0.527	0.0037	0.0119	0.2044	340	1.186	0.089
5/8	19	0.042	0.2299	0.1636	0.1416	0.262	0.541	0.0033	0.0105	0.2067	359	1.155	0.077
5/8	20	0.035	0.2419	0.1636	0.1453	0.221	0.555	0.0028	0.0091	0.2090	377	1.126	0.065
3/4	10	0.134	0.1825	0.1963	0.1262	0.833	0.482	0.0129	0.0344	0.2229	285	1.556	0.259
3/4	11	0.120	0.2043	0.1963	0.1335	0.809	0.510	0.0122	0.0326	0.2267	319	1.471	0.238
3/4	12	0.109	0.2223	0.1963	0.1393	0.747	0.532	0.0116	0.0309	0.2299	347	1.410	0.219
3/4	13	0.095	0.2463	0.1963	0.1466	0.665	0.560	0.0107	0.0285	0.2340	384	1.339	0.195
3/4	14	0.083	0.2679	0.1963	0.1529	0.592	0.584	0.0098	0.0262	0.2376	418	1.284	0.174
3/4	15	0.072	0.2884	0.1963	0.1587	0.522	0.606	0.0089	0.0238	0.2411	450	1.238	0.153
3/4	16	0.065	0.3019	0.1963	0.1623	0.476	0.620	0.0083	0.0221	0.2433	471	1.210	0.140
3/4	17	0.058	0.3157	0.1963	0.1660	0.429	0.634	0.0076	0.0203	0.2455	492	1.183	0.126
3/4	18	0.049	0.3339	0.1963	0.1707	0.367	0.652	0.0067	0.0178	0.2484	521	1.150	0.108
3/4	20	0.035	0.3632	0.1963	0.1780	0.268	0.680	0.0050	0.0134	0.2531	567	1.103	0.079
7/8	10	0.134	0.2894	0.2291	0.1589	1.062	0.607	0.0221	0.0505	0.2662	451	1.442	0.312
7/8	11	0.120	0.3167	0.2291	0.1662	0.969	0.635	0.0208	0.0475	0.2703	494	1.378	0.285
7/8	12	0.109	0.3390	0.2291	0.1720	0.893	0.657	0.0196	0.0449	02736	529	1.332	0.262
7/8	13	0.095	0.3685	0.2291	0.1793	0.792	0.685	0.0180	0.0411	0.2778	575	1.277	0.233
7/8	14	0.083	0.3948	0.2291	0.1856	0.703	0.709	0.0164	0.0374	0.2815	616	1.234	0.207
7/8	15	0.072	0.4197	0.2291	0.1914	0.618	0.731	0.0148	0.0337	0.2850	655	1.197	0.182
7/8	16	0.065	0.4359	0.2291	0.1950	0.563	0.745	0.0137	0.0312	0.2873	680	1.174	0.165
7/8	17	0.058	0.4525	0.2291	0.1987	0.507	0.759	0.0125	0.0285	0.2896	706	1.153	0.149
7/8	18	0.049	0.4742	0.2291	0.2034	0.433	0.777	0.0109	0.0249	0.2925	740	1.126	0.127
7/8	20	0.035	0.5090	0.2291	0.2107	0.314	0.805	0.0082	0.0187	0.2972	794	1.087	0.092
1	8	0.165	0.3526	0.2618	0.1754	1.473	0.670	0.0392	0.0784	0.3009	550	1.493	0.433
1	10	0.134	0.4208	0.2618	0.1916	1.241	0.732	0.0350	0.0700	0.3098	656	1.366	0.365
1	11	0.120	0.4536	0.2618	0.1990	1.129	0.760	0.0327	0.0654	0.3140	708	1.316	0.332
1	12	0.109	0.4803	0.2618	0.2047	1.038	0.782	0.0307	0.0615	0.3174	749	1.279	0.305
1	13	0.095	0.5153	0.2618	0.2121	0.919	0.810	0.0280	0.0559	0.3217	804	1.235	0.270
1	14	0.083	0.5463	0.2618	0.2183	0.814	0.834	0.0253	0.0507	0.3255	852	1.199	0.239
1	15	0.072	0.5755	0.2618	0.2241	0.714	0.856	0.0227	0.0455	0.3291	898	1.168	0.210
1	16	0.065	0.5945	0.2618	0.2278	0.650	0.870	0.0210	0.0419	0.3314	927	1.149	0.191
1	18	0.049	0.6390	0.2618	0.2361	0.498	0.902	0.0166	0.0332	0.3367	997	1.109	0.146
1	20	0.035	0.6793	0.2618	0.2435	0.361	0.930	0.0124	0.0247	0.3414	1060	1.075	0.106
1 1/4	7	0.180	0.6221	0.3272	0.2330	2.059	0.890	0.0890	0.1425	0.3836	970	1.404	0.605
1 1/4	8	0.165	0.6648	0.3272	0.2409	1.914	0.920	0.0847	0.1355	0.3880	1037	1.359	0.565
1 1/4	10	0.134	0.7574	0.3272	0.2571	1.599	0.982	0.0742	0.1187	0.3974	1182	1.273	0.470
1 1/4	11	0.120	0.8012	0.3272	0.2644	1.450	1.010	0.0688	0.1100	0.4018	1250	1.238	0.426
1 1/4	12	0.109	0.8365	0.3272	0.2702	1.330	1.032	0.0642	0.1027	0.4052	1305	1.211	0.391
1 1/4	13	0.095	0.8825	0.3272	0.2775	1.173	1.060	0.0579	0.0926	0.4097	1377	1.179	0.345
1 1/4	14	0.083	0.9229	0.3272	0.2838	1.036	1.084	0.0521	0.0833	0.4136	1440	1.153	0.304
1 1/4	16	0.065	0.9852	0.3272	0.2932	0.824	1.120	0.0426	0.0682	0.4196	1537	1.116	0.242
1 1/4	18	0.049	1.0423	0.3272	0.3016	0.629	1.152	0.0334	0.0534	0.4250	1626	1.085	0.185
1 1/4	20	0.035	1.0936	0.3272	0.3089	0.455	1.180	0.0247	0.0395	0.4297	1706	1.059	0.134
1 1/2	10	0.134	1.1921	0.3927	0.3225	1.957	1.232	0.1354	0.1806	0.4853	1860	1.218	0.575
1 1/2	12	0.109	1.2908	0.3927	0.3356	1.621	1.282	0.1159	0.1545	0.4933	2014	1.170	0.476
1 1/2	14	0.083	1.3977	0.3927	0.3492	1.257	1.334	0.0931	0.1241	0.5018	2180	1.124	0.369
1 1/2	16	0.065	1.4741	0.3927	0.3587	0.997	1.370	0.0756	0.1008	0.5079	2300	1.095	0.293
2	11	0.120	2.4328	0.5236	0.4608	2.412	1.760	0.3144	0.3144	0.6660	3795	1.136	0.709
2	12	0.109	2.4941	0.5236	0.4665	2.204	1.782	0.2904	0.2904	0.6697	3891	1.122	0.648
2	13	0.095	2.5730	0.5236	0.4739	1.935	1.810	0.2586	0.2586	0.6744	4014	1.105	0.569
2	14	0.083	2.6417	0.5236	0.4801	1.701	1.834	0.2300	0.2300	0.6784	4121	1.091	0.500

* Weights are based on low carbon steel with a density of 0.2833 lbs/cu in. For other metals multiply by the following factors:

Aluminum	0.35	Aluminum Bronze	1.04	Nickel	1.13
Titanium	0.58	Aluminum Brass	1.06	Nickel-Copper	1.12
A.I.S.I. 400 Series Stainless Steels	0.99	Nickel-Chrome-Iron	1.07	Copper and Cupro-Nickels	1.14
A.I.S.I. 300 Series Stainless Steels	1.02	Admiralty	1.09		

** Liquid Velocity $= \dfrac{\text{Lbs Per (Tube • Hour)}}{\text{(C) (Sp Gr of Liquid)}}$ in feet per sec (Sp Gr of Water at 60°F = 1.0)

Courtesy of TEMA

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FIG. 9-26

Tube Count vs. Diameter for Triangular Tube Pitch

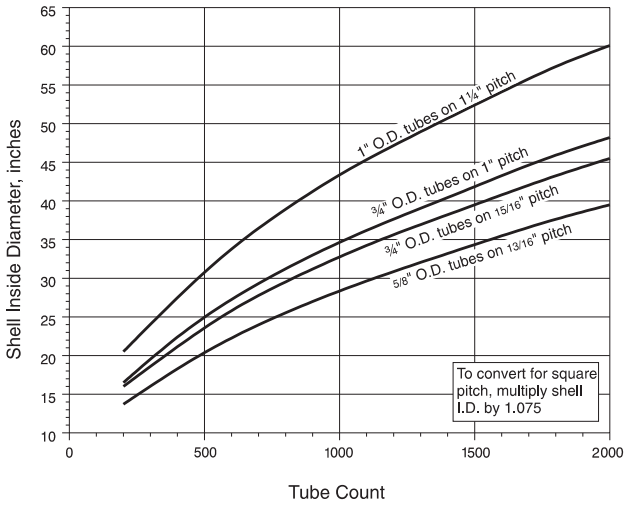


FIG. 9-27

Correction Factors for Number of Tube Passes

Shell Diameter, inches	Number of Tube Passes			
	Two	Four	Six	Eight
less than 12"	1.10	1.20	1.35	—
12" to 24"	1.03	1.08	1.12	1.25
25" to 41"	1.02	1.05	1.07	1.08
greater than 41"	1.01	1.03	1.04	1.06

FIG. 9-28

Adders to Shell Diameter

Shell Diameter, inches	Type of Construction		
	Fixed Tubesheet	Split Ring	Pull Through
less than 27"	None	5/8"	3-1/2"
27" to 41"	None	1"	4"
greater than 41"	None	1-5/8"	5"

so the exchanger can be assembled or retubed in the same way as a bare tube exchanger. The effect is to increase the heat transfer surface of the tube approximately 250% to result in a more compact exchanger for a given service compared to one using bare tubes. These tubes perform favorably in clean applications such as light hydrocarbon condensers where vapor velocity permits a condensate film to be distributed over more surface per tube. These tubes are available in metals commonly used in most heat exchangers.

High heat flux tubes with special coatings to create a porous surface are sometimes used where liquid velocities permit nucleate boiling to increase the heat flux per tube provided the porous surface remains exposed to the liquid.

For even more specialized considerations of fluid properties and operating requirements, a tube wall may be extruded at or near thickness to a variety of shapes. A convoluted spirally extruded tube wall offers a range for the hydraulic diameter that may be optimized for the fluids considered.

Other than low finned tubes, most enhanced surface tubes are limited to materials uniquely suited to the particular enhanced surfaces and special fabrication limitations. The limitations on application and availability as dictated in specific supplier's literature must be considered.

OPERATING CHARACTERISTICS

Inlet Gas Exchanger

The familiar feed-to-residue gas exchanger is characterized by a close temperature approach between the two streams over a long temperature range which requires countercurrent flow arrangement. For overall economy this service will have very long tubes and low pressure drops in an optimized design. Such design will include adequate protection from hydrate formation in the feed gas and a baffle arrangement suitable for low shellside pressure drop and no significant tube vibration.

In wet gas streams hydrate formation is normally prevented by spraying methanol or ethylene glycol on the face of the front tubesheet. Critical to the effectiveness of that injection is the spray coverage of the tube field and a tube side velocity sufficient to achieve annular (wet wall) flow in each tube as shown in Fig. 9-17.

To maintain countercurrent flow arrangement baffle variations may be considered to minimize shell side pressure drop. A variety of multisegmented baffles offer lower pressure drop per cross pass than the segmental type. Proprietary low pressure drop devices such as wire (or rod) web baffles may be appropriate if the loss in heat transfer is not significant. When tube vibration is a prime concern, a segmentally cut baffle arrangement with no tubes in the cut out window provides nonproprietary maximum tube support for a given pressure drop.

Tube Vibration

Tubes or tube bundles can be excited to sufficient movement to create noise, tube damage, and/or baffle damage. Some tube field geometries are particularly susceptible to acoustical resonance. Any tube has a natural frequency of vibration dependent on its supported span, size, and density. When velocity of a fluid induces cyclic forces approximating that natural frequency, vibration occurs. The first mode of vibration (lowest natural frequency) occurs at the half wave length equal to the supported span and is the usual case for analysis. However, higher modes of vibration are possible when multiple half wave lengths coincide with the supported span length. Since tube bundles have damping characteristics, damage may or may not occur at a particular mode of vibration. A substantial bibliography of analytical methods as well as calculation procedures for this subject are presented in the Recommended Good Practice section of TEMA standards.

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Evaluating Altered Performance

Exchanger performance will deviate when:

1. Process conditions are altered by feedstock, throughput, control/instrumentation, or mechanical failure of adjoining equipment.
2. Corrosion or foulant material achieves a limiting condition.
3. Internal leakage or mechanical failure has occurred in the exchanger.

Operating records and overall process analysis can address most problems except fouling, corrosion, internal leakage, and mechanical failure within the exchanger.

If a relief valve is overpressured on the low pressure side of an exchanger, it suggests interstream leakage or a near total flow restriction on the low pressure side. Substantial loss of pressure on the high pressure side confirms interstream leakage. The soundness of tubes, tubesheets, internally gasketed joints, and/or internal expansion joints must be tested and the failed components repaired, replaced, or plugged. A relief valve overpressuring on the high pressure side suggests a flow restriction downstream of the relief valve connection.

Flow restriction not accountable to operating changes in the process analysis is probably attributable to fouling debris somewhere in one or both stream systems. If such flow restriction occurs gradually (several days to several months), a systematic inspection with cleaning as needed is probably required. If such flow restriction occurs quickly (seconds to hours), mechanical failure or a process step-change probably occurred somewhere in the stream system. Only an available flow bypass around the exchanger can isolate and identify the flow restriction in the exchanger. A flow restriction anywhere in a stream system will alter an exchanger's heat transfer effect on both stream systems involved. The process analysis should indicate which stream has consequential limits and which stream is a problem source.

Perhaps the most difficult performance problem to isolate in operation is the discrimination between pass partition leakage and fouling, though fouling, being expected, is often presumed. Obviously in a new or clean exchanger, a bad gasket or fit-up might immediately come to mind; but when the unit is partially fouled, pressure drop data may or may not indicate which specific problem is occurring. Comparing pressure drop data to normal operation may be the best available indication while the unit is in service. In cases where continued operation would not have serious consequences in reaction products, product quality, corrosion, or economics due to unachieved heat transfer, this judgment may best be delayed until the unit is out-of-service. Though fouling may be observable, close examination of all pass plate edges, gaskets, flatness, and groove edges in tubesheets and channel covers should expose the problem.

Hairpin Heat Exchangers

Hairpin heat exchangers are designed in a hairpin shape and are fabricated in accordance with ASME code. The design consists of shell and tube closures proprietary for each vendor. Hairpins are divided into two major types: Double Pipe and Multi-tube.

The Double Pipe type, shown in Fig. 9-29, consists of a single tube or pipe, either finned or bare, inside a shell. The Multi-tube type, shown in Fig. 9-30, consists of several tubes, either finned or bare, inside a shell. The maximum pressure rating of hairpin exchangers depends on a number of key design considerations including nozzles, closures, and material of construction. Standard designs are available for pressures up to 5000 psig on tubeside and 500 psig on shellside, and special designs can be fabricated for higher pressures.

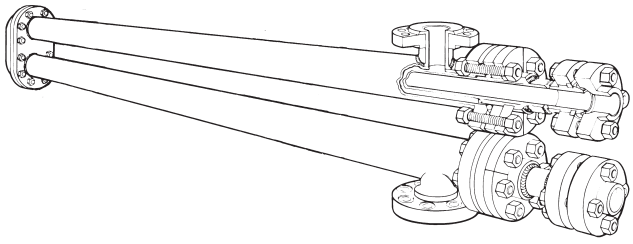
Hairpin sections are specially designed units which are normally not built to any industry standard other than ASME Code. However, TEMA tolerances are normally incorporated, wherever applicable.

Advantages

1. The use of longitudinal finned tubes will result in a compact heat exchanger for shellside fluids having a low heat transfer coefficient.
2. Countercurrent flow will result in lower surface area requirements for services having a temperature cross.
3. Potential need for expansion joint is eliminated due to U-tube construction.
4. Shortened delivery times can result from the use of stock components that can be assembled into standard sections.
5. Modular design allows for the addition of sections at a later time or the rearrangement of sections for new services.
6. Simple construction leads to ease of cleaning, inspection, and tube element replacement.

FIG. 9-29

Double Pipe Heat Exchanger



Courtesy of Brown Fintube Co.

Disadvantages

1. Multiple hairpin sections are not always economically competitive with a single shell and tube heat exchanger.
2. Proprietary closure design requires special gaskets.

Application Guidelines

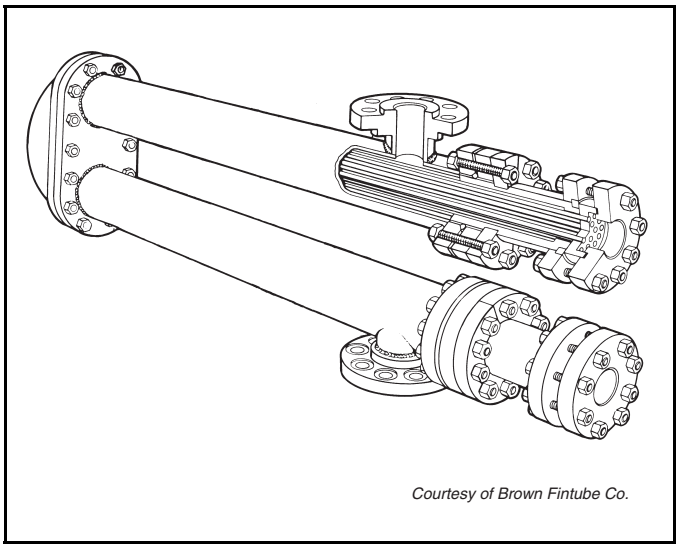
The suitability of using hairpin exchanger in a given application is frequently evaluated by computing the UA product from the basic heat transfer equation (see Eq 9-5):

$$UA = \frac{Q}{LMTD}$$

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FIG. 9-30

Multitube Heat Exchanger



Courtesy of Brown Fintube Co.

For preliminary evaluation UA = 150,000 may be considered to be the upper economical limit for applying hairpin type units.

- UA = 150,000 Btu/(hr • °F)

  Above this value, the unit may be uneconomical for a hairpin type design. If a hairpin is applied, it may require multiple 16" multitube sections.

- UA = 100,000 to 150,000 Btu (hr • °F)

  In this range, one or more 12" to 16" multitube sections will normally be required.

- UA = 50,000 to 100,000 Btu/(hr • °F)

  In this range, one or more 4" to 12" multitube sections will normally be required.

- UA = 50,000 Btu/(hr • °F)

  Below this value, both double pipe and multitube sections should be evaluated.

Fig. 9-31 lists typical sizes for hairpin type exchangers.

Tank Heaters

Tank heaters are required to maintain or increase the temperature of a tank's contents. Tank heaters are relatively inefficient in heat transfer since the fluid inside the tank normally does not flow with appreciable velocity over the heat transfer surface and the low velocities produce a high tankside film resistance. Many different methods exist for heating tanks. Some of the more common ones are described below.

Wall Mounted Coils or Panels

Wall mounted coils are common on small insulated tanks when the contents must be maintained at a constant temperature, usually at or near the inlet temperature. Design variations include simply tracing the tank with small tubing suitable for heating with steam or other hot fluid, to tanks whose walls are actually constructed from prefabricated heat transfer panels.

Internal Prefabricated Tank Heaters

Internal heaters are available in a wide variety of configurations. Steam is commonly used as the heating medium, but many other hot fluids have been used in actual practice. Externally finned pipe is commonly used to improve heat transfer efficiency. For large tanks, multiple units similar to that shown in Fig. 9-32 are supported near the bottom of the tank. For small tanks, the same tubing in a vertical configuration can be provided.

Internal Pipe Coils

Internal pipe coils are normally fabricated from 2 inch schedule 40 pipe supported near the bottom of the tank. Serpentine shaped coils are most common although circular coils are also used.

Prefabricated Stab-in Tube Bundle

Prefabricated stab-in bundles are similar to a kettle tube bundle. Removable U-tube bundles are mounted through an appropriately sized opening near the tank bottom.

Tank Suction Heaters

A tank suction heater is shown in Fig. 9-33. This type of unit raises the temperature and thus reduces the viscosity of a pumped fluid without heating the tank fluid that remains in the tank.

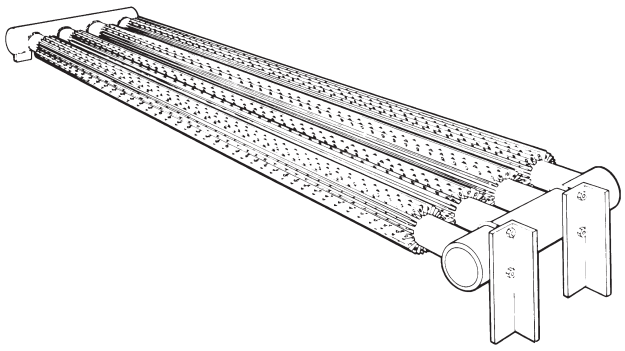
FIG. 9-31

Typical Hairpin Exchanger Sizes

	Double Pipe	Multitube
Shell dia (in.)	2" - 6"	3" - 16"
Tube dia (in.)	¾" - 4"	¾" - 1"
No. of Longitudinal Fins	0 - 72	0 - 24
Fin Height (in.)	0 - ¾"	0 - ½"
Surface (sq ft/20 ft)	10 - 40	75 - 1500

FIG. 9-32

Prefabricated Tank Heater

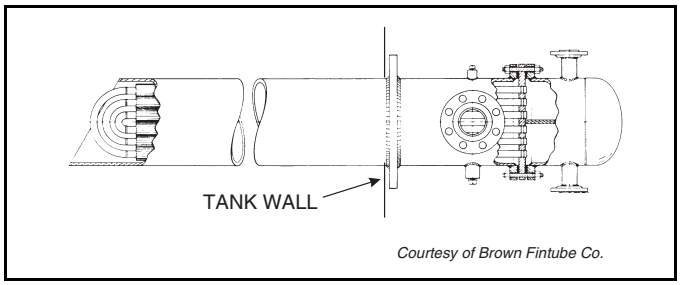


Courtesy of Brown Fintube Co.

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FIG. 9-33

Tank Suction Heater



Courtesy of Brown Fintube Co.

Plate-Fin Exchangers
(Brazed Aluminum)

Brazed aluminum heat exchangers have been employed in cryogenic gas processing plants since the 1950s. This section briefly describes the basic configuration, advantages, hardware capabilities, rough section criteria, and user considerations for brazed aluminum heat exchangers used in gas processing applications.

BASIC CONFIGURATION

A brazed aluminum heat exchanger is composed of alternating layers of corrugated fins and flat separator sheets called parting sheets. A stack of fins and parting sheets comprise the heat exchanger, sometimes referred to as the "core." The heat exchanger is normally specified by its outside dimensions in the following order: Width (W) x Stack Height (H) x Length (L). The number of layers, type of fins, stacking arrangement, and stream circuiting will vary depending on the application requirements. The basic aluminum components of the heat exchanger are shown in Fig. 9-34 for a typical three stream counterflow exchanger.

Nozzles — Nozzles are the pipe sections used to connect the heat exchanger headers to the piping.

Headers — Headers are the half cylinders which provide for the distribution of fluid from the nozzle to or from the ports of each appropriate layer within the heat exchanger.

Ports — Ports are the opening in either the side bar or the end bar, located under the headers, through which the fluids enter or leave the individual layers.

Distributor Fins — Distributor fins distribute the fluid between the port and the heat transfer fins.

Heat Transfer Fins — The heat transfer fin provides extended heat transfer surface. All fins, both heat transfer and distributor, provide connecting structure between the parting sheets, which is necessary for the structural and pressure holding integrity of the heat exchanger. Typical heat transfer fin thicknesses range from 0.006 to 0.023 inches.

Parting Sheets — The parting (separator) sheets contain the fluids within the individual layers in the exchanger and also serve as primary heat transfer surface. Typical parting sheet thicknesses range from 0.032 to 0.080 inches.

Outside Sheets — Outside (cap) sheets serve as the outside parting sheets. They are typically 1/4-inch thick and serve as an outer protective surface of the exchanger.

Bars — The side and end bars enclose the individual layers and form the protective perimeter of the exchanger. Solid extruded bars from 1/2 to 1 inch wide are typically used.

Support Angles — Support angles are typically 90 degree extruded aluminum angles welded to the bar face of the exchanger for the purpose of supporting or securing the exchanger in its installed position. Other support configurations are available.

Battery — A multiple exchanger assembly, sometimes referred to as a "battery," consists of two or more exchangers piped or manifolded together into a single assembly, with the individual exchangers arranged either in a parallel, series, or combination parallel-series arrangement.

Cold Box — Individual exchangers and batteries are often installed in a "cold box." A cold box consists of a welded, airtight carbon steel casing, rectangular or cylindrical in shape, which supports and houses the heat exchangers, piping, other related cryogenic equipment, and insulation material.

ADVANTAGES AND LIMITATIONS

Aluminum maintains excellent strength and ductility to temperatures as low as –452°F. Aluminum actually increases in strength at cold temperatures. Because of its relatively low melting temperature, however, aluminum is less resistant to fires and high temperatures than are some other materials. Aluminum is generally not employed for process temperatures above 150°F, especially when higher pressures are involved. Aluminum exchangers are less resistant to rough handling and mistreatment than steel equipment and are limited to use with fluids non-corrosive to aluminum.

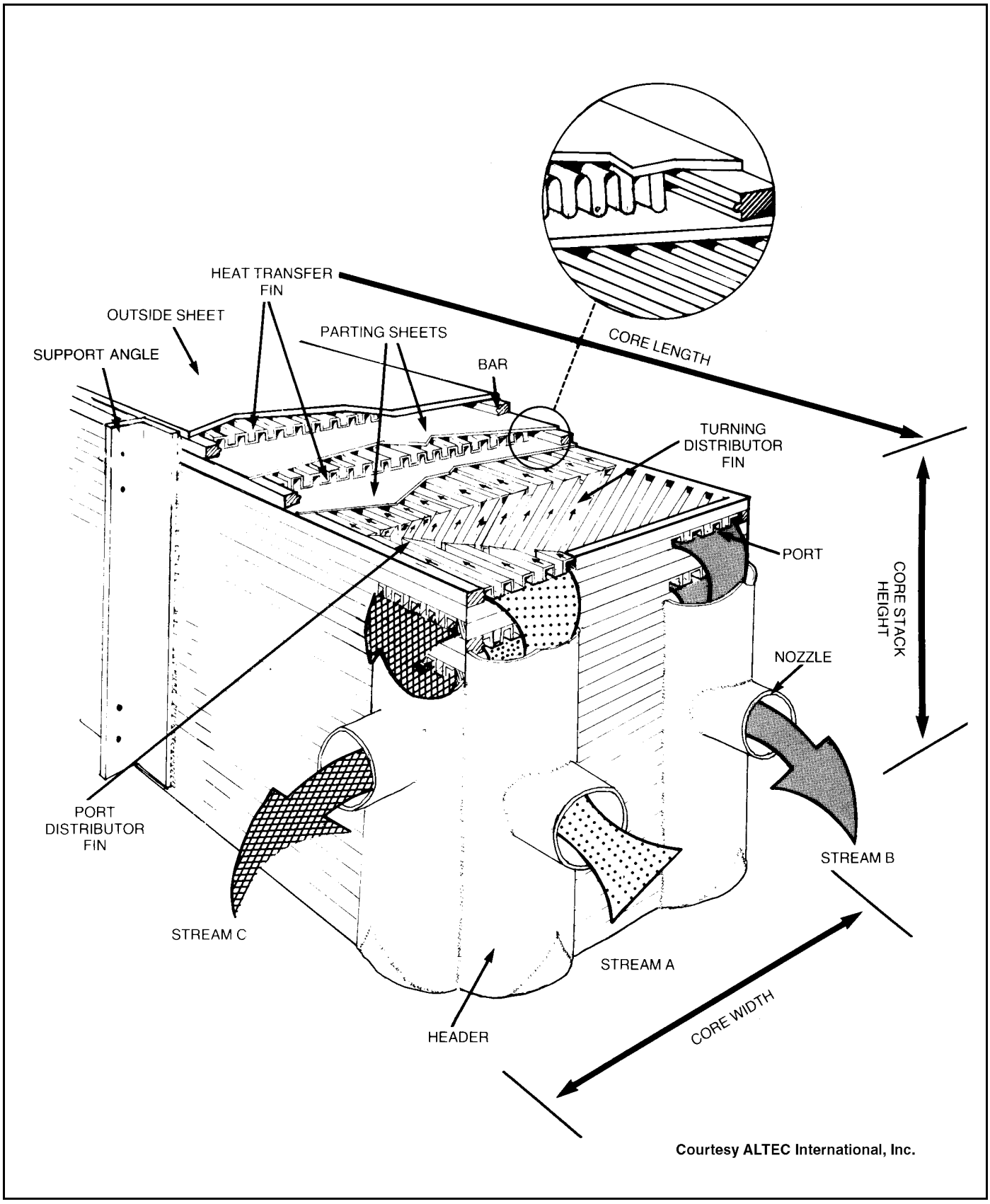
Brazed aluminum heat exchangers are compact and lightweight. A typical high pressure brazed aluminum heat exchanger (600-1,400 psig design pressure) will provide 300 to 400 square feet of heat transfer surface per cubic foot of exchanger volume. This is six to eight times the surface density of comparable shell and tube exchangers. Additionally, a typical high pressure brazed aluminum heat exchanger will have a density of 75 to 90 pounds per cubic foot of exchanger versus approximately 250 pounds per cubic foot for comparable shell and tube exchangers. The net effect of these differences is that a brazed aluminum heat exchanger will provide approximately 25 times more surface per pound of equipment than comparable shell and tube exchangers. This decrease in exchanger weight and volume reduces foundation, support, plot plan, and insulation requirements.

An important point to note when evaluating the size and efficiency of a brazed aluminum heat exchanger is that it is customary to include total surface in all streams, hot and cold. This is equivalent to counting both the inside and outside tube surfaces in a tubular heat exchanger. This method for specifying surface is used because brazed aluminum heat exchangers will often be designed with an unbalanced surface. More surface is provided on one stream of the heat exchanger than in the other stream(s) in order to balance the variation in heat transfer coefficients. Up to ten streams can be combined into a single brazed aluminum heat exchanger; combining counterflow, crossflow, and cross-counterflow circuiting. Temperature approaches of 3°F on single-phase fluids and 5°F on two-phase

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FIG. 9-34

Basic Components of a Three Stream Counterflow Brazed Aluminum Heat Exchanger



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fluids can be achieved. Typically, corrected mean temperature differences of 5°F to 10°F are employed in brazed aluminum heat exchanger applications.

Brazed aluminum heat exchangers should be used with clean fluids since they are more susceptible to plugging than other types of heat exchanger equipment; however, proper filters will prevent heat exchanger fouling. Brazed aluminum should not be used with fluids which are corrosive to aluminum. Mercury and caustic soda are extremely corrosive to aluminum and should not be introduced into the exchanger. Hydrogen sulfide and carbon dioxide are not a corrosion problem in streams with water dewpoint temperatures below the cold end temperature of the exchanger.

APPLICATIONS

Brazed aluminum heat exchangers are used in the following cryogenic natural gas processing applications.

Ethane Plus Recovery
Nitrogen Removal
Helium Recovery
Liquefied Natural Gas (LNG)

Within these applications, brazed aluminum heat exchangers are used for the following heat exchanger services:

Gas to Gas Exchangers
Demethanizer Reboilers
Demethanizer Reflux Condensers
Feed Gas Exchangers
Product Heaters
Propane Chillers

HARDWARE CAPABILITIES

Materials and Codes of Construction

Brazed aluminum heat exchangers are designed and constructed to comply with the "ASME Boiler and Pressure Vessel Code," Section VIII, Division I, or other applicable standards. The aluminum alloys used comply with ASME Section II, Part B, "Nonferrous Materials," or the requirements of the specified code authority.

Aluminum alloy 3003 is generally used for the parting sheets, corrugated fins, and bars which form the rectangular heat exchanger block. These parts are metallurgically bonded by a brazing process at temperatures of about 1100°F. The brazing alloy is an aluminum silicon metal and is provided on or with the parting sheets. Headers and nozzles are made from aluminum alloys 3003, 5054, 5083, 5086, 5454, or 6061-T6. Alloy 5083 is the most commonly used.

Maximum Working Temperature, Pressure, and Sizes

The maximum design temperature rating for brazed aluminum heat exchangers is typically 150°F; however, special designs are available for design temperatures up to 400°F. The minimum design temperature is –452°F.

ASME code approved brazed aluminum heat exchangers are available for pressure ratings from zero absolute to 1,400 psig. Different design pressures can be used for each stream in the exchanger. The maximum core size available will vary with the maximum design pressure as shown in Fig. 9-35. Some size variation from Fig. 9-35 will occur depending on a particular manufacturer's capabilities, specific design, and flow configuration. Batteries of exchangers are much larger and are limited in size by transportation capabilities.

Fins

Fins are available to cover a wide range of applications for a variety of heat transfer and pressure drop requirements at low, medium, and high pressure. The economic justification for using a particular fin type is unique for each application and is highly dependent on the cost of power relative to other considerations. Three major types of fins are shown in Fig. 9-36. These include plain (straight), serrated (lanced), and perforated. These and other more specialized fins can provide heat exchanger designs optimized for the best combination of heat transfer, pressure drop, compactness, and cost for a specific application.

Distributor and Passage Arrangements

There are a large number of distributor and passage arrangements available in brazed aluminum heat exchangers. Fin arrangements frequently used in gas processing applications are shown in Fig. 9-37 for a gas/gas exchanger. The "A" stream layers are shown with center distributors and provide for the residue gas to flow through the entire length of the heat exchanger. The "B & C" stream layers are arranged with side distributors and provide for each of the two high pressure feed gas streams to flow through only a portion of the overall heat exchanger length.

FIG. 9-35

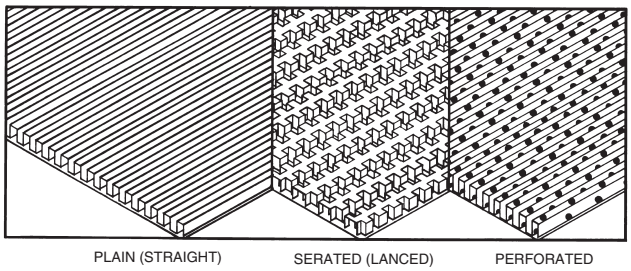
Approximate Maximum Plate-Fin Exchanger Sizes & Pressures

Max. ASME Design Pres. (psig)	Maximum Size (Inches)			Nominal Total Heat Transfer Surface (Square Ft)	Nominal Total Net Weight*, (Pounds)
	Width	Stack Ht	Length		
400	48	48	240	120,000	16,000
600	42	42	240	84,000	17,000
800	36	38	240	56,000	16,000
1000	36	38	240	56,000	17,000
1200	36	38	240	56,000	17,000
1400	25	25	144	15,000	5,000

*Without Flanges

FIG. 9-36

Three Basic Fin Types

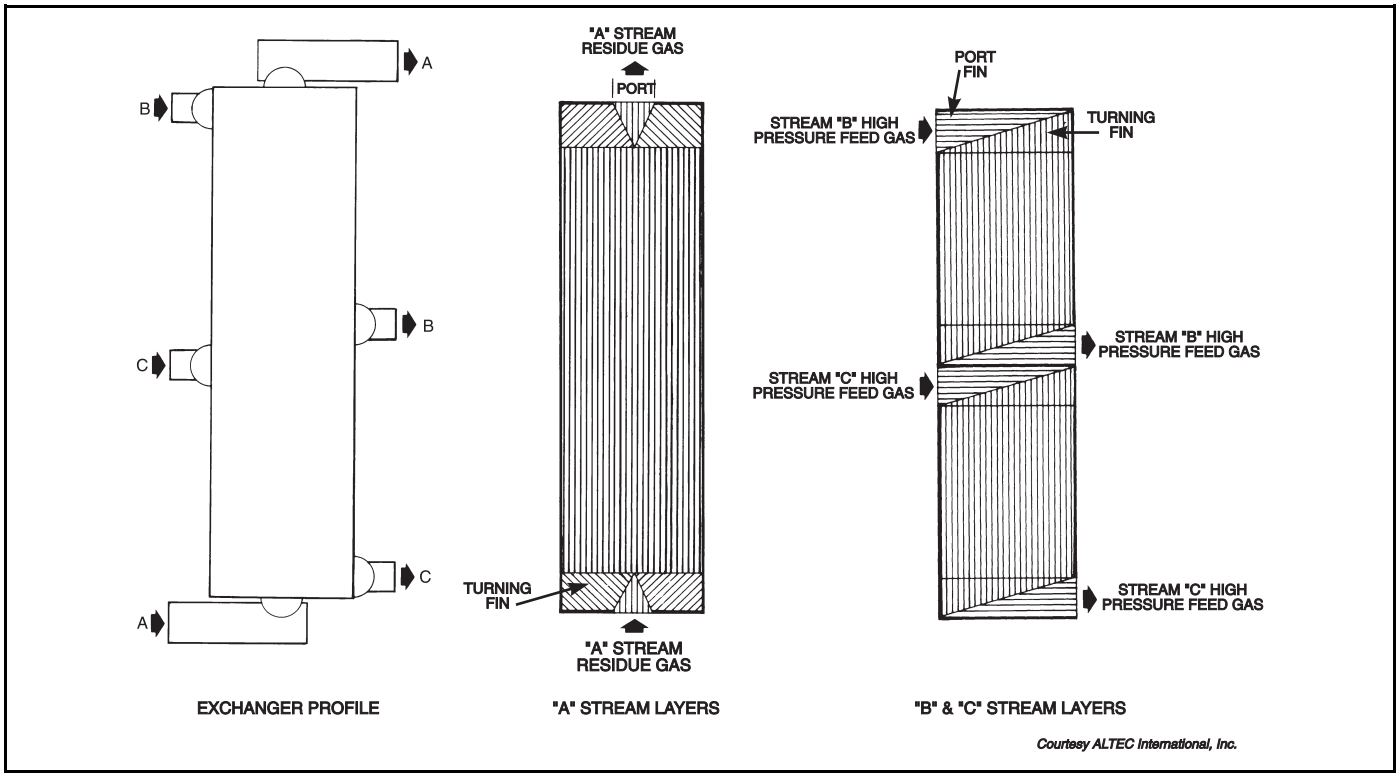


Courtesy ALTEC International, Inc.

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FIG. 9-37

Typical Fin Arrangements for Gas/Gas Exchanger



SELECTIONS

Brazed Aluminum Heat Exchanger Specifications

The design and specification of a brazed aluminum heat exchanger require thermodynamic and mechanical information.

Thermodynamic — Heat transfer duty, operating pressure and temperature, allowable pressure drop, flow rates, compositions, and the physical, thermodynamic, and transport properties of the fluids involved must be specified. A cooling (or load) curve should be supplied to the designer/manufacturer for two-phase applications, and it may be necessary for single phase streams operating over a wide temperature range.

Mechanical — Specifications should include information on applicable code authorities, design pressure and temperature, and requirements for connection size, type, and orientation. Exchanger support and package requirements should also be defined.

Fig. 9-38 is a sample manufacturer's specification sheet. This document communicates the details of the heat exchanger design between the manufacturer and user. Lines 1-37 define the minimum information required from the user. Other required information includes turndown conditions, off-design conditions, and any other special operating conditions, if applicable. Using this information, the manufacturer will design the heat exchanger and provide the information in lines 38-61.

Heat Load Curves

Generation of the heat load curve, commonly called the cooling curve, from a temperature-duty table is an important first step in the analysis of any heat exchanger. It illustrates the intended heat exchange process and is used to define the required heat exchanger conductance (UA). A cooling curve also shows bubble and dewpoints, regions of phase change, and close temperature approaches.

Fig. 9-39 shows the temperature-duty data and cooling curve for a three-stream gas-to-gas heat exchanger. For multistream heat exchange services (more than two streams), the cooling curve can be reduced to a classic two-stream case for purposes of calculating and corrected mean temperature difference (CMTD) and the UA required. This is called the combined cooling curve assumption and is normally used for simple sizing calculations. Fig. 9-39 shows how a three-stream exchanger cooling curve is reduced to two streams by combining the cold residue gas stream duty with the cold recycle stream duty at points of constant temperature to form a combined cold stream.

The CMTD is approximated by calculating the log mean temperature difference (LMTD) on portions of the combined cooling curve called zones. The UA required in each zone is then calculated from the zone LMTD. At this point, it is possible to make individual heat exchanger sizings for each zone, or where more approximate sizings are acceptable, to make a heat exchanger sizing based on the combined zones using the CMTD. The CMTD is approximated by adding the UA's and Q's of each zone and dividing as shown in Fig. 9-39.

Selection of the precise number and location of zones is a matter of choice for the designer. However, proper selection of the number and location of zones will increase the accuracy of

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FIG. 9-38

Brazed Aluminum Heat Exchanger Specifications

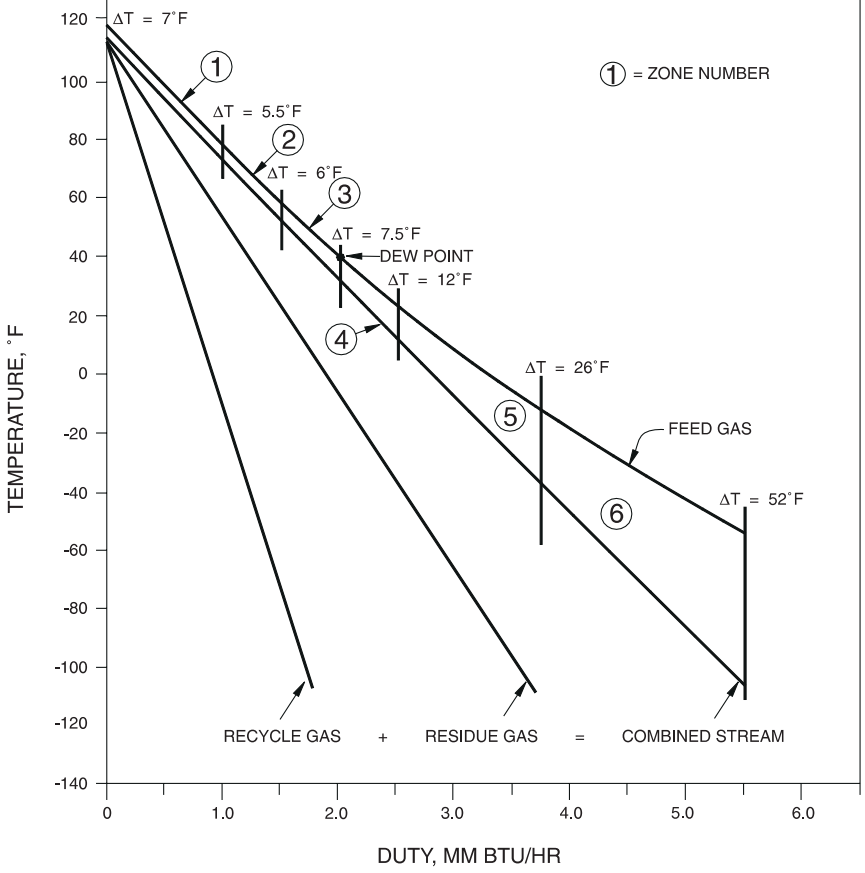
						Specification No. Typical Rev Pg 1 Of 1			
						Date			
						Customer			
						Project Job			
						Location			
Plant Service	GAS TO GAS EXCHANGER					Item No. E-1			
Stream		A) Feed Gas		B) Residue Gas		C) Recycle Gas			
Stream Components		Mole %		Mole %		Mole %		Mole %	
N2/CO2		4.0/0.6		4.5/0.4		4.0/0.4			
C1		83.8		93.7		92.5			
C2		6.3		1.3		2.1			
C3		3.1		0.1		1.0			
iC4/nC4		0.4/1.0		-		-			
iC5/nC5		0.2/0.6		-		-			
	Total	100.0		100.0		100.0			
Total Flow	lb/hr	40650		31320		14780			
Conditions At		Inlet	Outlet	Inlet	Outlet	Inlet	Outlet	Inlet	Outlet
Vapor Flow Rate	lb/hr	40650	32195	31320	31320	14780	14780		
Molecular Weight		19.5	17.9	16.9	16.9	17.2	17.2		
Density	lb/ft^3	2.79	4.84	1.02	0.56	1.32	0.80		
Viscosity	lb/ft^3-hr	0.031	0.028	0.019	0.029	0.020	0.029		
Liquid Flow Rate	lb/hr		8455						
Molecular Weight			30.1						
Density	lb/ft^3		28.0						
Viscosity	lb/ft^3-hr		0.15						
Temperature	°F	120	-54	-106	113	-106	113		
Pressure	PSIA	810		205		285			
Fluid Cond. or Vap.	lb/hr	8455 (Cond.)							
Avg. Specific Heat (Vap/Liq)	BTU/lb°F	0.638/0.578		0.520		0.532			
Avg. Thermal Cond. (Vap/Liq)	*	0.0204/0.570		0.0167		0.0165			
Latent Duty	BTU/hr								
Total Duty	BTU/hr	5,500,000		3,700,000		1,800,000			
Corrected MTD	°F	11.22							
Allowable Pressure Drop	PSI	10.0		7.0		7.0			
Fouling Resistance	hr-°F-ft^3-BTU	0.001		0.001		0.001			
Design Temp. (Max/Min)	°F	-300/+150		-300/+150		-300/+150			
Design Pressure (Max)	PSIG	1100		300		400			
Core Width (in) 25		No. Layers Core 63		Total No. Cores 1		In Parallel Series 1			
Core Stack Ht. (in) 23-1/4		? Sheet Thk (in) 0.080		No. Assemblies 1		Net Core (Ass'y) Wt. 5650			Lbs.
Core Length (in) 180		Outside Sheet Thk (in) 0.25		No. Cores per Ass'y 1		Shipping Core (Ass'y) Wt.. 6000			Lbs.
No. Layers per Core		21		30		12			
Fin Surface Type		Serrated		Serrated		Serrated			
Fin Height	in	0.280		0.280		0.280			
Fin Thickness & Spacing	in x FPI	0.016 x 17		0.016 x 17		0.016 x 17			
Effective Passage Width	in	23-3/8		23-3/8		23-3/8			
Total Free Flow Area	ft^2	0.65		0.93		0.37			
Effective Passage Length	in	163-3/8		162-3/8		162-3/8			
Total Heat Transfer Surface	ft^2	5910		8390		3360			
Frictional Pressure Drop	PSI	8.8		6.9		6.4			
Header Size	HPS	8	8	10	10	8	8		
Nozzle Size	HPS	6	6	8	8	6	6		
Manifold Size	HPS	-	-	-	-	-	-		
Type Connections		600 Lb.		300 Lb.		300 Lb.			
Test Pressure (Hydro Air Leak)	PSIG	1650/1375		450/375		600/500			
Special Testing		None		None		None			

Courtesy ALTEC International, Inc.

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FIG. 9-39

Heat Load Curve for a Three Stream Exchanger



TEMPERATURE DUTY DATA

FEED GAS (WARM STREAM)			RESIDUE GAS (COLD STREAM)			RESIDUE GAS (COLD STREAM)		
T (°F)	DUTY (MM BTU/HR)	Xv	T (°F)	DUTY (MM BTU/HR)	Xv	T (°F)	DUTY (MM BTU/HR)	Xv
120	0	1.0	113	0	1.0	113	0	1.0
96	0.585	1.0	64.6	0.824	1.0	64.6	0.394	1.0
72	1.179	1.0	16.2	1.636	1.0	16.2	0.783	1.0
48	1.785	1.0	– 32.2	2.443	1.0	– 32.2	1.173	1.0
24	2.504	0.977	– 80.6	3.252	1.0	– 80.6	1.570	1.0
0	3.339	0.931	– 106	3.700	1.0	– 106	1.800	1.0
24	4.230	0.879						
54	5.500	0.792						

HEAT EXCHANGE ZONES

ZONE	LMTD (°F)	DUTY (MM BTU/HR)	UA (MM BTU /(HR-°F))
1	6.22	1.0	0.1608
2	5.75	0.5	0.0870
3	6.72	0.5	0.0744
4	9.57	0.5	0.0522
5	18.1	1.25	0.0691
6	37.5	1.75	0.0467
TOTAL	CMTD	5.5	0.4902

$$\text{CMTD} = \frac{Q}{UA} = \frac{5.5}{0.4902} = 11.22°\text{F}$$

Courtesy ALTEC International, Inc.

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the estimate for the corrected mean temperature difference (CMTD) and the exchanger performance. The following are some useful guidelines for proper selection of a zone:

1. The temperature difference between the warm and cold composite streams must be essentially a linear function of duty; i.e., straight line.
2. The heat transfer coefficients in each stream must be nearly constant.
3. The properties of the fluids in each stream should be nearly constant. This is usually necessary in order for the heat transfer coefficient to remain essentially constant and for achieving reasonable accuracy in estimating stream pressure drop. This is particularly true for two-phase streams; therefore, special care should be taken in selecting zones when two-phase fluids are involved.

Heat leak in cryogenic heat exchangers is another factor which will affect the cooling curve. It acts as an unwanted heat flow into the heat exchange fluids and will reduce the CMTD. Heat leak and inaccuracies in the fluid thermodynamic data used to generate the cooling curve can significantly reduce the CMTD and increase the UA required for a particular process. For well insulated exchangers, heat leak normally has a negligible effect on the CMTD. However, the amount of heat leak should always be checked and combined as another warm stream on the cooling curve to determine its effect on the CMTD.

Design Considerations for Two-Phase Flow

Procedures for designing brazed aluminum heat exchangers for single-phase streams are well publicized by manufacturers and by Kays and London.[3] The design of brazed aluminum heat exchangers for two-phase streams is not as well published.

Brazed aluminum heat exchangers are often used with two-phase streams in gas processing applications. Generally, condensing is performed vertically downward and vaporization vertically upward.

Pressure drop is usually evaluated with the Lockhart and Martinelli[4] method which has been found to be reasonable for both vaporizing and condensing streams.

Since evaluation of the heat transfer coefficients for two-phase streams in brazed aluminum heat exchangers has not been well reported in the literature, most manufacturers use their own proprietary calculation procedures.

Two-phase (liquid-vapor) heat transfer in brazed aluminum heat exchangers is usually dominated by the forced convection mechanism. This convection mechanism tends to suppress any nucleate boiling.

The two-phase forced convection heat transfer coefficients for multi-component fluids are evaluated using the method described by Bell and Ghaly.[5] This method provides for reducing the two-phase heat transfer coefficient to account for the mass transfer resistance that is a characteristic of multi-component heat transfer.

Fluid distribution is always an important consideration when designing high effectiveness heat exchangers. Special care must be taken to ensure that the fluids maintain homogenous flow throughout the heat exchanger. This is especially important with two-phase streams, where fluid maldistribution can significantly reduce the performance of the heat exchanger. For this reason, special distributors are available for use with fluids which enter the exchanger in a two-phase condition. Fluid distribution for such an inlet condition is often handled by separating the vapor and liquid phases in a knock-out drum. The separated vapor and liquid phases are then distributed individually into the exchanger using conventional single-phase distributor arrangements. The vapor and liquid phases are then recombined inside the exchanger in the heat transfer zone. This method is the preferred arrangement for distributing fluids which enter the heat exchanger in a two-phase condition. However, sometimes alternative simple approaches to a two-phase inlet can be used. This choice depends on a thorough analysis of the effects of the potential vapor/liquid maldistribution on the cooling curve and performance of the exchanger.

Brazed aluminum heat exchangers are well suited for thermosyphon applications such as demethanizer reboilers and propane chillers. Hydraulic design considerations are the same as for shell and tube exchangers.

Approximate Sizing Procedure

The following is a quick and simple method for estimating the approximate size and performance of gas-to-gas exchangers and demethanizer reboilers used for ethane recovery. This short-cut method is applicable only for services condensing up to 30% (wt.) of feed gas or for reboilers vaporizing up to 20% (wt.) of feed liquid. For services outside these limits, a plate fin design specialist should be consulted.

The following sample problem illustrates the approximate sizing procedure for a gas-to-gas exchanger. Typically, this application involves a warm feed gas operating at pressures between 500 psia and 1,100 psia, which is cooled from above 100°F to below –100°F, and will partially condense up to 30% of its mass. The refrigeration is supplied by cold residue or recycle gas streams which normally operate at between 100 and 300 psia. Warm end temperature approaches for this exchanger are typically designed from 5 to 10°F. This sample problem is the same example used in the heat load curve (Fig. 9-39) and in the brazed aluminum heat exchanger specification (Fig. 9-39) for an optimized selection. The results of the rough selection agree well with the optimized selection. Lines 1 through 37 of Fig. 9-38 are given data, provided by the purchaser.

Example 9-4 — For purposes of simplifying this quick selection procedure, the following are assumed:

A fouling factor of 0.001 is included on each stream.

The CMTD was calculated assuming that heat leak was negligible, as per the example of Fig. 9-39.

The estimation of heat exchanger size by this quick procedure is reduced to a single-zone calculation. Precise selections by the manufacturers normally involve a multizone analysis of the heat exchanger.

Solution Steps

Step 1 — Determine Exchanger Cross Section

From Fig. 9-40, select the typical mass velocities (G) for each stream based on their respective operating pressures:

Feed Gas (810 psia)

G_H = 18.6 lb/(sq ft • sec)

Residue Gas (205 psia)

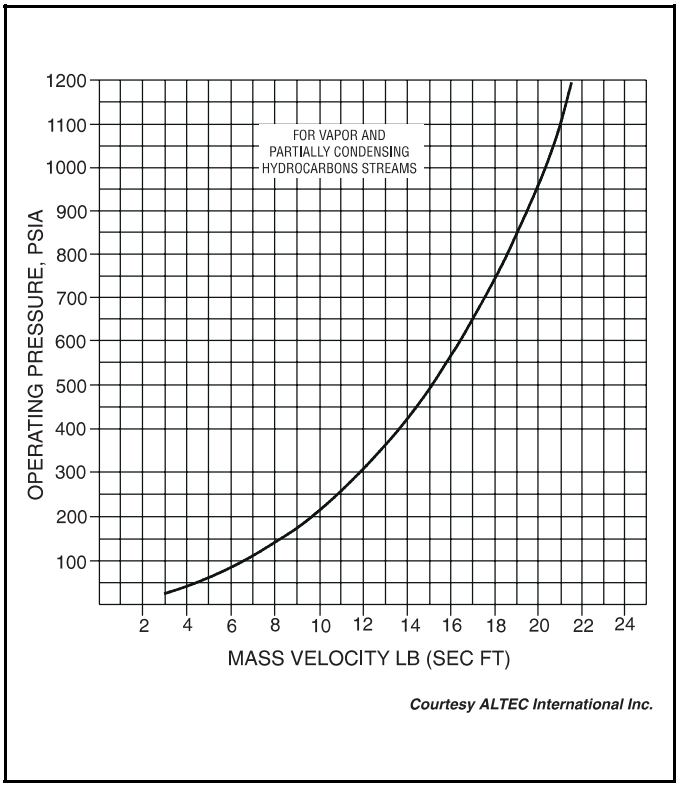
G_C = 9.7 lb/(sq ft • sec)

Recycle Gas (285 psia)

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FIG. 9-40

Typical Operating Mass Velocities
Gas Processing Exchangers



Courtesy ALTEC International Inc.

G_C = 11.6 lb/(sq ft • sec)

Using these G values, the exchanger cross section can be computed from the following equations:

$$H = \frac{0.083}{WN} \sum_{i=1}^{i=n} \left(\frac{\dot{m}_i}{G_i} \right), \text{ for gas to gas exchangers} \quad \textbf{Eq 9-15}$$

Where n = total number of all warm and cold streams

$$H = \frac{0.166}{WN} \left(\frac{\dot{m}_H}{G_H} \right), \text{ for demethanizer reboilers} \quad \textbf{Eq 9-16}$$

In order to use the above equations, initially choose a number of exchangers (N) and an exchanger width (W). Common exchanger widths are 12, 17, 25, 36, 42, and 48 inches. Select the cross section so that the stack height (H) is within the maximum size ranges shown in Fig. 9-35. For smaller cross section requirements, a good rule of thumb is to select a stack height which is nearly equal to the exchanger width. For the gas to gas exchanger, Example 9-4, assume one exchanger, 25 inches wide, and use equation 9-15:

$$H = \frac{0.083}{(25)\,(1)} \left[\frac{40650}{18.6} + \frac{31320}{9.7} + \frac{14780}{11.6} \right]$$

$$H = 22.2 \text{ inches}$$

The exchanger cross section has now been established at 25 inches wide x 22.2 inches high.

Step 2 — Determine Exchanger Length

First calculate the required UA from the heat load curve (see Fig. 9-39):

$$UA = UA_{zone\,1} + UA_{zone\,2} + \ldots + UA_{zone\,6} = \frac{Q}{CMTD}$$

$$= \frac{5{,}500{,}000}{11.22}$$

$$= 490{,}200 \text{ Btu/(hr • °F)}$$

From one of the following equations, depending on fin preference, calculate the exchanger length. When both perforated and serrated fins are used in the heat exchanger, use the average value obtained from Equations 9-17 and 9-18 (or Equations 9-20 and 9-21 for demethanizer reboilers). Serrated fins are high performance and yield shorter exchanger lengths with higher stream pressure drops. Perforated fins are lower performers and will yield longer exchangers with lower pressure drops. Plain fins are lowest performers and are normally only used in distributors, due to their low pressure drop characteristics. Normally, serrated fins provide the most optimum selection, unless pressure drop/operating cost is the controlling parameter.

For gas to gas exchangers (where feed gases condense up to 30% of their mass):

For serrated fin exchangers —

$$L = \frac{UA}{WHN\sqrt{G_{Total}}} + 0.65W \quad \textbf{Eq 9-17}$$

For perforated fin exchangers —

$$L = \frac{1.5\,UA}{WHN\sqrt{G_{Total}}} + 0.65W \quad \textbf{Eq 9-18}$$

Where $G_{Total} = (G_H)_{Avg} + (G_C)_{Avg}$ **Eq 9-19**

For demethanizer reboilers (where demethanizer liquids evaporate up to 20% of their mass):

For serrated fin exchangers —

$$L = \frac{0.67\,UA}{WHN\sqrt{G_H}} + 0.65W \quad \textbf{Eq 9-20}$$

For perforated fin exchangers —

$$L = \frac{UA}{WHN\sqrt{G_H}} + 0.65W \quad \textbf{Eq 9-21}$$

For one gas to gas exchanger, 25 inches (W) x 22.2 inches (H), using serrated fins, and a UA required of 490,200 Btu/(hr·°F), the required length to the nearest inch is:

$$L = \frac{490{,}200}{(25)\,(22.2)\,(1)\sqrt{18.6 + (9.7 + 11.6)/2}} + 0.65(25)$$

$$L = 180 \text{ inches}$$

The heat exchanger size is now established as:

One exchanger, 25 x 22.2 x 180 inches.

If exchanger length is too long for packaging and/or transportation considerations, lower the mass velocities (G) and return to Step 1. This will increase the core cross section and decrease the core length.

Step 3 — Check Stream Pressure Drops

The last step in rough sizing the heat exchanger is to verify that pressure drop for the exchanger size selected is within allowable levels for all streams. Pressure drop can vary widely depending on type and size of distributors chosen, the amount of phase change in two-phase streams, and other factors. The

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following equations will yield approximate stream frictional pressure drop which includes the heat transfer zone, distributors, and nozzles.

For Serrated Fin Exchangers:

For vapor streams —

$$\Delta P = \frac{(4.4)\,(10^{-4})\,(L+20)\,(G)^{1.8}}{\rho_m} \qquad \textbf{Eq 9-22}$$

For partially condensing streams (up to 30% of mass condensed) —

$$\Delta P = \frac{(8.8)\,(10^{-4})\,(L+40)\,(G)^{1.8}}{\rho_m} \qquad \textbf{Eq 9-23}$$

For Perforated Fin Exchangers:

For vapor streams —

$$\Delta P = \frac{(0.95)\,(10^{-4})\,(L+95)\,(G)^{1.8}}{\rho_m} \qquad \textbf{Eq 9-24}$$

For partially condensing streams (up to 30% of mass condensed) —

$$\Delta P = \frac{(1.9)\,(10^{-4})\,(L+190)\,(G)^{1.8}}{\rho_m} \qquad \textbf{Eq 9-25}$$

where $\rho_m = \dfrac{2(\rho_{in})\,(\rho_{out})}{\rho_{in}+\rho_{out}}$ **Eq 9-26**

And when inlet or outlet conditions are two-phase:

$$\rho_{2\Phi} = \left(\frac{1}{\frac{X_V}{\rho_V}+\frac{X_L}{\rho_L}}\right) \qquad \textbf{Eq 9-27}$$

Pressure drops for vaporizing liquids are not easily approximated by these rough estimating procedures. However, for these applications, the demethanizer liquid pressure drop will normally be within allowable levels when the exchanger is selected for feed gas mass velocities recommended in Fig. 9-40.

For example problem 9-4:

Feed Gas:

$$\rho_{out} = \rho_{2\Phi} = \left(\frac{1}{\frac{0.792}{4.84}+\frac{0.208}{28.0}}\right) = 5.85 \text{ lb/ft}^3$$

$$\rho_m = \frac{2(2.79)\,(5.85)}{2.79+5.85} = 3.78 \text{ lb/ft}^3$$

$$\Delta P = \frac{(8.8)\,(10^{-4})\,(180+40)\,(18.6)^{1.8}}{3.78}$$

$$= 9.9 \text{ psi (vs. 10.0 psi allowed)}$$

Residue Gas:

$$\rho_m = \frac{2(1.02)\,(0.56)}{1.02+0.56} = 0.72 \text{ lb/ft}^3$$

$$\Delta P = \frac{(4.4)\,(10^{-4})\,(180+20)\,(9.7)^{1.8}}{0.72}$$

$$= 7.3 \text{ psi (vs. 7.0 psi allowed)}$$

Recycle Gas:

$$\rho_m = \frac{2(1.32)\,(0.80)}{1.32+0.80} = 1.00 \text{ lb/ft}^3$$

$$\Delta P = \frac{(4.4)\,(10^{-4})\,(180+20)\,(11.6)^{1.8}}{1.00}$$

$$= 7.2 \text{ psi (vs. 7.0 psi allowed)}$$

For selections requiring more than one heat exchanger, the pressure drop for the manifold piping which interconnects the individual heat exchangers can be estimated according to the method developed by F. A. Zenz,[6] and must be added to the pressure drops for the individual exchanger(s) calculated by the above procedures to arrive at the total unit pressure drop. For the single core Example 9-4, this is not required, and the pressure drops are only slightly over allowable. If pressure drops were too excessive, it would be necessary to return to Step 1 (Equation 9-15) and to lower the mass velocities (G). This has the effect of increasing exchanger cross section until the desired pressure drop is achieved. Use the following equation for approximating a new mass velocity which will yield the allowable pressure drop.

$$G_{new} = G_{old}\left[\frac{\Delta P_{allowed}}{\Delta P_{old}}\right]^{0.56} \qquad \textbf{Eq 9-28}$$

The above sizing procedure should produce estimates of exchanger size which will be within plus or minus 15% and pressure drops which should be within plus or minus 25% of the final design.

INSTALLATION-OPERATION-MAINTENANCE

Mounting

Brazed aluminum heat exchangers are normally installed in a vertical orientation with the operational cold end down, and are supported with either aluminum support angles or an aluminum pedestal base supplied by the heat exchanger manufacturer. This type of support system is the most common for mounting the exchanger to steel framework or onto a platform. Other orientations of the heat exchangers and other support systems are sometimes permissible, but only when designed for special service by the manufacturer.

External loads on the heat exchanger can be imposed through the connecting piping due to mechanical or thermal loading or both. All support systems should be designed to minimize these loads and their effect on the heat exchanger. This is accomplished by providing sufficient pipe flexibility and by providing allowance for movement at the heat exchanger support member by using slotted bolt holes and bolts that are only finger tight. All support systems should be additionally safeguarded by use of sway bracing on the end of the exchanger opposite the main support system whenever the total external pipe loads on the exchanger will produce reaction forces at the main support members which exceed the actual weight of the exchanger.

Insulation

Since the exchangers are usually operating at cryogenic temperatures, highly efficient insulation is required to minimize heat leak. Typically, the exchanger is mounted in a cold box which is filled with perlite or rock wool. When the exchanger is not mounted inside a cold box, its exterior is normally insulated with rigid polyurethane foam. An alternative is Foamglas® insulation. These insulations are positioned and fastened around the exchanger and covered with a vapor barrier. Protective metal coverings or flashing can be used for this purpose. Some form of insulation (such as micarta spacers)

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should be used between the heat exchanger support member and the supporting beam or platform.

Field Testing and Repair

Maximum working pressures and temperatures are always specified on the manufacturer's nameplate. These values should not be exceeded during field testing or operation. Since it is extremely difficult to dry brazed aluminum heat exchangers in the field, only a clean dry gas should be used for leak testing.

Internal leaks in a brazed aluminum heat exchanger are generally indicated by a change of purity in any of the fluid streams. External leaks can be determined by sight, smell, audible sounds of leaking fluid, external gas monitoring equipment, or localized cold spots appearing on the external insulation. External leaks in exchangers mounted in a cold box are generally indicated by excessive venting through the cold box breather valves.

Several tests are available for locating external or internal leaks. An air-soap test is effective for locating external leaks. An air test with soap applied to nozzle connections or a nitrogen-freon test can be used to identify the streams involved in an internal or cross pass leak. Internal and external leaks usually can be repaired by blocking layers, making localized external welds, etc. Qualified manufacturer's representatives are usually required to establish the exact location of an internal leak and to make any repairs.

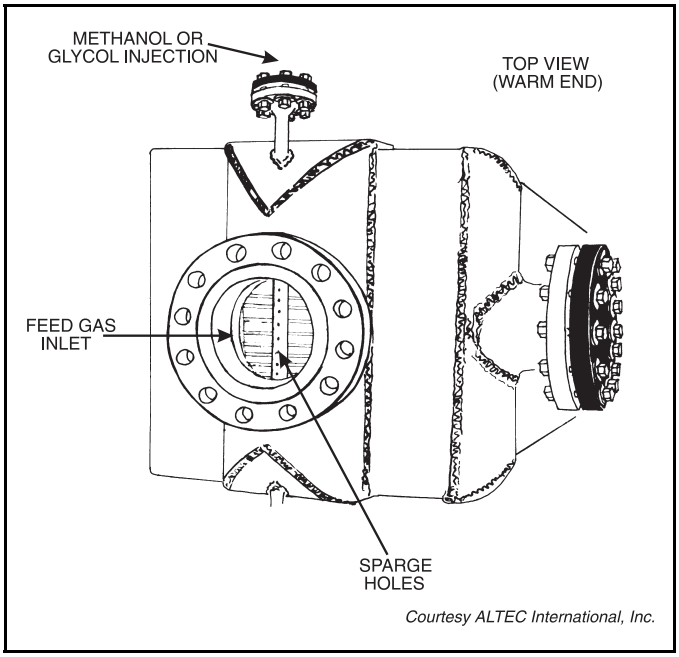
Hydrate Suppression

During start-up, upset, or even normal operating conditions, hydrates and/or heavy hydrocarbons may freeze out and block sections of the heat exchanger.

Injection sparge systems (see Fig. 9-41) are designed for injecting either methanol or glycol into the feed gas entering the exchanger. This method of hydrate suppression has proven effective.

FIG. 9-41

Typical Methanol or Glycol Injection Sparge System



Courtesy ALTEC International, Inc.

Cleaning

Only clean, dry fluids which are non-corrosive to aluminum should be used in brazed aluminum heat exchangers. The presence of particulates in the fluid resulting from start-up or mal-operation may not only lead to exchanger fouling but may also cause erosion in the high velocity areas of the exchanger. This can be prevented with proper filtering (177 micron screen-80 mesh Tyler standard, or finer) upstream of the brazed aluminum heat exchanger. A heavy duty, cleanable filter or strainer is strongly recommended on the inlet of all streams entering the exchanger.

Fouling which is caused by hydrate formation can be removed by warming the exchanger to ambient conditions. Deposits of heavy hydrocarbons, waxy materials, or compressor oils can be removed by a combination of warming and a solvent rinse. Solvents such as trichloroethane, toluene, or propylene are effective.

If plugging occurs, reverse gas flow, called puffing, is an effective method of removing particulate matter such as adsorbents, pipe scale, sand, or other solid debris. It involves the use of a calibrated rupture disk on the inlet nozzle of the plugged stream and one or more charges and ruptures to establish reverse flow with a dry gas until the particulate matter has been removed. Extreme care must be exercised to prevent exposure of personnel or equipment to explosive or toxic fluids and flying debris.

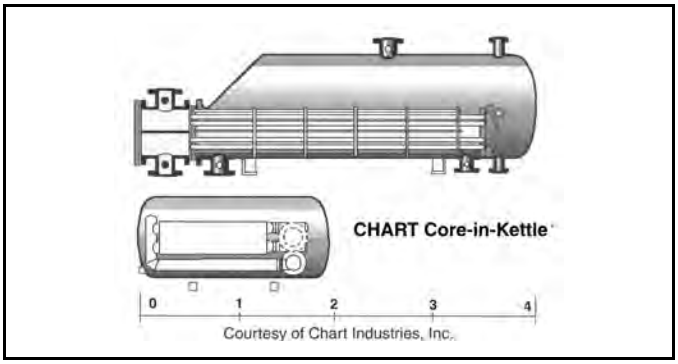
OTHER USES OF CORE BLOCKS

Core-in-Kettle©

These heat exchangers look and operate like "kettle" type shell and tube heat exchangers. Tube bundles are replaced with brazed aluminum cores. Because the cores contain about ten times more heat transfer surface area than comparable sized shell and tube exchangers, smaller core-in-kettle units can be used or greater heat transfer can be obtained which may also reduce system horsepower requirements. See Fig. 9-42. In a different physical configuration, cores can also be placed inside horizontal pressure vessels with standard elliptical heads. When these exchangers are used in reboiler applications, adequate vapor disengaging space must be provided above the core to prevent excessive liquid carryover. Brazed aluminum cores can also be placed at the top of a vertical distillation tower to allow overhead vapors to pass through on one side of a parting sheet with refrigerant on the other side.

FIG. 9-42

Relative Exchanger Sizes



Courtesy of Chart Industries, Inc.

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Any condensed liquid is allowed to run back down the exchanger and re-enter the tower proper as reflux.

As in brazed aluminum plate-fin exchangers, approach temperatures as low as 3°F on single phase fluids can be achieved. Applications for these heat exchangers include propane chillers, feed coolers, and tower reboilers.

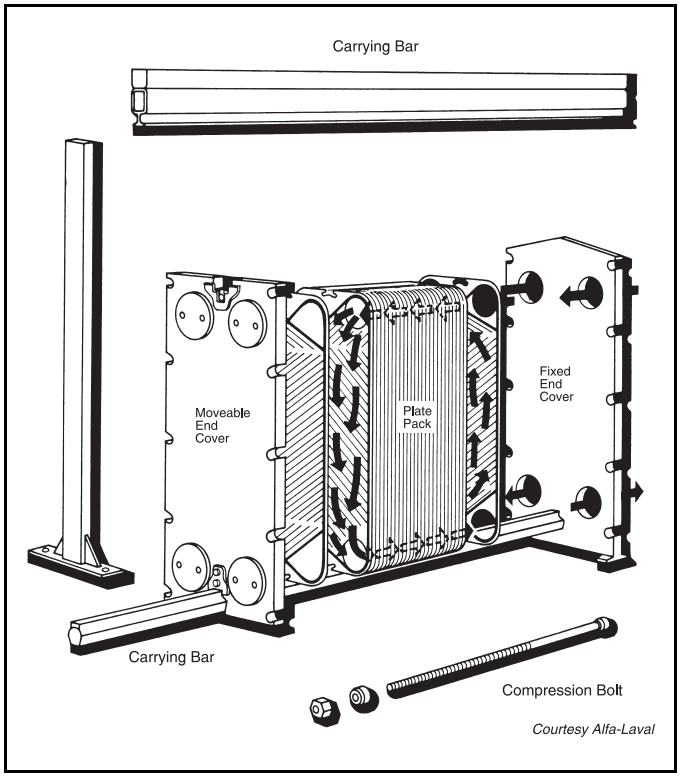
Plate Frame Heat Exchangers

A typical plate and frame heat exchanger "PHE" (sometimes referred to as a gasketed plate heat exchanger) is shown in an exploded view in Fig. 9-43. The PHE consists of an arrangement of gasketed pressed metal plates (heat transfer surface), aligned on two carrying bars, secured between two covers by compression bolts. Inlet and outlet ports for both hot and cold fluids are stamped into the corners of each plate. The ports are lined up to form distribution headers through the plate pack. All four fluid connections are usually located in the fixed end cover. This permits opening the exchanger without disconnecting any piping. Plates can be added and removed in the field should service requirements change. The plates are pressed into one of a number of available patterns and may be constructed of any material which can be cold formed to the desired pattern. The welding characteristics of the plate material are not of prime importance since very little or no welding is involved in plate construction.

Gasket grooves are pressed into the plates as they are formed. The gaskets are generally made of elastomers such as natural rubber, nitrile, butyl, neoprene, etc. The gasket material chosen depends on the temperature, pressure, and chemical characteristics of the fluid to which it will be exposed. The gasket cross-section varies with different plate designs and sizes. Rectangular, trapezoidal, or oval cross-sections are the most common. The width is generally 5-15 mm, depending on spacing. The height of the gasket before it is compressed is 15 to 50% higher than the spacing, depending on material, cross-section of gasket, gasket track, and gasket hardness. When the plate stack is compressed, the exposed surface of the gasket is very small. The gaskets are generally arranged in such a way that the through pass portal is sealed independently of the boundary gasket. Leaks from one fluid to the other cannot take place unless a plate develops a hole. Any leakage from the gaskets is to the outside of the exchanger where it is easily detected.

FIG. 9-43

Plate and Frame Heat Exchanger



Since the plates are generally designed to form channels giving highly turbulent flow, the PHE produces higher heat transfer coefficients for liquid flow than most other types. The high heat transfer coefficients are developed through the efficient use of pressure drop.

Advantages

The PHE has the following advantages over conventional shell and tube heat exchangers:

1. It can easily be disassembled for cleaning.
2. The plates can be rearranged, added to, or removed from the plate rack for difference service conditions.
3. The fluid residence time is short (low fluid volume to surface area ratio).
4. No hot or cold spots exist which could damage temperature sensitive fluids.
5. Fluid leakage between streams cannot occur unless plate material fails.
6. Fluid leakage due to a defective or damaged gasket is external and easily detected.
7. Low fouling is encountered due to the high turbulence created by the plates.
8. A very small plot area is required relative to a shell and tube type heat exchanger for the same service.
9. The maintenance service area required is within the frame size of the exchanger.

Disadvantages

1. Care must be taken by maintenance personnel to prevent damage to the gaskets during disassembly, cleaning, and reassembly.
2. A relatively low upper design temperature limitation exists.
3. A relatively low upper design pressure limitation exists.
4. Gasket materials are not compatible with all fluids.

Applications

The PHE is normally used in liquid services. This type of heat exchanger is considered to be a high heat transfer, high pressure drop device, but it can be used for services requiring a low pressure drop with the associated reduction in heat transfer coefficients.

Since the plates are thin, the PHE gives a relatively high heat transfer coefficient for the mass of material required. When alloy materials are required, the PHE is competitive with more conventional heat exchanger designs.

Materials of Construction

The frames are usually fabricated from carbon steel while the tension bolts are high tensile strength steel. Common plate materials include 304 and 316 stainless steel, titanium, In-

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coloy 825®, Hastelloy®, aluminum-bronze, tantalum, copper-nickel, aluminum, and palladium stabilized titanium. The PHE can be fabricated and stamped to the ASME Code.

Maximum Pressure and Temperature Ratings

Maximum allowable working pressure may be determined by frame strength, gasket retainment, or plate deformation resistance. Of these, it is often the frame that limits operating pressure, so that many manufacturers produce a low cost frame for low pressure duties (typically up to 85 psig) and a more substantial frame for higher pressures, for the same plate size. Maximum operating pressure for a typical PHE is normally 140-230 psig, but some types are designed to operate over 300 psig.

Normally, it is gaskets that limit the maximum operating temperature for a plate heat exchanger. Fig. 9-44 provides typical temperature limitations for common gasket materials not subject to chemical attack.

FIG. 9-44

Typical Gasket Material Temperature Limitations

Gasket Material	Temperature Limitation
Natural rubber, styrene, neoprene	160°F
Nitrile	210°F
Resin-cured butyl, viton	300°F
Ethylene/propylene, silicone	300°F
Compressed non-asbestos fiber	390°F

Size Limitations

The surface area per plate and number of plates per frame varies depending upon the manufacturer. Typically, surface area ranges between 0.4 and 39 square feet per plate. Frame sizes have been manufactured to contain up to 600 plates. When larger surface areas are required, multiple units are supplied.

Fouling Factors

Fouling factors required in the PHE are small compared with those commonly used in shell and tube designs for the following reasons:

1. High turbulence maintains solids in suspension.
2. Heat transfer surfaces are smooth. For some applications, a mirror finish may be available.
3. There are no “dead spaces” where fluids can stagnate, as for example, near the shell-side baffles in a tubular unit.
4. Since the plate is necessarily of a material not subject to massive corrosion, deposits of corrosion products to which fouling can adhere are absent.
5. High film coefficients lead to lower surface temperatures for the cold fluid (the cold fluid is usually the culprit as far as fouling is concerned).

Fig. 9-45 lists typical fouling factors for PHEs.

FIG. 9-45

Typical Fouling Factors for PHEs

Fluid	Fouling Factor Sq ft-°F-Hr/Btu
Water	
Demineralized or distilled	0.00001
Municipal supply (soft)	0.00002
Municipal supply (hard)	0.00005
Cooling tower (treated)	0.00004
Sea (coastal) or estuary	0.00005
Sea (ocean)	0.00003
River, canal, borehole, etc.	0.00005
Engine jacket	0.00006
Oils, lubricating	0.00002 to 0.00005
Solvents, organic	0.00001 to 0.00003
Steam	0.00001
Process fluids, general	0.00001 to 0.00006

PRINTED CIRCUIT HEAT EXCHANGERS

General

Printed Circuit Heat Exchangers (PCHEs) are highly compact, corrosion resistant heat exchangers capable of operating at pressures of several hundred atmospheres and temperatures ranging from cryogenic to several hundred degrees Fahrenheit.

PCHEs are constructed from flat alloy plates which have fluid flow passages photo-chemically machined (etched) into them. This process is similar to manufacturing electronic printed circuit boards, and gives rise to the name of the exchangers. In the case of PCHEs, it is fluid circuits which are formed by etching.

Stacks of etched plates, carrying flow passage designs tailored for each fluid, are interleaved and diffusion bonded together into solid blocks. Diffusion bonding is a solid state welding process in which the flat and clean metal surfaces are held together at high temperatures, resulting in interfacial crystal growth between the touching surfaces which gives rise to a bond strength equal to that of the bulk metal.

The thermal capacity of the exchanger is built to the required level by welding together diffusion bonded blocks to form the complete heat exchange core. Headers and nozzles are welded on to the core in order to direct the fluids to the appropriate sets of passages. Fig. 9-46 shows the complete construction of a two-fluid exchanger.

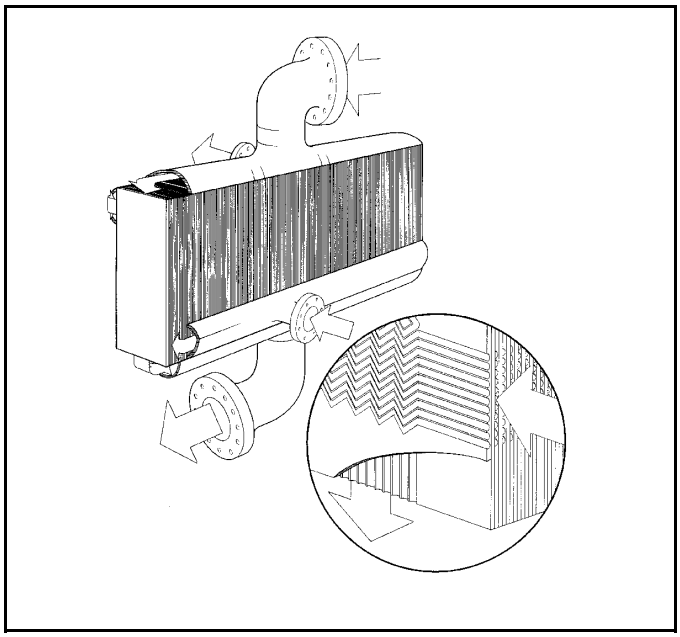
PCHEs are all-welded — there is no braze material employed in construction, and no gaskets are required. Hence the potential for leakage and fluid-compatability is reduced. In fact the high level of constructional integrity renders PCHEs exceptionally well suited to critical high pressure applications, such as high-pressure gas exchangers on offshore platforms.

The thermal design of PCHEs is subject to very few constraints. Fluids may be liquid, gas or two-phase, and they can exchange heat in counterflow, crossflow or coflow at any required pressure drop. Where energy is expensive, high heat exchange effectiveness can be achieved through very close temperature approaches in counterflow. To simplify control, or to further maximize energy efficiency, more than two fluids can exchange heat in a single core. Heat loads can vary from

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FIG. 9-46

Construction of a Two-fluid PCHE



a few Btu/hr to many Btu/hr, in exchangers weighing from a few pounds to over 50 tons.

Mechanical design is also flexible. Etching patterns can be adjusted to provide high pressure containment where required — design pressure may be several hundred atmospheres. The all-welded construction is compatible with very high temperature operation, and the use of austenitic stainless steels allows cryogenic application. It is worthy of note that vibration is absent from PCHEs as this can be an important source of failure in shell-and-tube exchangers.

Materials of construction include stainless steel and titanium as standard, with nickel and nickel alloys also commonly used.

Passages are typically of the order of 2000 microns semi-circular cross-section — that is, 2000 microns across and 1000 microns deep — for reasonably clean applications, although there is no absolute limit on passage size. The corrosion resistant materials of construction for PCHEs, the high wall shear stresses, and the absence of dead-spots assist in resisting fouling deposition.

Prime heat transfer surface densities, expressed in terms of effective heat transfer area per unit volume, can be up to 800 ft^2/ft^3. This is higher than prime surface densities in gasketed plate exchangers, and an order of magnitude higher than normal prime surface densities in shell-and-tube exchangers.

Design

Detailed thermal design of PCHEs is supported by proprietary design software developed by the manufacturers which allows for infinite geometric variation to passage arrangements in design optimization. Variations to passage geometry have negligible production cost impact since the only tooling required for each variation is a photographic transparency for the photo-chemical machining process.

Although the scope of PCHE capabilities is much wider, as a sizing guide it is safe to assume that channel patterns can be developed to mimic any j- and f-factors characteristics found in publications such as “Compact Heat Exchangers” by Kays and London for aluminum surfaces, or data presented by gasketed plate exchanger manufacturers.

It is rarely necessary to apply a correction factor substantially less than 1 to the LMTD calculated for an exchange, no matter how high the effectiveness required, because of the PCHE counterflow capabilities. Pressure drops can be specified at will; however, as with all heat exchangers, lower allowable pressure drops will result in low heat transfer coefficients and hence larger exchangers.

Applications

PCHEs extend the benefits of compact exchangers into applications where pressure, temperature or corrosivity prevent the use of conventional plate exchangers.

In hydrocarbons processing, PCHEs are employed with gas streams (possibly sour) in such areas as:

- compression aftercooling,
- gas/gas counterflow exchange for dewpoint control,
- cryogenic inerts removal and
- liquefaction.

The use of multi-stream contact in these duties is common. In refineries, suitable applications are to be found in light ends processing and feed-effluent exchange for platforming and HDS units.

Chemicals applications include duties requiring:

- high pressure capabilitiy, such as ammonia and methanol production,
- corrosion resistant materials, such as pure nickel for caustic soda and titanium for chloride environments, and
- high effectiveness counterflow contact, including heat recovery.

In power production, PCHEs are applicble as feedwater heaters, fuel gas heaters, water/water exchangers, and in various roles in non-conventional power production systems such as geothermal and solar.

EVAPORATIVE COOLING SYSTEMS

Evaporative cooling systems provide process fluid cooling without the use of a cooling tower and large water recirculating pumps. These systems combine the functions of an open cooling tower and heat exchanger by replacing the wet deck surface with a coil type heat exchanger. The liquid to be cooled is circulated through the coil or tube bundle, which is continually wetted on the outside by a recirculating water system. Air is simultaneously blown over the coil, causing a small portion of the recirculating water to evaporate. This evaporation removes heat from the water allowing for the cooling of the liquid in the coil. The process fluid outlet temperature’s approach to the ambient wet bulb temperature is determined on a case by case basis.

A variety of coil arrangements are available as well as alternate coil materials. ASME "U" stamping for coil construction is available where applicable. Fans provide the necessary air movement which can be either co-current or countercurrent to the water flow.

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SECTION 10

Air-Cooled Exchangers

An air-cooled exchanger (ACHE) is used to cool fluids with ambient air. Several articles have been published describing in detail their application and economic analysis. (See Bibliography at the end of this section.) This section describes the general design of air-cooled exchangers and presents a method of approximate sizing.

ARRANGEMENT AND MECHANICAL DESIGN

Figs. 10-2 and 10-3 show typical elevation and plan views of horizontal air-cooled exchangers as commonly used. The basic components are one or more tube sections served by one or more axial flow fans, fan drivers, speed reducers, and an enclosing and supporting structure.

Air-cooled exchangers are classed as forced draft when the tube section is located on the discharge side of the fan, and as induced draft when the tube section is located on the suction side of the fan.

Advantages of induced draft are:

- Better distribution of air across the section.
- Less possibility of the hot effluent air recirculating around to the intake of the sections. The hot air is dis-

FIG. 10-1

Nomenclature

A	=	area of heat transfer surface, sq ft
A_i	=	inside surface of tube, sq ft
A_b	=	outside bare tube surface, sq ft
A_x	=	outside extended surface of tube, sq ft
A_t	=	tube inside cross-sectional area, sq in. (see Fig. 9-25)
ACFM	=	actual cubic feet per minute
APF	=	total external area/ft of fintube, sq ft/ft
APSF	=	external area of fintube, sq ft/sq ft of bundle face area
AR	=	area ratio of fintube compared to the exterior area of 1 in. OD bare tube
B	=	correction factor, psi (see Fig. 10-14)
C_p	=	specific heat at average temperature, Btu/(lb • °F)
CMTD	=	corrected mean temperature difference, °F
dB(A)	=	overall weighted level of sound at a point distant from noise source based on "A" weighting system
D	=	fan diameter, ft
D_i	=	inside tube diameter, in.
D_o	=	outside tube diameter, in.
D_R	=	density ratio, the ratio of actual air density to the density of dry air at 70°F and 14.7 psia, 0.0749 lb/cu ft (see Fig. 10-16)
f	=	friction factor (see Fig. 10-15)
F	=	correction factor (see Fig. 10-8)
F_a	=	total face area of bundles, sq ft
F_p	=	air pressure drop factor, in. of water per row of tubes
FAPF	=	fan area per fan, ft^2/fan
FPM	=	fan tip speed, feet per minute
g	=	local acceleration due to gravity, ft/s^2
G	=	mass velocity, lb/(sq ft • s)
G_a	=	air face mass velocity, lb/(hr • sq ft) of face area
G_t	=	tubeside mass velocity, lb/(sq ft • s)
h_a	=	air side film coefficient Btu/(h • sq ft • °F)
h_s	=	shell side film coefficient based on outside tube area, Btu/(h • sq ft • °F)
h_t	=	tube side film coefficient based on inside tube area, Btu/(h • sq ft • °F)
HP	=	fan horsepower
J	=	J factor (see Fig. 10-13)
k	=	thermal conductivity, Btu/[(hr • sq ft • °F)/ft]
L	=	length of tube, ft
LMTD	=	log mean temperature difference, °F (see Fig. 9-3)
N	=	number of rows of tubes in direction of flow
N_f	=	number of fans
N_P	=	number of tube passes
N_R	=	modified Reynolds number, (in • lb/(sq ft • s • cp)
N_t	=	number of tubes
ΔP	=	pressure drop, psi
PF	=	fan total pressure, inches of water
ρ_a	=	density of air, lb/ cu ft
ρ_w	=	density of water, lb/ cu ft
P	=	temperature ratio (see Fig. 10-8)
PWL	=	sound pressure level
PWLN	=	PWL for N_f fans
Q	=	heat transferred, Btu/h
r_d	=	fouling resistance (fouling factor), (hr • ft^2• °F/Btu)
r_f	=	fluid film resistance (reciprocal of film coefficient)
r_{mb}	=	metal resistance referred to outside bare surface
r_{mx}	=	metal resistance referred to outside extended surface
R	=	distance in feet (see Eq. 10-6)
R	=	temperature ratio (see Fig. 10-8)
RPM	=	fan speed, rotations per minute
S	=	specific gravity (water = 1.0)
SPL	=	sound pressure level
t	=	temperature air side, °F
T	=	temperature tube side, °F
U	=	overall heat transfer coefficient, Btu/(h • ft^2 • °F)
W	=	mass flow, lb/hr

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FIG. 10-1

Nomenclature (Cont'd.)

Y	= correction factor, psi/ft (see Fig. 10-14)	**Subscripts:**	
Δt	= temperature change, °F	a	= air side
μ	= viscosity, cp	b	= bare tube surface basis
μ_w	= viscosity at average tube wall temperature, cp	s	= shell side
φ	= viscosity gradient correction	t	= tube side
		x	= extended tube surface basis
		1	= inlet
		2	= outlet

charged upward at approximately 2½ times the velocity of intake, or about 1500 ft/min.

- Less effect of sun, rain, and hail, since 60% of the face area of the section is covered.
- Increased capacity in the event of fan failure, since the natural draft stack effect is much greater with induced draft.

Disadvantages of induced draft are:

- Higher horsepower since the fan is located in the hot air.
- Effluent air temperature should be limited to 200°F, to prevent potential damage to fan blades, bearings, V-belts, or other mechanical components in the hot air stream.
- The fan drive components are less accessible for maintenance, which may have to be done in the hot air generated by natural convection.
- For inlet process fluids above 350°F, forced draft design should be used; otherwise, fan failure could subject the fan blades and bearings to excessive temperatures.

Advantages of forced draft are:

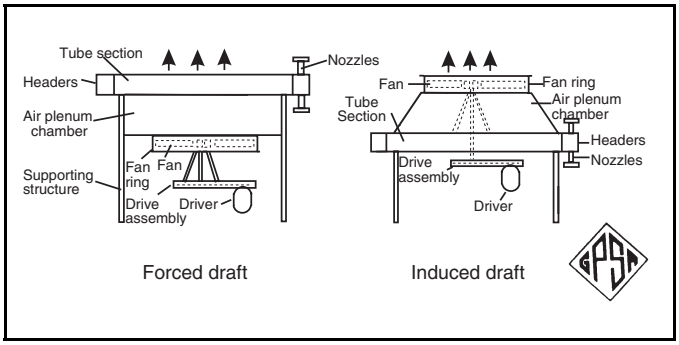
- Slightly lower horsepower since the fan is in cold air. (Horsepower varies directly as the absolute temperature.)
- Better accessibility of mechanical components for maintenance.
- Easily adaptable for warm air recirculation for cold climates.

The disadvantages of forced draft are:

- Poor distribution of air over the section.

FIG. 10-2

Typical Side Elevations of Air Coolers



- Greatly increased possibility of hot air recirculation, due to low discharge velocity from the sections and absence of stack.
- Low natural draft capability on fan failure due to small stack effect.
- Total exposure of tubes to sun, rain, and hail.

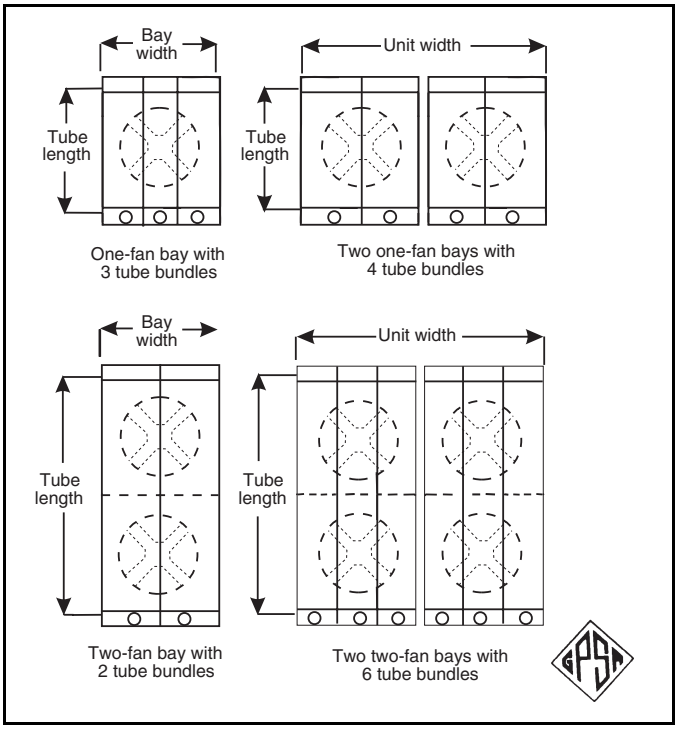
The horizontal section is the most commonly used air cooled section, and generally the most economical. For a fluid with freezing potential, the tubes should be sloped at least ⅛ in. per foot to the outlet header. Since in most cases there will be no problem associated with freezing, and it is more costly to design a sloped unit, most coolers are designed with level sections.

Vertical sections are sometimes used when maximum drainage and head are required, such as for condensing services.

Angled sections, like vertical sections, are used for condensing services, allowing positive drainage. Frequently, angle sections are sloped thirty degrees (30°) from the horizontal.

FIG. 10-3

Typical Plan Views of Air Coolers



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A-frames are usually sloped sixty degrees (60°) from the horizontal. See Fig. 10-4.

Fan sizes range from 3 ft to 28 ft diameter. However, 14 ft to 16 ft diameter is the largest diameter normally used. Fan drivers may be electric motors, steam turbines, hydraulic motors, or gas-gasoline engines. A speed reducer, such as a V-belt drive or reduction gear box, is necessary to match the driver output speed to the relatively slow speed of the axial flow fan. Fan tip speeds are normally 12,000 ft/min or less. General practice is to use V-belt drives up to about 30 bhp and gear drives at higher power. Individual driver size is usually limited to 50 hp.

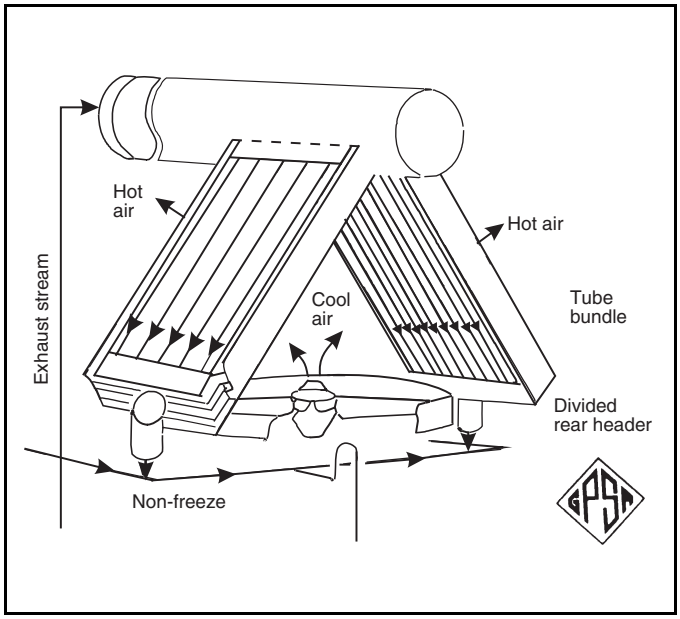
Two fan bays are popular, since this provides a degree of safety against fan or driver failure and also a method of control by fan staging. Fan coverage is the ratio of the projected area of the fan to the face of the section served by the fan. Good practice is to keep this ratio above 0.40 whenever possible because higher ratios improve air distribution across the face of the tube section. Face area is the plan area of the heat transfer surface available to air flow at the face of the section.

The heat-transfer device is the tube section, which is an assembly of side frames, tube supports, headers, and fin tubes. Aluminum fins are normally applied to the tubes to provide an extended surface on the air side, in order to compensate for the relatively low heat transfer coefficient of the air to the tube. Fin construction types are tension-wrapped, embedded, extruded, and welded.

Tension-wrapped is probably the most common fin type used because of economics. Tension wrapped tubing is common for continuous service with temperatures below 400°F. Extruded fin is a mechanical bond between an inner tube exposed to the process and an outer tube or sleeve (usually aluminum) which is extruded into a high fin. Embedded fin is an aluminum or steel fin grooved into the base tube. Embedded fins are used in cyclic and high temperature services. Other types of finned tubes available are soldered, edge wrapped, and serrated tension wrapped. Coolers are regularly manufactured in tube lengths from 6 ft to 50 ft and in bay widths from 4 ft to 30 ft.

FIG. 10-4

Angled Section Layout



Use of longer tubes usually results in a less costly design compared to using shorter tubes.

Base tube diameters are $\frac{5}{8}$ in. to $1\frac{1}{2}$ in. OD with fins from $\frac{1}{2}$ in. to 1 in. high, spaced from 7 to 11 per inch, providing an extended finned surface of 12 to 25 times the outside surface of the base tubing. Tubes are usually arranged on triangular pitch with the fin tips of adjacent tubes touching or separated by from $\frac{1}{16}$ in. to $\frac{1}{4}$ in. Matching of the tube section to the fan system and the heat transfer requirements usually results in the section having depth of 3 to 8 rows of fin tubes, with 4 rows the most typical.

A 1-in. OD tube is the most popular diameter, and the most common fins are $\frac{1}{2}$ in. or $\frac{5}{8}$ in. high. The data presented in Fig. 10-11 are for 1 in. OD tubes with $\frac{1}{2}$ in. high fins, 9 fins/in. (designated as $\frac{1}{2}$ x 9) and $\frac{5}{8}$ in. high fins, 10 fins/in. (designated as $\frac{5}{8}$ x 10).

Common materials of construction for headers are firebox quality carbon steel, ASTM SA-515-70, SA-516-70. Tubes are generally ASTM SA-214 (ERW), SA-179 (SMLS), carbon steel. Louvers are generally carbon steel, or aluminum with carbon steel construction being the most general and most economical. Fins are normally aluminum. Both stainless and brass alloys have their applications but are more expensive than carbon steel.

HEADER DESIGN

Plug header construction uses a welded box which allows partial access to tubes by means of shoulder plugs opposite the tubes. Plug headers are normally used as they are cheaper than the alternate cover plate design. Cover plate header construction allows total access to header, tube sheet, and tubes. This design is used in high fouling, low pressure service.

Fig. 10-5 shows typical designs for both plug header and cover plate header.

AIR-SIDE CONTROL

Air-cooled exchangers are sized to operate at warm (summer) air temperatures. Seasonal variation of the air temperature can result in over-cooling which may be undesirable. One way to control the amount of cooling is by varying the amount of air flowing through the tube section. This can be accomplished by using multiple motors, 2-speed drives, variable speed motors, louvers on the face of the tube section, or variable pitch fans.

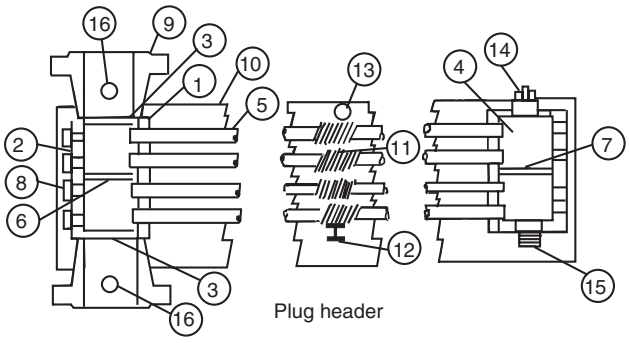
Staging of fans or fan speeds may be adequate for systems which do not require precise control of process temperature or pressure. Louvers will provide a full range of air quantity control. They may be operated manually, or automatically operated by a pneumatic or electric motor controlled from a remote temperature or pressure controller in the process stream. Louvers used with constant speed fans do not reduce fan power requirements.

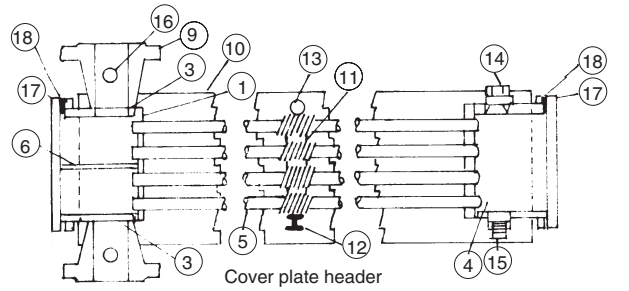
Auto-variable-pitch fans are normally provided with pneumatically operated blade pitch adjustment which may be controlled from a remote sensor. Blade pitch is adjusted to provide the required amount of air flow to maintain the process temperature or pressure at the cooler. The required blade angle decreases as ambient air temperature drops and this conserves fan power. Hydraulic variable speed drives reduce fan

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FIG. 10-5

Typical Construction of Tube Section with Plug and Cover Plate Headers





1. Tube sheet
2. Plug sheet
3. Top and bottom plates
4. End plate
5. Tube
7. Stiffener
8. Plug
9. Nozzle
10. Side frame
11. Tube spacer
13. Tube keeper
14. Vent
15. Drain
16. Instrument connection
17. Cover plate

speed when less air flow is required and can also conserve fan power.

A design consideration which might be required for satisfactory process fluid control is co-current flow. In extreme cases of high pour point fluids, no amount of air side control would allow satisfactory cooling and prevent freezing. Co-current flow has the coldest air cool the hottest process fluid, while the hottest air cools the coolest process fluid. This is done in order to maintain a high tube wall temperature. This gives a much poorer LMTD, but for highly viscous fluids is often the only way to prevent freezing or unacceptable pressure drops. With air coolers, the most common method of accomplishing co-current flow is to have the inlet nozzle on the bottom of the header with the pass arrangement upwards. This totally reverses the standard design, and may cause a problem with drainage during shut-downs. In addition, air side control is necessary with co-current designs.

WARM AIR RECIRCULATION

Extreme variation in air temperature, such as encountered in northern climates, may require special air recirculation features. These are needed to provide control of process stream temperatures, and to prevent freezing of liquid streams. Warm air recirculation varies from a standard cooler with one reversing fan to a totally enclosed system of automatic louvers and fans. These two widely used systems are termed “internal recirculation” and “external recirculation.”

A typical layout for internal recirculation is shown in Fig. 10-6. During low ambient operation, the manual fan continues to force air through the inlet half of the section. The auto-variable fan operates in a reversing mode, and draws hot air from the upper recirculation chamber down through the outlet end of the section. Because of the lower recirculation skirt, the manual fan mixes some of the hot air brought down by the auto-variable fan with cold outside air and the process repeats. The top exhaust louvers are automatically adjusted by a temperature controller sensing the process fluid stream. As the fluid temperature rises, the louvers are opened. During design ambient conditions, the louvers are full open and both fans operate in a standard forced draft mode.

A cooler with internal recirculation is a compromise between no recirculation and fully controlled external recirculation. It is cheaper than full external recirculation, and has less static pressure loss during maximum ambient temperature conditions. A cooler with internal recirculation is easier to erect, and requires less plot area than an external recirculation design. However, this latter design is more costly than a cooler with no recirculation, and cannot provide complete freeze protection. Because there is no control over air intake, and fans alone cannot fully mix air, stratified cold air may contact the section. With the fans off, high wind velocity during low ambient conditions could cause excessive cold air to reach the section.

A typical layout for external recirculation is shown in Fig. 10-7. During low ambient temperature conditions, two-speed motors on low speed, or auto-variable fans at low pitch, are normally used. For this design, the sides of the cooler are closed with manual louvers. Over both ends, a recirculation chamber projects beyond the section headers, and provides a duct for mixing cold outside air with warm recirculated air. As with the internal recirculation design, the top exhaust louvers are controlled by the temperature of the process fluid. However, this design provides for control of the inlet air temperature. As the inlet air louver closes, an internal louver in the end duct opens. These adjustments are determined by a controller which senses air temperature at the fan. Once the system reaches equilibrium, it automatically controls process temperature and prevents excessive cooling. During warm weather, the side manual louvers are opened, while close control is maintained by adjustment of the exhaust louvers.

The external recirculation design is preferred for critical control and prevention of freezing. Once operational, it requires little attention. Upon failure of power or air supply, the system closes automatically to prevent freezing. It can be designed to automatically reduce motor energy use when excess cooling is being provided. The main drawback for this type of system is its high cost. Several actuators and control devices are required, along with more steel and louvers. It is usually too large to be shop assembled, and requires more field assembly than an internal system. Because of the need to restrict air intake, this design increases the static pressure, causing greater energy use, and 20-25% larger motors than a standard cooler.

When designing an external recirculation unit, consideration must be given to the plenum depth and duct work to allow air mixing and prevent excessive static pressure loss. The louver intake area should be large enough to keep the air flow below 500 ft/min during maximum design conditions.

