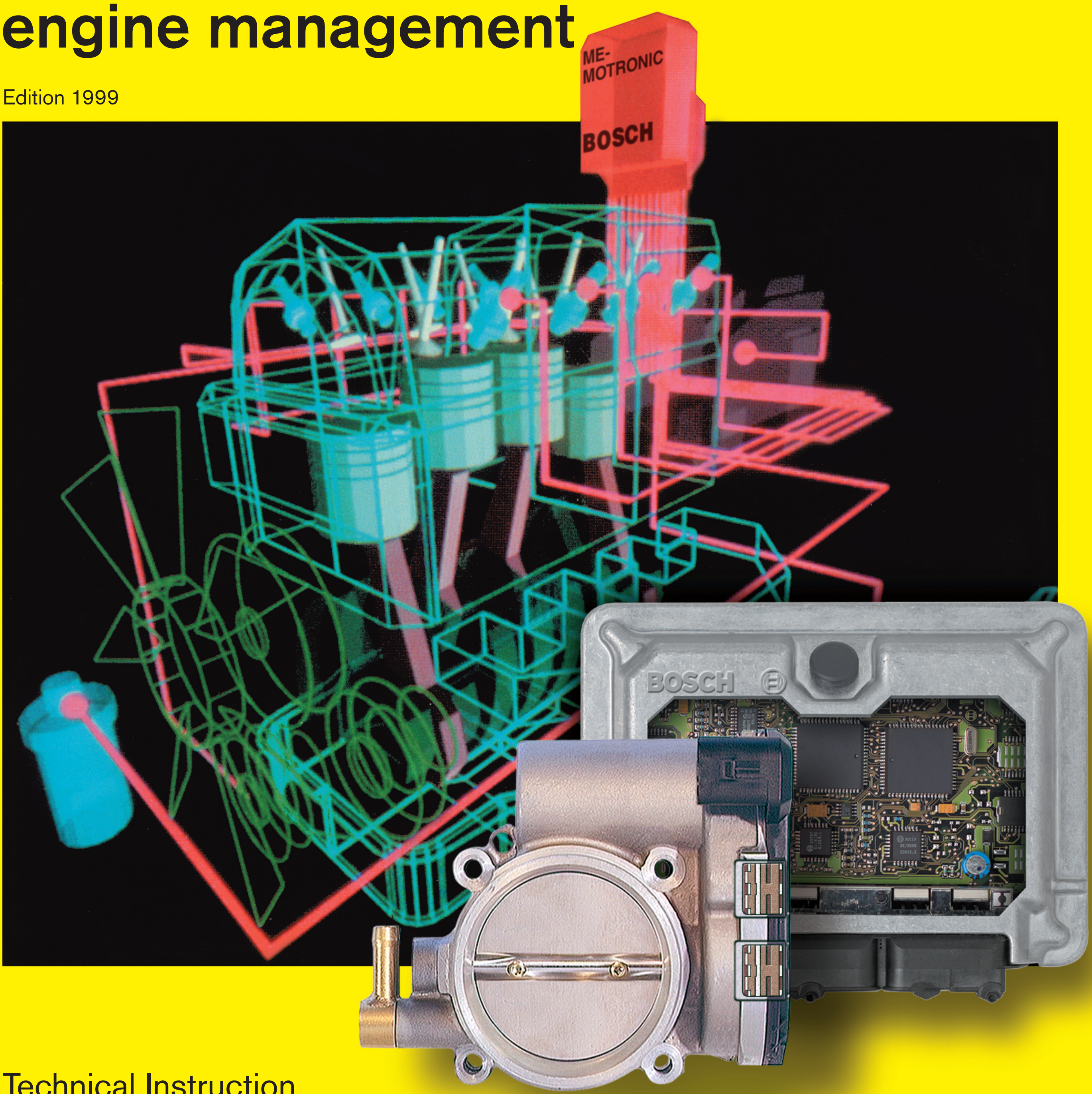
Gasoline-engine management

ME-Motronic engine management

Edition 1999



Technical Instruction

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ME-Motronic engine management

Electronic engine-management systems have advanced to become decisive factors in promoting fuel economy and enhancing the motor vehicle's environmental compatibility.
The engine-management system's primary assignment is to furnish the torque requested by driver demand while at the same time ensuring maximum fuel economy and minimum emissions. The ME-Motronic engine-management system for the gasoline engine (also known as the spark-ignition (SI) or Otto-cycle engine), unites all of the subsystems required to meet this challenge: The electronic throttle control (ETC, or "drive by wire") regulates the flow of induction air to satisfy instantaneous torque demand, while the fuel-injection subsystem regulates fuel mass. Meanwhile, the ignition subsystem governs ignition timing and the generation of spark energy.
ME-Motronic's capabilities extend even further to embrace coordinated action with other automotive systems designed to enhance comfort, convenience and safety for the user. An example is the way ME-Motronic adjusts torque levels to ensure maximum traction in response to demands from the ABS and ESP systems.
Progress in satisfying this highly variegated range of engine-management functions has been marked by ever-closer coordination of the individual subsystems. This brochure explains the design concept behind ME-Motronic engine management as well as how the system operates.
Conventional Motronic systems are described in a publication from this series entitled "M-Motronic engine management".

Combustion in the gasoline engine

The spark-ignition or Otto-cycle engine

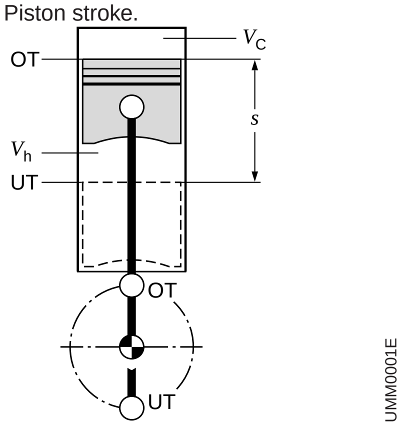
Operating concept

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 Augst 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.

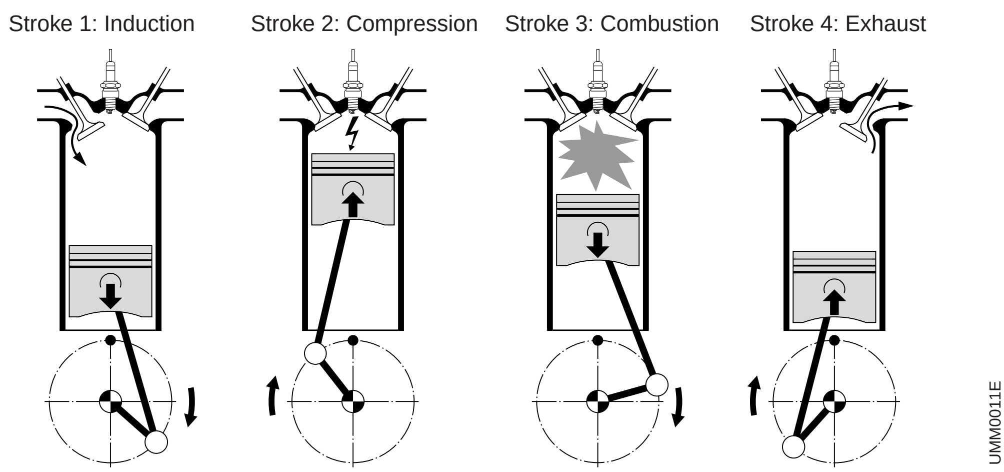
The ignition spark at the spark plug ignites the compressed air/fuel mixture, thus initiating combustion and the attendant temperature rise.
This raises pressure levels within the cylinder to propel the piston downward. The piston, in turn, exerts force against the crankshaft to perform work; this process 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.

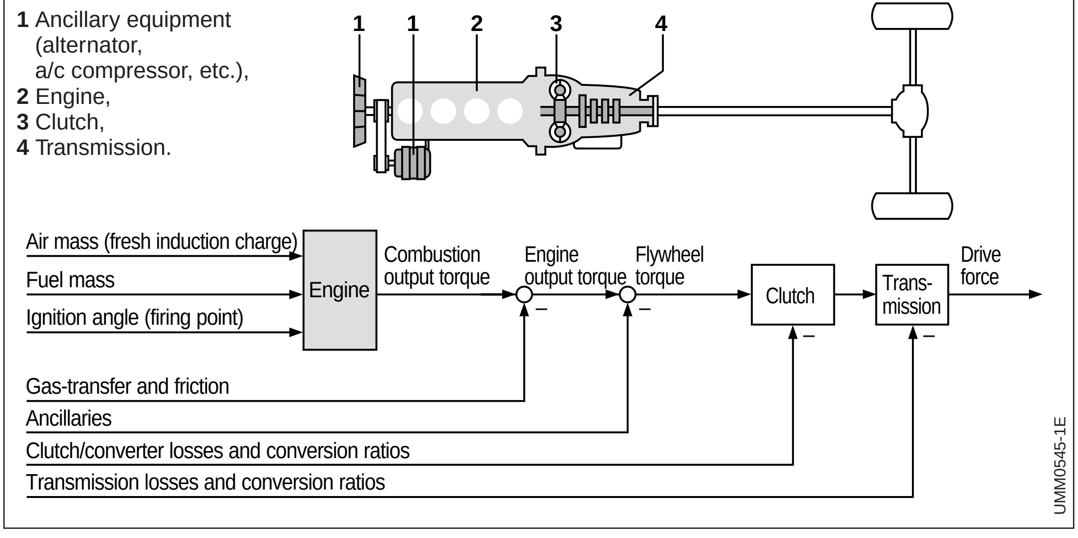
Cylinder-charge control
In Bosch engine-management systems featuring electronic throttle control (ETC), the "cylinder-charge control" subsystem determines the required induction-air mass and adjusts the throttle-valve opening accordingly. The driver exercises direct control over throttle-valve opening 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.

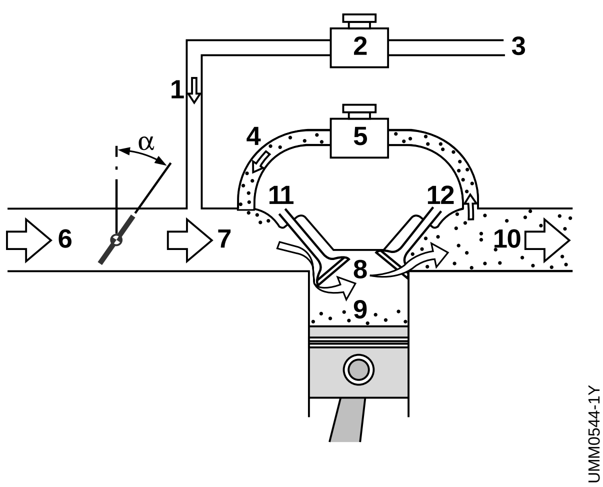
Residual gases
The portion of the charge consisting of residual gases is composed of
– The exhaust-gas mass that is not discharged while the exhaust valve is open and thus remains in the cylinder, and
– The mass of recirculated exhaust gas (on systems with exhaust-gas recirculation, Figure 2).

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

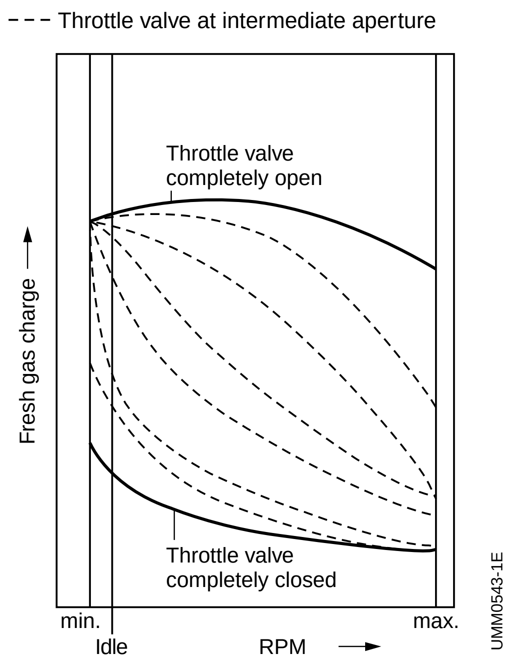
A supercharging (or boost) effect can be obtained by exploiting dynamics within the intake manifold. The actual degree of boost will depend upon the manifold's configuration as well as the engine's instantaneous operating point (essentially a function of the engine's speed, but also affected by load 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.

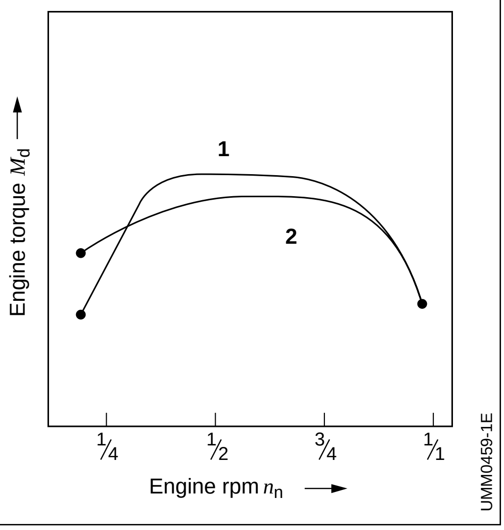
Exhaust-gas turbochargers

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).

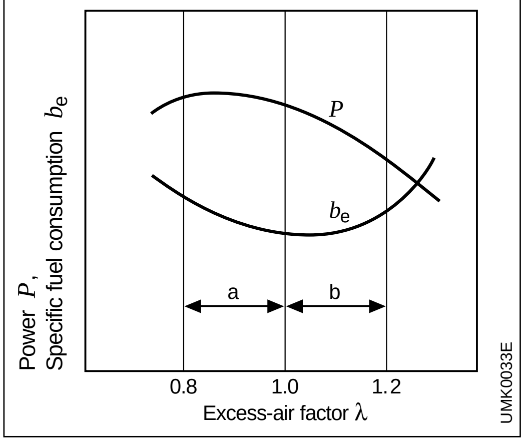
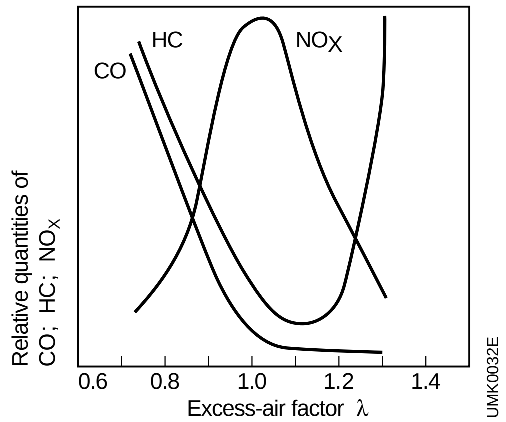


Fig. 2

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 over-heating 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 con-figuration 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).

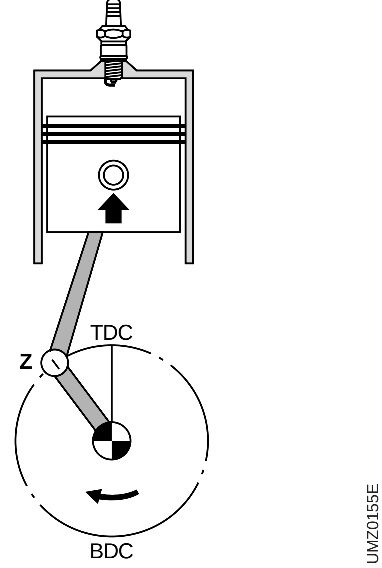
Ignition timing must be selected so that the following criteria are complied with:

– 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.

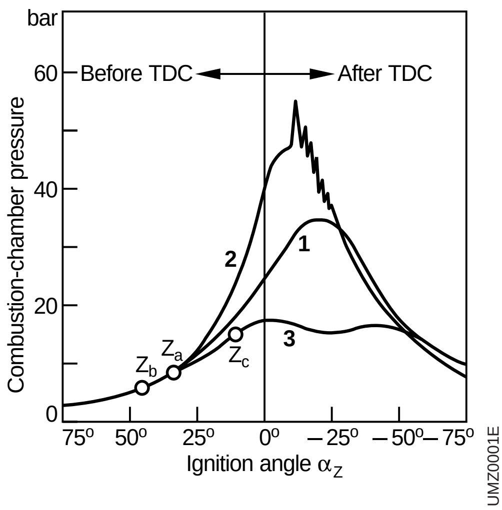
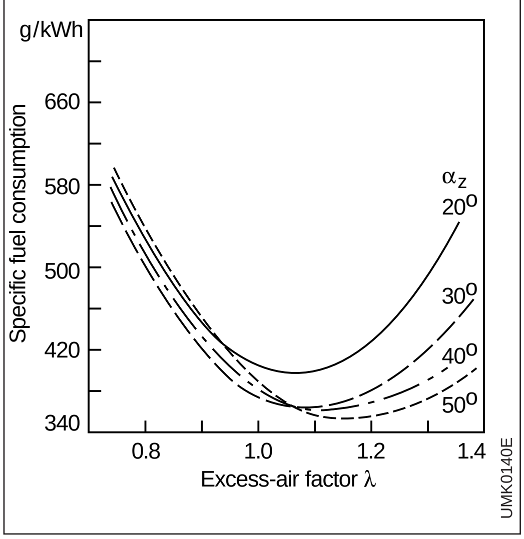


Fig. 3
Effect of excess-air factor λ and ignition timing α_Z on fuel consumption



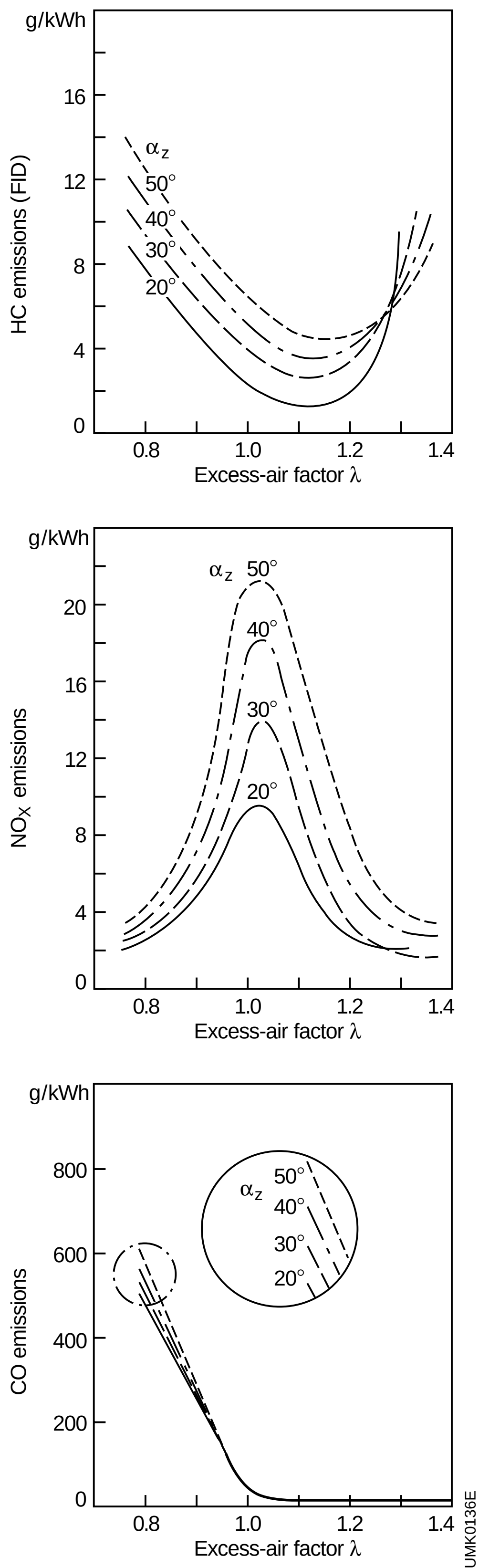
from $\lambda = 1.1...1.2$. Increases in the excess-air factor are accompanied by a corresponding increase in the optimal ignition advance angle, which is defined here as the timing that will minimize specific fuel consumption. The relationship between specific fuel consumption and excess-air factor (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.

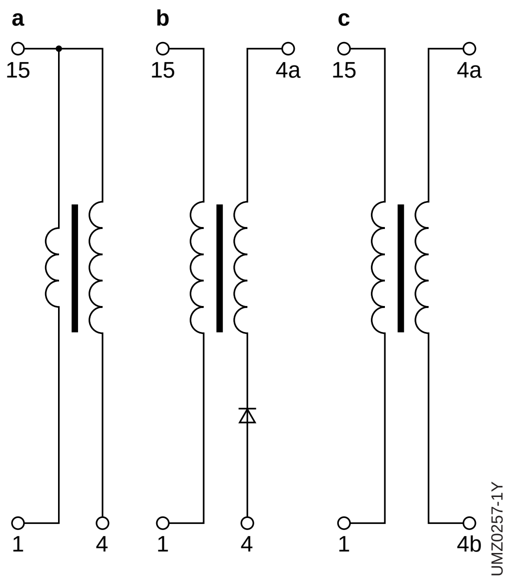
Design and function
Ignition-coil operation is based on an inductive concept. The coil consists of two magnetically coupled copper coils (primary and secondary windings). The energy stored in the primary winding's magnetic field is transmitted to the secondary side. Current and voltage are transformed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

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.

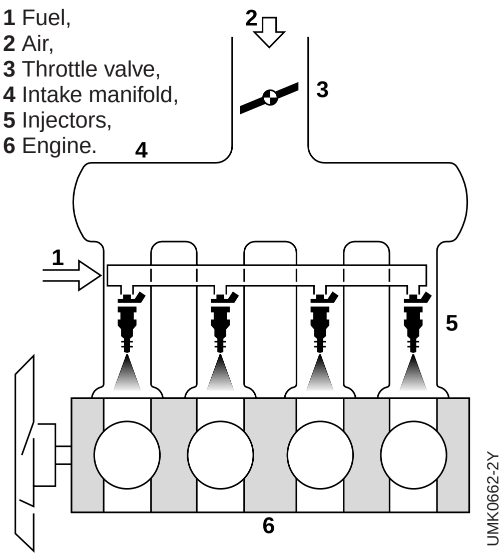
Combined mechanical-electronic fuel injection
Although the K-Jetronic layout served as the mechanical basis for the KE-Jetronic system, the latter employs expanded data-monitoring functions for more precise adaptation of injected fuel quantity to specific engine operating conditions.

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.

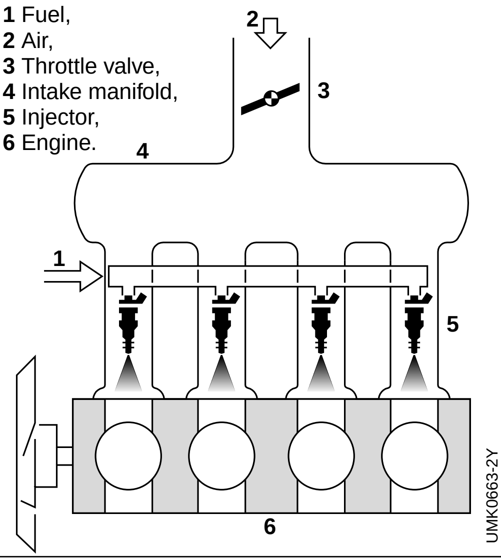
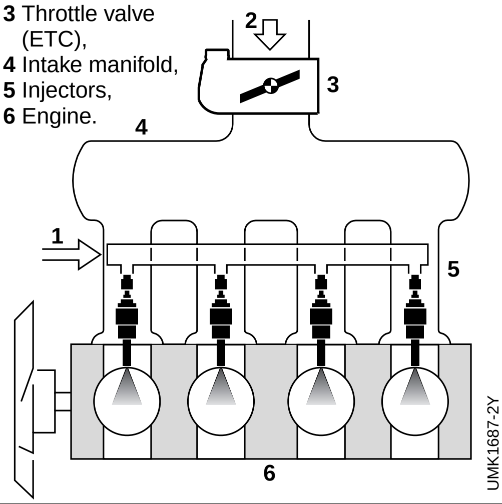


Fig. 3

Direct fuel injection (DI)

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



ME-Motronic engine management

The overall Motronic system

System overview

The Motronic system contains all of the actuators (servo units, final-control elements) required for intervening in the spark-ignition engine management, while monitoring devices (sensors) register current operating data for engine and vehicle. These sensor signals are then processed in the input circuitry of a central electronic control unit (ECU) before being transferred to the ECU microprocessor (function calculator). The information provided (Figs. 1 and 2) includes data on:

– Accelerator-pedal travel,
– Engine speed,
– Cylinder charge factor (air mass),
– Engine and intake-air temperatures,
– Mixture composition, and
– Vehicle speed.

The microprocessor employs these data as the basis for quantifying driver demand, and responds by calculating the engine torque required for compliance with the driver's wishes. Meanwhile, the driver or a transmission-shift control function selects the conversion ratio needed to help define engine speed.

The microprocessor generates the required actuator signals as the first stage in setting the stipulated operating status. These signals are then amplified in the driver circuit and transmitted to the actuators responsible for engine management. By ensuring provision of the required cylinder charge together with the corresponding injected fuel quantity, and the correct ignition timing, the system furnishes optimal mixture formation and combustion.

ME version

The following descriptions focus on a typical version of ME-Motronic. Within the type designation, the letter "M" stands for the classical Motronic function of coordinated control for injection and ignition, while "E" indicates integration of the ETC electronic throttle control.

