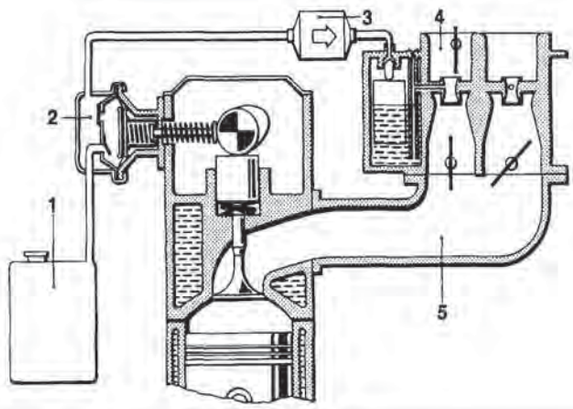
Schematic of a carburetor system
1 Fuel tank, 2 Fuel supply pump, 3 Fuel filter, 4 Carburetor, 5 Intake manifold.



Carburetors

Carburetor systems

The fuel is transported from the fuel tank to the carburetor by a fuel pump (generally a diaphragm unit) powered by the camshaft or distributor shaft. The system is designed to limit the maximum supply pressure. A fine-mesh fuel filter can be installed upstream or downstream from the pump as required.

Carburetor types

Downdraft carburetors

Downdraft carburetors are the most common type. Designs featuring optimized float chamber and metering-jet configurations result in efficient units. These designs work in conjunction with the corresponding intake-manifold layouts for optimum mixture formation and distribution.

Horizontal-draft carburetors

Horizontal-draft carburetors (familiar as fixed-venturi and constant-depression units) are useful for minimizing engine height.

Constant-depression carburetors feature venturi cross sections which vary in size during operation to maintain essentially constant vacuum levels at the fuel outlet. The variation in intake cross section is provided by a pneumatically-actuated plunger; attached to the plunger is a needle which regulates the fuel quantity.

Venturi configurations

The single-throat carburetor with one venturi is the least expensive design.

The two-stage carburetor featuring two venturis provides convenient tuning for individual applications and has become the standard in 4-cylinder applications. The first barrel controls part-throttle operation, while the second venturi opens for maximum power.

The double-barrel carburetor features two carburetor sections sharing a single float chamber and operating in parallel, making it ideal for use on 6-cylinder engines. The two-stage four-barrel carburetor has four venturi fed from a single float chamber.

Design and operating principles

The driver uses the accelerator pedal to vary the throttle valve's aperture so that the airflow into the engine is varied and with it the engine's power output. The carburetor varies the amount of fuel metered to the engine to reflect the current intake air flow. Together with the needle valve, the float regulates the fuel flow to the carburetor while maintaining a constant fuel level in the float chamber.

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Airflow is monitored by an air funnel designed to induce a venturi effect. The progressively narrower diameter increases the velocity of the air, producing a corresponding vacuum at the narrowest point. The resulting pressure differential relative to the float chamber – which can be further augmented with a boost venturi – is exploited to extract fuel from the float chamber. The jets and metering systems adapt fuel delivery to airflow.

Fuel-metering systems

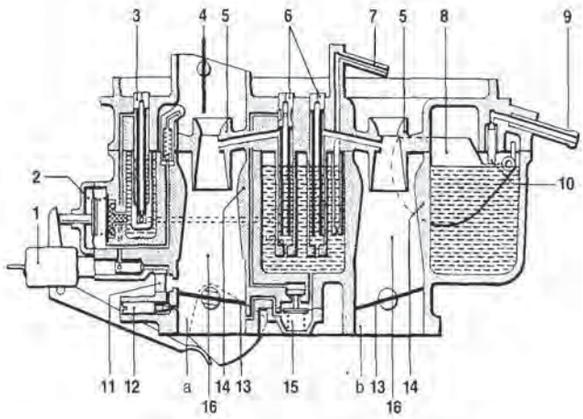
Main system
The fuel is metered by the main jet. Correction air is added as a delivery aid to the fuel through side orifices in the venturi tube.

Idle and progression system
At idle, the vacuum which the air stream produces at the fuel outlet is not sufficient to withdraw fuel from the main system. For this reason, there is a separate idle system with an outlet located downstream from the throttle valve at the point of maximum vacuum. The emulsion required for idling emerges from the idle circuit after initial processing by the idle fuel and air-correction jets.

During transitions to the main metering system the throttle valve controls a series of orifices, or a slit, drawing fuel from the idle circuit.

Other systems
These basic devices are supplemented by a range of additional systems. These are designed to adapt carburetor performance for warm operation (part-throttle control, full-throttle enrichment), to compensate for fuel accumulation with n the intake-manifold during acceleration (accelerator pump) and to meet the special engine requirements encountered during starting and in the warm-up phase. Other supplementary systems include lambda closed-loop mixture control and devices to deactivate the fuel supply during trailing-throttle operation.

Schematic of a two-stage carburetor
a) Primary stage, b) Secondary stage.
1 Idle cutoff valve, 2 Accelerator pump, 3 Idle circuit, 4 Choke, 5 Boost venturi, 6 Main systems with venturi tubes, 7 Full-throttle enrichment, 8 Float, 9 Fuel supply, 10 Needle valve, 11 Bypass plug, 12 Idle mixture screw, 13 Throttle valves, 14 Venturi wall, 15 Part-throttle control valve, 16 Venturi chamber.



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Electronically-controlled carburetor system (ECOTRONIC)

Basic carburetor

The basic carburetor is restricted to the throttle valve, float system, idle and transition systems, main system and choke. An idle-air control system with a choke-activated needle jet is also provided.

Additional components and actuators

The throttle-valve actuator is an electropneumatic servo device for controlling the cylinder charge. The actuator's plunger moves the throttle valve via a lever attached to the carburetor's throttle shaft.

The choke valve actuator is a final-control element designed to adapt the mixture in response to variations in engine operating conditions. This unit closes the choke valve to enrich the mixture by raising the pressure differential (vacuum) at the main jets while simultaneously increasing flow rates from the idle circuit.

Sensors

The throttle-valve potentiometer monitors the throttle valve's position and travel. One temperature sensor monitors the engine's operating temperature while a second sensor can be installed if necessary to monitor the temperature within the intake manifold.

The idle switch serves to identify trailing-throttle operation; it can be replaced by appropriate software in the electronic control unit (ECU).

Electronic control unit (ECU)

The ECU's input circuit converts ir coming analog signals into digital form. The processor performs further operations with the input data in order to calculate output values with reference to the programmed data map. The output signals control several functions, includir g regulation of the servo elements that operate the choke valve and main throttle valve.

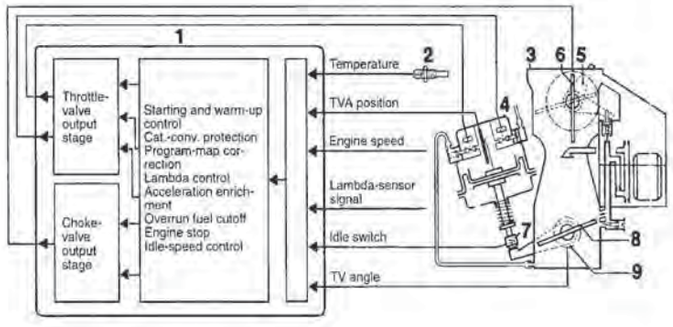
Basic functions

The basic carburetor determines the primary functions of the system. The idle, transition and full-throttle systems all contribute to matching performance to the programmed curves. The base calibrations can be intentionally "lean", as the choke-valve control can provide a corrective enrichment.

Electronic functions

Electronic open and closed-loop control circuits regulate a number of secondary operations within the ECU. Several of these are illustrated below. Further functions may include: ignition control, transmission-shift control, fuel consumption displays and diagnosis capabilities.

Schematic of an electronically controlled carburetor (ECOTRONIC)
1 ECU, 2 Temperature sensor, 3 Carburetor, 4 Throttle actuator, 5 Choke actuator, 6 Choke valve, 7 Idle switch, 8 Throttle valve, 9 Throttle potentiometer.



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Gasoline injection systems

Systems for external A/F mixture formation

Gasoline injection systems for external mixture formation are identified by the fact that the air-fuel mixture is created outside the combustion chamber (in the intake manifold).

Single-point injection (SPI)
Single-point injection is an electronically controlled injection system in which an electromagnetic fuel injector injects the fuel intermittently into the intake manifold at a central point ahead of the throttle valve. The Bosch single-point injection systems are called Mono-Jetronic and Mono-Motronic.

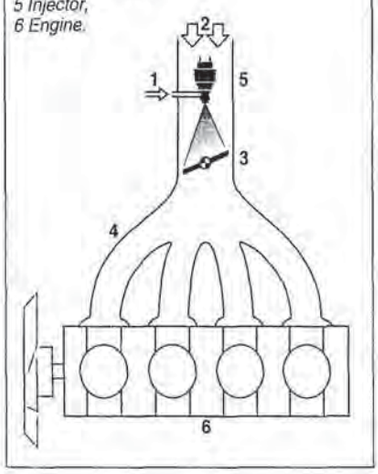
Multipoint injection (MPI)
Multipoint injection creates the ideal preconditions for satisfying the demands placed on a mixture-formation system. In multipoint injection systems, each cylinder is assigned a fuel injector, which injects the fuel directly ahead of that cylinder's intake valve. Examples of such systems are KE- and L-Jetronic with their respective variants.

Mechanical injection system:
The K-Jetronic system operates without a drive and injects the fuel continuously. The injected fuel mass is not determined by the fuel injector but rather prespecified by the fuel distributor.

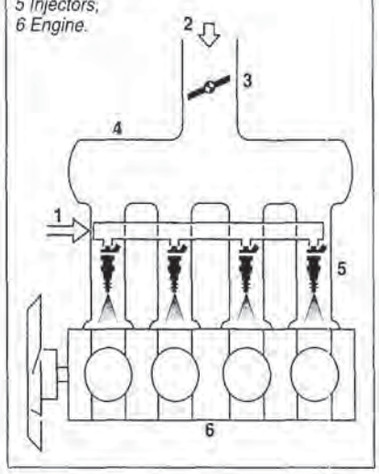
Combined mechanical-electronic injection system:
KE-Jetronic is based on the mechanical basic system of K-Jetronic. Thanks to the extended acquisition of operating data, this system facilitates electronically controlled supplementary functions in order to adapt the injected fuel quantity more exactly to the different engine operating states.

Electronic injection systems:
Electronically controlled injection systems inject the fuel intermittently with electromagnetically actuated fuel injectors. The injected fuel mass is determined by the injector opening time (for a given pressure drop across the injector).

Single-point fuel injection
1 Fuel,
2 Air,
3 Throttle valve,
4 Intake manifold,
5 Injector,
6 Engine.



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

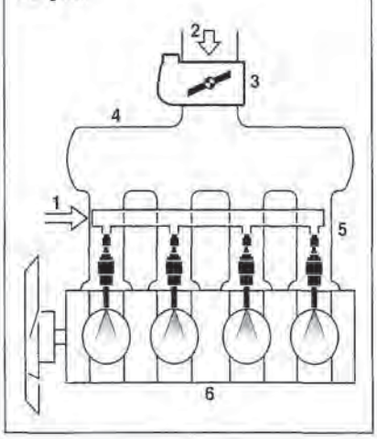


Examples: L-Jetronic, LH-Jetronic, and Motronic as an integrated engine management system.

The high standards required of a vehicle's smooth running and exhaust-emissions necessitate high demands being made on the A/F mixture composition of each working cycle. Precisely timed injection is significant as well as precise metering of the injected fuel mass in accordance with the air drawn in by the engine.

In modern multipoint injection systems, therefore, not only is each engine cylinder assigned an electromagnetic fuel injector but also this fuel injector is activated individually for each cylinder. In this way, both the fuel mass appropriate to each cylinder and the correct start of injection are calculated by the control unit (ECU). Injecting the precisely metered fuel mass directly ahead of the cylinder intake valve(s) at the correct moment in time improves mixture formation. This, in turn, helps to a large extent in preventing wetting of the intake-manifold walls with fuel, which can result in temporary deviations from the desired lambda value during transient engine operation. The advantages of multipoint injection can thus be fully exploited.

Direct injection (DI)
1 Fuel,
2 Air,
3 Throttle-valve (EGAS),
4 Intake manifold,
5 Injectors,
6 Engine.



The engine intake manifolds thus carry only the combustion air and can therefore be optimally adapted to the gas-dynamic requirements of the engine.

Systems for internal A/F mixture formation

In direct-injection systems for internal mixture formation, the fuel is injected directly into the combustion chambers by electromagnetically actuated fuel injectors. Each cylinder is assigned a fuel injector. Mixture formation takes place inside the cylinder. To ensure efficient combustion, it is essential that the fuel be finely atomized when leaving the injectors.

In normal operation, a direct-injection engine draws in only air and not an air-fuel mixture, as is the case in conventional injection systems. Herein lies an advantage of this new system: no fuel can precipitate on the intake-manifold walls. With external mixture formation, the air-fuel mixture is generally present throughout the entire combustion in a homogeneous state and in a stoichiometric ratio. On the other hand, formation of the mixture in the combustion chamber permits two completely different operating modes:

Stratified-charge operation

In stratified-charge operation, the mixture only has to be combustible in the area around the spark plug. The remaining section of the combustion chamber thus only contains fresh mixture and residual exhaust gas without unburnt fuel. In the idle and part-load ranges, this creates an altogether highly lean mixture and thus a reduction in the fuel consumption.

Homogeneous operation

In homogeneous operation, as with external mixture formation, there is a homogeneous mixture throughout the entire combustion chamber; and the entire fresh air available in the combustion chamber takes part in the combustion procedure. For this reason, this operating mode is used in the full-load range.

MED-Motronic is the management system for direct-injection gasoline engines.

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Single-point injection systems

Single-point fuel injection has advanced beyond the compact fuel-injection system stage to become part of a comprehensive engine-management system.

The various single-point injection systems differ in the design of the central-injection unit. All systems feature an injector located above the throttle plate; they differ from multipoint injection units in that they frequently operate at low pressure (0.7...1 bar). This means that an inexpensive, hydrodynamic electric fuel pump can be used which is generally in the form of an in-tank unit. The injector is flushed continuously by the fuel flowing through it in order to inhibit the formation of air bubbles. This arrangement is an absolute necessity in such a low-pressure system. The designation "Single-Point Injection (SPI)" corresponds to the terms Central Fuel Injection (CFI), Throttle Body Injection (TBI) and Mono-Jetronic (Bosch).

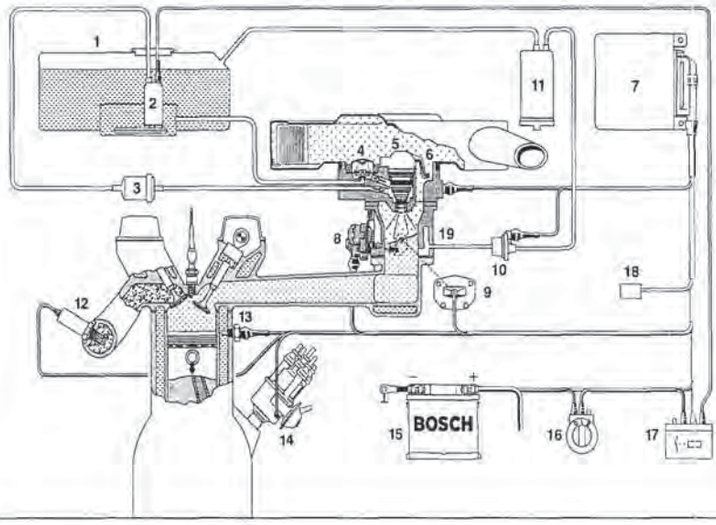
Mono-Jetronic

Mono-Jetronic is an electronically controlled, low-pressure single-point injection system for 4-cylinder engines, and features a centrally located solenoid-controlled fuel injector. At the heart of the system is the central injection unit, which uses the throttle valve to meter the intake air while injecting the fuel intermittently above the throttle valve. The intake manifold then distributes the fuel to the individual cylinders. Various sensors monitor all important engine operating data, which are then are used to calculate the triggering signals for the injectors and other system actuators.

Central injection unit

The injector is located above the throttle, in the intake-air path, in order to ensure homogeneous mixtures and consistent cylinder-to-cylinder distribution. The fuel spray is directed into the sickle-shaped orifice between the housing and throttle plate, whereby fuel wetting of the intake-tract walls is inhibited to a great extent, and the high pressure differential pro-

Schematic of a Mono-Jetronic system
1 Fuel tank, 2 Electric fuel pump, 3 Fuel filter, 4 Pressure regulator, 5 Injector, 6 Air-temperature sensor, 7 ECU, 8 Throttle actuator, 9 Throttle potentiometer, 10 Canister-purge valve, 11 Carbon canister, 12 Lambda sensor, 13 Coolant-temperature sensor, 14 Ignition distributor, 15 Battery, 16 Ignition switch, 17 Relay, 18 Diagnostic connector, 19 Central injection unit.



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motes optimum mixture formation. The injector operates at a system pressure of 1 bar (referred to atmospheric pressure). Efficient fuel atomization ensures consistently good mixture distribution, even in the critical full-load range. Injector triggering is synchronized with the ignition pulses.

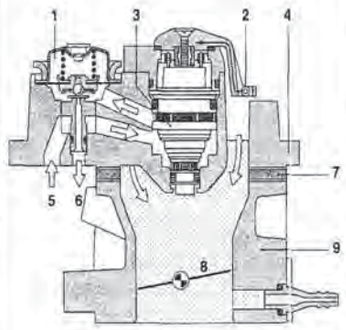
System control

In addition to the engine speed n, the main actuating variables for the injection system can include the air volume/air mass flow, the absolute manifold pressure, and the throttle position α. The (α/n) system applied with Mono-Jetronic can meet stringent emission requirements when used in conjunction with lambda closed-loop control and a 3-way catalytic converter. A self-adaptive system employs the signal from the lambda sensor as a reference to compensate for component tolerances and engine changes, thus maintaining high precision throughout the service life of the system.

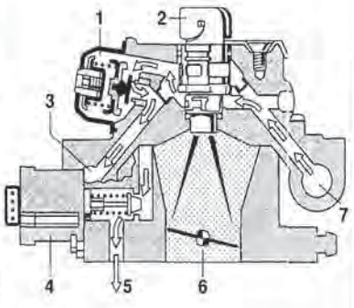
Adaptation functions

The injection time is extended to provide additional fuel for cold starts and during the post-start and warm-up phases. When the engine is cold, the throttle actuator adjusts the throttle position to supply more air to the engine, thus maintaining idle speed and exhaust emissions at a constant level. The throttle potentiometer recognizes the change in throttle position and initiates an increase in the fuel quantity via the ECU. The system regulates the enrichment for acceleration and full-throttle operation in the same way. The overrun fuel cutoff provides reductions in fuel consumption and in exhaust emissions during trailing-throttle operation. Adaptive idle-speed control lowers the idle speed and stabilizes it. For this purpose, the ECU issues a signal to the servomotor to adapt the throttle-valve position as a function of engine speed and temperature.

Mono-Jetronic central injection unit
1 Pressure regulator, 2 Air-temperature sensor, 3 Injector, 4 Upper part (hydraulics), 5 Fuel supply, 6 Fuel return, 7 Insulator plate, 8 Throttle valve, 9 Lower part.



Multec central injection unit (Opel)
1 Pressure regulator, 2 Injector, 3 Fuel return, 4 Stepper motor for idle-speed control, 5 To intake manifold, 6 Throttle valve, 7 Fuel inlet.



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Multipoint injection systems

K-Jetronic

Operating principle
– Continuous injection,
– Direct air-flow measurement.

K-Jetronic is a mechanical system which does not require an engine-driven injection pump. It meters a continuous supply of fuel proportional to the quantity of air being drawn into the engine.

Because of direct air-flow measurement, K-Jetronic also takes into account changes caused by the engine and permits the use of emission-control equipment, for which precise intake-air monitoring is an essential requirement.

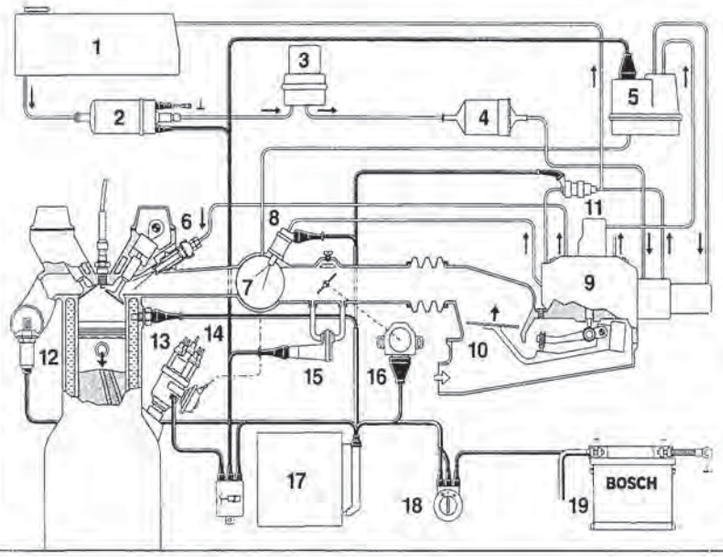
Operation
The intake air flows through the air filter, the air-flow sensor, and the throttle valve, before entering the intake manifold and continuing to the individual cylinders.

The fuel is delivered from the fuel tank by an electric (roller-cell) fuel pump. It then flows through the fuel accumulator and fuel filter to the fuel distributor. A pressure regulator in the fuel distributor maintains the fuel at a constant system pressure. The fuel flows from the fuel distributor to the injectors. Excess fuel not required by the engine is returned to the tank.

Mixture-control unit
The mixture-control unit consists of the air-flow sensor and the fuel distributor.

Air-flow sensor
The air-flow sensor consists of an air funnel and a pivoting air-flow sensor plate . A counterweight compensates for the weight of the sensor plate and pivot as-

Schematic of a K-Jetronic system
1 Fuel tank, 2 Electric fuel pump, 3 Fuel accumulator, 4 Fuel filter, 5 Warm-up regulator, 6 Injector, 7 Intake manifold, 8 Electric start valve, 9 Fuel distributor, 10 Air-flow sensor, 11 Frequency valve, 12 Lambda sensor, 13 Thermo-time switch, 14 Ignition distributor, 15 Auxiliary-air valve, 16 Throttle switch, 17 ECU, 18 Ignition switch, 19 Battery.

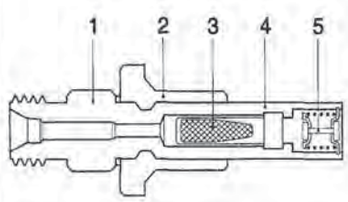


sembly. The sensor plate is displaced by the air flow, while the control plunger in the fuel distributor exerts hydraulic counterpressure to maintain the system in a balanced state. The position of the airflow sensor plate provides an index of intake air flow, and is transmitted to the fuel distributor's control plunger by a lever.

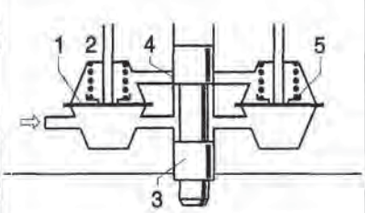
Fuel distributor

The amount of fuel supplied to the individual cylinders is regulated by varying the aperture of the metering slots in the fuel-distributor barrel. The number of rectangular-shaped metering slots in the barrel corresponds to the number of engine cylinders. The specific size of the metering-slot aperture depends on the control-plunger's position. In order to ensure constant pressure drop at the slots for various flow rates, a differential pressure regulator is located downstream of each metering slot.

Fuel injector
1 Hexagon, 2 Rubber molding, 3 Fine-mesh strainer, 4 Valve body, 5 Valve needle.



Fuel distributor in mixture-control unit
1 Diaphragm, 2 To injector, 3 Control plunger, 4 Metering slot, 5 Differential-pressure regulator.



Injector

The injector opens automatically at a pressure of approximately 3.8 bar, and has no metering function. It provides efficient mixture formation by opening and closing at a frequency of approx. 1500 Hz ("chatter").

It is held in place by a rubber molding. It is pressed, not screwed, into position. The hexagon serves to brace the injector when the fuel-supply line is screwed on.

Warm-up regulator

The warm-up regulator is controlled by an electrically-heated bimetallic element; it enriches the mixture in the warm-up phase by reducing the counterpressure (control pressure) exerted against the control plunger. A reduction in this control pressure means that the stroke of the airflow sensor plate for a given air flow increases (reflected by a correspondingly larger metering-slot aperture). The result is a richer mixture during warm-up.

Where desired, the warm-up regulator can be expanded to incorporate the following functions:
- full-throttle enrichment,
- acceleration enrichment,
- altitude compensation.

Auxiliary-air valve

The auxiliary-air valve, controlled by either a bimetallic spring or an expansion element, supplies the engine with additional air (which is monitored by the airflow sensor, but bypasses the throttle valve) during the warm-up phase. This supplementary air compensates for the cold engine's higher friction losses; it either maintains the normal idle speed or increases it in order to heat the engine and exhaust more quickly.

Electric start valve, thermo-time switch

The thermo-time switch activates the electric start valve as a function of engine temperature and elapsed time. During low-temperature starts, the start valve injects supplementary fuel into the intake manifold (cold-start enrichment).

Lambda closed-loop control

Open-loop control systems do not regulate the A/F ratio with enough accuracy to

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allow compliance with stringent emissions limits.

Lambda closed-loop control is required for operation of the 3-way catalytic converter. When it is installed, the K-Jetronic system must include an electronic control unit which uses the Lambda sensor's signal as its main input variable.

A solenoid frequency valve regulates the A/F mixture ratio by controlling the pressure differential at the metering slots. However, this principle cannot be applied to meet the more stringent emissions requirements scheduled for the future.

KE-Jetronic

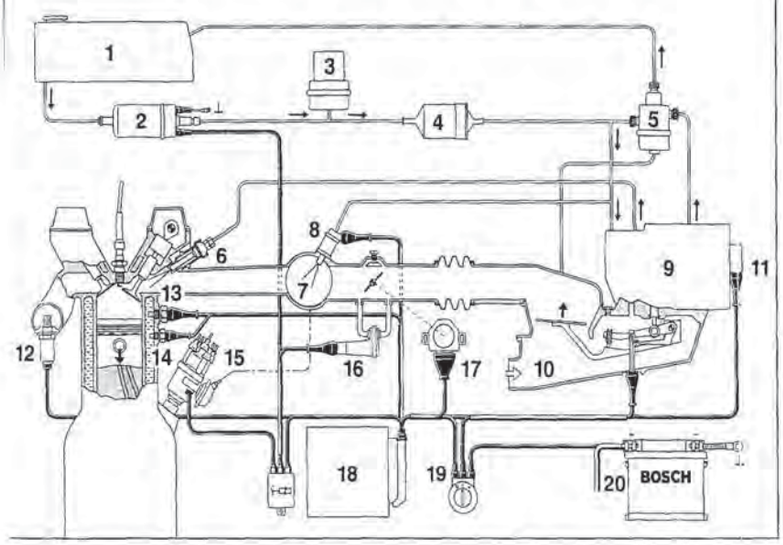
KE-Jetronic is an advanced version of the K-Jetronic system. KE-Jetronic includes an ECU for increased flexibility and supplementary functions. Additional components include:

- a sensor for the intake air flow,
- a pressure actuator for mixture ratio adjustment, and
- a pressure regulator which maintains system pressure at a constant level as well as providing a fuel-cutoff function when the engine is switched off.

Operation

An electric fuel pump generates the system pressure. The fuel flows through the fuel distributor, while a diaphragm regulator maintains the system pressure at a constant level. With K-Jetronic, the control circuit performs mixture corrections via the warm-up regulator. In contrast, with KE-Jetronic the primary pressure and the pressure exerted upon the control plunger are equal. The ratio is corrected by adjusting the pressure differential in all the fuel distributor's chambers simultaneously.

Schematic of a KE-Jetronic system
1 Fuel tank, 2 Electric fuel pump, 3 Fuel accumulator, 4 Fuel filter, 5 Fuel-pressure regulator, 6 Injector, 7 Intake manifold, 8 Electric start valve, 9 Fuel distributor, 10 Air-flow sensor, 11 Electrohydraulic pressure actuator, 12 Lambda sensor, 13 Thermo-time switch, 14 Coolant-temperature sensor, 15 Ignition distributor, 16 Auxiliary-air valve, 17 Throttle switch, 18 ECU, 19 Ignition switch, 20 Battery.

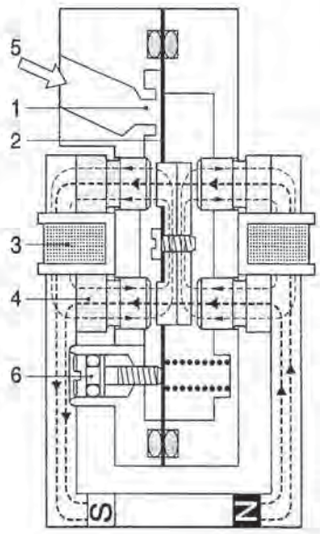


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The system pressure is present upstream from the metering slots, and applies a counterpressure to the control plunger. As with K-Jetronic, the control plunger is moved by an air-flow sensor flap. A damper unit prevents the oscillations that could be induced by the forces generated at the sensor flap. From the control plunger the fuel flows through the pressure actuator, the lower chambers of the differential-pressure valve, a fixed flow restrictor, and the pressure regulator, before returning to the fuel tank. Together with the flow restrictor, the actuator forms a pressure divider in which the pressure can be adjusted electrodynamically. This pressure is present in the lower chambers of the differential-pressure valves.

A pressure drop corresponding to the actuator current occurs between the actuator's two connections. This causes variations in the pressure differential at the metering slots, and alters the amount of fuel injected. The current can also be reversed to shut down the fuel supply completely. This feature can be employed for such functions as overrun fuel cutoff and engine-speed limitation.

Electrohydraulic pressure actuator
1 Nozzle, 2 Valve plate, 3 Coil, 4 Magnetic pole, 5 Fuel inlet, 6 Adjustment screw.



Electrohydraulic pressure actuator
This electrohydraulic actuator is flange-mounted on the fuel distributor. It is an electrically-controlled pressure regulator which operates using the nozzle/flapper-plate system. The mixture enrichment is directly proportional to the current flow.

Electronic control unit (ECU)
The ECU processes signals from the ignition (engine speed), temperature sensor (coolant temperature), throttle potentiometer (intake air flow), throttle switch (idle and overrun, WOT), starter switch, Lambda (O_2) sensor, pressure sensor and other sensors. Its most important functions are the control of:
- starting and post-start enrichment,
- warm-up enrichment,
- acceleration enrichment,
- full-throttle enrichment,
- overrun fuel cutoff,
- engine-speed limitation,
- idle-speed control,
- altitude compensation,
- closed-loop lambda control.

A coding switch (trim plug) makes it possible to select between operation with lambda control (with catalytic converter) and without it. This permits a choice between leaded and unleaded gasoline.

Lambda closed-loop control
The signal from the Lambda sensor is processed in the KE-Jetronic's ECU. The pressure actuator carries out the necessary adjustments.

L-Jetronic

Operating principle
- Air-flow measurement,
- Main controlled variables: air flow and engine speed,
- Intermittent injection.

L-Jetronic combines the advantages of direct air-flow measurement with the unique

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possibilities afforded by electronics. It is similar to K-Jetronic in that it recognizes all changes in engine condition (due to wear, combustion-chamber deposits, changes in valve setting). This ensures consistently good exhaust-gas composition.

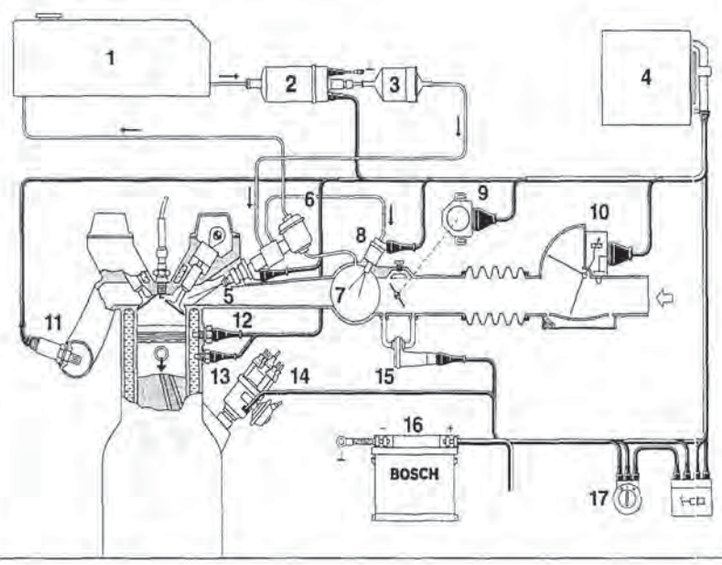
Operation

The fuel is injected through the engine's solenoid-operated injectors. A solenoid valve assigned to each cylinder is triggered once per crankshaft revolution. All of the injectors are wired in parallel to reduce the complexity of the electrical circuit. The pressure differential between fuel and intake-manifold pressures is maintained at a constant level of 2.5 or 3 bar such that the injected fuel quantity is only dependent on the opening period of the valves. For this purpose, the ECU delivers control pulses whose duration is dependent on the intake air flow, the engine speed, and other influencing variables. These are monitored by sensors and processed in the ECU.

Fuel supply

An electric fuel pump supplies the fuel and generates the injection pressure. The fuel is pumped from the fuel tank, through a paper filter, and into a high-pressure supply line at the other end of which there is a pressure regulator (spring-loaded diaphragm). The pressure regulator maintains a constant pressure at the metering orifice, regardless of the injected fuel quantity.

Schematic of an L-Jetronic system
1 Fuel tank, 2 Electric fuel pump, 3 Fuel filter, 4 ECU, 5 Injector, 6 Fuel-pressure regulator, 7 Intake manifold, 8 Electric start valve, 9 Throttle switch, 10 Air-flow sensor, 11 Lambda sensor, 12 Thermo-time switch, 13 Coolant-temperature sensor, 14 Ignition distributor, 15 Auxiliary-air valve, 16 Battery, 17 Ignition switch.

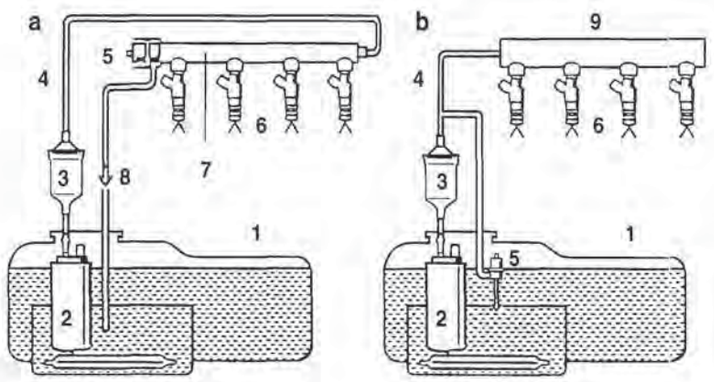


Standard system:
The fuel flows through the high-pressure line on its way to the engine-mounted fuel rail with the injectors. The pressure regulator is installed on the fuel rail. After flowing through the rail the portion of the fuel not required by the engine flows through the return line attached to the regulator and back to the tank. Because the returning fuel has been warmed on the way back from the engine, fuel temperatures within the tank rise.

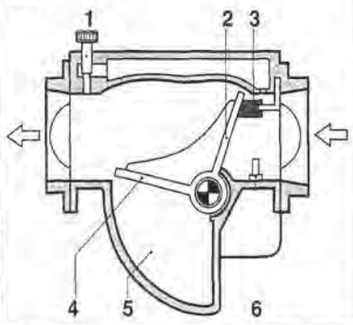
Fuel vapor is generated in the tank as a function of fuel temperature. For environmental purposes these vapors are routed through the tank ventilation system for storage in an activated charcoal canister until they can be returned through the intake manifold for combustion within the engine.

Returnless system:
A returnless fuel-supply system reduces the tendency of the fuel in the fuel tank to heat up, thus making it easier to comply with the legal requirements governing vehicular evaporative emissions. The pressure regulator is located either in the fuel tank or in its immediate vicinity, which means that the return line from the engine to the fuel tank can be dropped. The amount of fuel pumped to the fuel rail is limited to the quantity being used by the injectors. The excess flow volume emerging from the pump returns directly to the tank without the round trip to the engine and back. Assuming equivalent operating conditions, and depending upon the specific vehicular application, this system can reduce in-tank fuel temperatures by up to 10 K, cutting vaporization by roughly one third.

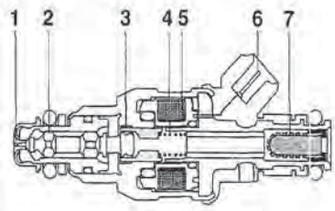
Fuel supply
a) Standard system, b) Returnless system.
1 Fuel tank, 2 Electric fuel pump, 3 Fuel filter, 4 Pressure line, 5 Fuel-pressure regulator, 6 Injectors, 7 Fuel rail (continuous flow), 8 Return line, 9 Fuel rail (no return flow).



Air-flow sensor
1 Idle-mixture adjustment screw, 2 Air-flow sensor flap, 3 Stop, 4 Compensation flap, 5 Damping chamber, 6 Air-temperature sensor.



Injector
1 Pintle, 2 Needle, 3 Armature, 4 Spring, 5 Solenoid winding, 6 Electrical terminals 7 Fuel strainer.



Air-flow sensor
The intake air flow deflects a sensor flap against the constant return force of a spring to a defined angular position, which is converted by a potentiometer into a voltage ratio. This voltage ratio determines the pulse length of a timing element in the ECU. A temperature sensor in the air-flow sensor indicates changes in air density caused by temperature variations.

Fuel injectors
Fuel injectors serve to meter and atomize the fuel. When the solenoid winding is energized, the nozzle needle is lifted a mere 0.05 mm off its seat.

Throttle-valve switch
This transmits a control signal to the ECU when the throttle valve is either completely closed (idle) or fully opened (full-throttle [WOT]).

Engine-temperature sensor
The engine-temperature sensor is designed as a temperature-sensitive resistor (thermistor) and controls the warm-up enrichment.

Auxiliary-air valve, electric start-valve, thermo-time switch
Design and function are similar to those of the corresponding K-Jetronic components.

Electronic control unit (ECU)
This ECU converts the engine variables into electrical pulses. Transmission intervals for these pulses are correlated with ignition timing, while their duration is basically a function of speed and intake air flow. Since all injectors are activated simultaneously, only a single driver stage is required. The temperature sensors respond to lower engine and air temperatures by increasing the injection duration. The throttle-switch signals allow mixture adaptation for idle and full-throttle operation.

Lambda closed-loop control
The ECU compares the signal from the Lambda sensor with a setpoint value before activating a two-state controller. The control adjustment is then performed, as are all corrections, by modifying the injection duration.

L3-Jetronic
L3-Jetronic incorporates functions extending beyond those provided by the L-Jetronic's analog technology. The L3 system's ECU employs digital technology to adjust the mixture ratios based on a load/engine-speed map. In order to save space, the ECU is installed in the engine compartment, directly on the air-flow sensor, where the two components form a single monitoring and control unit.

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LH-Jetronic

LH-Jetronic is closely related to L-Jetronic. The difference lies in the method of intake air-flow measurement, with LH-Jetronic using a hot-wire air-mass meter to measure the mass of the intake air. Thus, the results no longer depend on the air density, which varies with temperature and pressure.

The other LH-Jetronic components and the basic system concept are to a large extent the same as those in L-Jetronic.

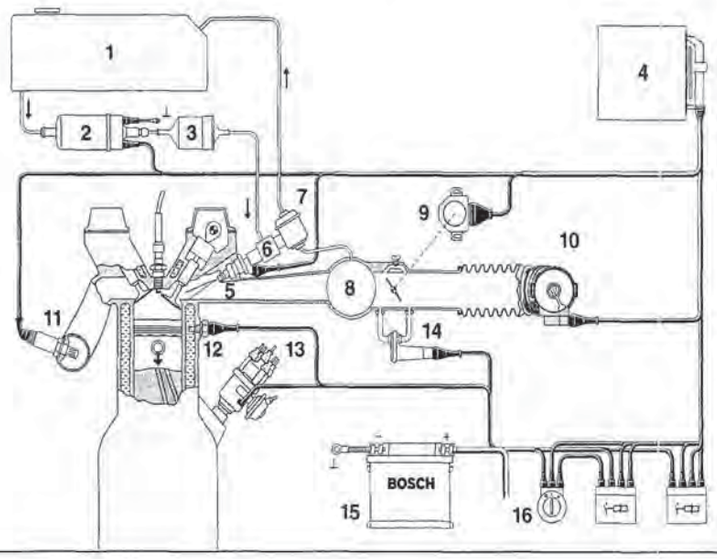
Operating-data processing in the ECU

LH-Jetronic is equipped with a digital ECU. Arrangements for adjusting the mixture ratio vary from those used with L-Jetronic in using a load/engine-speed map programmed for minimum fuel consumption and exhaust emissions. The ECU processes the sensor signals when calculating the injection duration that determines the injected fuel quantity. The ECU includes a microprocessor, a program and data memory, and an A/D converter. The microprocessor is provided with a suitable voltage supply and with a stable clock rate for data processing. The clock rate is defined by a quartz oscillator.

Hot-wire air-mass flow meter

The stream of intake air is conducted past a heated wire (hot wire). This wire forms part of an electrical bridge circuit. The flow of current through the wire serves to maintain it at a constant temperature above that of the intake air. This principle makes it possible to employ the current requirement as an index of the air mass being drawn into the engine. A resistor converts the heating current into a voltage signal, which the ECU then processes along with engine speed as a main input variable. A temperature sensor is mounted in the hot-wire air-mass flow meter to ensure that its output signal is not influenced by the temperature of the intake air. The A/F ratio at idle can be adjusted with a potentiometer. As contamination on

Schematic of an LH-Jetronic system
1 Fuel tank, 2 Electric fuel pump, 3 Fuel filter, 4 ECU, 5 Injector, 6 Fuel distributor, 7 Fuel-pressure regulator, 8 Intake manifold, 9 Throttle switch, 10 Hot-wire air-mass flow meter, 11 Lambda sensor, 12 Coolant-temperature sensor, 13 Ignition distributor, 14 Idle-speed actuator, 15 Battery, 16 Ignition switch.



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the surface of the hot wire could affect the output signal, each time the engine is shut down the wire is electrically heated for one second to burn-off any contamination. The hot-wire air-mass flow meter has no moving parts, and its aerodynamic resistance within the intake tract is negligible.

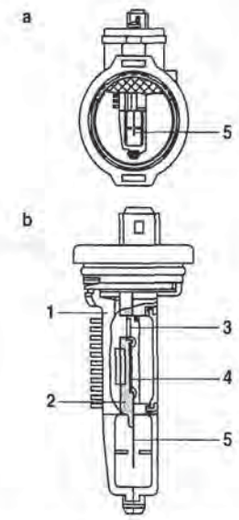
Hot-film air-mass flow meter

The operating principle of the hot-film air-mass flow meter is the same as that of the hot-wire sensor. However, in the interests of simplified design, a substantial portion of the electrical bridge circuit is installed on a ceramic substrate, in the form of thin-film resistors. In addition, there is no need to burn contaminants off the film. The contamination problem is solved by placing the areas on the sensor element which are decisive for thermal transmission at a downstream location. This prevents them from being affected by deposits on the sensor element's leading edge.

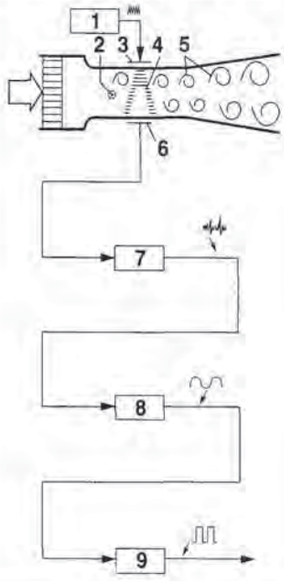
Kármán vortex volumetric flow meter

Yet another option for measuring intake air is provided by a sensor which uses the Kármán vortex principle to measure the volumetric flow rate. This meter monitors vortices generated as the intake air flows past vortex generators. The frequency of these vortices is a measure of the volumetric flow rate. This frequency is measured by emitting ultrasonic waves perpendicular to the direction of the intake-air flow. The propagation velocity of these waves as modified by the vortices is detected by an ultrasonic receiver and the resulting signals are evaluated in the ECU.

Hot-film air-mass flow meter
a Housing, b Hot-film sensor (installed in center of housing). 1 Heat sink, 2 Spacer, 3 Driver stage, 4 Hybrid, 5 Sensor element (metallic film).



Kármán vortex volumetric flow meter
1 Oscillator, 2 Vortex generator, 3 Transmitter, 4 Ultrasonic waves, 5 Eddy currents, 6 Receiver, 7 Amplifier, 8 Filter, 9 Pulse shaper.



Electromagnetic fuel injectors

Design and operation

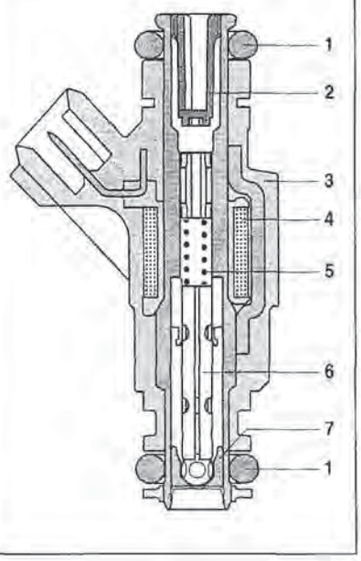
Fuel injectors essentially consist of a valve housing with current coil and electrical connection, a valve seat with spray-orifice disk and a moving valve needle with solenoid armature.

A filter strainer in the fuel feed protects the injector against contamination. Two O-rings seal the injector against the fuel-distribution pipe and the intake manifold. When the coil is de-energized, the spring and the force resulting from the fuel pressure press the valve needle against the valve seat to seal the fuel-supply system against the intake manifold (see Fig.).

When the injector is energized, the coil generates a magnetic field which attracts the armature and lifts the valve needle off of its seat to allow fuel to flow through the injector.

The injected fuel quantity per unit of time is essentially determined by the system pressure and the free cross-section of the spray orifices in the spray-orifice disk. The valve needle closes again when the field current is switched off.

Fuel injector EV6 *(example)*
1 O-rings, 2 Filter strainer, 3 Valve housing with electrical connection, 4 Current coil, 5 Spring, 6 Valve needle with solenoid armature, 7 Valve seat with spray-orifice disk.



Spray formation

The fuel injectors' spray formation, i.e. spray shape, spray angle and droplet size, influences the formation of the A/F mixture. Individual geometries of intake manifold and cylinder head make it necessary to have different types of spray formation.

Tapered spray:

Individual fuel sprays emerge through the openings in the spray-orifice disk. These fuel sprays combine to form a tapered spray. Tapered sprays can also be obtained by means of a pintle projecting through the injector needle tip. Tapered-spray injectors are typically used in engines with one intake valve per cylinder. The tapered spray is directed into the opening between the intake-valve disk and the intake-manifold wall.

Dual spray:

Dual-spray formation is used in engines with two intake valves per cylinder. The openings in the spray-orifice disk are arranged in such a way that two fuel sprays emerge from the injector. Each of these sprays supplies an intake valve.

Air-shrouding:

In the case of an air-shrouded injector, the pressure drop between intake-manifold and ambient pressures is used to improve mixture formation. Air is routed through an air-shrouding attachment into the outlet area of the spray-orifice disk. In the narrow air gap, the air is accelerated to a very high speed and the fuel is finely atomized when it mixes with it.

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Ignition

The ignition system's function is to initiate combustion in the flammable air-fuel mixture by igniting it at precisely the right moment. In the spark-ignition (Otto) engine, this is achieved with an electrical spark, i.e. an arc generated between the spark plug's electrodes. Consistently reliable ignition under all circumstances is essential for ensuring fault-free catalytic-converter operation.

Misfiring results in damage to or destruction of the catalytic converter due to overheating during afterburning of the uncombusted mixture.

Mixture ignition

Provided the composition of the mixture is stoichiometric, an energy of approximately 0.2 mJ is required for each individual ignition of the A/F mixture via electric spark. Over 3 mJ are required for a rich or lean mixture. This energy represents only a fraction of the total energy in the ignition spark, the actual ignition energy. If sufficient ignition energy is not available, there will be no ignition, the mixture cannot ignite, and misfiring will result.

The system must therefore deliver enough ignition energy to ensure consistently reliable ignition of the mixture, even under unfavorable conditions. Igniting a small flammable mixture cloud flowing past the spark can be enough to initiate the process.

This mixture cloud ignites, the flame spreads to the remaining mixture in the cylinder, and the fuel starts to combust. Ignitability is enhanced by efficient fuel atomization and good access of the mixture to the electrodes, as well as through extended spark duration and spark length (large electrode gap).

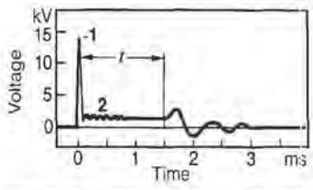
The spark plug determines the location and length of the spark; spark duration depends upon the type and design of the ignition system, as well as on the momentary ignition conditions.

Spark generation

Adequate voltage must be present before a spark will arc from one electrode to another. At the moment of ignition, the voltage across the electrodes abruptly rises from zero up to the arcing (ignition) voltage and the plug fires. Once the spark has ignited, the spark-plug voltage drops to the sparking voltage. The A/F mixture may ignite at any point during the firing period of the ignition spark (spark duration). Once the spark has broken away, the voltage is damped and drops to zero.

Although intense mixture turbulence is basically desirable, it can extinguish the spark, thus leading to incomplete combustion. The energy stored in the ignition coil should therefore suffice for one or more consecutive sparks, depending on individual requirements.

Spark-plug voltage characteristic with stationary or semi-stationary A/F mixture
1 Ignition voltage, 2 Spark voltage, t Spark duration.



High-voltage generation and energy storage

Battery-ignition systems generally employ an ignition coil to generate the high-tension voltage needed to generate the spark. The ignition coil operates as an autotransformer but within coil-ignition systems it also assumes the further important function of storing the ignition energy. When the contact breaker closes, energy from the vehicle's electrical system flows into the coil's primary winding. This energy is then stored in a magnetic field until the firing point, when the secondary winding discharges it to one of the engine's spark plugs. The ignition coil is designed to ensure that the available high-tension voltage in the coil is always well in excess of the spark plug's maximum-possible ignition-voltage requirement. Energy levels of 60...120 mJ within the coil correspond to an available voltage of 25...30 kV.

The operational reserves of high volt-

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age and ignition energy are sufficient to compensate for all electrical losses. Inadequate maintenance reduces these high-voltage reserves, and leads to ignition and combustion miss. Engine power drops and fuel consumption increases. In addition, this phenomenon can result in damage to or destruction of the catalytic converter, should one be installed. In extreme cases, the engine either fails to start – especially when cold – or stalls.

Ignition systems are also available with capacitive energy storage (CDI or Capacitor-Discharge Ignition) for use on high-performance and racing engines. These systems store the ignition energy in the electrical field of a capacitor before a special transformer transmits it to the spark plug in the form of a high-voltage ignition pulse.

Ignition timing and adjustment

Approximately two milliseconds elapse between the mixture's initial ignition and its complete combustion. The ignition spark must therefore arc early enough to ensure that main combustion, and thus the combustion-pressure peak in the cylinder, occur shortly after piston TDC. The ignition angle should therefore move further in the advance direction along with increasing engine speed. The chosen firing point should ensure that the following requirements are met:

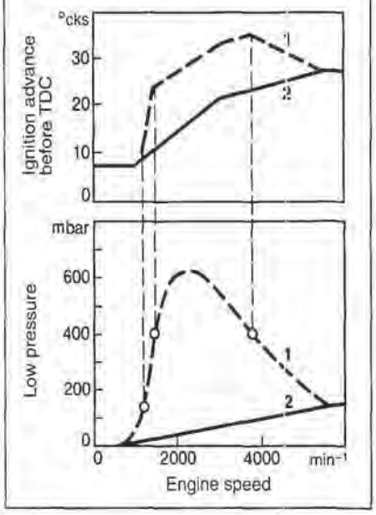
- maximum engine performance
- low fuel consumption
- no engine knock
- clean exhaust gas.

Since it is impossible to obtain optimal compliance with all of these requirements simultaneously; compromises must be found on a case-to-case basis. Optimal ignition timing is defined according to a variety of parameters. The most important are engine speed, engine load, engine design, fuel quality and momentary operating conditions (starting, idle and trailing throttle, etc.). In the simplest case, spark-advance mechanisms sensitive to variations in engine speed and intake-manifold vacuum adapt the ignition timing to suit the engine's current operating conditions.

In modern engine-management systems with extended functions, additional

Example of cumulative ignition timing consisting of centrifugal and vacuum advance

1 Part-load operation, 2 Full load.



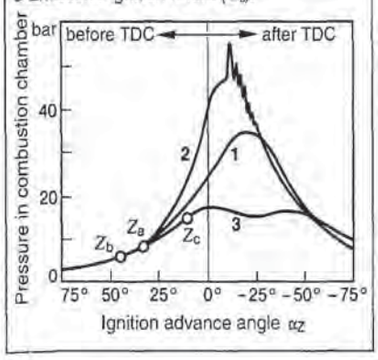
adjustments can be used e.g. for rapid torque adaptation or for swift heating of the catalytic converter.

All the adjustment strategies can operate either individually or simultaneously. The degree to which the ignition timing is advanced or retarded is determined by the ignition-advance curves calibrated specifically for each individual engine configuration.

At full load, the accelerator pedal is depressed fully and the throttle is wide open (WOT). Along with increasing engine speeds, ignition takes place earlier in order to maintain the combustion pressure at the levels required for optimal engine performance. The leaner A/F mixtures encountered during part-throttle operation are more difficult to ignite. Because this means that more time is required for ignition, it must be triggered earlier, with the timing being shifted further in the "advance" direction. The manifold vacuum employed to determine the necessary degree of spark advance is monitored downstream from the throttle valve. If the vacuum bore is located near the throttle valve (P. 489), the vacuum initially increases as the throttle is opened wider and begins to

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Combustion-pressure curve for various ignition firing points
1 Correct ignition advance (Z_a),
2 Excessive ignition advance (Z_b),
3 Excessive ignition retard (Z_c).



fall in the proximity of the full-throttle (WOT) position. The progressively wider throttle openings required to increase engine speed on the operating curve for part-throttle road operation are reflected in the relationship between vacuum and min^{-1} shown in the diagram.

Yet another diagram shows the curves for combustion-chamber pressure in a 4-stroke engine with correct and incorrect ignition timing. Even if the timing is initially correct, neglected maintenance can allow it to drift over the course of time. If the timing shifts towards a later firing point ("retard"), the result is a gradual drop in engine power and increased fuel consumption. Excessive "advance" may result in extreme cases in serious damage to spark plugs or to the engine if the engine knocks. The level of exhaust emissions also increases.

Ignition and emissions

Owing to the fact that it directly affects the various exhaust-gas components, the ignition has a significant effect upon exhaust emissions. Because various – and in this context sometimes mutually antagonistic – factors such as fuel economy, driveability, etc., are also potential optimization criteria, it is not always possible to specify the ideal ignition timing for minimum emissions.

Shifts in ignition timing induce mutually inverse response patterns in fuel consumption and exhaust emissions: While more spark advance increases power and reduces fuel consumption, it also raises HC and, in particular, NO_x emissions. Excessive spark advance can cause engine knock and lead to engine damage. Retarded ignition results in higher exhaust-gas temperatures, which can also harm the engine. Electronic engine-management systems featuring programmed ignition curves are designed to adapt ignition timing in response to variations in factors such as min^{-1}, load, temperature, etc. They can thus be employed to achieve the optimum compromise between these mutually antagonistic objectives.

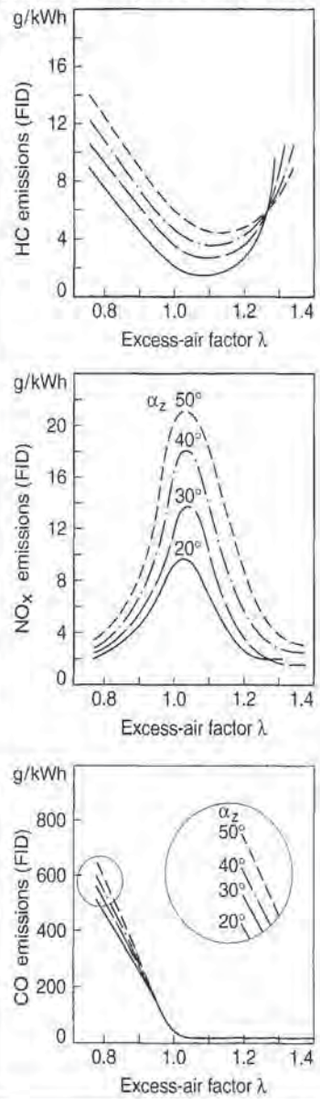
Overview of various ignition systems

Function	Ignition system			
	CI	TI	EI[1])	DLI
Designation	Coil ignition system	Transistorized ignition system	Engine management	
			Electronic ignition system	Distributorless semiconductor ignition system
High-voltage generation	Inductive			
Ignition triggering	Mechanical	Electronic		
Ignition angle determined from engine speed and load	Mechanical		Electronic	
Spark distribution to appropriate cylinder	Mechanical			Electronic

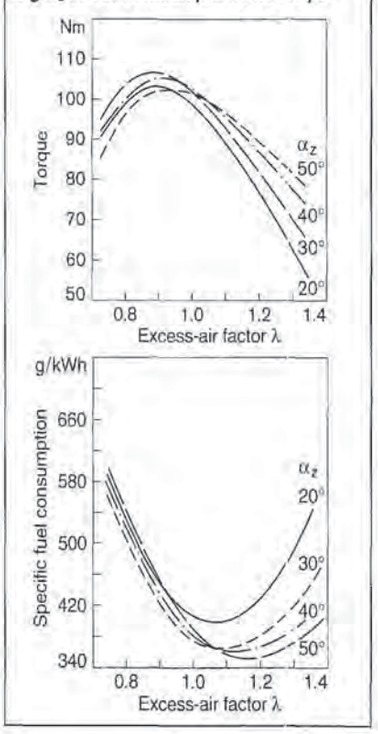
[1]) Any desired ignition timing is not possible with EI. DLI therefore dominates with integrated engine-management systems.

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Influence of excess-air factor λ and ignition point α_z on pollutant emissions



Influence of air factor λ and ignition timing α_z on fuel consumption and torque



Ignition energy

The ignition system generates a high-voltage spark at the spark plug to initiate combustion. An ignition-spark energy of approx. 0.2 mJ is adequate to ignite a stoichiometric air-fuel mixture, while richer or leaner mixtures require substantially higher levels of spark energy. Excess energy, i.e., from an ignition system designed to generate a high-energy spark of extended duration (transistorized or electronic ignition) stabilizes flame propagation and reduces the fluctuations from cycle to cycle. The reduction in fluctuations results in smoother engine operation and lower HC emissions. Increased spark projection, larger electrode gaps and thin electrodes also have a positive influence on the engine's smoothness and HC emissions.

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Ignition coil

The ignition coil functions as both an energy-storage device and a transformer. The coil, which is powered by DC voltage from the vehicle's electrical system, supplies the ignition pulses for the spark plugs at the required high voltage and discharge energy. The ignition driver stage with its defined deactivation current combines with a primary winding featuring specific resistance and inductance characteristics to determine the amount of energy stored within the ignition coil's magnetic field. The secondary winding can be designed to provide peak voltage, spark current and discharge duration in accordance with individual requirements.

The contact-breaker points used with coil ignition (CI) can only handle interrupt currents of up to approx. 5 A. TI, EI and DLI ignition systems and Motronic ECUs can handle much higher interrupt currents. The series resistors generally employed with coil ignition (they can be bypassed to increase energy during cold starts) can be omitted in electronic ignition systems. Here the electronic circuitry activates the ignition coil depending on battery voltage, engine speed and other influencing variables in such good time that full energy is available at the ignition point.

Each ignition coil is designed to meet the requirements of a particular application. It must charge quickly in order to furnish the voltages and ignition energies required at high engine speeds. Important priorities thus include low primary inductance and, in some cases, higher primary interrupt currents (for adequate energy storage).

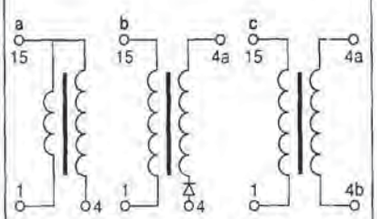
Design and operation

Traditional ignition coils with asphalt or oil insulation enclosed in metal casings are being increasingly replaced by units featuring an epoxy-resin filler. These not only allow more latitude in the selection of geometry, type and number of electrical terminals, but also provide more compact dimensions, better vibration resistance and lower weight. The ignition coil is generally attached by way of the iron core to the engine or vehicle body. Rod-type ignition coils are installed in the cylinder-head recess above the spark plug.

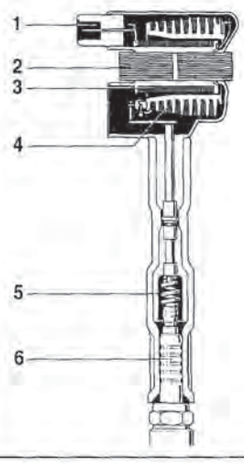
The coil's synthetic materials provide good adhesion between all of the high-voltage components and the molded epoxy resin, which penetrates into all the capillary spaces. Supplementary iron cores are sometimes embedded on the inside of the synthetic molding.

The secondary winding is mostly designed as a disk or sandwich coil, with the

Ignition coils (schematic)
Rotating distribution: a) Single-spark ignition coil. Distributorless ignition: b) Single-spark ignition coil, c) Dual-spark ignition coil.



Single-spark ignition coil
1 External low-voltage terminal, 2 Laminated iron core, 3 Primary winding 4 Secondary winding, 5 Internal high-voltage connection via spring contact, 6 Spark plug.



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windings distributed among a series of segments. Even distribution of stresses among the insulating elements in all chambers combines with high dielectric strength to permit compact dimensions while at the same time making foil and paper between wire layers redundant. The winding's self-capacitance is also reduced.

Because lower breakdown voltages are required for the negative (relative to engine ground) ignition spark, the positive terminals for the primary and secondary windings are generally combined on those ignition coils used with rotating high-voltage distribution.

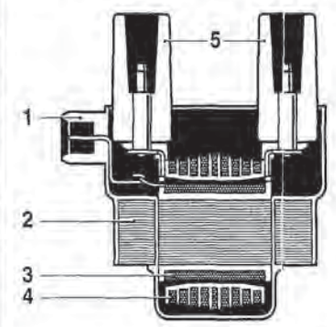
Single and dual-spark ignition coils are an alternative for use in ignition systems with distributorless ignition (DLI).

When a single-spark coil per spark plug is used, the primary current is controlled to furnish the relevant spark plug with an ignition pulse at precisely the right moment in time. High-voltage diodes are used to prevent the positive 1...2 kV high-voltage pulse generated when the primary current is activated from causing the spark plug to fire prematurely.

On the dual-spark coil, the secondary winding is galvanically insulated from the primary winding. Each of the two high-voltage outputs are connected to a spark plug. Ignition sparks are created at the two spark plugs when the primary current is deactivated. As with rotating high-voltage distribution, this system does not usually require any special precautions to prevent activation sparks.

Connection and installation are facilitated by combining several ignition coils in a common casing to form a single assembly. However, the individual coils continue to operate as independent units. The integration of output stages in the ignition coils means that short primary leads can be used (lower voltage drop). This arrangement also prevents power loss in the driver circuits from overheating the ECU.

Dual-spark ignition coil (distributorless ignition)
1 Low-voltage terminal, 2 Laminated iron core, 3 Primary winding, 4 Secondary winding, 5 High-voltage terminals.



Spark plug

Function

The spark plug introduces the ignition energy generated by the ignition coil into the combustion chamber. The high voltage creates an electric spark between the spark-plug electrodes which ignites the compressed A/F mixture. As this function must also be guaranteed under extreme conditions (cold starting, full load), the spark plug plays a decisive role in the optimum performance and reliable operation of a spark-ignition engine. These requirements remain the same over the entire service life of the spark plug.

Requirements

The spark plug must satisfy a variety of extreme performance demands: It is exposed to the varying periodic processes within the combustion chamber as well as external climatic conditions. However, the combustion chamber must remain sealed.

During spark-plug operation with electronic ignition systems, ignition voltages of up to 30,000 V may occur and must not damage the insulator. This insulation capability must also be guaranteed at temperatures in the region of 1000 °C.

Because the spark plug is subjected to mechanical stresses in the form of exposure to periodic pressure peaks (up to

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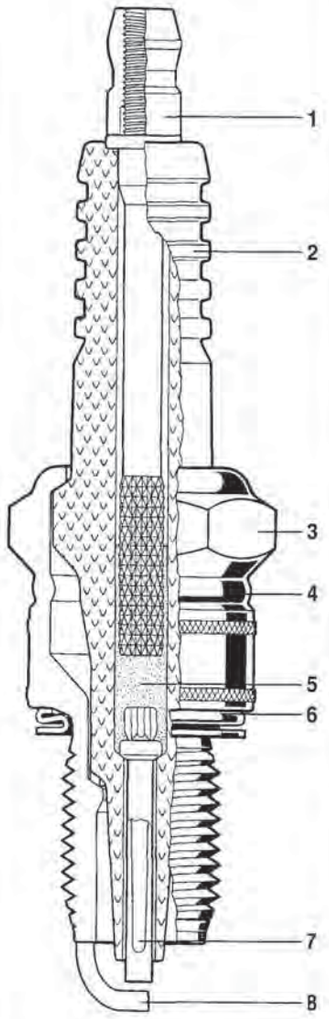
80 bar) within the combustion chamber, its materials must exhibit extreme resistance to thermal loads and continuous vibratory stress. At the same time, that section of the spark plug that protrudes into the combustion chamber is exposed to high-temperature chemical processes, making resistance to aggressive combustion deposits essential. Because it is subjected to rapid variations between the heat of the combustion gases and the cool A/F mixture, the spark-plug insulator must feature high resistance to thermal stresses (thermal shock). Effective heat dissipation at the electrodes and the insulator is also essential for reliable spark-plug performance.

Design

In a special high-grade ceramic insulator, an electrically conductive glass seal forms the connection between the center electrode and terminal stud. This glass element acts as a mechanical support for the components while providing a gas seal against the high-pressure combustion gases. It can also incorporate resistor elements for interference suppression and burn-off.

The connection end of the insulator is glazed for improved protection against contamination. The connection between it and the nickel-plated steel shell is gas-tight. The ground electrode, like the center electrode, is primarily manufactured using nickel-based alloys to cope with the high thermal stresses. It is welded to the shell. The thermal conduction properties of both the center and the ground electrodes are improved by using a nickel-alloy jacket material and a copper core. Silver and platinum, or platinum alloys, are employed as electrode material for special applications. The spark plugs have either an M4 or a standard SAE thread, depending upon the type of high-voltage connection. Spark plugs with metal shields are available for watertight systems and for maximum interference suppression.

Spark plug
1 High-voltage connector (terminal nut),
2 Al_2O_3 Ceramic insulator,
3 Shell,
4 Heat-shrinkage zone,
5 Conductive glass,
6 Captive gasket,
7 Composite center electrode Ni/Cu,
8 Ground electrode.



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Heat range

The operating temperature of the spark plug represents a balance between heat absorption and dissipation. The aim is to achieve a self-cleaning temperature of approx. 500 °C even at low engine power outputs.

If the temperature drops below this level, there is the danger that unburnt hydrocarbons and oil residue from incomplete combustion sequences will settle on the cold areas of the spark plugs (particularly when the engine is not at normal operating temperature, at low outside temperatures, and during repeated starts). This can create a conductive connection (shunt) between the center electrode and the spark-plug shell by way of which the ignition energy leaks away in the form of short-circuit current (risk of misfires). At higher temperatures, the residues containing carbons burn on the insulator nose; the spark plug thus "cleans" itself.

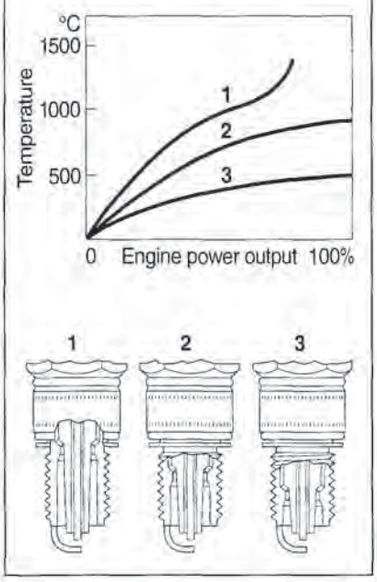
An upper temperature limit of approx. 900 °C should be observed because in this range wear of the spark-plug electrodes increases markedly (due to hot-gas corrosion) and if this limit is significantly exceeded this increases the risk of auto-ignition (ignition of the air-fuel mixture on hot surfaces). Such auto-ignition subjects the engine to extreme loads and can result in the engine's destruction within a short period of time. The spark plug must therefore be adapted accordingly in terms of its heat-absorption capability to the engine type. The identifying feature of a spark plug's thermal loading capacity is its heat range, which is defined by a code number and determined in comparison measurements with a reference standard source.

The Bosch ionic-current process presents possibilities for adapting the heat range to each engine. Characteristic changes in the combustion procedure due to increased thermal loading of the spark plugs can be detected using ionic current and used in the assessment of the auto-ignition process. The spark plug must be adapted in such a way as to preclude any possibility of premature ignition.

The use of center-electrode materials with high thermal conductivity (silver or nickel alloys with copper core) makes it possible to substantially extend the insulator nose without changing the plug's heat range, thus extending the plug's operating range downward into a lower thermal-load range and reducing the probability of fouling. These advantages are inherent in all Bosch Super (thermo-elastic) spark plugs.

Reducing the likelihood of combustion miss and ignition miss – with their attendant massive increases in hydrocarbon emissions – provides benefits in exhaust emissions and fuel consumption in part-throttle operation at low load factors.

Spark-plug temperature response
1 high, 2 medium, and 3 low heat range.



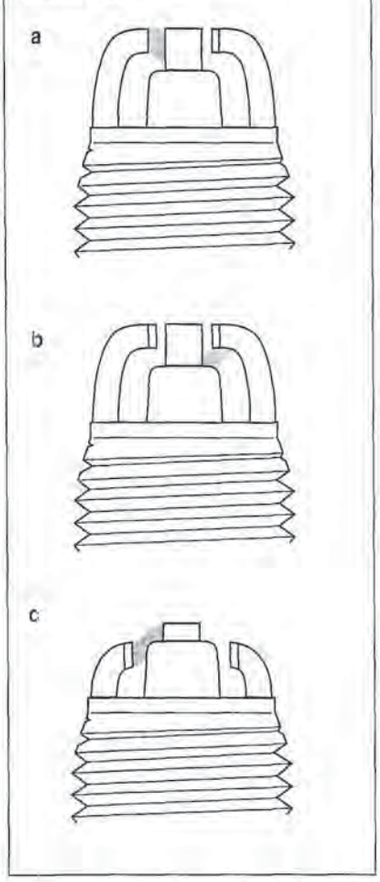
Electrode gap and ignition voltage

The electrode gap should on the one hand be as large as possible so that the ignition spark activates a large volume element and thus results in reliable ignition of the air-fuel mixture due to the development of a stable flame-core.

It is frequently impossible to achieve a smooth idle when the electrode gap is too

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Spark gap
a) Spark air gap,
b) Semi-surface gap,
c) Surface gap.



narrow. On the other hand, the electrode gap must be narrow enough to guarantee that the ignition voltage will continue to produce a reliable arc, even at the end of the spark plug's service life and under unfavorable circumstances.

The required ignition voltage is influenced not only by the size of the electrode gap, but also by the electrodes' shape and temperature and the materials used in their manufacture. Parameters specific to the combustion chamber such as mixture composition (lambda value), flow velocity, turbulence and density of the gas to be ignited, also play an important role.

On today's high-compression engines, which frequently feature high charge turbulence, electrode gaps must be carefully defined in order to guarantee reliable ignition and thus misfire-free operation throughout the required service life.

Spark gap
The configuration of the arcing path s defined by the mutual arrangement of the electrodes.

Spark air gap (a):
A linear spark arcing directly between the center and ground electrodes ignites the air/fuel mixture between the electrodes.

Semi-surface gap (b):
The semi-surface gap is created by positioning the ground electrodes (at a defined distance to the insulator end face) to the sides of the center electrode. Under certain conditions, the ignition spark forms between the center electrode and the surface of the insulator tip before arcing across a gas-filled gap to the ground electrode.

Shifting a portion of the spark's propagation travel to this shunt path extends the gap across which any given voltage can produce an arc.

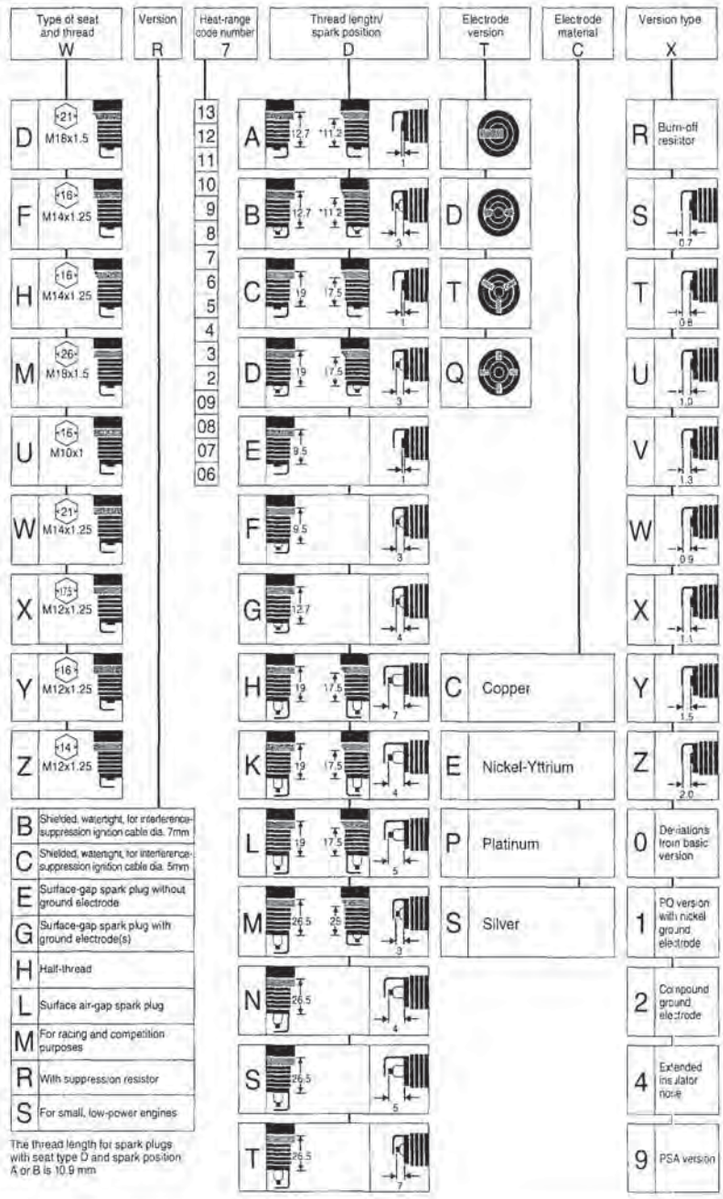
The increased electrode gap improves the ignition properties.

Surface gap (c):
On surface-gap spark plugs, the ground electrodes are positioned to the sides of the ceramic body. The sparks thus form over the surface of the insulator tip before arcing across a small gas-filled gap to the ground electrode. The arrangement of the ground electrode to the side of the ceramic body helps to reduce the quenching losses, i.e. the flame core can extend more effectively into the combustion chamber, thus enhancing the ignition properties of the plug.

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Designation codes for Bosch spark plugs

Individual spark-plug specifications are contained in the designation code. This code includes all vital spark-plug characteristics except the electrode gap, which is indicated on the package. Spark-plug specifications for individual engine applications are defined by Bosch and the engine manufacturers.



The thread length for spark plugs with seat type D and spark position A or B is 10.9 mm

Conventional coil ignition (CI)

Many vehicles are still equipped with conventional coil ignition. When the contact breaker closes with the ignition switched on, current from the battery or alternator flows through the ignition coil's primary winding, generating a powerful magnetic field in which the energy is stored. At the ignition point, the contact breaker interrupts the current, the magnetic field collapses and the high voltage necessary for ignition is induced in the secondary winding. This voltage is fed from terminal 4 to the ignition distributor via a high-tension cable and from there to the individual spark plugs.

The following is a basic definition of the relationship between the speed of a four-stroke SI engine and the number of sparks generated per minute:

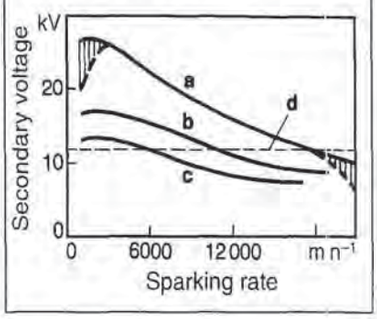
$$f = z \cdot n/2$$

f Spark-generation rate, z Number of cylinders, n Engine speed.

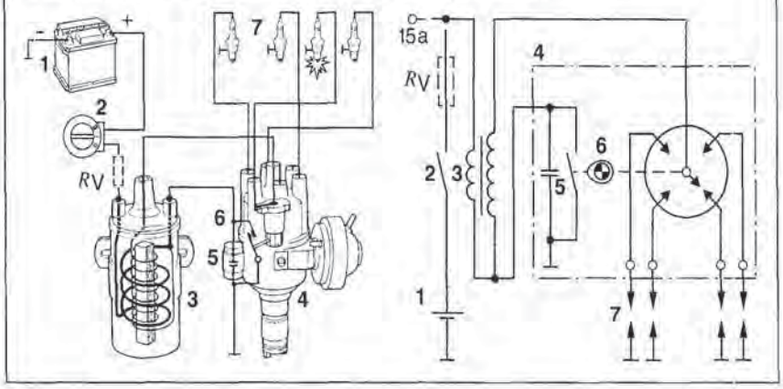
At low engine speeds, the contact-breaker points remain closed long enough to exploit the coil's full energy-storage potential. At higher engine speeds, this contact period – the dwell angle – is shorter, and the primary current is interrupted before maximum energy can be transferred to the coil. The resulting reduction in stored energy means that less high-tension current is then available from the coil.

In response, ignition coils are designed to provide high-tension voltage well in excess of the spark plugs' requirements, even at maximum engine speeds. Contamination on the insulating components acts as a capacitive and ohmic shunt, increasing the ignition loads placed upon the system, with combustion and ignition misfiring as the ultimate consequences.

Secondary voltage as a function of sparking rate
a *Without ohmic shunts (R > 10 MΩ),*
b *Shunt resistance 1 MΩ,*
c *Shunt resistance 0.5 MΩ,*
d *Required ignition voltage.*



***Conventional coil-ignition system (CI)**, components at left, circuit diagram at right*
1 Battery, 2 Ignition switch, 3 Coil, 4 Distributor, 5 Ignition condenser, 6 Contact breaker, 7 Spark plugs, R_V Ballast resistor for increased start voltage (optional).



Ignition coil
Description: see P. 481.

Ignition distributor
The distributor is a separate, self-contained component within the ignition system. It has the following functions:
- it distributes the ignition pulses to the engine's spark plugs in the defined sequence (CI, TI, and electronic ignition),
- triggers the ignition pulse, either when the contact breaker interrupts the primary current, or, with breakerless systems (CI, TI, EI in some cases), using a pulse generator,
- adjusts the ignition timing with a spark-advance mechanism on conventional ignition systems (CI, TI).

In modern electronic ignition systems, operating either alone or in combination with the fuel-injection system (Motronic), the distributor generally comprises only a rotor arm connected to the camshaft and the distributor cap with high-voltage cables.

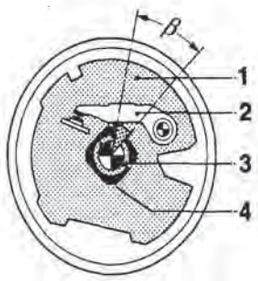
The contact-breaker points and the spark-advance mechanism perform separate functions from those of the distributor proper. They are combined with it in a single unit because they require a synchronized drive.

The ignition pulse passes through the center connection and the carbon brush or the center-tower spark gap to the distributor's rotor arm which then distributes this ignition energy by arcing it to fixed electrodes pressed into the periphery of the distributor cap. From here, the ignition pulses travel through the ignition cables to the spark plugs. A dust cover is sometimes installed to separate this high-voltage section from the rest of the unit.

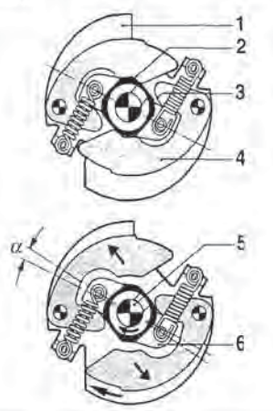
Contact breaker
A cam opens the contact-breaker points to interrupt the flow of primary current to the coil for ignition. The number of cam lobes corresponds to the number of engine cylinders. The portion of the distributor shaft's rotation during which the points remain closed is the dwell angle.

The contact-breaker points are subject to three types of wear:

Contact breaker
1 Moving breaker-plate assembly, 2 Breaker lever, 3 Distributor shaft, 4 Distributor cam.



Centrifugal advance mechanism, at rest (above), in operation (below)
1 Support plate, 2 Distributor cam, 3 Contact path, 4 Advance flyweight, 5 Distributor shaft, 6 Driver.



- contact pitting,
- contact arm (rubbing-block) wear,
- plastic deformation and local compression of the contact metal.

Contact pitting stems from the breaking sparks (residual arcing) induced by induction voltage during interruption of the primary current. The ignition condenser is designed to suppress this type of arcing, but residual sparks continue to occur. Although contact wear and rubbing-block wear are mutually counteractive, the effects of the latter are generally more pronounced, resulting in a tendency for the

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ignition to drift in the "retard" direction, toward a later ignition point.

Spark-advance mechanism
Ignition distributors are generally equipped with two spark-advance mechanisms: a speed-sensitive centrifugal advance mechanism and a load-dependent vacuum-controlled device.

Centrifugal advance mechanism
The centrifugal advance mechanism adjusts the ignition timing in response to changes in engine speed. The support plate upon which the flyweights are mounted rotates with the distributor shaft. The flyweights move outward as engine speed increases, thereby turning the driver over the contact path to the distributor shaft in the direction of rotation. In this way, the distributor cam also turns towards the distributor shaft by the ignition advance angle α. The point of ignition is advanced by this angle.

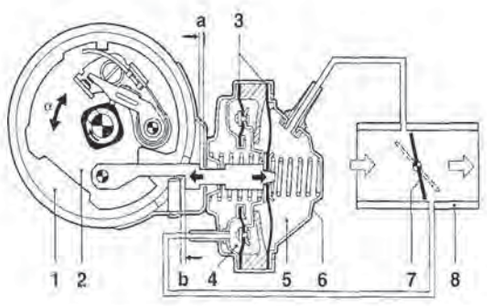
Vacuum adjustment mechanism
The vacuum mechanism adapts the ignition timing to changes in engine output and load factor. Intake manifold vacuum is monitored or tapped off in the vicinity of the throttle valve. The vacuum acts upon two aneroid capsules.

