Hydraulic accumulators

Objectives: Energy storage, impact and pulsation damping, operation as spring element.

The hydraulic accumulator consists of a shell, the interior of which is separated by a solid or flexible barrier. On one side of the barrier is gas, on the other fluid. There are three basic types of units: The bladder, the diaphragm and the piston accumulator.

Nitrogen is employed as the gas medium. During operation, the pressure from the fluid compresses the gas. The minimum operating pressure p_1 should lie at least 10% above the initial gas pressure p_0. The pressure variation between the initial gas pressure and the maximum operating pressure p_2 should not exceed the following: 1:4 in diaphragm and bladder accumulators; 1:10 in piston accumulators. The three operating states illustrated in the diagram are governed by the laws of polytropic changes in state:

$$p_0 \cdot V^n_0 = p_1 \cdot V^n_1 = p_2 \cdot V^n_2$$

For nitrogen, the polytropic exponent for isothermal changes of state is n = 1 and for adiabatic changes of state, n = 1.4. The available fluid volume between operating pressures is equivalent to the volume difference:

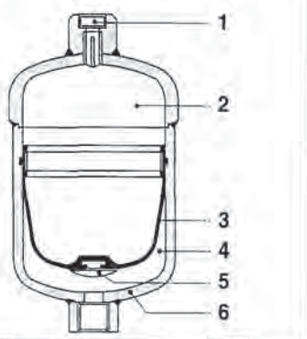
$$\Delta V = V_1 - V_2.$$

Auxiliary drives

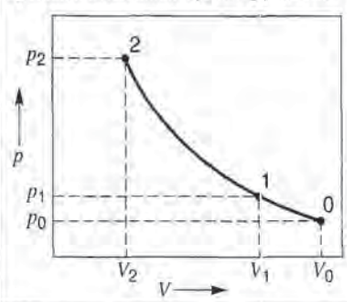
Electrohydraulic devices are employed as drive units in numerous automotive applications. The advantages associated with a positive power-to-weight ratio are accompanied by flexibility in installation. Electrohydraulic devices are used to control hoists and trailing axles, and for raising tractor axles. They are also used for controlling the steering and lifting mechanisms on industrial trucks and other vehicles. Platform lifts for loads of 500 to 5,000 kg represent a major area of application for electrohydraulic devices.

The motion of the platform is divided into both lowering and raising. While the upward or downward stroke is controlled by a single central or two outside cylinders, the tilt function is usually governed by two cylinders. These are either single-action cylinders with return springs, or double-action cylinders. The tilt-control functions are generally governed hydraulically. In addition to the "raise" and "lower" functions, other important features are a "floating" position for loading at fixed-position docks, positioning at an optional height (tilting under ful load), and compliance with specified "raising" and "lowering" speeds.

Diaphragm accumulator
1 Screw cap, 2 Gas chamber, 3 Diaphragm, 4 Fluid chamber, 5 Plug, 6 Steel case.



Operating states of hydraulic accumulator
p_0 Initial gas pressure, p_1 Minimum operating pressure, p_2 Maximum operating pressure, V_0 Volume at initial gas pressure, V_1 Volume at minimum operating pressure, V_2 Volume at maximum operating pressure.



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Hydrostatic fan drives

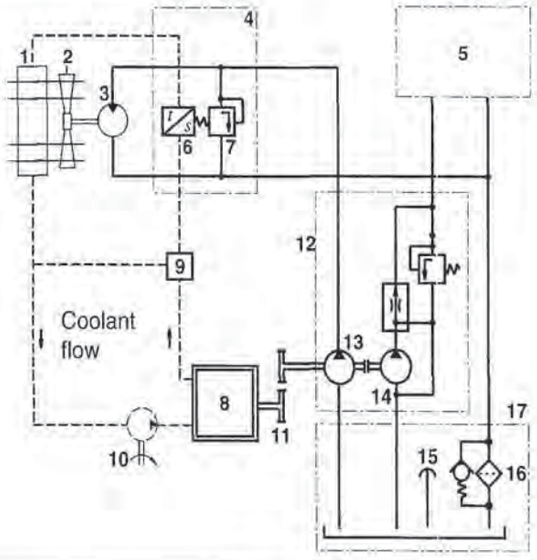
Air-cooling of engine coolant is regulated by thermostatic control of radiator-fan speed. In situations which demand greater scope in the siting of the radiator (due to space constraints, engine encapsulation) in positions where it is not in close proximity to the engine, hydrostatic fan drive systems are generally used for higher-power applications (busses, trucks, construction and agricultural machinery, fixed plant). Aside from the flexibility in positioning the radiator, these units also have the advantages of high power-to-weight ratios (low weight, compact component dimensions), uncomplicated control and regulation, reliability, and the reduced component wear which results from the hydraulic fluid's lubrication effect.

The essential components of the hydrostatic fan drive are the hydraulic pump, the motor (high-pressure gear or piston units), and the temperature-controlled valve in the bypass line to the hydraulic motor for controlling fan speed. The hydraulic pump is driven by the engine either directly or indirectly using a V-belt drive (conversion ratio). The pump, in turn, powers the hydraulic motor in the fan assembly. The motor speed depends on the fan's specific response properties ($n_L - \sqrt{\Delta p_M}$) and the effective pressure differential (Δp_M). If losses associated with transmitting the power are discounted, the speed will be directly proportional to system pressure (p).

Both continuous-action and discontinuous-action control are employed to govern the engine-coolant temperature. With two-point control (discontinuous), the bypass valve is in the form of an electrically triggered directional-control valve, with actuation controlled by a thermo-switch in the engine's coolant circuit. A pressure valve mounted in a parallel circuit determines the maximum fan speed – and thus cooling power – which will be obtained when the directional-control valve is closed. The precise regulating response is a function of the system pressure to which the valve is set (usually 200 bar). Continuous-control systems feature a bypass valve in the form of a pressure valve or throttle valve with a supplementary bypass-pressure valve for limiting system

Combined hydrostatic fan drive and power steering

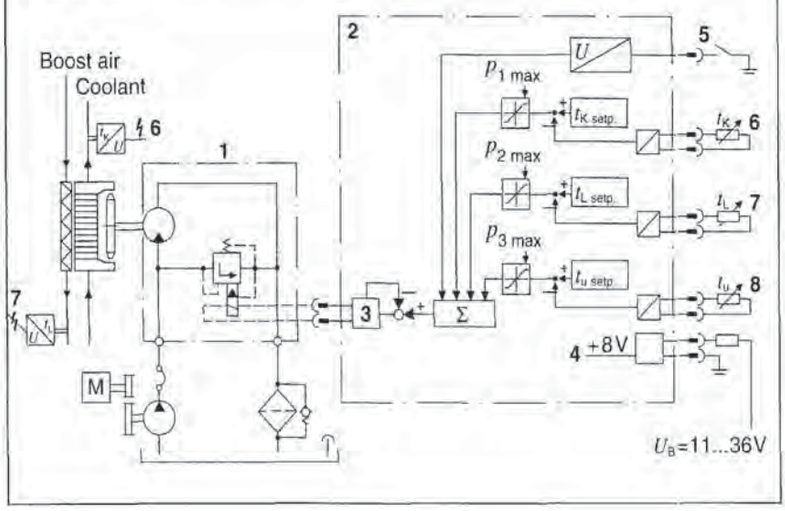
1 Radiator,
2 Fan,
3 Hydraulic motor,
4 Wastegate,
5 Hydrostatic steering,
6 Positioner,
7 Pressure valve,
8 Internal-combustion engine,
9 Coolant flow regulator (thermostat),
10 Coolant pump,
11 Transmission,
12 Hydraulic pump (tandem unit),
13 Hydraulic pump 1 for fan,
14 Hydraulic pump 2,
15 Air filter,
16 Filter,
17 Fluid reservoir.



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pressure. The unit is adjusted by a temperature-sensitive control mechanism with proportional response characteristics. It provides continuous, progressive control of system pressure (in the case of the throttle valve via outgoing bypass flow). The control mechanism can be a thermostatic element (expansion element with wax) located in the coolant stream. Electrohydraulic systems in which the valve is adjusted by a solenoid (proportional solenoid or solenoid with pulse-modulated switching) are becoming increasingly important. This type of solenoid is controlled by the output signal of an electrical temperature sensor located within the coolant stream. This type of unit provides proportional control of the system pressure as a function of temperature. It continually adjusts the fan speed for the required cooling power, remaining within a control range of plus/minus 2.5 °C. The proportion of operating time during which the fan operates at maximum speed is only roughly 5 %, meaning that the fan rotates at reduced speed most of the time. Reductions in fuel consumption and noise emissions are the result. The inherent system losses with this type of slip regulation, a maximum of roughly 15 %, are commensurate with the requirements of economy. In order to ensure adequate ventilation in the engine compartment beyond the base speed – of particular importance with compartments for encapsulated engines or low-noise vehicles – there is also a limit on minimum system pressure. The control electronics can also be expanded to process additional analog (such as internal and external temperature) and digital input signals. These can serve as the basis for generating output signals to the control solenoid, allowing additional adjustment of fan speed. An example is the combined application of the fan drive to regulate coolant, boost-air, and engine-compartment temperatures, and to switch the fan to maximum min^{-1} during retarder operation. Electrohydraulic systems can be integrated within the engine-management system. Hydrostatic fan-drive systems can also supply other systems or combine with additional ancillary drive systems such as clutch, transmission, compressor, water pump, alternator, hydraulic power

Electrohydraulic fan drive
1 Gear motor with proportional pressure valve, 2 Control unit, 3 Current regulator, 4 Voltage regulator, 5 Retarder operation.
Temperature sensors for: 6 Coolant, 7 Boost air, 8 Outside air.
U_B Battery voltage.



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steering, rear-axle steering, hydraulic systems for tippers, etc. By means of suitable system design or multiple pump combinations, function prioritization and safety-related demands can be satisfied.

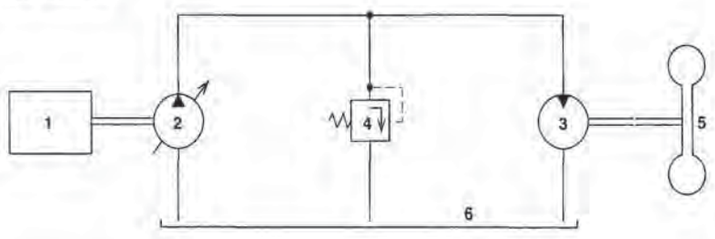
Hydrostatic drives

If the pressure outlet of an adjustable hydraulic pump is connected to a constant-speed or adjustable hydraulic motor (piston or gear motor), the result is an infinitely variable power-transmission device. The mechanical input power (torque × min^{-1}) emerges as mechanical output at another point. The specific conversion ratio is determined by the quotients of the preset pump flow volume and the motor's displacement volume. Parallel layouts incorporating several motors (differential effect) and series layouts (constant velocity) are also possible. However, a basic transmission within an open circuit can neither change direction nor apply braking force without the assistance of auxiliary mechanisms. Nevertheless, this type of layout is well suited to variable ancillary drive systems, such as fans, spreader plates, etc.

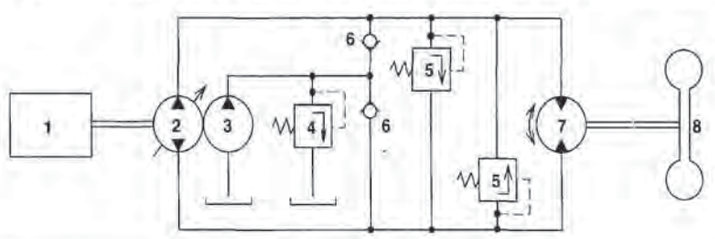
Main drives

Hydrostatic drive systems for automotive application must be able to cope with turning under power and with braking. For this reason, it is the closed circuit which has gained predominance. The main (reversible) pump is combined with a charge pump, which is usually flange-mounted. The flow from the charge pump into the low-pressure line compensates for leakage and losses in compression volume. Because there is always pressure on the low-pressure side, the main pump's maximum permissible speed is higher than in suction operation. At a constant conversion ratio, this type of transmission provides almost the same degree of positive drive as a mechanical unit. It is especially well-suited as a device for powering ma-

Open-circuit hydrostatic drive system
1 IC engine, 2 Adjustable hydraulic pump, 3 Hydraulic motor, 4 Pressure relief valve, 5 Output, 6 Fluid reservoir.



Basic layout for closed circuit
1 IC engine, 2 Adjustable hydraulic pump, 3 Charge pump, 4 Charge-pressure relief valve, 5 High-pressure relief valve, 6 Charge non-return valve, 7 Reversible hydraulic motor, 8 Output.



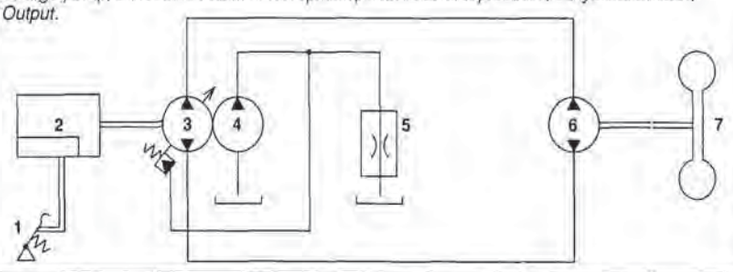
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chinery. Meanwhile, "automotive" controls have been developed to provide drive characteristics similar to those of cars (for industrial trucks, etc.). In these, the vehicle's IC engine and the transmission are governed by a single pedal. Most familiar are the circuits in which the operator uses the pedal to control the engine speed only. The engine power is directed through an auxiliary pump and a throttle circuit (generally incorporating several stages) to generate the control pressure which corresponds to the specific engine speed. This pressure, in turn, determines the main pump's flow volume via a control mechanism with proportional pressure response. This control concept is uncomplicated, and prevents the engine from stalling, as the pump responds to losses in input min^{-1} by switching down to lower flow rates requiring lower torques. However, more complicated circuits are required to satisfy more stringent demands for power control and fuel economy. Electrically adjustable pumps and modern sensor technology provide for elegant solutions using electronics.

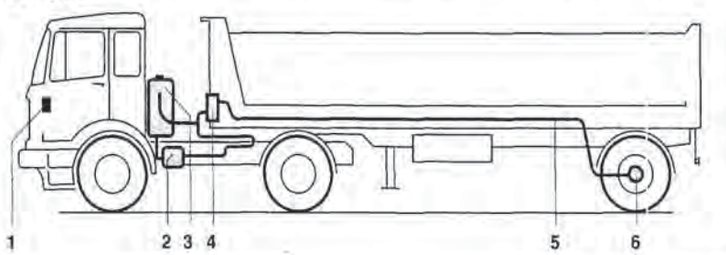
Auxiliary drives

Yet another application for hydrostatic drives is as auxiliary units on otherwise freewheeling truck axles for slow operation in difficult terrain. When required, this unit acts as a hydrostatic substitute for drive shaft and transfer case. For normal road operation, some form of low-loss switchoff is required for the unit. The solution is provided by a constant-displacement pump at the engine; this pump features variable ratios, and can also be disengaged completely. On the auxiliary-drive axle, there are slow-running hub motors whose pistons are retracted by springs when driving on the road. Thus the hydraulic system can be dimensioned for slow vehicle speeds. Neither rotating losses nor substantial friction losses are encountered under normal operating conditions.

Schematic diagram of "automotive" transmission
1 Driving pedal, 2 Engine with speed governor, 3 Hydraulic pump with adjustable pressure, 4 Charge pump, 5 Restrictor network for rpm-dependent control pressure, 6 Hydraulic motor, 7 Output.



Hydrostatic auxiliary drive
1 Electric control unit for normal/auxiliary drive, 2 Disengageable constant-displacement pump with incremental control, 3 Oil reservoir, 4 Disconnectable hose connections, 5 Hydraulic lines, 6 Hub motors



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Automotive pneumatics

Pneumatic systems are used in motor vehicles as sources of energy for
- Opening, closing and locking doors, hatches, etc.,
- Operation and control of brake systems (braking systems) P. 674, and
- Level control P. 613.

Operation of bus doors

Bus doors are operated by double-acting control cylinders. The action of the piston is transmitted to the door leaf. Alternating pressurization and depressurization of the two cylinder chambers opens and closes the door. Three drive systems are in use:
- The working cylinder's piston rod is connected to a lever attached to the door pivot. The door pivot is attached to the door. The extended piston rod responds to cylinder actuation by rotating the spindle to open the door.
- The piston cylinder is axially flange-mounted to the door spindle. The reciprocating motion of the cylinder piston is converted into rotary motion in the door spindle. This rotating motion opens and closes the flange-mounted door by pivoting it outward.
- The control cylinder (rotary drive) is actually a combined cylinder and rotary spindle. The piston stroke is transmitted to the door in the form of a rotating motion at the flange axis.

The doors should reach their travel limits without violent impacts and sudden stops. In order to damp the movement of the door during opening and closing, a pressure- or travel-sensitive cushioning device can be installed in the cylinder to reduce the speed of the door shortly before it reaches either end position. The end-position cushioning effect can be adjusted using a throttle screw.

To cite an example, a 4/2-way solenoid valve can be used to reverse the door's travel as follows. The driver triggers a current pulse by pushing the door-control button, causing the solenoid armature to move the rocker into the opposite position via a rod. The rocker responds by closing the supply valve and opening the discharge on one end of the cylinder, while the discharge is opened and the supply closed on the other end. Additional valve and control functions are specified in Germany by the requirements of the vehicle manufacturers as well as the safety requirements contained in § 35 e of the StVZO Road Licensing Regulations (FMVSS/CUR), the ZH1/494 guidelines for power windows, doors and hatches defined by the professional trade association, and the guidelines of the Association of Public Transport Services (VÖV) for standard city buses.

Power to closing doors must be deactivated or its direction reversed in response to any travel resistance (in Germany, VÖV standard city bus). The opening force must be limited to 150 N or be interrupted upon encountering resistance. After the emergency valve has been operated, the opening or closing force of the door must be canceled, so that the door can be operated by hand. After the emergency valve has been returned to its normal position, door movement must not begin until a separate pushbutton (located on the driver's console or in a box above the door) is pressed or adequate failsafe conditions have been satisfied. Abrupt door movements must be prevented.

Systems in urban buses

The doors in urban buses rely on any of several methods for travel reversal. These include pressure devices in the weatherstripping, differential-pressure switches, photocells, flexible doorshafts and potentiometers. Should a passenger get caught in the door the system responds by generating an electrical switching signal to reverse the door valve. During opening the system can react by depressurizing the control mechanism or, alternatively, by pressurizing both door-control cylinders simultaneously.

In buses with more than two doors, the rear third door must be controlled automatically. The driver only releases the

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door for operation. Door opening, open time and closing are electronically controlled on the basis of driver and passenger information.

It is often desirable for the front half of the front door to open while the other half remains closed. This function is achieved by a 2/2-way solenoid valve located in the closing line of the cylinder for the second door section.

Systems in touring coaches

Where the emergency valve is located upstream of the door valve (see diagram), the door-control button activates a solenoid to release the pushrod for the return mechanism.

When this power operation is triggered, violent closing is prevented by a flow restrictor. In normal operation the flow restrictor is kept open by secondary pressure.

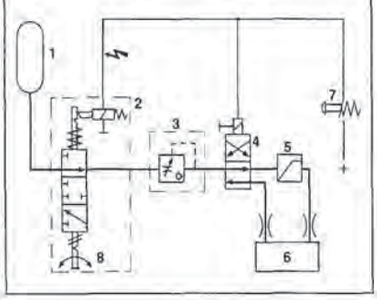
If the emergency valve is installed downstream of the door valve in the line leading to the closing chamber of the door cylinder, the emergency-valve lock is pneumatically actuated. Once a mean pressure is reached, the control rod is pneumatically released and the valve reverses position.

Door and hatch locking

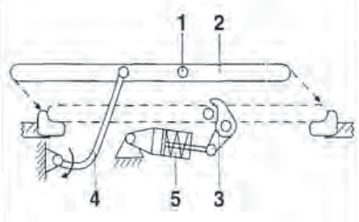
In large tour-bus doors which swing outward, it is essential that the doors remain locked during the journey. This is done either by lifting the door immediately after the closing process or by additional locking devices, with single-acting actuating cylinders installed in the door frame. At the conclusion of the door-closing process, these are activated by the door itself, for instance by way of a 3/2 shuttle control valve, and thus assist the door drive at the end of the closing process. This closure assistance and locking system is designed in such a way that it unlocks in the event of pressure loss. The door is then held only by the door lock which can be opened by hand in an emergency.

In contrast, springs engage the locking mechanisms for luggage-bay hatches when the control system is depressurized.

Door operating system (circuit diagram)
1 Air tank, 2 Emergency valve with solenoid release mechanism, 3 Flow restrictor, 4 Door valve, 5 Pressure-relief valve, 6 Control cylinder, 7 Push button, 8 Electric switch for acoustic signal.



Closing and locking device for touring coaches
1 Locking bolt, 2 Swinging door, 3 Safety catch, 4 Rocker arm, 5 Actuating cylinder.



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Symbols used in vehicle electrical systems

Standards: DIN 40900

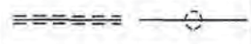
General symbols (selection from circuit diagrams on P. 839 ff.)

Connections

Conductor; conductor crossover, without/with connection

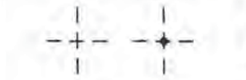


Conductor; conductor crossover, without/with connection



Mechanical linkage; electrical conductor (laid at later stage)

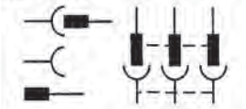
Crossovers (without/with connections)



Connection, general; separable connection (if indication necessary)



Plug-in connection; socket; connector; triple plug-in connection



Ground (equipment/vehicle ground)



Mechanical Function

Switch positions (basic position: solid line)

Manual actuation, actuation by follower (cam), thermal (bimetallic strip)

Detent; non-automatic/automatic return in direction of arrow (pushbutton)

Actuation, general (mech., pneum., hydraul.); piston actuation

Actuation by rotational speed n, pressure p, quantity Q, time t, temperature $t°$

Variability/adjustability, not intrinsic (external), general

Variability/adjustability, intrinsic, caused by applied physical variable, linear/non-linear

Variability/adjustability, general

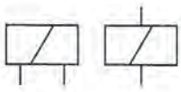
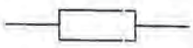
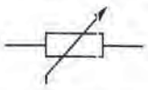
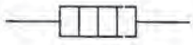
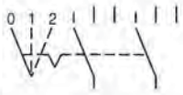
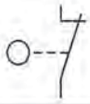
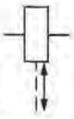
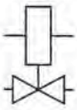
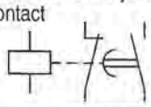
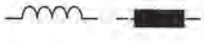
Switches

Momentary switch, NO/NC contact

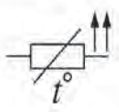
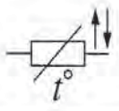
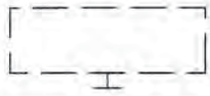
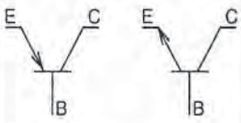
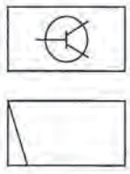
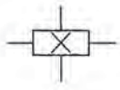
Detent switch, NO/NC contact

Changeover contact, non-bridging/bridging

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Switches	Various components	
Two-way make contact with neutral position (e.g. turn-signal switch) 	Actuators with one winding 	Resistor 
NO/NC contact 	Actuators with two windings acting in same direction 	Potentiometer (with three connections) 
Contact with two makes 	Actuators with two windings acting in opposing directions 	Heating resistor, glow plug, flame plug, heated rear window 
Multiple-position switch 	Electrothermal actuator, thermal relay 	Antenna/aerial 
Cam-operated switch (e.g. contact breaker) 	Electrothermal actuator, solenoid 	Fuse 
Thermostatic switch 	Solenoid valve, closed 	Permanent magnets 
Release/trip device 	Relay (actuator and switch) example: non-delayed-break NC contact and delayed-break NO contact 	Winding, inductive 

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Various components	Devices in motor vehicle	
PTC resistor 	Dotted/dashed line used to delineate or group together associated circuit sections 	Battery 
NTC resistor 	Shielded device, dashed line connected to ground 	Plug-and-socket connection 
Diode, general, current in direction of triangle tip 	Regulator, general 	Light, headlamp 
PNP transistor NPN transistor E=Emitter (arrow points in direction of flow C=Collector, positive B=Base (horizontal), negative 	Electronic control units 	Horn, fanfare horn 
		Heated rear window (general heating resistor) 
Light-emitting diode (LED) 	Indicating instrument, general; voltmeter; clock 	Switch, general; without indicator lamp 
Hall generator 	Rotational-speed indicator; temperature indicator; linear-speed indicator 	Switch, general; with indicator lamp 

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Devices in motor vehicle

Pressure switch



Spark plug



Motor with blower, fan



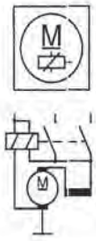
Relay, general



Ignition coil



Starter motor with solenoid switch (without/with internal circuitry)



Solenoid valve, fuel injector, cold-start valve



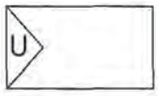
Ignition distributor, general



Thermo-time switch



Voltage regulator



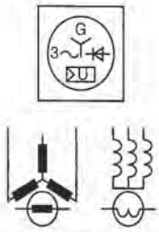
Wiper motor (one/two wiper speeds)



Throttle-valve switch



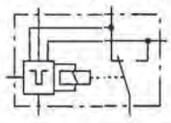
Alternator with regulator (without/with internal circuitry)



Rotary actuator



Intermittent-wiper relay



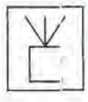
Auxiliary-air valve with electrothermal actuator



Electric fuel pump, motor drive for hydraulic pump



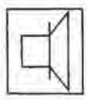
Car radio



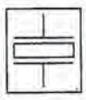
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Devices in motor vehicle

Speaker



Piezoelectric sensor



Linear-speed sensor



Voltage stabilizer



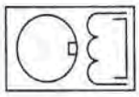
Resistance sensor



ABS rotational-speed sensor



Inductive sensor, controlled with reference mark



Air-flow sensor



Hall sensor



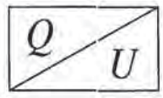
Turn-signal flasher, pulse generator, intermittent relay



Air-mass meter



Converter (rate, voltage)



Lambda oxygen sensor (unheated/heated)





Flow-quantity sensor, fuel-level sensor



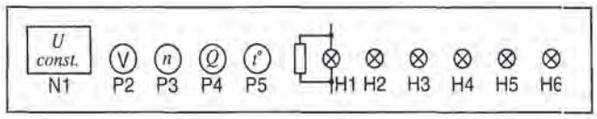
Temperature switch, temperature sensor



Inductive sensor



Instrument-cluster device (dashboard)



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Circuit diagrams

The circuit diagram is a drawn representation of electrical devices by means of symbols, and includes illustrations or simplified design drawings as necessary.

The diagram illustrates the relationship between the various devices and shows how they are connected to each other. The circuit diagram may be supplemented by tables, graphs, or descriptions. The type of circuit diagram actually used is determined by its particular purpose (e.g. illustrating the operation of a system) and by the way in which the circuit is represented.

The circuit diagram must comply with the requirements of the appropriate standards, and deviations are to be explained.

The current paths should be arranged so that signal flow or mechanical action takes place from left to right and/or from top to bottom.

In automotive electrical systems, block diagrams are usually shown with single inputs and outputs, and internal circuitry is omitted.

Schematic diagrams

Assignment

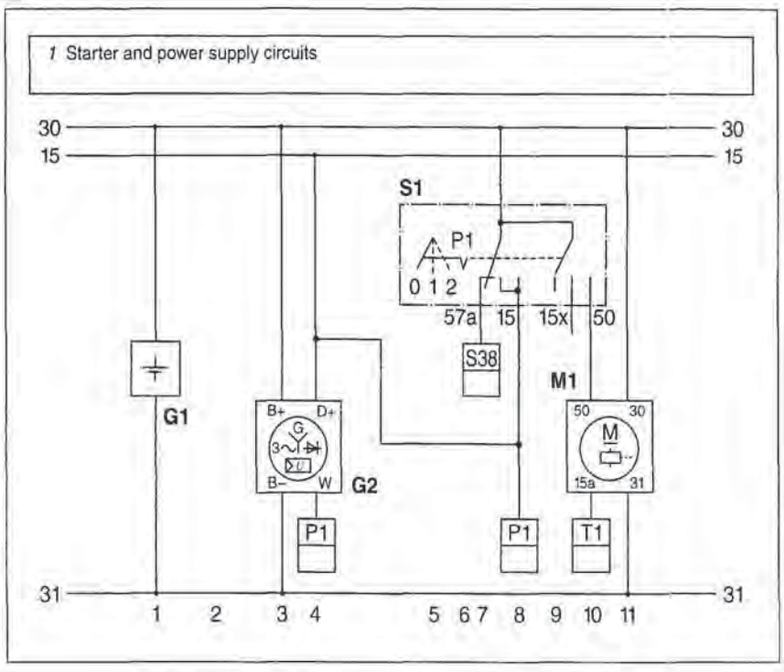
The schematic diagram is the detailed diagram of a circuit. By clearly showing the individual current paths, it explains the operation of an electric circuit. In a schematic diagram, the clear presentation of the circuit's operation, which makes the diagram easy to read, must not be interfered with by the presentation of the individual circuit components and their spatial relationships.

Design

The following schematic diagrams show examples of automotive circuits. They serve only to explain the text and cannot be used as the basis for design or installation.

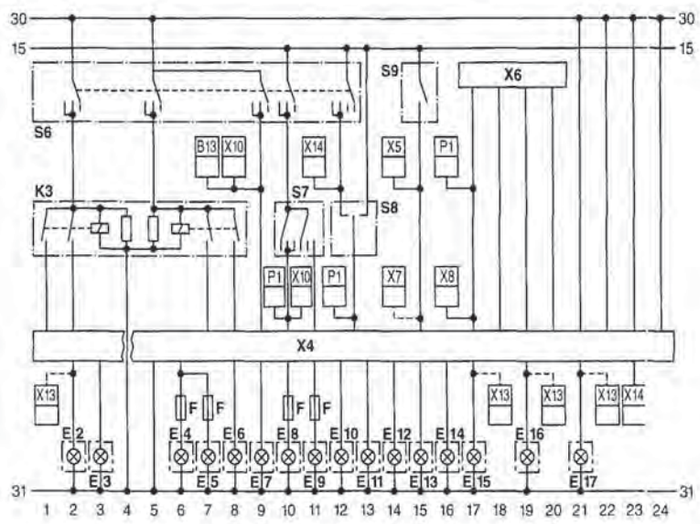
Examples of identification

G1: Equipment identification (DIN 40 719)
15: Terminal identification (D N 72 552)
1: Section identification (DIN 40 719)

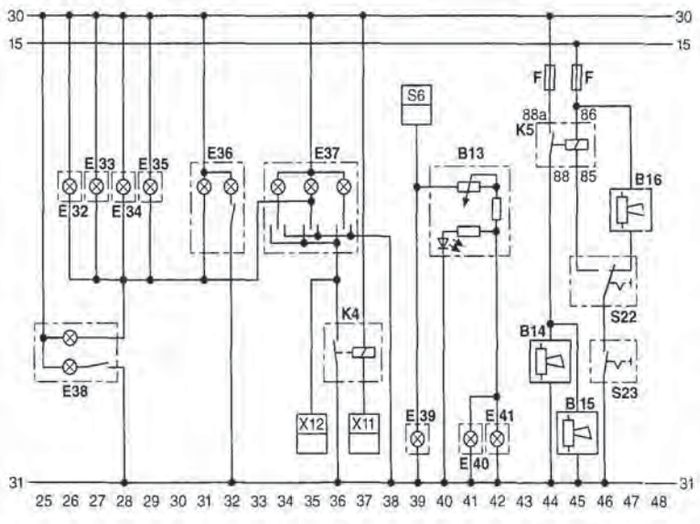


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2 Lighting, signaling system

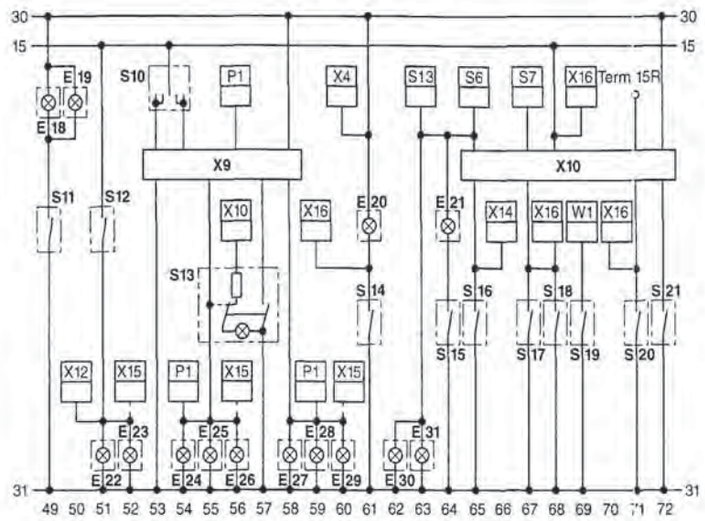


2 Lighting, signaling system, continued

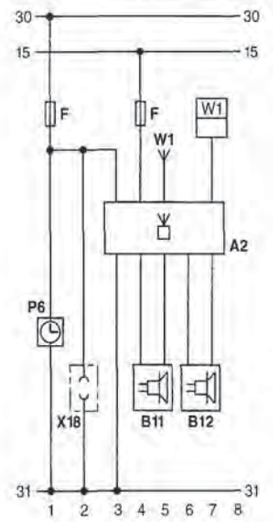


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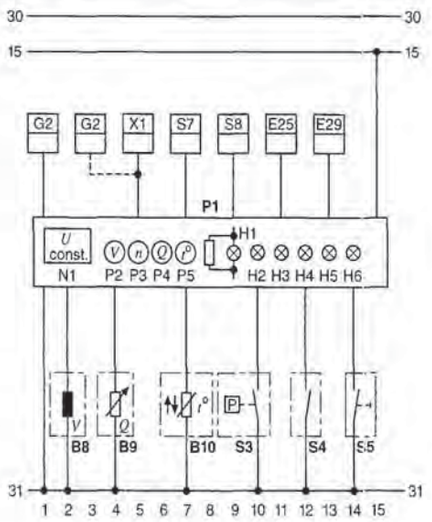
2 Lighting, signaling system, continued



3 Clock, radio

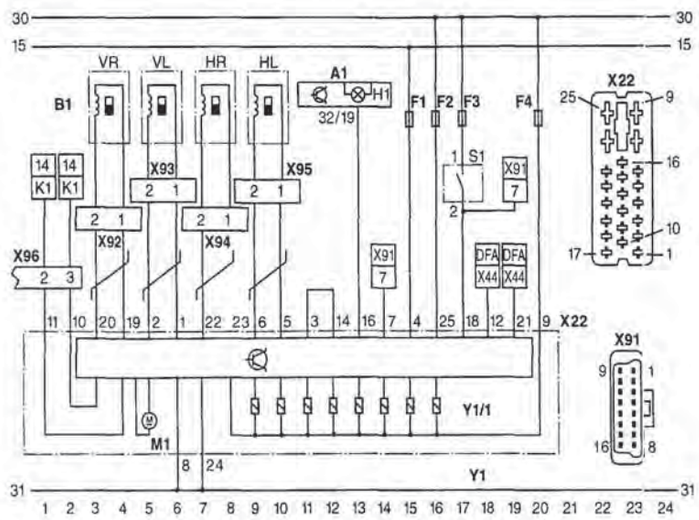


4 Display unit (instrument cluster)

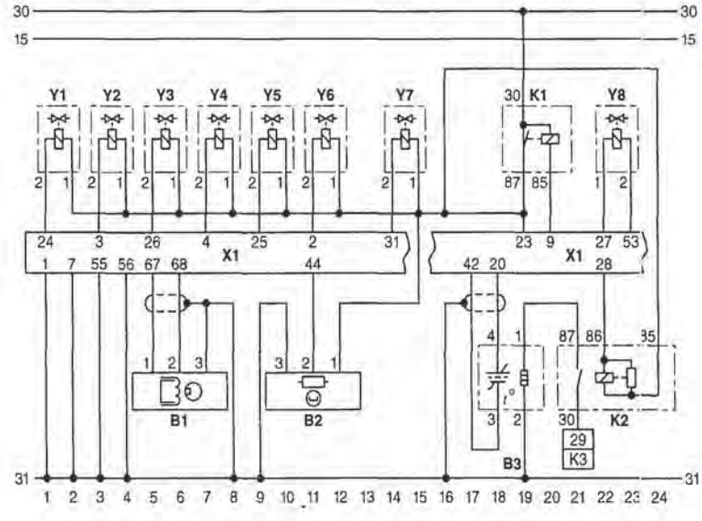


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5 ABS with CAN bus

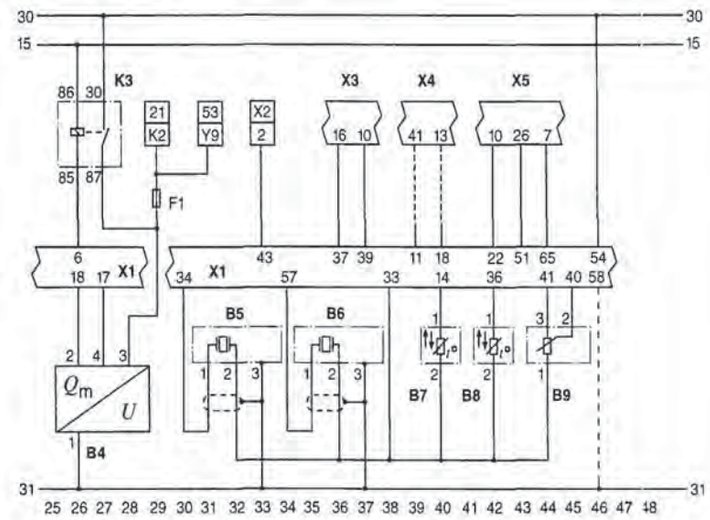


6 M-Motronic

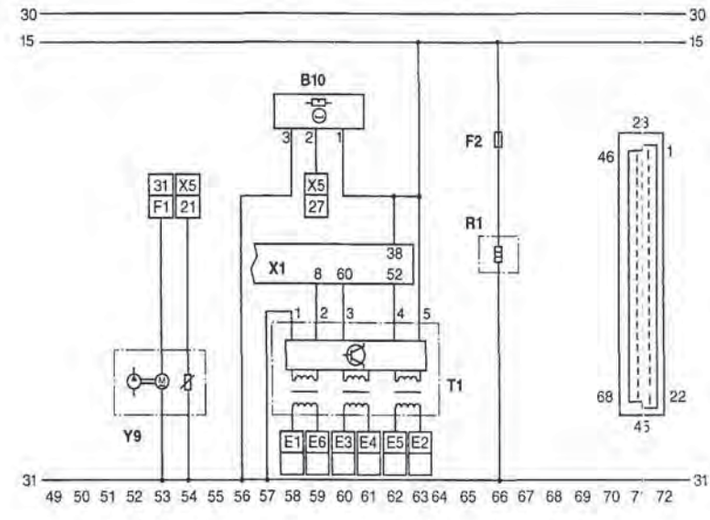


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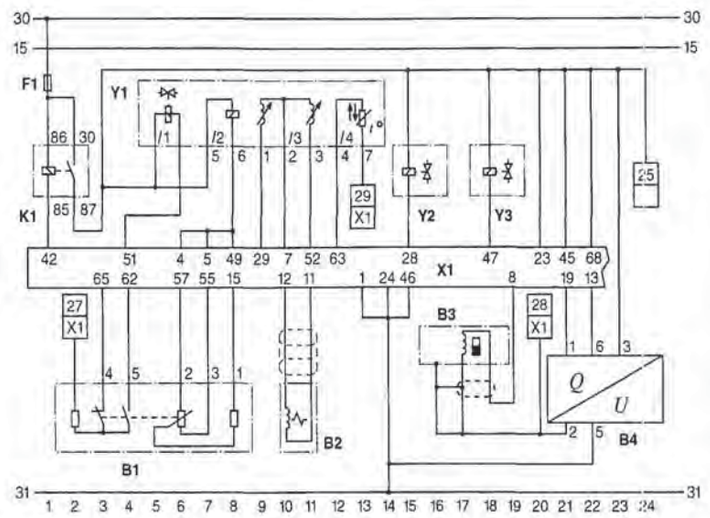
6 M-Motronic, continued



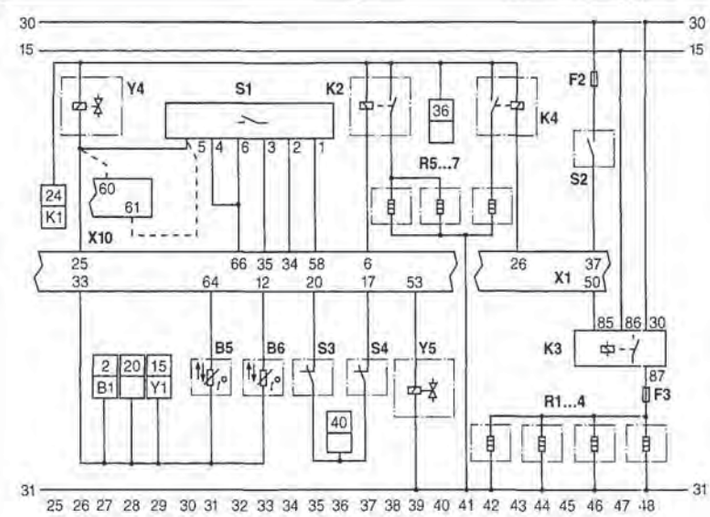
6 M-Motronic, continued



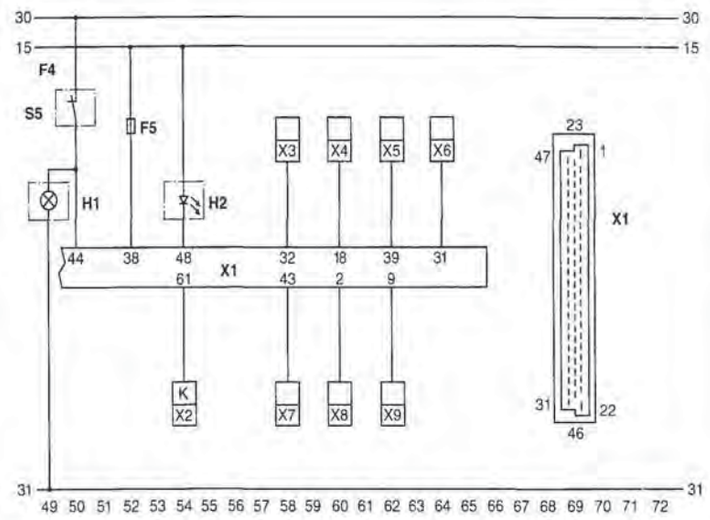
7 Distributor injection pump VE/EDC



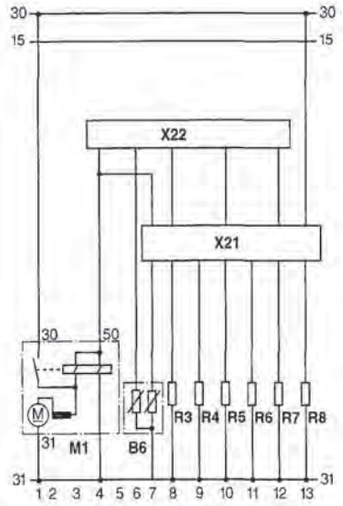
7 Distributor injection pump VE/EDC, continued



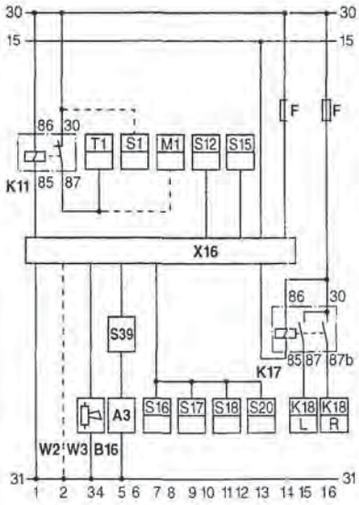
7 Distributor injection pump VE/EDC, continued



8 Start-assist system, diesel engine

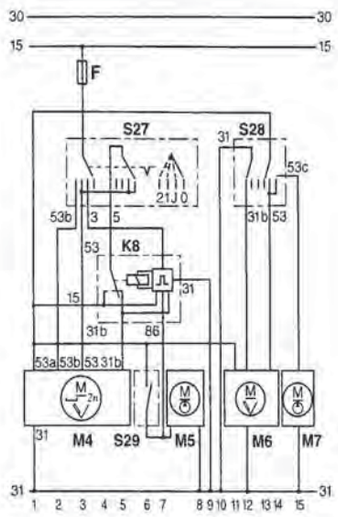


9 Car alarm

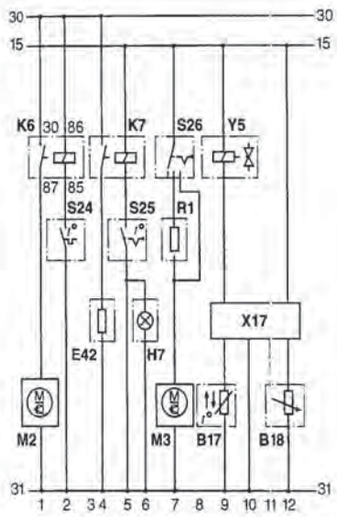


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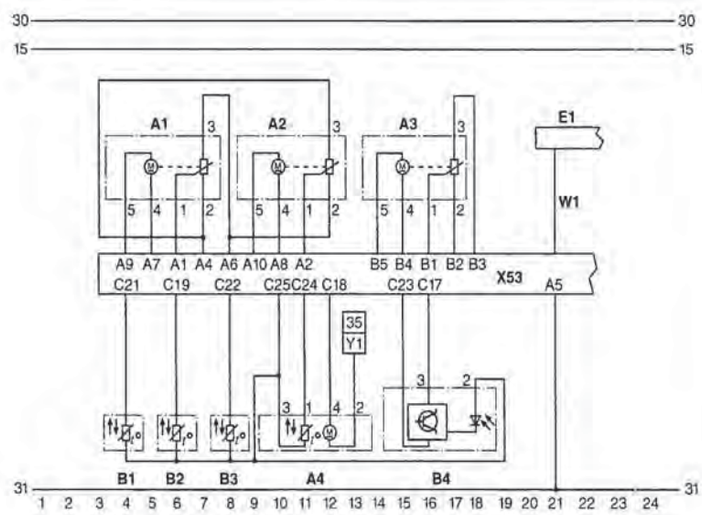
10 Wiper and washer systems



11 Fan, ventilation, heating

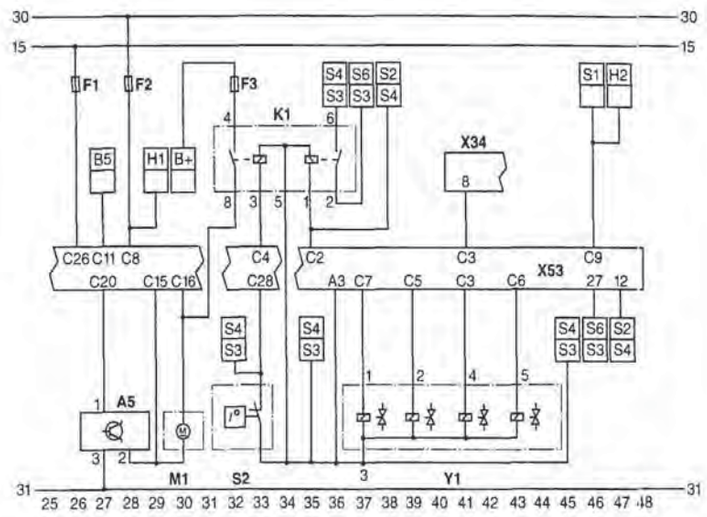


12 Climatronic automatic air-conditioning system

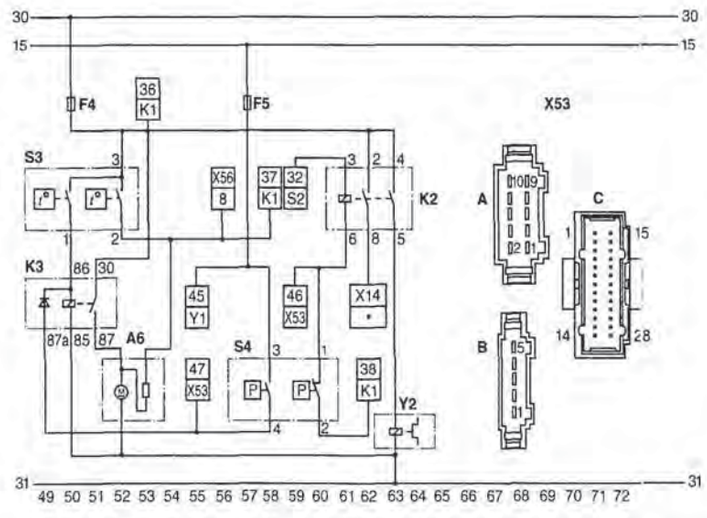


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12 Climatronic automatic air-conditioning system, continued

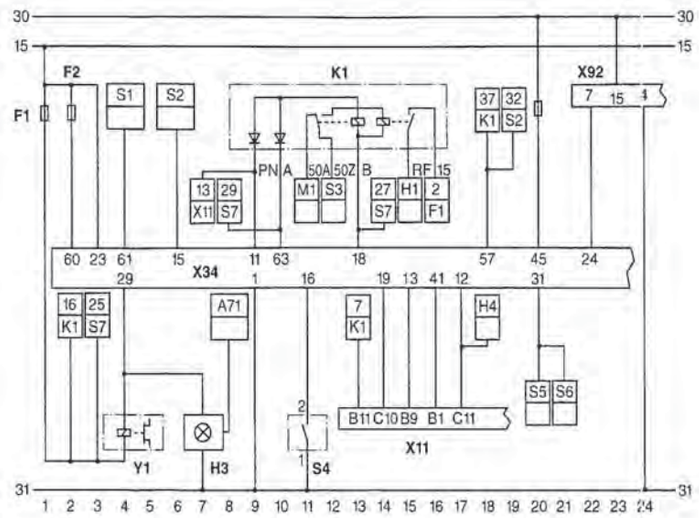


12 Climatronic automatic air-conditioning system, continued

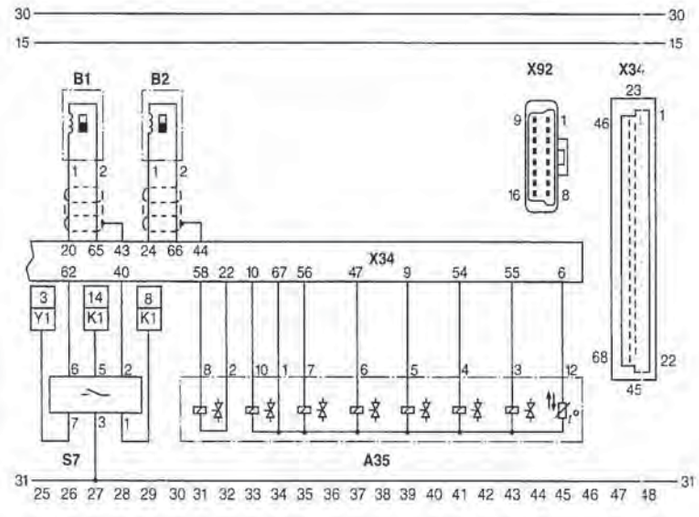


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13 Transmission-shift control system AG 4



13 Transmission-shift control system AG 4, continued



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Section designation and device identification

Table 1 details all section designations as listed in "Schematic diagrams" as from P. 839. The sections identify defined areas of the circuit diagrams which represent a particular system.

Table 2 lists electrical devices and their identification codes along with the number of the schematic-diagram section in which they appear.

Table 1 Sections

Section	System
1	Starting and power supply
2	Lighting and signaling system
3	Clock, radio
4	Display unit (instrument cluster)
5	ABS with CAN bus
6	M Motronic
7	Distributor injection pump VE/EDC
8	Start-assist system, diesel engine
9	Car alarm
10	Wiper and washer systems
11	Fan, ventilation, heating
12	Climatronic automatic air-conditioning system
13	AG4 Transmission-shift control system

Table 2 Device identification

ID Code	Device	Section
A1	Display-unit warning lamp	*5*
A1	Servomotor for center flap	*12*
A2	Radio	*3*
A2	Servomotor for dynamic-pressure flap	*12*
A3	Ignition system with knock control (EZK)	*9*
A3	Servomotor for temperature flap	*12*
A4	Instrument-panel temperature sensor and fan	*12*
A5	Fan control unit	*12*
A6	Radiator fan	*12*
A35	Transmission unit, electrical	*13*
B1	Engine-speed/reference-mark sensor	*5, 6*
B1	Accelerator-pedal sensor	*7*
B1	Coolant-temperature sensor	*12*
B1, 2	Road-speed sensor	*6, 13*
B2	Needle-movement sensor	*7*
B2	Outside-temperature sensor	*12*
B2	Transmission input-shaft speed sensor	*13*
B3	Lambda oxygen sensor	*6*
B3	Engine-speed/reference-mark sensor	*7*
B3	Intake-air temperature sensor	*12*
B4	Air-mass meter	*6, 7*
B4	Photosensor	*12*
B5	Knock sensor 1	*6*
B5, 7	Coolant-temperature sensor	*6, 7*
B6	Knock sensor 2	*6*
B6	Fuel-temperature sensor	*7, 8*
B8	Linear-speed sensor	*4*
B8	Intake-air temperature sensor	*6*
B9	Fuel-level sensor	*4*
B9	Throttle-valve potentiometer	*6*
B10	Coolant-temperature sensor	*4*
B10	Cylinder identification sensor	*6*
B11, 12	Speaker	*3*
B13	Dimmer for instrument lighting	*2*
B14, 15	Supertone horn	*2*
B16	Standard horn	*2, 9*
B17	Interior-temperature sensor	*11*
B18	Setting adjuster	*11*
E1	Climatronic display unit	*12*
E2, 3	Fog warning lamp, L/R	*2*
E4, 5	Driving lamp, L/R	*2*
E6, 7	Fog lamp, L/R	*2*
E8, 9	Low-beam headlamp, L/R	*2*
E10, 11	Side-marker lamp, L/R	*2*
E12, 13	License-plate lamp, L/R	*2*
E14, 17	Stop lamp L/R	*2*
E15, 16	Tail lamp L/R	*2*
E18	Trunk-lid lamp, L/R	*2*
E19	Trunk lamp	*2*
E20	Glove-box lamp	*2*
E21	Engine-compartment lamp	*2*
E22, 23	Backup lamp, L/R	*2*
E24, 26	Turn-signal lamp VL/HL	*2*
E25, 28	Auxiliary turn-signal lamp, L/R	*2*
E27, 29	Turn-signal lamp VR/HR	*2*
E30, 31	Ashtray lamp, front/rear	*2*
E32, 33	Footwell lamp, left front/rear	*2*
E34, 35	Footwell lamp, right front/rear	*2*
E36, 38	Rear reading lamp, R/L	*2*
E37	Front reading lamp	*2*
E39	Vanity-mirror lamp	*2*
E40	Instrument lighting	*2*
E41	Instrument-panel lighting	*2*
E42	Rear-window heating	*11*
F..	Fuses	

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ID Code	Device	Section
G1	Battery	1
G2	Alternator	1
H1	Charge-indicator lamp	4
H1	ABS-Warning lamp	5
H1	Indicator lamp	7
H2	Oil-pressure warning lamp	4
H2	Brake light	7
H3	Handbrake warning lamp	4
H3	Selector-lever illumination	13
H4	Brake-wear warning light	4
H5	High-beam warning lamp	4
H6	Turn-signal warning lamp	4
H7	Warning lamp for rear-window heating	11
K1	Main relay	6, 7
K1	Air-conditioning relay	12
K1	Starter locking relay	13
K2	Relay for oxygen-sensor heater	6
K2	Relay for small heater filament	7
K2	Air-conditioning compressor relay	12
K3	Side-marker lamp scanning relay	2
K3	Electric fuel pump relay	6
K3	Glow-plug relay	7
K3	Radiator-fan run-on relay	12
K4	Interior-light control relay	2
K4	Heater-filament relay	7
K5	Supertone-horn relay	2
K6	Engine-fan relay	11
K7	Rear-window heating relay	11
K8	Intermittent-wiper relay	10
K11	Starter locking/immobiliser relay	9
K17	Visual alarm relay	9
M1	Starter	1, 8
M1	Pump motor, hydraulic modulator	5
M1, 3	Ventilation-fan motor	11, 12
M2	Cooling-fan motor	11
M4	Wiper motor	10
M5	Windshield-washer motor	10
M6	Engine-fan relay	10
M7	Rear-window washer motor	10
N1	Voltage stabilizer	4
P1	Instrument cluster	4
P2	Electric speedometer	4
P3	Rev-meter	4
P4	Fuel gauge	4
P5	Engine-temperature display	4
P6	Clock	3
R1	Heater resistor	6
R1..4	Glow plugs	7
R1	Blower resistor	11
R5..7	Auxiliary heating (manual gearbox)	7

ID Code	Device	Section
R3..8	Glow plugs	8
S1	Ignition and starting switch	1
S1	Brake-light switch	5
S1	Operating controls for cruise control	7
S1	Lights switch	12
S2	Air-conditioning switch	7
S2	Evaporator temperature switch	12
S3	Oil-pressure switch	4
S3	Brake-pedal switch	7
S3	Radiator-fan temperature switch	12
S4	Handbrake switch	4
S4	Clutch-pedal switch	7
S4	Air-conditioning pressure switch	12
S4	Kick-down switch	13
S5	Brake-wear detector contact	4
S5	Brake-light switch	7
S6	Lights switch	2
S7	Fog-lamp switch	2
S7	Multi-function switch	13
S8	Dipswitch	2
S9	Brake-light switch	2
S10	Turn-signal switch	2
S11	Switch for tailgate lights	2
S12	Switch for backup lamps	2
S13	Hazard-warning and turn-signal switch	2
S14	Switch for glove-box lamp	2
S15	Switch for engine-compartment lamp	2
S16...18	Door position switch, front left, rear right, rear left	2
S19	Impact switch	2
S20	Door position switch, front right	2
S21	Door-handle switch	2
S22	Horn changeover switch	2
S23	Horn button	2
S24	Thermo-switch	11
S25	Rear-window heating switch	11
S26	Fan switch	11
S27	Wiper switch	10
S28	Rear-window wiper/washer switch	10
S29	Washer switch	10
S39	Alarm-system encoding switch	9
T1	Ignition coil	6
W1	Car radio antenna	3
W1	Connector for 16-core ribbon cable	12
W2, 3	Encoding wire	9
X1	ECU connector for Motronic/VE/EDC	6, 7

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ID Code	Device	Section
X3	ECU connector for air conditioning	6
X4	Plug, lamp-control module	2
X4	ECU connector for transmission-shift control	6
X5	Connecting plug for instrument cluster	6
X6	Plug, check-control	2
X9	Plug-in base, hazard-warning relay	2
X10	Connector for central body-electronics basic module	2, 7
X11	ECU connector for engine management	13
X16	ECU connector for alarm system	9
X17	ECU connector for air conditioning/heater control	11
X18	Diagnosis socket	3
X21	Connector for glow control unit	8
X22	ECU plug ABS/ABD[1])	5
X22	Diagnosis socket	8
X34	ECU connector for transmission-shift control	12, 13
X44	Navigation-system connector	5
X53	Connector for automatic air-conditioning system	12
X91, 92	Diagnosis socket	5, 13
Y1	Hydraulic pressure modulator	5
Y1	Fuel injector 1	6, 7
Y1	Valve block	12
Y1	Disabling solenoid	13
Y2	Air-conditioning output control	7
Y2	Air-conditioning compressor solenoid-operated coupling	12
Y2..5	Injectors 2..5	6, 7
Y5	Hot-water valve	11
Y6	Fuel injector 6	6
Y7	Canister-purge valve	6
Y8	Idle actuator	6
Y9	Electric fuel pump	6

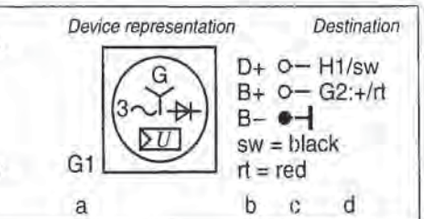
[1]) ABD = Automatic Brake Differential.

Wiring diagram: Detached representation

In the detached method of representation, the continuous connecting lines between the individual devices are omitted. The individual devices are represented by squares, rectangles, circles, symbols or illustrations and designated in accordance with DIN 40719 Part 2. The terminal designation of the device is also indicated. Each outgoing conductor from a device receives a code containing the terminal designation of the destination device as well as the device designation. If necessary, the conductor color is also given.

Example: Alternator

a) Device designation (code letter and code number),
b) Terminal designation on device,
c) Device to ground,
d) Destination, (code letter and code number, terminal designation, conductor color.



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Terminal designation as per DIN 72552

The purpose of the terminal-designation system for automotive electrical systems is to enable easy and correct connection of the conductors to the various devices, particularly in the event of repairs and equipment replacement.

The terminal designations do not identify the conductors, because devices with different terminal designations can be connected at the two ends of each conductor. For this reason, the terminal designations need not be written on the lines which designate conductors.

In addition to the terminal designations listed, designations according to DIN VDE standards may also be used on electrical equipment.

If in the case of multiple-contact connections the number of terminal designations as per DIN 72552 does not suffice, the terminals are consecutively numbered using numbers or letters whose representations of specific functions are not standardized.

Terminal designations: Examples

Term.	Definition
	Ignition coil, ignition distributor
1	Low voltage
	Ignition distributor with two separate circuits
1a	to contact breaker I
1b	to contact breaker II
	Ignition coil, ignition distributor
4	High voltage
	Ignition distributor with two separate circuits
4a	from ignition coil I, terminal 4
4b	from ignition coil II, terminal 4
	Switched (+) downstream of battery (output of ignition/
15	driving switch)
	Output on series resistor to igni-
15a	tion coil and to starter motor
	Glow plug and starter switch
17	Start
19	Preheat
	Battery
30	Input from battery positive (direct)
	Battery changeover 12/24 V
30a	Input from battery II positive
	Return from battery
31	Negative or ground (direct)
	Return to battery
31b	Negative or ground via switch or relay (switched negative)
	Battery changeover relay 12/24 V
31a	Return to battery II negative
31c	Return to battery I negative
	Electric motors
32	Return line[1])
33	Main terminal[1])

Term.	Definition
33a	Self-parking switchoff
33b	Shunt field
33f	For second lower-speed level
33g	For third lower-speed level
33h	For fourth lower-speed level
33L	Counterclockwise rotation
33R	Clockwise rotation
	Starters
45	Separate starter relay, output; starter, input (main current)
	Two-starter parallel operation Starting relay for engagement current
45a	Output, starter I, Input, starter I and II
45b	Output, starter II
48	Terminal on starter and on start-repeating relay for monitoring starting procedure
	Turn-signal flashers (pulse generators)
49	Input
49a	Output
49b	Output, second flasher circuit
49c	Output, third flasher circuit
	Starters
50	Starter control (direct)
	Battery changeover relay
50a	Output for starter control
	Starter control
50b	For parallel operation of two starters with sequential control
	Starting relay for sequential control of engagement current for parallel operation of two starters
50c	Input to starting relay for starter I

[1]) Polarity reversal terminal 32/33 possible

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Term.	Definition
50d	Input to starting relay for starter II
	Start-locking relay
50e	Input
50f	Output
	Start-repeating relay
50g	Input
50h	Output
	Wiper motors
53	Wiper motor, input (+)
53a	Wiper (+), self-parking
53b	Wiper (shunt winding)
53c	Electric windshield-washer pump
53e	Wiper (brake winding)
53i	Wiper motor with permanent magnet and third brush (for higher speed)
	Lighting
55	Fog lamp
56	Headlamp
56a	High beam and high-beam indicator
56b	Low beam
56d	Headlamp-flasher contact
57a	Parking lamp
57L	Parking lamp, left
57R	Parking lamp, right
58	Side-marker lamps, tail lamps, license-plate lamps and instrument-panel lamps
58L	ditto, left
58R	Right, license-plate lamp
	Alternators and voltage regulators
61	Alternator charge-indicator lamp
B+	Battery positive
B–	Battery negative
D+	Dynamo positive
D–	Dynamo negative
DF	Dynamo field
DF1	Dynamo field 1
DF2	Dynamo field 2
U,V,W	Alternator terminals
	Audio systems
75	Radio, cigarette lighter
76	Speakers
	Switches
	NC/changeover switch
81	Input
81a	1st output, NC switch side
81b	2nd output, NC switch side
	NO switch
82	Input
82a	1st output

Term.	Definition
82b	2nd output
82z	1st input
82y	2nd input
	Multi-position switch
83	Input
83a	Output, position 1
83b	Output, position 2
83L	Output, left position
83R	Output, right position
	Current relays
84	Input, actuator and relay contact
84a	Output, actuator
84b	Output, relay contact
	Switching relays
85	Output, actuator (winding end negative or ground)
86	Input, actuator (winding start)
86a	Winding start or 1st winding
86b	Winding tap or 2nd winding
	Relay contact for break (NC) and changeover contacts:
87	Input
87a	1st output (NC switch side)
87b	2nd output
87c	3rd output
87z	1st input
87y	2nd input
87x	3rd input
	Relay contact for make (NO) contact:
88	Input
	Relay contact for make (NO) and changeover contacts (make side):
88a	1st output
88b	2nd output
88c	3rd output
	Relay contact for make (NO) contact
88z	1st input
88y	2nd input
88x	3rd input
	Direction indicators (turn-signal flashers)
C	Indicator lamp 1
C0	Main terminal for monitoring circuits separate from flasher
C2	Indicator lamp 2
C3	Indicator lamp 3 (e. g. when towing two trailers)
L	Turn-signal lamps, left
R	Turn-signal lamps, right

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Assembled-representation diagrams

For trouble-shooting on complex and extensively networked systems with self-diagnosis function, Bosch has developed system-specific schematic diagrams. For other systems on a wide variety of vehicles, Bosch provides the assembled-representation diagrams on a "P" CD-ROM. They are fully integrated in ESI, the Electronic Service Information system from Bosch. This provides automotive repair shops with a useful tool for locating faults or connecting up retrofit equipment. An assembled-representation diagram is shown opposite.

In contrast with other schematic diagrams, the assembled-representation diagrams use American symbols that are supplemented by additional descriptions (see below). Those include component codes such as "A28" (Anti-theft system) that are explained in Table 1, and wire color codes (explained in Table 2). Both those tables are available on the "P" CD-ROM.

Additional descriptions on assembled-representation diagrams
1 *Wire color code,* **2** *Connector number,* **3** *Pin number (a dotted line between pins indicates that all pins are in the same connector).*

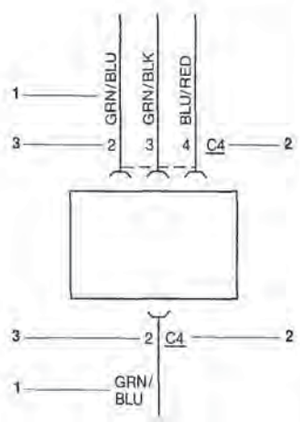


Table 1 Explanation of component codes

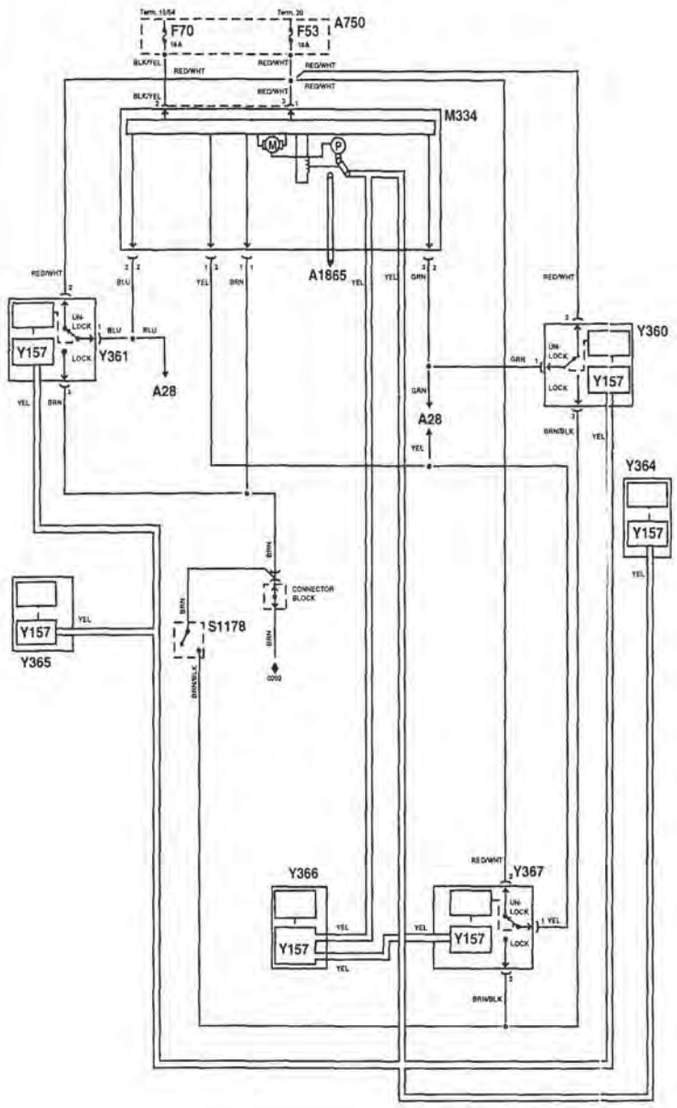
Code	Description
A1865	Electrically adjustable seat system
A28	Anti-theft system
A750	Fuse/relay box
F53	Fuse C
F70	Fuse A
M334	Fuel-supply pump
S1178	Warning-buzzer switch
Y157	Vacuum actuator
Y360	Actuator, right front door
Y361	Actuator, left front door
Y364	Actuator, right rear door
Y365	Actuator, left rear door
Y366	Actuator, fuel filler flap
Y367	Actuator, trunk lid/tailgate

Table 2 Explanation of wire color codes

Code	Description
BLK	Black
BLU	Blue
BRN	Brown
CLR	Clear
DKBLU	Dark blue
DKGRN	Dark green
GRN	Green
GRY	Gray
LTBLU	Light blue
LTGRN	Light green
NCA	Color not known
ORG	Orange
PNK	Pink
PPL	Purple
RED	Red
TAN	Tan
VIO	Violet
WHT	White
YEL	Yellow

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Assembled-representation diagram of a door locking system *(example)*



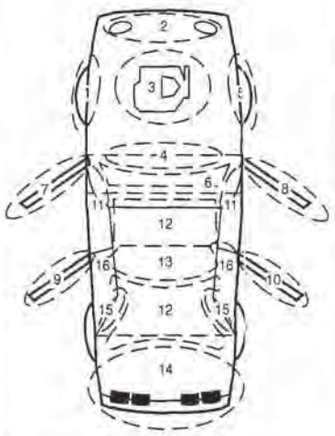
The assembled-representation diagrams are subdivided into system circuits and, if applicable, subsystems (see Table 3). As with other systems in ESI, system circuits are also allocated to one of the four vehicle subassemblies:
– engine,
– body,
– wheels and suspension, and
– drivetrain.

Table 3 System circuits

1	Engine management
2	Starting/charging
3	Air conditioning/heating (HVAC)
4	Radiator fan
5	ABS
6	Cruise control
7	Power windows
8	Central locking system
9	Instrument panel
10	Washer/wiper system
11	Headlamps
12	Exterior lights
13	Electricity supply
14	Grounding
15	Data line
16	Lockout
17	Theft-deterrence
18	Passive-safety systems
19	Power antenna
20	Warning system
21	Heated rear-window/mirrors
22	Supplementary safety systems
23	Interior lighting
24	Power steering
25	Mirror adjuster
26	Soft-top controls
27	Horn
28	Trunk, tailgate
29	Seat adjustment
30	Electronic damping
31	Cigarette lighter, socket
32	Navigation
33	Transmission
34	Active body components
35	Vibration damping
36	Mobile phone
37	Radio/Hi-Fi
38	Immobilizer

It is important to be aware of the grounding points, particularly when fitting additional accessories. For that reason, the "P" CD-ROM includes the vehicle-specific location diagram for the grounding points (see below) in addition to the assembled-representation circuit diagrams.

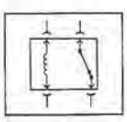
Grounding points
1 *Left front wing,* **2** *Front end,* **3** *Engine,* **4** *Bulkhead,* **5** *Right front wing,* **6** *Footwell bulkhead/dashboard,* **7** *Left front door,* **8** *Right front door,* **9** *Left rear door,* **10** *Right rear door,* **11** *A pillars,* **12** *Passenger compartment,* **13** *Roof,* **14** *Rear end,* **15** *C pillars,* **16** *B pillars.*

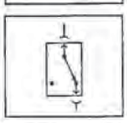


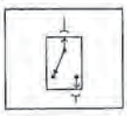
The assembled-representation diagrams use American diagram symbols which are different from the DIN/IEC symbols. A selection of the American diagram symbols is shown on the opposite page.

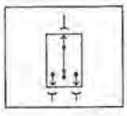
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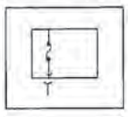
Selection of American diagram symbols

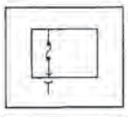

Relay

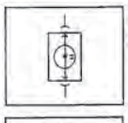

Normally closed contact

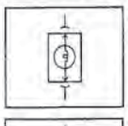

Normally open contact

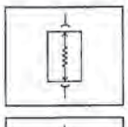

Two-way switch with zero position

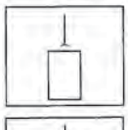

Fuse

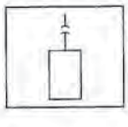

Power fuse

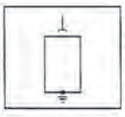

LED (light-emitting diode)


Incandescent lamp (bulb)


Resistor

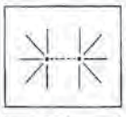

Plug, screw or soldered connection

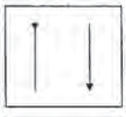

Component with fixed wiring harness

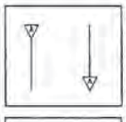

Connection box with ground terminal

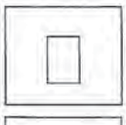

Ground cable


Line connection

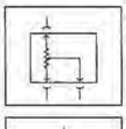

Dotted line: indicates a single connecting point

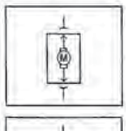

Circuit is continued in another circuit diagram

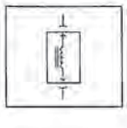

Circuit is continued at another point. Identical letters indicate the connecting point


The entire component is illustrated


Only the part of the component relevant to the system is illustrated


Potentiometer


Motor


Coil

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Calculation of conductor sizes

Units and symbols

Symbol		Unit
A	Conductor cross section	mm²
I	Electric current	A
l	Conductor length	m
P	Power required by load	W
R	Resistance (ohmic) (loads)	Ω
S	Current density of conductor	A/mm²
U_N	Nominal voltage	V
U_{vl}	Permissible voltage drop along insulated wire	V
U_{vg}	Permissible voltage drop across complete circuit	V
ϱ	Resistivity	Ω · mm²/m

Calculations

In determining the conductor cross section, allowance must be made for voltage drop and the effect of elevated temperatures.

1. Determine the current I of the load

   $I = P/U_N = U_N/R$

2. Calculate conductor cross-sectional area A using figures for U_{vl} given in Table 2 (for copper ϱ = 0.0185 Ω · mm²/m).

   $A = I \cdot \varrho\, l/U_{vl}$

3. Round-off the value for A to the next larger conductor cross section in accordance with Table 1.

Individual conductors with a cross-sectional area of less than 1 mm² are not to be recommended due to their inadequate mechanical strength.

4. Calculate the actual voltage drop U_{vl}

   $U_{vl} = I \cdot \varrho\, l/A$ and

5. Check current density S to make sure heat generation is within permissible limits (for short-duty operation $S < 30$ A/mm², see Table 1 for nominal conductor cross-sectional areas and permissible continuous currents for continuous duty).

   $S = I/A$

Table 1. Electrical copper conductors for motor vehicles
Single-core, untinned, PVC-insulated. Permissible working temperature: 70 °C.[2])

Nominal conductor cross-sectional area mm²	Approx. number of individual strands[1])	Maximum impedance per meter[1]) at +20 °C mΩ/m	Maximum conductor diameter[1]) mm	Nominal thickness of insulation[1]) mm	Maximum external diameter of insulated wire[1]) mm	Permissible continuous current (guide figure)[2]) at ambient temperature of + 30 °C A	+ 50 °C A
1	32	18.5	1.5	0.6	2.7	19	13.5
1.5	30	12.7	1.8	0.6	3.0	24	17.0
2.5	50	7.60	2.2	0.7	3.6	32	22.7
4	56	4.71	2.8	0.8	4.4	42	29.8
6	84	3.14	3.4	0.8	5.0	54	38.3
10	80	1.82	4.5	1.0	6.5	73	51.8
16	126	1.16	6.3	1.0	8.3	98	69.6
25	196	0.743	7.8	1.3	10.4	129	91.6
35	276	0.527	9.0	1.3	11.6	158	112
50	396	0.368	10.5	1.5	13.5	198	140
70	360	0.259	12.5	1.5	15.5	245	174
95	475	0.196	14.8	1.6	18.0	292	207
120	608	0.153	16.5	1.6	19.7	344	244

[1]) To DIN ISO 6722, Part 3.
[2]) To DIN VDE 0298, Part 4.

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The values given for U_{vl} in Table 2 are used to calculate the dimensions of the positive conductor. The voltage drop in the ground return is not taken into account. In the case of insulated ground cables, the total length in both directions should normally be used.

The U_{vg} values given in the table are test values and cannot be used for conductor calculations because they also include the contact resistance of switches, fuses, etc.

Notes

1 *In special cases where the starter-motor main lead is very long and the minimum starting temperature is reduced, the U_{vl} figure may possibly be exceeded.*
2 *If the starter-motor main return lead is insulated, the voltage drop along the return lead should not exceed that for the supply lead; the permissible figure is 4 % of the rated voltage in each case and 8 % in total.*
3 *The U_{vl} figures apply to starter solenoid temperatures of between 50 and 80 °C.*
4 *Also make allowance for the lead to the starter switch if necessary.*

Table 2. Recommended max. voltage drop

Type of conductor	Voltage drop across positive conductor U_v		Voltage drop across complete circuit U_{vg}		Remarks
Nominal voltage U_N	12 V	24 V	12 V	24 V	
Lighting cables From light switch terminal 30 to bulbs < 15 W to trailer socket From trailer socket to lights	0.1 V	0.1 V	0.6 V	0.6 V	Current at rated voltage and rated power
From light switch terminal 30 to bulbs > 15 W to trailer socket	0.5 V	0.5 V	0.9 V	0.9 V	
From light switch terminal 30 to headlamps	0.3 V	0.3 V	0.6 V	0.6 V	
Charging cable From 3-phase alternator terminal B + to battery	0.4 V	0.8 V	—	—	Current at rated voltage and rated power
Starter power cable	0.5 V	1.0 V	—	—	Starter short-circuit current at + 20 °C (Notes 1 and 2)
Starter control cable From ignition switch to starter terminal 50 Solenoid switch with single winding	1.4 V	2.0 V	1.7 V	2.5 V	Maximum control current (Notes 3 and 4)
Solenoid switch with engagement and hold winding	1.5 V	2.2 V	1.9 V	2.8 V	
Other control cables From switch to relay, horn, etc.	0.5 V	1.0 V	1.5 V	2.0 V	Current at rated voltage

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Connectors

Function and requirements

Electrical connectors must provide a reliable electrical connection between different system components and thereby ensure safe and reliable operation of the systems concerned under all operating conditions. They are designed to withstand the various stresses they can be expected to be subjected to during the service life of the vehicle. Examples of such stresses are:

- vibration,
- temperature fluctuation,
- extreme temperatures,
- damp and splash-water,
- corrosive fluids and gases, and
- micro-movement between contacts resulting in frictional corrosion.

Such stresses can increase the contact resistance between mating connectors and even cause complete loss of conduction. The insulation resistance can also be reduced, thereby causing short circuits between adjacent conductors.

Electrical connectors therefore must have the following characteristics:

- low contact resistance between current-carrying components,
- high insulation strength between current-carrying components with different electrical potentials,
- high sealing capabilities against water, damp and saline spray.

In addition to their physical properties, connectors must also satisfy other requirements specific to their particular area of application such as:

- straightforward and foolproof connection/disconnection in vehicle assembly/servicing situations, non-reversibility,
- secure and perceptible locking action, straightforward release,
- toughness and suitability for automated wiring-harness manufacture and transport.

Types of design

There is a range of design series spanning the variety of applications for Bosch electrical connectors. The various designs incorporate the types of contact specifically suited to the conditions in which they are used. Below are examples of two types of contact and their characteristics.

Bosch micro-contact

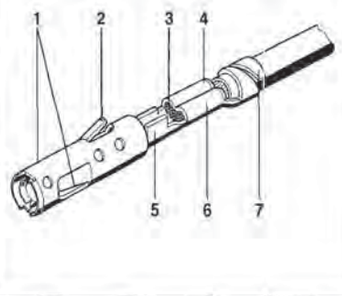
This tin or gold-plated contact which fits a 0.6 mm contact pin has been specially designed for a pin spacing of 2.5 mm, high thermal resistance (155 °C) and high resistance to vibration. It is suited to use in multi-pin connectors because it allows very compact connector dimensions.

The contact consists of two parts. One performs the conducting function and the other (steel spring sleeve) provides the contact pressure (normal contact force).

The steel spring sleeve maintains the contact pressure even at high temperatures and for the complete service life of the vehicle. The higher forces that are then necessarily required to connect and disconnect the connector are reduced by a special connector-locating facility. This also ensures the connector is precisely aligned so that the contacts or pins cannot be damaged by twisting or bending.

The complete connector is sealed against the male connector on the ECU concerned by a circumferential radial seal

Bosch micro-contact
1 Steel spring sleeve, 2 Locking spring (primary spring), 3 Single core (multi-strand), 4 Insertion radius, 5 Contact body, 6 Conductor crimp, 7 Insulation crimp.



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in the connector casing. This, together with three sealing lips, ensures a reliable seal against the ECU sealing collar.

The contacts are protected against the ingress of damp along the cable by a flat seal through which the contacts are passed with the wire crimped to them. The flat seal is made of a silicon gel and replaces the conventional individual core seal while at the same time making it possible to substantially reduce connector size and to vary contact usage (different numbers of contacts used).

The flat seal forms a reliable seal on its inner surfaces against the insulation of the wires.

When the connector is assembled, the contact with the wire attached is pushed through the flat seal that is already in place in the connector and the contact slides home into its position in the contact holder. There, it automatically locks into position by virtue of the locking spring. When all contacts are in position, a sliding pin provides a secondary locking mechanism. This is a supplementary locking facility and increases the security of the connector against inadvertent withdrawal of the wire and contact by force.

Bosch sensor/actuator contact

The Bosch sensor/actuator contact is used for 2...7-pin compact connectors which connect engine-compartment components (sensors and actuators) with the ECU. The pin spacing of 5 mm allows for the physical strength required.

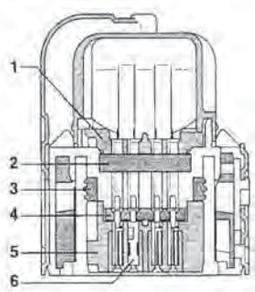
The Bosch sensor/actuator contact has an internal meander-shaped design which reliably prevents vibration transmission from the cable to the contacts. This ensures that there is no relative movement of the contact surfaces that might lead to corrosion.

The compact connector has individual core seals which prevent the ingress of dampness into the contact area. Three sealing lips on the connector casing provide the pressure to ensure there is an adequate seal against splash-water and other sources of damp.

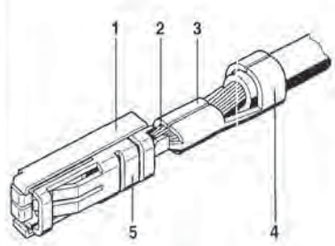
The self-locking snap-on connectors with additional release facility ensure ease of connection and disconnection in vehicle assembly and servicing situations. The connector is released by pressing a point marked by a ribbed surface.

The typical uses for this type of contact include connectors on diesel-engine components (e.g. common-rail pressure sensor, fuel injectors) and gasoline-engine components (e.g. fuel injectors, knock sensor).

Multi-pin connector with micro-contacts *(section)*
1 Pressure plate, 2 Flat seal, 3 Radial seal, 4 Sliding pin (secondary locking facility), 5 Contact holder, 6 Contact.



Bosch sensor/actuator contact
1 Steel spring sleeve, 2 Single core (multi-strand), 3 Conductor crimp, 4 Insulation crimp, 5 Meander.



