Internal combustion engine

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A prime mover that burns fuel inside the engine, in contrast to an external combustion engine, such as a steam engine, which burns fuel in a separate furnace. *See also:* ENGINE.

Most internal combustion engines are spark-ignition gasoline-fueled piston engines. These are used in automobiles, light- and medium-duty trucks, motorcycles, motorboats, lawn and garden equipment, and light industrial and portable power applications. Diesel engines are used in automobiles, trucks, buses, tractors, earthmoving equipment, as well as marine, power-generating, and heavier industrial and stationary applications. This article describes these types of engines. For other types of internal combustion engines *see* GAS TURBINE; ROCKET PROPULSION; TURBINE PROPULSION.

The aircraft piston engine is fundamentally the same as that used in automobiles but is engineered for light weight and is usually air-cooled. *See also:* RECIPROCATING AIRCRAFT ENGINE.

Engine types

Characteristics common to all commercially successful internal combustion engines include (1) the compression of air, (2) the raising of air temperature by the combustion of fuel in this air at its elevated pressure, (3) the extraction of work from the heated air by expansion to the initial pressure, and (4) exhaust.

Four-stroke cycle. William Barnett first drew attention to the theoretical advantages of combustion under compression in 1838. In 1862 Beau de Rochas published a treatise that emphasized the value of combustion under pressure and a high ratio of expansion for fuel economy; he proposed the four-stroke engine cycle as a means of accomplishing these conditions in a piston engine (**Fig. 1**). The engine requires two revolutions of the crankshaft to complete one combustion cycle. The first engine to use this cycle successfully was built in 1876 by N. A. Otto. *See also:* OTTO CYCLE.

Otto's engine, like almost all internal combustion engines developed at that period, burned coal gas mixed in combustible proportions with air prior to being drawn into the cylinder. The engine load was generally controlled by throttling the quantity of charge taken into the cylinder. Ignition was by a device such as an external flame or an electric spark, so that the timing was controllable. These are essential features of what has become known as the Otto or spark-ignition combustion cycle.

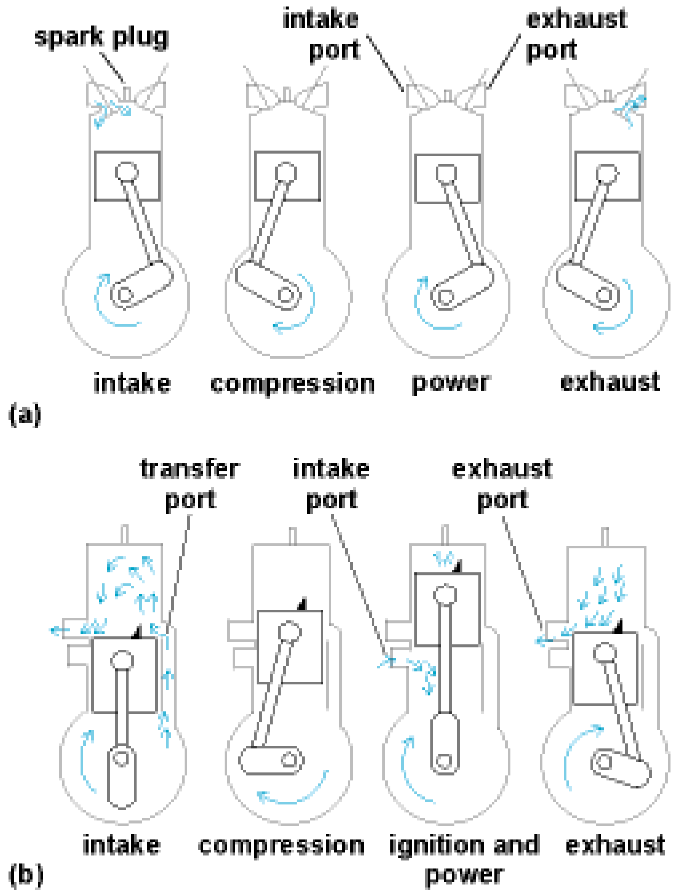

Fig. 1 Engine cycles (*a*) The four strokes of a four-stroke engine cycle. On intake stroke, the intake valve (left) has opened and the piston is moving downward, drawing air and gasoline vapor into the cylinder. On compression stroke, the intake valve has closed and the piston is moving upward, compressing the mixture. On power stroke, the ignition system produces a spark that ignites the mixture. As it burns, high pressure is created, which pushes the piston downward. On exhaust stroke, the exhaust valve (right) has opened and the piston is moving upward, forcing the burned gases from the cylinder. (*b*) Three-port two-cycle engine. The same action is accomplished without separate valves and in a single rotation of the crankshaft.

Two-stroke cycle. In 1878 Dugald Clerk developed the two-stroke engine cycle by which a similar combustion cycle required only one revolution of the crankshaft. In this cycle, exhaust ports in the cylinder were uncovered by the piston as it approached the end of its power stroke. A second cylinder then pumped a charge of air to the working cylinder through a check valve when the pump pressure exceeded that in the working cylinder.

