Basics of the gasoline (SI) engine

The gasoline or spark-ignition (SI) internal-combustion engine uses the Otto cycle [1]) and externally supplied ignition. It burns an air/fuel mixture and in the process converts the chemical energy in the fuel into kinetic energy.

For many years, the carburetor was responsible for providing an air/fuel mixture in the intake manifold which was then drawn into the cylinder by the downgoing piston.

The breakthrough of gasoline fuel injection, which permits extremely precise metering of the fuel, was the result of the legislation governing exhaust-gas emission limits. Similar to the carburetor process, with manifold fuel injection the air/fuel mixture is formed in the intake manifold.

Even more advantages resulted from the development of gasoline direct injection, in particular with regard to fuel economy and increases in power output. Direct injection injects the fuel directly into the engine cylinder at exactly the right instant in time.

[1]) Named after Nikolaus Otto (1832–1891) who presented the first gas engine with compression using the 4-stroke principle at the Paris World Fair in 1878.

Method of operation

The combustion of the air/fuel mixture causes the piston (Fig. 1, Pos. 8) to perform a reciprocating movement in the cylinder (9). The name reciprocating-piston engine, or better still reciprocating engine, stems from this principle of functioning.

The conrod (10) converts the piston's reciprocating movement into a crankshaft (11) rotational movement which is maintained by a flywheel at the end of the crankshaft. Crankshaft speed is also referred to as engine speed or engine rpm.

Four-stroke principle

Today, the majority of the internal-combustion engines used as vehicle power plants are of the four-stroke type. The four-stroke principle employs gas-exchange valves (5 and 6) to control the exhaust-and-refill cycle. These valves open and close the cylinder's intake and exhaust passages, and in the process control the supply of fresh air/fuel mixture and the forcing out of the burnt exhaust gases.

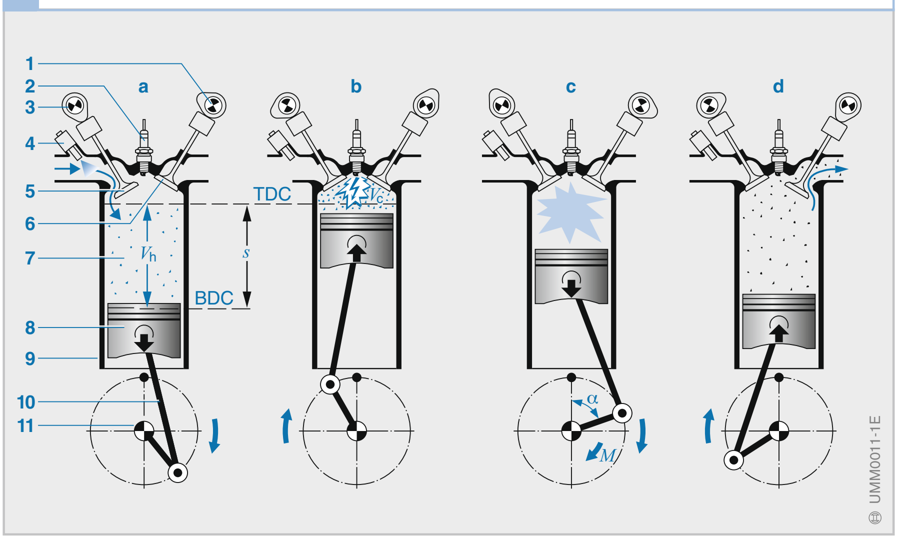
Fig. 1

a Induction stroke
b Compression stroke
c Power (combustion) stroke
d Exhaust stroke

1 Exhaust camshaft
2 Spark plug
3 Intake camshaft
4 Injector
5 Intake valve
6 Exhaust valve
7 Combustion chamber
8 Piston
9 Cylinder
10 Conrod
11 Crankshaft

M Torque
α Crankshaft angle
s Piston stroke
V_h Piston displacement
V_c Compression volume

1 Complete working cycle of the 4-stroke spark-ignition (SI) gasoline engine (example shows a manifold-injection engine with separate intake and exhaust camshafts)



K. Reif (Ed.), *Gasoline Engine Management,* Bosch Professional Automotive Information,
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1st stroke: Induction

Referred to Top Dead Center (TDC), the piston is moving downwards and increases the volume of the combustion chamber (7) so that fresh air (gasoline direct injection) or fresh air/fuel mixture (manifold injection) is drawn into the combustion chamber past the opened intake valve (5).

The combustion chamber reaches maximum volume ($V_h + V_c$) at Bottom Dead Center (BDC).

2nd stroke: Compression

The gas-exchange valves are closed, and the piston is moving upwards in the cylinder. In doing so it reduces the combustion-chamber volume and compresses the air/fuel mixture. On manifold-injection engines the air/fuel mixture has already entered the combustion chamber at the end of the induction stroke. With a direct-injection engine on the other hand, depending upon the operating mode, the fuel is first injected towards the end of the compression stroke.

At Top Dead Center (TDC) the combustion-chamber volume is at minimum (compression volume V_c).

3rd stroke: Power (or combustion)

Before the piston reaches Top Dead Center (TDC), the spark plug (2) initiates the combustion of the air/fuel mixture at a given ignition point (ignition angle). This form of ignition is known as externally supplied ignition. The piston has already passed its TDC point before the mixture has combusted completely.

The gas-exchange valves remain closed and the combustion heat increases the pressure in the cylinder to such an extent that the piston is forced downward.

4th stroke: Exhaust

The exhaust valve (6) opens shortly before Bottom Dead Center (BDC). The hot (exhaust) gases are under high pressure and leave the cylinder through the exhaust valve. The remaining exhaust gas is forced out by the upwards-moving piston.

A new operating cycle starts again with the induction stroke after every two revolutions of the crankshaft.

Valve timing

The gas-exchange valves are opened and closed by the cams on the intake and exhaust camshafts (3 and 1 respectively).
On engines with only 1 camshaft, a lever mechanism transfers the cam lift to the gas-exchange valves.

The valve timing defines the opening and closing times of the gas-exchange valves. Since it is referred to the crankshaft position, timing is given in "degrees crankshaft". Gas flow and gas-column vibration effects are applied to improve the filling of the combustion chamber with air/fuel mixture and to remove the exhaust gases. This is the reason for the valve opening and closing times overlapping in a given crankshaft angular-position range.

The camshaft is driven from the crankshaft through a toothed belt (or a chain or gear pair). On 4-stroke engines, a complete working cycle takes two rotations of the crankshaft. In other words, the camshaft only turns at half crankshaft speed, so that the step-down ratio between crankshaft and camshaft is 2:1.