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Automotive electrical systems

Energy supply

The vehicle must be able to draw upon sufficient reserves of electrical energy to satisfy a number of requirements. The available current supply must be capable of ensuring that the vehicle can be started and operated at all times. Operation of electrical accessories for a reasonable period of time with the engine off should not make subsequent engine starts impossible. Battery, starter, alternator and the electrical system as a whole must be designed for mutually compatible operation. The optimization criteria for the best possible combination are light weight, small volume and low fuel consumption. Low fuel consumption is generally the prime consideration. The following factors must receive special consideration:

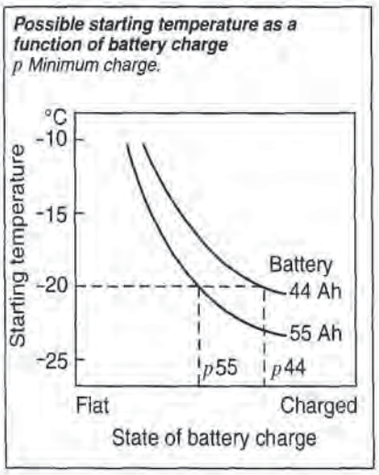
Starting temperature
The lowest temperature at which the engine can be started depends upon a number of factors including the battery (capacity, internal resistance, state of charge, etc.) and the starter (size, with or without intermediate transmission, electrical/permanent-magnet excitation, etc.). If the engine is to be started at a temperature of −20 °C, for example, the battery must have a given minimum state of charge p.

Alternator output
The current output of the alternator varies as a function of engine speed. The alternator can only supply a certain percentage of its rated current at idle n_L. If the current I_V consumed by the loads is greater than the current I_G supplied by the alternator, e. g. when the engine is idling, the battery will discharge. The voltage of the vehicle's electrical system will then drop.

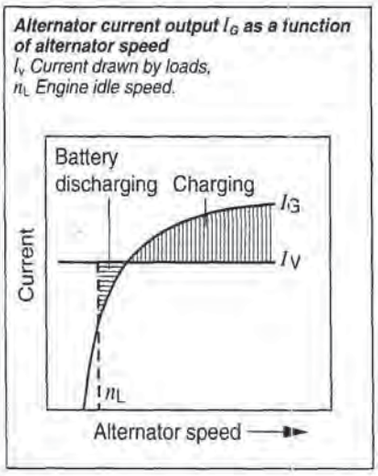
If on the other hand, the current draw I_V is less than the current I_G from the alternator, some of this current difference is used for battery charging I_B.

Driving
The speed at which the alternator turns depends upon the particular type of operation for the vehicle in question. The cumulative speed frequency distribution shows how often a certain speed is achieved or exceeded during vehicle operation.

Owing to traffic congestion or stops at traffic lights, a passenger car driven in commuter traffic is run at idle a high percentage of the time. Under highway driving conditions, on the other hand, the percentage of time the vehicle is run at idle is generally lower. Additional idle time is accumulated by city buses because



Possible starting temperature as a function of battery charge
p Minimum charge.



Alternator current output I_G as a function of alternator speed
I_V Current drawn by loads,
n_L Engine idle speed.

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they must make frequent stops. The state of charge of the battery is further reduced when loads must be left on with the engine turned off (e.g., at bus terminals). Long-distance tour buses generally run only a small percentage of the time at idle.

Electrical loads

Electrical loads have different duty cycles. A differentiation is made between continuous loads (ignition, fuel injection, etc.), long-time loads (lighting, heated rear window, etc.) and short-time loads (turn signals, stop lamps, etc.). Use of some electrical loads is seasonal (air conditioners in summer, seat heaters in winter). The on-time of electrical radiator fans depends upon temperature and driving conditions. In winter, lights are generally used when driving in commuter traffic.

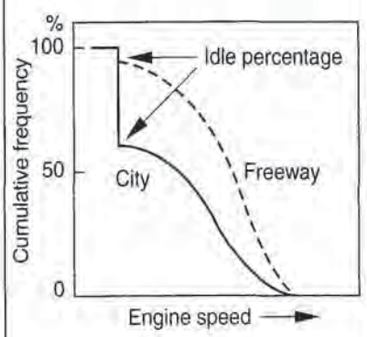
The electrical demands encountered during vehicle operation are not constant. The initial minutes following starting are generally characterized by high demand (for heating or cooling) with a subsequent sharp drop in current draw:

1. In the future, electrically heated catalytic converters will require additional power of 1...4 kW in order to reach operating temperatures of >300 °C within

Equipment power consumption as a function of duty cycle

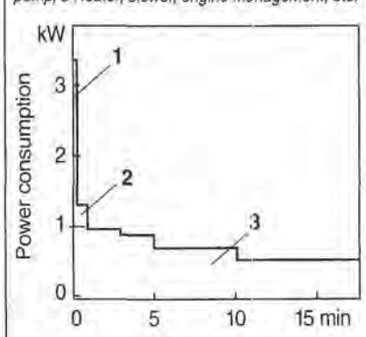
Load	Power consumption	Average power output
Motronic, electric fuel pump	250 W	250 W
Radio	20 W	20 W
Side-marker lamps	8 W	7 W
Low-beam headlamps	110 W	90 W
License-plate lamp, tail lamps	30 W	25 W
Warning lamp, instruments	22 W	20 W
Heated rear window	200 W	60 W
Interior heating, fan	120 W	50 W
Electric radiator fan	120 W	30 W
Windshield wipers	50 W	10 W
Stop lamps	42 W	11 W
Turn-signal lamps	42 W	5 W
Front fog lamps	110 W	20 W
Fog warning lamp	21 W	2 W
Total		
Installed loads	1145 W	
Average power consumed by loads		600 W

Cumulative engine-speed frequency for city and highway driving



Electrical power consumption as a function of driving time

1 Catalytic-converter heater, 2 Secondary-air pump, 3 Heater, blower, engine management, etc.



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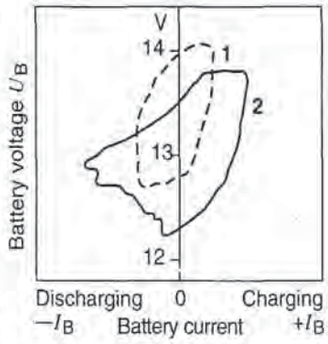
10...30 s after the vehicle is started (actual figures vary according to engine size and emissions-system design).
2. The secondary-air pump responsible for supplying air immediately downstream of the combustion chambers for exhaust-gas afterburning, can operate for up to 200 s after the engine is started.
3. Other electrical equipment such as heaters (including defrosters, etc.), blowers and lighting systems draw current for various periods, while the engine-management system remains in continuous operation.

Charging voltage

The battery charging voltage must be higher in cold weather and lower in warm weather in order to accommodate the chemical processes which take place inside the battery. The gassing voltage curve indicates the maximum permissible charging voltage at which the battery does not "gas".

Vehicle loads require a voltage which is as constant as possible. The voltage applied to lamp bulbs must have very close tolerances so that lamp-bulb service life and light intensity remain within specified limits. The voltage regulator limits the maximum voltage. It affects the lower voltage limit when the potential alternator current I_G is larger than the system's load-current requirement I_V. The voltage regulator is usually mounted on the alternator.

Dynamic system characteristics
(Envelopes for city driving)
1 with large alternator and small battery,
2 with small alternator and large battery.



In the event of significant deviations between the temperature of the voltage regulator and that of the battery electrolyte, it is advantageous to monitor the voltage-regulation temperature directly at the battery. It is possible to compensate for the voltage loss in the charge cable between the alternator and the battery by using a regulator to monitor actual voltage directly at the latter (over a supplementary wire).

Dynamic system characteristics

The interrelationships between the battery, alternator, loads, temperature, engine speed and engine/alternator speed ratio define the system's dynamic response pattern. It is dynamic because it varies along with each combination of parameters and each set of operating conditions. An xy-coordinate graph recorder can be connected to the battery terminals to plot dynamic response patterns.

Charging-balance calculation

The charging-balance calculation must take into consideration the above-mentioned influencing variables. A computer program is used to determine the state of battery charge at the end of a typical driving cycle. A typical passenger-car cycle consists of vehicle operation in commuter traffic (engine speeds are low) combined with winter operation (at which time charging-current consumption by the battery is low). Summer operation may place even greater loads on the system when the vehicle is equipped with an air conditioner (high current draw). The battery's charge at the end of the cycle must – at the very least – be sufficient to allow a subsequent engine start at the temperature in question.

Vehicle electrical system

The voltage level in the vehicle electrical system, and thus the battery charge, are also affected by the wiring between alternator, battery and electrical loads. If all loads are connected on the battery side, the total current in the charging cable is $I_G = I_B + I_V$. The charging voltage is lower owing to the high voltage drop. If all loads are connected on the alternator side, the voltage drop is less and the charging voltage is higher. This could be disadvanta-

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geous to loads (such as electronic circuitry) which are sensitive to voltage peaks or high voltage ripple. Those electrical devices which feature high current draw and relative insensitivity to overvoltage should therefore be connected at the alternator, and voltage-sensitive loads with low current draw should be connected at the battery. Voltage drops can be minimized by suitable conductor cross sections and good connections whose resistance remains low even after a long period.

Future automotive electrical systems

Within the standard 12-volt automotive electrical system, the battery represents a compromise between (to some extent) conflicting requirements: its capacity must be selected with reference to starting requirements as well as on-board current supply. During the engine starting sequence, the battery is subjected to high current loads (300...500 A). The associated voltage dip has an adverse effect on certain electrical consumer units (e.g. units with microcontroller). In contrast, current draw in standard operation – where battery capacity is the salient factor – is minimal. It is not possible to optimize both criteria within a single battery.

Future automotive electrical systems will feature two batteries (separate accumulators for starting and other electrical requirements); the "high power for starting" and "general-purpose electrical supply" functions will be separated. This will make it possible to avoid the voltage collapse that attends the starting process while ensuring reliable cold starts, even with the general-purpose battery discharged to 30 % of its capacity.

Starter battery

The starter battery must supply substantial current, but only for a limited period (during starting). Compact dimensions allow installation in the immediate vicinity of the starter motor with correspondingly short connection cables. Capacity can also be reduced. To ensure consistently high charge factors, this battery features a nominal rating of 10 volts. Because the voltage is lower than the 12 V of the other, general-purpose battery, the starter battery has charging priority.

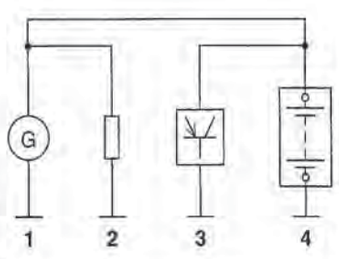
Charging and isolation module

This module isolates the starter battery with starter from the rest of the electrical system during starting and when the engine is off. This arrangement prevents the on-board voltage collapse associated with starting, while also protecting the starter battery against discharge when electrical systems operate with the engine off.

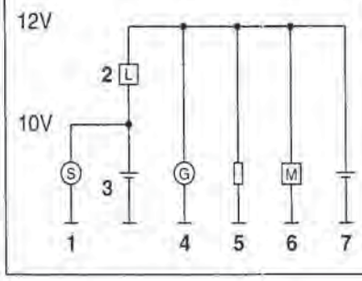
General-purpose battery

This battery is consigned exclusively to supplying the vehicle's electrical system (excluding the starter). It supplies rela-

Vehicle electrical system with loads connected to alternator and battery
1 Alternator, 2 High-draw electrical devices, 3 Low-draw electrical devices, 4 Battery.



Future electrical system
1 Starter, 2 Charging and insulation module, 3 Starter battery, 4 Alternator, 5 Electrical loads, 6 Engine management, 7 General-purpose battery.



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tively low currents (e.g., roughly 20 A for the engine-management system), but is also designed to deal with major cyclical variations; assuming adequate capacity and electrical demand that remains within the discharge range, this unit can store and furnish substantial amounts of energy. Design dimensions are selected based on capacity reserves for electrical equipment (e.g., parking lights, hazard-warning flashers), constant-draw devices and maximum specified discharge level.

Enhanced-performance versions

Progressively more powerful systems are envisaged for vehicles with extremely high power requirements. Among the potential design options could be:

- current supply to the engine-management system is from the starter battery instead of the general-purpose unit, or the system can alternate selectively between the two,
- the starter battery is also a 12 V unit; more complicated design is then needed to ensure that it retains charging priority,
- nominal voltage > 12 V,
- multiple-voltage networks featuring a –12 V (or –24 V) circuit installed parallel to the +12 V supply circuit, providing potential supplementary supplies of 24 V (or 36 V) from the outside conductor sections of the two circuits,
- application of two alternators.

Selection of a particular option will depend upon the precise objectives that the on-board circuit concept is to fulfill (for example, priority on avoiding voltage collapse during starting, weight savings, enhanced starting reliability).

Fuel consumption

A small portion of the fuel used by the vehicle is used to drive the alternator and transport the combined weight of starter, battery, and alternator (approx. 5 % in a medium-sized car).

Average fuel consumption per 100 km: For a weight of 10 kg: approx. 0.1 l; for 100 W of drive power: approx. 0.1 l. Alternators with high levels of efficiency at part load therefore help to reduce fuel consumption despite being slightly heavier.

Starter batteries

Requirements

Modern vehicles are placing increasingly exacting demands on starter batteries: Enhanced cold-starting power (higher currents, especially at low temperatures) is needed with diesel engines, while vehicles equipped with large numbers of sophisticated electrical accessories require large amounts of electrical energy, not only when driving, but also when stationary. Maintaining a positive charging balance is especially difficult with frequent town and commuter driving in winter because these are always accompanied by substantial current consumption.

Within the automotive electrical system, the battery assumes the role of a chemical storage unit for the electrical energy produced by the alternator. The battery must be capable of limited-duration high-current delivery for starting, and it must be able to furnish some or all of the electrical energy for other important system components for limited periods with the engine at idle or off. The lead-acid accumulator battery represents the usual means of meeting these demands. Typical system voltages are 12 V for passenger cars, and 24 V for commercial vehicles (achieved by connecting two 12 V batteries in series).

Batteries must be specifically designed to meet individual system requirements for starting power, capacity and current consumption at temperatures ranging from approx. –30...+70 °C. There are also additional specifications which must be satisfied for particular applications (e.g., maintenance-free batteries, vibration-proof batteries).

Lead-acid batteries

Charging and discharging

The active materials in a lead-acid battery are lead oxide (PbO_2) on the positive plates, spongy, highly porous lead (Pb) on the negative plates and the electrolyte, diluted sulfuric acid (H_2SO_4). The electrolyte is simultaneously an ion conductor for charging and discharging. In juxtaposition with the electrolyte, PbO_2 and Pb

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Density and freezing point of diluted sulfuric acid

State of charge	Battery design	Electrolyte density kg/l[1])	Freezing point °C
Charged	standard	1.28	–68
	tropicalized	1.23	–40
Half charged	standard	1.16/1.20[2])	–17 .. –27
	tropicalized	1.13/1.16[2])	–13 .. –17
Discharged	standard	1.04/1.12[2])	–3 ... –11
	tropicalized	1.03/1.06[2])	–2 ... –8

[1]) At 20 °C; The specific gravity of the electrolyte decreases as the temperature rises, and increases as the temperature drops at a rate of around 0.01 kg/l per 14 K difference in temperature.

[2]) Lower figure: High electrolyte utilization. Higher figure: Low electrolyte utilization.

Element	Charged	Discharged
Positive plate	PbO_2	$PbSO_4$
Electrolyte	$2 x H_2SO_4$	$2 x H_2O$
Negative plate	Pb	$PbSO_4$

adopt typical electrical voltages (individual potentials) the magnitudes of which (disregarding polarity) are equal to the sum of the cell voltages measurable from the outside. At rest, the cell voltage is approx. 2 V, increasing during charging and decreasing when the cell is subjected to a load. When the cell discharges, PbO_2 and Pb combine with H_2SO_4 to form $PbSO_4$ (lead sulfate). This conversion causes the electrolyte to lose SO_4 (sulfate) ions, and its density decreases. When the cell charges, the active materials PbO_2 and Pb are reconstituted from the $PbSO_4$.

If the charging voltage continues to be applied after the cell has reached a state of full charge, only the electrolytic decomposition of water occurs, producing oxygen at the positive plate and hydrogen at the negative plate (oxyhydrogen gas). Electrolyte density can be used as an indication of the state of charge of the battery. The accuracy of this relationship depends upon battery design (see table below showing "acidity levels") as well as electrolyte stratification and battery wear with a certain degree of irreversible sulfating and/or a high degree of shedding of the plate material.

Behavior at low temperatures

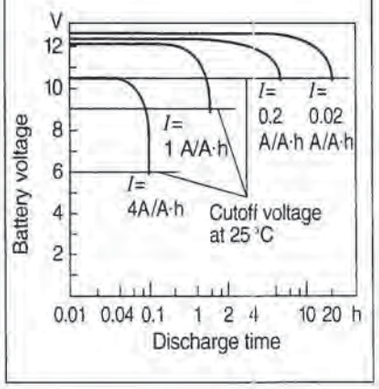
Basically, the chemical reactions in an accumulator battery take place more slowly at lower temperatures. The starting power even of a fully charged battery therefore decreases as the temperature drops. The more the battery discharges, the more diluted the electrolyte becomes. This means that freezing of the electrolyte in a discharged battery is very likely. Such a battery is only capable of supplying a low current which is not sufficient for starting the vehicle.

Battery characteristics

Designation

In addition to mechanical parameters such as dimensions, mounting and terminal design, batteries are primarily characterized by electrical ratings measured

Battery voltage curves as a function of discharge time for various discharge currents



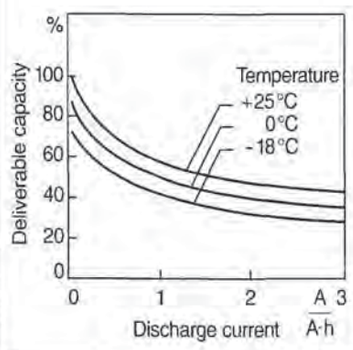
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according to specific testing standards (e.g. DIN EN 60096-1, previously DIN 43 539-2). Starter batteries manufactured in Germany are identified by a 9-digit type number, the rated voltage, the rated capacity and the cold-discharge test current as per DIN EN 60095-1/A11.
Example: 555 059 042 indicates 12 V, 55 A·h, a special type designation (059) and a cold-discharge test current of 420 A. In the future, this European standard will become a more widely used battery-identification convention.

Capacity
Battery capacity, rated in A·h, is the current which can be drawn from the battery under specified conditions. Capacity decreases as discharge current increases and temperature decreases.

Nominal capacity K_{20}
As defined by DIN EN, nominal capacity is the charge which the battery can deliver within 20 h at constant discharge current down to a cutoff voltage of 10.5 V (1.75 V/cell). The battery nominal capacity depends upon the quantities of active material used (positive mass, negative mass, electrolyte), and is relatively unaffected by the number of plates.

Deliverable capacity as a function of discharge current and temperature



Cold-discharge test current I_{CC} (previously I_{KP})
This figure provides an index of the battery's current-supply capability at low temperatures. According to DIN EN standards, the terminal voltage when discharging at I_{CC} and −18 °C must be at least 7.5 V (1.25 V/cell) 10 s after discharge starts. Further details relating to discharge duration are specified in the above standard. The short-time behavior of the battery when discharged at I_{CC} is largely determined by the number of plates, their surface area, and the gap between plates and separator material. Another variable which characterizes starting behavior is internal resistance R_i.

The following equation is true for a fully-charged battery (12V) at −18 °C: $R_i \leq 4000/I_{CC}$ (mΩ), where I_{CC} is given in A. The battery internal resistance and other resistances in the starter circuit determine the cranking speed.

Battery design
The 12-volt automotive battery contains six series-connected, individually partitioned cells in a polypropylene case. Each cell consists of a positive and negative plate set. These sets, in turn, are composed of the plates (lead grid and active mass) and the microporous material (separators) which insulates the plates of opposite polarities. The electrolyte is in the form of diluted sulfuric acid which permeates the pores in the plates and separators, and the voids in the cells. The terminals, the cell connectors, and the plate straps are made of lead; the openings in the partitions for the cellular connectors are tightly sealed. A hot-molding process is employed to permanently bond the one-piece cover to the battery case, providing the battery's upper seal. On conventional batteries, each cell is sealed by its own vent plug, which when removed permits initial battery filling and topping-up during service. When screwed in, the vent plugs allow charge gases to escape. Maintenance-free batteries frequently appear to be completely sealed units, but they also require escape vents.

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Battery designs

Maintenance-free battery

The maintenance-free battery defined in DIN standards features lead-alloy grids with a low antimony content to reduce gas generation and the attendant water losses during charging. This prolongs the electrolyte check intervals to:

- every 15 months or 25,000 km on low-maintenance batteries, and
- every 25 months or 40,000 km for maintenance-free batteries (as per DIN).

There is never any need (and generally no means provided) to check the electrolyte level on the completely maintenance-free (lead-calcium) battery. With the exception of two extremely small vent orifices, this type of battery is completely sealed. As long as the electrical system is operating normally (U = constant), water decomposition is minimal, and the electrolyte reserves above the plates will last for the life of the battery. A lead-calcium battery of this type also offers the advantage of very low self-discharge. This means that it can be stored for several months provided it is fully charged to begin with. When a maintenance-free battery is recharged remote from the vehicle's electrical system (i.e., with battery charger), the charge voltage is never to exceed 2.3...2.4 V per cell; overcharging at constant currents, or the use of chargers with a W charge curve will always lead to water consumption in the lead-acid battery.

Deep-cycle battery

Due to their particular design characteristics (thin plates, lightweight separator materials), starter batteries are poorly suited for applications in which frequent high levels of discharge occur, as this causes substantial wear at the positive plates (particularly through separation and sedimentation of the active material). In the deep-cycle battery, the separators have fiberglass mats to provide the positive mass with extra support and prevent premature shedding which would otherwise lead to sludge formation. The service life, as measured in charge/discharge cycles, is roughly twice as long as that of a standard battery. Deep-cycle batteries featuring pocket separators and felt layers have an even longer service life.

Vibration-proof battery

In the vibration-proof battery, an anchor of cast resin and/or plastic prevents the

Maintenance-free automotive battery
1 One-piece cover,
2 Terminal-post cover,
3 Cell connector,
4 Terminal post,
5 Frit,
6 Plate strap,
7 Case,
8 Bottom mounting rail,
9 Positive plates, inserted into envelope separators,
10 Negative plates.

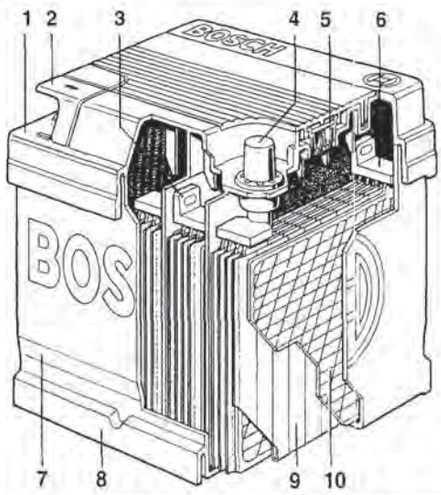


plate stacks from moving relative to the battery case. According to DIN requirements, this type of battery must be capable of withstanding a 20-hour sinusoidal vibration test (frequency 22 Hz) and a maximum acceleration of 6 $\cdot g$. In other words, the test is roughly 10 times as severe as for a standard battery. Vibration-proof batteries are used primarily in commercial vehicles, construction machinery, tractors, etc. and are identified by the code "Rf".

Heavy-duty battery
The heavy-duty battery combines the attributes of the deep-cycle and vibration-proof units. It is used in commercial vehicles which feature extreme vibration levels, and in which cyclic discharge patterns are commonplace. Designation: "HD".

"Kt" battery
The Kt (or "S") battery shares the basic design of the deep-cycle device, compared to which it has thicker plates, but fewer of them. Although no cold-discharge current is specified for the "Kt" battery, its starting power lies well below (35...40 %) that of comparably sized standard units. This battery is used in applications characterized by extreme cyclic variations, e.g., as a drive battery (P. 576, "Lead-acid batteries").

Operating states

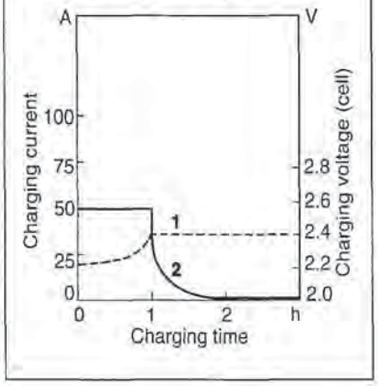
Charging
Voltage limiting is applied when the battery is charged in the vehicle. This corresponds to the IU charging pattern, in which the charge current drops automatically in response to increasing battery voltage. The IU method prevents harmful overcharging and ensures a long battery service life.

In contrast, many service-station and home chargers still operate with constant current or with a W-shaped charging curve (see diagram of "W" charging curve). In either case, there is little or no reduction in charge current once the battery is fully charged, resulting in substantial water consumption and corrosion at the positive grid.

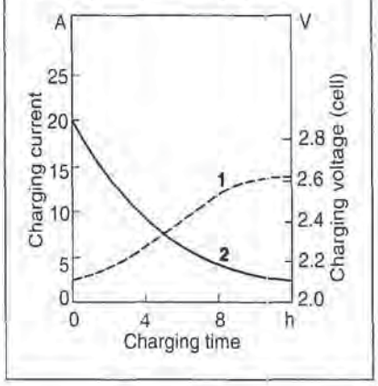
Discharge
Soon after the start of the discharge process, the battery's voltage drops by a specific increment; it then remains relatively consistent, falling slowly under continuing load. The ultimate voltage collapse occurs only just before complete discharge, upon exhaustion of one or several active components (positive mass, negative mass, electrolyte).

Self-discharge (see also "Battery maintenance" below)
Batteries discharge themselves continually, even when no loads (electrical de-

Battery charging: IU-curve
1 Charging voltage, 2 Charging current.



Battery charging: W-curve
1 Charging voltage, 2 Charging current.



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vices) are attached. At room temperature, a modern low-antimony battery will lose about 0.1...0.2% of its total charge each day. As the battery ages, the rate can increase to as much as 1% per day as a result of antimony transfer to the negative plate, and various other sources of contamination. This phenomenon can ultimately lead to battery failure. Rule of thumb regarding the influence of temperature: The rate of self-discharge doubles with every 10 °C increase in temperature.

The self-discharge rate of the lead-calcium battery is substantially lower (by a factor of 1:5), and it remains virtually constant throughout the life of the battery.

Battery maintenance

On low-maintenance batteries, the electrolyte level should be inspected in accordance with the manufacturer's instructions; when indicated, it should be replenished to the MAX mark with distilled or demineralized water. To minimize self-discharge, the battery should be kept clean and dry. An additional pre-winter examination of the electrolyte's specific gravity – or, should this be impossible, of its open-circuit voltage – is also advisable. The battery should be recharged when the specific gravity is below 1.20 g/ml, or the open-circuit voltage is under 12.2 V. Terminals, terminal clamps and installation clamps should be coated with acid-protection grease.

Batteries temporarily removed from service should be stored in a cool, dry place. The electrolyte's specific gravity and/or the open-circuit voltage should be checked every 3-4 months. The battery should be recharged whenever the figures drop below 1.20 g/ml or 12.2 V. Low-maintenance and maintenance-free batteries are best recharged with the IU method (see "Charging") at a maximum voltage of 14.4 V. This method allows adequate charging times in the order of 24 hours without any attendant risk of overcharging. If a constant-current or W-curve battery charger is used, the current (in A) should be reduced to max. 1/10 of the nominal capacity when gassing is seen to start. That is, for example, 6.6 A for a 66 Ah battery. The battery charger should be switched off about 1 hour afterwards. Ventilate the charging area (explosive gas formation, risk of explosion, no naked flames or sparks).

Battery malfunctions

Battery failures which are traceable to internal faults (such as short circuits accompanying separator wear or loss of active mass, broken connections between cells and plates) can rarely be rectified by repair. The battery has to be replaced. Internal short circuits are indicated by major variations in the specific-gravity readings between cells (difference between max. and min > 0.03 g/ml). It is frequently possible to charge and discharge a battery with defective cell connectors provided that the currents remain small, but attempts to start the engine will result immediately in total voltage collapse, even if the battery is fully charged.

If no defects can be found in a battery which consistently loses its charge (indication: low specific gravity in all cells, no starting power) or is overcharged (indication: high water loss), this suggests a malfunction in the vehicle's electrical system (alternator faulty, electrical equipment remains on when the engine is switched off due to faulty relays for instance, voltage regulator set too high or too low, or regulator completely inoperative). When a battery remains severely discharged for a relatively long period, the $PbSO_4$ crystals in the active mass become coarse, making the battery more difficult to recharge. Such a battery should be recharged by applying a minimal charge current (approx. 1/40 of the rated capacity in A) for roughly 50 hours.

Safety precautions

Handling

Before fitting a new starter battery, the manufacturer's instructions should be carefully read in order to avoid safety hazards resulting from incorrect handling. The possible dangers originate from the sulfuric acid in the battery and the explosive gas (mixture of hydrogen and oxygen) formed during battery charging. Tipping the battery for extended periods or carelessness during electrolyte checks

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can lead to burns from the sulfuric acid. Extra caution should be taken during charging and when connecting and disconnecting jumper cables immediately after charging due to the danger of the oxyhydrogen gas exploding.

The explosive gas mixture created by the charging process can be ignited by a sufficiently high level of thermal energy. For that reason, naked flames must not be permitted in the vicinity of batteries. Spark formation, e.g. by sudden opening or closing of electrical circuits, should be avoided and the generation of static electricity by transfer (carpets) or by rubbing with woolen or artificial fabrics must also be prevented. For the reasons listed above, battery-charging areas should always be well-ventilated, while protective goggles and gloves should be worn when handling the battery.

In order to prevent sparks when the battery is connected or disconnected, all electrical equipment must be switched off, and the terminals must be connected in the proper sequence. The rules are as follows:

- When installing the battery, always connect the positive cable first, and the negative cable last. When removing the battery, first disconnect the negative, and only then the positive cable (assuming that negative is ground).
- When connecting a charger or an external battery to boost a low battery which remains installed in the vehicle, always start by connecting the positive terminal of the battery being charged to the positive terminal of the external booster. Then connect the negative cable from the external charger or booster battery to an exposed metallic surface on the vehicle, at least 0.5 m away from the battery.
- Always disconnect the cable from the negative terminal before commencing work in the vicinity of the battery or on the vehicle's electrical system. Short circuits (with tools) generate sparks, and can also cause injury.

Testing starter batteries

The specifications and test procedures for standard automotive starter batteries are defined in DIN EN 60095-1. Those tests are suitable for determining and monitoring the quality of new starter batteries but do not claim in any way to fully reproduce the enormous variety of possible demands to which a battery might be subject in practical use.

Battery testers

The purpose of a battery test is to ascertain the starting power, condition, charge level and voltage of a battery. Testers for starter batteries make a distinction between the no-load test and the under-load test.

Under-load test

In the case of the under-load test, the battery is subjected to a current load comparable with that which occurs when starting the engine (up to several 100 A). The voltage dip ΔU while this load is applied, and the voltage rise ΔU during the recovery phase after the load is removed, are criteria for assessing the starting power and the condition of the battery. In order to be able to give a percentage figure for the starting power, the ΔU level measured is compared with the figure specified for a correctly functioning battery. This specified level is dependent on the size of the battery and is preselected by the user by entering the battery capacity or the cold-discharge test current. The charge level is ascertained on the basis of the battery voltage.

The advantage of the under-load test is the reliable detection of even the finest hairline cracks in the battery's cell connectors and lead plates.

The under-load period of around 30 seconds corresponds to the current draw for several starting attempts, and this has a negative effect on the battery charge level.

No-load test

In the no-load test, the battery is subjected for a few seconds only to a low-frequency, square-wave current of between approx. 0.25 A and 2 A. This load superimposes an alternating current in the millivolt range on the battery direct current. An assessment of the starting power and the condition of the battery can then be made on the basis of the signal amplitude

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and signal wave-form of this alternating current. In order to assess the battery condition and starting power using the no-load test, the cold-discharge test current of the battery being tested must be entered in the tester.

The advantages of the no-load test are minimal discharge of the battery being tested and the speed of the test.

Battery chargers

Charging curves

The most commonly used charging pattern is the "W" curve. The chargers in such cases are generally non-regulated. Due to the battery's and the charger's internal resistances, these chargers respond to increasing battery voltage by steadily lowering the charge current (charging times 12...24 hours).

Because these W-curve chargers do not incorporate any means for limiting the charging voltage, they are only conditionally suitable for use on maintenance-free batteries. For these, chargers should be used which operate according to the IU, IWU, or WU pattern.

With the IU curve, the lead-acid battery (2.4 V per cell) is supplied with constant charging current (to protect the charger against overload) until gassing commences. The charging voltage is then held constant and the charging current reduced sharply (to protect the battery against overcharging). Provided that the initial charging current is high enough, recharging times (to 80 % of full charge) of < 5 hours can be achieved with IU chargers.

Special variations of both the IU and the W charging patterns are available (e.g., Wa, WoW, IUW etc.). These can be applied in combination to meet individual demands for charging time, terminal voltage and freedom from maintenance.

Charging current and voltage settings

On chargers with controlled charging patterns (e.g. IU curve), a regulator continually monitors the momentary actual charging current and voltage levels (and also the ambient temperature if necessary). It compares the monitored (actual-value) data with battery-specific nominal values, and uses a final-control element to reduce the deviation to 0. This type of unit also compensates for fluctuations in its mains input voltage which could otherwise lead to variations in charging current. This has a beneficial effect on, among other things, the battery life and the maintenance intervals.

Charging current

In normal charging ($I_L = 1 \cdot I_{10}$) the battery is supplied with current corresponding to approximately 10 % of its capacity in A·h. Several hours are required to charge the battery completely. Boost charging ($I_L = 5 \cdot I_5$) can be used to bring an empty battery back up to about 80 % of its rated capacity with no damage. Once the gassing voltage is reached, the charging current must either be switched off (e.g., Wa curve) or reduced to a lower level (e.g., IU curve). These current-switching functions are controlled by an adjustable charge limiter or an automatic switchoff device. By the use of special electronic monitoring circuits, the fully charged point can also be determined on the basis of battery-specific voltage patterns in conjunction with the charging period (the battery voltage starts to drop again if it is overcharged).

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Alternators

Electrical-energy generation

The alternator must furnish the vehicle's electrical system with a sufficient supply of current under all operating conditions in order to ensure that the state of charge in the energy storage device (battery) is consistently maintained at an adequate level. The object is to achieve balanced charging, i.e., the curves for performance and speed-frequency response must be selected to ensure that the amount of current generated by the alternator under actual operating conditions is at least equal to the consumption of all electrical equipment within the same period.

The alternator actually produces alternating current. The vehicle's electrical system, on the other hand, requires direct current to recharge the battery and operate the electrical equipment. It is thus direct current that must ultimately be supplied to the electrical system.

The essential requirements are:

- Maintenance of a direct-current supply to all electrical equipment in the system,
- Supplementary charging reserves for (re)charging the battery, even with a constant load from electrical devices in continuous operation,
- Maintenance of a constant alternator voltage throughout the entire ranges of engine speeds and load conditions,
- Robust design capable of withstanding externally-imposed stresses such as vibration, high ambient temperatures, pronounced temperature variations, dirt, moisture, etc.,
- Low weight, compact dimensions and long service life,
- Minimal operating noise,
- High level of efficiency.

Design factors

Rotational speed

The alternator's operating efficiency (its power-to-weight ratio expressed as the ratio of energy generated to component mass in kg) increases as a function of rotational speed. This factor alone would dictate as high a conversion ratio as possible between the engine's crankshaft and the alternator.

The following factors though must also be considered:

- Increasing centrifugal forces at high alternator speeds,
- Alternator and fan noise,
- Efficiency fall-away at high speeds,
- Effect of high speeds on the service lives of the wear components (bearings, collector rings, carbon brushes),
- Mass inertia of the alternator relative to the crankshaft and the resulting load on the belt drive.

Typical conversion factors for automotive applications lie within a range of 1 : 2 to 1 : 3, with ratios of up to 1 : 5 being used in large commercial vehicles.

Engine idle represents roughly a third of the usage pattern of an alternator.

Temperatures

The losses that accompany energy conversion in any machine lead to high component temperatures.

High alternator temperatures are also a result of heat radiation from engine components and ancillaries (such as exhaust systems and turbochargers); the amount of heat depends upon the relative installation positions, and is greatest when the engine operates at high speeds and under high load factors. The supply of cooling air is generally drawn in from the engine compartment.

In view of the trend towards engine-compartment encapsulation which is becoming more and more common as a means of reducing noise, the ducting of fresh air towards the alternator is an appropriate method of reducing component temperatures. For situations where engine-compartment temperatures are extreme, there are liquid-cooled alternators available.

External influences

Installing the alternator on an IC engine means exposing it to extreme mechanical stresses. Depending upon the installation configuration and the engine's vibration characteristics, vibration levels of 500...800 $m \cdot s^{-2}$ can occur at the alternator. This subjects the alternator's mountings and components to extreme forces,

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and it is essential that critical natural frequencies in the alternator assembly are avoided.

Further detrimental influences include spray water, dirt, oil and fuel mist, and road salt in winter. These factors expose all components to the risk of corrosion. It is important that tracking between conducting parts be avoided, as electrolysis could otherwise lead to the early failure of vital operating components.

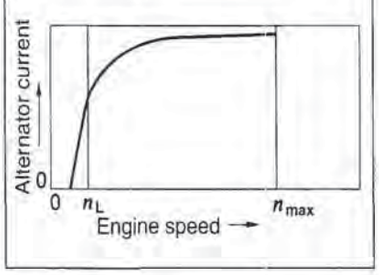
Characteristics and operation

In order to maintain an adequate charge in 12 V, 24 V or 36 V batteries, automotive alternators are designed to supply charge voltages of 14 V, 28 V for heavy commercial vehicles and in future also 42 V.

As direct current is required for charging the battery, a diode rectifier must be provided to convert the alternator's three-phase alternating current into direct current. The diodes also prevent the battery from discharging when the vehicle is stationary.

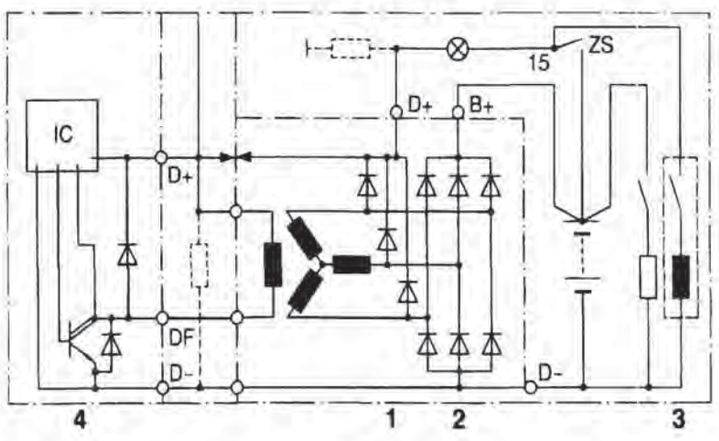
The current-generation curve has a sharp bend. There is no current production until after the so-called "0-ampere speed" is exceeded. At high speeds, the effect of the reverse magnetization field generated by the load current prevents the curve from climbing further. This characteristic means that even excessive loading cannot result in further increases in current flow, thereby protecting the alternator from the thermal damage associated with electrical overload.

Characteristic curve at maximum alternator current
Constant voltage.
n_L Idle speed,
n_{max} Maximum speed.



Alternators are self-excited 12- or 16-pole synchronous devices. The AC winding is wound in the stator's slots, while the excitation winding is housed within the rotor. The DC excitation current required by the excitation winding is conducted to the rotating rotor through sliding contacts (carbon brushes). The current produced

Alternator with standard regulator *(circuit diagram)*
1 Exciter diodes, 2 Power diodes (alternatively: Zener power diodes),
3 Vehicle electrical system, 4 Regulator.



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in the AC winding is channeled in two directions: Most of it flows through the positive diodes of the main rectifier bridge and into the vehicle's electrical system, from where it returns through the negative diodes.

Depending on the design of the alternator and regulator, the excitation current flows
a) through excitation diodes in the case of standard regulators, or
b) directly from the B+ terminal in the case of multifunction regulators.

With standard regulators, some of the current generated acts as excitation current, flowing through the three exciter diodes to terminal D+, through the regulator and collector rings to the rotating field winding and from there, back through the three negative diodes of the main rectifier.

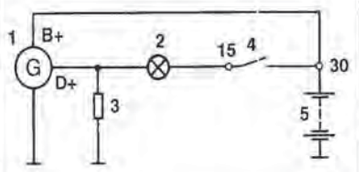
On alternators with multifunction regulators, there are no excitation diodes and the excitation current branches off immediately beyond the main rectifier. The voltage regulator switches on the excitation current only when the engine is started (which it detects by means of alternator rotation), thereby preventing battery discharge when the engine is not running.

The output of the alternator is adjusted to suit the demand from the vehicle's electrical system by varying the excitation current. The regulator is controlled by pulse-width modulation on the basis of constant terminal voltage.

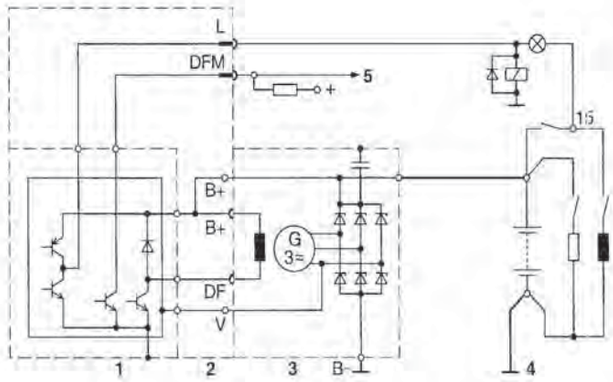
The D+ terminal assumes several functions: Firstly, the alternator is pre-excited from battery terminal B+ via the alternator indicator lamp and terminal D+. Secondly, terminal D+ is at a similar voltage level to B+ after excitation of the alternator. Specific consumer groups can be supplied with power via a relay.

The pre-excitation current determines the "on-line" speed at which initial excitation occurs when the engine is started.

Circuit with fault indication in the event of break in excitation circuit
1 Alternator, 2 Charge-indicator lamp, 3 Resistor R, 4 Ignition and starting switch, 5 Battery.



Alternator with multifunction regulator
Supply connections L, DFM.
1 Regulator IC, 2 Housing, 3 Alternator, 4 Vehicle electrical system, 5 Evaluation/Monitoring.



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This speed is well above the "zero-ampere speed", with the precise figure being largely determined by the power of the indicator lamp.

The charge indicator lamp should come on when the ignition is switched on prior to starting (monitoring function) and go out as the engine starts to run-up.

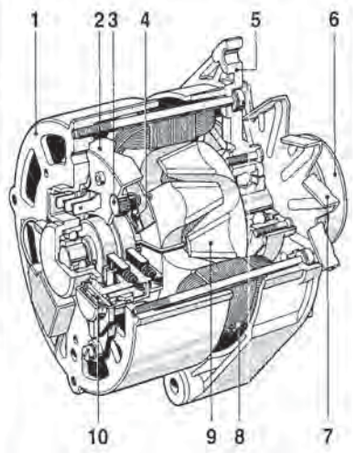
Design variations

Claw-pole alternators

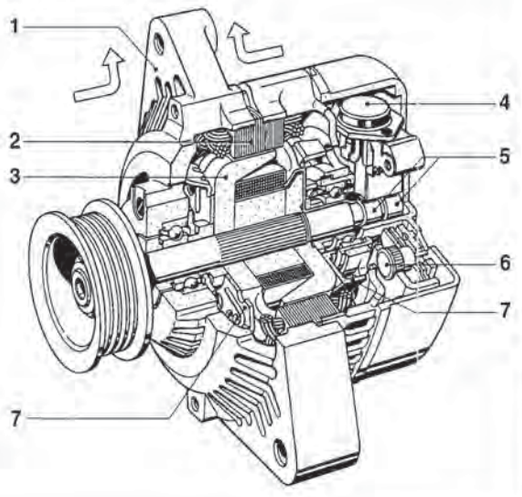
The already very familiar mechanical concept embodied in this alternator type has completely replaced the earlier DC generator as the standard design in automotive applications. Based on equal outputs for both concepts, the alternator weighs 50 % less, and is also less expensive to manufacture. Large-scale application only became feasible with the introduction of compact, powerful, inexpensive and reliable silicone diodes.

The leakage flux between the claw poles limits the length – and thus the output – that can be achieved for any given diameter with this type of layout. In order to increase output, two systems can be combined within the same housing in certain cases.

Claw-pole alternator *(compact-diode design)*
1 Collector-ring end shield, 2 Rectifier heat sink, 3 Power diode, 4 Exciter diode, 5 Drive end shield with mounting flanges, 6 Fan pulley, 7 External fan, 8 Stator, 9 Claw-pole rotor, 10 Transistor regulator.



Compact alternator
1 Casing,
2 Stator,
3 Rotor,
4 Electronic voltage regulator with brush holder,
5 Collector rings,
6 Rectifier,
7 Fan.



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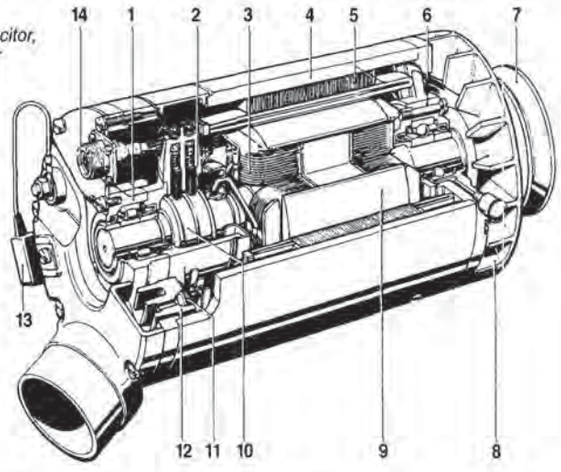
The classic alternator design is characterized by the external fan providing single-flow axial ventilation (compact-diode design).

Compact alternator (air-cooled)
The compact alternator is a new variant of the claw-pole concept based on dual-flow ventilation with two smaller internal fan elements. The cooling flow is extracted from the surrounding air in the axial plane, and exits the alternator radially in the vicinity of the stator winding heads, at the drive and collector-ring end shields. The major advantages of the compact alternator are:
- Higher maximum operating speeds for enhanced efficiency,
- Smaller fan diameters for reduced aerodynamic noise,
- Substantial reduction in magnetic noise,
- Longer carbon-brush life due to smaller collector-ring diameter.

Salient-pole alternator
Alternators based on the salient-pole concept are required in those special applications which are characterized by extreme power demands (as in touring coaches). The rotor is equipped with individual magnetic poles, each of which is provided with its own field winding. This layout allows the stator to be substantially longer (relative to its diameter) than would be possible with a claw-pole alternator. This means that higher outputs can be achieved without increasing diameter. The maximum speeds achievable, however, are lower than with claw-pole designs. Since the excitation currents required for the salient-pole alternator are substantially higher than those for the claw-pole unit, the attendant higher temperatures mean that the electronic regulator must be mounted in a special housing remote from the alternator.

Alternators with windingless rotors
The windingless-rotor alternator is a special design variation of the claw-pole unit in which only the claw poles rotate, while the excitation winding remains stationary. Instead of being connected directly to the shaft, one of the pole wheels is held in place by the opposite pole wheel via a nonmagnetic intermediate ring. The magnetic flux must cross two additional air gaps beyond the normal working gap. With this design, the rectifier supplies current to the excitation winding directly

Salient-pole alternator
1 Collector-ring end shield, 2 Contact brushes, 3 Excitation winding, 4 Casing, 5 Stator, 6 Drive end shield, 7 Drive-belt pulley, 8 Radial fan, 9 Salient-pole rotor, 10 Collector rings, 11 Power diode, 12 Heat sink, 13 Suppression capacitor, 14 Plug connector for lines to regulator.



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through the regulator; sliding contacts are not required. This arrangement obviates the wear factor represented by the collector-ring and carbon-brush assemblies, making it possible to design alternators for a substantially longer service life (important for construction equipment and railroad generators). The units weigh somewhat more than claw-pole alternators of comparable generating capacity due to the fact that additional iron is required to conduct the magnetic flux through two additional air gaps.

The windingless-rotor design is also obtainable as a liquid-cooled version (type LIF-B). Engine coolant flows around the complete circumference and rear of the alternator housing. The electronic components are mounted on the drive-end shield.

Operational limits

Ventilation

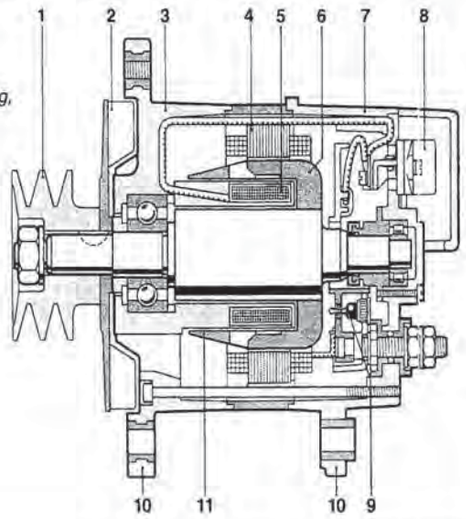
Air-cooled passenger-car alternators are almost always cooled by a flow of air supplied by an integral or attached radial fan. In certain cases where engine-compartment temperatures are very high, air is ducted in from outside. The dimensions of the ventilation arrangement must be adequate for ensuring that component temperatures remain below the specified limits under all conceivable operating conditions.

On alternators for heavy-duty vehicles, the entire collector-ring and carbon-brush assembly is usually encapsulated in order to prevent the entry of dust, dirt and water. Fresh-air induction is almost always beneficial, especially with higher outputs. In certain cases, sealed alternators with cooling fins on the outer casing are used. Other special applications may require the use of sealed alternators cooled by a liquid (e.g. oil).

Liquid-cooled alternators with a water jacket around the alternator casing are characterized in particular by

- lack of air-flow noise (–20 dBA),
- suitability for use in engine compartments with high temperatures,
- capability of integration in engine block,
- wading capabilities due to complete encapsulation, and
- contribution to engine warm-up by virtue of the heat dissipated by the alternator to the coolant.

Alternator with windingless rotor *(air-cooled)*
1 Pulley,
2 Fan,
3 Drive end shield with stationary inner pole,
4 Stator,
5 Stationary excitation winding,
6 Windingless rotor,
7 Rear end shield,
8 Regulator,
9 Power diode,
10 Bracket,
11 Conductive element.



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In contrast to air-cooled alternators, the external electrical connections are on the drive-pulley end. If a liquid-cooled alternator is fitted to the engine as a separate unit with its own housing, it has to have some means of connecting it to the cooling system (e.g. hoses). The reliable coolant supply must be designed-in during the overall application-engineering work.

Installation in the vehicle

Virtually all those engine-powered alternators which are driven by standard V-belts are installed on bracket assemblies which allow the belt tension to be adjusted by pivoting the alternator. If ribbed V-belts are used (Poly-V-belts), the alternator mounting is generally rigid, and belt tension is maintained by a separate mechanism. Larger alternators can be attached directly to the engine in special cradle-shaped mounts. The load imposed by the drive belt is the decisive factor in selecting the bearing dimensions for the alternator's drive side. The belt forces are determined by both the geometry of the drive layout and the power requirements of all the other devices being driven by the belt. Yet another factor is the effective radius of the pulley; larger radii can produce a substantial lever effect between the pulley's load-bearing surface and the drive-side bearings. The stresses emanating from these static factors are supplemented by the dynamic forces associated with torque and speed fluctuations.

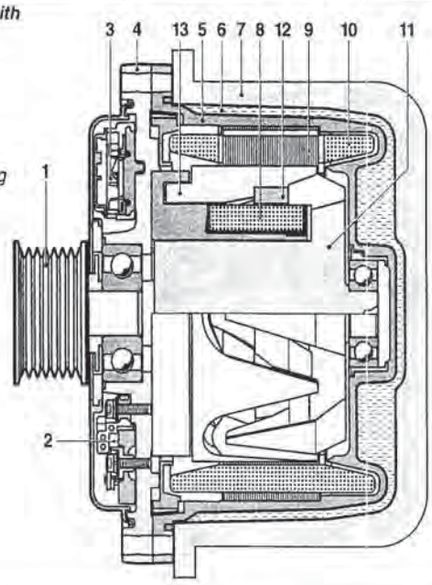
Belt pulleys with freewheel mechanisms allow the alternator shaft to be isolated from crankshaft vibration, thus making the belt drive substantially smoother for larger alternators with a greater mass inertia.

Drive layout

Although standard V-belts are usually employed to drive the alternator, ribbed V-belts are being increasingly used in automotive applications. Because this design allows tighter bend radii, smaller alternator pulleys and higher conversion ratios can be achieved. Railroad alternators are operated by helical-gear assemblies driven directly from the axle. It is imperative that special precautions be taken to dampen rotary oscillations when direct mechanical drive (without an intermediate belt, e.g., centrally at the crankshaft or via gears) is used to power the alternator.

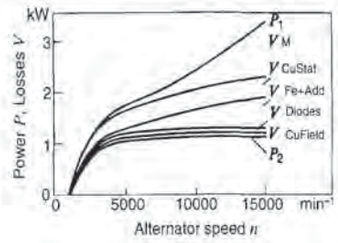
Liquid-cooled compact alternator with windingless rotor

1 Drive-belt pulley,
2 Rectifier,
3 Regulator,
4 Drive end shield,
5 Alternator housing,
6 Coolant,
7 Jacket casing for attached mounting or engine block,
8 Stationary excitation winding,
9 Stator core,
10 Stator winding,
11 Windingless rotor,
12 Nonmagnetic intermediate ring,
13 Conductive element.

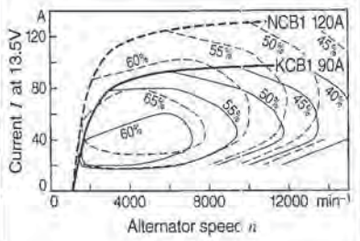


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Loss distribution of an alternator
P_1 Power input, P_2 Power output,
V_{mech} Air and bearing friction,
V_{CuStat} Copper losses in stator,
V_{Fe+Add} Iron and additional losses,
V_{Diodes} Rectifier losses,
$V_{CuField}$ Excitation losses.



Efficiency graphs for sizes NCB1 and KCB1 *(Compact alternators)*
Example of efficiency at part load:
I = 70 A, U = 14 V, n = 6000 min^{-1}.
η_{KCB1} = 57 % at 5.2 kg,
η_{NCB1} = 65 % at 6.3 kg.



Efficiency

Losses are an unavoidable by-product of all processes in which mechanical or kinetic energy is converted into electrical energy. The efficiency rating is the ratio between the power which is supplied to the unit and that which emerges. Iron losses result from the hysteresis and eddy currents produced by the alternating magnetic fields in the iron of the stator and rotor. Copper losses are produced by resistance in the rotor and stator windings. Their extent is proportional to the power-to-weight ratio, i.e., the ratio of generated electrical power to the mass of the effective components. The mechanical losses include the frictional loss in the roller bearings and at the contact brushes, the air resistance encountered by the rotor and, above all, the fan resistance which rises dramatically as speed increases.

In normal automotive operation, the alternator operates in the part-load range. Efficiency at medium speeds is then around 50%. The use of a larger (and heavier) alternator enables it to be operated in a more favorable partial-load efficiency band for the same electrical load. The efficiency gains provided by the larger alternator more than compensate for the losses in fuel economy associated with the greater weight. However, the higher mass inertia has to be taken into consideration.

The alternator is a typical example of an assembly which is in permanent operation all the time the vehicle is being driven, and when design measures are under consideration regarding maximum fuel economy, more emphasis should be placed on optimization of efficiency than upon optimization of its mass.

Noise

The quieter the modern-day vehicle becomes, the more the noise developed by the alternator becomes noticeable. The alternator's noise is compr sed of a magnetic and an aerodynamic component.

The magnetically induced high-pitched noise is audible mainly at low speeds (< 4000 min^{-1}). It can be counteracted by optimizing the alternator's magnetic circuit and its oscillation and radiation characteristics.

The aerodynamic noise occurs for the most part at high speeds. It can be reduced by the use of an optimized (e.g. asymmetrical) fan and optimisation of the air channels.

Alternator (voltage) regulation

Here too, the two options
– standard regulator and
– multifunction regulator

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have different basic purposes. The standard voltage regulator serves to keep the alternator voltage approximately constant in the face of wide fluctuations in alternator speed and load. The target level is usually temperature-dependent. The voltage is somewhat higher in winter to compensate for the fact that the battery is then more difficult to charge. In summer, the voltage regulator maintains the system voltage at a lower level to prevent the battery from overcharging. It is also possible for the voltage of the electrical system to drop below the regulation voltage when there is substantial current consumption at low min^{-1}.

Voltage regulators were formerly constructed using discrete components, but today they incorporate hybrid or monolithic circuits. When monolithic technology is applied, the control and regulator IC, the power transistor and the freewheeling diode are all located on a single chip.

Multifunction regulators perform special additional functions apart from the basic task of regulating the voltage.

The "load-response" (LR) function in particular is worthy of mention. It helps to improve the running and exhaust characteristics of the engine by means of a limited rate of output increase over time. A distinction is made between engine-running load response and engine-starting load response (alternator inactive for a defined period after starting).

Voltage regulators with digital interfaces represent a response to the increasing demands for greater mutual compatibility between engine-management and alternator-regulation systems. The interfaces used are chiefly of the bit-synchronous type. Coding is performed by way of the signal duration instead of its amplitude and is stored with a set protocol. The advantages here are the interference immunity and lack of temperature-susceptibility of the interface electronics.

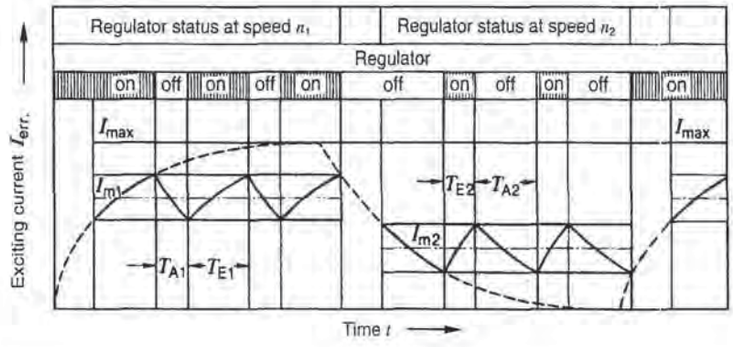
Interface regulators enable the load-response functions to be finely tuned to the engine operating status, the torque pattern to be optimized for the purposes of reducing consumption, and the charging voltage to be adjusted to improve the battery charge level.

Overvoltage protection

The electric strength of alternators and voltage regulators usually suffices to ensure that their semiconductor elements will operate reliably with the vehicle's battery connected. Emergency operation without the battery is characterized by extreme voltage peaks. Especially critical is the load-dump phenomenon in which the current to major consumers is suddenly interrupted. Thus additional measures

Function of regulator

I_{err} Excitation current, I_m Mean excitation current, T_E Duty period, T_A Off period, n_1 Lower speed, n_2 Higher speed.



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are necessary to ensure complete reliability in such cases.

There are three options available for protecting against overvoltage:

Zener diodes

Zener power diodes can be installed in the rectifier in place of the power diodes. Zener diodes limit the high-energy voltage peaks to levels which are harmless for the alternator or the regulator. In addition, Zener diodes can be used to provide remote protection for other voltage-sensitive equipment in the vehicle's electrical system. The response voltage of a rectifier which is equipped with Zener diodes ranges from 25 to 30 V. Compact alternators are 100%-equipped with Zener diodes.

Alternators and regulators with enhanced electric strength

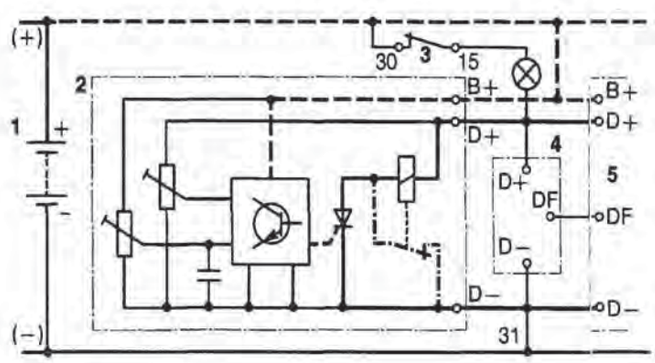
In such alternators and regulators, the semiconductor elements have higher electric-strength ratings. The enhanced electric strength of such alternators and regulators only protects the units themselves; these measures though furnish no additional protection for other electrical equipment in the system.

Overvoltage-protection devices

These semiconductor devices are connected to the alternator's D+ and D- (ground) terminals. The system responds to voltage peaks by shorting the alternator at the excitation winding. The main beneficiaries of these overvoltage protection devices are the alternator and regulator, with only secondary protection being provided for other voltage-sensitive equipment in the electrical system. Overvoltage-protection devices can be combined with other units specially designed to inhibit consequential damage. Such a layout prevents the battery from boiling-off its electrolyte should the regulator malfunction and remain in the "on" position.

Normally, alternators are not provided with reverse-polarity protection. Reversal of battery polarity (e.g. when using an external battery to start the vehicle) leads to destruction of the alternator diodes as well as endangering the semiconductor components of other assemblies in the vehicle.

Automatic overvoltage protection device *(Circuit diagram)*
1 Battery, 2 Overvoltage-protection device, 3 Ignition and starting switch, 4 Regulator, 5 Alternator.



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Controller Area Network (CAN)

Present-day motor vehicles are equipped with a large number of electronic control units (ECUs) which have to exchange large volumes of data with one another in order to perform their various functions. The conventional method of doing so by using dedicated data lines for each link is now reaching the limits of its capabilities. On the one hand, it makes the wiring harnesses so complex that they become unmanageable, and on the other the finite number of pins on the connectors becomes the limiting factor for ECU development. The solution is to be found in the use of specialized, vehicle-compatible serial bus systems among which the CAN has established itself as the standard.

Applications

There are four areas of application for CAN in the motor vehicle, each with its own individual requirements:

Real-time applications

Real-time applications, in which electrical systems such as Motronic, transmission-shift control, electronic stability-control systems are networked with one another, are used to control vehicle dynamics.

Typical data transmission rates range from 125 kbit/s to 1 Mbit/s (high-speed CAN) in order to be able to guarantee the real-time characteristics demanded.

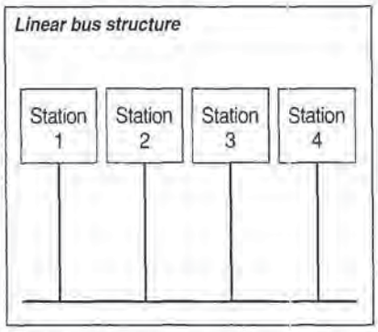
Multiplex applications

Multiplex applications are suitable for situations requiring control and regulation of body-component and luxury/convenience systems such as air conditioning, central locking and seat adjustment.

Typical data transmission rates are between 10 kbit/s and 125 kbit/s (low-speed CAN).

Mobile-communications applications

Mobile-communications applications connect components such as the navigation system, cellular phone or audio system with central displays and controls. The basic aim is to standardize control operations and to condense status information so as to minimize driver distraction.



Data transmission rates are generally below 125 kbit/s; whereby direct transmission of audio or video data is not possible.

Diagnostic applications

Diagnostic applications for CAN aim to make use of existing networking for the diagnosis of the ECUs incorporated in the network. The use of the "K" line (ISO 9141), which is currently the normal practice, is then no longer necessary.

The data rate envisaged is 500 kbit/s.

Bus configuration

CAN operates according to the multi-master principle, in which a linear bus structure connects several ECUs of equal priority rating. The advantage of this type of structure lies in the fact that a malfunction at one node does not impair bus-system access for the remaining devices. Thus the probability of a total system failure is substantially lower than with other logical architectures (such as ring or active star structures). When a ring or active star structure is employed, failure at a single node or at the CPU is sufficient to cause a total failure.

Content-based addressing

Addressing is message-based when using CAN. This involves assigning a fixed identifier to each message. The identifier classifies the content of the message (e.g., engine speed). Each station processes only those messages whose identifiers are stored in its acceptance list (message filtering). Thus CAN requires no station addresses for data transmission, and the nodes are not involved in ad-

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ministering system configuration. This facilitates adaptation to variations in equipment levels.

Logical bus states

The CAN protocol is based on two logical states: The bits are either "recessive" (logical 1) or "dominant" (logical 0). When at least one station transmits a dominant bit, then the recessive bits simultaneously sent from other stations are overwritten.

Priority assignments

The identifier labels both the data content and the priority of the message being sent. Identifiers corresponding to low binary numbers enjoy a high priority and vice versa.

Bus access

Each station can begin transmitting its most important data as soon as the bus is unoccupied. When several stations start to transmit simultaneously, the system responds by employing "Wired-AND" arbitration to sort out the resulting contentions over bus access. The message with the highest priority is assigned first access, without any bit loss or delay. Transmitters respond to failure to gain bus access by automatically switching to receive mode; they then repeat the transmission attempt as soon as the bus is free again.

Message format

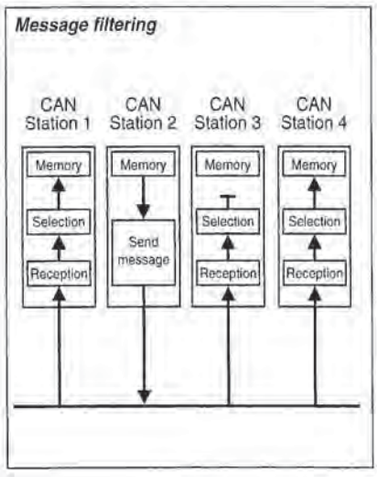
CAN supports two different data-frame formats, with the sole distinction being in the length of the identifier (ID). The standard-format ID is 11 bits, while the extended version consists of 29 bits. Thus the transmission data frame contains a maximum of 130 bits in standard format, or 150 bits in the extended format. This ensures minimal waiting time until the subsequent transmission (which could be urgent). The data frame consists of seven consecutive bit fields:

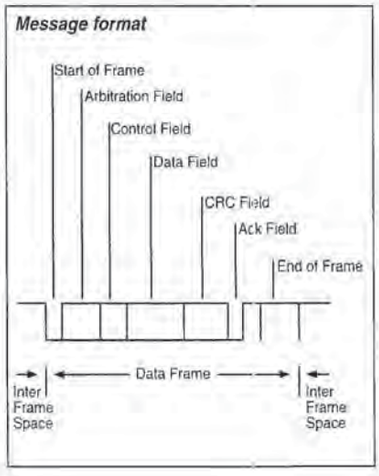
Start of frame indicates the beginning of a message and synchronizes all stations.

The arbitration field consists of the message's identifier and an additional control bit. While this field is being transmitted, the transmitter accompanies the transmission of each bit with a check to ensure that no higher-priority message is being transmitted (which would cancel the access authorization). The control bit determines whether the message is classified under "data frame" or "remote frame".

The control field contains the code indicating the number of data bytes in the data field.

The data field's information content comprises between 0 and 8 bytes. A message of data length 0 can be used to synchronize distributed processes.





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The CRC field (Cyclic Redundancy Check) contains the check word for detecting possible transmission interference.
The Ack field contains the acknowledgement signals with which all receivers indicate receipt of non-corrupted messages.
End of frame marks the end of the message.

Transmitter initiative
The transmitter will usually initiate a data transfer by sending a data frame. However, the receiver can also request data from the transmitter. This involves the receiver sending out a "remote frame". The data frame and the corresponding remote frame have the same identifier. They are distinguished from one another by means of the bit that follows the identifier.

Error detection
CAN incorporates a number of monitoring features for detecting errors. These include:
- 15-bit CRC (Cyclic Redundancy Check): Each receiver compares the CRC sequence which it receives with the calculated sequence.
- Monitoring: Each transmitter compares transmitted and scanned bit.
- Bit stuffing: Between "start of frame" and the end of the CRC field, each data frame or remote frame may contain a maximum of 5 consecutive bits of the same polarity. The transmitter follows up a sequence of 5 bits of the same polarity by inserting a bit of the opposite polarity in the bit stream; the receivers eliminate these bits as the messages arrive.
- Frame check: The CAN protocol contains several bit fields with a fixed format for verification by all stations.

Error handling
When a CAN controller detects an error, it aborts the current transmission by sending an error flag. An error flag consists of 6 dominant bits; it functions by deliberately violating the conventions governing stuffing and/or formats.

Fault confinement with local failure
Defective stations can severely impair the ability to process bus traffic. Therefore, the CAN controllers incorporate mechanisms which can distinguish between intermittent and permanent errors and local station failures. This process is based on statistical evaluation of error conditions.

Implementations
In order to provide the proper CPU support for a wide range of different requirements, the semiconductor manufacturers have introduced implementations representing a broad range of performance levels. The various implementations differ neither in the message they produce, nor in their arrangements for responding to errors. The difference lies solely in the type of CPU support required for message administration.

As the demands placed on the ECU's processing capacity are extensive, the interface controller should be able to administer a large number of messages and expedite data communications with, as far as possible, no demands on the CPU's computational resources. Powerful CAN controllers are generally used in this type of application.

The demands placed on the controllers by multiplex systems and present-day mobile communications are more modest. For that reason, more basic and less expensive chips are preferred for such uses.

Standardization
CANs for data exchange in automotive applications have been standardized both by the ISO and the SAE – in ISC 11519-2 for low-speed applications ≤ 125 kbit/s and in ISO 11898 and SAE J 2284 (cars) and SAE J 1939 (trucks and buses) for high-speed applications >125 kbit/s. There is also an ISO standard for diagnosis via CAN (ISO 15765 – Draft) in the course of preparation.

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CARTRONIC®

Networked vehicle systems
The continuing development of automotive electronic systems is driven by ever increasing demands. These include: Safety, comfort and environmental compatibility, tougher legal requirements and directives, integration of information and entertainment-system functions ("infotainment") and communication with external computers and data services via mobile telephone.

Under the influence of those demands combined with the unabated downward pressure on prices, individual vehicle systems (fuel injection, ABS, radio) have developed into a networked composite system in which information is exchanged via data busses (e. g. CAN) and reciprocal interaction is possible. Cross-marque standardization of individual components, subsystems and subfunctions within such a composite system is the basic requirement if development times are to be speeded-up. At the same time, reliability and system availability must be increased and, by sharing information between different vehicle systems, a reduction in the number of components required.

Composite systems (examples)
Present-day vehicles already have composite systems such as the traction control system (TCS) and the electronic stability program (ESP) which is an extension of it. The cross-system functions of those two systems are effected by the TCS control unit (ECU) informing the engine-management ECU when the wheels start to spin so that the latter can reduce the engine torque accordingly. Similarly, the air-conditioning system may inform the engine-management system that it is about to switch on, which will result in the need for an increase in engine torque/speed.

Requirements
The implementation of such cross-system functions through the interaction of subsystems requires agreement on the standardization of interfaces and subsystem functions. Definitions must be drawn up which specify what information a subsystem requires and what variables are to be controlled on the basis of that information. This is all the more important in view of the fact that the subsystems are developed separately (often by several different suppliers) and modification to suit a particular vehicle model or the requirements of a specific manufacturer is costly and susceptible to errors.

Concept
The above demands led to the development of the CARTRONIC as a classification and specification concept for all vehicle control and management systems. It contains definite rules for interaction between subsystems as well as expandable modular architectures for "function", "safety" and "electronics" on the basis of these formal rules. It thus provides the means for a description of the vehicle as an overall system. On that basis, suppliers can harmonize interaction between their products without having to be familiar with the internal processes of subsystems and without bringing about large-scale manufacturer- or model-related modifications.

Structuring, architecture
What is required is a universal structuring system and practical implementation of it in an appropriate structure. The function architecture at vehicle level encompasses all control and management tasks that arise within the vehicle. Logical components are defined which represent tasks of the composite system. The links and interfaces between components and the manner of their interaction are specified. The system architecture thus created has to be supplemented by a safety architecture which provides additional elements in order to ensure safe and reliable operation of the system as a whole. The composite system is then created by the transformation of the various logical and functional components into hardware form (electronic circuitry, ECUs, microcomputers). In contrast to the system architecture, the resulting optimized hardware topology is characterized by the specifics of the vehicle model (e.g. specified dimensions and siting of components).

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Architecture rules

The rules for the function architecture are designed to create a definition and layout for the composite system that is independent of the specific hardware topology, except from the point of view of logical and functional considerations. For that reason, the rules essentially define components and the permissible interaction in the sense of communication interrelationships.

Functional analysis

Conceptually, the functional analysis of a planned or existing composite system starts with the analysis of the previously autonomous individual systems. As even this takes place at the functional level (as yet separate from practical hardware development and partitioning), the descriptions are still independent of vehicle-specific design variations and therefore largely permit universally valid statements. Basic structuring at this level makes it possible to restrict the diversity of hardware and software and to use identical electronic modules (basic modules) for the basic functions of a large number of vehicle models.

Structural elements

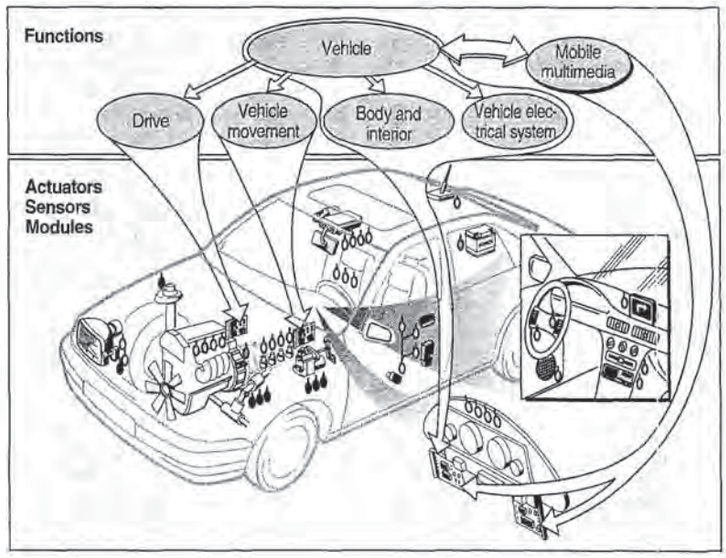
The elements of the architectures are systems, components and communication interrelationships which formally describe a composite system and also include the structuring and modeling rules for designing interaction and defining interdependencies.

Systems, components, interfaces

In those terms, a system is a combination of components that are linked to one another by means of communication mechanisms and which perform a higher-level function beyond their individual functions. The term component is explicitly not limited to interpretation as a physical unit but is understood rather to be a unit of function. CARTRONIC recognizes three types of component:

- components whose function is primarily coordination,
- components whose function is primarily operative, and
- components whose function is exclusively to generate, supply and forward information.

***CARTRONIC: Hardware topology** (example)*



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Interfaces on components refer to possible communication interrelationships that can be established with other components. Wherever possible, physical variables are defined as interfaces (e.g. transmission or engine torque).

Description of system
Therefore, a system is described by the representation of all functional components and their communication interrelationships and modes of reciprocal interaction.

Structuring rules
The structuring rules describe permissible communication interrelationships between different components within the system architecture. A hierarchical system concept is created according to the structure which starts from the vehicle as an entity and extends as far as the individual components. Accordingly, there are structuring rules for communication interrelationships between components on the same level and on different levels. There are also structuring rules for the forwarding of communications from one subsystem to another.

Modeling rules
The modeling rules consist of patterns which combine components and communication interrelationships for solving specific tasks which arise more than once within a vehicle system. Those patterns can then be repeated at various points within the structure of the vehicle.

Architectural features
A structure represented by means of the specified structuring and modeling rules displays standardized features and characteristics:
- a hierarchical work flow (jobs are accepted only on the same or a higher level),
- a clear distinction between coordinators and information suppliers (operating controls and sensors),
- clear demarcation between the individual components according to the "black box" principle (as visible as necessary and as concealed as possible).

Consequences
CARTRONIC represents a standardized concept for describing all vehicle functions. By virtue of the facility for defining generic functions, it can describe all commonly used vehicle control and management systems using standardized terminology. New functions will require appropriate extension of those function categories. The next step required is cross-marque definition of interfaces between components/subsystems at the functional and physical levels. This will enable implementation of complex function networks within the vehicle involving collaboration between multiple suppliers.

Outlook
The functional capabilities of present-day vehicle systems are determined to an increasing degree by software, and composite systems are becoming computer networks. The standardization of operating systems is making software applications portable, i. e. they can be used on different ECUs. The software architecture is thus becoming independent of the hardware topology. In order to keep individual software modules interchangeable/re-usable, the CARTRONIC architecture and interface rules will require further refinement and more precise definition. As in the field of computer applications, interfaces between different functions will be defined by "APIs" (application programming interfaces) which will then precisely define the CARTRONIC communication interrelationships. This course requires agreement between the different manufacturers and suppliers on industry standards. CARTRONIC has established the basic foundations for such a development.

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Electromagnetic compatibility (EMC) and interference suppression

The expression "electromagnetic compatibility" (EMC) defines an electrical system's ability to remain neutral in the vicinity of other systems. In other words it is compatible, and besides not interfering with other systems it also remains impervious to such interference as may emanate from them. In automotive applications, this means that the various electrical and electronic systems such as the ignition system, the electronic fuel injection, ABS/TCS, airbags, radio, car phone, navigation system, etc. must function in close physical proximity without interfering with each other beyond an allowable level. It also means that the vehicle in its role as a system must remain neutral within its environment; it is not to interfere electrically with other vehicles, and it must not interfere with broadcast or communications transmissions of any kind. At the same time, the vehicle must itself remain fully operational when exposed to strong electromagnetic fields from the outside (e.g., in the vicinity of radio transmitters).

It is in view of these considerations that automotive electrical systems and complete vehicles are designed to ensure electromagnetic compatibility.

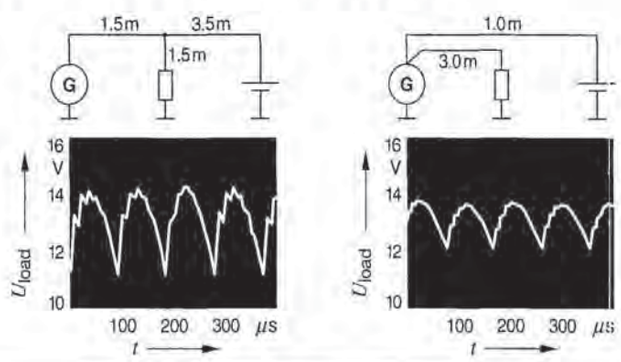
Sources of interference

On-board electrical system, ripple
The alternator supplies the vehicle's electrical system with rectified alternating current. Although the current is smoothed by the battery, a residual ripple remains. Its amplitude depends on the electrical system load and the wiring. Its frequency does not change according to alternator or engine speed and the fundamental oscillation is in the kHz range. It can penetrate into the vehicle's sound system – either directly (galvanically) or through inductance – where its presence is heard as a whine in the speakers.

On-board electrical system, pulses
Interference pulses are generated when electrical equipment in the vehicle is switched on and off. They are passed to adjacent systems directly through the power supply (conductive transfer) and indirectly by means of crossover from connecting leads (inductive and capacitative coupling). If the matching between the interference source and the system receiving it is inappropriate, these pulses can cause anything from malfunction to destruction of the adjacent systems.

The wide variety of pulses that occur in the vehicle can be classified in five basic groups. Classification according to amplitude provides the best basis for matching the interference sources and

Influence of electrical-system design on voltage ripple
Example: I_{Gen} = 130 A, n_{Gen} = 18,000 min^{-1}, Battery 12 V 55 A · h.



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the potentially susceptible equipment to achieve maximum compatibility between them.

This matching procedure may, for example, entail having all a vehicle's interference sources conform to Class II, while potentially susceptible devices (such as ECUs) can be designed to conform to Class III requirements (taking a safety margin into account). Reallocation to Classes I/II is advisable in cases where suppression at the source is easier than providing the corresponding protection at the susceptible devices.

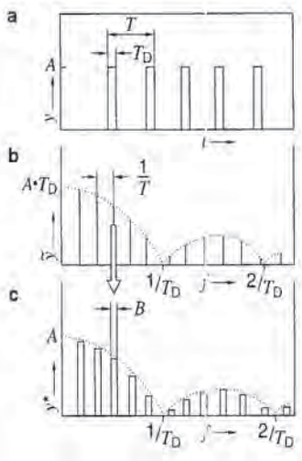
The principle can also be reversed; if protective measures at the potentially susceptible equipment are easier and less expensive then a move to Classes III/IV is advisable.

On-board electrical system, high-frequencies

High-frequency oscillations are generated inside many electromechanical and electronic components by the switching of voltages and currents (digital circuits, triggering of output stages, commutation). These oscillations travel through the components' circuits – especially the power-supply lines – and back into the vehicle's electrical system, where they arrive at variously attenuated intensities.

Depending upon whether the measured interference voltage's spectrum is continuous or a collection of individual curves, a distinction is drawn between two kinds of interference source: "broad-band" (electric motors, as used for wipers,

Sources of broad and narrow-band interference
a) Signal progression referred to time $y(t)$,
b) Corresponding spectrum $\bar{y}(j)$,
c) Observation of spectrum with monitoring device of bandwidth B: with $B \cdot T < 1$ (as shown in diagram) individual bars indicate "narrow-band" interference; $B \cdot T > 1$ continuous curve indicates "broad-band" interference.



Test pulses as per DIN 40839, Part 1 for 12V electrical systems

Test pulses				Classification of permissible pulse amplitudes			
Pulse shape	Source	Internal resistance	Pulse duration	I	II	III	IV
1	Switch-off of inductive loads, e.g. relays or valves.	10 Ω	2 ms	– 25 V	– 50 V	– 75 V	– 100 V
2	Switch-off of electric-motor loads such as fan motor which generate positive overvoltage when running on.	10 Ω	50 µs	+ 25 V	+ 50 V	+ 75 V	+ 100 V
3a	Overvoltages due to switching processes.	50 Ω	0.1 µs	– 40 V	– 75 V	– 110 V	– 150 V
3b				+ 25 V	+ 50 V	+ 75 V	+ 100 V
4	Supply-voltage curve during starting.	10 mΩ	up to 20 s	+ 12 V – 3 V	+ 12 V – 5 V	+ 12 V – 6 V	+ 12 V – 7 V
5	"Load dump"[1]	1 Ω	up to 400 ms	+ 35 V	+ 50 V	+ 80 V	+ 120 V

[1]) Load dump occurs when the alternator is feeding high current into the battery and the connection between the two is suddenly interrupted.

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fans, fuel pump, alternator) and "narrow-band" (ECUs with microprocessors). The classification depends upon the bandwidth of the test instrument being used.

Such high-frequency oscillations can represent a permanent source of interference for communications systems in the vehicle; they are located within the same frequency and amplitude ranges as the transmitted or received signals, and are thus easily able to enter the vehicle's communications system either directly at the sensor (antenna) or through the antenna cable. The narrow-band sources of interference are particularly critical as they have signal characteristics very similar to the spectrum of transmitters.

For broad-band interference sources such as electric motors, fans, etc., the interference output is assessed on the basis of the interference voltages on the supply lines in a defined test setup according to CISPR 25 or DIN/VDE 0879-2. The interference voltage gradings defined by interference-suppression levels simplify the matching of interference sources and devices susceptible to interference at the vehicle's OE-equipment stage.

Should the interference level initially approved for OE purposes prove to be too high upon subsequent installation of an additional communications system, there is a limited range of remedial measures which can be implemented:

- If the source of the interference derives its current directly from terminal 15 or 30, then the interference can be counteracted with suppression capacitors and filters of a type suitable for automotive application. The capacitors are generally connected directly to the source's terminal, with the ground wire being kept as short as possible. Coupling from conductors carrying interference voltage to other conductors can be reduced by passing such wires through a braided metal screen which is grounded as directly as possible at both ends.
- If the interference source is controlled by an ECU, it is generally forbidden to suppress it subsequently by means of suppression components, as this would modify the ECU switching response.
- The clock signals from the microprocessors in the ECUs act as narrow-band interference sources. Subsequent interference suppression is not generally possible with such components. Interference transmission can therefore be minimized as far as possible by appropriate circuitry (e.g. suppression capacitors), and suitable location and routing of components and wiring. If such interference-suppression measures are not sufficient, it is necessary to attempt to achieve satisfactory results by appropriate layout of the electrical system, and the choice of an advantageous antenna position combined with suitable routing of the antenna cable.