Basic functions

Motronic's primary function is to implement the engine's operational status in line with the driver's demands. The system's microprocessor responds to this demand by translating the accelerator-pedal travel into a specified engine output. When converting the required engine-output figure to the parameters for actually controlling engine output, that is

– The density of the cylinders' air charges,
– The mass of the injected fuel, and
– The ignition timing,

the system takes into account the extensive range of current operating data as monitored by its sensors:

Auxiliary functions

ME-Motronic complements these basic functions with a wide spectrum of supplementary open and closed-loop control functions, including:

– Idle-speed control,
– Lambda closed-loop air-fuel mixture control,
– Control of the evaporative-emissions control system,
– Exhaust-gas recirculation (EGR) for reductions in NO_X emissions,
– Control of the secondary-air injection to reduce HC emissions, and
– Cruise control.

These secondary functions have been rendered essential by a combination of

factors. While these include legal mandates for reduced exhaust emissions and a continued demand for further enhancements of fuel economy, they also embrace higher expectations now directed toward safety and driving comfort.
The system can also be expanded to incorporate the following supplements:

- Turbocharger and intake-manifold geometry-control functions (→ to enhance power output),
- Camshaft control for engines with variable valve timing (→ to enhance power output while simultaneously reducing both fuel consumption and exhaust emissions), and
- Knock control, and engine-speed control and vehicle-speed control (→ to protect engine and vehicle).

Torque-based control concept

The prime objective behind this torque-control strategy is to correlate this large and highly variegated range of objectives. This is the only way to allow flexible selection of individual functions for integration in the individual Motronic versions according to engine and vehicle type.

Torque coordination

Most of the above auxiliary open and closed-loop control functions exercise a feedback effect on engine torque. This frequently leads to the simultaneous appearance of mutually conflicting demands.
In a torque-based system, all of these functions reflect the driver's behavior in that they demand a specific engine torque. ME-Motronic's flexible-response torque-based control system can prioritize these mutually antagonistic requirements and implement the most important ones. This is the advantage of the torque-based structure. All functions submit individual and independent requests for torque.

Vehicle management

The CAN (Controller Area Network) bus system allows Motronic to maintain communications with the various control units governing other systems in the vehicle. One example of this cooperation is the way Motronic operates with the automatic transmission's ECU to implement torque reductions during gear changes, thus reducing wear on the transmission. If TCS (traction control system) is installed, its ECU responds to wheelslip by transmitting the corresponding data to the Motronic unit, which then reduces engine torque.
This is yet another benefit resulting from flexible torque-based control.

Diagnosis (OBD)

ME-Motronic is complemented by components designed for on-board monitoring (OBD), allowing it comply with the stringent emissions limits and the stipulations for integrated diagnosis.

Fig. 1

ME-Motronic – Schematic diagram

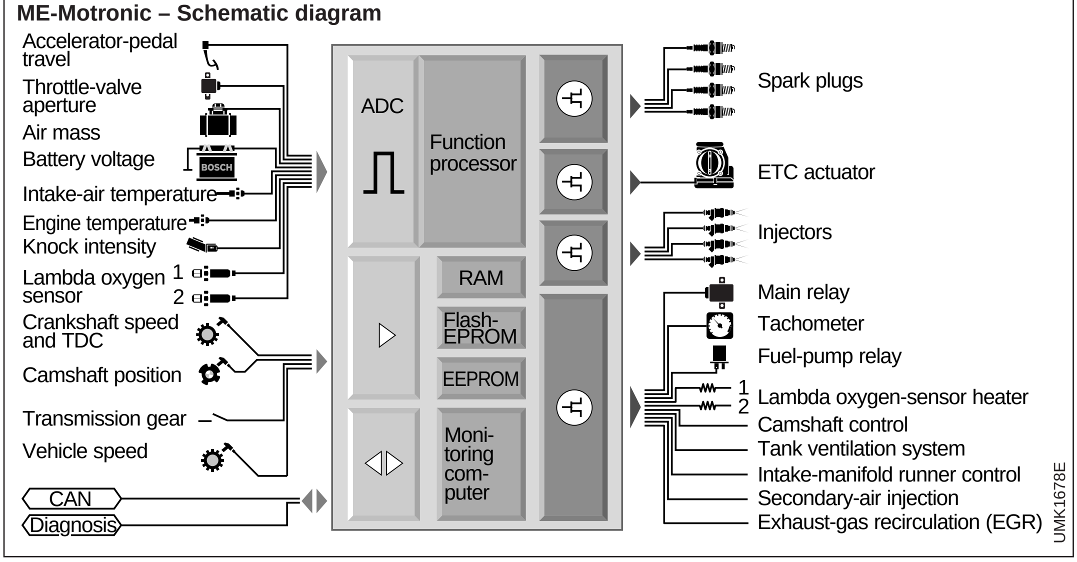
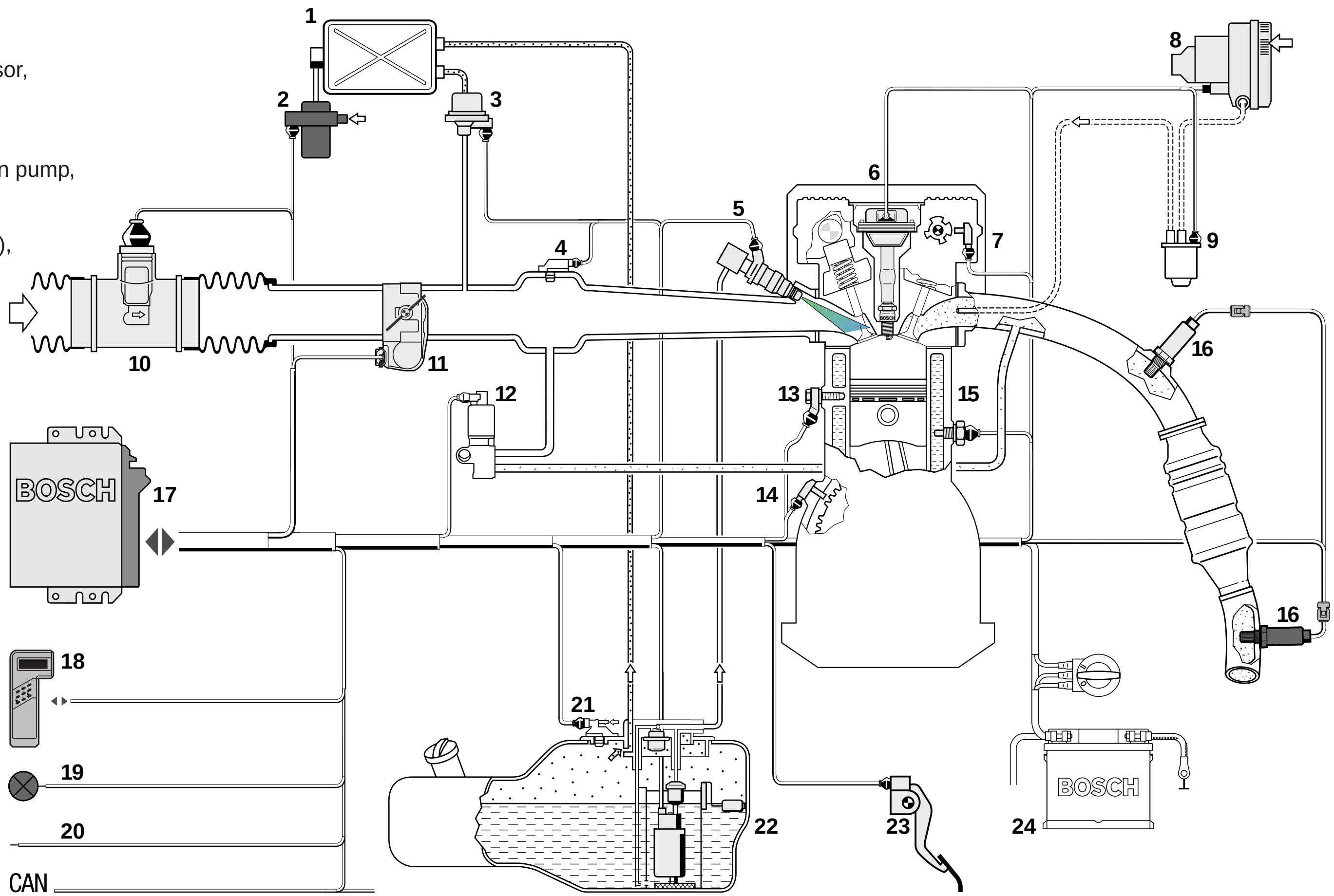


Fig. 2
ME-Motronic system diagram

ME-Motronic engine management ME7 (Example)

1 Activated-charcoal canister,
2 Check valve,
3 Canister-purge valve,
4 Intake-manifold pressure sensor,
5 Fuel rail/Injector,
6 Ignition coil/Spark plug,
7 Phase sensor,
8 Electric secondary-air injection pump,
9 Secondary-air injection valve,
10 Air-mass meter,
11 Throttle-valve assembly (ETC),
12 EGR valve,
13 Knock sensor,
14 RPM sensor,
15 Temperature sensor,
16 Lambda oxygen sensor,
17 Electronic control unit (ECU),
18 Diagnosis interface,
19 Diagnosis lamp,
20 Vehicle immobilizer,
21 Tank pressure sensor,
22 In-tank pump assembly,
23 Accelerator-pedal module,
24 Battery.



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Cylinder-charge control systems

Throttle-valve control

On spark-ignition engines with external mixture formation, the prime factor determining output force and thus power is the cylinder charge. The throttle valve controls cylinder charge by regulating the engine's induction airflow.

Conventional systems

Conventional layouts rely on mechanical linkage to control the throttle valve. A Bowden cable or linkage rod(s) translate accelerator-pedal travel into throttle-valve motion.

To compensate for the cold engine's higher levels of internal friction, a larger air mass is required and supplementary fuel must be injected. Increased air flow is also required to balance drive-power losses when ancillaries such as air-conditioning compressors are switched on. This additional air requirement can be met by an air-bypass actuator, which controls a supplementary air stream routed around the throttle valve (Figure 2). Yet another option is to use a throttle-valve actuator designed to respond to demand fluctuations by readjusting the throttle valve's minimum aperture. In both cases, the scope for electronic manipulation of airflow to meet fluctuating engine demand is limited to certain functions, such as idle control.

Principle of air control using air bypass valve

1 Idle valve (bypass valve), **2** ECU, **3** Throttle valve, **4** Bypass tract.

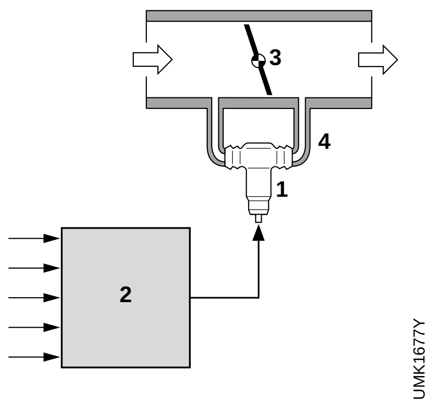


Fig. 2

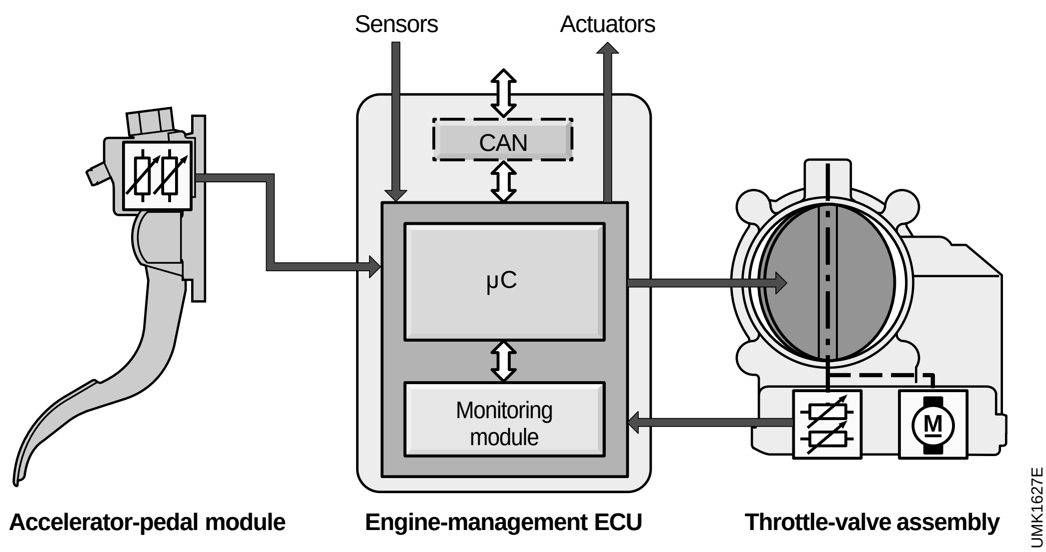
Systems with ETC

In contrast, ETC (electronic throttle control) employs an ECU to control throttle-valve travel. The throttle valve forms a single unit along with the throttle-valve actuator (DC motor) and the throttle-valve angle sensor: This is the throttle-valve assembly (Figure 1).

Two mutually-opposed potentiometers monitor accelerator-pedal travel as the basis for controlling this type of throttle-

Fig. 1

ETC system



valve assembly. The ECU calculates the throttle-valve aperture that corresponds to the driver's demand, makes any adjustments needed for adaptation to the engine's current operating conditions, and then generates a corresponding trigger signal for transmission to the throttle-valve actuator. The throttle-valve travel sensor with its two mutually-opposed potentiometers permits precise and accurate response to positioning commands.

The dual-potentiometer setup at the throttle valve is complemented by dual potentiometers to monitor accelerator pedal travel; this arrangement serves as an integral part of the overall ETC monitoring function by furnishing the desired system redundancy. This subsystem continuously checks and monitors all sensors and calculations that can affect throttle-valve aperture whenever the engine is running. The system's initial response to malfunctions is to revert to operation based on redundant sensors and process data. If no redundant signal is available, the throttle valve moves into its default position.

The ME-Motronic system integrates ETC control within the same engine-management ECU used to govern the ignition, injection and numerous auxiliary functions. This renders retention of a special dedicated ECU for ETC unnecessary. Figure 3 shows the components in an ETC system.

Gas-exchange control

Although throttle-valve control represents the primary method of regulating the flow of fresh air into the engine, a number of other systems are also capable of adjusting the mass of fresh and residual gases in the cylinder:

– Variable valve timing on both the intake and exhaust sides,
– Exhaust-gas recirculation (EGR),
– Variable-geometry intake manifold (dynamic boost), and
– Exhaust-gas turbocharger.

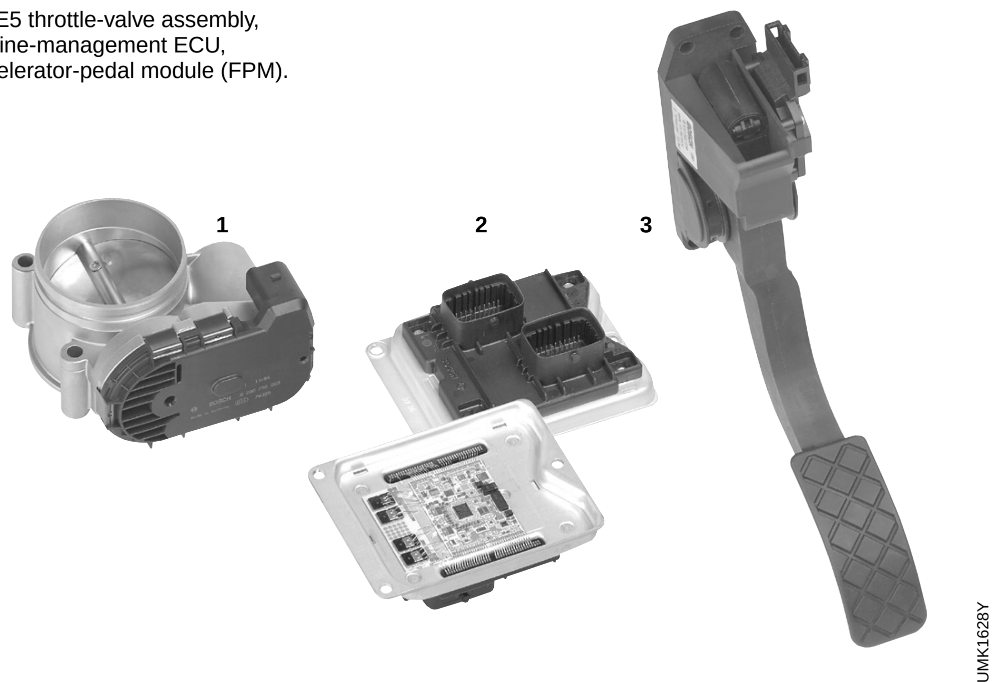
Variable valve timing

In defining valve timing it is important to recognize that fluctuations in factors such as engine speed and throttle-valve angle induce substantial variations in the flow patterns of the gas columns streaming into and out of the cylinder. This means that when invariable (fixed) valve timing

Fig. 3

ETC components

1 DV-E5 throttle-valve assembly,
2 Engine-management ECU,
3 Accelerator-pedal module (FPM).



Intake-camshaft rotation

1 Late, **2** Standard, **3** Early.

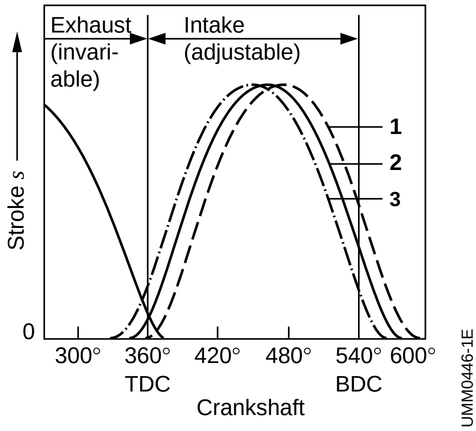


Fig. 4

Selective lobe activation

1 Standard, **2** Auxiliary lobe

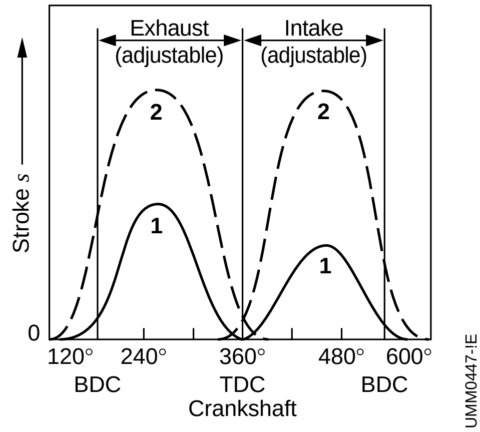


Fig. 5

is used the gas-exchange process can only be optimized for a single operating status. In contrast, variable valve timing can be employed to adapt gas flow to various engine speeds.

Camshaft adjustment

In conventional engines a chain or toothed timing belt serves as the mechanical link between the crankshaft and camshaft(s). On engines with adjustable camshaft, at least the angle of the intake-camshaft relative to the crankshaft can be varied. Nowadays, adjustment of both intake and exhaust camshafts relative to the crankshaft is being increasingly encountered. The adjustment process relies on electric or electrohydraulic actuators. Figure 4 shows how the open phase of the intake valve "shifts" relative to TDC when the intake camshaft's timing is modified. One option is to turn the camshaft to retard the "intake opens/closes" phase at idle to reduce residual gases and obtain smoother idling.

At high engine speeds the intake valve's closing point can be delayed to obtain maximum charge volumes. The same objective can be achieved at low to moderate engine speeds and/or in specific part-throttle ranges by varying the timing of the intake camshaft and shifting the entire intake cam-phase forward (advanced "intake opens/ closes").

Selective camshaft-lobe control

Systems with selective camshaft-lobe control modify valve timing by alternately activating cam lobes with two different ramp profiles.

The first lobe furnishes optimal valve timing and lift for intake and exhaust valves at the low end and in the middle of the engine's operating range.

A second cam lobe is available for increased valve lift and extended phase durations. This lobe is activated when the rocker arm to which it is connected locks onto the standard rocker arm once the engine crosses a specific speed threshold (Figure 5).

Infinitely-variable valve timing and valve-lift adjustment represents the optimum, but it is very complicated. This concept employs extended cam lobes featuring three-dimensional ramp profiles in conjunction with linear shifts in camshaft position, and grants maximum latitude for perfecting engine performance (Figure 6). This strategy can be used to obtain substantial torque increases throughout the engine's operating range.

Exhaust-gas recirculation (EGR)

Variable valve timing, as already mentioned in the section covering this subject, represents one way of influencing the mass of the residual gas remaining in the cylinder following combustion; this process is referred to as "internal" EGR.

Infinitely-variable valve timing and lift

a Minimum lift,
b Maximum lift.

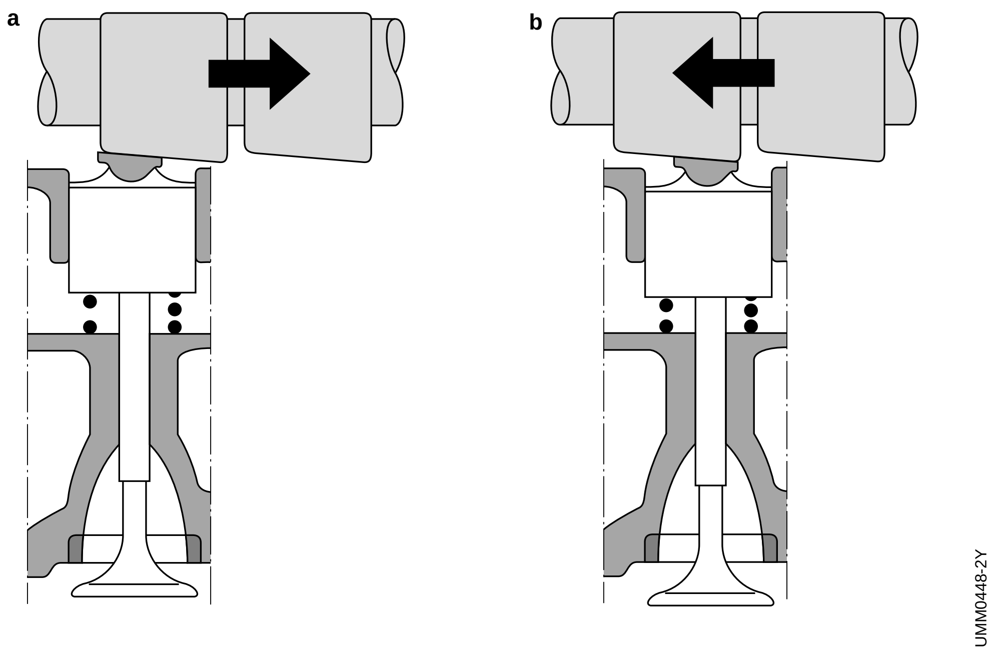
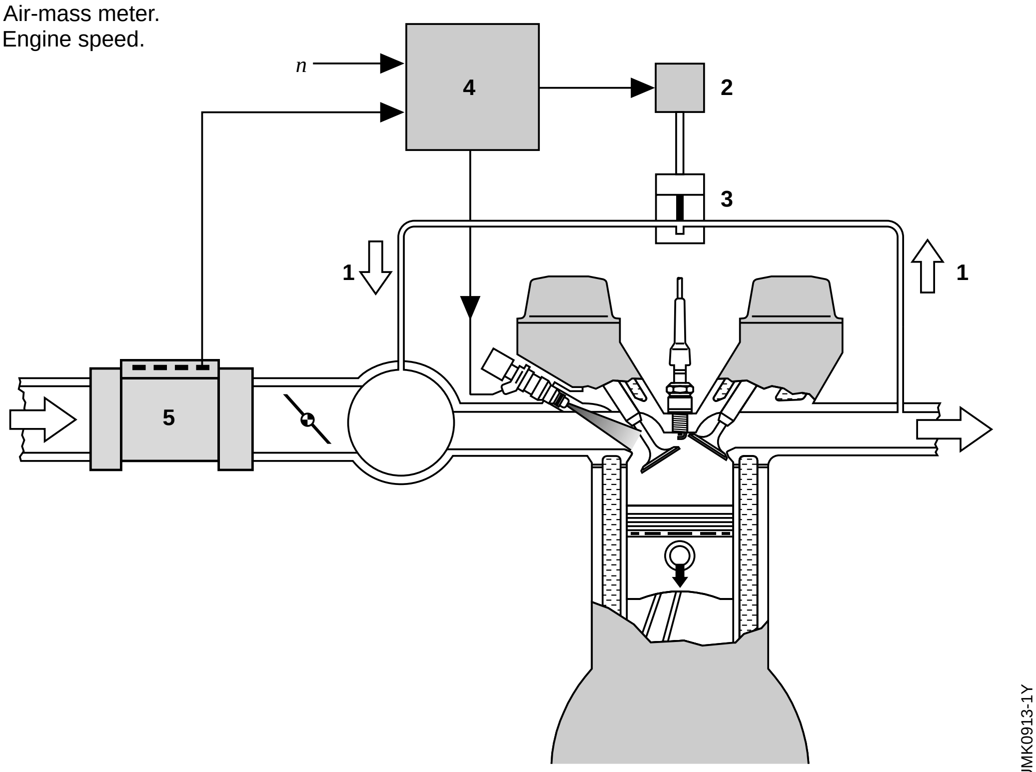


Fig. 6

Fig. 7

Exhaust-gas recirculation (example)

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



Another option available for varying the proportion of residual gases is to apply "external" EGR. Motronic controls this process by modifying the EGR valve's lift to reflect current engine operating conditions (Figure 7). The EGR system taps into the exhaust and, via the EGR valve, diverts a portion of the gases back into the fresh mixture. This is how the EGR valve defines the residual-gas component in the cylinder charge.
Exhaust-gas recirculation is an effective way to reduce emissions of nitrous oxides. Adding previously combusted exhaust gases to the fresh air-fuel mixture lowers peak combustion temperatures, and because generation of nitrous oxides is temperature-sensitive, NO_X emissions are reduced at the same time.
Assuming that the mass of the fresh-air charge remains constant, the overall charge will increase when exhaust gas is recirculated. This means the engine will produce the same torque at wider throttle-valve apertures (less throttling effect). The result is enhanced fuel economy.