Operation of advance mechanism
Because the air/fuel mixture combusts more slowly during operation at low load factors, it must be ignited earlier to compensate. Meanwhile, the proportion of those residual gases which have been burned but not discharged from the combustion chamber increases, and the mixture leans out. Vacuum for the advance mechanism is tapped off immediately downstream from the open throttle valve. As the engine load decreases, the vacuum in the advance unit rises, causing the diaphragm and its control arm to move to the right. The control arm turns the breaker-plate assembly against the distributor shaft's direction of rotation; the point of ignition is advanced still further.

Operation of retard mechanism
Here the connection with the intake manifold's internal vacuum is downstream from the closed throttle. The ring-shaped vacuum retard unit reduces exhaust emissions by reducing ignition advance under specific operating conditions (e.g. idle, trailing throttle). The ring diaphragm and its control arm move to the left when vacuum is applied. The control arm rotates the breaker-plate assembly together with the contact breaker in the distributor shaft's direction of rotation.

This spark-retard system operates independently of the advance mechanism. The advance mechanism has priority: simultaneous vacuum in both units during part-throttle operation shifts the unit to its "advance" position.

Vacuum advance mechanism with ignition advance and retard units
a Advance adjustment up to stop, b Retard adjustment up to stop
1 Ignition distributor, 2 Breaker-plate assembly, 3 Diaphragm, 4 Vacuum retard unit, 5 Vacuum advance unit, 6 Vacuum unit, 7 Throttle valve, 8 Intake manifold.



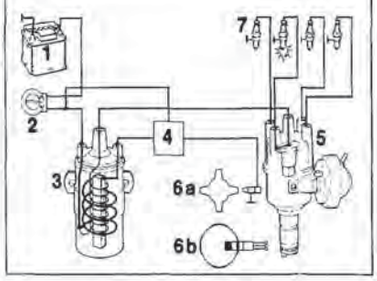
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Transistorized ignition (TI)

With conventional coil-ignition systems, ignition energy and maximum voltage are restricted by various electrical and mechanical factors limiting the breaker points' switching capacity. The demands placed upon battery-ignition systems are often more than the contact-breaker assembly can satisfy in its role as a power switch. In electronic ignition systems, the points are assisted or replaced entirely by wear-free control devices. Transistorized (coil) ignition is available in both breaker-triggered and breakerless versions.

Transistorized coil ignition with contact control is especially suitable for upgrading existing coil-ignition systems (CI). Breaker-triggered transistorized coil ignition systems are no longer installed as original equipment.

Breakerless transistorized ignition system
1 Battery, 2 Ignition switch, 3 Coil, 4 Electronic trigger box, 5 Ignition distributor with centrifugal and vacuum advance mechanism, 6a Induction-type pulse generator, 6b Hall-type pulse generator (alternative), 7 Spark plugs.



Breakerless transistorized ignition
On breakerless transistorized ignition systems, the cam-actuated contact breaker is replaced by a magnetic "pulse generator". This generates current and voltage pulses magnetically (without contacts) to trigger the high-voltage ignition pulse through the system electronics. The pulse generator is installed in the ignition distributor.

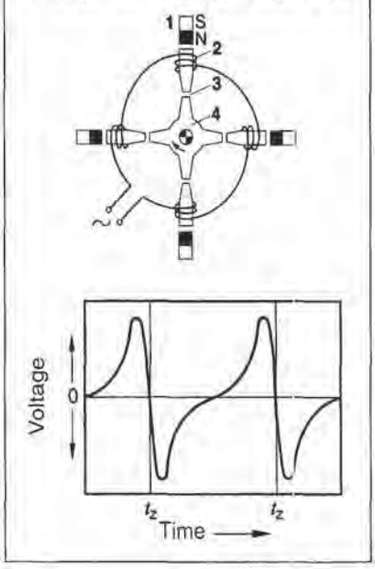
These triggering devices operate according to various principles.

Induction-type pulse generators (TI-I)
The induction-type pulse generator is a permanently-excited AC generator consisting of stator and rotor. The number of teeth or arms corresponds to the number of cylinders in the engine. The frequency and amplitude of the alternating current generated by the unit vary according to engine speed. The ECU processes this AC voltage and uses it for ignition control.

Hall-effect pulse generators (TI-H)
This type of ignition-pulse generator utilizes the Hall effect. A speed-sensitive magnetic field produces voltage pulses in an electrically charged semiconductor layer to control activation of the ECU's primary current.

Ignition pulse generators (impulsers) display clear benefits over mechanical contact breakers: They do not wear, and are thus maintenance-free. They allow precise control of ignition timing with attendant benefits in engine performance.

Ignition distributor with induction-type pulse generator
1 Permanent magnet, 2 Induction winding with core, 3 Variable air gap, 4 Trigger wheel.



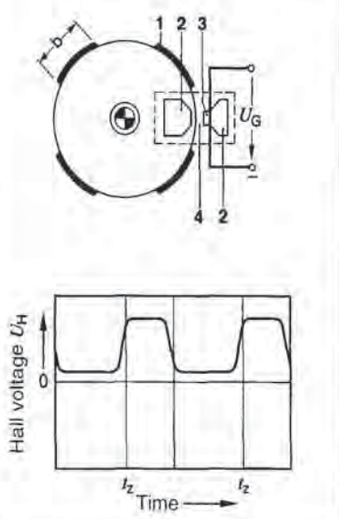
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Electronic control units
Virtually all of the electronic control units (trigger boxes) in use today are equipped with primary-current regulators and closed-loop dwell-angle control.

The primary-current regulator limits the current in order to protect the ignition coil and the driver stage. When used in conjunction with a coil featuring low primary resistance, it provides high starting current at low battery voltages. This makes it possible to dispense with series resistors upstream of the coil as well as with the bridging function for starting.

Closed-loop dwell-angle control ensures that the desired primary current is obtained in the control range as far as possible at the point of ignition. This reduces the power losses in the ECU. It also compensates for battery-voltage fluctuations and ignition-coil temperature effects. Depending on system design, this dwell-angle control is effective up to medium engine speeds. At high engine speeds, the dwell angle is determined by the break time required to achieve adequate arcing durations. The residual energy remaining in the coil after the break time promotes optimal coil charging with reduced dwell times.

Ignition distributor with Hall sensor
1 Vane with width b, 2 Soft-magnetic conductive elements, 3 Hall IC, 4 Air gap, U_G Sensor voltage (transformed Hall voltage).



The sparkless closed-circuit current deactivation switches off the primary current with the ignition on and the engine off to ensure that no sparks occur at the spark plug. However, there are also TC-I systems (with induction-type pulse-generator) with intrinsic closed-circuit current deactivation.

Transistorized ignition is sometimes employed together with auxiliary devices to adjust the spark advance. An example would be the idle-speed control which is installed between the Hall generator and the ECU; below idle speed it reacts to further decreases in engine min^{-1} by advancing the ignition, thus increasing torque and preventing engine speed from dropping any further. The electronic retard device reduces ignition advance at high engine speeds to prevent knocking. It is connected in parallel with the ECU. Today, both of these functions are integrated in the electronically adjusted ignition systems within the engine-management system.

Hybrid units have become the ECU standard for transistorized ignition systems owing to their ability to combine high packaging density with low weight and excellent reliability. Hybrid technology replaces the printed-circuit board, with an Al_2O_3 substrate bearing conductor paths and resistors applied in a silk-screening process. Semiconductor devices and capacitors in chip form complete the circuit. As the Darlington power-transistor chip is mounted insulated on the metallic base plate, cooling is excellent, permitting operation at high temperatures.

Ignition coils (description on P. 481)
The performance specifications of ignition coils for conventional ignition differ from those of ignition coils with electronic circuit-breakers (transistorized ignition, see P. 490). A coil designed for one application should never be employed in the other.

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Note: Unlike with breaker-triggered ignition, terminal 1 of the systems mentioned must not be shorted to ground (e.g. during compression testing) as this would overload the low-resistance primary winding of the ignition coil.

Capacitor-discharge ignition (CDI)

The operating concept behind CDI, or "thyristor ignition," as it is also called, differs from that of the ignition systems described above. CDI was developed for use with high-speed, high-output multicylinder reciprocating IC engines in high-performance and competition applications and for rotary-piston engines.

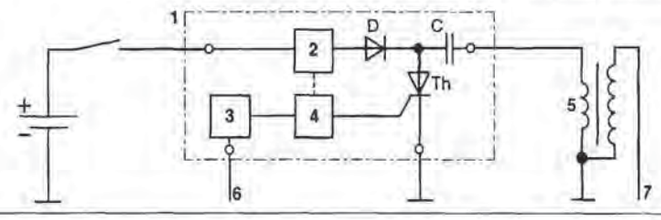
The salient characteristic of the CDI system is that it stores ignition energy in the electrical field of a capacitor. Capacitance and charge voltage of the capacitor determine the amount of energy which is stored. The ignition transformer converts the primary voltage discharged from the capacitor to the required high voltage. Capacitor-discharge ignition is available in both breaker-triggered and breakerless versions.

The major advantage of the CDI is that it generally remains impervious to electrical shunts in the high-voltage ignition circuit, especially those stemming from spark-plug contamination. For many applications, the spark duration of 0.1...0.3 ms is too brief to ensure that the air-fuel mixture will ignite reliably. Thus CDI is only designed for specific types of engine, and today its use is restricted to a limited application range, as transistorized ignition systems now afford virtually the same performance. CDI is not suited for aftermarket installations.

CDI can also be employed for distributorless ignition (DLI) with the installation of one ignition coil per cylinder, with energy distribution taking place at the medium-voltage level.

Capacitor-discharge ignition system with induction-type pulse generator, schematic

1 Control unit,
2 Charger,
3 Pulse shaper,
4 Control stage,
5 Ignition transformer,
6 To induction-type pulse generator,
7 To ignition distributor.



Accident hazard

All electronic ignition systems (including capacitor discharge, transistorized ignition and all systems with partial or comprehensive electronic control) are potentially dangerous. Always switch off the ignition or disconnect the battery before performing any service or maintenance including:
– Replacing components such as spark-plugs, coils, transformers, distributors, high-tension cables, etc.
– Connecting engine test devices such as stroboscopic lamps, dwell-tach testers, ignition oscillographs etc. Dangerously high voltage levels are present throughout the ignition system whenever it is turned on. All service operations should be performed exclusively by qualified technicians.

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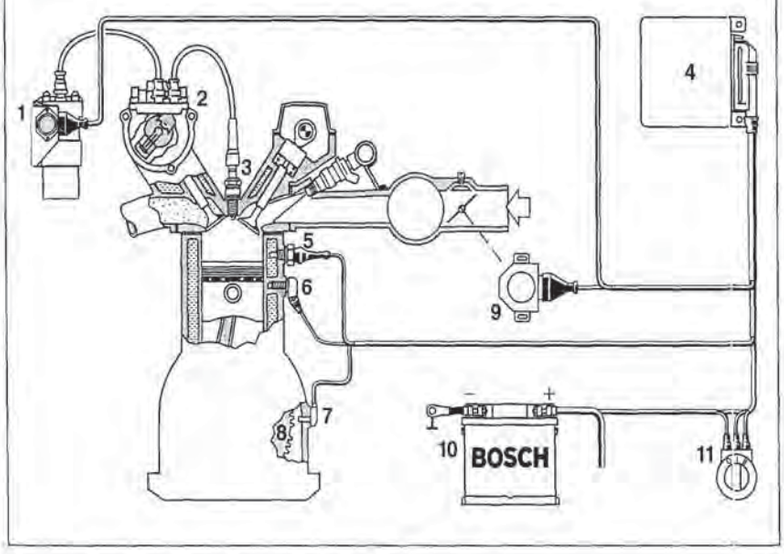
Electronic ignition (EI and DLI)

Electronic ignition derives its name from the fact that it calculates the ignition point electronically. The characteristic curves provided by the conventional distributor's centrifugal and vacuum-advance units are replaced by an optimized electronic ignition map. Mechanical high-tension distribution is retained with EI ignition. Fully electronic distributorless semiconductor ignition (DLI) uses stationary electronically controlled components to replace the mechanical, rotating high-tension distributor.

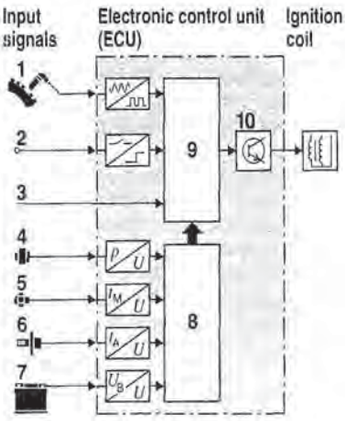
Electronic ignition systems operate more precisely than mechanical systems, with major benefits originating in the fact that the ignition process can be triggered from the crankshaft instead of from a distributor (distributor drive tolerances are no longer a factor). The limitations which mechanical adjustment mechanisms place upon the performance curve (summation of curves for load and engine speed in a single progression) are also avoided. The number of input variables is also theoretically unlimited, usually allowing extensions in the ignition angle's adjustment range. The fixed-drive ignition distributor's limitations regarding the engine's ignition-voltage requirements and ignition-angle adjustment range are such that it has difficulty coping with larger numbers of cylinders; efficient spark distribution cannot always be guaranteed. Corrective measures include dividing the ignition into two circuits (e.g., for 8- and 12-cylinder engines) and static voltage distribution.

Electronic ignition can be combined with electronic fuel-injection (Motronic), knock control, ASR, etc., making it possible to employ sensors and/or signals from other units in more than one system. A serial bus (CAN, P. 884) further reduces the number of inputs and processing circuits on the ECU's input-side.

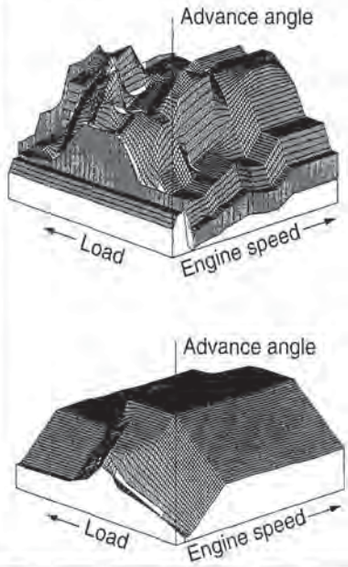
Schematic of an electronic ignition system (EI)
1 Ignition coil with ignition driver stage, 2 High-voltage distributor, 3 Spark plug, 4 ECU, 5 Engine-temperature sensor, 6 Knock sensor, 7 Engine-speed and reference-mark sensor, 8 Ring gear for sensor, 9 Throttle switch, 10 Battery, 11 Ignition switch.



Electronic ignition, signal processing
1 Engine speed, 2 Switch signals, 3 CAN (serial bus), 4 Intake-manifold pressure, 5 Engine temperature, 6 Intake-air temperature, 7 Battery voltage, 8 Microprocessor, 9 Analog/digital converter, 10 Driver stage.



Ignition maps
Above: electronically optimized, Below: mechanical advance system.



Operation

The engine's speed and crankshaft position are monitored directly at the ring gear, using either a separate rotor or a specific pin sequence employing an inductive, rod-type sensor, with two sensors being employed on older units. Triggering is either incremental or segmentary, according to whether the information is taken from teeth distributed evenly around the crankshaft or a crankshaft segment per cylinder pair:

- Beginning of segment = maximum spark advance angle,
- End of segment = starting angle.

In the incremental system illustrated here, the reference mark (shown as a tooth gap) represents a defined crankshaft position, providing a reference for electrically monitoring crankshaft angle using the ring-gear teeth. A distributor without an advance mechanism can also be employed for triggering; here the control signal is provided by a Hall generator. An absolute-pressure sensor in the intake manifold is the best way to monitor load. This provides a better gauge of cylinder charge than the spark-advance or retard bores on the throttle valve. It is also possible to incorporate load switches, throttle-valve potentiometers or electronic load signals from the mixture-preparation system. The microcomputer in the ECU processes the engine-speed and load signals prior to using them to calculate the precise ignition angle within the ignition map. The computer can also process other input variables, such as e.g. engine temperature, or information on trailing throttle or full-load operation from the throttle-valve switch, to derive correction values and regulate other vehicle-specific functions as required. The dwell angle for charging the ignition coil is also specified by the computer. The battery voltage is monitored here to enable voltage correction. The system responds to deviations in battery voltage from the specified baseline by extending or reducing coil-charging times accordingly. This ensures consistent availability of maximum voltage while at the same time limiting the accumulation of heat in ECU and coil to a minimum. At speeds below the starting

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speed, the computer interrupts the current flow through the ignition coil in order to prevent overheating. The driver stage can be either integrated inside the ECU or installed externally, e.g. on the ignition coil.

Signal processing in the ECU

After initial processing the digital signals go directly to the processor. Analog signals are first converted into digital form. There are also ECU's which can transmit supplementary digital or analog signals e.g., for overrun fuel cutoff and exhaust-gas recirculation). There exists a range of EI ignition-system versions of varying complexity. A comparison of the ignition map with the response curve of a distributor shows that it is possible to program each point in the ignition map independently from every other point. Thus the optimum ignition timing (e.g., for maximum fuel economy) can be selected for every operating condition, according to the limits imposed by factors such as exhaust emissions, pre-ignition limit and driveability. The entire system is completely maintenance-free and does not recuire any adjustments during the engine's service life.

Rotating voltage distribution

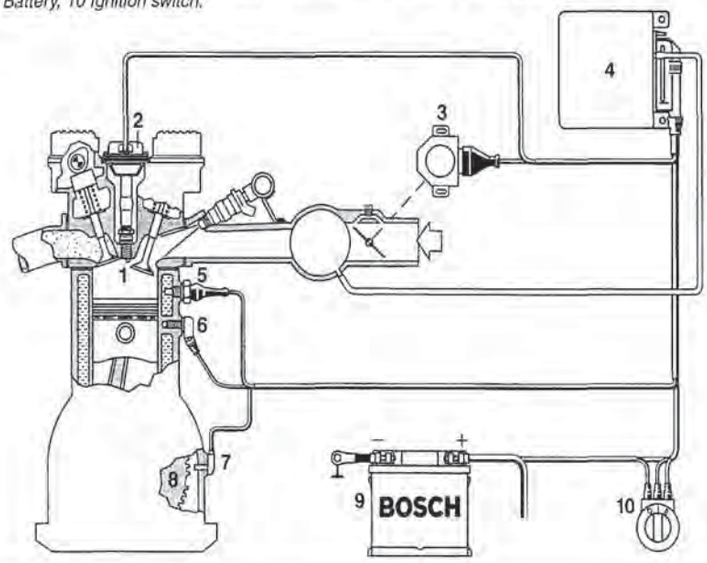
A high-voltage distributor distributes the ignition pulses to the spark plugs of the individual cylinders (as described earlier). If the distributor's adjustment range is insufficient for handling a larger number of cylinders, then two ignition circuits are employed, i.e., two 4-cylinder distributors can be used for one 8-cylinder engine. Synchronization through the crankshaft can be used for "two times 4 cylinders", whereas "two times 3 cylinders" with constant ignition-angle spacing (not employed up to now) must be controlled by the camshaft.

Distributorless (stationary) voltage distribution

Systems with single-spark ignition coil

Each cylinder has its own ignition coil with driver output stage, installed either di-

Schematic of a fully-electronic distributorless ignition system (DLI)
1 Spark plug, 2 Single-spark coil, 3 Throttle switch, 4 ECU, 5 Engine-temperature sensor, 6 Knock sensor, 7 Engine-speed and reference-mark sensor, 8 Ring gear for sensor, 9 Battery, 10 Ignition switch.



rectly above the spark plug or separately. Either synchronization with the camshaft sensor or a method for detecting the compression cylinder is required. On engines with an even number of cylinders, the system reverts to crankshaft triggering in the event of camshaft-sensor failure, although two coils are then always activated simultaneously (one of the sparks is discharged during an exhaust stroke). This system, suitable for engines with any number of cylinders, provides the greatest latitude for adjustment, as there is only one spark per cycle. All these advantages mean that the single-spark ignition coil is being increasingly used and is taking over from the dual-spark ignition coil in spite of costing more.

Systems with dual-spark ignition coil
One ignition coil is required for every two cylinders. The crankshaft can be used for synchronization. The high-voltage end of each ignition coil is connected to the spark plugs for two cylinders whose operating cycles are 360° out of phase with each other.

As there is an additional spark during the exhaust stroke, it is important to ensure that residual mixture or fresh mixture is not ignited. Furthermore, the dual-spark system is only suited for use with even numbers of cylinders. Owing to its cost advantage relative to the single-spark unit, the dual-spark ignition system is the most common distributorless ignition in use today.

Knock control

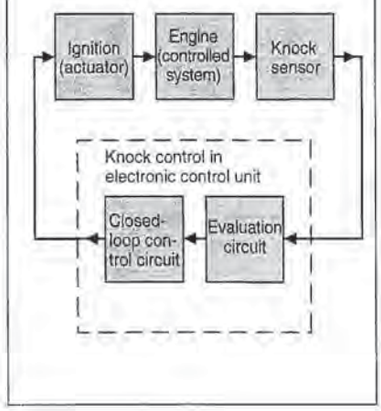
Application

Electronic control of the ignition point offers the possibility of controlling very precisely the ignition angle as a function of engine speed, load and temperature.

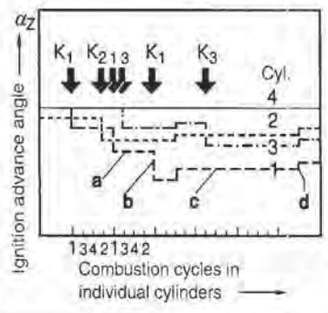
Nevertheless, without knock control a clear safety distance to the knock limit is required. This distance is necessary to ensure that even in the most knock-sensitive case with regard to engine tolerances, engine aging, environmental conditions and fuel quality, no cylinder reaches or exceeds the knock limit. The resulting engine design leads to lower compressions with retarded ignition points and thus worsening of fuel consumption and torque figures.

These disadvantages can be avoided through the use of knock control. Experience shows that knock control increases engine compression and significantly improves fuel consumption and torque. However, the pre-control ignition angle now no longer needs to be deter-

Schematic of knock-control system



Knock control *Control algorithm for ignition adjustments with a 4-cylinder engine*
$K_{1...3}$ knock in cylinders 1...3, Cylinder 4, no knock.
a Delay prior to ignition retard, b Retardation, c Delay before return to original ignition point, d Spark advance.



mined for the most knock-sensitive but rather for the most insensitive conditions (e.g. engine compression at lower tolerance limit, best possible fuel quality, most knock-insensitive cylinder). Each individual engine cylinder can now be operated over its entire service life in virtually all operating ranges at its knock limit and thus at optimal efficiency.

The essential precondition for this ignition-angle configuration is reliable knock detection for each individual cylinder as from a particular knock intensity, and over the entire engine operating range.

Combustion knock

The characteristic vibrations of combustion knock are detected by knock sensors, converted into electrical signals and forwarded to the Motronic.

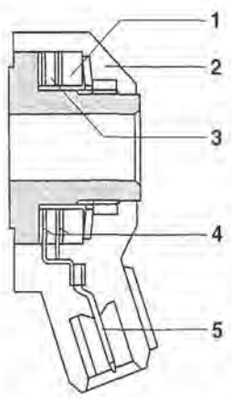
It is essential to work out carefully the number and installation positions of the required knock sensors. Reliable knock detection must be guaranteed for all cylinders and for all engine operating points, particularly at high engine speeds and loads. As a rule, 4-cylinder in-line engines are equipped with one knock sensor, 5- and 6-cylinder engines with two sensors and 8- and 12-cylinder engines with two or more sensors.

Knock detection, evaluation

For the purpose of knock detection, the vibrations characteristic to knocking are converted into electrical signals by one or more knock sensors mounted at appropriate positions on the engine, and then forwarded to the Motronic for evaluation. It is here that the corresponding evaluation algorithm is applied to detect knock for each cylinder and each combustion. Detected combustion knocks result in a retardation of the ignition point by a programmable amount at the cylinder in question. If knock stops, the ignition point is advanced again in stages up to the pre-control value.

The knock-detection and knock-control algorithms are matched in such a way as to eliminate any knocking that is audible and damaging to the engine.

Knock sensor
1 Seismic mass, 2 Potting compound, 3 Piezoceramic element, 4 Contacts, 5 Terminals.



Adaptation

Real engine operation produces different knock limits and thus also different ignition points for the individual cylinders. In order to adapt the ignition-point pre-control values to the particular knock limit, the ignition-point retardation values individual to each cylinder and dependent on the operating point are stored.

They are stored in non-volatile maps of the permanently powered RAM covering load and engine speed. In this way, the engine can also be operated in the event of rapid load and speed changes in each operating point at optimal efficiency and without audible combustion knocks.

The engine can even be approved for fuels with lower anti-knock properties.

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Combined ignition and injection system (Motronic)

The Motronic engine-management system has undergone substantial development since its introduction in 1979. Intially, integration was based on the basic gasoline-injection systems, in combination with distributorless semiconductor ignition, as were available in the following systems:
- KE-Motronic, based on KE continuous gasoline injection (P. 469),
- Mono-Motronic, based on intermittent single-point injection (P. 465),
- M-Motronic, based on intermittent multipoint injection (P. 471).

On the basis of M-Motronic, and by applying further integration steps, the Motronic system assumed control of all the manipulated variables of a spark-ignition engine that influence torque:
- ME-Motronic with electronic accelerator pedal (ETC) for controlling gasoline injection, ignition and fresh-air charge for manifold injection (P. 498).
- MED-Motronic with further integrated open- and closed-loop control functions for the high-pressure fuel circuit. With direct injection and realization of the various operating modes of this engine type (P. 507).

System overview

The Motronic system comprises all the sensors for recording the current engine and vehicle operating data (P. 501, Operating-data acquisition) and all the actuators for the adjustments to be carried out on the SI engine (P. 503, Operating-data processing).

Electronic control unit

The ECU employs sensors to monitor the relevant status of engine and vehicle at extremely short intervals (milliseconds).

Input circuits suppress sensor-signal interference and convert the signals to a single unified voltage scale. An analog-digital converter then transforms the conditioned signals into digital values. Further signals are received by way of a digital interface. Using this information, the microprocessor identifies the operating state desired by the driver and from it calculates for instance:
- the required torque,
- the resulting cylinder charge with the associated injected fuel quantity, and
- the correct ignition timing.

The low-level signal data from the microprocessor outputs are adapted by the driver stages to the levels required by the various actuators. A semiconductor memory chip stores all programs and performance maps, ensuring system consistency which remains completely impervious to fluctuations resulting from signal-level and component tolerances.

Digital accuracy is a function of word length, as well as of the consistency of the quartz's basic clock frequency and the types of algorithms used for the calculations. The consistency and precision of the reference voltages and the components installed in the analog input circuits influence the analog accuracy.

Program design must satisfy the engine's severe real-time demands: in an 8-cylinder engine at maximum speeds, less than 2.5 ms are available between two ignitions. All essential calculations must be completed within this period. In addition to these crankshaft-synchronous processes, there are also time-synchronous operations. Both types can be suspended by interrupts.

ME-Motronic

Design

Basic functions

In order to set the operating state desired by the driver (main function), the position of the gas pedal is converted by the microprocessor into a setpoint value for the engine torque. Then, while taking into consideration the highly varying current operating data from the ME-Motronic, these are converted into the variables for determining the engine torque:
- the charging of the cylinders with air,
- the mass of the injected fuel and
- the ignition angle.

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ME-Motronic system diagram
1 Carbon canister, 2 Shut-off valve, 3 Canister-purge valve, 4 Intake-manifold pressure sensor, 5 Fuel rail/fuel injector, 6 Ignition coil/spark plug, 7 Phase sensor, 8 Secondary-air pump, 9 Secondary-air valve, 10 Air-mass meter, 11 Throttle device (ETC), 12 EGR valve, 13 Knock sensor, 14 Engine-speed sensor, 15 Temperature sensor, 16 Lambda sensor, 17 ECU, 18 Diagnosis interface, 19 Diagnosis lamp, 20 Vehicle immobilizer, 21 Tank pressure sensor, 22 In-tank unit, 23 Accelerator-pedal module, 24 Battery.

Additional function
As well as the basic functions, the ME-Motronic has a large number of additional open- and closed-loop control functions. Examples of such functions are:
- idle-speed control,
- closed-loop lambda control,
- control of evaporative-emissions control system,
- exhaust-gas recirculation for reducing NO_X emissions,
- control of secondary-air system for reducing HC emissions, and
- cruise control.

These functions have been made necessary by the legislation covering the reduction of exhaust emissions, the calls for a decrease in fuel consumption but also increased demands placed on driving comfort and safety.

In addition, the ME-Motronic system can be further extended with the following functions:
- Control of turbocharger control and variable-tract intake manifold for increasing engine power output,
- Camshaft control for reducing exhaust emissions and fuel consumption and for increasing power,
- Knock control, engine-speed limitation and vehicle-speed limitation for protecting components in the engine and vehicle.

Torque-guided control concept
The torque-guidance principle is aimed at simplifying the many and sometimes very different assignments placed on the engine and the vehicle. Only then is it possible, depending on the engine or vehicle type, to determine the required functions in each case and to integrate them in the relevant Motronic version.

Most of the additional open- and closed-loop control functions likewise influence the engine torque. In a torque-guided system, all these functions behave like the driver: they request an engine torque irrespective of each other. The torque-guided ME-Motronic uses torque coordination to sort the often contradictory torque requirements and to then put the most important requirement into effect.

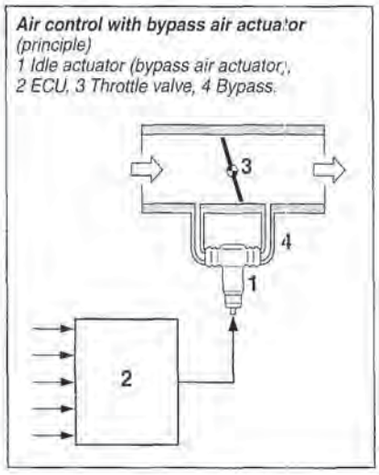
Vehicle management
The ME-Motronic can communicate via the CAN bus system (P. 884) with the ECUs of other vehicle systems. Among other things, in conjunction with the automatic-transmission ECU, it reduces torque at the gear-shift point so that the transmission is subjected to less loading and wear. When wheel spin occurs, the ECU belonging to the Traction Control System (TCS) signals to the ME-Motronic to reduce the generated torque.

Diagnosis
The ME-Motronic incorporates components for on-board monitoring (OBD). It therefore complies with the most stringent emission limits and the demands placed on the integrated diagnosis functions.

Cylinder-charge control systems

Throttle-valve control
In an SI engine with external A/F mixture formation, the cylinder charge is the factor that determines torque output and thus power output. The throttle valve controls the air-flow intake into the engine and thus the cylinder charge.

In conventional systems, a cable or a linkage transfers the movement of the accelerator pedal to the throttle valve. Requirements for additional air are covered by a bypass air actuator which directs an additional air flow around the



Air control with bypass air actuator
(principle)
1 Idle actuator (bypass air actuator), 2 ECU, 3 Throttle valve, 4 Bypass.

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throttle valve (see Fig.), or a throttle-valve actuator alters the stop for minimum air quantity. However, the air flow required by the engine can only be electronically influenced to a limited extent (idle-speed control).

Systems with ETC (electronic throttle control, integrated in the engine control unit for ignition, injection and other auxiliary functions) detect the position of the accelerator pedal with a pedal-travel sensor (potentiometer).

Taking into account the current engine operating status, the ECU calculates how far the throttle valve needs to open and actuates the throttle-valve drive accordingly (see Fig.). A throttle-valve angle sensor (potentiometer) monitors exact compliance with the desired throttle-valve position. Two potentiometers (for redundancy reasons) at the accelerator pedal and throttle device are part of the ETC monitoring system which, while the engine is running, continually checks all the sensors and calculations which affect throttle-valve opening.

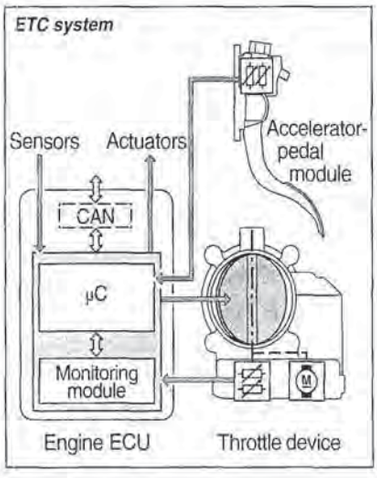
Charge-cycle control

The behavior of the gas columns flowing into and out of the cylinders changes considerably as a function, for instance, of engine speed or throttle-valve opening. With fixed valve timing, the charge cycle can only be optimized for a specific operating range.

Camshaft control or camshaft-lobe control (P. 387) permits not only variable valve timing, and thus adaptation to various engine speeds, but also enables the residual exhaust-gas mass in the cylinder to be influenced ("internal EGR").

("External") EGR effectively reduces the temperature-dependent NO_x emissions. Exhaust gases which have already been combusted are added to the A/F mixture in order to reduce peak combustion temperatures. For this purpose, the Motronic triggers the exhaust-gas recirculation valve as a function of the engine operating point. The EGR valve then adds a partial flow of exhaust gas to the fresh mixture, thereby increasing the total charge while the fresh-air charge remains constant. In this way, the engine must be throttled less heavily in order for a specific torque to be reached (lower fuel consumption).

The achievable torque, which is proportional to the fresh-mixture charge, can be increased still further by compressing the air in the cylinder. This is the purpose of dynamic supercharging, exhaust-gas turbocharging, or mechanical supercharging (P. 390).



Operating-data acquisition

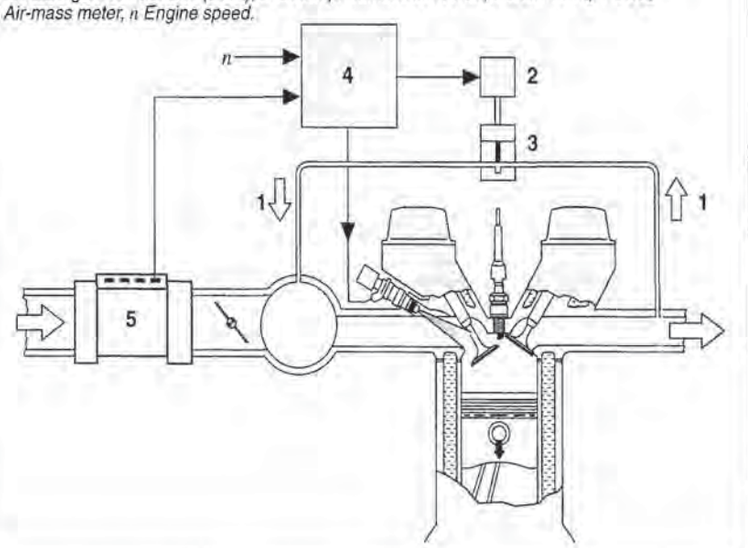
Vehicle-operator command

In the case of engine management with an electronic throttle valve (ETC), there is no longer a mechanical connection between the accelerator pedal and the throttle-valve actuator. Instead, the position of the accelerator pedal is sensed by a pedal-travel sensor or accelerator-pedal module and converted into an electrical signal. The engine-management system interprets this signal as the vehicle-operator command.

Air charge

In engine concepts with manifold injection, there is a linear relationship between the air charge and the torque generated by combustion. In the torque-guided ME-Motronic, the air charge is therefore not just one of the main factors involved in calculating the injected fuel quantity and

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



ignition angle, but also serves to calculate the torque currently being delivered by the engine. The following sensors can be used for this purpose:
– Hot-film air-mass meter,
– Intake-manifold pressure sensor,
– Ambient pressure sensor,
– Boost-pressure sensor (BPS) and
– Throttle-valve sensor.

Engine speed, crankshaft position and camshaft position
A speed sensor on the crankshaft monitors the crankshaft revolutions per minute (rotational speed). It then forwards this information on the crankshaft position, that is, the piston position of all cylinders, as an important input variable to the ME-Motronic. The cylinder's piston position is used, for instance, to determine the injection and ignition points.

To trigger the single-spark ignition coils, the ME-Motronic with static voltage distribution also requires additional information on the camshaft position. This information is mostly monitored with a Hall-effect sensor and forwarded to the ECU as a switch signal.

A/F mixture composition
The lambda factor (λ) is the figure which quantifies the A/F mass ratio of the mixture. The catalytic converter only performs optimally at $\lambda = 1$. The lambda sensor measures the oxygen concentration in the exhaust gas and provides information on the current value of the excess-air factor (P. 458).

Combustion knocks
The characteristic vibrations of combustion knocks are detected by knock sensors, converted into electrical signals and forwarded to the ME-Motronic (P. 498).

Engine and intake-air temperatures
The voltage at the NTC resistor of the engine-temperature sensor is a function of the coolant temperature. It is input via the analog-digital converter and is a measure of the temperature. In the same way, a sensor in the intake port measures the temperature of the intake air.

Operating-data processing

Torque guidance
Torque guidance is a feature of the ME-

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Motronic. Numerous subsystems within the Motronic (e.g. idle-speed control, engine-speed limitation), and the systems for management of the drivetrain (e.g. TCS or transmission-shift control) or of the complete vehicle (e.g. air-conditioner control), direct their torque requirements to the basic Motronic system with the aim of modifying the engine torque which has just been generated. The ME-Motronic evaluates and coordinates these requirements and implements the resulting setpoint torque using the available manipulated variables (optimum engine operation in terms of exhaust gas and fuel consumption at each operating point).

An essential precondition for torque guidance is the electronic accelerator pedal ETC with the throttle valve controlled independently of the accelerator pedal.

Calculation of setpoint torque

The basic variable for the ME-Motronic torque structure is the internal torque arising from combustion. The function of torque guidance is to set the internal torque, through appropriate selection of the manipulated engine variables, in such a way that the vehicle-operator command is satisfied, and all the losses and additional requirements are covered. Because the Motronic "knows" the optimum values for charge, injection time and ignition angle for each and every desired setpoint torque, it can ensure optimal engine operation in terms of exhaust gas and fuel consumption.

Setting of setpoint torque

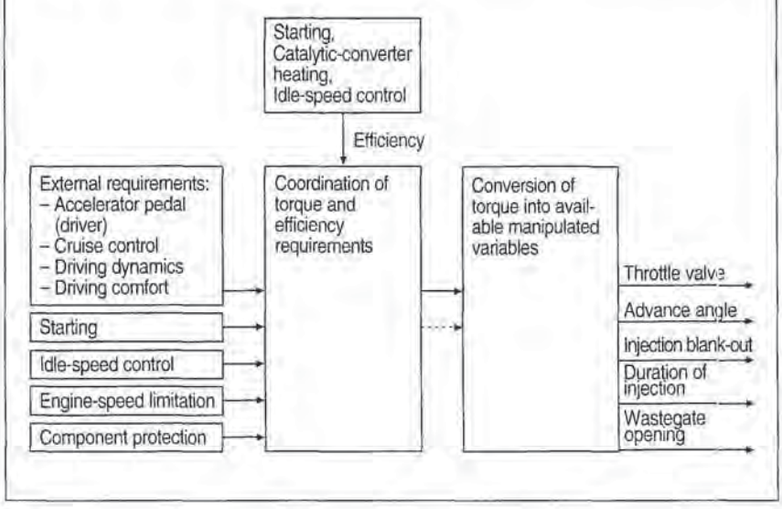
For setting the internal torque, the ME-Motronic's torque converter has a slow path (cylinder-charge path, steady-state operation) for the throttle valve (ETC), and a high-speed path (ignition-angle path, dynamic operation) for variation of the ignition angle and/or injection blank-out of individual cylinders.

Calculation of cylinder charge

The air mass in the cylinder after the intake valves have closed is termed the air charge. Referring the actual air charge to the maximum obtainable charge produces a variable that is independent of the engine piston displacement: the "relative (air) charge". The associated injected fuel quantity can be calculated from the relative air charge.

The relative charge cannot be measured directly but it can be determined from the available measurement signals.

Influencing torque in a gasoline engine



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For this purpose, an intake-manifold model calculation is used in which all the air-mass flows into the intake manifold and from the intake manifold to the combustion chamber are calculated. The most important measured variable in this model is either the air flow through the throttle valve (measured with a hot-film air-mass meter) or the absolute intake-manifold pressure (measured with a pressure sensor).

Starting out from this intake-manifold model, it is then also possible to determine the actual relative air charge.

Cylinder-charge control

In today's spark-ignition engines, the relative cylinder charge is also the main factor in influencing the engine torque, and is therefore used in the torque structure as a manipulated variable. The intake-manifold model is also used to control the cylinder charge by means of the throttle valve: for this purpose, a required cylinder charge is first of all calculated from the desired torque (which was calculated by ME-Motronic torque guidance). This cylinder charge is then converted into an associated throttle-valve angle and finally passed on as the setpoint value to the position controller of the throttle-valve actuator.

Calculation of injection period

The fuel mass for a stoichiometric air-fuel ratio can be calculated from the air charge in the cylinder. When the fuel-injector constant (this is a function of the fuel-injector properties) is taken into consideration, this furnishes the duration of injection.

The duration of injection is also dependent on the pressure differential between fuel-supply pressure (approx. 300 kPa) and injection back pressure. Fuel-supply systems with fuel recirculation keep the supply pressure constant with respect to the intake manifold. This ensures that, in spite of a changing intake-manifold pressure, the same pressure differential is applied at the fuel injectors. On the other hand, fuel-supply systems without fuel recirculation keep the supply pressure constant with respect to the environment. As the intake-manifold pressure changes, the pressure differential varies between fuel supply and intake manifold. A compensation function corrects this error.

A further adaptation factor individual to each cylinder serves to accommodate fuel-pressure pulsations caused by the opening and closing of the fuel injectors.

The effective opening duration calculated in this way is valid under the precondition that the fuel injector is already open. It must therefore be extended by the opening and closing time of the injector. An additional, battery-voltage-dependent duration of injection, which is added to the valve-opening period, compensates for this effect.

If the effective injection duration is too short, the influences of the valve-opening and closing time become excessive. In order to ensure exact fuel metering, the injection duration is restricted to a minimum value. This value is below the injection duration associated with the minimum possible cylinder charge.

To ensure optimum combustion, it is necessary to precisely define the injected fuel quantity and the exact point for the start of injection. The fuel is generally injected into the intake manifold while the intake valves are still closed. The end of injection is determined by the "fuel-hold angle", which is specified in "crankshaft degrees". The point at which the intake valve closes serves as the reference point. It is then possible to use the duration of injection for calculating, as a function of engine speed, the start of injection as an angle. The fuel-hold angle is determined taking into account the current operating conditions.

The ME-Motronic triggers a fuel injector for each cylinder. The fuel can thus be held (stored) separately for each cylinder to optimum effect (sequential injection).

Calculation of ignition and dwell angles

When calculating the ignition angle, the first step is to determine the "basic ignition angle" while taking into account the current engine operating conditions. This basic ignition angle can be adapted in specific operating states.

When calculating the dwell angle, the first step is to determine the dwell period from a speed- and battery-voltage-dependent map. Once a temperature-depen-

dent correction has been taken into account, the associated dwell angle is determined by means of a time/angle conversion.

The difference between the end of dwell (determined from the resulting ignition angle) and the dwell angle determines the start of dwell.

Operating states

In some operating states, the fuel requirement differs markedly from the steady-state requirements of an engine at normal operating temperature, with the result that corrective intervention in the mixture formation becomes necessary.

Starting

Intake-air adjustment, fuel injection and ignition are specially calculated during the starting procedure. An increased injected fuel quantity adapted to the engine temperature serves to build up a fuel film on the intake-manifold and cylinder walls and covers the increased fuel demand while the engine is running up. The ignition angle is likewise adapted to the starting procedure. The air charge while the engine is at a standstill is not affected by the throttle valve, which however is opened slightly in anticipation of the post-start phase.

Post-start

In this phase, the still increased charge and injected fuel quantity are reduced as a function of engine temperature and time at the end of the starting procedure. The ignition angle is adapted accordingly.

Warming-up

After starting at low engine temperatures, the engine's increased torque demand is covered up to a specific temperature threshold by adaptation of the charge, fuel injection and ignition.

Catalytic-converter heating

Highly retarded ignition angles generate very hot engine exhaust gas, thereby quickly bringing the catalytic converter up to operating temperature.

Idle

At idle, the engine-generated torque is just enough to permit sustained engine operation and running of the auxiliary systems. With idle-speed control, the desired idle speed remains stable under all conditions.

Full load

At full load, the throttle valve is wide open (WOT) and there are no throttling losses. The engine delivers the maximum torque relative to the current engine speed.

Acceleration and deceleration

Heavy acceleration and deceleration result in rapid changes in the intake-manifold pressure and thus in the conditions for the fuel film on the intake-manifold wall. In order to prevent leaning during acceleration, additional fuel is injected to build up the wall film. During deceleration, the injected fuel quantity is reduced accordingly.

Overrun fuel cutoff and restart

During the transition to no-combustion overrun conditions, the ME-Motronic cuts off the engine torque without jolting, and also provides for a smooth torque build-up when fuel feed starts again.

Additional closed- and open-loop control functions

Idle-speed control

Idle-speed control specifies a torque at which the desired engine speed is maintained under the given operating conditions. This torque increases as engine speed drops and decreases as engine speed rises.

Lambda closed-loop control

It is only possible to convert the noxious exhaust-gas constituents in the three-way catalytic converter in a very narrow range ("lambda window" where $\lambda = 0.99...1$). Lambda closed-loop control is essential in order to remain inside this window (P. 522).

Evaporative-emissions control system

The function of evaporative-emissions control systems is to limit the HC emissions. Such systems are equipped with a carbon canister, which accommodates

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the end of the vent line from the fuel tank. Degassing fuel is retained in the canister's activated carbon. When the engine is running, a vacuum is created in the intake manifold which causes air to be drawn in from the environment through the activated carbon and into the intake manifold. The fuel vapors trapped in the canister are entrained with this air and carried to the engine for combustion. A canister-purge valve meters this "scavenging flow".

Knock control

Knock sensors convert the vibrations characteristic to knocking into electrical signals. These are used by the Motronic to adjust the ignition point in order to prevent audible knocking and knocking which could damage the engine (P. 496).

Boost-pressure control

On engines with exhaust-gas turbocharging, the setpoint value of the desired boost pressure is adjusted by electronic boost-pressure control. It is converted into a setpoint value for the desired maximum charge and then by means of torque management into a setpoint value for the throttle-valve angle and a control duty factor for the "wastegate". In the "wastegate", by means of a change in the control pressure and stroke this signal leads to a change in the cross-section at the bypass valve.

Safety and security, comfort and convenience functions

The safety and security functions include:
- Engine-speed/vehicle-speed limitation,
- Torque/power limitation,
- Limitation of exhaust-gas temperature,
- Vehicle immobilizer.

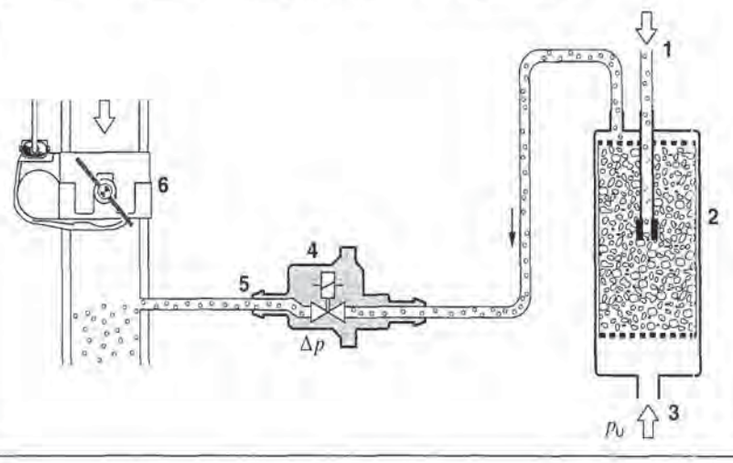
The comfort and convenience functions include:
- Load-change damping,
- Surge-damping function,
- Cruise control.

Integrated diagnosis (OBD)

"On-board diagnosis" (OBD) is a basic feature of the ME-Motronic. During normal operation, it continually compares the system's responses with the ECU's commands, as well as comparing the various sensor signals with each other for plausibility.

During vehicle inspection, a tester connected to a standardized interface reads out the stored faults and displays them. Due to the demands of the Californian Environmental Protection Agency, an ex-

Evaporative-emissions control system
1 Line from fuel tank to carbon canister, 2 Carbon canister, 3 Fresh air, 4 Canister-purge valve, 5 Line to intake manifold, 6 Throttle device with throttle valve.
Δ_p Difference between intake-manifold pressure p_S and ambient pressure p_U.



panded diagnosis system (OBD II; adapted for Europe: EOBD) monitors all those components which if they fail could result in a significant increase in noxious emissions (fault indication with diagnosis lamp).
Examples of systems subject to diagnosis:
- Air-mass meter,
- ETC throttle-valve actuator,
- Combustion misses,
- Catalytic converter,
- Lambda sensor,
- Fuel supply,
- Tank system,
- Secondary-air injection,
- Exhaust-gas recirculation (EGR).

Fault memory
If exhaust-gas-related faults are detected, an entry is made in the non-volatile fault memory. Apart from the officially stipulated fault codes, each entry includes a "freeze frame", which contains additional information on the general conditions under which the fault occurred (e.g. engine speed, engine temperature). Service-related faults can also be stored.

Limp-home operation
If a fault occurs during driving (e.g. failure of a component), this leads not only to an entry in the fault memory. Substitute values and emergency functions now come into force for calculating cylinder charge, fuel injection and ignition in such a way that the vehicle can continue to be driven, albeit with restricted comfort.

For this purpose, the ETC throttle-valve actuator has a limp-home position, in which the throttle valve is held in place by spring force. The engine speed is then restricted to low values so that the vehicle can continue to be driven (with impaired functions) in spite of this important actuating device having failed.

MED-Motronic

Requirements

With gasoline direct injection in SI engines, fuel consumption can be reduced by up to 20 % when compared with conventional manifold injection, and the CO_2 emissions caused by road traffic can be reduced with lasting effect.

In order for direct injection to be at all feasible, it must be possible during engine operation for coordinated alternation to take place between "charge stratification" at part load and operation with a homogeneous mixture at full load.

Essentially, the features required of the MED-Motronic engine-management system are:
- precise metering of the required injected fuel quantity,
- generation of the necessary injection pressure,
- definition of the start of injection, and
- introduction of the fuel directly and accurately into the engine's combustion chambers.

It must also coordinate the various torque demands on the engine in order then to be able to perform the required adjustments at the engine.

An important system interface delivers the internal torque from combustion. The torque-control structure is divided into the following areas of operation:
- torque demand,
- torque coordination and
- torque conversion.

The most important torque demand derives from the vehicle-operator command generated by the accelerator pedal, from the position of which the engine-management system interprets the demand to be made on the IC engine for a particular torque.

Further torque demands can also originate, for instance, from the transmission-shift control, the traction control system, or the electronic stability program.

Torque coordination is performed centrally in the engine-management system.

Design

As with manifold injection, the high-pressure direct-injection system is designed as a rail, or accumulator-type, injection system. In such systems, the fuel can be directly injected into the cylinders at any

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stipulated moment in time using electromagnetic high-pressure fuel injectors.

In comparison to the ME basic ECU, the ECU for the MED gasoline direct-injection system also incorporates a driver stage for triggering the pressure-control valves.

The intake air mass can be freely adjusted by the electronically controlled throttle valve (ETC). Precision measurement of air mass is made by means of a hot-film air-mass meter. The A/F mixture is monitored by universal lambda sensors LSF and LSU in the exhaust-gas flow upstream and downstream of the catalytic converter. They serve to control $\lambda = 1$ operation and lean-burn operation, and are also responsible for the precise control of catalytic-converter regeneration. It is important to set the exhaust-gas recirculation rate exactly, particularly in dynamic operation.

Operation

Fuel delivery and injection

The MED-Motronic accumulator injection system injects the fuel stored under pressure in the rail. Injection can take place at any moment in time, whereby electromagnetically controlled fuel injectors inject the fuel directly into the combustion chamber. It thus offers the following functions:
- free selection of the injection point,
- variable system pressure.

Low-pressure circuit:

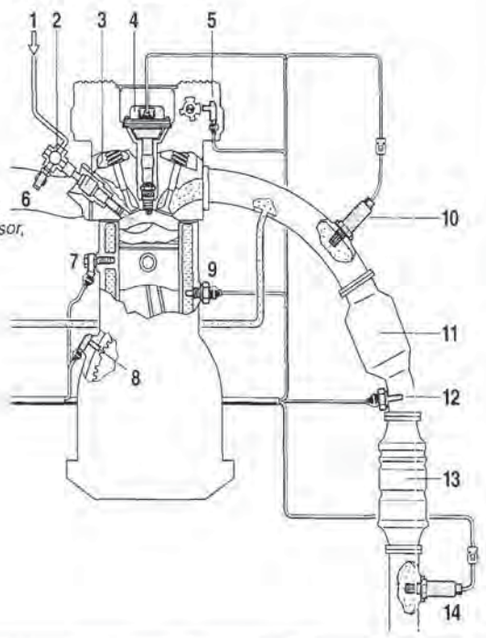
A primary pressure of 0.35 MPa (3.5 bar) is initially generated in a low-pressure circuit on the tank side which comprises an electric fuel pump with a mechanical pressure regulator connected in parallel. This feeds the high-pressure pump driven by the IC engine.

High-pressure circuit:

High-pressure pump: The function of the H.P. pump is to increase the fuel pressure

Direct-injection engine with MED-Motronic components

1 Fuel supply (high pressure),
2 Accumulator (rail),
3 Fuel injector,
4 Ignition coil with spark plug,
5 Phase sensor,
6 Pressure sensor,
7 Knock sensor,
8 Engine-speed sensor,
9 Engine-temperature sensor,
10 Lambda sensor (LSU),
11 Three-way catalytic converter,
12 Exhaust-gas temperature sensor,
13 NO_x catalytic converter,
14 Lambda sensor (LSF).



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from 0.35 MPa (3.5 bar) primary pressure to 12 MPa (120 bar), to ensure that pressure fluctuations in the rail remain at a minimum, and to ensure operation exclusively with fuel (to prevent any mixing with engine oil).

Rail: The rail must on the one hand feature sufficient elasticity to damp the pressure pulsations arising from the periodic fuel-extraction processes when injection takes place, and from the fuel-delivery pulsation generated by the H.P. fuel pump. On the other hand, it must be rigid enough for the rail pressure to adapt quickly to the requirements of engine operation.

Pressure sensor: The pressure sensor (welded diaphragm of high-grade steel as sensor element with measuring resistors) serves to register the pressure level in the rail.

Pressure-control valve: The pressure-control valve adjusts the system pressure across the entire engine operating range according to the map specifications, and irrespective of the injected fuel quantity and pump delivery rate. Excess fuel is returned to the intake side of the H.P. pump.

Fuel injectors: The fuel injectors are the central components of the gasoline direct-injection system and are directly connected to the rail. Start of injection and injected fuel quantity are defined by the triggering signal for the fuel injectors.

Mixture formation and combustion

A complex engine-management system is necessary for the gasoline direct-injection system to be exploited to the full in terms of low fuel consumption and high engine power output. A distinction is made between two basic operating modes in this respect:

Lower load range

In this range, the engine is run with a heavily stratified cylinder charge and a high level of excess air in order to achieve the lowest-possible fuel consumption. Retarded injection just before the ignition point is used with the aim of achieving the ideal state of having two zones in the combustion chamber: In one zone the combustible air-fuel mixture cloud at the spark plug, embedded in the second zone in the form of an insulating layer of air and residual exhaust gas. In this way, the engine can be run to a large extent unthrottled while gas-exchange losses are avoided. In addition, thermodynamic efficiency increases due to the avoidance of heat losses at the combustion-chamber walls.

Upper load range

As engine load and thus the injected fuel quantity increases, the stratified-charge cloud becomes increasingly richer. This would result in a deterioration in exhaust emissions, particularly with regard to soot. The engine is therefore operated with a homogeneous cylinder charge in this higher torque range.

Lambda coordination assumes the control function between $\lambda = 1$ and lean-burn operation. The fuel is injected during the intake stroke in order to ensure that fuel and air are thoroughly mixed.

Alternation between torque ranges

In the event of a switch between homogeneous and stratified-charge operation, it is crucial to control injected fuel quantity, air charge and ignition angle in such a way that the torque delivered by the engine to the transmission remains constant. The torque structure means that here also the important functions for controlling the electronic throttle valve have been transferred directly from the ME-Motronic. The throttle valve must be closed prior to the actual changeover from stratified to homogeneous operation.

Exhaust-gas treatment, catalytic-converter control

An important consideration with gasoline direct injection is that in stratified-charge operation the NO_X content in the very lean exhaust gas cannot be reduced by a three-way catalytic converter. Exhaust-gas recirculation with a high EGR rate helps to reduce the NO_X content of the exhaust gas by roughly 70 %. To comply with emissions-control legislation it is also absolutely essential to treat the NO_X emissions. To reduce this exhaust-gas content, the NO_X accumulator-type catalytic converter offers the greatest potential.

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Engine-test technology

Service and maintenance requirements for modern vehicles are declining steadily; electronic systems are essentially maintenance-free. But malfunctions can still occur. Factors such as wear, contamination and corrosion can impair the operation of engine and electronic systems, and settings can drift over time. Rapid and reliable diagnosis of malfunctions is the most important function of any service facility. It is important to distinguish between testing and diagnosis. Testing entails the determination of certain measured values for comparison with the prescribed specifications. Diagnostic procedures (e.g. engine diagnostics) attempt to correlate deviations from specified values with system functions, malfunction patterns and experience in order to determine the failure mode or to locate the defective component.

Engine diagnosis

Engine, ignition and fuel-metering systems are becoming more complex and less accessible. Universal, automated procedures, free from the effects of subjective influences, are thus essential elements of the computer-controlled testing program in the automotive service facility. These include:

- comparison of cylinder output by selective shorting of the ignition or smooth running analysis via current engine speed,
- comparison of compression via the shape of the starter current curve,
- determination of mixture distribution performed by selective measurement of the exhaust-gas HC content,
- analysis of the primary and secondary ignition-voltage patterns.

Electronic-system testing

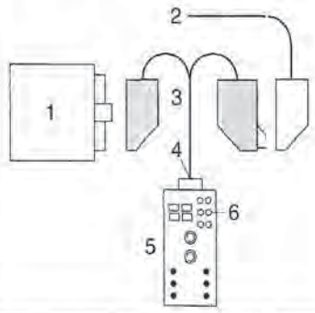
Testing of engine-related electronic systems is carried out using test equipment specifically designed to utilize on-board engine monitoring technology.

The test connection is established by inserting a universal test adapter between the plug and socket at the junction linking the peripheral device with the ECU. If only the peripheral device (sensor, actuator, wiring and power supply) is to be tested, then it is sufficient to connect the tester to the peripheral's plug, while the ECU is also connected for tests embracing dynamic operation. The individual electronic system requires only a single interchangeable, system-specific adapter cable. The program switches on the universal test adapter are then used to establish a logical test sequence specifically tailored to the requirements of the system.

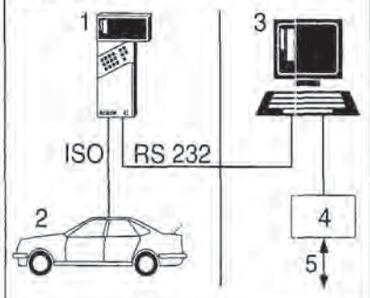
Once connected, the test unit provides a display of both measured values and of signals such as ignition and fuel-injection pulses.

When the ECU remains connected for operational testing, keys can be used to enter simulations of various operating conditions for test-unit evaluation.

Peripheral and functional testing
1 ECU, 2 System wiring harness, 3 System adapter cable, 4 Plug connection, 5 Universal test adapter, 6 Sockets for test unit



System test with ECU diagnostics
1 Test unit, 2 Test vehicle, 3 Information system, 4 Modem, 5 Long-distance data transmission.



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ECU diagnostics

The dominant role being assumed by electronic systems in the vehicle makes it necessary to devote increased attention to the problems associated with service. In addition, because essential vehicle functions are becoming increasingly dependent upon electronics, these systems must satisfy stringent reliability requirements, while emergency default programs are required to deal with system errors.

The solution is to incorporate electronic system-diagnosis functions in the ECU. These rely on the electronic "intelligence" already in place in the vehicle to continuously monitor the system, detect faults, store the fault data and perform diagnostics.

For instance, the ECU carries out its own self-check as follows: Programmed memory chips are provided with test patterns which can be retrieved and used for comparisons. For program memories, a comparison with test sums is employed to ensure that data and programs are correctly stored. The data and address buses are included in the test program.

Sensors are tested for plausibility within specified limits, while open and short circuits are also recognized. Final-control elements can be tested during activation using current-draw limits.

The "off-board test units" used for the evaluation of "on-board diagnostics" employ a communications interface as defined in ISO 9141. The serial port can maintain communications at rates ranging from 10 baud to 10 kbaud. It is designed as a single- or two-wire port, allowing connection of several control units to a central diagnosis plug.

A stimulation address is transmitted to all of the connected ECUs, whereupon each unit recognizes its address and responds by transmitting back a baud-rate recognition word.

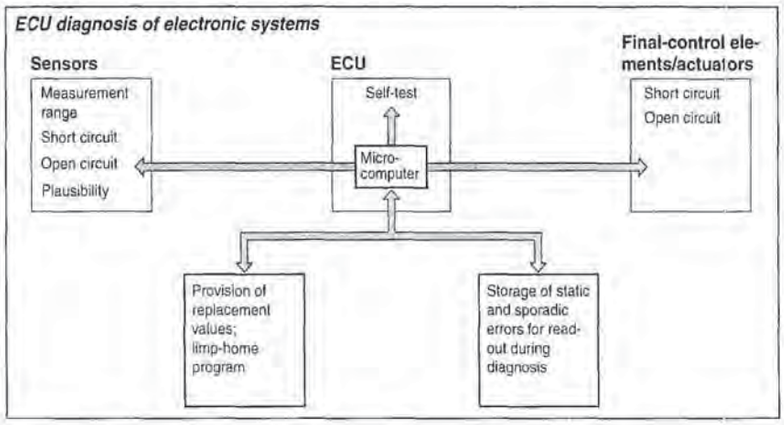
The test unit monitors the period between pulse flanks to determine the sender's baud rate, which it then adopts automatically. The key bytes that follow (assigned by the DIN Motor-Vehicle Committee) specify the protocol for subsequent data communications.

The programmable test unit converts the received data into diagnosis sequences and plain-text information designed specifically for the respective system.

The ECU diagnostic capabilities include:
- identification of system and ECU,
- recognition, storage and readout of static and sporadic malfunctions together with the error path, failure mode and associated parameters,
- readout of current actual values, switching conditions, specifications,
- stimulation of system functions, and
- programming of system variants.

Individual programs for the test unit are stored in plug-in modules, while updates and communications with data systems are supported by a data interface.

ECU diagnosis of electronic systems



LPG systems

Liquefied petroleum gas

Liquefied petroleum gas assumes a liquid state at pressures of 2...20 bar, depending on the propane/butane ratio and the temperature (see chapter on "Fuels", P. 238).

At the end of the '90s, LPG was being consumed by internal-combustion (IC) engines at a rate of approximately 10.3 million metric tons annually (of which 2.6 million tons were consumed in Europe). Should efforts to utilize the gas contained in petroleum succeed, then these figures could increase exponentially. The mineral-oil tax is a decisive factor in determining the profitability of liquefied petroleum gas.

Natural gas as engine fuel

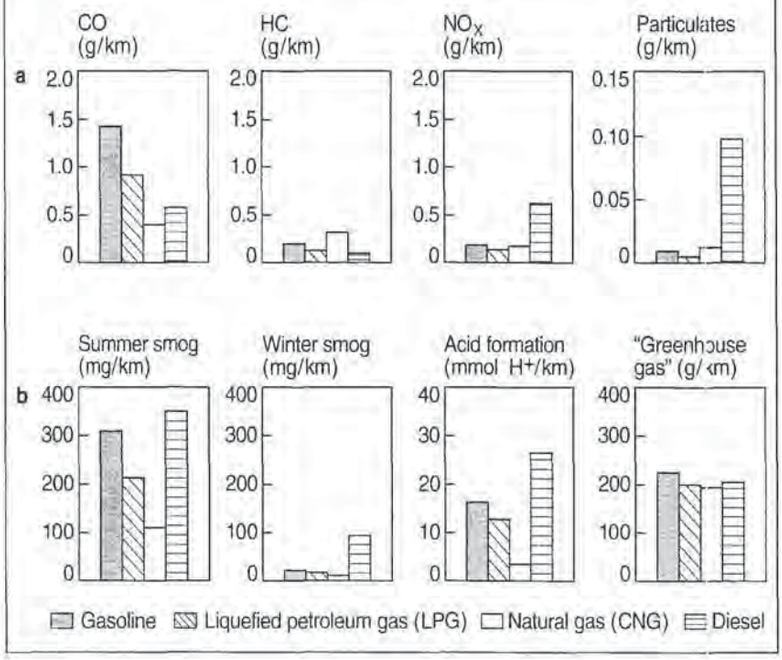
Reserves of natural gas are extensive. Together with it being used nowhere near as much as crude oil, this fact makes natural gas a very interesting alternative fuel for automotive applications. Both the LPG equipment configurations and the emissions produced would be similar to those for the combination of propane and butane known as LPG (see illustration). Natural gas can be transported in the vehicle either as a high-pressure gas (160...200 bar) or in liquefied form (at –160 °C) in an insulated tank; the disadvantage of the former mode lies in its limited operating range.

The only real difference between concepts based on natural gas and LPG is the way the gas is transported in the vehicle; actual differences in operating principles are minimal.

Operation on LPG

Any vehicle equipped with an IC engine can be converted for operation on LPG. For the most part, spark-ignition (SI) engines are re-equipped for dual-fuel operation (system can be switched between gasoline and LPG). LPG-powered taxis and buses are

Exhaust emissions and effects with gasoline, LPG, CNG and diesel
Empirical values taken from five spark-ignition and five diesel-engine vehicles, status 1993 in Europe, including indirect emissions (manufacture, transport, etc.).
a) Pollutant emissions, b) Pollutant effect ("greenhouse gas" assessment, status 1998).



generally set up for single-fuel (LPG only) operation, while regulations require this configuration on gas-powered industrial trucks intended for indoor use. When engines are converted, it should be remembered that they operate as naturally aspirated engines while running on LPG (consumption in liters increases by approx. 25% as compared to gasoline).

Exhaust emissions

Because LPG mixes well with air, emissions (of CO_2 and other components such as polycyclic aromatic hydrocarbons) are substantially lower than those produced by gasoline-burning engines, and are lower even than those produced by fuel-injection engines equipped with three-way closed-loop catalytic converters. LPG contains no lead or sulphur compounds. It's very good combustion characteristics are complemented by excellent mixture formation and distribution properties. These characteristics are even more significant at low temperatures.

Advantages

- Extremely economical for drivers who cover large distances.
- Assuming the same level of technology (electronic control systems, etc.), emissions from an LPG engine are substantially lower than those achieved with gasoline or diesel fuel.

Disadvantages

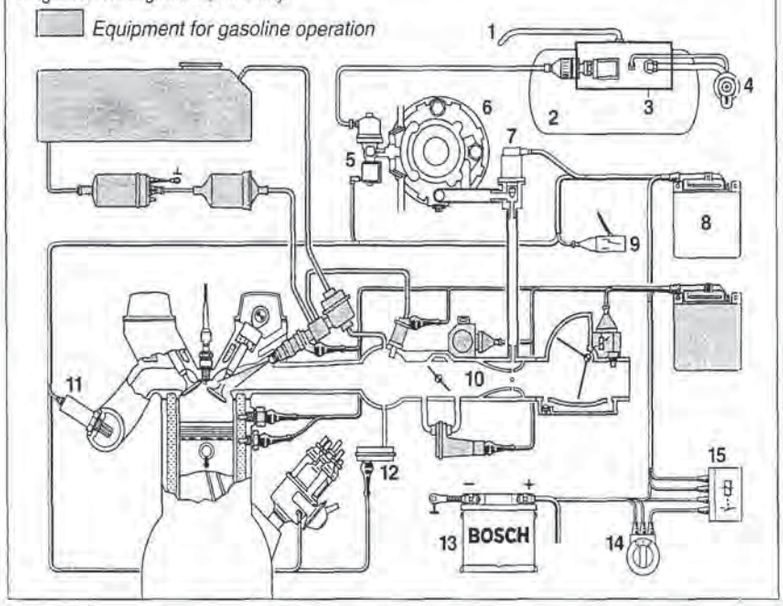
- Lower cruising ranges and increased fuel volumetric consumption compared to gasoline (although actual energy use is not higher than with gasoline, lowest with diesel engine).
- Special safety precautions are necessary, as LPG is pressurized.
- Pressurized gas cylinders require a lot of space, as the actual capacity is only 80% of cylinder volume (the remainder serving as expansion room for the gas).

LPG system

In Germany, professional installation of

Schematic diagram of an LPG system (carburetor principle)

1 Ventilation line for tank fittings, 2 LPG tank, 3 Housing with tank fittings, 4 External filler valve (designed to interrupt refueling at 80% of container capacity), 5 Gas shut-off valve, 6 Pressure regulator for evaporator, 7 Servomotor for gas control, 8 ECU, 9 Gas/gasoline changeover switch, 10 Venturi mixing device, 11 Lambda oxygen sensor, 12 Vacuum sensor, 13 Battery, 14 Ignition/starting switch, 15 Relay.



the LPG system in a specialist workshop is followed by a trip to the TÜV or TÜA (German inspection authorities) to secure operating approval. This inspection is based on the "Guidelines for inspecting vehicles with engines powered by liquefied gases" as issued by the Federal Minister of Transportation. Uniform regulations for Europe are being drawn up.

A modern LPG system will incorporate the following components:
- LPG tank,
- external filler valve designed to interrupt refueling at 80 % of container capacity,
- flow-interrupt valve,
- evaporator pressure regulator with cooling system,
- venturi mixing unit/injector,
- electronic control unit (ECU),
- servomotor for controlling gas flow,
- switch for alternating between LPG and gasoline operation.

The LPG flows from the LPG tank to the evaporator pressure regulator, where it is vaporized and its pressure reduced. The ECU processes the signals from the Lambda (O_2) and vacuum sensors, which serve as references for controlling the servomotor used to regulate the flow of liquefied gas to the venturi mixing unit.

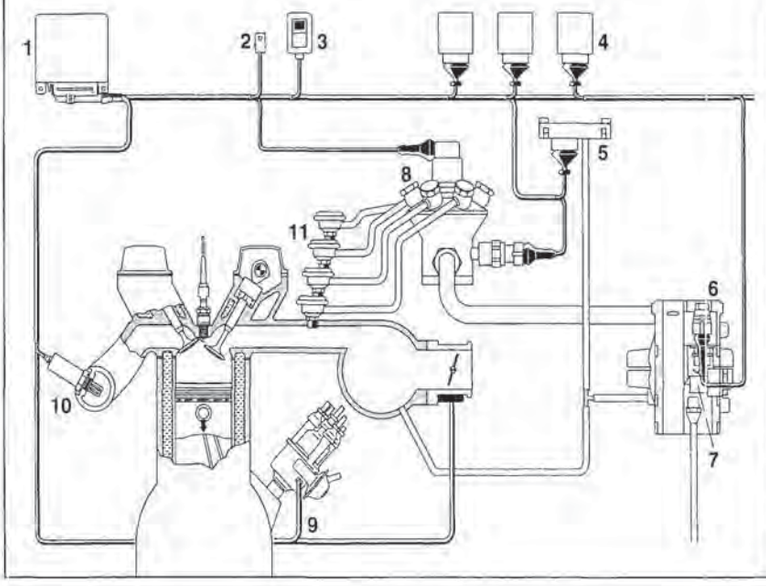
The flow-interrupt valve shuts immediately when the ignition is switched off. A gasoline/LPG switch installed in the instrument panel allows the operator to alternate between the two fuels.

LPG tanks

As LPG tanks are used to store pressurized gas, they are subject in Germany to the "TRG 380" technical regulations. At the factory, each tank receives official technical approval with certification.

They are equipped with an external filler valve (designed to interrupt refueling at 80 % of container capacity) as well as an electromagnetic discharge valve, and have a capacity of 40 ... 128 l for passenger cars.

Schematic diagram of an LPG system (injection principle)
1 ECU, 2 Diagnosis plug, 3 Fuel selection switch, 4 Relay, 5 Air-intake pressure sensor, 6 Evaporator pressure regulator, 7 Flow-interrupt valve, 8 Distributor with step motor, 9 RPM signal, 10 Lambda (O_2) sensor, 11 Gas injector nozzle.



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Natural-gas operation of spark-ignition engines

Against the backdrop of worldwide efforts to reduce CO_2 emissions and to comply with ever more stringent exhaust emission limits, natural gas is gaining increasing significance as an alternative fuel.

Properties and storage of natural gas

The main component of natural gas is methane (CH_4), making up 80...99 %. The remainder consists of inert gases such as carbon dioxide, nitrogen and other low-order hydrocarbons. A differentiation is made between L-gas (80...90 % methane) and H-gas (>90 %) depending on the gas quality.

Natural gas can be stored both in liquid form at −162 °C as LNG (Liquefied Natural Gas) or in compressed form at pressures of up to 200 bar as CNG (Compressed Natural Gas). In view of the great expense involved in storing the gas in liquid form, natural gas is used in compressed form in virtually all applications.The low energy density of natural gas is a particular disadvantage, making large storage tanks necessary. Metal-hydride storage tanks represent a further storage option, although they are not used for reasons of cost.

Mixture formation

In most systems, gas is injected into the intake manifold as in conventional multipoint gasoline-injection systems. A low-pressure common rail supplies the injector valves that inject intermittently into the intake manifold. Mixture formation is simplified by the completely gaseous supply of fuel, as natural gas does not condense on the intake manifolds and does not form a film on the walls. This has a favourable effect on emissions, particularly during the warm-up phase.

The output of the natural-gas engine is approx. 10...15 % lower than that of the gasoline engine due to the lower fuel mass necessary for stoichiometric combustion (17.2 : 1 ratio) as well as a lower volumetric efficiency due to the injected natural gas. Higher compression can boost performance while simultaneously increasing efficiency. The extremely high antiknock quality of natural gas (120 RON) enables very high compressions of approx. 13 : 1 (8 : 1 for regular gasoline). As a result, the natural-gas engine is ideally suited for turbocharging. As piston displacement decreases, efficiency increases due to additional de-throttling and reduced friction.

A further improvement in efficiency can be achieved by lean-mixture operation up to $\lambda = 1.7$. A lean mixture reduces combustion temperatures while at the same time further de-throttling the engine.

Emissions

Natural-gas vehicles are characterized by low CO_2 emissions due to the favourable hydrogen/carbon ratio (H/C ratio) of almost 4 : 1 (gasoline: 2.3 : 1) and the resulting shift in the main combustion products CO_2 and H_2O. Apart from the virtually particle-free combustion, in conjunction with a three-way closed-loop catalytic converter only very low levels of the pollutants NO_X, CO and NMHC ("non-methane hydrocarbons": the sum of all hydrocarbons minus methane) are emitted. Methane is classified as nontoxic, and is therefore not considered to be a pollutant.

In lean mixture mode the NO_X emissions are higher than in $\lambda = 1$ mode with a three-way catalytic converter. In the same way as in gasoline operation, this disadvantage can be largely eliminated by using more expensive exhaust-gas treatment methods (e.g. NO_X catalytic converter).

Natural-gas-engine applications

In view of their limited range, natural-gas engines are used almost exclusively in local public transportation fleets (e.g. buses and taxis). Bivalent systems which can be easily changed over from natural gas to gasoline operation are used primarily in passenger cars.

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Operation on alcohol (spark-ignition engines)

The limited availability of fossil fuels has led to an increased effort to develop engines and injection systems capable of using alcohols such as ethanol and methanol as alternative fuels ("Alternative fuels", P. 246). Due to its non-availability, virtually the only place where ethanol is used is Brazil. In the US (and in California in particular), increasing attention is being focused on methanol, which generates lower emissions: reduced NO_X and CO_2 along with reduced ozone and smog formation.

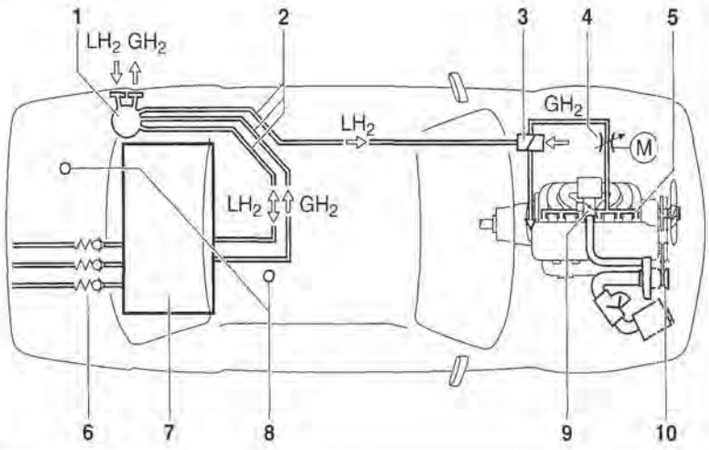
In the absence of comprehensive methanol-distribution networks to ensure universal availability, engines and engine-control systems must be designed for flexible dual-fuel operation (ranging from pure gasoline to max. 85% methanol). Alcohol places special, particularly critical demands upon engines and fuel-delivery components. Moisture, acids and gums contained in the fuel pose a hazard to metals, plastics and rubber. Because methanol has a high antiknock quality, engines designed to run exclusively on methanol have substantially higher compression ratios than gasoline engines, making them more efficient. On the other hand, methanol's low calorific value means that fuel consumption is almost doubled, necessitating higher fuel supply rates, greater tank volumes and special injectors.

Suitable Lambda (O_2) sensors can be employed for optimal emissions control with a catalytic converter. Special lubricants are able to maintain long-term stability in the face of this aggressive fuel and its combustion products.

Pilot control of the fuel mixture is facilitated by a fuel sensor which sends a signal to the ECU reflecting the proportion of methanol in the fuel.

Suitable programs implement the necessary mixture and ignition corrections applicable to a particular set of engine operating characteristics.

Hydrogen-powered passenger car with spark-ignition engine (BMW 735i)
LH_2 Liquid hydrogen, GH_2 Gaseous hydrogen. 1 Valve block for LH_2 fueling and GH_2 supply (vacuum-insulated), 2 Hydrogen lines, vacuum-insulated, 3 LH_2 Evaporator, 4 Metering valve for regulating power with electronic control, 5 Hydrogen injectors, 6 Overcurrent and safety valves, 7 Liquid-hydrogen tank with vacuum super-insulation, 8 Hydrogen sensors for automatic leak monitoring, 9 Throttle valve for gasoline operation with electronic control, 10 Variable-speed centrifugal supercharger.



Operation on hydrogen (spark-ignition engines)

Although the production of hydrogen, the required infrastructure and refueling all pose difficult problems, technically feasible solutions are on the horizon. Producing hydrogen by means of electrolysis necessitates sufficient quantities of electrical power in the form of solar energy or nuclear power.

Storing hydrogen in the vehicle

Gaseous storage in pressurized tanks
High pressures (300 bar) are required for storage in gaseous form. This results in high weight along with safety risks.

Liquid storage (cryogenic tank)
Liquid storage represents the best alternative with regard to both weight and energy density (present operating range approx. 300 km). The extremely low temperature required (–253 °C) places substantial demands on thermal insulation. Residual heat causes loss of hydrogen via the safety valves at a rate of about 2 % per day when the vehicle is parked. An electric evaporator maintains the specified tank pressure during operation.

Metal-hydride tanks
Hydrides are produced as hydrogen is absorbed by a metallic powder. This is an exothermic process, i. e. heat must be dissipated during fueling. There are no storage losses.

The disadvantages associated with the low energy density (range: 120 km) and high materials costs are to a degree offset by uncomplicated safety technology.

Methylcyclohexanol storage
This type of storage employs a catalyst to dehydrate the hydrogenous methylcyclohexanol at 500 °C. The by-products are hydrogen and recyclable toluene.

Mixture formation

Regardless of storage mode, up to now all systems inject gaseous hydrogen into the intake manifold. Although a number of advantages could be gained by injecting extremely low-temperature hydrogen directly into the combustion chamber (improved charge for higher output, cool mixture for low NO_X emissions, no danger of backfiring), the short injector-valve service life means that this type of system is not likely to appear in production in the near future.

Current external mixture-formation concepts rely on a continuous-injection system in which a central electric metering valve and a hydrogen distributor conduct the vaporized hydrogen to the individual intake tracts.

Backfiring into the intake passage is prevented by lean mixtures or supplementary water injection. A supercharging device can be used to compensate for a portion of the power loss associated with lean operation.

One alternative, based on intermittent sequential injection of hydrogen into the intake manifold, is currently in the development stage.

This system's unlimited range of options for injection timing allows it to inhibit backfiring almost completely, even with rich mixtures.

Injector valve and electronic-control system must operate with extreme precision and short valve-opening times the technical requirements are thus substantial.

Emissions

During combustion, pure hydrogen (H_2) oxidizes to form water (H_2O). No CO_2 is produced by the combustion process. Provided no fossil fuels are used in its production, H_2 is thus the only fuel which can be used to avoid all CO_2.

Electric drive is the only alternative which can make a similar claim. Future NO_X emission limits can be met by lean mixtures or a system for catalytic control of emissions (still to be developed).

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Exhaust emissions from spark-ignition engines

Combustion products

Complete combustion

The by-products of complete gasoline combustion are carbon dioxide and water.

Incomplete combustion

Unburned hydrocarbons:
C_nH_m (paraffins, olefins, aromatic hydrocarbons)
Partially-burned hydrocarbons:
$C_nH_m \cdot CHO$ (aldehydes),
$C_nH_m \cdot CO$ (ketones),
$C_nH_m \cdot COOH$ (carboxylic acids),
CO (carbon monoxide).
Thermal crack products and derivatives:
C_2H_2, C_2H_4, H_2 (acetylene, ethylene, hydrogen, etc.), C (soot), polycyclic hydrocarbons.

Combustion by-products

From atmospheric nitrogen: NO, NO_2 (nitrogen oxide). From fuel impurities: sulfurous oxides.

Oxidants

The following oxidants are produced when exhaust gas is exposed to sunlight: organic peroxides, ozone, peroxy-acetyl-nitrates.

Properties of exhaust-gas components

Major components

The major proportion of the exhaust gas is composed of the three components nitrogen, carbon dioxide and water vapor. These are nontoxic. However, emissions of CO_2 – largely the result of fuel combustion – are becoming more and more important due to their contribution to the "greenhouse effect".

Minor components

Carbon monoxide CO: A colorless, odorless and tasteless gas. Inhalation of air with a volumetric concentration of 0.3% carbon monoxide can result in death within 30 minutes. The CO-content of the exhaust gas from spark-ignition engines is especially high at idle. It is therefore imperative that the engine never be run in a closed garage!
Nitrogen monoxide NO: A colorless, tasteless and odorless gas; in air it is gradually converted into NO_2. Pure NO_2 is a poisonous, reddish-brown gas with a penetrating odor. The concentrations found in exhaust gases and in extremely polluted air can induce irritation in mucous membranes. NO and NO_2 are generally referred to collectively as oxides of nitrogen NO_X.
Hydrocarbons are present in exhaust gases in a variety of forms. When exposed to sunlight and nitrous oxide, they react to form oxidants which irritate the mucous membranes. Some hydrocarbons are considered to be carcinogenic.
Particulates (particulate matter) in accordance with American regulatory practice, are defined as all substances (except unbound water) which under normal conditions are present in exhaust gases in a solid (ash, carbon) or liquid state.