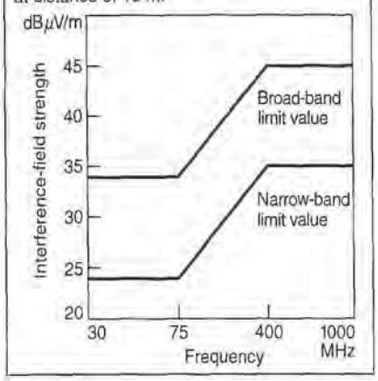
In the laboratory, assessment of the interference characteristics of electronic components either also uses line-conducting methods (e.g. measuring the interference voltage level on the supply lines) or antenna measurements in absorber rooms. The final verdict as to whether reception (radio or mobile com-

Permissible radio interference voltage level in dBµV for suppression levels in the individual frequency ranges as per DIN/VDE 0879-2 for broad-band interference (B) and narrow-band interference (S)

Interference-suppression level	Interference level									
	0.15...0.3 MHz (LW)		0.53...2.0 MHz (MW)		5.9...6.2 MHz (SW)		30...54 MHz		70 ..108 MHz (UHF)	
	B	S	B	S	B	S	B	S	B	S
1	100	90	82	66	64	57	64	52	48	42
2	90	80	74	58	58	51	58	46	42	36
3	80	70	66	50	52	45	52	40	36	30
4	70	60	58	42	46	39	46	34	30	24
5	60	50	50	34	40	33	40	28	24	18

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Limits for interference radiation by vehicles as per CISPR 12 and 95/54/EC
Broad-band and narrow-band levels, tested at distance of 10 m.



munications) will be possible in the vehicle depends upon interference-voltage measurements carried out at the receiver end of the antenna cable. This is done with the aid of a suitable testing circuit for matching the input impedance of the test receiver to the input impedance of the receiver unit. In order to obtain realistic results, those tests must if possible be performed using the original antenna in its original installation position. In order to isolate them from the external electromagnetic transmitter and interference signals, such measurements are carried out in shielded EMC chambers equipped with high-frequency absorbers.

The vehicle as an interference source

The ignition system is the major source of interference in the vehicle as a whole. The maximum permissible radiation from motor vehicles is limited by law (EU Directive 95/54/EC) in order to ensure that it does not interfere with the radio and TV reception of other vehicles and roadside residents. Maximum limits are specified for both broad-band and narrow-band interference.

The levels specified in the Directive represent the minimum requirements. Keeping interference to a level only just within the specified limits is not normally sufficient in practice to ensure interference-free reception in the same vehicle. For that reason, suppression has to be increased as appropriate for the vehicle design in order to achieve acceptable reception for an on-board radio, telephone, etc. Ignition systems can be fitted with suitable suppression components such as resistors in the HT connectors and resistor spark plugs to further reduce interference emission. For specialist vehicles (e.g. equipped with two-way radio for emergency services) it can even be necessary to partially or completely shield the ignition system. Such interference-suppression measures can have negative effects on the ignition system's secondary available voltage. In such cases, therefore, a thorough investigation into the permissibility of the measures is necessary.

Potentially susceptible devices

ECUs and sensors are susceptible to interference signals entering the system from outside. The interference signals emanate either from neighboring systems within the vehicle itself or from sources in the immediate vicinity (such as a powerful broadcast transmitter). Malfunctions start to occur at the point where the system loses the ability to distinguish between interference signals and useful signals.

The possibility of taking effective measures depends upon the characteristics of the useful and the interference signals.

The ECU is unable to distinguish between useful and interference signals if their signal characteristics are similar. This is the case for instance, when a pulse-shaped interference signal is similar to the signal from a wheel-speed sensor. Particularly critical here are frequencies in the vicinity of the useful-signal frequency ($f_S \approx f_N$) and in the same range as a few of the useful-frequency harmonics.

Non-modulated (or NF-modulated) sinusoidal HF signals (energy transmitted by interference sources) can be demodulated at the pn junctions in the electronic circuitry. This can lead to the direct components causing level shifts, or to the superimposition of interference signals which fluctuate as a function of time, as a result of the NF component of the interfer-

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$$U_b = \frac{k \cdot R_2 \cdot \sinh(\gamma \cdot l)}{(R_1 + R_2) \cdot \cosh(\gamma \cdot l) + W\left(k_a + \frac{R_1 \cdot R_2}{W^2} \cdot k_b\right) \cdot \sinh(\gamma \cdot l)} \cdot \left(\frac{R_1}{W} \cdot U - W \cdot I\right)$$

ence signal. Normally, the carrier frequency is a multiple of the useful frequency ($f_{S,HF} >> f_N$). The NF component of the interference signal is particularly critical if it is in the vicinity of the useful-signal frequencies ($f_{S,NF} = f_N$). Unwanted (interference) signals of a far lower frequency ($f_S << f_N$) can also lead to malfunctions due to intermodulations.

The necessary immunity to interference from electromagnetic fields is also defined by the EU Directive 95/54/EC. The field strengths to which a vehicle must be immune as specified by the Directive are, as with the interference emission limits, minimum requirements. In practice, vehicle manufacturers and suppliers provide significantly higher levels of interference immunity.

Interference coupling

Signals from interference sources penetrate susceptible devices in any of three ways:

- Direct (galvanic) coupling occurs when the source and the device affected by the interference share common current paths, a condition which can hardly be avoided with a common voltage source. The vehicle's wiring harness should be designed with the lowest-possible level of direct coupling in mind. Whether a parallel, serial or multipoint structure is best for the supply lines will depend upon the level of current, the frequency range, the impedance of the components and the general design of the system being connected.
- Coupling occurs on connecting lines when they are laid in parallel between the source and the susceptible device being exposed to the interference. In the model shown at the bottom of page 819, the voltage U_b which is coupled into the device receiving the interference is calculated as shown at the top of the page with the parameters

$$k = C/C_0;\ k_a = (C_a + C)/C_0;\ k_b = (C_b + C)/C_0$$

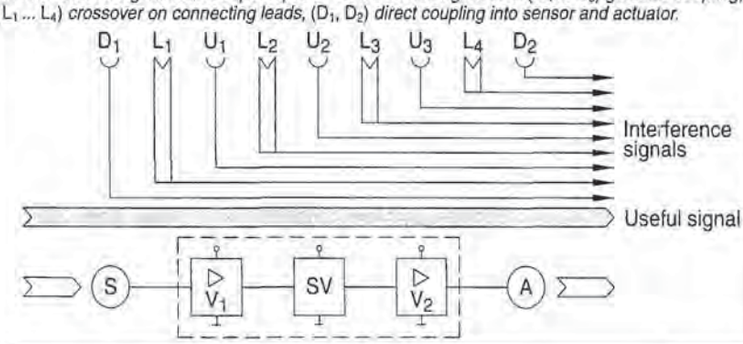
$$C_0 = \sqrt{C_a \cdot C_b + C \cdot (C_a + C_b)}$$

$$\gamma \cdot l = j(\omega/c) \cdot l;\ W = 1/(c \cdot C_0)$$

$c = 3 \cdot 10^8$ m/s (speed of light)

U_b consists of two parts: a "capacitive" component, which depends on the voltage U, and an "inductive" component, which is a function of the current I. If the wavelength of the interference signal is larger than the geometrical conductor length I, then the formula is simplified as follows:

Interference model
Electronic system: S *Sensor(s),* V_1 *Signal amplification and conditioning,* SV *Signal processing,* V_2 *Power amplification,* A *Actuator(s).*
An interference-signal flow is superimposed on the useful-signal flow. (U_1 ... U_3) *galvanic coupling,* (L_1 ... L_4) *crossover on connecting leads,* (D_1, D_2) *direct coupling into sensor and actuator.*



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$U_b \approx k \cdot (\gamma \cdot l) \cdot [U(R_1 \cdot R_2)/(R_1 + R_2) - W \cdot I \cdot R_2/(R_1 + R_2)]$

This indicates that coupling can be kept to a minimum the shorter the length l and the lower the standardized coupling capacity k.

k decreases as the distance between the conductors increases, and can be further reduced with a screen connected to ground at both ends.

- Direct interference is possible in cases where sensor S or actuator A (see Figure below) react directly to electromagnetic fields, e. g., if S is a radio antenna, a microphone or the magnetic head on a cassette player. Here the aim is to increase the physical distance between the interference source and the exposed susceptible equipment until the interference disappears.

Electrostatic discharge

The subject of the potential danger to components and electronic circuits from electrostatic discharge (ESD) also falls under the umbrella of EMC. Here, it is a case of protecting components and equipment from damage by static discharge from persons or from machinery on the production line. On the one hand, this involves the adoption of appropriate methods for handling equipment and, on the other, designing units in such a way that the extremely high voltages (several thousand volts) produced by electrostatic charges are reduced to resistible levels.

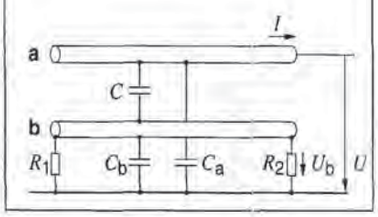
Measuring techniques

A wide variety of testing methods are in common use for testing the interference signal and the immunity to interference. Depending upon the type of assessment used for the interference phenomena, they can be roughly divided into methods operating in the time range (pulse generators, oscillographs) and those operating in the frequency range (sine-wave generators, measuring receivers, spectrum analyzers).

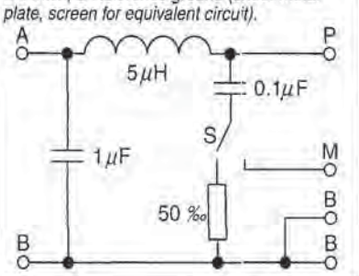
- In measurement techniques, the interference signals are generally registered as reference values in dB (decibels). The reference quantity for interference voltage is 1 µV; for electric field strength it is 1 µV/m, for power it is 1 mW, thus:
  $u^* = 20 \cdot \lg U$
  $e^* = 20 \cdot \lg E$
  $p^* = 10 \cdot \lg P$
  where u^*, e^*, p^* are given in dB; U in µV; E in µV/m; P in mW. In measurement techniques as used for interference-suppression measures, the quantities (pulse amplitude and energy radiated by interference source) are usually given directly (E in V/m; U in V; I in A).

EMC measurements are performed on individual systems in the laboratory and in the vehicle.

Model of coupling mechanism of line-conducted electromagnetic waves
a) Conductor on which an electromagnetic wave generated by the interference source is propagated, b) Target wire, component in interference-susceptible system.



Schematic diagram of the equivalent circuit of an automotive electrical system as per DIN/VDE 0879-2
Connections: P–B Unit being tested, A–B Power supply, M–B Radio-interference monitor, S Switch, B Reference ground (sheet-metal plate, screen for equivalent circuit).



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Laboratory test procedures

- Under standardized conditions, artificial networks are used to examine the pulses or high-frequency interference voltages which emanate from a device.
- Immunity to pulse-type interference is tested using special pulse generators which produce test pulses according to DIN 40839, Part 1, or ISO 7637, Parts 1 and 2.
- Wire-borne interference waves to be injected into the wiring harness of the electrical system under investigation, are produced either with the aid of a stripline, a TEM (transversal electromagnetic field) cell or BCI (bulk current injection). In the case of the stripline, the wiring harness is arranged in line with the direction of propagation of the electromagnetic wave between a strip conductor and a base plate. When the TEM cell is used, the ECU and a section of the wiring harness are arranged at right angles to the electromagnetic waves' direction of propagation. The BCI method involves superimposition of a current on the wiring harness by means of a power clip.
- For higher frequencies (> 400 MHz), the unit being tested and the wiring harness are irradiated by antennas and thereby directly exposed to an electromagnetic field.
- The emitted interference radiation is measured with broad-band antennas in screened measuring cells lined with absorbers.

Vehicle testing procedures

- The electronic system's resistance to the electromagnetic fields radiated by high-power transmitters is tested inside the vehicle which is located in a special anechoic chamber for the purpose. Here, appropriately high electrical and magnetic field strengths can be generated which are applied to the complete vehicle.
- The interference effects of the vehicle electrics and electronics on radio reception are measured using highly sensitive measuring receivers. As far as possible, the original vehicle antenna is left in place, and measurement is carried out at the input to the receiver.

Regulations and Standards

Automotive interference suppression (protection of non-mobile, permanently installed radio reception) has been mandatory in Europe since 1972. (EC Directive ECE10 / EU Directive 72/245/EEC). Since 1.1.1996 binding legal regulations have been in place concerning electromagnetic compatibility (EU/EEC Directive or German EMC laws) for all electrical products and installations which are sold on the market. For motor vehicles, the special EC Directive 95/54/EC, which supersedes the Directive 72/245/EEC, governs the protection of fixed-installation radio reception and the immunity of motor vehicles to interference from electromagnetic fields. In addition to the regulations governing the design of vehicles and components offered for sale to the public, this Directive also specifies the testing methods and maximum allowable limits.

The EMC measurement techniques are defined in a number of German and international Standards. Essentially, the national German Standards (DIN/VDE) agree with the international Standards (ISO/IEC-CISPR) and cover all aspects of automotive EMC.

Standards

Resistance to interference
DIN 40839, Part 1, ISO 7637-0/-1/-2,
DIN 40839, Part 3, ISO 7637-3,
DIN 40839, Part 4, ISO 11451/11452,
ISO/TR 10605.

Interference suppression
DIN/VDE 0879-2.
CISPR 12, CISPR 25.

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Alphabets and numbers

German alphabet

Gothic type

𝔄	𝔞	a	𝔍	𝔧	j	𝔖	𝔰	s
𝔅	𝔟	b	𝔎	𝔨	k	𝔗	𝔱	t
ℭ	𝔠	c	𝔏	𝔩	l	𝔘	𝔲	u
𝔇	𝔡	d	𝔐	𝔪	m	𝔙	𝔳	v
𝔈	𝔢	e	𝔑	𝔫	n	𝔚	𝔴	w
𝔉	𝔣	f	𝔒	𝔬	o	𝔛	𝔵	x
𝔊	𝔤	g	𝔓	𝔭	p	𝔜	𝔶	y
ℌ	𝔥	h	𝔔	𝔮	q	ℨ	𝔷	z
ℑ	𝔦	i	ℜ	𝔯	r			

Greek alphabet

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

Phonetic alphabets

	German	International	2-way radio
A	Anton	Amsterdam	Alpha
Å	Ärger	–	–
B	Berta	Baltimore	Bravo
C	Cäsar	Casablanca	Charlie
CH	Charlotte	–	–
D	Dora	Danmark	Delta
E	Emil	Edison	Echo
F	Friedrich	Florida	Foxtrott
G	Gustav	Gallipoli	Golf
H	Heinrich	Habana	Hotel
I	Ida	Italia	India
J	Julius	Jerusalem	Juliett
K	Kaufmann	Kilogram	Kilo
L	Ludwig	Liverpool	Lima
M	Martha	Madagascar	Mike
N	Nordpol	New York	November
O	Otto	Oslo	Oscar
Ö	Ökonom	–	–
P	Paula	Paris	Papa
Q	Quelle	Quebec	Quebec
R	Richard	Roma	Romeo
S	Samuel	Santiago	Sierra
Sch	Schule	–	–
T	Theodor	Tripoli	Tango
U	Ulrich	Uppsala	Uniform
Ü	Übermut	–	–
V	Viktor	Valencia	Victor
W	Wilhelm	Washington	Whiskey
X	Xanthippe	Xanthippe	X-Ray
Y	Ypsilon	Yokohama	Yankee
Z	Zeppelin	Zürich	Zulu

Cyrillic alphabet

Letter		Pronunciation	Letter		Pronunciation
А	а	a	Р	р	r
Б	б	b	С	с	ss
В	в	w	Т	т	t
Г	г	g	У	у	u
Д	д	d	Ф	ф	f
Е	е	je	Х	х	ch
Ё	ё	jo	Ц	ц	z
Ж	ж	sch (soft)	Ч	ч	tsch
З	з	s	Ш	ш	sch (hard)
И	и	i	Щ	щ	schtsch
Й	й	i (short)	Ъ	ъ	hard sign
К	к	k	Ы	ы	ü
Л	л	l	Ь	ь	soft sign
М	м	m	Э	э	ä
Н	н	n	Ю	ю	ju
О	о	o	Я	я	ja
П	п	p			

Roman numerals

I	1	XXX	30
II	2	XL	40
III	3	L	50
IV	4	LX	60
V	5	LXX	70
VI	6	LXXX	80
VII	7	XC	90
VIII	8	C	100
IX	9	CC	200
X	10	CD	400
XI	11	D	500
XX	20	DC	600
XXI	21	M	1000
XXIX	29	MVM	1995

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Passenger-car specifications

The following table contains data on selected passenger cars manufactured both inside and outside Germany (Validity: March 1, 1999), Source: "Automobil Revue Katalog 1999". The list is intended only as a representative selection; inclusion in or exclusion from the list is not intended as the reflection of a qualitative judgement of any kind.

Abbreviations

aa	All-wheel drive (AWD)
A	Automatic gearbox/transmission
ABS	Antilock braking system
ASR	Traction control
Bx	Opposed-cylinder engine
Ca	Convertible
Co	Coupé
Cp	Compact sedan/limousine
Db	Damper strut
DDQl	Double wishbone
De	De-Dion axle
DG	Double-link Macpherson-strut suspension
DQl	A-arm
DQu	Twin control arm
Ea	Independent suspension, rear
EBV	Electronic brake balancing
EDS	Electronic differential lock
El	Electronic gasoline injection
Eld	Electronic direct injection
EP/CR	Diesel fuel-injection pump, common-rail system, electronically controlled
EP/UIS	Unit Injector System UIS
EP/V	Diesel distributor-type fuel-injection pump
EP/eV	Diesel distributor-type fuel-injection pump, electronically controlled
ESP	Electronic Stability Program
F	Spring strut
Ge	Off-road vehicle
Gli	High-capacity sedan/minivan
h	Rear, on rear wheels
H	Manually-shifted transmissions
Hf	Hydropneumatic suspension
Hv	Hydrogas suspension
Ic	Intercooler
Ko	Station wagon
Kom	Compressor
Ku	Anti-roll bar
Li	Sedan/limousine
Ll	Trailing arm
Lue	Pneumatic suspension, electronically controlled
Ma	Multi-link axle
Mc	Microcar
Mm	Mid-engine
Mv	Microvan
P	Panhard roc
q	Transverse installation
Qf	Transverse leaf spring
Ql	Control arm
R	In-line engine
Ra	Multi-link independent-suspension axle (5 control arms per wheel)
Ro	Roadster
ROZ	Research Octane Number (RON)
Sa	Rigid axle
Sb	Disc brake
Sf	Coil spring
Sl	Semi-trailing arm
St	Continuously variable transmission (CVT)
T	Turbocharger
Ta	Torsion crank axle
Tb	Drum brake
Td	Telescopic shock absorber
Tf	Torsion bar
Tl	Trapezoidal arm suspension
v	Front, on front wheels
V	V-engine
Ve	Valves per cylinder
Vl	Four-joint trapezoidal-link axle
Vla	Trailing-arm torsion-beam axle
VR	Offset inline engine
Wa	Watt linkage
Ze	Single-point injection (TBI)
Zl	Supplementary link
2KE	2 spark plugs per cylinder
2Tlc	2 turbochargers and 2 intercoolers
4Sb	4 disk brakes
–	No details

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Explanations

Where available, the following information (lines 1–35) is given for each model. Not specifically listed are: Line 16: Mounted longitudinally; Line 20: Front-engined; Line 24: Water-cooled; Line 32: Independent front suspension.

1 **Vehicle body**
Example: 3/5 Ko 5 indicates a three- or five-door station wagon with five seats. The specifications are listed for the first alternative (3>/5 or 5>/3).

2 **Brake horsepower (BHP)**
The net engine output in kW/HP as per DIN 70020. If DIN data are not given, horsepower is referred to ECE (European Commission for Europe), SAE (Society of Automotive Engineers), or JIS (Japan Industry Standard)

4 **Maximum speed**
The maximum speed (in accordance with DIN 70 020) which the vehicle maintains over a measured distance of 1 km.

6 **Curb weight**
The weight of the vehicle when ready for use, excluding the driver.

8 **Weight-to-power ratio**
Calculated from the brake horsepower and weight of the vehicle when ready for use, excluding the driver (curb weight). The smaller the weight-to-power ratio, the greater the acceleration and climbing ability.

9 **Fuel consumption**
As specified by the EU Standard Stage II which has been in force for new vehicles since 1.1.1996, the fuel consumption is determined according to the NEDC (New European Driving Cycle) (P. 336). These tests reflect a certain degree of correspondence with real driving conditions by including "urban" (with cold start) and "out-of-town" components. The figures quoted may also originate from the Environmental Protection Agency in the USA. Some data originate from the ediorial staff (Ed.), or may still be figures determined according to DIN 70 030 (90 km/h – 120 km/h – urban).

15 **Specific power output**
Ratio of brake horsepower to swept volume (displacement). Also known as power output per liter. Conversion: 1 kW/*l* = 1.36 bhp/*l*.

16 **Engine type**
Order: Engine position, number of cylinders, cylinder configuration.
Examples: q4R indicates a transverse 4-cylinder in-line engine; 4Ve indicates 4 valves per cylinder.

18 **Alternator(14 V)**
Current rating quoted in A or power rating quoted in W.

19 **Battery(12 V)**
Battery capacity quoted in Ah or cold-discharge test current in A.

20 **Engine position/drive configuration**
In the case of mid-engine or rear-engine vehicles, the etters preceding the hyphen indicate the location of the engine, and those following it the location of the driven wheels, e.g., h-h indicates that both the engine and the driven wheels are at the rear of the vehicle. Mm-aa stands for a mid-engine vehicle with all-wheel drive, and v-h signifies a front-engine vehicle with rear-wheel drive.

21 **Transmission step-down ratio**
For the individual gears, transmission step-down ratios between engine and gearbox output shaft (standard equipment). See also line 23.

22 **Final drive ratio**
Conversion ratio between gearbox output shaft and axle drive shaft.

23 **Gearshift**
H indicates the number of forward speeds (gears). (A) indicates the number of automatic-gearbox speeds. Examples: 5 (4) signifies 5 forward gears, 4-speed automatic gearbox. 5/4 (4) signifies 5 or 4 forward gears or 4-speed automatic gearbox. Line 21 (transmission step-down ratio) always lists the first of the available options.

24 **Cooling**
Example: 6.5 means water-cooled, capacity 6.5 *l*.

32/ **Wheel suspension**
33 Example: QI-F means control arm with MacPherson strut, Sa-LI-Sf means rigid axle with trailing arm and coil spring.

34 **Tires**
See P. 630 for explanation of tire designations.

35 **Braking system**
Example: Sb-Tb-ABS indicates disk brakes front, drum brakes rear, and antilock braking system (ABS).

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	Country		Brazil		Germany		
	Vehicle make		Fiat	VW	Audi		
	Model		Palio 1.7 TD Weekend 70	Gol 1.0	A3 1.6	A3 1.9 TDi	TT 1.8
1	Body Style	–	5 Ko 5	5/3 Li 5	3/5 Li 5	5/3 Li 5	3 Co 2+2
2	Useful power at engine speed	kW (bhp) rpm	51 (69) 4500	51 (69) 5750	74 (101) 5600	66 (90) 3750	165 (224) 5900
3	Max. torque at engine speed	Nm rpm	134 2500	92 4500	145 3800	210 1900	280 2200
4	Maximum speed	km/h	164	161	188	181	243
5	Acceleration from 0 to 100 km/h	s	14.9	14.3	11	12.4	6.4
6	Curb weight	kg	1025	915	1090	1185	1395
7	Gross vehicle weight	kg	1620	1315	1650	1745	1765
8	Weight-to-power ratio	kg/kW	21.8	22.4	14.7	18	8.5
9	Fuel consumption	l/100 km	EU 5.1/8.6	(factory) 6.6/7.8	EU 5.9/10.5	EU 4.1/6.5	EU 7.3/12.3
10	Fuel octane number	RON	Diesel	92	95	Diesel	98
11	Fuel tank capacity	l	51	50	55	55	62
12	Engine swept volume	cm^3	1698	998	1595	1896	1781
13	Bore/stroke	mm	82.6/79.2	67.1/70.6	81/77.4	79.5/95.5	81/86.4
14	Compression ratio	–	20.3	10.5	10.3	19.5	8.9
15	Specific power output	kW/l	30	40.1	46.4	34.8	92.6
16	Engine type	–	q 4 R	4 R	q 4 R	q 4 R	4 R 5Ve
17	Fuel management	–	EP/V	Motronic	EI	EP/eV-T-Ic	Motronic-T-Ic
18	Alternator (14 V)	A	65/85	55	70	70/120	90
19	Battery (12 V)	Ah	60	36	44/60	80	60
20	(Engine pos.)/drive config.	–	v	v	v	v	aa
21	Transmission ratio	1st/2nd gear 3rd/4th gear 5th/6th gear	3.91/2.24 1.44/1.03 0.8	3.9/2.12 1.37/1.03 0.83	3.46/1.94 1.37/1.03 0.85	3.78/2.12 1.36/0.97 0.76	3.42/2.11 1.43/1.09 0.87/0.72
22	Final drive ratio	–	3.77	4.78	4.25	3.39	4.2
23	Gearshift	H (A)	5	5	5 (4)	5 (4)	6
24	Cooling	l	6.9	6	5	6	7
25	Vehicle length	mm	4130	3810	4150	4150	4040
26	Vehicle width	mm	1625	1650	1735	1735	1765
27	Vehicle height	mm	1520	1410	1425	1425	1350
28	Wheelbase	mm	2425	2470	2515	2515	2430
29	Track, front/rear	mm	1390/1385	1390/1380	1515/1495	1515/1495	1530/1505
30	Turning circle diameter	m	10.4	9.7	10.9	10.9	10.5
31	Ground clearance	mm	140	120	140	140	130
32	Suspension, front	–	DQl-F-Ku-Sf-Td	DQl-F	DQl-F-Sf-Ku-Td	DQl-F-Sf-Ku-Td	DQl-F-Sf-Ku-Td
33	Suspension, rear	–	Ll-Sf-Ku-Td	Vla-Ll-Sf-Td	Vla-Sf-Ku-Td	Vla-Sf-Ku-Td	Vla-Sf-Ku-Td
34	Tires	–	175/65 R 14	155 R 13	195/65 VR 15	195/65 VR 15	225/45 ZR 17
35	Braking system		Sb-Tb	Sb-Tb	4Sb-ABS-EBV	4Sb-ABS-EBV	4Sb-ABS-EBV

See PP. 898 for abbreviations and explanations.

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A4 Avant 2.4	S4 Avant 2.7 Biturbo	A6 1.9 TDi	A6 4.2	A8 3.7	A8 2.5 TDi	BMW Z3 1.9	
5 Ko 5	5 Ko 5	4 Li 5	4 Li 5	4 Li 5	4 Li 5	2 Ro 2	1
121 (165) 6000	195 (265) 5800	81 (110) 4150	220 (300) 6200	191 (260) 6000	110 (150) 4000	103 (140) 6000	2
230 3200	400 1850	235 1900	400 3000	350 3250	310 1500	180 4300	3
222	250	194	250	250	220	205	4
							5
8.6	5.7	12.6	6.9	8.1	9.9	9.5	
1360	1540	1415	1730	1510	1595	1200	6
1930	2090	1995	2310	2110	2195	1450	7
10.9	7.7	17.5	7.9	8.6	14.5	11.6	8
EU 7.5/14.9	EU 8.7/15.8	EU 4.6/7.5	EU 9.5/19	EU 8.4/16.9	EU 5.5/10.2	EU 6/11.4	9
95	98	Diesel	98	98	Diesel	95	10
62	62	70	82	90	90	51	11
2393	2671	1896	4172	3697	2496	1895	12
81/77.4	81/86.4	79.5/95.5	84.5/93	84.5/82.4	78.3/86.4	85/83.5	13
10.5	9.3	19.5	11	11	19.5	10.0	14
50.6	73	42.7	52.7	51.7	44.1	54.3	15
6 V 5Ve	6 V 5Ve	4 R	8 V 5Ve	8 V 5Ve	6 V 4Ve	4 R 4Ve	16
Motronic	Motronic-T-Ic	EP/eV-T-Ic	Motronic	Motronic	EP/eV-T-Ic	Motronic	17
90	120	120	120	120	120	80	18
70	70	80	92	92	92	70	19
v	aa	v	aa	v-ESP	v-ESP	h	20
3.5/1.94 1.3/0.94 0.79	3.5/1.89 1.23/0.97 0.81/0.68	3.5/1.94 1.23/0.84 0.68	3.57/2.2 1.51/1 0.8	3.67/2 1.41/1 0.74	3.75/2.06 1.32/0.93 0.73/0.6	4.23/2.52 1.66/1.22 1	21
3.89	4.11	3.7	2.91	3.09	3.88	3.45	22
5 (5)	6	5 (4)	(5)	(5)	6 (5)	5 (4)	23
8.5	6	7	11	11	10	6.5	24
4480	4485	4795	4835	5035	5035	4025	25
1735	1735	1810	1850	1880	1880	1690	26
1410	1400	1450	1450	1440	1440	1290	27
2620	2605	2760	2760	2880	2880	2445	28
1500/1480	1500/1490	1540/1570	1580/1590	1540/1580	1595/1585	1410/1425	29
11.1	11.4	11.7	11.7	12.3	12.3	10.0	30
110	110	120	120	120	120	110	31
DQu-Sf-Ku-Td	DQu-Sf-Ku-Td	Vl-F-Ku-Td	Vl-F-Ku-Td	Vl-Sf-Ku-Td	Vl-Sf-Ku-Td	DQl-F-Ku-Td	32
Vla-Ll-Sf-Ku-Td	DQu-Sf-Ku-Td	Vla-Sf-Ku-Td	Vla-DDQl-Sf-Ku-Td	Tl-Ql-Sf-Ku-Td	Tl-Ql-Sf-Ku-Td	Sl-Sf-Ku-Td	33
195/65 R 15	225/45 R 17	195/65 VR 15	235/50 R 16	225/60 R 16	225/60 R 16	205/60 R 15	34
4Sb-ABS-EBV	4Sb-ABS-EBV	4Sb-ABS-EBV	4Sb-ABS-EBV	4Sb-ABS	4Sb-ABS	4Sb-ABS	35

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	Country		Germany				
	Vehicle make		BMW (Continued)				
	Model		Z3 3.2	316i Compact	318tds Compact	320d	323i Touring
1	Body style	–	3 Co 2	3 Cp 5	3 Cp 5	4 Li 5	5 Ko 5
2	Useful power at engine speed	kW (bhp) rpm	236 (321) 7400	77 (105) 5300	66 (90) 4400	100 (136) 4000	125 (170) 5500
3	Max. torque at engine speed	Nm rpm	236 7400	165 2500	190 2000	280 1750	245 3950
4	Maximum speed	km/h	250	190	175	207	223
5	Acceleration from 0 to 100 km/h	s	5.4	11.9	13.9	9.9	8.3
6	Curb weight	kg	1390	1175	1215	1375	1365
7	Gross vehicle weight	kg	1660	1635	1675	1875	1865
8	Weight-to-power ratio	kg/kW	5.9	15.3	18.4	13.8	10.5
9	Fuel consumption	l/100 km	EU 7.9/16.6	EU 5.8/10.8	EU 4.9/8.5	EU 4.7/7.4	EU 7.3/15
10	Fuel octane number	RON	98	95	Diesel	Diesel	95
11	Fuel tank capacity	l	51	55	55	63	62
12	Engine swept volume	cm^3	3201	1895	1665	1951	2494
13	Bore/stroke	mm	86.4/91	85/83.5	80/82.5	84/88	84/75
14	Compression ratio	–	10.5	9.7	22	19	10.5
15	Specific power output	kW/l	73.7	40.6	39.6	51.2	50.1
16	Engine type	–	6 R 4Ve	4 R	4 R	4 R 4Ve	6 R 4Ve
17	Fuel managment	–	EI	EI	EP/eV-T	EP/eV-T-Ic	EI
18	Alternator (14 V)	A	115	80	95	120	80
19	Battery (12 V)	Ah	70	50	65	80	65
20	(Engine pos.)/drive config.	–	h	h	h	h	h
21	Transmission ratio	1st/2nd gear 3rd/4th gear 5th/6th gear	4.21/2.49 1.66/1.24 1	4.23/2.52 1.66/1.22 1	5.43/2.95 1.81/1.26 1	5.09/2 8 1.76/1 25 1	4.21/2.49 1.66/1.24 1
22	Final drive ratio	–	3.15	3.23	2.65	2.47	2.93
23	Gearshift	H (A)	5 (4)	5	5	5 (54)	5 (5)
24	Cooling	l	10.75	6.5	7.5	7	10.5
25	Vehicle length	mm	4025	4210	4210	4470	4430
26	Vehicle width	mm	1740	1700	1700	1740	1710
27	Vehicle height	mm	1280	1390	1390	1420	1390
28	Wheelbase	mm	2460	2700	2700	2725	2700
29	Track, front/rear	mm	1420/1490	1420/1425	1420/1425	1480/1490	1410/1430
30	Turning circle diameter	m	10.4	10.8	10.8	10.5	10.8
31	Ground clearance	mm	110	108	108	130	110
32	Suspension, front	–	DQl-F-Ku-Td	DQl-F-Ku-Td	DQl-F-Ku-Td	DQl-F-Ku-Td	DQl-F-Ku-Td
33	Suspension, rear	–	Sl-Sf-Ku-Td	Sl-Sf-Ku-Td	Sl-Sf-Ku-Td	Ma-Sf-Ku-Td	Ql-Ll-Sf-Ku-Td
34	Tires	–	1)	185/65 HR 15	185/60 HR 15	195/65 HR 15	205/60 VR 15
35	Braking system		4Sb-ABS	4Sb-ABS	4Sb-ABS	4Sb-ABS	4Sb-ABS

1) Front 225/45 R 17; Rear 245/40 R 17

see P. 898 for abbreviations and explanations.

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				Ford			
328i	530d	740d	840Ci	Ka 1.3	Puma 1.4	Fiesta 1.8D	
2 Ca 4	4 Li 5	4 Li 5	2 Co 4	3 Li 4	3 Co 2+2	5/3 Li 5	1
142 (193) 5300	135 (184) 4000	175 (238) 4000	210 (286) 5700	44 (60) 5000	66 (90) 5500	44 (60) 4800	2
280 3950	390 1750	560 1750	420 3900	103 2500	123 4000	105 2500	3
230	225	242	250	155	180	158	4
							5
7.7	8	6.4	6.6	15.4	11.9	17.6	
1430	1575	1960	1780	890	1035	1015	6
1830	2115	2495	2200	1265	1400	1510	7
10.1	11.7	11.2	8.5	20.2	15.7	24.2	8
EU 6.8/13.5	EU 5.7/9.8	EU 7/14.6	EU 9/19.2	EU 5.5/8.8	EU 5.9/9.5	EU 4.9/6.7	9
95	Diesel	Diesel	95	95	95	Diesel	10
62	70	85	90	42	42	42	11
2793	2926	3901	4398	1299	1388	1753	12
84/84	84/88	84/88	92/82.7	74/75.5	76/76.5	82.5/82	13
10.2	18	18	10	9.5	10.3	21.5	14
50.8	46.1	44.9	47.7	33.9	47.5	25.1	15
6 R 4Ve	6 R 4Ve	8 V 4Ve	8 V 4Ve	4 R	4 R 4Ve	q 4 R	16
EI	EP/CR-T-Ic	EP/CR-T-Ic	Motronic	EI	EI	EP/V	17
70/80	120	150	120	55	55	55	18
70	95	110	130	43	43	63	19
h	h	h	h	v	v-ASR	v	20
4.21/2.49 1.66/1.24 1	5.24/2.91 1.81/1.27 1	3.55/2.24 1.55/1 0.79	4.23/2.51 1.67/1.23 1/0.83	3.15/1.93 1.28/0.95 0.76	3.15/1.93 1.38/1.03 0.76	3.58/1.93 1.28/0.95 0.76	21
2.93	2.35	2.65	2.93	4.06	4.19	3.84	22
5 (5)	5 (5)	(5)	6 (5)	5	5	5	23
10.5	9.2	16	12	5.3	6	9.3	24
4435	4775	4985	4780	3620	3985	3830	25
1710	1800	1860	1855	1640	1670	1635	26
1350	1430	1430	1340	1400	1350	1330	27
2700	2830	2930	2685	2450	2445	2445	28
1410/1420	1510/1525	1550/1570	1555/1560	1395/1410	1450/1410	1430/1375	29
10.8	11.3	11.6	11.5	10.3	10.4	10.3	30
110	120	120	140	140	–	140	31
DQI-F-Ku-Td	DG-Sf-Td	QI-F-Ku-Td	DQI-F-Sf-Ku-Td	DQI-F-Ku-Td	DQI-F-Ku-Td	DQI-F-Ku-Td	32
QI-LI-Sf-Ku-Td	Ma-Sf-Td	Ma-Sf-Ku-Td	Ma-Sf-Ku-Td	Vla-Sf-Td	Vla-LI-Sf-Td	Vla-LI-Sf-Td	33
205/60 WR 15	225/65 R 15	235/60 WR 16	235/50 WR 16	155/70 R 13	195/50 R 15	155/70 R 13	34
4Sb-ABS	4Sb-ABS	4Sb-ABS	4Sb-ABS	Sb-Tb	Sb-Tb-ABS	Sb-Tb	35

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Country			Germany				
Vehicle make			Ford (Continued)				
Model			Escort 1.6	Focus 1.8	Focus 1.8TD Sedan	Mondeo 2.0	Cougar 2.5i
1	Body style	–	5 Li 5	5/3 Li 5	4 Li 5	4/5 Li 5	3 Co 5
2	Useful power at engine speed	kW (bhp) rpm	66 (90) 5250	85 (116) 5750	67 (91) 4000	96 (131) 5600	125 (170) 6250
3	Max. torque at engine speed	Nm rpm	138 3500	158 3750	200 2000	178 4000	220 4250
4	Maximum speed	km/h	177	198	184	206	225
5	Acceleration from 0 to 100 km/h	s	12.2	10.2	12.6	9.9	8.1
6	Curb weight	kg	1080	1125	1205	1245	1390
7	Gross vehicle weight	kg	1625	1620	1620	1815	1825
8	Weight-to-power ratio	kg/kW	16.4	13.4	18	12.9	11.1
9	Fuel consumption	l/100 km	EU 6.4/10.7	EU 6/10.3	EU 4/6.7	EU 5.9/11.6	EU 7.3/13.4
10	Fuel octane number	RON	95	95	Diesel	95	95
11	Fuel tank capacity	l	55	55	55	61.5	60
12	Engine swept volume	cm³	1597	1796	1753	1988	2544
13	Bore/stroke	mm	76/88	80.6/88	82.5/82	84.8/88	82.4/79.5
14	Compression ratio	–	10.3	10	19.4	10	9.7
15	Specific power output	kW/l	41.3	47.3	38.2	48.3	49.1
16	Engine type	–	q 4 R 4Ve	q 4 R 4Ve	q 4 R	q 4 R 4Ve	q 6 V 4Ve
17	Fuel managment	–	EI	EI	EP/eV-T-Ic	EI	EI
18	Alternator (14 V)	A	90	90	–	90	90
19	Battery (12 V)	Ah	48	48	–	48	48
20	(Engine pos.)/drive config.	–	v	v	v	v	v
21	Transmission ratio	1st/2nd gear 3rd/4th gear 5th/6th gear	3.58/1.91 1.28/0.95 0.76	3.15/1.93 1.28/0.95 0.76	3.66/2.05 1.26/0.86 0.67	3.42/2.14 1.45/1.03 0.77	3.42/2.14 1.48/1.11 0.85
22	Final drive ratio	–	3.82	4.06	3.56	3.84	3.82
23	Gearshift	H (A)	5	5	5	5 (4)	5 (4)
24	Cooling	l	6.6	6.6	6	6.6	7.5
25	Vehicle length	mm	4140	4150	4360	4560	4700
26	Vehicle width	mm	1700	1700	1700	1750	1770
27	Vehicle height	mm	1400	1480	1480	1420	1320
28	Wheelbase	mm	2525	2615	2615	2705	2705
29	Track, front/rear	mm	1440/1460	1495/1485	1495/1485	1505/1485	1505/1490
30	Turning circle diameter	m	10.5	10.9	10.9	10.9	10.9
31	Ground clearance	mm	–	–	–	120	–
32	Suspension, front	–	DQl-F-Ku-Td	DQl-F-Sf-Ku-Td	DQl-F-Sf-Ku-Td	DQl-F-Ku-Td	DQl-F-Sf-Ku-Td
33	Suspension, rear	–	Vla-Ll-Sf-Td	Ql-Ll-Sf-Td	Ll-Ql-Sf-Td	DQu-Ll-Ku-Td	Ql-Ll-F-Sf-Ku-Td
34	Tires	–	175/70 R 13	185/65 R 14	185/65 R 14	185/65 R 14	205/60 R 15
35	Braking system		Sb-Tb	4Sb-ABS	4Sb-ABS	4Sb-ABS	4Sb

See PP. 898 for abbreviations and explanations.

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	Mercedes-Benz						
Galaxy 1.9 TDI	A 140	A 160 CDI	SLK 200	SLK 230 Kompressor	C 240 T	C 220 CDI	
5 Gli 5-8	5 Li 5	5 Li 5	2 Ro 2	2 Ro 2	5 Ko 5	4 Li 5	1
81 (110) 4150	60 (82) 5000	44 (60) 3600	100 (136) 5500	142 (193) 5300	125 (170) 5900	92 (125) 4200	2
235 1900	130 3750	160 1500	190 3700	280 2500	225 3000	300 1800	3
172	170	150	208	231	216	198	4
							5
10.8	12.9	18.8	9.3	7.7	9.5	10.5	
1645	1020	1070	1195	1250	1470	1340	6
2400	1480	1530	1530	1585	1970	1890	7
20.2	17	23.9	12	8.8	11.4	14.6	8
EU 5.4/8.7	EU 5.6/9.8	EU 4.3/6	EU 6.9/12.9	EU 7/13.3	EU 7.5/14.3	EU 5/8.2	9
Diesel	95	Diesel	95	95	95	Diesel	10
70	54	54	53	53	62	62	11
1896	1397	1689	1998	2295	2398	2151	12
79.5/95.5	80/69.5	80/84	89.9/78.7	90.9/88.4	83.2/73.5	88/88.4	13
19.5	11	19	10.4	8.8	10	19	14
42.7	42.9	26	50.1	61.9	52.1	42.8	15
4 R	q 4 R	q 4 R 4Ve	4 R 4Ve	4 R 4Ve	6 V 3Ve	4 R 4Ve	16
EP/eV-T-Ic	EI	EP/CR-T	Motronic	Motronic-Kom-Ic	Motronic	EP/CR-T-Ic	17
90	90	90	90	90	115	90/125	18
61	46	62	62	62	74	74	19
v	v-ASR-ESP	v-ASR-ESP	h-ASR	h-ASR	h-ASR	h-ASR	20
3.58/2.05 1.35/0.92 0.67	3.27/1.92 1.26/0.88 0.7	3.63/2.09 1.31/0.9 0.72	3.91/2.17 1.37/1 0.8	3.86/2.18 1.38/1 0.8	3.86/2.18 1.38/1 0.8	4.1/2.18 1.38/1 0.8	21
4.36	4.06	3.05	3.91	3.46	3.67	3.07	22
5 (4)	5 (5)	(5) 5	5 (5)	5 (5)	5 (5)	5 (5)	23
11.7	6	6	8	8.5	8.5	8.5	24
4620	3575	3575	4000	4000	4515	4515	25
1810	1720	1720	1720	1720	1725	1725	26
1730	1580	1580	1290	1290	1430	1430	27
2835	2425	2425	2400	2400	2690	2690	28
1520/1505	1505/1450	1525/1470	1490/1470	1490/1470	1500/1465	1500/1465	29
11.7	10.7	10.7	10.6	10.6	10.8	10.8	30
150	–	–	150	150	150	150	31
QI-F-Ku-Td	DQI-F-Sf-Ku-Td	DQI-F-Sf-Ku-Td	DDQI-Sf-Ku-Td	DDQI-Sf-Ku-Td	DDQI-Sf-Ku-Td	DDQI-Sf-Ku-Td	32
SI-Sf-Ku-Td	VIa-Sf-Ku-Td	VIa-Sf-Ku-Td	Ra-Sf-Ku-Td	Ra-Sf-Ku-Td	Ra-Sf-Ku-Td	Ra-Sf-Ku-Td	33
195/65 R 15	195/50 R 15	155/70 R 15	205/60 R 15	205/55 R 16	195/65 R 15	195/65 R 15	34
4Sb-ABS	Sb-Tb-ABS	Sb-Tb-ABS	4Sb-ABS	4Sb-ABS	4Sb-ABS	4Sb-ABS	35

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	Country		Germany				
	Vehicle make		Mercedes-Benz (Continued)				
	Model		CL 500	E 280 T 4matic	E 430	CLK 200 Kompressor	S 320
1	Body style	–	2 Co 4/5	5 Ko 5-7	4 Li 5	2 Co 4	4 Li 5
2	Useful power at engine speed	kW (bhp) rpm	225 (306) 5600	150 (204) 5700	205 (279) 5750	141 (192) 5300	165 (224) 3199
3	Max. torque at engine speed	Nm rpm	460 2700	270 3000	400 3000	270 2500	315 3000
4	Maximum speed	km/h	250	220	250	233	240
5	Acceleration from 0 to 100 km/h	s	6.5	9.9	6.6	8.4	8.2
6	Curb weight	kg	1790	1665	1575	1320	1695
7	Gross vehicle weight	kg	2320	2290	2180	1820	2300
8	Weight-to-power ratio	kg/kW	8	9.8	7.7	9.4	10.3
9	Fuel consumption	l/100 km	EU -/13.4	EU 8.4/16	EU 8.1/16.8	EU 7.2/13.6	EU 8.2/17.1
10	Fuel octane number	RON	95	95	95	98	95
11	Fuel tank capacity	l	88	65	65	62	88
12	Engine swept volume	cm^3	4966	2799	4266	1998	3199
13	Bore/stroke	mm	97/84	89.9/73.5	89.9/84	89.9/78.7	89.9/84
14	Compression ratio	–	10	10	10	8.5	10
15	Specific power output	kW/l	45.3	53.6	48	70.6	51.6
16	Engine type	–	8 V 3Ve	6 V 3Ve	8 V 3Ve	4 R 4Ve	6 V 3Ve
17	Fuel managment	–	Motronic	Motronic	Motronic	Motronic-Kom-fc	Motronic
18	Alternator (14 V)	A	143/150	115	150	90	150
19	Battery (12 V)	Ah	100	74	100	62	74
20	(Engine pos.)/drive config.	–	h-ASR-ESP	aa	h-ASR-ESP	h-ASR	h-ASR-ESP
21	Transmission ratio	1st/2nd gear 3rd/4th gear 5th/6th gear	3.59/2.19 1.41/1 0.83	3.86/2.18 1.38/1 0.8	3.59/2.19 1.41/1 0.83	3.86/2.18 1.38/1 0.8	3.93/2.41 1.49/1 0.83
22	Final drive ratio	–	2.82	3.7	2.82	3.67	3.07
23	Gearshift	H (A)	(5)	5 (5)	(5)	5 (5)	(5)
24	Cooling	l	15	11	9	8	11
25	Vehicle length	mm	4995	4820	4800	4565	5040
26	Vehicle width	mm	1855	1800	1800	1720	1855
27	Vehicle height	mm	1400	1440	1440	1370	1450
28	Wheelbase	mm	2885	2830	2830	2690	2965
29	Track, front/rear	mm	1575/1580	1540/1540	1540/1540	1505/1475	1575/1575
30	Turning circle diameter	m	11.5	11.3	11.3	10.7	11.7
31	Ground clearance	mm	150	160	160	150	150
32	Suspension, front	–	DQl-Ql-Ku-Td-Lue	DDQl-Sf-Ku-Td	DDQl-Sf-Ku-Td	DDQl-Sf-Ku-Td	DQl-Ql-Ku-Td-Lue
33	Suspension, rear	–	Ra-Ku-Td-Lue	Ra-Sf-Ku-Td	Ra-Sf-Ku-Td	Ra-Sf-Ku-Td	Ra-Ku-Td-Lue
34	Tires	–	225/55 R 17	215/55 R 16	215/55 R 16	205/55 R 16	225/60 R 16
35	Braking system		4Sb-ABS	4Sb-ABS	4Sb-ABS	4Sb-ABS	4Sb-ABS

See PP. 898 for abbreviations and explanations.

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		Opel					
SL 600	ML 230	Corsa 1.0i	Corsa 1.6i	Astra 1.2i	Astra 2.0i	Zafira 1.8i	
2 Ro 2+2	5 Ge 5-7	3/5 Li 5	3/5 Li 5	3/4/5 Li 5	3/4/5 Li 5	5 Gli 5-7	1
290 (394) 5200	110 (150) 5400	40 (54) 5600	78 (106) 6000	48 (65) 5600	100 (136) 5600	85 (116) 5400	2
570 3800	220 3800	82 2800	148 4000	110 4000	188 3200	170 3400	3
250	180	150	192	165	208	184	4
							5
6.1	12.3	18	10.5	16	9	12	
1980	1855	865	980	1035	1170	1320	6
2320	2650	1360	1435	1555	1695	2040	7
6.8	16.2	21.6	12.5	21.6	11.7	11.5	8
EU 10.9/23.3	EU 10/17.1	EU 4.6/7.4	EU 6.1/10.8	EU 5/8	EU 6.6/12	EU 6.8/11.5	9
95	95	95	95	95	95	95	10
80	70	46	46	52	52	58	11
5987	2295	973	1598	1199	1998	1796	12
89/80.2	90.9/88.4	72.5/78.6	79/81.5	72.5/72.6	86/86	80.5/88.2	13
10	10.4	10.1	10.5	10.1	10.8	10.5	14
48.4	47.9	41.1	48.8	40	50	47.3	15
12 V 4Ve	4 R 4Ve	3 R 4Ve	4 R 4Ve	4 R 4Ve	4 R 4Ve	4 R 4Ve	16
Motronic	Motronic	Motronic	EI	Motronic	EI	EI	17
150	90	70	70	70	70	70/100	18
100	62	36	44	36	44	55	19
h-ASR-ESP	aa	v	v	v	v	v	20
3.59/2.19 1.41/1 0.83	3.86/2.18 1.38/1 0.8	3.73/2.14 1.41/1.12 0.89	3.73/2.14 1.41/1.12 0.89	3.73/2.14 1.41/1.12 0.89	3.58/2.14 1.48/1.12 0.89	3.73/2.14 1.41/1.12 0.89	21
2.65	4.73	3.74	3.74	3.94	3.74	4.19	22
(5)	5	5 (4)	5 (4)	5	5 (4)	5 (4)	23
18	8.5	4.3	5.6	5.2	7.2	6.5	24
4500	4590	3740	3740	4111	4111	4320	25
1810	1830	1610	1610	1710	1710	1740	26
1290	1780	1420	1420	1430	1430	1630	27
2515	2820	2445	2445	2610	2610	2690	28
1535/1525	1560/1560	1385/1390	1385/1390	1485/1460	1485/1460	1470/1490	29
11.3	11.9	10.1	10.1	10.8	10.8	11.2	30
150	200	140	140	130	130	150	31
DQI-Db-Sf-Ku-Td	DDQI-Tf-Ku-Td	LI-QI-F-Sf-Td	LI-QI-F-Sf-Td	DQI-F-Sf-Ku-Td	DQI-F-Sf-Ku-Td	DQI-F-Sf-Ku-Td	32
Ra-Sf-Ku-Td	DDQI-Sf-Ku-Td	Vla-LI-Sf-Td	Vla-LI-Sf-Td	Vla-LI-Sf-Ku-Td	Vla-LI-Sf-Ku-Td	Vla-LI-Sf-Ku-Td	33
245/45 R 17	225/75 R 16	165/70 R 13	185/60 R 14	175/70 R 14	195/60 R 15	195/65 R 15	34
4Sb-ABS	4Sb-ABS	Sb-Tb	Sb-Tb	Sb-Tb	4Sb	4Sb-ABS	35

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	Country		Germany				
	Vehicle make		Opel (continued)				
	Model		Vectra 2.5i	Omega 2.5 TD	Sintra 2.2i	Frontera 3.2i	Monterey 3.0 TD
1	Body style	–	4/5 Li 5	4 Li 5	5 Gli 5-8	5/3 Ge 5	3/5 Ge 5
2	Useful power at engine speed	kW (bhp) rpm	125 (170) 5800	96 (131) 4500	104 (141) 5400	151 (205) 5400	117 (159) 3900
3	Max. torque at engine speed	Nm rpm	230 3200	250 2200	202 2600	290 3000	333 2000
4	Maximum speed	km/h	230	200	183	192	160
5	Acceleration from 0 to 100 km/h	s	8.5	12	12.7	9.7	15.8
6	Curb weight	kg	1355	1550	1655	1780	1965
7	Gross vehicle weight	kg	1850	2120	2345	2450	2730
8	Weight-to-power ratio	kg/kW	10.6	16	15.9	11.8	16.4
9	Fuel consumption	l/100 km	EU 7.3/14.1	EU 6.1/10.9	EU 8.1/12.5	EU 10.1/17	EU 9/14
10	Fuel octane number	RON	95	Diesel	–	95	Diesel
11	Fuel tank capacity	l	60	75	70	65/75	85
12	Engine swept volume	cm^3	2498	2497	2198	3165	2999
13	Bore/stroke	mm	81.6/79.6	80/82.8	86/94.6	93.4/77	95.4/104.9
14	Compression ratio	–	10.8	22.5	10.5	9.4	19
15	Specific power output	kW/l	50	38.4	47.3	47.7	39
16	Engine type	–	6 V 4Ve	6 R	q 4 R 4Ve	6 V 4Va	4 R
17	Fuel management	–	Motronic	EP/eV-T-Ic	EI	Motronic	EP/CR-T-Ic
18	Alternator (14 V)	A	100	105	105	1013 W	60
19	Battery (12 V)	Ah	66	85	54	60	150
20	(Engine pos.)/drive config.	–	v	h	v	aa	aa
21	Transmission ratio	1st/2nd gear 3rd/4th gear 5th/6th gear	3.58/2.02 1.35/0.98 0.81	3.81/2.11 1.34/1 0.76	3.67/1.76 1.18/0.89 0.7	3.77/2.25 1.4/1 0.81	3.77/2.25 1.4/1 0.81
22	Final drive ratio	–	3.84	3.45	4.05	4.3	4.56
23	Gearshift	H (A)	5 (4)	5 (4)	5	5 (4)	5
24	Cooling	l	7.7	10.2	8.5	11.1	5.8
25	Vehicle length	mm	4495	4785	4670	4660	4795
26	Vehicle width	mm	1710	1785	1830	1785	1835
27	Vehicle height	mm	1420	1450	1780	1740	1840
28	Wheelbase	mm	2635	2730	2850	2700	2760
29	Track, front/rear	mm	1465/1470	1515/1530	1560/1610	1515/1520	1515/1520
30	Turning circle diameter	m	11.3	11.0	11.4	12.1	12.4
31	Ground clearance	mm	140	140	160	230	210
32	Suspension, front	–	DQl-F-Sf-Ku-Td	DQl-F-Sf-Ku-Td	DQl-F-Sf-Ku-Td	DDQl-Tf-Ku-Td	DDQl-Tf-Ku-Td
33	Suspension, rear	–	DQu-Sf-Ku-Td	Ql-Sl-Sf-Ku-Td	Vla-Ll-Sf-Ku-Td	Sa-Ll-?-Sf-Ku-Td	Sa-Ll-P-Ku-Td
34	Tires	–	195/65 R 15	205/65 R 15	205/65 HR 15	245/70 R 16	215/80 R 16
35	Braking system		4Sb-ABS	4Sb-ABS	4Sb-ABS	4Sb-ABS	4Sb-ABS

See PP. 898 for abbreviations and explanations.

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Porsche			VW (Volkswagen)				
Boxster 2.5i	911 Carrera 3.4	911 GT3 3.6	Lupo 1.0	Lupo 1.2 TDI	Polo 1.4	Polo Variant 1.9 TDI	
2 Ca 2	2 Co/Ca 2+2	2 Co/Ca 2+2	3 Li 4-5	3 Li 4-5	3/5 Li 5	5 Ko 5	1
150 (204) 6000	221 (300) 6800	265 (360) 6800	37 (50) 5000	45 (61) 4000	74 (101) 6000	66 (90) 4000	2
245 4500	350 4600	370 5000	86 3000	140 1800	128 4400	210 1900	3
240	280	> 280	152	165	188	180	4
							5
6.9	5.2	4.8	17.9	16.5	10.5	12.5	
1250	1320	1320	895	780	950	1135	6
1560	1720	1720	1350	1200	1450	1595	7
8.3	6	6	24.2	17.3	12.8	17.2	8
EU 7.1/14.3	EU 8.5/17.2	(Ed.) 10/22	EU 4.8/7.6	EU 2.7/3.7	EU 5.4/9.6	EU 4.1/6.4	9
98	98	98	95	Diesel	95	Diesel	10
57	64	64	34	34	45	45	11
2480	3387	3600	999	1196	1390	1896	12
85.5/72	96/78	100/76.4	67.1/70.6	76.5/86.7	76.5/75.6	79.5/95.5	13
11	11.3	11.3	10.7	19.5	10.5	19.5	14
60.5	65.2	73.6	37	37.6	53.2	34.8	15
6 Bx 4Ve	6 Bx 4Ve	6 Bx 4Ve	q 4 R	q 3 R	4 R 4Ve	4 R	16
Motronic	Motronic	Motronic	EI	EP/UIS-T-Ic	EI	EP/eV-T-Ic	17
115	120	120	70	70	70	90	18
60/75	70	70	36	36	44	61	19
h	h	h	v	v	v	v	20
3.5/2.12 1.43/1.03 0.79	3.82/2.2 1.52/1.22 1.02/0.84	3.82/2.15 1.56/1.21 0.97/0.75	3.46/1.96 1.36/1.05 0.86	3.46/1.96 1.82/0.81 0.64	3.45/2.1 1.45/1.1 0.89	3.3/1.94 1.31/0.92 0.72	21
3.89	3.44	3.44	4.23	3.33	3.86	3.16	22
5 (5)	6 (5)	6 (5)	5	5	5	5	23
–	22.5	22.5	5	5	5.5	6.5	24
4315	4430	4430	3525	3525	3715	4140	25
1780	1765	1780	1640	1640	1655	1640	26
1290	1300	1270	1460	1460	1420	1460	27
2415	2350	2350	2325	2325	2405	2440	28
1465/1525	1455/1500	1465/1480	1390/1400	1390/1400	1350/1380	1430/1390	29
10.9	12.1	12.1	9.8	9.8	10.4	10.9	30
–	–	–	110	110	110	110	31
QI-LI-F-Sf-Ku-Td	DQI-F-Sf-Ku-Td	DQI-F-Sf-Ku-Td	DQI-F-Sf-Td	DQI-F-Sf-Td	DQI-F-Sf-Ku-Td	DQI-F	32
QI-LI-F-Sf-Ku-Td	Ma-Sf-Ku-Td	Ma-Sf-Ku-Td	Vla-LI-Sf-Td	Vla-LI-Sf-Td	Vla-LI-Sf-Ku-Td	Vla-LI-Sf-Td	33
205/55 ZR 16	205/50 ZR 17	225/40 ZR 18	155/70 R 13	155/65 SR 14	185/55 R 14	185/60 R 14	34
4Sb-ABS	4Sb-ABS	4Sb-ABS	Sb-Tb	Sb-Tb-ABS	Sb-Tb-ABS	Sb-Tb-ABS	35

PETITIONER'S EXHIBIT 1002
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Country			Germany				
Vehicle make			VW (Volkswagen) (continued)				
Model			New Beetle 2.0	Golf IV 2.3 VR5	Bora 1.6	Passat Variant 2.8 Syncro VR6	Passat 1.9 TDI
1	Body style		2 Li 4	3/5 Li 5	4 Li 5	5 Ko 5	4 Li 5
2	Useful power at engine speed	kW (bhp) rpm	85 (116) 5200	110 (150) 6000	74 (101) 5600	142 (193) 6000	85 (116) 4000
3	Max. torque at engine speed	Nm rpm	170 2400	205 3200	145 3800	280 3200	285 1900
4	Maximum speed	km/h	185	216	188	238	200
5	Acceleration from 0 to 100 km/h	s	10.9	8.8	11.7	7.6	10.7
6	Curb weight	kg	1230	1245	1185	1495	1320
7	Gross vehicle weight	kg	1650	1800	1720	2040	1880
8	Weight-to-power ratio	kg/kW	14.6	11.3	16	10.2	15.5
9	Fuel consumption	l/100 km	EU 6.9/11.8	EU 7.2/13.2	EU 6/10.6	EU 8/15.2	EU 4.3/7
10	Fuel octane number	RON	95	95	95	95	Diesel
11	Fuel tank capacity	l	55	55	55	62	62
12	Engine swept volume	cm^3	1984	2324	1595	2771	1896
13	Bore/stroke	mm	82.5/92.8	81/90.2	81/77.4	82.5/86.4	79.5/95.5
14	Compression ratio	–	10.5	10.1	10.3	10.6	18
15	Specific power output	kW/l	42.8	47.3	46.4	51.2	44.8
16	Engine type	–	q 4 R	5 VR	4 R	6 VR 5Ve	4 R
17	Fuel management	–	Motronic	EI	EI	EI	EP/UIS-T-Ic
18	Alternator (14 V)	A	70/90	70	70	90	120
19	Battery (12 V)	Ah	44/60	60	44/60	320 A	80
20	(Engine pos.)/drive config.	–	v	v	v	aa	v
21	Transmission ratio	1st/2nd gear 3rd/4th gear 5th/6th gear	3.78/2.12 1.36/1.03 0.84	3.3/1.94 1.31/1.03 0.84	3.46/1.94 1.37/1.03 0.85	3.5/1.94 1.3/0.94 0.79	3.5/1.94 1.22/0.88 0.69
22	Final drive ratio	–	4.24	3.94	4.25	3.89	3.56
23	Gearshift	H (A)	5 (4)	5 (4)	5 (4)	5 (5)	5 (4)
24	Cooling	l	5.3	5.2	5.5	6.5	6.5
25	Vehicle length	mm	4080	4150	4375	4675	4675
26	Vehicle width	mm	1725	1735	1735	1740	1740
27	Vehicle height	mm	1500	1440	1440	1500	1460
28	Wheelbase	mm	2510	2510	2515	2705	2705
29	Track, front/rear	mm	1510/1495	1510/1490	1515/1495	1500/1500	1500/1500
30	Turning circle diameter	m	10.9	10.9	10.9	11.4	11.4
31	Ground clearance	mm	130	130	130	110	110
32	Suspension, front	–	DQl-F-Ku-Td	DQl-F-Sf-Ku-Td	DQl-F-Sf-Ku-Td	Ql-Ll-F-Sf-Ku-Td	Ql-Ll-F-Sf-Ku-Td
33	Suspension, rear	–	Vla-Sf-Ku-Td	Vla-Ll-Sf-Ku-Td	Vla-Ll-Sf-Ku-Td	Vla-Ll-Sf-Ku-Td	Vla-Ll-Sf-Ku-Td
34	Tires	–	205/55 R 16	205/55 WR16	195/65 R 15	205/55 R 16	195/65 R 15
35	Braking system		4Sb-ABS	4Sb-ABS	4Sb-ABS	4Sb-ABS	4Sb-ABS

See PP. 898 for abbreviations and explanations.

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	France						
	Citroën						
Sharan 1.8	Saxo 1.0i	Saxo 1.6i	Berlingo 1.4i	Berlingo 1.9D	Xsara 1.8i	Xsara 1.9TD	
5 Gli 5-7	5/3 Li 5	3/5 Li 5	3/4/5 Ko 5	3/4/5 Ko 5	5 Ko 5	5/3 Li 5	1
110 (150) 5700	37 (50) 6000	66 (90) 5600	55 (75) 5500	50 (68) 4600	66 (90) 5000	66 (90) 4000	2
210 1750	74 3700	137 3000	111 3400	120 2000	147 2600	196 2200	3
194	149	187	150	142	182	178	4
							5
12	19.1	11.4	14.6	16.3	13.5	12.7	
1660	825	920	1125	1185	1120	1140	6
2400	1240	1360	1780	1840	1650	1690	7
15.1	21.8	13.9	20.5	23.7	16.2	17.2	8
EU 7.9/13.9	EU 5.1/8.3	EU 5.8/9.9	EU 5.9/9.2	EU 5.4/8.6	EU 6.2/11	EU 5.2/8.8	9
95	95	95	95	Diesel	95	Diesel	10
70	45	45	55	55	54	54	11
1781	954	1587	1361	1905	1762	1905	12
81/86.4	70/62	78.5/82	75/77	83/88	83/81.4	83/88	13
9.5	9.4	9.6	10.2	23	9.5	21.8	14
61.7	38.8	40.9	40.4	26.2	37.4	34.6	15
4 R 5Ve	q 4 R	q 4 R	q 4 R	q 4 R	q 4 R	q 4 R	16
EI-T-Ic	Mono-Motronic	Motronic	EI	EP/V	EI	EP/V-T-Ic	17
70/90	50	60/70	60/70	70/80	60/70	70/80	18
280 A	250 A	250 A	200 A	300/400 A	250 A	300/400 A	19
v	v	v	v	v	v	v	20
3.58/2.05 1.35/0.97 0.81	3.42/1.81 1.28/0.98 0.77	3.42/1.95 1.36/1.05 0.85	3.42/1.95 1.36/1.05 0.85	3.42/1.95 1.36/1.05 0.85	3.45/1.87 1.28/0.95 0.8	3.45/1.87 1.15/0.83 0.66	21
4.06	4.29	3.56	4.54	4.53	3.67	4.05	22
5 (4)	5	5	5/4	5	5	5	23
6.5	6.1	6.1	6.5	8.5	7.5	9	24
4620	3720	3740	4110	4110	4355	4355	25
1810	1595	1620	1720	1720	1790	1790	26
1730	1370	1360	1800	1800	1420	1390	27
2845	2385	2385	2690	2690	2540	2540	28
1510/1500	1370/1300	1400/1320	1420/1440	1420/1440	1420/1430	1435/1440	29
11.7	10.0	10.9	10.7	10.7	11.4	11.4	30
150	120	120	140	140	110	110	31
DQI-F-Sf-	DQI-F-Ku-Td	DQI-F-Ku-Td	DQI-F	DQI-F	DQI-F-Sf-Ku-Td	DQI-F-Sf-Ku-Td	32
SI-Sf-Ku-Td	LI-Tf-Ku-Td	LI-Tf-Ku-Td	LI-Tf-Td	LI-Tf-Td	Ea-LI-Tf-Ku-Td	Ea-LI-Tf-Ku-Td	33
195/65 R 15	155/70 R 13	185/55 R 14	165/70 R 14	165/70 R 14	185/65 R 14	185/65 R 14	34
4Sb-ABS	Sb-Tb	Sb-Tb	Sb-Tb	Sb-Tb	Sb-Tb	Sb-Tb	35

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Country			France				
Vehicle make			Citroën (continued)				MCC
Model			Xsara Picasso 2.0 TDI	Xantia 2.9i	XM 2.5TD	Evasion 1.9TD	Smart 0.6
1	Body style	–	5 Gli 5	5 Li 5	5 Li 5	5 Gli 5+3	2 Mc 2
2	Useful power at engine speed	kW (bhp) rpm	80 (109) 4000	143 (194) 5500	96 (130) 4300	66 (90) 4000	33 (45) 5250
3	Max. torque at engine speed	Nm rpm	250 1750	273 4000	294 2000	196 2250	70 3000
4	Maximum speed	km/h	180	230	201	160	135
5	Acceleration from 0 to 100 km/h	s	12.5	8.2	12.1	16.8	18.9
6	Curb weight	kg	1300	1470	1580	1585	720
7	Gross vehicle weight	kg	–	1950	2130	2360	980
8	Weight-to-power ratio	kg/kW	16.3	10.5	16.6	24	21.8
9	Fuel consumption	l/100 km	(Ed.) 5/8	EU 8/15.9	EU 5.9/10.9	EU 6.7/10.6	(Ed.) 4.8/7
10	Fuel octane number	RON	Diesel	95	Diesel	Diesel	95
11	Fuel tank capacity	l	–	65	80	80	22
12	Engine swept volume	cm^3	1998	2946	2446	1905	599
13	Bore/stroke	mm	86/86	87/82.6	92/92	83/88	63.5/63
14	Compression ratio	–	18	10.5	22	21.8	9.5
15	Specific power output	kW/l	40	47.5	38.8	34.6	55
16	Engine type	–	4 R	q 6 V 4Ve	4 R 3Ve	q 4 R	3 R
17	Fuel management	–	EP/CR-T-Ic	Motronic	EP/eV-T-Ic	EP/V-T-Ic	Motronic
18	Alternator (14 V)	A	60/70	90	120	70/80	40
19	Battery (12 V)	Ah	250 A	300 A	450 A	300/400 A	35
20	(Engine pos.)/drive config.	–	v	v	v	v	h-h
21	Transmission ratio	1st/2nd gear 3rd/4th gear 5th/6th gear	–	3.25/1.78 1.19/0.88 0.7	3.42/1.86 1.18/0.86 0.64	3.42/1.94 1.25/0.75 0.67	3.38/2.45 1.76
22	Final drive ratio	–	–	4.31	4.06	4.54	4.21/1.67
23	Gearshift	H (A)	5	5 (4)	5 (4)	5	2x3 [SOFTIP]
24	Cooling	l	6.5	10	13	9	3.5
25	Vehicle length	mm	4275	4525	4710	4455	2500
26	Vehicle width	mm	1750	1755	1795	1820	1515
27	Vehicle height	mm	1640	1400	1390	1710	1529
28	Wheelbase	mm	2760	2740	2850	2825	1812
29	Track, front/rear	mm	1435/1450	1500/1455	1520/1445	1535/1540	1285/1355
30	Turning circle diameter	m	12.0	12.0	12.4	12.3	8.5
31	Ground clearance	mm	150	160	140	130	131
32	Suspension, front	–	DQl-F-Sf-Ku	Hf-DQl-F-Ku	Hf-DQl-F-Ku	DQl-F-Sf-Td	DQl-Qf-Ku-Td
33	Suspension, rear	–	Ea-Ll-Tf-Ku	Hf-Ll-Ku	Hf-Ll-Ku	Vla-Ll-P-Sf-Td	De-Wa-Sf-Ku-Td
34	Tires	–	185/65 R 15	205/60 R 15	205/65 R 15	205/65 R 15	1)
35	Braking system		Sb-Tb-ABS	4Sb-ABS	4Sb-ABS	Sb-Tb	Sb-Tb-ABS

1) Front 135/70 R 15; Rear 175/55 R 15

See P. 898 for abbreviations and explanations.

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MCC	Peugeot						
Smart 0.8 TD	106 1.4i	206 2.0i	306 1.8i	306 Break 2.0 TDI	Partner 1.8i	406 Break 2.9i	
2 Mc 2	3/5 Li 5	3/5 Li 5	4/3/5 Li 5	5 Ko 5	3/4/5 Ko 5	5 Ko 5	1
30 (40) 4000	55 (75) 5500	99 (135) 6000	81 (110) 5500	66 (90) 4000	66 (90) 5000	140 (190) 5500	2
100 2000	111 3400	190 4100	155 4250	205 1900	147 2600	267 4000	3
135	178	210	193	181	160	227	4
							5
17.5	13.2	8.4	11.6	13.2	12.2	8.5	
720	815	1050	1095	1200	1170	1505	6
980	1315	1560	1560	1655	1780	2175	7
–	14.8	10.6	13.5	17.2	17.7	10.4	8
(Ed.) 3.4/5.5	EU 5.4/9.4	EU 6.2/10.9	EU 6.3/12	EU 4.4/7.2	EU 7.1/11.9	EU 8/15.9	9
Diesel	95	95	95	Diesel	95	95	10
22	45	50	60	60	55	70	11
approx. 800	1361	1997	1762	1997	1762	2946	12
–	75/77	85/88	83/81.4	85/88	83/81.4	87/82.6	13
–	10.2	10.8	10.4	18	9.25	10.5	14
37.5	40.4	49.6	46	33	37.5	47.5	15
3 R	q 4 R	q 4 R 4Ve	4 R 4Ve	4 R	q 4 R	6 V 4Ve	16
EP/CR-T-Ic	EI	EI	EI	EP/CR-T	EI	Motronic	17
40	80	70	70	90/120/150	80	157	18
35	200 A	300 A	300 A	300 A	250 A	300 A	19
h-h	v	v	v	v	v	v	20
3.38/2.45 1.76	3.42/1.8 1.28/0.98 0.77	3.46/1.87 1.36/1.05 0.86	3.46/1.87 1.36/1.05 0.79	3.46/1.87 1.15/0.83 0.66	3.42/1.95 1.36/1,05 0.85	3.25/1.78 1.19/0.88 0.7	21
4.21/1.67	3.76	3.79	3.95/3.79	3.68	4.54	4.31	22
2x3 [SOFTIP]	5 (3)	5	5	5	5	5 (4)	23
3.5	6	7.8	6.5	7	7.5	7.8	24
2500	3680	3835	4265	4345	4110	4735	25
1515	1590	1650	1690	1690	1720	1765	26
1529	1380	1430	1380	1410	1800	1500	27
1812	2385	2440	2580	2580	2690	2700	28
1285/1355	1380/1310	1435/1430	1460/1430	1460/1430	1420/1440	1500/1480	29
8.5	11.0	10.2	11.0	11.0	10.7	12.0	30
131	130	110	130	130	140	130	31
DQI-Qf-Ku-Td	DQI-F	DQI-F-Td	DQI-F	DQI-F	DQI-F	DQI-F-Ku	32
De-Wa-Sf-Ku-Td	LI-Tf-Td	LI-Tf-Td	LI-Tf-Td	LI-Tf-Td	LI-Tf-Td	Ma-Sf-Ku-Td	33
1)	165/70 R 13	185/55 R 15	185/65 R 14	195/55 R 15	195/65 R 15	205/60 VR 15	34
Sb-Tb-ABS	Sb-Tb	4Sb	Sb-Tb	4Sb	Sb-Tb	4Sb-ABS	35

PETITIONER'S EXHIBIT 1002
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	Country		France				
	Vehicle make		Peugeot (continued)			Renault	
	Model		406 2.0 TDI	605 2.0i Turbo	806 2.1 TD	Kangoo 1.1i	Clio 1.6i
1	Body style	–	4 Li 5	4 Li 5	5 Gli 5-8	4 Ko 5	3/5 Li 5
2	Useful power at engine speed	kW (bhp) rpm	80 (109) 4000	108 (147) 5300	80 (109) 4300	43 (58) 5250	79 (107) 5750
3	Max. torque at engine speed	Nm rpm	250 1750	235 2500	250 2000	93 2500	148 3750
4	Maximum speed	km/h	191	213	175	140	195
5	Acceleration from 0 to 100 km/h	s	12.5	10	14.1	17.2	9.6
6	Curb weight	kg	1410	1515	1615	1020	995
7	Gross vehicle weight	kg	1950	1935	2385	1600	1515
8	Weight-to-power ratio	kg/kW	18.6	14	20.2	23.7	12.6
9	Fuel consumption	l/100 km	EU 4.5/7.3	EU 8/14.5	6.1/8.2/9	EU 6.1/8.3	EU 6/9.4
10	Fuel octane number	RON	Diesel	95	Diesel	95	95
11	Fuel tank capacity	l	70	80	80	52	50
12	Engine swept volume	cm^3	1997	1998	2088	1149	1598
13	Bore/stroke	mm	85/88	86/86	85/92	69/76.8	79.5/80.5
14	Compression ratio	–	18	8	21.5	9.6	10
15	Specific power output	kW/l	40.1	54	38.3	37.4	49.4
16	Engine type	–	4 R	q 4 R	q 4 R 3Ve	q 4 R	q 4 R 4Ve
17	Fuel management	–	EP/CR-T-Ic	Motronic-T-Ic	EP/eV-T-Ic	El	El
18	Alternator (14 V)	A	157	80	70/80/90	60	60
19	Battery (12 V)	Ah	300 A	300 A	400 A	40	50
20	(Engine pos.)/drive config.	–	v	v	v	v	v
21	Transmission ratio	1st/2nd gear 3rd/4th gear 5th/6th gear	3.42/1.78 1.12/0.8 0.61	3.17/1.82 1.25/0.97 0.77	3.42/1.78 1.12/0.8 0.61	3.72/2.05 1.39/1.03 0.82	3.36/1.86 1.32/1.03 0.82
22	Final drive ratio	–	4.06	4.06	4.79	4.21	4.07
23	Gearshift	H (A)	5 (4)	5 (4)	5	5	5
24	Cooling	l	8.5	7.8	10.5	6.7	6.7
25	Vehicle length	mm	4600	4765	4455	3995	3775
26	Vehicle width	mm	1765	1800	1835	1660	1640
27	Vehicle height	mm	1400	1410	1710	1830	1420
28	Wheelbase	mm	2700	2800	2825	2600	2470
29	Track, front/rear	mm	1500/1480	1525/1530	1535/1540	1400/1410	1390/1370
30	Turning circle diameter	m	12.0	12.0	12.3	10.9	11.1
31	Ground clearance	mm	130	100	140	140	120
32	Suspension, front	–	DQl-F-Ku	DQl-F-Ku-Td	DQl-F-Sf-Ku-Td	DQl-F-Sf-Ku-Td	DQl-F-Ku-Td
33	Suspension, rear	–	Ma-Sf-Ku-Td	DDQl-Ql-Sf-Ku-Td	Vla-Li-P-Sf-Ku-Td	Ea-Ll-Tf-Sf-Ku-Td	Ta-Sf-Ku-Td
34	Tires	–	195/65 HR 15	205/60 VR 15	205/65 R 15	165/70 R 13	165/70 R 13
35	Braking system		4Sb-ABS	4Sb-ABS	Sb-Tb	Sb-Tb	Sb-Tb-ABS

See PP. 898 for abbreviations and explanations.

UCLA SELFEMS Collection

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Clio 1.9 D	Sport Spider	Mégane 1.4i	Mégane Break 1.9 DTI	Mégane Scénic 2.0i	Mégane Cabriolet 2.0i	Laguna Break 1.8i	
5/3 Li 5	2 Ro 2	4/5 Li 5	5 Ko 5	5 Gli 5	2 Ca 4	5 Ko 5	1
47 (64) 4500	110 (150) 6000	70 (95) 6000	72 (98) 4000	84 (114) 5400	103 (140) 5500	88 (120) 5750	2
120 2250	186 4500	127 3750	200 2000	168 4250	200 4250	165 3750	3
161	215	184	183	185	210	203	4
							5
15.4	6.9	11.8	12.3	11.1	9	11.1	
975	930	1015	1135	1270	1135	1310	6
1550	1130	1580	1685	1800	1555	2075	7
20.7	7.2	15.6	15.5	15.1	11	14.4	8
EU 4.9/7.9	EU 7.4/9.7	EU 5.2/9.2	EU 4.4/6.8	EU 7.1/12	EU 6.1/10.2	EU 6.5/10.8	9
Diesel	–	95	Diesel	95	95	95	10
50	50	60	60	60	60	66	11
1870	1998	1390	1870	1998	1998	1783	12
80/93	82.7/93	79.5/70	80/93	82.7/93	82.7/93	82.7/83	13
21.5	9.8	10	18.3	9.7	10	9.8	14
25.1	55.1	50.3	38.5	42	51.5	49.3	15
q 4 R	4 R 4Ve	4 R 4Ve	4 R	4 R	4 R 4Ve	q 4 R 4Ve	16
EP/V	EI	EI	EP/eV-T	EI	EId	EI	17
70	70	70	70	70	70	90	18
70	50	50	70	50	50	50	19
v	Mm-h	v	v	v	v	v	20
3.36/1.86 1.32/1.03 0.79	3.36/1.86 1.39/1.09 0.89	3.09/1.86 1.32/1.02 0.79	3.73/2.05 1.32/1.03 0.75	3.37/2.05 1.36/1.1 0.82	3.36/1.86 1.32/1.03 0.82	3.72/2.05 1.39/1.1 0.89	21
3.57	4.06	4.06	3.29	4.06	3.94	3.87	22
5	5	5	5	5 (4)	5	5	23
7.4	5.5	6.5	7.4	7.1	6.5	7.3	24
3775	3795	4400	4440	4130	4030	4630	25
1640	1830	1700	1700	1720	1700	1750	26
1420	1250	1420	1420	1600	1370	1470	27
2470	2340	2580	2580	2580	2470	2670	28
1405/1385	1535/1540	1450/1430	1450/1430	1450/1470	1450/1430	1480/1460	29
10.7	9.8	11.4	11.4	11.4	11.2	11.5	30
120	–	120	120	120	120	120	31
DQI-F-Ku-Td	DDQI-Ku	DQI-F-Ku	DQI-F-Ku	DQI-F-Ku	DQI-F-Ku	DQI-F-Ku	32
Ta-Sf-Ku-Td	DQI-Ku	LI-TI-Ku	LI-TI-Ku	LI-TI-Ku	LI-TI-Ku	LI-TI-Ku-Td	33
165/70 R 13	205/50 VR 16	175/65 R 14	175/65 R 14	185/65 R 15	195/50 VR 16	195/65 R 15	34
Sb-Tb	4Sb	Sb-Tb	Sb-Tb	Sb-Tb	4Sb-ABS	Sb-Tb-ABS	35

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Country			France		Great Britain		
Vehicle make			Renault (continued)		AV	Aston Martin	Bentley
Model			Safrane 2.9i	Espace 2.2 TD	Aceca 4.6i	V8 Volante 5.3i Convertible	Arnage 4.4 TI
1	Body style	–	5 Li 5	5 Gli 5-7	2 Ca 2	2 Ca 4	4 Li 5
2	Useful power	kW (bhp)	140 (190)	83 (113)	240 (326)	261 (355)	260 (354)
	at engine speed	rpm	5750	4500	5800	6000	5500
3	Max. torque	Nm	267	250	427	500	570
	at engine speed	rpm	4000	2000	4600	4300	2500
4	Maximum speed	km/h	225	175	250	> 240	240
5	Acceleration from 0 to 100 km/h	s	9.5	14.5	5.7	5.9	6.5
6	Curb weight	kg	1580	1630	1510	2050	2300
7	Gross vehicle weight	kg	2125	2510	–	2370	2750
8	Weight-to-power ratio	kg/kW	11.3	19.6	6.3	7.5	8.8
9	Fuel consumption	l/100 km	EU 8.7/16.7	EU 6.3/9.6	EU 10.5/15.1	(Ed.) 14/22	EU 13.1/23.4
10	Fuel octane number	RON	95	Diesel	98	–	–
11	Fuel tank capacity	l	80	78	90	105	94
12	Engine swept volume	cm^3	2946	2188	4601	5341	4398
13	Bore/stroke	mm	87/82.6	87/92	90.2/90	100/85	92/82.7
14	Compression ratio	–	10.5	22	9.85	9.75	8.5
15	Specific power output	kW/l	47.5	37.9	52.2	48.9	59.1
16	Engine type	–	6 V 4Ve	q 4 R 3Ve	8 V 4Ve	8 V 4Ve	8 V 4Ve
17	Fuel management	–	El	EP/eV	El	El	Motronic-T-Ic
18	Alternator (14 V)	A	75	80	90	75	140
19	Battery (12 V)	Ah	300 A	65	60	68	90
20	(Engine pos.)/drive config.	–	v	v	h	h	h-ASR
21	Transmission ratio	1st/2nd gear	3.07/1.77	3.91/2.21	3.37/1.99	2.9/1.78	3.55/2.24
		3rd/4th gear	1.19/0.87	1.39/0.98	1.32/1	1.22/1	1.55/1
		5th/6th gear	0.7	0.76	0.67	0.85	0.79
22	Final drive ratio	–	4.73	3.81	3.23	3.54	2.93
23	Gearshift	H (A)	5 (4)	5 (4)	5	5/6 (4)	(5)
24	Cooling	l	7.8	7.1	16	18	15
25	Vehicle length	mm	4770	4515	4420	4945	5390
26	Vehicle width	mm	1820	1810	1870	1920	1930
27	Vehicle height	mm	1440	1690	1300	1380	1520
28	Wheelbase	mm	2770	2700	2470	2810	3115
29	Track, front/rear	mm	1530/1490	1535/1540	1590/1570	1510/1550	1610/1610
30	Turning circle diameter	m	11.4	11.4	10.5	12.8	12.9
31	Ground clearance	mm	120	110	–	100	140
32	Suspension, front	–	DQl-F-Ku	DQl-F-Ku	DDQl-Sf-Ku-Td	DQl-Sf-Ku-Td	DDQl-Sf-Ku-Td
33	Suspension, rear	–	Ql-Sf-Ku	Sa-Ll-P-Ku-Td	Ea-DDQl-Sf-Ku-Td	De-Wa-Ll-Sf-Td	Ea-DDQl-Sf-Ku-Td
34	Tires		205/60 R 15	205/65 R 15	–[1]	255/50 ZR 18	255/55 ZR17
35	Braking system		4Sb-ABS	Sb-Tb-ABS	4Sb-ABS	4Sb-ABS	4Sb-ABS

[1] Front 235/45 ZR 17; Rear 255/40 ZR 17

See P. 898 for abbreviations and explanations.