2 Valve timing diagram for a four-stroke gasoline-engine

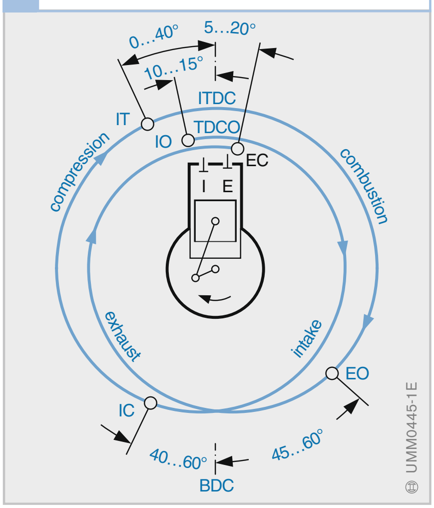


Fig. 2
- I Intake valve
- IO Intake valve opens
- IC Intake valve closes
- E Exhaust valve
- EO Exhaust valve opens
- EC Exhaust valve closes
- TDC Top Dead Center
- TDCO Overlap at TDC
- ITDC Ignition at TDC
- BDC Bottom Dead Center
- IT Ignition point

Compression

The difference between the maximum piston displacement V_h and the compression volume V_c is the compression ratio

$$\varepsilon = (V_h + V_c)/V_c.$$

The engine's compression ratio is a vital factor in determining

- Torque generation
- Power generation
- Fuel economy and
- Emissions of harmful pollutants

The gasoline-engine's compression ratio ε varies according to design configuration and the selected form of fuel injection (manifold or direct injection $\varepsilon = 7...13$). Extreme compression ratios of the kind employed in diesel powerplants ($\varepsilon = 14...24$) are not suitable for use in gasoline engines. Because the knock resistance of the fuel is limited, the extreme compression pressures and the high combustion-chamber temperatures resulting from such compression ratios must be avoided in order to prevent spontaneous and uncontrolled detonation of the air/fuel mixture. The resulting knock can damage the engine.

Air/fuel ratio

Complete combustion of the air/fuel mixture relies on a stoichiometric mixture ratio. A stoichiometric ratio is defined as 14.7 kg of air for 1 kg of fuel, that is, a 14.7 to 1 mixture ratio.

The air/fuel ratio λ (lambda) indicates the extent to which the instantaneous monitored air/fuel ratio deviates from the theoretical ideal:

$$\lambda = \frac{\text{induction air mass}}{\text{theoretical air requirement}}$$

The lambda factor for a stoichiometric ratio is λ 1.0. λ is also referred to as the excess-air factor.

Richer fuel mixtures result in λ figures of less than 1. Leaning out the fuel produces mixtures with excess air: λ then exceeds 1. Beyond a certain point the mixture encounters the lean-burn limit, beyond which ignition is no longer possible. The excess-air factor has a decisive effect on the specific fuel consumption (Fig. 3) and untreated pollutant emissions (Fig. 4).

Induction-mixture distribution in the combustion chamber

Homogeneous distribution

The induction systems on engines with manifold injection distribute a homogeneous air/fuel mixture throughout the combustion chamber. The entire induction charge has a single excess-air factor λ (Fig. 5a). Lean-burn engines, which operate on excess air under

3 Influence of the excess-air factor λ on the power P and on the specific fuel consumption b_e under conditions of homogeneous air/fuel-mixture distribution

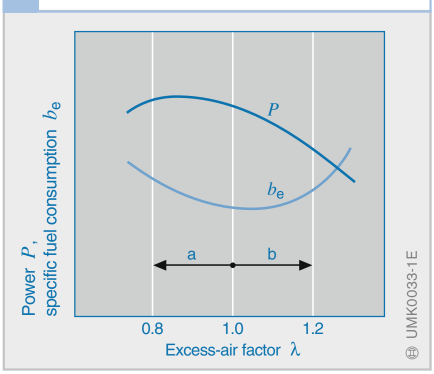
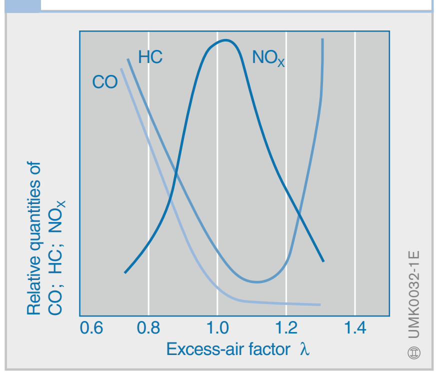


Fig. 3
a Rich air/fuel mixture (air deficiency)
b Lean air/fuel mixture (excess air)

4 Effect of the excess-air factor λ on the pollutant composition of untreated exhaust gas under conditions of homogeneous air/fuel-mixture distribution



specific operating conditions, also rely on homogeneous mixture distribution.

Stratified-charge concept

A combustible mixture cloud with $\lambda \approx 1$ surrounds the tip of the spark plug at the instant ignition is triggered. At this point the remainder of the combustion chamber contains either non-combustible gas with no fuel, or an extremely lean air/fuel charge. The corresponding strategy, in which the ignitable mixture cloud is present only in one portion of the combustion chamber, is the stratified-charge concept (Fig. 5b). With this concept, the overall mixture – meaning the average mixture ratio within the entire combustion chamber – is extremely lean (up to $\lambda \approx 10$). This type of lean operation fosters extremely high levels of fuel economy.

5 Induction-mixture distribution in the combustion chamber

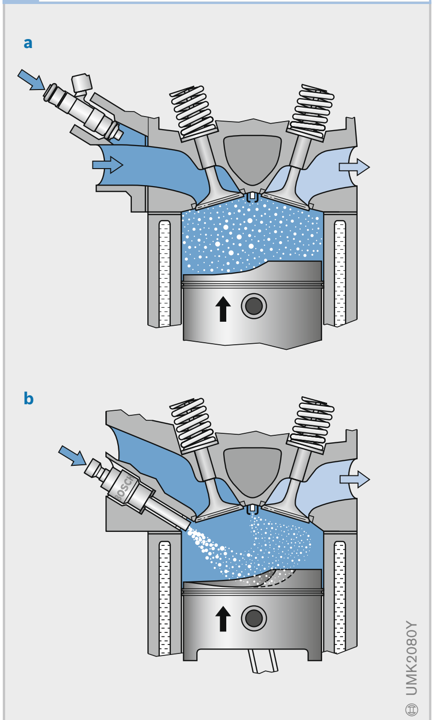


Fig. 5
a Homogeneous mixture distribution
b Stratified charge

Efficient implementation of the stratified-charge concept is impossible without direct fuel injection, as the entire induction strategy depends on the ability to inject fuel directly into the combustion chamber just before ignition.

