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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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Acknowledgments

Twelfth Edition

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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

FIG. 1-1

Typical Components of Industry Streams

	Components											
	CO ₂	H ₂ S	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 techni-

cal 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:

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

γ = relative density

associated gas

Gaseous hydrocarbons occurring 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.

barrel

A common English-unit measure 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

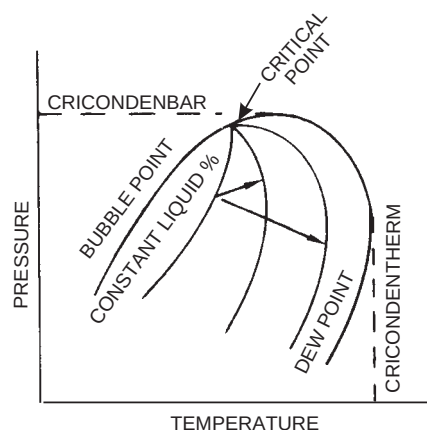
The liquid formed by the condensation of a vapor or gas; specifically, the hydrocarbon liquid separated from natural gas because of changes in temperature and pressure when the gas from the reservoir was delivered to the surface separators. In a steam system it may be water that is condensed and returned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



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-condensable 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.

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.

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.

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.

stripping

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 (SO₂)

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 CO₂ 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.

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

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

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

A.P.I. gravity	Baumé gravity	Specific gravity	Lb/ U.S. gal	U.S. gal/ lb
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:

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

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

For liquids heavier than water:

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

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 d₁ density

n = Volume proportion of oil of d₂ density

d₁ = Specific Gravity of m oil

d₂ = Specific Gravity of n oil

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

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)	$= [\text{SSU (at } 100^\circ\text{F)} (2.3210 \times 10^{-6})]$
centistokes (50–100 SSU)	$= [(\text{SSU}) (0.226)] - (205.3/\text{SSU})$
centistokes (100–350 SSU)	$= [(\text{SSU}) (0.220)] - (147.7/\text{SSU})$
centistokes (over 350 SSU)	$= [\text{SSU (at } 100^\circ\text{F or } 37.8^\circ\text{C)} (0.21576)]$
centistokes (over 350 SSU)	$= [\text{SSU (at } 210^\circ\text{F or } 98.9^\circ\text{C)} (0.21426)]$
centistokes (over 500 SSU)	$= [\text{SSU (at } 122^\circ\text{F or } 50^\circ\text{C)} (2.120)]$
centistokes (over 300 Redwood #1)	$= [\text{Redwood \#1 (Standard)} (0.255)]$
centistokes (over 50 Redwood #2)	$= [\text{Redwood \#2 (Admiralty)} (2.3392)]$
centistokes (over 18 Engler)	$= (\text{Engler}) (7.389)$
centistokes (over 20 Stormer)	$= (\text{Stormer}) (2.802)$
centistokes (over 1.0 Demler #10)	$= (\text{Demler \#10}) (31.506)$
centistokes (over 1.3 Demler #1)	$= (\text{Demler \#1}) (3.151)$
centistokes (over 14 Parlin #20)	$= (\text{Parlin Cup \#20}) (61.652)$
centistokes (over 230 Ford #4)	$= (\text{Ford Cup \#4}) (3.753)$
centistokes	$= 6200 \text{ Barbey}$

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) = lbf-sec/sq ft.

ABSOLUTE TO KINEMATIC VISCOSITY

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

KINEMATIC TO ABSOLUTE VISCOSITY

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

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:

gram-pound relationship

 Fahrenheit-Celsius scale relationship

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

1 therm = 100,000 Btu = 105.5056 x 10⁶ J
 = 105,505.6 kJ = 1.055 056 x 10⁵ kJ.
 (Btu denotes British Thermal Units)
 (ref: Physical Properties of Natural Gases,
 Gasunie, 1988 p. 23)

1 decatherm = 1,000,000 Btu = 1 MMBtu = 10⁶ Btu

1 thermie = 1 Mcal (15°C) = 4.1858 x 10⁶ J

1 terajoule = 1 TJ = 1 x 10¹² 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 m³/kmol at 15°C (288.15 K), 101.3250 kPa.
 GPA SI standard conditions.

379.49 std ft³/lb mol at 60°F (519.67°R), 14.696 psia.

Other standard conditions sometimes used are:

<u>P</u>	<u>T</u>
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 Nm³/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 m³ (15°C, 101.3250 kPa)

140.1 Nm³ (0°C, 101.3250 kPa)

5,229 std ft³ (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 Nm³ = 37.33 scf. Same mass on either side of this equation.

1 std m³ = 35.38 scf. Again, same mass on either side of this equation, but 1 std m³ (15°C) contains slightly smaller mass than in 1 Nm³ (0°C). Gas is less dense at 15°C than at 0°C.

Other Useful Relationships

Physical conversion of ft³ and m³ 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/m³

1 Grain/U.S. gallon = 17.118 06 g/m³

1 Hectare = 10 000 m² = 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/s² = 1 N

1 pascal pressure = 1 N/m² = 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)

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.

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 ₂ O	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 ₂ O	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	μ M=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

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 time-saving 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-

trogen and Carbon Dioxide by Temperature Programmed Gas Chromatography.

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

GPA Standard 2188 — 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.*

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 681.
- 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-C₇ 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. Chappellear 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 Elliott, Y. N. Lin, T. C. Chu, Patsy S. Chappellear 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. Chappellear 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 CH₄-CO₂ System at Low Temperatures* - S. C. Mraw, S. C. Hwang and Riki Kobayashi, Rice University, Houston, Texas. Project 739.

- 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: CH₂-m-Xylene, CO₂-m-Xylene, CH₄-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. Jensen, 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-41** *Vapor-Liquid Equilibrium Data on Water-Substitute Gas Components: N₂-H₂O, H₂-H₂O, CO-H₂O and H₂-CO-H₂O and H₂S-H₂O* - 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-H₂S, Mesitylene-CO₂* - 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 H₂-CH₄ 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 CH₄-CO₂-H₂S 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-*

*n*Pentane - 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-51 *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 N₂-Toluene, N₂-m-Xylene, and N₂-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-61 *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. Cediell, 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 Equilibrium 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 CO₂ EOR Projects* - Project supported jointly by the Gas Processors Association and special industry contributions. Jane Huey Hong and Riki Ko-

- 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 815-81.
- 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.

- RR-94** *Phase Equilibrium Studies for Processing of Gas from CO₂ EOR Projects (Phase III): A. The Effect of Toluene as an Extractive Agent; B. The Effect of Methane on V-L-E Behavior in CO₂-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 CO₂* - Y. Adachi, Patrick Malone, Toshikuni Yonemoto and Riki Kobayashi, Rice University, Houston, Texas. Project 856-85.
- RR-99** *The Water Content of CO₂-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 CO₂ in Aqueous Methyl-diethanolamine 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 Methyl-diethanolamine* - R. N. Maddox, A. H. Bhairi, James R. Diers and P. A. Thomas, Oklahoma State University, Stillwater, Oklahoma. Project 841.
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- RR-106** *The Influence of High Concentrations of Methanol on Hydrate Formation and the Distribution of Glycol in Liquid-Liquid Mixtures* - H-J Ng, C. J. Chen and D. B. Robinson, D. B. Robinson & Assoc., Ltd., Edmonton, Alberta. Project 825-86.
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OTHER SOURCES OF INFORMATION

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IP	Institute of Petroleum 61 New Cavendish Street London W1M 8AR, U.K. Phone: 44 0207 467 7100 Fax: 44 0207 255 1472 Email: ip@petroleum.co.uk	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
ISA	Instrument Society of America 67 Alexander Drive Box 12277 Research Triangle Park, NC 27709 (919) 549-8411 www.isa.org	NPGA	National Propane Gas Association 1150 17th St. NW, Ste. 310 Washington, DC 20036 (202) 466-7200 www.npga.org
ISO	International Organization for Standardization Case postale 56 CH-211 Geneva, Switzerland Phone: 4122 749 0111 Fax: 4122 733 3430	SPE	Society of Petroleum Engineers Box 833836 Richardson, TX 75083 (972) 952-9393 www.spe.org
NACE	National Association of Corrosion Engineers P.O. Box 201009 Houston, TX 77216-1009 (281) 492-0535 www.nace.org		Tech Street 1327 Jones Drive Ann Arbor, MI 48105 (800) 699-9277 www.techstreet.com
		TEMA	Tubular Exchanger Manufacturers Association 25 North Broadway Tarrytown, NY 10591 (914) 332-0040 www.tema.org

NOTES:

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."					

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%	85%
	Grade 34-25	Grade 34-40	Grade 34-55	Grade 34-70	
	Grade 30-25	Grade 30-40	Grade 30-55	Grade 30-70	
	Grade 26-25	Grade 26-40	Grade 26-55	Grade 26-70	
	Grade 22-25	Grade 22-40	Grade 22-55	Grade 22-70	
	Grade 18-25	Grade 18-40	Grade 18-55	Grade 18-70	
	Grade 14-25	Grade 14-40	Grade 14-55	Grade 14-70	

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.1Strip	50ppmw	No.1Strip
Carbon Dioxide				100ppmw	3,500ppmw	500ppmw
Sulfur	5ppmw	200ppmw	200ppmw			
Oxygen	300ppmw	—	—			
Moisture	75ppmw	Nofree	Nofree			
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.

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 Hachmuth⁴, 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:

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.

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 (H₂S). 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 H₂S 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.

FIG. 2-6

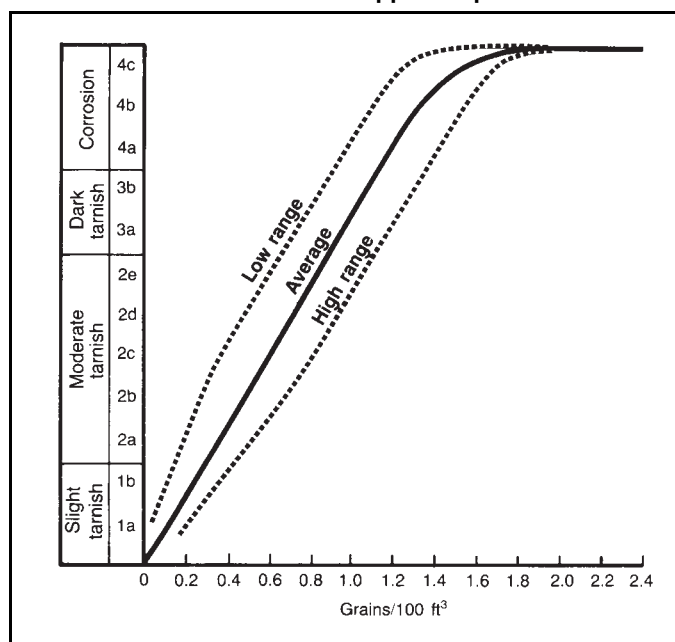
**Maximum Water Content of Dry Commercial
Liquid Propane***

Temp., °F	H ₂ O, ppmw	% Saturated
100	21	8
80	18	11
60	15	16
40	11	24
32	10	30
* These numbers are approximate		

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 al⁶ 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

FIG. 2-7

Concentration H₂S 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. Perry¹⁰ states that a No.1 copper strip is generally thought to represent hydrogen sulfide concentrations below 1-2 ppmw. As noted above, other research⁶ indicates a synergistic corrosion effect of extremely small amounts of H₂S if free sulfur is present in the stream.

From other observations, Pyburn et al⁶ report the following additional conclusions about corrosive elements in LP-gas that may affect the copper corrosion test:

1. H₂S and elemental sulfur are the corrosive agents.
2. H₂S 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 H₂S to grains per 100 ft³, measured at 60°F and 14.7 psia, use the following formula for any hydrocarbon:

$$\text{grains/100 ft}^3 = (\text{ppmw}_{\text{H}_2\text{S}})(\text{MW}_{\text{gas}})(0.001845) \quad \text{Eq 2-1}$$

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 58¹² 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 downstream 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

- 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

1. GPA Standard 2140-92: "Liquefied Petroleum Gas Specifications and Test Methods," Gas Processors Association.
2. GPA Standard 3132-84: "Natural Gasoline Specifications and Test Methods," Gas Processors Association.
3. Title 49, Code of Federal Regulations, Parts 173.301(f); 173.314(d).
4. Hachmuth, K. H., "Dehydrating Commercial Propane," Butane-Propane News, January, 1932.
5. Shock, D.A., "Research on Evaluation of Copper Strip for Corrosive Sulfur in LPG," Gas Processors Association, 1949.
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7. Poettman, F.H., and Dean, M.R., "Water Content of Propane," Petroleum Refiner, December, 1946.
8. Kobayashi, R. and Katz, D.L., "Vapor-Liquid Equilibria for Binary Hydrocarbon-Water Systems," Industrial and Engineering Chemistry, February, 1953.
9. Baumer, J.A., "Large Scale Liquid-Liquid DEA Treating for Fractionation Plants," Oil & Gas Journal, March 15, 1982.
10. Perry, C.R., "Several Treating Options Open for Ethane Recovery Plants," Oil & Gas Journal, May 23, 1977.
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.

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 cus-

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

FIG. 3-1 (Cont'd)

P_b = base pressure, psia	upstream transducer.
P_{f1}, P_{f2} = static pressure at either the upstream(1) or downstream(2) pressure tap, psia	T_b = base temperature, °F
P_p = operating or observed pressure, psig.	T_f = flowing temperature, °F
P_s = pressure at which the base volume of a meter prover was determined, usually 0 psig.	T_{meas} = reference temperature of the orifice plate bore, °F
P_1/P_2 = pressure ratio	UL = Minimum upstream meter tube length, in. (See Fig. 3-9)
Q = gas flow rate, cu ft/day	UL1 = See Fig. 3-9
Q_h = rate of flow, std. cu ft/hr or gal./hr	UL2 = See Fig. 3-9
R_h = maximum differential range, in. of water	UM = ultrasonic meter for measuring gas flow rates
R_p = maximum pressure range of pressure spring, psi	V = velocity of flowing gas.
S = square of supercompressibility	VOS = velocity of sound
SEP = Separation distance between elbows, in. (See Fig. 3-9)	X = axial distance separating transducers
SPU = Signal Processing Unit, the electronic microprocessor system of the multi-path ultrasonic meter.	Y = expansion factor to compensate for the change in density as the fluid passes through an orifice
t1 = transit time from upstream transducer to downstream transducer.	Y_{CR} = critical flow constant
t2 = transit time from downstream transducer to	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:

American Petroleum Institute
Publications and Distribution Section
1220 L Street Northwest
Washington, D.C. 20005
www.api.org

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

CHAPTER	API TITLE
1	Vocabulary
2	Tank Calibration
3	Tank Gauging
4	Proving Systems
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	Evaporation 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:

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.

FIG. 3-2

Flow Calculation Guide

		Gas				Liquid				Steam	
		Turbine or Displacement		Orifice		Orifice		Turbine or Displacement		Orifice	
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	√ <i>h_w</i>	—	—	•	•	•	•	—	—	•	—
Square Root of Static Pressure	√ <i>P_f</i>	—	—	—	•	—	—	—	—	—	—
Orifice Factor (KEY)	<i>F_n</i> • (<i>F_c</i> + <i>F_{sl}</i>)	—	—	•	•	•	•	—	—	•	3-13
Pressure Base Factor	<i>F_{pb}</i>	—	•	—	•	—	—	—	—	—	3-4
Flowing Temperature Factor	<i>F_{tf}</i>	—	—	—	•	—	—	—	—	—	3-4
Temperature Factor	<i>F_{tm}</i>	—	•	—	—	—	—	—	—	—	3-1
Temp Correction Factor (Liquids)	<i>C_{tl}</i>	—	—	—	—	—	—	—	•	—	3-1
Pressure Correction Factor (Liquids)	<i>C_{pl}</i>	—	—	—	—	—	—	—	•	—	3-4
Pressure Factor	<i>F_{pm}</i>	—	•	—	—	—	—	—	—	—	3-1
Supercompressibility Factor	<i>F_{pv}</i>	—	—	—	•	—	—	—	—	—	23-4
Square of Supercompressibility Factor	<i>S</i>	—	•	—	—	—	—	—	—	—	3-1
Density	ρ	•	—	—	—	—	—	•	—	—	3-1
Square Root of Density	√ρ	—	—	•	—	•	—	—	—	•	—
Specific Gravity Factor — Gas	<i>F_g</i>	—	—	—	•	—	—	—	—	—	3-4
Gravity-Temperature Factor — Liquid	<i>F_{gt}</i>	—	—	—	—	—	•	—	—	—	3-4
Meter Factor	<i>MF</i>	•	•	—	—	—	—	•	•	—	3-1
Count (Volume)	<i>CNT</i>	•	•	—	—	—	—	•	•	—	3-1
Constant				1.0618		1.0618					
Steam Factor	<i>F_s</i>									•	3-27/28
Expansion Factor	<i>Y</i>	—	—	•	•	—	—	—	—	•	3-1
Temperature Base Factor	<i>F_{tb}</i>	—	•	—	•	—	—	—	—	—	3-4
Orifice Thermal Expansion Factor	<i>F_a</i>	—	—	•	•	•	•	—	—	•	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.



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 $45^\circ \pm 15^\circ$ 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

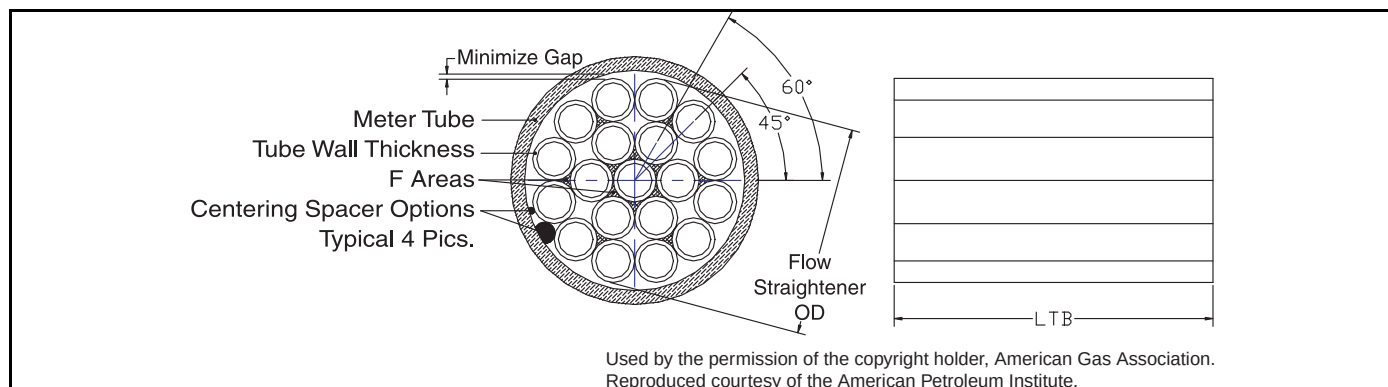
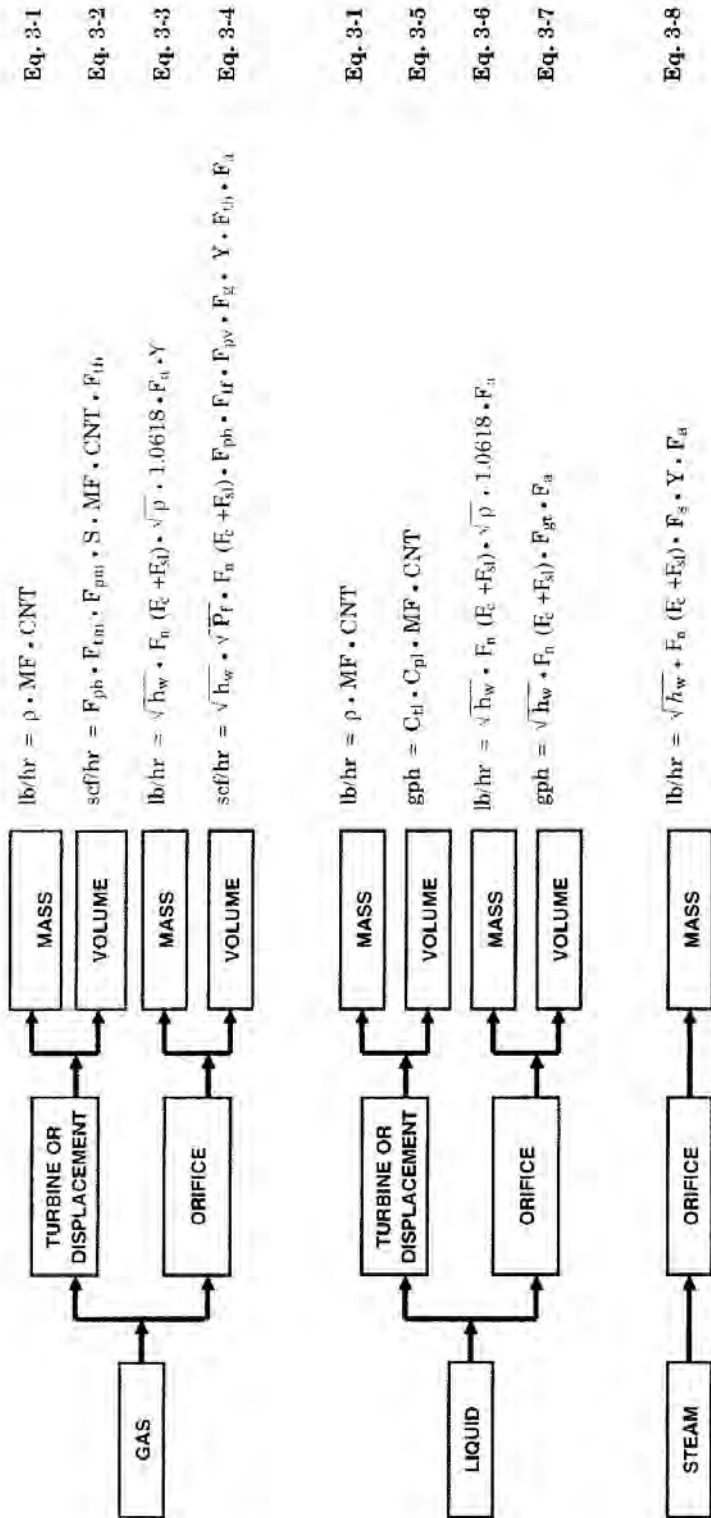


FIG. 3-4
Flow Calculation Guide Equations



Other Equations

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

$$C' = F_n (E_c + E_d) \cdot F_{ph} \cdot F_{th} \cdot F_{pv} \cdot F_a \cdot Y \cdot F_{th} \cdot F_a$$

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

$$C_{pl} = \frac{1}{1 - (F \cdot D_p)}$$

$$F_n (E_c + E_d) = \text{Key} = \text{orifice factor (Fig. 3-13)}$$

$$F_{pb} = \frac{14.73}{P_b}$$

$$F_{tb} = \frac{460 + T_b}{520}$$

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

$$F_g = \sqrt{\frac{1.0000}{G}} \quad P_{gt} = \frac{1.0057}{\sqrt{G_1}} \sqrt{\frac{G_f}{G_1}}$$

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



FIG. 3-5
Orifice Plate Holders

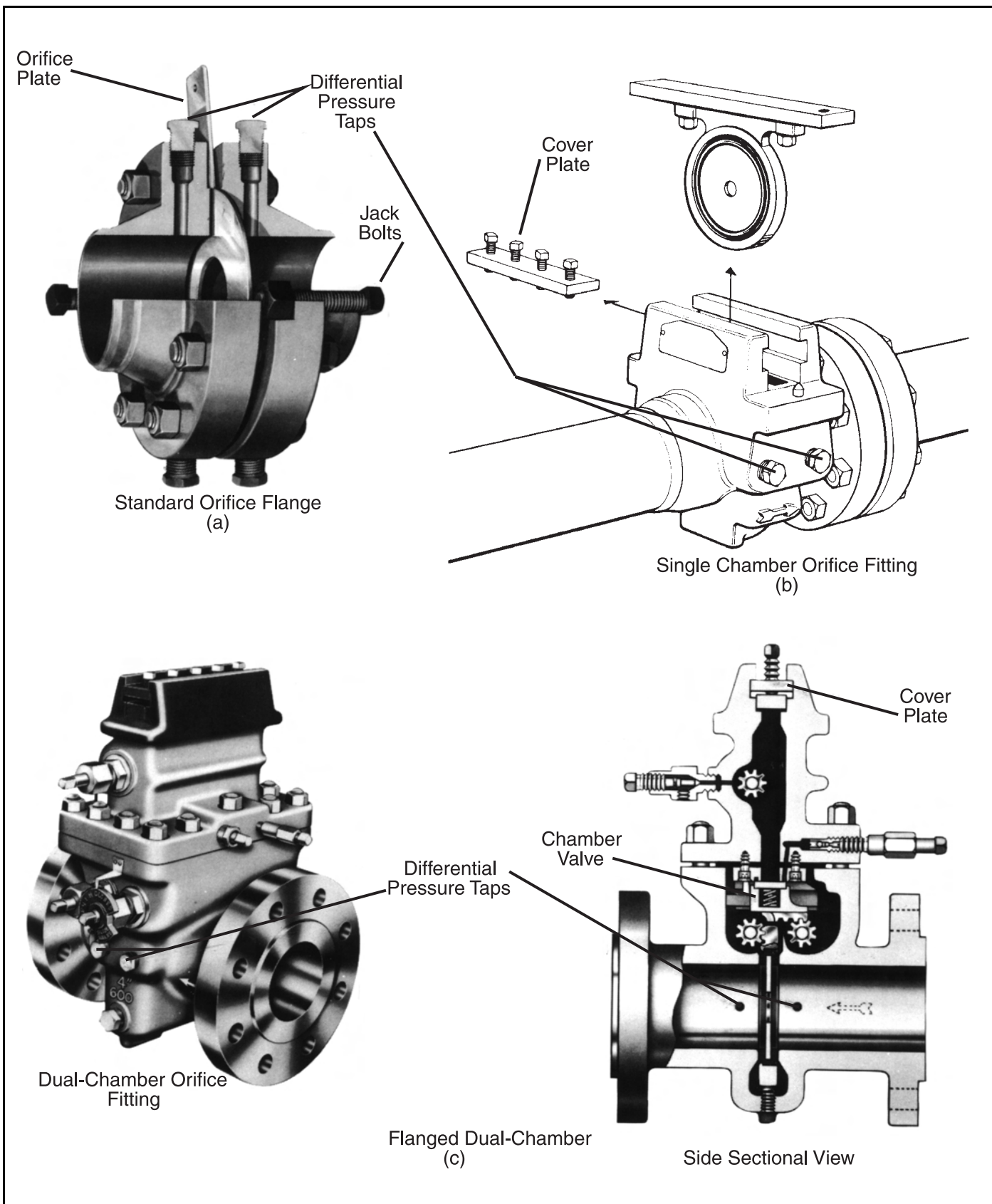
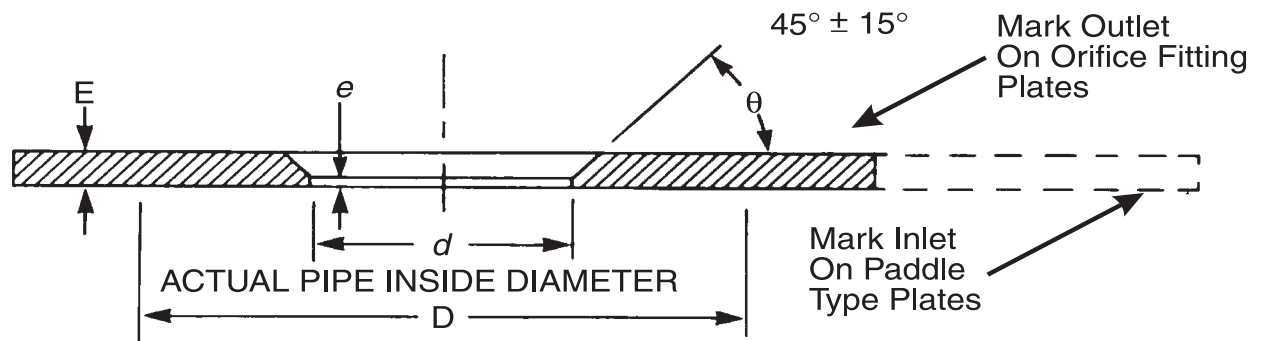


FIG. 3-6
Orifice Plate Thickness and Maximum Allowable Differential Pressure Based on the Structural Limit



Nominal Pipe Size (NPS) (inches)	Published Inside Pipe Diameter (inches)	Orifice Plate Thickness E (inches)			Maximum Allowable ΔP (In. H ₂ O) Orifice Fitting	Maximum Allowable ΔP (In. H ₂ O) 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
4	3.068	0.115	0.130	0.125	1000	1000
	3.152	0.115	0.130	0.125	1000	1000
	3.438	0.115	0.130	0.125	1000	1000
6	3.826	0.115	0.130	0.125	1000	1000
	4.026	0.115	0.130	0.125	1000	1000
	4.897	0.115	0.163	0.125	345	1000
8	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
10	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
12	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
16	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
20	14.688	0.175	0.490	0.375	465	1000
	15.000	0.175	0.500	0.375	465	1000

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 ₂ O) Orifice Fitting	Maximum Allowable ΔP (In. H ₂ O) 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

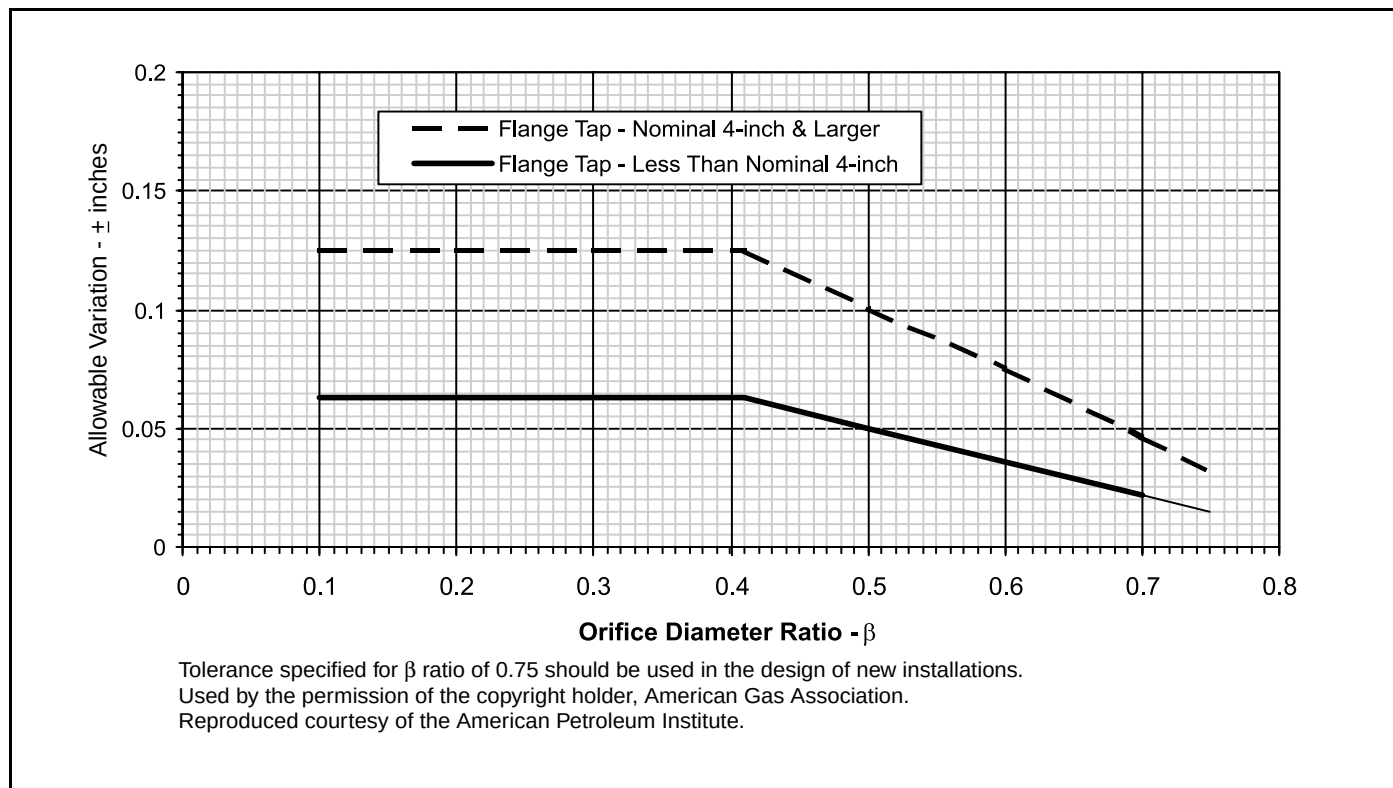


FIG. 3-8

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

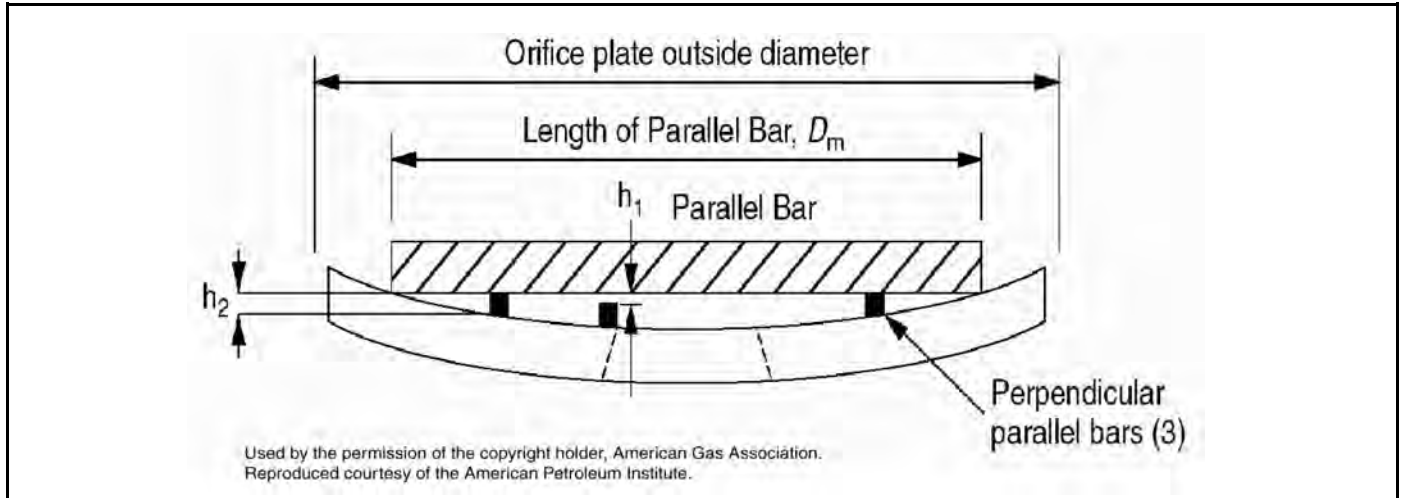


FIG. 3-9

Orifice Meter Tube Layout for Flanged or Welded Limit

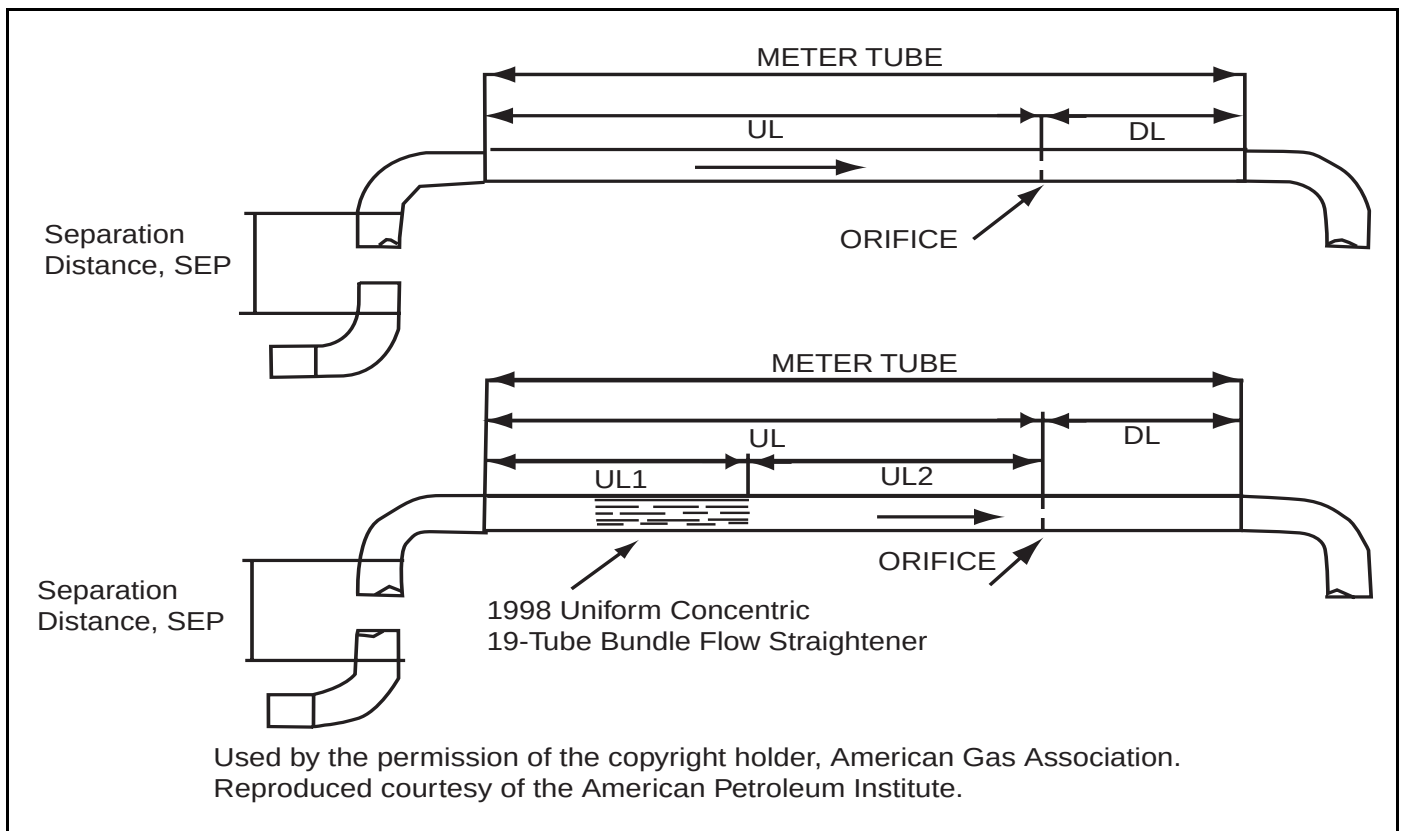


FIG. 3-10
Orifice Meter Installation Requirements without a Flow Conditioner

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

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 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 ,* 5D _i ≤ SEP ≤ 15D _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	10	50	19	30	17	6	70	2.8
0.30	11	10	12	12	50	32	30	19	6	108	3.0
0.40	16	10	13	13	50	44	30	21	6	145	3.2
0.50	30	30	18	18	95	44	30	25	7	145	3.5
0.60	44	44	30	30	95	44	30	30	9	145	3.9
0.67	44	44	44	44	95	44	44	35	11	145	4.2
0.75	44	44	44	44	95	44	44	44	13	145	4.5
Recommended length for maximum range $\beta \leq 0.75$	44	44	44	44	95	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 $\pm 0.25D_i$.

* – 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 $UL \geq 29D_i$.

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	5-25	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 $\beta \leq 0.75$	12-12.5 $\beta \leq 0.75$	12-13 $\beta \leq 0.75$	11-12.5 $\beta \leq 0.75$	13 $\beta \leq 0.75$	13 $\beta \leq 0.67$	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 $\pm 0.25D_i$.

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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;

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

Rearranging,

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

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

$$Key_g = 201.342$$

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,

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

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

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

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:

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

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

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

$$C_g = \sqrt{\frac{0.60}{G}} \quad \text{Eq 3-14}$$

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-

FIG. 3-13

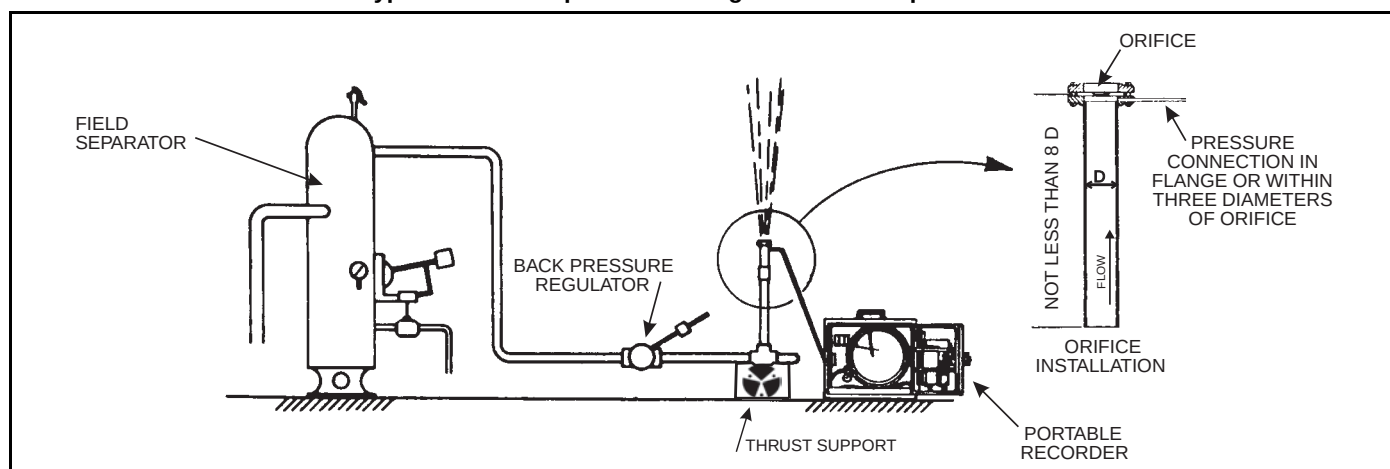
Plate Sizing and Approximate Flowrate, Natural Gas, Natural Gas Liquids and Steam (For Flange Taps)

Plate ID Inches	KEY								Plate ID Inches
	Nominal Meter Tube ID (Inches)								
	2	3	4	6	8	10	12	16	
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 • P _f / G) ^{0.5}	180.156		
KEY _l = 1.06 • (h _w • ρ _f) ^{0.5}		41.885	
KEY _s = 4.47 • (h _w • 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

FIG. 3-14

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



stream transducer to the downstream transducer with the flow (t_1), and from the downstream to the upstream transducers against the flow (t_2). 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 manufacturers 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).

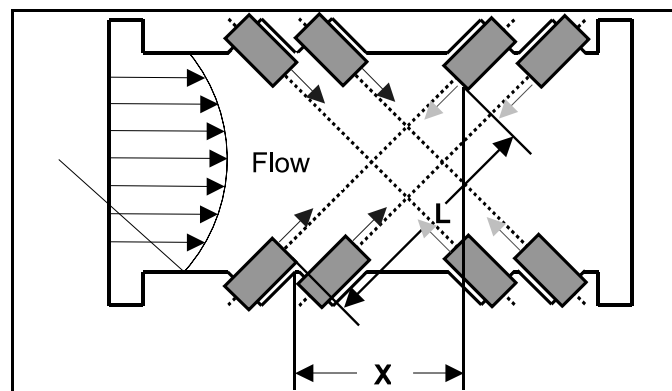
UM Meter Tube and Approach Piping Considerations

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 90° 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

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 contaminants 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](#)

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_1 is calculated using appropriate terms of Eq 3-7.

$$Q_h = Key_1 \cdot F_{gt} \cdot \sqrt{h_w}$$

$$\begin{aligned} \text{or } Key_1 &= \frac{Q_h}{F_{gt} \cdot \sqrt{h_w}} && \text{From Eq 3-7} \\ &= \frac{3400}{1.2193 \cdot \sqrt{50}} = 394.35 \end{aligned}$$

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_1 . A 1.5 inch orifice diameter would be selected, which has a 471 Key_1 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_1 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,

$$Q_h = Key_1 \cdot F_{gt} \cdot \sqrt{h_w} \quad \text{From Eq 3-7}$$

The value of Key_1 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,

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

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

- 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 further 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.

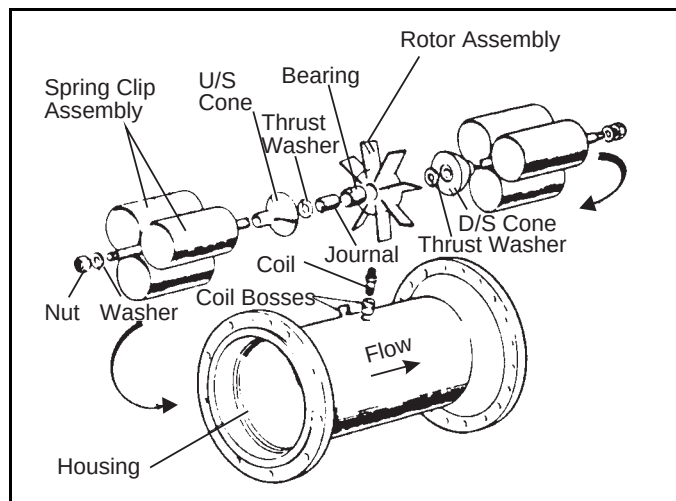
Several more different techniques and processes have been developed to directly measure the mass of a flowing fluid. The devices utilize the principle that angular momentum of a mass is directly proportional to the mass velocity. The resistance of a mass to change direction is measured by different types 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.

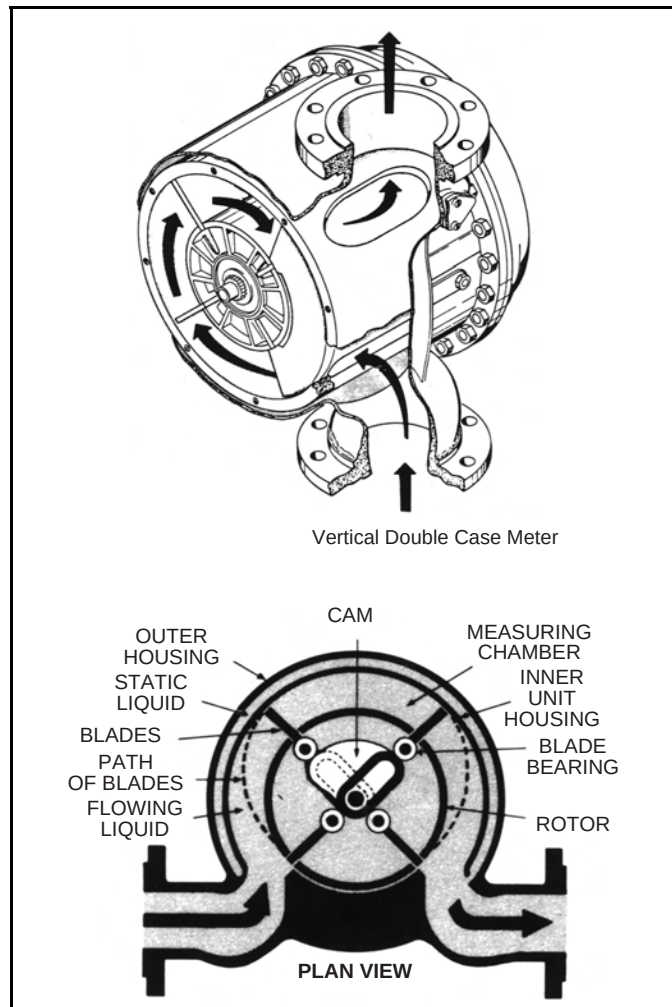
The *pick-off* coils are mounted on one tube while the magnets are mounted on the other. Each coil moves through the uniform magnetic field of the adjacent magnet as the two tubes move. The voltage generated from each pickoff coil creates a sine wave representing the motion of one tube relative to the 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



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

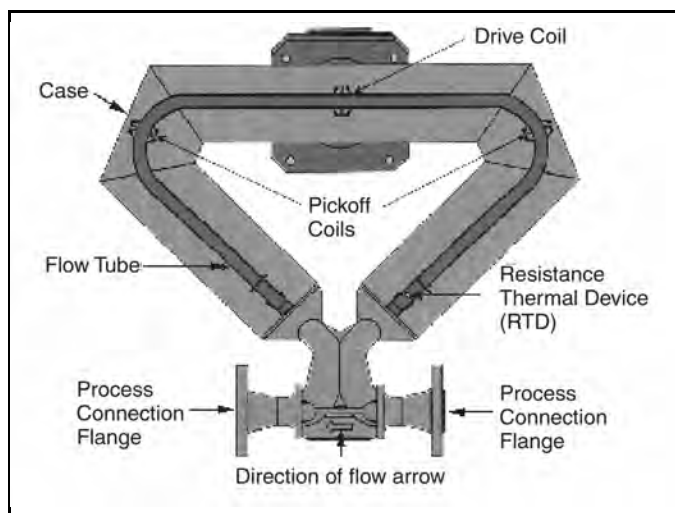
The purpose of proving a turbine or PD meter is to establish a factor which, when multiplied by the meter pulses and divided by the nominal K-factor, will give an accurate, corrected metered volume. Compensation for meter factors can be accomplished external to the meter through either mathematical or 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.							

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

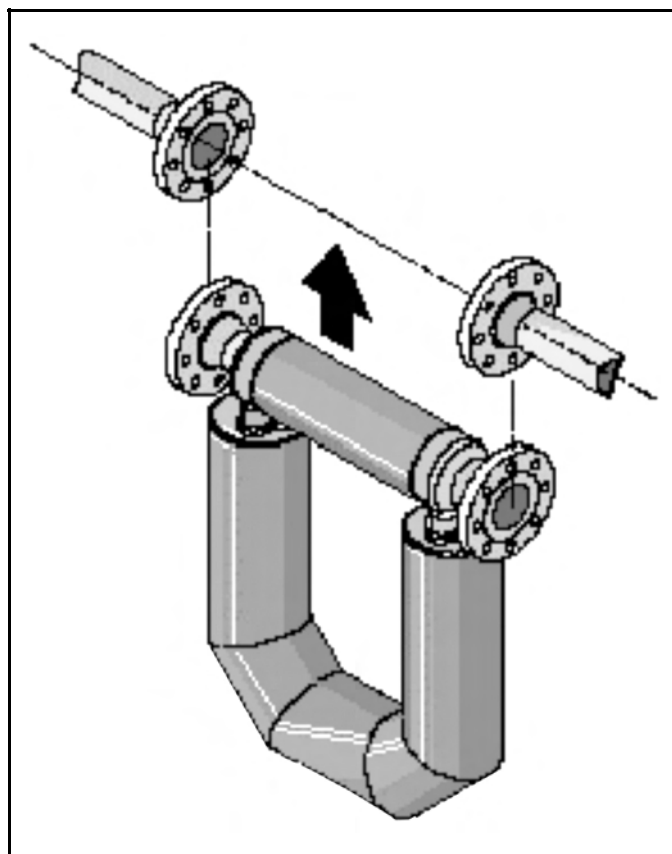


FIG. 3-21
Vibrating Tube Densitometer

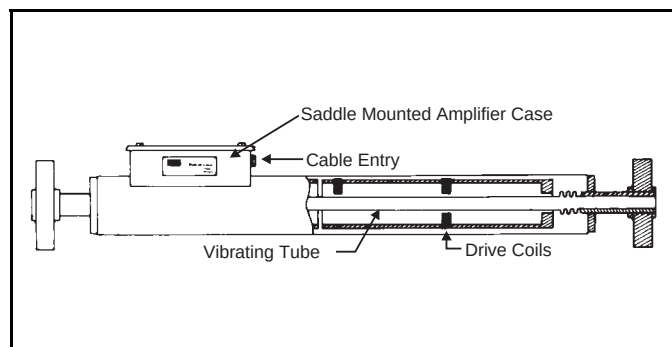
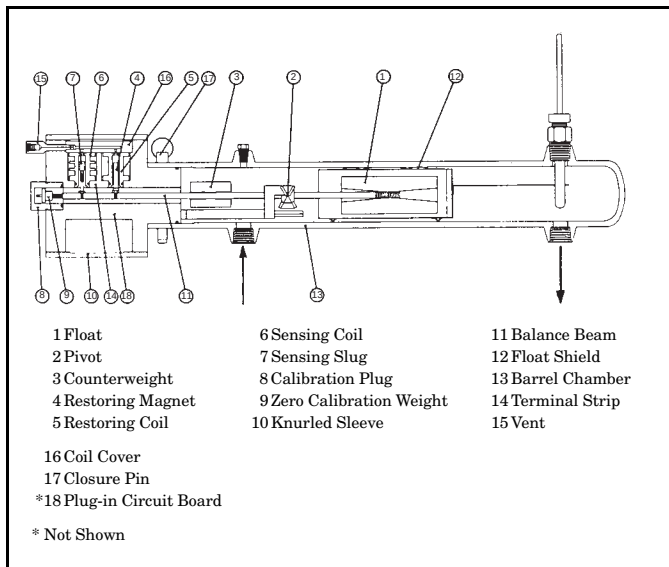


FIG. 3-22
Buoyant Force Densitometer



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
= $C_{ts} \cdot C_{ps} \cdot C_{tl} \cdot C_{pl} = 0.9863$
7. Corrected prover volume
= $0.9863 \times 2.0734 \text{ bbl} = 2.0450 \text{ bbl}$

Metered fluid corrections:

1. Uncorrected metered volume = $\frac{\text{Total meter pulses recorded}}{\text{manufacturer's pulses per barrel}} = \frac{28631}{13188} = 2.1710 \text{ bbl}$
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 = $C_{tl} \cdot C_{pl} \cdot 2.1710 = 2.1421 \text{ bbl}$
5. Meter factor = $\frac{\text{corrected prover volume}}{\text{corrected metered volume}} = \frac{2.0450}{2.1421} = 0.9547$

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.

FIG. 3-23

Temperature Correction Factors for Mild Steel, C_{ts}

C_{ts} for mild steel having a cubical coefficient of expansion of 1.86×10^{-5} per $^{\circ}\text{F}$

Observed Temperature, $^{\circ}\text{F}$	C_{ts} Value	Observed Temperature, $^{\circ}\text{F}$	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.

$$F_b = (F_{na}) (C) \cdot D^2 \quad \text{Eq 3-19}$$

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

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
1.0020										1.0020
1.0021										1.0021
1.0022										1.0022
1.0023										1.0023
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: $C_{ps} = 1 + \frac{P_p - P_s D}{E_m}$

FIG. 3-25
Meter Proving Report

LOCATION				DATE		AMBIENT TEMP.		REPORT NO.															
PROVER DATA				PREVIOUS REPORT																			
BASE VOLUME AT 60°F AND "0" psi.				SIZE		WALL		FLOW RATE		FACTOR		DATE											
bbl.								bbl/hr.															
METER DATA																							
SERIAL NO.		METER NO.		PULSES/bbl.		TEMP. COMP.		MANUF.		SIZE		MODEL											
FLOW RATE				NON-RESET TOTALIZER																			
bbl/hr.																							
RUN DATA																							
TEMPERATURE		PRESSURE		TOTAL PULSES		RUN NO.		C_{ts} = CORRECTION FOR TEMPERATURE ON STEEL C_{ps} = CORRECTION FOR PRESSURE ON STEEL C_{tl} = CORRECTION FOR TEMPERATURE ON LIQUID TABLE 6 OR TABLE 24 FOR LPG's C_{pl} = CORRECTION FOR PRESSURE ON LIQUID															
PROVER AVG.	METER	PROVER	METER																				
AVG.																							
LIQUID DATA																							
TYPE		API GRAVITY		SPECIFIC GRAVITY		R.V. PRESS		BATCH/TENDER NO.															
AT 60°F		AT 60°F		AT 60°F																			
FIELD CALCULATIONS																							
PROVER VOLUME		X		C_{ts}		X		C_{ps}		X		C_{tl}		X		C_{pl}		=		CORRECTED PROVER VOLUME			
AVERAGE PULSES		÷		PULSES/bbl.		=		GROSS METER VOL.		X		C_{tl} USE ONLY FOR NONTEMP. COMP. METER		X		C_{pl}		=		CORRECTED METER VOLUME			
CORRECTED PROVER VOLUME		÷		CORRECTED METER VOLUME		=		METER FACTOR		X		C_{pl} LIQUID CORR. FOR PRESS. AT METERING COND.		=		COMPOSITE FACTOR USE FOR CONSTANT PRESSURE APPLICATIONS							
REMARKS, REPAIRS, ADJUSTMENTS, ETC.,																							
SIGNATURE								DATE				COMPANY REPRESENTED											

Source: API MPMS Ch. 12.2

FIG. 3-26
Example Bidirectional Pipe Prover

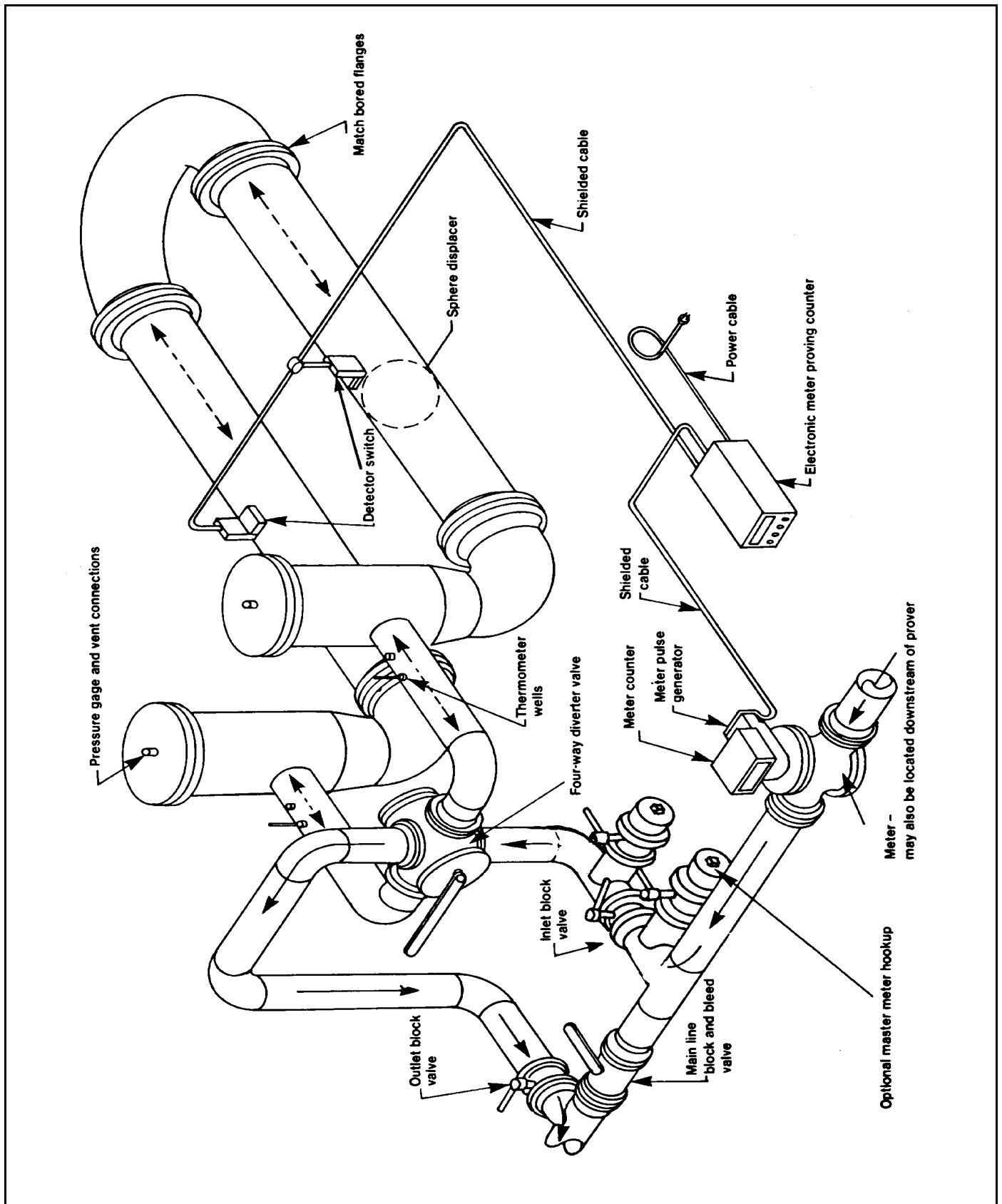


FIG. 3-27

Steam Coefficient Factors, Saturated Steam

$$F_s = \frac{1.0618}{\sqrt{P_{FV}}}, \text{ For Bellows-Type Meters}$$

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

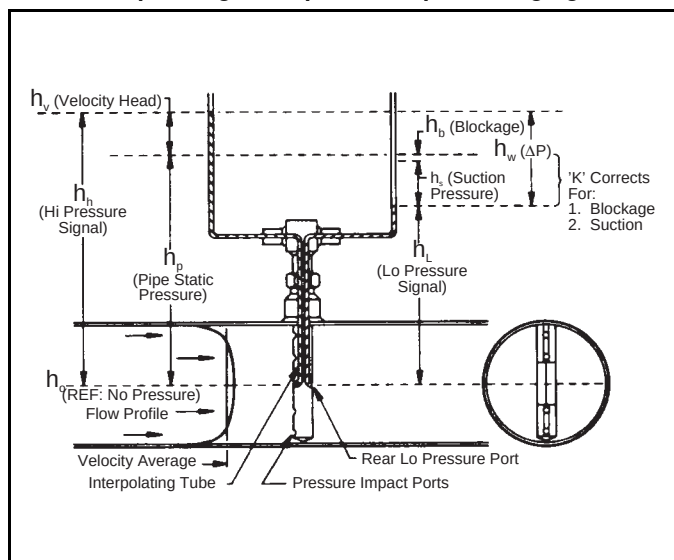
Steam Coefficient Factors, Superheated Steam

$$F_s = \frac{1.0618}{\sqrt{P_{FV}}}, \text{ For Bellows-Type Meters}$$

Pressure, psig	Total Steam Temperature, °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

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 indicated 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.

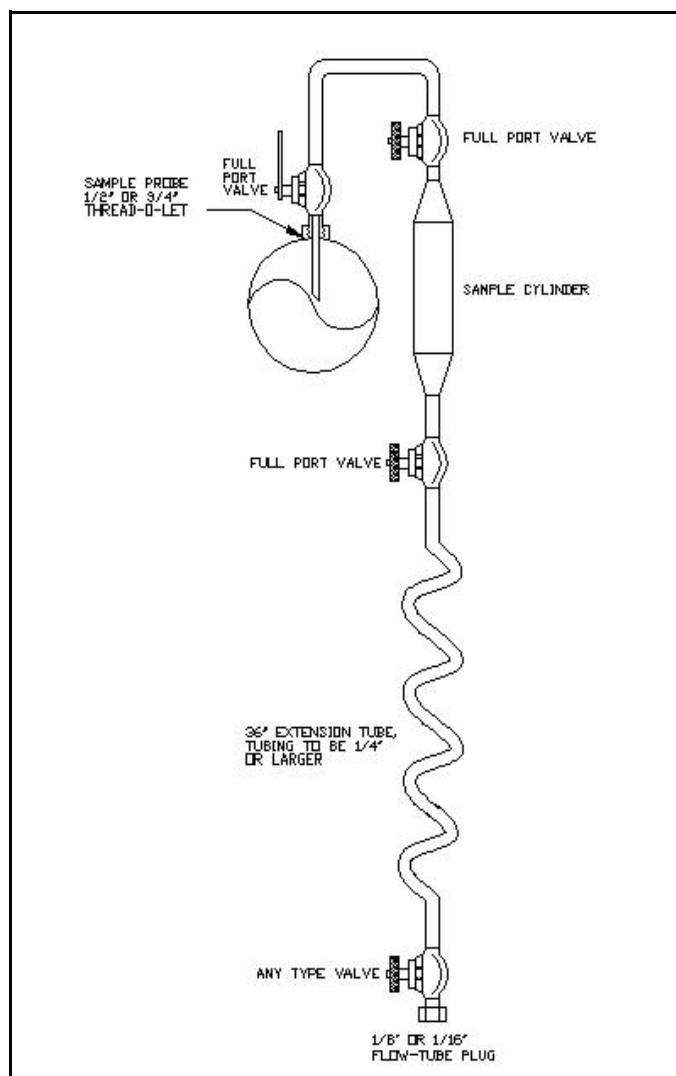
There are several other spot sampling methods detailed in the GPA 2166 standard and referenced in the API Chapter 14.1 standard. For relatively rich, unprocessed gas streams, the fill and empty, helium pop or the displacement methods should be used. If the gas is *continuously* lean and dry and sampled 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,

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 contaminants 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 pre-charged 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)

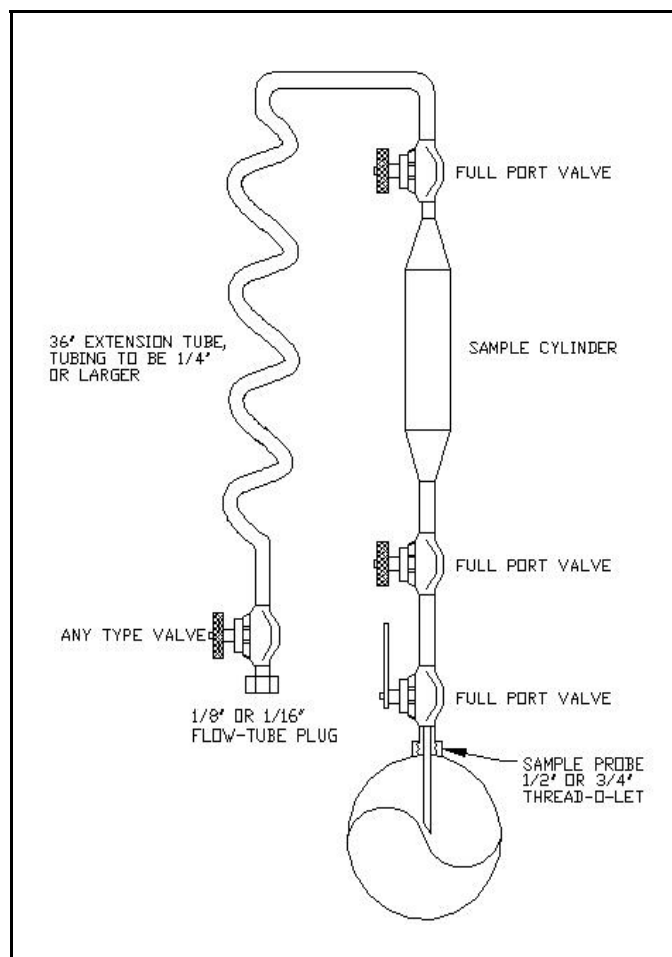
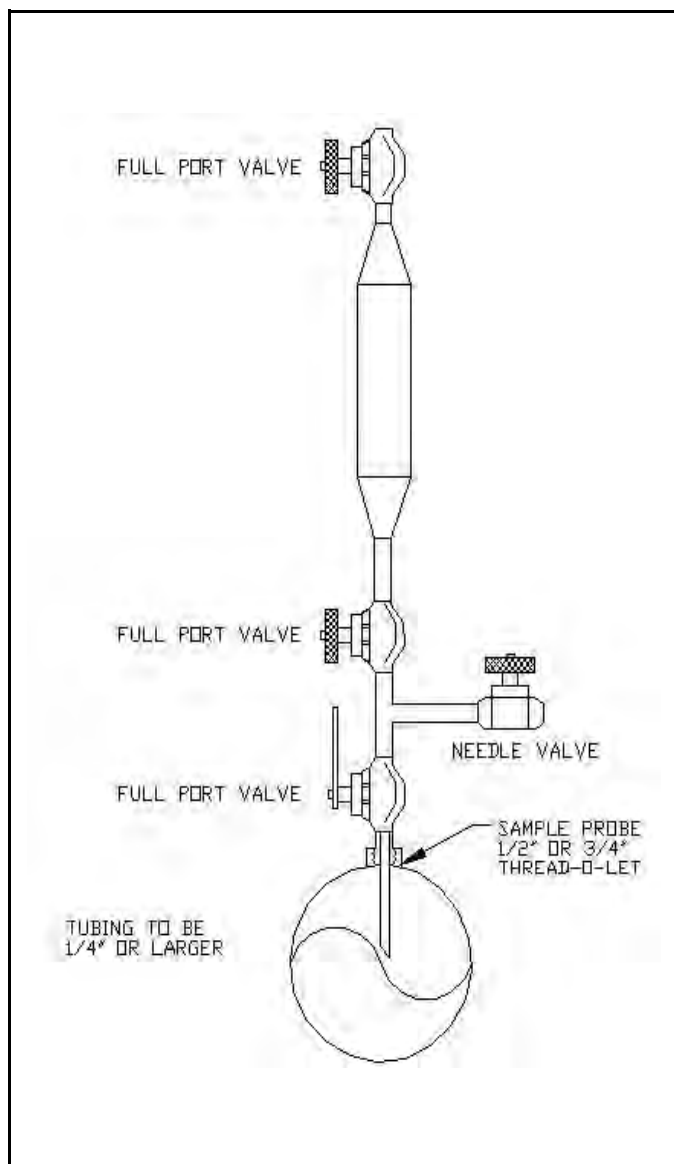


FIG. 3-32
Helium Pop Sampling Method



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

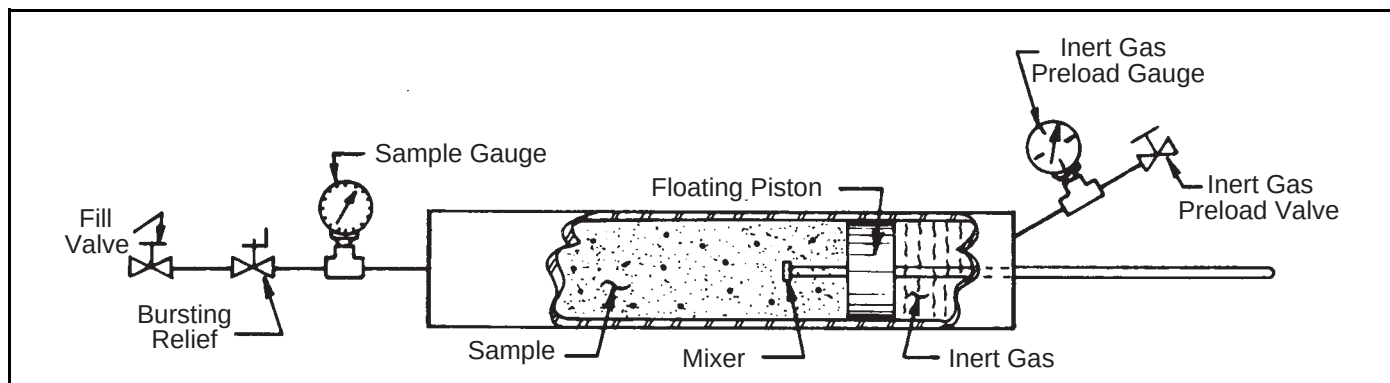
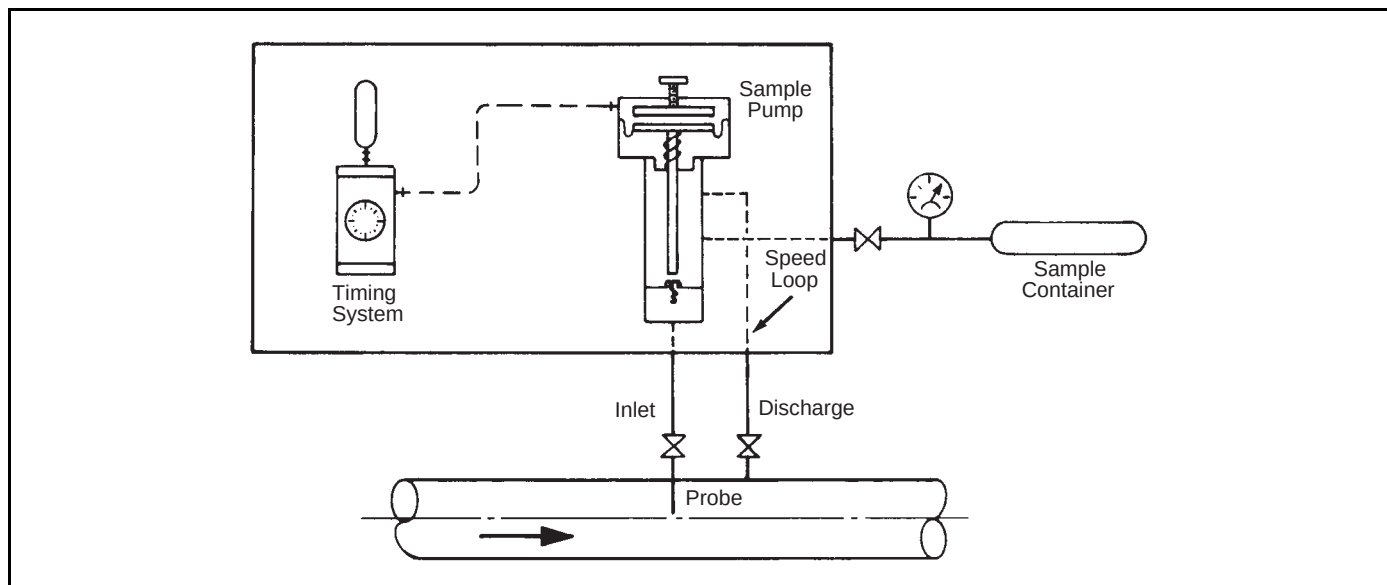


FIG. 3-34
Continuous Sampler



- 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 con-

sidered 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 °F 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.

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 homogeneous 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-

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.

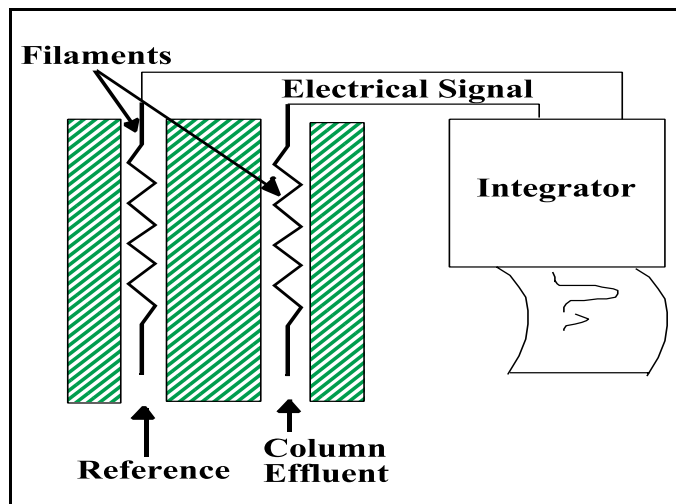
Detectors

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.

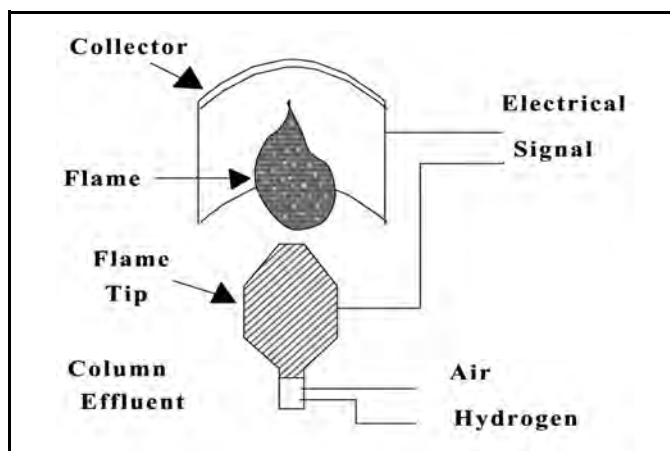
FIG. 3-35
Thermal Conductivity Detector



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 collec-

FIG. 3-36
Flame Ionization Detector



tor that develops an electrical signal proportional to the concentration of the component.

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.

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.

REFERENCES

1. Fluid Meters, Sixth Edition (1971), The American Society of Mechanical Engineers, New York, New York.
 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.
 4. GPA TP-25, Temperature Correction for the Volume of Light Hydrocarbons, Tables 24E and 23E, Gas Processors Association of 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	$N_1, N_2...N_x$ = numerical constants for units of measurement used
CO = controller output	P_c = absolute thermodynamic critical pressure
d = valve inlet diameter	P_u = ultimate time period
D = internal diameter of the pipe	P_v = vapor pressure of liquid at valve inlet temperature, psia
F_d = valve style modifier	P_1 = upstream absolute static pressure, measured two nominal pipe diameters upstream of valve-fitting assembly, psia
F_F = liquid critical pressure ratio factor, dimensionless	P_2 = downstream absolute static pressure, measured six nominal pipe diameters downstream of valve-fitting assembly, psia
F_k = ratio of specific heats factor, dimensionless	ΔP = pressure differential, $P_1 - P_2$
F_L = liquid pressure recovery factor of a valve without attached fittings, dimensionless	PB = proportional band setting
F_p = piping geometry factor, dimensionless	PB_u = ultimate proportional band setting
F_R = Reynolds number factor, dimensionless	PV = process variable measurement
G_r = liquid specific gravity at upstream conditions, ratio of density of liquid at flowing temperature to density of water at 60° F, dimensionless	(PV - SP) = error signal
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	q = volumetric flow rate
k = ratio of specific heats, dimensionless	Q_g = gas or vapor flow rate, lb/hr or cu ft/hr
K = gain	Q_L = liquid flow rate, gpm
K_c = cavitation index, dimensionless	SP = setpoint
K_d = derivative gain constant	T = absolute temperature of gas at inlet, °R
K_i = integral mode gain constant	T_c = time constant
K_n = normalization constant	T_d = derivative mode time constant
K_p = proportional mode gain constant	T_i = integral mode time constant
K_u = ultimate sensitivity	T_t = absolute upstream temperature (in degrees R)
M = molecular weight, atomic mass units	w = weight or mass flow rate
MO = manual-mode controller output	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

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

FIG. 4-1 (Cont'd)

Nomenclature

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 $CO = K_i[(PV - SP) dt]$.

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/Window: Also called "controller windup/window" 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.

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 mani-

fold 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.