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	Daimler	Jaguar		MG	Rolls-Royce	Rover	
Azure 6.8 Tl	V8 4.0i	S-Type 3.0i	XKR 4.0i Supercharged	MGF 1.8i	Silver Seraph 5.4i	214 1.4i	
2 Ca 4	4 Li 5	4 Li 5	2 Co 2+2	2 Ca 2	4 Li 5	5/3 Li 5	1
286 (389) 4000	216 (294) 6100	179 (243) 6800	276 (375) 6150	88 (120) 5500	240 (326) 5000	76 (103) 6000	2
750 2000	393 4250	300 4500	525 3600	166 3000	490 3900	127 3000	3
241	240	235	250	193	225	185	4
							5
6.7	7.3	7.5	5.4	9.2	7	10.7	
2610	1730	1630	1640	1060	2300	1000	6
2980	2260	2030	2010	1320	2750	1480	7
8.5	8.3	9.3	6.2	12	9.6	13.4	8
EU 13/25.7	EU 9/16.9	EU 8/14	EU 9.6/16.7	EU 5.7/10.2	EU 12.8/25.4	EU 5.4/9.9	9
–	95	95	95	95	95	95	10
108	81	69.5	75	50	94	50	11
6750	3996	2967	3996	1795	5379	1396	12
104.14/99.06	86/86	89/79.5	86/86	80/89.3	85/79	75/79	13
8	10.75	10.5	9	10.5	10	10.5	14
42.4	52.3	59	66.8	49	44.6	54.4	15
8 V	8 V 4Ve	6 V 4Ve	8 V 4Ve	4 R 4Ve	12 V	q 4 R 4Ve	16
El-T-Ic	El	El	El-Kom-Ic	El	Motronic	El	17
120	140	130	90/140	60	140	65	18
71	92	72	72/92	45	90	45	19
h-ASR	h-ASR	h-ASR	h-ASR	h	h-ASR	v	20
2.48/1.48 1/0.75 –	3.57/2.2 1.51/1 0.8	4.23/2.52 1.67/1.22 1	3.59/2.19 1.41/1 0.83	3.17/1.84 1.31/1.03 0.77	3.55/2.24 1.55/1 0.79	3.42/1.95 1.33/1.05 0.85	21
2.69	3.06	3.07	3.06	3.94	2.93	3.94	22
(4)	(5)	5 (5)	(5)	5	(5)	5	23
18	10	10	12	6	15	5.5	24
5340	5150	4860	4760	3920	5390	3970	25
1880	1800	1820	1830	1630	1930	1690	26
1480	1380	1450	1290	1260	1520	1420	27
3060	2995	2910	2590	2375	3115	2500	28
1550/1550	1500/1500	1545/1535	1550/1500	1400/1410	1610/1610	1470/1470	29
12.9	13.2	12.1	11.0	10.5	12.6	10.2	30
140	110	140	–	120	140	120	31
DQI-Sf-Ku	DDQI-Sf-Ku-Td	DDQI-Sf-Ku-Td	DDQI-Sf-Ku-Td	DQI-QI-Hv	DDQI-Sf-Ku	DQI-Sf-Ku-Td	32
Sl-Sf-Ku	QI-Sf-Ku-Td	Ea-DDQI-Sf-Ku-Td	QI-Sf-Ku-Td	QI-Hv	DDQI-Sf-Ku	Vla-Sf-Td	33
255/55 WR 17	225/60 R 16	225/55 R 16	[1]	205/50 R 15	235/65 VR 16	185/60 HR 14	34
4Sb-ABS	4Sb-ABS	4Sb-ABS	4Sb-ABS	4Sb-ABS	4Sb-ABS	Sb-Tb	35

[1]) Front 245/45 ZR 18; Rear 255/45 ZR 18

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

	Country		Great Britain			Italy	
	Vehicle make		Rover (continued)			Alfa Romeo	
	Model		420 2.0 TD	75 2.0i	75 2.0 TDI	145 1.6i T.S.	146 1.9 JTD
1	Body style	–	4/5 Li 5	4 Li 5	4 Li 5	3 Li 5	4 Li 5
2	Useful power at engine speed	kW (bhp) rpm	77 (105) 4200	110 (150) 6500	85 (116) 4000	88 (120) 6300	77 (105) 4000
3	Max. torque at engine speed	Nm rpm	210 2000	185 4000	260 2000	144 4500	255 2000
4	Maximum speed	km/h	185	210	193	195	187
5	Acceleration from 0 to 100 km/h	s	11	10.2	11.7	10.2	10.5
6	Curb weight	kg	1220	1370	1410	1165	1245
7	Gross vehicle weight	kg	1550	–	–	1685	1765
8	Weight-to-power ratio	kg/kW	15.8	12.5	16.6	13.2	15.7
9	Fuel consumption	l/100 km	EU 4.3/7.1	EU 7/13.5	(Ed.) 5/7	EU 6.5/11.1	EU 4.7/7.6
10	Fuel octane number	RON	Diesel	98	Diesel	95	Diesel
11	Fuel tank capacity	l	55	66	66	51	61
12	Engine swept volume	cm^3	1994	1991	1951	1598	1910
13	Bore/stroke	mm	84.5/88.9	80/66	84/88	82/75.65	82/90.4
14	Compression ratio	–	19.5	10.5	19.5	10.3	18.45
15	Specific power output	kW/l	38.6	55.2	43.6	55.1	40.3
16	Engine type	–	q 4 R	q 6 V 4Ve	q 4 R 4Ve	q 4 R 4Ve	q 4 R
17	Fuel management	–	EP/eV-T-Ic	EI	EP/CR-T-Ic	Motronic-2Ke	EP/CR-T-Ic
18	Alternator (14 V)	A	65	–	–	75/85	70
19	Battery (12 V)	Ah	65	–	–	45/50	60
20	(Engine pos.)/drive config.	–	v	v	v	v	v
21	Transmission ratio	1st/2nd gear 3rd/4th gear 5th/6th gear	3.42/1.95 1.33/1.05 0.85	3.58/2.02 1.35/1.03 0.81	3.58/2.02 1.29/0.92 0.69	3.91/2.24 1.52/1.16 0.97	3.91/2.24 1.44/1.03 0.77
22	Final drive ratio	–	3.94	4.41	3.74	3.56	3.05
23	Gearshift	H (A)	5	5 (5)	5 (5)	5	5
24	Cooling	l	8.4	–	–	8.4	8.9
25	Vehicle length	mm	4490	4745	4745	4060	4235
26	Vehicle width	mm	1700	1780	1780	1710	1710
27	Vehicle height	mm	1390	1430	1430	1430	1430
28	Wheelbase	mm	2620	2745	2745	2540	2540
29	Track, front/rear	mm	1480/1470	1505/1505	1505/1505	1470/1440	1470/1440
30	Turning circle diameter	m	10.5	12.0	12.0	10.5	10.5
31	Ground clearance	mm	150	120	120	120	120
32	Suspension, front	–	DQl-Ql-Ku	DQl-F-Sf-Ku-Td	DQl-F-Ku-Td	DQl-F-Ku-Td	DQl-F-Ku-Td
33	Suspension, rear	–	Ll-Ql-Zl-Sf-Td	Ma-Sf-Ku-Td	Ma-Sf-Ku-Td	Ll-Sf-Ku-Td	Ll-Sf-Ku-Td
34	Tires	–	175/65 R 14	195/65 R 15	195/65 R 15	185/60 R 14	185/60 R 14
35	Braking system		Sb-Tb	4Sb-ABS	4Sb-ABS	4Sb-ABS	4Sb-ABS

See PP. 898 for abbreviations and explanations.

PETITIONER'S EXHIBIT 1002
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156 2.5i	166 2.4 JTD	GTV 3.0i Spider	Ferrari 456 GT 5.5i	Fiat Seicento 0.9i	Punto 1.4i.e. Turbo	Coupé 2.0i Turbo	
4 Li 5	4 Li 5	2 Ca 2	2 Co 2+2	3 Li 5	3 Li 5	2 Co 4	1
140 (190) 6300	100 (136) 4000	162 (220) 6300	325 (442) 6250	29 (39) 5500	96 (131) 5600	162 (220) 5750	2
222 5000	304 2000	270 5000	550 4500	65 3000	200 3000	310 2500	3
230	202	250	> 300	140	200	250	4
							5
7.3	9.9	6.7	5.2	18	7.9	6.5	
1320	1490	1350	1790	705	1000	1310	6
1820	2000	1610	2200	1190	1450	1720	7
9.4	14.9	8.7	5.5	24.3	10.4	8.1	8
EU 8.4/16.5	EU 5.4/9.3	EU 8.7/16.8	11.6/12.7/29.8	EU 5.1/7.9	EU 6.4/11.7	EU 7.6/14.4	9
95	Diesel	95	98	95	95	95	10
63	72	70	110	35	47	63	11
2492	2387	2959	5474	899	1372	1998	12
88/68.3	82/90.4	93/72.6	88/75	65/67.7	80.5/67.4	82/75.65	13
10.3	18.45	10	10.6	8.8	7.8	8.5	14
56.2	41.9	54.7	59.4	32.2	70	81.1	15
q 6 V 4Ve	q 5 R	q 6 V 4Ve	12 V 4Ve	q 4 R	q 4 R	q 5 R 4Ve	16
Motronic	EP/CR-T-Ic	Motronic	Motronic	EI	Motronic-T-Ic	Motronic-T-Ic	17
120	100	120	140	55/65	65/85	100	18
70	100	70	70	40	40/50	50	19
v	v	v	h-h	v	v	v	20
3.5/2.24 1.52/1.16 0.97/0.82	3.8/2.24 1.36/0.97 0.76/0.61	3.5/2.24 1.52/1.16 0.97/0.82	2.92/1.9 1.37/1.07 0.87/0.74	3.91/2.05 1.34/0.98 0.84	3.91/2.24 1.52/1.16 0.87	3.8/2.24 1.52/1.16 0.91	21
3.94	3.18	3.56	3.64	4.07	3.35	3.11	22
6 (4)	6	6/5	6 (4)	5	5	5/6	23
9.2	9.1	11.7	20	4.8	6	7.2	24
4430	4720	4285	4730	3320	3760	4250	25
1745	1815	1780	1920	1510	1625	1770	26
1420	1420	1320	1300	1440	1450	1350	27
2595	2700	2540	2600	2200	2450	2540	28
1510/1500	1545/1530	1510/1530	1585/1605	1265/1255	1380/1360	1485/1470	29
11.6	11.6	12.0	–	8.8	9.7	11.9	30
140	140	120	130	150	150	150	31
DDQI-Sf-Ku-Td	DDQI-Sf-Ku-Td	DQI-F-Ku-Td	DDQI-Sf-Ku-Td	DQI-F-Td	DQI-F-Ku-Td	DQI-F-Ku-Td	32
DQu-LI-F-Ku-Td	QI-SI-Sf-Ku-Td	Ma-Sf-Ku-Td	Ea-DDQI-Sf-Ku-Td	LI-Sf-Ku-Td	LI-Sf-Ku-Td	LI-Sf-Ku-Td	33
215/60 R 15	205/55 R 16	195/60 R 15	[1]	145/70 R 13	185/55 R 14	205/50 R 16	34
4Sb-ABS	4Sb-ABS	4Sb-ABS	4Sb-ABS	Sb-Tb	4Sb-ABS	4Sb-ABS	35

[1] Front 255/45 ZR 17; Rear 285/40 ZR 17

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Country			Italy				
Vehicle make			**Fiat** (continued)				
Model			Barchetta 1.7i	Bravo 1.8i	Brava 1.9 JTD	Marea 2.4 TDI	Multipla 1.6i Bipower
1	Body style	–	2 Ca 2	3 Li 5	5 Li 5	4 Li 5	5 Gli 6
2	Useful power at engine speed	kW (bhp) rpm	96 (131) 6300	83 (113) 5800	77 (105) 4000	91 (124) 4000	76/68 (103/92) 5750
3	Max. torque at engine speed	Nm rpm	164 4300	154 4400	200 1500	265 2000	144 (Gas 130) 4000
4	Maximum speed	km/h	200	193	185	195	168 (Gas 157)
5	Acceleration from 0 to 100 km/h	s	8.9	10	10.6	10.5	13.5 (Gas 16)
6	Curb weight	kg	1060	1100	1180	1280	1300
7	Gross vehicle weight	kg	1260	1600	1570	1870	1870
8	Weight-to-power ratio	kg/kW	11	14.8	14.9	14.1	17.1
9	Fuel consumption	l/100 km	EU 6.5/11.6	EU 6.5/11.3	EU 4.5/7.3	EU 6/10.2	EU 7.2/11.1
10	Fuel octane number	RON	95	95	Diesel	Diesel	95 + gas
11	Tank capacity	l	50	50	50	63	38 (Gas 164)
12	Engine swept volume	cm^3	1747	1747	1910	2387	1581
13	Bore/stroke	mm	82/82.7	82/82.7	82/90.4	82/90.4	86.4/67.4
14	Compression ratio	–	10.3	10.3	18.45	20.7	10.5
15	Specific power output	kW/l	54.9	47.5	40.3	38.1	48.1
16	Engine type	–	q 4 R 4Ve	q 4 R 4Ve	q 4 R	5 R	q 4 R 4Ve
17	Fuel management	–	EI	EI	EP/CR-T-Ic	EP/eV-T-Ic	EI
18	Alternator (14 V)	A	75/85	75/85	75	75/85	80/90
19	Battery (12 V)	Ah	50	50	60	70	50
20	(Engine pos.)/drive config.	–	v	v	v	v	v
21	Transmission ratio	1st/2nd gear 3rd/4th gear 5th/6th gear	3.91/2.24 1.52/1.16 0.95	3.91/2.24 1.52/1.16 0.97	3.91/2.24 1.44/1.03 0.77	3.8/2.24 1.36/0.97 0.77	3.91/2.24 1.52/1.16 0.97
22	Final drive ratio	–	3.56	3.35	3.05	3.18	4.07
23	Gearshift	H (A)	5	5	5	5	5
24	Cooling	l	6.9	6.7	6.1	7.6	7
25	Vehicle length	mm	3915	4025	4190	4390	3995
26	Vehicle width	mm	1640	1755	1740	1740	1870
27	Vehicle height	mm	1270	1420	1410	1420	1700
28	Wheelbase	mm	2275	2540	2540	2540	2665
29	Track, front/rear	mm	1410/1410	1440/1440	1460/1460	1470/1440	1510/1520
30	Turning circle diameter	m	10.5	10.4	10.4	10.7.	11.0
31	Ground clearance	mm	120	150	150	120	120
32	Suspension, front	–	DQl-F-Ku-Td	DQl-F-Ku-Td	DQl-F-Ku-Td	DQl-F-Ku-Td	DQl-F-Ku-Td
33	Suspension, rear	–	Ll-Sf-Ku-Td	Ll-Sf-Ku-Td	Ll-Sf-Ku-Td	Ll-Sf-Ku-Tc	Ll-Sf-Ku-Td
34	Tires	–	195/55 R 15	175/65 R 14	175/65 R 14	175/70 R 14	185/65 R 15
35	Braking system		4Sb	Sb-Tb	Sb-Tb	4Sb-ABS	Sb-Tb

See PP. 898 for abbreviations and explanations.

PETITIONER'S EXHIBIT 1002
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	Lamborghini	Lancia					
Ulysse 2.1 TD	Diabolo SV 5.7i	Y 1.2 i.e.	Delta 2.0 i.e. Turbo HPE	Dedra 1.9 TD	Kappa 2.4 JTD	Z 2.0 i.e.	
5 Gli 5-7	2 Co 2	3 Li 5	3 Li 5	5 Ko 5	4 Li 5	5 Gli 5-8	1
80 (109) 4300	390 (530) 7100	63 (86) 6000	142 (193) 5500	66 (90) 4200	100 (136) 4000	97 (132) 5500	2
250 2000	605 5500	113 4500	290 3400	186 2500	304 2000	180 4200	3
175	320	177	225	174	202	186	4
							5
13.3	3.85	10.9	7.5	13.9	10	10.1	
1615	1530	910	1330	1285	1510	1555	6
2385	–	1390	1830	1765	2000	2360	7
20.2	3.9	14.4	9.4	18.9	15.1	16	8
EU 6.7/10.8	EU 17.3/37.9	EU 5.1/9.2	EU 8/14.6	EU 5.9/8.8	EU 6.2/10.2	EU 8.5/13.5	9
Diesel	95	95	95	Diesel	Diesel	95	10
80	100	47	61	60	70	80	11
2088	5707	1242	1995	1929	2387	1998	12
85/92	87/80	70.8/78.9	84/90	82.6/90	82/90.4	86/86	13
21.5	10	10.2	8	19.2	18.45	10.4	14
38.3	68.3	50.7	71.2	34.2	41.9	48.5	15
q 4 R 3Ve	12 V 4Ve	q 4 R 4Ve	4 R 4Ve	4 R	5 R	4 R 4Ve	16
EP/eV-T-Ic	EI	EI	EI	EP/V	EP/CR-T-Ic	Motronic	17
90	85	65/75	70	70	100/120	90	18
50/60	55	40	55	60	70	70	19
v	Mm-h	v	v	v	v	v	20
3.42/1.78 1.12/0.8 0.61	2.31/1.52 1.12/0.88 0.68	3.91/2.16 1.48/1.12 0.9	3.8/2.24 1.52/1.16 0.81	3.91/2.24 1.44/1.03 0.79	3.8/2.24 1.36/0.97 0.71	3.25/1.78 1.19/0.9 0.73	21
4.79	4.1	3.73	3.56	3.17	3.56	4.64	22
5	5	5	5	5	5	5	23
9	15	4.5	8.5	8.8	8.3	8.5	24
4455	4470	3725	4010	4345	4685	4470	25
1830	2040	1690	1760	1700	1825	1835	26
1710	1110	1440	1760	1450	1460	1710-1810	27
2825	2650	2380	2540	2540	2700	2825	28
1535/1540	1540/1640	1385/1370	1485/1415	1435/1415	1545/1535	1535/1540	29
11.8	13.0	9.6	11.6	11.4	10.9	11.8	30
130	140	150	150	150	140	–	31
DQI-F-Ku-Td	DQI-Db-Sf-Ku	DQI-F-Ku-Td	DQI-F-Ku-Td	DQI-F-Ku-Td	DQI-F-Sf-Ku-Td	DQI-F-Sf-Td	32
Vla-Sf-Ku-Td	DQI-Db-Sf-Ku	LI-Sf-Ku-Td	LI-Sf-Ku-Td	LI-Sf-Ku-Td	DQu-Sf-Ku-Td	Vla-P-LI-Sf-Ku-Td	33
195/70 R 14	1)	195/50 R 15	215/45 R 16	185/60 R 14	195/65 R 15	205/65 R 15	34
Sb-Tb	4Sb-ABS	Sb-Tb	4Sb-ABS	4Sb-ABS	4Sb-ABS	Sb-Tb	35

1) Front 235/35 ZR 18; Rear 335/30 ZR 18

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

Country			Italy	Japan			
Vehicle make			Maserati	Daihatsu	Honda		
Model			3200 GT BiTurbo	Charade 1.3i	Z 0.7 Ti	Logo 1.3	Civic Aerodeck 1.5i
1	Body style	–	2 Co 2+2	5/3/4 Li 5	3 Mv 4	5/3 Li 5	5 Ko 5
2	Useful power at engine speed	kW (bhp) rpm	271 (368) 6250	62 (84) 6500	47 (64) 6000	48 (65) 5000	84 (114) 6500
3	Max. torque at engine speed	Nm rpm	491 4500	105 5000	93 3700	108 2500	134 5200
4	Maximum speed	km/h	280	170	> 130	152	190
5	Acceleration from 0 to 100 km/h	s	5.1	11	–	14.5	11.5
6	Curb weight	kg	1590	830	960	940	1135
7	Gross vehicle weight	kg	1940	1350	–	1365	1610
8	Weight-to-power ratio	kg/kW	5.9	13.2	25.3	19.2	13.1
9	Fuel consumption	l/100 km	(Ed.) 11/26	EU 5.6/8.9	(Ed.) 6/10	EU 5.5/7.8	EU 5.7/8.8
10	Fuel octane number	RON	95	91	–	95	95
11	Fuel tank capacity	l	90	50	35	40	55
12	Engine swept volume	cm^3	3217	1296	657	1343	1493
13	Bore/stroke	mm	80/80	76/71.4	66/64	75/76	75/84.5
14	Compression ratio	–	8	9.5	8.5	9.2	9.6
15	Specific power output	kW/l	84.2	47.8	57.8	35.7	56.2
16	Engine type	–	8 V 4Ve	q 4 R 4Ve	Mm 3 R 4Ve	q 4 R	q 4 R 4Ve
17	Fuel management	–	El-2Tlc	El	El-T-lc	El	El
18	Alternator (14 V)	A	105	60	35	70	70/75
19	Battery (12 V)	Ah	60	36	32	36	36/55
20	(Engine pos.)/drive config.	–	h-ASR	v	aa	v	v
21	Transmission ratio	1st/2nd gear 3rd/4th gear 5th/6th gear	4.22/2.51 1.67/1.23 1/0.83	3.18/1.84 1.25/0.86 0.71	2.7/1.55 1.03/0.74	3.23/1.78 1.25/0.94 0.77	3.25/1.78 1.25/0.94 0.7
22	Final drive ratio	–	3.23	4.27	3.04	4.06	3.89
23	Gearshift	H (A)	6 (4)	5 (4)	(4)	5 (3+St)	5 (4+St)
24	Cooling	l	14	4.5	–	4.8	4.8
25	Vehicle length	mm	4510	3780	3395	3785	4425
26	Vehicle width	mm	1820	1620	1475	1645	1695
27	Vehicle height	mm	1310	1390	1680	1520	1440
28	Wheelbase	mm	2660	2395	2360	2360	2620
29	Track, front/rear	mm	1525/1540	1385/1390	1280/1290	1425/1400	1480/1490
30	Turning circle diameter	m	11.5	9.6	9.2	10.2	11.0
31	Ground clearance	mm	120	160	200	140	150
32	Suspension, front	–	DDQl-Sf-Ku-Td	DQl-F-Sf-Ku-Td	DQl-F-Sf-Ku-Td	Ql-F	DDQl-Sf
33	Suspension, rear	–	DDQl-Sf-Ku-Td	Ql-Ll-Sf-Ku-Td	De-P-Ll-Sf-Td	Vla-Sf-Ku-Td	Ql-Ll-Sf
34	Tires	–	1)	145 R 13	175 R 15	175/70 R13	175/70 R 13
35	Braking system		4Sb-ABS	Sb-Tb	Sb-Tb	Sb-Tb	Sb-Tb

1) Front 235/40 ZR 18; Rear 265/35 ZR 18

See P. 898 for abbreviations and explanations.

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Accord Coupé 2.0i	Capa 1.5i	Odyssey 3.5i	HR-V 1.6i	Lexus IS 200 2.0i	Lexus GS 300 3.0i	Mazda AZ Wagon 0.7 Ti	
2 Co 5	5 Mv 5	5 Gli 7	3 Ge 4	4 Li 5	4 Li 5	3/5 Mv 4	1
108 (147) 6000	72 (98) 6300	157 (213) 5200	77 (105) 6200	114 (155) 6200	163 (222) 5800	44 (60) 6000	2
184 5000	133 3500	310 4300	135 3400	195 4400	298 3800	83 4000	3
208	170	> 190	165	215	230	135	4
							5
10.1	–	–	11.2	9.5	8.2	–	
1355	1110	1910	1125	1300	1665	750	6
–	–	–	1530	–	2120	–	7
12.5	15.4	12.2	14.6	11.4	10.2	19.7	8
EU 7.1/12.2	(Ed.) 4.5/9	(Ed.) 9/13.1	EU 7.2/9.9	(Ed.) 7/13	EU 8.8/16	(Ed.) 4/8	9
95	95	95	95	95	–	95	10
65	40	75	55	70	75	30	11
1997	1493	3471	1590	1988	2997	657	12
85/88	75/84.5	89/93	75/90	75/75	86/86	65/66	13
10	9.4	9.4	9.6	9.6	10.5	10.5	14
54.1	48.2	45.2	48.4	57.3	56.1	57.8	15
4 R 4Ve	4 R 4Ve	6 V 4Ve	q 4 R 4Ve	6 R 4Ve	6 R 4Ve	3 R 4Ve	16
EI	EI	EI	EI	EI	EI	EI-T-Ic	17
90	75	95	75	–	60	300 W	18
52	36	52	20/45	–	60	36	19
v	v	v	aa (v)	h-ASR	h-ASR	v (aa)	20
3.29/1.96 1.34/1.03 0.81	2.47-0.45	2.43/1.37 0.86/0.58	3.5/1.96 1.34/1.07 0.87	– – –	3.36/2.18 1.43/1 0.75	3.38/2.06 1.28/0.89 0.72	21
4.26	5.81	3.94	4.56	–	3.62	5.94	22
5	(St)	(4)	5 (St)	6 (4)	(5)	5 (4/3)	23
–	4.8	7	4.8	–	8.4	4	24
4765	3775	5110	4010	4400	4810	3395	25
1785	1640	1920	1695	1720	1800	1475	26
1400	1650	1730	1590	1410	1450	1690	27
2670	2360	3000	2350	2670	2800	2360	28
1555/1535	1425/1420	1680/1680	1470/1455	1495/1485	1535/1510	1290/1290	29
11.1	9.6	11.5	10.8	–	11.0	8.4	30
150	150	160	190	150	150	–	31
DQI-QI-LI-Sf-Ku-Td	DQI-F-Sf-Ku-Td	DQI-QI-Sf-Ku-Td	DQI-F-Sf-Ku-Td	DQI-QI-LI-Sf-Ku-Td	DDQI-Sf-Ku-Td	QI-F-Ku	32
QI-LI-Sf-Ku-Td	Vla-LI-Sf-Ku-Td	QI-LI-Sf-Ku-Td	De-P-Sf-Ku-Td	DQI-DQu-LI-Sf-Ku-Td	DQI-QI-SI-Sf-Ku-Td	Sa-LI-P-Sf-Td	33
195/65 R 15	175/70 R 13	215/65 R 16	195/70 R 15	215/45 R 17	225/55 R 16	135 R 12	34
4Sb-ABS	Sb-Tb-ABS	Sb-Tb-ABS	Sb-Tb-ABS	4Sb-ABS	4Sb-ABS	Sb-Tb	35

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,581,565

	Country		Japan				
	Vehicle make		Mazda (continued)			Mitsubishi	
	Model		Demio 1.3i	323P 1.5i	626 Wagon 2.0i	Colt 1.6i	Carisma 1.8 GDI
1	Body style	–	5 Ko 5	3 Li 5	5 Ko 5	4/3 Li 5	4/5 Li 5
2	Useful power	kW (bhp)	53 (72)	65 (88)	100 (136)	129 (175)	92 (125)
	at engine speed	rpm	5500	5500	5800	7500	5500
3	Max. torque	Nm	105	132	181	167	174
	at engine speed	rpm	3500	4000	4500	7000	3750
4	Maximum speed	km/h	158	173	202	> 210	200
5	Acceleration from 0 to 100 km/h	s	13.2	11.8	10.5	7.5	10
6	Curb weight	kg	950	1010	1305	1020	1215
7	Gross vehicle weight	kg	1450	1540	1850	1445	1685
8	Weight-to-power ratio	kg/kW	17.9	16.5	11.8	7.9	13.2
9	Fuel consumption	l/100 km	EU 6/8.7	EU 6.3/9.7	EU 6.7/10.9	(Ed.) 8/14	EU 5/8.4
10	Fuel octane number	RON	95	95	95	91	95
11	Fuel tank capacity	l	43	50	64	50	60
12	Engine swept volume	cm^3	1324	1498	1991	1597	1834
13	Bore/stroke	mm	71/83.6	78/78.4	83/92	81/77.5	81/99
14	Compression ratio	–	9.4	9.4	9.7	11	12.5
15	Specific power output	kW/l	40	43.4	50.2	80.8	50.2
16	Engine type	–	q 4 R 4Ve	q 4 R 4Ve	q 4 R 4Ve	4 R 4Ve	4 R 4Ve
17	Fuel managment	–	EI	EI	L-Jetronic	EI	EId
18	Alternator (14 V)	A	70	70	80	50/70	75
19	Battery (12 V)	Ah	50	40	55/75	45/60	50
20	(Engine pos.)/drive config.	–	v	v	v	v	v
21	Transmission ratio	1st/2nd gear	3.42/1.84	3.42/1.84	3.31/1.83	3.07/1.95	3.58/1.95
		3rd/4th gear	1.29/0.97	1.29/0.97	1.31/0.97	1.38/1.03	1.27/0.97
		5th/6th gear	0.82	0.78	0.76	0.77	0.77
22	Final drive ratio	–	3.85	4.11	4.11	4.63	3.72
23	Gearshift	H (A)	5 (3)	5 (4)	5 (4)	5 (4)	5 (4)
24	Cooling	l	6	6	7	5	6
25	Vehicle length	mm	3810	4035	4660	4290	4435
26	Vehicle width	mm	1670	1695	1710	1690	1695
27	Vehicle height	mm	1540	1410	1520	1390	1400
28	Wheelbase	mm	2390	2505	2670	2500	2550
29	Track, front/rear	mm	1420/1420	1460/1460	1480/1470	1450/1460	1455/1475
30	Turning circle diameter	m	9.4	9.8	10.8	10.6	11.2
31	Ground clearance	mm	150	140	160	120	150
32	Suspension, front	–	QI-F-Ku-Td	DQI-F-Ku-Td	DQI-F-Ku	DQI-F-Ku	DQI-F-Ku
33	Suspension, rear	–	Vla-Sf-Td	QI-LI-F-Ku-Td	QI-LI-Sf-Ku	QI-LI-Sf	QI-LI-Sf
34	Tires	–	165/70 R 13	175/65 R 14	185/65 R 14	185/65 R 14	175/65 R 14
35	Braking system		Sb-Tb	Sb-Tb	4Sb-ABS	Sb-Tb	4Sb

See PP. 898 for abbreviations and explanations.

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	Nissan		Subaru	Suzuki		Toyota	
Space Star 1.3i	Almera 1.4i	Terrano II 2.4i	Legacy 2.0i	Wagon R 1.0i	X-90 1.6i	Yaris 1.0i	
5 Gli 5	3/4/5 Li 5	3/5 Ge 5	4 Li 5	5 Mv 4	2 Ge 2	5 Li 5	1
63 (86) 6000	64 (87) 6000	85 (116) 4800	92 (125) 5600	48 (65) 6500	71 (97) 5600	50 (68) 6000	2
117 3000	116 4000	191 3200	184 3600	181 3500	132 4000	90 4100	3
170	172	160	188	140	150	156	4
							5
13.4	12.6	13.7	10.8	–	–	12	
1120	1035	1630	1370	845	1100	820	6
1655	1545	2510	1815	1330	1360	–	7
17.8	16.2	18.7	15.7	15.9	15.6	16.4	8
EU 5.6/9	EU 5.8/9.2	EU 10.3/15.8	EU 7.7/12.3	EU 5/7.6	EU 6.7/9.8	(Ed.) 5/8	9
95	95	95	95	95	95	95	10
55	50	72	64	42	42	45	11
1299	1392	2389	1994	997	1590	998	12
71/82	73,6/81,8	89/96	92/75	68/68.6	75/90	69/66.7	13
10	9.5	8.6	10	10	9.5	10	14
48.5	46	35.6	46.1	48.2	44.7	50.1	15
q 4 R 4Ve	q 4 R 4Ve	4 R 3Ve	4 Bx 4Ve	4 R 4Ve	4 R 4Ve	q 4 R 4Ve	16
EI	EI	EI	EI	EI	EI	EI	17
50/70	70	50	90	300 W	70	70	18
45/60	55	60	48	26	48	50	19
v	v	aa	aa	v	aa	v	20
3.73/2.05 1.32/0.97 0.79	3.33/1.96 1.29/0.9 0.73	3.59/2.25 1.42/1 0.82	3.55/2.11 1.45/1.09 0.87	3.42/1.89 1.28/0.91 0.76	3.65/1.95 1.38/1 0.8	3.55/1.91 1.31/0.97 0.8	21
4.21	4.17	4.63	3.9	4.39	4.3	3.94	22
5	5	5	5 (4)	5 (4/3)	5 (4)	5 (4)	23
5	4	8	6	5	5.3	5	24
4030	4120	4185	4600	3410	3710	3610	25
1700	1690	1755	1700	1575	1695	1660	26
1520	1390	1830	1410	1700	1560	1500	27
2500	2535	2450	2650	2335	2200	2370	28
1475/1470	1470/1435	1455/1430	1460/1455	1360/1355	1425/1430	1450/1430	29
9.6	10.4	10.8	11.1	9.6	9.8	9.8	30
150	140	121	160	150	170	–	31
DQI-F-Ku-Td	DQI-F-Sf-Td	DDQI-Tf-Ku-Td	DQI-F-Sf-Ku-Td	QI-F-Sf-Ku-Td	DQI-Db-Sf-Ku-Td	DQI-F-Sf-Td	32
QI-LI-F-Ku-Td	Ta-LI-QI-Sf-Ku-Td	Sa-LI-P-Sf-Ku-Td	Ma-Sf-Ku-Td	Ta-P-Sf-Td	Sa-LI-Sf-Td	Vla-Sf-Td	33
175/65 R 14	175/70 R 13	235/75 R 15	195/60 R 15	155/65 R 13	195/65 R 15	175/65 R 14	34
Sb-Tb	Sb-Tb	Sb-Tb-ABS	4Sb-ABS	Sb-Tb	Sb-Tb-ABS	Sb-Tb	35

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	Country		Japan		Korea		
	Vehicle make		Toyota (Continued)		Daewoo		
	Model		Avensis 2.0 TD-CR	Picnic 2.0i	Matiz 0.8i	Lanos 1.5i	Nubira 2.0i
1	Body style	–	4/5 Li 5	5 Gli 6	5 Li 5	4/3/5 Li 5	5 Ko 5
2	Useful power at engine speed	kW (bhp) rpm	82 (112) 4000	94 (128) 5400	38 (52) 5900	55 (75) 5400	100 (136) 5600
3	Max. torque at engine speed	Nm rpm	250 2500	178 4400	69 4600	113 3400	184 4000
4	Maximum speed	km/h	180	180	144	170	195
5	Acceleration from 0 to 100 km/h	s	12	10.8	17	15	9
6	Curb weight	kg	1280	1335	725	1005	1285
7	Gross vehicle weight	kg	1830	2010	1210	1595	1860
8	Weight-to-power ratio	kg/kW	18.1	14.2	19.1	18.3	11
9	Fuel consumption	l/100 km	EU 5.2/8.5	EU 7.4/12.3	EU 5.2/8.4	EU 5.9/11.4	EU 7.4/15.1
10	Fuel octane number	RON	Diesel	95	95	91	–
11	Fuel tank capacity	l	62	60	35	48	62
12	Engine swept volume	cm^3	1975	1998	796	1349	1998
13	Bore/stroke	mm	86/85	86/86	86.5/72	76.5/73.4	86/86
14	Compression ratio	–	23	9.5	9.3	9.5	9.5
15	Specific power output	kW/l	33.4	47	47.7	40.8	50.1
16	Engine type	–	4 R	4 R 4Ve	q 3 R	q 4 R	q 4 R 4Ve
17	Fuel managment	–	EP/CR-T-Ic	EI	EI	EI	EI
18	Alternator (14 V)	A	70	80	65	75/100	75/85
19	Battery (12 V)	Ah	80	60	35	46/55	55
20	(Engine pos.)/drive config.	–	v	v	v	v	v
21	Transmission ratio	1st/2nd gear 3rd/4th gear 5th/6th gear	3.54/2.04 1.32/0.95 0.73	3.54/2.04 1.32/1.03 0.82	3.82/2.21 1.42/1.03 0.84	3.55/2.05 1.35/0.97 0.76	3.55/2.16 1.48/1.13 0.87
22	Final drive ratio	–	3.74	3.94	4.44	4.18	3.55
23	Gearshift	H (A)	5	5 (4)	5	5	5 (4)
24	Cooling	l	6.9	6.7	3.8	7	7
25	Vehicle length	mm	4490	4530	3595	4235	4515
26	Vehicle width	mm	1710	1695	1495	1680	1700
27	Vehicle height	mm	1430	1620	1480	1430	1430
28	Wheelbase	mm	2630	2735	2340	2520	2570
29	Track, front/rear	mm	1480/1450	1470/1450	1315/1280	1405/1425	1465/1455
30	Turning circle diameter	m	11.6	11.0	9.0	9.8	10.6
31	Ground clearance	mm	140	150	140	160	140
32	Suspension, front	–	DQl-F-Ku-Td	DQl-F-Sf-Ku-Td	Ql-F-Sf-Td	Ql-Ll-F-Sf-Ku-Td	DQl-F-Sf-Ku-Td
33	Suspension, rear	–	DQu-Ll-Ku-Td	Vla-Ll-Sf-Ku-Td	Vla-Ll-Sf-Td	Vla-Ll-Sf-Ku-Td	Vla-Sf-Ku-Td
34	Tires	–	185/65 HR 14	195/65 R 14	145/70 R 13	155 R 13	185/65 R 14
35	Braking system		Sb-Tb-ABS	Sb-Tb-ABS	Sb-Tb	Sb-Tb	4Sb-ABS

See PP. 898 for abbreviations and explanations.

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			Sweden				
Hyundai		**Kia**	**Saab**				
Atos 1.0i	Coupé 2.0i	Carnival 2.5i	9-3 2.0i	9-3 2.2 TD	9-5 2.0i Turbo	9-5 3.0i Turbo	
5 Mv 5	3 Co 4/5	5 Gli 5-7	5/3 Li 5	3/5 Li 5	4 Li 5	5 Ko 5	1
41 (56) 5500	102 (139) 6000	121 (165) 6500	96 (131) 6100	85 (116) 4300	110 (150) 5500	147 (200) 5000	2
82 3100	182 4900	222 4100	177 4300	260 1900	215 2500	310 2500	3
142	201	185	200	200	215	230	4
							5
15.1	8.6	–	11	10.9	9.8	8.7	
820	1250	1735	1295	1350	1485	1650	6
1220	1600	2425	1860	1910	–	2180	7
20	12.2	14.3	13.5	15.9	13.5	10.9	8
EU 5.4/8	EU 7.3/11.4	EU 8.9/15.3	EU 6.8/13.1	EU 4.6/8.9	EU 7.2/13.5	EU 8.5/17.2	9
95	95	95	95	Diesel	95	95	10
35	55	75	68	68	75	75	11
999	1975	2497	1985	2172	1985	2962	12
76/73	82/93.5	80/82.8	90/78	84/98	90/78	86/85	13
9.5	10.3	10.5	10.1	19.5	9.3	9.5	14
41	51.6	48.5	48.4	39.1	55.4	49.6	15
q 4 R 3Ve	q 4 R 4Ve	q 6 V 4Ve	4 R 4Ve	4 R 4Ve	q 4 R 4Ve	q 6 V 4Ve	16
EI	EI	EI	Motronic	EP/eV-T-Ic	Ze-T	Ze-T-Ic	17
60	75/90	80	70/130	120	90/130	90/130	18
40/45	68	54	60	85	60	60	19
v	v	v	v	v	v	v	20
3.54/1.95 1.31/0.92 0.78	3.31/1.95 1.39/1.06 0.84	3.75/1.95 1.3/0.94 0.75	3.38/1.76 1.12/0.89 0.7	3.38/1.76 1.12/0.89 0.7	3.38/1.76 1.18/0.89 0.66	2.58/1.41 1/0.74	21
4.53	3.84	4.31	4.45	4.45	4.05	4.28	22
5 (3)	5 (4)	5 (4)	5 (4)	5 (4)	5 (4)	(4)	23
6	6	8	10	10	9	9.7	24
3495	4340	4890	4630	4630	4810	4810	25
1495	1730	1895	1710	1710	1790	1790	26
1610	1300	1730	1430	1430	1450	1450	27
2380	2480	2905	2600	2600	2705	2705	28
1315/1300	1470/1450	1625/1600	1450/1440	1450/1440	1520/1520	1520/1520	29
9.1	10.0	13.0	11.1	11.1	10.8	10.8	30
160	150	170	140	140	–	–	31
Ql-F-Sf-Ku-Td	DQl-F-Sf-Ku-Td	Ql-F-Ku-Td	DQl-F-Ku	DQl-F-Ku	DQl-F-Sf-Ku-Td	DQl-F-Sf-Ku-Td	32
Ta-Ql-Ll-Sf-Td	DQu-Sf-Td	Sa-P-Ll-Sf-Ku-Td	Vla-Sf-Ku-Td	Vla-Sf-Ku-Td	DQu-Ll-Sf-Ku-Td	DQu-Ll-Sf-Ku-Td	33
155/70 R 13	205/50 R 15	205/70 R 15	185/65 HR 15	185/65 HR 15	205/65 R 15	205/60 R 15	34
Sb-Tb-ABS	4Sb-ABS	Sb-Tb	4Sb-ABS	4Sb-ABS	4Sb-ABS	4Sb-ABS	35

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	Country		Sweden				
	Vehicle make		Volvo				
	Model		S40 1.8i	V40 1.9 TDI	S70 2.0i	V70 2.4i	S80 2.8 T6
1	Body style	–	4 Li 5	5 Ko 5	4 Li 5	5 Ko 5	4 Li 5
2	Useful power	kW (bhp)	92 (125)	70 (95)	93 (126)	125 (170)	200 (272)
	at engine speed	rpm	5500	5000	6250	6100	5400
3	Max. torque	Nm	174	190	170	220	380
	at engine speed	rpm	3750	2000	4800	4700	2000
4	Maximum speed	km/h	200	180	195	210	250
5	Acceleration from 0 to 100 km/h	s	10.5	12.5	11.7	9.2	7.2
6	Curb weight	kg	1230	1250	1430	1470	1590
7	Gross vehicle weight	kg	1750	1790	1890	1970	2100
8	Weight-to-power ratio	kg/kW	13.4	14.4	15.4	11.4	8
9	Fuel consumption	l/100 km	5.7/6.9/9.1	5.2/6.3/8.3	EU 8.1/13.4	EU 7.8/15	EU 8/16.4
10	Fuel octane number	RON	95	Diesel	95	95	98
11	Fuel tank capacity	l	60	60	70	70	80
12	Engine swept volume	cm^3	1834	1870	1984	2435	2783
13	Bore/stroke	mm	81/89	80/93	81/77	83/90	81/90
14	Compression ratio	–	12.5	20.5	10	10.3	8.5
15	Specific power output	kW/l	50.2	35.3	46.9	51.3	71.9
16	Engine type	–	4 R 4Ve	4 R	q 5 R	q 5 R 4Ve	q 6 R 4Ve
17	Fuel management	–	EId	EP/V-T	EI	EI	Motronic-2Tlc
18	Alternator (14 V)	A	80	80	100	100	120
19	Battery (12 V)	Ah	62	72	520 A	520 A	600 A
20	(Engine pos.)/drive config.	–	v	v	v	v	v
21	Transmission ratio	1st/2nd gear	3.58/1.94	3.61/1.86	3.07/1.77	3.07/1.77	2.92/1.57
		3rd/4th gear	1.26/0.97	1.32/0.97	1.19/0.87	1.19/0.87	1/0.7
		5th/6th gear	0.76	0.76	0.7	0.7	
22	Final drive ratio	–	3.72	3.44	4.45	4.00	3.29
23	Gearshift	H (A)	5 (4)	5 (4)	5 (4)	5 (4)	(4)
24	Cooling	l	5	7.4	7.2	7.2	10.7
25	Vehicle length	mm	4480	4480	4720	4720	4820
26	Vehicle width	mm	1720	1720	1760	1760	1830
27	Vehicle height	mm	1390	1390	1400	1430	1450
28	Wheelbase	mm	2550	2550	2660	2660	2790
29	Track, front/rear	mm	1450/1470	1450/1470	1520/1470	1520/1470	1580/1560
30	Turning circle diameter	m	10.6	10.6	10.2	10.2	10.9
31	Ground clearance	mm	150	150	140	140	150
32	Suspension, front	–	DQl-F-Sf-Ku-Td	DQl-F-Sf-Ku-Td	DQl-F-Ku	DQl-F-Ku	DQl-F
33	Suspension, rear	–	Ql-Ll-Zl-Sf-Ku-Td	Ql-Ll-Zl-Sf-Ku-Td	Ma-Sf-Ku-Td	Ma-Sf-Ku-Td	Ma-Sf-Ku-Td
34	Tires	–	195/55 VR 15	195/55 VR 15	185/65 R 15	185/65 R 15	205/65 R 15
35	Braking system		4Sb-ABS	4Sb-ABS	4Sb-ABS	4Sb-ABS	4Sb-ABS

See PP. 898 for abbreviations and explanations.

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	Spain				Czech. Rep.	USA	
	Seat				Skoda	Buick	
C70 2.3i	Arosa 1.0i	Arosa 1.7 SDI	Ibiza 1.9 TDI	Cordoba 2.0i	Octavia 1.8 T	Le Sabre 3.8i	
2 Ca 4	3 Li 5	3 Li 5	5/3 Li 5	4 Li 5	5 Ko 5	4 Li 5-6	1
176 (239) 5100	37 (50) 5000	44 (60) 4200	81 (110) 4000	110 (150) 6000	110 (150) 5700	153 (208) 5200	2
330 2700	86 3000	115 2200	235 1900	180 4500	210 1750	312 3700	3
250	151	157	193	216	216	200	4
							5
6.9	17.4	16.8	10.2	8.5	9.1	–	
1560	865	960	1060	1120	1315	1620	6
1970	1265	1360	1520	1580	1830	–	7
–	23.4	21.8	13	10.2	11.8	10.6	8
8.4/11.3/16.3	EU 4.7/7.5	EU 3.6/5.9	EU 3.9/6.4	EU 6.4/11.9	EU 6.2/10.9	EPA 7.8/12.4	9
95	95	Diesel	Diesel	95	95	91	10
68	34	34	45	45	55	66	11
2319	999	1716	1896	1984	1781	3791	12
81/90	67.1/70.6	79.5/86.4	79.5/95.5	82.5/92.8	81/86.4	96.52/86.36	13
8.5	10.5	19.5	19.5	10.5	9.5	9.4	14
75.9	37	25.6	42.7	55.4	61.7	40.4	15
q 5 R 4Ve	4 R	4 R	q 4 R	4 R 4Ve	q 4 R 5Ve	q 6 V	16
EI-T	Motronic	EP/eV	EP/eV-T-Ic	Motronic	Motronic-T-Ic	EI	17
100	70	70	90	70/90	70/90	120	18
520 A	36	61	61	44/60	44/60	54	19
v-ASR	v	v	v	v-EDS	v	v-ASR	20
3.07/1.77 1.19/0.87 0.7	3.46/2.1 1.45/1.1 0.89	3.46/1.96 1.25/0.93 0.74	3.78/2.06 1.35/0.97 0.74	3.3/1.94 1.31/1.03 0.84	3.3/1.94 1.31/1.03 0.84	2.92/1.57 1/0.7	21
4.00	4.06	3.33	3.3	3.68	3.68	2.86	22
5 (4)	5	5	5	5	5	(4)	23
7	6	6	6.5	6.3	6.3	11	24
4720	3535	3535	3855	4145	4510	5080	25
1820	1640	1640	1640	1640	1730	1865	26
1410	1460	1460	1420	1420	1430	1450	27
2660	2325	2325	2445	2445	2510	2850	28
1520/1520	1370/1400	1370/1400	1430/1395	1430/1395	1515/1490	1580/1580	29
11.7	9.8	9.8	10.4	10.4	10.8	12.3	30
120	110	110	150	130	140	140	31
DQI-F-Sf-Ku-Td	DQI-F-Td	DQI-F-Td	DQI-F-Td	DQI-F-Ku-Td	DQI-F-Sf-Ku-Td	DQI-F-Sf-Ku-Td	32
Vla-QI-LI-Sf-Ku-Td	Vla-Sf-Td	Vla-Sf-Td	Vla-Sf-Td	Vla-Sf-Ku-Td	Vla-LI-Sf-Ku-Td	Ea-QI-SI-Sf-Ku-Td	33
225/45 ZR 17	155/70 R 13	155/70 R 13	185/55 R 15	195/45 R 16	175 R 14	215/70 R 15	34
4Sb-ABS	Sb-Tb	Sb-Tb	4Sb	4Sb-ABS	4Sb-ABS	4Sb-ABS	35

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Country			USA				
Vehicle make			**Cadillac**	**Chevrolet**	**Chrysler**		
Model			Catera 3.0i	Impala 3.4i	Stratus 2.5i	Sebring 2.0i	300 M 2.7i
1	Body style	–	4 Li 5	4 Li 5-6	4 Li 5	2 Co 5	4 Li 5
2	Useful power at engine speed	kW (bhp) rpm	149 (203) 6000	134 (182) 6000	120 (163) 5850	104 (141) 6000	149 (203) 5800
3	Max. torque at engine speed	Nm rpm	260 3600	279 4000	214 4400	177 4800	258 4850
4	Maximum speed	km/h	200	> 180	210	200	210
5	Acceleration from 0 to 100 km/h	 s	(0-97 km/h) 8.5	10	10.5	11	10.2
6	Curb weight	kg	1710	1540	1435	1335	1610
7	Gross vehicle weight	kg	–	–	1930	–	2120
8	Weight-to-power ratio	kg/kW	11.5	11.5	12.2	12.8	10.8
9	Fuel consumption	l/100 km	EPA 9.4/13.1	(Ed.) 9/15	EU 7.1/15.1	(Ed.) 8/12	EPA 9/11.2
10	Fuel octane number	RON	95	91	95	91	92
11	Fuel tank capacity	l	60	64.5	60	64	64
12	Engine swept volume	cm^3	2962	3350	2497	1996	2736
13	Bore/stroke	mm	86/85	92/84	83.5/76	87.5/83	86/78.5
14	Compression ratio	–	10	9.5	9.4	9.6	9.7
15	Specific power output	kW/l	50.3	40	48.1	52.1	54.5
16	Engine type	–	6 V 4Ve	q 6 V	q 6 V 4Ve	q 4 R 4Ve	6 V 4Ve
17	Fuel management	–	Motronic	EI	EI	EI	EI
18	Alternator (14 V)	A	120	105/140	90	90	125
19	Battery (12 V)	Ah	300 A	54	60	430 A	60
20	(Engine pos.)/drive config.	–	h-ASR	v-ASR	v	v	v
21	Transmission ratio	1st/2nd gear 3rd/4th gear 5th/6th gear	2.86/1.62 1/0.72	2.92/1.57 1/0.71	2.84/1.57 1/0.69	3.54/2.13 1.36/1.03 0.81	2.84/1.57 1/0.69
22	Final drive ratio	–	3.9	3.29	3.91	3.94	3.89
23	Gearshift	H (A)	(4)	(4)	(4)	5 (4)	(4)
24	Cooling	l	10	11.2	10	7	8.2
25	Vehicle length	mm	4930	5080	4750	4850	5000
26	Vehicle width	mm	1790	1855	1820	1770	1920
27	Vehicle height	mm	1430	1460	1380	1340	1420
28	Wheelbase	mm	2730	2805	2740	2635	2870
29	Track, front/rear	mm	1505/1520	1575/1555	1530/1530	1510/1505	1575/1565
30	Turning circle diameter	m	11.0	11.5	11.8	12.2	12.3
31	Ground clearance	mm	130	120	140	140	130
32	Suspension, front	–	DQl-F-Ku-Td	DQl-F-Sf-Ku-Td	DDQl-Sf-Ku-Td	DDQl-Sf-Ku-Td	Ql-Ll-F-Sf-Ku-Td
33	Suspension, rear	–	Ql-Sl-Sf-Ku-Td	Ql-Ll-F-Sf-Ku-Td	Ea-DQl-Ql-Ll-Sf-Ku-Td	Ea-DQl-Ql-Ll-Sf-Ku-Td	Ea-Ql-Ll-F-Sf-Ku-Td
34	Tires	–	225/55 R 16	225/60 R 16	185/65 R 15	195/70 HR 14	225/55 R 17
35	Braking system		4Sb-ABS	4Sb-ABS	4Sb	Sb-Tb	4Sb-ABS

See PP. 898 for abbreviations and explanations.

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Dodge Caravan 2.4i	**Ford** Windstar 3.8i	**Lincoln** Continental 4.6i	**Mercury** Cougar 2.5i	**Oldsmobile** Alero 2.4i	**Plymouth** Voyager 3.0i	**Pontiac** Sunfire 2.2i	
5 Gli 5-8	4 Gli 5-7	4 Li 5-6	3 Co 4	2 Co 5	5 Gli 7	2 Ca 5	1
112 (152) 5200	149 (203) 4900	205 (279) 5750	127 (173) 6250	112 (152) 5600	112 (152) 5200	86 (117) 5000	2
226 4000	326 3600	373 4750	224 4250	210 4400	239 4000	183 3600	3
170	> 175	215	225	200	175	175	4
							5
–	–	–	8.5	10.5	–	–	
1595	1690	1760	1375	1340	1605	1300	6
–	2440	–	–	–	–	–	7
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Sa-P-Td	Vla-Sf-Ku-Td	QI-LI-Ku-Lue	QI-LI-F-Sf-Ku-Td	DQu-LI-Sf-Ku-Td	Sa-P-Td	Vla-LI-Sf-Ku-Td	33
205/75 R 14	205/75 R 15	225/60 R 16	205/60 R 15	215/60 R 15	205/75 R 14	195/70 R 14	34
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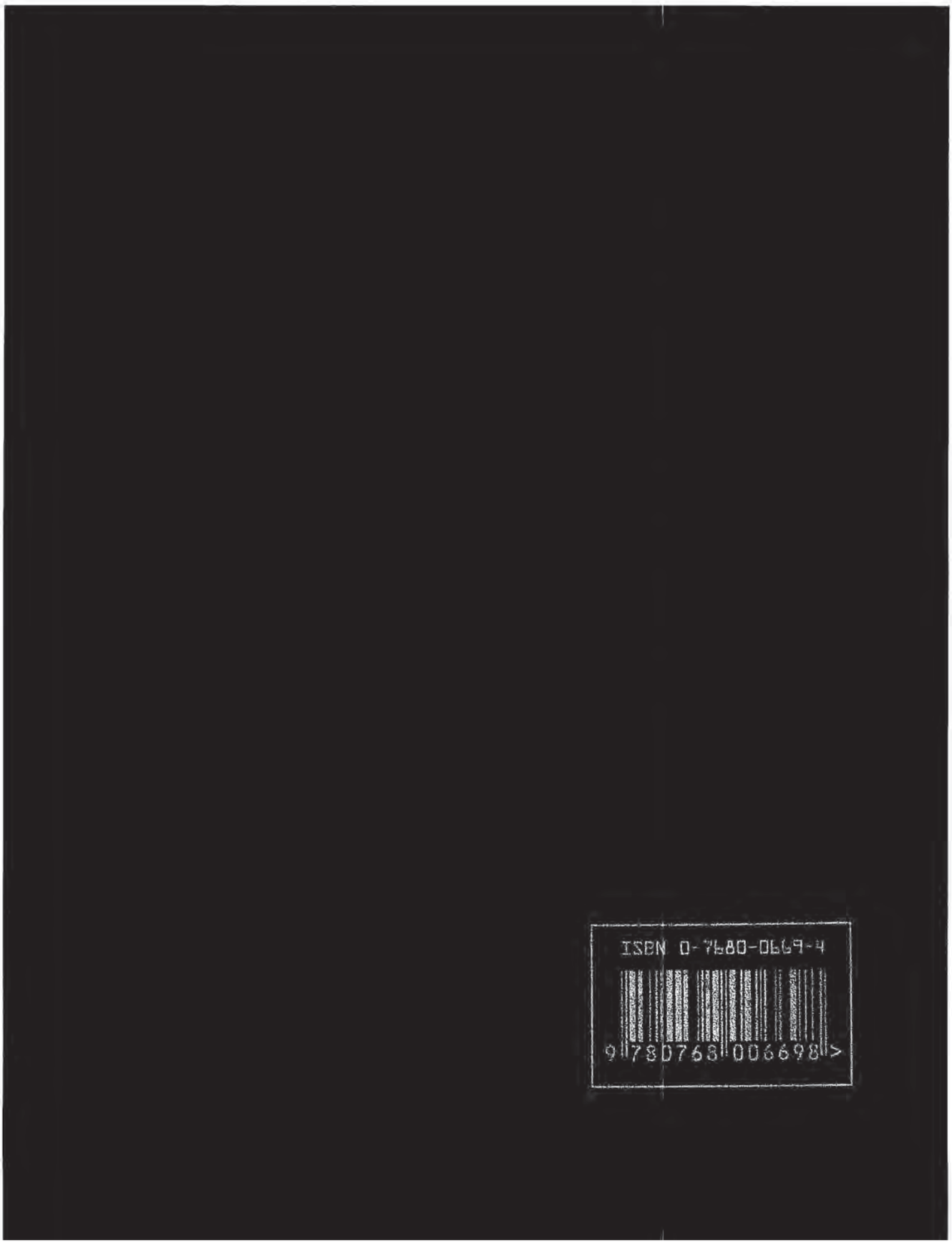
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PETITIONER'S EXHIBIT 1002
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Internal Combustion Engine Fundamentals



John B. Heywood

VW EX1014 (Part 1 of 4)
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John B. Heywood

Professor of Mechanical Engineering
Director, Sloan Automotive Laboratory
Massachusetts Institute of Technology

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ABOUT THE AUTHOR

Dr. John B. Heywood received the Ph.D. degree in mechanical engineering from the Massachusetts Institute of Technology in 1965. Following an additional post-doctoral year of research at MIT, he worked as a research officer at the Central Electricity Generating Board's Research Laboratory in England on magneto-hydrodynamic power generation. In 1968 he joined the faculty at MIT where he is Professor of Mechanical Engineering. At MIT he is Director of the Sloan Automotive Laboratory. He is currently Head of the Fluid and Thermal Science Division of the Mechanical Engineering Department, and the Transportation Energy Program Director in the MIT Energy Laboratory. He is faculty advisor to the MIT Sports Car Club.

Professor Heywood's teaching and research interests lie in the areas of thermodynamics, combustion, energy, power, and propulsion. During the past two decades, his research activities have centered on the operating characteristics and fuels requirements of automotive and aircraft engines. A major emphasis has been on computer models which predict the performance, efficiency, and emissions of spark-ignition, diesel, and gas turbine engines, and in carrying out experiments to develop and validate these models. He is also actively involved in technology assessments and policy studies related to automotive engines, automobile fuel utilization, and the control of air pollution. He consults frequently in the automotive and petroleum industries, and for the U.S. Government.

His extensive research in the field of engines has been supported by the U.S. Army, Department of Energy, Environmental Protection Agency, NASA, National Science Foundation, automobile and diesel engine manufacturers, and petroleum companies. He has presented or published over a hundred papers on

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his research in technical conferences and journals. He has co-authored two previous books: *Open-Cycle MHD Power Generation* published by Pergamon Press in 1969 and *The Automobile and the Regulation of Its Impact on the Environment* published by University of Oklahoma Press in 1975.

He is a member of the American Society of Mechanical Engineers, an associate fellow of the American Institute of Aeronautics and Astronautics, a fellow of the British Institution of Mechanical Engineers, and in 1982 was elected a Fellow of the U.S. Society of Automotive Engineers for his technical contributions to automotive engineering. He is a member of the editorial boards of the journals *Progress in Energy and Combustion Science* and the *International Journal of Vehicle Design.*

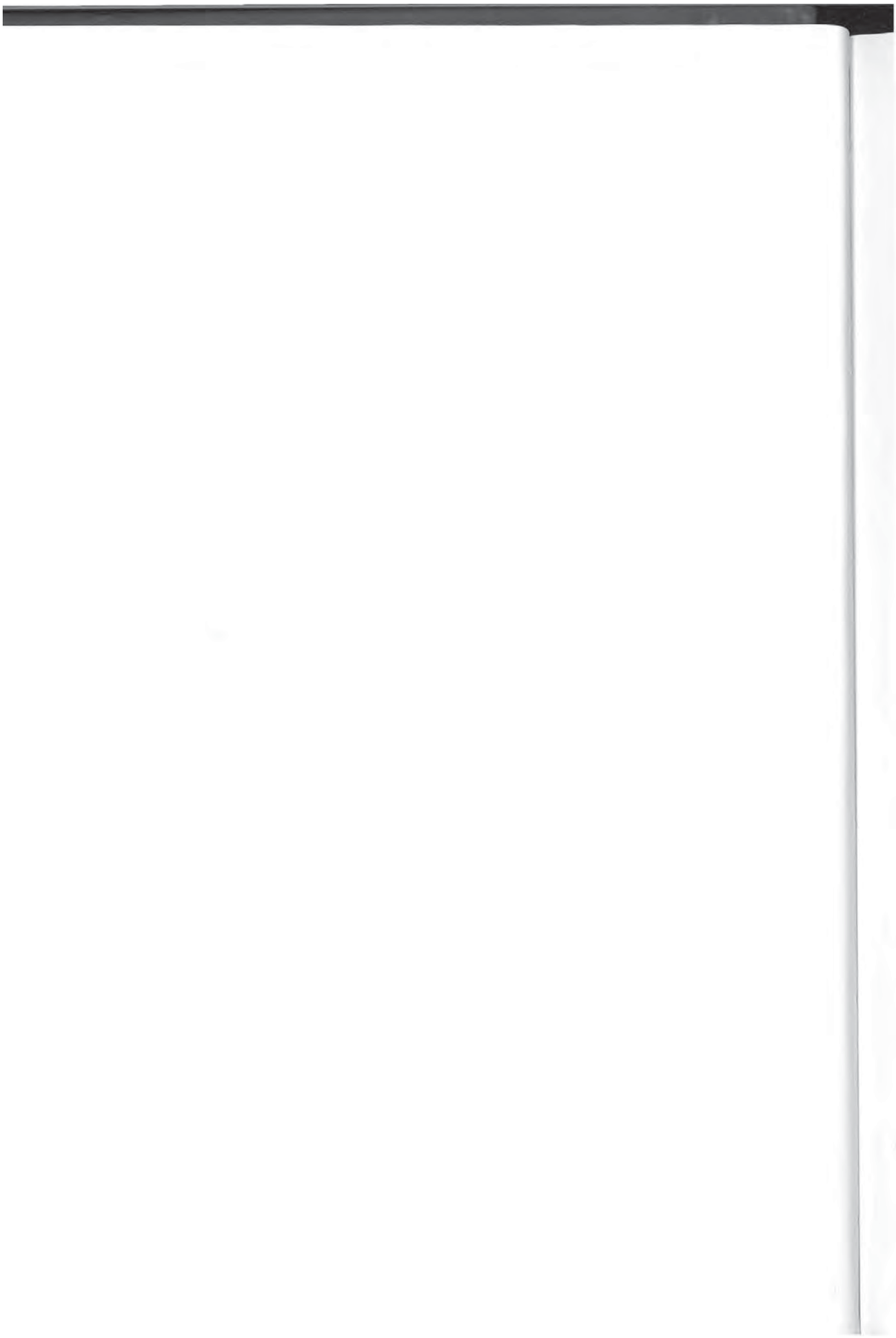
His research publications on internal combustion engines, power generation, and gas turbine combustion have won numerous awards. He was awarded the Ayreton Premium in 1969 by the British Institution of Electrical Engineers. Professor Heywood received a Ralph R. Teetor Award as an outstanding young engineering educator from the Society of Automotive Engineers in 1971. He has twice been the recipient of an SAE Arch T. Colwell Merit Award for an outstanding technical publication (1973 and 1981). He received SAE's Horning Memorial Award for the best paper on engines and fuels in 1984. In 1984 he received the Sc.D. degree from Cambridge University for his published contributions to engineering research. He was selected as the 1986 American Society of Mechanical Engineers Freeman Scholar for a major review of "Fluid Motion within the Cylinder of Internal Combustion Engines."

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THIS BOOK IS DEDICATED TO MY FATHER,
Harold Heywood:

I have followed many of the paths he took.

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PREFACE

Internal combustion engines date back to 1876 when Otto first developed the spark-ignition engine and 1892 when Diesel invented the compression-ignition engine. Since that time these engines have continued to develop as our knowledge of engine processes has increased, as new technologies became available, as demand for new types of engine arose, and as environmental constraints on engine use changed. Internal combustion engines, and the industries that develop and manufacture them and support their use, now play a dominant role in the fields of power, propulsion, and energy. The last twenty-five years or so have seen an explosive growth in engine research and development as the issues of air pollution, fuel cost, and market competitiveness have become increasingly important. An enormous technical literature on engines now exists which has yet to be adequately organized and summarized.

This book has been written as a text and a professional reference in response to that need. It contains a broadly based and extensive review of the fundamental principles which govern internal combustion engine design and operation. It attempts to provide a simplifying framework for the vast and complex mass of technical material that now exists on spark-ignition and compression-ignition engines, and at the same time to include sufficient detail to convey the real world dimensions of this pragmatic engineering field. It is the author's conviction that a sound knowledge of the relevant fundamentals in the many disciplines that contribute to this field, as well as an awareness of the extensive practical knowledge base which has been built up over many decades, are essential tools for engine research, development, and design. Of course, no one text can include everything about engines. The emphasis here is on the thermodynamics, combustion physics and chemistry, fluid flow, heat transfer, friction, and lubrication processes relevant to internal combustion engine design, performance, efficiency, emissions, and fuels requirements.

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From a fundamental point of view, how the fuel-air mixture within an internal combustion engine cylinder is ignited appropriately organizes the field. From the method of ignition—spark-ignition or compression-ignition—follows each type of engine's important features: fuel requirements, method of mixture preparation, combustion chamber design, details of the combustion process, method of load control, emission formation mechanisms, and performance and efficiency characteristics. While many engine processes (such as intake and exhaust flows, convective heat transfer, and friction) are similar in both types of engines, this distinction is fundamental and lies behind the overall organization of the book.

The book is arranged in four major sections. The first (Chapters 1 to 5) provides an introduction to, and overview of, the major characteristics of spark-ignition and compression-ignition engines, defines the parameters used to describe engine operation, and develops the necessary thermodynamics and combustion theory required for a quantitative analysis of engine behavior. It concludes with an integrated treatment of the various methods of analyzing idealized models of internal combustion engine cycles. The second section (Chapters 6 to 8) focuses on engine flow phenomena. The details of the gas exchange process—intake and exhaust processes in four-stroke and scavenging in two-stroke cycles—and the various methods of supercharging engines—are reviewed. Fuel metering methods for spark-ignition engines and air- and fuel-flow phenomena in intake manifolds are described. The essential features of the various types of fluid motion within the engine cylinder are then developed. These flow processes control the amount of air an engine will induct (and therefore its power), and largely govern the rate at which the fuel-air mixture will burn during combustion.

The third section of the book focuses on engine combustion phenomena. These chapters (9, 10, and 11) are especially important. The combustion process releases the fuel's energy within the engine cylinder for eventual conversion to useful work. What fraction of the fuel's energy is converted depends strongly on how combustion takes place. The spark-ignition and compression-ignition engine combustion processes (Chapters 9 and 10, respectively) therefore influence essentially all aspects of engine behavior. Air pollutants are undesirable byproducts of combustion. Our extensive knowledge of how the major pollutants form during these combustion processes and how such emissions can be controlled is reviewed in Chapter 11.

The last section of the book focuses on engine operating characteristics. First, the fundamentals of engine heat transfer and friction, both of which detract from engine performance, are developed in Chapters 12 and 13. Chapter 14 then focuses on the methods available for predicting important aspects of engine behavior based on realistic models of engine flow and combustion processes. Since the various thermodynamic-based and fluid-mechanic-based models which have been developed over the past fifteen years or so are increasingly used in engine research and development, a knowledge of their basic structure and capabilities is most important. Then, Chapter 15 presents a summary of how the operating characteristics—power, efficiency, and emissions—of spark-ignition and compression-ignition engines depend on the major engine design and oper-

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ating variables. These final two chapters effectively integrate the analytical understanding and practical knowledge of individual engine processes together to describe overall spark-ignition and compression-ignition engine behavior. Material on internal combustion engine fuels is distributed appropriately throughout the book. Each chapter is extensively illustrated and referenced, and includes problems for both undergraduate and graduate level courses.

While this book contains much advanced material on engine design and operation intended for the practitioner, each major topic is developed from its beginnings and the more sophisticated chapters have introductory sections to facilitate their use in undergraduate courses. The chapters are extensively cross-referenced and indexed. Thus several arrangements of the material for a course on engines can be followed. For example, an introductory course on internal combustion engines could begin with Chapters 1 and 2, which review the different types of engines and how their performance is characterized, and continue with the parts of Chapters 3 and 5, which introduce the key combustion concepts necessary to understand the effects of fuel/air ratio, and ideal cycle analysis. Selections from the introductory sections of Chapters 6, 9, 10, 11, and 15 could then be used to explain several of the practical and design aspects of spark-ignition and diesel engine intake and exhaust processes, combustion, emissions, and performance. A more advanced course would review this introductory material more rapidly, and then move on to those sections of Chapters 4 and 5, which cover fuel-air cycle analysis, a more extensive discussion of engine breathing using additional sections of Chapter 6, and more in-depth treatment of engine combustion and emissions processes based on the appropriate sections of Chapters 9, 10, and 11. Material on engine heat transfer and friction selected from Chapters 12 and 13 could be included next. While Chapter 14 on modeling the thermodynamics and fluid dynamics of real engine processes is primarily intended for the professional scientist and engineer, material from this chapter along with selections from Chapter 15 could be used to illustrate the performance, efficiency, and emissions characteristics of the different types of internal combustion engines. I have also used much of the more sophisticated material in Chapters 6 through 15 for review seminars on individual engine topics and more extensive courses for professional engineers, an additional important educational and reference opportunity.

Many individuals and organizations have assisted me in various ways as I have worked on this book over the past ten or so years. I am especially indebted to my colleagues in the Sloan Automotive Laboratory at M.I.T., Professors Wai K. Cheng, Ahmed F. Ghoniem, and James C. Keck, and Drs. Jack A. Ekchian, David P. Hoult, Joe M. Rife, and Victor W. Wong, for providing a stimulating environment in which to carry out engine research and for assuming additional burdens as a result of my writing. Many of the Sloan Automotive Laboratory's students have made significant contributions to this text through their research; their names appear in the reference lists. The U.S. Department of Energy provided support during the early stages of the text development and funded the work on engine cycle simulation used extensively in Chapters 14 and 15. I am grateful

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COMMONLY USED SYMBOLS, SUBSCRIPTS, AND ABBREVIATIONS†

1. SYMBOLS

a	Crank radius
	Sound speed
	Specific availability
$\mathbf{a}$	Acceleration
A	Area
A_C	Valve curtain area
A_{ch}	Cylinder head area
A_e	Exhaust port area
A_E	Effective area of flow restriction
A_i	Inlet port area
A_p	Piston crown area
B	Cylinder bore
	Steady-flow availability
c	Specific heat
c_p	Specific heat at constant pressure
c_s	Soot concentration (mass/volume)
c_v	Specific heat at constant volume
C	Absolute gas velocity

† Nomenclature specific to a section or chapter is defined in that section or chapter.

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C_s	Swirl coefficient
C_D	Discharge coefficient
	Vehicle drag coefficient
d	Diameter
d_n	Fuel-injection-nozzle orifice diameter
D	Diameter
	Diffusion coefficient
D_d	Droplet diameter
D_{SM}	Sauter mean droplet diameter
D_v	Valve diameter
e	Radiative emissive power
	Specific energy
E_A	Activation energy
f	Coefficient of friction
	Fuel mass fraction
F	Force
g	Gravitational acceleration
	Specific Gibbs free energy
G	Gibbs free energy
h	Clearance height
	Oil film thickness
	Specific enthalpy
h_c	Heat-transfer coefficient
h_p	Port open height
h_s	Sensible specific enthalpy
H	Enthalpy
I	Moment of inertia
J	Flux
k	Thermal conductivity
	Turbulent kinetic energy
k_i^+ , k_i^-	Forward, backward, rate constants for ith reaction
K	Constant
K_c	Equilibrium constant expressed in concentrations
K_p	Equilibrium constant expressed in partial pressures
l	Characteristic length scale
	Connecting rod length
l_T	Characteristic length scale of turbulent flame
L	Piston stroke
L_n	Fuel-injection-nozzle orifice length
L_v	Valve lift
m	Mass
$\dot{m}$	Mass flow rate
m_r	Mass of residual gas
M	Mach number
	Molecular weight

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n	Number of moles
	Polytropic exponent
n_R	Number of crank revolutions per power stroke
N	Crankshaft rotational speed
	Soot particle number density
	Turbocharger shaft speed
p	Cylinder pressure
	Pressure
P	Power
$\dot{q}$	Heat-transfer rate per unit area
	Heat-transfer rate per unit mass of fluid
Q	Heat transfer
$\dot{Q}$	Heat-transfer rate
Q_{ch}	Fuel chemical energy release or gross heat release
Q_{HV}	Fuel heating value
Q_n	Net heat release
r	Radius
r_c	Compression ratio
R	Connecting rod length/crank radius
	Gas constant
	Radius
R^+, R^-	One-way reaction rates
R_s	Swirl ratio
s	Crank axis to piston pin distance
	Specific entropy
S	Entropy
	Spray penetration
S_b	Turbulent burning speed
S_L	Laminar flame speed
S_p	Piston speed
t	Time
T	Temperature
	Torque
u	Specific internal energy
	Velocity
u'	Turbulence intensity
u_s	Sensible specific internal energy
u_T	Characteristic turbulent velocity
U	Compressor/turbine impellor tangential velocity
	Fluid velocity
	Internal energy
v	Specific volume
	Velocity
$\mathbf{v}$	Velocity
v_{ps}	Valve pseudo-flow velocity

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v_{sq}	Squish velocity
V	Cylinder volume
	Volume
V_c	Clearance volume
V_d	Displaced cylinder volume
w	Relative gas velocity
	Soot surface oxidation rate
W	Work transfer
W_c	Work per cycle
W_p	Pumping work
x, y, z	Spatial coordinates
x	Mass fraction
$\tilde{x}$	Mole fraction
x_b	Burned mass fraction
x_r	Residual mass fraction
y	H/C ratio of fuel
	Volume fraction
Y_α	Concentration of species α per unit mass
Z	Inlet Mach index
α	Angle
	Thermal diffusivity $k/(\rho c)$
β	Angle
γ	Specific heat ratio c_p/c_v
Γ_c	Angular momentum of charge
δ	Boundary-layer thickness
δ_L	Laminar flame thickness
$\Delta\tilde{h}^\circ_{f,i}$	Molal enthalpy of formation of species i
$\Delta\theta_b$	Rapid burning angle
$\Delta\theta_d$	Flame development angle
ε	$4/(4 + y)$: y = H/C ratio of fuel
	Turbulent kinetic energy dissipation rate
η_a	Availability conversion efficiency
η_c	Combustion efficiency
η_C	Compressor isentropic efficiency
η_{ch}	Charging efficiency
η_f	Fuel conversion efficiency
η_m	Mechanical efficiency
η_{sc}	Scavenging efficiency
η_t	Thermal conversion efficiency
η_T	Turbine isentropic efficiency
η_{tr}	Trapping efficiency
η_v	Volumetric efficiency
θ	Crank angle
λ	Relative air/fuel ratio
Λ	Delivery ratio

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μ	Dynamic viscosity
μ_i	Chemical potential of species i
ν	Kinematic viscosity μ/ρ
ν_i	Stoichiometric coefficient of species i
ξ	Flow friction coefficient
ρ	Density
$\rho_{a,0}$, $\rho_{a,i}$	Air density at standard, inlet conditions
σ	Normal stress
	Standard deviation
	Stefan-Boltzmann constant
	Surface tension
τ	Characteristic time
	Induction time
	Shear stress
τ_{id}	Ignition delay time
ϕ	Fuel/air equivalence ratio
Φ	Flow compressibility function [Eq. (C.11)]
	Isentropic compression function [Eq. (4.15*b*)]
ψ	Molar N/O ratio
	Throttle plate open angle
Ψ	Isentropic compression function [Eq. (4.15*a*)]
ω	Angular velocity
	Frequency

2. SUBSCRIPTS

a	Air
b	Burned gas
c	Coolant
	Cylinder
C	Compression stroke
	Compressor
cr	Crevice
e	Equilibrium
	Exhaust
E	Expansion stroke
f	Flame
	Friction
	Fuel
g	Gas
i	Indicated
	Intake
	Species i
ig	Gross indicated
in	Net indicated

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l	Liquid
L	Laminar
p	Piston
	Port
P	Prechamber
r, θ, z	r, θ, z components
R	Reference value
s	Isentropic
	Stoichiometric
T	Nozzle or orifice throat
	Turbine
	Turbulent
u	Unburned
v	Valve
w	Wall
x, y, z	x, y, z components
0	Reference value
	Stagnation value

3. NOTATION

Δ	Difference
$\bar{}$	Average or mean value
$\tilde{}$	Value per mole
[ ]	Concentration, moles/vol
{ }	Mass fraction
$\cdot$	Rate of change with time

4. ABBREVIATIONS

(A/F)	Air/fuel ratio
BC, ABC, BBC	Bottom-center crank position, after BC, before BC
CN	Fuel cetane number
Da	Damköhler number τ_T/τ_L
EGR	Exhaust gas recycle
EI	Emission index
EPC, EPO	Exhaust port closing, opening
EVC, EVO	Exhaust valve closing, opening
(F/A)	Fuel/air ratio
(G/F)	Gas/fuel ratio
IPC, IPO	Inlet port closing, opening
IVC, IVO	Inlet valve closing, opening
mep	Mean effective pressure
Nu	Nusselt number $h_c\, l/k$

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ON	Fuel octane number
Re	Reynolds number $\rho ul/\mu$
sfc	Specific fuel consumption
TC, ATC, BTC	Top-center crank position, after TC, before TC
We	Weber number $\rho_l u^2 D/\sigma$

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CHAPTER 1

ENGINE TYPES AND THEIR OPERATION

1.1 INTRODUCTION AND HISTORICAL PERSPECTIVE

The purpose of internal combustion engines is the production of mechanical power from the chemical energy contained in the fuel. In *internal* combustion engines, as distinct from *external* combustion engines, this energy is released by burning or oxidizing the fuel *inside* the engine. The fuel-air mixture before combustion and the burned products after combustion are the actual working fluids. The work transfers which provide the desired power output occur directly between these working fluids and the mechanical components of the engine. The internal combustion engines which are the subject of this book are spark-ignition engines (sometimes called Otto engines, or gasoline or petrol engines, though other fuels can be used) and compression-ignition or diesel engines.† Because of their simplicity, ruggedness and high power/weight ratio, these two types of engine have found wide application in transportation (land, sea, and air) and power generation. It is the fact that combustion takes place inside the work-

† The gas turbine is also, by this definition, an "internal combustion engine." Conventionally, however, the term is used for spark-ignition and compression-ignition engines. The operating principles of gas turbines are fundamentally different, and they are not discussed as separate engines in this book.

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producing part of these engines that makes their design and operating characteristics fundamentally different from those of other types of engine.

Practical heat engines have served mankind for over two and a half centuries. For the first 150 years, water, raised to steam, was interposed between the combustion gases produced by burning the fuel and the work-producing piston-in-cylinder expander. It was not until the 1860s that the internal combustion engine became a practical reality.[1, 2] The early engines developed for commercial use burned coal-gas air mixtures at atmospheric pressure—there was no compression before combustion. J. J. E. Lenoir (1822–1900) developed the first marketable engine of this type. Gas and air were drawn into the cylinder during the first half of the piston stroke. The charge was then ignited with a spark, the pressure increased, and the burned gases then delivered power to the piston for the second half of the stroke. The cycle was completed with an exhaust stroke. Some 5000 of these engines were built between 1860 and 1865 in sizes up to six horsepower. Efficiency was at best about 5 percent.

A more successful development—an atmospheric engine introduced in 1867 by Nicolaus A. Otto (1832–1891) and Eugen Langen (1833–1895)—used the pressure rise resulting from combustion of the fuel-air charge early in the outward stroke to accelerate a free piston and rack assembly so its momentum would generate a vacuum in the cylinder. Atmospheric pressure then pushed the piston inward, with the rack engaged through a roller clutch to the output shaft. Production engines, of which about 5000 were built, obtained thermal efficiencies of up to 11 percent. A slide valve controlled intake, ignition by a gas flame, and exhaust.

To overcome this engine's shortcomings of low thermal efficiency and excessive weight, Otto proposed an engine cycle with four piston strokes: an intake stroke, then a compression stroke before ignition, an expansion or power stroke where work was delivered to the crankshaft, and finally an exhaust stroke. He also proposed incorporating a stratified-charge induction system, though this was not achieved in practice. His prototype four-stroke engine first ran in 1876. A comparison between the Otto engine and its atmospheric-type predecessor indicates the reason for its success (see Table 1.1): the enormous reduction in engine weight and volume. This was the breakthrough that effectively founded the internal combustion engine industry. By 1890, almost 50,000 of these engines had been sold in Europe and the United States.

In 1884, an unpublished French patent issued in 1862 to Alphonse Beau de Rochas (1815–1893) was found which described the principles of the four-stroke cycle. This chance discovery cast doubt on the validity of Otto's own patent for this concept, and in Germany it was declared invalid. Beau de Rochas also outlined the conditions under which maximum efficiency in an internal combustion engine could be achieved. These were:

1. The largest possible cylinder volume with the minimum boundary surface
2. The greatest possible working speed

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TABLE 1.1
Comparison of Otto four-stroke cycle and Otto-Langen engines[2]

	Otto and Langen	Otto four-stroke
Brake horsepower	2	2
Weight, lb, approx.	4000	1250
Piston displacement, in^3	4900	310
Power strokes per min	28	80
Shaft speed, rev/min	90	160
Mechanical efficiency, %	68	84
Overall efficiency, %	11	14
Expansion ratio	10	2.5

3. The greatest possible expansion ratio
4. The greatest possible pressure at the beginning of expansion

The first two conditions hold heat losses from the charge to a minimum. The third condition recognizes that the greater the expansion of the postcombustion gases, the greater the work extracted. The fourth condition recognizes that higher initial pressures make greater expansion possible, and give higher pressures throughout the process, both resulting in greater work transfer. Although Beau de Rochas' unpublished writings predate Otto's developments, he never reduced these ideas to practice. Thus Otto, in the broader sense, was the inventor of the modern internal combustion engine as we know it today.

Further developments followed fast once the full impact of what Otto had achieved became apparent. By the 1880s several engineers (e.g., Dugald Clerk, 1854–1913, and James Robson, 1833–1913, in England and Karl Benz, 1844–1929, in Germany) had successfully developed two-stroke internal combustion engines where the exhaust and intake processes occur during the end of the power stroke and the beginning of the compression stroke. James Atkinson (1846–1914) in England made an engine with a longer expansion than compression stroke, which had a high efficiency for the times but mechanical weaknesses. It was recognized that efficiency was a direct function of expansion ratio, yet compression ratios were limited to less than four if serious knock problems were to be avoided with the available fuels. Substantial carburetor and ignition system developments were required, and occurred, before high-speed gasoline engines suitable for automobiles became available in the late 1880s. Stationary engine progress also continued. By the late 1890s, large single-cylinder engines of 1.3-m bore fueled by low-energy blast furnace gas produced 600 bhp at 90 rev/min. In Britain, legal restrictions on volatile fuels turned their engine builders toward kerosene. Low compression ratio "oil" engines with heated external fuel vaporizers and electric ignition were developed with efficiencies comparable to those of gas engines (14 to 18 percent). The Hornsby-Ackroyd engine became the most

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popular oil engine in Britain, and was also built in large numbers in the United States.[2]

In 1892, the German engineer Rudolf Diesel (1858–1913) outlined in his patent a new form of internal combustion engine. His concept of initiating combustion by injecting a liquid fuel into air heated solely by compression permitted a doubling of efficiency over other internal combustion engines. Much greater expansion ratios, without detonation or knock, were now possible. However, even with the efforts of Diesel and the resources of M.A.N. in Ausburg combined, it took five years to develop a practical engine.

Engine developments, perhaps less fundamental but nonetheless important to the steadily widening internal combustion engine markets, have continued ever since.[2–4] One more recent major development has been the rotary internal combustion engine. Although a wide variety of experimental rotary engines have been proposed over the years,[5] the first practical rotary internal combustion engine, the Wankel, was not successfully tested until 1957. That engine, which evolved through many years of research and development, was based on the designs of the German inventor Felix Wankel.[6, 7]

Fuels have also had a major impact on engine development. The earliest engines used for generating mechanical power burned gas. Gasoline, and lighter fractions of crude oil, became available in the late 1800s and various types of carburetors were developed to vaporize the fuel and mix it with air. Before 1905 there were few problems with gasoline; though compression ratios were low (4 or less) to avoid knock, the highly volatile fuel made starting easy and gave good cold weather performance. However, a serious crude oil shortage developed, and to meet the fivefold increase in gasoline demand between 1907 and 1915, the yield from crude had to be raised. Through the work of William Burton (1865–1954) and his associates of Standard Oil of Indiana, a thermal cracking process was developed whereby heavier oils were heated under pressure and decomposed into less complex more volatile compounds. These thermally cracked gasolines satisfied demand, but their higher boiling point range created cold weather starting problems. Fortunately, electrically driven starters, introduced in 1912, came along just in time.

On the farm, kerosene was the logical fuel for internal combustion engines since it was used for heat and light. Many early farm engines had heated carburetors or vaporizers to enable them to operate with such a fuel.