Ignition and flame propagation

The spark plug ignites the air/fuel mixture by discharging a spark across a gap. The extent to which ignition will result in reliable flame propagation and secure combustion depends in large part on the air/fuel mixture λ, which should be in a range extending from λ = 0.75...1.3. Suitable flow patterns in the area immediately adjacent to the spark-plug electrodes can be employed to ignite mixtures as lean as $\lambda \leq 1.7$.

The initial ignition event is followed by formation of a flame-front. The flame front's propagation rate rises as a function of combustion pressure before dropping off again toward the end of the combustion process. The mean flame front propagation rate is on the order of 15...25 m/s.

The flame front's propagation rate is the combination of mixture transport and combustion rates, and one of its defining factors is the air/fuel ratio λ. The combustion rate peaks at slightly rich mixtures on the order of λ = 0.8...0.9. In this range it is possible to approach the conditions coinciding with an ideal constant-volume combustion process (refer to section on "Engine efficiency"). Rapid combustion rates provide highly satisfactory full-throttle, full-load performance at high engine speeds.

Good thermodynamic efficiency is produced by the high combustion temperatures achieved with air/fuel mixtures of λ = 1.05...1.1. However, high combustion temperatures and lean mixtures also promote generation of nitrous oxides (NO_X), which are subject to strict limitations under official emissions standards.

Cylinder charge

An air/fuel mixture is required for the combustion process in the cylinder. The engine draws in air through the intake manifolds (Fig. 1, Pos. 14), the throttle valve (13) ensuring that the air quantity is metered. The fuel is metered through fuel injectors. Furthermore, usually part of the burnt mixture (exhaust gas) from the last combustion is retained as residual gas (9) in the cylinder or exhaust gas is returned specifically to increase the residual-gas content in the cylinder (4).

Components of the cylinder charge

The gas mixture trapped in the combustion chamber when the intake valve closes is referred to as the cylinder charge. This is comprised of the fresh gas and the residual gas.

The term "relative air charge *rac*" has been introduced in order to have a quantity which is independent of the engine's displacement. It describes the air content in the cylinder and is defined as the ratio of the current air quantity in the cylinder to the air quantity that would be contained in the engine displacement under standard conditions ($p_0 = 1013\,\text{hPa}$, $T_0 = 273\,\text{K}$). Accordingly, there is a relative fuel quantity *rfq*; this is defined in such a way that identical values for *rac* and *rfq* result in $\lambda = 1$, i.e., $\lambda = rac/rfq$, or with specified λ : $rfq = rac/\lambda$.

1 Cylinder charge in a gasoline engine

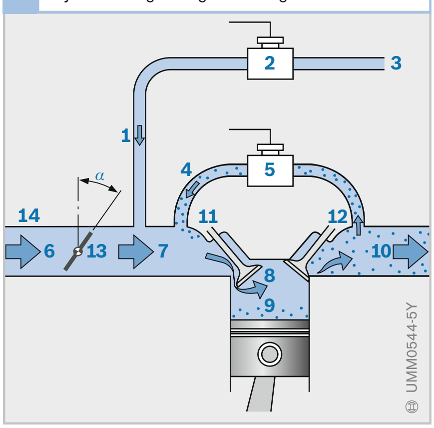


Fig. 1
1 Air and fuel vapor (from evaporative-emissions control system)
2 Canister-purge valve with variable valve-opening cross-section
3 Connection to evaporative-emissions control system
4 Returned exhaust gas
5 Exhaust-Gas Recirculation valve (EGR valve) with variable valve-opening cross-section
6 Air-mass flow (ambient pressure p_a)
7 Air-mass flow (manifold pressure p_m)
8 Fresh-gas charge (combustion-chamber pressure p_c)
9 Residual-gas charge (combustion-chamber pressure p_c)
10 Exhaust gas (exhaust-gas back pressure p_e)
11 Intake valve
12 Exhaust valve
13 Throttle valve
14 Intake manifold

α Throttle-valve angle

Fresh gas

The freshly introduced gas mixture in the cylinder is comprised of the fresh air drawn in and the fuel entrained with it. In a manifold-injection engine, all the fuel has already been mixed with the fresh air upstream of the intake valve. On direct-injection systems, on the other hand, the fuel is injected directly into the combustion chamber.

The majority of the fresh air enters the cylinder with the air-mass flow (Fig. 1, Pos. 6, 7) via the throttle valve (13). Additional fresh gas, comprising fresh air and fuel vapor, is directed to the cylinder via the evaporative-emissions control system (3, 2).

For homogeneous operation at $\lambda \leq 1$, the air in the cylinder directed via the throttle valve after the intake valve (11) has closed is the decisive quantity for the work at the piston during the combustion stroke and therefore for the engine's output torque. In this case, the air charge corresponds to the torque and the engine load. Here, changing the throttle-valve angle only indirectly leads to a change in the air charge. First of all, the pressure in the intake manifold must rise so that a greater air mass flows into the cylinder via the intake valves. Fuel can, on the other hand, be injected more contemporaneously with the combustion process and metered precisely to the individual cylinder. Therefore the injected fuel quantity is dependent on the current air quantity, and the gasoline engine is an air-directed system in "conventional" homogeneous mode at $\lambda \leq 1$.

During lean-burn operation (stratified charge), however, the torque (engine load) – on account of the excess air – is a direct product of the injected fuel mass. The air mass can thus differ for the same torque. The gasoline engine is therefore fuel-directed during lean-burn operation.

Almost always, measures aimed at increasing the engine's maximum torque and maximum output power necessitate an increase in the maximum possible fresh-gas charge. This can be achieved by increasing the engine displacement but also by supercharging (see section entitled "Supercharging").

Residual gas

The residual-gas share of the cylinder charge comprises that portion of the cylinder charge which has already taken part in the combustion process. In principle, one differentiates between internal and external residual gas. Internal residual gas is the exhaust gas which remains in the upper clearance volume of the cylinder after combustion or which, while the intake and exhaust valves are simultaneously open (valve overlap, see section entitled "Gas exchange"), is drawn from the exhaust port back into the intake manifold (internal exhaust-gas recirculation).

External residual gas is exhaust gas which is introduced via an exhaust-gas recirculation valve (Fig. 1, Pos. 4, 5) into the intake manifold (external exhaust-gas recirculation).