FIG. 4-2
Instrumentation Symbols

CONTROL VALVES AND REGULATORS	RELIEF VALVES AND ACTUATORS	FIRE AND SAFETY	PRIMARY FLOW ELEMENTS
GATE 3-WAY BALL 3-WAY BALL GLOBE ANGLE GLOBE 3-WAY GLOBE BUTTERFLY 4-WAY SOLENOID ANGLE SOLENOID 3-WAY SOLENOID 4-WAY SOLENOID PLUG 3-WAY PLUG DIAPHRAGM KNIFE SLIDE 3-WAY SLIDE DAMPER REDUCING PRESSURE REGULATOR SELF CONTAINED BACKPRESSURE REGULATOR SELF CONTAINED REDUCING PRESSURE REGULATOR W/ EXTERNAL TAP BACKPRESSURE REGULATOR W/ EXTERNAL TAP DIFFERENTIAL PRESSURE REDUCING REGULATOR TEMPERATURE REGULATOR FILLED SYSTEM TYPE	PRESSURE RELIEF VALVE VACUUM RELIEF VALVE PRESSURE/VACUUM RELIEF VALVE PILOT OPERATED RELIEF VALVE ATMOSPHERIC PRESSURE RELIEF VALVE RUPTURE DISC AUTO RECIRCULATION VALVE DIAPHRAGM ACTUATOR ACTUATOR WITH HANDWHEEL PRESSURE BALANCED DIAPHRAGM ACTUATOR SINGLE/DOUBLE ACTING CYLINDER ACTUATOR MOTOR OPERATED ACTUATOR DIGITAL ACTUATOR ELECTROHYDRAULIC ACTUATOR SINGLE SOLENOID ACTUATOR SINGLE SOLENOID ACTUATOR WITH MANUAL RESET HAND ACTUATOR OR HANDWHEEL OFF-LINE SINGLE SOLENOID WITH SINGLE ACTUATOR OFF-LINE ANGLE SOLENOID WITH SINGLE ACTUATOR OFF-LINE 3-WAY SOLENOID WITH SINGLE ACTUATOR OFF-LINE 4-WAY SOLENOID WITH SINGLE ACTUATOR OFF-LINE 4-WAY SOLENOID WITH DOUBLE ACTUATOR OFF-LINE SWITCHING VALVE	ALARM BOX FLAME DETECTOR HEAT DETECTOR SPEAKER/HORN SMOKE DETECTOR GAS DETECTOR INSTRUMENT SIGNAL LINES INSTRUMENT TO PROCESS CAPILLARY MECHANICAL LINK ELECTRIC PNEUMATIC HYDRAULIC GUIDED ELECTROMAGNETIC UNGUIDED ELECTROMAGNETIC SOFTWARE OR DCS LINK	IN-LINE ELEMENT INTEGRAL ORIFICE PLATE PRE-FAB METER RUN ORIFICE PLATE IN QUICK CHANGE FITTING AND PRE-FAB METER RUN ORIFICE PLATE IN QUICK CHANGE FITTING VENTURI AVERAGING PITOT SINGLE PORT PITOT FLOW NOZZLE ROTAMETER POSITIVE DISPLACEMENT TYPE TURBINE OR PROPELLER TYPE CORIOLIS TYPE TARGET TYPE VORTEX MAGNETIC ULTRASONIC FLUME WEIR SEGMENTAL WEDGE
		OTHER INSTRUMENTATION FIELD MOUNTED INSTRUMENT WITH IMPLIED PIPING VALVE INSTRUMENT COMPONENT LABEL INDICATES A PROVISION TO RESTRICT THE RELEASE OF FLAMMABLE LIQUIDS IS REQUIRED. SEE CSE STANDARDS GUIDE, SECTION 18 FOR DETAILS VENDOR INSTRUMENT FURNISHED BY EQUIPMENT VENDOR POSITIVE DISPLACEMENT TYPE FLOWMETER W/ PRESSURE REGULATOR FLOW STRAIGHTENING VANE	INSTRUMENT REMARKS - MAX 3 PER INSTRUMENT SP FIC 1234 PI 1234 FY 1234 FE 1234 HH - ALARM HIGH HIGH LABEL H - ALARM HIGH LABEL L - ALARM LOW LABEL LL - ALARM LOW LOW LABEL IMPLIED COMPONENT WITH A CHANGE IN DEFAULT NOMINAL DIAMETER FOR PIPING VALVE 1.5" CHEMICAL SEAL CONTROL VALVE LABEL CONTROL VALVE WITH NOMINAL DIAMETER AND FAIL ACTION LABEL 2" FO IF DEVICE HAS INTEGRAL TRANSMITTER, SHOW FT INSTEAD OF FE INSIDE TAG. TYPICAL FOR ALL PRIMARY ELEMENTS

FIG. 4-2 (Cont'd)
Instrumentation Symbols

INSTRUMENT OR FUNCTION SYMBOLS		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
	FIELD MOUNTED INSTRUMENT	A ANALYSIS		ALARM		
	INSTRUMENT MOUNTED ON MAIN CONTROL ROOM PANEL	B BURNER, COMBUSTION		USER'S CHOICE	USER'S CHOICE	USER'S CHOICE
	INSTRUMENT MOUNTED BEHIND MAIN CONTROL ROOM PANEL	C USER'S CHOICE			CONTROL	
	INSTRUMENT FUNCTION MOUNTED ON LOCAL PANEL	D USER'S CHOICE	DIFFERENTIAL			
	INSTRUMENT FUNCTION MOUNTED BEHIND LOCAL PANEL	E VOLTAGE		SENSOR (PRIMARY ELEMENT)		
	INSTRUMENT SHOWN ON ANOTHER DRAWING	F FLOW RATE	RATIO (FRACTION)			
	FIELD MOUNTED LIGHT	G USER'S CHOICE		GLASS, VIEWING DEVICE		
	LIGHT MOUNTED ON MAIN CONTROL ROOM PANEL	H HAND				HIGH
	LIGHT MOUNTED ON LOCAL PANEL	I CURRENT (ELECTRICAL)		INDICATE		
	LOCAL DCS SHARED CONTROL FUNCTION	J POWER	SCAN			
	PRIMARY DCS SHARED DISPLAY SHARED CONTROL FUNCTION ACCESSIBLE TO OPERATOR	K TIME, TIME SCHEDULE	TIME RATE OF CHANGE		CONTROL STATION	
	PRIMARY DCS SHARED DISPLAY SHARED CONTROL FUNCTION INACCESSIBLE TO OPERATOR	L LEVEL		LIGHT		LOW
	AUXILIARY DCS SHARED DISPLAY SHARED CONTROL FUNCTION ACCESSIBLE TO OPERATOR	M USER'S CHOICE	MOMENTARY			MIDDLE, INTERMEDIATE
	AUXILIARY DCS SHARED DISPLAY SHARED CONTROL FUNCTION INACCESSIBLE TO OPERATOR	N USER'S CHOICE		USER'S CHOICE	USER'S CHOICE	USER'S CHOICE
	HARD WIRED INTERLOCK	O USER'S CHOICE		ORIFICE, OPEN		
	LOCAL PLC SHARED CONTROL FUNCTION	P PRESSURE, VACUUM		POINT (TEST) CONNECTION		
	PRIMARY PLC SHARED DISPLAY SHARED CONTROL FUNCTION ACCESSIBLE TO OPERATOR	Q QUANTITY	INTEGRATE, TOTALIZE			
	PRIMARY PLC SHARED DISPLAY SHARED CONTROL FUNCTION INACCESSIBLE TO OPERATOR	R RADIATION		RECORD		
	PRIMARY COMPUTER SHARED DISPLAY AUXILIARY PLC SHARED DISPLAY SHARED CONTROL FUNCTION ACCESSIBLE TO OPERATOR	S SPEED, FREQUENCY	SAFETY		SWITCH	
	PRIMARY COMPUTER SHARED DISPLAY AUXILIARY PLC SHARED DISPLAY SHARED CONTROL FUNCTION INACCESSIBLE TO OPERATOR	T TEMPERATURE			TRANSMIT	
	LOCAL COMPUTER SHARED CONTROL FUNCTION	U MULTIVARIABLE		MULTIFUNCTION	MULTIFUNCTION	MULTIFUNCTION
	SHARED CONTROL FUNCTION ACCESSIBLE TO OPERATOR	V VIBRATION, MECHANICAL ANALYSIS			VALVE, DAMPER, LOUVER	
	SHARED CONTROL FUNCTION INACCESSIBLE TO OPERATOR	W WEIGHT, FORCE		WELL		
	AUXILIARY COMPUTER SHARED DISPLAY SHARED CONTROL FUNCTION ACCESSIBLE TO OPERATOR	X UNCLASSIFIED	X AXIS	UNCLASSIFIED	UNCLASSIFIED	UNCLASSIFIED
	AUXILIARY COMPUTER SHARED DISPLAY SHARED CONTROL FUNCTION INACCESSIBLE TO OPERATOR	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						
	SUMMING FUNCTION		ROOT EXTRACTION FUNCTION		REVERSE PROPORTIONAL FUNCTION	
	AVERAGING FUNCTION		EXPONENTIAL FUNCTION		CURRENT TO PNEUMATIC	
	DIFFERENCE FUNCTION		UNSPECIFIED FUNCTION		PNEUMATIC TO CURRENT	
	PROPORTIONAL FUNCTION		TIME FUNCTION		} BIAS FUNCTION	
	INTEGRAL FUNCTION		HIGH SELECTING FUNCTION			
	DERIVATIVE FUNCTION		LOW SELECTING FUNCTION			
	MULTIPLYING FUNCTION		HIGH LIMITING FUNCTION			
	DIVIDING FUNCTION		LOW LIMITING FUNCTION			

FIG. 4-3
Instrument Type Features

Pneumatic	Electronic
Advantages	
1. Intrinsically safe, no electrical circuits.	1. Greater accuracy.
2. Compatible with valves.	2. More compatible with computers.
3. Reliable during power outage for short period of time, dependent on size of air surge vessel.	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.	1. Contacts subject to corrosion.
2. Subject to air leaks.	2. Must be air purged, explosion proof, or intrinsically safe to be used in hazardous areas.
3. Mechanical parts may fail due to dirt, sand, water, etc.	3. Subject to electrical interference.
4. Signal boosters often needed on transmission lines of over 300 feet.	4. More difficult to provide for positive fail-safe operation.
5. Subject to freezing with moisture present.	5. Requires consideration of installation details to minimize points 1, 2, 3, and 4.
6. Control speed is limited to velocity of sound.	

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.

FIG. 4-5
Types of Manometers

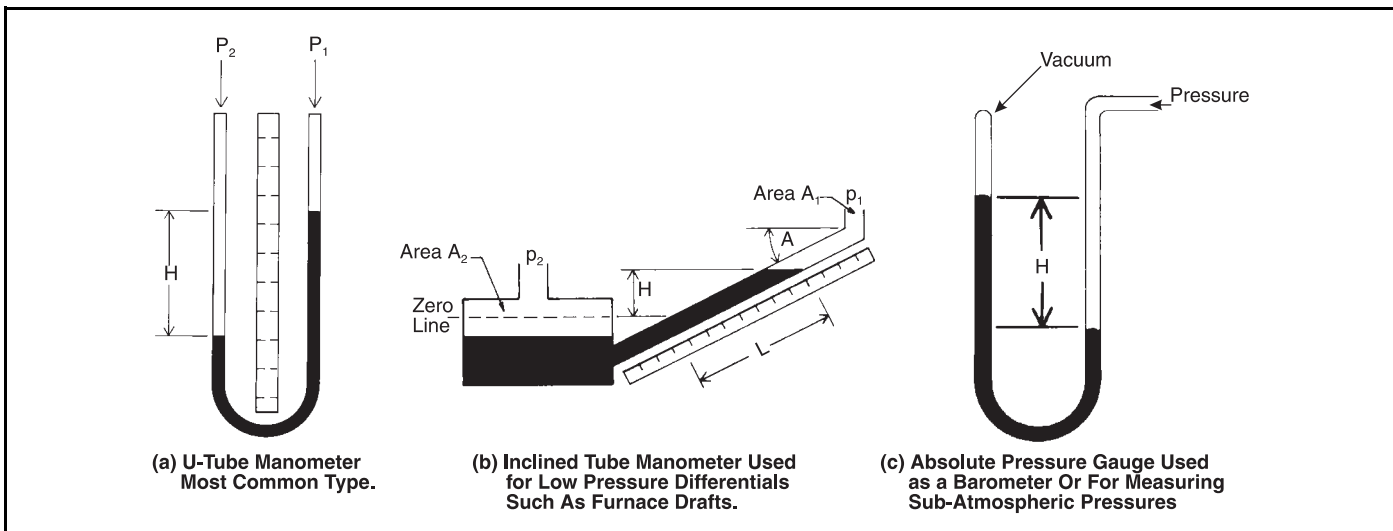


FIG. 4-6
Types of Bourdon Tubes

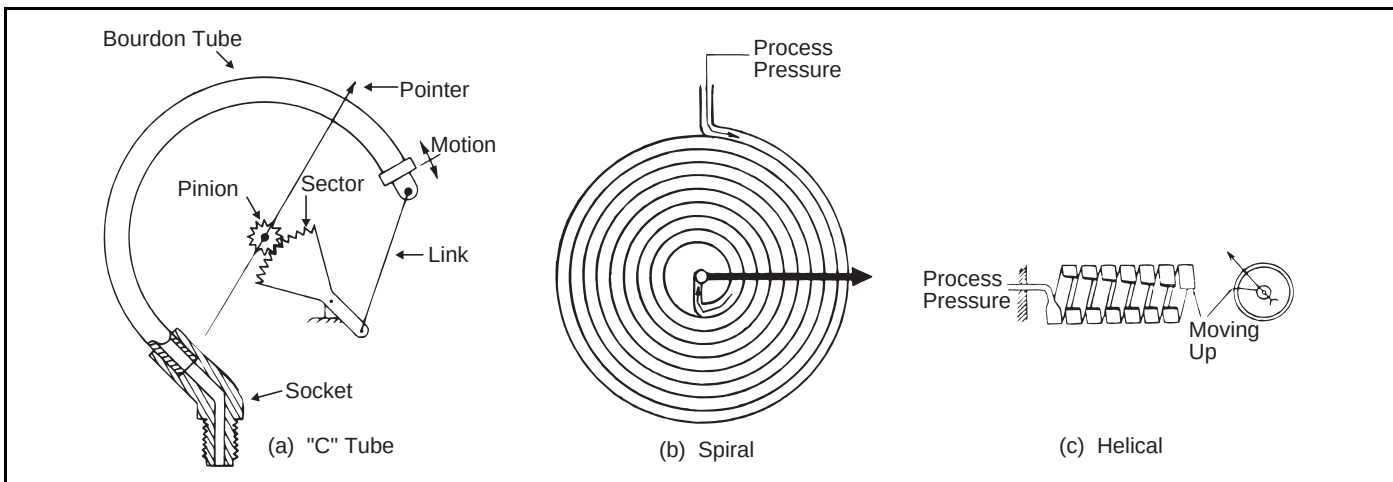
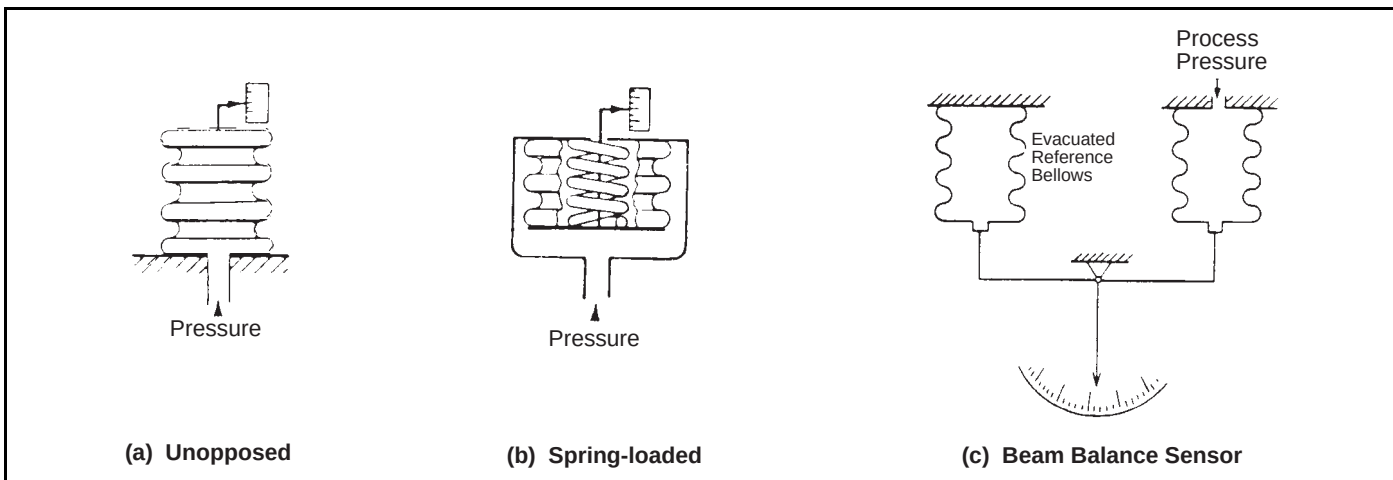


FIG. 4-7
Types of Bellows



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

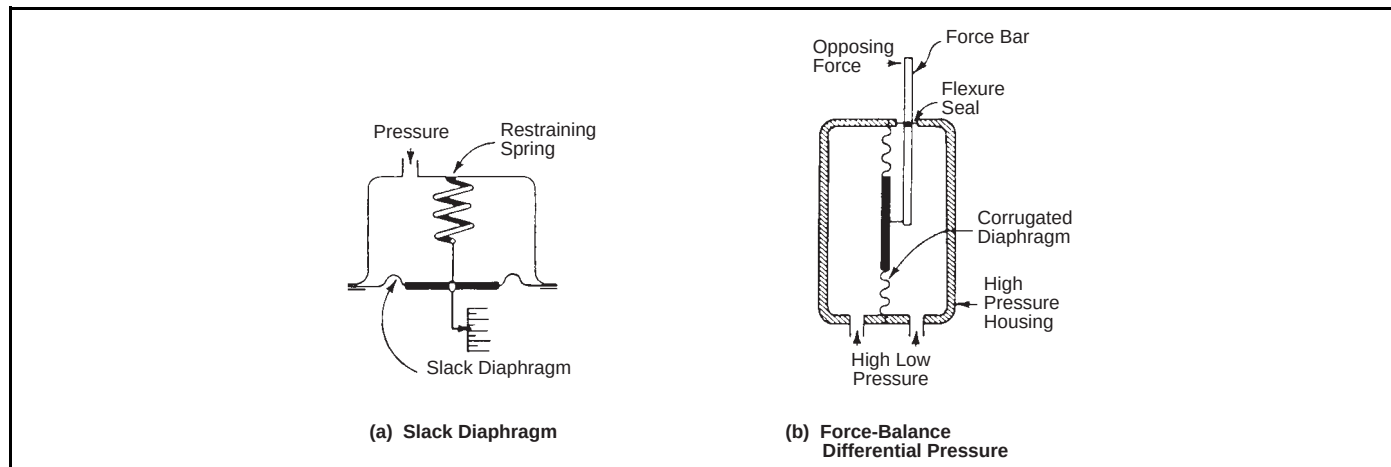


FIG. 4-9
Flat Glass Gauge Glasses

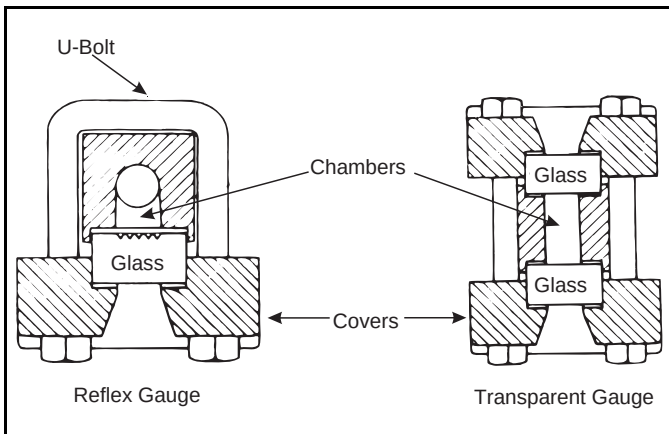


FIG. 4-11
Lever and Shaft Float Gauge

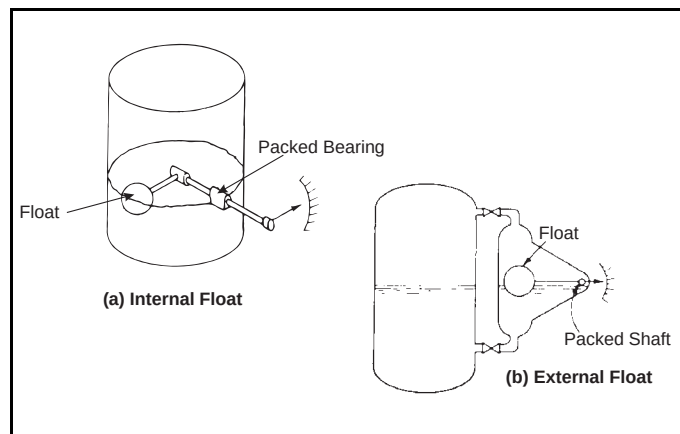


FIG. 4-10
Chain and Tape Float Gauge

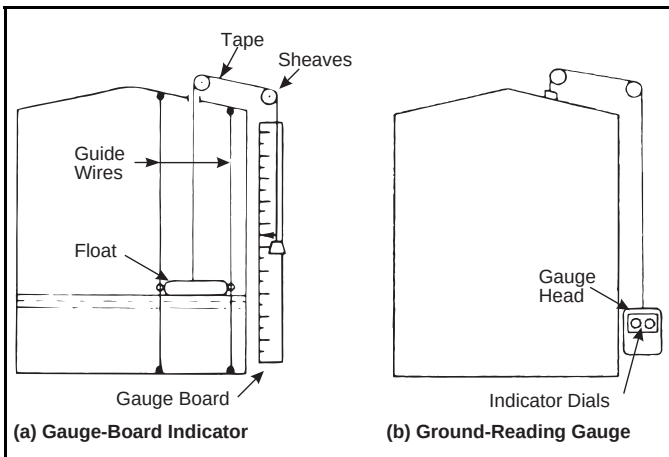


FIG. 4-12
Displacer Level Measuring Device

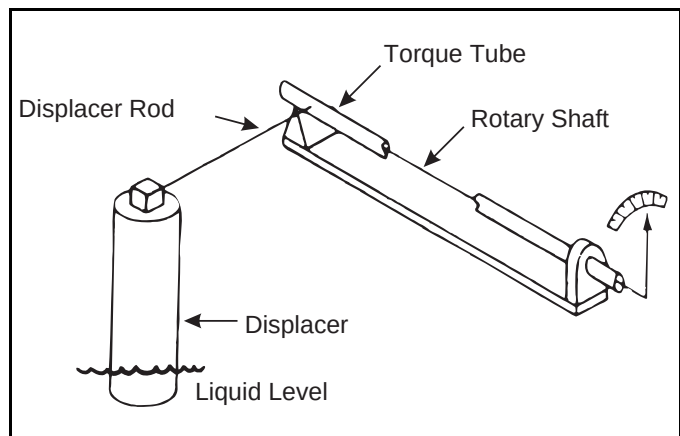
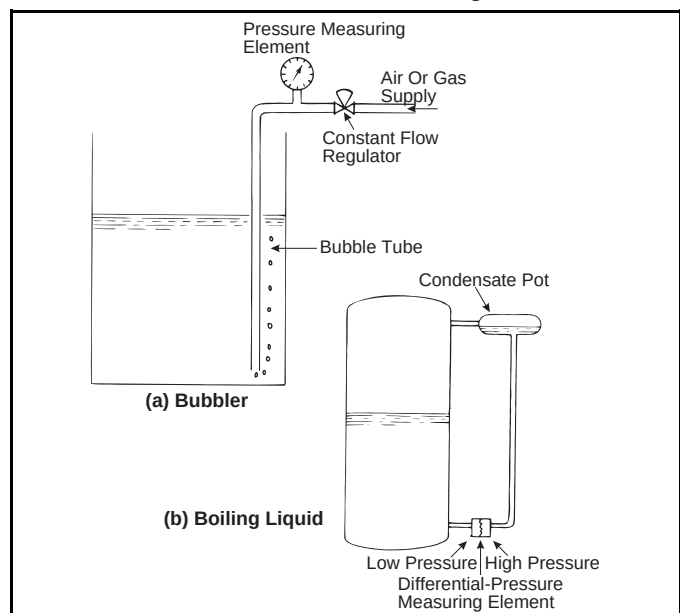


FIG. 4-13

Head Pressure Level Gauges



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.

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

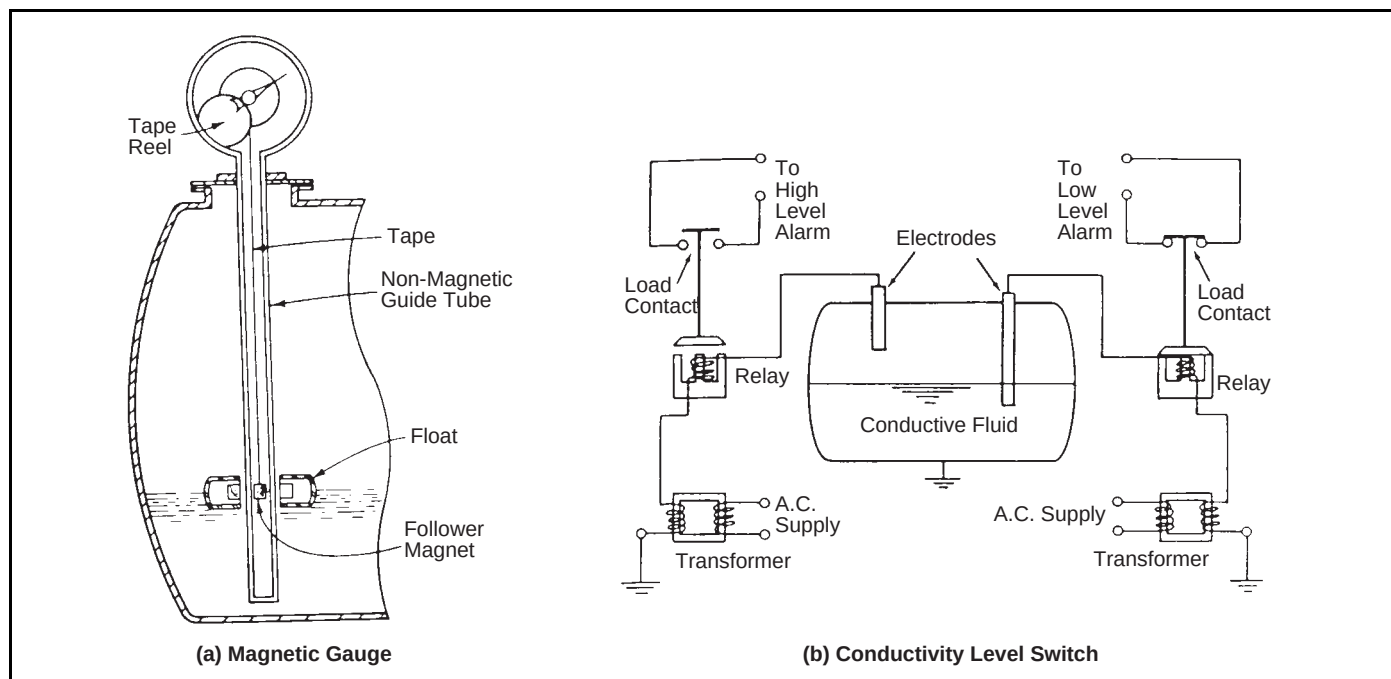


FIG. 4-15

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.

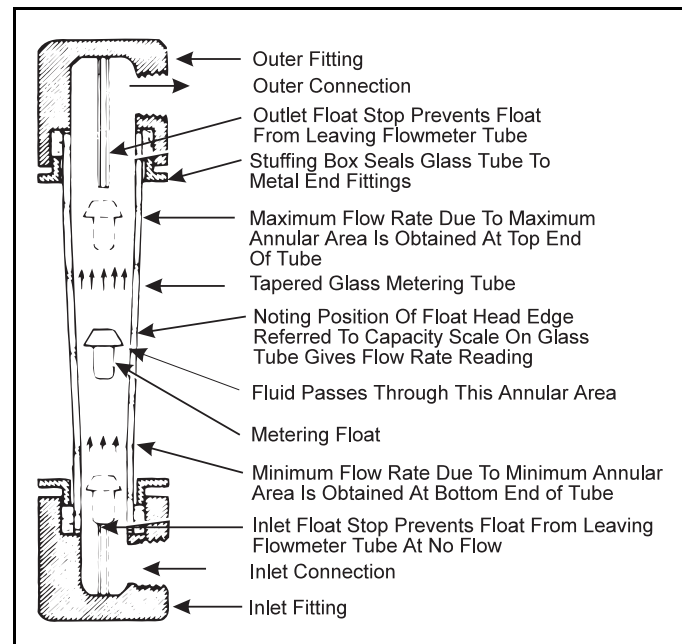
By installing more than one pair of transducers, a larger range of the flow profiles across the metering section can be covered and thereby increase the accuracy of the meter.

Other flowmeters — Some other flowmeter types occasionally encountered are:

- Doppler effect or ultrasonic flowmeters
- Vortex shedding flowmeters
- Laser velocimeters

FIG. 4-16

Rotameter



- 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.

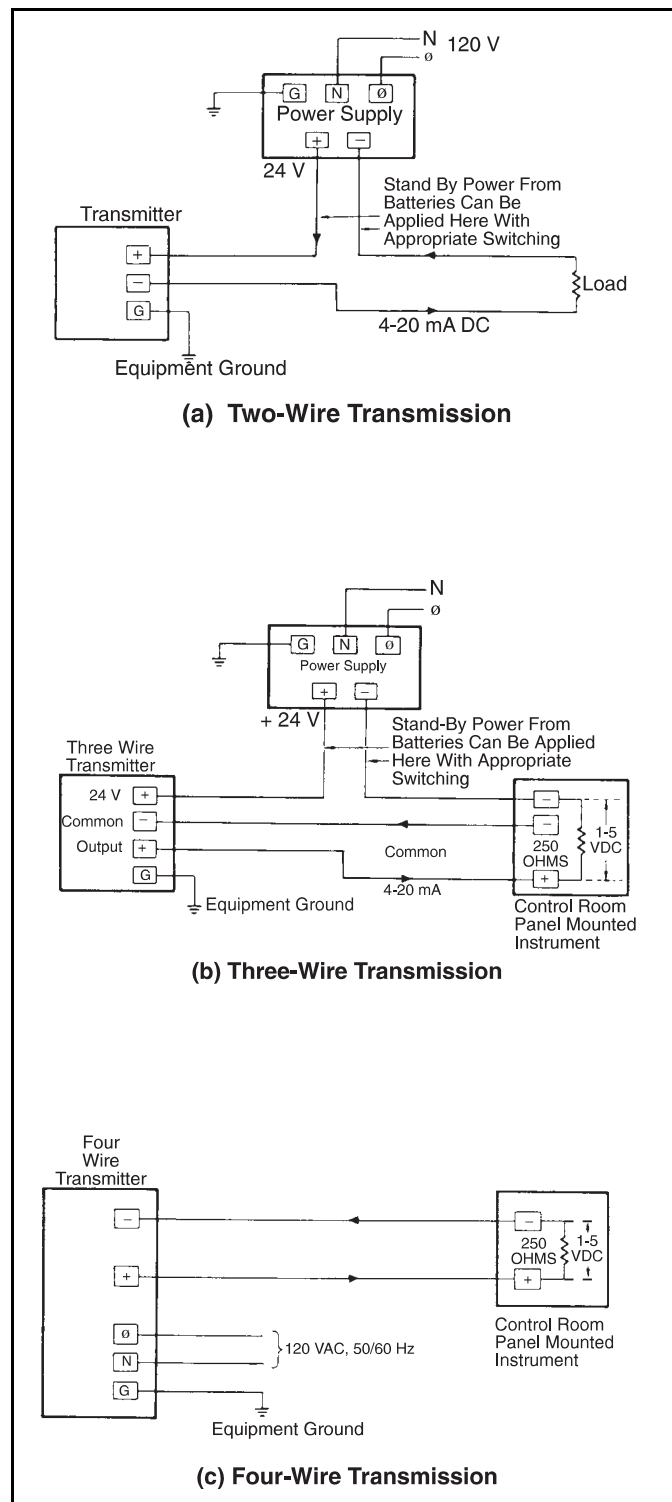
Connection Methods — Of great concern to the instrument engineer is the method by which electronic transmitters are connected in the instrumentation system. The “two-wire,” “three-wire,” and “four-wire” classifications are often used to describe the method of connection.

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



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

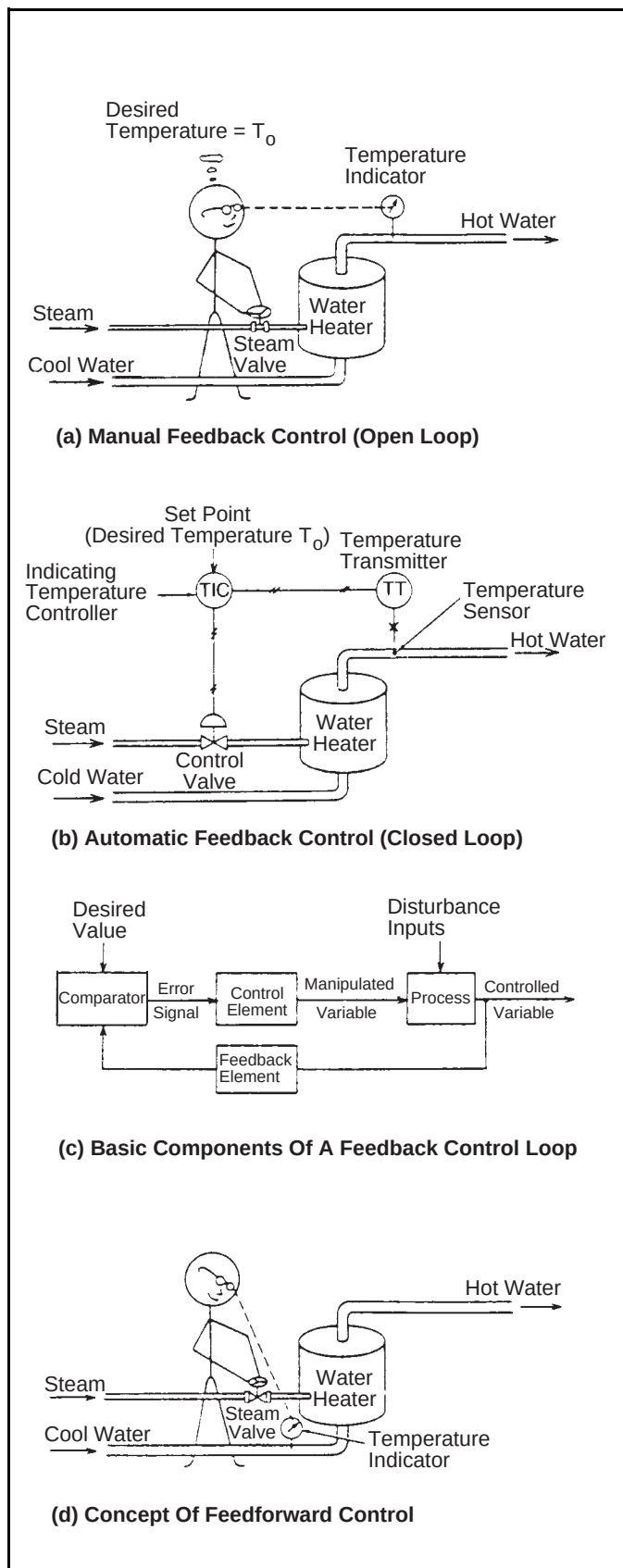
A control circuit is commonly referred to as a “loop.” A control loop may be classified as either “open” or “closed” depending upon whether the control adjustments are manual settings (open loop) or automatically determined by some type of feedback controller (closed-loop).

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.

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 “dead-band” 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

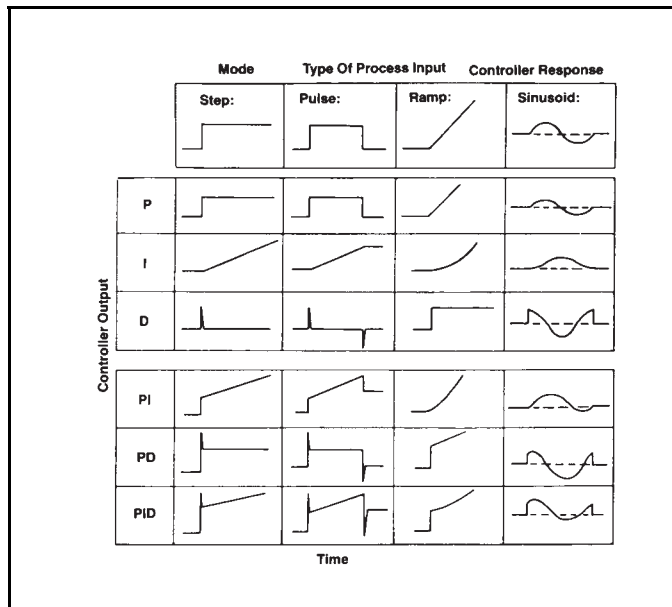
The error signal term ($PV - SP$) indicates “reverse” controller action (the controller output decreases when the measurement increases). Most controllers may be switched from one type of action to the other, and the choice depends upon the action of the final control element and the direction that the 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.

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:

$$CO = K_p (PV - SP) + MO \quad \text{Eq 4-1}$$

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:

$$\text{Proportional Band (PB \%)} = 100/\text{Gain} \quad \text{Eq 4-2}$$

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:

$$CO = K_p (PV - SP) + K_i \int (PV - SP) dt + MO \quad \text{Eq 4-3}$$

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:

$$CO = K_p [(PV - SP) + 1/T_i \int (PV - SP) dt] + MO \quad \text{Eq 4-4}$$

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:

$$CO = K_p (PV - SP) + K_d [d(PV - SP)/dt] + MO \quad \text{Eq 4-5}$$

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:

$$CO = K_p (PV - SP) + K_i \int (PV - SP) dt + K_d [d(PV - SP)/dt] + MO \quad \text{Eq 4-6}$$

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:

$$CO = K_p [(PV - SP) + 1/T_i \int (PV - SP) dt + T_d [d(PV - SP)/dt]] + MO \quad \text{Eq 4-7}$$

An alternate form used in many digital computer systems uses a normalization constant which “lumps” the units conversion factors in K_n :

$$\text{CO} = K_n [K_p(PV - SP) - 1/T_i \int (PV - SP) dt + T_d [d(PV - SP)/dt]] + MO \quad \text{Eq 4-8}$$

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.

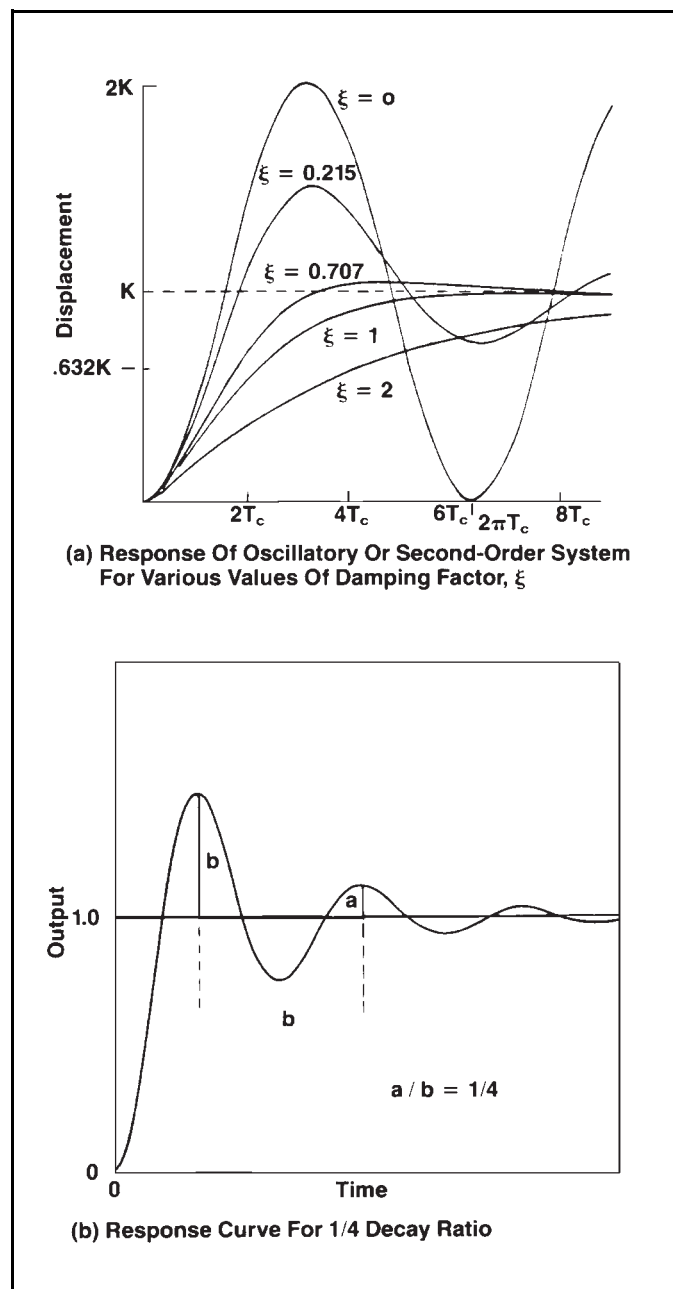
Ziegler-Nichols Method¹

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



2. Tune out all integral (reset) and derivative (rate) action. (set $T_i = \text{max.}$, set $T_d = \text{zero}$)
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).

FIG. 4-22

Typical Responses Obtained When Determining Ultimate Gain and Ultimate Period

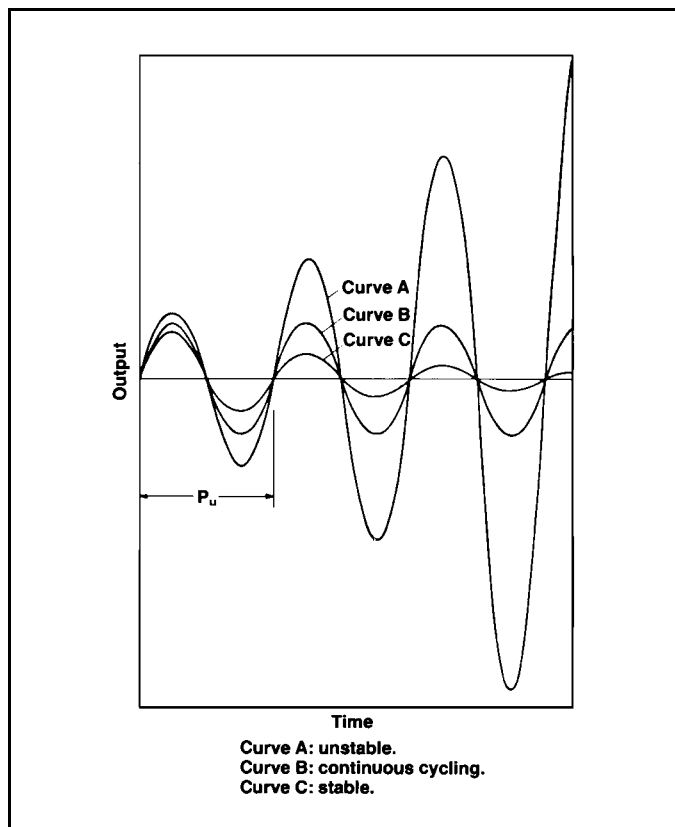


Fig. 4-23 gives relative controller gain, integral time, and derivative time for the various control mode combinations for quarter-decay response as related to ultimate controller gain setting, K_u , and ultimate period P_u . Gain settings are also shown in units of proportional band, PB.

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 :

$$K_p = 0.6 K_u = 0.6 (0.4 \text{ psi/}^\circ\text{F}) = 0.24 \text{ psi/}^\circ\text{F}$$

Integral time constant T_i :

$$T_i = P_u/2, T_i = 2/2 = 1.0 \text{ minute}$$

Derivative time constant T_d :

$$T_d = P_u/8, T_d = 2/8 = 0.25 \text{ minutes}$$

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 avail-

FIG. 4-23

Ziegler-Nichols Settings for 1/4 Decay Response¹

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

able.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.

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-

FIG. 4-25
Control Mode vs. Application

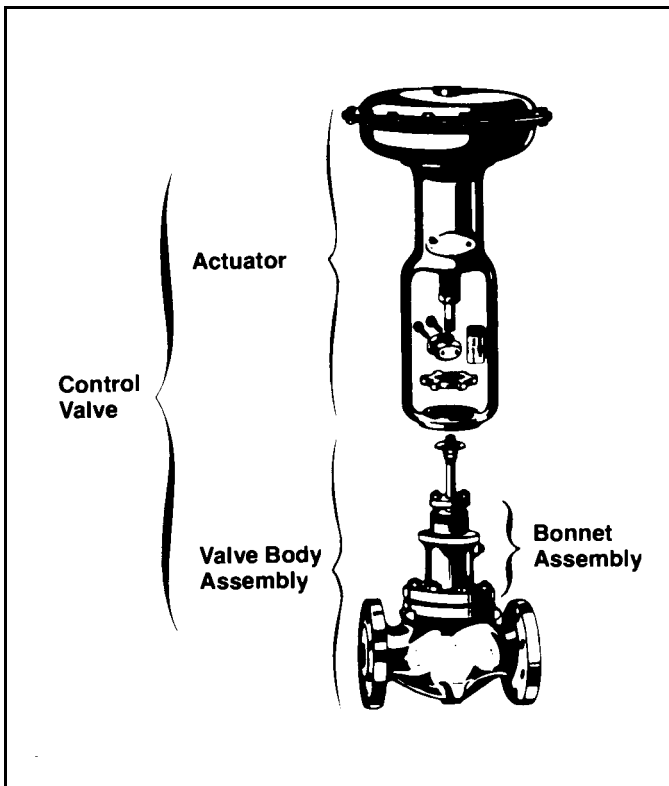
Control Mode	Process Reaction Rate	Load Changes		Applications
		Size	Speed	
On-off: Two-position with differential gap	Slow	Any	Any	Large-capacity temperature and level installations. Storage tanks, hot-water supply tanks, room heating, compressor suction scrubbers.
Floating, Single-speed with adjustable 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

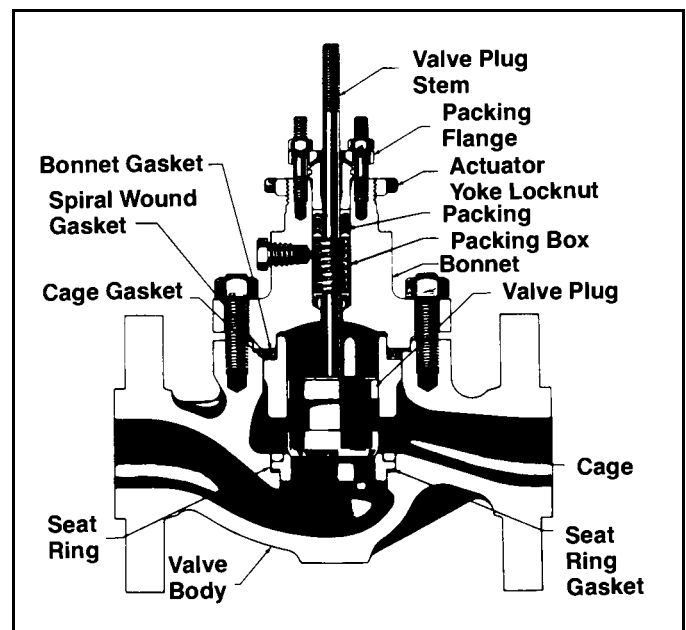


Control-Valve Bodies

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



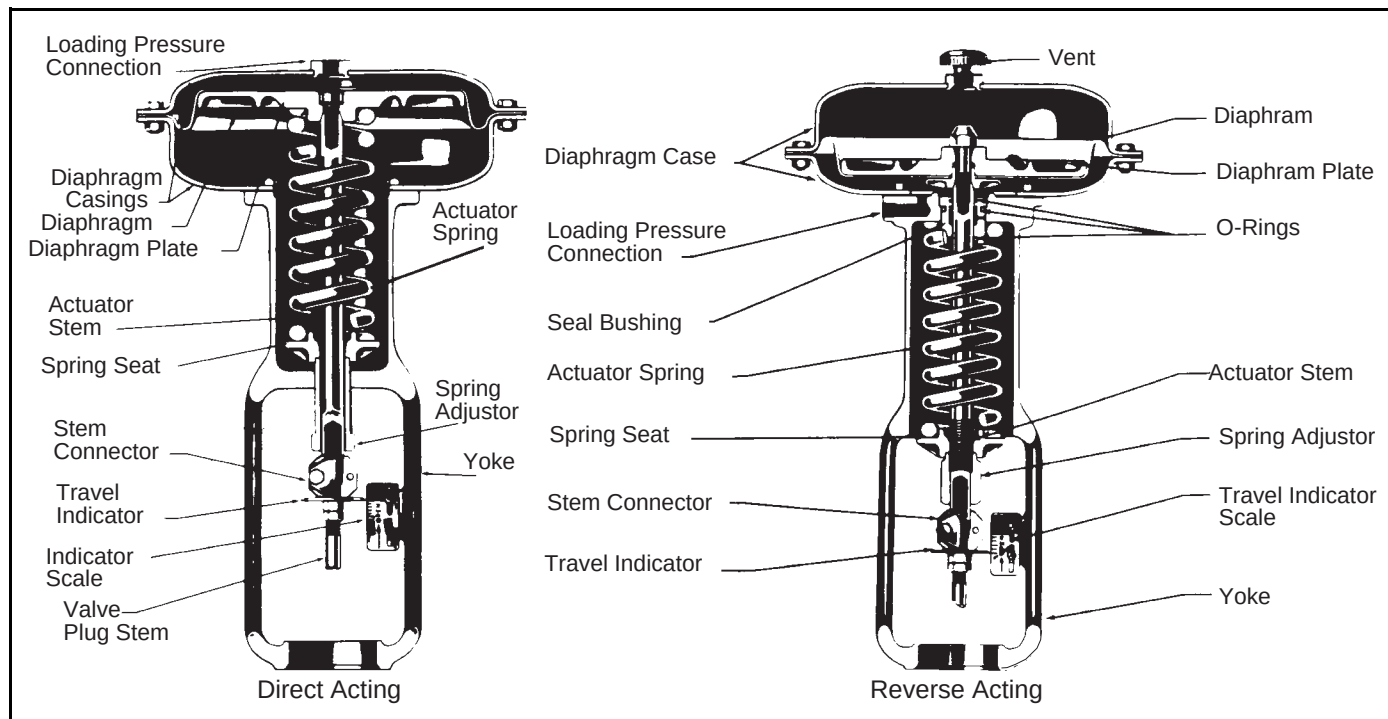
Control-Valve Actuators

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.

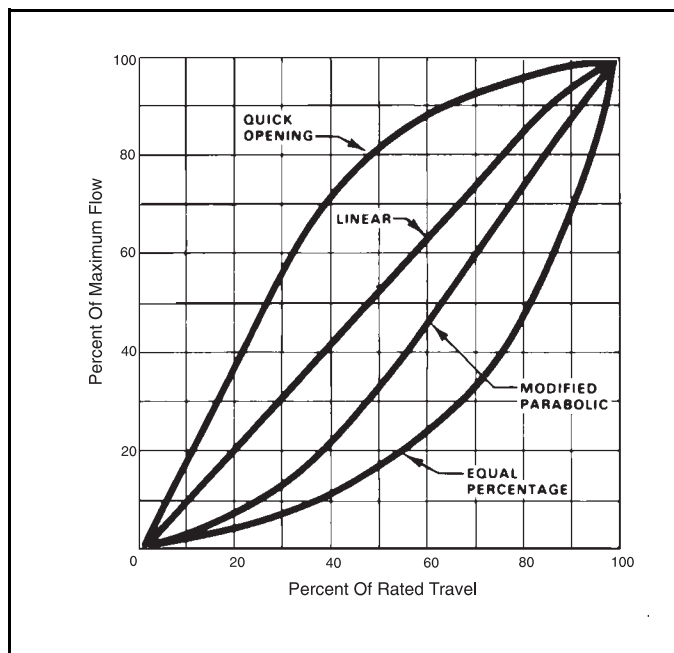
In the equal-percentage flow characteristic, equal increments of valve travel produce equal percentage changes in the existing flow. The change in flow rate is always proportional to the flow rate just before the change in position is made for a valve plug, disc, or ball position. When the valve plug, 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.

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.

$$Y = 1 - \frac{X}{3F_k X_c} \quad \text{Eq 4-9}$$

where,

$$F_k = \frac{k}{1.4} \quad \text{Eq 4-10}$$

Critical pressure drop, and thus critical flow, is realized when $X \geq F_k X_c$. 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.

$$Y = 1 - \frac{X}{3F_k X_c} = 1 - \frac{1}{3} = 0.67 \quad \text{Eq 4-11}$$

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 equa-

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	$w = N_8 F_p C_v P_1 Y \sqrt{\frac{X M}{T Z}}$
Volumetric Flow with Molecular Weight, M	$q = N_9 F_p C_v P_1 Y \sqrt{\frac{X}{M T Z}}$

FIG. 4-31

Numerical Constants for Gas and Vapor Flow Equations

Constant	Units Used in Equations						
	N	w	q^*	$p, \Delta p$	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.

tion. 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

with the listed C_v 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 manufacturer'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.							

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:

$$K_c = \frac{P_1 - P_v}{P_1 - P_2} \quad \text{Eq 4-12}$$

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-Newtonian 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

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

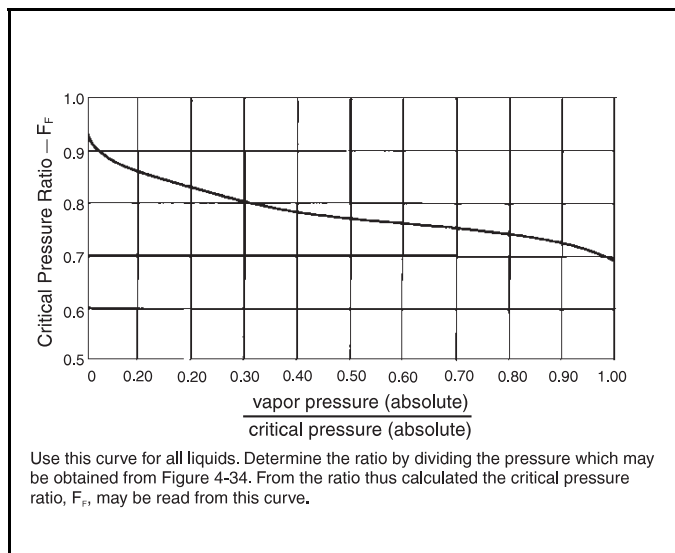


FIG. 4-34

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	$p, \Delta p$	d, D	γ_1	v
N_1	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.

- 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.

- 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

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 information 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-

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

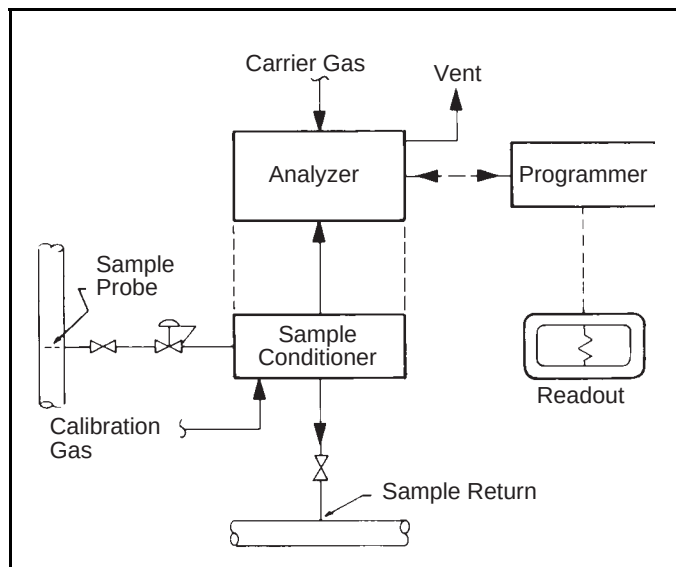


FIG. 4-40

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 ₂ S, 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:

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 re-

quirements 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	h_{L2} = enthalpy of saturated liquid at downstream pressure, Btu/lb
A = required discharge area of the valve, sq in. Use valve with the next larger standard orifice size/area	h_{G2} = enthalpy of vapor at downstream pressure, Btu/lb
A_B = bellows area, sq in.	H = height of vapor space of vessel, ft
A' = discharge area of the valve, sq in., for valve with next standard size larger than required discharge area	H_l = latent heat of the liquid exposed to fire, Btu/lb
A_c = total outside surface area of container, sq ft	H_S = flare stack height, ft
A_D = disk area, sq in.	I = radiation intensity at point X, Btu/(hr • ft ²)
A_N = nozzle seat area, sq in.	k = specific heat ratio, C_p/C_v (see Section 13)
A_P = piston area, sq in.	K_b = capacity correction factor due to back pressure
A_w = total wetted surface area of vessel, sq ft	K_c = combination correction for rupture disk = 0.9 = 1.0 no rupture disk installed
A_3 = vessel area exposed to fire, sq ft	K_d = coefficient of discharge, obtainable from the valve manufacturer
B = liquid expansion coefficient, 1/°F, at relieving temperature [or (Vol/Vol)/°F]	K_n = correction factor for Napier steam equation
C = drag coefficient	K_{sh} = correction factor due to the amount of superheat in the stream
C_p = specific heat at constant pressure, BTU/(lb • °F)	K_v = capacity correction factor due to viscosity
C_v = specific heat at constant volume, BTU/(lb • °F)	K_w = capacity correction factor due to back pressure (Fig. 5-14)
C_1 = coefficient determined by the ratio of specific heats of the gas or vapor at standard conditions	L = drum length, ft
d = flare tip diameter, inches	L/D = length to diameter ratio of pipe
D = particle diameter, ft	L_f = length of flame, ft
f = correction factor based on the ratio of specific heats	M = Mach number at pipe outlet
F = factor due to insulation (see Fig. 5-17)	MW = molecular weight of gas or vapor
F' = relief valve factor, dimensionless	$MABP$ = maximum allowable back pressure, psig
F_2 = coefficient for subcritical flow (Fig. 5-12)	NHV = net heating value of flare gas, Btu/lb
F_S = spring force, pounds	P = set pressure, psig
gpm = flow rate, gallons per minute at flowing temperature and pressure	P_{CF} = critical-flow pressure, psia
g = acceleration due to gravity, 32.2 ft/sec ²	P_n = normal operating gas pressure, psia
g_c = gravitational constant, 32.2 (ft • lbm)/(lbf • sec ²)	P_1 = upstream relieving pressure, psia. This is the set pressure plus the allowable overpressure plus the atmospheric pressure
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	P_2 = downstream pressure at the valve outlet, psia
h_{L1} = enthalpy of saturated liquid at upstream pressure, Btu/lb	P_b = back pressure, psig
	ΔP = pressure drop, psi
	ΔP_w = pressure drop, in. of water
	Q = heat input, Btu/hr

FIG. 5-1 (Cont'd)

Nomenclature

Q_a = minimum required capacity expressed as cfm of air at standard conditions	V'_w = wind velocity, mph
Q_r = heat released, Btu/hr	W = flow, lb/hr
Q_v = flow through valve, scfm	W_{hc} = hydrocarbon flow, lb/hr
r = ratio of downstream pressure to upstream pressure, P_2/P_1	W_{stm} = steam flow, lb/hr
R = distance from flame center to point X, ft	W_f = flare gas flow rate, lb/hr
Re = Reynolds number (dimensionless)	W_r = vapor rate to be relieved by the relief valve, lb/hr
R_o = universal gas constant (10.73) $\frac{\text{psia} \cdot \text{ft}^3}{\text{lb mole} \cdot ^\circ\text{R}}$	x_i = weight fraction of component i in total stream
S = specific heat, Btu/(lb $\cdot$ $^\circ\text{F}$)	X = distance from the base of the stack to another point at the same elevation, ft
t = temperature, $^\circ\text{F}$	X_c = see Fig. 5-21
T = absolute temperature of the inlet vapor, $^\circ\text{R}$	Y_c = see Fig. 5-21
T_n = normal operating gas temperature, $^\circ\text{R}$	Z = compressibility factor at flowing conditions
T_1 = gas temperature, $^\circ\text{R}$, at the upstream pressure	
T_w = vessel wall temperature, $^\circ\text{R}$	
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)	

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 shut-down 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.

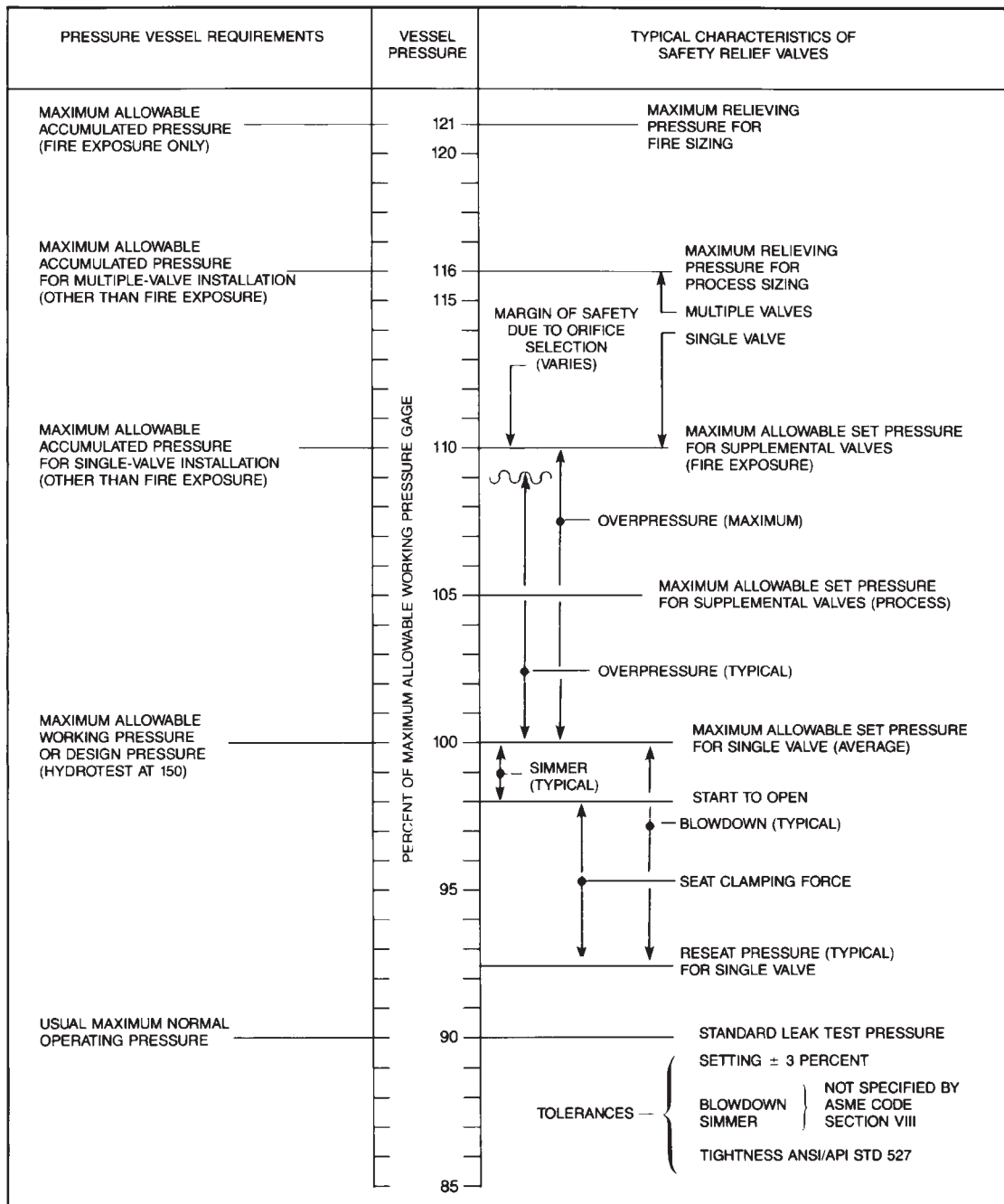
Fire Exposure

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.

FIG. 5-2
Characteristics of Safety Relief Valves for Vessel Protection¹



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

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 vac-

uum 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 Code² 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 B31.3³ or ANSI B31.8⁴, 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.

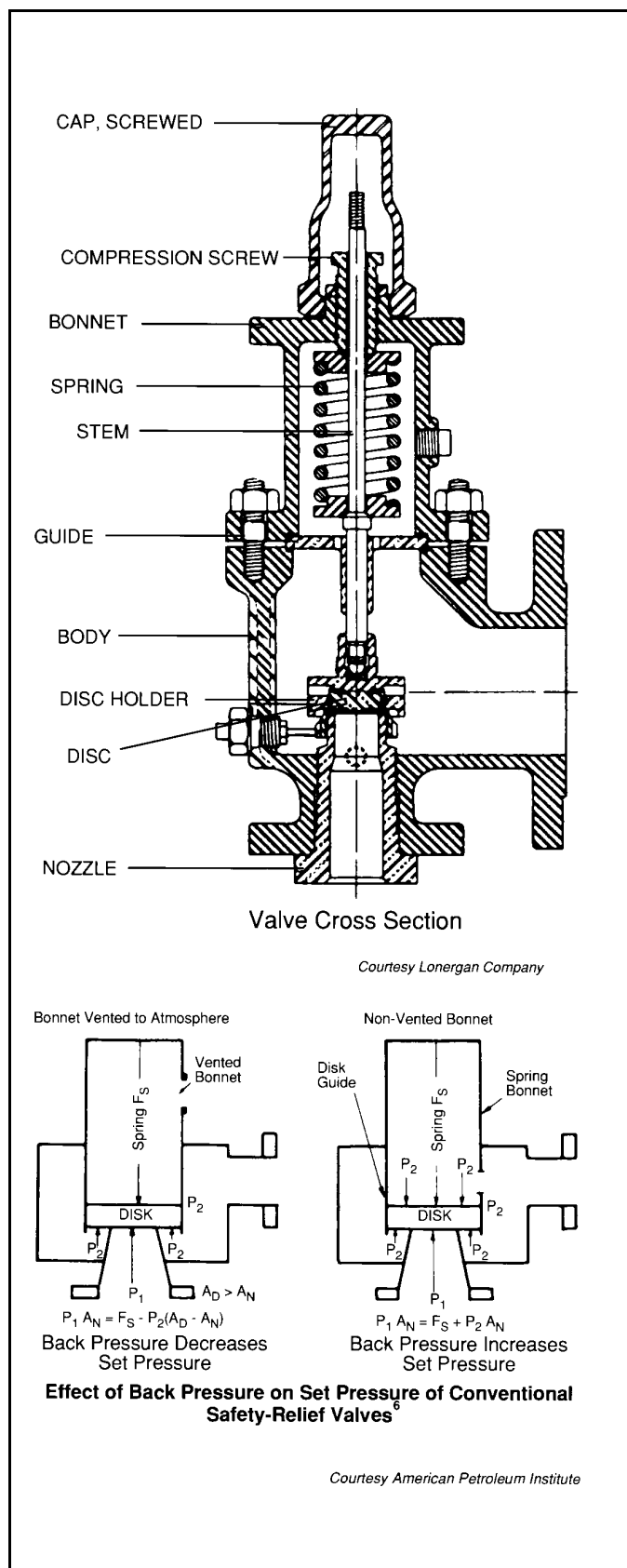
Balanced Relief Valves

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

FIG. 5-3

Conventional Safety-Relief Valve



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.

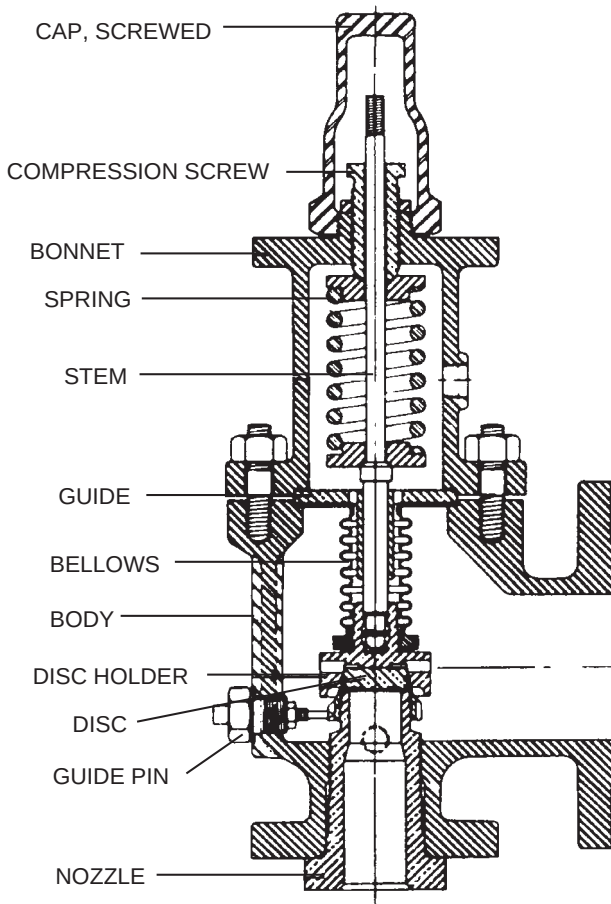
Resilient Seat Relief Valves

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

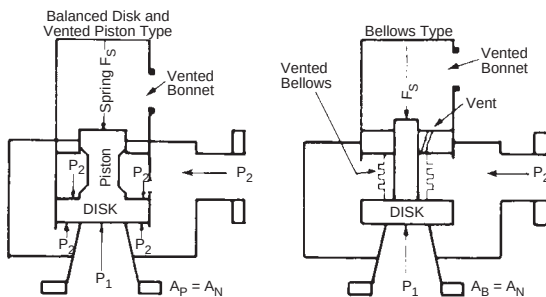
A rupture disk consists of a thin diaphragm held between flanges. The disk is designed to rupture and relieve pressure within tolerances established by ASME Code. Rupture disks can be used in gas processing plants, upstream of relief valves, to reduce minor leakage and valve deterioration. In these in-

FIG. 5-4
Balanced Safety-Relief Valve



Bellows Valve Cross Section

Courtesy Lonergan Company



Set press. $P_1 = \frac{F_S}{A_N} = \frac{\text{Spring Force}}{\text{Nozzle Seat Area}}$
Note: Back Pressure has very little effect on set pressure

Effect Of Back Pressure on Set Pressure of Balanced Safety-Relief Valves

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:

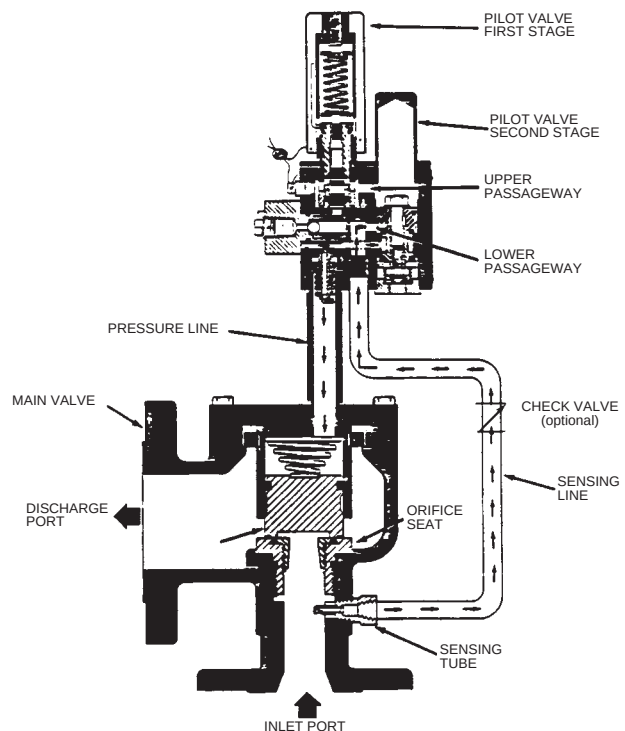
- (a) Rapid rates of pressure rise. A pressure relief valve system does not react fast enough or cannot be made large enough to prevent overpressure, e.g., an exchanger ruptured tube case or a runaway reaction in a vessel.
- (b) Large relieving area required. Because of extremely high flow rates and/or 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.

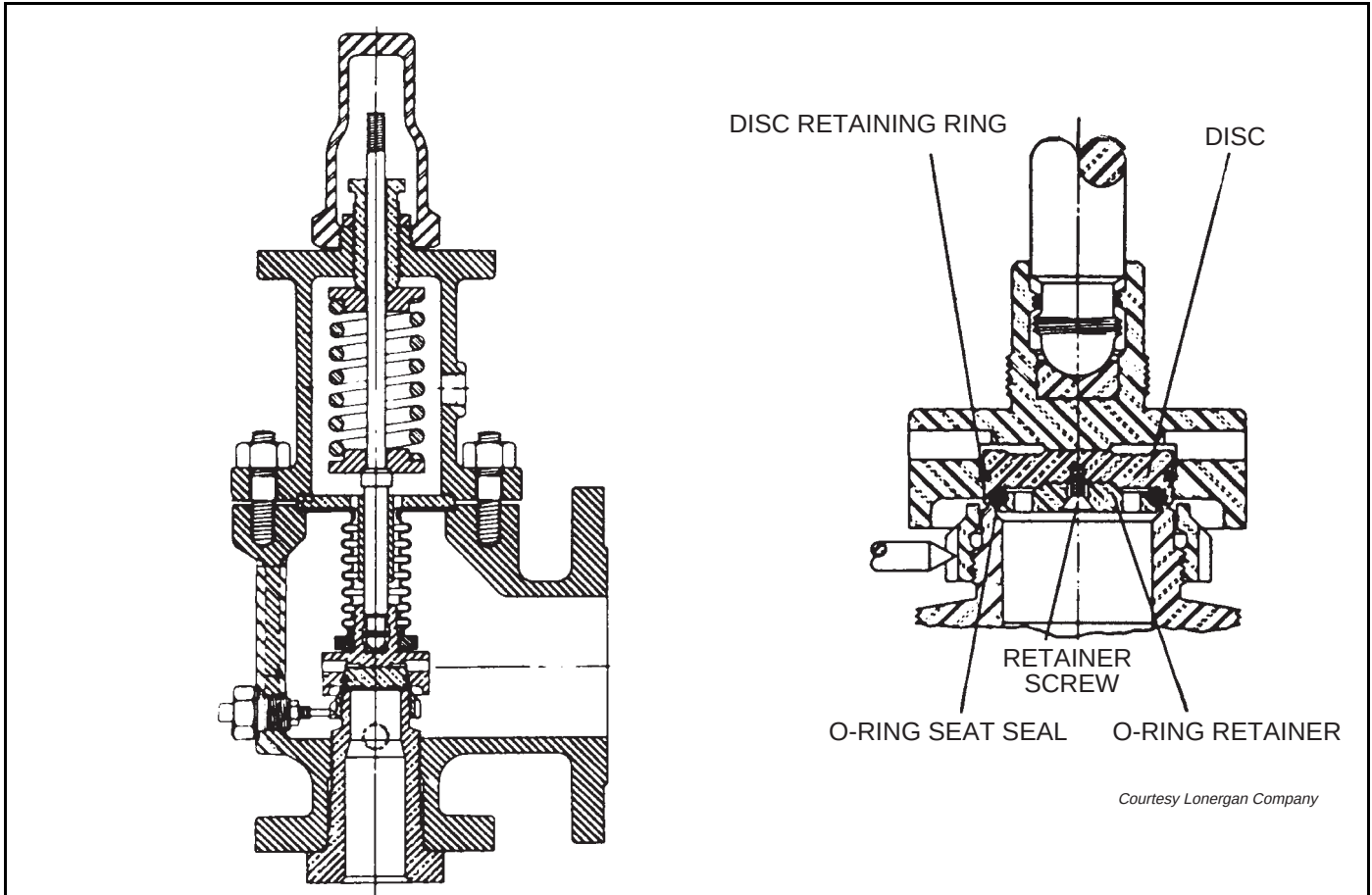
FIG. 5-5

Pilot Operated Relief Valve



Courtesy GPE Controls

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:

$$A = \frac{W \sqrt{(T_1) (Z)}}{(C_1) (K_d) (P_1) (K_b) (K_c) \sqrt{MW}} \quad \text{Eq 5-1}$$

$$A = \frac{Q_v \sqrt{(T_1) (MW) (Z)}}{(6.32) (C_1) (K_d) (P_1) (K_b) (K_c)} \quad \text{Eq 5-2}$$

$$C_1 = 520 \sqrt{k \left(\frac{2}{k+1} \right)^{\frac{k+1}{k-1}}} \quad \text{Eq 5-3}$$

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

FIG. 5-7
Relief Valve Designations

		Orifice Area cm ²	Orifice Area (in. ²)											
Standard Orifice Designation	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, P₂, in excess of the critical-flow pressure, P_{CF}, the relief valve orifice area can be calculated from

$$A = \frac{W \sqrt{Z T_1}}{(735) (F_2) (K) \sqrt{MW (P_1) (P_1 - P_2)}} \quad \text{Eq 5-4}$$

F₂ is taken from Fig. 5-12.

$$P_{CF} = P_1 \left(\frac{2}{k+1} \right)^{\frac{k}{k-1}} \quad \text{Eq 5-5}$$

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)

$$A = \frac{W}{(51.5) (P_1) (K_{sh}) (K_d) (K_c) (K_n) (K_b)} \quad \text{Eq 5-6}$$

$$K_n = \frac{0.1906 P_1 - 1000}{0.2292 P_1 - 1061} \quad \text{Eq 5-7}$$

where P₁ > 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 C₁⁶

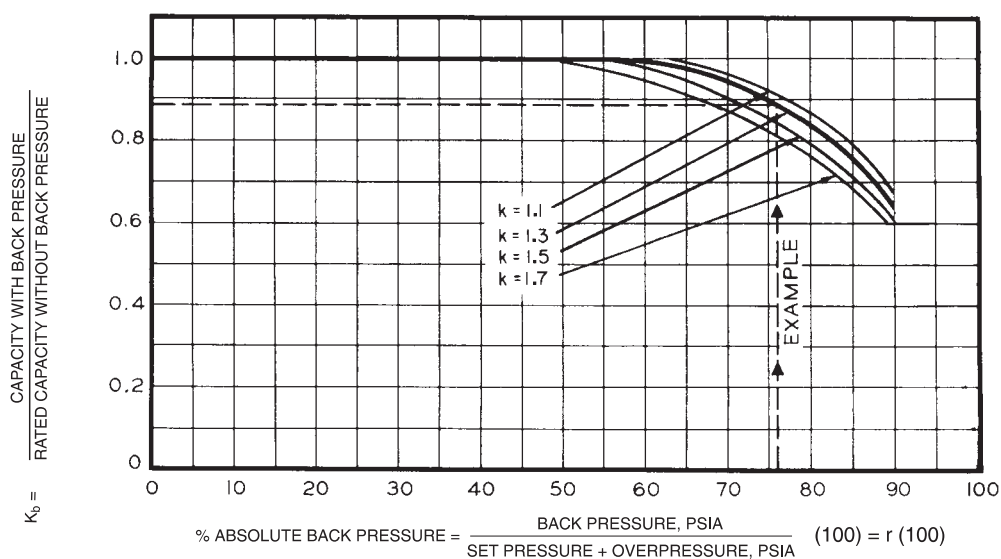
k	C ₁
0.4	216.9274
0.5	238.8252
0.6	257.7858
0.7	274.5192
0.8	289.494
0.9	303.0392
1.0	315.37*
1.1	326.7473
1.2	337.2362
1.3	346.9764
1.4	356.0604
1.5	364.5641
1.6	372.5513
1.7	380.0755
1.8	387.1823
1.9	393.9112
2.0	400.2962
2.1	406.3669
2.2	412.1494

*Interpolated values, since C₁ becomes indeterminate as k approaches 1.00
Note: Calculated from Eq. 5-3.

FIG. 5-9
Values of C₁ 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



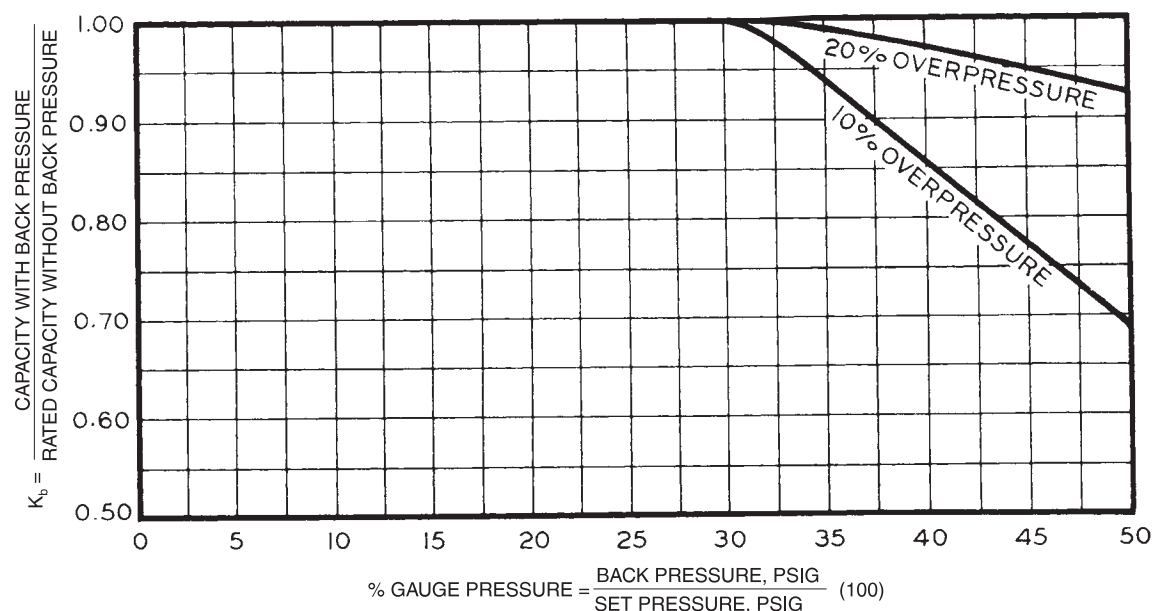
Example:

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	= $\left(\frac{(70 + 10 + 14.7)}{(100 + 10 + 14.7)} \right) (100) = 76 \text{ percent}$
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.

Courtesy American Petroleum Institute

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 .

Courtesy American Petroleum Institute

only when the manufacturer has approved the specific application.

$$A = \frac{(\text{gpm}) \sqrt{G}}{(38) (K_d) (K_c) (K_w) (K_v) \sqrt{(P_1 - P_b)}} \quad \text{Eq 5-8}$$

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:

$$Re = \frac{(\text{gpm}) (2,800) (G)}{\mu \sqrt{A'}} \quad \text{Eq 5-9}$$

$$Re = \frac{(12,700) (\text{gpm})}{\mu_s \sqrt{A'}} \quad \text{Eq 5-10}$$

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.

$$\text{gpm} = \frac{(B) (Q)}{500 (G) (S)}$$

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.

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.

$$\text{Weight Percent Flashing} = \left[\frac{h_{L1} - h_{L2}}{h_{G2} - h_{L2}} \right] (100) \quad \text{Eq 5-12}$$

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 Fauske⁷ 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 521¹, API Standard 2510⁸, NFPA 58⁹, 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.