The period following World War I saw a tremendous advance in our understanding of how fuels affect combustion, and especially the problem of knock. The antiknock effect of tetraethyl lead was discovered at General Motors,[4] and it became commercially available as a gasoline additive in the United States in 1923. In the late 1930s, Eugene Houdry found that vaporized oils passed over an activated catalyst at 450 to 480°C were converted to high-quality gasoline in much higher yields than was possible with thermal cracking. These advances, and others, permitted fuels with better and better antiknock properties to be produced in large quantities; thus engine compression ratios steadily increased, improving power and efficiency.

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During the past three decades, new factors for change have become important and now significantly affect engine design and operation. These factors are, first, the need to control the automotive contribution to urban air pollution and, second, the need to achieve significant improvements in automotive fuel consumption.

The automotive air-pollution problem became apparent in the 1940s in the Los Angeles basin. In 1952, it was demonstrated by Prof. A. J. Haagen-Smit that the smog problem there resulted from reactions between oxides of nitrogen and hydrocarbon compounds in the presence of sunlight.[8] In due course it became clear that the automobile was a major contributor to hydrocarbon and oxides of nitrogen emissions, as well as the prime cause of high carbon monoxide levels in urban areas. Diesel engines are a significant source of small soot or smoke particles, as well as hydrocarbons and oxides of nitrogen. Table 1.2 outlines the dimensions of the problem. As a result of these developments, emission standards for automobiles were introduced first in California, then nationwide in the United States, starting in the early 1960s. Emission standards in Japan and Europe, and for other engine applications, have followed. Substantial reductions in emissions from spark-ignition and diesel engines have been achieved. Both the use of catalysts in spark-ignition engine exhaust systems for emissions control and concern over the toxicity of lead antiknock additives have resulted in the reappearance of unleaded gasoline as a major part of the automotive fuels market. Also, the maximum lead content in leaded gasoline has been substantially reduced. The emission-control requirements and these fuel developments have produced significant changes in the way internal combustion engines are designed and operated.

Internal combustion engines are also an important source of noise. There are several sources of engine noise: the exhaust system, the intake system, the fan used for cooling, and the engine block surface. The noise may be generated by aerodynamic effects, may be due to forces that result from the combustion process, or may result from mechanical excitation by rotating or reciprocating engine components. Vehicle noise legislation to reduce emissions to the environment was first introduced in the early 1970s.

During the 1970s the price of crude petroleum rose rapidly to several times its cost (in real terms) in 1970, and concern built up regarding the longer-term availability of petroleum. Pressures for substantial improvements in internal combustion engine efficiency (in all its many applications) have become very substantial indeed. Yet emission-control requirements have made improving engine fuel consumption more difficult, and the removal and reduction of lead in gasoline has forced spark-ignition engine compression ratios to be reduced. Much work is being done on the use of alternative fuels to gasoline and diesel. Of the non-petroleum-based fuels, natural gas, and methanol and ethanol (methyl and ethyl alcohols) are receiving the greatest attention, while synthetic gasoline and diesel made from shale oil or coal, and hydrogen could be longer-term possibilities.

It might be thought that after over a century of development, the internal

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TABLE 1.2
The automotive urban air-pollution problem

Pollutant	Impact	Mobile source emissions as % of total†	Automobile emissions		Truck emissions††	
			Uncontrolled vehicles, g/km‡	Reduction in new vehicles, % ¶	SI engines, g/km	Diesel, g/km
Oxides of nitrogen (NO and NO_2)	Reactant in photochemical smog; NO_2 is toxic	40–60	2.5	75	7	12
Carbon monoxide (CO)	Toxic	90	65	95	150	17
Unburned hydrocarbons (HC, many hydrocarbon compounds)	Reactant in photochemical smog	30–50	10	90	17‡‡	3
Particulates (soot and absorbed hydrocarbon compounds)	Reduces visibility; some of HC compounds mutagenic	50	0.5§	40§	n	0.5

† Depends on type of urban area and source mix.

‡ Average values for pre-1968 automobiles which had no emission controls, determined by U.S. test procedure which simulates typical urban and highway driving. Exhaust emissions, except for HC where 55 percent are exhaust emissions, 20 percent are evaporative emissions from fuel tank and carburetor, and 25 percent are crankcase blowby gases.

§ Diesel engine automobiles only. Particulate emissions from spark-ignition engines are negligible.

¶ Compares emissions from new spark-ignition engine automobiles with uncontrolled automobile levels in previous column. Varies from country to country. The United States, Canada, Western Europe, and Japan have standards with different degrees of severity. The United States, Europe, and Japan have different test procedures. Standards are strictest in the United States and Japan.

†† Representative average emission levels for trucks.

‡‡ With 95 percent exhaust emissions and 5 percent evaporative emissions.

n = negligible.

combustion engine has reached its peak and little potential for further improvement remains. Such is not the case. Conventional spark-ignition and diesel engines continue to show substantial improvements in efficiency, power, and degree of emission control. New materials now becoming available offer the possibilities of reduced engine weight, cost, and heat losses, and of different and more efficient internal combustion engine systems. Alternative types of internal combustion engines, such as the stratified-charge (which combines characteristics normally associated with either the spark-ignition or diesel) with its wider fuel tolerance, may become sufficiently attractive to reach large-scale production. The engine development opportunities of the future are substantial. While they

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present a formidable challenge to automotive engineers, they will be made possible in large part by the enormous expansion of our knowledge of engine processes which the last twenty years has witnessed.

1.2 ENGINE CLASSIFICATIONS

There are many different types of internal combustion engines. They can be classified by:

1. *Application.* Automobile, truck, locomotive, light aircraft, marine, portable power system, power generation
2. *Basic engine design.* Reciprocating engines (in turn subdivided by arrangement of cylinders: e.g., in-line, V, radial, opposed), rotary engines (Wankel and other geometries)
3. *Working cycle.* Four-stroke cycle: naturally aspirated (admitting atmospheric air), supercharged (admitting precompressed fresh mixture), and turbocharged (admitting fresh mixture compressed in a compressor driven by an exhaust turbine), two-stroke cycle: crankcase scavenged, supercharged, and turbocharged
4. *Valve or port design and location.* Overhead (or I-head) valves, underhead (or L-head) valves, rotary valves, cross-scavenged porting (inlet and exhaust ports on opposite sides of cylinder at one end), loop-scavenged porting (inlet and exhaust ports on same side of cylinder at one end), through- or uniflow-scavenged (inlet and exhaust ports or valves at different ends of cylinder)
5. *Fuel.* Gasoline (or petrol), fuel oil (or diesel fuel), natural gas, liquid petroleum gas, alcohols (methanol, ethanol), hydrogen, dual fuel
6. *Method of mixture preparation.* Carburetion, fuel injection into the intake ports or intake manifold, fuel injection into the engine cylinder
7. *Method of ignition.* Spark ignition (in conventional engines where the mixture is uniform and in stratified-charge engines where the mixture is non-uniform), compression ignition (in conventional diesels, as well as ignition in gas engines by pilot injection of fuel oil)
8. *Combustion chamber design.* Open chamber (many designs: e.g., disc, wedge, hemisphere, bowl-in-piston), divided chamber (small and large auxiliary chambers; many designs: e.g., swirl chambers, prechambers)
9. *Method of load control.* Throttling of fuel and air flow together so mixture composition is essentially unchanged, control of fuel flow alone, a combination of these
10. *Method of cooling.* Water cooled, air cooled, uncooled (other than by natural convection and radiation)

All these distinctions are important and they illustrate the breadth of engine designs available. Because this book approaches the operating and emissions

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TABLE 1.3
Classification of reciprocating engines by application

Class	Service	Approximate engine power range, kW	Predominant type		
			D or SI	Cycle	Cooling
Road vehicles	Motorcycles, scooters	0.75–70	SI	2, 4	A
	Small passenger cars	15–75	SI	4	A, W
	Large passenger cars	75–200	SI	4	W
	Light commercial	35–150	SI, D	4	W
	Heavy (long-distance) commercial	120–400	D	4	W
Off-road vehicles	Light vehicles (factory, airport, etc.)	1.5–15	SI	2, 4	A, W
	Agricultural	3–150	SI, D	2, 4	A, W
	Earth moving	40–750	D	2, 4	W
	Military	40–2000	D	2, 4	A, W
Railroad	Rail cars	150–400	D	2, 4	W
	Locomotives	400–3000	D	2, 4	W
Marine	Outboard	0.4–75	SI	2	W
	Inboard motorcrafts	4–750	SI, D	4	W
	Light naval craft	30–2200	D	2, 4	W
	Ships	3500–22,000	D	2, 4	W
	Ships' auxiliaries	75–750	D	4	W
Airborne vehicles	Airplanes	45–2700	SI	4	A
	Helicopters	45–1500	SI	4	A
Home use	Lawn mowers	0.7–3	SI	2, 4	A
	Snow blowers	2–5	SI	2, 4	A
	Light tractors	2–8	SI	4	A
Stationary	Building service	7–400	D	2, 4	W
	Electric power	35–22,000	D	2, 4	W
	Gas pipeline	750–5000	SI	2, 4	W

SI = spark-ignition; D = diesel; A = air cooled; W = water cooled.
Source: Adapted from Taylor.[9]

characteristics of internal combustion engines from a fundamental point of view, the method of ignition has been selected as the primary classifying feature. From the method of ignition—spark-ignition or compression-ignition†—follow the important characteristics of the fuel used, method of mixture preparation, combustion chamber design, method of load control, details of the combustion process, engine emissions, and operating characteristics. Some of the other classifications are used as subcategories within this basic classification. The engine operating cycle—four-stroke or two-stroke—is next in importance; the principles of these two cycles are described in the following section.

Table 1.3 shows the most common applications of internal combustion

† In the remainder of the book, these terms will often be abbreviated by SI and CI, respectively.

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engines, the predominant type of engine used in each classification listed, and the approximate engine power range in each type of service.

1.3 ENGINE OPERATING CYCLES

Most of this book is about reciprocating engines, where the piston moves back and forth in a cylinder and transmits power through a connecting rod and crank mechanism to the drive shaft as shown in Fig. 1-1. The steady rotation of the crank produces a cyclical piston motion. The piston comes to rest at the top-center (TC) crank position and bottom-center (BC) crank position when the cylinder volume is a minimum or maximum, respectively.† The minimum cylinder volume is called the clearance volume V_c. The volume swept out by the

† These crank positions are also referred to as top-dead-center (TDC) and bottom-dead-center (BDC).

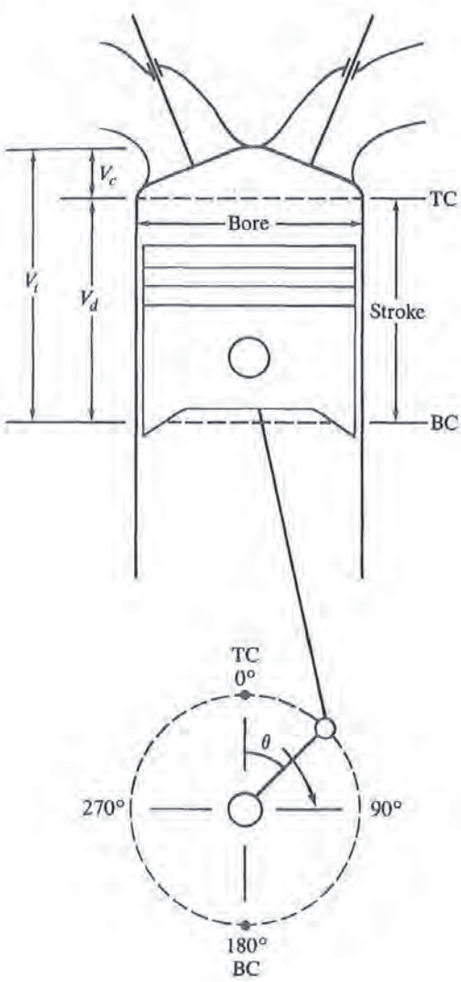


FIGURE 1-1
Basic geometry of the reciprocating internal combustion engine. V_c, V_d and V_t indicate clearance, displaced, and total cylinder volumes.

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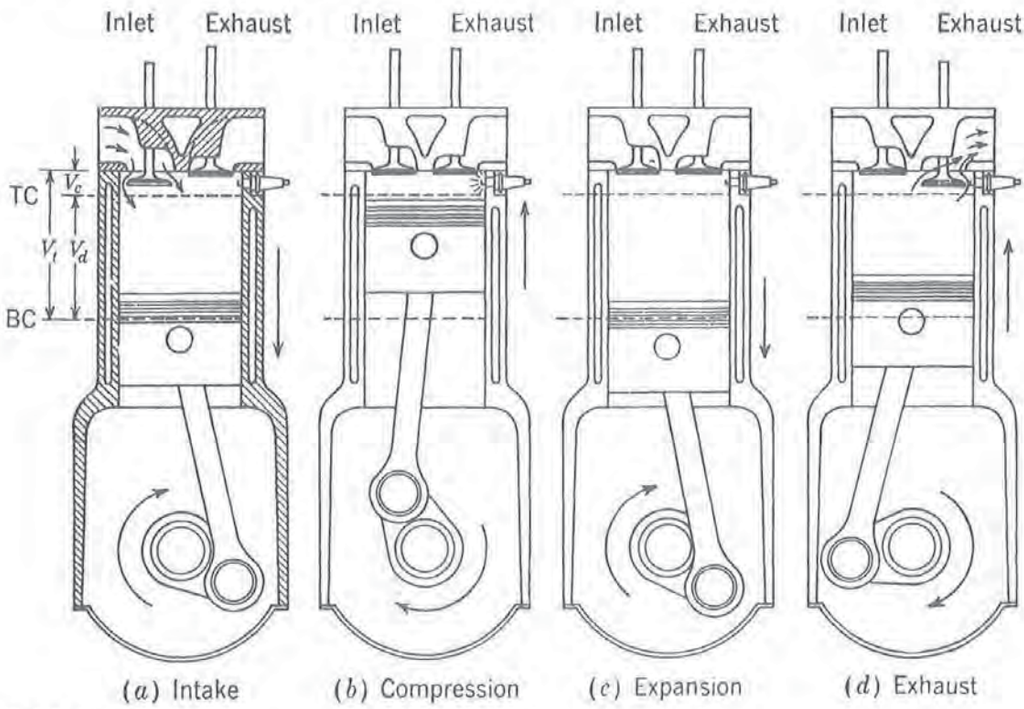


FIGURE 1-2
The four-stroke operating cycle.[10]

piston, the difference between the maximum or total volume V_t and the clearance volume, is called the displaced or swept volume V_d. The ratio of maximum volume to minimum volume is the compression ratio r_c. Typical values of r_c are 8 to 12 for SI engines and 12 to 24 for CI engines.

The majority of reciprocating engines operate on what is known as the *four-stroke cycle*. Each cylinder requires four strokes of its piston—two revolutions of the crankshaft—to complete the sequence of events which produces one power stroke. Both SI and CI engines use this cycle which comprises (see Fig. 1-2):

1. *An intake stroke*, which starts with the piston at TC and ends with the piston at BC, which draws fresh mixture into the cylinder. To increase the mass inducted, the inlet valve opens shortly before the stroke starts and closes after it ends.
2. *A compression stroke*, when both valves are closed and the mixture inside the cylinder is compressed to a small fraction of its initial volume. Toward the end of the compression stroke, combustion is initiated and the cylinder pressure rises more rapidly.
3. *A power stroke*, or expansion stroke, which starts with the piston at TC and ends at BC as the high-temperature, high-pressure, gases push the piston down and force the crank to rotate. About five times as much work is done on the piston during the power stroke as the piston had to do during compression.

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As the piston approaches BC the exhaust valve opens to initiate the exhaust process and drop the cylinder pressure to close to the exhaust pressure.

4. *An exhaust stroke*, where the remaining burned gases exit the cylinder: first, because the cylinder pressure may be substantially higher than the exhaust pressure; then as they are swept out by the piston as it moves toward TC. As the piston approaches TC the inlet valve opens. Just after TC the exhaust valve closes and the cycle starts again.

Though often called the Otto cycle after its inventor, Nicolaus Otto, who built the first engine operating on these principles in 1876, the more descriptive four-stroke nomenclature is preferred.

The four-stroke cycle requires, for each engine cylinder, two crankshaft revolutions for each power stroke. To obtain a higher power output from a given engine size, and a simpler valve design, the *two-stroke* cycle was developed. The two-stroke cycle is applicable to both SI and CI engines.

Figure 1-3 shows one of the simplest types of two-stroke engine designs. Ports in the cylinder liner, opened and closed by the piston motion, control the exhaust and inlet flows while the piston is close to BC. The two strokes are:

1. *A compression stroke*, which starts by closing the inlet and exhaust ports, and then compresses the cylinder contents and draws fresh charge into the crankcase. As the piston approaches TC, combustion is initiated.

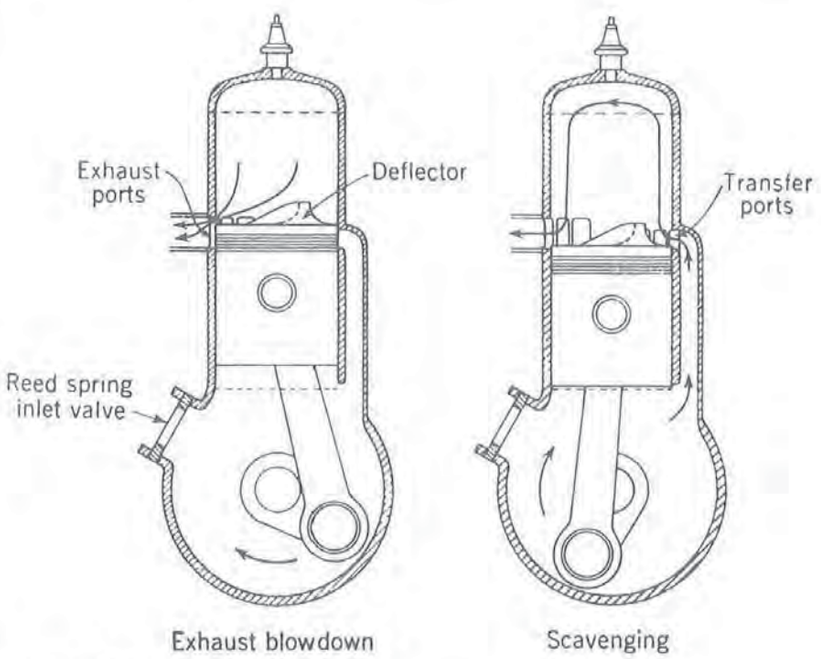


FIGURE 1-3
The two-stroke operating cycle. A crankcase-scavenged engine is shown.[10]

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2. *A power or expansion stroke*, similar to that in the four-stroke cycle until the piston approaches BC, when first the exhaust ports and then the intake ports are uncovered (Fig. 1-3). Most of the burnt gases exit the cylinder in an exhaust blowdown process. When the inlet ports are uncovered, the fresh charge which has been compressed in the crankcase flows into the cylinder. The piston and the ports are generally shaped to deflect the incoming charge from flowing directly into the exhaust ports and to achieve effective scavenging of the residual gases.

Each engine cycle with one power stroke is completed in one crankshaft revolution. However, it is difficult to fill completely the displaced volume with fresh charge, and some of the fresh mixture flows directly out of the cylinder during the scavenging process.† The example shown is a *cross-scavenged* design; other approaches use *loop-scavenging* or *uniflow* systems (see Sec. 6.6).

1.4 ENGINE COMPONENTS

Labeled cutaway drawings of a four-stroke SI engine and a two-stroke CI engine are shown in Figs. 1-4 and 1-5, respectively. The spark-ignition engine is a four-cylinder in-line automobile engine. The diesel is a large V eight-cylinder design with a uniflow scavenging process. The function of the major components of these engines and their construction materials will now be reviewed.

The engine cylinders are contained in the engine block. The block has traditionally been made of gray cast iron because of its good wear resistance and low cost. Passages for the cooling water are cast into the block. Heavy-duty and truck engines often use removable cylinder sleeves pressed into the block that can be replaced when worn. These are called *wet liners* or *dry liners* depending on whether the sleeve is in direct contact with the cooling water. Aluminum is being used increasingly in smaller SI engine blocks to reduce engine weight. Iron cylinder liners may be inserted at the casting stage, or later on in the machining and assembly process. The crankcase is often integral with the cylinder block.

The crankshaft has traditionally been a steel forging; nodular cast iron crankshafts are also accepted normal practice in automotive engines. The crankshaft is supported in main bearings. The maximum number of main bearings is one more than the number of cylinders; there may be less. The crank has eccentric portions (crank throws); the connecting rod big-end bearings attach to the crank pin on each throw. Both main and connecting rod bearings use steel-backed precision inserts with bronze, babbit, or aluminum as the bearing materials. The crankcase is sealed at the bottom with a pressed-steel or cast aluminum oil pan which acts as an oil reservoir for the lubricating system.

† It is primarily for this reason that two-stroke SI engines are at a disadvantage because the lost fresh charge contains fuel and air.

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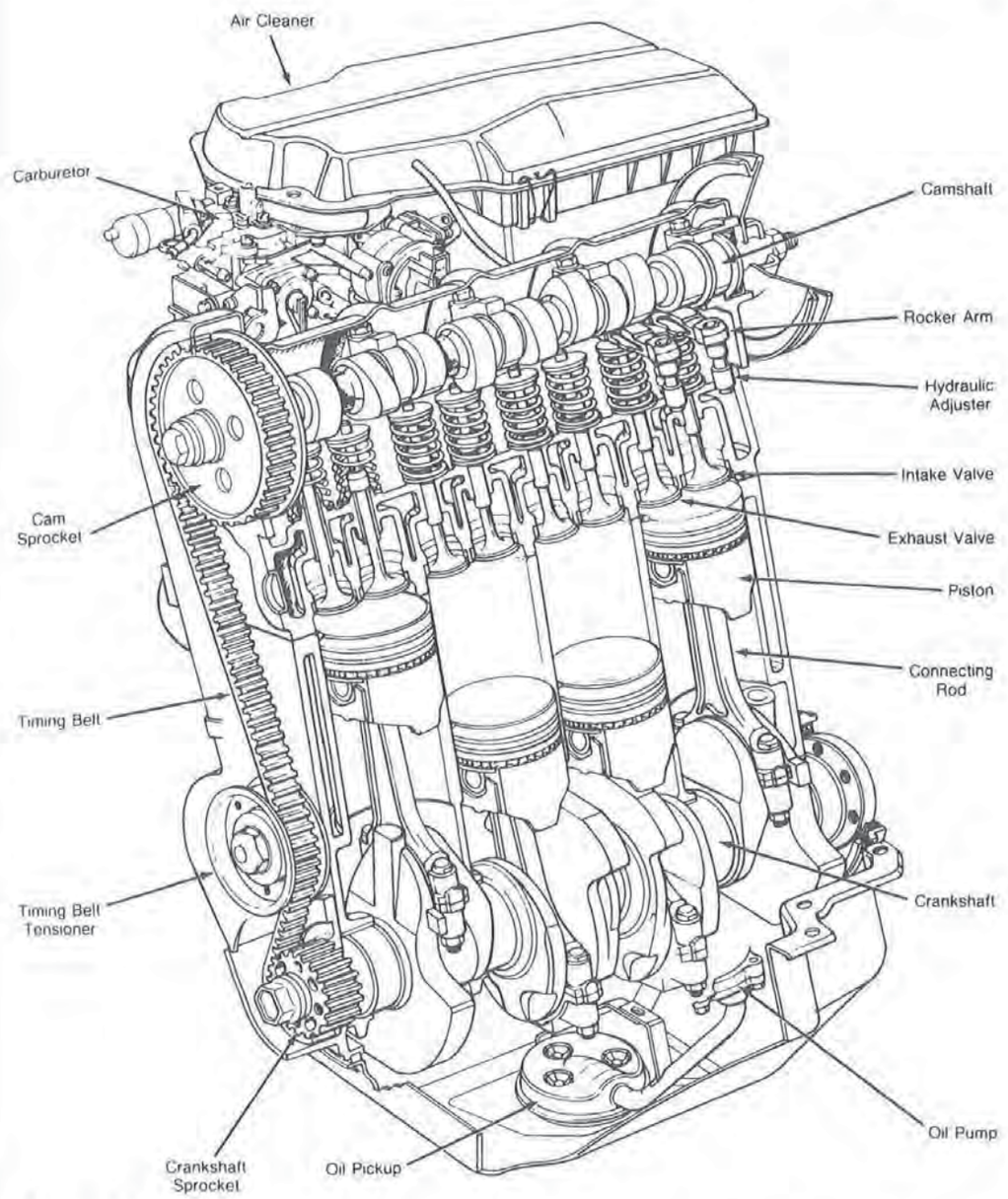


FIGURE 1-4
Cutaway drawing of Chrysler 2.2-liter displacement four-cylinder spark-ignition engine.[11] Bore 87.5 mm, stroke 92 mm, compression ratio 8.9, maximum power 65 kW at 5000 rev/min.

Pistons are made of aluminum in small engines or cast iron in larger slower-speed engines. The piston both seals the cylinder and transmits the combustion-generated gas pressure to the crank pin via the connecting rod. The connecting rod, usually a steel or alloy forging (though sometimes aluminum in small engines), is fastened to the piston by means of a steel piston pin through the rod upper end. The piston pin is usually hollow to reduce its weight.

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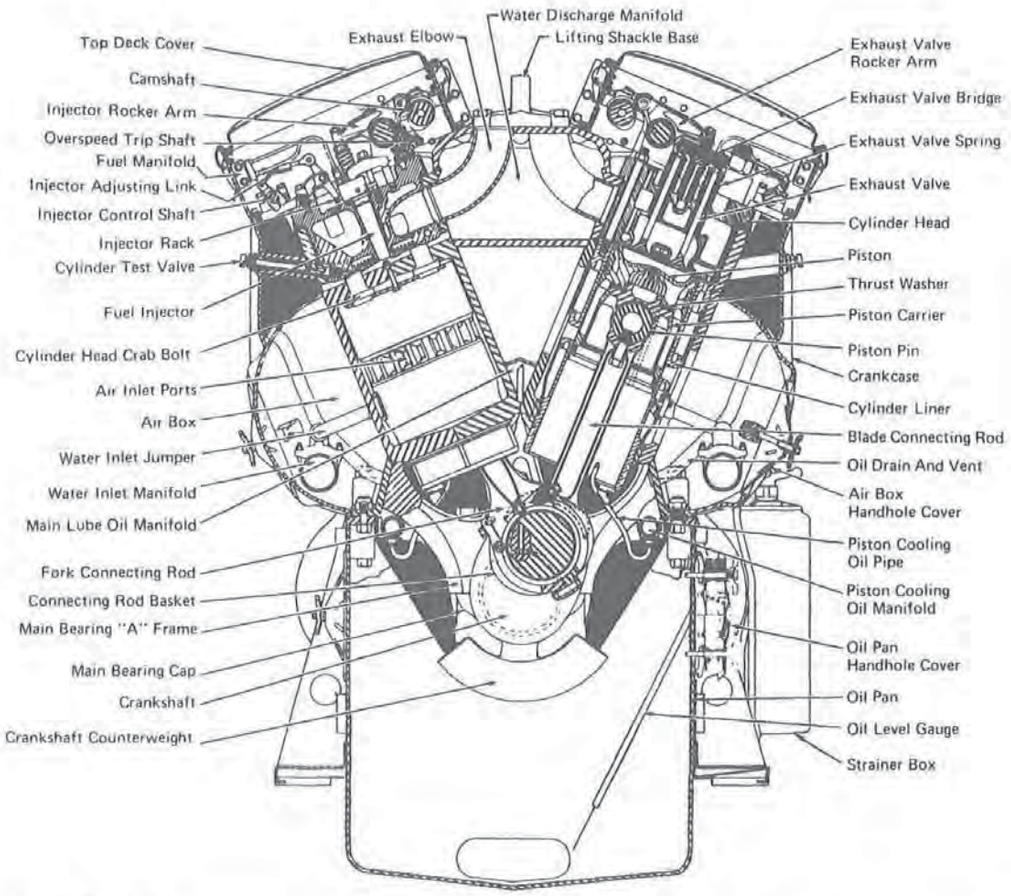


FIGURE 1-5
Cross-section drawing of an Electro-Motive two-stroke cycle diesel engine. This engine uses a uniflow scavenging process with inlet ports in the cylinder liner and four exhaust valves in the cylinder head. Bore 230.2 mm, stroke 254 mm, displaced volume per cylinder 10.57 liters, rated speed 750–900 rev/min. (*Courtesy Electro-Motive Division, General Motors Corporation.*)

The oscillating motion of the connecting rod exerts an oscillating force on the cylinder walls via the piston skirt (the region below the piston rings). The piston skirt is usually shaped to provide appropriate thrust surfaces. The piston is fitted with rings which ride in grooves cut in the piston head to seal against gas leakage and control oil flow. The upper rings are compression rings which are forced outward against the cylinder wall and downward onto the groove face. The lower rings scrape the surplus oil from the cylinder wall and return it to the crankcase. The crankcase must be ventilated to remove gases which blow by the piston rings, to prevent pressure buildup.

The cylinder head (or heads in V engines) seals off the cylinders and is made of cast iron or aluminum. It must be strong and rigid to distribute the gas forces acting on the head as uniformly as possible through the engine block. The cylinder head contains the spark plug (for an SI engine) or fuel injector (for a CI engine), and, in overhead valve engines, parts of the valve mechanism.

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The valves shown in Fig. 1-4 are poppet valves, the valve type normally used in four-stroke engines. Valves are made from forged alloy steel; the cooling of the exhaust valve which operates at about 700°C may be enhanced by using a hollow stem partially filled with sodium which through evaporation and condensation carries heat from the hot valve head to the cooler stem. Most modern spark-ignition engines have overhead valve locations (sometimes called valve-in-head or I-head configurations) as shown in Fig. 1-4. This geometry leads to a compact combustion chamber with minimum heat losses and flame travel time, and improves the breathing capacity. Previous geometries such as the L head where valves are to one side of the cylinder are now only used in small engines.

The valve stem moves in a valve guide, which can be an integral part of the cylinder head (or engine block for L-head engines), or may be a separate unit pressed into the head (or block). The valve seats may be cut in the head or block metal (if cast iron) or hard steel inserts may be pressed into the head or block. A valve spring, attached to the valve stem with a spring washer and split keeper, holds the valve closed. A valve rotator turns the valves a few degrees on opening to wipe the valve seat, avoid local hot spots, and prevent deposits building up in the valve guide.

A camshaft made of cast iron or forged steel with one cam per valve is used to open and close the valves. The cam surfaces are hardened to obtain adequate life. In four-stroke cycle engines, camshafts turn at one-half the crankshaft speed. Mechanical or hydraulic lifters or tappets slide in the block and ride on the cam. Depending on valve and camshaft location, additional members are required to transmit the tappet motion to the valve stem; e.g., in in-head valve engines with the camshaft at the side, a push rod and rocker arm are used. A recent trend in automotive engines is to mount the camshaft over the head with the cams acting either directly or through a pivoted follower on the valve. Camshafts are gear, belt, or chain driven from the crankshaft.

An intake manifold (aluminum or cast iron) and an exhaust manifold (generally of cast iron) complete the SI engine assembly. Other engine components specific to spark-ignition engines—carburetor, fuel injectors, ignition systems—are described more fully in the remaining sections in this chapter.

The two-stroke cycle CI engine shown in Fig. 1-5 is of the uniflow scavenged design. The burned gases exhaust through four valves in the cylinder head. These valves are controlled through cam-driven rocker arms. Fresh air is compressed and fed to the air box by a Roots blower. The air inlet ports at the bottom of each cylinder liner are uncovered by the descending piston, and the scavenging air flows upward along the cylinder axis. The fuel injectors are mounted in the cylinder head and are driven by the camshaft through rocker arms. Diesel fuel-injection systems are discussed in more detail in Sec. 1.7.

1.5 SPARK-IGNITION ENGINE OPERATION

In SI engines the air and fuel are usually mixed together in the intake system prior to entry to the engine cylinder, using a carburetor (Fig. 1-6) or fuel-injection system (Fig. 1-7). In automobile applications, the temperature of the air entering

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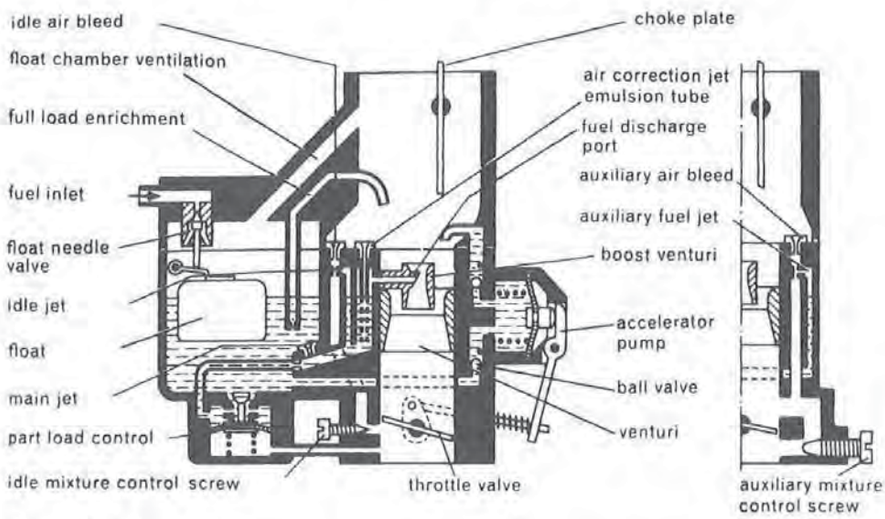


FIGURE 1-6
Cross section of single-barrel downdraft carburetor.[12] (*Courtesy Robert Bosch GmbH and SAE.*)

the intake system is controlled by mixing ambient air with air heated by contact with the exhaust manifold. The ratio of mass flow of air to mass flow of fuel must be held approximately constant at about 15 to ensure reliable combustion. The

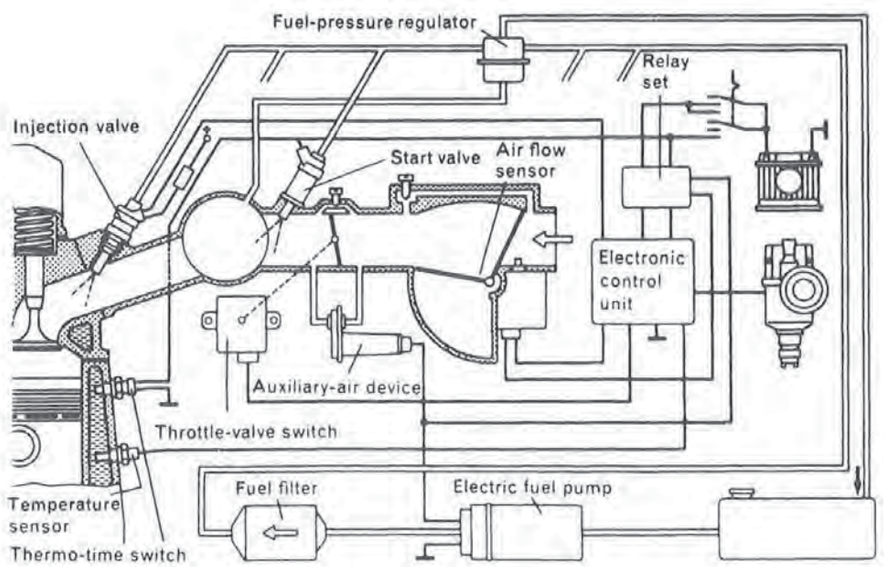


FIGURE 1-7
Schematic drawing of L-Jetronic port electronic fuel-injection system.[12] (*Courtesy Robert Bosch GmbH and SAE.*)

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carburetor meters an appropriate fuel flow for the engine air flow in the following manner. The air flow through the venturi (a converging-diverging nozzle) sets up a pressure difference between the venturi inlet and throat which is used to meter an appropriate amount of fuel from the float chamber, through a series of orifices, into the air flow at the venturi throat. Just downstream of the venturi is a throttle valve or plate which controls the combined air and fuel flow, and thus the engine output. The intake flow is throttled to below atmospheric pressure by reducing the flow area when the power required (at any engine speed) is below the maximum which is obtained when the throttle is wide open. The intake manifold is usually heated to promote faster evaporation of the liquid fuel and obtain more uniform fuel distribution between cylinders.

Fuel injection into the intake manifold or inlet port is an increasingly common alternative to a carburetor. With port injection, fuel is injected through individual injectors from a low-pressure fuel supply system into each intake port. There are several different types of systems: mechanical injection using an injection pump driven by the engine; mechanical, driveless, continuous injection; electronically controlled, driveless, injection. Figure 1-7 shows an example of an electronically controlled system. In this system, the air flow rate is measured directly; the injection valves are actuated twice per cam shaft revolution by injection pulses whose duration is determined by the electronic control unit to provide the desired amount of fuel per cylinder per cycle.[12] An alternative approach is to use a single fuel injector located above the throttle plate in the position normally occupied by the carburetor. This approach permits electronic control of the fuel flow at reduced cost.

The sequence of events which take place inside the engine cylinder is illustrated in Fig. 1-8. Several variables are plotted against crank angle through the entire four-stroke cycle. Crank angle is a useful independent variable because engine processes occupy almost constant crank angle intervals over a wide range of engine speeds. The figure shows the valve timing and volume relationship for a typical automotive spark-ignition engine. To maintain high mixture flows at high engine speeds (and hence high power outputs) the inlet valve, which opens before TC, closes substantially after BC. During intake, the inducted fuel and air mix in the cylinder with the *residual* burned gases remaining from the previous cycle. After the intake valve closes, the cylinder contents are compressed to above atmospheric pressure and temperature as the cylinder volume is reduced. Some heat transfer to the piston, cylinder head, and cylinder walls occurs but the effect on unburned gas properties is modest.

Between 10 and 40 crank angle degrees before TC an electrical discharge across the spark plug starts the combustion process. A distributor, a rotating switch driven off the camshaft, interrupts the current from the battery through the primary circuit of the ignition coil. The secondary winding of the ignition coil, connected to the spark plug, produces a high voltage across the plug electrodes as the magnetic field collapses. Traditionally, cam-operated breaker points have been used; in most automotive engines, the switching is now done electronically. A turbulent flame develops from the spark discharge, propagates

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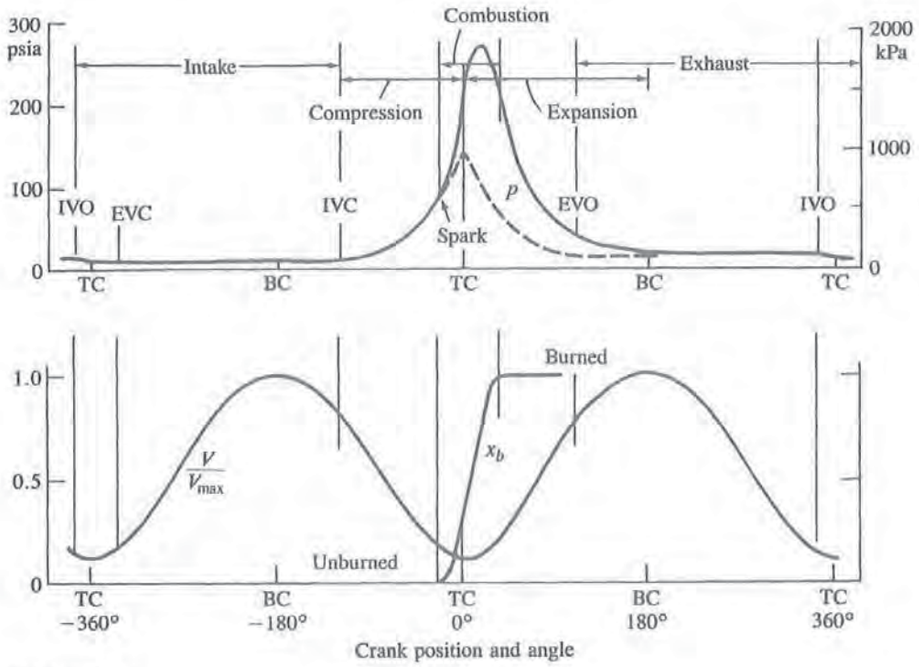


FIGURE 1-8
Sequence of events in four-stroke spark-ignition engine operating cycle. Cylinder pressure p (solid line, firing cycle; dashed line, motored cycle), cylinder volume V/V_{max}, and mass fraction burned x_b are plotted against crank angle.

across the mixture of air, fuel, and residual gas in the cylinder, and extinguishes at the combustion chamber wall. The duration of this burning process varies with engine design and operation, but is typically 40 to 60 crank angle degrees, as shown in Fig. 1-8. As fuel-air mixture burns in the flame, the cylinder pressure in Fig. 1-8 (solid line) rises above the level due to compression alone (dashed line). This latter curve—called the motored cylinder pressure—is the pressure trace obtained from a motored or nonfiring engine.† Note that due to differences in the flow pattern and mixture composition between cylinders, and within each cylinder cycle-by-cycle, the development of each combustion process differs somewhat. As a result, the shape of the pressure versus crank angle curve in each cylinder, and cycle-by-cycle, is not exactly the same.

There is an optimum spark timing which, for a given mass of fuel and air inside the cylinder, gives maximum torque. More advanced (earlier) timing or retarded (later) timing than this optimum gives lower output. Called *maximum*

† In practice, the intake and compression processes of a firing engine and a motored engine are not exactly the same due to the presence of burned gases from the previous cycle under firing conditions.

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brake-torque (MBT) timing,† this optimum timing is an empirical compromise between starting combustion too early in the compression stroke (when the work transfer is *to* the cylinder gases) and completing combustion too late in the expansion stroke (and so lowering peak expansion stroke pressures).

About two-thirds of the way through the expansion stroke, the exhaust valve starts to open. The cylinder pressure is greater than the exhaust manifold pressure and a *blowdown* process occurs. The burned gases flow through the valve into the exhaust port and manifold until the cylinder pressure and exhaust pressure equilibrate. The duration of this process depends on the pressure level in the cylinder. The piston then *displaces* the burned gases from the cylinder into the manifold during the exhaust stroke. The exhaust valve opens before the end of the expansion stroke to ensure that the blowdown process does not last too far into the exhaust stroke. The actual timing is a compromise which balances reduced work transfer to the piston before BC against reduced work transfer to the cylinder contents after BC.

The exhaust valve remains open until just after TC; the intake opens just before TC. The valves are opened and closed slowly to avoid noise and excessive cam wear. To ensure the valves are fully open when piston velocities are at their highest, the valve open periods often overlap. If the intake flow is throttled to below exhaust manifold pressure, then backflow of burned gases into the intake manifold occurs when the intake valve is first opened.

1.6 EXAMPLES OF SPARK-IGNITION ENGINES

This section presents examples of production spark-ignition engines to illustrate the different types of engines in common use.

Small SI engines are used in many applications: in the home (e.g., lawn mowers, chain saws), in portable power generation, as outboard motorboat engines, and in motorcycles. These are often single-cylinder engines. In the above applications, light weight, small bulk, and low cost in relation to the power generated are the most important characteristics; fuel consumption, engine vibration, and engine durability are less important. A single-cylinder engine gives only one power stroke per revolution (two-stroke cycle) or two revolutions (four-stroke cycle). Hence, the torque pulses are widely spaced, and engine vibration and smoothness are significant problems.

Multicylinder engines are invariably used in automotive practice. As rated power increases, the advantages of smaller cylinders in regard to size, weight, and improved engine balance and smoothness point toward increasing the number of

† MBT timing has traditionally been defined as the minimum spark advance for best torque. Since the torque first increases and then decreases as spark timing is advanced, the definition used here is more precise.

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cylinders per engine. An upper limit on cylinder size is dictated by dynamic considerations: the inertial forces that are created by accelerating and decelerating the reciprocating masses of the piston and connecting rod would quickly limit the maximum speed of the engine. Thus, the displaced volume is spread out amongst several smaller cylinders. The increased frequency of power strokes with a multicylinder engine produces much smoother torque characteristics. Multicylinder engines can also achieve a much better state of balance than single-cylinder engines. A force must be applied to the piston to accelerate it during the first half of its travel from bottom-center or top-center. The piston then exerts a force as it decelerates during the second part of the stroke. It is desirable to cancel these inertia forces through the choice of number and arrangement of cylinders to achieve a *primary balance*. Note, however, that the motion of the piston is more rapid during the upper half of its stroke than during the lower half (a consequence of the connecting rod and crank mechanism evident from Fig. 1-1; see also Sec. 2.2). The resulting inequality in piston acceleration and deceleration produces corresponding differences in inertia forces generated. Certain combinations of cylinder number and arrangement will balance out these secondary inertia force effects.

Four-cylinder in-line engines are the most common arrangements for automobile engines up to about 2.5-liter displacement. An example of this in-line arrangement was shown in Fig. 1-4. It is compact—an important consideration for small passenger cars. It provides two torque pulses per revolution of the crankshaft and primary inertia forces (though not secondary forces) are balanced. V engines and opposed-piston engines are occasionally used with this number of cylinders.

The V arrangement, with two banks of cylinders set at 90° or a more acute angle to each other, provides a compact block and is used extensively for larger displacement engines. Figure 1-9 shows a V-6 engine, the six cylinders being arranged in two banks of three with a 60° angle between their axis. Six cylinders are usually used in the 2.5- to 4.5-liter displacement range. Six-cylinder engines provide smoother operation with three torque pulses per revolution. The in-line arrangement results in a long engine, however, giving rise to crankshaft torsional vibration and making even distribution of air and fuel to each cylinder more difficult. The V-6 arrangement is much more compact, and the example shown provides primary balance of the reciprocating components. With the V engine, however, a rocking moment is imposed on the crankshaft due to the secondary inertia forces, which results in the engine being less well balanced than the in-line version. The V-8 and V-12 arrangements are also commonly used to provide compact, smooth, low-vibration, larger-displacement, spark-ignition engines.

Turbochargers are used to increase the maximum power that can be obtained from a given displacement engine. The work transfer to the piston per cycle, in each cylinder, which controls the power the engine can deliver, depends on the amount of fuel burned per cylinder per cycle. This depends on the amount of fresh air that is inducted each cycle. Increasing the air density prior to entry into the engine thus increases the maximum power that an engine of given dis-

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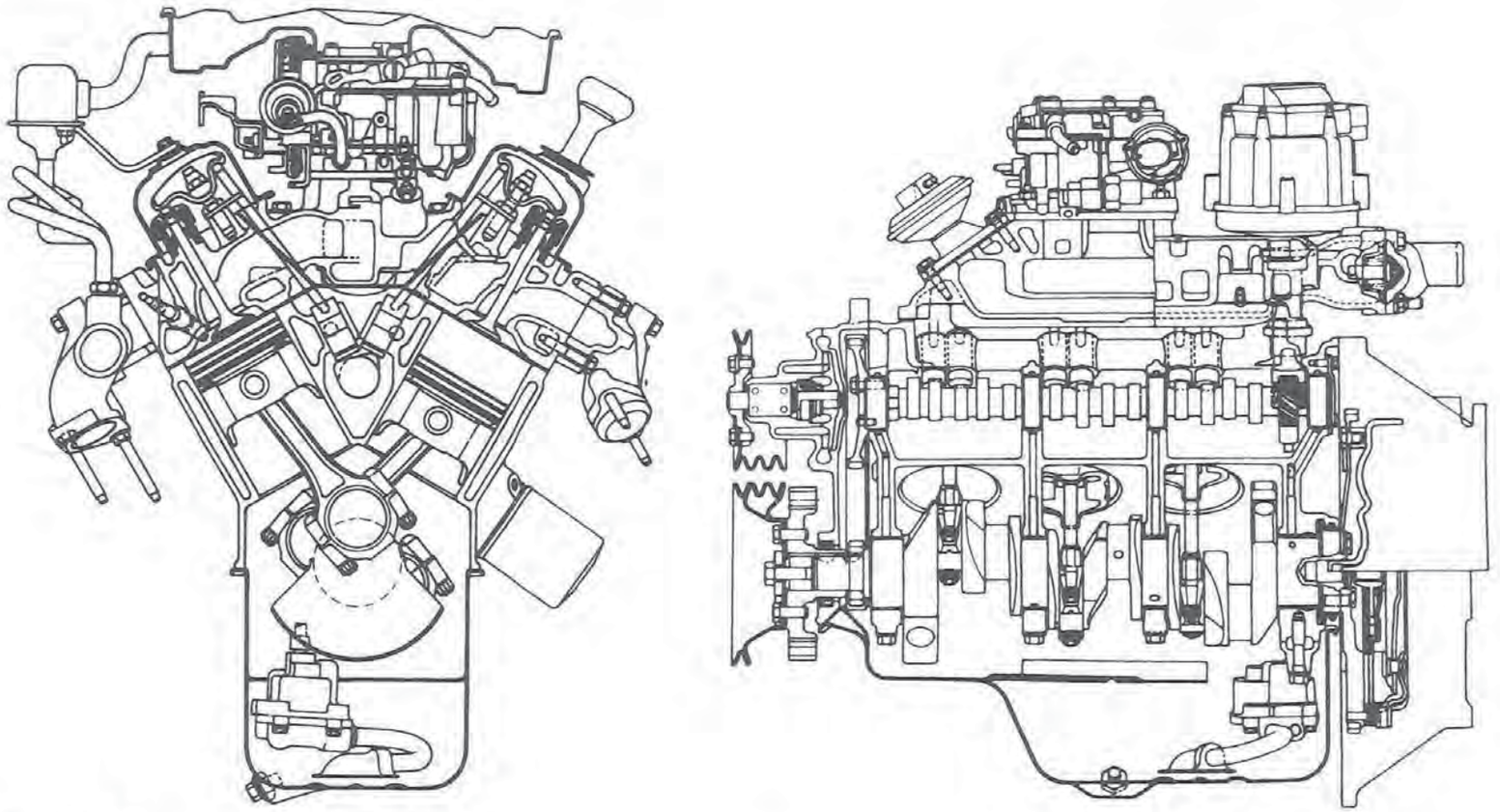


FIGURE 1-9
Cross-section drawings of General Motors 60 degree V-6 spark-ignition engine.[13] Displacement 2.8 liter, bore 89 mm, stroke 76 mm, compression ratio 8.5, maximum power 86 kW at 4800 rev/min.

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placement can deliver. Figure 1-10 shows an example of a turbocharged four-cylinder spark-ignition engine. The turbocharger, a compressor-turbine combination, uses the energy available in the engine exhaust stream to achieve compression of the intake flow. The air flow passes through the compressor (2), intercooler (3), carburetor (4), manifold (5), and inlet valve (6) as shown. Engine inlet pressures (or boost) of up to about 100 kPa above atmospheric pressure are typical. The exhaust flow through the valve (7) and manifold (8) drives the turbine (9) which powers the compressor. A wastegate (valve) just upstream of the turbine bypasses some of the exhaust gas flow when necessary to prevent the boost pressure becoming too high. The wastegate linkage (11) is controlled by a boost pressure regulator. While this turbocharged engine configuration has the carburetor downstream of the compressor, some turbocharged spark-ignition engines have the carburetor upstream of the compressor so that it operates at or below atmospheric pressure. Figure 1-11 shows a cutaway drawing of a small automotive turbocharger. The arrangements of the compressor and turbine

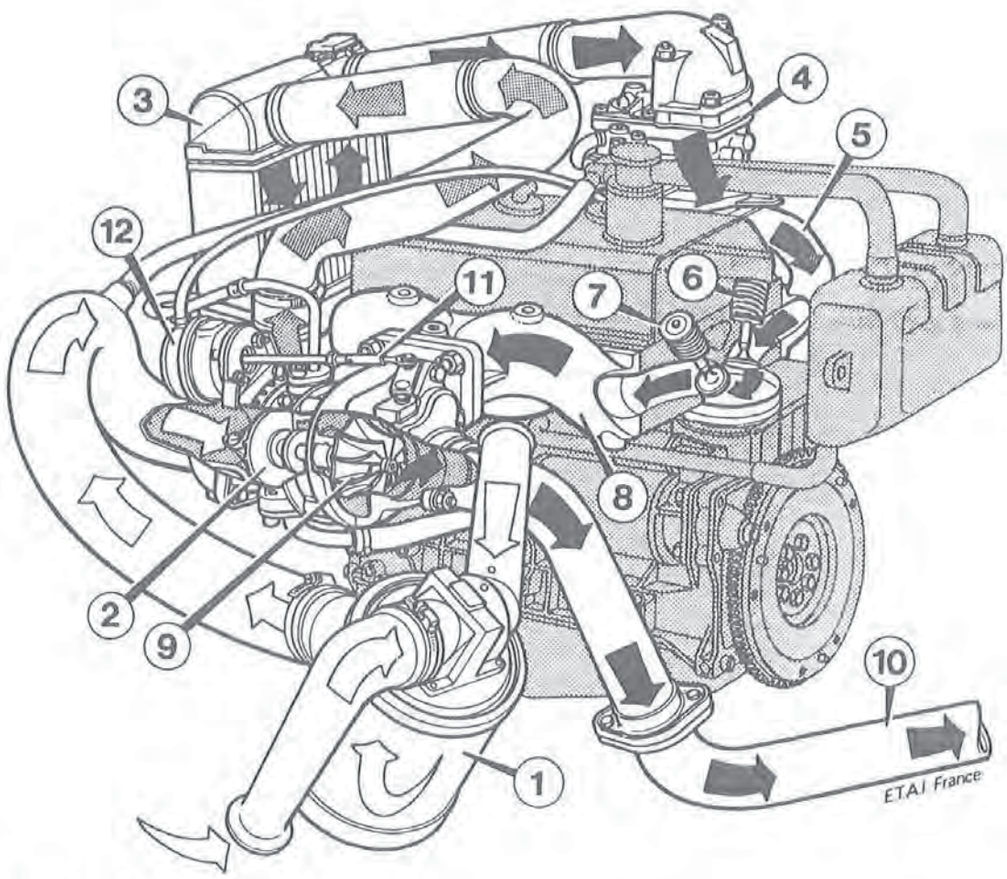


FIGURE 1-10
Turbocharged four-cylinder automotive spark-ignition engine. (*Courtesy Regie Nationale des Usines.*)

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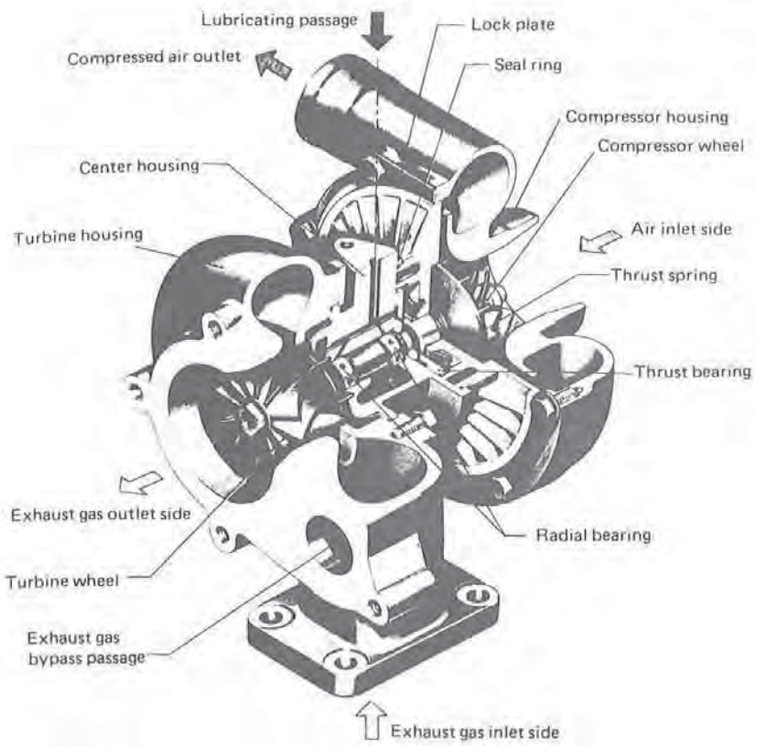


FIGURE 1-11
Cutaway view of small automotive engine turbocharger. (*Courtesy Nissan Motor Co., Ltd.*)

rotors connected via the central shaft and of the turbine and compressor flow passages are evident.

Figure 1-12 shows a two-stroke cycle spark-ignition engine. The two-stroke cycle spark-ignition engine is used for small-engine applications where low cost and weight/power ratio are important and when the use factor is low. Examples of such applications are outboard motorboat engines, motorcycles, and chain saws. All such engines are of the carburetor crankcase-compression type which is one of the simplest prime movers available. It has three moving parts per cylinder: the piston, connecting rod, and the crank. The prime advantage of the two-stroke cycle spark-ignition engine relative to the four-stroke cycle engine is its higher power per unit displaced volume due to twice the number of power strokes per crank revolution. This is offset by the lower fresh charge density achieved by the two-stroke cycle gas-exchange process and the loss of fresh mixture which goes straight through the engine during scavenging. Also, oil consumption is higher in two-stroke cycle engines due to the need to add oil to the fuel to lubricate the piston ring and piston surfaces.

The Wankel rotary engine is an alternative to the reciprocating engine geometry of the engines illustrated above. It is used when its compactness and higher engine speed (which result in high power/weight and power/volume ratios), and inherent balance and smoothness, offset its higher heat transfer, and

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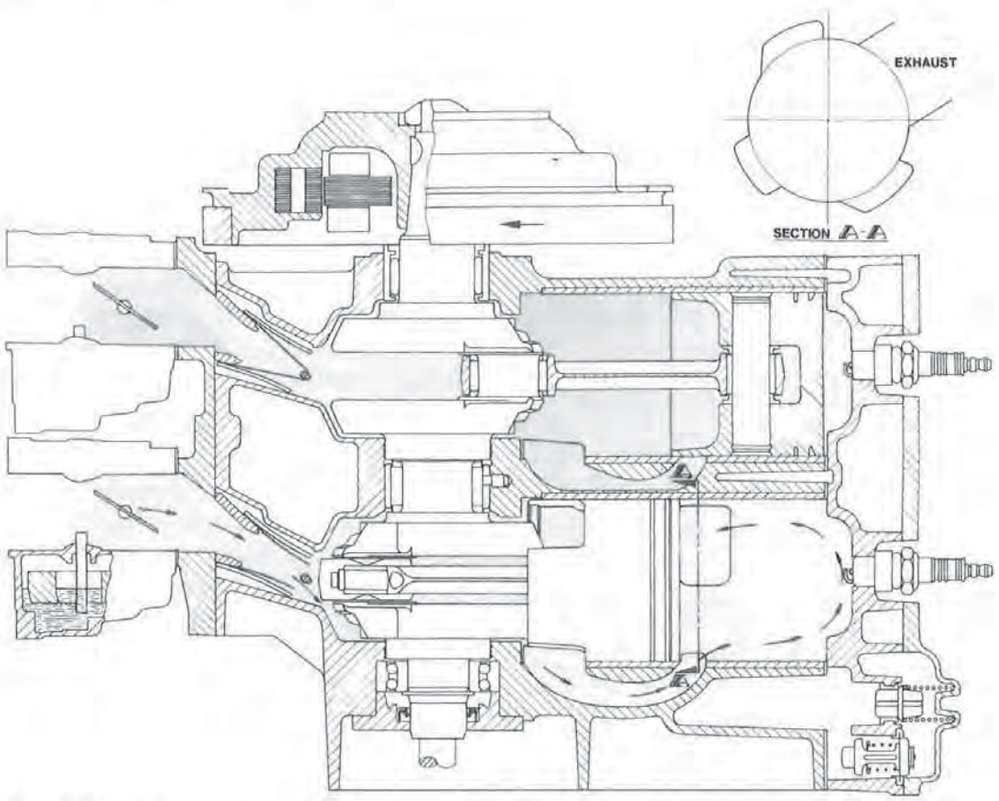


FIGURE 1-12
Cutaway drawing of two-cylinder two-stroke cycle loop-scavenged marine spark-ignition engine. Displaced volume 737 cm^3, maximum power 41 kW at 5500 rev/min. (*Courtesy Outboard Marine Corporation.*)

its sealing and leakage problems. Figure 1-13 shows the major mechanical parts of a simple single-rotor Wankel engine and illustrates its geometry. There are two rotating parts: the triangular-shaped rotor and the output shaft with its integral eccentric. The rotor revolves directly on the eccentric. The rotor has an internal timing gear which meshes with the fixed timing gear on one side housing to maintain the correct phase relationship between the rotor and eccentric shaft rotations. Thus the rotor rotates and orbits around the shaft axis. Breathing is through ports in the center housing (and sometimes the side housings). The combustion chamber lies between the center housing and rotor surface and is sealed by seals at the apex of the rotor and around the perimeters of the rotor sides. Figure 1-13 also shows how the Wankel rotary geometry operates with the four-stroke cycle. The figure shows the induction, compression, power, and exhaust processes of the four-stroke cycle for the chamber defined by rotor surface AB. The remaining two chambers defined by the other rotor surfaces undergo exactly the same sequence. As the rotor makes one complete rotation, during which the eccentric shaft rotates through three revolutions, each chamber produces one power "stroke." Three power pulses occur, therefore, for each rotor revolution;

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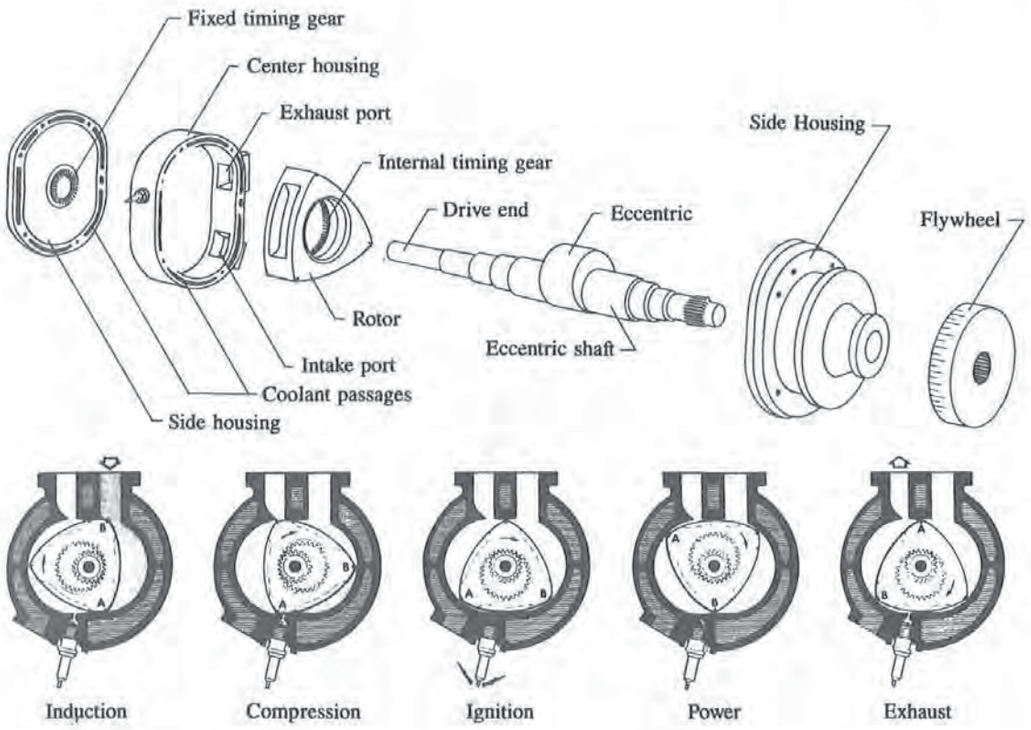


FIGURE 1-13
(*a*) Major components of the Wankel rotary engine; (*b*) induction, compression, power, and exhaust processes of the four-stroke cycle for the chamber defined by rotor surface AB. (*From Mobil Technical Bulletin*, Rotary Engines, © *Mobil Oil Corporation, 1971.*)

thus for each eccentric (output) shaft revolution there is one power pulse. Figure 1-14 shows a cutaway drawing of a two-rotor automobile Wankel engine. The two rotors are out of phase to provide a greater number of torque pulses per shaft revolution. Note the combustion chamber cut out in each rotor face, the rotor apex, and side seals. Two spark plugs per firing chamber are often used to obtain a faster combustion process.

1.7 COMPRESSION-IGNITION ENGINE OPERATION

In compression-ignition engines, air alone is inducted into the cylinder. The fuel (in most applications a light fuel oil, though heated residual fuel is used in marine and power-generation applications) is injected directly into the engine cylinder just before the combustion process is required to start. Load control is achieved by varying the amount of fuel injected each cycle; the air flow at a given engine speed is essentially unchanged. There are a great variety of CI engine designs in use in a wide range of applications—automobile, truck, locomotive, marine, power generation. Naturally aspirated engines where atmospheric air is inducted, turbocharged engines where the inlet air is compressed by an exhaust-driven

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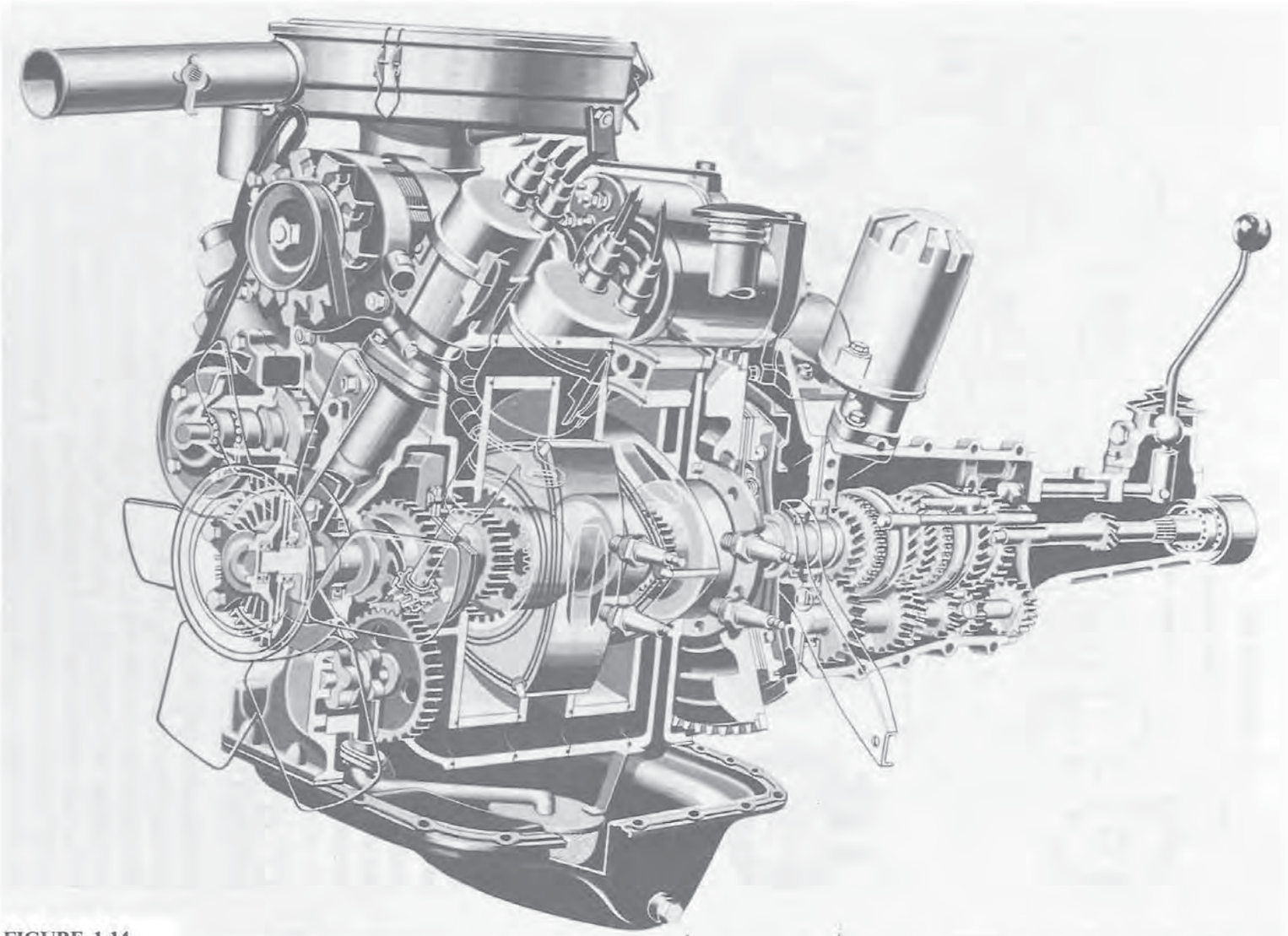


FIGURE 1-14
Cutaway drawing of two-rotor Wankel spark-ignition engine. Displacement of each working chamber 573 cm^3, compression ratio 9.4, maximum

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turbine-compressor combination, and supercharged engines where the air is compressed by a mechanically driven pump or blower are common. Turbocharging and supercharging increase engine output by increasing the air mass flow per unit displaced volume, thereby allowing an increase in fuel flow. These methods are used, usually in larger engines, to reduce engine size and weight for a given power output. Except in smaller engine sizes, the two-stroke cycle is competitive with the four-stroke cycle, in large part because, with the diesel cycle, only air is lost in the cylinder scavenging process.

The operation of a typical four-stroke naturally aspirated CI engine is illustrated in Fig. 1-15. The compression ratio of diesels is much higher than typical SI engine values, and is in the range 12 to 24, depending on the type of diesel engine and whether the engine is naturally aspirated or turbocharged. The valve timings used are similar to those of SI engines. Air at close-to-atmospheric pressure is inducted during the intake stroke and then compressed to a pressure of about 4 MPa (600 lb/in^2) and temperature of about 800 K (1000°F) during the compression stroke. At about 20° before TC, fuel injection into the engine cylinder commences; a typical rate of injection profile is shown in Fig. 1-15*b*. The liquid fuel jet atomizes into drops and entrains air. The liquid fuel evaporates; fuel vapor then mixes with air to within combustible proportions. The air temperature and pressure are above the fuel's ignition point. Therefore after a short *delay period*, spontaneous ignition (autoignition) of parts of the nonuniform fuel-air mixture initiates the combustion process, and the cylinder pressure (solid line in Fig. 1-15*c*) rises above the nonfiring engine level. The flame spreads rapidly through that portion of the injected fuel which has already mixed with sufficient air to burn. As the expansion process proceeds, mixing between fuel, air, and burning gases continues, accompanied by further combustion (see Fig. 1-15*d*). At full load, the mass of fuel injected is about 5 percent of the mass of air in the cylinder. Increasing levels of black smoke in the exhaust limit the amount of fuel that can be burned efficiently. The exhaust process is similar to that of the four-stroke SI engine. At the end of the exhaust stroke, the cycle starts again.

In the two-stroke CI engine cycle, compression, fuel injection, combustion, and expansion processes are similar to the equivalent four-stroke cycle processes; it is the intake and exhaust pressure which are different. The sequence of events in a loop-scavenged two-stroke engine is illustrated in Fig. 1-16. In loop-scavenged engines both exhaust and inlet ports are at the same end of the cylinder and are uncovered as the piston approaches BC (see Fig. 1-16*a*). After the exhaust ports open, the cylinder pressure falls rapidly in a blowdown process (Fig. 1-16*b*). The inlet ports then open, and once the cylinder pressure p falls below the inlet pressure p_i, air flows into the cylinder. The burned gases, displaced by this fresh air, continue to flow out of the exhaust port (along with some of the fresh air). Once the ports close as the piston starts the compression stroke, compression, fuel-injection, fuel-air mixing, combustion and expansion processes proceed as in the four-stroke CI engine cycle.

The diesel fuel-injection system consists of an injection pump, delivery pipes, and fuel injector nozzles. Several different types of injection pumps and

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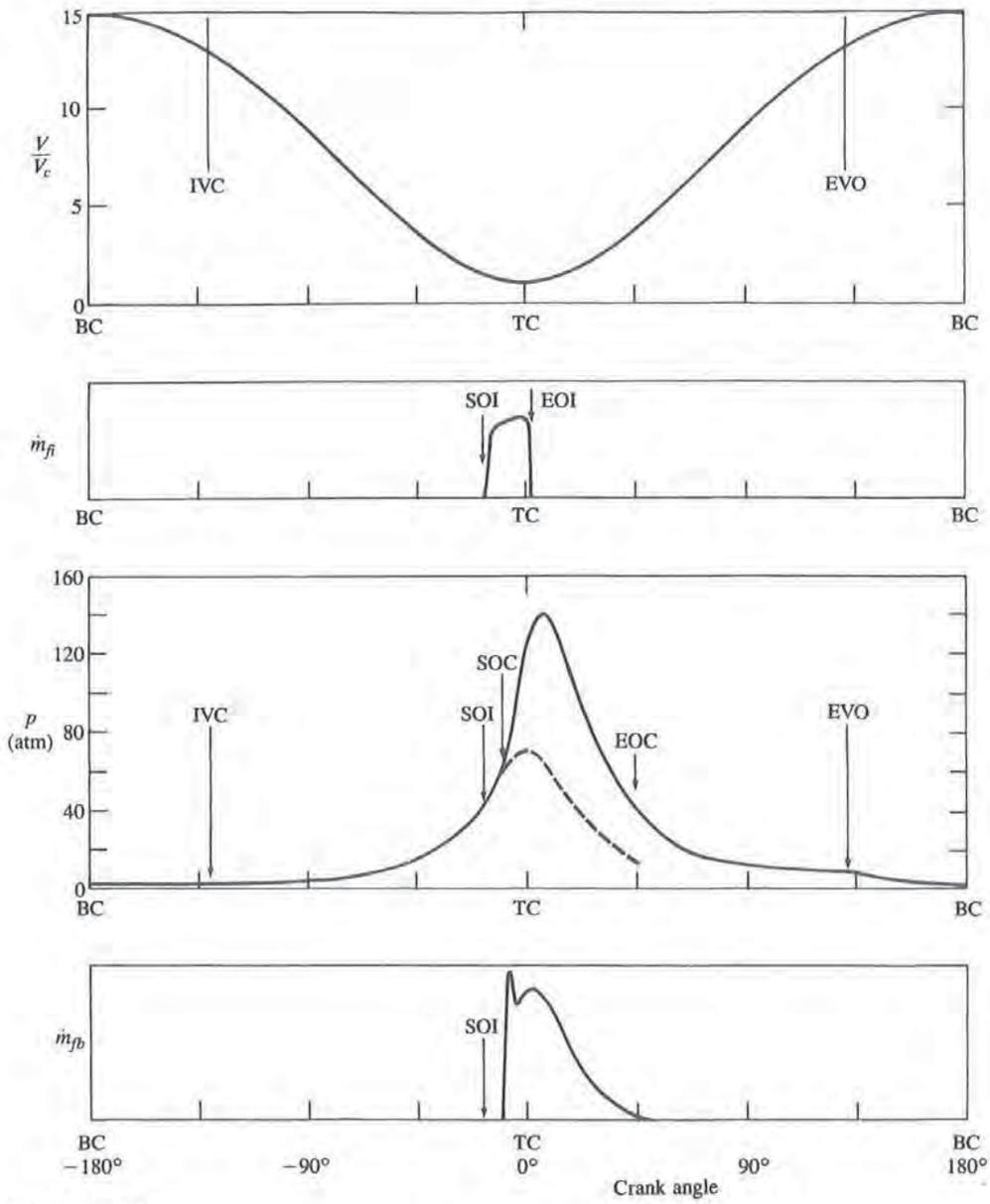


FIGURE 1-15
Sequence of events during compression, combustion, and expansion processes of a naturally aspirated compression-ignition engine operating cycle. Cylinder volume/clearance volume V/V_c, rate of fuel injection $\dot{m}_{fi}$, cylinder pressure p (solid line, firing cycle; dashed line, motored cycle), and rate of fuel burning (or fuel chemical energy release rate) $\dot{m}_{fb}$ are plotted against crank angle.

nozzles are used. In one common fuel pump (an in-line pump design shown in Fig. 1-17) a set of cam-driven plungers (one for each cylinder) operate in closely fitting barrels. Early in the stroke of the plunger, the inlet port is closed and the fuel trapped above the plunger is forced through a check valve into the injection

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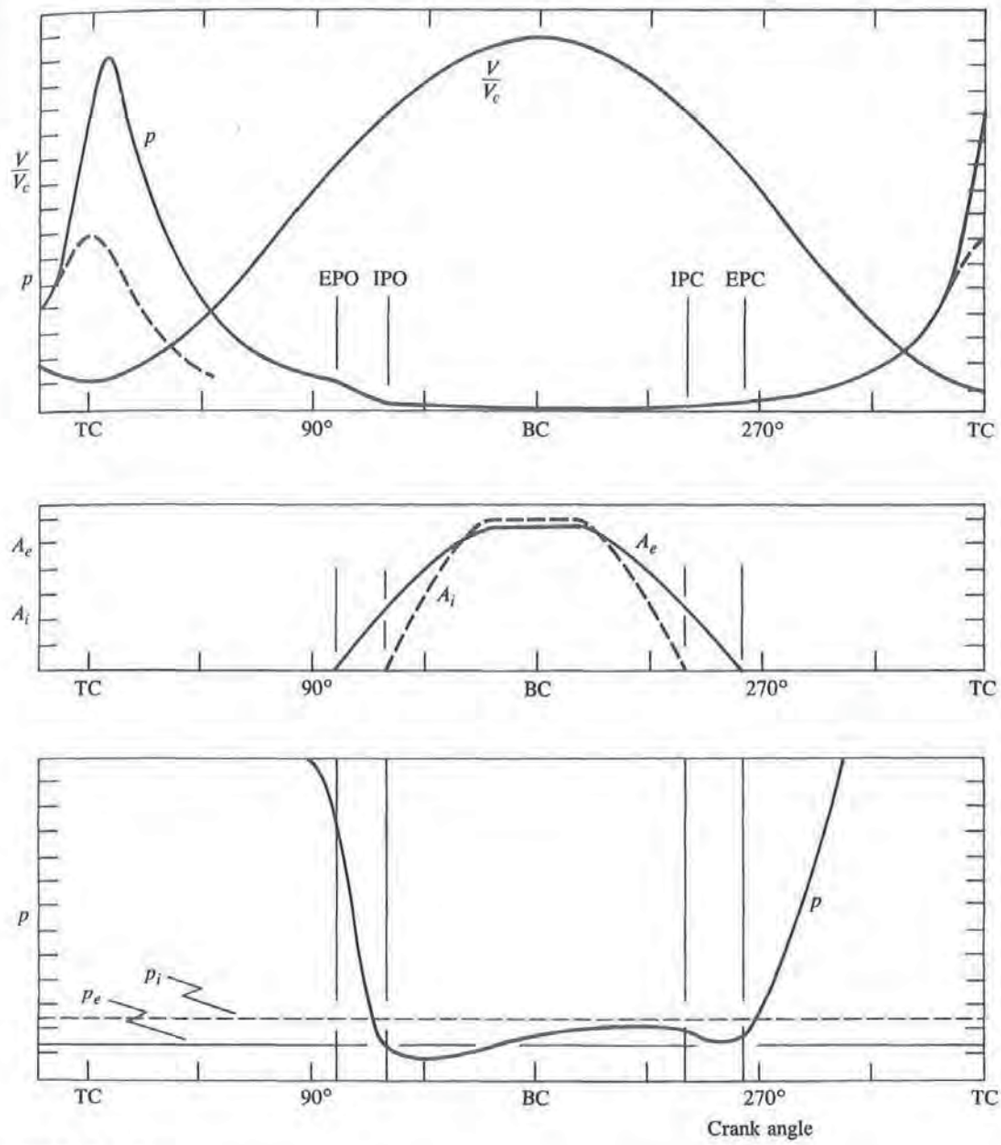


FIGURE 1-16
Sequence of events during expansion, gas exchange, and compression processes in a loop-scavenged two-stroke cycle compression-ignition engine. Cylinder volume/clearance volume V/V_c, cylinder pressure p, exhaust port open area A_e, and intake port open area A_i are plotted against crank angle.

line. The injection nozzle (Fig. 1-18) has one or more holes through which the fuel sprays into the cylinder. A spring-loaded valve closes these holes until the pressure in the injection line, acting on part of the valve surface, overcomes the spring force and opens the valve. Injection starts shortly after the line pressure begins to rise. Thus, the phase of the pump camshaft relative to the engine crankshaft controls the start of injection. Injection is stopped when the inlet port of the pump is uncovered by a helical groove in the pump plunger, because the high

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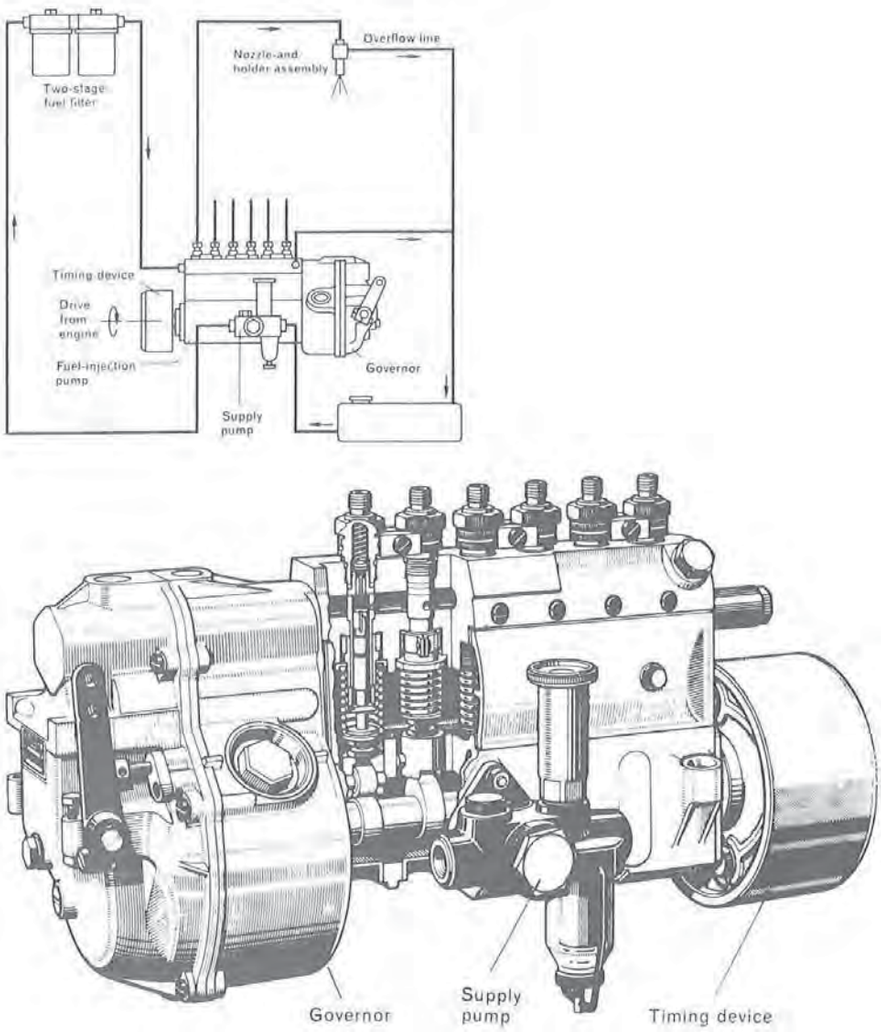
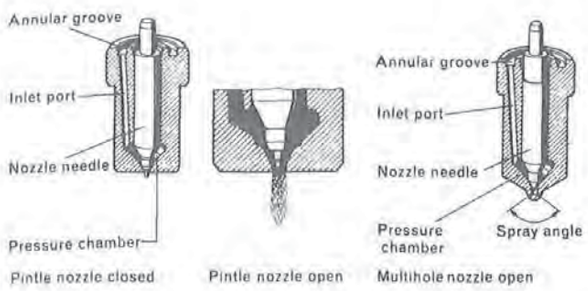


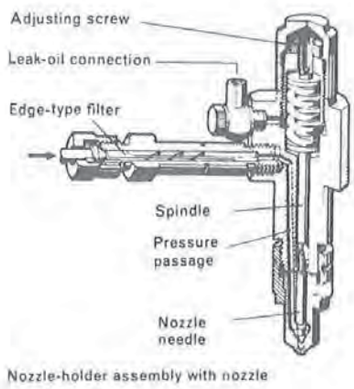
FIGURE 1-17
Diesel fuel system with in-line fuel-injection pump (type PE).[1,2] (*Courtesy Robert Bosch GmbH.*)

pressure above the plunger is then released (Fig. 1-18). The amount of fuel injected (which controls the load) is determined by the injection pump cam design and the position of the helical groove. Thus for a given cam design, rotating the plunger and its helical groove varies the load.

Distributor-type pumps have only one pump plunger and barrel, which meters and distributes the fuel to all the injection nozzles. A schematic of a distributor-type pump is shown in Fig. 1-19. The unit contains a low-pressure fuel pump (on left), a high-pressure injection pump (on right), an overspeed governor, and an injection timer. High pressure is generated by the plunger which is made to describe a combined rotary and stroke movement by the rotating eccentric disc or cam plate; the rotary motion distributes the fuel to the individual injection nozzles.