In 1891 Joseph Day simplified the two-stroke engine cycle by using the crankcase to pump the required air. The compression stroke of the working piston draws the fresh combustible charge through a check valve into the crankcase, and the next power stroke of the piston compresses this charge. The piston uncovers the exhaust ports near the end of the power stroke and slightly later uncovers intake ports opposite them to admit the compressed charge from the crankcase. A baffle is usually provided on the piston head of small engines to deflect

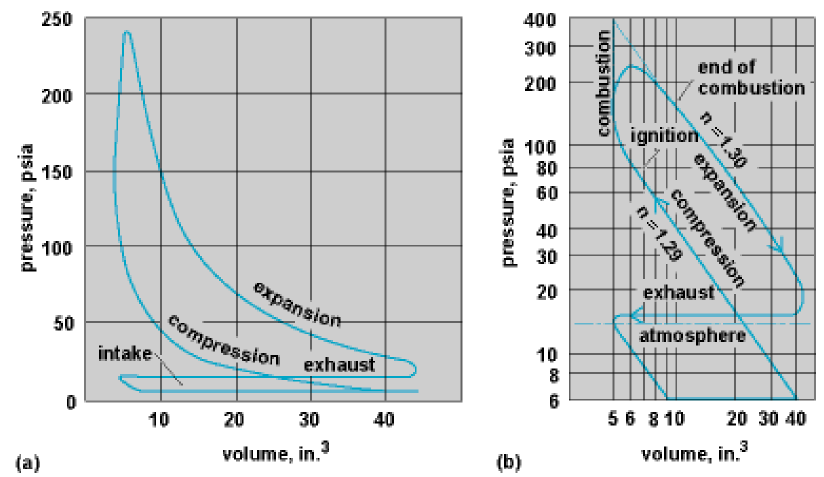

Fig. 2 **Typical Otto-cycle pressure–volume indicator card plotted on (*a*) rectangular coordinates and (*b*) logarithmic coordinates.**

the charge up one side of the cylinder to scavenge the remaining burned gases down the other side and out the exhaust ports with as little mixing as possible.

Modern engines using this two-stroke cycle have a third cylinder port known as the transfer port (Fig. 1*b*), instead of the crankcase check valve used by Day. Small engines of this type are widely used where fuel economy is not as important as mechanical simplicity and light weight. They do not need mechanically operated valves, and they develop one power impulse per cylinder for each crankshaft revolution.

Two-stroke-cycle engines do not develop twice the power of four-stroke-cycle engines with the same size of working cylinders at the same number of revolutions per minute (rpm). Principal reasons are (1) reduction in effective cylinder volume due to the piston movement required to cover exhaust ports; (2) appreciable mixing of burned (exhaust) gases with the combustible mixture; and (3) loss of some combustible mixture through the exhaust ports with the exhaust gases.

Otto-cycle engines

In the idealized four-stroke Otto cycle, combustion is instantaneous and at constant volume. This simplifies thermodynamic analysis, but combustion takes time. Gas pressure during the four strokes of the Otto cycle varies with the piston position as shown by the typical indicator card in **Fig. 2***a*. This is a pressure–volume (PV) card for an 8.7:1 compression ratio.

Engine power. To simplify calculations of engine power, the average net pressure during the working stroke, known as the mean effective pressure (mep), is frequently used. It may be obtained from the average net height of the card, which is found by measurement of the area and then division of this area by its length. Similar pressure–volume data may be plotted on logarithmic coordinates as in Fig. 2*b*, which develops expansion and compression relations as approximately straight lines. The slopes show the values of exponent n to use in equations for PV relationships.

The rounding of the plots at peak pressure, with the peak developing after the piston has started its power stroke, even with the spark occurring before the piston reaches the end of the compression stroke, is due to the time required for combustion. Changes in design can vary charge turbulence in the compression space prior to and during combustion. The greater the turbulence, the faster the combustion and the lower the antiknock or octane number required of the fuel, or the higher the compression ratio that may be used with a given fuel without knocking. The amount to which the turbulence can be raised is limited by the increased rate of pressure rise, which increases engine roughness. This must not exceed a level acceptable for automobile or other service. *See also:* AUTOMOBILE; AUTOMOTIVE ENGINE; COMBUSTION CHAMBER; COMPRESSION RATIO; MEAN EFFECTIVE PRESSURE; OCTANE NUMBER; SPARK KNOCK.

Detonation of a small part of the charge in the cylinder, after most of the charge has burned progressively, causes knock. This limits the compression ratio of an engine with a given fuel.

Compression ratio. According to classical thermodynamic theory, thermal efficiency η of the Otto combustion cycle is given by Eq. (1),

$$\eta = 1 - \frac{1}{r^{n-1}} \tag{1}$$

where the compression ratio r_c and expansion ratio r_e are the same ($r_c = r_e = r$). When theory assumes atmospheric air in the cylinder for extreme simplicity, exponent n is 1.4. Efficiencies calculated on this basis are almost twice as high as measured efficiencies. Logarithmic diagrams from experimental data show that n is about 1.3 (Fig. 2*b*). Even with this value, efficiencies achieved in practice are less than given by Eq. (1), because it assumes instantaneous combustion and 100% volumetric efficiency. This exponent should vary with the fuel-air mixture ratio, and to some extent with the compression ratio. For an 8:1 compression ratio, the exponent should vary from about 1.28 for a stoichiometric (chemically correct) mixture to about 1.31 for a lean mixture. Actual practice gives even lower thermal efficiencies. This is because of the assumed instantaneous changes in cyclic pressure (during combustion and exhaust) and the disregard of heat losses to the cylinder walls.