Mixture formation

The fuel used in spark-ignition (SI) engines is more volatile than diesel fuel, while the air-fuel mixing process prior to combustion also extends over a longer period than in a diesel engine. The result is that spark-ignition engines operate on a more homogenous mixture than their diesel counterparts. Spark-ignition engines run on a mixture in the stoichiometric range ($\lambda = 1$). Diesel engines, on the other hand, always operate with excess air ($\lambda > 1$). An air deficiency ($\lambda < 1$) results in increased soot, CO and HC emissions.

Combustion characteristic

The combustion characteristic defines the combustion as a function of time, and applies the ratio of the already combusted fuel to the fuel awaiting combustion.

Here, the efficiency, the combustion temperature, and therefore the fuel consumption and NO_X emissions are particularly influenced by the position of maximum energy conversion referred to piston TDC.

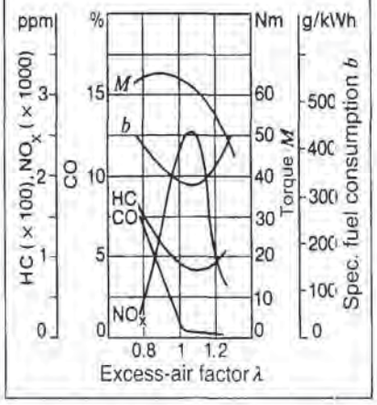
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In the spark-ignition engine, the ignition timing initiates the combustion process, while the start of injection initiates combustion in a diesel engine.

Emissions control

The methods used to influence the composition of the SI-engine exhaust gases are divided into two basic categories: engine-design measures and exhaust-gas treatment. The selection of procedures to be employed in any given country is determined by the legal regulations in force there. The major industrial nations, with their important markets, have been moving toward implementation of the stringent American exhaust emissions regulations (or have already implemented them). Compliance with this legislation necessitates using emission control systems which incorporate the 3-way catalytic converter, a principle which has already proven itself in the USA.

The excess-air factor λ influences:
Exhaust-gas composition (CO, NO_x, HC), torque (M) and specific fuel consumption (b). The values apply to the part-load range of a spark-ignition engine operating at a constant moderate speed and cylinder charge (λ values implemented in the vehicle engine range from approx. 0.85...1.15, depending on the operating point).



Engine-design measures

Setting the A/F ratio

The excess-air factor λ of the A/F mixture delivered to the engine has a dominating effect on the composition of the exhaust gas. The engine produces its maximum torque at approximately λ = 0.9; thus this ratio is generally programmed for full-load operation. A certain level of excess air is required for favourable fuel consumption. This coincides with the setting for low CO and HC emissions; oxides of nitrogen (NO_x) however, are at a maximum at this ratio. Excess-air factors of λ = 0.9...1.05 are selected at idle. An excessively lean mixture results in the engine's lean misfire limit (LML) being reached or exceeded, and as the mixture becomes progressively leaner, misfiring causes a rapid increase in HC emissions. For overrun (trailing-throttle) operation, it is frequently necessary to enrich the mixture (λ = 0.9) in order to maintain an ignitable mixture, while at the same time air is added to avoid excessive manifold vacuum.

Yet another option for overrun operation is to completely interrupt the fuel supply to the engine at speeds above idle (overrun fuel cutoff). Precise mixture control (P. 465) is achieved by the use of electronic fuel-injection systems (EFI).

Mixture formation

Mixture formation embraces not only setting the correct A/F ratio, but also the quality of the A/F mixture which actually enters the combustion chamber. The fuel's homogeneity, its stratification patterns, and its temperature at the instant of ignition are all essential factors in determining combustibility and combustion characteristic, with consequent effects upon exhaust-gas composition. Homogenous mixtures and controlled stratification (rich mixture at the spark plug, lean mixture in the vicinity of the combustion-chamber walls) are examples of two different development options.

Uniform distribution

Maximum engine efficiency can only be achieved if every cylinder is operated with the same excess-air factor. This necessitates a system which ensures that both air and fuel are distributed evenly among the individual cylinders.

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Exhaust-gas recirculation (EGR)
Exhaust gas can be conducted back to the combustion chamber to reduce peak combustion temperatures. Higher combustion temperatures induce an over-proportional increase in NO_X formation, and because exhaust-gas recirculation (EGR) reduces combustion temperatures, it represents a particularly effective means of controlling NO_X emissions. Extensive optimization of EGR can also lead to a reduction in fuel consumption. EGR can be implemented in either of two ways:
- Internal exhaust-gas recirculation is achieved with appropriate valve timing (overlap),
- External exhaust-gas recirculation employing controlled EGR valves.

Valve timing
Large valve overlaps (early opening of the inlet valve) increase the internal exhaust-gas recirculation, and can therefore help to reduce NO_X emissions. However, since the recirculated exhaust gas displaces fresh A/F mixture, early opening of the inlet valve also leads to a reduction in the maximum torque. In addition, excessively high exhaust-gas recirculation, particularly at idle, can lead to combustion misfire which in turn causes an increase in HC emissions. An optimum can be found with variable valve timing (P. 387), in which the valve timing is varied as a function of the operating point.

Compression ratio
It has long been recognized that the enhanced thermal efficiency associated with high compression ratios represents an effective means of improving fuel economy. However, the increase in peak combustion temperature also results in higher NO_X emissions.

Combustion-chamber design:
Low HC emissions are best achieved with a compact combustion chamber featuring a minimal surface area and no recesses. A centrally-located spark plug with short flame travel produces rapid and relatively complete combustion of the mixture, resulting in low HC emissions and reduced fuel consumption. Induced combustion-chamber turbulence also provides rapid combustion. Combustion chambers optimized in this way feature favourable HC emissions at $\lambda = 1$ and improve the engine's lean-mixture operation.

A thoroughly optimized combustion chamber design coupled with external measures (such as intake swirl) produce a lean-burn engine capable of running on mixtures in the range of $\lambda \approx 1.4...1.6$. Although the lean-burn engine features low exhaust emissions and excellent fuel economy, it does require catalytic exhaust-gas treatment in order to meet the most stringent emissions limits for CO, HC, and NO_X. Particularly due to the fact that developments in the aftertreatment of NO_X in lean exhaust gas are still in their infancy, the lean-burn engine has up to now only been successful in Europe and Japan, and only in the case of a few models using lean/mix concepts which compromised between emissions and fuel consumption. A new way of substantially improving the lean-running behavior of the spark-ignition engine is to inject fuel directly into the combustion chamber with stratified charge. In this case, accumulator-type catalytic converters are favoured for exhaust gas treatment.

The lean-running characteristics achieved by combustion-chamber design and induced turbulence can also be applied to implement high EGR rates for designs with $\lambda = 1$. Here, a reduction in fuel consumption can be easily achieved with exhaust-gas treatment, but not to the same degree as in lean-burn engines.

Ignition system
The design of the spark plug, its position in the combustion chamber, together with spark energy and spark duration all exercise a major influence on the ignition, the mixture's combustion characteristic, and therefore on emissions levels. The significance of these factors increases with the leanness ($\lambda > 1.1$) of the mixture. Ignition timing exerts a decisive effect on both exhaust emissions and fuel economy. Using the firing point for optimum fuel economy as a baseline, the timing is retarded to a point at which the exhaust valve opens before the combustion process is completed. Excess oxygen in the exhaust system can cause a thermal post-reaction to

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occur. Although this process reduces unburned hydrocarbons, it also leads to increased fuel consumption. NO_X emissions are low due to the low combustion-chamber temperatures. Fuel consumption and NO_X and HC emissions increase when the firing point is set earlier compared to the optimum.

Crankcase ventilation (blowby)

The concentration of hydrocarbons in the crankcase can be many times that found in the engine's exhaust gases. Control systems conduct these gases to a suitable point in the engine's intake tract, from where they are drawn into the combustion chamber for burning. Originally, these gases were allowed to escape untreated directly into the atmosphere; today, crankcase emission-control systems are a standard legal requirement.

Exhaust-gas treatment

Thermal afterburning

Before today's catalytic treatment of exhaust emissions became standard, initial attempts to reduce emissions utilized thermal afterburning. This method employs a specific residence time at high temperatures for burning the exhaust-gas components which failed to combust during normal combustion in the engine cylinders. In the rich range ($\lambda = 0.7...1.0$), the process must be supported with supplementary air injection (secondary air). In lean-burn engines ($\lambda = 1.05...1.2$), the residual oxygen in the exhaust gas supports the afterburning process.

In the past, mechanical pumps driven by belts directly from the engine were used for secondary-air injection. Since such air injection is only required during the engine's warm-up period, these pumps are switched using electromagnetic couplings. Lower-priced blower pumps powered by electric motors are quickly superseding the former mechanical versions.

Due to its lack of potential, particularly in maintaining low NO_X limits, thermal afterburning alone is currently considered to be of little significance. It can be used, however, to reduce emissions of HC and CO during the operating phase in which the catalytic converter has not yet reached its operating temperature. Because it greatly reduces the time taken for the catalytic converter to reach its operating temperature, thermal aftertreatment with air injection during the engine warm-up phase in combination with catalytic aftertreatment will play a major role in achieving compliance with more stringent emission limits in the future.

Catalytic afterburning

The catalytic converter is composed of a carrier substrate, which serves as a base for the catalytic material, mounted within a housing using vibration-proof, heat-insulated supports. Granulate and ceramic or metallic monolith structures are employed as substrate materials. The suitability of the monolith structure for automotive applications has been demonstrated over the course of an extended development period. It offers the following advantages: maximum utilization of catalytic surface, durability combined with physical strength, low thermal retention and limited exhaust back-pressure. The active catalytic layer consists of small quantities of noble metals (Pt, Rh, Pd), and is sensitive to lead. For this reason, it is essential that engines with catalytic converters be run on unleaded fuel exclusively, as lead destroys the effectiveness of the active layer. The conversion rate is largely a function of operating temperature; no significant conversion of pollutants takes place below an operating temperature of approx. 250 °C. Operating temperatures of approx. 400...800 °C provide ideal conditions for maximum efficiency and extended service life.

Installing the catalytic converter directly adjacent to the engine means that it more quickly reaches operating temperature since the exhaust gases are hotter. This results in improved efficiency, but at the cost of high thermal stresses. Because the maximum permissible operating temperature lies just slightly above 1000 °C, the units are generally installed at a less critical location under the floor of the vehicle. Engine malfunctions, such as ignition miss, can cause the temperature in the catalytic converter to increase to the point where the substrate melts, resulting in de-

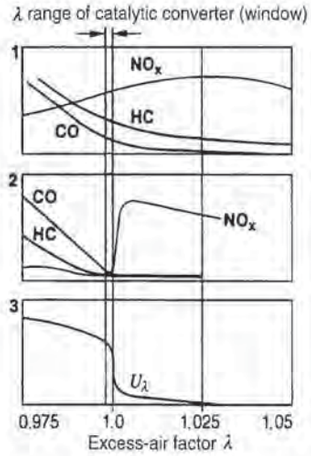
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struction of the unit. This must be prevented by using reliable, maintenance-free ignition systems. Oxidation-type catalytic converters oxidize CO and HC either by utilizing the excess air supplied by lean engine mixtures or by relying on secondary-air injection. Reduction-type converters on the other hand operate with air deficiency, and thus without air injection, to reduce NO_X levels.

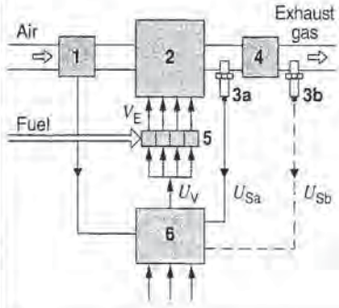
The reduction- and oxidation-type catalytic converters can also be combined in series to produce a dual-bed catalytic converter. This device uses air injection between the two converters to reduce not only NO_X emissions but also HC and CO levels. Its disadvantages include a more complex design (two converters, air injection) and the need to operate the engine in a range with higher fuel consumption ($\lambda = 0.9$).

The three-way or selective catalytic converter with lambda closed-loop control has proven to be the most effective concept for exhaust-gas aftertreatment. It is capable of providing the required reduction of all three pollutants provided the engine is operated with a stoichiometric mixture. The "window" for treatment of the three toxic components is narrow; this means that an open-loop fuel-metering system cannot be used with this concept.

Catalytic converter efficiency as a function of the excess-air factor λ
1 Exhaust emissions upstream of 3-way catalytic converter, 2 Exhaust emissions downstream of 3-way catalytic converter, 3 Electric signal from Lambda oxygen sensor. U_λ Sensor voltage.



Functional schematic of the Lambda closed-loop control
1 Air-mass meter, 2 Engine, 3a Lambda oxygen sensor 1, 3b Lambda oxygen sensor 2 (only if required), 4 Catalytic converter, 5 Fuel injectors, 6 ECU. U_S Sensor voltage, U_V Injector triggering voltage, V_E Injected fuel quantity.



Lambda closed-loop control

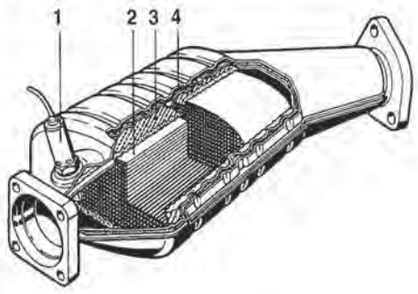
In the US, Europe and Japan, catalytic aftertreatment of the exhaust gas using a closed-loop three-way catalytic converter has proven to be a highly effective means of meeting the current low emission limits for CO, NO_X and HC.

Approximately 14.7 kg of air are required for complete combustion of 1 kg of gasoline. The excess-air factor λ (Lambda) is used to define this A/F ratio; it is the ratio of the actual A/F ratio to the stoichiometric A/F ratio. Two main control concepts for exhaust gas optimization are used in the spark-ignition engine:

Closed-loop control for λ = 1
This concept is the most effective for minimizing pollutants. The engine must be operated within a very narrow range in which $\lambda = 1 \pm 0.005$ (catalytic-converter window). Such precision can only be achieved with precise closed-loop control of the A/F mixture with a Lambda oxygen sensor installed upstream of the converter. A second Lambda sensor down-

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Two-bed, 3-way catalytic converter with Lambda oxygen sensor
1 Lambda oxygen sensor, 2 Ceramic monolith, 3 Wire screen, 4 Heat-resistant double shell.



stream of the catalytic converter increases the precision even further.

Closed-loop control for $\lambda > 1$ (lean-burn control)

The main advantage here lies in the reduction of fuel consumption as a result of lean-burn (non-throttled) operation. The success of this concept depends to a high degree upon the availability of catalytic converters which are able to reduce NO_X emissions during lean-burn operation. The lean-misfire limit (LML) for spark-ignition engines in the lean-burn range is reached at $\lambda \approx 1.7$ regardless of engine design measures.

Lambda oxygen sensors

Zirconium-dioxide sensor ($\lambda = 1$)

This sensor operates according to the principle of a galvanic oxygen concentration cell with solid electrolyte whose ceramic is made of zirconium dioxide and yttrium oxide. This oxide mix is a practically pure oxygen-ion conductor and separates the exhaust gas from the surrounding air.

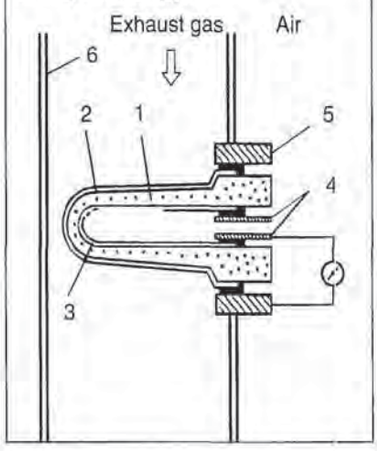
An electrical voltage is generated across the platinum-cermet electrodes in accordance with the Nernst principle.

$$U_s = R \cdot \frac{T}{4F} \cdot \ln\left(\frac{p_{O_2}''}{p_{O_2}}\right)$$

R general gas constant, F Faraday constant, T absolute temperature, p_{O_2}'' oxygen partial pressure of reference load, p_{O_2} oxygen partial pressure in exhaust gas.

The catalytically active electrodes must be in thermodynamic balance before measurement of the oxygen concentration can be applied for reaching conclusions on the Lambda values. In this case, the sensor's characteristic shows a jump at $\lambda = 1$. When high concentrations of exhaust-gas components which are not in thermodynamic balance are incompletely converted, the sensor's characteristic shows shifts in the excess-air factor λ.

Zirconium-dioxide oxygen sensor
1 ZrO_2 solid electrolyte, 2 Pt outer electrode, 3 Pt inner electrode, 4 Contacts, 5 Housing contact, 6 Exhaust pipe.



These must be compensated for by design measures taken at the engine. The sensor characteristic's dependence on temperature can be minimized by heating the sensor electrically.

Resistive sensor

Due to the change in the O_2 vacancy concentration of the oxide, oxidic semiconductors such as titanium oxide or strontium titanate change their volume conductivity.

The temperature-dependency of the conductivity is superimposed upon this effect, so that the determination of the Lambda value depends upon the quality of the required temperature control – open-loop or closed-loop (by varying the amount of heat generated).

Lean sensor

In its lean range (λ in the vicinity of 1), the "voltage-jump" (zirconium-dioxide) sensor as described above has only been used to a limited extent. Special measures taken to stabilize the sensor, together with the use of a high-capacity heater (18 W) permit operation to approx. $\lambda \leq 1.5$. The lean sensor using the limit-current principle permits the measurement of any values above $\lambda = 1$.

O_2 ions are pumped from the cathode to the anode when an external voltage is applied across two electrodes deposited on a ZrO_2 ceramic substrate. Since a diffusion barrier prevents the flow of O_2 molecules from the exhaust gas to the cathode, current saturation is reached above a given pump-voltage threshold value. By approximation, the resulting limit current is proportional to the oxygen concentration. This sensor principle is particularly suitable for lean-burn concepts. However, in lean/mix concepts in which a control setpoint $\lambda = 1$ is often desired, the broadband sensor is more suitable.

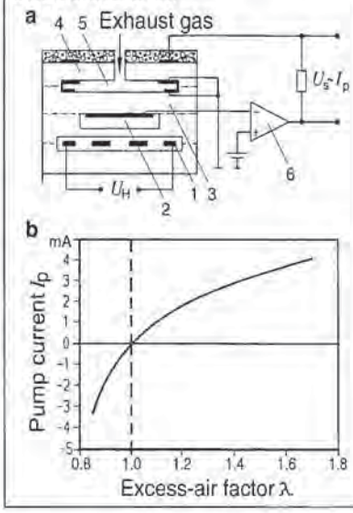
Broadband sensor

This is a combination of the lean sensor using the limit-current principle and the zirconium-dioxide sensor (Nernst oxygen concentration cell). As a two-cell sensor, and in combination with closed-loop control electronics, this sensor generates a clear signal which rises linearly within a broad lambda range ($0.7 < \lambda < 4$).

The pump cell and the concentration cell are made of ZrO_2, and each is coated with 2 porous platinum electrodes. The cells are so arranged that there is a measuring gap of 10...50 µm between them. This measuring gap is connected to the surrounding exhaust-gas atmosphere through an exhaust-gas opening in the solid electrolyte, and at the same time represents the diffusion barrier which determines the limit current. An electronic circuit controls the voltage applied to the pump cell so that the composition of the exhaust gas in the measuring gap remains constant at $\lambda = 1$. This corresponds to a voltage at the concentration cell of U_N = 450 mV. With lean exhaust gas, the pump cell pumps the oxygen from the measuring gap to the outside. On the other hand, when the exhaust gas is rich, the oxygen is pumped into the measuring gap from the surrounding exhaust gas (by decomposition of CO_2 and H_2O) and the direction of current flow is reversed. The pump current is proportional to the oxygen concentration or oxygen requirement. An integrated heater maintains an operating temperature of at least 600 °C.

Broadband sensor

a) Schematic design, b) Pump current as function of the excess-air factor λ. I_p Pump current, U_H Heater voltage, U_S Sensor voltage. 1 Heater, 2 Air reference, 3 Nernst cell, 4 Pump cell, 5 Diffusion barrier, 6 Control electronics.



Types of control

Two-step control

The zirconium-dioxide sensor as described above with its voltage-jump characteristic at $\lambda = 1$ is suitable for two-step controls. A manipulated variable composed of the voltage jump and the ramp changes its direction of control for each voltage jump which indicates a change from rich/lean or lean/rich. The typical amplitude of this manipulated variable has been set to the 2...3 % range. This results in a limited controller response which for the most part is determined by the sum of the dead times.

This sensor's typical "false measurement", caused by variations in exhaust-gas composition, can be compensated for by selective control. Here, the manipulated-variable curve is designed to incorporate an intended asymmetry, whereby the retention of the ramp value for a controlled dwell time t_v following the sensor's voltage jump is the preferred method.

Two-step control with reference sensor downstream of the catalytic converter

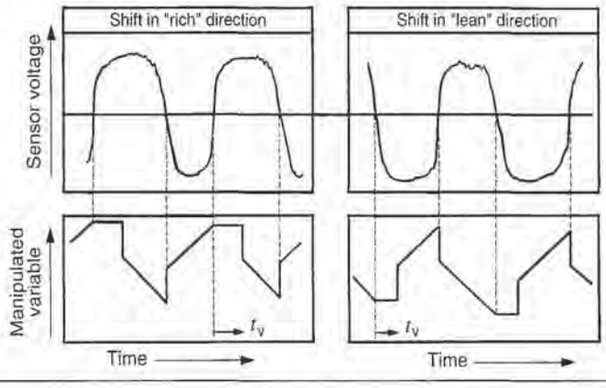
The effect of the disturbance on the accuracy at the voltage-jump point at $\lambda = 1$, as described above, has been minimized by a modified surface coating. Nevertheless, aging and environmental influences (poisoning) still have an effect. A sensor downstream of the catalytic converter is subjected to these influences to a considerably lesser extent. The principle of the two-step control is based upon the fact that the controlled rich or lean shift is changed additively by means of a "slow" corrective control loop.

The resulting long-term stability is of decisive importance for compliance with the exhaust-gas legislation now coming into force.

Continuous-action control with broadband sensor

The defined dynamic response of a two-step control can only be improved when the deviation from $\lambda = 1$ is actually measured. With the broadband sensor, it is possible to achieve continuous-action $\lambda = 1$ control with a stationary, very low amplitude together with high dynamic response. This control's parameters are calculated and adapted as a function of the engine's operating points. Above all, with this type of Lambda control, the unavoidable offset of the stationary and instationary pilot control can be compensated for far more quickly. If demanded by certain engine operating conditions (for instance, warm-up), further optimisation of the exhaust-gas emission applies the potential inherent in the control setpoints $\lambda \neq 1$, for instance, in the lean range. The expansion to a lean-mixture control system, with the advantages and disadvantages already dealt with, merely necessitates the definition of other control setpoints.

Manipulated-variable curve with closed-loop-controlled Lambda shift (two-step control)
t_v dwell time following voltage jump.



Testing exhaust and evaporative emissions

Test program

In order to precisely determine a passenger car's emission levels, the vehicle must be tested in an emissions test cell under conditions which accurately simulate actual driving conditions. Compared to on-road driving, operation in the test cell offers the advantage of allowing tests to be conducted at precisely predefined speeds without having to take traffic flow into consideration. This is the only means of ensuring that individual emissions tests remain mutually comparable.

The vehicle to be tested is parked with its drive wheels on rollers whose rotational resistance can be adjusted to simulate friction and aerodynamic drag. Inertial masses can be added or removed to simulate the weight of the vehicle. A blower mounted a short distance in front of the vehicle provides the necessary cooling.

The measurement of emission levels is based on a simulated driving pattern which progresses through a precisely defined driving cycle incorporating various vehicle speeds. The exhaust gases produced during this driving cycle are collected for analysis of the pollutant masses at the end of the cycle (see illustration).

The methods for collecting the exhaust gases, and the procedures for determining emissions, have largely been standardized in the various countries, but the driving cycles have not. In some countries, regulations governing exhaust emissions are supplemented by limits placed on evaporative losses from the fuel system.

Chassis dynamometer

In order to compare vehicle emissions, the speeds and forces acting on the vehicle during simulation on the chassis dynamometer and on the road must agree with respect to time. It is necessary to simulate the vehicle's moments of inertia, its rolling resistance, and its aerodynamic drag. For this purpose, eddy current brakes or DC motors produce a suitable speed-dependent retarding force. This force acts on the rollers, and must be overcome by the vehicle. Rapid couplings of various sizes are used to attach inertial masses to the rollers, thus simulating vehicle weight. It is imperative that the curve of braking load versus vehicle speed, and the required inertial masses, be maintained precisely. Deviations result in measuring errors. Ambient conditions such as atmospheric humidity, temperature and barometric pressure also influence test results.

Driving cycles

To obtain comparable exhaust-emission values, the predefined speeds on the dynamometer must agree with those on the road. Testing is based on a standardized driving cycle in which gearshifts, braking, idle phases, and standstill periods, have all been selected to provide a high level of correspondence with the velocities and acceleration that characterise typical driving in the normal traffic of a large town. Seven different test cycles are employed internationally. In Europe, the driving schedule has been abbreviated to EU Stage III (as of 01/2000; the 40 s pre-run is no longer used). The USA additionally uses the SFTP test for vehicles with air conditioning, as well as further sets of driving conditions.

Usually, a driver sits in the vehicle to maintain the speed at the levels indicated on a display screen.

Sampling and dilution procedures (CVS Method)

The European adoption of the constant-volume sampling method (CVS) in 1982 means that there is now basically a single standardized procedure in force worldwide for collecting exhaust gases.

Sampling and analysis of emissions

The dilution procedure operates according to the following principle:

The exhaust gases produced by the test vehicle are diluted with fresh air at a mean ratio of 1:10, and extracted using a special system of pumps such that the flow volume composed of exhaust gas and fresh air is maintained at a fixed ratio, i.e. the admixture of air is adjusted according to the vehicle's momentary ex-

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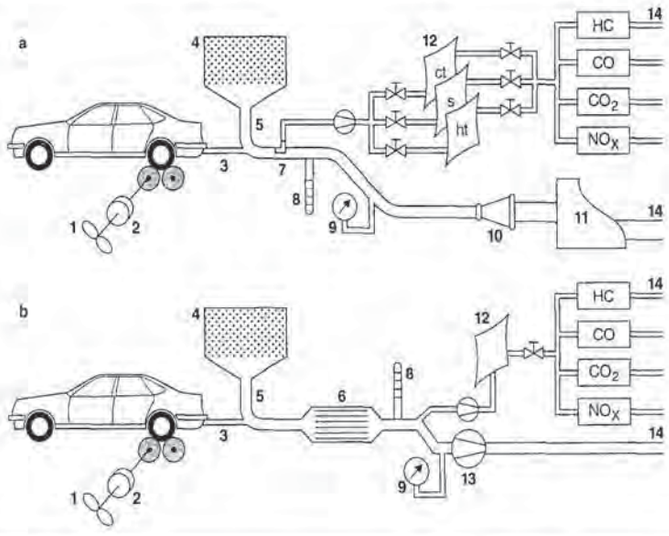
haust volume. Throughout the test a constant proportion of the diluted exhaust gas is extracted for collection in one or several sample bags. The pollutant concentration in the sample bags at the end of the test corresponds precisely to the mean concentration in the total quantity of fresh-air/exhaust mixture which has been extracted. Because the total volume of the fresh-air/exhaust mixture can be defined, the pollutant concentrations can be used as the basis for calculating the pollutant masses produced during the course of the test.

Advantages of this procedure: Condensation of the water vapor contained in the exhaust gas is avoided, which would otherwise cause a substantial reduction in the NO_X losses in the bag. In addition, dilution greatly inhibits the tendency of the exhaust components (especially hydrocarbons) to continue to react with one another. However, dilution does mean that the concentration of the pollutants decreases proportionally to the mean dilution ratio, necessitating the use of more sensitive analyzers. Standardized equipment is available for analysis of the pollutants in the bags.

Dilution systems

Either of two different but equally acceptable pump arrangements is generally used to maintain a constant flow volume for the test. In the first, a normal blower extracts the fresh-air/exhaust mixture through a venturi tube; the second arrangement makes use of a special rotary-piston blower (Roots blower). Either method is capable of measuring the flow volume with an acceptable degree of accuracy.

Test setups
a) For the US federal test (here with a Venturi system),
b) For the Europe test (here with rotary-piston blower).
1 Chassis dynamometer, 2 Inertial mass, 3 Exhaust gas, 4 Air filter, 5 Dilution air, 6 Radiator, 7 Sampling venturi tube, 8 Gas temperature, 9 Pressure, 10 Venturi tube, 11 Blower, 12 Sample bags, 13 Rotary-piston blower, 14 System outlet.
ct Emissions in transition phase, s Emissions in stabilized phase, ht Emissions during hot test



Determining evaporative losses from the fuel system

(Evaporation tests)

Apart from the combustion pollutants produced in the engine, a motor vehicle emits additional quantities of hydrocarbons (HC) through evaporation of fuel from the fuel tank and fuel system (dependent on design and fuel temperature). Some countries (e.g. the USA and Europe) have regulations which limit these evaporative losses.

SHED Test

The SHED (Sealed Housing for Evaporative Determination) test is the most common procedure for determining evaporative emissions. It comprises two test phases with varying conditioning processes in a gas-tight chamber.

The first part of the test is carried out with the fuel tank approx. 40% full. The actual testing of the HC concentration in the chamber begins at 15.5 °C as the test fuel is warmed (initial temperature: 10...14.5 °C). The test concludes after one hour when the temperature of the fuel has risen by 14 °C; at this time the HC concentration is measured once again. Evaporative emissions are determined by comparing the initial and final measurements. The vehicle's windows and trunk lid must remain open during the test.

For determining the evaporative emissions in the second portion of the test, the vehicle is first warmed up by being run through the test cycle valid for the country concerned. The engine is then turned off with the vehicle in the chamber. The increase in the HC concentration is measured for a period of one hour as the vehicle cools.

The sum of both measurements must lie below the current limit value of 2 g of evaporated hydrocarbons. Meanwhile, a more stringent SHED test has been decided upon for implementation in the USA.

ECE/EC test cycle and limits

The ECE/EC test cycle incorporates a synthetically generated driving curve (see illustration) calculated to provide a reasonable approximation of driver behaviour in urban traffic. Since 1993, the test cycle has been supplemented by a suburban section with speeds of up to 120 km/h. This new ECE/EC test cycle is presently required by law in the following countries: Germany, Netherlands, Belgium, Luxembourg, France, Denmark, Great Britain, Ireland, Italy, Spain, Finland, Austria, Sweden, Greece and Portugal.

The exhaust emissions test is conducted as follows:

After the vehicle has been appropriately conditioned (vehicle allowed to stand with engine off at a room temperature of 20...30 °C for at least 6 hours) the actual test begins after a cold start and 40 second pre-running phase (this phase does not apply as of EU Stage III). During measurement, exhaust gas is collected in a sample bag using the CVS method. The pollutant masses determined by analysis of the contents of the bag are converted in the Europe test to the driven distance. In addition, hydrocarbons and oxides of nitrogen are combined to form a composite limit ($HC+NO_X$). These substances are considered separately as of EU Stage III.

More stringent limits have been in force since 1992 irrespective of the vehicle engine displacement. This directive designated 91/441 EEC (EU Stage I) is listed in the Section "Exhaust emission limits",

ECE/EC test cycle with suburban section
1 Pre-run (no measurement): to date 40 s; does not apply as of EU Stage III.
Cycle distance: 11 km
Average speed: 32.5 km/h
Maximum speed: 120.0 km/h

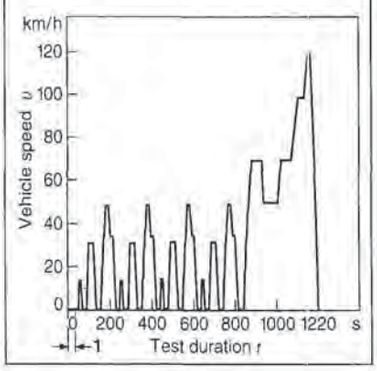


Table 1. It also places limits on evaporative losses. A further reduction of the limits was imposed in 1996/97 by directive 94/12/EEC (EU Stage II).

Further reductions of limits are planned for Europe (Stage III and IV, 2000 and 2005):
- Cold start at –7 °C (as of 2002),
- EOBD (European On-Board Diagnosis) of parts relevant to exhaust emissions,
- More stringent evaporative-emissions test,
- Durability (80,000; 100,000 km) and in-field monitoring,
- Exhaust-gas sampling immediately after start

Test cycles in the USA

FTP 75 test cycle

The FTP 75 test cycle (Federal Test Procedure), which consists of three test sections, represents actual speeds measured in the USA on the streets of Los Angeles in morning commuter traffic (see illustration 3a):

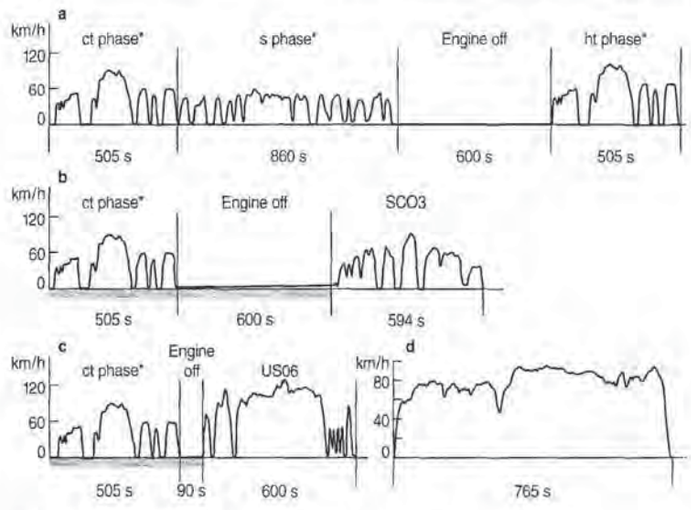
The vehicle to be tested is first conditioned (allowed to stand with engine off for 12 hours at a room temperature of 20...30 °C), then started and run through the prescribed test cycle:

Phase ct: Collection of diluted exhaust gas in bag 1 during the cold transition phase.

Phase s: Changeover to sample bag 2 at the beginning of the stabilized phase (after 505 s) without any interruption in the

USA test cycles

	a	b	c	d
Test cycle	**FTP 75**	**SC03**	**US06**	**Highway**
Cycle distance:	17.87 km	5.76 km	12.87 km	16.44 km
Cycle duration:	1877 s + 600 s pause	594 s	600 s	765 s
Average cycle speed:	34.1 km/h	34.9 km/h	77.3 km/h	77.4 km/h
Maximum cycle speed:	91.2 km/h	88.2 km/h	129.2 km/h	96.4 km/h



** ct Transition phase; s Stabilized phase; ht Hot test*
Phases during which exhaust gas is collected
Conditioning (other driving schedules are also possible)

driving cycle. The engine is turned off for a 10-minute pause immediately following the stabilized phase (after 1372 s).
Phase ht: The engine is restarted for the hot test (505 s in duration). The speeds used in this phase correspond directly to those in the cold transition phase. Exhaust gases are collected in a third sample bag. The bag samples from the previous phases are analyzed during the pause before the hot test, as the samples should not remain in the bags for longer than 20 minutes.

The exhaust-gas sample in the third sample bag is analyzed once the driving sequence has been completed. The weighted sums of the pollutant masses (HC, CO and NO_x; ct 0.43, s 1, ht 0.57) from all three bags are correlated with the distance covered during the test, and then expressed as emissions per mile. The limits on pollutant emissions vary among individual countries. This test procedure is used in the USA including California ("Emission limits", Table 2) and in other countries (Table 4).

SFTP schedules
The tests in accordance with the SFTP standard will be phased-in in stages between 2001 and 2004. Together they consist of three driving schedules: the FTP 75, the SC03 and the US06 schedule.

They are intended to additionally examine the following driving patterns (see illustration, USA test cycles b and c):
- aggressive driving,
- major changes in driving speed,
- engine start and driving-off,
- frequent minor changes in driving speed,
- parking times and
- operation with the air conditioner running.

In the SC03 and US06 schedules, the ct phase of the FTP 75 test cycle is applied after preconditioning with no collection of exhaust gases, however other conditioning procedures are also possible.

The SC03 schedule is carried out at a temperature of 30 °C and 40 % relative humidity (vehicles with air conditioning only). The individual driving schedules are weighted as follows:

- Vehicles with air conditioner:
  35 % FTP 75 + 37 % SC03 + 28 % US06
- Vehicles without air conditioner:
  72 % FTP 75 + 28 % US06

The SFTP and FTP 75 test cycles must be successfully completed independently of each other (Tables 2 and 3 in the section "Exhaust emission limits").

Test cycles for determining fleet fuel consumption
Every vehicle manufacturer must determine the fuel consumption of his vehicle fleet. If a manufacturer exceeds certain limits, he is fined. He is awarded a bonus if consumption lies below certain limits. Fuel consumption is determined from the exhaust emissions produced during two test cycles: the FTP 75 test cycle (55 %) and the highway test cycle (45 %).

The highway test cycle is conducted once after preconditioning (vehicle allowed to stand with engine off for 12 hours at 20...30 °C) without measurements being conducted. The exhaust emissions from a second test run are then collected. Fuel consumption is determined based on the emissions (test cycle d).

Every new vehicle is required to comply with the limits for 50,000 miles, regardless of vehicle weight and engine displacement. Under certain conditions, the USA grants exemptions for the various model years. The legislation imposes different limits at 50,000 and 100,000 miles; permissible limits are higher at 100,000 miles (deterioration factors).

In addition to many measures designed to protect the environment, the Clean Air Act stipulated more stringent emission limits for vehicles since 1994 (Table 2). California decided to enforce the more stringent limits as early as 1993, and is planning to take further drastic steps.

Cold-start enrichment, which is necessary when a vehicle is started at low temperatures, produces particularly high emissions. These cannot be measured in current emissions testing, which is conducted at ambient temperatures of 20...30 °C. The Clean Air Act attempts to limit these emissions as well by prescribing an emissions test at −6.7 °C.

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However, a limit is prescribed for carbon monoxide only.

Japanese test cycle

Two test cycles with differing hypothetical driving curves are combined to provide the complete test.

Following a cold start, the 11-mode cycle is run four times, with evaluation of all four cycles. The 10·15-mode test as a hot test is run through once (see illustration).

Preconditioning for the hot start also includes the stipulated idle emissions test, and is conducted as follows:

After warming up the vehicle at 60 km/h for approx. 15 minutes, the HC, CO and CO_2 concentrations in the exhaust pipe are measured. Following a secondary warm-up period of 5 minutes at 60 km/h, the 10·15-mode hot test is then started. In the 11-mode test as well as in the 10·15-mode test, analysis is performed using a CVS system. The diluted exhaust gas is in each case collected in a bag. In the cold test, the pollutants are specified in terms of g/test, whereas in the hot test they are correlated with the distance driven, i.e. they are converted to g/km (see Table 5 in the section "Exhaust emission limits").

The exhaust gas regulations in Japan include limits on evaporative emissions which are measured using the SHED method.

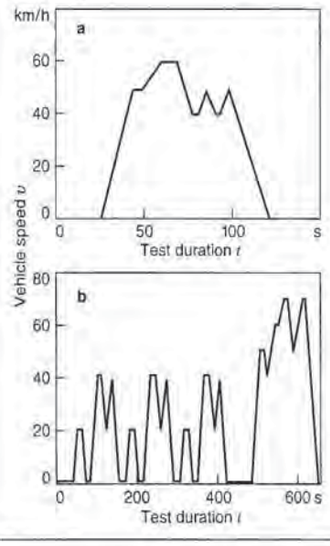
Japanese test cycles

a 11-mode cycle (cold test)

Cycle distance:	*1.021 km*
Cycles per test:	*4*
Average speed:	*30.6 km/h*
Maximum speed:	*60 km/h*

b 10·15-mode cycle (hot test)

Cycle distance:	*4.16 km*
Cycles per test:	*1*
Average speed:	*22.7 km/h*
Maximum speed:	*70 km/h*



Exhaust-gas analyzers

Legislation requires that emission testing also be carried out on vehicles which are already on the road.

In addition, exhaust-gas analyzers are also indispensable service tools in the workshop, necessary both for optimal A/F-mixture adjustment and for effective trouble-shooting on the engine.

Test procedure

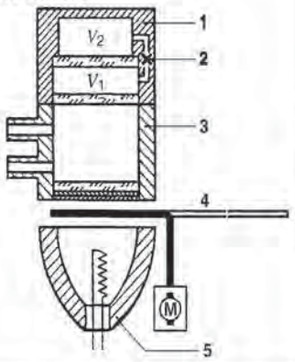
In automotive service operations, the infrared method has proved to be the only efficient and reliable means for testing exhaust gases.

This method bases on the fact that individual exhaust-gas components absorb infrared light at different specific rates, according to their characteristic wavelengths. The various designs include both single-component analyzers (e.g. for CO only) and multi-component analyzers (for CO/HC, CO/CO_2, CO/HC/CO_2 etc.).

Test chamber

Infrared radiation is transmitted by an emitter which has been heated to approx. 700 °C. The beam passes through a measuring cell before entering the receiver chamber. When measuring the CO content, the sealed receiver chamber contains gas with a defined CO content which absorbs a portion of the CO-specific radiation. This absorption causes the gas temperature to increase, which in turn causes gas with a volume of V_1 to flow via a flow sensor into the compensation volume with a volume of V_2. A rotating chopper disk induces a rhythmic interruption in the beam, producing an alternating basic flow between the two volumes V_1 and V_2. The flow sensor converts this motion into an alternating electrical signal. When a test gas with a variable CO content flows through the measuring cell, it absorbs radiant energy in a quantity proportional to its CO content; the energy is then no longer available in the receiver chamber. The result is a reduction of the base flow in the receiver chamber. The deviation from the alternating basic signal is therefore a measure of the CO content in the test gas.

Test chamber using infrared method (schematic diagram)
1 Receiver chamber with compensation volume V_1 and V_2, 2 Flow sensor, 3 Measuring cell, 4 Rotating chopper disk with motor, 5 Infrared emitter.



Catalytic-converter testing

A representative component can be used to obtain an indirect measurement of catalytic-converter operation on vehicles with closed-loop-controlled catalytic devices. The best-suited is CO, which must not exceed 0.3 % by volume downstream of the converter, whereby lambda must be exactly 1.00 (±0.01). Lambda, in turn, can be determined using the composition of the exhaust gas at the catalytic converter's outlet. The exhaust-gas analyzer calculates the value for lambda using the CO, HC, CO_2 and O_2 exhaust gas components, with constants used for NO and fuel composition HC_V. The O_2 content is measured by an electrochemical probe.

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Current (1998) emissions limits for gasoline engines

Table 1
EU emissions limits as measured in ECE/EU test cycle

Standards	Introduction	CO g/km	HC g/km	NO_x g/km	HC+NO_x g/km
EU Stage I	07.92	2.72	–	–	0.97
EU Stage II	01.96	2.2	–	–	0.5
EU Stage III	01.00	2.3	0.2	0.15	–
EU Stage IV	01.05	1.0	0.1	0.08	–

Table 2
US Federal (49 states) and California emissions limits. FTP 75 test cycle

	Model year	Standards	CO g/mile	HC g/mile	NO_x g/mile
US Federal	1994	Level 1	3.4	0.25	0.4
	2004 [1]	Level 2	1.7	0.125 [2]	0.2
California	[3]	TLEV [4]	3.4	0.125 [2]	0.4
	[3]	LEV [5]	3.4	0.075 [2]	0.2
	[3]	ULEV [6]	1.7	0.04 [2]	0.2

[1] Proposal. [2] NMOG = Non Methanic Organic Gases. [3] Introduction varies according to manufacturer's NMOG fleet average (both vehicle and total fleet are certified).
[4] Transitional Low Emission Vehicles. [5] Low Emission Vehicles. [6] Ultra Low Emission Vehicles.

Table 3
US emissions limits. STFP test cycle

	NMHC[1]+NO_x g/mile	$CO_{Composite}$ [2] g/mile	CO_{SC03} [2] g/mile	CO_{US06} [2] g/mile
up to 50,000 miles	0.65	3.4	3.0	9.0
50,000 to 100,000 miles	0.91	4.2	3.7	11.1

[1] Non Methane HC. [2] The manufacturer has the option of selecting $CO_{Composite}$ or CO_{SC03} and CO_{US06} limits.

Table 4
Emissions limits for Argentina, Australia, Brazil, Canada, Mexico, Norway, Switzerland, and South Korea measured with FTP 75 test cycle

Country	Introduction	CO g/km	HC g/km	NO_x g/km	Evap. emissions (HC) g/test
Argentina	01.97	2.0	0.3	0.6	6.0
Australia	01.97	1.9	0.24	0.57	1.9
Brazil	01.97	2.0	0.3	0.6	6.0
Canada	01.98	2.1	THC [2] 0.25; NMHC [3] 0.16	0.24	2.0
Mexico	01.95	2.1	0.25	0.62	2.0
Norway	01.89	2.1	0.25	0.62	2.0
Switzerland [1]	10.87	2.1	0.25	0.62	2.0
South Korea	01.91 01.00	2.1	0.25 0.16	0.62 0.25	2.0

[1] EU/ECE regulations recognized since 10/95. [2] THC = Total HC. [3] NMHC = Non Methane HC.

Table 5
Japanese emissions limits measured in Japanese test cycle

Test procedure	CO	HC	NO_x	Evap. emissions
10-15-mode (g/km)	2.1...2.7 (0.67)	0.25,..0.39 (0.08)	0.25...0.48 (0.08)	–
11-mode (g/test)	60.0...85.0 (19.0)	7.0...9.5 (2.2)	4.4...6.0 (1.4)	–
SHED (g/Test)	–	–	–	2.0

( ) planned figures

Diesel-engine management

Fuel metering

Requirements

To ensure efficient preparation of the A/F mixture, the fuel must not only be injected into the combustion chamber at a pressure of between 350 and 2000 bar, depending on the diesel combustion process, but it must also be metered with the greatest possible accuracy during each injection. In order to maintain the compromise between low fuel consumption and compliance with emission limits (exhaust gas and noise), it is necessary to precisely time the start of injection with an accuracy of approx. ±1° crankshaft.

The start of injection is adjusted in order to control the start of combustion and compensate for the pressure-wave propagation times in the pressure lines.

In mechanical control systems, an upstream timing device uses the upper helix on the injection-pump plunger to adjust the start of injection as a function of engine speed and load.

EDC systems (Electronic Diesel Control) have an integral timing function that adjusts the injection pump's start of delivery as a function of engine speed, load and engine temperature.

The injected fuel quantity is used for load- and speed-dependent control of diesel engines; the quantity of intake air is not throttled. Because the speed of an unloaded diesel engine could therefore increase to the point of self-destruction given sufficient injected fuel, an engine-speed governor is essential. Special control devices are also required to ensure stable idle behavior.

Fuel-injection process

The fuel can no longer be regarded as incompressible, considering the high pressures and short delivery times involved. Thus the injection processes are not static (i.e. they do not conform to the geometric laws of displacement), but rather dynamic (largely in accordance with the laws of acoustics).

In the case of cam-driven systems, a camshaft driven by the engine moves the injection-pump plunger in the direction of delivery. This causes pressure to build up in the high-pressure chamber of the pump element. The increasing pressure opens the delivery valve, and a pressure wave moves through the high-pressure delivery line to the injection nozzle at the speed of sound (approx. 1400 m/s). When the opening pressure of the nozzle is reached, the needle valve overcomes the force of the nozzle spring and lifts from its seat so that fuel can be injected through the spray orifices into the engine's combustion chamber. Fuel delivery ends when the pressure in the high-pressure chamber collapses due to the spill port being opened by the pump-plunger helix. The delivery valve then closes and reduces the pressure in the injection line. The pressure drop to static pressure between every two injection cycles is dimensioned such that

- the injection nozzle closes quickly to provide a clean break in fuel discharge and to prevent fuel dribble,
- residual pressure oscillations in the lines are effectively dampened. On the one hand to prevent the pressure peaks from reopening the nozzle, and on the other hand to ensure that the pressure drops do not cause cavitation damage.

Fuel-injection system

The fuel-injection system supplies fuel to the diesel engine, and consists of a low-pressure stage and a high-pressure stage.

The low-pressure stage consists of the fuel tank, fuel filter, mechanical or electric delivery pump with pressure relief valve, and the fuel lines.

The high-pressure stage comprises the high-pressure pump with delivery valve which generates the fuel pressure required for injection, the fuel-injection tubing, and the fuel injectors (nozzle-holder assemblies and nozzles). EDC (Electronic Diesel Control) systems also incorporate a control valve, which in today's systems generally takes the form of a solenoid valve. In the future, however, piezo-

valves will also be used.

Generally speaking, current state-of-the-art technology prescribes one of the following high-pressure injection systems for automotive diesels:

In-line fuel-injection pumps with mechanical governors or electronic actuators, and integral timing devices as dictated by the specific application.

The in-line fuel-injection pump is used primarily on commercial-vehicle engines. In this pump system, a number of pumping elements (also known as plunger-and-barrel assemblies) corresponding to the number of engine cylinders are accommodated in a housing and operated by means of a common camshaft. The range of in-line fuel-injection pumps also includes the control-sleeve pump, which allows the start of delivery in addition to the injected fuel quantity to be adjusted as desired.

The distributor injection pump incorporates a mechanical governor or electronic controller, and an integrated timing device.

The axial-piston pump is used in particular on high speed IDI and DI diesel engines for passenger cars and small commercial vehicles. In this type of pump, a central plunger driven by a cam plate generates pressure and distributes the fuel to the individual cylinders. A control sleeve or solenoid valve meters the injected fuel quantity.

The radial-piston pump is primarily used on modern high speed DI diesel engines for passenger cars and small commercial vehicles. Pressure is generated and fuel delivered by two to four plungers in a radial configuration driven by a cam ring. A solenoid valve controls the injection timing and meters the fuel.

In addition to the in-line and distributor injection pumps, there are also individual injection pumps which are driven directly by the engine camshaft. These are mostly used on large marine engines, construction machinery and small engines.

Another modern fuel-injection system is the UIS (Unit Injector System), in which the injection pump and fuel injector form one unit. One unit injector per cylinder is installed in the cylinder head. The unit is driven by the engine's camshaft, either directly via a push rod or indirectly via a rocker-arm assembly.

The UPS (Unit Pump System) operates according to the same principle as the Unit Injector System. However, in this type of pump a short high-pressure delivery line is connected between the pump and nozzle. This permits more design latitude, as the camshaft can be located either in the engine block or cylinder head.

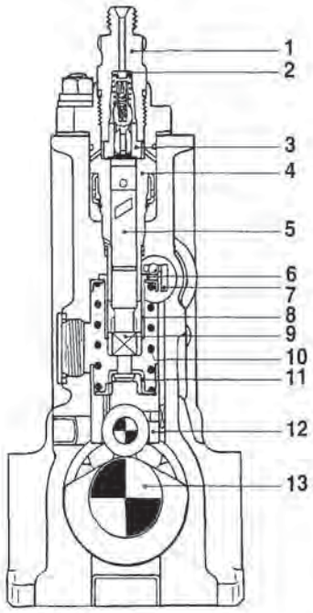
Common to all of these systems is the fact that the required injection pressure is generated at the moment each injection occurs. In the case of electronically controlled systems, however, the rate-of-discharge curve can be influenced not only by the cam contour but also by solenoid valve actuation. Nevertheless, the maximum possible pressure depends directly on engine speed and injected fuel quantity.

The injection pressure can be set independently of engine speed and load using an accumulator-type injection system. In this so-called Common Rail System (CRS), pressure generation and fuel injection are decoupled with respect to time and location. Injection pressure is generated by a separate high-pressure pump. This pump need not necessarily be driven synchronously with respect to the injection timing. The pressure can be set independently of engine speed and injected fuel quantity, and electrically operated injectors are used in place of pressure-controlled fuel injectors. The time and duration of injector actuation determine the start of injection and the injected fuel quantity. This system also offers a great deal of freedom with regard to the design of multiple or divided injection functions.

In order to achieve accurate open and closed-loop control of injected fuel quantity and start of injection, as well as minimum scatter between the individual cylinders and long service life, it is imperative that all injection systems are manufactured from high-precision, close-tolerance components.

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Size P in-line fuel-injection pump
1 Delivery-valve holder, 2 Spring seat, 3 Delivery valve, 4 Pump barrel, 5 Pump plunger, 6 Lever arm with ball head, 7 Control rack, 8 Control sleeve, 9 Plunger control arm, 10 Plunger return spring, 11 Spring seat, 12 Roller tappet, 13 Camshaft.



In-line fuel-injection pump (PE)

Fuel-supply pump

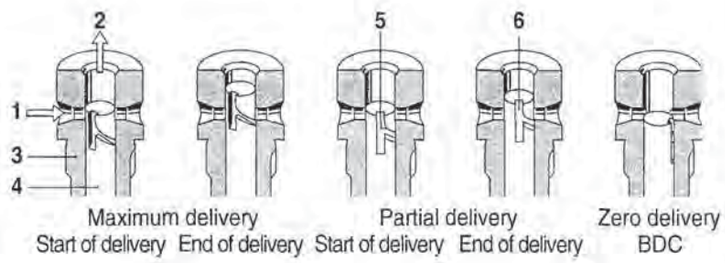
A piston pump delivers the fuel to the injection pump's fuel gallery at a pressure of 1...2.5 bar. The cam-driven supply-pump plunger travels to TDC on every stroke. It is not rigidly connected to the drive element; instead, a spring supplies the return pressure. The plunger spring responds to increases in line pressure by reducing the plunger's return trave to a portion of the full stroke. The greater the pressure in the delivery line, the lower the delivery quantity.

High-pressure pump

Every in-line fuel-injection pump has a plunger-and-barrel assembly (pumping element) for each engine cylinder. An engine-driven camshaft moves the plunger in the supply direction, and a spring presses it back to its initial position. Although the plunger has no seal, it is fitted with such precision (clearance: 3...5 µm) that its operation is virtually leak-free, even at high pressures and low engine speeds.

The plunger's actual stroke is constant. The delivery quantity is changed by altering the plunger's effective stroke. Inclined helices have been machined into the plunger for this purpose, so that the plunger's effective stroke changes when it is rotated. Active pumping starts when the upper edge of the plunger closes the in-

Fuel-delivery control in the in-line fuel-injection pump
1 From fuel gallery, 2 To nozzle, 3 Barrel, 4 Plunger, 5 Lower helix, 6 Vertical (stop) groove.



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take port. The high-pressure chamber above the plunger is connected by a vertical groove to the chamber below the helix. Delivery ceases when the helix uncovers the intake port.

Various helix designs are employed in the plunger. On plunger-and-barrel assemblies with a lower helix only, pumping always begins at the same stroke travel, the plunger being rotated to advance or retard the end of delivery. An upper helix can be employed to vary the start of delivery. There are also plunger-and-barrel assemblies on the market which combine upper and lower helices in a single unit.

In order of their suitability for use with high injection pressures, the major types of delivery valve currently in use are:
- Constant-volume valve,
- Constant-volume valve with return-flow restriction,
- Constant-pressure valve.

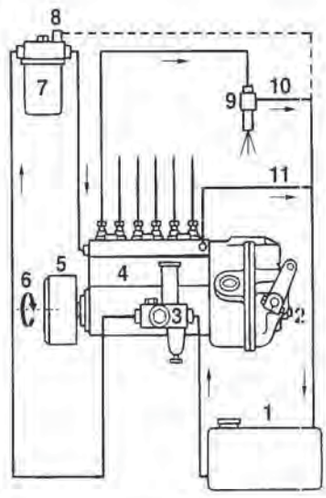
The delivery valve and pressure-relief characteristics must be specially designed for the specific application. Units incorporating a return-flow restriction or constant-pressure valve have an additional throttle element to damp the pressure waves reflected back from the injection nozzle, thus preventing it from opening again. The constant-pressure valve is employed to maintain stable hydraulic characteristics in high-pressure fuel-injection systems and on small, high-speed direct-injection engines.

In fuel-injection pumps which generate moderate pressures of up to 600 bar (e.g. Size A), the plunger-and-barrel assembly is installed in the pump housing in a fixed position, where it is held in place by the delivery valve and the delivery-valve holder.

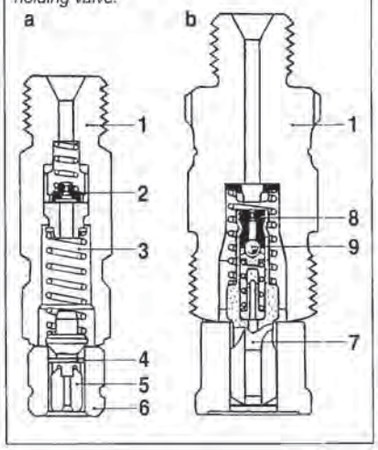
In pumps which generate injection pressures greater than approx. 600 bar, the plunger-and-barrel assembly, delivery valve and delivery-valve holder are screwed together to form a single unit, which means that the high sealing forces must no longer be accommodated by the pump housing (e.g. Sizes MW, P).

The in-line fuel-injection pump and the attached governor are connected to the engine's lube-oil system.

In-line fuel-injection pump with mechanical (flyweight governor)
1 Fuel tank, 2 Governor, 3 Fuel-supply pump, 4 Injection pump, 5 Timing device, 6 Drive from engine, 7 Fuel filter, 8 Vent, 9 Nozzle-and-holder assembly, 10 Fuel return line, 11 Overflow line.

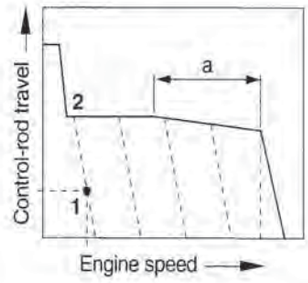


Delivery-valve holder with delivery valve
a) With constant-volume valve and return-flow restriction, b) With constant-pressure valve. 1 Delivery-valve holder, 2 Return-flow restriction, 3 Dead volume, 4 Retraction piston, 5 Valve ball, 6 Valve holder, 7 Supply valve, 8 Calibrated restriction, 9 Pressure-holding valve.

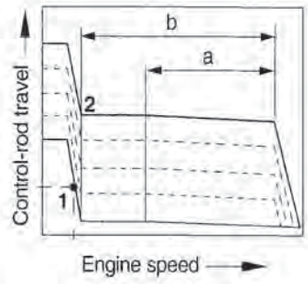


Governor characteristic curves
a Positive torque control in upper speed range, b Unregulated range, c Negative torque control.
1 Idle-speed setpoint, 2 Full-load curve, 3 Full-load curve, turbocharged engine, 4 Full-load curve, naturally-aspirated engine, 5 Full-load curve, naturally-aspirated engine with altitude compensation, 6 Intermediate engine-speed control, 7 Temperature-sensitive starting quantity.

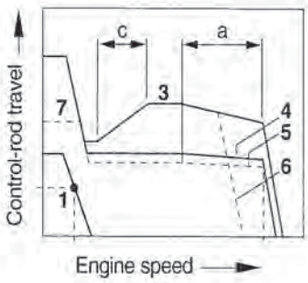
Variable-speed governor



Minimum-maximum-speed governor



Complex governor with additional control functions



Speed governing

The main function of the governor is to limit the maximum engine speed. In other words, it must ensure that the diesel engine does not exceed the maximum min^{-1} specified by its manufacturer. Depending upon type, the governor's functions may include maintaining specific, constant engine speeds, such as idle, or other speeds in the range between idle and maximum speed. The governor can also adjust full-load delivery in accordance with engine speed (adaptation), boost or atmospheric pressure, and it can be used to meter the extra fuel required for starting. The governor adapts the delivery quantity to these conditions by making corresponding adjustments in the position of the control rack.

Mechanical (flyweight) governors

The mechanical governor (also known as a flyweight or centrifugal governor) is driven by the engine's camshaft, and provides the performance curves described below. The flyweights, which act against the force of the governor springs, are connected to the control rack by a system of levers. During steady-state operation, centrifugal and spring forces are in a state of equilibrium, and the control rack assumes a position for fuel delivery corresponding to engine power output at that operating point. A drop in engine speed – for instance, due to increased load – results in a corresponding reduction in centrifugal force, and the governor springs move the flyweights, and with them the control rack, in the direction for increased delivery quantity until equilibrium is restored. Various functions are combined to produce the following types of governor:

Variable-speed governors

The variable-speed governor maintains a virtually constant engine speed in accordance with the position of the control lever. Applications: Preferably for commercial vehicles with auxiliary power take-off, for construction machinery, agricultural tractors, in ships and in stationary installations.

Minimum-maximum-speed governors

From the characteristic curve for the minimum-maximum-speed governor it can be seen that this type of governor is effective only at idle and when the engine reaches

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maximum min^{-1}. The torque in the range between these two extremes is determined exclusively by the position of the accelerator pedal. Applications: For road vehicles.

Combination governors

Combination governors are a synthesis of the two governor types described above. Depending upon the specific application, active control can be in the upper or lower engine-speed range.

Governor types

In the RQ and RQV governor, the flyweights act directly on the governor springs, and control-lever movements vary the transfer ratio at the fulcrum lever.

In the RSV and RSF governor, the governor spring is outside the flyweights; the transfer ratio at the fulcrum lever remains essentially constant.

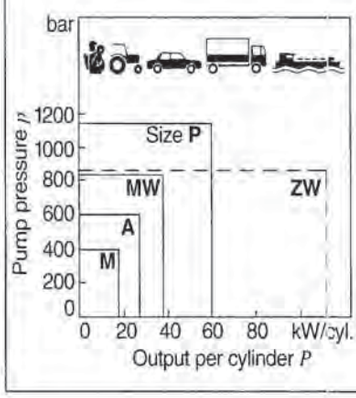
Speed droop

The governor's performance characteristics are essentially a function of the slope of the control curve, defined as speed droop:

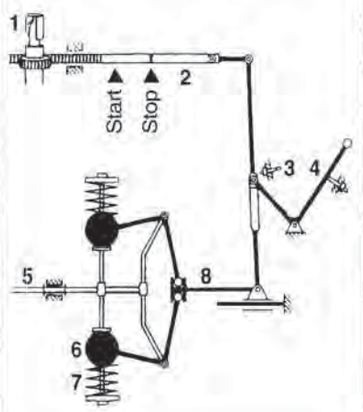
$$\delta = \frac{n_{LO} - n_{VO}}{n_{VO}} \cdot 100\,\%$$

The smaller the difference between the upper no-load speed (n_{LO}) and the upper full-load speed (n_{VO}), the lower the speed droop, i.e. the greater the precision with which the governor maintains a specific engine speed. Variable-speed governors in small high-speed engines generally achieve a full-load speed regulation (top-end breakaway consistency) of 6...10 %.

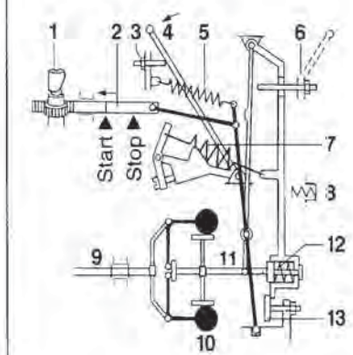
Applications for various types of in-line injection pump



RQ Minimum-maximum-speed governor
1 Pump plunger, 2 Control rack, 3 Full-load stop, 4 Control lever, 5 Injection-pump camshaft, 6 Flyweight, 7 Governor spring, 8 Sliding bolt.



RSV Variable-speed governor
1 Pump plunger, 2 Control rack, 3 Maximum-speed stop, 4 Control lever, 5 Start spring, 6 Stop or idle stop, 7 Governor spring, 8 Auxiliary idle spring, 9 Injection-pump camshaft, 10 Flyweight, 11 Sliding bolt, 12 Torque-control spring, 13 Full-load stop.

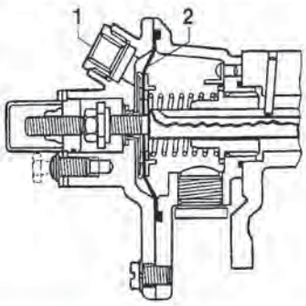


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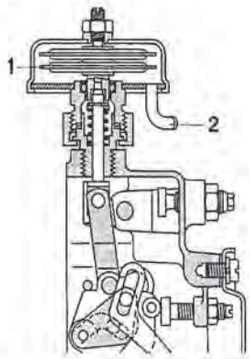
Mechanical add-on equipment

Torque control
An auxiliary spring (torque-control spring) is installed at a suitable position in the governor mechanism. The spring precisely adapts the governor's output curve to the diesel engine's full-load fuel requirements by lowering it slightly. When a given engine speed is reached, the spring compresses and causes the control rack to move in the direction for reduced fuel-delivery quantity (positive torque control). Negative torque control, which responds to increased engine speed by augmenting the fuel-delivery quantity, is also possible, albeit at the price of far more components and more complicated adjustment procedures.

Manifold-pressure compensator (LDA)
1 Boost-pressure connection, 2 Diaphragm.



Altitude-pressure compensator (ADA)
1 Pressure capsule, 2 Atmospheric-pressure connection.



Manifold-pressure compensator (LDA)
Due to the larger air mass, turbocharged engines are capable of converting a greater amount of fuel into engine torque as the boost pressure increases; A spring-loaded diaphragm is used to make a corresponding correction in the full-load fuel-delivery quantity. Increasing boost pressure acts on the working side of the diaphragm, which is connected to the control rack in such a way that the injected fuel quantity increases as boost pressure increases

Altitude-pressure compensator (ADA)
The altitude-pressure compensator is similar to the LDA. It reduces the full-load fuel-delivery in response to the ow atmospheric pressure (and low air density) encountered at high altitudes.

The unit includes a barometric capsule which displaces the control rack in the direction for lower fuel-delivery once atmospheric pressure drops by a specific increment.

Temperature-dependent starting device (TAS)
A cold engine requires a certain amount of additional fuel (enrichment) in order to start. This enrichment is not necessary on a warm engine and could lead to the emission of smoke.

The solution is TAS, which features a control-rack stop employing an expansion element to prevent enrichment during warm starts.

Rack-travel sensor (RWG)
The RWG monitors the control-rack position inductively.

After processing in an evaluation circuit, the signal can be used for such tasks as control of hydraulic or mechanical transmissions, for measuring fuel consumption, for exhaust-gas recirculation, and for diagnostics.

Port-closing sensor (FBG)
The FBG is an inductive unit which, with the engine running, monitors the point at which pump delivery starts (port closing). It can also check the timing device.

In addition, injection pumps equipped with this device can be supplied with the camshaft locked in the port-closing position. This setting facilitates simple and

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precise pump installation on the engine.

Timing devices
Centrifugally-controlled timing devices are positioned in the drivetrain between the engine and the injection pump. The flyweights respond to increasing engine speed by turning the injection pump's camshaft, with respect to the drive shaft, in the "delivery advance" direction. Front-mounted clutch-driven units and gear-driven in-pump devices, with an adjustment range of 3°...10° on the pump shaft are available.

Pump shutoff
A mechanical (stop lever), electric or pneumatic shutoff device is employed to shut down the diesel engine by interrupting the fuel supply.

Electronic governor (EDC)
Instead of flyweights, the electronic governor for the in-line fuel-injection pump uses a solenoid actuator with a non-contacting inductive position sensor to position the control rack. The solenoid actuator is triggered by an ECU.

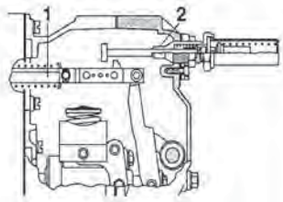
The ECU microprocessor compares accelerator position, min^{-1}, and a number of additional correction factors with the program maps stored in its memory in order to determine the correct injected fuel quantity, i.e. the correct rack position.

An electronic controller compares the monitored control-rack position with the specified setpoint in order to determine the required excitation-current input to the solenoid, which operates against a return spring. When deviations are detected, the excitation current is regulated to shift the control rack to precisely the specified position.

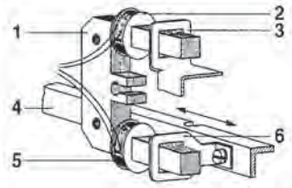
An inductive speed sensor monitors a camshaft-mounted pulse wheel; the ECU uses the pulse intervals to calculate engine speed.

Because it can monitor a number of engine and vehicle parameters and combine them to calculate the injected fuel quantity, an electronic governor has a number of advantages over a mechanical unit:
- Engine can be switched on and off with key,
- Complete freedom in determining full-load response,

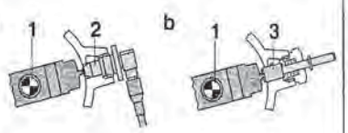
Temperature-dependent starting device
1 Control rack, 2 Start-quantity stop with expansion element.



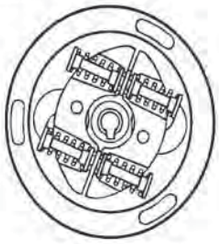
Rack-travel sensor (RWG)
1 Iron core, 2 Reference coil, 3 Fixed short-circuiting ring, 4 Control rack, 5 Measuring coil, 6 Moving short-circuiting ring



Port-closing sensor (FBG)
a) Measurement with sensor,
b) Blocking position;
1 Pump camshaft, 2 Sensor, 3 Blocking pin.



Timing device
Off position.



- Maximum injected fuel quantity can be precisely coordinated with the boost pressure in order to remain below the smoke limit,
- Corrections for air and fuel temperatures,
- Temperature-dependent start quantity,
- Engine-speed control for auxiliary power take-offs,
- Cruise-control facility,
- Regulation of maximum speed,
- Consistent, low idle speed,
- Active surge control,
- Option for intervention in traction control (TCS)/automatic transmission,
- Signal outputs for display of fuel consumption and engine speed,
- Service support through integral error diagnosis.

In-line control-sleeve fuel-injection pump

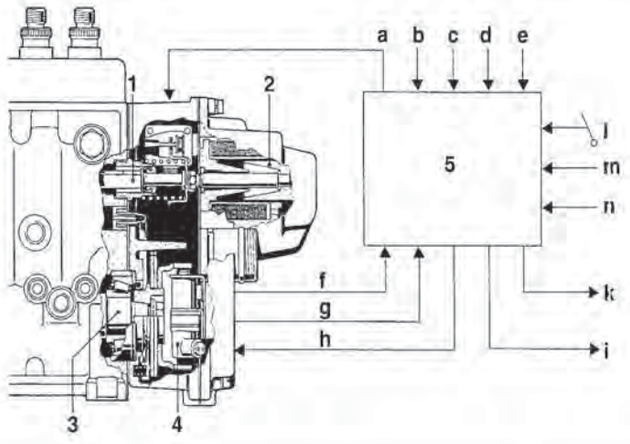
The in-line control-sleeve injection pump makes it possible to provide electronically-controlled adjustment of port closing (start of pump delivery). The spill port, which on conventional in-line pumps is in the housing and therefore immovable, is incorporated in the control sleeve which is a component in each plunger-and-barrel assembly. A control shaft with control-sleeve levers which engage the slide valves changes the positions of all sleeves at the same time.

Depending on the position of the control sleeve (up or down), the start of delivery is advanced or retarded relative to the position of the camshaft lobe. An electromagnetic actuator mechanism similar to that used in the electronically controlled in-line injection pumps turns the control shaft, albeit without position feedback.

A needle-motion sensor monitors the start of injection directly at the injection nozzle. It transmits a corresponding signal to the ECU, which compares it with the programmed value as a function of min^{-1}, injected fuel quantity, etc., in order to adjust the solenoid-excitation current to achieve congruence between the feedback and setpoint values for start of injection.

The engine-speed sensor obtains precise information on injection timing relative to TDC by monitoring pulses from reference marks on the engine's flywheel.

Electronic diesel control (EDC) for in-line fuel-injection pumps
1 Control rack, 2 Actuator, 3 Camshaft, 4 Engine-speed sensor, 5 ECU. Input/output quantities: a Redundant shutoff, b Boost pressure, c Vehicle speed, d Temperature (water, air, fuel), e Fuel-quantity command, f Engine speed, g Control-rack travel, h Solenoid position, i Fuel-consumption and engine-speed display, k Diagnostics, l Accelerator position, m Speed preset, n Clutch, brakes, engine brake.

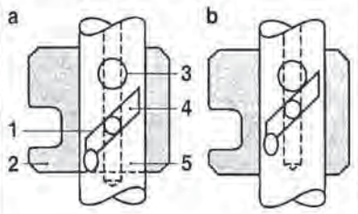


Distributor-type fuel-injection pump (VE)

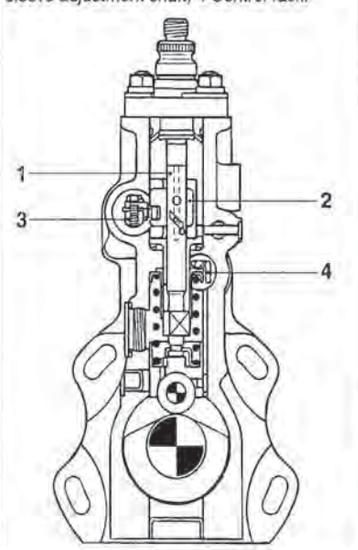
Distributor-type fuel-injection pumps are used in 3-, 4-, 5- and 6-cylinder diesel engines in passenger cars, tractors and light- and medium-duty commercial vehicles which generate up to 42 kW per cylinder, depending on engine speed and combustion system. Distributor-type fuel-injection pumps for direct-injection (DI) engines achieve a peak injection pressure of 1950 bar in the nozzle at speeds of up to 2400 min^{-1}.

A differentiation is made between distributor-type fuel-injection pumps with mechanical control, and those with electronic governing available in versions with a rotating-solenoid actuator and with solenoid valve open-loop control.

Plunger-and-barrel assembly with control sleeve
a) Start of delivery, b) End of delivery.
1 Control helix, 2 Control sleeve, 3 Spill port, 4 Control groove, 5 Pump plunger.



In-line control-sleeve pump
1 Pump plunger, 2 Control sleeve, 3 Control-sleeve adjustment shaft, 4 Control rack.



Mechanically controlled axial-piston distributor pumps (VE)

These mechanically controlled axial-piston distributor pumps comprise the following major assemblies:

Fuel-supply pump

If no presupply pump is present, this integral vane-type supply pump draws fuel from the tank and, together with a pressure-control valve, generates an internal pump pressure which increases with engine speed.

High-pressure pump

The VE distributor-type fuel-injection pump incorporates only one pumping element for all cylinders. The element's plunger displaces the fuel during its stroke while at the same time rotating to distribute it to the individual outlets. During each rotation of the drive shaft, the plunger completes a number of strokes equal to the number of engine cylinders to be supplied. Via the yoke, the injection pump's drive shaft turns the cam plate and the pump plunger fixed to it. The lobes on the bottom surface of the cam plate turn against the rollers of the roller ring, causing the cam plate and plunger to make a stroke movement in addition to their rotary movement (distribution and delivery). The pump delivers fuel for as long as the spill port in the plunger remains closed off during the working stroke. Delivery ends when the spill port is uncovered by the control collar. The position of the control collar thus determines the effective stroke and the injected fuel quantity. The governor determines the position of the sliding control collar on the plunger.

Mechanical (flyweight) governor

A ball pin connects the control collar with the governor levers, which are acted upon

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by the governor springs and the centrifugal force generated by the flyweights. Idling, transition ranges and max. engine speed can be adapted to meet engine requirements.

Speed droop, governor types: The description of speed droop and governor types (variable-speed governors, minimum-maximum-speed governors) for in-line fuel-injection pumps also applies to governors used with distributor-type fuel-injection pumps.

Load signal: On distributor fuel-injection pumps equipped with minimum-maximum speed governors, the position of the outer control lever can be monitored via microswitch or potentiometer to provide information on load.

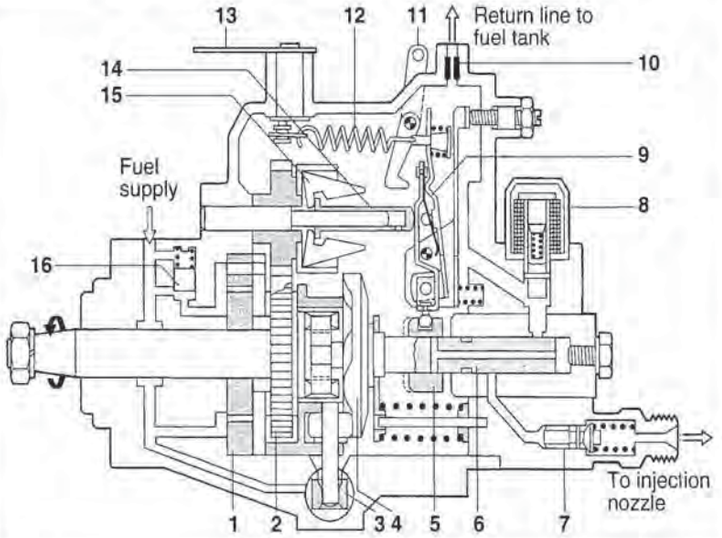
Add-on modules:
A number of add-on modules are available to process additional operating parameters for regulation of delivery quantity (such as manifold-pressure compensator (LDA), start quantity, hydraulic and mechanical full-load torque control) and for adjustment of the start of delivery – (e. g. cold-start accelerator, load-dependent start of delivery).

Hydromechanically controlled timing device
The speed-dependent supply-pump pressure (5...10 bar) acts on the front end of the spring-loaded timing-device plunger through a throttle bore. The plunger rotates the roller ring counter to the direction of rotation of the pump as a function of engine speed, thereby advancing the start of delivery.

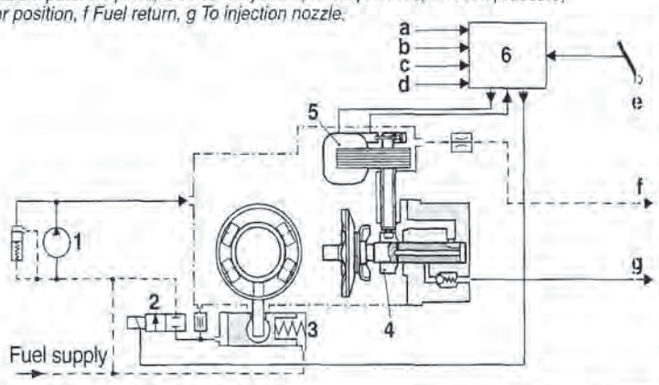
Pump shutoff
An electric shutoff device (solenoid valve) shuts off the diesel engine by interrupting the fuel supply.

Electronic Diesel Control (EDC): Distributor-type fuel-injection pump with rotating-solenoid actuator
In contrast to the mechanically controlled

VE Distributor-type fuel-injection pump (basic version)
1 Vane-type supply pump, 2 Governor drive, 3 Timing device, 4 Cam plate, 5 Control collar, 6 Distributor plunger, 7 Delivery valve, 8 Solenoid-actuated shutoff, 9 Governor lever mechanism, 10 Overflow throttle, 11 Mechanical shutoff device, 12 Governor spring, 13 Speed-control lever, 14 Control sleeve, 15 Flyweight, 16 Pressure-control valve.



Electronic diesel control (EDC) for distributor-type fuel-injection pumps
1 Supply pump, 2 Solenoid valve, 3 Timing device, 4 Control collar, 5 Rotary actuator with sensor, 6 ECU. Inputs/Outputs: a Speed, b Start of injection, c Temperature, d Boost pressure, e Accelerator position, f Fuel return, g To injection nozzle.



distributor-type fuel-injection pump, the EDC pump with rotating-solenoid actuator has an electronic governor and an electronically controlled timing device.

Electronic governor

An eccentrically-mounted ball pin provides the connection between the VE pump's control collar and the solenoid rotary actuator. The actuator's angular setting determines the position of the control collar, and with it the effective stroke of the pump. A non-contacting position sensor is connected to the rotary actuator. The ECU's microcomputer receives various signals from the sensors: accelerator-pedal position; engine speed; air, coolant and fuel temperature; boost pressure; atmospheric pressure; etc. It uses these input variables to determine the correct injected fuel quantity, which is then converted to a specific control-rack position with the aid of program maps stored in the unit's memory. The ECU varies the excitation current to the rotary actuator until it receives a signal indicating convergence between the setpoint and actual values for control-rack position.

Electronically-controlled timing device

In this device, the signal from a sensor in the nozzle-holder assembly, which indicates when the nozzle begins to open, is compared with a programmed setpoint value. A clocked solenoid valve connected to the working chamber of the plunger in the timing device varies the pressure above the plunger and thus the position of the timing device. The actuation clock ratio of the solenoid valve is varied until the setpoint and actual values agree.

Advantages of electronic versus mechanical control:

- Improved control of injected fuel quantity (fuel consumption, engine power, emissions),
- Improved control of engine speed (low idle speed, adjustment for air conditioner, etc.)
- Enhanced comfort (anti-surge control, smooth-running control),
- More precise start of injection (fuel consumption, emissions),
- Improved service possibilities (diagnostics).

The application options extend to embrace features such as open-loop and closed-loop control of exhaust-gas recirculation, boost-pressure control, glow-plug control, and interconnection with other on-board electrical systems.

Electronic Diesel Control (EDC): Solenoid-controlled distributor pumps

In the case of solenoid-controlled distributor pumps, the fuel is metered by a high-pressure solenoid valve which directly closes off the pump's element chamber.

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This permits even greater flexibility in fuel metering and in variation of the start of injection (see Figure for operating principle). The main assemblies of this new generation of distributor pumps are:
- the high-pressure solenoid valve,
- the ECU, and
- the incremental angle/time system for angle/time control of the solenoid valve using an angle of rotation sensor integrated into the pump.

The solenoid valve closes to define the start of the delivery, which then continues until the valve opens. The injected fuel quantity is determined by the length of time the valve remains closed. Solenoid-valve control enables rapid opening and closing of the element chamber irrespective of engine speed. In contrast to mechanically governed pumps and EDC pumps with a rotating-solenoid actuator, direct triggering by means of solenoid valves results in lower dead volumes, improved high-pressure sealing, and therefore greater efficiency.

The injection pump is equipped with its own, integral ECU for precise start-of-delivery control and fuel metering. Individual pump program maps and example-specific calibration data are stored in this ECU.

The engine ECU determines the start of injection and delivery on the basis of engine operating parameters, and sends this data to the pump ECU via the data bus. The system can control both the start of injection and the start of delivery.

The pump ECU also receives the injected fuel quantity signal via the data bus. This signal is generated by the engine ECU according to the accelerator-pedal signal and other parameters for required fuel quantity. In the pump ECU, the injected fuel quantity signal and the pump speed for a given start of delivery are taken as the input variables for the pump map on which the corresponding actuation period is stored as degrees of camshaft rotation.

And finally, the actuation of the high-pressure solenoid valve and the desired period of actuation are determined on the basis of the angle of rotation sensor integrated in the VE distributor pump.

The angle of rotation sensor in the pump is used for angle/time control. It consists of a magnetoresistive sensor and a reluctor ring divided into 3° increments interrupted by a reference mark for each cylinder.

The sensor determines the precise angle of camshaft rotation at which the solenoid valve opens and closes. This requires the pump ECU to convert timing data to angular position data and vice versa.

The low fuel-delivery rates at the start of injection, which result from the design of the VE distributor pump, are further reduced by the use of a two-spring nozzle holder. With a warm turbocharged engine, these low delivery rates permit low basic noise levels.