Dynamic pressure-charging

Because maximum possible torque is proportional to fresh-gas cylinder charge, maximum torque can be raised by compressing the intake air before it enters the cylinder.
The gas-exchange process is not governed solely by valve timing; intake and exhaust-tract configuration are also important factors. Periodic pressure waves are generated inside the intake manifold during cylinder intake strokes. These pressure waves can be exploited to boost the fresh-gas charge and maximize possible torque generation.
Intake manifolds for multipoint injection systems consist of individually tuned runners and the plenum chamber with its throttle valve. Careful selection of the length and diameter of runners and of plenum chamber dimensions can be employed to exploit pressure waves in the air traveling through the intake tract. This strategy can be used to increase the density – and with it the mass – of the fresh-gas charge.

Intake wave ram effect

The pressure waves generated by the reciprocating piston propagate through the intake runners and are reflected at their ends. The idea is to adapt the length and diameter of the runners to the valve timing in such a way that a pressure peak reaches the intake valve just before it closes. This supplementary pressurization effect increases the mass of the fresh gas entering the cylinder.

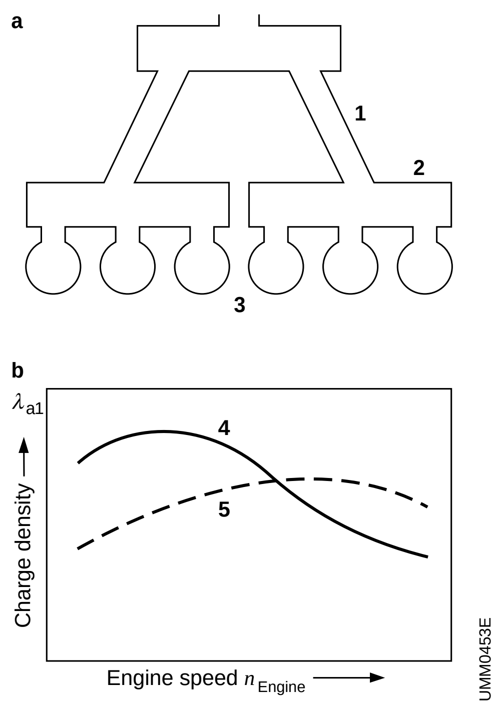
Resonance pressure charging

Resonance boost systems employ short runners as links between groups of cylinders with equal ignition intervals and 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 8). Sometimes the required plenum volumes are considerable, and the resulting storage effect can have a negative influence on dynamic response (in the form of mixture vacillation during sudden changes in load).

Fig. 8

Resonance boost

a Layout, **b** Air flow path.
1 Resonance tube, **2** Resonance chamber,
3 Cylinder, **4** With resonance boost,
5 With standard intake manifold.



Variable-geometry intake systems

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

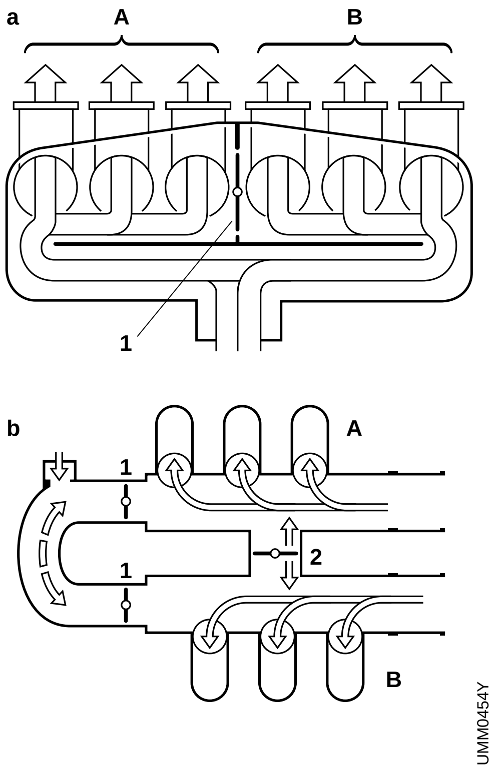
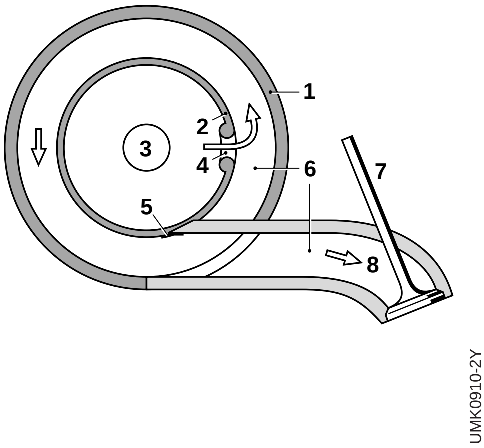


Fig. 9

Fig. 10

Infinitely-variable-length intake system

1 Fixed housing,
2 Rotating drum (air distributor),
3 Drum air-entry orifice,
4 Intake runner air-entry orifice,
5 Seal (e.g., leaf spring),
6 Intake runners,
7 Intake valve,
8 Induction air flow.



Brake mean effective pressure (bmep) over engine speed for three different lengths of an infinitely-variable intake system

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

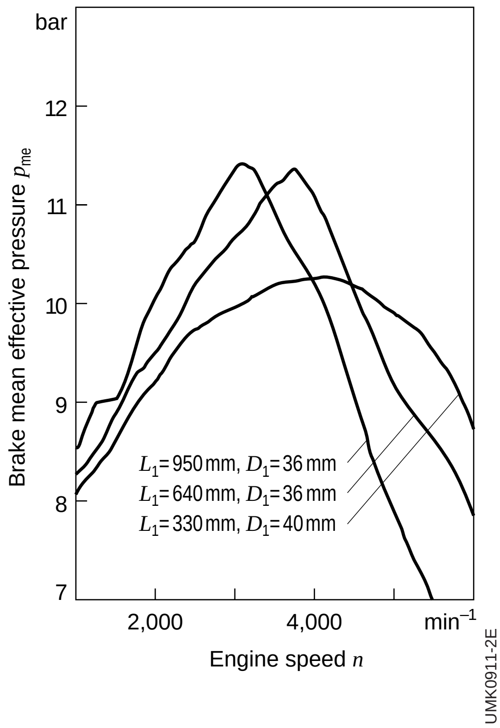


Fig. 11

Variable-geometry intake manifold

Both of the dynamic charge-flow enhancement concepts described above are suitable for increasing the maximum available charge volume, especially at the low end of the engine-speed range.

However, the variable-geometry intake manifold (selective intake-tract activation) can be used to obtain a virtually ideal torque curve. This concept opens up numerous possibilities, with adaptive response to variations in such factors as engine load factor, engine speed and throttle-valve angle:

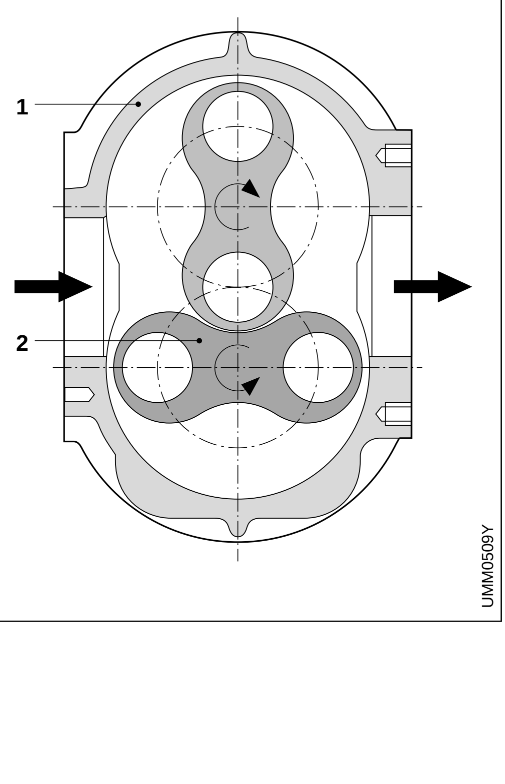
– Adjustment of runner length,
– Alternation between different runners of varying lengths and diameters,
– Selective deactivation of individual runners leading to individual cylinders in multitract systems, and
– Alternating selection of different plenum-chamber volumes.

For switching such variable-geometry air-intake systems, as engine speed varies flaps or similar devices are used to open and close the connections between the groups of cylinders (Fig. 9).
At low rpm the variable-length intake runners operate in combination with the initial resonance chamber. The length of the runners is then subjected to ongoing modification as engine speed rises before the process culminates in the opening of a second resonance chamber (Figure 10).
Figure 11 provides an index of air-flow efficiency by illustrating how variable intake-manifold geometry influences brake mean effective pressure (bmep) as a function of engine speed.

Exhaust-gas turbocharging
Yet another and even more effective option in the quest for higher induction-gas density levels is to install a supercharging device. Among the familiar techniques for furnishing forced induction on spark-ignition engines, the most widespread is turbocharging. Turbocharging makes it possible to extract high torques and powers from engines with minimal piston displacement by running them at high levels of volumetric efficiency. Compared to naturally-aspirated engines, if we presume identical power outputs, it is the turbocharged powerplant's lower weight and more compact dimensions that represent its salient advantages.
Until recently turbochargers were primarily viewed as a way to raise power-to-weight ratios, but today the focus is shifting toward achieving torque increases in the low and mid-range sectors of the engine's rev spectrum. This strategy is very often applied in conjunction with electronic boost-pressure control.
The exhaust-gas turbocharger's primary components are the impeller and the turbine, which are mounted at opposite ends of a common shaft.
The energy employed to drive the turbocharger is extracted from the engine's exhaust stream. Although, on the one hand, this process exploits energy that in naturally-aspirated engines would otherwise be wasted (owing to the inherent expansion limits imposed by the crankshaft assembly), on the other, the turbo process generates higher exhaust back-pressures as the price for impeller power.
The exhaust turbine converts a portion of the exhaust-gas' energy into rotational energy to drive the impeller, which draws in fresh air before compressing it and dispatching it through the intercooler, throttle-valve and intake manifold on its way to the engine.

Turbocharger boost-control actuator
Even at the bottom end of the rev range, passenger-car engines are expected to produce high torque. In response, turbine housings are basically designed for efficient operation with modest exhaust-gas mass flow rates, i.e., for WOT as low as $n \leq 2{,}000\ \text{min}^{-1}$.
To prevent the turbocharger from overboosting the engine when the exhaust-gas stream rises to higher mass flow rates, the unit must incorporate a bypass valve. This bypass valve, or

Fig. 12
Cross section of a Roots-type supercharger
1 Housing,
2 Rotor.



wastegate, diverts a portion of the exhaust gases in these higher ranges, routing them around the turbine and feeding the gas back into the exhaust system on the other side. This bypass valve is generally a flap valve integrated within the turbine housing, although less frequently it is in the form of a plate valve in a separate casing parallel to the turbine.

Mechanical supercharging

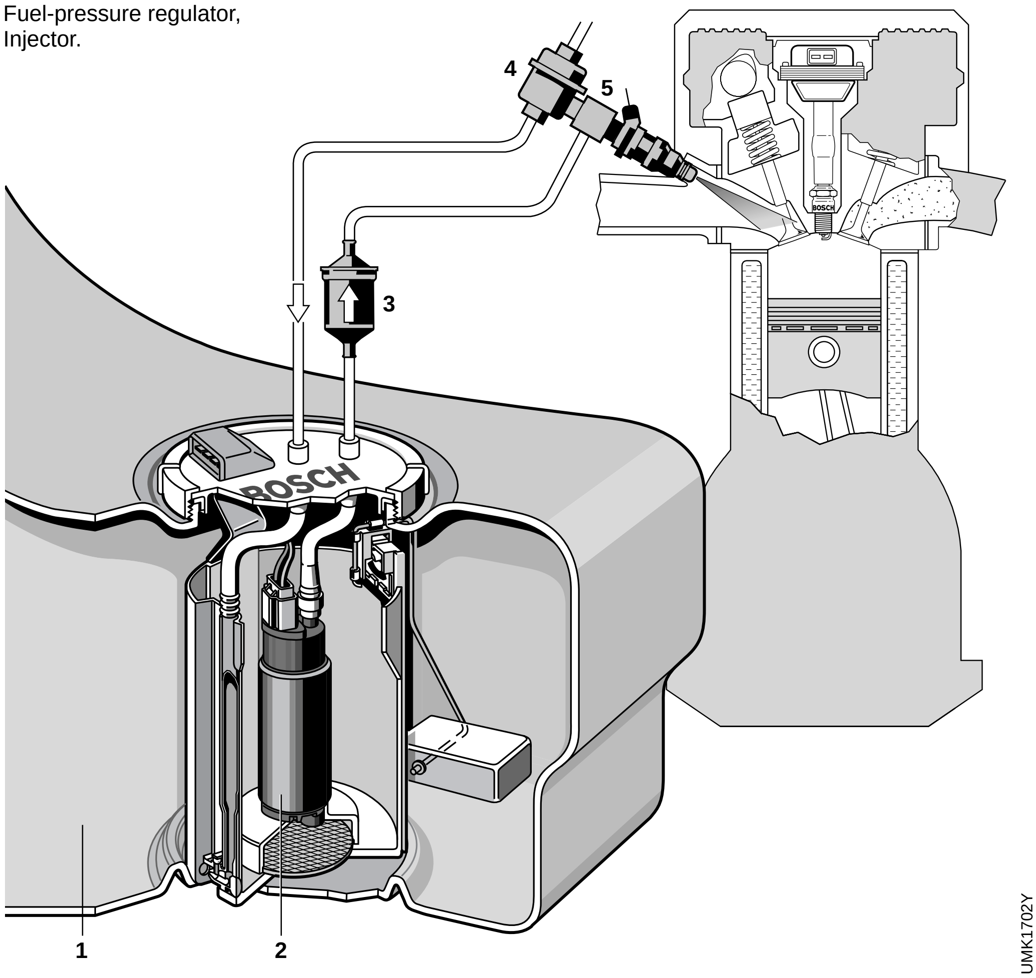
The motive force used to power the mechanical compressor is taken directly from the IC engine. Mechanical superchargers are available in a variety of designs, and Figure 12 shows the Roots compressor. A belt from the crankshaft usually drives the supercharger at a fixed ratio, so crankshaft and supercharger rotate at mutually invariable rates. One difference relative to the turbocharger is that the supercharger responds immediately to increases in rpm and load factor, with no lag while waiting for the impeller to accelerate. The result is higher engine torque in dynamic operation.

This advantage is relativized by the power needed to turn the compressor. This must be subtracted from the engine's effective net output, and leads to somewhat higher fuel consumption. However, Motronic can lessen the significance of this factor by controlling operation of a compressor clutch to switch off the supercharger at low rpm and under light loads.

Fig. 1

Fuel-supply system with return line

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



EKP 13.5 electric fuel pump

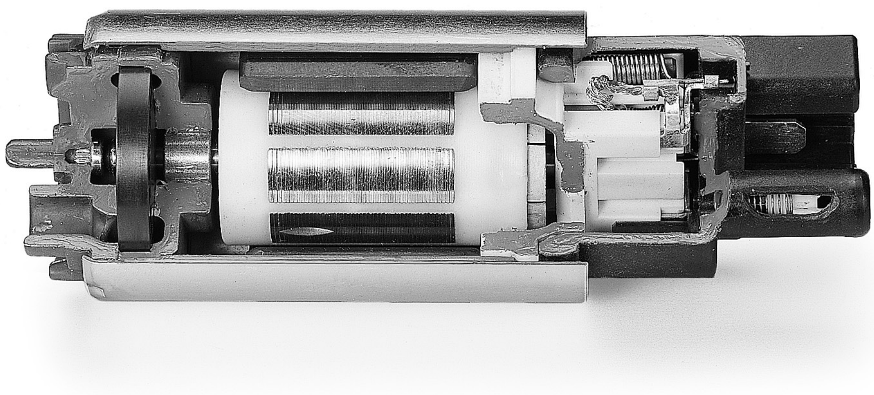


Fig. 2

Fuel system

Fuel supply and delivery

Systems with and without return lines

The fuel system's function is to ensure a consistently reliable supply of the fuel mass needed to meet the engine's requirements under any and all operating conditions. An electric pump draws the fuel from the tank and forces it through a filter for delivery to the fuel (distribution) rail with its solenoid-triggered injectors. The injectors then spray the fuel into the engine's intake manifold in precisely metered quantities. In systems equipped with a return line the excess fuel then flows through the pressure regulator and back to the fuel tank (Figure 1). Until quite recently this layout represented the state-of-the-art, but now returnless fuel-supply systems are becoming increasingly common.

Both systems rely on an electric fuel pump to provide the fuel circuit with a continuous supply of fuel from the tank. A pressure regulator, operating at a typical supply pressure of 300 kPa, maintains system pressure by controlling the flow of fuel returning to the tank. The high pressure inhibits formation of undesirable vapor bubbles in the fuel.

In the returnless system the pressure regulator is mounted immediately adjacent to the pump. This means that the fuel return line from the engine can be omitted, thus lowering production costs as well as fuel temperatures within the tank. These lower temperatures mean lower hydrocarbon emissions, and promote improved performance from the evaporative-emissions control system.

Electric fuel pump

The electric fuel pump maintains a continuous flow of fuel from the tank. It can be installed either within the tank itself (in-tank) or mounted externally in the fuel line (in-line).

The in-tank pumps currently in general use (Figure 2 shows the EKP 13.5 as a representative example) are integrated within pump assembly units along with the fuel-gauge sensor and a swirl baffle designed to remove vapor from the fuel in the return line. Hot-delivery problems in systems with in-line pumps can be solved by installing a supplementary booster pump within the tank to maintain a low-pressure fuel flow up to the main pump unit. The pump's maximum delivery capacity always exceeds the system's theoretical maximum requirement to ensure that it remains consistently capable of maintaining system pressure under all operating conditions.

The fuel pump is switched-on by the engine-management ECU. An interrupt circuit or software-based function stops fuel delivery whenever the engine is stationary with the ignition switched on.

Fuel filter

1 Paper element,
2 Strainer,
3 Support plate.

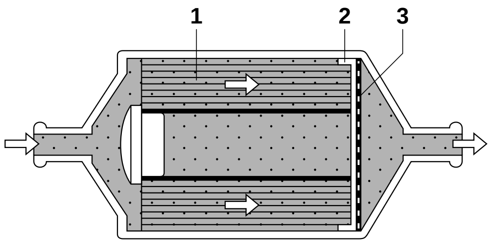


Fig. 3

Fig. 4

Fuel-pressure regulator

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

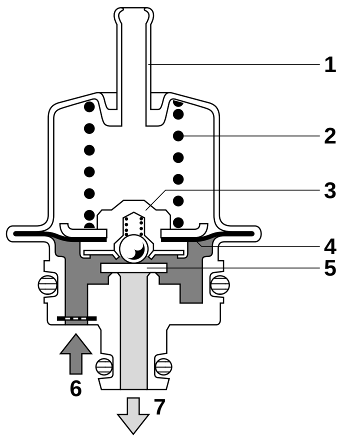
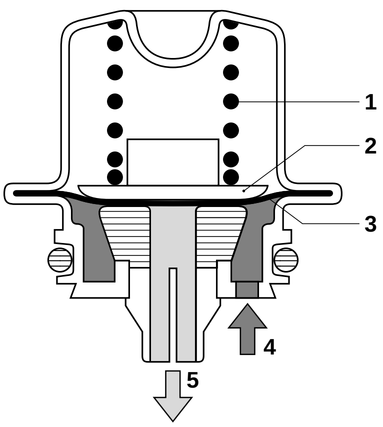


Fig. 5

Fuel-pressure attenuator

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



Fuel rail

The fuel then flows through the fuel rail, whence it is evenly distributed to all injectors. The injectors are mounted on the fuel rail, while rails in systems with a return line also include a fuel-pressure regulator. A pressure attenuator may also be installed. Fuel rails are carefully dimensioned to prevent local fuel-pressure variations caused by resonances which occur when the injectors open and close. This prevents irregularities in injection mass flow that might otherwise arise during load and rpm transitions. Depending upon the particular vehicle type and its special requirements, fuel rails can be manufactured in steel, aluminum or plastic. The rail may also incorporate a test valve, which can be used to bleed pressure during servicing as well as for test purposes.

Fuel filter

Fuel-borne contaminants can impair the operation of both pressure regulator and injectors. A filter is therefore installed in the line downstream from the electric fuel pump. This fuel filter contains a paper element with a mean pore diameter of 10 µm (Figure 3).

Fuel-pressure regulator

Injection quantities are determined by injection duration and the pressure differential between the fuel in the rail and the intake-manifold pressure.

Return-line systems utilise a pressure regulator to maintain the pressure differential between fuel system and intake manifold at a constant level. This pressure regulator regulates the amount of fuel returning to the tank to maintain a constant pressure drop across the injectors (Figure 4). In order to ensure that the fuel rail is efficiently flushed with fuel, the pressure regulator is generally mounted at its far end.

In systems without a return line the pressure regulator is installed within the in-tank pump assembly, whence it maintains the pressure within the fuel rail at a constant level relative to ambient pressure. As this implies, the system does not maintain a constant pressure differential between rail and manifold, so injection duration must be calculated accordingly.

Fuel-pressure attenuator
The injectors' cyclical operating phases and the periodic fuel-discharge characteristic of the positive-displacement fuel pump both induce pressure waves in the fuel system. Under unfavorable circumstances the mounts on the electric fuel pump, the fuel lines and the rail itself can transmit these vibrations to the fuel tank and to the vehicle's body. Noise stemming from these sources can be inhibited through the use of specially designed mounts and with fuel-pressure attenuators. The fuel-pressure attenuator (Figure 5) shares its general design layout with the pressure regulator, with a spring-loaded diaphragm separating the fuel from the air space in both cases.

Fuel injection

Uncompromising demands for running refinement and low emissions make it vital to provide consistently excellent air-fuel mixture quality in every engine cycle. The system must inject fuel in quantities precisely metered to reflect the induction air's mass (duration), while the start of injection is also a significant factor (timing). This is why multipoint injection systems feature a solenoid-operated injector for each individual cylinder. At the injection point specified by the ECU the injector sprays a precisely metered quantity of fuel into the area directly in front of the cylinder's intake valve(s). Thus condensation along the walls of the intake tract leading to undesired deviations from the prescribed lambda values is largely avoided. Because the engine's intake manifold conducts only combustion air, its geometry can be optimized solely on the basis of the engine's dynamic gas-flow requirements.

Solenoid injector

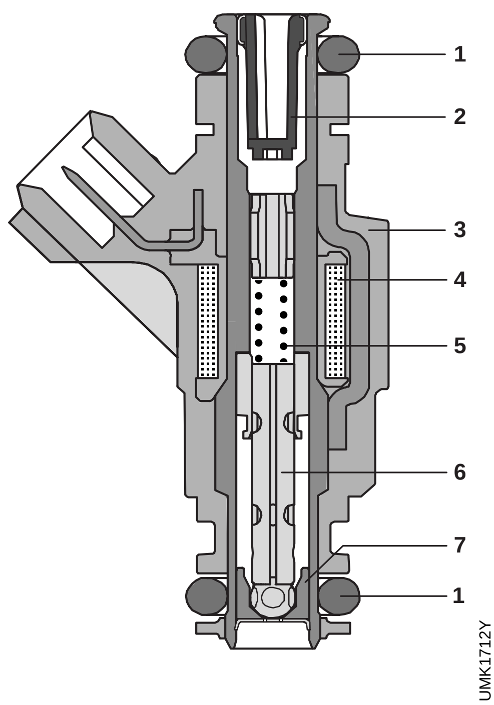
Design and function
The essential components of the injector are
- The valve casing with solenoid winding and electrical connection,
- Valve seat with injector-nozzle disc, and
- Reciprocating valve needle with solenoid armature.

A filter screen in the fuel supply line guards the valve against contamination, while two O-rings seal the injector against the fuel rail and the intake manifold. The spring and the force of fuel pressure against the valve seat insulate the fuel-supply system from the intake manifold for as long as the coil remains without current flow (Figure 6).
When the injector's solenoid winding is energized, the coil responds by

Fig. 6

EV6 injector design

1 O-rings,
2 Strainer,
3 Valve housing with electrical connection,
4 Coil winding,
5 Spring,
6 Valve needle with solenoid armature,
7 Valve seat with hole plate.



generating a magnetic field. This lifts the armature, the valve needle rises from the seat, and fuel flows through the injector. System pressure and the exit aperture defined by the orifices in the injector nozzle are the primary factors in determining the injected fuel quantity per unit of time. The valve needle closes again as soon as the trigger current ceases to flow.