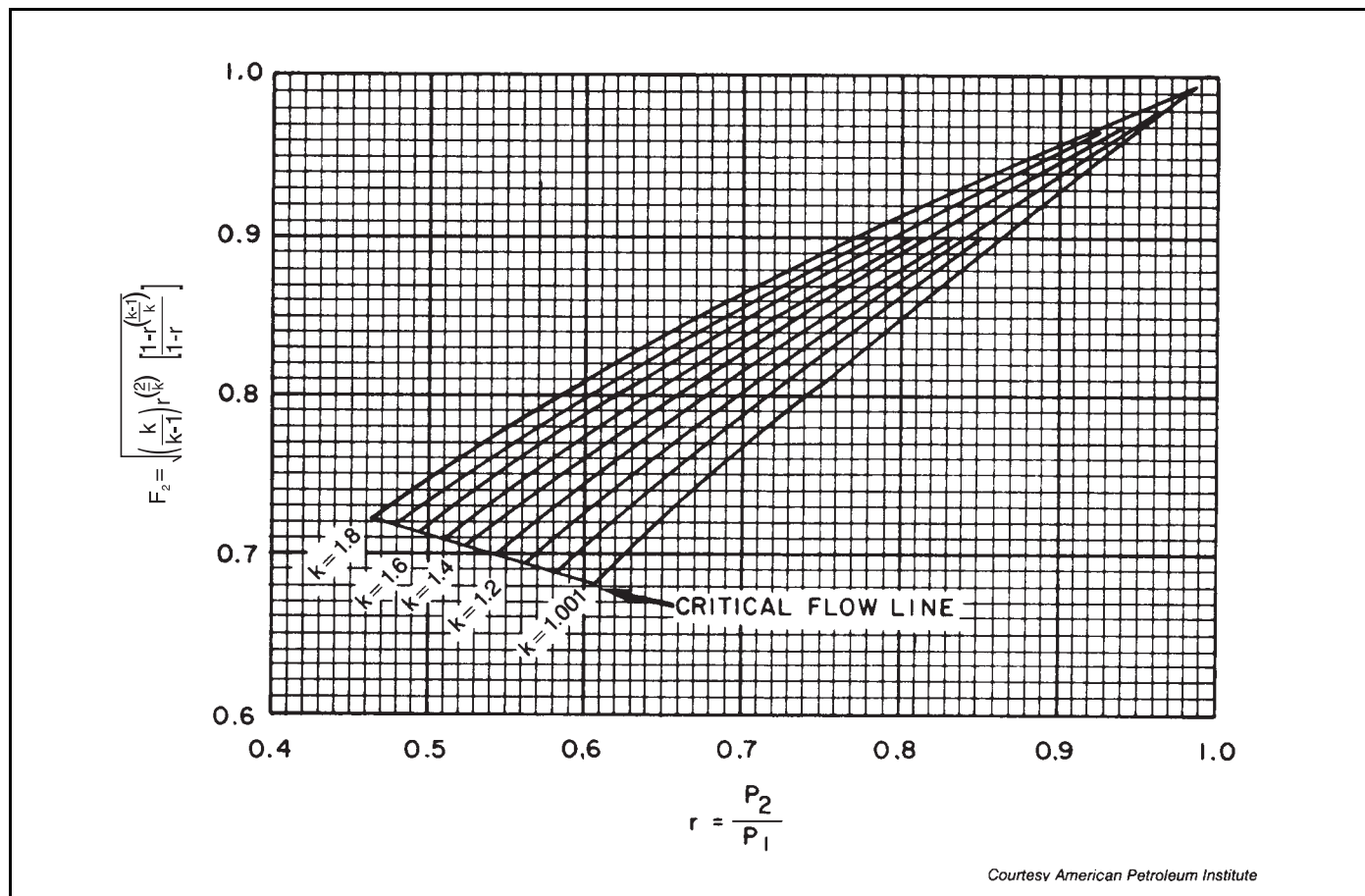
NFPA 58⁹ applies to LP-gas (propane and butane) systems. It presents the relief capacity requirements for vessels in terms of the rate of discharge of air, Q_a , and area of the vessel according to the equation:

$$Q_a = (53.632) (A_v)^{0.82} \quad \text{Eq 5-13}$$

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 Flow⁶



Courtesy American Petroleum Institute

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.

$$Q = (21,000) (F) (A_w)^{0.82} \quad \text{Eq 5-14}$$

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:

$$W = Q / H_l \quad \text{Eq 5-15}$$

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 520⁶. 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 fire²:

$$A = \frac{(F') (A_3)}{\sqrt{P_1}} \quad \text{Eq 5-16}$$

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

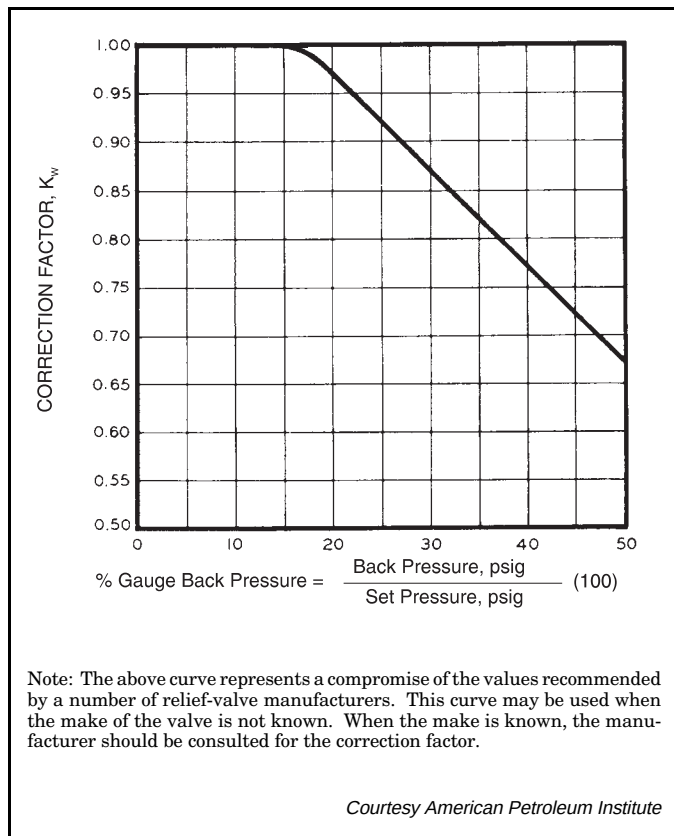
FIG. 5-13
Superheat Correction Factors for Safety Valves in Steam Service⁶

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

Courtesy American Petroleum Institute

FIG. 5-14

**Variable or Constant Back-Pressure Sizing Factor K_w on
Balanced Bellows Safety-Relief Valves (Liquids Only)⁶**



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

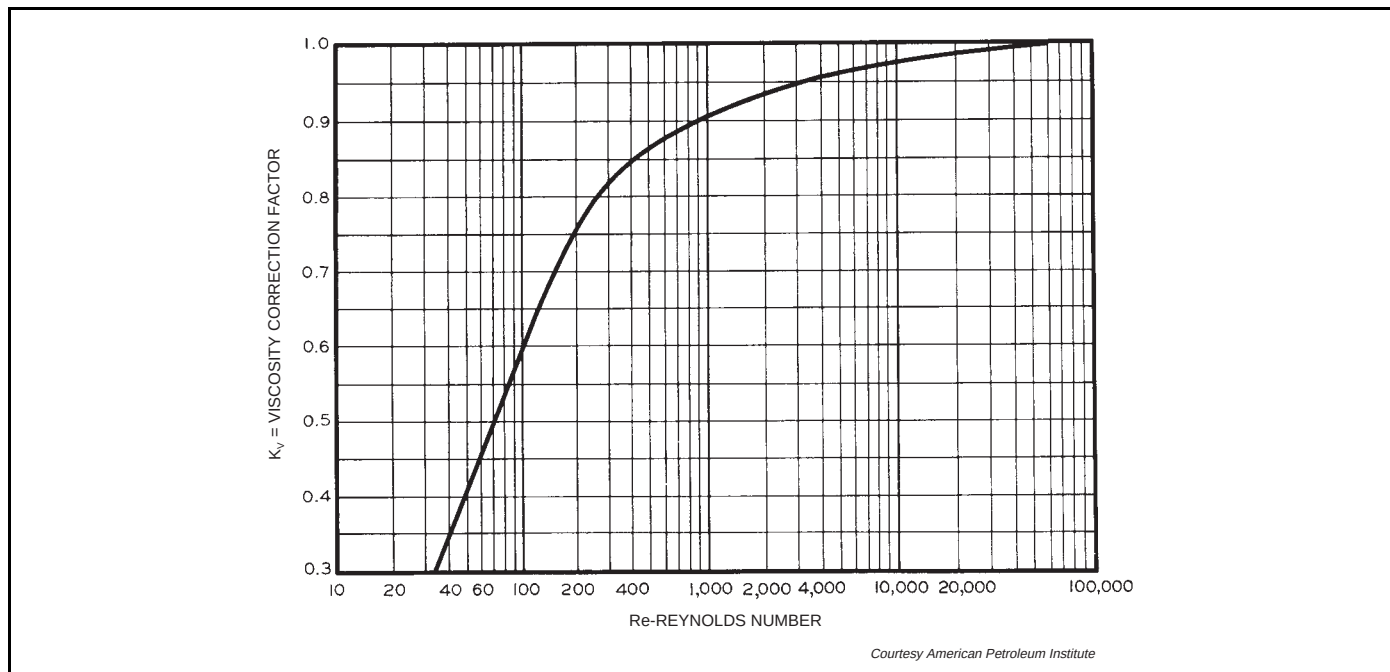


FIG. 5-16
Environmental Factors⁸

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.

Courtesy American Petroleum Institute

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 reseal 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

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

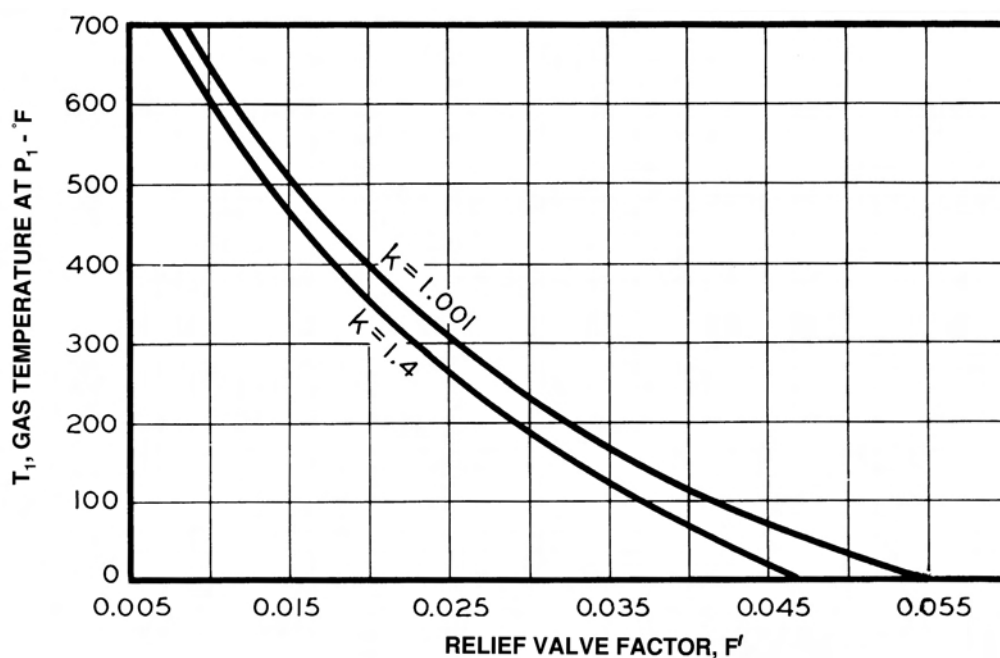
The maximum relief load anticipated for each relief system may be determined by tabulating the relief loads discharging to each header system at the conditions expected to exist in the system. For further discussion, see Section 5.2 of API RP 521.

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 Fires⁶



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.

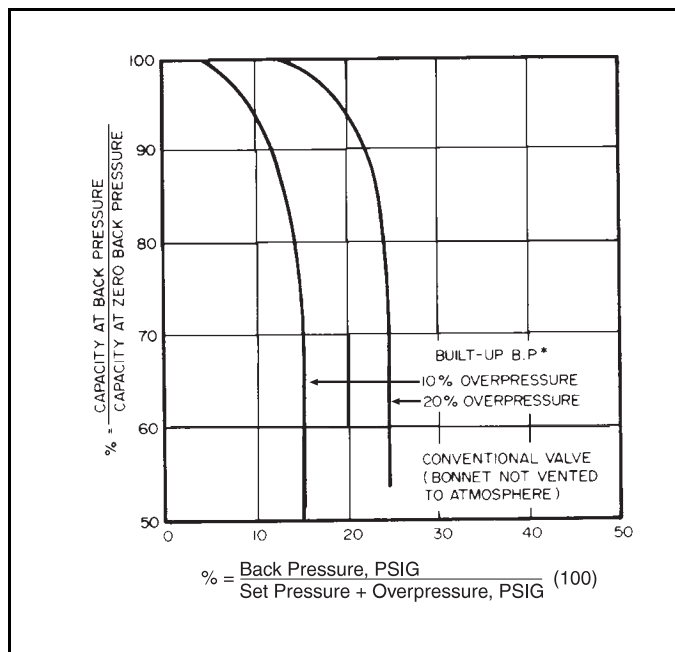
3. These curves conform to the relationship
$$F' = \left(\frac{0.1406}{C_1 K_d} \right) \left(\frac{\Delta T^{1.25}}{T_1^{0.6506}} \right)$$

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).

FIG. 5-18

Typical Effects of Variable Back Pressure on Capacity of Conventional Safety-Relief Valves⁶



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).

$$MW = \Sigma W_i / \Sigma (W / MW)_i \quad \text{Eq 5-17}$$

$$T = \Sigma W_i T_i / \Sigma W_i \quad \text{Eq 5-18}$$

$$\mu = \Sigma x_i \mu_i (MW)_i^{0.5} / \Sigma x_i (MW)_i^{0.5} \quad \text{Eq 5-19}$$

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:

$$U_d = (1.15) \sqrt{\frac{(g) (D) (\rho_L - \rho_v)}{(\rho_v) (C)}} \quad \text{Eq 5-20}$$

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 carry-over, noise, and ground level concentration of toxic material.

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 dan-

gerous 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:

$$I = \frac{(W_f) (NHV) (\epsilon)}{4 \pi (R^2)} \quad \text{Eq 5-21}$$

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 Coefficient¹

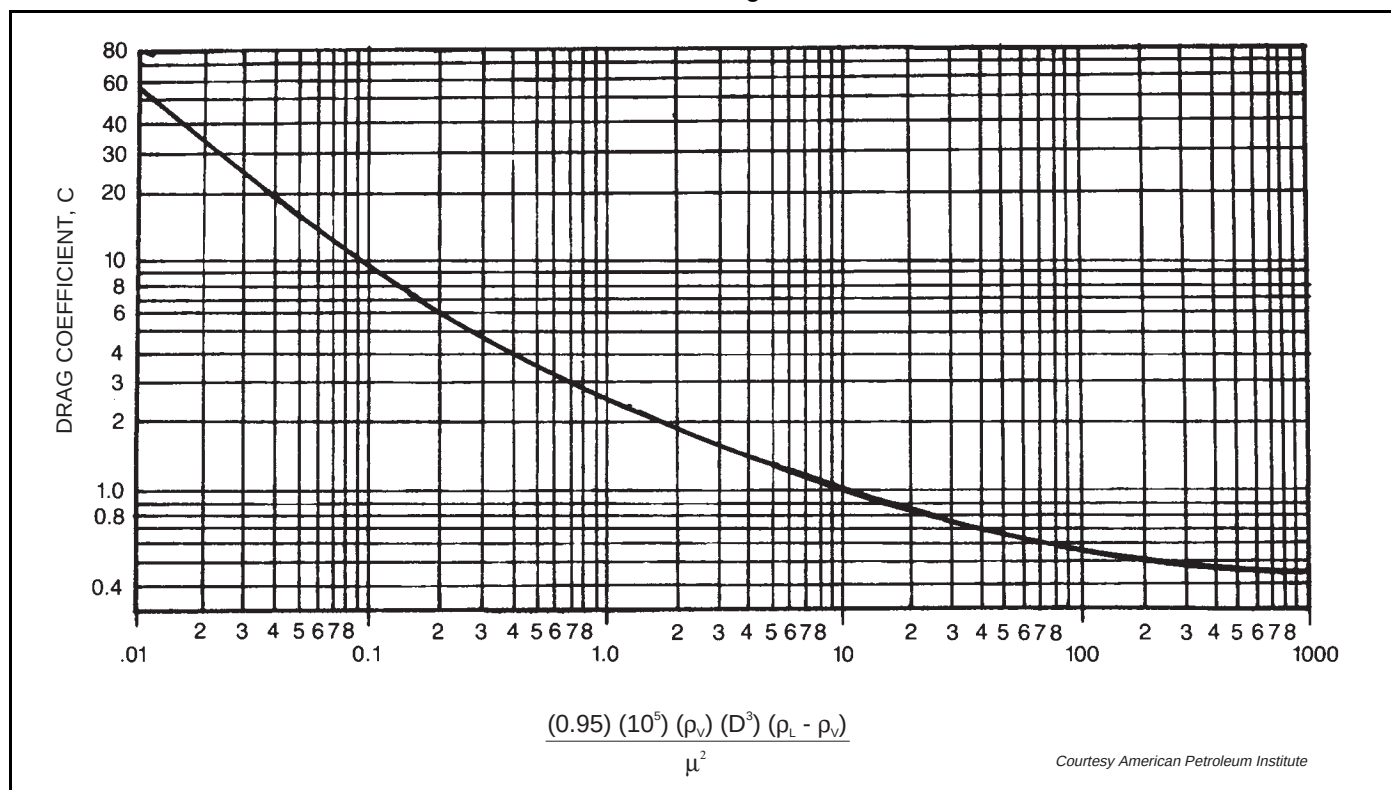


FIG. 5-20

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.

$$L_f = (10) (d) \sqrt{\frac{\Delta P_w}{55}} \quad \text{Eq 5-22}$$

or from API RP 521,

$$L_f = 3.94 [(Q_r) (10^{-6})]^{0.474} \quad \text{Eq 5-23}$$

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:

$$\Delta P_w = \frac{(27.7) \rho V^2}{(2 g_c) (144)} = \frac{\rho V^2}{344.8} \quad \text{Eq 5-24}$$

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.

$$d = \left(\sqrt{\frac{1.702 \cdot 10^{-5} \cdot W}{P_2 \cdot M} \cdot \left(\frac{Z \cdot T}{k \cdot MW} \right)^{0.5}} \right) \cdot 12 \quad \text{Eq 5-25}$$

Sonic velocity of a gas is given by:

$$a = 223 \sqrt{k \frac{T}{MW}} \quad \text{Eq 5-26}$$

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.

$$\theta = \tan^{-1} \left(\frac{V_w}{V_{ex}} \right) \quad \text{Eq 5-27}$$

$$V_{ex} = 550 \sqrt{\frac{\Delta P}{55}} \quad \text{Eq 5-28}$$

Note: API gives a greater lean angle

The coordinates of the flame center with respect to the tip are:

$$X_c = (L_f / 3) (\sin \theta) \quad \text{Eq 5-29}$$

$$Y_c = (L_f / 3) (\cos \theta) \quad \text{Eq 5-30}$$

The distance from any point on the ground level to the center of the flame is:

$$R = \sqrt{(X - X_c)^2 + (H_s + Y_c)^2} \quad \text{Eq 5-31}$$

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).

$$R^2 = (H_s + Y_c)^2 \quad \text{Eq 5-32}$$

$$R = (H_s + Y_c) \quad \text{Eq 5-33}$$

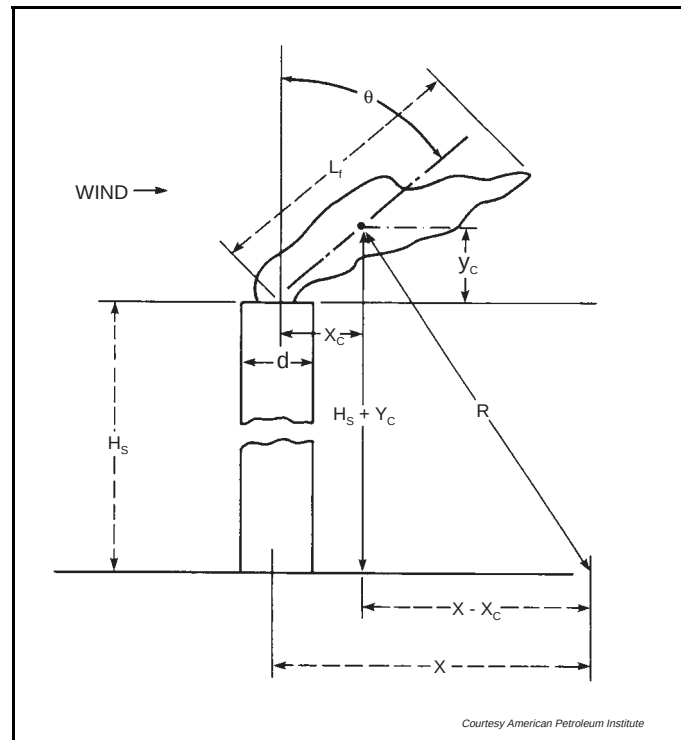
$$H_s = (R - Y_c) \quad \text{Eq 5-34}$$

$$H_s = R - [(L_f / 3) (\cos \theta)] \quad \text{Eq 5-35}$$

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



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 • sq ft).

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.

$$W_{\text{stm}} = W_{\text{hc}} \left[0.68 - \left(\frac{10.8}{\text{MW}} \right) \right] \quad \text{Eq 5-36}$$

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

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.

6. API RP 520 — Recommended Practice for the Design (Part I) and Installation (Part II) of Pressure Relieving Systems in Refineries, American Petroleum Institute, 1220 L Street, NW, Washington, DC 20005.
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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	x_i = mole fraction of component i in the liquid phase
A = surface area, ft^2	X = insulation thickness, in.
b = ellipse minor radius, ft	y_i = mole fraction of component i in the vapor phase
$B_{\max}$ = vapor pressure of liquid at maximum surface temperature, psia	Z = compressibility factor
$B_{\min}$ = vapor pressure of liquid at minimum surface temperature, psia	Greek
D = cylinder diameter, ft	α = radians
$f(Z_c)$ = cylinder partial volume factor, dimensionless	Δ = absolute internal tank pressure at which vacuum vent opens, psia
$f(Z_e)$ = head partial volume factor, dimensionless	π = 3.14159...
H = correction factor for horizontal surfaces	ϕ = required storage pressure, psia
H_n = depth of liquid in cylinder, ft	Σ = summation
H_p = height of liquid, ft	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)
H_T = height, ft	$MAWP$ = 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.
k = thermal conductivity, $\text{Btu}/[(\text{hr} \cdot \text{sq ft} \cdot ^\circ\text{F})/\text{in.}]$	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)
K = equilibrium constant, y/x , dimensionless	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 .
K_l = head coefficient, dimensionless	TVP = true vapor pressure is the pressure at which the gas and liquid in a closed container are in equilibrium at a given temperature.
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, $\text{Btu}/\text{sq ft} \cdot \text{hr}$	
R = gas constant, $10.73 \text{ psia} \cdot \text{ft}^3/(\text{R} \cdot \text{lb mole})$	
R_l = cylinder radius, ft	
R_i = thermal resistance of insulation (X/k), $(\text{hr} \cdot \text{sq ft} \cdot ^\circ\text{F})/\text{Btu}$	
T = temperature, $^\circ\text{R}$	
T_a = ambient air temperature, $^\circ\text{F}$	
T_c = critical temperature, $^\circ\text{R}$ or $^\circ\text{F}$	
T_f = temperature drop through surface air film, $^\circ\text{F}$	
T_h = hot face temperature, $^\circ\text{F}$	
T_i = temperature drop through insulation, $^\circ\text{F}$	
T_m = mean temperature of insulation, $^\circ\text{F}$	
$T_{\max}$ = maximum average temperature, $^\circ\text{F}$	
$T_{\min}$ = minimum average temperature, $^\circ\text{F}$	
T_R = reduced temperature, dimensionless	
T_s = outside surface temperature, $^\circ\text{F}$	
V = volume, ft^3	
W = width, ft	

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

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.

$$\phi = B_{\max} + (\Delta - B_{\min}) \frac{(T_{\max} + 460)}{(T_{\min} + 460)} - P_a \quad \text{Eq 6-1}$$

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,

$$\phi = B_{\max} - P_a \quad \text{Eq 6-2}$$

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:

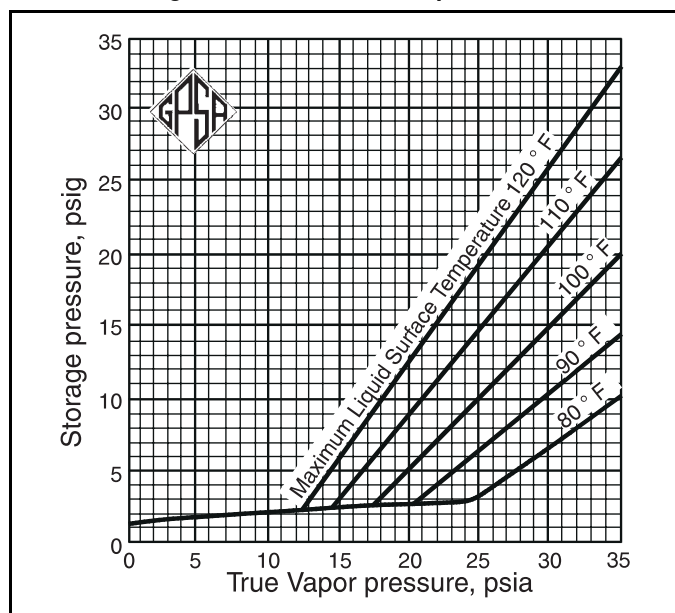
- Minimum liquid surface temperature is 10°F less than the maximum liquid surface temperature.
- Maximum vapor space temperature is 40°F greater than the maximum liquid surface temperature.
- Minimum vapor space temperature is 15°F less than the maximum liquid surface temperature.
- Stable ambient conditions (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 as-

FIG. 6-3

Storage Pressure vs. True Vapor Pressure



sumption that it is 10°F higher than the maximum temperature of the body of the liquid in a tank at that location.

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.

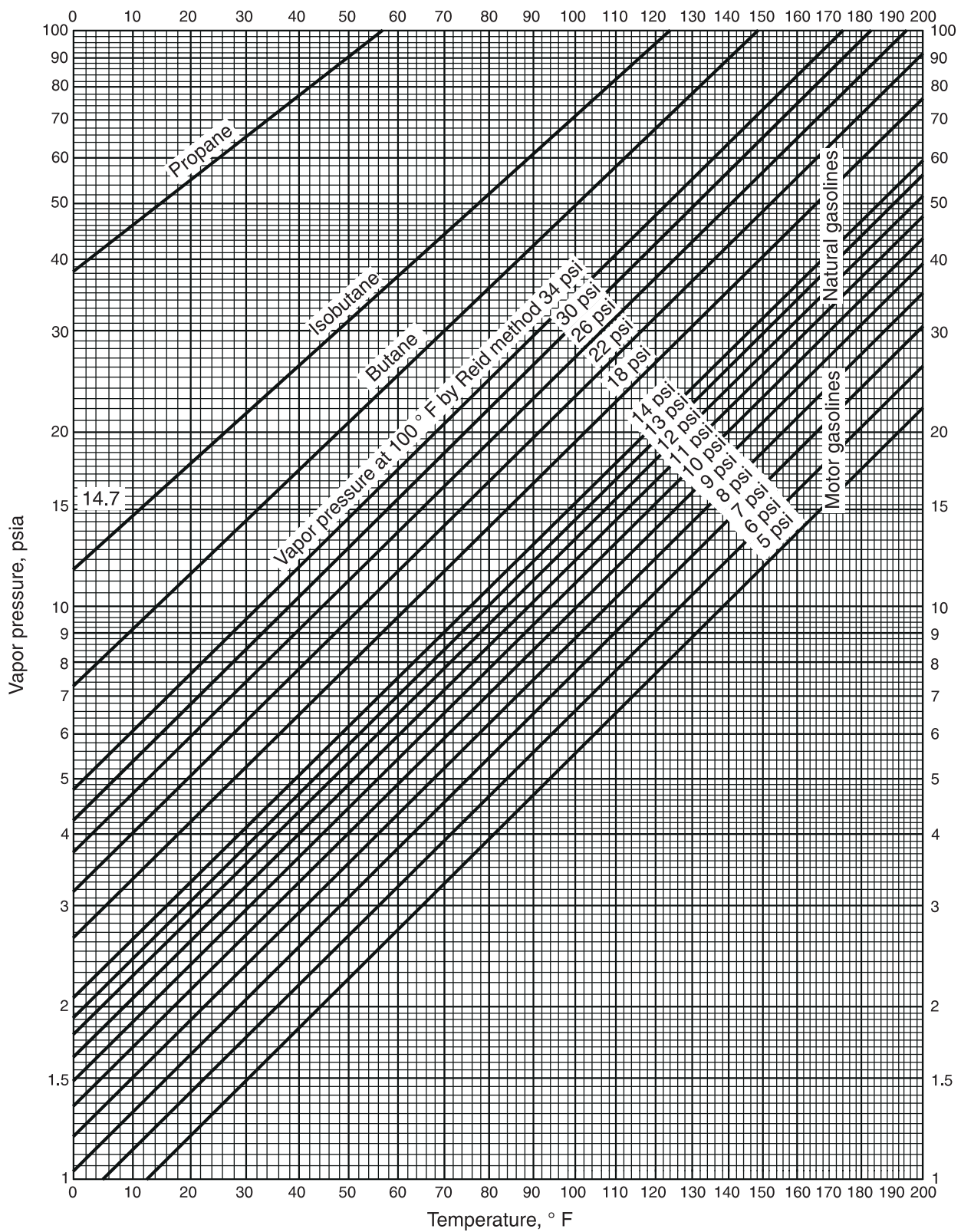
The graphical method of converting from RVP to TVP is an approximation and is generally more accurate for lighter components. Crude oils with very low RVPs could vary significantly from this graphical approach. This is due to the fact that during the Reid test the highest vapor pressure materials tend 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.

$$\text{TVP} = (1.07) (\text{RVP}) + 0.6 \quad \text{Eq 6-3}$$

Using this formula for the 12 psi RVP gasoline example would calculate a 13.4 psia TVP versus the 13.2 determined

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

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.

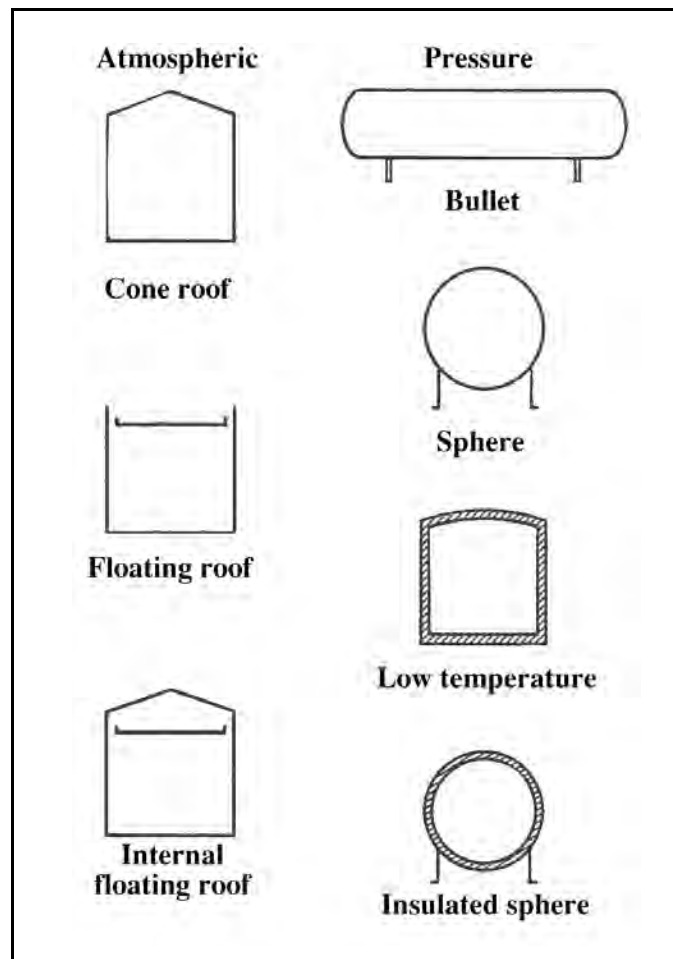
Both floating roofs and internal floating roofs are utilized to reduce vapor losses and aid in conservation of stored fluids.

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 requirements. 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.

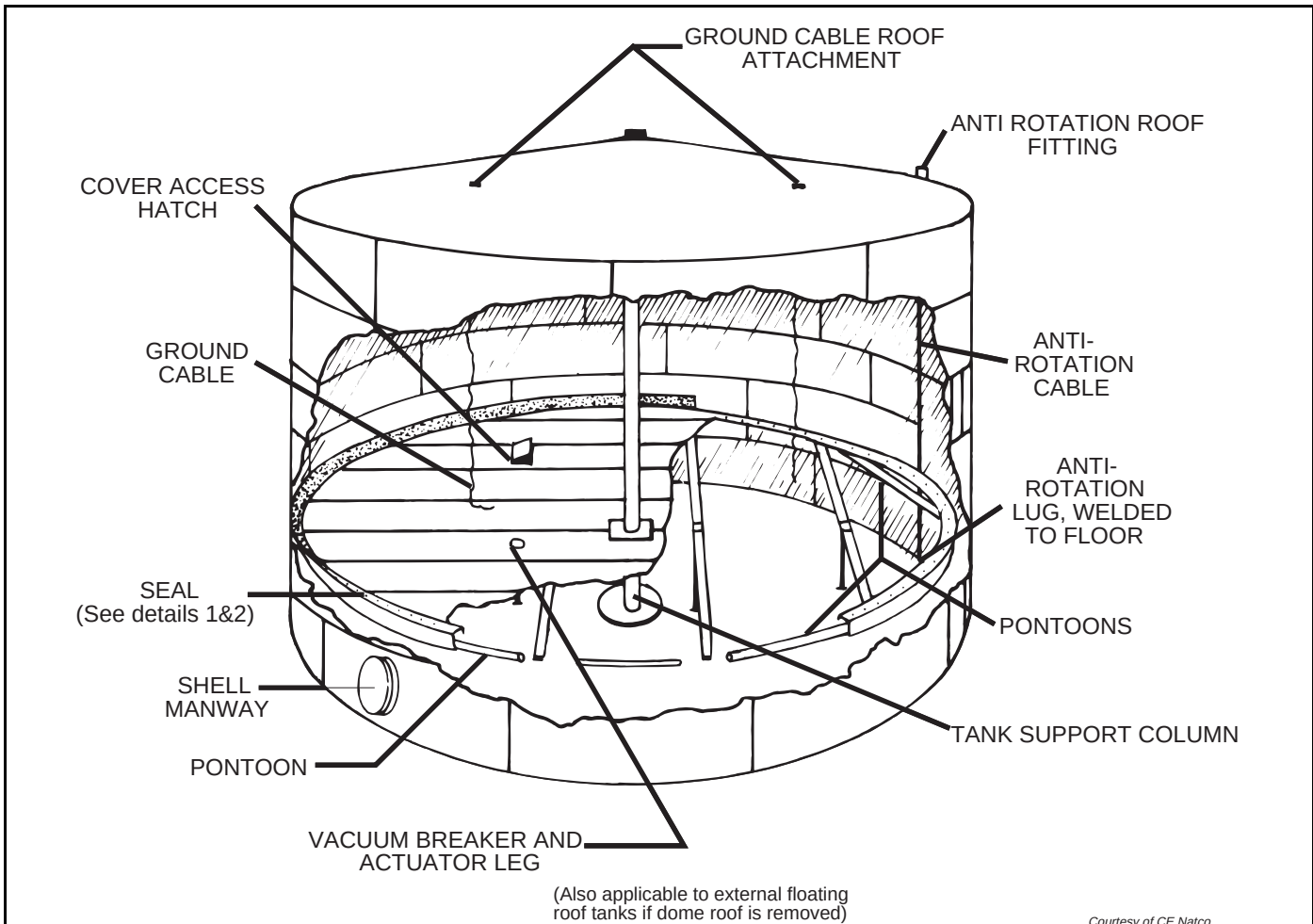
Specialty — Pipe Storage (Fig. 6-7) — Pipe that is used specifically for storing and handling liquid petroleum components or liquid anhydrous ammonia must be designed and constructed in accordance with any applicable codes.

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 Ponds² — Ponds are used for disposal, evaporation, or storage of liquids. Environmental considerations may pre-

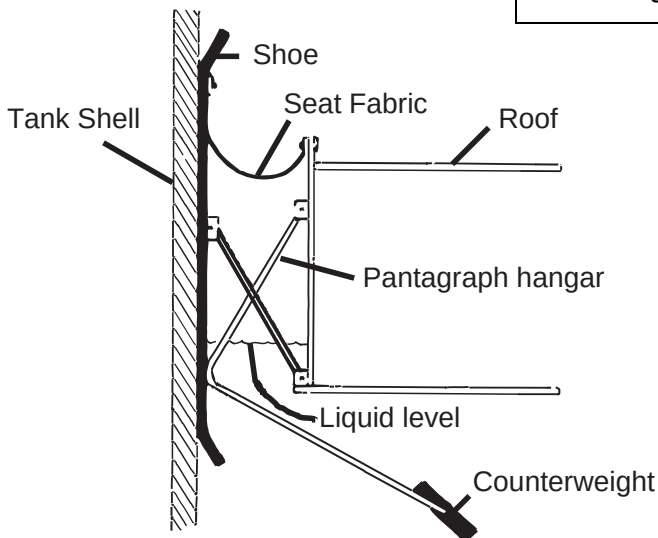
FIG. 6-6

Typical Arrangement of Internal Floating Roof Tank



Courtesy of CE Natco

Detail 1 - Shoe seal



Example Types of Seals for Floating Roof Tanks

Detail 2 - Tube Seal

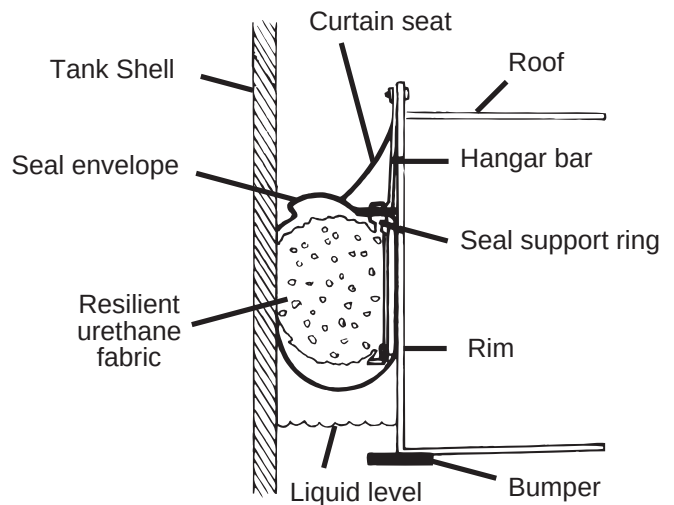
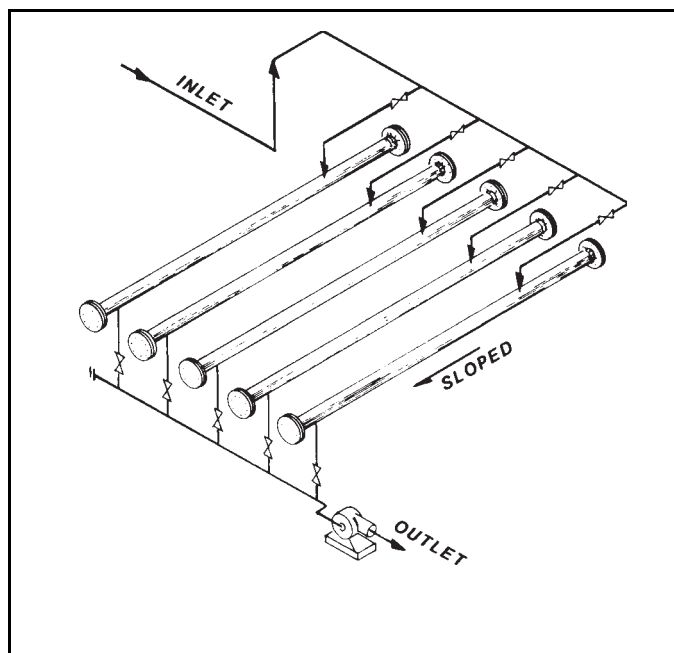


FIG. 6-7
Pipe Storage



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

Brine Displacement Cavern Operation (Solution Mined Cavern)

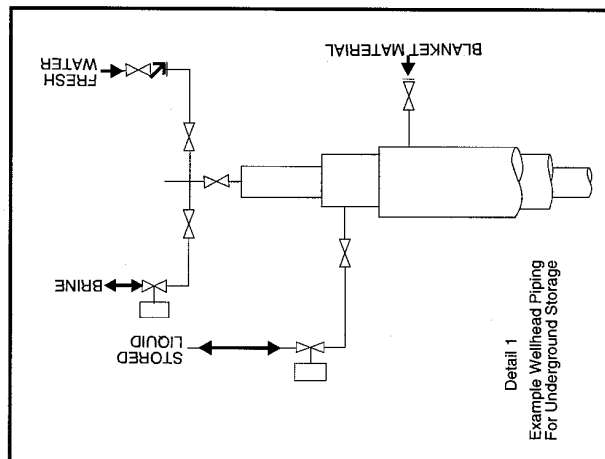


FIG. 6-9

Pump-Out Cavern Operation (Fracture Connected Solution Mined Cavern in Bedded Salt)

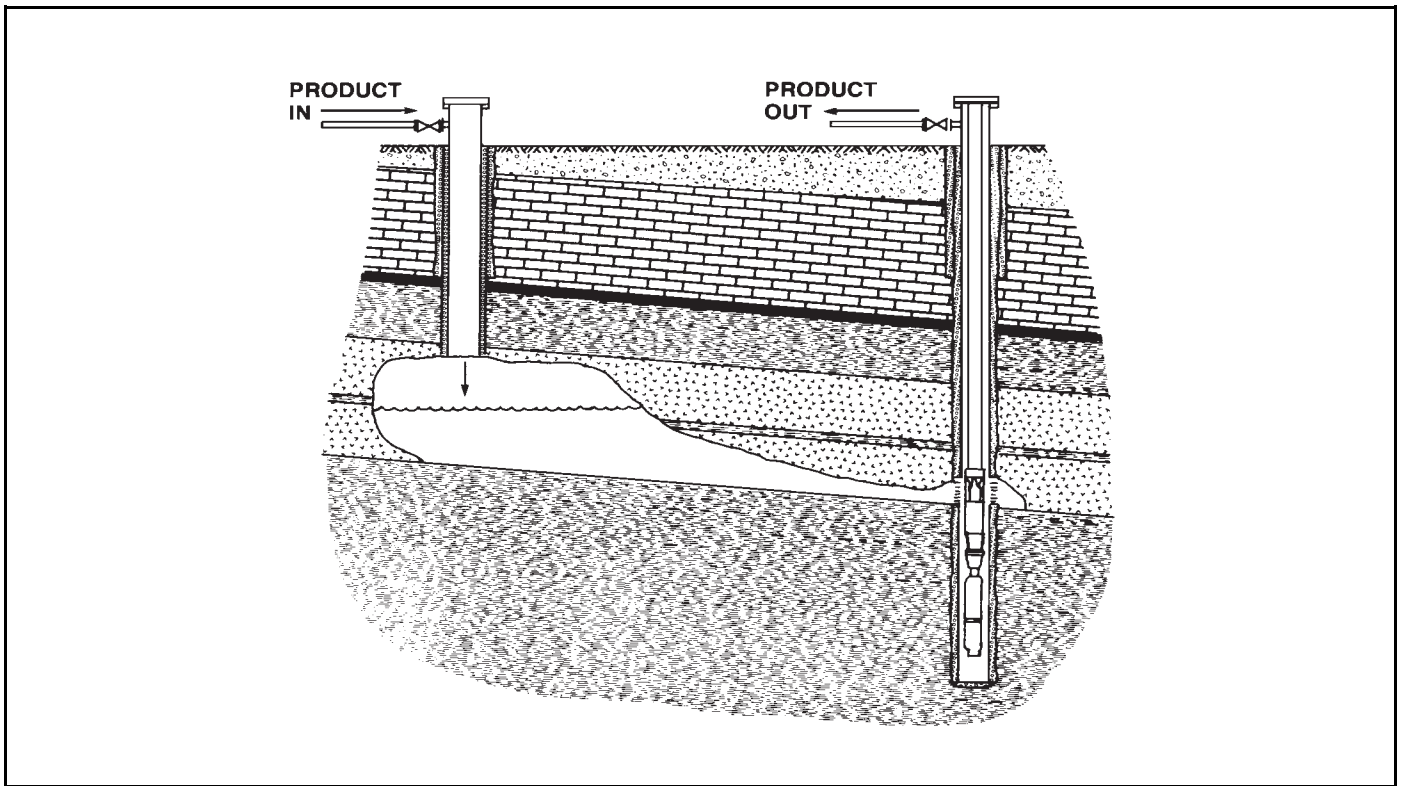
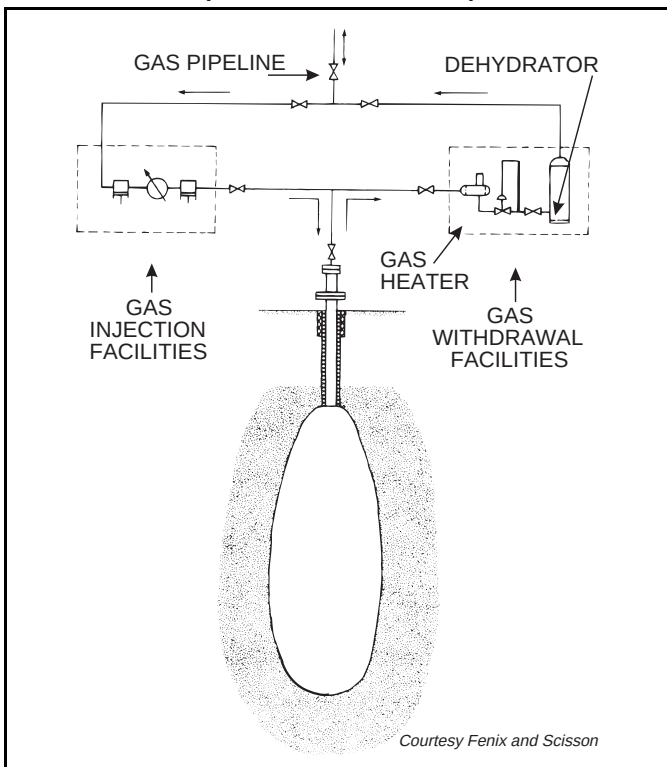


FIG. 6-10

Compression/Expansion Cavern Operation
(Solution Mined Cavern)



Courtesy Fenix and Scisson

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:

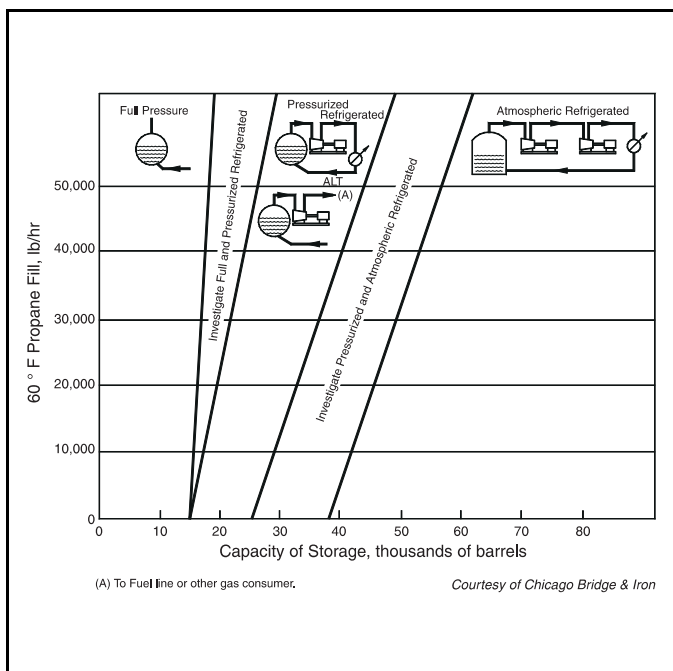
- Pump energy requirements
- Barometric pressure variations
- Product compositions
- Non-condensables
- Solar radiation effects
- Superheated products

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.

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

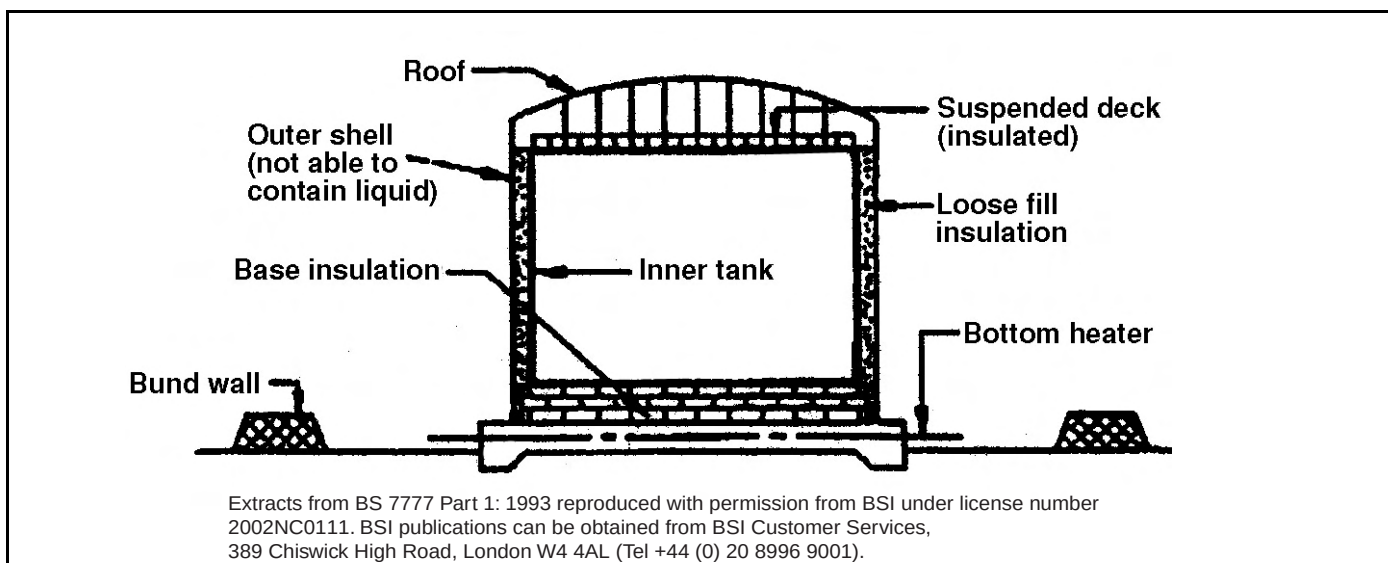
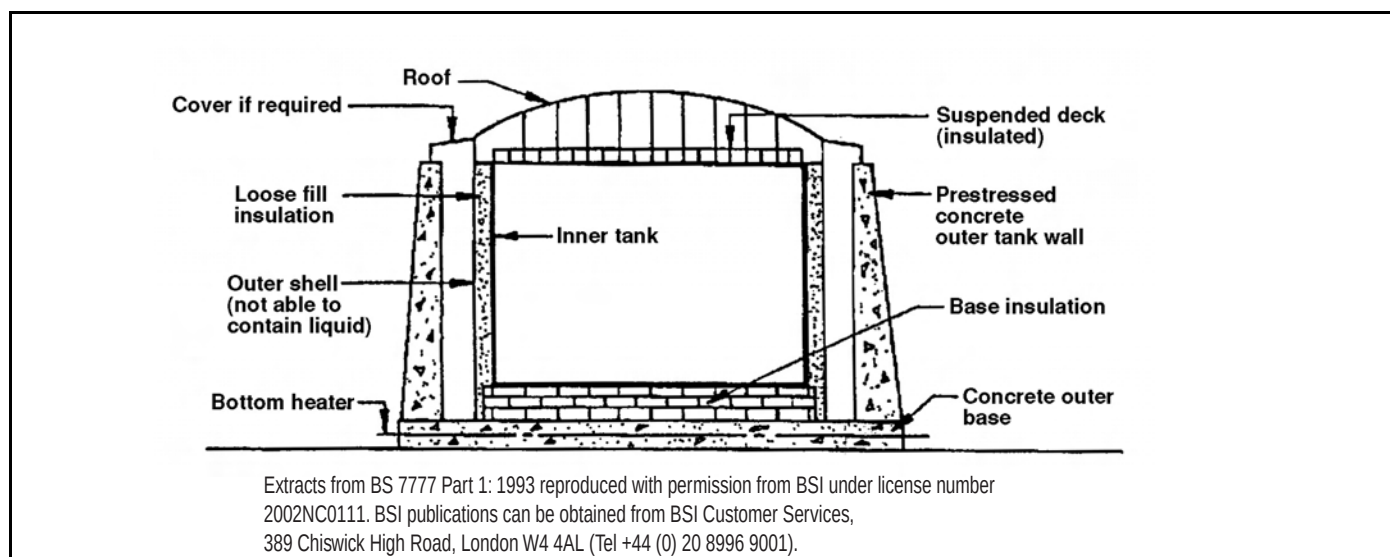


FIG. 6-13
Double Containment Tank



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.

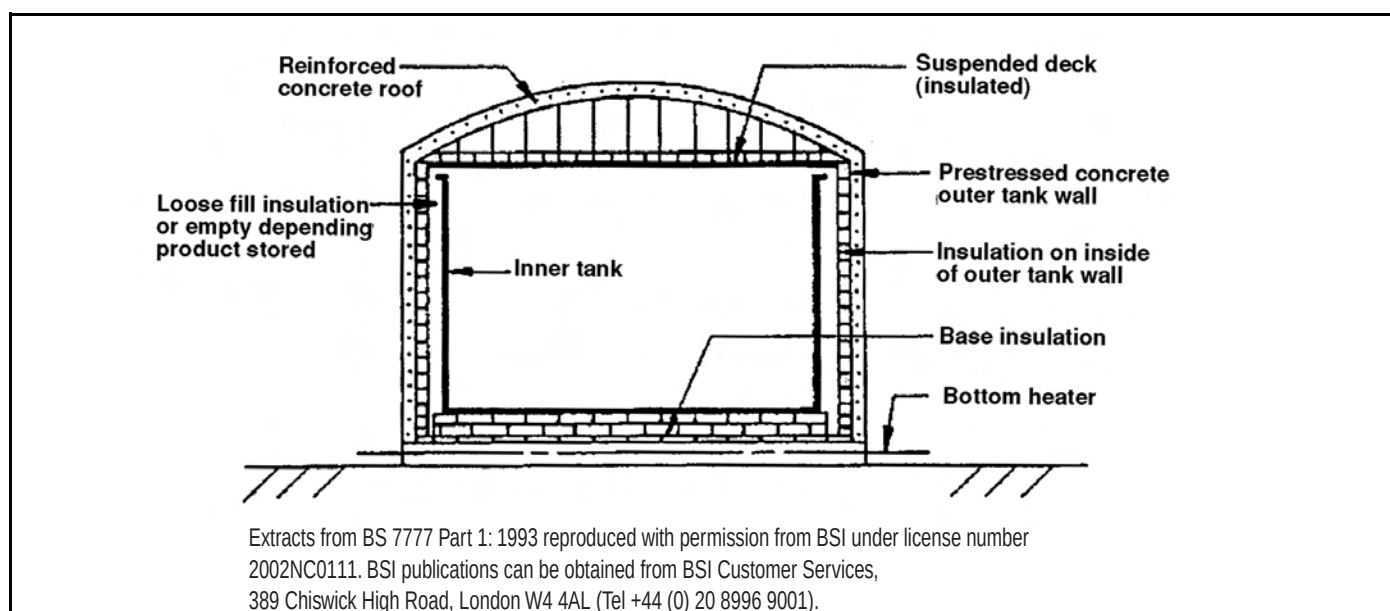
The inner tank contains the refrigerated liquid under normal operating conditions. The outer roof is supported by the outer tank. The outer tank is intended to be capable both of containing the refrigerated liquid and of controlled venting of the vapor resulting from product leakage after a credible 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



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 top-coats. 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-

FIG. 6-15

Constants for Determining Thermal Conductivity and Unit Heat-Transfer Rate for Some Common Insulating Materials

$$k = A' + B' T + C' T^2 + D' T^3 \quad \text{where } T = ^\circ\text{F}$$

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 Refractorios 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

$$\begin{aligned} T_h &= 450 & \Delta T_i &= 450 - 130 = 320 \\ T_s &= 130 & \Delta T_f &= 130 - 80 = 50 \\ T_a &= 80 & T_m &= (450 + 130)/2 = 290 \end{aligned}$$

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 $T_m = 290^\circ\text{F}$:

$$\begin{aligned} k &= 0.3488 + (5.038) (10^{-4}) (290) + (1.144) (10^{-7}) (290)^2 \\ &\quad + (7.172) (10^{-10}) (290)^3 \\ k &= 0.52 \end{aligned}$$

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 = $80 + 50 = 130^\circ\text{F}$. 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.

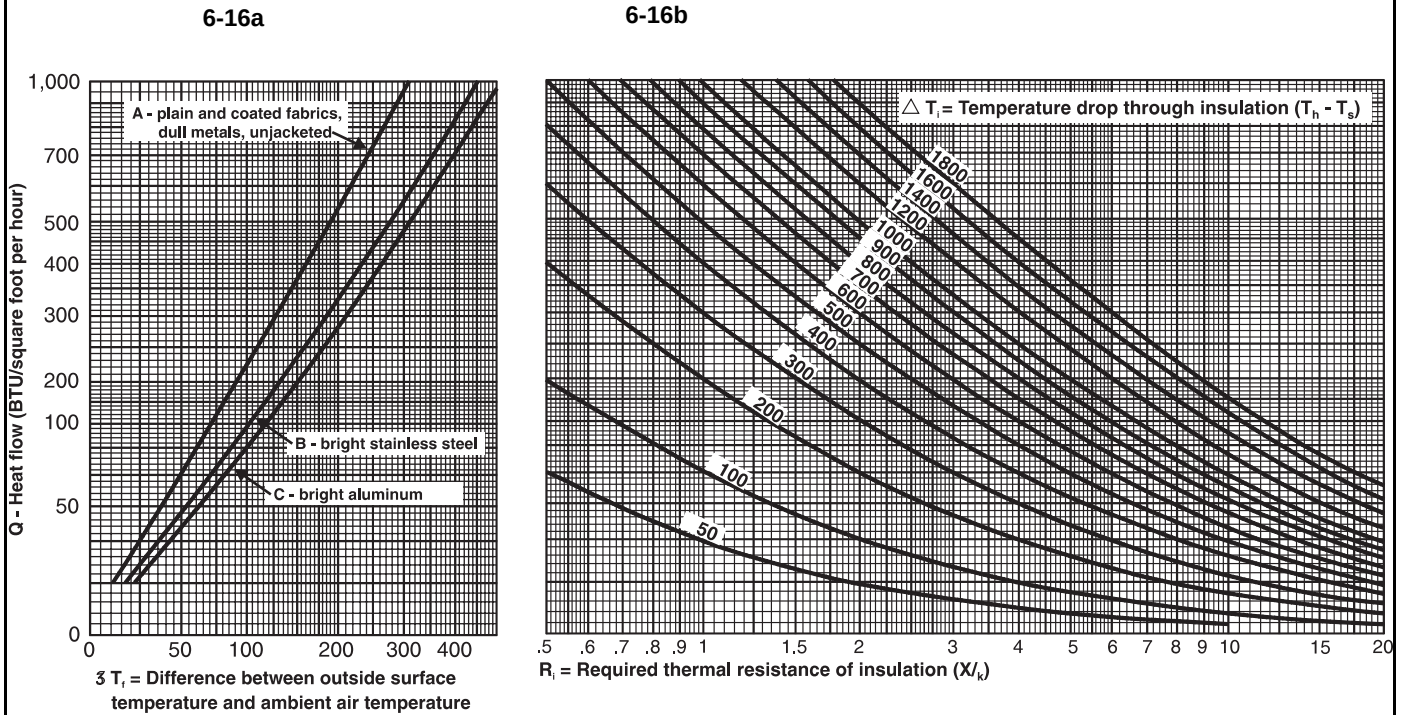
After determining the required thickness of 1.7 inches for a flat surface, go to Fig. 6-16c. Project horizontally from 1.7" for a flat surface to the line representing a 4" O.D. duct then vertically to an actual thickness of 1.35". The heat loss of 98 Btu/hr • sq ft of outside insulation surface remains the same.*

The heat loss per linear foot of outside duct surface (including insulation) is:

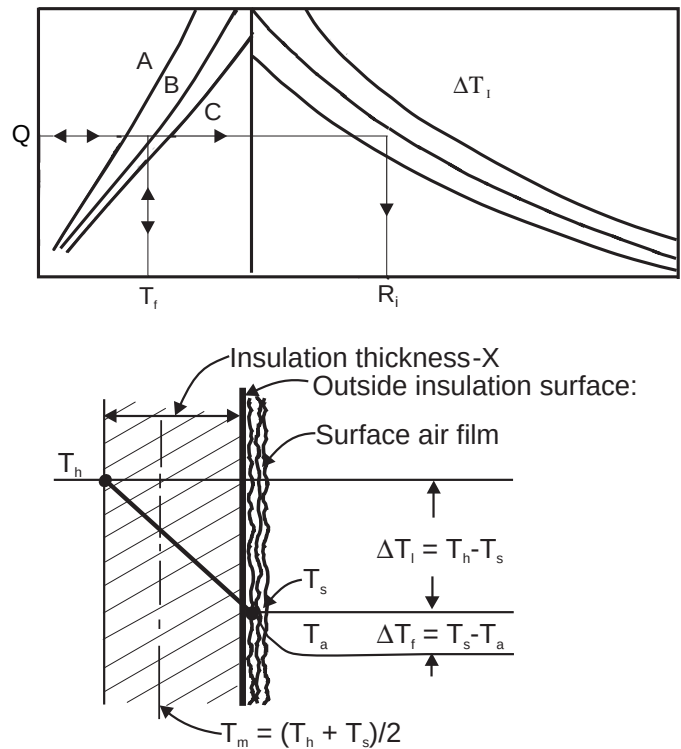
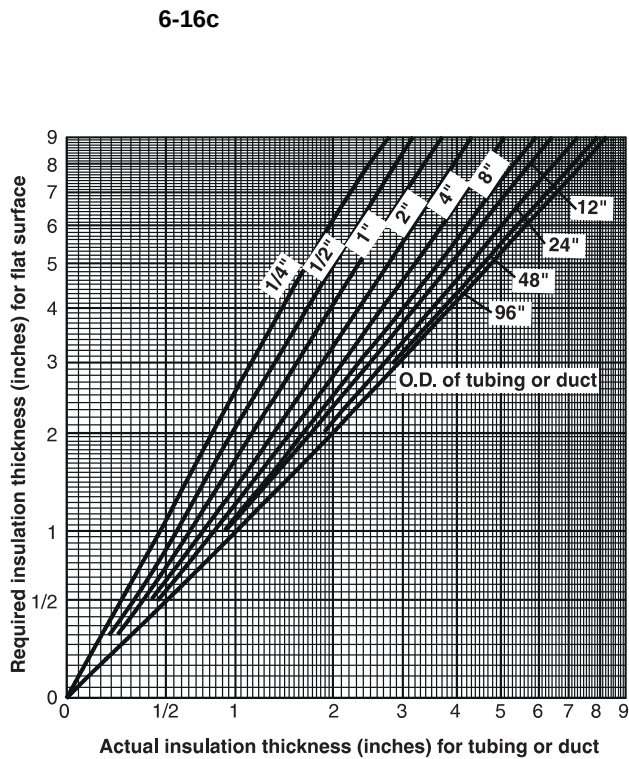
$$\frac{\pi \text{ OD}}{12} (Q) = \left(\frac{6.7 \cdot \pi}{12} \right) 98 = 171 \text{ Btu/hr / linear ft}$$

*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

FIG. 6-16
Heat Flow Through Insulation



How to Use [Figs. 6-16 a & b](#)



Courtesy John Mansville

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 $T_h = 1100$ $T_a = 75$
 $Q = 150 \text{ Btu/hr} \cdot \text{sq ft}$

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/ft³) 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.

$$\Delta T_i (\text{mineral wool}) (1100 - 500) = 600$$

$$T_m (\text{mineral wool}) (1100 + 500)/2 = 800$$

$$k (\text{from Fig. 6-15}) \text{ of mineral wool at } 800^\circ\text{F} (T_m) = 0.53$$

$$\Delta T_i (\text{cellular glass foam}) (500 - 180) = 320$$

$$T_m (\text{cellular glass foam}) (500 + 180)/2 = 340$$

$$k (\text{from Fig. 6-15}) \text{ of cellular glass foam at } 340^\circ\text{F} (T_m) = 0.56$$

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 $(4.0 \times .53) = 2.12$ 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 there-

fore 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.

FIG. 6-17

Summary of Specifications for Low-Temperature and Cryogenic Steels^{(1) (2)}

ASTM/AISI Spec & Grade	Lowest Usual Service Temp.	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.

- 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.,

$$\Sigma(y_i) = \Sigma(K_i x_i) = 1.0 \quad \text{Eq 6-4}$$

3. Calculate the compressibility factor for the vapor by either (a) or (b).
 - a. Compressibility factor charts, Section 23. Pseudo-critical 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:

$$PV = n_g ZRT, n_g = PV/ZRT = \text{total moles vapor} \quad \text{Eq 6-5}$$

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.

$$\Sigma[n_g(y_i) (\text{gal./mole})_i] = 60^\circ\text{F gallons in vapor phase} \quad \text{Eq 6-6}$$

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 = Σn_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.

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

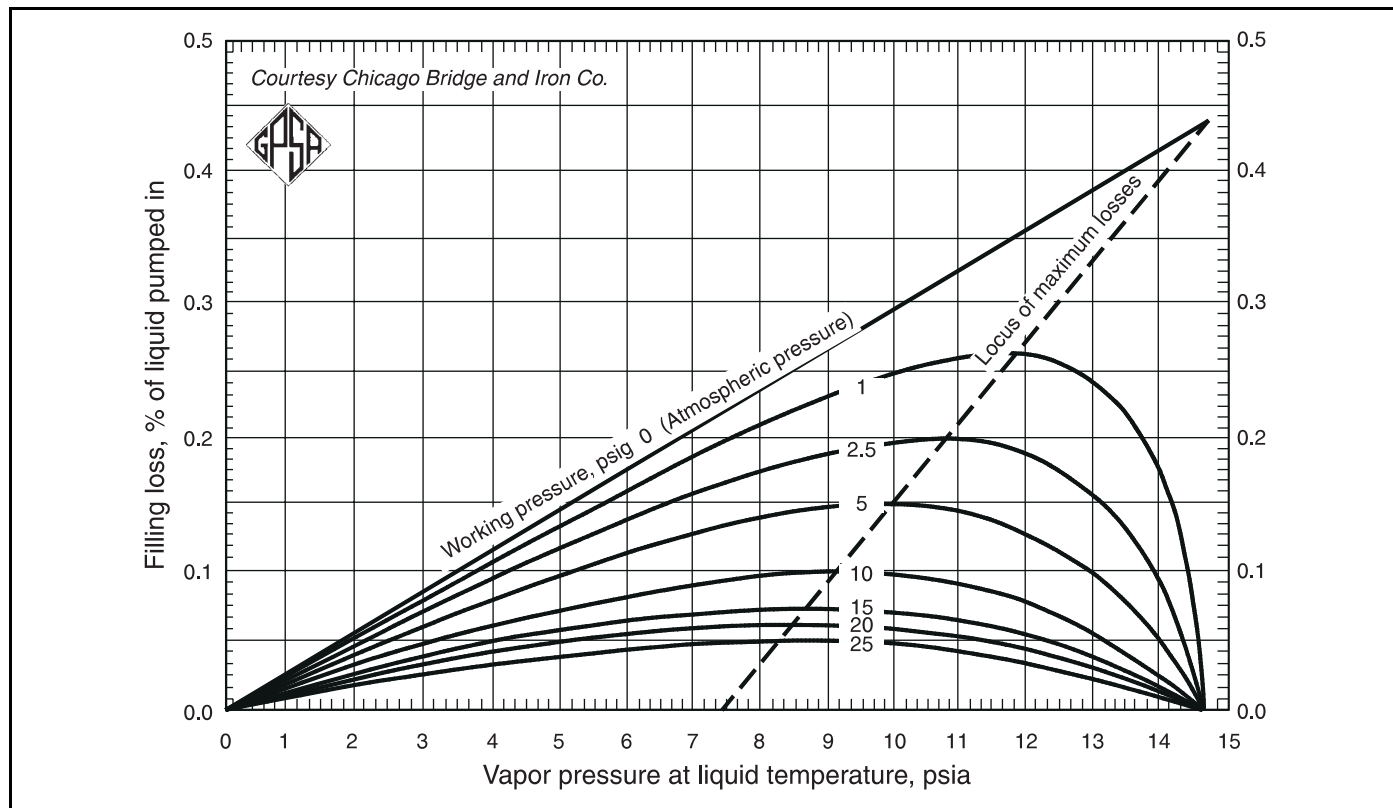


FIG. 6-19

Liquid Equivalent of Tank Vapor

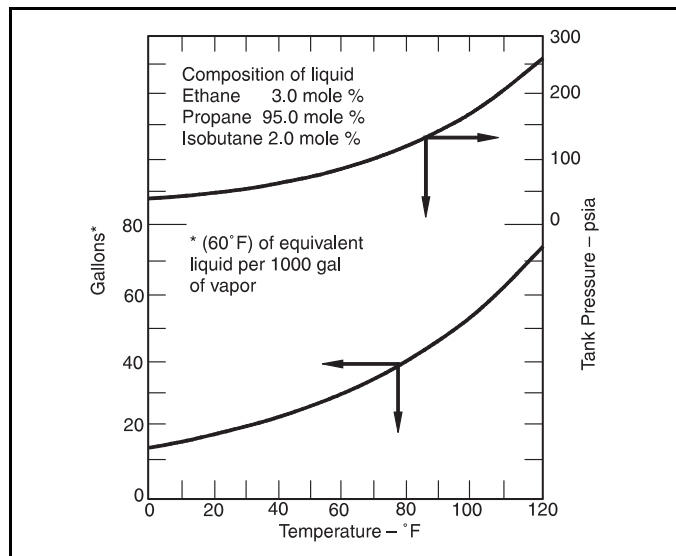
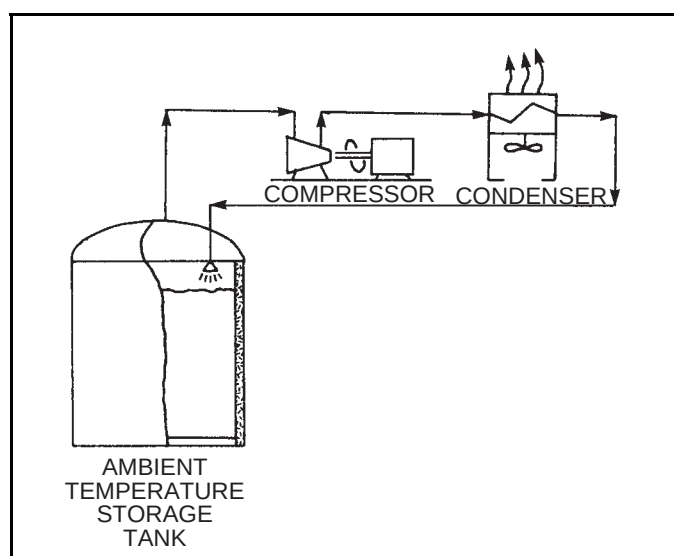


FIG. 6-20

Ambient Temperature Vapor Recovery Cycle



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.

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.

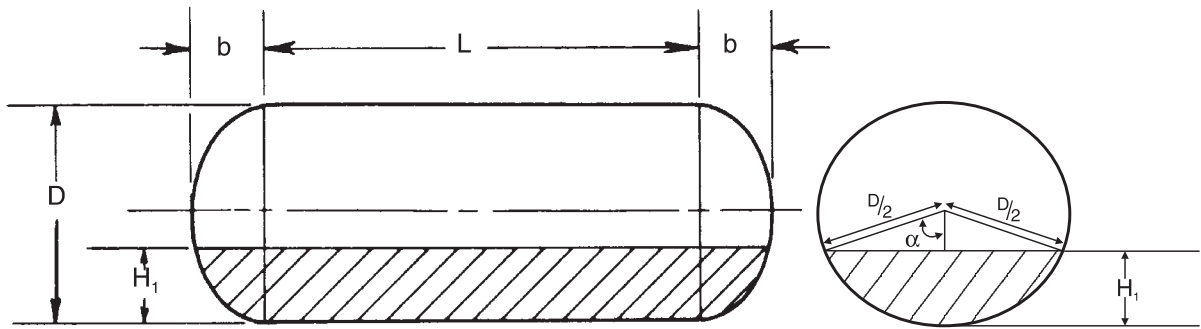
D = Diameter in Feet

0.1398854 D² = Barrels of 42 U.S. Gallons per vertical foot.5.875185 D² = U.S. Gallons per vertical foot.4.892148 D² = Imperial Gallons per vertical foot.0.022240 D² = Cubic Meters per vertical foot.0.785398 D² (D in meters) = Cubic meters per vertical meter.

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

$$= 1/6 \pi K_1 D^3 + 1/4 \pi D^2 L$$

$$K_1 = 2b/D \quad Ze = H_1/D \quad Zc = H_1/D$$

$$\text{Partial volume} = 1/6 \pi K_1 D^3 \times [f(Ze)] + 1/4 \pi D^2 L \times [f(Zc)]$$

$$f(Zc) = \text{Horizontal cylinder coefficient (see Fig. 6-24)} \quad \text{or} \quad f(Zc) = \left(\frac{\alpha - \sin(\alpha) \times \cos(\alpha)}{\pi} \right)$$

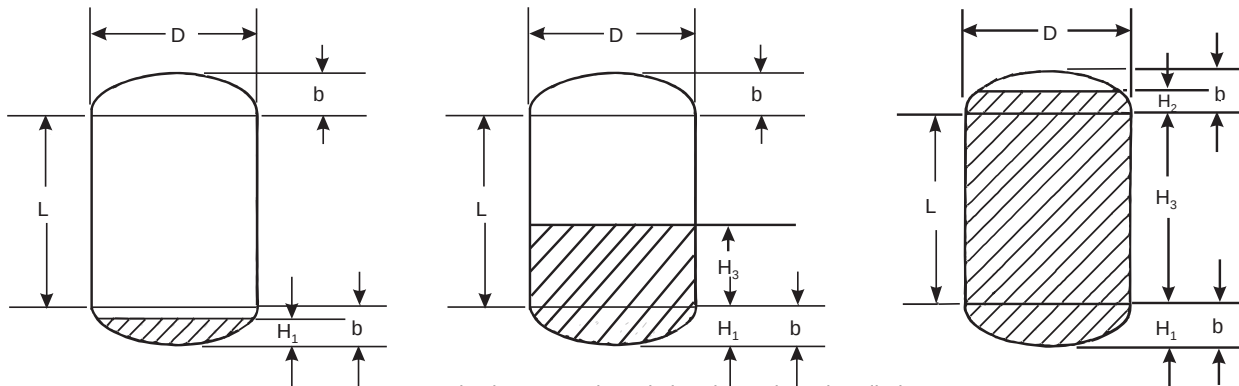
$$f(Ze) = \text{Ellipsoidal coefficient (see Fig. 6-25)} \quad \text{or} \quad f(Ze) = -\left(\frac{H_1}{D}\right)^2 \times \left(-3 + \frac{2H_1}{D}\right)$$

For elliptical 2:1 heads, $b = 1/4 D$, $K_1 = 1/2$

$$\alpha = 2 \times \text{Atan} \left(\frac{H_1}{\sqrt{\left(2 \times H_1 \times \frac{D}{2}\right) - H_1^2}} \right)$$

where α is in radians

VERTICAL CYLINDRICAL TANKS



$$\begin{aligned} \text{Total volume} &= \text{volume in heads} + \text{volume in cylinder} \\ &= 1/6 \pi K_1 D^3 + 1/4 \pi D^2 L \end{aligned}$$

$$\text{Partial volume} = 1/6 \pi K_1 D^3 \times [f(Ze)] + 1/4 \pi D^2 H_3$$

$$K_1 = 2b/D$$

$$Ze = (H_1 + H_2)/K_1 D$$

$$f(Ze) = \text{Ellipsoidal coefficient (see Fig. 6-25)} \quad \text{or} \quad f(Ze) = -\left(\frac{H_1 + H_2}{2b}\right)^2 \times \left(-3 + \left(\frac{H_1 + H_2}{b}\right)\right)$$

FIG. 6-24

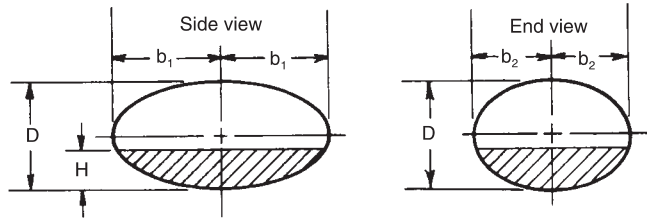
Coefficients for Partial Volumes of Horizontal Cylinders, $f(Z_c)$

Z_c	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									

FIG. 6-25

Table of Coefficients and Formulas for Determining Partial Volumes in Ellipsoids and Spheres

GENERAL ELLIPSOID



$$\text{Total volume of ellipsoid} = 1/6 \pi K_1 K_2 D^3$$

$$\text{Partial Volume} = 1/6 \pi K_1 K_2 D^3 [f(Ze)]$$

$$K_1 = 2b_1/D \quad K_2 = 2b_2/D$$

$$Ze = H/D \quad f(Ze) = \text{ellipsoidal coefficient (see below)} \quad \text{or} = -\left(\frac{H_1}{D}\right)^2 \times \left(-3 + \frac{2H_1}{D}\right)$$

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.

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 = $(3d - 2h) h^2 \times 0.0000539681$ where d is diameter of sphere and h is depth of liquid both in inches.

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 ftPartial Volume = $W \cdot L \cdot H_p$ = cu ft

* For imperial gallons, divide above capacities by 1.2

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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2. Oil Insurance Association Recommendations & Guidelines for Gasoline Plants - No. 301.
3. Perry, Robert H., *Perry's Chemical Engineer's Handbook*, 6th Edition, 1985.
4. Selby, Samuel M., *Standard Mathematical Tables*, 21st Edition, 1973.
5. Underwriters Laboratories (UL)
No. 142 Steel Above Ground Tanks
No. 58 Steel Underground Tanks

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 coa-

lescing. 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	MW = molecular weight, lb/lb mole
A_p = particle or droplet cross sectional area, ft^2	P = system pressure, psia
C = empirical constant for separator sizing, ft/hr	Q = estimated gas flow capacity, MMscfd per ft^2 of filter area
C^* = empirical constant for liquid-liquid separators, $(\text{bbl} \cdot \text{cp})/(\text{ft}^2 \cdot \text{day})$	Q_A = actual gas flow rate, ft^3/sec
C' = drag coefficient of particle, dimensionless (Fig. 7-3)	R = gas constant, $10.73 (\text{psia} \cdot \text{ft}^3)/(\text{°R} \cdot \text{lb mole})$
D_i = separator inlet nozzle diameter, in.	Re = Reynolds number, dimensionless
D_p = droplet diameter, ft	S_{hl} = specific gravity of heavy liquid, water = 1.0
D_v = inside diameter of vessel, ft	S_{ll} = specific gravity of light liquid, water = 1.0
G_m = maximum allowable gas mass-velocity necessary for particles of size D_p to drop or settle out of gas, $\text{lb}/(\text{hr} \cdot \text{ft}^2)$	T = system temperature, °R
g = acceleration due to gravity, 32.2 ft/sec^2	t = retention time, minutes
H_l = width of liquid interface area, ft	U = volume of settling section, bbl
J = gas momentum, $\text{lb}/(\text{ft} \cdot \text{sec}^2)$	V_t = critical or terminal gas velocity necessary for particles of size D_p to drop or settle out of gas, ft/sec
K = empirical constant for separator sizing, ft/sec	W = total liquid flow rate, bbl/day
K_{CR} = proportionality constant from Fig. 7-5 for use in Eq 7-5, dimensionless	W_{cl} = flow rate of light condensate liquid, bbl/day
L = seam to seam length of vessel, ft	Z = compressibility factor, dimensionless
L_l = length of liquid interface area, ft	Greek:
M = mass flow, lb/sec	ρ_g = gas phase density, lb/ft^3
M_p = mass of droplet or particle, lb	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.