Pilot injection

Pilot injection allows the combustion noise to be further reduced without sacrificing the objects of the system's design which aim at generating maximum power output at the rated-power operating point. Pilot injection does not require additional hardware. Within a matter of milliseconds, the ECU actuates the solenoid valve twice in rapid succession. The solenoid valve controls the injected fuel quantity with a high degree of precision and dynamic response (typical pilot-injection fuel quantity: 1.5 mm^3).

Solenoid-controlled axial-piston distributor pumps

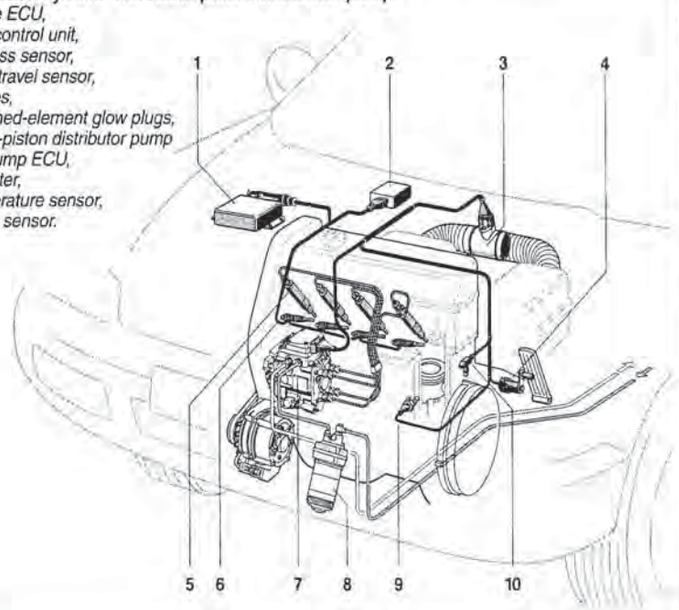
Axial-piston pumps use the same principle of pressure generation as EDC pumps with rotating-solenoid actuators. The injection pressure at the nozzle can be as high as 1500 bar, depending on the application.

By shifting the point at which delivery begins from the bottom dead center position of the pump plunger to a point within the plunger stroke, the pressure at low speeds can be increased and maximum pump torque can be substantially reduced at high speeds. For special applications, the variable start of delivery permits the start of injection to be advanced even at cranking speed.

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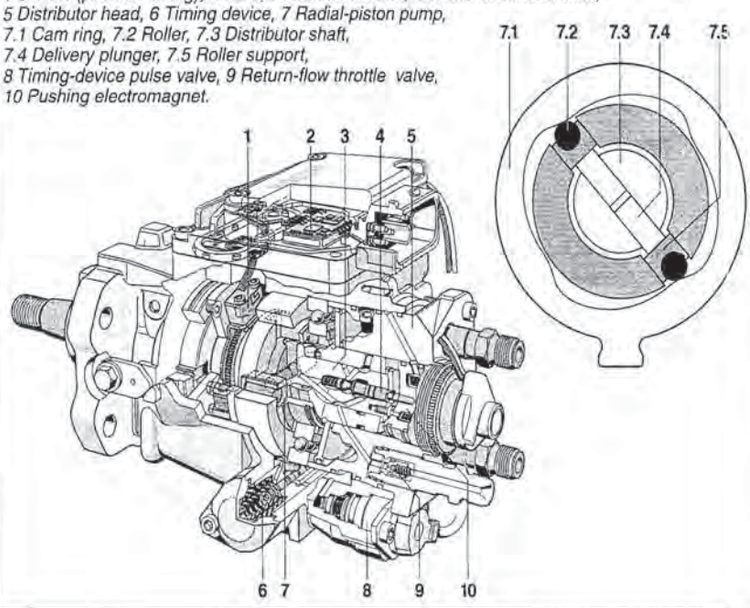
Fuel-injection system with radial-piston distributor pump

1 Engine ECU,
2 Glow-control unit,
3 Air-mass sensor,
4 Pedal-travel sensor,
5 Nozzles,
6 Sheathed-element glow plugs,
7 Radial-piston distributor pump with pump ECU,
8 Fuel filter,
9 Temperature sensor,
10 Speed sensor.



Solenoid-controlled radial-piston distributor pump

1 Sensor (position/timing), 2 ECU, 3 Distributor shaft, 4 Solenoid valve needle,
5 Distributor head, 6 Timing device, 7 Radial-piston pump,
7.1 Cam ring, 7.2 Roller, 7.3 Distributor shaft,
7.4 Delivery plunger, 7.5 Roller support,
8 Timing-device pulse valve, 9 Return-flow throttle valve,
10 Pushing electromagnet.



Solenoid-controlled radial-piston distributor pumps

Radial-piston pumps for high-performance direct-injection engines achieve element-chamber pressures of up to 1100 bar and nozzle pressures as high as 1950 bar.

Because the cam-drive design employs a direct, positive link, flexibility and compliance remain minimal, so the performance potential is greater. Fuel delivery is shared between at least two radial plungers. The small forces involved mean that steep (fast) cam profiles are possible. The fuel-delivery rate can be further increased by increasing the number of plungers.

Solenoid-controlled distributor pumps with integral ECU

The latest generation of distributor pumps are compact, self-contained systems incorporating an ECU to control both the pump and the engine-management functions. Because a separate engine ECU is no longer required, the injection system requires fewer plug-in connections and the wiring harness is less complex, thus making installation simpler.

The engine and complete fuel-injection system can be installed and tested as a self-contained system independent of vehicle type.

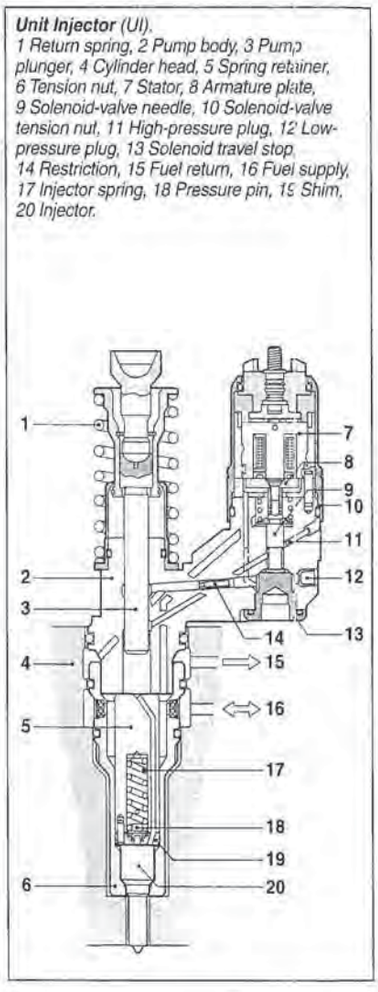
Time-controlled single-cylinder pump systems

The new modular, time-controlled single-cylinder pump systems include the electronically controlled "Unit Injector System" (UIS) and the "Unit Pump System" (UPS) which are used on modern direct-injection engines for commercial vehicles and passenger cars.

Unit injector system (UIS) for commercial vehicles

The electronically controlled unit injector is a single-cylinder fuel-injection pump with integral nozzle and solenoid valve which is installed directly in the cylinder head of the diesel engine. Each engine cylinder is allocated its own unit injector, which is operated by a rocker arm driven by an injection cam on the engine camshaft.

The start of injection and the injected fuel quantity are controlled by the high-speed solenoid valve. The values for these variables can be selected as desired from those stored in the program map. When de-energized, the solenoid valve is open. This means that fuel can flow freely from the fuel inlet of the low-pressure system through the pump and back into the low-pressure system in the engine cylinder head, thereby allowing the pump chamber to be filled during the pump plunger's suction stroke. Energizing the solenoid valve during the pump



***Unit Injector** (UI).*
1 Return spring, 2 Pump body, 3 Pump plunger, 4 Cylinder head, 5 Spring retainer, 6 Tension nut, 7 Stator, 8 Armature plate, 9 Solenoid-valve needle, 10 Solenoid-valve tension nut, 11 High-pressure plug, 12 Low-pressure plug, 13 Solenoid travel stop, 14 Restriction, 15 Fuel return, 16 Fuel supply, 17 Injector spring, 18 Pressure pin, 19 Shim, 20 Injector.

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plunger's delivery stroke closes this bypass, causing pressure to build up in the high-pressure system and fuel to be injected into the combustion chamber of the engine once the nozzle opening pressure is exceeded.

The compact design of the unit means that high-pressure volume is very small and hydraulic rigidity very high. As a result, injection pressures as high as 180 MPa (and 200 MPa in the future) can be achieved.

Such high injection pressures combined with electronic map-based control allow emission levels to be substantially reduced while simultaneously keeping fuel consumption low. The unit injector system is capable of meeting both present and future emission limits.

Electronic control enables this fuel-injection system to perform additional functions which are primarily intended to considerably improve driving smoothness.

By using adaptive cylinder matching, the complete drivetrain's rotational irregularity up to rated speed can be reduced. This ensures that the complete drivetrain runs much more smoothly. At the same time, this function can equalise the injected fuel quantity from the engine's individual injectors.

In the future, electrically controlled pilot injection (double triggering of the solenoid valve) will significantly reduce combustion noise and improve cold-starting characteristics.

In addition, the system permits the shutoff of individual cylinders. For instance, when the engine is running in the part-load range.

Unit pump system (UPS) for commercial vehicles

The unit pump system is also a modular, time-controlled single-cylinder high-pressure pump system, and is closely related to the UIS. Each of the engine's cylinders is supplied by a separate module with the following components:

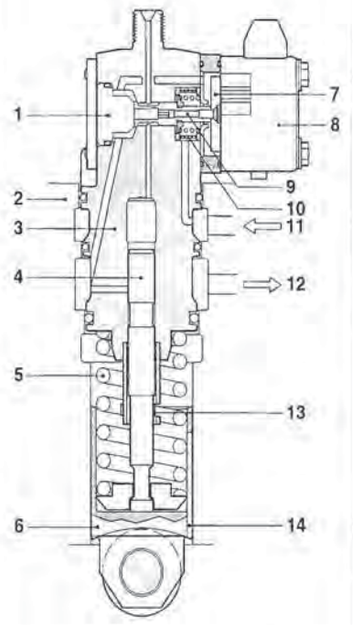
- high-pressure single-cylinder pump with integral high-speed solenoid valve,
- short high-pressure delivery line
- nozzle-holder assembly

The unit pump is integrated into the diesel-engine cylinder block and operated directly by an injection cam on the engine's camshaft via a roller tappet.

The method of solenoid valve actuation is the same as that of the UIS.

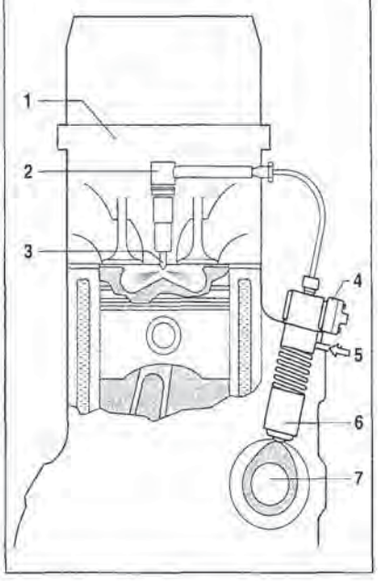
When the solenoid valve is open, the fuel can be drawn into the pump cylinder during the pump plunger's suction stroke, and return during the supply stroke. Only when the solenoid valve is energized, and thus closed, can pressure build up in the high-pressure system between the pump plunger and the nozzle during the pump plunger delivery stroke. Fuel is injected into the combustion chamber of the en-

***Unit Pump** (UP)*
1 Solenoid-valve needle-travel stop, 2 Engine block, 3 Pump body, 4 Pump plunger, 5 Return spring, 6 Roller tappet, 7 Armature plate, 8 Stator, 9 Solenoid-valve needle, 10 Filter, 11 Fuel supply, 12 Fuel return, 13 Retainer, 14 Locating groove.



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***Unit Pump System** (UPS)*
1 Injector nozzle holder, 2 Engine, 3 Injector nozzle, 4 Solenoid valve, 5 Supply, 6 High-pressure pump, 7 Cam.



gine once the nozzle opening pressure is exceeded.

Solenoid-valve closure thus defines the start of injection, and valve opening the injected fuel quantity.

The unit pump system can achieve injection pressures of up to 180 MPa (200 MPa in the future). As is the case with the UIS, these high injection pressures combined with electronic map-based control enable this injection system to meet both present and future emission limits while at the same time providing for low fuel consumption. This fuel-injection system also allows implementation of additional functions such as adaptive cylinder matching, electrically controlled pilot injection, and shutoff of individual cylinders.

Unit injector system (UIS) for passenger cars

The unit injector system for passenger cars is designed to meet the demands of modern direct-injection diesel engines with high levels of power density. It is characterized by its compact design, high injection pressures of up to 2000 bar, and mechanical-hydraulic pilot injection throughout the entire program-map range which substantially reduces combustion noise.

The unit injector system for passenger cars is also an individual-pump injection system, i.e. there is a separate unit injector (consisting of a high-pressure pump, nozzle and solenoid valve) for each engine cylinder. The unit injector is installed in the cylinder head between the valves, with the nozzle protruding into the combustion chamber. The unit injectors are operated by rocker arms driven by an overhead valve camshaft. The transverse mounting of the solenoid valve makes the unit more compact and achieves minimal high-pressure volume with correspondingly high hydraulic efficiency.

The injection system is filled during the pump-plunger suction stroke, while the solenoid valve is de-energized and thus open.

The injection period begins when the solenoid valve is energized (closed) during the pump-plunger delivery stroke. Pilot injection begins when pressure builds up in the high-pressure system and the nozzle opening pressure is reached. Pilot injection ends when a mechanical valve (bypass plunger) opens and abruptly reduces the pressure in the high-pressure chamber so that the nozzle closes. The stroke and shaft diameter of the bypass plunger determine the length of the interval, the so-called injection interval, between the end of pilot injection and the start of main injection. The movement of the bypass plunger also tensions the nozzle spring, which rapidly closes the nozzle at the end of pilot injection. Due to the strong hydraulic damping produced by a damper located between the nozzle needle and the nozzle spring, the opening stroke of the nozzle needle remains very short during pilot injection. Main injection begins when the nozzle opening pressure is reached. However, due to the additional force applied by the pretensioned nozzle spring, this pressure is twice as high as at the start of pilot injection. Injection ends when the solenoid valve is de-energized,

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and thus opened. The time interval between re-opening of the nozzle and opening of the solenoid valve therefore determines the quantity of fuel injected during the main-injection phase.

Electronic control allows the values for the start of injection and the injected fuel quantity to be selected as desired from those stored in the program map. This feature, together with the high injection pressures, makes it possible to achieve very high power densities combined with very low emission levels and exceptionally low fuel consumption.

The further reduction in size of the unit injectors will allow them to be used on 4-valve-per-cylinder engines in the future, thereby making it possible to reduce emissions even more. Its ability to further reduce emissions, coupled with further optimization of injection characteristics, makes the unit injector system capable of meeting future emission limits.

Electronic control unit (ECU)

The solenoid valves on the unit injector and unit pump are triggered by an ECU. The ECU analyses all of the relevant status parameters in the system relative to the engine and its environment, and defines the exact start of injection and injected fuel quantity for the operating state of the engine at any given time, thereby enabling environmentally friendly and economical engine operation. The start of injection is also controlled by a BIP signal (beginning of injection period) in order to balance out the tolerances in the overall system. Injection timing is synchronized with engine piston position by precise analysis of the signals from an incremental trigger wheel. In addition to the basic fuel-injection functions, there are a variety of additional functions for improving driving smoothness such as surge dampers, idle-speed governors, and adaptive cylinder matching. In order to meet strict safety requirements, the ECU automatically corrects and compensates for any faults and deviations that may occur in the system components and, when required, enables precise diagnosis of the injection system and the engine. The ECU communicates with other electronic systems on the vehicle such as the antilock braking system (ABS), the traction control system (TCS), and the transmission-shift control system via the high-speed CAN data bus.

Common-rail system (CRS)

Common-rail (accumulator) fuel-injection systems make it possible to integrate the injection system together with a number of its extended functions in the diesel engine, and thus increase the degree of freedom available for defining the combustion process.

The common-rail system's principal feature is that injection pressure is independent of engine speed and injected fuel quantity.

System design

The functions of pressure generation and injection are separated by an accumulator volume. This volume is the essential feature for the functioning of this system and is made up of volume components from the Common Rail itself, as well as from the fuel lines, and the injectors.

The pressure is generated by a high-pressure plunger pump. An in-line pump is used in trucks and a radial-piston pump in passenger cars. The pump operates at low maximum torques and thus substantially reduces drive-power requirements. For the high-pressure pumps in passenger cars, the required fuel-rail pressure is regulated by a pressure-control valve mounted on the pump or the rail. High-pressure pumps in commercial vehicles have a fuel-quantity control system. The latest generation of high-pressure pumps for passenger-car use also has a fuel-quantity control system. This reduces the temperature of the fuel within the system.

The system pressure generated by the high-pressure pump flows through a pressure-control circuit and is applied to the conventional injector. This injector serves as the core of this concept by ensuring correct fuel delivery into the combustion chamber. At a precisely defined instant the ECU transmits an activation signal to the injector solenoid to initiate fuel delivery. The injected fuel quantity is defined by the injector opening period and the system pressure.

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The ECU, sensors and most of the other system functions in the common-rail system are basically the same as in other time-controlled single-pump systems.

Hydraulic performance potential
This system enhances the latitude for defining combustion-process patterns by separating the pressurization and injection functions. Injection pressure can basically be selected from any point on the program map. The pressures currently used are 1350 bar in passenger-car systems and 1400 bar in commercial-vehicle systems.

Pilot injection and multiple injection can be used to further reduce exhaust and particularly noise emissions.

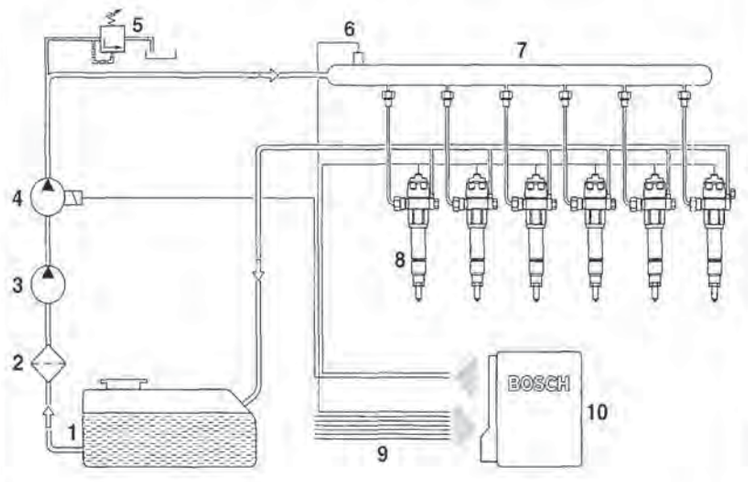
In the common-rail system, the movement of the nozzle needle, and thus the injection pattern, can be controlled within a defined range. The system can trigger the extremely fast solenoid several times in succession for multiple injection. Hydraulic pressure is used to augment injector-needle closing, ensuring rapid termination of the injection process.

System application engineering on the engine
No major modifications are required to adapt the diesel engine for operation with the common-rail system. A high-pressure pump replaces the injection pump, while the injector is integrated in the cylinder head in the same manner as a conventional nozzle-and-holder assembly. All of these features make the common-rail configuration yet another injection-system option.

Injection-pump test benches

Thorough testing and precise adjustment are indispensable if injection pumps and governors are to assist the diese engine in achieving its optimal consumption and output while at the same time allowing it to maintain compliance with today's ncreasingly stringent emissions requirements. This is where the injection-pump test bench is essential. The basic specifications for test procedure and test bench are stipulated in ISO standards, which place especially severe demands on the rigidity and drive uniformity of the drive unit.

Common-rail accumulator injection system
1 Fuel tank, 2 Filter, 3 Presupply pump, 4 High-pressure pump, 5 Pressure-control valve, 6 Pressure sensor, 7 Fuel rail, 8 Injectors, 9 Sensors, 10 ECU.



The injection pump under test is clamped to the test bench and its drive side is connected to the test-bench coupling. The test-bench drive unit comprises a special motor attached directly to the flywheel. Test-bench control is by means of a frequency converter with a vector control loop. Supply and return lines connect the injection pump to the test bench calibrating-oil supply, while pressure lines lead to the fuel-delivery measuring device. This consists of calibrating nozzles set to a precise opening pressure, which inject calibrating oil directly into the measuring system via injection dampers. The pressure and temperature of the calibrating fluid can be adjusted to comply with the test specifications.

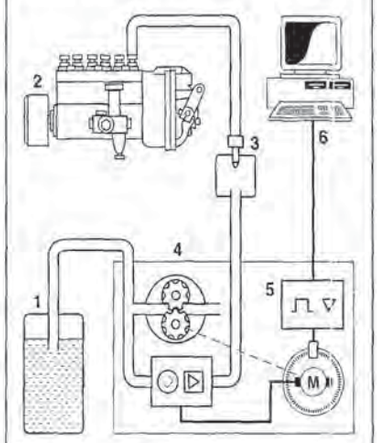
Continuous-flow delivery-quantity measurement

Using the continuous flow method of delivery-quantity measurement, the 12 measurement inputs are connected to two precision gear pumps via a hydraulic multiplexing device. The gear pump's speed is regulated so that the quantity of calibrating oil which it pumps corresponds with the amount of calibrating oil being discharged. The pump's speed thus provides an index for the flow quantity per unit of time. A microprocessor analyzes the measurement results and converts them to bar-graph form for display on a monitor. This test method is characterized by a high degree of accuracy and consistently reproducible test results.

Delivery-quantity measurement using measuring glasses

Quantity measurement with measuring glasses (graduates) starts with the calibrating oil from the test nozzles being routed past the graduates and back to the calibrating-oil reservoir. The control unit waits until the prescribed number of strokes has been entered at the stroke counter before starting the actual test by switching the calibrating-oil flow to the graduates. The flow is interrupted again once the prescribed number of strokes has been completed. The quantity of calibrating oil which has been discharged by the test nozzles can be read from the graduates.

Continuous-flow delivery-quantity measurement
1 Calibrating-fluid reservoir, 2 Injection pump, 3 Test nozzle, 4 Measurement cell, 5 Pulse counter, 6 Display monitor.



"Motor testers" for diesel engines

The diesel-pump tester is used to calibrate the pump precisely to the engine's requirements. The unit monitors port closing (start of delivery) and timing adjustment at specific engine speeds without any need to open the high-pressure lines. An inductive clamp sensor is attached to the injection line for cylinder no. 1. In conjunction with a stroboscope or TDC sensor for monitoring crankshaft position, the diesel-pump tester determines the port closing and the degree of timing adjustment.

If a port-closing sensor system is used, an inductive sensor is screwed into the governor housing. The sensor receives pulses from a pin when this moves past the sensor. This pin is attached to the governors flyweight housing. These pulses trail the signals from the TDC sensor at a specific interval which is used by the unit to calculate the start of delivery.

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Nozzles and nozzle holders

Functions

In a diesel-engine fuel-injection system, the nozzles connect the injection pump to the engine. Their functions are to:
- Assist in metering the fuel,
- Process the fuel,
- Define the rate-of-discharge curve,
- Seal off the combustion chamber.

In systems with separate injection pumps (inline, distributor and plug-in pumps) the nozzles are integrated in the nozzle holders. On unit injector systems (UIS) and common-rail systems (CRS), the nozzles are integral parts of the injectors.

Diesel fuel is injected at high pressure. Peak diesel fuel injection pressure can range as high as 2000 bar, a figure which will become even higher in the future. Under these conditions the diesel fuel ceases to behave as a solid, incompressible fluid, and becomes compressible. During the brief delivery period (in the order of 1 ms), the injection system is locally "inflated." For a given pressure, the nozzle cross section is one of the factors determining the quantity of fuel injected into the engine's combustion chamber.

The length and diameter of the nozzle spray hole (or orifice), the direction of the fuel jet and (to a certain degree) the shape of the spray hole affect mixture formation, and thus the engine's power output, fuel consumption, and emission levels.

Within certain limits, it is possible to achieve the required rate-of-discharge curve through optimal control of the injector's aperture (defined by the needle's stroke) and by regulating the injector needle's response. Finally, the injection nozzle must be capable of sealing the fuel-injection system against the hot, highly-compressed combustion gases with temperatures up to approx. 1000 °C. To prevent backflow of the combustion gases when the injection nozzle is open, the pressure in the injection nozzle's pressure chamber must always be higher than the combustion pressure. This requirement becomes particularly relevant toward the end of the injection sequence (when a stark reduction in injection pressure is accompanied by massive increases in combustion pressure), where it can only be ensured by carefully matching the injection pump, the injection nozzle and the nozzle needle for mutually satisfactory operation.

Designs

Diesel engines with divided or two-section combustion chambers (prechamber and whirl- (or turbulence) chamber engines) require nozzle designs differing from those used in single-section chambers (direct-injection engines).

In prechamber and whirl-chamber engines with divided combustion chambers, throttling-pintle nozzles are used which feature a coaxial spray pattern and which are generally equipped with needles which retract to open.

Direct-injection engines with single-section combustion chambers generally require hole-type nozzles.

Throttling-pintle nozzles

One injector (Type DN..SD..) and one injector holder (Type KCA for threaded socket installation) represent the standard combination for use with prechamber and whirl-chamber engines. The standard nozzle holder features M24x2 threads and uses a 27 mm wrench fitting. DN O SD.. nozzles with a needle diameter of 6 mm and a spray aperture angle of 0° are usually used; less common are nozzles with a defined spray dispersal angle (for example 12° in the DN 12 SD..). Smaller holders are used when only limited space is available (e.g., KCE holders).

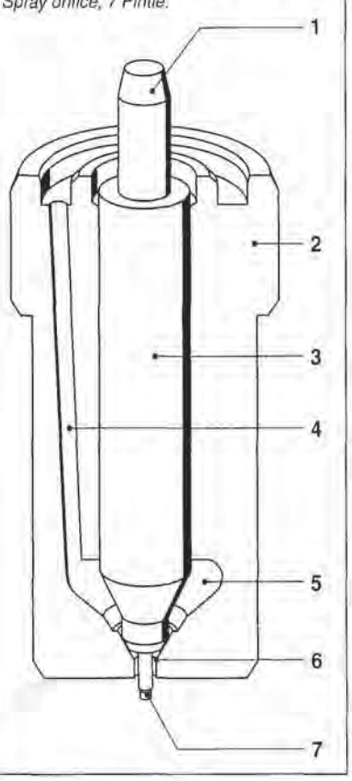
As a distinctive feature, throttling-pintle nozzles vary the discharge aperture – and thus the flow rate – as a function of needle stroke. The hole-type nozzle displays an immediate, sharp rise in aperture when the needle opens; in contrast, the throttling-pintle nozzle is characterized by an extremely flat aperture pro-

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gression at moderate needle strokes. Within this stroke range, the pintle (an extension at the end of the needle) remains in the spray orifice. The flow opening consists only of the small annular gap between the larger spray orifice and the throttling pintle. As needle stroke increases, the pintle completely opens the spray orifice, with an attendant substantial increase in the size of the aperture.

This stroke-sensitive aperture regulation can be employed to exert a certain amount of control on the rate-of-discharge curve (quantity of fuel injected into the engine within a specific period). At the start of injection, only a limited amount of fuel emerges from the injector nozzle, while a substantial quantity is discharged at the end of the cycle. This rate-of-discharge curve has a particularly positive effect on combustion noise.

Throttling-pintle nozzle
1 Pressure pin, 2 Nozzle body, 3 Needle, 4 Inlet passage, 5 Pressure chamber, 6 Spray orifice, 7 Pintle.



It must be remembered that excessively small apertures, i.e. excessively short needle strokes, cause the injection pump to more strongly push the nozzle needle in the "open" direction, thereby causing the needle to quickly emerge from the throttling stroke area. The injected fuel quantity per unit of time increases dramatically, and combustion noise rises accordingly.

Similarly, negative effects result from excessively small openings at the end of the injection cycle: the volume displaced by the closing nozzle needle is restricted by the narrow aperture. The result is an undesirable extension of the injection duration. Thus aperture configurations must accurately reflect both the injection-pump's delivery rate and the specific combustion conditions.

Special manufacturing processes are employed to produce spray holes to precise geometrical tolerances. During engine operation, substantial and unfortunately very irregular carbon deposits form in the throttle gap. The degree of deposit formation is determined by the quality of the fuel and the engine's operating conditions. In most cases only 30 % of the initial flow channel remains unobstructed.

Fewer and more even deposits are found on flat pintle nozzles, in which the annular opening between the nozzle body and the throttle pintle is almost zero. Here the throttle pintle utilises a machined surface to open the flow aperture. The resulting flow passage features reduced surface area relative to the flow opening, resulting in an enhanced self-cleaning effect. The machined surface is frequently parallel to the axis of the nozzle needle. Additional inclination can be employed to produce a more pronounced rise in the flat part of the flow curve, allowing a smoother transition to full nozzle opening. This expedient has a positive effect on part-load noise emissions and on operating characteristics.

Temperatures above 220 °C also promote deposit formation on injectors. Thermal shields are available to transfer the heat from the combustion chamber back to the cylinder head.

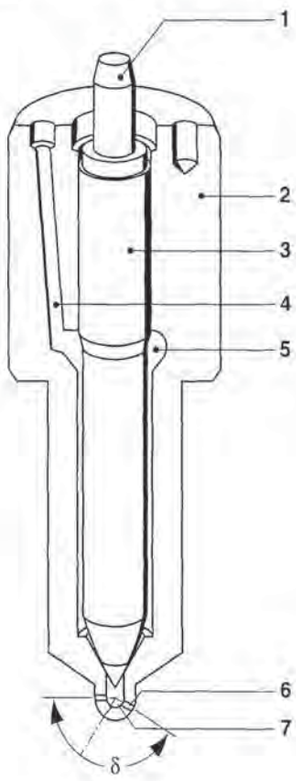
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Hole-type nozzles

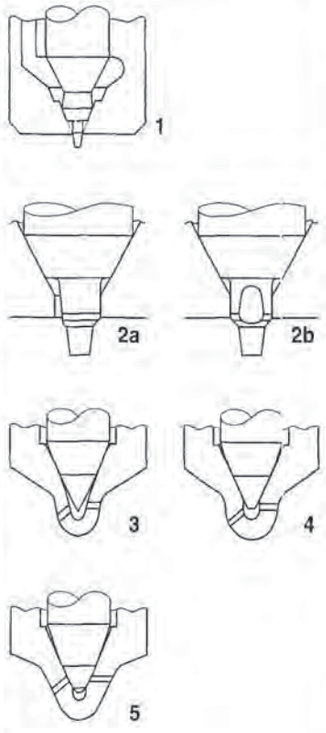
A wide range of nozzle-and-holder assemblies (DHK) is available for hole-type nozzles. In contrast to throttling pintle nozzles, hole-type nozzles must generally be installed in a specific position to ensure correct alignment between the orifices (which are at different angles in the nozzle body) and the engine combustion chamber. For this reason, lugs or hollow screws are usually employed to attach the nozzle-and-holder assemblies to the cylinder head while a locating device ensures the proper orientation.

Multihole nozzles are available with needle diameters of 6 and 5 mm (Size S) and 4 mm (Size P), with sac-less (vco) nozzles available in the latter size only. The nozzle springs must be suitable for use with the particular needle diameters and the normally extreme opening pressures (> 180 bar). At the end of the injection sequence there is a pronounced danger of the combustion gases being blown back into the nozzle, a development which would, in the course of time, result in destruction of the nozzle and hydraulic instability. The nozzle-needle diameter

Hole-type nozzle
1 Pressure pin, 2 Nozzle body, 3 Nozzle needle, 4 Inlet passage, 5 Pressure chamber, 6 Injection orifice, 7 Blind hole, δ Spray-hole cone angle. Dead-volume space = 6 + 7



Nozzle shapes
1 Throttling-pintle nozzle, 2 Throttling-pintle nozzle with flat-cut pintle, 2a Side view, 2b Front view, 3 Hole-type nozzle with conical blind hole, 4 Hole-type nozzle with cylindrical blind hole, 5 Sac-less (vco) nozzle.



and the compression spring are carefully matched to one another to ensure a good seal. In special cases, it is even necessary to allow for oscillation of the compression spring.

There are three different ways in which the spray orifices are arranged in the cone of the hole-type nozzle. These three designs differ in the amount of fuel which can freely evaporate into the combustion chamber at the end of the injection cycle. The designs with a cylindrical blind hole, conical blind hole, as well as the sac-less nozzle, have successively smaller fuel volumes in that order.

The engine hydrocarbon emissions decrease in the same order due to there being less residual fuel available for evaporation.

The length of the spray orifice is limited by the nozzle cone's mechanical integrity. At present, the minimum spray-orifice length is 0.6...0.8 mm for cylindrical and conical blind holes. The 1 mm minimum for sac-less (vco) nozzles is available only when special processing methods are employed to produce the spray orifices.

The tendency is toward shorter holes, as these generally allow better control of smoke emissions. Fuel flow tolerances of around ± 3.5% can be achieved in drilled hole-type nozzles. Additional hydro-erosive rounding of the inflow edges of the spray orifices can refine these tolerances to ± 2%.

Particularly for use in low-emission direct-injection diesel engines for cars, further refinement has been carried out on the injector nozzle. By optimizing the dead-volume space (see Fig. Hole-type nozzle, 6 and 7) in the nozzle body and modifying the injection-orifice geometry it has been possible to achieve maximum pressure at the injection orifice outlet in order to produce the optimum A/F mixture. In sac-less (vco) nozzles, spray-dispersal uniformity can be improved by the use of a double needle guide and a complex needle-tip geometry. The latter measure also improves performance stability throughout the nozzle's service life. These improvements required more advanced manufacturing processes, and in particular more sophisticated measuring methods.

The high-temperature strength of the material used in hole-type nozzles limits peak temperatures to approx. 300 °C. Thermal-protection sleeves are available for operation in especially difficult conditions, and there are even cooled injection nozzles for large-displacement engines.

Nozzle holders

Standard nozzle holders

The basic injector nozzle-and-holder assembly comprises the nozzle and the holder. The injector nozzle consists of two sections: the body and the needle. The nozzle needle moves freely within the body's guide bore while at the same time providing a positive seal against high injection pressures. At the bottom of the needle is a conical seal, which the nozzle spring presses against the body's correspondingly shaped sealing surface when the nozzle is closed. These two opposed conical surfaces exhibit a slight mutual variation in aperture angle, providing linear contact with high dynamic compression and a positive seal.

The diameter of the needle guide is greater than that of the seat. The hydraulic pressure from the injection pump acts against the differential surface between the needle diameter and the surface covered by the seat. The injection nozzle opens when the product of sealing surface and pressure exceeds the force of the nozzle spring in the holder. Because this process produces a sudden increase in pressurized surface area – with the seat suddenly joining the needle – a sufficiently high delivery rate will result in the injection nozzle snapping open very rapidly. It does not close again until the system has dropped from its opening pressure to below the (lower) closing pressure. This hysteresis effect is of particular significance when designing hydraulic stability into fuel-injection systems.

The opening pressure of a nozzle-and-holder combination (approx. 110...140 bar for a throttling pintle nozzle and 150...250 bar for a hole-type nozzle) is adjusted by placing shims under the compression spring.

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Closing pressures are then defined by the injection nozzle's geometry (ratio of needle diameter to seat diameter).

Dual-spring nozzle holders

These nozzle holders are primarily used in direct-injection (DI) engines, where precise pilot delivery patterns are the most important factors in reducing noise levels.

Pilot injection furnishes relatively gentle pressure rises for a quiet, stable idle along with a general reduction in combustion noise.

The dual-spring nozzle holder produces this effect by improving the rate-of-discharge curve, based on precise control and definition of

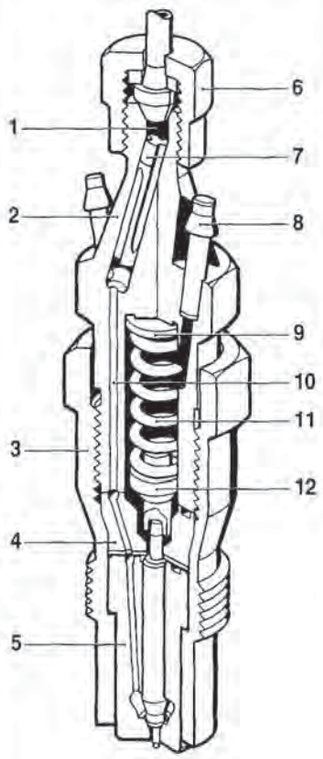
- opening pressure 1,
- opening pressure 2,
- prestroke, and
- overall stroke.

Opening pressure 1 is set and tested as with the single-spring holder. Opening pressure 2 is the sum of the pretension figures for spring 1 and auxiliary spring 2. Spring 2 is supported by a stop sleeve into which has been machined the dimensions of the prestroke (see Fig. Dual-spring nozzle holder KBEL..P.., H1). During injection, opening of the nozzle needle is initially restricted to the prestroke range. Common prestroke figures are 0.03...0.06 mm. As the pressure in the nozzle holder increases, the stop sleeve is lifted, allowing the nozzle needle to move to the end of its stroke. Also designed for use in dual-spring holders are the special-purpose injector nozzles in which the nozzle needle has no pintle, and the shoulder of the needle is level with the nozzle body.

In other words, the springs in the dual-spring holder are matched such that initially only a small quantity of fuel is injected into the combustion chamber, causing a slight initial rise in combustion pressure. The resulting extension in injection duration (with the main fuel delivery following the pilot delivery) serves to smooth out the combustion process.

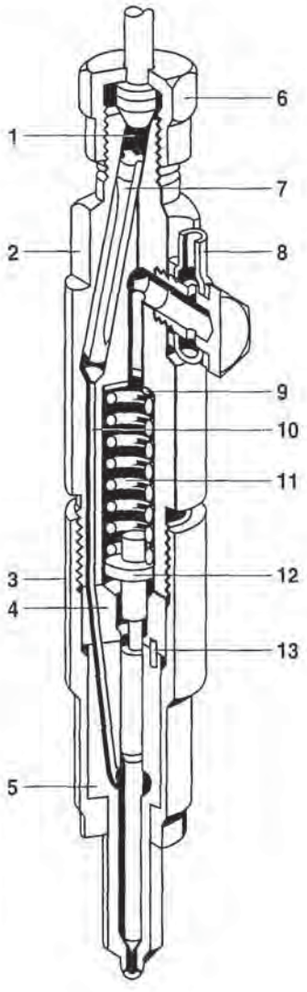
There are also dual-spring holders available for prechamber and whirl (turbulence) chamber engines. The setpoints are tailored to the respective injection system, with varying opening pressures of 130/180 bar and prestrokes of approx. 0.1 mm.

Nozzle-holder assembly *with throttling-pintle nozzle.*
1 Inlet, 2 Nozzle-holder body, 3 Nozzle-retaining nut, 4 Shim, 5 Injection nozzle, 6 Union nut with high-pressure line, 7 Edge filter, 8 Leak-off connection, 9 Pressure-adjusting shims, 10 Pressure passage, 11 Pressure spring, 12 Pressure pin.

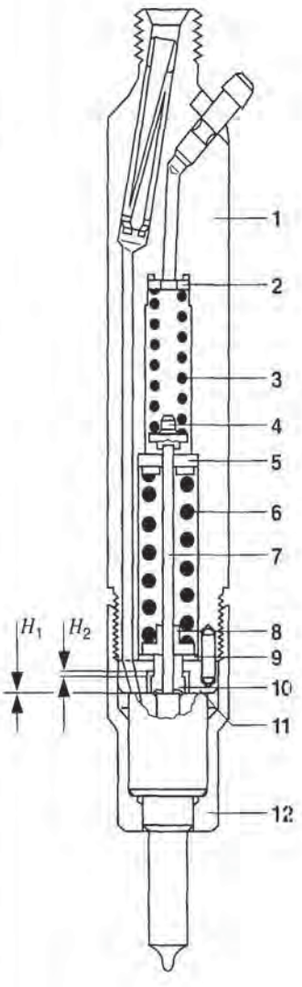


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Nozzle-holder assembly *with hole-type nozzle.*
1 Inlet , 2 Nozzle-holder body, 3 Nozzle-retaining nut, 4 Intermediate element, 5 Injection nozzle, 6 Union nut with high-pressure line, 7 Edge filter, 8 Leak-off connection, 9 Pressure-adjusting shims, 10 Pressure passage, 11 Pressure spring, 12 Pressure pin, 13 Locating pins.



KBEL...P... Dual-spring nozzle holder
H_1 Prestroke, H_2 Main stroke,
$H_{tot} = H_1 + H_2$ Total stroke.
1 Holder body, 2 Shim, 3 Compression spring 1, 4 Pressure pin, 5 Guide washer, 6 Compression spring 2, 7 Pressure pin, 8 Spring seat, 9 Shim, 10 Stop sleeve, 11 Spacer, 12 Nozzle-retaining nut.



Exhaust emissions from diesel engines

The primary and secondary constituents of the exhaust gases produced by diesel engines are listed together with the emissions from spark-ignition (SI) engines beginning on P. 518. Table 1 provides information on the composition and temperature of the exhaust gases from diesel engines.

Mixture formation

The fuel used in diesel engines has a higher boiling point than that used in gasoline engines. In addition, the A/F mixture in diesel engines is formed quickly just before combustion starts and is therefore less homogeneous. Diesel engines operate with excess air ($\lambda > 1$) across their entire operating range. An insufficient quantity of excess air results in increased particulate emissions (soot), and CO and HC emissions.

Combustion

The start of injection marks the initiation of the combustion process. The engine's efficiency is determined by the start of combustion and by the combustion characteristics. The characteristics (as a function of time) of the injected fuel quantity and the injection pressure, can be applied to control the combustion characteristics. These factors also determine the combustion temperature which, in turn, has a significant effect on the formation of nitrogen oxides (NO_x).

Emissions control

Measures at the engine

Combustion chamber

Exhaust-gas emission is affected by the design of the combustion chamber. Engines which have a divided combustion chamber (prechamber, swirl (turbulence) chamber) produce fewer nitrogen oxides than direct-injection engines, although these feature better fuel economy.

Careful adaptation of the air flow characteristics inside the combustion chamber to the fuel-jet pattern promotes better mixing of fuel and air, and thus more complete combustion. Reliable ignition requires a sufficiently high compression temperature.

Fuel injection

The start of injection, the rate-of-discharge curve, and the atomization of the fuel all have an effect on pollutant emissions. The point at which combustion starts is essentially a function of the start of injection (injection timing). Delayed injection reduces NO_x emissions, whereas excessive delay results in increased fuel consumption and HC emissions. With regard to the start of injection, a deviation of 1° (crankshaft) from the setpoint can increase NO_x emissions or

Table 1: Composition and temperature of diesel exhaust gas

Exhaust-gas components and temperature		at idle	at maximum output
Nitrous oxides (NO_x)	ppm	50...200	600...2500
Hydrocarbons (HC)	ppm C_1	50...500	<50
Carbon monoxide (CO)	ppm	100...450	350...2000
Carbon dioxide (CO_2)	Vol%	...3.5	12...16
Water vapor	Vol%	2...4	...11
Oxygen	Vol%	18	2...11
Nitrogen, etc.	Vol%	residual	residual
Smoke number, passenger cars		SZ ≈ <0.5	SZ ≈ 2...3
Exhaust-gas temperature downstream of exhaust valve	°C	100...200	550...750

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HC emissions by as much as 15%.

This high degree of sensitivity means that precise injection timing is essential. Electronic control systems are capable of maintaining optimum injection timing with a high degree of precision. Such devices control the timing-device setting, or the actuation of the injection-system solenoid valve relative to a crankshaft-angle mark (start-of-delivery control). Greater precision can be achieved by measuring the start of injection directly at the fuel injector by using a needle-motion sensor to detect the movement of the needle valve (start-of-injection control). In systems with solenoid valves, the start of injection can also be controlled by means of the current applied to the valve coil.

Any fuel entering the combustion chamber after combustion has terminated could be discharged directly into the exhaust system in unburned form, thus raising hydrocarbon emissions. To prevent this, the fuel volume between the injection nozzle seat and its injection orifices is held to a minimum. Sac-less (vco) nozzles completely seal off the injection orifice. In addition, "post-injection" must be avoided at all costs.

Finely atomized fuel promotes formation of a good A/F mixture which helps to reduce hydrocarbon and particulate emissions. Fine atomization is achieved with high injection pressures and optimum injection-orifice geometry.

The maximum injected fuel quantity for a given intake air mass must be limited such that the engine does not produce visible soot emissions. This requires an excess-air factor of at least 10...20% (λ = 1.1...1.2).

Intake-air temperature

As the temperature of the intake air increases the combustion temperature increases along with it, which in turn leads to an increase in NO_x emissions. Charge-air cooling (intercooling) is an effective means of reducing NO_x formation in turbocharged engines.

Exhaust-gas recirculation (EGR)

The exhaust gases mixed with the intake air reduce the amount of oxygen in the fresh intake charge while increasing its specific heat. Both factors result in a lower combustion temperature and thus decreased NOx production, and also reduce exhaust emissions. However, too much recirculated exhaust gas results in increased particulate and carbon-monoxide emissions due to air deficiency. The quantity of recirculated exhaust gases must therefore be limited to ensure that the combustion chamber receives sufficient oxygen to support combustion.

Exhaust-gas treatment

The use of noble-metal catalytic converters in the exhaust system reduces hydrocarbon emissions by burning a portion of the gaseous hydrocarbons and those bound to the soot (particulates) using the oxygen in the exhaust gases.

The catalytic converters used to reduce the NO_x emissions produced by gasoline engines must operate with either an oxygen deficit or a precise stoichiometric mixture (λ = 1). Diesel engines, however, can only be operated with excess air (λ > 1) because of the heterogeneous A/F mixture. Thus such conventional catalytic converters cannot be used on diesel engines. However, filters can be installed in the exhaust system to reduce particulate emissions (soot).

Emissions testing

In industrialized nations, and increasingly worldwide, harmful emissions from internal-combustion engines are limited by law. Exhaust-gas emissions tests are conducted under defined conditions.

Test layout

In general, the testing of passenger-car exhaust emissions takes place on a chassis dynamometer whereas commercial-vehicle emissions are tested on an engine test bench.

The prescribed method for dynamic test cycles and particulate-emission testing is the CVS (constant volume sampling) method. In this process, the exhaust gases produced by the vehicle are diluted with filtered ambient air and extracted with a blower during a standardized test cycle.

A constant volume of gas is delivered as determined either by a positive displacement pump (PDP) or a critical flow venturi (CFV). The temperature level specified for particulate-emissions testing

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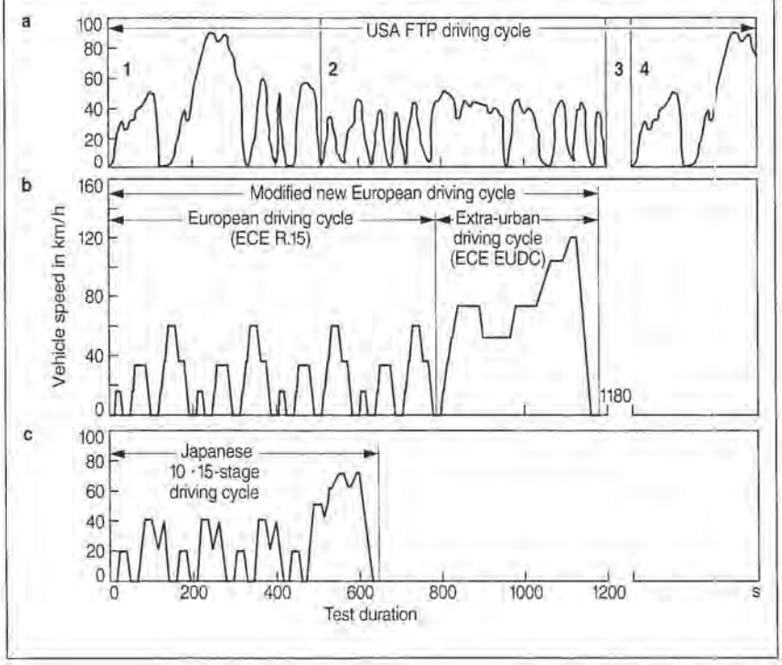
(max. 52 °C) must be maintained during the sampling procedure.

A sample of the diluted exhaust gas is passed through special filters whose increase in mass (determined by weighing on a microbalance before and after sampling) is used to calculate the level of particulate emissions.

A second sampling line (heated to 190 °C) leads to a flame ionization detector (FID), and a chemo-luminescence detector (CLD) for diesels, which continuously measure the hydrocarbon concentration (FID) and the nitrogen oxide concentration (CLD). A third sample is collected in exhaust-gas collection bags. Based on the contents of the bags at the end of the test, gas analyzers determine the CO, CO_2 and NO_x concentrations. The "ambient-air concentration" is determined by comparing the collected intake air with the exhaust so that the emissions originating from the engine can be distinguished from pollution already in the atmosphere. Calculations to determine the levels of the various exhaust-gas component emissions are based on the volume of mixed gas and the concentrations of the individual components.

In the USA, the same procedures and analyzers are employed in testing emissions produced by passenger-car and truck engines. The exhaust gases are usually diluted in a two-stage procedure to enable the larger volumes of gases to be processed using reasonably-sized dilution tunnels while simultaneously ensuring compliance with the legally prescribed conditions. In the European stationary test cycle, partial-flow dilution is also approved for use in measuring particulates. Testing for particulate levels is usually followed by an additional examination of exhaust-gas opacity in both stationary and dynamic full-throttle operation.

Driving cycles for emissions testing of diesel-powered passenger cars and light commercial vehicles
a) USA, b) Europe, c) Japan.
1 Transition phase (ct), 2 Stabilized phase (s), 3 Engine off for 600 s, 4 Hot test (ht) (curve as in ct).



Test cycles and exhaust-emission limits

Passenger cars and light utility vehicles

Europe

Emission-control legislation in the nations of the European Union is based on ECE Directive R15 along with EEC Directive 70/220 as revised. Existing emission limits (Euro 2) have been in effect since 1997 for production vehicles, and are expected to be lowered further in the year 2000 (Euro 3) and again in 2005 (Euro 4).

The original driving cycle according to ECE R15 has been completely replaced by the ECE R83 New European Driving Cycle (NEDC) with its non-urban component and speeds of up to 120 km/h, which in turn will be replaced beginning in the year 2000 (Euro 3) by the Modified New European Driving Cycle (MNEDC), in which the exhaust gas is collected immediately without a (40 s) delay. The entry of new nations into the European Union means that their emissions regulations will have to be revised so that they agree with EU regulations.

Japan

A 10·15 driving cycle is used to determine the concentrations of gaseous pollutants and particulates in diesel-engine exhaust gas. The driving cycle for passenger cars has been extended to include a high-speed section (similar to Europe).

USA

The FTP (Federal Test Procedure) 75 driving cycle is required for passenger cars and light commercial vehicles with a gross vehicle weight of less than 8500 lbs.

The speed curve corresponds to an urban operation cycle in the USA. Testing is performed on a chassis dynamometer, and the results are measured using the CVS method. Since it has now become enormously complex, emission-control legislation in the USA can only be described here in very abridged form. Additional test cycles have been integrated, including for instance SFTP-US 06 (maximum speed); SFTP-SC 03 (air-conditioner operation); HWFET (fuel economy) with averaging. A variety of service-life requirements, together with emission limit crediting, staggered phase-in of emission limits, and local regulations for vehicle-fleet operators in highly polluted regions of the country (California), etc. have also been incorporated.

Commercial vehicles

Europe

In Europe, until 1999 vehicles with a gross vehicle weight of > 3.5 tons and more than 9 seats were required to meet the stipulations of the 13-stage test in accordance with ECE R49. The test sequence stipulated a series of 13 different steady-state operating modes. An average emission level for both the gaseous exhaust components and the particulates was calculated by applying weighting factors to the measurements of emissions taken in the various operating modes.

Directive 91/542/EEC sets forth the limits which are currently applicable in the EU. The levels prescribed by the EURO 2 stage have been in effect since 1995. The 3rd stage (EURO 3), which will go into effect in the year 2000, also involves changes to the test cycle to bring it into agreement with the European Steady-state Cycle (ESC).

A European Transient Cycle (ETC) is planned for a further stage to begin in 2005 (EURO 4).

Japan

Pollutant emissions are measured using the Japanese 13-stage steady-state test, however the operating points, their order and their relative weighting differ from those of the European 13-stage test. Emission limits, which in Japan are still classified according to vehicle weight (over or under 12 t), are shown here for vehicles over 12 t.

USA

Since 1987, engines for heavy commercial vehicles have been tested on an engine test bench in a transient cycle, with emissions being measured according to the CVS method. The test cycle is based on highway operation under real-world conditions. The opacity of the exhaust gas

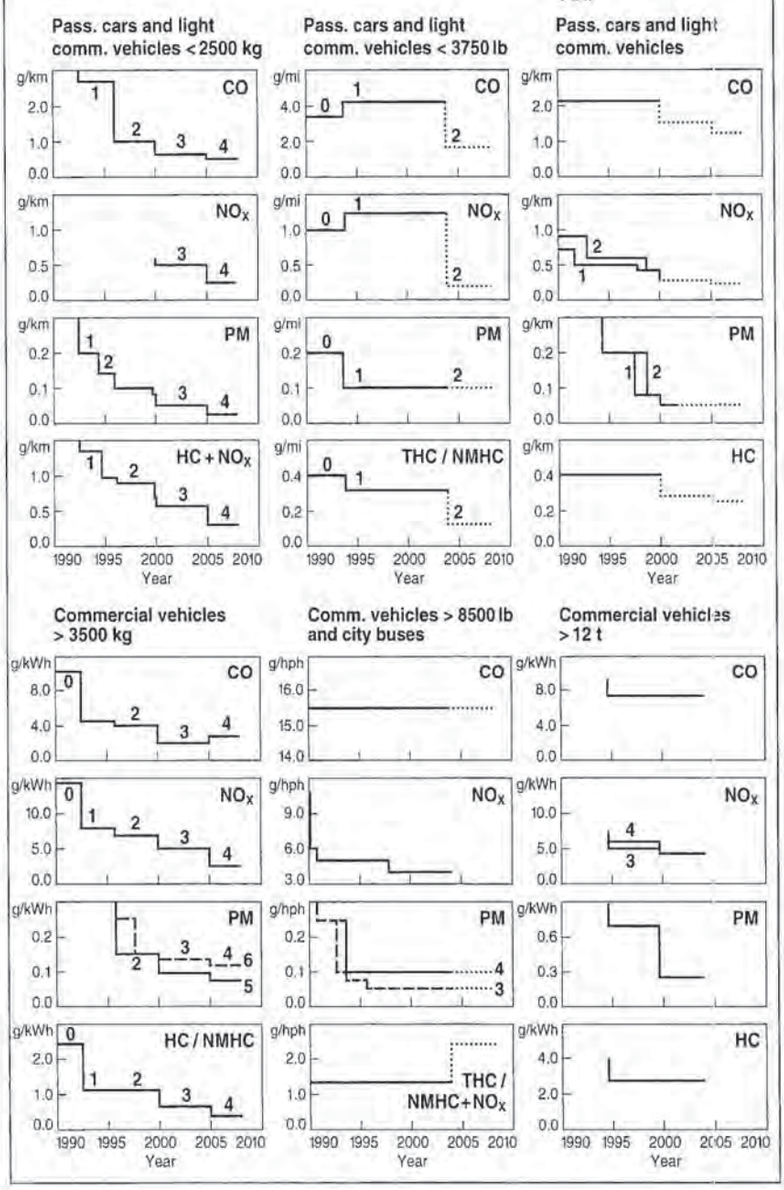
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Emission limits for diesel-powered passenger cars and commercial vehicles

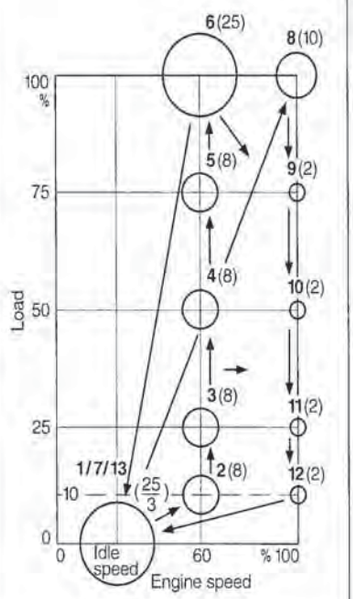
Europe
0...4 Euro 0...4,
5 >85 kW,
6 <85 kW.

USA
0...2 Level 0...2,
3 HDV,
4 UB.

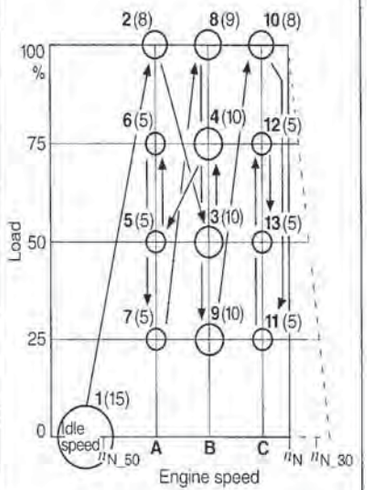
Japan
1 EIW >1265 kg,
2 EIW <1265 kg,
3 IDI,
4 DI.



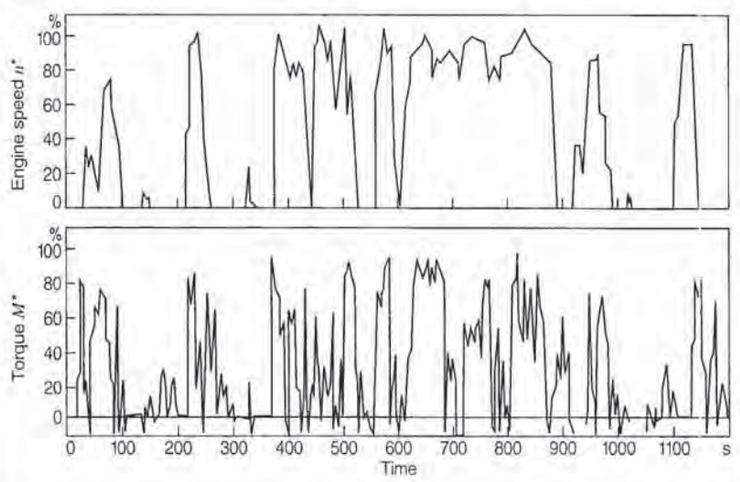
13-stage test ECE R.49 (Europe)
Order (bold) and Weighting
(% in parentheses) for test stages 1...13.



13-stage test EEC-ESC (Europe)
Order (bold) and Weighting
(% in parentheses) for test stages 1...13
$n_{N\_}$ *Rated speed (Rated output P_N),*
n_{N_30} *Engine speed at 30 % below P_N,*
n_{N_50} *Engine speed at 50 % below P_N.*



USA transient driving cycle for emissions testing of heavy commercial vehicle engines
Both the nominal engine speed n^* and the nominal torque M^* are taken from tables specified by legislation.



is monitored in a further test (Federal Smoke Cycle) under transient and quasi-steady-state operating conditions. California has different emission limits.

Smoke Emission Test Equipment

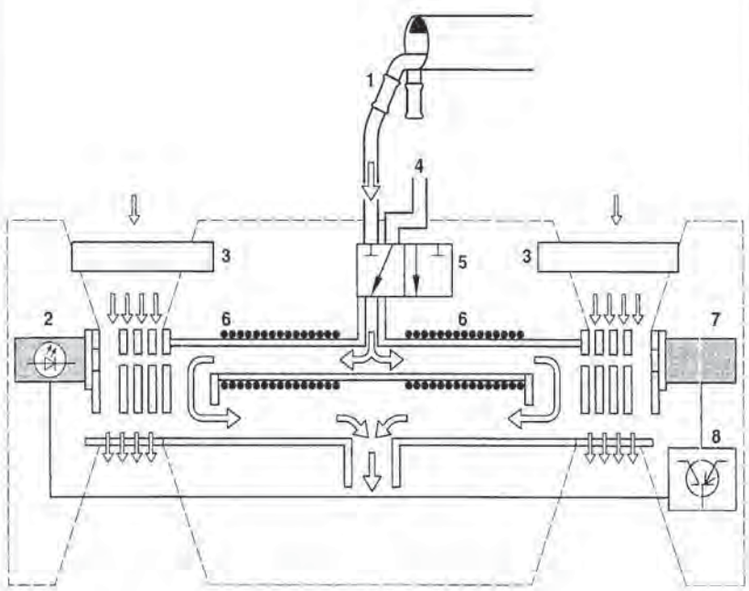
An important reason for testing diesel-engine exhaust emissions in the workshop, and for monitoring them while the vehicle is actually in operation on the road, is to determine the amount of soot (particulates) produced. This is indicated by the smoke number. Two standard procedures are used:

- In the filter method (measurement of reflected light), a specified quantity of exhaust gas is drawn through a filter element. The degree of filter discoloration then provides an indication of the amount of soot contained in the exhaust gas.
- In the absorption method (measurement of opacity), the opacity of the exhaust gas is indicated by the degree to which it blocks the passage of a beam of light which shines through it.

Measurement of diesel-engine smoke emissions is relevant only if the engine is under load, since it is only when the engine is operated under load that emissions of significant levels of particulates occur. Here, as well, two different test procedures are in common use:

- Measurement under full load, e. g. on a chassis dynamometer or over a specified test course, against the vehicle brakes.
- Measurements made under conditions of unhindered acceleration with a defined amount of accelerator pedal depression and load applied by the flywheel mass of the accelerating engine.

Smoke tester (opacity measurement)
1 Exhaust-sample probe, 2 Green LED, 3 Fan, 4 Purge air for calibration, 5 Calibrating valve, 6 Heater, 7 Receiver, 8 Electronic analyzer and display
→ Exhaust-gas path



As the results of testing for diesel smoke emissions vary according to both test procedure and type of load, they are not generally suitable for direct mutual comparisons.

Smoke tester (opacity measurement)
During unhindered acceleration, some of the exhaust gas is routed from the vehicle exhaust pipe via an exhaust-sample probe and sampling hose to the measuring chamber (without vacuum assistance). In particular, since temperature and pressure are controlled (Hartridge unit), this method prevents the test results from being affected by exhaust-gas back pressure and its fluctuation.

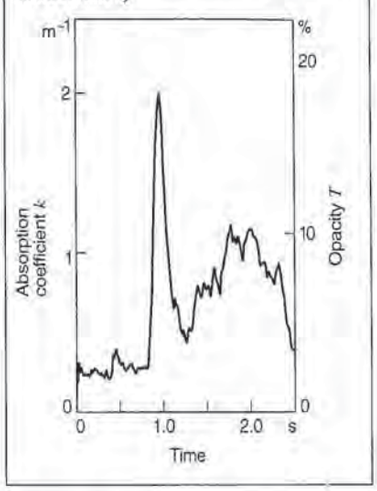
Inside the measuring chamber the diesel exhaust gas is penetrated by a beam of light. The attenuation of the light is measured photoelectrically, and is indicated as % opacity T or as a coefficient of absorption k. A high degree of accuracy and reproducibility of the test results requires that the length of the measuring chamber be precisely defined, and that the optical windows be kept free of soot (by air curtains, i.e. tangential air flows).

During testing under load, measurement and display are a continuous process. In the case of unhindered acceleration the entire test curve can be digitally stored. The tester automatically determines the maximum value and calculates the mean from several gas pulses.

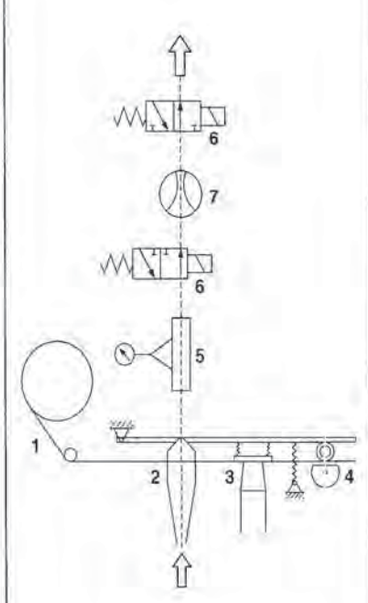
Smoke tester (filter method)
The smoke tester extracts a specified quantity (e.g. 0.1 or 1 l) of diesel exhaust gas through a strip of filter paper. Consistent, mutually comparable test results are achieved by recording the volume of gas processed in each test step; the device converts the results to a standardized form. The system also takes into account the effect of pressure and temperature, as well as the dead volume between the exhaust-sample probe and the filter paper.

The darkened filter paper is analyzed optoelectronically using a reflective photometer. The results are usually displayed as a Bosch smoke number or as a mass concentration (mg/m³).

Gas pulse measurement (opacity measurement)



Smoke tester (filter method)
1 Filter paper, 2 Gas passage, 3 Reflective photometer, 4 Paper transport, 5 Volume measurement, 6 Changeover valve for purge air, 7 Pump.



Start-assist systems

The colder the diesel engine, the more reluctant it is to start. Higher levels of internal friction combine with blowby and thermal losses to reduce compression pressures and temperatures to such a degree as to render starting impossible without the assistance of auxiliary start-assist devices. The individual temperature threshold for starting depends upon the specific engine design. Direct-injection (DI) engines, with their single-section combustion chambers and relatively low thermal losses, start more readily than prechamber or whirl-chamber engines (two-section combustion chamber) in which glow plugs are installed which extend into the secondary chamber. On DI engines the glow element extends into the main combustion chamber.

On large-displacement DI engines a flame plug or heater is employed to preheat the air in the intake tract.

Sheathed-element glow plugs

Design and characteristics

The main component in the sheathed-element glow plug is the tubular heating element. Firm, gas-tight installation in the glow-plug shell ensures that it can resist both corrosion and hot gases. The element contains a spiral filament embedded in magnesium-oxide powder. This spiral filament consists of several elements. The two resistor elements are installed in series. The tip-mounted heater coil maintains virtually constant electrical resistance regardless of temperature, while the control coil consists of a material with a positive temperature coefficient.

Circuit continuity is provided by welding the ground side of the heater coil to the inner tip of the glow tube, and by connecting the control coil to the terminal screw. The terminal screw, in turn, connects the glow plug to the vehicle's electrical system.

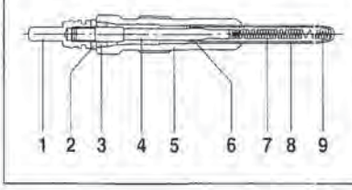
Operation

The glow plug responds to the initial application of voltage by converting most of the electrical energy into heat within the heater coil, producing a radical increase in the tip's temperature. The control coil heats more slowly. The resulting delayed rise in resistance reduces current draw and overall heat generation within the glow plug as it approaches its continuous-operation temperature. Individual heating patterns are defined by component dimensions.

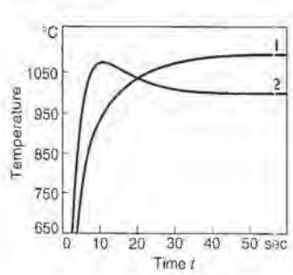
Start phase: Here the glow plug must heat to starting temperature (approx. 850 °C) as rapidly as possible. Plug locations within the combustion chamber are selected to ensure access to an ignitable mixture. Modern glow plugs heat to the required temperature in roughly 4 seconds.

Post-start phase: The glow plugs continue to operate briefly after the engine has started, improving initial engine operation while reducing blue-smoke emissions and combustion noise. The periods involved are no more than 180 s.

GSK2 sheathed-element glow plug
1 Terminal, 2 Insulator shim, 3 Double seal, 4 Terminal pin, 5 Casing, 6 Element seal, 7 Heater and control coils, 8 Glow tube, 9 Powder.

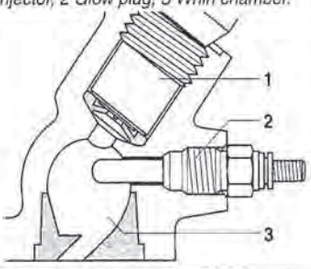


Temperature of sheathed-element glow plug in stationary air as function of time
1 S-RSK, 2 GSK2.



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Position of glow plug in the whirl chamber
1 Injector, 2 Glow plug, 3 Whirl chamber.



GSK2 sheathed-element glow plugs

Nickel control coils are used in conventional S-RSK sheathed-element glow plugs, while second-generation GSK2 glow plugs feature coils in CoFe alloy. These plugs reach ignition temperature more quickly and have a lower steady-state temperature. As a result, the preheating time prior to starting is shorter and the afterglow phase becomes possible.

Internal engine temperatures

The temperature at the glow plug changes according to the engine's operating mode. The examples shown illustrate the steady-state operating temperatures at a 13.5 V glow plug in a direct-injection engine. The maximum temperatures occur at low revolutions and high load (low air throughput resulting in less efficient cooling of the glow plug). In contrast, the highest temperatures in prechamber/whirl-chamber engines occur during operation at high loads and high min^{-1}.

Glow-control unit

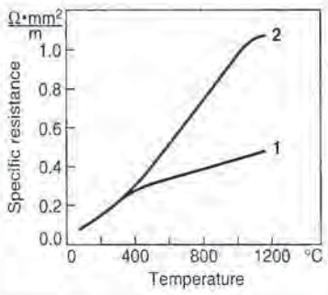
The complete glow-plug system incorporates not only the glow plugs themselves (only in rare cases are flame plugs used) but also a switching element for the high electrical glow-plug currents which is triggered by a special control unit. In addition, the system incorporates an indicator lamp (also controlled by the control unit) for signaling when the system is ready for engine start.

In the past, simple bi-metal switches were used but nowadays glow-plug systems have electronic control units. On more basic vehicles, independent glow-plug control units which handle all control and display functions are used. On modern vehicles, these functions are controlled by the central engine management system. Such units also perform safety and monitoring functions.

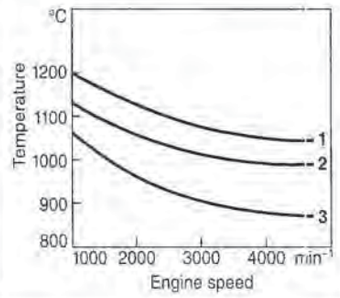
Design

The glow-control unit essentially consists of a power relay to regulate the glow-plug current, a printed-circuit board with the electronic circuitry to control glow times and triggering of the ready-to-start indicator, and the elements for the protective functions. The later generations of control units increasingly use semiconductor switches (Power MOSFET) instead of the electromechanical relay. The unit is enclosed in a plastic housing for protection against dust and water (this applies especially when it is installed in the engine compartment).

Specific control-coil resistance as function of temperature
1 S-RSK, 2 GSK2.

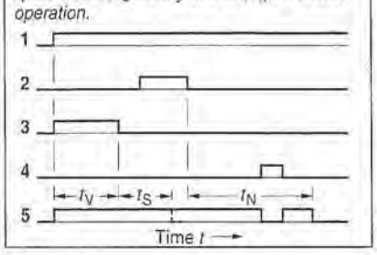


Glow-plug temperatures in DI engine
1 Full load, 2 50 % load, 3 No load.



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Typical glow sequence
1 Glow-start switch, 2 Starter, 3 Start-indicator lamp, 4 Load switch, 5 Glow plugs. t_V Glow time, t_S Ready to start, t_N Post-start operation.



Operation

The preheating and starting sequence is initiated by the glow-plug and starter switch in a similar manner to starting a gasoline engine. The preheating phase begins when the key is turned to "Ignition On".

Independent glow control units

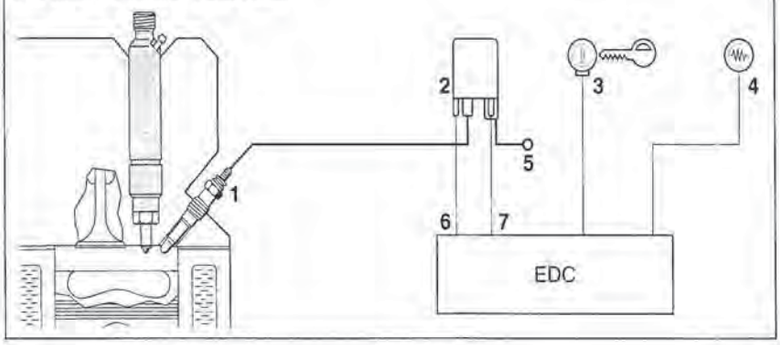
On basic units, a temperature sensor in the glow control unit controls the preheating period. This is designed to suit the specific requirements of the particular combination of engine and glow-plug so that the glow plug can reach the temperature necessary for efficient starting. At the end of the glow period, the start-indicator lamp goes out to signal that the engine can be started. The glow process continues for as long as the starter remains in operation, or until the safety override comes into effect (this limits the loads on battery and glow plugs). A strip fuse provides protection against short circuits.

With more sophisticated independent control units, an engine-temperature sensor (coolant NTC sensor) determines the heating periods more precisely. The glow control unit takes account of differences in battery voltage by adjusting the preheating period accordingly. Current continues to flow through the glow plugs once the engine has started. An engine-load monitor is used to interrupt or switch off the glow process. Protection against overvoltage and short circuits is provided by an electronic override circuit. A monitoring circuit detects glow-plug failure and relay errors. These are then displayed using the start-indicator lamp.

EDC-controlled glow control units

This type of unit receives informat on on when glow-plug operation is required, and when not, directly from the engine's central ECU. That unit provides (statically or by means of a serial data protocol) the information relating to the engine operating status (coolant temperature, engine speed and load) that is required for optimum control of the glow plugs. Similarly, the glow control unit also signals any faults it detects to the engine ECU via a diagnosis line or the serial interface. There, they are stored for servicing purposes or displayed if necessary for compliance with OBD requirements.

EDC-controlled glow-plug system on DI engine
1 Glow plug, 2 Glow control unit, 3 Glow-plug and starter switch, 4 Preheating indicator, 5 To battery, 6 Control, 7 Diagnosis.



Starting systems

Most starting systems for IC engines comprise a battery-driven DC motor (the actual starter motor), switchgear, control units and the associated wiring. The speeds at which an engine has to be cranked in order to start it (gasoline engines: approx. 60...100 min^{-1}; diesel engines: approx. 80...200 min^{-1}) are substantially lower than the speed at which the starter turns. The different speeds are matched to one another by the use of a suitable gearing ratio (between about $^1/_{10}$ and $^1/_{20}$) between the starter pinion and the engine's flywheel ring gear.

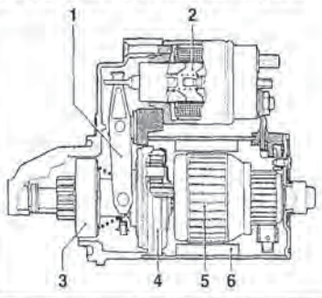
Design factors

The crankshaft torque and the minimum rotational speed required to start the engine depend among other things upon engine type, engine swept volume, number of cylinders, compression, bearing friction, additional loads driven by the engine, the fuel-management system, engine oil and temperature.

Both the torque and the rotational speed required for starting increase with declining temperatures, with the result that starter output must increase accordingly.

The battery internal resistance increases as its temperature and its state of charge drop. The battery no-load voltage decreases as temperature drops. The higher the battery output current and the lower the temperature, the lower is the battery terminal voltage. In addition, battery capacity decreases as temperature decreases and the battery discharge current increases. Therefore, the lower the temperature, the less power the battery can supply. In addition to the engine design characteristics, the minimum temperature at which the engine will start is a central factor in determining starting-system power.

Permanent-magnet reduction-gear starter
1 Engaging shift lever, 2 Engagement solenoid and solenoid switch, 3 Overrunning clutch with pinion, 4 Reduction gear (planetary gear), 5 Armature, 6 Permanent magnets.



Starters

A starter consists of an electric motor, a solenoid switch and a pinion-engaging drive mechanism whose functions are defined by the engine-starting sequence.

Initially, the pinion must engage the ring gear. As the engine starts and runs up to speed, it spins the pinion faster than the starter does and would eventually destroy the starter due to centrifugal force.

To prevent this, an overrunning clutch is installed between pinion and armature shaft which breaks the connection between them as soon as the engine "overtakes" the starter.

Electric motor

The advances made in present-day ferrite technology have enabled the development of permanent-magnet-energized starter motors which are proof against demagnetization. The form of energization is now virtually the standard design for car starters.

Reductions in starter weight and dimensions are achieved by operating the starter at higher speeds and lower armature torques. For such a starter to become a practical proposition, the crankshaft/starter-armature gear ratio must be increased. Since the ring-gear diameter cannot be increased, the higher gear ratio is achieved through the use of an additional transmission stage incorporated in the starter (reduction-gear starters)

On larger gasoline engines and passenger-car diesel engines, by far the most common type of starter motor is the permanent-magnet-energized reduction-gear design. For starter power in the 2.2 kW to 4 kW range, series-wound reduction-gear starters are beginning to supersede direct-drive versions.

Among the higher-power starters, as well as the direct-drive versions with a series-wound motor, there are also shunt-

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wound designs which provide smoother startup characteristics and limitation of the armature no-load speed.

Pinion-engaging systems

Inertia-drive starters

The inertia drive (as employed, e.g., in lawn mowers) is the simplest form of pinion-engaging drive. A helical spline in the shaft slides the overrunning clutch forward when the armature rotates.

When the starter is switched on, the unloaded armature begins to rotate freely. The pinion and overrunning clutch do not yet rotate due to their inertia, and are pushed forward by the spline. As soon as the pinion makes contact with the ring gear, it is kept from rotating and pushed further forward until making contact with the stop ring. At this point, the overrunning clutch begins to transmit the armature torque to the ring gear via the pinion, and the engine is cranked.

As soon as the engine begins to rotate the pinion at a speed above armature speed, although the overrunning clutch interrupts the transmission of force, the friction of the overrunning clutch attempts to accelerate it above armature speed. This causes the overrunning clutch and pinion to slide backward in the helical spline. This pinion-disengaging operation is assisted by the return spring which also holds the pinion in the disengaged position when the starter is not running.

Pre-engaged-drive starters

In pre-engaged-drive starters, an engagement solenoid engages the pinion with the ring gear.

The engagement solenoid incorporates the switching contacts for the starter current.

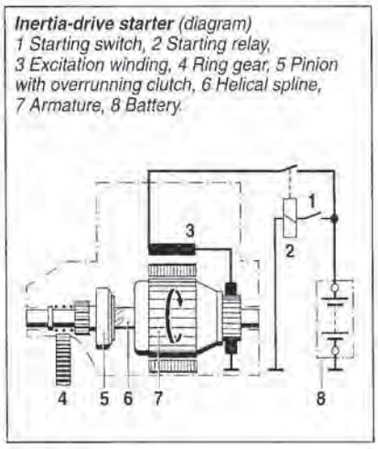
When the starting switch is closed, the hold-in winding H is energized, and current flows through the series circuit comprising pull-in winding E and electric motor. The starter motor solenoid picks up and moves the overrunning clutch and pinion gear assembly forwards by means of the thrust lever and the engaging spring.

If the pinion and ring gear happen to find themselves in mutually optimal positions, then the pinion will immediately mesh with a gap in the ring gear's teeth. In this case the pinion teeth's engagement depth in the ring gear increases until the travel limit is reached and the contact bridge in the solenoid switch hits the relay contacts; at this point the full voltage is applied to the starter motor.

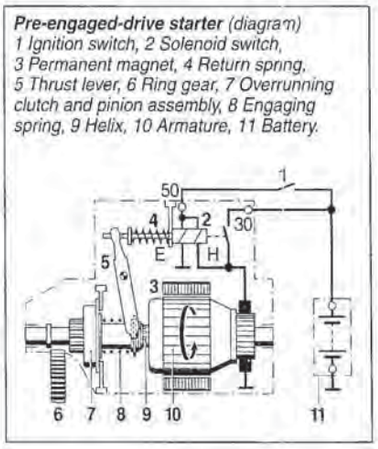
If the pinion teeth do not immediately mesh with the ring-gear gaps, the ring gear prevents the pinion from advancing any further. The engaging lever then compresses the meshing spring, and the main contact closes without the pinion having engaged.

The electric motor then turns the pinion which is in contact with the face of the ring gear until a pinion tooth finds a ring-gear gap and the meshing spring pushes the pinion and overrunning clutch forward.

When the solenoid winding is de-ener-



Inertia-drive starter *(diagram)*
1 Starting switch, 2 Starting relay, 3 Excitation winding, 4 Ring gear, 5 Pinion with overrunning clutch, 6 Helical spline, 7 Armature, 8 Battery.



Pre-engaged-drive starter *(diagram)*
1 Ignition switch, 2 Solenoid switch, 3 Permanent magnet, 4 Return spring, 5 Thrust lever, 6 Ring gear, 7 Overrunning clutch and pinion assembly, 8 Engaging spring, 9 Helix, 10 Armature, 11 Battery.

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gized, the return spring pushes the solenoid plunger and pinion with overrunning clutch back into the rest position.

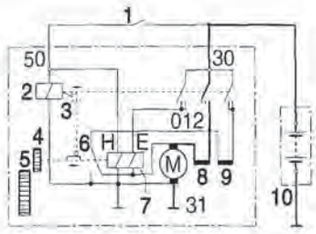
This disengaging operation is assisted by the helix when the engine speed overtakes the starter speed.

Sliding-gear starters

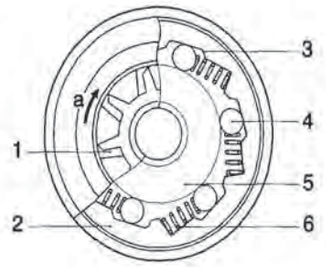
The sliding-gear drive switches the starter on in two stages.

When the starting switch is closed, battery voltage is applied across the hold-in winding H of the engagement solenoid and the control relay in parallel. The control relay picks up, but is held in contact position 1 (first stage) by a tripping lever and latch. Battery voltage is applied to the pull-in winding E of the engagement solenoid and the shunt winding of the motor which are connected in parallel with one another and in series with the armature. The starter begins to rotate, however, it can only generate low torque due to the high winding resistances in series with the armature winding. At the same time, the engagement solenoid pushes the pinion in the direction of the ring gear, so that it engages at low torque. Shortly before the end of meshing travel is reached, the engagement solenoid releases the latched control relay which immediately moves into contact position 2 (second stage). The starting current now flows through the series winding and the armature. The changeover contact on the engagement solenoid connects the shunt winding parallel to the armature and the series winding. The starter develops full torque.

Sliding-gear starter *(circuit diagram)*
1 Starting switch, 2 Control relay, 3 Tripping lever, 4 Pinion, 5 Ring gear, 6 Changeover contact, 7 Engagement solenoid, 8 Series winding, 9 Shunt winding, 10 Battery.



Roller-type overrunning clutch
1 Pinion, 2 Clutch shell, 3 Roller race, 4 Roller, 5 Pinion shaft, 6 Spring.
a Direction of rotation.



Overrunning clutch types

Overrunning clutches protect the starter armature from excessive speed when "overtaken" by the engine.

Roller-type overrunning clutch

Small and medium-sized starters generally have an overrunning clutch in which rollers are pushed by springs into wedge-shaped recesses between the clutch shell and the pinion shaft. When torque is applied in the opposite direction, the rollers are forced loose and the pinion is decoupled from the starter.