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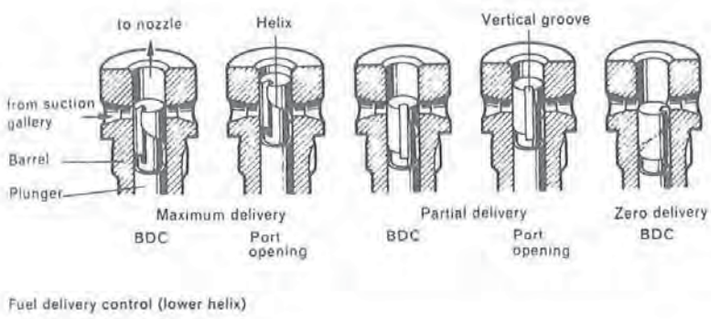


FIGURE 1-18
Details of fuel-injection nozzles, nozzle holder assembly and fuel-delivery control.[12] (*Courtesy Robert Bosch GmbH.*)

Distributor pumps can operate at higher speed and take up less space than in-line pumps. They are normally used on smaller diesel engines. In-line pumps are used in the mid-engine-size range. In the larger diesels, individual single-barrel injection pumps, close mounted to each cylinder with an external drive as shown in Fig. 1-5, are normally used.

1.8 EXAMPLES OF DIESEL ENGINES

A large number of diesel engine configurations and designs are in common use. The very large marine and stationary power-generating diesels are two-stroke

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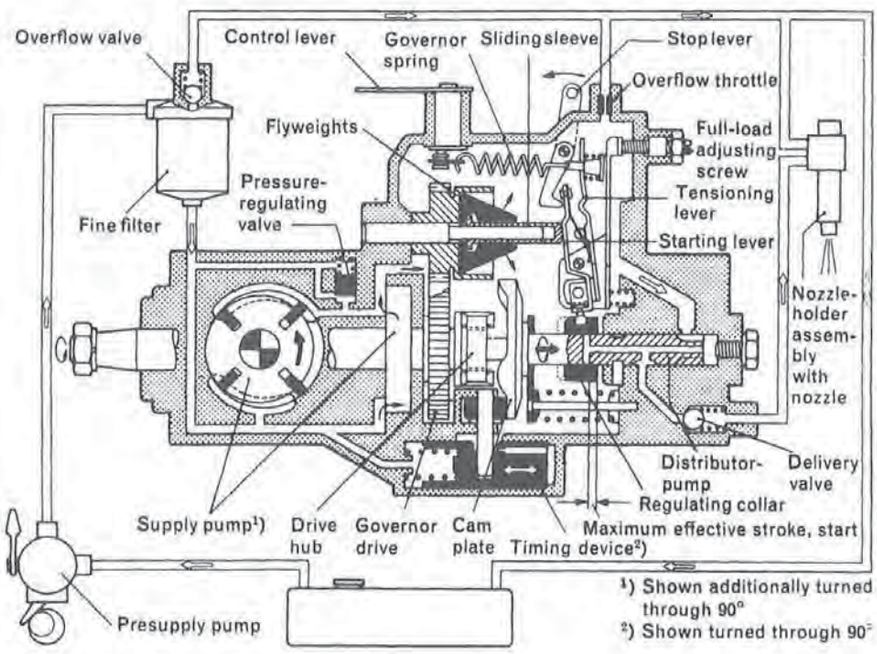


FIGURE 1-19
Diesel fuel system with distributor-type fuel-injection pump with mechanical governor.[1,2] (*Courtesy Robert Bosch GmbH.*)

cycle engines. Small- and medium-size engines use the four-stroke cycle. Because air capacity is an important constraint on the amount of fuel that can be burned in the diesel engine, and therefore on the engine's power, turbocharging is used extensively. All large engines are turbocharged. The majority of smaller diesels are not turbocharged, though they can be turbocharged and many are. The details of the engine design also vary significantly over the diesel size range. In particular, different combustion chamber geometries and fuel-injection characteristics are required to deal effectively with a major diesel engine design problem—achieving sufficiently rapid fuel-air mixing rates to complete the fuel-burning process in the time available. A wide variety of inlet port geometries, cylinder head and piston shapes, and fuel-injection patterns are used to accomplish this over the diesel size range.

Figure 1-20 shows a diesel engine typical of the medium-duty truck application. The design shown is a six-cylinder in-line engine. The drawing indicates that diesel engines are generally substantially heavier than spark-ignition engines because stress levels are higher due to the significantly higher pressure levels of the diesel cycle. The engine shown has a displacement of 10 liters, a compression ratio of 16.3, and is usually turbocharged. The engine has pressed-in cylinder liners to achieve better cylinder wear characteristics. This type of diesel is called a *direct-injection* diesel. The fuel is injected into a combustion chamber directly above the piston crown. The combustion chamber shown is a "bowl-in-piston" design, which puts most of the clearance volume into a compact shape. With this

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size of diesel engine, it is often necessary to use a swirling air flow rotating about the cylinder axis, which is created by suitable design of the inlet port and valve, to achieve adequate fuel-air mixing and fuel burning rates. The fuel injector, shown left-of-center in the drawing, has a multihole nozzle, typically with three to five holes. The fuel jets move out radially from the center of the piston bowl into the (swirling) air flow. The in-line fuel-injection pump is normally used with this type of diesel engine.

Figure 1-21 shows a four-cylinder in-line overhead-valve-cam design automobile diesel engine. The smallest diesels such as this operate at higher engine speed than larger engines; hence the time available for burning the fuel is less and the fuel-injection and combustion system must achieve faster fuel-air mixing rates. This is accomplished by using an *indirect-injection* type of diesel. Fuel is injected into an auxiliary combustion chamber which is separated from the main combustion chamber above the piston by a flow restriction or nozzle. During the latter stages of the compression process, air is forced through this nozzle from the

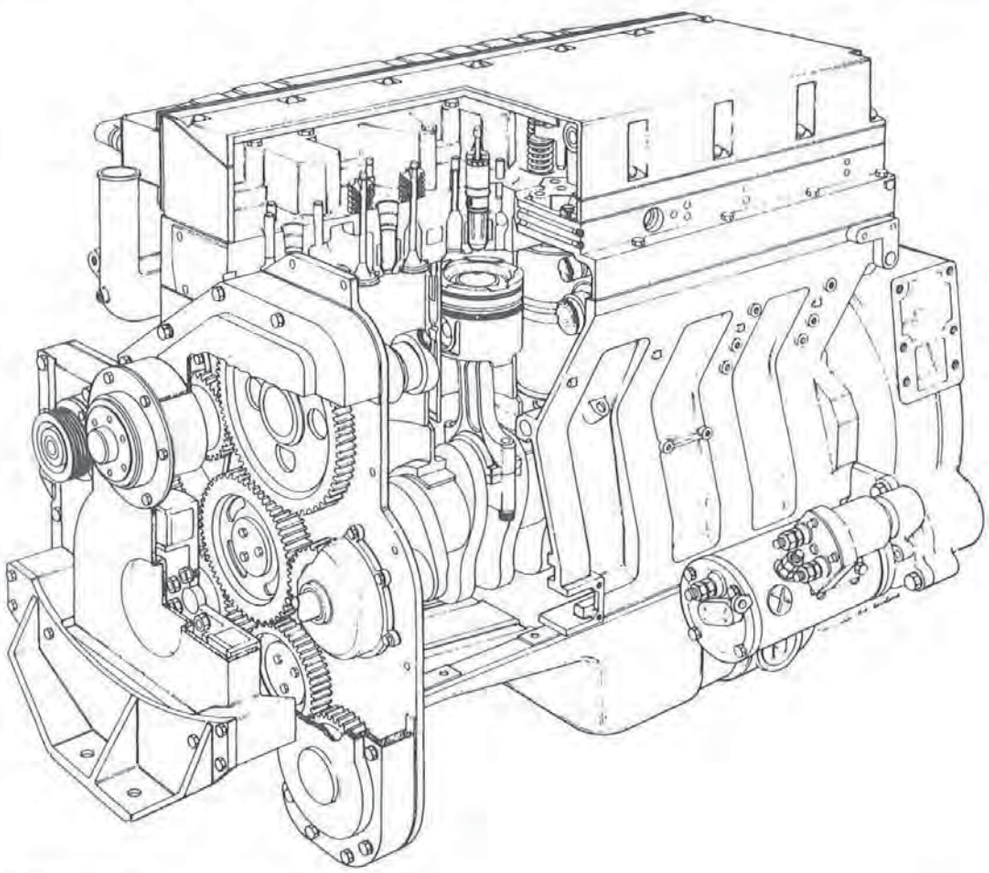


FIGURE 1-20
Direct-injection four-stroke cycle six-cylinder turbocharged Cummins diesel engine. Displaced volume 10 liters, bore 125 mm, stroke 136 mm, compression ratio 16.3, maximum power 168 to 246 kW at rated speed of 2100 rev/min. (*Courtesy Cummins Engine Company, Inc.*)

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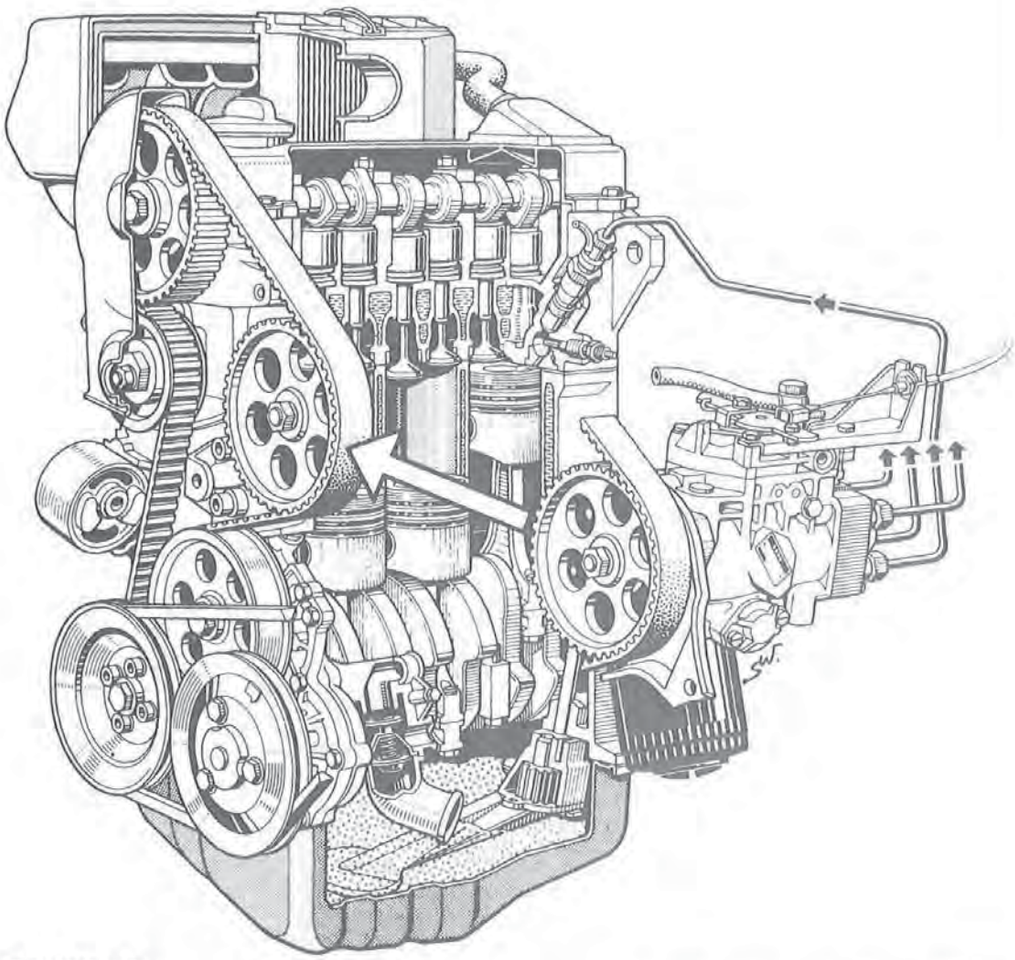


FIGURE 1-21
Four-cylinder naturally aspirated indirect-injection automobile Volkswagen diesel engine.[14] Displaced volume 1.47 liters, bore 76.5 mm, stroke 80 mm, maximum power 37 kW at 5000 rev/min.

cylinder into the prechamber at high velocity. Fuel is injected into this highly turbulent and often rapidly swirling flow in this auxiliary or prechamber, and very high mixing rates are achieved. Combustion starts in the prechamber, and the resulting pressure rise in the prechamber forces burning gases, fuel, and air into the main chamber. Since this outflow is also extremely vigorous, rapid mixing then occurs in the main chamber as the burning jet mixes with the remaining air and combustion is completed. A distributor-type fuel pump, which is normally used in this engine size range, driven off the camshaft at half the crankshaft speed by a toothed belt, is shown on the right of the figure. It supplies high-pressure fuel pulses to the pintle-type injector nozzles in turn. A glow plug is also shown in the auxiliary chamber; this plug is electrically heated prior to and during cold engine start-up to raise the temperature of the air charge and the fuel sufficiently to achieve autoignition. The compression ratio of this engine is 23. Indirect-injection diesel engines require higher compression ratios than direct-injection engines to start adequately when cold.

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Diesel engines are turbocharged to achieve higher power/weight ratios. By increasing the density of the inlet air, a given displaced volume can induct more air. Hence more fuel can be injected and burned, and more power delivered, while avoiding excessive black smoke in the exhaust. All the larger diesels are turbocharged; smaller diesels can be and often are. Figure 1-22 shows how a turbocharger connects to a direct-injection diesel.

All the above diesels are water cooled; some production diesels are air cooled. Figure 1-23 shows a V-8 air-cooled direct-injection naturally aspirated

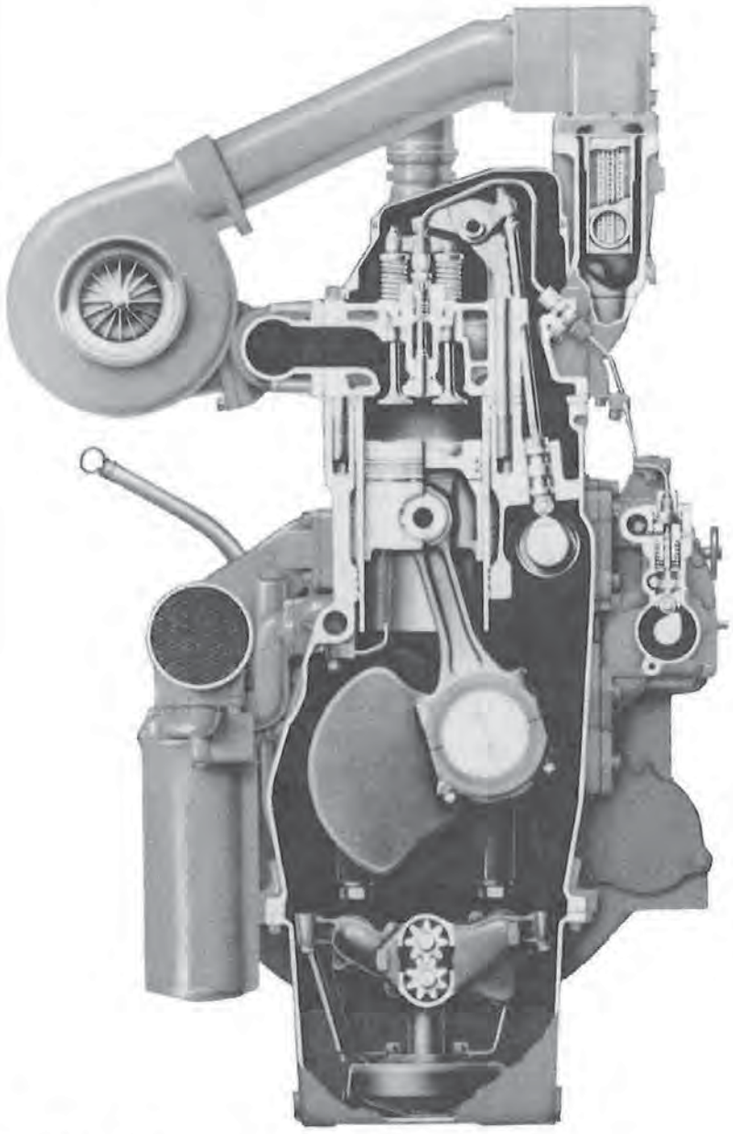


FIGURE 1-22
Turbocharged aftercooled direct-injection four-stroke cycle Caterpillar six-cylinder in-line heavy-duty truck diesel engine. Bore 137.2 mm, stroke 165.1 mm, rated power 200–300 kW and rated speed of 1600–2100 rev/min depending on application. (*Courtesy Caterpillar Tractor Company.*)

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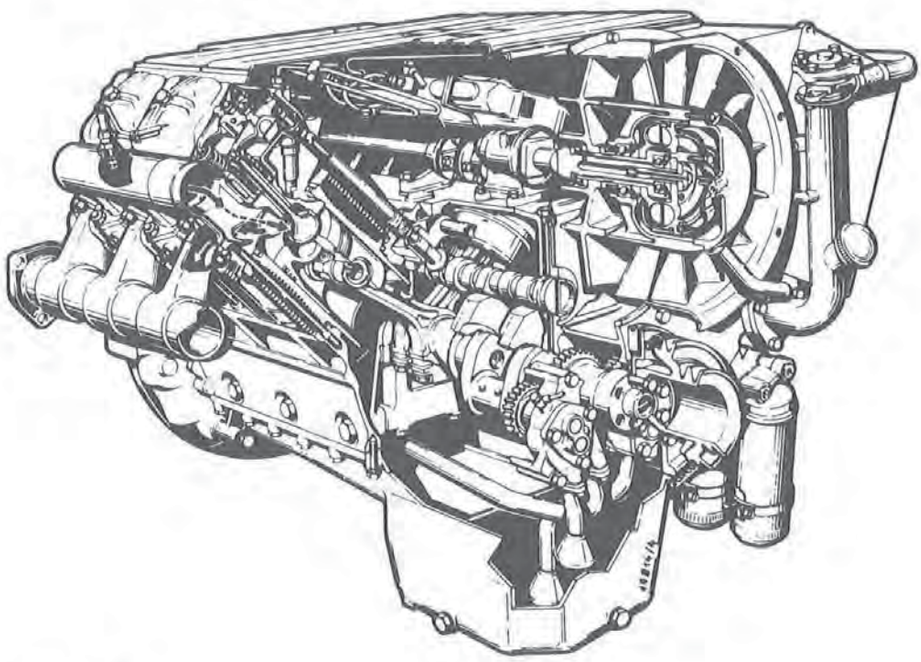


FIGURE 1-23
V-8 air-cooled direct-injection naturally aspirated diesel engine. Displacement 13.4 liter, bore 128 mm, stroke 130 mm, compression ratio 17, maximum rated power 188 kW at rated speed of 2300 rev/min. (*Courtesy Klöcker-Humboldt-Deutz AG.*[15])

diesel. The primary advantage compared to the water-cooled engines is lower engine weight. The fins on the cylinder block and head are necessary to increase the external heat-transfer surface area to achieve the required heat rejection. An air blower, shown on the right of the cutaway drawing, provides forced air convection over the block. The blower is driven off the injection pump shaft, which in turn is driven off the camshaft. The in-line injection pump is placed between the two banks of cylinders. The injection nozzles are located at an angle to the cylinder axis. The combustion chamber and fuel-injection characteristics are similar to those of the engine in Fig. 1-22. The nozzle shown injects four fuel sprays into a reentrant bowl-in-piston combustion chamber.

Diesels are also made in very large engine sizes. These large engines are used for marine propulsion and electrical power generation and operate on the two-stroke cycle in contrast to the small- and medium-size diesels illustrated above. Figure 1-24 shows such a two-stroke cycle marine engine, available with from 4 to 12 cylinders, with a maximum bore of 0.6–0.9 m and stroke of 2–3 m, which operates at speeds of about 100 rev/min. These engines are normally of the crosshead type to reduce side forces on the cylinder. The gas exchange between cycles is controlled by first opening the exhaust valves, and then the piston uncovering inlet ports in the cylinder liner. Expanding exhaust gases leave the cylinder via the exhaust valves and manifold and pass through the turbocharger

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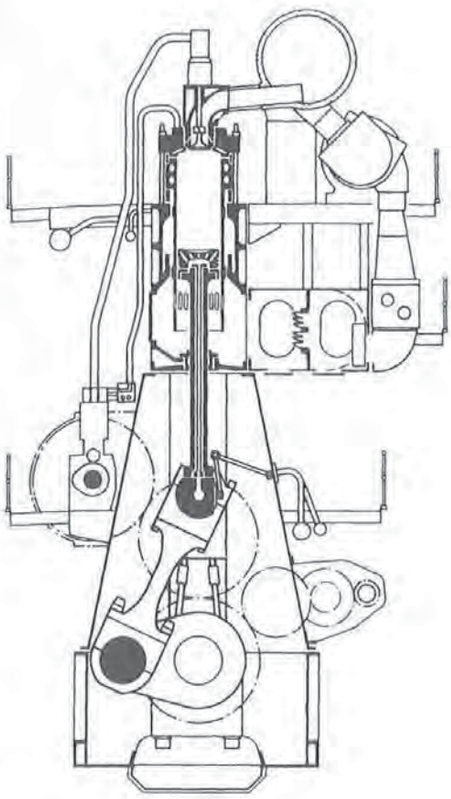


FIGURE 1-24
Large Sulzer two-stroke turbocharged marine diesel engine. Bore 840 mm, stroke 2900 mm, rated power 1.9 MW per cylinder at 78 rev/min, 4 to 12 cylinders. (*Courtesy Sulzer Brothers Ltd.*)

turbine. Compressed air enters via the inlet ports and induces forced scavenging; air is supplied from the turbocharger and cooler. At part load electrically driven blowers cut in to compress the scavenge air. Because these large engines operate at low speed, the motion induced by the centrally injected fuel jets is sufficient to mix the fuel with air and burn it in the time available. A simple open combustion chamber shape can be used, therefore, which achieves efficient combustion even with the low-quality heavy fuels used with these types of engines. The pistons are water cooled in these very large engines. The splash oil piston cooling used in medium- and small-size diesels is not adequate.

1.9 STRATIFIED-CHARGE ENGINES

Since the 1920s, attempts have been made to develop a hybrid internal combustion engine that combines the best features of the spark-ignition engine and the diesel. The goals have been to operate such an engine at close to the optimum compression ratio for efficiency (in the 12 to 15 range) by: (1) injecting the fuel directly into the combustion chamber during the compression process (and thereby avoid the knock or spontaneous ignition problem that limits conventional spark-ignition engines with their premixed charge); (2) igniting the fuel as it mixes with air with a spark plug to provide direct control of the ignition process

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(and thereby avoid the fuel ignition-quality requirement of the diesel); (3) controlling the engine power level by varying the amount of fuel injected per cycle (with the air flow unthrottled to minimize work done pumping the fresh charge into the cylinder). Such engines are often called *stratified-charge engines* from the need to produce in the mixing process between the fuel jet and the air in the cylinder a "stratified" fuel-air mixture, with an easily ignitable composition at the spark plug at the time of ignition. Because such engines avoid the spark-ignition engine requirement for fuels with a high antiknock quality and the diesel requirement for fuels with high ignition quality, they are usually fuel-tolerant and will operate with a wide range of liquid fuels.

Many different types of stratified-charge engine have been proposed, and some have been partially or fully developed. A few have even been used in practice in automotive applications. The operating principles of those that are truly fuel-tolerant or multifuel engines are illustrated in Fig. 1-25. The combustion chamber is usually a bowl-in-piston design, and a high degree of air swirl is created during intake and enhanced in the piston bowl during compression to achieve rapid fuel-air mixing. Fuel is injected into the cylinder, tangentially into the bowl, during the latter stages of compression. A long-duration spark discharge ignites the developing fuel-air jet as it passes the spark plug. The flame spreads downstream, and envelopes and consumes the fuel-air mixture. Mixing continues, and the final stages of combustion are completed during expansion. Most successful designs of this type of engine have used the four-stroke cycle. This concept is usually called a *direct-injection stratified-charge engine*. The engine can be turbocharged to increase its power density.

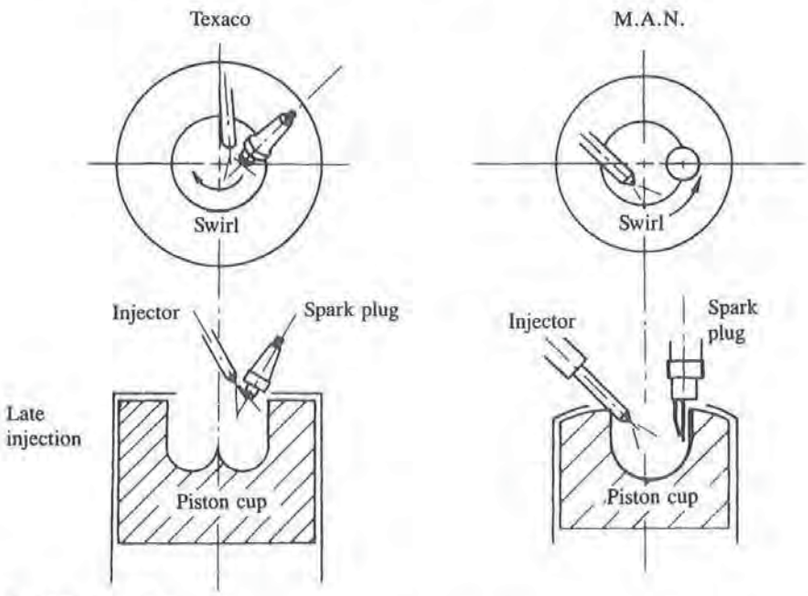


FIGURE 1-25
Two multifuel stratified-charge engines which have been used in commercial practice: the Texaco Controlled Combustion System (TCCS)[16] and the M.A.N.-FM System.[17]

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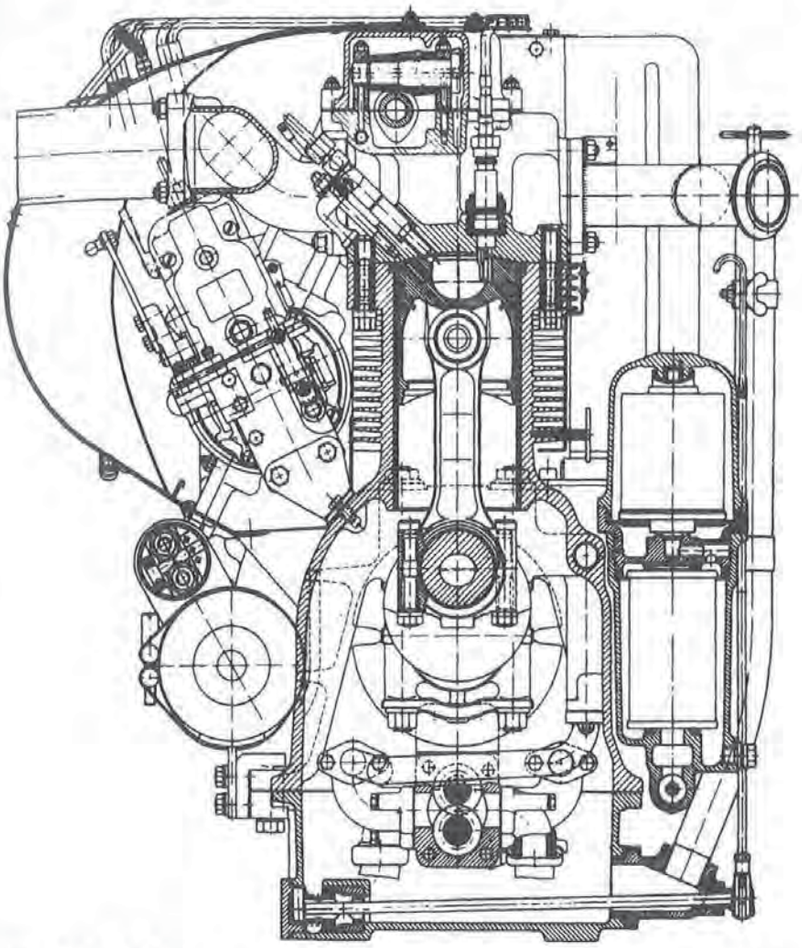


FIGURE 1-26
Sectional drawing of M.A.N. high-speed multifuel four-cylinder direct-injection stratified-charge engine. Bore 94.5 mm, stroke 100 mm, displacement 2.65 liters, compression ratio 16.5, rated power 52 kW at 3800 rev/min.[17]

A commercial multifuel engine is shown in Fig. 1-26. In this particular design, the fuel injector comes diagonally through the cylinder head from the upper left and injects the fuel onto the hot wall of the deep spherical piston bowl. The fuel is carried around the wall of the bowl by the swirling flow, evaporated off the wall, mixed with air, and then ignited by the discharge at the spark plug which enters the chamber vertically on the right. This particular engine is air cooled, so the cylinder block and head are finned to increase surface area.

An alternative stratified-charge engine concept, which has also been mass produced, uses a small *prechamber* fed during intake with an auxiliary fuel system to obtain an easily ignitable mixture around the spark plug. This concept, first proposed by Ricardo in the 1920s and extensively developed in the Soviet Union and Japan, is often called a *jet-ignition* or *torch-ignition* stratified-charge engine. Its operating principles are illustrated in Fig. 1-27 which shows a three-valve

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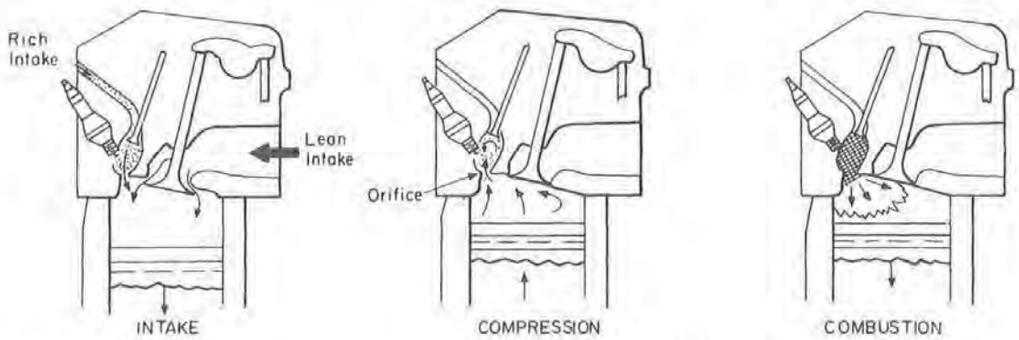


FIGURE 1-27
Schematic of three-valve torch-ignition stratified-charge spark-ignition engine.

carbureted version of the concept.[18] A separate carburetor and intake manifold feeds a fuel-rich mixture (which contains fuel beyond the amount that can be burned with the available air) through a separate small intake valve into the prechamber which contains the spark plug. At the same time, a very lean mixture (which contains excess air beyond that required to burn the fuel completely) is fed to the main combustion chamber through the main carburetor and intake manifold. During intake the rich prechamber flow fully purges the prechamber volume. After intake valve closing, lean mixture from the main chamber is compressed into the prechamber bringing the mixture at the spark plug to an easily ignitable, slightly rich, composition. After combustion starts in the prechamber, rich burning mixture issues as a jet through the orifice into the main chamber, entraining and igniting the lean main chamber charge. Though called a stratified-charge engine, this engine is really a jet-ignition concept whose primary function is to extend the operating limit of conventionally ignited spark-ignition engines to mixtures leaner than could normally be burned.

PROBLEMS

1.1. Describe the major functions of the following reciprocating engine components: piston, connecting rod, crankshaft, cams and camshaft, valves, intake and exhaust manifolds.

1.2. Indicate on an appropriate sketch the different forces that act on the piston, and the direction of these forces, during the engine's expansion stroke with the piston, connecting rod, and crank in the positions shown in Fig. 1-1.

1.3. List five important differences between the design and operating characteristics of spark-ignition and compression-ignition (diesel) engines.

1.4. Indicate the approximate crank angle at which the following events in the four-stroke and two-stroke internal combustion engine cycles occur on a line representing the full cycle (720° for the four-stroke cycle; 360° for the two-stroke cycle): bottom- and top-center crank positions, inlet and exhaust valve or port opening and closing, start of combustion process, end of combustion process, maximum cylinder pressure.

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1.5. The two-stroke cycle has twice as many power strokes per crank revolution as the four-stroke cycle. However, two-stroke cycle engine power outputs per unit displaced volume are less than twice the power output of an equivalent four-stroke cycle engine at the same engine speed. Suggest reasons why this potential advantage of the two-stroke cycle is offset in practice.

1.6. Suggest reasons why multicylinder engines prove more attractive than single-cylinder engines once the total engine displaced volume exceeds a few hundred cubic centimeters.

1.7. The Wankel rotary spark-ignition engine, while lighter and more compact than a reciprocating spark-ignition engine of equal maximum power, typically has worse efficiency due to significantly higher gas leakage from the combustion chamber and higher total heat loss from the hot combustion gases to the chamber walls. Based on the design details in Figs. 1-4, 1-13, and 1-14 suggest reasons for these higher losses.

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CHAPTER 2

ENGINE DESIGN AND OPERATING PARAMETERS

2.1 IMPORTANT ENGINE CHARACTERISTICS

In this chapter, some basic geometrical relationships and the parameters commonly used to characterize engine operation are developed. The factors important to an engine user are:

1. The engine's performance over its operating range
2. The engine's fuel consumption within this operating range and the cost of the required fuel
3. The engine's noise and air pollutant emissions within this operating range
4. The initial cost of the engine and its installation
5. The reliability and durability of the engine, its maintenance requirements, and how these affect engine availability and operating costs

These factors control total engine operating costs—usually the primary consideration of the user—and whether the engine in operation can satisfy environmental regulations. This book is concerned primarily with the performance, efficiency, and emissions characteristics of engines; the omission of the other factors listed above does not, in any way, reduce their great importance.

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Engine performance is more precisely defined by:

1. The maximum power (or the maximum torque) available at each speed within the useful engine operating range
2. The range of speed and power over which engine operation is satisfactory

The following performance definitions are commonly used:

Maximum rated power. The highest power an engine is allowed to develop for short periods of operation.

Normal rated power. The highest power an engine is allowed to develop in continuous operation.

Rated speed. The crankshaft rotational speed at which rated power is developed.

2.2 GEOMETRICAL PROPERTIES OF RECIPROCATING ENGINES

The following parameters define the basic geometry of a reciprocating engine (see Fig. 2-1):

Compression ratio r_c:

$$r_c = \frac{\text{maximum cylinder volume}}{\text{minimum cylinder volume}} = \frac{V_d + V_c}{V_c} \tag{2.1}$$

where V_d is the displaced or swept volume and V_c is the clearance volume.
Ratio of cylinder bore to piston stroke:

$$R_{bs} = \frac{B}{L} \tag{2.2}$$

Ratio of connecting rod length to crank radius:

$$R = \frac{l}{a} \tag{2.3}$$

In addition, the stroke and crank radius are related by

$$L = 2a$$

Typical values of these parameters are: $r_c = 8$ to 12 for SI engines and $r_c = 12$ to 24 for CI engines; $B/L = 0.8$ to 1.2 for small- and medium-size engines, decreasing to about 0.5 for large slow-speed CI engines; $R = 3$ to 4 for small- and medium-size engines, increasing to 5 to 9 for large slow-speed CI engines.

The cylinder volume V at any crank position θ is

$$V = V_c + \frac{\pi B^2}{4}(l + a - s) \tag{2.4}$$

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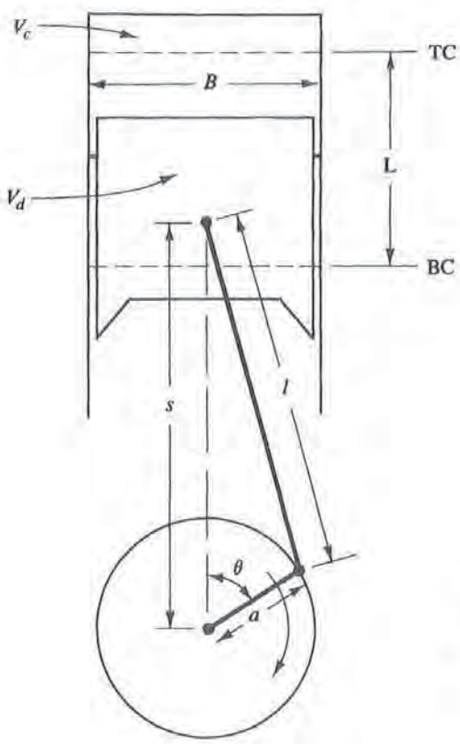


FIGURE 2-1
Geometry of cylinder, piston, connecting rod, and crankshaft where B = bore, L = stroke, l = connecting road length, a = crank radius, θ = crank angle.

where s is the distance between the crank axis and the piston pin axis (Fig. 2-1), and is given by

$$s = a \cos \theta + (l^2 - a^2 \sin^2 \theta)^{1/2} \tag{2.5}$$

The angle θ, defined as shown in Fig. 2-1, is called the *crank angle*. Equation (2.4) with the above definitions can be rearranged:

$$\frac{V}{V_c} = 1 + \tfrac{1}{2}(r_c - 1)[R + 1 - \cos \theta - (R^2 - \sin^2 \theta)^{1/2}] \tag{2.6}$$

The combustion chamber surface area A at any crank position θ is given by

$$A = A_{ch} + A_p + \pi B(l + a - s) \tag{2.7}$$

where A_{ch} is the cylinder head surface area and A_p is the piston crown surface area. For flat-topped pistons, $A_p = \pi B^2/4$. Using Eq. (2.5), Eq. (2-7) can be rearranged:

$$A = A_{ch} + A_p + \frac{\pi BL}{2}[R + 1 - \cos \theta - (R^2 - \sin^2 \theta)^{1/2}] \tag{2.8}$$

An important characteristic speed is the *mean piston speed* $\bar{S}_p$:

$$\bar{S}_p = 2LN \tag{2.9}$$

where N is the rotational speed of the crankshaft. Mean piston speed is often a

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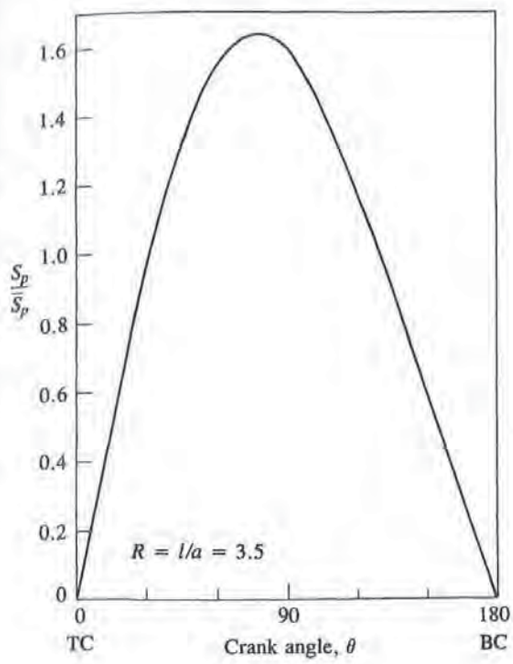


FIGURE 2-2
Instantaneous piston speed/mean piston speed as a function of crank angle for $R = 3.5$.

more appropriate parameter than crank rotational speed for correlating engine behavior as a function of speed. For example, gas-flow velocities in the intake and the cylinder all scale with $\bar{S}_p$. The *instantaneous* piston velocity S_p is obtained from

$$S_p = \frac{ds}{dt} \tag{2.10}$$

The piston velocity is zero at the beginning of the stroke, reaches a maximum near the middle of the stroke, and decreases to zero at the end of the stroke. Differentiation of Eq. (2.5) and substitution gives

$$\frac{S_p}{\bar{S}_p} = \frac{\pi}{2} \sin \theta \left[1 + \frac{\cos \theta}{(R^2 - \sin^2 \theta)^{1/2}} \right] \tag{2.11}$$

Figure 2-2 shows how S_p varies over each stroke for $R = 3.5$.

Resistance to gas flow into the engine or stresses due to the inertia of the moving parts limit the maximum mean piston speed to within the range 8 to 15 m/s (1500 to 3000 ft/min). Automobile engines operate at the higher end of this range; the lower end is typical of large marine diesel engines.

2.3 BRAKE TORQUE AND POWER

Engine torque is normally measured with a dynamometer.[1] The engine is clamped on a test bed and the shaft is connected to the dynamometer rotor. Figure 2-3 illustrates the operating principle of a dynamometer. The rotor is

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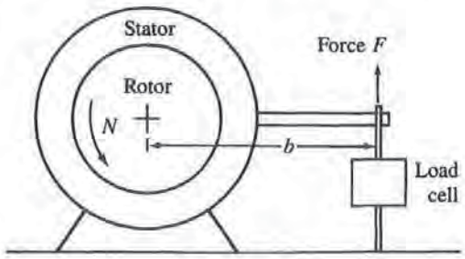


FIGURE 2-3
Schematic of principle of operation of dynamometer.

coupled electromagnetically, hydraulically, or by mechanical friction to a stator, which is supported in low friction bearings. The stator is balanced with the rotor stationary. The torque exerted on the stator with the rotor turning is measured by balancing the stator with weights, springs, or pneumatic means.

Using the notation in Fig. 2-3, if the torque exerted by the engine is T:

$$T = Fb \tag{2.12}$$

The power P delivered by the engine and absorbed by the dynamometer is the product of torque and angular speed:

$$P = 2\pi NT \tag{2.13a}$$

where N is the crankshaft rotational speed. In SI units:

$$P(\text{kW}) = 2\pi N(\text{rev/s})T(\text{N}\cdot\text{m}) \times 10^{-3} \tag{2.13b}$$

or in U.S. units:

$$P(\text{hp}) = \frac{N(\text{rev/min})\ T(\text{lbf}\cdot\text{ft})}{5252} \tag{2.13c}$$

Note that torque is a measure of an engine's ability to do work; power is the rate at which work is done.

The value of engine power measured as described above is called *brake power* P_b. This power is the usable power delivered by the engine to the load—in this case, a "brake."

2.4 INDICATED WORK PER CYCLE

Pressure data for the gas in the cylinder over the operating cycle of the engine can be used to calculate the work transfer from the gas to the piston. The cylinder pressure and corresponding cylinder volume throughout the engine cycle can be plotted on a p-V diagram as shown in Fig. 2-4. The *indicated work per cycle* $W_{c,i}$† (per cylinder) is obtained by integrating around the curve to obtain the

† The term indicated is used because such p-V diagrams used to be generated directly with a device called an engine indicator.

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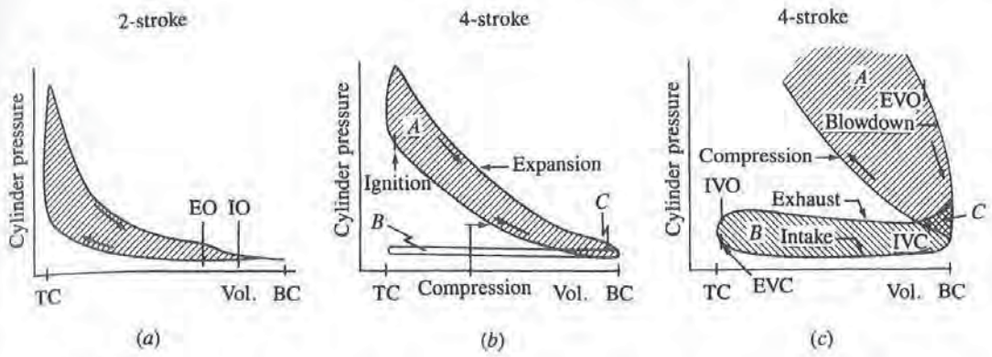


FIGURE 2-4
Examples of p-V diagrams for (*a*) a two-stroke cycle engine, (*b*) a four-stroke cycle engine; (*c*) a four-stroke cycle spark-ignition engine exhaust and intake strokes (pumping loop) at part load.

area enclosed on the diagram:

$$W_{c,i} = \oint p\, dV \tag{2.14}$$

With two-stroke cycles (Fig. 2-4*a*), the application of Eq. (2.14) is straightforward. With the addition of inlet and exhaust strokes for the four-stroke cycle, some ambiguity is introduced as two definitions of indicated output are in common use. These will be defined as:

Gross indicated work per cycle $W_{c,ig}$. Work delivered to the piston over the compression and expansion strokes only.

Net indicated work per cycle $W_{c,in}$. Work delivered to the piston over the entire four-stroke cycle.

In Fig. 2-4*b* and *c*, $W_{c,ig}$ is (area A + area C) and $W_{c,in}$ is (area A + area C) − (area B + area C), which equals (area A − area B), where each of these areas is regarded as a positive quantity. Area B + area C is the work transfer between the piston and the cylinder gases during the inlet and exhaust strokes and is called the *pumping work* W_p (see Chaps. 5 and 13). The pumping work transfer will be *to* the cylinder gases if the pressure during the intake stroke is less than the pressure during the exhaust stroke. This is the situation with naturally aspirated engines. The pumping work transfer will be *from* the cylinder gases to the piston if the exhaust stroke pressure is lower than the intake pressure, which is normally the case with highly loaded turbocharged engines.†

† With some two-stroke engine concepts there is a piston pumping work term associated with compressing the scavenging air in the crankcase.

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The power per cylinder is related to the indicated work per cycle by

$$P_i = \frac{W_{c,i} N}{n_R} \tag{2.15}$$

where n_R is the number of crank revolutions for each power stroke per cylinder. For four-stroke cycles, n_R equals 2; for two-stroke cycles, n_R equals 1. This power is the indicated power; i.e., the rate of work transfer from the gas within the cylinder to the piston. It differs from the brake power by the power absorbed in overcoming engine friction, driving engine accessories, and (in the case of gross indicated power) the pumping power.

In discussing indicated quantities of the four-stroke cycle engine, such as work per cycle or power, the definition used for "indicated" (i.e., gross or net) *should always be explicitly stated.* The gross indicated output, the definition most commonly used, will be chosen where possible in this book for the following reasons. Indicated quantities are used primarily to identify the impact of the compression, combustion, and expansion processes on engine performance, etc. The gross indicated output is, therefore, the most appropriate definition. It represents the sum of the useful work available at the shaft and the work required to overcome all the engine losses. Furthermore, the standard engine test codes[2] define procedures for measuring brake power and friction power (the friction power test provides a close approximation to the total lost power in the engine). The sum of brake power and friction power provides an alternative way of estimating indicated power; the value obtained is a close approximation to the gross indicated power.

The terms brake and indicated are used to describe other parameters such as mean effective pressure, specific fuel consumption, and specific emissions (see the following sections) in a manner similar to that used for work per cycle and power.

2.5 MECHANICAL EFFICIENCY

We have seen that part of the gross indicated work per cycle or power is used to expel exhaust gases and induct fresh charge. An additional portion is used to overcome the friction of the bearings, pistons, and other mechanical components of the engine, and to drive the engine accessories. All of these power requirements are grouped together and called *friction power* P_f.† Thus:

$$P_{ig} = P_b + P_f \tag{2.16}$$

Friction power is difficult to determine accurately. One common approach for high-speed engines is to drive or motor the engine with a dynamometer (i.e., operate the engine without firing it) and measure the power which has to be

† The various components of friction power are examined in detail in Chap. 13.

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supplied by the dynamometer to overcome *all* these frictional losses. The engine speed, throttle setting, oil and water temperatures, and ambient conditions are kept the same in the motored test as under firing conditions. The major sources of inaccuracy with this method are that gas pressure forces on the piston and rings are lower in the motored test than when the engine is firing and that the oil temperatures on the cylinder wall are also lower under motoring conditions.

The ratio of the brake (or useful) power delivered by the engine to the indicated power is called the *mechanical efficiency* η_m:

$$\eta_m = \frac{P_b}{P_{ig}} = 1 - \frac{P_f}{P_{ig}} \tag{2.17}$$

Since the friction power includes the power required to pump gas into and out of the engine, mechanical efficiency depends on throttle position as well as engine design and engine speed. Typical values for a modern automotive engine at wide-open or full throttle are 90 percent at speeds below about 30 to 40 rev/s (1800 to 2400 rev/min), decreasing to 75 percent at maximum rated speed. As the engine is throttled, mechanical efficiency decreases, eventually to zero at idle operation.

2.6 ROAD-LOAD POWER

A part-load power level useful as a reference point for testing automobile engines is the power required to drive a vehicle on a level road at a steady speed. Called *road-load power*, this power overcomes the rolling resistance which arises from the friction of the tires and the aerodynamic drag of the vehicle. Rolling resistance and drag coefficients, C_R and C_D, respectively, are determined empirically. An approximate formula for road-load power P_r is

$$P_r = (C_R M_v g + \tfrac{1}{2}\rho_a C_D A_v S_v^2) S_v \tag{2.18a}$$

where C_R = coefficient of rolling resistance ($0.012 < C_R < 0.015$)[3]
M_v = mass of vehicle [for passenger cars: curb mass plus passenger load of 68 kg (150 lbm); in U.S. units W_v = vehicle weight in lbf]
g = acceleration due to gravity
ρ_a = ambient air density
C_D = drag coefficient (for cars: $0.3 < C_D \lesssim 0.5$)[3]
A_v = frontal area of vehicle
S_v = vehicle speed

With the quantities in the units indicated:

$$P_r(\text{kW}) = [2.73 C_R M_v(\text{kg}) + 0.0126 C_D A_v(\text{m}^2) S_v(\text{km/h})^2] S_v(\text{km/h}) \times 10^{-3} \tag{2.18b}$$

or

$$P_r(\text{hp}) = \frac{[C_R W_v(\text{lbf}) + 0.0025 C_D A_v(\text{ft}^2) S_v(\text{mi/h})^2] S_v(\text{mi/h})}{375} \tag{2.18c}$$

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2.7 MEAN EFFECTIVE PRESSURE

While torque is a valuable measure of a particular engine's ability to do work, it depends on engine size. A more useful relative engine performance measure is obtained by dividing the work per cycle by the cylinder volume displaced per cycle. The parameter so obtained has units of force per unit area and is called the *mean effective pressure* (mep). Since, from Eq. (2.15),

$$\text{Work per cycle} = \frac{Pn_R}{N}$$

where n_R is the number of crank revolutions for each power stroke per cylinder (two for four-stroke cycles; one for two-stroke cycles), then

$$\text{mep} = \frac{Pn_R}{V_d N} \tag{2.19a}$$

For SI and U.S. units, respectively,

$$\text{mep(kPa)} = \frac{P(\text{kW})n_R \times 10^3}{V_d(\text{dm}^3)N(\text{rev/s})} \tag{2.19b}$$

$$\text{mep(lb/in}^2) = \frac{P(\text{hp})n_R \times 396{,}000}{V_d(\text{in}^3)N(\text{rev/min})} \tag{2.19c}$$

Mean effective pressure can also be expressed in terms of torque by using Eq. (2.13):

$$\text{mep(kPa)} = \frac{6.28n_R T(\text{N}\cdot\text{m})}{V_d(\text{dm}^3)} \tag{2.20a}$$

or

$$\text{mep(lb/in}^2) = \frac{75.4n_R T(\text{lbf}\cdot\text{ft})}{V_d(\text{in}^3)} \tag{2.20b}$$

The maximum brake mean effective pressure of good engine designs is well established, and is essentially constant over a wide range of engine sizes. Thus, the actual bmep that a particular engine develops can be compared with this norm, and the effectiveness with which the engine designer has used the engine's displaced volume can be assessed. Also, for design calculations, the engine displacement required to provide a given torque or power, at a specified speed, can be estimated by assuming appropriate values for bmep for that particular application.

Typical values for bmep are as follows. For naturally aspirated spark-ignition engines, maximum values are in the range 850 to 1050 kPa ($\sim$ 125 to 150 lb/in^2) at the engine speed where maximum torque is obtained (about 3000 rev/min). At the maximum rated power, bmep values are 10 to 15 percent lower. For turbocharged automotive spark-ignition engines the maximum bmep is in the 1250 to 1700 kPa (180 to 250 lb/in^2) range. At the maximum rated power, bmep is in the 900 to 1400 kPa (130 to 200 lb/in^2) range. For naturally aspirated four-stroke diesels, the maximum bmep is in the 700 to 900 kPa (100 to 130

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lb/in^2) range, with the bmep at the maximum rated power of about 700 kPa (100 lb/in^2). Turbocharged four-stroke diesel maximum bmep values are typically in the range 1000 to 1200 kPa (145 to 175 lb/in^2); for turbocharged aftercooled engines this can rise to 1400 kPa. At maximum rated power, bmep is about 850 to 950 kPa (125 to 140 lb/in^2). Two-stroke cycle diesels have comparable performance to four-stroke cycle engines. Large low-speed two-stroke cycle engines can achieve bmep values of about 1600 kPa.

An example of how the above engine performance parameters can be used to initiate an engine design is given below.

Example. A four-cylinder automotive spark-ignition engine is being designed to provide a maximum brake torque of 150 N·m (110 lbf·ft) in the mid-speed range ($\sim$ 3000 rev/min). Estimate the required engine displacement, bore and stroke, and the maximum brake power the engine will deliver.

Equation (2.20*a*) relates torque and mep. Assume that 925 kPa is an appropriate value for bmep at the maximum engine torque point. Equation (2.20*a*) gives

$$V(\text{dm}^3) = \frac{6.28 n_R T_{\max}(\text{N}\cdot\text{m})}{\text{bmep}_{\max}(\text{kPa})} = \frac{6.28 \times 2 \times 150}{925} = 2 \text{ dm}^3$$

For a four-cylinder engine, the displaced volume, bore, and stroke are related by

$$V_d = 4 \times \frac{\pi}{4} B^2 L$$

Assume $B = L$; this gives $B = L = 86$ mm.

The maximum rated engine speed can be estimated from an appropriate value for the maximum mean piston speed, 15 m/s (see Sec. 2.2):

$$\bar{S}_{\text{pmax}} = 2LN_{\max} \rightarrow N_{\max} = 87 \text{ rev/s (5200 rev/min)}$$

The maximum brake power can be estimated from the typical bmep value at maximum power, 800 kPa (116 lb/in^2), using Eq. (2.19*b*):

$$P_{\text{bmax}}(\text{kW}) = \frac{\text{bmep(kPa)} V(\text{dm}^3) N_{\max}(\text{rev/s})}{n_R \times 10^3} = \frac{800 \times 2 \times 87}{2 \times 10^3} = 70 \text{ kW}$$

2.8 SPECIFIC FUEL CONSUMPTION AND EFFICIENCY

In engine tests, the fuel consumption is measured as a flow rate—mass flow per unit time $\dot{m}_f$. A more useful parameter is the *specific fuel consumption* (sfc)—the fuel flow rate per unit power output. It measures how efficiently an engine is using the fuel supplied to produce work:

$$\text{sfc} = \frac{\dot{m}_f}{P} \tag{2.21}$$

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With units,

$$\text{sfc(mg/J)} = \frac{\dot{m}_f\text{(g/s)}}{P\text{(kW)}} \tag{2.22a}$$

or

$$\text{sfc(g/kW} \cdot \text{h)} = \frac{\dot{m}_f\text{(g/h)}}{P\text{(kW)}} = 608.3\ \text{sfc(lbm/hp} \cdot \text{h)} \tag{2.22b}$$

or

$$\text{sfc(lbm/hp} \cdot \text{h)} = \frac{\dot{m}_f\text{(lbm/h)}}{P\text{(hp)}} = 1.644 \times 10^3\ \text{sfc(g/kW} \cdot \text{h)} \tag{2.22c}$$

Low values of sfc are obviously desirable. For SI engines typical best values of brake specific fuel consumption are about 75 μg/J = 270 g/kW · h = 0.47 lbm/hp · h. For CI engines, best values are lower and in large engines can go below 55 μg/J = 200 g/kW · h = 0.32 lbm/hp · h.

The specific fuel consumption has units. A dimensionless parameter that relates the desired engine output (work per cycle or power) to the necessary input (fuel flow) would have more fundamental value. The ratio of the work produced per cycle to the amount of fuel energy supplied per cycle that can be released in the combustion process is commonly used for this purpose. It is a measure of the engine's efficiency. The fuel energy supplied which can be released by combustion is given by the mass of fuel supplied to the engine per cycle times the heating value of the fuel. The heating value of a fuel, Q_{HV}, defines its energy content. It is determined in a standardized test procedure in which a known mass of fuel is fully burned with air, and the thermal energy released by the combustion process is absorbed by a calorimeter as the combustion products cool down to their original temperature.

This measure of an engine's "efficiency," which will be called the *fuel conversion efficiency* η_f,† is given by

$$\eta_f = \frac{W_c}{m_f Q_{HV}} = \frac{(Pn_R/N)}{(\dot{m}_f n_R/N)Q_{HV}} = \frac{P}{\dot{m}_f Q_{HV}} \tag{2.23}$$

where m_f is the mass of fuel inducted per cycle. Substitution for $P/\dot{m}_f$ from Eq. (2.21) gives

$$\eta_f = \frac{1}{\text{sfc}\ Q_{HV}} \tag{2.24a}$$

† This empirically defined engine efficiency has previously been called thermal efficiency or enthalpy efficiency. The term fuel conversion efficiency is preferred because it describes this quantity more precisely, and distinguishes it clearly from other definitions of engine efficiency which will be developed in Sec. 3.6. Note that there are several different definitions of heating value (see Sec. 3.5). The numerical values do not normally differ by more than a few percent, however. In this text, the lower heating value at constant pressure is used in evaluating the fuel conversion efficiency.

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or with units:

$$\eta_f = \frac{1}{\text{sfc(mg/J)}Q_{\text{HV}}\text{(MJ/kg)}} \tag{2.24b}$$

$$\eta_f = \frac{3600}{\text{sfc(g/kW} \cdot \text{h)}Q_{\text{HV}}\text{(MJ/kg)}} \tag{2.24c}$$

$$\eta_f = \frac{2545}{\text{sfc(lbm/hp} \cdot \text{h)}Q_{\text{HV}}\text{(Btu/lbm)}} \tag{2.24d}$$

Typical heating values for the commercial hydrocarbon fuels used in engines are in the range 42 to 44 MJ/kg (18,000 to 19,000 Btu/lbm). Thus, specific fuel consumption is inversely proportional to fuel conversion efficiency for normal hydrocarbon fuels.

Note that the fuel energy supplied to the engine per cycle is not fully released as thermal energy in the combustion process because the actual combustion process in incomplete. When enough air is present in the cylinder to oxidize the fuel completely, almost all (more than about 96 percent) of this fuel energy supplied is transferred as thermal energy to the working fluid. When insufficient air is present to oxidize the fuel completely, lack of oxygen prevents this fuel energy supplied from being fully released. This topic is discussed in more detail in Secs. 3.5 and 4.9.4.

2.9 AIR/FUEL AND FUEL/AIR RATIOS

In engine testing, both the air mass flow rate $\dot{m}_a$ and the fuel mass flow rate $\dot{m}_f$ are normally measured. The ratio of these flow rates is useful in defining engine operating conditions:

$$\text{Air/fuel ratio } (A/F) = \frac{\dot{m}_a}{\dot{m}_f} \tag{2.25}$$

$$\text{Fuel/air ratio } (F/A) = \frac{\dot{m}_f}{\dot{m}_a} \tag{2.26}$$

The normal operating range for a conventional SI engine using gasoline fuel is $12 \leq A/F \leq 18$ $(0.056 \leq F/A \leq 0.083)$; for CI engines with diesel fuel, it is $18 \leq A/F \leq 70$ $(0.014 \leq F/A \leq 0.056)$.

2.10 VOLUMETRIC EFFICIENCY

The intake system—the air filter, carburetor, and throttle plate (in a spark-ignition engine), intake manifold, intake port, intake valve—restricts the amount of air which an engine of given displacement can induct. The parameter used to measure the effectiveness of an engine's induction process is the *volumetric efficiency* η_v. Volumetric efficiency is only used with four-stroke cycle engines which have a distinct induction process. It is defined as the volume flow rate of air into

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the intake system divided by the rate at which volume is displaced by the piston:

$$\eta_v = \frac{2\dot{m}_a}{\rho_{a,i} V_d N} \tag{2.27a}$$

where $\rho_{a,i}$ is the inlet air density. An alternative equivalent definition for volumetric efficiency is

$$\eta_v = \frac{m_a}{\rho_{a,i} V_d} \tag{2.27b}$$

where m_a is the mass of air inducted into the cylinder per cycle.

The inlet density may either be taken as atmosphere air density (in which case η_v measures the pumping performance of the entire inlet system) or may be taken as the air density in the inlet manifold (in which case η_v measures the pumping performance of the inlet port and valve only). Typical maximum values of η_v for naturally aspirated engines are in the range 80 to 90 percent. The volumetric efficiency for diesels is somewhat higher than for SI engines. Volumetric efficiency is discussed more fully in Sec. 6.2.

2.11 ENGINE SPECIFIC WEIGHT AND SPECIFIC VOLUME

Engine weight and bulk volume for a given rated power are important in many applications. Two parameters useful for comparing these attributes from one engine to another are:

$$\text{Specific weight} = \frac{\text{engine weight}}{\text{rated power}} \tag{2.28}$$

$$\text{Specific volume} = \frac{\text{engine volume}}{\text{rated power}} \tag{2.29}$$

For these parameters to be useful in engine comparisons, a consistent definition of what components and auxiliaries are included in the term "engine" must be adhered to. These parameters indicate the effectiveness with which the engine designer has used the engine materials and packaged the engine components.[4]

2.12 CORRECTION FACTORS FOR POWER AND VOLUMETRIC EFFICIENCY

The pressure, humidity, and temperature of the ambient air inducted into an engine, at a given engine speed, affect the air mass flow rate and the power output. Correction factors are used to adjust measured wide-open-throttle power and volumetric efficiency values to standard atmospheric conditions to provide a more accurate basis for comparisons between engines. Typical standard ambient

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conditions used are:

Dry air pressure	Water vapour pressure	Temperature
736.6 mmHg	9.65 mmHg	29.4°C
29.00 inHg	0.38 inHg	85°F

The basis for the correction factor is the equation for one-dimensional steady compressible flow through an orifice or flow restriction of effective area A_E (see App. C):

$$\dot{m} = \frac{A_E p_0}{\sqrt{RT_0}} \left\{ \frac{2\gamma}{\gamma - 1} \left[\left(\frac{p}{p_0} \right)^{2/\gamma} - \left(\frac{p}{p_0} \right)^{(\gamma+1)/\gamma} \right] \right\}^{1/2} \tag{2.30}$$

In deriving this equation, it has been assumed that the fluid is an ideal gas with gas constant R and that the ratio of specific heats ($c_p/c_v = \gamma$) is a constant; p_0 and T_0 are the total pressure and temperature upstream of the restriction and p is the pressure at the throat of the restriction.

If, in the engine, p/p_0 is assumed constant at wide-open throttle, then for a given intake system and engine, the mass flow rate of dry air $\dot{m}_a$ varies as

$$\dot{m}_a \propto \frac{p_0}{\sqrt{T_0}} \tag{2.31}$$

For mixtures containing the proper amount of fuel to use all the air available (and thus provide maximum power), the indicated power at full throttle P_i will be proportional to $\dot{m}_a$, the *dry* air flow rate. Thus if

$$P_{i,s} = C_F P_{i,m} \tag{2.32}$$

where the subscripts s and m denote values at the standard and measured conditions, respectively, the correction factor C_F is given by

$$C_F = \frac{p_{s,d}}{p_m - p_{v,m}} \left(\frac{T_m}{T_s} \right)^{1/2} \tag{2.33}$$

where $p_{s,d}$ = standard dry-air absolute pressure
p_m = measured ambient-air absolute pressure
$p_{v,m}$ = measured ambient–water vapour partial pressure
T_m = measured ambient temperature, K
T_s = standard ambient temperature, K

The rated *brake* power is corrected by using Eq. (2.33) to correct the *indicated* power and making the assumption that friction power is unchanged. Thus

$$P_{b,s} = C_F P_{i,m} - P_{f,m} \tag{2.34}$$

Volumetric efficiency is proportional to $\dot{m}_a/\rho_a$ [see Eq. (2.27)]. Since ρ_a is proportional to p/T, the correction factor for volumetric efficiency, C'_F, is

$$C'_F = \frac{\eta_{v,s}}{\eta_{v,m}} = \left(\frac{T_s}{T_m} \right)^{1/2} \tag{2.35}$$

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2.13 SPECIFIC EMISSIONS AND EMISSIONS INDEX

Levels of emissions of oxides of nitrogen (nitric oxide, NO, and nitrogen dioxide, NO_2, usually grouped together as NO_x), carbon monoxide (CO), unburned hydrocarbons (HC), and particulates are important engine operating characteristics.

The concentrations of gaseous emissions in the engine exhaust gases are usually measured in parts per million or percent by volume (which corresponds to the mole fraction multiplied by 10^6 or by 10^2, respectively). Normalized indicators of emissions levels are more useful, however, and two of these are in common use. *Specific emissions* are the mass flow rate of pollutant per unit power output:

$$sNO_x = \frac{\dot{m}_{NO_x}}{P} \tag{2.36a}$$

$$sCO = \frac{\dot{m}_{CO}}{P} \tag{2.36b}$$

$$sHC = \frac{\dot{m}_{HC}}{P} \tag{2.36c}$$

$$sPart = \frac{\dot{m}_{part}}{P} \tag{2.36d}$$

Indicated and brake specific emissions can be defined. Units in common use are μg/J, g/kW · h, and g/hp · h.

Alternatively, emission rates can be normalized by the fuel flow rate. An *emission index* (EI) is commonly used: e.g.,

$$EI_{NO_x} = \frac{\dot{m}_{NO_x}(g/s)}{\dot{m}_f(kg/s)} \tag{2.37}$$

with similar expressions for CO, HC, and particulates.

2.14 RELATIONSHIPS BETWEEN PERFORMANCE PARAMETERS

The importance of the parameters defined in Secs. 2.8 to 2.10 to engine performance becomes evident when power, torque, and mean effective pressure are expressed in terms of these parameters. From the definitions of engine power [Eq. (2.13)], mean effective pressure [Eq. (2.19)], fuel conversion efficiency [Eq. (2.23)], fuel/air ratio [Eq. (2.26)], and volumetric efficiency [Eq. (2.27)], the following relationships between engine performance parameters can be developed. For power P:

$$P = \frac{\eta_f m_a N Q_{HV}(F/A)}{n_R} \tag{2.38}$$

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For four-stroke cycle engines, volumetric efficiency can be introduced:

$$P = \frac{\eta_f \eta_v N V_d Q_{HV} \rho_{a,i}(F/A)}{2} \tag{2.39}$$

For torque T:

$$T = \frac{\eta_f \eta_v V_d Q_{HV} \rho_{a,i}(F/A)}{4\pi} \tag{2.40}$$

For mean effective pressure:

$$\text{mep} = \eta_f \eta_v Q_{HV} \rho_{a,i}(F/A) \tag{2.41}$$

The power per unit piston area, often called the *specific power*, is a measure of the engine designer's success in using the available piston area regardless of cylinder size. From Eq. (2.39), the specific power is

$$\frac{P}{A_p} = \frac{\eta_f \eta_v N L Q_{HV} \rho_{a,i}(F/A)}{2} \tag{2.42}$$

Mean piston speed can be introduced with Eq. (2.9) to give

$$\frac{P}{A_p} = \frac{\eta_f \eta_v \bar{S}_p Q_{HV} \rho_{a,i}(F/A)}{4} \tag{2.43}$$

Specific power is thus proportional to the product of mean effective pressure and mean piston speed.

These relationships illustrate the direct importance to engine performance of:

1. High fuel conversion efficiency
2. High volumetric efficiency
3. Increasing the output of a given displacement engine by increasing the inlet air density
4. Maximum fuel/air ratio that can be usefully burned in the engine
5. High mean piston speed

2.15 ENGINE DESIGN AND PERFORMANCE DATA

Engine ratings usually indicate the highest power at which manufacturers expect their products to give satisfactory economy, reliability, and durability under service conditions. Maximum torque, and the speed at which it is achieved, is usually given also. Since both of these quantities depend on displaced volume, for comparative analyses between engines of different displacements in a given engine category normalized performance parameters are more useful. The following measures, at the operating points indicated, have most significance:[4]

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TABLE 2.1
Typical design and operating data for internal combustion engines

	Operating cycle	Compression ratio	Bore, m	Stroke/ bore	Rated maximum: Speed, rev/min	Rated maximum: bmep, atm	Rated maximum: Power per unit volume kW/dm³	Weight/ power ratio, kg/kW	Approx. best bsfc, g/kW·h
Spark-ignition engines:									
Small (e.g., motorcycles)	2S,4S	6–11	0.05–0.085	1.2–0.9	4500–7500	4–10	20–60	5.5–2.5	350
Passenger cars	4S	8–10	0.07–0.1	1.1–0.9	4500–6500	7–10	20–50	4–2	270
Trucks	4S	7–9	0.09–0.13	1.2–0.7	3600–5000	6.5–7	25–30	6.5–2.5	300
Large gas engines	2S,4S	8–12	0.22–0.45	1.1–1.4	300–900	6.8–12	3–7	23–35	200
Wankel engines	4S	≈ 9	0.57 dm³ per chamber		6000–8000	9.5–10.5	35–45	1.6–0.9	300
Diesel engines:									
Passenger cars	4S	17–23	0.075–0.1	1.2–0.9	4000–5000	5–7.5	18–22	5–2.5	250
Trucks (NA)	4S	16–22	0.1–0.15	1.3–0.8	2100–4000	6–9	15–22	7–4	210
Trucks (TC)	4S	14–20	0.1–0.15	1.3–0.8	2100–4000	12–18	18–26	7–3.5	200
Locomotive, industrial, marine	4S,2S	12–18	0.15–0.4	1.1–1.3	425–1800	7–23	5–20	6–18	190
Large engines, marine and stationary	2S	10–12	0.4–1	1.2–3	110–400	9–17	2–8	12–50	180

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1. At maximum or normal rated point:

   Mean piston speed. Measures comparative success in handling loads due to inertia of the parts, resistance to air flow, and/or engine friction.

   Brake mean effective pressure. In naturally aspirated engines bmep is not stress limited. It then reflects the product of volumetric efficiency (ability to induct air), fuel/air ratio (effectiveness of air utilization in combustion), and fuel conversion efficiency. In supercharged engines bmep indicates the degree of success in handling higher gas pressures and thermal loading.

   Power per unit piston area. Measures the effectiveness with which the piston area is used, regardless of cylinder size.

   Specific weight. Indicates relative economy with which materials are used.

   Specific volume. Indicates relative effectiveness with which engine space has been utilized.

2. At all speeds at which the engine will be used with full throttle or with maximum fuel-pump setting:

   Brake mean effective pressure. Measures ability to obtain/provide high air flow and use it effectively over the full range.

3. At all useful regimes of operation and particularly in those regimes where the engine is run for long periods of time:

   Brake specific fuel consumption or *fuel conversion efficiency.*
   Brake specific emissions.

Typical performance data for spark-ignition and diesel engines over the normal production size range are summarized in Table 2.1.[4] The four-stroke cycle dominates except in the smallest and largest engine sizes. The larger engines are turbocharged or supercharged. The maximum rated engine speed decreases as engine size increases, maintaining the maximum mean piston speed in the range of about 8 to 15 m/s. The maximum brake mean effective pressure for turbocharged and supercharged engines is higher than for naturally aspirated engines. Because the maximum fuel/air ratio for spark-ignition engines is higher than for diesels, their naturally aspirated maximum bmep levels are higher. As engine size increases, brake specific fuel consumption decreases and fuel conversion efficiency increases, due to reduced importance of heat losses and friction. For the largest diesel engines, brake fuel conversion efficiencies of about 50 percent and indicated fuel conversion efficiencies of over 55 percent can be obtained.

PROBLEMS

2.1. Explain why the brake mean effective pressure of a naturally aspirated diesel engine is lower than that of a naturally aspirated spark-ignition engine. Explain why the bmep is lower at the maximum rated power for a given engine than the bmep at the maximum torque.

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2.2. Describe the impact on air flow, maximum torque, and maximum power of changing a spark-ignition engine cylinder head from 2 valves per cylinder to 4 valves (2 inlet and 2 exhaust) per cylinder.

2.3. Calculate the mean piston speed, bmep, and specific power of the spark-ignition engines in Figs. 1-4, 1-9, and 1-12 at their maximum rated power.

2.4. Calculate the mean piston speed, bmep, and specific power of the diesel engines in Figs. 1-20, 1-21, 1-22, 1-23, and 1-24 at their maximum rated power. Briefly explain any significant differences.

2.5. Develop an equation for the power required to drive a vehicle at constant speed up a hill of angle α, in terms of vehicle speed, mass, frontal area, drag coefficient, coefficient of rolling resistance, α, and acceleration due to gravity. Calculate this power when the car mass is 1500 kg, the hill angle is 15 degrees, and the vehicle speed is 50 mi/h.

2.6. The spark-ignition engine in Fig. 1-4 is operating at a mean piston speed of 10 m/s. The measured air flow is 60 g/s. Calculate the volumetric efficiency based on atmospheric conditions.

2.7. The diesel engine of Fig. 1-20 is operating with a mean piston speed of 8 m/s. Calculate the air flow if the volumetric efficiency is 0.92. If (F/A) is 0.05 what is the fuel flow rate, and the mass of fuel injected per cylinder per cycle?

2.8. The brake fuel conversion efficiency of a spark-ignition engine is 0.3, and varies little with fuel type. Calculate the brake specific fuel consumption for isooctane, gasoline, methanol, and hydrogen (relevant data are in App. D).

2.9. You are doing a preliminary design study of a turbocharged four-stroke diesel engine. The maximum rated power is limited by stress considerations to a brake mean effective pressure of 1200 kPa and maximum value of the mean piston speed of 12 m/s.

(*a*) Derive an equation relating the engine inlet pressure (pressure in the inlet manifold at the turbocharger compressor exit) to the fuel/air ratio at this maximum rated power operating point. Other reciprocating engine parameters (e.g., volumetric efficiency, fuel conversion efficiency, bmep, etc.) appear in this equation also.

(*b*) The maximum rated brake power requirement for this engine is 400 kW. Estimate sensible values for number of cylinders, cylinder bore, stroke, and determine the maximum rated speed of this preliminary engine design.

(*c*) If the pressure ratio across the compressor is 2, estimate the overall fuel/air and air/fuel ratios at the maximum rated power. Assume appropriate values for any other parameters you may need.

2.10. In the reciprocating engine, during the power or expansion stroke, the gas pressure force acting on the piston is transmitted to the crankshaft via the connecting rod. List the forces acting on the piston during this part of the operating cycle. Show the direction of the forces acting on the piston on a sketch of the piston, cylinder, connecting rod, crank arrangement. Write out the force balance for the piston (*a*) along the cylinder axis and (*b*) transverse to the cylinder axis in the plane containing the connecting rod. (You are not asked to manipulate or solve these equations.)

2.11. You are designing a four-stroke cycle diesel engine to provide a brake power of 300 kW naturally aspirated at its maximum rated speed. Based on typical values for brake mean effective pressure and maximum mean piston speed, estimate the required engine displacement, and the bore and stroke for sensible cylinder geometry and number of engine cylinders. What is the maximum rated engine speed (rev/min)

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for your design? What would be the brake torque (N · m) and the fuel flow rate (g/h) at this maximum speed? Assume a maximum mean piston speed of 12 m/s is typical of good engine designs.

2.12. The power per unit piston area P/A_p (often called the specific power) is a measure of the designer's success in using the available piston area regardless of size.

(*a*) Derive an expression for P/A_p in terms of mean effective pressure and mean piston speed for two-stroke and four-stroke engine cycles.

(*b*) Compute typical maximum values of P/A_p for a spark-ignition engine (e.g., Fig. 1-4), a turbocharged four-stroke cycle diesel engine (e.g., Fig. 1-22), and a large marine diesel (Fig. 1-24). Table 2-1 may be helpful. State your assumptions clearly.

2.13. Several velocities, time, and length scales are useful in understanding what goes on inside engines. Make estimates of the following quantities for a 1.6-liter displacement four-cylinder spark-ignition engine, operating at wide-open throttle at 2500 rev/min.

(*a*) The mean piston speed and the maximum piston speed.

(*b*) The maximum charge velocity in the intake port (the port area is about 20 percent of the piston area).

(*c*) The time occupied by one engine operating cycle, the intake process, the compression process, the combustion process, the expansion process, and the exhaust process. (*Note*: The word *process* is used here not the word *stroke*.)

(*d*) The average velocity with which the flame travels across the combustion chamber.

(*e*) The length of the intake system (the intake port, the manifold runner, etc.) which is filled by one cylinder charge just before the intake valve opens and this charge enters the cylinder (i.e., how far back from the intake valve, in centimeters, one cylinder volume extends in the intake system).

(*f*) The length of exhaust system filled by one cylinder charge after it exits the cylinder (assume an average exhaust gas temperature of 425°C).

You will have to make several appropriate geometric assumptions. The calculations are straightforward, and only approximate answers are required.

2.14. The values of mean effective pressure at rated speed, maximum mean piston speed, and maximum specific power (engine power/total piston area) are essentially independent of cylinder size for naturally aspirated engines of a given type. If we also assume that engine weight per unit displaced volume is essentially constant, how will the *specific weight* of an engine (engine weight/maximum rated power) at fixed total displaced volume vary with the number of cylinders? Assume the bore and stroke are equal.

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CHAPTER 3

THERMOCHEMISTRY OF FUEL-AIR MIXTURES

3.1 CHARACTERIZATION OF FLAMES

Combustion of the fuel-air mixture inside the engine cylinder is one of the processes that controls engine power, efficiency, and emissions. Some background in relevant combustion phenomena is therefore a necessary preliminary to understanding engine operation. These combustion phenomena are different for the two main types of engines—spark-ignition and diesel—which are the subject of this book. In spark-ignition engines, the fuel is normally mixed with air in the engine intake system. Following the compression of this fuel-air mixture, an electrical discharge initiates the combustion process; a flame develops from the "kernal" created by the spark discharge and propagates across the cylinder to the combustion chamber walls. At the walls, the flame is "quenched" or extinguished as heat transfer and destruction of active species at the wall become the dominant processes. An undesirable combustion phenomenon—the "spontaneous" ignition of a substantial mass of fuel-air mixture ahead of the flame, before the flame can propagate through this mixture (which is called the end-gas)—can also occur. This autoignition or self-explosion combustion phenomenon is the cause of spark-ignition engine knock which, due to the high pressures generated, can lead to engine damage.

In the diesel engine, the fuel is injected into the cylinder into air already at high pressure and temperature, near the end of the compression stroke. The autoignition, or self-ignition, of portions of the developing mixture of already

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injected and vaporized fuel with this hot air starts the combustion process, which spreads rapidly. Burning then proceeds as fuel and air mix to the appropriate composition for combustion to take place. Thus, fuel-air mixing plays a controlling role in the diesel combustion process.

Chapters 3 and 4 focus on the thermochemistry of combustion: i.e., the composition and thermodynamic properties of the pre- and postcombustion working fluids in engines and the energy changes associated with the combustion processes that take place inside the engine cylinder. Later chapters (9 and 10) deal with the phenomenological aspects of engine combustion: i.e., the details of the physical and chemical processes by which the fuel-air mixture is converted to burned products. At this point it is useful to review briefly the key combustion phenomena which occur in engines to provide an appropriate background for the material which follows. More detailed information on these combustion phenomena can be found in texts on combustion such as those of Fristrom and Westenberg[1] and Glassman.[2]

The combustion process is a fast exothermic gas-phase reaction (where oxygen is usually one of the reactants). A flame is a combustion reaction which can propagate subsonically through space; motion of the flame relative to the unburned gas is the important feature. Flame structure does not depend on whether the flame moves relative to the observer or remains stationary as the gas moves through it. The existence of flame motion implies that the reaction is confined to a zone which is small in thickness compared to the dimensions of the apparatus—in our case the engine combustion chamber. The reaction zone is usually called the flame front. This flame characteristic of spatial propagation is the result of the strong coupling between chemical reaction, the transport processes of mass diffusion and heat conduction, and fluid flow. The generation of heat and active species accelerate the chemical reaction; the supply of fresh reactants, governed by the convection velocity, limits the reaction. When these processes are in balance, a steady-state flame results.[1]

Flames are usually classified according to the following overall characteristics. The first of these has to do with the composition of the reactants as they enter the reaction zone. If the fuel and oxidizer are essentially uniformly mixed together, the flame is designated as *premixed*. If the reactants are not premixed and must mix together in the same region where reaction takes place, the flame is called a *diffusion* flame because the mixing must be accomplished by a diffusion process. The second means of classification relates to the basic character of the gas flow through the reaction zone: whether it is *laminar* or *turbulent*. In laminar (or streamlined) flow, mixing and transport are done by molecular processes. Laminar flows only occur at low Reynolds number. The Reynolds number (density × velocity × lengthscale/viscosity) is the ratio of inertial to viscous forces. In turbulent flows, mixing and transport are enhanced (usually by a substantial factor) by the macroscopic relative motion of eddies or lumps of fluid which are the characteristic feature of a turbulent (high Reynolds number) flow. A third area of classification is whether the flame is *steady* or *unsteady*. The distinguishing feature here is whether the flame structure and motion change with

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time. The final characterizing feature is the *initial phase* of the reactants—gas, liquid, or solid.

Flames in engines are unsteady, an obvious consequence of the internal combustion engine's operating cycle. Engine flames are turbulent. Only with substantial augmentation of laminar transport processes by the turbulent convection processes can mixing and burning rates and flame-propagation rates be made fast enough to complete the engine combustion process within the time available.

The conventional spark-ignition flame is thus a premixed unsteady turbulent flame, and the fuel-air mixture through which the flame propagates is in the gaseous state. The diesel engine combustion process is predominantly an unsteady turbulent diffusion flame, and the fuel is initially in the liquid phase. Both these flames are extremely complicated because they involve the coupling of the complex chemical mechanism, by which fuel and oxidizer react to form products, with the turbulent convective transport process. The diesel combustion process is even more complicated than the spark-ignition combustion process, because vaporization of liquid fuel and fuel-air mixing processes are involved too. Chapters 9 and 10 contain a more detailed discussion of the spark-ignition engine and diesel combustion processes, respectively. This chapter reviews the basic thermodynamic and chemical composition aspects of engine combustion.

3.2 IDEAL GAS MODEL

The gas species that make up the working fluids in internal combustion engines (e.g., oxygen, nitrogen, fuel vapor, carbon dioxide, water vapor, etc.) can usually be treated as ideal gases. The relationships between the thermodynamic properties of an ideal gas and of ideal gas mixtures are reviewed in App. B. There can be found the various forms of the ideal gas law:

$$pV = mRT = m\frac{\tilde{R}}{M}T = n\tilde{R}T \tag{3.1}$$

where p is the pressure, V the volume, m the mass of gas, R the gas constant for the gas, T the temperature, $\tilde{R}$ the universal gas constant, M the molecular weight, and n the number of moles. Relations for evaluating the specific internal energy u, enthalpy h, and entropy s, specific heats at constant volume c_v and constant pressure c_p, on a per unit mass basis and on a per mole basis (where the notation $\tilde{u}$, $\tilde{h}$, $\tilde{s}$, $\tilde{c}_v$, and $\tilde{c}_p$ is used) of an ideal gas, are developed. Also given are equations for calculating the thermodynamic properties of mixtures of ideal gases.

3.3 COMPOSITION OF AIR AND FUELS

Normally in engines, fuels are burned with air. Dry air is a mixture of gases that has a representative composition by volume of 20.95 percent oxygen, 78.09 percent nitrogen, 0.93 percent argon, and trace amounts of carbon dioxide, neon, helium, methane, and other gases. Table 3.1 shows the relative proportions of the major constituents of dry air.[3]

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TABLE 3.1
Principle constitutents of dry air

Gas	ppm by volume	Molecular weight	Mole fraction	Molar ratio
O_2	209,500	31.998	0.2095	1
N_2	780,900	28.012	0.7905	3.773
Ar	9,300	39.948		
CO_2	300	44.009		
Air	1,000,000	28.962	1.0000	4.773

In combustion, oxygen is the reactive component of air. It is usually sufficiently accurate to regard air as consisting of 21 percent oxygen and 79 percent inert gases taken as nitrogen (often called atmospheric or apparent nitrogen). For each mole of oxygen in air there are

$$\frac{1 - 0.2095}{0.2095} = 3.773$$

moles of atmospheric nitrogen. The molecular weight of air is obtained from Table 3.1 with Eq. (B.17) as 28.962, usually approximated by 29. Because atmospheric nitrogen contains traces of other species, its molecular weight is slightly different from that of pure molecular nitrogen, i.e.,

$$M_{aN_2} = \frac{28.962 - 0.2095 \times 31.998}{1 - 0.2095} = 28.16$$

In the following sections, *nitrogen* will refer to atmospheric nitrogen and a molecular weight of 28.16 will be used. An air composition of 3.773 moles of nitrogen per mole of oxygen will be assumed.

The density of dry air can be obtained from Eq. (3.1) with $R = 8314.3$ J/kmol·K and $M = 28.962$:

$$\rho(\text{kg/m}^3) = \frac{3.483 \times 10^{-3} p(\text{Pa})}{T(\text{K})} \tag{3.2a}$$

or

$$\rho(\text{lbm/ft}^3) = \frac{2.699 p(\text{lbf/in}^2)}{T(^\circ\text{R})} \tag{3.2b}$$

Thus, the value for the density of dry air at 1 atmosphere (1.0133×10^5 Pa, 14.696 lbf/in^2) and 25°C (77°F) is 1.184 kg/m^3 (0.0739 lbm/ft^3).