The residual gas is made up of inert gas [1)] and – in the event of excess air, i.e., during lean-burn operation – of unburnt air. The amount of inert gas in the residual gas is particularly important. This no longer contains any oxygen and therefore does not participate in combustion during the following power cycle. However, it does delay ignition and slows down the course of combustion, which results in slightly lower efficiency but also in lower peak pressures and temperatures. In this way, a specifically used amount of residual gas can reduce the emission of nitrogen oxides (NO_X). This then is the benefit of inert gas in lean-burn operation in that the three-way catalytic converter is unable to reduce the nitrogen oxides in the event of excess air.

In homogeneous engine mode, the fresh-gas charge displaced by the residual gas (consisting in this case of inert gas only) is compensated by means of a greater opening of the throttle valve. With a constant fresh-gas charge, this increases the intake-manifold pressure, therefore reduces the throttling losses (see section entitled "Gas exchange"), and in all results in reduced fuel consumption.

Gas exchange

The process of replacing the consumed cylinder charge (exhaust gas, also referred to in the above as residual gas) with fresh gas is known as gas exchange or the charge cycle. It is controlled by the opening and closing of the intake and exhaust valves in combination with the piston stroke. The shape and position of the camshaft cams determine the progression of the valve lift and thereby influence the cylinder charge.

The opening and closing times of the valves are called valve timing and the maximum distance a valve is lifted from its seat is known as the valve lift or valve stroke. The characteristic variables are Exhaust Opens (EO), Exhaust Closes (EC), Intake Opens (IO), Intake Closes (IC) and the valve lift. There are engines with fixed and others with variable timing and valve lifts (see chapter entitled "Cylinder-charge control systems").

The amount of residual gas for the following power cycle can be significantly influenced by a valve overlap. During the valve overlap, intake and exhaust valves are simultaneously open for a certain amount of time, i.e., the intake valve opens before the exhaust valve closes. If in the overlap phase the pressure in the intake manifold is lower than that in the exhaust train, the residual gas flows back into the intake manifold; because the residual gas drawn back in this way is drawn in again after Exhaust Closes, this results in an increase in the residual-gas content.

[1)] Components in the combustion chamber which behave inertly, that is, do not participate in the combustion process.

In the case of supercharging, the pressure before the intake valve can also be higher during the overlap phase; in this event, the residual gas flows in the direction of the exhaust train such that it is properly cleared away ("scavenging") and it is also possible for the air to flow through into the exhaust train.

When the residual gas is successfully scavenged, its volume is then available for an increased fresh-gas charge. The scavenging effect is therefore used to increase torque in the lower speed range (up to approx. 2000 rpm), either in combination with dynamic supercharging in naturally aspirated engines or with turbocharging.

Volumetric efficiency and air consumption

The success of the gas-exchange process is measured in the variables volumetric efficiency, air consumption and retention rate. The volumetric efficiency is the ratio of the fresh-gas charge actually remaining in the cylinder to the theoretically maximum possible charge. It differs from the relative air charge in that the volumetric efficiency is referred to the external conditions at the time of measurement and not to standard conditions.

The air consumption describes the total air-mass throughput during the gas-exchange process, likewise referred to the theoretically maximum possible charge. The air consumption can also include the air mass which is transferred directly into the exhaust train during the valve overlap. The retention rate, the ratio of volumetric efficiency to air consumption, specifies the proportion of the air-mass throughput which remains in the cylinder at the end of the gas-exchange process.

The maximum volumetric efficiency for naturally aspirated engines is 0.6...0.9. It depends on the combustion-chamber shape, the opened cross-sections of the gas-exchange valves, and the valve timing.

Pumping losses

Work is expended in the form of pumping losses or gas-exchange losses in order to replace the exhaust gas with fresh gas in the gas-exchange process. These losses use up part of the mechanical work generated and therefore reduce the effective efficiency of the engine. In the intake phase, i.e., during the downward stroke of the piston, the intake-manifold pressure in throttled mode is less than the ambient pressure and in particular the pressure in the piston return chamber. The piston must work against this pressure differential (throttling losses).

A dynamic pressure occurs in the combustion chamber during the upward stroke of the piston when the burnt gas is emitted, particularly at high engine speeds and loads; the piston must expend energy in order to overcome this pressure (push-out losses).

If with gasoline direct injection stratified-charge operation is used with the throttle valve fully opened or high exhaust-gas recirculation is used in homogeneous operation ($\lambda \leq 1$), this increases the intake-manifold pressure and reduces the pressure differential above the piston. In this way, the engine's throttling losses can be reduced, which in turn improves the effective efficiency.

Supercharging

The torque which can be achieved during homogenous operation at $\lambda \leq 1$ is proportional to the fresh-gas charge. This means that maximum torque can be increased by compressing the air before it enters the cylinder (supercharging). This leads to an increase in volumetric efficiency to values above 1.

Dynamic supercharging

Supercharging can be achieved simply by taking advantage of the dynamic effects inside the intake manifold. The supercharging level depends on the intake manifold's design and on its operating point (for the most part, on engine speed, but also on cylinder charge). The possibility of changing the intake-manifold geometry while the engine is running (variable intake-manifold geometry) means that dynamic supercharging can be applied across a wide operating range to increase the maximum cylinder charge.

Mechanical supercharging

The intake-air density can be further increased by compressors which are driven mechanically from the engine's crankshaft. The compressed air is forced through the intake manifold and into the engine's cylinders.

Exhaust-gas turbocharging

In contrast mechanical supercharging, the compressor of the exhaust-gas turbocharger is driven by an exhaust-gas turbine located in the exhaust-gas flow, and not by the engine's crankshaft. This enables recovery of some of the energy in the exhaust gas.

Charge recording

In a gasoline engine with homogeneous $\lambda = 1$ operation, the injected fuel quantity is dependent on the air quantity. This is necessary because after a change to the throttle-valve angle the air charge changes only gradually while the fuel quantity can be varied from injection to injection.

For this reason, the current available air charge must be determined for each combustion in the engine-management system (charge recording). There are essentially three systems which can be used to record the charge:

- A hot-film air-mass meter (HFM) measures the air-mass flow into the intake manifold.
- A model is used to calculate the air-mass flow from the temperature before the throttle valve, the pressure before and after the throttle valve, and the throttle-valve angle (throttle-valve model, α/n system [1])).
- A model is used to calculate the charge drawn in by the cylinder from the engine speed *(n)*, the pressure *(p)* in the intake manifold (i.e., before the intake valve), the temperature in the intake passage and further additional information (e.g., camshaft/valve-lift adjustment, intake-manifold changeover, position of the swirl control valve) (p/n system). Sophisticated models may be necessary, depending on the complexity of the engine, particularly with regard to the variabilities of the valve gear.

