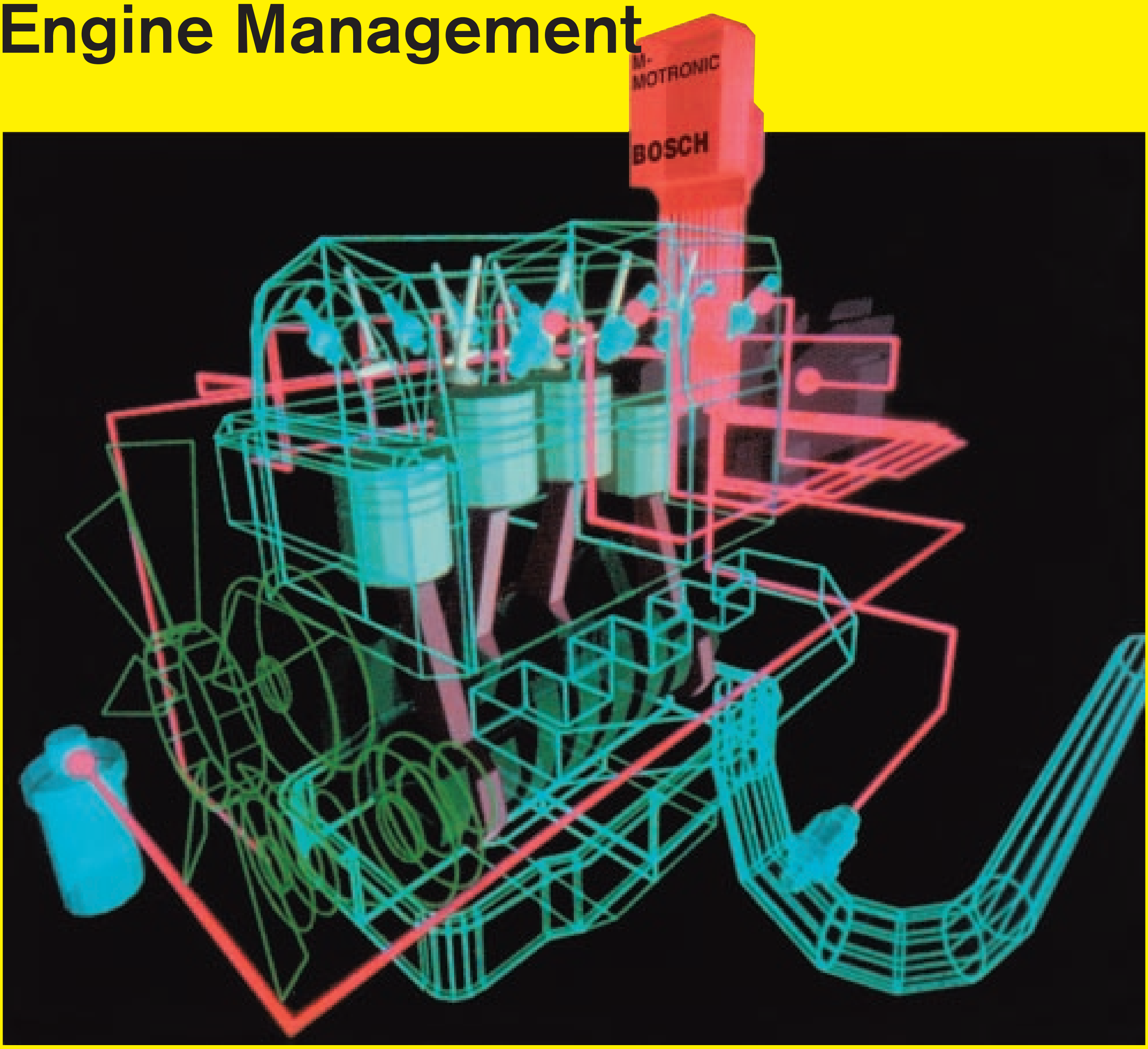
Gasoline-engine management

M-Motronic Engine Management



Technical Instruction

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M-Motronic
Engine Management

Modern electronics are opening up new perspectives in automotive design. The spark-ignition engine is being subjected to numerous, sometimes mutually-antagonistic demands. It is now possible to satisfy these demands – including high specific output, modest fuel consumption and low exhaust emissions – by using systems providing an optimal combination of operating characteristics.

Separate mixture-formation and ignition systems deal with parts of the problem: Jetronic controls fuel supply while the electronic ignition system provides optimal ignition control.

Motronic combines the two systems. A computer controls the injection and ignition systems with reference to shared optimization criteria.

Digital data processing and microprocessors make it possible to translate extensive operating information into program-map-controlled injection and ignition data.

Installation of a Lambda oxygen sensor and integration of a Lambda control unit in the CPU allow Motronic to meet tomorrow's emissions regulations today.

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Combustion in the gasoline engine

The spark-ignition or Otto-cycle engine

Operating concept

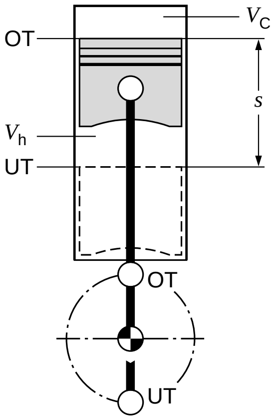
The spark-ignition or Otto-cycle[1]) powerplant is an internal-combustion (IC) engine that relies on an externally-generated ignition spark to transform the chemical energy contained in fuel into kinetic energy.

Today's standard spark-ignition engines employ manifold injection for mixture formation outside the combustion chamber. The mixture formation system produces an air/fuel mixture (based on gasoline or a gaseous fuel), which is then drawn into the engine by the suction generated as the pistons descend. The future will see increasing application of systems that inject the fuel directly into the combustion chamber as an alternate concept. As the piston rises, it compresses the mixture in preparation for the timed ignition process, in which externally-generated energy initiates combustion via the spark plug. The heat released in the combustion process pressurizes the cylinder, propelling the piston back down, exerting force against the crankshaft and performing work. After each combustion stroke the spent gases are expelled from the cylinder in preparation for ingestion of a fresh charge of air/fuel mixture. The primary design concept used to govern this gas transfer in powerplants for automotive applications is the four-stroke principle, with two crankshaft revolutions being required for each complete cycle.

Fig. 1

Reciprocating piston-engine design concept

OT = TDC (Top Dead Center); UT = BDC (Bottom Dead Center), V_h Swept volume, V_C Compressed volume, s Piston stroke.



The four-stroke principle

The four-stroke engine employs flow-control valves to govern gas transfer (charge control). These valves open and close the intake and exhaust tracts leading to and from the cylinder:

1st stroke: Induction,
2nd stroke: Compression and ignition,
3rd stroke: Combustion and work,
4th stroke: Exhaust.

Induction stroke
Intake valve: open,
Exhaust valve: closed,
Piston travel: downward,
Combustion: none.

The piston's downward motion increases the cylinder's effective volume to draw fresh air/fuel mixture through the passage exposed by the open intake valve.

Compression stroke
Intake valve: closed,
Exhaust valve: closed,
Piston travel: upward,
Combustion: initial ignition phase.

---

[1]) After Nikolaus August Otto (1832–1891), who unveiled the first four-stroke gas-compression engine at the Paris World Exhibition in 1876.

As the piston travels upward it reduces the cylinder's effective volume to compress the air/fuel mixture. Just before the piston reaches top dead center (TDC) the spark plug ignites the concentrated air/fuel mixture to initiate combustion.
Stroke volume V_h
and compression volume V_C
provide the basis for calculating the compression ratio
$\varepsilon = (V_h+V_C)/V_C$.
Compression ratios ε range from 7...13, depending upon specific engine design. Raising an IC engine's compression ratio increases its thermal efficiency, allowing more efficient use of the fuel. As an example, increasing the compression ratio from 6:1 to 8:1 enhances thermal efficiency by a factor of 12 %. The latitude for increasing compression ratio is restricted by knock. This term refers to uncontrolled mixture inflammation characterized by radical pressure peaks. Combustion knock leads to engine damage. Suitable fuels and favorable combustion-chamber configurations can be applied to shift the knock threshold into higher compression ranges.

Power stroke
Intake valve: closed,
Exhaust valve: closed,
Piston travel: upward,
Combustion: combustion/post-combustion phase.

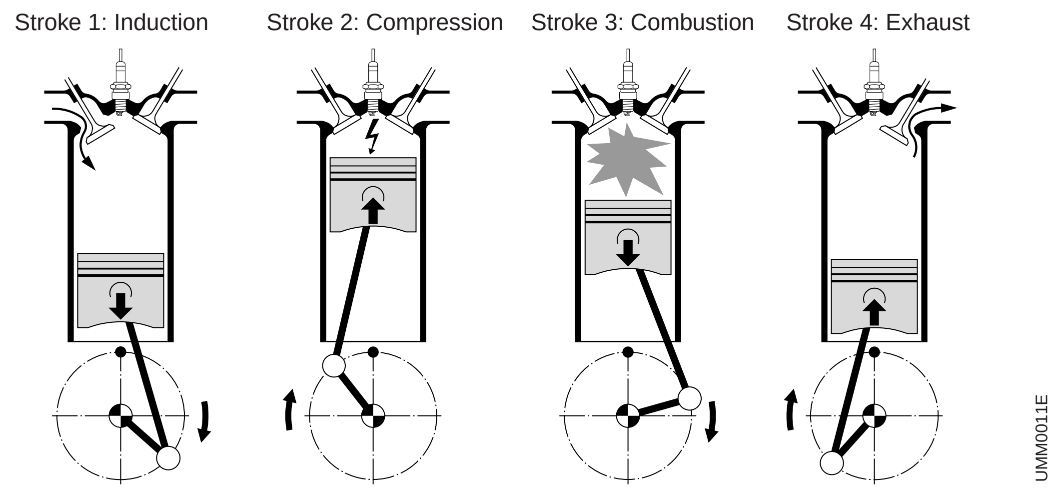
The ignition spark at the spark plug ignites the compressed air/fuel mixture, thus initiating combustion and the attendant temperature rise.
This raises pressure levels within the cylinder to propel the piston downward. The piston, in turn, exerts force against the crankshaft to perform work; this process is the source of the engine's power.
Power rises as a function of engine speed and torque ($P = M \cdot \omega$).
A transmission incorporating various conversion ratios is required to adapt the combustion engine's power and torque curves to the demands of automotive operation under real-world conditions.

Exhaust stroke
Intake valve: closed,
Exhaust valve: open,
Piston travel: upward,
Combustion: none.

As the piston travels upward it forces the spent gases (exhaust) out through the passage exposed by the open exhaust valve. The entire cycle then recommences with a new intake stroke. The intake and exhaust valves are open simultaneously during part of the cycle. This overlap exploits gas-flow and resonance patterns to promote cylinder charging and scavenging.

Fig. 2

Operating cycle of the 4-stroke spark-ignition engine



Gasoline-engine management

Technical requirements

Spark-ignition (SI) engine torque

The power P furnished by the spark-ignition engine is determined by the available net flywheel torque and the engine speed.

The net flywheel torque consists of the force generated in the combustion process minus frictional losses (internal friction within the engine), the gas-exchange losses and the torque required to drive the engine ancillaries (Figure 1).

The combustion force is generated during the power stroke and is defined by the following factors:

- The mass of the air available for combustion once the intake valves have closed,
- The mass of the simultaneously available fuel, and
- The point at which the ignition spark initiates combustion of the air/fuel mixture.

Primary engine-management functions

The engine-management system's first and foremost task is to regulate the engine's torque generation by controlling all of those functions and factors in the various engine-management subsystems that determine how much torque is generated.

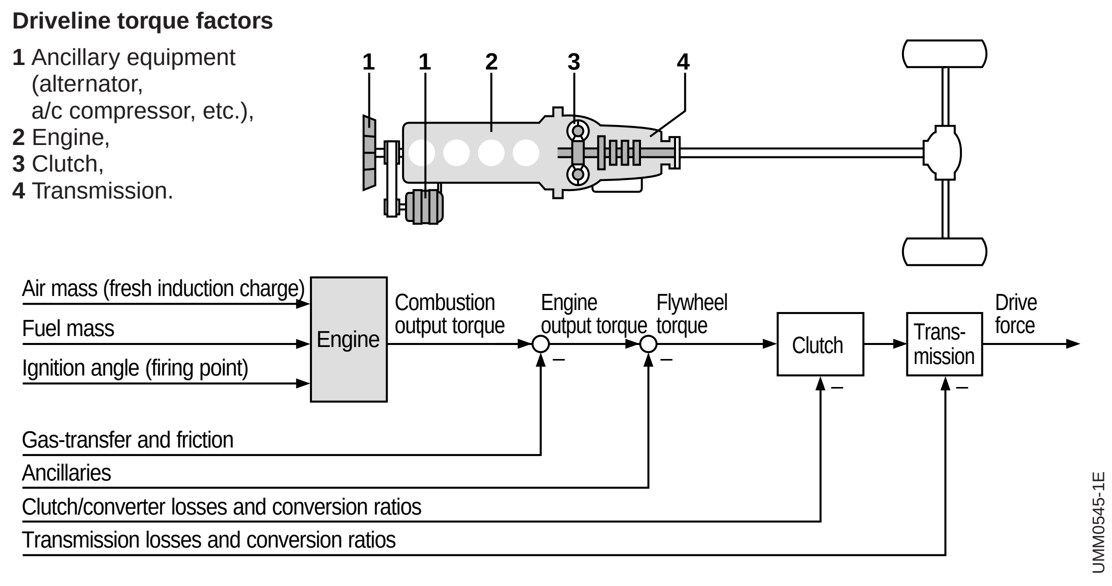
Cylinder-charge control

In Bosch engine-management systems featuring electronic throttle control (ETC), the “cylinder-charge control” subsystem determines the required induction-air mass and adjusts the throttle-valve opening accordingly. The driver exercises direct control over throttle-valve opening on conventional injection systems via the physical link with the accelerator pedal.

Mixture formation

The “mixture formation” subsystem calculates the instantaneous mass fuel requirement as the basis for determining the correct injection duration and optimal injection timing.

Fig. 1



Driveline torque factors

1 Ancillary equipment (alternator, a/c compressor, etc.),
2 Engine,
3 Clutch,
4 Transmission.

Ignition
Finally, the “ignition” subsystem determines the crankshaft angle that corresponds to precisely the ideal instant for the spark to ignite the mixture.

The purpose of this closed-loop control system is to provide the torque demanded by the driver while at the same time satisfying strict criteria in the areas of
- Exhaust emissions,
- Fuel consumption,
- Power,
- Comfort and convenience, and
- Safety.

Cylinder charge

Elements

The gas mixture found in the cylinder once the intake valve closes is referred to as the cylinder charge, and consists of the inducted fresh air-fuel mixture along with residual gases.

Fresh gas
The fresh mixture drawn into the cylinder is a combination of fresh air and the fuel entrained with it. While most of the fresh air enters through the throttle valve, supplementary fresh gas can also be drawn in through the evaporative-emissions control system (Figure 2). The air entering through the throttle-valve and remaining in the cylinder after intake-valve closure is the decisive factor defining the amount of work transferred through the piston during combustion, and thus the prime determinant for the amount of torque generated by the engine. In consequence, modifications to enhance maximum engine power and torque almost always entail increasing the maximum possible cylinder charge. The theoretical maximum charge is defined by the volumetric capacity.

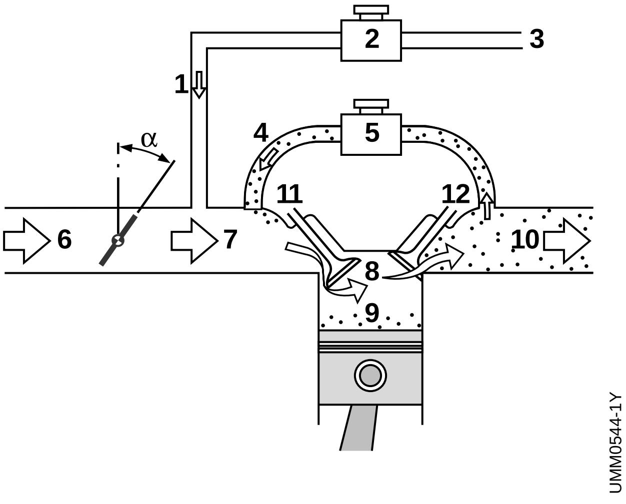
Residual gases
The portion of the charge consisting of residual gases is composed of
- The exhaust-gas mass that is not discharged while the exhaust valve is open and thus remains in the cylinder, and
- The mass of recirculated exhaust gas (on systems with exhaust-gas recirculation, Figure 2).

The proportion of residual gas is determined by the gas-exchange process.
Although the residual gas does not participate directly in combustion, it does influence ignition patterns and the actual combustion sequence. The effects of this residual-gas component may be thoroughly desirable under part-throttle operation.
Larger throttle-valve openings to compensate for reductions in fresh-gas filling

Fig. 2

Cylinder charge in the spark-ignition engine

1 Air and fuel vapor,
2 Purge valve with variable aperture,
3 Link to evaporative-emissions control system,
4 Exhaust gas,
5 EGR valve with variable aperture,
6 Mass airflow (barometric pressure p_U),
7 Mass airflow (intake-manifold pressure p_s),
8 Fresh air charge (combustion-chamber pressure p_B),
9 Residual gas charge (combustion-chamber pressure p_B),
10 Exhaust gas (back-pressure p_A),
11 Intake valve,
12 Exhaust valve,
α Throttle-valve angle.



are needed to meet higher torque demand. These higher angles reduce the engine’s pumping losses, leading to lower fuel consumption. Precisely regulated injection of residual gases can also modify the combustion process to reduce emissions of nitrous oxides (NO_X) and unburned hydrocarbons (HC).

Control elements

Throttle valve

The power produced by the spark-ignition engine is directly proportional to the mass airflow entering it. Control of engine output and the corresponding torque at each engine speed is regulated by governing the amount of air being inducted via the throttle valve. Leaving the throttle valve partially closed restricts the amount of air being drawn into the engine and reduces torque generation. The extent of this throttling effect depends on the throttle valve’s position and the size of the resulting aperture.

The engine produces maximum power when the throttle valve is fully open (WOT, or wide open throttle).

Figure 3 illustrates the conceptual correlation between fresh-air charge density and engine speed as a function of throttle-valve aperture.

Gas exchange

The intake and exhaust valves open and close at specific points to control the transfer of fresh and residual gases. The ramps on the camshaft lobes determine both the points and the rates at which the valves open and close (valve timing) to define the gas-exchange process, and with it the amount of fresh gas available for combustion.

Valve overlap defines the phase in which the intake and exhaust valves are open simultaneously, and is the prime factor in determining the amount of residual gas remaining in the cylinder. This process is known as "internal" exhaust-gas recirculation. The mass of residual gas can also be increased using "external" exhaust-gas recirculation, which relies on a supplementary EGR valve linking the intake and exhaust manifolds. The engine ingests a mixture of fresh air and exhaust gas when this valve is open.

Pressure charging

Because maximum possible torque is proportional to fresh-air charge density, it is possible to raise power output by compressing the air before it enters the cylinder.

Dynamic pressure charging

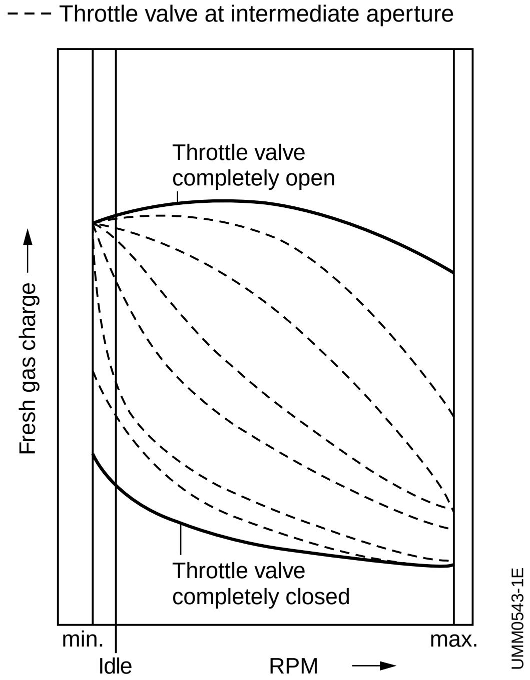
A supercharging (or boost) effect can be obtained by exploiting dynamics within the intake manifold. The actual degree of boost will depend upon the manifold’s configuration as well as the engine’s instantaneous operating point (essentially a function of the engine’s speed, but also affected by load factor). The option of varying intake-manifold geometry while the vehicle is actually being driven, makes it possible to employ dynamic precharging to increase the maximum available charge mass through a wide operational range.

Mechanical supercharging

Further increases in air mass are available through the agency of

Fig. 3

Throttle-valve map for spark-ignition engine



mechanically driven compressors powered by the engine’s crankshaft, with the two elements usually rotating at an invariable relative ratio. Clutches are often used to control compressor activation.

Exhaust-gas turbochargers

Here the energy employed to power the compressor is extracted from the exhaust gas. This process uses the energy that naturally-aspirated engines cannot exploit directly owing to the inherent restrictions imposed by the gas expansion characteristics resulting from the crankshaft concept. One disadvantage is the higher back-pressure in the exhaust gas exiting the engine. This back-pressure stems from the force needed to maintain compressor output.

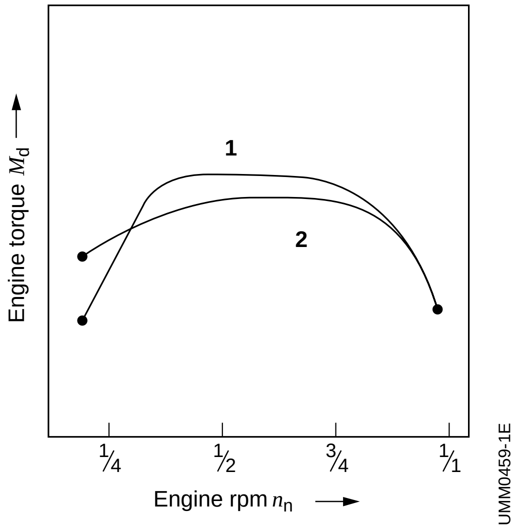
The exhaust turbine converts the exhaust-gas energy into mechanical energy, making it possible to employ an impeller to precompress the incoming fresh air. The turbocharger is thus a combination of the turbine in the exhaust-fas flow and the impeller that compresses the intake air.

Figure 4 illustrates the differences in the torque curves of a naturally-aspirated engine and a turbocharged engine.

Fig. 4

Torque curves for turbocharged and atmospheric-induction engines with equal power outputs

1 Engine with turbocharger,
2 Atmospheric-induction engine.



Mixture formation

Parameters

Air-fuel mixture

Operation of the spark-ignition engine is contingent upon availability of a mixture with a specific air/fuel (A/F) ratio. The theoretical ideal for complete combustion is a mass ratio of 14.7:1, referred to as the stoichiometric ratio. In concrete terms this translates into a mass relationship of 14.7 kg of air to burn 1 kg of fuel, while the corresponding volumetric ratio is roughly 9,500 litres of air for complete combustion of 1 litre of fuel.

The air-fuel mixture is a major factor in determining the spark-ignition engine’s rate of specific fuel consumption. Genuine complete combustion and absolutely minimal fuel consumption would be possible only with excess air, but here limits are imposed by such considerations as mixture flammability and the time available for combustion.

The air-fuel mixture is also vital in determining the efficiency of exhaust-gas treatment system. The current state-of-the-art features a 3-way catalytic converter, a device which relies on a stoichiometric A/F ratio to operate at maximum efficiency and reduce undesirable exhaust-gas components by more than 98 %.

Current engines therefore operate with a stoichiometric A/F ratio as soon as the engine’s operating status permits

Certain engine operating conditions make mixture adjustments to non-stoichiometric ratios essential. With a cold engine for instance, where specific adjustments to the A/F ratio are required. As this implies, the mixture-formation system must be capable of responding to a range of variable requirements.

Excess-air factor

The designation l (lambda) has been selected to identify the excess-air factor (or air ratio) used to quantify the spread between the actual current mass A/F ratio and the theoretical optimum (14.7:1):

λ = Ratio of induction air mass to air requirement for stoichiometric combustion.

λ = 1: The inducted air mass corresponds to the theoretical requirement.

λ < 1: Indicates an air deficiency, producing a corresponding rich mixture. Maximum power is derived from λ = 0.85...0.95.

λ > 1: This range is characterized by excess air and lean mixture, leading to lower fuel consumption and reduced power. The potential maximum value for λ – called the “lean-burn limit (LML)” – is essentially defined by the design of the engine and of its mixture formation/induction system. Beyond the lean-burn limit the mixture ceases to be ignitable and combustion miss sets in, accompanied by substantial degeneration of operating smoothness.

In engines featuring systems to inject fuel directly into the chamber, these operate with substantially higher excess-air factors (extending to λ = 4) since combustion proceeds according to different laws.

Spark-ignition engines with manifold injection produce maximum power at air deficiencies of 5...15 % (λ = 0.95...0.85), but maximum fuel economy comes in at 10...20 % excess air (λ = 1.1...1.2).

Figures 1 and 2 illustrate the effect of the excess-air factor on power, specific fuel consumption and generation of toxic emissions. As can be seen, there is no single excess-air factor which can simultaneously generate the most favorable levels for all three factors. Air factors of λ = 0.9...1.1 produce “conditionally optimal” fuel economy with “conditionally optimal” power generation in actual practice.

Once the engine warms to its normal operating temperature, precise and consistent maintenance of λ = 1 is vital for the 3-way catalytic treatment of exhaust gases. Satisfying this requirement entails exact monitoring of induction-air mass and precise metering of fuel mass.

Optimal combustion from current engines equipped with manifold injection relies on formation of a homogenous mixture as well as precise metering of the injected fuel quantity. This makes effective atomization essential. Failure to satisfy this requirement will foster the formation of large droplets of condensed fuel on the walls of the intake tract and in the combustion chamber. These droplets will fail to combust completely and the ultimate result will be higher HC emissions.

Fig. 1

Effects of excess-air factor λ on power P and specific fuel consumption b_e.

a Rich mixture (air deficiency),
b Lean mixture (excess air).

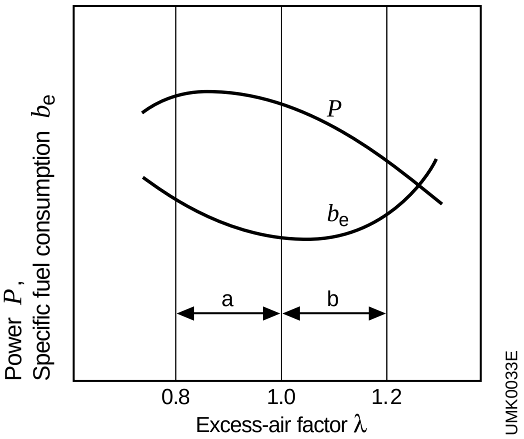
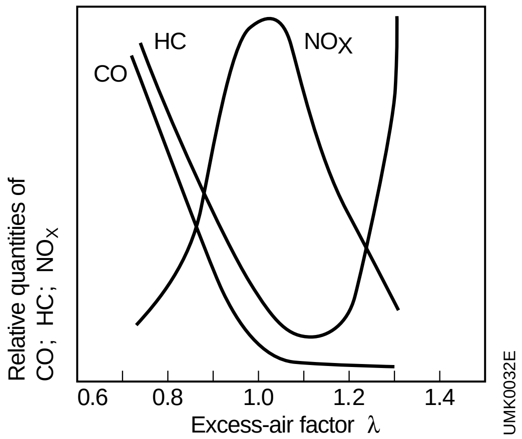


Fig. 2

Effect of excess-air factor λ on untreated exhaust emissions



Adapting to specific operating conditions

Certain operating states cause fuel requirements to deviate substantially from the steady-state requirements of an engine warmed to its normal temperature, thus necessitating corrective adaptations in the mixture-formation apparatus. The following descriptions apply to the conditions found in engines with manifold injection.

Cold starting

During cold starts the relative quantity of fuel in the inducted mixture decreases: the mixture “goes lean.” This lean-mixture phenomenon stems from inadequate blending of air and fuel, low rates of fuel vaporization, and condensation on the walls of the inlet tract, all of which are promoted by low temperatures. To compensate for these negative factors, and to facilitate cold starting, supplementary fuel must be injected into the engine.

Post-start phase

Following low-temperature starts, supplementary fuel is required for a brief period, until the combustion chamber heats up and improves the internal mixture formation. This richer mixture also increases torque to furnish a smoother transition to the desired idle speed.