A change in compression ratio causes little change in the mechanical efficiency, or the volumetric efficiency resulting from raising the compression ratio provides a corresponding increase in torque or mean effective pressure. This is frequently of more practical importance than the actual efficiency increase. *See also:* THERMODYNAMIC CYCLE.

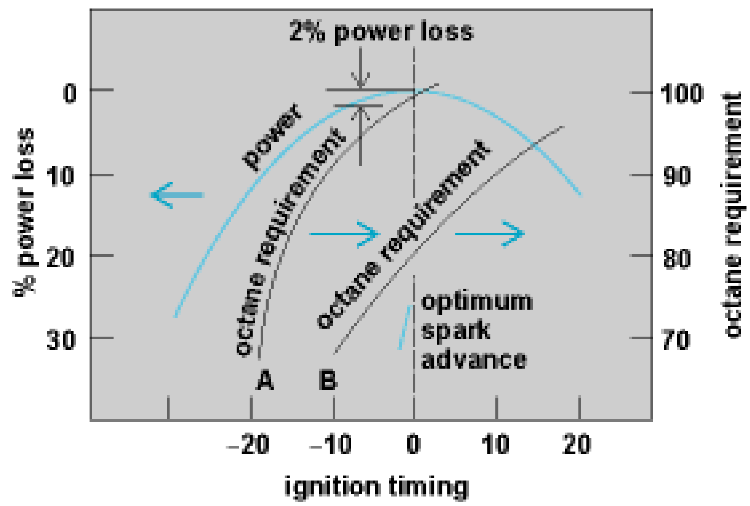

Fig. 3 **Effects of advancing or retarding ignition timing from optimum on engine power and resulting octane requirement of fuel in an experimental engine with a combustion chamber having typical turbulence (A) and a highly turbulent design (B) with the same compression ratio. Retarding the spark 7° for 2% power loss reduced octane requirement from 98 to 93 for design A.**

Engine load has little effect on indicated thermal efficiency, provided the fuel–air ratio remains constant and the ignition time is suitably advanced at reduced loads. This compensates for the slower rate of burning that results from dilution of the combustible charge with the larger percentages of burned gases remaining in the combustion space and the reduced turbulence at lower speeds.

High compression improves fuel economy because of improved thermal efficiency. However, the increased peak combustion temperature increases emissions of oxides of nitrogen in the exhaust gas. *See also:* AIR POLLUTION; SMOG.

Ignition timing. High thermal efficiency is obtained from high compression ratios at part loads, where engines normally run at automobile cruising speeds, with optimum spark advance. To avoid knock on available gasolines at wide-open throttle, a reduced or compromise spark advance is used. The tendency of an engine to knock at wide-open throttle is reduced appreciably when the spark timing is reduced 5–10° from optimum (**Fig. 3**). Advancing or retarding the spark timing from optimum results in an increasing loss in mean effective pressure for any normal engine, as shown by the heavy curve in Fig. 3. The octane requirement falls rapidly as the spark timing is retarded, the actual rate depending on the nature of the gasoline as well as on the combustion chamber design. Curves A and B show the effects on a given gasoline of the use of moderate- and high-turbulence combustion chambers, respectively, with the same compression ratio. Because the curve for mean effective pressure is relatively flat near optimum spark advance, retarding the spark for a 1–2% loss is normally acceptable because of the reduction in octane requirement.

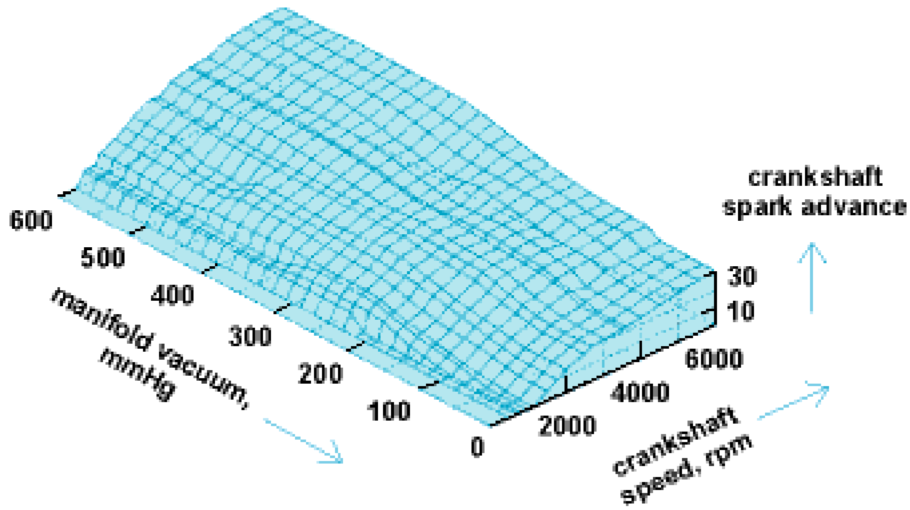

Fig. 4 Three-dimensional ignition map showing 576 timing points stored in the memory of a microprocessor system for a four-cylinder automotive engine. 1 mmHg = 133 Pa. (*Ford Motor Co.*)

In addition to the advantages of the higher compression ratio at cruising loads with optimum spark advance, the compromise spark at full load may be advanced toward optimum with higher-octane fuels. Then there is a corresponding increase in full-throttle mean effective pressure.