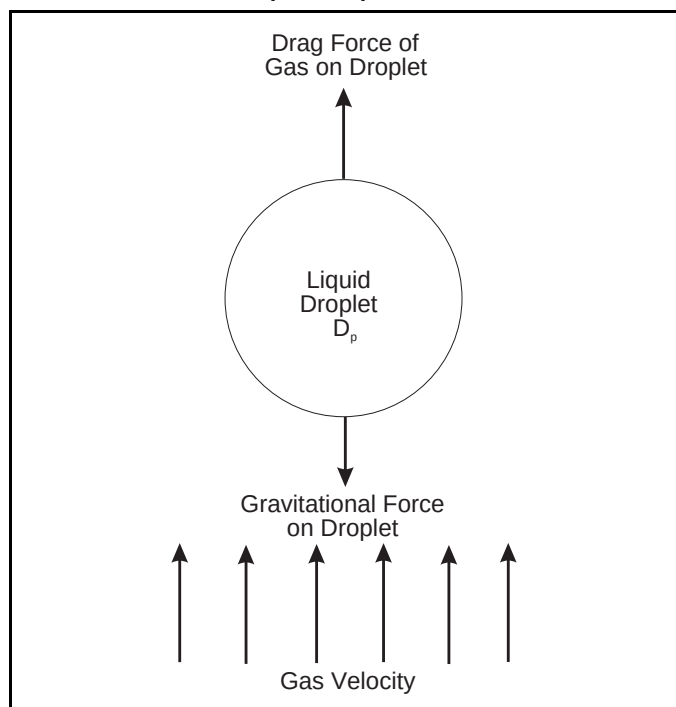
Slug Catcher: A particular separator design able to absorb sustained in-flow of large liquid volumes at irregular intervals. Usually found on gas gathering systems or other two-phase pipeline systems. A slug catcher may be a single large vessel or a manifolded system of pipes.

Three Phase Separator: A vessel used to separate gas and two immiscible liquids of different densities (e.g. gas, water, and oil).

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



$$V_t = \sqrt{\frac{2 g M_p (\rho_l - \rho_g)}{\rho_l \rho_g A_p C'}} = \sqrt{\frac{4 g D_p (\rho_l - \rho_g)}{3 \rho_g C'}} \quad \text{Eq 7-1}$$

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:

$$Re = \frac{1,488 D_p V_t \rho_g}{\mu} \quad \text{Eq 7-2}$$

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:

$$C' (Re)^2 = \frac{(0.95) (10^8) \rho_g D_p^3 (\rho_l - \rho_g)}{\mu^2} \quad \text{Eq 7-3}$$

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 Particles¹

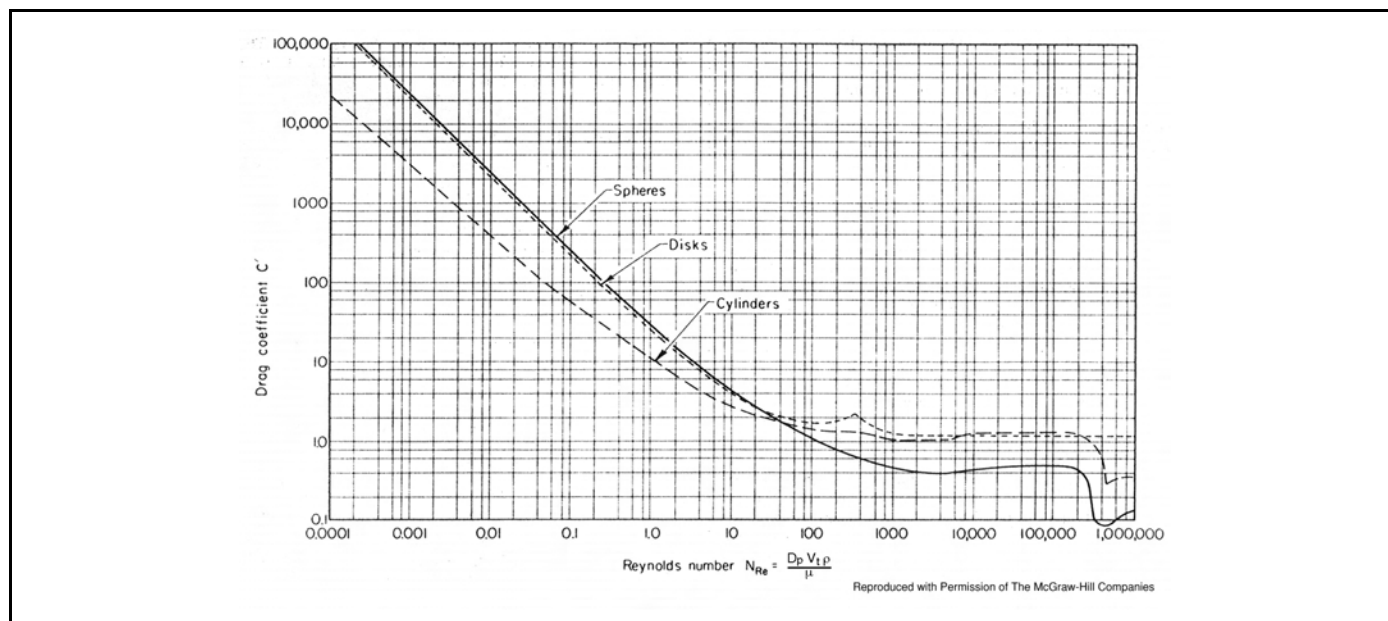
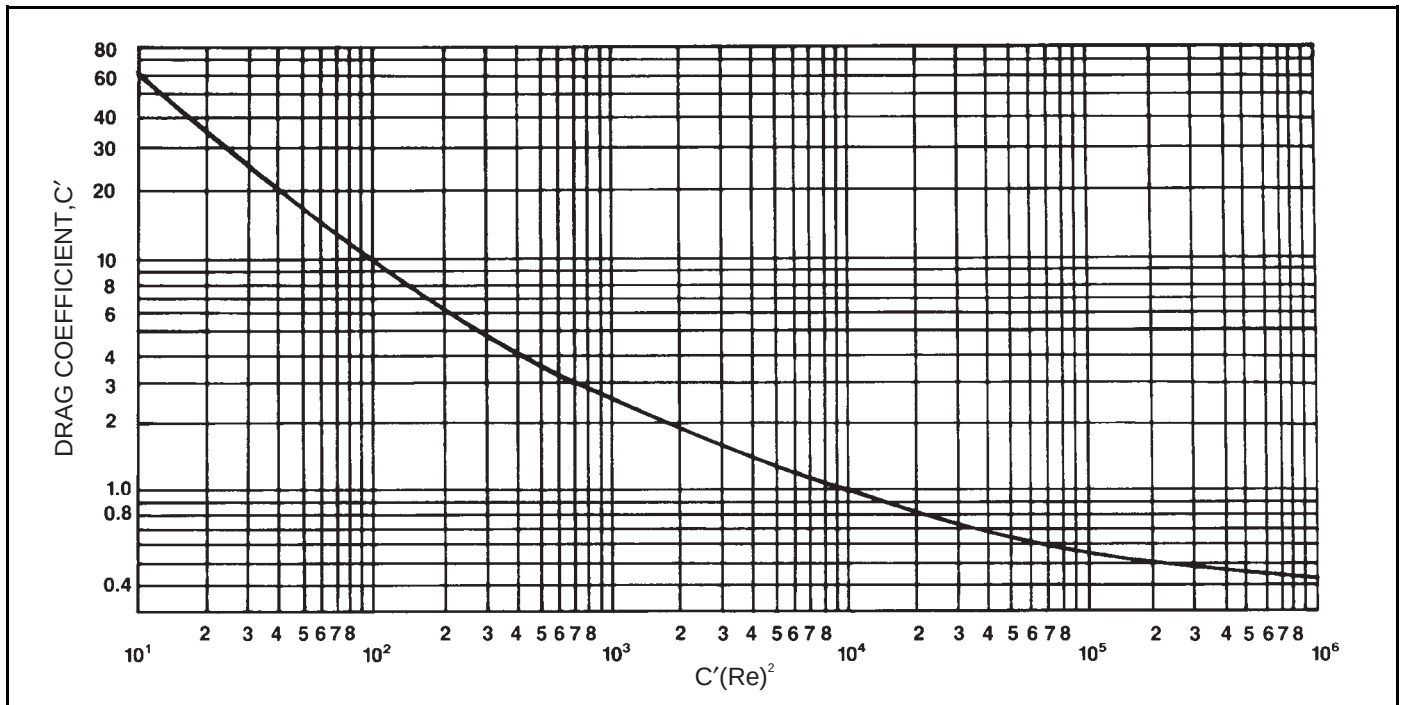


FIG. 7-4
Drag Coefficient of Rigid Spheres²



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:

$$V_t = \frac{1,488 \text{ g } D_p^2 (\rho_l - \rho_g)}{18 \mu} \quad \text{Eq 7-4}$$

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.

$$D_p = K_{CR} \left[\frac{\mu^2}{\text{g } \rho_g (\rho_l - \rho_g)} \right]^{0.33} \quad \text{Eq 7-5}$$

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:

$$V_t = \frac{3.49 \text{ g}^{0.71} D_p^{1.14} (\rho_l - \rho_g)^{0.71}}{\rho_g^{0.29} \mu^{0.43}} \quad \text{Eq 7-6}$$

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:

$$V_t = 1.74 \sqrt{\frac{\text{g } D_p (\rho_l - \rho_g)}{\rho_g}} \quad \text{Eq 7-7}$$

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 $K_{CR} = 18.13$.

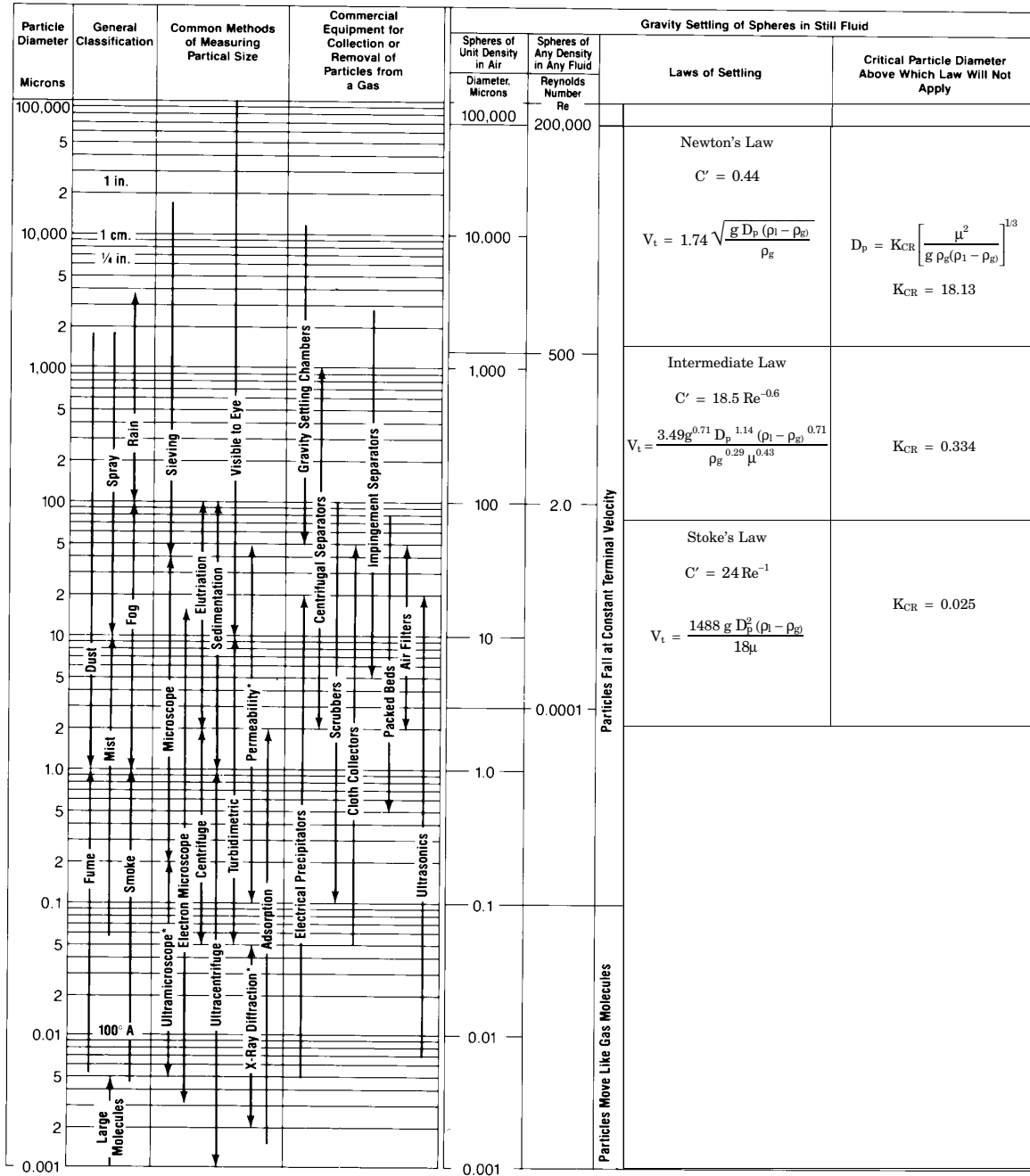
DROPLET SIZE DISTRIBUTIONS AND ENTRAINMENT LOADINGS

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.

FIG. 7-5
Gravity Settling Laws and Particle Characteristics



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

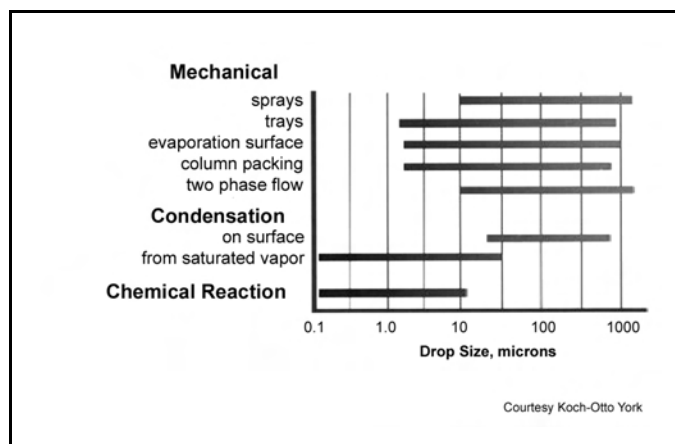
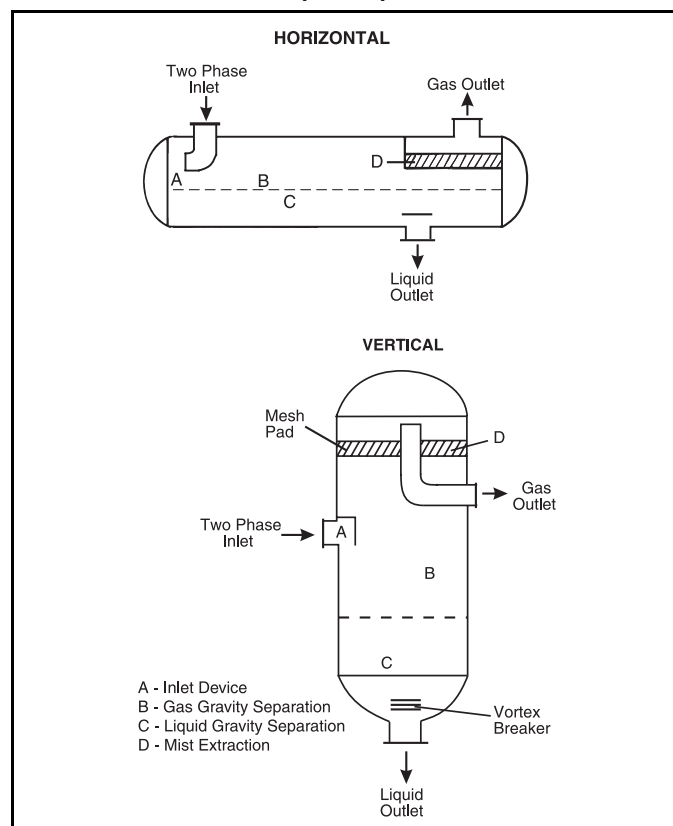


FIG. 7-7
Gas-Liquid Separators



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?

- 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

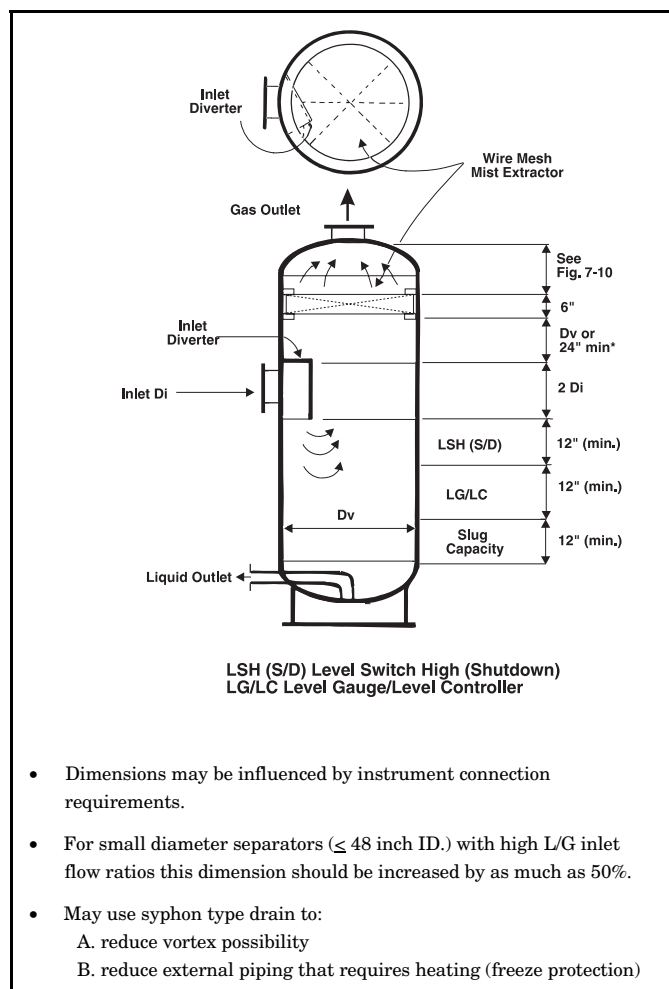
Horizontal Separators

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

FIG. 7-8

Vertical Separator with Wire Mesh Mist Extractor



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.

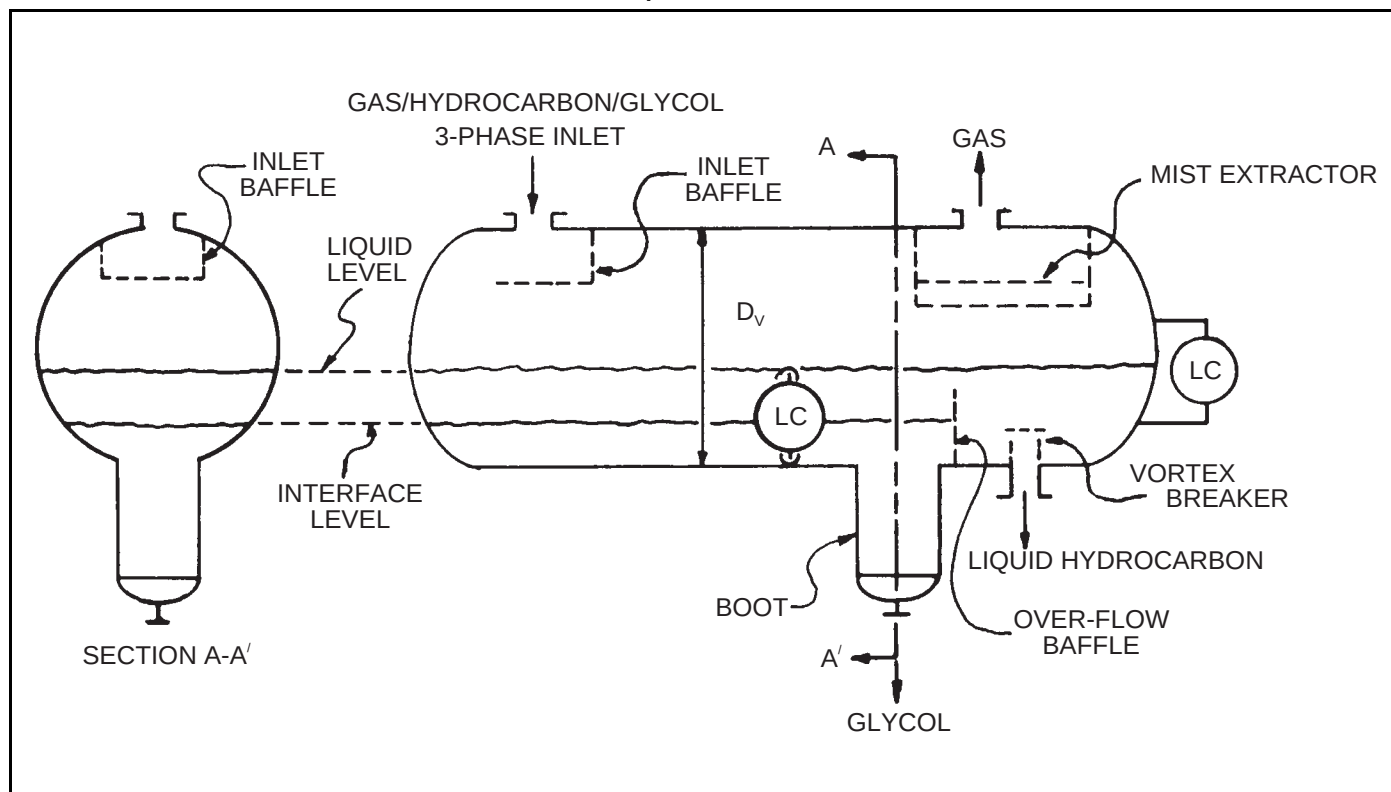
Typical L/D ratios for horizontal separators normally fall in the range of 2.5–5.

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. 7-9

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-

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/ft³ bulk density. Minimum recommended pad thickness is 4 inches⁴. 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 Brown⁵ equation given below:

$$V_t = K \sqrt{\frac{\rho_l - \rho_g}{\rho_g}} \quad \text{Eq 7-8}$$

The required mist extractor area is obtained from

$$A = \frac{Q_A}{V_t} \quad \text{Eq 7-9}$$

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/ft² 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

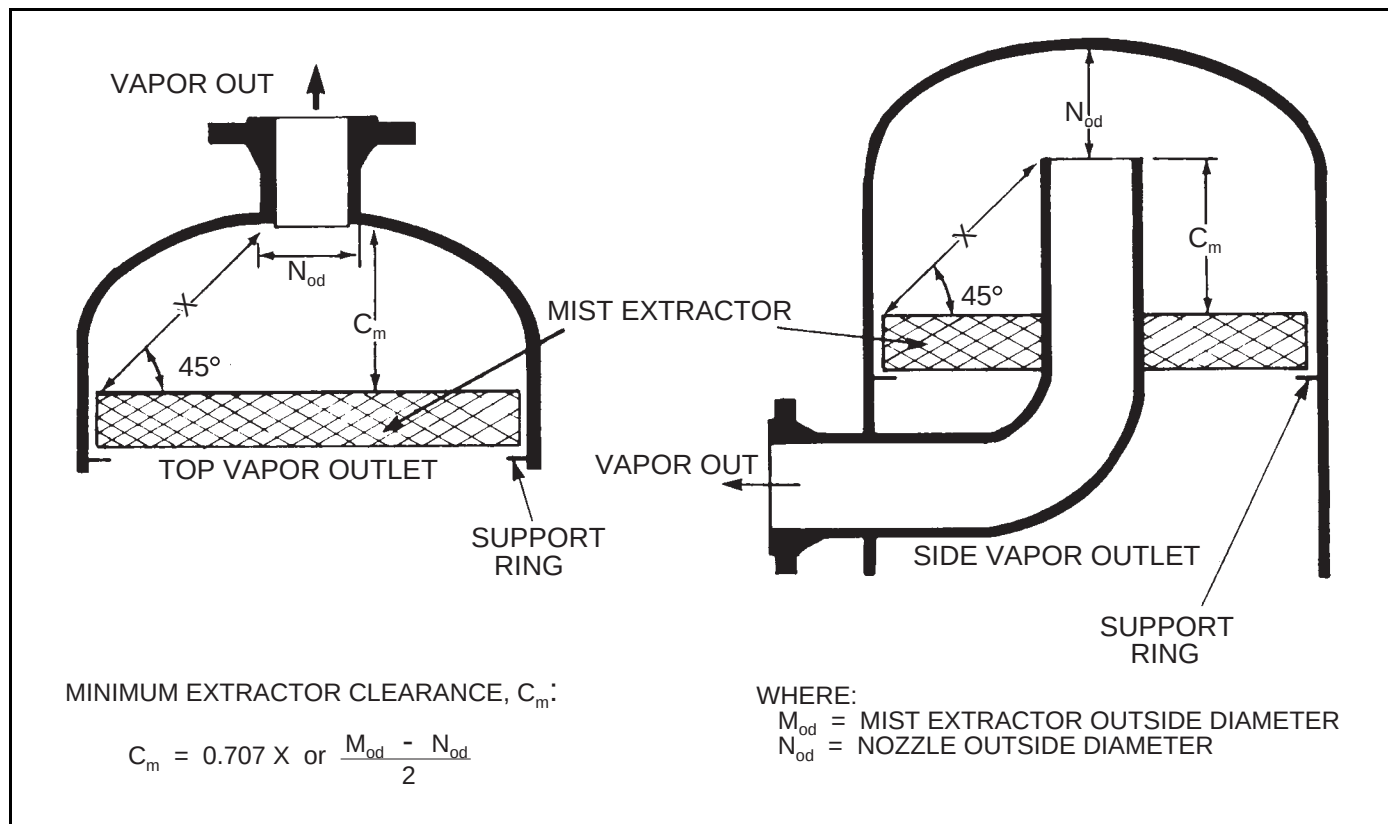


FIG. 7-11

Horizontal Separator with Knitted Wire Mesh Pad Mist Extractor and Lower Liquid Barrel

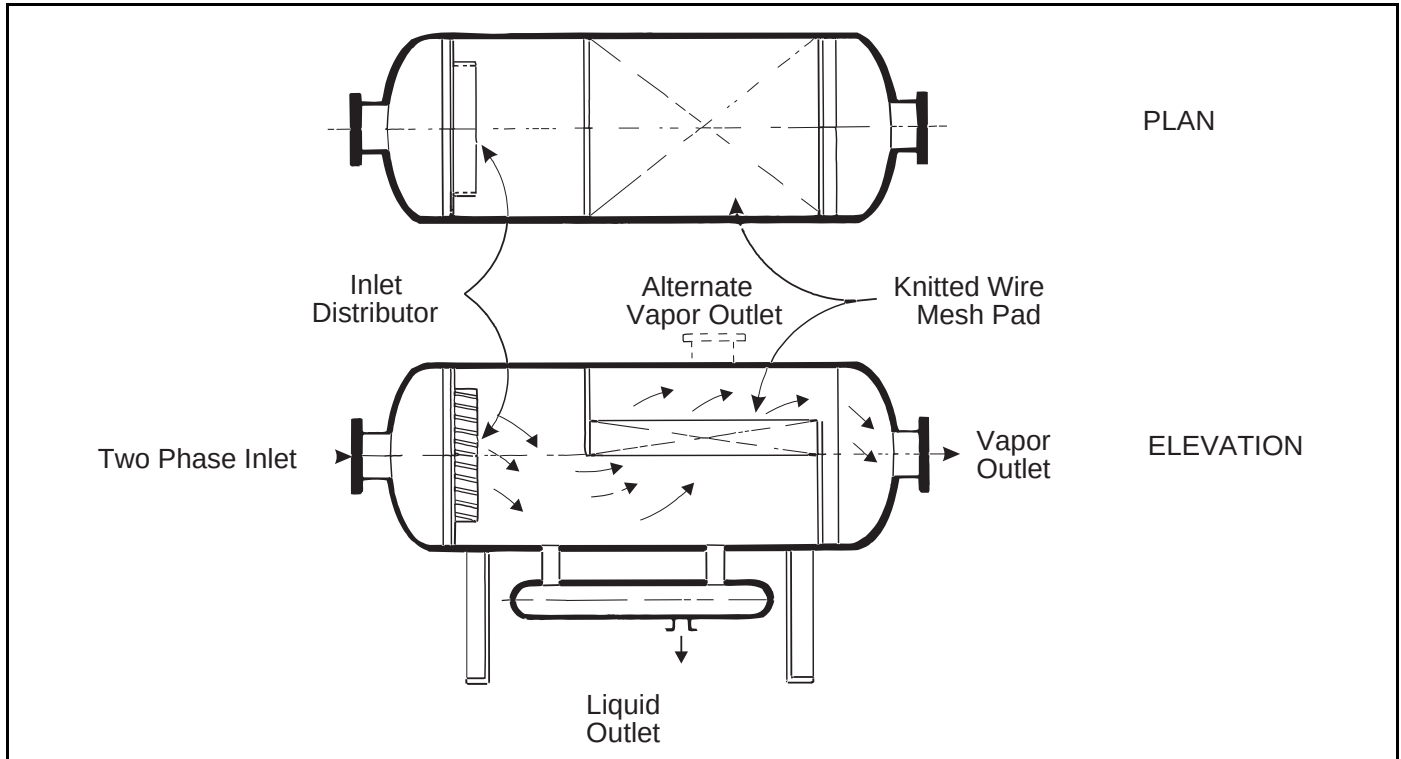


FIG. 7-12

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 Pressure⁶

Pressure, psig	Percent of Design Value
Atmospheric	100
150	90
300	85
600	80
1,150	75

combined 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.

FIG. 7-14

Cross-Section of Vane Element Mist Extractor Showing Corrugated Plates with Liquid Drainage Traps

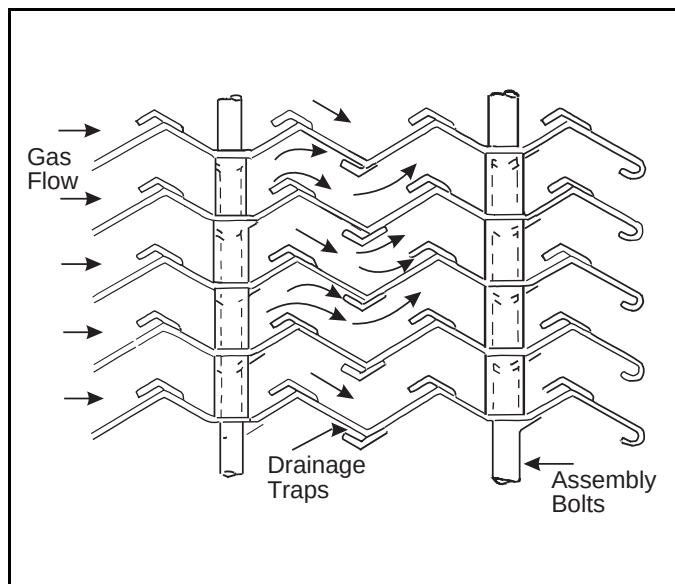


FIG. 7-15

Vertical Separator with Vane-Type Mist Extractor

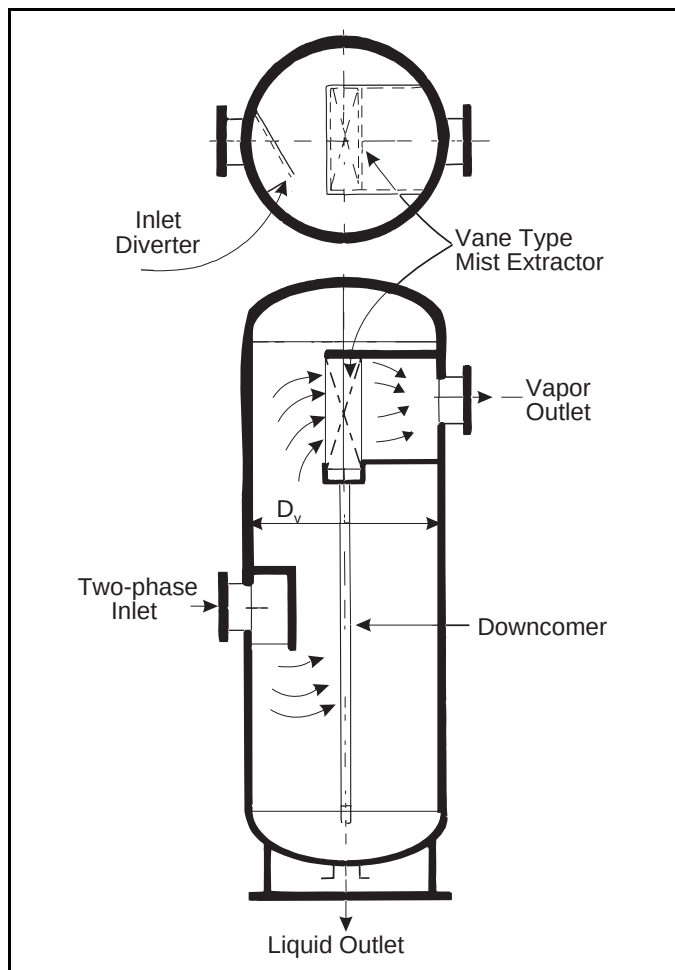


FIG. 7-16

Typical Vane Pack Separation Performance

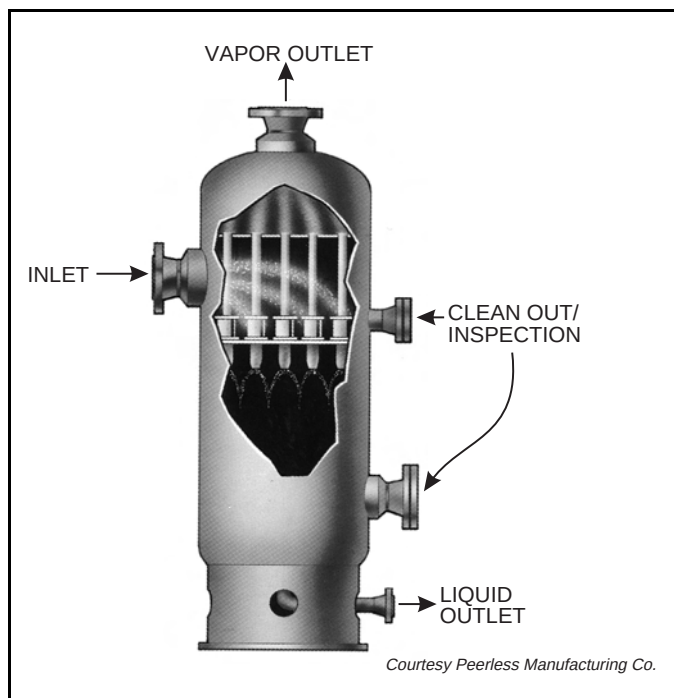
Droplet removal efficiency:	99% removal of droplets greater than 10–40 microns. Higher removal efficiency is for thicker packs, with closer vane spacings and more passes (bends).
Gas capacity, K, ft/sec	Horizontal flow: 0.9–1.0 Vertical up-flow: 0.4–0.5 The higher capacities are generally associated 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:

FIG. 7-17

Vertical Separator with Centrifugal Elements



- 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, turn-down, 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):

$$L = \frac{4 Q_A}{\pi V_t D_v} \quad \text{Eq 7-10}$$

If the separator is to be additionally used for liquid storage, this must also be considered in sizing the vessel.

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.

$$\text{Gas density, } \rho_g = \frac{P(\text{MW})}{RTZ} = \frac{(514.7)(21.72)}{(10.73)(560)(0.90)} = 2.07 \text{ lb/ft}^3$$

$$\text{Liquid density, } \rho_l = 0.5(62.4) = 31.2 \text{ lb/ft}^3$$

$$\text{Mass flow, } M = \frac{(60)(10^6)(21.72)}{(379)(24)(3600)} = 39.8 \text{ lb/sec}$$

$$\text{Particle diameter, } D_p = \frac{(150)(0.00003937)}{12} = 0.000492 \text{ ft}$$

From Eq 7-3,

$$C'(\text{Re})^2 = \frac{(0.95)(10^8)\rho_g D_p^3(\rho_l - \rho_g)}{\mu^2} = \frac{(0.95)(10^8)(2.07)(0.000492)^3(31.2 - 2.07)}{(0.012)^2} = 4738$$

From Fig. 7-4, Drag coefficient, $C' = 1.40$

$$\begin{aligned} \text{Terminal velocity, } V_t &= \sqrt{\frac{4gD_p(\rho_l - \rho_g)}{3\rho_g C'}} \\ &= \sqrt{\frac{4(32.2)(0.000492)(29.13)}{3(2.07)1.40}} \\ &= \sqrt{0.212} = 0.46 \text{ ft/sec} \end{aligned}$$

$$\text{Gas flow, } Q_A = \frac{M}{\rho_g} = \frac{39.80}{2.07} = 19.2 \text{ ft}^3/\text{sec}$$

Assume a diameter, $D_v = 3.5 \text{ ft}$

$$\begin{aligned} \text{Vessel length, } L &= \frac{4Q_A}{\pi V_t D_v} = \frac{4(19.2)}{\pi(0.46)(3.5)} \\ &= 15.2 \text{ ft} \end{aligned}$$

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?

$$A = \frac{Q_A}{V_t} = \frac{19.2}{0.46} = 41.7 \text{ ft}^2$$

$$D_v = 7.29 \text{ ft minimum}$$

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 12J⁷ 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 carry-over 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

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

$$K = \sqrt{\frac{4_g D_P}{3C'}} \quad \text{Eq 7-11}$$

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

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

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.

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

dimensioning purposes. Residence time establishes the separator volume required for the liquid as shown in Eq 7-12:

$$U = \frac{W(t)}{1440} \quad \text{Eq 7-12}$$

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

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

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.

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

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 must 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:

$$W_{cl} = C^* \left(\frac{S_{hl} - S_{ll}}{\mu} \right) (0.785) D_v^2 \quad \text{Eq 7-13}$$

Horizontal vessel:

$$W_{cl} = C^* \left(\frac{S_{hl} - S_{ll}}{\mu} \right) L_1 H_1 \quad \text{Eq 7-14}$$

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, $W_{cl} = C^*[(S_{hl} - S_{ll}) / \mu] \times 0.785 \times D_v^2$

From Fig. 7-23 for free liquids with water particle diameter = 200 microns, $C^* = 1,100$

$$600 \text{ bbl/day} = 1,100 \times [1.01 - 0.76] / 0.55 \times 0.785 \times D_v^2$$

$$D_v^2 = 660 / 392.5 = 1.53 \text{ ft}^2$$

$$D_v = 1.24 \text{ feet}$$

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

$$V_1 = 650 \times 3 / 1440 = 1.35 \text{ bbl}$$

$$V_1 = 1.35 \times 42 = 56.7 \text{ gallons}$$

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:

$$\text{Shell height} = V_1 / \text{vol/ft} = 56.7 / 13.1 = 4.3 \text{ feet}$$

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/ft².

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 load-

ing 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.

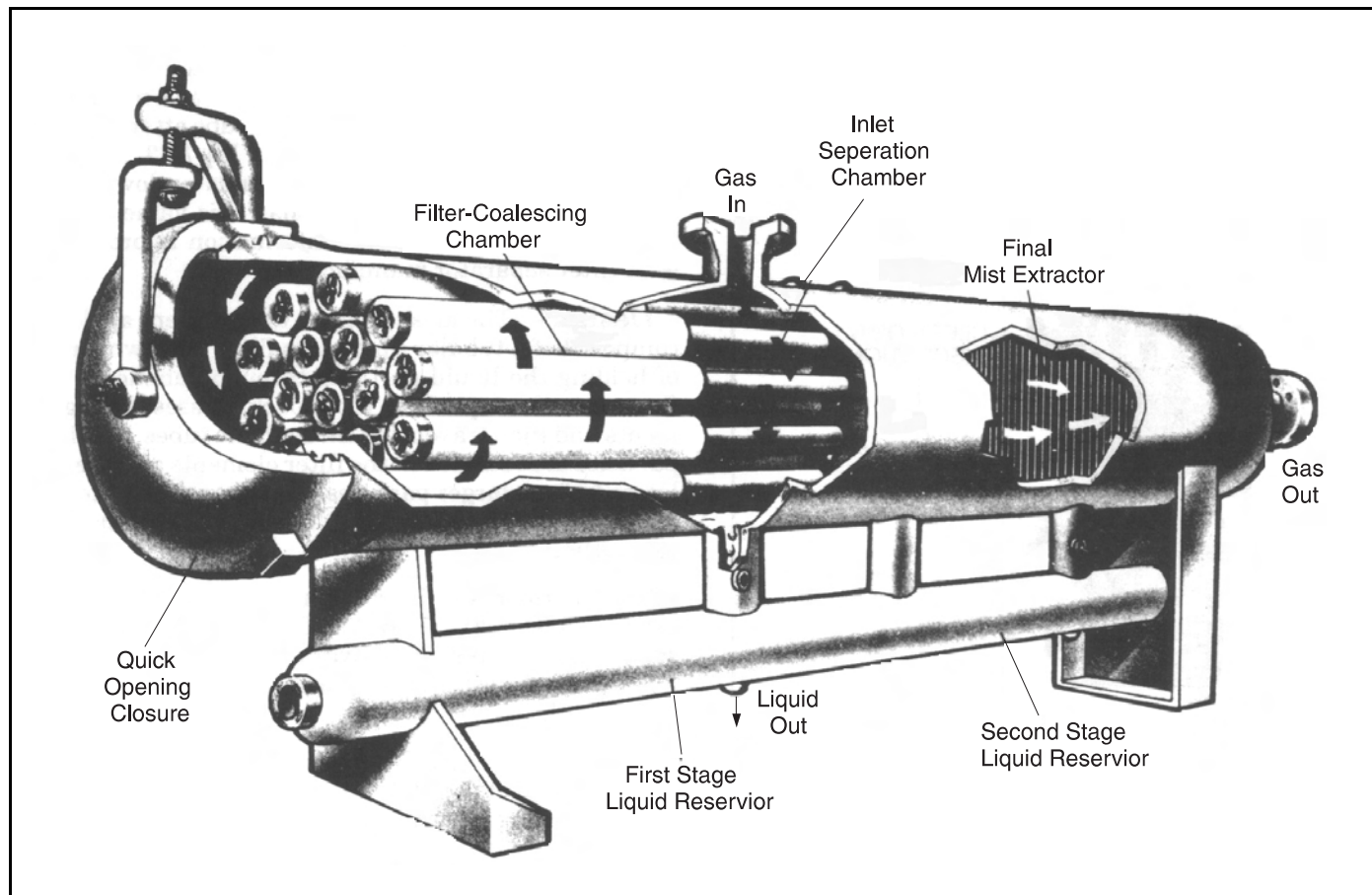
The second stage of a filter separator contains a mist extraction device. As for a conventional separator this may be a mesh pad, vane pack or multicyclone bundle. The same issues regarding mist extractor selection criteria, sizing, etc. apply as discussed previously. Mesh pads and vane pack are most 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

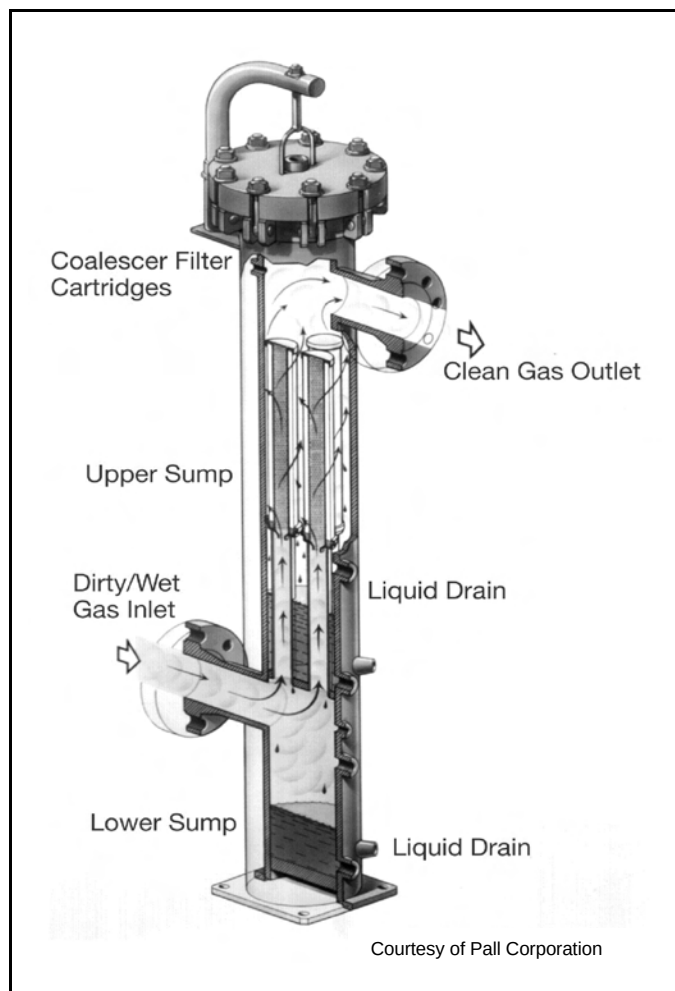


7-25 illustrates a typical coalescing filter. A coalescing filter is typically intended to remove fine liquid aerosols/mist from gas streams where entrained liquid loads are low. A typical filtration rating for coalescing filter elements is 0.3 microns absolute. This means that solid spherical particles 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

FIG. 7-25
Typical Coalescing Filter



commonly refers to the removal of solids and liquids from a gas stream.

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 coarse 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 Ratings¹²

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

given size in the effluent. This results in the following equation for relating the β value to removal efficiency:

$$\% \text{ removal} = (\beta - 1) / \beta \times 100 \quad \text{Eq 7-15}$$

Most “absolute” filters typically have a β of 10,000 (99.99% removal).

Special Applications – Slug Catcher Design¹³

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	L = length of heat transfer surface, ft; or = mean beam length, ft; or = characteristic dimension, ft
A = area, sq ft	LHV = lower or net heating value, Btu/SCF
AO = dry combustion air mols per dry fuel mols, for stoichiometric combustion, mols/mols	m = constant, Eq 8-4 and Eq 8-8
B = parameter defined by Eq 8-21	M = mass flow rate, lb/hr
c = number of carbon atoms in fuel molecule	MPH = velocity, miles/hr
C = constant, Eq 8-4 and Eq 8-8	n = number of fins per foot, number of tube rows
C _d = burner spud discharge coefficient	ni = number of nitrogen atoms in fuel molecule
C _p = specific heat, Btu/(lb • °F)	Nu = Nusselt number, dimensionless
CO = carbon monoxide mols in mols of dry flue gases, mols/mols	NHI = net heat input, Btu/hr
CO ₂ = carbon dioxide mols in mols of dry flue gases, mols/mols	NPS = nominal pipe size, in.
CO _{2o} = carbon dioxide mols in mols of dry flue gases for stoichiometric combustion, mols/mols	NTE = net thermal efficiency, Eq 8-17b
d = diameter of pipe or cylinder or fin, in.	o = number of oxygen atoms in fuel molecule
D = diameter of pipe or cylinder, ft	O ₂ = oxygen mols in mols of dry flue gases, mols/mols
EA = volume percentage of excess combustion air, %	O _{2o} = oxygen mols in mols of dry flue gases for stoichiometric combustion, mols/mols
f _r = Fanning friction factor	P = partial pressure of CO ₂ + H ₂ O, atm
F = view factor, dimensionless	P _B = barometric pressure, in. Hg @ 32°F
Fl _o = mols of dry flue gases per mols of dry fuel for stoichiometric combustion, mols/mols	ΔP = pressure difference, psi
g = acceleration due to gravity = 32.17 ft/sec ²	Pg = burner fuel gas pressure, psia
g _c = 32.17 lb mass • ft/(lb force • sec ²)	Pr = Prandtl number, dimensionless
G = Ratio of air to fuel, lb/lb; or = mass velocity, lb/(sec • sq ft)	q _{gs} = gas flow rate at standard conditions, SCF/day
GHI = gross heat input, Btu/hr	q _l = liquid flow rate, gals./min
GTE = gross thermal efficiency, Eq 8-17a	Q = heat transfer or heat input or heat content (rates), Btu/hr
Gr = Grashof number, dimensionless	r = ratio of flue gases to heat release, lb/MMBtu
h = heat transfer coefficient, Btu/(hr • sq ft • °F)	rf = fouling heat flow resistance, hr • sq ft • °F/Btu
H = heat content or enthalpy, Btu/lb; or = height of stack, ft; height of fin, in.	R = fraction of total heat liberation absorbed in radiant section (Fig. 8-20)
Hav = available draft, inches water column (gauge)	Re = Reynolds number, dimensionless
HHV = higher or gross heating value, Btu/SCF	s = number of sulfur atoms in fuel molecule
hy = number of hydrogen atoms in fuel molecule	S = tube spacing, in.
I = average tube radiant heat flux intensity (circumferential), Btu/(hr • sq ft)	SCF = standard cubic feet at 14.696 psia and 60°F
k = thermal conductivity, Btu/[(hr • sq ft • °F)/ft]	SG = specific gravity
	t = fin thickness, in.
	T = temperature, °R
	ΔT = temperature difference, °F

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:

$$Q = 12 \cdot k \cdot A \cdot \frac{\Delta T}{wt} \quad \text{Eq 8-1}$$

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:

$$Q = \frac{2 \cdot \pi \cdot L \cdot k \cdot \Delta T}{\ln(D_o/D_i)} = \frac{2 \cdot \pi \cdot L \cdot k \cdot \Delta T}{\ln(d_o/d_i)} \quad \text{Eq 8-2a}$$

For radial heat flow through a spherical vessel:

$$Q = \frac{2 \cdot \pi \cdot k \cdot \Delta T}{(1/D_i) + (1/D_o)} = \frac{\pi \cdot k \cdot \Delta T}{(6/d_i) + (6/d_o)} \quad \text{Eq 8-2b}$$

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.

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	
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	12
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	
USG K-FAC 19	1900	0.51	0.89				18.5	33	
Schuller Thermo 12	1200	0.45	0.64				15	160	
PARTEK Paroc 1212	1200	0.42	0.98				12		
Schuller 1000 SpinGlas	850	0.50					3		

NOTES

- Maximum Service Temperature listed has no safety factor included.
- 90-94% Alumina product for extreme temperature or high velocity service.
- Cast properties listed, gunning product available but properties will be different.
- Low silica product.
- For burner blocks and severe service.
- Can be used as one shot (single layer) lining.
- High performance medium weight lining.
- May be used as back-up insulation in 2 layer lining.
- 85% Alumina ramming plastic for burner blocks, etc.
- For external insulation only.
- May be used as hot face lining in dual layer system.
- Diatomaceous earth base.
- Mineral wool base.
- Calcium silicate base.
- Fiber glass base.
- Convert to Btu /[(hr • ft² • °F) /ft] before using in equations in this section.

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:

$$T_i = 250^\circ\text{F}$$

$$T_o = 50^\circ\text{F}$$

$$k = 0.0417 \text{ Btu}/[(\text{hr} \cdot \text{sq ft} \cdot ^\circ\text{F})/\text{ft.}]$$

(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

$$\begin{aligned} Q &= \frac{2 \cdot \pi \cdot k \cdot \Delta T}{\ln(d_o/d_i)} \\ &= \frac{2 \cdot \pi (0.0417)(250-50)}{\ln(10.625/8.625)} \\ &= 251.2 \text{ Btu/hr per linear ft} \end{aligned}$$

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

$$Q = h \cdot A \cdot \Delta T \quad \text{Eq 8-3}$$

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.

$$Nu = C (Gr \cdot Pr)^m \quad \text{Eq 8-4}$$

Where

$$Nu = \frac{h \cdot D_o}{k} = \frac{h \cdot d_o}{12 \cdot k} \quad \text{Eq 8-5a}$$

$$\text{or } Nu = \frac{h \cdot L}{k} \quad \text{Eq 8-5b}$$

$$\begin{aligned} Gr &= \frac{2.214 \cdot 10^6 \cdot D_o^3 \cdot \rho^2 \cdot g \cdot \beta \cdot \Delta T}{\mu^2} \\ &= \frac{1281 \cdot d_o^3 \cdot \rho^2 \cdot g \cdot \beta \cdot \Delta T}{\mu^2} \end{aligned} \quad \text{Eq 8-6a}$$

$$\text{or } Gr = \frac{2.214 \cdot 10^6 \cdot L^3 \cdot \rho^2 \cdot g \cdot \beta \cdot \Delta T}{\mu^2} \quad \text{Eq 8-6b}$$

$$Pr = \frac{2.419 \cdot C_p \cdot \mu}{k} \quad \text{Eq 8-7}$$

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

FIG. 8-4

**Heat Transfer Constants² 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 Facing Up Facing Down	$10^5 < Y < 2(10^7)$	0.54	0.25
	$2(10^7) < Y < 3(10^{10})$	0.14	0.33
	$3(10^5) < Y < 3(10^{10})$	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

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.

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

$$T_{\text{film}} = (190 + 114)/2 = 152^\circ\text{F} (612^\circ\text{R})$$

From Eq. 8-4, 8-5a, 8-6a and 8-7 and Fig. 8-4:

$$\begin{aligned} Nu &= 0.47 (Gr \cdot Pr)^{0.25} \\ \frac{h \cdot d_o}{12 \cdot k} &= 0.47 \left[\frac{1281 \cdot d_o^3 \cdot \rho^2 \cdot g \cdot \beta \cdot \Delta T \cdot C_p \cdot 2.419}{\mu \cdot k} \right]^{0.25} \end{aligned}$$

$$d_o = 3.5 \text{ in.}$$

$$\rho = 61.8 \text{ lb/cu ft}$$

$$g = 32.17 \text{ ft/sec}^2$$

$$\beta = 0.00035^\circ\text{F}^{-1}$$

$$\Delta T = 190 - 114 = 76^\circ\text{F}$$

$$C_p = 1.00 \text{ Btu}/(\text{lb} \cdot ^\circ\text{F})$$

$$\mu = 0.63 \text{ cp}$$

$$k = 0.365 \text{ Btu}/[(\text{hr} \cdot \text{sq ft} \cdot ^\circ\text{F})/\text{ft}]$$

The properties of water (ρ, β, μ, k) are based on the film temperature.

Substituting:

$$\begin{aligned} h &= \frac{(12)(0.47)(0.365)}{3.5} \left[\frac{1281(3.5)^3(61.8)^2(32.17)(0.00035)(76)(1.0)(2.419)}{(0.63)(0.365)} \right]^{0.25} \\ &= 0.588 [1.889 (10^9)]^{0.25} \\ &= 122.6 \text{ Btu}/(\text{hr} \cdot \text{sq ft} \cdot ^\circ\text{F}) \end{aligned}$$

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:

$$Nu = C \cdot Re^m \cdot Pr^{0.33} \quad \text{Eq 8-8a}$$

Utilization of a viscosity correction term gives the Sieder-Tate correlation:

$$Nu = C \cdot Re^m \cdot Pr^{0.33} (\mu_b/\mu_w)^{0.14} \quad \text{Eq 8-8b}$$

Where

$$\begin{aligned} Re &= \frac{1488 \cdot D \cdot V \cdot \rho}{\mu} = \frac{124 \cdot d \cdot V \cdot \rho}{\mu} \quad \text{Eq 8-9a} \\ &= \frac{0.5263 \cdot M}{D \cdot \mu} = \frac{6.316 \cdot M}{d \cdot \mu} \\ &= \frac{4.221 \cdot q_L \cdot \rho}{D \cdot \mu} = \frac{50.66 \cdot q_L \cdot \rho}{d \cdot \mu} \\ &= \frac{q_{gs} \cdot SG}{598 \cdot D \cdot \mu} = \frac{0.02007 \cdot q_{gs} \cdot SG}{d \cdot \mu} \end{aligned}$$

or

$$Re = \frac{1488 \cdot L \cdot V \cdot \rho}{\mu} \quad \text{Eq 8-9b}$$

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

$$T_{\text{film}} = (104 + 72)/2 = 88^\circ\text{F}$$

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)

$q_{gs} = 10$ (10⁶) scfd

$$Re = \frac{0.02 \cdot q_{gs} \cdot SG}{d \cdot \mu} =$$

$$= \frac{0.02 (10) (10^6) 0.6}{2.9 (0.019)} = 2.18 (10^6)$$

$$Pr = \frac{2.419 \cdot C_p \cdot \mu}{k} = \frac{2.419 \cdot 0.73 \cdot 0.019}{0.03}$$

$$= 1.118$$

$$h_i = \frac{(0.023) (12) (0.03)}{2.9} [2.18 (10^6)]^{0.8} (1.118)^{0.33}$$

$$= 349.9 \text{ Btu / (hr • sq ft • °F)}$$

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} .

$$Q = U \cdot A \cdot \Delta T_{LM}$$

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

$$U_o = \frac{1}{\frac{1}{h_o} + \frac{D_o}{D_i \cdot h_i} + \frac{D_o \cdot \ln(D_o/D_m)}{2 \cdot k_o} + \frac{D_o \cdot \ln(D_m/D_i)}{2 \cdot k_i} + r_{f_o} + r_{f_i} \cdot \frac{D_o}{D_i}}$$

$$= \frac{1}{\frac{1}{h_o} + \frac{d_o}{d_i \cdot h_i} + \frac{d_o \cdot \ln(d_o/d_m)}{24 \cdot k_o} + \frac{d_o \cdot \ln(d_m/d_i)}{24 \cdot k_i} + r_{f_o} + r_{f_i} \cdot \frac{d_o}{d_i}}$$

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

$$U_o = \frac{1}{\left[\frac{1}{122.6} + \left(\frac{3.50}{2.90 (349.9)} \right) + \frac{3.50 \cdot \ln(3.50/2.90)}{24 (26.2)} \right]}$$

$$U_o = 79.0 \text{ Btu}/(\text{hr} \cdot \text{sq ft} \cdot ^\circ\text{F})$$

$$Q = U_o A_o \Delta T_{LM} = 79.0 \left(\pi \frac{3.5}{12} \right) \frac{(190 - 60) - (190 - 84)}{\ln \left(\frac{130}{106} \right)}$$

$$= 8516 \text{ Btu/hr per linear ft}$$

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,

$$1/U_o = 0.01265 + 0.001 + \left(\frac{3.50}{2.90} \right) (0.001)$$

$$U_o = 67.3 \text{ Btu}/(\text{hr} \cdot \text{sq ft} \cdot ^\circ\text{F})$$

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:

$$A_o = \frac{\pi \cdot d_o}{12} (1 - n \cdot t/12) + \left(\frac{n \cdot \pi}{2} \right) \left(\frac{d_f^2 - d_o^2}{144} \right) \quad \text{Eq 8-12}$$

and

$$A_{cs} = \frac{d_o}{12} + \frac{n \cdot t (d_f - d_o)}{144} \quad \text{Eq 8-13}$$

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

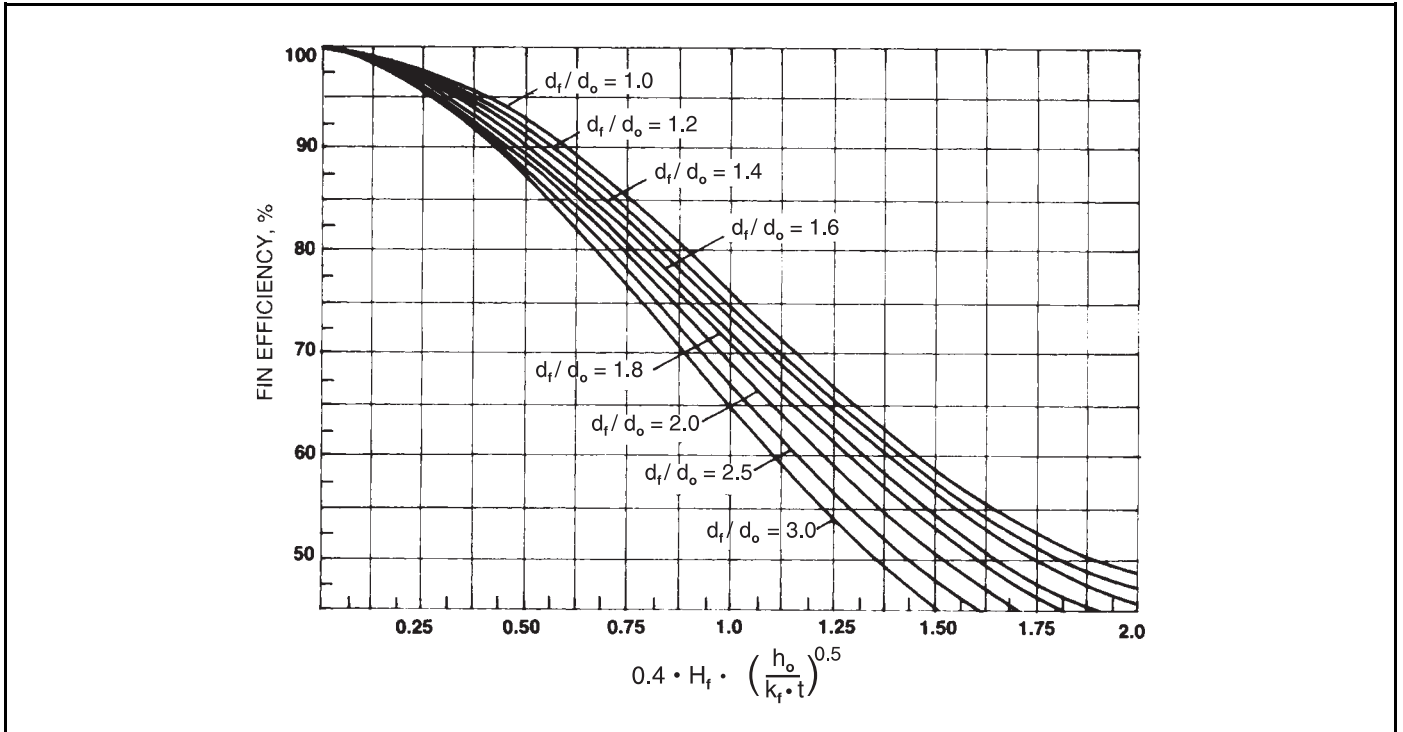
An estimate of the radiant heat flux between two surfaces is:

$$\frac{Q}{A} = \frac{\sigma \cdot F (T_1^4 - T_2^4)}{(1/\epsilon_1 + 1/\epsilon_2 - 1)} \quad \text{Eq 8-14}$$

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 recommended⁶ 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 ($P \cdot L$). See Figs. 8-10, 8-11, and 8-12.

FIG. 8-6
Fin Efficiency Chart⁴



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, $P_{\text{CO}_2 + \text{H}_2\text{O}} = 0.24$

From Fig. 8-11, $L = 2.0$ ft, so $P \cdot L = 0.48$

From Fig. 8-12, $\epsilon_1 = 0.12$

From Fig. 8-9, $\epsilon_2 = 0.79$ (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 Temperature⁵

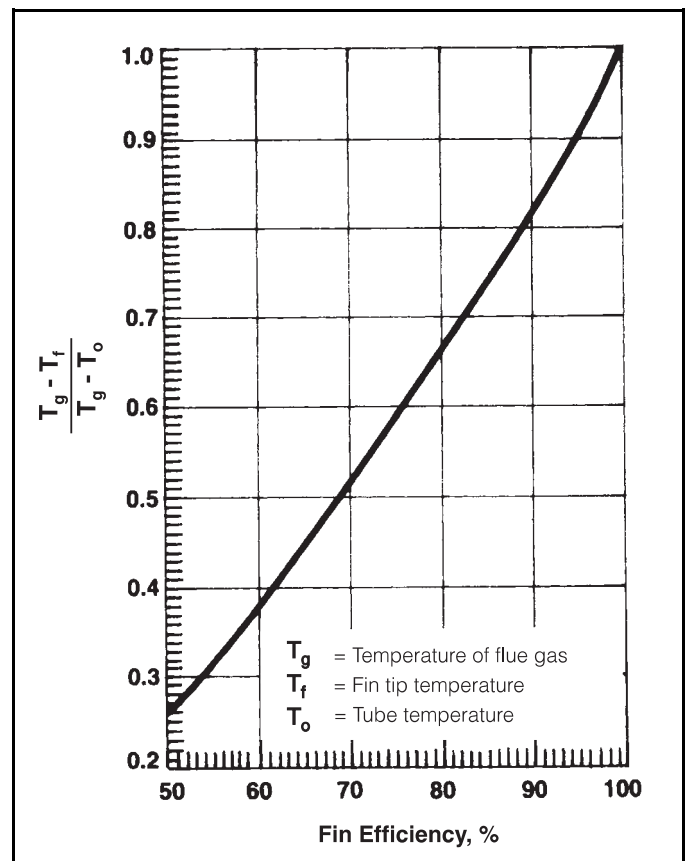


FIG. 8-8
Thermal Conductivity of Ferrous Materials⁴

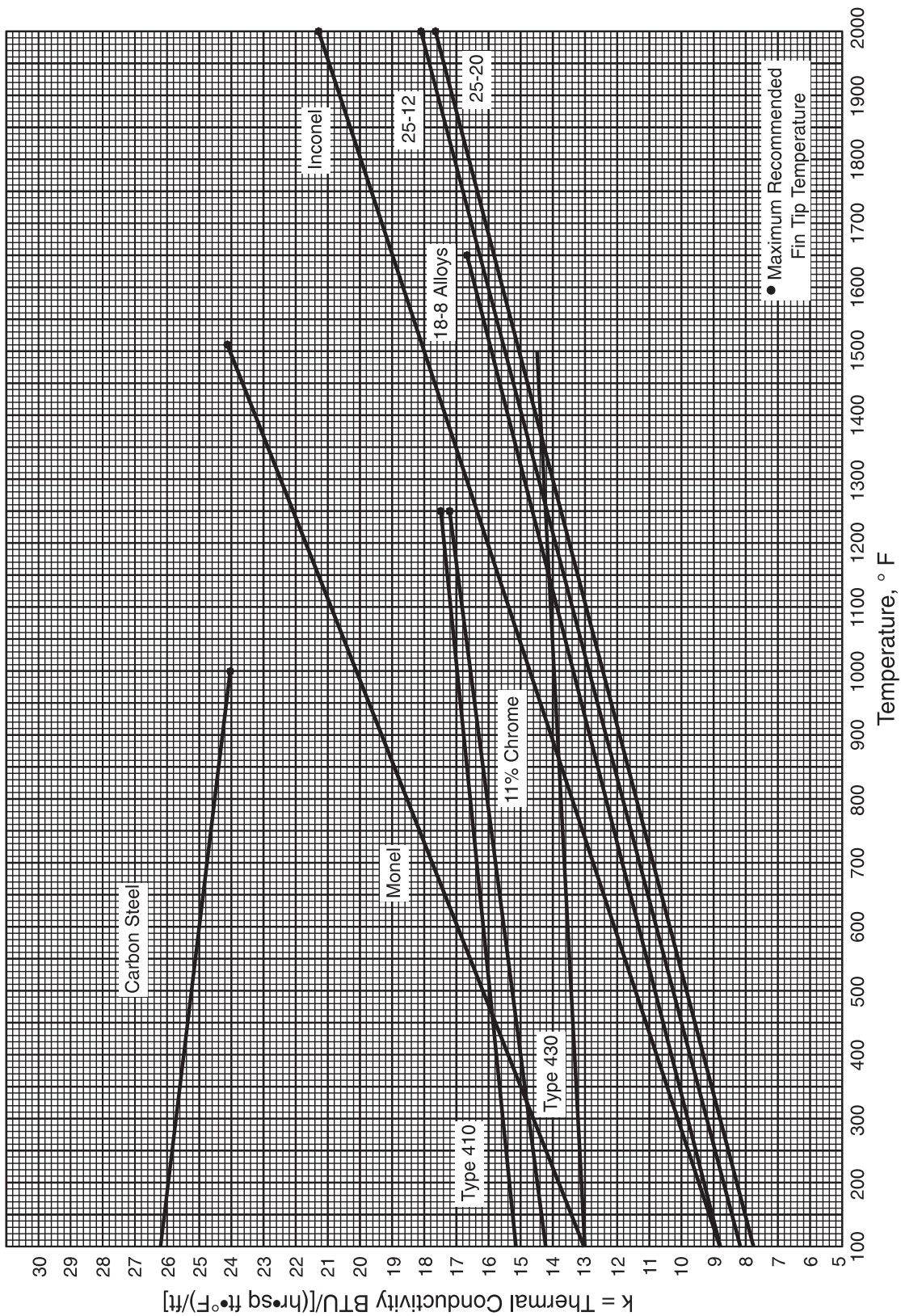


FIG. 8-9

Normal Total Emissivity of Various Surfaces³

A. Metals and Their Oxides

Surface	T, °F*	Emissivity*	Surface	T, °F*	Emissivity*
Aluminum			Steel, oxidized at 1100°F	390-1110	0.79
Highly polished plate, 98.3% pure	440-1070	0.039-0.057	Iron oxide	930-2190	0.85-0.89
Polished plate	73	0.040	Sheet steel, strong rough oxide layer	75	0.80
Rough plate	78	0.055	Dense shiny oxide layer	75	0.82
Oxidized at 1110°F	390-1110	0.11-0.19	Cast plate:		
Aluminum-surfaced roofing	100	0.216	Smooth	73	0.80
Calorized surfaces, heated at 1110°F:			Rough	73	0.82
Copper	390-1110	0.18-0.19	Cast iron, rough, strongly oxidized	100-480	0.95
Steel	390-1110	0.52-0.57	Wrought iron, dull oxidized	70-680	0.94
Brass			Steel plate, rough	100-700	0.94-0.97
Highly polished:			High temperature alloy steels		
73.2% Cu, 26.7% Zn	476-674	0.028-0.031	(see Nickel Alloys)		
62.4% Cu, 36.8% Zn, 0.4% Pb, 0.3% Al	494-710	0.033-0.037	Monel metal, oxidized at 1110°F	390-1110	0.41-0.46
82.9% Cu, 17.0% Zn	530	0.030	Nickel		
Chromium; (see Nickel Alloys for Ni-Cr steels)	100-1000	0.08-0.26	Electroplated on pickled iron, not polished	68	0.11
Copper			Plate, oxidized by heating at 1110°F	390-1110	0.37-0.48
Commercial, emiered, polished, but pits remaining	66	0.030	Nickel oxide	1200-2290	0.59-0.86
Commercial, scraped shiny but not mirror-like	72	0.072	Nickel alloys		
Polished	242	0.023	Chromnickel	125-1894	0.64-0.76
Plate, heated long time, covered with thick oxide layer	77	0.78	Nickelin (18-32 Ni; 55-68 Cu; 20 Zn), gray oxidized	70	0.262
Cuprous oxide	1470-2010	0.66-0.54	KA-2S alloy steel – (8% Ni; 18% Cr), light silvery, rough, brown, after heating	420-914	0.44-0.36
Iron and steel			After 42 hr. heating at 980°F	420-980	0.62-0.73
Metallic surfaces (or very thin oxide layer):			NCT-3 alloy (20% Ni; 25% Cr), brown, spotted, oxidized from service	420-980	0.90-0.97
Cast iron, polished	392	0.21	NCT-6 alloy (60% Ni; 12% Cr), smooth, black, firm adhesive oxide coat from service	520-1045	0.89-0.82
Ground sheet steel	1720-2010	0.55-0.61	Tin – bright tinned iron sheet	76	0.043 & 0.064
Smooth sheet iron	1650-1900	0.55-0.60	Zinc		
Oxidized surfaces:			Commercial, 99.1% pure, polished	440-620	0.045-0.053
Iron plate, pickled, then rusted red	68	0.612	Oxidized by heating at 750°F	750	0.11
Completely rusted	67	0.685	Galvanized sheet iron, fairly bright	82	0.228
Rolled sheet steel	70	0.657	Galvanized sheet iron, gray oxidized	75	0.276
Oxidized iron	212	0.736			
Cast iron, oxidized at 1100°F	390-1110	0.64-0.78			

B. Refractories, Building Materials, Paints, and Miscellaneous

Surface	T, °F*	Emissivity*	Surface	T, °F*	Emissivity*
Asbestos			Paints, lacquers, varnishes		
Board	74	0.96	Snowwhite enamel varnish on rough iron plate	73	0.906
Paper	100-700	0.93-0.945	Black shiny lacquer, sprayed on iron	76	0.875
Brick			Black shiny shellac on tinned iron sheet	70	0.821
Red, rough, but no gross irregularities	70	0.93	Black matte shellac	170-295	0.91
Silica, unglazed, rough	1832	0.80	Black lacquer	100-200	0.80-0.95
Silica, glazed, rough	2012	0.85	Flat black lacquer	100-200	0.96-0.98
Grog brick, glazed	2012	0.75	White lacquer	100-200	0.80-0.95
See Refractory Materials below.			Oil paints, sixteen different, all colors	212	0.92-0.96
Carbon			Aluminum paints and lacquers		
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	10% Al, 22% lacquer body, on rough or smooth surface	212	0.52
Enamel, white fused, on iron	66	0.897	26% Al, 27% lacquer body, on rough or smooth surface	212	0.3
Glass, smooth	72	0.937	Other Al paints, varying age and Al content	212	0.27-0.67
Gypsum, 0.02 in. thick on smooth or blackened plate	70	0.903	Al lacquer, varnish binder, on rough plate	70	0.39
Marble, light gray, polished	72	0.931	Al paint, after heating to 620°F	300-600	0.35
Oak, planed	70	0.895	Plaster, rough lime	50-190	0.91
Oil layers on polished nickel (lube oil)	68		Porcelain, glazed	72	0.924
Polished surface, alone		0.045	Quartz, rough, fused	70	0.932
+0.001-in. oil		0.27	Refractory materials, 40 different poor radiators	1110-1830	0.65-0.75
+0.002-in. oil		0.46	good radiators		0.70
+0.005-in. oil		0.72			0.80-0.85
Infinitely thick oil layer		0.82			0.85-0.90
Oil layers on aluminum foil (linseed oil)			Roofing paper	69	0.91
Al foil	212	0.087†	Rubber		
+1 coat oil	212	0.561	Hard, glossy plate	74	0.945
+2 coats oil	212	0.574	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.

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:

$$h_c + h_r = 4.5 \text{ Btu/(hr} \cdot \text{sq ft} \cdot ^\circ\text{F)}$$

$$Q = (4.5) \frac{(\pi 10.625)}{12} (50 - 30) = 250 \text{ Btu/hr per linear ft}$$

This agrees closely enough with the heat flow through the insulation — Example 8-1.

$$\text{Heat saved} = 2906 - 250 = 2656 \text{ Btu/hr per linear ft}$$

FIG. 8-10

Partial Pressure of CO₂ Plus H₂O

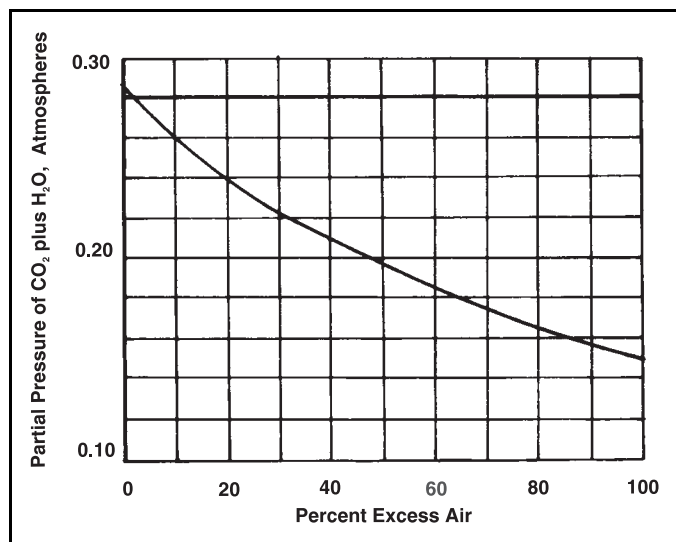


FIG. 8-11

Beam Length for Gas Radiation⁸

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	$(2/3) (\text{Furnace Volume})^{1/3}$
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	$(2/3) (\text{Diameter})$
1-2 to 1-∞	(1) (Diameter)

FIG. 8-12

Gas Emissivity¹⁰

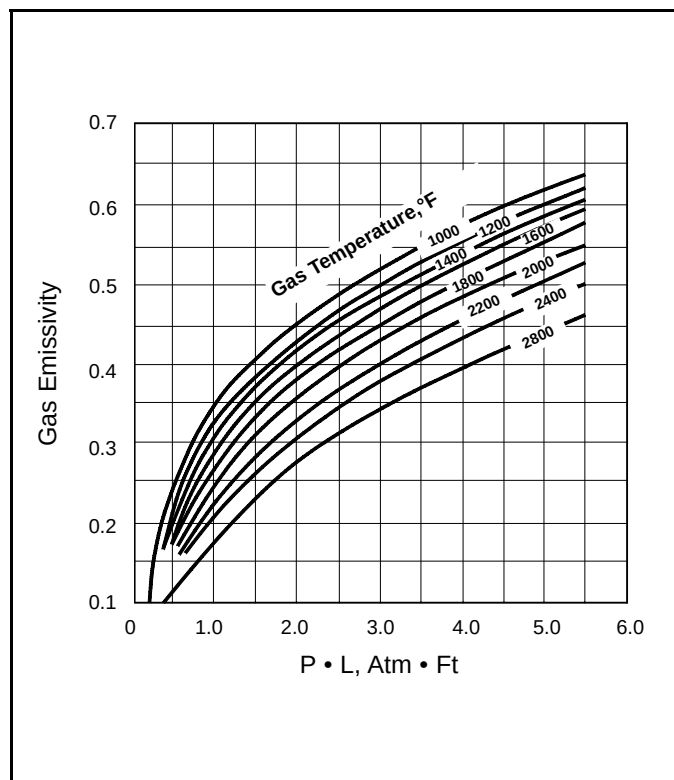
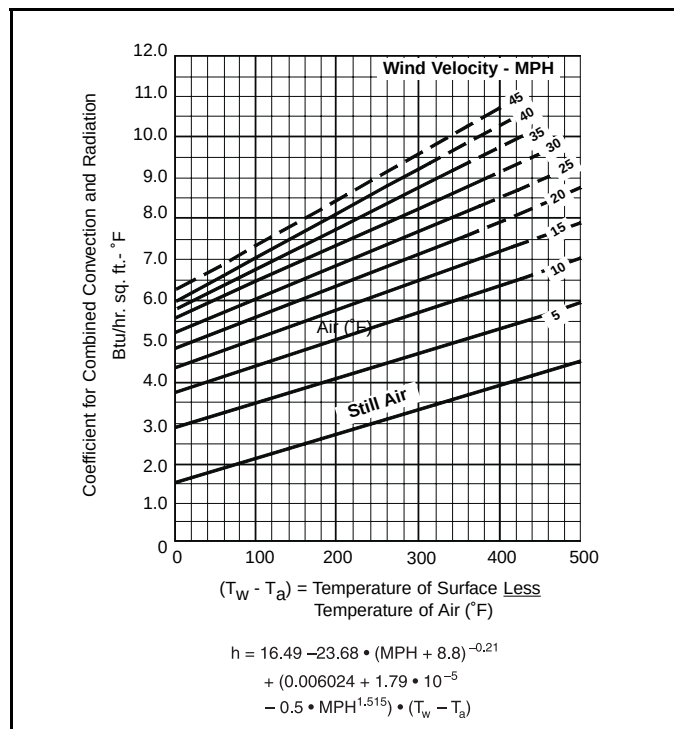


FIG. 8-13

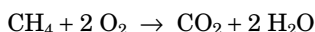
Combination Convection and Radiation Film Coefficients for Air in Contact with Vertical Walls or Surfaces¹¹



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.



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.

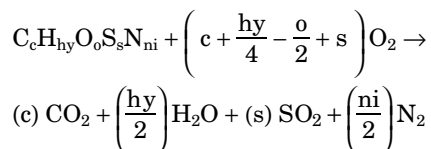


Fig. 8-14 shows how the composition of the flue gases depends on the amount of combustion air.

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.

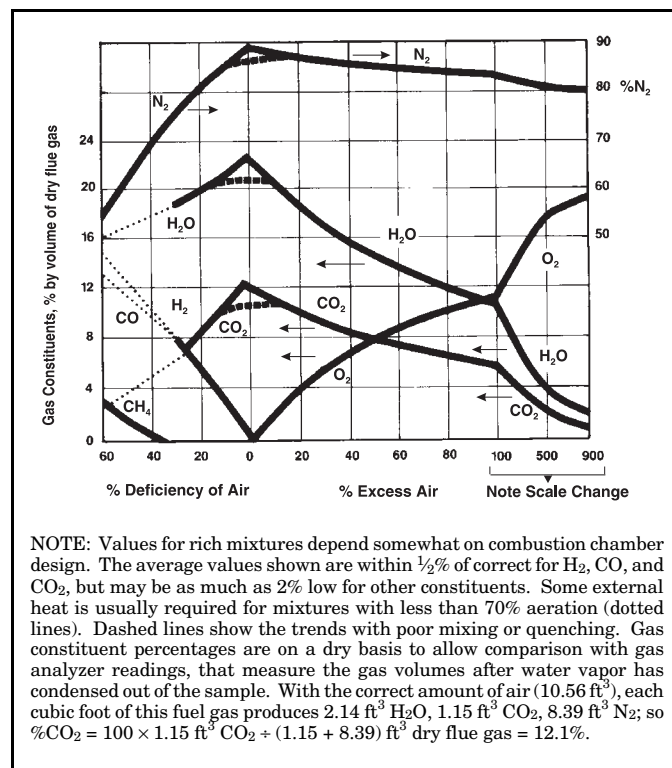
Air is about 20.9% oxygen on a dry basis, hence 4.77 mols (or scf) of air supply 1.0 mol (or scf) of oxygen. Applying this to methane, 9.54 mol (or scf) of air are needed for every mol (or scf) of methane.