Because only the mass flow passing into the intake manifold can be determined with a hot-film air-mass meter or a throttle-valve model, both these systems only provide a cylinder-charge value during stationary engine operation. Stationary means at constant intake-manifold pressure; because then the mass flows flowing into the intake manifold and off into the engine are identical.

In the event of a sudden load change (change in the throttle-valve angle), the inflowing mass flow changes spontaneously, while the off-flowing mass flow and with it the cylinder charge only change if the intake-manifold pressure has increased or reduced. The accumulator behavior of the intake manifold must therefore also be imitated *(intake-manifold model)*.

[1]) The designation α/n system is historically conditioned since originally the pressure after the throttle valve was not taken into account and the mass flow was stored in a program map covering throttle-valve angle and engine speed. This simplified approach is sometimes still used today.

Torque and power

Torques at the drivetrain

The power P delivered by a gasoline engine is defined by the available clutch torque M and the engine speed n. The clutch torque is the torque developed by the combustion process less friction torque (friction losses in the engine), pumping losses, and the torque needed to drive the auxiliary equipment (Fig. 1). The drive torque is derived from the clutch torque plus the losses arising at the clutch and transmission.

The combustion torque is generated in the power cycle and is determined in engines with manifold injection by the following variables:

- The air mass which is available for combustion when the intake valves close
- The fuel mass which is available at the same moment, and
- The moment in time when the ignition spark initiates the combustion of the air/fuel mixture

Direct-injection gasoline engines function at certain operating points with excess air (lean-burn operation). The cylinder thus contains air, which has no effect on the generated torque. Here, it is the fuel mass which has the most effect.

Generation of torque

The physical quantity torque M is the product of force F times lever arm s:

$$M = F \cdot s$$

The connecting rod utilizes the throw of the crankshaft to convert the piston's linear travel into rotary motion. The force with which the expanding air/fuel mixture drives the piston down the cylinder is converted into torque by the lever arm generated by the throw.

The lever arm l which is effective for the torque is the lever component vertical to the force (Fig. 2). The force and the leverage angle are parallel at Top Dead Center (TDC).

1 Torques at the drivetrain

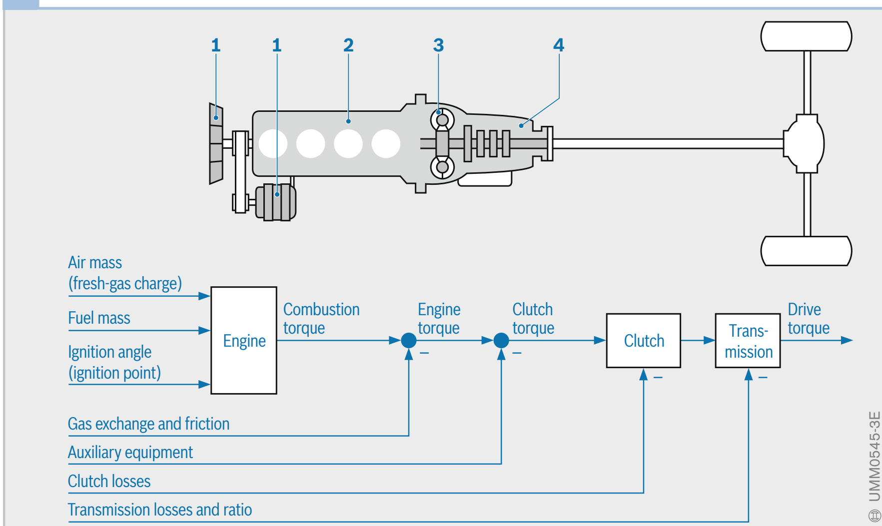


Fig. 1
1 Auxiliary equipment (A/C compressor, alternator, etc.)
2 Engine
3 Clutch
4 Transmission

This results in an effective lever arm of zero. The ignition angle must be selected in such a way as to trigger mixture ignition while the crankshaft is rotating through a phase of increasing lever arm (0...90 °crankshaft). This enables the engine to generate the maximum possible torque. The engine's design (for instance, piston displacement, combustion-chamber geometry, volumetric efficiency, charge) determines the maximum possible torque M that it can generate.

Essentially, the torque is adapted to the requirements of actual driving by adjusting the quality and quantity of the air/fuel mixture and the ignition angle. Fig. 3 shows the typical torque and power curves, plotted against engine speed, for a manifold-injection gasoline engine. As engine speed increases, full-load torque initially increases to its maximum M_{max}. At higher engine speeds, torque falls off again as the shorter opening times of the intake valves limits the cylinder charge.

Engine designers focus on attempting to obtain maximum torque at low engine speeds of around 2000 rpm. This rpm range coincides with optimal fuel economy. Engines with exhaust-gas turbochargers are able to meet these requirements.

Relationship between torque and power

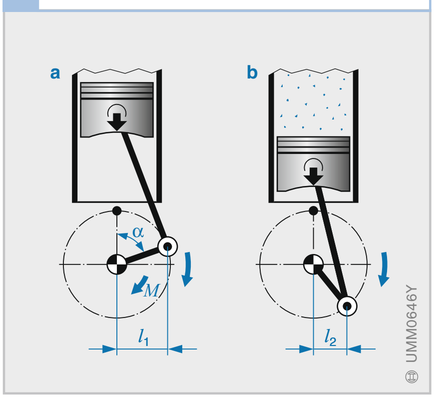
The engine's power output P climbs along with increasing torque M and engine speed n. The following applies:

$$P = 2 \cdot \pi \cdot M \cdot n$$

Engine power increases until it reaches its peak value at rated speed n_{rat} with rated power P_{rat}. Owing to the substantial decrease in torque, power generation drops again at extremely high engine speeds.

A transmission to vary conversion ratios is needed to adapt the gasoline engine's torque and power curves to meet the requirements of vehicle operation.