Many automotive engines have an electronic engine control system with an onboard microprocessor that controls spark timing electronically. The microprocessor continuously adjusts ignition timing for optimum fuel economy and drivability, while minimizing exhaust emissions. The microprocessor may be capable of retarding the timing if spark knock occurs. This allows the benefits of using a higher compression ratio and gasoline with a lower octane number without danger of engine-damaging detonation. In some systems, a digital map stored in memory provides a wide range of predetermined ignition settings (**Fig. 4**). *See also:* CONTROL SYSTEMS; IGNITION SYSTEM; MICROPROCESSOR.

Fuel–air ratio. A fuel–air mixture richer than that which develops maximum knock-free mep will permit use of higher compression ratios. However, the benefits derived from compromise or rich mixtures vary so much with mixture temperature and the sensitivity of the octane value of the particular fuel to temperature that this method is not generally practical. Nevertheless, piston-type aircraft engines may use fuel–air mixture ratios of 0.11 or even higher during takeoff, instead of about 0.08, which normally develops maximum mep in the absence of knock.

In automotive engines with an electronic engine control system, the microprocessor usually controls the amount of fuel delivered by either a feedback carburetor or a single-point (throttle-body) or a multipoint (port) electronic fuel-injection system. This maintains the mixture at or near the stoichiometric ratio, which minimizes exhaust emissions of hydrocarbons, carbon monoxide, and oxides of nitrogen. However, spark-ignition engines deliver

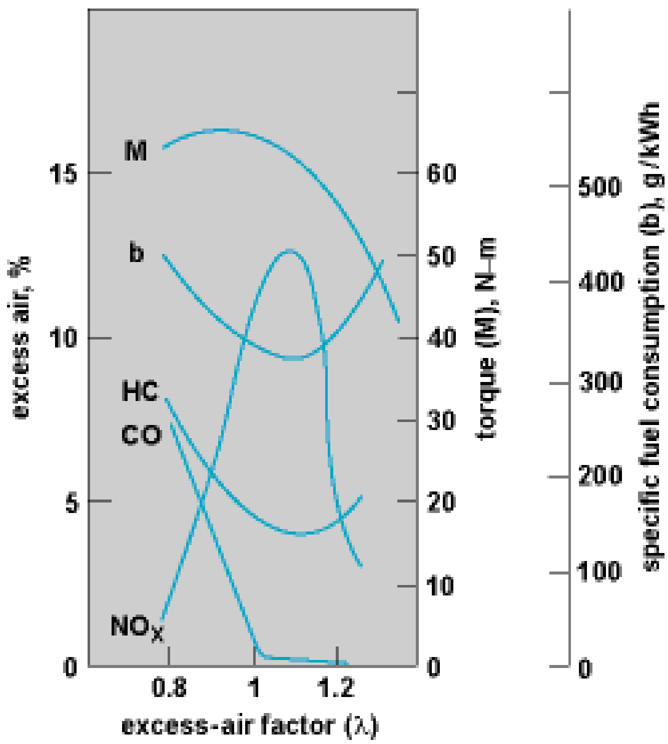

Fig. 5 **Graph showing how the excess-air factor affects exhaust-gas [composition carbon monoxide (CO), nitrogen oxides (NO_x), and hydrocarbons (HC)], torque (*M*), and specific fuel consumption (*b*) in the part-load range of an automotive spark-ignition engine running at a constant midrange speed and cylinder charge. (*Robert Bosch Corp.*)**

maximum power with an air deficiency of 0–10%, and minimum fuel consumption with about 10% excess air (**Fig. 5**).

Stroke–bore ratio. The ratio of the length of the piston stroke to the diameter of the cylinder bore has no appreciable effect on fuel economy and friction at corresponding piston speeds. Practical advantages that result from the short stroke include the greater rigidity of crankshaft from the shorter crank cheeks, with crankpins sometimes overlapping main bearings, and the narrower as well as lighter cylinder block that is possible. However, the higher rates of crankshaft rotation for an equivalent piston speed necessitate greater valve forces and require stronger valve springs. Also, the smaller depth of the compression space for a given compression ratio increases the surface-to-volume ratio and the proportion of heat lost by radiation during combustion. In automotive engines, stroke–bore ratios have decreased over the years.

Valve timing. The times of opening and closing the valves of an engine in relation to piston position are usually selected to develop maximum power over a desired speed range at wide-open throttle. The timing of these events is usually expressed as the number of degrees of crankshaft rotation before or after the piston reaches the end of one of its strokes.