Warm-up phase

The warm-up phase follows on the heels of the starting and immediate post-start phases. At this point the engine still requires an enriched mixture to offset the fuel condensation on the intake-manifold walls. Lower temperatures are synonymous with less efficient fuel processing (owing to factors such as poor mixing of air and fuel and reduced fuel vaporization). This promotes fuel precipitation within the intake manifold, with the formation of condensate fuel that will only vaporize later, once temperatures have increased. These factors make it necessary to provide progressive mixture enrichment in response to decreasing temperatures.

Idle and part-load

Idle is defined as the operating status in which the torque generated by the engine is just sufficient to compensate for friction losses. The engine does not provide power to the flywheel at idle. Part-load (or part-throttle) operation refers to the range of running conditions between idle and generation of maximum possible torque. Today’s standard concepts rely exclusively on stoichiometric mixtures for the operation of engines running at idle and part-throttle once they have warmed to their normal operating temperatures.

Full load (WOT)

At WOT (wide-open throttle) supplementary enrichment may be required. As Figure 1 indicates, this enrichment furnishes maximum torque and/or power.

Acceleration and deceleration

The fuel’s vaporization potential is strongly affected by pressure levels inside the intake manifold. Sudden variations in manifold pressure of the kind encountered in response to rapid changes in throttle-valve aperture cause fluctuations in the fuel layer on the walls of the intake tract. Spirited acceleration leads to higher manifold pressures. The fuel responds with lower vaporization rates and the fuel layer within the manifold runners expands. A portion of the injected fuel is thus lost in wall condensation, and the engine goes lean for a brief period, until the fuel layer restabilizes. In an analogous, but inverted, response pattern, sudden deceleration leads to rich mixtures. A temperature-sensitive correction function (transition compensation) adapts the mixture to maintain optimal operational response and ensure that the engine receives the consistent air/fuel mixture needed for efficient catalytic-converter performance.

Trailing throttle (overrun)

Fuel metering is interrupted during trailing throttle. Although this expedient saves fuel on downhill stretches, its primary purpose is to guard the catalytic converter against overheating stemming from poor and incomplete combustion (misfiring).

Ignition

Function

The function of the ignition system is to initiate combustion in the compressed air/fuel mixture by igniting it at precisely the right instant. In the spark-ignition engine, this function is assumed by an electric spark in the form of a short-duration discharge arc between the spark plug's electrodes.

Consistently reliable ignition is vital for efficient catalytic-converter operation. Ignition miss allows uncombusted gases to enter the catalytic converter, leading to its damage or destruction from overheating when these gases burn inside it.

Technical requirements

An electrical arc with an energy content of approximately 0.2 mJ is required for each sustainable ignition of a stoichiometric mixture, while up to 3 mJ may be needed for richer or leaner mixtures. This energy is only a fraction of the total (ignition) energy contained in the ignition spark. If the available ignition energy is inadequate, the mixture cannot ignite since ignition fails to take place, and the result is that the engine starts to misfire. This is why the system must supply levels of ignition energy that are high enough to always ensure reliable inflammation of the air/fuel mixture, even under the most severe conditions. A small ignitable mixture cloud passing by the arc is enough to initiate the process. The mixture cloud ignites and propagates combustion through the remaining mixture in the cylinder. Efficient mixture formation and easy access of the mixture cloud to the spark will improve ignition response, as will extended spark durations and larger electrode gaps (longer arcs). The location and length of the spark are determined by the spark plug's design dimensions. Spark duration is governed by the design and configuration of the ignition system along with the instantaneous ignition conditions.

Ignition timing

Ignition timing and its adjustment

Approximately two milliseconds elapse between the instant when the mixture ignites and its complete combustion. Assuming consistent mixture strength, this period will remain invariable. This means that the ignition spark must arc early enough to support generation of optimal combustion pressure under all operating conditions.

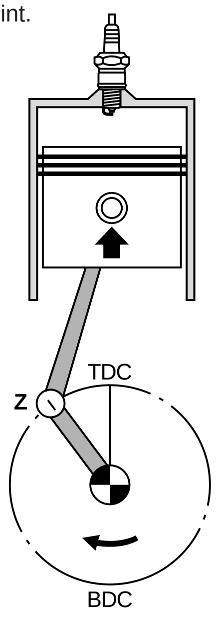
Standard practice defines ignition timing relative to top dead center, or TDC on the crankshaft. Advance angles are then quantified in degrees before TDC, with the corresponding figure being known as the ignition (timing advance) angle. Moving the ignition point back toward TDC is referred to as "retarding" the timing and displacing it forward toward an earlier ignition (firing) point is "advancing" it (Figure 1).

Ignition timing must be selected so that the following criteria are complied with:
- Maximum engine power,
- Maximum fuel economy,
- Prevention of engine knock, and
- "Clean" exhaust gas.

Fig. 1

Position of crankshaft and piston at the ignition (firing) point with advanced ignition

TDC Top Dead Center, BDC Bottom Dead Center, Z Ignition point.



It is impossible to fulfill all the above demands simultaneously, and a compromise must be reachd from case to case. The most favorable firing point at a given torque depends upon a variety of different factors. These are in particular, engine speed, engine load, engine design, fuel, and the particular operating conditions (e.g. starting, idle, WOT, overrun).

Engine knock is due to the abrupt combustion of portions of the air-fuel mixture which have not yet been reached by the advancing flame front triggered by the ignition spark. In this case, the firing point is too far advanced. Combustion knock not only leads to increases in combustion-chamber temperature, which in turn can cause pre-ignition, but also to marked increases in pressure. Such abrupt ignition events generate pressure oscillations which are superimposed on the normal pressure characteristic (Fig. 2).

Today, the high compressions employed in spark-ignition engines involve a far greater risk of combustion knock than was the case with the compression ratios which were common in the past. One differentiates between two different forms of “knock”:

- Acceleration knock at low engine speeds and high load (clearly audible as pinging), and
- High-speed knock at high engine speeds and high load.

For the engine, high-speed knock is a particularly critical factor, since the other engine noises generated at such speeds make it inaudible. This is why audible knock is not a faithful index of preignition tendency. At the same time, electronic means are available for precise detection. Consistent knock causes severe engine damage (destruction of cylinder-head gaskets, bearing damage, “holed” piston crowns) as well as spark-plug damage.

Preignition tendency depends upon such factors as engine design (for instance: combustion chamber layout, homogenous air-fuel mixture, efficient induction flow passages) and fuel quality.

Ignition timing and emissions

The effects of the excess-air factor λ and ignition timing on specific fuel consumption and exhaust emissions are demonstrated in Figures 3 and 4. Specific fuel consumption responds to leaner mixtures with an initial dip before rising

Fig. 2

Combustion-chamber pressure curves for different firing points

1 Ignition Z_a with ideal advance,
2 Ignition Z_b advanced too far (knock),
3 Ignition Z_c too late.

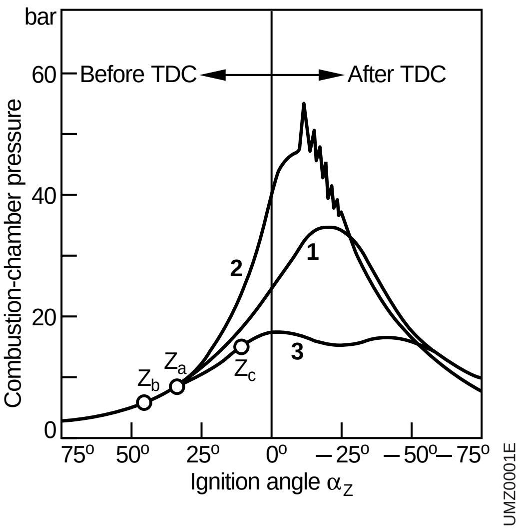
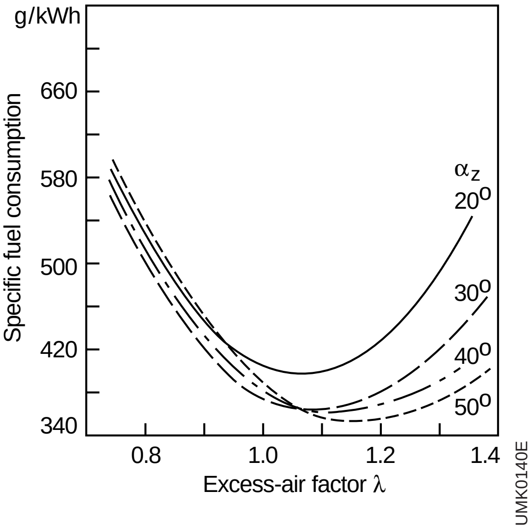


Fig. 3

Effect of excess-air factor λ and ignition timing α_Z on fuel consumption



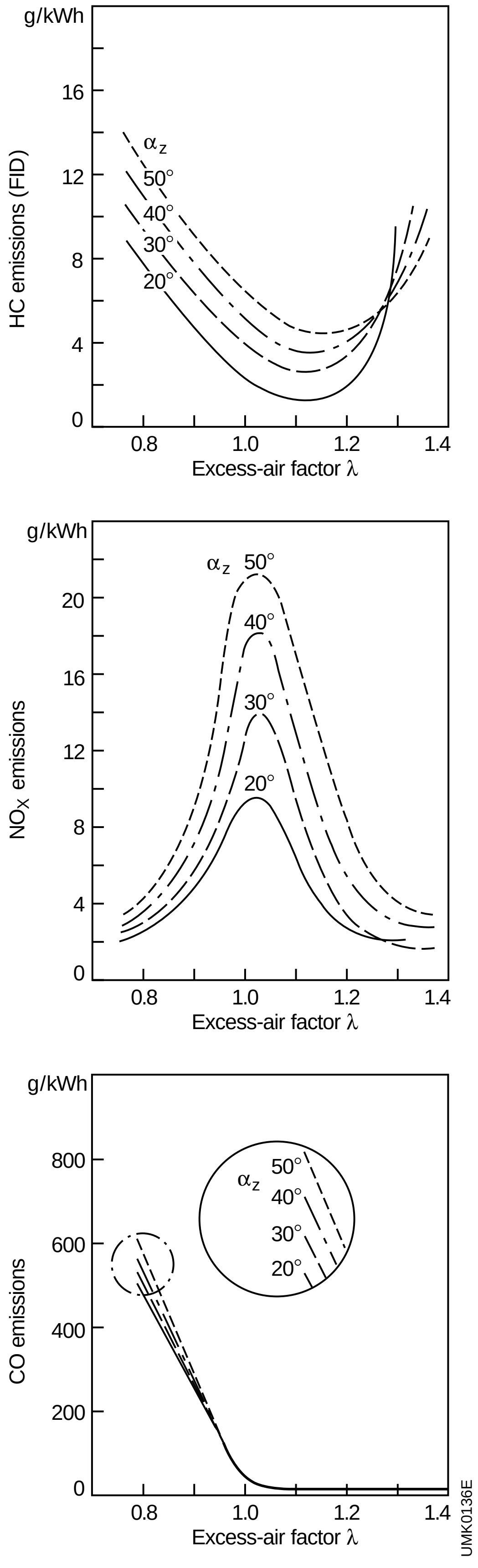
from $\lambda = 1.1...1.2$. Increases in the excess-air factor are accompanied by a corresponding increase in the optimal ignition advance angle, which is defined here as the timing that will minimize specific fuel consumption. The relationship between specific fuel consumption and excess-air factor (assuming optimal ignition timing) can be explained as follows: The air deficiency encountered in the “fuel-rich” range leads to incomplete combustion, while substantial shifts toward the lean misfire limit (LML) will start to cause delayed combustion and misfiring, ultimately leading to higher levels of specific fuel consumption. The optimal ignition advance angle increases at higher excess-air ratios owing to the slower rate of flame-front propagation enountered in lean mixtures; the ignition timing must be advanced to compensate for these delays.

HC emissions, which bottom out at $\lambda = 1.1$, display a similar response pattern. The initial rise within the lean range can be attributed to the flame being extinguished due to the cooling on the walls of the combustion chamber. Extremely lean mixtures produce delayed combustion and failure to ignite, phenomena which occur with increasing frequency as the lean misfire limit is approached. Below $\lambda =$ 1.2, further ignition advance will lead to higher HC emissions, but it will also shift the lean misfire limit to accomodate mixtures with even less fuel. This is why an increase in ignition advance lowers the levels of HC emissions in the lean range beyond $\lambda = 1.25$.

Emissions of nitrous oxides (NO_x) display a completely different pattern by rising in response to higher oxygen (O_2) concentrations and maximum peak combustion temperatures. The result is the characteristic bell-shaped curve for NO_x emissions. These rise up to $\lambda \approx 1.05$ in response to the accompanying increases in O_2 concentrations and peak combustion temperatures. Then, beyond $\lambda = 1.05$, NO_x generation displays a sharp drop as the mixture continues further into the lean range, owing to the

Fig. 4

Excess-air factor λ and ignition timing α_z on exhaust emissions



rapid reduction in peak temperatures that accompanies higher levels of mixture dilution. This response pattern also accounts for the extreme sensitivity with which NO_x emissions respond to changes in ignition timing, escalating sharply as advance is increased.
Because a mixture of $\lambda = 1$ is needed to implement emissions-control concepts relying on the 3-way catalytic converter, adjusting the ignition advance angle is the only remaining option for optimizing emissions.

Inductive ignition systems

The spark-ignition engine's inductive (coil) ignition system generates the high-tension voltage to provide the energy then employed to create an arc at the spark plug. While inductive ignition systems rely on coils to store ignition energy, an available alternative is storage in a condenser (→ so-called high-voltage capacitor-discharge ignition/CDI). The inductive ignition circuit's components are the driver (output amplifier) stage, the coil and the spark plug

Ignition coil

Function

The ignition coil stores the required ignition energy and generates the high voltages required to produce an arc at the firing point.

Design and function

Ignition-coil operation is based on an inductive concept. The coil consists of two magnetically coupled copper coils (primary and secondary windings). The energy stored in the primary winding's magnetic field is transmitted to the secondary side. Current and voltage are transformed in accordance with the turns ratio of the primary and secondary windings (Fig. 1).

Modern ignition coils feature an iron core, composed of individual metal plates inside a synthetic casing. Within this casing the primary winding is wound around a bobbin mounted directly on the core. These elements are concentrically enclosed by the secondary winding, which is designed as a disc or chamber winding for improved insulation resistance. For effective insulation of core and windings, these elements are all enclosed in epoxy resin inside the casing. Specific design configurations are selected to reflect individual operational requirements.

Ignition driver stage

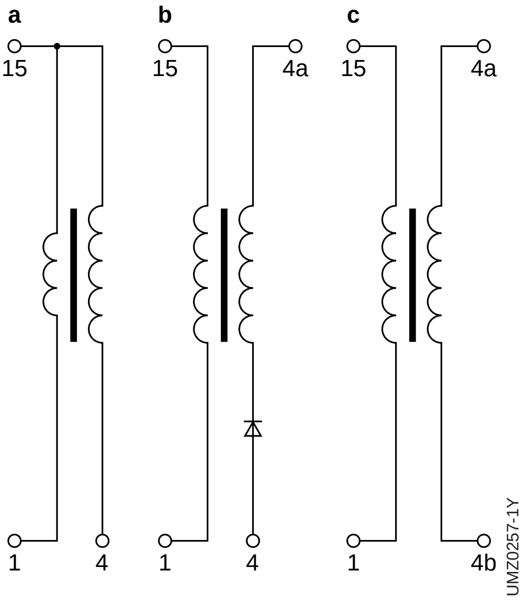
Assignment and function

Ignition driver stages featuring multi-stage power transistors switch the flow of primary current through the coil, replacing the contact-breaker points employed in earlier systems.
In addition, this ignition driver stage is also responsible for limiting primary current and primary voltage. The primary voltage is limited to prevent excessively steep increases of secondary voltage,

Fig. 1

Ignition coils (schematic)

Rotating distribution: **a** Single-spark coil.
Static distribution: **b** Single-spark coil,
c Dual-spark coil.



which could damage components within the high-tension circuit. Restrictions on primary current hold the ignition system's energy output to the specified level.

The ignition system's driver stage may be internal (integrated within the ignition ECU) or external (mounted locally).

High-voltage generation

The ignition ECU switches on the ignition driver stage for the calculated dwell period. It is within this period that the primary current within the coil climbs to its specified intensity.

The energy for the ignition system is stored in the coil's magnetic field and defined by the levels of the coil's primary current and primary inductance.

At the firing point the ignition driver stage interrupts the current flow through the primary winding to induce flux in the magnetic field and generate secondary voltage in the coil's secondary winding.

The ultimate level of secondary voltage (secondary voltage supply) depends upon a number of factors. These include the amount of energy stored in the ignition system, the capacity of the windings and the coil's transformation ratio as well as the secondary load factor and the restrictions on primary voltage imposed by the ignition system's driver stage.

The secondary voltage must always exceed the level required to produce an arc at the spark plug (ignition-voltage requirement), and the spark energy must always be high enough to reliably initiate combustion in the mixture, even in the face of secondary arcing.

When primary current is switched on, this induces an undesired voltage (switch-on voltage) of roughly 1...2 kV in the secondary winding whose polarity opposes that of the high voltage. It is essential that this is prevented from generating an arc (switch-on arc) at the spark-plug.

In systems with conventional rotating voltage distribution, this switch-on spark is effectively suppressed by the distributor's spark gap. On distributorless ignition systems with non-rotating (static) voltage distribution featuring dedicated ignition coils, a diode in the high-voltage circuit performs this function.

With distributorless (static) spark distribution and dual-spark coils, the high arcing voltage associated with two spark plugs connected in series effectively suppresses the switch-on spark without any need for supplementary countermeasures.

Voltage distribution

High-tension voltage must be on hand at the spark plug at the moment of ignition (firing point). This function is the responsibility of the high-voltage distribution system.

Rotating voltage distribution

Systems using a rotating voltage-distribution concept rely on a mechanical ignition distributor to relay the high voltage from a single ignition coil to the individual cylinders. This type of voltage-distribution has ceased to be relevant in the current generation of engine-management systems.

Static voltage distribution

Distributorless ignition (otherwise known as static or electronic ignition) is available in two different versions:

<u>System equipped with single-spark ignition coils</u>

Each cylinder is equipped with its own ignition coil and driver stage, which the engine-management ECU triggers sequentially in the defined firing order. Because internal voltage loss within a distributor is no longer a consideration, the coils can be extremely compact. The preferred installation location is directly above the spark plug. Static distribution with single-spark ignition coils is universally suited for use with any number of cylinders. While there are no inherent restrictions on adjusting ignition advance (timing), these units do require a supplementary synchronization arrangement furnished by a camshaft sensor.

System equipped with dual-spark ignition coils

Each set of two cylinders is supplied by a single ignition driver stage and one coil, with each end of the latter's secondary winding being connected to a different spark plug. The cylinders are paired so that the compression stroke on one will coincide with the exhaust stroke on the other.

When the ignition fires an arc is generated at both spark plugs simultaneously. Because it is important to ensure that the spark produced during the exhaust stroke will ignite neither residual nor fresh incoming gases, this system is characterized by restrictions on adjusting ignition advance (timing). This system does not require a synchronization sensor at the camshaft.

Connectors and interference suppressors

High-voltage cables

The high voltage from the ignition coil must be able to reach the spark plugs. On coils not mounted in direct electrical contact with the spark plugs this function is performed by special high-voltage cables featuring outstanding high-voltage strength and synthetic insulation. Fitted with the appropriate terminals, these cables provide the electrical connections between the high-voltage components.

Because every high-voltage lead represents a capacitive load for the ignition system and reduces the available supply of secondary voltage accordingly, cables should always be as short as possible.

Interference resistors, interference suppression

The pulse-shaped, high-tension discharge that characterizes every arc at the spark plug also represents a source of radio interference. The current peaks associated with discharge are limited by suppression resistors in the high-voltage circuit. To hold radiation of interference emanating from this circuit to a minimum, the suppression resistors should be installed as close as possible to the actual interference source.

Resistors (capacitors) for interference suppression are generally installed in the spark-plug cable terminals, while rotating distributors also include rotor-mounted resistors. Spark plugs with integral suppression resistors are also available. It is important to remember that higher levels of resistance in the secondary circuit are synonymous with corresponding energy loss in the ignition circuit, and result in a reduction in the energy available for firing the spark plug.

Partial or comprehensive encapsulation of the ignition system can be implemented to obtain further reductions in interference radiation.

Spark plug

The spark plug creates the electrical arc that ignites the air-fuel mixture within the combustion chamber.

The spark plug is a ceramic-insulated, high-voltage conductor leading into the combustion chamber. Once arcing voltage is reached, electrical energy flows between the center and ground electrodes to convert the remainder of the ignition-coil energy into a spark.

The level of the voltage required for ignition depends upon a variety of factors including electrode gap, electrode geometry, combustion-chamber pressure, and the instantaneous A/F ratio at the firing point.

Spark-plug electrodes are subject to wear in the course of normal engine operation, and this wear leads to progressively higher voltage requirements. The ignition system must be capable of providing enough secondary voltage to ensure that adequate ignition voltage always remains available, regardless of the operating conditions encountered in the intervals between spark-plug replacements.

Gasoline-injection systems

Carburetors and gasoline-injection systems are designed for a single purpose: To supply the engine with the optimal air-fuel mixture for any given operating conditions. Gasoline injection systems, and electronic systems in particular, are better at maintaining air-fuel mixtures within precisely defined limits, which translates into superior performance in the areas of fuel economy, comfort and convenience, and power. Increasingly stringent mandates governing exhaust emissions have led to a total eclipse of the carburetor in favor of fuel injection.

Although current systems rely almost exclusively on mixture formation outside the combustion chamber, concepts based on internal mixture formation – with fuel being injected directly into the combustion chamber – were actually the foundation for the first gasoline-injection systems. As these systems are superb instruments for achieving further reductions in fuel consumption, they are now becoming an increasingly significant factor.

Overview

Systems with external mixture formation

The salient characteristic of this type of system is the fact that it forms the air-fuel mixture outside the combustion chamber, inside the intake manifold.

Multipoint fuel injection

Multipoint fuel injection forms the ideal basis for complying with the mixture-formation criteria described above. In this type of system each cylinder has its own injector discharging fuel into the area directly in front of the intake valve. Representative examples are the various versions of the KE and L-Jetronic systems (Figure 1).

Mechanical injection systems

The K-Jetronic system operates by injecting continually, without an external drive being necessary. Instead of being determined by the injection valve, fuel mass is regulated by the fuel distributor.

Combined mechanical-electronic fuel injection

Although the K-Jetronic layout served as the mechanical basis for the KE-Jetronic system, the latter employs expanded data-monitoring functions for more precise adaptation of injected fuel quantity to specific engine operating conditions.

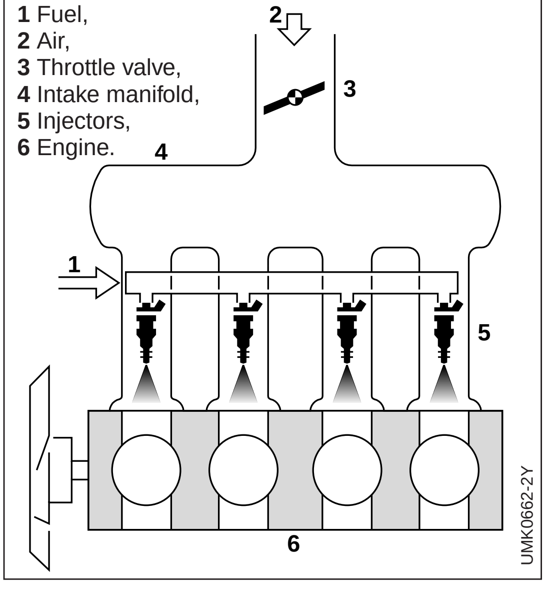
Electronic injection systems

Injection systems featuring electronic control rely on solenoid-operated injection

Fig. 1

Multipoint fuel injection (MPI)

1 Fuel,
2 Air,
3 Throttle valve,
4 Intake manifold,
5 Injectors,
6 Engine.



valves for intermittent fuel discharge. The actual injected fuel quantity is regulated by controlling the injector's opening time (with the pressure-loss gradient through the valve being taken into account in calculations as a known quantity).
Examples: L-Jetronic, LH-Jetronic, and Motronic as an integrated engine-management system.

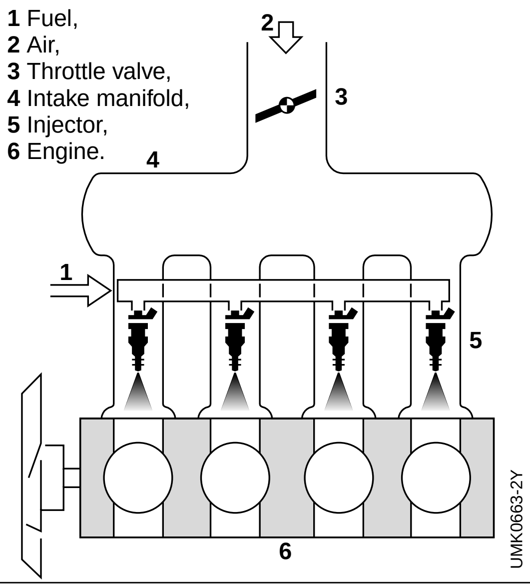
Single-point fuel injection
Single-point (throttle-body injection (TBI)) fuel injection is the concept behind this electronically-controlled injection system in which a centrally located solenoid-operated injection valve mounted upstream from the throttle valve sprays fuel intermittently into the manifold. Mono-Jetronic and Mono-Motronic are the Bosch systems in this category (Figure 2).

Systems for internal mixture formation

Direct-injection (DI) systems rely on solenoid-operated injection valves to spray fuel directly into the combustion chamber; the actual mixture-formation process takes place within the cylinders, each of which has its own injector (Figure 3). Perfect atomization of the fuel emerging from the injectors is vital for efficient combustion.
Under normal operating conditions, DI engines draw in only air instead of the combination of air and fuel common to conventional injection systems. This is one of the new system's prime advantages: It banishes all potential for fuel condensation within the runners of the intake manifold.
External mixture formation usually provides a homogenous, stoichiometric air-fuel mixture throughout the entire combustion chamber. In contrast, shifting the mixture-preparation process into the combustion chamber provides for two distinctive operating modes:
With stratified-charge operation, only the mixture directly adjacent to the spark plug needs to be ignitable. The remainder of the air-fuel charge in the combustion chamber can consist solely of fresh and residual gases, without unburned fuel. This strategy furnishes an extremely lean overall mixture for idling and part-throttle operation, with commensurate reductions in fuel consumption.
Homogenous operation reflects the conditions encountered in external mixture formation by employing uniform consistency for the entire air-fuel charge throughout the combustion chamber. Under these conditions all of the fresh air within the chamber participates in the combustion process. This operational mode is employed for WOT operation.
MED-Motronic is used for closed-loop control of DI gasoline engines.

Fig. 2

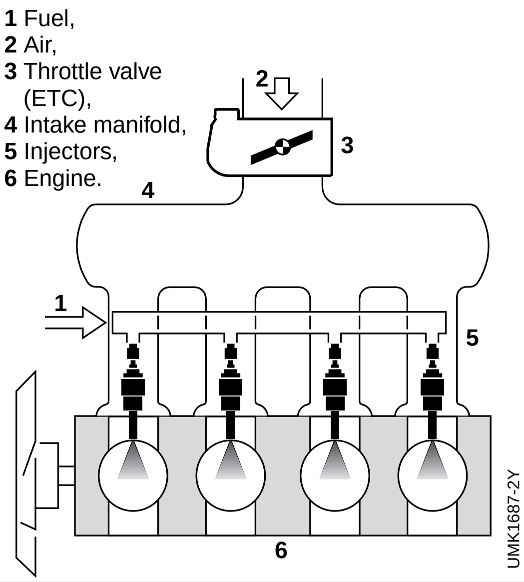
Throttle-body fuel injection (TBI)



1 Fuel,
2 Air,
3 Throttle valve,
4 Intake manifold,
5 Injector,
6 Engine.

Fig. 3

Direct fuel injection (DI)



1 Fuel,
2 Air,
3 Throttle valve (ETC),
4 Intake manifold,
5 Injectors,
6 Engine.

]

M-Motronic engine management

The M-Motronic system

System overview

M-Motronic combines all the electronic systems for engine control in a single control unit (ECU) which, in turn, governs the actuating systems used to control the spark-ignition engine. Engine-mounted monitoring devices (sensors) gather the required operating data and relay the information to input circuits for:

- Ignition (on/off),
- Camshaft position,
- Vehicle speed,
- Gear selection,
- Transmission control,
- Air conditioner, etc.