Multiplate overrunning clutch

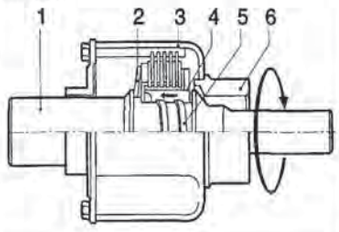
The multiplate overrunning clutch is used in large commercial-vehicle starters. The driver with the outer plates and the starter armature, and on the other side the driving shaft and the pinion are positively connected to one another. The inner plates fit in an inner clutch race which rides axially in a helical spline in the driver shaft. Under no-load conditions the plate stack is lightly compressed by the compression spring, and can only transmit low torque. As load increases, the inner clutch race is moved by the helical spline in the direction of the compression spring, which is thus more strongly compressed and presses the plates more tightly together. The multiplate overrunning clutch is therefore able

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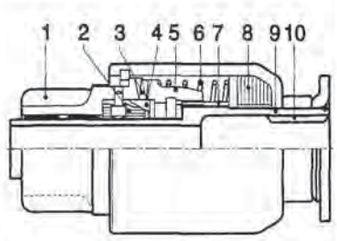
to transmit increasing torque as the starter load increases.

Radial-tooth overrunning clutch
The radial-tooth overrunning clutch is also used in large commercial-vehicle starters. The entire clutch system is coupled to the armature shaft, on which it slides axially (meshing), by means of spur toothing in the dirt sleeve. The outer surface of the dirt sleeve has a helical spline, and transmits the torque to a clutch nut which further transmits the torque to the pinion by means of the steep flanks of the saw-tooth-shaped radial teeth. During overrun, the pinion pushes the clutch nut backwards by means of the shallow flanks of the radial teeth, and interrupts the transmission of force. The disengaging ring is also moved backward, and held in the disengaging position by the flyweights. The centrifugal force developed by the flyweights at low pinion speeds no longer suffices to keep the overrunning clutch in its disengaged position, and the spring again pushes the clutch nut into the pinion.

Multiplate overrunning clutch
1 Drive shaft (connected to pinion), 2 Compression spring, 3 Driver with outer plates, 4 Inner clutch race with inner plates, 5 Helical spline, 6 Drive end (connected to armature).



Radial-tooth overrunning clutch
1 Pinion, 2 Centrifugal weight, 3 Spur gear, 4 Thrust ring, 5 Clutch nut, 6 Spring, 7 Helix, 8 Damper, 9 Dirt sleeve, 10 Straight gear.



Starter protection

Starter power is also a function of the size of the vehicle battery. The thermal loading of the current-carrying parts and the mechanical loading of the torque transmission parts increase as battery size increases. It is for this reason that a maximum permissible battery size is usually specified for each starter type. Although the starter is designed to operate for only short periods of time, its design must allow for longer cranking times and thus increased thermal load at low temperatures.

This is why extended periods of starter operation should be followed by cooling-off periods.

In the case of larger starters, excessively long operating times are prevented by built-in thermo-switches (e.g., in the carbon brushes). In remote-control starting systems (e.g., rear-engine buses, emergency power generator sets, diesel railroad cars, etc.) on the other hand, the starting procedure cannot always be monitored by the driver.

Operator errors can damage the starter or ring gear.

Start-locking relays
These prevent inadvertent engagement of the starter when the engine is running or excessive overrunning after the engine has started. The rise in alternator voltage that occurs when the engine is running is used as the variable for indicating that the engine has started. As the engine slows down after the ignition switch is turned off, the alternator no longer produces a usable voltage "signal"; in this case a timer in the start-locking relay blocks for a few seconds any repeated attempt to use the starting system.

Start-repeating relays
Start-repeating relays interrupt the starting procedure if the pinion is still unable to engage the ring gear, but the starter remains switched on.

In this way, they prevent overloading of the solenoid switch winding.

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Electric drives

The electric drive is quiet, produces no exhaust emissions and is very efficient. Whereas in purely electrically powered vehicles the electric drive alone powers the wheels, hybrid vehicles have at least two different sources of drive energy, at least one of which is usually an electric drive.

In contrast to internal-combustion engine vehicles, on electric-only vehicles the energy accumulator generally determines the vehicle's performance. The capacity of the electric motor is matched to the maximum output of the energy accumulator. The energy accumulator may take the form of an electrochemical battery or a fuel cell and its associated fuel tank.

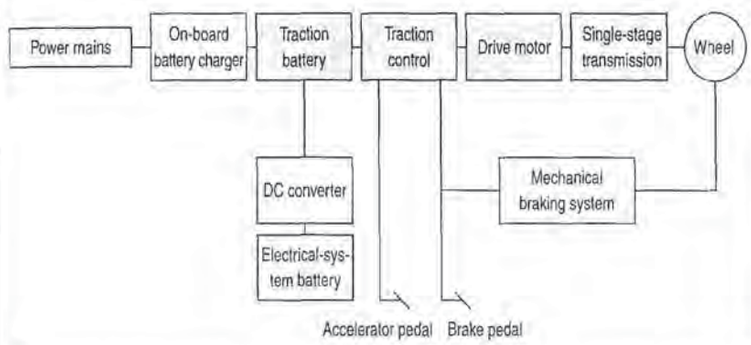
Depending on the intended application, battery-powered electric vehicles can be classified as either road vehicles or industrial trucks. Industrial trucks are used for transporting goods on company premises, and are generally not licensed for use on public roads. Their top speed is below 50 km/h. Due to the low power density of the batteries, the range of battery-powered on-road vehicles is significantly less than that of vehicles powered by internal-combustion engines. The maximum speed of such vehicles is also normally limited to around 130 km/h. Whereas more than half of all new industrial trucks are electrically powered, the percentage of electrically powered on-road vehicles s very low.

Energy supply

There is no shortage of power for electric vehicles that are recharged by plugging them into a wall socket. If electric vehicles in Germany were largely recharged at night, existing power plants could provide enough energy to charge more than 10 million vehicles. That number of electric vehicles would require less than 5 % of Germany's total electricity output.

Any household power outlet can be used to charge the batteries. However, these outlets can provide only 3.7 kW of electrical power, which means that an hour of charging would provide enough power to drive a distance of no more than about 20 km. Shorter recharging periods can be achieved by using a three-phase AC power source (as for industrial trucks). Compared to the refueling times for diesel vehicles in particular, the recharging periods required by comparable electric vehicles in order to cover the same distances are roughly 100 times longer, even in the case of very high charging capacity.

Electric drive unit in a battery-powered electric on-road vehicle *(block diagram)*



Batteries

Whereas costs considerations dictate that the lead-acid battery is the power source used most often in industrial trucks, in modern electric cars it is increasingly being replaced by nickel and lithium batteries.

Lead-acid battery

Although the basic design of the lead-acid battery is the same as that of the starter battery (see "Starter Battery", P. 866), the combinations of materials and the cell design are specially adapted to the particular requirements of traction operation. The batteries commonly used in industrial trucks are generally combinations of individual cells, whereas a modular design with 3 or 6 cells per module is used in most electric on-road vehicles due to the higher energy density.

Industrial trucks generally use lead-acid batteries with a liquid electrolyte which must be topped up with water on a regular basis. In the case of electric on-road vehicles, this level of maintenance is not acceptable for the vehicle's user. Consequently, maintenance-free batteries with a solid electrolyte (gel) have become standard in these applications. Under real-world conditions, vehicles equipped with lead-acid batteries have a range of 50...70 km per battery charge in city driving. The daily range of an electric vehicle can be increased through intermediate charging of the batteries when the vehicle is parked.

The amount of energy which can be drawn from a lead-acid battery decreases as battery temperature drops. This means that a battery heating system is required by electric on-road vehicles in some climates in order to prevent a reduction in vehicle range during the winter months.

Because the electrolyte takes part in the chemical reaction inside a lead-acid battery, the available capacity varies as a function of discharge time. For example, if the discharge time is reduced from two hours to one, the available capacity is reduced by roughly 20 %.

Batteries in industrial-truck applications can achieve service lives of 7...8 years with 1200...1500 cycles. Fleet experience with electric passenger cars indicates that lead-acid batteries can be expected to last for around 5 years and roughly 700 cycles. The shorter service life in electric on-road vehicle applications is primarily a result of the much greater battery load. In these vehicles the battery is discharged in an average of 2 hours or less, whereas discharge times in industrial trucks are generally in the 7...8 hour range.

Battery systems

Properties	**Lead-acid system** open/sealed	**Nickel systems** Nickel-cadmium (Ni/Cd) Nickel-metal hydride (NiMH)	**Lithium systems** Lithium-ion Lithium-polymer
Cell voltage	2 V	1.2 V	3...4 V
Energy density	25...30 Wh/kg	35... 100 Wh/kg	60...150 Wh/kg
Energy efficiency without heating/cooling	75...85 %	60...85 %	85...90 %
Power density	100...200 W/kg	100...500 W/kg	300...1500 W/kg
Service life in cycles	600...900	> 1000	> 1000 projected
Operating temperature	10...55 °C	–20...55 °C	–10...50 or 60 °C
Maintenance-free	Depending on design	Depending on design	yes

Commercially available vehicles (examples)

Vehicle type	Type of battery	Engine power	Acceleration 0...50 km/h	Maximum speed	Typical range per charge	Typical line-power consumption
Passenger car	Ni/Cd	21 kW	9 s	90 km/h	80 km	18 kWh/100 km
Passenger car	NiMH	49 kW	7 s	130 km/h	200 km	26 kWh/100 km
Passenger car	Lithium-ion	62 kW	6 s	120 km/h	200 km	23 kWh/100 km
Van	Lead-acid	80 kW	7 s	120 km/h	90 km	35 kWh/100 km

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Nickel-based batteries

Nickel-cadmium batteries and, increasingly, nickel-metal hydride batteries with an alkaline electrolyte are now used in many electrical appliances. Because cadmium is harmful to the environment, it is likely that in the foreseeable future the nickel-cadmium system will be replaced by the nickel-metal hydride system. Whereas electrical appliances normally use sealed batteries, in traction applications open nickel-cadmium cells are often used. These cells, like open lead-acid batteries, must be refilled with water at regular intervals. Nickel-metal hydride batteries must be sealed due to the inherent characteristics of the system. The low cell voltage of only 1.2 V means that the relative density of non-active components in a 6 V nickel-cadmium module, for example, must be higher than in a lead-acid battery. A battery service life of up to 10 years or 2000 cycles has been demonstrated in a number of applications. The higher costs resulting from the use of relatively expensive materials and the complex manufacturing process are partially offset by a much longer service life than that of lead-acid batteries.

Nickel-cadmium and nickel-metal hydride batteries are cooled when used in electric on-road vehicles; heating is required only at temperatures below −20 °C. Available capacity is virtually independent of discharge time. The alkaline battery's higher energy density can be exploited both to increase the payload and to extend the vehicle's radius of action. Electric cars typically have a range of approx. 80...100 km using nickel-cadmium batteries.

In the nickel-metal hydride system, cadmium is replaced by hydrogen. This hydrogen must be stored in a multimetal medium. The nickel-metal hydride battery has a higher energy density and a somewhat longer service life than the nickel-cadmium battery. Nickel-based batteries also have a higher power density, making them particularly interesting for use in hybrid vehicles.

Lithium-based batteries

Lithium systems allow energy densities of over 100 Wh/kg and power densities of over 300 W/kg in vehicle traction batteries. They can be operated at ambient temperature or slightly higher temperatures, and are characterized by high cell voltages of over 4 V. In the demanding electrical-appliance battery market (for products such as laptops and video recorders) the lithium-ion system has already become successfully established.

Lithium systems do not show any memory effect as do nickel-cadmium systems. A disadvantage of the lithium batteries is that they require a relatively complex battery protection system. For example, each individual cell must be monitored because they are not proof against overcharging. In order to prevent endangerment of the environment, these batteries must also be specially protected against short circuits.

Lithium-ion battery

A lithium-ion battery stores lithium ions in electrically reversible form on the negative electrode in a graphite lattice. The positive electrode contains cobalt as the main component, along with lithium making the system rather expensive. Attempts are thus being made to use more economical materials such as manganese.

Organic material is used as the electrolyte; aqueous electrolytes cannot be used because lithium reacts strongly with water.

Lithium-polymer battery

Another very promising lithium system is the lithium polymer battery. It consists of a thin lithium film, a polymer electrolyte and a positive film electrode made primarily of either vanadium or manganese. Individual cells are formed by rolling or folding the film, which has an overall thickness of approx. 0.1 mm. The working temperature is approx. 60 °C.

Drivetrains

The drivetrain in an electric vehicle generally consists of the power controller, the motor and the transmission. The power controller translates the position of the accelerator pedal into the appropriate motor current and voltage. In most cases the drive torque is a function of the accelera-

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tor pedal position, as in the case of IC engines.

The cost of the motor depends largely on the required maximum torque; thus it is advantageous to use the highest-possible step-down gear ratio between the engine and the drive wheels. The step-down gearing may consist of one or two stages, depending on the desired hill-climbing ability and the vehicle's maximum speed for the given maximum torque and variable speed range of the drive train. Modern electric cars have single-stage reduction gears.

A difference between electric drive units and combustion engines is the necessary distinction between short-term and extended-duty performance. Short-term performance is usually limited by the maximum setting of the power controller. The maximum power available over longer periods is defined by the half-hourly output in the case of on-road electric vehicles, which is generally limited by the permissible motor temperature. This distinction also applies to most batteries. Depending on the type of drive, short-term and extended-duty ratings vary by a factor of 1 to 3. Maximum drive power must therefore be monitored and adjusted, if necessary, in accordance with the characteristic thermal limits of the power controller, motor or batteries.

This distinction between short-term and half-hourly operation has among other things also led to the adoption of two maximum-speed ratings for electric on-road vehicles: maximum speed over a distance of 2x1 kilometer and maximum speed over a period of 30 minutes.

Series-wound direct-current drive

This type of drive unit has the simplest type of power controller. The motor voltage is set in accordance with the desired current by applying the battery voltage to the motor in a variable on/off ratio and/or chopper frequency by means of a circuit breaker (thyristor or transistor(s)).

For the recovery of braking energy, the power controller must operate as a step-up chopper, which means that additional components are needed. Because the field and armature of the motor are in series, drive power drops in proportion to the square of the motor speed with the full battery voltage applied.

Although its efficiency is relatively low, this type of drive is still used in most industrial trucks today because of its simple design and low cost. The low top speeds of these vehicles make it possible to use single-stage reduction gears.

Separately-excited direct-current drive

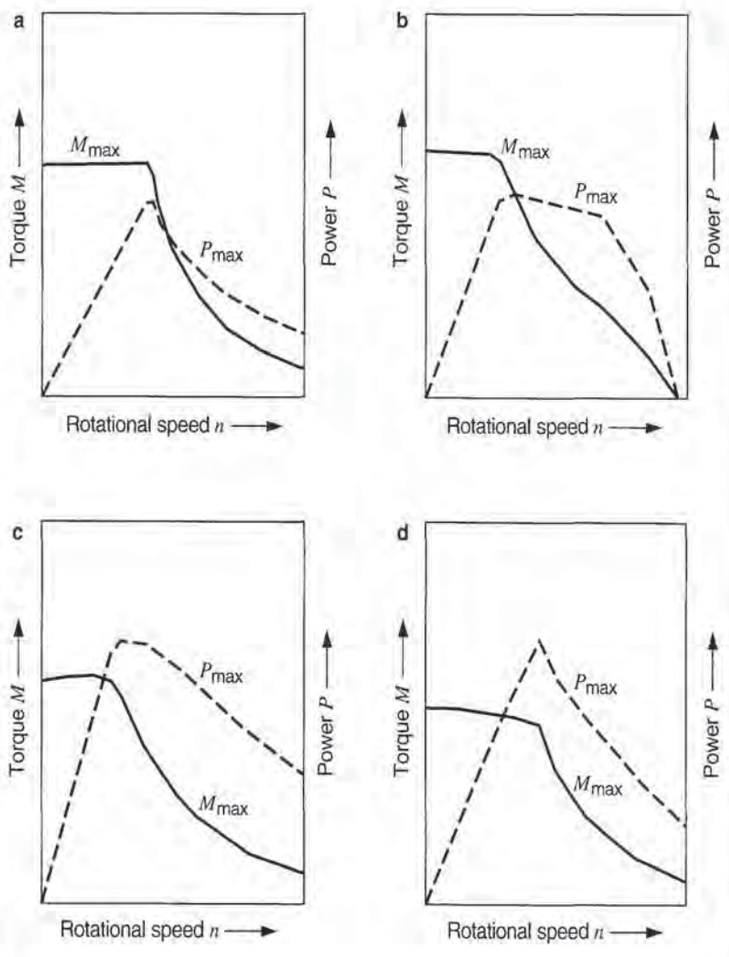
In this type of drive unit, the motor's magnetic excitation is provided by its own controller (field rheostat). Depending on the size of the motor, the field can be weakened in a ratio of up to approx. 1 4. Field strength starts to diminish at a nominal motor speed obtained with full motor voltage at the armature and maximum field current. During initial acceleration with maximum field current an electronic armature-control device limits the motor current until the motor reaches its nominal speed, with the full motor voltage applied to the armature. In the reduced field-strength range, consistent armature currents produce relatively constant power outputs. Because commutation becomes more difficult as the field current drops, the armature current must usually be reduced before the maximum speed is reached.

Because commutating poles are required, this design is somewhat more complex than that of a series-wound motor. The mechanical commutator limits rotational speed to roughly 7000 min^{-1}. This type of drive unit can be used with a multi-stage transmission to reduce motor cost and weight. Efficient energy recovery during braking is possible without requiring additional components.

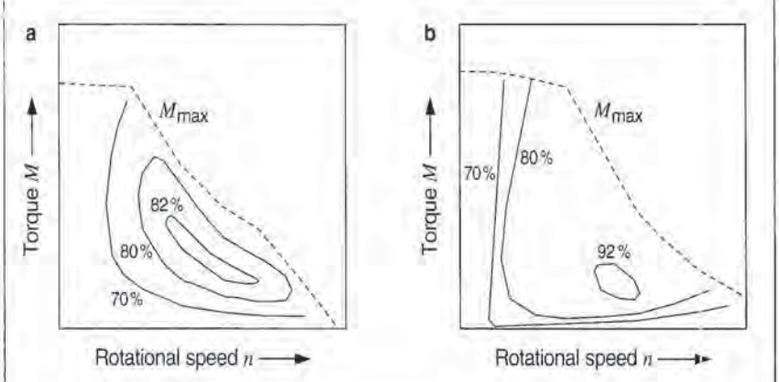
However, very few electric cars today are being equipped with direct-current drive units. Three-phase AC asynchronous or synchronous drives are now the norm, due in part to their low maintenance requirements. The carbon brushes in DC motors must be regularly replaced, albeit at relatively long intervals.

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Torque and power as a function of rotational speed for various types of drive
a) Series-wound DC drive,
b) Separately-excited DC drive,
c) Asynchronous drive,
d) Permanently-excited synchronous drive.
M_{max} *Maximum torque,*
P_{max} *Maximum power.*



Efficiency curves
a) Separately-excited DC drive, b) Permanently-excited synchronous drive.
M_{max} Maximum torque.



Asynchronous drive
The motor in an asynchronous drive unit is the simplest and most economical in design, and is also considerably smaller in size and weight than a direct-current motor. In principle, however, the controller in a three-phase drive unit is more complex than that used in DC drives. As with the separately-excited DC motor, operation with reduced field current is possible. Because these motors have no mechanical commutator, they can operate at speeds of up to 20,000 min^{-1} if appropriately designed. This means that single-stage transmissions can be used, even in on-road vehicles. These drives are more efficient than direct-current drives, but not quite as efficient as synchronous drives with permanent magnets. Braking energy can also be recovered with a high degree of efficiency.

Permanently-excited synchronous drive
This type of drive is characterized by very-high efficiencies also in the part-load range because it uses permanent magnets to generate the excitation field. Rare-earth magnets with a high energy density allow very compact dimensions combined with high torque. However, rare-earth magnets make the motor more expensive than asynchronous designs, for example. This type of motor is not capable of operation with reduced field current. Nevertheless, quasi reduced-field-current operation in which the torque-generating component is reduced by increasing the reactive-current component of the stator current enables virtually constant-output operation, so that a single-stage reduction gear is generally sufficient here as well.

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Hybrid drives

In the broadest sense, the term "hybrid drives" is used to denote vehicle drives with more than one drive source. Hybrid drives can incorporate several similar or dissimilar types of energy stores and/or power converters. The goal of hybrid-drive developments is to combine different drive components, such that the advantages of each are utilized under varying operating conditions in such a manner that the overall advantages outweigh the higher technical outlay associated with hybrid drives.

Classification of hybrid drives

In terms of available performance and range, the IC engine as a drive source is superior to all other drive systems. Its disadvantages – drop in efficiency at part load, and the generation of toxic emissions – led to the development of hybrid drives which incorporate the IC engine. IC engines in some hybrid drives are thus designed for use over the medium-power range, whereby the differences between generated power and the power required at any given time are made up by the additional mechanical or electrical energy store.

Hybrid drive with mechanical energy store

A hybrid drive incorporating an IC engine and flywheel, can be operated in a number of ways by engaging and disengaging the three couplings K1, K2 and K3. The IC engine can be directly coupled to the power-shift transmission or the flywheel. There are 2 kinetic-energy storage devices, the flywheel and the vehicle mass.

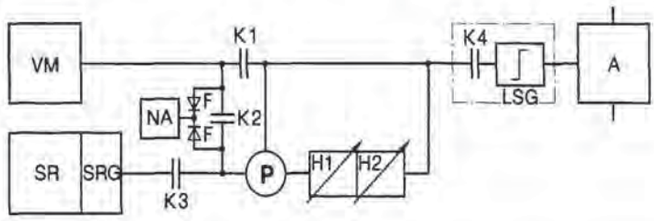
In addition, the internal-combustion engine can start the two kinetic-energy storage devices from standstill via the continuously variable transmission. This configuration also permits parallel operation with the two drive sources, diesel engine and flywheel. Here, only the power delivered by the flywheel is transmitted to the power-shift transmission via the loss-intensive continuously variable transmission using hydrostatic converters, whereas the engine power is fed directly to the transmission input.

Classification of hybrid drives
△ Research proposals, ● Prototype vehicles, ◆ Tested in practical applications

Drive source	Internal-combustion engine	●	●					◆	◆	◆
	Electric motor			●	◆	△	△	◆	◆	◆
Mechanical energy store	Flywheel	●			◆	△				◆
	Pressure accumulator		●							
Electrical energy sources	Battery			●		△	△	◆		
	Fuel cell						△			
	External supply (trolley)			●	◆				◆	
Design examples		DB MAN Gyro-bus	MAN Hydro-bus	Esslingen DUO-bus	Basel Neoplan Trolley bus			DB hybrid electric bus, MB/MAN delivery truck	Essen DUO-bus	Munich Magnet motor-bus

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Hybrid drive incorporating internal-combustion engine with flywheel
VM *IC engine,* SR *Flywheel,* SRG *Flywheel reduction gear,* NA *Auxiliary drive,* F *Freewheel,* K *Coupling,* P *Planetary gear,* H1, 2 *Continuously-variable hydrostatic transmission,* LSG *Power-shift transmission (automatic planetary gear),* A *Powered axle.*



Hybrid drive with electrical energy store

Hybrid drives without IC engines which use only electrical drive components are designed to apply their hybrid components in order to avoid the disadvantages of the purely battery-powered electric drive. The useful energy stored in the battery allows only a limited driving range, and is further reduced as power demand increases.

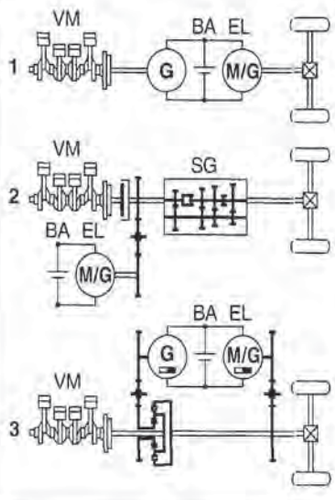
Combining a mechanical energy store with the battery prevents it being affected by power peaks, thus contributing to more efficient utilization of the battery energy. A hybrid system which uses a combination of two different electrochemical energy sources (battery and fuel cell) separates the energy sources such that one source has a high power, and the other has good energy-storage capability. In the case of hybrid electric drives with an external supply of energy (trolley systems), the vehicle's own energy store is used as a short-time storage medium for short distances of travel without the overhead power supply. This configuration reduces the high costs associated with overhead contact wires and increase versatility in traffic.

Hybrid drive designs

Hybrid drives which combine an internal-combustion (IC) engine and an electric drive are the only hybrid drives which have warranted serious attention to date. The electric drive component of such drives is powered either by an on-board battery or via an overhead contact wire and current-collector system. The diagram at the right shows the various basic drive configurations. The battery indicated in each configuration can be replaced by the external power-supply arrangement.

The main difference among the various configurations is the series, parallel or mixed interconnection of the power sources. In the series configuration (1)

Hybrid drive configurations
1 Series configuration, 2 Parallel configuration, 3 Mixed configuration VM *IC engine,* EL *Electric drive (operated as a motor or alternator/generator),* BA *Battery or external power supply,* SG *Manually shifted transmission.*



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the individual drive components are connected in series, whereas in the parallel configuration (2) the drive power of both drive sources is mechanically added. The letters M and G indicate whether the electric drive is operating in "motor" or "generator" mode.

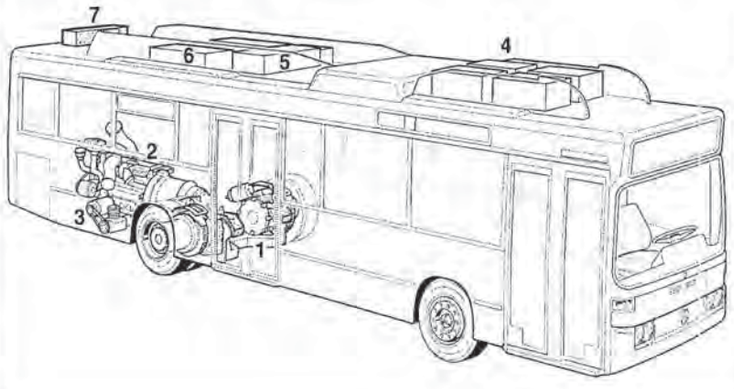
Because the diesel engine in the series configuration is mechanically decoupled from the vehicle drive, the diesel engine can be operated at a constant speed, i.e. at its optimum operating point in terms of efficiency and emissions. Despite the advantages of the series configuration, its disadvantage is that energy must be converted several times. Including battery storage efficiency, the mechanical efficiency between the diesel engine and the powered axle is hardly greater than 55%.

The parallel hybrid configuration (2) has the advantage that when operated in the mode which incorporates an IC engine, it is just as efficient as the engine in a conventional vehicle. In configuration 2, the change-speed transmission required by the diesel-engine drive is also part of the electric drive branch. In this type of drive the speed of the electric motor therefore must be varied only within a specific range above a basic speed, in a manner similar to the way in which the diesel engine is operated. Moreover, in this configuration the electric drive also profits from the torque conversion by the downstream transmission, as a result of which the electric motor must only be dimensioned for low drive torque. This leads to an equivalent reduction in motor mass which is roughly proportional to motor torque.

The mixed configuration (3) represents a combination of configurations 1 and 2, and corresponds to a splitter transmission with an infinitely-variable transmission ratio. On the one hand, the power of the IC engine is mechanically transmitted directly to the driving wheels, while on the other, the rotation of the IC engine is decoupled from the rotation of the driving wheels by the speed overlay in the planetary gear.

The drive configuration in a hybrid electric bus is the same as that in type 1 described above. In order to obtain the low floor height required, the electric drive takes the form of wheel hub motors. Due to the low floor level, the electrical energy store – a high-temperature battery – must be mounted together with its peripherals on the roof of the vehicle.

Hybrid electric bus
1 Wheel hub motors, 2 Diesel engine with alternator/generator, 3 Air compressor and power-steering pump, 4 Traction batteries, 5 Vehicle electrical system converter, 6 Inverter, 7 Braking resistor.

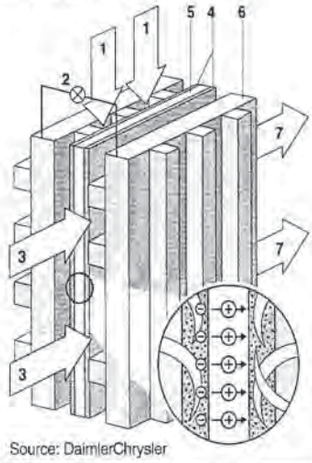


Fuel Cells

Fuel cells are electrochemical cells in which the chemical energy of a suitable fuel is continuously converted into electrical energy using atmospheric oxygen (O_2). The most common fuels which lend themselves to such applications are hydrogen (H_2), methanol (CH_3OH) and, to a more limited degree, methane (at very high temperatures). Because conventional fuels can not be used directly, they must be converted into H_2 in a chemical gas-reforming reaction. Fuel-cell operation is very efficient, and produces low levels of harmful emissions. They are modular in design, and can therefore be used over a wide power range from a few watts to several megawatts.

Due to those characteristics and promising new developments in the field of low-temperature fuel cells, many automobile manufacturers now see the fuel-cell drive as a serious alternative to the internal-combustion engine for automotive applications. For that reason, the major vehicle manufacturers in particular are working intensively on the development of fuel cells suitable for automotive use.

Polymer electrolyte fuel cell *(principle of operation)*
1 Hydrogen, 2 Electrical load, 3 Air (oxygen), 4 Catalyst, 5 Electrolyte, 6 Bipolar plate, 7 Water vapor and residual air.



Source: DaimlerChrysler

However, a realistic assessment of the fuel-cell drive in terms of its environmental and customer benefit is only possible by looking at the whole picture. As far as emissions are concerned, not only must the direct emissions from the vehicle be taken into account, but also those produced in the fuel-cell manufacturing process. The same applies to the system's efficiency, which can only be compared with other types of drive if the overall efficiency of the entire process from the primary energy source to the driving wheels is considered.

The most important application for fuel cells to date has been as a means of generating electrical energy in spacecraft and submarines.

Design variations

In contrast to combustion engines, fuel-cell operation does not require a specific (high) temperature; some fuel cells operate at room temperature, while others are designed for temperatures of up to approximately 1000 °C (Table 1). The various designs differ from one another above all in the type of electrolyte used, which depends on the temperature. Up to around 90°C the electrolyte is aqueous or contains water. For mid-range temperatures (500 ... 700 °C) molten alkaline carbonate electrolytes have become the standard, while for high temperatures (800 ... 1000 °C), only ceramic-based solid electrolytes (e.g. zirconium dioxide) can be used. Apart from the differences in the type of electrolyte used, fuel cells also differ according to their electrode materials.

Fuel cells are often referred to by the acronyms shown in Table 1.

Fuel conditioning

Although attempts have long been made to operate fuel cells directly using various fuels, the fuel cells available today must use H_2 as their energy source. At present, H_2 is generally obtained from natural gas or other fossil fuels by means of a chemical gas-reforming process. For mobile applications, H_2 must either be stored in the

vehicle or derived from another on-board fuel.

Hydrogen storage medium
For automotive applications, H_2 can be stored and transported in gaseous form in cylinders at pressures of up 300 bar or in liquid form in cryotanks at −253 °C. For low-power applications or in submarines, hydrogen is stored in metal hydrides or even in special modified carbon compounds. If H_2 is stored as a gas under pressure or as a liquid, it must be remembered that a considerable portion of the primary energy is required simply to compress or liquefy the H_2. Furthermore, the energy density of an H_2 storage medium is less than that of a conventional fuel tank.

Methanol reforming
CH_3OH is produced from natural gas with an efficiency of approx. 65%. Its advantage over H_2 is that it can be dispensed in liquid form similar to conventional vehicle fuels. However, a separate infrastructure must be made available to handle CH_3OH; it cannot be stored in existing fuel tanks because it is considerably more corrosive than gasoline or diesel fuel. CH_3OH can be converted into H_2, CO_2 and CO in a catalytic reforming process using water vapor at temperatures of 250...450 °C. The CO combines with water to produce H_2 and CO_2 in a subsequent catalytic conversion stage. The residual CO must be removed in a gas purifier because it chemically inhibits the fuel-cell electrodes.

Gasoline reforming
The advantages of gasoline are its high energy density and widespread availability through an already existing infrastructure, however it is considerably more difficult to reform gasoline into H_2 than it is to reform CH_3OH, for example. Conversion involves partial oxidation in the presence of air and water at temperatures of 800... 900 °C, producing H_2, CO_2 and CO. The CO is converted in two subsequent catalytic stages using H_2O into H_2 and CO_2. The residual CO must be separated out in a gas purifier in this case as well, because it inhibits the fuel-cell electrodes. The problems associated with gasoline reformation concern primarily the complex system which must be controlled at high temperatures, and the inhibition of the catalysts by the formation of coke.

Thermodynamics and kinetics

The electrochemical reactions which take place in fuel cells are essentially the same as those in galvanic cells (e. g. batteries), however in fuel cells only gaseous or liquid fuels are used. The oxidizing agent is generally O_2 or atmospheric oxygen. For that reason, fuel cells require special porous electrode structures.

Table 2 gives the reaction equations and the calculated fuel-cell data (theoretical cell voltage E_0 and thermodynamic efficiency η_{th}) for the two fuels of interest, H_2 and CH_3OH, at different temperatures. The first reaction equation describes the

Table 1. Types of Fuel Cells

Fuel Cell Designation		Electrolyte	Temperature °C	Cell Efficiency (Load Partial Load) %	Type of Application
Alkaline Fuel Cell	AFC	Aqueous KOH	60...90	50...60	Mobile, stationary
Polymer Electrolyte Fuel Cell	PEFC	Polymer electrolyte Membrane	50...80	50...60	Mobile, stationary
Direct Methanol Fuel Cell	DMFC		110...130	30...40	Mobile
Phosphoric Acid Fuel Cell	PAFC	H_3PO_4	160...220	55	Stationary
Molten Carbonate Fuel Cell	MCFC	Alkaline carbonates	620...660	60...65	Stationary
Solid Oxide Fuel Cell	SOFC	ZrO_2	800...1000	55...65	Stationary

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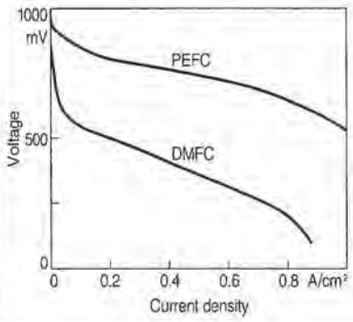
familiar electrolytic gas reaction, which in fuel cells is a controlled process ("cold combustion") rather than an explosive one. This reaction takes place in a controlled manner in fuel cells because the important reaction processes (H_2 oxidation and O_2 reduction) occur at physically separate electrodes and at a relatively low temperature (Figure and Table 1). It is therefore possible in fuel cells to directly and completely convert the chemical energy, which corresponds to the decrease in free enthalpy ΔG_R of the reaction, into electrical work A_e in accordance with Equation 1:

$$A_e = -\Delta G_R = n \cdot F \cdot E_0 \qquad \text{(Equation 1)}$$

(n Number of electrons converted per fuel molecule, F Faraday constant).

Because in this type of energy conversion the customary indirect path via the generation of heat is avoided, fuel-cell efficiency is not limited by the relatively poor efficiency of the Carnot cycle.

Typical current/voltage characteristics for low-temperature fuel cells



As shown in Table 2, the cell voltages E_0 for the two fuels are close to one another (approx. 1.2 V) and the thermodynamic efficiency η_{th} calculated according to Equation 2 is also < 1 for CH_3OH if liquid water is formed as the reaction product:

$$\eta_{th} = \Delta G_R / \Delta H_R \qquad \text{(Equation 2)}$$

At temperatures above 100 °C (if CH_3OH and H_2O are present in gaseous form), the CH_3OH/O_2 fuel cell shows that in principle fuel cells can also attain a thermodynamic efficiency of > 1.

In practice the theoretical fuel-cell figures are not reached with any of the fuels, even at high temperatures. The reasons for this are primarily to be found in the kinetic inhibition at the two electrodes (e.g. charge penetration and material transfer) as well as in the mixed potential formation at the positive electrode and the electrolyte impedance. The resulting polarization can be reduced but not completely eliminated by using noble metals (platinum, ruthenium) as catalysts, specially structured porous electrodes and small electrode gaps. This applies even in the absence of current flow when a steady-state voltage of only approx. 1 V is measured at an H_2/O_2 PEFC (see Table 1) instead of the expected 1.23 V. When a load is applied, fuel-cell voltage and efficiency drop by varying degrees, depending on the fuel, as current increases due to increasing polarization (see graph). Other variables which have a significant effect on the shape of the current/voltage characteristic of a fuel cell are temperature, gas pressure (1.5 ... 3 bar) and the noble-metal electrode coating, which is now as little as 0.1 ... 0.5 mg Pt/cm².

The typical working voltage of an H_2/air

Table 2. Reaction equations and calculated data for hydrogen (H_2) and methanol (CH_3OH) as energy sources for fuel cells

Reaction equations	State of H_2O/CH_3OH	Temperature °C	E_0 V	η_{th} %
$H_2 + {}^1/_2O_2 \rightarrow H_2O$	Liquid	25	1.23	83
	Liquid	127	1.15	81
	Gaseous	127	1.16	92
	Gaseous	227	1.11	87
$CH_3OH + {}^3/_2O_2 \rightarrow CO_2 + 2\,H_2O$	Liquid/liquid	25	1.21	97
	Liquid/liquid	127	1.20	99
	Gaseous/gaseous	127	1.20	103
	Gaseous/gaseous	227	1.21	104

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PEFC at rated continuous output (0.4...0.5 W/cm² of electrode surface area) is around 0.75...0.70 V (see graph) which, according to Equation 2, results in an efficiency of 51...48%. With CH_3OH as the PEFC energy source, despite the higher temperature (110...130 °C), lower power density (0.1...0.2 W/cm²) and greater amount of Pt on the electrodes, a lower voltage (0.50...0.35 V) and a correspondingly lower thermodynamic efficiency (41...30%) is produced than in the case of an H_2/air PEFC. However, this does not take into account the fact that, in the case of CH_3OH, its permeation through the electrolyte membrane results in a loss at the anode due to which the efficiency of a direct methanol fuel cell is even further reduced.

Fuel cells in motor vehicles

As clean, high-efficiency energy converters, from the point of view of environmental protection and conservation of resources, fuel cells represent an interesting alternative to the traditional methods of generating electrical power. Of the types of fuel cells described here, PEFCs in particular show the most promise in both fixed and mobile applications by virtue of their low operating temperature and compact and robust design. In the area of mobile applications, driven by the increasingly stringent exhaust-emission regulations in the USA and Europe, fuel-cell power units are clearly at the forefront of development efforts. In addition, however, the possibility of using PEFCs as the electrical power source in vehicles with conventional engines is also being investigated, and initial experiments have been carried out. A vehicle electrical system with a power supply independent of the engine which provides the motive power would enable implementation of desirable timer-controlled or remotely controlled auxiliary functions (e.g. preheating of the engine/catalytic converter, air-conditioner operation without the engine running).

All of the automobile manufacturers engaged in comprehensive fuel-cell development work have focused on two key problems, which in principle affect all fuel-cell applications: the high cost of PEFCs and the availability of pure H_2, which is essential for the operation of PEFCs and for which there is no infrastructure at present which could be used to supply vehicles powered by such fuel cells. An additional disadvantage of H_2 as a fuel for fuel-cell powered vehicles is the fact that its energy density is satisfactory only if it is stored in pressurized or liquefied form. For reasons of safety alone, there are serious concerns regarding the storage of H_2 in those forms in private passenger cars.

When using fuel cells as automotive drives, efficiency drops to around 30% from the H_2 storage tank to the wheels. This loss is attributable in part to the auxiliary systems required for monitoring and operation of the fuel cells (e.g. air compressor, coolant pumps, fan cooler control equipment and, where applicable, gas reformers). The electrical power required by the secondary loads can amount to as much as 25% of the fuel-cell output, depending on the nature of the peripherals and the size and operating point of the fuel cell. In addition to fuel-cell efficiency, the efficiency of the electric drive unit must also be taken into consideration. When considering the entire energy chain from the primary energy source to the vehicle's powered wheels, and given the technology available at present, the overall efficiency of modern diesel and fuel-cell-powered vehicles is comparable for vehicles of equal power-to-weight ratios.

No predictions can be made at this time regarding the service life of PEFCs under the dynamic conditions of motor-vehicle operation. As in all applications involving catalysts, fuel cells can also be expected to suffer from a reduction in the catalytic action of the electrodes over time. The result is a gradual decline in the voltage and efficiency of the fuel cell which is referred to as degradation. It is assumed, however, that over the long term the service life of fuel-cell drives will be similar to that of internal-combustion engines.

Bibliography

Karl Kordesch, Günther Simander, Fuel Cells and Their Applications, VCH Weinheim 1996.

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Drivetrain

Units and symbols

Quantity		Unit
a	Acceleration	m/s²
c_w	Drag coefficient	–
e	Rotational inertia coefficient	–
f	Rolling resistance coefficient	–
g	Gravitational acceleration	m/s²
i	Transmission ratio	–
m	Vehicle mass	kg
n	Speed of rotation	min⁻¹
r	Dynamic tire radius	m
s	Wheel slip	–
v	Road speed	m/s
A	Frontal area	m²
D	System diameter	m
I	Overall conversion range	–
J	Moment of inertia	kg · m²
M	Torque	N · m
P	Power	kW
α	Angle of inclination	°
φ	Overdrive factor	–
η	Efficiency	–
λ	Performance index	–
μ	Conversion	–
ρ	Density	kg/m³
ω	Angular velocity	rad/s
ν	Speed ratio	–

Subscripts:			
eff	Effective	eng	Engine
tot	Total	o	Towards maximum output
hydr	Hydraulic	dt	Drivetrain
max	Maximum	G	Gearbox
min	Minimum	P	Pump
fd	Final drive	R	Roadwheel
		T	Turbine

Assignment

The function of the automotive drivetrain is to provide the thrust and tractive forces required to induce motion. Energy in chemical (fuels) or electrical (batteries, solar cells) form is converted into mechanical energy within the power unit, with spark-ignition and diesel internal-combustion engines representing the powerplants of choice. Every power unit operates within a specific revolution range as defined by two extremities: the idle speed and the maximum min⁻¹. Torque and power are not delivered at uniform rates throughout the operating range; the respective maxima are available only within specific bands. The drivetrain's conversion ratios adapt the available torque to the momentary requirement for tractive force.

Design

The dynamic condition of an automobile is described by the motion-resistance equation. It equates the forces generated by the drive train with the forces required at the driving wheels (resistance to motion).

From the motion-resistance equation it is possible to calculate acceleration, maximum speed, climbing ability and also the overall conversion range I of the transmission.

$$I = \frac{(i/r)_{max}}{(i/r)_{min}} = \frac{\tan \alpha_{max} \cdot v_o}{(P/(m \cdot g))_{eff.} \cdot \varphi}$$

Equilibrium relation between drive forces and tractive resistance

The equation defining the equilibrium between drive forces and resistance factors is applied to determine various quantities, such as acceleration, top speed, climbing ability, etc.

Available power = Tractive resistance at drive wheels (power requirement)

$$M_{eng} \cdot \frac{i_{tot}}{r} \cdot \eta_{tot} = m \cdot g \cdot f \cdot \cos \alpha + m \cdot g \cdot \sin \alpha + e \cdot m \cdot a + c_w \cdot A \cdot \frac{\rho}{2} \cdot v^2$$

Driving force applied to tyre footprint	Rolling resistance	Gravitational resistance	Acceler-ation	Wind resistance

Where rotational inertia coefficient $e = 1 + \frac{J}{m \cdot r^2}$ and mass inertia $J = J_R + i_{fd}^2 \cdot J_{dt} + i_{fd}^2 \cdot i_G^2 \cdot J_{eng}$

The overdrive factor φ is

$$\varphi = \frac{(i/r)_{min}}{\omega_o / v_o}.$$

Calculations of effective specific output should always be based on the power P which is actually available for tractive application (net power minus driven ancillaries, power losses, altitude loss). Special conditions, such as automobile trailer towing, must be factored into the weight $m \cdot g$. $\varphi = 1$ is true, when the curve for cumulative running resistance in top gear directly intersects the point of maximum output. The φ factor determines the relative positions of the curves for running-resistance and engine output in top-gear operation, it also defines the efficiency level at which the engine operates.

$\varphi > 1$ displaces operation into an inefficient engine-performance range, but also enhances acceleration reserves and hill-climbing ability in top gear. In contrast, selecting $\varphi < 1$ will increase fuel economy, but only at the price of substantially slower acceleration and lower climbing reserves. Minimum fuel consumption is achieved on the operating curve η_{opt}. $\varphi > 1$ reduces, $\varphi < 1$ increases the required transmission conversion range I.

Drivetrain configurations

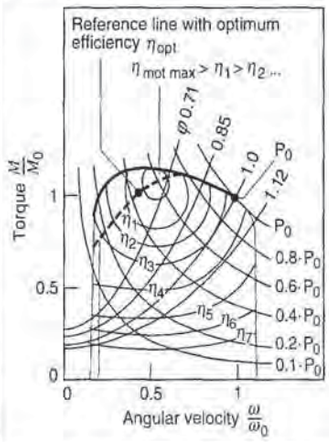
The layout of the automotive drivetrain varies according to the position of the engine and the drive axle:

Drive configuration	Engine position	Driven axle
Standard rear-wheel drive	front	Rear-axle
Front-wheel drive	front, longit. or transverse	Front axle
All-wheel drive	front, occasionally rear or middle	Front-axle and rear-axle
Rear-wheel drive with rear-mounted engine	rear	Rear-axle

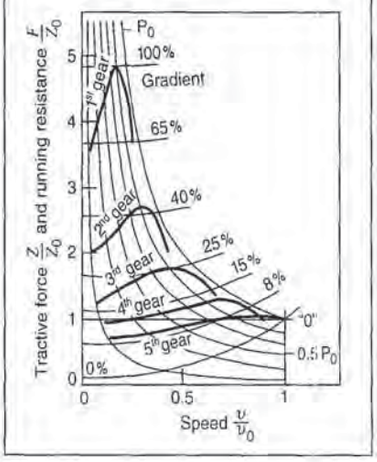
Drivetrain elements

The elements of the drive train must perform the following functions:
- remaining stationary even with the engine running,
- achieving the transition from a stationary to a mobile state,
- converting torque and rotational speed,
- providing for forward and reverse motion,
- compensating for wheel-speed variations in curves,
- ensuring that the power unit remains within a range on the operating curve commensurate with minimum fuel consumption and exhaust emissions.

Engine performance curve with curves for running resistance



Tractive force/speed diagram



Stationary idle, transition to motion and interruption of the power flow are all made possible by the clutch. The clutch slips to compensate for the difference in the rotational speeds of engine and drivetrain when the vehicle is being set in motion. When different conditions demand a change of gear, the clutch disengages the engine from the transmission while the gear shift operation takes place. With automatic transmissions, the hydrodynamic torque converter is responsible for the take-up of power. The transmission (gearbox) modifies the engine's torque and min^{-1} to adapt them to the vehicle's momentary tractive requirements.

The overall conversion of the drivetrain is the product of the constant transmission ratio of the axle differential and the variable transmission ratio of the gearbox – assuming there are no other transmission stages involved. Gearboxes are almost always multiple fixed-ratio gearboxes though some are of the continuously variable ratio type.

Gearboxes generally fall into one of two categories: manual gearboxes with spur gears and a layshaft arrangement and load-actuated automatic transmissions with planetary gears. The transmission also allows the selection of different rotational directions for forward and reverse operation.

The differential allows laterally opposed axles and wheels to rotate at varying rates during cornering while providing uniform distribution of the driving forces. Limited-slip final drives respond to slippage at one of the wheels by limiting the differential effect, shifting additional power to the wheel at which traction is available.

Torsion dampers, hydrodynamic transmission elements, controlled-slip friction clutches or mass-suspension systems dissipate high vibration amplitudes, as well as protecting against overload and providing added ride comfort.

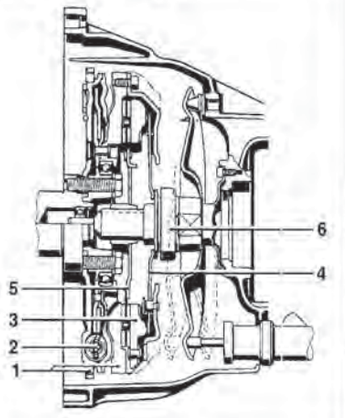
Power take-up elements

Dry-plate friction clutch

The friction clutch consists of a pressure plate, a clutch disk – featuring bonded or riveted friction surfaces – and the second friction surface represented by the engine-mounted flywheel. The flywheel and pressure plate provide the thermal absorption required for friction operation of the clutch; flywheel and pressure plate are connected directly to the engine, while the clutch disk is mounted on the transmission's input shaft.

A spring arrangement, frequently in the form of a central spring plate, applies the force which joins the flywheel, pressure plate and clutch disk for common rotation; in this state, the clutch is engaged for positive torque transfer. To disengage the clutch (e.g., for gear shifting), a mechanically or hydraulically actuated throwout bearing applies force to the center of the pressure plate, thereby releasing the pressure at the periphery. The clutch is controlled either with a clutch pedal or with an electrohydraulic, electropneumatic or electromechanical final-control element. A single- or multi-stage torsion damper, with or without a predamper, may be integrated in the clutch plate to absorb vibration.

Clutch with dual-mass flywheel
1 Dual-mass flywheel, 2 Flexible element, 3 Pressure plate, 4 Spring plate, 5 Friction plate, 6 Thrust bearing



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A two-section (dual-mass) flywheel featuring a flexible intermediate element can be installed forward of the clutch for maximum insulation against vibrations. The resonant frequency of this sprung mass system is below the exciter frequency (ignition frequency) of the engine at idling speed and therefore outside the operating speed range. It acts as a vibration insulating element between the engine and the other drive-train components (low-pass filter).

When used together with electronic control units, the automatic clutch can provide either gradual engagement for starting off, or it can be applied in conjunction with a servo-operated shifting mechanism to form a fully-automatic transmission unit. Traction control and disengagement of power transmission during braking are also possible.

Wet-plate friction clutch

The wet-plate friction clutch has the advantage over the dry-plate version that its thermal performance is better as oil can be passed through to assist heat dissipation. However, its drag losses when disengaged are considerably higher than with a dry clutch. Use in combination with synchromesh gearboxes presents problems due to the increased synchronous load when changing gear. The wet clutch was introduced as a standard component on continuously variable car transmissions. It has space-saving advantages particularly when one or more friction-drive gear-shift components (multiplate clutch or clutch stop) that are present in any case can also be used for the power take-up process.

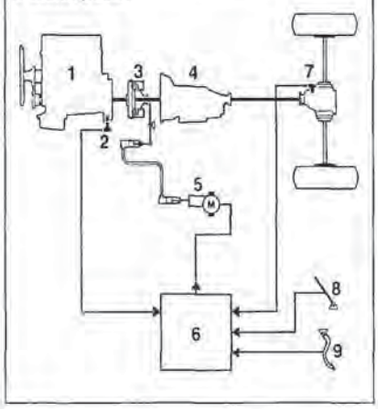
Hydrodynamic torque converter

The hydrodynamic torque converter consists of the impeller which is the driving component, the turbine which is the driven component and the stator which assists the torque converter function. The torque converter is filled with oil and transmits the engine torque by means of the viscosity of the oil. It compensates for the speed difference between the engine and the other drive-train components and is therefore ideally suited to a power take-up function. An impeller converts the mechanical energy into fluid energy; and a second transformation, back into mechanical energy, takes place at the blades within the turbine.

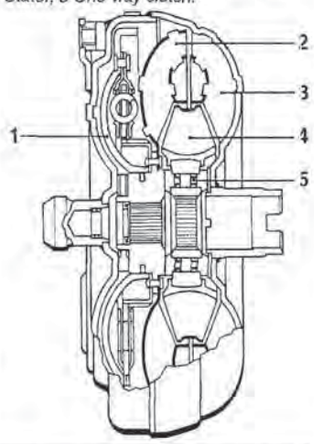
The impeller's input torque M_p and input power P_p are calculated as follows:

$$M_p = \lambda \cdot \rho \cdot D^5 \cdot \omega^2_p$$
$$P_p = \lambda \cdot \rho \cdot D^5 \cdot \omega^3_p$$

Automatic clutch, limited to clutch actuation
1 Engine, 2 Engine rpm sensor, 3 Clutch, 4 Transmission, 5 Servomotor, 6 ECU, 7 Speed sensor, 8 Accelerator pedal, 9 Clutch pedal.



Hydrodynamic converter with lockup clutch
1 Lockup clutch, 2 Turbine, 3 Impeller, 4 Stator, 5 One-way clutch.



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λ Performance index
ρ Density of medium
($\approx$ 870 kg/m³ for hydraulic fluid)
D Circuit diameter in m
ω_P Angular velocity of impeller

A stator located between impeller and turbine diverts the hydraulic oil back to the input side of the impeller. This raises the torque beyond the initial engine output as exerted at the impeller. Torque conversion is then

$$\mu = -M_T/M_P$$

The factor ν is defined as the ratio of turbine speed to impeller speed; it has a determining influence on both the performance index λ and the conversion factor μ:

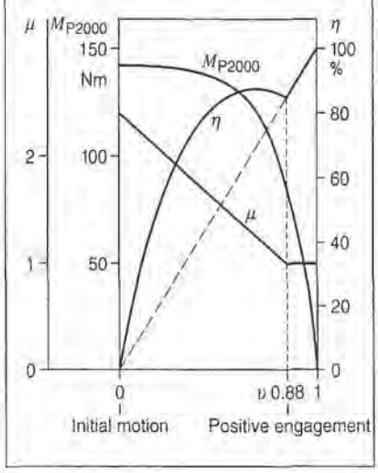
$$\nu = \omega_T / \omega_P.$$

The slip factor $s = 1 - \nu$ and the force conversion together determine the hydraulic efficiency:

$$\eta_{hydr} = \mu\,(1 - s) = \mu \cdot \nu.$$

Maximum torque multiplication is achieved at $\nu = 0$, i.e., with the turbine at stall speed. Further increases in turbine speed are accompanied by a virtually linear drop in multiplication until a torque ratio of 1:1 is reached at the coupling point. Above this point the stator, which is housing-mounted with a one-way clutch, freewheels in the flow.

Trilok converter
(typical passenger-car performance curve)
$M_{P2000} = M_P$ at n = 2000 min⁻¹



In motor-vehicle applications, the two-phase Föttinger torque converter with centripetal flow through the turbine, the "Trilok converter", has become the established design. The geometrical configuration of this unit's blades is selected to provide torque multiplication in the range of 1.7...2.5 at stall speed ($\nu = 0$). The curve defining the hydraulic efficiency factor $\eta_{hydr} = \nu \cdot \mu$ in the conversion range is roughly parabolic. Above the coupling point, which is at 10...15% slip, the efficiency is equal to the speed ratio ν and reaches levels of around 97% at high speeds.

The hydrodynamic torque converter is a fully automatic infinitely variable transmission with virtually zero-wear characteristics; it eliminates vibration peaks and absorbs vibration highly effectively.

However, its conversion range and efficiency, particularly at high levels of slip, are not sufficient for motor-vehicle applications so that the torque converter can only be usefully employed in combination with multi-speed or continuously variable gearboxes.

Converter lockup clutch

In order to improve efficiency, the impeller and turbine can be locked together by a converter lockup clutch once power take-up is complete. The converter lockup clutch consists of a plunger with friction surface, which is connected to the turbine hub. The transmission's valve body regulates the direction in which the fluid flows through the converter to regulate coupling engagement.

The converter lockup clutch normally requires additional means of vibration absorption such as
- a torsion damper,
- controlled-slip operation of the converter lockup clutch at critical vibration levels or
- both of the above in combination.

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Multi-speed gearbox

Multi-speed gearboxes have become the established means of power transmission in motor vehicles. Good efficiency characteristics dependent upon the number of gears and engine torque characteristics, satisfactory to good adaptation to the traction hyperbola and easily mastered technology are the essential reasons for its success.

Gear shifting on multi-speed gearboxes is performed using either disengagement of power transmission (positively interlocking mechanism) or under load by a friction mechanism. The first group includes manual and semi-automatic gearboxes while the second group encompasses automatic transmissions.

The manually-shifted transmissions installed in passenger cars and in most heavy vehicles are dual-shaft units with main and countershaft (layshaft, idler gears). Transmissions in heavy commercial vehicles sometimes incorporate two or even three countershafts. In such cases, special design features are required in order to ensure that power is evenly distributed to all countershafts.

Automatic transmissions for cars and commercial vehicles are, in the majority of cases, planetary gear transmissions and only in rare cases are countershaft designs used. The planetary gears generally take the form of a planetary-gear link mechanism. They frequently involve the use of Ravigneaux or Simpson planetary gears.

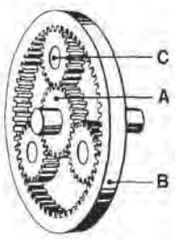
Planetary-gear sets

The basic planetary-gear set consists of the sun gear, internal ring gear and the planet gears with carrier. Each element can act as input or output gear, or may be held stationary. The coaxial layout of the three elements makes this type of unit ideal for use with friction clutches and brake bands, which are employed for selective engagement or fixing of individual elements. The engagement pattern can be changed – and a different conversion ratio selected – without interrupting torque flow; this capability is of particular significance in automatic transmissions.

As several gear wheels mesh under load simultaneously, planetary-gear transmissions are very compact. They have no free bearing forces, permit high torque levels, power splitting or power combination, and feature very good efficiency levels.

Planetary-gear set with various conversion ratios
A Sun gear, B Internal ring gear, C Planet gears with carrier, Z Number of teeth.

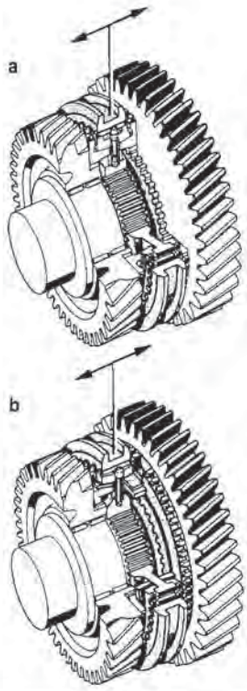
Basic equation for planetary-gear sets: $n_A + (Z_B / Z_A) \cdot n_B - [1 + Z_B / Z_A] \cdot n_C = 0$



Input	Output	Fixed	Transmission ratio	Remarks
A	C	B	$i = 1 + Z_B / Z_A$	$2.5 \le i \le 5$
B	C	A	$i = 1 + Z_A / Z_B$	$1.25 \le i \le 1.67$
C	A	B	$i = \frac{1}{1 + Z_B / Z_A}$	$0.2 \le i \le 0.4$ overdrive
C	B	A	$i = \frac{1}{1 + Z_A / Z_B}$	$0.6 \le i \le 0.8$ overdrive
A	B	C	$i = -Z_B / Z_A$	Non-automotive with reversible direction of rotation $-0.4 \le i \le -1.5$
B	A	C	$i = -Z_A / Z_B$	Non-automotive with reversible direction of rotation $-0.25 \le i \le -0.67$

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Manually-shifted transmission
a) Single-band synchromesh,
b) Dual-band synchromesh.



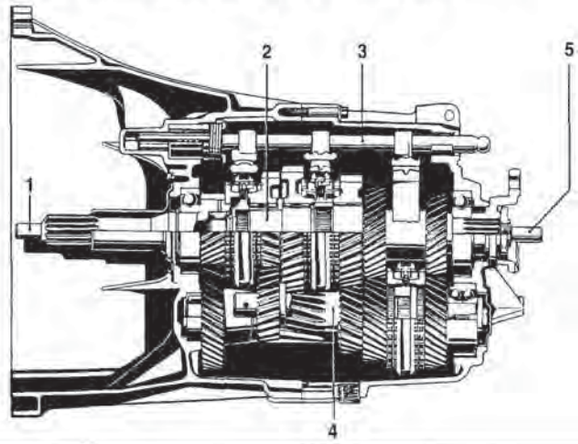
Manually-shifted transmissions

The basic elements of the manually-shifted transmission are:
- single or multiplate dry clutch for interrupting and engaging the power flow; actuation may be power-assisted to deal with high operating forces,
- variable-ratio gear transmission unit featuring permanent-mesh gears in one or several individual assemblies,
- shift mechanism with shift lever.

The force required for gear selection is transmitted via shift linkage rods or cable, while dog clutches or synchronizer assemblies lock the active gears to the shafts. Before a shift can take place, it is necessary to synchronize the rotating speeds of the transmission elements being joined. When the transmission incorporates dog clutches (of the type still sometimes used in transmissions for heavy commercial vehicles), the driver performs this task by double-clutching on both upshifts and downshifts, with the latter being accompanied by the application of throttle.

Virtually all passenger-car transmissions, and the majority of those in commercial vehicles, employ locking synchronizer assemblies. These include a friction

5-speed transmission for passenger car with conventional drive layout *(ZF S5-31).*
1 Input shaft, 2 Main shaft, 3 Selector rail, 4 Idler shaft, 5 Output shaft



coupling for initial equalization of rotating speed and a lockout mechanism to prevent positive gear engagement prior to completion of the synchronization process. By far the majority use single-cone synchromesh clutches. In cases where there are particularly high demands for performance and/or reduction of gear-shifting force, double or even triple-cone synchromesh clutches or multi-plate synchromesh clutches are used.

Most transmissions in passenger cars include 5, occasionally even 6 forward ratios. The transmission-ratio range (depending on the number of gears and closeness of the ratios) is approximately between 4 and 5.5 while the transmission efficiency can be as high as 99%. The gearbox layout depends on the vehicle's drive configuration (standard rear-wheel drive, front-wheel drive with inline or transverse engine, or four-wheel drive). Thus the input and output shafts may share a single axis, or they may be mutually offset; the final-drive and differential assembly may also be included in the unit.

Transmissions in commercial vehicles can have between 5 and 16 gears, depending upon the type of vehicle and the specific application. For up to 6 gears, the transmission consists of a single gearbox. The transmission-ratio range is between 4 and 9. Transmissions with between 7 and 9 gears are two-box transmissions in which the range-selector box is pneumatically operated. The conversion range extends to 13.

For still higher numbers of ratios – up to 16 – three transmission elements are employed: the main transmission, a splitter group and the range group, with pneumatic actuation for the latter two units. The conversion range is as high as 16.

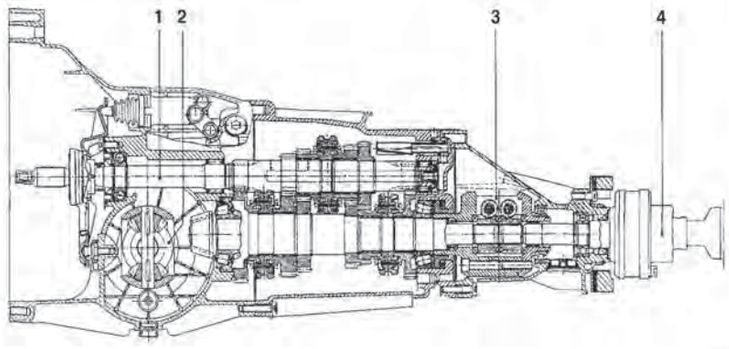
Power take-offs (auxiliary drives)

Commercial-vehicle transmissions are fitted with a variety of power take-off connections for driving auxiliary equipment. A basic distinction is made between clutch and engine-driven PTO's. The individual selection depends upon the specific application.

Retarders

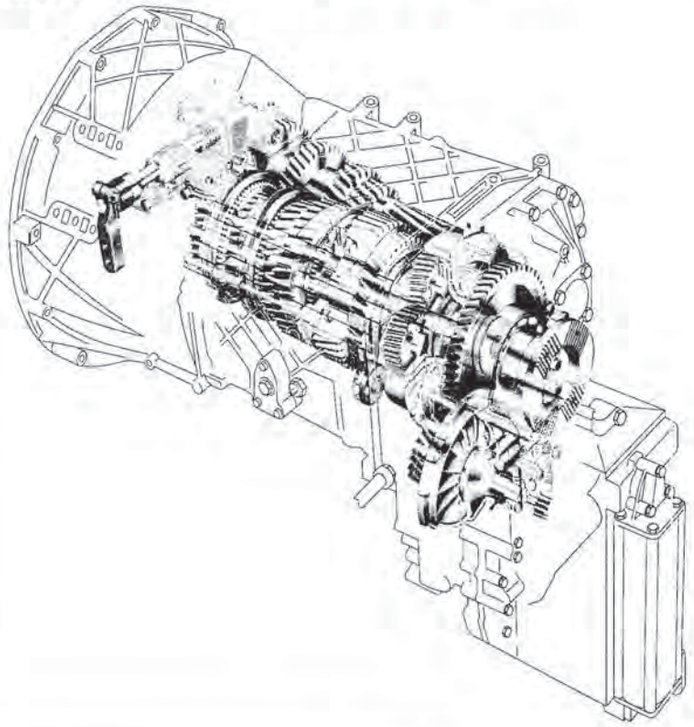
Hydrodynamic or electrodynamic retarders are non-wearing auxiliary brakes for reducing the thermal load on the road-wheel brakes under continuous braking. They can be fitted either on the drive input side (primary retarders) or the drive output side (secondary retarders) integral with the transmission, or as a separate unit between the transmission output shaft and the driving axle. The advantages of the integrated designs are com-

5-speed transmission for longitudinal-engine passenger car with 4wd *(Audi quattro)*
1 Input shaft, 2 Front-axle differential, 3 AWD transfer box with Torsen locking differential, 4 Rear-axle drive shaft

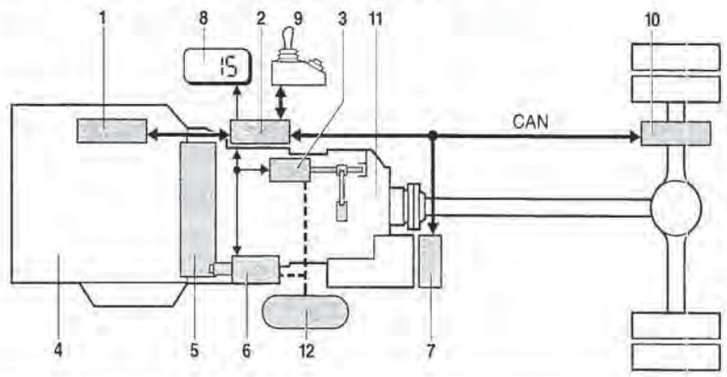


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16-speed multiple transmission with integral retarder for heavy trucks
(ZF Ecosplit 16 S221)



Semi-automatic transmission *(schematic)*
1 Engine electronics (EDC), 2 Transmission electronics, 3 Transmission actuator, 4 D iesel engine, 5 Dry-plate friction clutch, 6 Clutch actuator, 7 Intarder control unit, 8 Display, 9 Gear selector, 10 ABS/TCS, 11 Gearbox, 12 Air cylinder.
—— Electrical, ----- Pneumatic, —— CAN communication



pact dimensions, low weight and fluid shared with the transmission in a single circuit. Primary retarders have specific advantages when braking at low speeds and are therefore widely used on public transport busses. Secondary retarders have particular advantages on long-distance trucks for adjustment braking at higher speeds or when traveling downhill.

Automatic transmissions

There are two types of automatic transmission (or gearbox) which are distinguished according to their effect on vehicle handling dynamics:

- Semi-automatic transmissions are manual gearboxes on which all operations normally performed by the driver when changing gear are carried out by electronically controlled actuator systems. In terms of vehicle dynamics, this means that a gear change always involves disengagement of a clutch and therefore of the drive to the driving wheels.
- Fully automatic transmissions, usually referred to simply as automatic transmissions, change gear under load, i.e. the power continues to be transmitted to the driving wheels during a gear-shift operation.

That difference in vehicle handling dynamics is the essential factor which determines the types of application for these two transmission types. Fully automatic transmissions are used in situations where disengagement of the power transmission would be associated with a significant reduction in comfort (above all on cars with powerful acceleration), or where it is unacceptable for reasons of handling dynamics (above all on off-road vehicles). Semi-automatic transmissions are found on long-distance haulage trucks, passenger coaches and more recently also on small cars, racing cars and fast production sports cars.

Semi-automatic transmissions

Partially or fully automated gear-shifting systems contribute substantially to simplifying control of the gears and increasing economy. Particularly when used on trucks, the disadvantages inherent in the interruption of power transmission are compensated for by a number of decisive advantages:

- narrower spacing of ratios, with up to 16 gears,
- enhanced-efficiency power transfer,
- reduced costs,
- same basic transmission unit for manual and semi-automatic designs.

Operation

A positioner module on the transmission converts the electrical signals into hydraulic or pneumatic signals and controls the actuator cylinders for the gear-shift operation. The electrical control signals are generated by an intermediate ECU.

Design variations

On the simplest systems, the mechanical gear-shift linkage is simply replaced by a remote control system. The gear-shift lever then merely sends out electrical signals. Drive-engagement and clutch-operation procedures are as with a standard manually-shifted transmission. More complex versions combine these systems with a recommended shift-point function. Advantages are:

- reduced shifting effort,
- simplified installation (no shift linkage),
- prevention of incorrect operation (over-revving of engine).

On fully automated gear-shifting systems both the transmission and the drive-engagement mechanism are automated. The driver's control device consists of either a lever or pushbuttons, with an override provision in the shape of a driver-selected manual mode or +/– buttons.

Complex shift programs are required to control a multi-ratio transmission. A system which engages the gears according to a fixed pattern will not be adequate. Current running resistance (as determined by load and road conditions) must be factored in to achieve the optimal balance between driveability and fuel economy. This task is performed by a microcomputer control system along with control of synchromesh for gear changing. To that end the engine speed is adjusted by

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the electronic throttle-control system (ETC or EGAS) to the speed called for by the transmission control system via the data communication bus. As a result, mechanical synchromesh systems can be partly or entirely dispensed with in the gearbox.
Advantages are:
- optimum operating economy through automatic, computer-controlled shifting,
- reduced driver workload,
- lower weight and smaller dimensions,
- enhanced safety for both driver and vehicle.

Fully automatic transmissions
Fully automatic transmissions perform the drive-engagement and ratio selection (shifting) operations with no additional driver input. The power take-up unit is always a hydrodynamic torque converter which almost always has a converter lockup clutch on cars and generally does on commercial vehicles. The power-transmission efficiency of such automatic transmissions is inherently slightly lower than that of manual and semi-automatic transmissions. However, in many applications this is compensated for by shift programs designed to keep engine operation inside the maximum economy range.
The components are:
- hydrodynamic torque converter (always employed with passenger-car transmissions, while commercial vehicles generally use the Trilok design); for starting off, for torque multiplication and for absorption of harmonic vibrations.
- a number of planetary-gear sets downstream from the hydrodynamic torque converter. Number and arrangement according to number of gears and ratios.
- hydraulically-actuated multiplate clutches, plate or band brakes. Assigned to the individual elements within the planetary-gear sets to execute shifts without interrupting the flow of power.
- one-way clutches together with shift elements for optimal activation under load.
- a transmission-control system to define gear selections and shift points and to regulate demand-response shifting, as dictated by the driver-selected shift program (selector lever), accelerator pedal, engine operating conditions and vehicle speed. The transmission control system is an electronic-hydraulic system.
- an engine-driven hydraulic-fluid pump (occasionally supplemented by a second fluid pump at the output end) provides hydraulic pressure for valve body and shift elements as well as supplying fluid to the hydrodynamic torque converter. It also supports lubrication and cooling in the transmission.

Design variations
Present-day automatic car transmissions have 4 or 5 gears while future designs will feature 6 speeds. The mechanical conversion range lies between 3.5 (4-speed transmission) and 5.0 (5-speed unit). Figures for driveoff conversion extend from 1.7 to 2.5.

Automatic transmissions in commercial vehicles can have between 3 and 6 forward gears. The mechanical conversion range extends from 2 to 8. These transmissions frequently have integral hydrodynamic retarders, as the requisite peripheral systems (fluid pump, large fluid pan, fluid cooler) are already in place.

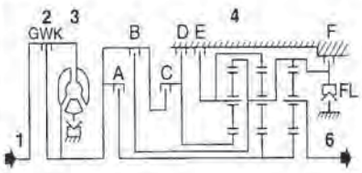
Electronic transmission control

Automatic gearboxes are nowadays almost exclusively controlled by electronically operated hydraulic systems. Hydraulic actuation is retained for the clutches, while the electronics assume responsibility for gear selection and for adapting the pressures in accordance with the torque flow. The advantages are:
- a number of gear-shift programs,
- smoother gear shifting,
- adaptability to different types of vehicle,
- simpler hydraulic control, and
- ability to dispense with one-way clutches.

Sensors detect the transmission output-shaft speed, the engine load and speed, the selector-lever position and the positions of the program selector and the kick-down switch. The control signal

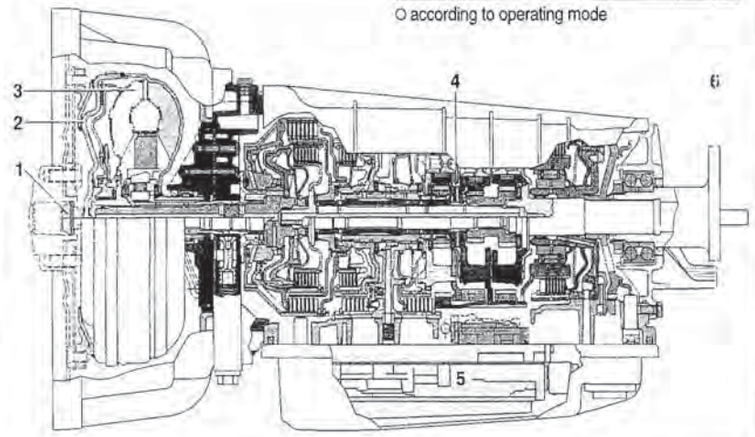
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5-speed passenger-car automatic gearbox *(ZF 5 HP24)*
1 Input shaft, 2 Controlled-slip converter lock up clutch (GWK), 3 Hydrodynamic torque converter, 4 5-speed planetary gears with three multiplate brakes, three multiplate clutches and one one-way clutch, 5 Electronic-hydraulic transmission-shift control, 6 Output shaft

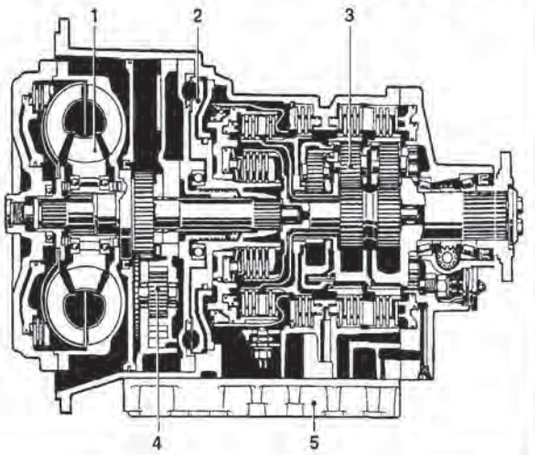


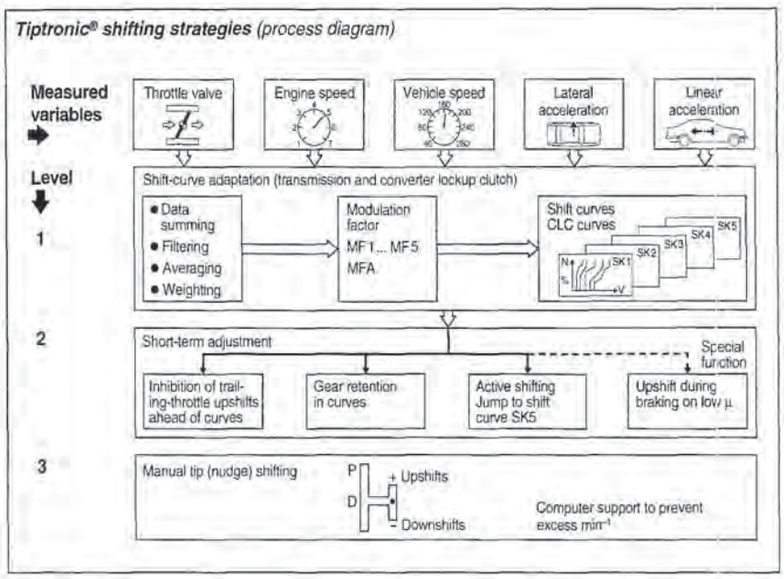
Gear	Clutch			Brake			Overrun	Ratio
	A	B	C	D	E	F	FL	i
1	●					○	●	3.57
2	●				●			2.20
3	●			●				1.51
4	●	●						1.00
5		●		●				0.80
R			●			●		-4.10

○ according to operating mode



Automatic transmission with integral primary retarder for busses, trucks and special vehicles
(ZF Ecomat 5HP 500).
1 Hydrodynamic torque converter with lockup clutch,
2 Hydrodynamic retarders,
3 5-speed planetary gears,
4 Oil pump,
5 Transmission-shift control.





processes this information according to a predefined program and uses the results in determining the control variables to be transmitted to the gearbox. Electrohydraulic converter elements form the link between the electronic and hydraulic circuits, while standard solenoid valves activate and disengage the clutches. Analog or digital pressure regulators ensure precise control of pressure levels at the friction surfaces. A typical system includes:

Shift-point control
In selecting the gear to be engaged, the system refers to the rotating speeds of the transmission output shaft and of the engine before triggering the appropriate solenoid valves. The driver may select from among different shift programs (e.g., for maximum fuel economy or for maximum performance). In addition, the selector lever allows manual input from the driver.

"Intelligent" shift programs improve driveability by supplementing the standard transmission-control data with additional parameters such as forward and lateral acceleration, and the speed with which accelerator and brake pedal are pressed. A complex control program selects the appropriate gear for the current operating conditions and driving style using such expedients as suppressing trailing-throttle upshifts on the approach to and during corners, or automatically responding to slow throttle openings by activating a shift program for low- min^{-1} upshifts. Concepts which combine the high level of convenience of such "intelligent" gear-shift programs with facilities for active adaptation to individual driver preferences have become very widespread. In addition to the normal positions for neutral, drive and reverse, the selector levers for such systems have a second parallel channel in which one touch of the lever produces an immediate gear change (provided no engine-speed limits would be exceeded).

Converter lockup
A mechanical lockup clutch can be employed to improve the efficiency of the transmission unit by eliminating torque-converter slip. The variables employed to determine when conditions are suitable for activation of the converter lockup mechanism are engine load and transmission output-shaft min^{-1}.

Control of shift quality
The accuracy with which the pressure at the friction elements is adjusted to the

level of torque being transmitted (determined with reference to engine load and min^{-1}) has a decisive influence on shift quality; this pressure is regulated by a pressure regulator. Shifting comfort can be further enhanced by briefly reducing engine output for the duration of the shift (e.g., by retarding the ignition timing); this also reduces friction loss at the clutches and extends component service life.

Safety circuits
Special monitoring circuits prevent transmission damage stemming from operator error, while the system responds to malfunctions in the electrical system by reverting to a backup mode.

Final-control elements
Electrohydraulic converter elements such as solenoid valves and pressure regulators form the link between the electronic and hydraulic circuits.

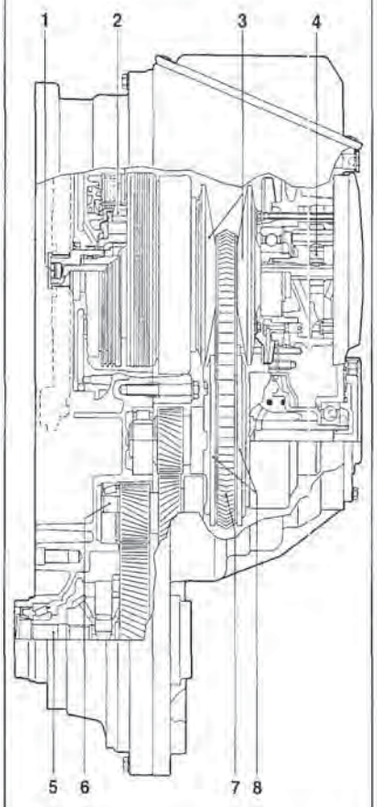
Continuously-variable transmissions

The continuously-variable transmission (CVT) can convert every point on the engine's operating curve to an operating curve of its own, and every engine operating curve into an operating range within the field of potential driving conditions. Its advantage over conventional fixed-ratio transmissions lies in the potential for enhancing performance and fuel economy while reducing exhaust emissions (e.g., by maintaining the engine in the performance range for maximum fuel economy).

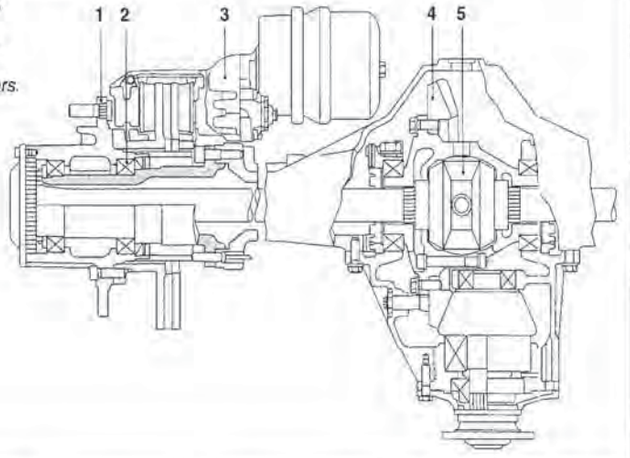
The continuously variable transmission (CVT) can operate mechanically (belt or friction-roller), hydraulically or electrically. The highest level of development has been achieved with mechanical continuously variable units employing steel belts (as installed in low-output series-production cars), where the unit's offset axes make it ideal for installation in front-wheel drive cars with transverse-mounted engines. The conversion range is 5.5...6. The major elements in the continuously variable mechanical belt transmissions for passenger cars are:

- engagement mechanism for starting off (wet multiplate clutch, hydrodynamic torque converter or magnetic-particle coupling),
- primary and secondary disks with axially adjustable taper-disk sections and power transfer via steel bands (CVT)
- electronic/hydraulic transmission control,

Continuously variable passenger-car transmission *(ZF VT1)*
1 Input shaft,
2 Reversing-wheel set with power take-up components,
3 Axial-shift cones (input side),
4 Oil pump,
5 Final drive unit with differential,
6 Intermediate shaft,
7 Linked steel belt,
8 Axial-shift cones (output side).



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Direct-drive axle for busses (ZFA131).
1 Wheel hub,
2 Wheel bearing,
3 Brake caliper (air-operated),
4 Final drive,
5 Differential gears.

– reversing mode, and
– final drive unit with differential.

We can expect continuously variable transmissions (CVTs) using steel belts to be fitted on more powerful cars as well in the foreseeable future.

Friction-roller transmissions have not been used in production vehicles to date. Form the point of view of configuration, they are better suited to rear-wheel-drive vehicles.

Electrical transmissions are used on some public transport vehicles. Apart from with diesel engines, they can be combined with alternative power sources such as batteries, overhead power lines or, in the future, fuel cells. Other advantages of electrical bus transmissions on busses are non-integrated (distributed) design of the overall power train, single-wheel drive and, in particular, simpler realisation of low-floor vehicle designs.

Hydrostatic-mechanical indirect-drive transmissions are in use on production agricultural tractors. Use on road-going vehicles is unlikely due to the high noise levels produced.

Final-drive units

The overall conversion ratio between engine and drive wheels is produced by several elements operating in conjunction: a transmission with several fixed ratios (automatic or manual), an intermediate transmission in some applications (AWD transfer case) and the final-drive unit.