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



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.

$$\text{AO} = 4.77 \cdot \left[c + \frac{hy}{4} - \frac{o}{2} + s \right] \quad \text{Eq 8-15}$$

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.

$$\text{EA} = \frac{\text{Fl}_o}{\text{AO}} \left[\frac{\text{CO}_{2o}}{\text{CO}_2 + \text{CO}} \right] \cdot 100$$

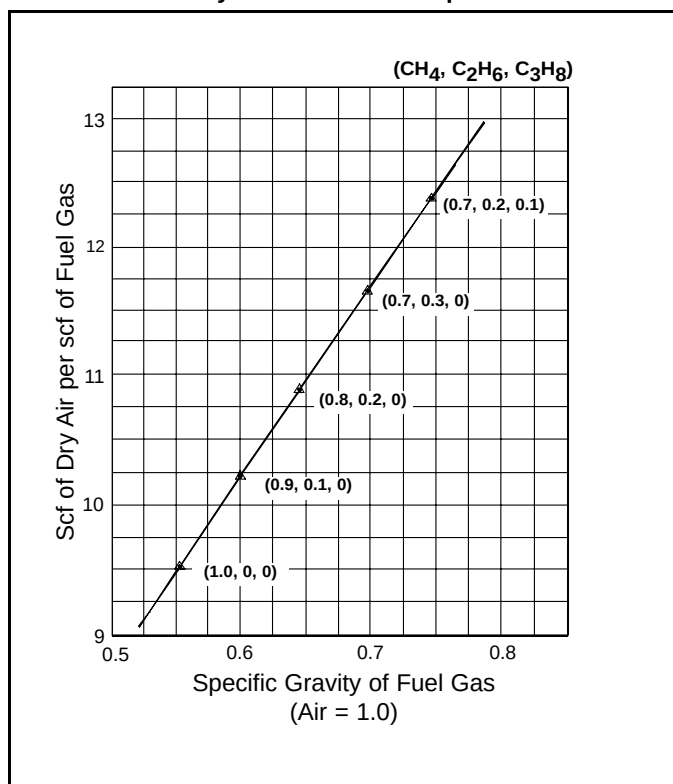
$$= \frac{\text{Fl}_o}{\text{AO}} \left[\frac{\text{O}_{2o}}{20.9 - \text{O}_2 + \text{CO}} \right] \cdot 100 \quad \text{Eq 8-16}$$

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

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:

$$\text{HHV} = 1568.72 \cdot \text{SG} - 2524.88 \cdot X_{\text{CO}_2} - 1658.37 \cdot X_{\text{N}_2} + 141.05$$

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

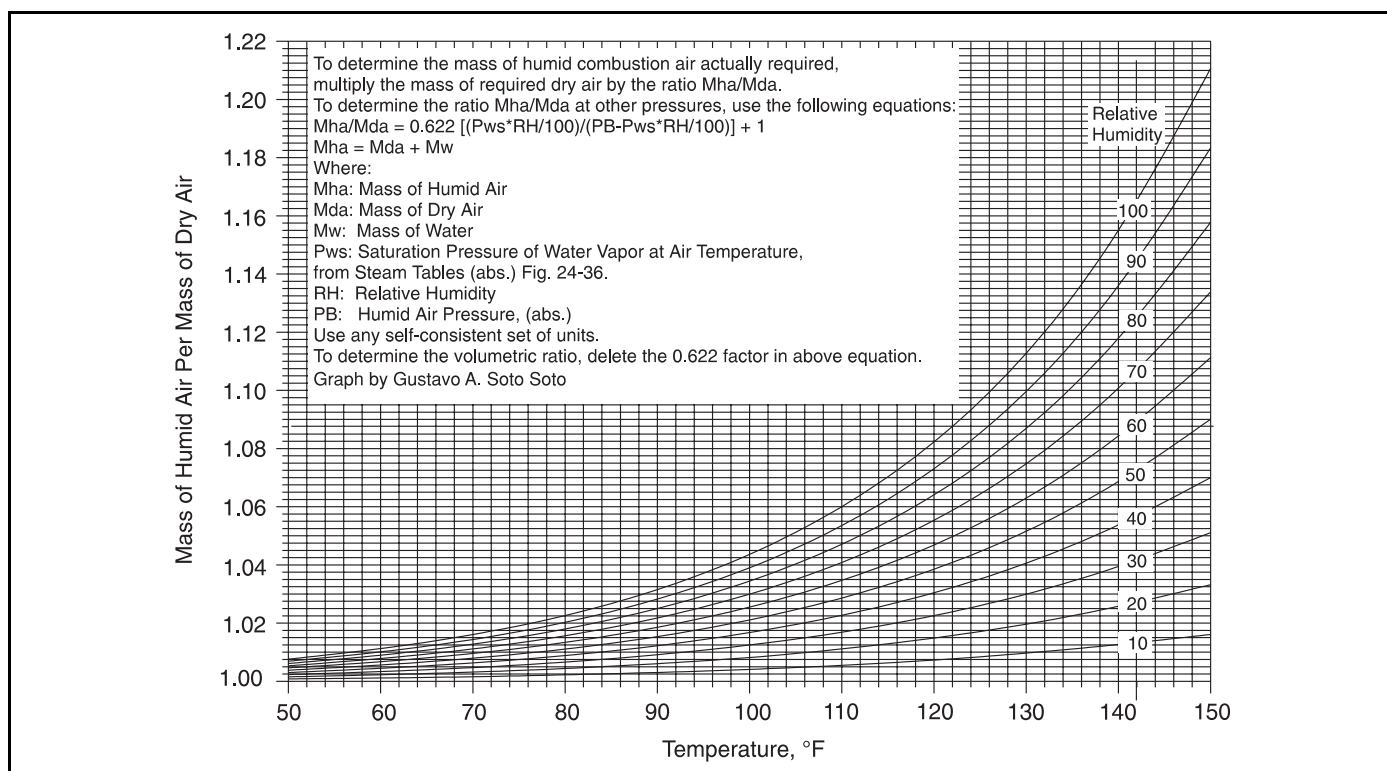


FIG. 8-16

Humid Air Ideal Density

760 mm Hg Air Pressure (Absolute) and Indicated Dry-Bulb Temperature and Relative Humidity

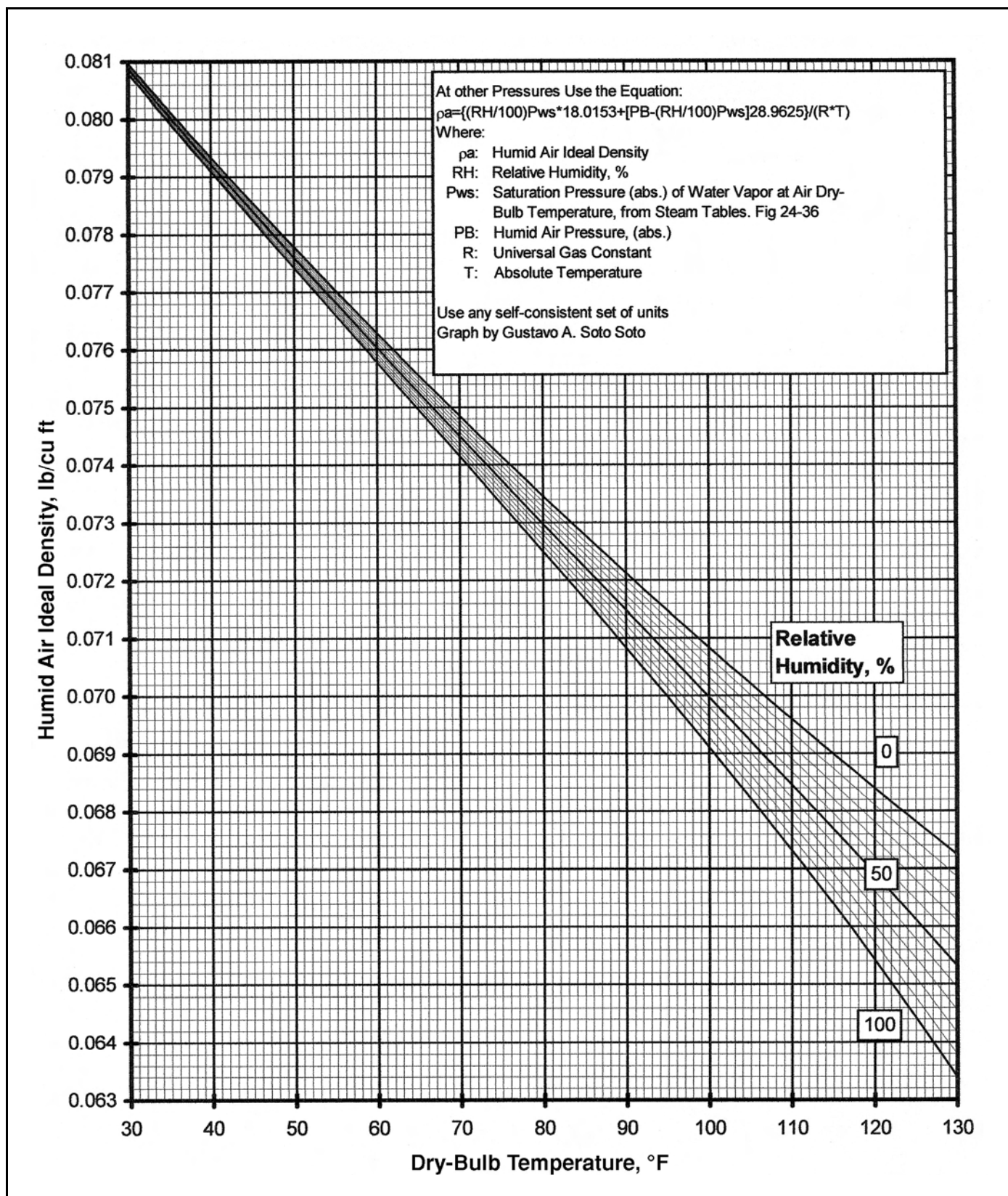
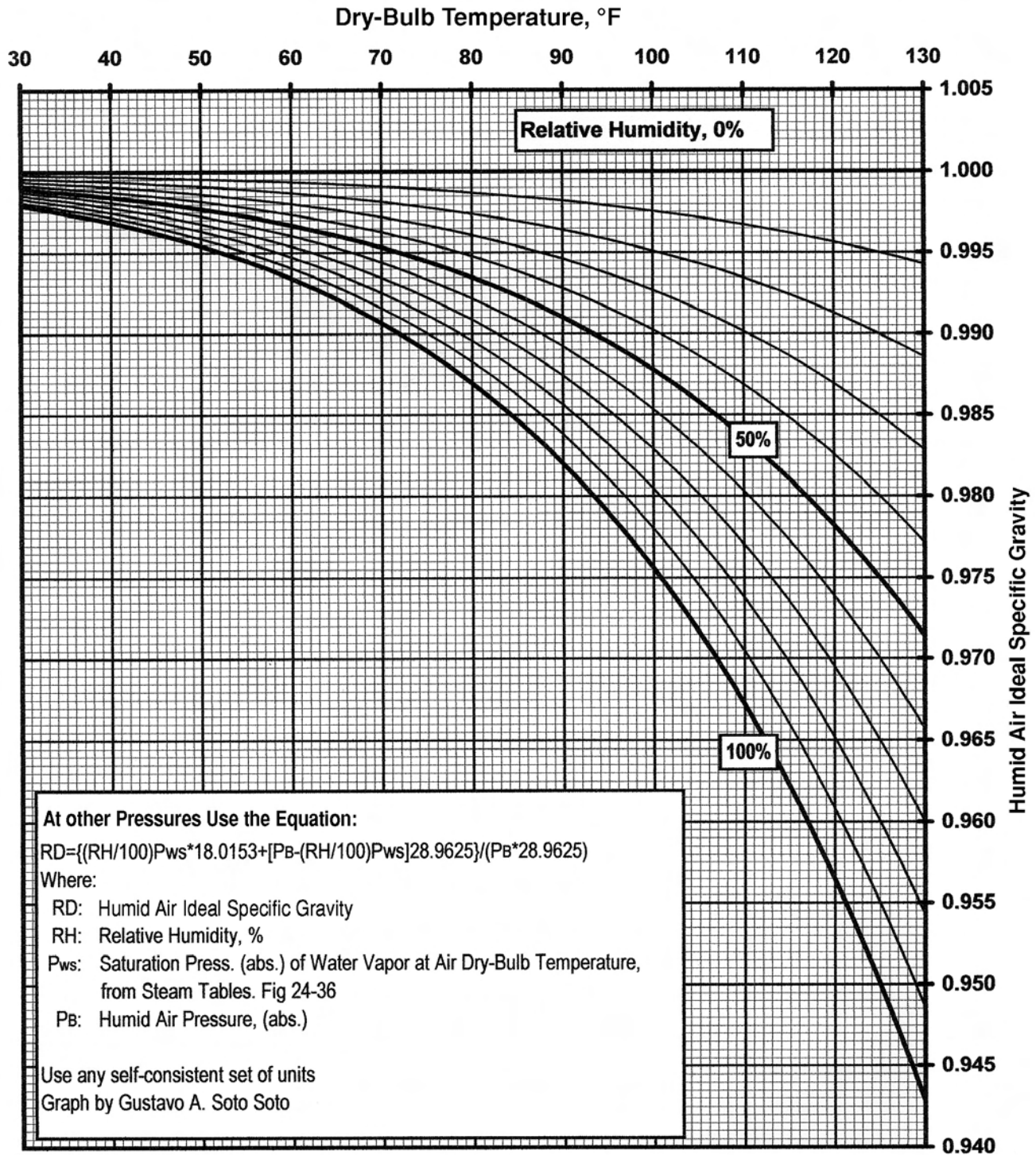


FIG. 8-17

Humid Air Ideal Specific Gravity
760 mm Hg Air Pressure (Absolute) and Indicated Dry-Bulb Temperature and Relative Humidity



Thermal Efficiency

The two ways to express the heat released during combustion result in two definitions for the thermal efficiency, gross and net.

$$\text{GTE} = \frac{\text{UHT}}{\text{GHI}} \quad \text{Eq 8-17a}$$

$$\text{NTE} = \frac{\text{UHT}}{\text{NHI}} \quad \text{Eq 8-17b}$$

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

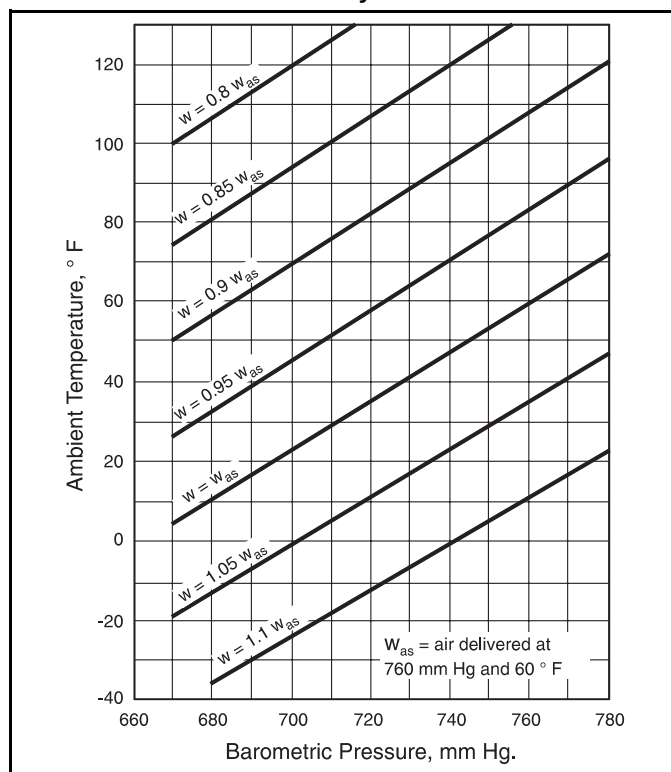


FIG. 8-19

Gross Thermal Efficiency for a Gas with HHV = 1000 Btu/scf

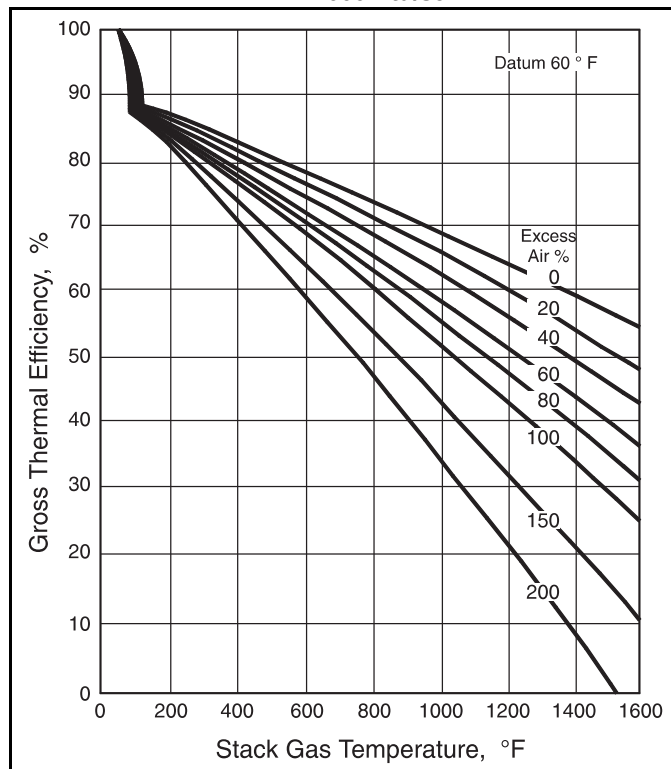
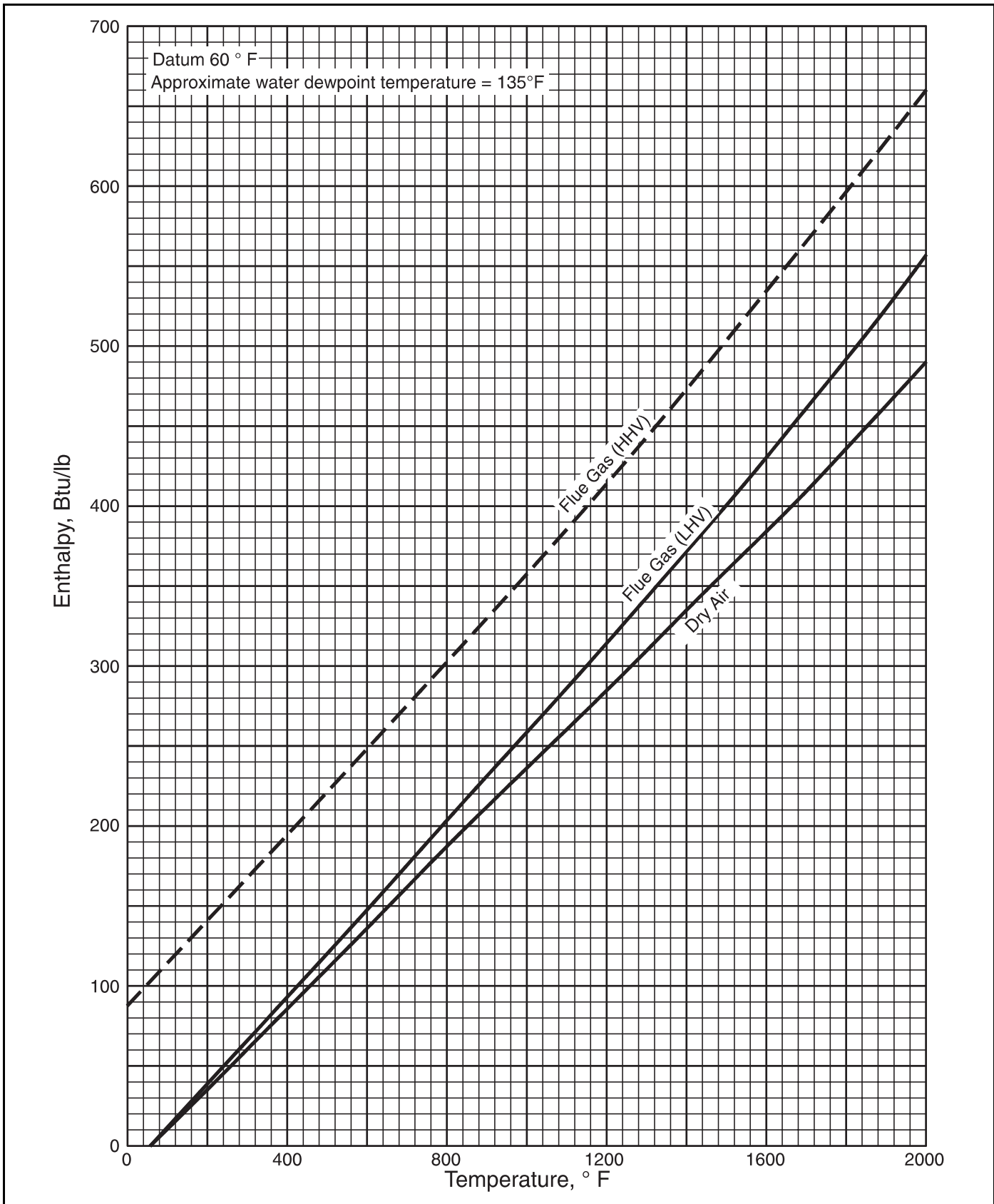


FIG. 8-20

Typical Enthalpy of Combustion Gases for a Dry Natural Gas Fuel and 20% Excess Dry Air



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).

$$\begin{aligned} H_{av} = & 0.192 \cdot H_s (\rho_g - \rho_a) \\ & + 0.0029 \cdot V^2 \cdot \rho_g \left(4 \cdot f_f \cdot \frac{H_s}{D_s} + 1 \right) \end{aligned} \quad \text{Eq 8-18}$$

Also

$$\begin{aligned} H_{av} = & 0.2545 \cdot H_s \cdot P_B \cdot \left(\frac{SG_g}{T_g} - \frac{SG_a}{T_a} \right) + \\ & + \frac{2.737 \cdot T_g}{P_B \cdot SG_g \cdot D_s^4} \cdot \left(\frac{M}{100,000} \right)^2 \left(4 \cdot f_f \cdot \frac{H_s}{D_s} + 1 \right) \end{aligned} \quad \text{Eq 8-19}$$

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.

$$\frac{GHI_2}{GHI_1} = \frac{Y_2 \cdot \sqrt{\frac{P_{g2} \cdot \Delta P_2}{T_{g2}}} \cdot \left(\frac{HHV_2}{\sqrt{SG_2}} \right)}{Y_1 \cdot \sqrt{\frac{P_{g1} \cdot \Delta P_1}{T_{g1}}} \cdot \left(\frac{HHV_1}{\sqrt{SG_1}} \right)} \quad \text{Eq 8-20}$$

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 SO₂ when present in the flue gas because of the sulfur content of some fuels, and therefore temperature control of the flue gas above the SO₂ dew-point 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.

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:

$$\frac{1000}{T_{DP}} = 3.9526 - 0.1863 \cdot \ln(P_{H_2O}) + 0.000867 \cdot \ln(P_{SO_2}) - 0.000913 \cdot \ln(P_{H_2O}) \cdot \ln(P_{SO_2})$$

For sulfuric acid:

$$\frac{1000}{T_{DP}} = 2.276 - 0.0294 \cdot \ln(P_{H_2O}) - 0.0858 \cdot \ln(P_{H_2SO_4}) - 0.0062 \cdot \ln(P_{H_2O}) \cdot \ln(P_{H_2SO_4})$$

For nitric acid:

$$\frac{1000}{T_{DP}} = 3.6614 - 0.1446 \cdot \ln(P_{H_2O}) - 0.0827 \cdot \ln(P_{HNO_3}) + 0.00756 \cdot \ln(P_{H_2O}) \cdot \ln(P_{HNO_3})$$

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 Fuels⁴²

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 560²³ 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.

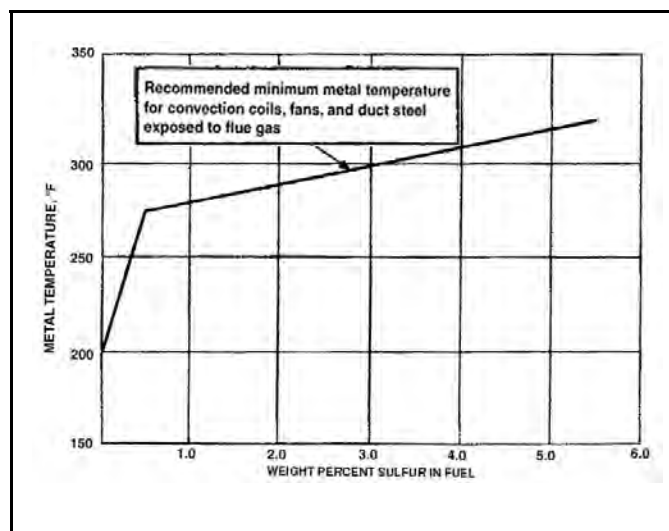
NO_x Control

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 gener-

FIG. 8-21

Recommended Minimum Metal Temperatures²³



ally 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% O₂, 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

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:

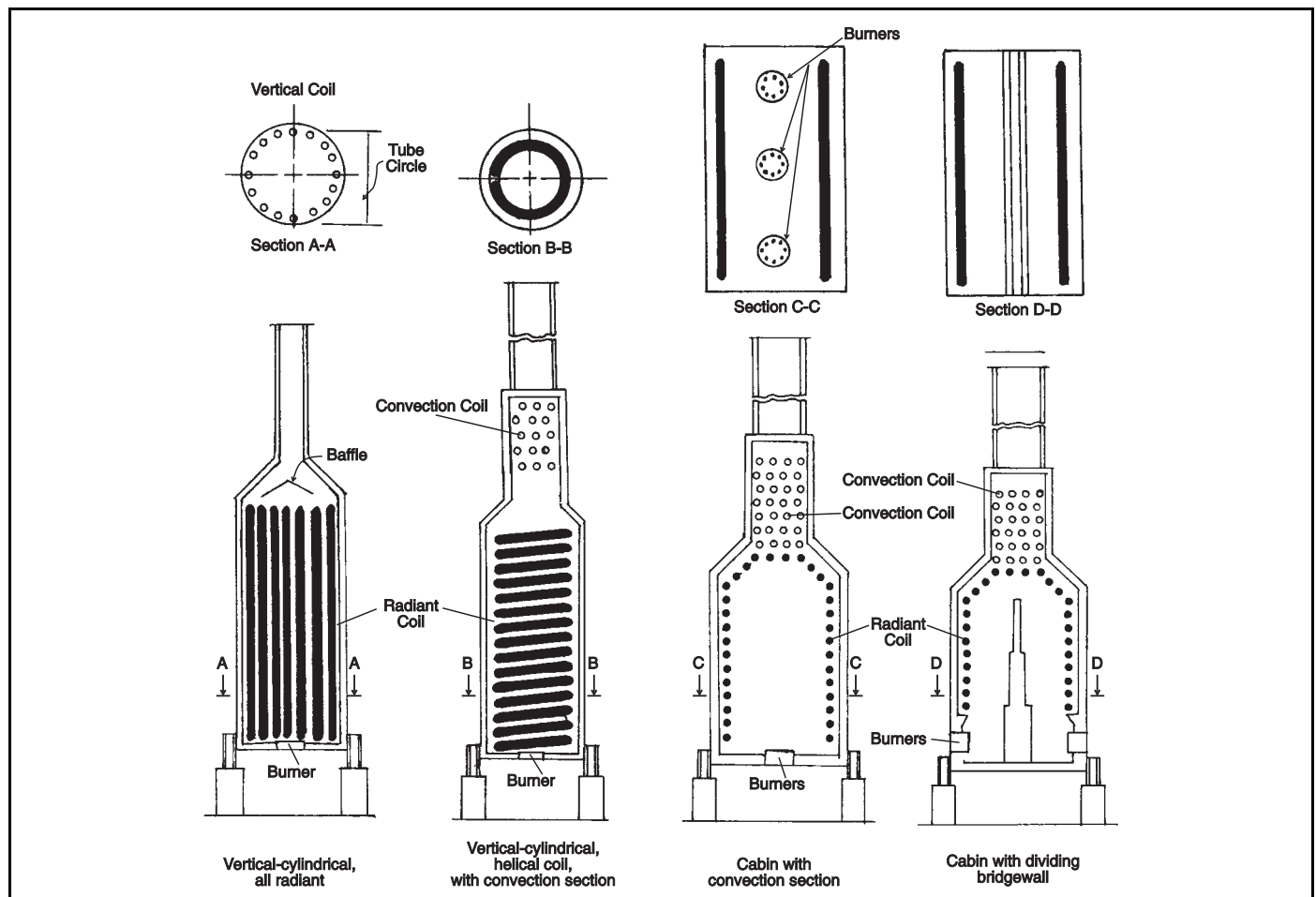
- They require the smallest plot area for a given duty.
- The cost is usually 10% to 15% lower in the larger sizes.

- 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



- 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.

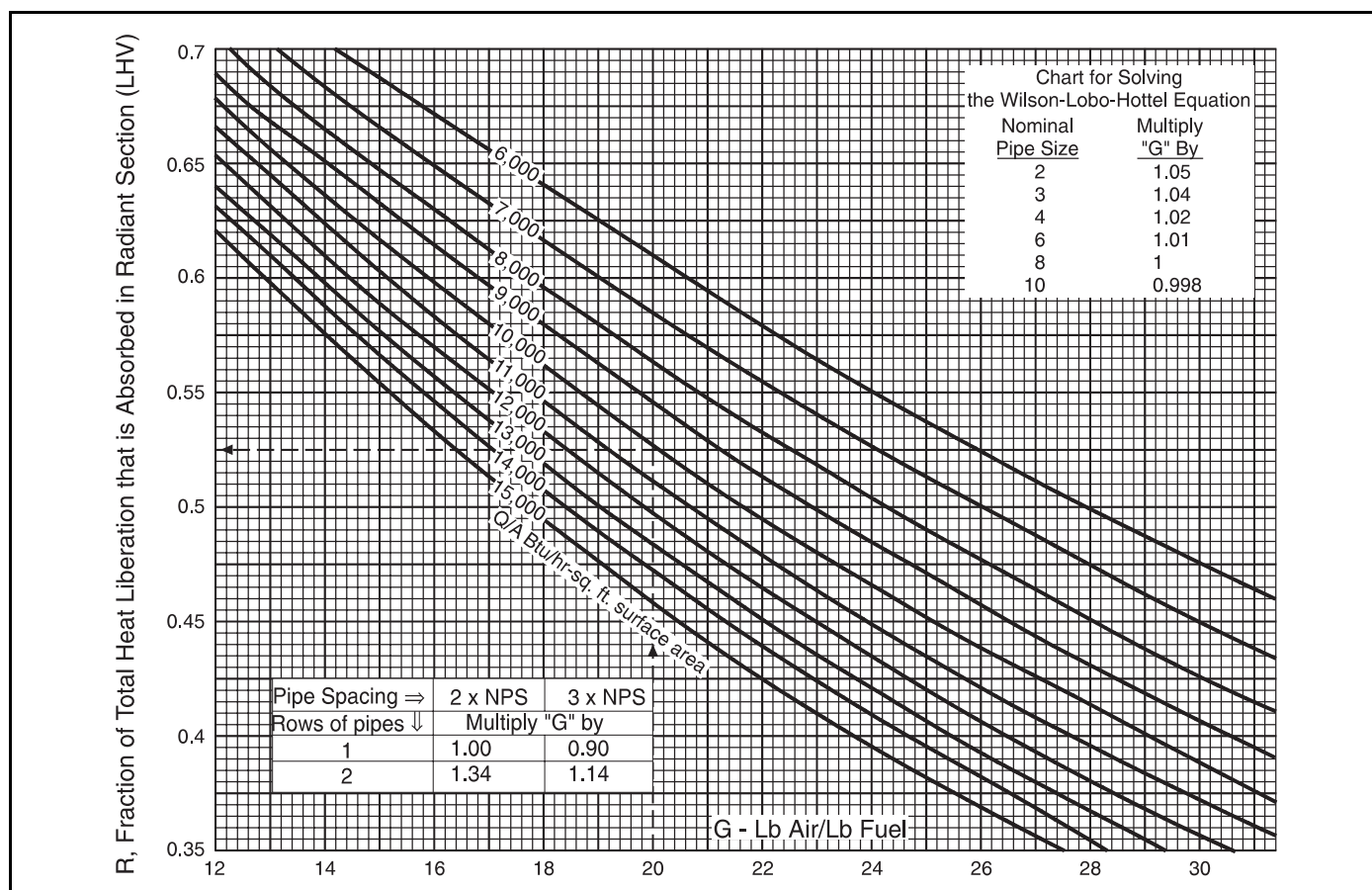
Equations 8-21 and 8-22, as well as Fig. 8-23, may be used to obtain an estimate of the absorbed heat in the radiant section of a fired heater, expressed as a fraction of the total net heat liberation, in terms of the average heat flux to the tubes (circumferential), the arrangement of the tubes, and the air to fuel weight ratio. These equations are solutions of the Wilson-Lobo and Hottel equation.

$$B = \frac{d_o \cdot n \cdot \pi \cdot I \cdot G^2}{S \cdot a} \quad \text{Eq 8-21}$$

$$R = 1 - \frac{\sqrt{(B^2 + 70.56 \cdot 10^6 \cdot B)} - B}{35.28 \cdot 10^6} \quad \text{Eq 8-22}$$

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



where “a” is a constant depending on the arrangement of the tubes. The “a” value is:

No. of rows	Tube spacing	
	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

$$r = 970 \text{ lb flue gas/MMBtu (Fig. 8-24).}$$

$$\begin{aligned} r \cdot \text{LHV} &= 970 \frac{\text{lb flue gas}}{\text{MMBtu}} \cdot 1000 \frac{\text{Btu}}{\text{scf fuel gas}} \\ &= \frac{970 \text{ lb flue gas}}{1000 \text{ scf fuel gas}} \end{aligned}$$

$$\text{Weight of 1000 scf fuel gas} = \frac{(1000)(0.61)(29)}{379.5} = 46.6 \text{ lb}$$

$$\text{Weight of combustion air} = 970 - 46.6 = 923.4 \text{ lb}$$

$$G = 923.4/46.6 = 19.8 \text{ lb air/lb fuel}$$

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

$$R = 1 - \frac{\sqrt{(7.87 \cdot 10^6)^2 + (70.56 \cdot 10^6)(7.87 \cdot 10^6)} - 7.87 \cdot 10^6}{35.28 \cdot 10^6}$$

$$R = 0.519 \text{ (Note: Fig. 8-23 yields } R = 0.525, \text{ so use the average of } R = 0.522.)}$$

$$Q_r = \frac{\text{UHT} \cdot R}{\text{NTE}} = \frac{(10)(0.522)}{0.8} = 6.525 \text{ MMBtu/hr}$$

$$\text{Heat transfer area} = \frac{Q_r}{I} = \frac{6.525 \cdot 10^6}{10,000} = 652.5 \text{ sq ft}$$

$$\text{The surface area of 4 in. pipe is 1.178 sq ft/linear ft.}$$

$$\text{Total tube length} = 652.5/1.178 = 553.9 \text{ ft}$$

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.

$$\text{Tube length} = 553.9/36 = 15.4 \text{ ft}$$

Convection Section

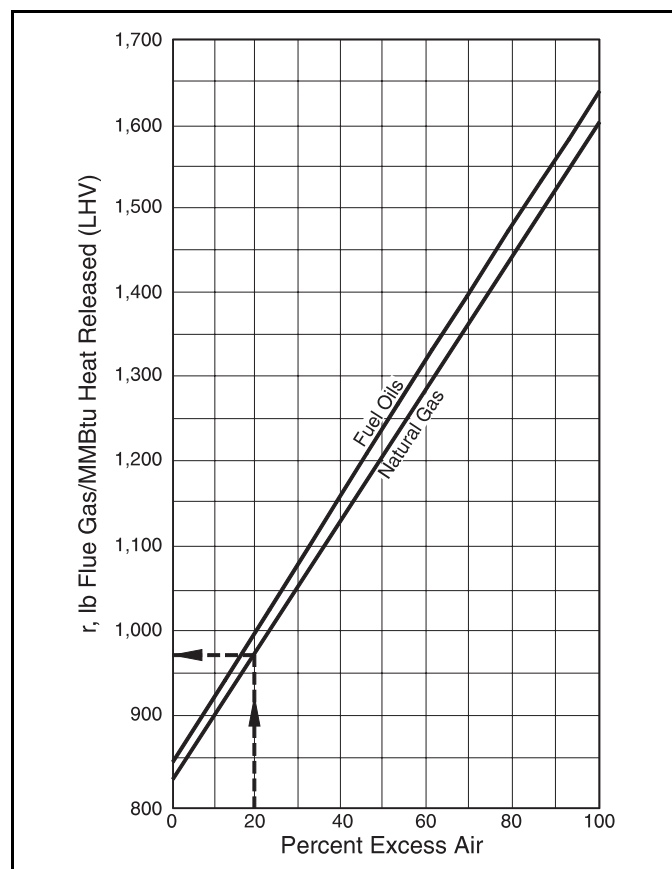
The purpose of the convection section is to transfer as much heat as possible from the combustion gases leaving the radiant section. As always there is the trade-off between capital cost, i.e., adding more tubes, and operating cost, i.e., improved thermal efficiency.

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 Rates⁹



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.

$$Q_{\text{total}} = \text{duty}/\text{NTE} = 10 (10^6)/0.80 = 12.5 \text{ MMBtu/hr}$$

$$r = 970 \text{ lb flue gas/MMBtu (Fig. 8-24)}$$

$$\text{Flue gases flow rate} = 12.5(970) = 12,125 \text{ lb/hr}$$

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:

$$Q_{\text{radiant exit}} = 12.5 - 6.525 - 0.25 = 5.725 \text{ MMBtu/hr}$$

The enthalpy (without latent heat) of the exit gas from radiant section:

$$H = 5.725 (10^6)/12,125 = 472.1 \text{ Btu/lb}$$

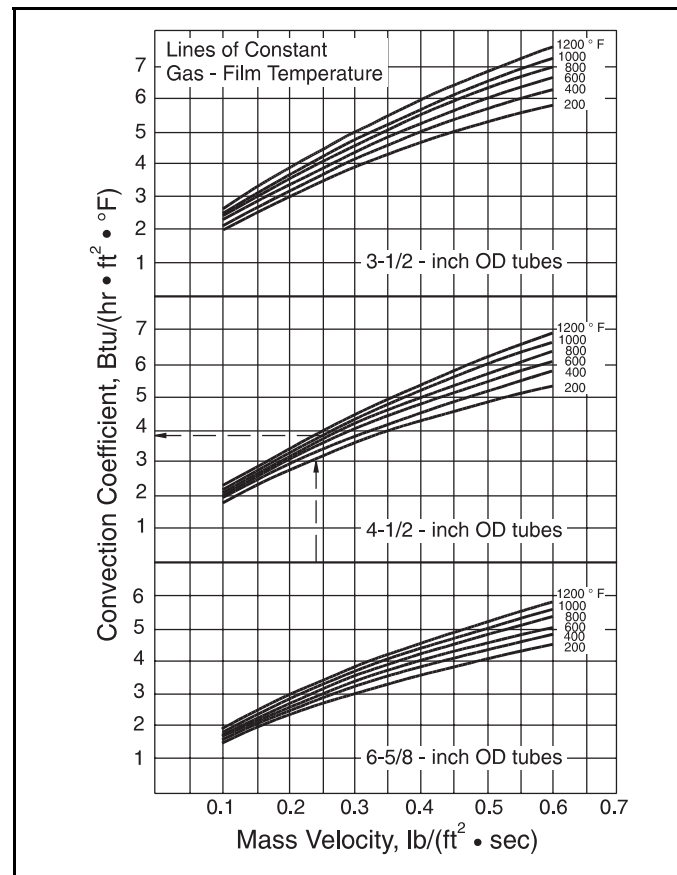
$$T_g = 1730^\circ\text{F (Fig. 8-20, Flue Gas – LHV)}$$

Convection Section:

$$\text{Area for gas flow} = (\text{no. of tubes}) (L) (\text{spacing} - D)$$

FIG. 8-25

Flue Gas Convection-Coefficients for Flow Across Staggered Banks of Bare Tubes⁹



$$G_g = 12,125/(14.0) (3600) = 0.241 \text{ lb/(sec • sqft)}$$

First shock row. Assume the average gas temperature is 1625°F and tube wall temperature is 500°F.

$$T_{g \text{ mean}} = \frac{500 + 1625}{2} = 1062^\circ\text{F}$$

$$h_o = 3.8 \text{ Btu/(hr • sqft • °F) (Fig. 8-25)}$$

$$A = 1.178 \text{ sq ft per linear ft (Example 8-8)}$$

$$A_{\text{tubes}} = 48 (1.178) = 56.54 \text{ sq ft}$$

$$Q_c = h_o A \Delta T = (3.8) (56.54) (1625 - 500)$$

$$= 0.242 \text{ MMBtu/hr}$$

$$I = Q/A = 10,000 \text{ Btu/(hr • sq ft) (Example 8-8)}$$

$$Q_r = (Q/A) (A) = 10,000 (56.54) = 0.565 \text{ MMBtu/hr}$$

$$Q_c + Q_r = (0.242 + 0.565) 10^6 = 0.807 \text{ MMBtu/hr}$$

$$Q_{\text{exit gases}} = (5.725 - 0.807) 10^6 = 4.918 \text{ MMBtu/hr}$$

$$H_{\text{exit gases}} = 4.918 (10^6)/12,125 = 405.6 \text{ Btu/lb}$$

$$T_{g(\text{exit})} = 1520^\circ\text{F (Fig. 8-20, Flue Gas – LHV)}$$

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)

$$Q_r = (3333)(56.54) = 0.188 \text{ MMBtu/hr}$$

$$\text{With } h_o = 3.7 \text{ Btu/(hr • sq ft • °F)}$$

$$Q_c = (3.7)(56.54)(1465 - 480) = 0.206 \text{ MMBtu/hr}$$

$$Q_c + Q_r = 0.206 + 0.188 = 0.394 \text{ MMBtu/hr}$$

$$Q_{\text{exit gases}} = (4.918 - 0.394) 10^6 = 4.524 \text{ MMBtu/hr}$$

$$H_{\text{exit gases}} = 4.524 (10^6)/12,125 = 373.1 \text{ Btu/lb}$$

$$T_g(\text{exit}) = 1410^\circ\text{F} \text{ (Fig. 8-20, Flue Gas - LHV)}$$

Finned rows. The combustion gas mass velocity increases because of the decreased cross sectional area of finned pipe. From Eq 8-13.

$$A_{cs} = \frac{4.5}{12} + \frac{36(0.105)(7.0 - 4.5)}{144}$$

$$= 0.441 \text{ sq ft/linear ft}$$

$$G_g = \frac{12,125}{[48(0.667 - 0.441)(3600)]}$$

$$= 0.310 \text{ lb/(sec • sq ft)}$$

$$Q_f = 10.0 - 6.525 - 0.807 - 0.394 = 2.274 \text{ MMBtu/hr}$$

$$Q_{\text{exit gases}} = 12.50 - 10.00 - 0.25 = 2.25 \text{ MMBtu/hr}$$

$$H_g(\text{exit}) = 2.25 (10^6)/12,125 = 185.6 \text{ Btu/lb}$$

$$T_g(\text{exit}) = 730^\circ\text{F} \text{ (Fig. 8-20, Flue Gas - LHV)}$$

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:

$$T_{p1} = 200^\circ\text{F}; \quad T_{p2} = 470^\circ\text{F}; \quad T_{p \text{ av}} = 335^\circ\text{F}$$

$$T_{g1} = 1410^\circ\text{F}; \quad T_{g2} = 730^\circ\text{F}; \quad T_{g \text{ av}} = 1070^\circ\text{F}$$

$$\Delta T_{LM} = [(1410 - 470) - (730 - 200)] / \ln(940/530) = 715^\circ\text{F}$$

$$T_{\text{film}} = (335 + 1070)/2 = 703^\circ\text{F}$$

$$h_o = 4.0 \text{ Btu/(hr • sq ft • °F)} \text{ (Fig. 8-25)}$$

$$Q_f = h_o A_o \Delta T_{LM} \text{ (Eq 8-10)}$$

$$A_o = 2.274 (10^6)/(4.0)(715) = 795 \text{ sq ft}$$

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.

$$L_{\text{pipe}} = 795/[(0.87)(12.10)] = 75.5 \text{ ft}$$

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

$$\text{Assume } \mu_g = \mu_a \text{ from Fig. 23-26.}$$

$$\text{Re} = \frac{0.5263 \cdot M}{D \cdot \mu} = \frac{(0.5263)(12,125)}{(2)(0.024)} = 132,946$$

For an absolute roughness of 0.002 feet, the relative roughness is 0.001 then,

$$f = 0.005 \text{ (Fig. 17-2)}$$

From Eq 8-18

$$H_{av} = 0.192 \cdot H_s (\rho_g - \rho_a) + 0.0029 \cdot V^2 \cdot \rho_g \left(4 \cdot f_f \cdot \frac{H_s}{D_s} + 1 \right)$$

Calculate ρ_g and ρ_a from the ideal gas law.

$$V = \frac{M}{3600 \cdot A \cdot \rho} = \frac{12,125}{3600 \cdot \pi \cdot (0.0335)} = 32.0 \text{ ft/sec}$$

$$H_{av} = 0.192 (20) (0.0335 - 0.0766) + 0.0029 (32)^2 0.0335$$

$$[4(0.005) (20/2) + 1]$$

$$= -0.046 \text{ in. } \text{H}_2\text{O}$$

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 $\text{RD}=0.6$ 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 lum-nite (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 lum-nite, 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

FIG. 8-26
10 MMBtu/hr Regeneration 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

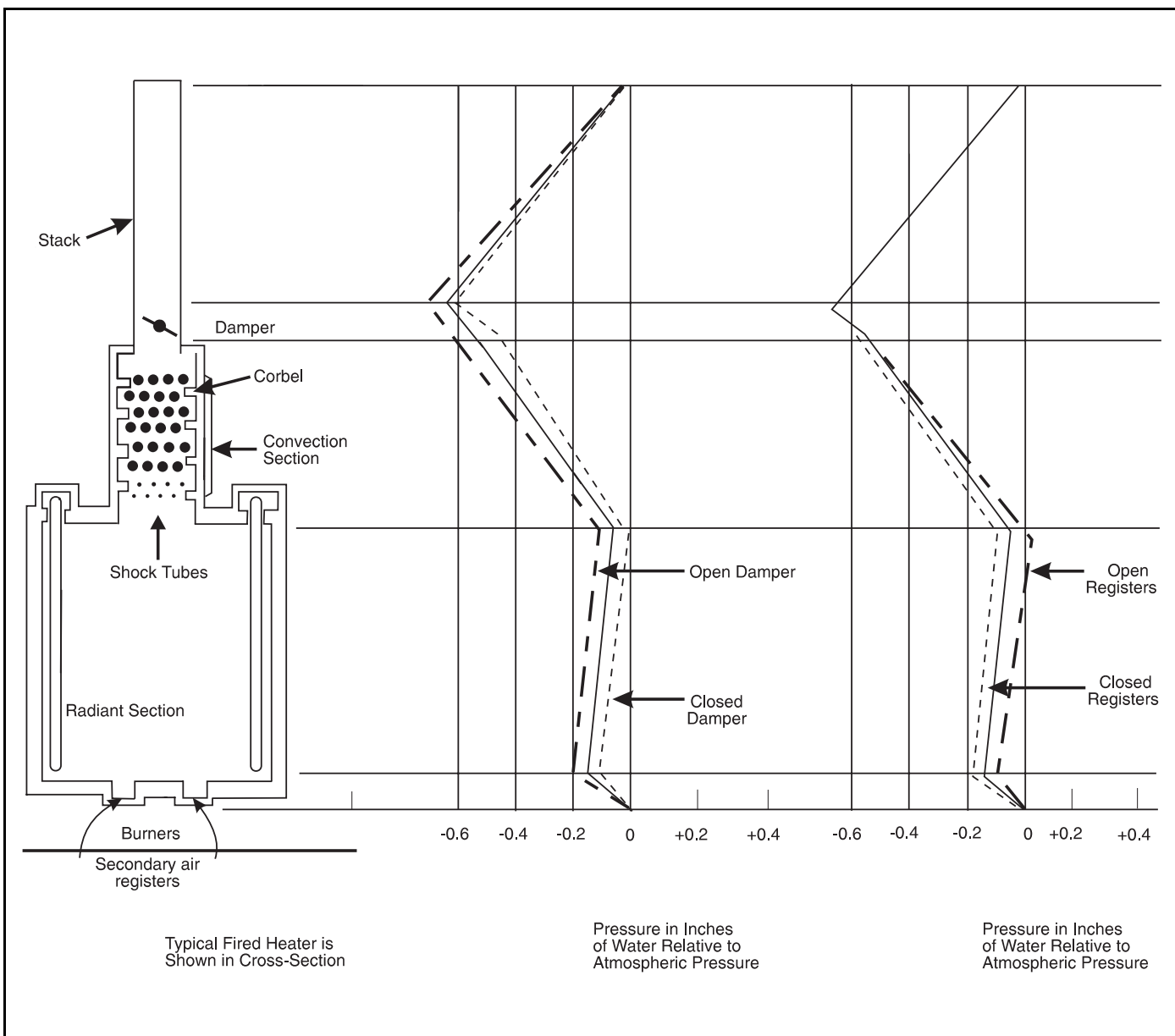
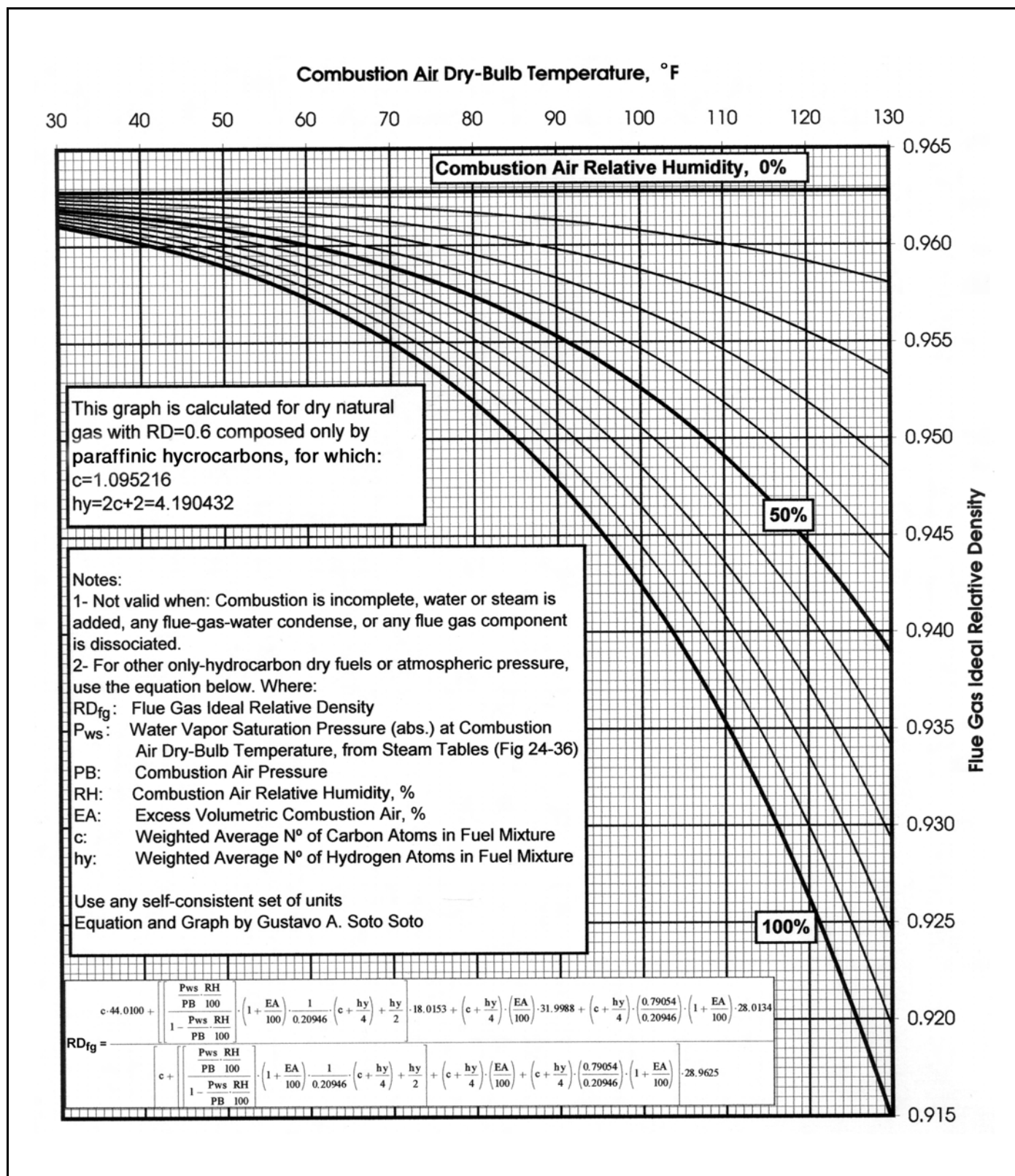


FIG. 8-28

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



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 CO₂ 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.

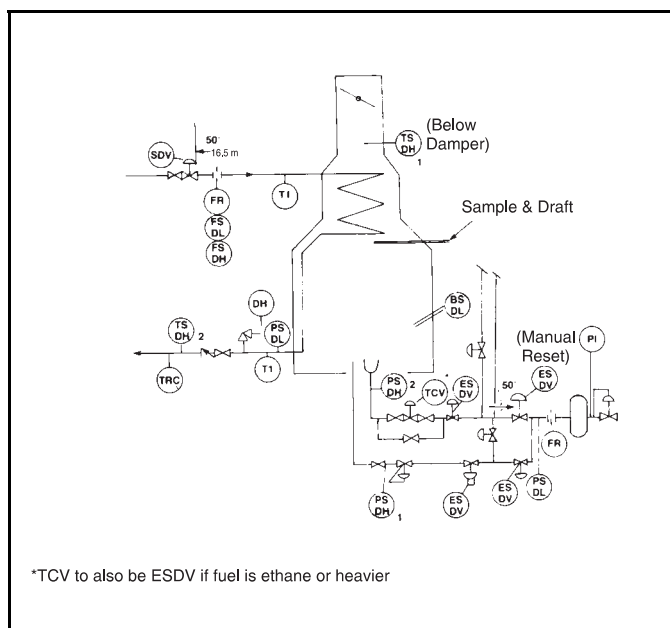
Draft gauges — These are needed to set the stack damper or burner air registers.

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.

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

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

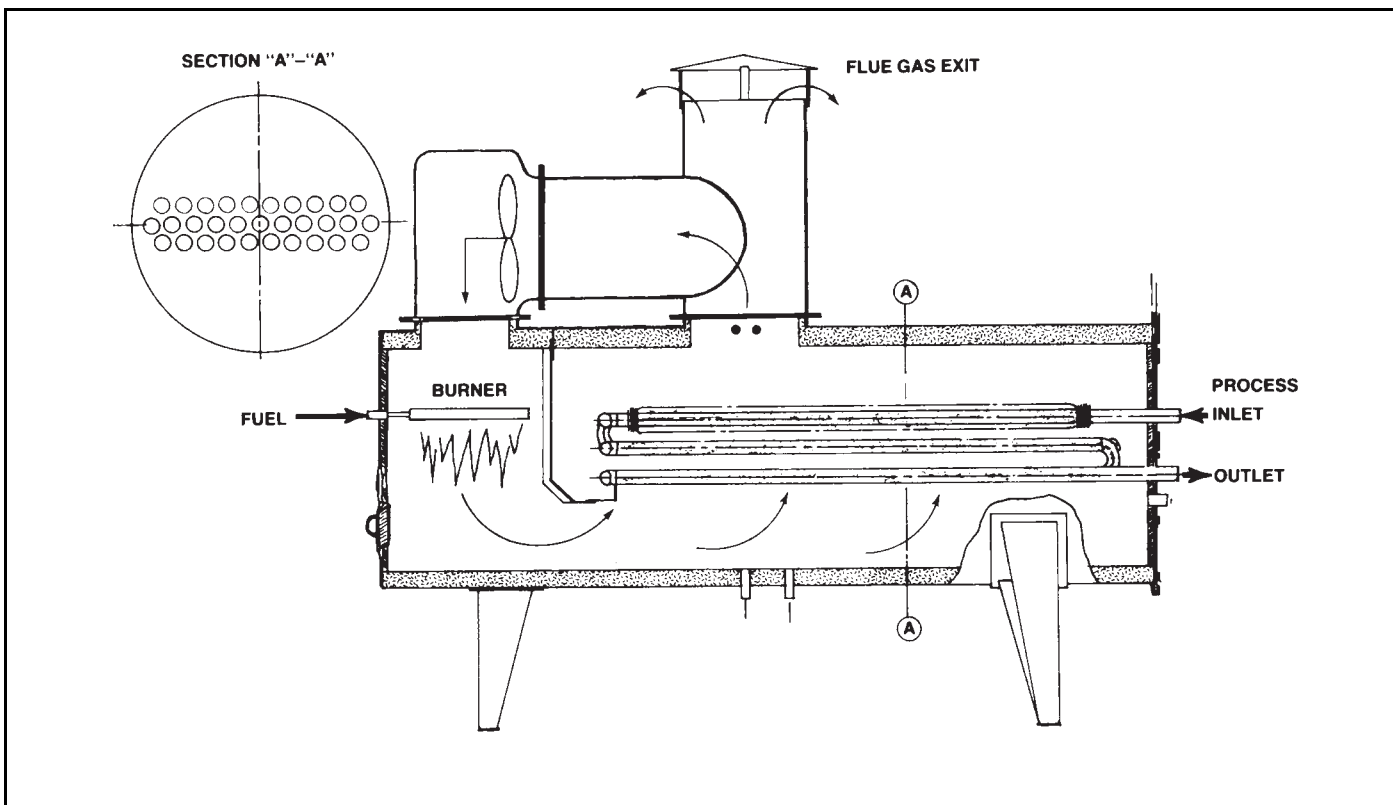
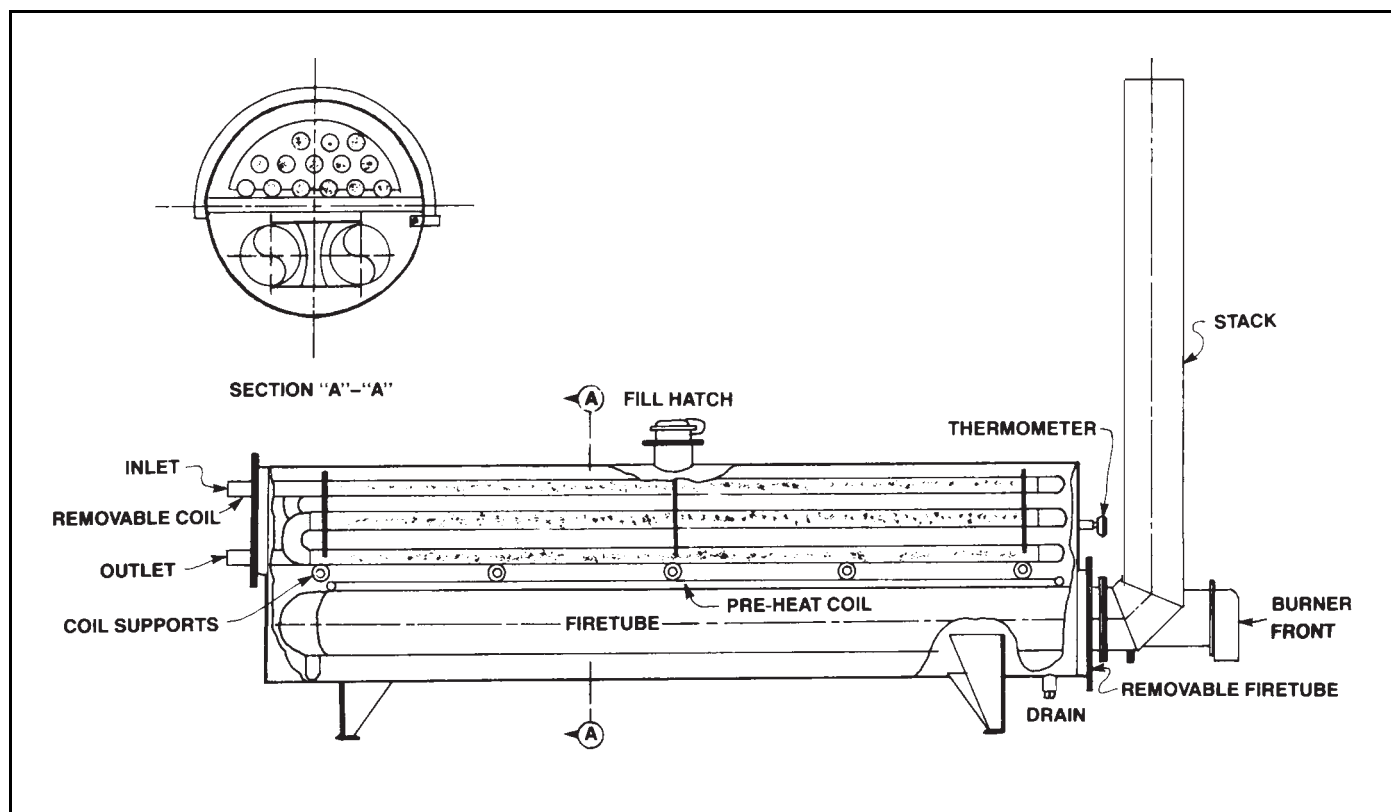


FIG. 8-32
Indirect Fired Water Bath Heater



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.

$$q_l = 50 \text{ gpm}$$

$$M = (50) (4.24) (60) = 12720 \text{ lb/hr}$$

$$\text{UHT} = M (H_2 - H_1)$$

$$= (12720) (462 - 287) = 2.23 \text{ MMBtu/hr}$$

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 pre-heat 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 CH₄ 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

temperature. This is the minimum coil area that corresponds to an operable path. So:

$$\text{Preheat duty: } (476 - 448) = 28 \text{ Btu/lb CH}_4$$

$$\text{Postheat duty: } (510 - 476) = 34 \text{ Btu/lb}$$

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

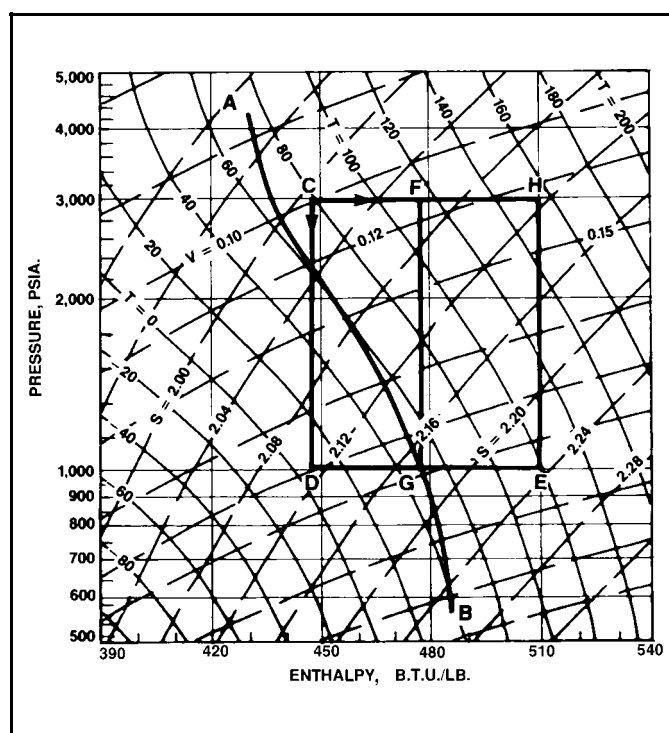
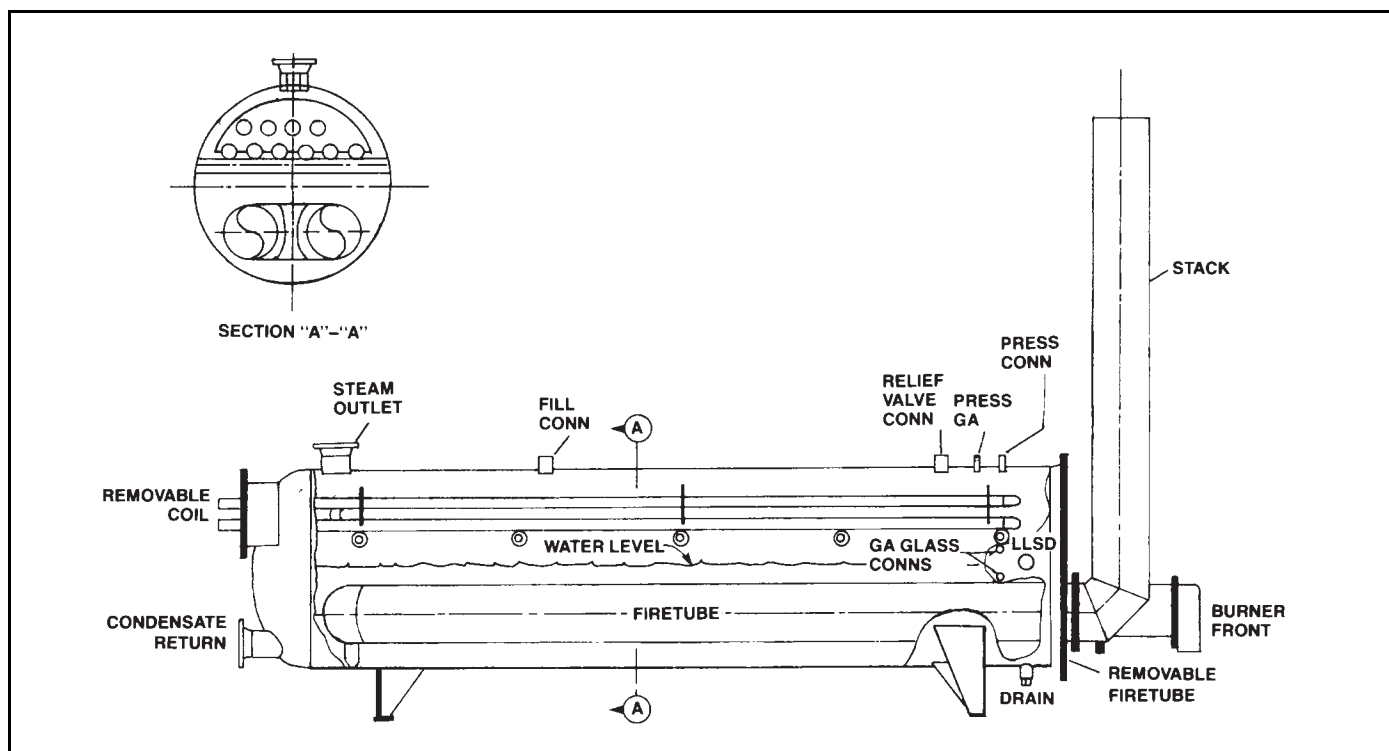


FIG. 8-34

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

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

- 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.

$$r = 970 \text{ lb flue gas/MMBtu heat release (Fig. 8-24)}$$

$$\text{Combustion gas flow is } 4(970) = 3880 \text{ lb/hr}$$

From Eq 8-7, 8-8b and 8-9a and Fig. 8-5

$$Nu = 0.023 \cdot Re^{0.8} \cdot Pr^{0.33} \cdot (\mu_b/\mu_w)^{0.14}$$

$$k = 0.038 \text{ Btu/(hr • sq ft • °F/ft)} \text{ at average } T = 1550^\circ\text{F}$$

$$C_p = 0.248 \text{ Btu/(lb • °F)}$$

$$\mu_b = 0.045 \text{ cp}$$

$$\mu_w = 0.023 \text{ cp}$$

$$\frac{h \cdot 2.0}{0.038} = 0.023 \left[\frac{0.5263(3880)}{2(0.045)} \right]^{0.8} \left[\frac{2.419 \cdot 0.248 \cdot 0.045}{0.038} \right]^{0.33} \left[\frac{0.045}{0.023} \right]^{0.14}$$

$$h = \frac{0.038(0.023)}{2.0} \cdot (3052.5)(0.8933)(1.0985)$$

$$h = 1.31 \text{ Btu/(hr • sq ft • °F)}$$

$$Q/A_c = 1.31(2800 - 300) = 3,275 \text{ Btu/(hr • sq ft)}$$

$$\text{Total heat flux} = 16,990 + 3,275 = 20,265 \text{ Btu/(hr • sq ft)}$$

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. H₂O, 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

FIG. 8-36
Typical Physical Properties of Hot Oil

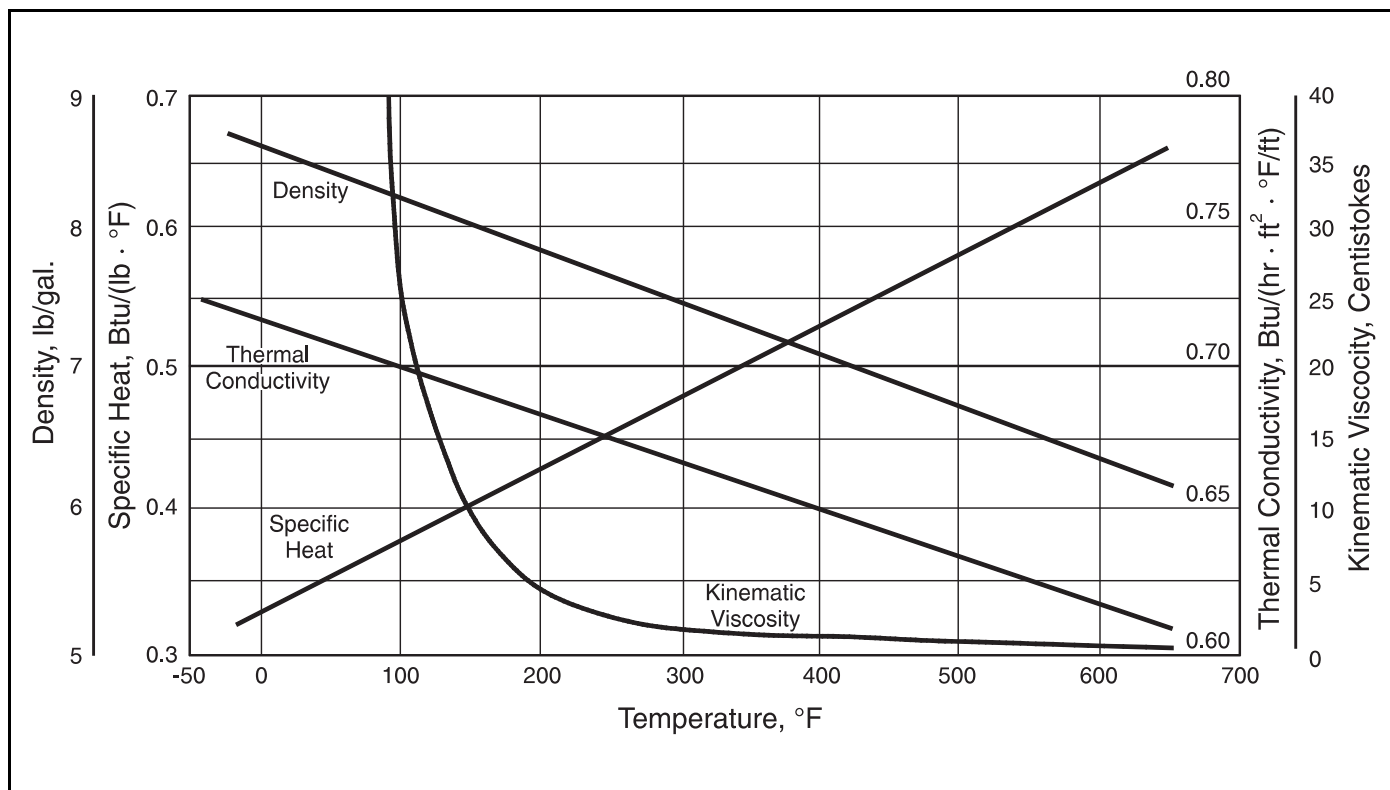


FIG. 8-37
Salt Bath Heater

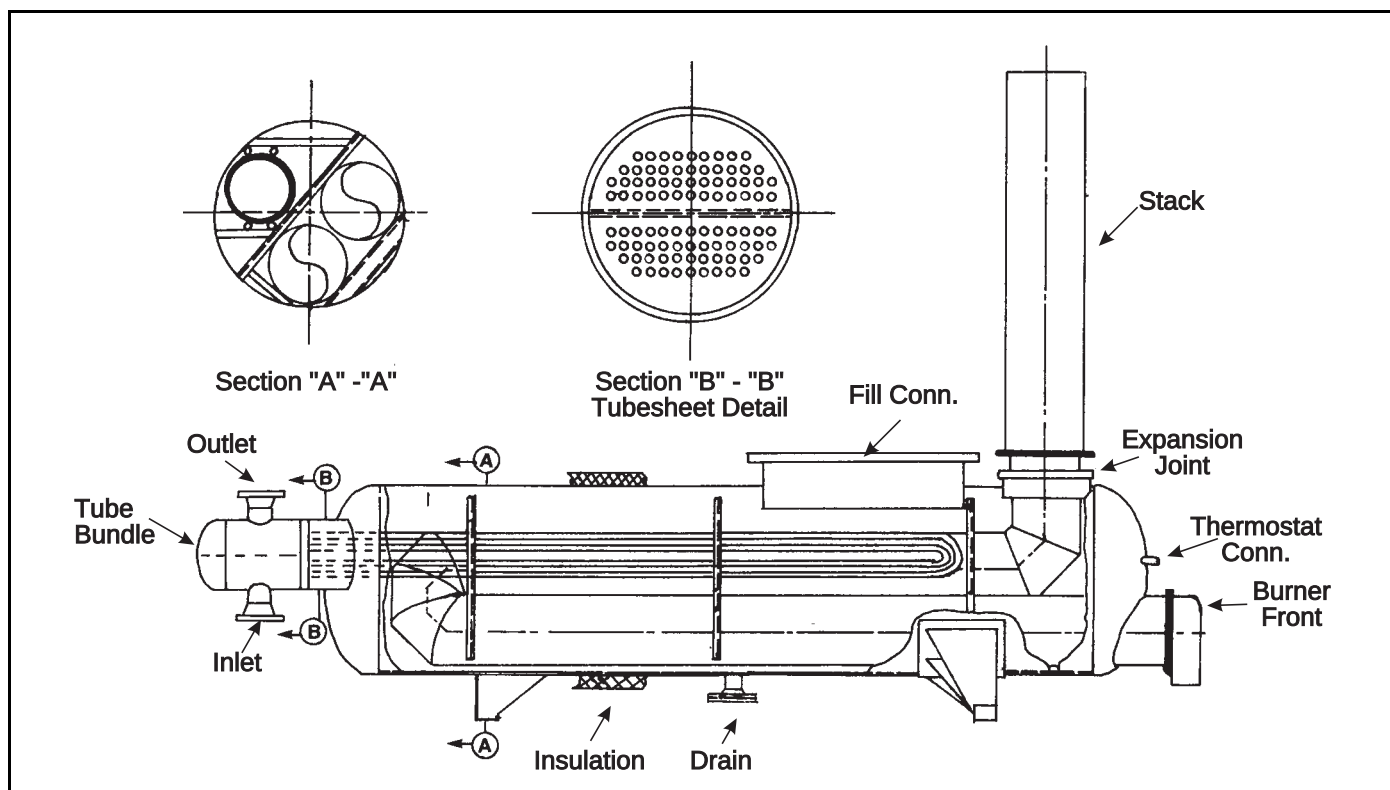
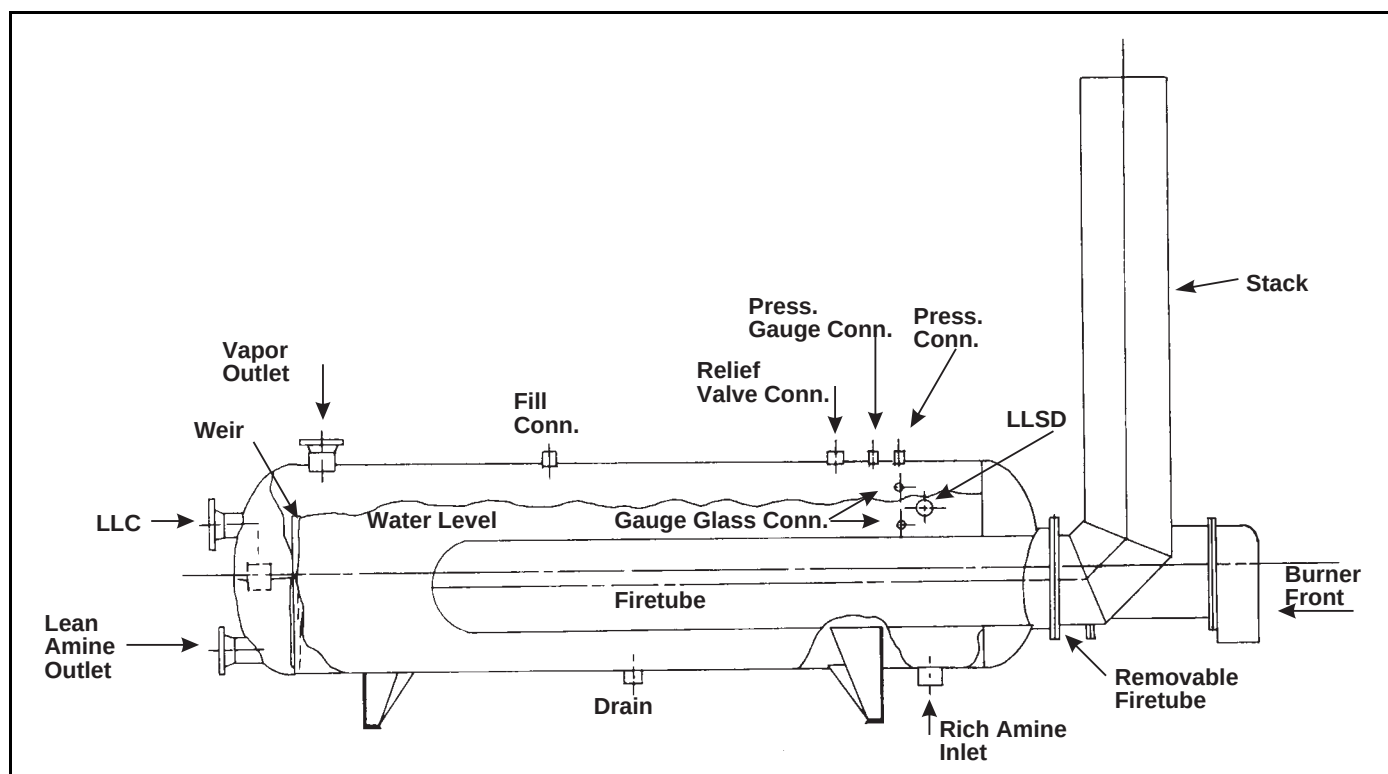


FIG. 8-38
Amine Reboiler



“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 in-

crease 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

FIG. 8-39
Indirect Fired Heater

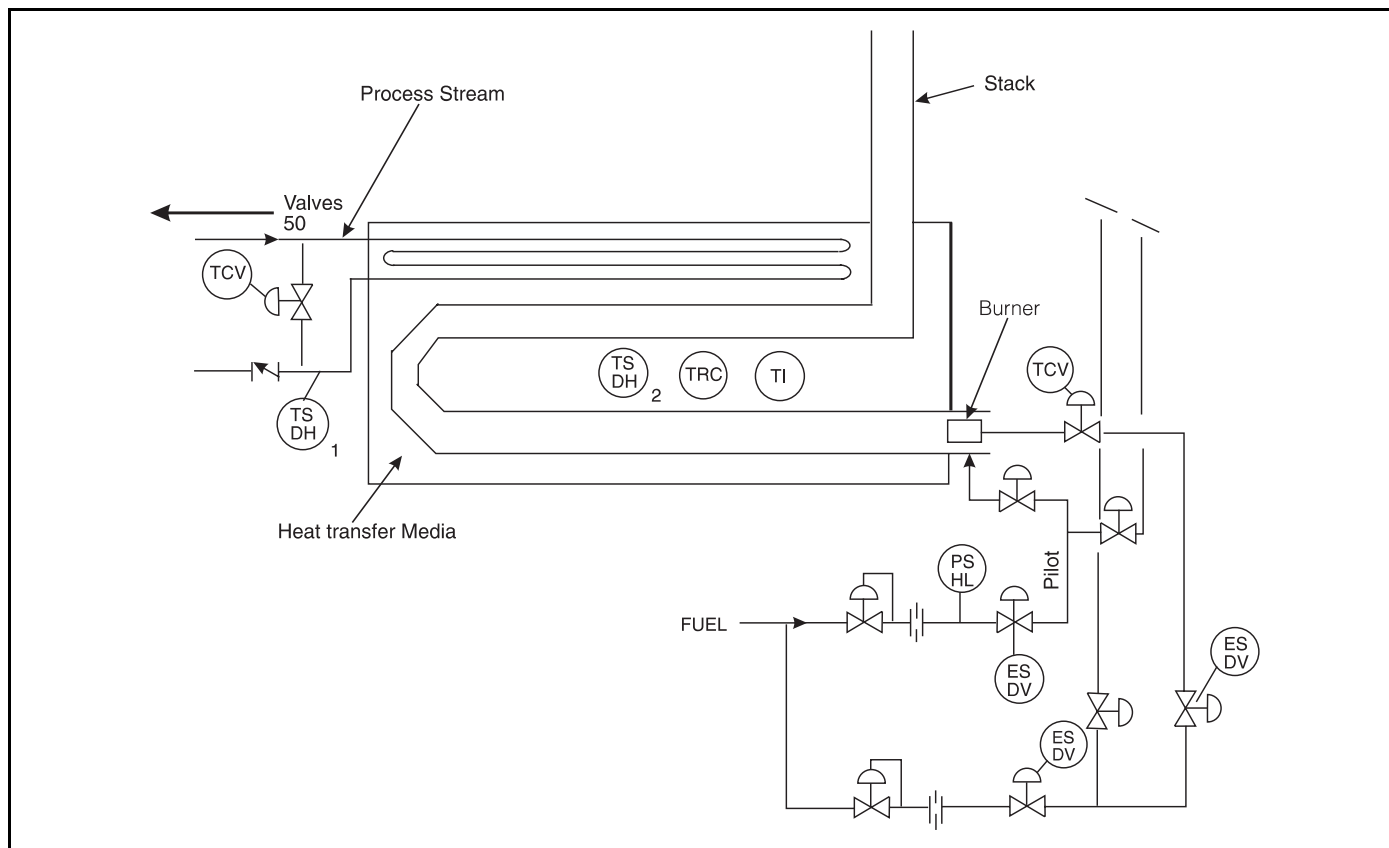


FIG. 8-40
Bath Heater Alarm/Shutdown Description

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.