Actual air normally contains water vapor, the amount depending on temperature and degree of saturation. Typically the proportion by mass is about 1 percent, though it can rise to about 4 percent under extreme conditions. The *relative humidity* compares the water vapor content of air with that required to saturate. It is defined as:

> The ratio of the partial pressure of water vapor actually present to the saturation pressure at the same temperature.

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Water vapor content is measured with a wet- and dry-bulb psychrometer. This consists of two thermometers exposed to a stream of moist air. The dry-bulb temperature is the temperature of the air. The bulb of the other thermometer is wetted by a wick in contact with a water reservoir. The wet-bulb temperature is lower than the dry-bulb temperature due to evaporation of water from the wick. It is a good approximation to assume that the wet-bulb temperature is the adiabatic saturation temperature. Water vapor pressure can be obtained from observed wet- and dry-bulb temperatures and a psychrometric chart such as Fig. 3-1.[4] The effect of humidity on the properties of air is given in Fig. 3-2.[5]

The fuels most commonly used in internal combustion engines (gasoline or petrol, and diesel fuels) are blends of many different hydrocarbon compounds obtained by refining petroleum or crude oil. These fuels are predominantly carbon and hydrogen (typically about 86 percent carbon and 14 percent hydrogen by weight) though diesel fuels can contain up to about 1 percent sulfur. Other fuels of interest are alcohols (which contain oxygen), gaseous fuels (natural gas and liquid petroleum gas), and single hydrocarbon compounds (e.g., methane, propane, isooctane) which are often used in engine research. Properties of the more common internal combustion engine fuels are summarized in App. D.

Some knowledge of the different classes of organic compounds and their

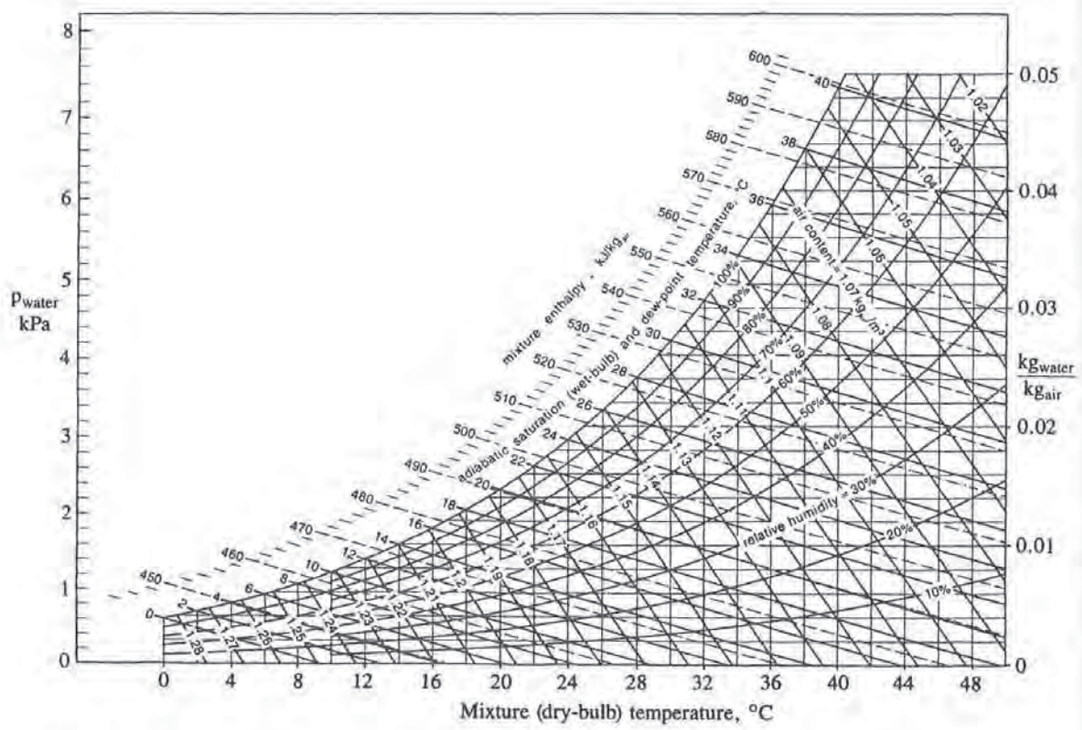


FIGURE 3-1
Psychrometric chart for air-water mixtures at 1 atmosphere. (*From Reynolds.*[4])

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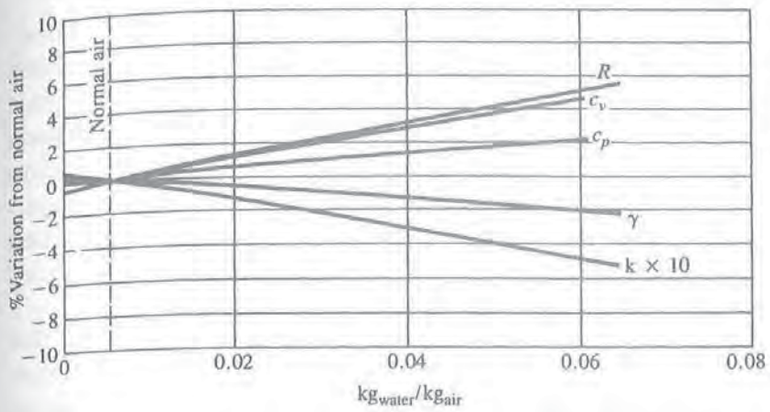
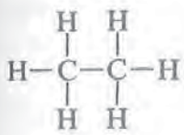


FIGURE 3-2
Effect of humidity on properties of air: R is the gas constant; c_v and c_p are specific heats at constant volume and pressure, respectively; $\gamma = c_p/c_v$; k is the thermal conductivity. (*From Taylor.*[5])

molecular structure is necessary in order to understand combustion mechanisms.[6] The different classes are as follows:

Alkyl Compounds

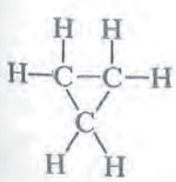
Paraffins
(alk*anes*)



C_nH_{2n+2}

Single-bonded open-chain saturated hydrocarbon molecules: i.e., no more hydrogen can be added. For the larger molecules straight-chain and branched-chain configurations exist. These are called normal (*n*-) and iso compounds, respectively. Examples: CH_4, methane; C_2H_6, ethane; C_3H_8, propane; C_8H_{18}, *n*-octane and isooctane. There are several "isooctanes," depending on the relative position of the branches. By isooctane is usually meant 2,2,4-trimethylpentane, indicating five carbon atoms in the straight chain (pentane) with three methyl (CH_3) branches located respectively at C-atoms 2, 2, and 4. Radicals deficient in one hydrogen take the name methyl, ethyl, propyl, etc.

Cycloparaffins
or *napthenes*
(*cyclanes*)

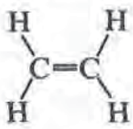


C_nH_{2n}

Single bond (no double bond) ring hydrocarbons. Unsaturated, since ring can be broken and additional hydrogen added. Examples: C_3H_6, cyclopropane (three C-atom ring); C_4H_8, cyclobutane (four C-atom ring); C_5H_{10}, cyclopentane (five C-atom ring).

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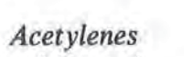
Olefins
(alk*enes*)



C_nH_{2n}

Open-chain hydrocarbons containing a double bond; hence they are unsaturated. Examples are: C_2H_4, ethene (or ethylene); C_3H_6, propene (or propylene); C_4H_8, butene (or butylene); . . . . From butene upwards several structural isomers are possible depending on the location of the double bond in the basic carbon chain. Straight- and branched-chain structures exist. Diolefins contain two double bonds.

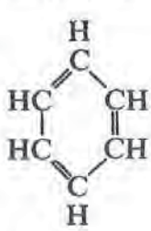
Acetylenes
(alk*ynes*)



C_nH_{2n-2}

Open-chain unsaturated hydrocarbons containing one carbon-carbon triple bond. First member is acetylene, H—C≡C—H. Additional members of the alkyne series comprise open-chain molecules, similar to higher alkenes but with each double bond replaced by a triple bond.

Aromatics

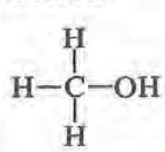


C_nH_{2n-6}

Building block for aromatic hydrocarbons is the benzene (C_6H_6) ring structure shown. This ring structure is very stable and accommodates additional $-CH_2$ groups in side chains and not by ring expansion. Examples: C_7H_8, toluene; C_8H_{10}, xylene (several structural arrangements); . . . . More complex aromatic hydrocarbons incorporate ethyl, propyl, and heavier alkyl side chains in a variety of structural arrangements.

Alcohols

Monohydric alcohols



$C_nH_{2n+1}OH$

In these organic compounds, one hydroxyl (—OH) group is substituted for one hydrogen atom. Thus methane becomes methyl alcohol, CH_3OH (also called methanol); ethane becomes ethyl alcohol, C_2H_5OH (ethanol); etc.

3.4 COMBUSTION STOICHIOMETRY

This section develops relations between the composition of the reactants (fuel and air) of a combustible mixture and the composition of the products. Since these relations depend only on the conservation of mass of each chemical element in the reactants, only the relative elemental composition of the fuel and the relative proportions of fuel and air are needed.

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If sufficient oxygen is available, a hydrocarbon fuel can be completely oxidized. The carbon in the fuel is then converted to carbon dioxide CO_2 and the hydrogen to water H_2O. For example, consider the overall chemical equation for the complete combustion of one mole of propane C_3H_8:

$$C_3H_8 + aO_2 = bCO_2 + cH_2O \tag{3.3}$$

A carbon balance between the reactants and products gives $b = 3$. A hydrogen balance gives $2c = 8$, or $c = 4$. An oxygen balance gives $2b + c = 10 = 2a$, or $a = 5$. Thus Eq. (3.3) becomes

$$C_3H_8 + 5O_2 = 3CO_2 + 4H_2O \tag{3.4}$$

Note that Eq. (3.4) only relates the elemental composition of the reactant and product species; it does not indicate the process by which combustion proceeds, which is much more complex.

Air contains nitrogen, but when the products are at low temperatures the nitrogen is not significantly affected by the reaction. Consider the complete combustion of a general hydrocarbon fuel of average molecular composition C_aH_b with air. The overall complete combustion equation is

$$C_aH_b + \left(a + \frac{b}{4}\right)(O_2 + 3.773N_2) = aCO_2 + \frac{b}{2}H_2O + 3.773\left(a + \frac{b}{4}\right)N_2 \tag{3.5}$$

Note that only the *ratios* of the numbers in front of the symbol for each chemical species are defined by Eq. (3.5); i.e., only the relative proportions on a molar basis are obtained. Thus the fuel composition could have been written CH_y where $y = b/a$.

Equation (3.5) defines the *stoichiometric* (or chemically correct or theoretical) proportions of fuel and air; i.e., there is just enough oxygen for conversion of all the fuel into completely oxidized products. The stoichiometric air/fuel or fuel/air ratios (see Sec. 2.9) depend on fuel composition. From Eq. (3.5):

$$\left(\frac{A}{F}\right)_s = \left(\frac{F}{A}\right)_s^{-1} = \frac{(1 + y/4)(32 + 3.773 \times 28.16)}{12.011 + 1.008y}$$

$$= \frac{34.56(4 + y)}{12.011 + 1.008y} \tag{3.6}$$

The molecular weights of oxygen, atmospheric nitrogen, atomic carbon, and atomic hydrogen are, respectively, 32, 28.16, 12.011, and 1.008. $(A/F)_s$ depends only on y; Fig. 3-3 shows the variation in $(A/F)_s$ as y varies from 1 (e.g., benzene) to 4 (methane).

Example 3.1. A hydrocarbon fuel of composition 84.1 percent by mass C and 15.9 percent by mass H has a molecular weight of 114.15. Determine the number of moles of air required for stoichiometric combustion and the number of moles of products produced per mole of fuel. Calculate $(A/F)_s$, $(F/A)_s$, and the molecular weights of the reactants and the products.

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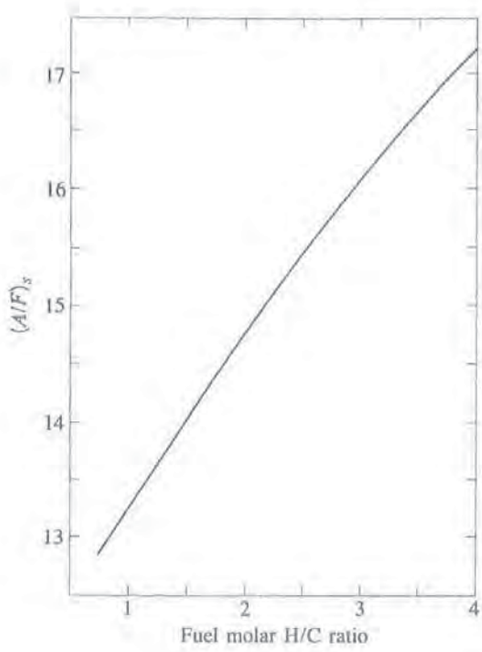


FIGURE 3-3
Stoichiometric air/fuel ratio for air–hydrocarbon fuel mixtures as a function of fuel molar H/C ratio.

Assume a fuel composition $C_a H_b$. The molecular weight relation gives

$$114.15 = 12.011a + 1.008b$$

The gravimetric analysis of the fuel gives

$$\frac{b}{a} = \frac{15.9/1.008}{84.1/12.011} = 2.25$$

Thus

$$a = 8 \qquad b = 18\dagger$$

The fuel is octane C_8H_{18}. Equation (3.5) then becomes

$$\underset{\text{Fuel}}{C_8H_{18}} + 12.5(\underset{\text{Air}}{O_2 + 3.773N_2}) = \underset{\text{Products}}{8CO_2 + 9H_2O + 47.16N_2}$$

In moles:

$$1 + 12.5(1 + 3.773) = 8 + 9 + 47.16$$

$$1 + 59.66 = 64.16$$

Relative mass:

$$114.15 + 59.66 \times 28.96 = 8 \times 44.01 + 9 \times 18.02 + 47.16 \times 28.16$$

$$114.5 + 1727.8 = 1842.3$$

† Note that for fuels which are mixtures of hydrocarbons, a and b need not be integers.

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Per unit mass fuel:

$$1 \quad + \quad 15.14 \quad = 16.14$$

Thus for stoichiometric combustion, 1 mole of fuel requires 59.66 moles of air and produces 64.16 moles of products. The stoichiometric $(A/F)_s$ is 15.14 and $(F/A)_s$ is 0.0661.

The molecular weights of the reactants M_R and products M_P are

$$M_R = \frac{1}{n}\sum n_i M_i = \frac{1}{60.66}(1 \times 114.15 + 59.66 \times 28.96)$$

$$M_P = \frac{1}{n}\sum n_i M_i = \frac{1}{64.16}(8 \times 44.01 + 9 \times 18.02 + 47.16 \times 28.16)$$

or $$M_R = 30.36 \qquad M_P = 28.71$$

Fuel-air mixtures with more than or less than the stoichiometric air requirement can be burned. With excess air or fuel-lean combustion, the extra air appears in the products in unchanged form. For example, the combustion of isooctane with 25 percent excess air, or 1.25 times the stoichiometric air requirement, gives

$$C_8H_{18} + 1.25 \times 12.5(O_2 + 3.773N_2) = 8CO_2 + 9H_2O + 3.13O_2 + 58.95N_2 \tag{3.7}$$

With less than the stoichiometric air requirement, i.e., with fuel-rich combustion, there is insufficient oxygen to oxidize fully the fuel C and H to CO_2 and H_2O. The products are a mixture of CO_2 and H_2O with carbon monoxide CO and hydrogen H_2 (as well as N_2). The product composition cannot be determined from an element balance alone and an additional assumption about the chemical composition of the product species must be made (see Secs. 4.2 and 4.9.2).

Because the composition of the combustion products is significantly different for fuel-lean and fuel-rich mixtures, and because the stoichiometric fuel/air ratio depends on fuel composition, the ratio of the actual fuel/air ratio to the stoichiometric ratio (or its inverse) is a more informative parameter for defining mixture composition. The *fuel/air equivalence ratio* ϕ,

$$\phi = \frac{(F/A)_{\text{actual}}}{(F/A)_s} \tag{3.8}$$

will be used throughout this text for this purpose. The inverse of ϕ, the *relative air/fuel ratio* λ,

$$\lambda = \phi^{-1} = \frac{(A/F)_{\text{actual}}}{(A/F)_s} \tag{3.9}$$

is also sometimes used.

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For fuel-lean mixtures: $\phi < 1, \lambda > 1$
For stoichiometric mixtures: $\phi = \lambda = 1$
For fuel-rich mixtures: $\phi > 1, \lambda < 1$

When the fuel contains oxygen (e.g., with alcohols), the procedure for determining the overall combustion equation is the same except that fuel oxygen is included in the oxygen balance between reactants and products. For methyl alcohol (methanol), CH_3OH, the stoichiometric combustion equation is

$$CH_3OH + 1.5(O_2 + 3.773N_2) = CO_2 + 2H_2O + 5.66N_2 \tag{3.10}$$

and $(A/F)_s = 6.47$. For ethyl alcohol (ethanol), C_2H_5OH, the stoichiometric combustion equation is

$$C_2H_5OH + 3(O_2 + 3.773N_2) = 2CO_2 + 3H_2O + 11.32N_2 \tag{3.11}$$

and $(A/F)_s = 9.00$.

If there are significant amounts of sulfur in the fuel, the appropriate oxidation product for determining the stoichiometric air and fuel proportions is sulfur dioxide, SO_2.

For hydrogen fuel, the stoichiometric equation is

$$H_2 + \tfrac{1}{2}(O_2 + 3.773N_2) = H_2O + 1.887N_2 \tag{3.12}$$

and the stoichiometric (A/F) ratio is 34.3.

Note that the composition of the products of combustion in Eqs. (3.7) and (3.10) to (3.12) may not occur in practice. At normal combustion temperatures significant dissociation of CO_2 and of H_2O occurs (see Sec. 3.7.1). Whether, at low temperatures, recombination brings the product composition to that indicated by these overall chemical equations depends on the rate of cooling of the product gases. More general relationships for the composition of unburned and burned gas mixtures are developed in Chap. 4.

The stoichiometric (A/F) and (F/A) ratios of common fuels and representative single hydrocarbon and other compounds are given in App. D along with other fuel data.

3.5 THE FIRST LAW OF THERMODYNAMICS AND COMBUSTION†

3.5.1 Energy and Enthalpy Balances

In a combustion process, fuel and oxidizer react to produce products of different composition. The actual path by which this transformation takes place is understood only for simple fuels such as hydrogen and methane. For fuels with more complicated structure, the details are not well defined. Nonetheless, the first law

† The approach used here follows that developed by Spalding and Cole.[7]

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of thermodynamics can be used to relate the end states of mixtures undergoing a combustion process; its application does not require that the details of the process be known.

The first law of thermodynamics relates changes in internal energy (or enthalpy) to heat and work transfer interactions. In applying the first law to a system whose chemical composition changes, care must be exercised in relating the reference states at which zero internal energy or enthalpy for each species or groups of species are assigned. We are not free, when chemical reactions occur, to choose independently the zero internal energy or enthalpy reference states of chemical substances transformed into each other by reaction.

Consider a system of mass m which changes its composition from reactants to products by chemical reaction as indicated in Fig. 3-4. Applying the first law to the system between its initial and final states gives

$$Q_{R-P} - W_{R-P} = U_P - U_R \tag{3.13}$$

Heat transfer Q_{R-P} and work transfer W_{R-P} due to normal force displacements may occur across the system boundary. The standard thermodynamic sign convention for each energy transfer interaction—positive for heat transfer *to* the system and positive for work transfer *from* the system—is used.

We will consider a series of special processes: first, a *constant volume* process where the initial and final temperatures are the same, T'. Then Eq. (3.13) becomes

$$Q_{R-P} = U'_P - U'_R = (\Delta U)_{V,T'} \tag{3.14}$$

The internal energy of the system has changed by an amount $(\Delta U)_{V,T'}$ which can be measured or calculated. Combustion processes are exothermic [i.e., Q_{R-P} and $(\Delta U)_{V,T'}$ are negative]; therefore the system's internal energy decreases. If Eq. (3.14) is expressed per mole of fuel, then $(\Delta U)_{V,T'}$ is known as the increase in

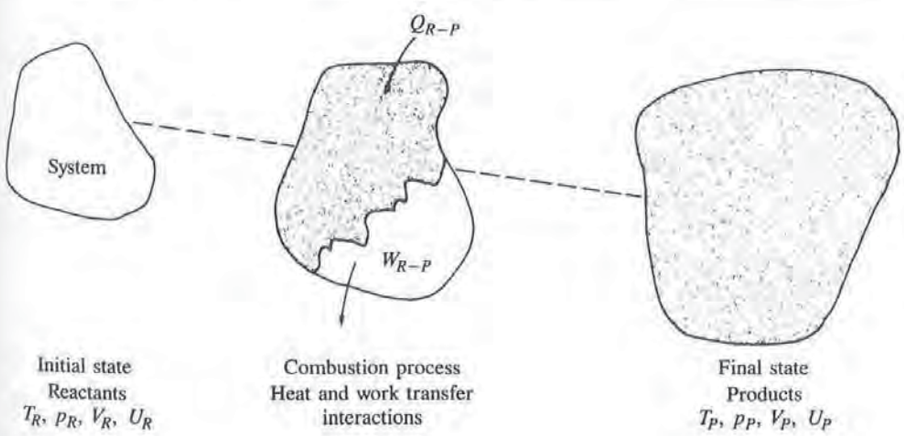


FIGURE 3-4
System changing from reactants to products for first law analysis.

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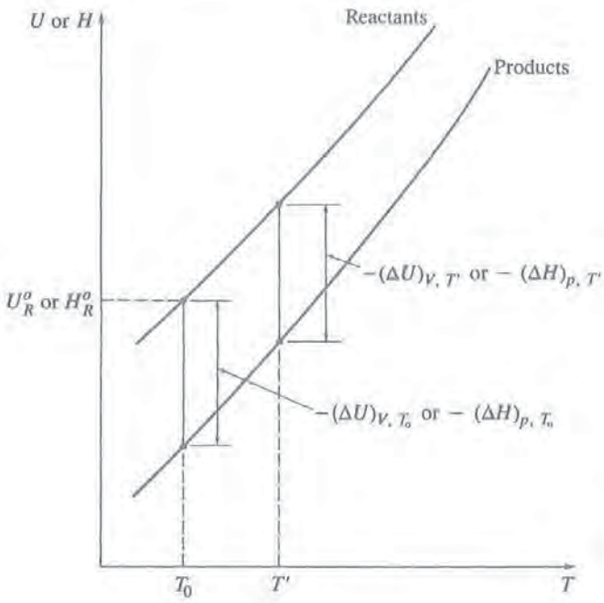


FIGURE 3-5
Schematic plot of internal energy (U) or enthalpy (H) of reactants and products as a function of temperature.

internal energy at constant volume, and $-(\Delta U)_{V,T'}$ is known as the *heat of reaction at constant volume* at temperature T'.

Next, consider a *constant pressure* process where the initial and final temperatures are the same, T'. For a constant pressure process

$$W_{R-P} = \int_R^P p\,dV = p(V_P - V_R) \tag{3.15}$$

so Eq. (3.13) becomes

$$Q_{R-P} - p(V'_P - V'_R) = U'_P - U'_R$$

or

$$\begin{aligned} Q_{R-P} &= (U'_P + pV'_P) - (U'_R + pV'_R) \\ &= H'_P - H'_R = (\Delta H)_{p,T'} \end{aligned} \tag{3.16}$$

The enthalpy of the system has changed by an amount $(\Delta H)_{p,T'}$, which can be measured or calculated. Again for combustion reactions, $(\Delta H)_{p,T'}$ is a negative quantity. If Eq. (3.16) is written per mole of fuel, $(\Delta H)_{p,T'}$ is called the increase in enthalpy at constant pressure and $-(\Delta H)_{p,T'}$ is called the *heat of reaction at constant pressure* at T'.

These processes can be displayed, respectively, on the internal energy or enthalpy versus temperature plot shown schematically in Fig. 3-5. If U (or H) for the reactants is arbitrarily assigned a value U_R° (or H_R°) at some reference temperature T_0, then the value of $(\Delta U)_{V,T_0}$[or $(\Delta H)_{p,T_0}$] fixes the relationship between $U(T)$ or $H(T)$, respectively, for the products and the reactants. Note that the slope of these lines (the specific heat at constant volume or pressure if the diagram is expressed per unit mass or per mole) increases with increasing tem-

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perature; also, the magnitude of $(\Delta U)_{V,\,T'}$ [or $(\Delta H)_{p,\,T'}$] decreases with increasing temperature because c_v (or c_p) for the products is greater than for the reactants.

The difference between $(\Delta H)_{p,\,T'}$ and $(\Delta U)_{V,\,T'}$ is

$$(\Delta H)_{p,\,T'} - (\Delta U)_{V,\,T'} = p(V_P - V_R) \tag{3.17}$$

Only if the volumes of the products and reactants in the constant pressure process are the same are $(\Delta H)_{p,\,T'}$ and $(\Delta U)_{V,\,T'}$ equal. If all the reactant and product species are ideal gases, then the ideal gas law Eq. (3.1) gives

$$(\Delta H)_{p,\,T'} - (\Delta U)_{V,\,T'} = \bar{R}(n'_P - n'_R)T' \tag{3.18}$$

Note that any inert gases do not contribute to $(n'_P - n'_R)$.

With a hydrocarbon fuel, one of the products, H_2O, can be in the gaseous or liquid phase. The internal energy (or enthalpy) of the products in the constant volume (or constant pressure) processes described above in Fig. 3-5 will depend on the relative proportions of the water in the gaseous and liquid phases. The limiting cases of all vapor and all liquid are shown in Fig. 3-6*a* for a U-T plot. The internal energy differences between the curves is

$$|(\Delta U)_{V,\,T',\,H_2O\ liq}| - |(\Delta U)_{V,\,T',\,H_2O\ vap}| = m_{H_2O}\,u'_{fg\,H_2O} \tag{3.19}$$

where m_{H_2O} is the mass of water in the products and $u'_{fg\,H_2O}$ is the internal energy of vaporization of water at the temperature and pressure of the products. Similar

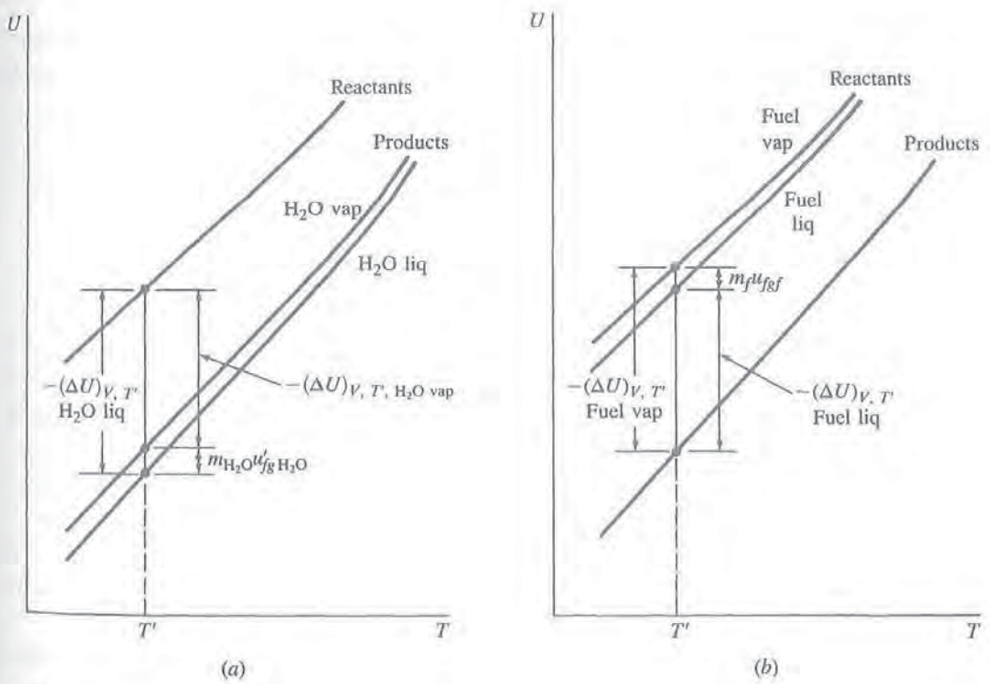


FIGURE 3-6
Schematic plots of internal energy of reactants and products as a function of temperature. (*a*) Effect of water in products as either vapor or liquid. (*b*) Effect of fuel in reactants as either vapor or liquid.

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curves and relationships apply for enthalpy:

$$|(\Delta H)_{p,\,T',\,H_2O\ liq}| - |(\Delta H)_{p,\,T',\,H_2O\ vap}| = m_{H_2O}\, h'_{fg\,H_2O} \tag{3.20}$$

For some fuels, the reactants may contain the fuel as either liquid or vapor. The U-T (or H-T) line for the reactants with the fuel as liquid or as vapor will be different, as indicated in Fig. 3-6*b*. The vertical distance between the two reactant curves is $m_f u_{fgf}$ (or $m_f h_{fgf}$) where the subscript f denotes fuel.

3.5.2 Enthalpies of Formation

For fuels which are single hydrocarbon compounds, or where the precise fuel composition is known, the internal energies or enthalpies of the reactants and the products can be related through the enthalpies of formation of the reactants and products.

The *enthalpy of formation* $\Delta\tilde{h}_f^\circ$ of a chemical compound is the enthalpy increase associated with the reaction of forming one mole of the given compound from its elements, with each substance in its thermodynamic standard state at the given temperature.

The *standard state* is the state at one atmosphere pressure and the temperature under consideration. We will denote the standard state by the superscript °.

Since thermodynamic calculations are made as a difference between an initial and a final state, it is necessary to select a *datum state* to which all other thermodynamic states can be referred. While a number of datum states have been used in the literature, the most common datum is 298.15 K (25°C) and 1 atmosphere. We will use this datum throughout this text. Elements at their *reference state* are arbitrarily assigned zero enthalpy at the datum temperature. The reference state of each element is its stable standard state; e.g., for oxygen at 298.15 K, the reference state is O_2 gas.

Enthalpies of formation are tabulated as a function of temperature for all commonly occurring species. For inorganic compounds, the *JANAF Thermochemical Tables* are the primary reference source.[8] These tables include values of the molar specific heat at constant pressure, standard entropy, standard Gibbs free energy (called free energy in the tables), standard enthalpy, enthalpy of formation and Gibbs free energy of formation, and $\log_{10}$ equilibrium constant for the formation of each species from its elements. Some primary references for thermodynamic data on fuel compounds are Maxwell,[9] Rossini *et al.*,[10] and Stull *et al.*[11] Enthalpies of formation of species relevant to hydrocarbon fuel combustion are tabulated in Table 3.2. Selected values of thermodynamic properties of relevant species are tabulated in App. D.

For a given combustion reaction, the enthalpy of the products at the standard state relative to the enthalpy datum is then given by

$$H_P^\circ = \sum_{\text{products}} n_i \Delta\tilde{h}_{f,i}^\circ \tag{3.21a}$$

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TABLE 3.2
Standard enthalpies of formation

Species	State†	$\Delta\tilde{h}_f^\circ$, MJ/kmol
O_2	Gas	0
N_2	Gas	0
H_2	Gas	0
C	Gas	0
CO_2	Gas	−393.52
H_2O	Gas	−241.83
H_2O	Liquid	−285.84
CO	Gas	−110.54
CH_4	Gas	−74.87
C_3H_8	Gas	−103.85
CH_3OH	Gas	−201.17
CH_3OH	Liquid	−238.58
C_8H_{18}	Gas	−208.45
C_8H_{18}	Liquid	−249.35

† At 298.15 K (25°C) and 1 atm.

and the enthalpy of the reactants is given by

$$H_R^\circ = \sum_{\text{reactants}} n_i \Delta\tilde{h}_{f,i}^\circ \tag{3.21b}$$

The enthalpy increase, $(\Delta H)_{p,\,T_0}$, is then obtained from the difference $(H_P^\circ - H_R^\circ)$. The internal energy increase can be obtained with Eq. (3.17).

Example 3.2. Calculate the enthalpy of the products and reactants, and the enthalpy increase and internal energy increase of the reaction, of a stoichiometric mixture of methane and oxygen at 298.15 K.

The stoichiometric reaction is

$$CH_4 + 2O_2 = CO_2 + 2H_2O$$

Thus, for H_2O gas, from Table 3.2 and Eq. (3.21*a*, *b*):

$$H_P^\circ = -393.52 + 2(-241.83) = -877.18 \text{ MJ/kmol } CH_4$$

For H_2O liquid:

$$H_P^\circ = -393.52 + 2(-285.84) = -965.20 \text{ MJ/kmol } CH_4$$

$$H_R^\circ = -74.87 \text{ MJ/kmol } CH_4$$

Hence for H_2O gas:

$$(\Delta H)_P^\circ = -877.18 + 74.87 = -802.31 \text{ MJ/kmol } CH_4$$

and for H_2O liquid:

$$(\Delta H)_P^\circ = -965.20 + 74.87 = -890.33 \text{ MJ/kmol } CH_4$$

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Use Eq. (3.18) to find $(\Delta U)^\circ_V$. With H_2O gas, the number of moles of reactants and products are equal, so

$$(\Delta U)^\circ_V = (\Delta H)^\circ_p = -802.3 \text{ MJ/kmol } CH_4$$

For H_2O liquid:

$$(\Delta U)^\circ_V = -890.33 - 8.3143 \times 10^{-3}(1 - 3)298.15 \text{ MJ/kmol } CH_4$$

$$(\Delta U)^\circ_V = -885.4 \text{ MJ/kmol } CH_4$$

Note that the presence of nitrogen in the mixture or oxygen in excess of the stoichiometric amount would not change any of these calculations.

3.5.3 Heating Values

For fuels where the precise fuel composition is not known, the enthalpy of the reactants cannot be determined from the enthalpies of formation of the reactant species. The *heating value* of the fuel is then measured directly.

The heating value Q_{HV} or calorific value of a fuel is the magnitude of the heat of reaction at constant pressure or at constant volume at a standard temperature [usually 25°C (77°F)] for the complete combustion of unit mass of fuel. Thus

$$Q_{HV_p} = -(\Delta H)_{p, T_0} \tag{3.22a}$$

and

$$Q_{HV_V} = -(\Delta U)_{V, T_0} \tag{3.22b}$$

Complete combustion means that all carbon is converted to CO_2, all hydrogen is converted to H_2O, and any sulfur present is converted to SO_2. The heating value is usually expressed in joules per kilogram or joules per kilomole of fuel (British thermal units per pound-mass or British thermal units per pound-mole). It is therefore unnecessary to specify how much oxidant was mixed with the fuel, though this must exceed the stoichiometric requirement. It is immaterial whether the oxidant is air or oxygen.

For fuels containing hydrogen, we have shown that whether the H_2O in the products is in the liquid or gaseous phase affects the value of the heat of reaction. The term *higher heating value* Q_{HHV} (or gross heating value) is used when the H_2O formed is all condensed to the liquid phase; the term *lower heating value* Q_{LHV} (or net heating value) is used when the H_2O formed is all in the vapor phase. The two heating values at constant pressure are related by

$$Q_{HHV_p} = Q_{LHV_p} + \left(\frac{m_{H_2O}}{m_f}\right) h_{fg\,H_2O} \tag{3.23}$$

where (m_{H_2O}/m_f) is the ratio of mass of H_2O produced to mass of fuel burned. A similar expression with $u_{fg\,H_2O}$ replacing $h_{fg\,H_2O}$ applies for the higher and lower heating value at constant volume.

The heating value at constant pressure is the more commonly used; often the qualification "at constant pressure" is omitted. The difference between the heating values at constant pressure and constant volume is small.

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Heating values† of fuels are measured in calorimeters. For gaseous fuels, it is most convenient and accurate to use a continuous-flow atmosphere pressure calorimeter. The entering fuel is saturated with water vapor and mixed with sufficient saturated air for complete combustion at the reference temperature. The mixture is burned in a burner and the combustion products cooled with water-cooled metal tube coils to close to the inlet temperature. The heat transferred to the cooling water is calculated from the measured water flow rate and water temperature rise. The heating value determined by this process is the higher heating value at constant pressure.

For liquid and solid fuels, it is more satisfactory to burn the fuel with oxygen under pressure at constant volume in a bomb calorimeter. A sample of the fuel is placed in the bomb calorimeter, which is a stainless-steel container immersed in cooling water at the standard temperature. Sufficient water is placed in the bomb to ensure that the water produced in the combustion process will condense. Oxygen at 30 atmospheres is admitted to the bomb. A length of firing cotton is suspended into the sample from an electrically heated wire filament to act as a source of ignition. When combustion is complete the temperature rise of the bomb and cooling water is measured. The heating value determined by this process is the higher heating value at constant volume.

The heating values of common fuels are tabulated with other fuel data in App. D. The following example illustrates how the enthalpy of a reactant mixture relative to the enthalpy datum we have defined can be determined from the measured heating value of the fuel.

Example 3.3. Liquid kerosene fuel of the lower heating value (determined in a bomb calorimeter) of 43.2 MJ/kg and average molar H/C ratio of 2 is mixed with the stoichiometric air requirement at 298.15 K. Calculate the enthalpy of the reactant mixture relative to the datum of zero enthalpy for C, O_2, N_2, and H_2 at 298.15 K.

The combustion equation per mole of C can be written

$$CH_2 + \tfrac{3}{2}(O_2 + 3.773N_2) = CO_2 + H_2O + 5.660N_2$$

or

$$14 \text{ kg fuel} + \left.\begin{matrix} 7.160 \text{ kmol} \\ 207.4 \text{ kg} \end{matrix}\right\} \text{air} \rightarrow \left.\begin{matrix} 7.66 \text{ kmol} \\ 221.4 \text{ kg} \end{matrix}\right\} \text{products}$$

where $M = 28.962$ for atmospheric air.

The heating value given is at constant volume, $-(\Delta U)^\circ_V$. $(\Delta H)^\circ_p$ is obtained from Eq. (3.18), noting that the fuel is in the liquid phase:

$$(\Delta H)^\circ_p = -43.2 + 8.3143 \times 10^{-3}\,(7.66 - 7.160) \times \frac{298.15}{14}$$

$$= -43.2 + 0.09 = -43.1 \text{ MJ/kg fuel}$$

† Standard methods for measuring heating values are defined by the American Society for Testing Materials.

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The enthalpy of the products per kilogram of mixture is found from the enthalpies of formation (with H_2O vapor):

$$h_P = \frac{1(-393.52) + 1(-241.83)}{221.4}$$

$$= 2.87 \text{ MJ/kg}$$

The enthalpy of the reactants per kilogram of mixture is then

$$h_R = h_P - (\Delta h)_p^\circ = 2.87 + \frac{43.1 \times 14}{221.4} = 5.59 \text{ MJ/kg}$$

3.5.4 Adiabatic Combustion Processes

We now use the relationships developed above to examine two other special processes important in engine analysis: constant-volume and constant-pressure adiabatic combustion. For an adiabatic constant-volume process, Eq. (3.13) becomes

$$U_P - U_R = 0 \tag{3.24}$$

when U_P and U_R are evaluated relative to the same datum (e.g., the enthalpies of C, O_2, N_2, and H_2 are zero at 298.15 K, the datum used throughout this text).

Frequently, however, the tables or graphs of internal energy or enthalpy for species and reactant or product mixtures which are available give internal energies or enthalpies relative to the species or mixture value at some reference temperature T_0, i.e., $U(T) - U(T_0)$ or $H(T) - H(T_0)$ are tabulated. Since

$$U_P(T_0) - U_R(T_0) = (\Delta U)_{V, T_0}$$

it follows from Eq. (3.24) that

$$[U_P(T) - U_P(T_0)] - [U_R(T) - U_R(T_0)] = -(\Delta U)_{V, T_0} \tag{3.25}$$

relates the product and reactant states. Figure 3-7 illustrates the adiabatic constant-volume combustion process on a U-T diagram. Given the initial state of the reactants (T_R, V) we can determine the final state of the products (T_P, V).

For an adiabatic constant-pressure combustion process, Eq. (3.13) gives

$$H_P - H_R = 0$$

which combines with Eq. (3.16) to give

$$[H_P(T) - H_P(T_0)] - [H_R(T) - H_R(T_0)] = -(\Delta H)_{p, T_0} \tag{3.26}$$

Figure 3-7 illustrates this process also. Given the initial reactant state (T_R, p) we can determine the final product state (T_P, p).

Note that while in Figs. 3-5, 3-6, and 3-7 we have shown U and H for the reactants and products to be functions of T only, in practice for the products at high temperature U and H will be functions of p and T. The analysis presented here is general; however, to determine the final state of the products in an adia-

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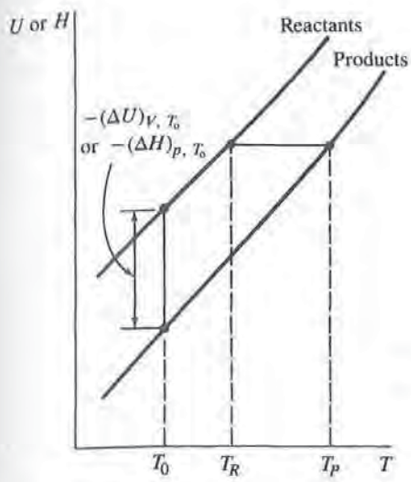


FIGURE 3-7
Adiabatic constant-volume combustion process on U-T diagram or adiabatic constant-pressure combustion process on H-T diagram.

batic combustion process, the constant-volume or constant-pressure constraint must also be used explicitly.

The final temperature of the products in an adiabatic combustion process is called the *adiabatic flame temperature*. Examples of typical adiabatic flame temperatures are shown later in Fig. 3-11.

3.5.5 Combustion Efficiency of an Internal Combustion Engine

In practice, the exhaust gas of an internal combustion engine contains incomplete combustion products (e.g., CO, H_2, unburned hydrocarbons, soot) as well as complete combustion products (CO_2 and H_2O) (see Sec. 4.9). Under lean operating conditions the amounts of incomplete combustion products are small. Under fuel-rich operating conditions these amounts become more substantial since there is insufficient oxygen to complete combustion. Because a fraction of the fuel's chemical energy is not fully released inside the engine during the combustion process, it is useful to define a *combustion efficiency*. The engine can be analyzed as an open system which exchanges heat and work with its surrounding environment (the atmosphere). Reactants (fuel and air) flow into the system; products (exhaust gases) flow out. Consider a mass m which passes through the control volume surrounding the engine shown in Fig. 3-8; the *net chemical energy release* due to combustion within the engine is given by

$$[H_R(T_A) - H_P(T_A)] = m\left(\sum_{i,\text{ reactants}} n_i \Delta\tilde{h}^\circ_{f,i} - \sum_{i,\text{ products}} n_i \Delta\tilde{h}^\circ_{f,i}\right)$$

Enthalpy is the appropriate property since $p_R = p_P = p_{\text{atm}}$. n_i is the number of moles of species i in the reactants or products per unit mass of working fluid and $\Delta\tilde{h}^\circ_{f,i}$ is the standard enthalpy of formation of species i at ambient temperature T_A.

The amount of fuel energy *supplied* to the control volume around the engine which can be released by combustion is $m_f Q_{\text{HV}}$. Hence, the *combustion*

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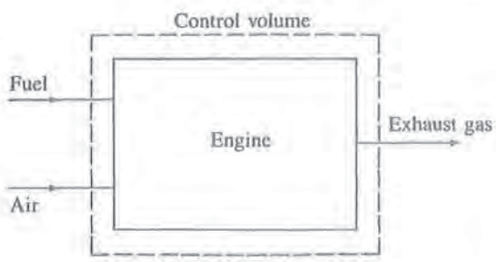


FIGURE 3-8
Control volume surrounding engine.

efficiency—the fraction of the fuel energy supplied which is released in the combustion process—is given by[12]

$$\eta_c = \frac{H_R(T_A) - H_P(T_A)}{m_f Q_{HV}} \tag{3.27}$$

Note that m and m_f could be replaced by the average mass flow rates $\dot{m}$ and $\dot{m}_f$.

Figure 3-9 shows how combustion efficiency varies with the fuel/air equivalence ratio for internal combustion engines. For spark-ignition engines, for lean equivalence ratios, the combustion efficiency is usually in the range 95 to 98 percent. For mixtures richer than stoichiometric, lack of oxygen prevents complete combustion of the fuel carbon and hydrogen, and the combustion efficiency steadily decreases as the mixture becomes richer. Combustion efficiency is little affected by other engine operating and design variables, provided the engine com-

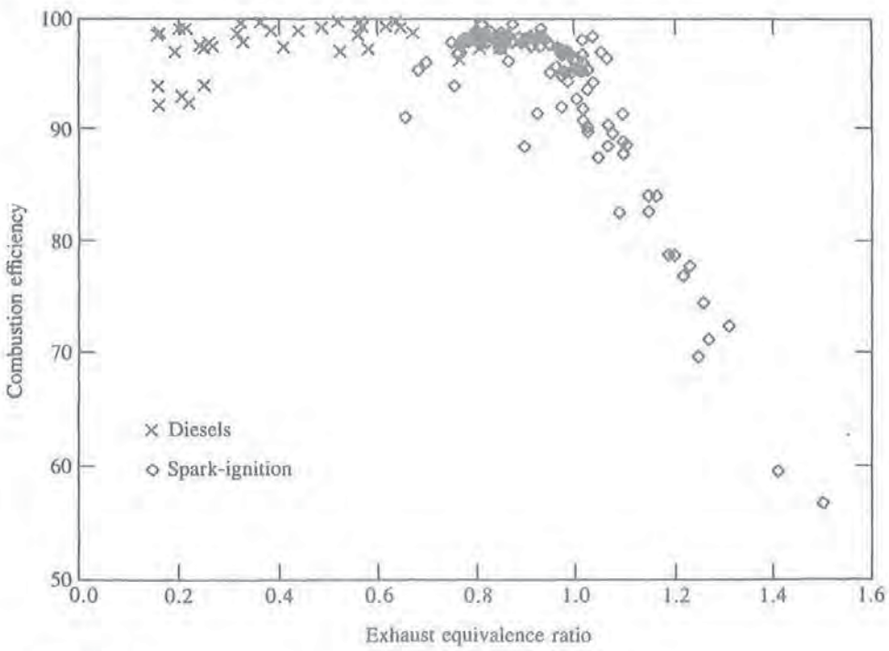


FIGURE 3-9
Variation of engine combustion efficiency with fuel/air equivalence ratio.

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bustion process remains stable. For diesel engines, which always operate lean, the combustion efficiency is normally higher—about 98 percent. Details of exhaust gas composition, on which these combustion efficiencies are based, can be found in Sec. 4.9.

3.6 THE SECOND LAW OF THERMODYNAMICS APPLIED TO COMBUSTION

3.6.1 Entropy

In App. B, it is shown how the entropy of a mixture of ideal gases of known composition can be calculated. The discussion earlier relating enthalpies (or internal energies) of reactant and product mixtures applies to entropy also. The standard state entropies of chemical species are tabulated in the JANAF tables relative to zero entropy at 0 K. If the entropies of the elements at a datum temperature are arbitrarily set equal to zero, then the values of the entropy of a reactant mixture of given composition and of the resulting product mixture of given composition are both determined.

3.6.2 Maximum Work from an Internal Combustion Engine and Efficiency

An internal combustion engine can be analyzed as an open system which exchanges heat and work with its surrounding environment (the atmosphere). Reactants (fuel and air) flow into the system; products (exhaust gases) flow out. By applying the second law of thermodynamics to a control volume surrounding the engine, as illustrated in Fig. 3-8, we can derive an expression for the maximum useful work that the engine can deliver.

Consider a mass m of fluid as it passes through the control volume surrounding the engine. The first law gives

$$\Delta Q - \Delta W_U = \Delta H$$

where ΔW_U is the useful work transfer (i.e., non-$p\,dV$ work) to the environment and $\Delta H = H_P - H_R$. Since the heat transfer ΔQ occurs only with the atmosphere at temperature T_A, from the second law

$$\frac{\Delta Q}{T_A} \leq \Delta S$$

These equations combine to give

$$\Delta W_U \leq -(\Delta H - T_A \Delta S) = -\Delta B$$

where B is the steady-flow availability function, $H - T_A S$.[13] Usually $p_R = p_A$ and $T_R = T_A$. The maximum work will be obtained when $p_P = p_A$ and $T_P = T_A$.

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TABLE 3.3
Enthalpies and free energies of combustion reactions

Reaction†	$\Delta\tilde{h}^\circ_{298}$, MJ/kmol	$\Delta\tilde{g}^\circ_{298}$, MJ/kmol
$C + O_2 \rightarrow CO_2$	−393.52	−394.40
$H_2 + \frac{1}{2}O_2 \rightarrow H_2O$	−240.91	−232.78
$CH_4 + 2O_2 \rightarrow CO_2 + 2H_2O$	−802.30	−800.76
$CH_4O(l) + \frac{3}{2}O_2 \rightarrow CO_2 + 2H_2O$	−638.59	−685.35
$C_3H_8(g) + 5O_2 \rightarrow 3CO_2 + 4H_2O$	−2044.0	−2074.1
$C_6H_6(l) + \frac{15}{2}O_2 \rightarrow 6CO_2 + 3H_2O$	−3135.2	−3175.1
$C_8H_{18}(l) + \frac{25}{2}O_2 \rightarrow 8CO_2 + 9H_2O$	−5074.6	−5219.9

† H_2O (gas) in products.

Under these conditions,

$$\Delta W_U \leq -[(H - TS)_{P_{T_A, p_A}} - (H - TS)_{R_{T_A, p_A}}] = -(\Delta G)_{T_A, p_A}$$

or
$$\Delta W_{U\,\max} = -(\Delta G)_{T_A, p_A} \tag{3.28}$$

G is the Gibbs free energy, $H - TS$, and $(\Delta G)_{T_A, p_A}$ is the Gibbs free energy increase in the reaction of the fuel-air mixture to products at atmospheric temperature and pressure. $-(\Delta G)_{T_A, p_A}$ will be a maximum when combustion is complete.

A fundamental measure of the effectiveness of any practical internal combustion engine is the ratio of the actual work delivered compared with this maximum work. This ratio will be called the *availability conversion efficiency* η_a:

$$\eta_a = \frac{\Delta W}{\Delta W_{U\,\max}} = -\frac{\Delta W}{(\Delta G)_{T_A, p_A}} \tag{3.29}$$

The property *availability* is the maximum useful work transfer that can be obtained from a system atmosphere (or control-volume atmosphere) combination at a given state. This efficiency therefore defines the fraction of the availability of the unburned fuel and air which, passing through the engine and interacting only with the atmosphere, is actually converted to useful work. Availability analysis of engine operation is proving valuable in identifying where the significant irreversibilities or losses in availability occur. This topic is discussed more fully in Sec. 5.7.

$(\Delta G)_{T_A, p_A}$, or $(\Delta g)_{T_A, p_A}$, is not an easy quantity to evaluate for practical fuels; it is the heating value, $-(\Delta h)_{T_A}$, which is usually measured. Values of $(\Delta g)^\circ_{298}$ and $(\Delta h)^\circ_{298}$ for selected fuel combustion reactions are given in Table 3.3. For the pure hydrocarbons they are closely comparable because at 298 K, $\Delta\tilde{s}^\circ \ll \Delta\tilde{h}^\circ/T$. For hydrogen and methanol the differences are larger, however. Because for practical fuels $-(\Delta h)^\circ_{298}$ is measured directly as the heating value of the fuel, it is standard practice to use the following definition of efficiency:

$$\eta_f = \frac{W_c}{m_f Q_{HV}} \tag{3.30}$$

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which was defined as the *fuel conversion efficiency* in Sec. 2.8. Note that sometimes the higher heating value is used in Eq. (3.30) and sometimes the lower heating value. Whichever value is used should be explicitly stated. The normal practice in internal combustion engine analysis is to use the lower heating value at constant pressure, since the engine overall is a steady flow device and the water in the exhaust is always in vapor form. We will use Q_{LHV_p} in Eq. (3.30) throughout this text. The fuel conversion efficiency is the most commonly used definition of engine efficiency because it uses an easily measured quantity, the heating value, to define the usable fuel energy supplied to the engine. For hydrocarbon fuels, since $\Delta\tilde{h}^\circ \approx \Delta\tilde{g}^\circ$, the fuel conversion efficiency and the availability conversion efficiency are closely comparable in value.

In practice, not all the fuel energy supplied to the engine is released by the combustion process since combustion is incomplete: the combustion efficiency [Eq. (3.27)] is less than unity. It is sometimes useful to separate out the effects of incomplete combustion by defining an efficiency which relates the actual work per cycle to the amount of fuel chemical energy released in the combustion process. We will call this the *thermal conversion efficiency* η_t:

$$\eta_t = \frac{W_c}{H_R(T_A) - H_P(T_A)} = -\frac{W_c}{(\Delta H)_{T_A}} = \frac{W_c}{\eta_c m_f Q_{\mathrm{HV}}} \tag{3.31}$$

Obviously the fuel conversion, thermal conversion, and combustion efficiencies are related by

$$\eta_f = \eta_c \eta_t \tag{3.32}$$

It is important to understand that there is a fundamental difference between availability conversion efficiency as defined by Eq. (3.29) [and the fuel conversion efficiency for internal combustion engines, Eq. (3.30), which closely approximates it] and the efficiency of a thermodynamic heat engine. The second law limit to the availability conversion efficiency is unity. For a thermodynamic heat engine (which experiences heat-transfer interactions with at least two heat reservoirs) the efficiency is limited to a value substantially less than unity by the temperatures of the heat reservoirs available.[13]

3.7 CHEMICALLY REACTING GAS MIXTURES

The working fluids in engines are mixtures of gases. Depending on the problem under consideration and the portion of the engine cycle in which it occurs chemical reactions may: (1) be so slow that they have a negligible effect on mixture composition (the mixture composition is essentially "frozen"); (2) be so rapid that the mixture state changes and the composition remains in chemical equilibrium; (3) be one of the rate-controlling processes that determine how the composition of the mixture changes with time.

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3.7.1 Chemical Equilibrium

It is a good approximation for performance estimates in engines to regard the burned gases produced by the combustion of fuel and air as in chemical equilibrium.† By this we mean that the chemical reactions, by which individual species in the burned gases react together, produce and remove each species at equal rates. No net change in species composition results.

For example, if the temperature of a mass of carbon dioxide gas in a vessel is increased sufficiently, some of the CO_2 molecules *dissociate* into CO and O_2 molecules. If the mixture of CO_2, CO, and O_2 is in equilibrium, then CO_2 molecules are dissociating into CO and O_2 at the same rate as CO and O_2 molecules are *recombining* in the proportions required to satisfy the equation

$$CO + \tfrac{1}{2}O_2 = CO_2$$

In combustion products of hydrocarbon fuels, the major species present at low temperatures are N_2, H_2O, CO_2, and O_2 or CO and H_2. At higher temperatures (greater than about 2200 K), these major species dissociate and react to form additional species in significant amounts. For example, the adiabatic combustion of a stoichiometric mixture of a typical hydrocarbon fuel with air produces products with species mole fractions of: $N_2 \sim 0.7$; H_2O, $CO_2 \sim 0.1$; CO, OH, O_2, NO, $H_2 \sim 0.01$; H, O ~ 0.001; and other species in lesser amounts.

The second law of thermodynamics defines the criterion for chemical equilibrium as follows. Consider a system of chemically reacting substances undergoing a constant-pressure, constant-temperature process. In the absence of shear work (and electrical work, gravity, motion, capillarity), the first law gives

$$\delta Q = dH$$

The second law gives

$$\delta Q \leq T\, dS$$

Combining these gives

$$dH - T\, dS \leq 0$$

Since we are considering constant-temperature processes, this equation holds for finite changes:

$$\Delta H - T\, \Delta S = \Delta G \leq 0$$

Thus, reactions can only occur (at constant pressure and temperature) if G $(= H - TS)$ for the products is less than G for the reactants. Hence at equilibrium

$$(\Delta G)_{p,\,T} = 0 \tag{3.33}$$

† This assumption is not valid late in the expansion stroke and during the exhaust process (see Sec. 4.9). Nor does it take account of pollutant formation processes (see Chap. 11).

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Consider a reactive mixture of ideal gases. The reactant species M_a, M_b, etc., and product species M_l, M_m, etc., are related by the general reaction whose stoichiometry is given by

$$\nu_a M_a + \nu_b M_b + \cdots = \nu_l M_l + \nu_m M_m + \cdots \tag{3.34a}$$

This can be written as

$$\sum_i \nu_i M_i = 0 \tag{3.34b}$$

where the ν_i are the stoichiometric coefficients and by convention are positive for the product species and negative for the reactant species.

Let an amount δn_a of M_a react with δn_b of M_b, etc., and produce δn_l of M_l, δn_m of M_m, etc. These amounts are in proportion:

$$\delta n_i = \nu_i \delta n \tag{3.35}$$

The change in Gibbs free energy of a mixture of ideal gases, at constant pressure and temperature, as the composition changes is given by

$$(\Delta G)_{p,\,T} = \sum_i \tilde{\mu}_i \delta n_i \tag{3.36}$$

where δn_i is the change in number of moles of species i and $\tilde{\mu}$ is the *chemical potential*. The chemical potential, an intensive property, is defined as

$$\tilde{\mu}_i = \left(\frac{\partial G}{\partial n_i}\right)_{p,\,T,\,n_j(j\neq i)} \tag{3.37}$$

It is equal in magnitude to the specific Gibbs free energy at a given temperature and pressure. For an ideal gas, it follows from Eqs. (B.13), (B.15) and (3.37) that

$$\tilde{\mu}_i = \tilde{\mu}_i^\circ(T) + \tilde{R}T \ln \frac{p_i}{p_0} \tag{3.38}$$

where $\tilde{\mu}_i^\circ$ equals $\tilde{g}_i^\circ$, the standard specific Gibbs free energy of formation. The standard state pressure p_0 is usually one atmosphere.

Substitution in Eq. (3.36) gives, at equilibrium,

$$\sum \left(\tilde{\mu}_i^\circ + \tilde{R}T \ln \frac{p_i}{p_0}\right)\delta n_i = 0$$

or

$$\sum \left(\tilde{\mu}_i^\circ + \tilde{R}T \ln \frac{p_i}{p_0}\right)\nu_i \delta n = 0$$

We can divide by δn and rearrange, to obtain

$$\sum \ln \left(\frac{p_i}{p_0}\right)^{\nu_i} = -\frac{(\sum \tilde{\mu}_i^\circ \nu_i)}{\tilde{R}T} = -\frac{\Delta G^\circ}{\tilde{R}T} = \ln K_p \tag{3.39}$$

K_p is the equilibrium constant at constant pressure:

$$K_p = \prod_i \left(\frac{p_i}{p_0}\right)^{\nu_i} \tag{3.40}$$

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It is obtained from the Gibbs free energy of the reaction which can be calculated from the Gibbs free energy of formation of each species in the reaction, as indicated in Eq. (3.39) above. It is a function of temperature only.

In the JANAF tables,[8] to simplify the calculation of equilibrium constants, values of $\log_{10}(K_p)_i$, the equilibrium constants of formation of one mole of each species from their elements in their standard states, are tabulated against temperature. The equilibrium constant for a specific reaction is then obtained via the relation

$$\log_{10}(K_p)_{\text{reaction}} = \sum_i \nu_i \log_{10}(K_p)_i \tag{3.41}$$

With the JANAF table values of $(K_p)_i$, the pressures in Eqs. (3.40) and (3.41) must be in atmospheres.

The effect of pressure on the equilibrium composition can be deduced from Eq. (3.40). Substitution of the mole fractions $\tilde{x}_i$ and mixture pressure p gives

$$\prod_i \left(\frac{p_i}{p_0}\right)^{\nu_i} = \prod_i \left(\tilde{x}_i \frac{p}{p_0}\right)^{\nu_i} = \left(\frac{p}{p_0}\right)^{\sum_i \nu_i} \prod_i \tilde{x}_i^{\nu_i} = K_p$$

If $\sum_i \nu_i = 0$, changes in pressure have no effect on the composition. If $\sum_i \nu_i > 0$ (dissociation reactions), then the mole fractions of the dissociation products decrease as pressure increases. If $\sum_i \nu_i < 0$ (recombination reactions), the converse is true.

An equilibrium constant, K_c, based on concentrations (usually expressed in gram moles per cubic centimeter) is also used:

$$K_c = \prod_i [\mathrm{M}_i]^{\nu_i} \tag{3.42}$$

Equation (3.40) can be used to relate K_p and K_c:

$$K_p = K_c(\tilde{R}T)^{\sum_i \nu_i} \tag{3.43}$$

for $p_0 = 1$ atmosphere. For $\sum_i \nu_i = 0$, K_p and K_c are equal.

Example 3.4. A stoichiometric mixture of CO and O_2 in a closed vessel, initially at 1 atm and 300 K, is exploded. Calculate the composition of the products of combustion at 2500 K and the gas pressure.

The combustion equation is

$$\mathrm{CO} + \tfrac{1}{2}\mathrm{O_2} = \mathrm{CO_2}$$

The JANAF tables give $\log_{10} K_p$ (equilibrium constants of formation from the elements in their standard state) at 2500 K of CO_2, CO, and O_2 as 8.280, 6.840, and 0, respectively. Thus, the equilibrium constant for the CO combustion reaction above is, from Eq. (3.41),

$$\log_{10} K_p = 8.280 - 6.840 = 1.440$$

which gives $K_p = 27.5$.

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If the degree of dissociation in the products is α (i.e., a fraction α of the CO_2 formed by complete combustion is dissociated), the product composition is

$$CO_2, (1-\alpha)\ ; \qquad CO, \alpha; \qquad O_2, \frac{\alpha}{2}$$

For this mixture, the number of moles of reactants n_R is $\frac{3}{2}$; the number of moles of products n_P is $(1 + \alpha/2)$.

The ideal gas law gives

$$p_R V = n_R \tilde{R} T_R \qquad p_P V = n_P \tilde{R} T_P$$

Thus

$$\frac{p_P}{n_P} = \frac{1}{1.5} \times \frac{2500}{300} = 5.555 \text{ atm/mol}$$

The equilibrium relation [Eq. (3.40)] gives

$$\frac{1-\alpha}{\alpha(\alpha/2)^{1/2}} \left(\frac{n_P}{p_P}\right)^{1/2} = 27.5$$

which can be solved to give $\alpha = 0.074$.

The composition of the products in mole fractions is, therefore,

$$x_{CO_2} = \frac{1-\alpha}{n_P} = 0.893$$

$$x_{CO} = \frac{\alpha}{n_P} = 0.071$$

$$x_{O_2} = \frac{\alpha/2}{n_P} = 0.037$$

The pressure of the product mixture is

$$p = 5.555 n_p = 5.76 \text{ atm}$$

Example 3.5. In fuel-rich combustion product mixtures, equilibrium between the species CO_2, H_2O, CO, and H_2 is often assumed to determine the burned gas composition. For $\phi = 1.2$, for C_8H_{18}–air combustion products, determine the mole fractions of the product species at 1700 K.

The reaction relating these species (often called the water gas reaction) is

$$CO_2 + H_2 = CO + H_2O$$

From the JANAF tables, $\log_{10} K_p$ of formation for these species at 1700 K are: CO_2, 12.180; H_2, 0; CO, 8.011; $H_2O(g)$, 4.699. The equilibrium constant for the above reaction is, from Eq. (3.41),

$$\log_{10} K_p = 8.011 + 4.699 - 12.180 = 0.530$$

from which $K_p = 3.388$.

The combustion reaction for C_8H_{18}–air with $\phi = 1.2$ can be written

$$C_8H_{18} + \frac{12.5}{1.2}(O_2 + 3.773N_2) \rightarrow aCO_2 + bH_2O + cCO + dH_2 + 39.30N_2$$

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A carbon balance gives: $a + c = 8$

A hydrogen balance gives: $2b + 2d = 18$

An oxygen balance gives: $2a + b + c = 20.83$

The equilibrium relation gives $(bc)/(ad) = 3.388$ (since the equilibrated reaction has the same number of moles as there are reactants or products, the moles of each species can be substituted for the partial pressures).

These four equations can be solved to obtain

$$c^2 - 19.3c + 47.3 = 0$$

which gives $c = 2.89$, $a = 5.12$, $b = 7.72$, and $d = 1.29$. The total number of moles of products is

$$a + b + c + d + 39.3 = 56.3$$

and the mole fractions of the species in the burned gas mixture are

CO_2, 0.0908; H_2O, 0.137; CO, 0.051; H_2, 0.023; N_2, 0.698

Our development of the equilibrium relationship for one reaction has placed no restrictions on the occurrence of simultaneous equilibria. Consider a mixture of N reacting gases in equilibrium. If there are C chemical elements, conservation of elements will provide C equations which relate the concentrations of these N species. Any set of $(N - C)$ chemical reactions, each in equilibrium, which includes each species at least once will then provide the additional equations required to determine the concentration of each species in the mixture. Unfortunately, this complete set of equations is a coupled set of C linear and $(N - C)$ nonlinear equations which is difficult to solve for cases where $(N - C) > 2$. For complex systems such as this, the following approach to equilibrium composition calculations is now more widely used.

Standardized computer methods for the calculation of complex chemical equilibrium compositions have been developed. A generally available and well-documented example is the NASA program of this type.[14] The approach taken is to minimize explicitly the Gibbs free energy of the reacting mixture (at constant temperature and pressure) subject to the constraints of element mass conservation. The basic equations for the NASA program are the following.

If the stoichiometric coefficients a_{ij} are the number of kilomoles of element i per kilomole of species j, b_i^* is the number of kilomoles of element i per kilogram of mixture, and n_j is the number of kilomoles of species j per kilogram of mixture, element mass balance constraints are

$$\sum_{j=1}^{n} a_{ij} n_j - b_i^* = 0 \qquad \text{for } i = 1, 2, \ldots, l \tag{3.44}$$

The Gibbs free energy per kilogram of mixture is

$$g = \sum_{j=1}^{n} \tilde{\mu}_j n_j \tag{3.45}$$

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For gases, the chemical potential $\tilde{\mu}_j$ is

$$\tilde{\mu}_j = \tilde{\mu}_j^\circ + \tilde{R}T \ln\left(\frac{n_j}{n}\right) + \tilde{R}T \ln p \tag{3.46}$$

where $\tilde{\mu}_j^\circ$ is the chemical potential in the standard state and p is the mixture pressure in atmospheres. Using the method of lagrangian multipliers, the term

$$G = g + \sum_{i=1}^{l} \lambda_i \sum_{j=1}^{n} (a_{ij} n_j - b_i^*)$$

is defined. The condition for equilibrium then becomes

$$\delta G = \sum_{j=1}^{n} \left(\mu_j + \sum_{i=1}^{l} \lambda_i a_{ij}\right)\delta n_j + \sum_{i=1}^{l} \sum_{j=1}^{n} (a_{ij} n_j - b_i^*)\delta\lambda_i = 0 \tag{3.47}$$

Treating the variations δn_j and $\delta\lambda_i$ as independent gives

$$\tilde{\mu}_j + \sum_{i=1}^{l} \lambda_i a_{ij} = 0 \qquad \text{for } j = 1, \ldots, n \tag{3.48}$$

and the original mass balance equation (3.44). Equations (3.44) and (3.48) permit the determination of equilibrium compositions for thermodynamic states specified by a temperature T and pressure p.

In the NASA program, the thermodynamic state may be specified by other pairs of state variables: enthalpy and pressure (useful for constant-pressure combustion processes); temperature and volume; internal energy and volume (useful for constant-volume combustion processes); entropy and pressure, and entropy and volume (useful for isentropic compressions and expansions). The equations required to obtain mixture composition are not all linear in the composition variables and an iteration procedure is generally required to obtain their solution. Once the composition is determined, additional relations, such as those in App. B which define the thermodynamic properties of gas mixtures, must then be used.

For each species, standard state enthalpies $\tilde{h}^\circ$ are obtained by combining standard enthalpies of formation at the datum temperature (298.15 K) $\Delta\tilde{h}_{f298}^\circ$ with sensible enthalpies $(\tilde{h}^\circ - \tilde{h}_{298}^\circ)$, i.e.,

$$\tilde{h}^\circ = \Delta\tilde{h}_{f298}^\circ + (\tilde{h}^\circ - \tilde{h}_{298}^\circ) \tag{3.49}$$

For the elements in their reference state, $\Delta\tilde{h}_{f298}^\circ$ is zero [the elements important in combustion are C (solid, graphite), $H_2(g)$, $O_2(g)$, $N_2(g)$].

For each species, the thermodynamic quantities specific heat, enthalpy, and entropy as functions of temperature are given in the form:

$$\frac{\tilde{c}_p^\circ}{\tilde{R}} = a_1 + a_2 T + a_3 T^2 + a_4 T^3 + a_5 T^4 \tag{3.50a}$$

$$\frac{\tilde{h}^\circ}{\tilde{R}T} = a_1 + \frac{a_2}{2} T + \frac{a_3}{3} T^2 + \frac{a_4}{4} T^3 + \frac{a_5}{5} T^4 + \frac{a_6}{T} \tag{3.50b}$$

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$$\frac{\tilde{s}^\circ}{\tilde{R}} = a_1 \ln T + a_2 T + \frac{a_3}{2} T^2 + \frac{a_4}{3} T^3 + \frac{a_5}{4} T^4 + a_7 \qquad (3.50c)$$

The coefficients are obtained by least-squares matching with thermodynamic property data from the JANAF tables. Usually two sets of coefficients are included for two adjacent temperature intervals (in the NASA program these are 300 to 1000 K and 1000 to 5000 K) (see Sec. 4.7).

In some equilibrium programs, the species to be included in the mixture must be specified as an input to the calculation. In the NASA program, all allowable species are included in the calculation, though species may be specifically omitted from consideration.

For each reactant composition and pair of thermodynamic state variables, the program calculates and prints out the following:

1. *Thermodynamic mixture properties* (obtained from the equilibrium composition and the appropriate gas mixture rule; see App. B). p, T, ρ, h, s, M, $(\partial \ln V/\partial \ln p)_T$, $(\partial \ln V/\partial \ln T)_p$, c_p, γ_s, and a (sound speed)
2. *Equilibrium composition.* Mole fractions of each species (which are present in significant amounts), $\tilde{x}_i$

Figure 3-10 shows how the equilibrium composition of the products of combustion of isooctane-air mixtures at selected temperatures and 30 atm pressure varies with the equivalence ratio. At low temperatures, the products are N_2, CO_2, H_2O, and O_2 for lean mixtures and N_2, CO_2, H_2O, CO, and H_2 for rich mixtures. As temperature increases, the burned-gas mixture composition becomes much more complex with dissociation products such as OH, O, and H becoming significant.

Figure 3-11 shows adiabatic flame temperatures for typical engine conditions as a function of the equivalence ratio, obtained with the NASA program using the methodology of Sec. 3.5.4. The isooctane-air unburned mixture state was 700 K and 10 atm. Flame temperatures for adiabatic combustion at constant pressure (where p_R and H_R are specified) and at constant volume (where V_R and U_R are specified) are shown. Flame temperatures at constant volume are higher, because the final pressure is higher and dissociation is less. Maximum flame temperatures occur slightly rich of stoichiometric.

3.7.2 Chemical Reaction Rates

Whether a system is in chemical equilibrium depends on whether the time constants of the controlling chemical reactions are short compared with time scales over which the system conditions (temperature and pressure) change. Chemical processes in engines are often not in equilibrium. Important examples of nonequilibrium phenomena are the flame reaction zone where the fuel is oxidized, and the air-pollutant formation mechanisms. Such nonequilibrium processes are controlled by the rates at which the actual chemical reactions which convert

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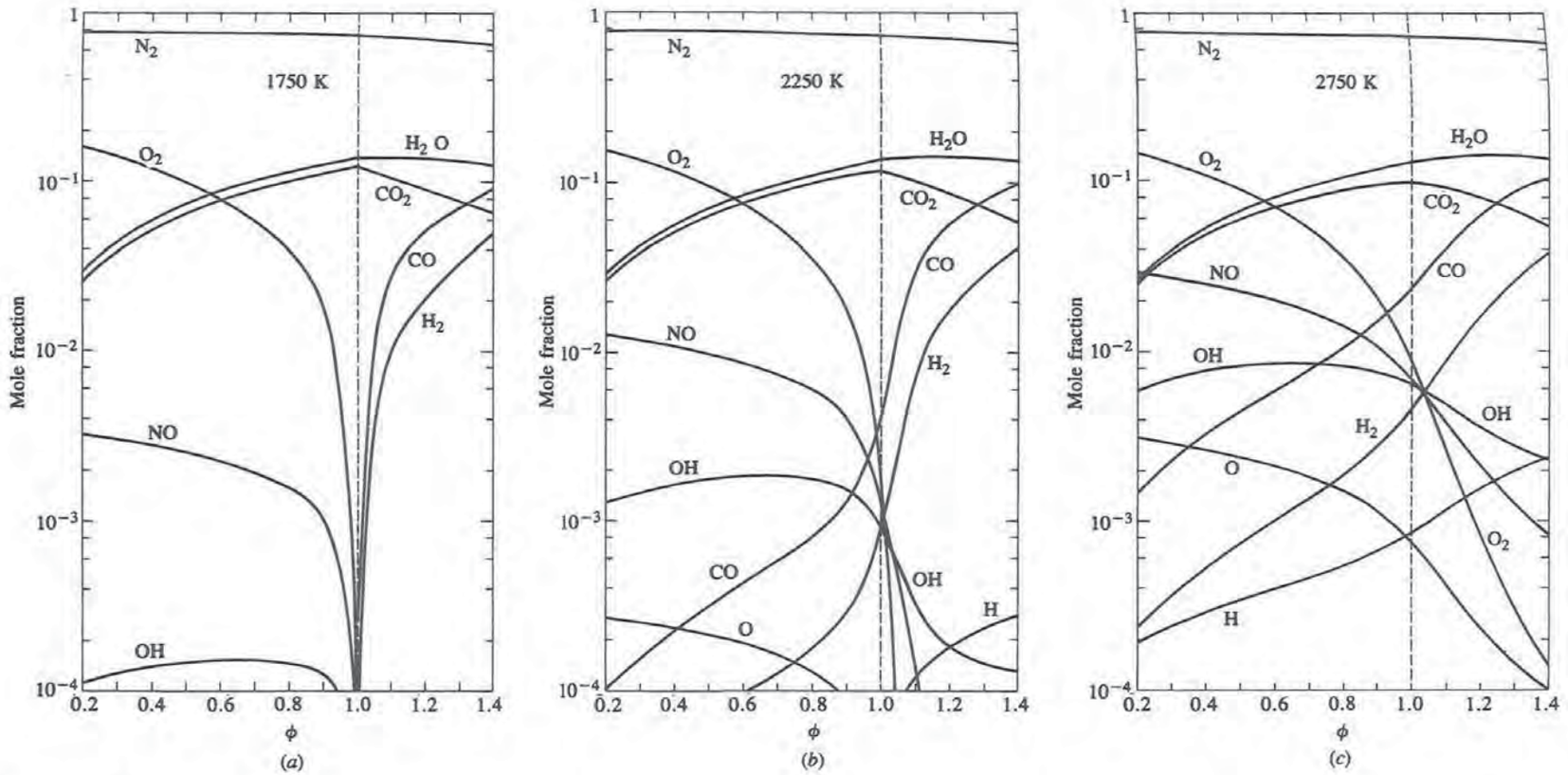


FIGURE 3-10
Mole fractions of equilibrium combustion products of isooctane-air mixtures as a function of fuel/air equivalence ratio at 30 atmospheres and (*a*) 1750 K; (*b*) 2250 K; and (*c*) 2750 K.

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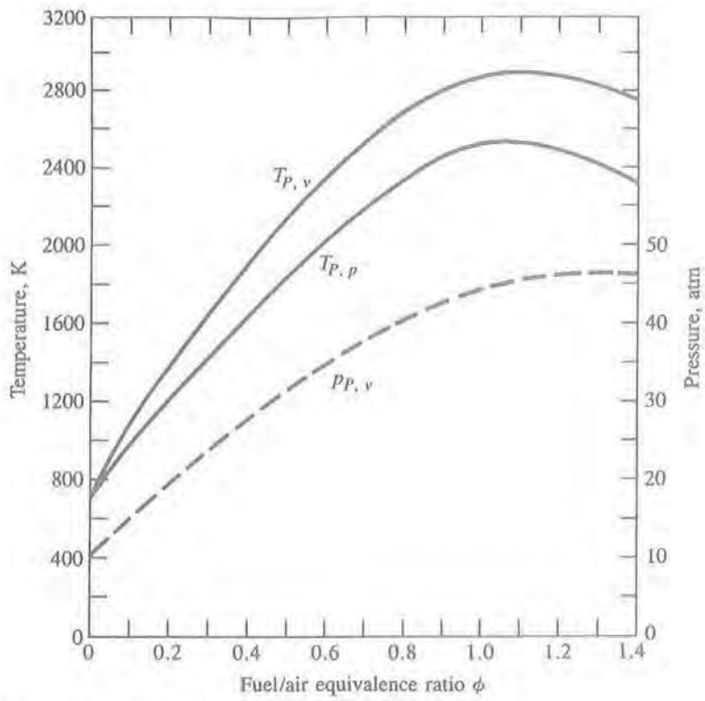


FIGURE 3-11
Equilibrium product temperatures for constant-volume ($T_{P,v}$) and constant-pressure ($T_{P,p}$) adiabatic combustion of isooctane-air mixture initially at 700 K and 10 atm, as a function of fuel/air equivalence ratio. Pressure ($p_{P,v}$) is equilibrium pressure for adiabatic constant-volume combustion.

reactants to products occur. The rates at which chemical reactions proceed depend on the concentration of the reactants, temperature, and whether any catalyst is present. This field is called chemical kinetics and some of its basic relations will now be reviewed.[2]

Most of the chemical reactions of interest in combustion are binary reactions, where two reactant molecules, M_a and M_b, with the capability of reacting together collide and form two product molecules, M_c and M_d; i.e.,

$$M_a + M_b = M_c + M_d \tag{3.51}$$

An important example of such a reaction is the rate-controlling step in the process by which the pollutant nitric oxide, NO, forms:

$$O + N_2 = NO + N$$

This is a second-order reaction since the stoichiometric coefficients of the reactants ν_a and ν_b are each unity and sum to 2. (The only first-order reactions are decomposition processes.) Third-order reactions are important in combustion, also. Examples are the recombination reactions by which radical species such as H, O, and OH combine during the final stage of the fuel oxidation process: e.g.,

$$H + H + M = H_2 + M^* \tag{3.52}$$

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M is any molecule (such as N_2) which takes part in the collision and carries away the excess energy.

The law of mass action states that the rate at which product species are produced and the rate at which reactant species are removed is proportional to the product of the concentrations of reactant species, with the concentration of each species raised to the power of its stoichiometric coefficient ν_i. Thus, for reaction (3.51) above, the reaction rate R^+ in the forward (+) direction, reactants to products, is given by

$$R^+ = -\frac{d[M_a]^+}{dt} = \frac{d[M_c]^+}{dt} = k^+[M_a][M_b] \tag{3.53}$$

If the reaction can also proceed in the reverse (−) direction, then the backward rate R^- is given by

$$R^- = -\frac{d[M_c]^-}{dt} = \frac{d[M_a]^-}{dt} = k^-[M_c][M_d] \tag{3.54}$$

k^+ and k^- are the rate constants in the forward and reverse directions for this reaction. The *net rate* of production of products or removal of reactants is, therefore,

$$\begin{aligned} R^+ - R^- &= \frac{d[M_c]^+}{dt} + \frac{d[M_c]^-}{dt} = -\frac{d[M_a]^+}{dt} - \frac{d[M_a]^-}{dt} \\ &= k^+[M_a][M_b] - k^-[M_c][M_d] \end{aligned} \tag{3.55}$$

These results can be stated more generally as follows. Any reaction can be written as

$$\sum_{i=1}^{n} \nu_{R_i} M_{R_i} = \sum_{i=1}^{m} \nu_{P_i} M_{P_i} \tag{3.56}$$

where ν_i is the stoichiometric coefficient of species M_i, subscripts R and P denote reactants and products, respectively, and there are n reactant species and m product species. The forward reaction rate R^+ and the reverse reaction rate R^- are given by

$$\begin{aligned} R^+ &= k^+ \prod_{i=1}^{n} [M_{R_i}]^{\nu_{Ri}} \\ R^- &= k^- \prod_{i=1}^{m} [M_{P_i}]^{\nu_{Pi}} \end{aligned} \tag{3.57}$$

The net rate of removal of reactant species M_{R_i} is

$$-\frac{d[M_{R_i}]}{dt} = \nu_{R_i}(R^+ - R^-) \tag{3.58a}$$

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and the net rate of production of product species M_{P_i} is

$$\frac{d[M_{P_i}]}{dt} = \nu_{P_i}(R^+ - R^-) \tag{3.58b}$$

The rate constants, k, usually follow the Arrhenius form:

$$k = A \exp\left(-\frac{E_A}{RT}\right) \tag{3.59}$$

where A is called the frequency or preexponential factor and may be a (moderate) function of temperature; E_A is the activation energy. The Boltzmann factor $\exp(-E_A/RT)$ defines the fraction of all collisions that have an energy greater than E_A—i.e., sufficient energy to make the reaction take place. The functional dependence of k on T and the constants in the Arrhenius form, Eq. (3.59), if that is appropriate, are determined experimentally.

At equilibrium, the forward and reverse reaction rates are equal. Then, from Eq. (3.55), with $R^+ - R^- = 0$:

$$\frac{k^+}{k^-} = \frac{[M_c][M_d]}{[M_a][M_b]} = K_c$$

where K_c is the equilibrium constant based on concentrations defined by Eq. (3.42). It can be related to K_p, the equilibrium constant based on partial pressures, by Eq. (3.43).

The chemical reaction mechanisms of importance in combustion are much more complex than the above illustrations of rate-controlled processes. Such mechanisms usually involve both parallel and sequential interdependent reactions. The methodology reviewed above still holds; however, one must sum algebraically the forward and reverse rates of all the reactions which produce (or remove) a species of interest. In such complex mechanisms it is often useful to assume that (some of) the reactive intermediate species or radicals are in *steady state*. That is, these radicals react so quickly once they are formed that their concentrations do not rise but are maintained in steady state with the species with which they react. The net rate at which their concentration changes with time is set equal to zero.

PROBLEMS

3.1. Isooctane is supplied to a four-cylinder spark-ignition engine at 2 g/s. Calculate the air flow rate for stoichiometric combustion. If the engine is operating at 1500 rev/min, estimate the mass of fuel and air entering each cylinder per cycle. The engine displaced volume is 2.4 liters. What is the volumetric efficiency?

3.2. Calculate the exhaust gas composition of a butane-fueled spark-ignition engine operating with equivalence ratio of 0.9. Assume the fuel is fully burned within the cylinder. Butane is C_4H_{10}.

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3.3. The molar composition of dry exhaust gas of a propane-fueled SI engine is given below (water was removed before the measurement). Calculate the equivalence ratio.

$$CO_2 = 10.8\%, \quad O_2 = 4.5\%, \quad CO = 0\%, \quad H_2 = 0\%$$

3.4. Evaluate and compare the lower heating values per unit mass of stoichiometric mixture and per unit volume of stoichiometric mixture (at standard atmospheric conditions) for methane, isooctane, methyl alcohol, and hydrogen. Assume the fuel is fully vaporized.

3.5. The measured engine fuel flow rate is 0.4 g/s, air flow rate is 5.6 g/s, and exhaust gas composition (measured dry) is $CO_2 = 13.0\%$, $CO = 2.8\%$ with O_2 essentially zero. Unburned hydrocarbon emissions can be neglected. Compare the equivalence ratio calculated from the fuel and air flow with the equivalence ratio calculated from exhaust gas composition. The fuel is gasoline with a H/C ratio of 1.87. Assume a H_2 concentration equal to one-third the CO concentration.

3.6. The brake fuel conversion efficiency of an engine is 0.3. The mechanical efficiency is 0.8. The combustion efficiency is 0.94. The heat losses to the coolant and oil are 60 kW. The fuel chemical energy entering the engine per unit time, $\dot{m}_f Q_{HV}$, is 190 kW. What percentage of this energy becomes (*a*) brake work; (*b*) friction work; (*c*) heat losses; (*d*) exhaust chemical energy; (*e*) exhaust sensible energy.

3.7. An upper estimate can be made of the amount of NO formed in an engine from considering the equilibrium of the reaction $N_2 + O_2 = 2NO$. Calculate the NO concentration at equilibrium at 2500 K and 30 atm. $\log_{10} K_p = -1.2$ for this reaction at 2500 K. Assume N/O ratio in the combustion products is 15. N_2, O_2, and NO are the only species present.