2 Lever arm for crankshaft drive



3 Torque and power curves

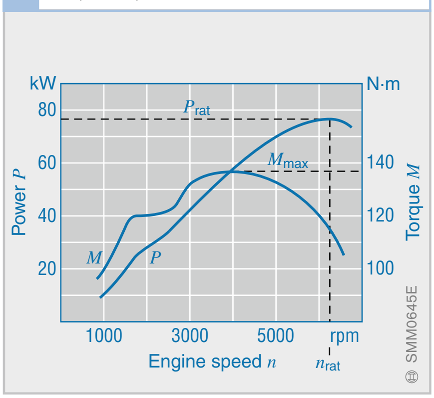


Fig. 2
Changing the effective lever arm during the power cycle
a Increasing lever arm l_1
b Decreasing lever arm l_2

Fig. 3
Typical curves for a manifold-injection gasoline engine

Engine efficiency

Thermal efficiency

The internal-combustion engine does not convert all the energy which is chemically available in the fuel into mechanical work, and some of the added energy is lost. This means that an engine's efficiency is less than 100 % (Fig. 1). Thermal efficiency is one of the important links in the engine's efficiency chain.

Pressure-volume diagram (*p-V* diagram)

The *p-V* diagram is used to display the pressure and volume conditions during a complete working cycle of the 4-stroke IC engine.

The ideal cycle

Figure 2 (curve A) shows the compression and power strokes of an ideal process as defined by the laws of Boyle/Mariotte and Gay-Lussac. The piston travels from BDC to TDC (point 1 to point 2), and the air/fuel mixture is compressed without the addition of heat (Boyle/Mariotte). Subsequently, the mixture burns accompanied by a pressure rise (point 2 to point 3) while volume remains constant (Gay-Lussac).

From TDC (point 3), the piston travels towards BDC (point 4), and the combustion-chamber volume increases. The pressure of the burnt gases drops whereby no heat is released (Boyle/Mariotte). Finally, the burnt mixture cools off again with the volume remaining constant (Gay-Lussac) until the initial status (point 1) is reached again.

The area inside the points 1 – 2 – 3 – 4 shows the work gained during a complete working cycle. The exhaust valve opens at point 4 and the gas, which is still under pressure, escapes from the cylinder. If it were possible for the gas to expand completely by the time point 5 is reached, the area described by 1 – 4 – 5 would represent usable energy. On an exhaust-gas-turbocharged engine, the part above the atmospheric line (1 bar) can to some extent be utilized (1 – 4 – 5′).

Real p-V diagram

Since it is impossible during normal engine operation to maintain the basic conditions for the ideal cycle, the actual *p-V* diagram (Fig. 2, curve B) differs from the ideal *p-V* diagram.

Measures for increasing thermal efficiency

The thermal efficiency rises along with increasing air/fuel-mixture compression. The higher the compression, the higher the pressure in the cylinder at the end of the compression phase, and the larger is the enclosed area in the *p-V* diagram. This area is an indication of the energy generated during the combustion process. When selecting the compression ratio, the fuel's antiknock qualities must be taken into account.

Manifold-injection engines inject the fuel into the intake manifold onto the closed intake valve, where it is stored until drawn into the cylinder. During the formation of the air/fuel mixture, the fine fuel droplets vaporize. The energy needed for this process is in the form of heat and is taken from the air and the intake-manifold walls. On direct-injection engines the fuel is injected into the combustion chamber, and the energy needed for fuel-droplet vaporization is taken from the air trapped in the cylinder which cools off as a result. This means that the compressed air/fuel mixture is at a lower temperature than is the case with a manifold-injection engine, so that a higher compression ratio can be chosen.

Thermal losses

The heat generated during combustion heats up the cylinder walls. Part of this thermal energy is radiated and lost. In the case of gasoline direct injection, the stratified-charge air/fuel mixture cloud is surrounded by a jacket of gases which do not participate in the combustion process. This gas jacket hinders the transfer of heat to the cylinder walls and therefore reduces the thermal losses.

Further losses stem from the incomplete combustion of the fuel which has condensed onto the cylinder walls. Thanks to the insulating effects of the gas jacket, these losses are reduced in stratified-charge operation. Further thermal losses result from the residual heat of the exhaust gases.

Losses at $\lambda = 1$

The efficiency of the constant-volume cycle climbs along with increasing excess-air factor (λ). Due to the reduced flame-propagation velocity common to lean air/fuel mixtures, at $\lambda > 1.1$ combustion is increasingly sluggish, a fact which has a negative effect upon the SI engine's efficiency curve. In the final analysis, efficiency is the highest in the range λ = 1.1...1.3. Efficiency is therefore less for a homogeneous air/fuel-mixture formation with λ = 1 than it is for an air/fuel mixture featuring excess air. When a 3-way catalytic converter is used for emissions control, an air/fuel mixture with $\lambda = 1$ is absolutely imperative for efficient operation.

Pumping losses

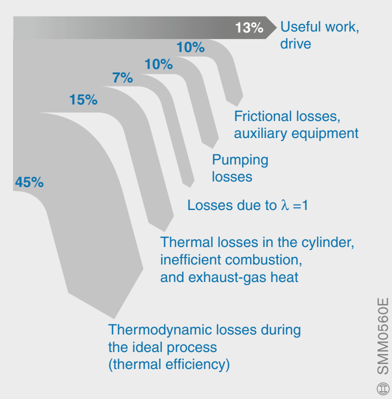
During the exhaust and refill cycle, the engine draws in fresh gas during the 1st (induction) stroke. The desired quantity of gas is controlled by the throttle-valve opening.
A vacuum is generated in the intake manifold which opposes engine operation (throttling losses). Since with a gasoline direct-injection engine the throttle valve is wide open at idle and part load, and the torque is determined by the injected fuel mass, the pumping losses (throttling losses) are lower.

In the 4th stroke, work is also involved in forcing the remaining exhaust gases out of the cylinder.

Frictional losses

The frictional losses are the total of all the friction between moving parts in the engine itself and in its auxiliary equipment. For instance, due to the piston-ring friction at the cylinder walls, the bearing friction, and the friction of the alternator drive.

1 Efficiency chain of an SI engine at $\lambda = 1$



2 Sequence of the motive working process in the p-V diagram

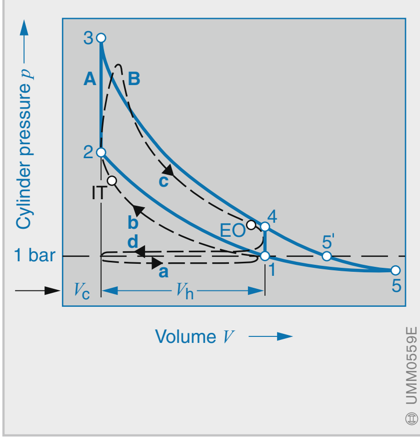


Fig. 2
A Ideal constant-volume cycle
B Real p-V diagram

a Induction
b Compression
c Work (combustion)
d Exhaust

IT Ignition point
EO Exhaust valve opens

Specific fuel consumption

Specific fuel consumption b_e is defined as the mass of the fuel (in grams) that the internal-combustion engine requires to perform a specified amount of work (kW · h, kilowatt hours). This parameter thus provides a more accurate measure of the energy extracted from each unit of fuel than the terms liters per hour, litres per 100 kilometers or miles per gallon.