Monitored analog data include:

- Battery voltage,
- Engine temperature,
- Intake-air temperature,
- Air quantity,
- Throttle-valve angle,
- Lambda oxygen sensor,
- Knock sensor, etc.
  as well as
- Engine speed.

Input circuits located within the ECU convert these data for subsequent operations in the microprocessor. The microprocessor, in turn, uses these data to determine the engine's momentary operating conditions; this information serves as the basis for the ECU's command signals, which are amplified by power-output stages before being transmitted to the final-control elements used to control the engine. This system combines fuel injection, highest-quality mixture preparation and the correct ignition timing to provide mutual support over the entire range of operating conditions encountered in the spark-ignition engine.

M-Motronic versions

The descriptions and illustrations on the following pages refer to a typical M-Motronic configuration (Figure 1). Other M-Motronic systems are available to meet the special demands arising from specific national regulations as well as the requirements of the individual automobile manufacturers.

Basic functions
Control of the ignition and fuel-injection processes is (independent of version) at the core of the M-Motronic system.

Auxiliary functions
Additional open and closed-loop control functions – required in response to legislation aimed at reductions in exhaust emissions and fuel consumption – supplement the basic M-Motronic functions while making it possible to monitor all components exercising an influence on the composition of the exhaust gases.
These include:

- Idle-speed control,
- Lambda oxygen control,
- Control of the evaporative emissions control system,
- Knock control,
- Exhaust-gas recirculation (EGR) for reducing NO_X emissions, and
- Control of the secondary-air injection to reduce HC emissions.

The system can also be expanded to meet special demands from automobile manufacturers by including the following:

- Open-loop turbocharger control as well as control of variable-tract intake manifolds for increased engine power output,

- Camshaft control for achieving reductions in exhaust emissions and fuel consumption as well as enhanced output, and
- Knock control along with engine and vehicle-speed governing functions, to protect engine and vehicle.

The acquisition and processing of the measured information is described in the chapters dealing with acquisition and processing of the operating data.

Vehicle management

M-Motronic supports the control units in other vehicle systems. It can, for instance, operate together with the automatic transmission's control unit to provide reductions in torque during shifting, thereby lessening transmission wear. M-Motronic can also work together with the ABS control unit to provide traction control (TCS) for enhanced vehicle safety.

The schematic system illustration below shows a maximal-configuration M-Motronic system. This type of system can be employed to satisfy

- The stringent emissions limits, and
- The requirement for an integral, onboard diagnosis (OBD) system for California vehicles from 1993 onward.

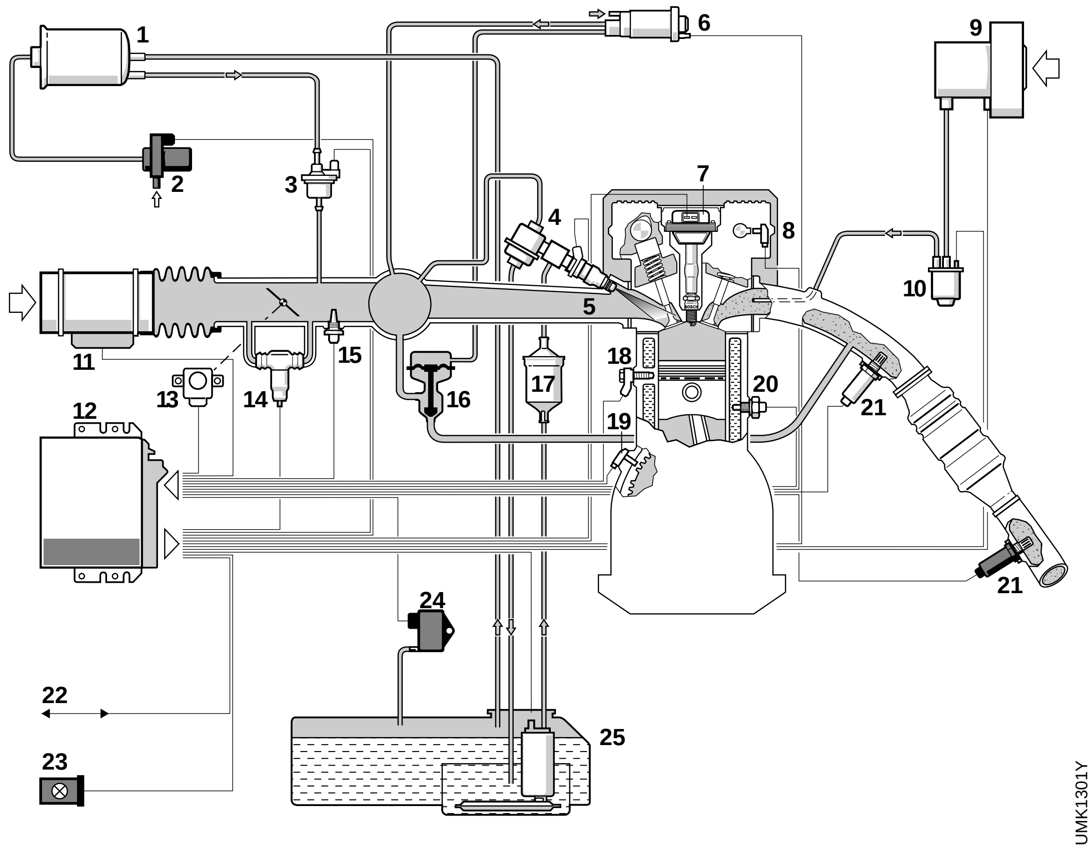


Fig. 1

System diagram of M-Motronic M5 with integrated diagnostics (OBD)

1 Carbon canister,
2 Shutoff valve,
3 Canister-purge valve,
4 Fuel-pressure regulator,
5 Injector,
6 Pressure actuator,
7 Ignition coil,
8 Phase sensor,
9 Secondary-air pump,
10 Secondary-air valve,
11 Air-mass meter,
12 Control unit (ECU),
13 Throttle-valve sensor,
14 Idle actuator,
15 Air-temperature sensor,
16 EGR valve,
17 Fuel filter,
18 Knock sensor,
19 Engine-speed sensor,
20 Engine-temperature sensor,
21 Lambda oxygen sensor,
22 Diagnosis interface,
23 Diagnosis lamp,
24 Pressure differential sensor,
25 Electric fuel pump.

Fuel system

Fuel supply

Fuel-supply system

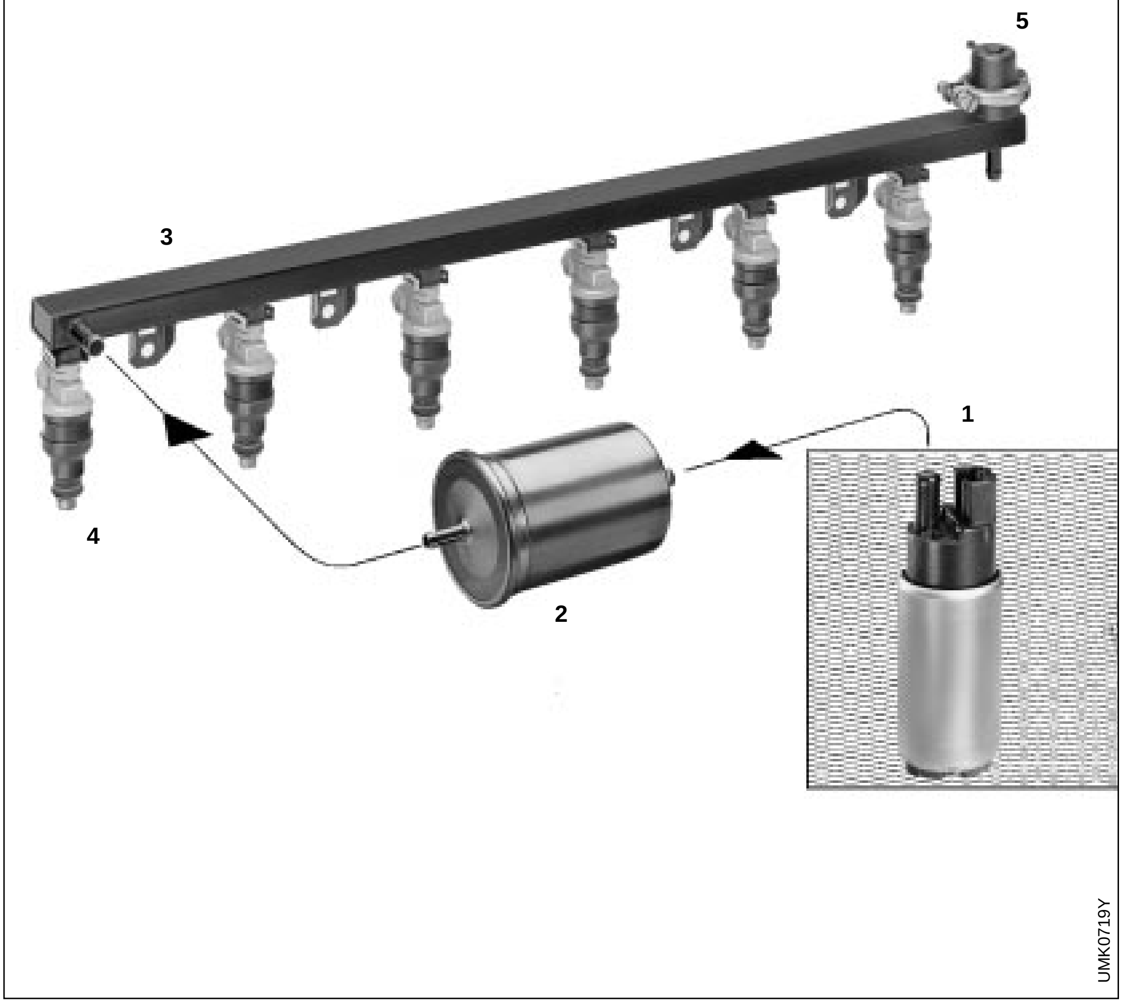
The fuel-supply system must be capable of providing the engine with the required quantity of fuel under all operating conditions. An electric pump draws the fuel through a filter while extracting it from the tank for delivery to the fuel-distribution rail with its electromagnetic injectors. The injectors spray the fuel into the engine's intake tract in precisely metered quantities. The excess fuel flows through the pressure regulator and back to the fuel tank (Figure 1).

The pressure regulator generally employs the pressure within the intake manifold as its reference. This characteristic pressure works in combination with the constant flow through the fuel rail (cooling effect) to prevent vapor bubbles from forming in the fuel. The resulting pressure differential at the injector usually remains constant in the 300 kPa range.
Where necessary, the fuel-supply system can also be designed to incorporate pressure attenuators to reduce pulsation in the fuel line.

Fig. 1

Fuel supply system

1 Electric fuel pump (in-tank),
2 Fuel filter,
3 Fuel rail,
4 Injector,
5 Fuel-pressure regulator.



Electric fuel pump

Function

The electric fuel pump supplies a continuous flow of fuel from the tank. It can be installed either within the fuel tank itself ("in-tank") or at an external location in the fuel line ("in-line").

The in-tank pumps currently in general use (Figures 2 and 3) are integrated within the fuel tank's installation assembly along with the level sensor and a swirl plate to remove vapor bubbles from the fuel return line. When an in-line pump is used, hot delivery problems can be solved by using a supplementary in-tank booster pump to supply fuel from the tank at low pressure. To ensure that the delivery pressure in the system is maintained at the required level, the maximum supply capacity is always greater than the system's theoretical maximum requirement.

The fuel pump is activated by the engine-management ECU. A safety circuit interrupts fuel delivery when the engine is stationary with the ignition on.

Fig. 2

Two-stage electric fuel pump (side-channel and internal-gear pump)

1 First stage (side-channel pump), **2** Main stage (internal-gear pump), **3** Motor armature, **4** Commutator, **5** Non-return valve, **6** Electrical connection.

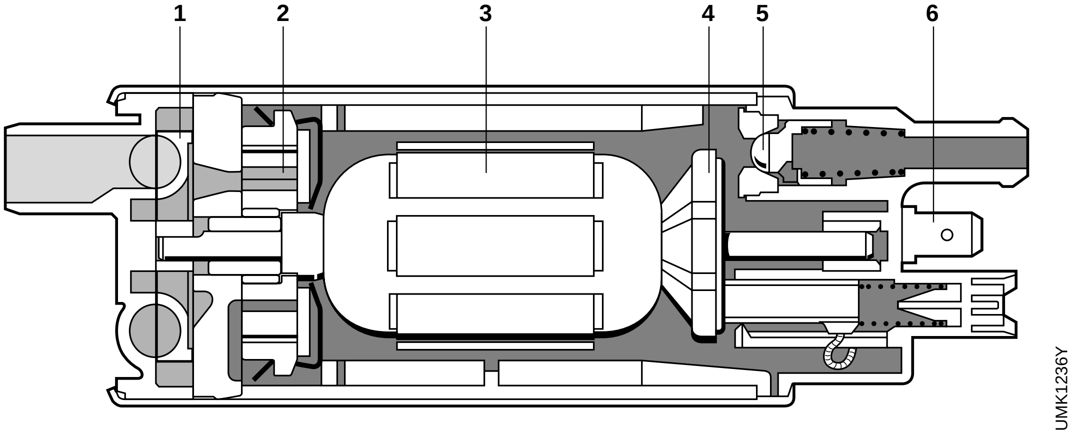
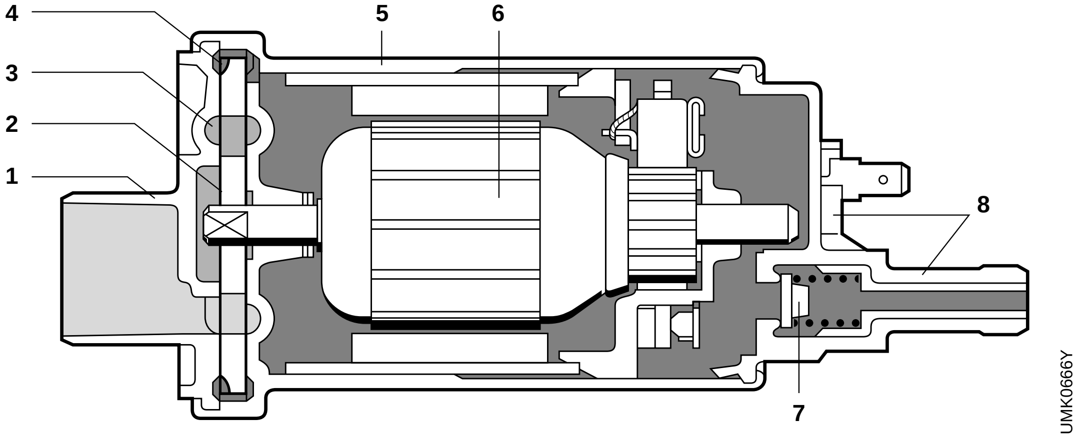


Fig. 3

Two-stage electric fuel pump (side-channel and peripheral pump)

1 Suction cover with supply connection, **2** Impeller, **3** First stage (side-channel pump), **4** Main stage (peripheral pump), **5** Pump housing, **6** Motor armature, **7** Non-return valve, **8** End cover with pressure connection.



Design

The electric fuel pump consists of the following elements:

– Pump assembly,

– Electric motor and end cover.

The electric motor and pump assembly are located in a common housing, where they are immersed in circulating fuel. This arrangement provides effective cooling for the electric motor. Because no oxygen is present, it is impossible for an ignitable mixture to form; there is no danger of an explosion. The end cover contains the electrical connections, the non-return valve and the hydraulic connection for the pressure side. The non-return valve maintains system pressure for a period of time after the unit is shut down to prevent vapor bubbles from forming. Interference-suppression devices can also be included in the end-cover assembly.

Design variations

Various design principles are employed to satisfy individual system demands (Figure 4).

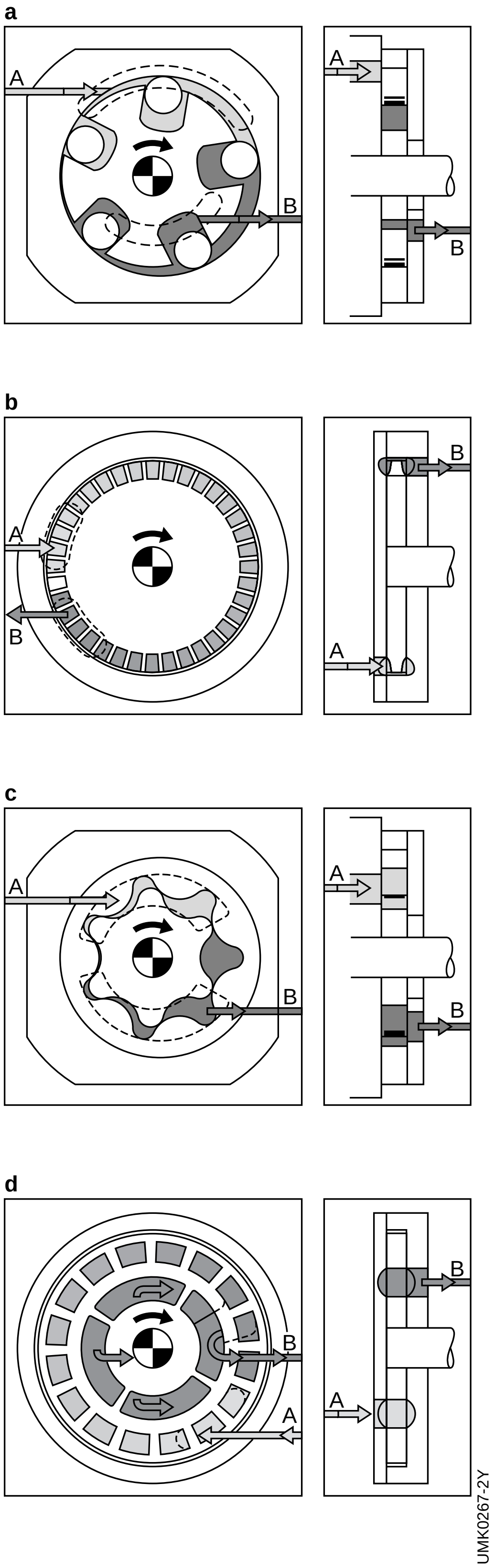
Positive-displacement pumps

Roller-cell (RZP) and internal-gear pumps (IZP) are both classified as positive-displacement designs. Both types of pump operate by using variable-sized, circulating chambers to expose a supply orifice and draw in fuel as their volume expands. Once the maximum volume is reached, the supply orifice closes and the discharge orifice opens. The fuel is now forced out as the effective volume in the chamber decreases. The chambers on the roller-cell pump are formed by rollers circulating in a rotor plate. A combination of centrifugal force and fuel pressure forces them outward against the eccentric roller path. The eccentricity between rotor plate and roller path provides the constant increase and decrease in chamber volume.

Fig. 4

Principles of operation

a Roller-cell pump,
b Peripheral pump,
c Internal-gear pump,
d Side-channel pump.



The internal-gear pump consists of an internal drive gear that moves against the surface of an eccentrically-mounted ring gear; the ring gear is equipped with one tooth more than the drive gear. As the mutually-sealed tooth flanks turn, variable chambers are formed between them. Roller-cell pumps can be used to obtain fuel pressures in excess of 650 kPa. Internal-gear pumps can supply up to 400 kPa, a figure adequate for virtually all Motronic applications.

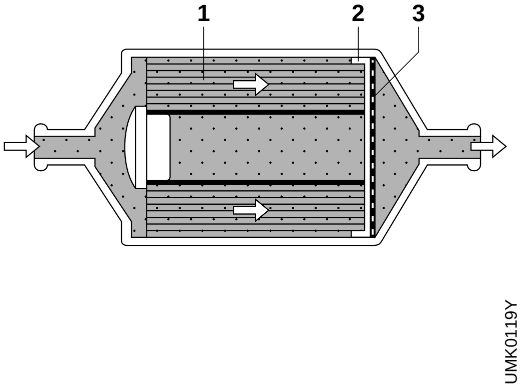
Hydrokinetic flow pumps

The peripheral and side-channel pumps are both classified as hydrokinetic flow pumps. In these pumps an impeller accelerates the fuel particles before discharging them into the tract where they generate pressure via pulse exchange. The peripheral pump differs from its side-channel counterpart in its larger number of impeller blades and the shape of the impellers, as well as in the positions of the side channels, which – unlike those of the side-channel unit – are located on the circumference or periphery. Although peripheral pumps are only capable of generating maximum fuel pressures in the 400 kPa range, they do supply a continuous, virtually pulseless flow of fuel. This makes them particularly attractive for use in those vehicular applications where limiting noise is a major priority. Side-channel pumps can only produce pressures of up to 30 kPa. One important use for this type of unit is as a booster pump in systems with in-line main pumps; another major application is as the primary stage in a two-stage in-tank pump of the kind installed in vehicles susceptible to hot starting problems and/or with single-point fuel injection.

Fig. 5

Fuel filter

1 Paper element,
2 Strainer,
3 Support plate.



Fuel filter

Contaminants in the fuel can impair the operation of both pressure regulator and injectors. A filter is therefore installed downstream from the electric fuel pump. This fuel filter contains a paper element featuring a mean pore diameter of 10 µm. A backplate retains it in its housing. The replacement intervals are determined by the filter’s volume and contamination levels in the fuel (Figure 5).

Fuel rail

The fuel flows through the fuel rail where it is evenly distributed to all injectors. The injectors are mounted on the fuel rail, which also usually includes a fuel-pressure regulator. A pressure attenuator may also be present. The dimensions of the fuel rail are selected to inhibit the local fluctuations in fuel pressure that could otherwise be triggered as the injectors run through their operating cycles. This prevents the injection quantities from reacting to changes in load and engine speed. Depending upon the particular vehicle type and its special requirements, the fuel rail can be made of steel, aluminum or plastic. It may also include an integral test valve, which can be used to bleed pressure for servicing as well as for test purposes.

Fuel-pressure regulator

Injection quantity should be determined exclusively by injection duration. Thus the difference between the fuel pressure in the distribution rail and the pressure in the intake tract must remain constant. A means is thus required for adjusting the fuel pressure to reflect variations in the load-sensitive manifold pressure. The fuel-presure regulator regulates the amount of fuel returning to the tank to maintain a constant pressure drop across the injectors. The pressure regulator is generally positioned at the far end of the fuel rail to avoid impairing the flow within the rail. However, it can also be mounted in the fuel-return line.

The fuel-pressure regulator is designed as a diaphragm-controlled overflow pressure regulator (Figure 6). A rubber-fiber diaphragm divides the pressure regulator into two sections: fuel chamber and pressure chamber. The spring presses against a valve holder integrated within the diaphragm. This force causes a flexibly mounted valve plate to push against a valve seat. When the pressure exerted against the diaphragm by the fuel exceeds that of the spring, the valve opens and allows fuel to flow directly back to the tank until the diaphragm assembly returns to a state of equilibrium, with equal pressure exerted on both of its sides. A pneumatic line is provided between the spring chamber and the intake manifold downstream from the throttle valve, allowing the chamber to respond to changes in manifold vacuum. Thus the pressures at the diaphragm correspond to those at the injectors. As a result, the pressure drop at the injectors remains constant, as it is determined solely by the spring force and surface area of the diaphragm.

Fuel-pressure attenuator

The injectors' operating cycles and the periodic discharge of fuel that characterize the positive-displacement fuel pump both induce fluctuations in fuel pressure. Under unfavorable circumstances, the mountings for the electric fuel pump, fuel lines and fuel rail can transmit these vibrations to the vehicle's body. Noise from this source can be prevented using specially designed mounting elements and fuel-pressure attenuators. The layout of the fuel-pressure attenuator (Figure 7) is similar to that of the pressure regulator. In both cases a spring-loaded diaphragm separates the fuel from the air space. The spring force is calculated to lift the diaphragm from its seat as soon as the fuel reaches operating pressure. This provides a variable fuel chamber which can accept fuel to ease pressure peaks and then release it again when pressure falls. The spring chamber can be fitted with a manifold-vacuum line to stay within the optimum operating range in the face of fluctuations in the fuel's absolute pressure. The pressure attenuator also shares the pressure regulator's installation flexibility, as it can also be mounted in the rail or in the fuel-return line.

Fig. 6

Fuel-pressure regulator

1 Intake-manifold connection, **2** Spring, **3** Valve holder, **4** Diaphragm, **5** Valve, **6** Fuel supply, **7** Fuel return.

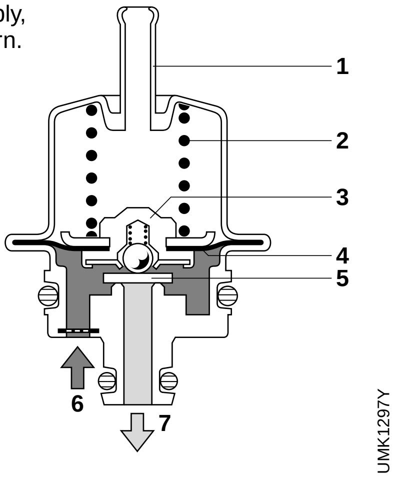
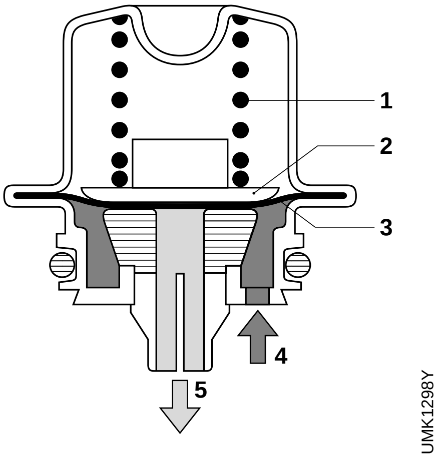


Fig. 7

Fuel-pressure attenuator

1 Spring, **2** Spring plate, **3** Diaphragm, **4** Fuel supply, **5** Fuel return.



Fuel injection

Uncompromising demands for smooth running and low emissions in automobiles have made it necessary to provide thorough and precise mixture formation for every single work cycle. The fuel mass must be injected in quantities that are precisely metered to match the amount of intake air; in today's applications exact injection timing is acquiring increasing significance. For this reason, every cylinder is assigned an electromagnetic injector. The injector sprays the fuel – in precise quantities at a point in time determined by the ECU – directly toward the cylinder intake valve(s). Thus condensation along the walls of the intake tract of the kind that leads to deviations from the desired Lambda value is largely avoided. Because the engine's intake manifold conducts only combustion air, its geometry can be optimized to meet the engine's dynamic gas-flow requirements.

Electromagnetic injector

The electromagnetic injector contains a solenoid armature mounted on a valve needle (Figures 8 and 9), and travels through precise motions within the valve body. When the unit is at rest, a coil spring presses the valve needle against the seat to seal off the flow of fuel through

Fig. 8

Injector (top-feed)

1 Filter strainer in fuel supply, **2** Electrical connection, **3** Solenoid winding, **4** Valve housing, **5** Armature, **6** Valve body, **7** Valve needle.