Designs

Four injector configurations are employed to cover virtually all current Motronic systems (Figure 7):

– The EV1 injector has been in production since the early 70s. Thanks to ongoing development, this injector continues to satisfy all of the essential requirements encountered in modern gasoline-injection systems.

– The EV6 is based on the EV1. It features superior hot-starting for improved performance in returnless fuel-supply systems. This is important, considering the fact that the fuel supplied to the injector is hotter in a returnless system than it is in systems with fuel return. There are also a range of different EV6 installation lengths available.

– So-called “air shrouding” can be added to further improve mixture formation with the EV6 (refer to Figure 9).

– The EV12 has been developed from the EV6. For intake manifolds having difficult geometries, the injection point can be shifted forward 20 mm to obtain an ideal position.

Spray formation

The injector’s spray pattern as determined by discharge geometry and angle, as well as droplet size, influence the air-fuel mixture formation process. Different spray-patterns are needed to adapt the injector for use with specific geometrical configurations in cylinder head and intake manifold. Various spray concepts are available to meet these requirements (Figure 8).

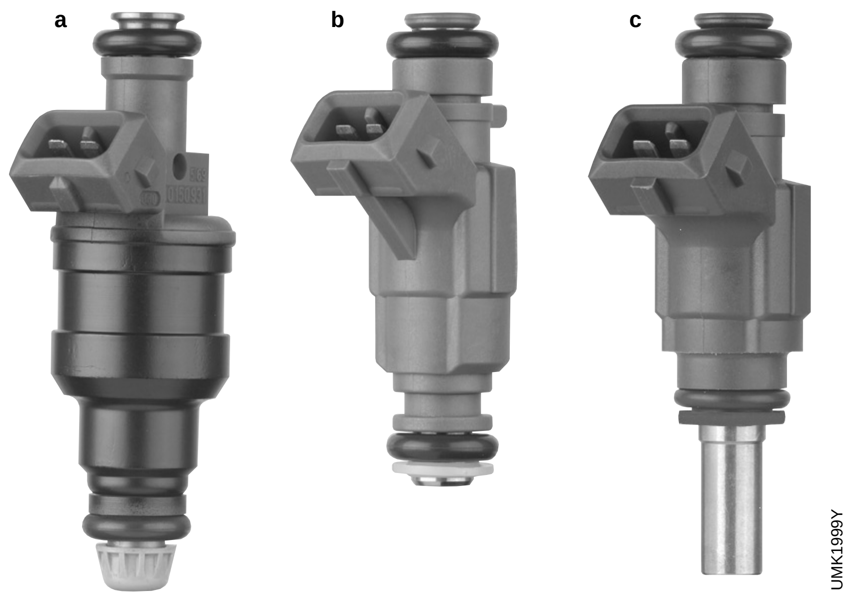
Tapered spray pattern

After being discharged through individual orifices in the nozzle plate, the emerging fuel streams converge to form a tapered spray cloud. A pintle at the bottom end of the needle and extending beyond the nozzle can also be employed to obtain a tapered form. Engines with a single

Fig. 7

View of different injectors

a EV1 injector,
b EV6 injector,
c EV12 injector.



intake valve are a typical application area for tapered-spray injectors, which focus their discharge toward the opening between the intake valve and the wall of the intake manifold.

Dual-stream injector
Dual-stream patterns are employed in engines with two intake valves. The nozzle plate's holes are arranged to concentrate the emerging fuel in two spray patterns. Each of these clouds serves one intake valve.

Air shrouding
Air-shrouded valves exploit the pressure differential between intake manifold and the ambient atmosphere to enhance the quality of mixture formation. Air is conducted through a shroud in the discharge region of the nozzle plate. This air accelerates to extremely high velocities as it travels through this narrow passage before then emerging and promoting more intense atomization in the air-fuel mixture (Figure 9).

Fig. 8

Discharge patterns

a Tapered pattern,
b Dual-stream pattern.

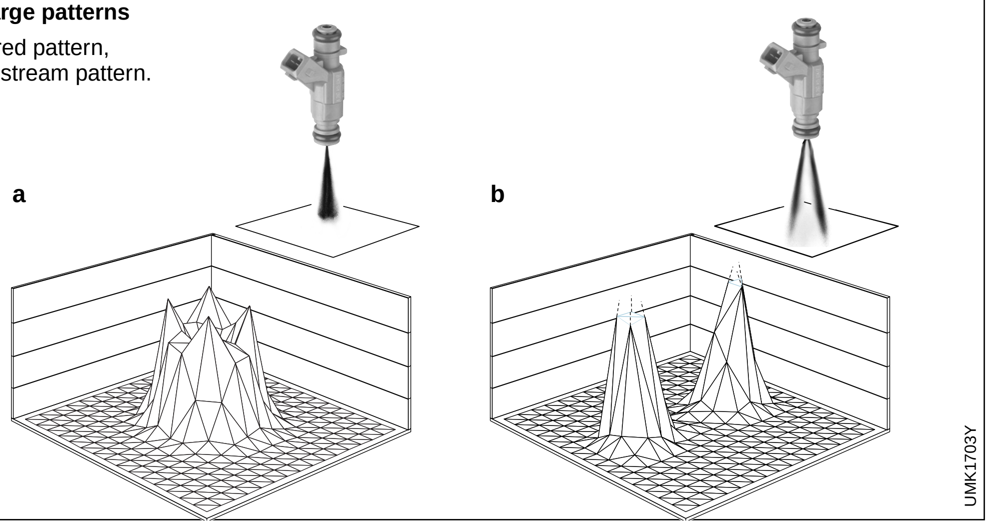
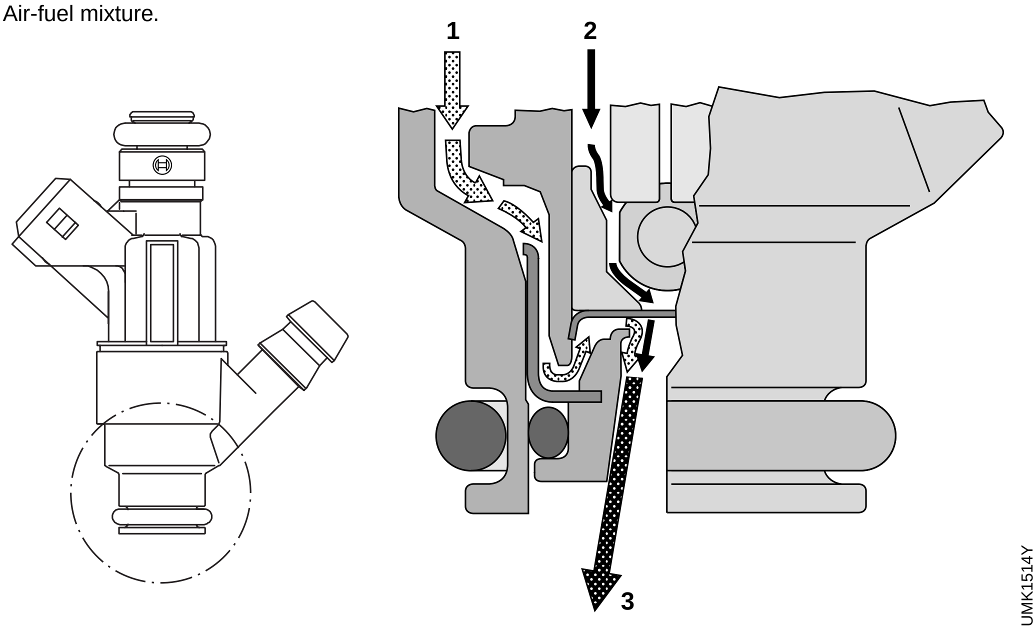


Fig. 9

EV6 with external air shroud

1 Air,
2 Fuel,
3 Air-fuel mixture.



Operating-data acquisition

Driver demand

There is no mechanical linkage to connect the accelerator pedal with the throttle valve in engine-management systems with ETC throttle control. Instead, the accelerator pedal's position is monitored by an accelerator-pedal travel sensor for transformation into an electrical signal within the pedal module. The engine-management system interprets this signal as "driver demand."

The accelerator-pedal module is an operational unit unifying all required pedal functions and mechanical components. Although this concept renders all in-vehicle adjustment redundant, the fact that a wide variety of individual installation geometries together with very cramped conditions are frequently encountered often makes it necessary to resort to specific configurations according to vehicle type.

The accelerator-pedal travel sensor incorporates a redundancy feature (two potentiometers) to facilitate diagnosis and ensure backup operation. The sensors operate with separate reference voltages from mutually independent sources, and their signals are also processed separately within the engine ECU.

Air charge

Engines with design concepts based on manifold injection display a linear relationship between air-charge density and the force generated in the combustion process, which, in turn, corresponds to the engine's load factor. This is why air charge is more than simply a primary parameter for calculating injection quantity and ignition timing in the ME-Motronic system. In a torque-based system such as ME-Motronic, cylinder charge also serves as the basis for calculating instantaneous engine torque generation.

Using the following sensors, the ME-Motronic monitors cylinder charge with the following sensors:

– Hot-film air-mass meter (HFM),
– Intake-manifold pressure sensor (DS-S),
– Ambient-pressure sensor (DS-U),
– Boost-pressure sensor (DS-L, on turbocharged powerplants), and
– Throttle-valve sensor (DKG).

Concepts for monitoring induction charge vary according to engine, and all sensors are not present in all systems. Values for those parameters that are not directly monitored are derived from other, monitored data.

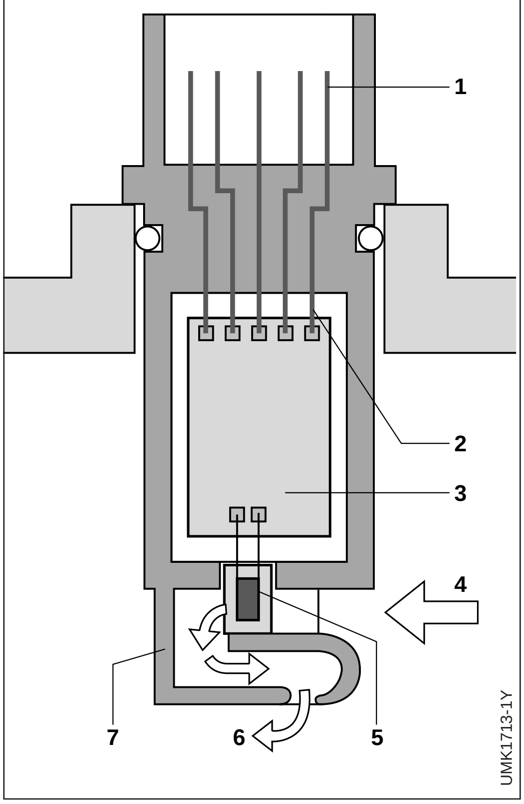
HFM5 hot-film air-mass meter

The hot-film sensor is a "thermal" air-flow monitor. Basically speaking, this meter (or sensor) is positioned somewhere between air filter and throttle valve,

Fig. 1

Sensor element in hot-film air-mass meter

1 Electrical connections, **2** Electrical terminals, **3** Evaluation electronics, **4** Air intake, **5** Sensor element, **6** Air discharge, **7** Casing.



although it can for example be located as an “insert” in the air-filter housing or in a sensor tube in the air-intake tract. Figure 1 shows the meter’s design.

The air-flow meter must register the engine’s mass air intake (kg/h) with the utmost precision. At high load factors, in particular, there is a tendency for reverse flow pulses from the piston to generate waves in the air upstream from the throttle valve. These reverse undulations should not be allowed to detract from the air-mass meter’s monitoring accuracy.

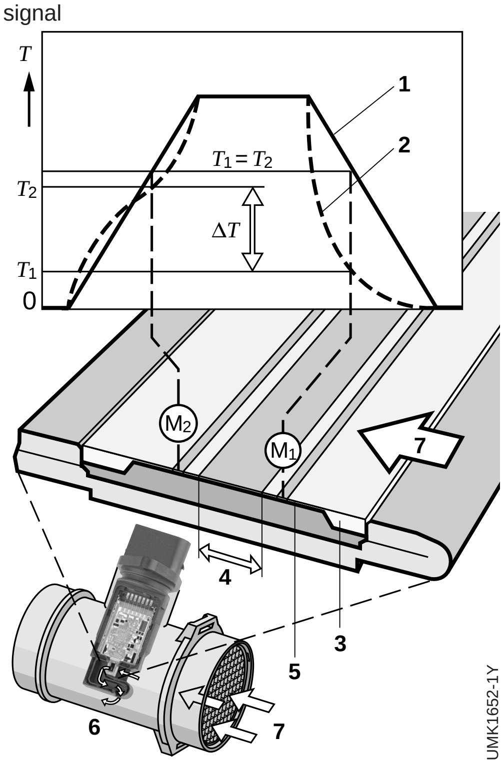
The HFM5 hot-film air-mass meter is a micromechanical device featuring a hot zone heated to a specific temperature. Temperatures drop on each side of this zone. If no air flow is present, then the descending thermal gradients in the sectors on each side of the hot zone will be identical. Air-flow leads to a more radical temperature progression curve for the induction side, in response to the cooling effect that the incoming air exerts on the sensor element. Although the air flowing past the opposite (engine) side of the sensor also has a cooling effect, the air warmed by the heater element ultimately raises relative temperatures on this side. The result is the thermal progression pattern in the illustration. Temperatures T_1 and T_2 are found at the two monitoring points M_1 and M_2, respectively, with the actual differential being determined by induction-air mass. This differential ΔT is then converted into a voltage.

This monitoring concept also makes it possible to register reverse flow, a situation reflected by T_2 being lower than T_1.

Fig. 2

Monitoring concept of hot-film air-mass meter

1 Temperature curve, no air flow,
2 Temperature curve, with air flow,
3 Sensor, **4** Heated zone, **5** Diaphragm,
6 HFM5 with measurement tube, **7** Air current.
M_1, M_2 Scan points, T_1, T_2 Temperatures,
ΔT Temperature differential generates sensor signal



HFM5 signal voltage over air mass

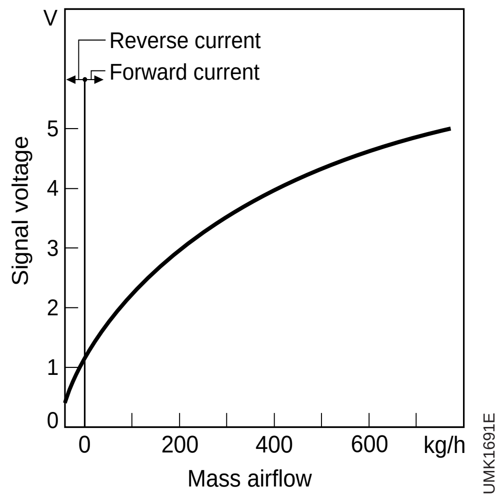


Fig. 3

The temperature differential is a direct index of induction-air mass. The relationship between air mass and the voltage generated by the processing circuit defines the sensor’s characteristic curve (Figure 3), which has sectors for both forward and reverse flow. Monitoring precision is further enhanced by using a voltage provided by the Motronic ECU as a reference for the sensor signal. The characteristic curve has also been plotted to aid the Motronic’s integrated diagnosis in recognizing problems such as open circuits.

A separate intake-air temperature sensor can also be integrated in the system.

Pressure sensor (for integration in ECU)

1 Pressure connection,
2 Pressure cell with sensor elements,
3 Seal rim,
4 Evaluation circuit,
5 Thick-layer hybrid (ceramic substrate).

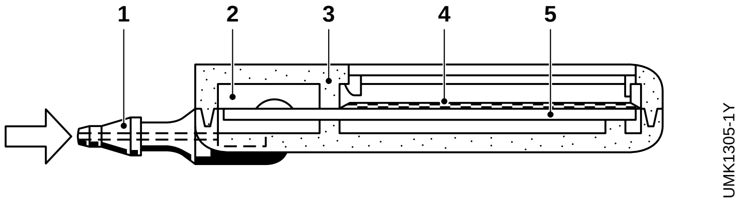


Fig. 4

The sensor tube is available in a variety of diameters designed to reflect the engine's specific airflow requirements.

Intake-manifold pressure sensor
A pneumatic link allows the pressure sensor to monitor absolute pressure levels (kPa) within the intake manifold. The sensor's monitoring range extends from 2...115 kPa (20...1,150 mbar).
This micromechanical sensor is available either for integration within the ECU or as a remote sensor for local mounting on or near the manifold.
This sensor, with its pneumatic link to the manifold, consists of a pressure cell with two sensor elements and a space for the processing circuitry, both of which are mounted on a single ceramic substrate (Figure 4). The processing circuit amplifies the electrical voltages generated at various levels as a reflection of pressure variations. It also compensates for thermal effects and converts the pressure curve to linear form. The signal from the processing circuit is transmitted to the ECU.

Ambient-pressure sensor
The ambient-pressure sensor shares its design with the manifold-pressure sensor, but is located inside the ECU. Barometric pressure readings are needed by systems that rely on throttle-valve aperture instead of an air-mass meter to monitor the incoming air supply. The precise data on ambient air density registered by this sensor serve as a significant factor in numerous diagnosis functions.

Boost-pressure sensor
In order to regulate boost pressure, this must first of all be measured. This function is taken over by a pressure sensor with a measurement range extending up to 250 kPa (2,500 mbar).

Throttle-valve sensor
ME-Motronic uses the throttle valve to control engine output. The throttle-valve sensor is employed to verify that the throttle valve responds to commands by assuming the specified angle (closed-loop position control).
The throttle-valve sensor forms an integral component within the throttle-valve assembly, with system redundancy provided by sensors incorporating two separate potentiometers operating with individual reference voltages.

Engine speed, crankshaft and camshaft angles

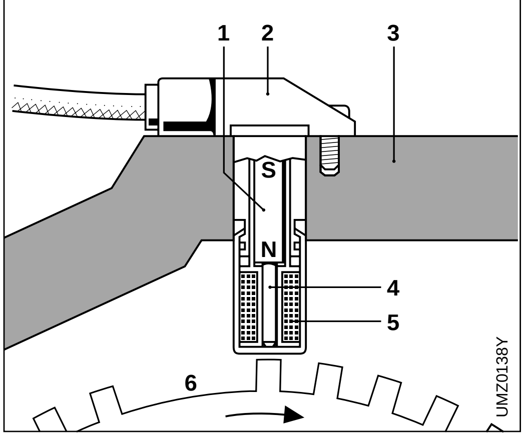
Engine speed and crankshaft angle
The momentary piston position (travel) within the cylinder serves the system as one parameter for defining the ignition firing point. Because each piston in each cylinder is joined to the crankshaft via a connecting rod, the crankshaft or engine-speed sensor can provide data indicating piston travel in each cylinder. The rate at which the crankshaft changes its position is the engine speed, quantified in the number of crankshaft revolutions per minute (rpm). Engine speed, which is yet another vital operating parameter for Motronic, is also calculated from crankshaft position.

<u>Generating the crankshaft-angle signal</u>
Installed on the crankshaft is a ferro-magnetic sensor rotor with a theoretical

capacity for 60 teeth, but with a 2-tooth gap, giving an actual sequence of 58 teeth. This 58-tooth sequence is scanned by an inductive rpm sensor featuring a soft-iron core and a permanent magnet (Figure 5). The sensor's magnetic field responds to the passing teeth by generating AC voltage. The amplitude of this voltage rises radically at progressively higher engine speeds. Adequate amplitudes for operation are available at minimal engine speeds, extending as low as 20 min^{-1}.
Tooth and sensor-pole geometries must be precisely matched. The processing circuit in the ECU converts the sinus-wave voltage with its highly inconsistent amplitudes into a constant-amplitude square-wave pattern.

Determining crankshaft angle
The flank data for the square-wave voltage are transmitted to the computer through 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 sectors. This tooth gap corresponds to a specific crankshaft angle, with cylinder no. 1 as the reference. The computer uses this point as its crankshaft synchronization reference. It then counts 6 degrees further for each subsequent negative tooth flank. Because the ignition

Fig. 5
Inductive RPM sensor
1 Permanent magnet, **2** Housing, **3** Engine block, **4** Soft-iron core, **5** Winding, **6** Toothed rotor with reference point (gap).



must be triggered based on more minute increments, the interflank periodicity is divided by eight. This new unit can then be multiplied by one, two, three, etc., and appended to a flank position as the basis for determining the firing point, allowing timing adjustments in increments of 0.75 degrees.

Calculating segment duration and engine speed from engine-speed sensor signal
The cylinder offset in the four-stroke engine results in two crankshaft revolutions (720 degrees) elapsing between the start of each new cycle on cylinder no. 1. This period defines the mean ignition interval, and the intermediate duration is the segment time T_S. Equal distribution of the intervals results in:

Table 1

Interval	Degrees	Teeth
2 cylinders	360	60
3 cylinders	240	40
4 cylinders	180	30
5 cylinders	144	24
6 cylinders	120	20

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

Camshaft position
The camshaft controls the engine's intake and exhaust valves while rotating at half the rate of the crankshaft.
When a piston travels to top dead center, it is the positions of the intake and exhaust valves, as defined by the camshaft, that determine the cylinder's actual cycle phase. TDC can mark the end of the compression phase and the start of ignition, or, alternatively, the end of the exhaust stroke. Crankshaft data alone cannot indicate which.
Motronic systems with static (distributorless) ignition and dedicated coils – such as ME-Motronic – differ from

systems with rotating spark distribution by requiring supplementary data on cycle phase. The ECU must decide which coil and spark-plug combination are to be triggered, and it derives this information from the camshaft's position.
Data for camshaft position are also needed by systems that adapt injection timing for each cylinder individually, such as those with sequential injection.

Hall-sensor signal

Camshaft position is usually monitored with a Hall sensor. This consists of a Hall element with a semiconductor wafer through which current flows. This ferromagnetic Hall element responds to activation by a trigger wheel rotating in unison with the camshaft by generating voltage at right angles to the direction of the current passing through it.

Determining camshaft position

As the Hall voltage lies in the millivolt range, it must be processed in the sensor before being transmitted to the ECU in the form of a switching signal. The basic procedure is for the microprocessor to respond to trigger-wheel gaps by checking for Hall voltage and determining whether or not cylinder no. 1 is on its power stroke.
Special trigger-wheel designs make it possible to use the camshaft signal as an emergency back-up in the event of crankshaft sensor failure. However, the resolution provided by the camshaft signal is much too low to allow its use as a permanent replacement for the crankshaft rpm sensor.

Mixture composition

Excess-air factor λ

The lambda excess-air factor (λ) quantifies the relative masses of the air and fuel in the mixture. Optimal performance is obtained from the catalyst at λ = 1. Because the lambda, or O_2 sensor, monitors the concentration of oxygen in the exhaust gas, its signals are an index of the excess-air factor lambda (λ).

Oxygen (lambda) sensor

The oxygen sensor (lambda probe) is a "dual-threshold" unit capable of indicating both rich (λ<1) and lean (λ>1) mixtures. The radical transitions that characterize this sensor's response curve (Figure 7) facilitate mixture adjustments to achieve λ=1.
The wide-band sensor provides information on the current excess-air factor, and can also be used to maintain richer and leaner mixtures.

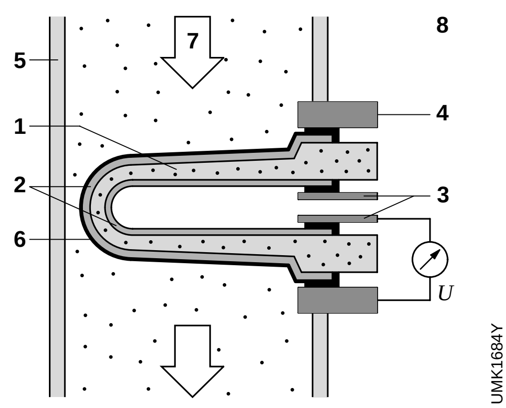
Two-state oxygen (lambda) sensor based on the Nernst concept

The outside of the oxygen sensor's electrode extends into the exhaust stream, while the inside remains in contact with the surrounding air (Figure 6). The essential component of the oxygen sensor is a special ceramic body featuring gas-permeable platinum electrodes on its surface. Sensor operation relies on the porous nature of the ceramic material, which allows oxygen in the air to diffuse (solid electrolyte). This ceramic becomes conductive when heated. Voltage is generated at the electrodes in response to differences in the oxygen levels on the inside and outside of the sensor. A stoichiometric air-fuel ratio of λ = 1 produces a characteristic jump (step function) in the response curve (Figure 7).

Fig. 6

Position of lambda oxygen sensor in exhaust pipe (schematic)

1 Ceramic coating, **2** Electrodes, **3** Contacts, **4** Housing contacts, **5** Exhaust pipe, **6** Ceramic shield (porous), **7** Exhaust gas, **8** Ambient air, *U* Voltage.



The oxygen sensor's voltage and internal resistance are both sensitive to temperature. Reliable control operation becomes possible once exhaust-gas temperatures exceed 350 °C with unheated sensors, or 200 °C with heated units.