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FIG. 10-6

Internal Recirculation Design

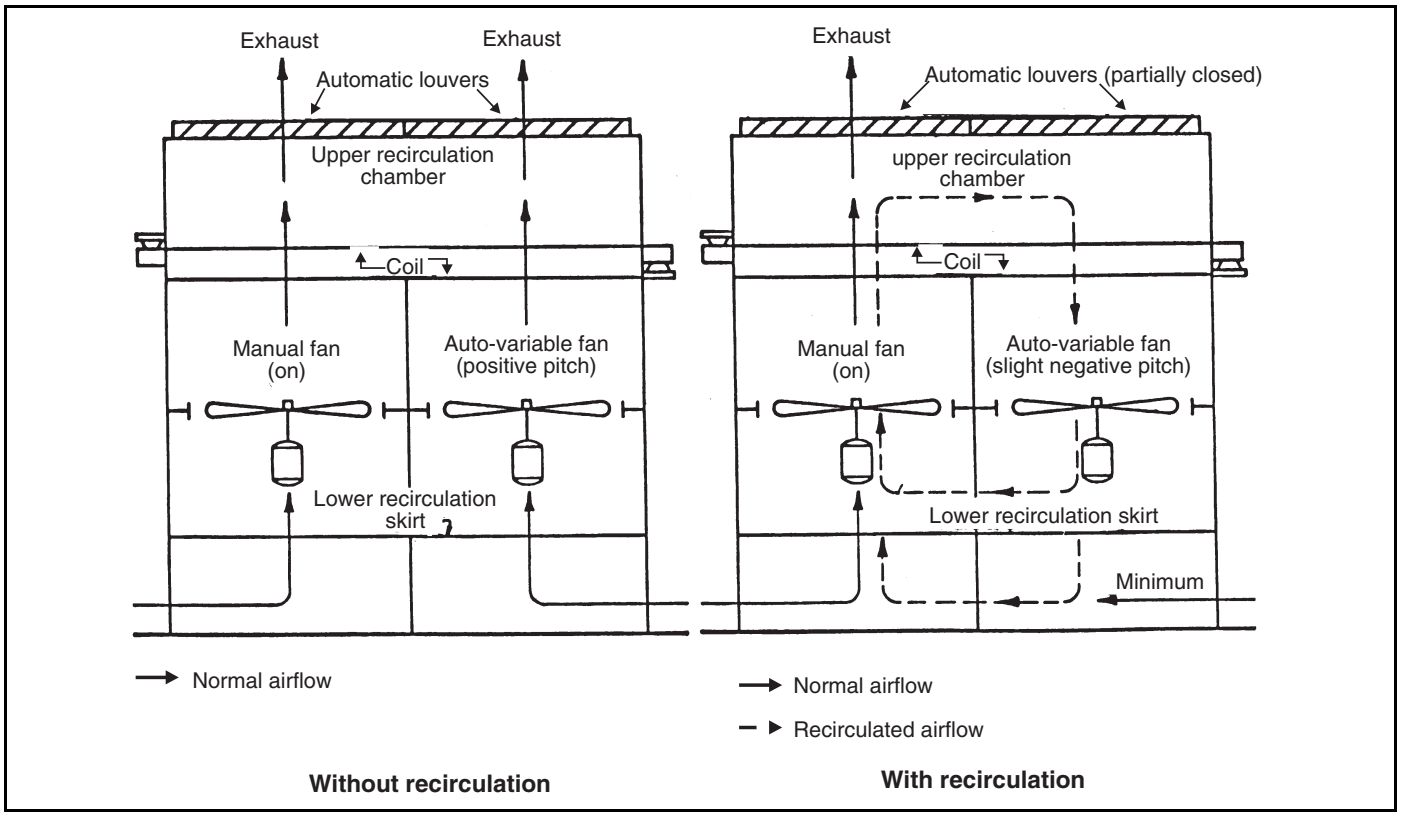
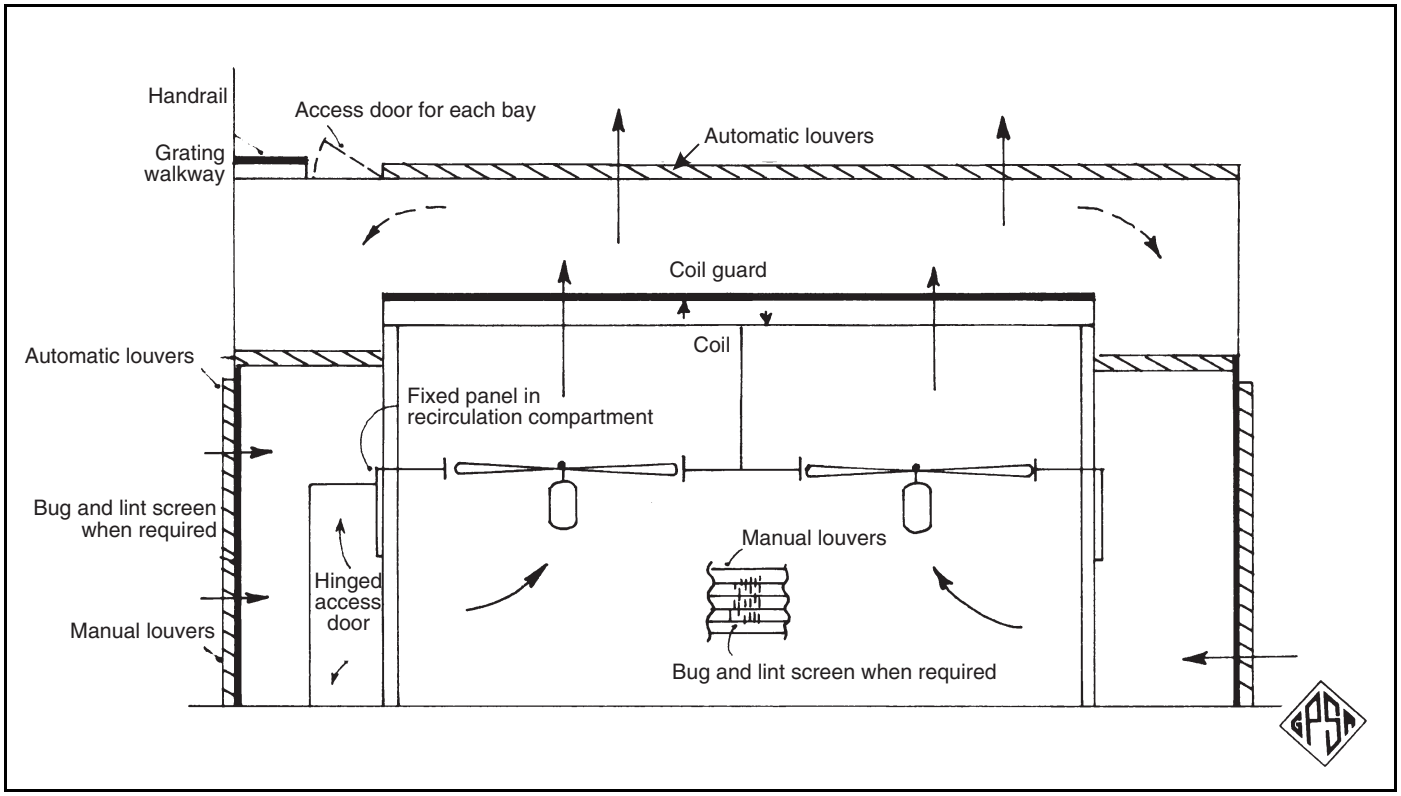


FIG. 10-7

External Recirculation Design



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AIR EVAPORATIVE COOLERS

Wet/dry type (air evaporative coolers) air coolers may be a good economical choice when a close approach to the ambient temperature is required. In these systems, the designer can take advantage of the difference between the dry bulb and wet bulb temperatures. There are two general types of air evaporative cooler combinations used although other combinations are possible:

Wet air type — In this type, the air is humidified by spraying water into the air stream on the inlet side of the air cooler. The air stream may then pass through a mist eliminator to remove the excess water. The air then passes over the finned tubes at close to its wet-bulb temperature. If the mist eliminator is not used, the spray should be clean, treated water or the tube/fin type and metallurgy should be compatible with the water.

Wet tube type — An air evaporative cooler may be operated in series with an air cooler if there is a large process fluid temperature change with a close approach to the ambient. The process fluid enters a dry finned tube section and then passes into a wet, plain tube section (or appropriate finned tube section). The air is pulled across the wet tube section and then, after dropping out the excess moisture, passes over the dry tube section.

SPECIAL PROBLEMS IN STEAM CONDENSERS

There are often problems with steam condensers which need special attention at the design stage.

Imploding (collapsing bubbles) or knocking can create violent fluid forces which may damage piping or equipment. These forces are created when a subcooled condensate is dumped into a two-phase condensate header, or when live steam passes into subcooled condensate. This problem is avoided by designing the steam system and controls so that steam and subcooled condensate do not meet in the system.

Non-condensable gas stagnation can be a problem in the air cooled steam condenser any time there is more than one tube row per pass. The temperature of the air increases row by row from bottom to top of the air cooled section. The condensing capacity of each row will therefore vary with each tube row in proportion to the ΔT driving force. Since the tubes are connected to common headers and are subject to the same pressure drop, the vapor flows into the bottom rows from both ends. The non-condensables are trapped within the tube at the point of lowest pressure. The non-condensables continue to accumulate in all but the top rows until they reach the tube outlet. The system becomes stable with the condensate running out of these lower tube rows by gravity. This problem can be eliminated in several ways:

- By assigning only one tube row per pass.
- By connecting the tube rows at the return end with 180° return bends and eliminating the common header.

AIR COOLER LOCATION

Circulation of hot air to the fans of an air cooler can greatly reduce the cooling capacity of an air cooler. Cooler location should take this into consideration.

Single Installations

Avoid locating the air-cooled exchanger too close to buildings or structures in the downwind direction. Hot air venting from the air cooler is carried by the wind, and after striking the obstruction, some of the hot air recycles to the inlet. An induced draft fan with sufficient stack height alleviates this problem, but locating the air cooler away from such obstructions is the best solution.

An air cooler with forced draft fans is always susceptible to air recirculation. If the air cooler is located too close to the ground, causing high inlet velocities relative to the exhaust air velocity leaving the cooler, the hot air recirculation can become very significant. Forced draft coolers are preferably located above pipe lanes relatively high above the ground. Induced draft coolers are less likely to experience recirculation because the exhaust velocities are normally considerably higher than the inlet velocities.

Banks of Coolers

Coolers arranged in a bank should be close together or have air seals between them to prevent recirculation between the units. Mixing of induced draft and forced draft units in close proximity to each other invites recirculation. Avoid placing coolers at different elevations in the same bank.

Avoid placing the bank of coolers downwind from other heat generating equipment.

Since air can only enter on the ends of coolers in a bank, the bank should be located above ground high enough to assure a reasonably low inlet velocity.

The prevailing summer wind direction can have a profound effect on the performance of the coolers. Normally the bank should be oriented such that the wind flows parallel to the long axis of the bank of coolers, and the items with the closest approach to the ambient temperature should be located on the upwind end of the bank.

These generalizations are helpful in locating coolers. The use of Computational Fluid Dynamics to study the effect of wind direction, velocity, obstructions, and heat generating objects should be considered to assure the best location and orientation of air cooled heat exchangers, especially for large installations.

MULTIPLE SERVICE DISCUSSION

If different services can be placed in the same plot area without excessive piping runs, it is usually less expensive to combine them on one structure, with each service having a separate section, but sharing the same fan and motors. Separate louvers may be placed on each service to allow independent control. The cost and space savings makes this method common practice in the air cooler industry.

In designing multiple service coolers, the service with the most critical pressure drop should be calculated first. This is because the pressure drop on the critical item might restrict the maximum tube length that the other services could tolerate. The burden of forcing more than one service into a single tube length increases the possibility of design errors. Several trial calculations may be needed to obtain an efficient design.

After all service plot areas have been estimated, combine them into a unit having a ratio of 2 or 3 to 1 in length to width (assuming a two fan cooler). After assuming a tube length, calculate the most critical service for pressure drop using the assumed number and length of tubes and a single pass. If the

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drop is acceptable or very close, calculate the critical service completely. Once a design for the most critical service has been completed, follow the same procedure with the next most critical service. After the second or subsequent services have been rated, it is often necessary to lengthen or shorten the tubes or change the overall arrangement. If tubes need to be added for pressure drop reductions in already oversurfaced sections, it might be more cost effective to add a row(s) rather than widen the entire unit. The fan and motor calculations are the same as for a single service unit, except that the quantity of air used must be the sum of air required by all services.

CONDENSING DISCUSSION

The example given covers cooling problems and would work with straight line condensing problems that have the approximate range of dew point to bubble point of the fluid. Where de-superheating or subcooling or where disproportionate amounts of condensing occur at certain temperatures, as with steam and non-condensables, calculations for air coolers should be done by "zones." A heat release curve developed from enthalpy data will show the quantity of heat to be dissipated between various temperatures. The zones to be calculated should be straight line zones; that is, from the inlet temperature of a zone to its outlet, the heat load per degree temperature is the same.

After the zones are determined, an approximate rate must be found for each zone. Do this by taking rates from vapor cooling, condensing, and liquid cooling, then average these based on the percent of heat load for that phase within the zone. Next, calculate the LMTD of each zone. Begin with the outlet zone using the final design outlet temperature and the inlet temperature of that zone. Continue to calculate the zone as if it were a cooler, except that only one pass and one or two rows should be assumed, depending on the percentage of heat load in that zone. In calculating the pressure drop, average conditions may be used for estimating.

If the calculations for zone one (or later a succeeding zone) show a large number of short tubes with one pass, as is usually the case with steam and non-condensables, recalculate the zone with multiple rows (usually four) and short tubes having one pass that uses only a percentage of the total pressure drop allowed. The total cooler will be calculated as if each zone were a cooler connected in series to the next one, except that only tube pressure drops should be calculated for the middle zones. Thus, each zone must have the same number of tubes and true ambient must be used in calculating the LMTD. Only the tube length may vary, with odd lengths for a zone acceptable as long as overall length is rounded to a standard tube length.

If the calculations for zone one (and succeeding zones) fit well into a longer tube length, the LMTD must be weighted. After the outlet zone has been calculated, calculate zone two using the inlet temperature for it and its outlet temperature, which is the inlet temperature of zone one. The "ambient" used to find the zone two LMTD will be the design ambient plus the air rise from zone one. Continue in this manner, always using the previous zone's outlet air temperature in calculating the current zone's LMTD. After the cooler size and configuration have been determined, the fan and motor calculations will be made in the normal manner.

The ultimate pressure drop is the sum of the drops for each zone or approximately the sum of the drop for each phase using the tube length and pass arrangement for each phase. An estimated overall tube side coefficient may be calculated by estimating the coefficient for each phase. Then take a weighted average based on the percentage of heat load for each phase. The total LMTD must be the weighted average of the calculated zone LMTDs.

THERMAL DESIGN

The basic equation to be satisfied is the same as given in Section 9, Heat Exchangers:

$$Q = UA\ CMTD \qquad \textbf{Eq 10-1}$$

Normally Q is known, U and CMTD are calculated, and the equation is solved for A. The ambient air temperature to be used will either be known from available plant data or can be selected from the summer dry bulb temperature data given in Section 11, Cooling Towers. The design ambient air temperature is usually considered to be the dry bulb temperature that is exceeded less than 5 percent of the time in the area where the installation is required.

A complication arises in calculating the LMTD because the air quantity is a variable, and therefore the air outlet temperature is not known. The procedure given here starts with a step for approximating the air-temperature rise. After the air-outlet temperature has been determined, the corrected LMTD is calculated in the manner described in the shell and tube section, except that MTD correction factors to be used are from Figs. 10-8 and 10-9 which have been developed for the cross-flow situation existing in air-cooled exchangers.

Fig. 10-8 is for one tube pass. It is also used for multiple tube passes if passes are side by side. Fig. 10-9 is for two tube passes and is used if the tube passes are over and under each other. A MTD correction factor of 1.0 is used for four or more passes, if passes are over and under each other. A correction factor of 1.0 may be used as an approximation for three passes, although the factor will be slightly lower than 1.0 in some cases.

The procedure for the thermal design of an air cooler consists of assuming a selection and then proving it to be correct. The typical overall heat transfer coefficients given in Fig. 10-10 are used to approximate the heat transfer area required. The heat transfer area is converted to a bundle face area using Fig. 10-11 which lists the amount of extended surface available per square foot of bundle area for two specific fin tubes on two different tube pitches for 3, 4, 5, and 6 rows. After assuming a tube length, Fig. 10-11 is also used to ascertain the number of tubes. Both the tube side and air side mass velocities are now determinable.

The tube-side film coefficient is calculated from Figs. 10-12 and 10-13. Fig. 10-17 gives the air-side film coefficient based on outside extended surface. Since all resistances must be based on the same surface, it is necessary to multiply the reciprocal of the tube-side film coefficient and tube-side fouling factor by the ratio of the outside surface to inside surface. This results in an overall transfer rate based on extended surface, designated as U_x. The equation for overall heat transfer rate is:

$$\frac{1}{U_x} = \left(\frac{1}{h_t}\right)\left(\frac{A_x}{A_i}\right) + r_{dt}\left(\frac{A_x}{A_i}\right) + r_{mx} + \frac{1}{h_a} \qquad \textbf{Eq 10-2}$$

The basic equation will then yield a heat transfer area in extended surface, A_x, and becomes:

$$Q = (U_x)\,(A_x)\,CMTD$$

Either method is valid and each is used extensively by thermal design engineers. Fig. 10-10 gives typical overall heat

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FIG. 10-8

MTD Correction Factors (1 Pass – Cross Flow, Both Fluids Unmixed)

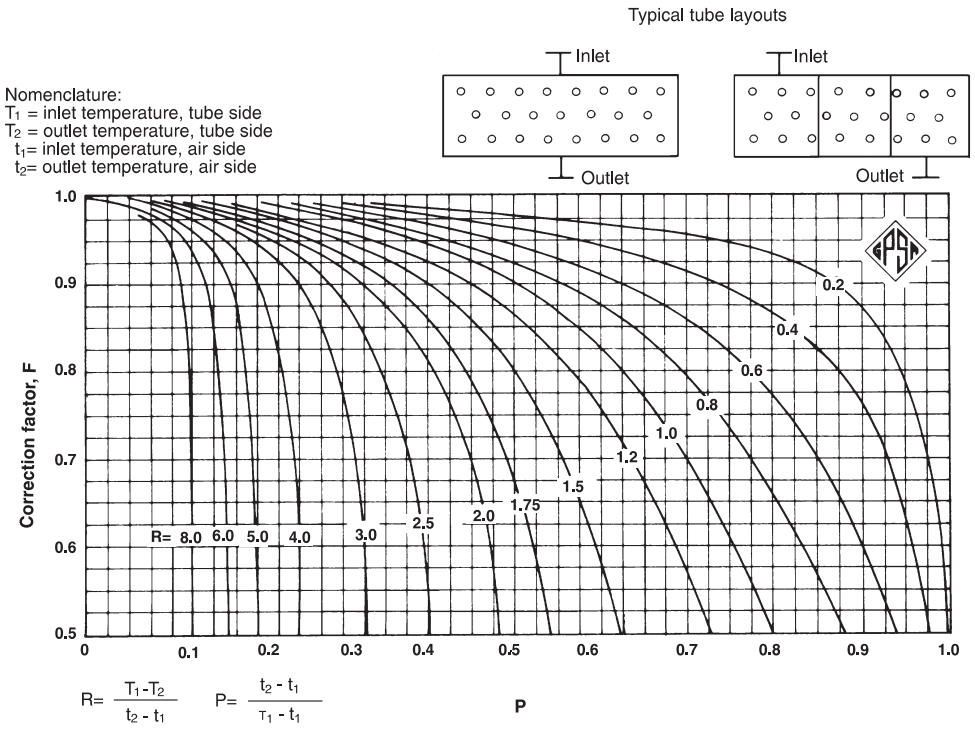
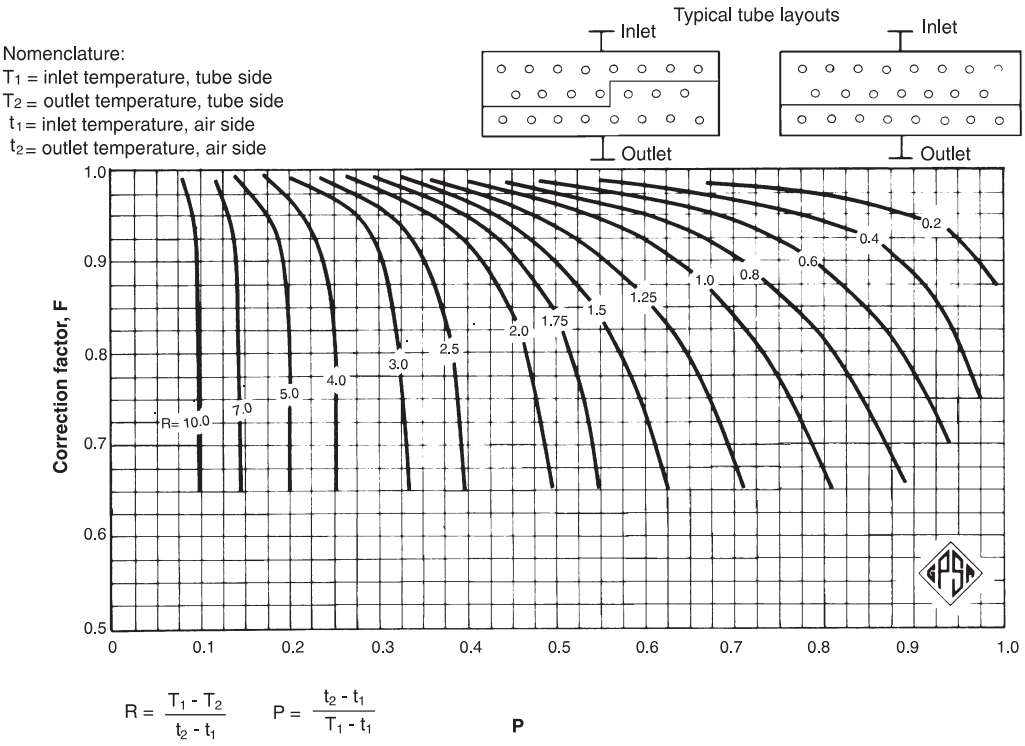


FIG. 10-9

MTD Correction Factors (2 Pass – Cross Flow, Both Fluids Unmixed)



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FIG. 10-10

Typical Overall Heat-Transfer Coefficients for Air Coolers

Service	1 in. Fintube			
	½ in. by 9		⅝ in. by 10	
	U_b	U_x	U_b	U_x
1. Water & water solutions				
	(See note below)			
Engine jacket water (r_d = 0.001)	110	7.5	130	6.1
Process water (r_d = 0.002)	95	6.5	110	5.2
50-50 ethylene glycol-water (r_d = 0.001)	90	6.2	105	4.9
50-50 ethylene glycol-water (r_d = 0.002)	80	5.5	95	4.4
2. Hydrocarbon liquid coolers				
Viscosity, cp, at avg. temp.	U_b	U_x	U_b	U_x
0.2	85	5.9	100	4.7
0.5	75	5.2	90	4.2
1.0	65	4.5	75	3.5
2.5	45	3.1	55	2.6
4.0	30	2.1	35	1.6
6.0	20	1.4	25	1.2
10.0	10	0.7	13	0.6
3. Hydrocarbon gas coolers				
Pressure, psig	U_b	U_x	U_b	U_x
50	30	2.1	35	1.6
100	35	2.4	40	1.9
300	45	3.1	55	2.6
500	55	3.8	65	3.0
750	65	4.5	75	3.5
1000	75	5.2	90	4.2
4. Air and flue-gas coolers Use one-half of value given for hydrocarbon gas coolers.				
5. Steam Condensers (Atmospheric pressure & above)				
	U_b	U_x	U_b	U_x
Pure Steam (r_d = 0.0005)	125	8.6	145	6.8
Steam with non-condensibles	60	4.1	70	3.3
6. HC condensers				
Condensing* Range, °F	U_b	U_x	U_b	U_x
0° range	85	5.9	100	4.7
10° range	80	5.5	95	4.4
25° range	75	5.2	90	4.2
60° range	65	4.5	75	3.5
100° & over range	60	4.1	70	3.3
7. Other condensers				
	U_b	U_x	U_b	U_x
Ammonia	110	7.6	130	6.1
Freon 12	65	4.5	75	3.5

Notes: U_b is overall rate based on bare tube area, and U_x is overall rate based on extended surface.

Based on approximate air face mass velocities between 2600 and 2800 lb/(hr.sq ft of face area).

*Condensing range = hydrocarbon inlet temperature to condensing zone minus hydrocarbon outlet temperature from condensing zone.

transfer coefficients based on both extended surface and outside bare surface, so either method may be used. The extended surface method has been selected for use in the example which follows. The air-film coefficient in Fig. 10-17 and the air static pressure drop in Fig. 10-18 are only for 1 in. OD tubes with ⅝ in. high fins, 10 fins per inch on 2¼ in. triangular pitch. Refer to Bibliography Nos. 2, 3, and 5 for information on other fin configurations and spacings.

The minimum fan area is calculated in Step 16 using the bundle face area, number of fans, and a minimum fan coverage of 0.40. The calculated area is then converted to a diameter and rounded up to the next available fan size. The air-side static pressure is calculated from Fig. 10-18 and the fan total pressure is estimated using gross fan area in Step 20. Finally, fan horsepower is calculated in Step 21 assuming a fan efficiency of 70%, and driver horsepower is estimated by assuming a 92%-efficient speed reducer.

Example 10-1 — Procedure for estimating transfer surface, plot area, and horsepower

Required data for hot fluid

Name and phase: 48°API hydrocarbon liquid

Physical properties at avg temp = 200°F

C_p = 0.55 Btu/(lb • °F)

μ = 0.51 cp

k = 0.0766 Btu/[(hr • sq ft • °F)/ft]

(From this Data Book Section 23)

Heat load: Q = 15,000,000 Btu/hr

Flow quantity: W_t = 273,000 lb/hr

Temperature in: T_1 = 250°F

Temperature out: T_2 = 150°F

Fouling factor r_{dt} = 0.001 (hr • sq ft • °F)/Btu

Allowable pressure drop: ΔP_t = 5 psi

Required data for air

Ambient temperature: t_1 = 100°F

Elevation: Sea level (see Fig. 10-16 for altitude correction)

C_{Pair} = 0.24 Btu/(lb • °F)

Basic assumptions

Type: Forced draft, 2 fans

Fintube: 1 in. OD with ⅝ in. high fins

Tube pitch: 2 ½ in. triangular (Δ)

Bundle layout: 3 tube passes, 4 rows of tubes, 30 ft long tubes

First trial

1. Pick approximate overall transfer coefficient from Fig. 10-10. U_x = 4.2

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FIG. 10-11

Fintube Data for 1-in. OD Tubes

Fin Height by Fins/inch	½ in. by 9		⅝ in. by 10		
APF, sq ft/ft	3.80		5.58		
AR, sq ft/sq ft	14.5		21.4		
Tube Pitch	2 in. Δ	2¼ in. Δ	2¼ in. Δ	2⅜ in. Δ	2½ in. Δ
APSF (3 rows)	68.4	60.6	89.1	84.8	80.4
(4 rows)	91.2	80.8	118.8	113.0	107.2
(5 rows)	114.0	101.0	148.5	141.3	134.0
(6 rows)	136.8	121.2	178.2	169.6	160.8

Notes: APF is total external area/ft of fintube in sq ft/ft. AR is the area ratio of fintube compared to the exterior area of 1 in. OD bare tube which has 0.262 sq ft/ft. APSF is the external area in sq ft/sq ft of bundle face area.

2. Calculate approximate air temperature rise

$$\Delta t_a = \left(\frac{U_x + 1}{10}\right)\left(\frac{T_1 + T_2}{2} - t_1\right)$$

$$\Delta t_a = \left(\frac{4.2 + 1.0}{10}\right)\left(\frac{250 + 150}{2} - 100\right) = 52°F$$

3. Calculate CMTD

Hydocarbon	250	→	150
Air	152	←	100
	98		50

LMTD = 71.3°F (see Fig. 9-3)

CMTD = (71.3) (1.00) = 71.3°F
(3 tube passes assumed)

4. Calculate required surface

$$A_x = \frac{Q}{(U_x)\,(CMTD)}$$

$$A_x = \frac{15{,}000{,}000}{(4.2)\,(71.3)} = 50{,}090 \text{ sq ft}$$

5. Calculate face area using APSF factor from Fig. 10-11

$$F_a = \frac{A_x}{APSF}$$

$$F_a = \frac{50{,}090}{107.2} = 467 \text{ sq ft (4 rows assumed)}$$

6. Calculate unit width with assumed tube length

$$\text{Width} = \frac{F_a}{L}$$

$$\text{Width} = \frac{467}{30} = 15.57 \text{ ft}$$

For simplification round this answer to 15.5, thus F_a = 465 (30-ft-long tubes assumed)

7. Calculate number of tubes using APF factor from Fig. 10-11

$$N_t = \frac{A_x}{(APF)\,(L)}$$

8. Calculate tube-side mass velocity from assumed number of passes and reading A_t from Fig. 9-25 for a 1 in. OD x 16 BWG tube

A_t = 0.5945 sq in.

$$G_t = \frac{(144)\,(W_t)\,(N_p)}{(3600)\,(N_t)\,(A_t)}$$

$$G_t = \frac{(0.04)\,(273{,}000)\,(3)}{(299)\,(0.5945)} = 184 \text{ lb/(ft}^2 \bullet \text{sec)}$$

9. Calculate modified Reynolds number

$$N_R = \frac{(D_i)\,(G_t)}{\mu} = \frac{(0.87)\,(184)}{0.51} = 314$$

10. Calculate tube-side pressure drop using equation from Fig. 10-14 and from Fig. 10-15

$$\Delta P_t = \frac{fYLN_p}{\phi} + BN_p$$

$$\Delta P_t = \frac{(0.0024)\,(14.5)\,(30)\,(3)}{0.96} + (0.25)\,(3) = 4.0 \text{ psi}$$

(ϕ is a difficult function to calculate rigorously, see Fig. 10-19)

11. Calculate tube-side film coefficient using equation from Fig. 10-13 and

$$k\left(\frac{C_p\mu}{k}\right)^{⅓}$$ from Fig. 10-12

$$h_t = \frac{Jk\left(\frac{C_p\mu}{k}\right)^{⅓}\phi}{D_i} = \frac{(1900)\,(0.12)\,(0.96)}{0.87} = 252$$

12. Calculate air quantity

$$W_a = \frac{Q}{(0.24)\,(\Delta t_a)}$$

$$W_a = \frac{15{,}000{,}000}{(0.24)\,(52)} = 1{,}200{,}000 \text{ lb/hr}$$

13. Calculate air face mass velocity

$$G_a = \frac{W_a}{F_a} = \text{lb/(hr} \bullet \text{sq ft of face area)}$$

$$G_a = \frac{1{,}200{,}000}{465} = 2{,}581$$

14. Read air-side film coefficient from Fig. 10-17

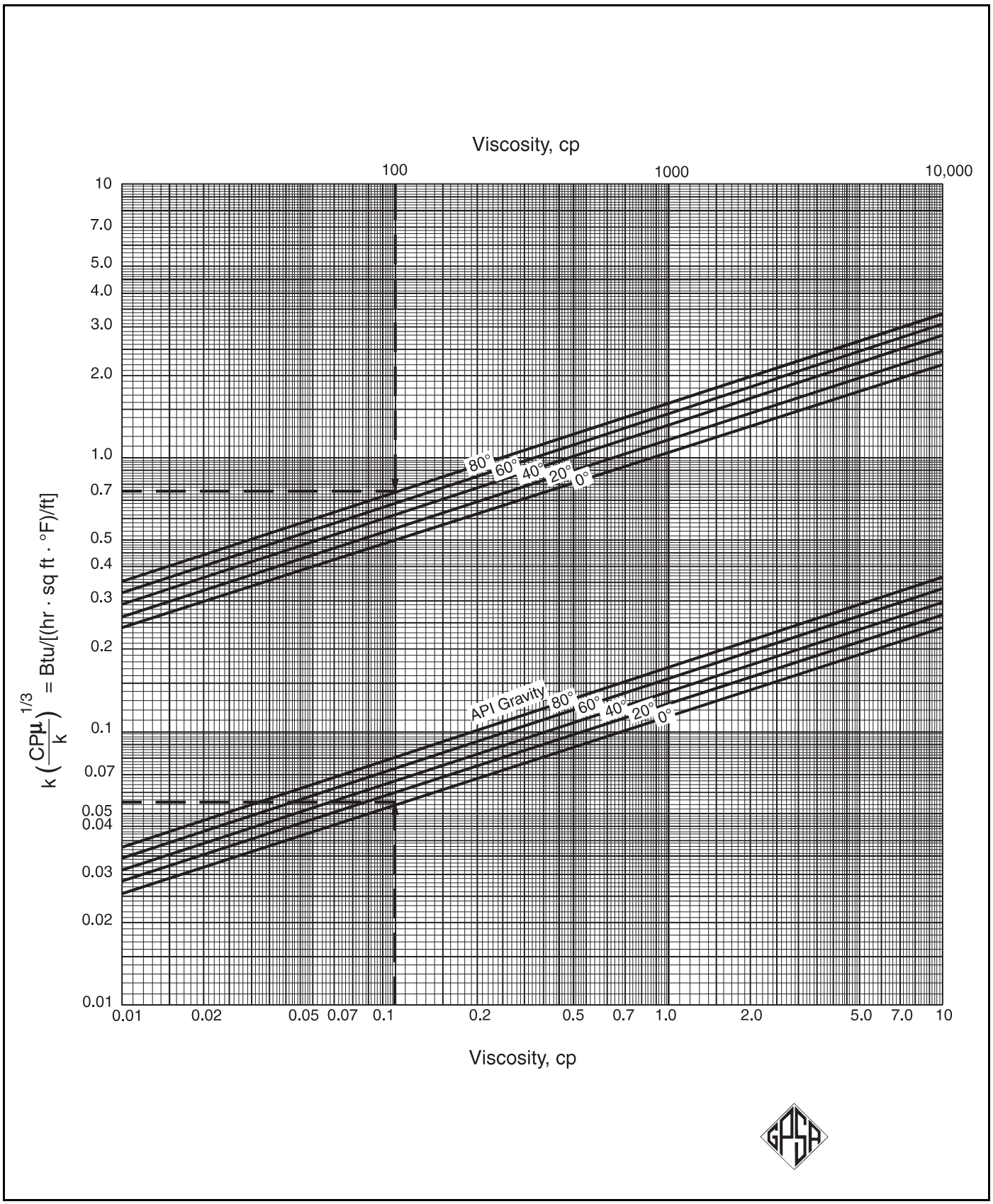
h_a = 8.5

15. Calculate overall transfer coefficient

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FIG. 10-12

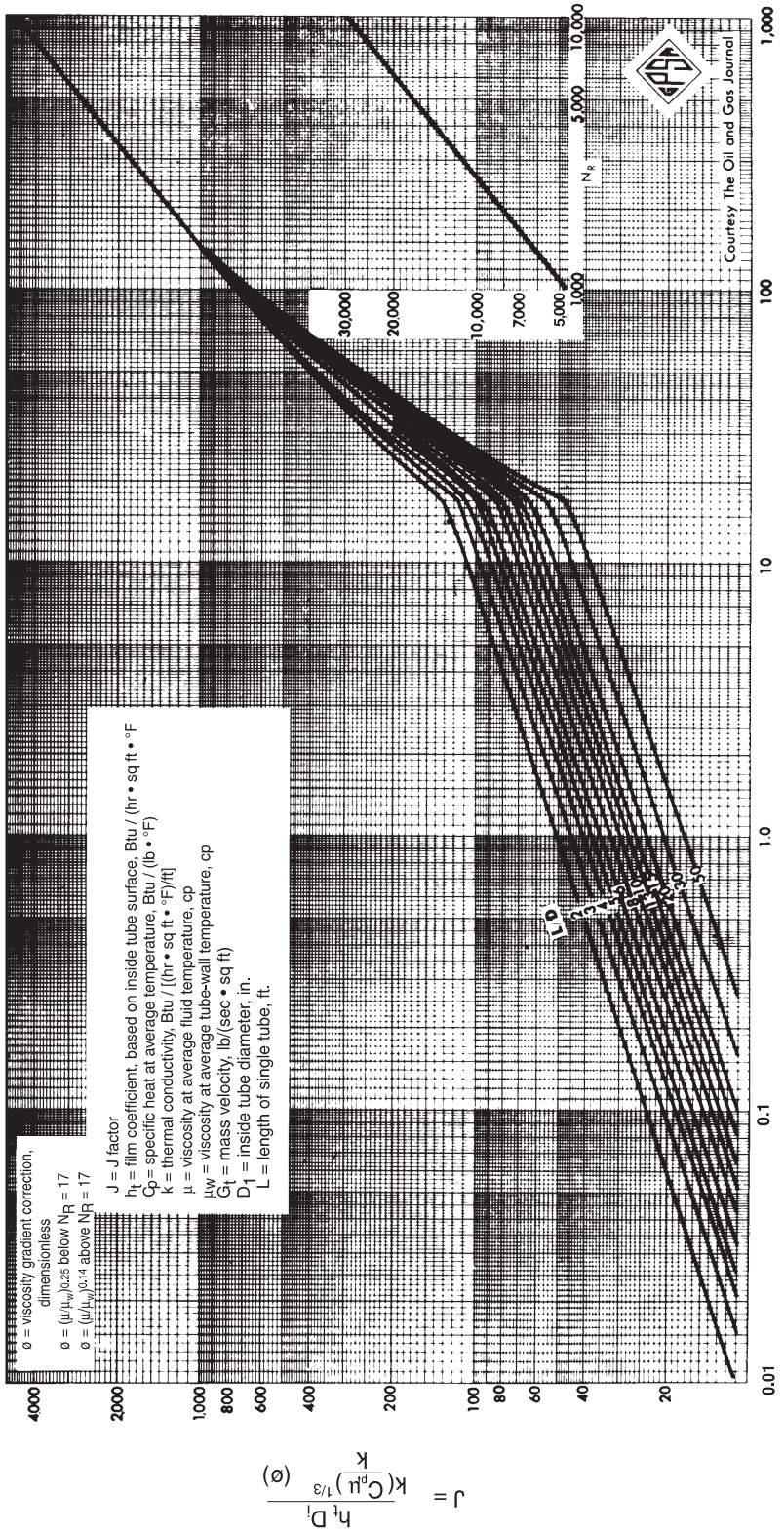
Physical Property Factor for Hydrocarbon Liquids



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FIG. 10-13

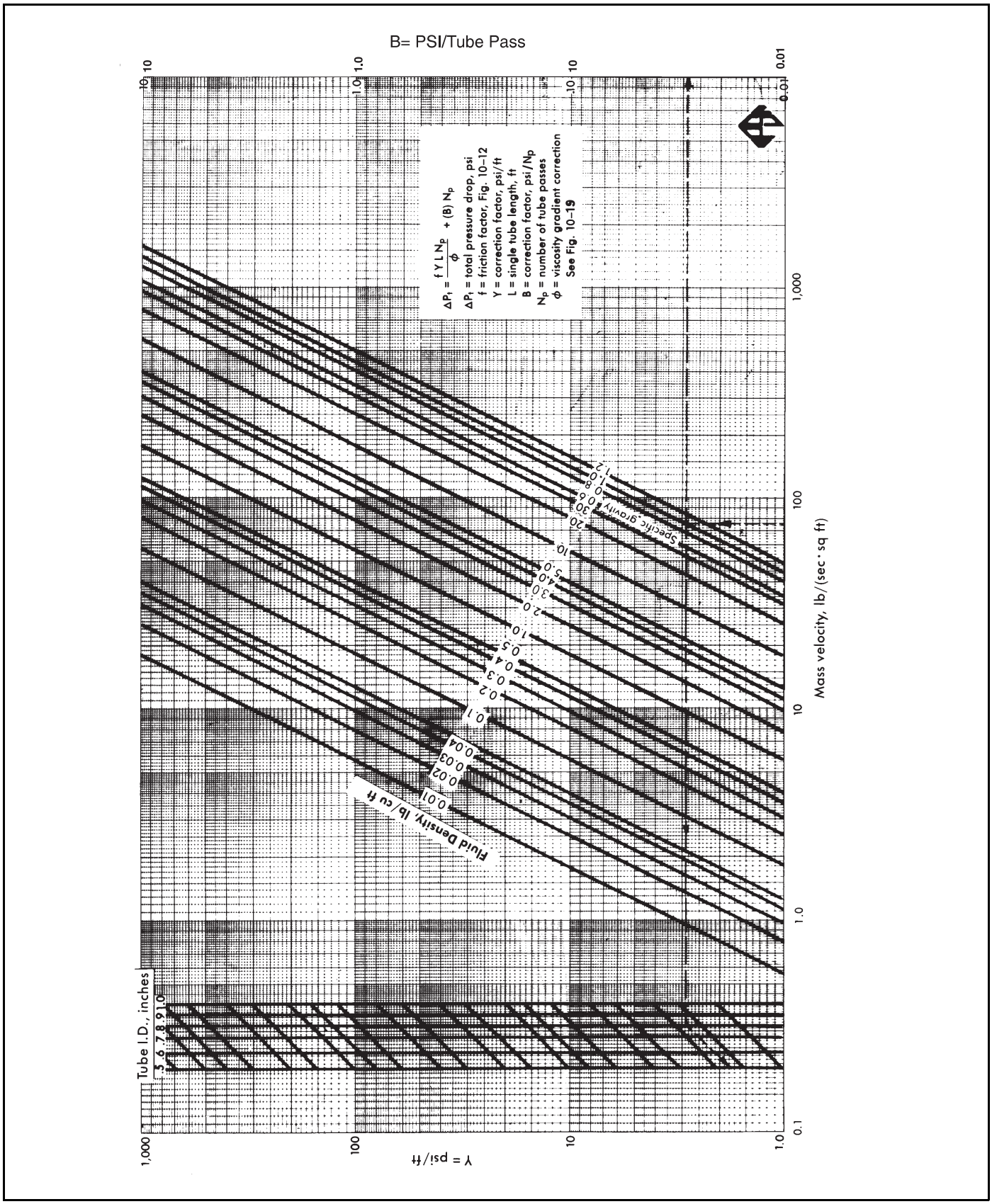
J Factor Correlation to Calculate Inside Film Coefficient, h_t



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FIG. 10-14

Pressure Drop for Fluids Flowing Inside Tubes



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$$\frac{A_x}{A_i} = \frac{(AR)\,(D_o)}{D_i}$$

$$\frac{A_x}{A_i} = \frac{(21.4)\,(1.0)}{0.87} = 24.6$$

$$\frac{1}{U_x} = \left(\frac{1}{h_t}\right)\left(\frac{A_x}{A_i}\right) + r_{dt}\left(\frac{A_x}{A_i}\right) + r_{mx} + \frac{1}{h_a}$$

$$\frac{1}{U_x} = \left(\frac{1}{252}\right)(24.6) + (0.001)\,(24.6) + \frac{1}{8.5}$$

$$U_x = 4.17$$

(r_{mx} is omitted from calculations, since metal resistance is small compared to other resistances)

Second and subsequent trials. If U_x calculated in Step 15 is equal or slightly greater than U_x assumed in Step 1, and calculated pressure drop in Step 9 is within allowable pressure drop, the solution is acceptable. Proceed to Step 16. Otherwise, repeat Steps 1-15 as follows:

1. Assume new U_x between value originally assumed in Step 1 and value calculated in Step 15.
2. Adjust Δt_a by increasing Δt_a if calculated U_x is higher than assumed U_x, or decreasing Δt_a if calculated U_x is lower than assumed U_x.

3.-15. Recalculate values in Steps 3-15 changing assumed number of passes in Steps 3 and 8, and tube length in Step 6, if necessary to obtain a pressure drop as calculated in Step 9 as high as possible without exceeding the allowable.

16. Calculate minimum fan area.

$$\text{Fan area/fan} = \text{FAPF} = \frac{(0.40)\,(F_a)}{(N_f)}$$

$$\text{FAPF} = \frac{(0.40)\,(465)}{2} = 93\ \text{ft}^2\ (2\ \text{fans assumed})$$

17. $\text{Fan diameter} = [4\,(\text{FAPF})/\pi]^{0.5} = [4\,(93)/3.1416]^{0.5}$
$= 11$ ft (rounded up)

18. Calculate air static pressure drop using F_p from Fig. 10-18 and D_R at avg air temp from Fig. 10-16.

$$T_a, \text{avg} = \frac{100°\text{F} + 152°\text{F}}{2} = 126°\text{F}$$

$$\Delta P_a = \frac{(F_p)\,(N)}{(D_R)}$$

$$\Delta P_a = \frac{(0.10)\,(4)}{0.90} = 0.44\ \text{inches of water}$$

19. Calculate actual air volume using D_R of air at fan inlet.

$t_1 = 100°\text{F}$

$$\text{ACFM} = \frac{W_a}{(D_R)\,(60)\,(0.0749)}$$

$$\text{ACFM} = \frac{1{,}200{,}000}{(0.94)\,(60)\,(0.0749)} = 284{,}000\ \text{Total}$$

or 142,000 / Fan

20. Approximate fan total pressure using D_R of air at fan and fan area.

$$\text{PF} = \Delta P_a + \left[\frac{\text{ACFM}}{4005\left(\frac{\pi D^2}{4}\right)}\right]^2 (D_R)$$

Where: $4005 = \sqrt{\frac{2\,g\,\rho_w\,(3600)}{\rho_a \cdot 12}}$ at 70°F

$$\text{PF} = 0.44 + \left(\frac{142{,}000}{(4005)\,(0.785)\,(11^2)}\right)^2 (0.94)$$

$= 0.57$ inches of water

21. Approximate brake horsepower per fan, using 70% fan efficiency.

$$\text{bhp} = \frac{(\text{ACFM/fan})\,(\text{PF})}{(6356)\,(0.70)}$$

Where the conversion factor

$$6356 = \left(\frac{33{,}000\ \text{ft–lb}}{\text{min} \cdot \text{hp}}\right)\left(\frac{12\ \text{in.}}{\text{ft}}\right)\left(\frac{\text{ft}^3}{62.3\ \text{lb}}\right)$$

Note: 62.3 is the weight of one cubic foot of water at 60°F.

$$\text{bhp} = \frac{(142{,}000)\,(0.57)}{(6356)\,(0.70)} = 18.2$$

Actual fan motor needed for 92% efficient speed reducer is 18.2/0.92 = 19.8 hp. For this application, 20 hp drivers would probably be selected.

Solution:

$(15.5\ \text{ft})\,(30\ \text{ft}) = 465$ sq ft

(465 sq ft) (APSF) = extended surface area

$(465)\,(107.2) = 49{,}848$ sq ft

Therefore, one unit having 49,848 sq ft of extended surface, two 11 ft diameter fans, and two 20 hp fan drivers, is required.

Fig. 10-20 has been included to aid the air cooler designer in choosing the proper pressure for the air density calculation at elevations higher than sea level.

MAINTENANCE AND INSPECTION

Attention to the design of the air cooler, and the choice of materials, is essential to provide low maintenance operation. Major factors to be considered are atmospheric corrosion, climatic conditions, and temperature cycling of fluid being cooled.

Scheduled preventive maintenance and inspection is the key to trouble-free air cooler operation. A check of all fans for vibration should be made regularly. At the first sign of undue vibration on a unit, the unit should be shut down at the earliest opportunity for thorough examination of all moving parts. A semi-annual inspection and maintenance program should:

- Check and replace worn or cracked belts.
- Inspect fan blades for deflection and for cracks near hubs.
- Grease all bearings.
- Change oil in gear drives.

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FIG. 10-15

Friction Factor for Fluids Flowing Inside Tubes

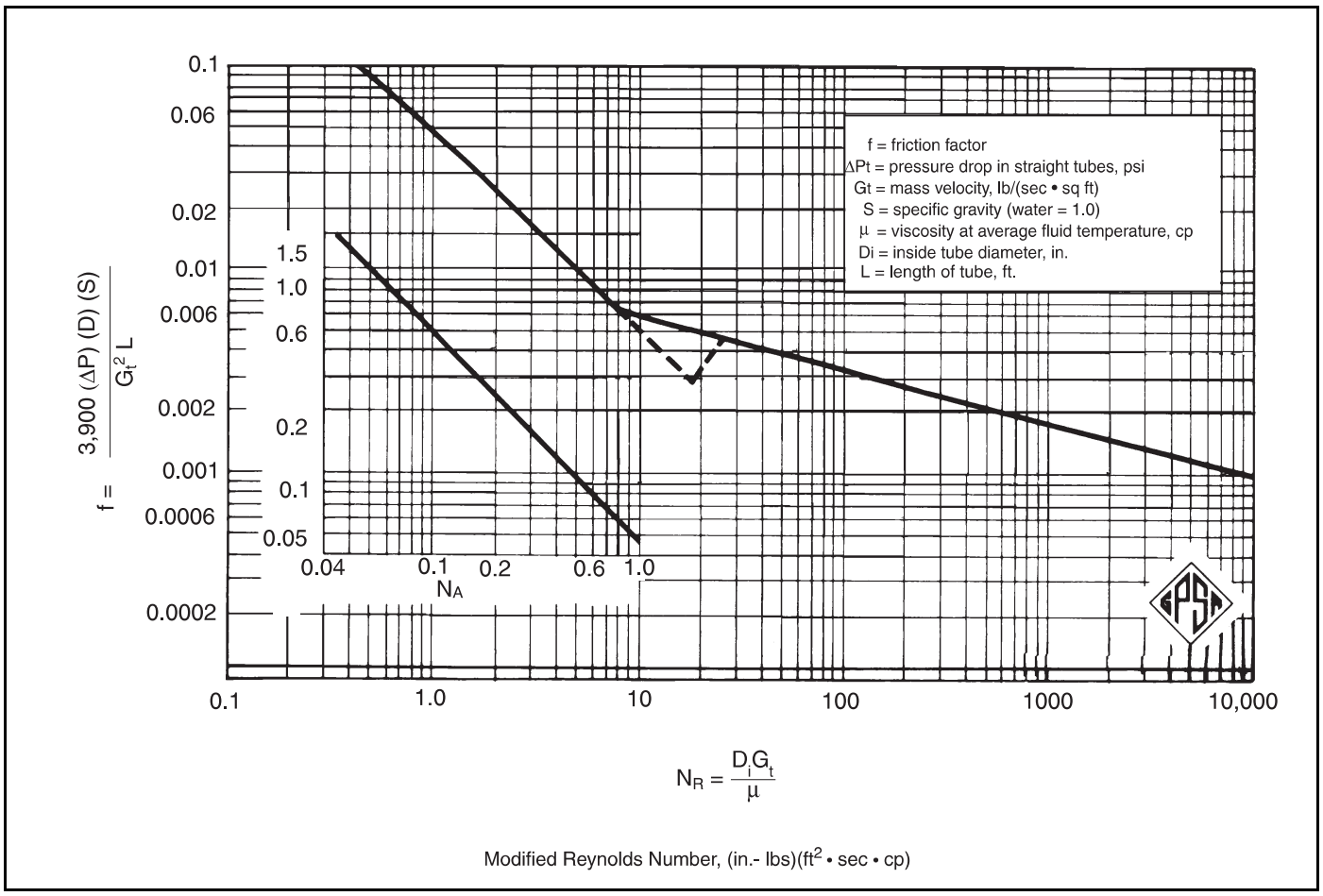


FIG. 10-16

Air-Density Ratio Chart

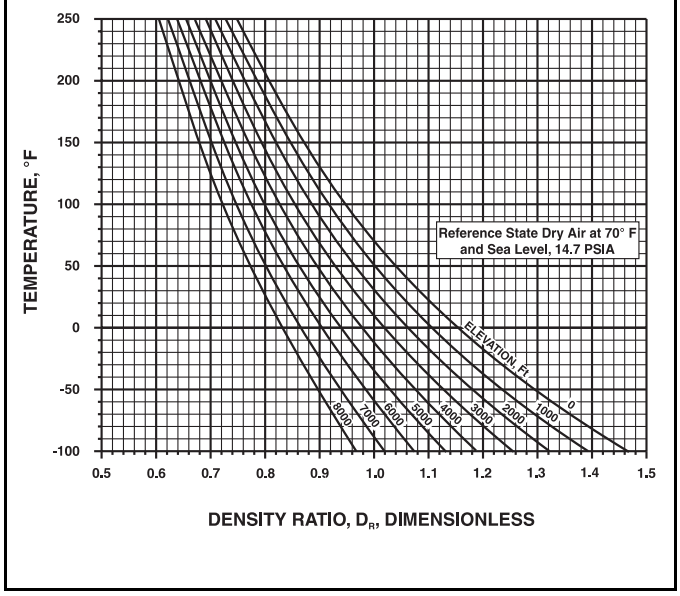
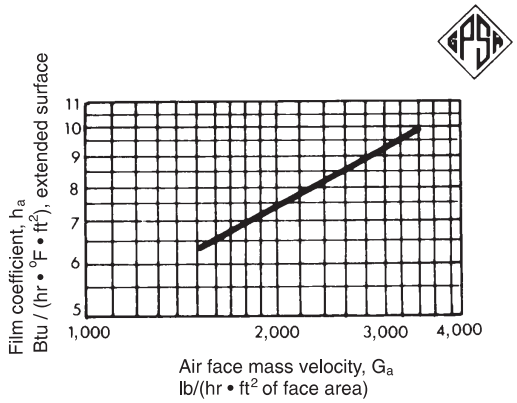


FIG. 10-17

Air Film Coefficient



Note: Figs 10-17 and 10-18 apply only to 1 in. OD, 5/8 in. high, 10 fpi, 2 1/2 in. Δ pitch fintube banks.

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- Check the inside of tube section for accumulation of grease, dirt, bugs, leaves, etc., and schedule cleaning before tubes become packed with such debris.

NOISE CONSIDERATIONS

Fan noise can be elusive requiring sophisticated equipment to measure accurately. Fan noise control must begin at the air-cooled heat exchanger design stage. Noise control, as an after thought, can result in a very costly fan and drive component retrofit, and possible addition of heat transfer surface or acoustic barriers. Acoustic barriers may increase the pressure drop the fan must overcome; hence, bigger fans and more horsepower will be required.

FIG. 10-18

Air Static-Pressure Drop

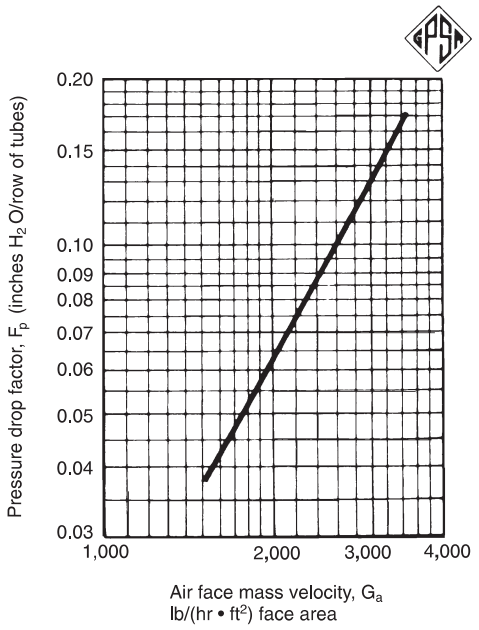


FIG. 10-19

Correction Factor for Fluid Viscosity Within the Tubes

Correction factor * when $\varphi = \left(\frac{\mu}{\mu_w}\right)^{0.14}$ (See Fig. 10-15)

	Correction Factor, ϕ
1. Hydrocarbon vapor; steam; water	1.0
2. Hydrocarbon liquids (18 to 48 API), MEA/DEA solutions	0.96
3. Water/glycol solutions; heat transfer fluids	0.92
4. Lube oils; heavy petroleum fractions (10 to 18 API)	0.85

* When $N_r < 17$, $\varphi = (\frac{\mu}{\mu_w})^{0.25}$ A Reynolds number of less than 17 is only likely for lube oils or heavy petroleum fractions. The minimum recommended value of ϕ to use in Step 10 is 0.80, even though the calculated value may be lower.

Since noise (Sound Pressure Level) is a function of tip speed, slowing fan rotation can reduce noise.

Unfortunately, a fan's pressure capability decreases with the square of the speed. Therefore, the fan's pressure capability must increase to maintain the required airflow.

To increase the pressure capability of a fan, the fan's solidity ratio must be increased, by adding more blades, or using blades with a wider chord, such as a low-noise blade design. Unfortunately, increasing the number of blades can reduce fan efficiency.

Noise requirements are often more restrictive at night. If this is a consideration, slowing the fan as ambient temperature drops using a variable-speed drive can be one solution to reducing noise. As the nighttime ambient temperature drops, required airflow is reduced, therefore fan speed may be slowed to lower noise levels.

Noise-Related Nomenclature

Decibel:

A number representing relative sound intensity expressed with respect to a reference pressure or power level. The usual reference for sound pressure level is of 20 micro newtons per square meter (20 μ/m^2). A decibel is a logarithm (base 10) of a ratio of the power values abbreviated "dB".

Frequency:

Sound vibration rate per second in Hertz (cycles per second).

Low-Noise Fans:

A fan able to operate at low speed due to its high-pressure capability. Fan pressure capability is a function of its solidity ratio. Therefore, a low-noise fan will generally have more or wider blades than would be required if the fan operated at normal tip speeds.

Octave Bands:

Noise is categorized by dividing it into separate frequency bands of octaves or 1/3 octaves. Generally, 63, 125, 250, 500, 1K, 2K, 4K and 8K center frequencies are used to define noise bands in Hertz (cycles/sec).

Sound Power Level:

Acoustical power (energy) can be expressed logarithmically in decibels with respect to a reference power, usually 10^{-12} watts. The relationship is given as: Sound Power Level.

$$PWL = 10 \log \left(\frac{W}{10^{-12}\ watts}\right) \qquad \textbf{Eq 10-3}$$

Sound power level cannot be measured directly but must be calculated from sound pressure levels (SPL) dB. In metric terms, this is known as Lw.

Sound pressure level, known as SPL, or Lp in metric terminology, is the audible noise given in decibels that relates to intensity at a point some distance from the noise source. It can be related to octave bands or as an overall weighted level dB(A).

Weighted sound levels relate the decibel (loudness) to a frequency. Ears can easily pick up high-frequency noises (both intensity and direction) but are relatively insensitive to low-frequency noise. For a stereo system high-frequency speakers must be very carefully located in a room for best results, but low-frequency bass speakers can be placed anywhere, even out of sight.

There are three basic weighting systems: A, B and C. The "A" system, dB(A), most closely relates to our ear, the "B" sys-

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FIG. 10-20

Altitude and Atmospheric Pressures[8]

Altitude above Sea Level		Barometer		Atmospheric Pressure	
meters	**feet**	**mm Hg abs.**	**in Hg abs.**	**kPa (abs)**	**psia**
0	0	760.0	29.92	101.325	14.696
153	500	746.3	29.38	99.49	14.43
305	1000	733.0	28.86	97.63	14.16
458	1500	719.6	28.33	95.91	13.91
610	2000	706.6	27.82	94.18	13.66
763	2500	693.9	27.32	92.46	13.41
915	3000	681.2	26.82	90.80	13.17
1068	3500	668.8	26.33	89.15	12.93
1220	4000	656.3	25.84	87.49	12.69
1373	4500	644.4	25.37	85.91	12.46
1526	5000	632.5	24.90	84.32	12.23
1831	6000	609.3	23.99	81.22	11.78
2136	7000	586.7	23.10	78.19	11.34
2441	8000	564.6	22.23	75.22	10.91
2746	9000	543.3	21.39	72.39	10.50
3050	10 000	522.7	20.58	69.64	10.10

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tem, dB(B), has some specific uses and the "C" system, dB(C), is considered unweighted.

The dB(A) is the most common weighting system. It expresses sound levels with a single number instead of having to specify the noise of each octave band.

Note that the sound range of ACHEs (at close range) is typically between 80 and 105 dB(A).

ACHE Noise

Whether the concern is for overall plant noise or the noise exposure of plant workers in the vicinity of the fans, a different type of noise specification must be used.

Overall, noise limitations from an ACHE are typically a sound power level (PWL) specification for each unit. This limits the contribution of each unit (typically two fans) to the plant noise as a whole. This is usually needed where noise at the plant boundary is considered. Contributions of each part of the plant must be carefully controlled if overall plant noise is limited. PWLs can be expressed as weighted level dB(A) or sometimes even by limitations on each octave band.

If worker protection is the main concern, a limitation of sound pressure level at 3 ft or 1 m below the bundle will probably be imposed as "SPL dB(A) at 3 ft". The OSHA limitation is 90 dB(A) for eight-hour exposure, but 85 dB(A) down to 80 dB(A) is not uncommon.

Predicting Fan Noise

Each fan manufacturer has proprietary equations for predicting fan noise. API Guidelines use the general formula:

$$PWL = 56 + 30 \log \left(\frac{FPM}{1000}\right) + \log HP \qquad \textbf{Eq 10-4}$$

This calculates PWL as dB(A)

Proprietary noise equations are based on actual tests at various speeds and operating conditions considering the following effects:

- Fan diameter
- Fan tip speed
- Blade Type
- Blade pitch angle
- Inlet conditions
- Horsepower

Note: Logs are common logs (base 10).

For example:

14 ft fan

237 RPM (10,424 FPM tip speed)

25.1 HP

Find sound power level

PWL= 56 + 30 log 10.4 + log 25.1

= 100.5 dB(A)

- When considering multiple noise sources (fans) use the relation:

$$PWL_N = PWL + 10 \log N \qquad \textbf{Eq 10-5}$$

The sound power level for 2 adjacent fans is the PWL of one fan plus 10 log 2 or $PWL_2 = PWL + 3$.

A doubling of the noise source adds 3 dB.

- Noise attenuates with distance by the equation:

$$SPL \text{ (at distance R)} = PWL - 20 \log R \qquad \textbf{Eq 10-6}$$

Where R is in feet from the center of the source. Measure R as a "line of sight" distance.

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Consider the noise an observer at grade hears at 50 ft from an operating ACHE with the fan on and the line-of-sight distance from grade to the center of the fan is actually 62 ft. Remember what the ear hears is SPL, the noise energy is PWL.

Assume PWL = 100.5 dB(A).

SPL = 100.5 - 20 log 62 = 64.7 dB(A)

This also assumes background noise is at least 10 dB quieter.

Note: If both fans were running, the SPL would have been 67.7 dB(A).

When considering the noise at 3 ft beneath the unit, the drive system and motor noise become dominant at lower tip speeds.

Factors that influence this noise are:

- Motor noise
- Belt or gear noise
- Bearing noise
- Reflected noise from supports
- Background noise

Gear noise is especially significant in a forced draft unit.

Noise Testing

Frequently, the ACHE must be tested for confirmation that its noise does not exceed specifications imposed by the purchaser. There are two basic types of tests normally performed before shipment:

a) Measure SPL dB(A) (Sound Pressure Level) at "3 ft below the bundle or fan guard" — depending on whether the unit is induced or forced draft.

b) Measure PWL (Sound Power Level) using "hemispherical power level test." PWL is specified as either a dB(A) weighted value or by octave bands.

The "SPL at 3 ft" test is by far the most common and least expensive. Usually, only one or two measurements are required. The answer is immediate and read directly from the noise meter.

The hemispherical test is far more complicated and expensive. Several hours, many technicians and a large crane are required to perform this test. Full details of the test are given in API Recommended Practice 631 M, June, 1981.[7]

REFERENCES

1. A.P.I. Standard 661, “Air Cooled Heat Exchangers for General Refinery Services.”
2. Briggs, D. E., Young, E. H., “Convection Heat Transfer and Pressure Drop of Air Flowing Across Triangular Pitch of Tubes,” Chemical Engineering Progress Symposium Series, Volume 59, No. 41, 1963.
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5. Robinson, K. K., Briggs, D. E., “Pressure Drop of Air Flowing Across Triangular Pitch Banks of Finned Tubes,” Chemical Engineering Progress Symposium Series, Volume 62, No. 64, 1966.
6. Rubin, Frank L., “Winterizing Air Cooled Heat Exchangers,” Hydrocarbon Processing, October 1980, pp. 147-149.
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SECTION 11

Cooling Towers

INTRODUCTION

The purpose of this section is to provide a basic understanding of the design and operation of a cooling tower. A cooling tower allows water to be cooled by ambient air through evaporation.

COOLING TOWER PSYCHROMETRICS

Psychrometry is the study of cooling by evaporation. Maximum evaporation takes place when water, in the form of tiny droplets, is exposed to the maximum air flow for the longest possible time. The process of evaporation through removal of

FIG. 11-1

Nomenclature

Acfm	=	actual volumetric flow rate of air-vapor mixture, cu ft/min
ahp	=	air horsepower, hp
AWB	=	ambient wet bulb temperature, °F
B	=	combined water loss through blowdown and windage, % of circulating water or cu ft/min
CWT	=	cold water temperature, °F
DB	=	dry bulb temperature, °F
E	=	water evaporated, % of circulating water or cu ft/min
gpm	=	gallons per minute
G	=	air rate, lb/(sq ft • hr)
h_a	=	specific enthalpy of dry air, BTU/lb
h_{as}	=	$h_s - h_a$, BTU/lb
h_s	=	enthalpy of moist air at saturation per lb of dry air, BTU/lb
HWT	=	hot water temperature, °F
lb_a	=	pounds of dry air
lb_w	=	pounds of water
L	=	water rate, lb/(sq ft • hr)
LG	=	liquid to gas ratio, lb/lb
Q	=	cu ft/min
PF	=	performance factor, dimensionless
R	=	cooling tower range, °F
V	=	air velocity, ft/min
v_a	=	specific volume of dry air, cu ft/lb
v_{as}	=	$v_s - v_a$, cu ft/lb
v_s	=	volume of moist air at saturation per lb of dry air, cu ft/lb
Ws	=	lb_w/lb_a, humidity ratio at saturation
WB	=	wet bulb temperature, °F

Cold Water Temperature: Temperature of the water leaving the collection basin, exclusive of any temperature effects incurred by the addition of makeup and/or the removal of blowdown.

Collection Basin: Chamber below and integral with the tower where water is transiently collected and directed to the sump or pump suction line.

Counterflow: Air flow direction through the fill is countercurrent to that of the falling water.

Crossflow: Air flow direction through the fill is essentially perpendicular to that of the falling water.

Distribution Basin: Shallow pan-type elevated basin used to distribute hot water over the tower fill by means of orifices in the basin floor. Application is normally limited to crossflow towers.

Double-Flow: A crossflow cooling tower where two opposed fill banks are served by a common air plenum.

Drift: Circulating water loss from the tower as liquid droplets entrained in the exhaust air stream. Units percent of circulating water rate or gpm. [For more precise work, an L/G parameter is used, and drift becomes pounds of water per million pounds of exhaust air (ppmw).]

Drift Eliminators: An assembly of baffles or labyrinth passages through which the air passes prior to its exit from the tower, for the purpose of removing entrained water droplets from the exhaust air.

Dry-Bulb Temperature: The temperature of the entering or ambient air adjacent to the cooling tower as measured with a dry-bulb thermometer.

Evaporation Loss: Water evaporated from the circulating water into the air stream in the cooling process.

Air Horsepower: The power output developed by a fan in moving a given air rate against a given resistance.

Air Inlet: Opening in a cooling tower through which air enters. Sometimes referred to as the louvered face on induced draft towers.

Air Rate: Mass flow of dry air per square foot of cross-sectional area in the tower's heat transfer region per hour.

Air Velocity: Velocity of air-vapor mixture through a specific region of the tower (i.e. the fan).

Ambient Wet-Bulb Temperature: The wet-bulb temperature of the air encompassing a cooling tower, not including any temperature contribution by the tower itself. Generally measured upwind of a tower, in a number of locations sufficient to account for all extraneous sources of heat.

Approach: Difference between the cold water temperature and the entering wet-bulb temperature.

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FIG. 11-1 (Cont'd)

Nomenclature

Blowdown: Water discharged from the system to control concentrations of salt or other impurities in the circulating water.

Capacity: The amount of water that a cooling tower will cool through a specified range, at a specified approach and wet-bulb temperature.

Cell: Smallest tower subdivision which can function as an independent unit with regard to air and water flow; it is bounded by either exterior walls or partition walls. Each cell may have one or more fans and one or more distribution systems.

Circulation Rate: Actual water flow rate through a given tower.

Fan Cylinder: Cylindrical or venturi-shaped structure in which a propeller fan operates. Sometimes referred to as a fan "stack" on larger towers.

Fan Deck: Surface enclosing the top of an induced draft cooling tower, exclusive of the distribution basins on a crossflow tower.

Fan Pitch: The angle which the blades of a propeller fan make with the plane of rotation, measured at a prescribed point on each blade.

Fill: That portion of a cooling tower which constitutes its primary heat transfer surface. Sometimes referred to as "packing."

Forced Draft: Refers to the movement of air under pressure through a cooling tower. Fans of forced draft towers are located at the air inlets to "force" air through the tower.

Hot Water Temperature: Temperature of circulating water entering the cooling tower's distribution system.

Induced Draft: Refers to the movement of air through a cooling tower by means of an induced partial vacuum. Fans of induced draft towers are located at the air discharges to "draw" air through the tower.

Liquid-to-Gas Ratio: A ratio of the total mass flows of water and dry air in a cooling tower. (See Air Rate and Water Rate)

Louvers: Blade or passage type assemblies installed at the air inlet face of a cooling tower to control water splashout and/or promote uniform air flow through the fill. In the case of film-type crossflow fill, they may be integrally molded to the fill sheets.

Makeup: Water added to the circulating water system to replace water lost by evaporation, drift, windage, blowdown, and leakage.

Natural Draft: Refers to the movement of air through a cooling tower purely by natural means. Typically, by the driving force of a density differential.

Net Effect Volume: That portion of the total structural volume within which the circulating water is in intimate contact with the flowing air.

Performance Factor: Variable used in determining performance characteristics in cooling towers.

Psychrometer: An instrument incorporating both a dry-bulb and a wet-bulb thermometer, by which simultaneous dry-bulb and wet-bulb temperature readings can be taken.

Range: Difference between the hot water temperature and the cold water temperature.

Recirculation: Describes a condition in which a portion of the tower's discharge air re-enters the air inlets along with the fresh air. Its effect is an elevation of the average entering wet-bulb temperatures compared to the ambient.

Water Rate: Mass flow of water per square foot of fill plan area of the cooling tower per hour.

Wet-Bulb Temperature: the temperature of the entering or ambient air adjacent to the cooling tower as measured with a wet-bulb thermometer.

Wet-Bulb Thermometer: A thermometer whose bulb is encased within a wetted wick.

Windage: Water lost from the tower because of the effects of wind.

Wind Load: The load imposed upon a structure by a wind blowing against its surface.

latent heat allows the water to be cooled below the ambient dry-bulb temperature. The dry air enters the cooling tower and begins to gain moisture and enthalpy in an effort to reach equilibrium with the water. The water may be cooled 15°F or more while the air mass dry-bulb temperature may increase only slightly. A psychrometric chart (Fig. 11-2) may be used to illustrate the relationships between wet- and dry-bulb temperatures. All nomenclature is indicated in Fig. 11-1.

Wet-bulb Temperature

The basis for thermal design of an evaporative type cooling tower is the wet-bulb temperature of the air entering the tower. Water cannot be cooled below the wet-bulb temperature by evaporation. The air entering the tower at a temperature of 75°F and 100% relative humidity has a 75°F "wet-bulb" temperature. The wet-bulb temperature is usually measured using a sling psychrometer. Wet bulb and dry bulb data for various locations around the world are shown in Figs. 11-3a and 11-3b.

Types of Cooling Systems

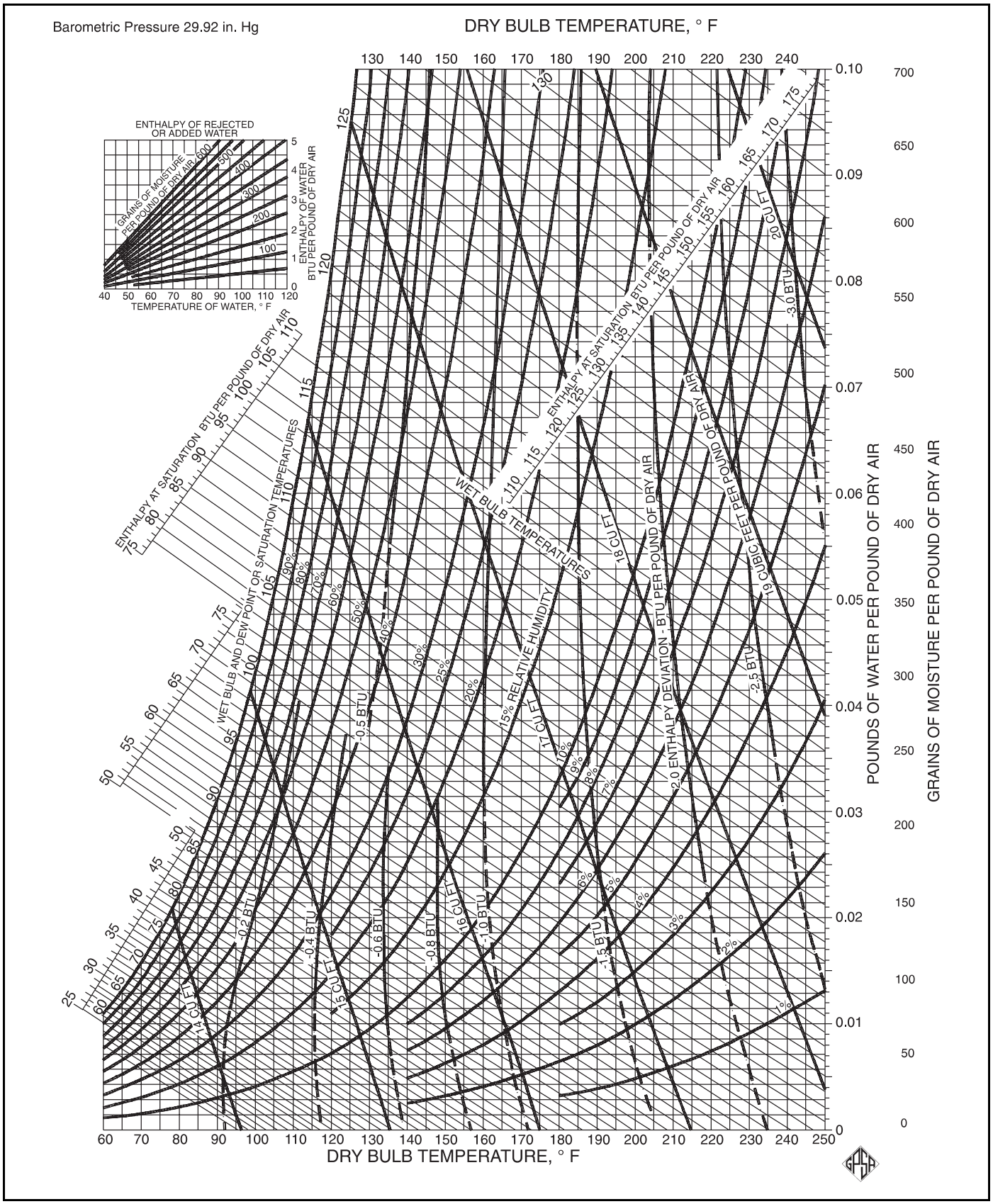
Cooling water is circulated through equipment to absorb and carry away heat. The basic cooling systems are shown in Fig. 11-4. The open recirculating system is the most common for industrial plants.

The open recirculating system routes cooled water through the heat-exchange equipment. Effluent warm water then is cooled by contact with air in a cooling tower. The cooling effect is produced by evaporation of a portion of the circulating water, and this evaporation causes the dissolved solids in the water to become concentrated. Water lost by evaporation must be replaced by makeup water. But water lost through entrainment of droplets in the circulating air (windage or drift) tends to limit the degree of concentration since the entrained droplets also contain dissolved solids. Windage loss varies with the type of tower.

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FIG. 11-2

Psychrometric Chart



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FIG. 11-3a

Dry Bulb/Wet Bulb Temperature Data [2]

U.S. State and Stations	Design Dry-Bulb °F 5%	Design Wet-Bulb °F 5%
ALABAMA		
Alexander City	91	78
Anniston	92	78
Auburn	91	78
Birmingham	92	76
Decatur	91	76
Dothan	91	78
Florence	92	76
Gadsden	92	76
Huntsville	91	76
Mobile AP	91	78
Mobile CO	91	78
Montgomery	93	78
Selma-Craig	93	79
Talladega	92	78
Tuscaloosa	94	77
ALASKA		
Anchorage	66	57
Barrow	49	47
Fairbanks	75	60
Juneau	67	58
Kodiak	62	56
Nome	59	55
ARIZONA		
Douglas	93	68
Flagstaff	80	59
Fort Huachuca	90	67
Kingman	97	69
Nogales	94	69
Phoenix	105	75
Prescott	92	64
Tucson	100	71
Winslow	93	64
Yuma	107	77
ARKANSAS		
Blytheville	91	78
Camden	94	78
El Dorado	94	78
Fayetteville	92	75
Fort Smith	95	78
Hot Springs	94	78
Jonesboro	91	78
Little Rock	94	78
Pine Bluff	95	80
Texarkana	93	78
CALIFORNIA		
Bakersfield	98	70
Barstow	102	70
Blythe	108	74
Burbank	88	69
Chico	98	68
Concord	94	68
Covina	92	70
Crescent City	63	59
Downey	86	70
El Cajon	78	68
El Centro	108	78
Escondido	82	69
Eureka/Arcata	63	59
Fairfield-Travis AFB	91	67
Fresno	97	70
Hamilton	80	67
Laguna Beach	77	68
Livermore	93	68
Lompoc, Vandenburg AFB	67	60
Long Beach	77	68
Los Angeles AP	77	68
Los Angeles CO	86	70
Merced-Castle AFB	96	70
Modesto	95	69
Monterey	68	61
Napa	92	68
Needles	108	74
Oakland	75	63
Oceanside	77	68
Ontario	96	71
Oxnard	77	67
Palmdale	98	66
Palm Springs	108	73
Pasadena	92	70
Petaluma	87	68
Pomona	95	71
Redding	100	68
Redlands	96	71
Richmond	75	63
Riverside-March AFB	95	70
Sacramento	94	70
Salinas	67	59
San Bernardino, Norton AFB	96	71
San Diego	78	68
San Fernando	88	69
San Francisco AP	73	62
San Francisco CO	69	61
San Jose	77	65
San Luis Obispo	84	70
Santa Ana	82	69
Santa Barbara	75	66
Santa Cruz	68	61
Santa Maria	73	63
Santa Monica	77	68
Santa Paula	84	68
Santa Rosa	91	67
Stockton	94	68
Ukiah	91	67
Visalia	97	70
Yreka	89	64
Yuba City	99	68
COLORADO		
Alamosa	80	60
Boulder	89	62
Colorado Springs	86	61
Denver	89	62
Durango	85	62
Fort Collins	89	62
Grand Junction	92	62
Greeley	92	63
La Junta	95	69
Leadville	78	54
Pueblo	92	65
Sterling	90	65
Trinidad	89	64
CONNECTICUT		
Bridgeport	81	73
Hartford, Brainard Field	85	74
New Haven	82	74
New London	83	74
Norwalk	81	73
Norwich	83	74
Waterbury	82	72
Windsor Locks, Bradley Field	85	73
DELAWARE		
Dover AFB	87	76
Wilmington	87	75
DISTRICT OF COLUMBIA		
Andrews AFB	87	75
Washington National	89	76
FLORIDA		
Belle Glade	89	78
Cape Kennedy	87	79
Daytona Beach	88	78
Fort Lauderdale	90	79
Fort Myers	91	79
Fort Pierce	89	79
Gainesville	92	78
Jacksonville	92	78
Key West	89	79
Lakeland	89	78
Miami	89	78
Miami Beach	88	78
Ocala	92	78
Orando	91	78
Panama City, Tyndall AFB	89	79
Pensacola	91	79
St. Augustine	87	79
St. Petersburg	90	78
Sanford	91	78
Sarasota	90	78
Tallahassee	90	78
Tampa	90	78
West Palm Beach	90	79
GEORGIA		
Albany, Turner AFB	93	78
Americus	92	77
Athens	90	76
Atlanta	90	75
Augusta	93	78
Brunswick	87	79
Columbus, Lawson AFB	91	77
Dalton	91	77
Dublin	91	77
Gainesville	89	75
Griffin	88	76
La Grange	89	76
Macon	91	77
Marietta, Dobbins AFB	90	76
Moultrie	92	78
Rome	91	77
Savannah-Travis	91	78
Valdosta-Moody AFB	92	78
Waycross	91	78
HAWAII		
Hilo	82	74
Honolulu	85	74
Kaneohe Bay	83	75
Wahiawa	84	73
IDAHO		
Boise	91	65
Burley	92	61
Coeur d'Alene	83	61
Idaho Falls	84	61
Lewiston	90	64
Moscow	84	62
Mountain Home AFB	94	63
Pocatello	89	61
Twin Falls	92	61
ILLINOIS		
Aurora	88	76
Belleville, Scott AFB	89	76
Bloomington	88	75
Carbondale	90	77
Champaign/Urbana	90	75
Chicago, Midway	88	74
Chicago, O'Hare	86	74
Chicago	88	75
Danville	88	75
Decatur	88	75
Dixon	88	75
Elgin	86	75
Freeport	87	74
Galesburg	88	75
Greenville	89	76
Joliet	88	75
Kankakee	88	75
La Salle/Peru	88	75
Macomb	89	76
Moline	88	75
Mt. Vernon	89	76
Peoria	87	75
Quincy	90	77
Rantoul, Chanute AFB	89	75
Rockford	87	74
Springfield	89	76
Waukegan	87	75
INDIANA		
Anderson	89	76
Bedford	89	76
Bloomington	89	76
Columbus, Bakalar AFB	90	76
Crawfordsville	88	76
Evansville	91	77
Fort Wayne	87	74
Goshen	86	74
Hobart	85	74
Huntington	87	74
Indianapolis	87	75
Jeffersonville	90	76
Kokomo	88	74
Lafayette	88	75
La Porte	87	75
Marion	88	74
Muncie	87	75
Peru, Bunker Hill AFB	86	74
Richmond	87	75
Shelbyville	88	75
South Bend	86	74
Terre Haute	89	76
Valparaiso	87	75
Vincennes	90	76
IOWA		
Ames	87	75
Burlington	88	75
Cedar Rapids	86	75
Clinton	87	75
Council Bluffs	88	75
Des Moines	88	75
Dubuque	86	74
Fort Dodge	86	74
Iowa City	87	76
Keokuk	89	76
Marshalltown	88	75
Mason City	85	74
Newton	88	75
Ottumwa	88	75
Sioux City	89	75
Waterloo	86	75
KANSAS		
Atchison	91	77
Chanute	94	76
Dodge City	95	71
El Dorado	96	75
Emporia	94	76
Garden City	94	71
Goodland	93	68
Great Bend	95	75
Hutchinson	97	74
Liberal	94	71
Manhattan, Fort Riley	92	76
Parsons	94	76
Russell	95	75
Salina	97	75
Topeka	93	76
Wichita	96	75
KENTUCKY		
Ashland	89	75
Bowling Green	89	76
Corbin	89	75
Covington	88	74
Hopkinsville, Campbell AFB	89	76
Lexington	88	75
Louisville	90	76
Madisonville	90	77
Owensboro	91	77
Paducah	92	77
LOUISIANA		
Alexandria	92	78

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FIG. 11-3a (Cont'd)

Dry Bulb/Wet Bulb Temperature Data[2]

U.S. State and Stations	Design Dry-Bulb °F 5%	Design Wet-Bulb °F 5%
Baton Rouge	92	79
Bogalusa	92	79
Houma	92	79
Lafayette	92	79
Lake Charles	92	79
Minden	94	78
Monroe	94	78
Natchitoches	93	78
New Orleans	90	79
Shreveport	94	78
MAINE		
Augusta	82	70
Bangor, Dow AFB	80	69
Caribou	78	67
Lewiston	82	70
Millinocket	80	68
Portland	81	70
Waterville	81	70
MARYLAND		
Baltimore AP	89	76
Baltimore CO	87	76
Cumberland	87	75
Frederick	88	76
Hagerstown	89	75
Salisbury	88	76
MASSACHUSETTS		
Boston	85	72
Clinton	84	72
Fall River	81	72
Framingham	83	71
Gloucester	83	72
Greenfield	82	71
Lawrence	84	73
Lowell	85	73
New Bedford	80	72
Pittsfield	81	70
Springfield, Westover AFB	84	72
Taunton	83	73
Worcester	81	70
MICHIGAN		
Adrian	85	73
Alpena	83	70
Battle Creek	85	73
Benton Harbor	85	72
Detroit	86	73
Escanaba	80	69
Flint	85	72
Grand Rapids	85	72
Holland	83	72
Jackson	85	73
Kalamazoo	85	73
Lansing	84	72
Marquette	77	68
Mt. Pleasant	84	72
Muskegon	82	72
Pontiac	85	73
Port Huron	83	73
Saginaw	84	72
Sault St. Marie	77	68
Traverse City	83	71
Ypsilanti	86	72
MINNESOTA		
Albert Lea	84	73
Alexandria	85	72
Bemidji	81	69
Brainerd	84	71
Duluth	79	68
Fairbault	85	73
Fergus Falls	85	72
International Falls	80	68
Mankato	85	73
Minneapolis/St. Paul	86	73
Rochester	84	73
St. Cloud	85	72
Virginia	80	68
Willmar	85	72
Winona	85	74
MISSISSIPPI		
Biloxi, Keesler AFB	90	80
Clarksdale	92	78
Columbus AFB	91	78
Greenville AFB	91	78
Greenwood	91	78
Hattiesburg	92	79
Jackson	93	78
Laurel	92	79
McComb	92	78
Meridian	93	78
Natchez	92	79
Tupelo	92	78
Vicksburg	93	79
MISSOURI		
Cape Girardeau	92	77
Columbia	91	76
Farmington	90	75
Hannibal	90	77
Jefferson City	92	76
Joplin	94	76
Kansas City	93	76
Kirksville	90	76
Mexico	91	76
Moberly	91	76
Poplar Bluff	92	78
Rolla	89	76
St. Joseph	91	77
St. Louis AP	91	76
St. Louis CO	91	76
Sedalia, Whiteman AFB	90	76
Sikeston	92	77
Springfield	91	75
MONTANA		
Billings	88	64
Bozeman	84	60
Butte	80	57
Cut Bank	82	61
Glasgow	85	64
Glendive	89	65
Great Falls	85	60
Havre	87	65
Helena	85	61
Kalispell	84	62
Lewiston	83	62
Livingston	84	60
Miles City	92	67
Missoula	85	62
NEBRASKA		
Beatrice	92	76
Chadron	91	68
Columbus	92	75
Fremont	92	76
Grand Island	91	73
Hastings	91	73
Kearney	90	72
Lincoln	92	76
McCook	91	71
Norfolk	90	75
North Platte	90	71
Omaha	88	75
Scottsbluff	90	67
Sidney	90	67
NEVADA		
Carson City	89	60
Elko	90	60
Ely	85	58
Las Vegas	104	69
Lovelock	93	64
Reno AP	90	61
Reno CO	91	61
Tonopah	90	61
Winnemucca	92	61
NEW HAMPSHIRE		
Berlin	81	70
Claremont	83	71
Concord	84	71
Keene	83	71
Laconia	83	71
Manchester, Grenier AFB	85	72
Portsmouth, Pease AFB	83	72
NEW JERSEY		
Atlantic City	86	75
Long Branch	87	75
Newark	88	75
New Brunswick	86	75
Paterson	88	75
Phillipsburg	86	74
Trenton	85	75
Vineland	86	75
NEW MEXICO		
Alamogordo, Holloman AFB	94	67
Albuquerque	92	64
Artesia	97	70
Carlsbad	97	70
Clovis	91	67
Farmington	91	64
Gallup	86	61
Grants	85	61
Hobbs	97	69
Las Cruces	94	67
Los Alamos	85	60
Raton	87	63
Roswell, Walker AFB	96	69
Santa Fe	86	61
Silver City	91	63
Socorro	93	65
Tucumcari	95	68
NEW YORK		
Albany AP	85	72
Albany CO	85	72
Auburn	84	72
Batavia	84	72
Binghamton	81	70
Buffalo	83	72
Cortland	82	71
Dunkirk	83	72
Elmira	83	71
Geneva	84	72
Glen Falls	82	71
Gloversville	83	72
Hornell	82	72
Ithaca	82	71
Jamestown	83	71
Kingston	85	73
Lockport	84	73
Massena	80	70
Newburg-Stewart AFB	85	73
NYC-Central Park	87	74
NYC-Kennedy AP	84	74
NYC-LaGuardia AP	87	74
Niagara Falls	84	73
Olean	81	71
Oneonta	80	70
Oswego	80	72
Plattsburg AFB	80	70
Poughkeepsie	86	74
Rochester	85	72
Rome-Griffiss AFB	83	71
Schenectady	84	72
Suffolk County AFB	80	73
Syracuse	84	72
Utica	82	71
Watertown	81	72
NORTH CAROLINA		
Asheville	85	72
Charlotte	91	76
Durham	90	76
Elizabeth City	89	78
Fayetteville, Pope AFB	90	77
Goldsboro, Seymour-Johnson AFB	89	77
Greensboro	89	75
Greenville	89	77
Henderson	90	77
Hickory	88	73
Jacksonville	88	78
Lumberton	90	77
New Bern	88	78
Raleigh/Durham	90	76
Rocky Mount	89	77
Wilmington	89	79
Winston-Salem	89	74
NORTH DAKOTA		
Bismark	88	70
Devil's Lake	85	69
Dickinson	87	68
Fargo	85	72
Grand Forks	84	70
Jamestown	87	71
Minot	86	68
Williston	85	68
OHIO		
Akron-Canton	84	72
Ashtabula	83	72
Athens	90	74
Bowling Green	86	73
Cambridge	87	75
Chillicothe	90	74
Cincinnati	88	74
Cleveland	86	73
Columbus	87	74
Dayton	86	73
Defiance	88	74
Findlay	87	74
Fremont	85	73
Hamilton	87	73
Lancaster	88	74
Lima	88	74
Mansfield	85	73
Marion	88	74
Middletown	87	73
Newark	89	74
Norwalk	85	73
Portsmouth	89	75
Sandusky	88	73
Springfield	87	74
Steubenville	84	72
Toledo	85	73
Warren	85	71
Wooster	84	72
Youngstown	84	71
Zanesville	87	75
OKLAHOMA		
Ada	95	75
Altus AFB	98	75
Ardmore	95	76
Bartlesville	95	76
Chickasha	95	76
Enid-Vance AFB	97	76
Lawton	96	76
McAlester	93	75
Muskogee	95	77
Norman	94	75
Oklahoma City	95	76
Ponca City	94	76
Seminole	94	75

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FIG. 11-3a (Cont'd)

Dry Bulb/Wet Bulb Temperature Data [2]

U.S. State and Stations	Design Dry-Bulb °F 5%	Design Wet-Bulb °F 5%
Stillwater	93	75
Tulsa	95	77
Woodward	94	75
OREGON		
Albany	86	66
Astoria	68	62
Baker	86	61
Bend	84	60
Corvallis	86	66
Eugene	86	66
Grants Pass	93	68
Klamath Falls	84	60
Medford	91	67
Pendleton	90	63
Portland AP	81	66
Portland CO	82	66
Roseburg	87	66
Salem	84	66
The Dalles	85	67
PENNSYLVANIA		
Allentown	86	73
Altoona	84	72
Butler	85	73
Chambersburg	87	75
Erie	83	72
Harrisburg	88	75
Johnstown	80	70
Lancaster	87	75
Meadville	83	71
New Castle	86	73
Philadelphia	87	75
Pittsburgh AP	84	72
Pittsburgh CO	86	72
Reading	86	73
Scranton/Wilkes-Barre	84	72
State College	84	72
Sunbury	86	73
Uniontown	85	74
Warren	83	72
West Chester	86	75
Williamsport	86	73
York	88	75
RHODE ISLAND		
Newport	82	73
Providence	83	73
SOUTH CAROLINA		
Anderson	90	75
Charleston AFB	89	79
Charleston	90	79
Columbia	93	77
Florence	90	78
Georgetown	88	79
Greenville	89	75
Greenwood	91	76
Orangeburg	93	77
Rock Hill	92	76
Spartanburg	89	75
Sumter-Shaw AFB	90	77
SOUTH DAKOTA		
Aberdeen	88	73
Brookings	89	73
Huron	90	73
Mitchel	90	73
Pierre	92	72
Rapid City	89	67
Sioux Falls	88	73
Watertown	88	73
Yankton	88	74
TENNESSEE		
Athens	90	75
Bristol-Tri City	87	73
Chattanooga	91	76
Clarksville	90	76
Columbia	91	76
Dyersburg	91	78
Greenville	88	74
Jackson	92	77
Knoxville	90	75
Memphis	93	78
Murfreesboro	91	76
Nashville	91	76
Tullahoma	91	75
TEXAS		
Abilene	97	74
Alice	95	79
Amarillo	93	70
Austin	97	77
Bay City	92	79
Beaumont	91	80
Beeville	95	79
Big Springs	95	72
Brownsville	92	79
Brownwood	96	75
Bryan	94	78
Corpus Christi	92	79
Corsicana	96	77
Dallas	97	77
Del Rio, Laughlin AFB	97	76
Denton	97	76
Eagle Pass	98	77
El Paso	96	68
Fort Worth	97	76
Galveston	88	80
Greenville	97	76
Harlingen	93	79
Houston AP	92	79
Houston CO	93	79
Huntsville	96	77
Killeen-Gray AFB	95	75
Lamesa	94	71
Laredo	99	77
Longview	95	78
Lubbock	94	71
Lufkin	94	78
McAllen	94	79
Midland	96	71
Mineral Wells	97	76
Palestine	96	78
Pampa	94	70
Pecos	96	71
Plainview	94	70
Port Arthur	91	80
San Angelo, Goodfellow AFB	97	73
San Antonio	96	76
Sherman Perrin AFB	95	76
Snyder	96	72
Temple	97	77
Tyler	95	78
Vernon	97	75
Victoria	94	79
Waco	97	77
Wichita Falls	98	75
UTAH		
Cedar City	89	62
Logan	88	63
Moab	96	63
Ogden	88	64
Price	89	62
Provo	94	64
Richfield	89	62
St. George	99	67
Salt Lake City	92	64
Vernal	86	62
VERMONT		
Barre	78	70
Burlington	82	71
Rutland	81	71
VIRGINIA		
Charlottesville	88	75
Danville	90	75
Fredericksburg	90	76
Harrisonburg	88	73
Lynchburg	88	75
Norfolk	89	77
Petersburg	90	77
Richmond	90	77
Roanoke	88	73
Staunton	88	73
Winchester	88	75
WASHINGTON		
Aberdeen	73	62
Bellingham	74	63
Bremerton	75	63
Ellensburg	87	63
Everett-Paine AFB	73	63
Kennewick	92	67
Longview	81	66
Moses Lake, Larson AFB	90	64
Olympia	79	64
Port Angeles	67	61
Seattle-Boeing Fld	77	65
Seattle CO	78	65
Seattle-Tacoma AP	76	63
Spokane	87	62
Tacoma-McChord AFB	79	64
Walla Walla	90	66
Wenatchee	92	65
Yakima	89	65
WEST VIRGINIA		
Beckley	79	70
Bluefield	79	70
Charleston	87	74
Clarksburg	87	74
Elkins	82	71
Huntington	89	75
Martinsburg	88	75
Morgantown	85	74
Parkersburg	88	75
Wheeling	84	72
WISCONSIN		
Appleton	83	72
Ashland	79	68
Beloit	88	75
Eau Claire	86	73
Fond du Lac	84	72
Green Bay	83	72
La Crosse	85	74
Madison	85	73
Manitowoc	83	72
Marinette	82	71
Milwaukee	84	73
Racine	85	74
Sheboygan	83	74
Stevens Point	86	73
Waukesha	84	73
Wausau	85	72
WYOMING		
Casper	87	60
Cheyenne	84	60
Cody	83	61
Evanston	82	57
Lander	85	61
Laramie	79	59
Newcastle	84	66
Rawlins	81	60
Rock Springs	82	57
Sheridan	88	63
Torrington	88	63

The dry bulb and wet bulb temperatures represent values which have been equalled or exceeded by 5% of the total hours during the months of June through September in the northern hemisphere and the months December through March in the southern hemisphere.

The data for Canadian stations are based on the month of July only.

AP = airport AFB = air force base CO = urban offices

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FIG. 11-3b

Dry Bulb/Wet Bulb Temperature Data[2]

Country and Station	Design Dry-Bulb °F 5%	Design Wet-Bulb °F 5%
ADEN		
Aden	98	82
AFGHANISTAN		
Kabul	93	64
ALGERIA		
Algiers	89	75
ARGENTINA		
Buenos Aires	86	75
Córdoba	93	74
Tucuman	96	74
AUSTRALIA		
Adelaide	91	68
Alice Springs	100	72
Brisbane	86	75
Darwin	91	81
Melbourne	86	68
Perth	93	73
Sydney	80	72
AUSTRIA		
Vienna	83	67
AZORES		
Lajes (Terceira)	77	71
BAHAMAS		
Nassau	88	79
BELGIUM		
Brussels	77	67
BELIZE		
Belize	89	81
BERMUDA		
Kindley AFB	85	78
BOLIVIA		
La Paz	68	56
BRAZIL		
Belem	87	78
Belo Horizonte	83	75
Brazilia	86	75
Curitiba	82	74
Fortaleza	89	78
Porto Alegre	89	75
Recife	86	77
Rio de Janeiro	90	78
Salvador	86	78
São Paulo	82	74
BULGARIA		
Sofia	84	69
BURMA		
Mandalay	101	80
Rangoon	95	82
CAMBODIA		
Phnom Penh	94	82
CANADA (Alberta)		
Calgary	79	62
Edmonton	79	65
Grande Prairie	78	62
Jasper	77	63
Lethbridge	84	65
McMurray	79	65
Medicine Hat	87	66
Red Deer	78	64
CANADA (British Columbia)		
Dawson Creek	76	62
Fort Nelson	78	64
Kamloops	88	65
Nanaimo	77	65
New Westminster	78	66
Penticton	87	67
Prince George	77	62
Prince Rupert	61	57
Trail	86	65
Vancouver	74	66
Victoria	70	60
CANADA (Manitoba)		
Brandon	83	70
Churchill	74	63
Dauphin	81	70
Flin Flon	79	67
Portage la Prairie	83	71
The Pas	79	68
Winnepeg	84	71
CANADA (New Brunswick)		
Campbellton	79	68
Chatham	82	69
Edmundston	80	69
Fredericton	82	70
Moncton	79	69
Saint John	75	66
CANADA (Newfoundland)		
Corner Brook	71	65
Gander	77	66
Goose Bay	77	64
St. John's	73	66
Stephenville	71	65
CANADA (Northwest Terr.)		
Fort Smith	78	65
Frobisher	59	51
Inuvik	75	61
Resolute	51	46
Yellowknife	74	62
CANADA (Nova Scotia)		
Amherst	79	68
Halifax	74	66
Kentville	80	69
New Glasgow	77	69
Sydney	77	68
Truro	78	70
Yarmouth	70	65
CANADA (Ontario)		
Belleville	82	73
Chatham	85	74
Cornwall	84	72
Hamilton	83	73
Kapuskasing	80	69
Kenora	80	70
Kingston	82	73
Kitchener	83	72
London	83	73
North Bay	79	68
Oshawa	84	73
Ottawa	84	72
Owen Sound	80	70
Peterborough	83	72
St. Catharines	83	73
Sarnia	84	73
Sault Ste. Marie	79	70
Sudbury	81	68
Thunder Bay	80	68
Timmins	81	68
Toronto	85	73
Windsor	86	74
CANADA (Prince Edward Island)		
Charlottetown	76	68
Summerside	77	68
CANADA (Quebec)		
Bagotville	80	68
Chicoutimi	80	68
Drummondville	82	71
Granby	83	72
Hull	84	72
Mégantic	81	71
Montréal	83	72
Québec	81	70
Rimouski	76	67
St. Jean	84	72
St. Jérome	83	72
Sept. Iles	70	63
Shawnigan	82	71
Sherbrooke	81	71
Thetford Mines	81	71
Trois Riviéres	82	71
Val d'Or	80	68
Valleyfield	84	72
CANADA (Saskatchewan)		
Estevan	86	69
Moose Jaw	86	68
North Battleford	82	66
Prince Albert	81	67
Regina	84	68
Saskatoon	83	67
Swift Current	87	67
Yorkton	80	68
CANADA (Yukon Territory)		
Whitehorse	74	58
CEYLON		
Colombo	88	80
CHILE		
Punta Arenas	64	54
Santiago	88	69
Valparaiso	77	65
CHINA		
Chungking	95	79
Shanghai	90	80
COLOMBIA		
Baranquilla	93	82
Bogotá	69	58
Cali	79	68
Medellin	84	72
CONGO		
Brazzaville	91	80
Kinasha (Leopoldville)	90	80
Stanleyville	90	80
CUBA		
Guantanamo Bay	92	80
Havana	89	80
CZECHOSLOVAKIA		
Prague	83	64
DENMARK		
Copenhagen	74	64
DOMINICAN REPUBLIC		
Santo Domingo	88	80
ECUADOR		
Guayacil	89	79
Quito	71	62
EL SALVADOR		
San Salvador	95	75
ETHIOPIA		
Addis Ababa	81	64
Asmara	80	63
FINLAND		
Helsinki	72	63
FRANCE		
Lyon	86	69
Marseilles	84	69
Nantes	80	67
Nice	83	72
Paris	83	67
Strasbourg	80	67
FRENCH GUIANA		
Cayenne	90	82
GERMANY		
Berlin	78	66
Hamburg	73	65
Hannover	75	65
Mannheim	82	68
Munich	80	64
GHANA		
Accra	89	79
GIBRALTER		
Gibralter	86	74
GREECE		
Athens	91	71
Thessaloniki	91	75
GREENLAND		
Narssarssuaq	61	52
GUATEMALA		
Guatemala City	81	67
GUYANA		
Georgetown	87	79
HAITI		
Port au Prince	93	80
HONDURAS		
Tegucigalpa	85	71
HONG KONG		
Hong Kong	90	80
HUNGARY		
Budapest	84	70
ICELAND		
Reykjavik	56	53
INDIA		
Ahmenabad	105	78
Bangalore	93	74
Bombay	92	81
Calcutta	96	82
Madras	101	83
Nagpur	107	78
New Delhi	105	82
INDONESIA		
Djakarta	88	78
Kupang	92	80
Makassar	88	79
Medan	90	79
Palembang	90	79
Surabaya	89	79
IRAN		
Abadan	110	81
Meshed	93	66
Tehran	98	73
IRAQ		
Baghdad	108	72
Mosul	110	72
IRELAND		
Dublin	70	62
Shannon	71	63
ISRAEL		
Jerusalem	92	69
Tel Aviv	91	72
ITALY		
Milan	84	74
Naples	86	72
Rome	89	72
IVORY COAST		
Abidjan	88	81
JAPAN		
Fukuoka	89	79
Sapporo	80	72
Tokyo	87	79
JORDAN		
Amman	92	68
KENYA		
Nairobi	78	65
KOREA		
Pyongyang	85	76
Seoul	87	78
LEBANON		
Beirut	90	76
LIBERIA		
Monrovia	88	81
LIBYA		
Bengasi	91	75
MADAGASCAR		
Rananarive	83	71
MALAYSIA		
Kuala Lumpur	92	81
Penang	92	80
Singapore	90	80
MARTINIQUE		
Fort de France	88	80
MEXICO		
Guadalajara	89	66
Mérida	94	77
Mexico City	79	59
Monterrey	93	77
Vera Cruz	88	82

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FIG. 11-3b (Cont'd)

Dry Bulb/Wet Bulb Temperature Data[2]

Country and Station	Design Dry-Bulb °F 5%	Design Wet-Bulb °F 5%
MOROCCO		
Casablanca	86	70
NEPAL		
Katmandu	86	76
NETHERLANDS		
Amsterdam	73	63
NEW GUINEA		
Manokwari	87	81
Point Moresby	90	79
NEW ZEALAND		
Auckland	76	65
Christ Church	76	66
Wellington	72	64
NICARAGUA		
Managua	92	79
NIGERIA		
Lagos	90	81
NORWAY		
Bergen	73	65
Oslo	74	64
PAKISTAN		
Chittagong	89	81
Karachi	95	81
Lahore	105	81
Peshwar	103	79
PANAMA & CANAL ZONE	91	80
Panama City		
PARAGUAY		
Asunción	96	80
PERU		
Lima	84	74
PHILIPPINES		
Manilla	91	81
POLAND		
Kraków	78	66
Warsaw	78	68
PORTUGAL		
Lisbon	83	67
PUERTO RICO		
San Juan	87	79
RUMANIA		
Bucharest	89	70
SAUDI ARABIA		
Dhahran	108	84
Jedda	100	83
Riyadh	106	76
SENEGAL		
Dakar	91	80
SOMALIA		
Mogadiscio	89	81
SOUTH AFRICA		
Capetown	86	70
Johannesburg	81	69
Pretoria	85	68
SOVIET UNION		
Alma Ata	83	67
Archangel	68	57
Kaliningrad	77	65
Krasnoyarsk	76	60
Kiev	81	67
Kharkov	82	67
Kuibyshev	81	66
Leningrad	72	63
Minsk	74	65
Moscow	78	65
Odessa	82	68
Petropavlovsk	65	56
Rostov on Don	84	68
Sverdlovsk	72	60
Tashkent	90	69
Tblisi	83	66
Vladivostok	74	68
Volgograd	86	69
SPAIN		
Barcelona	84	73
Madrid	89	67
Valencia	88	73
SUDAN		
Khartoum	104	75
SURINAM		
Paramaribo	90	81
SWEDEN		
Stockholm	72	60
SWITZERLAND		
Zurich	78	66
SYRIA		
Damascus	98	70
TAIWAN		
Tainan	90	82
Taipei	90	81
TANZANIA		
Dar es Salaam	88	81
THAILAND		
Bangkok	93	81
TRINIDAD		
Port of Spain	89	79
TUNISIA		
Tunis	96	74
TURKEY		
Adana	95	77
Ankara	89	66
Istanbul	86	73
Izmir	94	73
UNITED ARAB REPUBLIC		
Cairo	98	74
UNITED KINGDOM	69	62
Belfast		
Birmingham	73	63
Cardiff	73	62
Edinburgh	68	61
Glasgow	68	61
London	76	65
URUGUAY		
Montevideo	85	71
VENEZUELA		
Caracas	81	69
Maracaibo	95	83
VIET NAM		
Da Nang	93	85
Hanoi	95	84
Saigon	89	83
YUGOSLAVIA		
Belgrade	86	72

The dry bulb and wet bulb temperatures represent values which have been equalled or exceeded by 5% of the total hours during the months of June through September in the northern hemisphere and the months December through March in the southern hemisphere.

The data for Canadian stations are based on the month of July only.

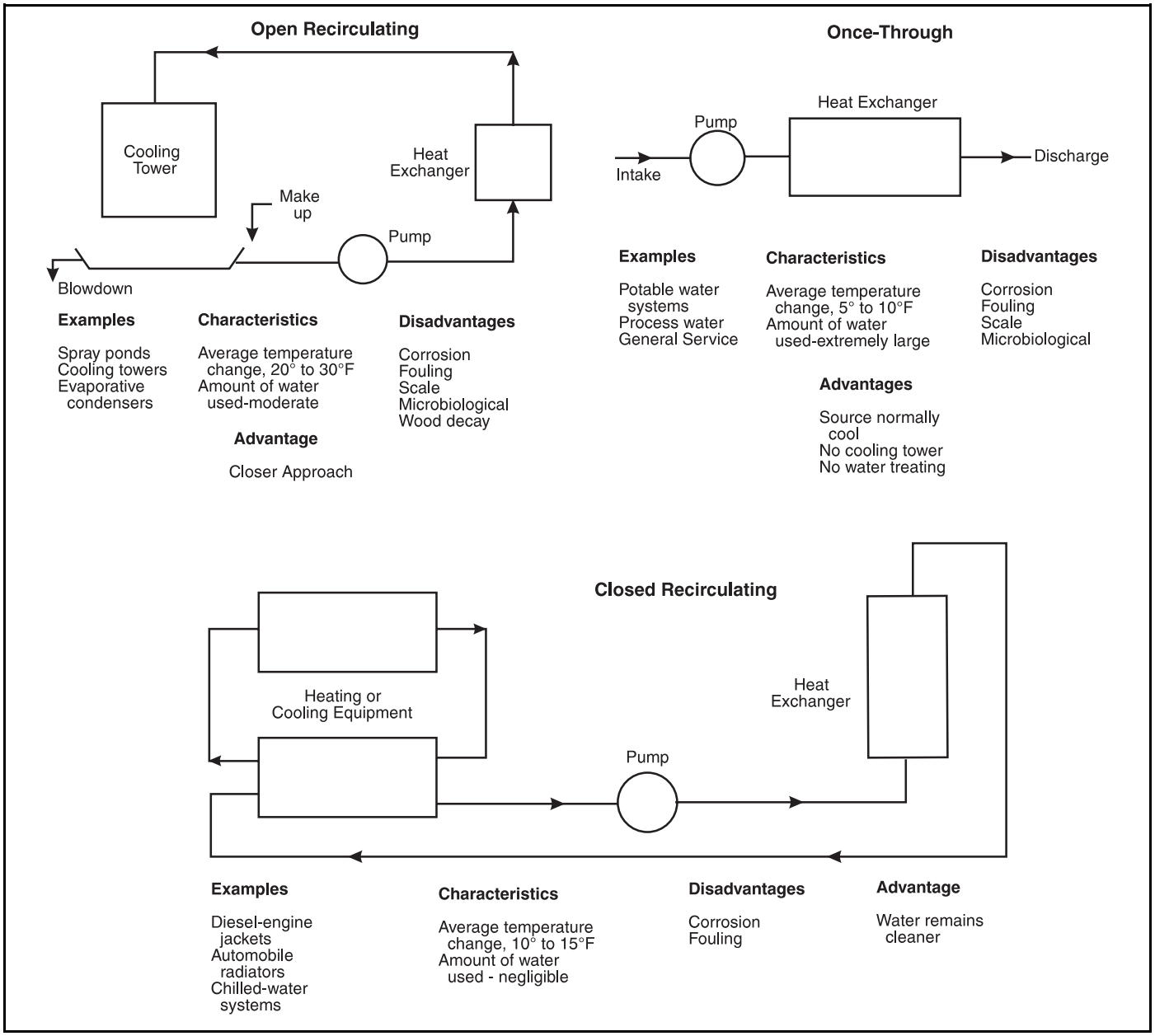
AP = airport AFB = air force base CO = urban offices

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FIG. 11-4

Cooling System Characteristics



Typical windage losses, expressed as percentages of the total system water circulation rate, for different evaporative equipment are as follows:

Spray ponds	1.0 to 5.0%
Atmospheric-draft towers	0.3 to 1.0%
Mechanical-draft towers	0.1 to 0.3%

Tower Location

Local heat sources upwind of the cooling tower can elevate the wet-bulb temperature of the air entering the tower. Interference occurs when a portion of the saturated air upwind of the tower contaminates the ambient air of a downwind tower. Drift and condensed water can cause corrosion problems with downwind equipment.

PERFORMANCE CHARACTERISTICS

The performance characteristics of various types of towers will vary with height, fill configuration, and flow arrangement — crossflow or counterflow; however, these factors have been taken into consideration in preparation of the Performance Characteristic Nomograph Fig. 11-5. When accurate characteristics of a specific tower are required, the cooling tower manufacturer should be consulted.