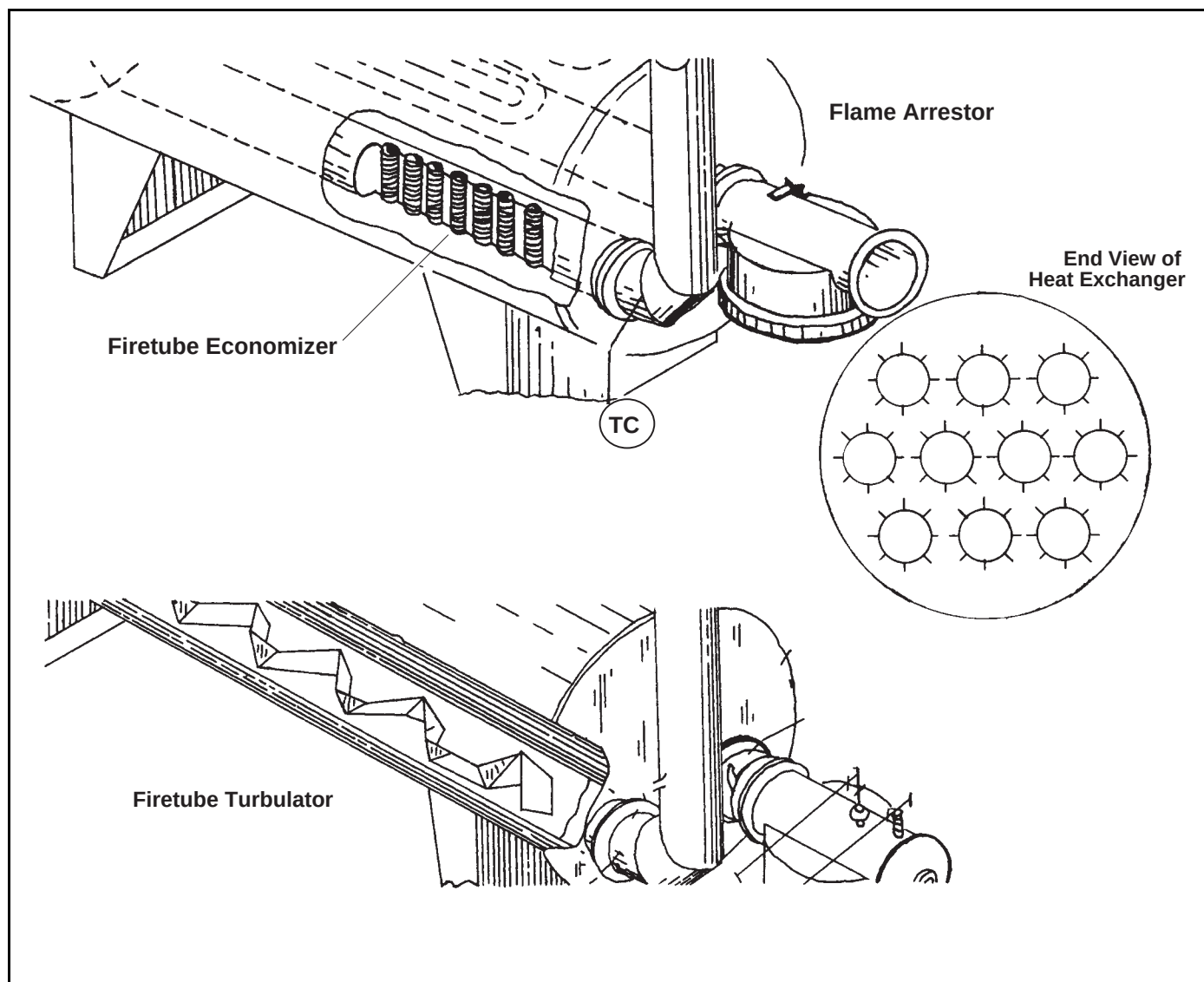
Schematic Label	Alarm/Shutdown Description	Line Heater	Hydrocarbon Reboiler	Low Pressure Steam Heater	Hot Oil or Salt Heater	Glycol Reboiler	Amine Reboiler
TSDH-2	High Bath Temperature	No Note 1, 2	Yes	No	Yes	Yes	Yes
LSL	Low Bath Level	No Note 2	No	Yes	Yes Note 3	No Note 2, 3	Yes Note 2, 3
PSL	Low Fuel 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.

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,

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 CO₂ 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 re-firing.

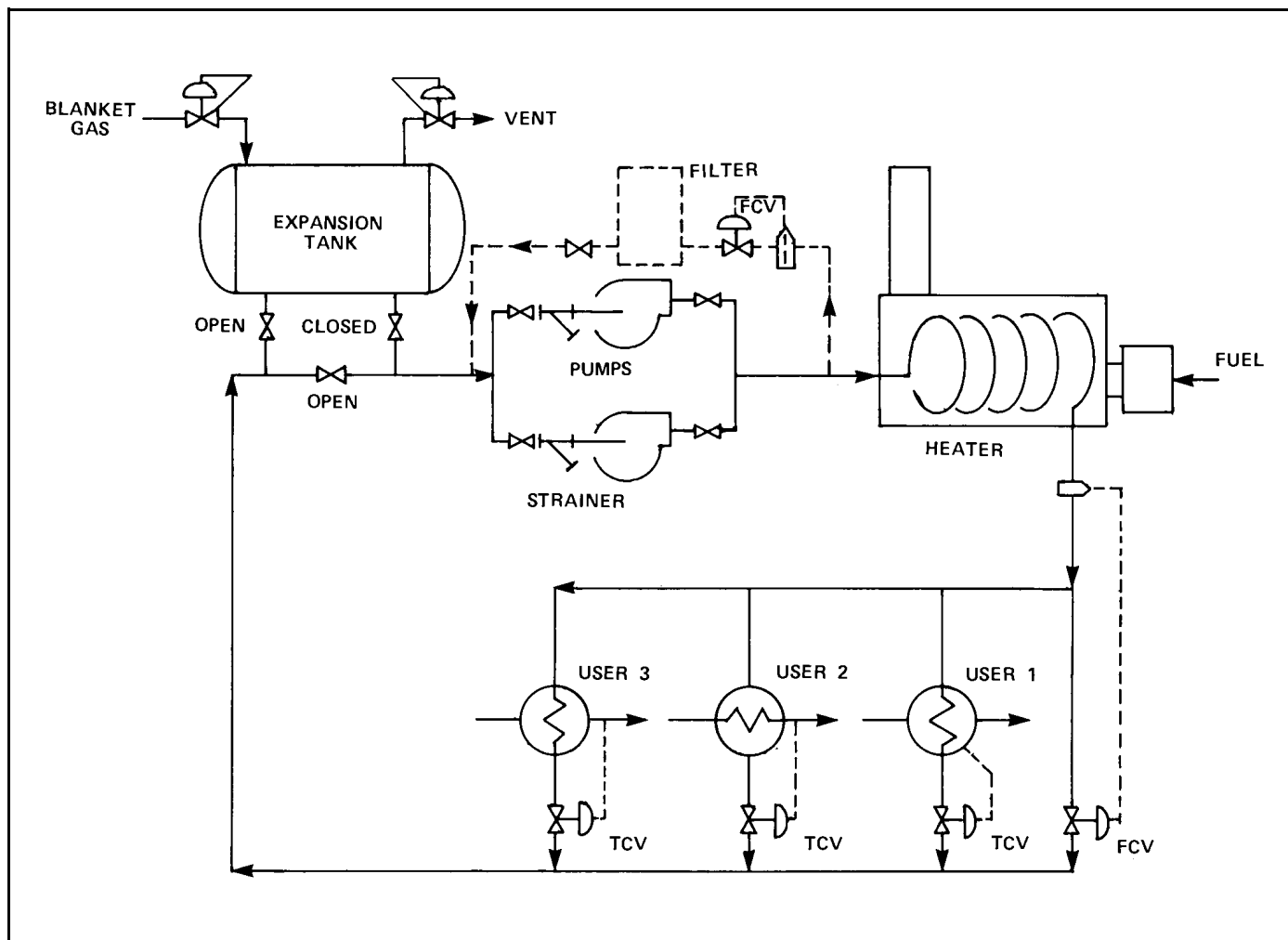
Heat transfer systems may have:

- 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

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 oxidation, 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

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 refring 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 considera-

tion 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:

$$Q = \dot{m}_H C_{pH} (T_{H1} - T_{H2}) \quad \text{Eq 9-1}$$

For isothermal condensing of the hot fluid:

$$Q = \dot{m}_H \lambda_H \quad \text{Eq 9-2}$$

For no phase change of the cold fluid:

$$Q = \dot{m}_C C_{pC} (T_{C2} - T_{C1}) \quad \text{Eq 9-3}$$

For isothermal boiling of the cold fluid:

$$Q = \dot{m}_C \lambda_C \quad \text{Eq 9-4}$$

Basic Heat Transfer Relations

$$Q = UA \text{ (LMTD) for single-pass designs} \quad \text{Eq 9-5a}$$

$$Q = UA \text{ (CMTD) for multi-pass designs} \quad \text{Eq 9-5b}$$

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 = nth 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

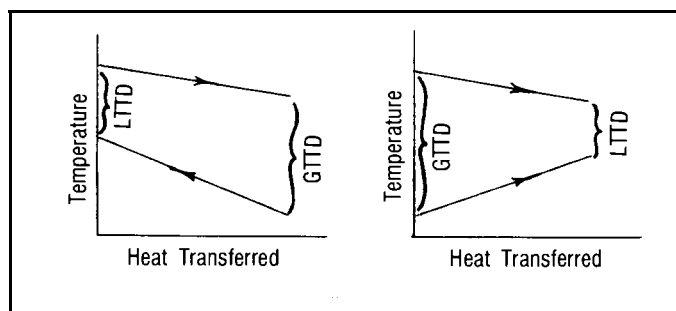


Fig. 9-3 defines LMTD in terms of Greatest Terminal Temperature Difference (GTTD) and Least Terminal Temperature Difference (LTTD), where “terminal” refers to the first or last point of heat exchange in the heat exchanger.

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.

$$\text{CMTD} = (\text{LMTD}) (F) = \text{Corrected Mean Temperature Difference} \quad \text{Eq 9-6}$$

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 n^{th} zone of the exchanger.

$$Q_{\text{Total}} = U_{\text{wtd}} A_{\text{Total}} (\text{WTD}) \quad \text{Eq 9-7}$$

$$Q_n = U_n A_n (\text{LMTD})_n \quad \text{Eq 9-8}$$

Then the weighted temperature difference may be defined as:

$$(\text{WTD}) = \frac{\sum [U_n A_n (\text{LMTD})_n]}{\sum [U_n A_n]} = \frac{Q_{\text{Total}}}{\sum [Q_n / (\text{LMTD})_n]} \quad \text{Eq 9-9}$$

And the weighted overall heat transfer coefficient becomes:

$$U_{\text{wtd}} = \frac{Q_{\text{Total}}}{A_{\text{Total}} (\text{WTD})} = \frac{\sum [Q_n / (\text{LMTD})_n]}{A_{\text{Total}}} \quad \text{Eq 9-10}$$

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

$$U_o = \frac{1}{\left[\frac{1}{h_o} + \left(\frac{A_o}{A_i} \right) \left(\frac{1}{h_i} \right) + r_w + r_{fo} + \left(\frac{A_o}{A_i} \right) r_{fi} \right]} \quad \text{Eq 9-11}$$

Metal Resistance for Plain Tubes

The metal resistance is calculated by the following equation:

$$r_w = \left(\frac{D_o}{24k_w} \right) \ln \left(\frac{D_o}{D_i} \right) \quad \text{Eq 9-12}$$

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).

FIG. 9-3
LMTD Chart

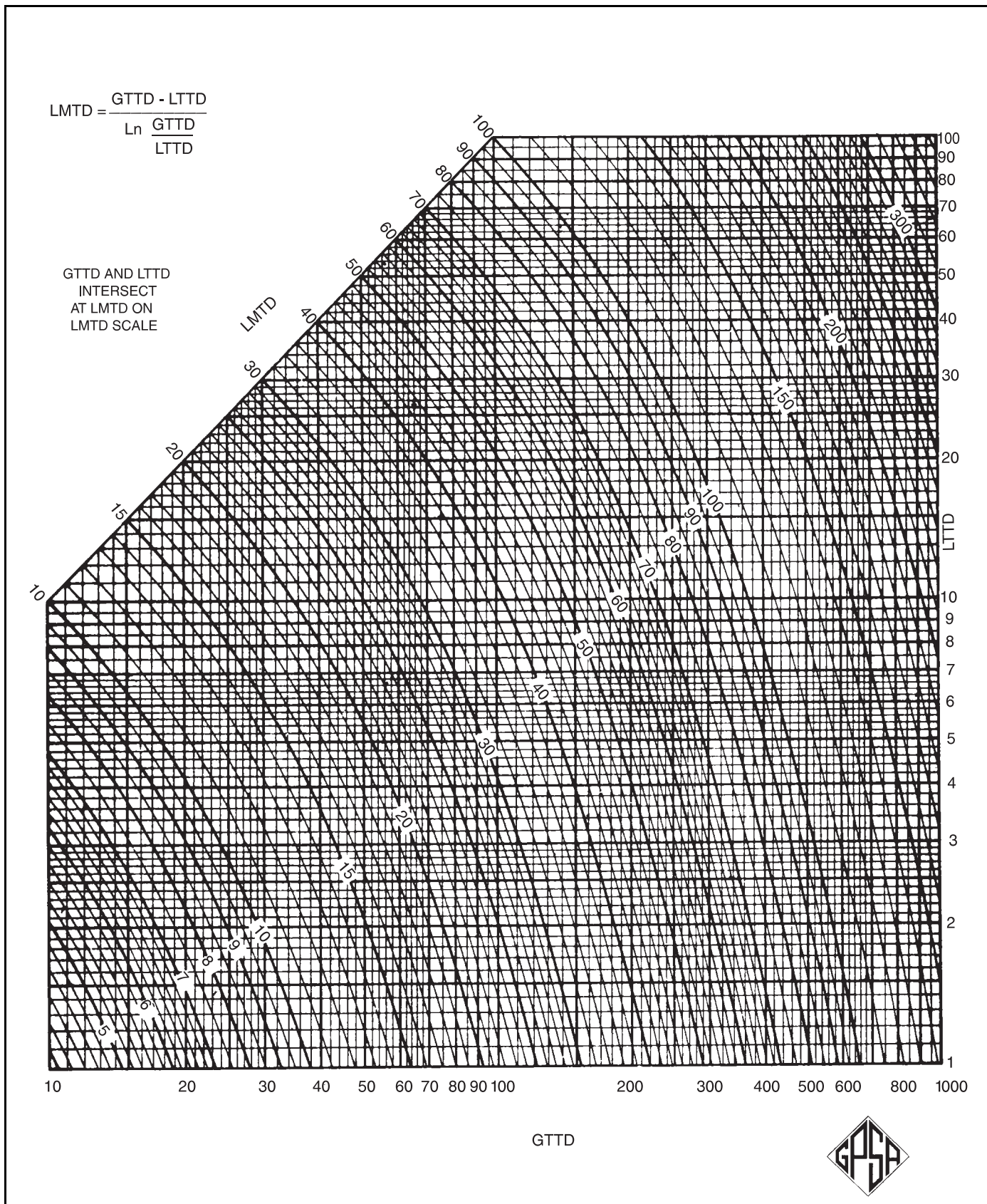


FIG. 9-4

LMTD Correction Factor (1 shell pass; 2 or more tube passes)

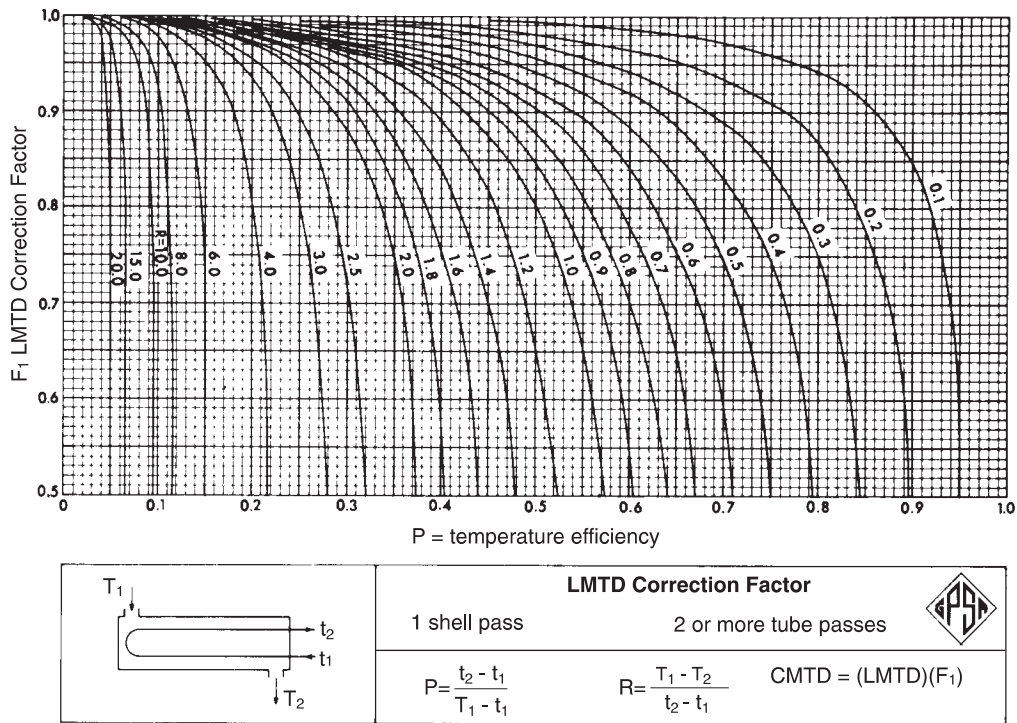


FIG. 9-5

LMTD Correction Factor (2 shell passes; 4 or more tube passes)

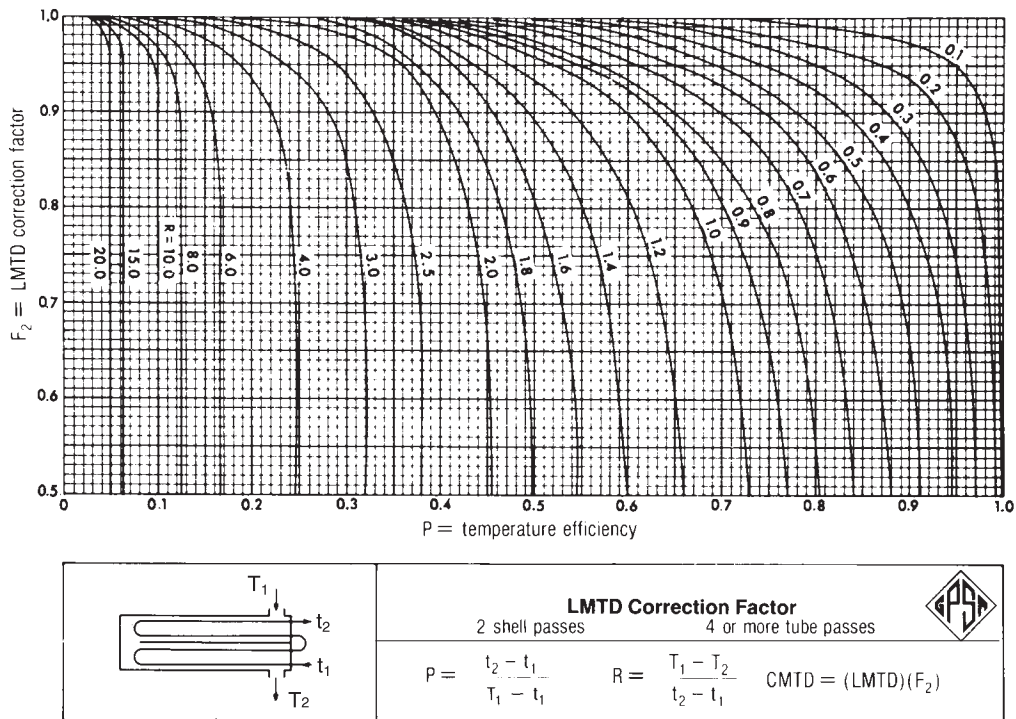


FIG. 9-6

LMTD Correction Factor (3 shell passes; 6 or more tube passes)

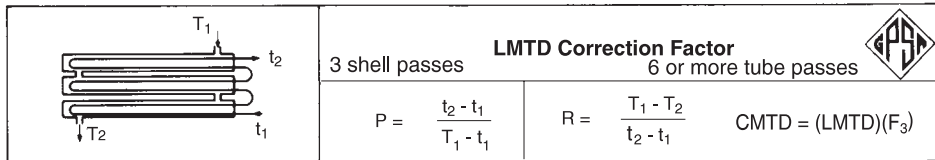
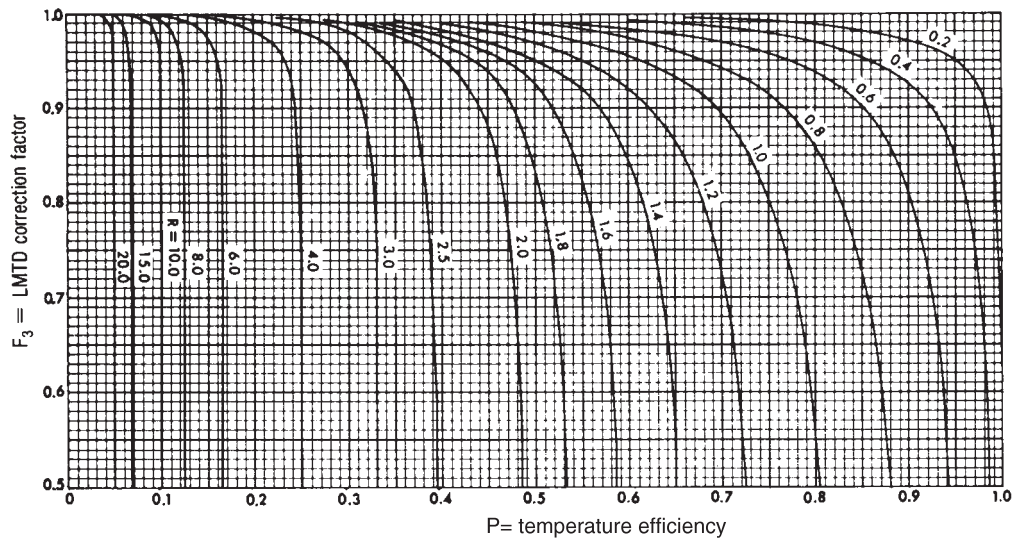


FIG. 9-7

LMTD Correction Factor (4 shell passes; 8 or more tube passes)

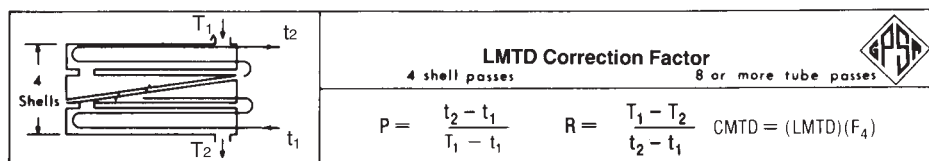
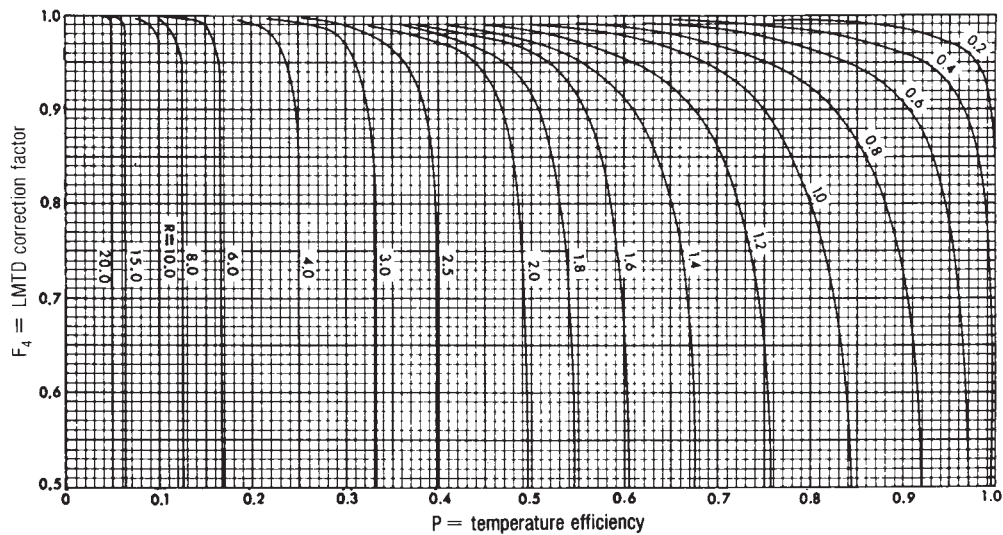


FIG. 9-8

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:

$$\frac{h_2}{h_1} = \frac{r_2}{r_1} = \left(\frac{\mu_2}{\mu_1} \right)^{0.47} \left(\frac{k_1}{k_2} \right)^{0.67} \left(\frac{C_{p1}}{C_{p2}} \right)^{0.33} \left(\frac{G_1}{G_2} \right)^{0.8} \left(\frac{D_{i2}}{D_{i1}} \right)^{0.2} \quad \text{Eq 9-13}$$

and,

$$\frac{\Delta P_2}{\Delta P_1} = \left(\frac{\mu_2}{\mu_1} \right)^{0.2} \left(\frac{G_2}{G_1} \right)^{1.8} \left(\frac{\rho_1}{\rho_2} \right) \left(\frac{D_{i1}}{D_{i2}} \right)^{1.2} \left(\frac{N_{p2}}{N_{p1}} \right) \quad \text{Eq 9-14}$$

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)

$$\begin{aligned} Q_H/Q_C &= \frac{[\dot{m}_H C_{pH} (T_{H1} - T_{H2})]}{[\dot{m}_C C_{pC} (T_{C2} - T_{C1})]} \\ &= \frac{[475,723(0.542) (197 - 100)]}{[650,860(0.60) (124 - 60)]} = 1.0 \end{aligned}$$

2. Calculate the LMTD.

$$\begin{aligned} 197 &\rightarrow 100 \\ \frac{124}{73} &\leftarrow \frac{60}{40} \\ \text{LMTD} &= \frac{(73 - 40)}{\ln(73/40)} = 54.9^\circ\text{F} \end{aligned}$$

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

$$= \frac{[784 \pi (D_i)^2]}{4} = (784) (0.3019) = 237 \text{ sq in.}$$

$$G = \frac{(650,860)}{[(3600)(237/144)]} = 109.8 \text{ lb/(sq ft • sec)}$$

$$\text{Re} = \frac{(123.9)(0.62)(109.8)}{0.21} = 40,200$$

Therefore, it is turbulent flow since $\text{Re} > 2000$.

From Fig. 9-10 (see Note †), the ratio of the second to the first resistance is:

$$(r_i)_2 = (f) (r_i)_1$$

FIG. 9-10
Variables in Exchanger Performance

Variable*	Flow Regime	$r_2 = (f) \cdot (r_1)^\dagger$		$\Delta P_2 = (f) \cdot (\Delta P_1)^\dagger$	
		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 (or mass flowrate)	Turbulent	$(G_1/G_2)^{0.6}$	$(G_1/G_2)^{0.8}$	$(G_2/G_1)^{1.85}$	$(G_2/G_1)^{1.8}$
Density	Streamline		$(G_1/G_2)^{0.33}$		(G_2/G_1)
Tube diameter	Turbulent	$(D_{o2}/D_{o1})^{0.4}$	$(D_{i2}/D_{i1})^{0.2}$	(ρ_1/ρ_2) $(D_{o1}/D_{o2})^{0.15}$	ρ_1/ρ_2 $(D_{i1}/D_{i2})^{1.2}$
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)

Use base values from Fig. 9-11 for $(r_i)_1$ conditions.

$$f = \left(\frac{\mu_2}{\mu_1}\right)^{0.47} \left(\frac{k_1}{k_2}\right)^{0.67} \left(\frac{C_{p1}}{C_{p2}}\right)^{0.33} \left(\frac{G_1}{G_2}\right)^{0.8} \left(\frac{D_{i2}}{D_{i1}}\right)^{0.2}$$

$$= \left(\frac{0.21}{0.726}\right)^{0.47} \left(\frac{0.0788}{0.078}\right)^{0.67} \left(\frac{0.50}{0.600}\right)^{0.33} \left(\frac{185}{109.8}\right)^{0.8} \left(\frac{0.62}{0.495}\right)^{0.2}$$

$$= 0.840$$

Basis: (Inside Area)

$$(r_i)_2 = f(r_i)_1 \text{ and } (r_i)_1 = 0.0038 \text{ from Fig. 9-11}$$

$$= (0.840)(0.0038)$$

$$= 0.0032 \text{ (hr} \cdot \text{sq ft} \cdot ^\circ\text{F)/Btu}$$

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.

$$f = \left(\frac{\mu_2}{\mu_1}\right)^{0.2} \left(\frac{G_2}{G_1}\right)^{1.8} \left(\frac{\rho_1}{\rho_2}\right) \left(\frac{D_{i1}}{D_{i2}}\right)^{1.2} \left(\frac{N_{p2}}{N_{p1}}\right)$$

$$= \left(\frac{0.21}{0.726}\right)^{0.2} \left(\frac{109.8}{185}\right)^{1.8} \left(\frac{46.9}{38.3}\right) \left(\frac{0.495}{0.62}\right)^{1.2} \left(\frac{1}{1}\right)$$

$$= 0.285$$

$$(\Delta P_i)_2 = (f)(\Delta P_i)_1 = (0.285)(0.061) = 0.017 \text{ psi/ft}$$

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.

$$G = \frac{(475,723)}{(3600)(156.8/144)} = 121.4 \text{ lb/(sq ft} \cdot \text{sec)}$$

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.

$$f = \left(\frac{\mu_2}{\mu_1}\right)^{0.27} \left(\frac{k_1}{k_2}\right)^{0.67} \left(\frac{C_{p1}}{C_{p2}}\right)^{0.33} \left(\frac{G_1}{G_2}\right)^{0.6} \left(\frac{D_{o2}}{D_{o1}}\right)^{0.4}$$

$$= \left(\frac{0.34}{0.549}\right)^{0.27} \left(\frac{0.0762}{0.077}\right)^{0.67} \left(\frac{0.577}{0.542}\right)^{0.33} \left(\frac{132.4}{121.4}\right)^{0.6} \left(\frac{0.75}{0.625}\right)^{0.4}$$

$$= 0.998$$

$$(r_o)_2 = (f)(r_o)_1 = (0.998)(0.0028) = 0.0028 \text{ (hr} \cdot \text{sq ft} \cdot ^\circ\text{F)/Btu}$$

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.

$$RC_2 = 23 \text{ (RC}_1 = 10 \text{ per note on Fig. 9-11)}$$

Obtain the number of crossflow spaces, which is one more than the number of baffles, from Fig. 9-12.

$$SP_2 = 19 \text{ [} SP_1 = 1 \text{ per note on Fig. 9-11 since } (\Delta P_o)_1 \text{ is for one baffle space.]}$$

$$f = \left(\frac{\mu_2}{\mu_1}\right)^{0.15} \left(\frac{G_2}{G_1}\right)^{1.85} \left(\frac{\rho_1}{\rho_2}\right) \left(\frac{D_{o1}}{D_{o2}}\right)^{0.15} \left(\frac{SP_2}{SP_1}\right) \left(\frac{RC_2}{RC_1}\right)$$

$$= \left(\frac{0.34}{0.549}\right)^{0.15} \left(\frac{121.4}{132.4}\right)^{1.85} \left(\frac{46.8}{41.2}\right) \left(\frac{0.625}{0.750}\right)^{0.15} \left(\frac{19}{1}\right) \left(\frac{23}{10}\right)$$

$$= 38.3$$

$$(\Delta P_o)_2 = (f)(\Delta P_o)_1 = (38.3)(0.25) = 9.58 \text{ psi}$$

6. Calculate the tube metal resistance.

$$D_o = 0.75 \text{ in.}$$

$$D_i = 0.62 \text{ in.}$$

$$k_w = 30 \text{ Btu/[(hr} \cdot \text{sq ft} \cdot ^\circ\text{F)/ft]}, \text{ from Fig. 9-8.}$$

$$r_w = \frac{D_o}{24 k_w} \left[\ln \left(\frac{D_o}{D_i} \right) \right] = \frac{0.75}{(24)(30)} \left[\ln \left(\frac{0.75}{0.62} \right) \right]$$

$$= 0.0002 \text{ (hr} \cdot \text{sq ft} \cdot ^\circ\text{F)/Btu}$$

7. Calculate the overall heat transfer coefficient.

$$\Sigma r = r_i \left(\frac{A_o}{A_i} \right) + r_o + r_w + r_{f0} + r_{fi} \left(\frac{A_o}{A_i} \right)$$

$$= 0.0032 \left(\frac{0.1963}{0.1623} \right) + 0.0028 + 0.0002 + 0.0020$$

$$+ 0.001 \left(\frac{0.1963}{0.1623} \right) = 0.0101$$

$$U = \frac{1}{\Sigma r} = \frac{1}{0.0101} = 99.0 \text{ Btu/(hr} \cdot \text{sq ft} \cdot ^\circ\text{F)}$$

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.

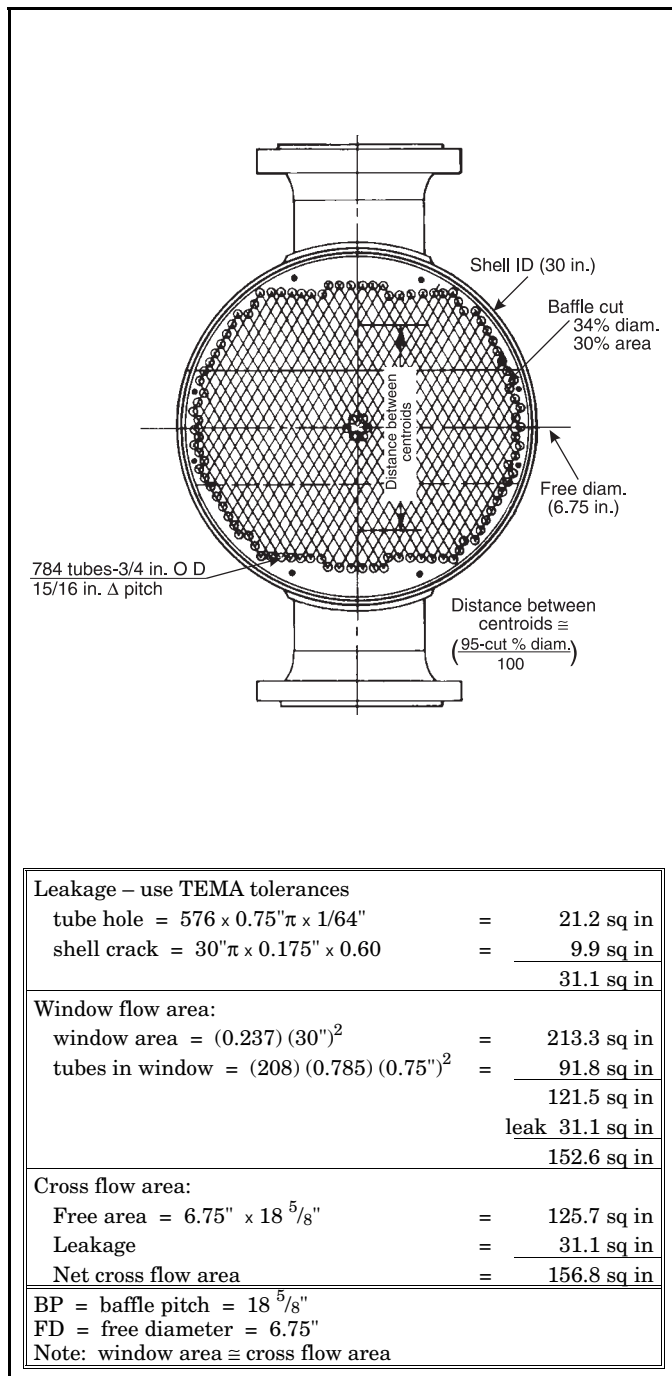
Condensing the overhead vapors of a distillation column is an example of condensing a mixed vapor stream. A vertical exchanger flanged directly to the top of the column might be used. The condensed liquid drains back into the column counter-current to the vapor entering the condenser. The major concerns 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 determination¹ 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.

FIG. 9-12

Heat Exchanger Specification Sheet

1											Job No.
2	Customer										Reference No.
3	Address										Proposal No.
4	Plant Location										Date Rev.
5	Service of Unit Lean Oil to Rich Oil Exchanger										Item No.
6	Size	30-360	Type	NEN	(Hor/Vert)	Connected In		Parallel	Series		
7	Surf./Unit (Gross/Eff.)	4525	Sq Ft.	Shells/Unit	One	Surf/Shell (Gross/Eff.)		4525	SqFt		
8	PERFORMANCE OF ONE UNIT										
9	Fluid Allocation					Shell Side			Tube Side		
10	Fluid Name					Lean Oil			Rich Oil		
11	Fluid Quantity, Total					Lb/Hr	475,723			650,860	
12	Vapor (In/Out)										
13	Liquid					475,723	475,723	650,860	650,860		
14	Steam										
15	Water										
16	Noncondensable										
17	Temperature (In/Out)					°F	197	100	60	124	
18	Density (#/ft3) AVG.						41.2		38.3		
19	Viscosity, Liquid (AVG.)						0.34		0.21		
20	Molecular Weight, Vapor										
21	Molecular Weight, Noncondensable										
22	Specific Heat (AVG.)					Btu/Lb °F	0.542		0.600		
23	Thermal Conductivity					Btu Ft / Hr Sq Ft °F	0.077 (AVG.)		0.078 (AVG.)		
24	Latent Heat					Btu/Lb @ °F					
25	Inlet Pressure					Psia	124.7		449.7		
26	Velocity					Ft/S					
27	Pressure Drop, Allow./Calc.					PSI	12 / 9.58		10 / 0.51		
28	Fouling Resistance (Min.)						0.0020		0.0010		
29	Heat Exchanged					25,000,000	Btu/Hr: MTD (Corrected)		54.9	°F	
30	Transfer Rate, Service					100.6	Clean		148.4	Btu/Hr Sq Ft °F	
31	CONSTRUCTION OF ONE SHELL										Sketch (Bundle/Nozzle Orientation)
32						Shell Side			Tube Side		
33	Design/Test Pressure					Psig	185 /Code	500 /Code			
34	Design Temperature					°F	650	650			
35	No. Passes per Shell						One	One			
36	Corrosion Allowance					In.	0.0625	0.0625			
37	Connections					In	10"	14"			
38	Size & Rating					Out	10"	14"			
39						Intermediate					
40	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
41	Tube Type SA-214 (welded steel)					Material					
42	Shell SA-516-70 ID 30 OD In.					Shell Cover					(Integ.) (Remov.)
43	Channel or Bonnet (2) SA-516-70 (Int.)					Channel Cover (2) SA-516-70 (Rem.)					
44	Tubesheet-Stationary (2) SA-516-70					Tubesheet -Floating					
45	Floating Head Cover					Impingement Protection					
46	Baffles-Cross SA-283-C Type Segmented					% Cut (Diam/Area)	34	Spacing: c/c		18-5/8	
47	Baffles-Long					Seal Type					
48	Supports-Tube SA-283-C					U-Bend	Type				
49	Bypass Seal Arrangement Spacers					Tube-Tubesheet Joint					
50	Expansion Joint NONE					Type					
51	pv ² -Inlet Nozzle 1717					Bundle Entrance	2220	Bundle Exit		1210	
52	Gaskets-Shell Side					Tube Side					
53	-Floating Head										
54	Code Requirements ASME Sec. VIII Div 1 (with Stamp)					TEMA Class "C"					
55	Weight/Shell 20,600					Filled with Water	29,900	Bundle		Lb.	
56	Remarks										

FIG. 9-13
Heat Exchanger Detail Design Results



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 ("dry-wall" 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)

$$U_v = 69.9 \text{ Btu/(hr} \cdot \text{sq ft} \cdot ^\circ\text{F)}$$

$$h_v = 111.1 \text{ Btu/(hr} \cdot \text{sq ft} \cdot ^\circ\text{F)}$$

Condensing: (108°F to 108°F)

$$U_c = 140.0 \text{ Btu/(hr} \cdot \text{sq ft} \cdot ^\circ\text{F)}$$

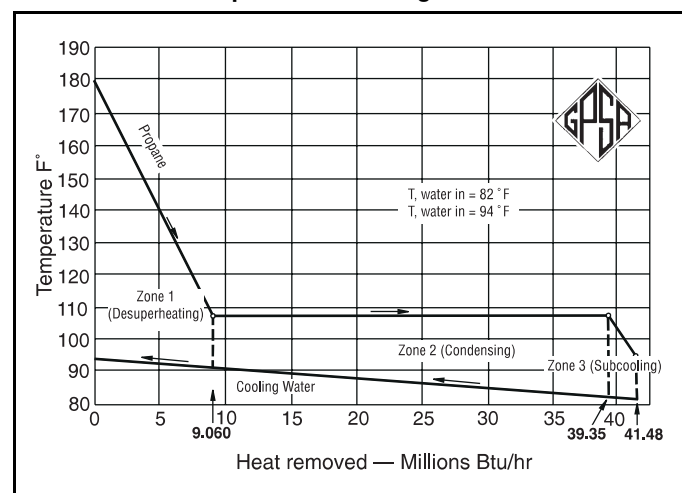
Subcooling: (108°F to 95°F)

$$U_L = 114.5 \text{ Btu/(hr} \cdot \text{sq ft} \cdot ^\circ\text{F)}$$

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



$$T_{wo} = T_v - [U_v (T_v - T_c)/h_v]$$

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.

$$T_{wo} = 180 - \left[\frac{69.9 (180 - 94)}{111.1} \right] = 125.90^\circ\text{F}$$

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 .

$$Q_v = 9,060,000 \text{ Btu/hr from Fig. 9-14.}$$

$$Q_1 = \frac{(180 - 133)}{(180 - 108)} (9,060,000) \\ = 5,910,000 \text{ Btu/hr (Heat removed from vapor between } 180^\circ\text{F and } 133^\circ\text{F)}$$

$$T_c = 94 - \left[\frac{5,910,000}{41,480,000} (94 - 82) \right] = 92.3^\circ\text{F}$$

(T_c is the water temperature at $Q_v - Q_1$, see Fig. 9-14)

$$T_{wo} = 133 - \left[\frac{69.9}{111.1} (133 - 92.3) \right] = 107.4^\circ\text{F}$$

Include the remainder of the desuperheating duty in the condensing zone. (Zone 2.)

ZONE 1

$$Q = Q_1 = 5,910,000 \text{ Btu/hr from above.}$$

$$\begin{array}{ccc} 180 & \rightarrow & 133.0 \\ \frac{94}{86} & \leftarrow & \frac{92.3}{40.7} \end{array}$$

$$LMTD = \frac{86 - 40.7}{\ln(86/40.7)} = 60.6^\circ\text{F}$$

$$A = \frac{5,910,000}{(69.9)(60.6)} = 1,395 \text{ ft}^2$$

ZONE 2

$$Q = Q_v - Q_1 + Q_c \quad (Q_c \text{ from Fig. 9-14})$$

$$= 9,060,000 - 5,910,000 + 30,300,000$$

$$= 33,450,000 \text{ Btu/hr}$$

$$T_c = 94 - \frac{(5,910,000 + 33,450,000)}{41,480,000} (94 - 82) = 82.6^\circ\text{F}$$

(NOTE: You do **not** use 133°F as the inlet temperature to this zone.)

$$\begin{array}{ccc} 108.0 & \rightarrow & 108.0 \\ \frac{92.3}{15.7} & \leftarrow & \frac{82.6}{25.4} \end{array}$$

$$LMTD = \frac{25.4 - 15.7}{\ln(25.4/15.7)} = 20.2^\circ\text{F}$$

$$A = \frac{33,450,000}{(140.0)(20.2)} = 11,830 \text{ ft}^2$$

ZONE 3

$$Q = Q_L = 2,120,000 \text{ Btu/hr from Fig. 9-14}$$

$$\begin{array}{ccc} 108.0 & \rightarrow & 95 \\ \frac{82.6}{25.4} & \leftarrow & \frac{82}{13} \end{array}$$

$$LMTD = \frac{25.4 - 13}{\ln(25.4/13)} = 18.5^\circ\text{F}$$

$$A = \frac{2,120,000}{(114.5)(18.5)} = 1000 \text{ ft}^2$$

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.

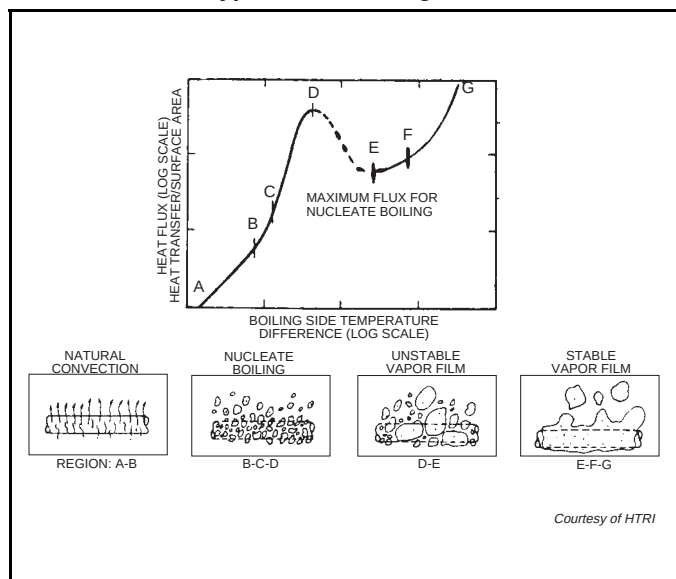
REBOILERS AND VAPORIZERS

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)

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

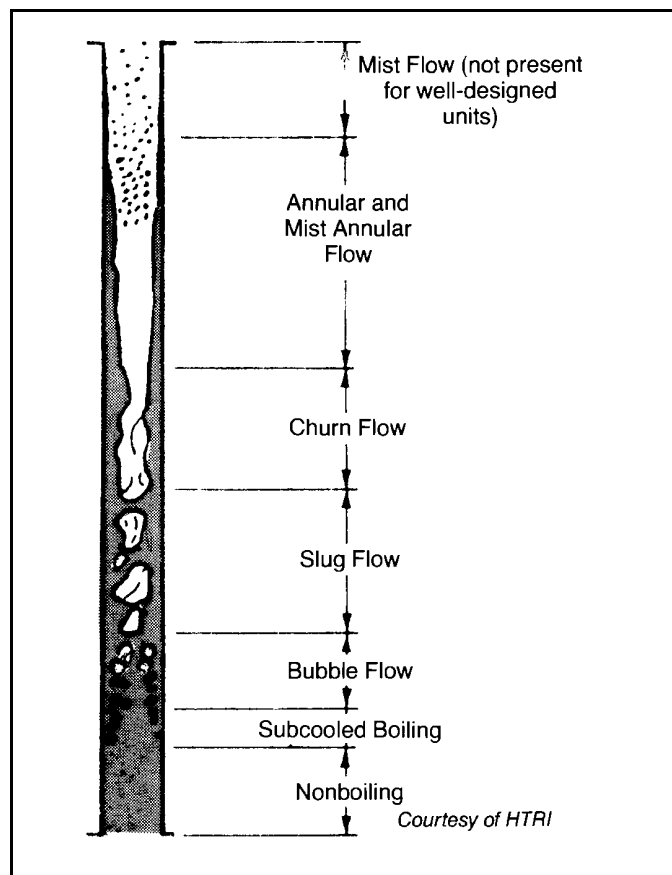
When the geometric flowpath of a boiling fluid is well defined (all boiling except pool boiling), the effects of liquid and vapor velocities are part of a design or operating analysis. Liquid and vapor co-exist in 'regimes' as illustrated in Fig. 9-17. Typically, these regimes progress to termination 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

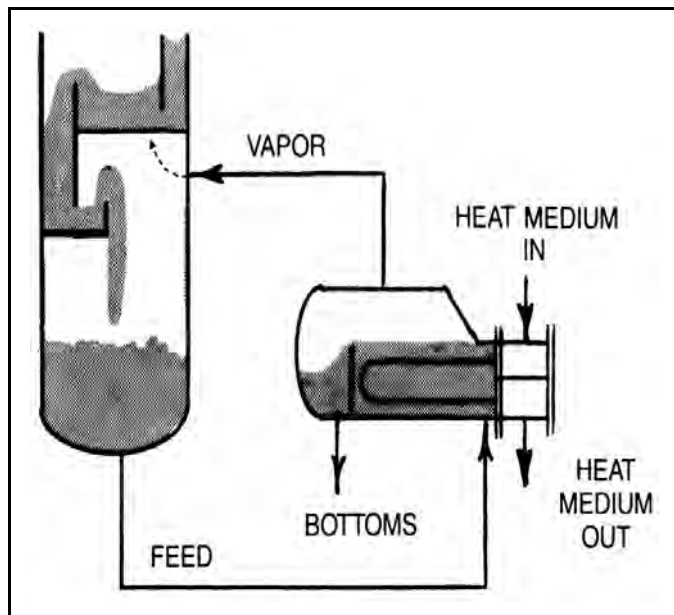
Two-Phase Flow Regimes in Vertical Tubes



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

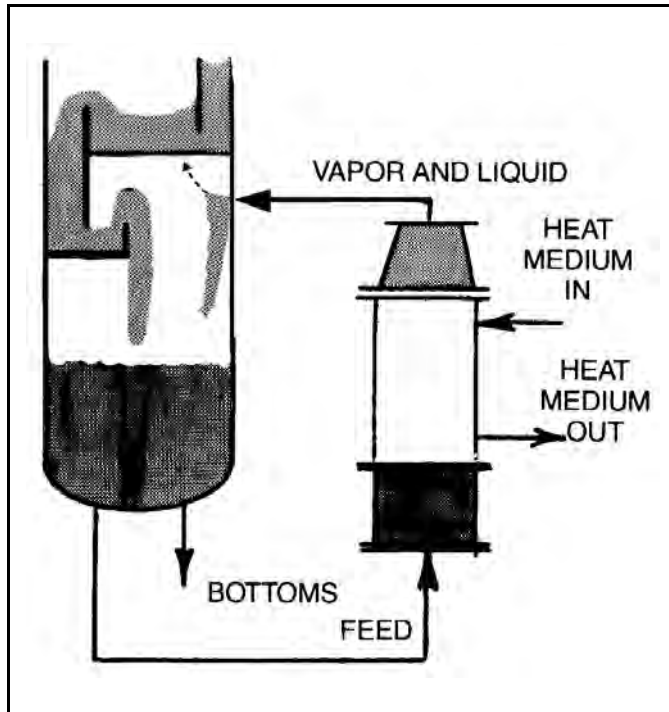


FIG. 9-20

Once-Through Reboiler with Bottom Tray Feed

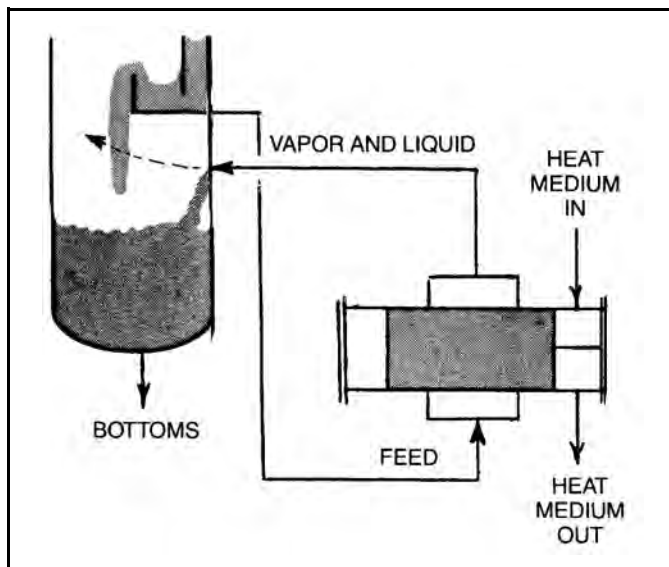
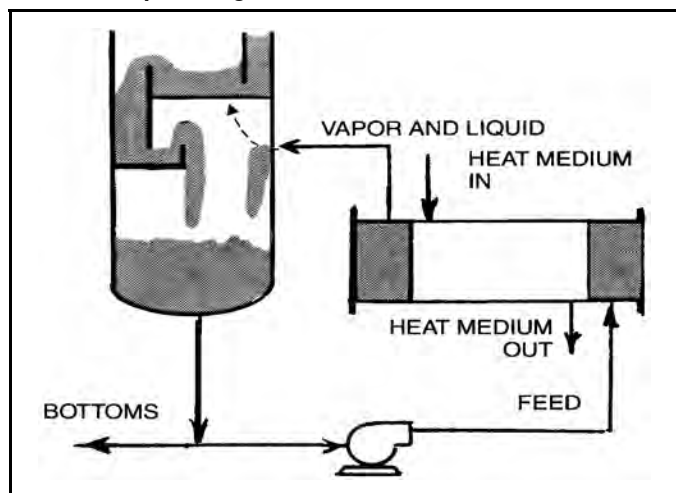


FIG. 9-21

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, gen-

eral 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.

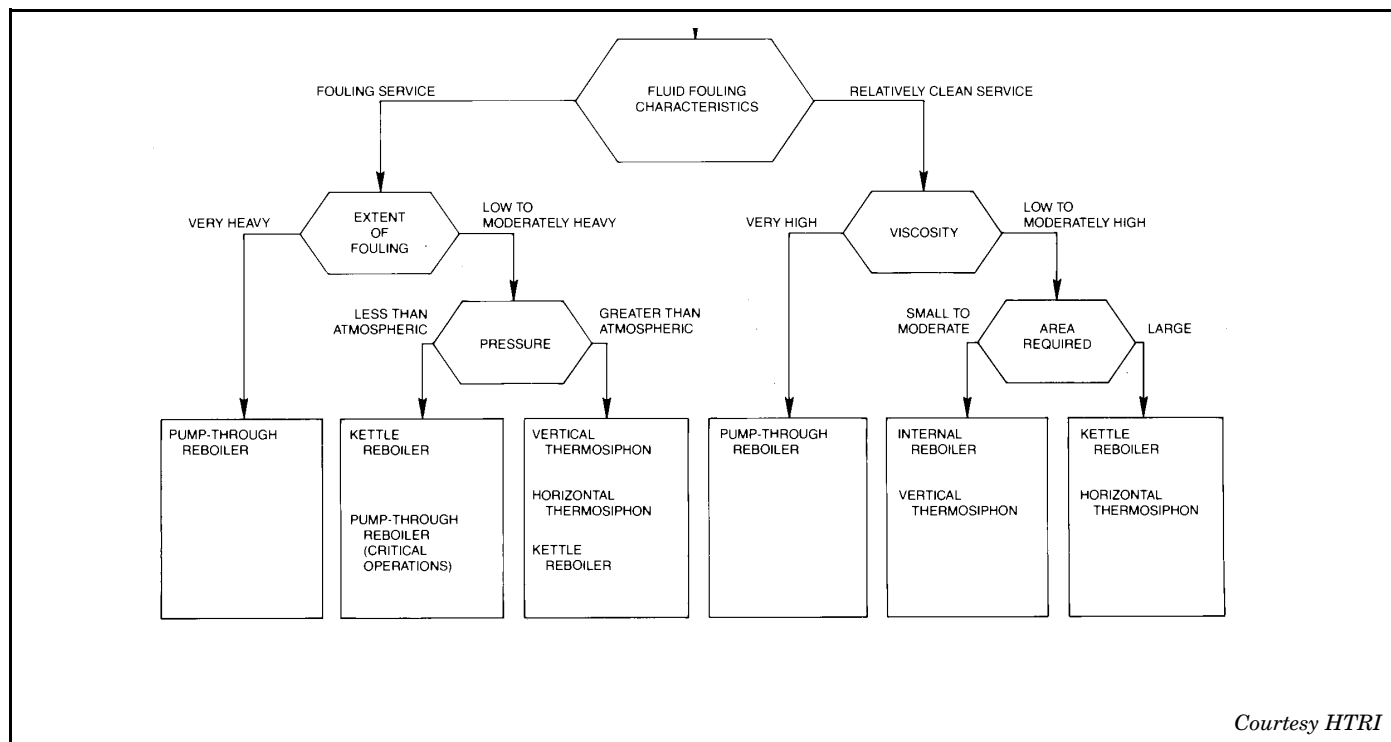
Detailed understanding of shell and tube exchangers for use in the process industry requires an understanding of the TEMA Standards. Other industry standards as may be offered by ASME, API, or ANSI can be applied in a particular situation with or without TEMA Standards. The purchase order and specification 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

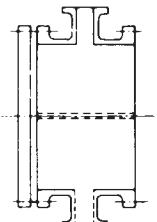
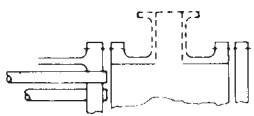
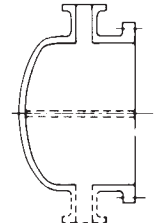
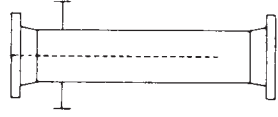
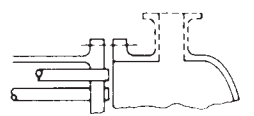
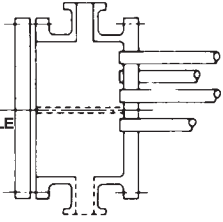
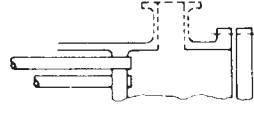
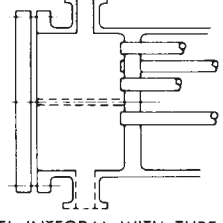
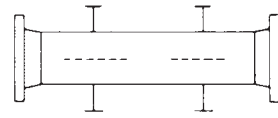
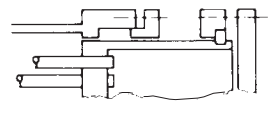
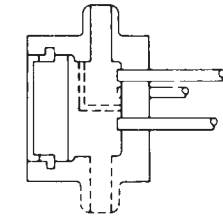
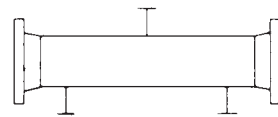
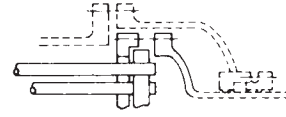
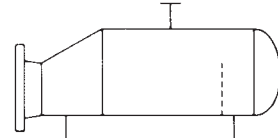
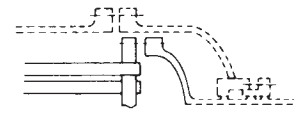
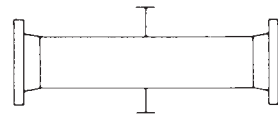
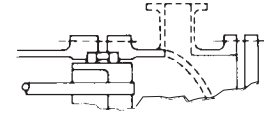
Reboiler Selection Chart



Courtesy HTRI

FIG. 9-23

Shell and Tube Exchanger Nomenclature

FRONT END STATIONARY HEAD TYPES		SHELL TYPES		REAR END HEAD TYPES	
A		E		L	
	CHANNEL AND REMOVABLE COVER		ONE PASS SHELL		FIXED TUBESHEET LIKE "A" STATIONARY HEAD
					
	BONNET (INTEGRAL COVER)		TWO PASS SHELL WITH LONGITUDINAL BAFFLE		FIXED TUBESHEET LIKE "B" STATIONARY HEAD
					
C	REMOVABLE TUBE BUNDLE ONLY	G	SPLIT FLOW	N	FIXED TUBESHEET LIKE "N" STATIONARY HEAD
N		H		P	
	CHANNEL INTEGRAL WITH TUBE- SHEET AND REMOVABLE COVER		DOUBLE SPLIT FLOW		OUTSIDE PACKED FLOATING HEAD
D		J		S	
	SPECIAL HIGH PRESSURE CLOSURE	J	DIVIDED FLOW		FLOATING HEAD WITH BACKING DEVICE
		K		T	
			KETTLE TYPE REBOILER	T	PULL THROUGH FLOATING HEAD
		X		U	
			CROSS FLOW	U	U-TUBE BUNDLE
				W	
				W	EXTERNALLY SEALED FLOATING TUBESHEET

Courtesy of TEMA

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 $1.075 \times 1.05 \times 1.02 = 1.15$.
Partially corrected diameter = $25" \times 1.15 = 28.75"$.
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

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	0.2736	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:

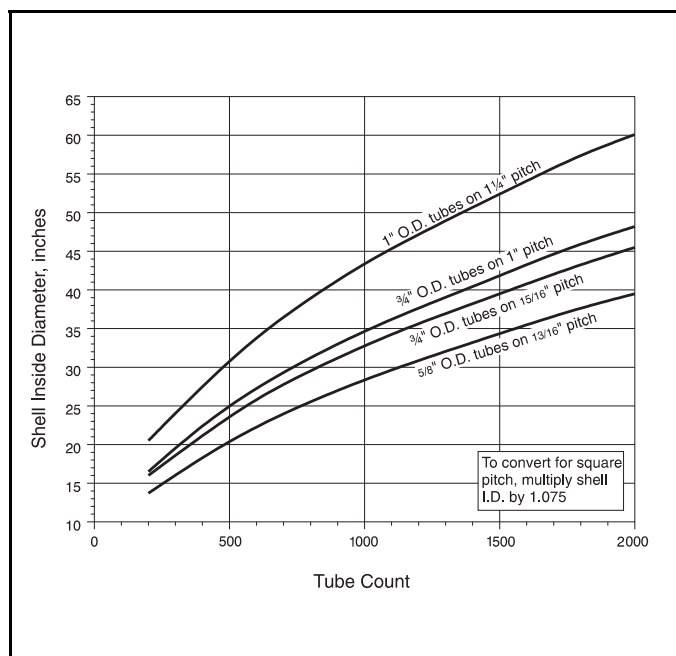
Aluminum	0.35	Aluminum Bronze	1.04	Nickel	1.13
Titanium	0.58	Aluminum Brass	1.06	Nickel-Copper	1.12
A.I.S.I. 400 Series Stainless Steels	0.99	Nickel-Chrome-Iron	1.07	Copper and Cupro-Nickels	1.14
A.I.S.I. 300 Series Stainless Steels	1.02	Admiralty	1.09		

** Liquid Velocity = $\frac{\text{Lbs Per (Tube} \cdot \text{Hour)}}{(\text{C}) (\text{Sp Gr of Liquid})}$ in feet per sec (Sp Gr of Water at 60°F = 1.0)

Courtesy of TEMA

FIG. 9-26

Tube Count vs. Diameter for Triangular Tube Pitch



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 for-

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"

mation 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 non-proprietary 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.

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.

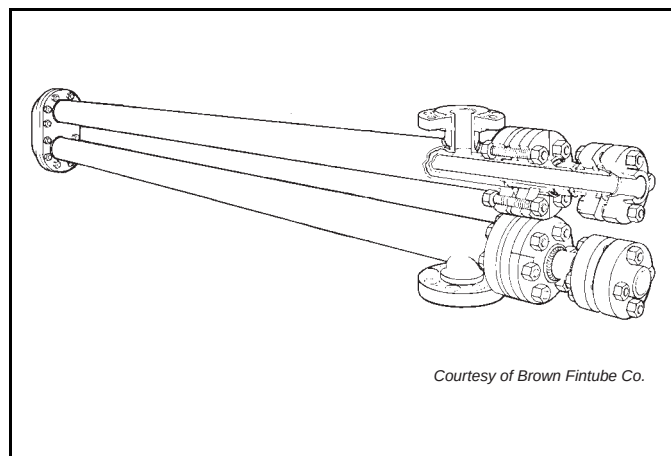
Hairpin sections are specially designed units which are normally not built to any industry standard other than ASME Code. However, TEMA tolerances are normally incorporated, wherever applicable.

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



Disadvantages

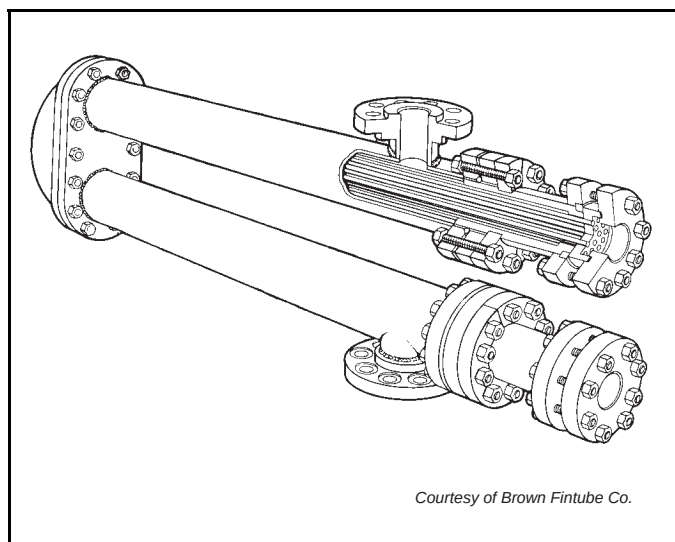
1. Multiple hairpin sections are not always economically competitive with a single shell and tube heat exchanger.
2. Proprietary closure design requires special gaskets.

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):

$$UA = \frac{Q}{LMTD}$$

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.

- $UA = 150,000 \text{ Btu}/(\text{hr} \cdot ^\circ\text{F})$

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.

- $UA = 100,000 \text{ to } 150,000 \text{ Btu}/(\text{hr} \cdot ^\circ\text{F})$

In this range, one or more 12" to 16" multitube sections will normally be required.

- $UA = 50,000 \text{ to } 100,000 \text{ Btu}/(\text{hr} \cdot ^\circ\text{F})$

In this range, one or more 4" to 12" multitube sections will normally be required.

- $UA = 50,000 \text{ Btu}/(\text{hr} \cdot ^\circ\text{F})$

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.

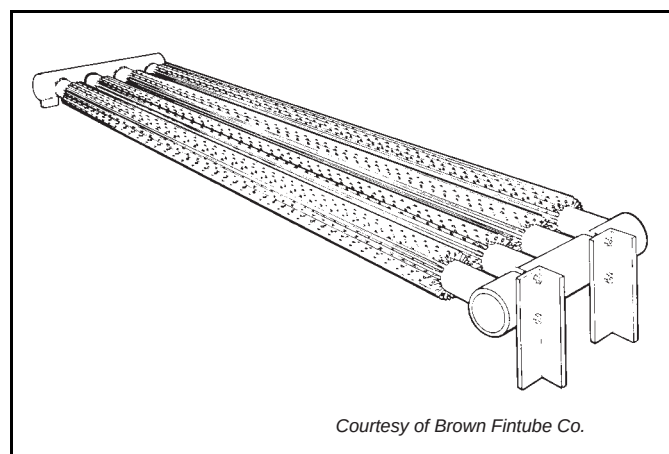
Tank Suction Heaters

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.)	$\frac{3}{4}$ " - 4"	$\frac{3}{4}$ " - 1"
No. of Longitudinal Fins	0 - 72	0 - 24
Fin Height (in.)	0 - $\frac{3}{4}$ "	0 - $\frac{1}{2}$ "
Surface (sq ft/20 ft)	10 - 40	75 - 1500

FIG. 9-32
Prefabricated Tank Heater



Courtesy of Brown Fintube Co.

FIG. 9-33
Tank Suction Heater

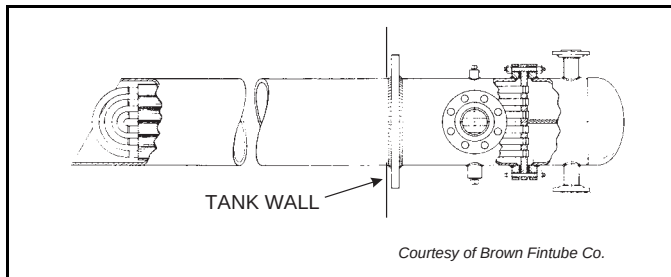


Plate-Fin Exchangers (Brazed Aluminum)

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

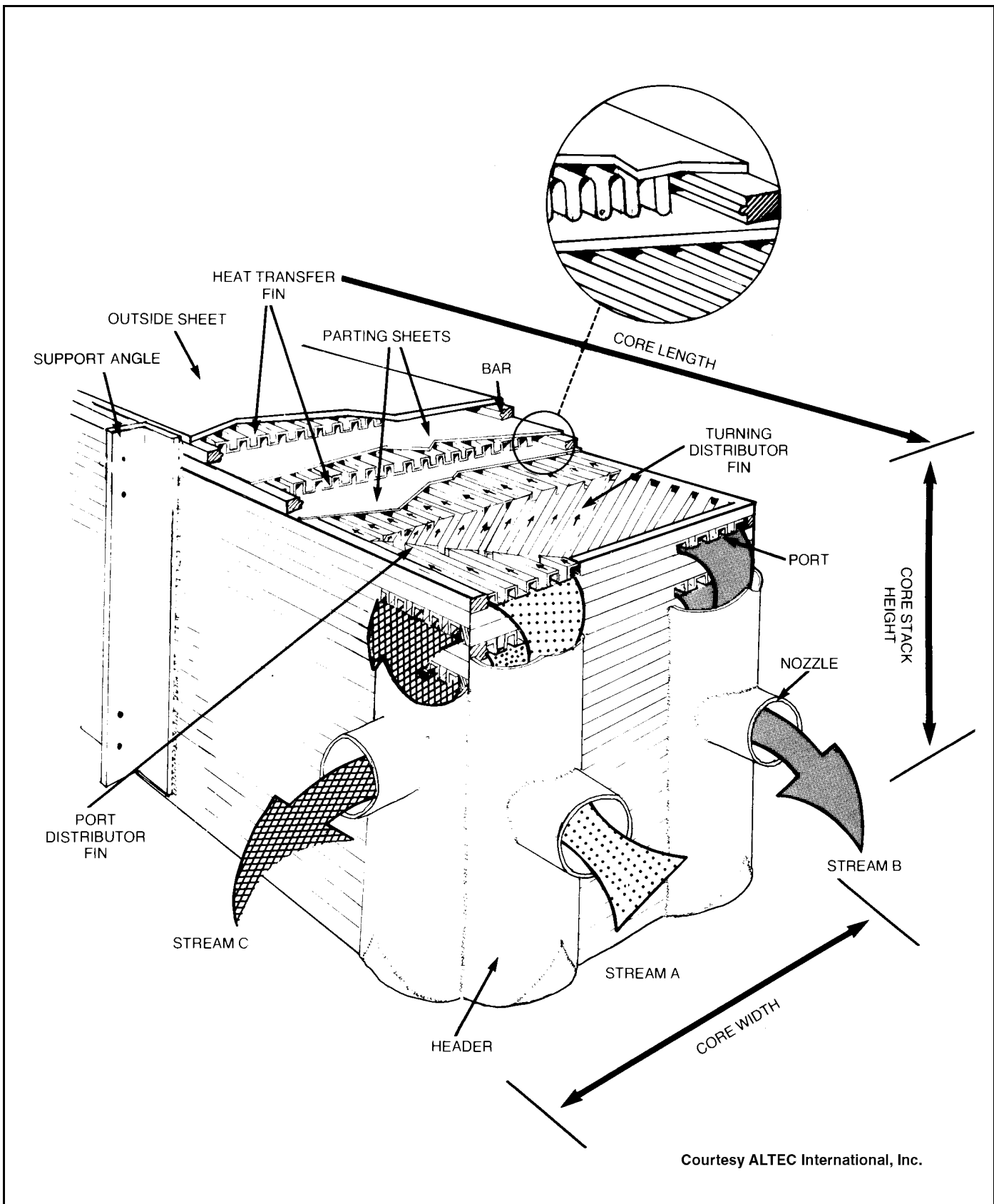
Aluminum maintains excellent strength and ductility to temperatures as low as -452°F . Aluminum actually increases in strength at cold temperatures. Because of its relatively low melting temperature, however, aluminum is less resistant to fires and high temperatures than are some other materials. Aluminum 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

FIG. 9-34

Basic Components of a Three Stream Counterflow Brazed Aluminum Heat Exchanger



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

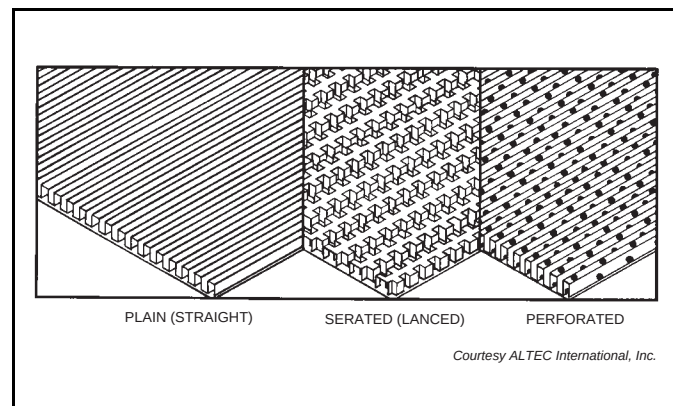
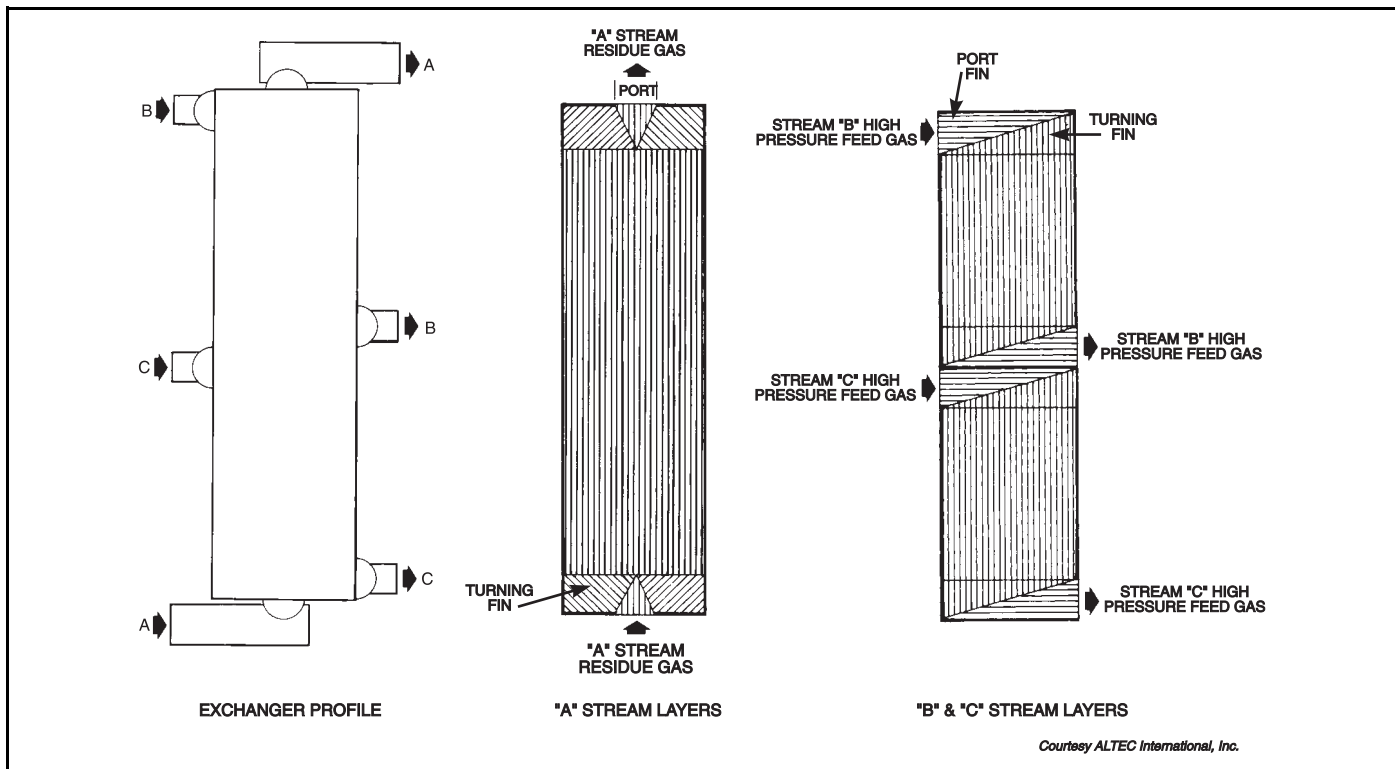


FIG. 9-37

Typical Fin Arrangements for Gas/Gas Exchanger



SELECTIONS

Braze Aluminum Heat Exchanger Specifications

The design and specification of a braze 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

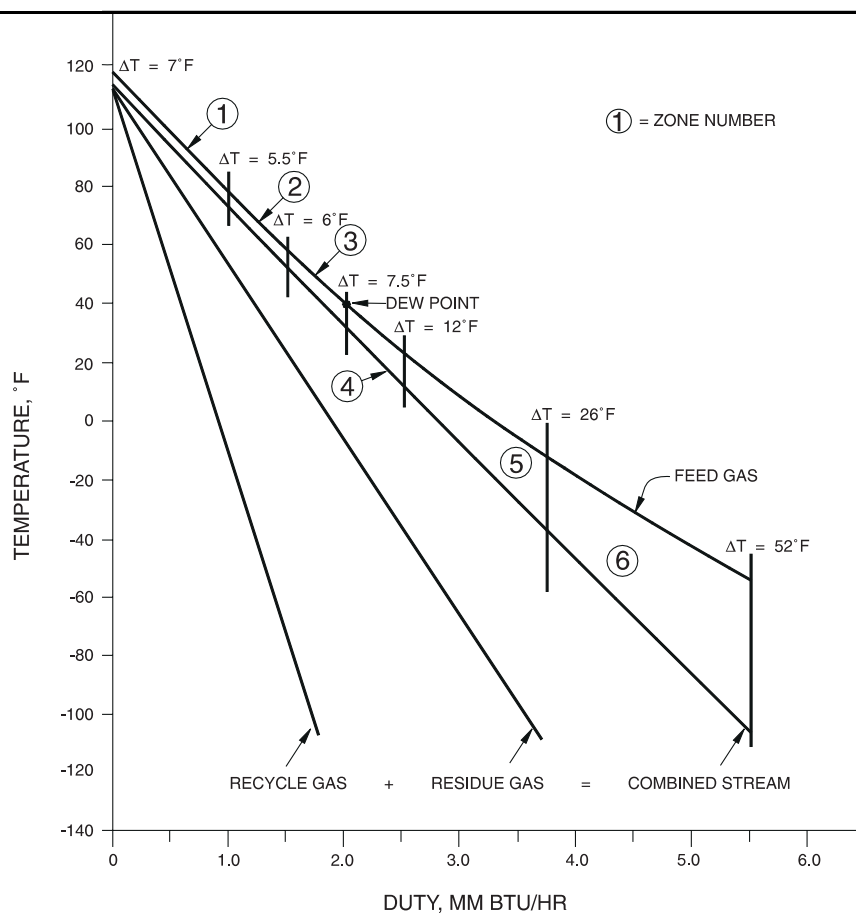
FIG. 9-38

Braze Aluminum Heat Exchanger Specifications

1							Specification No.	Typical	Rev	Pg 1	Of 1
2							Date				
3							Customer				
4							Project Job				
5							Location				
6	Plant Service	GAS TO GAS EXCHANGER					Item No.	E-1			
7	Stream	A) Feed Gas		B) Residue Gas		C) Recycle Gas					
8	Stream Components	Mole %		Mole %		Mole %		Mole %			
9	N ₂ /CO ₂	4.0/0.6		4.5/0.4		4.0/0.4					
10	C1	83.8		93.7		92.5					
11	C2	6.3		1.3		2.1					
12	C3	3.1		0.1		1.0					
13	iC ₄ /nC ₄	0.4/1.0		-		-					
14	iC ₅ /nC ₅	0.2/0.6		-		-					
15	Total	100.0		100.0		100.0					
16	Total Flow	lb/hr	40650		31320		14780				
17	Conditions At	Inlet	Outlet	Inlet	Outlet	Inlet	Outlet	Inlet	Outlet		
18	Vapor Flow Rate	lb/hr	40650	32195	31320	31320	14780	14780			
19	Molecular Weight		19.5	17.9	16.9	16.9	17.2	17.2			
20	Density	lb/ft ³	2.79	4.84	1.02	0.56	1.32	0.80			
21	Viscosity	lb/ft ³ -hr	0.031	0.028	0.019	0.029	0.020	0.029			
22	Liquid Flow Rate	lb/hr		8455							
23	Molecular Weight			30.1							
24	Density	lb/ft ³		28.0							
25	Viscosity	lb/ft ³ -hr		0.15							
26	Temperature	°F	120	-54	-106	113	-106	113			
27	Pressure	PSIA	810		205		285				
28	Fluid Cond. or Vap.	lb/hr	8455 (Cond.)								
29	Avg. Specific Heat (Vap/Liq)	BTU/lb°F	0.638/0.578		0.520		0.532				
30	Avg. Thermal Cond. (Vap/Liq)	*	0.0204/0.570		0.0167		0.0165				
31	Latent Duty	BTU/hr									
32	Total Duty	BTU/hr	5,500,000		3,700,000		1,800,000				
33	Corrected MTD	°F	11.22								
34	Allowable Pressure Drop	PSI	10.0		7.0		7.0				
35	Fouling Resistance	hr-°F-ft ³ -BTU	0.001		0.001		0.001				
36	Design Temp. (Max/Min)	°F	-300/+150		-300/+150		-300/+150				
37	Design Pressure (Max)	PSIG	1100		300		400				
38	Core Width (in)	25	No. Layers Core	63	Total No. Cores	1	In Parallel Series	1			
39	Core Stack Ht. (in)	23-1/4	? Sheet Thk (in)	0.080	No. Assemblies	1	Net Core (Ass'y) Wt.	5650		Lbs.	
40	Core Length (in)	180	Outside Sheet Thk (in)	0.25	No. Cores per Ass'y	1	Shipping Core (Ass'y) Wt.	6000		Lbs.	
41	No. Layers per Core		21		30		12				
42	Fin Surface Type		Serrated		Serrated		Serrated				
43	Fin Height	in	0.280		0.280		0.280				
44	Fin Thickness & Spacing	in x FPI	0.016 x 17		0.016 x 17		0.016 x 17				
45	Effective Passage Width	in	23-3/8		23-3/8		23-3/8				
46	Total Free Flow Area	ft ²	0.65		0.93		0.37				
47	Effective Passage Length	in	163-3/8		162-3/8		162-3/8				
48	Total Heat Transfer Surface	ft ²	5910		8390		3360				
49	Frictional Pressure Drop	PSI	8.8		6.9		6.4				
50	Header Size	HPS	8	8	10	10	8	8			
51	Nozzle Size	HPS	6	6	8	8	6	6			
52	Manifold Size	HPS	-	-	-	-	-	-			
53	Type Connections		600 Lb.		300 Lb.		300 Lb.				
54	Test Pressure (Hydro Air Leak)	PSIG	1650/1375		450/375		600/500				
55	Special Testing		None		None		None				

Courtesy ALTEC International, Inc.