Heated oxygen (lambda) sensor

The heated sensor reduces the waiting period between engine start and initiation of effective closed-loop control by providing reliable performance with cooler exhaust gas (e.g., at idle). Because heated sensors warm more quickly, they also have shorter reaction times, reducing lag in closed-loop operation. This type of sensor also offers greater latitude in the selection of installation positions.

A ceramic heater element warms the sensor's active ceramic layer from the center to ensure that the active material remains hot enough for operation, even while exhaust-gas temperatures remain modest.

The heated sensor is protected by a guard tube incorporating a restricted flow opening to prevent its ceramic material from being cooled through exposure to low-temperature exhaust gases.

Wide-band oxygen (Lambda) sensor

While the dual-threshold sensor signals rich and lean mixtures, the wide-band sensor can actually be used to measure the excess-air factor via the exhaust gas. This facility promotes improved dynamic response in the closed-loop control system, regardless of whether the momentary mixture specification is $\lambda = 1$.

The wide-band sensor expands on the principle of the Nernst cell by incorporating a second electrochemical cell, the pump cell. It is through a small slot in this pump cell that the exhaust gas enters the actual monitoring chamber (diffusion gap) in the Nernst cell. Figure 8 shows a schematic diagram of the sensor's design. This configuration contrasts with the layout used in the dual-threshold cell by maintaining a consistently stoichiometric air-fuel ratio in the chamber. Application of pumping voltage to the pump cell results in oxygen discharge when the exhaust gas is lean, and oxygen induction if it is rich. The resulting pumping current is an index of the excess-air factor in the exhaust gas. The basic progression of the curve for pumping current is portrayed in Figure 9. Lean exhaust leads to a positive pumping current, which is needed to maintain a stoichiometric composition in the atmosphere within the diffusion gap,

Fig. 7

Two-state lambda oxygen sensor voltage curve at 600 °C operating temperature

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

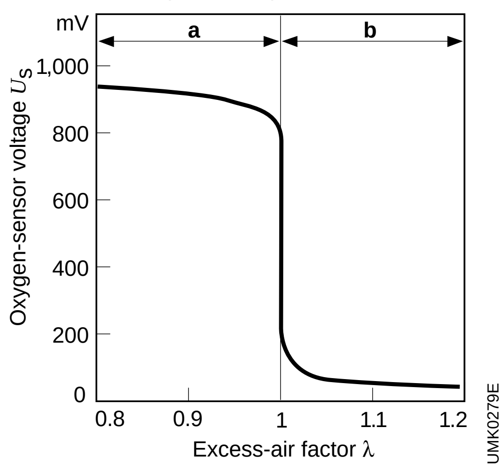
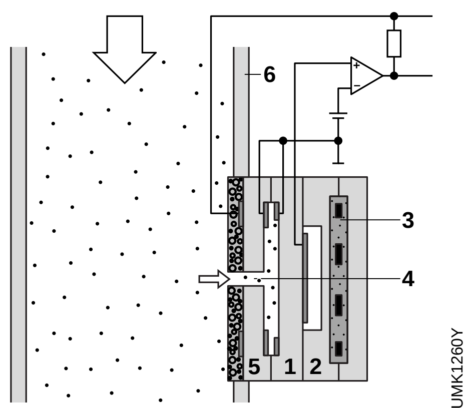


Fig. 8

Schematic portrayal of continuous wide-band lambda oxygen sensor and position of sensor in exhaust pipe

1 Nernst cell, **2** Reference cell, **3** Heater,
4 Diffusion gap, **5** Pump cell, **6** Exhaust pipe.



Pump current I_p in a wide-band lambda oxygen sensor over exhaust-gas air factor

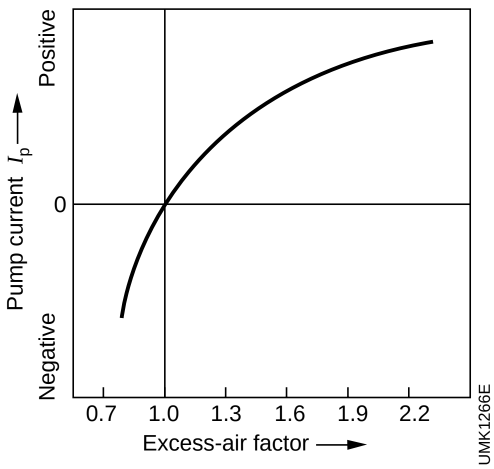


Fig. 9

while rich exhaust is indicated by a negative pumping current.
The wide-band sensor thus differs substantially from its dual-threshold counterpart. While the dual-threshold unit uses the voltage at the Nernst cell as a direct measurement signal, the wide-band sensor employs special processing and control circuitry to adjust the pumping current. The resulting current is then monitored as an index of the exhaust gas' excess-air factor. Because sensor operation is no longer dependent on the step-function response of the Nernst cell, air factor can be monitored as a continuous progression.

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 undesirable combustion phenomenon marks the outer limits of ignition timing advance, and thus, at the same time, defines the boundaries of power-generation potential and efficiency. It occurs when fresh mixture preignites in spontaneous combustion before being reached by the expanding flame front. During an otherwise normally initiated combustion event, the pressure and temperature peaks created by the piston's compressive force generate self-ignition in the end gas (remaining unburned mixture). This process can be accompanied by flame velocities in excess of 2000 m/s, as compared to speeds of roughly 30 m/s for normal combustion. This abrupt combustion leads to substantial pressure rises in the end gas, and the resulting pressure wave continues to propagate until halted by the cylinder walls.
Chronic preignition is characterized by pressure pulses and high thermal stresses acting on the cylinder-head gasket, the piston and around the valves. It can produce mechanical damage in all of these locations.
The characteristic oscillations induced by combustion knock can be monitored by knock sensors for conversion into electrical signals, which can then be transmitted to the Motronic ECU (Figures 10 and 11).
Both the number and installation positions of the knock sensors must be defined with the utmost care. Reliable knock detection must be guaranteed for all cylinders and under all operating conditions, with special emphasis on high engine speeds and load factors. As a general rule, 4-cylinder engines are equipped with one, 5 and 6-cylinder engines with two, 8 and 12-cylinder engines with two or more knock sensors.

Engine and intake-air temperatures

The engine-temperature sensor incorporates a thermally sensitive resistor which extends into the coolant circuit it monitors. Figure 12 shows the design structure of this sensor.
The electrical resistor is characterized by a Negative Temperature Coefficient (NTC), indicating that its electrical resistance is inversely proportional to temperature. Figure 13 shows the basic response curve of resistance over temperature. The NTC resistor forms part of a voltage divider circuit operating on a 5-volt power supply. The voltage from the NTC resistor varies with temperature, and an analog-digital

converter registers these data as an index of thermal conditions in the coolant. Compensation for the non-linear relationship between voltage and temperature is provided by a table stored in the computer's database, which matches each reading with a corresponding temperature. The sensor in the intake tract monitors the temperature of the induction air according to the same concept.

Engine-temperature sensor

1 Electrical terminals, **2** Housing,
3 NTC resistor, **4** Coolant.

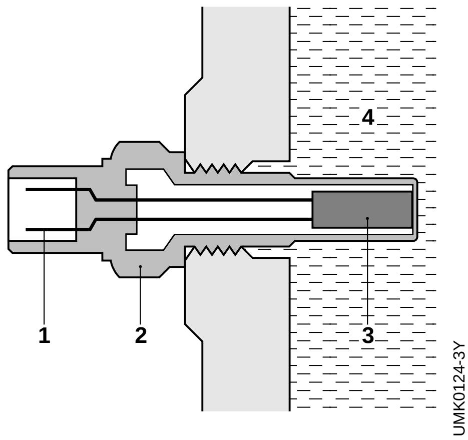


Fig. 12

Fig. 10

Knock sensor

1 Seismic mass, **2** Casting,
3 Piezoelectric ceramic layer, **4** Contact paths,
5 Electrical terminals.

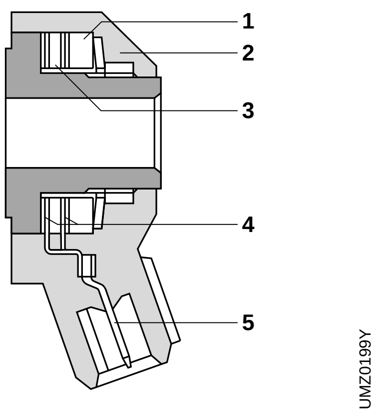


Fig. 13

Temperature sensor (NTC) response curve

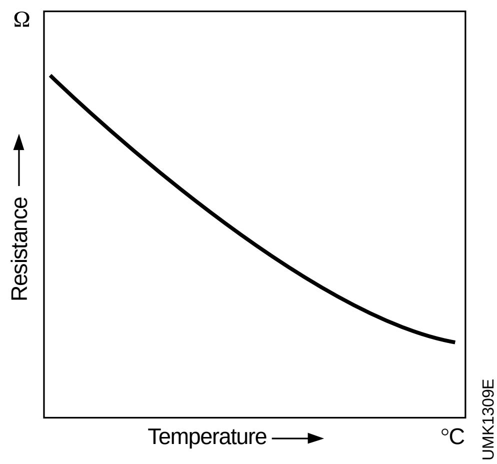
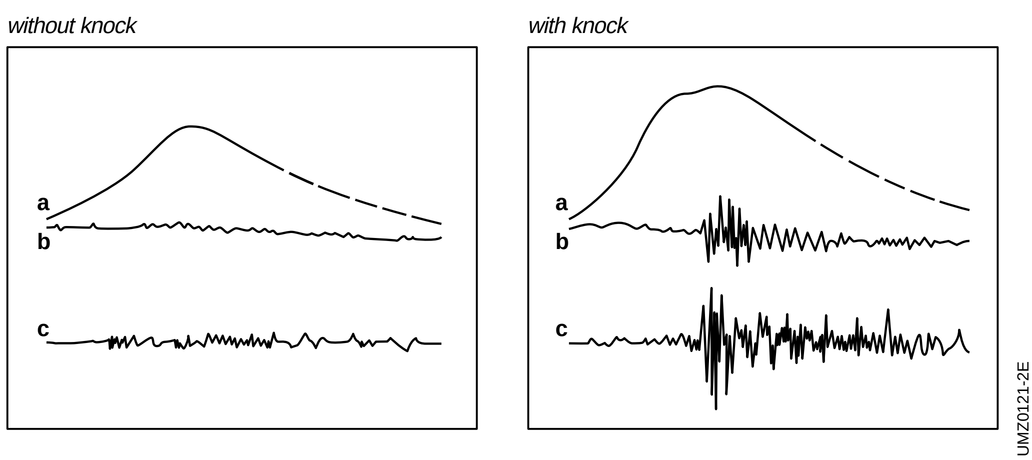


Fig. 11

Knock-sensor signals
The knock sensor transmits signal c which reflects pressure-wave pattern **a** in the cylinder.
The filtered pressure signal is portrayed in **b**.



Operating-data processing

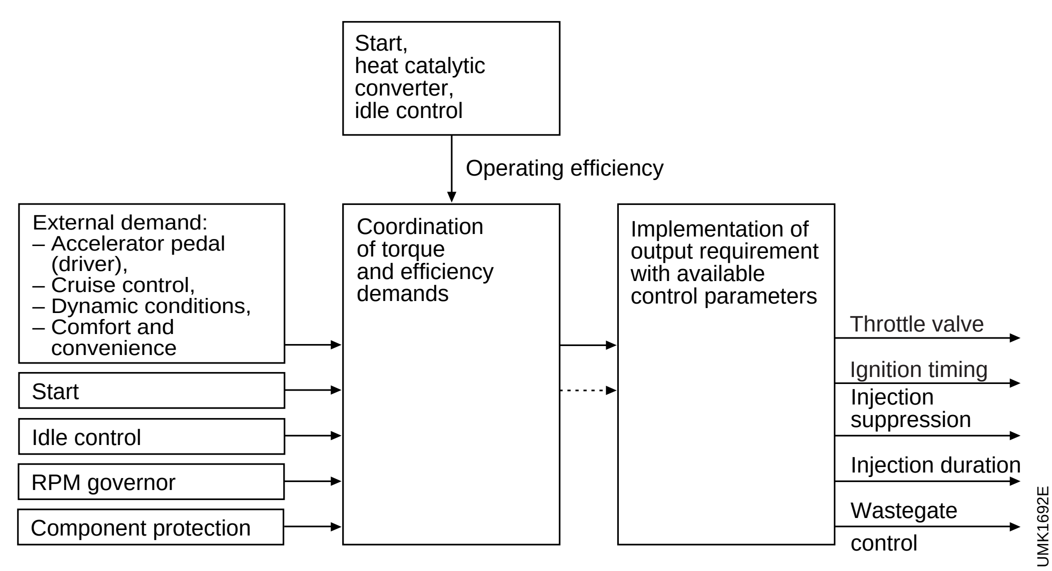
Torque-led control concept

Purpose

The engine-management system's primary assignment is to convert driver demand into engine power and torque. The driver needs this engine power to overcome running resistance during steady-state operation as well as for accelerating the vehicle.
On today's spark-ignition engines this entails calculating the air-charge requirement along with the corresponding injection quantity and optimal ignition timing. Once these parameters have been defined, the system can proceed to active control of the actuators that regulate them (throttle-valve assembly, injectors, ignition coils).
In addition to governing cylinder charge, injection and timing, engine-management systems have also assumed control of a number of auxiliary functions, many of which also consume engine power.
A distinguishing feature of ME-Motronic is its torque-led control concept. Numerous subcomponents within the overall Motronic system (idle control, rpm governor, etc.) join the systems controlling the drivetrain (e.g., TCS, transmission-shift control) and general vehicular functions (such as air-conditioner operation) in relaying requests for adaptation of current engine output to the basic Motronic system. To cite one example, the air-conditioner control system requests an increase in engine output prior to engaging the a/c compressor clutch.
Earlier practice entailed direct implementation of such commands at the control-parameter level (cylinder charge, fuel mass and ignition timing) on an uncoordinated, individual basis. ME-Motronic goes a step further by prioritizing and coordinating individual demands before using the available control parameters to implement the resulting specified torque (Figure 1). This coordinated control strategy makes it possible to obtain optimal emissions and fuel consumption from the engine at every operational coordinate.
An essential element within the torque-based control concept is the ETC electronic accelerator pedal which permits the throttle-valve control mechanism to advance beyond merely reflecting pedal inputs. Formerly the driver used the accelerator pedal to activate a mechanical linkage and determine throttle-valve aperture. The driver

Fig. 1

Torque adjustments in the gasoline engine



exercised direct control over the cylinder charge, and the engine-management system's options for initiating separate adjustments were limited to activation of a bypass routed around the throttle valve.

Calculating specified torque

The basic parameter underlying ME-Motronic's torque-led control concept is the internal torque produced during combustion. This is the physical force produced by gas pressure during the compression and power strokes. The engine's actual net torque production is obtained by subtracting such factors as friction, pumping (gas-transfer) losses and drive power for ancillary equipment (water pump, alternator, etc.) from this internal force.

The ultimate goal of the torque-led control system is to select precisely the engine control parameters needed for accurate response to driver demand while simultaneously compensating for all losses and supplementary requirements. Because Motronic "knows" the optimal specifications for charge density, injection duration and ignition timing for any desired torque, it can consistently maintain optimal emissions and fuel economy.

Adjustment of actual torque

ME-Motronic's torque coordinator has two potential control paths to choose from when regulating internal torque generation (Figure 2). One path, furnishing gradual reaction, is controlled by triggering the throttle valve (ETC), while the rapid-response path relies on manipulation of ignition timing and/or deactivating the injection at individual cylinders. The slower path, also known as the charge-control path, is responsible for static operation. The charge requirement calculated for a given torque generation determines the cylinder charge, which is then provided by the throttle valve. The rapid-response (ignition timing) path can react very quickly to dynamic variations in torque generation.

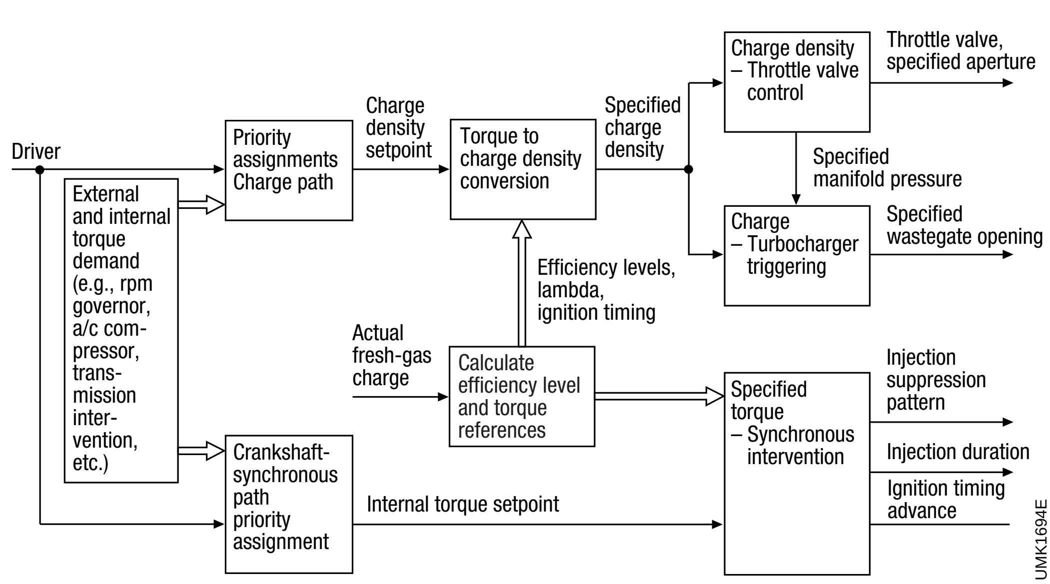
Calculating cylinder charge

The air mass within the cylinder following closure of the intake valve is the air charge. There is also a "relative (air) charge" which is independent of piston displacement. It is defined as the ratio of the current charge to a charge obtained under specified standard conditions (p_0 = 1,013 hPa, T_0 = 273 K).

The relative charge must be known in order to calculate injected fuel quantity.

Fig. 2

Coordinating output with charge path and crankshaft-synchronous paths (ignition timing path)



On today's spark-ignition engines it is also the primary parameter for influencing engine output, which is why it is incorporated in the torque structure as a control parameter. Because no means for directly monitoring charge density is available, it must be calculated from available sensor signals with the aid of a simulation model. The requirements for the charge model are:
– Precise determination of charge density under all operating conditions (dynamic, selective-flow intake manifold, variable valve timing, etc.),
– Accurate response to exhaust-gas components in systems with variable-rate EGR (controlled external or internal EGR),
– Calculation of the control command parameter for "throttle-valve aperture" corresponding to any given charge-density requirement.

Intake-manifold simulation model

The actual mass of the air within the cylinder is relevant for fuel metering and torque calculations. In the absence of a method to directly monitor cylinder charge, it is calculated using an intake-manifold simulation model. Depending on the system's induction-charge sensor (air-mass meter or intake-manifold pressure sensor), the raw data for this model is either monitored directly or simulated.

Induction air mass

Here, the decisive parameter is the mass of the incoming air. While charge density can be calculated directly from induction air mass during static engine operation, abrupt variations in throttle-valve aperture result in a time lag. This stems from the fact that (for example) the first response to an opening throttle valve is for the intake manifold to fill with air. A disparity exists between the air mass actually entering the combustion chamber and the air monitored by a sensor such as the hot-wire air-mass meter (HFM) for the duration of this lag period. It is only after pressure levels in the manifold start to rise that more air can flow into the combustion chamber.

Intake-manifold pressure

These considerations elevate manifold pressure to the status of a primary factor; the relationship between the relative cylinder charge – the relevant factor – and the intake-manifold pressure can be portrayed using a linear equation (Figure 3).

The linear equation's offset is defined by the partial pressure emanating from internal residual gases, making it a function of valve overlap, rpm and ambient barometric pressure, while the gradient is determined by engine speed, valve overlap and combustion-chamber temperature.

Other flow into the manifold

A supplementary mass flow, over and above the air entering through the throttle valve, results from activation of such systems as the evaporative-emissions control. The regeneration flow required by this system can be varied with the aid of a tank vent valve (purge valve). With manifold pressure as a known quantity, it is possible to calculate the regeneration flow for use in the intake-manifold model simulation process.

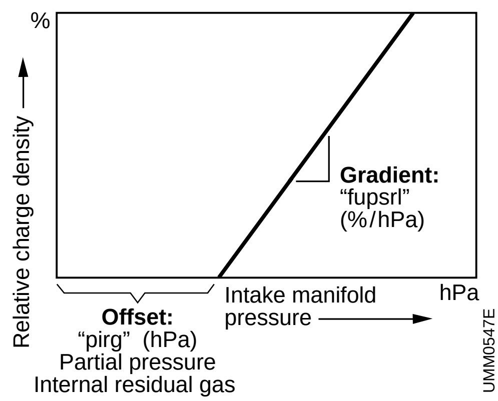
Monitoring charge density with the HFM

When an HFM is fitted this directly measures the air mass entering the intake manifold. This process entails

Fig. 3

Relationship between manifold pressure and relative charge density

Note: Each parameter set ("pirg", "fupsrl") is valid for a single rpm, one valve overlap and one manifold geometry.



multiplying the mean mass airflow monitored during an intake stroke (segment) by the intake stroke's duration for conversion into a relative charge density. The other parameters required for intake-manifold model simulation (such as intake-air temperature) are either monitored directly or calculated in the modeling process (in this case intake-manifold pressure, but also secondary parameters such as combustion-chamber temperature).

Monitoring charge density with a manifold-pressure sensor

If an intake-manifold pressure sensor is present to act as a "primary charge-density sensor" then manifold pressure can be monitored directly. The system calculates the mass of the air entering the intake manifold based on manifold pressure.

Cylinder charge control

The density of the charge entering the cylinder is also controlled with the intake-manifold model, exploiting the fact that gas flow through valves (in this case the throttle valve) can be formulated as an equation. The main factors are the entry pressure immediately in front of the throttle valve, the pressure drop, the temperature and the effective aperture; all parameters calculated with the intake-manifold model. Other parameters relevant for specific throttle valves (such as friction losses in the air current) must be quantified using test-stand measurements.

Now the intake-manifold model can be "turned around" to calculate a throttle-valve aperture from the desired cylinder charge density (which has been calculated by ME-Motronic's torque-led control facility). This aperture is transmitted to the throttle-valve actuator's position controller as a command value.

Calculating injection timing

Calculating injection duration

Cylinder-charge density can be used as the basis for calculating the fuel mass required to obtain a stoichiometric air-fuel ratio. The injector constant, which varies according to injector design, can then be incorporated into the calculations to produce the injection duration.

Injection duration is also affected by the differential between the fuel's supply pressure and injection counterpressure. The standard fuel supply pressure is 300 kPa (3 bar). This pressure can be maintained using any of a variety of reference sources. Fuel-supply systems with return lines maintain constant supply pressures relative to manifold pressure. This strategy ensures that the pressure differential across the injectors remains constant in the face of changing manifold pressures, so that roughly consistent flow rates result. Returnless fuel systems rely on a different concept, maintaining their 300 kPa supply pressure relative to ambient pressure. Fluctuations in the pressure within the intake manifold produce variations in the differential between its own pressure and that of the fuel supply. A compensation function corrects this potential error source.

As the injectors open and close they induce pressure waves in the fuel-supply system. This leads to flow-rate inconsistencies when the injector is opened. An adaptation factor correlated with engine speed and injection duration is used to compensate.

The opening duration calculated up to this point will be valid if we assume that the injector has already opened and is discharging fuel at a constant flow rate, but the injector's opening time must also be considered in real-world operation. This opening duration displays significant variations depending on the voltage being supplied by the battery. There may be substantial lag before the valve opens completely, especially in the starting phase or when the battery is partially discharged. A supplementary injection duration based on battery voltage is added to the base duration to compensate for this effect.

Excessively short injection durations would lend disproportionate influence to the valve opening and closing times. This is why a minimum injection duration is defined to guarantee precise fuel metering. This minimal duration is less than the injection period required for minimum potential cylinder charging.

Injection timing

Optimal combustion depends on correct injection timing as well as precise metering. The fuel is usually injected into the intake manifold while the intake valve is still closed. Termination of the injection period is defined by something known as the injection advance, which is indicated in crankshaft degrees, and uses intake-valve closure as a reference. The injection duration can then be correlated with engine speed to obtain a point for initiating injection defined as an angle. Current operating conditions are also reflected in the calculations to define the injection advance angle.

ME-Motronic triggers an individual injector for each cylinder, making it possible to preposition a separate fuel charge for each cylinder (sequential injection). This option is not available with systems that rely on only one injection valve (single-point injection) or simultaneous activation of several injectors at once (group injection).

Calculating the ignition angle

The "reference ignition angle" is calculated based on the engine's current steady-state operating status. Its essential determinants are instantaneous cylinder charge, engine speed, and mixture composition (as indicated by the excess-air factor λ). The ignition angle is corrected to compensate for the particular operating conditions encountered during starting and in the warm-up phase. A simplified representation of the "reference ignition angle" in ME-Motronic would define it as the earliest potential ignition angle under any given operating conditions. Under standard operating conditions, with the engine warmed to its normal running temperature, this angle is defined by a minimum interval separating it from the knock threshold.