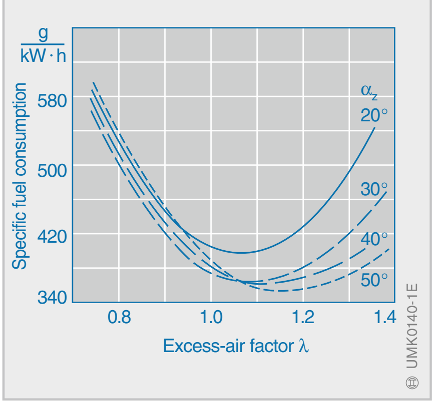
Effects of excess-air factor

Homogeneous mixture distribution

When engines operate on homogeneous induction mixtures, specific fuel consumption initially responds to increases in excess-air factor λ by falling (Fig. 1). The progressive reductions in the range extending to $\lambda = 1.0$ are explained by the incomplete combustion that results when a rich air/fuel mixture burns with inadequate air.

The throttle plate must be opened to wider apertures to obtain a given torque during operation in the lean range ($\lambda > 1$). The resulting reduction in throttling losses combines with enhanced thermodynamic efficiency to furnish lower rates of specific fuel consumption.

1 Effects of excess-air factor λ and ignition timing α_z on fuel consumption during operation with homogeneous mixture distribution



As the excess-air factor is increased, the flame front's propagation rate falls in the resulting, progressively leaner mixtures.
The ignition timing must be further advanced to compensate for the resulting lag in ignition of the combustion mixture.

As the excess-air factor continues to rise, the engine approaches the lean-burn limit, where incomplete combustion takes place (combustion miss). This results in a radical increase in fuel consumption. The excess-air factor that coincides with the lean-burn limit varies according to engine design.

Stratified-charge concept

Engines featuring direct gasoline injection can operate with high excess-air factors in their stratified-charge mode. The only fuel in the combustion chamber is found in the stratification layer immediately adjacent to the tip of the spark plug. The excess-air factor within this layer is approximately $\lambda = 1$.

The remainder of the combustion chamber is filled with air and inert gases (exhaust-gas recirculation). The large throttle-plate apertures available in this mode lead to a reduction in pumping losses. This combines with the thermodynamic benefits to provide a substantial reduction in specific fuel consumption.

Effects of ignition timing

Homogeneous mixture distribution

Each point in the cycle corresponds to an optimal phase in the combustion process with its own defined ignition timing (Fig. 1). Any deviation from this ignition timing will have negative effects on specific fuel consumption.

Stratified-charge concept

The range of possibilities for varying the ignition angle is limited on direct-injection gasoline engines operating in the stratified-charge mode. Because the ignition spark must be triggered as soon as the mixture cloud reaches the spark plug, the ideal ignition point is largely determined by injection timing.

Achieving ideal fuel consumption

During operation on homogeneous induction mixtures, gasoline engines must operate on a stoichiometric air/fuel ratio of $\lambda = 1$ to create an optimal operating environment for the 3-way catalytic converter. Under these conditions using the excess-air factor to manipulate specific fuel consumption is not an option. Instead, the only available recourse is to vary the ignition timing. Defining ignition timing always equates with finding the best compromise between maximum fuel economy and minimal levels of raw exhaust emissions. Because the catalytic converter's treatment of toxic emissions is very effective once it is hot, the aspects related to fuel economy are the primary considerations once the engine has warmed to normal operating temperature.

Fuel-consumption map

Testing on an engine dynamometer can be used to determine specific fuel consumption in its relation to brake mean effective pressure and to engine speed. The monitored data are then entered in the fuel consumption map (Fig. 2). The points representing levels of specific fuel consumption are joined to form curves. Because the resulting graphic portrayal resembles a sea shell, the lines are also known as shell or conchoid curves.

As the diagram indicates, the point of minimum specific fuel consumption coincides with a high level of brake mean effective pressure p_{me} at an engine speed of roughly 2600 rpm.

Because the brake mean effective pressure also serves as an index of torque generation M, curves representing power output P can also be entered in the chart. Each curve assumes the form of a hyperbola. Although the chart indicates identical power at different engine speeds and torques (operating points A and B), the specific fuel consumption rates at these operating points are not the same. At Point B the engine speed is lower and the torque is higher than at Point A. Engine operation can be shifted toward Point A by using the transmission to select a gear with a higher conversion ratio.

2 Fuel-consumption map for gasoline engine with homogeneous induction mixture

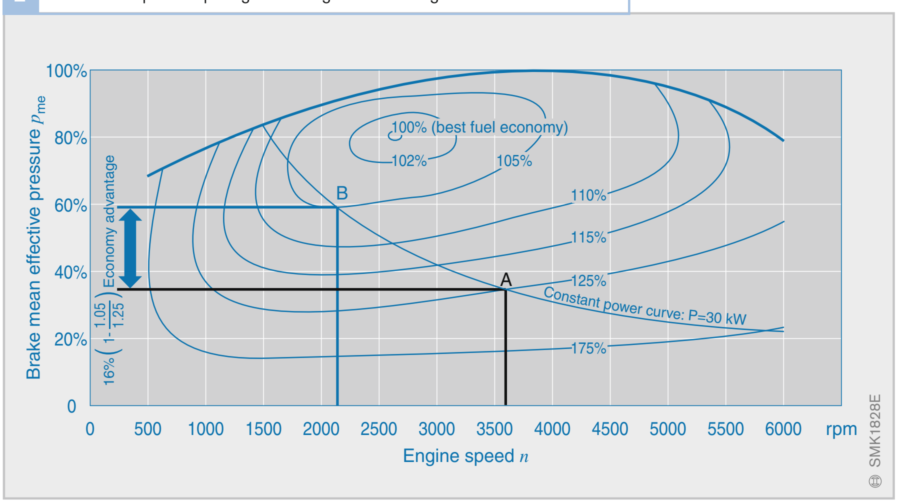


Fig. 2
Engine data:
4-cylinder gasoline engine
Displacement:
$V_H = 2.3$ litres
Power:
$P = 110$ kW at 5400 rpm
Torque peak:
$M = 220$ N·m at 3700...4500 rpm
Brake mean effective pressure:
$p_{me} = 12$ bar (100 %)

Calculating torque M and power P with numerical value equations:
$M = V_H \cdot p_{me} / 0.12566$
$P = M \cdot n / 9549$

M in N·m
V_H in dm³
p_{me} in bar
n in rpm
P in kW

Combustion knock

Among the factors imposing limits on the latitude for enhancing an engine's thermodynamic efficiency and increasing power-plant performance are spontaneous pre-ignition and detonation. This highly undesirable phenomenon is frequently accompanied by an audible "pinging" noise, which is why the generally applicable term for this condition is "knock". Knock occurs when portions of the mixture ignite spontaneously before being reached by the flame front. The intense heat and immense pressure peaks produced by combustion knock subject pistons, bearings, cylinder head and head gasket to enormous mechanical and thermal loads. Extended periods of knock can produce blown head gaskets, holed piston crowns and engine seizure, and leads to destruction of the engine.