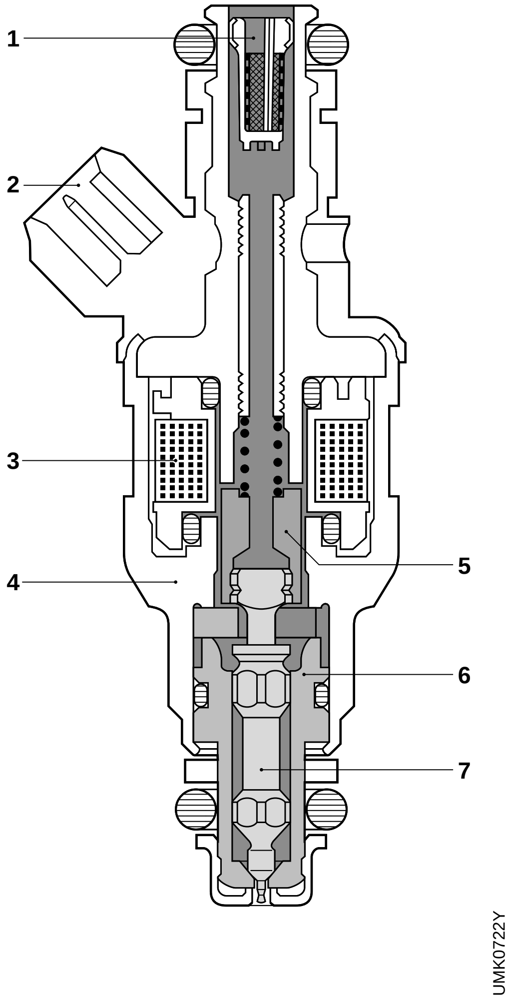
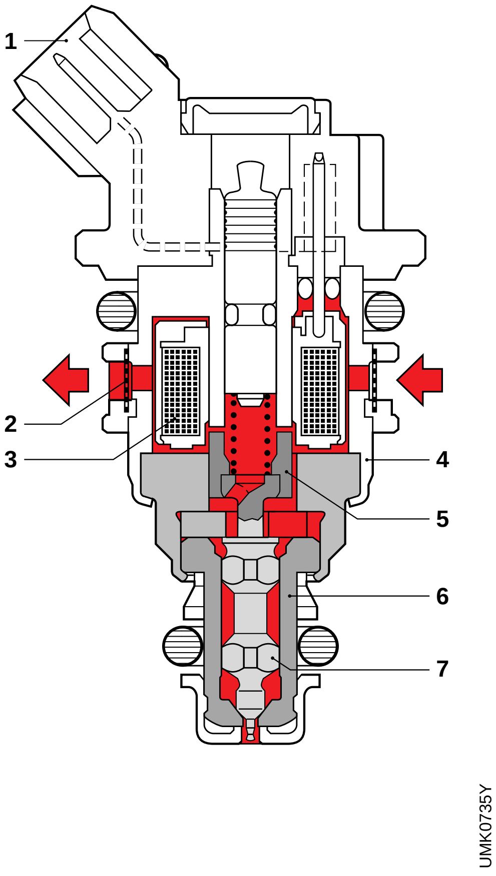


Fig. 9

Injector (bottom-feed)

1 Electrical connection, **2** Filter screen in fuel supply, **3** Solenoid winding, **4** Valve housing, **5** Armature, **6** Valve body, **7** Valve needle.



the outlet orifice and into the intake manifold. When the control transmits an activation current to the solenoid winding in the valve housing, the solenoid armature rises between 60 and 100 µm, lifting the valve needle in the process; the fuel can now flow through the calibrated orifice. The response times lie between 1.5 ... 18 ms at a control frequency of 3...125 Hz, depending upon the type of injection and the momentary engine speed and load conditions.

Different injector designs are employed to meet varying requirements:

Top-feed injector

Fuel enters the top-feed injector from above and flows through its vertical axis. This unit is installed in a specially-formed opening in the fuel rail. Sealing is provided by an upper seal ring, while a clip holds the unit in place. The lower end, which also has a seal ring, extends into the engine's intake manifold (Figure 8).

Bottom-feed injector

The "bottom-feed" injector is integrated within the fuel-rail assembly, where it is constantly immersed in flowing fuel. The fuel supply enters the unit from the side ("bottom-feed"). The fuel rail itself is mounted directly on the intake manifold.

Fig. 10

Injectors (bottom-feed) integrated in fuel rail

1 Fuel supply,
2 Injector,
3 Electrical connection,
4 Contact rail,
5 Pressure regulator,
6 Fuel return.

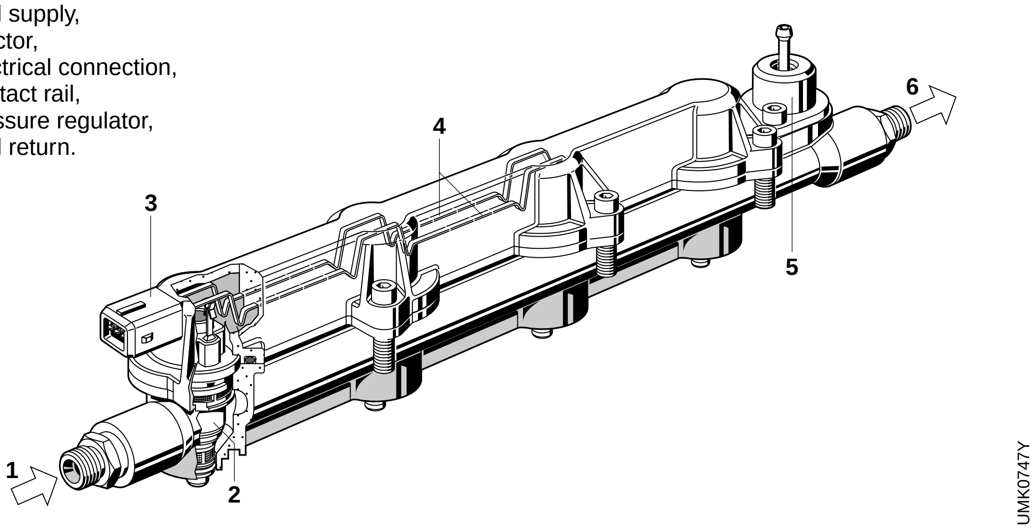
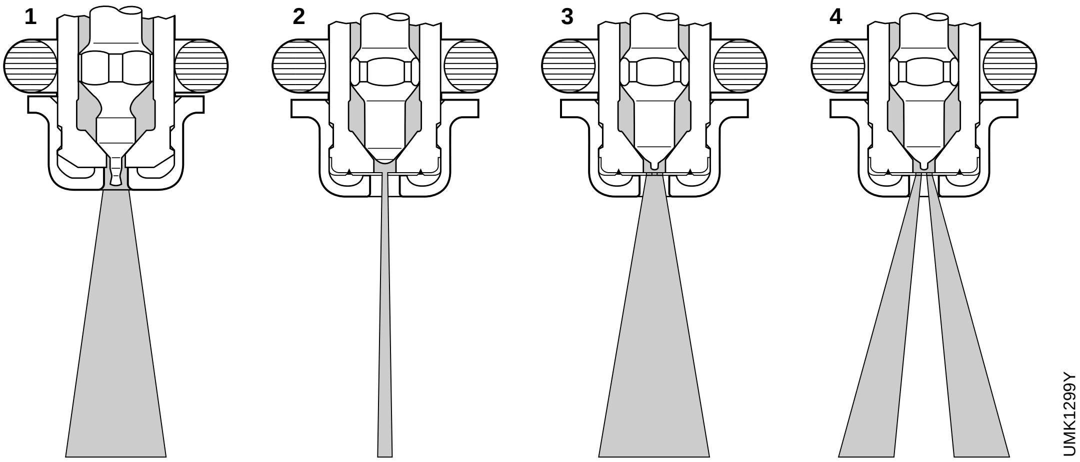


Fig. 11

Metering layouts and fuel preparation

1 Ring-gap metering, **2** Single-orifice metering, **3** Multi-orifice metering,
4 Multi-orifice metering with dual-stream injector.



The injector is retained in the fuel rail by either a clip, or a cover on the rail which can also house the electrical connections. Two seal rings prevent the fuel from escaping. This type of modular design provides several advantages; these include good starting and driving response with hot engines as well as low installation height (Figures 9 and 10).

Mixture formation

A variety of different fuel-metering arrangements are employed to satisfy the demand for the effective fuel atomization necessary for ensuring maximum homogeneity in the air-fuel mixture while simultaneously holding intake-tract condensation to a minimum. The injector's discharge orifice is specially calibrated to fulfill these requirements in the respective applications (Figure 11).

On units with ring-gap metering, a section of the valve needle (pintle) extends through the valve body. The resulting ring gap forms the calibrated fuel-discharge orifice. The lower end of the pintle features a machined breakaway edge where the fuel atomizes before emerging in a tapered pattern.

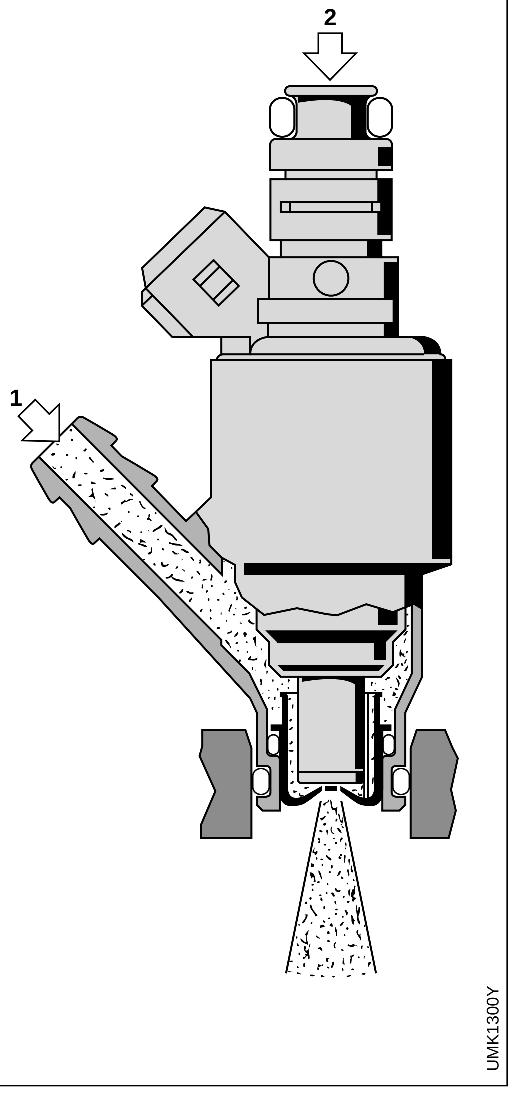
On injectors with single-orifice metering, the pintle is replaced by a thin injection-orifice disk with a calibrated opening. Virtually none of the thin jet of fuel lands on the walls of the intake tract. However, fuel atomization is limited. Injectors featuring multi-orifice metering are fitted with an injection-orifice disk of the kind used in the single-orifice units, the difference being that the multi-orifice disk contains numerous calibrated openings. These are arranged to provide a tapered spray pattern similar to that achieved with annular-orifice metering devices, and provide comparable fuel atomization. The orifices can also be designed to provide two or more spray patterns. This makes it possible to achieve optimum fuel distribution via separate injection into the individual inlet runners on multi-valve engines. Meanwhile, air-shrouded injectors can provide even better mixture formation.

Combustion air traveling at the speed of sound is extracted from the intake manifold at a location upstream from the throttle valve; it then proceeds through a calibrated opening located directly on the injection-orifice disk. The interaction between fuel and air molecules provides thorough atomization. To allow air to be drawn in through the opening, a partial vacuum referred to atmospheric pressure is required in the intake manifold. The air-shrouded design is thus most effective during part-throttle operation (Figure 12).

Fig. 12

Air-shrouded injector

1 Air supply, **2** Fuel supply.



Operating-data acquisition

Engine load

One of the most important variables used for determining injection quantity and ignition advance angle is the engine's load state (load monitoring).

The various Motronic systems employ the following load sensors to monitor engine load:

- Air-flow sensor,
- Hot-wire air-mass meter,
- Hot-film air-mass meter,
- Intake manifold pressure sensor, and
- Throttle-valve sensor.

In the Motronic systems the throttle-valve sensor generally assumes the function of a secondary load sensor, supplementing one of the primary load sensors listed above. It is also employed as a primary load sensor in some isolated cases.

Air-flow sensor

The air-flow sensor is located between the air filter and the throttle valve, where it monitors the volumetric flow rate [m^3/h] of the air being drawn into the engine. The force of the air stream acts against the constant return force of a spring, and the air flap's deflection angle is monitored via potentiometer. The potentiometer voltage is transmitted to the ECU for comparison with the potentiometer's initial supply voltage. The resulting voltage ratio serves as an index of the induction air's volumetric flow rate. The ECU ensures accuracy by compensating for the effects of potentiometer aging and temperature when processing the resistance (Figure 1).

In order to prevent pulsation in the intake air stream from setting up oscillations in the air-flow sensor flap, the system also includes a counterflap and a “damping volume.” The air-flow sensor is equipped with a temperature sensor. This transmits a temperature-sensitive resistance value to the control unit, allowing it to compensate for variations in air density arising from changes in the temperature of the intake air.

The air-flow sensor is still a component in many of the M-Motronic and L-Jetronic systems currently in production. The load sensors described in the following section are preferably installed, and will replace the flap-controlled air-flow sensor in future systems.

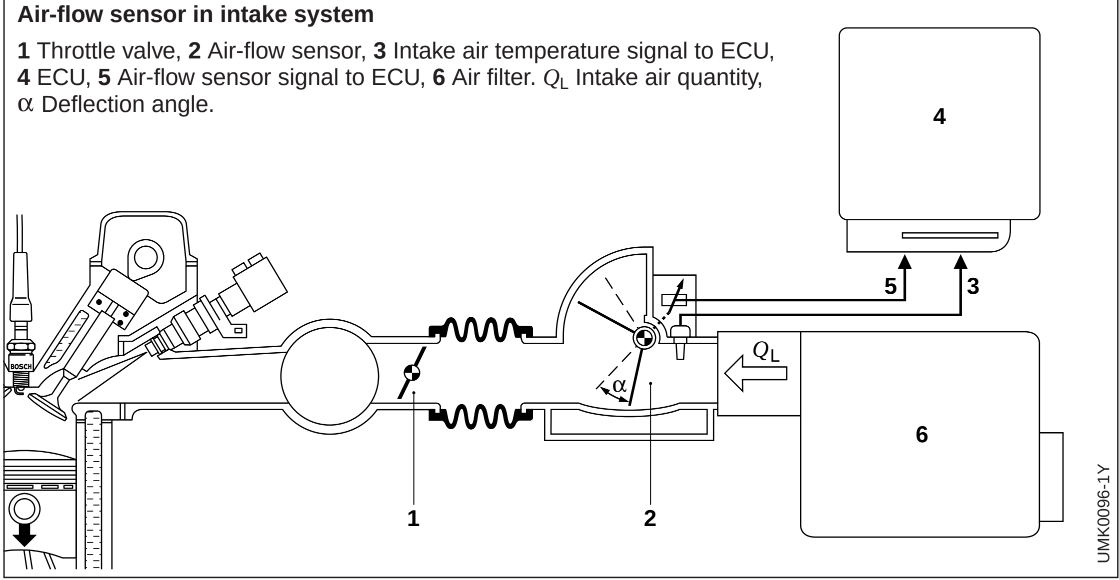
Air-mass meters

The hot-wire and hot-film air-mass meters are both “thermal” load sensors. They are installed between the air filter and the throttle valve, where they monitor the mass flow [kg/h] of the air being drawn into the engine. The two meters operate according to a common principle.

Fig. 1

Air-flow sensor in intake system

1 Throttle valve, **2** Air-flow sensor, **3** Intake air temperature signal to ECU, **4** ECU, **5** Air-flow sensor signal to ECU, **6** Air filter. Q_L Intake air quantity, α Deflection angle.



An electrically heated element is mounted in the intake-air stream, where it is cooled by the flow of incoming air. A control circuit modulates the flow of heating current to maintain the temperature differential between the heated wire (or film) and the intake air at a constant level. The amount of heating current required to maintain the temperature thus provides an index for the mass air flow. This concept automatically compensates for variations in air density, as this is one of the factors that determines the amount of warmth that the surrounding air absorbs from the heated element.

Hot-wire air-mass meter

The heated element on the hot-wire air-mass meter is a platinum wire only 70 µm in diameter. A temperature sensor is integrated within the hot-wire air-mass meter to provide compensation data for intake-air temperature. The main components in the control circuit are a bridge circuit and an amplifier. The heated wire and the intake-air temperature sensor both act as temperature-sensitive resistors within the bridge (Figures 2 through 4). The heating current generates a voltage signal, proportional to the mass air flow, at a precision resistor. This is the signal transmitted to the ECU.

Fig. 2

Components of the heated-wire air-mass meter

1 Temperature sensor, **2** Sensor ring with hot wire, **3** Precision resistor.
Q_M Mass flow.

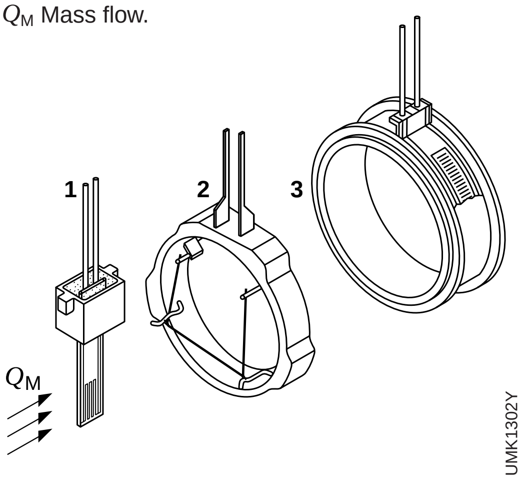


Fig. 3

Bridge circuit in heated-wire air-mass meter

R_H Hot wire, R_K Compensation resistor, R_M Measurement resistor, R_1, R_2 Adjustable resistors. U_M Measurement voltage, Q_M Incoming air mass per unit of time.

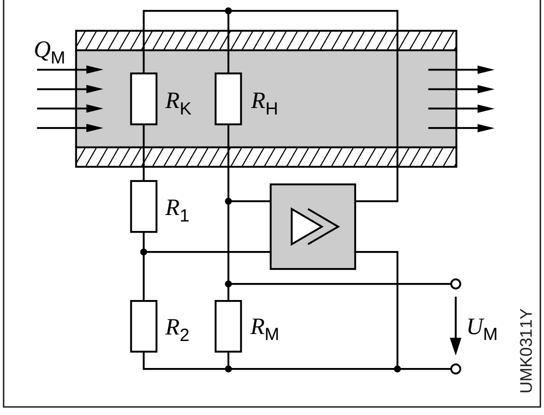
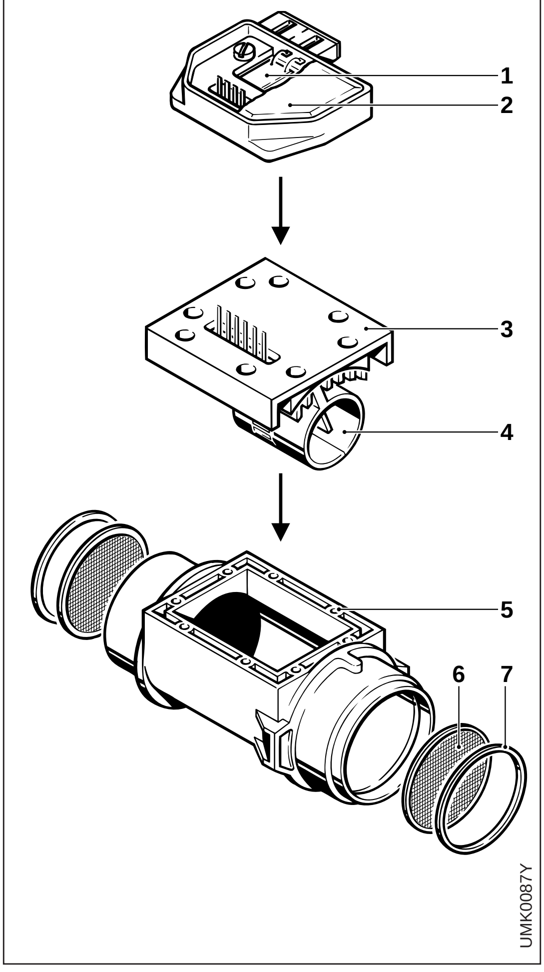


Fig. 4

Hot-wire air-mass meter

1 Hybrid circuit,
2 Cover,
3 Metal insert,
4 Inner tube with hot wire,
5 Housing,
6 Screen,
7 Retainer.



To prevent the “drift” that could result from contaminant deposits on the platinum wire, the wire is heated up to “burn-off” temperature for one second after the engine is switched off. This process vaporizes and/or splits off the deposits and cleans the wire.

Hot-film air-mass meter

The heated element on the hot-film air-mass meter is a platinum film resistor (heater). It is located on a ceramic plate together with the other elements in the bridge circuit. The temperature of the heater is monitored by a temperature-sensitive resistor (flow sensor) also included in the bridge.

The separation of heater and flow sensor facilitates design of the control circuitry. Saw cuts are employed to ensure thermal decoupling between the heating element and the intake-air temperature sensor.

The complete control circuitry is located on a single layer. The voltage at the heater provides the index for the mass air flow. The hot-film air-mass meter’s electronic circuitry then converts the voltage to a level suitable for processing in the ECU (Figures 5 through 7).

This device does not need a burn-off process to maintain its measuring precision over an extended period. In recognition of the fact that most deposits collect on the sensor element’s leading edge, the essential thermal-transfer elements are located downstream on the ceramic layer. The sensor element is also designed to ensure that deposits will not influence the flow pattern around the sensor.

Intake-manifold pressure sensor

A pneumatic passage connects the intake-manifold to this pressure sensor, which monitors the absolute pressure [kPa] within the intake manifold.

The unit can be constructed as an installation component for the ECU or as a remote sensor for mounting on or near the intake manifold. A hose connects the installation unit to the manifold.

Fig. 5

Hot-film air-mass meter

a Housing, **b** Hot-film sensor
(installed in center of housing).
1 Heat sink, **2** Intermediate component,
3 Power chip, **4** Hybrid circuit, 5 Sensor element.

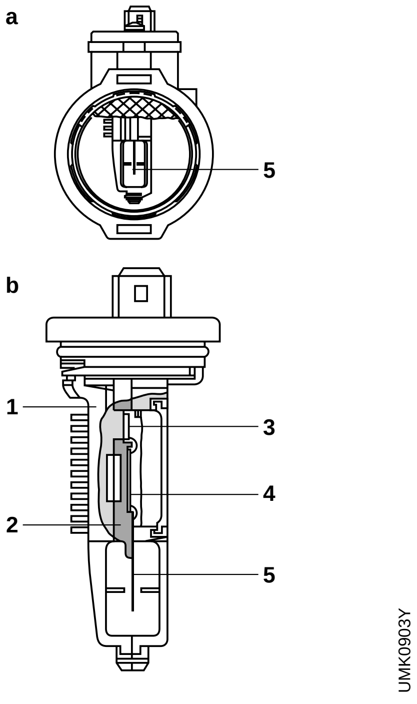
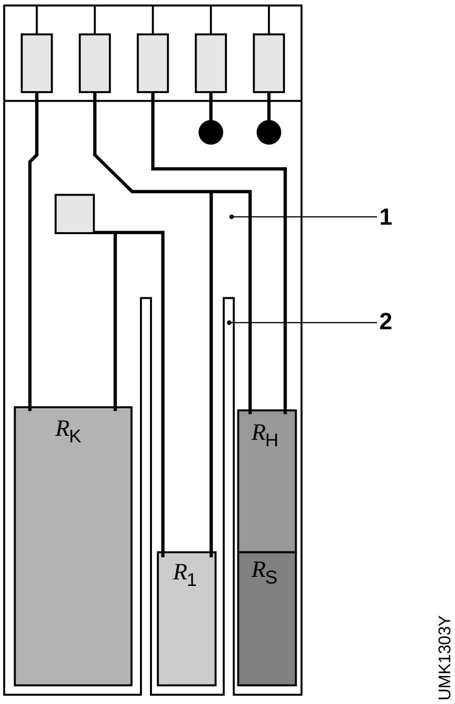


Fig. 6

Hot-film sensor element

1 Ceramic substrate, **2** Sawcut.
R_K Temperature-compensation sensor,
R_1 Bridge resistor, R_H Heater resistor,
R_S Sensor resistor.



The sensor is divided into a pressure cell with two sensor elements and a chamber for the evaluation circuitry. Sensor elements and evaluation circuitry are located on a single ceramic layer (Figure 8).

The sensor element consists of a bell-shaped thick-layer diaphragm enclosing a reference volume with a specific internal pressure. The diaphragm's deflection is determined by the pressure in the intake manifold.
A series of piezo-resistive resistor elements is arranged on the diagram; the conductivity of these elements varies in response to changes in mechanical tension. These resistors are incorporated in a bridge circuit in such a manner that any deflection at the diaphragm will lead to a change in the bridge balance. The bridge voltage thus provides an indication of intake-manifold pressure (Figure 9).

The evaluation circuit amplifies the bridge voltage, compensates for temperature effects and linearizes the pressure response curve. The output signal from the evaluation circuit is transmitted to the ECU.

Throttle-valve sensor

This sensor provides a secondary load signal based on the angle of the throttle valve. The applications for this secondary load signal include providing information for dynamic functions, load-range recognition (idle, full or part-throttle), and serving as a backup signal in the event of main-sensor failure.

The throttle-valve sensor is attached to the throttle-valve assembly where it shares a common shaft with the throttle valve. A potentiometer evaluates the throttle valve's deflection angle and transmits a voltage ratio to the ECU via a resistance circuit (Figures 10 and 11).

Circuit of hot-film air-mass meter

R_K Temperature-compensation sensor,
R_H Heater resistor, R_1, R_2, R_3 Bridge resistors,
U_M Measurement voltage, I_H Heater current,
t_L Air temperature, Q_M incoming air mass per unit of time.

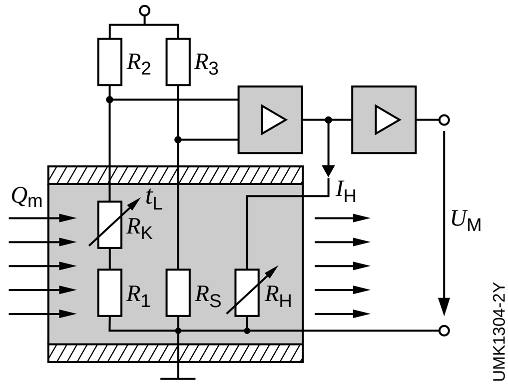


Fig. 7

Fig. 8

Pressure sensor (for installation in ECU)

1 Pressure connection, **2** Pressure cell with sensor elements, **3** Sealing edge, **4** Evaluation circuitry, **5** Thick-film hybrid.

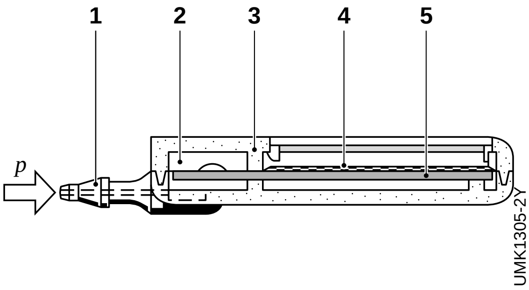
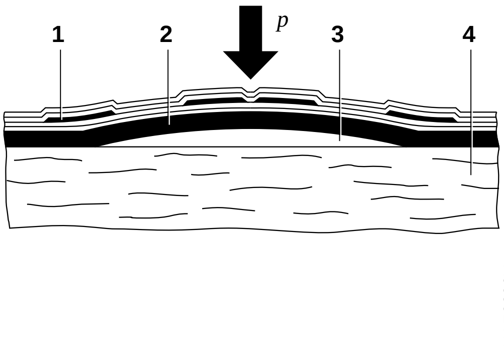


Fig. 9

Thick-film diaphragm in pressure sensor

1 Piezoresistive resistors, **2** Base diaphragm,
3 Reference pressure chamber, **4** Ceramic substrate.
p Pressure.



More exacting precision is required when the throttle-valve sensor is used as the primary load sensor. This higher level of precision is obtained by using a throttle-valve sensor incorporating two potentiometers (two angle ranges) as well as improved suspension.

The control unit determines the mass of the intake air by monitoring throttle-valve angle and engine speed. Data from temperature sensors allows the unit to respond to variations in the air mass due to temperature change.

Fig. 10

Throttle-valve sensor

1 Throttle-valve shaft, **2** Resistor track 1,
3 Resistor track 2, **4** Wiper arm with wiper,
5 Electrical connection.

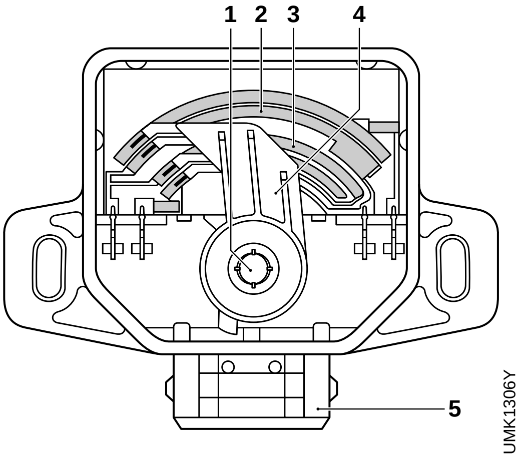
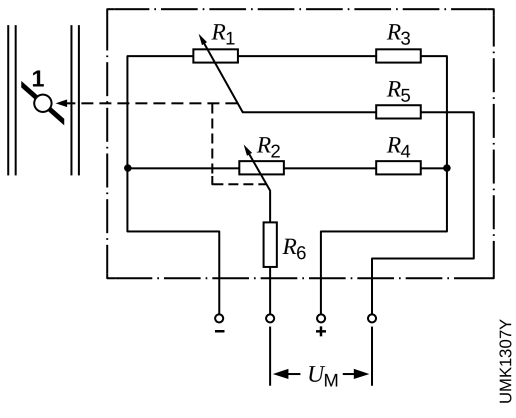


Fig. 11

Throttle-valve sensor circuitry

U_M Measurement voltage, R_1, R_2 Resistor tracks 1 and 2, R_3, R_4, R_5 adjustment resistors.
1 Throttle valve.