This reference ignition angle can then be further retarded by the knock control (to avoid combustion knock) and the crankshaft-synchronized torque-guidance output (to reduce torque).

The reference ignition angle is combined with the correction factors listed above to produce the so-called "basic ignition angle".

The actual ignition angle specified by the system reflects the addition of a supplementary correction factor designed to compensate for phase error in the engine-speed sensor.

Calculating the dwell angle

The purpose of the ignition system is to supply enough energy to ensure complete combustion of the air-fuel mixture at precisely the right instant. Energy availability is essentially defined by the dwell period for charging the primary circuit; the end of this period usually coincides with the firing point. The ECU specifies a dwell angle corresponding to the ignition coil's charge requirement. It activates the coil's primary current at the start of the dwell period and then interrupts it to initiate ignition at the firing point. This is how ME-Motronic controls distributorless ignition systems (DLI).

The system refers to a program map to determine the dwell angles for specific engine speeds and battery voltages, while its final output also includes a temperature correction.

The start of the dwell period is defined by the difference between the end of dwell and the dwell angle. The dwell angle is calculated from the dwell period using a time/angle conversion equation. The end of the dwell period is defined to coincide with the firing point (ignition timing).

The system basically has two options for defining the start and the end of the dwell period:

– As an angle,
– As a period of time.

When defined as an angle, the segment time is used to convert the dwell period into an angle. During dynamic variations in engine speed this produces a timing error, as the segment times used for calculating angle position are already outdated. Positive dynamic engine-speed changes (acceleration) lead to attenuated dwell periods, while negative dynamics (deceleration) produce extended dwell angles. Compensation for the dwell period reductions that accompany acceleration is provided by an injection advance, which must always be added to the basic duration. This dynamic injection advance declines as engine speed increases. In contrast, pronounced dynamic changes at low rpm can retard the dwell timing to such an extent that the dwell period becomes too brief for recharging the coil. The response is to transmit the dwell period termination point as a time function at low rpm. This ensures generation of adequate ignition energy regardless of dynamic fluctuations.

Operating conditions

The various engine operating conditions are primarily distinguished by variations in torque generation and engine speed. Figure 1 shows the different ranges.
Conditions characterized by high rates of dynamic change in rpm and load facor are especially significant, as they place special demands on the mixture-formation system (i.e., condensation and vaporization of the fuel film on the manifold walls). Other important elements are starting phase, and the subsequent transition phase that continues until engine and exhaust system warm to their normal operating temperatures.

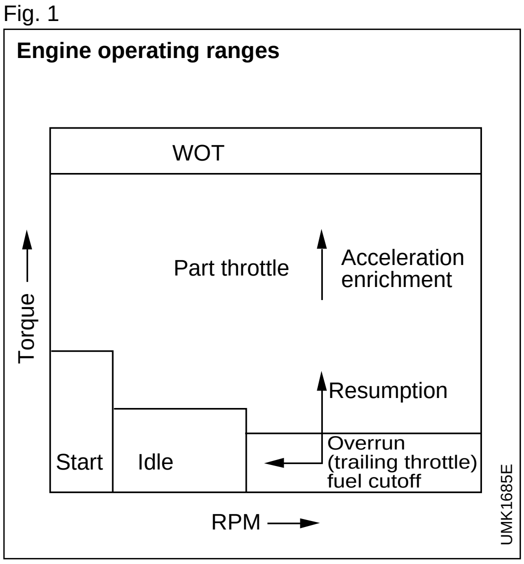
Starting

Special calculations are employed to regulate charge control, injection and ignition timing for the duration of the starting process. At the outset of this process the air within the intake manifold is stationary, and the manifold's internal pressure reflects that of the surrounding atmosphere. This rules out using the manifold simulation model for throttle-valve control. Instead, throttle-valve position is specified as a fixed parameter based on starting temperature.
In an analogous process special "injection timing" is specified for the initial injection pulses.
The injected fuel quantity is augmented in accordance with engine temperature to promote formation of a fuel film on the walls of the intake manifold and the cylinders, thereby compensating for the engine's higher fuel requirement as it runs up to speed.
The system immediately starts to fade out the start-enrichment as soon as the engine turns over, until it is completely cancelled upon termination of the starting phase (600...700 min^{-1}).
The ignition angle is also adapted to the starting process. It is adjusted as a function of engine temperature, intake-air temperature, and engine speed.

Cylinder recognition

Before the first ignition spark can be generated, the system must reliably identify which cylinder is currently on its compression stroke. Ignition during the intake stroke could initiate backfiring through the manifold, leading to component damage.

Fig. 1



Signal patterns for ignition, crankshaft and camshaft on a 6-cylinder engine with standard sensor rotor

a Ignition-coil secondary voltage, **b** Crankshaft rpm-sensor signal, **c** Hall-sensor signal (standard sensor rotor) at camshaft.
1 Close, **2** Ignition.
A Ignition at Cylinder No. 1, **B** Ignition at Cylinder No. 5, **C** Ignition at Cylinder No. 3, **D** Ignition at Cylinder No. 6, **E** Ignition at Cylinder No. 4.

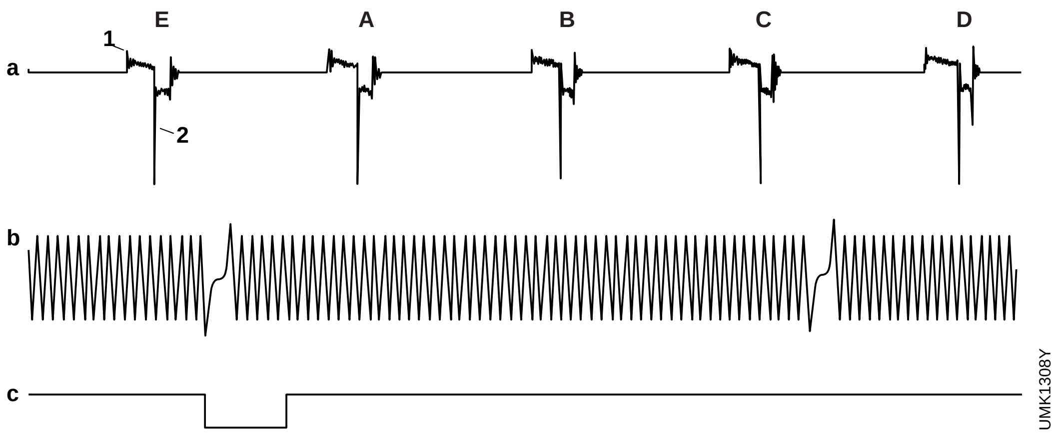


Fig. 2

The system determines cylinder phases by correlating the respective signals from crankshaft and camshaft sensors. The camshaft sensor rotor features at least one segment. The signal progression for one camshaft rotation is shown in Figure 2, which also shows the signal pattern generated by the crankshaft sensor as it scans through all 58 rotor teeth on each rotation. When the gap in the signal from this sensor (Figure 2, Curve b) coincides with the depression in curve c, this indicates that cylinder no. 1 is on its compression stroke, meaning that it is next in line to fire (Ignition A in Figure 2). The first ignition spark in the start phase cannot be triggered until the level of the camshaft signal at the first gap in the crankshaft-sensor signal has been evaluated; this is how the system determines which cylinder is next in the firing order (Figure 2: Cyl. 1 or 6).

Fig. 3

Rapid-start sensor rotor

Rapid start

The “rapid-start” mode reduces the time that elapses between starter engagement and cylinder recognition. This shortens overall starting times for enhanced convenience while also reducing the loads placed on starter and battery.

The rapid start relies on equipment such as a special “rapid-start” sensor rotor mounted on the camshaft (Figure 3). This sensor rotor generates a unique flank profile (Figure 4), enabling the system to recognize cylinder phases – and thus transmit the first ignition spark – before the first gap in the crankshaft signal is reached.

RPM and phase-sensor signal (camshaft signal) from rapid-start sensor rotor on 4-cylinder engine

a Filtered rpm-sensor signal,
1...4 Initiation of the calculations for Cylinders 1 through 4.
2...30 Pulses in one half of a crankshaft rotation.
b Rapid-start sensor-rotor signal with four equidistant negative phase flanks.
A Equidistant to next negative phase flank.
The pulse sequence indicates the engine's current rotation cycle.

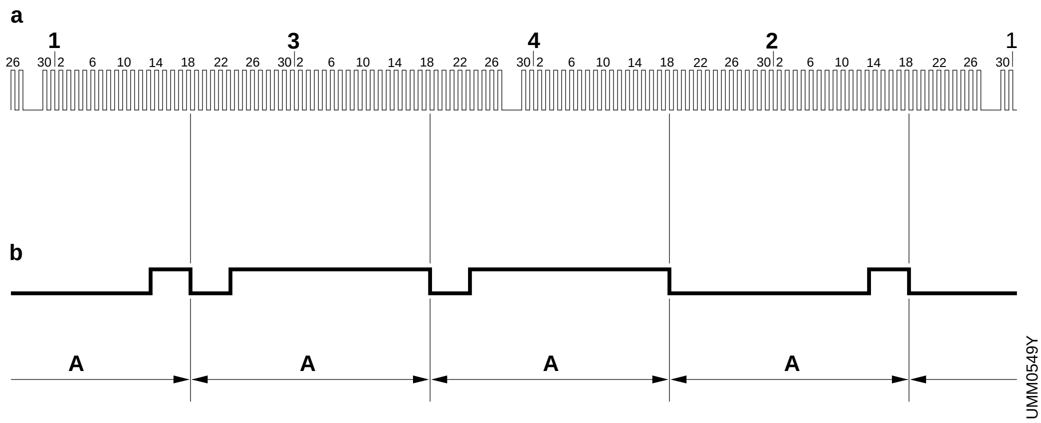


Fig. 4

Post-start phase

The post-start phase (immediately following termination of the starting phase) is marked by further reductions in the charge densities and injected-fuel quantities employed for starting. System response in this phase is defined by the rise in engine temperature and the period that has elapsed since the starting phase ended.
Ignition angles are also adjusted to correspond to the revised injected-fuel quantities and different operating status. The post-start phase trails off in a smooth transition to the warm-up phase.

Warm-up and catalytic-converter heating

After starting at low engine temperatures, cylinder charge, injection and ignition are all adjusted to compensate for the engine's greater torque requirements; this process continues up to a suitable temperature threshold.
The prime concern in this phase is rapid warming of the catalytic converter, as quick transition to catalytic-converter operation permits drastic reductions in exhaust emissions. The strategy employs a portion of the exhaust gas for "cat-converter heating" during this phase, while accepting the resulting sacrifices in engine efficiency.
There are basically two concepts:
– Secondary air injection into a rich mixture with retarded ignition timing, and
– Lean warm-up with extremely retarded (late) ignition timing.

Both concepts entail using retarded ignition timing to operate the engine at a low level of efficiency. The initial results are higher exhaust-gas temperatures and reduced torque generation. The torque-based control automatically compensates for this loss by prescribing higher cylinder-charge densities. This produces a larger quantity of hot exhaust gas for use in heating the catalytic converter with minimal delay. The catalytic converter's rapid warm-up and the consequent early onset of operation furnish a substantial reduction in exhaust emissions.

<u>Lean warm-up</u>
The combination of lean warm-up with the extremely retarded ignition point leads to the post-oxidization of the

Effects of secondary-air injection on CO and HC emissions (6-cylinder engine, 2.8 litre, 145 kW).
1 Without secondary-air injection, **2** With secondary-air injection.
v Vehicle velocity.

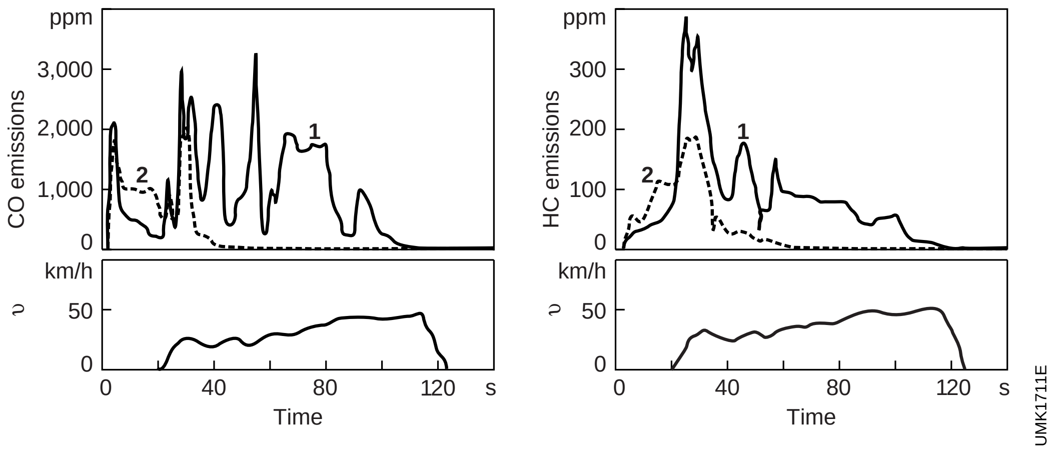


Fig. 5

unburnt hydrocarbons which result from inefficient combustion.
The term “lean warm-up” stems from the use of a slightly lean base mixture to supply the oxygen required to support this oxidation process.
Although this concept’s advantage is the freedom to dispense with supplementary components, limits on its potential for heat generation mean that the catalytic converter must be installed close to the engine to minimize thermal losses.

Secondary-air injection
This concept expands on the low-efficiency strategy by operating the engine on high levels of excess fuel (λ < 0.6) to increase the carbon monoxide (CO) and hydrocarbon (HC) content of the exhaust gas. Fresh (“secondary” air which is not involved in the internal combustion process) is then injected directly downstream of the exhaust valves to support oxidation of CO and HC. This produces heat energy, which then flows to the catalytic converter, enabling it to reach operating temperature with minimal delay.
An electric vacuum pump draws the required secondary air from within the air-filter housing or through a special coarse filter. Injection into the exhaust system is then regulated by a deactivation valve and a check valve designed to prevent hot exhaust gases from flowing back into the secondary-air injection system. ME-Motronic triggers the secondary-air pump and air valve at the indicated intervals. A wide-band Lambda oxygen sensor facilitates precise diagnosis of the secondary-air pump.
This process produces enough heat for use with catalytic converters situated further away from the engine. Figure 5 shows the respective curves for hydrocarbon and carbon-monoxide emissions in the initial seconds of the emissions test with and without secondary-air injection.

Idle

The engine generates no torque at idle; the power generated in the combustion process being needed to sustain engine operation and to drive the ancillary devices. Under these conditions, the torque that the engine needs to remain in operation combines with the idle speed to define fuel consumption. Because a substantial portion of the fuel consumed by vehicles in heavy stop-and-go traffic is actually burned in this kind of use, it pays to hold friction losses during idling at the lowest possible level. This translates into specifying low idle speeds.

ME-Motronic's closed-loop idle control reliably maintains a stable idle at the defined level regardless of variations in operating conditions. These variations can stem from from factors such as fluctuating current draw in the electrical system, air-conditioner compressors, gear engagement on automatic-transmission vehicles, active power steering, etc.

WOT (full load)

At Wide-Open Throttle (WOT) there are no throttling losses, and the engine produces the maximum potential power available at any given rpm.

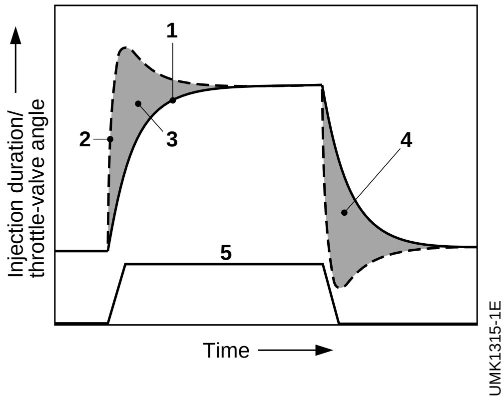
Transition response

Acceleration/deceleration
A portion of the fuel discharged into the intake manifold does not reach the cylinder in time for the subsequent combustion process. Instead, it forms a condensation layer along the walls of the intake manifold. The actual quantity of fuel stored in this film rises radically in response to higher load factors and extended injection durations.
A portion of the fuel injected when the throttle valve opens is absorbed for this film. As a result, a corresponding quantity of supplementary fuel must be injected to compensate and prevent the mixture from going lean under acceleration. Because the additional fuel retained in the wall film is released again once the load factor drops, injection durations must also be reduced by a corresponding increment during deceleration.
Figure 6 shows the corresponding curves for injection duration.

Fig. 6

Transitional injection duration

1 Injection signal from charge-density calculations, **2** Corrected injection duration, **3** Supplementary injected fuel quantity, **4** Reduction of injected fuel quantity, **5** Throttle valve angle α_{TV}.



Overrun fuel cutoff/renewed fuel flow
Overrun, or trailing throttle, indicates a condition in which the power being provided by the engine at the flywheel is negative. Under these conditions the engine's friction and gas-flow losses can be exploited to slow the vehicle. The engine can continue to run with or without active fuel injection.
For passive, injectionless trailing-throttle operation the injection is deactivated to reduce fuel consumption and exhaust emissions. ME-Motronic's torque-based control can regulate suppression of the fuel-injection pulses to prevent radical torque jumps during the transition to trailing throttle by relying on gradual instead of abrupt reductions in specified output.
Injection resumes once rpm falls to a specified reactivation speed located at a point above idle. Actually, the ECU is programmed with a range of reactivation speeds. These vary to reflect changes in parameters such as engine temperature and dynamic variations in engine speed, and are calculated to prevent the rpm from falling below the defined minimum threshold.
Once injection resumes, the system starts by using the initial injection pulses to discharge supplementary fuel and rebuild the wall fuel layer. When fuel injection is resumed, the slow, controlled increase of engine torque by the torque-based control ensures that torque buildup is smooth (gentle transition).

Closed-loop idle-speed control

Purpose

The engine does not furnish torque at the flywheel during idling. To ensure consistent idling at the lowest possible level, the closed-loop idle-speed control system must maintain a balance between torque generation and the engine's "power consumption."
Power generation is needed at idle in order to satisfy load requirements from a number of quarters. These include internal friction at the engine's crankshaft and valve-train assemblies, as well as such ancillary equipment as the water pump.
The engine's internal friction losses are subject to substantial variation in response to temperature fluctuations, while friction also changes, albeit at a much slower rate, over the course of the engine's service life.
The load imposed by external factors (such as the a/c compressor) also fluctuates through a wide range as ancillaries are switched on and off. Modern engines are especially sensitive to these variations, owing to their lower reciprocating and flywheel masses as well as higher intake-manifold (storage) volumes.

Operating concept

ME-Motronic's torque-based concept relies on closed-loop idle-speed control to quantify the output needed to maintain the desired idle speed under any operating conditions. This output rises as engine speed decreases, and drops as it increases.
The system responds to recognition of new "interference factors" such as activation of the a/c compressor or engagement of a drive range in an automatic transmission by requesting more torque. Torque demand must also be increased at low engine temperatures to compensate for higher internal friction losses and/or maintain a higher idle speed.
The sum of all these output demands is relayed to the torque coordinator, which then proceeds to calculate the corresponding charge density, mixture composition and ignition timing.

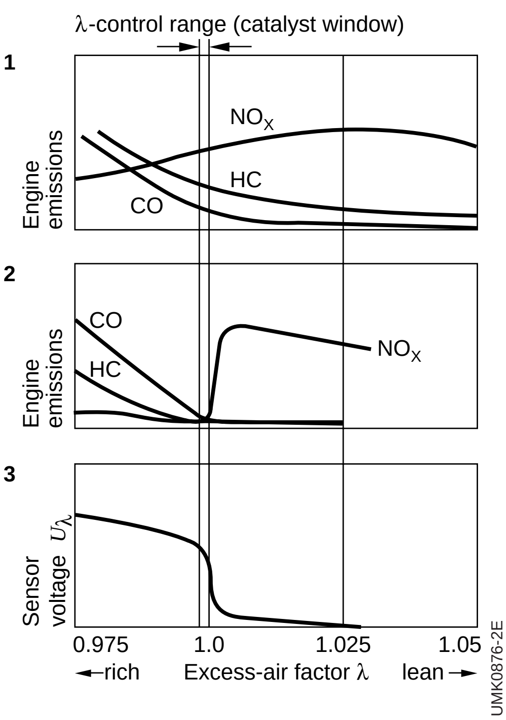
Lambda closed-loop control

Post-treatment of exhaust gases in a 3-way catalytic converter represents an effective means of reducing concentrations of harmful exhaust pollutants. The converter can reduce hydrocarbons (HC), carbon monoxide (CO) and oxides of nitrogen (NO_X) by 98 % and more (Figure 1), converting them to water (H_2O), carbon dioxide (CO_2) and nitrogen (N_2). This level of efficiency is contingent on engine operation within a very narrow scatter range surrounding the stoichiometric air-fuel ratio of $\lambda = 1$.

Fig. 1

Catalytic efficiency and lambda-sensor voltage relative to excess-air factor λ

1 Without catalytic post-combustion treatment,
2 With catalytic post-combustion treatment,
3 Voltage curve with 2-threshold λ sensor.



Two-state lambda closed-loop control

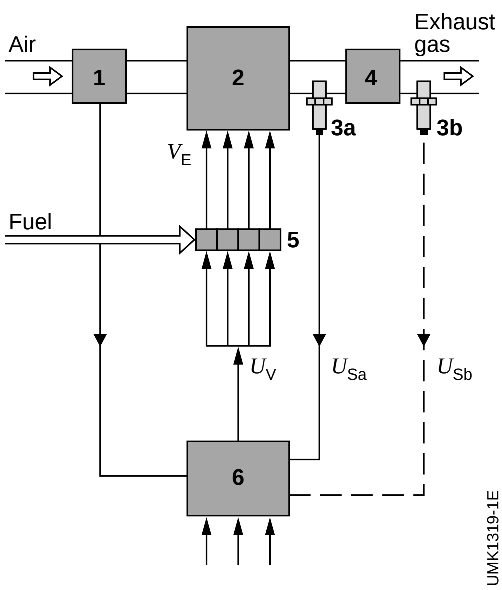
Control range

The “lambda window,” corresponding to the range available for effective simultaneous processing of all three “classical” exhaust-gas components, is extremely restricted. Closed-loop lambda control is needed to maintain operation within this window (λ = 0.99...1).
The two-state oxygen sensor monitors the exhaust stream’s oxygen content from a position on the engine side of the catalytic converter. Lean mixtures (λ > 1) induce sensor voltages of approx. 100 mV, while rich mixtures (λ < 1) generate roughly 800 mV. At λ = 1 the sensor voltage suddenly jumps from one level to the other.
ME-Motronic includes this signal from the oxygen sensor in its calculations of injection duration. Figure 2 is a schematic portrayal of the circuit’s configuration.

Fig. 2

Lambda closed-loop control: Operational schematic

1 Air-mass meter,
2 Engine,
3a Lambda oxygen sensor 1,
3b Lambda oxygen sensor 2 (with two-state control only),
4 Catalytic converter,
5 Injectors,
6 ECU.
U_S Sensor voltage,
U_V Injector control voltage.



Operation

A closed-loop lambda control system can only function in tandem with a fully operational oxygen sensor. An auxiliary processing circuit monitors the sensor on a continuing basis.
A cold oxygen sensor or damaged circuitry (short or open circuits) will lead to implausible voltage signals, which the ECU will reject. Depending upon individual configuration and installation position, the heated oxygen sensors found today in most systems can assume operation after only 15 to 30 seconds.
Cold engines require a richer mixture (λ < 1) to run smoothly. This is why the closed-loop lambda control circuit is only released for active intervention once a defined temperature threshold has been reached.
Once the lambda control assumes operation, the ECU uses a comparator to convert the sensor signal into binary form.
The lambda closed-loop control reacts to incoming signals (λ > 1 = mixture too lean, or λ < 1 = mixture too rich) by modifying the control variables, generating a control factor for use as a multiplication factor when modifying the injection duration.
Injection duration is adjusted (lengthened or reduced) and the control factor reacts by settling into a state of constant oscillation (Figure 3).
Continuing oscillation in a range with λ = 1 as its focal point is the only way to achieve optimal lambda control with a dual-state system. The precision of the closed-loop control process depends upon the speed with which the control system can adjust the control factor to counteract shifts in the excess-air factor. While waiting fuel is constantly being discharged into the combustion chamber, the O_2 sensor is located elsewhere, further back in the exhaust system. The resulting gas-transit times translate into response lag within the control circuit, with the actual delay depending on the

engine's load factor and speed. The ultimate reaction to mixture adjustments can only be measured once the lag period has elapsed. This leads to a minimum phase duration (periodicity) for the cyclical revisions in the control factor. Processing times and the sensor's response delay increase lag even further. The duration of the oscillation periods is determined by the transit times of the gas, while the ramp climb maintains largely constant amplitudes throughout the engine's load and speed range, despite variations in gas-transit times.
Radical steps in control factor during mixture adjustments (sensor jump) accelerate the reaction process, making it possible to shorten the oscillation period.