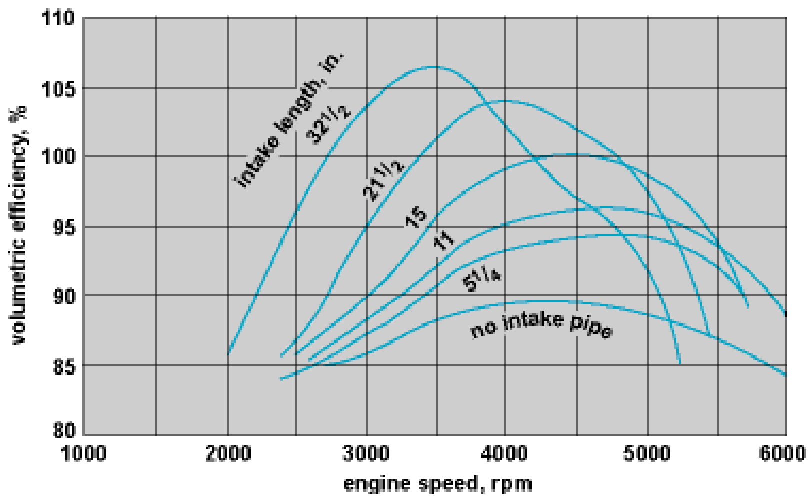

Fig. 6 **Effect of intake-pipe length and engine speed on volumetric efficiency of one cylinder of a six-cylinder engine. 1 in. = 2.5 cm.**

Because of the time required for the burned gas to flow through the exhaust valve at the end of the power stroke of a piston, the exhaust valve usually starts opening considerably before the end of the stroke. If the valve opens when the piston is nearer the lower end of its stroke, power is lost at high engine speeds because the piston on its exhaust stroke has to move against gas pressure remaining in the cylinder. If the valve opens before necessary, the burned gas is released while it is still at sufficient pressure to increase the work done on the piston.

For any engine, there is an optimum time for opening the exhaust valve that will develop the maximum power at some particular speed. The power loss at other speeds does not increase rapidly. Therefore when an engine is throttled at part load, there is less gas to discharge through the exhaust valve and less need for the valve to be opened as early as at wide-open throttle.

The timing of intake valve events is normally selected to trap the largest possible quantity of combustible mixture (air in a diesel engine) in the cylinder when the valve closes at some desired engine speed and at wide-open throttle. The intermittent flow through the intake valve undergoes alternate accelerations and decelerations, which require time. During the intake stroke, the mass of air moving through the passage to the intake valve is given velocity energy that may be converted to a slight pressure at the valve when the air mass still in the passage is stopped by its closure. Advantage of this phenomenon may be obtained at some engine speed to increase the air mass which enters the cylinder.

The engine speed at which the maximum volumetric efficiency is developed varies with the relative valve area, closure time, and other factors, including the diameter and length of the passage. The curves in **Fig. 6** show the characteristic falling off at high speeds from the inevitable throttling action as air flows at increased velocities through any restriction such as a valve or intake passage and venturi of a carburetor.

Volumetric efficiency has a direct effect on the mean effective pressure developed in a cylinder, on the torque, and on the power that may be realized at a given speed. Since power is a product of speed and torque, the peak power of an engine occurs at a higher speed than for maximum torque, where the rate of torque loss with any further increase in speed will exceed the rate of speed increase. An engine develops maximum power at a speed about twice that for maximum torque.

To obtain maximum torque and power, intake-valve closing may be delayed until the piston has traveled almost half the length of the compression stroke. At engine speeds below those where maximum torque is developed by this valve timing, some of the combustible charge that has been drawn into the cylinder on the intake stroke will be driven back through the intake valve before it closes. This reduces the effective compression ratio at wide-open throttle. The engine has an increasing tendency to develop spark knock as the speed and the resulting gas turbulence are reduced. *See also:* ENGINE MANIFOLD.

Supercharging spark-ignition engines. Volumetric efficiency and thus the mep of a four-stroke spark-ignition engine may be increased over a part of or the whole speed range by supplying air to the engine intake at higher than atmospheric pressure. This is usually accomplished by a centrifugal or rotary pump. The indicated power (power developed in the cylinder) of an engine increases directly with the absolute pressure in the intake manifold. Because fuel consumption increases at the same rate, the indicated specific fuel consumption (fuel flow rate per unit power output) is generally not altered appreciably by supercharging.

The three principal reasons for supercharging four-cycle spark-ignition engines are (1) to lessen the tapering off of mep at higher engine speed; (2) to prevent loss of power due to diminished atmospheric density, as when an airplane (with piston engines) climbs to high altitudes; and (3) to develop more torque at all speeds.

In a normal engine characteristic, torque rises as speed increases but falls off at higher speeds because of the throttling effects of such parts of the fuel intake system as valves and carburetors. If a supercharger is installed so as to maintain the volumetric efficiency at the higher speeds without increasing it in the middle-speed range, peak horsepower can be increased.