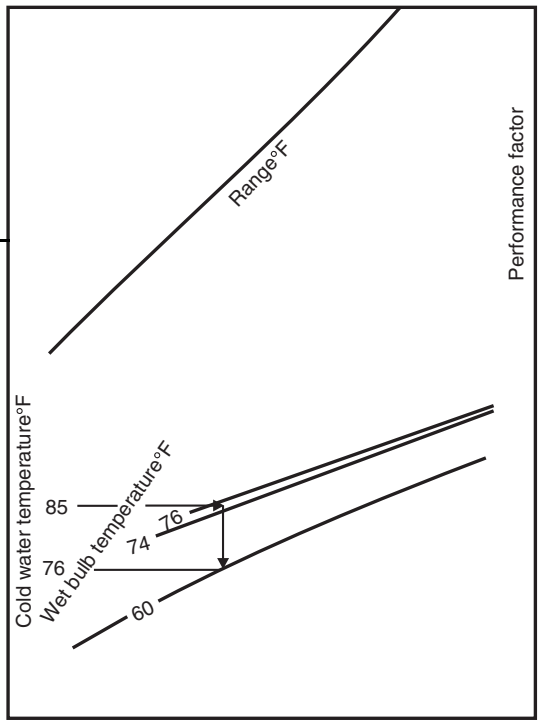
Performance tests on a cooling tower should be done in accordance with the Cooling Tower Institute (CTI) Acceptance

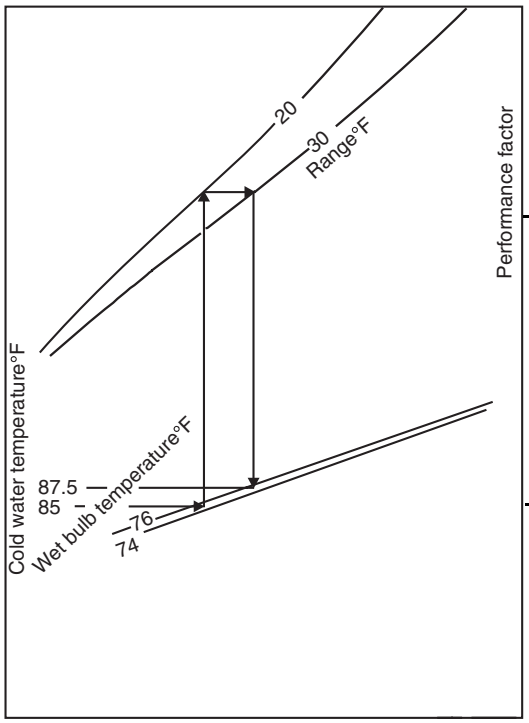
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Example 11-1 — Effect of Varying WB Temperature on Cold Water Temperature (CWT).

What is new CWT when WB changes from 75° to 60° with gpm and range remaining constant?

Enter Nomograph at 85° CWT, go horizontally to 75° WB, then vertically down to 60° WB, read new CWT of 76°.





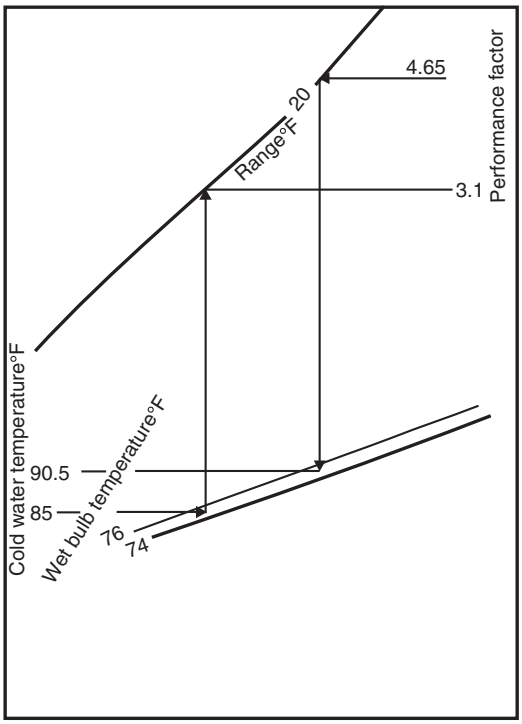
Example 11-2 — Effect of Varying Cooling Range on Cold Water Temperature.

What is new CWT when cooling range is changed from 20° to 30° (50% increase in heat load) with gpm and WB held constant?

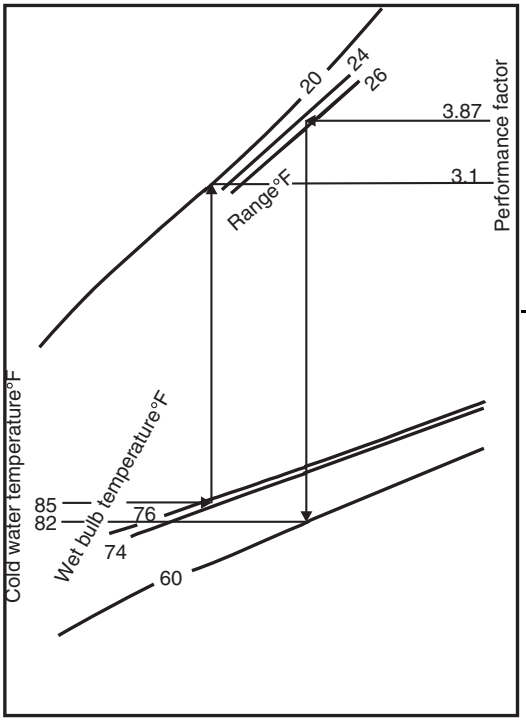
Enter Nomograph at 85° CWT, go horizontally to 75° WB, vertically to 20° R, horizontally to 30° R, vertically downward to 75° WB, read new CWT 87.5°.

Example 11-3 — Effect of Varying Water Circulating Rate and Heat Load on Cold Water Temperature.

What is new CWT when water circulation is changed from 1000 gpm to 1500 gpm (50% change in heat load at constant Range). Varying water rate, particularly over wide ranges, may require modifications to the distribution system. Enter Nomograph at 85° CWT, go horizontally to 75° WB, vertically to 20° R, horizontally to Performance Factor of 3.1. Obtain new PF by multiplying (3.1) (1500/1000) = 4.65, then enter Nomograph at PF of 4.65, go horizontally to 20° R, vertically down to 75° WB, read new CWT 90.5°.

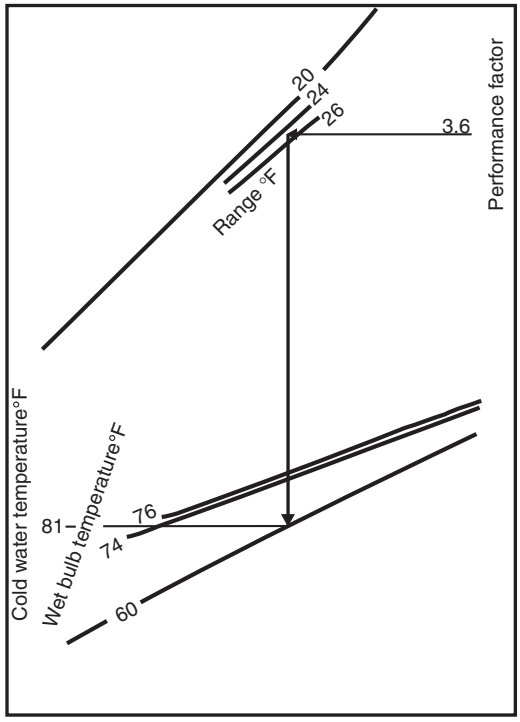


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Example 11-4 — Effect of Varying WB Temperature, Range, and Water Circulating Rate on Cold Water Temperature.

What is new CWT when the WB changes from 75° to 60°, R changes from 20° to 25°, gpm changes from 1000 to 1250 (25% change in heat load.) Enter Nomograph at 85° CWT, go horizontally to 75° WB, vertically to 20° R, horizontally read PF 3.1 then multiply (3.1) (1250/1000) = 3.87 (new PF). Enter Nomograph at PF = 3.87, go horizontally to 25° R, vertically down to 60° WB, read 82° new CWT.



Example 11-5 — Effect of Varying Fan HP Input on Cold Water Temperature.

What is new CWT if motor is changed from 20 HP to 25 HP in Example 11-4? The air flow rate varies as the cube root of the horsepower and performance varies almost directly with the ratio of water rate to air rate, therefore the change in air flow rate can be applied to the Performance Factor. Increasing the air flow rate (by installing a larger motor) decreases the Performance Factor. PF correction factor = $(25\ HP/20\ HP)^{1/3}$ = 1.077. Divide PF by PF correction factor to get new PF. Applying this to Example 11-4, we get 3.87/1.077 = 3.6. Enter Nomograph at 3.6 PF (instead of 3.87 PF) go horizontally to 25° R, vertically down to 60° WB, read 81° CWT.

Test Code and the American Society of Mechanical Engineers (ASME) test code.

Examples

The use of the Nomograph is illustrated by the following examples covering typical changes in operating conditions.

Assume a cooling tower is operating at known conditions of:

Flow	=	1000 gpm
Hot Water	=	105°F
Cold Water	=	85°F
Wet Bulb	=	75°F

This is commonly referred to as 105-85-75 or 20° Range (105° – 85°) and 10° Approach (85° – 75°).

Example 11-6 — The correction factor shown in Example 11-5 could also be used to increase gpm instead of decreasing CWT, as was done in Example 11-5. In Example 11-4, we developed a new CWT of 82° when circulating 1250 gpm at 25° R and 60°WB. If motor HP is increased from 20 to 25 under these conditions with PF correction factor = 1.077 (as shown in Example 11-5), GPM could be increased from 1250 to (1250) (1.077) = 1346 gpm.

Example 11-7 — Calculate the concentrations and blowdown rate for the following cooling tower:

Circulation Rate = 10,000 gpm

Water Temperature Drop Through Tower = 20°F

Type of Tower = Mechanical Induced Draft

Blowdown Rate = 20 gpm, or 0.2% of circulation rate

Therefore:

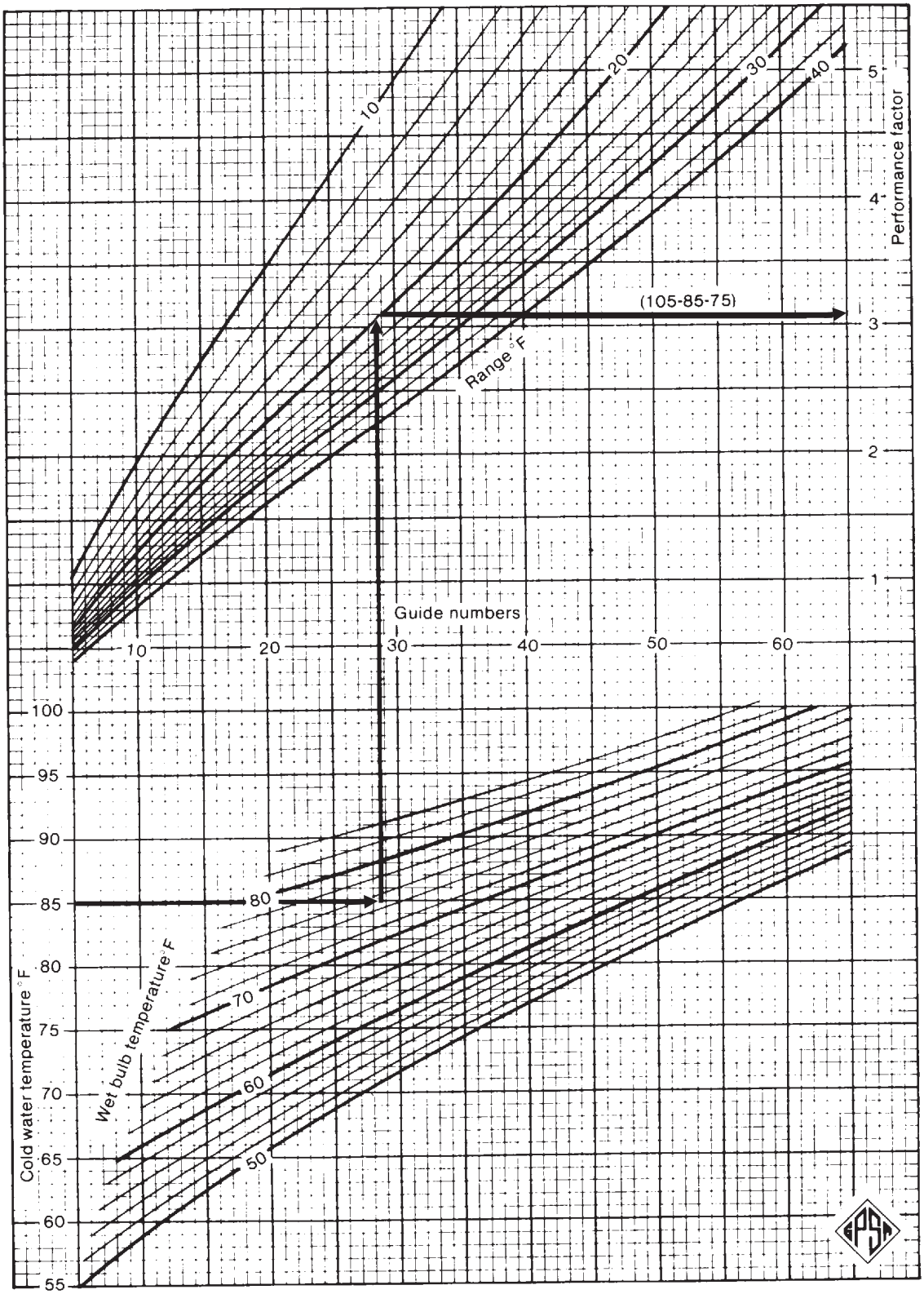
Evaporation Loss = 2% (1% for each 10° temperature drop)
(all rates are based on a percent of circulation rate)

Windage Loss = 0.3% (maximum for mechanical draft tower, p. 11-2)

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FIG. 11-5

Performance Characteristic Nomograph



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$$\text{Number of Concentrations (cycles)} = \frac{E + B}{B}$$

$$= \frac{2.0 + (0.2 + 0.3)}{(0.2 + 0.3)} = 5.0$$

If the resultant concentrations are excessive and a desired concentration of 4.0 is required, what must the blowdown rate be?

$$B = \frac{E}{\text{Cycles} - 1}$$

$$= \frac{2.0}{4.0 - 1} = 0.67\%$$

The windage component of B is 0.3%, therefore the blowdown rate required would be 0.67 – 0.3 = 0.37% or

(10,000 gpm) (0.0037) = 37 gpm

CONCENTRATION CYCLES

The concentration of compounds occurring in circulating water systems that can cause scaling or corrosion of equipment must be controlled at a desirable level. This concentration level, developed on each system, is based on the quality of makeup water and the water treating chemicals used to control corrosion or scaling. The concentration is usually reported as concentration cycles and refers to the number of times the compounds in the makeup water are concentrated in the blowdown water. For example, if the concentration in the makeup water were 125 ppm and the concentration of the blowdown were 563, the concentration cycles would be 563/125 or 4.5 cycles. The compounds are concentrated by the loss of water through evaporation and windage. The evaporation loss is based on the fact that 1,000 Btu are required to evaporate one (1) pound of water. The heat of evaporation is furnished by cooling the circulating water. One hundred (100) pounds of water must be cooled 10° to furnish this amount of heat, therefore, evaporation loss is 1% of tower circulation for each 10° temperature drop through the tower.

TYPES OF COOLING TOWERS

Cooling towers have two types of air flow: crossflow and counterflow. In crossflow towers, the air moves horizontally across the downward flow of water. In counterflow towers, the air moves vertically upward against the downward fall of the water.

There are many types and sizes of cooling towers:

Mechanical Draft Towers

Fans are used to move the air through the mechanical draft tower. The performance of the tower has a greater stability because it is affected by fewer psychrometric variables. The fans provide a means of regulating the air flow. Mechanical draft towers are characterized as either forced draft or induced draft.

Forced draft towers (Fig. 11-6) — The fan is located on the air stream entering the tower. This tower is characterized by high air entrance velocities and low exit velocities, therefore, the towers are susceptible to recirculation thus having a lower performance stability. The fans can also be subject to icing under conditions of low ambient temperature and high humidity.

FIG. 11-6

Mechanical Forced Draft Counterflow Tower

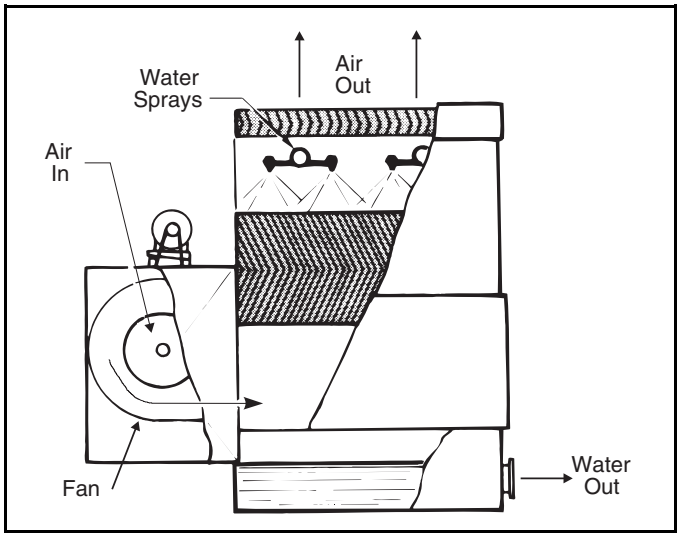


FIG. 11-7a

Mechanical Induced Draft Counterflow Tower

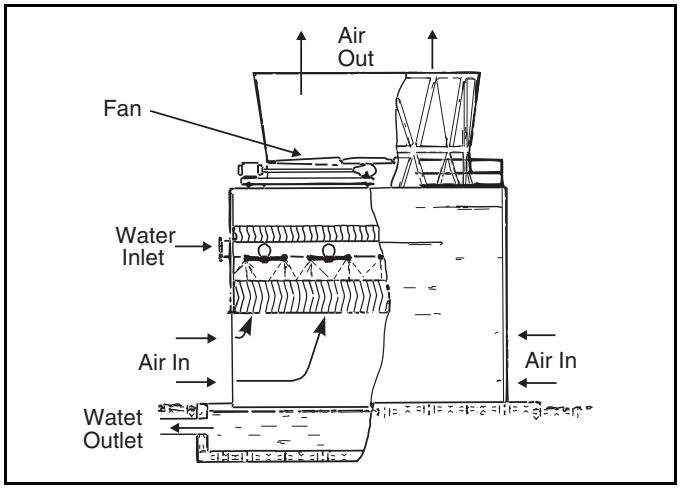
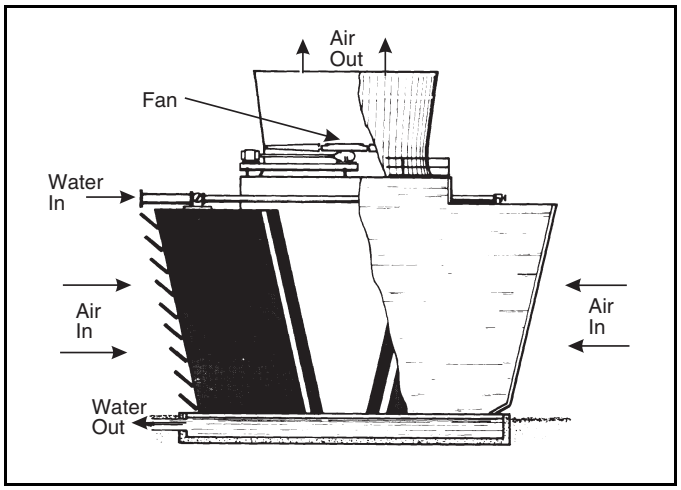


FIG. 11-7b

Mechanical Induced Draft Cross Flow Tower



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FIG. 11-8

Mechanical Draft Coil Shed Tower

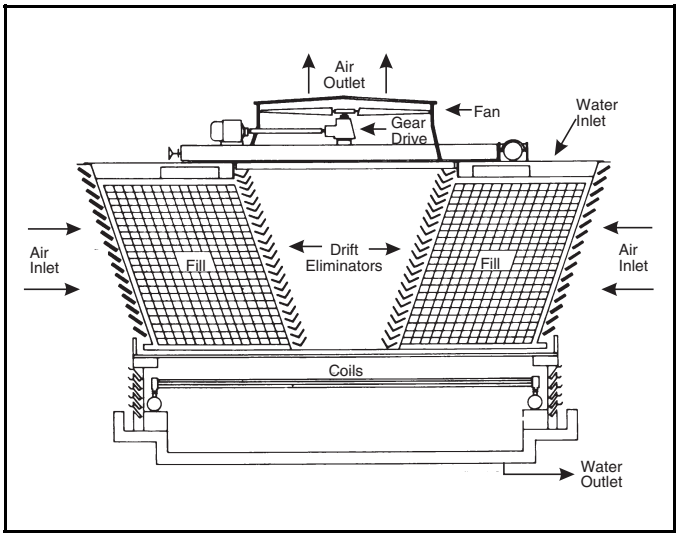
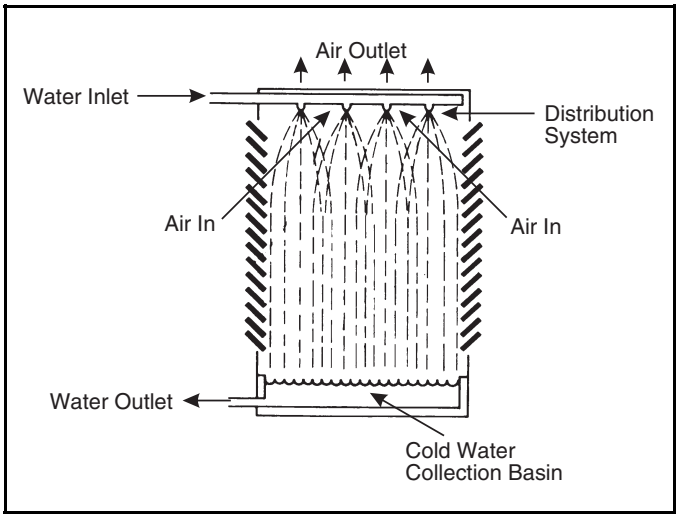


FIG. 11-9

Atmospheric Spray Tower

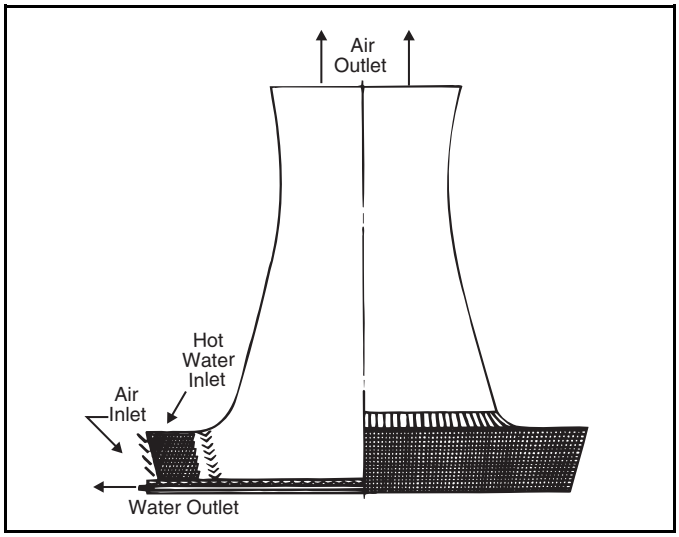


Induced draft towers (Fig. 11-7) — The fan is located on the air stream leaving the tower. This causes air exit velocities which are three to four times higher than their air entrance velocities. This improves the heat dispersion and reduces the potential for recirculation. Induced draft towers require about one (1) horsepower of input for every 8000 cfm of air.[3]

Coil shed towers (Fig. 11-8) — This application exists in many older cooling towers. The atmospheric coils or sections are located in the basin of the cooling tower. The sections are cooled by flooding the surface of the coils with cold water. Reasons for discontinued use were scaling problems, poor temperature control, and construction costs. This type tower can exist both as mechanical or natural draft.

FIG. 11-10

Hyperbolic Natural Draft Tower



Natural Draft Towers

Atmospheric spray towers (Fig. 11-9) — Cooling towers of this type are dependent upon atmospheric conditions. No mechanical devices are used to move the air. They are used when small sizes are required and when low performance can be tolerated.

Hyperbolic natural draft towers (Fig. 11-10) — These towers are extremely dependable and predictable in their thermal performance. A chimney or stack is used to induce air movement through the tower.

REFERENCES AND BIBLIOGRAPHY

1. American Society of Heating, Refrigerating and Air-Conditioning Engineers, "ASHRAE Handbook—Fundamentals," Table 1, Chapter 6, Atlanta, Georgia.
2. American Society of Heating, Refrigerating and Air-Conditioning Engineers, "ASHRAE Handbook—Fundamentals," Tables 1, 2 and 3, Chapter 24, Atlanta, Georgia.
3. Evans, Frank L., Jr., "Equipment Design Handbook for Refineries and Chemical Plants," 2nd ed., Gulf Publishing Company, Houston, Texas.

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FIG. 11-11

Properties of Saturated Air[1]

Temp.	Humidity Ratio	Volume ft³/lb dry air			Enthalpy Btu/lb dry air		
°F	W_s lbw/lb$_a$	v_a	v_{as}	v_s	h_a	h_{as}	h_s
–80	0.0000049	9.553	0.000	9.553	–19.221	0.005	–19.215
–78	0.0000057	9.604	0.000	9.604	–18.740	0.006	–18.734
–76	0.0000067	9.655	0.000	9.655	–18.259	0.007	–18.252
–74	0.0000078	9.705	0.000	9.705	–17.778	0.008	–17.770
–72	0.0000090	9.756	0.000	9.756	–17.298	0.010	–17.288
–70	0.0000104	9.807	0.000	9.807	–16.817	0.011	–16.806
–68	0.0000120	9.857	0.000	9.858	–16.336	0.013	–16.324
–66	0.0000139	9.908	0.000	9.908	–15.856	0.015	–15.841
–64	0.0000160	9.959	0.000	9.959	–15.375	0.017	–15.359
–62	0.0000184	10.009	0.000	10.010	–14.895	0.019	–14.876
–60	0.0000212	10.060	0.000	10.060	–14.414	0.022	–14.392
–58	0.0000243	10.111	0.000	10.111	–13.933	0.025	–13.908
–56	0.0000279	10.161	0.000	10.162	–13.453	0.029	–13.424
–54	0.0000319	10.212	0.001	10.213	–12.972	0.033	–12.939
–52	0.0000365	10.263	0.001	10.263	–12.492	0.038	–12.454
–50	0.0000416	10.313	0.001	10.314	–12.011	0.043	–11.968
–48	0.0000475	10.364	0.001	10.365	–11.531	0.050	–11.481
–46	0.0000541	10.415	0.001	10.416	–11.050	0.056	–10.994
–44	0.0000615	10.465	0.001	10.466	–10.570	0.064	–10.505
–42	0.0000699	10.516	0.001	10.517	–10.089	0.073	–10.016
–40	0.0000793	10.567	0.001	10.568	–9.609	0.083	–9.526
–38	0.0000898	10.617	0.002	10.619	–9.128	0.094	–9.034
–36	0.0001017	10.668	0.002	10.670	–8.648	0.106	–8.541
–34	0.0001150	10.719	0.002	10.721	–8.167	0.120	–8.047
–32	0.0001298	10.769	0.002	10.772	–7.687	0.136	–7.551
–30	0.0001465	10.820	0.003	10.822	–7.206	0.154	–7.053
–28	0.0001650	10.871	0.003	10.873	–6.726	0.173	–6.553
–26	0.0001858	10.921	0.003	10.924	–6.245	0.195	–6.051
–24	0.0002088	10.972	0.004	10.976	–5.765	0.220	–5.545
–22	0.0002346	11.022	0.004	11.027	–5.284	0.247	–5.038
–20	0.0002632	11.073	0.005	11.078	–4.804	0.277	–4.527
–18	0.0002950	11.124	0.005	11.129	–4.324	0.311	–4.013
–16	0.0003303	11.174	0.006	11.180	–3.843	0.348	–3.495
–14	0.0003694	11.225	0.007	11.232	–3.363	0.390	–2.973
–12	0.0004128	11.276	0.007	11.283	–2.882	0.436	–2.447
–10	0.0004608	11.326	0.008	11.335	–2.402	0.487	–1.915
–8	0.0005139	11.377	0.009	11.386	–1.922	0.543	–1.378
–6	0.0005726	11.427	0.010	11.438	–1.441	0.606	–0.835
–4	0.0006373	11.478	0.012	11.490	–0.961	0.675	–0.286
–2	0.0007088	11.529	0.013	11.542	–0.480	0.751	0.271
0	0.0007875	11.579	0.015	11.594	0.0	0.835	0.835
2	0.0008742	11.630	0.016	11.646	0.480	0.928	1.408
4	0.0009695	11.680	0.018	11.699	0.961	1.030	1.991
6	0.0010743	11.731	0.020	11.751	1.441	1.143	2.584
8	0.0011895	11.782	0.022	11.804	1.922	1.266	3.188
10	0.0013158	11.832	0.025	11.857	2.402	1.402	3.804
12	0.0014544	11.883	0.028	11.910	2.882	1.550	4.433
14	0.0016062	11.933	0.031	11.964	3.363	1.714	5.077
16	0.0017724	11.984	0.034	12.018	3.843	1.892	5.736
18	0.0019543	12.035	0.038	12.072	4.324	2.088	6.412
20	0.0021531	12.085	0.042	12.127	4.804	2.303	7.107
22	0.0023703	12.136	0.046	12.182	5.285	2.537	7.822
24	0.0026073	12.186	0.051	12.237	5.765	2.793	8.558
26	0.0028660	12.237	0.056	12.293	6.246	3.073	9.318
28	0.0031480	12.287	0.062	12.349	6.726	3.378	10.104
30	0.0034552	12.338	0.068	12.406	7.206	3.711	10.917
32	0.003790	12.389	0.075	12.464	7.687	4.073	11.760
34	0.004109	12.439	0.082	12.521	8.167	4.420	12.587
36	0.004452	12.490	0.089	12.579	8.648	4.793	13.441
38	0.004820	12.540	0.097	12.637	9.128	5.194	14.322
40	0.005216	12.591	0.105	12.696	9.609	5.624	15.233
42	0.005640	12.641	0.114	12.756	10.089	6.086	16.175
44	0.006094	12.692	0.124	12.816	10.570	6.582	17.152
46	0.006581	12.743	0.134	12.877	11.050	7.114	18.164
48	0.007103	12.793	0.146	12.939	11.531	7.684	19.215
50	0.007661	12.844	0.158	13.001	12.012	8.295	20.306
52	0.008259	12.894	0.171	13.065	12.492	8.949	21.441
54	0.008897	12.945	0.185	13.129	12.973	9.648	22.621
56	0.009580	12.995	0.200	13.195	13.453	10.397	23.850
58	0.010309	13.046	0.216	13.262	13.934	11.197	25.131
60	0.011087	13.096	0.233	13.329	14.415	12.052	26.467
62	0.011919	13.147	0.251	13.398	14.895	12.966	27.862
64	0.012805	13.198	0.271	13.468	15.376	13.942	29.318
66	0.013750	13.248	0.292	13.540	15.856	14.983	30.840
68	0.014758	13.299	0.315	13.613	16.337	16.094	32.431
70	0.015832	13.349	0.339	13.688	16.818	17.279	34.097
72	0.016976	13.400	0.365	13.764	17.299	18.543	35.841
74	0.018194	13.450	0.392	13.843	17.779	19.889	37.668
76	0.019491	13.501	0.422	13.923	18.260	21.323	39.583
78	0.020871	13.551	0.453	14.005	18.741	22.851	41.592
80	0.022340	13.602	0.487	14.089	19.222	24.479	43.701
82	0.023902	13.653	0.523	14.175	19.702	26.211	45.913
84	0.025563	13.703	0.561	14.264	20.183	28.055	48.238
86	0.027329	13.754	0.602	14.356	20.664	30.017	50.681
88	0.029208	13.804	0.646	14.450	21.145	32.105	53.250
90	0.031203	13.855	0.692	14.547	21.626	34.325	55.951
92	0.033323	13.905	0.742	14.647	22.107	36.687	58.794
94	0.035577	13.956	0.795	14.751	22.588	39.199	61.787
96	0.037972	14.006	0.852	14.858	23.069	41.871	64.940
98	0.040516	14.057	0.912	14.969	23.550	44.711	68.260
100	0.043219	14.107	0.976	15.084	24.031	47.730	71.761
102	0.046090	14.158	1.045	15.203	24.512	50.940	75.452
104	0.049140	14.208	1.118	15.326	24.993	54.354	79.346
106	0.052383	14.259	1.196	15.455	25.474	57.986	83.460
108	0.055826	14.309	1.279	15.588	25.955	61.844	87.799
110	0.059486	14.360	1.367	15.727	26.436	65.950	92.386
112	0.063378	14.411	1.462	15.872	26.917	70.319	97.237
114	0.067512	14.461	1.562	16.023	27.398	74.964	102.362
116	0.071908	14.512	1.670	16.181	27.879	79.906	107.786
118	0.076586	14.562	1.784	16.346	28.361	85.169	113.530
120	0.081560	14.613	1.906	16.519	28.842	90.770	119.612
122	0.086860	14.663	2.037	16.700	29.323	96.742	126.065
124	0.092500	14.714	2.176	16.890	29.805	103.102	132.907
126	0.098504	14.764	2.325	17.090	30.286	109.877	140.163
128	0.104910	14.815	2.485	17.299	30.767	117.111	147.878
130	0.111738	14.865	2.655	17.520	31.249	124.828	156.076
132	0.119023	14.916	2.837	17.753	31.730	133.066	164.796
134	0.126804	14.966	3.033	17.999	32.212	141.873	174.084
136	0.135124	15.017	3.242	18.259	32.693	151.294	183.987
138	0.144019	15.067	3.467	18.534	33.175	161.374	194.548
140	0.153538	15.118	3.708	18.825	33.656	172.168	205.824
142	0.163748	15.168	3.967	19.135	34.138	183.754	217.892
144	0.174694	15.219	4.245	19.464	34.620	196.183	230.802
146	0.186460	15.269	4.545	19.815	35.101	209.550	244.651
148	0.199110	15.320	4.869	20.189	35.583	233.932	259.514
150	0.212730	15.370	5.218	20.589	36.064	239.426	275.490
152	0.227429	15.421	5.596	21.017	36.546	256.158	292.705
154	0.243309	15.471	6.005	21.477	37.028	274.245	311.273
156	0.260512	15.522	6.450	21.972	37.510	293.849	331.359
158	0.279166	15.572	6.933	22.505	37.992	315.120	353.112
160	0.29945	15.623	7.459	23.082	38.474	338.263	376.737
162	0.32156	15.673	8.034	23.707	38.956	363.501	402.457
164	0.34572	15.724	8.664	24.388	39.438	391.095	430.533
166	0.37220	15.774	9.355	25.129	39.920	421.352	461.271
168	0.40131	15.825	10.117	25.942	40.402	454.630	495.032
170	0.43343	15.875	10.959	26.834	40.884	491.372	532.256
172	0.46905	15.926	11.894	27.820	41.366	532.138	573.504
174	0.50867	15.976	12.937	28.913	41.848	577.489	619.337
176	0.55294	16.027	14.103	30.130	42.331	628.197	670.528
178	0.60274	16.078	15.418	31.496	42.813	685.260	728.073
180	0.65911	16.128	16.909	33.037	43.295	749.871	793.166
182	0.72331	16.178	18.609	34.787	43.778	823.487	867.265
184	0.79703	16.229	20.564	36.793	44.260	908.061	952.321
186	0.88251	16.280	22.834	39.113	44.742	1006.149	1050.892
188	0.98272	16.330	25.498	41.828	45.225	1121.174	1166.399
190	1.10154	16.381	28.661	45.042	45.707	1257.614	1303.321
192	1.24471	16.431	32.477	48.908	46.190	1422.047	1468.238
194	1.42029	16.481	37.161	53.642	46.673	1623.758	1670.430
196	1.64070	16.532	43.046	59.578	47.155	1877.032	1924.188
198	1.92472	16.583	50.636	67.218	47.638	2203.464	2251.102
200	2.30454	16.633	60.793	77.426	48.121	2640.084	2688.205

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SECTION 12

Pumps & Hydraulic Turbines

Pumps

The most common types of pumps used in gas processing plants are centrifugal and positive displacement. Occasionally regenerative turbine pumps, axial-flow pumps, and ejectors are used.

Modern practice is to use centrifugal rather than positive displacement pumps where possible because they are usually less costly, require less maintenance, and less space. Conventional centrifugal pumps operate at speeds between 1200 and 8000 rpm. Very high speed centrifugal pumps, which can operate

FIG. 12-1

Nomenclature

A = cross-sectional area of plunger, piston, or pipe, sq in.
a = cross-sectional area of piston rod, sq in.
AC = alternating current
bbl = barrel (42 U.S. gallons)
bhp = brake horsepower
C = constant (Fig. 12-19)
C_p = specific heat at average temperature, Btu/(lb • °F)
cfs = cu ft/sec
D = displacement of reciprocating pump, gpm
DC = direct current
d = impeller diameter, in.
e = pump efficiency, fraction
g = 32.2 ft/sec^2 (acceleration of gravity)
gpm = U.S. gallons/minute
H = total equipment head, ft of fluid
h = head, ft of fluid pumped
hyd hp = hydraulic horsepower
k = factor related to fluid compressibility (Fig. 12-16)
K = type of pump factor (Eq. 12-18)
L = length of suction pipe, ft
L_s = stroke length, in.
m = number of plungers or pistons
NPSH = net positive suction head of fluid pumped, ft
NPSHA = NPSH available, ft
NPSHR = NPSH required, ft
n = speed of rotation, revolutions/minute (rpm)
n_s = specific speed (See Fig. 12-2 for units)
N = Polytropic exponent of charge gas. (For nitrogen, N = 1.4)
ΔP = differential pressure, psi
P = pressure, psia or psig
P_{vp} = liquid vapor pressure at pumping temperature, psia
psi = lb/sq in.
psia = lb/sq in. absolute
psig = lb/sq in. gauge
Q = rate of liquid flow, gpm
r = ratio of internal volume of fluid between valves, when the piston or plunger is at the end of the suction stroke, to the piston or plunger displacement.
RD = relative density to water at standard temperature
s = slip or leakage factor for reciprocating and rotary pumps
S = suction specific speed (units per Eq 12-7)
sp gr = specific gravity at average flowing conditions. Equal to RD
T = torque, ft lb
t_r = temperature rise, °F
u = impeller peripheral velocity, ft/sec
VE = volumetric efficiency, fraction
VE_o = overall volumetric efficiency
VE_ρ = volumetric efficiency due to density change
VE_l = volumetric efficiency due to leakage
V_{pd} = pulsation dampener volume, in^3
v = liquid mean velocity at a system point, ft/sec
z = elevation of a point of the system above (+) or below (–) datum of the pump. For piping, the elevation is from the datum to the piping centerline; for vessels and tanks, the elevation is from the datum to the liquid level.

Greek:

ρ = density at average flowing conditions, lb/ft^3
ρ_i = inlet density, lb/ft^3
ρ_o = outlet density, lb/ft^3
Δ = allowable pressure fluctuations as a percentage of mean pressure

Subscripts:

a = acceleration
ave = with P, average pressure in pulsating flow
bep = best efficiency point, for maximum impeller diameter
c = compression
d = discharge of pump
dv = discharge vessel
D = displacement
f = friction
i = inlet of equipment
l = leakage
max = with P, maximum acceptable peak pressure in pulsating flow
min = with P, minimum acceptable valley pressure in pulsating flow
o = outlet of equipment
ov = overall
p = pressure
r = rise

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FIG. 12-1 (Cont'd)

Nomenclature

s = static, suction of pump, specific, or stroke
sv = suction vessel
v = velocity
vp = vapor pressure
w = water
x = point x in the system
y = point y in the system
1 = impeller diameter or speed 1
2 = impeller diameter or speed 2

Alignment: The straight line relation between the pump shaft and the driver shaft.

Casing, Axially Split: A pump case split parallel to the pump shaft.

Casing, Radially Split: A pump case split transverse to the pump shaft.

Cavitation: A phenomenon that may occur along the flow path in a pump when the absolute pressure equals the liquid vapor pressure at flowing temperature. Bubbles then form which later implode when the pressure rises above the liquid vapor pressure.

Coupling: A device for connecting the pump shaft to the driver shaft consisting of the pump shaft hub and driver shaft hub, usually bolted together.

Coupling, Spacer: A cylindrical piece installed between the pump shaft coupling hub and driver shaft coupling hub, to provide space for removal of the mechanical seal without moving the driver.

Cutwater: The point of minimum volute cross-sectional area, also called the volute tongue.

Datum Elevation: The reference horizontal plane from which all elevations and heads are measured. The pumps standards normally specify the datum position relative to a pump part, e.g. the impeller shaft centerline for centrifugal horizontal pumps.

Diffuser: Pump design in which the impeller is surrounded by diffuser vanes where the gradually enlarging passages change the liquid velocity head into pressure head.

Displacement: The calculated volume displacement of a positive displacement pump with no slip losses.

Double Acting: Reciprocating pump in which liquid is discharged during both the forward and return stroke of the piston.

Duplex: Pump with two plungers or pistons.

Efficiency, Mechanical: The ratio of the pump hydraulic power output to pump power input.

Efficiency, Volumetric: The ratio of the pump suction or discharge capacity to pump displacement.

Head: The flowing liquid column height equivalent to the flowing liquid energy, of pressure, velocity or height above the datum, whose sum is the total head. Also used to express changes of energy such as the friction losses, the equipment total head and the acceleration head.

Head, Acceleration: The head equivalent to the pressure change due to changes in velocity in the piping system.

HPRT: Hydraulic power recovery turbine.

Impeller: The bladed member of the rotating assembly of a centrifugal pump which imparts the force to the liquid.

NPSHA: The total suction absolute head, at the suction nozzle, referred to the standard datum, minus the liquid vapor absolute pressure head, at flowing temperature available for a specific application. For reciprocating pumps it includes the acceleration head. NPSHA depends on the system characteristics, liquid properties and operating conditions.

NPSHR: The minimum total suction absolute head, at the suction nozzle, referred to the standard datum, minus the liquid vapor absolute pressure head, at flowing temperature, required to avoid cavitation. For positive displacement pumps it includes internal acceleration head and losses caused by suction valves and effect of springs. It does not include system acceleration head. NPSHR depends on the pump characteristics and speed, liquid properties and flow rate and is determined by vendor testing, usually with water.

Pelton Wheel: A turbine runner which turns in reaction to the impulse imparted by a liquid stream striking a series of buckets mounted around a wheel.

Recirculation Control: Controlling the quantity of flow through a pump by recirculating discharge liquid back to suction.

Rotor: The pump or power recovery turbine shaft with the impeller(s) mounted on it.

Rotor, Francis-type: A reverse running centrifugal pump impeller, used in a hydraulic power recovery turbine, to convert pressure energy into rotational energy.

Run-out: The point at the end of the head-capacity performance curve, indicating maximum flow quantity and usually maximum brake horsepower.

Runner: The shaft mounted device in a power recovery turbine which converts liquid pressure energy into shaft power.

Shut-off: The point on the pump curve where flow is zero, usually the point of highest total dynamic head.

Simplex: Pump with one plunger or piston.

Single Acting: Reciprocating pump in which liquid is discharged only during the forward stroke of the piston.

Slip: The quantity of fluid that leaks through the internal clearances of a positive displacement pump per unit of time. Sometimes expressed on a percentage basis.

Surging: A sudden, strong flow change often causing excessive vibration.

Suction, Double: Liquid enters on both sides of the impeller.

Suction, Single: Liquid enters one side of the impeller.

Throttling: Controlling the quantity of flow by reducing the cross-sectional flow area, usually by partially closing a valve.

Triplex: Pump with three plungers or pistons.

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Vanes, Guide: A series of angled plates (fixed or variable) set around the circumference of a turbine runner to control the fluid flow.

Volute, Double: Spiral type pump case with two cutwaters 180° apart, dividing the flow into two equal streams.

Volute, Single: Spiral type pump case with a single cutwater to direct the liquid flow.

Vortex Breaker: A device used to avoid vortex formation in the suction vessel or tank which, if allowed, would cause vapor entrainment in the equipment inlet piping.

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up to 23,000 rpm and higher, are used for low-capacity, high-head applications. Most centrifugal pumps will operate with an approximately constant head over a wide range of capacity.

Positive displacement pumps are either reciprocating or rotary. Reciprocating pumps include piston, plunger, and diaphragm types. Rotary pumps are: single lobe, multiple lobe, rotary vane, progressing cavity, and gear types. Positive displacement pumps operate with approximately constant capacities over wide variations in head, hence they usually are installed for services which require high heads at moderate capacities. A special application of small reciprocating pumps in gas processing plants is for injection of fluids (e.g. methanol and corrosion inhibitors) into process streams, where their constant-capacity characteristics are desirable.

Axial-flow pumps are used for services requiring very high capacities at low heads. Regenerative-turbine pumps are used for services requiring small capacities at high heads. Ejectors are used to avoid the capital cost of installing a pump, when a suitable motive fluid (frequently steam) is available, and are usually low-efficiency devices. These kinds of pumps are used infrequently in the gas+processing industry.

Fig. 12-1 provides a list of symbols and terms used in the text and also a glossary of terms used in the pump industry. Fig. 12-2 is a summary of some of the more useful pump equations. Fig. 12-3 provides guidance in selecting the kinds of pumps suitable for common services.

EQUIPMENT AND SYSTEM EQUATIONS

The energy conservation equation for pump or hydraulic turbine systems comes from Bernoulli's Theorem and relates the total head in two points of the system, the friction losses between these points and the equipment total head. Elevations are measured from the equipment datum.

The total head at any system point is:

$$h = z + h_p + h_v = z + \frac{2.31 \bullet P}{sp\ gr} + \frac{v^2}{2 \bullet g} \qquad \textbf{Eq 12-1}$$

The system friction head is the inlet system friction head plus the outlet system friction head:

$$h_f = h_{fx} + h_{fy} \qquad \textbf{Eq 12-2}$$

The equipment total head is the outlet nozzle total head minus the inlet nozzle total head:

$$H = h_o - h_i = z_o - z_i + \frac{2.31\ (P_o - P_i)}{sp\ gr} + \frac{v_o^2 - v_i^2}{2 \bullet g} \qquad \textbf{Eq 12-3}$$

When the elevation and size of inlet and outlet nozzles are the same, the equipment total head (H) equals the difference of pressure heads. H is positive for pumps and negative for HPRTs.

When using any suction-and-discharge-system points the following general equation applies:

$$z_x + \frac{2.31 \bullet P_x}{sp\ gr} + \frac{v_x^2}{2 \bullet g} - h_{fx} + H = z_y + \frac{2.31 \bullet P_y}{sp\ gr} + \frac{v_y^2}{2 \bullet g} + h_{fy} \qquad \textbf{Eq 12-4}$$

When the points are located in tanks, vessels or low velocity points in the piping, the velocity head is normally negligible, but may not be negligible in equipment nozzles. Note that the subscripts "i" and "o" are used for variables at pumps and HPRTs inlet and outlet nozzles, respectively, while the subscripts "s" and "d" are used only for variables at pumps suction and discharge nozzles. The subscripts "x" and "y" are used for variables at points in each inlet and outlet subsystem and usually are suction and discharge vessels. Also "x" and "y" are used for friction head from point "x" to equipment inlet nozzle and from equipment outlet nozzle to point "y".

The work done in compressing the liquid is negligible for practically incompressible liquids and it is not included in above equations. To evaluate the total head more accurately when handling a compressible liquid, the compression work should be included. If a linear relationship between density and pressure is assumed, the liquid compression head that substitutes for the difference of pressure heads in the above equations is:

$$H_c = 1.155\ (P_o - P_i)\left[\frac{1}{sp\ gr_o} + \frac{1}{sp\ gr_i}\right] \qquad \textbf{Eq. 12-5}$$

When the differential pressure is sufficiently high to have a density change of more than 10%, or when the pressure is near the fluid's critical pressure, the change in fluid density and other properties with pressure is not linear. In these cases, Equations 12-3 to 12-5 may not be accurate. A specific fluid properties relationship model is required in this case. For pure substances, a pressure-enthalpy-entropy chart may be used for estimating purposes by assuming an isentropic process. The pump manufacturer should be consulted for the real process, including the equipment efficiency, heat transfer, etc. to determine the equipment performance.

NET POSITIVE SUCTION HEAD

See NPSH definition in Fig. 12-1. There should be sufficient net positive suction head available (NPSHA) for the pump to work properly, without cavitation, throughout its expected capacity range. Ususally a safety margin of about 2 to 3 ft. of NPSHA above NPSHR is adequate. Cavitation causes noise, impeller damage, and impaired pump performance. Consideration must also be given to any dissolved gases which may affect vapor pressure. For a given pump, NPSHR increases with increasing flow rate. If the pump suction nozzle pressure is known

$$NPSHA = \frac{2.31 \bullet (P_i - P_{vp})}{sp\ gr} + z_i + \frac{v_i^2}{2 \bullet g} \qquad \textbf{Eq 12-6a}$$

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FIG. 12-2

Common Pump Equations

FLOW RATE						
Given ⇒ multiply by to get ⇓	ft^3/sec	bbl/day	bbl/h	UK gal./min	m^3/h	lb/h
US gal./min	449	0.0292	0.700	1.2009	4.40	1/(500 • sp gr)

PRESSURE							
Given ⇒ multiply by to get ⇓	kPa	ft water at 39.2°F	m water at 0°C	ft liquid	bar	* std atm 760 mm Hg at 0°C	kgf/cm^2
lb/in^2	0.145	0.4335	1.422	sp gr / 2.31	14.5038	14.6959	14.2233

DENSITY						
Given ⇒ multiply by to get ⇓	kg/m^3	lb/US gal	lb/UK gal	kg/lt	API gravity	Baumé gravity
lb/ft^3	0.062428	7.48047	6.22884	62.428	See Fig. 1-3	

$$h_p = \frac{144 \bullet P}{\rho} = \frac{2.31 \bullet P}{\text{sp gr}}$$

$$h_v = \frac{v^2}{2 \bullet g}$$

$$u = \frac{(d)(n)}{229}$$

$$v = \frac{(Q)\,(0.321)}{A}$$

$$\text{hyd hp} = \frac{Q \bullet H \bullet \text{sp gr}}{3{,}960} = \frac{Q \bullet \Delta P}{1{,}714} \quad **$$

$$\text{bhp} = \frac{\text{hyd hp}}{e} \text{ (for pumps)}$$

bhp = hyd hp • e (for turbines)

sp gr = specific gravity

$$\text{bhp} = \frac{Q \bullet H \bullet \text{sp gr}}{(3{,}960)(e)} = \frac{Q \bullet \Delta P}{1{,}714 \bullet e} \quad ** \text{ (for pumps)}$$

$$T = \frac{(\text{bhp})(5252)}{n}$$

$$n_s = \frac{n \bullet \sqrt{Q_{bep}}}{(H_{bep})^{3/4}} = \frac{n \bullet H_{bep}^{1/4} \bullet \sqrt{Q_{bep}}}{H_{bep}}$$

1 HP = 0.7457 kW
= 550 ft • lbf/s
= 33,000 ft • lbf/min

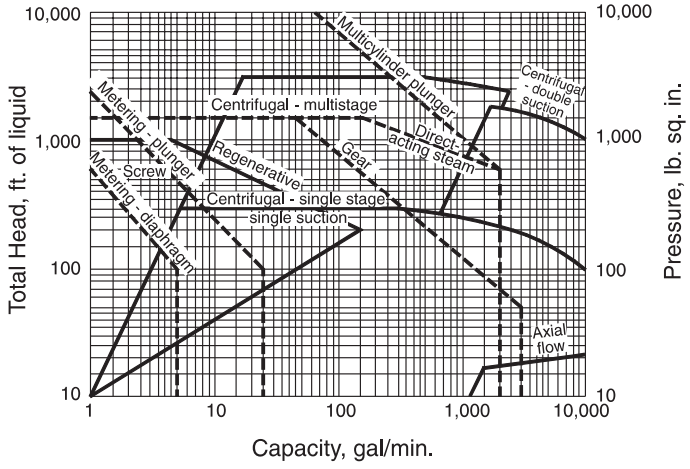
Water density at 60°F = 62.37 lb/ft^3
Standard gravity acceleration:
$g = 9.80665\ m/s^2 = 32.1740\ ft/s^2$

See Fig. 1-7 for viscosity relationships
*Standard atmospheric pressure:
1 atm = 760 mm Hg = 101.325 kPa = 14.696 psi
**See Eq. 12-3 and 12-4.

CENTRIFUGAL PUMPS AFFINITY LAWS 1: Values at initial conditions 2: Values at new conditions			
CHANGE ⇒	**SPEED**	**DIAMETER**	**SPEED AND DIAMETER**
Q_2 =	$Q_1 (n_2/n_1)$	$Q_1 (d_2/d_1)$	$Q_1 (d_2/d_1) (n_2/n_1)$
h_2 =	$h_1 (n_2/n_1)^2$	$h_1 (d_2/d_1)^2$	$h_1 [(d_2/d_1) (n_2/n_1)]^2$
bhp_2 =	$bhp_1 (n_2/n_1)^3$	$bhp_1 (d_2/d_1)^3$	$bhp_1 [(d_2/d_1) (n_2/n_1)]^3$
$NPSHR_2$ =	$NPSHR_1 (n_2/n_1)^2$	—	$NPSHR_1 (n_2/n_1)^2$

FIG. 12-3

Pump Selection Guide



Pump coverage chart based on normal ranges of operation of commercially available types.
Solid lines - use left ordinate, head scale. Broken lines - use right ordinate, pressure scale.

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When the pump suction nozzle pressure is not known, but the pressure at any point (x) of the suction system is known, NPSHA may be calculated with the following equation, where h_{fx} is the head friction loss from the point x to the suction nozzle. With commonly used suction pipe diameters, the velocity head may be negligible.

$$NPSHA = \frac{2.31 \bullet (P_x - P_{vp})}{sp\ gr} + z_x + \frac{v_x^2}{2 \bullet g} - h_{fx} \qquad \textbf{Eq 12-6b}$$

Moreover, when the suction system point is the specific case of the suction vessel, the equation is the following, where h_{fv} is the head friction loss from the suction vessel to the suction nozzle.

$$NPSHA = \frac{2.31 \bullet (P_{sv} - P_{vp})}{sp\ gr} + z_{sv} - h_{fsv} \qquad \textbf{Eq 12-6c}$$

The pressures in the above equations must be both absolute or gage; when using gage pressure both must be relative to the same atmospheric pressure. To convert a system pressure gage reading to absolute pressure add the existing local atmospheric pressure. The fluid vapor pressure must be at operating temperature. If the fluid vapor pressure is given in gage pressure, check which atmospheric pressure is reported. The use of the true local atmospheric pressure is very important in the cases of high altitude locations, and of a close margin of NPSHA over the NPSHR.

The pressure shall be measured at the pipe or nozzle centerline height; otherwise, adequate correction shall be made. Pay special attention to large pipe or nozzle diameters and the elevation of gage attached to them, pole or panel mounted instrument elevation, and different density fluid in the instrument line, see Hydraulic Institute Standards.[5]

To avoid vapor formation in the suction system, there must also be a Net Positive Pipe Pressure (NPPP) along it. Therefore, for every suction line point and operating condition the line pressure, at the top of the pipe must be higher than the fluid vapor pressure, being the pressure determined taking in account the pipe elevation.

The entrained and dissolved air or gases in the pump suction affects the pump performance, both mechanically and hydraulically, especially when the suction nozzle pressure is smaller than the suction vessel pressure. In centrifugal pumps it causes the reduction of capacity and discharge pressure, because of the reduced overall density; and also, at low flow, the impeller centrifugal action separates the gas from the liquid resulting in the cessation of the liquid flow. For these cases, specially designed centrifugal pumps with better tolerance to gas entrainment or rotary pumps may be considered, even though their capacity is affected by entrained and dissolved air or gases. See Hydraulic Institute Standards.[5]

Datum — The pump datum elevation is a very important factor to consider and should be verified with the manufacturer. Some common references are shown in Fig. 12-4. Some manufacturers provide two NPSHR curves for vertical can pumps, one for the first stage impeller suction eye and the other for the suction nozzle.

NPSH Correction Factors —NPSHR is determined from tests by the pump manufacturer using water near room temperature and is expressed in height of water. When hydrocarbons or high-temperature water are pumped, less NPSH is required than when cold water is pumped. Hydraulic Institute correction factors for various liquids are reproduced in Fig. 12-5. Some users prefer not to use correction factors to assure a greater design margin of safety.

NPSH and Suction Specific Speed — Suction specific speed is an index describing the suction capabilities of a first stage impeller and can be calculated using Eq. 12-7. Use half of the flow for double suction impellers.

$$S = \frac{n\sqrt{Q_{bep}}}{NPSHR_{\ bep}^{\ 3/4}} \qquad \textbf{Eq. 12-7}$$

Pumps with high suction speed tend to be susceptible to vibration (which may cause seal and bearing problems) when they are operated at other than design flow rates. As a result, some users restrict suction specific speed, and a widely accepted maximum is 11,000. For more details on the significance of suction specific speed, consult pump vendors or references listed in the References section.

Submergence — The suction system inlet or the pump suction bell should have sufficient height of liquid to avoid vortex formation, which may entrain air or vapor into the system and cause loss of capacity and efficiency as well as other problems such as vibration, noise, and air or vapor pockets. Inadequate reservoir geometry can also cause vortex formation, primarily in vertical submerged pumps. Refer to the Hydraulic Institute Standards[5] for more information.

CALCULATING THE REQUIRED DIFFERENTIAL HEAD

The following procedure is recommended to calculate the head of most pump services encountered in the gas processing industry. See Example 12-1.

1. Prepare a sketch of the system in which the pump is to be installed, including the upstream and downstream vessels (or some other point at which the pressure will not be affected by the operation of the pump). Include all components which might create frictional head loss (both suction and discharge) such as valves, orifices, filters, and heat exchangers.
2. Show on the sketch:
   - — The datum position (zero elevation line) according to the proper standard. See Fig. 12-4.
   - — The pump nozzles sizes and elevations.
   - — The minimum elevation (referred to the datum) of liquid expected in the suction vessel.
   - — The maximum elevation (referred to the datum) to which the liquid is to be pumped.
   - — The head loss expected to result from each component which creates a frictional pressure drop at design capacity.
3. Use appropriate equations (Eq 12-1 to Eq 12-4).
4. Convert all the pressures, frictional head losses, and static heads to consistent units (usually pounds per square inch or feet of head). In 5 and 6 below, any elevation head is negative if the liquid level is below the datum. Also, the vessel pressures are the pressures acting on the liquid surfaces. This is very important for tall towers. In the case of partitioned vessels, be sure to use the corresponding chamber pressure and liquid level elevation. And when the liquid is not a continuous phase, or it is not clear where the liquid level is, as in the case of

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packed fractionating towers, consider only the piping and exclude such vessels from the system.

5. Add the static head to the suction vessel pressure, then subtract the frictional head losses in the suction piping. This gives the total pressure (or head) of liquid at the pump suction flange.
6. Add the discharge vessel pressure, the frictional head losses in the discharge piping system, and the discharge static head. This gives the total pressure (or head) of liquid at the pump discharge. According to the type of capacity and head controls, pump type and energy conservation, required for the particular situation, provide a head and/or a flow additional margin to provide a good control. A control valve to throttle the discharge or to recirculate the flow, or a variable speed motor, etc. may be the options to provide good control.
7. Calculate the required pump total head by subtracting the calculated pump suction total pressure from the calculated pump discharge total pressure and converting to head.
8. It is prudent to add a safety factor to the calculated pump head to allow for inaccuracies in the estimates of heads and pressure losses, and pump design. Frequently a safety factor of 10% is used, but the size of the factor used for each pump should be chosen with consideration of:
    - The accuracy of the data used to calculate the required head
    - The cost of the safety factor
    - The problems which might be caused by installing a pump with inadequate head.

Example 12-1 — Liquid propane, at its bubble point, is to be pumped from a reflux drum to a depropanizer. The maximum flow rate is expected to be 360 gpm. The pressures in the vessels are 200 and 220 psia respectively. The specific gravity of propane at the pumping temperature (100°F) is 0.485. The elevations and estimated frictional pressure losses are shown on Fig.12-6. The pump curves are shown in Fig. 12-7. The pump nozzles elevations are zero and the velocity head at nozzles is negligible.

Required differential head is determined as follows:

Absolute Total Pressure at Pump Suction

Reflux drum		200.0 psia
Elevation	20 ft. • 0.485/2.31 =	+4.2 psi
Friction	piping	–0.5 psi
	valves	–0.2 psi
		203.5 psia
		= 188.8 psig

Absolute Total Pressure at Pump Discharge

Tower		220.0 psia
Elevation	74 ft • 0.485/2.31 =	+15.5 psi
Friction	piping	+3.0 psi
	valves	+2.0 psi
	orifice	+1.2 psi
	filter	+13.0 psi
	check valve	+1.0 psi
	control valve	+9.0 psi
		264.7 psia
		= 250.0 psig

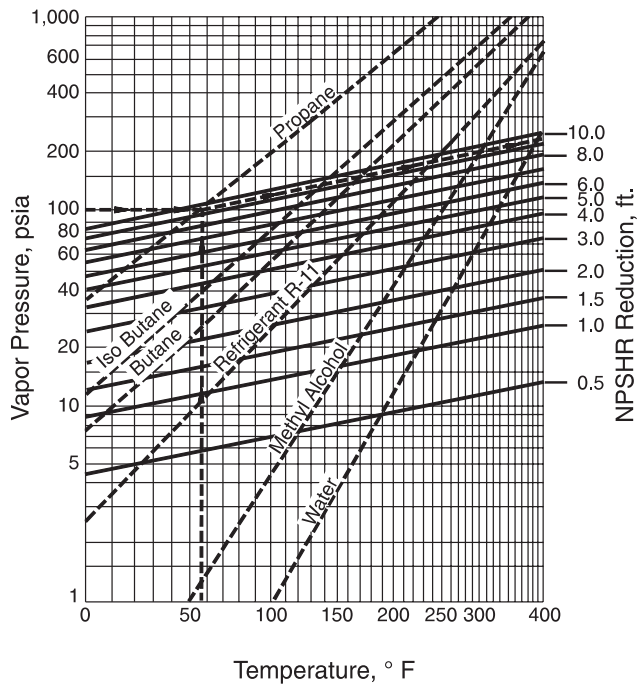
FIG. 12-4

Datum elevation

Pump type	Standard	Datum elevation
Centrifugal, horizontal	API 610[1] Hydraulic Institute[5]	Shaft centerline
Centrifugal, vertical in-line	API 610[1]	Suction nozzle centerline
Centrifugal, other vertical	API 610[1]	Top of the foundation
Centrifugal, vertical single suction, volute and diffused vane type	Hydraulic Institute[5]	Entrance eye to the first stage impeller
Centrifugal, vertical double suction	Hydraulic Institute[5]	Impeller discharge horizontal centerline
Vertical turbine. Line shaft and submersible types	AWWA E101[18]	Underside of the discharge head or head baseplate
Reciprocating	Hydraulic Institute[5]	Suction nozzle centerline

FIG. 12-5

NPSHR Reduction for Centrifugal Pumps Handling Hydrocarbon Liquids and High Temperature Water



This chart has been constructed from test data obtained using the liquids shown. For applicability to other liquids refer to the Hydraulic Institute Standards.[5]

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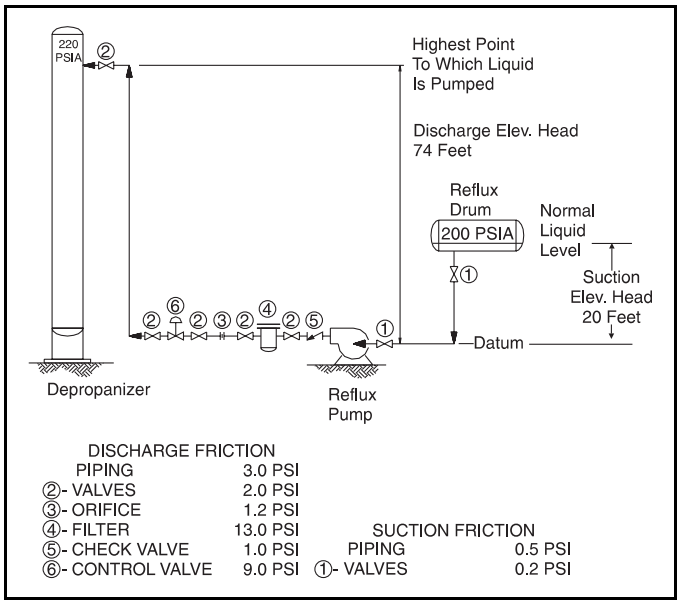
Differential pressure = 250.0 – 188.8 = 61.2 psi

$$\text{Differential head} = H = \frac{(61.2)\,(2.31)}{0.485} = 292 \text{ ft}$$

10% safety factor	30 ft
Required differential head (H)	322 ft

FIG. 12-6

Example 12-1 Depropanizer



Calculation of NPSHA

Reflux drum pressure		200.0 psia
Elevation	20 ft • 0.485/2.31 =	+4.2 psi
Friction	valves =	–0.2 psi
	piping =	–0.5 psi
Fluid vapor pressure		–200.0 psia
		3.5 psi
NPSHA	$\frac{3.5 \cdot 2.31}{0.485}$	= 16.7 ft

This NPSHA result is adequate when compared to the 9 ft. of NPSHR in the curve shown in Fig. 12-7.

Calculation of Hydraulic Power

$$\text{hyd hp} = \frac{Q \cdot H \cdot \text{sp gr}}{3960} \quad \textbf{(from Fig. 12-2)}$$

$$\text{hyd hp} = \frac{(360)\,(322)\,(0.485)}{3960} = 14.2 \text{ hp}$$

Calculation of Actual Horsepower

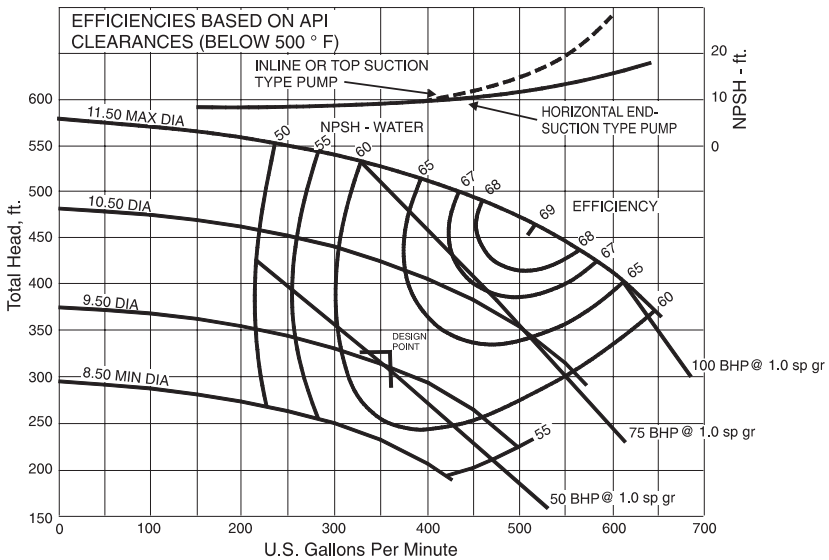
$$\text{bhp} = \frac{\text{hyd hp}}{e} \quad \textbf{(from Fig. 12-2)}$$

Fig. 12-7 is the performance curve of the selected pump. The efficiency at rated capacity and required head is 62%, with a brake horsepower calculated as follows:

$$\text{bhp} = \frac{14.2 \text{ hp}}{0.62} = 22.9 \text{ bhp}$$

FIG. 12-7

Depropanizer Reflux Pump for Example 12-1



DEPROPANIZER REFLUX PUMP			
MAX IMPELLER DIA 11.50 IN	3 X 4 X 11.5 SINGLE STAGE PUMP		
MIN DIA 8.50 IN	DIA IMPELLER 9.75 IN	IMPELLER PATT	3,560 RPM
EYE AREA 14.2 SQ IN	NPSH REQUIRED 9 FT	REFERENCE	CURVE NO. CA-307-3

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Motor Sizing — The maximum flow is 500 gpm with a head of 240 feet for this particular pump impeller size, which results in a brake horsepower requirement of 26.2 bhp at run-out (i.e., end of head curve). Therefore a 30 hp motor is selected for the pump driver to provide "full curve" protection.

CENTRIFUGAL PUMPS

Figs. 12-8a through 12-8e are cross-sectional drawings showing typical configurations for five types of centrifugal pumps. A guide to selecting centrifugal pumps is shown in Fig. 12-9. Horizontal centrifugal pumps are more common; however, vertical pumps are often used because they are more compact and, in cold climates, may need less winterizing than horizontal pumps. The total installed cost of vertical pumps is frequently lower than equivalent horizontal pumps because they require smaller foundations and simpler piping systems.

Vertical can pumps are often used for liquids at their bubble-point temperature because the first stage impeller is located below ground level and therefore requires less net positive suction head at the suction flange. The vertical distance from the suction flange down to the inlet of the first stage impeller provides additional NPSHA.

Centrifugal Pump Theory

Centrifugal pumps increase the pressure of the pumped fluid by action of centrifugal force on the fluid. Since the total head produced by a centrifugal pump is independent of the density of the pumped fluid, it is customary to express the pressure increase produced by centrifugal pumps in feet of head of fluid pumped.

Operating characteristics of centrifugal pumps are expressed in a pump curve similar to Fig. 12-7. Depending on impeller design, pump curves may be "drooping," "flat," or "steep." Fig. 12-10 shows these curves graphically. Pumps with drooping curves tend to have the highest efficiency but may be undesirable because it is possible for them to operate at either of two flow rates at the same head. The influence of impeller design on pump curves is discussed in detail in Hydraulic Institute Standards.[5]

Specific Speed

Specific speed gives an indication of the impeller shape and pump characteristics, as can be seen in the Fig. 12-11, from the Hydraulic Institute Standards. The ratios of major dimensions vary uniformly with specific speed. Specific speed is given by the equation in Fig 12-2.

Affinity Laws

The relationships between rotational speeds, impeller diameter, capacity, head, power, and NPSHR for any particular pump are defined by the affinity laws (See Fig. 12-2 for affinity laws). These equations are to predict new curves for changes in impeller diameter and speed.

The capacity of a centrifugal pump is directly proportional to its speed of rotation and its impeller diameter. The total pump head developed is proportional to the square of its speed and its impeller diameter. The power consumed is proportional to the cube of its speed and its impeller diameter. The NPSHR is proportional to the square of its speed.

These equations apply in any consistent set of units but only apply exactly if there is no change of efficiency when the rotational speed is changed. This is usually a good approximation if the change in rotational speed is small.

A different impeller may be installed or the existing modified. The modified impeller may not be geometrically similar to the original. An approximation may be found if it is assumed that the change in diameter changes the discharge peripheral velocity without affecting the efficiency. Therefore, at equal efficiencies and rotational speed, for small variations in impeller diameter, changes may be calculated using the affinity laws.

These equations do not apply to geometrically similar but different size pumps. In that case dimensional analysis should be applied.

The affinity equations apply to pumps with radial flow impellers, that is, in the centrifugal range of specific speeds, below 4200. For axial or mixed flow pumps, consult the manufacturer. See Fig. 12-2 for specific speed equation.

Efficiency

Hydraulic Institute charts reproduced in Fig 12-13 and Fig 12-14, provide centrifugal pumps optimum generally attainable efficiency vs. capacity for several pump types, corrected according to specific speed and maximum and minimum deviation.

Viscosity

Most liquids pumped in gas processing plants have viscosities in the same range as water. Thus they are considered "nonviscous" and no viscosity corrections are required. Occasionally fluids with viscosities higher than 5 centistokes are encountered (e.g. triethylene glycol, 40 cs at 70°F) and corrections to head, capacity, and power consumption may be required.

Viscosity correction charts and the procedures for using them are included in Hydraulic Institute Standards.[5]

Matching the Pump to the System Requirements

A pump curve depicts the relationship between the head and capacity of a pump. A system curve shows the relationship between the total head difference across the system and the flow rate through it. The total head difference consists of three components: static (gravity) head, pressure head, and headloss due to friction. Static and pressure heads do not change with flow. However, frictional losses usually increase approximately as the square of the flow rate through the system. If the system curve is plotted with the same units as the pump curve, it can be superimposed as shown in Fig. 12-12.

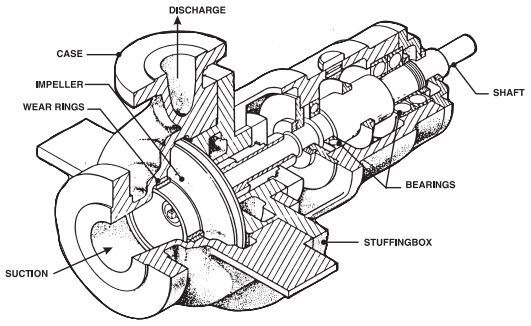
For pump selection, the shape and slope of the pump curve shall be considered in its position with respect to the system curve. When the curves are approximately perpendicular to each other, the change in the operating point position due to deviations in the curves will be minimum. In addition, the shape and slope shall be considered when several pumps are used in series and/or parallel operation to produce the desired range of flow and/or operating pressure. Refer to Fig. 12-15 and Fig. 12-16 for series and parallel operation.

Throttling Control — If a centrifugal pump and a system were matched as shown in Fig. 12-12, the flow rate through the system will be "A" unless some kind of flow control is provided. Control usually is provided by throttling a valve in the discharge piping of the pump, which creates extra frictional

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FIG. 12-8a

Horizontal Single Stage Process Pump



Pump Type: Overhung Impeller, Separately Coupled, Single Stage, Centerline Support, API 610

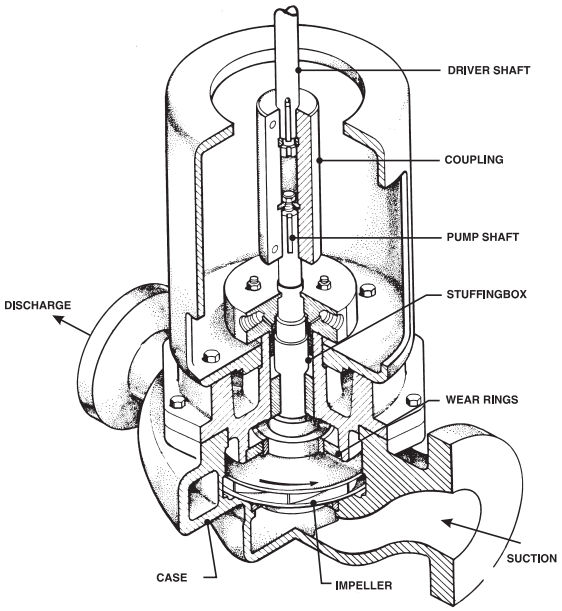
Range: 10-10,000 USGPM; 50-700 feet TH; 1200-3600 rpm

Typical Application: Reflux, circulation, booster, boiler feed

Courtesy of Bingham – Willamette Ltd.

FIG. 12-8b

Vertical Inline Pump



Pump Type: Inline, Separately Coupled, Single Stage, Rigid Coupling, API 610

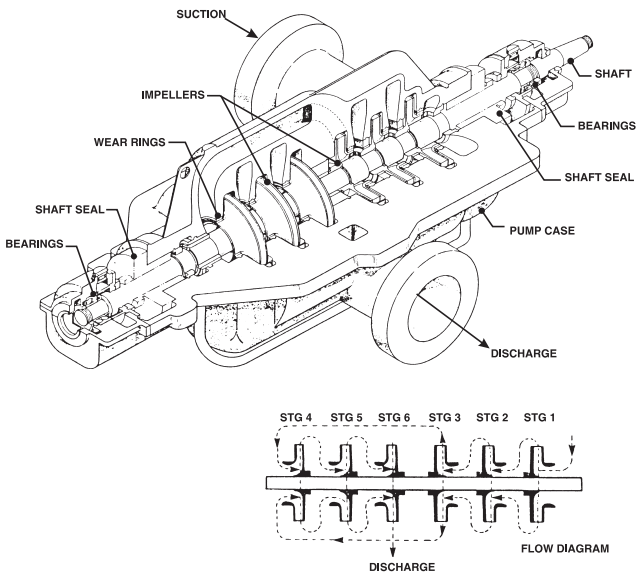
Range: 10-3500 USGPM; 50-700 feet TH; 1200-3600 rpm

Typical Application: Reflux, circulation, booster, boiler feed

Courtesy of Bingham – Willamette Ltd.

FIG. 12-8c

Horizontal Multi-Stage Pump



Pump Type: Horizontal, multi-stage, opposed impellers, axial split case, separately coupled

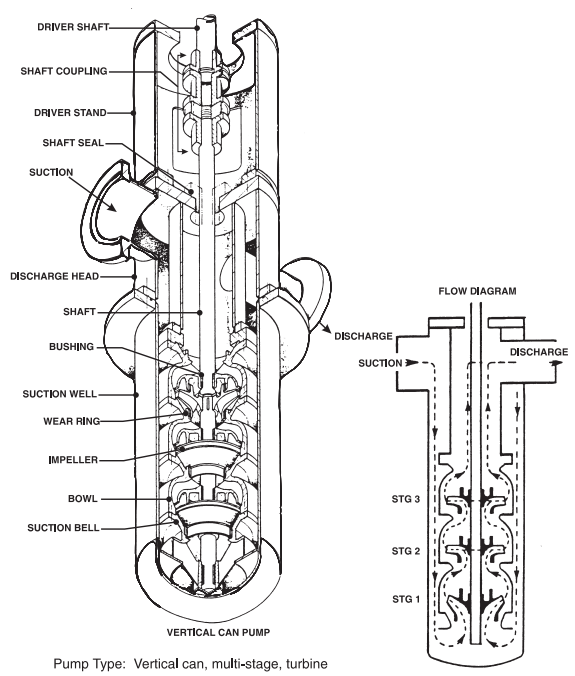
Range: 10-3000 USGPM; 800-10,000 feet TH; 1800-7000 rpm

Typical Application: Treating solution charge, absorption oil, boiler feed, pipeline, water and NGL injection

Courtesy of Bingham – Willamette Ltd.

FIG. 12-8d

Vertical Can Pump



Pump Type: Vertical can, multi-stage, turbine

Range: 50-10,000 USGPM; 100-3,000 feet TH; 1800-3600 rpm

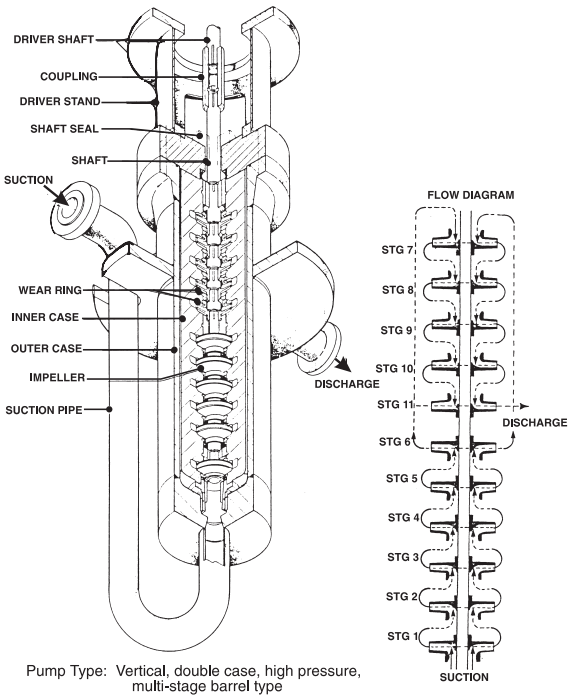
Typical Application: Loading, transfer, pipeline booster, boiler feed

Courtesy of Bingham – Willamette Ltd.

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FIG. 12-8e

Vertical, High Pressure, Double Case, Multi-Stage Pump



Pump Type: Vertical, double case, high pressure, multi-stage barrel type

Range: 50-1000 USGPM; 500-8000 feet TH; 3600 rpm

Typical Application: High pressure injection, ethane product, miscible flood, boiler feed

Courtesy of Bingham – Willamette Ltd.

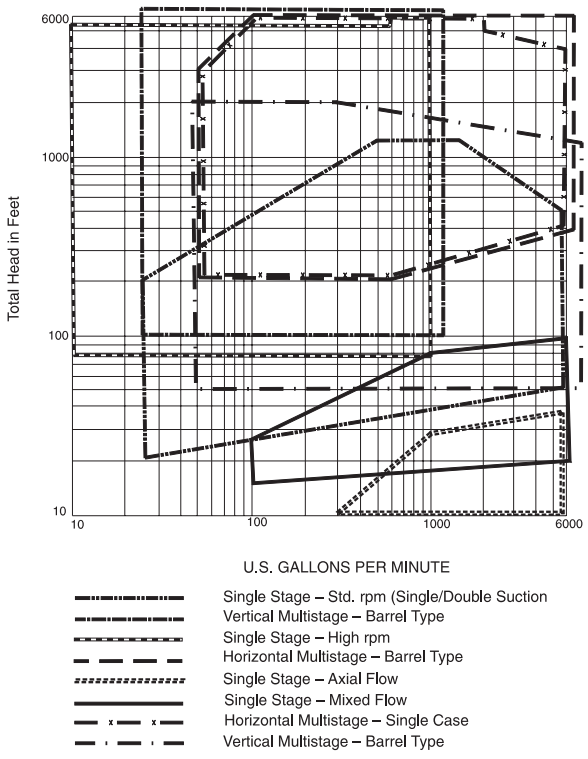
losses so that pump capacity is reduced to that required. In Fig. 12-12, the required flow rate is represented by "B." Required amount of extra frictional losses to achieve a flow rate of "B" is represented on Fig. 12-12 by the difference between "H_B-PUMP" and "H_B-SYSTEM." Frequently the throttling valve is an automatic control valve which holds some plant condition constant (such as liquid level, flow rate, or fluid temperature). This control method consumes energy since it artificially increases the system resistance to flow.

Recirculation Control — Pump capacity can also be controlled by recirculating a portion of the pumped fluid back to the suction. This control method is used more frequently for positive displacement pumps than for centrifugal pumps, since the discharge of most positive displacement pumps should not be throttled. This control method should be used with caution for centrifugal pumps, since a wide-open recirculation may result in a head so low that the pumped fluid will be circulated back to the suction at an extremely high rate, causing high power consumption, increase in fluid temperature, and possibly cavitation, as well as possibly overloading the driver.

Speed Control — Another way of regulating centrifugal pump capacity is to adjust the rotational speed of the pump. This is frequently not easily done because most pumps are driven by fixed-speed motors. However, pumps controlled by adjusting the rotational speed often consume substantially less energy than those controlled in other ways. The changed power consumption can be calculated by Eq. 12-8, which assumes that the frictional head is proportional to the square of the flow rate.

FIG. 12-9

Pump Selection Guide — Centrifugal Pumps



Courtesy of Bingham – Willamette Ltd.

$$bhp_2 = bhp_1\left(\frac{e_1}{e_2}\right)\left[\frac{h_s\,(Q_2/Q_1) + h_{f1}\,(Q_2/Q_1)^3}{h_s + h_{f1}}\right] \qquad \textbf{Eq 12-8}$$

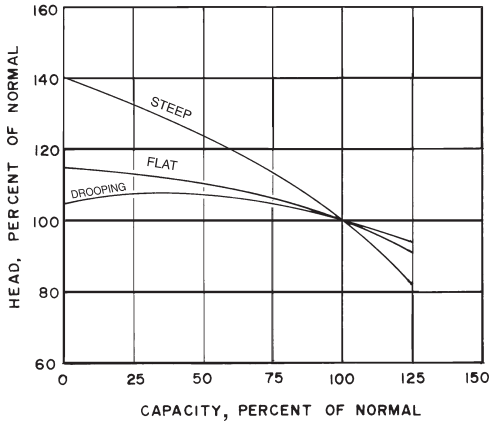
subscript 1 refers to initial flow rate

subscript 2 refers to the changed flow rate

h_s (static) is equivalent to the zero flow system total head

FIG. 12-10

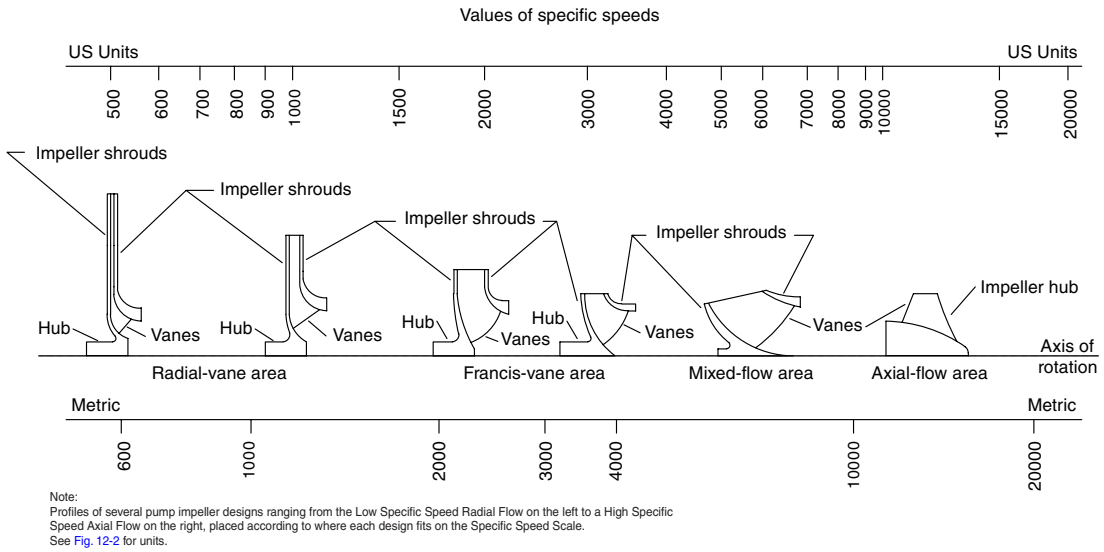
Example Centrifugal Pump Head Curves



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FIG. 12-11

Values of Specific Speeds (n_S)



Note:
Profiles of several pump impeller designs ranging from the Low Specific Speed Radial Flow on the left to a High Specific Speed Axial Flow on the right, placed according to where each design fits on the Specific Speed Scale.
See Fig. 12-2 for units.

On-Off Control — Pump capacity can be controlled by starting and stopping the pump manually or by an automatic control such as pressure, level or temperature switches.

Temperature Rise Due to Pumping

When a liquid is pumped, its temperature increases because the energy resulting from the inefficiency of the pump appears as heat.

$$t_r = \frac{H\left(\frac{1}{e} - 1\right)}{778\,C_p} \qquad \textbf{Eq 12-9}$$

Usually when the pump is running normally, the temperature rise is negligible. However, if the pump discharge is shut off, all energy is converted to heat and since there is no fluid flow through the pump to carry the heat away, the liquid in the pump will heat rapidly and eventually vaporize. This can produce catastrophic failures, particularly in large multistage pumps.

Pump vendors should be requested to provide data on minimum flow.

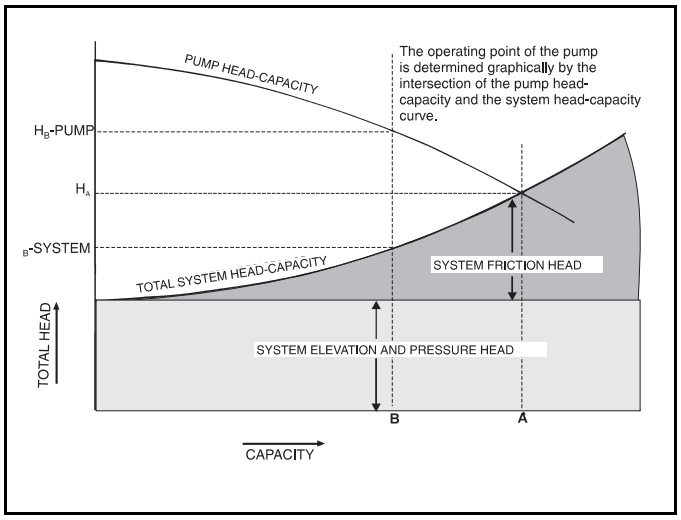
Expensive pumps, such as large multistage units, can be protected by installing minimum flow recirculation which will ensure an adequate flow through the pump.

Series and Parallel Operation

Often pumps are installed in series or in parallel with other pumps. In parallel, the capacities at any given head are added; in series, the heads at any given capacity are added. A multistage pump is in effect a series of single stage units. Figs. 12-15 and 12-16 show series and parallel pumps curves, a system curve, and the effect of operating one, two or three pumps in a system. In both figures, the operating points for both pumps "A" and "B" are the same only when one pump is operating. For 2 or 3 pumps operating, the points are not the same because of the pump curve shapes. Hence, due considertion should be given to the pump curve shape when selecting pumps for series or parallel operation.

FIG. 12-12

Example Combined Pump-System Curves



Parallel operation is most effective with identical pumps; however, they do not have to be identical, nor have the same shut-off head or capacity to be paralleled. When pumps are operating in parallel it is imperative that their performance curves rise steadily to shut-off. A drooping curve gives two possible points of operation, and the pump load may oscillate between the two causing surging.

Drivers

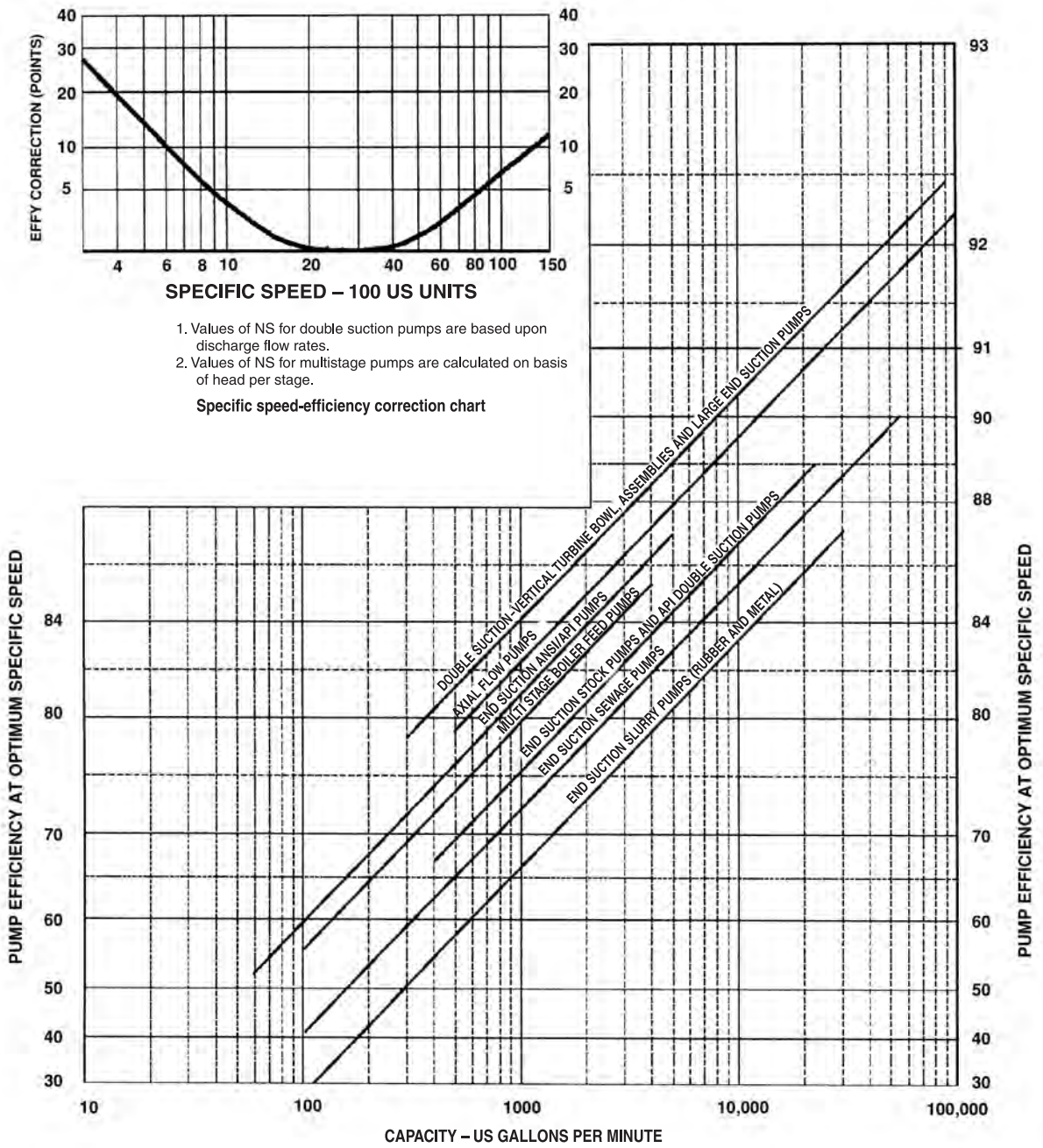
Most pumps used in gas processing service are driven by electric motors, usually fixed speed induction motors.

API Standard 610, Section 3.1.4. (Drivers), states:

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FIG. 12-13

Optimum Generally Attainable Efficiency Chart



NOTES:

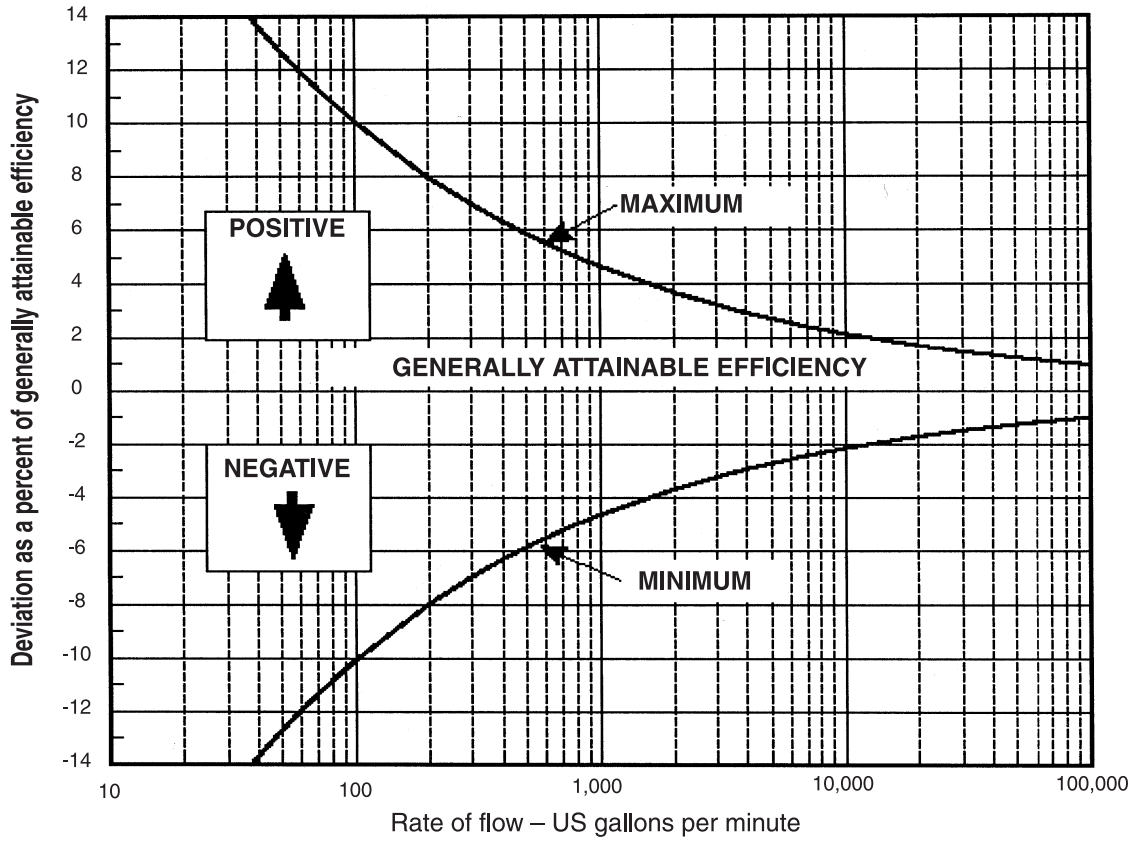
1. The above charts depict the generally attainable efficiency levels of centrifugal pumps at best efficiency point with maximum diameter impeller when pumping clear water at 85˚F.
2. The above charts relate to industrial class pumps designed, manufactured and tested in accordance with recognized industry standards.

Courtesy of the Hydraulic Institute, www.pumps.org

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FIG. 12-14

Deviation from Generally Attainable Efficiency



Courtesy of the Hydraulic Institute, www.pumps.org

"Motors shall have power ratings, including the service factor (if any), at least equal to the percentages of power at pump rated conditions given in. . ." the next table. "However, the power at rated conditions shall not exceed the motor nameplate rating. Where it appears that this procedure will lead to unnecessary oversizing of the motor, an alternate proposal shall be submitted for the purchaser's approval."

Motor Nameplate Rating		Percentage of Rated Pump Power
kW	hp	
<22	<30	125
22-55	30-75	115
>55	>75	110

Alternatives to electric motor drivers are:

- internal combustion engines
- gas turbines
- steam turbines
- hydraulic power-recovery turbines

Usually the speed of rotation of these drivers can be varied to provide control.

Variable Speed Drives — Fig. 12-18 lists various types of adjustable speed drives, their characteristics and their application.

Materials of Construction

Pumps manufactured with cast-steel cases and cast-iron internals are most common in the gas processing industry. API Std 610 is a good reference for material selection. The material selections in this document can be over-ridden as required to reflect experience.

Experience is the best guide to selection of materials for pumps. Process pump manufacturers can usually provide suggestions for materials, based on their experience and knowledge of pumps.

Shaft Seals

Mechanical seals are the most common sealing devices for centrifugal pumps in process service. The purpose of the seal is to retain the pumped liquid inside the pump at the point

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FIG. 12-15

Series Pumps Selection

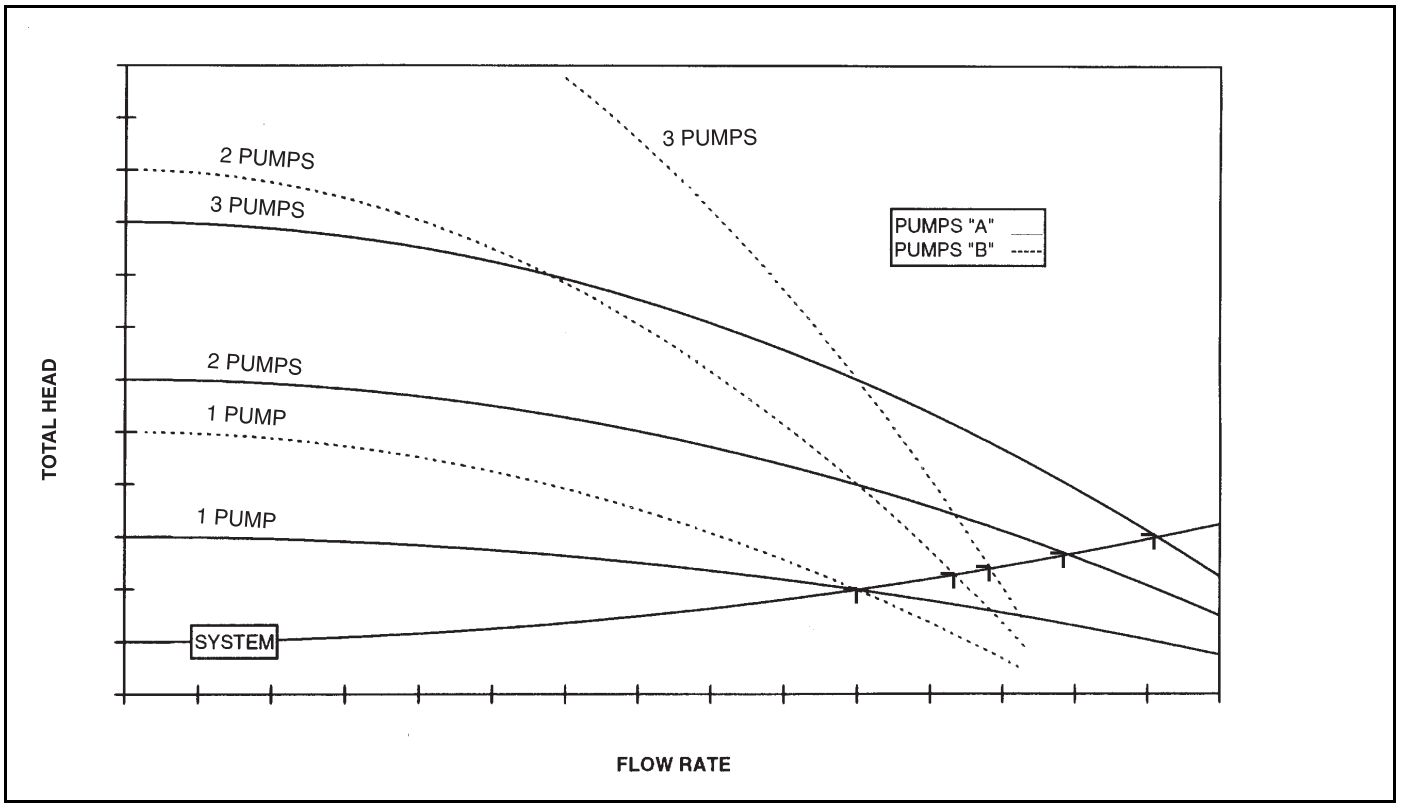
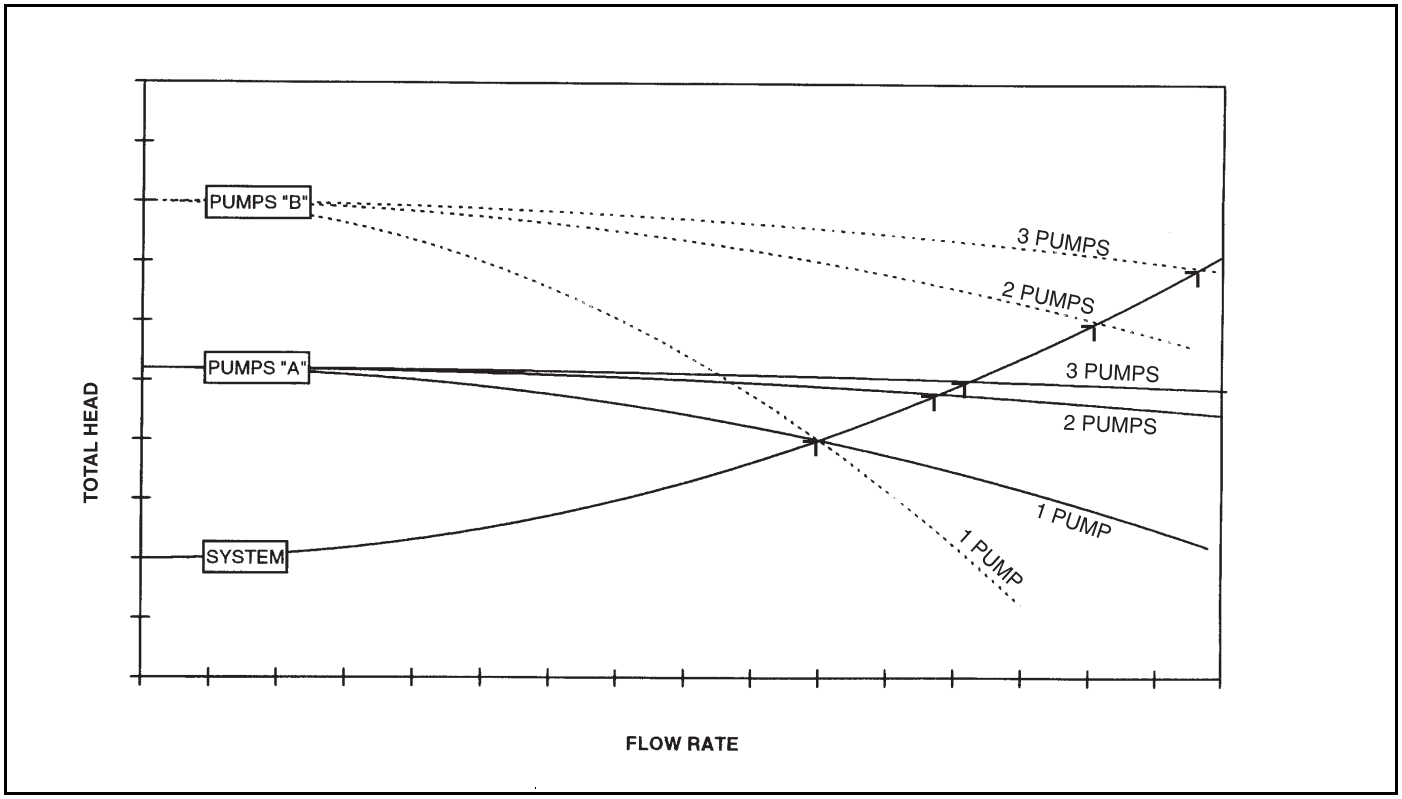


FIG. 12-16

Parallel Pumps Selection



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where the drive shaft penetrates the pump body. Mechanical seals consist of a stationary and a rotating face, and the actual sealing takes place across these very smooth, precision faces. Seal faces may require cooling and lubrication. API Std 610 describes seal flush systems used to cool the seal faces and remove foreign material. Seal manufacturers can provide application and design information.

Alignment, Supports, and Couplings

The alignment of the pump and driver should be checked and adjusted in accordance with the manufacturer's recommendations before the pump is started. If the operating temperature is greatly different from the temperature at which the alignment was performed, the alignment should be checked, and adjusted if necessary, at the operating temperature.

Pump and piping supports should be designed and installed so that forces exerted on the pump by the piping will not cause pump misalignment when operating temperature changes or other conditions occur.

The shaft coupling should be selected to match the power transmitted and the type of pump and driver. A spacer type coupling should be used if it is inconvenient to move either the pump or the driver when the seal (or other component) requires maintenance.

Piping

Pump requirements, nozzle size, type of fluid, temperature, pressure and economics determine materials and size of piping.

Suction lines should be designed to keep friction losses to a minimum. This is accomplished by using an adequate line size, long radius elbows, full bore valves, etc. Pockets where air or vapor can accumulate should be avoided. Suction lines should be sloped, where possible, toward the pump when it is below the source, and toward the source when it is below the pump. Vertical downward suction pipes require special care to avoid pulsation and vibrations that can be caused by air or vapor entrainment. Elbows entering double suction pumps should be installed in a position parallel to the impeller. Sufficient liquid height above the suction piping inlet, or a vortex breaker, should be provided to avoid vortex formation which may result in vapors entering the pump. Suction vessel tangential inlets and centrifugal pumps may induce a vortex in the vessel and pump suction line, opening a vapor core that feeds into the pump suction. Whatever the cause, vortexes can be eliminated with a straightening cross, also called a vortex breaker, installed at the vessel outlet nozzle.

For discharge piping, sizing is determined by the available head and economic considerations. Velocities range from 3 to 15 ft/sec. A check valve should be installed between the discharge nozzle and the block valve to prevent backflow.

Auxiliary piping (cooling, seal flushing and lubrication) is a small but extremely important item. API Standard 610, "Centrifugal Pumps for General Refinery Service," or applicable national standard should be followed. Provisions for piping of stuffing box leakage and other drainage away from the pump should be provided.

Pump Protection

The following protection may be considered:

- low suction pressure
- high discharge pressure
- low suction vessel (or tank) level
- high discharge vessel (or tank) level
- low flow
- flow reversal
- high temperature of bearings, case, etc.
- vibrations
- lack of lubrication
- overspeed

Protection may be considered for the pump driver and may be combined with pump protections.

Installation, Operation, Maintenance

Installation, operation, and maintenance manuals should be provided by the pump manufacturer and are usually application specific. See Fig. 12-17 for a checklist of pump troubles and causes.

Driver rotation and alignment should be checked before the pump is operated.

A typical starting sequence for a centrifugal pump is:

- Ensure that all valves in auxiliary sealing, cooling, and flushing system piping are open, and that these systems are functioning properly.
- Close discharge valve.
- Open suction valve.
- Vent gas from the pump and associated piping.
- Energize the driver.
- Open discharge valve slowly so that the flow increases gradually.
- Note that, on larger multistage pumps, it is very important that flow through the pump is established in a matter of seconds. This is frequently accomplished by the previously mentioned minimum flow recirculation.