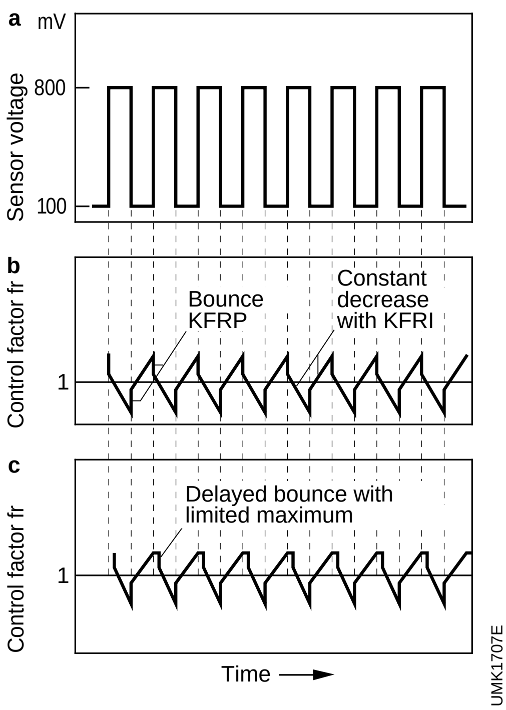
Lambda shift

Because the sensor's response pattern varies depending upon the direction of the monitored mixture transition (viz., rich

Fig. 3

Dynamic two-state lambda control with typical oscillation patterns

a Signal from two-state sensor (simplified),
b Oscillation pattern of control factor,
c Switching lag with limited maximum.



to lean, or lean to rich), a symmetrical control arrangement would produce the slightly lean exhaust mixture portrayed in Figure 3b. Because catalytic converter efficiency is optimal in the λ = 0.99...1.0 range, the control system must be able to counteract this tendency. An asymmetrical controller oscillation pattern can shift the mixture into the optimal conversion range.
The required asymmetry is obtained either by delaying the switch-over of the control factor after the voltage jump (from lean to rich) at the oxygen sensor, or with an asymmetrical step function. Maxima are limited to maintain the controller's dynamic response.

Adapting the Lambda pilot control to the Lambda closed-loop control

The lambda closed-loop control system corrects each consecutive injection event in the sequence based on previous monitoring data from the O_2 sensor.
As a result, a certain time shift arising from gas-transit times is unavoidable, and the approach to new operating points defined with maladjusted pilot control is characterized by deviations from λ = 1. This condition continues until the system's cyclical control can reestablish equilibrium.
Thus a special default (or reference) control mechanism is needed to maintain compliance with emissions limits. The pilot control is programmed when the system is adapted to the engine, and a corresponding lambda control map is stored in a ROM (program memory). However, subsequent revisions in the default control may be needed to compensate for the effects of drift factors during the vehicle's service life, including variations in the density and quality of the fuel.
If the lambda controller starts to consistently implement a single set of corrections during operation in a particular engine speed and load range, the pilot control's adaptation function will register this fact and respond by programming corresponding corrections into the system's non-volatile memory

(RAM or EEPROM with constant current supply). The corrected pilot control is then ready for immediate implementation at the next start, assuming duty until the lambda closed-loop control becomes active.
Interruptions in the power supply to the non-volatile memory are also registered; adaptation then recommences using neutral pilot-control values as a starting point.

Dual-sensor lambda closed-loop control
Installing the oxygen sensor at the back end of the catalytic converter ("cat-back" position) helps guard it against contaminants in the exhaust gas while also reducing the thermal stresses imposed on it. This type of auxiliary sensor can generate a second, overlapping control signal to augment the one from the main, ("cat-forward") sensor on the engine side and ensure stable air-fuel mixture composition over an extended period.
The superimposed control system modifies the asymmetry of the constant oscillation pattern that characterizes control mechanisms based solely on a cat-forward oxygen sensor; thus compensating for the lambda shift.
A lambda control strategy based exclusively on a sensor mounted behind the converter ("cat-back") would be handicapped by excessive control lag produced by extended gas-transit times.
While helping maintain the lambda control system's long-term operational stability, the second, "cat-back" sensor also can be employed as a tool for assessing the catalytic converter's effectiveness.

Continuous lambda closed-loop control

While the two-state sensor can only indicate two states – rich and lean – with a corresponding voltage jump, the wide-band sensor monitors deviations from $\lambda = 1$ by transmitting a continuous signal. In other words, this wide-band sensor makes it possible to implement lambda control strategies based on continuous instead of dual-state information.
The advantages are:
– A substantial improvement in dynamic response, with quantified data on deviations from the specified gas composition, and
– The option of adjusting to any values, i.e., also air factors other than $\lambda = 1$.
The second option is especially significant for strategies seeking to exploit the fuel-savings potential of lean operation (lean-burn concepts).

Evaporative-emissions control system

Source of fuel vapors
The fuel in the tank is warmed by:
– Heat radiated from external sources, and
– Excess fuel from the system return line, which is heated during its passage through the engine compartment.
This results in HC emissions which primarily emerge from the fuel tank in the form of vapor.

Limiting HC emissions
Evaporative emissions are limited by legal mandate. Limitation is by means of evaporative-emissions control systems equipped with an activated-charcoal filter (carbon canister) installed at the end of the tank's vent line. The activated charcoal in the canister binds the fuel vapors, allowing only air to escape into the atmosphere, while simultaneously providing the pressure-relief function. To support ongoing regeneration in the charcoal filter, an additional line leads from the canister to the intake manifold.
Vacuum, produced in the intake manifold whenever the engine is running, draws in a current of atmospheric air; this air flows through the charcoal on its way to the manifold. The air stream absorbs the fuel vapors stored in the activated charcoal and

Evaporative-emissions control system

1 Line from fuel tank to activated-charcoal canister, **2** Activated-charcoal canister, **3** Fresh air, **4** Canister-purge valve, **5** Line to intake manifold, **6** Throttle assembly with throttle valve. Δp Difference between manifold pressure p_s and ambient barometric pressure p_u.

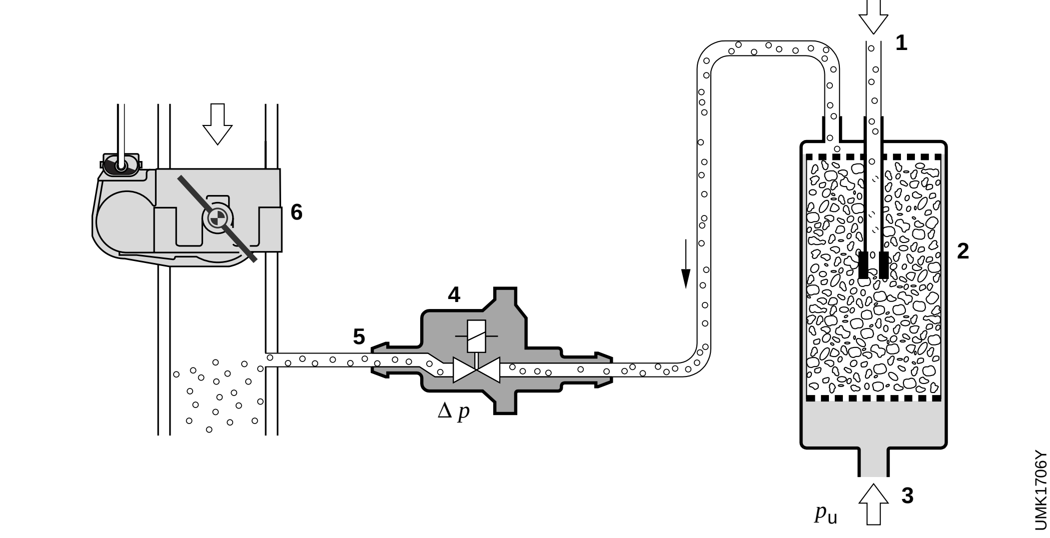


Fig. 1

takes them to the engine for combustion. A purge-valve installed in the line to the manifold meters this regenerative or "cleansing" flow (Figure 1).

Regeneration flow

The regenerative stream is an air-fuel mixture of necessarily unknown composition, as it can contain fresh air as well as substantial concentrations of gasoline extracted from the activated charcoal.

The regenerative flow thus represents a major interference factor for the lambda closed-loop control system. A regenerative flow representing 1% of the intake stream and consisting solely of fresh air will lean out the overall intake mixture by 1%. A flow with a substantial gasoline component on the other hand can enrichen the mixture by something in the order of 30%, owing 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 ensures adequate ventilation of the carbon canister while holding lambda deviations to a minimum (Figure 2).

ECU control functions

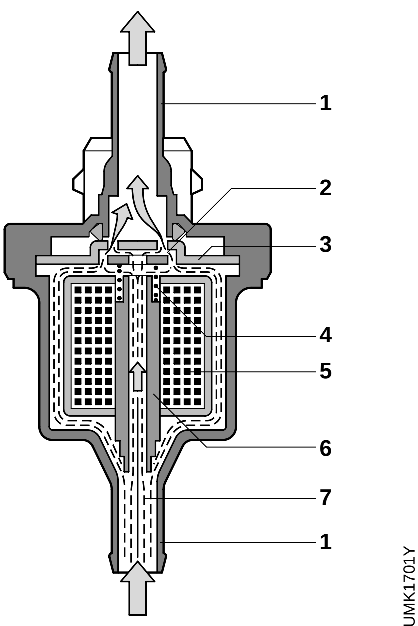
The canister-purge valve closes at regular intervals in order to allow the mixture adaptation process to proceed without interference from the tank's ventilation flow.

During active regeneration the system selects the optimal purge quantity for

Fig. 2

Canister-purge valve

1 Hose fitting, **2** Seal flange, **3** Armature, **4** Spring, **5** Solenoid winding, **6** Solenoid core with flow channel, **7** Flow path.



instantaneous operating conditions, with the ECU generating signals to open the valve in a ramp pattern. The purge stream absorbs a specified gaseous "fuel load," defined using data from the previous regeneration cycle. At the same time, the system reduces injection durations to compensate for the anticipated fuel content in the purge stream. Because the mixture-adaptation function is a separate process, the system now interprets any deviations from lambda as changes in "fuel load" and responds with corrective adjustment to the initial specification.
For the "load-sensitive" control of this purge flow, ME-Motronic uses parameters familiar from the intake-manifold model, which define such factors as the manifold's internal pressure and temperature. This facilitates precise calculation of the purge flow. The system is designed to operate with up to 40 % of the total fuel coming from the regeneration current.
With the lambda control system inactive, only a minimal regenerative current is allowed into the induction system, as until the lambda comes online there is no control mechanism capable of compensating for the mixture deviations that regeneration produces. In order to prevent unburned fuel vapors from entering the catalytic converter, the purge valve closes immediately in response to the interruption of fuel supply which occurs when the throttle is released (overrun fuel cutoff).

Knock control

Electronic control of ignition timing allows extremely precise adjustment of advance angles based on engine rpm, temperature and load factor.
Despite this precision, conventional systems must still operate with a substantial safety margin to avoid approaching the knock threshold. This margin is necessary to ensure that no cylinder will reach or go beyond the

Knock control

Control algorithm for ignition intervention on a 4-cylinder engine.
$K_{1...3}$ Knock at Cylinder 1...3 (no knock on Cyl. no. 4).
a Adjusted for retarded ignition timing,
b Increment for advanced ignition timing,
c Advanced ignition timing.

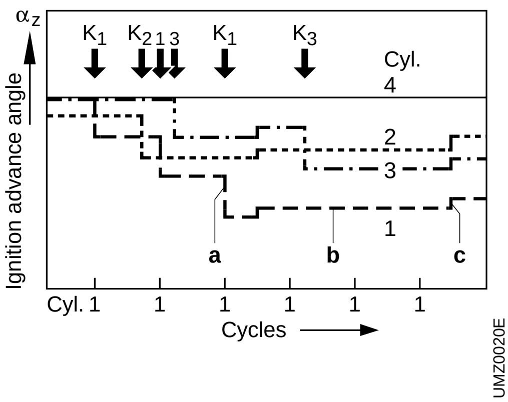


Fig. 1

preignition limit, even when susceptibility is increased by risk factors such as engine tolerances, aging, environmental conditions and fuel quality. The engine design which results when these factors are taken into consideration features a lower compression ratio with retarded ignition which lead to sacrifices in fuel consumption and torque.
These disadvantages can be avoided by using a knock-control system. Experience confirms that such a system allows higher compression ratios, with considerable improvements in both fuel economy and torque. With this system, it is no longer necessary to specify pilot ignition-timing angles defined to reflect worst-case scenaria. Instead, ideal conditions (engine compression at tolerance threshold, maximum fuel quality, cylinder least prone to preignition) can serve as the basis for specifying ignition timing. This makes it possible for each cylinder to be operated at the preignition limit, which coincides with optimal efficiency, in virtually all ranges, and throughout the life of the engine.
The essential prerequisite for this kind of knock-control system is reliable detection of any and all preignition exceeding a specified intensity. This must embrace

every cylinder and extend throughout the engine's entire operating range.
Preignition is detected by sensors designed to register solid-borne sonic waves. Installed at one or several suitable points on the engine, these knock sensors detect the characteristic oscillation patterns produced by knock and transform them into electrical signals suitable for transmission to the Motronic ECU for subsequent processing (refer to the section on ignition for additional information). The ECU employs a special processing algorithm to detect incipient preignition in every combustion cycle and in every cylinder. Detection of knock triggers a specified, programmed reduction in ignition advance. When the knock danger subsides, the ignition for the affected cylinder is then gradually advanced back toward the pilot ignition-timing angle.
The knock-recognition and knock-control algorithms are designed to prevent the kind of preignition that results in audible knock and engine damage (Figure 1).

Adaptation

Real-world engine operation is characterized by different knock limits in different cylinders, and ignition timing must be adjusted accordingly. In order to adapt the pilot-ignition timing to reflect the individual knock limits under varying operating conditions, individual ignition-retard increments are stored for each cylinder.
These data for specific engine speeds and load factors are stored in non-volatile program maps in permanently-powered RAMs. This strategy permits the engine to be operated at maximum efficiency under all conditions without any danger of audible combustion knock, even during abrupt changes in load and rpm.
The engine can even be approved to run on low-octane fuels. Standard practice is to adapt the engine to run on premium fuel.
Operation with regular-grade gasoline can also be approved.

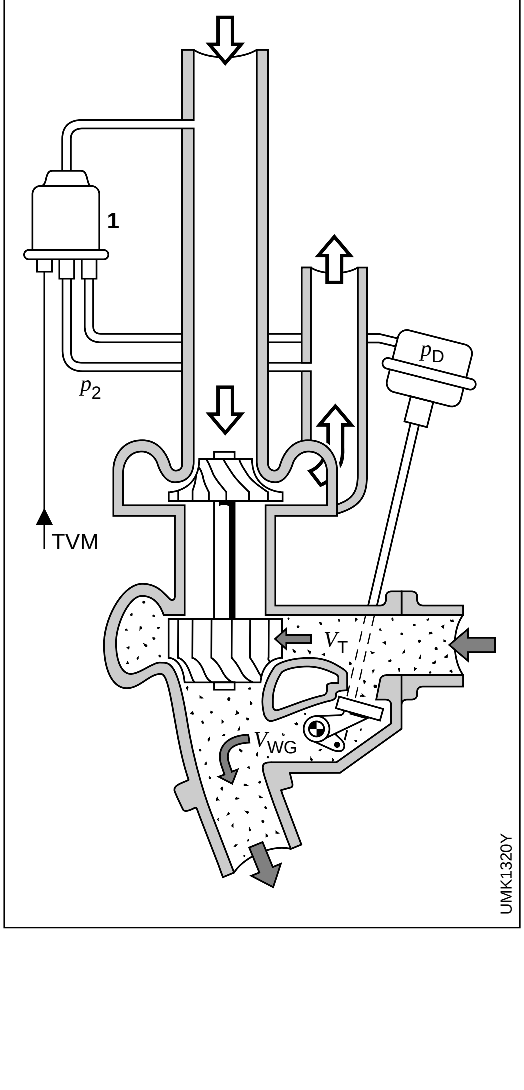
Boost-pressure control

Boost-pressure control mechanisms that rely on pneumatically-triggered mechanical layouts use actuators (wastegates) that are directly exposed to the pressure in the the impeller outlet. This concept allows only very limited definition of torque response as a function of engine speed. Load control is limited to the full-load bypass. There is no provision for compensation of the full-load boost

Fig. 1

Actuator for electronic boost-pressure control

1 Cycle valve.
p_2 Boost pressure,
p_D Pressure on diaphragm,
TVM Cycle valve triggering signal from ECU,
V_T Flow volume through turbine,
V_{WG} Flow volume through wastegate.



tolerances, and at part load, the closed wastegate impairs operating efficiency. Acceleration from low rpm can be marked by a delay in turbocharger response (a very pronounced “turbo lag”). These problems can be avoided with electronic boost-pressure control (Figure 1). This system can provide reductions in specific fuel consumption under some part-throttle operating conditions, controlling the wastegate’s opening pattern to obtain the following results:

– The engine’s back-pressure losses and the impeller’s output both drop,
– Pressure and temperature at the impeller’s discharge orifice fall, and
– The pressure gradient at the throttle valve is reduced.

The exhaust-gas turbocharger and its boost-control device must be precisely matched to the engine as the primary requirements for achieving these improvements.
The affected components in the boost-control device are:

– The electropneumatic cycle valve,
– The effective diaphragm surface, stroke and spring in the aneroid capsule, and
– The cross section of the valve head or valve flap in the wastegate.

ME-Motronic employs electronic boost control to regulate induction pressure to the specified value. This specified boost pressure is converted into a specification for the desired maximum cylinder charge. The torque-based control function converts this specification into a setpoint for throttle-valve aperture and a pulse-duty factor for the wastegate. The signal modifies the wastegate’s control pressure and stroke to regulate the bypass opening.
Control-circuit elements compensate for the difference between the setpoint defined by current operating conditions (program map) and the actual, monitored boost pressure. The calculated value at the controller output is then included in the process used to define the maximum cylinder charge.
On turbocharged engines, the temperature of the exhaust gas between engine and turbine should not exceed certain limits. This is why Motronic’s boost control operates exclusively in conjunction with knock control, as the latter represents the only means for operating the engine with maximum ignition advance throughout its service life. A result of using optimal ignition timing at all operating coordinates is extremely low exhaust-gas temperatures. Further reductions in exhaust temperature are available through intervention in cylinder charge, meaning boost pressure in this case, and/or air-fuel mixture.

Protective functions

Limiting vehicle and engine speed

Extremely high engine speeds can lead to powerplant demolition (valve train, pistons). The rpm limiting function prevents the maximum approved engine speed (redline) from being exceeded.
Incorporation of a vehicle-speed limiter may be necessary in response to specific equipment specifications as defined for vehicles in certain markets (i.e., tires, suspension). In addition, several German manufacturers have made a voluntary commitment to limit the maximum speeds of their vehicles to 250 km/h.
The functions for restricting vehicle and engine speeds operate according to the same principles. A control agorithm reduces the permitted engine output once a specified threshold is crossed. This output limit is included in ME-Motronic’s torque-based control function.

Torque and power limits

It is sometimes necessary to restrict torque generation in order to reduce the loading on certain drivetrain components (such as the transmission). ME-Motronic’s torque-based control function provides for the definition of such a limit. It is also possible to restrict ultimate output by governing engine speed and torque.

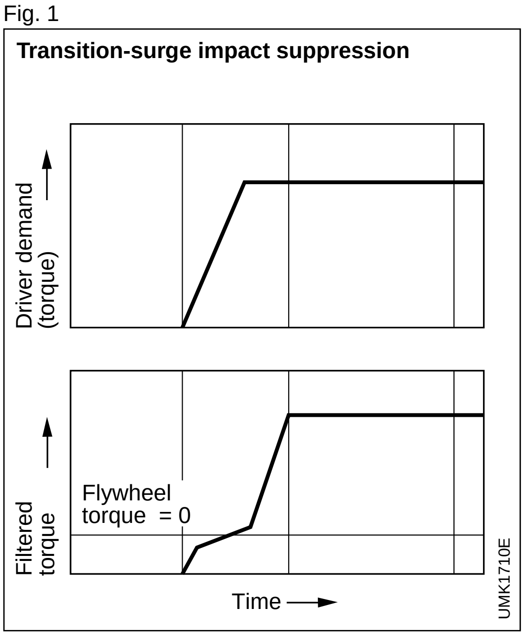
Limiting exhaust-gas temperatures

High exhaust-gas temperatures can damage exhaust-system components. Therefore, a model incorporated within the ECU is employed to simulate these temperatures. Extreme requirements for monitoring precision can be satisfied by installing a temperature sensor. Temperatures beyond a defined threshold trigger mixture enrichment, which cools the exhaust by extracting heat energy to vaporize the fuel. Limiting charge density and torque are additional options.

Vehicle immobilizer

To prevent unauthorized vehicle use, the Motronic ECU incorporates a feature that prevents the engine from being started until the ECU itself has been released via a special control line. The actual release mechanism is an encoded signal prepared by an external control unit. This second control unit verifies user authorization by analyzing the signal from a transmitter in the ignition key or a keypad entry code, etc.

Fig. 1



Improved drivability

Transition surge-impact suppression

Positive and negative load shifts – initiated by abruptly depressing or releasing the accelerator pedal – can produce jolts in the driveline. This effect is especially pronounced when the torque reversal transfers forces to mounting bushings or the transmission. An example is the engine, which shifts from one engine mount to the other during transitions from power-on to power-off.
This force transfer can be prevented, or at least reduced in intensity, by controlling the rates of torque rise and reduction in order to achieve gentler transitions. In order to adjust flywheel torque (Fig. 1) this strategy relies on manipulation of ignition timing and cylinder charge.

Surge-damping function

The fact that the engine and drivetrain represent a spring-mass system means that during operation this system can start to oscillate. The surge-damping function detects these oscillations and suppresses them by intervening in engine output torque in the respective phase. Oscillation recognition is based on a comparison between a reference rpm derived from driver demand and the current rpm. Intervention is through adjustments to ignition timing. Effective suppression of driveline oscillations entails implementing the torque intervention at opposed phases to the torque oscillation.

Cruise control

The cruise control’s function is to compensate for changes in rolling/aerodynamic resistance and maintain a constant vehicle speed without the driver

having to press the accelerator pedal. In addition to maintaining current vehicular velocity, these systems provide a range of supplementary functions. Cruise control thus enhances driving convenience on long trips while also facilitating compliance with posted speed limits.
Because ME-Motronic's throttle-valve actuator is already integrated in the ETC function, the supplementary effort required to integrate cruise control within the system is minimal.

Functions:

The driver uses a stalk/switch to control the following functions:
- Adopt and then maintain the current road-speed (set),
- Accelerate and then maintain predefined road-speed,
- Decelerate and then maintain predefined road-speed,
- Accelerate to a stored target speed (resume),
- Graduated incremental increase in defined road-speed (tip up),
- Graduated incremental decrease in defined road-speed (tip down),
- Cruise control deactivation at main switch and/or tip-off switch.

Control elements

The driver can operate the cruise-control system with a single control element which incorporates switches for the functions:
- Set,
- Resume,
- Accelerate, and
- Decelerate.

Depending upon its particular configuration, an individual switch may govern more than one function. Thus one button may be used for set/decelerate and one for resume/accelerate.
In such a case, the function to be implemented by the system when the switch is activated will depend upon the system's current status and how long pressure is applied to the switch. The tip-up and tip-down functions are triggered by brief pressure on the switch for acceleration and deceleration. In addition to the switch components for specific functions, the control element may also include an optional main switch and a cruise-control cancelation switch. Where present, the main switch must be switched on before entries at the function switch will be registered. When the main switch is deactivated, any speeds stored previously will be lost. Active cruise control can also be interrupted by applying pressure to the brake or clutch pedal.

Fig. 1
Cruise-control operating stalk

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Integrated diagnosis

Diagnostic procedure

An "On-Board Diagnosis" (OBD) system is standard equipment with Motronic. This integral diagnostic unit monitors ECU commands and system responses while also checking sensor signals for plausibility. This test program proceeds continually during normal vehicle operation.
The ECU stores recognized errors together with the operating conditions under which they arose. When the vehicle is serviced, a tester can then be used to read out and display the stored error data through a standardized interface. This information facilitates fault diagnosis procedures for service personnel.
Diagnosis processes extending far beyond those in earlier systems have been developed to comply with mandates issued by the California Air Resources Board. All components whose failure could cause a substantial increase in harmful emissions must be monitored, and detected faults must trigger a diagnosis lamp in the instrument panel. This expanded diagnosis is designated OBD II.
An OBD system adapted for European conditions is designated EOBD.

Diagnosed areas

Air-mass meter

The process for monitoring operation of the air-mass meter is an example of Motronic's self-diagnosis function. While cylinder charge is being determined based on the mass of the induction air, a supplementary reference is calculated at the same time from throttle-valve angle and engine speed. If the ECU detects excessive variation between the two, its initial response is to record the error. As vehicle operation continues, plausibility checks determine which of the sensors is defective. The ECU does not store the designated error code until it has unequivocally determined which sensor is at fault.

ETC throttle-valve actuator

Because engine output is manipulated by adapting cylinder charge, the throttle-valve actuator must satisfy stringent demands for reliability and diagnosis. The actuator monitors current throttle-valve position with two, mutually counterrotational potentiometers. The signals these produce are then compared. If a deviation occurs, the system falls back on the intake-manifold model as a basis for extrapolating throttle-valve position and restoring signal plausibility.