The rapid fall of atmospheric pressure at increased altitudes causes a corresponding decrease in the power of unsupercharged piston-type aircraft engines. For example, at 20,000 ft (6 km) the air density, and thus the absolute manifold pressure and indicated torque of an aircraft engine, would be only about half as great as at sea level. The useful power developed would be still less because of the friction and other mechanical power losses which are not affected appreciably by volumetric efficiency. By the use of superchargers, which are usually of the centrifugal type, sea-level air density may be maintained in the intake manifold up to considerable altitudes. Some aircraft engines drive these superchargers through gearing which may be changed in flight, from about 6.5 to 8.5 times engine speed. The speed change avoids oversupercharging at medium altitudes with corresponding power loss. Supercharged aircraft engines must be throttled at sea level to avoid damage from detonation or excessive overheating caused by the high mep which would otherwise be developed. *See also:* SUPERCHARGER.

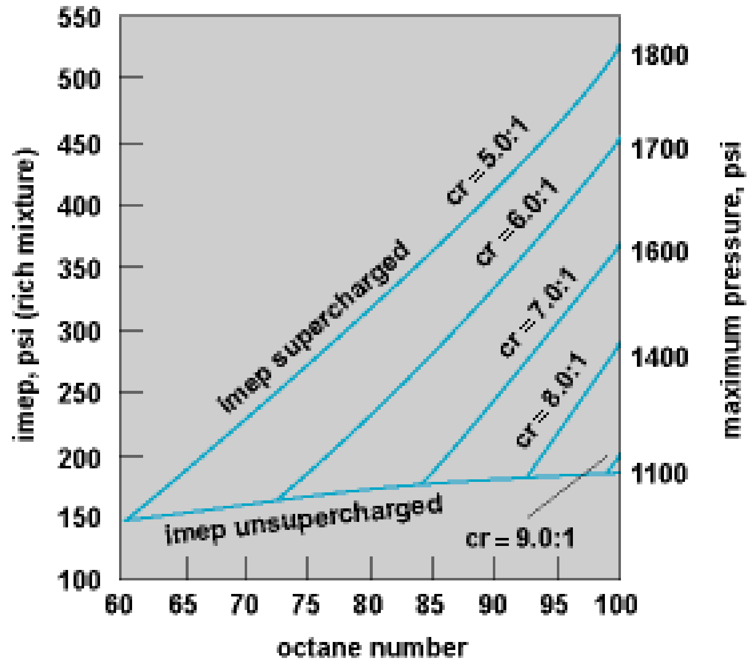

Fig. 7 **Graph showing the relationship between compression ratio and knock-limited imep for given octane numbers, obtained by supercharging a laboratory engine. (*After H. R. Ricardo, The High-Speed Combustion Engine, 4th ed., Blackie, 1953*)**

Normally an engine is designed with the highest compression ratio allowable without knock from the fuel expected to be used. This is desirable for the highest attainable mep and fuel economy from an atmospheric air supply. Any increase in the volumetric efficiency of such an engine would cause it to knock unless a fuel of higher octane number were used or the compression ratio were lowered. When the compression ratio is lowered, the knock-limited mep may be raised appreciably by supercharging but at the expense of lowered thermal efficiency. There are engine uses where power is more important than fuel economy, and supercharging becomes a solution. The principle involved is illustrated in **Fig. 7** for a given engine. With no supercharge this engine, when using 93-octane fuel, developed an indicated mean effective pressure (imep; an average pressure forcing the piston down the cylinder) of 180 pounds per square inch (psi; 1240 kilopascals) at the borderline of knock at 8:1 compression ratio. If the compression ratio were lowered to 7:1, the mep could be raised by supercharging along the 7:1 1 curve to 275 imep before it would be knock-limited by the same fuel. With a 5:1 compression ratio it could be raised to 435 imep. Thus the imep could be raised until the cylinder became thermally limited by the temperatures of critical parts, particularly of the piston head.

Engine balance

Rotating masses such as crank pins and the lower half of a connecting rod may be counterbalanced by weights attached to the crankshaft. The vibration which would result from the reciprocating forces of the pistons and their associated masses is usually minimized or eliminated by the arrangement of cylinders in a multicylinder engine so that the reciprocating forces in one cylinder are neutralized by those in another. Where these forces are in different planes, a corresponding pair of cylinders is required to counteract the resulting rocking couple.

If piston motion were truly harmonic, which would require a connecting rod of infinite length, the reciprocating inertia force at each end of the stroke would be as in Eq. (2),

$$F = 0.000456WN^2s \tag{2}$$

where *W* is the total weight of the reciprocating parts in one cylinder, *N* is the rpm, and *s* is the stroke in inches. Both *F* and *W* are in pounds. But the piston motion is not simple harmonic because the connecting rod is not infinite in length, and the piston travels more than half its stroke when the crankpin turns 90° from firing dead center. This distortion of the true harmonic motion is due to the so-called angularity *a* of the connecting rod, shown by Eq. (3),

$$a = \frac{r}{l} = \frac{s}{2l} \tag{3}$$

where *r* is the crank radius, *s* the stroke, and *l* the connecting rod length, all in inches.