RECIPROCATING PUMPS

The most common reciprocating pump in gas plants is the single-acting plunger pump which is generally employed in services with moderate capacity and high differential pressure. These pumps fill on the backstroke and exhaust on the forward stroke. They are available with single (simplex) or multi-plungers (duplex, triplex, etc.), operating either horizontally or vertically. Examples of plunger pump service in gas plants are: high pressure chemical or water injection, glycol circulation, and low capacity, high pressure amine circulation, and pipeline product pumps.

Double-acting piston pumps which fill and exhaust on the same stroke have the advantage of operating at low speeds and can pump high viscosity liquids which are difficult to handle with normal centrifugal or higher speed plunger pumps.

Pump Calculations

Power requirement bhp: see equation in Fig. 12-2.

Displacement for single-acting pump

$$D = \frac{A \bullet m \bullet L_s \bullet n}{231} \qquad \textbf{Eq 12-10}$$

Displacement for double-acting pump

$$D = \frac{(2A - a)\, m \bullet L_s \bullet n}{231} \qquad \textbf{Eq 12-11}$$

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FIG. 12-17

Check List for Centrifugal Pump Troubles and Causes

Trouble:	Possible Causes:
1. Failure to deliver liquid	a. Wrong direction of rotation b. Pump not primed c. Suction line not filled with liquid d. Air or vapor pocket in suction line e. Inlet to suction pipe not sufficiently submerged f. Available NPSH not sufficient g. Pump not up to rated speed h. Total head required greater than head which pump is capable of delivering
2. Pump does not deliver rated capacity	a. Wrong direction of rotation b. Suction line not filled with liquid c. Air or vapor pocket in suction line d. Air leaks in suction line or stuffing boxes e. Inlet to suction pipe not sufficiently submerged. f. Available NPSH not sufficient g. Pump not up to rated speed h. Total head greater than head for which pump designed j. Foot valve too small k. Foot valve clogged with trash m. Viscosity of liquid greater than that for which pump designed n. Mechanical defects: (1) Wearing rings worn (2) Impeller damaged (3) Internal leakage resulting from defective gaskets o. Discharge valve not fully opened
3. Pump does not develop rated discharge pressure	a. Gas or vapor in liquid b. Pump not up to rated speed c. Discharge pressure greater than pressure for which pump designed d. Viscosity of liquid greater than that for which pump designed e. Wrong rotation f. Mechanical defects: (1) Wearing rings worn (2) Impeller damaged (3) Internal leakage resulting from defective gaskets
4. Pump loses liquid after starting	a. Suction line not filled with liquid b. Air leaks in suction line or stuffing boxes c. Gas or vapor in liquid d. Air or vapor pockets in suction line e. Inlet to suction pipe not sufficiently submerged f. Available NPSH not sufficient g. Liquid seal piping to lantern ring plugged h. Lantern ring not properly located in stuffing box
5. Pump overloads driver	a. Speed too high b. Total head lower than rated head c. Excessive recirculation d. Either or both the specific gravity and viscosity of liquid different from that for which pump is rated e. Mechanical defects: (1) Misalignment (2) Shaft bent (3) Rotating element dragging (4) Packing too tight
6. Vibration	a. Starved suction (1) Gas or vapor in liquid (2) Available NPSH not sufficient (3) Inlet to suction line not sufficiently submerged (4) Gas or vapor pockets in suction line b. Misalignment c. Worn or loose bearings d. Rotor out of balance (1) Impeller plugged (2) Impeller damaged e. Shaft bent f. Improper location of control valve in discharge line g. Foundation not rigid
7. Stuffing boxes overheat	a. Packing too tight b. Packing not lubricated c. Wrong grade of packing d. Insufficient cooling water to jackets e. Box improperly packed.
8. Bearings overheat	a. Oil level too low b. Improper or poor grade of oil c. Dirt in bearings d. Dirt in oil e. Moisture in oil f. Oil cooler clogged or scaled g. Failure of oiling system h. Insufficient cooling water circulation i. Insufficient cooling air k. Bearings too tight m. Oil seals too close fit on shaft n. Misalignment
9. Bearings wear rapidly	a. Misalignment b. Shaft bent c. Vibration d. Excessive thrust resulting from mechanical failure inside the pump e. Lack of lubrication f. Bearings improperly installed g. Dirt in bearings h. Moisture in oil j. Excessive cooling of bearings

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FIG. 12-18

Adjustable Speed Drives[3] and Power Transmissions

Type	Characteristics	Applications
Electric Drivers		
Solid State AC drives	• high efficiency • good speed regulation • low maintenance • complex controls • high cost • can be explosion proof • can retrofit	• 50 to 2500+ bhp • larger pumps where good speed regulation over not too wide a range is required • hazardous areas
Solid State DC drives	• similar to AC except speed regulation good over a wider range	• 50 to 500+ bhp • non-hazardous areas
Electromechanical		
Eddy Current Clutch	• efficient, proportional to slip • poor speed regulation • require cooling	• 5 to 500+ bhp • smaller centrifugal pumps where speed is usually near design • non-hazardous areas
Wound-Rotor Motor	• poor speed regulation • reasonable efficiency	• 50 to 500+ bhp • larger pumps non-hazardous areas
Mechanical		
Rubber Belt	• wide range of speed regulation possible	• fractional to 100 bhp • small centrifugal and positive displacement pumps
Metal Chain	• low to medium efficiency	• chemical feed pumps • non-hazardous areas
Hydraulic Power Recovery Turbines	• medium efficiency • continuously variable speed • reversible use as pump	• available hydraulic head

Notes:

1. Actual capacity (Q) delivered by pump is calculated by multiplying displacement by the volumetric efficiency.
2. The combination of mechanical and volumetric efficiency for reciprocating pumps is normally 90% or higher for noncompressible fluids.
3. In double-acting pumps with guided piston (rod in both sides), change "a" to "2a" in Eq 12-11.

Example 12-2 — Calculate the power required for a simplex plunger pump delivering 10 gpm of liquid of any specific gravity at 3000 psi differential pressure and mechanical efficiency of 90%.

$$\text{bhp} = \frac{(10)\,(3000)}{(1714)\,(0.90)} = 19.4 \text{ hp}$$

Volumetric Efficiency, Compressible Fluids — Unlike water, lighter hydrocarbon liquids (e.g. ethane, propane, butane) are sufficiently compressible to affect the performance of reciprocating pumps.

The theoretical flow capacity is never achieved in practice because of leakage through piston packing, stuffing boxes, or valves and because of changes in fluid density when pumping compressible fluids such as light hydrocarbons.

The ratio of real flow rate to theoretical flow rate (pump displacement) is the volumetric efficiency. The volumetric efficiency depends on the size, seals, valves and internal configuration of each pump, the fluid characteristics and operating conditions.

When pumping compressible liquids, the volumetric efficiency should be stated with reference to the flow rate measured in a specific side of the pump (suction or discharge side).

The relationship of overall suction and discharge volumetric efficiency, displacement, and suction and discharge flow rate of a reciprocating pump is defined in Eq 12-12. When the leakage is not considered, the overall efficiencies may be substituted by the density change efficiencies.

$$D = \frac{Q_s}{VE_{sov}} = \frac{Q_d}{VE_{dov}} \qquad \textbf{Eq 12-12}$$

The following equations are based on the discharge flow rate. Similar equations may be written for the suction side, and conversions may be made by multiplying them by the discharge to suction densities ratio.

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The overall discharge volumetric efficiency is a combination of volumetric efficiency due to leakage and discharge volumetric efficiency due to fluid density change.

$$VE_{dov} = VE_l \bullet VE_{d\rho} \qquad \textbf{Eq 12-13}$$

The volumetric efficiency due to leakage is related to slip as follows:

$$VE_l = 1 - s \qquad \textbf{Eq 12-14}$$

The effect of the difference in the leakage flow rate measured at suction pressure vs discharge pressure is neglected here, assuming that all leakages are internal.

The discharge volumetric efficiency due to density change is:

$$VE_{d\rho} = 1 - r\left[1 - \frac{\rho_i}{\rho_o}\right] \qquad \textbf{Eq 12-15}$$

When the change in fluid density is linear with the change in pressure and is smaller than 10%, and the temperature change is negligible, Equation 12-16 may be used to calculate hydraluic power. H_c comes from Eq 12-5. Additionally, approximately 2 to 5% of power may be required for the work done during the piston cycle, in compressing and in decompressing the fluid that is held in the pump chamber without flowing through the pump.

$$\text{hyd hp} = \frac{Q_d \bullet \text{sp gr}_o \bullet H_c}{3960} \qquad \textbf{Eq 12-16}$$

When the differential pressure is sufficiently high to cause a density change of more than 10%, or when the pressure is near the fluid's critical pressure, or when temperature change is not negligible, this equation may not be accurate. In such cases the pump manufacturer should be consulted. See Equipment and System Equations, last paragraph.

Data on density change with pressure and temperature can be found in Section 23, "Physical Properties."

Example 12-3 — For a 3" diameter and a 5 inch stroke triplex plunger pump pumping propane with a suction density 31.4 lb/cu ft and a discharge density 32.65 lb/cu ft and given that r = 4.6 and s = 0.03, find the overall discharge volumetric efficiency.

Discharge volumetric efficiency due to density change:

$$VE_{d\rho} = 1 - 4.6\left(1 - \frac{31.4}{32.65}\right) = 0.824$$

Volumetric efficiency due to leakage

$$VE_l = 1 - 0.03 = 0.97$$

Overall discharge volumetric efficiency:

$$VE_{dov} = (0.824) \bullet (0.97) = 0.799$$

Suction System Considerations

The suction piping is a critical part of any reciprocating pump installation. The suction line should be as short as possible and sized to provide not more than three feet per second fluid velocity, with a minimum of bends and fittings. A centrifugal booster pump is often used ahead of a reciprocating pump to provide adequate NPSH which would also allow higher suction line velocities.

NPSH required for a reciprocating pump is calculated in the same manner as for a centrifugal pump, except that additional allowance must be made for the requirements of the reciprocating action of the pump. The additional requirement is termed acceleration head. This is the head required to accelerate the fluid column on each suction stroke so that this column will, at a minimum, catch up with the receding face of the piston during its filling stroke.

Acceleration Head — Acceleration head is the fluctuation of the suction head above and below the average due to the inertia effect of the fluid mass in the suction line. With the higher speed of present-day pumps or with relatively long suction lines, this pressure fluctuation or acceleration head must be taken into account if the pump is to fill properly without forming vapor which will cause pounding or vibration of the suction line.

With the slider-crank drive of a reciprocating pump, maximum plunger acceleration occurs at the start and end of each stroke. The head required to accelerate the fluid column (h_a) is a function of the length of the suction line and average velocity in this line, the number of strokes per minute (rpm), the type of pump and the relative elasticity of the fluid and the pipe, and may be calculated as follows:

$$h_a = \frac{L \bullet v \bullet n \bullet C}{k \bullet g} \qquad \textbf{Eq 12-17}$$

where C and k are given in Fig. 12-19.

Example 12-4 — Calculate the acceleration head, given a 2" diameter x 5" stroke triplex pump running at 360 rpm and displacing 73 gpm of water with a suction pipe made up of 4′ of 4" and 20′ of 6" standard wall pipe.

Average Velocity in 4" Pipe = 1.84 fps

Average Velocity in 6" Pipe = 0.81 fps

Acceleration Head in 4" Pipe

$$h_{a4} = \frac{(4)\,(1.84)\,(360)\,(0.066)}{(1.5)\,(32.2)} = 3.62 \text{ ft}$$

Acceleration Head in 6" Pipe

$$h_{a6} = \frac{(20)\,(0.81)\,(360)\,(0.066)}{(1.5)\,(32.2)} = 7.97 \text{ ft}$$

Total Acceleration Head

$$h_a = 3.62 + 7.97 = 11.6 \text{ ft}$$

Karassik et al[9] recommend that the NPSHA exceed the NPSHR by 3 to 5 psi for reciprocating pumps.

Pulsation — A pulsation dampener (suction stabilizer) is a device installed in the suction piping as close as possible to the pump to reduce pressure fluctuations at the pump. It consists of a small pressure vessel containing a cushion of gas (sometimes separated from the pumped fluid by a diaphragm). Pulsation dampeners should be considered for the suction side of any reciprocating pump, but they may not be required if the suction piping is oversized and short, or if the pump operates at less than 150 rpm. A properly installed and maintained pulsation dampener should absorb the cyclical flow variations so that the pressure fluctuations are about the same as those that occur when the suction piping is less than 15′ long.

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Similar pressure fluctuations occur on the discharge side of every reciprocating pump. Pulsation dampeners are also effective in absorbing flow variations on the discharge side of the pump and should be considered if piping vibration caused by pressure fluctuations appears to be a problem. Pulsation dampener manufacturers have computer programs to analyze this phenomenon and should be consulted for reciprocating pump applications over 50 hp. Discharge pulsation dampeners minimize pressure peaks and contribute to longer pump and pump valve life. The need for pulsation dampeners is increased if multiple pump installations are involved.

Ensure that bladder type pulsation dampeners contain the correct amount of gas. The following equation may be used for sizing estimation of bladder and diaphragm-type pulsation dampeners, where the volume, length and area must be in self-consistent units.

$$V_{pd} = \frac{A \cdot L_s \cdot K \cdot (100/(100-\Delta))^{1/N}}{1-(100/(100+\Delta))^{1/N}} = \frac{A \cdot L_s \cdot K \cdot (P_{ave}/P_{min})^{1/N}}{1-(P_{ave}/P_{max})^{1/N}} \qquad \textbf{Eq 12-18}$$

Where K has a value of:		
	Single Acting	Double Acting
Simplex	0.60	0.25
Duplex	0.25	0.15
Triplex	0.13	0.06
Quadruplex	0.10	0.06
Quintuplex	0.06	0.02

Capacity Control — Manual or automatic capacity control for one pump or several parallel pumps can be achieved by one or a combination of the following methods:

- on-off control
- recirculation
- variable speed driver or transmission
- variable displacement pump

Drivers — Two types of mechanisms are commonly used for driving reciprocating pumps; one in which the power of a motor or engine is transmitted to a shaft and there is a mechanism to convert its rotative movement to alternating linear movement to drive the pumping piston or plunger. In the other type, there is a power fluid, such as steam, compressed air, or gas acting on a piston, diaphragm or bellow linked to the pumping piston or plunger.

Piping — Suction and discharge piping considerations are similar to those for centrifugal pumps. In addition, acceleration head must be included for pipe sizing. For piping materials and thickness selection, pressure pulsations amplitude and fatigue life should be considered.

ROTARY PUMPS

The rotary pump is a positive displacement type that depends on the close clearance between both rotating and stationary surfaces to seal the discharge from the suction. The most common types of rotary pumps use gear or screw rotating elements. These types of positive displacement pumps are commonly used for viscous liquids for which centrifugal or reciprocating pumps are not suitable. Low viscosity liquids with poor lubricating properties (such as water) are not a proper application for gear or screw pumps.

REGENERATIVE PUMPS

Regenerative pumps are also called peripheral pumps. The unit has a rotary wheel or impeller with vanes on both sides of its periphery, which rotates in an annular shaped chamber in the pump casing. The fluid moves outwards through the vanes, at the vanes tips the fluid passes to the chamber and is recirculated back to a lower point on the impeller vanes; thus there are two fluid helical paths around the impeller and chamber, recirculating the fluid from vane to vane, from the suction to the discharge ports, on both sides of the impeller. The recirculation increases the head developed in each stage, so the head is a function of the number of recirculation cycles. Capacity, head and power, and speed follow fan laws.

Typically, the performance curve is a downward slope straight line; therefore, a throttling valve in a regenerative pump will permit more precise changes in flow than in centrifugal pumps. The maximum shut-off head developed may be up to 6 times the shut-off head of a single stage centrifugal pump running at the same speed.

Because of close clearances, regenerative pumps can not be used to pump liquids containing solid particles. They can pump liquids containing vapors and gases, if they contain sufficient liquid to seal the close clearances.

FIG. 12-19

Reciprocating Pump Acceleration Head Factors

C = 0.200 for simplex double-acting
= 0.200 for duplex single-acting
= 0.115 for duplex double-acting
= 0.066 for triplex single or double-acting
= 0.040 for quintuplex single or double-acting
= 0.028 for septuplex single or double-acting
= 0.022 for nonuplex single or double-acting

k = a factor related to the fluid compressibility

hot oil	2.5
most hydrocarbons	2.0
amine, glycol, water	1.5
deaerated water	1.4
liquid with small amounts of entrained gas	1.0

Note: "C" will vary from the listed values for unusual ratios of connecting rod length to crank radius over 6.

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DIAPHRAGM PUMPS

Diaphragm pumps are reciprocating, positive displacement type pumps, utilizing a valving system similar to a plunger pump. These pumps can deliver a small, precisely controlled amount of liquid at a moderate to very high discharge pressure. Diaphragm pumps are commonly used as chemical injection pumps because of their controllable metering capability, the wide range of materials in which they can be fabricated, and their inherent leakproof design.

MULTIPHASE PUMPS

Multiphase pumps can pump immiscible liquids such as oil and water with gas. There are screw types and rotodynamic types. A progressive cavity design is used along the flow path to accommodate gas volume reduction caused by increased pressure. A full range of gas/liquid ratios can be handled. This class of pumps is of interest in applications where conventional pumps and separate compressors with or without separate pipelines are not economically feasible.

LOW TEMPERATURE PUMPS

Two types of centrifugal pumps have been developed for cryogenic applications: the external motor type and the submerged motor type.

External motor type — These pumps are of conventional configuration with a coupled driver and can be single or multistage. The pump assembly is usually mounted in a vessel from which it pumps.

Submerged motor type — This type of pump is characterized by being directly coupled to its motor, with the complete unit being submerged in the fluid.

Hydraulic Turbines

Many industrial processes involve liquid streams which flow from higher to lower pressures. Usually the flow is controlled with a throttling valve, hence the hydraulic energy is wasted. Up to 80% of this energy can be recovered by passing the liquid through a hydraulic power recovery turbine (HPRT). To justify the installation of an HPRT, an economic analysis of the power savings versus added equipment and installation costs should be performed.

TYPES OF HPRTs

Two major types of centrifugal hydraulic power recovery turbines are used.

1. Reaction—Single or multistage Francis-type rotor with fixed or variable guide vanes.
2. Impulse—Pelton Wheel, usually specified for relatively high differential pressures.

HPRTs with Francis-type rotors are similar to centrifugal pumps. In fact, a good centrifugal pump can be expected to operate with high efficiency as an HPRT when the direction of flow is reversed.

The Pelton Wheel or impulse runner type HPRT is used in high head applications. The impulse type turbine has a nozzle which directs the high pressure fluid against bowl-shaped buckets on the impulse wheel. This type of turbines' performance is dependent upon back pressure, while the reaction type is less dependent upon back pressure.

Power Recovered by HPRTs

The theoretical energy which can be extracted from a high pressure liquid stream by dropping it to a lower pressure through an HPRT can be calculated using the hydraulic horsepower. See Fig. 12-2 for bhp equation. Since some of the energy will be lost because of friction, the hydraulic horsepower must be multiplied by the efficiency of the HPRT.

In applications where the fluid that enters the HPRT has large dissolved gas content, the available power is larger than the power that may be calculated using the liquid equations, so, the power shall be calculated using an adequate two-phase calculation method.

The amount of power recovered by an HPRT is directly proportional to the efficiency rather than inversely proportional as is the case when calculating the power required by a pump. Thus, if a fluid is pumped to a high pressure and then reduced to its original pressure using an HPRT, the proportion of the pumping energy which can be supplied by the HPRT is equal to the efficiencies of the pump and turbine multiplied together. Typically, good centrifugal pumps and good HPRTs have efficiencies of between 70% and 80%. Thus, the HPRT can be expected to provide between 50% and 60% of the energy required for pumping.

Usually the high-pressure liquid contains a substantial amount of dissolved gas. The gas comes out of solution as the liquid pressure drops. This does not cause damage to the HPRT, presumably because the fluid velocity through the HPRT is high enough to maintain a froth-flow regime. The term NPSHR does not apply to HPRTs.

Applications

HPRTs may be used to drive any kind of rotating equipment (e.g. pumps, compressors, fans, electrical generators). The main problems are matching the power required by the driven load to that available from the HPRT and speed control. Both the power producer and the speed can be controlled by:

- throttling the liquid flow, either downstream or upstream from the HPRT
- allowing a portion of the liquid to bypass the HPRT
- adjusting inlet guide vanes installed in the HPRT

Sometimes HPRTs are installed with a "helper" driver. If this is an electric motor, the speed will be controlled by the motor speed.

Typical gas-processing streams for which HPRTs should be considered are:

- Rich sweetening solvents (e.g. amines, etc.)
- Rich absorption oil
- High-pressure crude oil.
- Condensed high pressure natural gas liquids.
- Liquid refrigerant letdown, in mechanical refrigeration cycles.

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- High pressure LNG letdown, in natural gas liquefaction.

The lower limit of the power recovery which can be economically justified with single-stage HPRTs is about 30 hp and with multistage, about 100 hp. HPRTs usually pay out their capital cost in from one to three years.

Frequently, when an HPRT is to be used to drive a pump, both devices are purchased from one manufacturer. This has the advantage of ensuring that the responsibility for the entire installation is assumed by a single supplier.

The available pressure differential across the HPRT is calculated using a technique similar to that used to calculate the differential head of centrifugal pumps.

Example 12-5 — Specify an HPRT driven pump for a gas sweetening process.

Given:

lean DEA flow	1000 gpm
lean DEA temperature	110°F
lean DEA specific gravity	1.00
lean DEA vapor pressure at 120°F	1.7 psia
rich DEA flow	1000 gpm
rich DEA temperature	160°F
rich DEA specific gravity	1.01
pump suction total pressure	75 psig
pump discharge total pressure	985 psig
HPRT inlet total pressure	960 psig
HPRT outlet total pressure	85 psig

Solution:

For this example, the suction and discharge pressures have already been calculated using a technique similar to that suggested for centrifugal pumps.

$$\text{NPSHA for pump} = \frac{2.31\,(75 + 14.7 - 1.7)}{1.00} = 203\text{ ft}$$

$$\text{Required head for pump} = \frac{2.31\,(985 - 75)}{1.00} = 2102\text{ ft}$$

The pump selected is a 5-stage unit. From the pump curve (Fig. 12-21), the expected efficiency of the pump is 78.5%. Hence, the required power will be:

$$\text{bhp for pump} = \frac{(1000)\,(2102)\,(1.00)}{(3960)\,(0.785)} = 676\text{ hp}$$

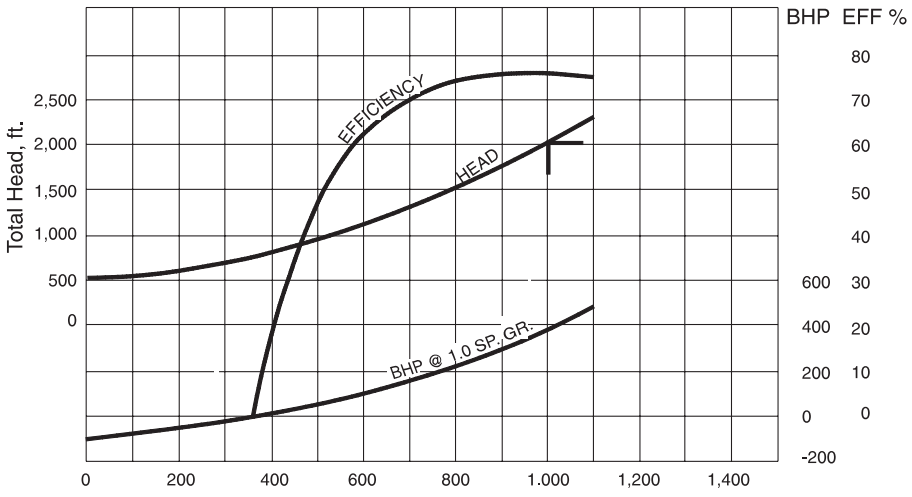
$$\text{Available head for HPRT} = \frac{2.31\,(960 - 85)}{1.01} = 2001\text{ ft}$$

The HPRT selected is a 3-stage unit. From the performance curve (Fig. 12-20), the expected efficiency of the HPRT is 76%. Hence, the available power will be:

$$\text{bhp from HPRT} = \frac{(1000)\,(2001)\,(1.01)\,(0.76)}{3960} = 388\text{ hp}$$

FIG. 12-20

Rich DEA Pressure Letdown

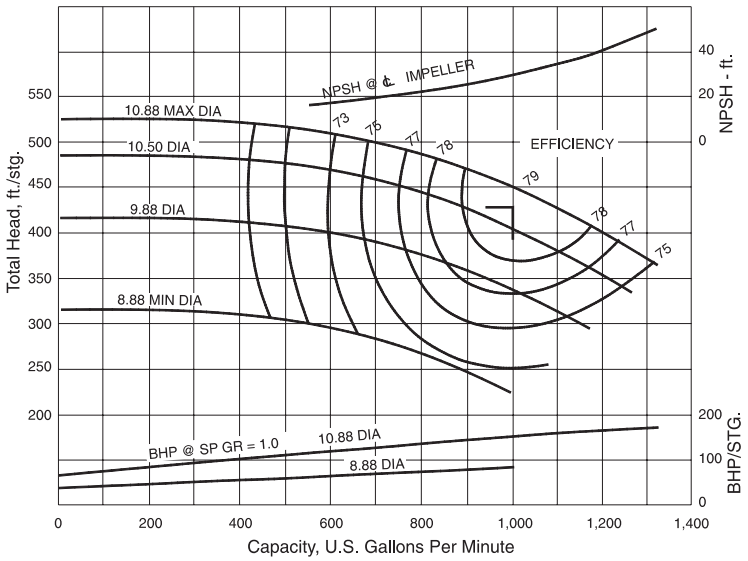


RICH DEA PRESSURE LETDOWN			
IMPELLER MAX DIA 10.06 IN	4 X 6 X 10 HORIZONTAL MULTISTAGE TURBINE		
MIN DIA	DIA IMPELLER 10.06 IN	IMPELLER PATT	3,560 RPM
EYE AREA SQ IN	NPSH REQUIRED	REFERENCE	DRAWING NO. HT-3X

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FIG. 12-21

Lean Amine Charge Pump



LEAN AMINE CHARGE PUMP					
IMPELLER MAX DIA	10.88	IN	4 X 6 X 10.5 HORIZONTAL MULTISTAGE PUMP		
MIN DIA	8.88	IN	DIA IMPELLER 10.63	IN	IMPELLER PATT 3,560 RPM
EYE AREA	13.0	SQ IN	NPSH REQUIRED	REFERENCE	CURVE NO. MD-38

Another driver, such as an electric motor, would be required for the pump to make up the difference in bhp between the pump and HPRT. The other driver would have to be capable of providing at least 288 hp. It is good practice to provide an electric motor driver large enough to drive the pump by itself to facilitate startups. The pump, HPRT, and electric motor driver (helper or full size) would usually be direct connected. In some cases, a clutch is used between the pump and HPRT, so the unit is independent of the HPRT.

The pump and HPRT are similar in hydraulic design except that the pump has five stages and the HPRT, three stages. In this case, the HPRT is a centrifugal pump running backwards.

CODES & ORGANIZATIONS

API Std 610 8th Edition—Centrifugal Pumps for General Refinery Service

ANSI B73.1—Horizontal End-Suction Centrifugal Pumps

ANSI B73.2—Vertical Inline Centrifugal Pumps

Hydraulic Institute—Centrifugal, Reciprocating & Rotary Pumps

API Std 674—Positive Displacement Pumps – Reciprocating

API Std 675—Positive Displacement pumps – Controlled Volume

API Std 676—Positive Displacement Pumps – Rotary

API Std 682—Shaft Sealing Systems for Centrifugal and Rotary Pumps.

ANSI/AWWA E101-88—Vertical Turbine Pumps – Line Shaft and Submersible Types

NEMA, EMMAC, UL, CSA—Electric Motor Drivers

UL, ULC, NFPA, FM—Fire Water Pumps

AIChE—American Institute of Chemical Engineers

API—American Petroleum Institute

ANSI—American National Standards Institute

AWWA—American Water Works Association

CSA—Canadian Standards Association

EMMAC—Electrical Manufacturers Association of Canada

FM—Factory Mutual

NEMA—National Electrical Manufacturers Association

NFPA—National Fire Prevention Association

UL—Underwriters Laboratory

ULC—Underwriters Laboratory of Canada

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NOTES:

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SECTION 13

Compressors and Expanders

Compressors

Depending on application, compressors are manufactured as positive-displacement, dynamic, or thermal type (Fig. 13-2).

Positive displacement types fall in two basic categories: reciprocating and rotary.

The reciprocating compressor consists of one or more cylinders each with a piston or plunger that moves back and forth, displacing a positive volume with each stroke.

The diaphragm compressor uses a hydraulically pulsed flexible diaphragm to displace the gas.

FIG. 13-1

Nomenclature

ACFM = actual cubic feet per minute (i.e. at process conditions)
A_p = cross sectional area of piston,sq in
A_r = cross sectional area of piston rod, sq in
Bhp = brake or shaft horsepower
C = cylinder clearance as a percent of cylinder volume
C_p = specific heat at constant pressure, Btu (lb • °F)
C_v = specific heat at constant volume, Btu (lb • °F)
D = cylinder inside diameter, in
d = piston rod diameter, in
EP = extracted horsepower of expander
F = an allowance for interstage pressure drop, Eq 13-4
Ghp = gas horsepower, actual compression horsepower, excluding mechanical losses, bhp
H = head, ft • lb/lb
h = enthalpy, Btu/lb
ICFM = inlet cubic feet per minute, usually at suction conditions
k = isentropic exponent, C_p/C_v
MC_p = molar specific heat at constant pressure, Btu (lb mole • °F)
MC_v = molar specific heat at constant volume, Btu (lb mole • °F)
MW = molecular weight, lb/lb mole
N = speed, rpm
N_m = molar flow, moles/min
n = polytropic exponent or number of moles
P = pressure, psia
P_c = critical pressure, psia
PD = piston displacement, ft^3/min
P_L = pressure base used in the contract or regulation, psia
pP_c = pseudo critical pressure, psia
P_R = reduced pressure, P/P_c
pT_c = pseudo critical temperature, °R
P_V = partial pressure of contained moisture, psia
p = pressure, lb/ft^2
Q = inlet capacity (ICFM)
Q_g = standard gas flow rate, MMSCFD
R = universal gas constant = $10.73 \frac{\text{psia} \cdot \text{ft}^3}{\text{lb mole} \cdot °\text{R}}$
= $1545 \frac{\text{lb/ft}^3}{\text{l mole} \cdot °\text{R}}$ or $\frac{\text{ft} \cdot \text{lb}}{\text{lb mole} \, °\text{R}}$
= $1.986 \frac{\text{Btu}}{\text{lb mole} \cdot °\text{R}}$
r = compression ratio, P_2/P_1
s = entropy, Btu/(lb • °R)
SCFM = cubic feet per minute measured at 14.7 psia and 60°F
stroke = length of piston movement, in
T = absolute temperature, °R
T_c = critical temperature, °R
T_R = reduced temperature, T/T_c
t = temperature, °F
V = specific volume, ft^3/lb
VE = volumetric efficiency, percent
W = work, ft • lb
w = weight flow, lb/min
X = temperature rise factor
y = mole fraction
Z = compressibility factor
Z_{avg} = average compressibility factor = $(Z_s + Z_d)/2$
η = efficiency, expressed as a decimal

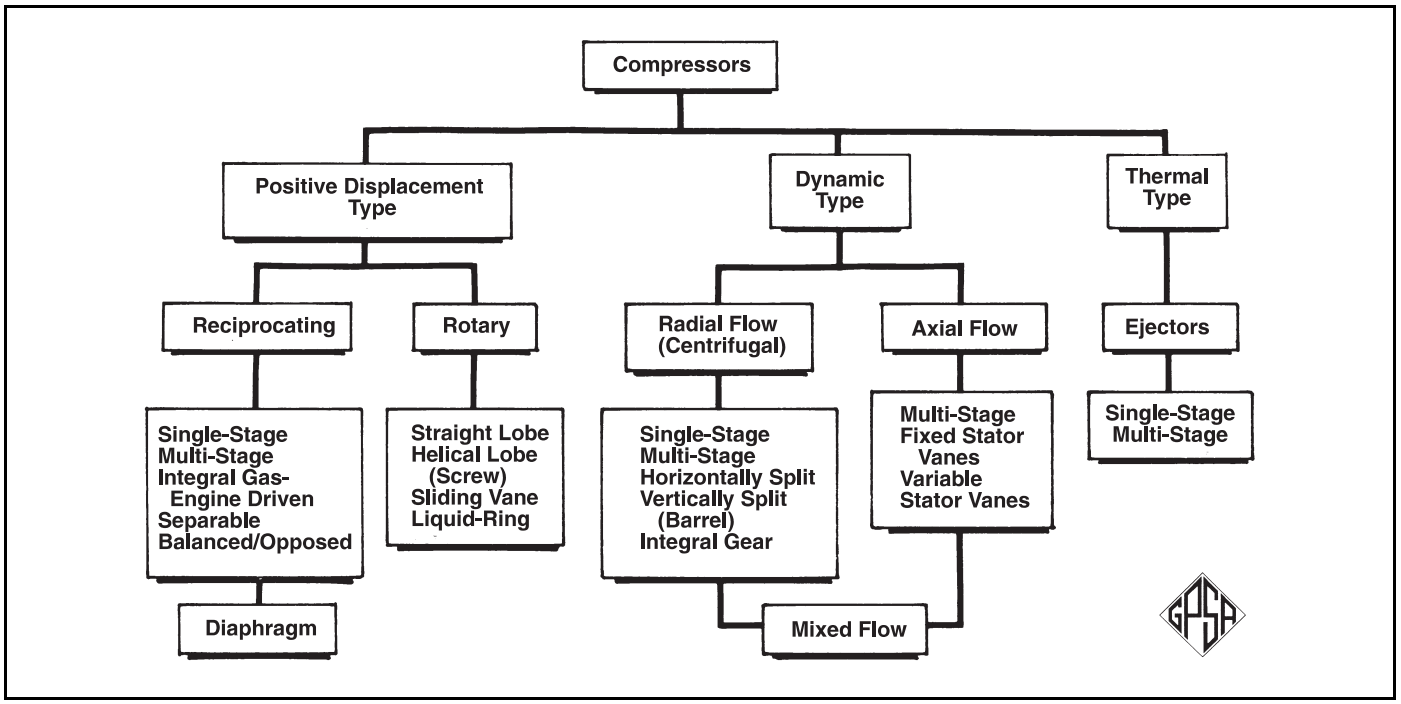
Subscripts

avg = average
d = discharge
g = gas
is = isentropic process
L = standard conditions used for calculation or contract
p = polytropic process
S = standard conditions, usually 14.7 psia,60°F
s = suction
t = total or overall
1 = inlet conditions
2 = outlet conditions

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FIG. 13-2

Types of Compressors



Rotary compressors cover lobe-type, screw-type, vane-type, and liquid ring type, each having a casing with one or more rotating elements that either mesh with each other such as lobes or screws, or that displace a fixed volume with each rotation.

The dynamic types include radial-flow (centrifugal), axial-flow, and mixed flow machines. They are rotary continuous-flow compressors in which the rotating element (impeller or bladed rotor) accelerates the gas as it passes through the element, converting the velocity head into static pressure, partially in the rotating element and partially in stationary diffusers or blades.

Ejectors are “thermal” compressors that use a high velocity gas or steam jet to entrain the inflowing gas, then convert the velocity of the mixture to pressure in a diffuser.

Fig. 13-3 covers the normal range of operation for compressors of the commercially available types.

The advantages of a centrifugal compressor over a reciprocating machine are:

1. Lower installed first cost where pressure and volume conditions are favorable,
2. Lower maintenance expense,
3. Greater continuity of service and dependability,
4. Less operating attention,
5. Greater volume capacity per unit of plot area,
6. Adaptability to high-speed low-maintenance-cost drivers.

The advantages of a reciprocating compressor over a centrifugal machine are:

1. Greater flexibility in capacity and pressure range,
2. Higher compressor efficiency and lower power cost,
3. Capability of delivering higher pressures,
4. Capability of handling smaller volumes,
5. Less sensitive to changes in gas composition and density.

RECIPROCATING COMPRESSORS

Reciprocating compressor ratings vary from fractional to more than 40,000 hp per unit. Pressures range from low vacuum at suction to 30,000 psi and higher at discharge for special process compressors.

Reciprocating compressors are furnished either single-stage or multi-stage. The number of stages is determined by the overall compression ratio. The compression ratio per stage (and valve life) is generally limited by the discharge temperature and usually does not exceed 4, although small-sized units (intermittent duty) are furnished with a compression ratio as high as 8.

Gas cylinders are generally lubricated, although a non-lubricated design is available when warranted; example: nitrogen, oxygen, and instrument air.

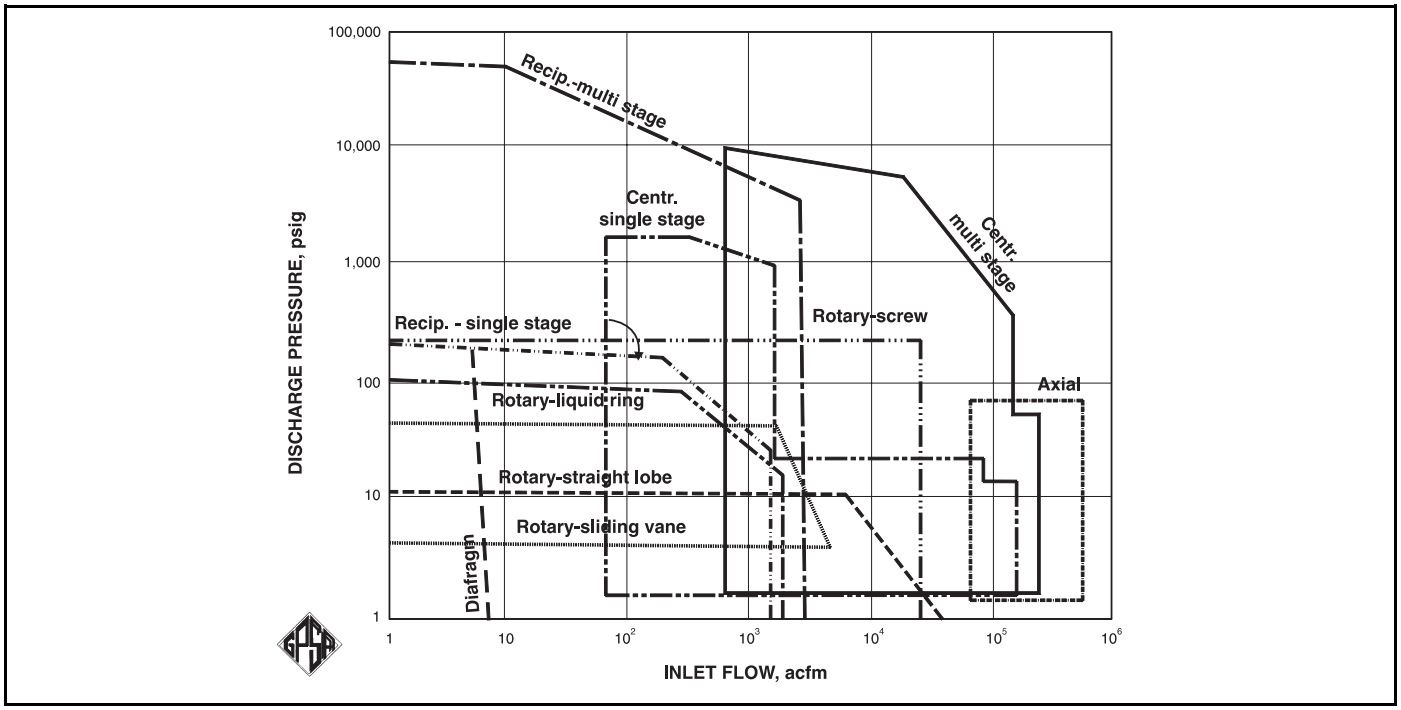
On multistage machines, intercoolers are normally provided between stages. These are heat exchangers which remove the heat of compression from the gas and reduce its temperature to approximately the temperature existing at the compressor intake. Such cooling reduces the actual volume of gas going to the high-pressure cylinders, reduces the horsepower required for compression, and keeps the temperature within safe operating limits.

Reciprocating compressors should be supplied with clean gas as they cannot satisfactorily handle liquids and solid particles that may be entrained in the gas. Liquids and solid particles tend to destroy cylinder lubrication and cause excessive

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FIG. 13-3

Compressor Coverage Chart



wear. Liquids are non-compressible and their presence could rupture the compressor cylinder or cause other major damage.

Reciprocating compressors are typically designed to one of the following industry standard specifications:

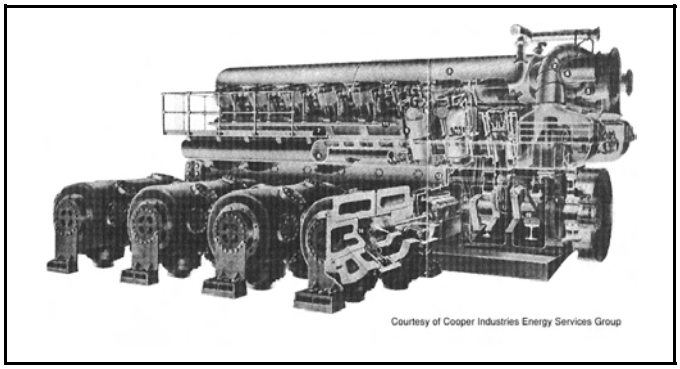
API Standard 618 "Reciprocating Compressors for Petroleum, Chemical, and Gas Industry Services."

API Specification 11P "Specification for Packaged Reciprocating Compressors for Oil & Gas Production Services."

API 618 covers low to moderate speed compressors, typically in the 300–700 rpm speed range. These compressors may be of a separable design, typically electric motor driven, although an "integral" design refers to a compressor driven by a gas engine where the power cylinders of the engine that turn the crankshaft are in the same housing as the gas compression cylinders (see Fig. 13-4).

FIG. 13-4

Integral Engine Compressor



Separable API 618 compressors have historically been used in refineries and chemical plants, normally electric motor driven, although they are also used in gas plant service. In the past, integral compressors were often used in pipeline boosting service as well as inlet compression service at field gas plants.

API 11P covers packaged high-speed separable compressors with speeds from 900–1800 rpm. These compressors are normally driven by electric motors or gas engines, and are often used for field gas compression applications, including gas plants.

Comparison of API 11P and 618 Compressors		
	API 618	**API 11P**
Horsepower	1500–34 000	35–2200
Speed	300–700	900–1800
Size & weight	Larger	Smaller
Efficiency	High	Moderate
Operating life	Longer	Shorter
Maintenance costs	Lower	Higher
Purchase cost	Higher	Lower
Installation cost	Higher	Lower

Performance Calculations

The engineer in the field is frequently required to:

1. determine the approximate horsepower required to compress a certain volume of gas from some intake conditions to a given discharge pressure, and

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2. estimate the capacity of an existing compressor under specified suction and discharge conditions.

The following text outlines procedures for making these calculations from the standpoint of quick estimates and also presents more detailed calculations. For specific information on a given compressor, consult the manufacturer of that unit.

For a compression process, the enthalpy change is the best way of evaluating the work of compression. If a P-H diagram is available (as for propane refrigeration systems), the work of compression would always be evaluated by the enthalpy change of the gas in going from suction to discharge conditions. Years ago the capability of easily generating P-H diagrams for natural gases did not exist. The result was that many ways of estimating the enthalpy change were developed. They were used as a crutch and not because they were the best way to evaluate compression horsepower requirements.

Today the engineer does have available, in many cases, the capability to generate that part of the P-H diagram required for compression purposes. This is done using equations of state on a computer. This still would be the best way to evaluate the compression horsepower. The other equations are used only if access to a good equation of state is not available.

Section 13 continues to treat reciprocating and centrifugal machines as being different so far as estimation of horsepower requirements is concerned. This treatment reflects industry practice. The only difference in the horsepower evaluation is the efficiency of the machine. Otherwise the basic thermodynamic equations are the same for all compression.

The reciprocating compressor horsepower calculations presented are based on charts. However, they may equally well be calculated using the equations in the centrifugal compressor section, particularly Eqs. 13-25 through 13-43. This also includes the mechanical losses in Eqs. 13-37 and 13-38.

There are two ways in which the thermodynamic calculations for compression can be carried out — by assuming:

1. isentropic reversible path — a process during which there is no heat added to or removed from the system and the entropy remains constant, pv^k = constant
2. polytropic reversible path — a process in which changes in gas characteristics during compression are considered, pv^n = constant

Fig. 13-5 shows a plot of pressure vs. volume for each value of exponent n. The work, W, performed in proceeding from p_1 to p_2 along any polytropic curve (Fig. 13-5) is

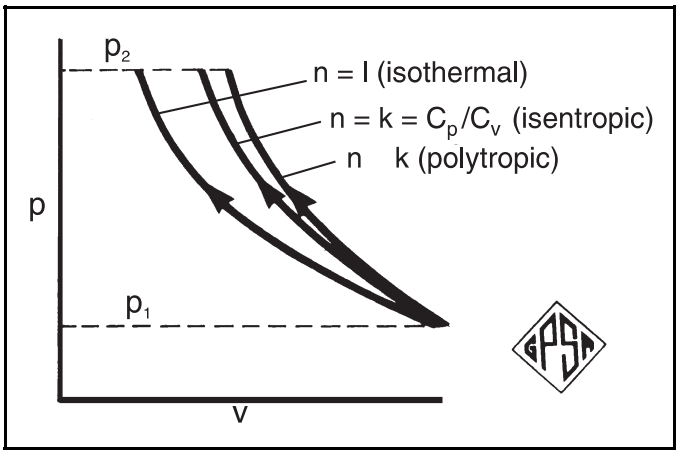
$$W = \int_1^2 V \cdot dp = \int_{p_1}^{p_2} V \cdot dp \qquad \textbf{Eq 13-1}$$

The amount of work required is dependent upon the polytropic curve involved and increases with increasing values of n. The path requiring the least amount of input work is n = 1, which is equivalent to isothermal compression, a process during which there is no change in temperature. For isentropic compression, n = k = ratio of specific heat at constant pressure to that at constant volume.

It is usually impractical to build sufficient heat-transfer equipment into the design of most compressors to carry away the bulk of the heat of compression. Most machines tend to operate along a polytropic path which approaches the isentropic. Most compressor calculations are therefore based on an efficiency applied to account for true behavior.

FIG. 13-5

Compression Curves



A compression process following the middle curve in Fig. 13-5 has been widely referred to in industry as “adiabatic”. However, all compression processes of practical importance are adiabatic. The term adiabatic does not adequately describe this process, since it only implies no heat transfer. The ideal process also follows a path of constant entropy and should be called “isentropic”, as will be done subsequently in this chapter.

Eq 13-3 which applies to all ideal gases can be used to calculate k.

$$MC_p - MC_v = R = 1.986 \text{ Btu/(lbmol} \cdot \text{°F)} \qquad \textbf{Eq 13-2}$$

By rearrangement and substitution we obtain:

$$k = \frac{C_p}{C_v} = \frac{MC_p}{MC_v} = \frac{MC_p}{MC_p - 1.986} \qquad \textbf{Eq 13-3}$$

To calculate k for a gas we need only know the constant pressure molar heat capacity (MC_p) for the gas. Fig. 13-6 gives values of molecular weight and ideal-gas state heat capacity (i.e. at 1 atm) for various gases. The heat capacity varies considerably with temperature. Since the temperature of the gas increases as it passes from suction to discharge in the compressor, k is normally determined at the average of suction and discharge temperatures.

For a multi-component gas, the mole weighted average value of molar heat capacity must be determined at average cylinder temperature. A sample calculation is shown in Fig. 13-7.

The calculation of pP_c and pT_c in Fig. 13-7 permits calculation of the reduced pressure $P_R = P/pP_c$ mix and reduced temperature $T_R = T/pT_c$ mix. The compressibility Z at T and P can then be determined using the charts in Section 23.

If only the molecular weight of the gas is known and not its composition, an approximate value for k can be determined from the curves in Fig. 13-8.

Estimating Compressor Horsepower

Eq 13-4 is useful for obtaining a quick and reasonable estimate for compressor horsepower. It was developed for large slow-speed (300 to 450 rpm) compressors handling gases with a specific gravity of 0.65 and having stage compression ratios above 2.5.

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FIG. 13-6

Molar Heat Capacity MC_p (Ideal-Gas State), Btu/(lb mol • °R)

*Data source: Selected Values of Properties of Hydrocarbons, API Research Project 44; MW updated to agree with Fig. 23-2

Gas	Chemical formula	Mol wt	Temperature							
			0°F	50°F	60°F	100°F	150°F	200°F	250°F	300°F
Methane	CH_4	16.043	8.23	8.42	8.46	8.65	8.95	9.28	9.64	10.01
Ethyne (Acetylene)	C_2H_2	26.038	9.68	10.22	10.33	10.71	11.15	11.55	11.90	12.22
Ethene (Ethylene)	C_2H_4	28.054	9.33	10.02	10.16	10.72	11.41	12.09	12.76	13.41
Ethane	C_2H_6	30.070	11.44	12.17	12.32	12.95	13.78	14.63	15.49	16.34
Propene (Propylene)	C_3H_6	42.081	13.63	14.69	14.90	15.75	16.80	17.85	18.88	19.89
Propane	C_3H_8	44.097	15.65	16.88	17.13	18.17	19.52	20.89	22.25	23.56
1-Butene (Butlyene)	C_4H_8	56.108	17.96	19.59	19.91	21.18	22.74	24.26	25.73	27.16
cis-2-Butene	C_4H_8	56.108	16.54	18.04	18.34	19.54	21.04	22.53	24.01	25.47
trans-2-Butene	C_4H_8	56.108	18.84	20.23	20.50	21.61	23.00	24.37	25.73	27.07
iso-Butane	C_4H_{10}	58.123	20.40	22.15	22.51	23.95	25.77	27.59	29.39	31.11
n-Butane	C_4H_{10}	58.123	20.80	22.38	22.72	24.08	25.81	27.55	29.23	30.90
iso-Pentane	C_5H_{12}	72.150	24.94	27.17	27.61	29.42	31.66	33.87	36.03	38.14
n-Pentane	C_5H_{12}	72.150	25.64	27.61	28.02	29.71	31.86	33.99	36.08	38.13
Benzene	C_6H_6	78.114	16.41	18.41	18.78	20.46	22.45	24.46	26.34	28.15
n-Hexane	C_6H_{14}	86.177	30.17	32.78	33.30	35.37	37.93	40.45	42.94	45.36
n-Heptane	C_7H_{16}	100.204	34.96	38.00	38.61	41.01	44.00	46.94	49.81	52.61
Ammonia	NH_3	17.0305	8.52	8.52	8.52	8.52	8.52	8.53	8.53	8.53
Air		28.9625	6.94	6.95	6.95	6.96	6.97	6.99	7.01	7.03
Water	H_2O	18.0153	7.98	8.00	8.01	8.03	8.07	8.12	8.17	8.23
Oxygen	O_2	31.9988	6.97	6.99	7.00	7.03	7.07	7.12	7.17	7.23
Nitrogen	N_2	28.0134	6.95	6.95	6.95	6.96	6.96	6.97	6.98	7.00
Hydrogen	H_2	2.0159	6.78	6.86	6.87	6.91	6.94	6.95	6.97	6.98
Hydrogen sulfide	H_2S	34.08	8.00	8.09	8.11	8.18	8.27	8.36	8.46	8.55
Carbon monoxide	CO	28.010	6.95	6.96	6.96	6.96	6.97	6.99	7.01	7.03
Carbon dioxide	CO_2	44.010	8.38	8.70	8.76	9.00	9.29	9.56	9.81	10.05

* Exceptions: Air - Keenan and Keyes, Thermodynamic Properties of Air, Wiley, 3rd Printing 1947. Ammonia - Edw. R. Grabl, Thermodynamic Properties of Ammonia at High Temperatures and Pressures, Petr. Processing, April 1953. Hydrogen Sulfide - J. R. West, Chem. Eng. Progress, 44, 287, 1948.

FIG. 13-7

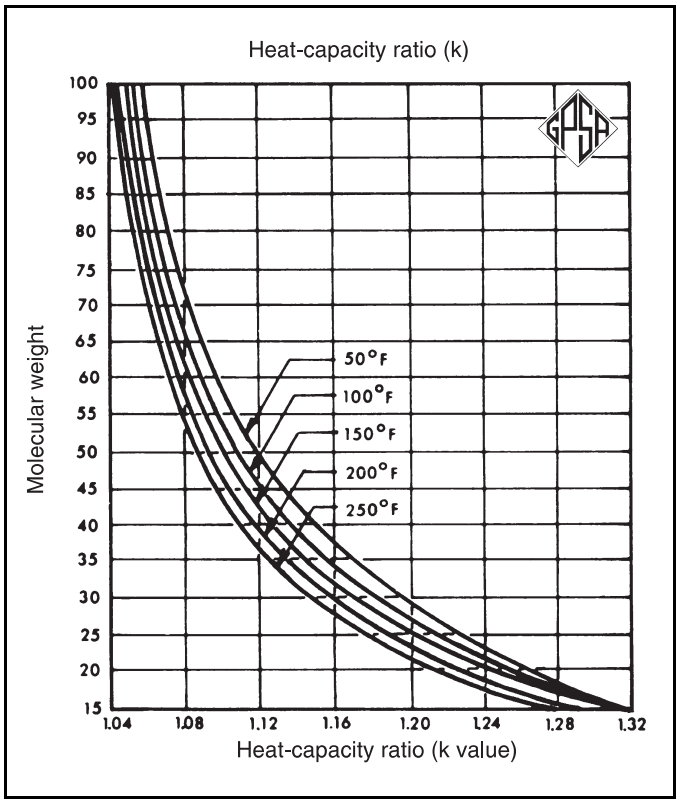
Calculation of k

Example gas mixture		Determination of mixture mol weight		Determination of MC_p, Molar heat capacity		Determination of pseudo critical pressure, pP_c, and temperature, pT_c			
Component name	Mol fraction y	Individual Component Mol weight MW	y • MW	Individual Component MC_p @ 150°F*	y • MC_p @ 150°F	Component critical pressure P_c psia	y • P_c	Component critical temperature T_c°R	y • T_c
methane	0.9216	16.04	14.782	8.95	8.248	666	615.6	343	316.1
ethane	0.0488	30.07	1.467	13.78	0.672	707	34.6	550	26.8
propane	0.0185	44.10	0.816	19.52	0.361	616	11.4	666	12.3
i-butane	0.0039	58.12	0.227	25.77	0.101	528	2.1	734	2.9
n-butane	0.0055	58.12	0.320	25.81	0.142	551	3.0	765	4.2
i-pentane	0.0017	72.15	0.123	31.66	0.054	490	0.8	829	1.4
Total	1.0000	MW =	17.735	MC_p =	9.578	pP_c =	667.5	pT_c =	363.7
$MC_v = MC_p - 1.986 = 7.592$						$k = MC_p/MC_v = 9.578/7.592 = 1.26$			
*For values of MC_p other than @ 150°F, refer to Fig. 13-6									

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FIG. 13-8

Approximate Heat-Capacity Ratios of Hydrocarbon Gases



CAUTION: Compressor manufacturers generally rate their machines based on a standard condition of 14.4 psia rather than the more common gas industry value of 14.7 psia.

Due to higher valve losses, the horsepower requirement for high-speed compressors (1000 rpm range, and some up to 1800 rpm) can be as much as 20% higher, although this is a very arbitrary value. Some compressor designs do not merit a higher horsepower allowance and the manufacturers should be consulted for specific applications.

$$\text{Brake horsepower} = (22)\left(\frac{\text{ratio}}{\text{stage}}\right)(\#\text{ of stages})\,(\text{MMcfd})\,(F) \qquad \textbf{Eq 13-4}$$

Where:

MMcfd = Compressor capacity referred to 14.4 psia and intake temperature

F = 1.0 for single-stage compression
1.08 for two-stage compression
1.10 for three-stage compression

Eq 13-4 will also provide a rough estimate of horsepower for lower compression ratios and/or gases with a higher specific gravity, but it will tend to be on the high side. To allow for this tendency use a multiplication factor of 20 instead of 22 for gases with a specific gravity in the 0.8 to 1.0 range; likewise, use a factor in the range of 16 to 18 for compression ratios between 1.5 and 2.0.

Curves are available which permit easy estimation of approximate compression-horsepower requirements. Fig. 13-9 is typical of these curves.

Example 13-1 — Compress 2 MMcfd of gas at 14.4 psia and intake temperature through a compression ratio of 9 in a 2-stage compressor. What will be the horsepower?

Solution Steps

Ratio per stage = $\sqrt{9} = 3$

From Eq 13-4 we find the brake horsepower to be:

(22) (3) (2) (2) (1.08) = 285 Bhp

From Fig. 13-9, using a k of 1.15, we find the horsepower requirement to be 136 Bhp/MMcfd or 272 Bhp. For a k of 1.4, the power requirement would be 147 Bhp/MMcfd or 294 total horsepower.

The two procedures give reasonable agreement, particularly considering the simplifying assumptions necessary in reducing compressor horsepower calculations to such a simple procedure.

Detailed Calculations

There are many variables which enter into the precise calculation of compressor performance. Generalized data as given in this section are based upon the averaging of many criteria. The results obtained from these calculations, therefore, must be considered as close approximations to true compressor performance.

Capacity

Most gases encountered in industrial compression do not exactly follow the ideal gas equation of state but differ in varying degrees. The degree in which any gas varies from the ideal is expressed by a compressibility factor, Z, which modifies the ideal gas equation:

$$PV = nRT \qquad \textbf{Eq 13-5}$$

to $$PV = nZRT \qquad \textbf{Eq 13-6}$$

Compressibility factors can be determined from charts in Section 23 using the pP_R and pT_R of the gas mixture. For pure components such as propane, compressibility factors can be determined from the P-H diagrams, although the user would be better advised to determine the compression horsepower using the P-H diagram (see Section 24).

For the purpose of performance calculations, compressor capacity is expressed as the actual volumetric quantity of gas at the inlet to each stage of compression on a per minute basis (ICFM).

From SCFM

$$Q = \text{SCFM}\left(\frac{14.7}{520}\right)\left(\frac{T_1 Z_1}{P_1 Z_L}\right) \qquad \textbf{Eq 13-7}$$

From weight flow (w, lb/min)

$$Q = \frac{10.73}{MW}\left(\frac{w\,T_1\,Z_1}{P_1\,Z_L}\right) \qquad \textbf{Eq 13-8}$$

From molar flow (Nm, mols/min)

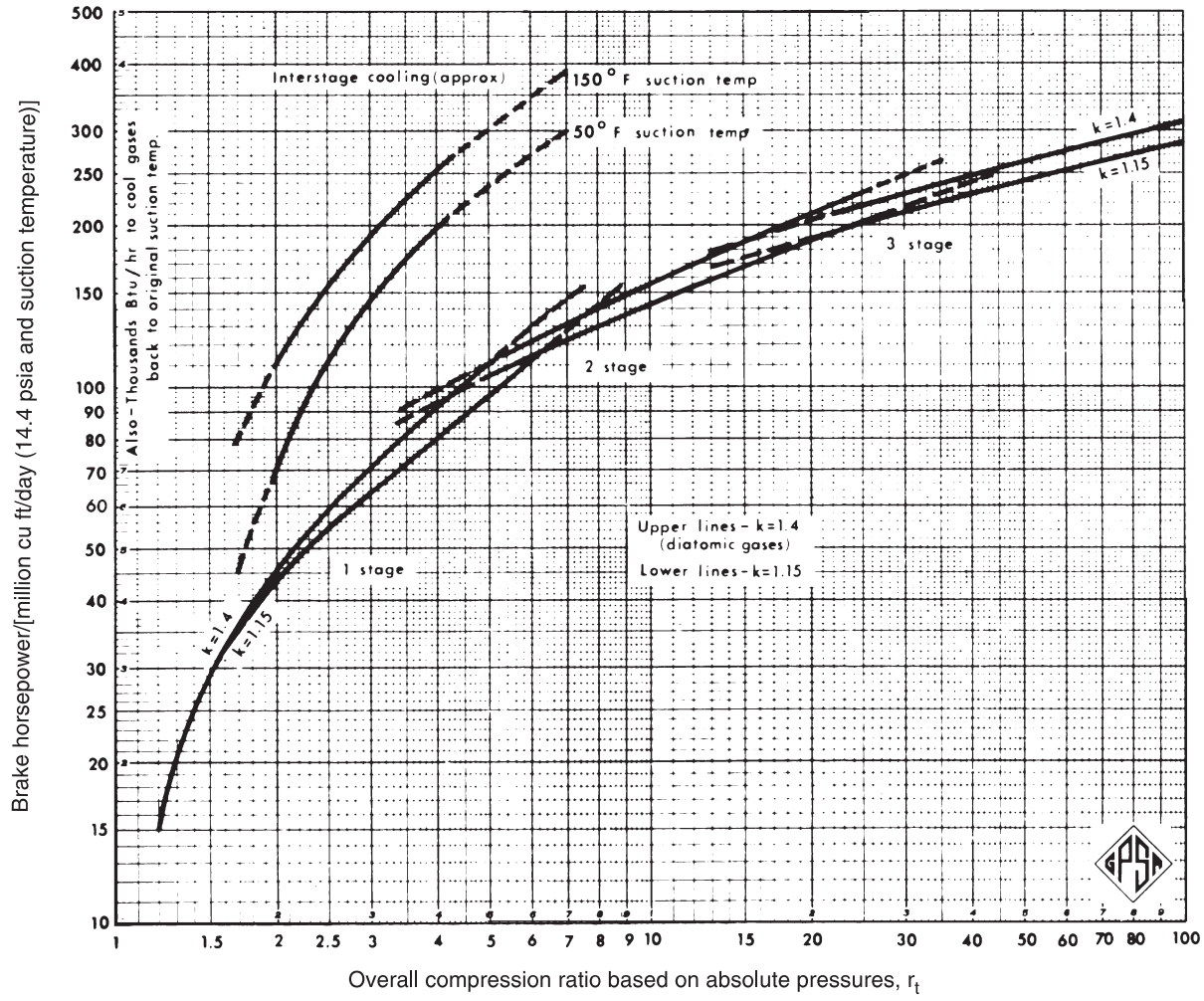
$$Q = \left(\frac{379.5 \cdot 14.7}{520}\right)\left(\frac{N_m\,T_1\,Z_1}{P_1\,Z_L}\right) \qquad \textbf{Eq 13-9}$$

From these equations, inlet volume to any stage may be calculated by using the inlet pressure P_1 and temperature T_1. Moisture should be handled just as any other component in the gas.

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FIG. 13-9

Approximate Horsepower Required to Compress Gases



In a reciprocating compressor, effective capacity may be calculated as the piston displacement (generally in cu ft/min) multiplied by the volumetric efficiency.

The piston displacement is equal to the net piston area multiplied by the length of piston sweep in a given period of time. This displacement may be expressed:

For a single-acting piston compressing on the outer end only,

$$PD = \frac{(\text{stroke}) (N) (D^2) \pi}{(4) \cdot (1728)} \qquad \textbf{Eq 13-10}$$

$$= 4.55 (10^{-4}) (\text{stroke}) (N) (D^2)$$

For a single-acting piston compressing on the crank end only,

$$PD = \frac{(\text{stroke}) (N) (D^2 - d^2) \pi}{(4) \cdot (1728)} \qquad \textbf{Eq 13-11}$$

$$= 4.55 (10^{-4}) (\text{stroke}) (N) (D^2 - d^2)$$

For a double-acting piston (other than tail rod type),

$$PD = \frac{(\text{stroke}) (N) (2 D^2 - d^2) \pi}{(4) \cdot (1728)} \qquad \textbf{Eq 13-12}$$

$$= 4.55 (10^{-4}) (\text{stroke}) (N) (2 D^2 - d^2)$$

Volumetric Efficiency

In a reciprocating compressor, the piston does not travel completely to the end of the cylinder at the end of the discharge stroke. Some clearance volume is necessary and it includes the space between the end of the piston and the cylinder head when the piston is at the end of its stroke. It also includes the volume in the valve ports, the volume in the suction valve guards, and the volume around the discharge valve seats.

Clearance volume is usually expressed as a percent of piston displacement and referred to as percent clearance, or cylinder clearance, C.

$$C = \frac{\text{clearance volume, cu in.}}{\text{piston displacement, cu in.}} (100) \qquad \textbf{Eq 13-13}$$

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For double acting cylinders, the percent clearance is based on the total clearance volume for both the head end and the crank end of a cylinder. These two clearance volumes are not the same due to the presence of the piston rod in the crank end of the cylinder. Sometimes additional clearance volume (external) is intentionally added to reduce cylinder capacity.

The term "volumetric efficiency" refers to the actual pumping capacity of a cylinder compared to the piston displacement. Without a clearance volume for the gas to expand and delay the opening of the suction valve(s), the cylinder could deliver its entire piston displacement as gas capacity. The effect of the gas contained in the clearance volume on the pumping capacity of a cylinder can be represented by:

$$VE = 100 - r - C\left[\frac{Z_s}{Z_d}(r^{1/k}) - 1\right] \qquad \textbf{Eq 13-14}$$

Volumetric efficiencies as determined by Eq. 13-14 are theoretical in that they do not account for suction and discharge valve losses. The suction and discharge valves are actually spring-loaded check valves that permit flow in one direction only. The springs require a small differential pressure to open. For this reason, the pressure within the cylinder at the end of the suction stroke is lower than the line suction pressure and, likewise, the pressure at the end of the discharge stroke is higher than line discharge pressure.

One method for accounting for suction and discharge valve losses is to reduce the volumetric efficiency by an arbitrary amount, typically 4%, thus modifying Eq. 13-14 as follows:

$$VE = 96 - r - C\left[\frac{Z_s}{Z_d}(r^{1/k}) - 1\right] \qquad \textbf{Eq 13-15}$$

When a non-lubricated compressor is used, the volumetric efficiency should be corrected by subtracting an additional 5% for slippage of gas. This is a capacity correction only and, as a first approximation, would not be considered when calculating compressor horsepower. The energy of compression is used by the gas even though the gas slips by the rings and is not discharged from the cylinder.

If the compressor is in propane, or similar heavy gas service, an additional 4% should be subtracted from the volumetric efficiency. These deductions for non-lubricated and propane performance are both approximate and, if both apply, cumulative.

Fig. 13-10 provides the solution to the function $r^{1/k}$. Values for compression ratios not shown may be obtained by interpo-

FIG. 13-10

Values of $r^{1/k}$

Compression Ratio	k, isentropic exponent C_p/C_v								
	1.10	1.14	1.18	1.22	1.26	1.30	1.34	1.38	1.42
1.2	1.180	1.173	1.167	1.161	1.156	1.151	1.146	1.141	1.137
1.4	1.358	1.343	1.330	1.318	1.306	1.295	1.285	1.276	1.267
1.6	1.533	1.510	1.489	1.470	1.452	1.436	1.420	1.406	1.392
1.8	1.706	1.675	1.646	1.619	1.594	1.572	1.551	1.531	1.513
2.0	1.878	1.837	1.799	1.765	1.733	1.704	1.677	1.652	1.629
2.2	2.048	1.997	1.951	1.908	1.870	1.834	1.801	1.771	1.742
2.4	2.216	2.155	2.100	2.050	2.003	1.961	1.922	1.886	1.852
2.6	2.384	2.312	2.247	2.188	2.135	2.086	2.040	1.999	1.960
2.8	2.550	2.467	2.393	2.326	2.264	2.208	2.156	2.109	2.065
3.0	2.715	2.621	2.537	2.461	2.391	2.328	2.270	2.217	2.168
3.2	2.879	2.774	2.680	2.595	2.517	2.447	2.382	2.323	2.269
3.4	3.042	2.926	2.821	2.727	2.641	2.563	2.492	2.427	2.367
3.6	3.204	3.076	2.961	2.857	2.764	2.679	2.601	2.530	2.465
3.8	3.366	3.225	3.100	2.987	2.885	2.792	2.708	2.631	2.560
4.0	3.526	3.374	3.238	3.115	3.005	2.905	2.814	2.731	2.655
4.2	3.686	3.521	3.374	3.242	3.124	3.016	2.918	2.829	2.747
4.4	3.846	3.668	3.510	3.368	3.241	3.126	3.021	2.926	2.839
4.6	4.004	3.814	3.645	3.493	3.357	3.235	3.123	3.022	2.929
4.8	4.162	3.959	3.779	3.617	3.473	3.342	3.224	3.116	3.018
5.0	4.319	4.103	3.912	3.740	3.587	3.449	3.324	3.210	3.106
5.2	4.476	4.247	4.044	3.863	3.700	3.554	3.422	3.303	3.193
5.4	4.632	4.390	4.175	3.984	3.813	3.659	3.520	3.394	3.279
5.6	4.788	4.532	4.306	4.105	3.925	3.763	3.617	3.485	3.364
5.8	4.943	4.674	4.436	4.224	4.035	3.866	3.713	3.574	3.448
6.0	5.098	4.815	4.565	4.343	4.146	3.968	3.808	3.663	3.532
6.2	5.252	4.955	4.694	4.462	4.255	4.069	3.902	3.751	3.614
6.4	5.406	5.095	4.822	4.579	4.363	4.170	3.996	3.839	3.696
6.6	5.560	5.235	4.949	4.696	4.471	4.270	4.089	3.925	3.777
6.8	5.713	5.374	5.076	4.813	4.578	4.369	4.181	4.011	3.857

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lation. The closest k value column may be safely used without a second interpolation.

Volumetric efficiencies for "high speed" separable compressors in the past have tended to be slightly lower than estimated from Eq 13-14. Recent information suggests that this modification is not necessary for all models of high speed compressors.

In evaluating efficiency, horsepower, volumetric efficiency, etc., the user should consider past experience with different speeds and models. Larger valve area for a given swept volume will generally lead to higher compression efficiencies.

Equivalent Capacity

The net capacity for a compressor, in cubic feet per day @ 14.4 psia and suction temperature, may be calculated by Eq. 13-16a which is shown in dimensioned form:

$$\text{MMcfd} = \frac{\text{PD}\ \frac{\text{ft}^3}{\text{min}} \cdot 1400\ \frac{\text{min}}{\text{d}} \cdot \frac{\text{VE\%}}{100} \cdot \text{P}_s\ \frac{\text{lb}}{\text{in}^2} \cdot 10^{-6} \frac{\text{MMft}^3}{\text{ft}^3} \cdot Z_{14.4}}{14.4\ \frac{\text{lb}}{\text{in}^2} \cdot Z_s}$$

Eq 13-16a

which can be simplified to Eq. 13-16b when $Z_{14.4}$ is assumed to equal 1.0.

$$\text{MMcfd} = \frac{\text{PD} \cdot \text{VE} \cdot \text{P}_s \cdot 10^{-6}}{Z_s} \qquad \textbf{Eq 13-16b}$$

For example, a compressor with 200 cu ft/min piston displacement, a volumetric efficiency of 80%, a suction pressure of 75 psia, and suction compressibility of 0.9 would have a capacity of 1.33 MMcfd at 14.4 psia and suction temperature. If compressibility is not used as a divisor in calculating cu ft/min, then the statement "not corrected for compressibility" should be added.

In many instances the gas sales contract or regulation will specify some other measurement standard for gas volume. To convert volumes calculated using Equation 13-16 (i.e. at 14.4 psia and suction temperature) to a P_L and T_L basis, Eq 13-17 would be used:

$$\text{MMscfd at } P_L, T_L = (\text{MMcfd from Eq 13–16}) \left(\frac{14.4}{P_L}\right)\left(\frac{T_L}{T_s}\right)\left(\frac{Z_L}{Z_s}\right)$$

Eq 13-17

Discharge Temperature

The temperature of the gas discharged from the cylinder can be estimated from Eq 13-18, which is commonly used but not recommended. (Note: the temperatures are in absolute units, °R or K.) Eqs 13-31 and 13-32 give better results.

$$T_d = T_s\,(r^{(k-1)/k}) \qquad \textbf{Eq 13-18}$$

The discharge temperature determined from Eq 13-18 is the theoretical value. While it neglects heat from friction, irreversibility effects, etc., and may be somewhat low, the values obtained from this equation will be reasonable field estimates.

Rod Loading

Each compressor frame has definite limitations as to maximum speed and load-carrying capacity. The load-carrying capacity of a compressor frame involves two primary considerations: horsepower and rod loading.

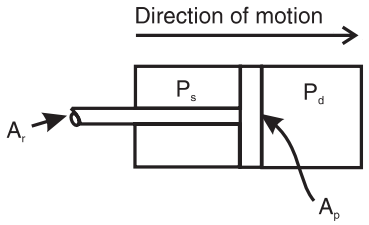
The horsepower rating of a compressor frame is the measure of the ability of the supporting structure and crankshaft to withstand torque (turning force) and the ability of the bearings to dissipate frictional heat. Rod loads are established to limit the static and inertial loads on the crankshaft, connecting rod, frame, piston rod, bolting, and projected bearing surfaces.

Good design dictates a reversal of rod loading during each stroke. Non-reversal of the loading results in failure to allow bearing surfaces to part and permit entrance of sufficient lubricant. The result will be premature bearing wear or failure.

Rod loadings may be calculated by the use of Eqs 13-19 and 13-20.

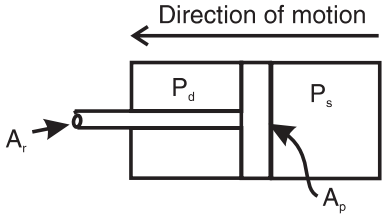
$$\begin{aligned}\text{Load in compression} &= P_d\,A_p - P_s\,(A_p - A_r) \\ &= (P_d - P_s)\,A_p + P_s\,A_r \end{aligned} \qquad \textbf{Eq 13-19}$$

Rod in Compression



$$\begin{aligned}\text{Load in tension} &= P_d\,(A_p - A_r) - P_s\,A_p \\ &= (P_d - P_s)\,A_p - P_d\,A_r \end{aligned} \qquad \textbf{Eq 13-20}$$

Rod in Tension



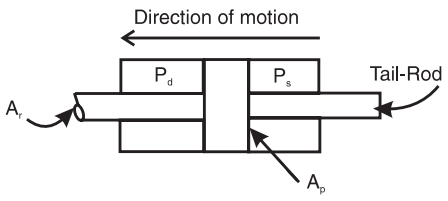
Using Eqs. 13-19 and 13-20, a plus value for the load in both compression and tension indicates a reversal of loads based on gas pressure only. Inertial effects will tend to increase the degree of reversal.

The true rod loads would be those calculated using internal cylinder pressures after allowance for valve losses. Normally, the operator will know only line pressures, and because of this, manufacturers generally rate their compressors based on line-pressure calculations.

A further refinement in the rod-loading calculation would be to include inertial forces. While the manufacturer will consider inertial forces when rating compressors, useful data on this point is seldom available in the field. Except in special cases, inertial forces are ignored.

A tail-rod cylinder would require consideration of rod cross-section area on both sides of the piston instead of on only one side of the piston, as in Eqs 13-19 and 13-20.

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Detailed Horsepower Calculation

A more detailed calculation of reciprocating compressor power requirements can be performed using the following equation:

$$\text{BHP/stage} = 3.03 \bullet Z_{avg} \bullet [Q_g T_s/E] \bullet (k/(k-1)) \bullet \left(\frac{P_L}{T_L}\right) \bullet [(P_d/P_s)^{((k-1)/k)}-1] \qquad \textbf{Eq 13-21}$$

BHP = brake horsepower
Q_g = gas flow rate, MMSCFD
T_s = suction temperature, °R
Z_{avg} = $(Z_s+Z_d)/2$
Z_s = suction compressibility factor
Z_d = discharge compressibility factor
E = overall efficiency
High speed reciprocating units — 0.82
Low speed reciprocating units — 0.85
K = ratio of specific heats, Cp/Cv
P_s = suction pressure, psia
P_d = discharge pressure, psia
P_L= standard pressure, psia
T_L= standard temperature, °R

The total horsepower for the compressor is the sum of the horsepower required for each of the stages that are utilized. For multistage machines an allowance should be made for the interstage pressure drop associated with piping, cooler, scrubber, etc., typically 5–10 psi.

Procedure

1. Calculate overall compression ratio ($r_t = P_{dfinal} - P_s$).
2. Calculate the compression ratio per stage, r, by taking the s root of r_t, where s is the number of compression stages. The number of stages, s, should be increased until the ratio per stage, r, is < ~ 4. This should generally result in stage discharge temperatures of < 300°F depending on the interstage cooler outlet temperature assumed.
3. Multiplying r by the absolute suction pressure of the stage being considered will give you discharge pressure of the stage.
4. Calculate the horsepower required for the stage using Eq 13-21.
5. Substract the assumed interstage pressure loss from the discharge pressure of the preceding stage to obtain the suction pressure for the next stage.
6. Repeat steps 4 and 5 until all stages have been calculated.
7. Sum the stage horsepowers to obtain the total compressor power required.

Example 13-2 — Compress 2 MMscfd of gas measured at 14.65 psia and 60°F. Intake pressure is 100 psia, and intake temperature is 100°F. Discharge pressure is 900 psia. The gas has a specific gravity of 0.80 (23 MW). What is the required horsepower, assuming a high speed compressor?

1. Compression ratio is

   $$\frac{900 \text{ psia}}{100 \text{ psia}} = 9$$

   This would be a two-stage compressor; therefore, the ratio per stage is $\sqrt{9}$ or 3.

2. 100 psia x 3 = 300 psia (1st stage discharge pressure)

   300 psia = 295 psia (suction to 2nd stage)

   Where the 5 psi represents the pressure drop between first stage discharge and second stage suction.

   $$\frac{900 \text{ psia}}{295 \text{ psia}} = 3.05 \text{ (compression ratio for 2nd stage)}$$

   It may be desirable to recalculate the interstage pressure to balance the ratios. For this sample problem, however, the first ratios determined will be used.

3. From Fig. 13-8 a gas with specific gravity of 0.8 at 150°F would have an approximate k of 1.21. For most compression applications, the 150°F curve will be adequate. This should be checked after determining the average cylinder temperature.

4. Determine the discharge temperature for the first stage from Eq 13-18. For a compression ratio of 3, discharge temperature = approximately 220°F. Average cylinder temperature = 160°F.

5. In the same manner, discharge temperature for the second stage (with r = 3.05 and assuming interstage cooling to 120°F) equals approximately 244°F. Average cylinder temperature = 182°F.

6. From the physical properties section (Section 23), estimate the compressibility factors at suction and discharge pressure and temperature of each stage.

   1st stage: $Z_s = 0.98$
   $Z_d = 0.97$
   $Z_{avg} = 0.975$
   2nd stage: $Z_s = 0.94$
   $Z_d = 0.92$
   $Z_{avg} = 0.93$

7. Calculate the horsepower required for the first and second stages from Eq 13-21:

$$\text{BHP for 1st stage} = 3.03 \bullet (0.975) \bullet [2 \bullet 560/0.82] \bullet [1.21/(1.21-1)] \bullet \left(\frac{14.65}{520}\right) \bullet [(300/100)^{(1.21-1)/1.21} - 1] = 137.6$$

$$\text{BHP for 2nd stage} = 3.03 \bullet (0.93) \bullet [2 \bullet 580/0.82] \bullet [1.21/(1.21-1)] \bullet \left(\frac{14.65}{520}\right) \bullet [(900/295)^{(1.21-1)/1.21} - 1] = 138.2$$

Total BHP required = 137.6 + 138.2 = 275.8

Note that in Example 13-1 the same conditions result in a compression power of 285 Bhp which is close agreement.

Limits to compression ratio per stage — The maximum ratio of compression permissible in one stage is

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usually limited by the discharge temperature or by rod loading, particularly in the first stage.

When handling gases containing oxygen, which could support combustion, there is a possibility of fire and explosion because of the oil vapors present.

To reduce carbonization of the oil and the danger of fires, a safe operating limit may be considered to be approximately 300°F. Where no oxygen is present in the gas stream, temperatures of 350°F may be considered as the maximum, even though mechanical or process requirements usually dictate a lower figure.

Packing life may be significantly shortened by the dual requirement to seal both high pressure and high temperature gases. For this reason, at higher discharge pressures, a temperature closer to 250°F or 275°F may be the practical limit.

In summary, and for most field applications, the use of 300°F maximum would be a good average. Recognition of the above variables is, however, still useful.

Economic considerations are also involved because a high ratio of compression will mean a low volumetric efficiency and require a larger cylinder to produce the same capacity. For this reason a high rod loading may result and require a heavier and more expensive frame.

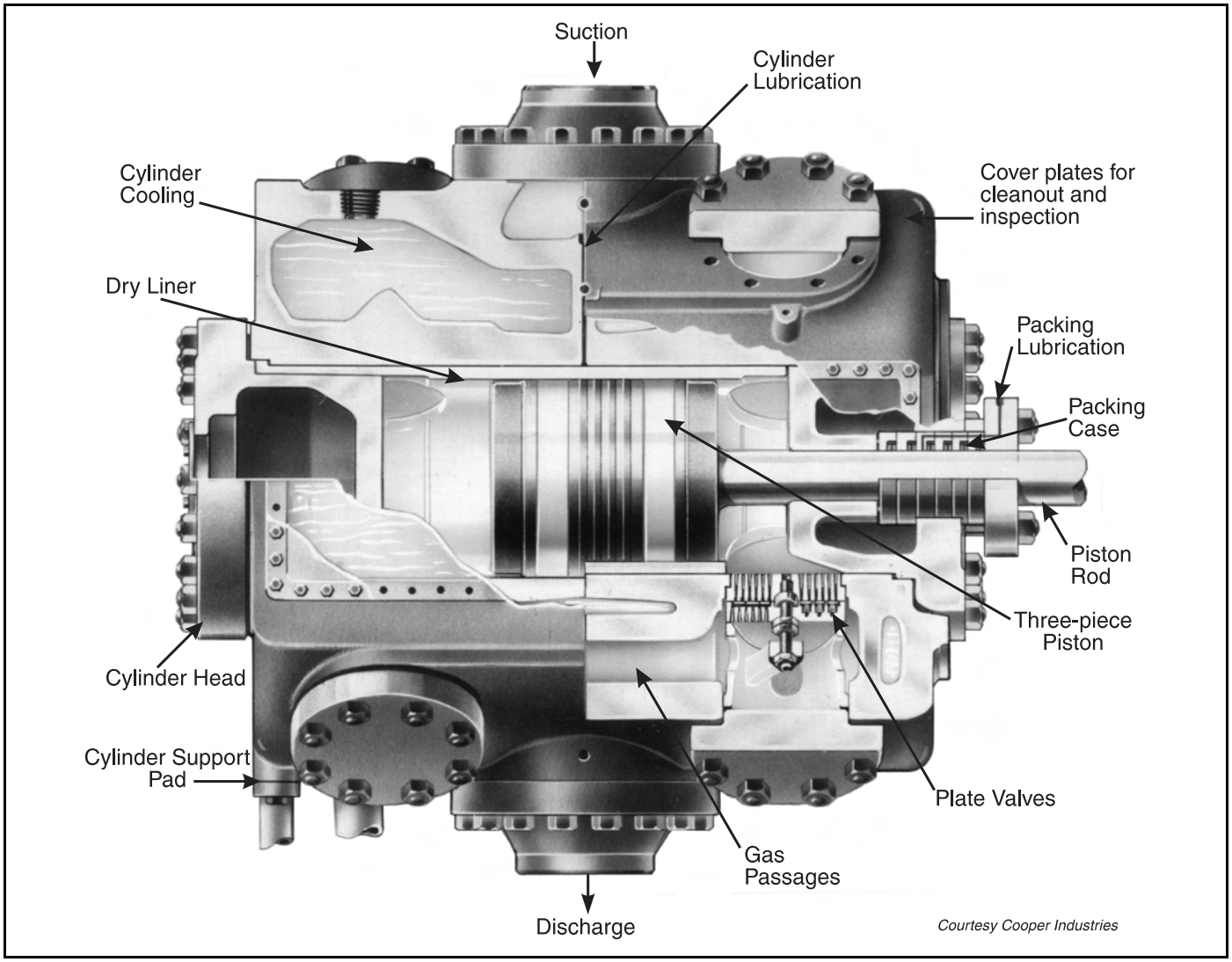
Where multi-stage operation is involved, equal ratios of compression per stage are used (plus an allowance for piping and cooler losses if necessary) unless otherwise required by process design. For two stages of compression the ratio per stage would approximately equal the square root of the total compression ratio; for three stages, the cube root, etc. In practice, especially in high-pressure work, decreasing the compression ratio in the higher stages to reduce excessive rod loading may prove to be advantageous.

Cylinder Design

Depending on the size of the machine and the number of stages, reciprocating compressors are furnished with cylinders fitted with either single- or double-acting pistons, see examples in Figs. 13-11 through 13-13.

FIG. 13-11

Low Pressure Cylinder with Double-Acting Piston



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In the same units, double-acting pistons are commonly used in the first stages and often single-acting in the higher stages of compression.

Cylinder materials are normally selected for strength; however, thermal shock, mechanical shock, or corrosion resistance may also be a determining factor. The table below shows discharge pressure limits generally used in the gas industry for cylinder material selection.

Cylinder Material	Discharge Pressure (psig)
Cast Iron	up to 1,200
Nodular Iron	about 1,500
Cast Steel	1,200 to 2,500
Forged Steel	above 2,500

API standard 618 recommends 1000 psig as the maximum pressure for both cast iron and nodular iron.

Cylinders are designed both as a solid body (no liner) and with liners. Cylinder liners are inserted into the cylinder body to either form or line the pressure wall. There are two types. The wet liner forms the pressure wall as well as the inside wall of the water jacket. The dry type lines the cylinder wall and is not required to add strength.

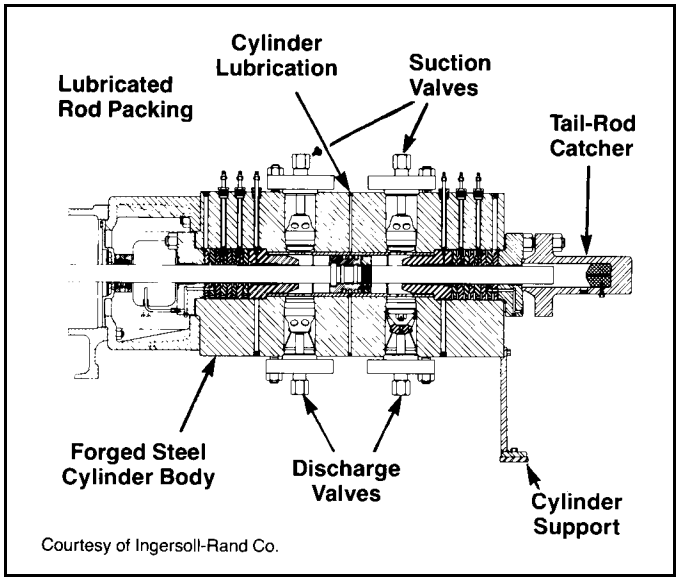
Standard cylinder liners are cast iron. If cylinders are required to have special corrosion or wear resistance, other materials or special alloys may be needed.

Most compressors use oils to lubricate the cylinder with a mechanical, force-feed lubricator having one or more feeds to each cylinder.

The non-lubricated compressor has found wide application where it is desirable or essential to compress air or gas without contaminating it with lubricating oil.

FIG. 13-12

High Pressure Cylinder with Double-Acting Piston and Tail-Rod



For such cases a number of manufacturers furnish a "non-lubricated" cylinder (Fig. 13-14). The piston on these cylinders is equipped with piston rings of graphitic carbon or plastic as well as pads or rings of the same material to maintain the proper clearance between the piston and cylinder. Plastic packing of a type that requires no lubricant is used on the stuffing box. Although oil-wiper rings are used on the piston rod where it leaves the compressor frame, minute quantities of oil might conceivably enter the cylinder on the rod. Where even such small amounts of oil are objectionable, an extended cylinder connecting piece can be furnished. This simply

FIG. 13-13

Single-Acting Plunger Cylinder Designed for 15,000 psig Discharge

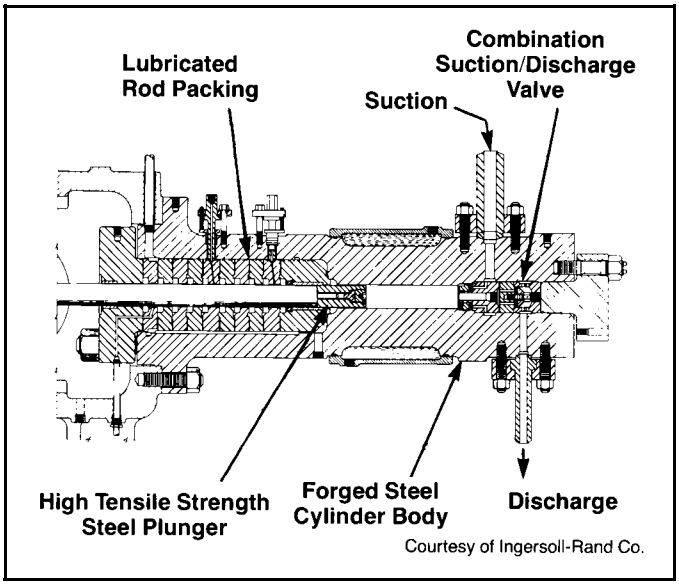
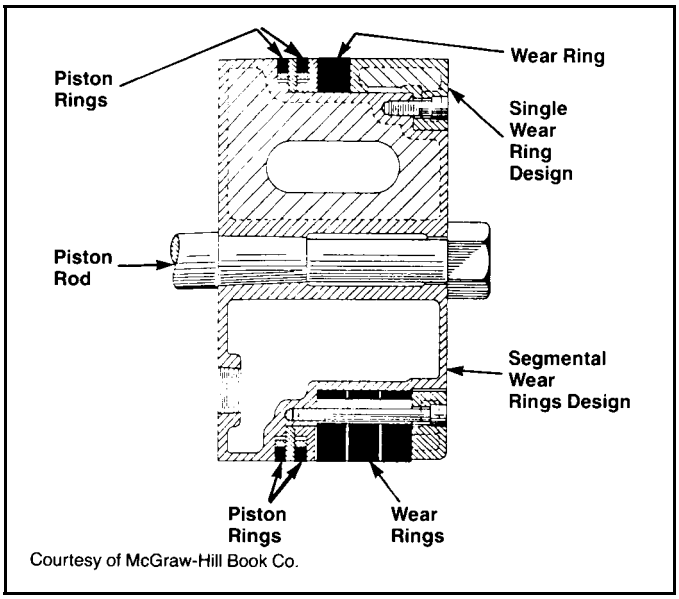


FIG. 13-14

Piston Equipped with Teflon® Piston and Wear Rings for a Single-Acting Non-Lubricated Cylinder



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lengthens the piston rod so that no lubricated portion of the rod enters the cylinder.

A small amount of gas leaking through the packing can be objectionable. Special distance pieces are furnished between the cylinder and frame, which may be either single-compartment or double-compartment. These may be furnished gas tight and vented back to the suction, or may be filled with a sealing gas or fluid and held under a slight pressure, or simply vented.

Compressor valves for non-lubricated service operate in an environment that has no lubricant in the gas or in the cylinder. Therefore, the selection of valve materials is important to prevent excessive wear.

Piston rod packing universally used in non-lubricated compressors is of the full-floating mechanical type, consisting of a case containing pairs of either carbon or plastic (TFE) rings of conventional design.

When handling oxygen and other gases such as nitrogen and helium, it is absolutely necessary that all traces of hydrocarbons in cylinders be removed. With oxygen, this is required for safety, with other gases to prevent system contamination. Deoiling schemes are discussed in Refrigeration, Section 14.

High-pressure compressors with discharge pressures from 5,000 to 30,000 psi usually require special design and a complete knowledge of the characteristics of the gas.

As a rule, inlet and discharge gas pipe connections on the cylinder are fitted with flanges of the same rating for the following reasons:

- Practicality and uniformity of casting and machinery,
- Hydrostatic test, usually at 150% design pressure,
- Suction pulsation bottles are usually designed for the same pressure as the discharge bottle (often federal, state, or local government regulation).

Reciprocating Compressor Control Devices

Output of compressors must be controlled (regulated) to match system demand.

In many installations some means of controlling the output of the compressor is necessary. Often constant flow is required despite variations in discharge pressure, and the control device must operate to maintain a constant compressor capacity. Compressor capacity, speed, or pressure may be varied in accordance with the requirements. The nature of the control device will depend on the regulating variable — whether pressure, flow, temperature, or some other variable — and on type of compressor driver.

Unloading for Starting — Practically all reciprocating compressors must be unloaded to some degree before starting so that the driver torque available during acceleration is not exceeded. Both manual and automatic compressor startup unloading is used. Common methods of unloading include: discharge venting, discharge to suction bypass, and holding open the inlet valves using valve lifters.

Capacity Control — The most common requirement is regulation of capacity. Many capacity controls, or unloading devices, as they are usually termed, are actuated by the pressure on the discharge side of the compressor. A falling pressure indicates that gas is being used faster than it is being compressed and that more gas is required. A rising pressure indicates that more gas is being compressed than is being used and that less gas is required.

A common method of controlling the capacity of a compressor is to vary the speed. This method is applicable to steam-driven compressors and to units driven by internal combustion engines. In these cases the regulator actuates the steam-admission or fuel-admission valve on the compressor driver to control the speed.

Electric motor-driven compressors usually operate at constant speed, although variable speed drives are becoming increasingly more common. For constant speed motors other methods of controlling the capacity are necessary. On reciprocating compressors up to about 100 hp, two types of control are usually available. These are automatic-start-and-stop control and constant-speed control.

Automatic-start-and-stop control, as its name implies, stops or starts the compressor by means of a pressure-actuated switch as the gas demand varies. It should be used only when the demand for gas will be intermittent.

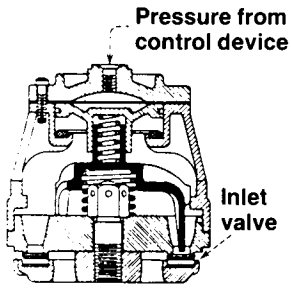
Constant-speed control permits the compressor to operate at full speed continuously, loaded part of the time and fully or partially unloaded at other times. Two methods of unloading the compressor with this type of control are in common use: inlet-valve unloaders, and clearance unloaders. Inlet-valve unloaders (Fig. 13-15) operate to hold the compressor inlet valves open and thereby prevent compression. Clearance unloaders (Fig. 13-16) consist of pockets or small reservoirs which are opened when unloading is desired. The gas is compressed into them on the compression stroke and expands into the cylinder on the return stroke, reducing the intake of additional gas.

Motor-driven reciprocating compressors above 100 hp in size are usually equipped with a step control. This is in reality a variation of constant-speed control in which unloading is accomplished in a series of steps, varying from full load down to no load.

Five-step control (full load, three-quarter load, one-half load, one-quarter load, and no load) is accomplished by means of clearance pockets. On some makes of machines inlet-valve and clearance control unloading are used in combination.

FIG. 13-15

Inlet Valve Unloader



Courtesy of McGraw-Hill Book Co.

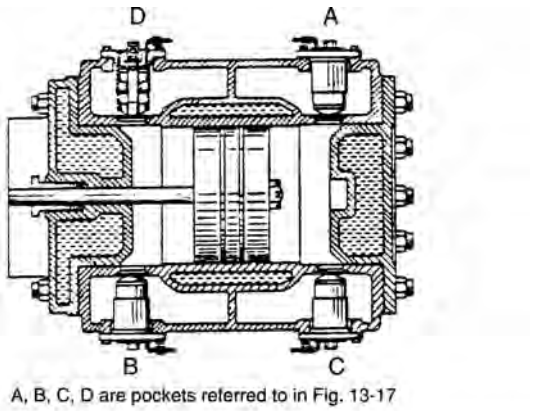
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A common practice in the natural gas industry is to prepare a single set of curves for a given machine unless there are side loads or it is a multi-service machine.

Fig. 13-17 shows indicator cards which demonstrate the unloading operation for a double acting cylinder at three capacity points. The letters adjacent to the low-pressure diagrams represent the unloading influence of the respective and cumulative effect of the various pockets as identified in Fig. 13-21. Full load, one-half, and no load capacity is obtained by holding corresponding suction valves open or adding sufficient clearance to produce a zero volumetric efficiency. No-capacity operation includes holding all suction valves open.

FIG. 13-16

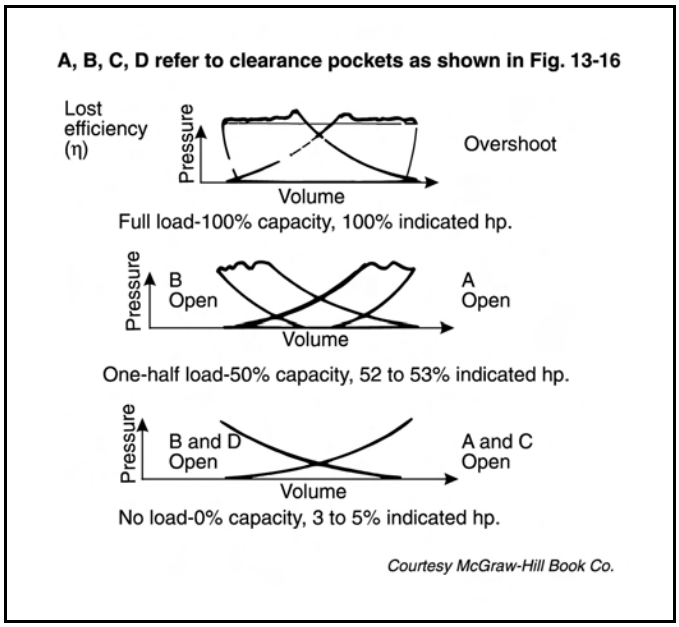
Pneumatic Valves Controlling Four Fixed Pockets in Compressor for Five-Step Control



Courtesy McGraw-Hill Book Co.

FIG. 13-17

Indicator Diagram for Three Load Points of Operation



Courtesy McGraw-Hill Book Co.

Fig. 13-18 shows an alternative representation of compressor unloading operation with a step-control using fixed volume clearance pockets. The curve illustrates the relationship between compressor capacity and driver capacity for a varying compressor suction pressure at a constant discharge pressure and constant speed. The driver can be a gas engine or electric motor.

The purpose of this curve is to determine what steps of unloading are required to prevent the driver and piston rods from serious overloading. All lines are plotted for a single stage compressor.

The driver capacity line indicates the maximum allowable capacity for a given horsepower. The cylinder capacity lines represent the range of pressures calculated with all possible combinations of pockets open and cylinder unloading, as necessary, to cover the capacity of the driver.

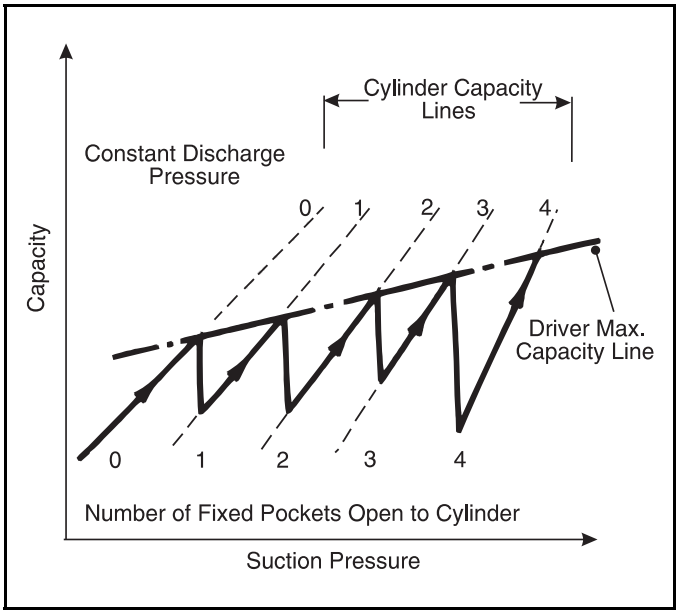
Starting at the end (line 0-0) with full cylinder capacity, the line is traced until it crosses the driver capacity line at which point it is dropped to the next largest cylinder capacity and follow until it crosses the driver line, etc. This will produce a "saw tooth" effect, hence the name "saw tooth" curve. The number of "teeth" depends upon the number of combinations of pockets (opened or closed) required for unloading.

The same method is followed for multi-stage units. For each additional stage another "saw tooth" curve must be constructed, i.e., for a two stage application, two curves are required to attain the final results.

Although control devices are often automatically operated, manual operation is satisfactory for many services. Where manual operation is provided, it often consists of a valve, or valves, to open and close clearance pockets. In some cases, a movable cylinder head is provided for variable clearance in the cylinder (Fig. 13-19).

FIG. 13-18

"Saw Tooth" Curve for Unloading Operation



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Gas Pulsation Control

Pulsation is inherent in reciprocating compressors because suction and discharge valves are open during only part of the stroke.

Pulsation must be damped (controlled) in order to:

a. provide smooth flow of gas to and from the compressor,

b. prevent overloading or underloading of the compressors, and

c. reduce overall vibration.

There are several types of pulsation chambers. The simplest one is a volume bottle, or a surge drum, which is a pressure vessel, unbaffled internally and mounted on or very near a cylinder inlet or outlet.

A manifold joining the inlet and discharge connections of cylinders operating in parallel can also serve as a volume bottle.

Performance of volume bottles is not normally guaranteed without an analysis of the piping system from the compressor to the first process vessel.

Volume bottles are sized empirically to provide an adequate volume to absorb most of the pulsation. Several industry methods were tried in an effort to produce a reasonable rule-of-thumb for their sizing. Fig. 13-20 may be used for approximate bottle sizing.

Example 13-3

Indicated suction pressure = 600 psia

Indicated discharge pressure = 1400 psia

Cylinder bore = 6 in

Cylinder stroke = 15 in

Swept volume = π (6^2/4) (15) = 424 cu in

From Fig. 13-25:

At 600 psi inlet pressure, the suction bottle multiplier is approximately 7.5. Suction-bottle volume = (7.5) (424) = 3,180 cu in.

NOTE: When more than one cylinder is connected to a bottle, the sum of the individual swept volumes is the size required for the common bottle.

FIG. 13-19

Sectional View of a Cylinder Equipped with a Hand-Operated Valve Lifter and Variable-Volume Clearance

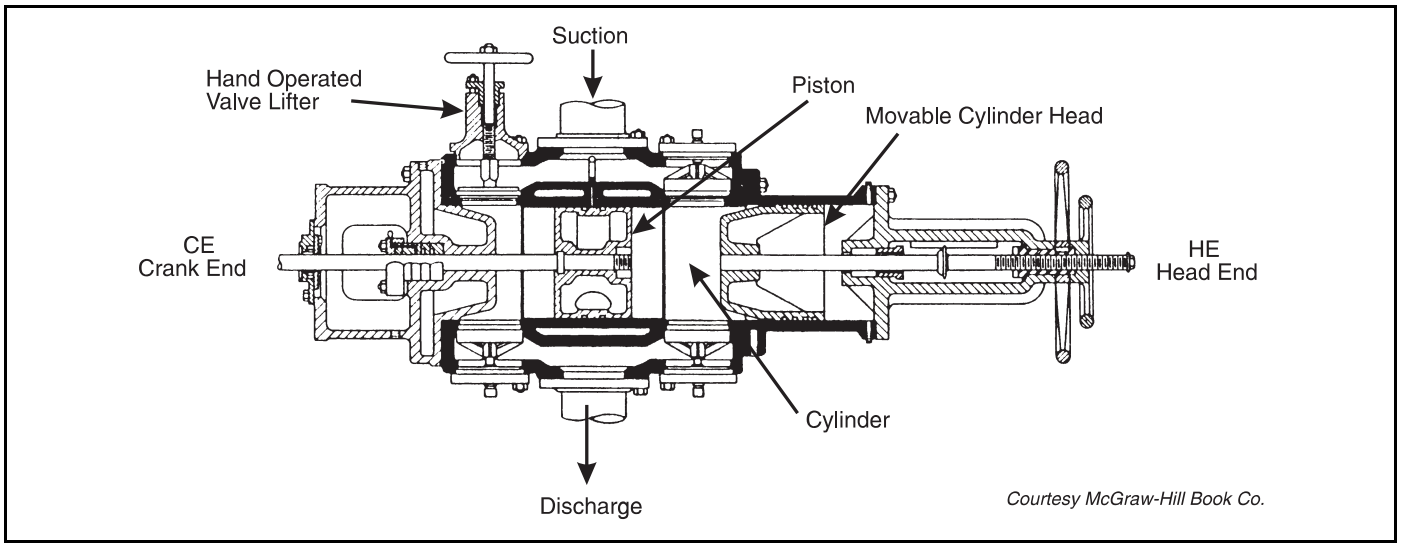
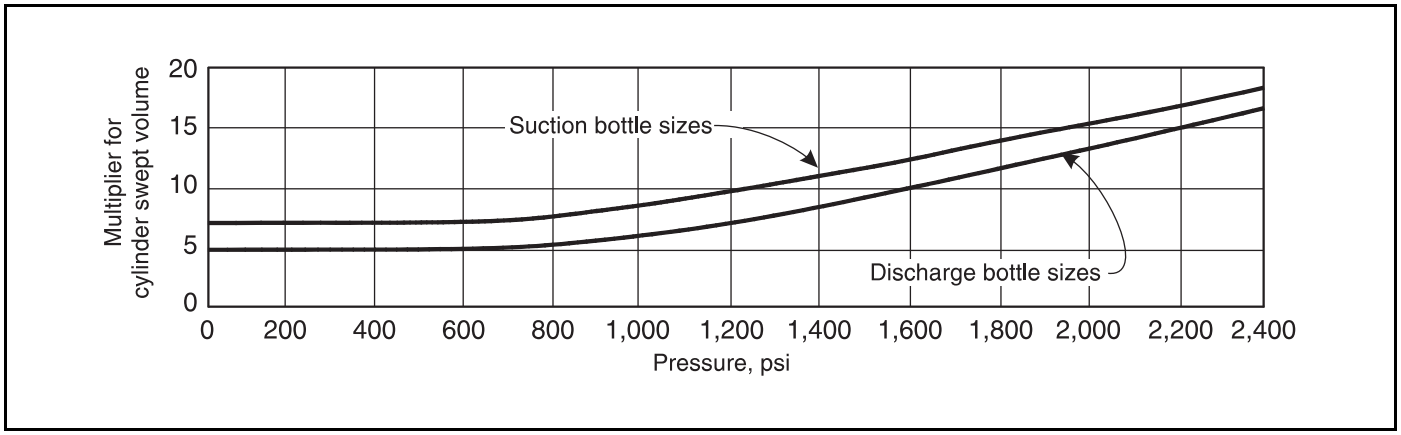


FIG. 13-20

Approximate Bottle Sizing Chart



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For more accurate sizing, compressor manufacturers can be consulted. Organizations which provide designs and/or equipment for gas-pulsation control are also available.

Having determined the necessary volume of the bottle, the proportioning of diameter and length to provide this volume requires some ingenuity and judgment. It is desirable that manifolds be as short and of as large diameter as is consistent with pressure conditions, space limitations, and appearance.

A good general rule is to make the manifold diameter 1-1/2 times the inside diameter of the largest cylinder connected to it, but this is not always practicable, particularly where large cylinders are involved.

Inside diameter of pipe must be used in figuring manifolds. This is particularly important in high-pressure work and in small sizes where wall thickness may be a considerable percentage of the cross sectional area. Minimum manifold length is determined from cylinder center distances and connecting pipe diameters. Some additions must be made to the minimum thus determined to allow for saddle reinforcements and for welding of caps.

It is customary to close the ends of manifolds with welding caps which add both volume and length. Fig. 13-21 gives approximate volume and length of standard caps.

Pulsation Dampeners (Snubbers)

A pulsation dampener is an internally-baffled device. The design of the pulsation dampening equipment is based on acoustical analog evaluation which takes into account the specified operating speed range, conditions of unloading, and variations in gas composition.

Analog evaluation is accomplished with an active analog that simulates the entire compressor, pulsation dampeners, piping and equipment system and considers dynamic interactions among these elements.

Pulsation dampeners also should be mounted as close as possible to the cylinder, and in large volume units, nozzles should be located near the center of the chamber to reduce unbalanced forces.

Pulsation dampeners are typically guaranteed for a maximum residual peak-to-peak pulsation pressure of 2% of average absolute pressure at the point of connection to the piping system, and pressure drop through the equipment of not more than 1% of the absolute pressure. This applies at design condition and not necessarily for other operating pressures and flows. A detailed discussion of recommended design approaches for pulsation suppression devices is presented in API Standard 618, Reciprocating Compressors for General Refinery Services.