Engine speed, crankshaft and camshaft positions

Engine speed and crankshaft position

The degree of piston travel within the cylinder is employed as a measured variable for determining the firing point. The pistons in all cylinders are connected to the crankshaft via the connecting rods. A sensor at the crankshaft thus provides the information on the locations of the pistons in the cylinders.

The speed at which the crankshaft changes its position is the engine speed, defined in the number of crankshaft revolutions per minute (rpm). This also represents another important Motronic input variable for the Motronic unit, and is calculated from the crankshaft position signal. Although the signal from the crankshaft sensor basically indicates crankshaft position, which is then converted to an engine-speed signal in the ECU, the device has come to be known as the engine-speed, or rpm sensor.

Generating the crankshaft position signal

Installed on the crankshaft is a ferromagnetic ring gear with a theoretical capacity of 60 teeth, whereby two teeth are missing on the gear in question (tooth gap). An inductive speed sensor registers the 58-tooth sequence. This sensor consists of a permanent magnet and a soft-iron core with a copper winding (Figure 12).

The magnetic flux field at the sensor responds as the teeth on the sensor gear pass by, generating AC voltage (Figure 13). The amplitude of this AC voltage decreases as the interval between sensor and sensor gear increases, and rises in response to higher engine speeds. Sufficient amplitude is already available at a minimal engine speed (20 min^{-1}). Pole and tooth geometry must be matched. The evaluation circuit in the ECU converts the sinus voltage with its highly varying amplitudes into square-wave voltage with a constant amplitude.

Calculating the crankshaft position

The flanks of the square-wave voltage are transmitted to the computer via an interrupt input. A gap in the tooth pattern is registered at those points where the flank interval is twice as large as in the previous and subsequent periods. The tooth gap corresponds to a specific crankshaft position for cylinder no. 1. The computer synchronizes the crankshaft position according to this point in time. It then counts 3 degrees further for every subsequent positive or negative tooth flank. The ignition signals, however, must be transmitted in smaller stages. The duration between two tooth flanks is thus further divided by four. The time unit thus derived can be multiplied by two, three or four and added to a tooth flank for the ignition advance angle (allowing steps of 0.75 degrees).

Calculating segment duration and engine speed from the engine-speed sensor signal

The relatonship between the cylinders of the four-stroke engine is such that two crankshaft rotations (720 degrees) elapse between the start of each new cycle at cylinder no. 1.
This interval is the mean ignition interval, and is referred to as the segment time T_s. When the interval is distributed equally, the result is:

Interval	Degrees	Teeth
2 cylinders	360	60
3 cylinders	240	40
4 cylinders	180	30
5 cylinders	144	24
6 cylinders	120	20
8 cylinders	90	15
12 cylinders	60	10

Fig. 12

Engine-speed sensor

1 Permanent magnet, **2** Housing,
3 Engine housing, **4** Soft-iron core, **5** Winding,
6 Ring gear with reference point (tooth gap).

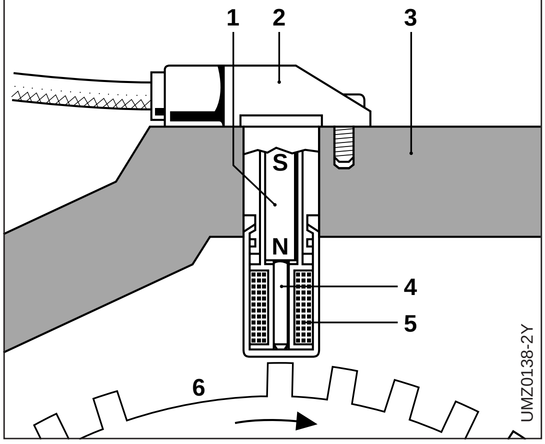
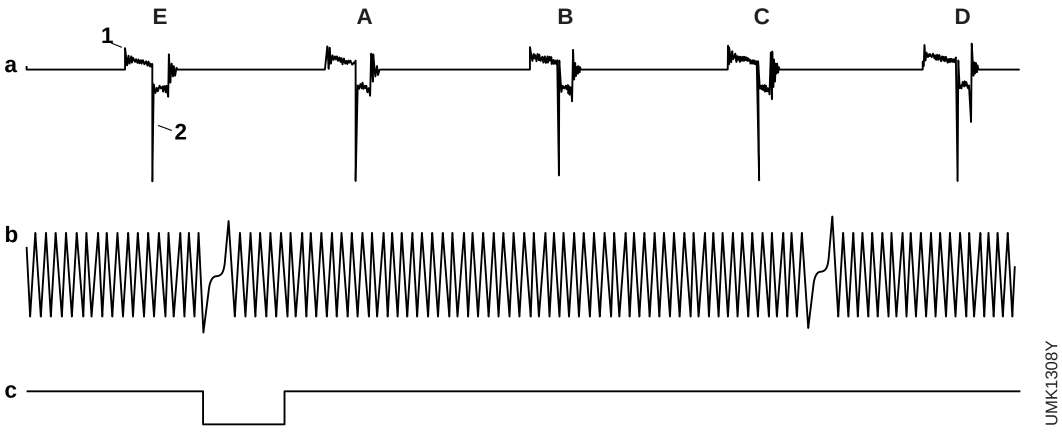


Fig. 13

Signal patterns for ignition, crankshaft and camshaft

a Ignition-coil secondary voltage, **b** Crankshaft rpm sensor signal,
c Camshaft Hall-sensor signal.
1 Close, 2 Ignition.



Ignition, injection and the engine speed derived from the segment time are recalculated for every new interval. The figure for rotational speed describes the mean crankshaft rpm within the segment time and is proportional to its reciprocal.

Camshaft position

The camshaft controls the engine's intake and exhaust valves while operating at half the rotating speed of the crankshaft. When a piston travels to top dead center, the camshaft uses the settings of the intake and exhaust valves to determine whether the cylinder is in the compression phase to be followed by ignition, or in the exhaust phase. This information cannot be derived from the crankshaft position.

If the ignition is equipped with a high-voltage distributor with a direct mechanical link to the camshaft, the rotor will point to the correct cylinder automatically: the ECU does not require supplementary information on the position of the camshaft. In contrast, Motronic systems featuring stationary voltage distribution and single-spark ignition coils require additional information, as the ECU must be able to decide which ignition coil and spark plug are due to be triggered. To do so, it must be informed as to the camshaft's position.

The position of the camshaft must also be monitored in those systems where separate injection timing is used for each individual cylinder, as is the case with sequential injection (SEFI).

Hall-sensor signal

Camshaft position is usually monitored with a Hall sensor. The monitoring device itself consists of a Hall element with a semiconductor wafer through which current flows. This element is controlled by a trigger wheel that turns together with the camshaft. It consists of a ferromagnetic material and generates voltage at right angles to the direction of the current as it passes the Hall element (Figure 13).

Calculating camshaft position

As the Hall voltage lies in the millivolt range, it is processed within the sensor before being transmitted to the ECU in the form of a switching signal. In the simplest case, the computer responds to trigger-wheel gaps by checking to see whether Hall voltage is present and whether or not cylinder no. 1 is on its power stroke.

Special trigger-wheel designs make it possible to use the camshaft signal as a backup for emergency operation in case of crankshaft (engine-speed) sensor failure. However, the resolution provided by the camshaft signal is too imprecise to allow it to be employed as a permanent replacement for the crankshaft speed sensor.

Mixture composition

Excess-air factor λ

The lambda oxygen sensor monitors the excess-air factor λ. Lambda defines the number for the mixture's A/F ratio. The catalytic converter functions best at $\lambda = 1$.

Lambda oxygen sensor

The Lambda oxygen sensor's outer electrode extends into the exhaust stream, while the inner electrode is exposed to the surrounding air (Figure 14).

The essential constituent of the oxygen sensor is a special-ceramic body featuring gas-permeable platinum electrodes on its surface. Sensor operation is based on the ceramic material's porosity, which allows oxygen in the air to diffuse (solid electrolyte). The ceramic material becomes conductive at higher temperatures. Voltage is generated at the electrodes when different oxygen levels are present on the respective sides. A stoichiometric air/fuel ratio of $\lambda = 1$ produces a characteristic jump (jump function) in the response curve (Figure 16).

The oxygen sensor's voltage and internal resistance are both sensitive to temperature. Reliable control operation is possible with exhaust-gas temperatures exceeding 350 °C (unheated sensor), or 200 °C (heated sensor).

Heated Lambda oxygen sensor

The design of the heated Lambda oxygen sensor is largely the same as that of the unheated version (Figure 15). A ceramic heater element warms the sensor's active ceramic layer from the inside, ensuring that its ceramic material remains hot enough for operation – even at low exhaust-gas temperatures. The heated sensor is protected by a tube with a restricted flow opening to prevent the sensor's ceramics from being cooled by low-temperature exhaust gases.

The heated sensor reduces the waiting period between engine start and effective closed-loop control. It also provides reliable control with lower-temperature exhaust gases (e.g., at idle). Heated sensors offer shorter reaction times for reduced closed-loop response intervals. This type of sensor also offers greater latitude in selecting the installation position.

Fig. 14

Lambda oxygen sensor location in exhaust pipe

1 Special ceramic coat, **2** Electrodes, **3** Contact, **4** Housing contact, **5** Exhaust pipe, **6** Ceramic shield (porous), **7** Exhaust gas, **8** Air.

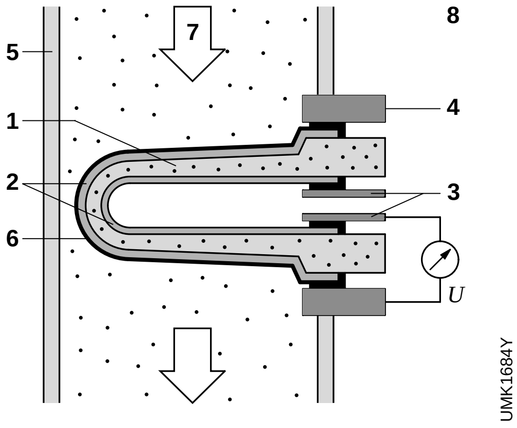


Fig. 16

Lambda oxygen sensor voltage curve at 600 °C operating temperature

a Rich mixture (air deficiency), **b** Lean mixture (excess air).

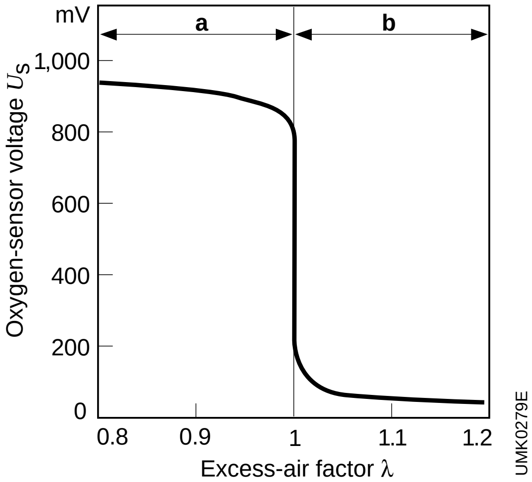
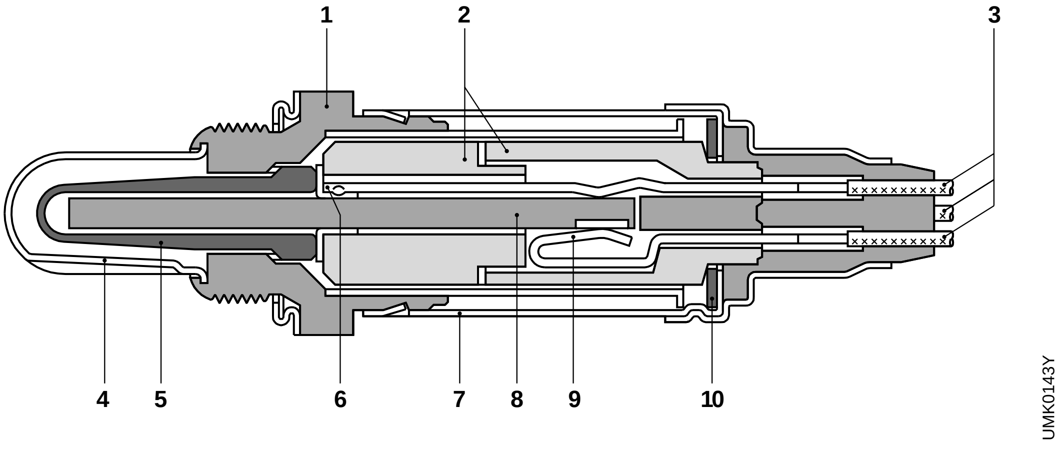


Fig. 15

Heated Lambda oxygen sensor

1 Probe housing, **2** Ceramic shield tube, **3** Electrical connections, **4** Shield tube with slits, **5** Active ceramic sensor layer, **6** Contact, **7** Shield, **8** Heating element, **9** Clamp connections for heater element, **10** Disc spring.



Combustion knock

Under certain conditions combustion in the spark-ignition engine can degenerate into an abnormal process characterized by a typical “knocking” or “pinging” sound. This phenomenon is an undesirable combustion process known as “knocking”, which limits the engine’s output and specific efficiency levels. It occurs when fresh mixture preignites in spontaneous combustion before being reached by the expanding flame front.

Normally initiated combustion and the piston’s compressive force lead to the pressure and temperature peaks that produce self-ignition in the end gas (remaining unburned mixture). Flame velocities in excess of 2,000 m/s can occur, as compared to speeds of roughly 30 m/s for normal combustion. This abrupt combustion process produces substantial local pressure increases in the end gas. The resulting pressure wave propagates until stopped by impact with the cylinder walls representing the outer extremity of the combustion chamber.

Chronic preignition is accompanied by pressure waves and increased thermal stresses at the cylinder-head gasket, piston and in the vicinity of the valves. All of these factors can lead to mechanical damage.

Fig. 17

Knock sensor

1 Seismic mass, **2** Cast mass,
3 Piezoelectric ceramic, **4** Contacts,
5 Electrical connection.

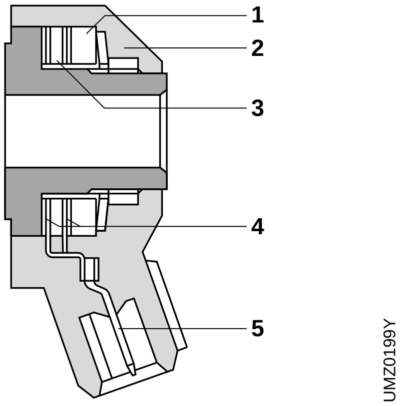


Fig. 19

“Listening” positions for the knock sensors

1 The knock sensor is located between the second and third cylinders. **2** If two knock sensors are fitted, these are located between the first and second, and the third and fourth cylinders.

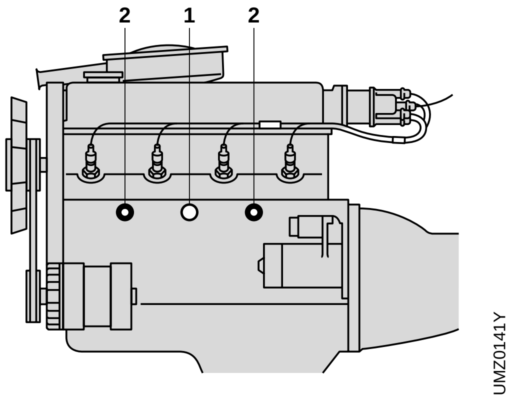
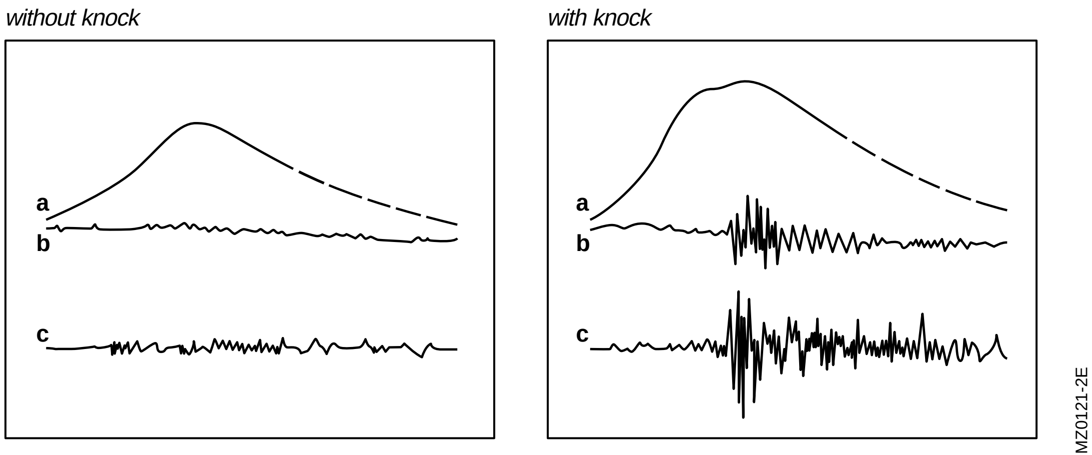


Fig. 18

Knock-sensor signals

The knock sensor supplies a signal **c** corresponding to the pressure pattern **a** in the cylinder.
The filtered pressure signal is shown in **b**.



The characteristic vibration patterns generated by combustion knock can be monitored by knock sensors for conversion into electrical signals, which are then transmitted to the Motronic ECU (Figures 17 and 18). Both the number and positions of the knock sensors must be carefully selected. Reliable knock recognition must be guaranteed for all cylinders and under all engine operating conditions, with special emphasis on high loads and engine speeds. As a general rule, 4-cylinder engines are equipped with one, 5 and 6-cylinder engines with two, and 8 and 12-cylinder engines with two or more knock sensors (Figure 19).

Fig. 20

Engine-temperature sensor

1 Electrical connection, **2** Housing,
3 NTC resistor.

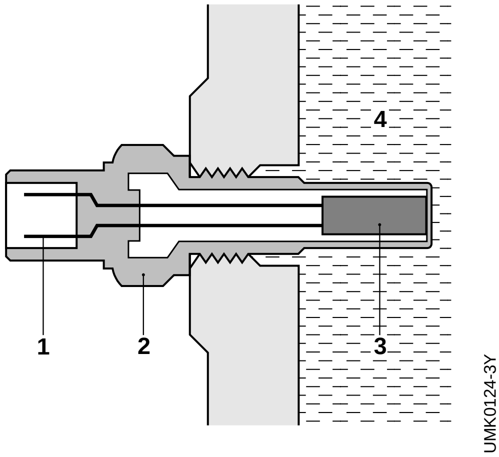
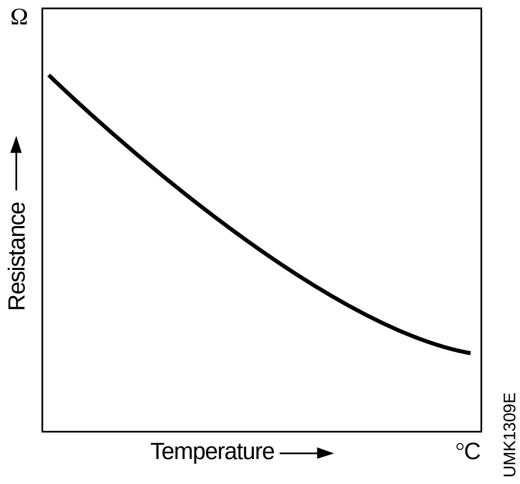


Fig. 21

Temperature-sensor response curve (NTC)



Engine and intake-air temperatures

The engine-temperature sensor incorporates a temperature-sensitive resistor which extends into the coolant circuit whose temperature it monitors. A sensor in the intake tract registers the intake-air temperature in the same fashion (Figure 20).

The resistor is of the negative temperature coefficient type (NTC, see Figure 21) and forms part of a voltage-divider circuit operating with a 5 V supply. An analog-digital converter monitors the resistor's voltage drop, which provides an index of the temperature. Compensation for the non-linear relationship between voltage and temperature is provided by a table stored in the computer's memory; the table matches each voltage reading with a corresponding temperature.

Battery voltage

The electromagnetic injector's opening and closing times are affected by the battery's voltage. Should voltage fluctuations occur in the vehicle's electrical system, the ECU will prevent response delays by adjusting the duration of the injection process. At low battery voltages the ignition circuit's dwell times must be extended to provide the coil with the opportunity to accumulate sufficient ignition energy.

Operating-data processing

Processing load signals

Monitored variables

The ECU uses the signals for load and engine speed to calculate a load signal corresponding to the air mass inducted into the engine during each stroke. This load signal serves as the basis for calculations of injection duration and for addressing the programmed response curves for ignition advance angle (Figure 1).

Monitoring air mass

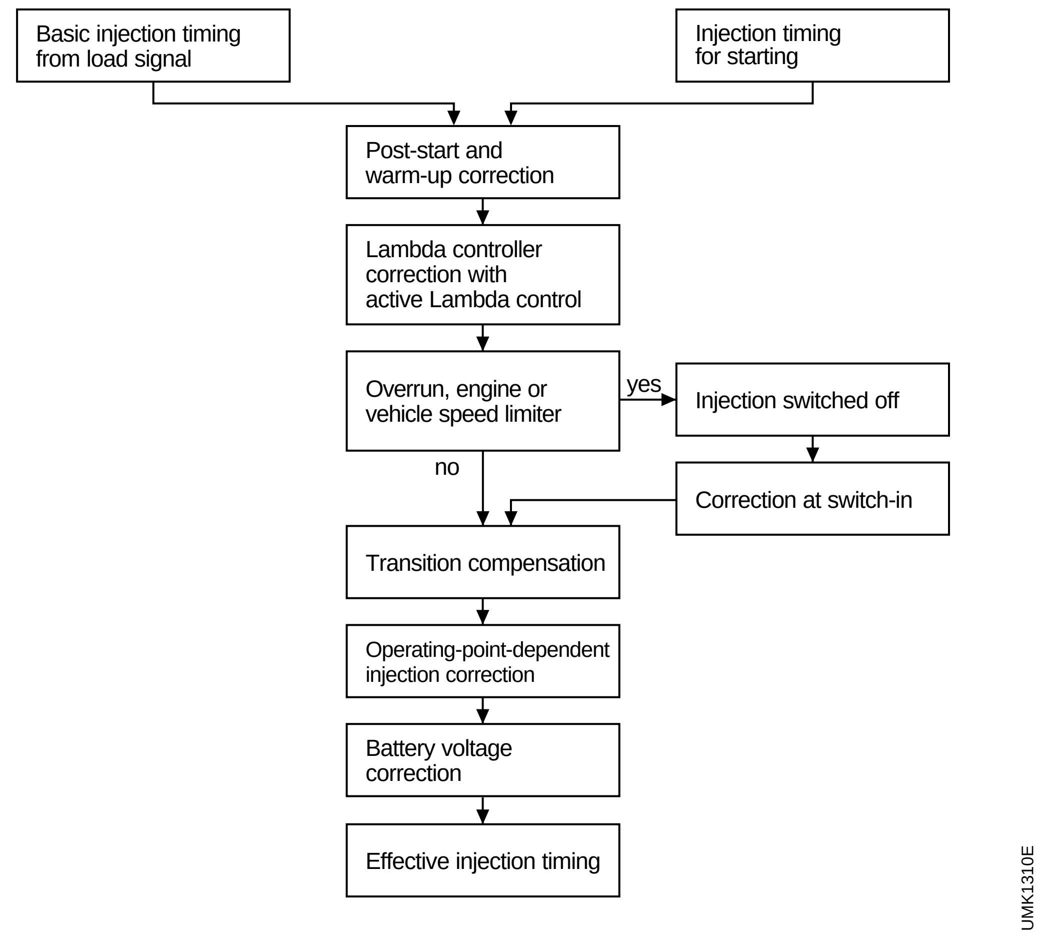
Hot-wire or hot-film air-mass meters measure the air mass directly, producing a signal that is suitable for use as a parameter in load-signal calculations. When an air-flow meter is used, density correction is also required before air mass and load signal can be determined.
In special cases monitoring errors due to heavy pulsation in the intake manifold also receive compensation in the form of a pulsation correction.

Monitoring pressure

Pressure-monitoring systems (using a pressure sensor to determine load) differ from air-mass monitoring systems in that no direct formulas are available for defining the relationship between intake pressure and the air-mass intake. The ECU therefore calculates the load signal with the aid of corrections stored in a program map.
Subsequent compensation is provided for changes in temperature and residual gas relative to the initial state.

Fig. 1

Calculating injection timing



Measuring throttle-valve angle

When a throttle-valve sensor is used, the ECU determines the load signal with reference to engine speed and throttle-valve angle. Compensation for variations in air density are based on readings for temperature and ambient pressure.

Calculating injection timing

Base injection timing

The base injection timing is calculated directly from the load signal and from the injector constants, and defines the relationship between the duration of the activation signal and the flow quantity at the injector. This constant thus varies according to injector design. When the injection duration is multiplied by the injector constant the result will be a fuel mass corresponding to a particular air mass for each stroke.

The base setting is selected for an excess-air factor of $\lambda = 1$.

This remains valid for as long as the pressure differential between fuel and intake manifold stays constant. When it varies, a λ correction map compensates for this influence on injection times.

Meanwhile, a battery-voltage corrector compensates for the effects that fluctuations in battery voltage have on the injectors' opening and closing times.

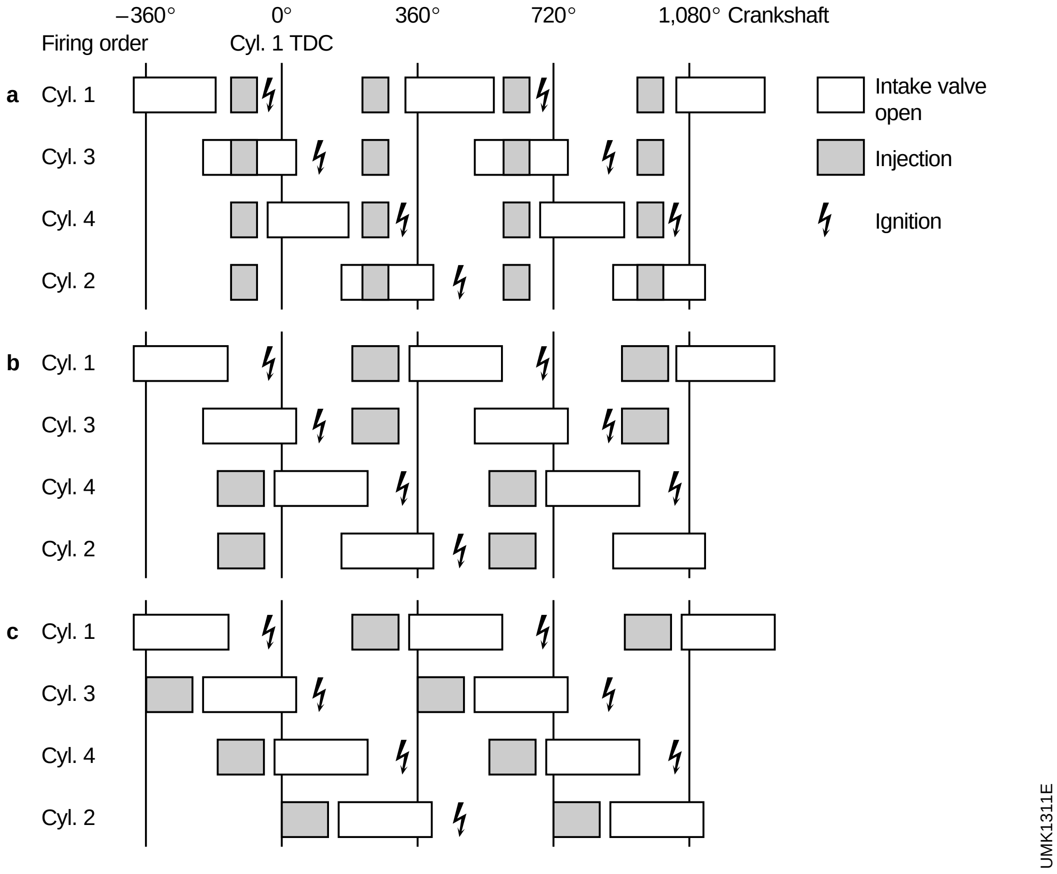
Effective injection time

The effective injection time results when the correction factors are included in the calculations. The correction factors are determined in corresponding special functions and provide adjustment data for varying engine operating ranges and conditions. The correction factors are used both individually and in combinations according to applicable parameters.