Reciprocating inertia forces act in line with the cylinder axis and may be considered as combinations of a primary force—the true harmonic force from Eq. (2)—oscillating at the same frequency as the crankshaft rpm and a secondary force oscillating at twice this frequency having a value of *Fa*, which is added to the primary at top dead center and subtracted from it at bottom dead center. In general, harmonics above the second order may be disregarded. Therefore, for a connecting rod with the angularity $a = 0.291$, the inertia force caused by a piston at top dead center is about 1.29 times the pure harmonic force, and at bottom dead center it is about 0.71 times as large.

Where two pistons act on one crankpin, with the cylinders in 90° V arrangements, the resultant primary force is radial and of constant magnitude, and it rotates around the crankshaft with the crankpin. Therefore, it may be compensated for by an addition to the weight required to counterbalance the centrifugal force of the revolving crankpin and its associated masses. The resultant of the secondary force of the two pistons is 1.41 times as large as for one cylinder, and reciprocates in a horizontal plane through the crankshaft at twice crankshaft speed.

In four-cylinder inline engines with crankpins in the same plane, the primary reciprocating forces of the two inner pistons in cylinders 2 and 3 cancel those of the two outer pistons in cylinders 1 and 4, but the secondary forces from all pistons are added. Therefore, they are equivalent to the force resulting from a weight about 4*a* times the weight of one piston and its share of the connecting rod, oscillating parallel to the piston movement, having the same stroke, but moving at twice the frequency. A large a for this type of engine is advantageous. Where the four cylinders are arranged alternately on each side of a similar crankshaft, and in the same plane, both primary and secondary forces are in balance. Six cylinders in line also balance both primary and secondary forces.

V-8 engines use a crank arrangement in which the crankpins are in two planes 90° apart. Staggering the crankpins for pistons 1 and 2 90° from each other equalizes secondary forces, but the forces are in different planes. The couple this introduces is canceled by an opposite couple from the pistons operating on the crankpins for pistons 3 and 4.

Torsion dampers. In addition to vibrational forces from rotating and reciprocating masses, vibration may develop from torsional resonance of the crankshaft at various critical speeds. The longer the shaft for given bearing diameters, the lower the speeds at which these vibrations develop. On automotive engines, such vibrations are dampened by a viscous vibration damper or by a bonded-rubber vibration damper that is similar to a small flywheel coupled to the crankshaft through a rubber ring. The vibration damper may be combined with the pulley for an engine-accessory drive belt. *See also:* MECHANICAL VIBRATION.

Firing order. The firing order is the sequence in which the cylinders deliver their power impulses to the crankshaft. It is determined by such factors as engine design, ignition intervals, and crankshaft loading. Cylinder arrangements are generally selected for even firing intervals and torque impulses, as well as for balance.

Figure 8 shows various cylinder arrangements and firing orders that have been used in automotive engines. The Society of Automotive Engineers (SAE) Standard for engine rotation and cylinder numbering provides that standard rotation is counterclockwise rotation of the crankshaft as viewed from the principal output end of the engine. If power can be delivered from either end, rotation shall be as viewed from the flywheel end.

Excluding radial engines or those with coplanar cylinder bore axes, cylinders may be numbered by either of two methods: (1) In single or multibank engines, the cylinders are numbered in the sequence in which the connecting rods are mounted along the crankshaft, beginning with the cylinder farthest from the principal output end. (2) In multibank engines, the cylinders may be numbered in sequence in each bank, starting with the cylinder farthest from the principal output end and designated right or left bank by suffixes R and L, such as 1R and 1L.

Cylinder bank and accessory locations are described as right or left when the engine is viewed from the flywheel or principal output end.

Compression-ignition engines

In 1897, about 20 years after Otto first ran his engine, Rudolf Diesel successfully demonstrated an entirely different method of igniting fuel. Air is compressed to a pressure high enough for the adiabatic temperature to reach or exceed the ignition temperature of the fuel. Because this temperature is 1000°F (538°C) or higher, compression ratios of 12:1 to 23:1 are used commercially with compression pressures from about 440 to 800 psi