As pressure vessels, all pulsation chambers (volume bottles and dampeners) are generally built to Section VIII of ASME Code and suitable for applicable cylinder relief valve set pressure.

Suction pulsation chambers are often designed for the same pressure as the discharge units, or for a minimum of 2/3 of the design discharge pressure.

Troubleshooting

Minor troubles can normally be expected at various times during routine operation of the compressor. These troubles are most often traced to dirt, liquid, and maladjustment, or to operating personnel being unfamiliar with functions of the various machine parts and systems. Difficulties of this type can usually be corrected by cleaning, proper adjustment, elimination of an adverse condition, or quick replacement of a relatively minor part.

Major trouble can usually be traced to long periods of operation with unsuitable coolant or lubrication, careless operation and routine maintenance, or the use of the machine on a service for which it was not intended.

A defective inlet valve can generally be found by feeling the valve cover. It will be much warmer than normal. Discharge valve leakage is not as easy to detect since the discharge is always hot. Experienced operators of water-cooled units can usually tell by feel if a particular valve is leaking. The best indication of discharge valve trouble is the discharge temperature. This will rise, sometimes rapidly, when a valve is in poor condition or breaks. This is one very good reason for keeping a record of the discharge temperature from each cylinder.

FIG. 13-21

Welding Caps

Pipe size	Standard weight		Extra strong		Double Extra strong	
	Volume, cu in.	Length, in.	Volume, cu in.	Length, in.	Volume, cu in.	Length, in.
4"	24.2	2 1/2	20.0	2 1/2	15	3
6"	77.3	3 1/2	65.7	3 1/2	48	4
8"	148.5	4 11/16	122.3	4 11/16	120	5
10"	295.6	5 3/4	264.4	5 3/4		
12"	517.0	6 7/8	475.0	6 7/8		
14"	684.6	7 13/16	640.0	7 13/16		
16"	967.6	9	911.0	9		
18"	1432.6	10 1/16	1363.0	10 1/16		
20"	2026.4	11 1/4	1938.0	11 1/4		
24"	3451.0	13 7/16	3313.0	13 7/16		

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Recording of the interstage pressure on multistage units is valuable because any variation, when operating at a given load point, indicates trouble in one or the other of the two stages. If the pressure drops, the trouble is in the low pressure cylinder. If it rises, the problem is in the high pressure cylinder.

Troubleshooting is largely a matter of elimination based on a thorough knowledge of the interrelated functions of the various parts and the effects of adverse conditions. A complete list of possible troubles with their causes and corrections is impractical, but the following list of the more frequently encountered troubles and their causes is offered as a guide (Fig. 13-22).

CENTRIFUGAL COMPRESSORS

This section is intended to supply information sufficiently accurate to determine whether a centrifugal compressor should be considered for a specific job. The secondary objective is to present information for evaluating compressor performance.

Fig. 13-23 gives an approximate idea of the flow range that a centrifugal compressor will handle. A multi-wheel (multistage) centrifugal compressor is normally considered for inlet volumes between 500 and 200,000 inlet acfm. A single-wheel (single stage) compressor would normally have application between 100 and 150,000 inlet acfm. A multiwheel compressor can be thought of as a series of single wheel compressors contained in a single casing.

Most centrifugal compressors operate at speeds of 3,000 rpm or higher, a limiting factor being impeller stress considerations as well as velocity limitation of 0.8 to 0.85 Mach number at the impeller tip and eye. Recent advances in machine design have resulted in production of some units running at speeds in excess of 40,000 rpm.

Figs. 13-24 through 13-26 provide cross-sectional drawings and identification of major components for typical centrifugal compressors.

Centrifugal compressors are usually driven by electric motors, steam or gas turbines (with or without speed-increasing gears), or turboexpanders.

There is an overlap of centrifugal and reciprocating compressors on the low end of the flow range, see Fig. 13-3. On the higher end of the flow range an overlap with the axial compressor exists. The extent of this overlap depends on a number of things. Before a technical decision could be reached as to the type of compressor that would be installed, the service, operational requirements, and economics would have to be considered.

Design requirements for centrifugal compressors are covered by API Standard 617.

Performance Calculations

The operating characteristics must be determined before an evaluation of compressor suitability for the application can be made. Fig. 13-27 gives a rough comparison of the characteristics of the axial, centrifugal, and reciprocating compressor.

The centrifugal compressor approximates the constant head-variable volume machine, while the reciprocating is a constant volume-variable head machine. The axial compressor, which is a low head, high flow machine, falls somewhere in between. A compressor is a part of the system, and its performance is dictated by the system resistance. The desired system capability or objective must be determined before a compressor can be selected.

Fig. 13-28 is a typical performance map which shows the basic shape of performance curves for a variable-speed centrifugal compressor. The curves are affected by many variables, such as desired compression ratio, type of gas, number of wheels, sizing of compressor, etc.

With variable speed, the centrifugal compressor can deliver constant capacity at variable pressure, variable capacity at constant pressure, or a combination variable capacity and variable pressure.

Basically the performance of the centrifugal compressor, at speeds other than design, is such that the capacity will vary directly as the speed, the head developed as the square of the speed, and the required horsepower as the cube of the speed. As the speed deviates from the design speed, the error of these rules, known as the affinity laws, or fan laws, increases. The fan laws only apply to single stages or multi-stages with very low compression ratios or very low Mach numbers.

Fan Laws:

$$Q \propto N;\ \text{i.e.,}\ \frac{Q_{110}}{N_{110}} = \frac{Q_{100}}{N_{100}} = \frac{Q_{90}}{N_{90}} \qquad \textbf{Eq 13-22}$$

$$H \propto N^2;\ \text{i.e.,}\ \frac{H_{110}}{(N_{110})^2} = \frac{H_{100}}{(N_{100})^2} = \frac{H_{90}}{(N_{90})^2} \qquad \textbf{Eq 13-23}$$

$$Bhp \propto N^3;\ \text{i.e.,}\ \frac{Bhp_{110}}{(N_{110})^3} = \frac{Bhp_{100}}{(N_{100})^3} = \frac{Bhp_{90}}{(N_{90})^3} \qquad \textbf{Eq 13-24}$$

By varying speed, the centrifugal compressor will meet any load and pressure condition demanded by the process system within the operating limits of the compressor and the driver. It normally accomplishes this as efficiently as possible, since only the head required by the process is developed by the compressor. This compares to the essentially constant head developed by the constant speed compressor.

Fig. 13-28 depicts typical performance curves with a small compression ratio. The system resistance has been superimposed on the chart: Line A represents typical system resistance of a closed system, such as a refrigeration unit where there is a relatively constant discharge pressure. Line B is an open-end system, such as pipeline application where pressure increases with capacity.

Fig. 13-29 shows a higher compression ratio. The range of stable operation is reduced because of the larger compression ratio. This is indicated by the surge line in Fig. 13-29 being further to the right than in Fig. 13-28.

Estimating Performance

Figs. 13-30 through 13-37 may be used for estimating compressor performance. These curves are suitable for estimating only and are not intended to take the place of a “wheel-by-wheel” selection by the compressor manufacturer, nor should the curves be used to calculate performance using field data in an attempt to determine a variance from predicted performance based on manufacturer’s data. Fig. 13-30 is used to convert scfm to icfm. All centrifugal compressors are based on flows that are converted to inlet or actual cubic feet per minute. This is done because the centrifugal wheel is sensitive to inlet volume, compression ratio (i.e., head), and specific speed.

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FIG. 13-22

Probable Causes of Reciprocating Compressor Trouble

Trouble	Probable Cause(s)
Compressor Will not Start	1. Power supply failure. 2. Switchgear or starting panel. 3. Low oil pressure shut down switch. 4. Control panel.
Motor Will Not Synchronize	1. Low voltage. 2. Excessive starting torque. 3. Incorrect power factor. 4. Excitation voltage failure.
Low Oil Pressure	1. Oil pump failure. 2. Oil foaming from counterweights striking oil surface. 3. Cold oil. 4. Dirty oil filter. 5. Interior frame oil leaks. 6. Excessive leakage at bearing shim tabs and/or bearings. 7. Improper low oil pressure switch setting. 8. Low gear oil pump by-pass/relief valve setting. 9. Defective pressure gauge. 10. Plugged oil sump strainer. 11. Defective oil relief valve.
Noise In Cylinder	1. Loose piston. 2. Piston hitting outer head or frame end of cylinder. 3. Loose crosshead lock nut. 4. Broken or leaking valve(s). 5. Worn or broken piston rings or expanders. 6. Valve improperly seated/damaged seat gasket. 7. Free air unloader plunger chattering.
Excessive Packing Leakage	1. Worn packing rings. 2. Improper lube oil and/or insufficient lube rate (blue rings). 3. Dirt in packing. 4. Excessive rate of pressure increase. 5. Packing rings assembled incorrectly. 6. Improper ring side or end gap clearance. 7. Plugged packing vent system. 8. Scored piston rod. 9. Excessive piston rod run-out.

Trouble	Probable Cause(s)
Packing Over-Heating	1. Lubrication failure. 2. Improper lube oil and/or insufficient lube rate. 3. Insufficient cooling.
Excessive Carbon On Valves	1. Excessive lube oil. 2. Improper lube oil (too light, high carbon residue). 3. Oil carryover from inlet system or previous stage. 4. Broken or leaking valves causing high temperature. 5. Excessive temperature due to high pressure ratio across cylinders.
Relief Valve Popping	1. Faulty relief valve. 2. Leaking suction valves or rings on next higher stage. 3. Obstruction (foreign material, rags), blind or valve closed in discharge line.
High Discharge Temperature	1. Excessive ratio on cylinder due to leaking inlet valves or rings on next higher stage. 2. Fouled intercooler/piping. 3. Leaking discharge valves or piston rings. 4. High inlet temperature. 5. Fouled water jackets on cylinder. 6. Improper lube oil and/or lube rate.
Frame Knocks	1. Loose crosshead pin, pin caps or crosshead shoes. 2. Loose/worn main, crankpin or crosshead bearings. 3. Low oil pressure. 4. Cold oil. 5. Incorrect oil. 6. Knock is actually from cylinder end.
Crankshaft Oil Seal Leaks	1. Faulty seal installation. 2. Clogged drain hole.
Piston Rod Oil Scraper Leaks	1. Worn scraper rings. 2. Scrapers incorrectly assembled. 3. Worn/scored rod. 4. Improper fit of rings to rod/side clearance.

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FIG. 13-23

Centrifugal Compressor Flow Range

Nominal flow range (inlet acfm)	Average polytropic efficiency	Average isentropic efficiency	Speed to develop 10,000 ft head/wheel
100- 500	0.70	0.67	20,500
500- 7,500	0.80	0.78	10,500
7,500- 20,000	0.86	0.83	8,200
20,000- 33,000	0.86	0.83	6,500
33,000- 55,000	0.86	0.83	4,900
55,000- 80,000	0.86	0.83	4,300
80,000 -115,000	0.86	0.83	3,600
115,000 -145,000	0.86	0.83	2,800
145,000 -200,000	0.86	0.83	2,500

Fig. 13-31 is a useful curve to find inlet (actual) cfm when the weight flow in lb/min is known. Actual cfm and inlet cfm both denote the gas at suction conditions. These terms are often used interchangeably. This curve can be used in reverse to determine mass flow.

Fig. 13-32 is used to determine the approximate discharge temperature that is produced by the compression ratio. Discharge temperatures above the 400°F range should be checked since mechanical problems as well as safety problems may exist. This curve includes compressor efficiencies in the range of 60 to 75%.

Example 13-4 — Given: $r = 10.0$; $Q_1 = 10{,}000$ icfm

$k = 1.15$; $t_1 = 0°C$

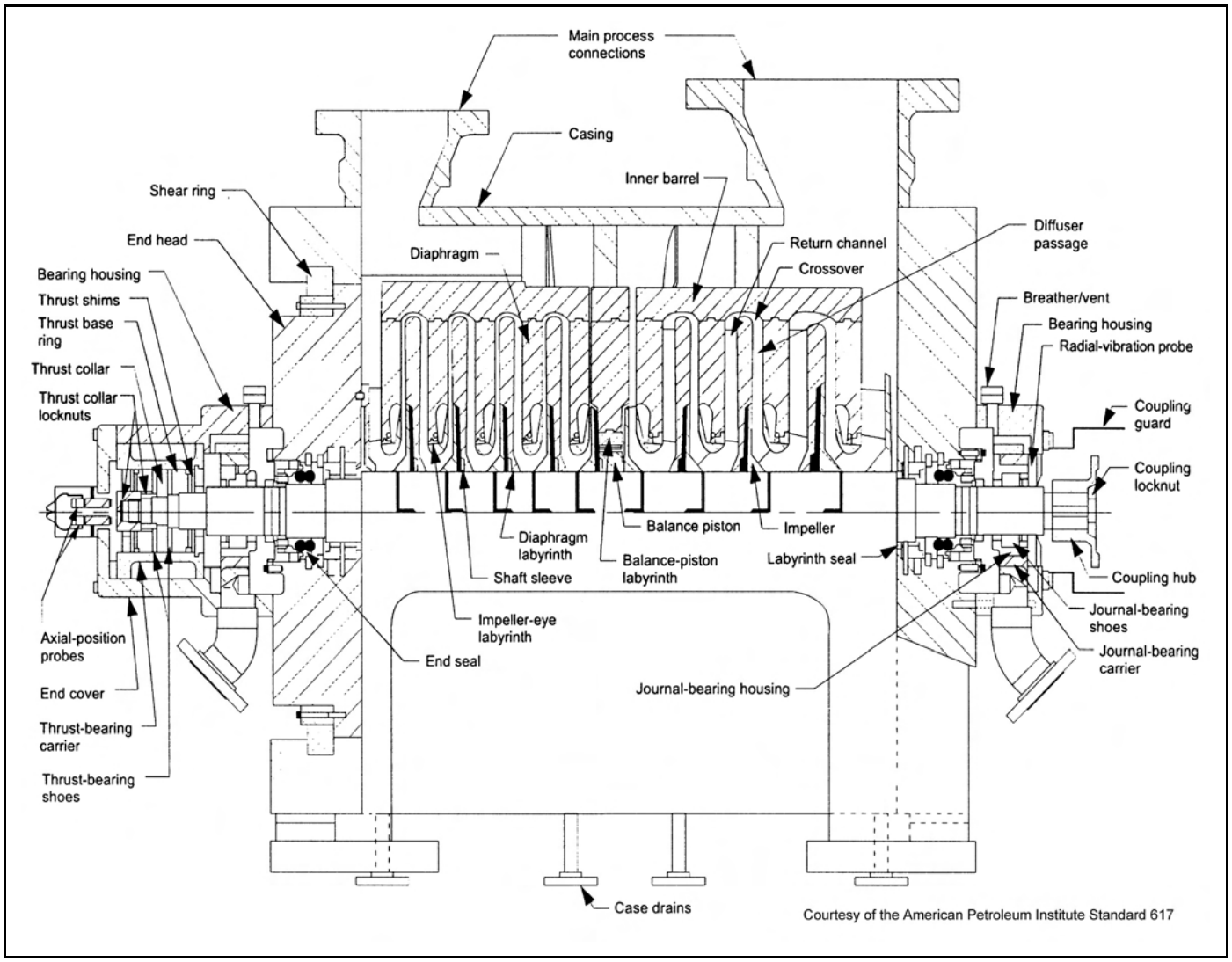
Find: Discharge temperature

Answer: $t_2 = 230°F$ (approximately) from Fig. 13-32.

Fig. 13-34 gives the approximate horsepower required for the compression. It includes compressor efficiencies in the range of 60 to 70%.

FIG. 13-24

Typical Centrifugal Compressor Showing Nomenclature of Key Parts



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Example 13-5 — Given: Weight flow, w, = 1,000 lb/min

head = 70 000 ft-lb/lb

Find: Horsepower

Answer: Ghp = 3,000 from Fig. 13-34.

Fig. 13-37 predicts the approximate number of compressor wheels required to produce the head. If the number of wheels is not a whole number, use the next highest number.

Calculating Performance

When more accurate information is required for compressor head, gas horsepower, and discharge temperature, the equations in this section should be used. This method applies to a gas mixture for which a P-H diagram chart is not available. To calculate the properties of the gas, see Figs. 13-6 and 13-7. All values for pressure and temperature in these calculation procedures are the absolute values. Unless otherwise specified, volumes of flow in this section are actual volumes.

To calculate the inlet volume:

$$Q = \frac{(w)\,(1{,}545)\,(T_1)\,(Z_1)}{(MW)\,(P_1)\,(144)} \qquad \textbf{Eq 13-25}$$

If we assume the compression to be isentropic (reversible adiabatic, constant entropy), then:

$$H_{is} = \frac{ZRT}{MW\,(k-1)/k}\left[\left(\frac{P_2}{P_1}\right)^{(k-1)/k} - 1\right] \qquad \textbf{Eq 13-26}$$

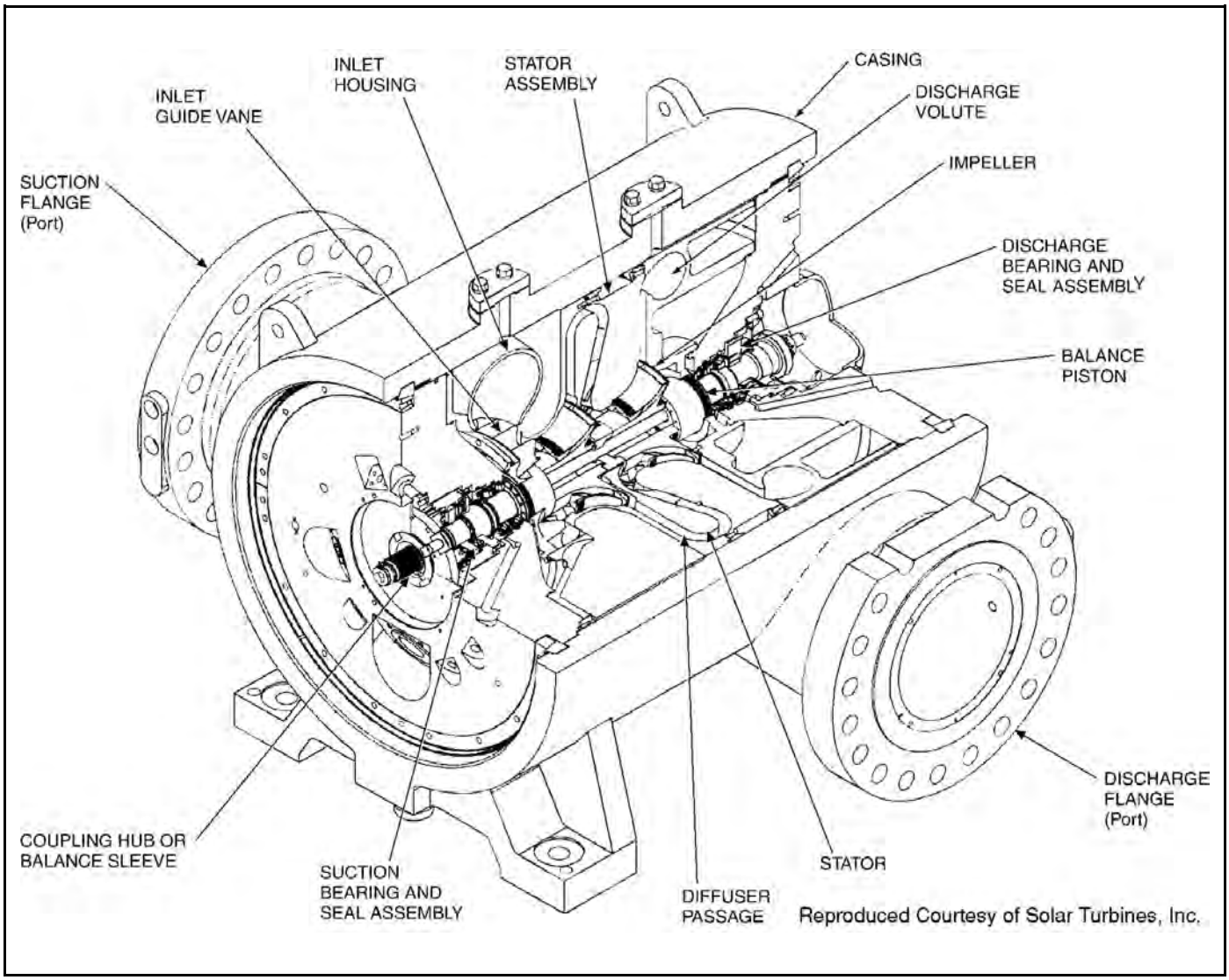
Since these calculations will not be wheel-by-wheel, the head will be calculated across the entire machine. For this, use the average compressibility factor:

$$Z_{avg} = \frac{Z_1 + Z_2}{2}$$

The heat capacity ratio, k, is normally determined at the average suction and discharge temperature (see Figs. 13-7 and 13-8).

FIG. 13-25

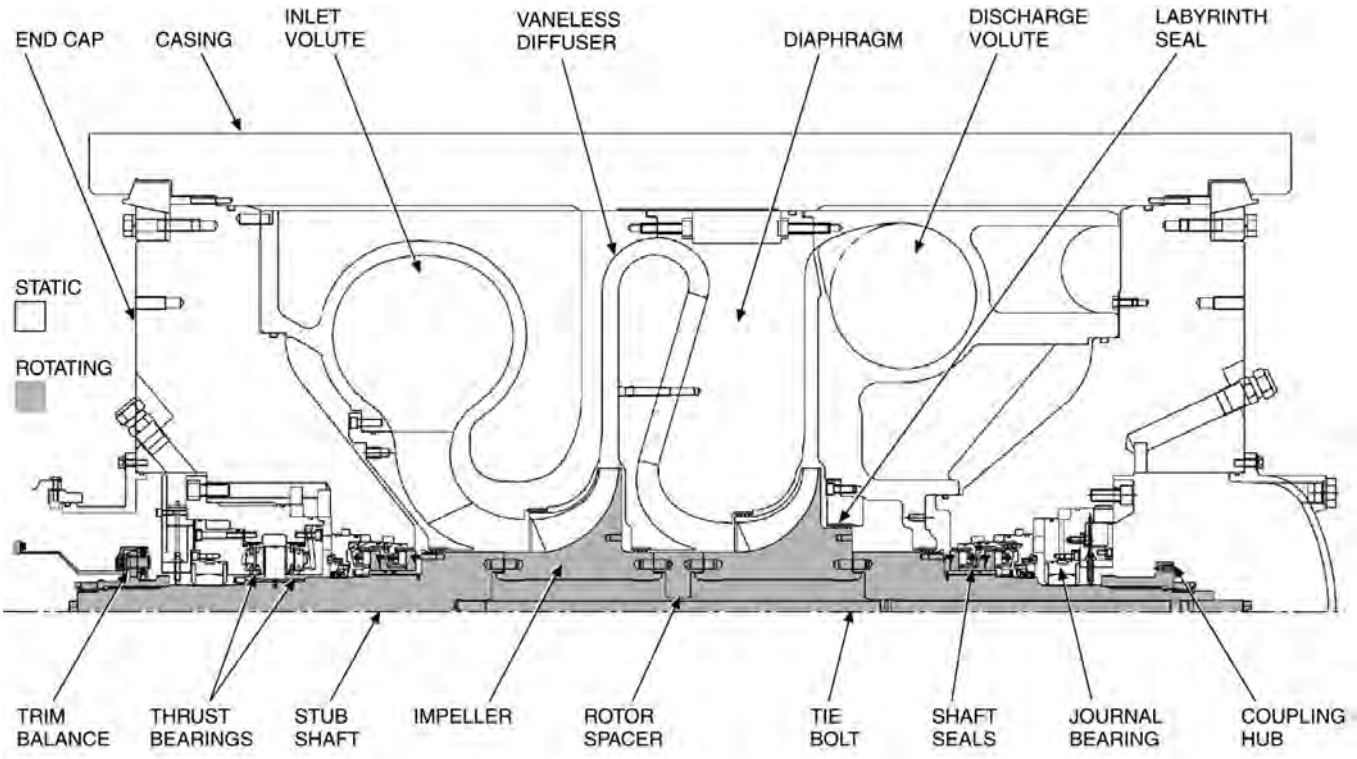
Typical Centrifugal Compressor Cutaway

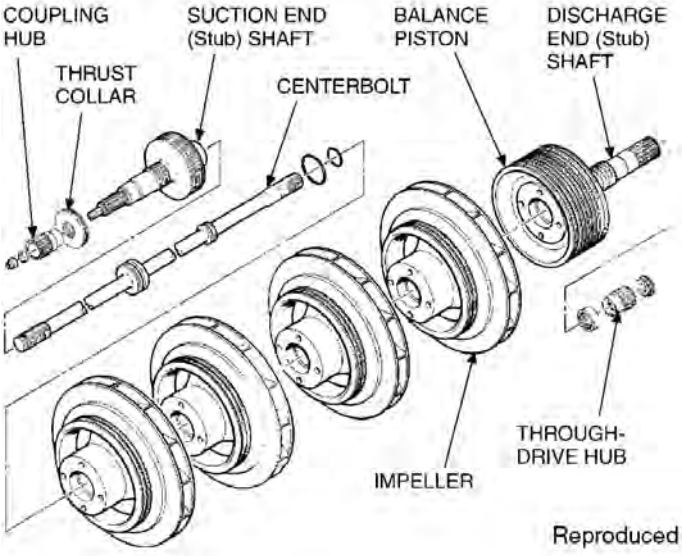


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FIG. 13-26

Centrifugal Compressor Cross Section



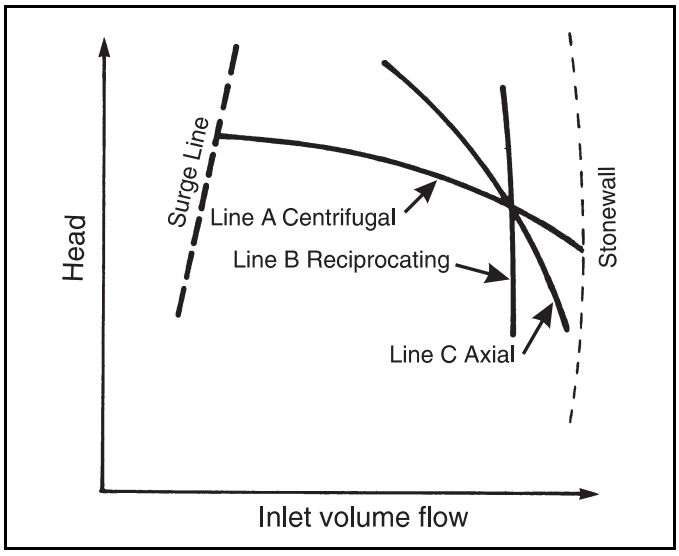


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FIG. 13-27

Compressor Head



Isentropic Calculation

To calculate the head:

$$H_{is} = \frac{Z_{avg} R T_1}{MW\,(k-1)/k}\left[\left(\frac{P_2}{P_1}\right)^{(k-1)/k} - 1\right] \qquad \textbf{Eq 13-27a}$$

which can also be written in the form:

$$H_{is} = \frac{1545}{MW}\,\frac{Z_{avg} T_1}{(k-1)/k}\left[\left(\frac{P_2}{P_1}\right)^{(k-1)/k} - 1\right] \qquad \textbf{Eq 13-27b}$$

The gas horsepower can now be calculated from:

$$Ghp = \frac{(w)\,(H_{is})}{(\eta_{is})\,(33{,}000)} \qquad \textbf{Eq 13-28}$$

The approximate theoretical discharge temperature can be calculated from:

$$\Delta T_{ideal} = T_1\left[\left(\frac{P_2}{P_1}\right)^{(k-1)/k} - 1\right] \qquad \textbf{Eq 13-29}$$

$$T_2 = T_1 + \Delta T_{ideal} \qquad \textbf{Eq 13-30}$$

FIG. 13-28

Compressor Performance, Low Compression Ratio

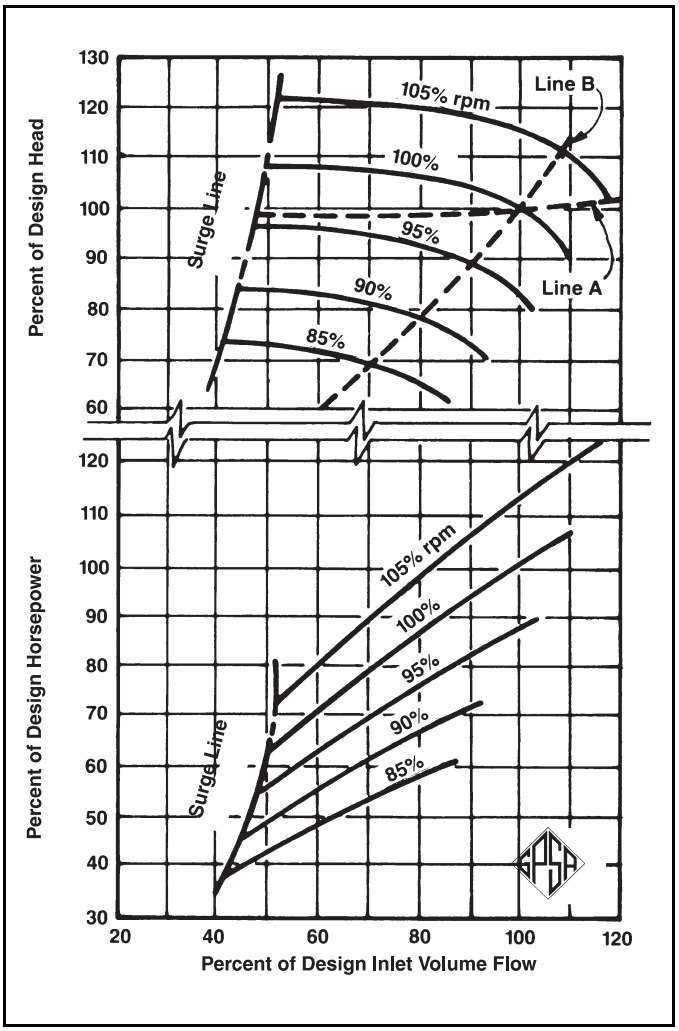
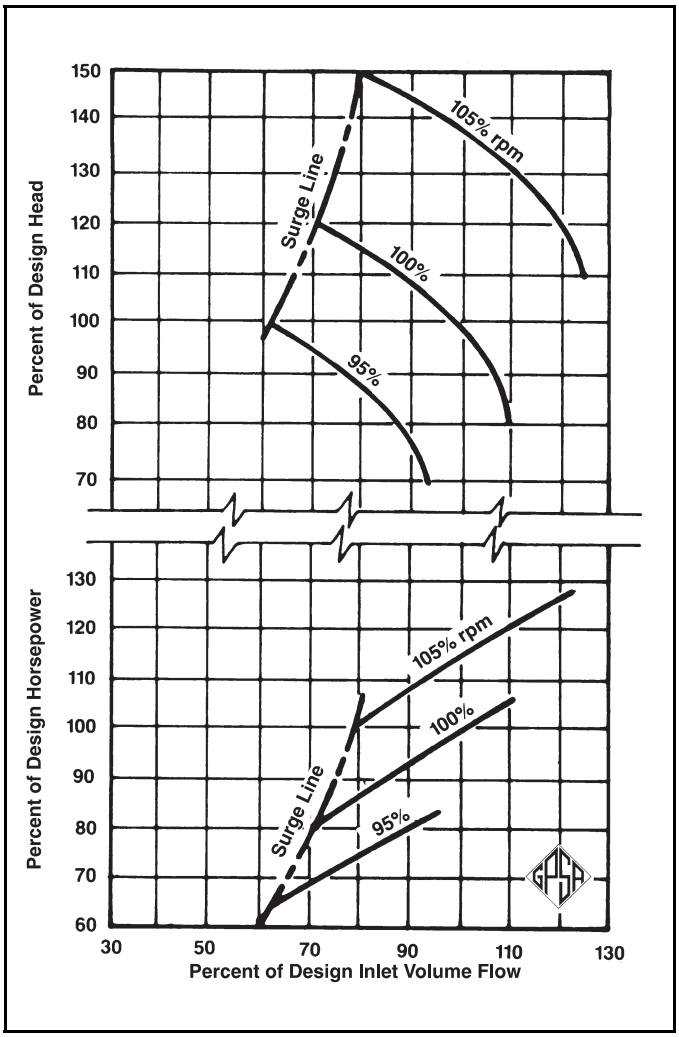


FIG. 13-29

Compressor Performance, Higher Compression Ratio

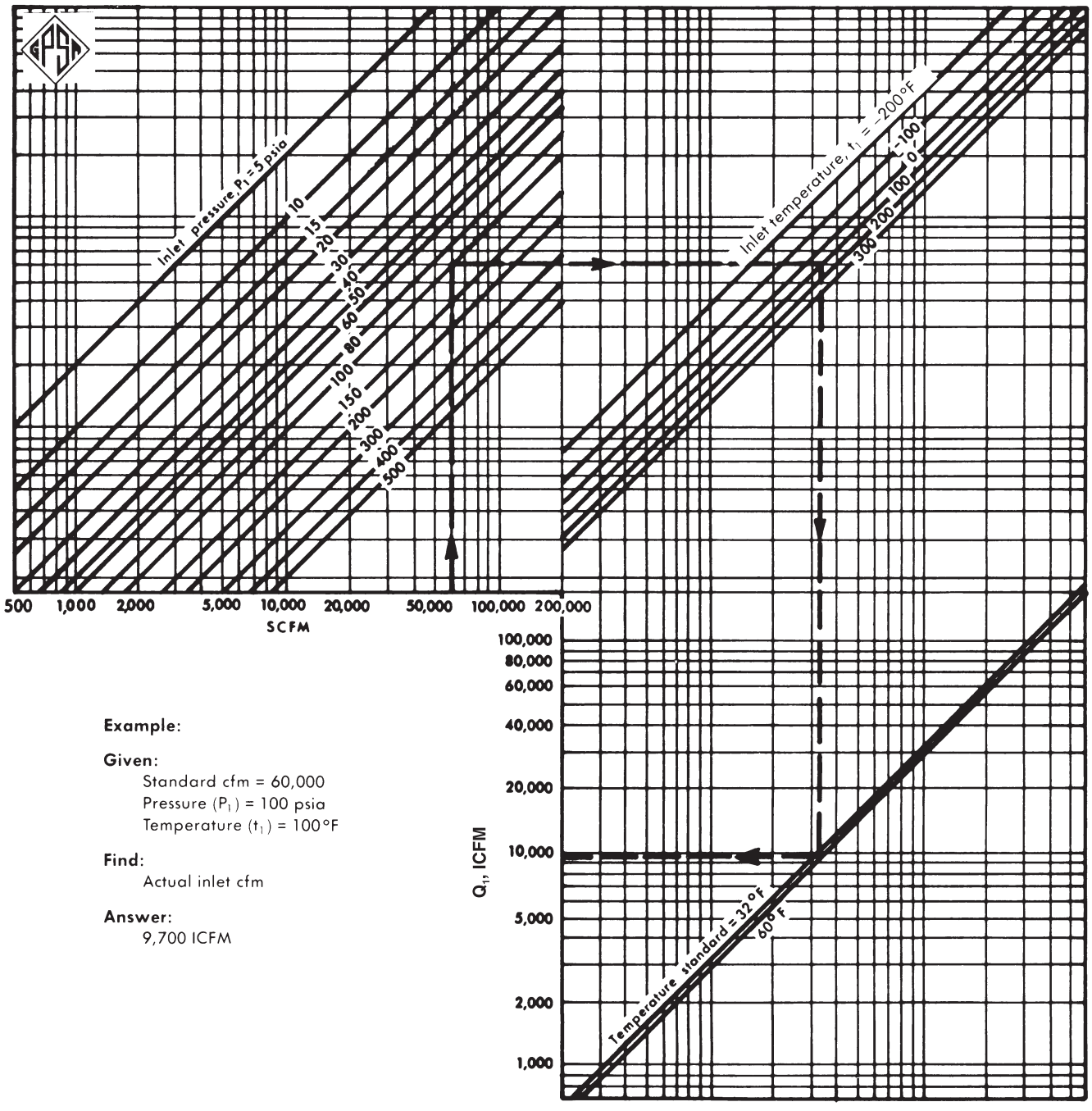


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FIG. 13-30

ICFM to SCFM

Z = 1



Example:

Given:

Standard cfm = 60,000
Pressure (P_1) = 100 psia
Temperature (t_1) = 100°F

Find:

Actual inlet cfm

Answer:

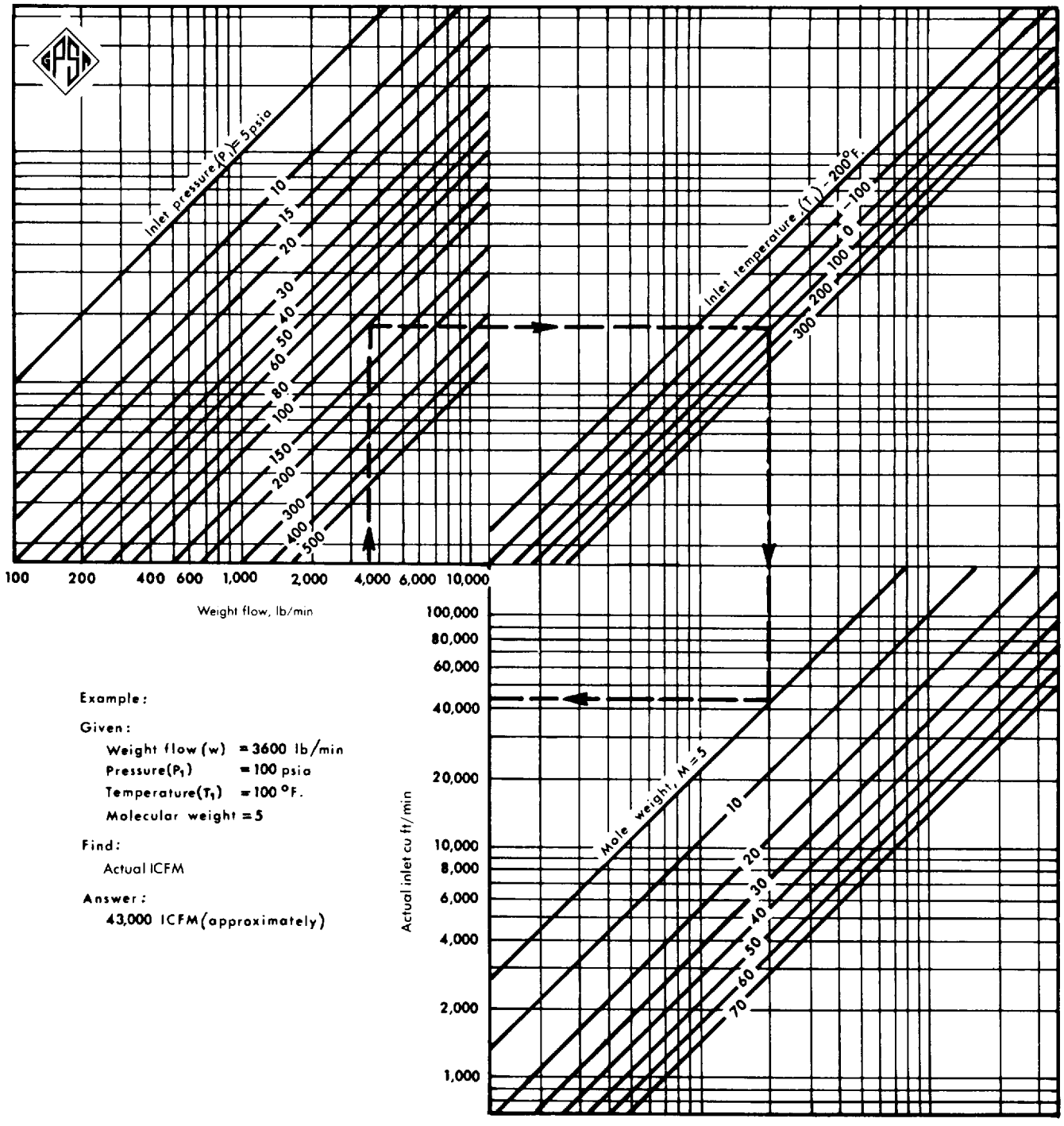
9,700 ICFM

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FIG. 13-31

Mass Flow to Inlet Volume Flow

Z = 1



Example:

Given:

Weight flow (w) = 3600 lb/min

Pressure(P_1) = 100 psia

Temperature(T_1) = 100 °F.

Molecular weight = 5

Find:

Actual ICFM

Answer:

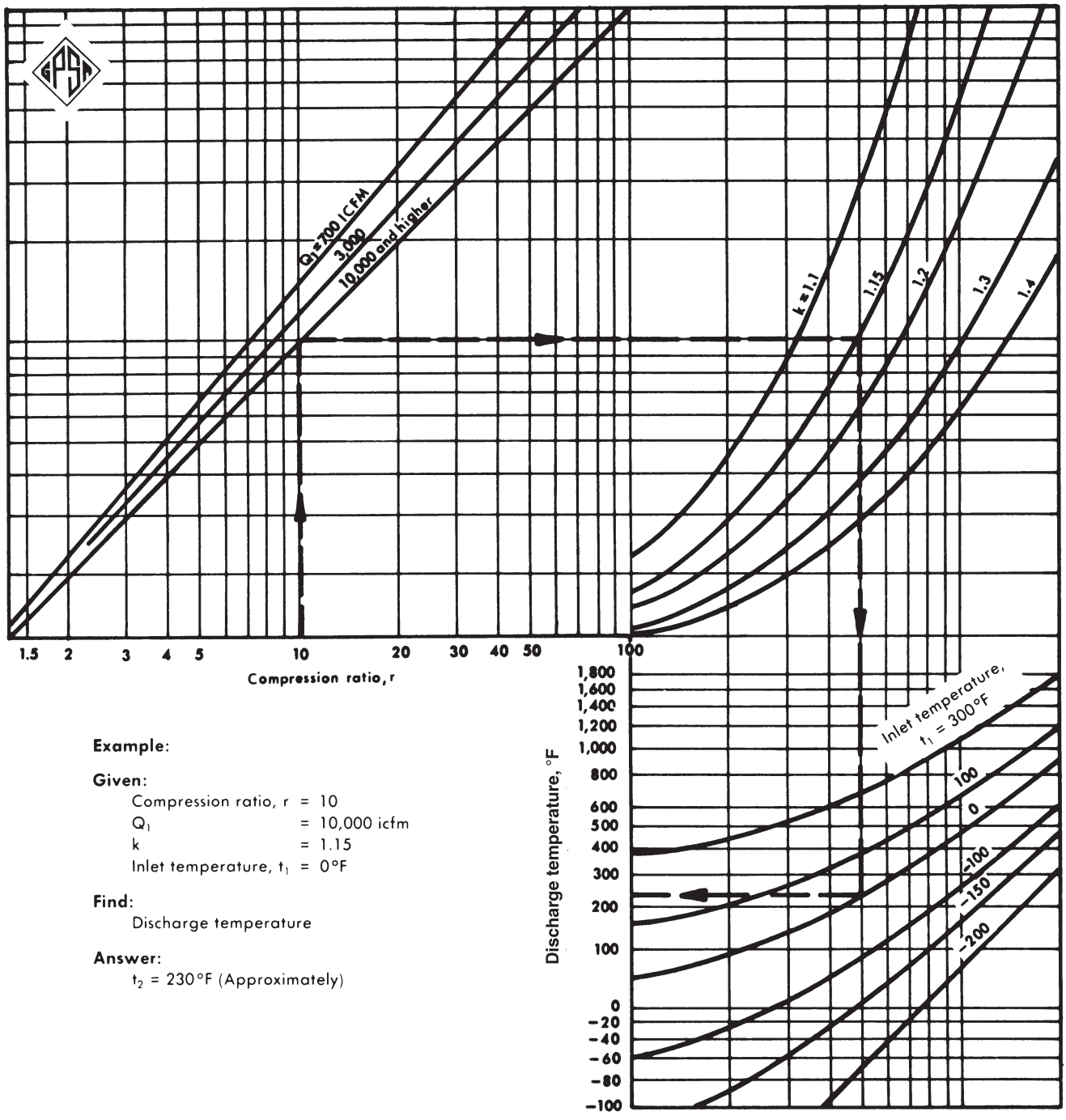
43,000 ICFM (approximately)

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FIG. 13-32

Discharge Temperature

Z = 1



Example:

Given:

Compression ratio, r = 10
Q_1 = 10,000 icfm
k = 1.15
Inlet temperature, t_1 = 0°F

Find:

Discharge temperature

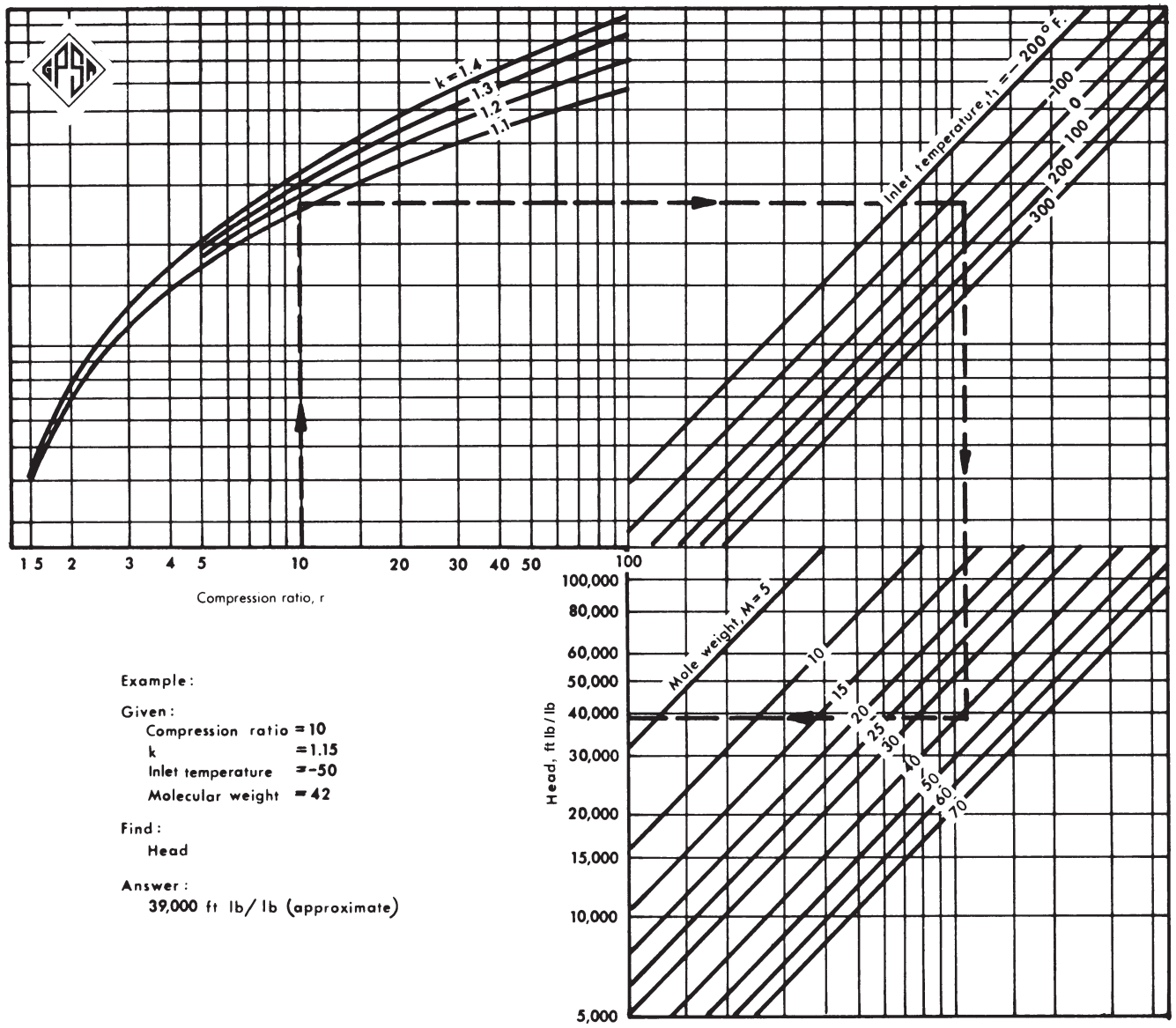
Answer:

t_2 = 230°F (Approximately)

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FIG. 13-33

Head
Z = 1



Example:

Given:
- Compression ratio = 10
- k = 1.15
- Inlet temperature = –50
- Molecular weight = 42

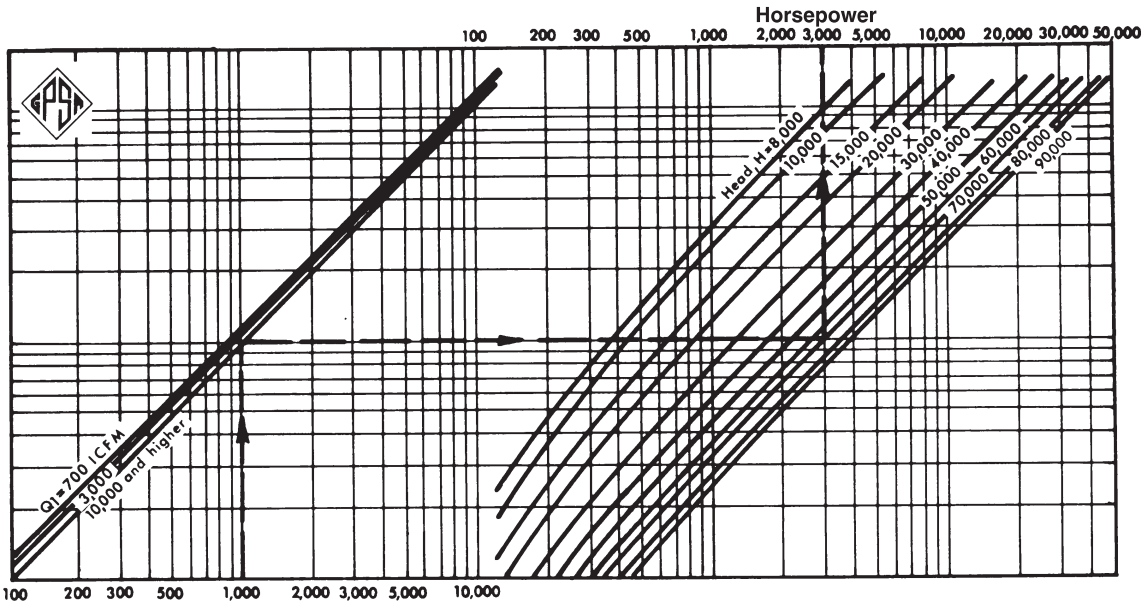
Find:
- Head

Answer:
- 39,000 ft lb/ lb (approximate)

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FIG. 13-34

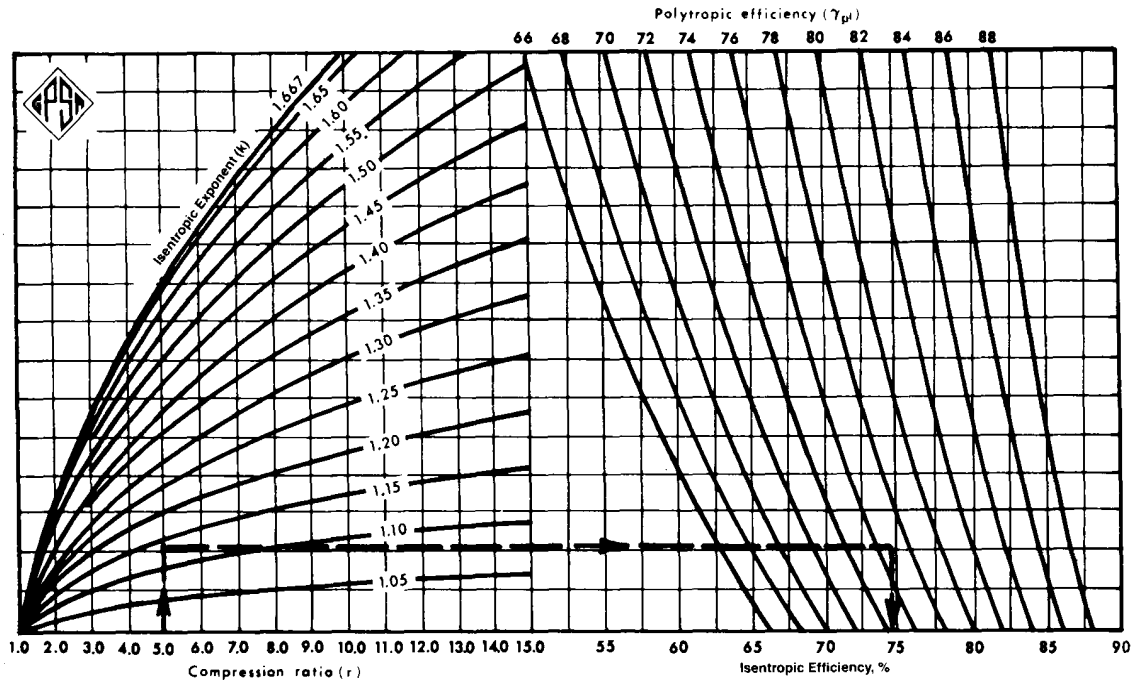
Horsepower Determination



Example: **Given:** Weight flow, w = 1,000 lb/min; Head = 70,000 ft-lb/lb — **Find:** Horsepower — **Answer:** Hp = 3,000

FIG. 13-35

Efficiency Conversion



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The actual discharge temperature can be approximated:

$$\Delta T_{actual} = T_1 \frac{\left[\left(\frac{P_2}{P_1}\right)^{(k-1)/k} - 1\right]}{\eta_{is}} \qquad \textbf{Eq 13-31}$$

$$T_2 = T_1 + \Delta T_{actual} \qquad \textbf{Eq 13-32}$$

Polytropic Calculation

Sometimes compressor manufacturers use a polytropic path instead of isentropic. Polytropic efficiency is defined by:

$$\frac{n}{(n-1)} = \left[\frac{k}{(k-1)}\right]\eta_p \qquad \textbf{Eq 13-33}$$

(See Fig. 13-35 for conversion of isentropic efficiency to polytropic efficiency.)

The equations for head and gas horsepower based upon polytropic compression are:

$$H_p = \frac{Z_{avg}RT_1}{MW\,(n-1)/n}\left[\left(\frac{P_2}{P_1}\right)^{(n-1)/n} - 1\right] \qquad \textbf{Eq 13-34a}$$

which also can be written in the form:

$$H_p = \frac{1545}{MW}\,\frac{Z_{avg}T_1}{(n-1)/n}\left[\left(\frac{P_2}{P_1}\right)^{(n-1)/n} - 1\right] \qquad \textbf{Eq 13-34b}$$

$$Ghp = \frac{(w)\,(H_p)}{(\eta_p)\,(33{,}000)} \qquad \textbf{Eq 13-35}$$

Polytropic and isentropic head are related by

$$H_p = \frac{H_{is}\,\eta_p}{\eta_{is}} \qquad \textbf{Eq 13-36}$$

The approximate actual discharge temperature can be calculated from:

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{\left(\frac{n-1}{n}\right)} \qquad \textbf{Eq 13-37}$$

Mechanical Losses

After the gas horsepower has been determined by either method, horsepower losses due to friction in bearings, seals, and speed increasing gears must be added.

Fig. 13-36 shows losses related to the shaft speed and casing size for conventional multistage units.

Bearings and seal losses can also be roughly computed from Scheel's equation:

$$\text{Mechanical losses} = (Ghp)^{0.4} \qquad \textbf{Eq 13-38}$$

To calculate the brake horsepower:

$$Bhp = Ghp + \text{mechanical losses} \qquad \textbf{Eq 13-39}$$

Compressor Speed

The basic equation for estimating the speed of a centrifugal compressor is:

$$N = (N_{nominal})\sqrt{\frac{H\,total}{(No.\,of\,wheels)\,(H\,max/\,wheel)}} \qquad \textbf{Eq 13-40}$$

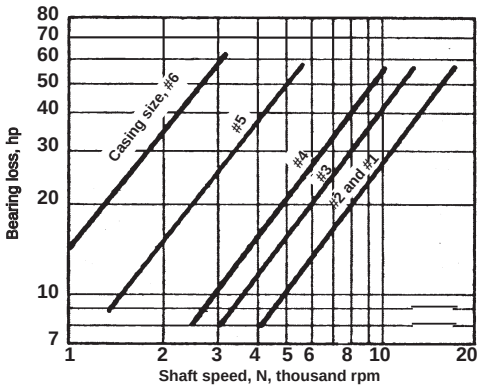
where the number of wheels is determined from Fig. 13-37.

FIG. 13-36

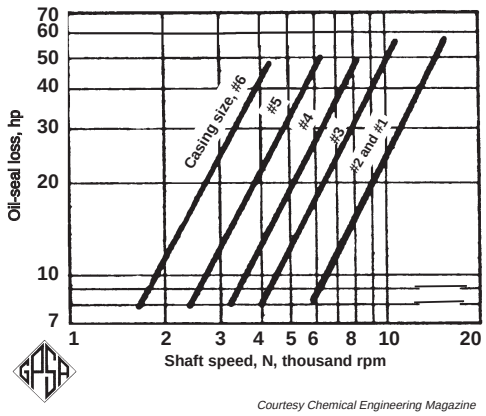
Mechanical Losses

Casing Size	Max Flow (inlet acfm)	Nominal Speed (rpm)
1	7,500	10,500
2	20,000	8,200
3	33,000	6,400
4	55,000	4,900
5	115,000	3,600
6	150,000	2,800

a. Bearing horsepower losses



b. Oil-seal horsepower losses



Courtesy Chemical Engineering Magazine

Nominal speeds to develop 10,000 feet of head/wheel can be determined from Fig. 13-23. However, to calculate the maximum head per wheel, the following equation based on molecular weight (or more accurately, density) can be used.

$$H\ max/stage = 15{,}000 - 1{,}500\,(MW)^{0.35} \qquad \textbf{Eq 13-41}$$

This equation will give a head of 10,000 ft for a gas when MW = 30 and 11,000 ft when MW = 16.

P-H Diagram

When a P-H diagram is available for the gas to be compressed, the following procedure should be used. Fig. 13-38 represents a section of a typical P-H diagram.

For the given inlet conditions, the enthalpy can be shown as point 1 on the P-H diagram. Starting from Point 1 follow the line of constant entropy to the required discharge pressure

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(P_2), locating the isentropic discharge state point (2_{is}). With these two points located the differential isentropic enthalpy can be calculated from the following equation:

$$\Delta h_{is} = h_{2is} - h_1 \qquad \textbf{Eq 13-42}$$

To convert to isentropic head, the equation is:

$$H_{is} = \Delta h_{is} (778 \text{ ft} \cdot \text{lb/Btu}) \qquad \textbf{Eq 13-43}$$

To find the discharge enthalpy:

$$h_2 = \frac{\Delta h_{is}}{\eta_{is}} + h_1 \qquad \textbf{Eq 13-44}$$

The actual discharge temperature can now be obtained from the P-H diagram. The gas horsepower can be calculated using Eq 13-28 and Eq 13-35.

FIG. 13-37

Wheels Required

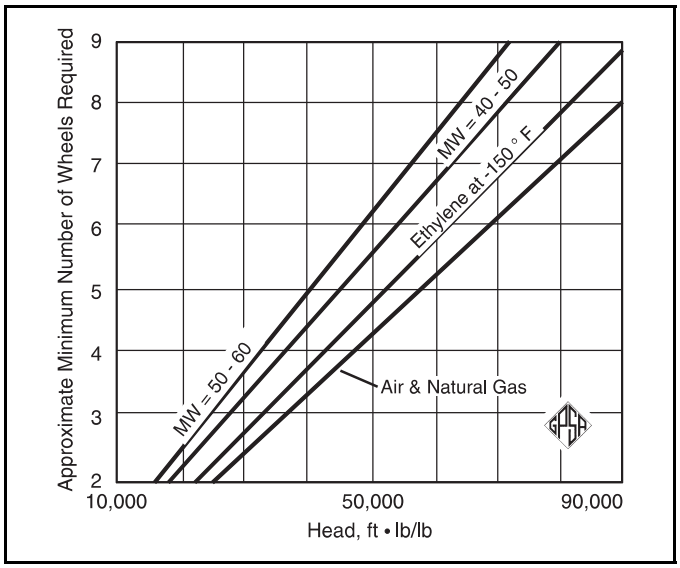
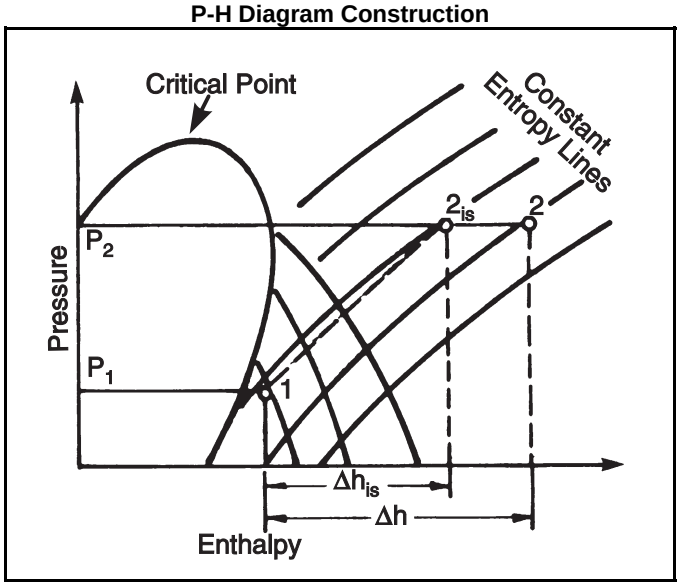


FIG. 13-38

P-H Diagram Construction



From Fig. 13-37 and Eqs. 13-40 and 13-41, the speed and number of wheels can be estimated.

To convert to polytropic head it will be necessary to assume a polytropic efficiency. See Fig. 13-23 for an efficiency corresponding to the inlet flow. Fig. 13-35 will give a corresponding adiabatic efficiency. The polytropic head may now be determined from Eq. 13-36.

When a P-H diagram is available, it is the fastest and most accurate method of determining compressor horsepower and discharge temperature.

Centrifugal Refrigeration Compressors

Compression ratio per wheel will vary on the order of 1.5 to 2.75 per wheel depending on the refrigerant and speed.

Due to the ease of applying external side loads to centrifugal machines, it is quite common to flash refrigerant from the condenser en route to the evaporators and/or to accept side loads from product being cooled by refrigerant at higher pressures than the lowest evaporator level.

Since side-loading is the practice rather than the exception, it is common to let the centrifugal compressor manufacturer obtain the desired performance characteristics from the following data: evaporator temperature levels; refrigeration loads required in MMBtu/hr; heat rejection medium (air or water); and type of driver.

Refrigeration compressors are also discussed in Section 14.

GENERAL

Flow Limits

Two conditions associated with centrifugal compressors are surge (pumping) and stone-wall (choked flow).

At some point on the compressor's operating curve there exists a condition of minimum flow/maximum head where the developed head is insufficient to overcome the system resistance. This is the surge point. When the compressor reaches this point, the gas in the discharge piping back-flows into the compressor. Without discharge flow, discharge pressure drops until it is within the compressor's capability, only to repeat the cycle.

The repeated pressure oscillations at the surge point should be avoided since it can be detrimental to the compressor. Surging can cause the compressor to overheat to the point the maximum allowable temperature of the unit is exceeded. Also, surging can cause damage to the thrust bearing due to the rotor shifting back and forth from the active to the inactive side.

"Stonewall" or choked flow occurs when sonic velocity is reached at any point in the compressor. When this point is reached for a given gas, the flow through the compressor cannot be increased further without internal modifications.

Interstage Cooling

Multistage compressors rely on intercooling whenever the inlet temperature of the gas and the required compression ratio are such that the discharge temperature of the gas exceeds about 300°F.

There are certain processes that require a controlled discharge temperature. For example, the compression of gases

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such as oxygen, chlorine, and acetylene requires that the temperature be maintained below 200°F.

The thermal stress within the horizontal bolted joint is the governing design limitation in a horizontally split compressor case. The vertically split barrel-type case, however, is free from the thermal stress complication.

Substantial power economy can be gained by precooling the gas before it enters the interstage impellers. Performance calculations indicate that the head and the horsepower are directly proportional to the absolute gas temperature at each impeller.

The gas may be cooled within the casing or in external heat exchangers.

Two methods of cooling within the casing are used — water cooled diaphragms between successive stages and direct liquid injection into the gas.

Diaphragm cooling systems include high-velocity water circulation through cast jackets in the diffuser diaphragms. The diaphragm coolers are usually connected in series.

Liquid injection cooling is the least costly means of controlling discharge temperatures. It involves injecting and atomizing a jet of water or a compatible liquid into the return channels. In refrigeration units, liquid refrigerant is frequently used for this purpose. Injected liquid also functions as a solvent in washing the impellers free of deposits. Nevertheless, the hazards of corrosion, erosion, and flooding present certain problems resulting in possible replacement of the compressor rotor.

External intercoolers are commonly used as the most effective means of controlling discharge temperatures. The gas is discharged from the compressor casing after one or more stages of compression and, after being cooled, is returned to the next stage or series of stages for further compression.

Intercoolers usually are mounted separately. When there are two or more compressor casings installed in series, individual machines may or may not be cooled or have intercoolers. In some cases, it may be advantageous to use an external cooler to precool gas ahead of the first wheel.

Journal and Thrust Bearings

Radial journal bearings are designed to handle high speeds and heavy loads and incorporate force-feed lubrication. They are self-aligning, straight sleeve, multi-lobe sleeve, or tilting pad type, each sized for good damping characteristics and high stability.

Tilting pad bearings have an advantage over the sleeve type as they eliminate oil whip or half-speed oil whirl which can cause severe vibrations.

Bearing sleeves or pads are fitted with replaceable steel-backed babbitted shells or liners.

Axial thrust bearings are bidirectional, double faced, pivoted-shoe type designed for equal thrust capacity in both directions and arranged for force-feed lubrication on each side.

Thrust bearings are sized for continuous operation at maximum differential pressure including surge thrust loads, axial forces transmitted from the flexible coupling and electric motor thrust.

On units where the thrust forces are low, a tapered land thrust bearing may be used but must be selected for proper rotation direction. At times a combination of pivoted-shoe and tapered land is recommended.

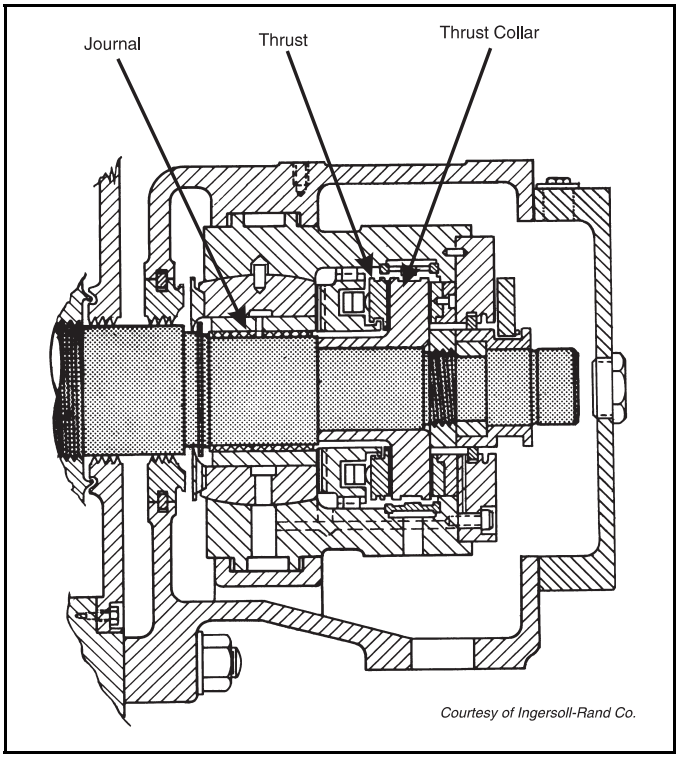
Compressor designs with impellers arranged in one direction usually have a balance drum (piston) mounted on the discharge end of the shaft to minimize axial loads on the thrust bearing.

Fig. 13-39 illustrates typical journal and thrust bearings generally used in horizontally and vertically split casings.

Bearing supports are cast integral with, or bolted to, the case with isolated bearing chambers to prevent lubricating oil leakage into the gas system or contamination of the oil or the gas. Bearing housings are horizontally split and readily accessible for inspection and maintenance. Provisions are made to accommodate pick-ups and sensors for vibration and temperature monitoring.

FIG. 13-39

Journal and Thrust Bearing Assembly



Magnetic Bearings

Magnetic bearings are a relatively new development that are gaining in popularity. An active magnetic bearing comprises two main components — a mechanical part and an electronic part. The mechanical parts of the bearing are similar to an electric motor with a rotor and stator. An iron core in the stator is wound with coils through which is fed an electric current, thereby inducing a magnetic field. This magnetic field produces the forces that support the compressor shaft.

The electronic part of the active magnetic bearing is the digital control system. It includes sensors that measure the exact position of the shaft. Deviations from the desired posi-

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tion of the shaft will trigger the software in the control system to adjust the current flowing through the electromagnets that determine the strength of the magnetic field. The currents are adjusted according to a set algorithm that corrects the deviation. Magnetic bearings are available in radial and axial/thrust designs.

Advantages

- Reduced space/weight requirements due to elimination of the need for a bearing lube oil system
- Reduced long term costs for maintenance and repairs
- Reduced bearing-related losses (near zero friction)
- Increased reliability and availability
- Improved machine monitoring/diagnostic capabilities
- Higher speeds possible.

Limitations

- Generally physically larger than "conventional" bearings
- Higher complexity
- Requires electrical power

See Fig. 13-40 for a schematic of a typical magnetic bearing arrangement.

Shaft Seals

Shaft seals are provided on all centrifugal compressors to limit, or completely eliminate, gas leakage along the shaft where it passes through the casing.

With the wide range of temperature, pressure, speed, and operating conditions encountered by compressors, there can be no one universal seal, or seal system, to handle all applications.

Basically, the designs of seals available are: labyrinth (gas), restrictive ring (oil or gas), liquid film (oil), and mechanical (contact) (oil or gas).

A mechanical (contact) seal, Fig. 13-41, has the basic elements similar to the liquid film seal. The significant difference

FIG. 13-40

Active Magnetic Bearing System

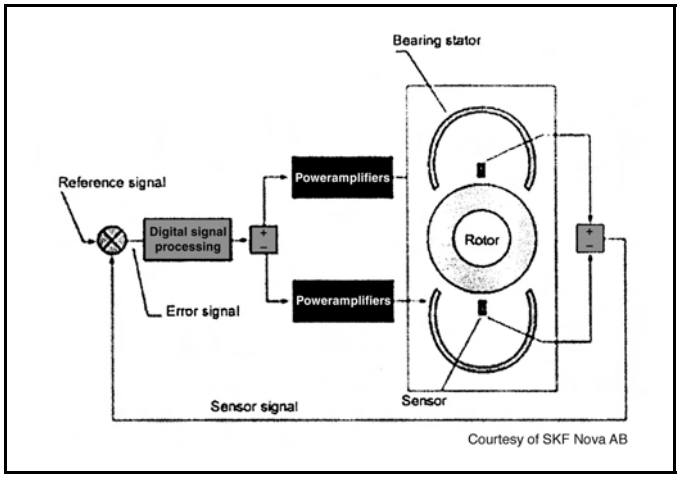
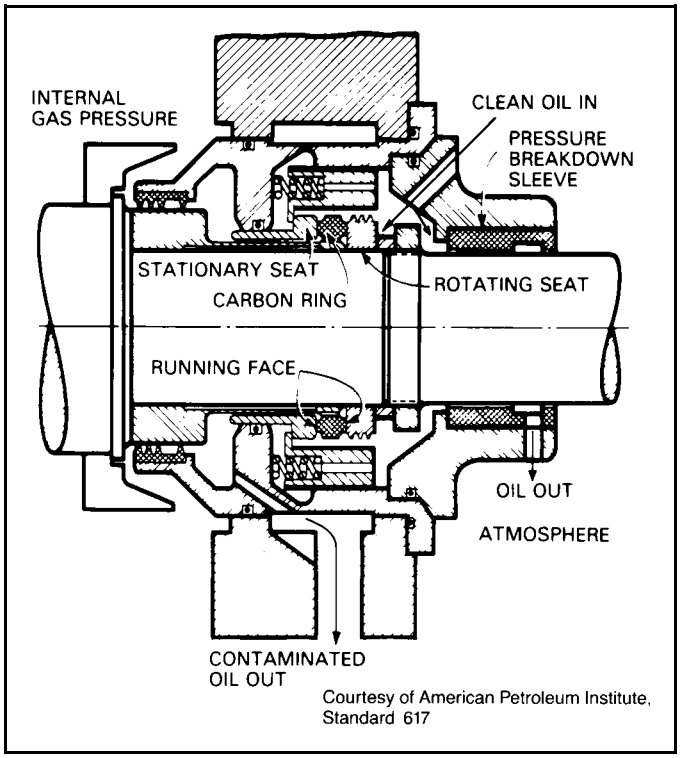


FIG. 13-41

Mechanical (Contact) Shaft Seal



is that clearances in this seal are reduced to zero. The seal operates with oil pressure 35 to 50 psi above internal gas pressure as opposed to 5 psi in the liquid film seal.

The mechanical (contact) seal can be applied to most gases, but finds its widest use on clean, heavier hydrocarbon gases, refrigerant gases, etc.

A mechanical gas seal uses the process gas as working fluid to eliminate the seal oil system. See Figs. 13-42 through 13-47.

The liquid film seal, Figs. 13-45 and 13-46, was also developed for the severe conditions of service but requires higher oil circulation rate than the mechanical (contact) type.

The seal consists of two sleeves which run at close clearance to the shaft with a liquid injected between the sleeves to flow to the seal extremities. The sleeves are lined with babbitt or a similar non-galling material which is compatible with the properties of the compressed gas and the sealing liquid.

The sealing liquid, usually a lubricating oil, is introduced between the two rings at a controlled differential pressure of about 5 psi above the internal gas pressure, presenting a barrier to direct passage of gas along the shaft. This fluid also performs the very important functions of lubricating the sleeves and removing heat from the seal area.

Lubrication and Seal-oil Systems

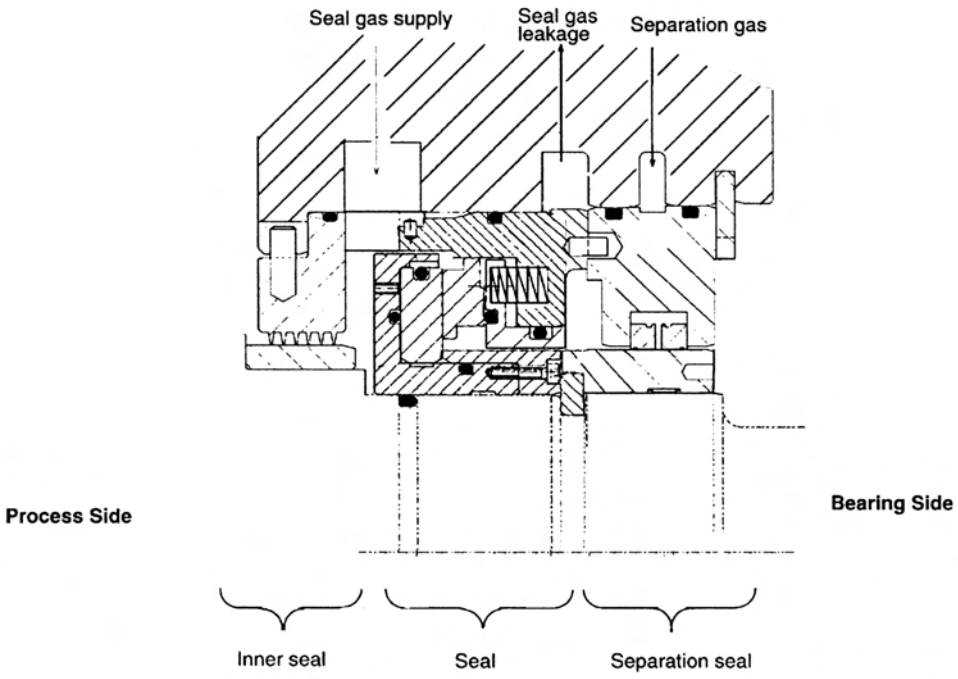
On all centrifugal compressors that have force-feed lubricated bearings, a lubrication oil system is required. When oil-film or mechanical (contact) seals are used, a pressurized seal-oil system must be provided.

Each system is designed for continuous operation with all the elements (oil reservoir, pumps with drivers, coolers, filters,

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FIG. 13-42

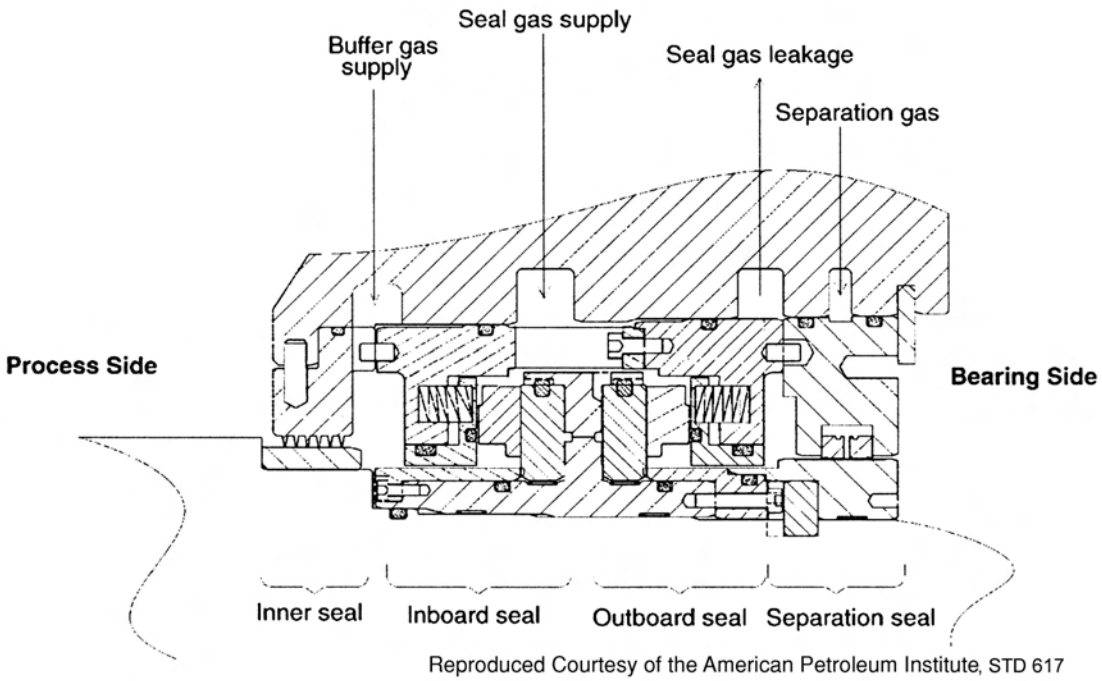
Single Gas Seal



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FIG. 13-43

Double Gas Seal

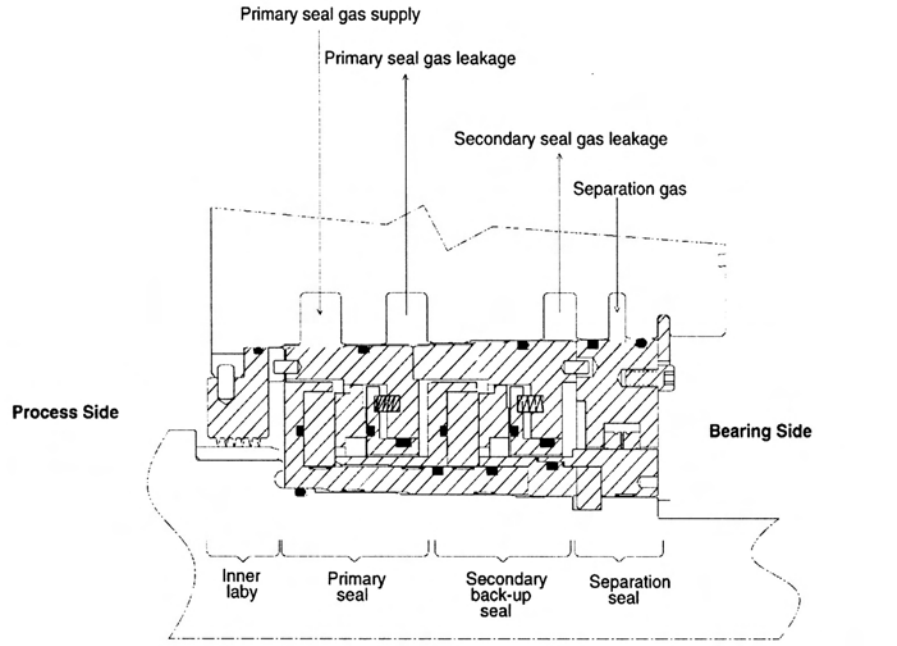


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FIG. 13-44

Tandem Gas Seal



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pressure gauges, control valves, etc.) piped and mounted on a flat steel fabricated base plate located adjacent to the compressor. The compressor manufacturer normally supplies both systems in order to have overall unit responsibility.

Depending on the application, lubrication and seal-oil systems may be furnished as combined into one system, or as one lubrication system having booster pumps to increase the pressure of only the seal oil to the required sealing level. In service involving heavily contaminated gases, separate lube-oil and seal-oil systems should be used.

The lubrication system may supply oil to both compressor and driver bearings (including gear) couplings (if continuously lubricated), as well as turbine governor, trip and throttle valve, and hydraulic control system.

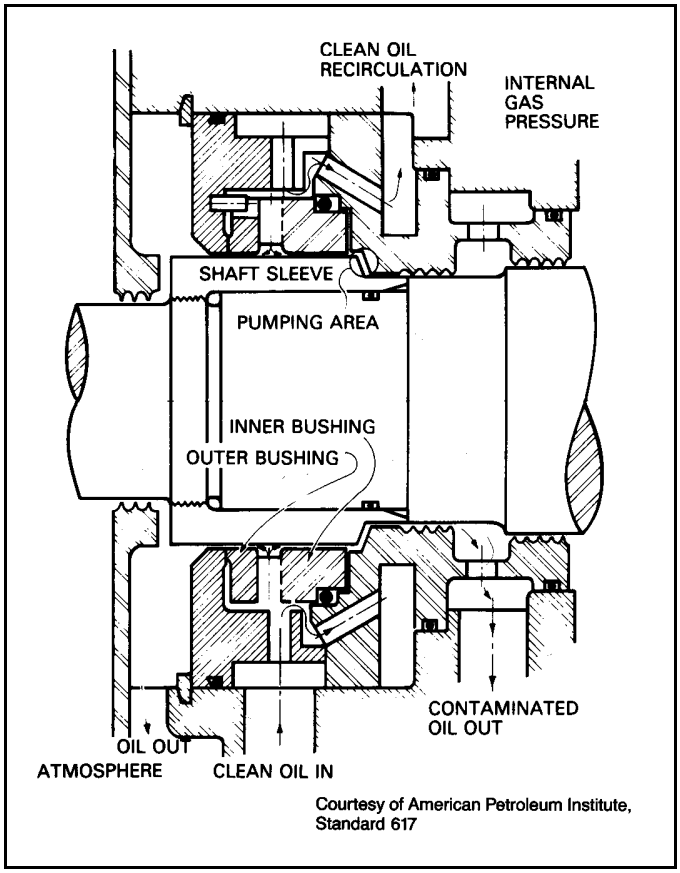
A single lubricant shall be used in all system equipment, usually an oil, having approximate viscosities of 150 Saybolt Universal Seconds (SUS) at 100°F and 43 SUS at 210°F.

In addition to all the elements of a common pressurized lubrication system, the seal oil system requires a collection system for the oil. Depending on the gas composition, a degassing tank may be installed in the seal oil trap return line to remove the oil-entrained gas prior to return of the seal oil to the common oil reservoir. The flow past the outer sleeve passes through an atmospheric drain system and is returned to the reservoir. The relatively low flow through the inner sleeve is collected in a drain trap or continuous drainer and may be returned to the reservoir or discarded, depending upon the degree and type of contamination which occurred while it was in contact with the internal gas.

Compressors using only liquid film seals should be provided with a seal-oil system which incorporates an overhead surge

FIG. 13-45

Liquid Film Shaft Seal with Pumping Bushing



Courtesy of American Petroleum Institute, Standard 617

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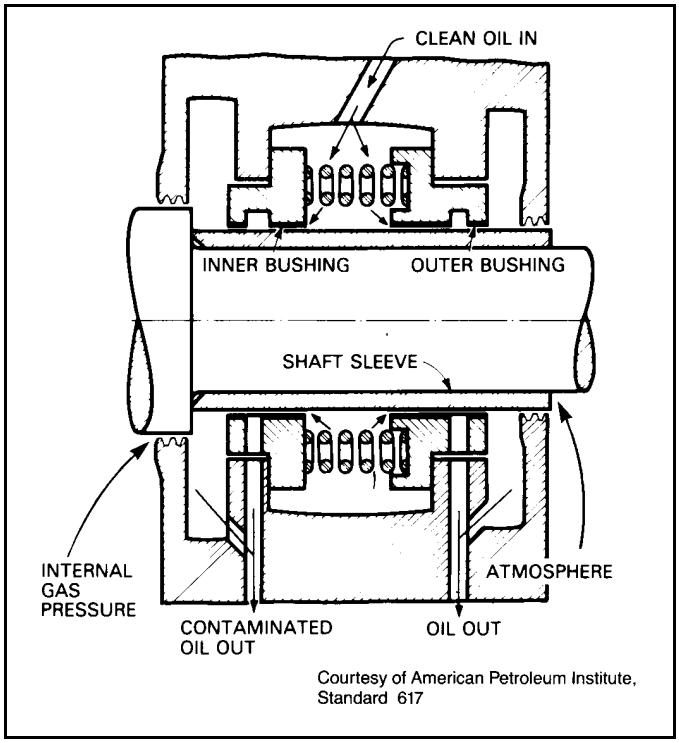
tank. The surge tank provides seal-oil capacity for coastdown in case of a compressor shutdown.

In combined seal-oil and lube-oil systems when large amounts of contaminants are present in the process gas, the seal-oil design may call for buffer gas injection to form a barrier between the compressed gas and the seal oil.

Fig. 13-47 shows clean sweet buffer gas being injected into the center of a labyrinth seal preceding the oil film seal with seal oil supplied between the two sleeves. Part of the seal oil flows across the inner sleeve and mixes with buffer gas and then drains into the seal oil trap. The other part of the seal oil flows across the outer sleeve, mixes with the bearing lube oil drain flow, and returns to the common lube- and seal-oil reservoir.

FIG. 13-46

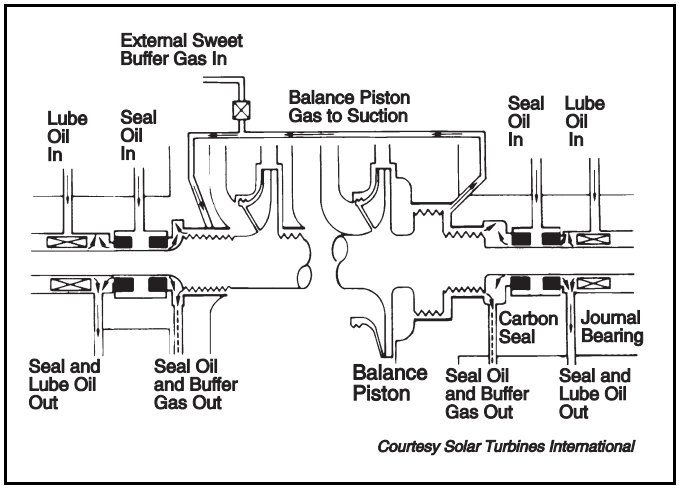
Liquid Film Shaft Seal with Cylindrical Bushing



Courtesy of American Petroleum Institute, Standard 617

FIG. 13-47

Combined Seal-Oil and Lube-Oil System with External Sweet Buffer Gas



Courtesy Solar Turbines International

Drivers — Centrifugal compressors can be driven by a wide variety of prime movers including electric motors, steam turbines, gas combustion turbines, and gas-expander turbines. Each driver has its own design parameters. A motor drive presents limitations in operation of the compressor due to constant and low speed. The constant speed restriction is minimized by suction or discharge throttling. The low speed restriction is corrected by introduction of a speed increasing gear. A steam turbine, on the other hand, has variable speed capability that allows more control of the compressor capacity or discharge pressure, and its high speed permits the compressor to be directly connected to the driver. In the case of a single-shaft gas turbine, the power output is limited at a reduced speed.

Drivers are discussed in Section 15.

CONTROL SYSTEMS

Centrifugal compressor controls can vary from the very basic manual recycle control to elaborate ratio controllers. The driver characteristics, process response, and compressor operating range must be determined before the right controls can be selected.

The most efficient way to match the compressor characteristic to the required output is to change speed in accordance with the fan laws (affinity laws, see Eqs 13-22 to 13-24):

$$\frac{N_1}{N_2} = \frac{Q_1}{Q_2} = \frac{\sqrt{H_1}}{\sqrt{H_2}}$$

One of the principal advantages of using steam or gas turbines as drivers for compressors is that they are well suited to variable-speed operation. With such drivers, the speed can be controlled manually by an operator adjusting the speed governor on the turbine or, alternatively, the speed adjustment can be made automatically by a pneumatic or electric controller that changes the speed in response to a pressure or flow signal.

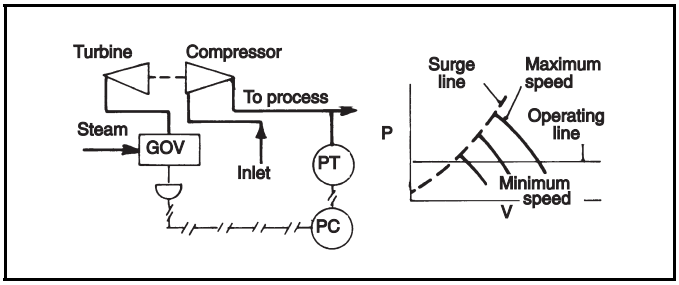
Pressure Control at Variable Speed

The control system operates as follows:

The pressure transmitter (PT) in Fig. 13-48 senses the process pressure. It converts this signal to a signal proportional to

FIG. 13-48

Pressure Control at Variable Speed



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the process pressure and sends it to the pressure controller (PC).

The pressure controller amplifies the transmitter signal and sends a modified signal to the final element. Depending on system requirements the controller may require additional correction factors called reset and rate.

The final element in this case is speed control. This mechanism varies the turbine-governor speed setting over a predetermined range.

As the load decreases, pressure will rise. An increase in process pressure above the set value will cause the signal to reach the governor and reduce the speed, maintaining the desired system pressure.

Volume Control at Variable Speed

If the nature of the process requires constant volume delivered, then the arrangement shown in Fig. 13-49 would be used.

Here, the flow transmitter (FT) senses the process flow, converts the signal to a signal proportional to the process flow, and sends it to the flow controller (FC).

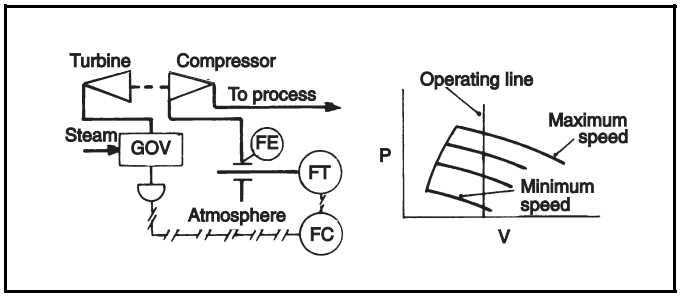
The flow controller amplifies the transmitter signal and sends a modified signal to the final element. Reset and rate correction factors may be needed.

The final element is speed control, which is accomplished by a mechanism that varies the turbine-governor speed setting. An increase in flow over set point would cause a signal to reach the governor and reduce the speed to maintain the desired system flow.

When using electric motors as constant speed drivers, the centrifugal compressor is normally controlled by a suction throttling device such as butterfly valve or inlet guide vanes. Throttling the suction results in a slightly lower suction pressure than the machine is designed for, and thus requires a higher total head if the discharge pressure remains constant. This can be matched to the compressor head-capacity curve, i.e., higher head at reduced flow. In throttling the inlet, the density of the gas is reduced, resulting in a matching of the required weight flow to the compressor inlet-volume capabilities at other points on the head/capacity curve.

FIG. 13-49

Volume Control at Variable Speed



Pressure Control at Constant Speed

The control system shown in Fig. 13-50 has the pressure signal sensed and amplified in a similar manner as described in the scheme for variable speed control (Fig. 13-48).

FIG. 13-50

Pressure Control at Constant Speed

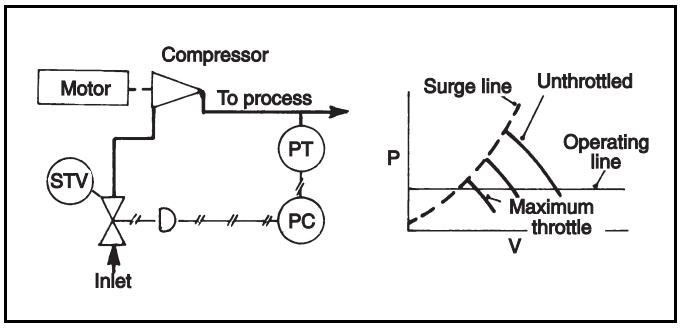
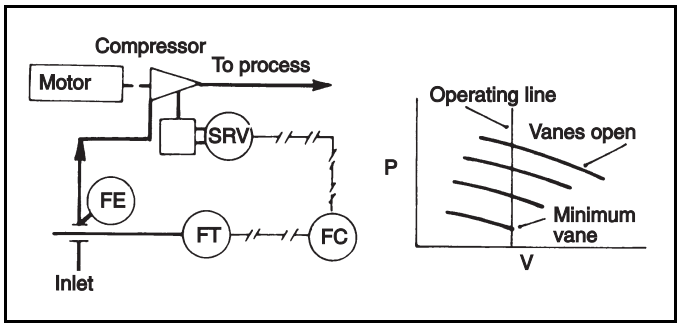


FIG. 13-51

Volume Control at Constant Speed



The final element is a suction throttle valve (STV) that reduces the flow of gas into the compressor.

A process pressure increase over a set value would cause a signal to reach the suction throttle valve (STV) and would partially close the valve in order to reduce the inlet pressure.

Volume Control at Constant Speed

The control scheme for this arrangement is shown in Fig. 13-51.

The flow transmitter (FT) senses the process flow using an orifice or venturi as the primary flow element (FE), converts this to a signal that is proportional, and sends this signal to the flow controller (FC). The flow controller amplifies the transmitter signal and sends a modified signal to the final element. Reset and derivative controller actions may be required.

The final element is the compressor guide-vane mechanism. The guide vanes are adjusted by means of a positioning cylinder. This cylinder is operated by a servo-valve (SRV) that receives a signal from the flow controller.

Here, an increase in flow above the set point causes a signal to reach the final element, which will result in the closing of the guide vanes to decrease flow.

Adjustable Inlet Guide Vanes — The use of adjustable inlet guide vanes is the most efficient method of controlling a constant speed compressor. The vanes are built into the inlet of the 1st stage, or succeeding stages, and can be controlled through the linkage mechanism either automatically or manually.

The vanes adjust the capacity with a minimum of efficiency loss and increase the stable operating range at design pressure. This is accomplished by pre-rotation of the gas entering

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the impeller which reduces the head-capacity characteristics of the machine. Fig. 13-52 illustrates the effect of such control at various vane positions.

Prior to control selection, the economics of inlet guide vanes must be considered because of their higher initial cost, complex mechanism, maintenance, and requirement for frequent adjustment.

Anti-surge Control

It is essential that all centrifugal compressor control systems be designed to avoid possible operation in surge which usually occurs below 50% to 70% of the rated flow.

Compressor surge is a large pressure and volume fluctuation that takes place when attempting to operate at a higher pressure ratio than design maximum. The surge limit line (see Fig. 13-52) can be reached from a stable operating point by either reducing flow or decreasing suction pressures. An anti-surge system senses conditions approaching surge, and maintains the unit pressure ratio below the surge limit by recycling some flow to the compressor suction. Care must be taken to cool this recycle stream.

Volume, pressure rise, or pressure ratio may be used as control parameters to sense an approaching surge condition. Such a condition will be established by the characteristic curve of the compressor.

A volume-controlled anti-surge system is shown in Fig. 13-53. The flow transmitter (FT) senses the process flow using an orifice or venturi as a primary flow element (FE). It converts the signal to one that is proportional to the process flow and sends it to the surge controller (SC).

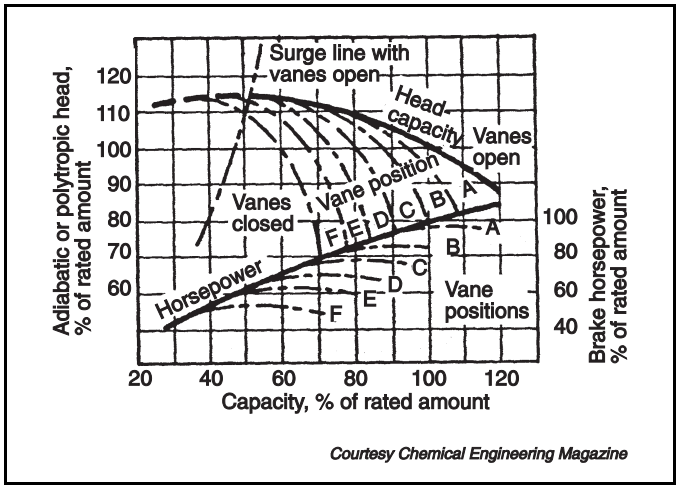
The surge controller compares the transmitted signal to the set-point signal. If the set-point signal is exceeded, the controller amplifies the signal difference and sends this modified signal to the final element. Reset and rate correction factors may be needed.

The final element is a surge control valve (SCV). The valve releases pressure buildup at the discharge of the compressor.

As flow decreases to less than the minimum volume set-point, a signal will cause the surge control valve to open. The valve opens, as required, to keep a minimum volume flowing through the compressor.

FIG. 13-52

Effect of Adjustable Inlet Guide Vanes on Compressor Performance



Courtesy Chemical Engineering Magazine

FIG. 13-53

Anti-Surge Control–Minimum Volume

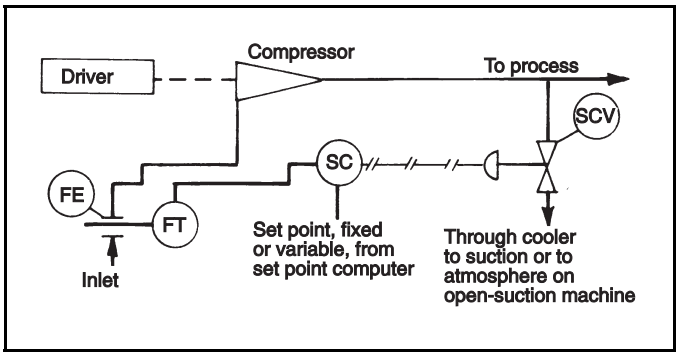
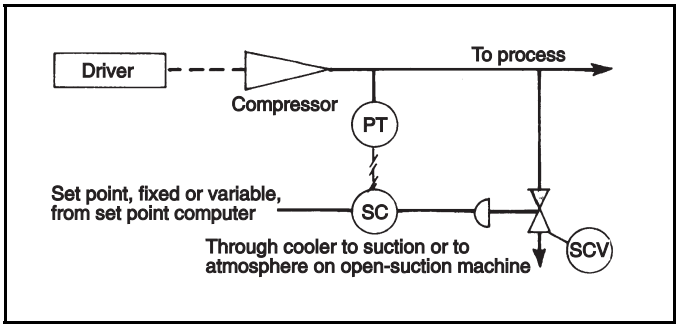


FIG. 13-54

Anti-Surge Control–Pressure Limiting



For a pressure-limiting anti-surge control system, see Fig. 13-54. The pressure transmitter (PT) senses the process pressure. It converts this to a signal that is proportional to process pressure and sends it to the surge controller (SC).

The surge controller compares the transmitted signal to the set-point signal. If the set-point signal is exceeded, the controller amplifies the signal difference and sends this modified signal to the surge control valve. Reset and rate correction factors may also be required. The blow-off valve relieves a pressure buildup on the discharge end of the compressor.

A process pressure increase over the pressure set-point signal will cause the blowoff valve to open. The valve opens as required to keep the pressure limited to a minimum volume of gas flowing through the compressor.

A set-point computer may be required on multiparameter compressor-control systems where the surge set point may change. Suction temperature, suction pressure, and speed are examples of parameters that cause the surge point of the compressor to change.

In the design of a compressor system, any changes that may occur from a single operating point in the overall process must be carefully considered. In many instances, automatic controls are needed to maintain the highest degree of system performance. In any event, the choice of manual or automatic controls is dictated by the operating pattern of the centrifugal compressor.

The user, compressor manufacturer, and contractor should work closely together to determine the minimum control sys-

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tem required. Various phases of operation, such as start-up, shut-down, initial runs, and normal runs, should be investigated to make certain that a workable system has been designed.

Vibration Control System

This control system may be provided to monitor the driver behavior at the shaft bearings for detection of excessive lateral vibration and axial movement and for protection against possible machinery failure through alarm and/or shutdown devices.

The system may protect not only the compressor but also the driver, such as a steam or gas turbine, that usually runs at the same high speed as the compressor. When a speed increasing or reducing gear unit is furnished between the compressor and driver, also consider monitoring vibration at the gear shaft bearings.

The main system components are: variation transducer(s), signal amplifier(s) with d-c power supply, and vibration monitor and/or analyzer.

Vibration transducers fall into three categories: displacement probe, velocity pick-up, and accelerometer.

The displacement probe is most commonly used for equipment with high value, as it can measure shaft vibration relative to bearing housing. Output signal from each transducer is small and, therefore, it must be amplified before being transmitted to a vibration monitor or analyzer.

FIG. 13-55

Vibration Severity Chart[1]

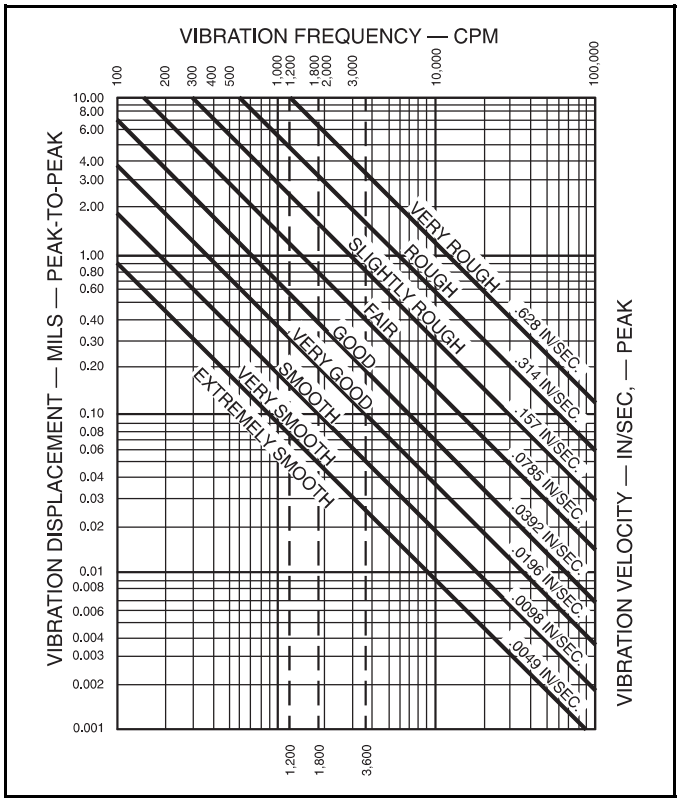


Fig. 13-55 shows a vibration severity chart for use as a guide in judging vibration levels as a warning of impending trouble.

For more information on vibration monitoring systems, see API Standard 670, Noncontacting Vibration and Axial Position Monitoring System, and API Standard 678, Accelerometer-Based Vibration Monitoring System.

OPERATIONAL CONSIDERATIONS

Rotor Dynamics and Critical Speeds

The demand for smooth-running turbomachinery requires careful analysis of rotor dynamics taking into account bearing performance, flexibility, critical speed, and rotor response.

Equally important is to analyze the dynamic behavior of the compressor for sudden changes in load due to start-up, shutdown, or loss of power supply.

Successful rotor design is the result of accurate calculation of critical speeds. A critical speed occurs at a condition when the rotor speed corresponds to a resonant frequency of the rotor-bearing support system. Under no circumstances should the compressor be allowed to run at a critical speed for a prolonged length of time as the rotor vibrations amplified by this condition can cause machinery failure.

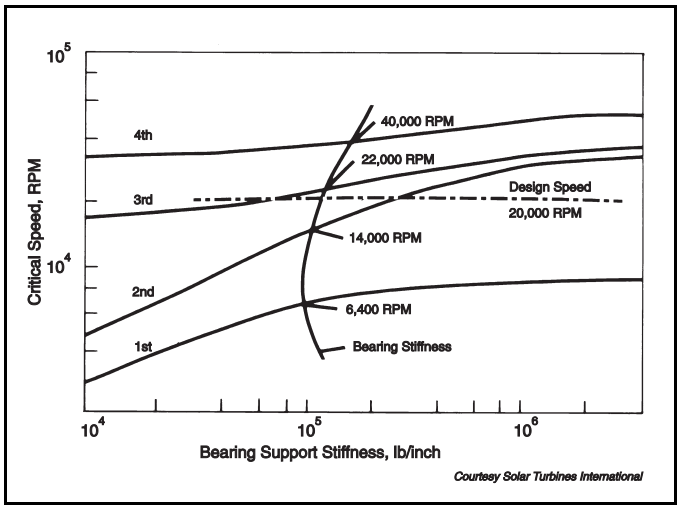
Critical Speed Map

A critical speed map is one of various methods used to predict the operational behavior of the rotor. First, the critical speeds for a given rotor geometry are calculated for a range of assumed bearing support stiffness values. The result is a map like that shown in Fig. 13-56. The bearing stiffness characteristics are determined from the geometry of the bearing support system, and cross-plotted on the critical speed map.

The map depicts the values of the undamped critical speeds and how they are influenced by bearing stiffness. The intersections of the bearing stiffness curve and the critical speed lines represent the undamped critical speeds. The intersection points generally indicate margins between the criticals and the operating speed range.

FIG. 13-56

Undamped Critical Speed Map



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However, the use of this map is very limited because it is based on a simplified undamped, circular synchronous analysis with no cross-coupled or unbalance effects. It is a good trending tool showing a machine's basic dynamic characteristics. It may not accurately depict peak response frequencies.

The critical speed map is used extensively because it enables determination of bearing or support stiffness by correlating test-stand data.

Unbalance Response Analysis

This method predicts rotor-bearing system resonances to greater accuracy than the critical speed map. Here, bearing support stiffness and damping are considered together with synchronous vibration behavior for a selected imbalance distribution. A computer is normally required to solve the resulting differential equations. Satisfactory results depend on the accurate input of bearing stiffness and damping parameters.

Several runs are usually made with various amounts and locations of unbalance. The plot of results of a typical unbalance response study is shown in Fig. 13-57. Each curve represents the rotor behavior at a particular station or axial location such as those corresponding to the midspan, bearings, and overhangs.

No rotor can be perfectly balanced and, therefore, it must be relatively insensitive to reasonable amounts of unbalance.

The unbalance-response results predict the actual amplitudes that permit calculations of the unbalance sensitivity. This is expressed in mils of vibration amplitude per ounce-inch or gram-inch of unbalance.

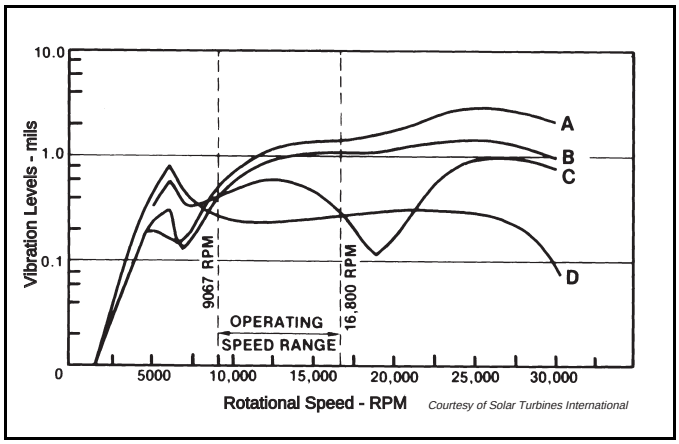
The peaks of the response curves represent the critical speed locations. Fig. 13-58 shows limits of placement of critical speeds as specified in the API Standard 617, Centrifugal Compressors for General Refinery Services.