Fig. 2

Comparison of injection types

a Simultaneous injection, **b** Group injection, **c** Sequential injection.



The process for calculating the injection time is illustrated in Figure 1. The individual operating ranges and conditions will be explained in more detail in the following chapters.
Once cylinder filling drops below a certain level, the mixture will cease to ignite. Restricting the injection time thus prevents the formation of unburned hydrocarbons in the exhaust gas.
For starting, the injection time is calculated separately using criteria independent of the calculated load signal.

Injection mode
In addition to the injection time, the injection mode is yet another important parameter for fuel economy and exhaust emissions. The range of options depends upon the type of injection system (Figure 2):

- Simultaneous injection,
- Group injection, or
- Sequential injection.

Simultaneous injection
With simultaneous injection, the injection process is triggered twice per cycle at a specific point in time at all injectors, that is, once for each camshaft revolution, or once for every two crankshaft revolutions. The injection mode is static.

Group injection
Group injection combines the injectors in two groups, with each group being triggered once per cycle. The time interval between the two triggering points is equal to one crankshaft rotation period. This arrangement makes it possible to use the engine operating point as the essential criterion in selecting the injection mode while also preventing undesirable spray through the open intake valves throughout a wide range in the program map.

Sequential injection
This type of injection provides the highest degree of design latitude. Here, the injection processes from the individual injectors take place independently of each other at the same point in the cycle referred to the respective cylinder. There are no restrictions on injection timing, which can be freely adapted to correspond with the optimization criteria.

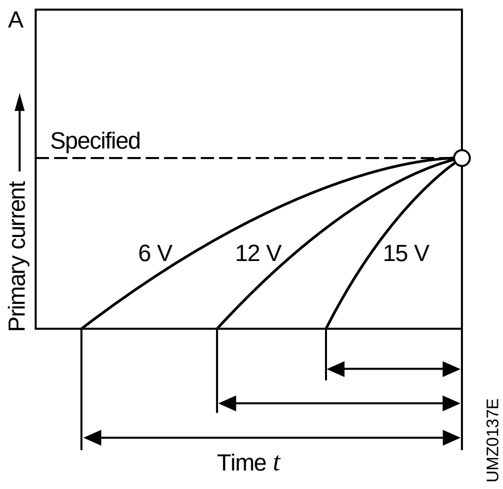
Comparison
Group and sequential injection require a wider injector variation range (range extending from the minimum quantity at idle to the maximum under full throttle) than simultaneous injection.

Controlling dwell angle

The dwell angle varies the ignition coil’s current-flow period according to engine speed and battery voltage. The dwell angle is selected to ensure availability of the required primary current at the end of the current flow time throughout the widest possible range of operating conditions.
The dwell angle is based on the ignition coil’s charge time, which, in turn, depends upon battery voltage (Figure 3). A supplementary dynamic reserve makes it possible to supply the required current even during sudden shifts to high engine speeds.
The charge time is restricted in the upper rpm range to maintain an adequate arcing time at the spark plug.

Fig. 3

Primary current pattern at various system voltages

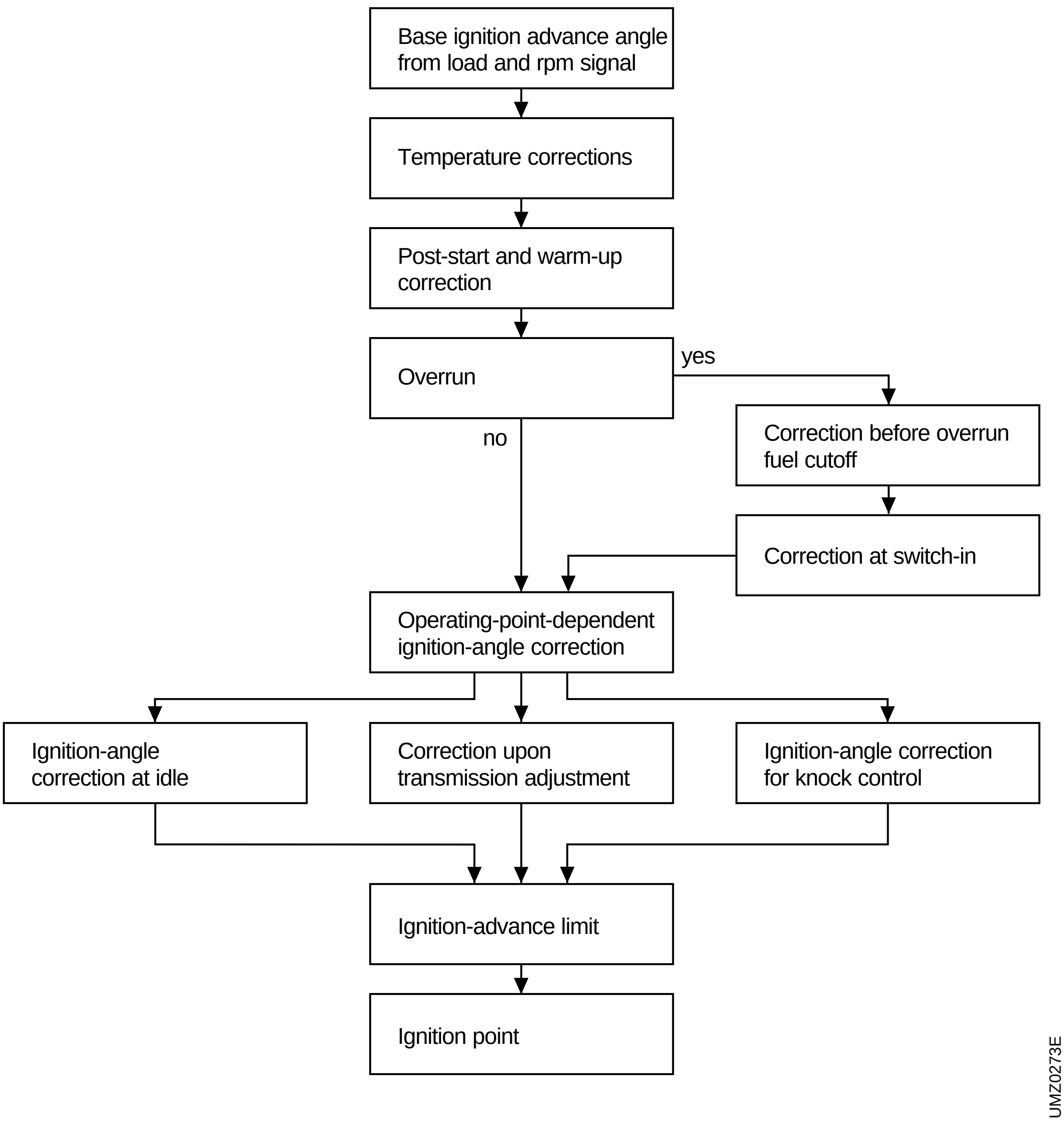


Controlling ignition advance angle

A program map containing the basic ignition timing for various engine loads and speeds is stored in the memory of the M-Motronic ECU. This ignition advance angle is optimized for minimal fuel consumption and exhaust emissions.
Data for engine and intake-air temperature (monitored via sensors) provide the basis for corrections to compensate for temperature variations. The unit can supply additional corrections and/or revert to other program maps to adapt to all operating conditions. Thus the mutual effects of torque, emissions, fuel consumption, preignition tendency and driveability can all be taken into account. Special ignition-angle correction factors are active during operation with secondary air injection or exhaust-gas recirculation (EGR) as well as in dynamic vehicle operation (e.g., when accelerating).
The various operating ranges (idle, part throttle, full throttle, start and warm-up) continue to be taken into account. Figure 4 shows a flow chart describing ignition-angle processing.

Fig. 4

Calculating ignition timing



Operating conditions

Start

Special calculations are employed to determine the injection quantity for the duration of the starting procedure.

In addition, special injection timing is used for the initial injection pulses. The injection quantity is augmented in accordance with engine temperature to promote formation of a fuel film on the walls of the intake manifold, thereby compensating for the engine's higher fuel requirements as it runs up to speed. As soon as the engine starts to turn over, the quantity of supplementary fuel is reduced and then cancelled once the engine starts to run.

Ignition advance angle is also specially adjusted for starting. The adjustment occurs with reference to engine temperature and engine speed.

Post-start phase

The post-start phase is characterized by further reductions in the supplementary injection quantity. The reductions are based on engine temperature and the elapsed time since the end of the starting process. The ignition advance angle is also adjusted to correspond to the revised fuel quantities and the different operating conditions. The post-start phase terminates with a smooth transition to the warm-up phase.

Warm-up phase

Different strategies can be employed for the warm-up phase, depending upon engine and emissions-control design. The decisive criteria are drivability, emissions and improved fuel economy. A lean warm-up combined with retarded ignition timing raises the temperature of the exhaust gas. Another way to obtain high exhaust temperatures is to employ a rich warm-up mixture together with secondary-air injection. Here air is injected into the exhaust system downstream from the exhaust valves for a brief period after the engine starts. A secondary-air pump can provide this additional air. When the temperature is high enough, this excess air supports oxidation of HC and CO in the exhaust system while simultaneously generating the desired high exhaust temperatures (Figure 1).

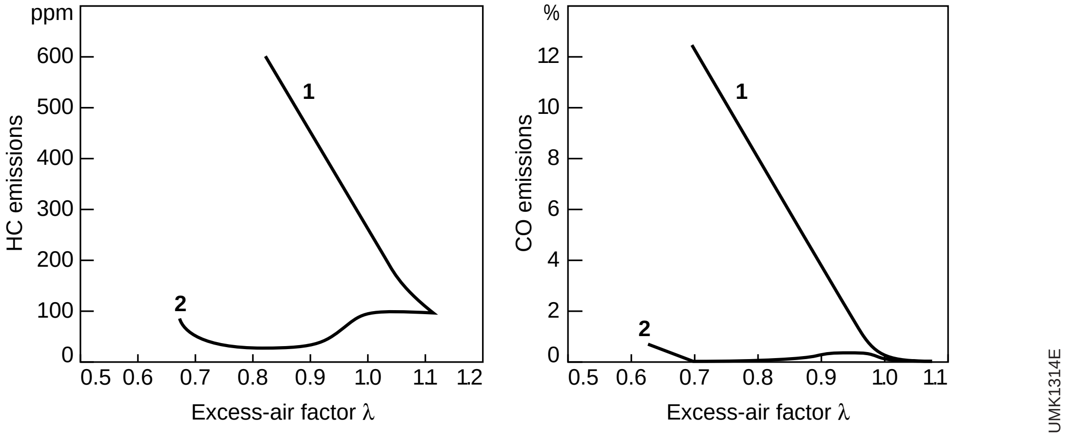
Both of these measures help the catalytic converter to begin effective operation sooner.

The effects of the adjustments to ignition angle and injection timing can be supplemented by higher idle speeds. These are provided by a specially designed air injection unit, and also result in shorter warm-up times at the catalytic converter.

Fig. 1

Effect of secondary air on HC and CO

1 Without secondary air,
2 With secondary air.



Once the converter reaches operating temperature the injection is governed to $\lambda = 1$. This is accompanied by a corresponding adjustment in ignition angle.

Transition compensation

Acceleration/deceleration

A portion of the fuel sprayed into the intake manifold does not reach the cylinder in time for the next combustion process. Instead, it forms a condensation layer along the walls of the intake tract. The actual quantity of fuel stored in this film increases radically in response to higher loads and extended injection times.

A portion of the fuel injected when the throttle valve opens is used for this film. A corresponding amount of supplementary fuel must thus be injected to compensate and prevent the mixture from leaning out. Because the additional fuel retained in the wall film is released as the load factor drops, the injection time must be reduced by a corresponding amount during deceleration.

Figure 2 shows the resulting curve for injection times.

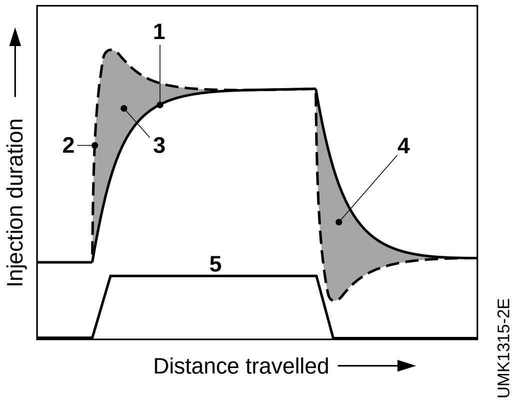
Overrun fuel cutoff/ renewed fuel flow

When the throttle closes, the injection is switched off in the interests of reduced fuel consumption and lower exhaust emissions.

Fig. 2

Transition injection timing

1 Injection timing from load signal,
2 Effective injection timing, **3** Additional fuel,
4 Fuel reduction, **5** Throttle-valve angle α_{DK}.



The injection stop is preceded by a reduction in the ignition advance to attenuate torque jump during the transition to trailing throttle.

The injection starts again once a specific reactivation speed – located above idle speed – is reached. Various activation speeds are stored in the ECU. These vary according to different parameters such as engine temperature and rpm dynamics, and are calculated to prevent excessive drops in engine speed, regardless of operating conditions.

When the injection resumes, it sprays in supplementary fuel to rebuild the fuel wall layer. The ignition advance angle is also adjusted to provide a smooth torque increase.

Closed-loop idle-speed control

Idle

The fuel consumption at idle is largely determined by the engine’s efficiency and by the idle speed. A substantial proportion of the fuel consumed by vehicles in heavy urban traffic is actually burned at idle. The idle speed should thus be as low as possible. At the same time, the idle should never fall so far that rough running or stalling occur, even under additional loads such as electrical equipment, air conditioner, automatic transmission in gear, power steering, etc.

Idle-speed control

The idle-speed control must maintain a balance between torque generation and engine load in order to ensure a constant idle speed. The load on the idling engine is a combination of numerous elements, including internal friction within the engine's crankshaft and valvetrain assemblies as well as the ancillary drives (for instance, for the water pump). The idle-speed control compensates for this internal friction, which, in turn, changes over the life of the engine. These loads are also extremely sensitive to temperature variations.

In addition to these internal sources of friction, there are also external factors such as the load from the air conditioner that was mentioned above. The load from these external factors is subject to substantial variations as ancillary devices are switched on and off. Modern engines with small flywheel weights and large-volume intake manifolds are especially sensitive to these load fluctuations.

Input variables

In addition to the signal from the engine-speed sensor, the idle-speed control circuit also requires information on throttle-valve angle in order to recognize the idle state (foot off accelerator pedal). Engine temperature is also monitored to allow advance compensation for the effects of temperature. An air mass is specified with reference to engine temperature and the target idle-speed; this idle speed is then corrected in closed-loop operation. Where present, the input signals from the air conditioner and automatic transmission also facilitate the correction process and provide supplementary support for closed-loop idle-speed control.

Fig. 3

Bypass actuator with hose connections

Fig. 4

Manifold-mounted bypass actuator

Actuator adjustments

Physically, for idle-speed control there are three possibilities for adjustment intervention:

Air control

The proven control procedure is to regulate air flow by means of a bypass around the throttle valve, or to adjust the throttle valve itself using either a variable throttle stop or a direct actuator unit as found in “Electronic Throttle Control (ETC).”

On bypass actuators designed for hose connection, the bypass around the throttle valve consists of air hoses and an actuator (Figure 3). More modern are bypass actuators for direct installation; this type of bypass-air regulation device is flange-mounted directly on the throttle-valve assembly.

Figure 4: Example of a single-winding rotary actuator for direct installation.

One disadvantage associated with bypass actuators is that they add to the throttle-valve’s own leakage air. Once the engine is well run-in, the combined air flowing through the throttle valve and the bypass actuator could well exceed the air quantity that the engine needs at idle. At this point effective idle regulation is no longer possible. This liability disappears when adjustments to the throttle valve itself are employed to regulate the air flow. The idle throttle device uses an electric motor and gear drive to vary the position of the throttle valve’s idle stop (Figure 5). Delays in idle response are encountered when adjustments to air flow are used in systems with large-volume intake manifolds.

Adjustments to ignition advance angle

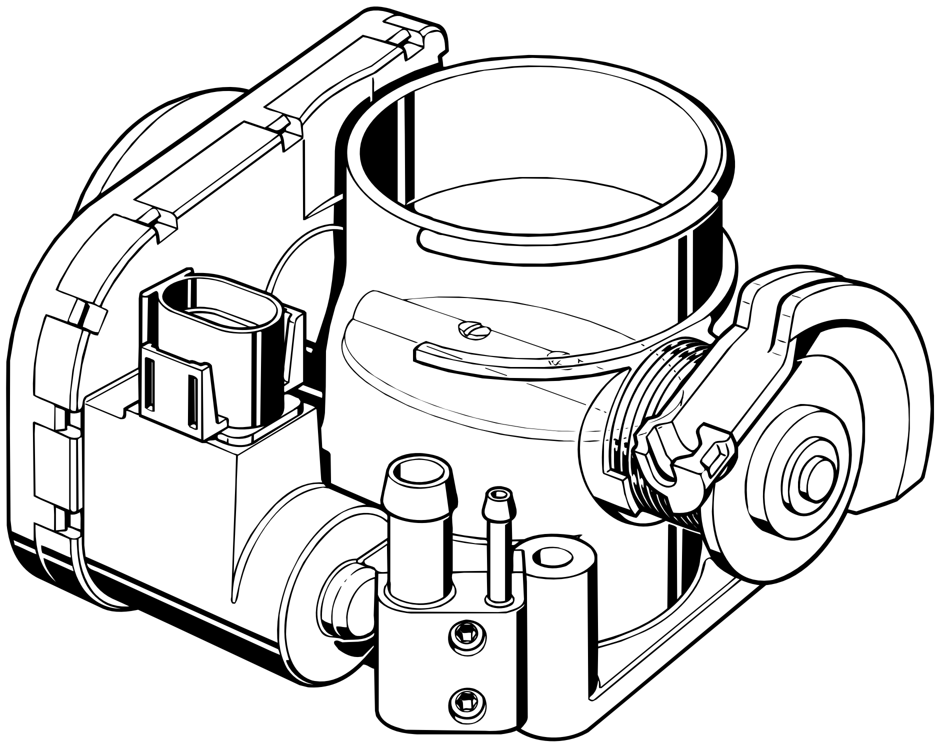
The second (and faster reacting) option is to adjust the ignition advance angle. Systems with an rpm-sensitive ignition advance angle react to sinking engine speeds by increasing the ignition advance to provide a boost in torque.

Mixture composition

Strict emissions-control regulations and a limited range of practical possibilities relegate the mixture-adjustment option to virtual insignificance.

Fig. 5

Throttle-valve assembly with integral idle actuator



Lambda closed-loop control

Post-treatment of exhaust gases in a three-way catalytic converter is an effective means for reducing concentrations of harmful exhaust emissions. The converter transforms the three pollutants CO, HC and NO_X into H_2O, CO_2 and N_2.

Control range

The range available for simultaneous conversion of all three of the above components is extremely narrow, and is termed the “lambda window” (λ = 0.99 . . . 1). This means that closed-loop Lambda control is essential.

A Lambda oxygen sensor is installed in the exhaust system upstream from the catalytic converter, where it monitors the exhaust-gas oxygen content.

Lean mixtures ($\lambda > 1$) produce a sensor voltage of approx. 100 mV, while a rich mixture ($\lambda < 1$) generates approx. 800 mV. At $\lambda = 1$ the sensor voltage jumps abruptly from one level to the other (Figure 6).

The ECU uses the signal from the air-mass meter and the monitored engine speed to generate an injection signal. At the same time, it also produces a supplementary Lambda-control factor from the Lambda-sensor signal for use in correcting the injection time.

Operation

The Lambda oxygen sensor must be operational before the Lambda closed-loop control circuit can function. An auxiliary evaluation circuit monitors this factor on a continuing basis. A cold oxygen sensor or damaged circuitry (short or open circuits) will generate implausible voltage signals which are rejected by the ECU. The heated Lambda sensors used in most systems are ready for operation after only 30 seconds.

Cold engines require a richer mixture ($\lambda < 1$) to idle smoothly. For this reason the Lambda closed-loop control circuit can only be activated once a set temperature threshold has been passed.

Once the lambda control is activated, the ECU uses a comparator to convert the signal from the sensor into a binary signal.

The controller reacts to the transmitted signal ($\lambda > 1$ = mixture too lean, or $\lambda < 1$ = mixture too rich), by modifying the control variables (with an initial jump followed by a ramp progression). The injection time is adjusted (lengthened or shortened), and the control factor reacts to the continuing data transfer by settling into a constant oscillation (Figure 7). The duration of the oscillation periods is determined by the flow times of the gas, while the “ramp climb” maintains largely constant amplitudes within the load

Fig. 6

Closed-loop control range for Lambda oxygen sensor with emissions reduction

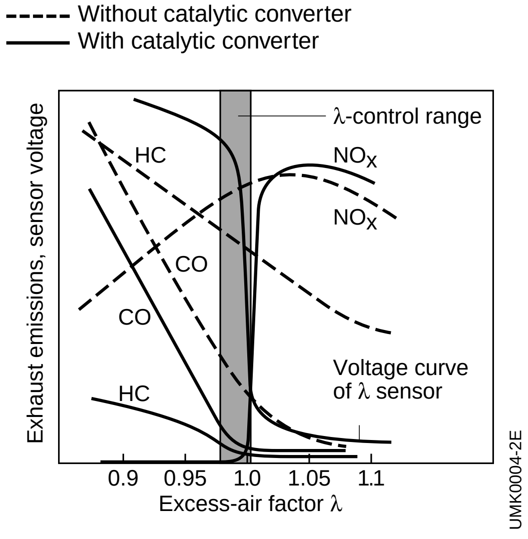
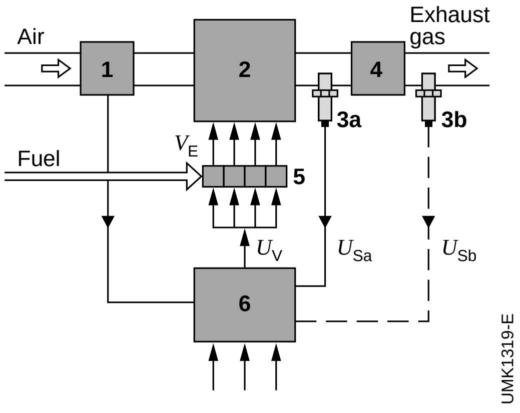


Fig. 7

Diagram of Lambda closed-loop control circuit

1 Air-mass meter, **2** Engine, **3a** Lambda oxygen sensor 1, **3b** Lambda oxygen sensor 2 (as required), **4** Catalytic converter, **5** Injectors, **6** ECU. U_S Oxygen-sensor voltage, U_V Injector control voltage, V_E Injected fuel quantity.



speed range despite variations in the travel time of the gas.

Lambda shift

The optimum conversion range and the voltage jump at the oxygen sensor do not coincide precisely. An asymmetrical control oscillation pattern can be used to shift the mixture into the optimal range ($\lambda = 1$). The asymmetry is obtained either by delaying the switch in the control factor after the voltage jump (from lean to rich) at the oxygen sensor or by providing an asymmetrical jump. This is the case when the voltage jump at the oxygen sensor during the transition from lean to rich is different from that produced at the change from rich to lean.

Adapting the pilot settings to the Lambda closed-loop control

Lambda closed-loop control corrects the subsequent injection process on the basis of the previous measurement at the oxygen sensor, whereby an unavoidable time lag results from the gas travel times. For this reason, the approach to a new operating point is accompanied by deviations from $\lambda = 1$ due to incorrect pilot control, a condition alleviated once the closed-loop control picks up the new cycle. Thus a special pilot control mechanism is needed to maintain compliance with emissions limits. The pilot-control is defined during adaptation to the engine and the Lambda response curve is stored in the ROM. However, revisions may become necessary due to the drift that can occur during the life of the vehicle. Among the drift factors are variations in the density and quality of the fuel. When the Lambda controller consistently repeats the same corrections in a certain load and engine-speed range, the pilot-control adaption mechanism recognizes this state. It corrects the pilot control for this range and records the correction in a RAM memory chip (with an uninterrupted current supply). The corrected pilot control is thus ready to respond immediately at the next start, assuming duty until the Lambda closed-loop control becomes operative. Power interruptions in the current to the permanent memory are also recognized; the adaptation process then commences again with neutral default values providing the initial basis for operation.

Dual-sensor Lambda closed-loop control

Installing the Lambda oxygen sensor downstream of the catalytic converter provides better protection against contamination from the exhaust gas (here, “downstream” = on the tailpipe side). This type of backup sensor can provide a second control signal to augment the one from the main sensor upstream of the converter (here, “upstream” = on the engine side). The second signal is superimposed on the first to provide stable mixture composition over an extended period (Figure 7).

The superimposed control modifies the asymmetry in the constant oscillation pattern that is associated with control concepts based on the oxygen sensor mounted upstream from the converter; this compensates for the lambda shift. A Lambda closed-loop control strategy based solely on the downstream-mounted oxygen sensor would feature excessive response delays due to the gas travel times.

Evaporative-emissions control systems

Origins of fuel vapors

The fuel in the fuel tank is heated by:

- – Heat radiation from outside sources, and
- – The excess fuel from the system return line which was heated during its passage through the engine compartment.

The result is the HC emissions that are usually emitted from the fuel tank in the form of vapor.

Limiting HC emissions

Evaporative emissions are subject to legal limits.

Evaporative-emissions control systems restrict these emissions. These systems are equipped with an activated charcoal filter (the so-called carbon canister) located at the end of the fuel tank's vent line. The activated charcoal in the canister binds the fuel vapors and allows only air to escape into the atmosphere, as well as serving as a pressure-release device. In order to ensure that the charcoal can continually regenerate, an additional line leads from the carbon canister to the intake manifold. Vacuum is produced in this line when the engine runs, causing a stream of atmospheric air to flow through the charcoal on its way to the manifold. The fuel vapors stored in the activated charcoal are entrained by the air stream and conducted to the engine for combustion. A so-called canister-purge valve in the line to the intake manifold meters the flow of this regeneration or "cleansing" air (Figure 8).

Regeneration flow

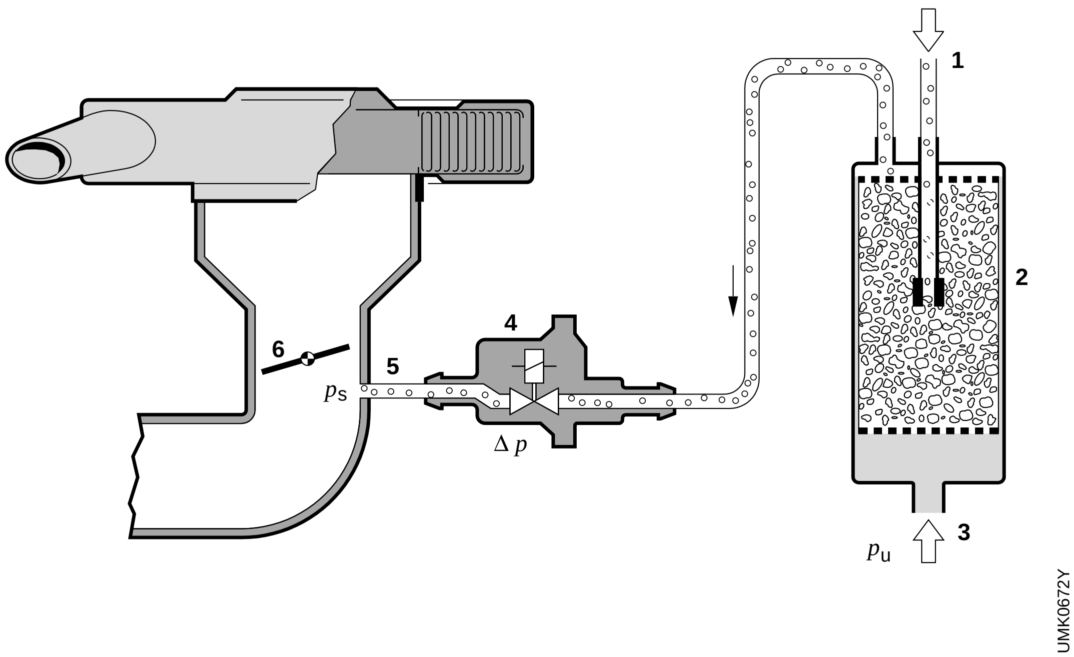
The regeneration flow is an air-fuel mixture of necessarily indeterminate composition, since fresh air as well as air containing substantial concentrations of gasoline vapor can come from the carbon canister. The regeneration flow thus represents a major interference factor for the Lambda control system. A regeneration flow representing 1 % of the intake air and consisting solely of fresh air will

Fig. 8

Evaporative-emissions control system

1 Line from fuel tank to carbon canister,
2 Carbon canister,
3 Fresh air,
4 Canister-purge valve,
5 Line to intake manifold,
6 Throttle valve.