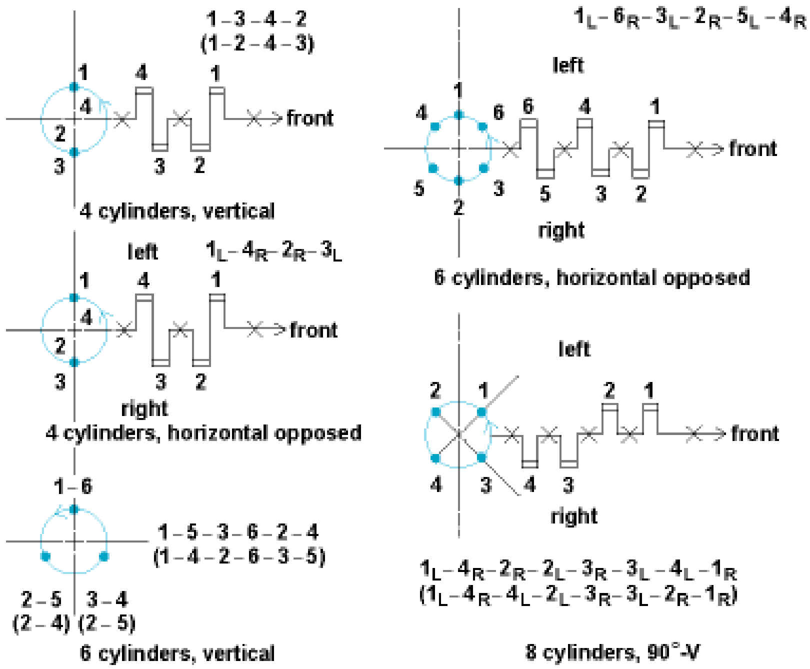

Fig. 8 **Various cylinder arrangements and firing orders.**

(3 to 5.5 megapascals). The fuel is injected into the cylinders shortly before the end of the compression stroke, at a time and rate suitable to control the rate of combustion.

Compression ratio and combustion. The idealized diesel engine cycle assumes combustion at constant pressure. Like the Otto cycle, thermal efficiency increases with compression ratio, but also varies with the amount of heat added (at the constant pressure) up to the cutoff point where the pressure begins to drop from adiabatic expansion. *See also:* DIESEL CYCLE; DIESEL ENGINE.

Fuel injection. Early diesel engines used air injection of the fuel to develop extremely fine atomization and a good distribution of the spray. But the need for injection air at pressures of about 1500 psi (10 MPa) required expensive and bulky multistage air compressors and intercoolers.

A simpler fuel-injection method was introduced by James McKechnie in 1910. He atomized the fuel as it entered the cylinder by use of high fuel pressure and suitable spray nozzles. After considerable development, it became possible to atomize the fuel sufficiently to minimize the smoky exhaust that had been characteristic of the early airless or solid-injection engines. By 1930, solid injection had become the generally accepted method of injecting fuel in diesel engines.

During the 1980s, electronically controlled fuel injection began replacing the mechanical system. Electronically controlled mechanically actuated unit injectors allowed injection pressure of 22,000 psi (150 MPa). *See also:* FUEL INJECTION.

Supercharged diesel engines. Combustion in a four-stroke diesel engine is improved by supercharging. Fuels that would smoke heavily and misfire at low loads will burn otherwise satisfactorily with supercharging. The indicated mean effective pressure rises directly with the supercharging pressure, until it is limited by the rate of heat flow from the metal parts surrounding the combustion chamber, and the resulting temperatures.

When superchargers of either the centrifugal or positive-displacement type are driven mechanically by the engine, the power required becomes an additional loss to the engine output. There is a degree of supercharge for any engine that develops maximum efficiency. A supercharge that is too high absorbs more power in the supercharger than is gained by the engine, especially at low loads. Another means of driving the supercharger is by an exhaust turbine, which recovers some of the energy that would otherwise be wasted in the exhaust. This may be accomplished with so small an increase of back pressure that little power is lost by the engine. The result is an appreciable increase in efficiency at loads high enough to develop the necessary exhaust pressure. *See also:* TURBOCHARGER.

Supercharging a two-cycle diesel engine requires some means of restricting or throttling the exhaust to build up cylinder pressure at the start of the compression stroke, and is used on a few large engines. Most medium and large two-stroke diesel engines are usually equipped with blowers to scavenge the cylinders after the working stroke and to supply the air required for the subsequent cycles. These blowers, in contrast to superchargers, do not build up appreciable pressure in the cylinder at the start of compression. If the capacity of such a blower is greater than the engine displacement, it will scavenge the cylinder of practically all exhaust products, even to the extent of blowing some air out through the exhaust ports. Such blowers, like superchargers, may be driven by the engine or by exhaust turbines.

Contrast between diesel and Otto engines

There are many characteristics of the diesel engine which are in direct contrast to those of the Otto engine. The higher the compression ratio of a diesel engine, the less the difficulties with ignition time lag. Too great an ignition lag results in a sudden and undesired pressure rise which causes an audible knock. In contrast to an Otto engine, knock in a diesel engine can be reduced by use of a fuel of higher cetane number, which is equivalent to a lower octane number. *See also:* CETANE NUMBER.

The larger the cylinder diameter of a diesel engine, the simpler the development of good combustion. In contrast, the smaller the cylinder diameter of the Otto engine, the less the limitation from detonation of the fuel.

High intake-air temperature and density materially aid combustion in a diesel engine, especially of fuels having low volatility and high viscosity. Some engines have not performed properly on heavy fuel until provided with a supercharger. The added compression of the supercharger raised the temperature and, what is more important, the density of the combustion air. For an Otto engine, an increase in either the air temperature or density increases the tendency of the engine to knock and therefore reduces the allowable compression ratio.

Diesel engines develop increasingly higher indicated thermal efficiency at reduced loads because of leaner fuel-air ratios and earlier cutoff. Such mixture ratios may be leaner than will ignite in an Otto engine. Furthermore, the reduction of load in an Otto engine requires throttling, which develops increasing pumping losses in the intake system.

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