Longer distances between transmission and final drive are bridged by the drive shaft (in one piece or in several sections with intermediate bearings). Angular offsets resulting from non-aligned connecting drive shafts are compensated for by means of universal joints, constant-velocity joints and flexible-disk joints.

The central component of a car final drive is either a hypoid bevel-gear crownwheel and pinion (in-line engine) or a spur-gear crownwheel and pinion (transverse engine) while the layout of the components may take the form of a separate differential (on rear-wheel-drive and four-wheel-drive vehicles) or an integral gearbox and differential (on front-wheel-drive vehicles).

The chief components of a final drive differential are the crown wheel and pinion, the planetary gears, bearings, drive shaft and halfshaft flanges and differential housing. The final-drive transmission ratio is usually between 2.6 and 4.5 to 1.

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The crown wheel is normally bolted to the differential case which holds the planetary gears, and the pinion shaft and differential case run in taper roller bearings. To reduce the transmission of structure-borne noise to the bodyshell, the final drive unit is attached to the vehicle frame by flexible (rubber) mountings.

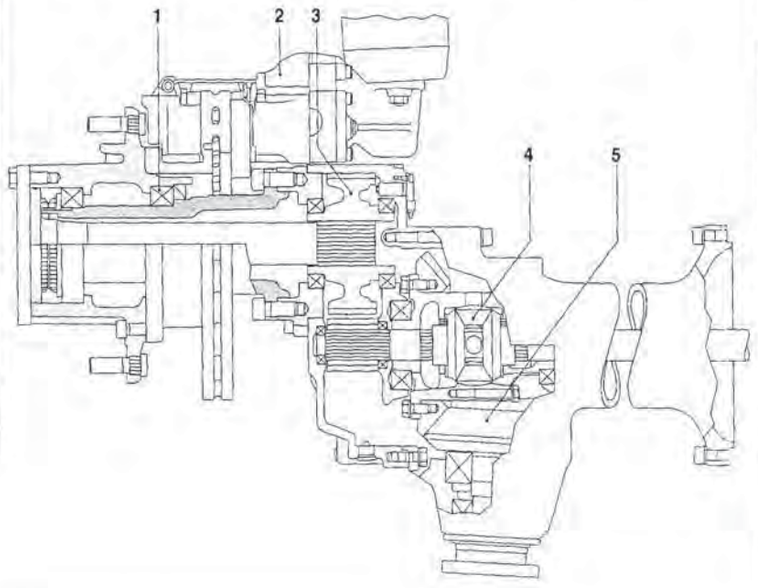
In addition to its torque transmission capabilities, mechanical efficiency and weight, the noise-producing characteristics of a final drive unit have become a decisive criterion in modern automobile development. In this regard, the crown wheel and pinion are of primary significance in terms of noise generation. The quietness of the mechanism is essentially dependent on the way in which the gears are manufactured. A distinction is made between conventional manufacturing methods and grinding of the teeth whereby the imperfections produced by heat treatment (case hardening) are eliminated by appropriate grinding so that in contrast with conventional methods (e.g. lapping) precise (i.e. maximum possible correspondence between calculated tooth-profile topography and actual geometry produced on the machine) and reproducible tooth-profile topographies are produced.

On commercial vehicles, direct-drive axles with hypoid bevel gears are most commonly used. The final-drive transmission ratio ranges between 3 and 6 to 1. In cases where smooth-running characteristics are particularly important, e.g. on busses, the gears are ground.

On public transport busses, which nowadays are almost always low-floor designs, portal (hub reduction) axles are used as they allow very low floor levels. In addition to the helical bevel-gear differential there is also an indirect-drive spur-gear reducer stage. This indirect-drive arrangement allows the required high torque levels to be transmitted with the limited reducer offset.

If there are special ground clearance requirements (e.g. construction-site vehi-

Hub reduction axle for busses *(ZF AV132)*
1 Wheel bearing, 2 Brake caliper, 3 Reduction gearing, 4 Differential gears, 5 Final drive.



cles), planetary axles are used. Propeller shaft and halfshafts can be made smaller by splitting the power transmission, and the available space is increased.

Differential

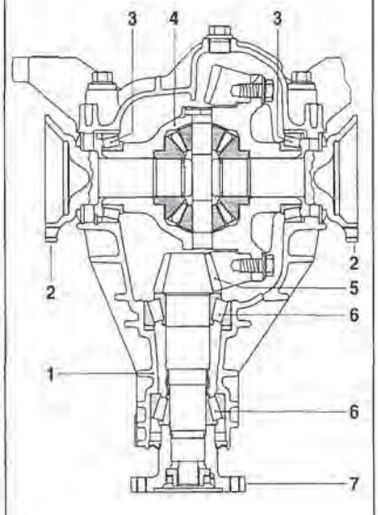
The differential unit compensates for discrepancies in the respective rotation rates of the drive wheels: between inside and outside wheels during cornering, and between different drive axles on AWD vehicles.

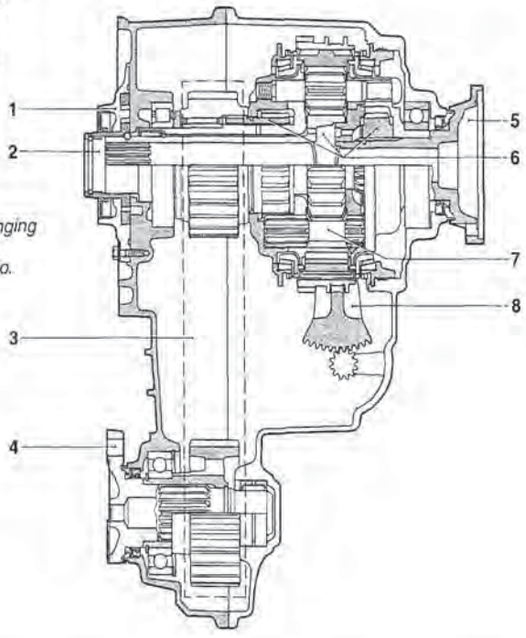
Apart from a few special cases, bevel gears are the preferred design for differentials. The differential gears act as a balance arm to equalize the distribution of torque to the left and right wheels. When lateral variations in the road surface produce different coefficients of friction at the respective wheels ("μ-split"), this balance effect limits the effective drive torque to a level defined as twice the tractive force available at the wheel (tire) with the lower coefficient of friction. If torque then exceeds frictional resistance, that wheel will spin.

This undesirable effect can be eliminated by locking the differential either by a positively interlocking or friction mechanism. Positively interlocking differential locks are switched-in by the driver. Their disadvantage lies in the stress applied to the drivetrain which occurs during cornering. Friction-type differential locks operate automatically using friction plates, cones or a combination of worm and spur-gear drives and thus have a variable locking action depending on torque. The friction-type locking action can also be achieved with a viscous coupling and is then dependent on the differential speed.

The importance of self-locking differentials is declining. They are increasingly being displaced by electronic systems which slow down a spinning wheel by applying the brake and thereby transfer power transmission to the wheel with more grip (e.g. Traction Control System (TCS)).

Passenger-car rear-axle final-drive unit (ZF HAG210)
1 Housing (two-piece),
2 Halfshaft flange,
3 Differential-case bearings,
4 Differential gears,
5 Crownwheel and pinion,
6 Pinion bearing,
7 Propeller-shaft flange.





Two-stage AWD transfer box for light off-road vehicles
(Steyr-Daimler-Puch ITC)
1 Pump,
2 Transfer-box input shaft,
3 Chain,
4 Front-axle driveshaft,
5 Rear-wheel driveshaft,
6 Torque-sensing transfer gearing,
7 2-speed planetary gearing,
8 Shift mechanism for changing between highway and off-road transmission ratio.

All-wheel drive and transfer case

All-wheel drive (AWD) improves traction on cars, off-road vehicles and commercial vehicles on wet and slippery road surfaces and rough terrain. The following types of system are available:

- Disengageable all-wheel drive with either direct transmission to front and rear axles, or with transfer box. The transfer box and final-drive differentials can also have a disengageable lock. Transfer cases for off-road vehicles incorporate an additional driver-controlled conversion range for steep gradients and low speeds.
- Permanent all-wheel drive whereby all wheels are driven at all times. Central transfer box, either non-locking or with friction-type torque-dependent locking mechanism, Torsen lock or viscous coupling. Torque distribution between front and rear axles 50:50 or asymmetrical. Additional crawler-gear ratios are also possible.

Designs with viscous coupling or electronically controlled multiplate clutch instead of the central all-wheel-drive transfer box also come under the umbrella of permanent all-wheel drive.

On more recent vehicles, additional locks on the AWD transfer box or axle differentials are increasingly being dispensed with in favor of intelligently controlled operation of the brakes.

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TCS traction control

During standing-start and moving acceleration and under braking, the efficiency with which forces can be transferred to the road depends upon the traction available between tires and road surface. The adhesion/ slip curves for acceleration and braking display the same basic pattern as each other (see illustration). The vast majority of acceleration and braking operations entail only limited amounts of slip, allowing response to remain within the stable range in the diagram where up to a certain point any rise in slip is accompanied by a corresponding increase in available adhesion. Beyond this point, further increases in slip take the cuves through the maxima and into the instable range, in which any further increase in slip will generally result in a reduction of adhesion. Under braking, this will result in the wheel locking within a few tenths of a second. Under acceleration, one or both of the driving wheels will start to spin more and more as the drive torque exceeds the adhesion by an ever increasing amount.

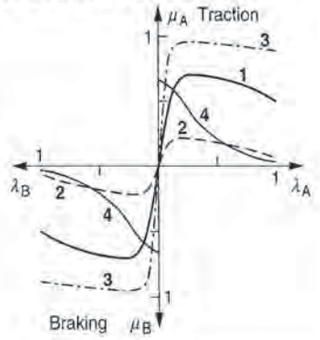
ABS responds to the first case (braking) by inhibiting wheel lock. ASR reacts to the second scenario by holding acceleration slip within acceptable levels to prevent wheel spin. The system actually performs two functions:
– enhancing traction
– maintaining vehicle stability (true tracking).

Traction-control systems for passenger cars

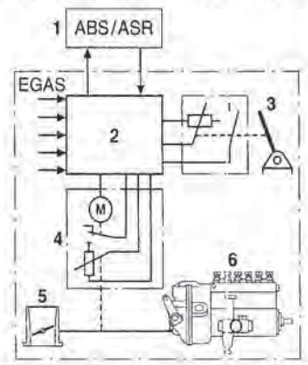
ABS/TCS 2I (Bosch)

To provide optimal closed-loop control of torque at the drive wheels, the mechanical connection between accelerator pedal and throttle valve (or pedal and injection-pump control lever on diesel engines) is replaced by the ETC electronic throttle control (EGAS or electronic "drive-by-wire" accelerator). A sensor converts the position of the accelerator pedal into an electrical signal, which the ECU then uses to generate a control voltage. A servomotor responds to this signal by repositioning the throttle valve (or injection-pump control lever on diesels); it then transmits a position report back to the ECU. Brief, simultaneous activation of the service brakes is employed to supplement the ETC (improved tractive performance via limited-slip effect). The standard ABS hydraulic modulator is expanded to include a TCS section, both to provice additional hydraulic energy for brake-force application and for switching to TCS operation. The ABS solenoid valves in the hydraulic modulator switch between three positions – "pressure buildup", "maintain pressure" and "discharge" – to regulate the flow of

Adhesion/slip curves
Coefficient of friction under acceleration/ braking $\mu_{A,B}$, Torque under acceleration/ braking $\lambda_{A,B}$, Lateral-force coefficient μ_S, 1 Wet asphalt $\mu_{A,B}$, 2 Hard packed snow $\mu_{A,B}$, 3 Concrete $\mu_{A,B}$, 4 Wet asphalt μ_S.

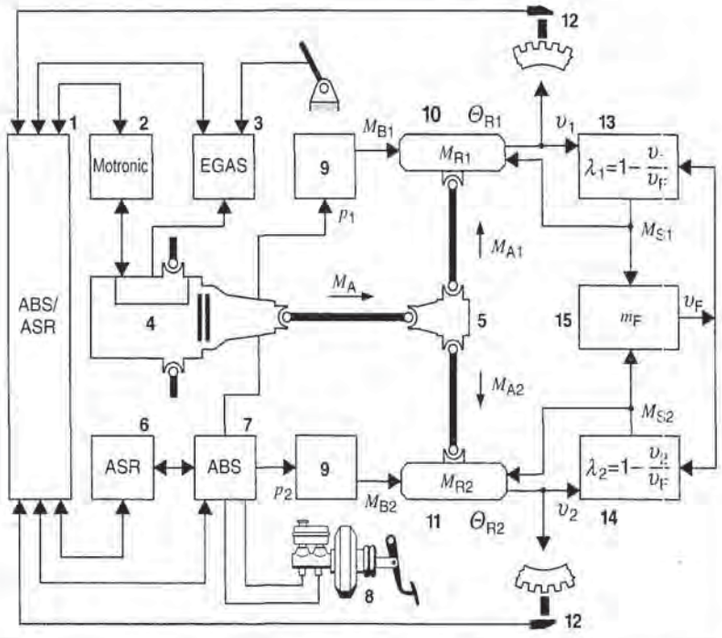


EMS electronic engine-power control for TCS
1 ABS/TCS control unit, 2 ETC control unit, 3 Accelerator pedal, 4 Servomotor, 5 Throttle valve, or 6 Diesel injection pump.

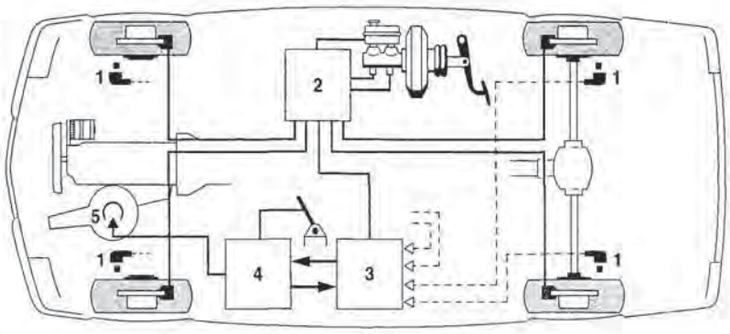


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ABS/TCS schematic diagram for cars
1 ABS/TCS control unit, 2 Motronic ECU, 3 ETC control unit, 4 Engine, clutch, transmission, 5 Differential, 6 TCS pressure source, 7 ABS hydraulic modulator, 8 Brake master cylinder with brake booster, 9 Wheel brakes, 10 Wheel 1, 11 Wheel 2, 12 Wheel-speed sensor, 13 Road surface, Wheel 1, 14 Road surface, Wheel 2, 15 Vehicle mass m_F, p Brake pressure, υ Wheel speed, υ_F Vehicle speed, λ Slip, θ_R Wheel inertia, M_A Drive force, M_B Braking force, M_R Total forces acting on drive wheel, M_S Surface forces. Indices 1, 2: Wheel 1, 2.



ABS/TCS system components for cars
1 Speed sensor, 2 ABS/TCS hydraulic modulator, 3 ABS/TCS control unit, 4 ETC control unit, 5 Throttle (gasoline engine)/Diesel fuel-injection pump control lever.



system pressure and furnish the pressure modulation required for rapid and precise control of braking force at the drive wheels.

The ETC control unit, which is connected to the ABS/TCS control unit via an interface, takes over control of the throttle or the diesel fuel-injection pump control lever in order to control engine torque.

On spark-ignition engines, system response is improved by retarding ignition timing to reduce the relatively extended delays encountered when engine torque is controlled exclusively with the throttle valve.

ABS/TCS5 (Bosch)

By designing this ABS system around solenoid valves featuring only 2 hydraulic connections and 2 switch positions (2/2 solenoid valve) it was possible to achieve further reductions in both weight and costs – while combining the complete range of standard functions with the advantages of sealed braking circuits.

Because the hydraulic unit's design is based on a modular concept, it is easy to adapt for use with both standard brake-system configurations (II and X pattern) and with different vehicle drive layouts (including all-wheel drive).

The design of the 2/2 solenoid valves allows the sealed hydraulic valve section to be inserted into its energizer coil. This considerably simplifies the design of an add-on ECU for the hydraulic modulator, and separate testing of the two components now becomes possible.

This direct connection of the ECU with the solenoid valve coils substantially simplifies the wiring-harness layout in the vehicle by reducing the number of connectors and leads required. This in turn considerably reduces susceptibility to faults and thereby the probability of failure.

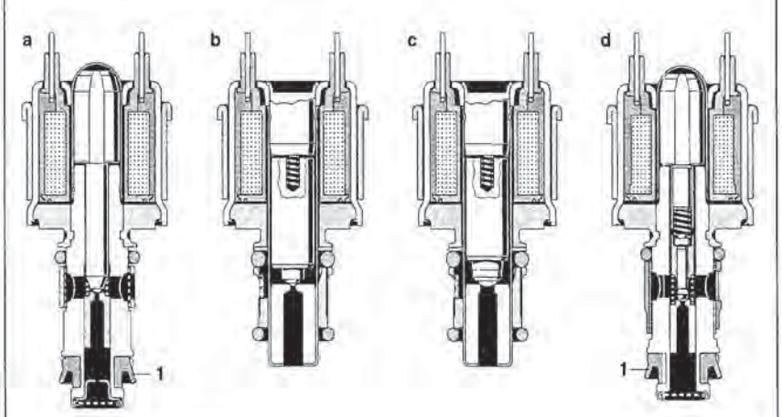
The engine-control functions remain largely independent of the hydraulic system, and the current design was adopted for ABS/TCS5 without modification.

MSR engine drag-torque control

The TCS unit installed in passenger cars can be expanded to include the MSR engine drag-torque control system. On slick road surfaces, a downshift or sudden throttle closure can cause excessive engine braking at the drive wheels.

MSR responds to these conditions by gently adjusting the throttle-valve for slightly increased engine torque in order to reduce the braking forces and help maintain maximum vehicle stability.

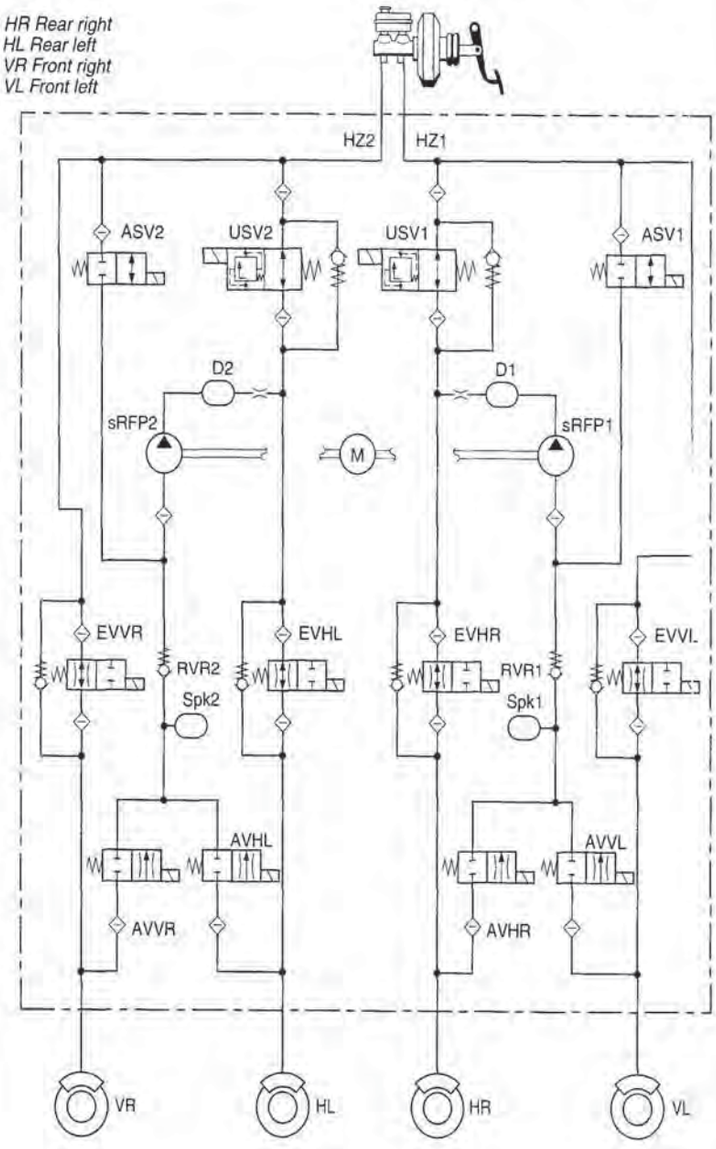
2/2 Solenoid valves for ABS/TCS5
a Inlet valve, b Outlet valve, c Intake valve, d Pilot valve.
1 Non-return valve sleeve.



Schematic diagram of ABS/TCS5 hydraulics for X brake-circuit configuration

HZ1, 2 *Main-cylinder lines,*
USV1, 2 *Circuit-control valves,*
D1, D2 *Attenuator chambers,*
RVR1, 2 *Vacuum/safety valves,*
AV (HR, HL, VR, VL*) *Pressure-relief valves,*
ASV1, 2 *Suction valves,*
sRFP1, 2 *Self-priming return pump,*
Spk 1, 2 *Accumulator chambers,*
EV (HR, HL, VR, VL*) *Pressurization valves,*
HR, HL, VR, VL *Wheel brakes.*

* *HR Rear right*
HL Rear left
VR Front right
VL Front left



TCS traction-control for commercial vehicles

This closed-loop traction-control system is integrated within the ABS control unit for shared use of ABS components such as wheel-speed sensors and pressure-control valves.

The control system consists of a brake-control circuit and an engine-control circuit. The TCS brake-control circuit also requires a two-way directional-control valve and a TCS valve (2/2 solenoid valve), and the engine-control circuit a final control element for reducing the engine torque.

Brake-control circuit

When the vehicle moves off on a low-traction or μ-split (left/right variations in traction) surface, the wheelspin that accompanies excessive throttle will frequently be limited to a single drive wheel. Due to the low coefficient of friction at this wheel, only minimal tractive forces will be available to move the vehicle. The brake controller responds to this situation by applying braking pressure to the spinning wheel; this force is conveyed through the differential and acts as drive torque at the (still) stationary wheel.

First, the ECU switches the TCS valve to the open position for initial braking at the spinning wheel. The control circuit then uses the ABS pressure-control valve to control the pressure at the wheel cylinder as a function of wheel behavior. The brake pressure is controlled in such a way that the driving wheels are synchronized. The result is a locking effect comparable to that provided by a mechanical limited-slip or locking differential. In order to achieve the same forward drive, however, the engine torque must be greater than required with a mechanical differential lock by an amount equivalent to the braking force applied by the TCS brake controller.

This is one reason why mechanical differential locks are frequently employed in difficult terrain (e.g., construction sites). The brake control function is then employed under those conditions where the driver might have difficulty in determining whether the differential lock is providing increased traction. It is often possible to dispense with the mechanical differential lock entirely on vehicles not used in demanding off-road conditions.

The benefits of the brake controller are generally felt when starting off, when accelerating, or in alpine conditions on "μ-split" surfaces. The control circuit must supply high braking forces to the low-traction wheel when a fully-loaded vehicle operates on extreme μ-split on mountain roads. To prevent excessive thermal loading of the brakes, TCS incorporates two default modes:

a) The brake controller remains inactive at speeds above 30 km/h,

b) Controller activity and wheel speed are used to estimate the thermal load at the brakes; if a defined limit is exceeded, the controller responds with deactivation.

Using the brake-control function described above, the driving wheels can also be synchronized so that a mechanical differential lock, if present, can be automatically engaged, e.g. with the aid of a pneumatic cylinder. The ABS/TCS control unit calculates the correct point and conditions for releasing the differential lock.

Engine-control circuit

Drive wheels on surfaces affording equal but limited traction will respond to excess throttle by spinning on both sides. The residual tractive force available for propelling the vehicle then shifts into the unstable section of the adhesion/slip curve with its diminishing adhesion coefficients (see Fig. "Adhesion/slip curve") Attempts to accelerate a stationary or crawling vehicle on ice or snow "polish" the surface, resulting in a further, substantial reduction of traction. At the same time vehicle stability is sacrificed. The engine-control circuit responds to these conditions by reducing the drive slip to an acceptable level, thereby enhancing traction and vehicular stability.

Both electric and pneumatic servo elements are available for controlling the reduction in engine torque.

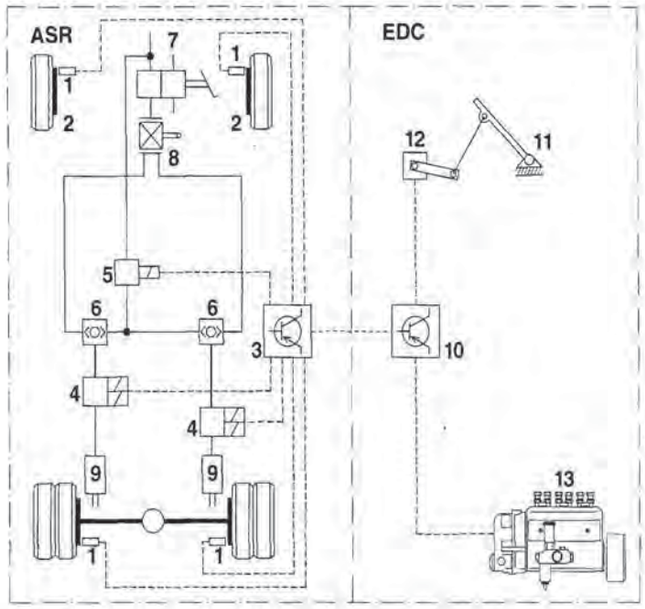
The following section illustrates the principles involved by describing two options for electric control:

- an interface connected to electronic engine-control devices,
- direct control of an electrical servomotor.

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Traction control system (TCS) for commercial vehicles with electronic injection-pump control (EDC)

1 Wheel-speed sensor, 2 Pulse ring, 3 ABS/TCS control unit, 4 Pressure control valve (single circuit), 5 2/2 Solenoid valve, 6 Two-way valve, 7 Service-brake valve, 8 Brake-pressure regulator, 9 Wheel cylinders, 10 EDC control unit, 11 Accelerator pedal, 12 Pedal-position sensor, 13 Injection pump.



Interface (refer to CAN on page 884): The ABS/TCS control unit receives the signals reflecting driver commands from one of the engine-management ECUs (e.g., accelerator-pedal position or desired fuel-injection quantity). The ABS/TCS control unit consults both this signal and various other data such as wheel slip in calculating the torque reduction requirement to be implemented by the engine-management ECU. Examples of engine-management ECUs are the ETC (Engine Throttle Control) and EDC (Electronic Diesel Control) units. They incorporate all engine management functions (e.g. vehicle-speed control, engine-speed limiting, idle-speed control, etc.) and immediately carry out the TCS request to reduce engine speed with the required degree of accuracy.

Servomotor the ABS/TCS control unit operates the servomotor directly. The servomotor is a DC device with integral position feedback for precise position control; regulation thus remains unaffected by variables such as positioning forces at the injection pump and friction in the throttle linkage. The TCS can only use the linkage to reduce throttle, thereby excluding the possibility of inadvertent opening of the throttle. A vehicle-speed limiter (FGB) integrated in the ECU either holds the vehicle's road speed at a statutory speed limit (v_{max} control) or at a speed set by the driver by means of a button (v_{set} control). Here, the driver must depress the accelerator beyond the position which would otherwise correspond to the desired v_{max} or v_{set} velocity. The ABS/TCS control unit compensates for the excess throttle. These FGB limiters have been mandatory equipment on buses (AGVW > 10 metric tons) and trucks (AGVW > 12 tons) since 1 January, 1994.

Suspension

Types of oscillation

Suspension springing and damping operate chiefly on the vertical oscillations of the vehicle. Driving comfort (oscillatory loads on passengers and cargo) and operating safety (distribution of forces against the road surface as wheel-load factors fluctuate) are determined by the suspension. Several spring-damper systems will serve to illustrate the synergetic operation of the vehicle components.
Driving comfort is largely determined by

Dual-mass model as quarter-vehicle model
h Excitation amplitude, z Oscillation amplitude, c Spring resistance, k Damping constants, m Mass. Subscripts: 1 Tires and axle, 2 Body.

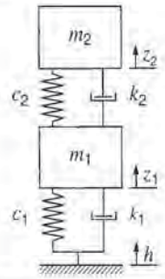


Table 1.
Effects of design parameters on the vehicle's vertical oscillations

Design variable	Effect at body natural frequency	Effect in intermediate frequency range	Effect at suspension natural frequency
Body specifications Spring constant	Large on ride comfort	Medium on ride comfort	Small on handling safety
harder	Frequency and amplitude increase, comfort decreases		Frequency increases, amplitude decreases slightly
softer	Frequency and maximum amplitude decrease, comfort increases		Amplitude increases slightly at low excitation frequencies
Damper constant	Major on ride comfort Optimization required		Major on wheel-load fluctuation
higher (damper harder)	Acceleration decreases	Acceleration increases	Acceleration increases, dynamic wheel-load fluctuation decreases
lower (damper softer)	Acceleration increases	Acceleration decreases	Amplitude decreases slightly, dynamic wheel-load fluctuation increases
Mass	Minor on amplification factor or wheel-load fluctuation; as payload increases, the acceleration amplification factor decreases (ride comfort is worse and relative wheel-load fluctuation higher with vehicle empty than with vehicle fully loaded)		
Wheel and tyre specifications Suspension (with tires getting softer)	Natural frequency and amplitude change hardly at all		Natural frequency and amplitude of body acceleration and wheel-load fluctuation decrease roughly in proportion to the reduction in vertical tire rigidity
Damping	Frequency and amplitude do not change with change in tire damping		Amplitude with body acceleration and wheel-load fluctuation decreases minimally with harder damping
	Due to heat generation, tire damping should be held to a minimum in order to allow long spring travel for soft tires		
Wheel mass	Reducing wheel mass hardly affects ride comfort		Small wheel mass increases handling safety

the degree of body oscillation. Root mean square of vertical body acceleration:

$\ddot{z}_2 / h$

Axle oscillations (and thus, indirectly, fluctuations in wheel load) are the salient factor in determining driving safety. Root mean square of vertical axle oscillations:

$\ddot{z}_1 / h$

Both types of motion are characterized by specific frequency relationships in the form of amplitude ratios.

Table 1 enumerates the relative effects of parameter variations in a dual-mass model (also applicable to actual vehicles).

Body-mounted springs and dampers also affect pitch and roll of the vehicle body as well as its vertical vibration characteristics.

Pitch: Gyration around the vehicle's transverse axis, such as that encountered under acceleration from rest (front-suspension expansion and rear-suspension compression). Pitching movements under acceleration or braking are reduced by the chosen suspension kinematics (suspension-link configuration).

Roll: Gyration around a longitudinal axis which generally runs from the lower front to the upper rear of the vehicle; roll motion occurs in response to steering inputs (suspension compresses at the outside and expands on the inside of the curve). Stabilizers (anti-roll bars) at front and rear axles reduce this effect.

Suspension design elements

(See Table 2)

Controlled suspension systems

Level-control system

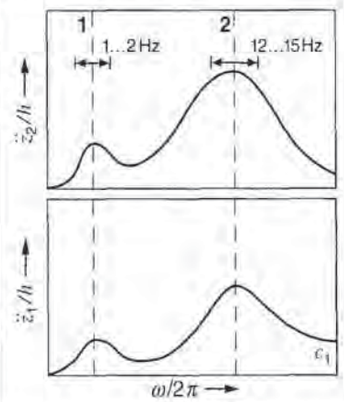
Partially loaded systems

The use of soft springs (comfort) results in long spring travel, such as that encountered with the vehicle loaded. In order to maintain vehicle-body height at an acceptable level, auxiliary air springs or hydropneumatic springs are employed.

The springing element is provided by the volume of the gas, while vehicle level is monitored mechanically directly at the suspension. Valves are used to control the input or output of the air or hydraulic fluid to/from the suspension element; the

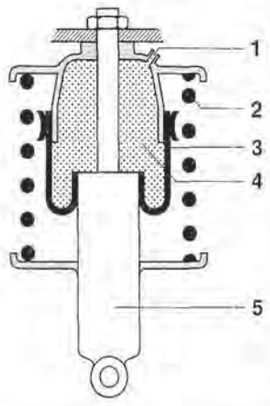
Effect of frequency on kinematic amplitudes

$\ddot{z}_1$ Amplitude of axle acceleration, $\ddot{z}_2$ Amplitude of body acceleration, h Excitation amplitude. 1 Natural body frequency, 2 Natural axle frequency.



Air-suspension level-control system (partially loaded system)

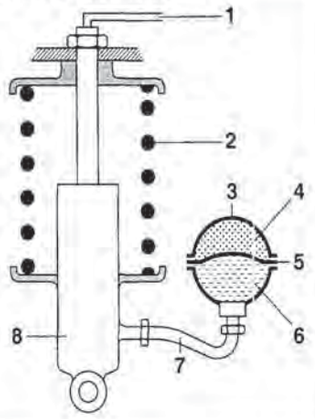
1 Air connection, 2 Steel spring, 3 Auxiliary air-spring element, 4 Gas volume, 5 Shock absorber.



system can also incorporate intermediate electronic level-control units acting upon solenoid valves.
Advantages of the electronic system:
- Reduced energy consumption achieved by avoiding control cycles during braking, acceleration and in curves,
- System reacts to increased vehicle speed by reducing body ride height, resulting in fuel savings,
- Increase in vehicle-body ride height on poor road surfaces,
- Enhanced stability in curves achieved through lateral blocking of the suspension elements on a given axle.

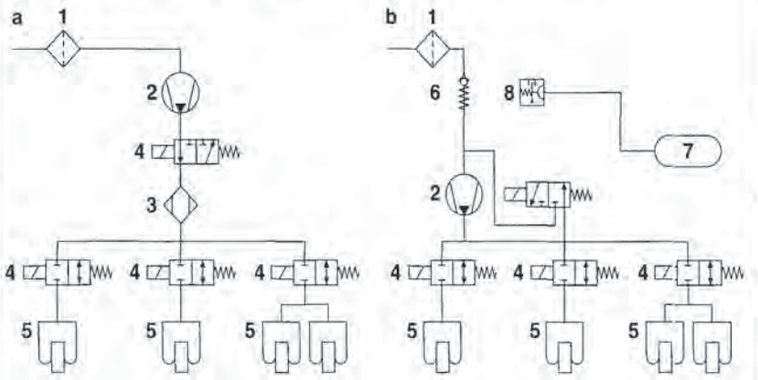
Additional advantages for commercial vehicles:
- Automatic limitation of suspension travel in order to accomodate interchangeable bodies and containers,
- Vehicle-body height can be adjusted as desired, e.g., to align load floor with loading ramps,
- Control of lifting axles (ride height raised automatically when lifting axle is elevated), lowers automatically when approved drive-axle weight is exceeded, lift axle raised briefly (2...3 minutes) to increase weight on drive axle (drive-traction increase).

Hydropneumatic level-control system (partially loaded system)
1 Fluid supply, 2 Steel spring, 3 Accumulator, 4 Gas volume, 5 Rubber diaphragm. 6 Fluid, 7 Hose, 8 Shock absorber.



Fully loaded suspension systems
The cushioning effect is provided by the gas suspension element alone and coil springs are dispensed with. Either a single axle (generally the rear axle) or both axles of the vehicle can be controlled. If all

Level-control system (fully loaded systems)
a) Open system, b) Closed system. 1 Filter, 2 Compressor, 3 Dryer, 4 Directional-control valve, 5 Bellows, 6 Non-return valve, 7 Compressed-air reservoir, 8 Pressure switch.



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axles are to be controlled, then the system must include an ECU with a specific control strategy which responds to factors such as fluctuations in axle load (to prevent the vehicle from tilting), while monitoring control times and recognizing system errors.

Open system:
Advantages: Relatively simple design and control.

Disadvantages: High compressor output required for brief periods of active control, air dryer required, noise during suction and blow-off.

Closed system:
Advantages: Low compressor output (minimal pressure differential between accumulator and suspension element), no problems with moisture.

Disadvantages: Relatively complicated design (accumulator, pressure switch, non-return valve).

Air-filled suspension elements weigh less than their hydropneumatic counterparts.

Active suspension

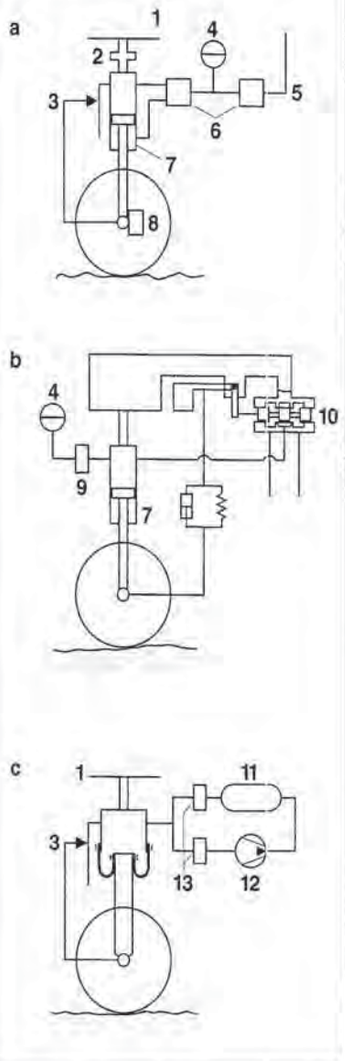
Active suspension controls both the "springing" and the "damping" functions. Various types of design concepts have been realized.

Designs incorporating hydraulic cylinder

A source of auxiliary power generates the energy needed for rapid adjustments using high-speed hydraulic cylinders, while sensors provide the link between the cylinders and the vehicle body. Sensors for wheel load, travel and acceleration transmit signals to an ECU featuring a control cycle of just a few milliseconds.

The control system achieves virtually constant wheel-load factors while maintaining a constant mean vehicle-body height. Steel springs or hydropneumatic suspension elements are employed to support the static wheel load.

Active suspension
a) Hydraulic cylinder,
b) Hydropneumatic suspension,
c) Air suspension.
1 Vehicle body, 2 Wheel-load sensor, 3 Travel sensor, 4 Accumulator, 5 Pumping circuit, 6 Servo valve, 7 Actuating cylinder, 8 Acceleration sensor, 9 Damper, 10 Valve, 11 Compressed-air reservoir, 12 Compressor, 13 Solenoid valve.



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Designs incorporating hydropneumatic suspension (fluid control)

Vehicle-body oscillations are compensated for by selective control of the hydraulic fluid in the hydropneumatic suspension circuit. The hydraulic fluid is either directed to the hydropneumatic element or discharged from the spring strut. In order to reduce energy requirements, the system's action is restricted to smoothing wide-spaced undulations; a gas accumulator installed adjacent to the spring strut is responsible for higher-frequency spring reaction.

The damping element can be set to concentrate on wheel movements.

Air-suspension designs

Controlled supply of compressed air to the air-spring bellows smoothes out the body vibrations. The sealed air-spring system only handles low-frequency vibrations and steering-related body roll.

Because the system compensates for transverse forces, it is also suitable for application with spring-strut axles.

Shock absorbers

Telescopic shock absorbers convert oscillations in body and suspension into heat. The resilient mountings used to attach them to body and axle also provide for noise insulation.

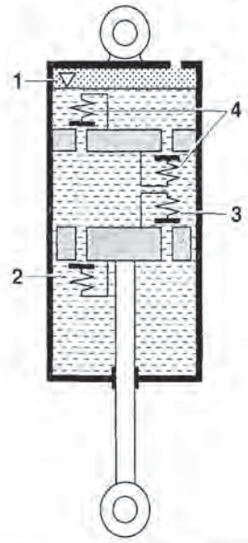
Single-tube shock absorbers

A sliding separating-piston and gas cushion form the gas-pressure damper in the single-tube shock absorber.

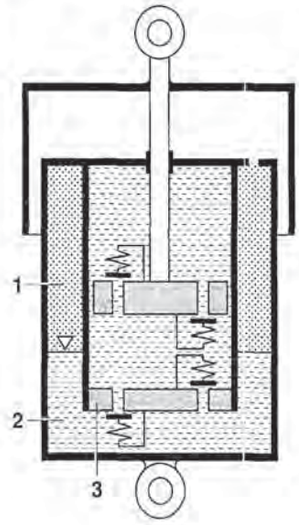
Advantages: Easy to tailor to specific applications, as the large piston diameter allows low working pressures. Sufficient room for valves and passages. Heat is dissipated directly via the outer tube. The shock absorber can be installed in any position.

Disadvantages: Length. The outer tube, which acts as a guide cylinder for the piston, is susceptible to damage from stone throw etc. Cannot be shortened in order to fit in positions where space is limited. The piston-rod seal is subjected to the damping pressure.

Single-tube shock absorber
1 Atmospheric pressure, 2 Valve, soft spring, 3 Valve, hard spring, 4 Non-return valve.



Twin-tube shock absorber
1 Atmospheric pressure, 2 Reserve chamber, 3 Base valve.



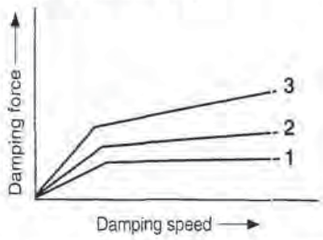
Twin-tube shock absorber

This type of damper is available either as the atmospheric or low-pressure twin-tube type.

Advantages: Insensitive to stone throw. Tube can be adapted to fit in locations where space is limited. This shock-absorber type is short, as the balance chamber is next to the working cylinder.

Disadvantages: These shock absorbers are sensitive to overloading (damping stops). Only certain specific installation positions are possible.

Shock-absorber curves (rebound modes)
1 Comfort, 2 Standard, 3 Sport.



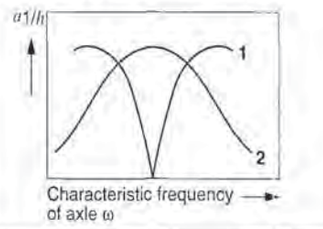
Damping characteristics

The damping characteristics are the result of the cumulative function of orifice damping and of the spring-loaded valve which closes the passage; the spring responds to pressure by increasing the free aperture of the outlet orifice. Piston bore and spring can be specifically tailored to provide linear to mildly degressive damping curves. An internal adjustment mechanism can be used to obtain several performance curves from a single shock absorber. The compression-stage values are frequently only 30...50 % of those for the rebound mode.

Electronically-controlled adjustable shock absorbers (active adaptation to operating conditions) can be used to enhance driving comfort and safety. Fixed damping parameters, on the other hand, result in defined relationships between comfort and safety.

The control law is frequently a semi-active "Skyhook" type, in which the shock absorber adjusts with reference to body speed.

Amplification function of axle acceleration
1 With absorber, 2 Without absorber.
a_1 Amplitude of z_1, h Excitation amplitude.

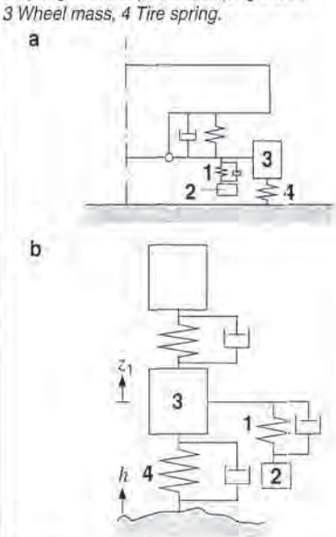


Vibration absorber

The vibration absorber is a supplementary mass providing both springing and damping. The vibration absorber assimilates the oscillations of the main system, i. e., the main system ceases to oscillate – the motion is restricted to the vibration damper (see "Oscillations" P. 29).

Vibration dampers affect body vibrations, their effect on the suspension is minimal.

Vibration absorption
a) Installation on vehicle,
b) Mechanical equivalent system.
1 Spring and damper, 2 Damping mass,
3 Wheel mass, 4 Tire spring.



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Suspension design elements (Table 2)

Suspension design elements	Diagram		Effect of vehicle load on natural frequency of body	Characteristics
Steel springs				
Leaf spring	Car leaf spring	Truck leaf spring with auxiliary spring	$\frac{\nu_{loaded}}{\nu_{empty}} = \sqrt{\frac{m_{empty}}{m_{loaded}}}$	Single or multi-leaf. Is also used to determine wheel travel in some applications. Interleaf friction in some types, can be reduced with plastic inserts (noise possible). Generally without inserts in trucks, maintenance required. Good transfer of forces to chassis.
Coil spring	Cylindrical coil spring	Barrel-shaped coil spring	Natural frequency decreases with increase of payload. Characteristics are generally linear.	Progressive characteristic achieved by change of pitch or conical coil. Shock absorber can be fitted inside spring. No self-damping. Spring noise possible. Advantages: Limited space requirement, low weight, maintenance-free. Disadvantages: special suspension links required.
Torsion bar				Made of round or flat-bar steel (round section is lighter). Vehicle height adjustment possible, depending on design. Wear and maintenance-free. Flat-steel bundle employed for additional flexural loads.
Air spring				
Roll-gaiter air spring	Element with constant gas volume. *1 Chassis frame, 2 Roll-gaiter air spring, 3 Piston, 4 Compressed air, 5 Mounting plate.*		$\frac{\nu_{loaded}}{\nu_{empty}} = 1$ Natural frequency remains constant when vehicle loaded. Characteristics are dependent on gas properties, piston shape, direction of plies in the gaiter.	Used as suspension strut or spring particularly on commercial vehicles and busses. Increasingly widespread use in passenger cars for level control at rear axle and for suspension on all wheels. Achieves high vertical compliance (increased comfort). Wheel travel must be defined by separate suspension components. Low pressure (<10 bar) demands large volume. Low vertical spring rate not achievable due to geometry of toroid air bellows.
Toroid bellows	Roll bellows	Toroid bellows		

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Hydropneumatic suspension Hydraulic diaphragm accumulator Piston accumulator	Hydropneumatic spring element with constant gas mass. *1 Gas, 2 Oil, 3 Diaphragm, 4 Steel spring.* Hydraulic diaphragm accumulator Piston accumulator 	$\frac{v_{loaded}}{v_{empty}} = \sqrt{\frac{m_{loaded}}{m_{empty}}}$ Natural frequency increases with payload. Characteristics are progressive and dependent on initial charge pressure of reservoir.	The gas volume in a reservoir (separated from oil by a piston) determines its spring characteristics. The fluid compresses the gas according to wheel load. Damper valves are integrated in the shock absorber and the junction between suspension strut and reservoir. The rubber diaphragm requires maintenance due to gas diffusion.
Rubber springs		Natural frequency is affected by payload due to the non-linear spring characteristics.	Vulcanized rubber compression spring between metal components, increasingly with integral hydraulic damper. Used as engine/transmission mountings, suspension-arm mountings and auxiliary springs.
Stabilizer		No effect if both sides simultaneously moved in same direction. If one side moved up/down half of anti-roll bar rigidity effective, if two sides moved in opposing directions full rigidity of anti-roll bar effective.	Reduces body-roll tendency and influences vehicle cornering characteristics (over or understeer). Generally made of U-shaped bar or tube, sides frequently flattened to cope with flexural loads. Attachment points must be at extreme outside of axle to allow minimal stabilizer diameter. Suspension link fulcrums must be arranged relative to the anti-roll bar in such a way that it is subjected only to torsion and not bending.

Suspension linkage

The suspension layout employed to connect the individual wheels to the vehicle structure defines the suspension-linkage design. The linkage enables the wheel for the most part to move vertically in order to compensate for irregularities in the road surface. In addition, the front wheels are steered, together with the rear wheels in the case of rear-wheel-steered vehicles. The front and rear suspension linkage systems differ due to the special demands imposed by the steering mechanism. Appropriate design measures are taken for the suspension geometry and springing so as to limit vertical body travel while also reducing pitch and roll.

Suggested reading:
Wolfgang Matschinsky.
Radführungen der Straßenfahrzeuge.
("Suspension linkage for road vehicles").
2nd. Edition; Springer-Verlag 1998.

Kinematics

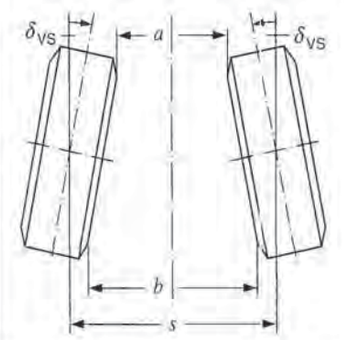
The front wheels pivot around an inclined axis whose position is determined by the joints and the suspension components.

The following kinematic data are of essential importance for the wheel's response to steering input and the transfer of forces between tire and road surface:

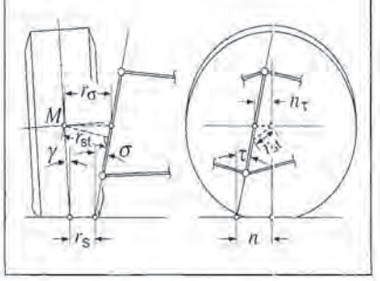
Toe-in δ_{vs}
Toe-in is the angle between the vehicle's longitudinal axis and a plane through the center of the (steered-wheel) tire. It can also be defined as half the difference in the distance between the wheel's front and rear rim flange. Toe-in affects the straight-ahead running as well as the steering, and in the case of front-wheel-drive vehicles compensates for the resulting elastokinematic changes in track. For standard-drive vehicles the toe-in is approx. 5...20'. For FWD vehicles, the toe-out is up to −20' (to compensate for drive forces).

Deflection-force lever arm r_{st}
The deflection-force lever arm is the shortest distance between the wheel center and the inclination angle of the steering axis. On FWD vehicles, its length provides an index of the effect of the drive forces on the steering and therefore of the rolling-resistance forces.

Toe-in
δ_{vs} Toe-in angle, Distance between wheels: a Front, b Rear, b–a Toe-in (in mm), s Track.



Wheel position
M Center of wheel, r_{st} Radius of deflection leverage, n_τ Caster offset, n Positive caster, τ Caster angle, r_σ Kingpin inclination offset, r_s Kingpin offset, γ Camber angle, σ Kingpin angle.



Caster n
Positive caster is the distance between the wheel's contact point and the point at which the steering axis intersects the road as viewed from the side. Caster determines the degree of self-centering action in the steering as well as influencing straight-line stability and steering force in curves.

Caster angle τ
The caster angle is defined as the angle between the steering axis and the vertical plane as viewed from the side. In conjunction with the kingpin angle, it influ-

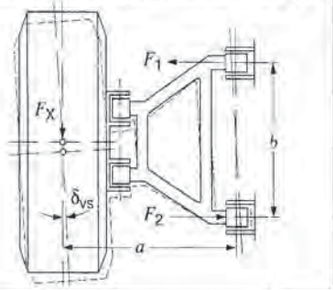
ences the camber change as a function of the steering angle, as well as the self-centering characteristics of the steering.

Kingpin offset r_s
The kingpin offset, or steering offset, is the distance between the point of contact between wheel and road surface, and the point at which the steering axis intersects the road surface. It is negative when the point of steering-axis intersection is between the center and the outside of the wheel. Kingpin offset combines with the effects of braking forces to generate steering motions and self-centering action at the steering wheel (driver information). Negative kingpin offset results in self-stabilizing steering angles.

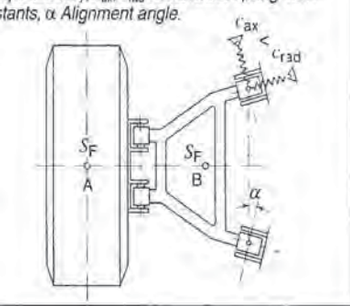
Camber angle γ
Camber angle is the inclination of the wheel axis towards the road surface in the vertical plane. The camber angle is negative when the top of the wheel is inclined toward the center of the vehicle. It influences lateral control (today's vehicles generally employ moderate cambers of –3...0° out of consideration for the tires).

Kingpin angle σ
The kingpin angle is the angle between the steering axis and the vehicle's longitudinal plane, as measured in the vehicle's transverse plane. It influences steering force (steering feel) along with caster angle and offset, and the steering roll radius.

Wheel displacement by longitudinal force F_x due to system flexibility
F_1, F_2 Suspension-arm mounting forces, a Distance between wheel axes and suspension arm, b Suspension arm mounting separation, δ_{vs} Toe-in angle.



Positioning of suspension-arm mountings to counteract steering deflection due to flexibility
S_F Center of gravity (A Reduction, B Complete equalization), c_{ax}, c_{rad} Axial/radial spring constants, α Alignment angle.



Basic types

(see Tables on following pages).

Elastokinematics

The wheel's rack and camber are influenced by the kinematic properties of the suspension geometry and by the variations in kinematic properties attendant upon spring compression. The forces acting on the suspension (acceleration, deceleration, vertical and lateral forces, rolling resistance) also have an effect upon the wheel's dynamic position due to compliance in mountings and in the components themselves (kinematic response due to elastic deformation: elastokinematics). The usual objective is to aim at specific changes in wheel position by employing kinematic and elastokinematic effects. To this end, kinematic and elastokinematic properties are designed into the axle which are calculated to provide a self-compensating response to force and spring action. Specially-aligned component mountings result in cardanic angles during springing. According to the design of the rubber mount they define the options for the compensation of elastokinematic effects.

Some modern rear suspensions employ elastokinematics to reduce reaction to changes in dynamic load. Differing longitudinal and vertical forces at the wheels allow the flexibly-mounted axle carrier or individual locating member to pivot, providing the wheels at the outside of the curve with more toe-in (stabilizing steering effect at the rear wheels).

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Basic suspension types and their characteristics

Rigid axles				
Leaf spring	A-arm	Watt linkage	Panhard rod	Panhard rod
	Trailing arm	A-bracket	A-bracket Trailing arm	Trailing arm
Used for rear axle with conventional (rear-wheel) drive; for front and rear axle on heavy and off-road vehicles				
Track, toe-in, camber remain constant relative to road surface, even with body roll, good tracking				
Low manufacturing costs, axle tramp, high unsprung masses, unfavorable deformation under lateral force and torsion	No lateral body movements during suspension travel, no unfavorable wheel displacement by transverse and longitudinal forces or torsion, large space requirements			
	Unrestricted choice of pitch fulcrum High cost and weight		Panhard rod causes lateral body movements when vehicle is driven	Unrestricted choice of pitch fulcrum
Semi-rigid axles				
Torsion-beam axle	Torsion-beam trailing-arm axle	Trailing-arm torsion-beam axle		
Employed for rear axle in conjunction with front-wheel drive				
Large distance between mounts minimizes structural stresses, favorable force transfer at rigid longitudinal members, simple manufacture, two attachment points, simple assembly, extremely robust, limited kinematic possibilities				
RA depends on position of Panhard rod above wheel centre	RA below center of wheel (depending on link position)	RA on road surface (all RAs aligned with vehicle center)	RA = Roll axis	

Independent suspension

Trailing arm

Semi-trailing arm

Semi-trailing arm

Swing axle

Swing axle

Rear-axle designs for front or rear-wheel drive

Small space requirements, low costs, limited kinematic versatility, change in camber, large change in caster trail, position of torque fulcrum, high stresses

Low manufacturing costs, favorable options in terms of kinematics, elasto-kinematically poor, lateral force and circumferential force oversteer, high suspension-link forces

Low costs, limited options in terms of kinematics, body is lifted by lateral force when cornering, vertical bracing effect with positive camber

Mc Pherson strut

Transverse link

Upper and lower A-arms

Lateral and transverse links

Used for front or rear suspension on FWD and standard RWD vehicles;

Small space requirements (vehicle width), low body forces due to large support base, small number of joints, easy to fit, lightweight, broad tolerances, limited kinematic versatility with regard to change in camber, kingpin angle, transient and pitch fulcrum, space required for springs, tire width, height

Greatest degree of kinematic latitude; high cost due to large number of suspension links; limited structural tolerances (without subframe); due to relatively small distances between mounting points, rigid mountings are required in order to prevent excessive wheel deflection (loss of comfort)

Used for front suspension on FWD and standard RWD vehicles

Transfer of force from upper suspension arm to rigid bulkhead

Wheels

Modern vehicle wheels generally comprise a rim and nave or wheel disc. The rim is that part of the wheel on which the tire is mounted. The nave joins the rim to the wheel hub. Wheel size is primarily determined by the wheel-brake diameter and by the load-bearing capacity of the tire. The most important terms are: rim width, rim diameter, center hole, hole circle diameter, number of mounting holes, countersunk and stud design, and rim offset.

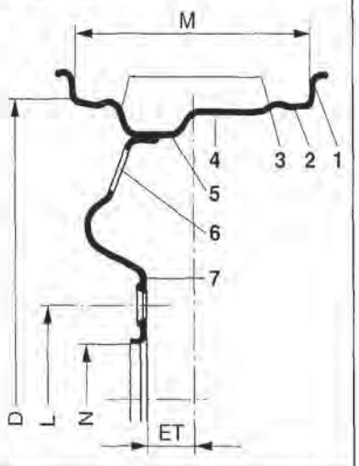
Rim designs
Rims differ from one another (depending upon the type of tire) in terms of number of parts and cross-sectional rim shape. The most important rim details are: rim flange, rim bead seat and rim base. Rims have one of the following cross-sectional shapes:
- Drop center,
- Flat base,
- 5° tapered bead seat or
- 15° tapered bead seat.

Wheel mounting
Mounting the wheel to the wheel hub fulfills two requirements: Centering the wheel in order to ensure that it runs true, and transmission of the wheel forces to the wheel hub. Correct wheel mounting affects the service life of the wheel system as a whole.

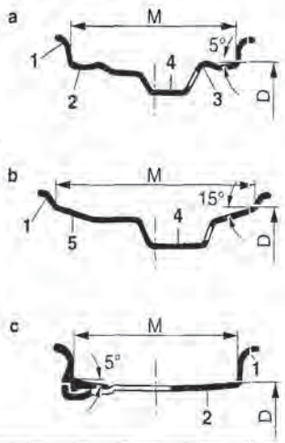
Passenger-car wheels

Mass-produced wheels are made of sheet steel, with forged or cast aluminum and (less commonly) magnesium being used primarily for special wheels and aftermarket wheels. Sheet aluminum, although used in some cases, has not become popular for reasons of cost.
The wheels in particular feature considerable potential when it comes to complying with the general trend towards lightweight construction in the automobile industry. The sheet-aluminum wheel (with the wheel cap in the form of a full cover or painted in silver as a styling variant), and the split wheel derived from the classical forged wheel, are among the lightest wheel versions while at the same time be-

Disc wheel *(e.g. 6J x 14 H 2)*
1 Rim flange (e.g. "J" section), 2 5° tapered bead seat, 3 Hump (e.g. double hump H 2), 4 Rim, 5 Rim well, 6 Vent hole, 7 Wheel disc. D *Rim diameter (e.g. 14"),* L *Hole circle diameter,* M *Rim width (e.g. 6"),* N *Center hole,* ET *Rim offset.*



Rim designs
a) Passenger-car hump rim as per DIN 7817, b) Truck 15° sloping bead seat rim (tubeless) as per DIN 78022, c) Truck 5° tapered bead seat rim as per DIN 7820.
1 Flange, 2 5° tapered bead seat, 3 Hump, 4 Rim well, 5 15° tapered bead seat. M *Rim width,* D *Diameter.*



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ing reasonably priced considering the weight savings involved.

Whereas the manufacturing process for the two-piece sheet-aluminum wheels and for the steel wheel are practically identical, a difference is made between two different versions of 1-piece split wheel:

1. An extruded section formed by a forging process. This is rolled and split at its circumference.
2. A circular blank stamped from the coil and deep-drawn.

There are further important lightweight-wheel versions on the market although these have still not become very widespread. They include the cast-aluminum wheel with pressure-rolled rim and the structured wheel which is designed primarily for maximum force transfer and less with regard to an attractive appearance. Since wheel covers in different materials, shapes and colors can be fitted, this provides a wide range of design possibilities.

The use of plastic as a material for wheels is still in the developmental stage due in particular to insufficient high-temperature strength and difficult wheel mounting and manufacture.

The disc and rim are welded together on sheet-steel wheels; in the case of forged and cast light-alloy wheels, these two components are usually manufactured in one piece. Multiple-piece designs, even those which are made of different materials (e.g., magnesium disc and aluminum rim), are available only in special cases and for racing vehicles. Car-wheel rims are almost always drop-centre rims with double humps H 2 (rarely with flat hump FH), tapered bead seats and "J" section rim flange. The lower flange shape is frequently found on smaller vehicles; the higher flange shapes JK and K are only rarely seen, and only on higher-weight vehicles.

More recent rim developments which have been put into series manufacture on a limited basis are the TR rim (with metric dimensions) developed by MICHELIN for use with matching TRX tires, which permit more room for the brakes, as well as the rim with "Denloc" groove, which also requires a matched tire. The Dunlop rim and tire combination provides improved safety in case of loss of air. The TD system (TRX-Denloc) brings together both wheel/tire systems. As opposed to common practice, in all three above designs incorporating other rim and tire versions is either not possible at all, or possible only to a very limited extent.

In a completely new development, the tire grips the outside of the rim (CONTINENTAL). This design enables the brake diameter to be increased considerably, although at the same time changing the way some tire/rim properties react with one another. It allows the driver to drive at reduced speed for several hundred kilometers with a flat tire or tires. This could eliminate the need for a spare tire This version has also not become popular on the market and its use is confined to special applications.

Based on American efforts to reduce the amount of space taken up by the spare tire as well as – to a very limited extent – to reduce overall vehicle weight, European vehicles are increasingly being fitted with a space-saving spare wheel, which has a matching, lower-performance tire. Opinions on the usefulness of this substandard combination vary.

Design criteria for passenger-car wheels include high component strength, efficient brake cooling, reliable wheel mounting, high concentricity, small space requirements, good corrosion protection, low weight, low cost, easy tire mounting, a good tire seat, good balance-weight attachment and aesthetically pleasing design (particularly in the case of light-alloy wheels). Recent attention has also been given to wheel designs which reduce the vehicle drag coefficient c_w.

Wheels are usually mounted to the vehicle by means of three to five wheel lugs or lug nuts whose collars have a special shape which conforms to the wheel which is usually centered. A high level of true-running is achieved by means of a central wheel mount at the wheel hub using an alignment shoulder. A central wheel mount designed as a Rudge hub or incorporating driving pins is only used with racing wheels.

Wheel trims are fixed onto the wheels with clips or (less frequently) by the wheel nuts in order to

– improve visual appearance,
– improve the drag coefficient,
– cool the wheels/and or the brakes.

Traditional materials for wheel covers are steel and aluminum. Plastics have re-

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cently been used much more frequently in order to reduce weight and cost.

Commercial-vehicle wheels

The primary requirements to be met by commercial-vehicle wheels are:
- high fatigue strength and long service life in order to maximize traffic safety,
- lowest possible wheel weight because the wheel, as an unsprung, rotating mass, influences the oscillation system "vehicle",
- high load capacity via appropriate wheel shape and use of optimum materials,
- reduction in wheel-disk unevenness,
- reduction of radial and lateral run-out and wheel imbalance,
- easy assembly of tire to rim during manufacture and in practical use.

15° tapered-bead seat rim
A modern technical design is the one-piece 15° tapered-bead seat rim for tubeless commercial-vehicle tyres.

Advantages:
- The one-piece wheel allows a reduction in wheel weight of up to 10% as compared to the two-piece rim, and exhibits improved true-running and less lateral run-out,
- increased rim diameter,
- sufficient free space,
- unified valve in a precise position and at a sufficient distance from the brake drum or caliper,
- unified balance-weight shapes,
- introduction of central centering and
- use of fully- or semi-automatic tire-fitting equipment.

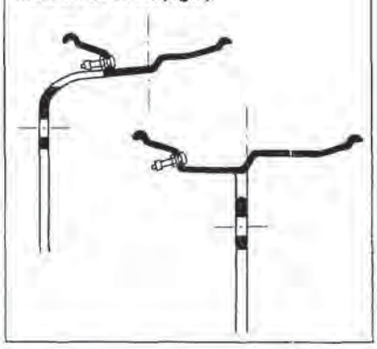
Radial and lateral run-out, imbalance
Radial run-out of the wheel is one of the main causes of vehicle vibrations. With today's commonly used 15° tapered-bead seat, the reduction of the permissible radial run-out to 1.25 mm (peak-to-peak) results in a marked improvement compared to the flat-bed rims. As opposed to radial run-out, lateral run-out is a less critical problem. Wheel imbalance is also not nearly as much of a problem as tire imbalance, which is of a greater magnitude. Maximum permissible static imbalance is limited to 2000 cmg.

Wheel centering
Lug centering via spherical washers or solely via ball-seat nuts has been replaced by central centering in order to reduce excessive radial run-out; this arrangement also permits different maximum and minimum play depending upon wheel size (especially with 22.5" wheels). This requires tolerances which are as close as possible.

Flatness of the contact surface
Any unevenness of the wheel contact surface (waviness, inclination, shielding, etc.) is transmitted to the brake drum when the wheel nuts are tightened, causing fluctuations in braking force as the wheel rotates. These fluctuations, in turn, cause vibrations in the steering system. Driving tests have led to the establishment of a maximum value for waviness of 0.15 mm and 0.2 mm for inclination from the center of the wheel outward.

Trilex wheel system
The Trilex wheel system with cast steel "spoked wheels" comprises a rim and spider. The removable rims are connected to the spider by means of clips and screws. In order to simplify mounting and dismounting of tube tires, the rim consists of three parts with transverse pitch. The same spider can be used in conjunction with the so-called Tublex rim for mounting tubeless tires.

MONOLEX wheel with rim offset (left) and without rim offset (right)



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A more recent development is the one-piece cast MONOLEX 15° tapered-bead seat wheel which in comparison with optimized-weight steel wheels is neutral in terms of weight effect.

Light-alloy wheels

One reason for the use of light-alloy wheels is the reduction in weight for 15° tapered-bead seat wheels. Light-alloy wheels are available with cast or forged rims. In spite of weight reduction and adequate strength characteristics, for reasons of cost, light-alloy wheels are used on commercial vehicles only in special cases.

Commercial-vehicle wheel loading

Prestress

Prestress occurs through the combination of stresses caused by assembling the wheel with those which result from inflating the tire.

Static nominal wheel stress

If the wheel is allowed to roll slowly on a perfectly flat road surface under static nominal wheel load, the stress in the wheel section in question varies periodically as the wheel rotates.

Additional dynamic stresses

These similar additional stresses are caused by dynamic wheel forces which result as the vehicle is driven straight ahead over an uneven road surface. Quasi-static wheel forces are then produced which occur during maneuvers such as cornering, steering while stationary, braking and accelerating.

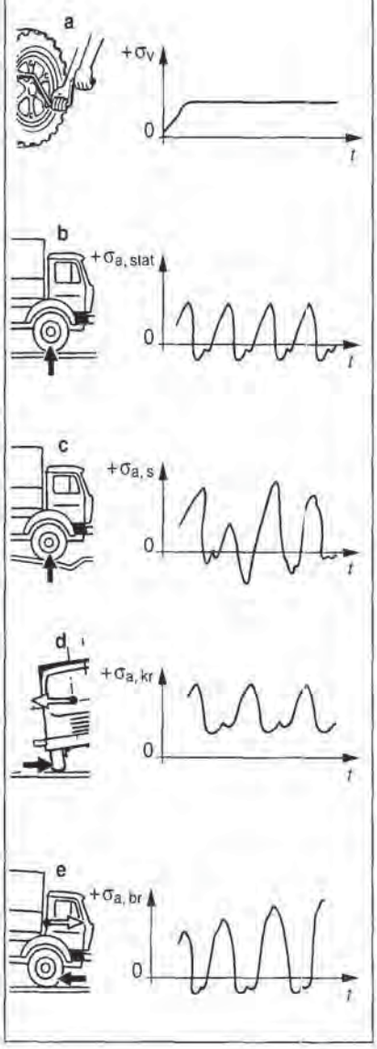
The resulting group of stresses caused by the above-mentioned types of wheel loading is used today as the basis for wheel dimensioning and testing.

Significant weak points

The highly stressed parts of the wheel which are thus most susceptible to cracking are the disc flange, the venting holes, the welded joint between disc and rim, and the rim-well radius.

The flange joint is particularly susceptible. Hub-centered wheels most often crack tangentially above the centre hole. With bolt-centered wheels, the cracks generally spread out radially from the bolt holes.

Types of wheel loading and stress distributions
a) Prestress σ_v
b) Static wheel load $\sigma_{a,stat}$
c) Straight-ahead travel over bumps $\sigma_{a,s}$
d) Cornering $\sigma_{a,kr}$
e) Braking $\sigma_{a,br}$



Tires

Tire categories

Tires are classified with reference to the respective requirements of various vehicle types and sizes, and operating conditions. The essential data – tire dimensions, load ratings, specified inflation pressures and authorized speeds – are standardized in the interests of interchangeability in the 7 tire groups or categories listed in Table 1. In addition to pneumatic tires, solid tires are also in use which are approved for speeds of up to 25 km/h (up to 16 km/h for unsprung drive wheels). The classifications in tire groups 1...4 are based upon road conditions:

- "Standard" highway (summer) tires,
- "Special" tyres for use off-road or for mixed use on and off the road,
- "M+S" (winter) tyres.

The same basic set of operating requirements applies for all tires (Table 2), although it should be noted that the emphasis shifts toward the final three criteria (esp. No. 6) on heavier vehicles.

Tire design

Today, passenger cars are equipped exclusively with radial tires. Bias-ply tires are now only fitted on motorcycles, bicycles, earthmoving equipment, and industrial and agricultural machines. For commercial vehicles they are increasingly becoming less important.

The once-dominant bias-ply tire received its name from the fact that referred to the tread, the cord threads of the tire casing run diagonally (or with a bias) so that they cross each other (cross ply). However, the more complex radial tire, with its radial design comprising two main sections, represents the only means of satisfying the increasingly variegated range of operating capabilities demanded of the tires used on today's passenger cars and heavy commercial vehicles.

The cords in the radial tire's casing layers run in the shortest and most direct path – radially – from bead to bead. A belt surrounds the relatively thin, elastic cas-

[1]) The corresponding European standards can be found in the "Data Book of Tires and Rims" of the ETRTO (European Tire and Rim Technical Organization, Brussels).
[2]) Guideline of the Economic Association of the German Rubber Industry, Frankfurt.

Table 1. Tire categories and applicable standards

No.	Tire application	German Standards (selection)[1]) DIN	WdK[2])
1	**Engine-driven, two-wheeled vehicles** Motorcycles, motor scooters, motorcycles of less than 50 cm³ engine displacement, mopeds	7801, 7802, 7810	119
2	**Passenger cars** Including station wagons and special spare tires	7803	128, 203
3	**Light-duty commercial vehicles** Including delivery trucks	7804	132, 133
4	**Commercial vehicles** Including multipurpose vehicles	7805, 7793	134, 135, 142, 143, 144, 153
5	**Earthmoving machines** Transport vehicles, loaders, graders	7798, 7799	145, 146
6	**Industrial trucks** Including solid rubber tires	7811, 7845	171
7	**Agricultural vehicles and machinery** Tractors, machines, implements, trailers	7807, 7808, 7813	156, 161

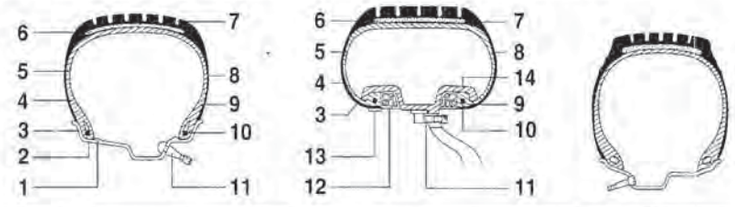
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Examples of radial-tire construction (tubeless version)
1 Hump, 2 Bead seat, 3 Flange, 4 Casing, 5 Airtight inner liner, 6 Belt, 7 Tread, 8 Sidewall, 9 Bead, 10 Bead core, 11 Valve, 12 Cover ring, 13 Balance weight, 14 Rim shoulder.

CT tires

Car radial tyres
Carcass: *two radially orientated rayon-cord plies* ***Belt:*** *two crossover steel-cord plies and two circumferential nylon-cord plies*

Truck radial tyres
Carcass: *one radially orientated steel-cord p'y*
Belt: *four crossover steel-cord plies*



ing in order to provide sufficient stability. The "bias-belted" design, more widespread in the US than elsewhere, incorporates an additional belt outside the diagonal casing. The performance of this type of tire is nowhere near as good as that of the radials.

CT tires feature run-flat properties. When deflated, the tyre presses against the inverted bead seat and can thus be driven for several 100 km when flat.

In the TRX/Denloc System, a specially-designed protrusion in the bead fits into a corresponding groove in the wheel rim. This arrangement locks the bead in position to prevent it from slipping into the center of the wheel, and thus provides limited run-flat properties.

Tubeless tires, already popular on passenger cars, are being used with increasing frequency on heavy commercial vehicles. The essential requirement is a single-piece, airtight wheel rim, or a multi-piece rim capable of accepting the required flexible rim-seal rings. Tubeless tyres have a vulcanized-on airtight inner layer instead of an inner tube.

The bead of the tubeless tire must press more tightly against the wheel rim in order to provide the necessary seal. Special, supplementary bead seals in the form of elastomer rings are sometimes used.

Dispensing with the inner tube reduces weight and simplifies mounting procedures.

Table 2. Service requirements

No.	Main criteria	Sub-criteria
1	Ride comfort	Soft suspension, low noise, smooth running (low out-of-roundness)
2	Steering behavior	Steering-force, steering precision[3])
3	Driving stability	Straight-ahead stability[3]), cornering stability[3])
4	Driving safety	Tire seat on rim, tire/road adhesion[3])
5	Durability	Structural stability, high-speed performance, bursting pressure, puncture resistance
6	Economy	Expected service life (mileage), wear pattern, sidewall wear, rolling resistance, retreadability

[3]) Main criteria for driving on roads in winter.

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Diameter and design of the wheel rim
With tyre types 3 and 4, non-sectional 15° tapered-bead seat rims (for tubeless tyres) have displaced the sectional 5° tapered-bead seat rim. This type of rim is identified by the diameter code (1 unit = 25.4 mm) ending in 0.5, as in 17.5, 19.5, 22.5.

Important codes for 5° tapered-bead seat rims are 16 and 20. For car tyres there are not only the normal drop-centre rims with whole-number diameter codes 10, 12, 13, etc. but also special designs with diameter measurements in mm.

Aspect ratio *H/W* (height-to-width ratio)
H:W ratio = (*H/W*) · 100
H Tire cross-sectional height
W Tire cross-sectional width

The aspect ratio (*H/W* = Height relative to Width) on today's standard passenger-car tires ranges between 80 and 50; the figure ranges down to 25 for sports-car tires, and is between 100 and 45 for the tires on heavy commercial vehicles.

Passenger-car tires with low *H/W* aspect ratios provide high levels of cornering stability. The different aspect ratios are all based on a single outside diameter in order to facilitate interchangeability. At a constant internal diameter for tire and wheel rim, a tire with a low *H/W* aspect ratio will be wider, with a larger contact surface, and it will be distinguished by a more impressive appearance.

A low *H/W* aspect ratio makes it possible to maintain the width of the footprint while increasing the internal diameter of the wheel rim; thus providing more space for the brakes. The introduction of the tapered drop-center rim for heavy commercial vehicles was only made possible by the development of a suitable tubeless tire with a low *H/W* aspect ratio. This was because a reduction of the rim-base diameter was impossible due to the brake-drum diameter having to remain unchanged.

Wider tires with low *H/W* aspect ratios also represent the only feasible option for applications where minimum tire diameters (in other words, maximum usable height) are stipulated, e.g., for container transport.

Tire designation symbols

The tire's designation is stamped on the sidewall (see Table 5), and reflects standards mandatory in a number of European countries. These include ECE Directive Number 30 for passenger-car tires; Number 54 for tires for heavy commercial vehicles (speeds of 80 km/h and above); and ECE Directive Number 75 concerning tires for motorized two-wheeled vehicles (vehicles with engine displacement exceeding 50 cm³ or attaining speeds in excess of 40 km/h).

Tires tested for compliance with ECE regulations are identified by the code molded into the sidewall adjacent to the bead. The code consists of a circle containing a large "E" followed by the code number of the responsible agency, which is in turn followed by an approval number.

Example: (E4) 020 427.

The small "e" which can be molded into the sidewall in accordance with the European guidelines 92/23 is of equal significance to the ECE marking

Example: (e4) 00321.

Appropriate European guidelines are being prepared for agricultural machinery.

The tire's width, its construction (R = Radial; "—" = Cross-ply; B = Bias-belted), and its rim diameter represent the minimum information requirement for tire designation. The diameter of the tire is also usually included on tires for industrial trucks. On tires for two-wheeled vehicles, passenger cars and heavy commercial vehicles, this information is frequently supplemented by the *H/W* aspect ratio in % which is appended directly behind the width information from which it is separated by a slash. ECE regulations require this information for all new tires. Although it is not stipulated by the ECE Directives, tires for passenger cars and two-wheeled vehicles may also feature a speed-rating code letter behind the *H/W* aspect ratio or tire width. On cross-ply tires the code letter replaces the horizontal line. On VR, VB, ZR and ZB tires, it is mandatory that the code letter forms an integral part of the tire-size designation.

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The PR (Ply Rating) follows the tire size, and is now employed as a code to distinguish the load ratings for various versions of the same size. Originally it indicated the number of plies in the tire carcass.

The service description is an additional suffix combining the load index (LI) and the speed-rating symbol (GSY). ECE Directives prescribe it as a replacement for the PR number or the speed-rating letters in the tire size. Specific values are assigned to the codes in the service description (Table 4).

For passenger-car tires:
Rated speed = Top speed. On vehicles whose design only allows top speeds of 60 km/h and below, the tire load ratings may be increased as shown in the table below so that higher loads can be carried.

Table 3. Load-rating increase

km/h	Load-rating increase in %	Air pressure increase in bar
60	10	0.1
50	15	0.2
40	25	0.3
30	35	0.4
25	42	0.5

Reductions in tire load can be traded for higher top speeds on most tires for motorscooters and heavy commercial vehicles. Conversely, both types of tire may be used to carry higher loads throughout virtually the entire speed range when operated below their rated (reference) speed, provided that the vehicle be specifically designed to operate at the lower maximum speed. Again, increased maximum load ratings are approved for tires on passenger-car trailers for speeds up to 100 km/h, and on certain commercial vehicles used for short-distance haulage and transportation.

The correct inflation pressure for a particular tire size and PR rating or service description are specified in the standards and/or the manuals provided by the tire's manufacturer (Table 1). The suffix containing the speed-rating symbol always indicates the actual rated speed.

Special passenger-car spare tires (low-weight and "space-saver" designs) are identified on the sidewall as intended for temporary use at limited speeds.

Correspondence between the speed rating for M+S tires on passenger cars, heavy commercial vehicles and motorcycles, and the respective maxima of the vehicles upon which they are mounted is not mandatory. However, a sticker indicating the lower speed for which the tires are approved must be affixed to the inside of the vehicle within the driver's field of vision. The following categories of tires may feature additional data prescribed by US highway safety legislation. These data, molded into the sidewall adjacent to the bead, are valid for Canada, and are also used in Israel:
- FMVSS 109 for passenger-car tires,
- FMVSS 119 for two-wheeled and heavy commercial vehicles.

These data are stamped adjacent to the letters "DOT", following which come the tire identification code and the date of manufacture, as well as further data on maximum load rating, maximum inflation pressure, and the cord plies which make up the casing and belts.

Australian Safety Regulation ADR 23, which applies to passenger-car tires, employs the identification codes from FMVSS 109 and ECE-R30.

Table 4. Service description codes (examples)

Load index

LI	50	51	88	89	112	113	145	149	157
kg	190	195	560	580	1120	1150	2900	3250	4125

Speed symbol

SS	F	G	J	K	L	M	N	P	Q	R	S	T	H	V	W	Y
km/h	80	90	100	110	120	130	140	150	160	170	180	190	210	240	270	300

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Table 5. Tire identification examples

Tire category		Identification example				Example includes details of			
		Tire designation symbols	PR[3] number	Usage code [4] LI	[5] GSY	Tyre dia. *A*	Tire width *B*	*H/W* %	Rim dia. *d*
	Suitability								
MC	Moped	2¼–16 Moped	–	–	–	–	Code	–	Code
	Small motor cycles	3–17 reinforced[2]	–	51	J	–	Code	–	Code
	Motorcycles	3.00–17 reinforced[2]	–	50	P	–	Code	–	Code
		110/80 R 18	–	58	H	–	mm	80	Code
		120/90 B 18	–	65	H	–	mm	90	Code
	Motor scooters	3.50–10	–	51	J	–	Code	–	Code
Passenger cars		165 R 14 M+S	–	84	Q	–	mm	–	Code
		195/65 R 15 reinforced[2]	–	95	T	–	mm	–	Code
		200/60 R 365	–	88	H	–	mm	60	mm
		205/60 ZR 15	–	91	W	–	mm	60	Code
		CT 235/40 ZR 475	–	–	–	–	mm	40	mm
CV	Delivery vans	185 R 14 C[1]	8 PR	102/100	M	–	mm	–	Code
	Light-duty trucks	245/70R 17.5	–	143/141	J	–	Code	–	Code
	Trucks	11/70 R 22.5	–	146/143	K	–	Code	70	Code
	Trailers	14/80 R 20	–	157	K	–	Code	80	Code
	Buses	295/80 R 22.5	–	149/145	M	–	mm	80	Code
MPV	Multi-purp. veh.	10.5 R 20 MPT[8]	14 PR	134	G	–	Code	–	Code
EM	Transporters	18.00–25 EM[9]	32 PR	–	–	–	Code	–	Code
	Loaders	29.5–29 EM[9]	28 PR	–	–	–	Code	–	Code
IT	Industrial trucks	6.50–10[6]	10 PR	–	–	–	Code	–	Code
	Carts	21 x 4[6]	4 PR	–	–	Code	Code	–	–
	Industrial trucks	28 x 9–15[7]	14 PR	–	–	Code	Code	–	Code
		300 x 15[7]	18 PR	–	–	–	mm	–	Code
AS	Tractors	480/70 R 34	–	143	A 8	–	mm	–	Code
		7.50–60 AS[10] Front	6 PR	–	–	–	Code	–	Code
	Equipment[11]	11.0/65–12 Impl.	6 PR	–	–	–	Code	65	Code

Example for passenger-car tires: **175/70R13 82S**
175 mm nominal tire width, **70** % H/W ratio, **R**adial tires, Code **13** Rim diameter (approx. 330 mm [13 inches]), Load index **82** (475 kg), Speed symbol **S** = 180 km/h.

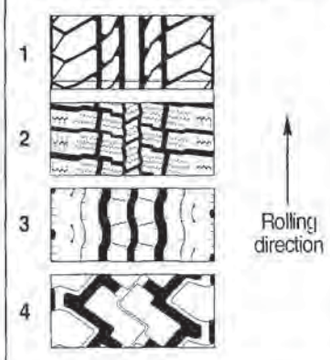
[1]) C = light-duty truck (delivery-van) tires (also for high-load-capacity motor-scooter tires). [2]) reinforced = additional designation for reinforced tires for two-wheeled motor vehicles and passenger cars. [3]) PR = load-range class. [4]) Load-range code for single/dual tires. [5]) Speed code for vehicle nominal (reference) speed. [6]) Pneumatic tires. [7]) Solid rubber tires. [8]) MPT = Multi-purpose tire. [9]) EM = Earthmoving machinery. [10]) AS = Agricultural tractor. [11]) Tires for equipment and trailers.