Δp Difference between manifold pressure p_s and atmospheric pressure p_u.



lean out the intake mixture by 1 %. A flow with a substantial gasoline component can enrichen the mixture by something in the order of 30 %, due to the effects on the A/F ratio λ of fuel vapor with a stoichiometric factor of 14.7. In addition, the specific density of fuel vapor is twice that of air.

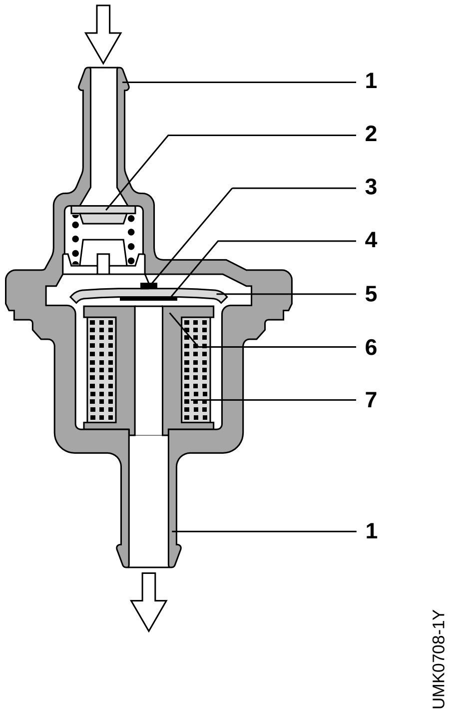
Canister-purge valve
The canister-purge valve’s control mechanism must ensure adequate air-purging of the carbon canister while holding Lambda deviations to a minimum (Figure 9).

ECU control operations
The canister-purge valve closes at regular intervals in order to allow the mixture adaptation process to function without being interfered with by tank ventilation.
The canister-purge valve opens in a “ramp-shaped” pattern. The ECU “memorizes” the resulting Lambda control deviations as mixture corrections from the fuel-regeneration system. The system is designed to operate with up to 40 % of the total fuel coming from the regeneration flow.

Fig. 9

Canister-purge valve

1 Hose connection, **2** Non-return valve,
3 Leaf spring, **4** Sealing element,
5 Solenoid armature, **6** Sealing seat,
7 Solenoid coil.



With the Lambda control system inactive, only small regeneration quantities are accepted, as there would be no control mechanism capable of compensating for the mixture deviations that would occur.
The valve closes immediately in the overrun fuel cutoff mode to prevent unburned gasoline vapors from entering the catalytic converter.

Knock control

Electronic control of the ignition timing allows extremely precise adjustments of the ignition-advance angle based on engine rpm, temperature and load factor. Nevertheless, a substantial safety margin to the knock limit must be maintained. This margin is necessary in order to ensure that no cylinder will reach or go beyond its preignition limit, even when susceptibility is increased by risk factors such as engine tolerances, aging, environmental conditions and fuel quality. The resulting engine design, with its fixed safety margin, is characterized by lower compression and reduced ignition advance. The ultimate results are sacrifices in fuel economy and torque.
These liabilities can be avoided by using a knock sensor, whereby experience has shown that the compression ratio can be raised with accompanying improvements in both fuel economy and torque. With this device, it is no longer necessary to select the default ignition advance angle with the most knock-sensitive conditions in mind. Instead, a best-case scenario can be used (e.g., engine compression at lower tolerance limit, optimum fuel quality, cylinder with minimum preignition tendency). This makes it possible to operate each individual cylinder at the preignition limit for optimum efficiency in virtually all operating ranges for the life of the vehicle.
The essential prerequisite allowing this kind of ignition-angle program is reliable recognition of all preignition beyond a specified intensity, extending to every

cylinder and throughout the engine's entire operating range. The knock sensors are solid-body sonic detectors installed at one or several suitable points on the engine. Here they detect the characteristic oscillation patterns that accompany knock and transform them into electrical signals before transmitting them to the Motronic ECU for processing. This ECU employs a special processing algorithm to detect incipient preignition in any of the combustion cycles in the respective cylinders. When this condition is recognized, the ignition advance angle is reduced by a programmed increment. When the knock danger subsides, the ignition for the affected cylinder is then slowly advanced back to the default setting. The knock recognition and knock-control algorithms are designed to prevent the kind of preignition that results in engine damage as well as audible knock (Fig. 10).

Adaptation

Real-world engine operation results in the individual cylinders having different knock limits, and therefore different ignition points. In order to adapt the default values to reflect the respective knock limits under varying operating conditions, individual ignition-retard increments are stored for each cylinder.

The data are stored in the non-volatile memory programs in the permanent RAM for engine speed and load factor. This allows the engine to be operated at optimum efficiency under all operating conditions without any danger of audible combustion knock, even during abrupt changes in load and rpm.

The engine can even be approved for operation with fuels having a low anti-knock quality. Generally, the engine is adjusted for use with premium-grade gasoline. Operation with regular-grade gasoline can also be approved.

Knock control on turbocharged engines

Systems combining boost pressure and knock control are especially effective on engines with exhaust-gas turbochargers.

Closed-loop knock control

Control algorithm with active ignition intervention on 4-cylinder engine.
$K_{1...3}$ Knock at cylinders 1...3 (no knock at cylinder 4)
a Ignition retard, **b** Step width for advance,
c Advance.

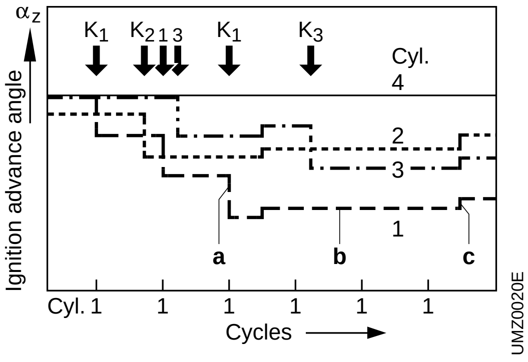


Fig. 10

The initial response to ignition knock is to reduce the timing advance angle. Further action to reduce the pre-ignition tendency in the form of a reduction in boost pressure is initiated only once the ignition-retard limit – which varies according to the temperature of the exhaust gas – has been reached. This makes it possible to maintain exhaust-gas temperatures within acceptable limits while operating the turbocharged engine at the knock limit for optimum efficiency.

Boost-pressure control

Turbocharger boost

The exhaust-gas turbocharger has prevailed in the face of competition from other supercharging methods, such as pressure-wave and mechnical supercharging. Turbochargers make it possible to achieve high torque and output from small-displacement, high-efficiency powerplants. Because a turbocharged engine can be smaller than its naturally-aspirated counterpart producing the same amount of power, it boasts a higher power-to-weight ratio.

Automotive industry research has demonstrated that compared to a standard naturally aspirated engine, a small-displacement turbocharged engine with electronic boost control can provide improvements in fuel economy equal to those achieved with a prechamber diesel. The main components of the exhaust-gas turbocharger are the compressor and the exhaust-gas impeller mounted on the other side of the same shaft. The exhaust-gas turbocharger transforms a portion of the energy in the exhaust gas into the rotation energy used to power the compressor. This, in turn, draws in fresh air and compresses it before blowing it through the intercooler, throttle valve, intake manifold and into the engine.

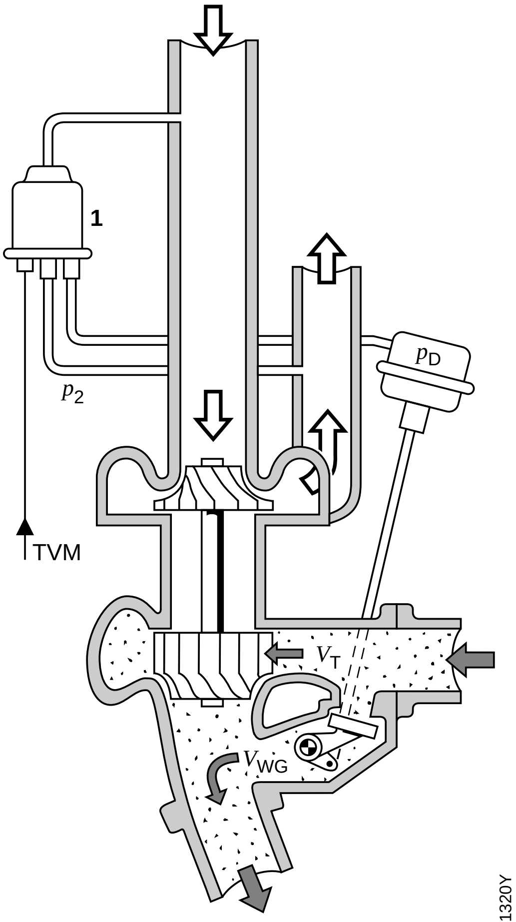
Actuators for exhaust-gas turbochargers

Passenger-car engines must be capable of generating substantial torque at low engine speeds. This is the reason why the turbocharger housing is dimensioned to operate most efficiently with lower mass exhaust-gas flow rates (for example, full load at n = 2,000 min^{-1}). To prevent the turbocharger from overloading the engine at higher exhaust-gas mass flow rates, a bypass mechanism (waste gate) must be included in the housing to divert a portion of the flow around the turbine and into the downstream exhaust system. The bypass valve generally assumes the form of a flap in the turbine housing. Less common is a disk valve installed parallel to the turbine in a separate housing. Variable turbine geometry has still not been used for spark-ignition engines, but could also be combined with boost-pressure control (Figure 11).

Fig. 11

Actuator for electronic boost-pressure control

1	Pulse valve,
p_2	Boost pressure,
p_D	Pressure in diaphragm unit,
TVM	Triggering signal from ECU to pulse valve,
V_T	Flow volume through turbine,
V_{WG}	Flow volume through waste gate.



Boost-pressure (closed-loop) control

Pneumatic-mechanical closed-loop control systems use a turbocharger actuator directly exposed to the boost pressure at the turbocharger outlet. This layout provides only limited latitude for tailoring the progression of the torque curve through the engine speed range. There is only a full-load limit. It is not possible to provide control compensation for tolerances at full-load boost. At part load the closed bypass valve reduces efficiency. Acceleration from low engine speeds can be accompanied by delayed turbocharger reaction (turbo lag).

These disadvantages can be avoided with electronic boost-pressure control (Fig. 11). The specific fuel consumption can be reduced in some part-throttle operating ranges. The system operates by opening the bypass valve, with the following results:

- Residual work on the part of the engine and turbine output are reduced,
- Pressure and temperature at the compressor outlet are lowered, and
- The pressure differential at the throttle valve is lowered.

A linear relationship between torque curve and throttle-valve angle is also obtained, with improved sensitivity at the accelerator pedal. In order to provide the improvements listed above, the exhaust-gas turbocharger and the actuator must be perfectly adapted for use in the individual engine. The affected elements in the actuator are:

- The electropneumatic pulse valve,
- The effective diaphragm surface, stroke and spring in the diaphragm unit, and
- The diameter of the valve disk/flap at the wastegate.

Depending upon the load sensor, the setpoints stored in the M-Motronic ECU with electronic boost-pressure control are for pressure, air quantity or air mass. The setpoints for various engine speeds and throttle-valve angles are stored in a program map.

Actuators within the closed-loop control circuit adjust the monitored actual value to coincide with the value prescribed for the particular operating conditions. The calculated value is transmitted through the controller output in the form of a signal (pulse-width modulated) to the pulse valve. Within the actuator this signal modifies the control pressure and the stroke to change the effective opening at the bypass valve.

The temperature of the exhaust gases between the turbocharged engine and the turbine should never be allowed to exceed certain limits. This is why Bosch only uses boost-pressure control in conjunction with knock control. Knock control is the only means of allowing the engine to run with the maximum potential ignition advance throughout its service life. Running the engine at the optimal ignition advance angle for the specific operating conditions results in extremely low exhaust-gas temperatures.

Additional adjustments to boost pressure and/or mixture can be used to lower the temperature of the exhaust gas even further.

Limiting engine and vehicle speed

Extremely high engine speeds can lead to destruction of the engine (valve train, pistons). Engine-speed limiting prevents the maximum approved engine speed (redline) from being exceeded.
M-Motronic provides the option of restricting engine and vehicle speed by phasing out the injection.

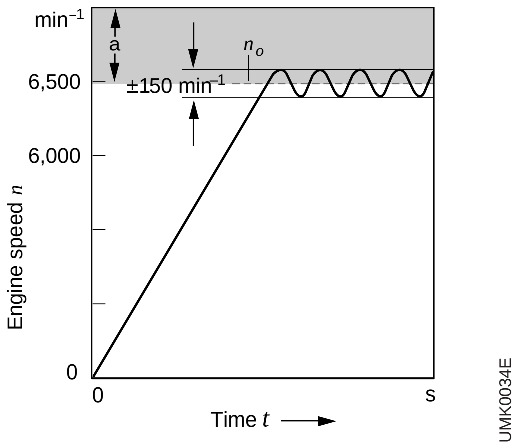
When the maximum engine speed n_0 (or the maximum vehicle speed) is exceeded, the unit responds by suppressing the injection signals. This limits the speed of the engine (or vehicle).

The injection resumes normal operation once the speed falls below a narrow threshold.
This process is repeated at rapid intervals within an engine-speed tolerance range located around the prescribed maximum.

A reduction in operating response and smoothness calls the driver's attention to the engine speed and provides the motivation for an appropriate response.
Figure 12 illustrates the engine-speed curve's reaction to the engine-speed limitation.

Fig. 12

Limiting maximum engine speed n_0 by suppressing injection pulses
a Fuel-cutoff range.



Exhaust-gas recirculation (EGR)

During valve overlap a certain amount of residual gas is returned from the combustion chamber to the intake manifold. This recirculated gas is then entrained along with the fresh air on the next intake stroke.

The actual amount of recirculated gas is determined by the valve overlap, and is thus a fixed variable relative to the various points on the operating curve.

The proportion of recirculated gas can be adjusted using either “external” exhaust-gas recirculation (EGR) with a M-Motronic-controlled exhaust-gas recirculation valve (Fig. 13), or with variable camshaft timing.

Up to a certain point, at least, larger amounts of recirculated gas can exercise a positive effect on energy conversion and thus on fuel consumption. An increase in recirculated gas also leads to lower combustion-chamber temperatures with corresponding reductions in the NO_x emission.

At the same time, once a certain point is passed a higher residual-gas component will lead to incomplete combustion. The results here are higher emissions of unburned hydrocarbons, increased fuel consumption and rough idling (Figure 14).

Effect of residual exhaust-gas recirculation on fuel consumption and emissions

1 Excess-air factor λ (residual exhaust-gas component RG = constant),
2 Recirculated gas component RG (λ = constant).

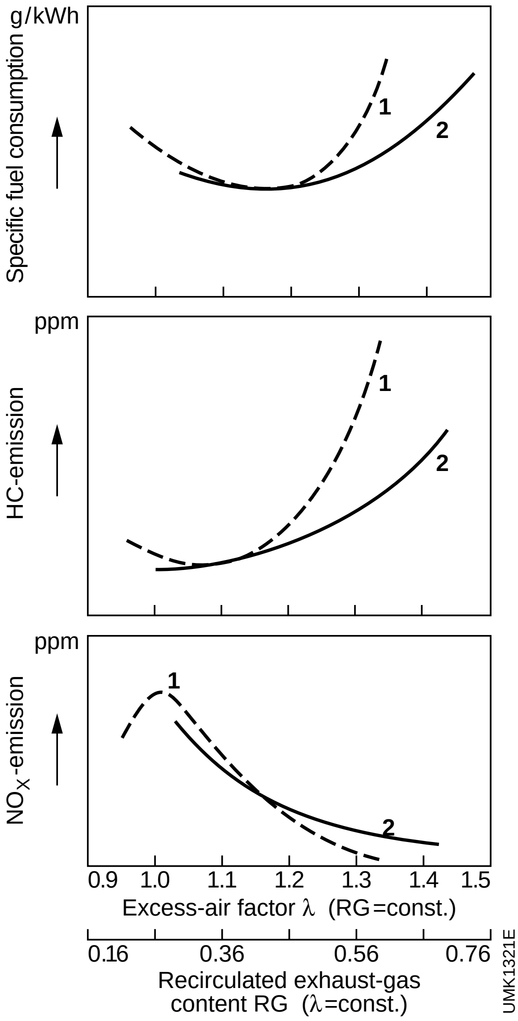
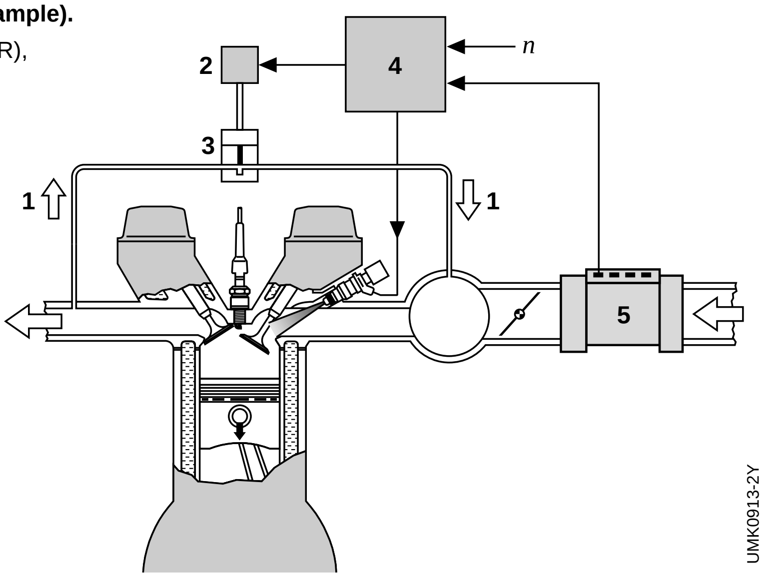


Fig. 14

Fig. 13

Exhaust-gas recirculation (example).

1 Exhaust-gas recirculation (EGR),
2 Electropneumatic converter,
3 EGR valve,
4 ECU,
5 Air-mass meter.
n Engine speed.



Camshaft timing

Camshaft timing can influence the spark-ignition engine in a variety of ways:

- Higher torque and output, lower emissions and fuel consumption,
- Control of the mixture composition, and,
- Graduated or infinitely-variable intake and exhaust adjustment.

“Intake closes” timing plays a decisive role in determining the amount of cylinder charge for a given engine speed. When the intake valve closes early, maximum air will be inducted at low rpm, while longer intake durations shift the maximum toward higher engine speed ranges.

The phase in which the valves overlap (at “intake opens” and “exhaust closes”) determines the amount of internal residual-gas recirculation.

Longer “valve open” durations due to advanced intake-valve timing will raise the proportion of recirculated gas, as they increase the mass of the gas returned to the intake manifold for reinduction. This reduces the mass of fresh air being drawn in at a given throttle-valve opening; at any given load point the throttle valve must open further to compensate. This “dethrottling” effect reduces the gas-exchange circuit to improve efficiency and lower fuel consumption.

The proportion of recirculated gas sinks when the intake cycle is shifted toward “retard.” This provides improvements in fuel economy and operating smoothness.

Variable camshaft angle

Hydraulic or electric actuators turn the camshaft by increments corresponding to specified engine speeds or operating points (this system requires that at least one intake and one exhaust camshaft be located in the cylinder head). This varies the timing for “intake/exhaust opens” and/or “intake/exhaust closes” (Figure 15).

As an example, if the actuators turn the intake camshaft to delay “intake opens/closes” at idle or at high rpm, the result will be a reduction in the proportion of recirculated gases at idle, or enhanced cylinder charging at higher engine speeds.

When the intake camshaft is turned toward earlier “intake opens/closes” at low or moderate engine speeds or in certain load ranges, the result is a higher maximum air charge to the cylinder.

This would also lead to a larger proportion of recirculated gas in the part-throttle range, with corresponding effects on fuel consumption and exhaust emissions.

Fig. 15

Intake camshaft rotation

1 Retarded, **2** Normal, **3** Advanced.

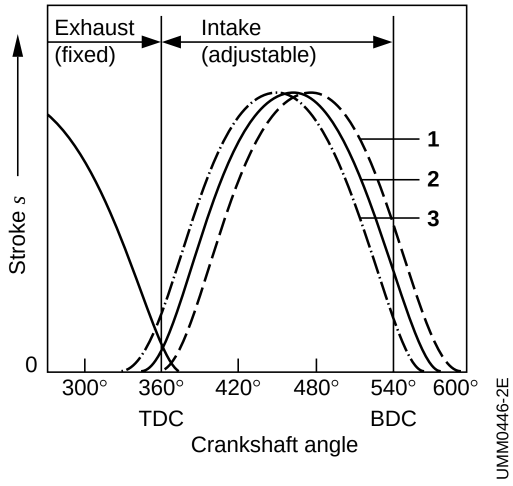
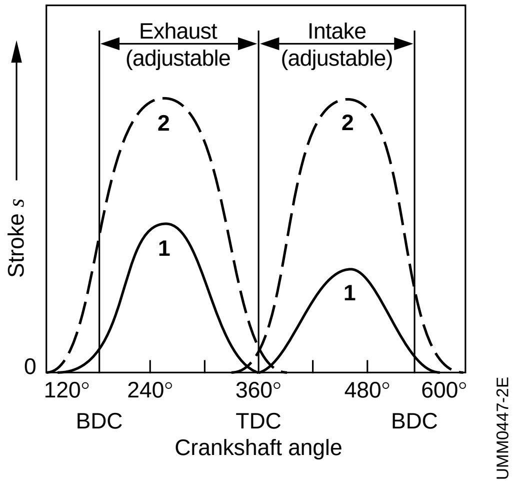


Fig. 16

Variable camshaft timing

1 Standard, **2** Auxiliary lobes.



Camshaft lobe control

Systems with camshaft lobe control modify the valve timing by alternately activating cam lobes with two different shapes. The first lobe supplies optimum valve timing and lift for intake and exhaust valves during lower and mid-range operation. A second cam lobe provides longer valve-opening times and lift, and becomes operational when the rocker arm to which it is connected is locked onto the standard rocker arm in response to engine speed (Figure 16).

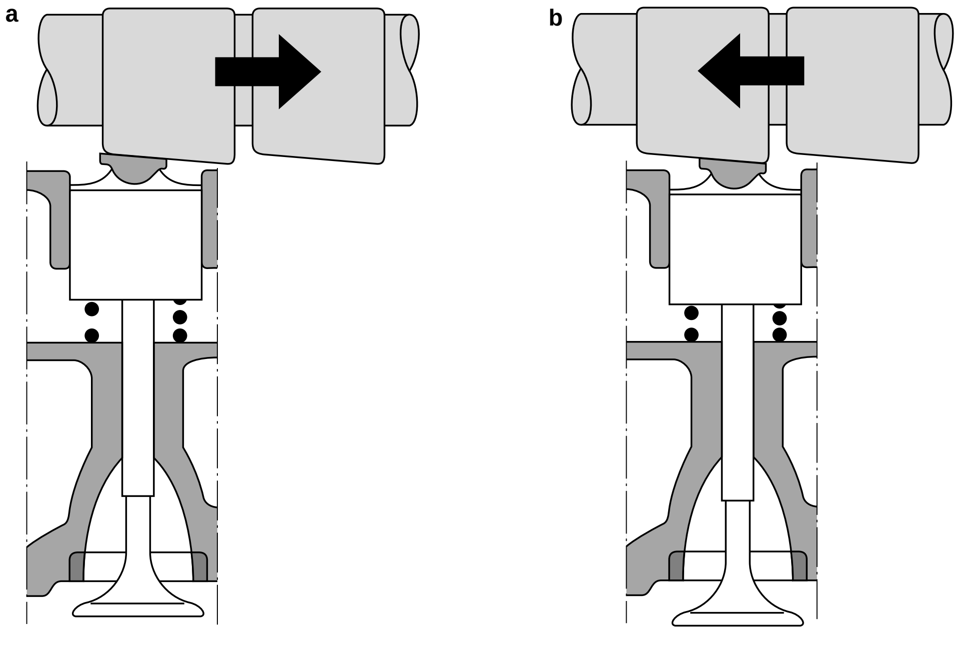
An optimal but complicated process is infinitely-variable valve timing and lift adjustment. This concept employs cam lobes with 3-dimensional geometry and sliding camshafts to provide maximum latitude in engine design (Figure 17).

Operating conditions

Fig. 17

Infinitely-variable valve timing and lift adjustment

a minimum,
b maximum lift.



Variable-geometry intake manifold

The tandem objectives of engine design are maximum torque at low engine speeds and high output at the rated maximum. The engine's torque curve is proportional to the mass of the intake air as a function of engine speed.

One effective means of influencing torque is to provide the intake manifold with the appropriate geometrical configuration. The simplest method for providing intake boost is to exploit the dynamics of the incoming air. To ensure balanced distribution of the air-fuel mixture, intake manifolds for carburetor or single-point (Mono-Jetronic) injection systems need short intake runners with minimal variations in lengths.

The intake runners for multipoint systems transport only air; the injectors discharge the fuel. This arrangement offers a wider range of options in intake-manifold design. The standard manifold for a multipoint injection system consists of individual curved runners and a plenum chamber with throttle valve.

General principles:

- Short curved runners allow high maximum output accompanied by sacrifices in torque at lower engine speeds; whereby long runners provide an inverse response pattern.
- Large-volume plenum chambers can provide resonance effects in certain engine-speed ranges, leading to improved cylinder filling. They are also subject to potential faults in dynamic response; these assume the form of mixture variations under rapid load change.

Variable intake-manifold geometry can be used to obtain an almost ideal torque curve. Geometry can be changed for instance as a function of engine load, engine speed, and throttle-valve setting. Various changes are possible:

Fig. 18

Resonance "supercharging"

a Configuration, **b** Curve of air input.
1 Resonator runner, **2** Resonance chamber,
3 Cylinders, **4** With resonance boost,
5 Normal intake manifold.

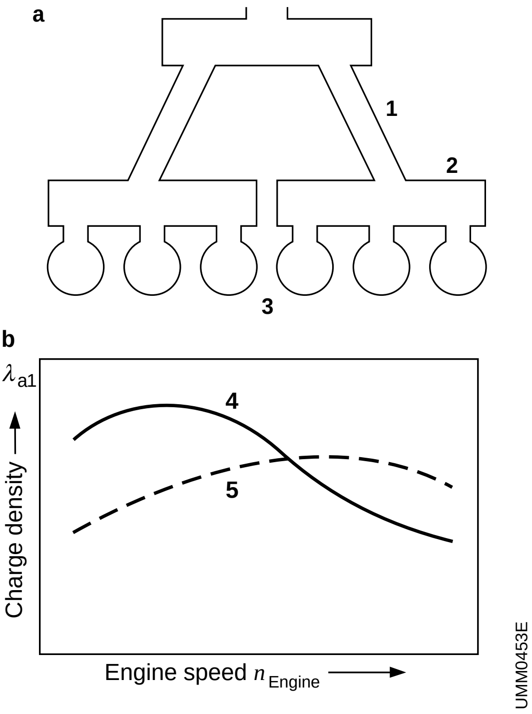
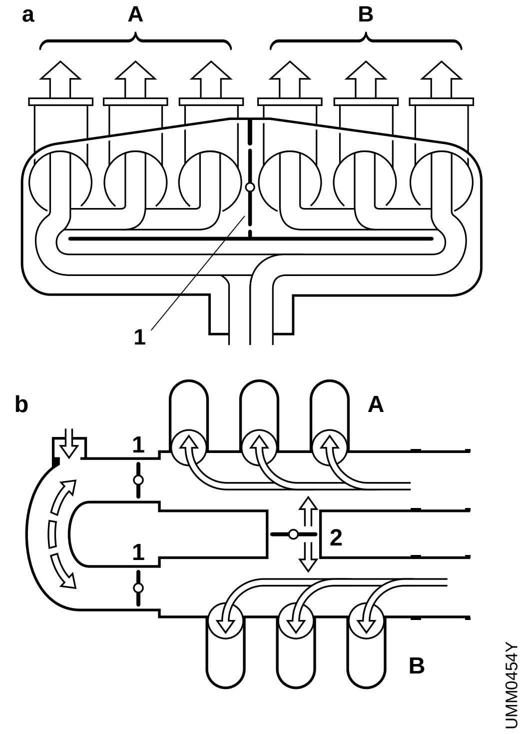


Fig. 19

Variable-geometry intake systems

Switchable: **a** two-stage, **b** three-stage.
A, B Cylinder groups; **1, 2** Flaps, engine speed determines opening time.