3.8. Carbon monoxide reacts with air at 1 atm and 1000 K in an exhaust gas reactor. The mole fractions of the exhaust gas-air mixture flowing into the reactor are CO, 3%; O_2, 7%; N_2, 74%; CO_2, 6%; H_2O, 10%.

(*a*) Calculate the concentration of CO and O_2 in gram moles per cm^3 in the entering mixture.

(*b*) The rate of reaction is given by

$$d[CO]/dt = -4.3 \times 10^{11} \times [CO][O_2]^{0.25} \exp[-E/(RT)]$$

[ ] denotes concentration in gram moles per cm^3, $E/R = 20{,}000$ K. Calculate the initial reaction rate of CO, $d[CO]/dt$: time is in seconds.

(*c*) The equilibrium constant K_p for the reaction $CO + \frac{1}{2}O_2 = CO_2$ at 1000 K is 10^{10}. Find the equilibrium CO concentration.

(*d*) Determine the time to reach this equilibrium concentration of CO using the initial reaction rate. (The actual time will be longer but this calculation indicates approximately the time required.)

3.9. The exhaust gases of a hydrogen-fueled engine contain 22.3 percent H_2O, 7.44 percent O_2, and 70.2 percent N_2. At what equivalence ratio is it operating?

3.10. Gas is sampled at 1 atmosphere pressure from the exhaust manifold of an internal combustion engine and analyzed. The mole fractions of species in the exhaust are:

$$H_2O, 0.0468; \quad CO_2, 0.0585; \quad O_2, 0.123; \quad N_2, 0.772$$

Other species such as CO and unburned hydrocarbons can be neglected.

(*a*) The fuel is a synthetic fuel derived from coal containing only carbon and hydrogen. What is the ratio of hydrogen atoms to carbon atoms in the fuel?

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(*b*) Calculate the fuel/air equivalence ratio at which this engine is operating.
(*c*) Is the internal combustion engine a conventional spark-ignition or a diesel engine? Explain.
(*d*) The engine has a displaced volume of 2 liters. Estimate approximately the percentage by which the fuel flow rate would be increased if this engine were operated at its maximum load at this same speed (2000 rev/min). Explain briefly what limits the equivalence ratio at maximum load.

3.11. The following are approximate values of the relative molecular mass (molecular weights): oxygen O_2, 32; nitrogen N_2, 28; hydrogen H_2, 2; carbon C, 12. Determine the stoichiometric fuel/air and air/fuel ratios on a mass basis, and the lower heating value per unit mass of stoichiometric mixture for the following fuels:

Methane (CH_4), isooctane (C_8H_{18}), benzene (C_6H_6), hydrogen (H_2), methyl alcohol (CH_3OH)

Heating values for these fuels are given in App. D.

3.12. Liquid petroleum gas (LPG) is used to fuel spark-ignition engines. A typical sample of the fuel consists of

70 percent by volume propane C_3H_8
5 percent by volume butane C_4H_{10}
25 percent by volume propene C_3H_6

The *higher heating values* of the fuels are: propane, 50.38 MJ/kg; butane, 49.56 MJ/kg; propylene (propene), 48.95 MJ/kg.
(*a*) Work out the overall combustion reaction for stoichiometric combustion of 1 mole of LPG with air, and the stoichiometric F/A and A/F.
(*b*) What are the higher and lower heating values for combustion of this fuel with excess air, per unit mass of LPG?

3.13. A spark-ignition engine is operated on isooctane fuel (C_8H_{18}). The exhaust gases are cooled, dried to remove water, and then analyzed for CO_2, CO, H_2, O_2. Using the overall combustion reaction for a range of equivalence ratios from 0.5 to 1.5, calculate the mole fractions of CO_2, CO, H_2, and O_2 in the dry exhaust gas, and plot the results as a function of equivalence ratio. Assume:
(*a*) that all the fuel is burnt inside the engine (almost true) and that the ratio of moles CO to moles H_2 in the exhaust is 3 : 1, and
(*b*) that there is no hydrogen in the exhaust for lean mixtures.
For high-power engine operation the air/fuel ratio is 14 : 1. What is the exhaust gas composition, in mole fractions, *before* the water is removed?

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CHAPTER 4

PROPERTIES OF WORKING FLUIDS

4.1 INTRODUCTION

The study of engine operation through an analysis of the processes that occur inside the engine has a long and productive history. The earliest attempts at this analysis used the constant-volume and constant-pressure ideal cycles as approximations to real engine processes (see Chap. 5). With the development of high-speed digital computers, the simulation of engine processes has become much more sophisticated and accurate (see Chap. 14). All these engine simulations (from the simplest to the most complex) require models for the composition and properties of the working fluids inside the engine, as well as models for the individual processes—induction, compression, combustion, expansion, and exhaust—that make up the engine operating cycle. This chapter deals with models for the working fluid composition, and thermodynamic and transport properties.

The composition of the working fluid, which changes during the engine operating cycle, is indicated in Table 4.1. The unburned mixture for a spark-ignition engine during intake and compression consists of air, fuel, and previously burned gases. It is, therefore, a mixture of N_2, O_2, CO_2, H_2O, CO, and H_2 for fuel-rich mixtures, and fuel (usually vapor). The composition of the unburned mixture does not change significantly during intake and compression. It is suffi-

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TABLE 4.1
Working fluid constituents

Process	Spark-ignition engine	Compression-ignition engine
Intake	Air Fuel† Recycled exhaust‡ Residual gas§	Air Recycled exhaust‡ Residual gas§
Compression	Air Fuel vapor Recycled exhaust Residual gas	Air Recycled exhaust Residual gas
Expansion	Combustion products (mixture of N_2, H_2O, CO_2, CO, H_2, O_2, NO, OH, O, H, ...)	Combustion products (mixture of N_2, H_2O, CO_2, CO, H_2, O_2, NO, OH, O, H, ...)
Exhaust	Combustion products [mainly N_2, CO_2, H_2O, and either O_2 ($\phi < 1$) or CO and H_2 ($\phi > 1$)]	Combustion products (mainly N_2, CO_2, H_2O, and O_2)

† Liquid and vapor in the intake; mainly vapor within the cylinder.
‡ Sometimes used to control NO_x emissions (see Secs. 11.2, 15.3.2, and 15.5.1).
§ Within the cylinder.

ciently accurate to assume the composition is *frozen*. For the compression-ignition engine, the unburned mixture prior to injection contains no fuel; it consists of air and previously burned gas.

The combustion products or burned mixture gases, during the combustion process and much of the expansion process, are close to *thermodynamic equilibrium*. The composition of such mixtures has already been discussed (Sec. 3.7.1). As these combustion products cool, recombination occurs as indicated in Fig. 3-10. Towards the end of the expansion process, the gas composition departs from the equilibrium composition; recombination can no longer occur fast enough to maintain the reacting mixture in equilibrium. During the exhaust process, reactions are sufficiently slow so that for calculating thermodynamic properties the composition can be regarded as *frozen*.

The models used for predicting the thermodynamic properties of unburned and burned mixtures can be grouped into the five categories listed in Table 4.2. The first category is only useful for illustrative purposes since the specific heats of unburned and burned mixtures are significantly different. While the specific heats of the working fluids increase with increasing temperature in the range of interest, a constant-specific-heat model can be matched to the thermodynamic data over a limited temperature range. This approach provides a simple *analytic* model which can be useful when moderate accuracy of prediction will suffice. The appropriateness of frozen and equilibrium assumptions has already been discussed above. Approximations to thermodynamic equilibrium calculations are useful because of

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TABLE 4.2
Categories of models for thermodynamic properties

	Unburned mixture	Burned mixture
1.	Single ideal gas throughout operating cycle with c_v (and hence c_p) constant	
2.	Ideal gas; $c_{v,u}$ constant	Ideal gas; $c_{v,b}$ constant
3.	Frozen mixture of ideal gases; $c_{v,i}(T)$	Frozen mixture of ideal gases; $c_{v,i}(T)$
4.	Frozen mixture of ideal gases; $c_{v,i}(T)$	Approximations fitted to equilibrium thermodynamic properties
5.	Frozen mixture of ideal gases; $c_{v,i}(T)$	Mixture of reacting ideal gases in thermodynamic equilibrium

Note: Subscripts i, u, and b denote species i in the gas mixture, the unburned mixture, and burned mixture properties, respectively.

the savings in computational time, relative to full equilibrium calculations, which can result from their use.

Values of thermodynamic properties of unburned and burned mixtures relevant to engine calculations are available from charts, tables, and algebraic relationships developed to match tabulated data. A selection of this material is included in this chapter and App. D. The references indicate additional sources.

4.2 UNBURNED MIXTURE COMPOSITION

The mass of charge trapped in the cylinder (m_c) is the inducted mass per cycle (m_i), plus the residual mass (m_r) left over from the previous cycle. The residual fraction (x_r) is

$$x_r = \frac{m_r}{m_c} \tag{4.1}$$

Typical residual fractions in spark-ignition engines range from 20 percent at light load to 7 percent at full load. In diesels, the residual fraction is smaller (a few percent) due to the higher compression ratio, and in naturally aspirated engines is approximately constant since the intake is unthrottled. If the inducted mixture is fuel and air (or air only), then the *burned gas fraction* (x_b) in the unburned mixture during compression equals the residual fraction.

In some engines, a fraction of the engine exhaust gases is recycled to the intake to dilute the fresh mixture for control of NO_x emissions (see Sec. 11.2). If

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the percent of exhaust gas recycled (%EGR) is defined as the percent of the total intake mixture which is recycled exhaust,†

$$\text{EGR}(\%) = \left(\frac{m_{\text{EGR}}}{m_i}\right) \times 100 \tag{4.2}$$

where m_{EGR} is the mass of exhaust gas recycled, then the burned gas fraction in the fresh mixture is

$$x_b = \frac{m_{\text{EGR}} + m_r}{m_c} = \left(\frac{\text{EGR}}{100}\right)(1 - x_r) + x_r \tag{4.3}$$

Up to about 30 percent of the exhaust can be recycled; the burned gas fraction during compression can, therefore, approach 30 to 40 percent.

The composition of the burned gas fraction in the unburned mixture can be calculated as follows. The combustion equation for a hydrocarbon fuel of average molar H/C ratio y [e.g., Eq. (3.5)] can be written per mole O_2 as

$$\varepsilon\phi C + 2(1-\varepsilon)\phi H_2 + O_2 + \psi N_2 \rightarrow n_{CO_2}CO_2 + n_{H_2O}H_2O + n_{CO}CO + n_{H_2}H_2 + n_{O_2}O_2 + n_{N_2}N_2 \tag{4.4}$$

where ψ = the molar N/O ratio (3.773 for air)

$$\varepsilon = \frac{4}{4 + y}$$

y = the molar H/C ratio of the fuel
ϕ = fuel/air equivalence ratio
n_i = moles of species i per mole O_2 reactant

The n_i are determined using the following assumptions:

1. For lean and stoichiometric mixtures ($\phi \leq 1$) CO and H_2 can be neglected.
2. For rich and stoichiometric mixtures ($\phi \geq 1$) O_2 can be neglected.
3. For rich mixtures, either (*a*) the water gas reaction

$$CO_2 + H_2 = CO + H_2O$$

---

† An alternative definition of percent EGR is also used based on the ratio of EGR to fresh mixture (fuel and air):

$$\text{EGR}^*(\%) = \left(\frac{m_{\text{EGR}}}{m_a + m_f}\right) \times 100$$

The two definitions are related by

$$\frac{\text{EGR}^*}{100} = \frac{\text{EGR}}{100 - \text{EGR}} \quad \text{and} \quad \frac{\text{EGR}}{100} = \frac{\text{EGR}^*}{100 + \text{EGR}^*}$$

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can be assumed to be in equilibrium with the equilibrium constant $K(T)$:

$$K(T) = \frac{n_{H_2O} n_{CO}}{n_{CO_2} n_{H_2}}$$

where $K(T)$ can be determined from a curve fit to JANAF table data:[8]

$$\ln K(T) = 2.743 - \frac{1.761 \times 10^3}{T} - \frac{1.611 \times 10^6}{T^2} + \frac{0.2803 \times 10^9}{T^3} \qquad (4.5)$$

where T is in K, or (*b*) K can be assumed constant over the normal engine operating range. A value of 3.5 is often assumed (see Sec. 4.9), which corresponds to evaluating the equilibrium constant at 1740 K.

The n_i obtained from an element balance and the above assumptions are shown in Table 4.3. The value of c is obtained by solving the quadratic:

$$(K-1)c^2 - c\{K[2(\phi - 1) + \varepsilon\phi] + 2(1 - \varepsilon\phi)\} + 2K\varepsilon\phi(\phi - 1) = 0 \qquad (4.6)$$

The mole fractions are given by

$$\tilde{x}_i = \frac{n_i}{n_b}$$

where $n_b = \sum_i n_i$ is given in the bottom line of Table 4.3.

While Eq. (4.4) is for a fuel containing C and H only, it can readily be modified for alcohols or alcohol-hydrocarbon blends. For a fuel of molar composition $CH_y O_z$, the reactant mixture

$$CH_y O_z + \frac{1}{\phi}\left(1 + \frac{y}{4} - \frac{z}{2}\right)(O_2 + \psi N_2)$$

can be rearranged per mole of O_2 reactant as

$$\zeta\phi\varepsilon C + 2\zeta\phi(1 - \varepsilon)H_2 + O_2 + \left(1 - \frac{\varepsilon z}{2}\right)\zeta\psi N_2 \qquad (4.7a)$$

TABLE 4.3
Burned gas composition under 1700 K

	n_i, moles/mole O_2 reactant	
Species	$\phi \le 1$	$\phi > 1$†
CO_2	$\varepsilon\phi$	$\varepsilon\phi - c$
H_2O	$2(1 - \varepsilon)\phi$	$2(1 - \varepsilon\phi) + c$
CO	0	c
H_2	0	$2(\phi - 1) - c$
O_2	$1 - \phi$	0
N_2	ψ	ψ
Sum: n_b	$(1 - \varepsilon)\phi + 1 + \psi$	$(2 - \varepsilon)\phi + \psi$

† c defined by Eq. (4.6).

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where

$$\zeta = \frac{2}{2 - \varepsilon z(1 - \phi)} \tag{4.7b}$$

If we write

$$\phi^* = \zeta\phi \qquad \text{and} \qquad \psi^* = \left(1 - \frac{\varepsilon z}{2}\right)\zeta\psi \tag{4.7c}$$

the reactant expression (4.7*a*) becomes

$$\phi^*\varepsilon C + 2\phi^*(1 - \varepsilon)H_2 + O_2 + \psi^* N_2$$

which is identical in form to the reactant expression for a hydrocarbon fuel (4.4). Thus Table 4.3 can still be used to give the composition of the burned gas residual fraction in the unburned mixture, except that ϕ^* replaces ϕ and ψ^* replaces ψ in the expressions for n_i.

Now consider the unburned mixture. The number of moles of fuel per mole O_2 in the mixture depends on the molecular weight of the fuel, M_f. If the average molecular formula of the fuel is $(CH_y)_\alpha$ then

$$M_f = \alpha(12 + y)$$

The fresh fuel-air mixture (not yet diluted with EGR or residual),

$$\varepsilon\phi C + 2(1 - \varepsilon)\phi H_2 + O_2 + \psi N_2$$

then becomes

$$\frac{4}{M_f}(1 + 2\varepsilon)\phi(CH_y)_\alpha + O_2 + \psi N_2$$

The unburned mixture (fuel, air, and a burned gas fraction), per mole O_2 in the mixture, can be written:

$$(1 - x_b)\left[\frac{4}{M_f}(1 + 2\varepsilon)\phi(CH_y)_\alpha + O_2 + \psi N_2\right] + x_b(n_{CO_2} + n_{H_2O} + n_{CO} + n_{H_2} + n_{O_2} + n_{N_2})$$

The number of moles of each species in the unburned mixture, per mole O_2, is summarized in Table 4.4. The mole fractions of each species are obtained by dividing by the total number of moles of unburned mixture n_u,

$$n_u = (1 - x_b)\left[\frac{4(1 + 2\varepsilon)\phi}{M_f} + 1 + \psi\right] + x_b n_b \tag{4.8}$$

where n_b is given in Table 4.3.

The molecular weights of the (low-temperature) burned and unburned

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TABLE 4.4
Unburned mixture composition

	n_i, moles/mole O_2 reactant	
Species	$\phi \leq 1$	$\phi > 1$
Fuel	$4(1 - x_b)(1 + 2\varepsilon)\phi/M_f$	
O_2	$1 - x_b\phi$	$1 - x_b$
N_2	ψ	ψ
CO_2	$x_b\varepsilon\phi$	$x_b(\varepsilon\phi - c)$
H_2O	$2x_b(1 - \varepsilon)\phi$	$x_b[2(1 - \varepsilon\phi) + c]$
CO	0	$x_b c$
H_2	0	$x_b[2(\phi - 1) - c]$
Sum†	n_u	n_u

† Given by Eq. (4.8).

mixture can now be determined. The mass of mixture (burned or unburned) per mole O_2 in the mixture, m_{RP}, is given by

$$m_{RP} = 32 + 4\phi(1 + 2\varepsilon) + 28.16\psi \tag{4.9}$$

The molecular weight of the burned mixture, M_b, is therefore

$$M_b = \frac{m_{RP}}{n_b} \tag{4.10}$$

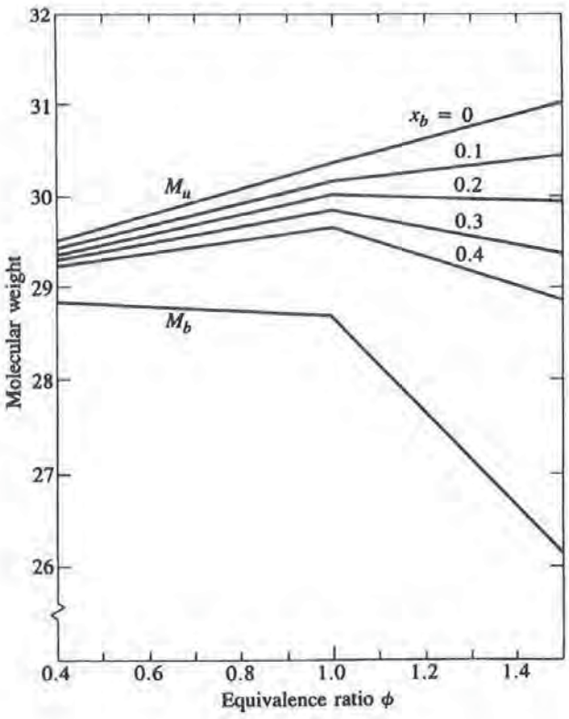


FIGURE 4-1
Molecular weight of unburned and low-temperature burned isooctane-air mixtures as a function of fuel/air equivalence ratio and burned gas fraction.

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TABLE 4.5
Factors for relating properties on molar and mass basis

Quantity, per mole O_2 in the mixture	General equation‡		Equation for C_8H_{18}-air mixtures
Moles of burned mixture n_b	$n_b = (1-\varepsilon)\phi + 1 + \psi,$ $n_b = (2-\varepsilon)\phi + \psi,$	$\phi \le 1$ $\phi > 1$	$n_b = 0.36\phi + 4.773$ $n_b = 1.36\phi + 3.773$
Moles of unburned mixture n_u	$(1-x_b)\left[\frac{4(1+2\varepsilon)\phi}{M_f} + 1 + \psi\right] + x_b n_b$	$\begin{cases}\phi \le 1\\ \phi > 1\end{cases}$	$n_u = 0.08\phi + 4.773 + 0.28x_b\phi$ $n_u = 0.08\phi + 4.773 + x_b(1.28\phi - 1)$
Mass of mixture† (burned or unburned)	$m_{RP} = 32 + 4\phi(1+2\varepsilon) + 28.16\psi$		$138.2 + 9.12\phi$
Mass of air†	$32 + 28.16\psi$		138.2

† Units: kg/kmol or lbm/lb · mol.

‡ For hydrocarbon fuels, ψ for air = 3.773; for fuels containing oxygen, ϕ^* and ψ^* given by Eq. (4.7c) are substituted for ϕ and ψ, respectively.

The molecular weight of the unburned mixture, M_u, is

$$M_u = \frac{m_{RP}}{n_u} \tag{4.11}$$

Figure 4-1 gives M_u and M_b for a range of ϕ and x_b for air, isooctane, burned gas mixtures.

Frequently, thermodynamic properties of unburned and burned mixtures are expressed per unit mass of *air* in the *original* mixture (for burned mixture this is the mixture before combustion). To obtain properties in these units, we need the mass of original air, per mole O_2 in the mixture, which is

$$(32 + 28.16\ \psi)$$

with units of kilograms per kilomole or pound-mass per pound-mole.

Table 4.5 summarizes the factors needed to relate properties expressed on a molar and a mass basis.

4.3 GAS PROPERTY RELATIONSHIPS

The individual species in the unburned and burned gas mixtures can with sufficient accuracy be modeled as ideal gases. Ideal gas relationships are reviewed in App. B. The most important relationships for property determination for engine calculations are summarized below.

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Since internal energy and enthalpy are functions of temperature only, the specific heats at constant volume and constant pressure are given by

$$c_v = \left(\frac{\partial u}{\partial T}\right)_v = \frac{du}{dT} = c_v(T) \tag{4.12a}$$

$$c_p = \left(\frac{\partial h}{\partial T}\right)_p = \frac{dh}{dT} = c_p(T) \tag{4.12b}$$

and

$$u - u_0 = \int_{T_0}^{T} c_v\, dT \tag{4.13a}$$

$$h - h_0 = \int_{T_0}^{T} c_p\, dT \tag{4.13b}$$

The entropy $s(T, v)$ or $s(T, p)$ is given by

$$s - s_0 = \int_{T_0}^{T} c_v \frac{dT}{T} + R \ln \frac{v}{v_0} \tag{4.14a}$$

$$s - s_0 = \int_{T_0}^{T} c_p \frac{dT}{T} - R \ln \frac{p}{p_0} \tag{4.14b}$$

The integrals in Eqs. (4.14*a*, *b*) are functions of temperature only, and are useful in evaluating entropy changes and in following isentropic processes. If we define

$$\Psi(T) = \int_{T_0}^{T} c_v(T) \frac{dT}{T} \tag{4.15a}$$

and

$$\Phi(T) = \int_{T_0}^{T} c_p(T) \frac{dT}{T} \tag{4.15b}$$

then

$$s - s_0 = \Psi + R \ln \left(\frac{v}{v_0}\right) \tag{4.16a}$$

$$s - s_0 = \Phi - R \ln \left(\frac{p}{p_0}\right) \tag{4.16b}$$

Thus, for example, the entropy change between states (T_1, p_1) and (T_2, p_2) is

$$s_2 - s_1 = \Phi_2 - \Phi_1 - R \ln \left(\frac{p_2}{p_1}\right) \tag{4.17}$$

For an isentropic process,

$$\ln \left(\frac{p_2}{p_1}\right) = \frac{\Phi_2 - \Phi_1}{R} \tag{4.18}$$

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In these equations, the units of u and h can be on a per unit mass or molar basis [i.e., joules per kilogram (British thermal units per pound-mass) or joules per kilomole (British thermal units per pound-mole)]; similarly, s, c_v, c_p, R, Ψ, and Φ can be in joules per kilogram-kelvin (British thermal units per pound-mass-degree Rankine) or joules per kilomole-kelvin (British thermal units per pound-mole-degree Rankine).

For gas mixtures, once the composition is known, mixture properties are determined either on a mass or molar basis from

$$u = \sum x_i u_i \qquad \tilde{u} = \sum \tilde{x}_i \tilde{u}_i \tag{4.19a}$$

$$h = \sum x_i h_i \qquad \tilde{h} = \sum \tilde{x}_i \tilde{h}_i \tag{4.19b}$$

$$s = \sum x_i s_i \qquad \tilde{s} = \sum \tilde{x}_i \tilde{s}_i \tag{4.19c}$$

and

$$c_v = \sum x_i c_{v,i} \qquad \tilde{c}_v = \sum \tilde{x}_i \tilde{c}_{v,i} \tag{4.20a}$$

$$c_p = \sum x_i c_{p,i} \qquad \tilde{c}_p = \sum \tilde{x}_i \tilde{c}_{p,i} \tag{4.20b}$$

4.4 A SIMPLE ANALYTIC IDEAL GAS MODEL

While the first category of model listed in Table 4.2 is too inaccurate for other than illustrative purposes, the second category—constant but different specific heats for the unburned and burned gas mixtures—can with careful choice of specific heat values be made much more precise. The advantages of a simple analytic model may be important for certain problems.

Figure 4-2 shows an internal energy versus temperature plot for a stoichiometric mixture. It is a quantitative version of Fig. 3-5. The unburned mixture line is for a burned gas fraction of 0.1. The fuel is isooctane. Data to construct such graphs can be obtained from charts or tables or computer programs (see Secs. 4.5 to 4.7). The units for u are kilojoules per kilogram of air in the original mixture (the units of the charts in Sec. 4.5). The datum is zero enthalpy for O_2, N_2, H_2, and C (solid) at 298 K. Note that the specific heats of the unburned and burned mixtures (the slopes of the lines in Fig. 4-2) are a function of temperature; at high temperatures, the internal energy of the burned mixture is a function of temperature and pressure.

However, the temperature range of interest for the unburned mixture in an SI engine is 400 to 900 K (700 to 1600°R); for the burned gas mixture, the extreme end states are approximately 2800 K, 35 atm (5000°R, 500 lb/in^2 abs) and 1200 K, 2 atm (2200°R, 30 lb/in^2 abs). Linear approximations to the unburned and burned mixture curves which minimize the error in u over the temperature (and pressure) ranges of interest are shown as dashed lines. The error in T for a given u is less than 50 K.

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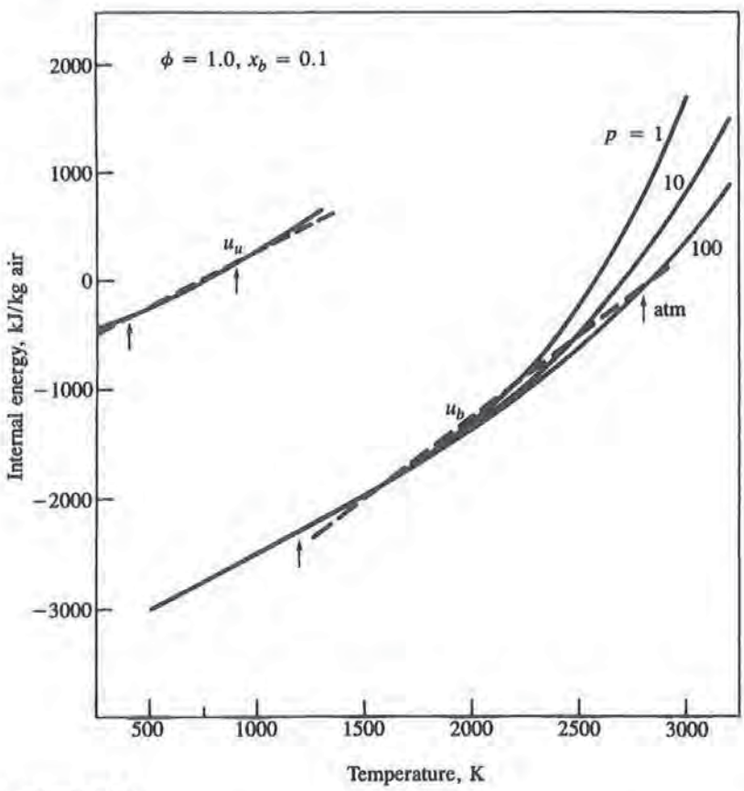


FIGURE 4-2
Internal energy versus temperature plot for stoichiometric unburned and burned gas mixtures: iso-octane fuel; unburned residual fraction 0.1.

The basis for this ideal gas model is

$$u_u = c_{v,u} T_u + h_{f,u} \qquad h_u = c_{p,u} T_u + h_{f,u} \tag{4.21a, b}$$

$$u_b = c_{v,b} T_b + h_{f,b} \qquad h_b = c_{p,b} T_b + h_{f,b} \tag{4.22a, b}$$

where $h_{f,u}$ and $h_{f,b}$ are the enthalpies of formation of unburned and burned gas mixture, respectively, at 0 K.

Then, for a constant-volume adiabatic combustion process,

$$u_u = u_b$$

or

$$c_{v,u} T_u + h_{f,u} = c_{v,b} T_b + h_{f,b}$$

If we solve for T_b and use the relations $(R_b/R_u) = (M_u/M_b)$ and $c_v/R = 1/(\gamma - 1)$, we obtain

$$T_b = (\gamma_b - 1)\left(\frac{M_b}{M_u}\right)\left(\frac{T_u}{\gamma_u - 1} + \frac{\Delta h_f}{R_u}\right) \tag{4.23}$$

where $\Delta h_f = h_{f,u} - h_{f,b}$.

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For a constant-pressure adiabatic combustion process,

$$h_u = h_b$$

and it can similarly be shown that

$$T_b = \frac{\gamma_b - 1}{\gamma_b}\left(\frac{M_b}{M_u}\right)\left(\frac{\gamma_u}{\gamma_u - 1}\,T_u + \frac{\Delta h_f}{R_u}\right) \tag{4.24}$$

To use the model, suitable values of γ_u, γ_b, M_u, (M_b/M_u), and $\Delta h_f/R_u$ must be determined. Values for M_u and M_b can be obtained from Eqs. (4.10) and (4.11).† Values of γ_u, γ_b, and $\Delta h_f/R_u$ can be obtained from graphs such as Fig. 4-2 (see Example 4.1 below). Values for γ_u, γ_b, and $\Delta h_f/R_u$ are available in the literature (e.g., Refs. 1 and 2) for a range of ϕ and x_b. However, values used for computations should always be checked over the temperature range of interest, to ensure that the particular linear fit to $u(T)$ used is appropriate.

Example 4.1. Determine the values of γ_u, γ_b, and $\Delta h_f/R_u$ which correspond to the straight-line fits for $u_u(T)$ and $u_b(T)$ in Fig. 4-2.

Equations for the straight lines in Fig. 4-2 are

$$u_u \text{ (kJ/kg air)} = 0.96T(\text{K}) - 700$$

and

$$u_b \text{ (kJ/kg air)} = 1.5T(\text{K}) - 4250$$

From Table 4.5, for isooctane fuel with $\phi = 1.0$ and $x_b = 0.1$, the number of moles of unburned mixture per mole O_2 in the mixture is

$$n_u = 0.08 \times 1 + 4.773 + 0.28 \times 0.1 \times 1 = 4.881$$

The mass of air per mole O_2 in the mixture is 138.2. Thus, the number of moles of unburned mixture per unit mass of air in the original mixture is

$$\frac{4.881}{138.2} = 0.0353$$

The molar specific heat of the unburned mixture $\tilde{c}_{v,u}$ is therefore

$$\tilde{c}_{v,u} = \frac{0.96}{0.0353} = 27.2 \text{ kJ/kmol} \cdot \text{K}$$

Since $\tilde{R} = 8.314$ kJ/kmol · K,

$$\gamma_u = \frac{27.2 + 8.314}{27.2} = 1.31$$

The number of moles of burned mixture per mole O_2 is (from Table 4.5)

$$n_b = 0.36 \times 1 + 4.773 = 5.133$$

† The error in ignoring the effect of dissociation on M_b is small.

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The number of moles of burned mixture per unit mass of air in the original mixture is

$$\frac{5.133}{138.2} = 0.0371$$

The molar specific heat $\tilde{c}_{v,b}$ is therefore

$$\tilde{c}_{v,b} = \frac{1.5}{0.0371} = 40.4 \text{ kJ/kmol} \cdot \text{K}$$

and γ_b is

$$\gamma_b = \frac{40.4 + 8.314}{40.4} = 1.21$$

To find $\Delta h_f / R_u$, R_u is given by

$$R_u = 8.314 \times 0.0353 = 0.293 \text{ kJ/kg air} \cdot \text{K}$$

and so

$$\frac{\Delta h_f}{R_u} = \frac{(-700) - (-4250)}{0.293} = 1.2 \times 10^4 \text{ K}$$

4.5 THERMODYNAMIC CHARTS

One method of presenting thermodynamic properties of unburned and burned gas mixtures for internal combustion engine calculations is on charts. Two sets of charts are in common use: those developed by Hottel *et al.*[3] and those developed by Newhall and Starkman.[4, 5] Both these sets of charts use U.S. units. We have developed a new set of charts in SI units, following the approach of Newhall and Starkman. Charts are no longer used extensively for engine cycle calculations; computer models for the thermodynamic properties of working fluids have replaced the charts. Nonetheless, charts are useful for illustrative purposes, and afford an easy and accurate method where a limited number of calculations are required. The charts presented below are for isooctane fuel, and the following equivalence ratios: $\phi = 0.4, 0.6, 0.8, 1.0, 1.2$.

4.5.1 Unburned Mixture Charts

The thermodynamic properties of each unburned fuel-air mixture are represented by two charts. The first chart is designed to relate the mixture temperature, pressure, and volume at the beginning and at the end of the compression process; the second gives the mixture internal energy and enthalpy as functions of temperature.

The following assumptions are made:

1. The compression process is reversible and adiabatic.

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TABLE 4.6
Unburned mixture composition for charts

Equivalence ratio ϕ	(F/A)	Kilograms of mixture per kilogram of air	Moles of mixture per mole of O_2	Kilomole of mixture per kilogram of air	$n_u \bar{R}$,† J/kg air · K
0.4	0.0264	1.0264	$4.805 + 0.112x_b$	$0.0348 + 0.00081x_b$	289
0.6	0.0396	1.0396	$4.821 + 0.168x_b$	$0.0349 + 0.00122x_b$	290
0.8	0.0528	1.0528	$4.837 + 0.224x_b$	$0.0350 + 0.00162x_b$	291
1.0	0.0661	1.0661	$4.853 + 0.280x_b$	$0.0351 + 0.00203x_b$	292
1.2	0.0792	1.0792	$4.869 + 0.536x_b$	$0.0352 + 0.00388x_b$	292

† For $x_b = 0$. Error in neglecting x_b is usually small.

2. The fuel is in the vapor phase.
3. The mixture composition is homogeneous and frozen (no reactions between the fuel and air).
4. Each species in the mixture can be modeled as an ideal gas.
5. The burned gas fraction is zero.†

It proves convenient to assign zero internal energy or enthalpy to the unburned mixture at 298.15 K. Internal energy and enthalpies relative to this datum are called *sensible internal energy* u_s *or sensible enthalpy* h_s. By sensible we mean changes in u or h which result from changes in temperature alone, and we exclude changes due to chemical reaction or phase change.

Table 4.6 provides the basic composition data for the unburned mixture charts. Equations (4.13*a*, *b*) provide the basis for obtaining the $u_{s,u}(T)$ and $h_{s,u}(T)$ curves shown in Fig. 4-3.

Equations (4.15) and (4.16) provide the basis for following a reversible adiabatic (i.e., isentropic) compression process. Between end states 1 and 2, we obtain, per kilogram of air in the mixture,

$$\Psi(T_2) = \Psi(T_1) - n_u \bar{R} \ln\left(\frac{v_2}{v_1}\right) \tag{4.25a}$$

$$\Phi(T_2) = \Phi(T_1) + n_u \bar{R} \ln\left(\frac{p_2}{p_1}\right) \tag{4.25b}$$

where n_u is the number of moles of unburned mixture per kilogram of air. Values

† This assumption introduces negligible error into calculations of the compression process for mixtures with normal burned gas fractions, since the major constituent of the residual is N_2. The burned gas fraction must, however, be included when the unburned mixture properties are related to burned mixture properties in a combustion process.

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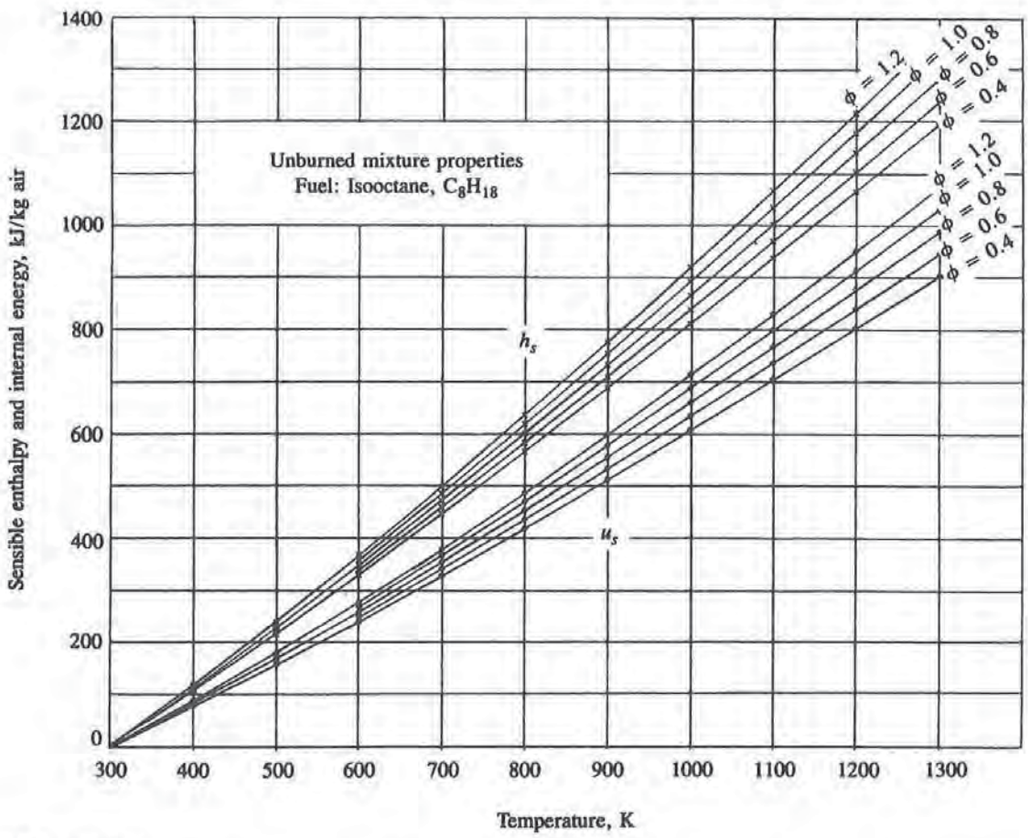


FIGURE 4-3
Sensible enthalpy and internal energy of unburned isooctane-air mixtures as function of temperature. Units: kJ/kg air in mixture.

of n_u and $n_u \tilde{R}$ are given in Table 4.6. $\Psi(T)$ and $\Phi(T)$ are given in Fig. 4-4. Note that v, p, and T are related by

$$p(\text{Pa})v(\text{m}^3/\text{kg air}) = n_u\tilde{R}(\text{J/kg air} \cdot \text{K})T(\text{K}) \tag{4.26}$$

Example 4.2. The compression process in an internal combustion engine can be modeled approximately as adiabatic and reversible (i.e., as an isentropic process). A spark-ignition engine with a compression ratio of 8 operates with a stochiometric fuel vapor–air mixture which is at 350 K and 1 atm at the start of the compression stroke. Find the temperature, pressure, and volume per unit mass of air at the end of the compression stroke. Calculate the compression stroke work.

Given $T_1 = 350$ K at the start of compression, find T_2 at the end of compression using the isentropic compression chart, Fig. 4-4, and Eq. (4.25*a*). For $T_1 = 350$ K, $\Psi_1 = 150$ J/kg air · K. From Eq. (4.25*a*),

$$\Psi_2(T_2) = \Psi_1(T_1) - n_u\tilde{R} \ln\left(\frac{v_2}{v_1}\right) = 150 - 292 \ln\left(\frac{1}{8}\right) = 757 \text{ J/kg air} \cdot \text{K}$$

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Figure 4-4 then gives

$$T_2 = 682 \text{ K}$$

The ideal gas law [Eq. (4.26)] gives

$$v_1 = \frac{292 \times 350}{1 \times 1.013 \times 10^5} = 1.0 \text{ m}^3/\text{kg air}$$

and

$$p_2 = p_1\left(\frac{T_2}{T_1}\right)\left(\frac{v_1}{v_2}\right) = \frac{682}{350} \times 8 = 15.5 \text{ atm}$$

$$v_2 = \frac{1.0}{8} = 0.125 \text{ m}^3/\text{kg air}$$

Note that p_2 can also be obtained from Fig. 4-4 and Eq. (4.25*b*):

$$\ln\left(\frac{p_2}{p_1}\right) = \frac{\Phi_2 - \Phi_1}{n_u \tilde{R}} = \frac{980 - 180}{292} = 2.74$$

$$p_2 = 15.5 \text{ atm} = 1.57 \text{ MPa}$$

The compression stroke work, assuming the process is adiabatic and using the data in Fig. 4-3, is

$$-W_{1-2} = u_s(T_2) - u_s(T_1) = 350 - 40 = 310 \text{ kJ/kg air}$$

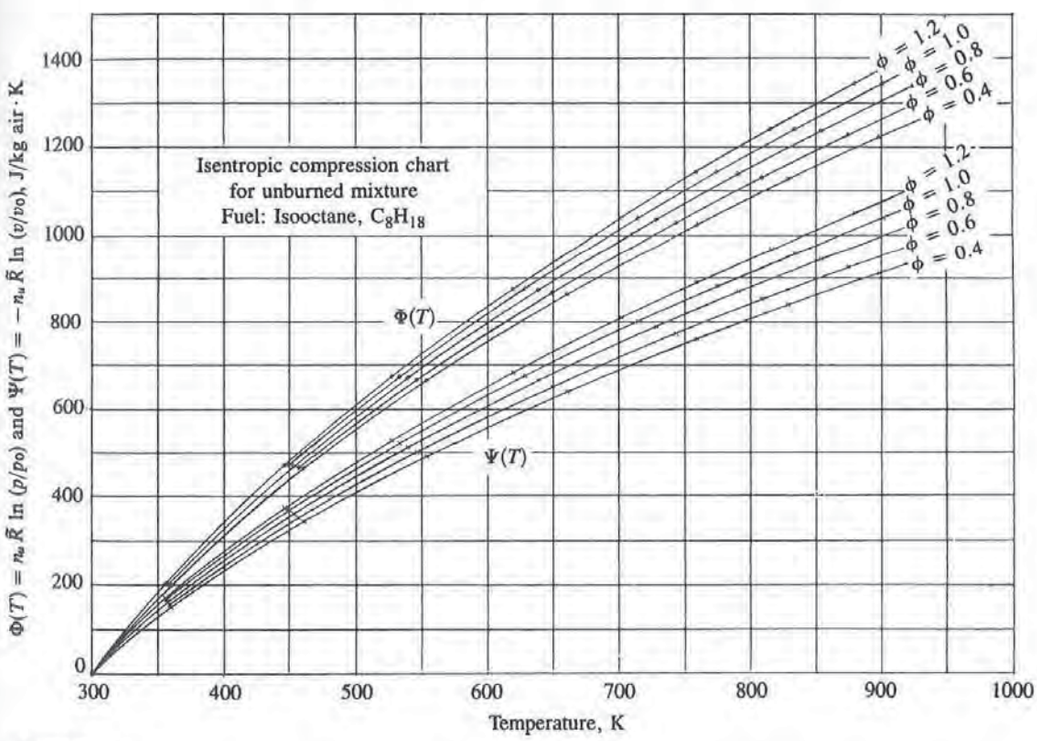


FIGURE 4-4
Isentropic compression functions, Φ and Ψ, as function of temperature for unburned isooctane-air mixtures. Units: J/kg air · K.

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4.5.2 Burned Mixture Charts

The primary burned mixture charts are for the products of combustion at high temperatures, i.e., for the working fluid during the expansion process. The following assumptions are made:

1. Each species in the mixture can be modeled as an ideal gas.
2. The mixture is in thermodynamic equilibrium at temperatures above 1700 K; the mixture composition is frozen below 1700 K.
3. *Datum.* At the datum state of 298.15 K (25°C or 77°F) and 1 atm the chemical elements in their naturally occurring form (N_2, O_2, H_2 as diatomic gases and C as solid graphite) are assigned zero enthalpy and entropy.

The charts were prepared with the NASA equilibrium program described in Sec. 3.7.[9,10] The C/H/O/N ratio of the mixture is specified for each chart. The extensive properties (internal energy, enthalpy, entropy, and specific volume) are all expressed per unit mass of air in the original mixture; i.e., they correspond to the combustion of 1 kg of air with the appropriate mass of fuel. The mass basis for the unburned and burned mixture charts are the same.

Figures 4-5 to 4-9 are property charts for the high-temperature burned gas; each is a plot of internal energy versus entropy for a particular fuel and equivalence ratio. Lines of constant temperature, pressure, and specific volume are drawn on each chart. An illustration of the use of these charts follows.

Example 4.3. The expansion process in an internal combustion engine, following completion of combustion, can be modeled approximately as an adiabatic and reversible process (i.e., isentropic). Under full-load operation, the pressure in the cylinder of a spark-ignition engine at top-center immediately following combustion is 7100 kPa. Find the gas state at the end of the expansion stroke and the expansion stroke work. The compression ratio is 8, the mixture is stoichiometric, and the volume per unit mass of air at the start of expansion is 0.125 m^3/kg air.

Locate $p_1 = 7100$ kPa and $v_1 = 0.125$ m^3/kg air on the $\phi = 1.0$ burned gas chart (Fig. 4-8). This gives $T_1 = 2825$ K, $u_1 = -5$ kJ/kg air, and $s_1 = 9.33$ kJ/kg air · K. The gas expands at constant entropy to $v_2 = 8 \times v_1 = 1$ m^3/kg air. Following a constant entropy process from state 1 on Fig. 4-8 gives

$$T_2 = 1840 \text{ K}, \qquad p_2 = 570 \text{ kPa}, \qquad \text{and} \qquad u_2 = -1540 \text{ kJ/kg air}$$

The expansion stroke work, assuming the process is adiabatic, is

$$W_{1-2} = -(u_2 - u_1) = 1540 - 5 = 1535 \text{ kJ/kg air}$$

As the burned gases in an engine cylinder cool during the expansion process, the composition eventually "freezes"—becomes fixed in composition—because the chemical reactions become extremely slow. This is usually assumed to occur at about 1700 K (see Sec. 4.9). The equilibrium assumption is then no longer valid. For lean and stoichiometric mixtures this distinction is not important because the mole fractions of dissociated species below this temperature are

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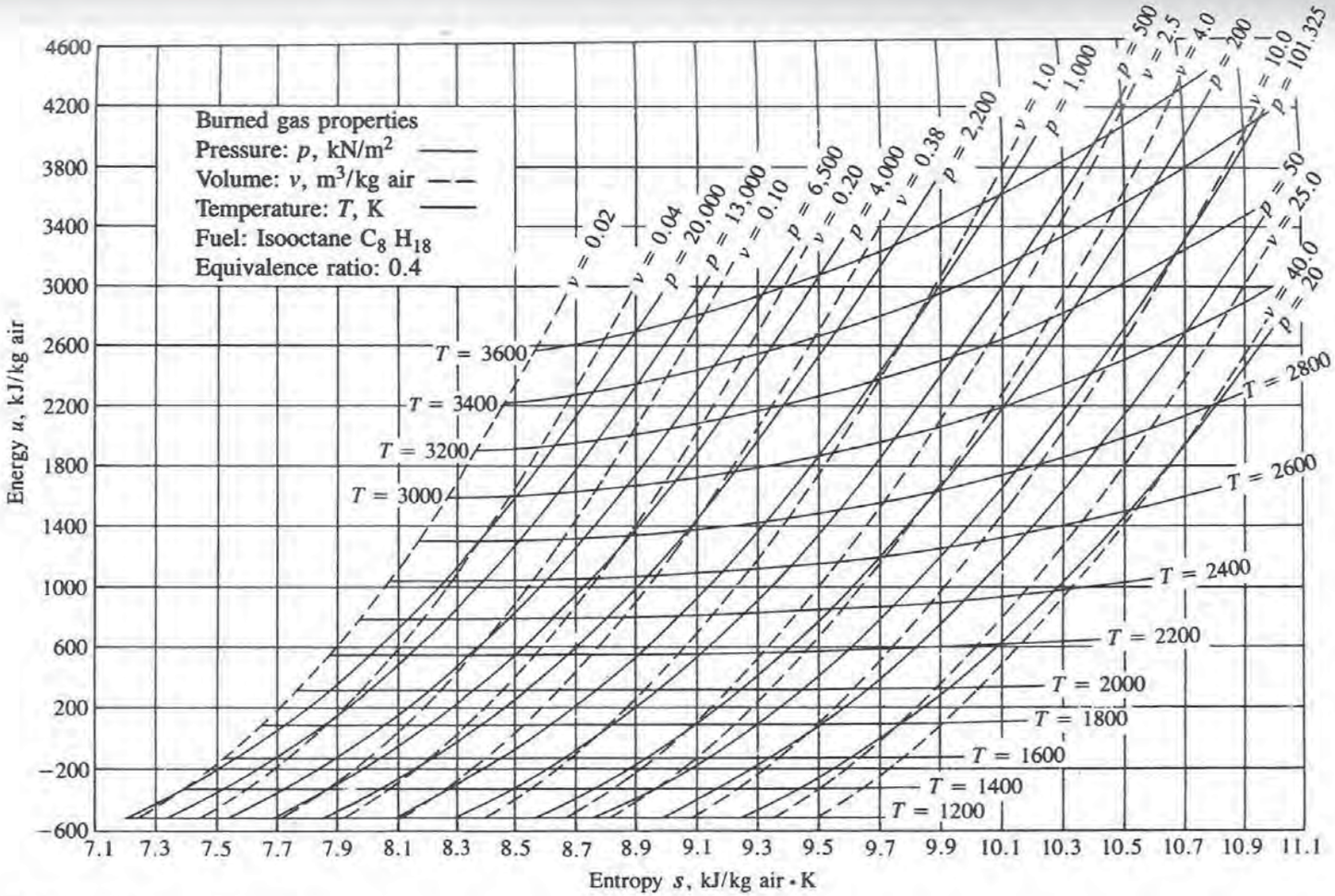


FIGURE 4-5
Internal energy versus entropy chart for equilibrium burned gas mixture, isooctane fuel; equivalence ratio 0.4.

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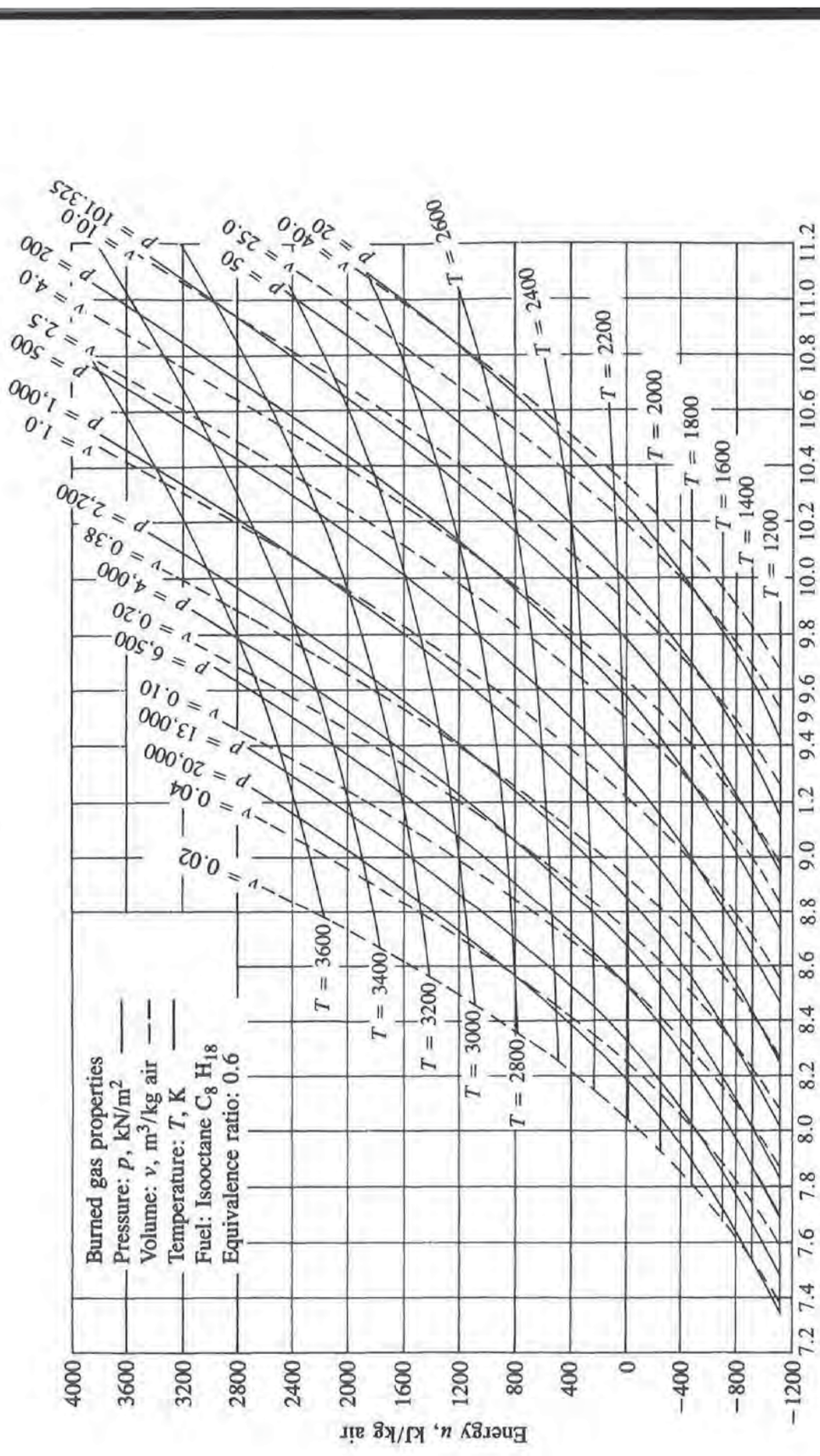


FIGURE 4-6
Internal energy versus entropy chart for equilibrium burned gas mixture, isooctane fuel; equivalence ratio 0.6.

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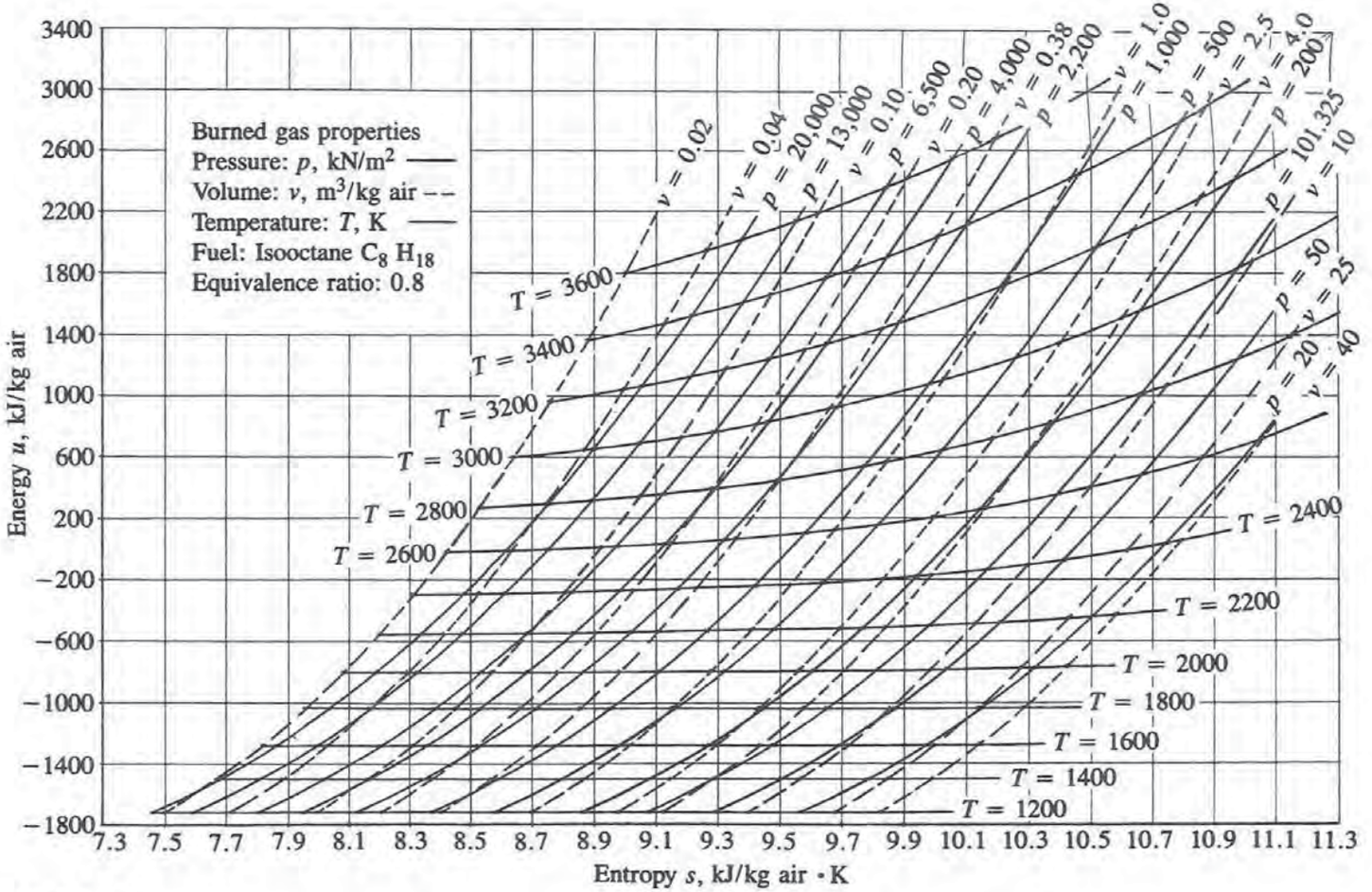


FIGURE 4-7
Internal energy versus entropy chart for equilibrium burned gas mixture, isooctane fuel; equivalence ratio 0.8.

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FIGURE 4-8
Internal energy versus entropy chart for equilibrium burned gas mixture, isooctane fuel; equivalence ratio 1.0.

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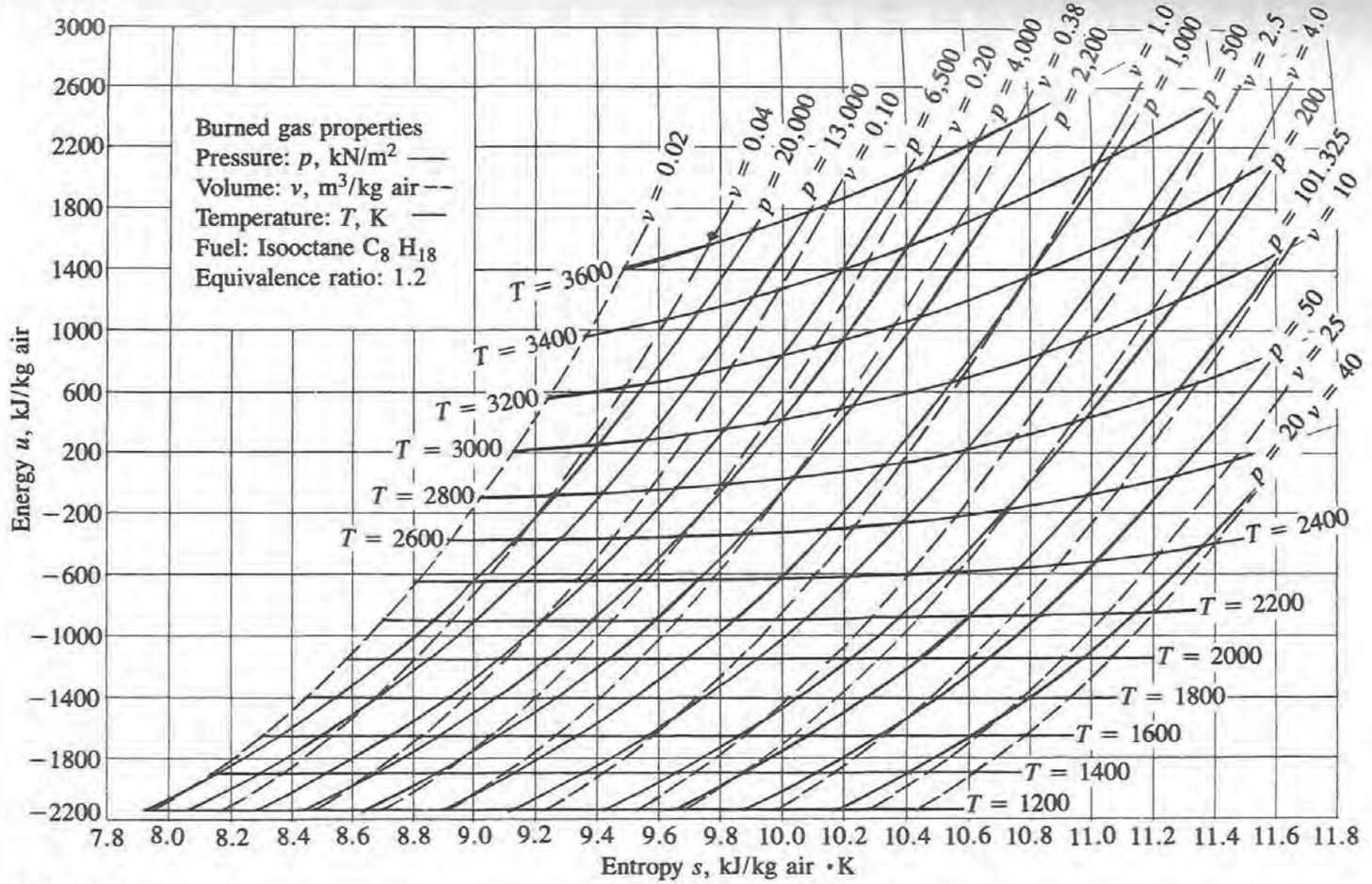


FIGURE 4-9
Internal energy versus entropy chart for equilibrium burned gas mixture, isooctane fuel; equivalence ratio 1.2.

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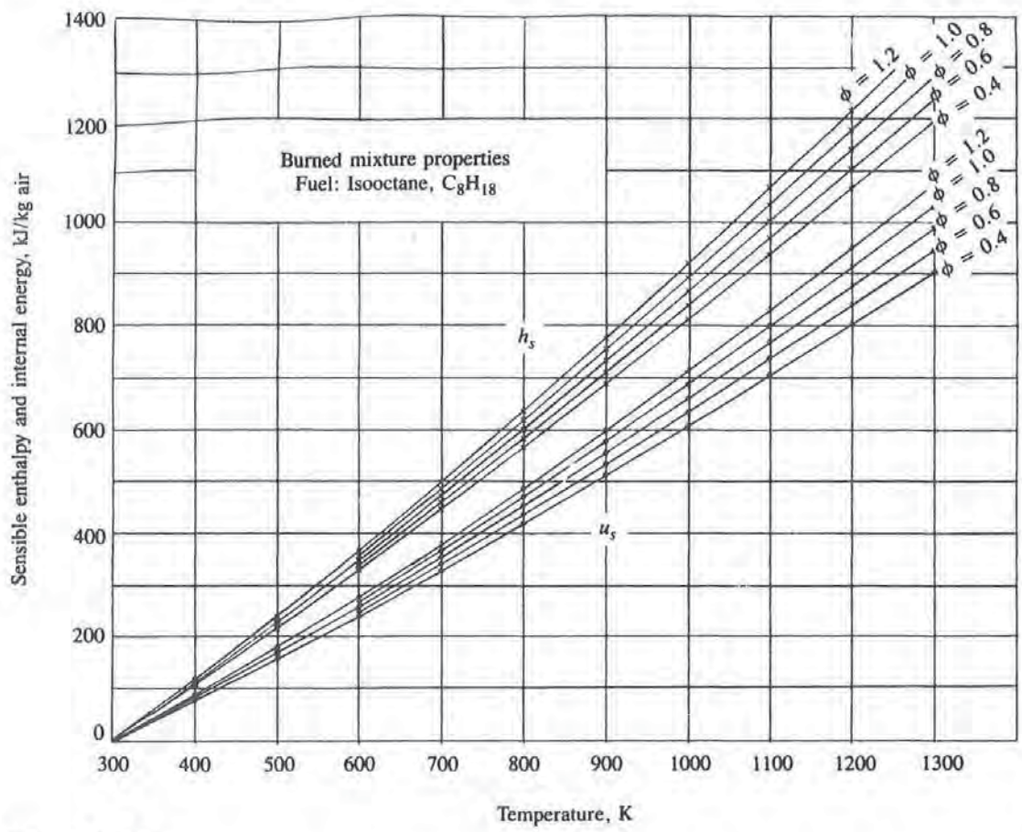


FIGURE 4-10
Sensible enthalpy and internal energy of low-temperature burned gases as function of temperature, isooctane fuel. Units: kJ/kg air in original mixture.

TABLE 4.7
Frozen burned gas composition: C_8H_{18}-air combustion

ϕ	CO_2	H_2O	CO	H_2	O_2	N_2	Sum	Units
0.4	0.0521	0.0586	—	—	0.122	0.767	1.000	mole fractions
	1.85	2.08	—	—	4.34	27.3	35.6	mol/kg air‡
0.6	0.0770	0.0866	—	—	0.0802	0.756	1.000	mole fractions
	2.78	3.13	—	—	2.89	27.3	36.1	mol/kg air‡
0.8	0.101	0.113	—	—	0.0395	0.746	1.000	mole fractions
	3.70	4.14	—	—	1.45	27.3	36.6	mol/kg air‡
1.0	0.125	0.140	—	—	—	0.735	1.000	mole fractions
	4.64	5.2	—	—	—	27.3	37.1	mol/kg air‡
1.2†	0.0905	0.138	0.0516	0.0224	—	0.698	1.000	mole fractions
	3.54	5.38	2.02	0.876	—	27.3	39.1	mol/kg air‡

† $K(T)$ in Eq. (4.6) evaluated at 1740 K; $K = 3.5$.
‡ Note mol/kg air; multiply by 10^{-3} for kmol/kg air.

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small. For rich mixtures, a frozen composition must be selected and used because the mole fractions of CO_2, CO, H_2O, and H_2 would continue to change if equilibrium is assumed as the temperature decreases. Internal energy and enthalpy, per kilogram of air in the original mixture, of the frozen burned mixture are plotted against temperature in Fig. 4-10. The assumed frozen burned mixture compositions are listed in Table 4.7. These are sensible internal energies and enthalpies, given relative to their values at 298.15 K.

4.5.3 Relation between Unburned and Burned Mixture Charts

We now address the questions: Given unburned mixture at T_1, p_1, v_1, what is the state of the burned mixture following (1) constant-volume adiabatic combustion or (2) constant-pressure adiabatic combustion?

The datum for internal energy and enthalpy for the unburned mixture in Fig. 4-3 is different from the datum for internal energy and enthalpy for the burned mixture. For the unburned mixture, zero internal energy and enthalpy for the *mixture* at 298.15 K was assumed. For the burned mixture, zero enthalpy for the gaseous species O_2, N_2, and H_2, and C (solid graphite) at 298.15 K was assumed. These data can be related through the enthalpies of formation, from O_2, N_2, H_2, and C, of each species in the unburned mixture.

If $\Delta\tilde{h}^\circ_{f,i}$ is the enthalpy of formation of species i at 298.15 K, per kilomole, and $\Delta h^\circ_{f,u}$ is the enthalpy of formation of the unburned mixture at 298.15 K, per kilogram of air in the original mixture, then

$$\Delta h^\circ_{f,u} = \sum_i n_i \Delta\tilde{h}^\circ_{f,i} \tag{4.27}$$

where n_i is the number of kilomoles of species i per kilogram of air. The unburned mixture enthalpy h_u, with the same datum as the burned mixture enthalpy, is therefore given by the sum of the sensible enthalpy $h_{s,u}$ and $\Delta h^\circ_{f,u}$:

$$h_u = h_{s,u} + \Delta h^\circ_{f,u} \tag{4.28}$$

Similarly, the internal energy u_u is given by

$$u_u = u_{s,u} + \Delta u^\circ_{f,u} \tag{4.29}$$

$\Delta u^\circ_{f,u}$ can be obtained from

$$\Delta u^\circ_{f,u} = \sum_i n_i \Delta\tilde{u}^\circ_{f,i} \tag{4.30}$$

Alternately, Eq. (3.18) can be used to obtain $\Delta u^\circ_{f,u}$ from $\Delta h^\circ_{f,u}$:

$$\Delta u^\circ_{f,u} = \Delta h^\circ_{f,u} - (n_P - n_R)\tilde{R}T \tag{4.31}$$

Enthalpies and internal energies of formation of the relevant burned gas species and individual fuel compounds are given in Table 4.8 and App. D. Values of n_i are obtained from Tables 4.4 and 4.7. Following the procedure used in Example 4.4 below, expressions for $\Delta h^\circ_{f,u}$ and $\Delta u^\circ_{f,u}$ in kilojoules per kilogram of

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TABLE 4.8
Standard enthalpies and internal energies of formation†

	$\Delta\tilde{h}^\circ_{f,i}$, MJ/kmol	$\Delta\tilde{u}^\circ_{f,i}$, MJ/kmol
CO_2	−393.5	−393.5
H_2O (gas)	−241.8	−240.6
CO	−110.5	−111.7
C_8H_{18} (gas)	−224.1	−204.3

† At 298.15 K. $\Delta\tilde{h}^\circ_{f,i}$ for O_2, N_2, and H_2 are zero by definition.
Sources: JANAF tables,[8] Rossini *et al.*[16]

air can be obtained. For the charts of Figs. 4-3 and 4-5 to 4-9, these expressions are:

$\phi = 0.4$:

$$\Delta h^\circ_{f,u} = -51.9 - 1181x_b \qquad \Delta u^\circ_{f,u} = -47.3 - 1183x_b$$

$\phi = 0.6$:

$$\Delta h^\circ_{f,u} = -77.8 - 1771x_b \qquad \Delta u^\circ_{f,u} = -70.9 - 1774x_b$$

$\phi = 0.8$:

$$\Delta h^\circ_{f,u} = -103.8 - 2361x_b \qquad \Delta u^\circ_{f,u} = -94.6 - 2365x_b \tag{4.32}$$

$\phi = 1.0$:

$$\Delta h^\circ_{f,u} = -129.7 - 2951x_b \qquad \Delta u^\circ_{f,u} = -118.2 - 2956x_b$$

$\phi = 1.2$:

$$\Delta h^\circ_{f,u} = -155.6 - 2759x_b \qquad \Delta u^\circ_{f,u} = -141.9 - 2769x_b$$

Example 4.4. Calculate $\Delta h^\circ_{f,u}$, the enthalpy of formation of the unburned mixture, and $\Delta u^\circ_{f,u}$, the internal energy of formation of the unburned mixture, for a C_8H_{18}-air mixture with $\phi = 1.0$ and burned gas fraction x_b.

Table 4.4 gives the moles of each species in the unburned mixture, per mole O_2 with $\phi = 1.0$, as

C_8H_{18}, $0.08(1 - x_b)$	CO_2, $0.64x_b$
O_2, $1 - x_b$	H_2O, $0.72x_b$
N_2, 3.773	CO and H_2, 0

Table 4.5 gives the mass of air per mole O_2 as 138.2 kg/kmol. Thus the number of kilomoles of each species per kilogram of air is

C_8H_{18}, $5.787 \times 10^{-4}\,(1 - x_b)$	CO_2, $4.629 \times 10^{-3}x_b$
O_2, $7.233 \times 10^{-3}\,(1 - x_b)$	H_2O, $5.208 \times 10^{-3}x_b$
N_2, 2.729×10^{-2}	CO and H_2, 0

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With $\Delta\tilde{h}^\circ_{f,i}$ from Table 4.8, Eq. (4.27) gives

$$\Delta h^\circ_{f,u} = 5.787 \times 10^{-4} \times (-224.1 \times 10^6)(1 - x_b) + x_b[4.629 \times 10^{-3} \times (-393.5 \times 10^6) + 5.208 \times 10^{-3} \times (-241.8 \times 10^6)]$$

$$\Delta h^\circ_{f,u} = (-129.7 - 2951x_b) \times 10^3 \quad \text{J/kg air}$$

With $\Delta\tilde{u}^\circ_{f,i}$ from Table 4.8, Eq. (4.30) gives

$$\Delta u^\circ_{f,u} = 5.787 \times 10^{-4} \times (-204.3 \times 10^6)(1 - x_b) + x_b[4.629 \times 10^{-3} \times (-393.5 \times 10^6) + 5.208 \times 10^{-3} \times (-240.6 \times 10^6)]$$

$$\Delta u^\circ_{f,u} = (-118.2 - 2956x_b) \times 10^3 \quad \text{J/kg air}$$

Alternatively, we can determine $\Delta u^\circ_{f,u}$ from $\Delta h^\circ_{f,u}$ using Eq. (4.31). For this calculation, the "product" gas is the unburned mixture and the "reactant" gas is the mixture of elements from which the unburned mixture is formed. The number of gaseous moles in the unburned mixture n_P, per mole O_2 in the original mixture, is (from Table 4.5 for $\phi \leq 1$)

$$n_P = (1 - x_b)\left[4(1 + 2\varepsilon)\frac{\phi}{M_f} + 1 + \psi\right] + x_b[(1 - \varepsilon)\phi + 1 + \psi]$$

The elemental reactant mixture from which the unburned mixture is formed is, from Eq. (4.4),

$$\varepsilon\phi C + 2(1 - \varepsilon)\phi H_2 + O_2 + \psi N_2$$

Thus, n_R, the moles of *gaseous* elements, is

$$n_R = 2(1 - \varepsilon)\phi + 1 + \psi$$

For air, $\psi = 3.773$; for C_8H_{18} fuel, $\varepsilon = 0.64$ and $M_f = 114$. For $\phi = 1$,

$$n_P - n_R = -0.64 + 0.28x_b \quad \text{moles/mole } O_2$$

and

$$(n_P - n_R)\tilde{R}T = (-0.64 + 0.28x_b) \times 8.3143 \times 10^3 \times \frac{298.15}{138.2}$$

or

$$(n_P - n_R)\tilde{R}T = (-11.5 + 5.0x_b) \times 10^3 \quad \text{J/kg air}$$

Since

$$\Delta u^\circ_{f,u} = \Delta h^\circ_{f,u} - (n_P - n_R)\tilde{R}T$$

$$\Delta u^\circ_{f,u} = (-129.7 - 2951x_b) \times 10^3 - (-11.5 + 5.0x_b) \times 10^3$$

$$\Delta u^\circ_{f,u} = (-118.2 - 2956x_b) \times 10^3 \quad \text{J/kg air}$$

The combustion process links the unburned and burned mixture properties as follows:

For an adiabatic constant-volume combustion process,

$$u_b = u_u = u_{s,u} + \Delta u^\circ_{f,u} \tag{4.33}$$

and

$$v_b = v_u$$

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Thus, given $u_{s,u}$ and v_u, the state of the burned mixture can be determined from the appropriate burned mixture chart.

For an adiabatic constant-pressure combustion process,

$$h_b = h_u = h_{s,u} + \Delta h^\circ_{f,u} \tag{4.34}$$

Since

$$u_b = h_b - pv_b$$

given $h_{s,u}$ and p, u_b and v_b must be found by trial and error along the specified constant-pressure line on the appropriate burned mixture chart.

Example 4.5. Calculate the temperature and pressure after constant-volume adiabatic combustion and constant-pressure adiabatic combustion of the unburned mixture (with $\phi = 1.0$ and $x_b = 0.08$) at the state corresponding to the end of the compression process examined in Example 4.2.

The state of the unburned mixture at the end of the compression process in Example 4.2 was

$$T_u = 682 \text{ K}, \qquad u_{s,u} = 350 \text{ kJ/kg air}, \qquad p_u = 1.57 \text{ MPa}, \qquad v_u = 0.125 \text{ m}^3\text{/kg air}$$

For an adiabatic *constant-volume* combustion process [Eq. (4.33)],

$$u_b = u_u = u_{s,u} + \Delta u^\circ_{f,u}$$

For $\phi = 1.0$, $\Delta u^\circ_{f,u}$ is given by Eq. (4.32) as

$$\Delta u^\circ_{f,u} = -118.2 - 2956x_b = -118.2 - 236.5 = -355 \text{ kJ/kg air}$$

Hence

$$u_b = 350 - 355 = -5 \text{ kJ/kg air}$$

Also

$$v_b = v_u = 0.125 \text{ m}^3\text{/kg air}$$

Locating (u_b, v_b) on the burned gas chart (Fig. 4-8) gives

$$T_b = 2825 \text{ K}, \qquad p_b = 7100 \text{ kPa}$$

For a *constant-pressure* combustion process [Eq. (4.34)],

$$h_b = h_u = h_{s,u} + \Delta h^\circ_{f,u}$$

For $\phi = 1.0$, $\Delta h^\circ_{f,u}$ is given by Eq. (4.32) as

$$\Delta h^\circ_{f,u} = -129.7 - 2951x_b = -129.7 - 236 = -366 \text{ kJ/kg air}$$

At $T_u = 682$ K, $h_{s,u} = 465$ kJ/kg air, so

$$h_b = 465 - 366 = 99 \text{ kJ/kg air}$$

Since $p_b = p_u = 1.57$ MPa, the internal energy u_b is given by

$$u_b = h_b - p_b v_b = 99 - 1.57 \times 10^3 v_b \qquad \text{kJ/kg air}$$

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A trial-and-error solution for v_b and u_b along the $p = 1570$ kPa line on Fig. 4-8 gives

$$u_b = -655 \text{ kJ/kg air}, \qquad T_b = 2440 \text{ K}, \qquad v_b = 0.485 \text{ m}^3/\text{kg air}$$

(Use the ideal gas law to estimate p, T, or v more accurately.)

4.6 TABLES OF PROPERTIES AND COMPOSITION

Tables of thermodynamic properties of air are useful for analysis of motored engine operation, diesels and compressors. Keenan, Chao, and Kaye's *Gas Tables*[6] are the standard reference for the thermodynamic properties of air at low pressures (i.e., at pressures substantially below the critical pressure when the ideal gas law is accurate). These gas tables are in U.S. and SI units. A set of tables for air in SI units has been prepared by Reynolds[7] following the format of the Keenan *et al.* tables. A condensed table of thermodynamic properties of air, derived from Reynolds, is given in App. D. It contains:

h = enthalpy, kJ/kg
u = internal energy, kJ/kg
$\Psi = \int_0^T \left(\frac{c_v}{T}\right) dT$, kJ/kg · K
$\Phi = \int_0^T \left(\frac{c_p}{T}\right) dT$, kJ/kg · K
p_r = relative pressure
v_r = relative volume
c_p = specific heat at constant pressure, kJ/kg · K
c_v = specific heat at constant volume, kJ/kg · K
γ = ratio of specific heats

all as a function of T(K).

Φ is the standard state entropy at temperature T and 1 atm pressure, relative to the entropy at 0 K and 1 atm pressure. The entropy at pressures other than 1 atm is obtained using Eq. (4.14*b*).

The relative pressure p_r is defined by

$$\ln p_r = \frac{\Phi}{R} \tag{4.35}$$

and is a function of T only. Along a given isentropic, it follows from Eq. (4.18) that the ratio of actual pressures p_2 and p_1 corresponding to temperatures T_2 and T_1 is equal to the ratio of relative pressures, i.e.,

$$\left(\frac{p_2}{p_1}\right)_{s=\text{const}} = \left(\frac{p_{r2}}{p_{r1}}\right) \tag{4.36}$$

This affords a means of determining T_2, for an isentropic process, given T_1 and p_2/p_1 (see Example 4.6).

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The relative volume v_r is defined by

$$v_r = \frac{RT}{p_r} \tag{4.37}$$

The units are selected so that v_r is in cubic meters per kilogram when T is in kelvins and p_r is in pascals. Along a given isentropic, the ratio of actual volumes V_2 and V_1 (for a fixed mass) at temperatures T_2 and T_1, from Eq. (4.37), is equal to the ratio of relative volumes

$$\left(\frac{V_2}{V_1}\right)_{s=\text{const}} = \left(\frac{v_{r2}}{v_{r1}}\right) \tag{4.38}$$

This affords a means of determining T_2 for an isentropic process, given T_1 and V_2/V_1 (see Example 4.6).

Tables giving the composition and thermodynamic properties of combustion products have been compiled. They are useful sources of property and species concentrations data in burned gas mixtures for a range of equivalence ratios, temperatures, and pressures. Summary information on four generally available sets of tables is given in Table 4.9. The most extensive set of tables of combustion product composition and thermodynamic properties is the AGARD set, *Properties of Air and Combustion Products with Kerosene and Hydrogen Fuels*, by Banes *et al.*[12] Note, however, that their enthalpy datum differs from the usual datum (enthalpy for O_2, N_2, H_2, and C is zero at 298.15 K). The elements in their reference state at 298.15 K were assigned arbitrary positive values for enthalpy to avoid negative enthalpies for the equilibrium burned gas mixture.

Example 4.6. In a diesel engine, the air conditions at the start of compression are $p_1 = 1$ atm and $T_1 = 325$ K. At the end of compression $p_2 = 60$ atm. Find the temperature T_2 and the compression ratio V_1/V_2.

Air tables (see App. D), at $T_1 = 325$ K, give

$$p_{r1} = 97.13 \qquad \text{and} \qquad v_{r1} = 960.6$$

Use Eq. (4.36),

$$\frac{p_{r2}}{p_{r1}} = \frac{p_2}{p_1} = 60$$

to give

$$p_{r2} = 5828$$

Tables then give

$$T_2 = 992 \text{ K} \qquad \text{and} \qquad v_{r2} = 48.92$$

The compression ratio is given by

$$\frac{V_1}{V_2} = \frac{v_{r1}}{v_{r2}} = \frac{960.6}{48.92} = 19.6$$

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TABLE 4.9
Tables of properties of air and combustion products

Source	Properties P, composition C	Mixture	Units	ϕ range	T range	p range	Enthalpy datum
Keenan, Chao, and Kaye[6]	P	Air	U.S., SI		100–3600 K	Low	$h = 0$ at 0 K
		$(CH_2)_n$-air		0.25, 0.5, 1.0	100–2000 K	Low	$\tilde{h}_b = 0$ at 0 K
Reynolds[7]	P	Air	SI		200–1500 K	Low	
General Electric[11]	P and C	Air-$(CH_2)_n$	U.S.	0.25–4	600–5000°R	0.01–30 atm	h of C, H_2, N_2, O_2 zero at 0°R
AGARD[12]	P and C	Air	SI		100–6000 K	1–800 atm	Arbitrary, to keep $h_b > 0$
		$(CH_2)_n$-air		0.2–2			
		H_2-air		0.2–2			

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4.7 COMPUTER ROUTINES FOR PROPERTY AND COMPOSITION CALCULATIONS

When large numbers of computations are being made or high accuracy is required, engine process calculations are carried out on a computer. Relationships which model the composition and/or thermodynamic properties of unburned and burned gas mixtures have been developed for computer use. These vary considerably in range of application and accuracy.

The most complete models are based on polynomial curve fits to the thermodynamic data for each species in the mixture and the assumptions that (1) the unburned mixture is frozen in composition and (2) the burned mixture is in equilibrium. The approach used as the basis for representing JANAF table thermodynamic data[8] in the NASA equilibrium program[9,10] (see Sec. 3.7) will be summarized here because it is consistent with the approach used throughout to calculate unburned and burned mixture properties.

For each species i in its standard state at temperature T(K), the specific heat $\tilde{c}_{p,i}$ is approximated by

$$\frac{\tilde{c}_{p,i}}{\tilde{R}} = a_{i1} + a_{i2}\,T + a_{i3}\,T^2 + a_{i4}\,T^3 + a_{i5}T^4 \tag{4.39}$$

The standard state enthalpy of species i is then given by

$$\frac{\tilde{h}_i}{\tilde{R}T} = a_{i1} + \frac{a_{i2}}{2}\,T + \frac{a_{i3}}{3}\,T^2 + \frac{a_{i4}}{4}\,T^3 + \frac{a_{i5}}{5}\,T^4 + \frac{a_{i6}}{T} \tag{4.40}$$

The standard state entropy of species i at temperature T(K) and pressure 1 atm, from Eq. (4.14), is then

$$\frac{\tilde{s}_i}{\tilde{R}} = a_{i1}\ln T + a_{i2}\,T + \frac{a_{i3}}{2}\,T^2 + \frac{a_{i4}}{3}\,T^3 + \frac{a_{i5}}{4}\,T^4 + a_{i7} \tag{4.41}$$

Values of the coefficients a_{ij} for CO_2, H_2O, CO, H_2, O_2, N_2, OH, NO, O, and H from the NASA program are given in Table 4.10. Two temperature ranges are given. The 300 to 1000 K range is appropriate for unburned mixture property calculations. The 1000 to 5000 K range is appropriate for burned mixture property calculations. Figure 4-11 gives values of c_p/R for the major species, CO_2, H_2O, O_2, N_2, H_2, and CO, as a function of temperature.

4.7.1 Unburned Mixtures

Polynomial functions for various fuels (in the vapor phase) have been fitted to the functional form:[13–15]

$$\tilde{c}_{p,f} = A_{f1} + A_{f2}\,t + A_{f3}\,t^2 + A_{f4}\,t^3 + \frac{A_{f5}}{t^2} \tag{4.42}$$

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