Critical speeds should not encroach upon operating speed ranges, and the separation margin of encroachment (SM) from all lateral modes is required to be at least:

1. Twenty (20) percent over the maximum continuous speed for rigid shaft rotor systems.
2. Fifteen (15) percent below any operating speed and twenty (20) percent above the maximum continuous speed for flexible shaft rotor system.

FIG. 13-57

Unbalance Response Plot



Field Performance

Once the compressor has been installed, quite often the performance of the system is to be tested. The same information is required for evaluation as was supplied for the initial selection. This information is essential:

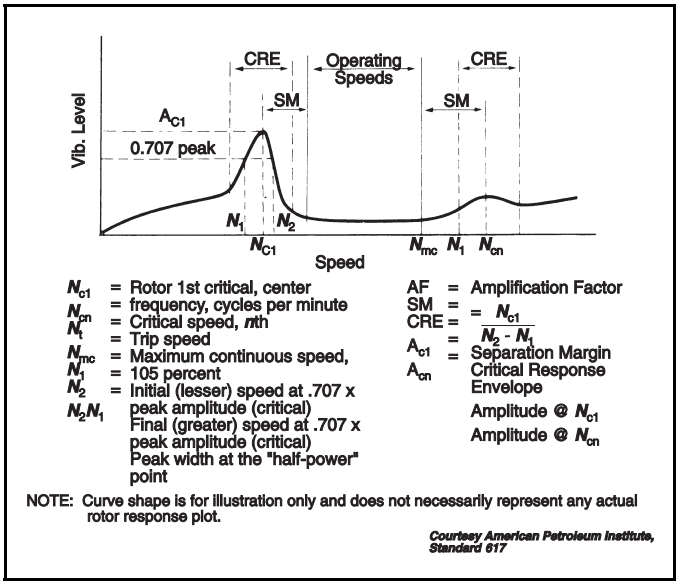
1. Inlet conditions
   a. Flow (scfm, acfm, or lb/min)
   b. Gas analysis
   c. Pressure, psia
   d. Temperature, °F
2. Intermediate conditions
   a. Pressure at intermediate nozzles, psia
   b. Temperature at intermediate nozzles, °F
3. Discharge conditions
   a. Pressure, psia
   b. Temperature, °F
4. Compressor speed, rpm
5. Power requirement from driver if available (steam flow, amperes, etc.)

If all these conditions were given except flow, it would not be possible to determine the compressor operating point. The system resistance curve imposed on the typical performance is not always constant. Discharge temperature of the compressor is a good indication as to the operation of the compressor. However, such items as recycle (internal or external), different gas analysis, different suction conditions, and different flow can affect the values. The efficiency at part load or overload will normally be lower than at design conditions.

Test procedures should be agreed upon between the manufacturer and user before a performance test is run. A test point should be run several times to see if the results can be dupli-

FIG. 13-58

Rotor Response Plot



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cated. Calibrated instruments should be used to improve the accuracy of the test data.

Troubleshooting

Operational troubles occurring in service may be due to a variety of causes.

If the trouble cannot be traced to adverse gas flow conditions or liquid "slugs" present in the system, Fig. 13-59 can be used as a guide for troubleshooting frequently encountered problems.

Careless operation and maintenance needs little comment. Lack of proper care of any machine is bound to result in a succession of minor troubles eventually leading to a major breakdown.

INTEGRALLY GEARED COMPRESSORS

An integrally geared compressor utilizes a central driven bull gear with typically 2–4 high speed pinion driven shafts. One or two impellers can be mounted on each pinion shaft. See Figures 13-60 and 13-61. This forms a compact unit for the multistage compression of a wide range of gases.

Integrally geared compressors offer the following potential advantages:

FIG. 13-59

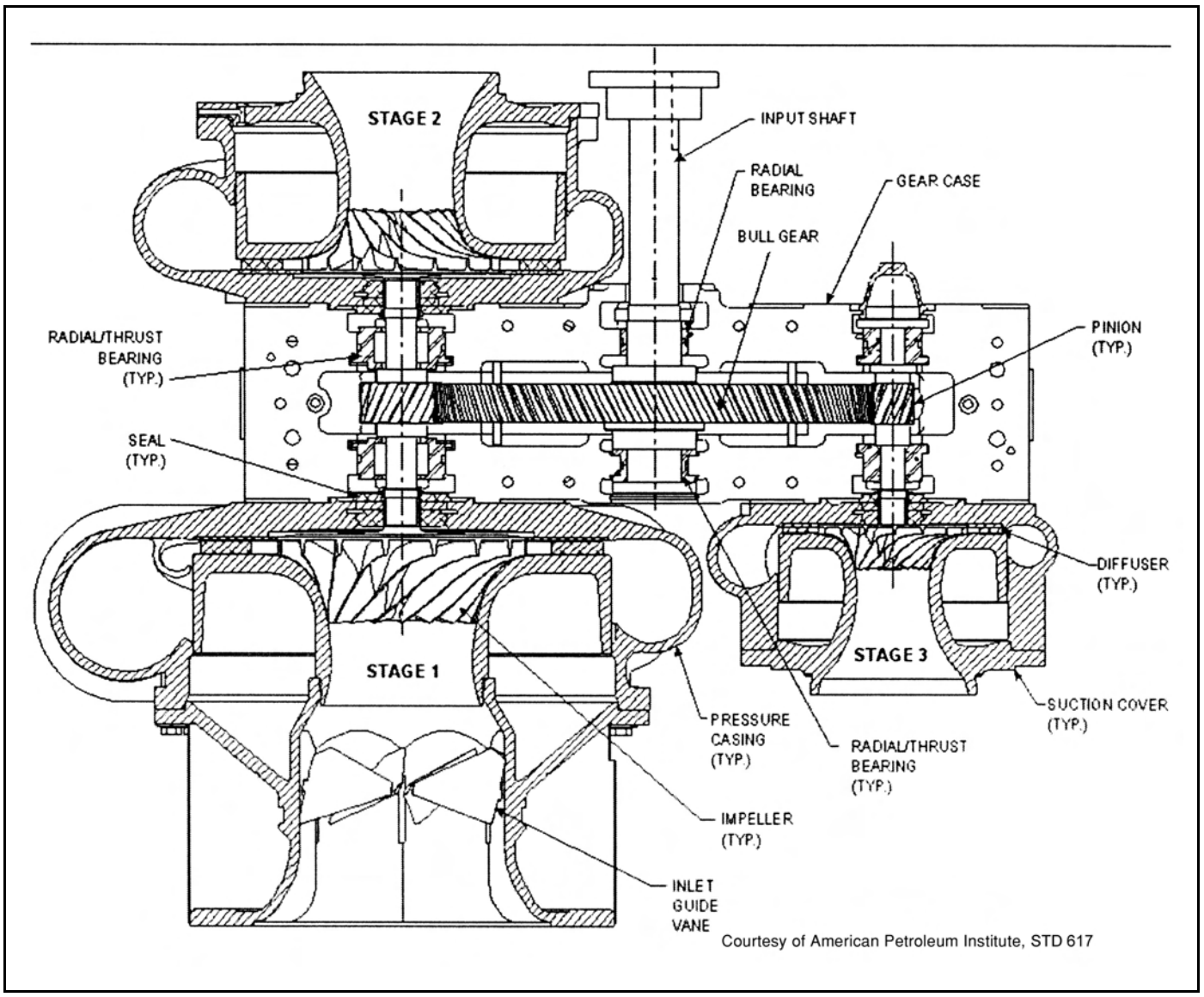
Probable Causes of Centrifugal Compressor Trouble

Trouble	Probable Cause(s)
Low Discharge Pressure	1. Compressor not up to speed. 2. Excessive compressor inlet temperature. 3. Low inlet pressure. 4. Leak in discharge piping. 5. Excessive system demand from compressor.
Compressor Surge	1. Inadequate flow through the compressor. 2. Change in system resistance due to obstruction in the discharge piping or improper valve position. 3. Deposit buildup on rotor or diffusers restricting gas flow.
Low Lube Oil Pressure	1. Faulty lube oil pressure gauge or switch. 2. Low level in oil reservoir. 3. Oil pump suction plugged. 4. Leak in oil pump suction piping. 5. Clogged oil strainers or filters. 6. Failure of both main and auxiliary oil pumps. 7. Operation at a low speed without the auxiliary oil pump running (if main oil pump is shaft-driven). 8. Relief valve improperly set or stuck open. 9. Leaks in the oil system. 10. Incorrect pressure control valve setting or operation. 11. Bearing lube oil orifices missing or plugged.
Shaft Misalignment	1. Piping strain. 2. Warped bedplate, compressor or driver. 3. Warped foundation. 4. Loose or broken foundation bolts. 5. Defective grouting.
High Bearing Oil Temperature Note: Lube oil temperature leaving bearings should never be permitted to exceed 180°F.	1. Inadequate or restricted flow of lube oil to bearings. 2. Poor conditions of lube oil or dirt or gummy deposits in bearings. 3. Inadequate cooling water flow lube oil cooler. 4. Fouled lube oil cooler. 5. Wiped bearing. 6. High oil viscosity. 7. Excessive vibration. 8. Water in lube oil. 9. Rough journal surface.
Excessive Vibration Note: Vibration may be transmitted from the coupled machine. To localize vibration, disconnect coupling and operate driver alone. This should help to indicate whether driver or driven machine is causing vibration.	1. Improperly assembled parts. 2. Loose or broken bolting. 3. Piping strain. 4. Shaft misalignment. 5. Worn or damaged coupling. 6. Dry coupling (if continuously lubricated type is used). 7. Warped shaft caused by uneven heating or cooling. 8. Damaged rotor or bent shaft. 9. Unbalanced rotor or warped shaft due to severe rubbing. 10. Uneven build-up of deposits on rotor wheels, causing unbalance. 11. Excessive bearing clearance. 12. Loose wheel(s) (rare case). 13. Operating at or near critical speed. 14. Operating in surge region. 15. Liquid "slugs" striking wheels. 16. Excessive vibration of adjacent machinery (sympathetic vibration).
Water In Lube Oil	1. Condensation in oil reservior. 2. Leak in lube oil cooler tubes or tube-sheet.

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FIG. 13-60

Typical Integrally Geared Compressor Showing Nomenclature of Key Parts



- low power consumption due to different impeller speeds, tailored aerodynamics and optimized auxiliaries.
- wide operating range and improved part load efficiencies due to adjustable inlet guide vanes at the first or at all compression stages.
- multiservice capability
- packaged designs available. A package includes the compressor, process coolers, lube oil console, process piping and all tubing and wiring.

Design requirements of integrally geared compressors are covered by API Standard 617.

AXIAL COMPRESSORS

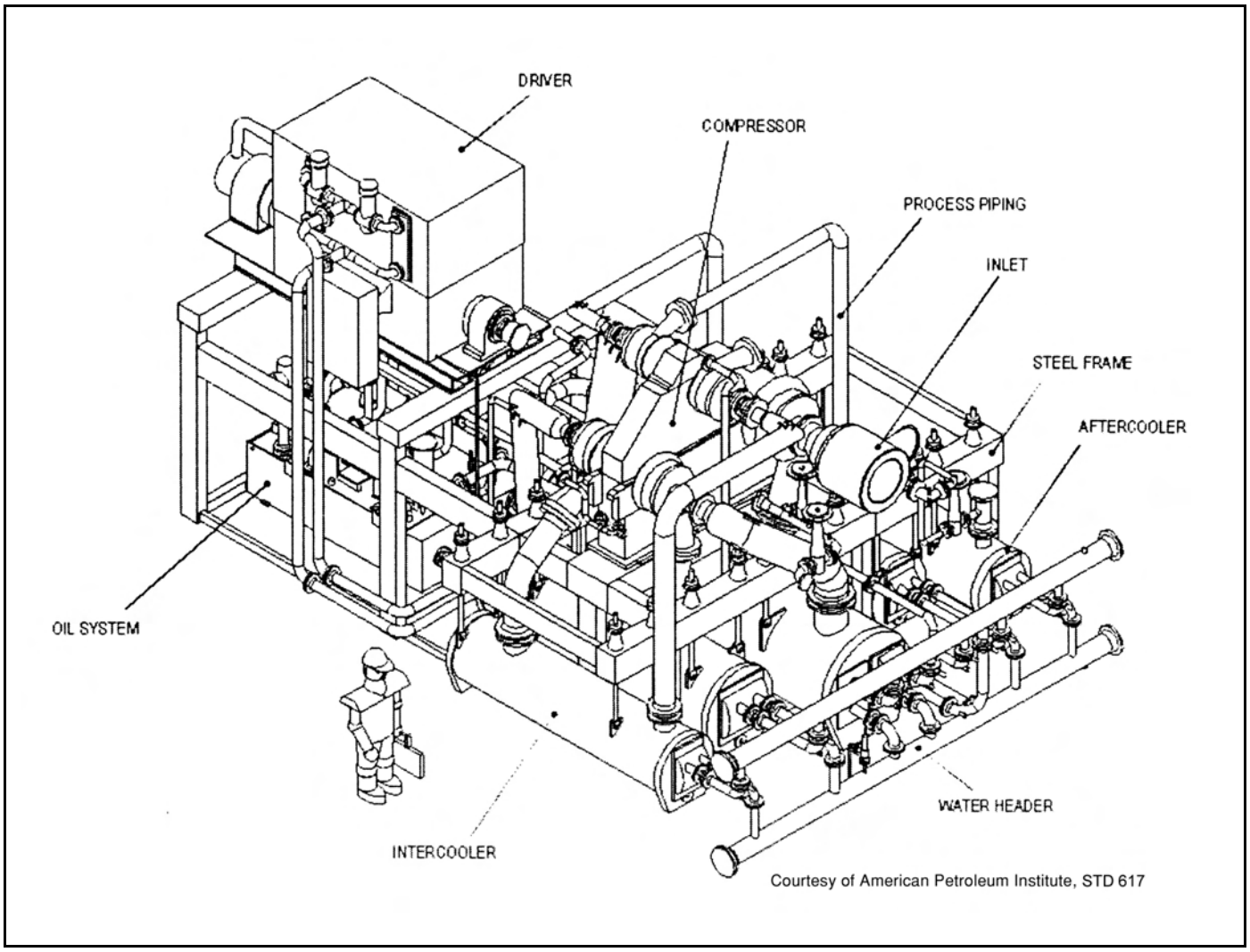
Axial compressors are basically high-flow, low-pressure machines, in contrast to the lower flow, high-pressure centrifugal compressors (the axial compressors used in gas turbines are often designed for higher pressures and compression ratios). Axial compressors are generally smaller and significantly more efficient than comparable centrifugal compressors. The characteristic feature of an axial compressor, as its name implies, is the axial direction of flow through the machine. An axial flow compressor requires more stages than a centrifugal due to the lower pressure rise per stage. In general, it takes approximately twice as many stages to achieve a given pressure ratio as would be required by a centrifugal. Although the axial compressor requires more stages, the diametral size of an axial is typically much lower than for a centrifugal. The axial compressor's capital cost is usually higher than that of a centrifugal but may be justified based on efficiency and size.

The axial compresor utilizes alternating rows of rotating and stationary blades to transfer the input energy from the rotor to the gas in order to generate an increase in gas pressure. A multistage axial flow compressor has two or more rows of rotating blades operating in series on a single rotor in a single

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FIG. 13-61

Typical Integrally Geared Compressor Arrangement Showing Nomenclature of Key Elements



casing. The casing contains the stationary vanes (stators) for directing the air or gas to each succeeding row of rotating blades. These stationary vanes, or stators, can be fixed or variable angle, or a combination of both.

A cross-sectional view of a typical axial flow compressor is shown in Fig. 13-62.

Performance Capabilities — The volume range of the axial compressor starts at approximately 30,000 cfm with a typical upper end of the flow range at 400,000 cfm. Much larger axial machines have been built. As can be seen in Fig. 13-3, the flow range for the axial overlaps the higher end of the range for typical centrifugal compressor coverage. At the lower end of the axial's flow range, a thorough evaluation of axial vs centrifugal must normally be made. However, at the higher end flows, the axial compressor often becomes the obvious choice. As stated previously, the physical size of the axial is far smaller than the comparable centrifugal machine that would be required, and the efficiency of the axial is usually better. In many high flow applications, the axial is often a better match for the drivers that would typically be selected.

Because of the low pressure rise per stage, axial compressors are always manufactured as multistage machines. Axial compressors are in general low pressure machines. Typical discharge pressures are usually less than approximately 100 psig. They are very commonly utilized in refineries and other industrial processes for high volume, low pressure air supply applications. The most common application of axial compressors, besides aircraft jet engine use, is in gas turbines. In gas turbine applications, the axial air compressor is often designed to operate at final discharge pressures of up to around 500 psig.

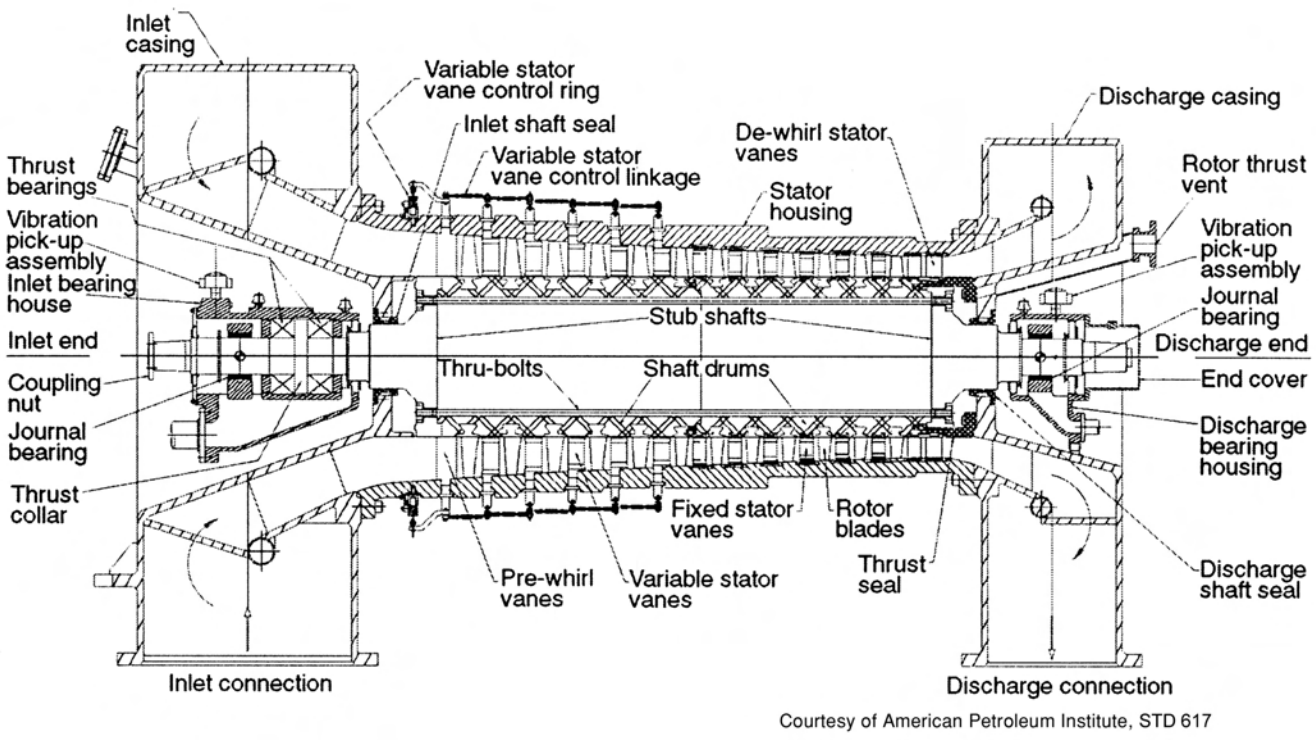
Horsepower requirements for axial flow compressors in process service typically range from 3,000 to 65,000 HP for single casing units, depending on flow and pressure ratio requirements. Efficiencies for axial compressors are high, especially for larger machines, and can reach 90% (adiabatic).

Design requirements for centrifugal compressors are covered by API Standard 617.

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FIG. 13-62

Typical Axial Compressor Showing Nomenclature of Key Parts



Courtesy of American Petroleum Institute, STD 617

SCREW COMPRESSORS

Screw compressors, also known as helical lobe compressors, fall into the category of rotary positive displacement compressors. Fig. 13-63 shows a cutaway cross-section of a typical rotary screw compressor.

Rotary screw compressors are available in oil-free (dry) or oil-injected designs. Oil-free compressors typically use shaft-mounted gears to keep the two rotors in proper mesh without contact. Applications for oil-free compressors include all processes that cannot tolerate contamination of the compressed gas or where lubricating oil would be contaminated by the gas. Oil-injected screw compressors are generally supplied without timing gears. The injected lubricant provides a layer separating the two screw profiles as one screw drives the other. Oil-injected machines generally have higher efficiencies and utilize the oil for cooling as well, which allows for higher compression ratios in a single screw compressor stage.

Although originally intended for air compression, rotary screw compressors are now compressing a large number of gases in the hydrocarbon processing industries. In particular, screw compressors are widely used in refrigeration service and are gaining in popularity in the gas production business in booster and gas gathering applications.

Gas compression is achieved by the intermeshing of the rotating male and female rotors. Power is applied to the male rotor and as a lobe of the male rotor starts to move out of mesh with the female rotor a void is created and gas is taken in at the inlet port. As the rotor continues to turn, the intermesh space is increased and gas continues to flow into the compressor until the entire interlobe space is filled. Continued rotation brings a male lobe into the interlobe spacing compressing and moving the gas in the direction of the discharge port. The volume of gas is progressively reduced as it increases in pressure. Further rotation uncovers the discharge port and the compressed gas starts to flow out of the compressor. Continued rotation then moves the remaining trapped gas out while a new charge is drawn into the suction of the compressor into the space created by the unmeshing of a new pair of lobes as the compression cycle begins again. Fig. 13-64 provides a sequence of drawings showing compression process. Screw compressors are usually driven by constant speed motors, with capacity control normally achieved via an internal regulating device known as a slide valve. By moving the slide in a direction parallel to the rotors, the effective length of the rotors can be shortened. This provides smooth control of flow from 100 percent down to 10 percent of full compressor capacity.

Rotary screw compressors in use today cover a range of suction volumes from 180 to 35,000 acfm, with discharge pressures up to 580 psig. Typical adiabatic efficiency will be in the range of 70 to 80%.

Design requirements for screw compressors are covered under API Standard 619.

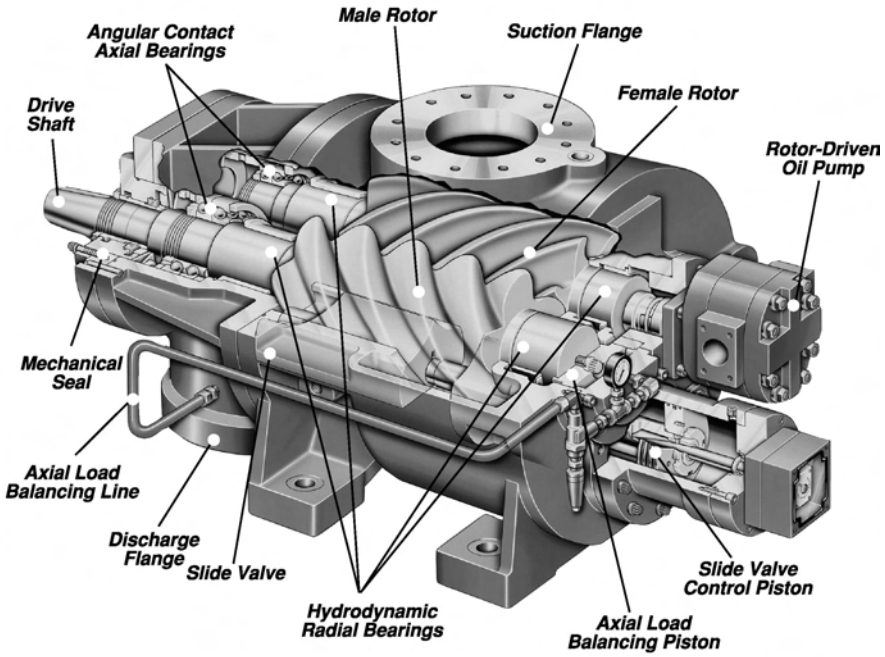
ROTARY-SLIDING VANE COMPRESSORS

Rotary-sliding vane compressors (Fig. 13-65) are positive displacement machines. They have several applications, in-

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FIG. 13-63

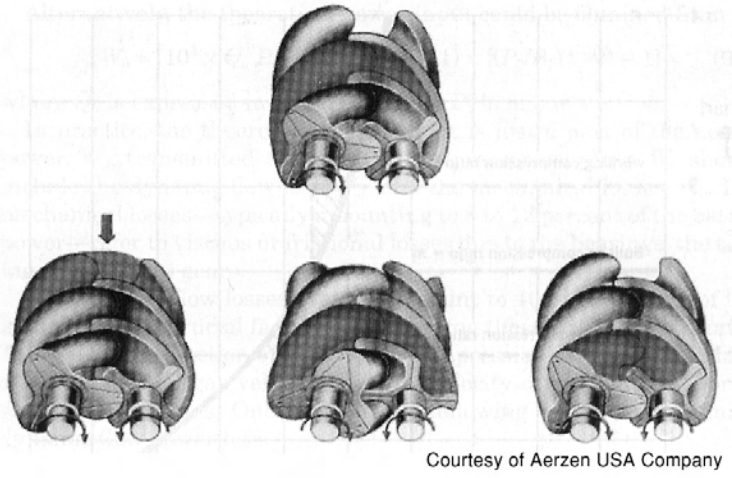
Rotary Screw Compressor



Courtesy of Ariel Corporation

FIG. 13-64

Working Phases of Rotary Screw Compressor



Courtesy of Aerzen USA Company

(a)

Suction intake Gas enters through the intake aperture and flows into the helical grooves of the rotors which are open

(b)

Compression process As rotation of the rotors proceeds, the air intake aperture closes, the volume diminishes and pressure rises.

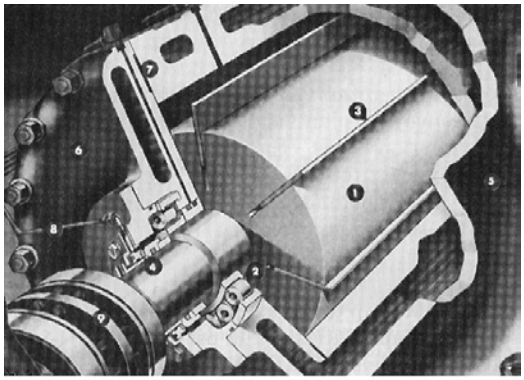
(c)

Discharge The compression process is completed, the final pressure attained, the discharge commences.

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FIG. 13-65

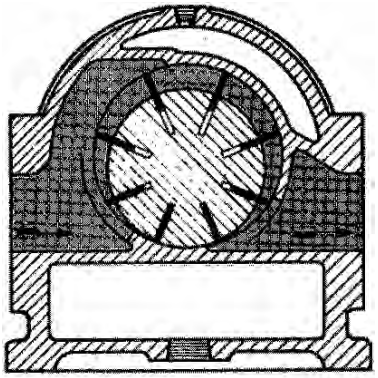
Sliding Vane Compressor and Principal Components: Rotor and Shaft (1), Bearings (2), Blades (3), Mechanical Seals (4), Cylinder and Housing (5), Heads and Covers (6), Gaskets (7), Lube Supply Line (8), Coupling (9)



Courtesy of A-C Compressor Corporation

FIG. 13-66

Operating Principle of Sliding Vane Compressor



Courtesy of A-C Compressor Corporation

cluding vapor recovery and vacuum service. Each unit has a rotor eccentrically mounted inside a water jacketed cylinder. The rotor is fitted with blades that are free to move radially in and out of longitudinal slots. These blades are forced against the cylinder wall by centrifugal force. Fig. 13-66 illustrates how individual pockets are thus formed by the blades, and how the gas inside these pockets is compressed as the rotor turns. Oil is injected into the flow stream to lubricate the vanes, and is recovered via a downstream scrubber and recycled to the inlet.

Sliding vane compressors are available in single- and multistage configurations. Typical single-stage capacities are ranging through 3200 cfm and 50 psig; two-stage compressors deliver pressures from 60 to 150 psig and flows up to approximately 1800 cfm. Most applications of rotary-sliding vane compressors in oil and gas service involve fairly small units, normally under 150 HP.

EJECTORS

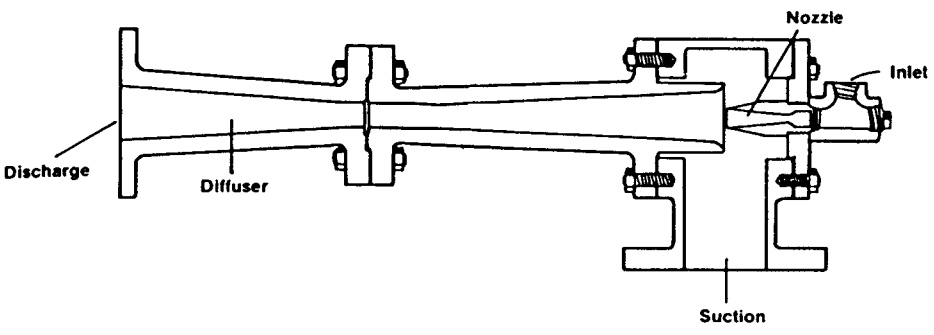
An ejector works by converting the pressure energy of a motive fluid into kinetic energy (velocity) as it flows through a relatively small converging — diverging nozzle. See Fig. 13-67. The increased velocity of the motive fluid causes a corresponding reduction in pressure creating suction in the mixing chamber, into which the process fluid is drawn. The process fluid mixes with and becomes entrained in the motive fluid stream. The mixed fluid then passes through the converging — diverging diffuser, where the velocity is converted back into pressure energy. The resultant discharge pressure is higher than the suction pressure of the ejector.

Historically, ejectors have been primarily used to generate vacuum. Ejectors can be staged in series to achieve deep vacuum levels. They can use many different types of motive fluid. Air and steam are the most common. To avoid contamination and other problems, it is important to choose a motive fluid compatible with the process fluid. Ejectors are one of the few compressor types that are relatively immune to liquid carryover in the suction gas.

An ejector is not as efficient as most types of mechanical compressors but has the advantage of simplicity and no moving parts. This helps make ejectors very reliable with minimal maintenance costs.

FIG. 13-67

Cross Section of an Ejector



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Turboexpanders

The use of turboexpanders in gas processing plants began in the early sixties. By 1970, most new gas processing plants for ethane or propane recovery were being designed to incorporate the particular advantages characteristic of an expander producing usable work. The trend in the gas processing industry continues toward increased use of the turboexpander.

Selection of a turboexpander process cycle is indicated when one or more of the following conditions exist:

1. "Free" pressure drop in the gas stream.
2. Lean gas.
3. High ethane recovery requirements (i.e., over 30% ethane recovery).
4. Compact plant layout requirement.
5. High utility costs.
6. Flexibility of operation (i.e., easily adapted to wide variation in pressure and products).

There are multiple factors in addition to the ones listed above that affect a final process selection. If two or more of the above conditions are coexistent, generally a turboexpander process selection will be the best choice.

Fig. 13-68 shows a typical low temperature turboexpander process for recovering ethane and heavier hydrocarbons from a natural gas stream.

Fig. 13-69 represents the pressure-temperature diagram for this expander process. The solid curve represents the plant inlet gas. The solid line on the right is the dew point line. At a fixed pressure and, if the temperature of the gas is to the right of this dew point line, the gas is 100 percent vapor. If the gas is cooled, liquid starts to condense when the temperature reaches the dew point line. As cooling continues, more liquid is condensed until the bubble point line is reached — the solid line on the left. At this point, all of the gas is liquid. Additional cooling results in colder liquid.

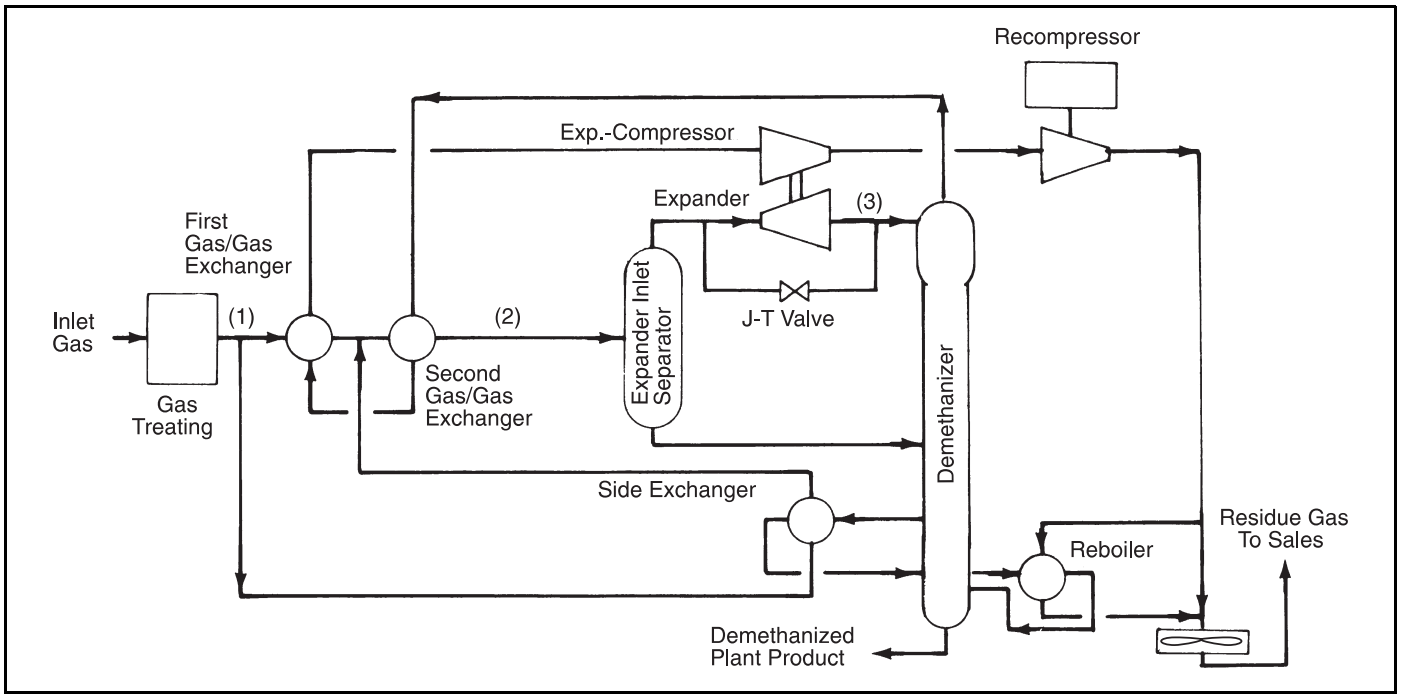
Downstream of the gas treating facilities, the inlet gas is represented by point 1 on both Fig. 13-68 and 13-69. As the gas is cooled by the gas/gas exchangers and demethanizer side exchanger, its temperature moves along the dotted line to point 2 (Fig. 13-69). At 2, the gas enters the expander inlet separator where the condensed liquid is separated from the vapor. This vapor now has its own pressure-temperature diagram, as represented by the dashed curve. At the expander inlet, the gas is on its dew point line.

As the gas flows through the expander, its pressure-temperature path is shown by the dashed line from point 2 to point 3. Point 3 represents the outlet of the expander. The importance of using the expander as a driver for a compressor can be seen in Fig. 13-69. If the gas had been expanded without doing any driver work, the expansion path would be from point 2 to point 4. This is called a Joule-Thomson, or constant enthalpy expansion. The outlet temperature and pressure would be higher than that accomplished in the expander (nearly isentropic) expansion process.

Note that the pressure at Point 4 is not as low as that attained by flow through the expander (Point 3). This is because it has been assumed for this example that, without the expander running (therefore the brake compressor also not running), the process cannot restore the demethanizer overhead vapor to the residue gas pressure using the separate recompressor alone.

FIG. 13-68

Example Expander Process



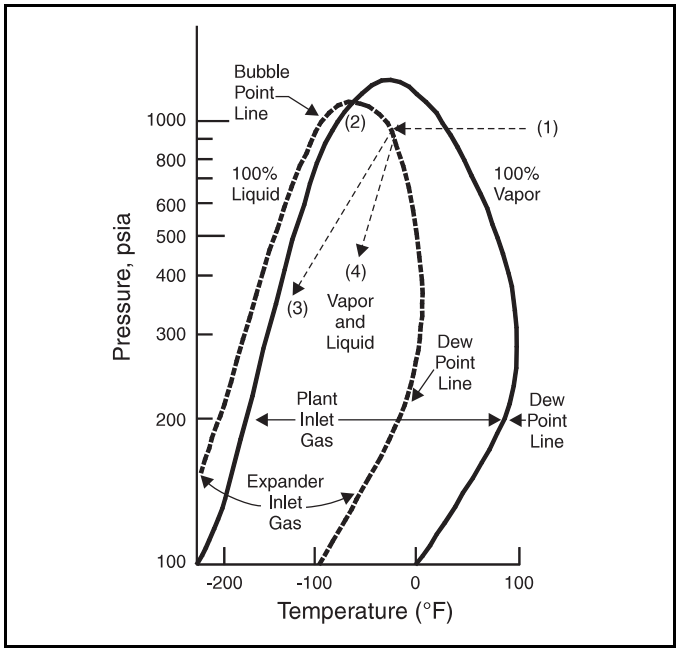
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Also, because the path to Point 4 is adiabatic without the gas doing work, the gas does not cool to as low a temperature as the path to Point 3. That is, the path (2) to (3) is isentropic expansion producing work and thereby cooling the gas more than the simple isentropic expansion path.

The higher temperature results in a reduction of product recovery. The use of the expander brake compressor to boost the residue gas pressure will allow a lower expansion pressure without the use of more residue compression.

FIG. 13-69

Pressure-Temperature Diagram for Expander Process



THERMODYNAMICS

A turboexpander recovers useful work from the expansion of a gas stream. The process operates isentropically in the ideal case and produces something less than the theoretical work in the real case. In the process of producing work, the expander lowers the bulk stream temperature which can result in partial liquefaction of the bulk stream. A simple schematic of an expander is given in Fig. 13-70.

An example calculation of an expander operating in a natural gas plant follows. The GPSA method is a hand calculation process which relies on various K-value and thermodynamic property charts available in the GPSA Data Book. Due to the limits of accuracy in reading these charts, it is impossible to duplicate all of the numbers in the example problem in spite of using each time the same step-by-step method. For all intent and purposes, the example calculation can be regarded as sufficiently correct with a discrepancy of less than 3% for the final result of the expander horsepower.

A simplified 3-component mixture will be used, as the example is general for all mixtures. The example involves the use of the K-value charts (see Section 25) and the calculation of stream enthalpies and entropies.

Thermodynamic properties are taken from Section 24. K-values are calculated from the data on the binary systems methane-ethane, and methane-propane, and the infinite dilution ternary system.

Gas inlet conditions (t_1, P_1) to the expander are generally set by process balances. The outlet pressure P_2 from the expander is set by process and recompressor power considerations. Fig. 13-61 gives stream conditions with an example calculation.

Outlet conditions for the expander must be determined by trial-and-error calculations. Assume three different outlet temperatures, t′, t″, t‴ at the set outlet pressure of P_2 [2000 kPa (abs)], from the expander. At each assumed temperature and P_2, calculate the equilibrium conditions of the bulk stream from the expander, and determine the stream bulk enthalpy (h) and entropy (s).

For most natural gas applications, including this example, a limiting boundary value for the outlet temperature can be obtained from the methane P-H diagram. From this chart, determine the ideal outlet temperature t_2 by isentropic expansion from inlet t_1, P_1 to outlet P_2. In most real gas mixtures, t_2 will be somewhat warmer than this. In the example case, t_2 for pure methane is –159°F. Therefore, assume t′ = –150°F, t″ = –140°F, t‴ = –130°F.

Draw a plot of $\Delta h(h_{t1P1} - h_{t2P2})$ versus temperature at P_2, and s versus outlet temperature at P_2, using stream bulk properties as calculated in Fig. 13-71. This plot is illustrated in Fig. 13-72.

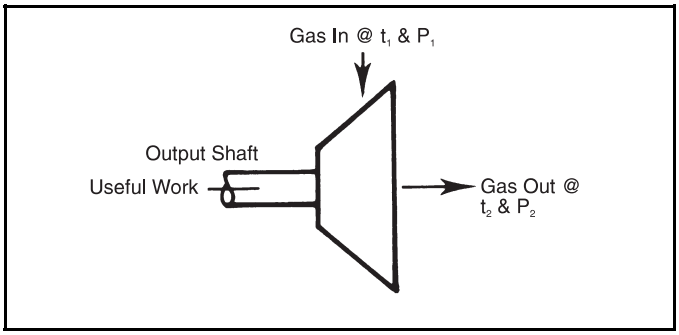
From the plot of the calculated s versus outlet temperature at P_2 (300 psia), determine the isentropic outlet temperature, using s_1 (32.58) from the inlet conditions. From this outlet temperature, read the corresponding Δh ideal, which is the ideal work available from a 100% efficient isentropic process. Since the expander operates at something less than ideal conditions, an efficiency must be used to determine actual expander outlet conditions. If an expander efficiency of 80% is used, the actual $\Delta h = 0.8 \times \Delta h$ ideal $= \Delta h_A$. Using Δh_A from the plot, read an outlet temperature of t_2 (–135.2°F). Now make an equilibrium flash calculation at conditions t_2, P_2 –135.2°F, (300 psia) to find the real outlet conditions for the expander, operating at 80% efficiency.

The total work produced by the expander based on the assumed efficiency can be calculated as shown in Fig. 13-71.

A schematic P-H diagram path for the example expander calculation is shown in Fig. 13-73.

FIG. 13-70

Simple Expander



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FIG. 13-71

Expander Example Calculation

Flow: 60 MMscfd t_1 = –60°F

P_1 = 900 psia

P_2 = 300 psia

	mols/hr
Methane	6,185.8
Ethane	263.5
Propane	138.3
Total mols/hr	6,587.6
Total lbs/hr	113,261.0

Inlet Conditions:

All enthalpies (h) are calculated from Figs. 24-3, 24-6, and 24-7, and all entropies (s) from Figs. 24-18, 24-19, and 24-20. See Fig. 24-17 for example calculation of h and Fig. 24-2 for example calculation of s.

Inlet Enthalpy, (h)	118.134 Btu/lb
Inlet Entropy, (s)	32.58 Btu/(mol • °R)

Outlet Conditions

	-150°F 300 psia	-140°F 300 psia	-130°F 300 psia
K-value for methane	1.1589	1.3680	1.6023
K-value for ethane	0.0813	0.0948	0.1127
K-value for propane	0.00974	0.01022	0.01235
Outlet liquid, lbs/hr	48,380	23,958	15,720
Outlet vapor, lbs/hr	64,882	89,303	97,541
Outlet liquid enthalpy (h) Btu/lb	–53.202	–68.675	–77.500
Outlet vapor enthalpy (h) Btu/lb	120.702	127.901	134.301
Outlet liquid entropy (s) Btu/(mol • °R)	25.557	25.916	26.220
Outlet vapor entropy (s) Btu/(mol • °R)	33.172	33.688	34.147
Mixture h, Btu/lb	46.419	86.319	104.904
Mixture s, Btu/(mol • °R)	30.198	32.383	33.366
h inlet – h outlet = Δ h, Btu/lb	71.715	31.815	13.230

See Fig. 13-72

Δ h ideal = 27.688 Btu/lb

Δ h actual = 0.8 efficiency x 27.688 = 22.150 Btu/lb

t_2 at 80% expander efficiency = –135.2°F @ 300 psia.

Flash @ Actual Outlet Conditions:

	K at –135.2°F 300 psia	Liquid	Vapor
Methane	1.4726	567	5,619
Ethane	0.1019	156	107
Propane	0.0104	129	9
Total mols/hr		852	5,735
Total lbs/hr		19,477	93,784

Work produced = 22.150 Btu/lb x 113,261 lb/hr = 2,509,000 Btu/hr

Horsepower = 2,509,000 ÷ 2,545 = 986 hp

In many applications the loading device for the turboexpander is a centrifugal compressor and its performance can be calculated as shown in this section. Shaft and bearing losses in the order of 2% are usually deducted to calculate net power input to the driven end from the expander.

FIG. 13-72

T-h and T-s Diagram

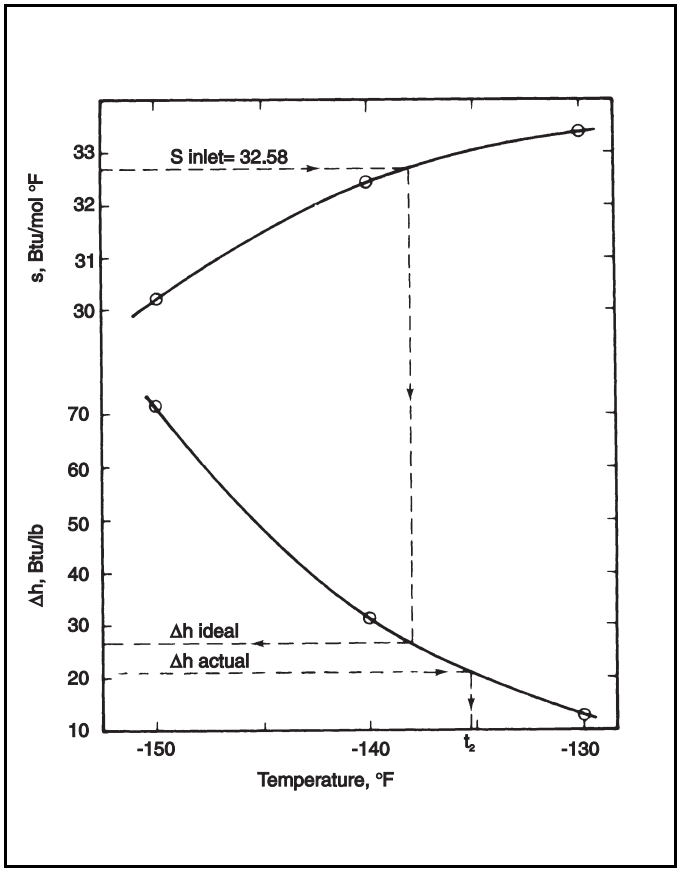
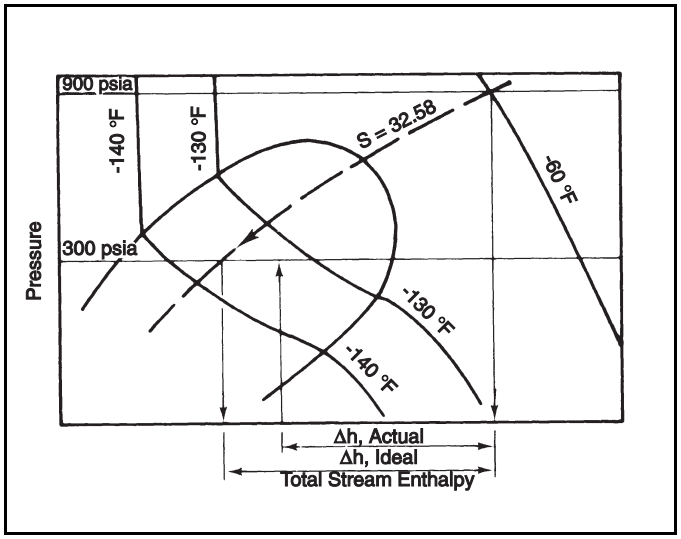


FIG. 13-73

Schematic P-H Diagram for Expander



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MECHANICAL

Mechanical design of the turboexpander is the business of several manufacturers. Any specific information must come from such supplier.

Of the various general turbine types available, the radial reaction turbine design is dominant in cryogenic turboexpander natural gas plant applications. These units operate over wide ranges of inlet flow and pressure conditions, by utilizing variable inlet guide vanes. They operate at very high rotating speeds and thus are subject to the design and operating cautions common to similar sophisticated rotating equipment.

The most common configuration is a turboexpander-compressor where the expander power is used to compress gas in the process. In this case, the compressor wheel operates on the same shaft as the expander wheel. Other applications of the power recovery are expander-pump or expander-generator drives. These normally require gearing to reduce the expander speed to that required for the driven unit.

Since power recovery and refrigeration effect are primary benefits of expander applications, rotating speeds are set to optimize the expander efficiency. This will usually result in a compromise in the compressor end design and lower compressor efficiencies. Usual efficiencies quoted for radial type units are 75 to 85% for the expander and 65 to 80% for the compressor.

Some areas requiring extra attention in the installation of turboexpanders are listed below. The list is by no means comprehensive, but these items require more than the normal amount of concern in designing the installation of a turboexpander unit for cryogenic operation.

1. The expander inlet gas stream must be free of solid or liquid entrainment. Liquids are removed in a high pressure separator vessel. An inlet screen of fine mesh is usually required for solids removal. Monitoring of the pressure drop across this screen is recommended. Formation of solids (ice, carbon dioxide, amines, heavy oils) will often occur here first and can be detected by an increase in pressure drop across the screen.
2. Source of the seal gas, particularly during start-up, is an important consideration. The stream must be clean, dry, sweet, and of sufficient pressure to meet the system requirements.
3. Normally a quick closure shutoff valve is required on the expander inlet. Selection of this valve and actuator type must take into account start-up, operating, and shutdown conditions.
4. Vibration detection instrumentation is useful but not mandatory. Its application is normally an owner and vendor option and influenced by operating economics.
5. Loading of the flanges by the process piping system must be within prescribed limits to avoid distortion of the case, resulting in bearing or wheel rubbing problems.
6. Failures due to mechanical resonance have occurred in turboexpanders. Even though the manufacturer will exert his best efforts at the manufacturing stage to avoid this problem, in-plant operation may uncover an undesirable resonance. This must be solved in conjunction with the manufacturer and may involve a redesign of the wheels, bearing modifications, vane or diffuser redesign, etc.

The installation of a turboexpander-compressor unit also requires the proper design of a lube system, instrumentation, etc., in common with other industrial rotating equipment. It is common practice to install a turboexpander-compressor with no special anti-surge instrumentation for the compressor unit. This is acceptable if it can be determined that the gas flow through the compressor is balanced with flow through the expander and the two will vary simultaneously.

Auxiliary Systems

Both lubricated and non-lubricated turboexpander designs are available.

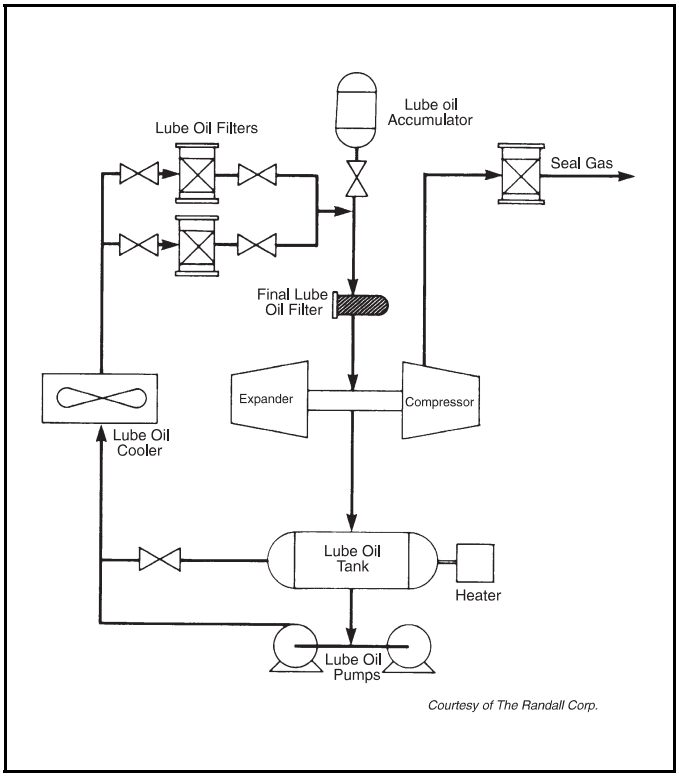
Lubrication System — The lubrication system circulates cooled and filtered lube oils to the turboexpander bearings as shown on Fig. 13-74. The principle components of the system are monitored on the lube console and normally consist of two electric motor-driven lube oil pumps, an oil cooler, a dual filter valve, a bladder type with switching coastdown accumulator, and a pressurized reservoir with mist eliminator.

The lube oil pumps (one stand-by) must maintain a constant flow to the radial and thrust bearings. Absence of oil, or improper filtration, can cause bearing damage. Most manufacturers recommend a light turbine oil (315 SSU at 100°F) for best machine performance.

The lube oil cooler is an integral part of the system to reject heat that is generated across the bearings. It can be of a fan air cooled type or shell and tube design, water cooled. If the cooling water is scale forming, duplicate coolers (one stand-by) are recommended.

FIG. 13-74

Lube Oil Schematic



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Lube oil filtration is extremely important due to close tolerances between bearing surfaces.

The lube-oil reservoir serves as a surge tank to enhance pump suction as well as to serve as a degassing drum permitting process seal gas to be released from the oil. If necessary, the reservoir should be equipped with a heater to bring the oil up to temperature for a "cold" start.

Seal Gas System — The seal gas system prevents loss of process gas and assures protection against entry of lube oil into process gas areas. To accomplish this, a stream of "seal gas" is injected into each labyrinth shaft seal at a pressure higher than that of the process gas. The leaking seal gas is collected in the oil reservoir, then returned through a mist eliminator to the fuel gas system, or put back into the compressor suction end.

The system for seal gas injection consists of a liquid collector, electric heater (if required), twin filters, and differential pressure regulators.

If recompression is necessary for the gas processing plant, sales gas is ideal for use as seal gas. If no recompression is provided, a stream can be taken from the expander inlet separator, warmed and used as seal gas. A minimum seal gas temperature (about 70°F) is required to prevent oil thickening.

Seal gas filtration is essential because of close clearances provided between the shaft and seals.

Seal gas flow requirements are determined by the expander manufacturers as a part of their performance rating.

Control Systems

Process — Control of the process streams begins with proper dehydration and filtering. Generally a final protective screen upstream of the expander is designed into the piping system to form a protective barrier against carbon dioxide or water freezing.

As a further protection against water freezing, methanol injection connections are incorporated into the system upstream of the expander.

Machine — The expander speed is established by the manufacturer, given the process conditions. For maximum efficiency, the expander manufacturer determines the wheel diameter and specific speed.

As plant operating conditions change, the expander speed may change. Fig. 13-75 shows the change in efficiency as a function of change in design flow rate.

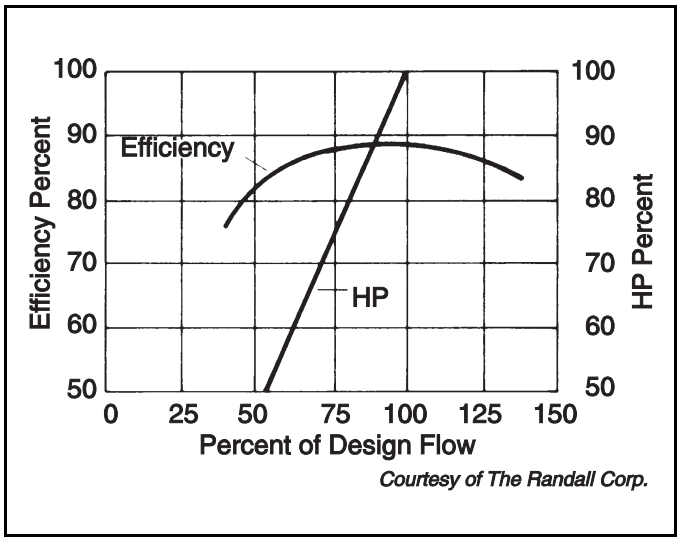
Gas entering the expander is directed by adjustable nozzles into the impeller. About one-half of the pressure drop across the expander takes place in the nozzles, imparting kinetic energy to the gas which is converted to shaft horsepower by the expander wheel. Pressure reductions are normally limited to 3-4 ratios. Greater ratios reduce expander efficiency to the extent that 2-stage expansion may be advisable.

The adjustable nozzles function as pressure control valves. A pneumatic operator takes a split range signal (3 to 9 psi) to stroke the nozzles. On increasing flow beyond the full open nozzle position, a 9 to 15 psi signal from a pressure controller opens a bypass control valve. This valve is called the J-T (Joule-Thomson) valve.

Thrust bearing force imbalance is caused by difference in pressures between the expander discharge and compressor suction. With a differential of the order of 20 psi, the thrust loads are usually within the capabilities of the thrust bearings. At higher pressure differentials, it is essential that steps be taken to control the thrust loads against each other, thereby the net thrust load will not exceed the thrust bearing capacity.

FIG. 13-75

Example Change in Efficiency with Flow Rate



Courtesy of The Randall Corp.

This is done by providing a force-measuring load-meter on each thrust bearing, Fig. 13-76, and a thrust control valve which controls the thrust by control of pressure behind the thrust balancing drums or behind one of the seals. These two load-meters indicate thrust bearing oil film pressure (proportional to bearing load) and the third shows the pressure behind the balancing drum as controlled by the valve in its vent as a means of adjusting the thrust load.

Vibration comes from an unbalanced force on one of the rotating components, or it could come from an outside source such as pipe vibration or gas pulsation.

Most expanders are supplied with monitoring and shutdown devices for shaft vibration. These devices are set to shut down the expander before damage occurs.

Lube Oil — The lube oil must be filtered. Most systems use a primary and secondary filtering system. Controls are provided to ensure oil flow to bearings at proper pressure and temperature. Two (2) lube oil pumps are furnished, the second pump serving as a standby. The standby oil pump is controlled automatically to cut in to provide oil pressure upon failure of the main pump or reduction in pressure for other reasons.

Generally an oil flow bypass valve is included to permit excess flow to bypass the expander bearings and return to the reservoir.

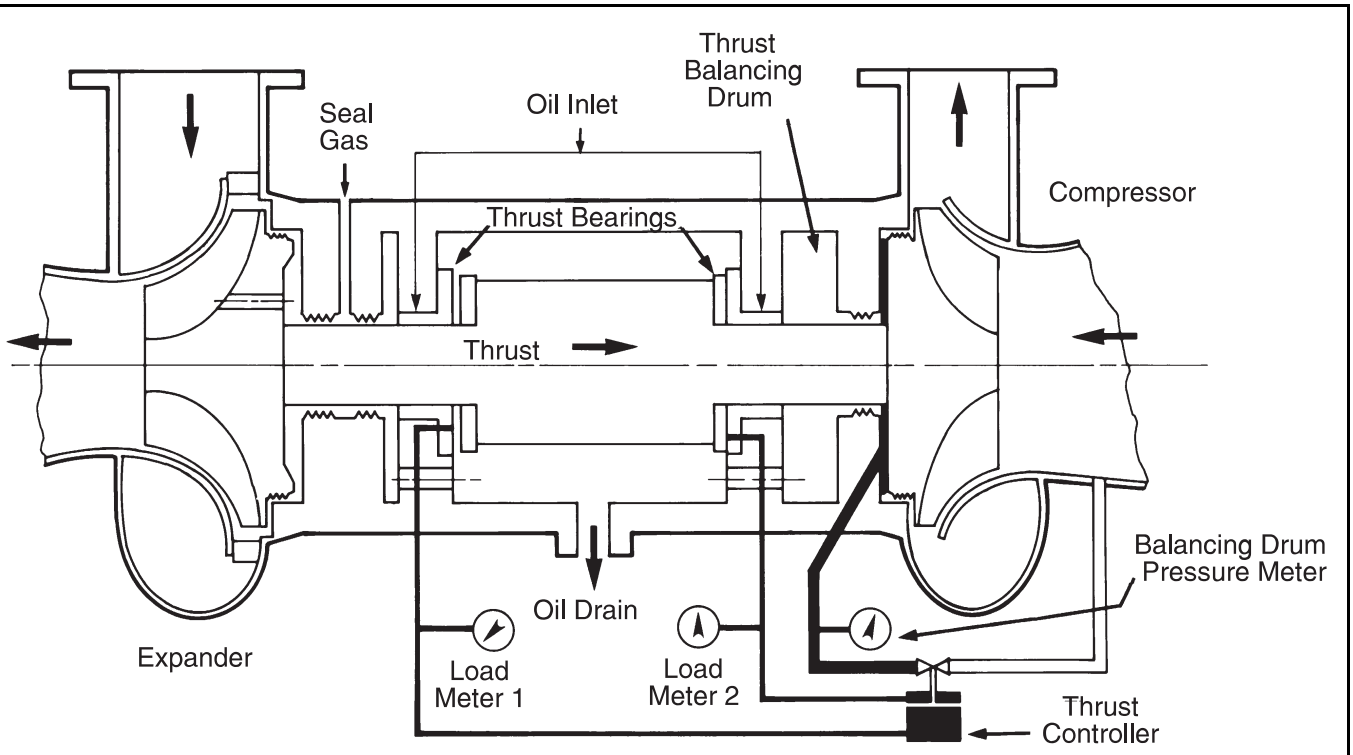
For temperature control, the oil must be cooled to prohibit heat buildup which occurs through the bearings. Also, a temperature control bypass is included in the circuit for an extra measure of control to keep the oil from getting too cool.

Seal Gas — Use a suitable gas stream with filtering and pressure control to maintain proper gas pressure at the shaft seals.

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FIG. 13-76

Typical Expander/Compressor Cross-Section with Thrust Balancing Schematic



Courtesy of The Randall Corp.

If the seal gas is delivered from a cold supply point (expander inlet separator) then a means of heating the gas is necessary.

The seal gas should be introduced before the lube oil system is started because there might be a pressure upset which would put enough oil into the process to cause a problem.

Each of the main rotating components (radial bearings, thrust bearings, and shaft seals) can be damaged or eroded by improper oil filtration, lack of oil flow, improper gas dehydration, and improper seal gas filtration.

Shutdown — A number of conditions during the operation of expanders justify prompt shutdown to avoid serious damage.

Some of these conditions are:

- High Vibration
- Low Lube Oil Flow
- High Inlet Separator Level
- High Inlet Screen Pressure Drops
- High Thrust
- High Lube Oil Temperature
- Low Lube Oil Pressure
- High Speed

Two primary actions of a shutdown signal are to block gas flow to the expander and the compressor. This is accomplished by actuating quick acting shutdown valves at the expander inlet and outlet and the compressor inlet. Simultaneously, a pressurized bladder supplies oil to the bearings during the expander coast down. The expander bypass valve (J-T) opens automatically and is positioned by the split-range pressure controlled to keep the plant on-line in the J-T mode.

Field Performance — Field measurements can be made to check efficiencies and horsepower of the expander. The process of calculations is just the reverse of selecting a machine performance.

Knowing the gas composition, mass flow (lbs/hr), inlet and outlet conditions (pressure, temperature) for the expander, the actual difference in enthalpy can be determined for each unit.

Thus:

$$\Delta h_{actual} = h_{t_2P_2} - h_{t_1P_1}$$

$$\eta = \frac{\Delta h \text{ actual}}{\Delta h \text{ ideal}}$$

$$EP_{actual} = \frac{\Delta h \text{ actual} \bullet \text{lbs/hr}}{2{,}545}$$

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API 614 - “Lubrication, Shaft-Sealing and Control Oil Systems for Special-Purpose Applications”.

API 617 - “Axial and Centrifugal Compressors and Expander Compressors for Petroleum, Chemical and Gas Industry Services."

API 618 - “Reciprocating Compressors for General Refinery Services”.

API 619 - "Rotary Type Positive Displacement Compressors for Petroleum, Chemical and Gas Industry Services."

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NOTES

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SECTION 14

Refrigeration

Refrigeration systems are common in the natural gas processing industry and processes related to the petroleum refining, petrochemical, and chemical industries. Several applications for refrigeration include NGL recovery, LPG recovery, hydrocarbon dew point control, reflux condensation for light hydrocarbon fractionators, and LNG plants.

Selection of a refrigerant is generally based upon temperature requirements, availability, economics, and previous experience. For instance, in a natural gas processing plant, ethane and propane may be at hand; whereas in an olefins plant, ethylene and propylene are readily available. Propane or propylene may not be suitable in an ammonia plant because of the

FIG. 14-1
Nomenclature

GHP = gas horsepower defined by Eq 14-7b
h = enthalpy, Btu/lb
h'_{VD} = isentropic enthalpy, Btu/lb
k = specific heat ratio
m = refrigerant flow, lb/hr
n = number of stages
P = pressure, psia
Q = heat duty, Btu/hr
r = compression ratio
S = entropy, Btu/(lb • °R)
T = temperature, °F
W = work of compression, Btu/hr
X = weight fraction
η = isentropic efficiency
σ = surface tension, dynes/cm
ρ = density, lb/ft^3
Δh = enthalpy change, Btu/lb

Subscripts

L = liquid state
V = vapor state
A, B, C, D = denote unique points of operation on P-H diagrams
b = bypass
i = isentropic
cd = condenser
ref = refrigeration
s = suction
d = discharge
n = stage number
T = Total
1, 2 = stage number

Accumulator: a storage vessel for liquid refrigerant; also known as surge drum.

Bubble point: the temperature at which the vapor pressure of the liquid refrigerant equals the absolute external pressure of the liquid-vapor interface.

Capacity, refrigerating system: the cooling effect produced by the total enthalpy change between the refrigerant entering the evaporator and the refrigerant leaving the evaporator.

Chiller, Evaporator: a heat exchanger in which the liquid refrigerant is vaporized by a process stream which is in turn cooled.

Compression ratio: ratio of outlet to inlet absolute pressures for a compressor.

Condenser: a heat exchanger in which the refrigerant, compressed to a suitable pressure, is condensed by rejection of heat to a cooling medium.

Cooling medium: any substance whose temperature is such that it is used, with or without change of state, to lower the temperature of refrigerant either during condensing or subcooling.

Effect, refrigerating: the rate of heat removal by a refrigerant in a refrigeration system. It is equal to the difference in specific enthalpies of the refrigerant at two designated thermodynamic states.

Expansion valve: a valve for controlling the flow of refrigerant to an evaporator or chiller.

Flash gas: the gas resulting from the instantaneous evaporation of refrigerant by a pressure reducing device, such as a control valve.

Frost Plug: small diameter closed nozzle protruding from the side of an insulated vessel which indicates liquid level in the vessel by accumulation of frost.

Halocarbons: a family of refrigerants consisting of fluorinated and/or chlorinated hydrocarbons.

Hot gas bypass: warm discharge gas recycled to chiller for maintaining system's operating integrity at minimum load conditions.

Liquid refrigerant receiver: a vessel in a refrigeration system designed to ensure the availability of adequate liquid refrigerant for proper functioning of the system and to store the liquid refrigerant when the system is pumped down.

Refrigerant: the fluid used for heat transfer in a refrigeration system, which absorbs heat at a low temperature and low pressure and rejects heat at a higher temperature and a higher pressure.

Ton of refrigeration: amount of heat required to melt 1 ton of ice in 24 hours, equivalent to 12,000 Btu/hr at 32°F.

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risk of contamination, while ammonia may very well serve the purpose. Halocarbons have been used extensively because of their nonflammable characteristics.

Fig. 14-1 provides a nomenclature and glossary for this section.

MECHANICAL REFRIGERATION

Refrigeration Cycle

The refrigeration effect can be achieved by using one of these cycles:

- Vapor compression-expansion
- Absorption
- Steam jet (water-vapor compression)

By utilizing the Pressure-Enthalpy (P-H) diagram, the refrigeration cycle can be broken down into four distinct steps:

- Expansion
- Evaporation
- Compression
- Condensation

The vapor-compression refrigeration cycle can be represented by the process flow and P-H diagram shown in Fig. 14-2.

Expansion Step — The starting point in a refrigeration cycle is the availability of liquid refrigerant. Point A in Fig. 14-2 represents a bubble point liquid at its saturation pressure, P_A, and enthalpy, h_{LA}. In the expansion step, the pressure and temperature are reduced by flashing the liquid through a control valve to pressure P_B. The lower pressure, P_B, is determined by the desired refrigerant temperature, T_B (point B).

At point B the enthalpy of the saturated liquid is h_{LB}, while the corresponding saturated vapor enthalpy is h_{VB}. Since the expansion step (A – B) occurs across an expansion valve and no energy has been exchanged, the process is considered to be isenthalpic. Thus the total stream enthalpy at the outlet of the valve is the same as the inlet, h_{LA}.

Since point B is inside the envelope, vapor and liquid coexist. In order to determine the amount of vapor formed in the expansion process, let X be the fraction of liquid at pressure P_B with an enthalpy h_{LB}. The fraction of vapor formed during the expansion process with an enthalpy h_{VB} is (1-X). Equations for the heat balance and the fraction of liquid formed are:

$$(X)\, h_{LB} + (1 - X)\, h_{VB} = h_{LA} \qquad \textbf{Eq 14-1}$$

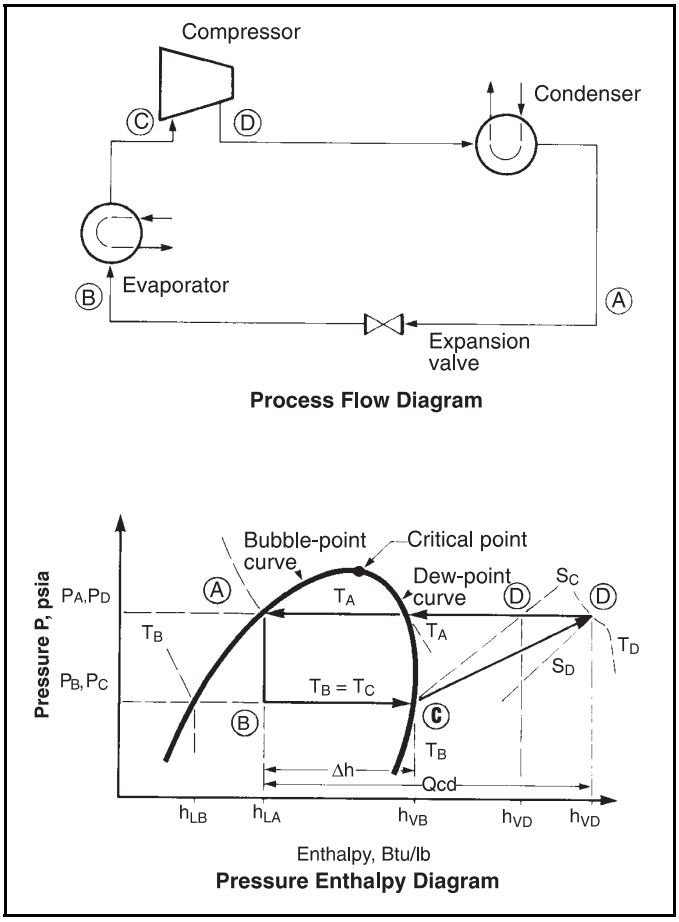
$$X = \frac{(h_{VB} - h_{LA})}{(h_{VB} - h_{LB})} \qquad \textbf{Eq 14-2}$$

$$(1 - X) = \frac{(h_{LA} - h_{LB})}{(h_{VB} - h_{LB})} \qquad \textbf{Eq 14-3}$$

Evaporation Step — The vapor formed in the expansion process (A-B) does not provide any refrigeration to the process. Heat is absorbed from the process by the evaporation of the liquid portion of the refrigerant. As shown in Fig. 14-2, this is a constant temperature, constant pressure step (B-C). The enthalpy of the vapor at point C is h_{VB}.

Physically, the evaporation takes place in a heat exchanger referred to as an evaporator or a chiller. The process refrigeration is provided by the cold liquid, X, and its refrigerant effect

FIG. 14-2
Process Flow Diagram and Pressure-Enthalpy Diagram



can be defined as X ($h_{VB} - h_{LB}$) and substituting from Eq. 14-2, the effect becomes:

$$\text{Effect} = h_{VB} - h_{LA} \qquad \textbf{Eq 14-4}$$

The refrigeration duty (or refrigeration capacity) refers to the total amount of heat absorbed in the chiller by the process, generally expressed as "tons of refrigeration," or Btu/unit time. The refrigerant flow rate is given by:

$$m = \frac{Q_{ref}}{(h_{VB} - h_{LA})} \qquad \textbf{Eq 14-5}$$

Compression Step — The refrigerant vapors leave the chiller at the saturation pressure P_C. The corresponding temperature equals T_C at an enthalpy of h_{VB}. The entropy at this point is S_C. These vapors are compressed isentropically to pressure P_A along line C – D′ (Fig. 14-2).

The isentropic (ideal) work, W_i, for compressing the refrigerant from P_B to P_A is given by:

$$W_i = m\,(h'_{VD} - h_{VB}) \qquad \textbf{Eq 14-6}$$

The quantity h'_{VD} is determined from refrigerant properties at P_A and an entropy of S_C. Since the refrigerant is not an ideal fluid and since the compressors for such services do not operate ideally, isentropic efficiency, η_i, has been defined to compensate for the inefficiencies of the compression process. The actual work of compression, W, can be calculated from:

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$$W = \frac{W_i}{\eta_i} = \frac{m\,(h'_{VD} - h_{VB})}{\eta_i} = m\,(h_{VD} - h_{VB}) \qquad \textbf{Eq 14-7}$$

The enthalpy at discharge is given by:

$$h_{VD} = \frac{(h'_{VD} - h_{VB})}{\eta_i} + h_{VB} \qquad \textbf{Eq 14-7a}$$

The work of compression can also be expressed as:

$$GHP = \frac{W}{2544.4} \qquad \textbf{Eq 14-7b}$$

where 2544.4 Btu/hr = 1 hp.

(See Section 13 for a discussion on compressors.)

Condensation Step — The superheated refrigerant leaving the compressor at P_A and T_D (Point D in Fig. 14-2) is cooled at nearly constant pressure to the dew point temperature, T_A, and refrigerant vapors begin to condense at constant temperature.

During the desuperheating and condensation process, all heat and work added to the refrigerant during the evaporation and compression processes must be removed so that the cycle can be completed by reaching Point A (the starting point) on the P-H diagram, as shown in Fig. 14-2.

By adding the refrigeration duty to the heat of compression, we calculate the condensing duty, Q_{cd}, from:

$$Q_{cd} = m\,[(h_{VB} - h_{LA}) + (h_{VD} - h_{VB})]$$

$$= m\,(h_{VD} - h_{LA}) \qquad \textbf{Eq 14-8}$$

The condensing pressure of the refrigerant is a function of the cooling medium available — air, cooling water, or another refrigerant. The cooling medium is the heat sink for the refrigeration cycle.

Because the compressor discharge vapor is superheated, the refrigerant condensing curve is not a straight line. It is a combination of desuperheating and constant temperature condensing. This fact must be considered for proper design of the condenser.

System Pressure Drop — Some typical values for pressure drops that must be considered are:

Condenser pressure drop	3.0 to 7.0 psi
Line hydraulic losses	
Evaporator to Compressor*	0.1 to 1.5 psi
Compressor to Condenser	1.0 to 2.0 psi
Condenser to Receiver	0.5 to 1.0 psi

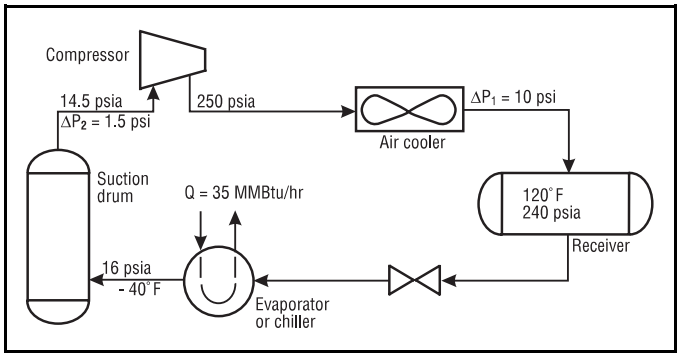
* This is an important consideration in refrigeration services with low suction pressure to compressor.

Refrigeration Stages

Refrigeration systems utilizing one, two, three, or four stages of compression have been successfully operated in various services. The number of levels of refrigeration generally depends upon the number of compression stages required, interstage heat loads, economics, and the type of compression.

One-Stage System — A typical one-stage refrigeration system is shown in Fig. 14-3 where the data are for pure propane refrigerant. Fig. 14-4 illustrates a process application of a single level chiller and the associated cooling curve.

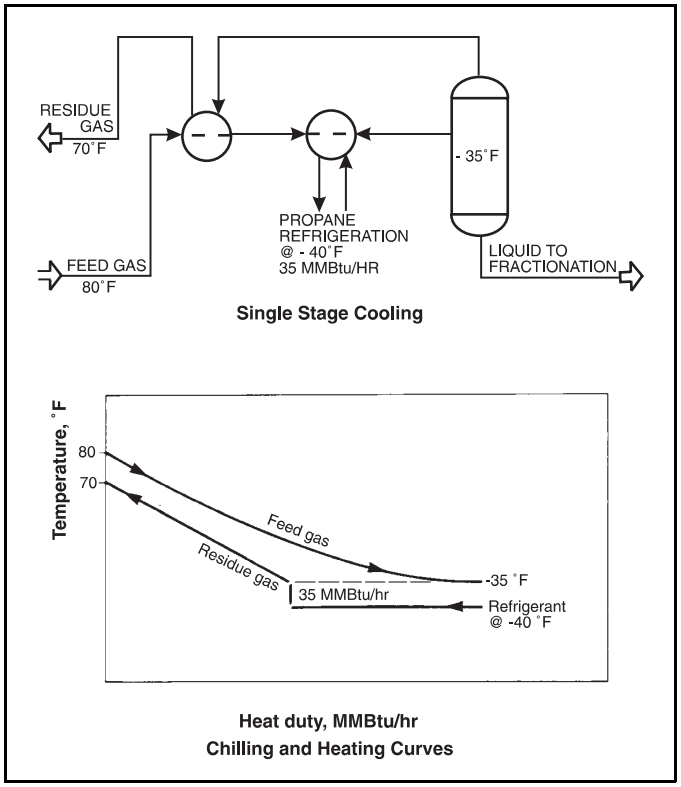
FIG. 14-3
One-Stage Refrigeration System



Two-Stage System—Savings in the 20% range can often be achieved with a two-stage refrigeration system and interstage flash economizer. Additional savings can be realized by removing process heat at the interstage level rather than at the low stage level. A typical two-stage system with an intermediate load is shown in Fig. 14-5 with data for pure propane.

Three-Stage System — Additional horsepower savings can be achieved by using a three-stage compression system. As with a two-stage system, flash economization and/or an intermediate heat load can be used. The savings, while not as dramatic as the two stage versus one-stage, can still be significant enough to justify the additional equipment. A typical three stage propane system is shown in Fig. 14-6.

FIG. 14-4
Single-Stage Cooling, Chilling and Heating Curves



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FIG. 14-5
Two-Stage Refrigeration System

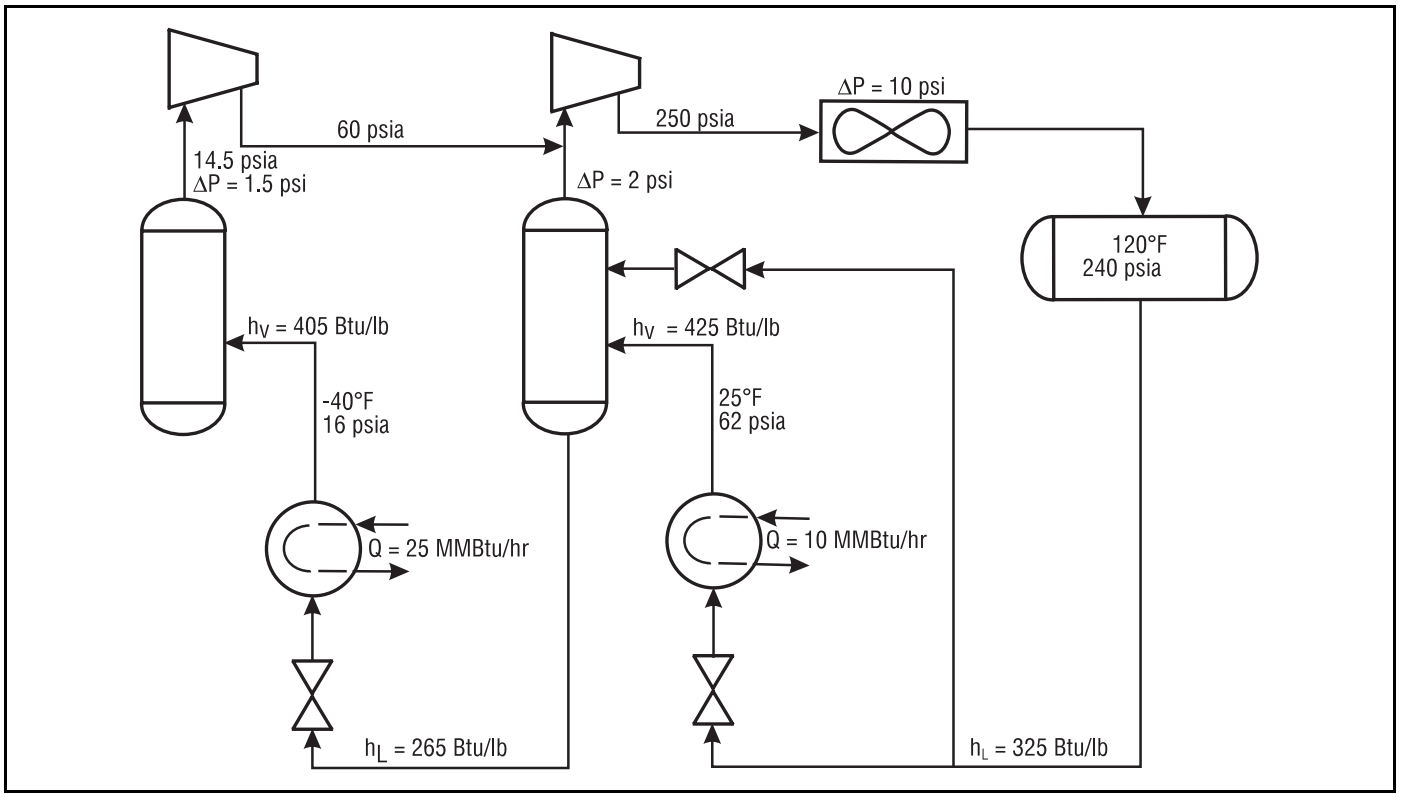
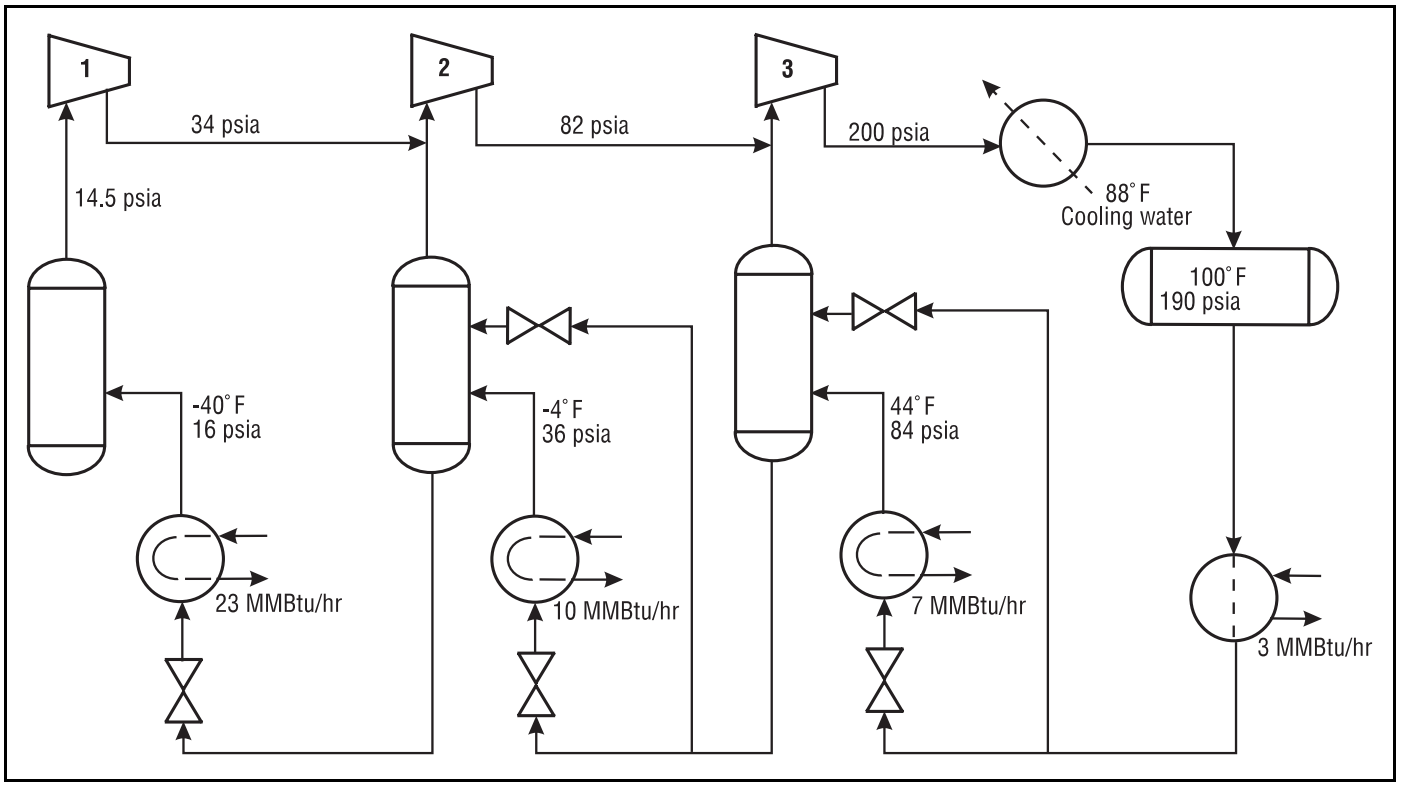


FIG. 14-6
Three-Stage Refrigeration System



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System Configuration — Energy consumption is frequently reduced as the number of stages is increased. For a propane refrigeration system, Fig. 14-7 illustrates the effect of interstages without using refrigeration at intermediate levels. However, the installation cost of such refrigeration systems increases as the number of stages increases. The optimum overall cost will be a function of the specific system and has to be determined for a set of economic criteria.

FIG. 14-7

Effect of Staging on a Propane Refrigeration System

	Stages, n		
	1	2	3
Refrigeration Duty, MMBtu/hr	1.0	1.0	1.0
Refrigeration Temperature, °F	–40	–40	–40
Refrigerant Condensing Temperature, °F	100	100	100
Compression Requirements, hp	292	236	224
Reduction in hp, %	Base	19.2	23.3
Condenser Duty, MMBtu/hr	1.743	1.600	1.575
Change in condenser duty, %	Base	–8.2	–9.6

The compression horsepower for refrigeration can be reduced further by shifting refrigerant load from cooler levels to warmer levels. Fig. 14-8 shows a refrigeration system using two levels of chilling. The gas is initially chilled to 30°F with 25°F propane and then to –35°F with –40°F propane. The selection of the 25°F level results from equal compression ratios for each stage. The interstage pressure and corresponding refrigerant temperature may be fixed by either equipment or process conditions. Equal compression ratios per stage are chosen whenever possible to minimize horsepower.

Example 14-1 — Calculate the horsepower and condenser duty required for the process shown in Fig. 14-8 using propane refrigeration. Design condensing temperature is 120°F. The pressure drop from the chillers to the compressor suction is 1.5 psi. The pressure drop from compressor discharge to the receiver is 10 psi.

Solution Steps:

In order to determine the interstage refrigeration level for a two-stage system, determine the ratio per stage:

$$r = \left(\frac{P_d}{P_s}\right)^{1/n} \qquad \textbf{Eq 14-9}$$

From the propane vapor pressure curve:

$$P_d = 240 \text{ psia} + 10 \text{ psi} = 250 \text{ psia}$$

$$P_s = 16 \text{ psia} - 1.5 \text{ psi} = 14.5 \text{ psia}$$

$$r = \left(\frac{250}{14.5}\right)^{1/2} = 4.15$$

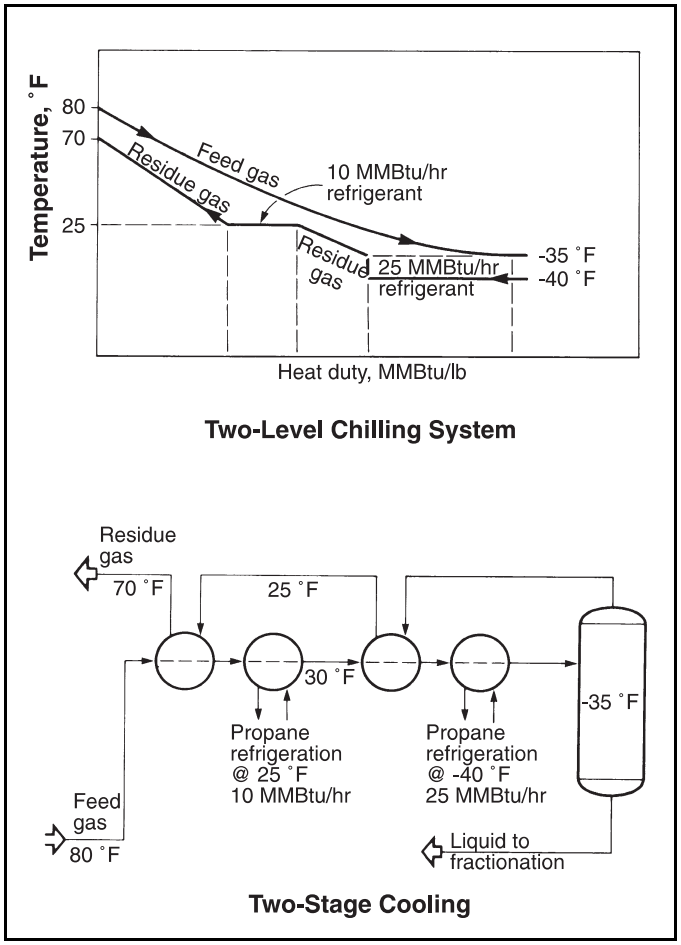
Thus the second stage suction pressure is:

$$P_{s2} = (14.5)(4.15) = 60 \text{ psia}$$

The first stage discharge pressure is:

FIG. 14-8

Two-Level Chilling, Two-Stage Cooling System



$$P_{dl} = 60 + 2.0 = 62 \text{ psia}$$

From the vapor pressure curve for propane, the refrigeration temperature at 62 psia is 25°F. Substituting enthalpy values from Section 24, Fig. 24-27 into Eq 14-5, we find the refrigerant flowrate through each chiller:

$$m_1 = \frac{(25)(10^6)}{(405 - 265)} = 178{,}570 \text{ lb / hr}$$

$$m_2 = \frac{(10)(10^6)}{(425 - 325)} = 100{,}000 \text{ lb / hr}$$

where m_1 is the flowrate through the first stage chiller, and m_2 is the flowrate through the second stage chiller.

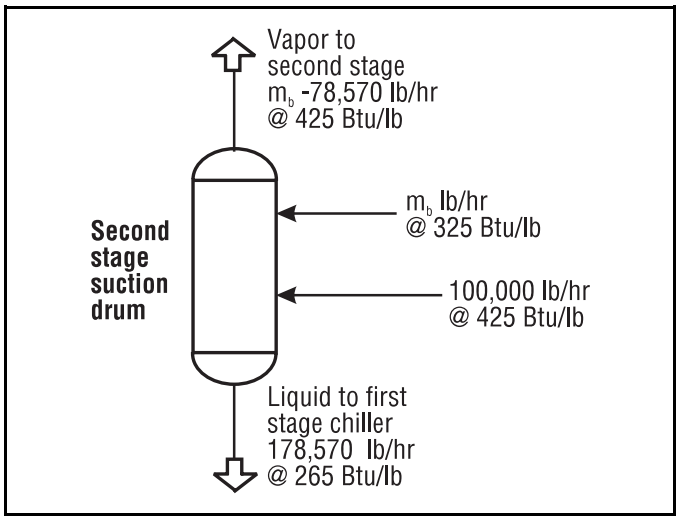
Liquid flow to the first-stage chiller (178,570 lb/hr) is provided by flashing the liquid refrigerant from the refrigerant receiver at 120°F and bypassing the second-stage chiller.

In order to determine the flow of liquid refrigerant from the receiver, consider the heat and material balances shown in Fig. 14-9. Here, let m_b (lb/hr) denote the refrigerant bypassing the second-stage chiller. The chiller produces 100,000 lb/hr of refrigerant vapor at 25°F. These vapors flow through the second stage suction drum, and leave overhead. The liquid required from the second stage flash drum for the first stage chiller comes from the quantity m_b.

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FIG. 14-9

Data for Heat and Material Balances



By material balance, we find the vapors leaving the second stage suction drum as $m_b + 100{,}000 - 178{,}570$ or $m_b - 78{,}570$ lb/hr. By heat balance around the suction drum, we can determine the amount of refrigerant, m_b:

$$(m_b - 78{,}570)\,(425) + (178{,}570)\,(265) = m_b\,(325) + (100{,}000)\,(425)$$

$$m_b = 285{,}712 \text{ lb/hr}$$

In order to calculate isentropic work for the first stage, it is necessary to determine the isentropic enthalpy at 60 psia. Fig. 24-27, the first stage inlet entropy equals 0.93 Btu/(lb•°R), and the corresponding isentropic enthalpy at 60 psia is 440 Btu/lb.

The ideal change in enthalpy = 440 – 410 = 30 Btu/lb

For propane refrigerant k = 1.13, compression ratio, r, of 4.15 and the isentropic efficiency, η_i of 0.75, the required compression power for the first stage is obtained from Eq 14-7b:

$$GHP_1 = \frac{(30)\,(178{,}570)}{(0.75)\,(2{,}544.4)} = 2807 \text{ hp}$$

Using Eq 14-7a we determine the first stage discharge enthalpy is:

$$h_{vld} = \frac{30}{0.75} + 410 = 450 \text{ Btu/lb}$$

A material balance around the second compression stage yields the total refrigerant flow:

$$m_T = m_1 + (m_b - 78{,}570) = 178{,}570 + (285{,}712 - 78{,}570)$$

$$= 385{,}712 \text{ lb/hr}$$

A heat balance at the second compression stage entrance yields the second stage inlet enthalpy:

$$h_{v2s} = \frac{(450)\,(178{,}570) + (425)\,(285{,}712 - 78{,}570)}{(385{,}712)}$$

$$= 437 \text{ Btu/lb}$$

From Section 24, the inlet entropy at 60 psia and 437 Btu/lb is 0.93 Btu/(lb • °R), and the isentropic enthalpy at 250 psia is 465 Btu/lb.

Substituting into Eq 14-6, the ideal enthalpy change across the second stage as:

$$\Delta h = 465 - 437 = 28 \text{ Btu/lb}$$

The required compression power for the second stage is determined from Eq 14-7b:

$$GHP_2 = \frac{(28)\,(385{,}712)}{(0.75)\,(2{,}544.4)} = 5659 \text{ hp}$$

Hence, the compression required for the two-stage propane refrigeration system becomes:

$$GHP_T = 2807 + 5659 = 8466 \text{ hp}$$

Using Eq 14-7a, the second stage discharge enthalpy is:

$$H_{V2d} = \frac{28}{0.75} + 437 = 474 \text{ Btu/lb}$$

Substituting into Eq 14-8 yields the condenser duty for the two-stage propane refrigeration system:

$$Q_{cd} = (474 - 325)\,(385{,}712) = 57.5 \text{ MMBtu/hr}$$

From Fig. 24-27 the second stage discharge temperature at 250 psia and enthalpy of 474 Btu/lb is 176°F.

Condensing Temperature

Condensing temperature has a significant effect on the compression horsepower and condensing duty requirements. Mehra[3] illustrated the effect of the condensing temperature on refrigeration requirements for one, two, and three stage systems. Results for a one-stage propylene refrigeration system are summarized in Fig. 14-10.

Fig. 14-10 illustrates that the colder the condensing temperature, the lower the horsepower requirements for a given refrigeration duty. Traditionally, the heat sinks for most refrigeration systems have been either cooling water or ambient air. If cooling water or evaporative condensing is utilized, an 80 to 100°F temperature can be achieved. For most U.S. Gulf Coast locations, a condensing temperature of 115 to 125°F is common when using ambient air for cooling. Section 11 provides wet and dry bulb temperature data for other parts of the United States. Fig. 14-10 also indicates, to a certain extent, the effect on operations between summer and winter conditions as well as between day and night operations.

FIG. 14-10

Effect of Condensing Temperature

Condensing Temperature, °F	60	80	100	120	140
Refrigeration Duty, MMBtu/hr	1.0	1.0	1.0	1.0	1.0
Refrigeration Temperature, °F	–50	–50	–50	–50	–50
Compression Requirement, hp	211	267	333	429	554
Change in hp, %	–36.6	–19.8	Base	28.8	66.4
Condenser Duty, MMBtu/hr	1.54	1.68	1.84	2.09	2.42
Change in Condenser Duty, %	–16.3	–8.7	Base	13.6	31.5

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Refrigerant Subcooling

Subcooling liquid refrigerants is common in refrigeration systems. Subcooling the refrigerant reduces the energy requirements. It is carried out when an auxiliary source of cooling is readily available, and the source stream needs to be heated. Subcooling can be accomplished by simply installing a heat exchanger on the appropriate refrigerant and process streams.

Example 14-2 — Consider installing a 3 MMBtu/hr subcooler on the liquid propane refrigerant from the receiver at 120°F in Example 14-1 for the two-stage propane refrigeration system. The second stage of this system is shown in Fig. 14-11.

Solution Steps:

By performing the heat balance around the subcooler and the second stage suction drum, the liquid refrigerant flowrate to the subcooler is determined to be 315,712 lb/hr. When comparing this to the earlier flowrate of 385,712 lb/hr, the refrigerant flow is reduced by 70,000 lb/hr.

By heat balance around the subcooler, we determine the enthalpy of liquid propane refrigerant leaving the subcooler is 315 Btu/lb which corresponds to a temperature of 110°F.

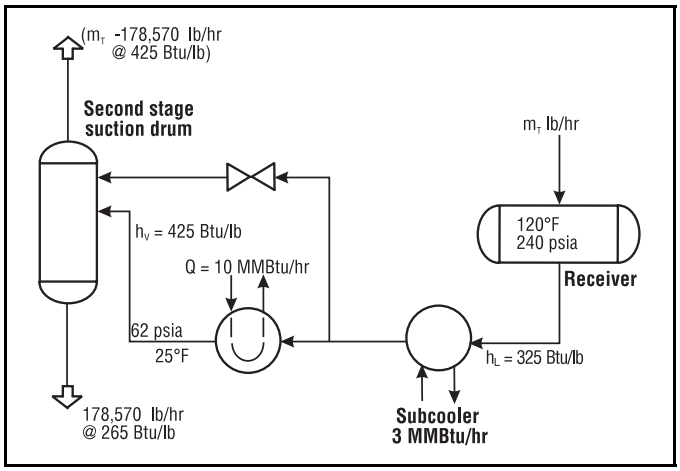
The flowrate of refrigerant through the second stage chiller becomes

$$m_2 = \frac{(10)\,(10^6)}{(425 - 315)} = 90{,}909 \text{ lb/hr}$$

As a result of subcooling, the flow of refrigerant through the second stage chiller has been reduced from 100,000 lb/hr to 90,909 lb/hr. The lower flowrates result in reduced compression horsepower, condenser duty, and reduced size of piping and equipment. These benefits must be balanced against the installed cost of the subcooler exchanger.

FIG. 14-11

Refrigerant Subcooling



Refrigerant For Reboiling

Refrigerants have been successfully used for reboiling services wherever applicable conditions exist. Reboiling is similar in concept to subcooling — heat is taken out of the refrigeration cycle.

In reboiling service, the heat removed from the refrigerant condenses the refrigerant vapor at essentially constant temperature and pressure. The liquid refrigerant produced in a reboiler service is flashed to the next lower pressure stage to produce useful refrigeration. The refrigerant condensing pressure is a function of the reboiling temperature.

Refrigerant Cascading

In the cascading of refrigerants, warmer refrigerants condense cooler ones. Based on the low temperature requirements of a process, a refrigerant that is capable of providing the desired cold temperature is selected. For example, the lowest attainable temperature from ethane refrigerant is –120°F (for a positive compressor suction pressure), whereas the lowest temperature level for propane is –40°F (for a similar positive pressure).

In a refrigeration cycle, energy is transferred from lower to higher temperature levels economically by using water or ambient air as the ultimate heat sink. If ethane is used as a refrigerant, the warmest temperature level to condense ethane is its critical temperature of about 90°F. This temperature requires unusually high compression ratios — making an ethane compressor for such service complicated and uneconomical. Also in order to condense ethane at 90°F, a heat sink at 85°F or lower is necessary. This condensing temperature is a difficult cooling water requirement in many locations. Thus a refrigerant such as propane is cascaded with ethane to transfer the energy from the ethane system to cooling water or air.

An example of a cascaded system is shown in Fig. 14-12, where an ethane system cascades into a propane system. The condenser duty for the ethane system is 30.71 MMBtu/hr. This duty becomes a refrigeration load for the propane system along with its 23 MMBtu/hr refrigeration at –40°F. Therefore, the propane refrigeration system has to be designed to provide a total of 53.71 MMBtu/hr at –40°F in addition to 10 MMBtu/hr at –-4°F and 7 MMBtu/hr at 44°F.

Refrigerant Properties

Physical properties of pure component refrigerants in common use are given in Fig. 14-13. The vapor pressure curves for ethane, ethylene, propane, propylene, and Refrigerant 22 (R-22) are available in Sections 23 and 24 or references 2, 5, 9, and 10. Vapor pressure data for other refrigerants can be obtained from refrigerant manufacturers.

Enthalpy data are necessary in designing any refrigeration system. Pressure-enthalpy diagrams for pure ethane, ethylene, propane, propylene, and R-22 are available in Section 24 of this data book or references 2, 5, 9, and 10. Enthalpy data for other refrigerants can be obtained from refrigerant manufacturers.

Horsepower and Condenser Duty Estimation

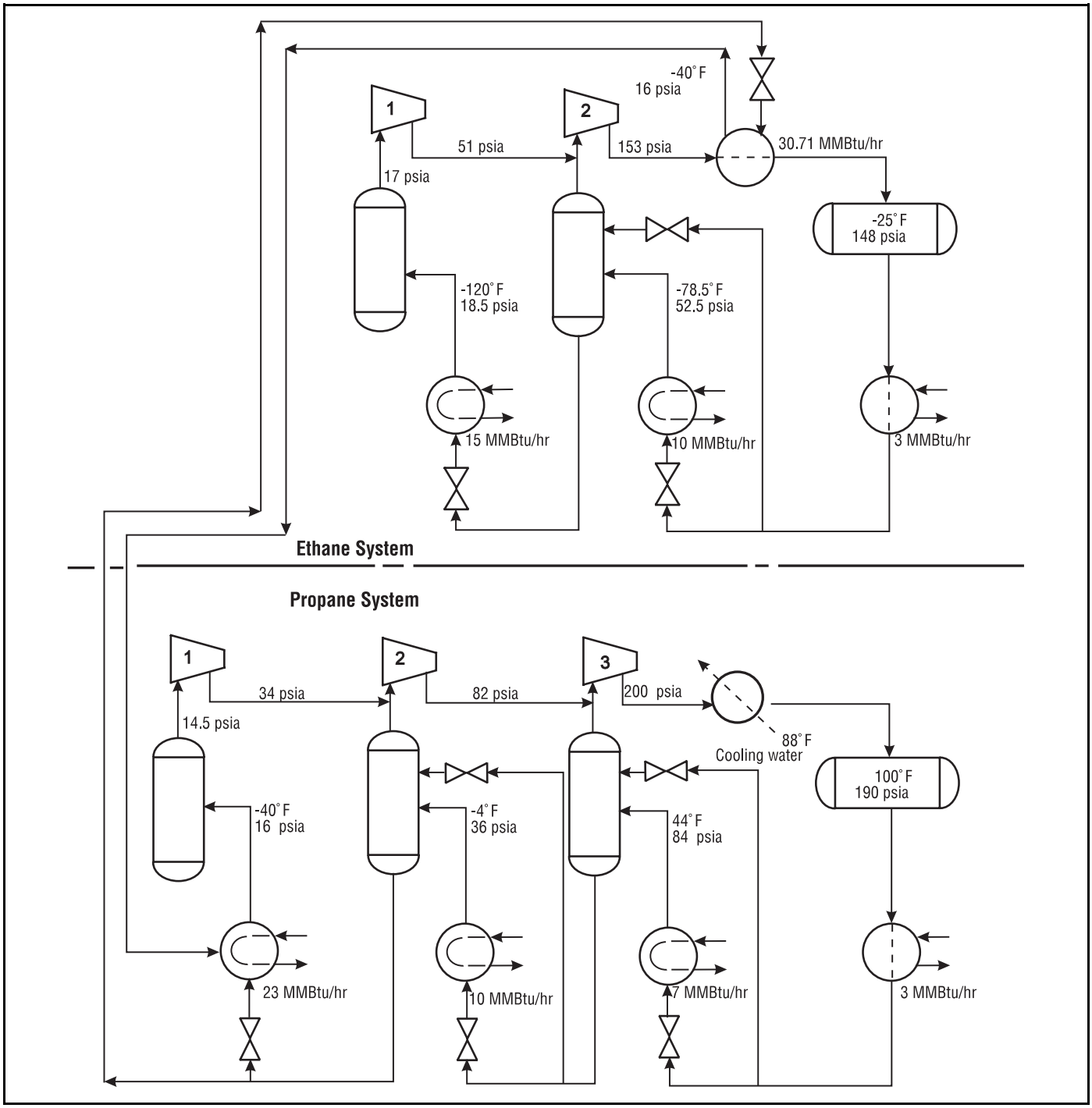
Since many gas processing plants require mechanical refrigeration, generalized charts[5] were developed to aid in a modular approach for designing refrigeration systems.

Because of the complexity of generalizing refrigeration systems, the charts have been developed for four of the most common refrigerants: ethylene, propylene, propane, and Refrigerant 22.

In order to apply these curves to most of the commercially available compressors, a polytropic efficiency of 0.77 was assumed. The polytropic efficiency was converted into an isentropic efficiency[1] to include the effects of compression ratio and specific heat ratio ($k = C_p/C_v$) for a given refrigerant. For well

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FIG. 14-12
Cascade Refrigeration System



balanced and efficient operation of the compressor, an equal compression ratio between stages was employed.

The refrigeration level is defined as the temperature of the dew point vapor leaving the evaporator. The pressures at the compressor suction and side load inlet nozzles were adjusted by 1.5 psi to allow for pressure drop. These charts also include a 5 psi pressure drop across the refrigerant condenser for ethylene, and a 10 psi drop for propane, propylene, and Refrigerant 22.

Before developing any system, one must define refrigerant temperature and condensing temperature of the refrigerant based on the medium used for condensing.

To achieve maximum energy conservation and minimum energy cost, it is necessary to match the process conditions and refrigeration compressor design to obtain the best efficiency.

After defining the lowest refrigerant level and the condensing temperature, the pressure at the evaporator and con-

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FIG. 14-13

Physical Properties of Common Refrigerants [1, 4, 9, 10, 11]

ASHRAE Refrigerant Number	Chemical Name	Chemical Formula	Molecular Weight	Normal Boiling Point °F @ 14.696 psia	Critical Temperature °F	Critical Pressure psia	Freezing Point °F @ 14.696 psia	Liquid Viscosity Centipoise	Liquid Thermal Conductivity Btu / (hr • sq ft • °F)/ft	Specific Heat Ratio $k = C_p/C_v$	Toxicity UL Group Classification
11	Trichloro-fluoromethane	$CC1_3F$	137.4	74.8	388.4	640.0	–168	0.421 @ NBT 0.395 @ 86°F	0.0506 @ NBT 0.0498 @ 86°F	1.13	5
114	Dichlorotetra-fluroethane	$CC1F_2OC1F_2$	170.0	38.4	294.3	474.0	–137	0.44 @ NBT 0.32 @ 86°F	0.0405 @ NBT 0.0366 @ 86°F	1.09	6
12	Dichlorodifluoro methane	$CC1_2F_2$	120.9	–21.6	233.6	597.0	–252	0.358 @ NBT 0.206 @ 86°F	0.0518 @ NBT 0.0392 @ 86°F	1.14	6
22	Chlorodifluoro methane	$CHC1F_2$	86.5	–41.4	204.8	716.0	–256	0.33 @ NBT 0.192 @ 86°F	0.0695 @ NBT 0.0495 @ 86°F	1.18	5a
600	N-Butane	C_4H_{10}	58.1	31.1	305.6	550.7	–217	0.213 @ NBT 0.159 @ 86°F	0.0663 @ NBT 0.061 @ 86°F	1.09	5b
290	Propane	C_3H_8	44.1	–43.7	206.0	616.3	–305	0.21@ NBT 0.101 @ 86°F	0.076 @ NBT 0.056 @ 86°F	1.14	5b
1270	Propylene	C_3H_6	42.1	–53.9	197.1	667.2	–301	0.15 @ NBT 0.089 @ 86°F	0.082 @ NBT 0.057 @ 86°F	1.15	5b
170	Ethane	C_2H_6	30.1	–127.4	9.01	707.8	–297	0.168 @ NBT 0.039 @ 86°F	0.082 @ NBT 0.048 @ 86°F	1.19	5b
1150	Ethylene	C_2H_4	28.1	–154.8	48.6	731.1	–272	0.17 @ NBT 0.07 @ 86°F	0.111 @ NBT 0.031 @ 86°F	1.24	5b
50	Methane	CH_4	16.0	–258.7	–116.7	667.8	–296	0.118 @ NBT	0.110 @ NBT	1.305	5b
717	Ammonia	NH_3	17.0	–28.0	270.4	1636.0	–108	0.25 @ 5°F 0.207 @ 86°F	0.29 @ 32°F 0.29 @ 86°F	1.29	2

denser can be established from the vapor-pressure curve for a specific refrigerant. All examples and data in this section are based upon pure component properties. In actual practice, pure hydrocarbon refrigerants are not always available. Impurities may cause significant deviations in design and performance.