- Runner-length adjustment,
- Change-over between runner lengths or runner diameters,
- In case of multiple runners, the selective cutoff of an individual runner for each cylinder,
- Changeover between different plenum-chamber volumes.

Intake oscillation boost

Each cylinder has an individual, fixed-length intake runner, usually connected to a plenum chamber. The energy balance is defined by a process in which the induction force from the piston is converted into kinetic energy in the gas column upstream from the intake valve. This kinetic energy then serves to compress the fresh charge.

Resonance boost

Resonance boost systems use short runners to connect groups of cylinders with equal ignition intervals to resonance chambers. These, in turn, are connected via resonance tubes to the atmosphere or a plenum chamber, allowing them to act as Helmholtz resonators (Figure 18).

Variable-geometry intake systems

Both types of dynamic supercharging augment the achievable cylinder charge, especially in the lower engine-speed range.

Variable-response intake systems use devices such as flaps to separate and connect system areas assigned to various groups of cylinders (Figure 19).
Variable-length intake runners operate with the first resonance chamber at low rpm. The length of the runner changes as engine speed increases, at which point a second resonance chamber opens (Figure 20).
Figure 21 shows the effects of variable intake-runner geometry on the brake mean effective pressure (bmep), which is used as an index for cylinder charging.

Fig. 20

Infinitely-variable length intake system

1 Fixed housing,
2 Rotating drum (air distributor),
3 Drum air-supply opening,
4 Air-supply opening for intake runners,
5 Seal (e.g., leaf spring),
6 Intake runners,
7 Intake valve,
8 Intake air stream.

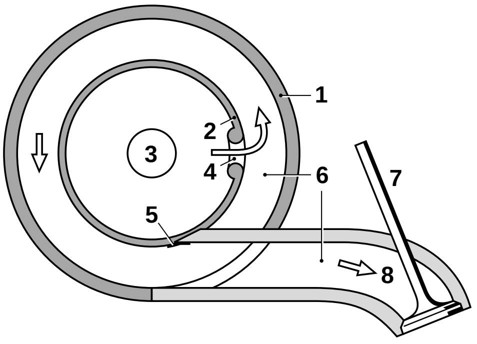


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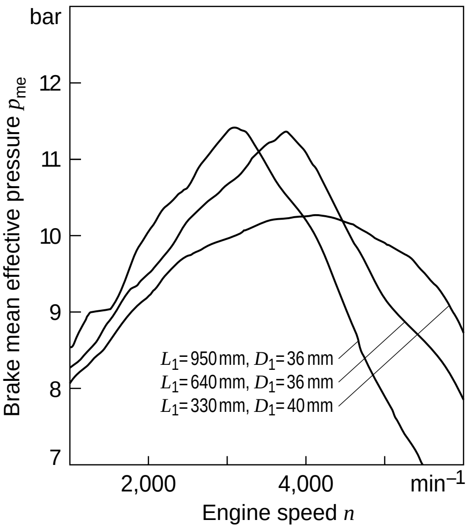
Fig. 21

Brake mean effective pressure relative to engine speed at three lengths of the variable-geometry intake system

L_1 Effective intake-runner length,
D_1 Intake-runner diameter.



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Integrated diagnosis

Diagnostic procedure

An “on-board diagnosis (OBD) system” is standard equipment with M-Motronic. This integral diagnostic unit monitors ECU commands and system responses. It also checks the individual sensor signals for plausibility. This test procedure is carried out constantly during normal vehicle operation.

The ECU stores recognized errors together with the operating conditions under which they occurred. When the vehicle is serviced, a tester can be used to read out and display the stored errors through a standardized interface. The information facilitates trouble-shooting for the service personnel.

In response to the demands of the California Air Resources Board, (CARB), diagnosis procedures extending far beyond those in earlier tests have been developed. All components whose failure could cause a substantial increase in harmful emissions must be monitored.

Diagnosis areas

Air-mass meter

The process for monitoring the air-mass meter provides an example of the M-Motronic system’s self-diagnosis function. While the injection duration is being determined on the basis of intake-air mass, a supplementary comparison injection time is calculated from throttle-valve angle and engine speed. If the ECU discovers an excessive variance between the two, its initial response is to store a record of the error. As vehicle operation continues, plausibility checks determine which of the sensors is defective. The control unit does not store the corre-sponding error code until it has unequivocally determined which sensor is at fault.

Combustion miss

Combustion miss, as can result from factors such as worn spark plugs or faulty electrical contacts, allows unburned mixture to enter the catalytic converter. This mixture can destroy the converter, and represents an extra load on the environment. Because even isolated combustion failures result in higher emissions, the system must be able to recognize them. Figure 1 shows the effects of combustion miss on emissions of hydrocarbons (HC), carbon monoxide (CO) and oxides of nitrogen (NO_X).

Many potential methods of detecting combustion miss were tested, and monitoring the running response of the crankshaft proved to be the most suitable. Combustion miss is accompanied by a shortfall in torque equal to the increment which would normally have been generated in the cycle where the error occured. The result is a reduction in rotation speed. At high speeds and low loads the interval from ignition to ignition (period duration) is extended by only 0.2 %. This means that the rotation must be monitored with extreme precision, while extensive computations are also required to distinguish combustion miss from other interference factors.

Catalytic converter

Yet another diagnostic function monitors the efficiency of the catalytic converter. For this purpose the Lambda oxygen sensor upstream of the catalytic converter is supplemented by a second downstream oxygen sensor. A correctly-operating converter will store oxygen, thus attenuating the Lambda control oscillations. As the catalytic converter ages, this response deteriorates until finally the signal pattern from the upstream sensor approaches that received from the downstream sensor. A comparison of the signals from the oxygen sensors thus provides the basis for determining the catalytic converter’s condition. A warning lamp alerts the driver in the event of a defect.

Exhaust emissions relative to combustion-miss rate

Engine: 6-cylinder, 2.8 litre
US 94 emissions limits
HC = 0.25 g/mile
CO = 3.40 g/mile
NO_x = 0.40 g/mile

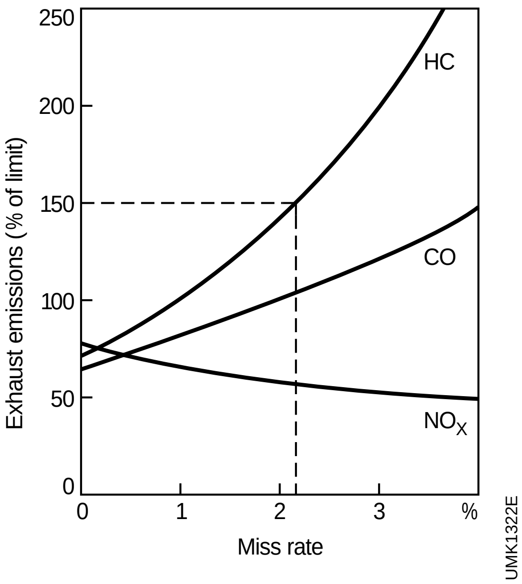
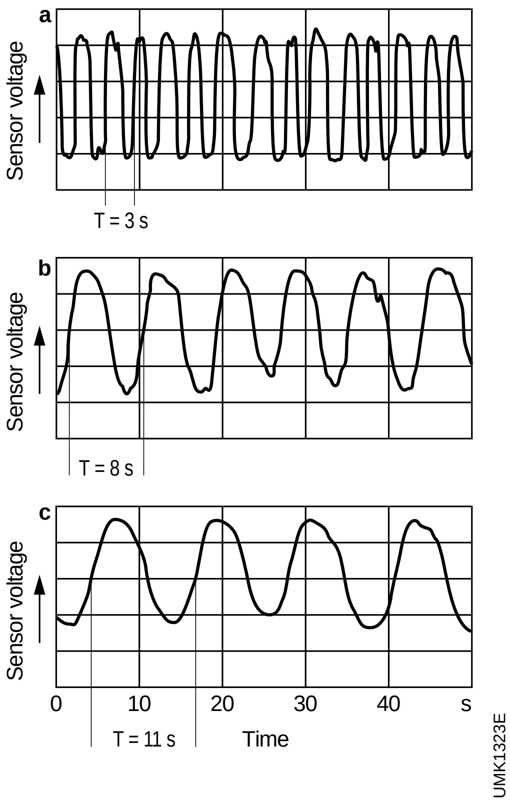


Fig. 1

Fig. 2

Monitoring the Lambda oxygen sensor's dynamic response pattern

a New sensor, **b** Aged Type II sensor,
c Aged Type III sensor.



Lambda oxygen sensor

A stoichiometric air-fuel mixture must be maintained if the catalytic converter is to perform to its full potential. This is taken care of by the signals from the Lambda oxygen sensors. The fact that two oxygen sensors are fitted in each exhaust tract makes it possible to use the sensor downstream from the converter to check for control variations in the upstream unit.

A Lambda oxygen sensor that has been exposed to excessive heat for a considerable period of time may react more slowly to changes in the air-fuel mixture. This increases the period duration for the Lambda control's two-state controller (Figure 2). A diagnosis function monitors this control frequency and informs the driver of excessive delays in sensor response via warning lamp.

The Lambda oxygen sensors' heating resistance is checked by measuring current and voltage. The M-Motronic ECU controls the heater resistance element directly, with no relay in between, to allow this test to be performed. The sensor signal is subjected continuously to plausibility checks, and the system responds to implausible signals by denying access to other functions depending upon the Lambda control. The appropriate error code is also stored in the fault memory.

Fuel supply

When the air-fuel mixture deviates from stoichiometric for extended periods of time, this condition is taken into account together with the mixture adaptation. If the deviations exceed specific predefined limits, this indicates that a fuel-system component or a fuel-metering device has moved outside specification tolerances. An example would be a faulty pressure regulator or load sensor, whereby the error could also stem from a leak in the intake manifold or exhaust system.

Secondary-air injection

The secondary-air injection activated after cold starts must also be monitored, as its failure would also influence emissions. This can be done using the signals from the Lambda oxygen sensors when the secondary-air injection is active, or it can be activated and observed at idle using a Lambda control test function.

Exhaust-gas recirculation (EGR)

Various options are available for diagnosing the exhaust-gas recirculation system, whereby two are in general use. With the first option, a sensor monitors the temperature increase at the location where the hot exhaust gases return to the intake manifold while the EGR is operating.

With the second procedure, the exhaust-gas recirculation valve (EGR valve) is opened all the way with the vehicle on trailing throttle (overrun fuel cutoff). The exhaust gases flowing into the intake manifold cause its internal pressure to rise. A pressure sensor measures and evaluates the increase in manifold pressure.

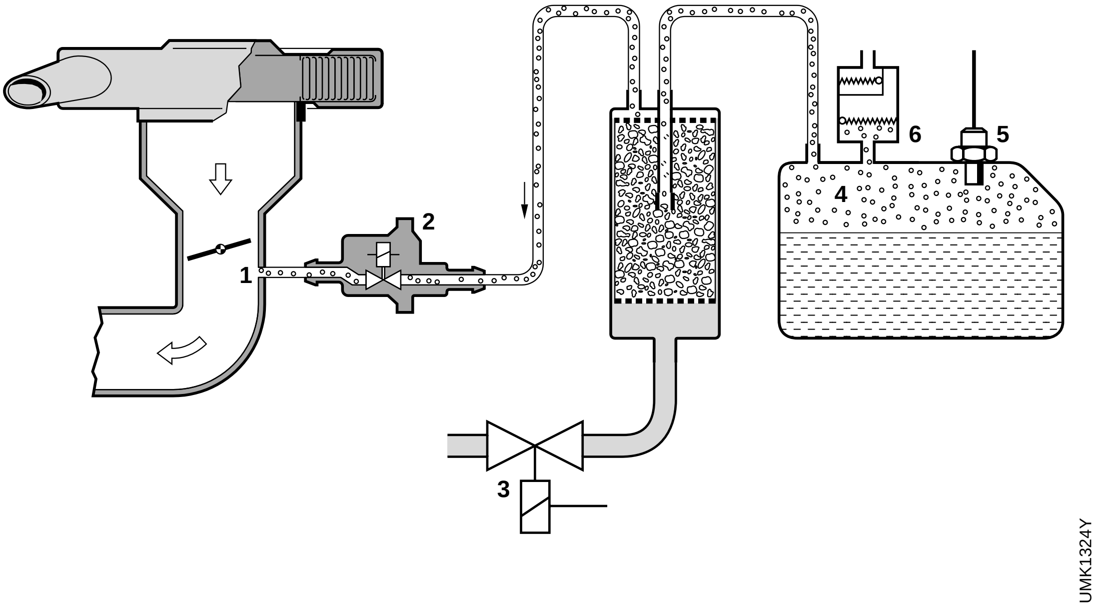
Tank system

Emissions emanating from the exhaust system are not the only source of environmental concern; fuel vapors from the fuel tank are also a problem. In the near term the legal requirements will be limited to a relatively simple check of canister-purge valve operation. A means of recognizing leaks in the evaporative-emissions storage system will be required at a later date. Figure 3 illustrates the basic principle employed for this diagnosis. A cutoff valve closes off the storage system.
Then, preferably with the engine idling, the canister-purge valve is opened and the intake-manifold pressure spreads through the system. An in-tank pressure sensor monitors the pressure build-up to determine whether leaks are present.

Fig. 3

Vacuum test to detect leaks in evaporative-emissions control system

1 Intake manifold,
2 Canister-purge valve,
3 Shutoff valve,
4 Fuel tank,
5 Pressure differential sensor,
6 Safety valve.



Other monitoring devices
The main emphasis of the new statutes applies to the engine-management system, but other systems (for instance, automatic transmission) are also monitored. These report any faults to the engine-management ECU, which then assumes responsibility for triggering the diagnosis lamp. Greater system complexity and ever more stringent environmental regulations are making diagnosis increasingly important.

Emergency running mode (limp-home)

In the interval between the initial occurrence of a fault and vehicle service, default settings and emergency-running functions assume responsibility for the ignition and air-fuel mixture. This allows the vehicle to continue operating, albeit with sacrifices in driving comfort. The ECU responds to errors that have been recognized in an input circuit by replacing the missing information or reverting to a default value.
When an output unit fails, specific backup measures corresponding to the individual problem are implemented. Thus the ECU reacts to a defect in the ignition circuit by switching off the fuel injection at the affected cylinder in order to prevent damage to the catalytic converter.

When the vehicle is serviced, the Bosch engine tester can be used to read out and display the faults and errors detected during operation (Figure 4).

Fig. 4
Bosch engine tester

ECU

Function

The ECU is the “computer and control center” for the engine-management system. It employs stored functions and algorithms (processing programs) to process the input signals transmitted by the sensors. These signals serve as the basis for calculating the control signals to the actuators (e.g., ignition coil, injectors) which it manages directly via power output stages (Figure 1).

Physical design

The ECU is a metal housing containing a printed-circuit board with electronic componentry.
A multiple-terminal plug connector provides the link between ECU and sensors, actuators and power supply. Depending upon the specific ECU and the number of system functions, the plug can be of 35-, 55- or 88-pole design.

The amplifiers and power output components for direct actuator control are installed on heat sinks in the ECU. Efficient heat transfer to the bodywork is necessary due to the the amount of heat that these components produce.

Environmental conditions

The ECU must withstand temperature extremes, moisture and mechanical loads with absolutely no impairment of operation. Resistance to electromagnetic interference, and the ability to suppress radiation of high-frequency static, must also be of a high order.

The ECU must be capable of errorless signal processing within an operating range extending from –30 °C to +60 °C, at battery voltages that range from 6 V (during starting) to 15 V.

Power supply

A voltage regulator provides the ECU with the constant 5 V operating voltage needed for the digital circuitry.

Signal inputs

Various processes are employed to transmit the input signals to the ECU. The signals are conducted through protective circuits, while signal converters and amplifiers may also be present. The microprocessor can process these signals directly.

An analog/digital converter (A/D) within the microprocessor transforms analog signals (for instance, information on intake-air quantity, temperature of engine and intake air, battery voltage, Lambda oxygen sensor) into digital form.
The signal from the inductive sensor with information on engine speed and crankshaft reference point is processed in a special circuit to suppress interference pulses.

Signal processing

The input signals are processed by the microprocessor within the ECU. In order to function, this microprocessor must be equipped with a signal-processing program stored in a non-volatile memory (ROM or EPROM). This memory also contains the specific individual performance curves and program maps used for engine control.
Due to the large number of engine and vehicle variations, some ECU’s are equipped with a special version-code feature. This allows the manufacturer or service technician to feed supplementary program data into the program maps stored in the EPROM, making it possible to provide the operating characteristics desired for the particular version. Other types of ECU are designed to allow complete data banks to be programmed into the EPROM at the end of production (end-of-line programming). This reduces the number of individual ECU configurations required by the manufacturer.
A read/write memory component (RAM) is needed for storing calculated values and adaptation factors as well as any system errors that may be detected (diagnosis). This RAM requires an uninterrupted power supply to function properly. This memory chip will lose all

data if the vehicle battery is disconnected. The ECU must then recalculate the adaptation factors after the battery is reconnected. To prevent this, some units therefore use an EEPROM instead of a RAM to store these required variables.

Transmitting the signal

The output stages triggered by the microprocessor supply sufficient power for direct control of the actuators. These output stages are protected against short circuits to ground, irregularities in battery voltage and electrical overload that could destroy them.

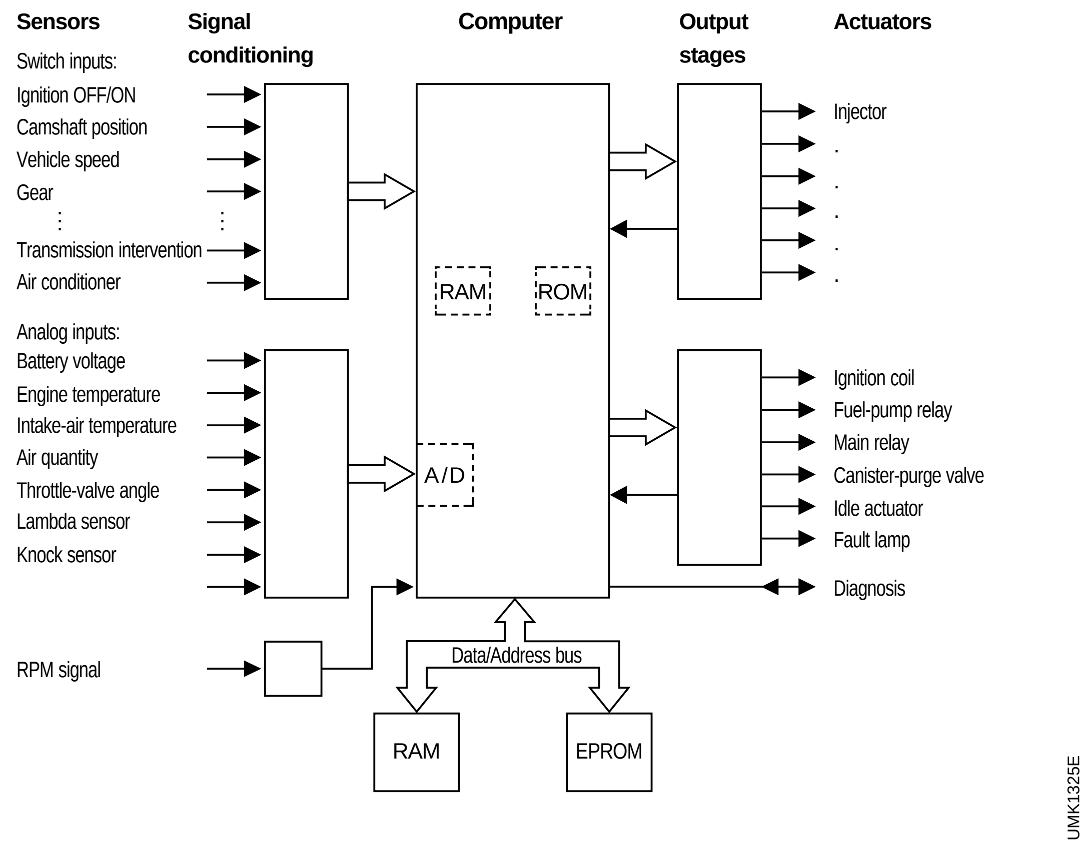
At several output stages, the OBD diagnosis function recognizes errors and reacts by deactivating (where necessary) the defective output. The error entry is stored in the RAM. The service technician can then read out the error using a tester connected to the serial interface.

Another protective circuit operates independently of the ECU to switch off the electric fuel pump when the engine-speed signal falls below a certain level.

When some ECU's are switched off at the ignition/steering lock (Terminal 15, or "ignition off"), a holding circuit holds the main relay open until program processing can be completed.

Fig. 1

Motronic block diagram



Interfaces to other systems

System overview

Increased application of electronic control systems in vehicles in areas such as
- Transmission control,
- Electronic throttle control (EMS, E-Gas, drive-by-wire),
- Electronic engine management (Motronic),
- Antilock braking system (ABS),
- Traction control system (TCS),
- On-board computer, etc.,

has made it necessary to combine the respective ECU’s in networks. Data communications between control systems reduce the number of sensors and allow better exploitation of the individual system potentials.

The interfaces can be divided into two categories:
- Conventional interfaces, with binary signals (switch inputs), pulse-duty factors (pulse-width-modulated signals),
- Serial data transmission, e.g., Controller Area Network (CAN).

Conventional interfaces

In conventional automotive data-communications systems, each signal is assigned to a single line. Binary signals can only be transmitted as one of the two conditions “1” or “0” (binary code), for instance, air-conditioning compressor “ON” or “OFF”.

Pulse-duty factors (potentiometer) can be employed to relay more detailed data, such as throttle-valve aperture.

The increasing levels of data exchange between the various electronic components in the vehicle means that conventional interfaces are no longer capable of providing satisfactory performance. The complexity of today’s wiring harnesses is already difficult to manage, while the requirements for data communications between ECU’s are on the rise (Figure 1). These problems can be solved by using CAN (Controller Area Network), a bus system (bus bar) specially designed for automotive use.

Provided that the ECU’s are equipped with a serial CAN interface, CAN can be used to relay the signals from the sources listed above.

Serial data transmission (CAN)

There are three basic applications for CAN in motor vehicles:
- To connect ECU’s,
- Bodywork and convenience electronics (Multiplex),
- Mobile communications.

The following is limited to a description of communications between ECU’s.

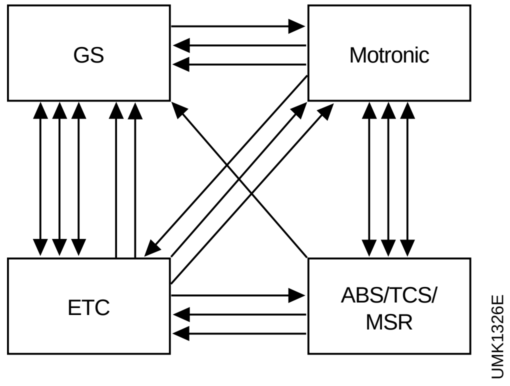
ECU networking

Here electronic systems such as M-Motronic, electronic transmission control, etc. are combined within a single network. Typical transmission times lie between approx. 125 kBit/s and 1MBit/s. These times must be high enough to maintain the required real-time response. One of the advantages that serial data transmission enjoys over conventional interfaces (e.g., pulse-duty factor, switching and analog signals) is the high speeds achieved without high loads on the central processing units (CPU’s) in ECU’s.

Fig. 1

Conventional data communications

GS Transmission control, ETC Electronic throttle control, ABS Antilock braking system,
TCS Traction control system,
MSR Engine-drag torque control.



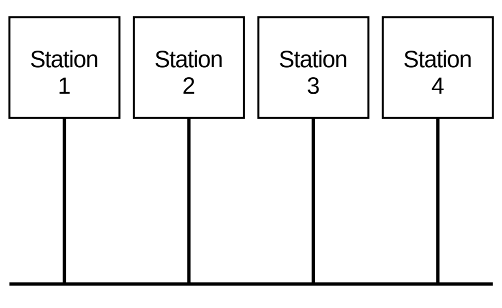
Bus configuration
CAN works on the “multiple master” principle. This concept combines several equal-priority ECU’s in a linear bus structure (Figure 2). The advantage of this structure is the fact that failure of one subscriber will not affect access for the others. The probability of total failure is thus much lower than with other logical arrangements (such as loop or star structures). With loop or star architecture, failure in one of the subscribers or the central ECU will result in total system failure.

Content-keyed addressing
The CAN bus system addresses the data according to content. Each message is assigned a permanent 11-bit identifier. This identifier indicates the contents of the message (e.g., engine speed). Each station processes only those data whose identifiers are stored in its acceptance list (acceptance check). This means that CAN does not need station addresses to transmit data, and the junctions do not need to administer system configuration.

Bus arbitration
Each station can begin transmitting its highest priority message as soon as the bus is free. If several stations start transmitting simultaneously, the resulting bus-access conflict is resolved using a “wired-and” arbitration arrangement. This arrangement assigns first access to the message with the highest priority, with no loss of either time or data bits. When a station loses the arbitration, it automatically reverts to standby status and repeats its transmission attempt as soon as the bus indicates that it is no longer occupied.

Fig. 2

Linear bus structure



Message format
A data frame of less than 130 bits in length is created for transmissions to the bus. This ensures that the queue time until the next – possibly extremely urgent – data transmission is held to a minimum. The data frames consist of seven consecutive fields.

Standardization
The International Standards Organisation (ISO) has recognized CAN as a standard for use in automotive applications with data streams of over 125 kBit/s, along with two additional protocols for data rates up to 125 kBit/s.

The Program

	Order Number
Gasoline-engine management	
Emission Control (for Gasoline Engines)	1 987 722 102
Gasoline Fuel-Injection System K-Jetronic	1 987 722 159
Gasoline Fuel-Injection System KE-Jetronic	1 987 722 101
Gasoline Fuel-Injection System L-Jetronic	1 987 722 160
Gasoline Fuel-Injection System Mono-Jetronic	1 987 722 105
Ignition	1 987 722 154
Spark Plugs	1 987 722 155
M-Motronic Engine Management	1 987 722 161
ME-Motronic Engine Management	1 987 722 178
Diesel-engine management	
Diesel Fuel-Injection: An Overview	1 987 722 104
Diesel Accumulator Fuel-Injection System Common Rail CR	1 987 722 175
Diesel Fuel-Injection Systems Unit Injector System / Unit Pump System	1 987 722 179
Radial-Piston Distributor Fuel-Injection Pumps Type VR	1 987 722 174
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Diesel In-Line Fuel-Injection Pumps PE	1 987 722 162
Governors for Diesel In-Line Fuel-Injection Pumps	1 987 722 163
Automotive electrics/Automotive electronics	
Alternators	1 987 722 156
Batteries	1 987 722 153
Starting Systems	1 987 722 170
Electrical Symbols and Circuit Diagrams	1 987 722 169
Lighting Technology	1 987 722 176
Safety, Comfort and Convenience Systems	1 987 722 150
Driving and road-safety systems	
Compressed-Air Systems for Commercial Vehicles (1): Systems and Schematic Diagrams	1 987 722 165
Compressed-Air Systems for Commercial Vehicles (2): Equipment	1 987 722 166
Brake Systems for Passenger Cars	1 987 722 103
ESP Electronic Stability Program	1 987 722 177

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