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Tire applications

Tire selection based upon the recommendations of the tire manufacturer is essential for obtaining satisfactory performance. Optimal operating characteristics can only be obtained when tires of one single design are mounted on all wheels (for instance, radial tires). This is obligatory for private cars. For commercial vehicles with a gross weight of over 2.8 t, tires on the same axle must be of identical design. During seasonal storage of the tires, inner tubes and bead bands tend to age and become brittle more rapidly when exposed to direct sunlight. Moving air promotes this process. Intact packaging is particularly important for ensuring that the tube remains in good condition. The storage area should thus be cool, dry and dark. Any contact with oil or grease is to be avoided. Particular care is required when mounting tires. They must never be mounted on anything other than undistorted, undamaged, rust-free wheel rims which show no signs of more than minimal wear. When a loose flange is used, that side should receive especially critical attention. New valves, and, where applicable, new tubes and bead bands are always to be installed with new tires. Caution is also advised when used inner tubes are refitted following repairs: Inner tubes expand during use, a condition which can result in dangerous folds forming when the tube is reinstalled. If there are any doubts at all, new tubes should be used.

Tire-tread examples
1 Passenger-car tire, 2 Passenger-car M+S tire, 3 Commercial-vehicle tire, 4 Commercial-vehicle tire, high-traction tread design.



Tire tread (see examples)
It is illegal to regroove the tread on tires for two-wheeled vehicles and passenger cars; manufacturer regulations must be complied with for other tire categories.
Tire rotation is recommended to deal with variations in tread wear between axles. The less tread there is on the tire, the thinner is the protective layer covering belts and casing. This aspect should be considered when planning operation under demanding operating conditions extending over a long period of time. In addition, reduced tread depth results in a more than proportional increase in braking distances on wet road surfaces. This applies in particular to passenger cars and fast commercial vehicles. For instance, for a light FWD passenger car and a heavy rear-wheel drive car, the figures apply in Table 6 below when the vehicle is braked from 100 km/h (highly dependent upon the road surface, tire profile, and the tire's rubber mixture).

Table 6. Tread depth and braking distances (braking from 100 km/h)

Vehicle		Light passenger car with front-wheel drive					Heavy passenger car with rear-wheel drive (ABS)			
Tread depth	in mm	8	4	3	2	1	8	3	1.6	1
Braking distance	in m	76	99	110	129	166	59	63	80	97
	in %	100	130	145	170	218	100	107	135	165
Increased braking distance per mm of wear	in %	7	15	25	48		1.4	20	50	

Tire traction

Units and symbols

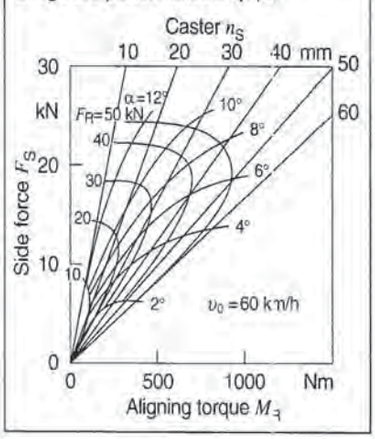
Symbol		Unit
f	Frequency	Hz
F_B	Braking force	kN
F_R	Wheel load	kN
F_S	Side force	kN
M_R	Aligning torque	N · m
n_S	Caster	mm
p_i	Tire pressure	bar
v_0	Test velocity	km/h
α	Slip angle	°
γ	Camber angle	°
λ	Slip	–

It is essential that accurate tire-performance maps are available for the engineering and optimization of the vehicle's handling, driveability, and comfort, and for the reduction of its drivetrain vibration.

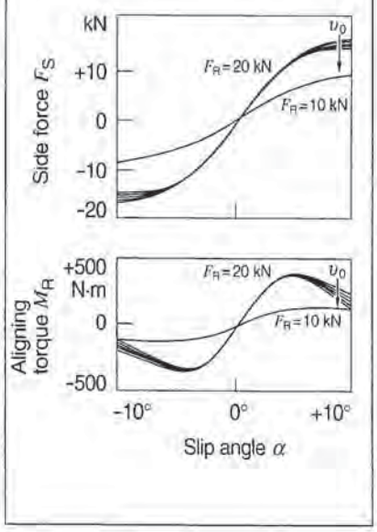
Representative tire-performance maps for passenger-cars and light commercial vehicles are already familiar and are available in various publications [1, 3, 4]. For this reason, the following will concentrate on tires for heavy-duty commercial vehicles with the dimensions 11 R 22.5. These tires are in widespread use [2].

All data in the performance maps refer to the Michelin XZA 11 R 22.5 tire.

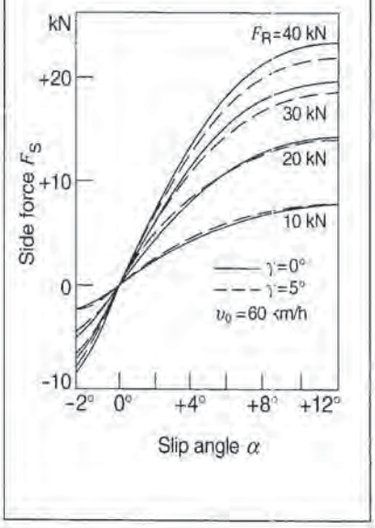
***Gough tire-performance map** (B)*



***Side force and aligning torque vs. slip angle** (A)*
Parameter: Speed 20 km/h ≤ v_0 ≤ 100 km/h.



***Side force vs. slip angle** (C)*
Parameters: Wheel load and camber angle.



Tire performance with wheel rotating freely at a given slip angle

When the tire rotates at a slip angle, side forces are generated as a function of the slip angle. These side forces are accompanied by an aligning torque (Diagram B). The Gough diagram [3] is frequently employed to illustrate this phenomenon. The side force is dependent on the slip angle and increases degressively as higher loads are applied to the wheel (Diagram C). The maximum side force is inversely proportional to speed, while the influence of speed increases as a function of wheel load (Diagram A).

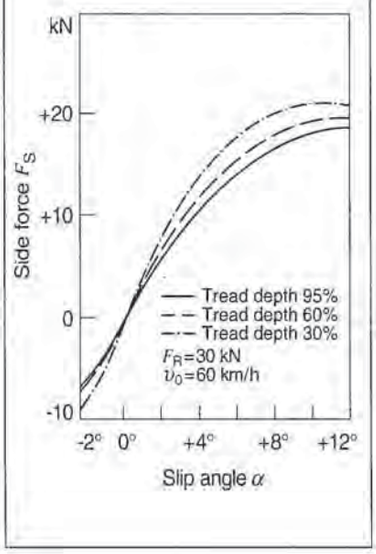
When dealing with tires for passenger-cars and light commercial vehicles, the situation is thus: If a wheel rotating at a given slip angle is subjected to camber, the camber and side forces cause a parallel displacement of the side-force/slip-angle curves. Tires on heavy-duty commercial vehicles also display additional displacements of the side-force/slip-angle curves due to the side forces accompanying the camber; this phenomenon only occurs at larger slip angles. The result is that practically all curves intersect the coordinate origin (Diagram C).

On dry roads, reduced tread depth results in steeper side-force/slip-angle curves, with an accompanying increase in the maximum side forces which can be transferred (Diagram D).

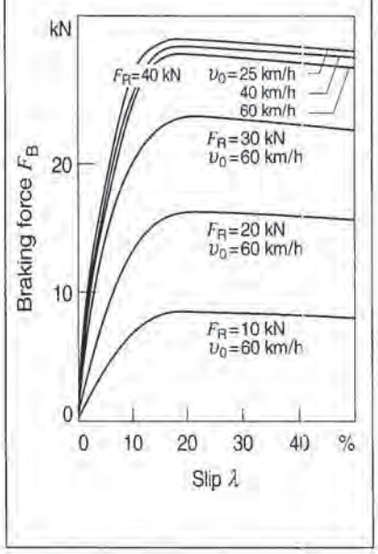
Tire performance under acceleration and braking with wheel rolling straight ahead

The tire's response to "slip" is similar to its reaction to slip angles (Diagram E). The maximum circumferential force will generally lie within the range of 10 to 20% slip on a dry road surface. The coefficient of adhesion responds to higher wheel loads with a less striking, more degressive increase in the circumferential than in the lateral direction. On larger tires, the influence which speed exerts on the level of the lateral coefficient of adhesion is less pronounced in the normal speed range for heavy-duty commercial vehicles than with tires similar to those used on passenger cars (Diagram E).

***Side force vs. slip angle** (D)*
Parameter: Tread depth.



***Braking force vs. slip** (E)*
Parameters: Wheel load and speed.



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Tire pressure has only a minimal effect on maximum circumferential forces at low wheel loads. At higher wheel loads, reduced tire pressures result in a substantially more pronounced rate of increase in the maximum circumferential force (Diagram G).

Side and circumferential forces respond to high tire loads by displaying mutually opposed reactions to variations in tire pressure (Diagrams F, G).

Tire performance at slip angles and slip

If a tire operating under circumferential forces or slip is subjected to an additional slip angle, at all slip rates the usable circumferential forces will decrease with increasing slip angle. The higher the slip angle, the more the curve for maximum circumferential force is displaced toward higher slip rates (Diagram I).

The elliptical curves for side force vs. circumferential force (braking force) vary according to wheel load (Diagram H). For a specific wheel load, on ABS-equipped vehicles this curve represents the maximum available adhesion limits for the vehicle dynamics.

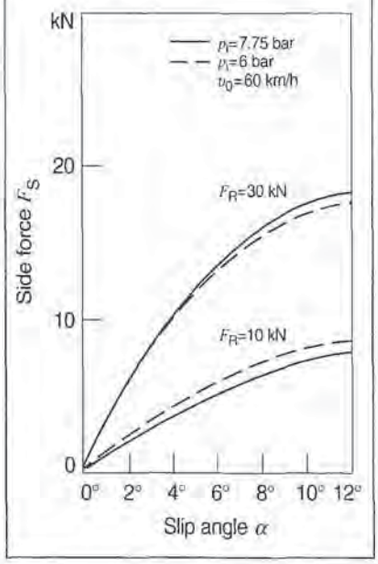
The tire-performance curves resulting from measurements actually taken, indicate the propagation of side force as a function of braking force within the slip-angle range of 0...10°. The parameters for wheel load, speed and tire pressure remain constant (Diagram K).

Dynamic tire-performance curves

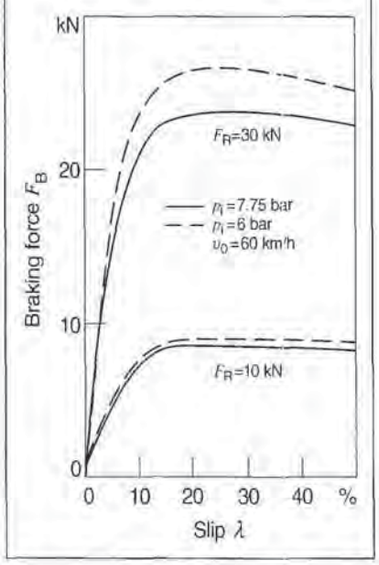
The illustrated tire-performance curves are based on parameters which are only subject to gradual change during the course of the measurements, i.e. quasi-static conditions. Actual operation, on the other hand, is characterized by dynamic processes. Increases in the speed at which the influencing parameters vary will induce certain maneuver-specific changes in tire response, the magnitude of which makes it imperative that they be considered.

The most significant influencing parameters are dynamic changes in:
– Slip angle,
– Trackwidth,

Side force vs. slip angle on heavy-duty commercial-vehicle tires *(F)*
Parameters: Tire pressure and wheel load.

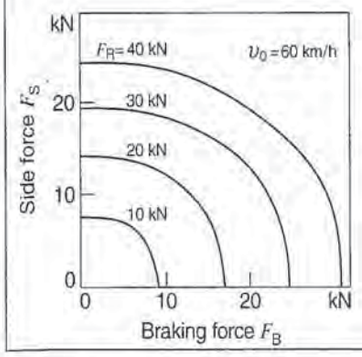


Braking force vs. slip on heavy-duty commercial-vehicle tires *(G)*
Parameters: Tire pressure and wheel load.

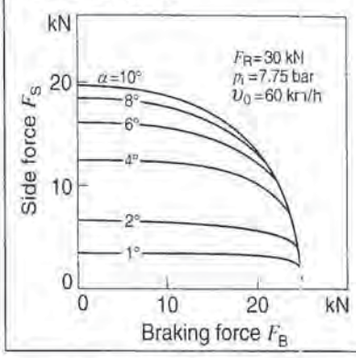


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Lateral force vs. braking force *(H)*
Maximum achievable adhesion at constant wheel load.



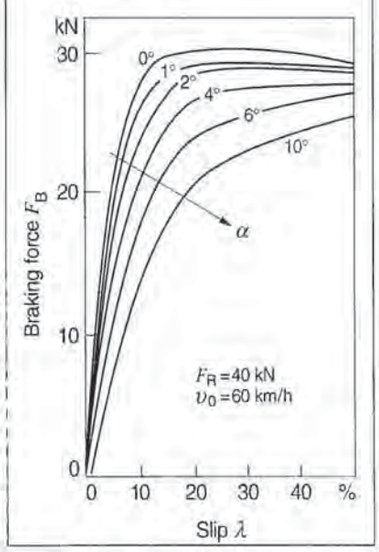
Tire-performance curves measured at a wheel load of 30 kN *(K)*
Side force as function of braking force, Parameter: Slip angle.



– Camber,
– Slip,
– Wheel load.

Generally, the tire's response to these rapidly changing parameters is shown graphically as a function of frequency. That is, amplitudes and phase angles of the tire forces and moments are illustrated relative to frequency, as a function of the forces acting on the tire, while including the frequency-dependent progression of mean values for tire forces [5, 6].

Braking force vs. slip *(I)*
Parameter: Slip angle.



Literature

[1] Gengenbach, W.: Experimentelle Untersuchung von Reifen auf nasser Fahrbahn (Experimental investigation of tires on wet road surface). ATZ, 1968, Hefte 3, 8, and 9,

[2] von Glasner, E.C.: Einbeziehung von Prüfstandsergebnissen in die Simulation des Fahrverhaltens von Nutzfahrzeugen (Including test-bench results in the simulation of commercial-vehicle handling). Habilitation, Universität Stuttgart, 1987.

[3] Gough, V. E.: Cornering Characteristics of Tires. Automobile Engineer, 1954, Vol. 44.

[4] Weber R.: Beitrag zum Übertragungsverhalten zwischen Schlupf- und Reifenführungskräften (Contribution concerning the transmission behavior between slip and circumferential/side forces). AI, 1981, Vol. 4.

[5] Fritz, G. Seitenkräfte und Rückstellmomente von Personenwagenreifen bei periodischer Änderung der Spurweite, des Sturz- und Schräglaufwinkels (Side forces and aligning torques of passenger-car tires under the influence of periodic change of track width, camber, and slip angle). Dissertation, Universität Karlsruhe, 1978.

[6] Weber, R.: Reifenführungskräfte bei schnellen Änderungen von Schräglauf und Schlupf (Circumferential/side forces as a function of rapid changes in slip angle and slip). Habilitation, Universität Karlsruhe, 1981.

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Steering

The steering mechanism converts the driver's rotational input at the steering wheel into a change in the steering angle of the vehicle's steering road wheels.

Requirements on steering systems

In accordance with European Directive 70/311/EWG the steering system must guarantee easy and safe steering of the vehicle: The maximum permissible operating time and operating force for a fully serviceable steering system, and for a faulty system, are to be taken from these Regulations (Table). They must be complied with when the vehicle is turned into a spiral path from a straight-ahead path at a speed of 10 km/h.

Steering behavior

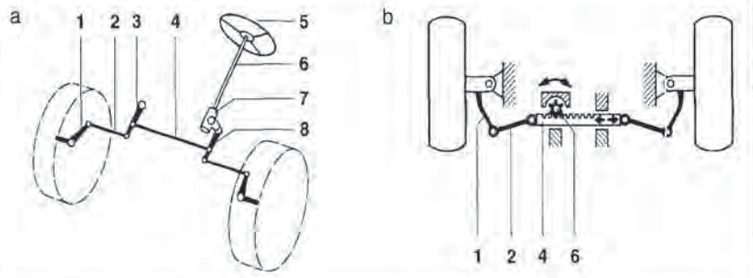
The requirements in terms of steering behavior can be summarized as follows:
1. Jolts from irregularities in the road surface must be damped as much as possible during transmission to the steering wheel. However, such damping must not cause the driver to lose contact with the road.
2. The basic design of the steering kinematics must satisfy the Ackermann conditions: the extensions of the wheel axes of the left and right front wheels, when at an angle, intersect on an extension of the rear axle.
3. By means of suitable directness of the steering system (particularly if rubber-elastic connections are used) the vehicle must react to minute steering corrections.
4. When the steering wheel is released, the wheels must return automatically to the straight-ahead position and must remain stable in this position.

Regulations for steering operating force

Vehicle category	Fully serviceable steering system			Faulty steering system		
	Maximum applied force in daN	Time in s	Turning radius in m	Maximum applied force in daN	Time in s	Turning radius in m
M_1	15	4	12	30	4	20
M_2	15	4	12	30	4	20
M_3	20	4	12	45	6	20
N_1	20	4	12	30	4	20
N_2	25	4	12	40	4	20
N_3	20	4	12[1]	45[2]	6	20

[1] Or steering lock in case this value is not reached.
[2] 50 daN for non-articulated vehicles, with two or more steered axles, excluding friction-steered axles.

Steering assembly (diagram)
a) Basic principle, b) Rack-and-pinion steering
1 Steering arm, 2 Drag link, 3 Idler arm, 4 Tie rod/rack, 5 Steering wheel, 6 Steering shaft, 7 Steering box, 8 Pitman arm.



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5. The steering should have as low a ratio as possible (number of steering-wheel turns from lock to lock) in order to obtain ease of handling. The steering forces involved are determined not only by the steering ratio but also by the front suspension load, the turning circle, the suspension geometry (caster angle, kingpin angle, kingpin offset), the properties of the tire tread and the road surface.

Types of steering box

A steering box must have the following qualities:
- no play in the straight-ahead position,
- low friction, resulting in high efficiency,
- high rigidity,
- readjustability.

For these reasons, only two types have become established to date:

Rack-and-pinion steering
Basically, as the name implies, the rack-and-pinion steering consists of a rack and a pinion. The steering ratio is defined by the ratio of pinion revolutions (= steering-wheel revolutions) to rack travel. Suitable toothing of the rack allows the ratio to be made variable over the travel. This lowers the actuating force or reduces the travel for steering corrections.

Recirculating-ball steering
The forces generated between steering worm and steering nut are transmitted via a low-friction recirculating row of balls. The steering nut acts on the steering shaft via gear teeth. A variable ratio is possible with this steering box.

Steering kinematics

Steering kinematics and axle design must be such that, although the driver receives feedback on the adhesion between wheels and road surface, the steering wheel is not subjected to any forces from the spring motion of the wheels or from driving forces (front-wheel drive) (see P. 620).

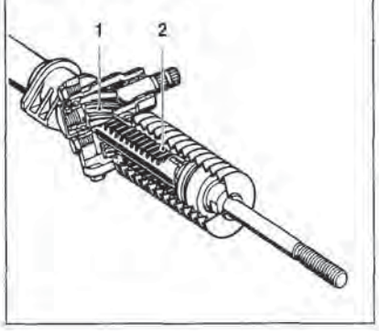
The steering-axis inclination causes the front section of the vehicle to lift when the wheels are at an angle. This leads to a steering-angle-dependent caster return torque.

The toe-in (toe-out) is a slip angle present even during straight-ahead driving which tensions the linkages and causes a rapid build-up of transverse forces when the wheels are at an angle.

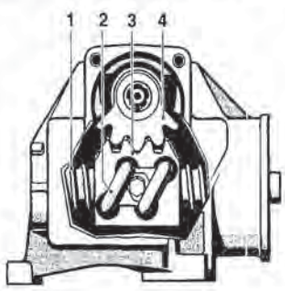
The caster produces a lever arm for lateral forces: i.e., a speed-dependent return torque.

The steering roll radius determines the extent to which the steering system is affected by disturbance forces (brakes pulling unevenly, driving forces under traction/overrun conditions with front-wheel drive). Today, the goal is to achieve a steering roll radius which is "zero" to "slightly negative".

Rack-and-pinion steering
1 Pinion, 2 Rack.



Recirculating-ball steering
1 Steering worm, 2 Recirculating balls, 3 Steering nut, 4 Steering shaft with gear teeth.



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Classification of steering systems

European Directive 70/311/EEC distinguishes three types of steering system for front wheels:

- Muscular-energy steering systems in which the steering force is produced exclusively by the driver (see mechanical steering boxes on P. 639)
- Power steering systems in which the steering force is produced exclusively by an energy source in the vehicle.
- Power-assisted steering systems in which the steering force is produced by the muscular energy of the driver and by an energy source. Used for high-speed vehicles.

Hydraulic power-assisted steering

Energy source

The energy source consists of a vane pump (generally driven by the engine) with an integral oil-flow regulator, an oil reservoir and connecting hoses and pipes.

The pump must be dimensioned so that it generates sufficient pressure to enable rotation of the steering wheel at a speed of at least 1.5 n/s even when the engine is only idling.

The compulsory pressure-limiting valve required on hydraulic systems is usually integrated.

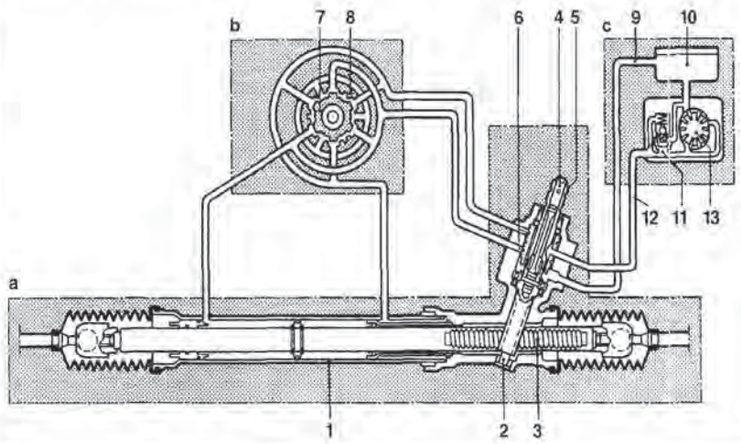
The pump and the system components must be designed such that the operating temperature of the hydraulic fluid does not rise to an excessive level (<100 °C) and such that no noise is generated and the oil does not foam.

On small cars, electrically powered gear or roller pumps are occasionally used as well. Such units are powered by the vehicle's electrical system and are more versatile with regard to their location, as well as facilitating modular vehicle design. An ECU achieves energy saving effects by varying the pump speed. Higher-voltage electrical systems currently under development would also allow larger cars to use this type of unit.

Control valve

The control valve provides the steering cylinder with an oil pressure which corresponds to the rotary motion of the steer-

Hydraulic rack-and-pinion steering with rotary distributor
a) Steering assembly, b) Rotary distributor cross-section (enlarged), c) Oil supply (source of energy)
1 Steering cylinder, 2 Drive pinion, 3 Rack, 4 Torsion bar, 5 Steering spindle, 6 Control port, 7 Rotary distributor, 8 Control bushing, 9 Return line, 10 Oil reservoir, 11 Pressure- and oil-flow-limiting valve, 12 Pressure line, 13 Vane-type pump.



ing wheel. A flexible torque detector (e.g. torsion bar, spiral spring, leaf spring) translates the applied torque, proportionally in most cases, precisely and without any degree of play into as small an actuator travel as possible. The control edges, which are in the form of chamfers or bevels, move as a result of the actuator travel and thus form the corresponding opening cross section for the oil flow.

Control valves are usually built according to the "open center" principle, i.e., when the control valve is not actuated, the oil delivered by the pump flows back to the oil reservoir at zero pressure.

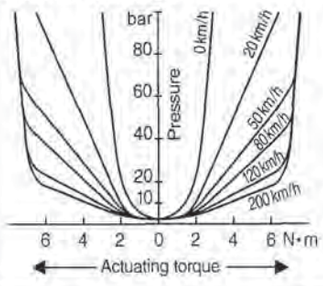
Steering cylinder

The double-acting steering cylinder converts the applied oil pressure into an assisting force which acts on the rack and which intensifies the steering force exerted by the driver. The steering cylinder is normally integrated in the steering box. As the steering cylinder has to be extremely low-friction, particularly high demands are made on the piston and rod seals.

Parameterizable hydraulic power-assisted steering

Increasing demands regarding vehicle operating comfort and safety have led to the introduction of power-assisted steering systems with modulation capability. The electronically controlled power-assisted rack-and-pinion steering system is an example. It operates dependent on speed, i.e., the vehicle speed as measured by the electronic speedometer controls the actuating force of the steering system. A control unit evaluates the speed signals and determines the level of hydraulic reaction, and therefore the actuating force on the steering wheel. This level of hydraulic reaction is transmitted to the steering-system control valve via an electrohydraulic converter, in the course of which the hydraulic reaction changes in relation to the vehicle speed. The special design of the steering characteristic means that when parking, and when moving the steering wheel while standing still, only minimal forces need be applied to the steering wheel, whilst the level of assistance is reduced as speed increases. Thus precise and accurate steering is made possible at high speeds. With this system, it is important that oil pressure and volume flow are at no time reduced, so that they can be called on immediately in emergency situations. These qualities permit outstanding steering precision and safety, while at the same time providing optimum steering comfort.

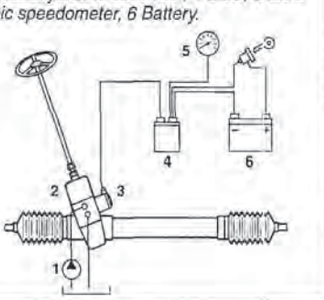
Parameterizable power-assisted steering *(characteristic curves)*
The characteristic curve form can be adjusted to the vehicle data.



Electric power-assisted steering

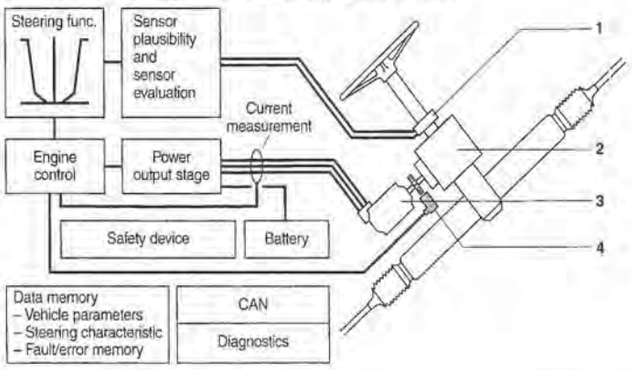
The SERVOLECTRIC (Bosch/ZF designation) is currently under development for medium-sized and smaller cars. This electric power-assisted steering system features an electric motor which is powered by the vehicle's electrical system and can be fitted in one of three possible arrangements either as a steering-column, pinion-drive or rack-and-pinion drive unit.
The SERVOLECTRIC's ECU provides the

Speed-dependent power-assisted steering with modulation capability *(diagram)*
1 Pressure-oil pump, 2 Steering-valve housing, 3 Electrohydraulic converter, 4 ECU, 5 Electronic speedometer, 6 Battery.



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Schematic diagram of electric power-assisted steering
1 Torque sensor, 2 Reduction gear, 3 Motor sensor, 4 Electric motor.



system's programmability (e.g. dynamic handling variables), the automatic steering-axis feedback amplification, as well as offering considerable steering-energy savings potential (approx. 85 % compared with hydraulic power-assisted steering with engine-driven pump).

The SERVOLECTRIC system allows the vehicle to be steered unassisted (using greater effort) in the event of failure of the power assistance.

Power-assisted steering for commercial vehicles

Power-assisted steering with all-hydraulic transmission

With "hydrostatic steering" there is no mechanical connection between steering wheel and road wheels. The steering force is hydraulically boosted and is transmitted exclusively by hydraulic means. Located in the ECU is a metering pump which supplies the steering cylinder with an oil pressure corresponding to the movement of the steering wheel. Owing to unavoidable leakage losses in the metering pump, the straight-ahead position of the steering wheel is no longer defined. This is why the use of this steering system is confined to machines.

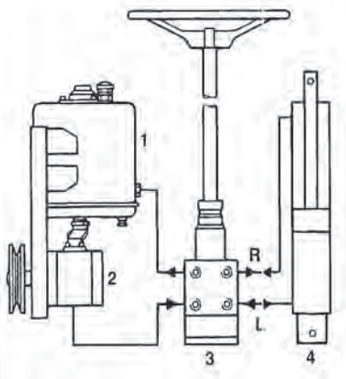
The maximum permissible speed is 25 km/h in many European countries; in Germany it is 50 km/h and, with dual-circuit design, could be increased to 62 km/h.

Dual-circuit power-assisted steering system for heavy commercial vehicles

Dual-circuit steering systems are necessary where the actuating forces needed at the steering wheel exceed 450 N in the case of failure of the power assistance (EWG 70/311).

These steering systems are characterized by hydraulic redundancy. Both steer-

Power-assisted steering with all-hydraulic transmission
1 Oil reservoir, 2 Steering pump, 3 ECU with metering pump, 4 Steering cylinde:. Connections: L Oil flow for left lock, R Oil flow for right lock



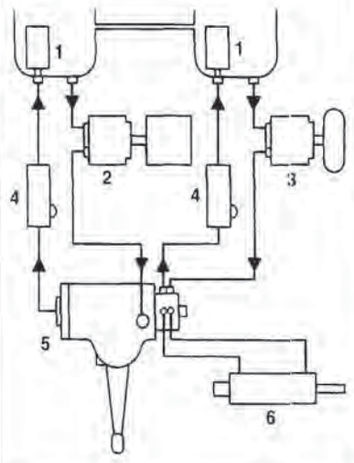
ing circuits of such a system are functionally tested by means of flow indicators. The steering-circuit supply pumps must be driven in differing ways (e.g., engine-dependent, vehicle-speed-dependent or electrically). In accordance with legislation, if the engine or one of the steering circuits fails, the vehicle can be steered with the still functioning circuit.

Single-circuit power-assisted steering system for commercial vehicles

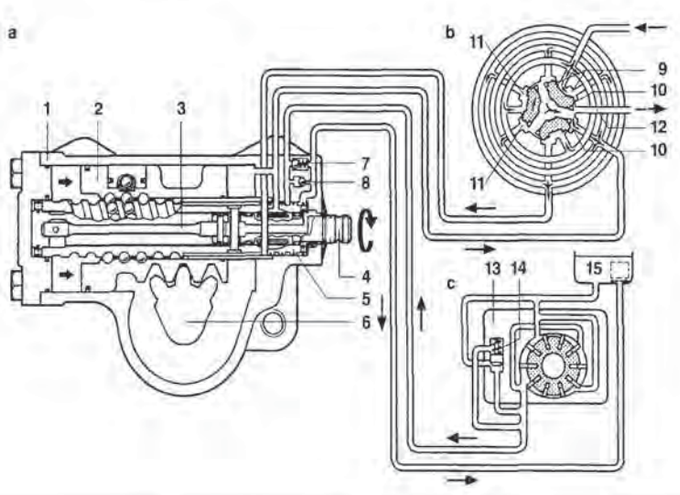
Commercial vehicles are usually fitted with recirculating-ball power steering. In modern systems the control valve is integrated into the steering worm, which leads to compact design and optimum weight.

Only minor modification of the control valve components is necessary, to permit the actuating force in modern recirculating-ball power-steering systems to be adjusted with the aid of control electronics to vehicle speed and to other parameters such as lateral acceleration or load condition.

Dual-circuit power-assisted steering system
1 Oil reservoir, 2 Engine-driven pump, 3 Wheel-driven pump, 4 Flow indicator, 5 Dual-circuit hydraulic steering, 6 Steering cylinder.



Recirculating-ball hydraulic steering system
a) Steering assembly, b) Rotary distributor cross-section (enlarged), c) Oil supply (source of energy) 1 Housing, 2 Piston, 3 Torsion bar, 4 Rotary distributor/steering spindle, 5 Control bushing/worm, 6 Sector shaft, 7 Pressure-limiting valve, 8 Replenishing valve, 9 Inlet slot, 10 Return slot, 11 Axial groove, 12 Return groove, 13 Vane-type pump, 14 Pressure and oil-flow-limiting valve, 15 Oil reservoir.



Braking systems

Definitions, Principles
(Based on ISO 611)

Braking equipment
All the braking systems fitted to a vehicle, whose function is to reduce its speed or bring it to a halt, or to hold the vehicle stationary if already halted.

Braking systems

Service braking system
All the elements, the action of which may be modulated, allowing the driver to reduce, directly or indirectly, the speed of a vehicle during normal driving or to bring the vehicle to a halt.

Secondary braking system
All the elements, the action of which may be modulated, allowing the driver to reduce, directly or indirectly, the speed of a vehicle or to bring the vehicle to a halt in case of failure of the service braking system.

Parking braking system
All the elements allowing the vehicle to be held stationary mechanically even on an inclined surface, and particularly in the absence of the driver.

Additional retarding braking system
All the elements allowing the driver directly or indirectly to stabilize or to reduce the speed of the vehicle, particularly on a long incline.

Automatic braking system
All the elements which automatically brake the towed vehicle as a result of intended or accidental separation from the towing vehicle.

Antilock braking system (ABS)
Aggregate of all devices within the service-braking system which provide automatic slip control (in the direction of rotation) for one or several wheels under braking conditions. The braking force at wheels featuring direct control is controlled using data provided by the wheel's own wheel-speed sensor, while indirectly controlled wheels rely on data provided by the wheel-speed sensor(s) at one or more of the other wheels. An ABS system with "Select High" control has directly and indirectly controlled wheels. With an ABS system with "Select Low" control, all wheels with speed sensors are directly controlled.

Constituent elements

Energy supplying device
Parts of a braking system which supply, regulate and, if necessary, condition the energy required for braking. It terminates at the point where the transmission device starts, i.e. where the various circuits of the braking systems, including the circuits of accessories if fitted, are protected either towards the energy supplying device or from each other.

The energy source is the part of the energy-supplying device which generates the energy. It may be located away from the vehicle (for example in the case of a compressed-air braking system for a trailer) and can also be the muscular strength of the driver.

Control device
Parts of a braking system which initiate the operation and control the effect of this braking system. The control signal can be conveyed within the control device by, for example, mechanical, pneumatic, hydraulic or electrical means, including the use of auxiliary or non-muscular energy.

The control device is defined as starting at that component to which the control force is directly applied, and can be operated:
- by direct action of the driver, either by hand or foot;
- by indirect action of the driver or without any action (only in the case of towed vehicles);
- by variation of the pressure in a connecting pipe, or of the electric current in a cable, between the towing and the towed vehicle at the time of operation of one of the braking systems of the towing vehicle, or in the case of a failure; and
- by the inertia of the vehicle or by its weight or that of one of its main constituent elements.

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The control device is defined as ending at that point at which the braking energy is distributed, or where a portion of the energy is diverted to control braking energy.

Transmission device
Parts of a braking system which transmit the energy distributed by the control device. It starts either at the point where the control device terminates or at the point where the energy supplying device terminates. It terminates at those parts of the braking system in which the forces opposing the vehicle's movement, or its tendency towards movement, are generated. It can, for example, be of the mechanical, hydraulic, pneumatic (pressure above or below atmospheric), electric, or combined (for example hydromechanical, hydropneumatic) type.

Brake
Parts of a braking system in which the forces opposing the vehicle's movement, or its tendency towards movement, are developed, such as friction brakes (disk or drum) or retarders (hydrodynamic or electrodynamic retarders, engine brakes).

Supplementary device of the towing vehicle for the towed vehicle
Parts of a braking system on a towing vehicle which are intended for the supply of energy to, and control of, the braking systems on the towed vehicle. It comprises the components between the energy supplying device of the towing vehicle and the supply-line coupling head (inclusive), and between the transmission device(s) of the towing vehicle and the control-line coupling head (inclusive).

Definitions relating to the energy supplying device

Muscular-energy braking system
Braking system in which the energy necessary to produce the braking force is supplied solely by the physical effort of the driver.

Energy-assisted braking system
Braking system in which the energy necessary to produce the braking force is supplied by the physical effort of the driver and one or more energy supplying devices.

Non-muscular-energy braking system
Braking system in which the energy necessary to produce the braking force is supplied by one or more energy supplying devices excluding the physical effort of the driver. This is used only to control the system. Note: However, a braking device in which the driver can increase the braking force, in the total failed energy condition, by muscular effort acting on this device, is not included in the above definition.

Inertia braking system
Braking system in which the energy necessary to produce the braking force arises from the approach of the trailer to its towing vehicle.

Gravity braking system
Braking system in which the energy necessary to produce the braking force is supplied by the lowering of a constituent element of the trailer (e.g. trailer drawbar), due to gravity.

Definitions of braking systems relating to the arrangement of the transmission device

Single-circuit braking system
Braking system having a transmission device embodying a single circuit. The transmission device comprises a single circuit if, in the event of a failure in the transmission device, no energy for the production of the application force can be transmitted by this transmission device.

Multi-circuit braking system
Braking system having a transmission device embodying several circuits. The transmission device comprises several circuits if, in the event of a failure in the transmission device, energy for the production of the application force can still be transmitted, wholly or partly, by this transmission device.

Definitions of braking systems relating to vehicle combinations

Single-line braking system
Assembly in which the braking systems of the individual vehicles act in such a way that the single line is used both for the energy supply to, and for the control of, the braking system of the towed vehicle.

Two- or multi-line braking systems
Assembly in which the braking systems of the individual vehicles act in such a way that several lines are used separately and simultaneously for the energy supply to, and for the control of, the braking system of the towed vehicle.

Continuous braking system
Combination of braking systems for vehicles forming a vehicle combination. Characteristics:
- from the driving seat, the driver can operate by a single operation and with a variable degree of force a directly operated control device on the towing vehicle and an indirectly operated control device on the towed vehicle,
- the energy used for the braking of each of the vehicles forming the combination is supplied by the same energy source (which can be the muscular effort of the driver) and
- simultaneous or suitably phased braking of the individual units of a vehicle combination.

Semi-continuous braking system
Combination of braking systems for vehicles forming a vehicle combination. Characteristics:
- the driver, from his driving seat, can operate gradually by a single operation a directly operated control device on the towing vehicle and an indirectly operated control device on the towed vehicle,
- the energy used for the braking of each of the vehicles forming the combination is supplied by at least two different energy sources (one of which can be the muscular effort of the driver) and
- simultaneous or suitably phased braking of the individual units of a vehicle combination.

Non-continuous braking system
Combinations of the braking systems of the vehicles forming a combination which is neither continuous nor semi-continuous.

Braking-system control lines

Wiring & conductors: These are employed to conduct electrical energy.
Tubular lines: Rigid, semi-rigid or flexible tubes/pipes used to transfer hydraulic or pneumatic energy.

Lines connecting the braking equipment of vehicles in a vehicle combination

Supply line: A supply line is a special feed line transmitting energy from the towing vehicle to the energy reservoir of the towed vehicle.
Control line: A control line is a special pilot line by which the energy essential for control is transmitted from the towing vehicle to the towed vehicle.
Common supply and control line: Line serving equally as supply line and as control line (single-line braking system).
Secondary line: Special actuating line transmitting from the towing vehicle to the towed vehicle the energy essential for the secondary braking of the towed vehicle.

Braking mechanics
Mechanical phenomena occurring between the initiation of actuation of the control device and the end of the braking action.

Gradual braking
Braking which, within the normal range of operation of the control device, permits the driver, at any moment, to increase or reduce, to a sufficiently fine degree, the braking force by operation of the control device. When an increase in braking force is obtained by action of the control device, an inverse action shall lead to a reduction of that force.

Braking-system hysteresis: Difference in control forces between application and release for the same braking torque.

Brake-hysteresis: Difference in applica-

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tion force between application and release for the same braking torque.

Forces and torques

Control force F_c: Force exerted on the control device.

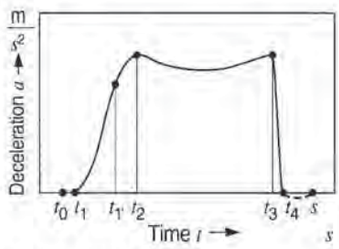
Application force F_s: In friction brakes, the total force applied to a brake lining, which causes the braking force by friction effect.

Braking torque: Product of the frictional forces resulting from the application force and the distance between the points of application of these forces and the axis of rotation of the wheels.

Total braking force F_t: Sum of the braking forces at the interfaces between all the wheels and the ground, produced by the effect of the braking system, which oppose the movement or the tendency to movement of the vehicle.

Braking-force distribution: Specification of braking force according to axle, given as % of the total braking force F_t. Example: Front axle 60 %, rear axle 40 %.

Times and deceleration during braking to a stop



before t_0: Reaction time
t_0: *Initial application of force on control device*
t_1: *Start of deceleration*
$t_{1'}$: *Buildup time completed*
t_2: *Fully developed deceleration*
t_3: *Full deceleration ends*
t_4: *Braking completed (vehicle stationary)*
$t_1 - t_0$: *Initial response time*
$t_{1'} - t_1$: *Buildup time*
$t_3 - t_2$: *"Mean fully developed deceleration" range*
$t_4 - t_1$: *Active braking time*
$t_4 - t_0$: *Total braking time*

Brake coefficient C^*: Defines the relationship between the total circumferential force of a given brake and the respective brake's application force.

$$C^* = F_u/F_s$$

F_u Total circumferential force, F_s Application force. The mean is employed when there are variations in the application forces at the individual brake shoes.

$$F_s = \sum F_{si}/i$$

i Number of brake shoes

Times (see diagram)

Reaction time
The time which elapses between perception of the state or object which induces the response, and the point at which the control device is actuated (t_0).

Actuating time of the brakes
Elapsed time between the moment when the component of the control device (t_0) on which the control force acts starts to move, and the moment when it reaches its final position corresponding to the applied control force (or its travel). (This is equally true for application and release of the brakes).

Initial response time $t_1 - t_0$
Elapsed time between the moment when the component of the control device on which the control force acts starts to move and the moment when the braking force takes effect.

Buildup time $t_{1'} - t_1$
Period that elapses between the point at which the braking force starts to take effect and the point at which a certain level is reached (75 % of asymptotic pressure in the wheel cylinder as per EU Directive 71/320 EEC, App. III/2.4).

Response and buildup time
The sum of the initial response and buildup times is used to assess how the brake system behaves referred to time, until the moment at which the full braking effect is reached.

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Active braking time $t_4 - t_1$
Elapsed time between the moment when braking force takes effect and the moment at which it ceases. If the vehicle stops before braking force ceases, the time of the end of movement is the end of the active braking time.

Release time
Elapsed time between the moment when the actuating time for release starts and the moment when the braking force ceases.

Total braking time $t_4 - t_0$
Elapsed time between the moment when the control device component on which the control force acts starts to move, and the moment when the braking force ceases. If the vehicle stops before the braking force ceases, the time of the end of movement constitutes the end of the total braking time. (See the graph on P. 647)

Braking distance s
Distance traveled by the vehicle during the total braking time. If the time of the end of movement constitutes the end of the total braking time, this distance is called the "stopping distance".

Braking work W
Integral of the product of the instantaneous total braking force, F_t, and the elementary movement, d_s, over the distance travelled during braking, s.

$$W = \int_0^s F_t \cdot d_s$$

Instantaneous braking power P
Product of the momentary total braking force F_t and the vehicle's road speed υ

$$P = F_t \cdot \upsilon$$

Braking deceleration
Reduction of speed obtained by the braking system in the considered time, t. The following can be identified:
Momentary deceleration: $a = d\upsilon/dt$

Mean deceleration: $a_{ms} = \upsilon_0^2/(2 \cdot s_0)$

The mean deceleration value over the stopping distance, whereby υ_0 and s_0 refer to the point t_0.

Mean deceleration a_{mt} within a section of time:
Deceleration between two points in time t_i and t_j. This formula is used for the evaluation of the braking efficiency of retarders.

$$a_{mt} = (\upsilon_i - \upsilon_j)/(t_j - t_i)$$

Mean fully developed deceleration a_{mf}:
Mean value of deceleration (a_{mt}) over subdivision $t_3 - t_2$ of the fully developed deceleration.

$$a_{mf} = \frac{1}{t_3 - t_2} \cdot \int_{t_2}^{t_3} a \cdot dt$$

Braking factor z
Ratio between the total braking force, F_t, and the static weight, G_s, on the axle or the axles of the vehicle:

$$z = F_t/G_s$$

Legal regulations

The testing of braking equipment pursuant to the issuance of the type approval for a vehicle in Germany may take place in accordance with one of the following three groups of regulations as chosen by the vehicle manufacturer. They are:
- National regulations and statutory requirements for the testing of brakes applicable in the country of use,
- Council Directive of the European Communities: Directive 71/320/EEC and the Amending Directives and Annexes or
- ECE Regulation 13, 13H and 78 of the UN Economic Commission in Geneva.

In Europe, the national regulations of individual countries may specify introduction of the EU Directive or ECE Regulations. Essentially, they may be very similar. However, the ECE Regulations 13 and 13H are likely to be more recent and contain, for example, regulations for braking systems with electrical control systems (electronically controlled braking systems).

Braking systems conforming to EU Directives, and ECE Regulations 13 and 78 (for classification refer to P. 717).

Category L vehicles (less than 4 wheels)
Motorized two- and three-wheeled vehi-

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cles must be equipped with 2 mutually independent braking systems. In the case of heavy category L_5 3-wheel vehicles, the two braking systems must both act on two wheels. Additionally, these vehicles must be equipped with a parking braking system.

Category M and N vehicles

The categories M and N vehicles must meet the requirements which pertain to the service braking system, secondary braking system and parking braking system. These 3 braking systems may have common components. Such vehicles must have at least 2 mutually independent control devices.

The apportioning of braking force among the individual axles is prescribed. Vehicles in the categories M_2 and N_2 or higher must be fitted with an antilock braking system (ABS) (some exceptions may still apply for older models).

Additional retarding braking systems can be used in order to fulfill braking requirements on long inclines. Vehicles in category M_3 used for local or long-distance duty must be able to meet the requirements for braking on such "descents" using the continuous-operation braking system only.

Category O trailers

Category O_1 trailers are not required to have a braking system, however, there are requirements governing the safety connection to the towing vehicle. Beginning with category O_2, trailers must be fitted with a service braking system and a parking braking system which may have common components. It must be possible for the parking braking system to be operated by a person standing next to the vehicle. The apportioning of braking force among the individual axles is prescribed.

An antilock braking system (ABS) is prescribed for some trailers in Class O_3 and above.

Inertia braking systems are permissible for trailers up to category O_2.

The trailer must brake automatically if it becomes decoupled from the tractor vehicle while moving, or (for trailers < 1.5 t) it must be equipped with a safety connection to the towing vehicle.

Vehicles equipped with antilock braking systems (ABS)

ABS systems must conform to Appendix X of EU Directive 71/320/EEC or Annex 13 of ECE Regulation 13 (on Category 1 vehicles of classes M_2, M_3, N_2 and N_3). The essential requirements are:

- locking of the directly controllec wheels under braking on any road surface at speeds of over 15 km/h must be prevented,
- operating stability and steering control are to be maintained and
- grip on road surfaces offering the same or (in the case of Category 1) different levels of adhesion to wheels on different sides (μ split) must be utilized, and
- there must be a visual warning system for indicating electrical faults.

Only minimum standards are in force for trailer ABS. The ABS on tractor and trailer should be mutually compatible in order to ensure safety and prevent excessive tire wear.

Tractor vehicles and trailers with compressed-air braking systems

The compressed-air connections must be of the two- or multi-line design. When the service brakes on the tractor unit are operated, the service brakes on the trailer must also be operated with a var able degree of force. If any fault occurs on the service brakes of the tractor unit, that part of the system not affected by the fault must be cable of braking the trailer with a variable degree of force. If one of the lines between tractor unit and trailer is separated or develops a leak, it must still be possible to brake the trailer, or it must brake automatically.

The braking efficiency of the vehicle when fully loaded (and when empty for vehicles without ABS) is specified according to the pressure at the control-line coupling head. The service brakes of the trailer must only be capable of operation in conjunction with the service, secondary or parking brake system on the tractor unit.

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Requirements of StVZO (Germany), EU Directive 71/320 EEC and ECE Regulations

Vehicle class (For classification, refer to P. 717)		Passenger cars & motor coaches M_1	M_2	M_3	Commercial vehicles N_1	N_2	N_3	Trailers O_1	O_2	O_3	O_4
Service braking system		Acting on all wheels Prescribed force distribution to the axles						No braking system or as O_2	Inertia braking system or as O_3		
ABS as per EC Dir. or ECE[1]) ($v_{max} \geq 25$ km/h)		–	+	+	–	+	+	–	–	+	+
Type O test (drive disengaged)											
Test speed	km/h	80	60	60	80	60	60	–	60	60	60
Braking distance	≤m	50.7	36.7	36.7	61.2	36.7	36.7		$z \geq 0.30$, Semitrailer $z \geq 0.45$		
Formula for		$0.1v + \frac{v^2}{150}$	$0.15v + \frac{v^2}{130}$								
Mean fully developed deceleration	≥m/s²	5.8	5.0								
Actuating force	≤n	500	700						at ≤6.5 bar		
Type O test (drive engaged)		Behaviour of vehicle under braking from 30% – 80% v_{max} and braking efficiency						–	–	–	–
Test velocity v = 80% v_{max}, but	≤km/h	160	100	90	120	100	90				
Braking distance	≤m	212.9	111.6	91.8	157.1	111.6	91.8				
Braking distance formula		$0.1v + \frac{v^2}{130}$	$0.15v + \frac{v^2}{103.5}$								
Mean fully developed deceleration	≥m/s²	5.0	4.0								
Actuating force	≤n	500	700								
Type I test		Repeated braking at 3 m/s² loaded, drive engaged						–	Continuous braking, loaded 40 km/h 7% downhill gradient 1.7 km $z \geq 0.36$ and $z \geq 60$% of figure measured in Type O test at 40 km/h		–
v_1 = 80% v_{max}, but	≤km/h	120	100	60	120	60	60				
$v_2 = {}^1/_2\, v_1$											
Braking cycle repetitions	n	15	15	20	15	20	20				
Braking cycle duration	s	45	55	60	55	60	60				
Effectiveness of hot brakes at end of Type I Test		≥80% of brake efficiency specified for Type O test (drive disengaged) and ≥60% of the brake efficiency achieved in Type O test (drive disengaged)									
Type II Test on long descents		Energy corresponding to 30 km/h, 6% downhill gradient and 6 km, loaded, engine drive engaged, continuous-operation braking system in operation. Measurement as for Type O test (drive disengaged)						–	–	–	at 40 km/h
Effectiveness of hot brakes at end of Type II Test											
Braking distance formula		M_3: $0.15v + \frac{1.33v^2}{130}$ N_3: $0.15v + \frac{1.33v^2}{115}$									
Braking distance	≤m			45.8			50.6				
Mean fully developed deceleration	≥m/s²	–	–	3.75	–	–	3.3				$z \geq 0.33$

[1]) Exceptions may still be allowed for older models.

Vehicle class (For classification, refer to P. 717)		Passenger cars & motor coaches M_1	M_2	M_3	Commercial vehicles N_1	N_2	N_3	Trailers O_1	O_2	O_3	O_4
Type IIa Test For continuous-operation braking systems		Energy corresponding to 30 km/h, 7% downhill gradient, 6 km, loaded, only continuous-operation braking system in operation. Only permitted for M_3[2]) and towing O_4, N_3.									
Type III Test		–						–	–	--	[3])
Residual braking effect after failure in the transmission device/circuit failure, engine drive disengaged								The brakes of the trailer must be capable of full or partial operation using variable force.			
Test speed	km/h	80	60	60	70	50	40				
Braking distance, loaded	≤m	150.2	101.3	101.3	152.5	80.0	52.4				
Braking distance, empty	≤m	178.7	119.8	101.3	180.9	94.5	52.4				
Mean fully developed deceleration, loaded	≥m/s²	1.7	1.5	1.5	1.3	1.3	1.3				
empty	≥m/s²	1.5	1.3	1.5	1.1	1.1	1.3				
Actuating force	≤n	700	700	700	700	700	700				
Secondary braking system (tested as for Type O Test, drive disengaged)								The brakes of the trailer must be operated using variable force.			
Test speed	km/h	80	60	60	70	50	40				
Braking distance	≤m	93.3	64.4	64.4	95.7	54.0	38.3				
Braking-distance formula		$0.1v + \frac{2v^2}{150}$	$0.15v + \frac{2v^2}{130}$		$0.15v + \frac{2v^2}{115}$						
Mean fully developed deceleration	≥m/s²	2.9	2.5		2.2						
Actuating force, by hand	≤n	400	600		600						
with foot	≤n	500	700		700						
Parking braking system (Loaded test)											
Holding stationary on incline (downgrade or upgrade)	≥%	18			18			–	18		
Together with unbraked vehicle of class O	≥%	12			12			–	--		
Actuating force, by hand	≤n	400	600		600			–	600		
with foot	≤n	500	700		700			–	--		
Type O Test[4]) (drive disengaged, loaded)											
Test speed	km/h	80	60	60	70	50	40	–			
Mean fully developed deceleration and deceleration prior to standstill	≥m/s²	1.5			1.5		–				
Automatic braking system With compressed-air systems, automatic trailer braking in case of pressure loss in supply line											
Test speed	km/h							–	40		
Braking factor	≥%							–	13.5		

[2]) Except public transport busses
[3]) Repeated braking as for Type I test for N_3. Afterwards, braking efficiency ≥ 40 and ≥ 60% of level achieved in Type O Test.
[4]) With parking braking system or via auxiliary control device for service braking system.

Design and components of a braking system

Basic components of a braking system:

The braking system consists of:
- Energy supply,
- A control device,
- A transmission device for controlling braking force, and for activating the engine brake, parking brake, and retarder,
- Additional equipment in the towing vehicle for braking the trailer.
- Wheel brakes.

Each of those components affects the braking forces which are decisive for slowing down the vehicle or vehicle combination.

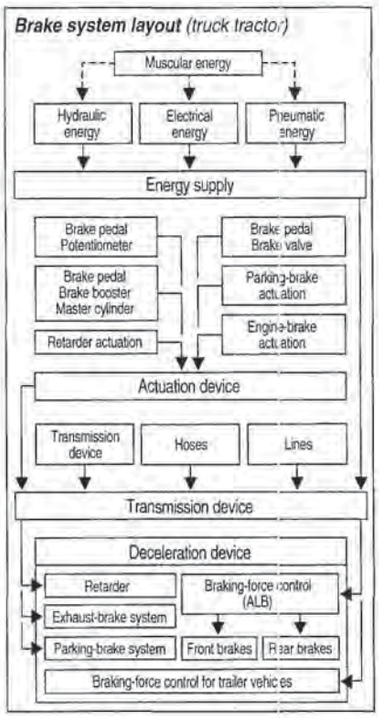
Different ranges of applications for different vehicle types result in highly variegated demands being placed on braking systems. The inescapable result is a multiplicity of highly diversified braking systems, differing from one another in both design and application.

Braking-system applications

Legal regulations stipulate that the braking system on a heavy commercial vehicle will consist of:
- service brakes,
- secondary brakes,
- parking brake, and (in some cases)
- additional retarding braking systems, and
- self-actuated braking system.

The service and parking brakes are equipped with separate individual control and transmission devices. The service brakes are generally applied with the foot, while the parking brake can be actuated with either hand or foot. The secondary braking system frequently shares components with the service or parking brakes. For instance when one circuit in a dual-circuit service-braking system also functions as secondary brake. The additional retarding braking system, which acts as a supplementary, wear-free unit, is especially useful for relieving the service brakes on long downgrades (see "Additional retarding braking system", page 682). Self-actuated (automatic) braking systems apply to trailers only.

Brake system layout *(truck tractor)*



Type of energy and transmission media

Depending upon the type of energy applied to control the braking system, a distinction is drawn between:
- muscular-energy braking systems,
- energy-assisted braking systems,
- non-muscular-energy braking systems,
- inertia braking systems.

The various systems can also be installed in combination. In contrast to the non-muscular-energy braking system, for instance, the energy-assisted system also depends to some degree on the force exerted at the pedal.

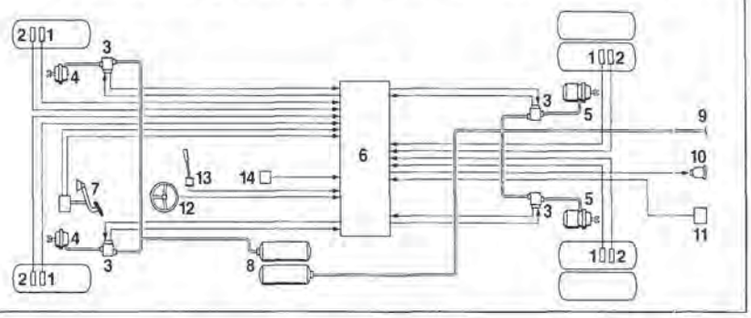
Energy-assisted and non-muscular-energy systems are distinguished not only by energy type, but also by the mediums used to transmit the energy. Pneumatic (vacuum, compressed air) and hydraulic energy are the most common, electrical energy is sometimes also employed.

Type of transmission device

The means employed to transmit energy within the braking system may be me-

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Electronic/compressed-air braking system for two-axle tractor
1 Wheel-speed sensor, 2 Brake-pad wear sensor, 3 Control valve, 4 Front-wheel cylinder, 5 Rear-wheel cylinder, 6 ECU, 7 Brake pedal, 8 Compressed-air cylinder, 9 Compressed-air supply to trailer, 10 Trailer control line, 11 Coupling-force sensor, 12 Steering-wheel position sensor, 13 Control for retarder and engine-braking system, 14 Yaw-rate/lateral-acceleration sensor



chanical, hydraulic, pneumatic or electric/electronic. Hybrid combinations may also be used in transmitting the force to the wheel brakes. Electric/electronic transmission mechanisms will play an important role in the electronic-pneumatic and electronic-hydraulic braking systems for the next generation of vehicles.

Braking-system design

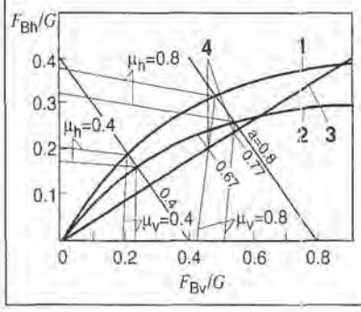
The braking system is designed with reference to both the requirements of the vehicle and the intrinsic imperatives of the system itself. In the case of the vehicle-oriented design, the vehicle's center of gravity and the specified distribution of braking force to the front and rear axles, determine the amount of braking force which can be applied before the wheels lock at any specific level of adhesion between tire and road surface. The braking-force-distribution diagram is used to illustrate this relationship. The coordinate axes show the braking force at front and rear axles relative to weight. The intersection of the straight lines representing equal adhesion coefficients at front and rear axles form the parabola defining "ideal" braking-force distribution. Lines representing constant braking complete the diagram.

If no braking-force metering device is fitted, then the distribution of the braking force as installed in the unit also forms a straight line. The slope is the ratio of the braking forces at front and rear axles as determined by the dimensions of the brakes. The wheels will always lock on the front axle first as long as the line for as-installed distribution remains below the ideal distribution (stable distribution of braking forces). The point at which the front wheels lock is found at the intersection of "as-installed distribution" and the lines representing the respective coefficient of adhesion.

The essential design criteria are:

– regulations governing minimum braking force required before the onset of lock, and locking sequence,

Distribution of braking force without braking-force metering device
Ideal distribution of braking force at 1 for approved gross vehicle weight, 2 for curb weight, with driver, 3 Braking-force distribution as installed, 4 Front brakes locked. F_{Bh} Rear braking force, F_{Bv} Front braking force, G Weight, a Braking. Adhesion coefficients: rear μ_{HFH}, front μ_{HFV}.



- distribution of load weight,
- influence of fading,
- engine braking,
- failure of a brake circuit,
- braking-force metering device (if fitted),
- retarder (if fitted).

System-oriented design concentrates on the dimensions for the wheel brakes and on the control devices.

Design criteria for the wheel brakes:
- brake type (disc or drum),
- endurance (resistance to wear and severe use),
- space available for installation,
- acceptable pressure levels,
- rigidity (on hydraulic brakes: volume of brake fluid required for actuation).

Design criteria for the control device:
- Pedal travel and pedal force for normal braking, emergency braking, and in the event of failure of a brake circuit or the brake servo.
- comfort requirements,
- installation space,
- combination with systems for brake-pressure regulation.

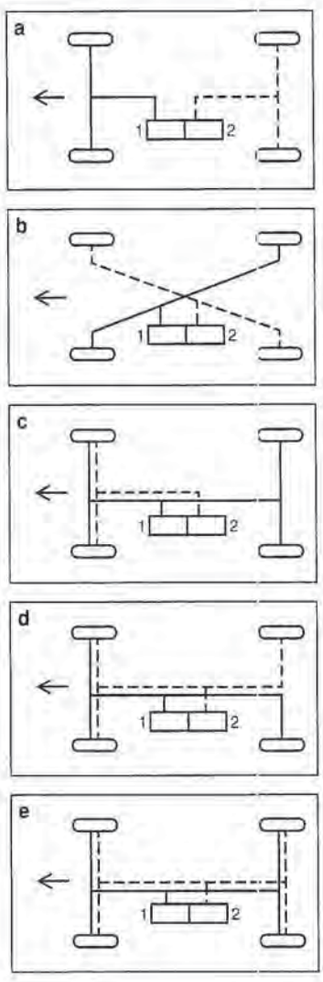
Brake-circuit configurations

Legal regulations stipulate a dual-circuit transmission system as mandatory.

Of the five available options (DIN 74 000, see below), versions II and X have become standard. As the brake lines, hoses, connections, and static and dynamic seals remain at a low level of complication, the probability of failure due to leaks is comparable to that achieved with a single-circuit system. The potential response to failure of a circuit due to overheating at one wheel points up a serious weakness in the HI, LL and HH distribution patterns, where loss of both brake circuits on a wheel could lead to total brake-system failure.

Vehicles with a forward weight bias use distribution pattern X to fulfill the regulatory requirements. The II distribution pattern is an excellent solution for vehicles with rear weight bias and mid-range and heavy commercial vehicles.

Brake-circuit configuration: Variants
a) II distribution, b) X distribution, c) HI distribution, d) LL distribution, e) HH distribution.
1 Brake circuit 1, 2 Brake circuit 2.
← Direction of travel.



II distribution pattern
Front-axle/rear-axle split. One circuit brakes the front axle and the other the rear axle.

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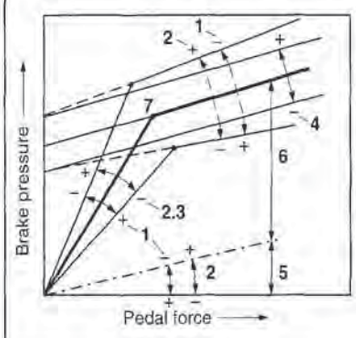
X distribution pattern
Diagonal distribution pattern. Each circuit brakes a given front wheel and the diagonally opposite rear wheel.

HI distribution pattern
Front-axle and rear-axle/front-axle split. One circuit brakes the front and rear axles, and one circuit brakes only the front axle.

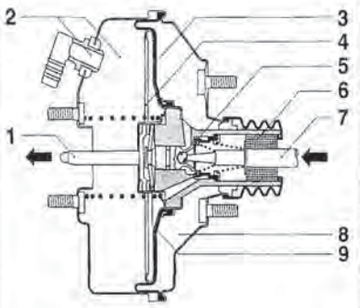
LL distribution pattern
Front-axle and rear-wheel/front-axle and rear-wheel split. Each circuit brakes the front axle and one rear wheel.

HH distribution pattern
Front-axle and rear-axle/front-axle and rear-axle split. Each circuit brakes the front axle and the rear axle.

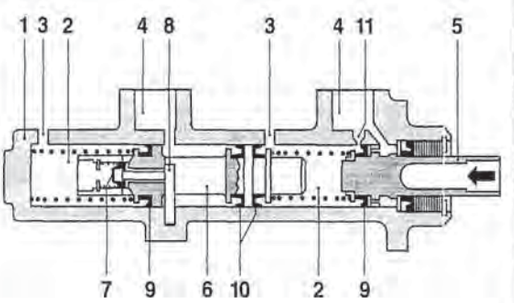
Vacuum-operated brake booster
Effect of design parameters.
1 Master-cylinder surface area, 2 Pedal leverage, 3 Boost factor, pedal leverage, 4 Diaphragm surface area, vacuum level, 5 Effect of pedal force, 6 Effect of servo assist, 7 Output point.



Vacuum-operated brake booster
1 Push rod, 2 Vacuum chamber with vacuum connection, 3 Diaphragm, 4 Piston, 5 Bell valve, 6 Air filter, 7 Piston rod, 8 Rear chamber, 9 Backing plate.



Tandem master cylinder with central valve in secondary circuit
1 Cylinder body,
2 Pressure chamber,
3 Pressure connection,
4 Connection for compensating reservoir,
5 Thrust-rod piston,
6 Secondary piston,
7 Central valve,
8 Stop for central valve,
9 Primary seal,
10 Separating seal,
11 Balancing port.



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Braking systems for passenger cars and light utility vehicles

Control devices

The control device consists of:
- brake pedal,
- vacuum brake booster,
- master cylinder,
- brake-fluid reservoir,
- device to warn of a brake-circuit failure and/or low brake-fluid level.

In addition to the basic equipment listed above, hydraulic boosters or hydraulic non-muscular-energy braking systems may be used in certain applications. In non-muscular-energy systems, the brake booster and master cylinder are replaced by a brake valve. The force at the pedal is modulated to achieve the desired brake pressure. High-pressure pumps and accumulators are included to generate and to store the requisite energy.

A vacuum brake booster is generally used due to its inexpensive and uncomplicated design. On this type of booster, the force applied at the pedal regulates the amount of outside air which is applied to a diaphragm, while vacuum remains present on the diaphragm's other side. The pressure differential at the diaphragm generates force to supplement that applied at the pedal. The simplified diagram provides a schematic illustration of the main factors which influence the braking pressure; working losses and efficiency levels are not considered:
- Pedal conversion,
- boost factor,
- diaphragm surface area,
- vacuum pressure,
- surface area of master cylinder.

The brake pressure is the result of a combination of the force at the pedal and an auxiliary assist. The proportion represented by the assist increases steadily up to full boost; the designed-in boost factor determines the precise rate. At full boost, the maximum pressure difference between outside air and vacuum has been reached. Additional augmentation of the output force is only possible via an unaccustomed increase in the force applied at the pedal. Thus it is important that the booster be designed to ensure that high rates of deceleration can be achieved without exceeding full boost to any appreciable degree.

The major determinant for output force is the surface area of the diaphragm. Two diaphragms in tandem are employed to satisfy higher pressure requirements. Technical considerations limit the maximum feasible diaphragm diameter to approx. 250 mm. The maximum negative pressure, as obtained at the intake manifold of a spark-ignition engine with the throttle closed, is approx. 0.8 bar. A vacuum pump is required on diesel engines.

As the demand for boost pressure in heavy vehicles is characteristically greater, the logical choice is a hydraulic booster, which can be designed to function on the same principles.

The energy is frequently provided by the power-steering pump, with an intermediate hydraulic accumulator being incorporated in the circuit to reduce the tendency of brakes and steering to influence each other.

A push rod carries the output force directly to the piston in the tandem master cylinder. The hydraulic pressure thus generated is transmitted to the "floating" intermediate piston, resulting in roughly equal pressures in the two chambers which supply the respective circuits.

Failure in one of the brake circuits can have one of two results: Either the push rod comes up against the intermediate piston, or hydraulic force presses the intermediate piston back against the wall of the master cylinder. This condition will be felt at the pedal, which will continue moving with virtually no resistance.

A master cylinder which responds in several stages has proven a useful expedient on vehicles with the II distribution pattern. The intermediate piston, which has a smaller diameter than the push-rod piston, applies pressure to the rear-axle circuit. The system responds to failure in the front-axle circuit by increasing the pressure which is transmitted to the rear circuit at a constant pedal pressure. The degree of pressure increase is based on the ratio of the piston areas of the push-rod and intermediate pistons.

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A brake-fluid reservoir is connected to the master cylinder to compensate for the effects of brake-lining wear and leakage. When the brake is released, either a centrally positioned valve in the master-cylinder piston opens or the piston seal opens a balancing port. This arrangement ensures that the brake system is not under pressure when released, while also providing compensation for fluid losses. The chief disadvantage of this simple layout lies in the fact that vapor bubbles in the brake fluid, which have been generated due to overheating, cause the fluid to drain from the affected brake circuit when the brakes are released. This could make it impossible to build up pressure when the brakes are applied again.

In order to prevent complete drainage in the event of a major leak, the brake-fluid reservoir is designed (at least as from a given brake-fluid level) with two circuits. One or two float-actuated switches trigger an optical display once the fluid falls below a certain level. The float-actuated switches can be replaced by differential pressure switches on the master cylinder, these then indicate failure of a brake circuit.

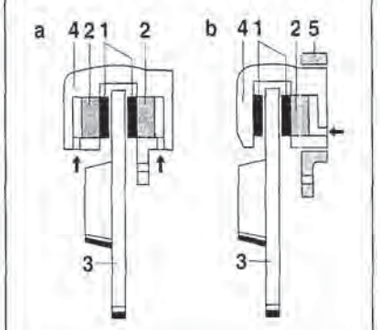
Wheel brakes

The wheel brakes must meet the following requirements:
- uniform effectiveness,
- smooth, graduated response,
- resistance to contamination and corrosion,
- extreme reliability,
- durability,
- resistance to wear,
- ease of maintenance.

Whereas on small passenger cars and commercial vehicles, various types of drum brakes fulfill these demands satisfactorily; disc brakes represent the only means of achieving even response and good control on heavy, high-speed passenger cars.

Gray cast-iron brake discs with bilaterally acting calipers have proven to be the most satisfactory layout. The brake disc is usually located within the well of the wheel rim. An arrangement which makes it necessary to provide for adequate heat dissipation through radiation, convection and thermal conductance. Additional expedients such as internally-ventilated brake discs, air ducts and optimal-flow wheel designs are employed to reduce disc temperatures, particularly on high-performance vehicles.

Disc brakes
a) Fixed caliper, b) Floating caliper.
1 Friction pads, 2 Piston, 3 Brake disc, 4 Caliper, 5 Support.



Brake calipers fall into one of two categories: Fixed calipers or floating calipers.

In the case of the fixed caliper, the housing is rigid and "grips" the brake disc from both sides. When the brakes are applied, two pistons in the caliper housing, one on each side of the brake disc, force the brake pads up against the brake disc.

Two basic subcategories of floating caliper have established themselves. The sliding caliper and the so-called Mark II caliper. With both designs, the piston or pistons act directly against brake pad on the inner side of the disc. The sliding-caliper frame or the caliper then pulls the outer pad against the disc. Compared to fixed calipers, the floating units offer the following advantages:
- Modest space requirement between brake disc and wheel nave (convenient where suspension employs small or negative steering-roll radius),
- Reduced thermal stress on the fluid, as no fluid lines are located in the critical area directly above the brake disc.

Constructive measures effectively alleviate inherent disadvantages (tendency to rattle and squeak, uneven wear of friction pads, corrosion in transmission elements).

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Braking-force metering device

The braking-force metering device is not a closed-loop control element, like the brake-pressure regulating valve as used for ABS, but rather an open-loop control element. The individual metering devices differ with respect to their function as braking-force limiters or reducers, or their control parameters, such as brake pressure, axle load or rate of deceleration.

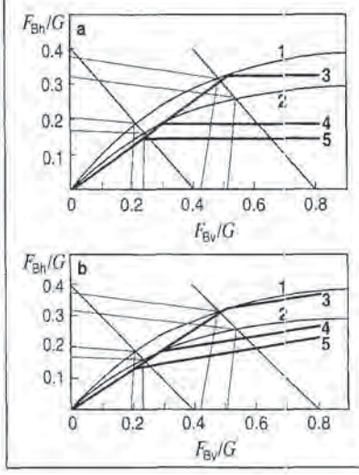
The braking force is apportioned between the front and rear axles in accordance with the dimensions of the particular brakes. It is the job of the metering device to adjust this braking-force apportionment in order to achieve a closer approximation to the ideal distribution, i.e., the parabolic curve. The ideal braking-force distribution is determined solely by the vehicle's center of gravity and the nature of the particular braking maneuver. These relationships can be shown in a dimensionless braking-force-distribution diagram. The weight-related braking forces at the front and rear axles are entered on the coordinate axes. The lines for identical braking appear as straight lines with a negative slope (–1). The ideal braking-force-distribution curves for the vehicle conditions "curb weight" and "approved gross vehicle weight" are in the form of parabolas. Diagram "a" is for a braking-force limiter and diagram "b" a braking-force reducer.

Pressure-sensitive metering valves achieve good approximation of ideal distribution with the vehicle in the "curb weight" state. On the other hand, under "approved gross vehicle weight" conditions (upper parabola), they deviate from the ideal once the limiter or reducer becomes operative (bend in the curve), i.e., the proportion of the total braking force directed toward the rear axle decreases as the rear-axle load increases.

The load-sensitive apportioning valve responds to increased loads by displacing the triggering point upward, allowing a reasonable approximation of ideal braking-force distribution under all load conditions.

The deceleration-sensitive apportioning valve is triggered by a specific rate of deceleration, and is thus basically insensitive to load.

Braking-force-distribution diagram
a) Braking-force limiter, b) Braking-force reducer. F_{Bh} Rear braking force, F_{Bv} Front braking force, G Weight. 1 Loaded, 2 Empty, 3 Loaded, according to load, 4 Empty, according to pressure; empty according to deceleration, and loaded; empty according to load, 5 Loaded, according to pressure.



The metering valve must be designed to ensure that the distribution of braking force remains on or below the ideal curve. The potential effects of fluctuations of the pad friction coefficient, as well as of engine torque and tolerances of the valve itself, must all be considered in preventing over-brake of the rear axle. In practice, this means that actual installed distribution (with bend in curve) should remain well below the ideal.

The criteria according to which the metering valve is designed include the following:

- ABS compatibility,
- Complexity in the case of split rear-axle braking circuits (e.g. X distribution),
- Bypass function for dealing with brake-circuit failure, especially with braking-force limiters,
- Facility for testing of setting and operation.

Vehicles with balanced load conditions are not necessarily equipped with a metering device, as the disadvantages of an undetected defect in the device outweigh its minimal advantages.

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ABS antilock braking systems for passenger cars

ABS antilock braking systems are closed-loop control devices within the braking system which prevent wheel lock-up during braking and, as a result, retain the vehicle's steerability and stability. The main ABS components are:

Hydraulic modulator, wheel-speed sensors, and the ECU for signal processing and control and triggering of the signal lamp and of the actuators in the hydraulic modulator.

Basic closed-loop control process

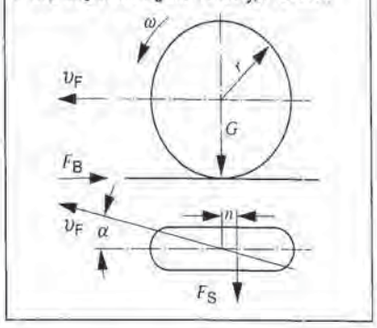
On initial braking, the brake pressure is increased; the brake slip λ rises and at the maximum point on the adhesion/slip curve, it reaches the limit between the stable and unstable ranges. From this point on, any further increase in brake pressure or braking torque does not cause any further increase in the braking force F_B. In the stable range, the brake slip is largely deformation slip, it increasingly tends towards skidding in the unstable range.

Brake slip $\lambda = (v_F - v_R)/v_F \cdot 100\,\%$
Wheel speed $v_R = r \cdot \omega$
Braking force $F_B = \mu_{HF} \cdot G$
Lateral force $F_S = \mu_S \cdot G$
μ_{HF} Coefficient of friction,
μ_S Lateral-force coefficient.

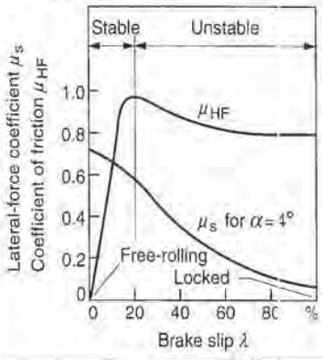
There is a more or less sharp drop in the coefficient of friction μ_{HF}, depending upon the shape of the slip curve. The resulting excess torque causes the wheel to lock-up very quickly (when braking without ABS); this is expressed as a sharp increase in wheel deceleration.

The wheel-speed sensors monitor the motion of the wheels. If one of the wheels shows signs of lock-up, there is a sharp rise in peripheral wheel deceleration and in wheel slip. If these exceed defined critical values, the controller sends commands to the solenoid-valve unit to stop or reduce the buildup of wheel-brake pressure until the danger of lock-up has passed. The brake pressure must then be built up again in order to ensure that the wheel is not underbraked. During automatic brake control, it is constantly necessary for the stability or instability of the wheel motion to be detected, and the

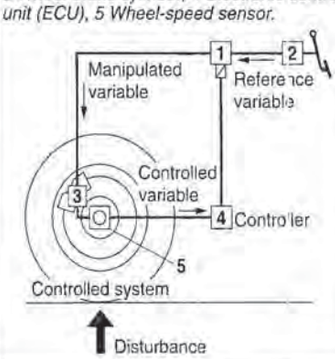
Forces at the braked wheel
G Force due to weight, F_B Braking force, F_S Lateral force, v_F Vehicle speed, α Slip angle, ω Angular velocity, n Caster.



Adhesion/slip curve
The curve shape differs greatly as a function of road surface and tire condition.



ABS control loop
1 Solenoid-valve unit, 2 Master cylinder, 3 Wheel-brake cylinder, 4 Electronic control unit (ECU), 5 Wheel-speed sensor.



wheel must be kept in the slip range with maximum braking force by a succession of pressure-buildup, pressure-reduction and pressure-holding phases.

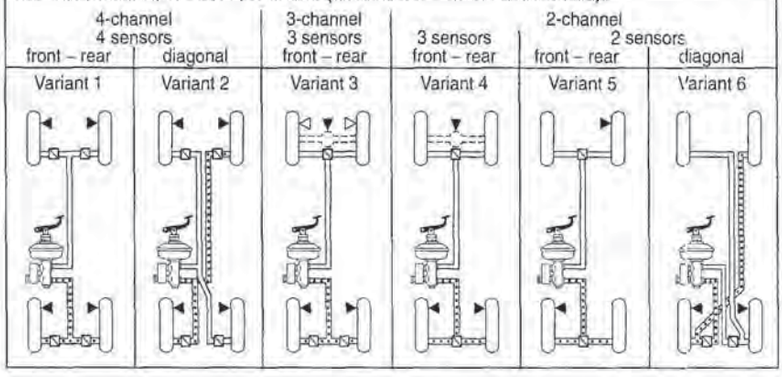
Disturbances in the closed control loop
The ABS system must take the following disturbances into account:
- Changes in the adhesion between the tires and the road surface caused by different types of road surface and changes in the wheel loadings, e.g. when cornering,
- Irregularities in the road surface causing the wheels and axles to vibrate,
- Out-of-roundness, brake hysteresis, brake fading,
- Variations in the pressure input to the master cylinder caused by the driver's brake-pedal actuation,
- Differences in wheel circumferences, for instance when the spare wheel is fitted.

Criteria of control quality
The following criteria for control quality must be fulfilled by efficient antilock braking systems:
- Maintenance of driving stability through provision of sufficient lateral guiding forces at the rear wheels.
- Maintenance of steerability through provision of adequate lateral guiding forces at the front wheels.
- Reduction in stopping distance as opposed to braking with locked-up wheels through optimum utilization of the adhesion between tires and road.
- Rapid matching of the braking force to different adhesion coefficients, for instance when driving through deep water or driving over patches of ice or hard snow.
- Guaranteeing low braking-torque control amplitudes to prevent vibrations in the running gear.
- High level of comfort due to silent actuators and low feedback through the brake pedal.

ABS system variants
A variety of versions are available depending upon the braking-force-distribution concept, the type of vehicle drive concerned, functional stipulations, and costs factor. The overview below shows six system variants, which are described in the following according to the number of channels and sensors.

4-channel systems (variants 1, 2)
These systems permit the individual control of the wheel-brake pressure of each wheel for the rear axle/front axle (II) and for the diagonal (X) braking-force-distribution concepts. When braking on split-coefficient road surfaces though, measures must be taken to ensure that the yaw moment (torque around the vertical axis) cannot adversely affect driving stability. The solution here is to control the wheels on the front axle individually and those on the rear axle in accordance with the "select-low" principle (that is, the rear wheel

ABS system variants
☒ Control channel, ◀ Sensor, ◁ Sensor (alternative to sensor on differential).



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with the lowest coefficient of friction determines the braking pressure applied to both rear wheels).

3-channel system (variant 3)
The yaw moment when braking on split-coefficient road surfaces is reduced to such an extent (due to the system's operating principle) that passenger cars with a long wheelbase, and a high moment of mass inertia about the vertical axis, are well able to control this braking situation.

In the case of the 3- and 4-channel systems, passenger cars with a short wheelbase and a low moment of mass inertia, however, require electronic delay of the yaw-moment buildup. When braking on split-coefficient road surfaces, this causes a delayed buildup of the braking torque at the front wheel with the high coefficient of friction. As a result, the driver has enough time to correct the yaw by an appropriate steering adjustment.

2-channel systems (variants 4, 5, 6)
The 2-channel systems on the one hand need less components than the 3- and 4-channel versions, and this makes them less costly to manufacture. On the other hand though, this is accompanied by a number of functional limitations.

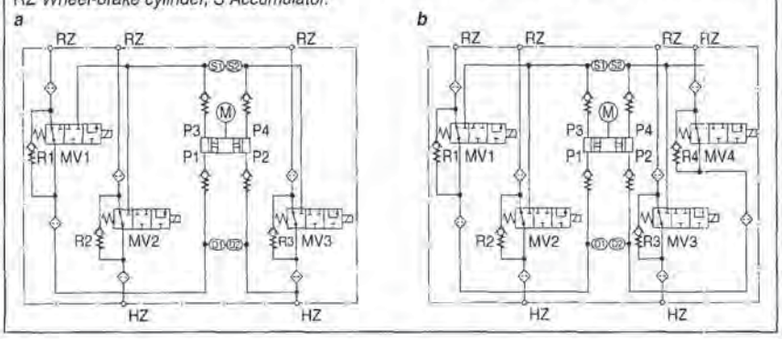
With variant 4 in the select-high mode (the front wheel with the higher coefficient of friction determines the brake pressure applied jointly to both front wheels), practically every time the driver hits the brakes hard (panic braking), one of the front wheels locks up. This is accompanied by a high level of tire wear and reduced steerability. With version 5, this always occurs when the "sensed" front wheel suddenly encounters a higher level of friction coefficient than the "non-sensed" wheel. Variant 6 can be used only with diagonal braking-force distribution (X). In this version, the brake pressures at the front wheels are controlled individually, while the brake pressures at the rear wheels are jointly controlled. Because the front-to-rear distribution of braking force bears the responsibility for ensuring that the rear wheels do not lock, this system provides somewhat lower deceleration rates than a 3- or 4-channel system.

Note: For some light-truck models with front-rear brake-circuit split in the American market there is a simple system consisting of a sensor on the rear differential and a control channel (without return pump) which prevents the rear wheels locking. The front wheels can lock up with sufficiently high braking pressures with the consequent loss of steering function. This system is not an ABS with the range of functions described at the beginning of this section and is used only for certain specialized types of vehicle.