One-Stage Systems — Figs. 14-14 through 14-17 provide data for estimating gas horsepower and condenser duty requirements for one-stage refrigeration systems using ethylene, propane, propylene, and R-22 refrigerants.

Two-Stage Systems — The data for estimating gas horsepower and refrigerant condenser duty requirements for two-stage refrigeration systems utilizing ethylene, propane, propylene, and R-22 are shown in Figs. 14-19 through 14-21.

Three-Stage Systems — The data for estimating gas horsepower and condenser duty requirements for three-stage refrigeration systems utilizing ethylene, propane, propylene, and R-22 are presented in Figs. 14-23 through 14-26.

Example 14-3 — Estimate the horsepower and condenser duty requirements for a single stage propylene refrigeration system that will provide 25 MMBtu/hr of process chilling at a refrigerant level of –20°F.

Solution Steps

The unit GHP for this example from Fig. 14-17 is 222 hp per MMBtu/hr of refrigeration duty at an evaporator temperature of –20°F and a condenser temperature of 100°F. And, from Fig. 14-17, the condenser duty factor equals 1.565 MMBtu/hr per MMBtu/hr of refrigeration duty for the same evaporator and condenser temperatures. Hence, the total power and condenser duty are:

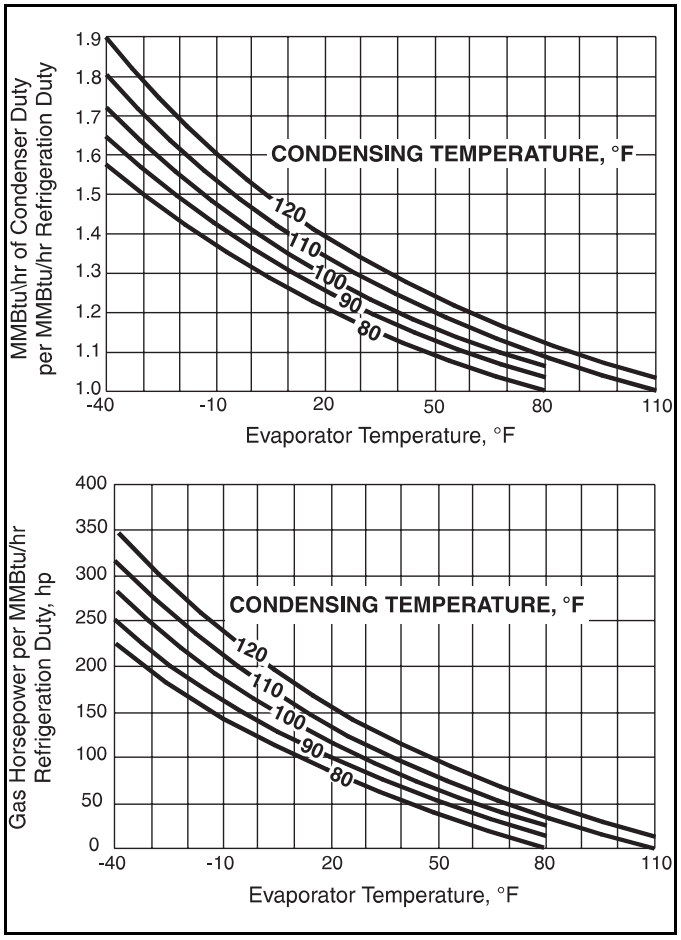
$$GHP = (25)(222) = 5{,}550 \text{ hp}$$

$$Q_{cd} = (25)(1.565) = 39.13 \text{ MMBtu/hr}$$

Heat Exchanger Economizing — An alternative to flash economizing of the refrigeration cycle is to use a heat ex-

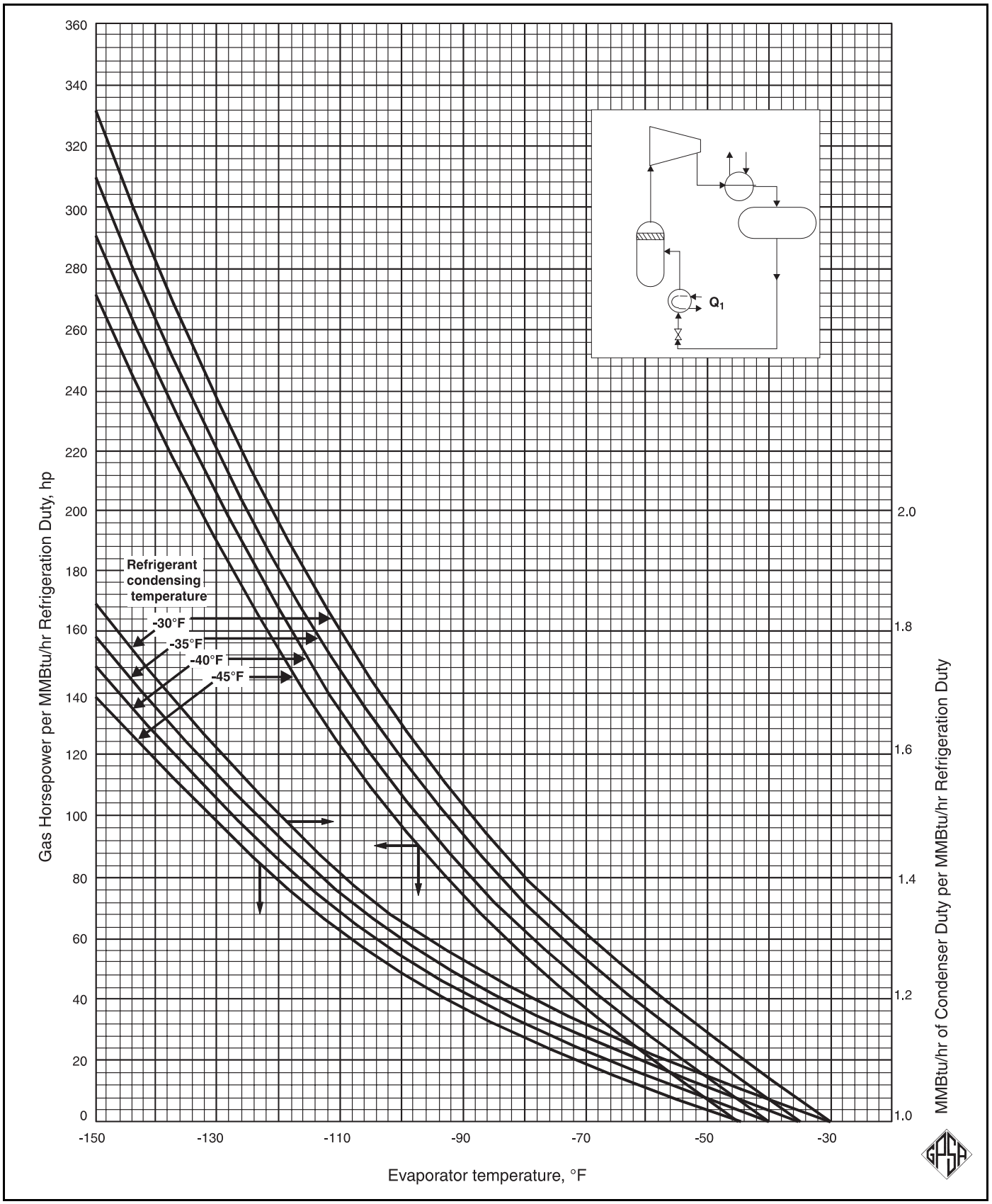
FIG. 14-14

Condenser Duty and Gas Horsepower for One Stage R-22 Refrigerant



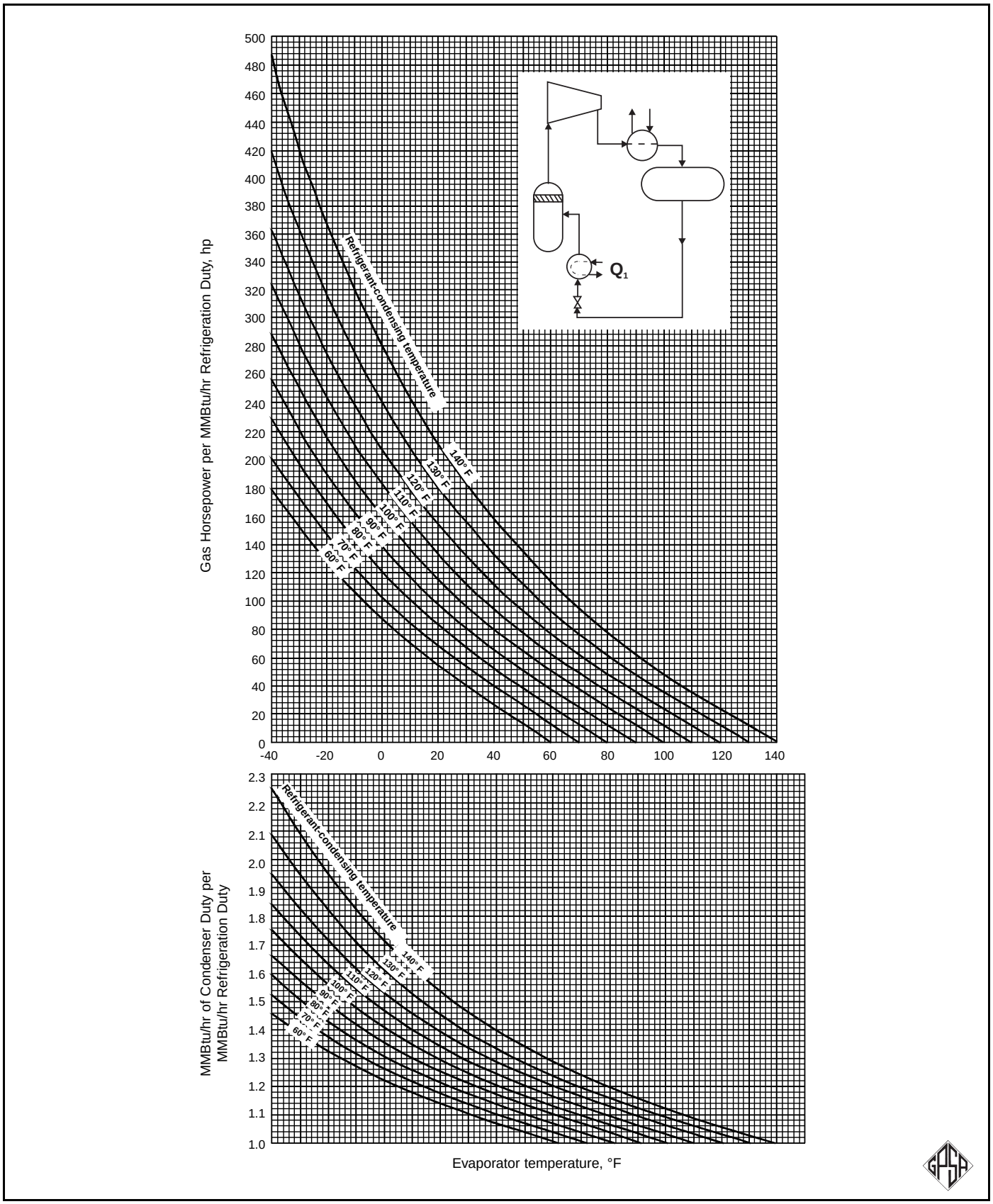
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FIG. 14-15
Single-Stage Ethylene Refrigeration System



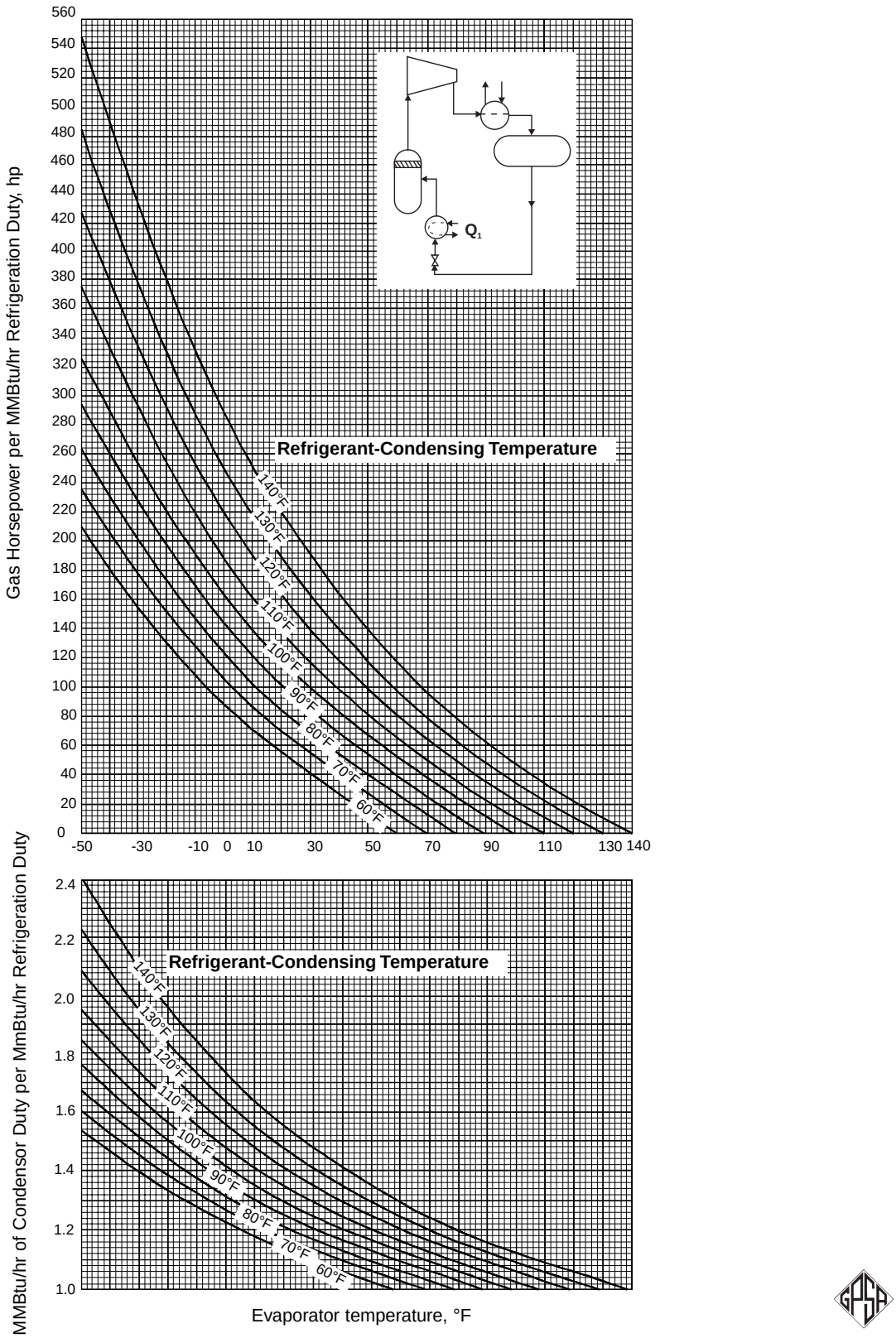
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FIG. 14-16
Single-Stage Propane Refrigeration System



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FIG. 14-17
Single-Stage Propylene Refrigeration System



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changer to accomplish an economizing step. Fig. 14-18 shows an example economizer using a heat exchanger. The heat exchanger is a chiller which uses some of the condensed refrigerant to subcool the balance of the condensed refrigerant stream. The refrigerant used for the chilling is then fed to the interstage (or second stage) of the refrigeration compressor. The subcooled refrigerant is then used for process chillers. The subcooled refrigerant produces less unusable vapor when flashed to suction drum conditions than a refrigerant stream that is not subcooled. Thus the use of the heat exchanger effectively shifts vapor from the low stage of compression to the high stage, thus saving power. The resultant process impact is very similar to the flash economization previously discussed.

Design and Operating Considerations

The following are some of the important parameters that should be considered while designing any refrigeration system to provide a safe, reliable, and economical operation.

Oil Removal — Oil removal requirements from evaporators are related to the type of the refrigerant, lubricant, evaporator, and compressor used in the refrigeration cycle. Fig. 14-26 illustrates the application of an oil reclaimer in a propane refrigerant cycle. In order to remove oil from the refrigerant, a slip stream of refrigerant from the bottom of the chiller is drained into the reclaimer where hot propane refrigerant from the compressor discharge is used to evaporate the refrigerant into the compressor suction. The oil is removed from the bottom of the reclaimer. Similar arrangements can be utilized for other hydrocarbon and ammonia refrigerants. Operation may be designed for either manual or automatic.

Where halocarbon refrigerants and/or synthetic lubricants are employed, it is imperative that the oil reclaimer system be approved by the compressor manufacturer.

Liquid Surge and Storage — All refrigeration systems should have a liquid surge and storage vessel, commonly called a receiver. A surge vessel is required on all systems where the operating charge in the evaporator(s) and the condenser(s) varies due to variable load conditions. In addition to accommodating a varying refrigerant charge, the receiver drains the condenser(s) of liquid so that the effective condensing surface is not reduced by liquid backing up. The refrigerant charge in a receiver may vary over a wide range, from a minimum at full load to a maximum at no load.

Systems with inadequate surge vessels often cause problems as they lose the liquid seal due to load variations that always occur. Surge vessels or receivers are relatively inexpensive and when sizing them, consideration should be given to: (1) a volume equal to 100% of the system inventory at 80% full level, and (2) the availability and quantity of refrigerant makeup.

Vacuum Systems — Refrigeration systems can operate with a suction pressure below atmospheric pressure. These vacuum systems require special considerations:

- Where hydrocarbons are used with reciprocating compressors (which employ rod "packing"), air can enter the compressor and possibly form a hazardous mixture. Extreme care should be taken where such systems are used. These systems must have a manual or automatic purge system. Double acting packing should be employed.
- Where halocarbons such as R-11, R-114, R-113, and other low pressure, high volume refrigerants are employed with centrifugal compressors, the deep vacuums may "draw" air and moisture through flanges, seals, etc. This water-oxygen combination in the presence of halocarbons forms acid and causes "crevice corrosion" of the tubes along with some other problems. A positive purge system must be employed and frequent monitoring of the moisture content in the refrigerant is suggested.
- High pressure halocarbons R-12, R-22, and others are employed in vacuum systems with reciprocating, centrifugal, and screw compressors. These systems will suffer from the same corrosion problems as defined above, but to a lesser extent. However, since they are generally

FIG. 14-18
Heat Exchanger Economizing

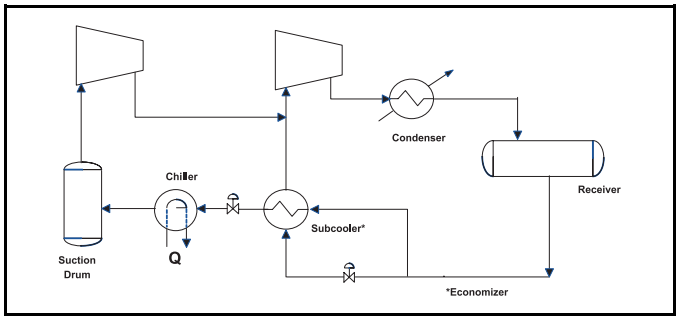
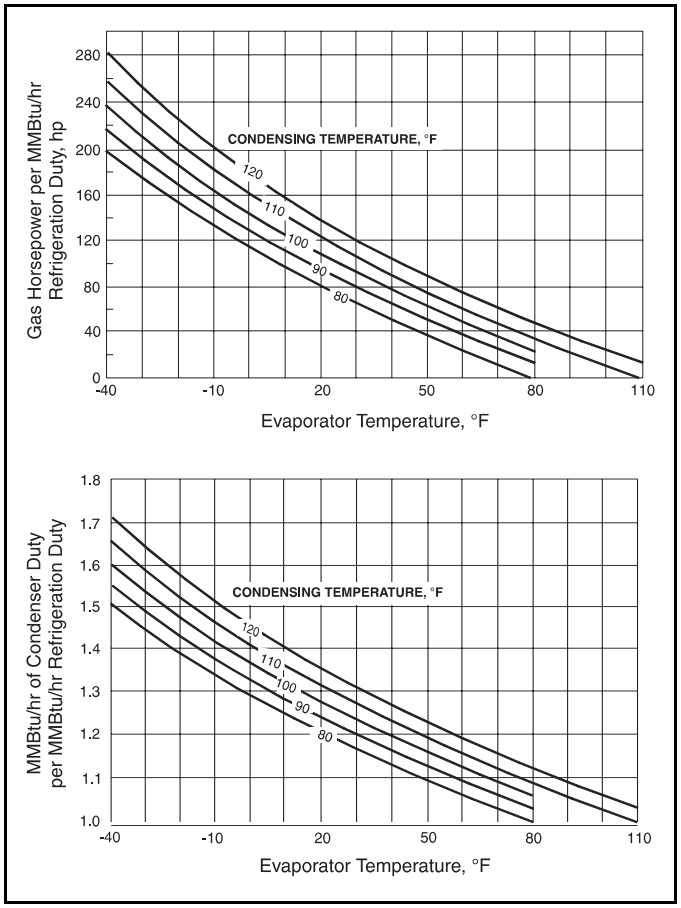
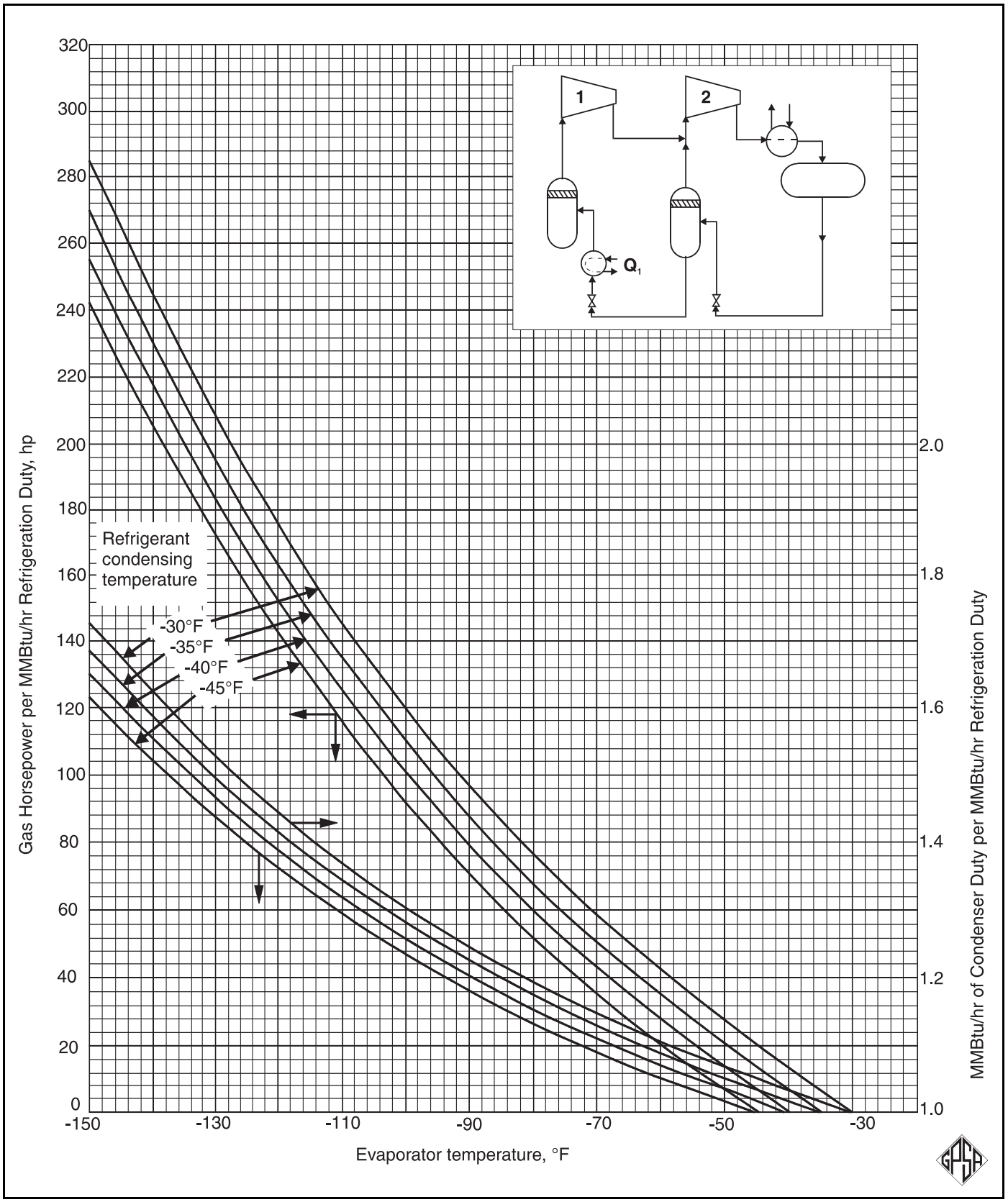


FIG. 14-19
Gas Horsepower and Condenser Duty for Two Stage R-22 Refrigeration



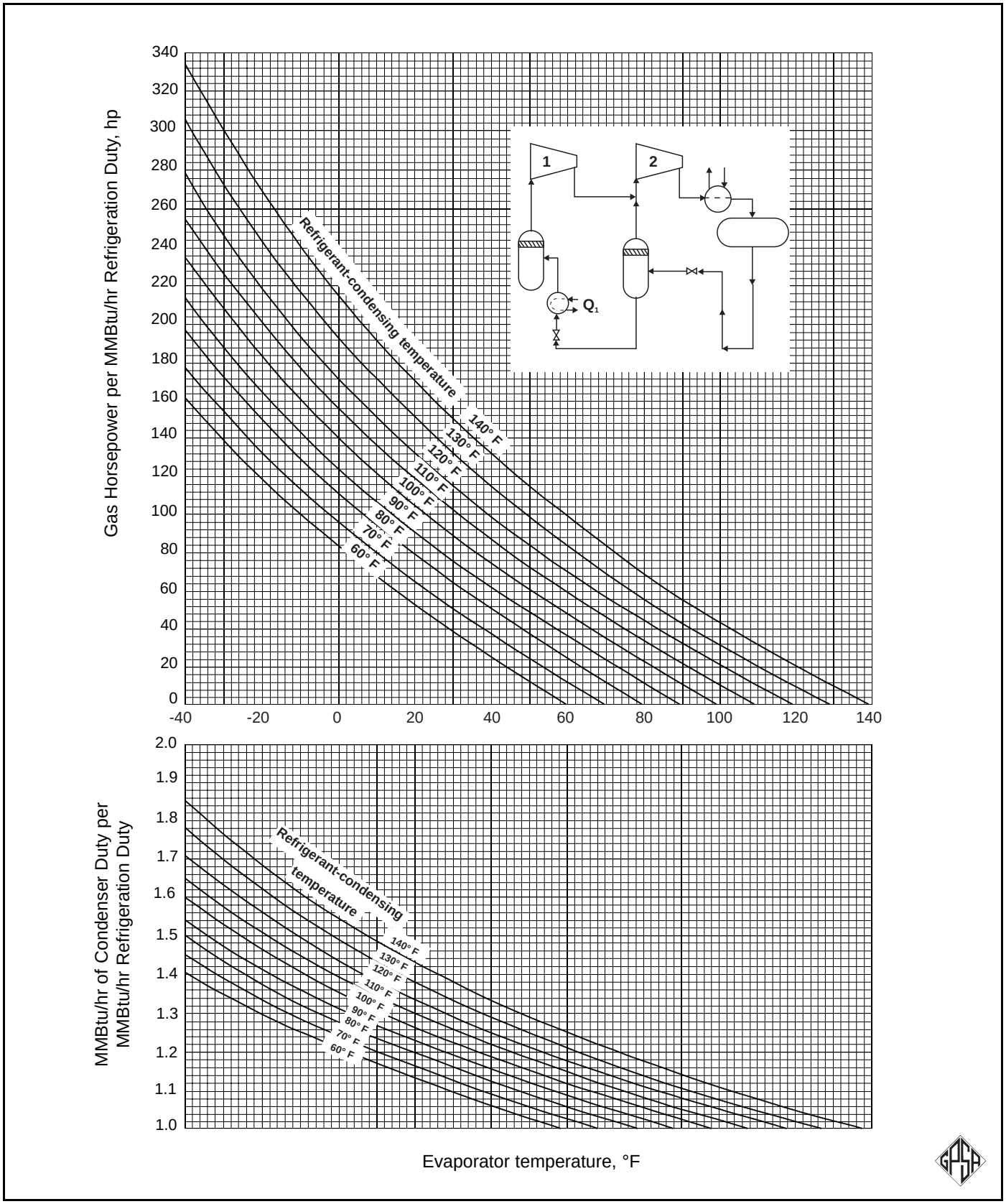
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FIG. 14-20
Two-Stage Ethylene Refrigeration System



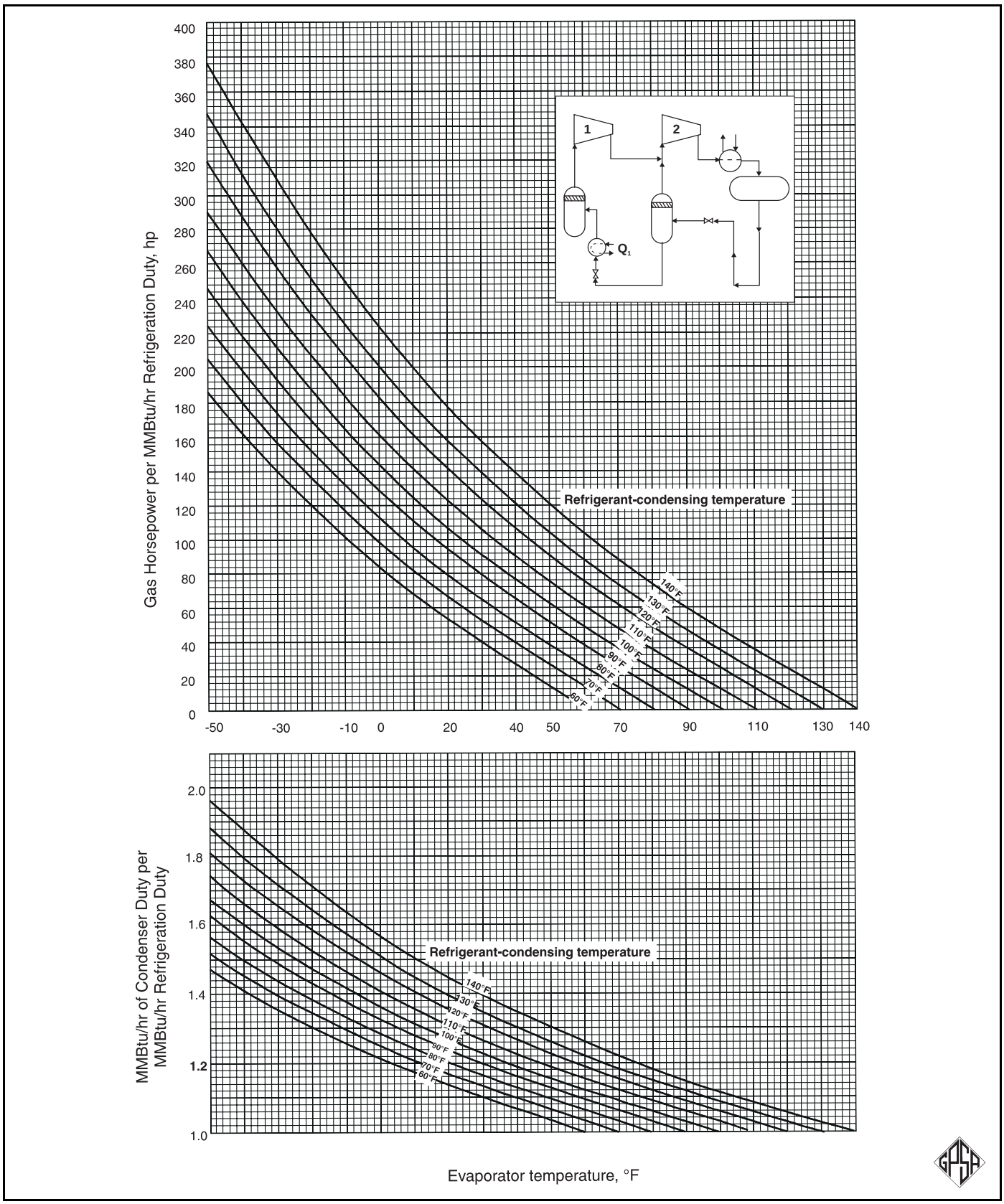
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FIG. 14-21
Two-Stage Propane Refrigeration System



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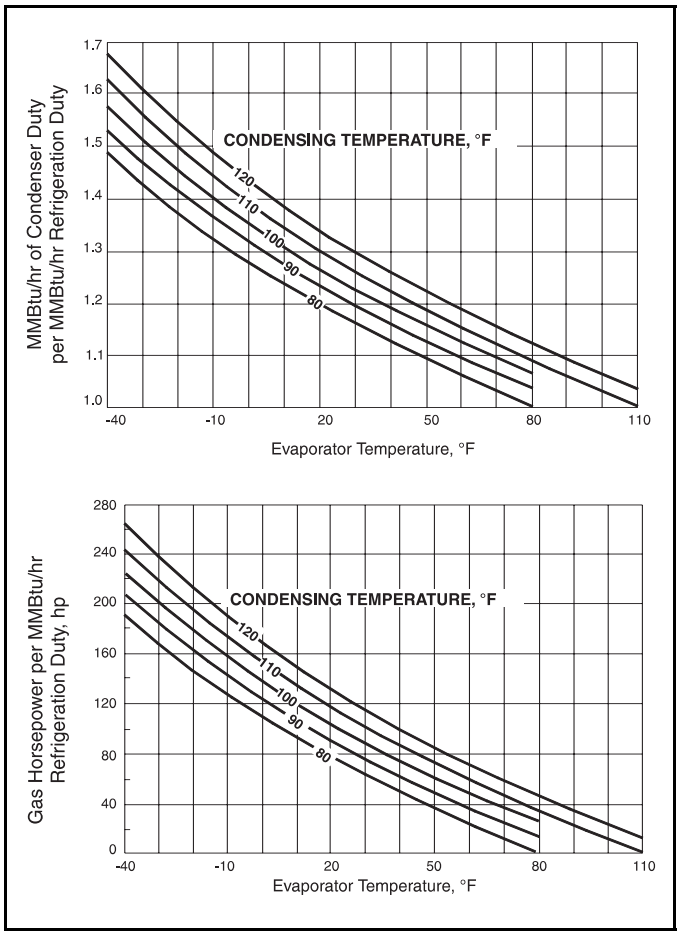
FIG. 14-22
Two-Stage Propylene Refrigeration System



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FIG. 14-23

Condenser Duty and Gas Horsepower for Three Stage R-22 Refrigerant



used at lower temperatures, water in the system can freeze the control valve and in the evaporator. Refrigerant dryers are required in these systems. A good purge system is also required.

- Ammonia has been employed with reciprocating, centrifugal, and screw compressors in vacuum service for many years. Since water will not freeze in the presence of ammonia and the aqua-ammonia formed is only slightly corrosive, this type system has few problems during operation. A good purge system is recommended.

Considerations for Vacuum Refrigeration Systems:

1. Eliminate all flanges where possible.
2. Weld all piping.
3. Use weld in-line valves.
4. Use steel "refrigeration type" stop valves with "back seating" feature and seal caps in lieu of hand wheels. All suction line valves should be angle valves to reduce pressure drop.
5. Install and maintain a good dryer on halocarbon and hydrocarbon systems.
6. Install a purge unit to keep the system free of air and noncondensables.
7. All control valves should have bellows-type seals in lieu of packing if the internal pressure is below atmospheric.
8. An oxygen analyzer is required to detect the presence of air.

Materials of Construction — Materials of construction for refrigeration systems relate specifically to the type of refrigerant used. Some of the basic guidelines are:

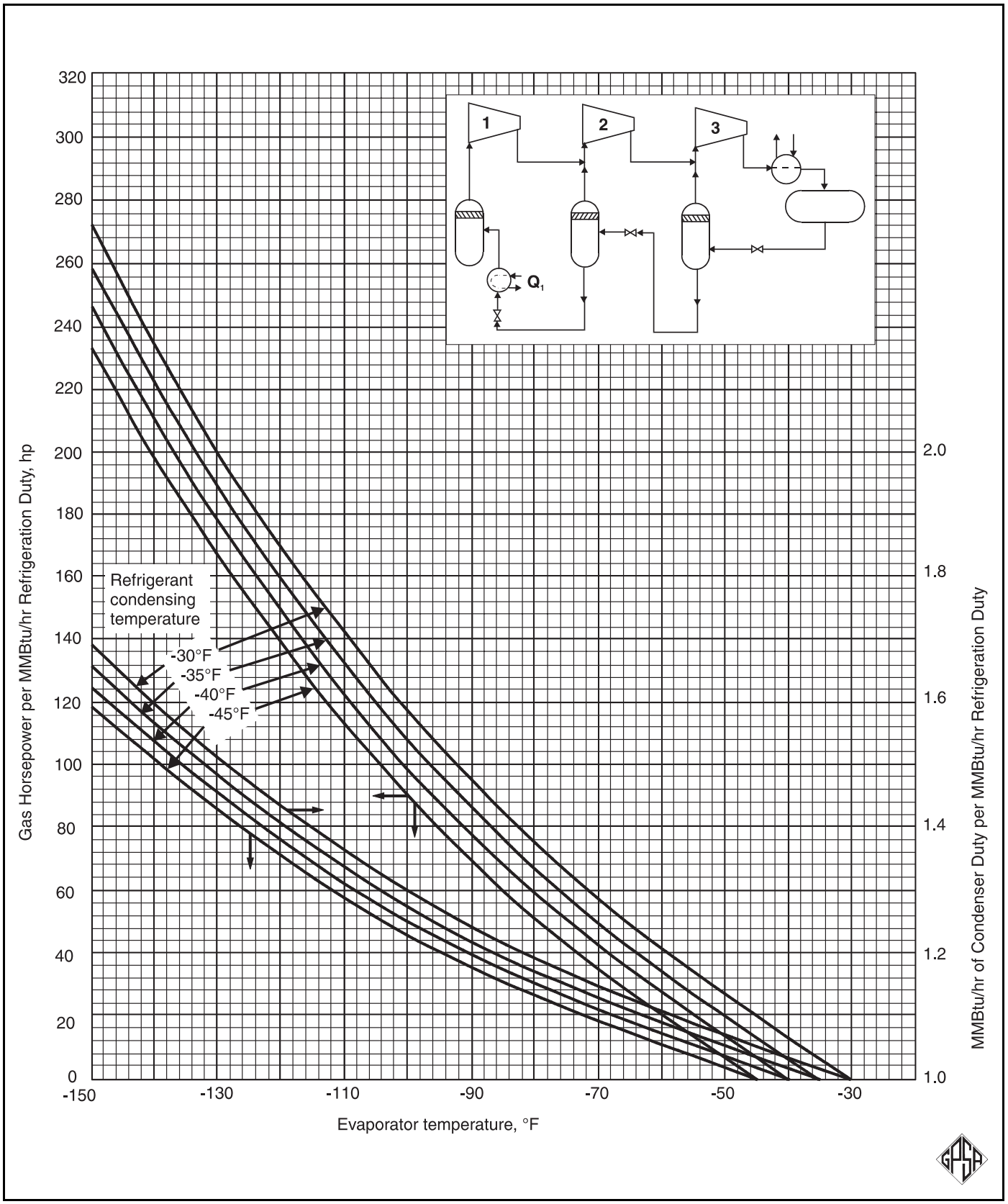
- No copper or copper-based alloys can be used with ammonia refrigeration systems. Where SO_2, H_2S, or similar corrosive chemicals are in the process side or are present in the atmosphere, copper or copper-based alloys are unsuitable.
- Generally, copper and copper alloys can be employed with hydrocarbon and halocarbon refrigerants; however, for most systems steel piping and components are recommended.
- Due to the ambient temperature vapor pressure of most refrigerants, the refrigeration system is normally designed for 250 psig or greater. The low temperature components of the system will operate at temperatures and coincident pressures far below the design pressures. Generally speaking, carbon steel can be used to –20°F. The ASME pressure vessel code section 8 addresses the issue of material selection and impact testing for systems operating below –20°F. Certain provisions in that code may allow carbon steel use for components which operate at pressures less than 25% of design pressure. In many applications, Charpy impact testing will be required.
- The ANSI B31.3 pipe code is generally used for most refrigeration systems in gas processing plants. This code has some provisions for use of materials to –50°F which are normally limited to –20°F and should be consulted for application in this area. There also exists an ANSI B31.5 "Refrigeration Piping Code" which has been used in some applications.
- Various alloys and aluminum are normally used for low temperature operations and may be economically advantageous at temperatures above –20°F.

Refrigerant Purity — Refrigerant contaminants can consist of several components:

- Lubricating oil tends to accumulate in the chiller. Lube oil contamination is reduced by controlling the amount of compressor cylinder lubrication, using synthetic lubricants, providing a good compressor discharge vapor separator to eliminate free oil, and providing a good reclaimer to remove oil accumulation. Fig. 14-27 illustrates one method for eliminating lube oil from the refrigeration system.
- Lighter constituents in the refrigerant charge, such as ethane for a propane system, tend to accumulate in the refrigerant receiver, causing higher condensing pressure. Light component contamination is controlled by the type of refrigerant which is purchased. It can be further reduced by purging the receiver vapors. If the process plant inlet pressure is sufficiently low, the accumulator can be purged into the plant inlet for re-recovery of the hydrocarbons.
- Butane and heavier constituents in a propane refrigeration system tend to accumulate in the chiller. Heavy component contamination is normally not a severe problem, and it is best controlled by draining from the bottom of the lowest temperature chiller.

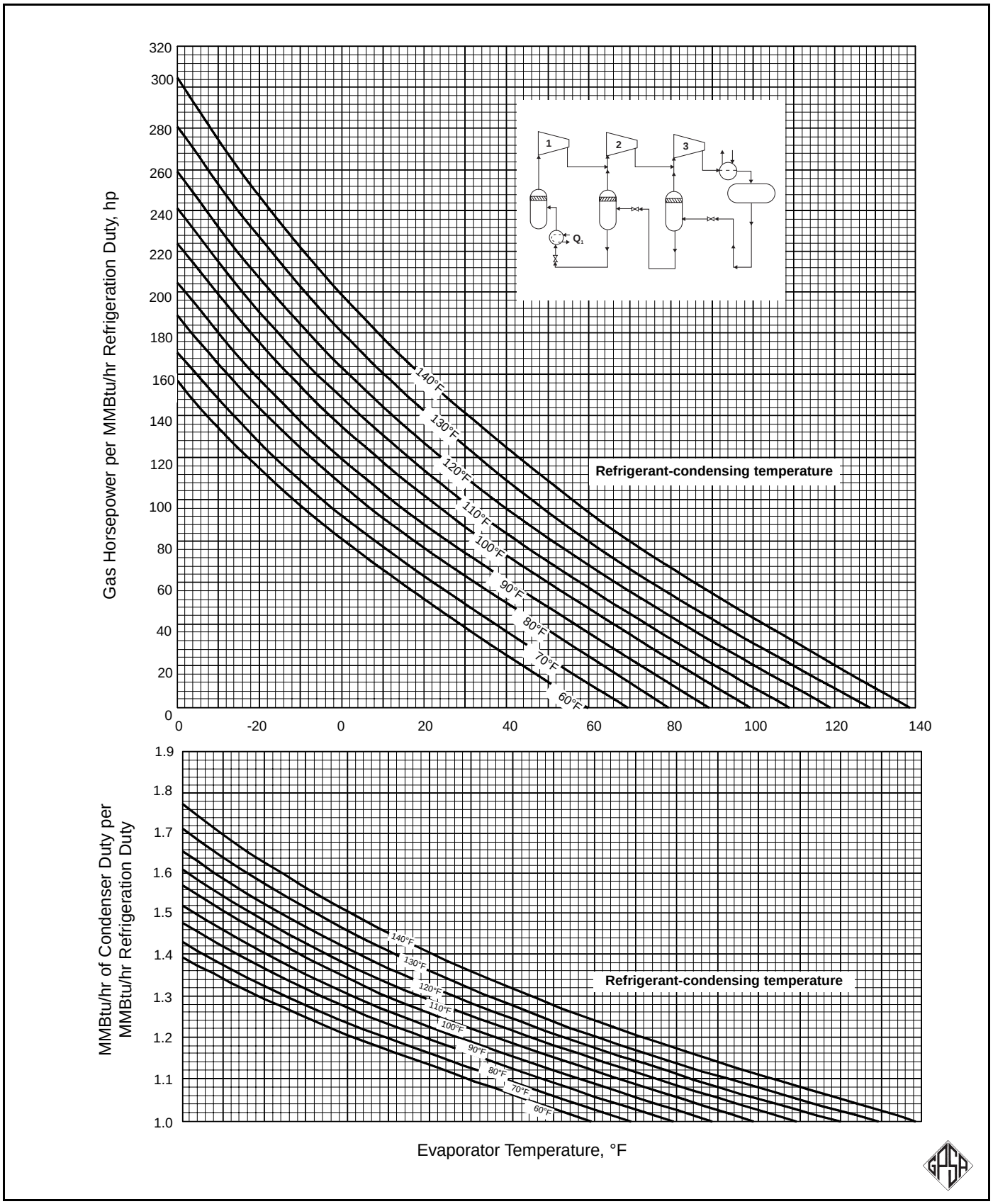
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FIG. 14-24
Three-Stage Ethylene Refrigeration System



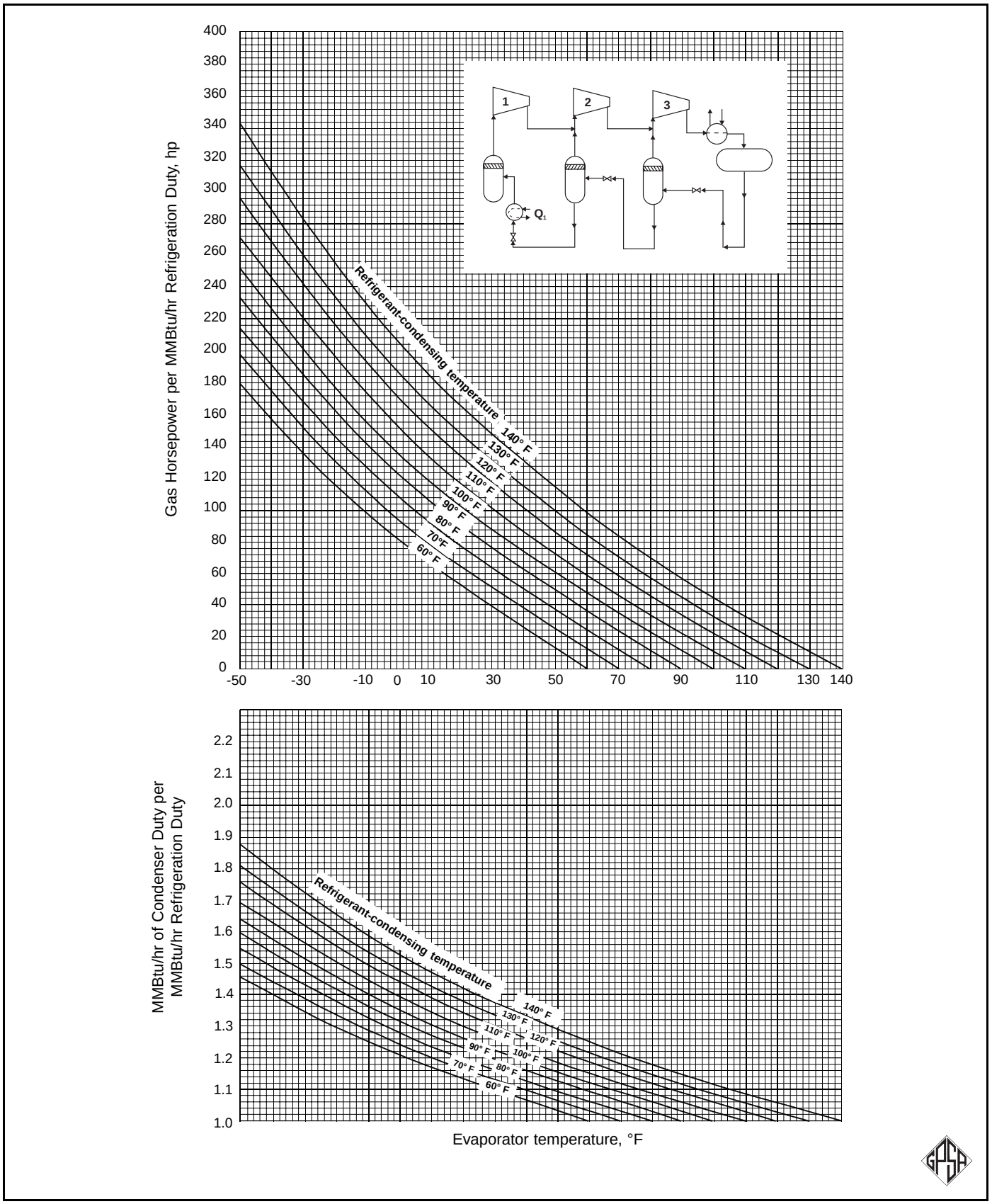
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FIG. 14-25
Three-Stage Propane Refrigeration System



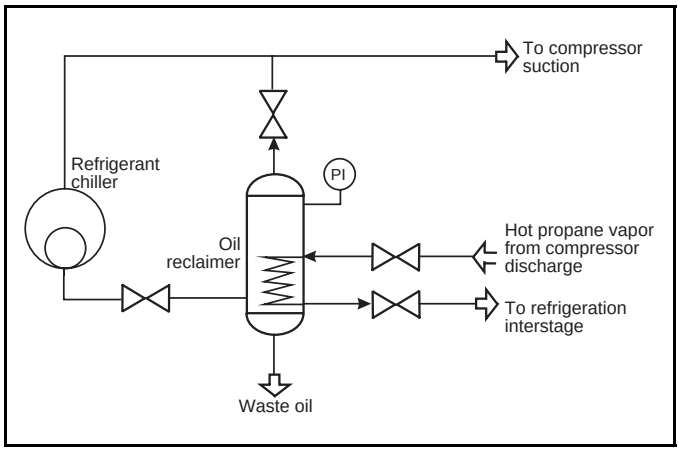
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FIG. 14-26
Three-Stage Propylene Refrigeration System



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FIG. 14-27
Oil Reclaimer



- Process fluid constituents may leak into the refrigerant in the chiller.
- Air can be introduced through the compressor packing if cylinder pressures are below atmospheric.
- Moisture, if present, will form ice and plug up the system either at the control valves or in the chiller. Moisture normally enters the system with the purchased refrigerant charge; it can be the source of considerable operating problems until it is removed. Some refrigeration systems employ a continuous dryer, some only a moisture indicator. The problem can usually be eliminated by injecting methanol in the system and draining it from the chiller. Moisture must also be removed prior to the start-up of a new system, normally by evacuating the system, purging the system with nitrogen or dry gas, injecting methanol, or a combination of these.

Seal Gas and Lube Oil System — Centrifugal compressors are sealed at the shaft by labyrinth seals. To protect against possible entry of oil into the refrigerant, an outward leakage of gas is permitted out of each labyrinth seal. However, to prevent a loss of valuable refrigerant, there is a provision at the middle of the compressor end labyrinth seal to inject a stream of less valuable gas (referred to as seal gas). The seal gas is supplied at a pressure higher than the pressure behind the compressor wheel at a rate slightly higher than that leaking out through the seals. It is the seal gas that leaks out, thus retaining the more valuable refrigerant. The seal gas leaks by the seals and combines with the lube oil leaving the unit and enters the lube oil surge tank.

A seal gas filter is usually provided at the inlet to the refrigerant compressor. The filter element should be changed out at regular intervals. The seal gas flow for a given pressure on the seal gas is set by factory clearances designed into each labyrinth seal.

Types of Compressors

There is a variety of compressors to choose from for refrigeration services. The most common are centrifugal, reciprocating, and screw compressors.

The type of refrigerant used also influences the compressor choice. Small, up to 200 hp, halocarbon-type refrigeration systems normally use reciprocating compressors which have the crankcase vented to the compressor suction. This type compressor can also be used for propane, but the problem of propane solubility in the oil at higher temperatures requires special lubricating oil and a crankcase heater.

Refrigeration load also influences the compressor selection. Centrifugal compressors are not normally economical below about 500 hp with motor drivers, or about 800 hp with turbine drives. Above 1,000 hp, and particularly where the load is close to an even multiple of this number, the use of centrifugal compressors becomes more economical. For lower horsepowers, reciprocating, screw, and rotary compressors are commonly used.

Centrifugal Compressors — At the normal process temperatures encountered in gas processing, a three or four wheel centrifugal compressor is normally required for refrigeration service. This offers the opportunity of utilizing multiple interstage flash economizers and permits multiple chiller temperature levels for further reductions in horsepower.

Centrifugal compressor capacity is controlled by speed variation or suction or discharge pressure throttling. Discharge throttling can cause surge. It is also possible to recirculate refrigerant discharge vapors to the compressor suction during operation at lower loading in order to avoid surge problems. Such recirculation results in wasted horsepower and is one of the primary drawbacks to utilizing centrifugal units.

For more details on Centrifugal Compressors, refer to Section 13.

Reciprocating Compressors — Process temperatures generally dictate two stage compression in a reciprocating machine. This affords the opportunity for one interstage economizer, and also one additional level of chilling. In a conventional refrigeration system, the first stage cylinder is normally quite large as a result of the low suction pressure. The economizer also reduces first stage volume, cylinder diameter, and consequently rod load. Capacity adjustment is accomplished by speed variation, variable clearance on the cylinders, valve lifters, and recirculation of refrigerant vapor to the suction. As with centrifugal compressors, recirculation does result in wasted horsepower. It is also possible to throttle the refrigerant suction pressure between the chiller and compressor in order to reduce cylinder capacity. However, suction pressure control can result in wasted horsepower and the possibility of below atmospheric suction pressure, which should be avoided.

For more details on Reciprocating Compressors, refer to Section 13.

Screw Compressors — Screw compressors have been used in refrigeration systems for many years. They can be employed with all refrigerants. The limitation for suction pressure is about 3 psia with standard discharge pressures at 350 psig. Discharge pressures of over 750 psig are also available.

Screw compressors are gaining popularity in the gas processing industry. Screws can operate over a wide range of suction and discharge pressures without system modifications. There are essentially no compression ratio limitations with ratios up to 10 being used. They operate more efficiently in the 2 to 7 ratio and are comparable in efficiency to reciprocating compressors within this range. Automatic capacity control can provide capacity adjustments from 100% down to 10% with comparable reduction in power requirements.

Screw compressors normally operate at 3600 rpm direct coupled to motor drives. However, they can operate over a

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range of speeds from 1500 to 4500 rpm. Engine drives, gas turbines, and expanders can also be used as drivers.

Rotary Compressors — There is a limited application for large rotary compressors. This is the low-temperature field in which the rotary serves the purpose of a high volume low-stage or booster compressor. These booster compressors are applied at saturated suction conditions ranging from –125°F to –5°F with R-12, R-22, ammonia, and propane refrigerants. Available units range in horsepower from 10 to 600 hp and in displacement from 60 to 3600 cfm in a single unit.

Mixed Refrigerants

Cryogenic processes which remove heat below ambient temperature generally use pure compounds as refrigerants in a closed mechanical refrigeration system. However, when it is not necessary to remove the heat at a practically constant temperature, it may be advantageous to use a mixture of refrigerants. In a proper composition, a mixed refrigerant can minimize temperature differences between the process stream and the refrigerant during heat exchange. This match provides an efficient chilling system.

The most common conventional refrigerants, R-22 and propane, exhibit atmospheric boiling temperatures of –41°F and –44°F, respectively. Lower temperatures can be obtained utilizing propylene, ethane, and ethylene, which have atmospheric boiling temperatures of –54°F, –128°F, and –155°F, respectively. However, these refrigerants require the use of a cascade system because condensation at ambient temperatures is not feasible. One alternative is the use of a mixed refrigerant; for example, ethane-propane. The ethane lowers the evaporation temperature while still permitting condensation at ambient temperatures, albeit at considerably higher pressures.

Fig. 14-28 compares the shape of process chilling curves for an ethylene refrigerant cycle with a mixed refrigerant cycle.[6] The shape of the refrigerant vaporization curve is a function of the composition of the mixed refrigerant. In Fig. 14-28 the composition of the mixed refrigerant is methane 8 mol %, ethylene 37 mol %, and propane 55 mol %.

Some of the design parameters[7] to be considered while evaluating the application of a mixed refrigerant cycle include:

- Compressor suction pressure.
- Shape of vaporization curve.
- Compressor discharge pressure and compression ratio.
- Type of controls.
- Type of compressor.

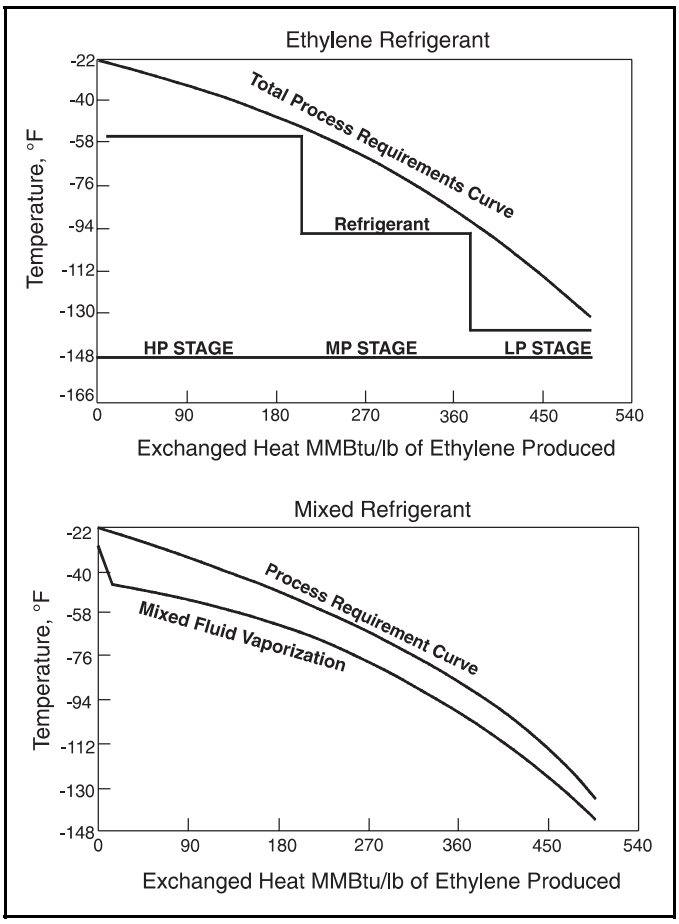
Mixed refrigerants present the problem of component segregation with the lighter components concentrating in the receiver, and the heavier components concentrating in the chiller unless the refrigerant is totally vaporized. Because of the high condensing pressure, mixed refrigerants significantly increase the horsepower per ton of refrigeration.

Chillers

Kettle Type Chiller — The most common type of chiller employed in the gas processing industry is the kettle type. The refrigerant is expanded into the shell of the kettle where a level is maintained to completely submerge the process tube bundle. A level control maintains the proper amount of refrigerant in the kettle.

When using a kettle type chiller, care should be taken to provide adequate vapor disengaging space above the operating level of liquid refrigerant. This type chiller improperly designed and operated is probably the largest single cause of compressor failure due to liquid carryover. The following equation allows approximation of allowable refrigerant load:

FIG. 14-28
Process Chilling Curves



Allowable refrigerant load in lb/hr per cu ft vapor space

$$= \frac{(\text{S.F.})\,(\rho_V)\,(3980)}{(0.869)} \sqrt{\frac{\sigma}{\rho_L - \rho_V}} \qquad \textbf{Eq 14-10}$$

where S.F. = Safety Factor = ½

HTRI has a detailed method for determining the chiller sizing taking into account the vapor space requirement.

Plate-Fin Chillers — Modern cryogenic plants frequently employ plate-fin exchangers for gas cooling and chilling. When the design calls for a hot gas-gas exchanger, a gas chiller, and a cold gas-gas exchanger in sequence, then it may be convenient to put these services in single plate-fin exchanger. Also, plate-fin exchangers offer significant savings for low temperature application where stainless steel is needed for shell and tube units. Significant pressure drop savings can be realized by using single or multiple units for chilling services.

For other types of heat exchangers, refer to Sections 9 and 10.

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System Controls

Level Controls — External cage (displacer-type) level controls are the most commonly used in refrigeration services and are probably the most reliable and easy to maintain instruments. However, because the float chamber is external to the refrigeration system, it is imperative that the float chamber and connecting lines to the chiller be adequately sized and well insulated. Vaporization of refrigerant (due to heat leak) in the float chamber can result in difficulty in maintaining proper level. Internal float level controls eliminate this problem, but present some problems in instrument maintenance.

A differential pressure device is also frequently used for chiller level control; it affords good control when properly installed. The high pressure side connection from the liquid phase should be large, well insulated, and installed in such a way that lubricating oil cannot accumulate and cause erroneous readings. The low pressure side connection to the vapor phase should be uninsulated and possibly even liquid sealed or heated to prevent liquid accumulation.

The same problem exists in level indication. External gauge glasses should have large connecting lines to the chillers and good insulation. Bull's-eye sight glasses are much better for direct indication of chiller level and normally do not present any maintenance problem other than cleaning the glass. Frost plugs are sometimes used and give an approximate level indication while requiring no maintenance.

Pressure Controls — Refrigerant compressor high suction pressure control may be desirable when there are multiple refrigerant compressors in the system. Without high suction pressure control, loss of one refrigerant compressor can result in overloading of the other compressors and loss of all units in an unattended operation. However, suction pressure control can also result in horsepower waste if the compressor suction is throttled unnecessarily.

The refrigerant compressor hot gas bypass is used to prevent compressor suction pressure from getting too low. If the process load decreases, the hot gas bypass will open to maintain a satisfactory compressor suction pressure in an unattended plant. If hot gas bypass remains open, the compressor capacity should be adjusted to reduce bypassing in order to conserve energy. Screw compressors need no such arrangements as they can be automatically unloaded to satisfy the suction pressure settings.

Evaporator Temperature — The evaporator (or chiller) temperature is normally controlled by controlling the refrigerant pressure on the chiller. This may be accomplished by using back pressure valves, refrigerant compressor speed, or hot gas bypass around the compressor.

Low Ambient Controls — All refrigeration systems should have low ambient controls where ambient temperature is below 40°F. These controls, which maintain a preset pressure differential between the condenser and the evaporator pressures, are necessary for continuous operation and for start-up at low ambients. There are several approaches to these controls:

1. For air coolers used as condensers, louvers, air recirculation systems, and fan cycling are employed.
2. For both shell and tube condensers and air coolers, condensing pressure can be controlled by installing a pressure regulating valve actuated by condensing pressure set at a minimum predetermined pressure in the line between the condenser and the liquid receiver. In addition, a small pressure regulator set at a predetermined pressure is installed in a line between the discharge line and the liquid receiver. This regulator will direct enough hot gas to the receiver to keep the pressure high enough to operate the evaporator liquid control valve.
3. Where a shell and tube condenser is used, a water flow control valve operated by condenser pressure can be utilized. This type control may cause sediment and scaling in the condenser.
4. Where evaporative condensers are used, several methods can be employed to control condensing pressure depending upon the ambient temperature and type of installation.
    - The condenser can be selected to operate as an air cooler at temperatures below 32°F (water system shut down and drained) while employing a fan cycling controller.
    - A system as described above in item 2 can be employed.
    - Where the system is indoors with the condenser outdoors, an indoor water sump can be employed with a fan cycling controller.

Refrigerant System Troubleshooting

Figure 14-30 contains a check list for troubleshooting refrigeration systems. This is not an exhaustive list but rather a handy guide to prompt inspection of the system depending on the operating problems experienced.

Freon (CFC) Refrigerant Phase Out

Clorinated fluorocarbons (commonly called Freon) have been used for many years as effective refrigerants in many applications. However, the stability of these compounds, coupled with their chlorine content, has linked them to the depletion of the earth's protective ozone layer. As a result, these compounds have been phased out of production and usage globally. Hydrofluorocarbons (HFC) have been developed as an alternative.

One compound, HFC-134a has been developed to replace dichlorodifluoromethane (CFC-12 or R-12). This compound is reasonably close to R-12 in performance but differences in equipment design and operation must be taken into account in the replacement. Figure 14-29 shows a comparison of HFC-134a and R-12 for an example application. One of the important differences is the higher compression ratio necessary for this refrigerant. Properties for HFC-134a are given in Figures 14-31 through 14-33.

ABSORPTION REFRIGERATION

Even though absorption refrigeration has seen little use in the gas processing industry, it does have application. In areas where there is low cost natural gas, where a low level heat source is available, or where electrical rates have risen dramatically, absorption refrigeration may be an economical way to attain modest temperature level refrigeration. In circumstances where unused boiler capacity is available in summer months, absorption units can be utilized to produce refrigeration.

Lithium Bromide-Water Systems

The lithium bromide absorption refrigeration cycle[8] operates on the simple principle that, under low absolute pressure, water will boil at a low temperature. Fig. 14-34 shows a schematic arrangement of lithium bromide-water system. The system uses heat to efficiently produce refrigeration. The

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FIG. 14-29

Theoretical Cycle Comparison of R-12 and HFC-134a*

	R-12	HFC-134a
Capacity (as % of R-12)	100	99.7
Compressor		
Exit temperature F	188.2	181.5
Exit Pressure, psia	195.6	213.7
Compression Ratio	4.1	4.7

*Conditions: Condenser, 130F; Evaporator, 35F; Comp. Suction, 80F; Expansion Device,125F

lower shell is divided into absorber and evaporator sections while the upper shell consists of the generator and condenser sections. The evaporator section contains the refrigerant, water. A coil, through which the cooling system water circulates, is inserted into the evaporator to establish a heat exchange.

The refrigerant gains heat from the cooling system water, and because of low pressure maintained in the evaporator, quickly reaches saturation temperature and vaporizes, cooling the system water. The remainder of the cycle deals with reclaiming this refrigerant.

The affinity of lithium bromide for water causes the refrigerant vapor to be absorbed by the strong solution in the absorber section. The diluted (weak) solution is pumped into the generator, where steam or hot water is used to drive the water out of the solution as a vapor. The vapor passes into the condenser and changes back to liquid which returns to the evaporator to be reused. Meanwhile, the strong solution left in the generator flows back to the absorber to complete the cycle.

The lowest chilled water temperature achieved by this system is 42°F and typically the unit operates between 42°F and 50°F with varying refrigeration capacity loads.

Aqueous Ammonia System

Refrigeration can be provided by using waste heat with the water-ammonia absorption cycle. This cycle was originally employed in the 1800's and has been refined over the years. It lost its economic value in the 1930's as the more efficient centrifugal and reciprocating compressor systems became inexpensive. Due to its basic inefficiency, the ammonia absorption system cannot be justified unless low level waste heat is available, such as low pressure steam or hot process streams.

Ammonia absorption capacities have been designed in sizes from a minimum of 2.4 MMBtu/hr at –50°F and 3.6 MMBtu/hr at 20°F to a maximum of 30 MMBtu/hr at –50°F and 60 MMBtu/hr at 20°F. Most systems would employ shell and tube condensers and absorbers; however, evaporative cooled absorbers have been used. Air cooled condensers and absorbers could also be used. Various schemes can be used for supplying the water requirements of the condensers and absorbers other than series flow. Parallel flow can be used to reduce absorber size and the heat input to the system.

The heat source will govern the generator design. The generator may be finned-surface heat exchangers with aqueous solution pumped through the tubes for vapor heating mediums or double-pipe heat exchangers for liquid heating mediums. Fig. 14-35 shows a flow sheet of an ammonia absorption system.

Reliability — Ammonia absorption systems are normally installed with spare aqua pumps and spare reflux pumps offering a comparison to centrifugal, reciprocating, and screw compressor systems that have a spare compressor-motor train. Downtime from failure of mechanical items is negligible due to the 100% spare pumps.

FIG. 14-30

Refrigeration System Checklist

Indication	Causes
High Compressor Discharge Pressure	Check accumulator temperature. If the accumulator temperature is high, check: 1. Condenser operation for fouling. 2. High air or water temperature. 3. Low fan speed or pitch. 4. Low water circulation. If condensing temperature is normal, check for: 1. Non-condensables in refrigerant. 2. Restriction in system which is creating pressure drop.
High Process Temperature	Check refrigerant temperature from chiller. If refrigerant temperature is high and approach temperature on chiller is normal, check: 1. Chiller pressure. 2. Refrigerant composition for heavy ends contamination. 3. Refrigerant circulation or kettle level (possible inadequate flow resulting in superheating of refrigerant). 4. Process overload of refrigerant system. If refrigerant temperature is normal, and approach to process temperature is high, check: 1. Fouling on refrigerant side (lube oil or moisture). 2. Fouling on process side (wax or hydrates). 3. Process overload of chiller capacity.
Inadequate Compressor Capacity	Check: 1. Process overload of refrigerant system. 2. Premature opening of hot gas bypass. 3. Compressor valve failure. 4. Compressor suction pressure restriction. 5. Low compressor speed.
Inadequate Refrigerant Flow to Economizer or Chiller	Check: 1. Low accumulator level. 2. Expansion valve capacity. 3. Chiller or economizer level control malfunction. 4. Restriction in refrigerant flow (hydrates or ice).

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FIG. 14-31
Pressure vs. Temperature of HFC-134a[12]

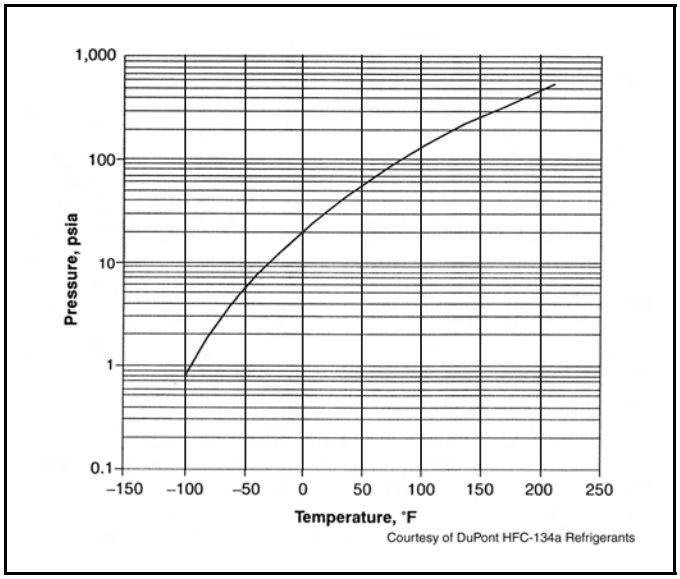


FIG. 14-32
Vapor Thermal Conductivity of HFC-134a at Atmospheric Pressure[12]

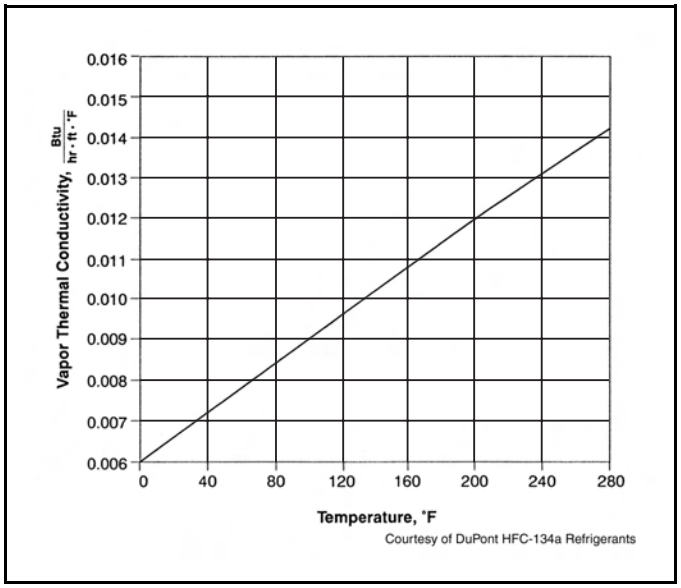
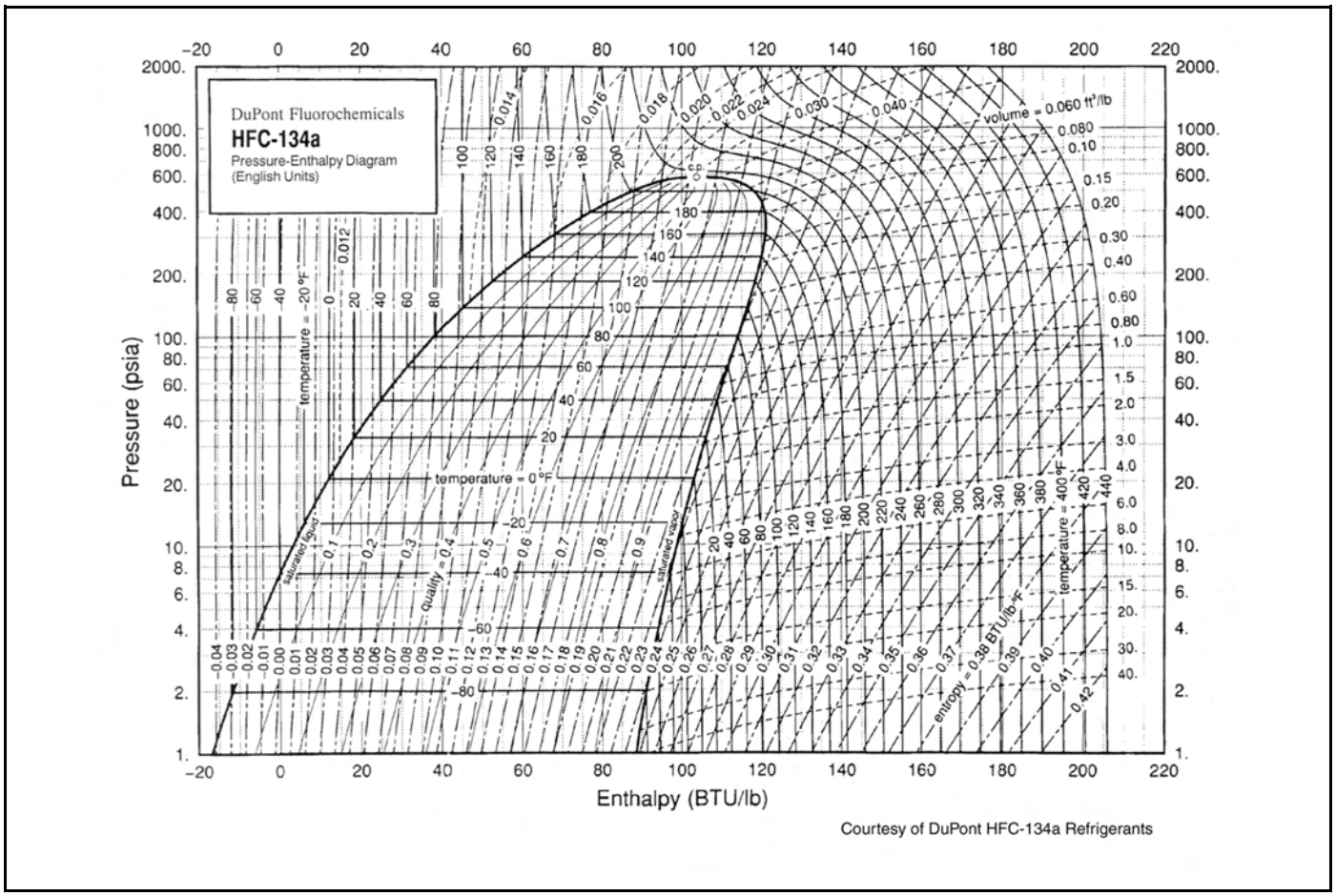


FIG. 14-33
Pressure-Enthalpy Diagram for HFC-134a[12]



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FIG. 14-34

Lithium Bromide-Water Refrigeration System

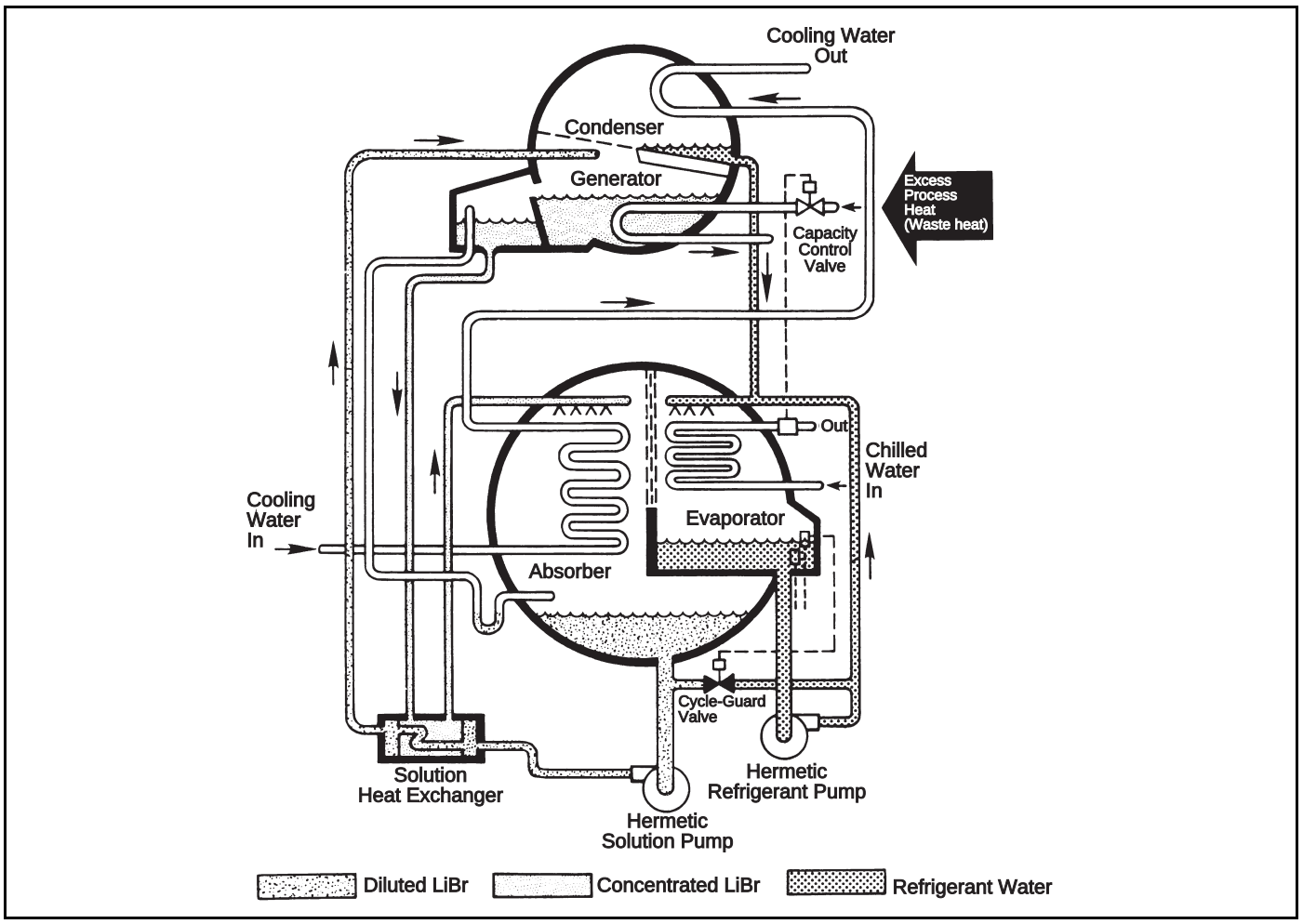
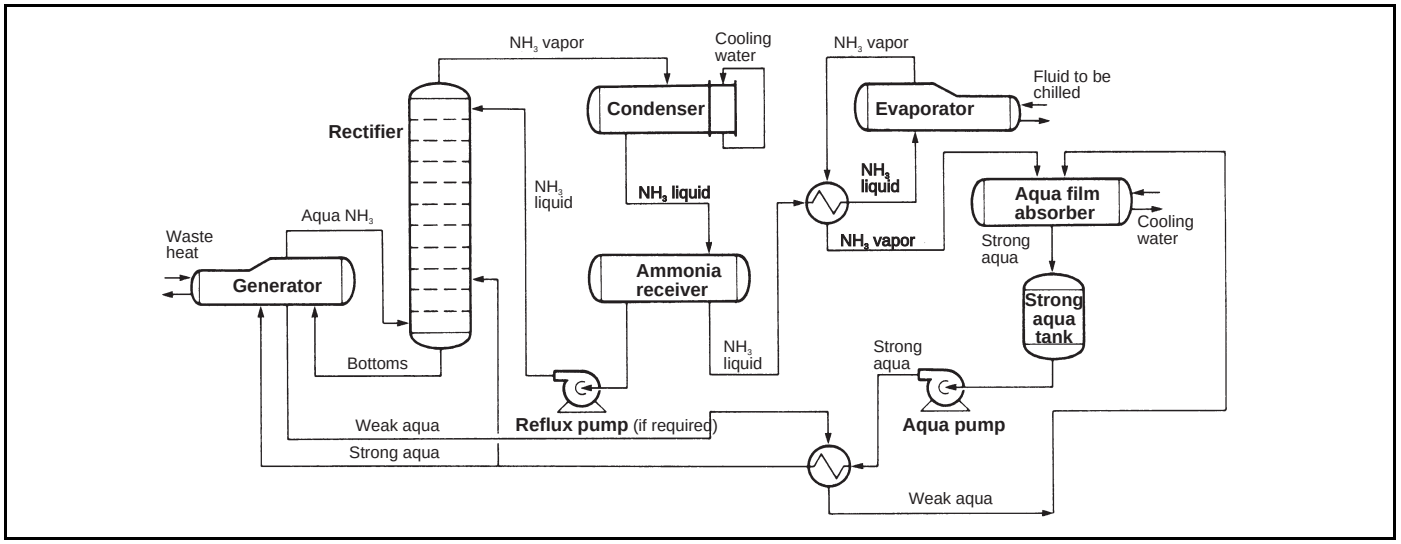


FIG. 14-35

Flow Sheet of an Ammonia Absorption System



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Design Flexibility — Ammonia absorption systems are usually custom designed for each specific application. Evaporator temperatures down to –60°F are possible. Systems can be designed one stage or two stage for several different evaporator temperatures. Systems can be increased in size and evaporator temperatures raised or lowered by the addition of heat exchange surface. Evaporator temperatures are related to heat input temperature. Raising the temperature of the heat source lowers the possible evaporator temperature. The evaporators remain 100% efficient at all times as the refrigeration is oil-free. There is no need to add oil fouling factors to the evaporator design, thus saving 5 to 10% in the evaporator cost. The choice of evaporator design has no limitations.

Applications — The ammonia absorption system has many applications. It can produce refrigeration from waste heat for almost any kind of application in the chemical and petroleum industry. Waste steam has been used as the heat source in many installations in the chemical and petroleum industry providing temperatures from 50°F to –50°F. Process vapor streams and hot oil have also been used as heat sources. Exhaust gases from gas turbines would be an excellent source of heat and this heat would normally be capable of providing low temperature refrigeration due to its high temperatures. Supplemental firing can also be added for peak loads. Stack gases of many kinds could also be used as a heat source.

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SECTION 15

Prime Movers For Mechanical Drives

"Prime movers for mechanical drives" are used as a common name for machines made for transferring mechanical energy to pumps and compressors, including:

- Steam turbines
- Gas turbines
- Electrical motors
- Internal combustion engines

Special considerations for the use of prime movers as drives for generators are not included in this chapter.

Mechanical Drive Steam Turbines

TURBINE TYPES

Mechanical drive steam turbines are major prime movers for compressor, blower, and pump applications. Steam turbines are available for a wide range of steam conditions, horsepower, and speeds. Typical ranges for each design parameter are:

Inlet Pressure, psig	30 – 2000
Inlet Temperature, °F	saturated – 1000
Exhaust Pressure, psig	saturated – 700
Horsepower	5 – 100,000
Speed, rpm	1800 – 14,000

Steam turbines used as process drivers are usually required to operate over a range of speeds in contrast to a turbine used to drive an electric generator which runs at nearly constant speed. Significant hardware differences exist between these two applications. Only variable speed process drivers will be covered here.

Mechanical drive steam turbines are categorized as:

- Single-stage or multi-stage
- Condensing or non-condensing exhausts
- Extraction or admission
- Impulse or reaction

Single Stage/Multi-Stage

In a single-stage turbine, steam is accelerated through one cascade of stationary nozzles and guided into the rotating blades or buckets on the turbine wheel to produce power. A Rateau design has one row of buckets per stage (Fig. 15-2). A Curtis design has two rows of buckets per stage and requires a set of turning vanes between the first and second row of buckets to redirect the steam flow (Fig. 15-3). A multi-stage turbine utilizes either a Curtis or Rateau first stage followed by one or more Rateau stages.

Single-stage turbines are usually limited to about 2500 horsepower although special designs are available for larger units. Below 2500 horsepower the choice between a single and a multi-stage turbine is usually an economic one. For a given shaft horsepower, a single-stage turbine will have a lower capital cost but will require more steam than a multi-stage turbine because of the lower efficiency of the single-stage turbine.

Condensing/Non-Condensing

The energy available in each pound of steam which flows through the turbine is a function of the overall turbine pressure ratio (inlet pressure/exhaust pressure) and inlet temperature. Condensing turbines are those whose exhaust pressure is below atmospheric. They offer the highest overall turbine pressure ratio for a given set of inlet conditions and therefore require the lowest steam flow to produce a given horsepower. A cooling medium is required to totally condense the steam.

Non-condensing or back-pressure turbines exhaust steam at pressures above atmospheric and are usually applied when the exhaust steam can be utilized elsewhere.

Extraction/Admission

Some mechanical drive steam turbines are either extraction or admission machines. Steam is extracted from, or admitted to, the turbine at some point between the inlet and exhaust

FIG. 15-1

Nomenclature

A	= area, sq in.	N	= number of power strokes per min
ASR	= actual steam rate, lb/(hp • hr)	P	= number of magnetic poles in motor
BMEP	= brake mean effective pressure, psi	s	= specific entropy of superheated steam, Btu/(lb • °F)
D	= diameter, in.	s_f	= specific entropy of saturated water, Btu/(lb • °F)
F	= steam low, lb/hr	s_g	= specific entropy of saturated steam, Btu/(lb • °F)
f	= frequency, Hz	S	= piston stroke, ft
h	= specific enthalpy of superheated steam, Btu/lb	TSR	= theoretical steam rate, lb/(hp • hr)
h_f	= specific enthalpy of saturated water, Btu/lb	v	= velocity, ft/sec
h_g	= specific enthalpy of saturated steam, Btu/lb	ρ	= density, lb/cu ft

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FIG. 15-2

Rateau Design

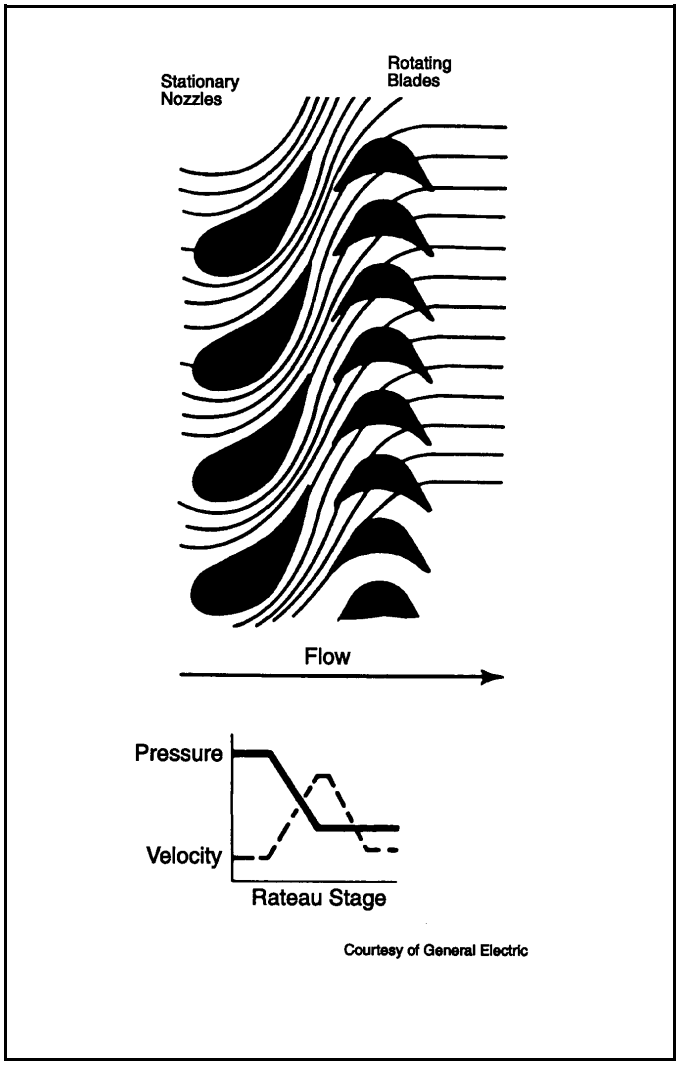


FIG. 15-3

Curtis Design

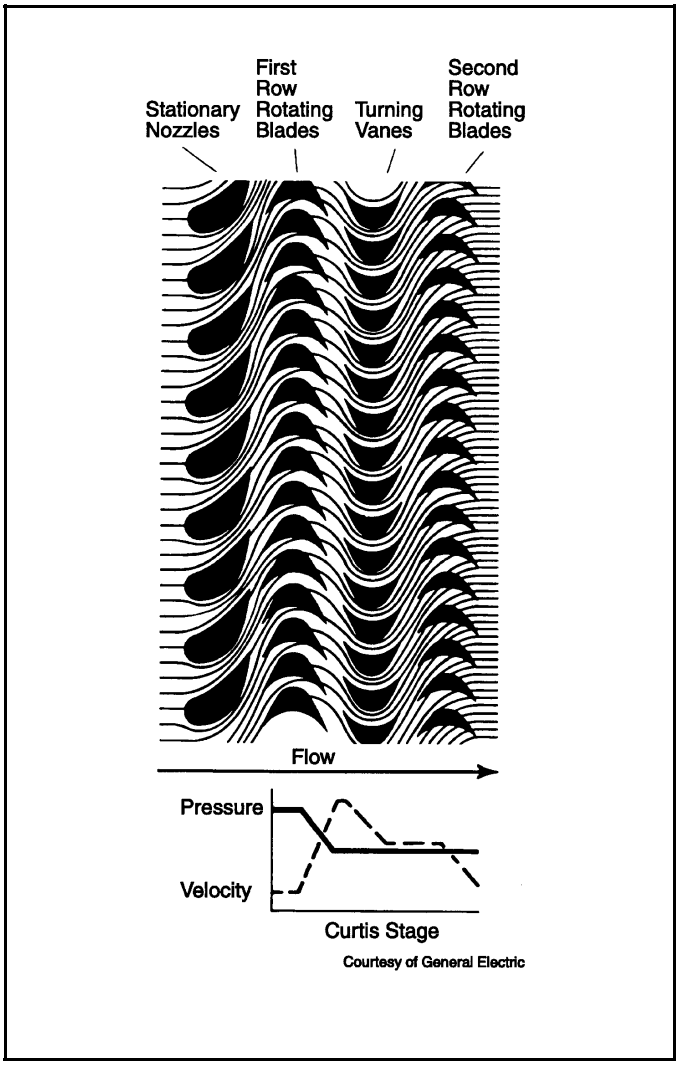
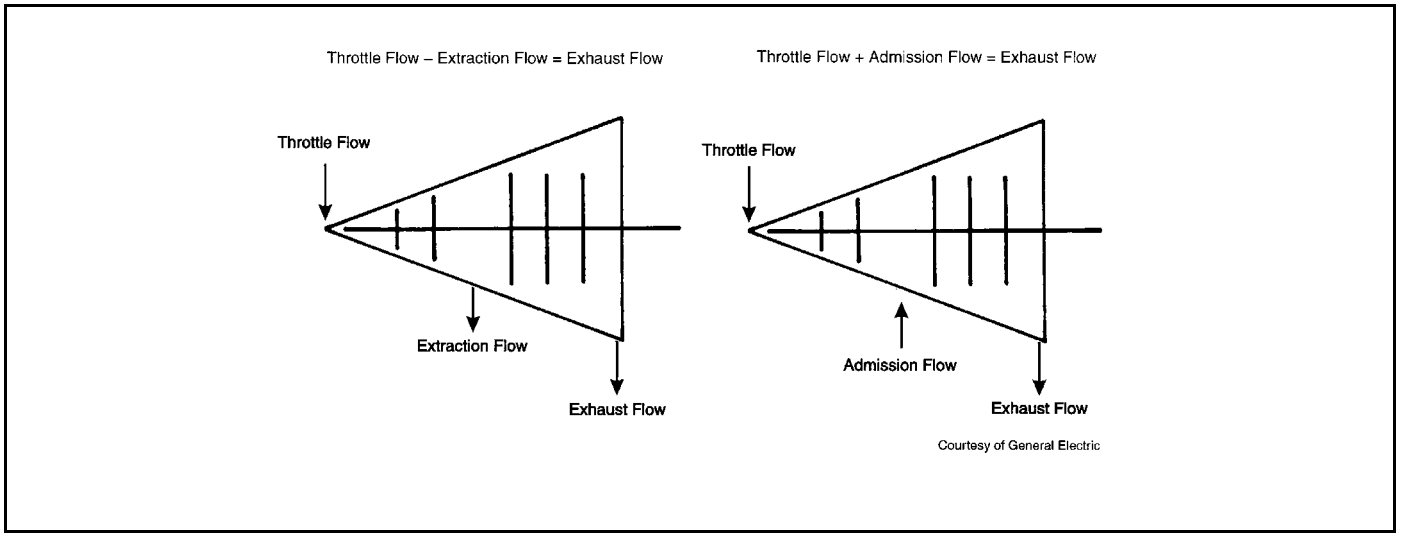


FIG. 15-4

Extraction / Admission Flow Turbines



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(Fig. 15-4). Admission or extraction units may be either controlled or uncontrolled. An uncontrolled turbine accepts or provides steam based only on the characteristics of the steam system to which the extraction or admission line is connected. A controlled turbine will control the flow of extraction or admission steam based on some process measurement such as pressure or flow. In general, if the horsepower associated with the extraction or admission flow is greater than 15% of the total turbine horsepower, a controlled extraction (or admission) turbine is used.

Impulse/Reaction

Turbines are further categorized by the philosophy employed in the steam path design and are divided into two major design concepts: impulse and reaction. In an impulse turbine the pressure drop for the entire stage takes place across the stationary nozzle. In reaction designs, the pressure drop per stage is divided equally between the stationary nozzles and the rotating blades (Fig. 15-5). For given horsepower, speed and steam conditions, a reaction turbine will, in general, employ approximately three times more stages than an impulse turbine in the same turbine span. Most U.S. mechanical drive steam turbines are of the impulse type.

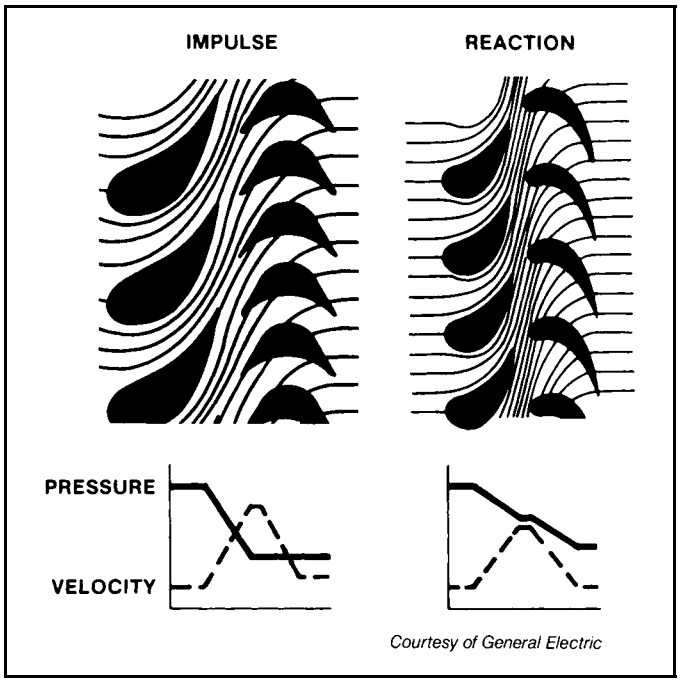
TURBINE COMPONENTS

Trip and Throttle Valve/Stop (Block) Valve

A trip-and-throttle valve or stop valve, or both, may be positioned between the steam supply and the turbine inlet control valve(s) (Fig. 15-6). During normal operation this valve remains fully open and its primary function is to shut off the steam supply in response to a trip (shutdown) signal. In addition a trip-and-throttle valve can be used to modulate the steam flow during start-up and can be either manually or hydraulically positioned from zero lift to 100% lift. The stop valve can only be positioned either in the closed or fully open positions. In order to minimize the pressure drop through the trip-and-throttle valve, maximum inlet velocities are usually limited to 150 ft/sec. Velocities above this level will usually result in high pressure drops which will reduce turbine efficiency.

FIG. 15-5

Turbine Types



Inlet Control Valves

The primary function of the inlet control valve(s) is regulation of the steam flow to provide the appropriate horsepower and speed. These valves may also close in response to a shutdown signal. Throttling which occurs across the control valve(s) reduces the thermal performance of the turbine. This efficiency loss is a function of the control valve design and overall turbine pressure ratio. For a given amount of throttling, turbines with large pressure ratios suffer smaller efficiency losses than turbines with smaller pressure ratios (Fig. 15-7).

Multi-stage turbines may have a single inlet control valve or several control valves to regulate the inlet steam. Typical multi-valve steam turbines will have from three to eight control valves (Fig. 15-8). Multi-valve turbines have higher efficiencies at reduced loads because only the flow through one of the control valves is incurring a throttling loss (Fig. 15-9).

FIG. 15-6

Single Valve with Hand Valves

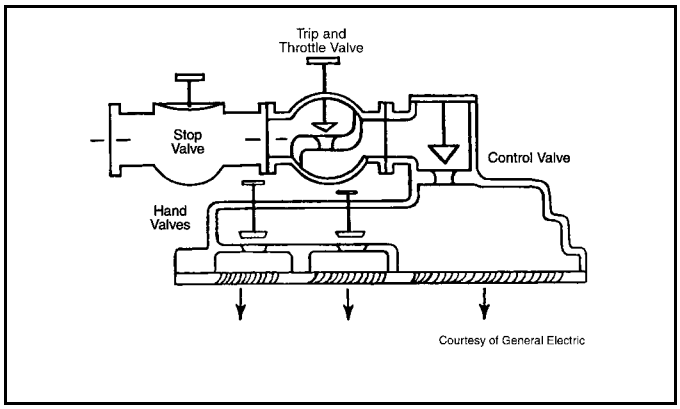
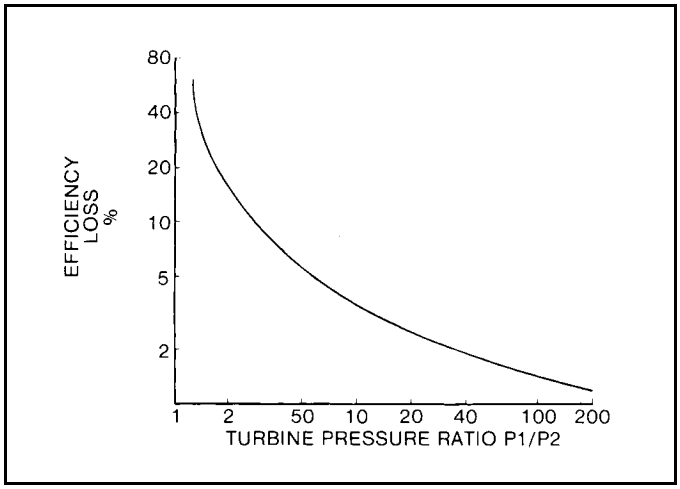


FIG. 15-7

Loss in Available Energy of Steam Due to 10% Throttling



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Turbines with a single control valve will often employ hand valves to improve efficiency at reduced loads. For the turbine shown in Fig. 15-6 both hand valves would be open at or near full load. As the load on the unit is reduced one or both of these hand valves can be closed to reduce throttling loss. Fig. 15-10 shows the efficiency advantage at reduced loads.

Nozzles/Blades (Buckets)

On constant speed turbines a design objective is to avoid all bucket resonances at the operating speed. On variable speed turbines, although the design objectives remain the same, it is seldom possible to avoid all blade resonance because of the wide operating speed range. In these cases it is important to identify all blade resonance and to verify that all stresses are well below the material strength.

Exhaust Casings

Turbine exhaust casings are categorized by pressure service (condensing or non-condensing) and number of rows of the last stage buckets (single flow, double flow, triple flow). Non-condensing exhausts are usually cast steel with most of the applications between 50 and 700 psig exhaust pressure. Most condensing exhausts are steel fabrications although some utilize cast iron construction. Maximum exhaust flange velocities are typically 450 ft/sec. Velocities above this level will usually result in substantial increases in exhaust hood losses and will decrease turbine efficiency.

Moisture Protection

As steam expands through the turbine both the pressure and temperature are reduced. On most condensing and some non-condensing exhaust applications, the steam crosses the saturation line thereby introducing moisture into the steam path. The water droplets which are formed strike the buckets and can cause erosion of the blades. In addition, as the water

FIG. 15-8

Multi-Valve Inlet

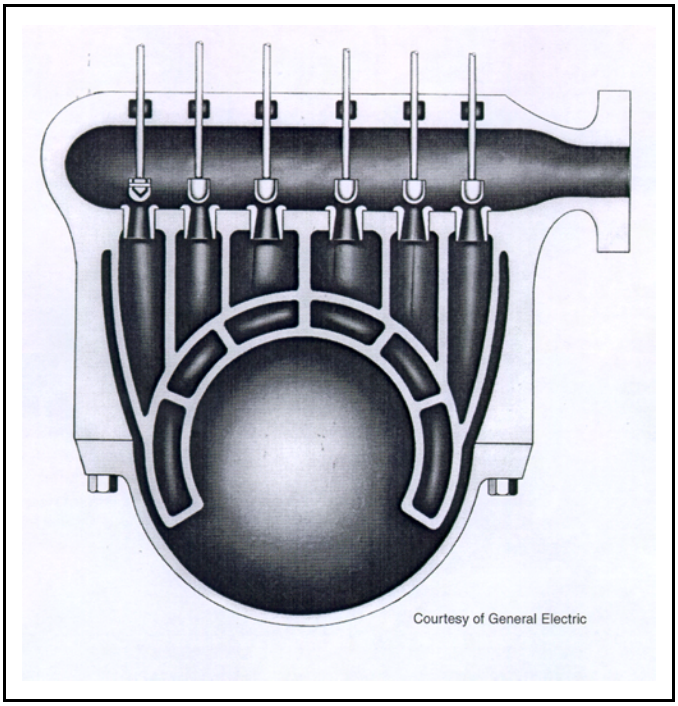


FIG. 15-9

Multi-Valve vs Single-Valve Performance Characteristic
(Typical Non-Condensing Turbine)
Turbine Pressure Ratio = 8.0

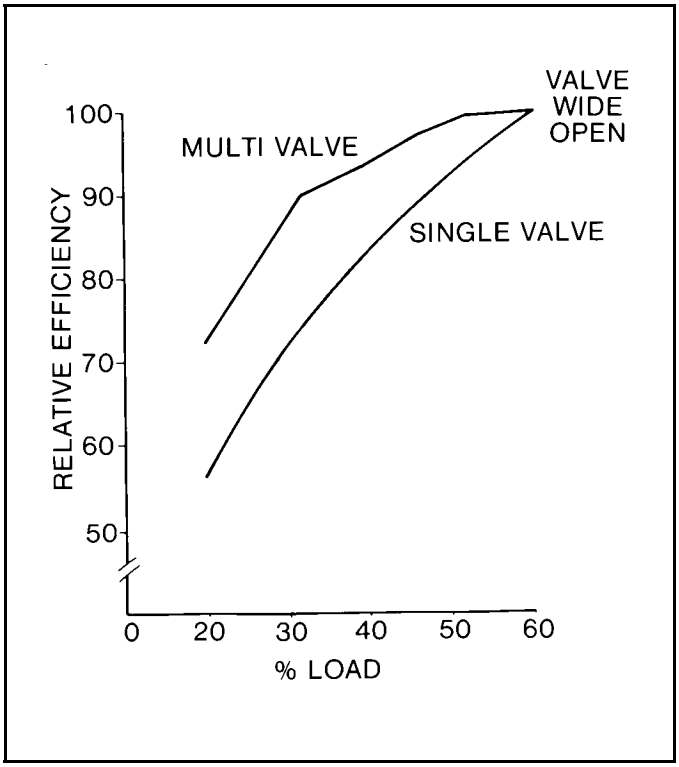
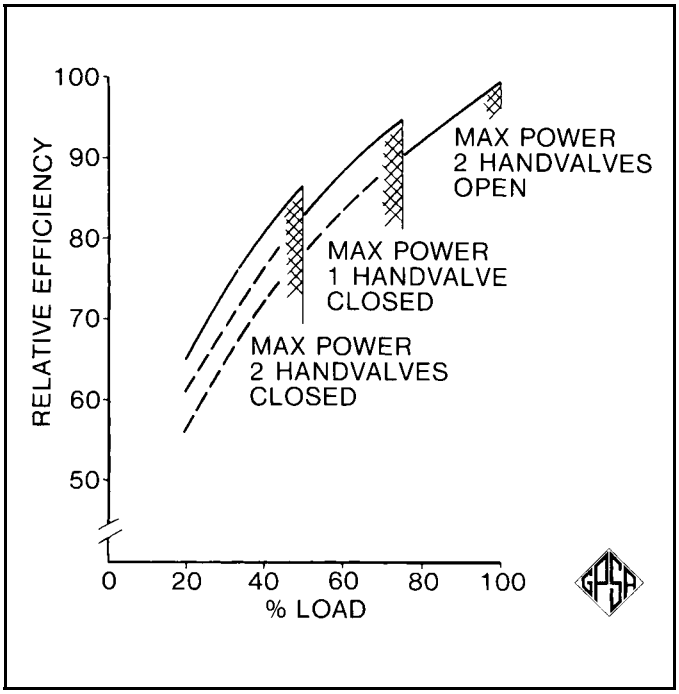


FIG. 15-10

Single Valve with Hand Valves Performance Characteristic
(Typical Non-Condensing Turbine)
Turbine Pressure Ratio = 8.0



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is centrifuged from the blades, the water droplets strike the stationary components, also causing erosion. Where the moisture content is greater than 4%, moisture separators, which are internal to the turbine, can be used to remove a large percentage of the moisture, improving the turbine efficiency and reducing the impact erosion on the buckets. Stainless steel moisture shields can also be used to minimize the impact erosion of the stationary components.

Control Systems

Mechanical governors were the first generation control systems employed on mechanical drive turbines. Shaft speed is sensed by a fly-ball governor with hydraulic relays providing the input to the control valve. A second generation control system was developed and utilized analog control circuitry with the fly-ball governor replaced by speed pick-ups and the hydraulic relays with electronic circuit boards. A third generation control system was developed and replaced the electronic circuitry with digital logic. A microprocessor is used and the control logic is programmed into the governor. The major advantage of this system is the ability to utilize two governors simultaneously, each capable of governing the turbine alone. If the primary governor incurs a fault, the back-up governor assumes control of the turbine and provides diagnostic information to the operator.