The sources of combustion knock

The spark plug ignites the air/fuel mixture toward the end of the compression stroke, just before the piston reaches Top Dead Centre (TDC). Because several milliseconds can elapse until the entire air/fuel mixture can ignite (the precise ignition lag varies according to engine speed), the actual combustion peak occurs after TDC.

The flame front extends outward from the spark plug. After being compressed during the compression stroke, the induction mixture is heated and pressurized as it burns within the combustion chamber. This further compresses any unburned air/fuel mixture within the chamber. As a result, some portions of the compressed air/fuel mixture can attain temperatures high enough to induce spontaneous auto-ignition (Fig. 1). Sudden detonation and uncontrolled combustion are the results.

When this type of detonation occurs it produces a flame front with a propagation rate 10 to 100 times that associated with the normal combustion triggered by the spark plug (approximately 20 m/s). This uncontrolled combustion generates pressure pulses which spread out in circular patterns from the core of the process. It is when these pulsations impact against the walls of the cylinder that they generate the metallic pinging sound typically associated with combustion knock.

Other flame fronts can be initiated at hot spots within the combustion chamber. Among the potential sources of this hot-spot ignition are spark plugs which during operation heat up excessively due to their heat range being too low. This type of pre-ignition produces engine knock by initiating combustion before the ignition spark is triggered.

Engine knock can occur throughout the engine's speed range. However, it is not possible to hear it at extremely high rpm, when its sound is obscured by the noise from general engine operation.

Factors affecting tendency to knock

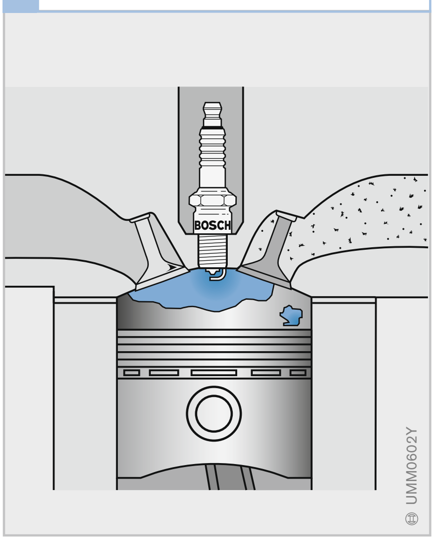
Substantial ignition advance: Advancing the timing to ignite the mixture earlier produces progressively higher combustion-chamber temperatures and correspondingly extreme pressure rises.
High cylinder-charge density: The charge density must increase as torque demand rises (engine load factor). This leads to high temperatures during compression.
Fuel grade: Because fuels with low octane ratings furnish only limited resistance to knock, compliance with manufacturer's specifications for fuel grade(s) is vital.
Excessively high compression ratio: One potential source of excessively high compression would be a cylinder head gasket of less than the specified thickness. This leads to higher pressures and temperatures in the air/fuel mixture during compression. Deposits and residue in the combustion chamber (from ageing, etc.) can also produce a slight increase in the effective compression ratio.
Cooling: Ineffective heat dissipation within the engine can lead to high mixture temperatures within the combustion chamber.
Geometry: The engine's knock tendency can be aggravated by unfavorable combustion-chamber geometry. Poor turbulence and swirl characteristics caused by unsatisfactory intake-manifold tract configurations are yet another potential problem source.

Engine knock with direct gasoline injection

With regard to engine knock, when operating with homogeneous air/fuel mixtures direct-injection gasoline engines behave the same as manifold-injected power plants. One major difference is the cooling effect exerted by the evaporating fuel during direct injection, which reduces the temperature of the air within the cylinder to levels lower than those encountered with manifold injection.

During operation in the stratified-charge mode it is only in the area immediately adjacent to the spark plug tip that an ignitable mixture is present. When the remainder of the combustion chamber is filled with air or inert gases, there is no danger of spontaneous ignition and engine knock. Nor is there any danger of detonation when an extremely lean air/fuel mixture is present within these outlying sections of the combustion chamber. The ignition energy required to generate a flame in this kind of lean mixture would be substantially higher than that needed to spark a stoichiometric combustion mixture. This is why stratified-charge operation effectively banishes the danger of engine knock.

1 The sources of combustion knock



Avoiding consistent engine knock

To effectively avoid pre-ignition and detonation, ignition systems not equipped with knock detection rely on ignition timing with a safety margin of 5...8 degrees (crankshaft) relative to the knock limit.

Ignition systems featuring knock detection employ one or several knock sensors to monitor acoustic waves in the engine. The engine-management ECU detects knock in individual combustion cycles by analysing the electrical signals relayed by these sensors. The ECU then responds by retarding the ignition timing for the affected cylinder to prevent continuous knock. The system then gradually advances the ignition timing back toward its original position. This progressive advance process continues until the ignition timing is either back at the initial reference point programmed into the engine's software map, or until the system starts to detect knock again. The engine management regulates the timing advance for each cylinder individually.

The limited number of combustion events with mild knock of the kind that also occur with knock control are not injurious to the health of the engine. On the contrary: They help dissolve deposits formed by oil and fuel additives within the combustion chamber (on intake and exhaust valves, etc.), allowing them to be combusted and/or discharged with the exhaust gases.

Advantages of knock control

Thanks to reliable knock recognition, engines with knock control can use higher compression ratios. Co-ordinated control of the ignition's timing advance also makes it possible to do without the safety margin between the timing point and the knock threshold; the ignition timing can be selected for the "best case" instead of the "worst case" scenario. This provides benefits in terms of thermodynamic efficiency. Knock control

- reduces fuel consumption,
- enhances torque and power, and
- allows engine operation on different fuels within an extended range of octane ratings (both premium and regular unleaded, etc.).