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)	X _v	T (°F)	DUTY (MM BTU/HR)	X _v	T (°F)	DUTY (MM BTU/HR)	X _v
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

$$\text{CMTD} = \frac{Q}{UA} = \frac{5.5}{0.4902} = 11.22^{\circ}\text{F}$$

Courtesy ALTEC International, Inc.

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 Martinelli⁴ 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 homogeneous 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)

$$G_H = 18.6 \text{ lb/(sq ft} \cdot \text{sec)}$$

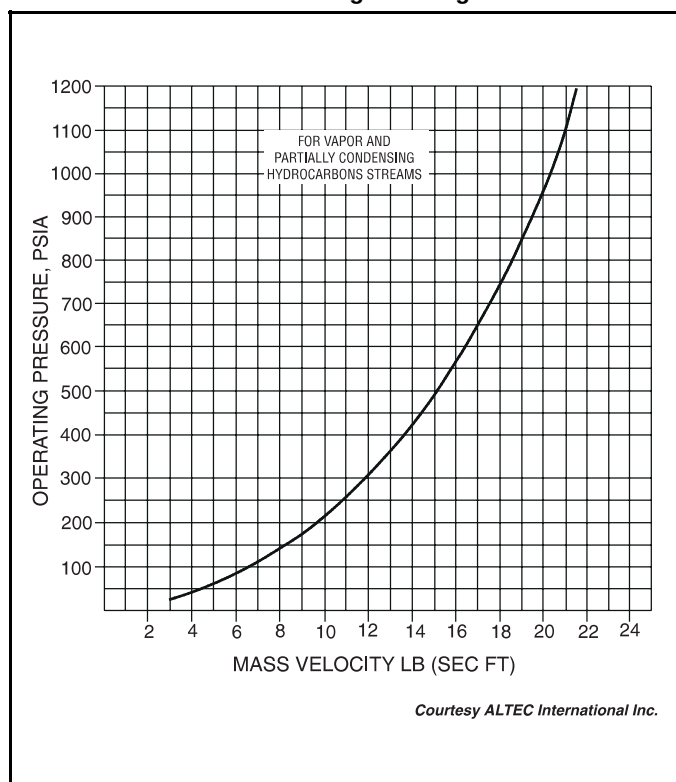
Residue Gas (205 psia)

$$G_C = 9.7 \text{ lb/(sq ft} \cdot \text{sec)}$$

Recycle Gas (285 psia)

FIG. 9-40

Typical Operating Mass Velocities
Gas Processing Exchangers



$$G_C = 11.6 \text{ lb/(sq ft} \cdot \text{sec)}$$

Using these G values, the exchanger cross section can be computed from the following equations:

$$H = \frac{0.083}{WN} \sum_{i=1}^{i=n} \left(\frac{\dot{m}_i}{G_i} \right), \text{ for gas to gas exchangers} \quad \text{Eq 9-15}$$

Where n = total number of all warm and cold streams

$$H = \frac{0.166}{WN} \left(\frac{\dot{m}_H}{G_H} \right), \text{ for demethanizer reboilers} \quad \text{Eq 9-16}$$

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):

$$\begin{aligned} UA &= UA_{\text{zone 1}} + UA_{\text{zone 2}} + \dots + UA_{\text{zone 6}} = \frac{Q}{\text{CMTD}} \\ &= \frac{5,500,000}{11.22} \\ &= 490,200 \text{ Btu/(hr} \cdot \text{°F)} \end{aligned}$$

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 —

$$L = \frac{UA}{WHN \sqrt{G_{\text{Total}}}} + 0.65W \quad \text{Eq 9-17}$$

For perforated fin exchangers —

$$L = \frac{1.5 UA}{WHN \sqrt{G_{\text{Total}}}} + 0.65W \quad \text{Eq 9-18}$$

Where $G_{\text{Total}} = (G_H)_{\text{Avg}} + (G_C)_{\text{Avg}}$

Eq 9-19

For demethanizer reboilers (where demethanizer liquids evaporate up to 20% of their mass):

For serrated fin exchangers —

$$L = \frac{0.67 UA}{WHN \sqrt{G_H}} + 0.65W \quad \text{Eq 9-20}$$

For perforated fin exchangers —

$$L = \frac{UA}{WHN \sqrt{G_H}} + 0.65W \quad \text{Eq 9-21}$$

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

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 —

$$\Delta P = \frac{(4.4) (10^{-4}) (L + 20) (G)^{1.8}}{\rho_m} \quad \text{Eq 9-22}$$

For partially condensing streams (up to 30% of mass condensed) —

$$\Delta P = \frac{(8.8) (10^{-4}) (L + 40) (G)^{1.8}}{\rho_m} \quad \text{Eq 9-23}$$

For Perforated Fin Exchangers:

For vapor streams —

$$\Delta P = \frac{(0.95) (10^{-4}) (L + 95) (G)^{1.8}}{\rho_m} \quad \text{Eq 9-24}$$

For partially condensing streams (up to 30% of mass condensed) —

$$\Delta P = \frac{(1.9) (10^{-4}) (L + 190) (G)^{1.8}}{\rho_m} \quad \text{Eq 9-25}$$

$$\text{where } \rho_m = \frac{2(\rho_{in}) (\rho_{out})}{\rho_{in} + \rho_{out}} \quad \text{Eq 9-26}$$

And when inlet or outlet conditions are two-phase:

$$\rho_{2\Phi} = \left(\frac{1}{\frac{X_V}{\rho_V} + \frac{X_L}{\rho_L}} \right) \quad \text{Eq 9-27}$$

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.

$$G_{new} = G_{old} \left[\frac{\Delta P_{allowed}}{\Delta P_{old}} \right]^{0.56} \quad \text{Eq 9-28}$$

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)

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.

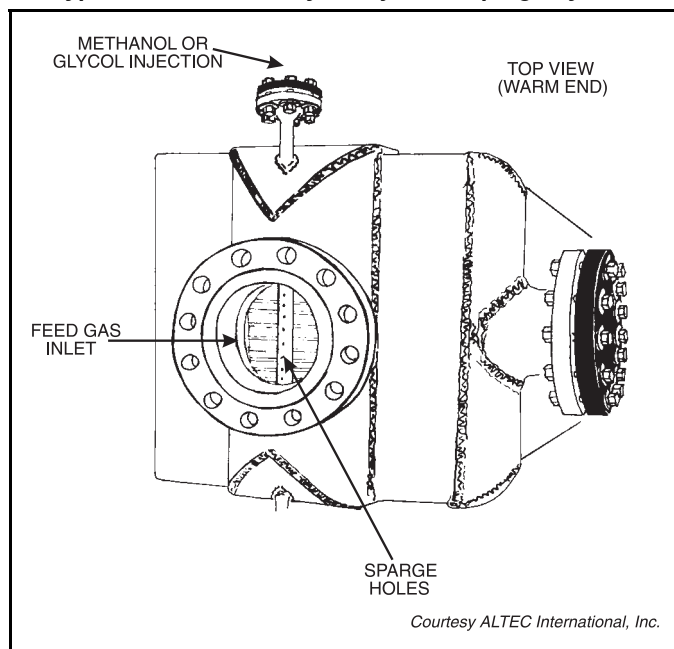
Hydrate Suppression

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



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.

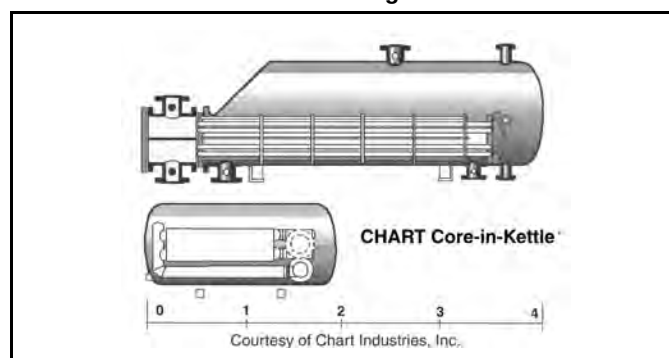
OTHER USES OF CORE BLOCKS

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



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.

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.

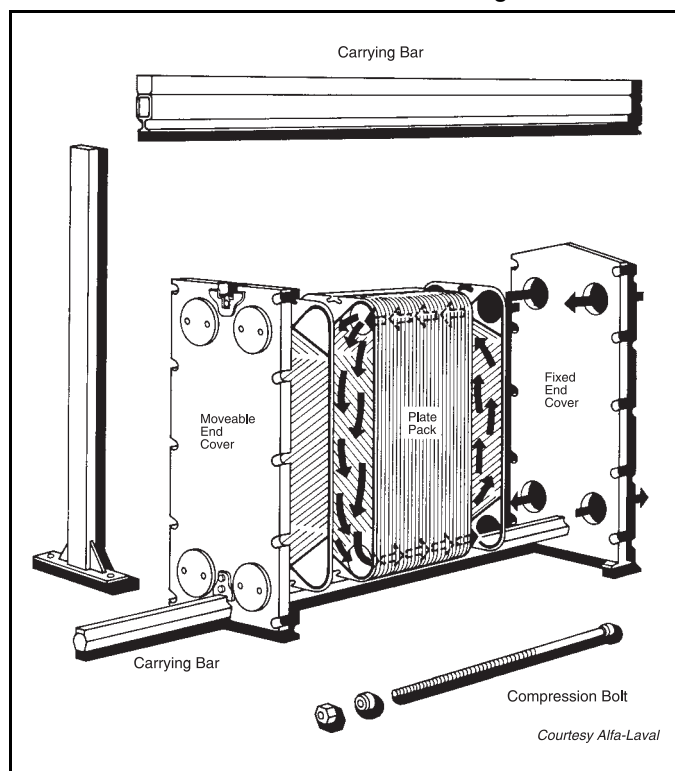
Since the plates are thin, the PHE gives a relatively high heat transfer coefficient for the mass of material required. When alloy materials are required, the PHE is competitive with more conventional heat exchanger designs.

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-

FIG. 9-43

Plate and Frame Heat Exchanger



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.

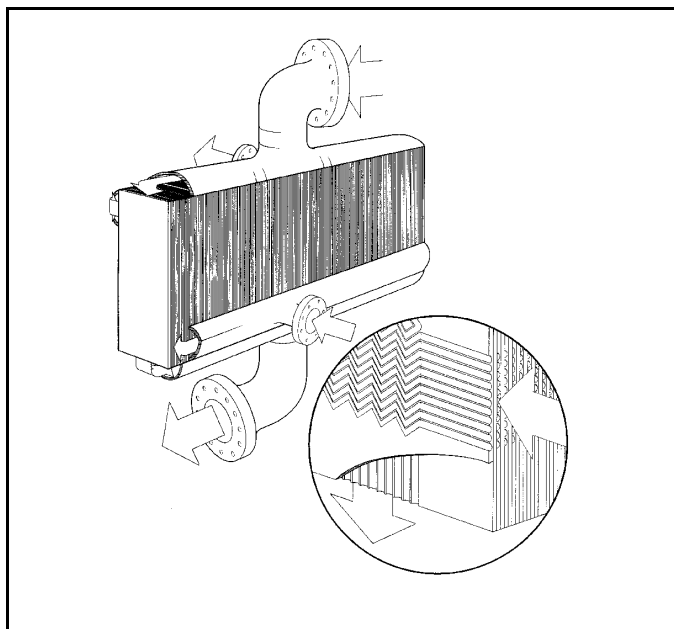
The thermal capacity of the exchanger is built to the required level by welding together diffusion bonded blocks to form the complete heat exchange core. Headers and nozzles are welded on to the core in order to direct the fluids to the appropriate sets of passages. Fig. 9-46 shows the complete construction 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-compatibility 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

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 ft²/ft³. 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 capability, 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 applicable 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	h_t = tube side film coefficient based on inside tube area, Btu/(h • sq ft • °F)
A_i = inside surface of tube, sq ft	HP = fan horsepower
A_b = outside bare tube surface, sq ft	J = J factor (see Fig. 10-13)
A_x = outside extended surface of tube, sq ft	k = thermal conductivity, Btu/[(hr • sq ft • °F)/ft]
A_t = tube inside cross-sectional area, sq in. (see Fig. 9-25)	L = length of tube, ft
ACFM = actual cubic feet per minute	LMTD = log mean temperature difference, °F (see Fig. 9-3)
APF = total external area/ft of fintube, sq ft/ft	N = number of rows of tubes in direction of flow
APSF = external area of fintube, sq ft/sq ft of bundle face area	N_f = number of fans
AR = area ratio of fintube compared to the exterior area of 1 in. OD bare tube	N_p = number of tube passes
B = correction factor, psi (see Fig. 10-14)	N_R = modified Reynolds number, (in • lb/(sq ft • s • cp))
C_p = specific heat at average temperature, Btu/(lb • °F)	N_t = number of tubes
CMTD = corrected mean temperature difference, °F	ΔP = pressure drop, psi
dB(A) = overall weighted level of sound at a point distant from noise source based on "A" weighting system	PF = fan total pressure, inches of water
D = fan diameter, ft	ρ_a = density of air, lb/ cu ft
D_i = inside tube diameter, in.	ρ_w = density of water, lb/ cu ft
D_o = outside tube diameter, in.	P = temperature ratio (see Fig. 10-8)
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)	PWL = sound pressure level
f = friction factor (see Fig. 10-15)	PWLN = PWL for N_f fans
F = correction factor (see Fig. 10-8)	Q = heat transferred, Btu/h
F_a = total face area of bundles, sq ft	r_d = fouling resistance (fouling factor), (hr • ft ² • °F/Btu)
F_p = air pressure drop factor, in. of water per row of tubes	r_f = fluid film resistance (reciprocal of film coefficient)
FAPF = fan area per fan, ft ² /fan	r_{mb} = metal resistance referred to outside bare surface
FPM = fan tip speed, feet per minute	r_{mx} = metal resistance referred to outside extended surface
g = local acceleration due to gravity, ft/s ²	R = distance in feet (see Eq. 10-6)
G = mass velocity, lb/(sq ft • s)	R = temperature ratio (see Fig. 10-8)
G_a = air face mass velocity, lb/(hr • sq ft) of face area	RPM = fan speed, rotations per minute
G_t = tubeside mass velocity, lb/(sq ft • s)	S = specific gravity (water = 1.0)
h_a = air side film coefficient Btu/(h • sq ft • °F)	SPL = sound pressure level
h_s = shell side film coefficient based on outside tube area, Btu/(h • sq ft • °F)	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

FIG. 10-1
Nomenclature (Cont'd.)

Y = correction factor, psi/ft (see Fig. 10-14)
 Δt = temperature change, °F
 μ = viscosity, cp
 μ_w = viscosity at average tube wall temperature, cp
 ϕ = viscosity gradient correction

Subscripts:

a = air side
b = bare tube surface basis
s = shell side
t = tube side
x = extended tube surface basis
1 = inlet
2 = outlet

charged upward at approximately $2\frac{1}{2}$ 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.

- 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 $\frac{1}{8}$ 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

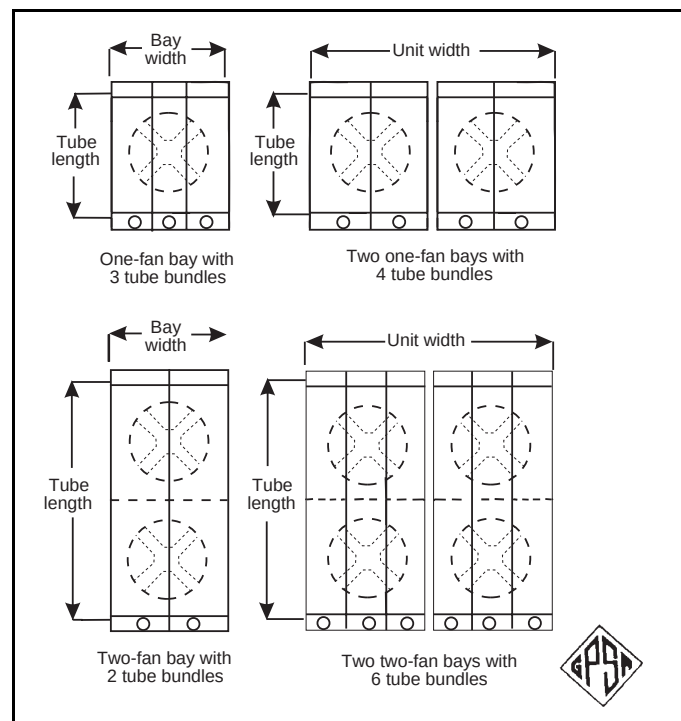
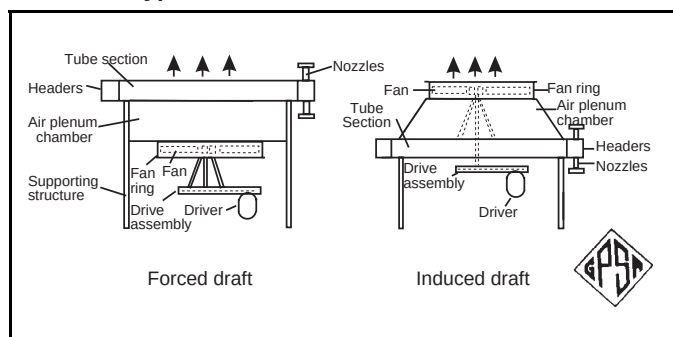


FIG. 10-2

Typical Side Elevations of Air Coolers



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.

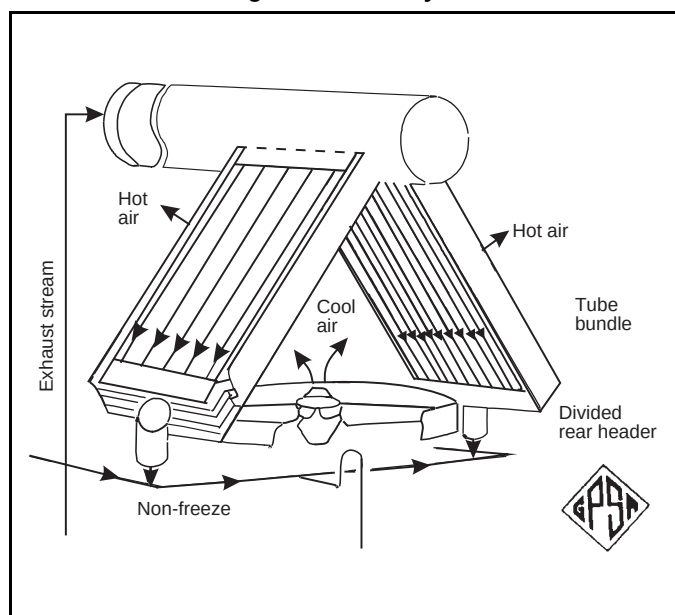
Two fan bays are popular, since this provides a degree of safety against fan or driver failure and also a method of control by fan staging. Fan coverage is the ratio of the projected area of the fan to the face of the section served by the fan. Good practice is to keep this ratio above 0.40 whenever 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 $\frac{1}{2} \times 9$) and $\frac{5}{8}$ in. high fins, 10 fins/in. (designated as $\frac{5}{8} \times 10$).

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

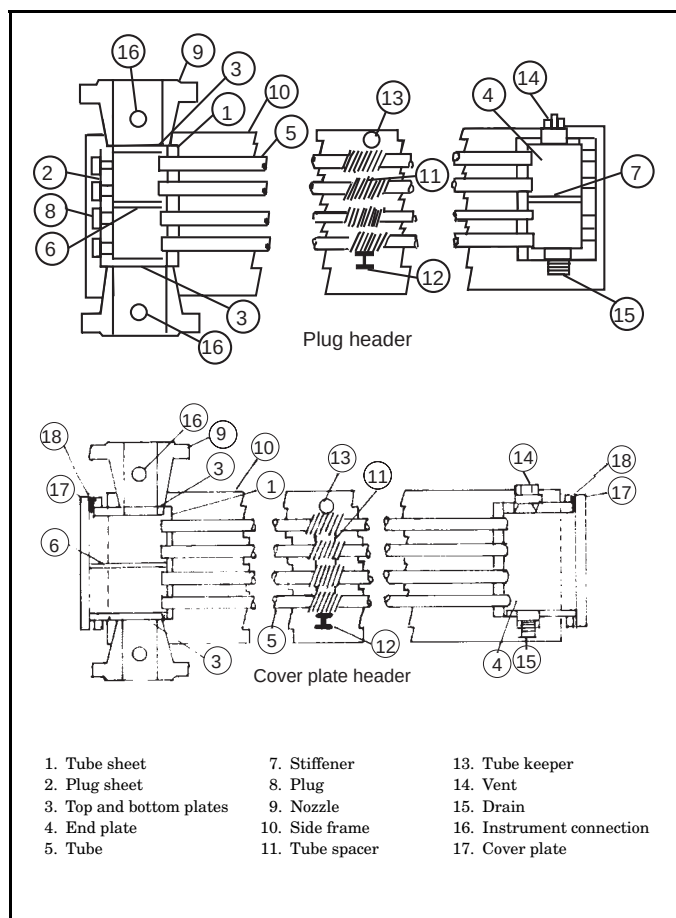
Air-cooled exchangers are sized to operate at warm (summer) air temperatures. Seasonal variation of the air temperature can result in over-cooling which may be undesirable. One way to control the amount of cooling is by varying the amount of air flowing through the tube section. This can be accomplished 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

FIG. 10-5

Typical Construction of Tube Section with Plug and Cover Plate Headers



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.

FIG. 10-6
Internal Recirculation Design

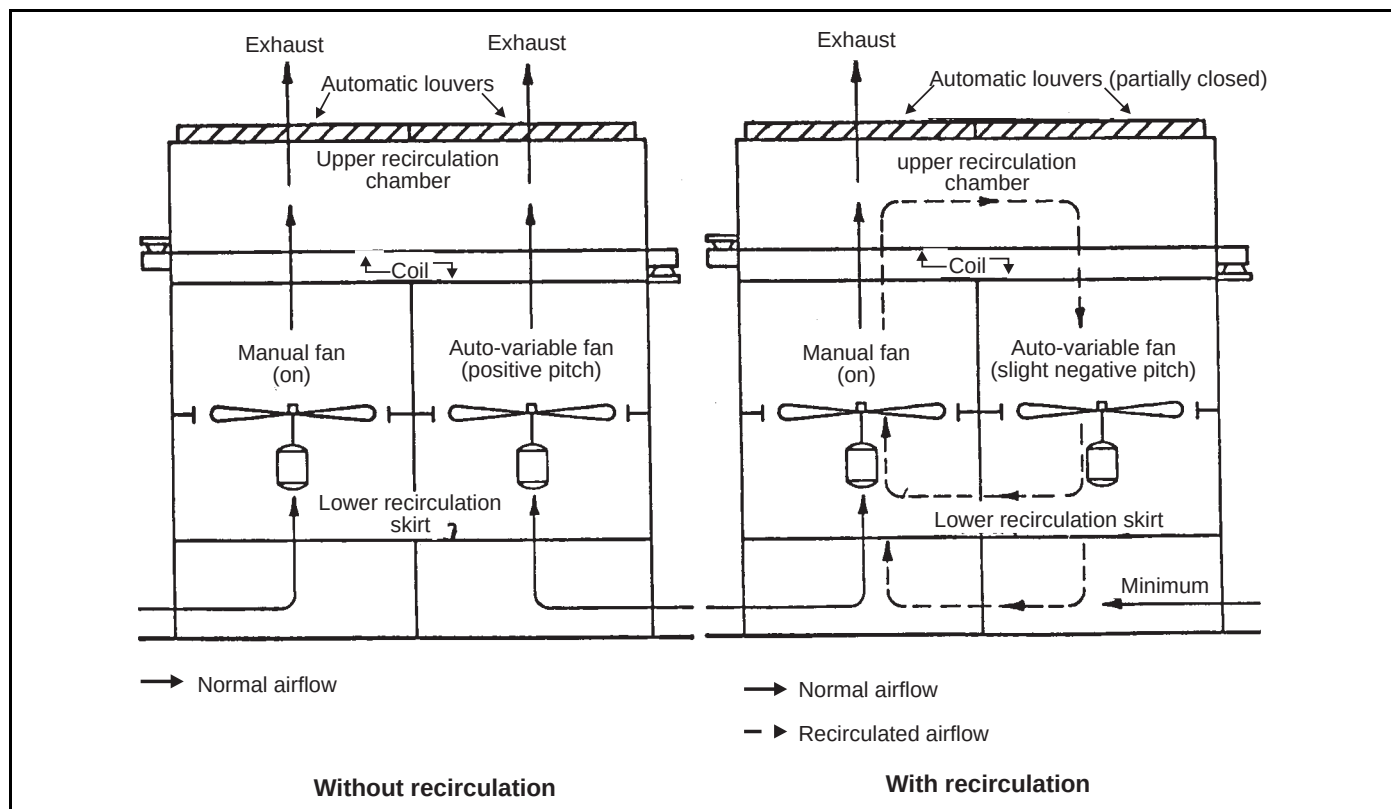
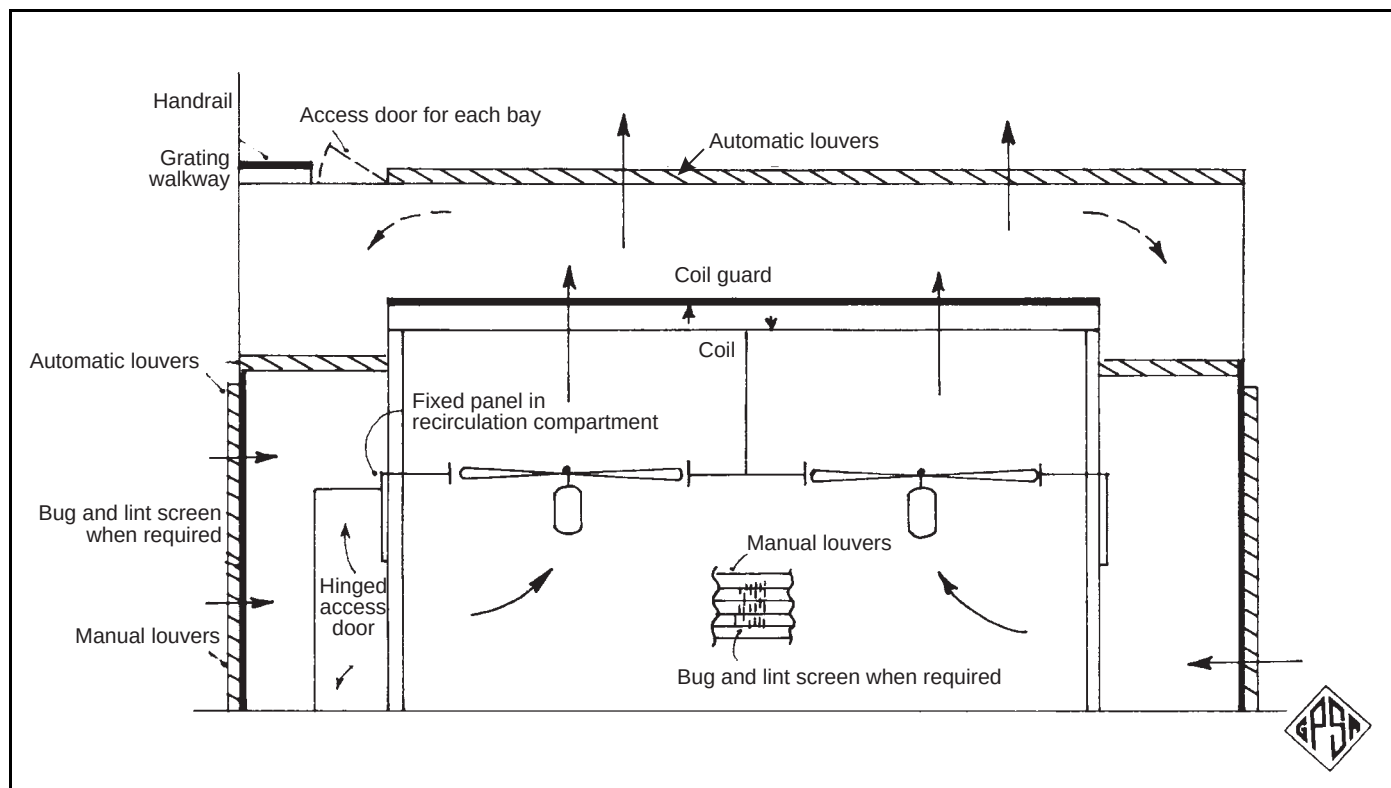


FIG. 10-7
External Recirculation Design



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

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 es-

timating 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:

$$Q = UA \text{ CMTD} \quad \text{Eq 10-1}$$

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:

$$\frac{1}{U_x} = \left(\frac{1}{h_t} \right) \left(\frac{A_x}{A_i} \right) + r_{dt} \left(\frac{A_x}{A_i} \right) + r_{mx} + \frac{1}{h_a} \quad \text{Eq 10-2}$$

The basic equation will then yield a heat transfer area in extended surface, A_x , and becomes:

$$Q = (U_x) (A_x) \text{ CMTD}$$

Either method is valid and each is used extensively by thermal design engineers. Fig. 10-10 gives typical overall heat

FIG. 10-8

MTD Correction Factors (1 Pass – Cross Flow, Both Fluids Unmixed)

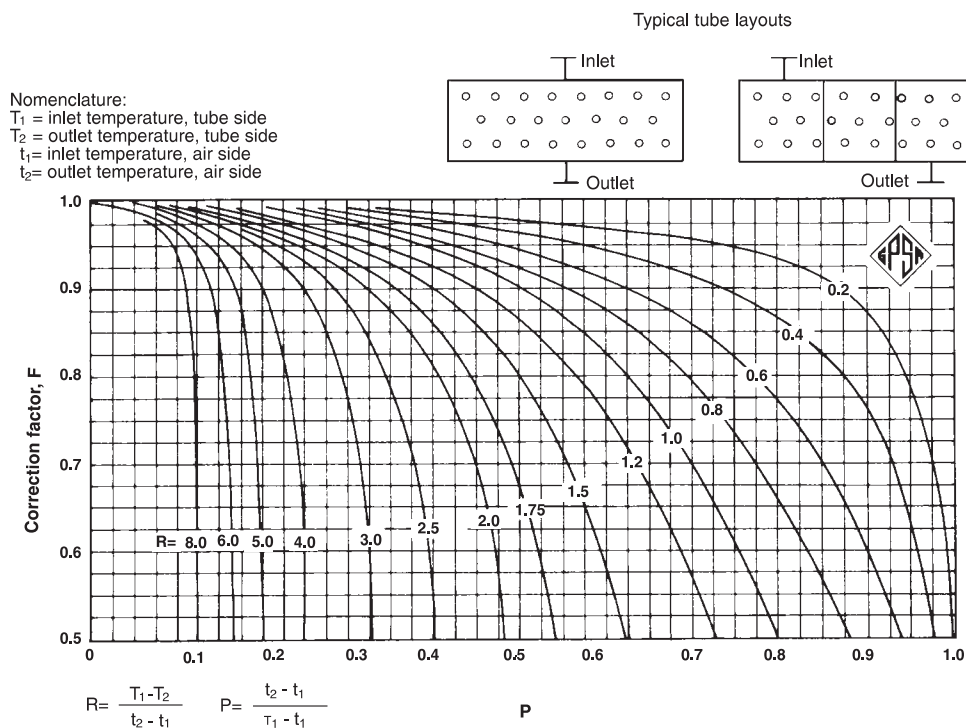


FIG. 10-9

MTD Correction Factors (2 Pass – Cross Flow, Both Fluids Unmixed)

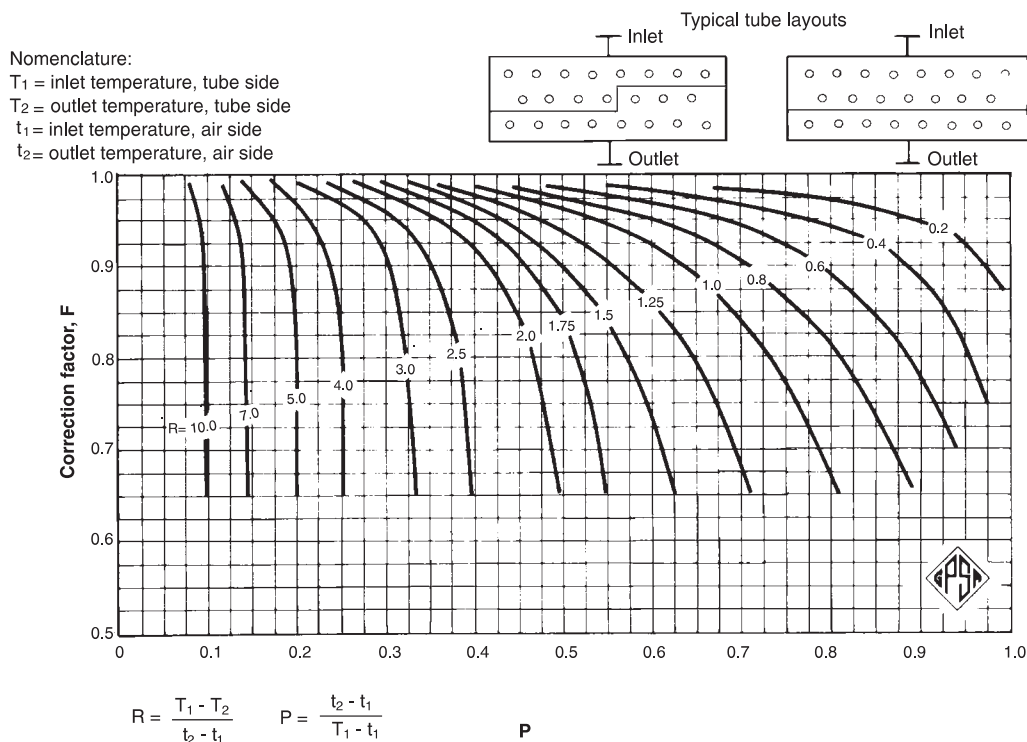


FIG. 10-10

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

$$C_p = 0.55 \text{ Btu/(lb} \cdot ^\circ\text{F)}$$

$$\mu = 0.51 \text{ cp}$$

$$k = 0.0766 \text{ Btu/[(hr} \cdot \text{sq ft} \cdot ^\circ\text{F)/ft]}$$

(From this Data Book Section 23)

Heat load: $Q = 15,000,000 \text{ Btu/hr}$

Flow quantity: $W_t = 273,000 \text{ lb/hr}$

Temperature in: $T_1 = 250^\circ\text{F}$

Temperature out: $T_2 = 150^\circ\text{F}$

Fouling factor $r_{dt} = 0.001 \text{ (hr} \cdot \text{sq ft} \cdot ^\circ\text{F)/Btu}$

Allowable pressure drop: $\Delta P_t = 5 \text{ psi}$

Required data for air

Ambient temperature: $t_1 = 100^\circ\text{F}$

Elevation: Sea level (see Fig. 10-16 for altitude correction)

$$C_{\text{Pair}} = 0.24 \text{ Btu/(lb} \cdot ^\circ\text{F)}$$

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. $U_x = 4.2$

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

$$\Delta t_a = \left(\frac{U_x + 1}{10} \right) \left(\frac{T_1 + T_2}{2} - t_1 \right)$$

$$\Delta t_a = \left(\frac{4.2 + 1.0}{10} \right) \left(\frac{250 + 150}{2} - 100 \right) = 52^\circ\text{F}$$

3. Calculate CMTD

$$\begin{array}{lll} \text{Hydcarbon} & 250 & \rightarrow 150 \\ \text{Air} & \frac{152}{98} & \leftarrow \frac{100}{50} \end{array}$$

$$\text{LMTD} = 71.3^\circ\text{F} \text{ (see Fig. 9-3)}$$

$$\text{CMTD} = (71.3)(1.00) = 71.3^\circ\text{F}$$

(3 tube passes assumed)

4. Calculate required surface

$$A_x = \frac{Q}{(U_x)(\text{CMTD})}$$

$$A_x = \frac{15,000,000}{(4.2)(71.3)} = 50,090 \text{ sq ft}$$

5. Calculate face area using APSF factor from Fig. 10-11

$$F_a = \frac{A_x}{\text{APSF}}$$

$$F_a = \frac{50,090}{107.2} = 467 \text{ sq ft (4 rows assumed)}$$

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

$$N_t = \frac{A_x}{(\text{APF})(L)}$$

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

$$A_t = 0.5945 \text{ sq in.}$$

$$G_t = \frac{(144)(W_t)(N_p)}{(3600)(N_t)(A_t)}$$

$$G_t = \frac{(0.04)(273,000)(3)}{(299)(0.5945)} = 184 \text{ lb/(ft}^2 \cdot \text{sec)}$$

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

$$\Delta P_t = \frac{f_{YL} N_p}{\phi} + B N_p$$

$$\Delta P_t = \frac{(0.0024)(14.5)(30)(3)}{0.96} + (0.25)(3) = 4.0 \text{ psi}$$

(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

$$k \left(\frac{C_p \mu}{k} \right)^{1/3} \text{ from Fig. 10-12}$$

$$h_t = \frac{Jk \left(\frac{C_p \mu}{k} \right)^{1/3} \phi}{D_i} = \frac{(1900)(0.12)(0.96)}{0.87} = 252$$

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

$$G_a = \frac{W_a}{F_a} = \text{lb/(hr} \cdot \text{sq ft of face area)}$$

$$G_a = \frac{1,200,000}{465} = 2,581$$

14. Read air-side film coefficient from Fig. 10-17

$$h_a = 8.5$$

15. Calculate overall transfer coefficient

FIG. 10-12
Physical Property Factor for Hydrocarbon Liquids

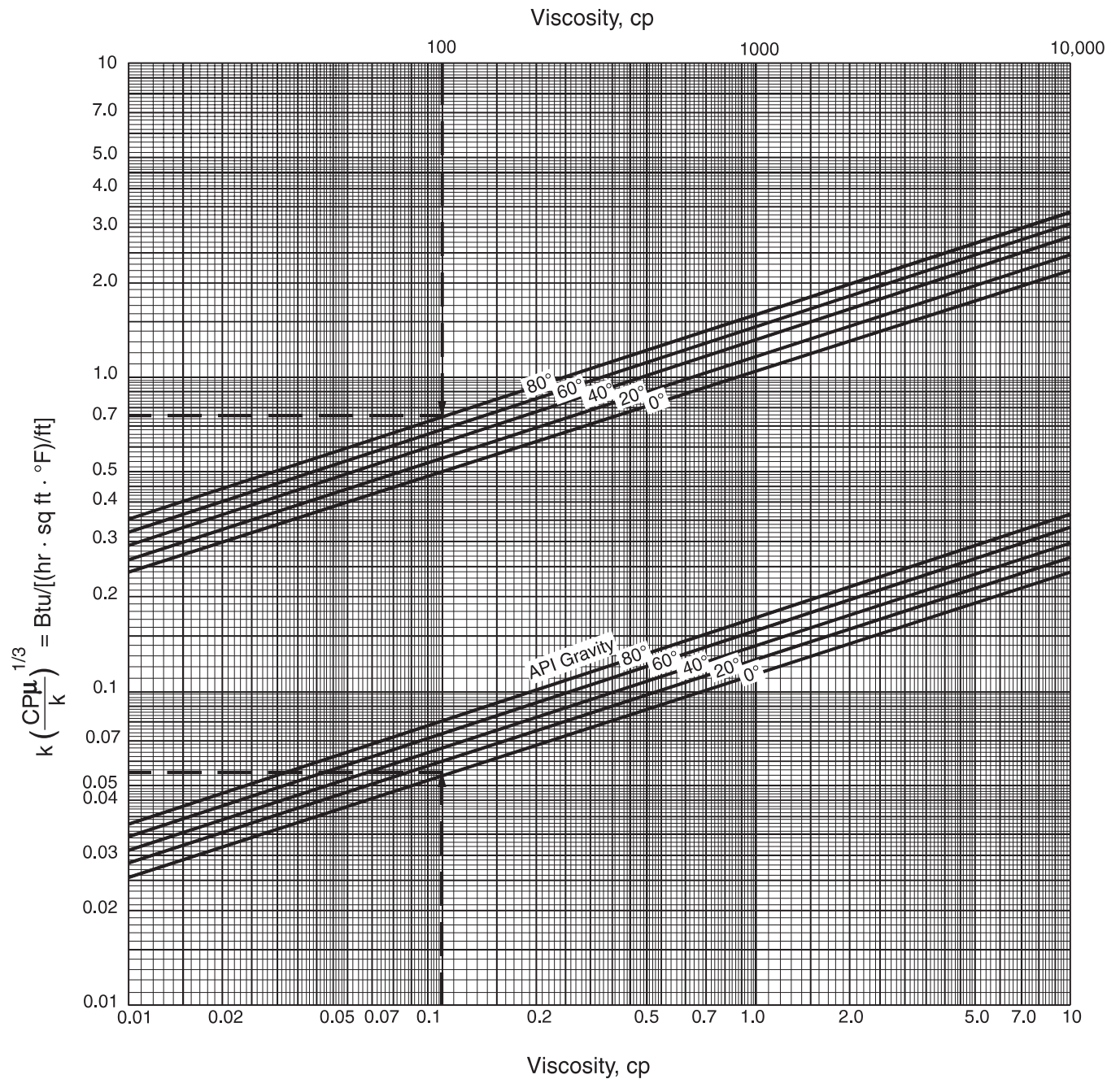


FIG. 10-13

J Factor Correlation to Calculate Inside Film Coefficient, h_i

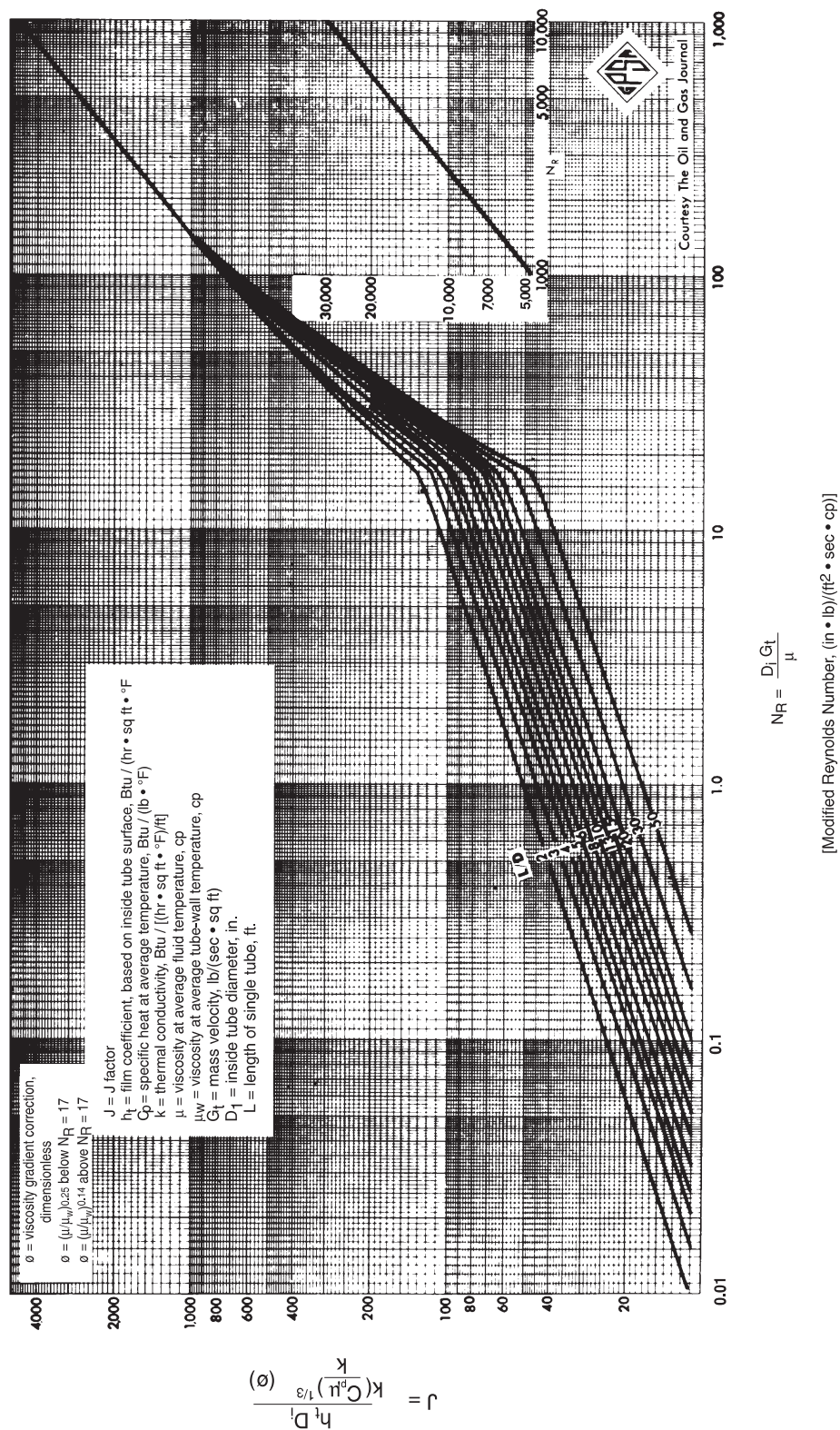


FIG. 10-14
Pressure Drop for Fluids Flowing Inside Tubes



$$\frac{A_x}{A_i} = \frac{(AR) (D_o)}{D_i}$$

$$\frac{A_x}{A_i} = \frac{(21.4) (1.0)}{0.87} = 24.6$$

$$\frac{1}{U_x} = \left(\frac{1}{h_t} \right) \left(\frac{A_x}{A_i} \right) + r_{dt} \left(\frac{A_x}{A_i} \right) + r_{mx} + \frac{1}{h_a}$$

$$\frac{1}{U_x} = \left(\frac{1}{252} \right) (24.6) + (0.001) (24.6) + \frac{1}{8.5}$$

$$U_x = 4.17$$

(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.

$$\text{Fan area/fan} = \text{FAPF} = \frac{(0.40) (F_a)}{(N_t)}$$

$$\text{FAPF} = \frac{(0.40) (465)}{2} = 93 \text{ ft}^2 \text{ (2 fans assumed)}$$

$$17. \text{ Fan diameter} = [4 (\text{FAPF})/\pi]^{0.5} = [4 (93)/3.1416]^{0.5} = 11 \text{ ft (rounded up)}$$

18. Calculate air static pressure drop using F_p from Fig. 10-18 and D_R at avg air temp from Fig. 10-16.

$$T_{a, \text{ avg}} = \frac{100^\circ\text{F} + 152^\circ\text{F}}{2} = 126^\circ\text{F}$$

$$\Delta P_a = \frac{(F_p) (N)}{(D_R)}$$

$$\Delta P_a = \frac{(0.10) (4)}{0.90} = 0.44 \text{ inches of water}$$

19. Calculate actual air volume using D_R of air at fan inlet.

$$t_1 = 100^\circ\text{F}$$

$$\text{ACFM} = \frac{W_a}{(D_R) (60) (0.0749)}$$

$$\text{ACFM} = \frac{1,200,000}{(0.94) (60) (0.0749)} = 284,000 \text{ Total}$$

$$\text{or } 142,000 / \text{ Fan}$$

20. Approximate fan total pressure using D_R of air at fan and fan area.

$$\text{PF} = \Delta P_a + \left[\frac{\text{ACFM}}{4005 \left(\frac{\pi D^2}{4} \right)} \right]^2 (D_R)$$

$$\text{Where: } 4005 = \sqrt{\frac{2 g \rho_w (3600)}{\rho_a \cdot 12}} \text{ at } 70^\circ\text{F}$$

$$\text{PF} = 0.44 + \left(\frac{142,000}{(4005) (0.785) (11^2)} \right)^2 (0.94) = 0.57 \text{ inches of water}$$

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:

$$(15.5 \text{ ft}) (30 \text{ ft}) = 465 \text{ sq ft}$$

$$(465 \text{ sq ft}) (\text{APSF}) = \text{extended surface area}$$

$$(465) (107.2) = 49,848 \text{ sq ft}$$

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.

FIG. 10-15
Friction Factor for Fluids Flowing Inside Tubes

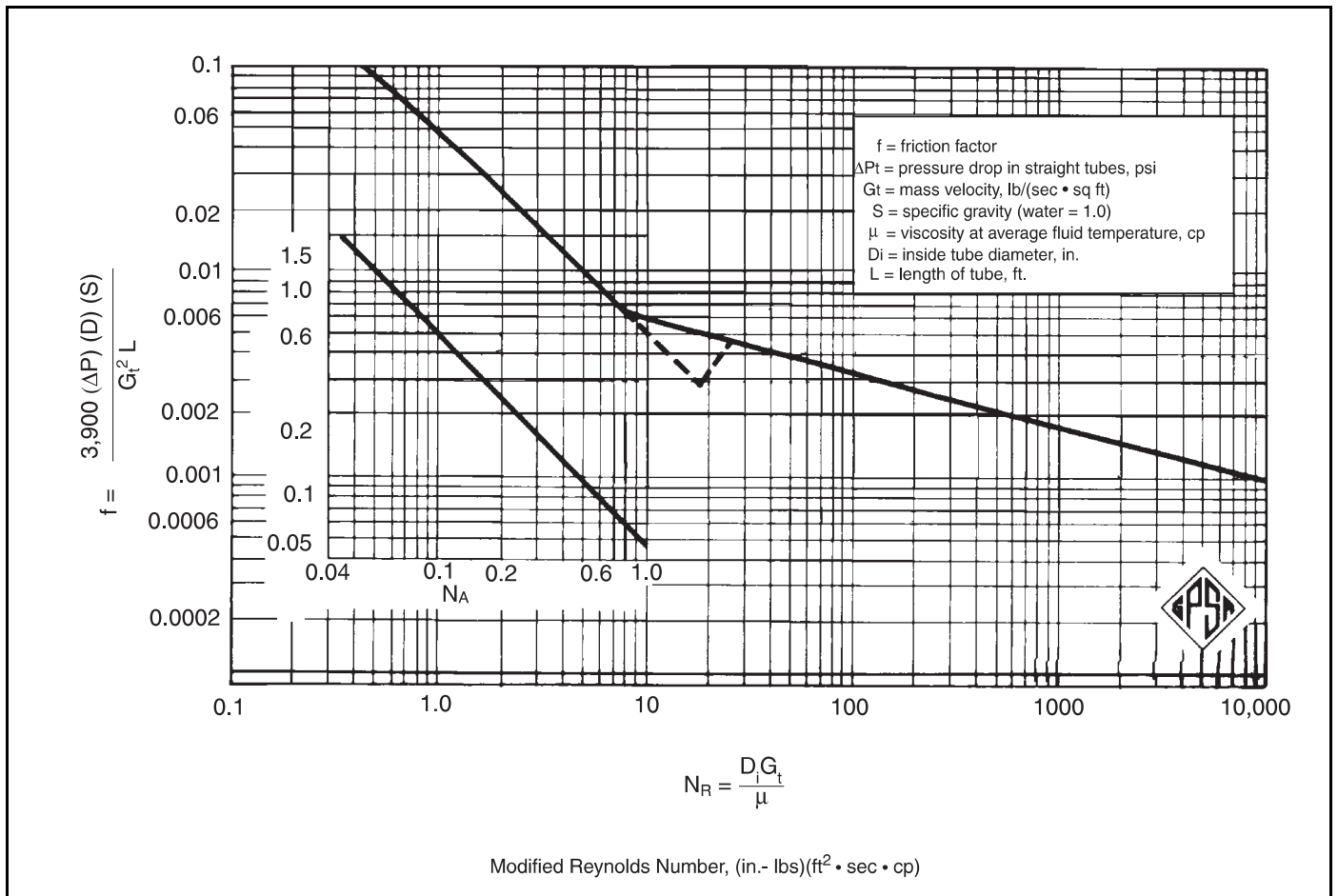


FIG. 10-16
Air-Density Ratio Chart

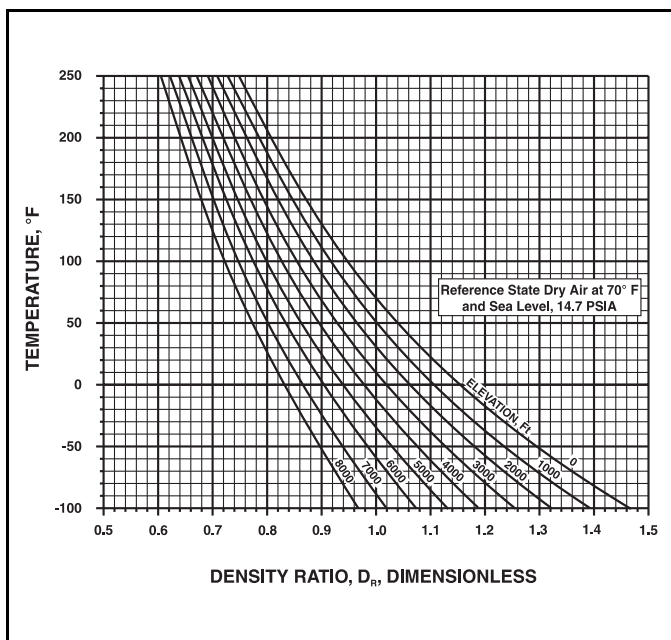
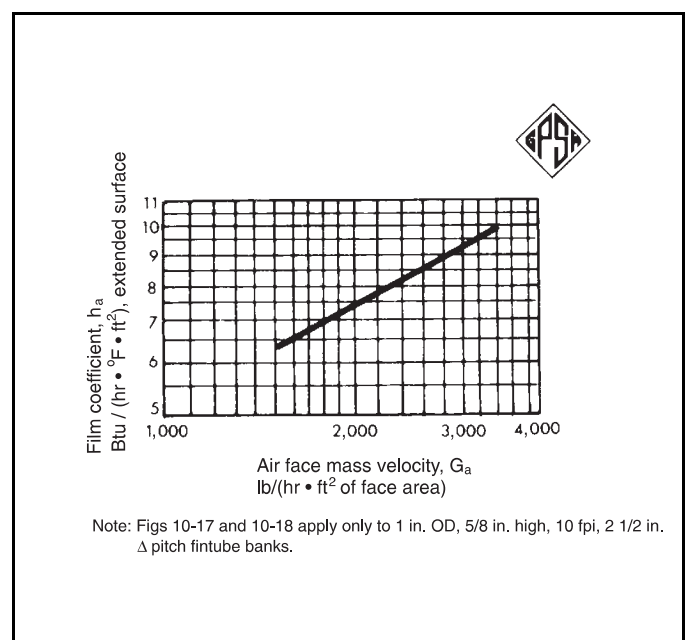


FIG. 10-17
Air Film Coefficient



- 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

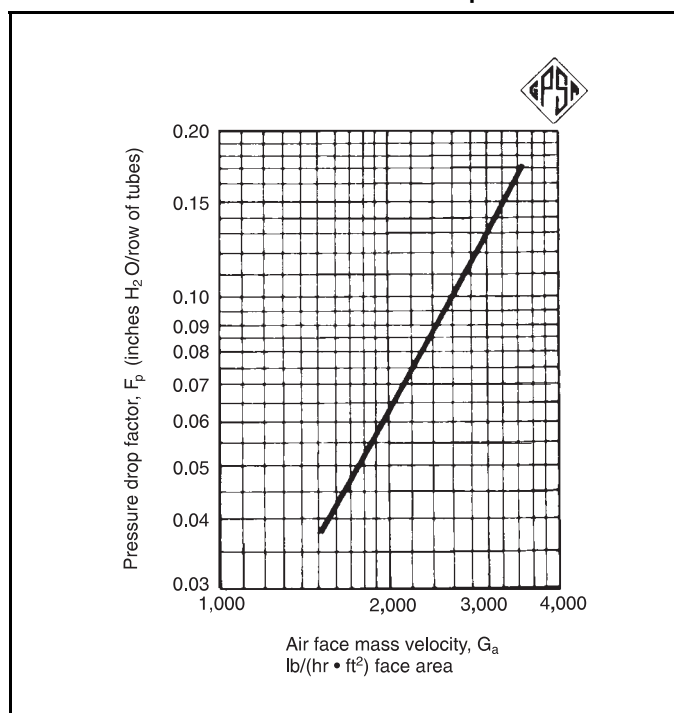


FIG. 10-19

Correction Factor for Fluid Viscosity Within the Tubes

Correction factor * when $\phi = \left(\frac{\mu}{\mu_w}\right)^{0.14}$ (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 $N_r < 17$, $\phi = \left(\frac{\mu}{\mu_w}\right)^{0.25}$ 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 ($20 \mu\text{m}^2$). 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.

$$\text{PWL} = 10 \log \left(\frac{W}{10^{-12} \text{ watts}} \right) \quad \text{Eq 10-3}$$

Sound power level cannot be measured directly but must be calculated from sound pressure levels (SPL) dB. In metric terms, this is known as L_w .

Sound pressure level, known as SPL, or L_p 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-

FIG. 10-20
Altitude and Atmospheric Pressures⁸

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:

$$PWL = 56 + 30 \log \left(\frac{FPM}{1000} \right) + \log HP \quad \text{Eq 10-4}$$

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

$$PWL = 56 + 30 \log 10.4 + \log 25.1 \\ = 100.5 \text{ dB(A)}$$

- When considering multiple noise sources (fans) use the relation:

$$PWL_N = PWL + 10 \log N \quad \text{Eq 10-5}$$

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:

$$SPL \text{ (at distance R)} = PWL - 20 \log R \quad \text{Eq 10-6}$$

Where R is in feet from the center of the source. Measure R as a "line of sight" distance.

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 $PWL = 100.5 \text{ dB(A)}$.

$SPL = 100.5 - 20 \log 62 = 64.7 \text{ dB(A)}$

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

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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 counter-current 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.

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.

FIG. 11-2
Psychrometric Chart

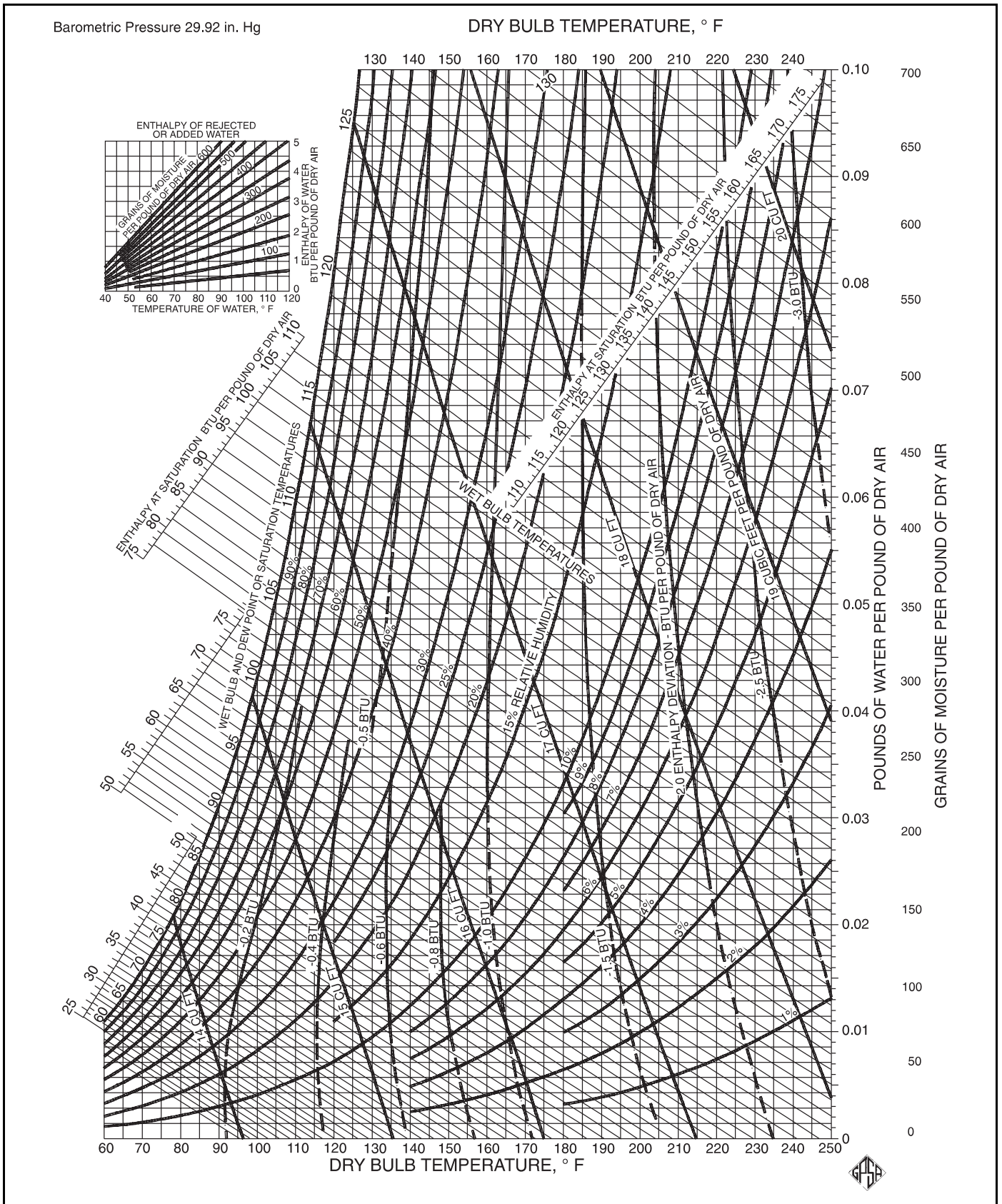


FIG. 11-3a
Dry Bulb/Wet Bulb Temperature Data ²

U.S. State and Stations	Design Dry-Bulb °F 5%	Design Wet-Bulb °F 5%	U.S. State and Stations	Design Dry-Bulb °F 5%	Design Wet-Bulb °F 5%	U.S. State and Stations	Design Dry-Bulb °F 5%	Design Wet-Bulb °F 5%	U.S. State and Stations	Design Dry-Bulb °F 5%	Design Wet-Bulb °F 5%
ALABAMA			Ontario	96	71	Miami Beach	88	78	Springfield	89	76
Alexander City	91	78	Oxnard	77	67	Ocala	92	78	Waukegan	87	75
Anniston	92	78	Palmdale	98	66	Orlando	91	78	INDIANA		
Auburn	91	78	Palm Springs	108	73	Panama City,	89	79	Anderson	89	76
Birmingham	92	76	Pasadena	92	70	Tyndall AFB			Bedford	89	76
Decatur	91	76	Petaluma	87	68	Pensacola	91	79	Bloomington	89	76
Dothan	91	78	Pomona	95	71	St. Augustine	87	79	Columbus,	90	76
Florence	92	76	Redding	100	68	St. Petersburg	90	78	Bakalar AFB		
Gadsden	92	76	Redlands	96	71	Sanford	91	78	Crawfordsville	88	76
Huntsville	91	76	Richmond	75	63	Sarasota	90	78	Evansville	91	77
Mobile AP	91	78	Riverside-March AFB	95	70	Tallahassee	90	78	Fort Wayne	87	74
Mobile CO	91	78	Sacramento	94	70	Tampa	90	78	Goshen	86	74
Montgomery	93	78	Salinas	67	59	West Palm Beach	90	79	Hobart	85	74
Selma-Craig	93	79	San Bernardino,	96	71	GEORGIA			Huntington	87	74
Talladega	92	78	Norton AFB			Albany, Turner AFB	93	78	Indianapolis	87	75
Tuscaloosa	94	77	San Diego	78	68	Americus	92	77	Jeffersonville	90	76
ALASKA			San Fernando	88	69	Athens	90	76	Kokomo	88	74
Anchorage	66	57	San Francisco AP	73	62	Atlanta	90	75	Lafayette	88	75
Barrow	49	47	San Francisco CO	69	61	Augusta	93	78	La Porte	87	75
Fairbanks	75	60	San Jose	77	65	Brunswick	87	79	Marion	88	74
Juneau	67	58	San Luis Obispo	84	70	Columbus,	91	77	Muncie	87	75
Kodiak	62	56	Santa Ana	82	69	Lawson AFB			Peru, Bunker	86	74
Nome	59	55	Santa Barbara	75	66	Dalton	91	77	Hill AFB		
ARIZONA			Santa Cruz	68	61	Dublin	91	77	Richmond	87	75
Douglas	93	68	Santa Maria	73	63	Gainesville	89	75	Shelbyville	88	75
Flagstaff	80	59	Santa Monica	77	68	Griffin	88	76	South Bend	86	74
Fort Huachuca	90	67	Santa Paula	84	68	La Grange	89	76	Terre Haute	89	76
Kingman	97	69	Santa Rosa	91	67	Macon	91	77	Valparaiso	87	75
Nogales	94	69	Stockton	94	68	Marietta,	90	76	Vincennes	90	76
Phoenix	105	75	Ukiah	91	67	Dobbins AFB			IOWA		
Prescott	92	64	Visalia	97	70	Moultrie	92	78	Ames	87	75
Tucson	100	71	Yreka	89	64	Rome	91	77	Burlington	88	75
Winslow	93	64	Yuba City	99	68	Savannah-Travis	91	78	Cedar Rapids	86	75
Yuma	107	77	COLORADO			Valdosta-Moody AFB	92	78	Clinton	87	75
ARKANSAS			Alamosa	80	60	Waycross	91	78	Council Bluffs	88	75
Blytheville	91	78	Boulder	89	62	HAWAII			Des Moines	88	75
Camden	94	78	Colorado Springs	86	61	Hilo	82	74	Dubuque	86	74
El Dorado	94	78	Denver	89	62	Honolulu	85	74	Fort Dodge	86	74
Fayetteville	92	75	Durango	85	62	Kaneohe Bay	83	75	Iowa City	87	76
Fort Smith	95	78	Fort Collins	89	62	Wahiawa	84	73	Keokuk	89	76
Hot Springs	94	78	Grand Junction	92	62	IDAHO			Marshalltown	88	75
Jonesboro	91	78	Greeley	92	63	Boise	91	65	Mason City	85	74
Little Rock	94	78	La Junta	95	69	Burley	92	61	Newton	88	75
Pine Bluff	95	80	Leadville	78	54	Coeur d'Alene	83	61	Ottumwa	88	75
Texarkana	93	78	Pueblo	92	65	Idaho Falls	84	61	Sioux City	89	75
CALIFORNIA			Sterling	90	65	Lewiston	90	64	Waterloo	86	75
Bakersfield	98	70	Trinidad	89	64	Moscow	84	62	KANSAS		
Barstow	102	70	CONNECTICUT			Mountain Home AFB	94	63	Atchison	91	77
Blythe	108	74	Bridgeport	81	73	Pocatello	89	61	Chanute	94	76
Burbank	88	69	Hartford,	85	74	Twin Falls	92	61	Dodge City	95	71
Chico	98	68	Brainard Field			ILLINOIS			El Dorado	96	75
Concord	94	68	New Haven	82	74	Aurora	88	76	Emporia	94	76
Covina	92	70	New London	83	74	Belleville, Scott AFB	89	76	Garden City	94	71
Crescent City	63	59	Norwalk	81	73	Bloomington	88	75	Goodland	93	68
Downey	86	70	Norwich	83	74	Carbondale	90	77	Great Bend	95	75
El Cajon	78	68	Waterbury	82	72	Champaign/Urbana	90	75	Hutchinson	97	74
El Centro	108	78	Windsor Locks,	85	73	Chicago, Midway	88	74	Liberal	94	71
Escondido	82	69	Bradley Field			Chicago, O'Hare	86	74	Manhattan, Fort Riley	92	76
Eureka/Arcata	63	59	DELAWARE			Chicago	88	75	Parsons	94	76
Fairfield-Travis AFB	91	67	Dover AFB	87	76	Danville	88	75	Russell	95	75
Fresno	97	70	Wilmington	87	75	Decatur	88	75	Salina	97	75
Hamilton	80	67	DISTRICT OF			Dixon	88	75	Topeka	93	76
Laguna Beach	77	68	COLUMBIA			Elgin	86	75	Wichita	96	75
Livermore	93	68	Andrews AFB	87	75	Freeport	87	74	KENTUCKY		
Lompoc,	67	60	Washington National	89	76	Galesburg	88	75	Ashland	89	75
Vandenberg AFB			FLORIDA			Greenville	89	76	Bowling Green	89	76
Long Beach	77	68	Belle Glade	89	78	Joliet	88	75	Corbin	89	75
Los Angeles AP	77	68	Cape Kennedy	87	79	Kankakee	88	75	Covington	88	74
Los Angeles CO	86	70	Daytona Beach	88	78	La Salle/Peru	88	75	Hopkinsville,	89	76
Merced-Castle AFB	96	70	Fort Lauderdale	90	79	Macomb	89	76	Campbell AFB		
Modesto	95	69	Fort Myers	91	79	Moline	88	75	Lexington	88	75
Monterey	68	61	Fort Pierce	89	79	Mt. Vernon	89	76	Louisville	90	76
Napa	92	68	Gainesville	92	78	Peoria	87	75	Madisonville	90	77
Needles	108	74	Jacksonville	92	78	Quincy	90	77	Owensboro	91	77
Oakland	75	63	Key West	89	79	Rantoul,	89	75	Paducah	92	77
Oceanside	77	68	Lakeland	89	78	Chanute AFB			LOUISIANA		
			Miami	89	78	Rockford	87	74	Alexandria	92	78

FIG. 11-3a (Cont'd)
Dry Bulb/Wet Bulb Temperature Data²

U.S. State and Stations	Design Dry-Bulb °F 5%	Design Wet-Bulb °F 5%	U.S. State and Stations	Design Dry-Bulb °F 5%	Design Wet-Bulb °F 5%	U.S. State and Stations	Design Dry-Bulb °F 5%	Design Wet-Bulb °F 5%	U.S. State and Stations	Design Dry-Bulb °F 5%	Design Wet-Bulb °F 5%
Baton Rouge	92	79	Virginia	80	68	Tonopah	90	61	Watertown	81	72
Bogalusa	92	79	Willmar	85	72	Winnemucca	92	61	NORTH CAROLINA		
Houma	92	79	Winona	85	74	NEW HAMPSHIRE			Asheville	85	72
Lafayette	92	79	MISSISSIPPI			Berlin	81	70	Charlotte	91	76
Lake Charles	92	79	Biloxi, Keesler AFB	90	80	Claremont	83	71	Durham	90	76
Minden	94	78	Clarksdale	92	78	Concord	84	71	Elizabeth City	89	78
Monroe	94	78	Columbus AFB	91	78	Keene	83	71	Fayetteville	90	77
Natchitoches	93	78	Greenville AFB	91	78	Laconia	83	71	Pope AFB		
New Orleans	90	79	Greenwood	91	78	Manchester	85	72	Goldsboro, Seymour-	89	77
Shreveport	94	78	Hattiesburg	92	79	Grenier AFB			Johnson AFB		
MAINE			Jackson	93	78	Portsmouth,	83	72	Greensboro	89	75
Augusta	82	70	Laurel	92	79	Pease AFB			Greenville	89	77
Bangor, Dow AFB	80	69	McComb	92	78	NEW JERSEY			Henderson	90	77
Caribou	78	67	Meridian	93	78	Atlantic City	86	75	Hickory	88	73
Lewiston	82	70	Natchez	92	79	Long Branch	87	75	Jacksonville	88	78
Millinocket	80	68	Tupelo	92	78	Newark	88	75	Lumberton	90	77
Portland	81	70	Vicksburg	93	79	New Brunswick	86	75	New Bern	88	78
Waterville	81	70	MISSOURI			Paterson	88	75	Raleigh/Durham	90	76
MARYLAND			Cape Girardeau	92	77	Phillipsburg	86	74	Rocky Mount	89	77
Baltimore AP	89	76	Columbia	91	76	Trenton	85	75	Wilmington	89	79
Baltimore CO	87	76	Farmington	90	75	Vineland	86	75	Winston-Salem	89	74
Cumberland	87	75	Hannibal	90	77	NEW MEXICO			NORTH DAKOTA		
Frederick	88	76	Jefferson City	92	76	Alamogordo,	94	67	Bismark	88	70
Hagerstown	89	75	Joplin	94	76	Holloman AFB			Devil's Lake	85	69
Salisbury	88	76	Kansas City	93	76	Albuquerque	92	64	Dickinson	87	68
MASSACHUSETTS			Kirkville	90	76	Artesia	97	70	Fargo	85	72
Boston	85	72	Mexico	91	76	Carlsbad	97	70	Grand Forks	84	70
Clinton	84	72	Moberly	91	76	Clovis	91	67	Jamestown	87	71
Fall River	81	72	Poplar Bluff	92	78	Farmington	91	64	Minot	86	68
Framingham	83	71	Rolla	89	76	Gallup	86	61	Williston	85	68
Gloucester	83	72	St. Joseph	91	77	Grants	85	61	OHIO		
Greenfield	82	71	St. Louis AP	91	76	Hobbs	97	69	Akron-Canton	84	72
Lawrence	84	73	St. Louis CO	91	76	Las Cruces	94	67	Ashtabula	83	72
Lowell	85	73	Sedalia,	90	76	Los Alamos	85	60	Athens	90	74
New Bedford	80	72	Whiteman AFB			Raton	87	63	Bowling Green	86	73
Pittsfield	81	70	Sikeston	92	77	Roswell, Walker AFB	96	69	Cambridge	87	75
Springfield,	84	72	Springfield	91	75	Santa Fe	86	61	Chillicothe	90	74
Westover AFB			MONTANA			Silver City	91	63	Cincinnati	88	74
Taunton	83	73	Billings	88	64	Socorro	93	65	Cleveland	86	73
Worcester	81	70	Bozeman	84	60	Tucumcari	95	68	Columbus	87	74
MICHIGAN			Butte	80	57	NEW YORK			Dayton	86	73
Adrian	85	73	Cut Bank	82	61	Albany AP	85	72	Defiance	88	74
Alpena	83	70	Glasgow	85	64	Albany CO	85	72	Findlay	87	74
Battle Creek	85	73	Glendive	89	65	Auburn	84	72	Fremont	85	73
Benton Harbor	85	72	Great Falls	85	60	Batavia	84	72	Hamilton	87	73
Detroit	86	73	Havre	87	65	Binghamton	81	70	Lancaster	88	74
Escanaba	80	69	Helena	85	61	Buffalo	83	72	Lima	88	74
Flint	85	72	Kalispell	84	62	Cortland	82	71	Mansfield	85	73
Grand Rapids	85	72	Lewiston	83	62	Dunkirk	83	72	Marion	88	74
Holland	83	72	Livingston	84	60	Elmira	83	71	Middletown	87	73
Jackson	85	73	Miles City	92	67	Geneva	84	72	Newark	89	74
Kalamazoo	85	73	Missoula	85	62	Glen Falls	82	71	Norwalk	85	73
Lansing	84	72	NEBRASKA			Gloversville	83	72	Portsmouth	89	75
Marquette	77	68	Beatrice	92	76	Hornell	82	72	Sandusky	88	73
Mt. Pleasant	84	72	Chadron	91	68	Ithaca	82	71	Springfield	87	74
Muskegon	82	72	Columbus	92	75	Jamestown	83	71	Steubenville	84	72
Pontiac	85	73	Fremont	92	76	Kingston	85	73	Toledo	85	73
Port Huron	83	73	Grand Island	91	73	Lockport	84	73	Warren	85	71
Saginaw	84	72	Hastings	91	73	Massena	80	70	Wooster	84	72
Sault St. Marie	77	68	Kearney	90	72	Newburg-	85	73	Youngstown	84	71
Traverse City	83	71	Lincoln	92	76	Stewart AFB			Zanesville	87	75
Ypsilanti	86	72	McCook	91	71	NYC-Central Park	87	74	OKLAHOMA		
MINNESOTA			Norfolk	90	75	NYC-Kennedy AP	84	74	Ada	95	75
Albert Lea	84	73	North Platte	90	71	NYC-LaGuardia AP	87	74	Altus AFB	98	75
Alexandria	85	72	Omaha	88	75	Niagara Falls	84	73	Ardmore	95	76
Bemidji	81	69	Scottsbluff	90	67	Olean	81	71	Bartlesville	95	76
Brainerd	84	71	Sidney	90	67	Oneonta	80	70	Chickasha	95	76
Duluth	79	68	NEVADA			Oswego	80	72	Enid-Vance AFB	97	76
Fairbault	85	73	Carson City	89	60	Plattsburg AFB	80	70	Lawton	96	76
Fergus Falls	85	72	Elko	90	60	Poughkeepsie	86	74	McAlester	93	75
International Falls	80	68	Ely	85	58	Rochester	85	72	Muskogee	95	77
Mankato	85	73	Las Vegas	104	69	Rome-Griffiss AFB	83	71	Norman	94	75
Minneapolis/St. Paul	86	73	Lovelock	93	64	Schenectady	84	72	Oklahoma City	95	76
Rochester	84	73	Reno AP	90	61	Suffolk County AFB	80	73	Ponca City	94	76
St. Cloud	85	72	Reno CO	91	61	Syracuse	84	72	Seminole	94	75
						Utica	82	71			