Combustion miss

Combustion miss, resulting from such factors as worn spark plugs, allows unburned mixture to enter the catalytic converter. This mixture can destroy the catalyst, and is also detrimental to the environment. Because even an isolated combustion miss produces higher emissions, the system must be able to detect it.

Fig. 1

Exhaust emissions as a function of combustion miss

Engine: 6 cyl., 2.8 litre.
US emissions limits.

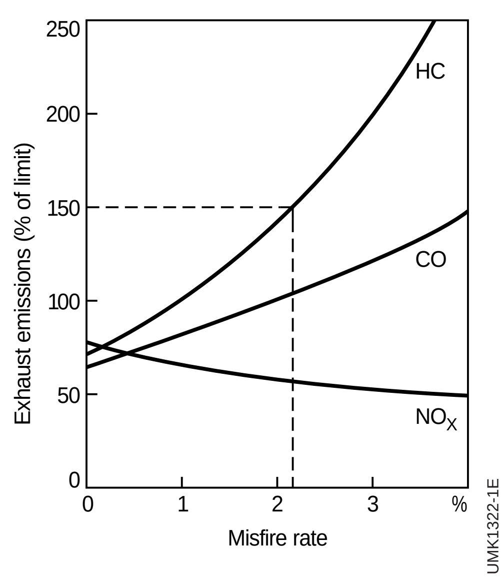


Figure 1 shows the effects of combustion miss on emissions of hydrocarbons (HC), carbon monoxide (CO) and oxides of nitrogen (NO_X).

In the search for a method to monitor combustion miss, the monitoring of inconsistencies in the crankshaft's rotational speed has emerged as the best compromise between complexity and practicability. Combustion miss is accompanied by a shortfall in torque equal to the increment that would normally have been produced in the cycle. The result is a reduction in rotation speed. At high speeds and low load factors the intervals separating firing points (periodicity) will be extended by only 0.2 %. This means that 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. This process relies on a second oxygen sensor mounted downstream of the converter as a supplement to the first, cat-forward unit. A healthy catalytic converter will store oxygen, thus attenuating the lambda control oscillations. As the catalyst ages, this effect deteriorates until finally the signal patterns from the two sensors start to converge. Comparison of the signals from the two O_2 sensors thus serves as the basis for assessing the catalytic converter's condition. A warning lamp alerts the driver when a defect is detected.

Lambda-Sonde

<u>Two-state sensor (Nernst probe)</u>

A precisely stoichiometric air-fuel mixture is vital for optimal operation of the catalytic converter.

The lambda closed-loop control system uses the signals from the oxygen sensors to maintain this mixture.

Monitoring dynamic response of lambda oxygen sensors

a New sensor,
b Aged Type II sensor,
c Aged Type III sensor,
T Phase duration.

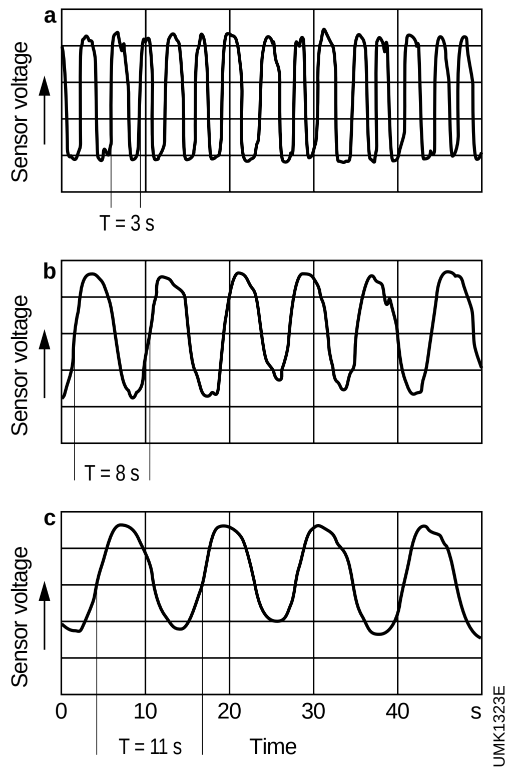


Fig. 2

The oxygen sensors are diagnosed for:

– Electrical plausibility: The system continuously assesses the sensor signal's plausibility. It reacts to implausible signals by deactivating other lambda-related control functions while simultaneously entering the corresponding error code in the fault memory.

– Sensor dynamic response (phase duration, Figure 2): An oxygen sensor exposed to excessively high temperatures for an extended period of time may start to react more slowly to changes in the air-fuel mixture. This leads to extended phase durations (periodicity) in the two-state control pattern. A diagnosis function monitors this control frequency and triggers a diagnosis lamp to alert the driver to excessive response lag in the sensor.

– Control range: The presence of two oxygen sensors in each exhaust line makes it possible to use the post-catalyst (“cat-back”) sensor to check the engine-side (“cat-forward”) sensor for drift in its effective response range.

– Heater: The system checks current and voltage to the oxygen sensor’s heater resistor. For this check the Motronic ECU must rely on a direct link to the heater resistor, instead of controlling it through a relay.

LSU wide-band sensor

With the introduction of the LSU wide-band oxygen sensor it is now possible to monitor specified mixtures other than $\lambda = 1$. The ongoing lambda control process uses a “cat-forward” control circuit with an LSU, complimented by a super-imposed “cat-back” circuit featuring a two-state sensor. This strategy makes it possible to assess the LSU’s operation with the aid of the two-state sensor. The diagnosis process includes the following elements:

– Electrical plausibility: One feature that distinguishes the LSU sensor from its two-state counterpart is that the potential spectrum of plausible signals extends throughout the entire voltage range. In addition to checking upper and lower limit values, the system also compares its signal with that being transmitted from the sensor mounted downstream of the catalytic converter.

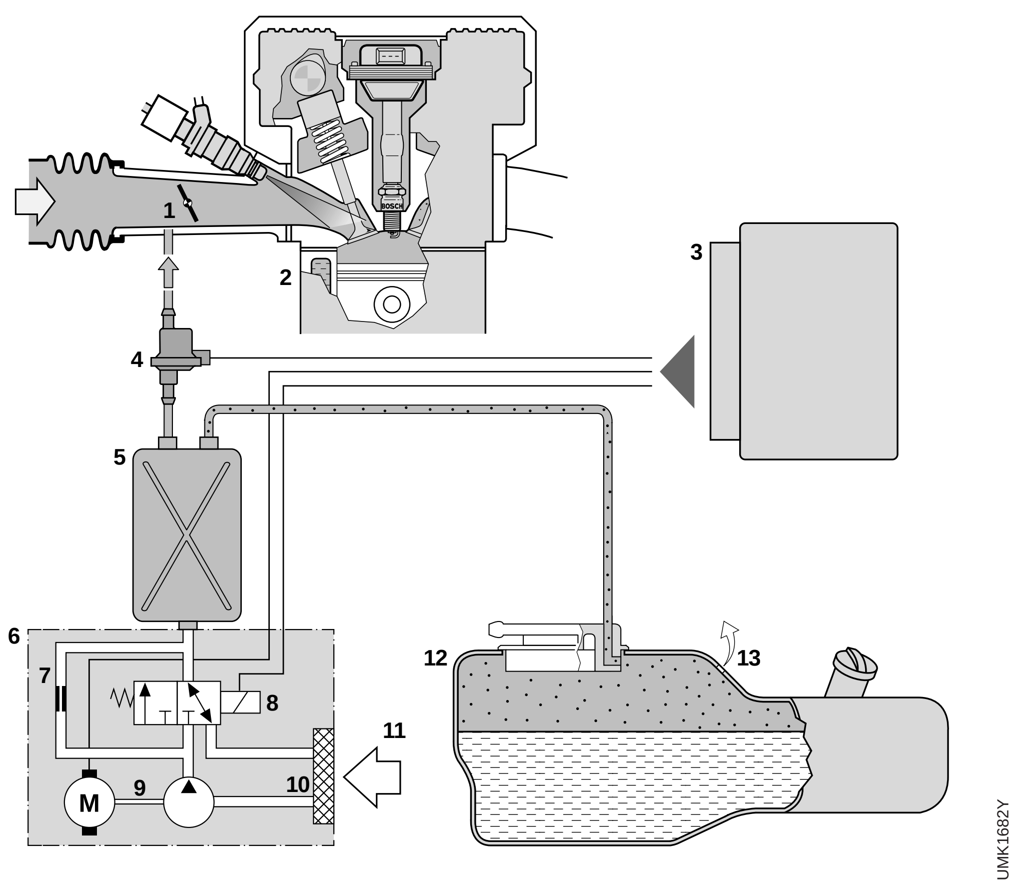
– Sensor dynamic response: Diagnosis based on assessment of a mandatory, superimposed amplitude.

– Control range: The second, “cat-back” sensor is used to verify compliance with a delta lambda threshold.

Fig. 3

Reference leak test for detecting fuel-system leakage

1 Throttle valve, **2** Engine, **3** ECU, **4** Canister-purge valve, **5** Activated-charcoal canister, **6** Diagnosis module, **7** Reference leak, **8** Circuit-control valve, **9** Electric air pump, **10** Filter, **11** Fresh air, **12** Fuel tank, 13 Leak.



– Heater: This process is the same one used with the two-state sensor, while ongoing variations in the lambda signal are also evaluated.

Fuel supply
When the air-fuel ratio deviates from stoichiometric for extended periods of time, the system recognizes this state, with the mixture adaptation serving as one reference. If the deviations exceed specific programmed limits, this indicates that a fuel-system or fuel-metering component has shifted outside its specified tolerance range. An example would be a faulty pressure regulator or cylinder-charge sensor (for instance, the hot-film air-mass meter), while other potential error sources include leaks at the intake manifold or in the exhaust system.

Tank system
Emissions emanating from the exhaust system are not the sole element of ecological concern; vapors from the fuel tank are also a problem.
While current requirements for the European market are limited to a relatively simple check to verify correct operation of the purge valve, US mandates already demand a means of detecting leakage at any and all points within the evaporative-emissions control system.

<u>Vacuum test</u>
The diagnosis process relies on analysis under vacuum. A shut-off valve is used to block the fresh-air supply to the carbon canister and seal off the vapor retention system. Then, preferably with the engine running at idle, the purge valve is opened and vacuum from the intake manifold propagates throughout the system. A tank-mounted pressure sensor monitors the subsequent pressure curve to determine whether leakage is present.

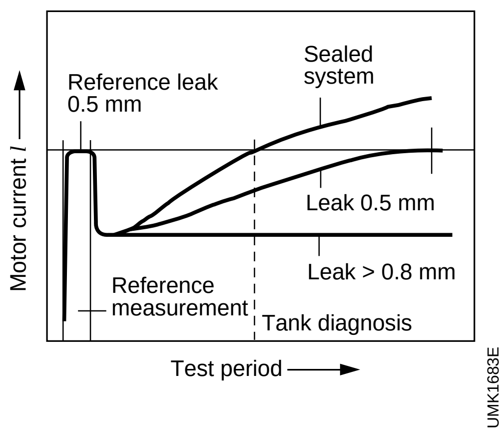
<u>Reference leak procedure</u>
Yet another process for diagnosing fuel tank leakage (Figure 3) relies on an electric air pump (9) to pressurize the tank (12). Instead of monitoring pressure with a pressure sensor, this test uses the pump's current draw as its test parameter. The first step is calibration, based on simulating a reference leak (7) with a defined flow at the purge valve. Then a circuit-control valve (8) is used to link the pump with the activated-charcoal canister (5). The resulting current pattern will point to any leakage present in the fuel system (Figure 4).

Secondary-air injection
The secondary-air injection activated following cold starts must also be monitored, as its failure would also effect emissions. This can be done using the signals from the lambda oxygen sensor while the secondary-air injection is in operation, or the injection can be activated and observed at idle using a special lambda test function.

Exhaust-gas recirculation
Because exhaust-gas recirculation (EGR) permits reductions in the concentrations of nitrous oxides in the exhaust gas, the operational integrity of this system must also be monitored.
Opening the EGR valve conducts part of the exhaust-gas flow back into the intake manifold. As this supplementary flow of residual gases enters the manifold and

Fig. 4

Schematic portrayal of pump current curve during pressurization test on fuel system



then the cylinders, it initially affects manifold vacuum, and then, subsequently, the combustion process. This characteristic opens two options for diagnosing the EGR system:

Diagnosis based on manifold vacuum
The EGR valve is briefly closed under part-throttle operation. If ETC is used to hold the flow of induction air passing through the hot-film air-mass meter at a constant level, the valve will shift the intake-manifold vacuum. This change is monitored with the manifold pressure sensor, and its magnitude is an index of the EGR system's condition.

Diagnosis based on idling stability
This method is employed on systems without hot-film air-mass meter or supplementary intake-manifold pressure sensor. At idle, the EGR valve is opened slightly. The increased mass of residual gas leads to rougher engine operation. This, in turn, is detected by the system's smooth-running monitoring system which applies the rough running to diagnose the EGR system.

Other monitoring functions
While the new emissions statutes focus on the engine-management system, other systems (such as the electronic transmission-shift control) are also monitored. These relay data on any faults to the engine-management ECU, which then assumes responsibility for triggering the diagnosis lamp.
Greater system complexity and increasingly stringent environmental legislation are making diagnosis increasingly important.

Diagnosis sequence scheduling
OBD II demands completion of all diagnosis functions at least once in the course of the emissions test cycle. The former scheduling concept triggered the individual diagnosis functions according to an invariable scheme; the pattern was dictated by the operating conditions that characterized the individual stages of the emissions test cycle. Under normal operating conditions, this strategy could lead to extended delays before the right operating conditions for initiating a specific diagnosis process were encountered. In practice this means that some diagnostic procedures might be performed only rarely, as individual driving habits could render it impossible for the system to adhere to the scheduled diagnosis sequence.
Diagnosis System Management can respond to operating conditions and initiate dynamic adjustments in the diagnostic sequence to allow optimal diagnosis processing in daily operation.

Fault memory
Detection of emissions-relevant malfunctions leads to an entry in the non-volatile fault memory. While this entry reflects the officially mandated error codes, it also includes a “freeze frame” consisting of supplementary information on the operating conditions under which the error occurs (i.e., engine speed and temperature). Depending on the system project, malfunctions of significance for vehicle servicing are also stored, although not included in the OBD II catalog.
Error codes can be read out by connecting the client's own service tester or a Bosch Motortester (Figure 5) to the ECU interface. The same equipment can be used to record test data (such as engine speed). OBD II prescribes standardized protocols for storing malfunction data as defined by the SAE (Society of Automotive Engineers). This makes it possible to access the fault memory using standard, commercially-available “scan tools.”

Emergency (limp-home) mode

In the interval elapsing between initial occurrence of a fault and the subsequent vehicle workshop visit, ignition timing and air-fuel mixtures can be processed based on default values and emergency-running functions. This allows continued vehicle operation, albeit with sacrifices in comfort and convenience. The ECU responds to recognized errors in an input line by substituting data from a simulation model or by falling back on a redundant signal.

Failure in units on the output side initiate implementation of specific backup measures corresponding to the individual problem. As an example, the system can react to a defect in the ignition circuit by suppressing fuel injection at the affected cylinder to prevent damage to the catalytic converter.

The ETC throttle-valve actuator has a spring-loaded default position. This maintains engine operation at low rpm, allowing ME systems to continue in restricted operation in the event of failure in the vital ETC circuit.

Actuator diagnosis

During normal driving, many Motronic functions (e.g., EGR) are operational only under specific conditions. This renders it impossible to activate all actuators (such as the EGR valve) for operational checks while on the road.

Actuator diagnosis represents a unique case within the range of diagnostic processes. It proceeds with the engine off, and never during normal operation. This test mode relies on the engine tester, which serves as the triggering device for operational checks in the service workshop. The actual test process entails activation of each actuator in sequence. Operation can then be verified using acoustic or other means.

In this mode the injectors should only be triggered with extremely brief pulses (< 1 ms). Although this is not enough time for the injector to open completely, and no fuel is injected into the manifold, the sound can be clearly heard.

Fig. 5

Bosch KTS 500 Motortester in use

ECU

Purpose
This ECU (Figure 1) serves as the “processing and control center” for the engine-management system. It employs stored functions and algorithms (processing programs) to process the incoming signals from the sensors. These signals serve as the basis for calculating the control signals to the actuators (such as ignition coil and injectors) which it controls directly through its driver stages (3).

Physical design
The ECU has a metal housing containing a printed circuit board (2) with electronic componentry. Compact units which feature hybrid technology are available for installation directly on the engine. They are specially designed to withstand higher thermal stresses.

A multi-terminal plug connects the ECU to its sensors and actuators, as well as to its power supply. The number of individual terminals within this interface varies according to the number of functions covered by the ECU. More than 100 terminals must usually be included in the ME-Motronic connector plugs.

The PCB has a metal base underneath the output amplifier circuitry, where through-hole contacts provide good thermal transfer to its lower side. From here the heat generated by the output amplifier circuits is transferred through heat bridges to the housing.

Environmental conditions
The ECU must withstand extreme temperatures, humidity and physical stresses. Resistance to incoming electromagnetic interference and the ability to suppress outward-bound radiation of high-frequency static must also be of a high order.

Under normal operating conditions the ECU must be capable of errorless signal processing at environmental temperatures ranging from –30 °C to +60 °C, at with battery voltages that extend from 6 V (during starting) to 15 V.

Power supply
A voltage regulator (10) provides the ECU with the constant 5 V operating power needed for the digital circuitry.

Signal entry
The sensor signals enter the ECU via protective circuits, and through signal converters and amplifiers where indicated:

– Analog-digital (A/D) converters integrated in the microprocesors (4, 7) transform analog input signals (e.g., information on accelerator-pedal position, induction air mass, engine and intake-air temperature, battery voltage, air-fuel mixture ratio, etc.) into digitalized data.

– Digital input signals (such as switching signals from the air conditioner or transmission selector lever, but also including digital signals such as rpm pulses from Hall sensors) are suitable for direct processing in the microprocessor.

– Incoming signals in pulse form transmitted by inductive sensors furnish information on vehicle speed as well as on crankshaft speed and angle. These are processed in a special circuit (10) and converted into square-wave signals.

Depending upon the system’s integration level, initial signal processing may be partially or completely performed in the sensor itself. Input information arriving through the data bus (CAN) also arrives without need for preliminary processing.

Signal processing
Input signals are handled 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, 5). This memory also contains the specific individual performance curves and program maps (data) used to govern the engine-management system.

Owing to the large number of engine and vehicle variations with their extensive

individual ranges of data requirements, the ECUs are not programmed until they reach the end of the production line. There is no need to open up the ECU for this procedure. This reduces the number of ECU configurations required by any single vehicle manufacturer.
A read/write memory (RAM) is needed for storing calculated values and adaptation factors along with any system errors that may be detected (diagnosis). RAMs rely on an uninterrupted power supply for continued operation, and lose all data if the vehicle's battery is disconnected. The ECU must then recalculate the adaptation factors after the battery is reconnected. To get around this problem, some units store required variables in an EEPROM (6, non-volatile memory) instead of a RAM.

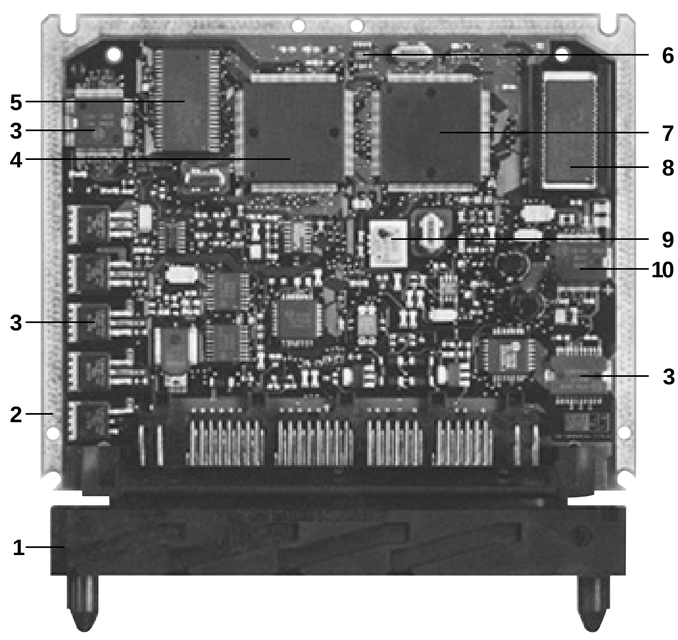
Signal output
The microprocessor-controlled output amplifier (driver) circuits supply sufficient power for direct operation of the actuators. These driver circuits are protected against shorts to ground, fluctuations in battery voltage and the electrical overloads that could destroy them. The OBD diagnosis function is able to recognizes errors in the driver stages, and reacts by deactivating the respective output (where necessary). The error entry is stored in the RAM. The service technician can then access the error information by connecting a tester to the serial interface.
Another protective circuit operates independently of the ECU to deactivate the electric fuel pump when engine speed falls below a specified level. When the Terminal 15 power supply is interrupted at the ignition switch ("ignition off"), some ECUs rely on a holding circuit to keep the main relay open until program processing can be completed.

Fig. 1

ME7 Electronic control unit

1 Multi-terminal plug connection,
2 Printed circuit board,
3 Driver stages,
4 Microprocessor with ROM (function processor),
5 Flash EPROM (supplementary memory with program for specific vehicle),
6 EEPROM,
7 Microprocessor with ROM (expansion processor),
8 Flash EPROM (program memory for expansion processor),
9 Barometric-pressure sensor,
10 CJ910 peripheral chip (integrated 5V voltage supply and inductive sensor evaluation circuit).

RAM is on the underside of the PCB and thus invisible in this figure.



Interfaces to other systems

System overview

Increasingly widespread application of electronic control systems for automotive functions such as
- Electronic engine management (Motronic),
- Electronic transmission-shift control,
- Electronic vehicle immobilizers,
- Antilock braking systems (ABS),
- Traction control system (TCS), and
- On-board computers

has made it vital to interconnect the individual control circuits by means of networks. Data transfer between the various control systems reduces the number of sensors while also promoting exploitation of the performance potential in the individual systems.

The interfaces can be divided into two categories:
- Conventional interfaces, with binary signals (switch inputs), pulse-duty factors (pulse-width modulated signals), and
- 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 two conditions: “1” or “0” (binary code). An example would be the a/c compressor, which can be “on” or “off.”
Pulse-duty factors can be employed to relay more detailed data, such as throttle-valve aperture.
Increasing data traffic between various on-board electronic components means that conventional interfaces are no longer capable of providing satisfactory performance. The complexity of current wiring harnesses is already difficult to manage, and the requirements for data communications between ECUs are on the rise (Figure 1).

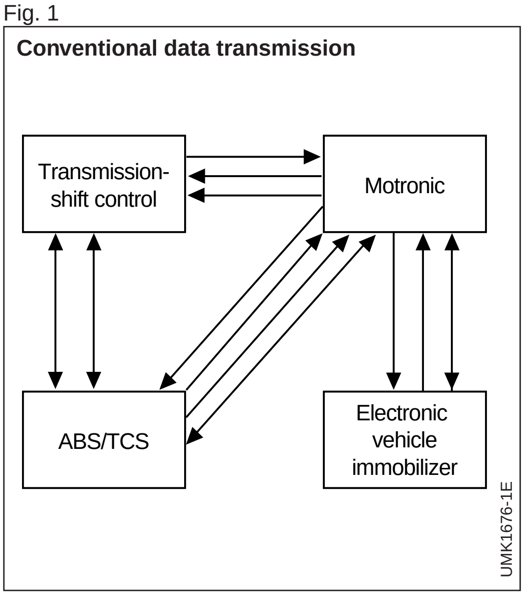
Serial data transmission (CAN)

These problems can be solved with a CAN, that is, a bus system (bus bar) specially designed for automotive applications.
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.

There are three basic applications for CAN in motor vehicles:
- To link ECU’s,
- Body-related and convenience electronics (multiplex), and
- Mobile communications.

The following is limited to a description of communications between ECUs.

ECU networking
This strategy links electronic systems such as Motronic, electronic transmission-shift control, etc. Typical transmission rates lie between approximately 125 kBit/s and 1 MBit/s, and must be high enough to maintain the required real-time response. One of the advantages that distinguishes serial data transfer from conventional interfaces (pulse-duty factors, switching and analog signals,

Fig. 1


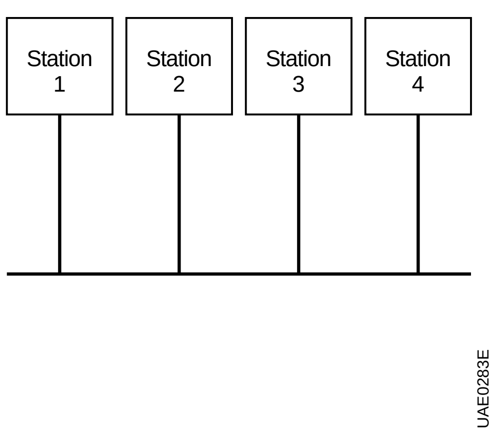
etc.) is the high speeds achieved without placing major burdens on the central processing units (CPU's).

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

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

Fig. 2

Linear bus structure



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

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 Organization for Standardisation (ISO) has recognized a CAN standard for use in automotive applications with data rates of over 125 kBit/s, and along with two other protocols for data rates of up to 125 kBit/s.

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