EFFICIENCY

Factors Affecting Efficiency

The objective of the steam turbine is to maximize the use of the available steam energy where the available steam energy is defined as the difference between the inlet and exhaust energies (enthalpies) for a 100% efficient constant entropy (i.e., isentropic) process. There are numerous loss mechanisms which reduce the efficiency from the isentropic such as throttling losses, steam leakage, friction between the steam and the nozzles/buckets, bearing losses, etc. Efficiency can range from a low of 40% for a low horsepower single-stage turbine to a high approaching 90% for a large multistage, multi-valve turbine.

Techniques to Improve Efficiency

Various techniques are employed to maximize turbine efficiency, each designed to attack a specific loss mechanism. For example, the number of stages utilized can range from the fewest possible to develop the load reliably to the thermodynamically optimum selection. Spill bands can be utilized to minimize throttling losses. High efficiency nozzle/bucket profiles are available to reduce friction losses. Exhaust flow guides are available to reduce the pressure within the exhaust casing.

The specific features employed on a given application are usually based on the trade-off between capital investment and the cost to produce steam over the life of the turbine.

Operation at Part Load

Most equipment driven by steam turbines are centrifugal machines where horsepower varies as the cube of speed. Part load efficiency varies as a function of speed, flow, and the number of stages. By assuming horsepower to vary as the cube of speed the turbine part load efficiency can be approximated as a percentage of the design efficiency (Fig. 15-11).

EXAMPLES

Figs. 15-11 through 15-19 and 24-37 and 24-38 allow estimates to be made of steam rate, turbine efficiency, number of stages, and the inlet and exhaust nozzle diameters. The following examples illustrate the use of these figures:

FIG. 15-11

Part Load Efficiency Correction Factor vs Percent Power Multi-Valve Steam Turbines

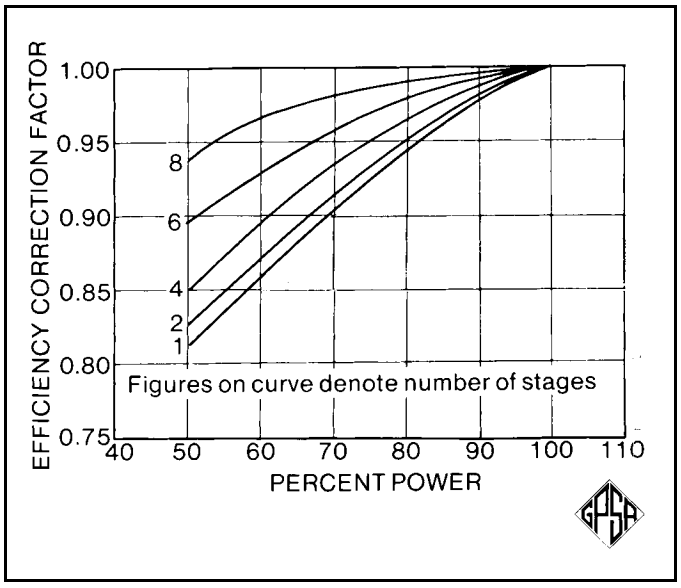
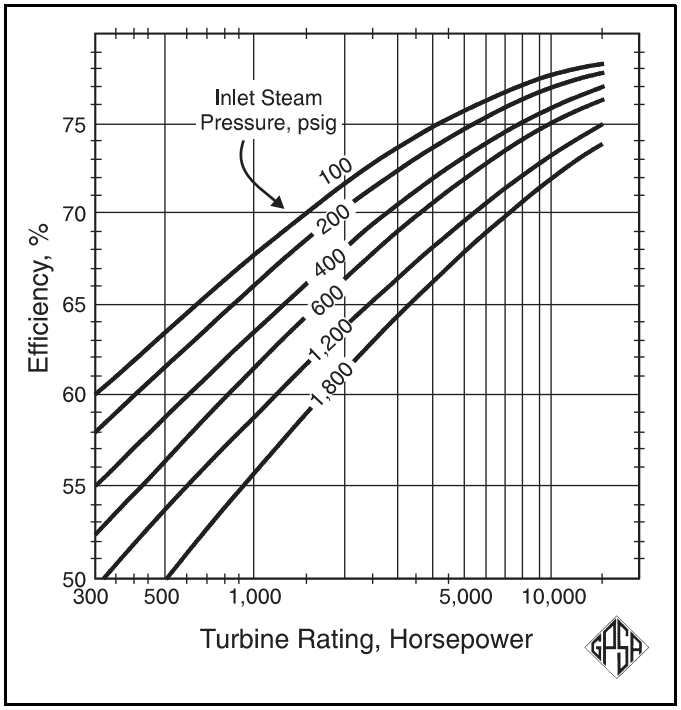


FIG. 15-12

Basic Efficiency of Multi-Valve, Multi-Stage Condensing Turbines



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Example 15-1 — Given a steam turbine application with the following characteristics:

Inlet Pressure	600 psia
Inlet Temperature	750°F
Exhaust Pressure	2 psia
Required Horsepower	6000
Speed	7000 rpm

Determine

The actual steam rate (ASR)

The inlet and exhaust nozzle diameters

The approximate number of stages

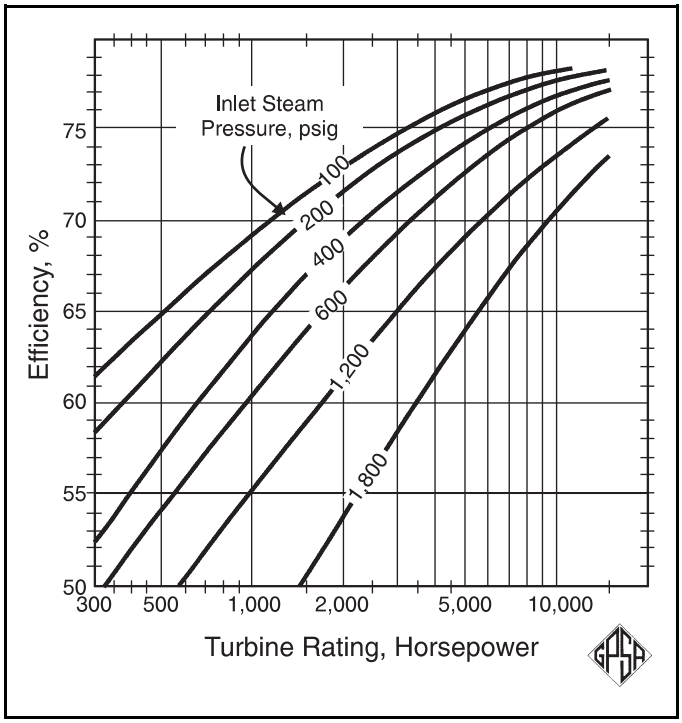
The steam rate at a partial load of 4000 hp and 6100 rpm

Solution Steps

Using Figs. 24-37 and 38, the theoretical steam rate (TSR) may be determined from the difference in the inlet enthalpy and the theoretical exhaust enthalpy (i.e. isentropic exhaust enthalpy), but first the inlet and exhaust states should be confirmed. Fig. 24-38 for superheated steam indicates that the inlet is superheated (i.e., 750°F is above the saturation temperature of 486.2°F), and gives an inlet entropy of 1.6109 Btu/(lb • °F). From Fig. 24-37, for saturated steam at the turbine exhaust pressure of 2 psia absolute, the liquid and vapor entropies are 0.1750 and 1.9200 Btu/(lb • °F). Since the inlet entropy is within this range, the theoretical exhaust must be two-phase. Had the exhaust-vapor entropy been equal to the inlet entropy, the exhaust would be single-phase vapor (i.e. at its dewpoint). Had the exhaust-vapor entropy been below the inlet entropy, the assumed two-phase exhaust would have been incorrect and Fig. 24-38 instead of 24-37 would be applicable.

FIG. 15-13

Basic Efficiency of Multi-Valve, Multi-Stage Non-Condensing Turbines



Inlet conditions at 600 psia and 750°F (the average of the values at 700°F and 800°F on Fig. 24-38):

$s = 1.6109$ Btu/(lb • °F)

$h = 1379.4$ Btu/lb

Exhaust conditions at 2.0 psia:

$s_f = 0.1750$ Btu/(lb • °F)

$s_g = 1.9200$ Btu/lb • °F)

$h_f = 94.03$ Btu/lb

$h_g = 1116.2$ Btu/lb

FIG. 15-14

Superheat Efficiency Correction Factor for Condensing Turbines

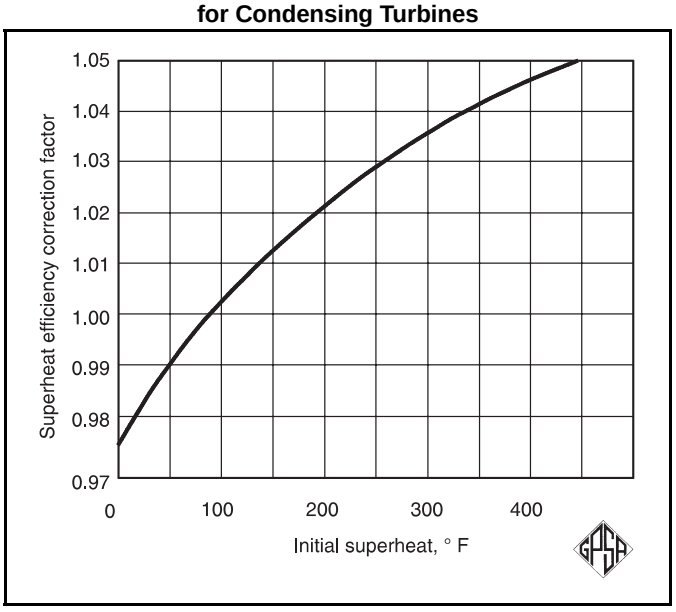
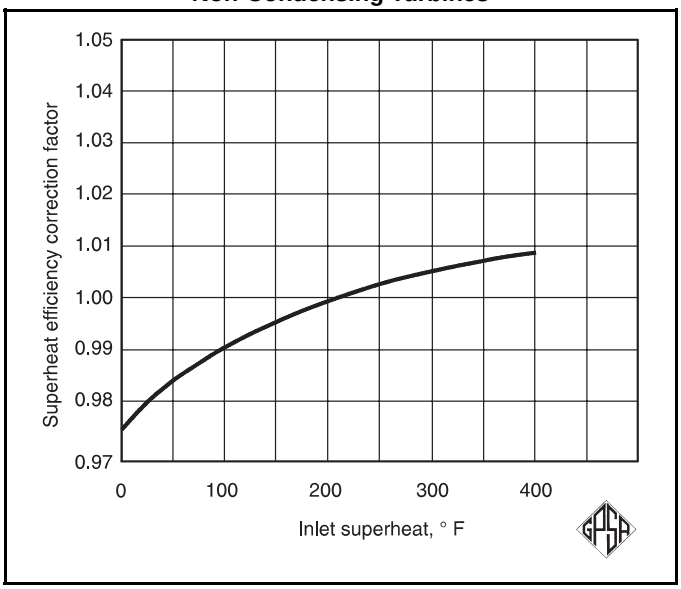


FIG. 15-15

Superheat Efficiency Correction Factor for Non-Condensing Turbines



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Letting x equal the liquid fraction in the exhaust, and equating the inlet and exhaust entropies:

$1.6109 = x\,(0.1750) + (1 - x)(1.9200)$

$x = 0.1771$

$1 - x = 0.8229$ (vapor fraction in the exhaust)

Exhaust enthalpy $= (0.1771)(94.03) + (0.8229)(1116.2)$

$= 935.2$ Btu/lb

Enthalpy change $= 935.2 - 1379.4$

$= -444.2$ Btu/lb

Substituting Btu = (hp • hr) /2544:

Enthalpy change = (-444.2/2544) = (-1/5.727)(hp • hr)/lb

TSR = the absolute value of the inverse of the enthalpy change

= 5.727 lb/(hp • hr)

Basic efficiency = 0.729 (Fig. 15-12)

Inlet saturation temperature = 486.2°F (first column Fig. 24-38)

Inlet superheat = 750 - 486 = 264°F

Superheat efficiency-correction factor = 1.03 (Fig. 15-14)

Speed efficiency-correction factor = 0.957 (Fig. 15-16)

Corrected efficiency = (0.729)(1.03)(0.957) = 0.719

ASR = 5.727/0.719 = 7.97 lb/(hp • hr)

F = (6000 hp) 7.97 lb/(hp • hr)

= 47,800 lb/hr

The inlet and exhaust diameters may be estimated from the equation:

$$D = \sqrt{\frac{(0.051)\,(F)}{(\rho v)}} \qquad \textbf{Eq 15-1}$$

A reasonable rule of thumb for maximum velocity of the inlet steam is 150 (ft/sec).

$\rho = 0.88$ lb/ft^3 @ 600 psia and 750°F

$$D = \sqrt{\frac{(0.051)\,(47{,}800)}{(0.88)\,(150)}}$$

$D = 4.3$ in.

A 4 in. NPS (minimum) inlet nozzle would be selected.

For exhaust sizing a maximum steam velocity of 450 ft/sec is a reasonable rule of thumb.

$\rho = 0.0057$ lb/ft^3 @ 2 psia

$$D = \sqrt{\frac{(0.051)\,(47{,}800)}{(0.0057)\,(450)}}$$

$D = 30.8$ in.

A 30 in. exhaust nozzle would be selected.

The number of stages may be estimated using Fig. 15-18. Drawing a horizontal line from the 7000 RPM indicates that between 1.5 and 2 stages per 100 Btu/lb of available energy would be acceptable.

Available Energy (theoretical)(i.e., the isentropic enthalpy change calculated above)

= 444.2 Btu/lb

FIG. 15-16

Speed Efficiency Correction Factor for Condensing and Non-Condensing Turbines

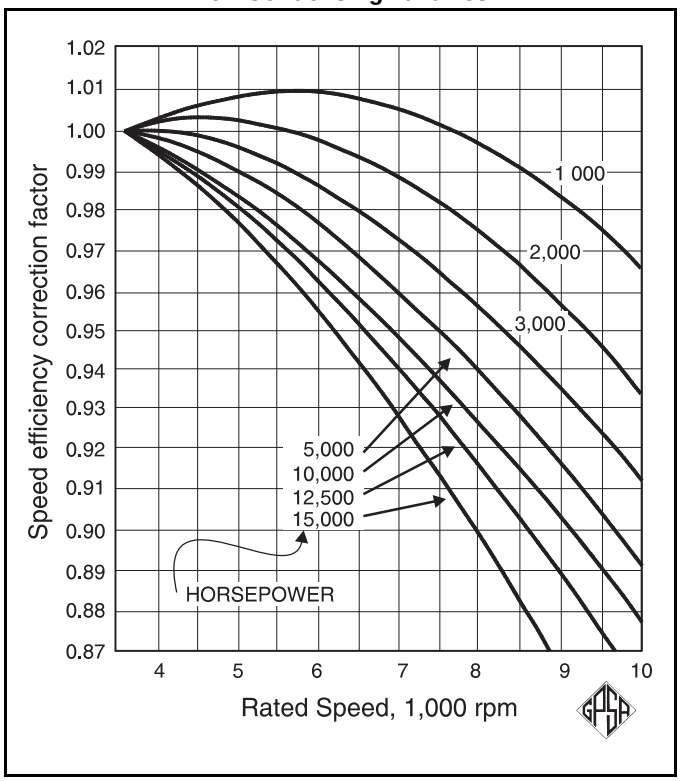
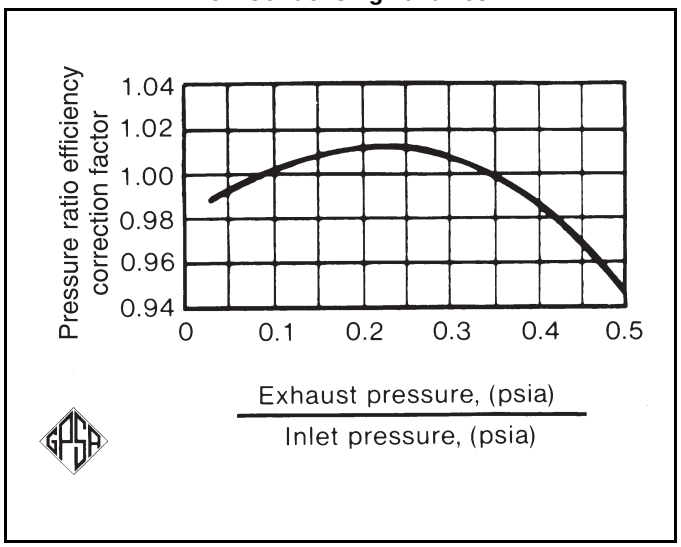


FIG. 15-17

Pressure Ratio Efficiency Correction Factor, Non-Condensing Turbines



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Number of Stages

$$= \frac{(1.5)\,(444)}{(100)} = 7 \text{ (approximately)}$$

or, Number of Stages

$$= \frac{(2)\,(444)}{(100)} = 9 \text{ (approximately)}$$

Nine stages would provide increased efficiency but at additional cost.

At partial load of 4000 hp and 6,100 RPM and assuming seven stages from Fig. 15-11, a part load efficiency factor of approximately 0.96 is obtained. From Fig. 15-12, the basic efficiency at 4000 hp and 6,100 RPM is estimated to be 0.71.

Efficiency = (0.96) (0.71) = 0.68

Actual Steam Rate = 5.73/0.68 = 8.43 lb/(hp • hr)

F = (4000) (8.43) = 33,700 lb/hr

Example 15-2 — Determine the ASR and total steam requirements for a multi-stage turbine and a single-stage turbine at the following conditions:

Inlet Pressure	250 psig
Outlet Pressure	100 psig
Inlet Temperature	500°F
Horsepower	900 hp
Speed	5000 rpm

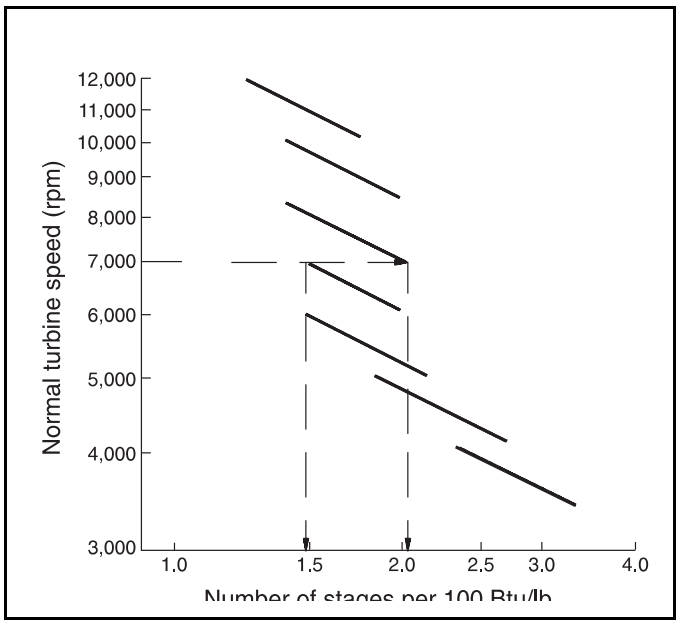
Solution Steps

For a multi-stage turbine:

Examining Figs. 24-37 and 38 in the same way as in Example 15-1, the turbine inlet is superheated, and the exhaust is two-phase.

FIG. 15-18

Stages Required per 100 Btu/lb of Available Energy as a Factor of Normal Turbine Speed



Inlet conditions at 250 psig (264.7 psia) and 500°F (interpolating linearly between 240 and 260 psia on Fig. 24-38):

s = 1.5873 Btu/(lb • °F)

h = 1261.8 Btu/lb

Exhaust conditions at 100 psig (i.e. 114.7 psia). From Fig. 24-37 interpolating linearly between 89.64 psia at 320°F and 117.99 psia at 340°F, get the following for 114.7 psia:

s_f = 0.4872 Btu/(lb • °F)

s_g = 1.5918 Btu/(lb • °F)

h_f = 308.9 Btu/lb

h_g = 1189.5 Btu/lb

Letting x equal the liquid fraction in the exhaust, and equating the inlet and exhaust entropies:

1.5873 = x(0.4872) + (1–x)(1.5918)

x = 0.0041

1–x = 0.9959 (fraction vapor in exhaust)

Exhaust enthalpy = (0.0041)(308.9) + (0.9959)(1189.5)
= 1185.9

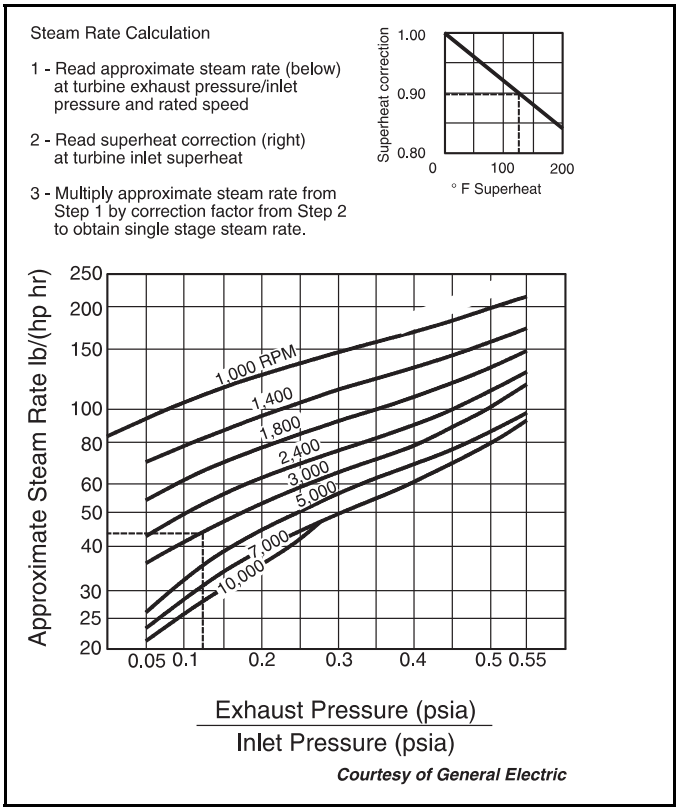
Enthalpy change = 1185.9 - 1261.8 = -75.9 Btu/lb

Substituting 1 Btu = (hp • hr) /2544:

Enthalpy change = (-75.9/2544) = (-1/33.5)(hp • hr) /lb)

FIG. 15-19

Single-Stage Application



Courtesy of General Electric

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TSR = the absolute value of the inverse of the enthalpy change

= 33.5 lb/(hp • hr)

Basic efficiency = 66% (Fig. 15-13)

Inlet saturation temperature = 406.0 °F (interpolating between 260 and 280 psia on Fig. 24-38)

Inlet superheat = 500 - 406 (Fig. 24-38) = 94°F

Efficiency-correction factor for superheat = 0.99 (Fig. 15-15)

Efficiency-correction factor for speed = 1.01 (Fig. 15-16)

Pressure ratio = (114.7 psia)/(264.7 psia) = 0.433

Efficiency-correction factor for pressure ratio = 0.97 (Fig. 15-17)

ASR = [33.5 lb/(hp • hr)]/[(0.66) (0.99) (1.01) (0.97)]

= 52.3 lb/(hp • hr)

F = [52.3 lb/(hp • hr)] (900 hp)

= 47,100 lb/hr

For a single-stage turbine

ASR = [75 lb/(hp • hr)] (0.93) (Fig. 15-19)

= 70 lb/(hp • hr)

F = [70 lb/(hp • hr)] (900 hp)

= 63,000 lb/hr

Mechanical Drive Gas Turbines

GENERAL

Gas turbines are extensively used in all phases of the gas industry as a source of shaft power. They are used to drive compressors, generators, and other equipment required to produce, process, and transport natural gas. The main advantages of gas turbines are:

- Compact, light weight design
- Minimal maintenance
- Short installation time

Compact, Lightweight Design

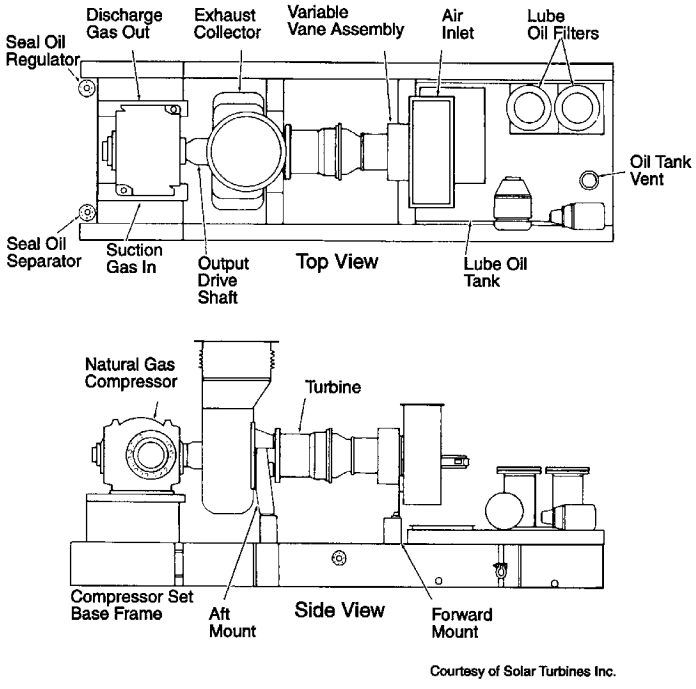
The compact, lightweight design of gas turbines makes them ideally suited for offshore platform installations, portable generating sets, remote sites, or any application where size and weight are important considerations.

Maintenance

Once installed, the gas turbine requires a minimum of routine maintenance. It is important to monitor the operating parameters of the turbine (pressures, temperatures, speed, vibration levels, etc.). This can often be done by an operator at a location remote from the actual turbine installation.

FIG. 15-20

Typical Gas Turbine Skid Layout



Courtesy of Solar Turbines Inc.

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Installation

The relatively light weight, compact size, and simple design of gas turbines make them an attractive choice where power must be quickly installed in the field. The gas turbine is often delivered on an integral one-piece baseplate with all auxiliary equipment installed and tested by the manufacturer. Thus, construction and start-up time are minimized.

GAS TURBINE TYPES

The gas turbine was first widely used as an aircraft power plant. However, as they became more efficient and durable, they were adapted to the industrial marketplace. Over the years the gas turbine has evolved into two basic types for high-power stationary applications: the industrial or heavy-duty design and the aircraft derivative design.

Heavy Duty

The industrial type gas turbine is designed exclusively for stationary use. Where high power output is required, 35,000 hp and above, the heavy duty industrial gas turbine is normally specified. The industrial gas turbine has certain advantages which should be considered when determining application requirements. Some of these are:

- Less frequent maintenance.
- Can burn a wider variety of fuels.
- Available in larger horsepower sizes.

Aircraft Derivative

An aircraft derivative gas turbine is based on an aircraft engine design which has been adapted for industrial use. The engine was originally designed to produce shaft power and later as a pure jet. The adaptation to stationary use was relatively simple.

Some of the advantages of the aircraft derivative gas turbines are:

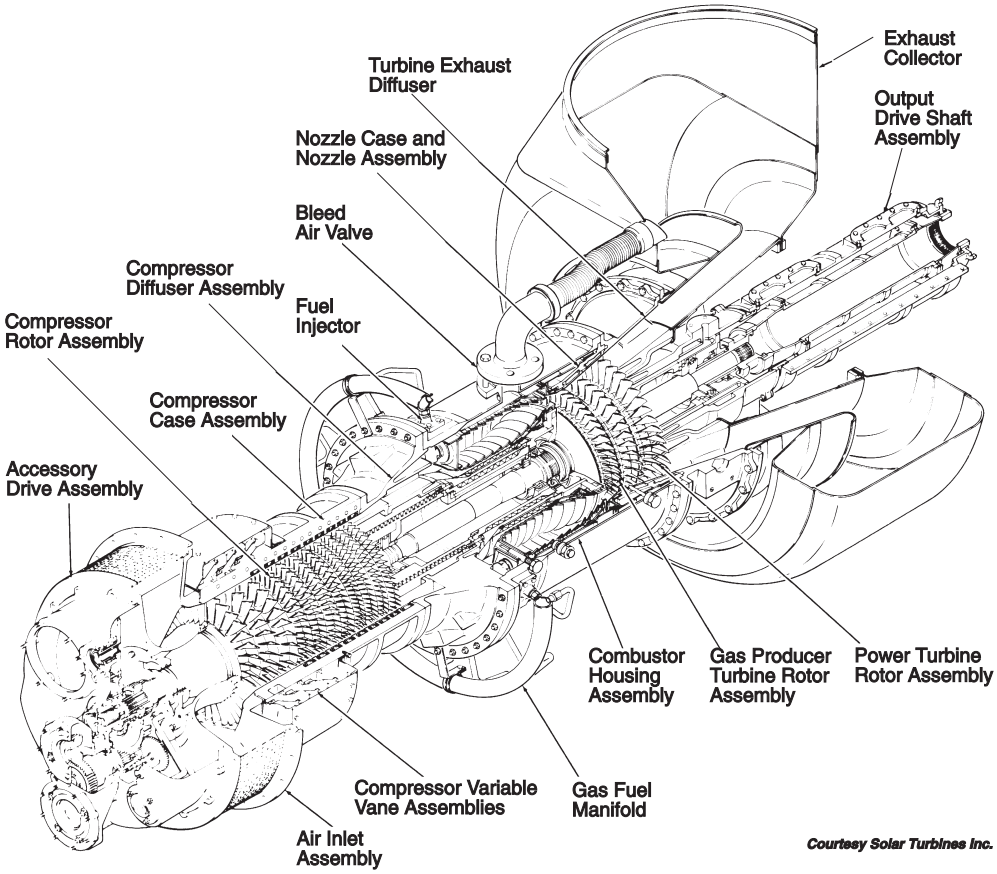
- Higher efficiency than industrial units.
- Quick overhaul capability.
- Lighter and more compact, an asset where weight limitations are important such as offshore installations.

Single Shaft/Split Shaft

Gas turbine designs are also differentiated by shaft configuration. In a single shaft design, all rotating components of the gas turbine are mounted on one shaft. In a split shaft design, the air compressor rotating components are mounted on one shaft, and the power turbine rotating components are mounted on another shaft. The driven equipment is connected to the power turbine shaft. The single shaft design is simpler, requiring fewer bearings, and is generally used where the speed range of the driven equipment is narrow or fixed (as in generator sets). It requires a powerful starting system since all the rotating components (including the driven equipment) must be accelerated to idle speed during the start cycle.

FIG. 15-21

Gas Turbine Internals



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A split shaft design is advantageous where the driven equipment has a wide speed range or a high starting torque. The air compressor is able to run at its most efficient speed while the power turbine speed varies with the driven equipment. The split shaft design allows a much smaller starting system since only the air compressor shaft is accelerated during the start cycle.

GAS TURBINE CYCLES

The basic gas turbine cycle is termed the Brayton cycle. The ideal Brayton cycle is a closed cycle consisting of an isentropic compression process; a constant pressure external heating process; an isentropic expansion process; and finally a constant pressure external cooling process which returns the working substance to the inlet state of the compression process. A schematic and TS diagram of the ideal Brayton cycle are shown in Fig. 15-22. The turbomachinery used in the process includes an axial flow or centrifugal compressor and an axial or radial flow turbine.

Simple Open Cycle

The simple open cycle gas turbine takes atmospheric air into the compressor as the working substance. Following compression, the air enters the combustion chamber where the temperature is raised by the combustion of fuel. The gaseous combustion products are then expanded back to the atmosphere through a turbine. A diagram of this cycle is shown in Fig. 15-23. The turbine in this system derives enough power from the high temperature gas to drive both the compressor and load.

Regenerative Ideal Brayton Cycle

The use of a regenerator in an ideal Brayton cycle acts to reduce the amount of available energy lost by external heat exchange. The system schematic is illustrated in Fig. 15-24. This available energy loss is due to irreversible heat input and is illustrated in Fig. 15-25. A heat exchanger or regenerator is placed in the system to transfer heat internally from the hot exhaust gas to the cooler air leaving the compressor. This preheating of the combustion air thus reduces the amount of external heat input needed to produce the same work output.

Combined Cycle

Instead of using the hot exhaust gas for regeneration, this approach uses exhaust gas to generate steam. This steam can be used either as a supplement to the plant steam system or to generate additional horsepower in a Rankine cycle. In the basic Rankine cycle, the hot exhaust gas passes successively through the superheater, evaporator, and economizer of the steam generator before being exhausted to the atmosphere. The steam leaving the boiler is expanded through a steam turbine to generate additional power. The cycle is closed by the addition of a condenser and feed water pump completing a basic Rankine cycle. Since the steam cycle does not require any additional fuel to generate power, the overall thermal efficiency is increased. Fig. 15-26 shows schematically a typical installation and its TS diagrams.

FIG. 15-22

Ideal Brayton Cycle

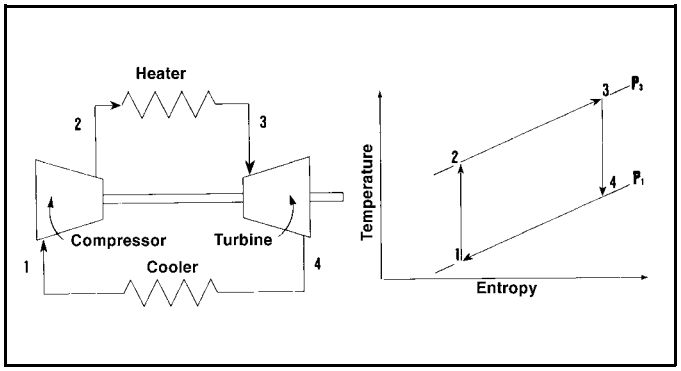


FIG. 15-23

Simple Open Cycle

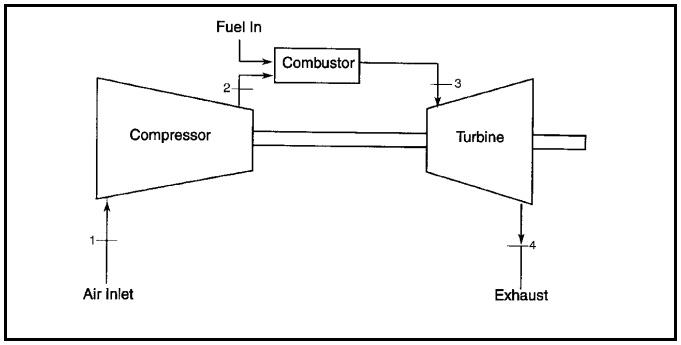


FIG. 15-24

Regenerative Ideal Brayton Cycle

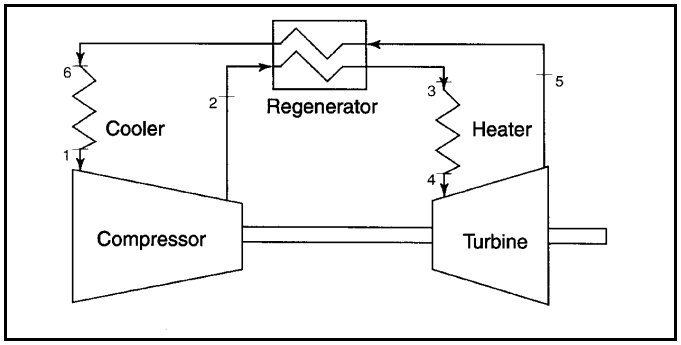
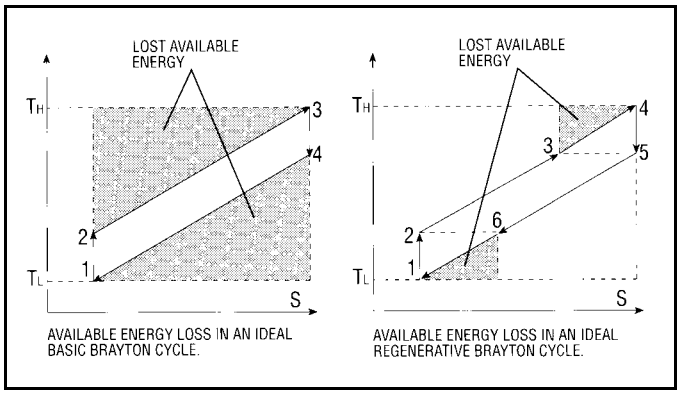


FIG. 15-25

Ideal Brayton Cycle Available Energy



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AUXILIARY SYSTEMS

Lube Systems

Two types of oils are used in lubricating gas turbine equipment. They are mineral and fire-resistant synthetic based oils. The oil type used depends on the bearing construction of the particular turbine.

Babbitt type sleeve and thrust bearings, typical of heavy duty turbines, use a mineral based oil. Driven equipment such as compressors, gear, and generators also use this type oil, thus a common, combined lube system can be provided for the train.

Aircraft derivative gas generators all incorporate anti-friction type ball and/or roller bearings. A synthetic oil is used in this service and is provided in a separate system from the mineral oil system used to lubricate the driven equipment. An oil scavenging system is also typical of these gas generators. Engine mounted pumps are used to scavenge oil from the main bearing pumps and return it to the reservoir.

Air Filtration

The primary reason for inlet air filtration is to prevent unwanted dirt from entering the gas turbine. By reducing the contaminants which contribute to corrosion, erosion, and fouling, the gas turbine life is extended. For a more complete discussion of filtration see Section 7.

There are various types of filters. The main types are as follows:

Inertial — This type removes the larger particulates from the inlet air.

Prefilters — These are medium filters usually made of cotton fabrics or spun-glass fibers, used to extend the life of a high efficiency filter further downstream.

Coalescers — These filters are used to remove moisture from the inlet air system.

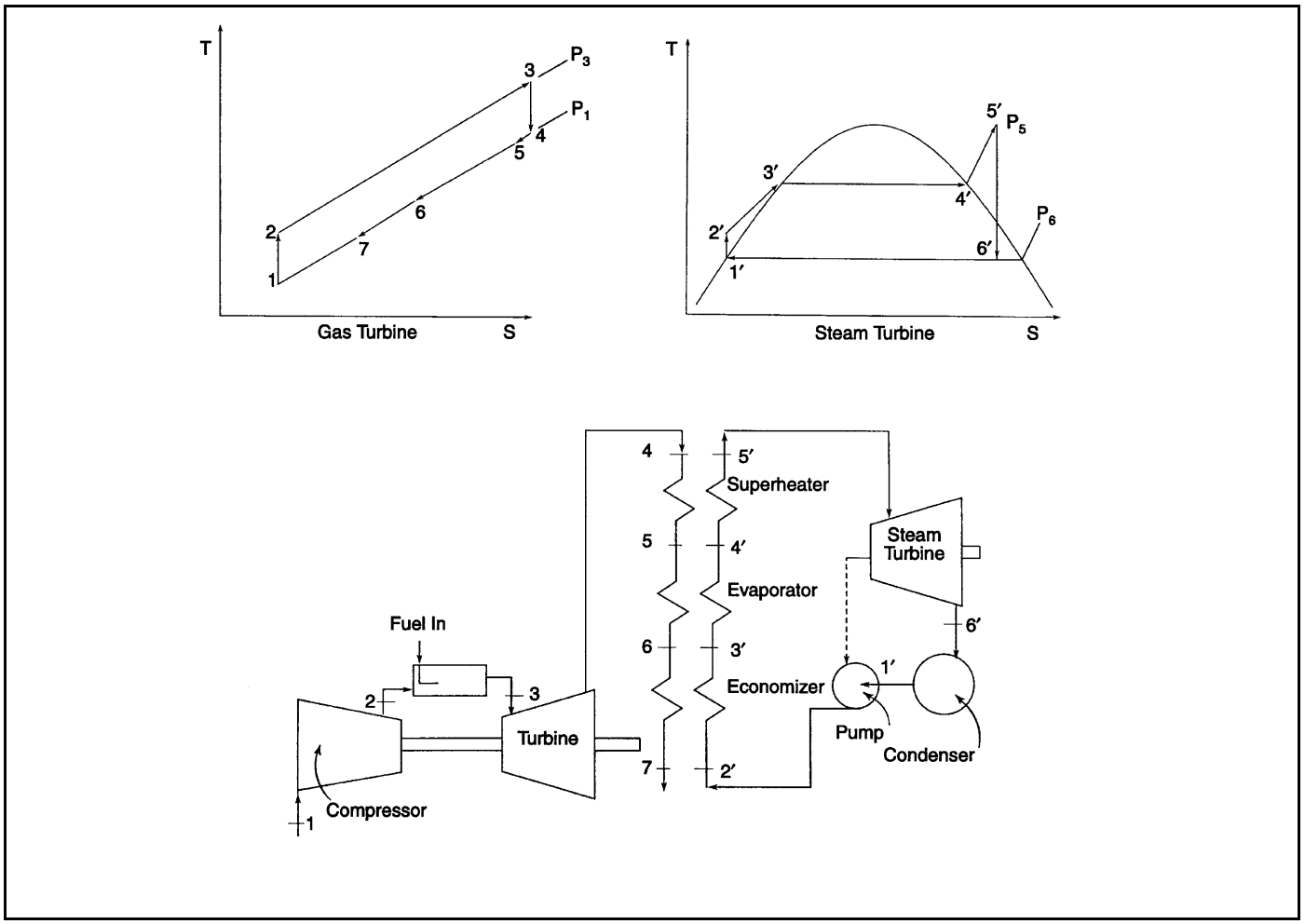
High Efficiency Media — These filters remove smaller dirt particles from the inlet air.

Marine or Demister — These filters are used in marine environments to remove both moisture and salt.

Self-Cleaning — These filters are composed of a number of high efficiency media filter "banks." Air is drawn through the media at a low velocity and, at a predetermined pressure drop across the system, a reverse blast of air removes built-up dirt on the filter and lowers the pressure drop. This filter can be used in any environment. It is particularly useful in colder

FIG. 15-26

Combined Cycle



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FIG. 15-27

Gas Turbine Air Filtration

Type of Environment	Suggested Filtration
Rural Country	High Efficiency Media
Urban/Industrial	Inertial & High Efficiency Media
Desert	Inertial and Media or Self-Cleaning
Tropical	Inertial & Media
Arctic	High Efficiency Media with Anti-Icing or Self-Cleaning
Offshore	Demisters

climates where ice build-up is a problem. The reverse blast of air also removes any ice that has built up on the filter.

Another method of eliminating icing problems is to install an anti-icing system. In this system, heated air from the gas generator discharge is introduced through distribution manifolds immediately downstream of the inlet air silencer.

The selection of a filtration system is largely dependent on the site location and operating conditions. Fig. 15-27 suggests filtration for various types of environments.

Since filters do protect the gas turbine and help extend its useful life, some type of filtration is always recommended.

Acoustics

The noise created by a gas turbine engine is considerable and must be reduced to protect plant personnel and minimize environmental impact. The main sources of noise in a gas turbine installation are the intake, the exhaust, and casing radiated noise.

The noise associated with the intake is characterized as high frequency noise. This type of noise is the loudest and most disturbing to the ear since it is in a range where hearing is most sensitive. The second most objectionable noise is produced by the gas generator and power turbine and is radiated from the casing. Although the exhaust noise contains more energy, the casing noise is more objectionable since it contains more noise in a frequency range where the ear is most sensitive. The exhaust noise is a low frequency noise which is only slightly audible. It does, however, possess a considerable amount of energy which results in a detectable pressure change.

A variety of methods can be used to attenuate noise. The most common are the use of silencers and enclosures. The inlet noise is the first area considered since this is where the largest amount of sound power is produced. Inlet noise is the loudest directly in front of the inlet opening. Consequently, the least expensive method for obtaining some of the required noise reduction is to place an elbow at the inlet. Additional silencing is usually necessary and can be attained by the use of acoustic baffles before the elbow.

Casing radiated noise can be reduced by using an acoustical enclosure over the turbine. If an enclosure is used, it is necessary to provide gas and fire detection and fire extinguishing equipment inside the enclosure.

The last major source of noise to be silenced is the gas turbine exhaust noise. Since most turbines exhaust vertically, there is generally no need for an elbow. However, a silencer with acoustic baffles is needed and the exhaust ducting should be sound insulated.

Gas Turbine Performance

The performance of a gas turbine is usually expressed in terms of power and heat rate. Power is the net power available at the output shaft of the turbine after all losses and power take-offs have been subtracted.

Heat rate is a measure of thermal efficiency or the amount of heat energy (in the form of fuel) which must be input to the gas turbine to produce the output power. Heat rate is usually expressed in terms of Btu/(hp • hr) or Btu/(kW • hr) based on the lower heating value of the fuel. Heat rate and thermal efficiency are related as follows:

$$\text{Thermal efficiency} = \frac{2544}{\text{Heat Rate, } \frac{\text{Btu (LHV)}}{\text{hp} \cdot \text{hr}}}$$

$$= \frac{3414}{\text{Heat Rate, } \frac{\text{Btu (LHV)}}{\text{kW} \cdot \text{hr}}}$$

Power and heat rate both vary depending on environmental conditions such as ambient air temperature, altitude, barometric pressure, and humidity. Therefore, when performance is stated for a gas turbine, the ambient conditions must be defined. In order to compare different gas turbines, a set of standard conditions known as ISO (International Standards Organization) conditions have been defined as follows:

ISO Conditions: Ambient Temperature = 59°F = 15°C
Altitude = 0 ft (sea level)
Ambient Pressure = 29.92 in. Hg
Relative Humidity = 60%

All gas turbine performance is stated in ISO conditions. To arrive at site rated horsepowers, the ISO conditions must be corrected for the following:

Altitude (Fig. 15-28)
Inlet Losses (Fig. 15-29)
Exhaust Losses (Fig. 15-30)
Temperature (Fig. 15-31)
Humidity (below)

For changing relative humidity, the power output does not change, and the heat rate changes only slightly. For example, for an increase in relative humidity from 60 to 100 percent, a typical correction factor for the heat rate is 1.0016. For a decrease to zero percent, a typical correction factor is 0.9979.

Performance is also affected by other installation variables including power take-offs and type of fuel used. Inlet loss is the pressure drop which occurs as the outside air passes through the inlet filters and plenum. Similarly, exhaust loss is the pressure drop through the exhaust stack, silencers, and heat recovery equipment (if any) which creates a back pressure on the turbine. Power take-offs include any devices such as oil pumps, generators, etc. which are directly driven from the gas turbine output shaft and thus reduce the available output power. Sometimes it is necessary to correct power and/or heat rate for the type of fuel used in the gas turbine. The turbine manufacturer's performance brochure should be consulted for necessary corrections.

The following example shows the method of calculating performance for a gas turbine at site conditions using data typically supplied in the manufacturer's performance brochure.

Example 15-3 — Calculate maximum available site power and heat rate for the example gas turbine at the following conditions:

Turbine ISO Horsepower = 27,500

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FIG. 15-28

Altitude Correction Factor

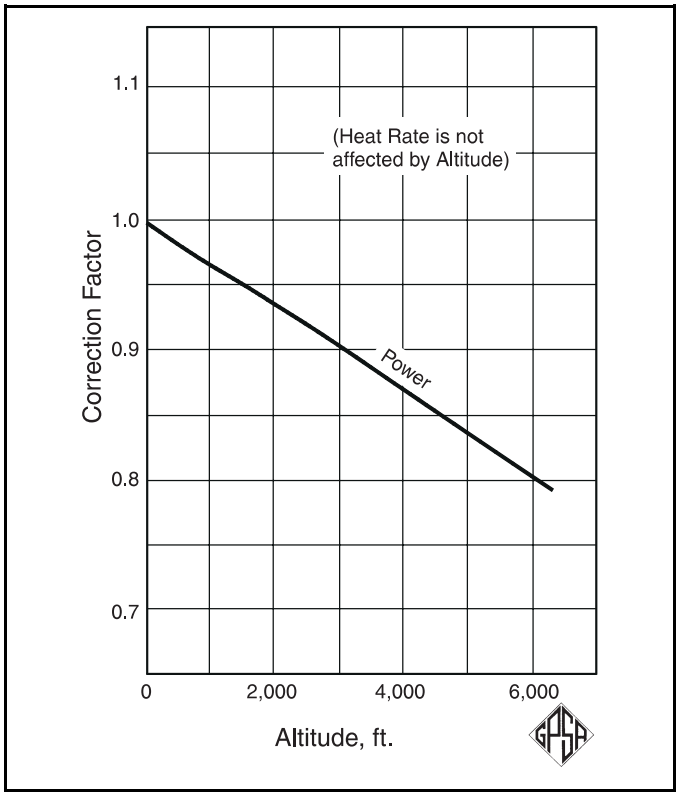


FIG. 15-29

Inlet Loss Correction Factor

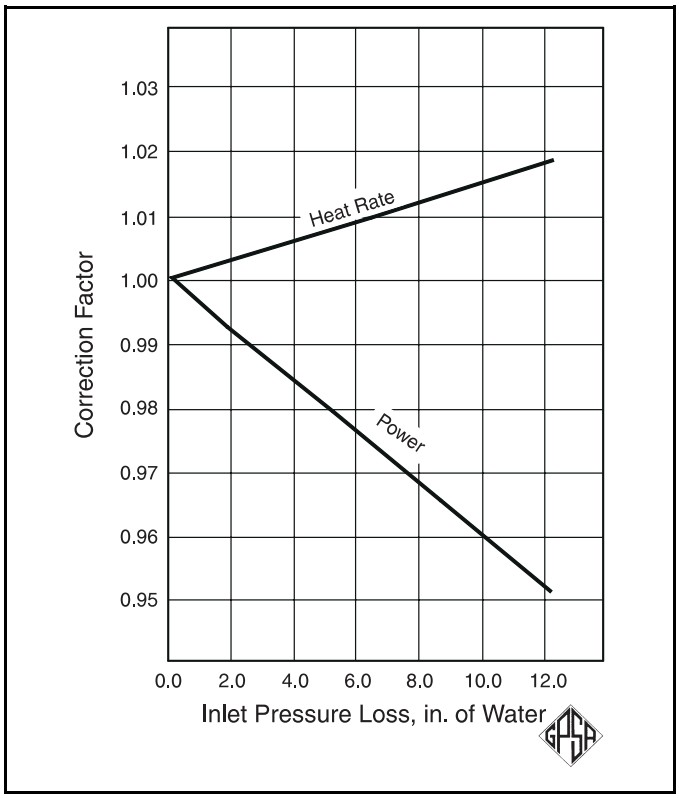


FIG. 15-30

Exhaust Loss Correction Factor

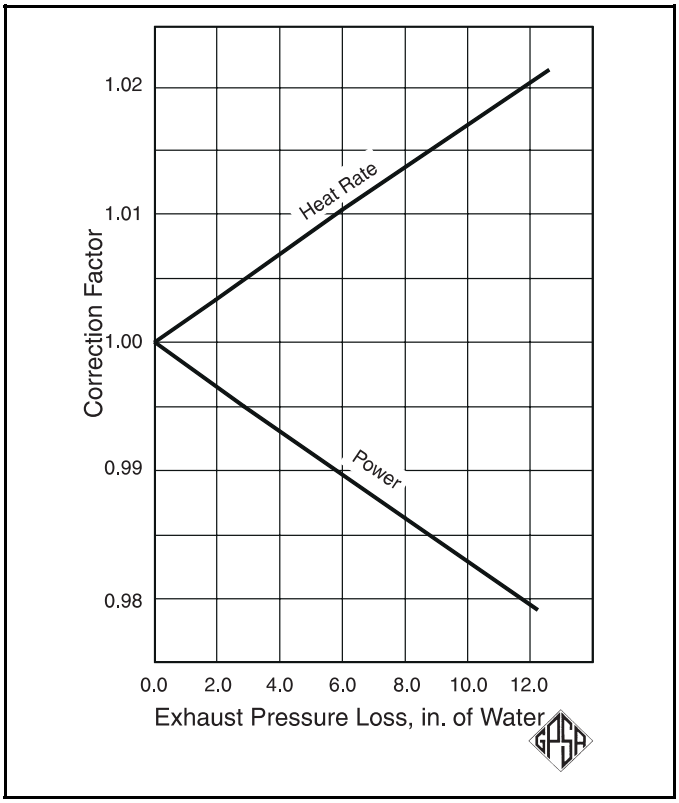
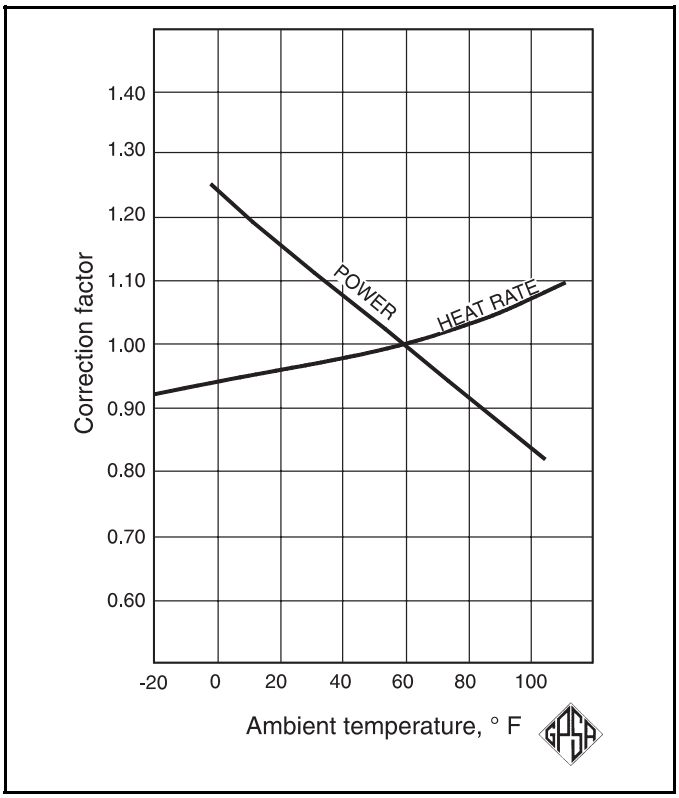


FIG. 15-31

Ambient Temperature Correction Factor



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Turbine ISO Heat Rate	=	7,090 Btu/(hp • hr)
Ambient Temperature	=	80°F
Altitude	=	1000 ft (above sea level)
Inlet Pressure Drop	=	4 in. H_2O
Exhaust Pressure Drop	=	2 in. H_2O
Relative Humidity	=	60%
Fuel	=	Natural Gas

Solution Steps

Find the power altitude correction factor from Fig. 15-28. For 1000 ft altitude, the correction factor is 0.965.

Find power inlet loss correction factor from Fig. 15-29. For 4 inches of water, the correction factor is 0.984.

Find power exhaust loss correction factor from Fig. 15-30. For 2 inches of water, the correction factor is 0.9965.

Find the power ambient temperature correction factor from Fig. 15-31. For 80°F the correction factor is 0.915.

Since relative humidity is 60% and fuel is natural gas, no corrections are required.

Calculate the maximum available site power by multiplying maximum-no-loss power by each of the correction factors.

Power (site) = power (0.965) (0.984) (0.9965) (0.915)
Power (site) = 27,500 (0.965) (0.984) (0.9965) (0.915)
Power (site) = 23,800 hp

For the heat rate find the inlet loss correction factor, exhaust loss correction factor, and ambient temperature correction factor from Figs. 15-29, 15-30, and 15-31, respectively. (Note: Heat rate is not affected by altitude.)

Inlet loss factor	=	1.0065
Exhaust loss factor	=	1.003
Temperature factor	=	1.03

Calculate site heat rate by multiplying no-loss heat rate by the correction factors.

Heat rate (site) = (Heat rate) (1.0065) (1.003) (1.03)
Heat rate (site) = [7090 Btu/(hp • hr)](1.0065)(1.003)(1.03)
Heat rate (site) = 7370 Btu/(hp • hr)

The above calculation procedures may vary slightly with different manufacturers but will follow the same principles.

Basic specifications for some of the commonly used gas turbine engines are shown in Fig. 15-32.

Gas Turbine Emissions

The gas turbine, in general, is a low emitter of exhaust gas pollutants relative to other heat engines in similar service. This is because the fuel is burned with ample excess air to ensure complete combustion at all but minimum load conditions. It is unique in its ability to burn a wide variety of fuels making each application unique in terms of exhaust emissions. However, gas turbine engine emissions recently have become a major factor in the design, selection, and operation of the unit. Various federal, state, and local authorities have issued standards and codes to control pollution of the atmosphere.

Carbon monoxide (CO) emissions occur because of incomplete combustion of fuel carbon. CO emissions for distillate and other liquid fuels are generally higher than for natural gas.

Unburned hydrocarbons (UHC) are formed by the incomplete combustion of fuel. Like CO emissions, they are directly related to combustion efficiency. However, because most gas turbine units on the market today have good combustor designs, the CO and UHC emissions are of secondary importance to NO_x emissions.

Sulfur oxides (SO_x) exhausted from gas turbines are a direct function of sulfur content in the fuel.

The high temperature and oxygen content during combustion tends to favor the formation of SO_3 and SO_2 at equilibrium. Sulfur oxide emissions from pipeline natural gas are virtually zero while wellhead gases, process gases, coal gases, and other fuels may contain significant quantities of sulfur in the form of H_2S.

Gas turbine particulate emissions are influenced by the fuel properties and combustion conditions. Particulates generally refer to visible smoke, ash, ambient non-combustibles, and products of erosion and corrosion in the hot gas path. Particulate and smoke emissions are usually small when burning natural gas, but are a significant consideration when operating on liquid fuels.

Of the exhaust components the most significant are the oxides of nitrogen (NO_x). The amount of NO_x produced is a function of the fuel burned, firing temperature, compressor discharge temperature, and residence time in the combustion zone. Since the trend towards high turbine efficiencies leads to higher pressure ratios and firing temperatures, the emission rates of NO_x are higher for these units.

Nitrogen oxides are categorized into two areas according to the mechanism of formation. NO_x formed by oxidation of free nitrogen in the combustion air or fuel is called "thermal NO_x," while that due to oxidation of organically bound nitrogen in the fuel is referred to as "organic NO_x." As implied by the name, thermal NO_x are mainly a function of the stoichiometric flame temperature. The formation of thermal NO_x is on the order of parts per million (by volume) or ppmv; however, the conversion of organic NO_x is virtually 100%. Efforts to reduce thermal NO_x by reducing flame temperatures have little effect on, and actually may increase, organic NO_x.

UHC emissions can be reduced by proper combustor design for maximum efficiency. Sulfur oxides can be eliminated by removing sulfur compounds from the fuel. Similarly, particulates can be minimized by appropriate fuel treatment. However, reduction of NO_x formation also produces increased inefficiency.

Two general approaches are used for NO_x reduction:

- The use of an inert heat sink such as water or steam injection
- Modifications of fuel-air ratios and combustor design

In addition to these two methods, catalytic combustion is being researched.

Water or Steam Injection — Water or steam injection is an effective way to reduce NO_x exhaust emissions. Two areas of caution in the design of this system must be considered. The first is the dynamic effect that water injection has on the combustion zone in terms of flame stability and dynamic pressures. Inadequate design could adversely affect hardware life. Also the rate of CO emissions increases with the rate of water injection. It is not effective in reducing organic NO_x emissions and may actually contribute to them.

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FIG. 15-32

2003 Basic Specifications – Gas Turbine Engines

Model	Power Rating (ISO Rating) hp	Heat Rate (LVH) Btu/hp-hr	Pressure Ratio	Power Shaft RPM	At ISO RATING CONDITIONS		
					Turbine Inlet Temp. °F	Exhaust Flow lb/s	Exhaust Temp °F
Alstom Power							
TYPHOON	6626	7950	13.0	13000	2016	39.6	1013
TORNADO	10288	7593	12.6	10450	1897	64.9	912
CYCLONE	17986	7027	16.8	9500	2300	86.8	1031
GT35C	22885	7925	12.3	6300	1562	204.8	711
GT35C	23287	7760	12.0	3450	1562	202.8	709
GT10B	34102	7250	14.0	7700	–	177.2	1009
GT10C	40392	6820	17.6	6500	–	200.6	964
Dresser-Rand							
DR60	19109	7020	20.1	7900	–	104.5	916
DR61	30982	6785	18.8	5500	–	150.1	986
DR61G	31235	6772	18.8	3600	–	152.1	975
DR61GP	42067	6442	22.2	3600	–	192.2	959
DR61P	42067	6276	22.3	6200	–	186.1	955
DR63G	59335	5941	28.5	3600	–	282.2	824
GE Aero-Energy Products							
LM1600 (PA)	19110	7016	20.2	7900	–	104.3	916
LM2000 (PE)	24300	7117	16.0	3600	–	1056.0	894
LM2500 (PE)	30400	7017	18.1	3000	–	1177.3	993
LM2500 (PK)	40044	6702	22.6	3000	–	1144.2	966
LM6000 (PC)	60000	5938	29.1	3600	–	1056.0	894
GE Power Systems, GE Oil & Gas							
PGT5	7295	9500	8.6	10290	–	56.9	991
PGT10	14295	7950	13.8	7900	–	93.3	910
GE10	15675	7810	15.2	7900	–	103.4	903
PGT16	19120	7020	20.1	7900	–	104.5	919
PGT25	31200	6750	17.9	6500	–	151.9	977
PGT25+	42060	6180	21.5	6100	–	185.8	932
LM6000	60000	5980	28.5	3600	–	280.0	853
MS3002R	14520	7300	7.3	6500	–	117.1	698
MS3002	15140	9500	7.1	6500	–	117.1	1008
MS5002R	35600	6990	8.7	4670	–	276.7	693
MS5002C	38005	8700	8.8	4670	–	274.0	963
MS5002D	44680	8410	10.8	4670	–	311.7	950
MS6001B	58380	7650	11.9	5111	–	319.7	1011
MS7001EA	115330	7700	12.5	3600	–	661.4	1000
GE Power Systems							
M3142R(J)	14520	7386	7.3	6500	–	116.8	698
M3142(J)	15140	9500	7.1	6500	–	116.8	1008
M5261(RA)	26400	9380	7.5	4860	–	202.8	988
M5322R(B)	32000	7068	8.2	4670	–	251.3	666
M5352(B)	35000	8828	8.6	4670	–	271.2	916
M5352R(C)	35600	6990	8.7	4670	–	266.8	693
M5382(C)	38000	8701	8.9	4670	–	277.8	959
M6581(B)	58380	7650	12.0	5111	–	308.6	1011
M7121(EA)	115630	7722	11.9	3600	–	659.2	999
Kawasaki Heavy Industries, Ltd.							
S2A-01	915	12189	8.5	1500/1800	–	11.2	885
M1A-13	2081	9976	9.4	1500/1800	–	17.6	968
M1A-13D	2083	10078	9.5	1500/1800	–	17.4	986
M1T-13	4097	10137	9.4	1500/1800	–	35.5	968

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FIG. 15-32 (Cont'd.)

2003 Basic Specifications – Gas Turbine Engines

Model	Power Rating (ISO Rating) hp	Heat Rate (LVH) Btu/hp-hr	Pressure Ratio	Power Shaft RPM	At ISO RATING CONDITIONS		
					Turbine Inlet Temp. °F	Exhaust Flow lb/s	Exhaust Temp °F
M1T-13D	4104	10229	9.5	1500/1800	–	35.1	986
M7A-01	7607	8342	12.7	1500/1800	–	48.0	1013
M7A-01D	7245	8450	12.7	1500/1800	–	48.0	1008
M7A-01ST	8956	7481	12.7	1500/1800	–	48.9	981
M7A-02	9531	8120	15.9	1500/1800	–	59.5	972
M7A-02D	9276	8162	15.9	1500/1800	–	59.5	955
L20A	24138	7260	18.0	1500/1800	–	127.4	1013
MAN Turbomaschinen AG GHH Borsig							
THM1203A	8046	10871	7.8	7800	1724	77.6	959
THM1203AR	7711	7951	7.8	7800	1724	77.6	–
THM1304-9	12069	8835	9.6	8600	1760	99.0	925
THM1304-10	13008	8715	10.0	8600	1787	99.9	932
THM1304-10R	12606	7012	10.0	8600	1787	99.9	–
THM1304-11	15019	8206	11.3	8600	1823	108.2	941
FT8	34690	6615	19.5	5500	–	188.5	856
FT8 Plus	37940	6580	20.2	5500	–	193.3	898
Pratt & Whitney							
ST6L-721	681	10885	6.8	33000	–	6.6	957
ST6L-795	909	10320	7.4	33000	–	7.1	1092
ST6L-813	1137	10108	8.5	30000	–	8.6	1051
ST18A	2630	8411	14.0	20000	–	17.6	990
ST30	4479	7987	15.0	14875	–	28.7	955
ST40	5416	7689	15.0	14875	–	30.8	1011
FT8	34690	6615	19.5	5500	–	188.5	856
FT8 Plus	37940	6580	20.2	5500	–	193.3	898
Rolls-Royce							
501-KC5	5500	8495	9.4	13600	–	34.2	1060
501-KC7	7400	7902	13.5	13600	–	46.1	968
AVON 2648	20360	8660	8.8	5500	–	169.8	828
AVON 2656	21000	8405	8.8	4950	–	169.8	819
RB211-6556	34900	7100	20.1	4950	–	202.8	910
RB211-6562	39600	6705	20.8	4800	–	208.3	916
RB211-6762	40750	6565	21.5	4800	–	210.5	918
RB211-6761	44500	6290	21.5	4850	–	207.2	937
TRENT 50	70470	5910	35.0	3000/3600	–	341.7	931
TRENT 60	79050	6140	35.0	3000/3600	–	359.6	802
Solar Turbines Incorporated							
Saturn 20	1589	10366	6.2	22300	–	14.3	968
Centaur 40	4694	9104	10.2	15500	–	41.9	842
Centaur 50	6155	8482	10.6	16500	–	41.7	959
Centaur 50L	6303	8206	10.5	14300	–	41.0	950
Taurus 60	7697	7948	12.0	14300	–	47.8	950
Taurus 70	10353	7287	16.8	12000	–	58.9	923
Mars 90	13222	7651	16.5	9500	–	88.6	869
Mars 100	15006	7489	17.6	9500	–	92.2	905
Tital 130	19498	7121	16.4	8850	–	108.0	914
Vericor Power Systems							
ASE 8	704	11764	10.5	1800/1500	1666	7.9	909
ASE 40(VPS3)	4407	9001	8.8	15400	2028	28.2	1108
ASE 50(VPS4)	5064	8313	10.2	16000	1911	31.1	1044

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Modification of Fuel-Air Ratio and Combustor Design — An increased number of gas turbines are available with low NO_x design based on modifications to fuel-air ratio and combustor design. NO_x content in flue gas as low as 10–25 ppm can be achieved by this method.

Different vendors have different approaches to effectively control the fuel-air ratio to reduce NO_x content. However, low NO_x, turbines are available, both for single-fuel and dual-fuel systems from most vendors.

Low NO_x designs make the gas turbines more complex, especially for dual-fuel systems. It is therefore recommended to carefully review the need for dual-fuel low NO_x machines.

In summary, the characteristics of gas turbine emissions must be considered for each application, since each is unique to the turbine, installation, fuel, and operation. All these factors are important in matching the gas turbine to the job.

Electric Motors

INTRODUCTION

Electric motor drives offer efficient operation and add flexibility to the design of petroleum refineries, petrochemical plants, and gas processing plants. Electric motors can be built with characteristics to match almost any type of load. They can be designed to operate reliably in outdoor locations where exposed to weather and atmospheric contaminants.

Proper motor application is essential if reliable performance is to be achieved. Critical items to consider are load characteristics for both starting and running conditions, load control requirements, power system voltage and capacity, and any conditions at the plant site that could affect the type of motor enclosure.

A-C MOTOR TYPE AND SELECTION

One of the first considerations in motor selection is to choose between a squirrel cage induction and a synchronous motor. The induction motor has the advantage of simplicity. It is a rugged machine and has an outstanding record for dependability. In general it can accelerate higher load inertias than synchronous motors and usually will do so in less time. Induction motor control is simple and no excitation equipment is required. Its principal disadvantages are that it operates at lagging power factor and has higher inrush (starting) current. Up to about 5000 horsepower, induction motors are normally preferred. Above this, induction and synchronous motor costs converge, and synchronous motors are often chosen.

The constant-speed synchronous motor has inherent advantages that often make it the logical choice for industrial drive applications at lower power ratings. Load speed can be exact. Torque characteristics of the motor can be varied by design to match the requirements of the driven load and the available power supply. Starting, pull-in, and pull-out torques can be selected over a wide range. Power factor improvement is available with rated power factor of unity, leading, or even lagging.

ELECTRICAL SYSTEM

Induction Motors

In general, induction motors tend to draw more starting current at a lower power factor than synchronous motors of the same size and speed. This results in a greater voltage drop on the system when the motor is started. If the motor is driving a high inertia load, such as a fan or compressor, the lower terminal voltage will increase the temperature rise of the squirrel cage winding during acceleration. As a rule of thumb, voltage drops on starting greater than 20% may require special motor designs. These designs may reduce the full load efficiency of the motor during normal operation by one or two percent. It is best to furnish a power supply that will limit the voltage drop to 20% or less when starting the largest motor on a fully loaded bus.

Once started the induction motor is a stable machine. Most motors can easily ride through a 25 to 30% dip in system voltage caused by external faults or switching. Overall system stability and continuity may be achieved by using large induction motors. This is, however, accomplished at the expense of lower power factor and efficiency.

Synchronous Motors

The synchronous motor is usually easier to start than an induction machine. The system voltage drop on starting is less for a given horsepower motor. However, synchronous motors have less thermal capacity in their windings and may be more severely taxed when accelerating high inertia loads.

Once synchronized and running, synchronous motors present special system problems. They may tend to pull out of synchronism on voltage dips that induction motors can ride through. A gradually increasing load from zero to 125% of rated load will be easily accommodated. A suddenly applied load of 125% can easily cause the motor to pull out of synchronism with the electrical system.

When applying large synchronous machines to a system it is important to perform a transient load study. This will help ascertain if the electrical system is capable of supporting the motor demands under transient conditions.

Speed

3000 to 3600 rpm — Synchronous motors are seldom economical for this range because of the high cost of rotor construction. Although large (i.e., greater than 22,000 hp) two-pole 3600 rpm synchronous motors have been built, slower speed motors with step-up gears, or squirrel-cage induction motors are usually a more economic choice.

900 to 1800 rpm — Synchronous motors above 5000 hp are widely used for pumps and for centrifugal compressors with speed increasers. The need for power-factor correction, high efficiency, low inrush, or constant speed may favor synchronous motors below 5000 hp.

For applications above 2500 hp requiring speed increasers, 1200 rpm unity-power-factor synchronous motors should be

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evaluated against 1800 rpm motors since the lower-speed motors are slightly more energy efficient. For 1200 rpm loads, synchronous motors may be more economical at 1250 hp and above. For 900 rpm loads, synchronous motors should be considered at 1000 hp and above.

Motor requirements below 500 hp in the 900 to 1800 rpm speed range are normally handled by standard induction motors.

514 to 720 rpm — Synchronous motors are often selected above 1 hp per rpm, such as 800 hp at 720 rpm, 700 hp at 600 rpm, and 600 hp at 514 rpm.

Below 514 rpm — The synchronous motor should be considered for sizes down to 200 hp because of higher efficiency, improved power factor, and possible lower cost. At high voltages (4 kV and above), the synchronous motor becomes more economical at even lower horsepowers.

Motor Voltage

The proper selection of voltage for a given motor drive can vary from a routine procedure to a complex study requiring a complete electrical system analysis. In many instances the in-plant distribution system is well established at a particular voltage, say 2300 (2.3 kV). The new machine may be small compared to available system capacity on the 2.3 kV bus so no problem is involved in purchasing a standard motor of that voltage. In more complicated cases additional substation capacity may be necessary to accommodate the new machine. However, when very large units are to be added, many factors must be considered. A new distribution voltage level, a new transmission line, or a higher voltage transmission from the electric utility might be necessary.

MOTOR ENCLOSURES

Motor enclosure selection should be predicated upon the environmental conditions under which the motor must operate. Directly related to this is the amount of maintenance required to provide long-term reliability and motor life. In general, the more open the enclosure is to the atmosphere, the lower the first cost of the machine but the higher the maintenance costs that may be necessary. Enclosures frequently used in a-c motors are listed below.

Drip-Proof

These are generally used only indoors or in enclosed spaces not exposed to severe environmental conditions. Maintenance requirements will depend upon general cleanliness of the location and any chemical contaminants in the area.

Weather-Protected Type I

This is the least costly outdoor machine. It is essentially a drip-proof guarded motor with heaters and outdoor bearing seals and is very susceptible to weather and atmospheric contamination. Considerable maintenance may be required to ensure satisfactory winding and bearing life.

Weather-Protected Type II

This is the more commonly used outdoor enclosure. It is more expensive than the WP-I but minimizes the entrance of water and dirt. Maintenance is less than for WP-I types. Chemical contaminants in gaseous form may be carried into a WP-II machine with the ventilating air and attack parts that are vulnerable to them.

Totally Enclosed Forced Ventilated (TEFV)

TEFV enclosures can be used indoors or outdoors in dirty or hazardous environments. Since the motor cooling air is piped in from a separate source the influx of dirt and gaseous contaminants is minimized. Maintenance is minimal depending upon the cleanliness of the cooling air.

Totally Enclosed Water-to-Air Cooled (TEWAC)

The totally enclosed water-to-air cooled machine uses an air to water heat exchanger to remove heat generated by motor losses. It is the quietest enclosure available and will usually result in the lowest maintenance costs. It will breathe during shutdown but often a breather filter is used to remove particulate contaminants. It is more efficient than a TEFC motor because it does not have the external fan to drive. Its first cost

FIG. 15-33

Energy Evaluation Chart
NEMA Frame Size Motors, Induction

HP	Approx. Full Load RPM	Amperes Based on 460V		Efficiency in Percentage at Full Load	
		Standard Efficiency	High Efficiency	Standard Efficiency	High Efficiency
1	1,800	1.9	1.5	72.0	84.0
	1,200	2.0	2.0	68.0	78.5
1-1/2	1,800	2.5	2.2	75.5	84.0
	1,200	2.8	2.6	72.0	84.0
2	1,800	2.9	3.0	75.5	84.0
	1,200	3.5	3.2	75.5	84.0
3	1,800	4.7	3.9	75.5	87.5
	1,200	5.1	4.8	75.5	86.5
5	1,800	7.1	6.3	78.5	89.5
	1,200	7.6	7.4	78.5	87.5
7-1/2	1,800	9.7	9.4	84.0	90.2
	1,200	10.5	9.9	81.5	89.5
10	1,800	12.7	12.4	86.5	91.0
	1,200	13.4	13.9	84.0	89.5
15	1,800	18.8	18.6	86.5	91.0
	1,200	19.7	19.0	84.0	89.5
20	1,800	24.4	25.0	86.5	91.0
	1,200	25.0	24.9	86.5	90.2
25	1,800	31.2	29.5	88.5	91.7
	1,200	29.2	29.1	88.5	91.0
30	1,800	36.2	35.9	88.5	93.0
	1,200	34.8	34.5	88.5	91.0
40	1,800	48.9	47.8	88.5	93.0
	1,200	46.0	46.2	90.2	92.4
50	1,800	59.3	57.7	90.2	93.6
	1,200	58.1	58.0	90.2	91.7
60	1,800	71.6	68.8	90.2	93.6
	1,200	68.5	69.6	90.2	93.0
75	1,800	92.5	85.3	90.2	93.6
	1,200	86.0	86.5	90.2	93.0
100	1,800	112.0	109.0	91.7	94.5
	1,200	114.0	115.0	91.7	93.6
125	1,800	139.0	136.0	91.7	94.1
	1,200	142.0	144.0	91.7	93.6
150	1,800	167.0	164.0	91.7	95.0
	1,200	168.0	174.0	91.7	94.1
200	1,800	217.0	214.0	93.0	94.1
	1,200	222.0	214.0	93.0	95.0

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is greater than WP-II but less than TEFC, excluding any additional capital cost for a cooling water system. Operating costs are higher because of the necessity to continuously supply it with cooling water.

Totally Enclosed Fan Cooled (TEFC)

This is the highest degree of enclosure for an air cooled machine. In large sizes, the TEFC motor has an air-to-air heat exchanger. Internal motor air is recirculated around the outside of the tubes while outside air is driven through the tubes by a shaft driven fan. These motors are quite expensive especially in large sizes because of the high volume of cooling air required relative to motor size. These motors are indicated for use in very dirty or hazardous locations.

The TEFC enclosure minimizes the maintenance required for these very dirty applications. However, the machines will breathe when shut down and vapor and gaseous contaminants can be drawn into them. TEFC motors are usually noisy because of the large external fan.

Explosion-Proof

An explosion-proof machine is a totally enclosed machine whose enclosure is designed and constructed to withstand an internal explosion. It is also designed to prevent the ignition of combustibles surrounding the machine by sparks, flashes, or explosions which may occur within the machine casing.

THE INDUCTION GENERATOR

The induction generator can be used as a convenient means of recovering industrial process energy that would otherwise be wasted. Excess steam or compressed gas can often drive such a generator to convert useless energy to valuable kilowatts.

An induction generator is simply an induction motor driven above its synchronous speed by a suitable prime mover. This results in production rather than consumption of electric energy. Normally the induction generator does not differ in any aspect of electrical or mechanical construction from an induction motor. Only the operating speed range separates one mode of behavior from the other.

Important differences exist between the induction generator and the more widely used synchronous generator. These are basically the same as the differences between induction and synchronous motors. Besides low cost and simplicity of control an important benefit is that the induction machine is instantly convertible from generator to motor operation or vice versa.

The synchronous generator needs precise prime-mover speed control to maintain its output at correct frequency. When connected to a public utility system such a machine cannot be allowed to deviate more than a fraction of a cycle from rated frequency without being tripped off the line. However, speed changes do not affect the voltage or power output of the generator — only the frequency.

For the induction machine, voltage and frequency remain constant, set by the connected power system, whatever the driven speed. The speed change does directly affect the power output of the generator and therefore the temperature of its windings. Unless other machines are coupled into the same drive to dampen speed swings, close control of rpm is almost as necessary to the induction generator as to the alternator.

Smaller generators (down to 300 kW) are finding many uses. Among them:

- Recovering energy of compression on the downhill side of a natural gas pipeline.
- Producing electric power from the expansion of geothermal steam.
- Generating power through expansion of compressed gases in cryogenic production.
- Recovering energy from single-stage waste steam turbines in the 5-175 psig inlet pressure range.

SPEED VARIATION

Because of the continuing increase in the cost of electric energy, variable speed drives offer an economical means of reducing energy requirements in many areas of operation.

Variable Frequency Electric Motors

For many years variable speed applications relied on either d-c motors or a constant speed a-c motor coupled to various mechanical systems to provide the range of speeds required. Solid-state electronics provide an effective means of speed control for a-c motors by changing the frequency of the electrical signal. They can be used with both induction and synchronous motors.

A standard a-c motor operating at 60 hertz will operate at a constant speed, depending upon the number of magnetic poles it has in accordance with the formula:

$$\text{rpm} = \frac{120f}{P} \qquad \textbf{Eq 15-2}$$

However, if the input frequency can be varied in accordance with the speed requirements, then a wide range of speeds can be obtained. For example, with a frequency range from 50 to 120 hertz, a 4-pole motor has a speed range from 1500 through 3600 rpm.

Fixed Speed Electric Motors With Fluid Couplings

The speed of an equipment item driven by a fixed-speed electric motor can be varied with a fluid coupling. This is essentially a pump discharging to a power-recovery turbine, both in the same casing. The pump is connected to the driver shaft and the turbine to the driven shaft. The turbine speed is varied by varying the amount of fluid in the casing. Increasing the fluid increases the circulation between the pump and turbine, thereby increasing the speed of the turbine. A fluid coupling costs less than electronic speed control but is less efficient. For example, the typical efficiency for electronic speed control is about 92–95% from minimum to normal speed, but for a fluid coupling is about 95% at normal speed, and can range from 50 to 70% at 50% speed, depending on the coupling make and type.

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Internal Combustion Engines

Internal combustion engines are classified according to the type of fuel used and the method of fuel ignition. Many subclassifications are used to describe engines according to their speed, cycle arrangements, mechanical configuration, and other design characteristics.

ENGINE TYPES

Spark ignition and compression ignition are the two methods of initiating combustion used in reciprocating internal combustion engines. In practice the ignition method also defines the fuels or range of fuels used.

Spark Ignition

Natural gas, liquefied petroleum gas (LPG), or gasoline are the fuels used in spark ignition engines. They are often referred to as gas engines or gasoline engines and resemble in appearance (except perhaps for size) and operation the engines used in automobiles. High voltage electrical energy fires one or more spark plugs per cylinder to ignite the air/fuel mixture. Most spark ignition engines can be easily modified to burn any of the above fuels. Often the fuel delivery system is the only part of the engine requiring significant changes.

Compression Ignition (Diesel)

Engines that use heat of compression as the ignition source are almost always referred to as diesel engines. A broad range of liquid fuels can be burned in a diesel engine provided proper attention is paid to the handling and preparation of the fuel as well as to the design of the engine. The type and quality of the fuel can have a significant effect on the service life of the engine. Diesel fuels are classified under an ASTM designation D-975. This specification covers limits for three grades of fuel which can be purchased commercially (Fig. 15-34). Engine manufacturers may also publish limits for the fuels they recommend be used in specific engines.

Dual-Fuel

Dual-fuel engines may operate in one of two modes. One mode is as an ordinary diesel engine. It may also operate on a gaseous fuel with a pilot injection of liquid diesel fuel for ignition. The pilot fuel provides less than 10% of the total fuel energy at full load.

Four-Stroke-Cycle

Most spark ignition engines use a four-stroke-cycle which is completed in two crankshaft revolutions and consists of the following piston strokes:

1. An intake stroke to draw the fuel/air mixture into the engine cylinder.
2. A compression stroke which raises the pressure and temperature of the mixture.
3. The expansion or power stroke from the ignition and combustion of the fuel mixture.
4. An exhaust stroke to free the cylinder of combustion products.

The four cycle diesel engine operates in a similar fashion. During the intake stroke only air is introduced into the cylinder. Compression of air alone causes a higher temperature to be reached in the cylinder. The fuel is injected into the cylinder at the very beginning of the expansion stroke and spontaneously ignites.

Two-Stroke-Cycle

The four cycle engine requires two revolutions of the crankshaft for each power stroke. To get a higher output from the same size engine, the two-stroke-cycle was developed. This cycle is applicable both to compression ignition and spark ignition engines. The two-stroke-cycle is completed in one revolution of the crankshaft and consists of two piston strokes: the compression stroke and the expansion stroke. Combustion air intake occurs at the end of the expansion and the beginning of the compression stroke. Ignition and combustion occurs at the end of the compression and beginning of the expansion stroke.

Supercharged Engines

A supercharged engine has a compressor to increase the density of the combustion air before it is inducted into the cylinder. Supercharging increases the power output from a given cylinder size by increasing the engine mean effective pressure.

Two types of supercharging are common: mechanical compressors driven by an engine auxiliary output shaft or a separate prime mover and exhaust turbine driven compressor which obtains its power from expansion of the engine exhaust. This later type of supercharger is commonly called a turbocharger.

Speed

Most process plant engines are used to drive equipment with a limited range of speed requirements, and which can be selected to operate near the point of highest efficiency. Speed increasing or decreasing gears may be used to match an engine with a particular service. Internal combustion engines are classified according to speed in the following broad categories:

High speed — above 1500 rpm
Medium speed — 700 to 1500 rpm
Low speed — below 700 rpm

High speed engines can offer weight and space advantages but will usually require more maintenance than a medium or low speed engine. High speed engines are often selected for standby or intermittent applications. As a general rule the lower the speed the longer the service life. Although internal combustion engines are usually selected to run over a limited speed range, they will operate well over large ranges of speed just as an automobile engine does.

PERFORMANCE RATING

Several measurements of performance can be used to compare engines. Four commonly used measurements are:

1. Specific fuel consumption, (lb or Btu)/(bhp • hr)
2. BMEP, psi
3. Specific weight, lb/bhp
4. Output per unit of displacement, bhp/cu in.

The relationship between brake mean effective pressure (BMEP) and brake horsepower (bhp) is given below.

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$$BMEP = \frac{(bhp)\ (33{,}000)}{(S)\ (A)\ (N)} \qquad \textbf{Eq 15-3}$$

The value of N is equivalent to RPM for two-stroke-cycle engines, and RPM divided by two for four-stroke cycle. BMEP indicates how much turbocharging increases the brake horsepower which is the power delivered to the driven equipment by the engine output shaft.

The intended use of the engine will determine the most important measure of performance. For an aircraft engine the first and third items may be the most important; while for a stationary engine in continuous service with no space or weight limitations, the first item would be of primary importance.

The power delivered is directly related to atmospheric conditions. Operation in areas of low atmospheric pressure (high altitudes) will reduce the power output. High inlet air temperature will also reduce the power output. Engines are rated for various altitudes above sea level (i.e. barometric pressures) and ambient temperatures (e.g. 1500/3000 feet and 90°F according to DEMA; 1500 feet and 85°F; and so forth). A rule of thumb for derating naturally aspirated engines is 3-1/2% reduction in power for each 1000 ft above the rating altitude, and 1% reduction for every 10°F above the rating temperature. For exact deration of naturally aspirated engines, or for turbocharged engines, the manufacturers must be consulted.

Following are gas-engine design parameters. The values vary considerably depending on the engine type, make and model, and on the site conditions, but ranges of typical values are given.

- Fuel-gas requirements (i.e. heat rate) [6500 to 8500 Btu/(bhp • hr), LHV].
- Heat rejection at the power-end exhaust manifold [1500 to 3000 Btu/(bhp • hr) with jacket water cooling, or 800 to 1500 without].
- Heat rejection at a turbo aftercooler if applicable [100 to 500 Btu/(bhp • hr)].
- Heat rejection at the lube-oil cooler [300 to 900 Btu/(bhp • hr)].

New technologies have reduced specific weights (i.e. lb/bhp), increased fuel efficiencies, lengthened the periods between overhauls, and reduced emissions. Precisely programmed electronically controlled fuel injection incorporates ambient and other important operating conditions to minimize fuel consumption and emissions over full operating ranges. Many engine designs include pre-combustion chambers that jet flames into the main combustion chambers effectively igniting leaner air/fuel mixtures (i.e. "lean burn") resulting in higher efficiencies and lower emissions. New thermal-barrier coatings (TBCs) insulate many engine components from thermal shock and reduce heat losses that would otherwise decrease thermal efficiencies.

FIG. 15-34

Grades of Diesel Fuel, ASTM D-975 (1995) Classification

		1-D	2-D	4-D
Flash point, °F	Min	100	125	130
Carbon Residue, %	Max	0.15	0.35	–
Water and Sediment, % by Vol	Max	0.05	0.05	0.50
Ash, % by Wt	Max	0.01	0.01	0.10
Distillation °F 90% Pt	Max	550	640	–
	Min	–	540	–
Viscosity at 104°F Centistokes	Min	1.3	1.9	5.5
	Max	2.4	4.1	24.0
Sulfur, % by Wt		0.05	0.05	2.0
Cetane No.	Min	40	40	30
Aromaticity, % by Vol	Max	35	35	–

Engine Energy Balance

A gas engine converts the combustion energy in the fuel to mechanical power and heat. The combustion energy is usually distributed as follows:

	% Range
Mechanical power	30–40
Heat rejected to cylinder cooling	25–40
Heat rejected to oil cooler	3–5
Heat rejected to turbo aftercooler	4–9
Heat rejected to exhaust	25–30
Heat rejected to atmosphere (i.e. surface heat loss)	3–6

The mechanical power is the sum of the brake horsepower (bhp) (i.e. available shaft power), and the power to drive such engine auxiliaries as a lube-oil pump, cooling-water pump, radiator fan, and alternator (for a spark ignition engine).

Fig. 15-35 includes engine power ratings, specific fuel requirements (i.e., "heat rates"), heat rejections and exhaust conditions for a variety of gas engines. The values are based on full design operating power at the speeds noted for various altitudes above sea level and ambient temperatures.

An engine's power efficiency, typically called "thermal efficiency," is calculated from the following equation:

$$\%\ \text{Thermal Efficiency} = \frac{100 \times 2544}{\text{Heat Rate (Btu/(bhp • hr), LHV)}}$$

The total heat rejected is calculated from the following equation:

$$\text{Heat Rejected} = (\text{Heat Rate} - 2544)\ \text{Btu/(bhp • hr)}$$

The heat rejected to the engine exhaust gas is calculated from the following equation:

Exhaust Heat Btu/(bhp • hr) = Total heat rejected minus the sum of the heat rejected to cylinder cooling, oil cooling, turbo aftercooling, and engine-surface heat loss to the atmosphere

It is technically feasible to recover part of the heat. Low temperature heat at about 180°F, for such as space heating, can be recovered from the cooling circuits for cylinder jackets, lube oil and turbo charged air. Higher level heat at above 300°F can be recovered by heat exchange with engine exhaust. Below 300°F water vapor will condense with CO_2 absorption, acid formation, and resulting corrosion. A heat recovery arrangement is illustrated in Fig. 15-36. Technical feasibility depends upon the economic criteria and improves as the engine size increases. Heat recovery can increase the overall thermal efficiency to as high as 75%. For example, an engine's thermal efficiency can be increased from a typical regular value of 33% to 75% by recovering about 60% of the heat normally rejected to the coolant and exhaust.

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FIG. 15-35

Engine Ratings and Operating Parameters

Note: Figures may be approximate due to variations in engine services and are representative of new engines only. Refer to manufacturer for exact information.

							Heat Rejection			Btu / (bhp · hr)			
ENGINE	Full Power at Full Speed (bhp)	Full Speed (rpm)	Strokes Per Cycle	BMEP (psi)	Fuel Reqmt (Btu/(bhp·hr)) LHV	Jacket Water Cooler	Cylinder Cooling	Oil Cooler	Turbo Intercooler/ Aftercooler	Atmosphere i.e. Surface Heat Loss	Exhaust rate (lb/(bhp·hr)	Exhaust temp (F)
Caterpillar												
G3406TA	325	1800	4	Not Avail.	7103	2803	Note (1)	Note (1)	157	284	6.60	964
G3408TA	400	1800	4	Not Avail.	7485	3250	Note (1)	Note (1)	68	283	6.55	925
G3412TALE	637	1800	4	Not Avail.	7541	2705	Note (1)	Note (1)	354	302	9.34	838
G3508TALE	633	1400	4	Not Avail.	7619	2275	Note (1)	Note (1)	410	302	9.86	847
G3512TALE	945	1400	4	Not Avail.	7570	2159	Note (1)	Note (1)	412	270	10.02	860
G3516TALE	1265	1400	4	Not Avail.	7548	2217	Note (1)	Note (1)	391	252	9.71	869
G3606TALE	1775	1000	4	Not Avail.	6781	606	305	449	237	11.73	867	
G3608TALE	2370	1000	4	Not Avail.	6761	606	304	445	237	11.69	865	
G3612TALE	3550	1000	4	Not Avail.	6761	616	304	449	237	11.71	858	
G3616TALE	4735	1000	4	Not Avail.	6736	608	303	433	236	11.65	864	
Cummins												
G/GTA5.9	152	2200	4	Not Avail.	7382	2473	Not Avail.	319	279	Not Avail.	Not Avail.	
G/GTA8.3	200	1800	4	Not Avail.	7964	2394	Not Avail.	256	299	Not Avail.	Not Avail.	
G/GTA855	304	1800	4	Not Avail.	7910	2506	Not Avail.	317	283	Not Avail.	Not Avail.	
KTA19GC	420	1800	4	Not Avail.	7376	2011	Not Avail.	340	Not Avail.	Not Avail.	Not Avail.	
Dresser-Rand												
412KVSR	2600	330	4	139	7100	1850	325	800	Not Avail.	11.90	750	
410KVR	3750	350	4	170	6450	850	350	550	Not Avail.	11.30	750	
510TLAD	4000	300	2	122.4	6850	Not Avail.	Not Avail.	Not Avail.	Not Avail.	Not Avail.	Not Avail.	
512KVR	4500	350	4	170	6450	850	350	550	Not Avail.	11.30	760	
612TCV	5600	330	2	122.9-130.8	6700	Not Avail.	Not Avail.	Not Avail.	Not Avail.	Not Avail.	550	
612TCVC	7150	330-345	2	133.4-133.9	6700	Not Avail.	Not Avail.	Not Avail.	Not Avail.	Not Avail.	550	

Note: (1) The heat rejected to the oil cooler is included with that to the jacket-water cooler.

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FIG. 15-35 (Cont'd.)

Engine Ratings and Operating Parameters

Note: Figures may be approximate due to variations in engine services and are representative of new engines only. Refer to manufacturer for exact information.

							Heat Rejection		Btu / (bhp · hr)			
ENGINE	Full Power at Full Speed (bhp)	Full Speed (rpm)	Strokes Per Cycle	BMEP (psi)	Fuel Reqmt (Btu/(bhp·hr)) LHV	Jacket Water Cooler	Oil Cooler	Cylinder Cooling	Turbo Intercooler/ Aftercooler	Atmosphere i.e. Surface Heat Loss	Exhaust rate (lb/(bhp·hr)	Exhaust temp (F)
Waukesha												
F18GL	400	1800	4	160	7524	1980	325		485	211	9.36	874
H24GL	530	1800	4	160	7800	2147	323		547	188	10.26	851
L36GL	800	1800	4	160	7129	1839	238		421	144	9.14	838
P48GL	1065	1800	4	160	7106	1892	233		422	134	9.12	836
F2895GL	607	1200	4	138	7282	1960	329		395	264	10.81	700
F3521GL	738	1200	4	138	7377	2003	306		373	253	10.90	703
F3524GSI	840	1200	4	158	7825	2435	337		149	410	6.64	1135
L5108GL	1072	1200	4	138	7295	1926	354		368	275	10.78	680
L5790GL	1215	1200	4	138	7284	1967	329		398	239	10.03	679
L5794GSI	1380	1200	4	158	7650	2275	355		110	473	6.41	1135
L7042GL	1478	1200	4	138	7274	1905	291		367	222	10.75	709
L7044GSI	1680	1200	4	158	7876	2318	337		107	432	6.73	1181
P9390GL	1963	1200	4	138	7221	1750	316		387	182	10.19	763
8LAT27GL	2090	1000	4	190	6556	1187	280		510	125	11.98	677
12VAT27GL	3130	1000	4	190	6568	1238	239		569	90	11.04	703
16VAT27GL	4500	1000	4	205	6583	719	290		539	204	12.21	776
Wartsila												
12V34SG-CD (1)	5,552	750	4	252	5,780	725 (2)	295		233 (3)	94	9.82	779

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Example 15-4 —Calculate the thermal efficiency, total heat rejected, and total exhaust heat for a Waukesha L7042GL at 1200 RPM, 77°F and sea level, and its full power rating.

Solutions Steps

From Fig. 15-35

Full Power	=	1478 bhp
Heat rate	=	7274 Btu/(bhp • hr), LHV
Heat rejected to water cooling, oil cooling, turbo intercooling, and radiation	=	1905 + 291 + 367 + 222 = 2785 Btu/(bhp • hr)

Therefore:

Thermal efficiency	=	$\frac{100 \times 2544}{7274}$ = 35.0%
Heat rejected per bhp	=	7274 – 2544 = 4730[Btu/(bhp • hr)
Total heat rejected	=	4730 [Btu/(bhp • hr)] 1478 bhp = 7.0 MMBtu/hr
Exhaust heat per bhp	=	4730 – 2785 = 1945 Btu/(bhp • hr)
Total exhaust heat	=	1945 [Btu/(bhp•hr)] 1478 bhp = 2.87 MMBtu/hr

Auxiliaries

BEARINGS

Hydrodynamic journal bearings are found in all types of industrial turbomachinery, which include pumps, electric motors, steam turbines, electric generators, and gas compressors. The hydrodynamic bearing types most commonly found in turbomachinery are:

- Plane cylindrical
- Pressure dam
- Tilting pad

For all bearing types, the fundamental geometric parameters are journal diameter, pad arc angle, length-to-diameter ratio, and running clearance. Some bearing types, such as tilting pad bearings, have additional geometric variations including number of pads, preload, pad pivot offset angle, and orientation of the bearing (on or between pads). The key operating conditions are oil viscosity, oil density, rotating speed, gravity load at the bearing, and applied external loads (such as gear mesh or pump volute loadings). A machinery expert should be consulted for further details concerning types of bearings and their applications and designs.

GEARS

There are many different types of open gears such as spur, helical, spiral bevel, and worm. This section will focus on enclosed high speed helical gear reducers or increasers commonly used in the natural gas, refinery, and petrochemical industries.

Speed Increasers and Reducers

Speed increasers are usually used on centrifugal compressors, axial compressors, blowers, and centrifugal pumps driven by motors, turbines, and industrial combustion engines. Speed reducers are used on reciprocating compressors, rotary positive displacement compressors, centrifugal pumps, generators, and fans driven by turbines and motors.

High Speed Gears

High-speed gears are generally defined as having either or both of the following:

1. Pinion speed of at least 2,900 rpm.
2. Pitch line velocities above 5,000 ft/min.

There are units operating with pitch line velocities in excess of 35,000 ft/min and transmitting 30,000 hp.

Gearing

High speed gears can be selected with either single helical gearing (used extensively in Europe) or opposed double helical (i.e., “herringbone”) gearing (predominant in the United States). Pros and cons of each type of gear design are numerous with double helical gearing being more efficient because there is only one thrust bearing required. The thrust bearing is usually on the low speed shaft.

Surface Finish

High speed gears are classified as precision quality gears. Fig. 15-37 shows a minimum surface finish and quality required for various pitch line velocities as recommended in Figure 1, page 14 of AGMA 2001-C95, Fundamental Rating Factors and Calculation Methods for Involute Spur and Helical Gear Teeth.

GEAR RATINGS

Various parameters affecting the durability, strength ratings, and scoring temperatures include:

Horsepower — The horsepower rating of high speed gears is determined from the durability rating and strength rating on the gear or pinion as specified in AGMA 6011-G92, Specifications for High Speed Helical Gear Units. In addition, the rating is limited by the scuffing temperature as determined in accordance with AGMA 217.01, Information Sheet, Gear Scoring Design Guide for Aerospace Spur and Helical Power Gears, and Annex A of AGMA 2001-C95.

Durability — The durability hp rating for a specific gear set is primarily dependent on the speed and allowable contact stress of the gear and does not vary significantly with tooth size. The allowable contact stress is dependent on the surface hardness of the gear or pinion tooth and varies with material composition and mechanical properties.

Strength — The strength hp rating for a specific gear set varies mainly with the speed, allowable fatigue stress of material, and with tooth thickness. The tooth form, pressure angle, filet radius, number of teeth, helix angle, and pitch line velocity also affect the strength horsepower rating.

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Allowable fatigue stress is dependent on the tensile strength of the material and varies with heat treatment and chemical composition.

Scuffing Temperatures — The scuffing or flash temperature index is the calculated temperature of the oil in the gear mesh. This temperature is arrived at by calculating the temperature rise of the lubricant in the mesh and adding it to the inlet oil temperature. The temperature rise for a given set of gears increases with the tooth loading, speed, and surface finish (i.e. increasing roughness).

The temperature of the gears will increase as the pitch is decreased, pressure angle is decreased, or helix angle is increased.

Design Factors

The following design factors must be considered for high speed drives.

Housings — must be of rugged design for strength and rigidity to maintain precise alignment of gears and bearings.

Bearings — should be split-sleeve, babbitt lined, steel-backed precision journal bearings with thrust faces for axial loads. Fixed pad or tilting pad (Kingsbury type) should be used where required. Tilt pad radial bearings may also be required for high rpm, high load applications.

Shafts — precision machined from heat treated, high quality alloy (4140 is common) steel. Adequately sized to rigidly maintain gear alignment and protect from overload.

Pinions — normally cut integral with shaft from a high quality forging that is through hardened or surface hardened by carburizing or nitriding. Grinding is the most common finishing method but precision hobbing, shaving, or lapping are also used.

Gears — usually made from a high quality forging that is through hardened or surface hardened by carburizing or nitriding and is separate from the low speed shaft. Gear may be integral with the shaft when operating conditions require. Grinding is the most common finish method but precision hobbing, shaving, or lapping are also used.

Dynamic Balance — balance all rotating elements to assure smooth operation at high rpm.

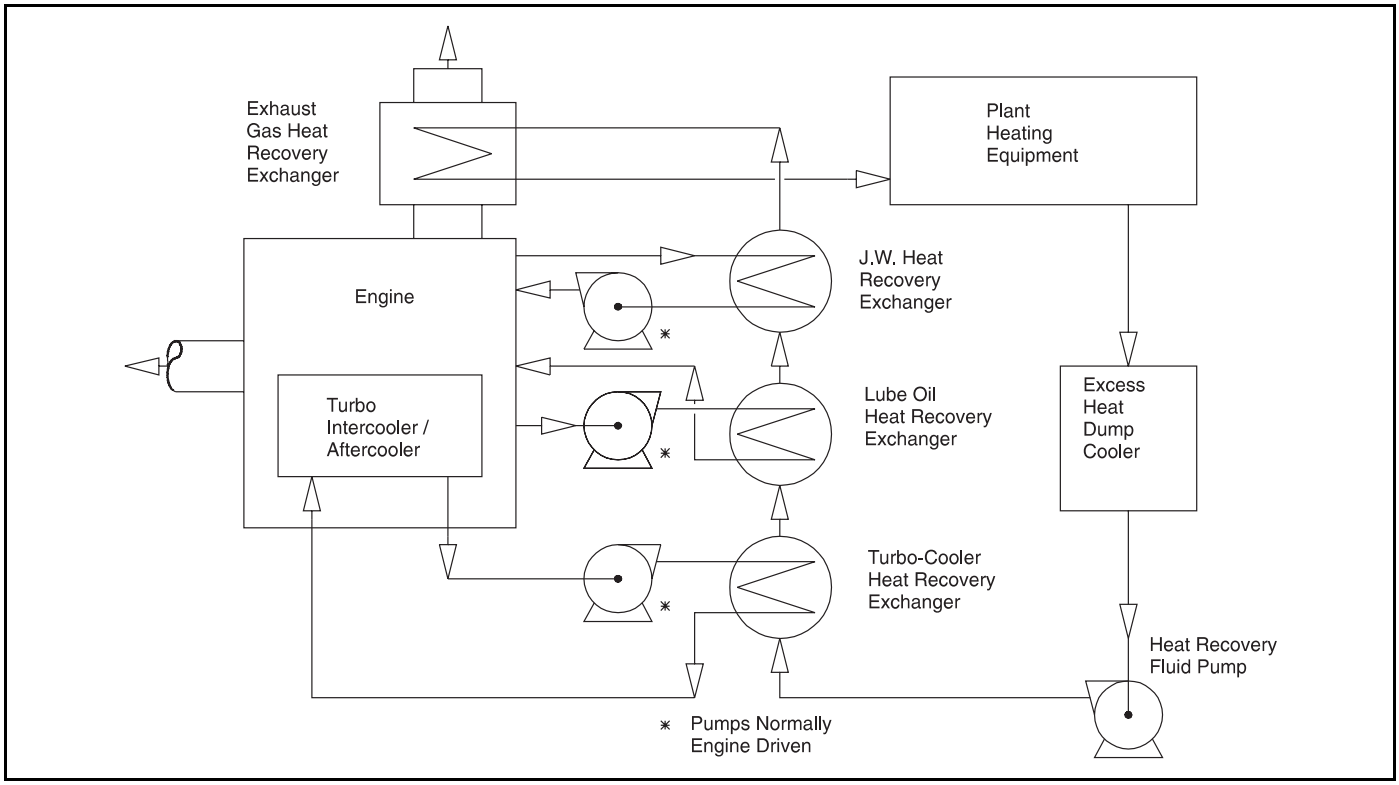
Seals — shaft seal should be of the labyrinth type, with clearance between shaft and seal of 0.020 to 0.030 inch. To prevent oil leakage through the clearance, the labyrinth is made interlocking with grooves machined in the cap to create air back pressure during rotation to retain the lubricant.

High speed gears are usually used on critical process trains where down time is quite costly and catastrophic failure must be avoided at all costs. Therefore, gear drives are becoming more and more instrumented. Optional monitoring equipment often specified by users include:

- Vibration probes and proximitors (to measure shaft vibration)
- Keyphasors (provide timing and phase reference)
- Accelerometers (measure casing acceleration)
- Direct reading dial type thermometers in stainless thermowells (measure bearing temperature)
- Resistance temperature detectors (RTDs) and thermocouples (measure bearing temperature)

FIG. 15-36

Example Engine Heat Recovery Arrangement



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- Temperature and pressure switches (alarm and shutdown functions)

Lubrication

The majority of high horsepower, high speed gears are lubricated from a common sump which also lubricates the driving and the driven equipment. These systems are normally designed to operate with a high grade turbine oil with a minimum viscosity of 150 SSU at 100°F. A good operating pressure range for the oil is 25 to 50 psi, with 25 micron filtration.

COUPLINGS

A coupling is required to connect a prime mover to a piece of driven machinery. The purpose of a coupling is to transmit rotary motion and torque from one piece of machinery to another. A coupling may also serve a secondary purpose such as accommodating misalignment of the two pieces of equipment. There are two general categories of couplings: rigid and flexible.

Rigid Couplings

Rigid couplings are used when the two machines must be kept in exact alignment or when the rotor of one machine is used to support the rotor of another machine. Very precise alignment of machine bearings is necessary when using this type of coupling. Manufacturing tolerances are also extremely important. One common application for rigid couplings is in the pump industry where the prime mover, generally an electric motor, is positioned vertically above the pump.

Flexible Couplings

Flexible couplings, in addition to transmitting torque, accommodate unavoidable misalignment between shafts. Mechanically flexible couplings provide for misalignment by clearances in the design of the coupling. The most common type of mechanically flexible coupling is the gear-type. Material flexible couplings use the natural flexing of the coupling element to compensate for shaft misalignment. Metal, elastomer, or plastic having sufficient resistance to fatigue failure may be used for the flexing element of the coupling. Many types of flexible couplings are in common use and selection for a particular application depends on many factors including cost, horsepower, shaft speed, and reliability. A specialist should always be consulted for proper selection on any critical piece of equipment.

FIG. 15-37

Gear Quality

Pitch Line Velocity (ft/min)	Surface Quality (RMS) (micro inches)	Minimum Gear Quality Number
Under 8,000	45	10
8,001 - 10,000	32	11
10,001 - 20,000	32	12
20,001 - 30,000	20	12-13
Over 30,000	16	12-14

VIBRATION MONITORING

The oldest and most basic type of vibration measurement involved the use of the human senses to feel and listen to a machine. The basic approach has not changed, just the method. It was always difficult to justify enough time for one individual, or a group of individuals, to acquire periodic measurements on a large number of machines. Also, with the advent of high speed, high performance machines, failures can occur faster than personnel can react. In addition, very subtle changes can occur over a long period of time, making it difficult to realize by the human senses, but still affecting the machine's mechanical stability and safety. Vibration monitoring is simply the full-time electronic measurement and monitoring of vibration levels from a given machine. Typically, the monitoring responds to the overall signal input from the transducer regardless of the source of vibration (in-balance, bearing wear, coupling problems, misalignment, etc.).

A typical vibration monitor provides two levels of alarm: alert and danger, that can be adjusted to fit the characteristics of a given machine. These set points have associated relays which can be connected to external audible or visual annunciators on the control panel. If the alert or danger set point is exceeded, the monitor and annunciator will alert operations and maintenance personnel of this event. Ideally, the alert alarm will indicate that the machine condition has changed significantly, but allow some discrete time before the machine is in a dangerous condition. For most applications, if the machine does reach the danger level of vibration and continued operation would probably result in machine failure, automatic shutdown is mandatory regardless of the time lag that has occurred between alert and danger signal.

There are three types of vibration sensors: (1) accelerometers, (2) velocity transducers, and (3) proximity probes. For most large critical machinery, and certainly for machinery with fluid film-type bearings, the important measurement to be made is rotor motion relative to the machine bearing or bearing support. For this application, the proximity probe transducer has proven to be the most reliable indicator of machinery malfunctions.

The proximity probe is a noncontacting transducer, typically installed on the bearing or bearing housing, and observes the rotor radial dynamic motion and position with respect to the bearing clearance. This same type of proximity probe can be used to measure axial position and vibration as well.

For machines which exhibit significant amounts of casing motion, it may be necessary to add to this system a seismic transducer measuring machine casing vibration. Some unique applications dictate that measurements are necessary in the high frequency region, where accelerometers are typically employed.

The American Petroleum Institute (API) has published a specification describing vibration monitoring systems, API 670, "Vibration, Axial Position, and Bearing Temperature Monitoring Systems."

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