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%	U.S. State and Stations	Design Dry-Bulb °F 5%	Design Wet-Bulb °F 5%	U.S. State and Stations	Design Dry-Bulb °F 5%	Design Wet-Bulb °F 5%	U.S. State and Stations	Design Dry-Bulb °F 5%	Design Wet-Bulb °F 5%
Stillwater	93	75	Rock Hill	92	76	Laredo	99	77	Kennewick	92	67
Tulsa	95	77	Spartanburg	89	75	Longview	95	78	Longview	81	66
Woodward	94	75	Sumter-Shaw AFB	90	77	Lubbock	94	71	Moses Lake,	90	64
OREGON			SOUTH DAKOTA			Lufkin	94	78	Larson AFB		
Albany	86	66	Aberdeen	88	73	McAllen	94	79	Olympia	79	64
Astoria	68	62	Brookings	89	73	Midland	96	71	Port Angeles	67	61
Baker	86	61	Huron	90	73	Mineral Wells	97	76	Seattle-Boeing Fld	77	65
Bend	84	60	Mitchel	90	73	Palestine	96	78	Seattle CO	78	65
Corvallis	86	66	Pierre	92	72	Pampa	94	70	Seattle-Tacoma AP	76	63
Eugene	86	66	Rapid City	89	67	Pecos	96	71	Spokane	87	62
Grants Pass	93	68	Sioux Falls	88	73	Plainview	94	70	Tacoma-	79	64
Klamath Falls	84	60	Watertown	88	73	Port Arthur	91	80	McChord AFB		
Medford	91	67	Yankton	88	74	San Angelo,	97	73	Walla Walla	90	66
Pendleton	90	63	TENNESSEE			Goodfellow AFB			Wenatchee	92	65
Portland AP	81	66	Athens	90	75	San Antonio	96	76	Yakima	89	65
Portland CO	82	66	Bristol-Tri City	87	73	Sherman Perrin AFB	95	76	WEST VIRGINIA		
Roseburg	87	66	Chattanooga	91	76	Snyder	96	72	Beckley	79	70
Salem	84	66	Clarksville	90	76	Temple	97	77	Bluefield	79	70
The Dalles	85	67	Columbia	91	76	Tyler	95	78	Charleston	87	74
PENNSYLVANIA			Dyersburg	91	78	Vernon	97	75	Clarksburg	87	74
Allentown	86	73	Greenville	88	74	Victoria	94	79	Elkins	82	71
Altoona	84	72	Jackson	92	77	Waco	97	77	Huntington	89	75
Butler	85	73	Knoxville	90	75	Wichita Falls	98	75	Martinsburg	88	75
Chambersburg	87	75	Memphis	93	78	UTAH			Morgantown	85	74
Erie	83	72	Murfreesboro	91	76	Cedar City	89	62	Parkersburg	88	75
Harrisburg	88	75	Nashville	91	76	Logan	88	63	Wheeling	84	72
Johnstown	80	70	Tulahoma	91	75	Moab	96	63	WISCONSIN		
Lancaster	87	75	TEXAS			Ogden	88	64	Appleton	83	72
Meadville	83	71	Abilene	97	74	Price	89	62	Ashland	79	68
New Castle	86	73	Alice	95	79	Provo	94	64	Beloit	88	75
Philadelphia	87	75	Amarillo	93	70	Richfield	89	62	Eau Claire	86	73
Pittsburgh AP	84	72	Austin	97	77	St. George	99	67	Fond du Lac	84	72
Pittsburgh CO	86	72	Bay City	92	79	Salt Lake City	92	64	Green Bay	83	72
Reading	86	73	Beaumont	91	80	Vernal	86	62	La Crosse	85	74
Scranton/Wilkes-	84	72	Beeville	95	79	VERMONT			Madison	85	73
Barre			Big Springs	95	72	Barre	78	70	Manitowoc	83	72
State College	84	72	Brownsville	92	79	Burlington	82	71	Marinette	82	71
Sunbury	86	73	Brownwood	96	75	Rutland	81	71	Milwaukee	84	73
Uniontown	85	74	Bryan	94	78	VIRGINIA			Racine	85	74
Warren	83	72	Corpus Christi	92	79	Charlottesville	88	75	Sheboygan	83	74
West Chester	86	75	Corsicana	96	77	Danville	90	75	Stevens Point	86	73
Williamsport	86	73	Dallas	97	77	Fredericksburg	90	76	Waukesha	84	73
York	88	75	Del Rio,	97	76	Harrisonburg	88	73	Wausau	85	72
RHODE ISLAND			Laughlin AFB			Lynchburg	88	75	WYOMING		
Newport	82	73	Denton	97	76	Norfolk	89	77	Casper	87	60
Providence	83	73	Eagle Pass	98	77	Petersburg	90	77	Cheyenne	84	60
SOUTH CAROLINA			El Paso	96	68	Richmond	90	77	Cody	83	61
Anderson	90	75	Fort Worth	97	76	Roanoke	88	73	Evanston	82	57
Charleston AFB	89	79	Galveston	88	80	Staunton	88	73	Lander	85	61
Charleston	90	79	Greenville	97	76	Winchester	88	75	Laramie	79	59
Columbia	93	77	Harlingen	93	79	WASHINGTON			Newcastle	84	66
Florence	90	78	Houston AP	92	79	Aberdeen	73	62	Rawlins	81	60
Georgetown	88	79	Houston CO	93	79	Bellingham	74	63	Rock Springs	82	57
Greenville	89	75	Huntsville	96	77	Bremerton	75	63	Sheridan	88	63
Greenwood	91	76	Killeen-Gray AFB	95	75	Ellensburg	87	63	Torrington	88	63
Orangeburg	93	77	Lamesa	94	71	Everett-Paine AFB	73	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 Data²

Country and Station	Design Dry-Bulb °F 5%	Design Wet-Bulb °F 5%	Country and Station	Design Dry-Bulb °F 5%	Design Wet-Bulb °F 5%	Country and Station	Design Dry-Bulb °F 5%	Design Wet-Bulb °F 5%	Country and Station	Design Dry-Bulb °F 5%	Design Wet-Bulb °F 5%
ADEN			Portage la Prairie	83	71	Trois Rivières	82	71	GUATEMALA		
Aden	98	82	The Pas	79	68	Val d'Or	80	68	Guatemala City	81	67
AFGHANISTAN			Winnipeg	84	71	Valleyfield	84	72	GUYANA		
Kabul	93	64	CANADA (New Brunswick)			CANADA (Saskatchewan)			Georgetown	87	79
ALGERIA			Campbellton	79	68	Estevan	86	69	HAITI		
Algiers	89	75	Chatham	82	69	Moose Jaw	86	68	Port au Prince	93	80
ARGENTINA			Edmundston	80	69	North Battleford	82	66	HONDURAS		
Buenos Aires	86	75	Fredericton	82	70	Prince Albert	81	67	Tegucigalpa	85	71
Córdoba	93	74	Moncton	79	69	Regina	84	68	HONG KONG		
Tucuman	96	74	Saint John	75	66	Saskatoon	83	67	Hong Kong	90	80
AUSTRALIA			CANADA			Swift Current	87	67	HUNGARY		
Adelaide	91	68	(Newfoundland)			Yorkton	80	68	Budapest	84	70
Alice Springs	100	72	Corner Brook	71	65	CANADA (Yukon Territory)			ICELAND		
Brisbane	86	75	Gander	77	66	Whitehorse	74	58	Reykjavik	56	53
Darwin	91	81	Goose Bay	77	64	CEYLON			INDIA		
Melbourne	86	68	St. John's	73	66	Colombo	88	80	Ahmenabad	105	78
Perth	93	73	Stephenville	71	65	CHILE			Bangalore	93	74
Sydney	80	72	CANADA (Northwest Terr.)			Punta Arenas	64	54	Bombay	92	81
AUSTRIA			Fort Smith	78	65	Santiago	88	69	Calcutta	96	82
Vienna	83	67	Frobisher	59	51	Valparaiso	77	65	Madras	101	83
AZORES			Inuvik	75	61	CHINA			Nagpur	107	78
Lajes (Terceira)	77	71	Resolute	51	46	Chungking	95	79	New Delhi	105	82
BAHAMAS			Yellowknife	74	62	Shanghai	90	80	INDONESIA		
Nassau	88	79	CANADA (Nova Scotia)			COLOMBIA			Djakarta	88	78
BELGIUM			Amherst	79	68	Baranquilla	93	82	Kupang	92	80
Brussels	77	67	Halifax	74	66	Bogotá	69	58	Makassar	88	79
BELIZE			Kentville	80	69	Cali	79	68	Medan	90	79
Belize	89	81	New Glasgow	77	69	Medellin	84	72	Palembang	90	79
BERMUDA			Sydney	77	68	CONGO			Surabaya	89	79
Kindley AFB	85	78	Truro	78	70	Brazzaville	91	80	IRAN		
BOLIVIA			Yarmouth	70	65	Kinasha	90	80	Abadan	110	81
La Paz	68	56	CANADA (Ontario)			(Leopoldville)			Meshed	93	66
BRAZIL			Belleville	82	73	Stanleyville	90	80	Tehran	98	73
Belem	87	78	Chatham	85	74	CUBA			IRAQ		
Belo Horizonte	83	75	Cornwall	84	72	Guantanamo Bay	92	80	Baghdad	108	72
Brazilia	86	75	Hamilton	83	73	Havana	89	80	Mosul	110	72
Curitiba	82	74	Kapuskasing	80	69	CZECHOSLOVAKIA			IRELAND		
Fortaleza	89	78	Kenora	80	70	Prague	83	64	Dublin	70	62
Porto Alegre	89	75	Kingston	82	73	DENMARK			Shannon	71	63
Recife	86	77	Kitchener	83	72	Copenhagen	74	64	ISRAEL		
Rio de Janeiro	90	78	London	83	73	DOMINICAN REPUBLIC			Jerusalem	92	69
Salvador	86	78	North Bay	79	68	Santo Domingo	88	80	Tel Aviv	91	72
São Paulo	82	74	Oshawa	84	73	ECUADOR			ITALY		
BULGARIA			Ottawa	84	72	Guayacil	89	79	Milan	84	74
Sofia	84	69	Owen Sound	80	70	Quito	71	62	Naples	86	72
BURMA			Peterborough	83	72	EL SALVADOR			Rome	89	72
Mandalay	101	80	St. Catharines	83	73	San Salvador	95	75	IVORY COAST		
Rangoon	95	82	Sarnia	84	73	ETHIOPIA			Abidjan	88	81
CAMBODIA			Sault Ste. Marie	79	70	Addis Ababa	81	64	JAPAN		
Phnom Penh	94	82	Sudbury	81	68	Asmara	80	63	Fukuoka	89	79
CANADA (Alberta)			Thunder Bay	80	68	FINLAND			Sapporo	80	72
Calgary	79	62	Timmins	81	68	Helsinki	72	63	Tokyo	87	79
Edmonton	79	65	Toronto	85	73	FRANCE			JORDAN		
Grande Prairie	78	62	Windsor	86	74	Lyon	86	69	Amman	92	68
Jasper	77	63	CANADA (Prince Edward Island)			Marseilles	84	69	KENYA		
Lethbridge	84	65	Charlottetown	76	68	Nantes	80	67	Nairobi	78	65
McMurray	79	65	Summerside	77	68	Nice	83	72	KOREA		
Medicine Hat	87	66	CANADA (Quebec)			Paris	83	67	Pyongyang	85	76
Red Deer	78	64	Bagotville	80	68	Strasbourg	80	67	Seoul	87	78
CANADA (British Columbia)			Chicoutimi	80	68	FRENCH GUIANA			LEBANON		
Dawson Creek	76	62	Drummondville	82	71	Cayenne	90	82	Beirut	90	76
Fort Nelson	78	64	Granby	83	72	GERMANY			LIBERIA		
Kamloops	88	65	Hull	84	72	Berlin	78	66	Monrovia	88	81
Nanaimo	77	65	Mégantic	81	71	Hamburg	73	65	LIBYA		
New Westminster	78	66	Montréal	83	72	Hannover	75	65	Bengasi	91	75
Penticton	87	67	Québec	81	70	Mannheim	82	68	MADAGASCAR		
Prince George	77	62	Rimouski	76	67	Munich	80	64	Rananarive	83	71
Prince Rupert	61	57	St. Jean	84	72	GHANA			MALAYSIA		
Trail	86	65	St. Jérôme	83	72	Accra	89	79	Kuala Lumpur	92	81
Vancouver	74	66	Sept. Iles	70	63	GIBRALTER			Penang	92	80
Victoria	70	60	Shawnigan	82	71	Gibraltar	86	74	Singapore	90	80
CANADA (Manitoba)			Sherbrooke	81	71	GREECE			MARTINIQUE		
Brandon	83	70	Thetford Mines	81	71	Athens	91	71	Fort de France	88	80
Churchill	74	63				Thessaloniki	91	75	MEXICO		
Dauphin	81	70				GREENLAND			Guadalajara	89	66
Flin Flon	79	67				Narsarsuaq	61	52	Mérida	94	77
									Mexico City	79	59
									Monterrey	93	77
									Vera Cruz	88	82

FIG. 11-3b (Cont'd)
Dry Bulb/Wet Bulb Temperature Data²

Country and Station	Design Dry-Bulb °F 5%	Design Wet-Bulb °F 5%	Country and Station	Design Dry-Bulb °F 5%	Design Wet-Bulb °F 5%	Country and Station	Design Dry-Bulb °F 5%	Design Wet-Bulb °F 5%	Country and Station	Design Dry-Bulb °F 5%	Design Wet-Bulb °F 5%
MOROCCO			PHILIPPINES			Minsk	74	65	TRINIDAD		
Casablanca	86	70	Manilla	91	81	Moscow	78	65	Port of Spain	89	79
NEPAL			POLAND			Odessa	82	68	TUNISIA		
Katmandu	86	76	Kraków	78	66	Petropavlovsk	65	56	Tunis	96	74
NETHERLANDS			Warsaw	78	68	Rostov on Don	84	68	TURKEY		
Amsterdam	73	63	PORTUGAL			Sverdlovsk	72	60	Adana	95	77
NEW GUINEA			Lisbon	83	67	Tashkent	90	69	Ankara	89	66
Manokwari	87	81	PUERTO RICO			Tblisi	83	66	Istanbul	86	73
Point Moresby	90	79	San Juan	87	79	Vladivostok	74	68	Izmir	94	73
NEW ZEALAND			RUMANIA			Volgograd	86	69	UNITED ARAB		
Auckland	76	65	Bucharest	89	70	SPAIN			REPUBLIC		
Christ Church	76	66	SAUDI ARABIA			Barcelona	84	73	Cairo	98	74
Wellington	72	64	Dhahran	108	84	Madrid	89	67	UNITED		
NICARAGUA			Jedda	100	83	Valencia	88	73	KINGDOM	69	62
Managua	92	79	Riyadh	106	76	SUDAN			Belfast		
NIGERIA			SENEGAL			Khartoum	104	75	Birmingham	73	63
Lagos	90	81	Dakar	91	80	SURINAM			Cardiff	73	62
NORWAY			SOMALIA			Paramaribo	90	81	Edinburgh	68	61
Bergen	73	65	Mogadiscio	89	81	SWEDEN			Glasgow	68	61
Oslo	74	64	SOUTH AFRICA			Stockholm	72	60	London	76	65
PAKISTAN			Capetown	86	70	SWITZERLAND			URUGUAY		
Chittagong	89	81	Johannesburg	81	69	Zurich	78	66	Montevideo	85	71
Karachi	95	81	Pretoria	85	68	SYRIA			VENEZUELA		
Lahore	105	81	SOVIET UNION			Damascus	98	70	Caracas	81	69
Peshwar	103	79	Alma Ata	83	67	TAIWAN			Maracaibo	95	83
PANAMA & CANAL ZONE			Archangel	68	57	Tainan	90	82	VIET NAM		
Panama City	91	80	Kaliningrad	77	65	Taipei	90	81	Da Nang	93	85
PARAGUAY			Krasnoyarsk	76	60	TANZANIA			Hanoi	95	84
Asunción	96	80	Kiev	81	67	Dar es Salaam	88	81	Saigon	89	83
PERU			Kharkov	82	67	THAILAND			YUGOSLAVIA		
Lima	84	74	Kuibyshev	81	66	Bangkok	93	81	Belgrade	86	72
			Leningrad	72	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.

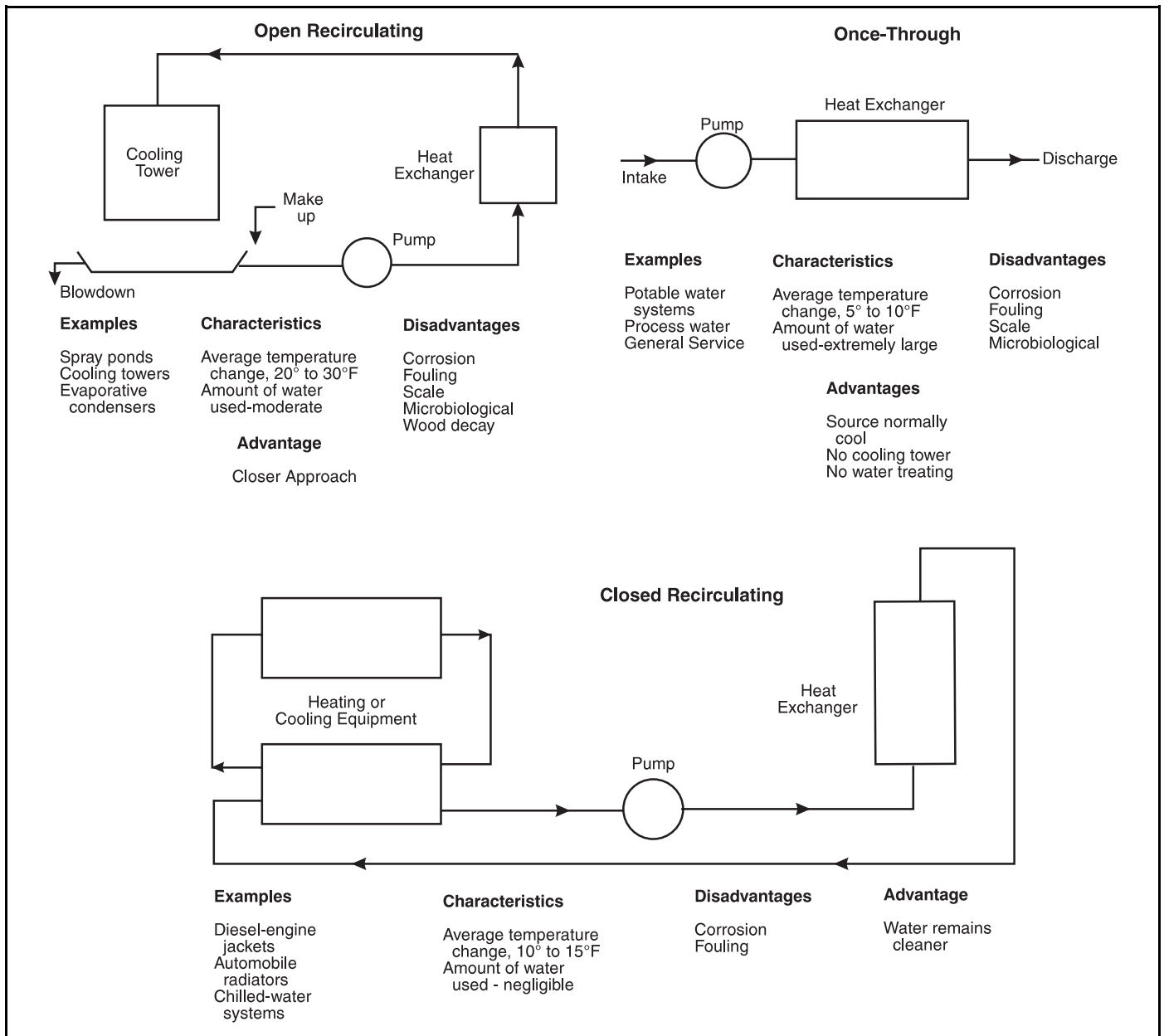
The data for Canadian stations are based on the month of July only.

AP = airport AFB = air force base CO = urban offices

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FIG. 11-4

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

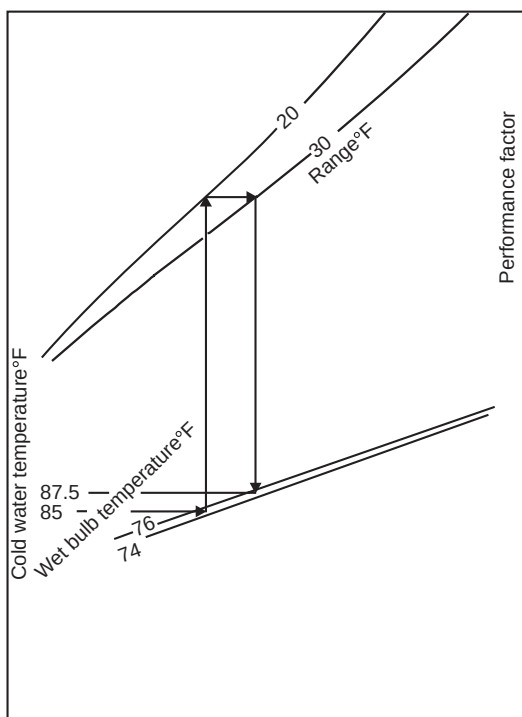
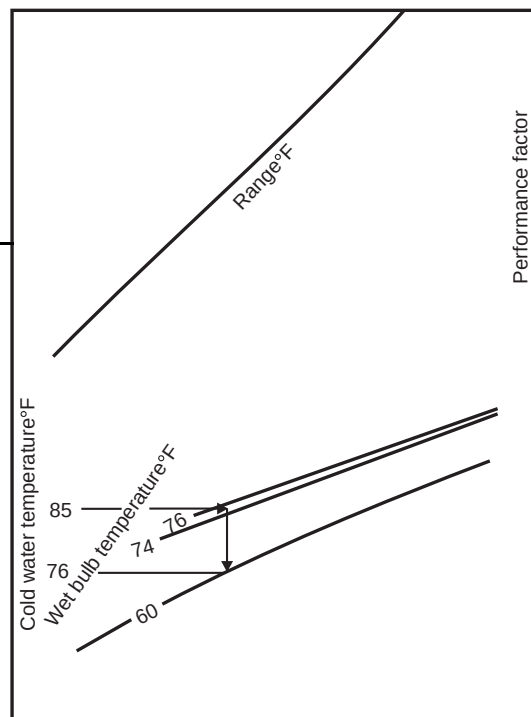
The performance characteristics of various types of towers will vary with height, fill configuration, and flow arrangement — crossflow or counterflow; however, these factors have been taken into consideration in preparation of the Performance Characteristic Nomograph Fig. 11-5. When accurate characteristics 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

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?

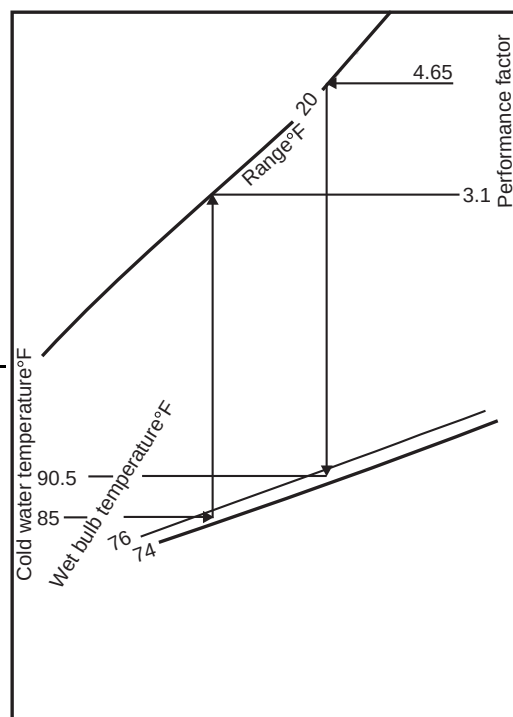
Enter Nomograph at 85° CWT, go horizontally to 75° WB, then vertically down to 60° WB, read new CWT of 76°.



Example 11-2 — Effect of Varying Cooling Range on Cold Water Temperature.

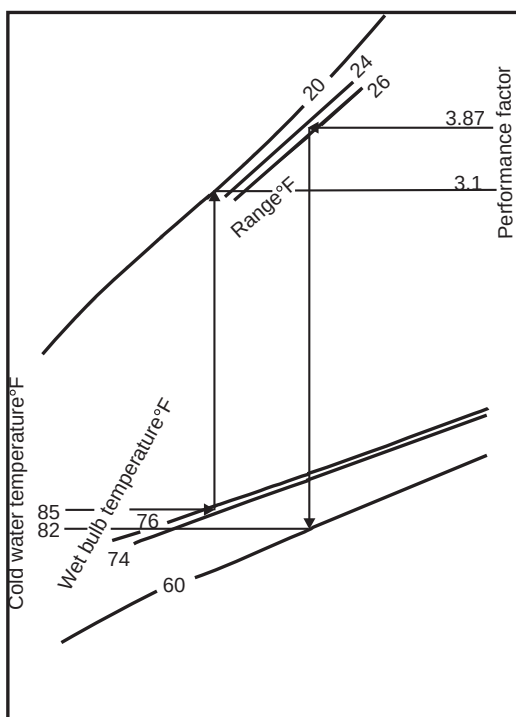
What is new CWT when cooling range is changed from 20° to 30° (50% increase in heat load) with gpm and WB held constant?

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°.

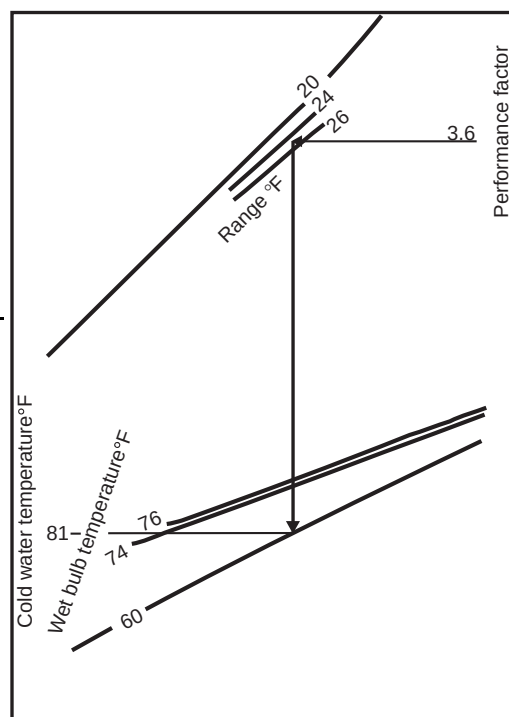


Example 11-4 — Effect of Varying WB Temperature, Range, and Water Circulating Rate on Cold Water Temperature.

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 = $(25 \text{ HP}/20 \text{ HP})^{1/3} = 1.077$. 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:

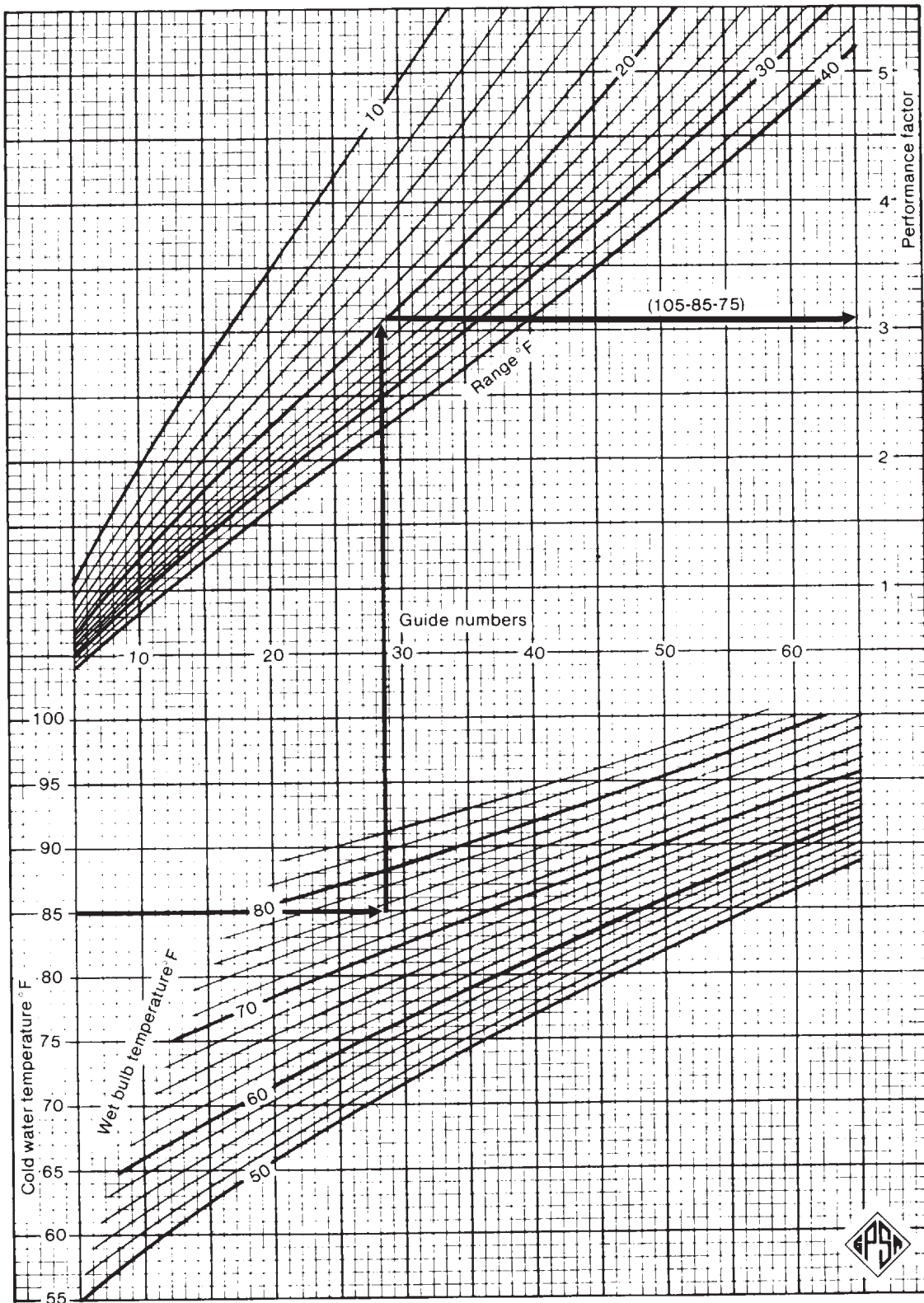
Circulation Rate	=	10,000 gpm
Water Temperature Drop Through Tower	=	20°F
Type of Tower	=	Mechanical Induced Draft
Blowdown Rate	=	20 gpm, or 0.2% of circulation rate

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)

FIG. 11-5
Performance Characteristic Nomograph



$$\begin{aligned}\text{Number of} \\ \text{Concentrations (cycles)} &= \frac{E + B}{B} \\ &= \frac{2.0 + (0.2 + 0.3)}{(0.2 + 0.3)} = 5.0\end{aligned}$$

If the resultant concentrations are excessive and a desired concentration of 4.0 is required, what must the blowdown rate be?

$$\begin{aligned}B &= \frac{E}{\text{Cycles} - 1} \\ &= \frac{2.0}{4.0 - 1} = 0.67\%\end{aligned}$$

The windage component of B is 0.3%, therefore the blowdown rate required would be $0.67 - 0.3 = 0.37\%$ or

$$(10,000 \text{ gpm}) (0.0037) = 37 \text{ gpm}$$

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.

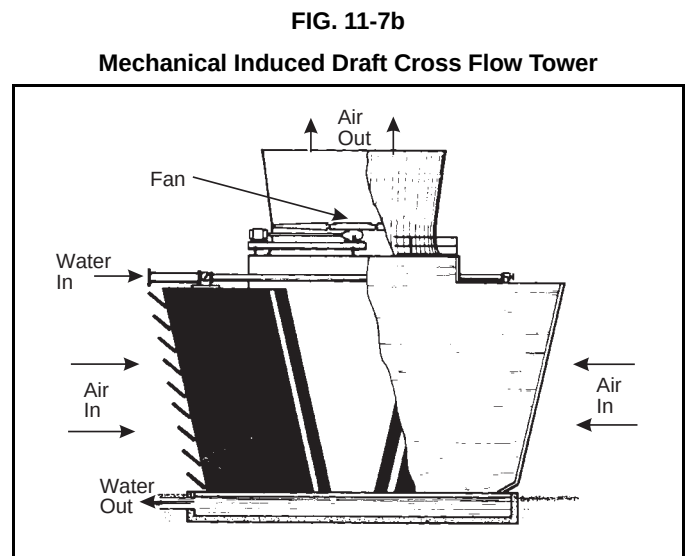
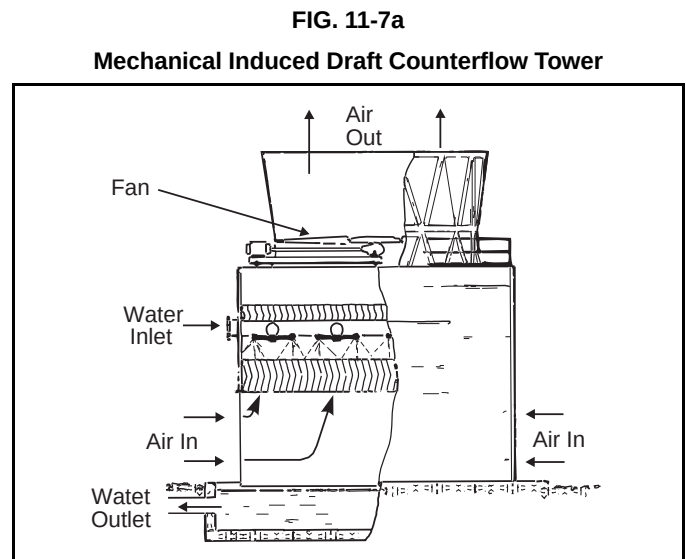
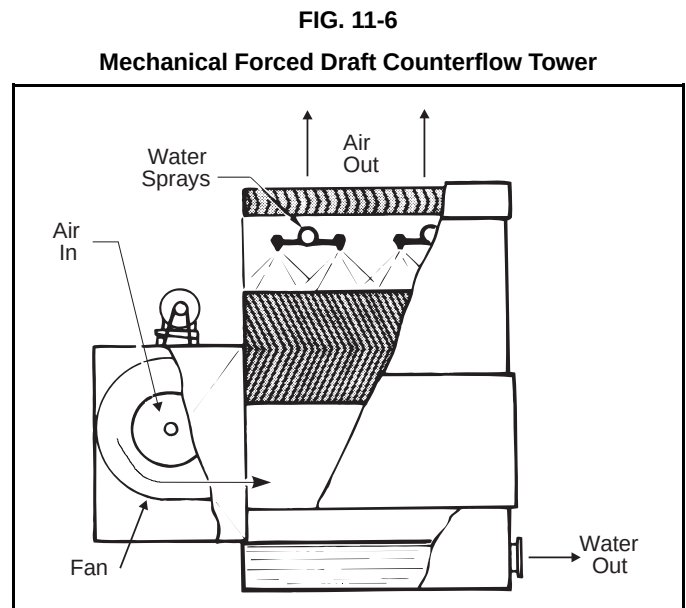


FIG. 11-8

Mechanical Draft Coil Shed Tower

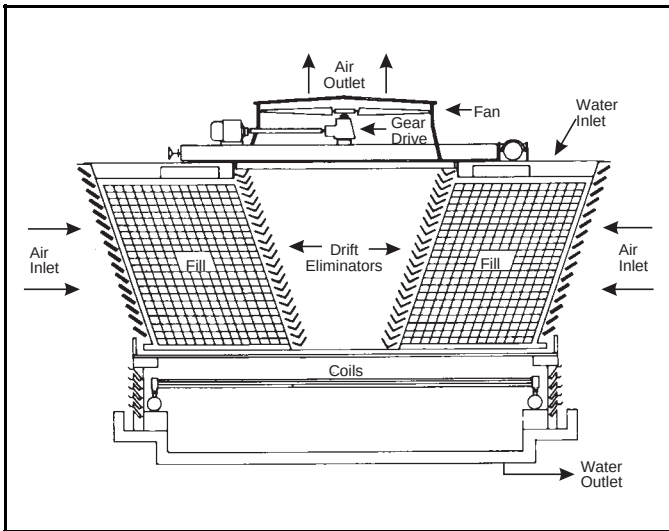
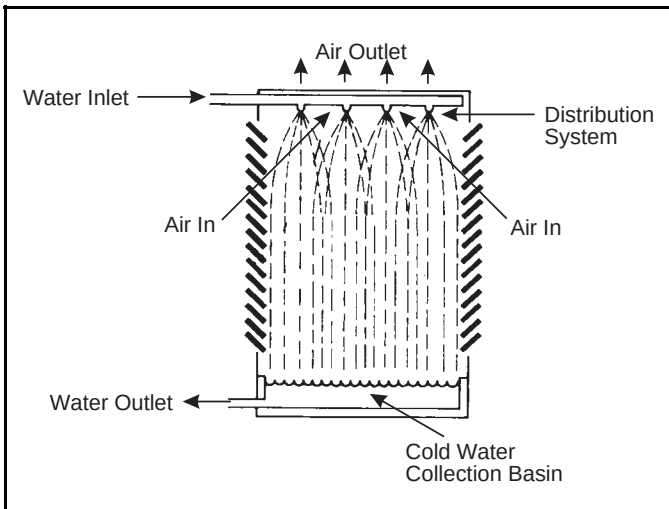


FIG. 11-9

Atmospheric Spray Tower

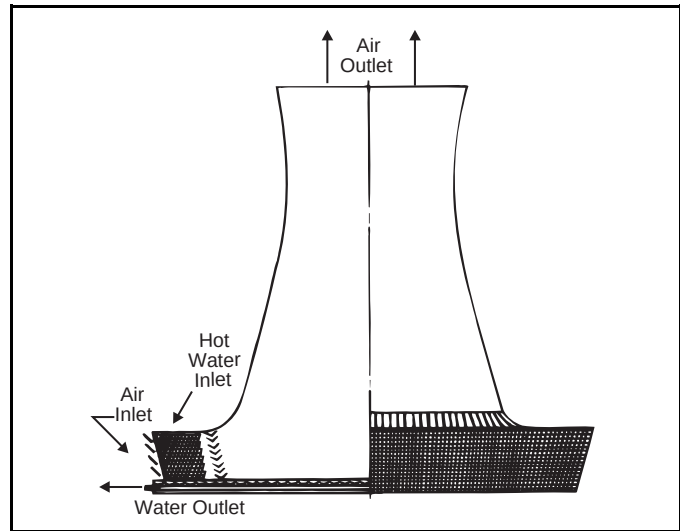


Induced draft towers (Fig. 11-7) — The fan is located on the air stream leaving the tower. This causes air exit velocities which are three to four times higher than their air entrance velocities. This improves the heat dispersion and reduces the potential for recirculation. Induced draft towers require 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.

FIG. 11-11
Properties of Saturated Air¹

Temp. °F	Humidity Ratio lbw/lb _a		Volume ft ³ /lb dry air			Enthalpy Btu/lb dry air		
	W _s	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	

Temp. °F	Humidity Ratio lbw/lb _a		Volume ft ³ /lb dry air			Enthalpy Btu/lb dry air		
	W _s	v _a	v _{as}	v _s	h _a	h _{as}	h _s	
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	223.932	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	0.7233							

NOTES:

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.	sp gr = specific gravity at average flowing conditions. Equal to RD
a = cross-sectional area of piston rod, sq in.	T = torque, ft lb
AC = alternating current	t_r = temperature rise, °F
bbl = barrel (42 U.S. gallons)	u = impeller peripheral velocity, ft/sec
bhp = brake horsepower	VE = volumetric efficiency, fraction
C = constant (Fig. 12-19)	VE _o = overall volumetric efficiency
C _p = specific heat at average temperature, Btu/(lb • °F)	VE _p = volumetric efficiency due to density change
cfs = cu ft/sec	VE _l = volumetric efficiency due to leakage
D = displacement of reciprocating pump, gpm	V _{pd} = pulsation dampener volume, in ³
DC = direct current	v = liquid mean velocity at a system point, ft/sec
d = impeller diameter, in.	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.
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)	
	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

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.

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.

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:

$$h = z + h_p + h_v = z + \frac{2.31 \cdot P}{sp \text{ gr}} + \frac{v^2}{2 \cdot g} \quad \text{Eq 12-1}$$

The system friction head is the inlet system friction head plus the outlet system friction head:

$$h_f = h_{fx} + h_{fy} \quad \text{Eq 12-2}$$

The equipment total head is the outlet nozzle total head minus the inlet nozzle total head:

$$H = h_o - h_i = z_o - z_i + \frac{2.31 (P_o - P_i)}{sp \text{ gr}} + \frac{v_o^2 - v_i^2}{2 \cdot g} \quad \text{Eq 12-3}$$

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:

$$z_x + \frac{2.31 \cdot P_x}{sp \text{ gr}} + \frac{v_x^2}{2 \cdot g} - h_{fx} + H = z_y + \frac{2.31 \cdot P_y}{sp \text{ gr}} + \frac{v_y^2}{2 \cdot g} + h_{fy}$$

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:

$$H_c = 1.155 (P_o - P_i) \left[\frac{1}{sp \text{ gr}_o} + \frac{1}{sp \text{ gr}_i} \right] \quad \text{Eq. 12-5}$$

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. Usually 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

$$NPSHA = \frac{2.31 \cdot (P_i - P_{vp})}{sp \text{ gr}} + z_i + \frac{v_i^2}{2 \cdot g} \quad \text{Eq 12-6a}$$

FIG. 12-2

Common Pump Equations

FLOW RATE						
Given $\Rightarrow$ multiply by to get $\downarrow$	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 $\Rightarrow$ multiply by to get $\downarrow$	kPa	ft water at 39.2°F	m water at 0°C	ft liquid	bar	* std atm 760 mm Hg at 0°C
lb/in ²	0.145	0.4335	1.422	sp gr / 2.31	14.5038	14.6959
						kgf/cm ²
						14.2233
DENSITY						
Given $\Rightarrow$ multiply by to get $\downarrow$	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	

$$h_p = \frac{144 \cdot P}{\rho \cdot v^2} = \frac{2.31 \cdot P}{\text{sp gr}}$$

$$h_v = \frac{v^2}{2 \cdot g}$$

$$u = \frac{(d)(n)}{229}$$

$$v = \frac{(Q)(0.321)}{A}$$

$$1 \text{ HP} = 0.7457 \text{ kW}$$

$$= 550 \text{ ft} \cdot \text{lb}/\text{s}$$

$$= 33,000 \text{ ft} \cdot \text{lb}/\text{min}$$

$$\text{hyd hp} = \frac{Q \cdot H \cdot \text{sp gr}}{3,960} = \frac{Q \cdot \Delta P}{1,714}^{**}$$

$$\text{bhp} = \frac{\text{hyd hp}}{e} \text{ (for pumps)}$$

$$\text{bhp} = \text{hyd hp} \cdot e \text{ (for turbines)}$$

sp gr = specific gravity

$$\text{Water density at } 60^\circ\text{F} = 62.37 \text{ lb/ft}^3$$

$$\text{Standard gravity acceleration:}$$

$$g = 9.80665 \text{ m/s}^2 = 32.1740 \text{ ft/s}^2$$

$$\text{bhp} = \frac{Q \cdot H \cdot \text{sp gr}}{(3,960)(e)} = \frac{Q \cdot \Delta P}{1,714 \cdot e}^{**} \text{ (for pumps)}$$

$$T = \frac{(\text{bhp})(5252)}{n}$$

$$n_s = \frac{n \cdot \sqrt{Q_{\text{bep}}}}{(H_{\text{bep}})^{3/4}} = \frac{n \cdot H_{\text{bep}}^{1/4} \cdot \sqrt{Q_{\text{bep}}}}{H_{\text{bep}}}$$

See Fig. 1-7 for viscosity relationships

*Standard atmospheric pressure:

$$1 \text{ atm} = 760 \text{ mm Hg} = 101.325 \text{ kPa} = 14.696 \text{ psi}$$

**See Eq. 12-3 and 12-4.

CENTRIFUGAL PUMPS AFFINITY LAWS

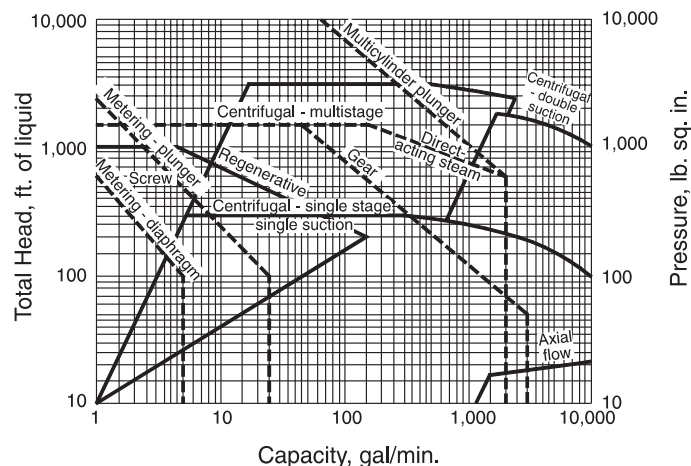
1: Values at initial conditions

2: Values at new conditions

CHANGE $\Rightarrow$	SPEED	DIAMETER	SPEED AND DIAMETER
$Q_2 =$	$Q_1 (n_2/n_1)$	$Q_1 (d_2/d_1)$	$Q_1 (d_2/d_1) (n_2/n_1)$
$h_2 =$	$h_1 (n_2/n_1)^2$	$h_1 (d_2/d_1)^2$	$h_1 [(d_2/d_1) (n_2/n_1)]^2$
$\text{bhp}_2 =$	$\text{bhp}_1 (n_2/n_1)^3$	$\text{bhp}_1 (d_2/d_1)^3$	$\text{bhp}_1 [(d_2/d_1) (n_2/n_1)]^3$
$\text{NPSHR}_2 =$	$\text{NPSHR}_1 (n_2/n_1)^2$	—	$\text{NPSHR}_1 (n_2/n_1)^2$

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.

Courtesy of McGraw Hill Publishing Company

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.

$$\text{NPSHA} = \frac{2.31 \cdot (P_x - P_{vp})}{\text{sp gr}} + z_x + \frac{v_x^2}{2 \cdot g} - h_{fx} \quad \text{Eq 12-6b}$$

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.

$$\text{NPSHA} = \frac{2.31 \cdot (P_{sv} - P_{vp})}{\text{sp gr}} + z_{sv} - h_{fsv} \quad \text{Eq 12-6c}$$

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.

$$S = \frac{n\sqrt{Q_{bep}}}{\text{NPSHR}_{bep}^{3/4}} \quad \text{Eq. 12-7}$$

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 Standards⁵ 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

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 ves-

sels 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

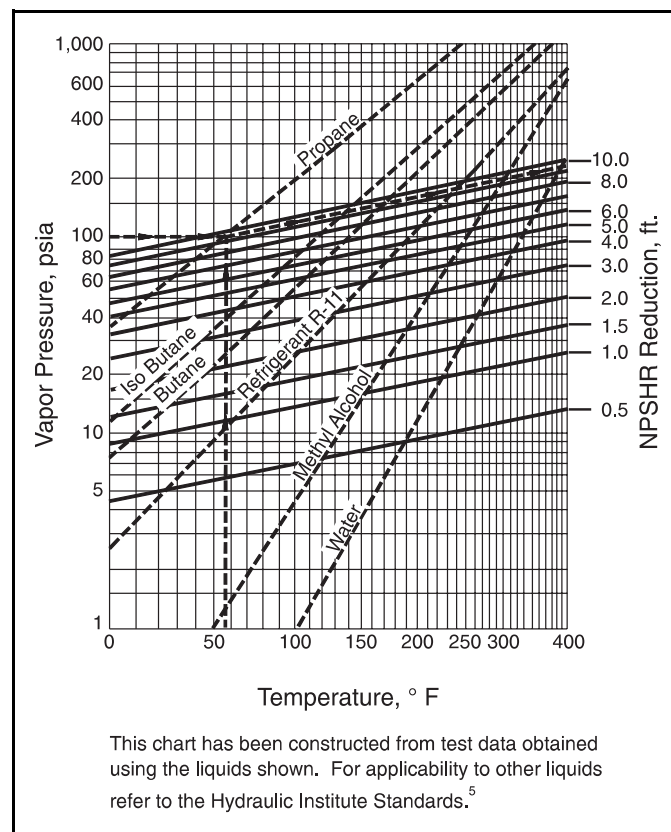
Absolute Total Pressure at Pump Discharge

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



Differential pressure = $250.0 - 188.8 = 61.2$ psi

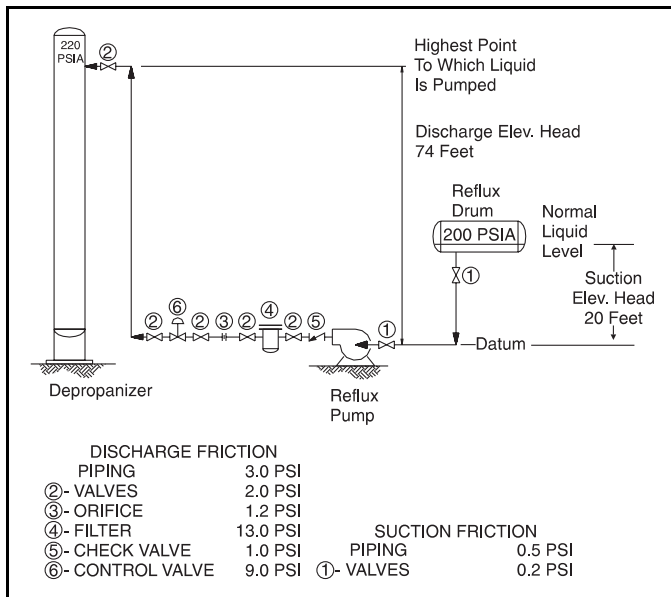
Differential head = $H = \frac{(61.2)(2.31)}{0.485} = 292$ ft

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	$20 \text{ ft} \cdot 0.485/2.31 =$	+4.2 psi
Friction	valves =	-0.2 psi
	pipng =	-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

$$\text{hyd hp} = \frac{Q \cdot H \cdot \text{sp gr}}{3960} \quad (\text{from Fig. 12-2})$$

$$\text{hyd hp} = \frac{(360)(322)(0.485)}{3960} = 14.2 \text{ hp}$$

Calculation of Actual Horsepower

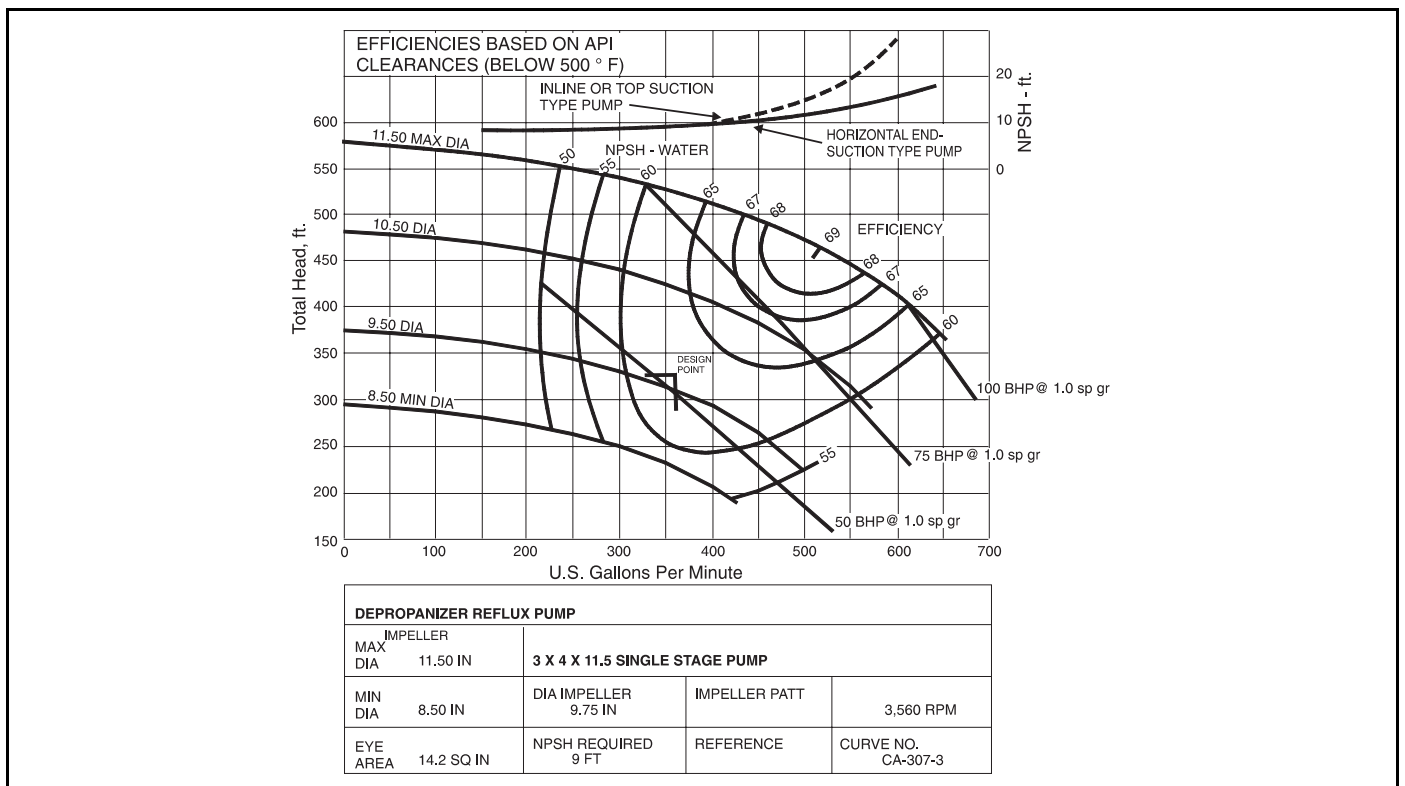
$$\text{bhp} = \frac{\text{hyd hp}}{e} \quad (\text{from Fig. 12-2})$$

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



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 rota-

tional 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 head-loss 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

FIG. 12-8a

Horizontal Single Stage Process Pump

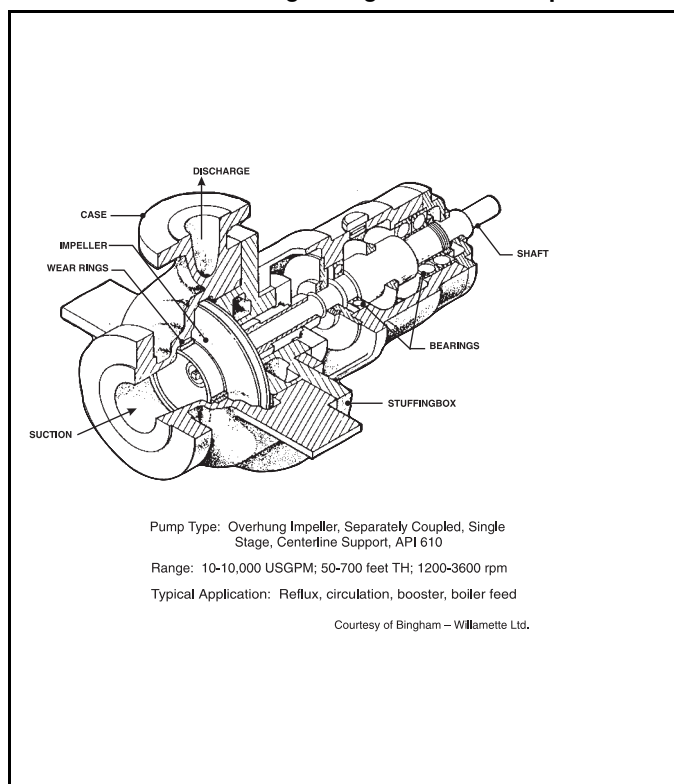


FIG. 12-8b

Vertical Inline Pump

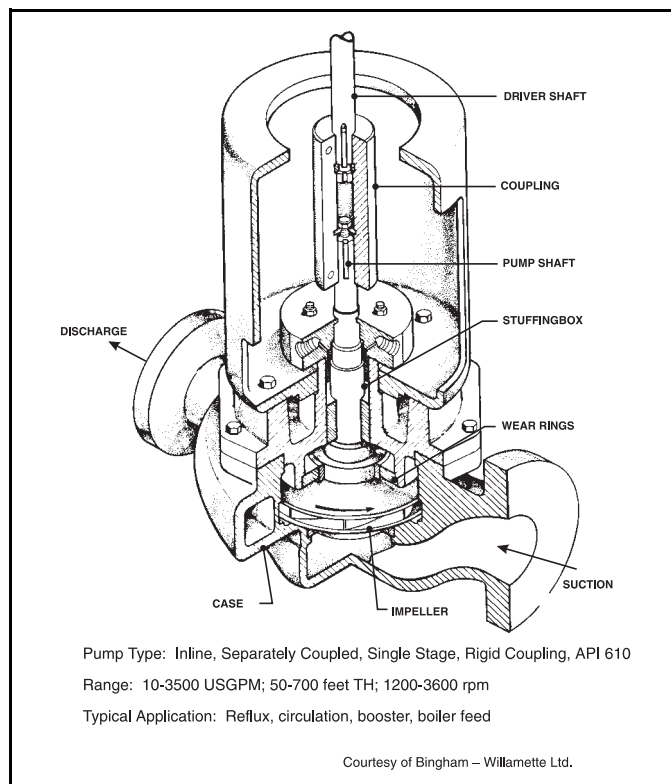


FIG. 12-8c

Horizontal Multi-Stage Pump

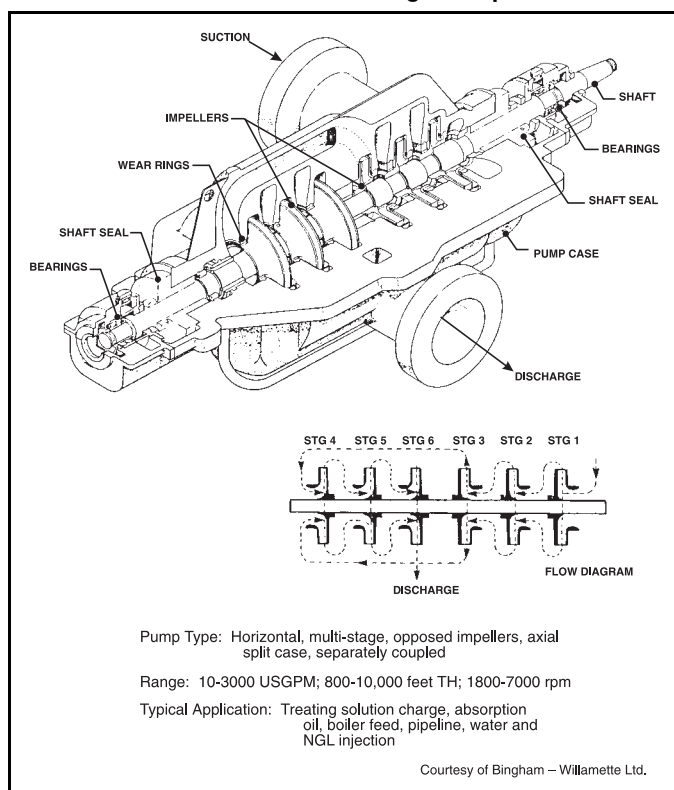


FIG. 12-8d

Vertical Can Pump

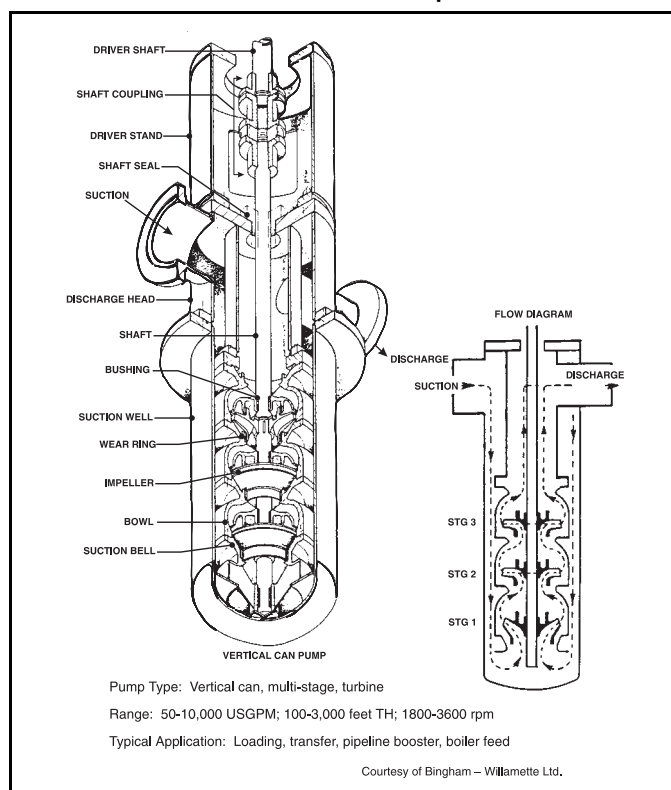
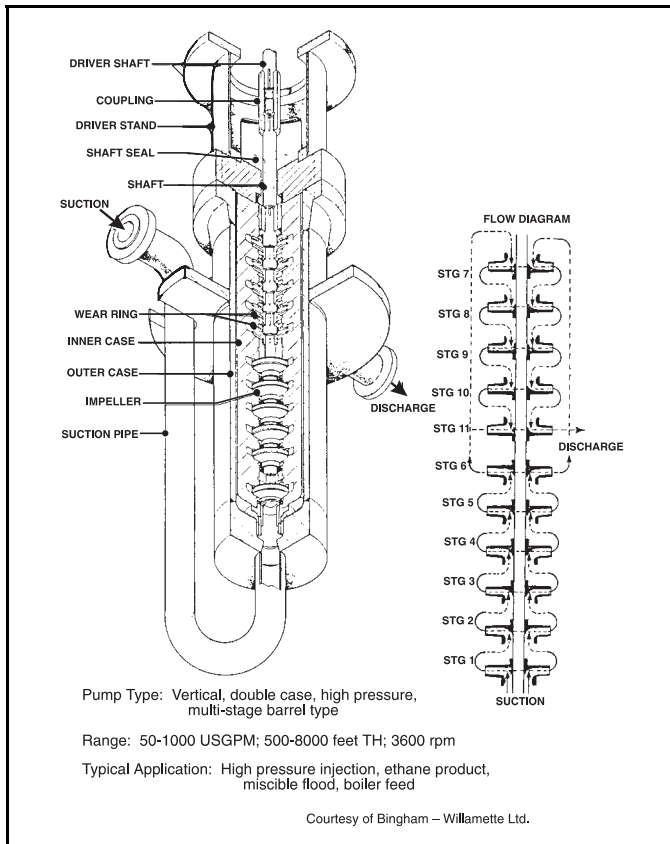


FIG. 12-8e

Vertical, High Pressure, Double Case, Multi-Stage Pump



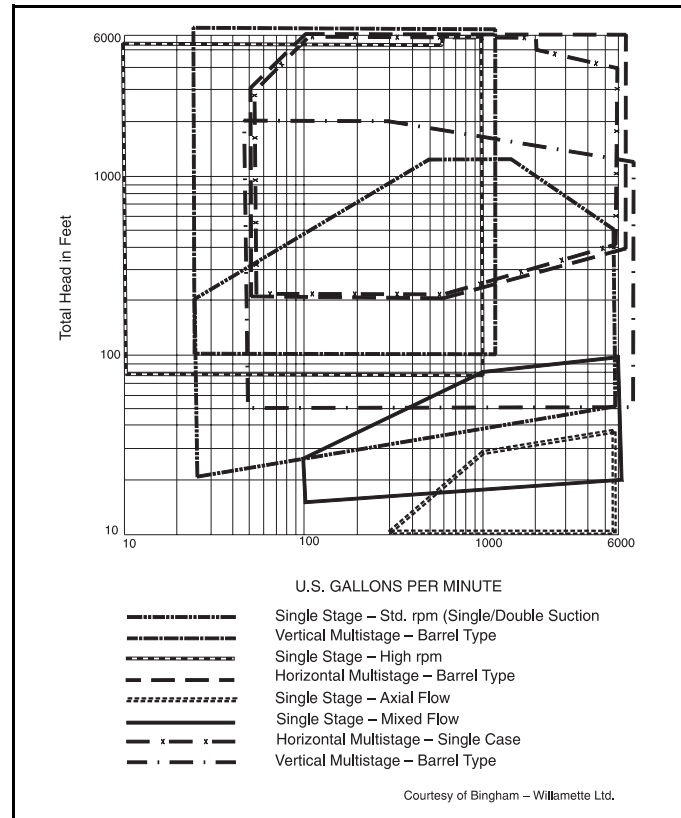
losses so that pump capacity is reduced to that required. In Fig. 12-12, the required flow rate is represented by "B." Required amount of extra frictional losses to achieve a flow rate of "B" is represented on Fig. 12-12 by the difference between "H_{B-PUMP}" and "H_{B-SYSTEM}." Frequently the throttling 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



$$bhp_2 = bhp_1 \left(\frac{e_1}{e_2} \right) \left[\frac{h_s (Q_2/Q_1) + h_{f1} (Q_2/Q_1)^3}{h_s + h_{f1}} \right] \quad \text{Eq 12-8}$$

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

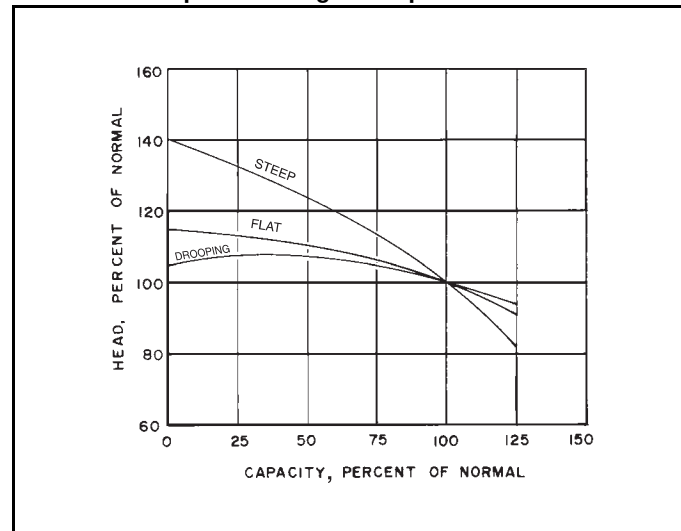
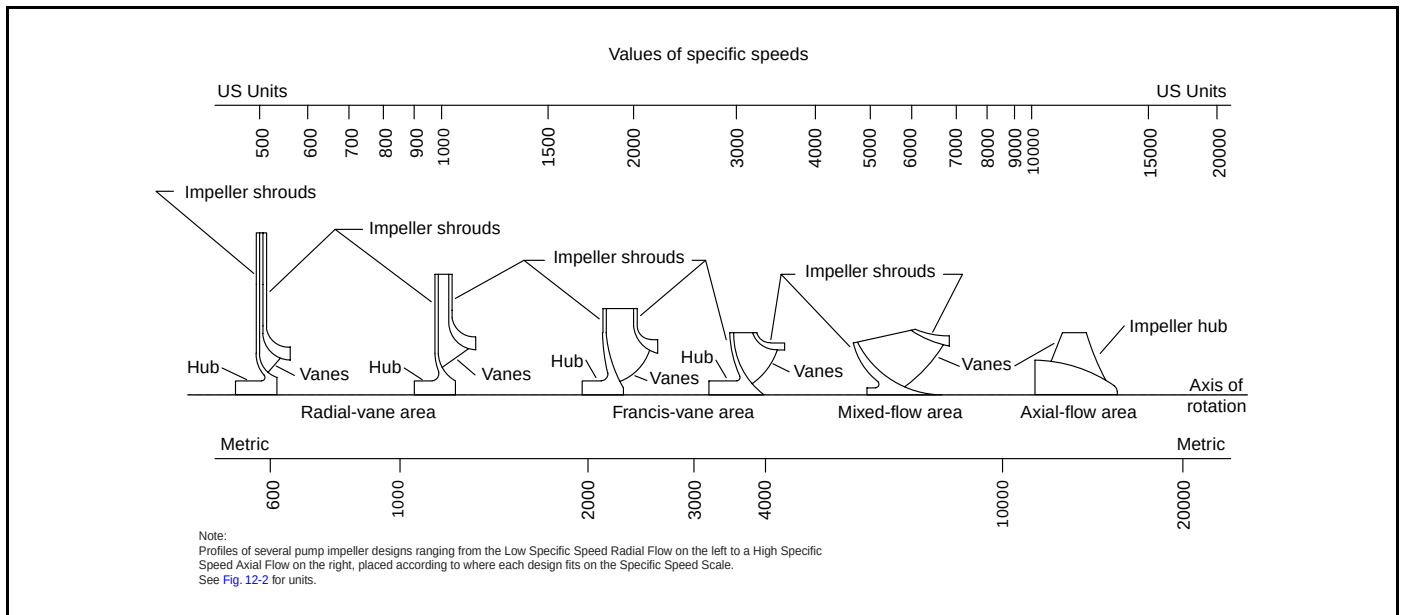


FIG. 12-11
Values of Specific Speeds (n_s)



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.

$$t_r = \frac{H \left(\frac{1}{e} - 1 \right)}{778 C_p} \quad \text{Eq 12-9}$$

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.

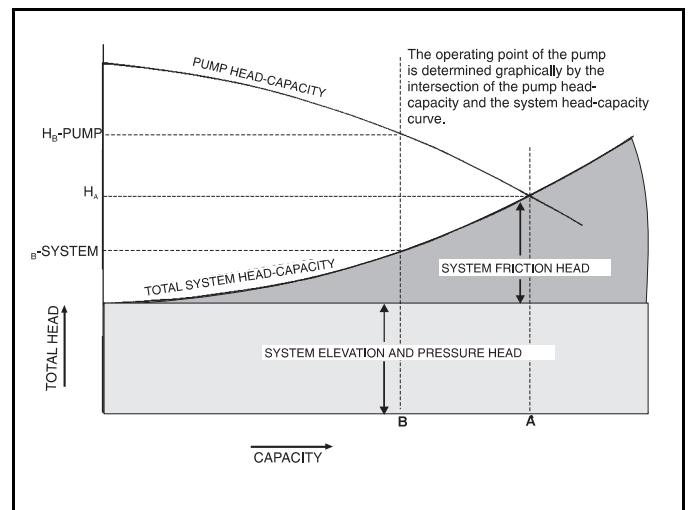
Pump vendors should be requested to provide data on minimum flow.

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 consideration

FIG. 12-12
Example Combined Pump-System Curves



should be given to the pump curve shape when selecting pumps for series or parallel operation.

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.

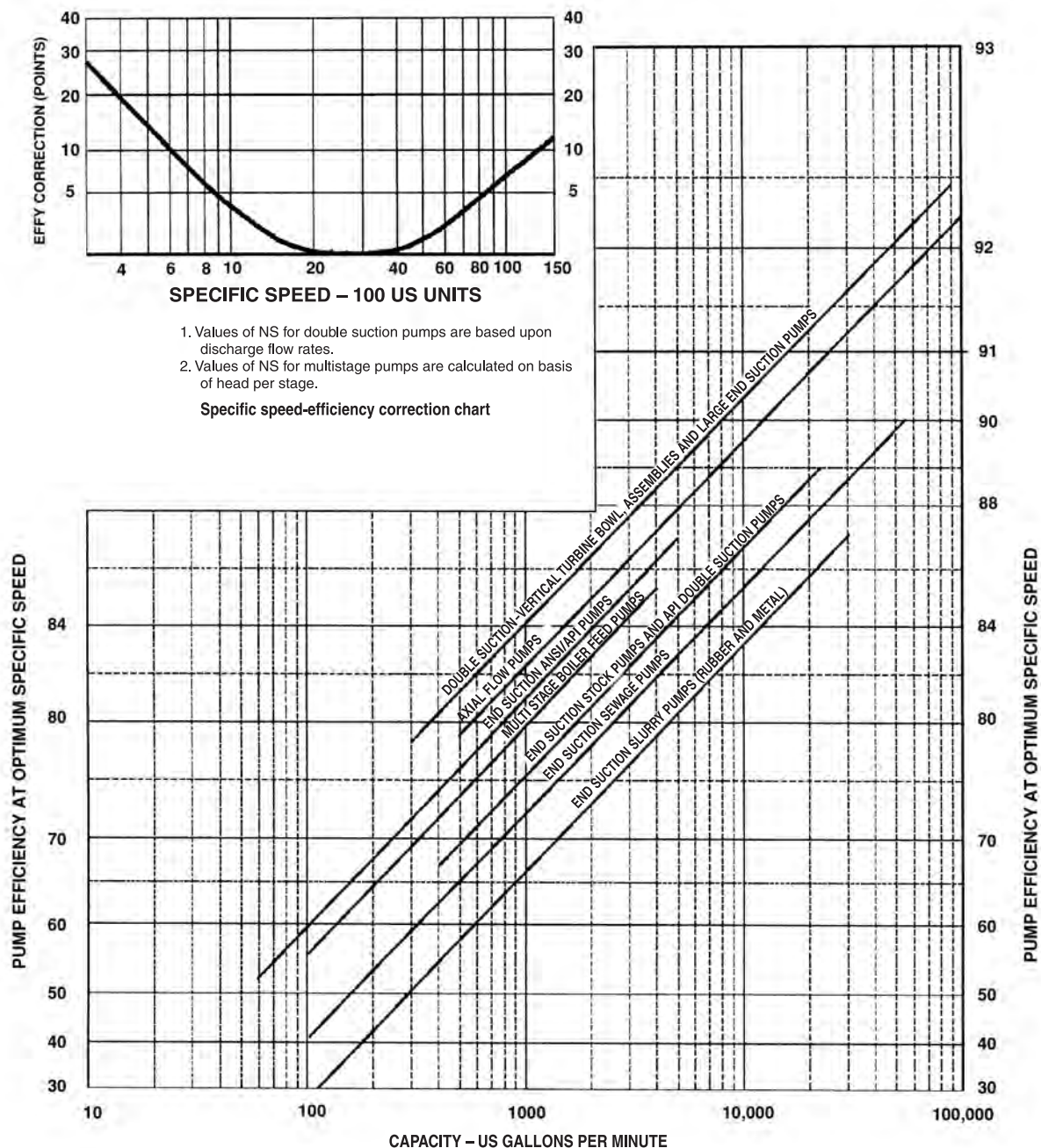
Drivers

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:

FIG. 12-13

Optimum Generally Attainable Efficiency Chart



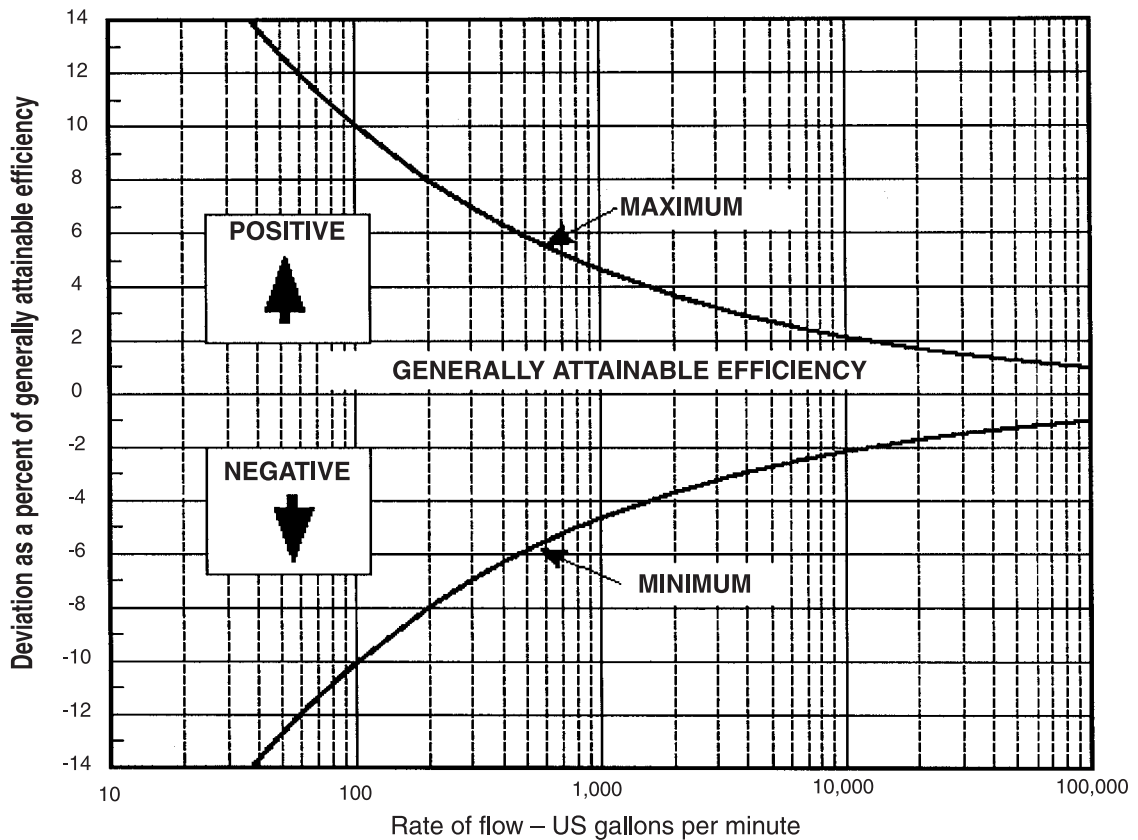
NOTES:

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

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

FIG. 12-15
Series Pumps Selection

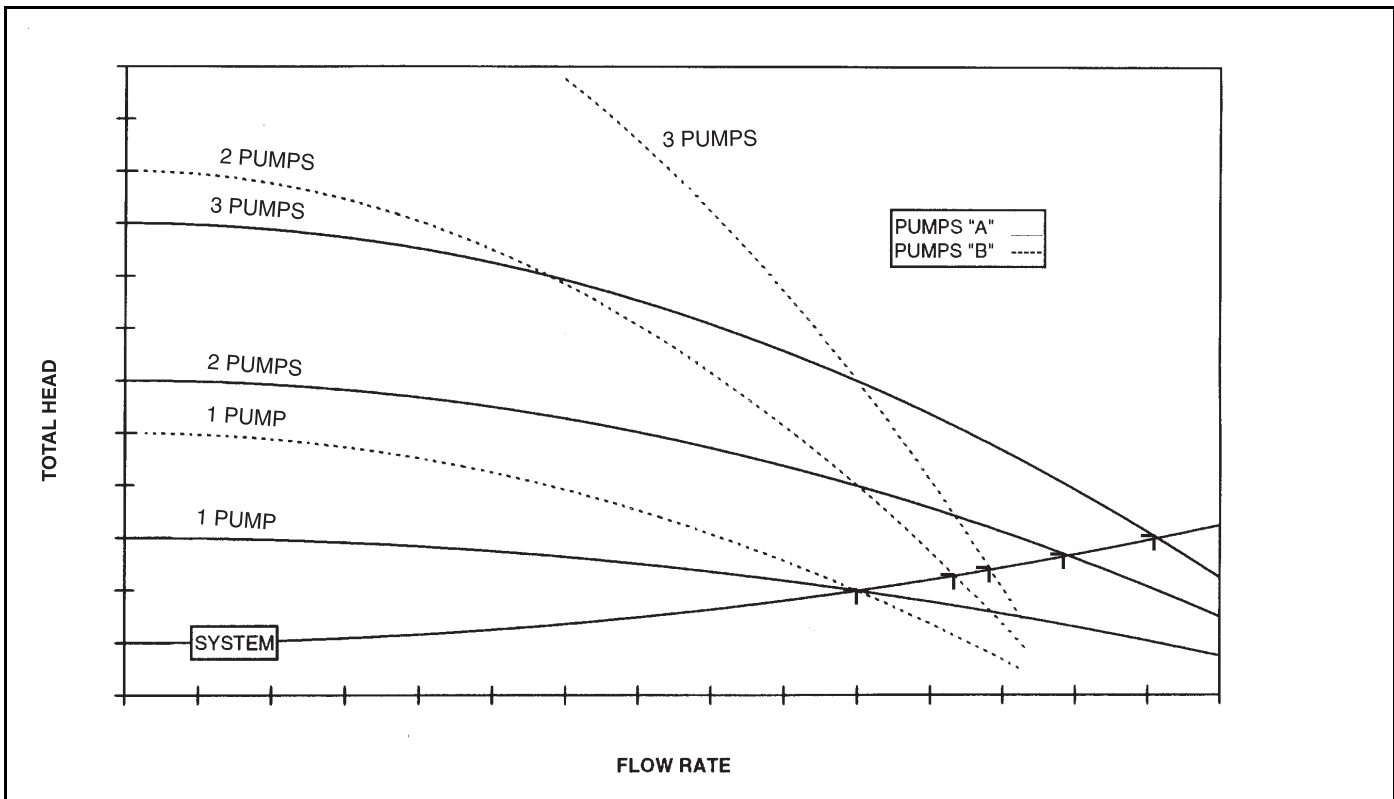


FIG. 12-16
Parallel Pumps Selection