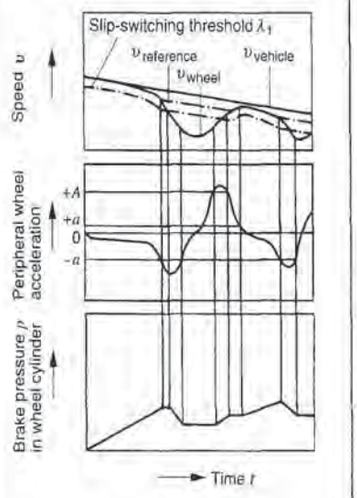
ABS versions

ABS 2S-3-channel/-4-channel systems (Bosch)
In this system, the ABS and the brake booster are separate units.

Functional diagrams
a) Three-channel hydraulic modulator, b) Four-channel hydraulic modulator.
D *Damper*, HZ *Master cylinder*, M *Electric motor*, MV *Solenoid valve*, P *Pump*, *R Control channel*, RZ *Wheel-brake cylinder*, S *Accumulator.*



ABS control cycle for high friction coefficients



The 3-channel hydraulic modulator for the II braking-force-distribution version (front/rear) comprises three solenoid valves with three possible positions, and a return pump with electric drive motor.

In the first, de-energized position, there is an unhindered passage from the master cylinder to the wheel-brake cylinder, with the result that the wheel-brake pressure rises during initial braking and during automatic brake control. In the second, semi-energized position, the passage from the master cylinder to the wheel-brake cylinder is interrupted, with the result that the wheel-brake pressure is kept constant. In the third, fully energized position, the wheel-brake cylinder is connected to the return and the wheel-brake pressure drops.

Pressure dissipation takes only about 20 ms so that the wheel does not lock up; pressure build-up, on the other hand, takes place in several stages with intervening holding phases and typically takes around 200 ms.

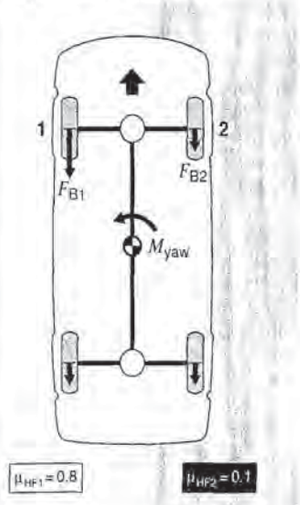
The 4-channel hydraulic-modulator valve for diagonal braking-force distribution calls for four solenoid valves, because the rear-wheel brakes belong to different brake circuits. However, both the rear-wheel valves are jointly energized, so that there is the same pressure in each of the rear-wheel-brakes, and the "select-low" control mode is easy to implement.

The depicted control cycle shows automatic brake control in the case of a high friction coefficient. The change in wheel speed (deceleration) is calculated in the ECU. After the value falls below the $(-a)$ threshold, the hydraulic modulator valve unit is switched to the pressure-holding mode. If the wheel speed then also drops below the slip-switching threshold λ_1, the valve unit is switched to pressure reduction; this is done as long as the $(-a)$ signal is present. During the subsequent pressure-holding phase, the peripheral wheel acceleration increases until the $(+a)$ threshold is exceeded; thereupon, the brake pressure continues to be kept constant. After the relatively high $(+A)$ threshold has been exceeded, the brake pressure is increased, so that the wheel is not accelerating excessively as it enters the stable range of the adhesion/slip curve. After the $(+a)$ signal has dropped out, the brake pressure is slowly raised until, when the wheel acceleration again falls below the $(-a)$ threshold, the second control cycle is initiated, this time with an immediate pressure reduction. In the first control cycle, a short pressure-holding phase was necessary initially for the filtering of disturbances. In the case of high wheel moments of inertia, low friction coefficient and slow pressure rise in the wheel-brake cylinder (cautious initial braking, e.g., on black ice), the wheel might lock-up without the deceleration switching threshold having responded. In this case, therefore, the wheel slip, too, is used in automatic brake control.

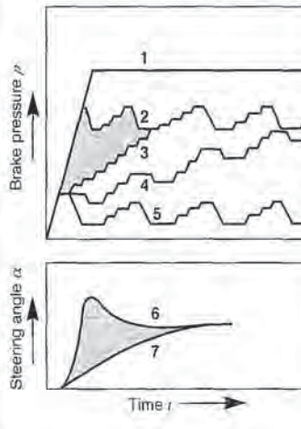
Under certain road-surface conditions, passenger cars with all-wheel drive and with differential locks engaged pose problems when the ABS system is in operation; this calls for special measures to support the reference speed, lower the wheel-deceleration thresholds, and reduce the engine drag torque.

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Buildup of yaw moment induced by large differences in friction coefficients
M_{yaw} Yaw moment, F_B Braking-force, μ_{HF} Coefficient of friction. 1 "High" wheel, 2 "Low" wheel.



Curves for brake pressure/steering angle with delayed yaw-moment buildup *(GMA) 1 Master-cylinder pressure p_{MC}, 2 Braking pressure p_{high} w/o GMA, 3 p_{high} with GMA 1, 4 p_{high} with GMA 2, 5 p_{low} at "Low" wheel, 6 Steering angle α w/o GMA, 7 Steering angle α with GMA.*



Brake control with yaw-moment buildup delay

When the brakes are applied on an asymmetrical road surface (for instance, left wheels on dry asphalt, right wheels on ice), the result is vastly different braking forces at the front wheels. This difference induces a turning motion (yaw moment) around the vehicle's vertical axis.

When a heavy passenger car with ABS is braked, the resulting yaw sets in slowly and the driver has time for corrective steering maneuvers. On smaller cars, the ABS must be supplemented by an additional yaw-moment buildup-delay device (GMA) to ensure that control is maintained during panic stops on asymmetrical surfaces. GMA delays the pressure buildup in the wheel cylinder of the front wheel with the higher coefficient of braking force at the road surface ("high" wheel).

The GMA concept is demonstrated in the diagram: Curve 1 represents the master-cylinder pressure p_{MC}. Without GMA, the wheel on asphalt soon arrives at the pressure p_{high} (Curve 2), while the wheel which is running on ice goes to p_{low} (Curve 5); each wheel achieves the maximum retardation available under the given circumstances (individual control).

The GMA 1 System (Curve 3) is suited for use with vehicles with a less critical response pattern, while GMA 2 is designed for cars which display an especially marked tendency toward yaw-induced instability (Curve 4).

In all cases in which GMA comes into effect, the "High" wheel is under-braked at first. This means that the GMA must always be very carefully adapted to the vehicle in question in order to limit increases in stopping distances.

ABS 2E (Bosch)

This economical "full ABS" design (a 3-position solenoid valve with electronic control stage is replaced by a plunger/floating piston arrangement) offers the same safety features and functions as the ABS2S system with only minimal loss of comfort in terms of brake pedal feedback and noise.

Under normal braking, without active ABS, brake fluid flows to the right rear

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wheel through the rear-axle solenoid valve (4), while the plunger's central valve (6) supplies the left rear wheel.

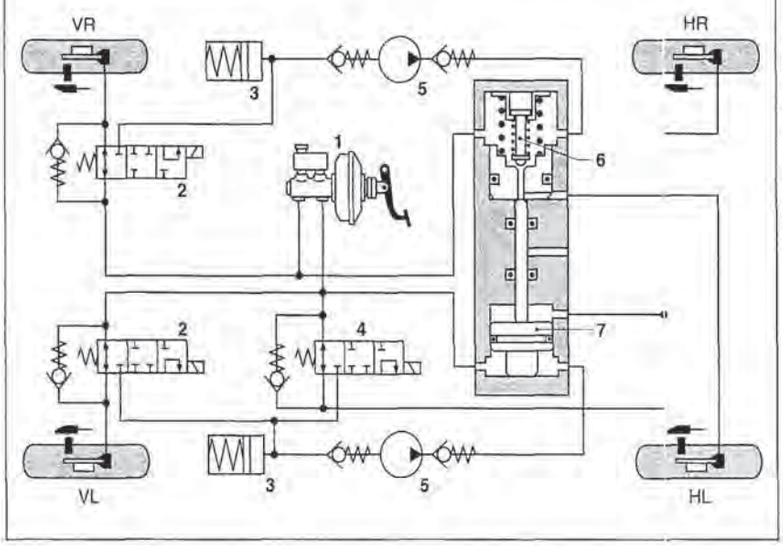
When the ABS is activated, each of the left-side solenoid valves (2) controls a front brake, while the rear-axle solenoid (4) directly controls the rear right wheel (HR). The rear-axle solenoid valve (4) switches to position 3 upon receiving the "pressure-reduce" command from the ECU. In the process, fluid is released from the rear right wheel (HR) into the accumulator chamber (3) via the valve (4) so that the pressure at the wheel brake (HR) is reduced. Since the upper side of the floating piston (7) is hydraulically connected to the wheel brake (HR), this pressure reduction is also applied above the floating piston (7). The underside of the piston (7) is subjected to pressure from the master cylinder (1), and the resulting force on the piston (7) causes the floating piston assembly to move upwards. The upper plunger now comes into contact with the central valve pin (6) so that the central valve closes and volume can be accepted from the rear brake (HL) when upward movement continues. At the same time, the pressure in the brake (LR) and on the underside of the top plunger also drops as required. The movement of the plunger then comes to a standstill when the resulting force on the plunger assembly has dropped to zero. This is the case when the pressure in the two brakes (HL) and (HR) is equal. The processes for "pressure hold" and "pressure buildup" are analogous to the pressure-reduction process and are controlled by the position of the solenoid valve (4).

ABS5 Family (Bosch)

As a result of the development of solenoid valves with two hydraulic positions it has been possible to thoroughly revolutionize the ABS system while retaining the range of functions of the ABS2S. The ABS5 is based upon the tried and proven return principle as used in the ABS2S. It contains the following components for each brake circuit (with X and II braking-force distribution):

- Return pump,
- Accumulator,

ABS2E hydraulic system *(X braking-force distribution)*
1 Brake booster, 2 Solenoid valves, 3 Accumulator chambers, 4 Rear-axle solenoid valve, 5 2-circuit return pump, 6 Central valve, 7 Plunger piston.
V front, H rear, R right, L left.



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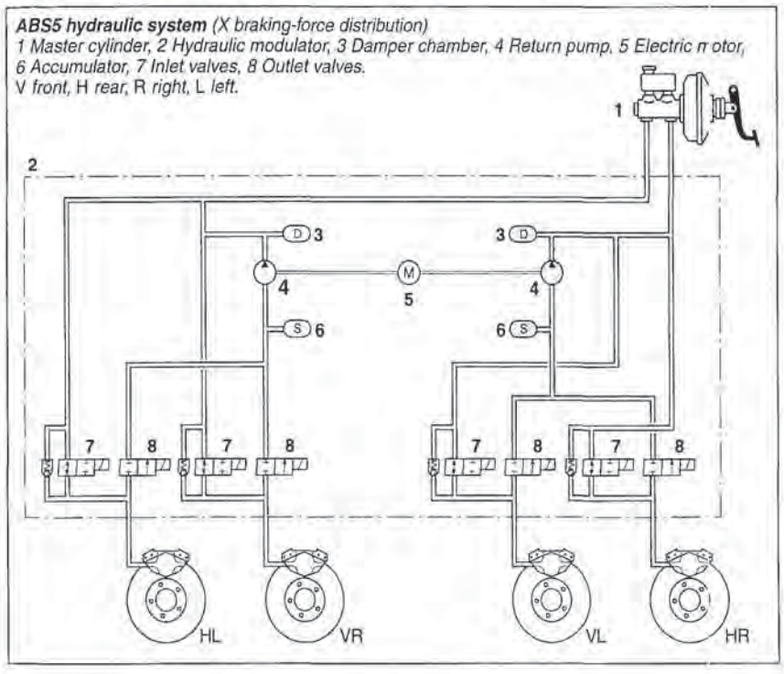
- Damper chamber, and
- 2/2 solenoid valves with two hydraulic positions and two hydraulic connections.

Each wheel (or the rear axle in the case of II braking-force distribution) is allocated a solenoid-valve pair, a solenoid valve (EV – open when de-energized) for the pressure increase (inlet) and a solenoid valve (AV – closed when de-energized) for pressure reduction (outlet). In order to achieve rapid release of pressure at the brakes when the pedal is released, the inlet valves each have a non-return valve which is integrated in the valve body (e.g. non-return valve sleeves or unsprung non-return valves).

Pressure increase and decrease are each allocated a solenoid valve which features only 1 active (energized) position. This leads to compact valve designs with only 2 hydraulic connections, smaller volume, lower weight, and low magnetic forces. The design means that only a single switching transistor is needed for the electrical triggering, with the attendant reduced electrical power loss in the solenoid coils and in the ECU. The valve block for mounting the 2/2 solenoid valves results in a smaller and lighter hydraulic modulator and means that the ECU can be attached directly to the hydraulic modulator. This has the advantage that the vehicle's wiring harness needs less lines. Because of their compact design, the 2/2 solenoid valves enable shorter electrical switching times and even pulse-width-modulated cyclic operation which substantially improves function (e.g. adaptation to changes in coefficient of friction) and control convenience (e.g. smaller delay fluctuations with the aid of pressure stages and lower levels of valve noise).

By coordinated development of mechanical valve design and pulse-width-modulated control (mechatronic optimization), it has been possible to produce an ABS system with "quiet" solenoic valves which represents a new milestone in terms of noise generation and pedal feedback.

Electric-motor variants, together with hydraulic throttling (restriction) of the flow



ABS5 hydraulic system *(X braking-force distribution)*
1 Master cylinder, 2 Hydraulic modulator, 3 Damper chamber, 4 Return pump, 5 Electric motor, 6 Accumulator, 7 Inlet valves, 8 Outlet valves.
V *front,* H *rear,* R *right,* L *left.*

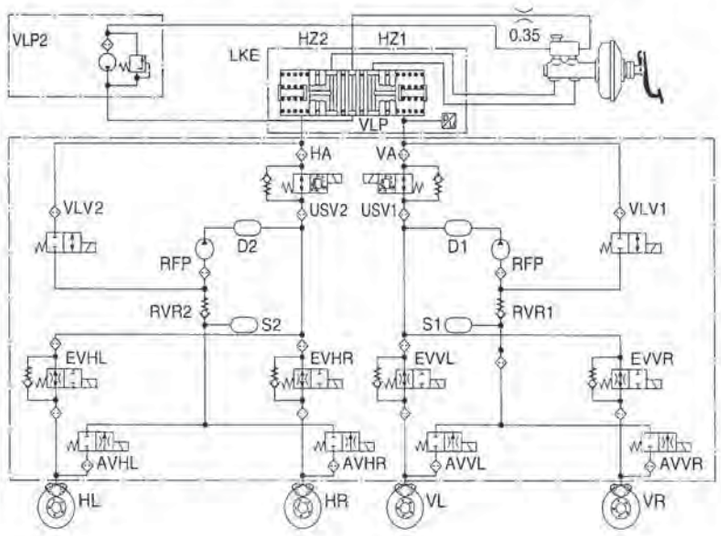
cross sections in the solenoid valves, and variations in the accumulator volumes, mean that the ABS5-family represents a new generation of ABS systems suitable for installation in all series-production passenger cars.

ABS with TCS (Bosch)
See chapter "TCS traction control system," P. 606

ABS and ESP hydraulics (Bosch)
Proper control of vehicle-handling dynamics by means of ESP Electronic Stability Program (previously known as VDC Vehicle Dynamics Control) requires sufficiently rapid build-up of braking force in the brakes. On the one hand, low temperatures and low coefficients of friction are the conditions under which the ESP is called upon most often. On the other though, at low temperatures the brake-fluid viscosity increases considerably, thus making it necessary to modify the hydraulic concept compared to that used with ABS/TCS. The ABS/TCS5 system as described in the Chapter "ABS with TCS (Bosch)" was taken as the basic system. Here, the pre-charge pump (VLP) was incorporated to ensure the return-pump delivery capacity when the ESP is needed under cold-weather conditions. The requirement for closed and separate brake circuits meant that this pre-charge pump could not be connected hydraulically directly upstream of the return pump. Instead it is applied to a charge-plunger unit (LKE) connected between the master cylinder and the ABS/TCS hydraulics. As soon as brake pressure is needed, the pre-charge pump switches on (together with the pre-charge valves (V_V) and the changeover valves (USV)) and delivers brake fluid to the charge-plunger unit (LKE) so that the two plungers in the unit move apart.

This mechanically closes the central valves of the LKE and the brake-fluid volume from the two LKE cylinders is forced through the opened pre-charge valves to the return pumps of the ABS/TCS hydraulics. The pre-charging of the return

Hydraulic system for ABS and ESP
VLP *Pre-charge pump*, VLV *Pre-charge valve*, RFP *Return pump*, RVR *Non-return valve*, LKE *Charge-plunger unit*, USV *Change-over valve*, D *Damper*, AV *Outlet valve*, EV *Inlet valve*, HZ *Master cylinder*, S *Accumulator*, VA *Front axle*, HA *Rear axle*, V *Front*, H *Rear*, R *Right*, L *Left*.



pumps means that sufficiently rapid brake-pressure buildup is ensured at the wheel brakes even at low temperatures. A separate line with a small restriction is provided between the LKE and the reservoir in order that the brake fluid in the pre-charge circuit can be bled, and the LKE plungers return to their initial position when the pre-charge pump is switched off.

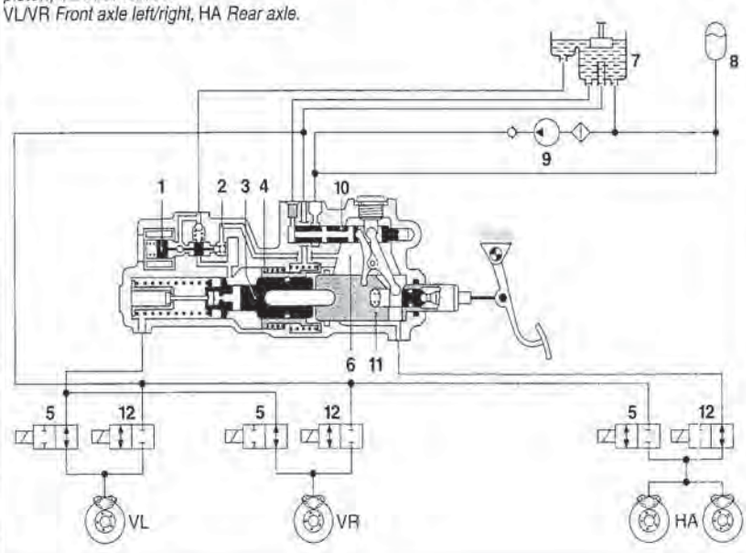
By further optimization of the master cylinder and the ABS/TCS5 induction valves with the aim of increasing the hydraulic flow cross-section, it has been possible to produce an ESP hydraulic modulator that enables adequate pressure generation dynamics even without a precharger pump at low temperatures and thus further reduces the number of system components.

MK 2 (Teves)
The hydraulic components – brake booster and ABS – form a compact integrated unit for installation on the firewall. Under normal braking, the booster piston impels brake fluid directly to the rear brakes while pushing the master cylinder piston to the left, supplying brake fluid to the front brakes.

When the ABS is activated, the main valve opens, connecting the booster chamber with the primary side of the master cylinder piston while closing off the connection between the primary side and the reservoir. Brake fluid flows from the booster chamber to the front brakes via the connecting line and seal of the master-cylinder piston. During ABS-controlled braking, booster pressure is exerted against the left side of the positioning sleeve, maintaining master cylinder and booster piston in a middle position. This ensures that sufficient piston travel remains available for front-wheel braking in the event of ABS failure.

The supply and discharge valves provide optimal regulation of the pressures in the wheel cylinders during ABS-controlled braking maneuvers, with the brake fluid which is ejected from the wheel cylinder flowing back to the reservoir.

Schematic diagram: ABS MK 2 *(Teves)*
Brakes not actuated.
1 Main valve, 2 Connecting line, 3 Master-cylinder piston, 4 Positioning sleeve, 5 Outlet valves, 6 Servo chamber, 7 Fluid reservoir, 8 Pressure accumulator, 9 Pump, 10 Brake valve, 11 Servo piston, 12 Inlet valves.
VL/VR *Front axle left/right,* HA *Rear axle.*



The front brakes are controlled individually. The rear brakes are regulated together, whereby the wheel with the lower coefficient of traction determines the level of pressure (select low).

MK 4 with TCS (Teves)
This version is used with a conventional vacuum booster to provide "separate ABS". The system can also be extended to include TCS.

When the start of active ABS control makes it necessary to reduce the brake pressure at one of the wheels, the discharge valve is opened while the supply valve remains closed, allowing brake fluid to flow back to the reservoir from the brake. When the supply valve opens to increase the pressure, brake fluid flows from the master-cylinder chamber, and the pedal gives way somewhat. The hydraulic unit must supply new energy, as repeated cycles would otherwise cause the pedal to drop too far.

The ABS hydraulic energy-supply unit employs an electrically-driven, dual piston pump which is triggered when the system recognizes incipient wheel lock. The pump extracts brake fluid from the reservoir and pumps it through the supply-valve orifice and to the brake at a suitably increased pressure.

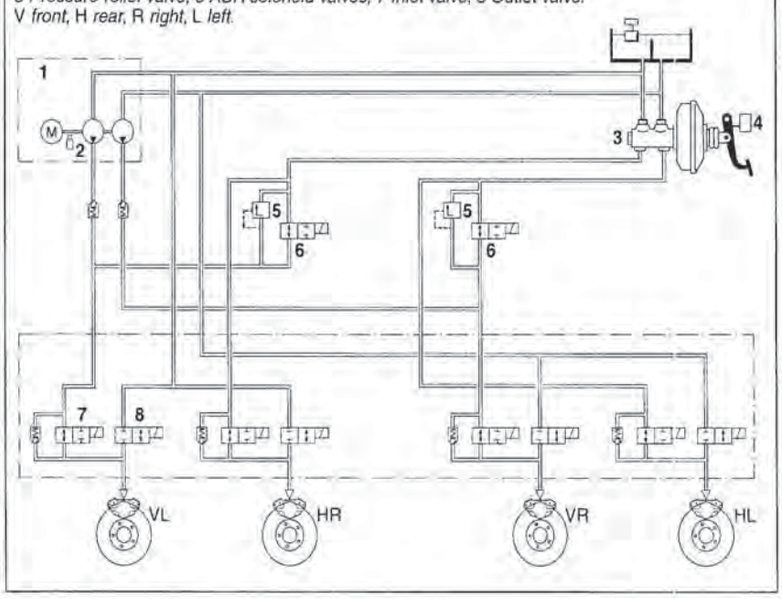
The excess flow quantity from the pump flows into the chambers of the master cylinder, where it presses back the master-cylinder piston and the brake pedal. The position sensor triggers and switches off the pump so as to ensure that the intermittent flow which it produces results in adequate feel and travel at the pedal.

The hydraulics can be expanded for traction control by adding 2 TCS solenoid valves, which serve to isolate the master cylinder from the TCS, and 2 pressure-relief valves, responsible for regulating pressure in the TCS system.

The rotation sensor is a security device for monitoring the operation of the pump motor.

MK 20 Family (Teves)
In contrast to the MK4, the MK 20 system operates according to the return principle using sealed brake circuits. Pressure

Schematic diagram: ABS/TCS MK 4 *(Teves)*
1 Energy supply, 2 Wheel-speed sensor, 3 Master-cylinder piston, 4 Position sensor, 5 Pressure-relief valve, 6 ASR solenoid valves, 7 Inlet valve, 8 Outlet valve.
V *front*, H *rear*, R *right*, L *left*.



modulation is by means of a pair of 2/2 solenoid valves (inlet/outlet) for each wheel and the system can be upgraded to TCS/ESP by the addition of two changeover valves and two induction valves. The hydraulic components thus correspond exactly to the arrangement employed by the Bosch ABS5 Family.

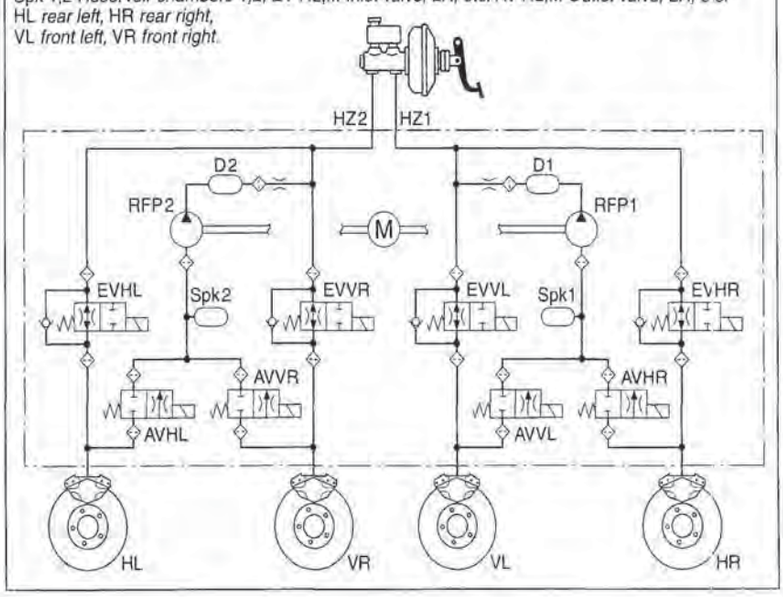
Pumpless ABS (Denso)

For a certain section of the Japanese market, a greatly simplified ABS system was developed which dispenses with damper chambers and return pumps together with their electric motors and control systems. It operates according to the following principle:

If the wheel-speed sensors detect that wheels are about to lock under braking, the appropriate inlet valves are closed in order to prevent further increase of pressure in the wheel cylinders. If this does not succeed in preventing the wheels locking, the corresponding pressure release valve is opened until the wheel stops slipping and is then closed again. The brake fluid that flows out of the wheel cylinders in the process returns to the reservoir chambers. Once the wheel speed becomes stable again, the inlet valves concerned are opened again. The process described repeats itself continually until the driver stops braking or the vehicle comes to a standstill. When the brake pedal is released by the driver, the brake fluid that has collected in the reservoir chambers flows back through the reservoir-chamber springs and the non-return valves to the master cylinder. The system is then ready for ABS braking again.

The smaller number of components compared with conventional systems offers considerable cost savings but the system has significant functional deficiencies. Among other things, under extended periods of ABS braking (e.g. from high speeds on road surfaces with poor adhesion or in situations where adhesion fluctuates dramatically), the reservoir chambers can reach their capacity while at the same time the brake-pedal travel increases considerably in an unfamiliar way. Apart from the loss in pedal-opera-

Schematic diagram: ABS MK 20 *(Teves)*
HZ 1,2 *Master cylinder 1,2,* D 1,2 *Damper chamber 1,2,* RFP 1,2 *Return pump 1,2,*
Spk 1,2 *Reservoir chambers 1,2,* EV HL,... *Inlet valve, LR, etc.* AV HL,... *Outlet valve, LR, etc.*
HL *rear left,* HR *rear right,*
VL *front left,* VR *front right.*



tion comfort, this means that the pressure in the wheel cylinders can not be reduced any further with the result that the wheels could lock up. For that reason, this type of system is only used on certain vehicles with limited top speeds and has not as yet become established in the European or American automobile markets.

ABS components (Bosch)

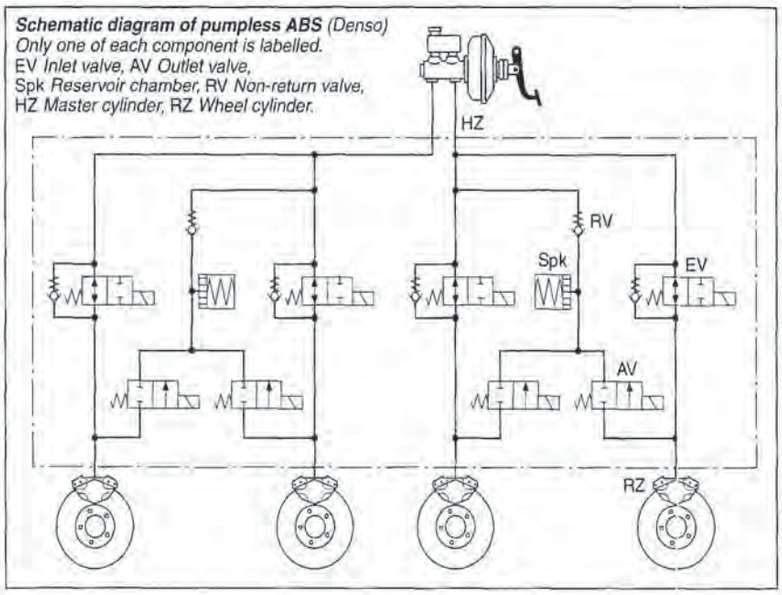
Wheel-speed sensor
The inductive wheel-speed sensor provides the control unit (ECU) with information on wheel speed.

ECU unit with vehicle-specific LSI circuits
The ECU shown in schematic form in a 4-channel system receives, filters and amplifies the speed-sensor signals and ascertains from them the degree of wheel slip and the acceleration of the individual wheels as well as the reference speed, which is the best possible calculation of vehicle road speed.

Input circuit:
The input circuit consists of a low-pass filter and input amplifier; the circuit suppresses interference and amplifies the signals from all wheel-speed sensors (Channels 1...4).

Digital controller:
The digital controller consists in the case of ABS2S of two identical but independent digital vehicle-specific LSI circuits which each process the information from two wheels (channels 1+2 or 3+4) simultaneously and perform the logical processes. Once processed, the wheel frequency data continues in the circuit to a serial arithmetic-logic unit. This logic unit, in turn, uses the data to calculate the values for "wheel slip" and "circumferential deceleration" or "circumferential acceleration" required for closed-loop control. A self-adapting, complex controller logic converts the control signals into position commands for the solenoid valves. A serial interface, connected to the input stage, arithmetic-logic unit and controller logic via data link, maintains data communications between the two digital LSI circuits.

Yet another function block contains the monitoring circuit for error recognition and analysis. Should the ECU malfunction, a warning lamp informs the vehicle operator that the ABS is no longer opera-



Schematic diagram of pumpless ABS *(Denso)*
Only one of each component is labelled.
EV *Inlet valve,* AV *Outlet valve,*
Spk *Reservoir chamber,* RV *Non-return valve,*
HZ *Master cylinder,* RZ *Wheel cylinder.*

tional. However, the braking system retains full normal operating capability when the ABS is deactivated.

Output circuits:
Two output circuits function as current regulators for Channels 1 + 2 and 3 + 4 while receiving the position commands employed for solenoid excitation from the LSI circuits.

Output stage:
The output stage employs the input from the current regulators in the two output circuits in providing the excitation current for the solenoid valves.

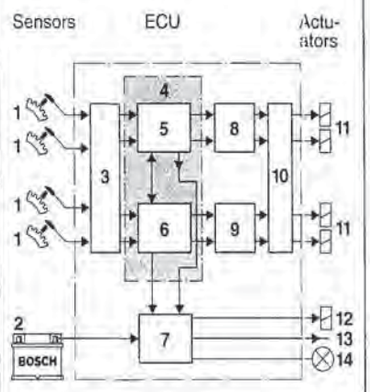
Voltage stabilizer, fault memory:
This function block stabilizes the supply voltage and monitors it to ensure that it remains within the tolerances required for reliable operation. The block also incorporates undervoltage recognition, which reacts to insufficient on-board voltage by shutting down the unit, as well as relays and the warning-lamp control circuit.

ECU unit with microcontrollers
Instead of the vehicle-specific LSI circuits, this control unit has two microcontrollers which take care of signal processing, "running" the controller program and self-monitoring on the part of the ABS system. The unit also carries out diagnosis in accordance with ISO standards, making it possible to track down defective ABS components with the aid of either the warning lamp or an "intelligent" tester.

The use of microcontrollers could achieve substantial optimization of controller algorithms incorporating customization to vehicle and driver requirements. For example, more precise calculation of wheel slip enables early detection of excessive rear-wheel braking which can then be corrected in time by changes in brake pressure with the result that, on some vehicles, the rear-wheel pressure reducer can be dispensed with. This achieves better overall vehicle braking (electronic brake balancing).

Further hybridization of the electronic components has succeeded in considerably reducing their number and thereby the size of the control unit as well.

***ECU** (Bosch, 4-channel layout).*
1 Wheel-speed sensors, 2 Battery, 3 Input circuit, 4 Digital controller, 5 LSI circuit 1, 6 LSI circuit 2, 7 Voltage stabilizer/fault memory, 8 Output circuit 1, 9 Output circuit 2, 10 Output stage, 11 Solenoid valves, 12 Safety relay, 13 Stabilized battery voltage, 14 Warning lamp.



Hydraulic modulator (for ABS2S and ABS5S with II and X braking-force distribution)
For each brake circuit, the hydraulic modulator consists of:
– Electric-motor-driven return pump P,
– Accumulator chamber S,
– Damper chamber D, and
– Various solenoid valves.

Return pump P:
The return pump returns the brake fluid from the wheel-brake cylinders to the master cylinder.

Accumulator S:
The accumulators provide temporary storage of the large quantity of brake fluid which accompanies pressure reduction.

Damper chamber D:
The dampers and their downstream throttling devices serve to smooth the high levels of pulsation which occur during the return of brake-fluid to the master cylinder.

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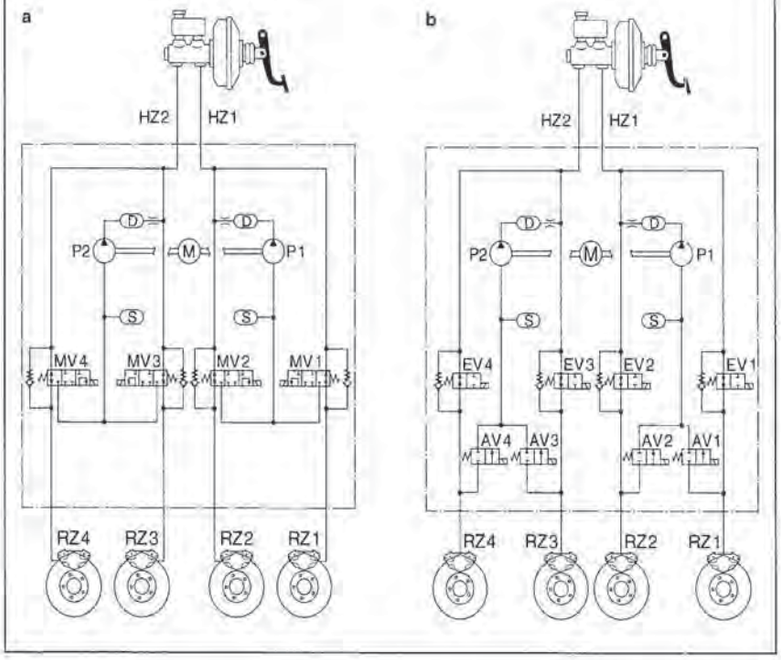
They ensure that the noise level is kept to a minimum.

3/3 solenoid valves on ABS2S:
Each wheel is allocated a 3/3 solenoid valve. The valve serves to modulate the pressure in the wheel cylinders during active ABS control. The modulation takes place in 3 modes (build-up, hold, and reduce).

2/2 solenoid valves on ABS5:
Each wheel is allocated an EV/AV valve pair. The same pressure-modulation modes as above can be achieved for each wheel brake by appropriate control of the valves.

In addition to the effect of the damper chambers, PWM control of the 2/2 solenoid valves allows the required pressure changes in the wheel cylinders to be achieved with a greater degree of driver comfort in terms of noise, initiation of suspension vibration and pedal feedback.

Schematic diagram of hydraulic modulator
a) ABS2S system, b) ABS5 system.
HZ *Master cylinder,* M *Electric motor,* P *Plunger pump,* D *Damper,* S *Volume accumulator,* MV *3/3 solenoid valve,* EV *2/2 inlet valves,* AV *2/2 outlet valve,* RZ *Wheel cylinder.*



Electrohydraulic Brakes (EHB)

In contrast to antilock braking systems (ABS), the traction control system (TCR) and electronic stability program (ESP) can generate pressure in the wheel cylinders independently of driver action. This capability forms an important basis for electrohydraulic brakes.

Function

In conventional car braking systems, the force applied by the driver is transmitted mechanically by the lever action of the brake pedal to the vacuum brake servo and from there in amplified form to the master cylinder. The pressure thus generated is used to achieve the desired braking effect with the individual brakes on each wheel. With electrohydraulic brakes, that purely mechanical-hydraulic sequence of actions is broken and replaced by sensors, an ECU and a hydraulic pressure supply. Under normal operating conditions, there is no mechanical link between the brake pedal and the wheel brake.

Design

An electrohydraulic braking system consists of the following components (see diagram):

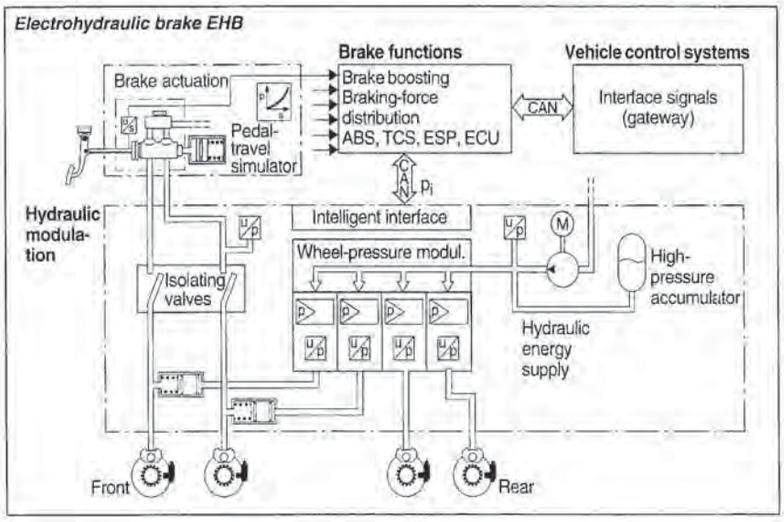
- Actuator unit,
- Hydraulic pressure modulator
- Sensors (e. g. wheel-speed sensor),
- Add-on ECU (at the hydraulic pressure modulator),
- or separate ECU,
- Control and pressure lines.

Mode of operation

For safety reasons, two separate sensors (one on the actuator unit for detecting the pedal travel and a pressure sensor on the hydraulic modulator) are used to detect a "braking request" and transmit it to the ECU. Using software, this ECU also incorporates the brake servo, ABS, TCS and ESP functions. The other sensors in the ABS, TCS and ESP systems provide the ECU with data relating to aspects of vehicle dynamics such as road speed, cornering, and the motion of the wheels. Using this information, the ECU calculates the signals to be sent to the hydraulic modulator which are converted by the wheel pressure modulators into the brake pressures for the individual wheels. An electrically driven hydraulic pump with a high-pressure accumulator and a pressure monitoring system provides the hydraulic pressure supply.

For safety reasons, in the event of a fault in the system it switches to an operating mode in which the vehicle can be braked without power assistance.

Electrohydraulic brake EHB



Braking systems for commercial vehicles above 7.5 t laden weight

System and configuration

Power-braking systems of medium and heavy-duty commercial vehicles mostly operate with:

- Compressed air (normal pressure level) as a medium for energy supply and as a transmission device.
- Pneumatic-hydraulic power transmission in the service-braking system and pneumatic power transmission in the parking-brake system, or
- Pneumatic high pressure level (Figs. B & C).

Service-braking system for towing (tractor) vehicles

For practical operations, in medium and heavy commercial vehicles, the force exerted by the driver's foot alone is not sufficient to generate adequate braking deceleration. For this reason, compressed-air power-brake systems are principally used; these systems use compressed air as a stored form of energy to control or actuate the service-braking system. Diaphragm actuators create the brake application forces on the wheel brakes. "Air-over-hydraulic" brake systems are becoming less and less common, as the air pressure required to control the wheel brakes must be converted into hydraulic pressure by means of brake servo-unit cylinders, compact brake assemblies or actuating cylinders.

A modern dual-circuit compressed-air brake system with trailer-brake connection, spring-type brake actuator, secondary and parking-brake system consists of the following main components:

- Energy supply,
- Compressed-air reservoir,
- Brake valves,
- Braking-force controller,
- Wheel brakes, and
- Control and air supply for the trailer braking system (diagram D).

The energy supply comprises a compressor and pressure regulator. Where necessary, antifreeze pumps, automatic water drainage, air filters, air dryers and intermediate reservoir are added, providing clean and well-drained air.

As there may be a number of air-consuming installations apart from the braking system in the tractor-trailer combinations used in modern-day commercial-vehicle operations, it is advisable to install considerably more powerful compressors than those, for instance, required by EC/ECE brake regulations.

The four-circuit protection valve is located at the interface point between the energy supply and the compressed air reservoir. In the event of a fault, this protection valve ensures the security and continuing supply of the individual brake circuits and of priority secondary consumers.

The actuating mechanism for the service brakes starts with the brake pedal and ends at the mechanically operated components of the dual-circuit brake valve.

Sometimes, in commercial vehicles, the limited space available makes it necessary to fit the brake valve on the frame behind the cab instead of in the (tipping) cab itself. The problem of left-hand and right-hand drive trucks with the brake valve in the same position on the frame is solved by means of a dual-circuit hydraulic transmission device for the braking-circuit control. In this process an electrical warning system monitors the level of fluid in the compensating reservoirs of both circuits.

Additional installations in towing vehicles for pulling trailers/semitrailers with compressed-air brake systems (see "Basic components of a braking system") help to supply air to the trailer/semitrailer braking system, and provide a controlling function with regard to the braking effect (trailer control valve, coupling heads, etc.).

Two-line braking system for trailers

In the case of the two-line braking system, which is the standard European design version, one line (supply line) connects the energy reservoirs in the towing vehicle with those in the trailer/semitrailer; and this line is permanently under pressure.

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The second line (brake line) leads from the trailer control valve in the towing vehicle to the trailer brake valve in the trailer/semitrailer (diagram E). Braking occurs as a result of pressure increase. Automatic braking when the trailer/semitrailer accidentally comes loose from the towing vehicle is ensured in this system by the supply line. If the supply line comes loose or breaks, air flows out of it and the brake valve in the trailer/semitrailer activates the brakes. With the aid of a dual-circuit trailer control valve and the four-circuit protection valve, it is possible to continue the supply of air to the trailer/semitrailer and to control braking, even if one circuit of the dual-circuit braking system in the towing vehicle fails. The standardized coupling heads for "supply" and "brake" are fitted with an automatic shut-off element, which opens during the coupling process.

Automatic load-sensitive device for braking-force metering

The automatic device for correcting braking forces as a function of load (ALB) is a vital element of the braking-system transmission device. With the vehicle partially-loaded or empty, braking-force metering valves permit the braking forces to be adjusted to the reduced axle loading e.g., by sensing of the axle ride clearances) and thus permit a correction of braking-force metering to the vehicle's axles ("kinked load-sensitive braking-force metering"), or they permit a set level of braking (important for vehicles in road train or tractor-trailer operation). There are two kinds of load-sensitive braking-force metering valves (Fig. A):

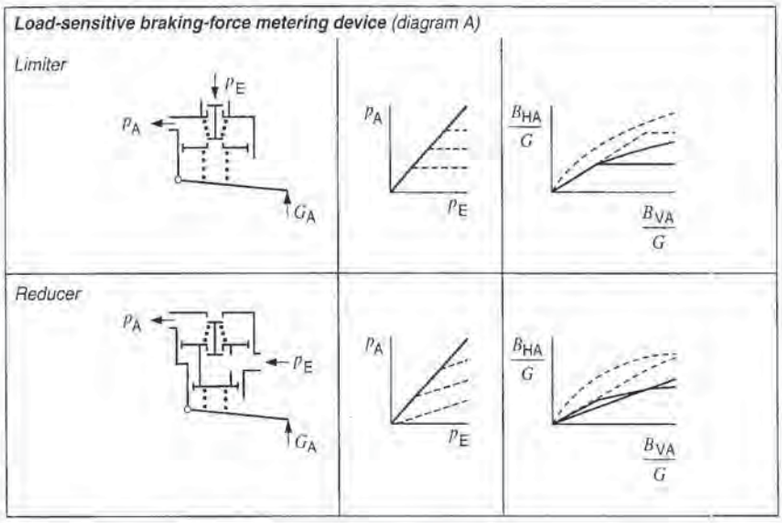
1. Braking-force limiters

Above a given "switchover point", the braking-force limiter restricts the increase of braking force (e.g., on the rear axle), that is to say there is a "kink" in the braking-force metering characteristic.

2. Braking-force reducers

Even under most unfavorable load conditions, load-sensitive braking-force reducers permit braking-force metering which approaches the parabola of the dynamic (ideal) braking-force metering characteristic (see "Design of a braking system"). In the area above the switchover point the braking forces on the particular axle are reduced with respect to the original braking-force metering level. Here, the installed braking-force metering is dependent on the transmission ratio and on the switchover pressure (which in turn is dependent on axle load) of the braking-force metering valve.

Load-sensitive braking-force metering device *(diagram A)*

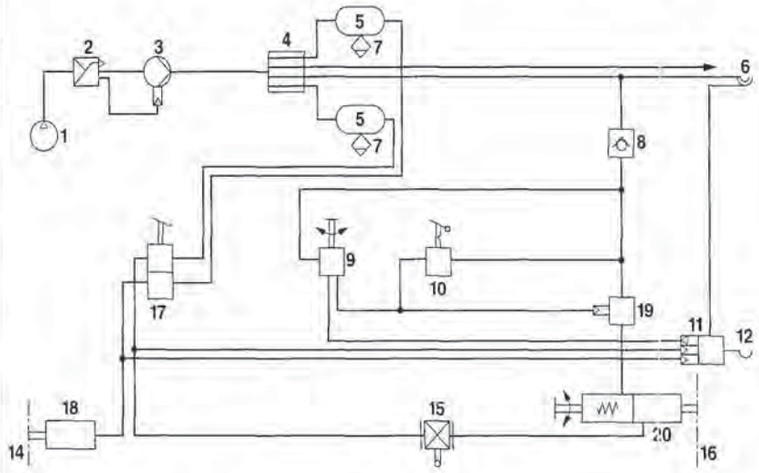


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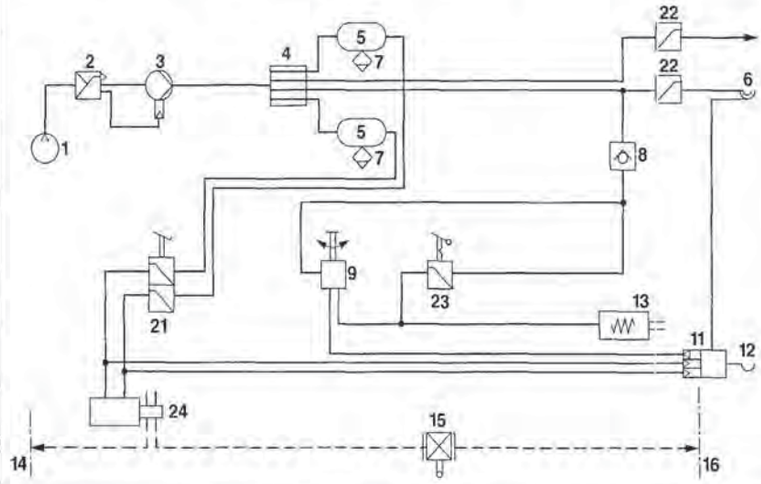
Compressed-air braking systems for trucks

1 Compressor,
2 Pressure regulator,
3 Anti-freeze pump,
4 Four-circuit safety valve,
5 Compressed-air cylinder,
6 Coupling head with automatic shutoff valve,
7 Water drain valve,
8 Non-return valve,
9 Test valve,
10 Parking-brake valve,
11 Trailer-control valve,
12 Coupling head without shutoff valve,
13 Spring-type brake cylinder,
14 Front wheels,
15 Automatic Load-sensitive Braking-force metering (ALB),
16 Rear wheels,
17 Service-brake valve,
18 Brake cylinder

Dual-circuit two-line power-brake system with pneumatic transmission device *(Fig. B)*

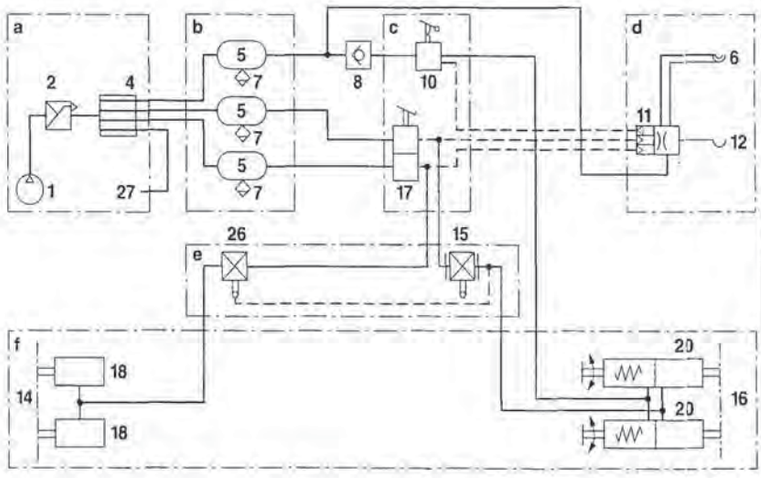


Dual-circuit two-line power-brake system ("Air-over-hydraulic" braking system) with hydraulic transmission device *(Fig. C)*

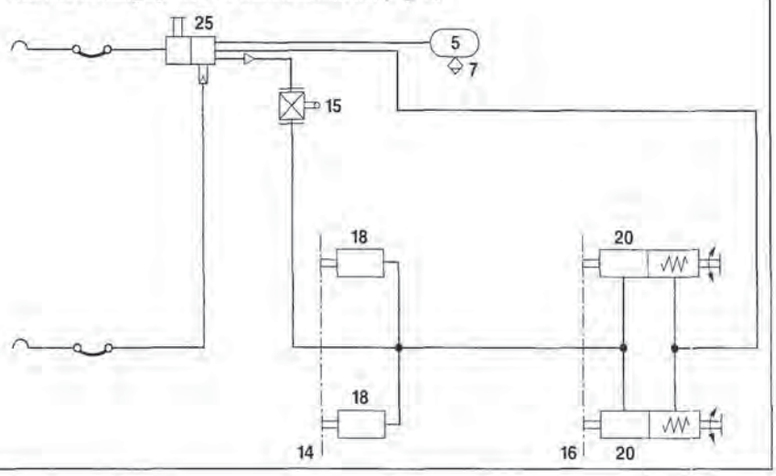


19 Relay valve,
20 Combination brake cylinder,
21 Service-brake valve with pressure limiter,
22 Pressure limiter valve,
23 Parking-brake valve with pressure limiter,
24 Precharger cylinder, dual-circuit,
25 Trailer-brake valve,
26 Loaded/empty valve,
27 Ancillary systems, (e.g. Engine braking system).

Main components of a modern compressed-air power-brake system *(Fig. D)*
a) Energy supply device, b) Reservoir, c) Brake valves, d) Trailer control and supply device, e) Braking-force control, f) Wheel brakes.



Two-line braking system for trailer/semitrailer *(Fig. E)*



Basic comments on "kinked" braking-force metering

Load-sensitive braking-force metering devices adjust the installed braking-force metering to the dynamic (ideal) metering level and prevent the wheels of a given axle from locking prematurely. However, at low adhesion coefficients and low rear-axle load, it is possible for the wheels to lock. This is because the metering device's switchover point can enter the unstable range of braking-force metering if high engine braking torque (retarder braking torque), tolerance fluctuations in the metering device and/or high fluctuations at the wheel brakes occur.

A braking system which works precisely under all braking conditions is only possible if braking-force metering is optimized. Commercial vehicles with extreme differences between empty and fully laden conditions require braking systems with automatic load-sensitive braking-force control (ALB) on the rear axle together with empty/loaded valves (in order to increase the working range of the ALB, Fig. F).

Wheel brakes

In future, disc brakes will be fitted more and more on medium- to heavy-duty commercial vehicles (at least on the towing-vehicle's front axles). Currently, drum brakes are prevalent worldwide.

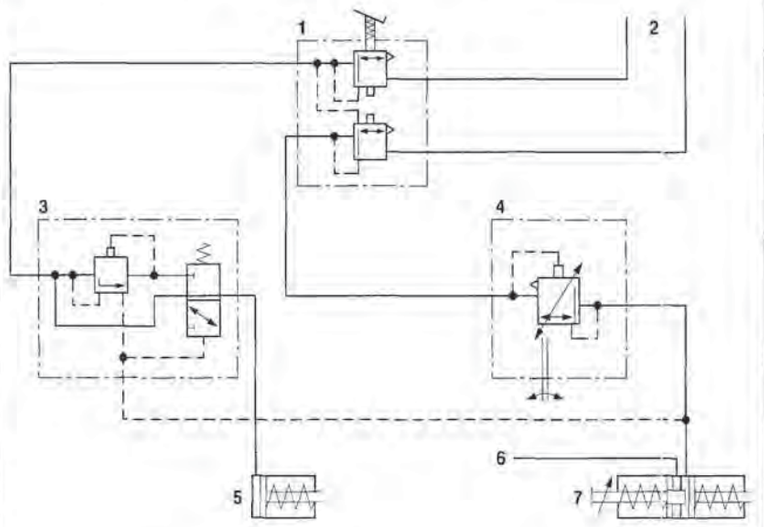
The brake factor C^* as an assessment criterion for brake performance, indicates the ratio of braking force to actuating force. This value takes into account the influence of the internal transmission ratio of the brake as well as the friction coefficient, which in turn is mainly dependent on the parameters speed, brake pressure and temperature (Fig. G).

Drum brakes

The actual construction of the drum brake depends upon the requirements imposed by brake-shoe actuation, anchorage, and adjustment.

Simplex drum brakes (Fig. H): These differ in particular according to their type of application (floating, fixed) and type of support or anchorage (rotating shoes,

ALB function diagram for four-wheel vehicles *(Fig. F)*
1 Service-brake valve, 2 From pressure supply, 3 Load/empty valve (FA), 4 ALB valve (RA), 5 Diaphragm actuator (FA), 6 From parking brake, 7 Combination brake cylinder (RA). RA = Rear axle, FA = Front axle.



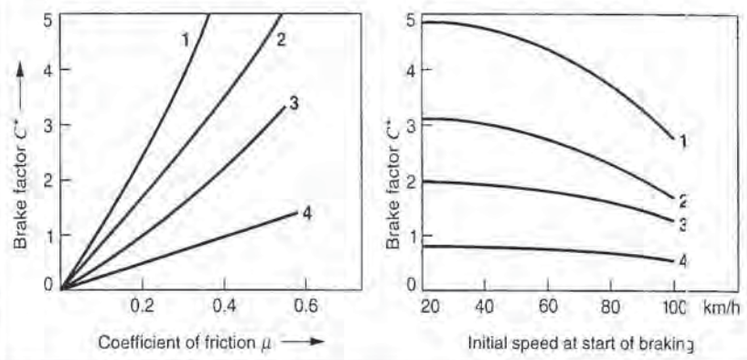
sliding shoes). Wheel brakes with floating-brake application and rotating-shoe support are common. In the case of hydraulic braking-force actuation, for instance, the brakes are applied by means of floating pressure pistons whose travel is not fixed, and which develop actuating forces which are equal in both directions. One of the shoes is the leading shoe and the other the trailing shoe. In the former, the frictional forces between the brake lining and the brake drum support the actuating force, whereas in the latter the frictional forces oppose it.

In the case of the Simplex drum brake, C^* is the sum of the values for the individual shoes, and is = 2.0 (referred to a coefficient of friction of $\mu = 0.38$; it always appears in the following C^* observations as the basis value). A disadvantage of this design is the considerable difference in the braking effect between the two brake shoes, and the resulting greatly increased wear on the leading shoe as compared to the trailing shoe.

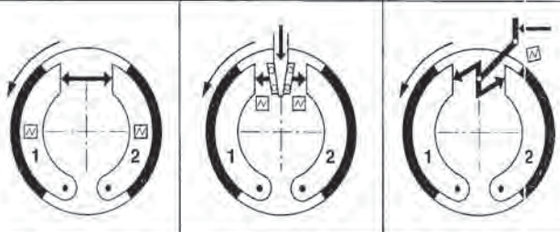
For this reason, the trailing shoe often has a much thinner lining than the leading shoe.

The simplex drum brake can also be actuated by means of a wedge unit (with integrated adjusting mechanism). This has become more and more prevalent,

Brake factor C* as a function of the coefficient of friction and initial road speed *(Fig. G)*
1 Duo-servo drum brake, 2 Duo-duplex drum brake, 3 Simplex drum brake, 4 Disc brake.



Simplex drum brakes *(Fig. H)*

Design	Rotating shoe	Wedge	S-cam
Operating principle			
Brake factor	$C^* = C_1 + C_2$		$C^* = 4/(1/C_1 + 1/C_2)$
Brake shoes	1 Leading shoe, 2 Trailing shoe		

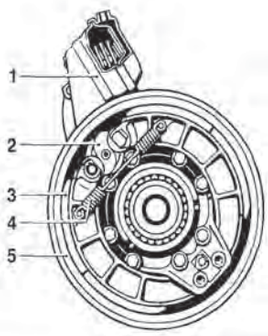
particularly in light and medium commercial vehicles with compressed-air braking systems (Figs. H, K).

The type of wheel brake used most often in heavy-duty commercial vehicles is the pneumatic S-cam simplex drum brake with fixed application (Fig. J).

Advantages:
- Uniform lining wear on leading and trailing shoes as a result of fixed application,
- Long lining life,
- An application mechanism which is simple, reliable and insensitive to temperature. It comprises diaphragm cylinder, automatic slack adjuster, brake shaft and S-cams,
- Little change in brake factor C^*,
- Simple operation of the parking-brake system via spring-brake actuators,
- Precise adjustment by means of automatic slack adjusters.

Disadvantages:
- High internal forces and thus relatively heavy brake construction, as unequal cam forces occur and lead to high free bearing forces,
- Relatively low brake factor C^*, which means considerable application work when braking,
- Due to the roughly equal application travel of leading and trailing shoes, compared to individual shoes, the application forces behave in the opposite manner.
- For the same coefficient of friction, the brake factor C^* is somewhat lower than that of simplex brakes with hydraulic or pneumatic application.

Simplex drum brake with S-cam *(Fig. J)*
1 Diaphragm actuator, 2 S-cam, 3 Brake shoes, 4 Return spring, 5 Brake drum.



Duo-duplex drum brakes: The Duo-duplex brake with two leading shoes (for instance with wedge-unit control) (Fig. K) is rarely used nowadays.

This brake features floating application and the resulting sliding shoe anchorage. An advantage of this brake type is the practically equal brake-lining wear on both shoes and the significantly higher internal transmission ratio in comparison to simplex drum brakes. Twin leading shoes achieve characteristics of $C^* \approx 3.0$ which, however, can not be sustained constantly for long periods due to the tendency of this type of drum brake to fade.

Duo-servo drum brakes: In the past, these were widely used in light commercial vehicles (particularly on the rear axle). This brake's primary characteristic is the fact that both when driving forwards and when reversing, the support force of the primary shoe is used as the actuation force for the secondary shoe. The brake factor is $C^* \approx 5.0$.

The popularity of the duo-servo brake lies in the fact that the high brake factors achieved permit even relatively heavy vans and light-duty trucks up to a weight of approx. 7.5 t to be equipped with vacuum-assisted braking systems. At the same time, the manually operated parking brake incorporated in such systems generates a braking torque of considerable magnitude. However, under conditions of high thermal stress, significant brake-factor fluctuations occur. This fact limits the range of application of this type of brake and necessitates a braking-force metering system which is precisely adapted to the vehicle in question. In future wheel-braking systems, duo-servo drum brakes will for the most part no longer be used for the service-braking system.

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Disc brakes

Apart from their use in express coaches, disc brakes are currently fitted to medium and heavy commercial vehicles.

The advantages of disc brakes in comparison to drum brakes are:
- Brakes can be applied with far more sensitivity,
- Equal wear of the inboard and outboard brake pads if the appropriate degree of heat dissipation is provided,
- Less tendency to brake noise,
- Relatively constant characteristics with minimal fade tendency.

Disadvantages:
- Shorter brake-lining life,
- Usually higher acquisition and operating costs (in comparison to drum brakes).

The high degree of adaptive braking required at high highway speeds is handled better by disc brakes. The brake discs are less susceptible to cracking than the drums of the drum brakes, apart from which disc brakes are less subject to fading. The brake factor of the disc brake is $C^* \approx 0.76$, referred to the basis value of $\mu = 0.38$.

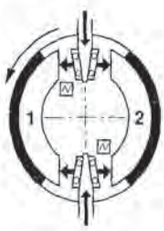
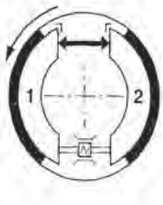
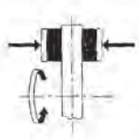
Floating-caliper disc brakes are currently replacing the fixed-caliper brakes used in the past. This development is the result of efforts to design lighter and cheaper brake assemblies which are more temperature-resistant. Floating calipers have a positive effect on variability and consistency of braking effect.

Automatic adjustment of wheel brakes

Brake-lining wear increases the clearance between brake lining and brake drum, and thus increases the braking distance. If the clearance is not adjusted correctly, in extreme cases the piston stroke in the brake cylinder may increase to such an extent that there is no braking effect. Automatic adjustment to the correct clearance is effected when the wheel brake is released.

When the vehicle is braked, the piston stroke in the brake cylinder necessary to bridge the total clearance can be divided into three sections:
- Preset constructive clearance between brake lining and brake drum/disc,
- Clearance resulting from lining wear,
- Clearance dependent on the elasticity of the brake drum/disc and of the brake linings, as well as on the transmission of force between brake cylinder and wheel brake ("elasticity clearance").

Duo-drum brakes and disc brake *(Fig. K)*

Design	***Duo-Duplex*** *Wedge*	***Duo-Servo*** *Positive-action adjustment*	***Disc brake***
Operating principle			
Brake factor	$C^* = C_1 + C_2$	$C^* = C_1 + C_2(k_1 + k_2 \cdot C_1)$	$C^* = 2 \cdot \mu$
Brake shoes	1 Leading shoe, 2 Trailing shoe		–

A slack adjuster automatically ensures the correct adjustment (Fig. L).

Parking-brake system

The braking systems with spring-type brake actuators which are usual in commercial vehicles above 7.5 t are a convenient form of parking-brake and service-braking system. In these, in a purely compressed-air braking system, spring-type brake cylinders from the parking-brake system and diaphragm brake cylinders from the service-braking system are combined.

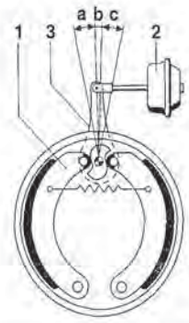
In the disengaged position the four-circuit protection valve and the handbrake valve connect the reservoirs of the service-braking system with the spring compression chamber, and maintain the spring under tension. In vehicles with a connection for the trailer/semitrailer braking system a buffer reservoir is also located in this line. When the handbrake valve is actuated, the pressure in the compression chamber is reduced. Consequently partial braking occurs at first and, with the handbrake valve still actuated, a pressure reduction down to the "surrounding conditions" takes place, and thus full braking of the spring-brake actuator (secondary braking), occurs. Further actuation defines a "Parking setting". With the aid of a further lever setting, in vehicle combinations only the tractor vehicle is braked, and not the entire tractor-trailer rig. As well as this test setting for checking the efficiency of the mechanical parking brake with a trailer attached, EU/ECE legislation for instance stipulates a safeguard emergency-air supply, as well as a nine-fold actuation and release using the energy reserve, a warning device indicating that the spring-brake is beginning to function, and an auxiliary release device.

Retarder braking systems (additional retarding braking systems)

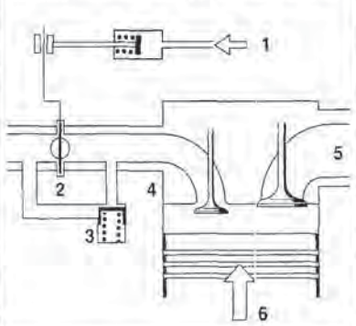
The wheel brakes used in passenger cars and commercial vehicles are not designed for continued retarding operation. In a prolonged period of braking (e.g., when driving downhill) the brakes can be thermally over-stressed, causing the braking effect to be reduced ("fading"). In extreme cases (particularly if the service-braking system has been badly maintained) this may even lead to complete braking-system failure. In order to permit continuous downhill braking, therefore, particularly vehicles with a high laden weight are frequently fitted with a wear-free supplementary retarding braking system in addition to their normal wheel brakes. This system is independent of the wheel brakes, and is also used for braking the vehicle so as to comply with speed limits.

This reduces brake wear as well as increasing braking comfort for the driver. In commercial vehicles two basic types of

Brake clearances *(Fig. L)*
a Due to elasticity. b Due to wear. c Constructive. 1 Brake shoe, 2 Diaphragm cylinder, 3 Automatic slack adjuster.



Exhaust brake with additional pressure-control valve *(Fig. M)*
1 Exhaust-flap actuation (compressed air), 2 Bypass, 3 Pressure-control valve, 4 Discharge, 5 Intake, 6 Piston (4th exhaust stroke).



additional retarding braking system are used:

1. Exhaust (engine) braking system

The exhaust braking power is comprised of the drag power together with the braking power (effected by the throttling of exhaust-gas flow in the exhaust stroke). The maximum drag power of standard engines is 5...7 kW/l depending upon engine swept volume. In contrast, standard engines with conventional exhaust brakes ("exhaust-flap brakes") achieve braking powers of 14...20 kW/l. A further increase in exhaust braking power is only possible by means of additional design modification. Decompression exhaust brakes (e.g., "C-brake", "Jake brake", "Dynatard" or "Powertard") as well as the constant-throttle exhaust brake can significantly improve braking power (Fig. P).

Exhaust brake system with exhaust flap:

The exhaust-flap brake is still the most common system in use today. In this system the driver is able to close a butterfly valve in the exhaust tract by means of compressed air (relatively little constructional outlay is involved). As a result, a counterpressure is generated in the exhaust-gas system which must be overcome by each piston during its exhaust stroke (Fig. M).

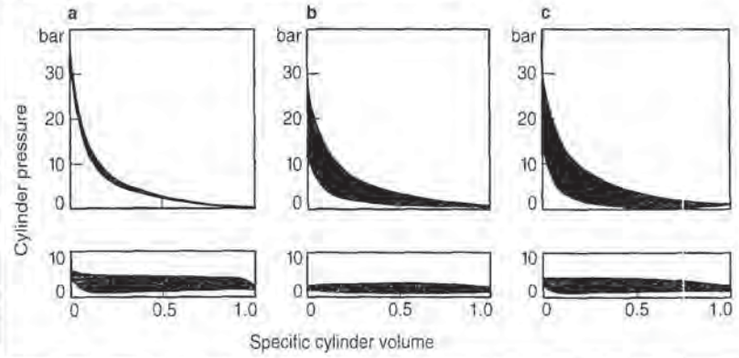
Through the use of a pressure-regulator valve in the bypass, in conjunction with an exhaust flap, the braking power can be increased in the engine's lower and medium speed ranges. At higher revs, the pressure-regulator valve prevents pressure increases beyond the limits which could lead to valve or valve-gear damage.

Exhaust brake system with constant throttle:

The conventional exhaust brake with exhaust flap utilizes only the energy available in the engine's gas-exchange process, i.e. during the 4th (exhaust) and 1st (intake) working strokes. Specific decompression during the 2nd and 3rd working strokes releases part of the compression energy. The "pressure-volume" diagrams below show the pressure curves in a cylinder for the "exhaust flap" and "constant throttle" exhaust braking systems as well as for the combination of both systems (Fig. N). The braking power when using the constant throttle system, as opposed to the flap system, is obtained principally in the engine's high-pressure cycle.

The installation of a small restriction valve in the bypass to the exhaust valve permits an increase in braking power. This valve is actuated by compressed air, in the same way as in the servo-cylinder of the exhaust-brake flap. During exhaust-brake operation the valve can remain open, thus providing a constant throttle cross-section (Fig. O).

Principle of operation of exhaust (engine) braking systems as illustrated by "pressure-volume" diagram (n_M = 1700 min^{-1}) *(Fig. N)*
a) Exhaust-brake flap closed, b) Constant throttle actuated,
c) Exhaust-brake flap closed and constant throttle actuated.



2. Retarders

Retarders are increasingly being used as wear-free additional retarding braking systems in trucks and buses. In this way, the demands of legislation are met, active vehicle safety is increased due to the reduction of the stress placed on the service-braking system, and vehicle economy is increased as a result of higher average speeds and the reduction of brake-lining wear.

Retarders can be fitted between the engine and the transmission (primary retarders) or between the transmission and the driven axle(s) (secondary retarders). The disadvantage of primary retarders lies in the unavoidable interruption of power transmission, and thus braking effect, which occurs during gear changing with manual transmissions. Primary retarders can be employed in conjunction with power-shift transmissions, and in the case of steep downhill gradients which are negotiated at low speeds, this provides for a slight advantage in comparison to the secondary-retarder system. (Fig. R).

Currently, two basic design concepts represent the state-of-the-art:

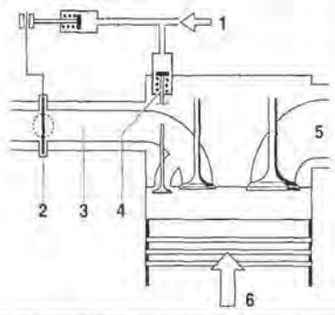
Hydrodynamic retarders: This retarder works in the same way as the Foettinger clutch (Fig. S). The rotor converts the mechanical energy supplied by the drive shaft into kinetic energy of a fluid. This kinetic energy is in turn converted into heat at the stator, which means that the fluid used must be cooled.

A hand lever or the brake pedal (in the case of an integrated retarder) transmits the driver's braking-power requirements. In conjunction with an electronic control circuitry, a defined control air pressure is set. By means of a corresponding quantity of air, this control air pressure then forces a quantity of oil into the retarder's working area between rotor and stator. The flow energy absorbed by the oil as a result of the vehicle speed and the associated rotor motion is braked by the stator's fixed blades. This in turn effects braking of the rotor and thus of the entire vehicle. Characteristics are:

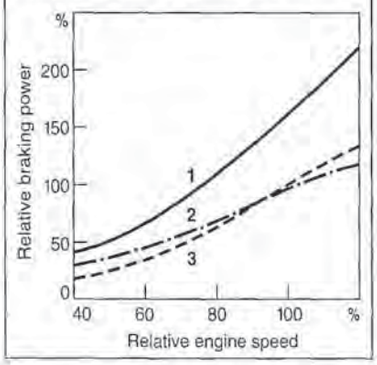
- Adequate cooling-circuit dimensioning is necessary in order to dissipate to the engine cooling circuit the heat generated by braking. An oil/water heat exchanger is used,
- Relatively complex design,
- Low weight of the hydrodynamic retarder which is integrated directly into the transmission,
- High specific braking powers,
- Very sensitive control of applied braking torque,
- Fan losses occurring when the retarder is not running must also be taken into consideration in retarder design.

In the hydrodynamic secondary retarder, an almost constant braking torque is available over a broad propshaft-

Exhaust brake with exhaust flap and constant throttle *(Mercedes-Benz) (Fig. O)*
1 Compressed air, 2 Exhaust flap, 3 Exhaust, 4 Constant throttle, 5 Intake, 6 Piston (2nd working stroke).



Braking-power curves *(V8 engine) (Fig. P)*
1 Exhaust flap and constant throttle, 2 Constant throttle, 3 Exhaust flap.

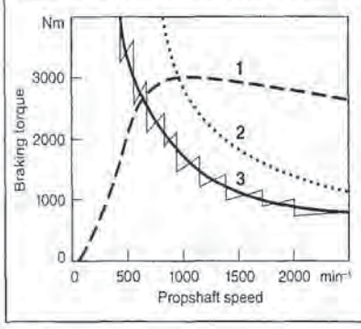


speed range (Fig. T). Below approx. 1,000 min^{-1} the braking torque drops steeply. As a result of this characteristic, conventionally designed hydrodynamic retarders are particularly suited to high-speed transport vehicles (overland transport).

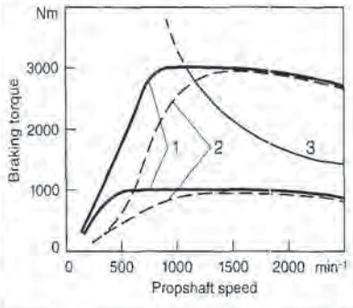
Modern retarder designs rectify the unfavorable braking torque characteristic of the secondary retarder design described above by providing high braking torques even at still low propshaft speeds. A spur-gear stage with a transmission ratio of approx. 1 : 2 drives such "boost retarders", which are fitted to the side of the transmission (Fig. S). Even in the propshaft lower speed range, a microprocessor control ensures acceptable braking-torque levels via a proportional valve.

Hydrodynamic retarders can only be used as continuous-operation braking systems in certain situations for limited

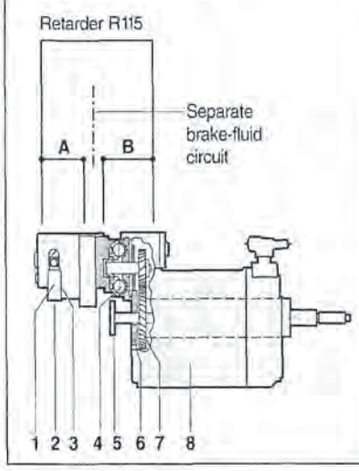
Braking-torque characteristics of primary and secondary retarders *(Fig. R)*
1 Secondary retarder, 2 Limit of cooling effect under continuous load (300 kW), 3 Primary retarder on input side of 16-speed gearbox.



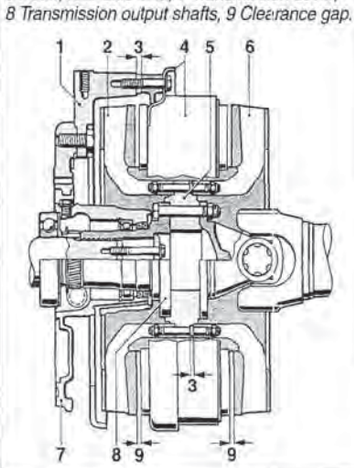
Hydrodynamic retarder performance characteristics *(Fig. T)*
1 Boost retarder, 2 Conventional retarder, 3 Cooling-power limit at continuous load (300 kW).



Boost retarder *(Fig. S)*
1 Transmission, 2 Spur gears (transmission ratio), 3 Gearbox flange, 4 Rotor, 5 Stator, 6 Compressed-air cylinder, 7 Proportional valve, 8 Wiring connector.



Electrodynamic retarder *(Fig. U)*
1 Star-shaped bracket, 2 Rotor, transmission side, 3 Spacers (for adjusting clearance), 4 Stator with coils, 5 Intermediate flange, 6 Rotor, rear-axle side, 7 Transmission cover, 8 Transmission output shafts, 9 Clearance gap.



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periods. The maximum cooling power of modern diesel engines is approx. 300 kW. Thus, as a result of the coupling of the engine and retarder cooling circuits, there exists a risk for both engine and retarder if additional safety measures are not implemented. For this reason, thermo switches are used to restrict the retarder's braking power so that thermal equilibrium is ensured.

Electrodynamic retarders: Currently common electrodynamic retarders have a mounting in the form of a stator, on which field coils are mounted (Fig. U). The rotors mounted on both sides of the drive shaft are ribbed for better heat dissipation. In order to brake the vehicle, voltage is applied to the field coils (from the battery or alternator) which thus generate a magnetic field which induces eddy currents in the rotors as they pass through the field. This generates a braking-torque level which is dependent on the stator excitation and on the air gap between rotor and stator.

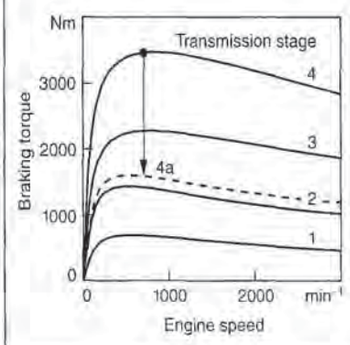
Characteristics are:
- Dissipation to the atmosphere of the heat produced,
- Relatively simple design,
- Relatively heavy construction,
- Trouble-free operation only with adequate current supply,
- Heating of the retarder leads to a reduction in braking torque,
- High braking powers even at low vehicle speeds,
- Braking power influenced by rotor blading, air-flow conditions around the eddy-current brake, and by ambient temperature.

In contrast to the conventional hydrodynamic secondary retarders, electrodynamic retarders provide relatively high braking torques at low driveshaft speeds (Fig. V).

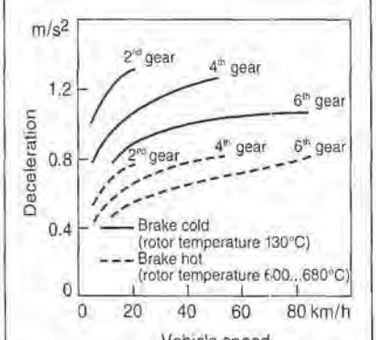
The significant reduction in braking torques of the electrodynamic retarder as the rotor temperature increases result from the thermal safeguard (Fig. W). Vehicle deceleration is reduced as thermal stress on the electrodynamic retarder increases.

In order to prevent temperature-related destruction of the retarder when the vehicle is being braked, a bi-metal switch restricts the current supply to half of the eight coils when the stator temperature reaches approx. 250 °C (Fig. W).

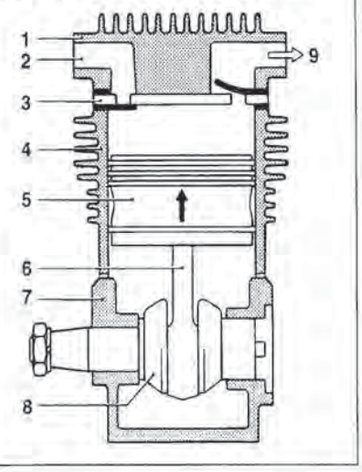
Braking-torque characteristic of an electrodynamic retarder *(Fig. V)*
4a Braking power when the cooling power limit has been reached (switching stage 4).



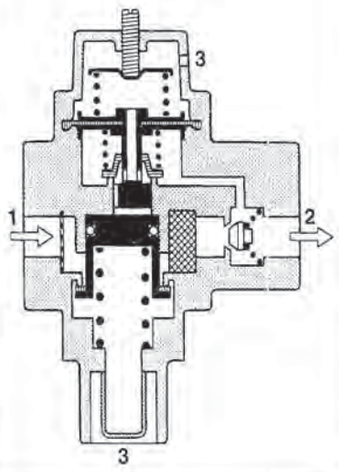
Influence of transmission ratio and rotor temperature on the performance of electrodynamic retarders *(17 t commercial vehicle, laden) (Fig. W).*



Air compressor (compression and discharge)
1 Cylinder head, 2 Air intake, 3 Intermediate plate (with intake and outlet valve), 4 Cylinder, 5 Piston, 6 Connecting rod, 7 Crankcase, 8 Crankshaft, 9 To compressed-air system.



Pressure regulator
1 From air compressor, 2 To air reservoirs, 3 Blow-off.



Components for compressed-air brakes

Energy-supplying device

The energy-supplying device comprises the following:
- Energy source
- Pressure regulator
- Air conditioning

Energy source: The energy source is a continuously running air compressor which is driven by the engine via V-belts or toothed gears. It consists of:
- Crankcase with crankshaft (driver for power-steering pump at the free end of the shaft), bearing assembly and connections for circulating engine lubrication;
- Cylinder with piston and connecting rod;
- Intermediate plate with intake and outlet valves;
- Cylinder cover with suction and pressure connections for air and, if appropriate, the fittings required for liquid-cooled versions.

Energy-economy controls are increasingly being used in order to reduce the losses during idle operation (opening and flow resistances in the valves, lines, and pressure regulator). These are integrated in the valve plate and actuated pneumatically via an actuator. At idle, either a bypass from the compressor's pressure chamber to the input side is opened, or the intake bore is opened by rotating/shifting the intake valve.

The air compressor is usually mounted on the vehicle engine by means of base or flange attachment. In some cases it is integrated in the engine block.

During its downward stroke, the piston draws in air after the intake valve has automatically opened as a result of the vacuum. The intake valve closes at the beginning of the return movement of the piston. The air is now compressed and, after reaching a set pressure, it is conveyed via the outlet valve, which also opens automatically, into the downstream compressed-air system.

In terms of delivery rate, the aim is a volumetric efficiency of 70 % and in terms of oil consumption a maximum of 0.5 g/h is desirable.

Pressure regulation: The pressure regulator ensures that the desired pressure level is maintained. Two principal types of regulation are used:

1. Regulation in which the pressure regu-

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lator has no influence on the energy source (in compressors with maximum speeds >2,500 min^{-1}).

The pressure regulator switches off when the desired maximum operating pressure is reached, and returns the air supplied by the compressor to atmosphere during the subsequent no-load period. If the pressure in the air reservoirs reaches the lower operating-pressure limit the pressure regulator switches on again and supplies the air delivered by the compressor to the air reservoirs.

2. Regulation in which the pressure regulator influences the energy source (in compressors with maximum speeds <2,500 min^{-1}).

When the desired maximum operating pressure is reached, the regulator applies pressure to a plunger in the compressor and opens its intake valve. Without being supplied to the air reservoirs, the intake air is expelled through the intake fitting. When the pressure in the air reservoirs reaches the lower operating-pressure limit, the regulator switches and the intake valve can again automatically open and close, and the air reservoirs are filled.

Pressure level: In towing vehicles, values between 7 and 12 bar (low pressure) and between 14 and 20 bar (high pressure) are used. In the two-line braking system, the pressure in the lines connecting towing vehicle and trailer is between 6 and 8 bar.

Compressed-air conditioning: The air must be conditioned in order to ensure proper operation of the downstream braking-system components. Impurities in the air can cause leaks in the control valves, and water in the compressed air leads to corrosion or icing-up during frost. To combat these problems, an air drier is connected downstream of the air compressor. Such a configuration makes it unnecessary to add antifreeze to the system.

Single-box air drier

Basically, an air drier consists of a desiccant box and a housing. The housing incorporates the air passage, a bleeder valve, and a control element for granulate regeneration. Normally, the granulate is regenerated by switching in a regeneration air reservoir with integral regenerating throttle.

Function: When the bleeder valve is closed, the compressed air from the air compressor flows through the desiccant box and from there to the supply-air reservoirs. When flowing through the desiccant box, water is removed from the moist compressed air by means of condensation and adsorption. At the same time a regeneration-air reservoir with a volume of approx. 4...6 liters is filled with dry compressed air.

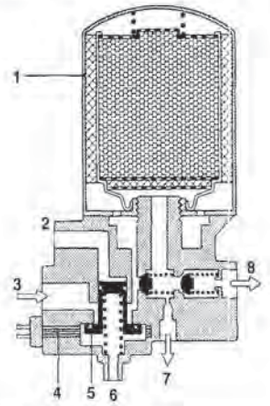
The granulate in the desiccant box has a limited water absorption capacity and must therefore be regenerated at regular intervals. In the reverse process, dry compressed air from the regeneration-air reservoir is reduced to atmospheric pressure via the regenerating throttle, flows back through the moist granulate from which it draws off the moisture, and flows as moist air via the opened bleeder valve into the atmosphere. On air driers with an integrated pressure regulator, its control element is fitted to port 4 of the air-drier valve body.

Control device

Normally, the control device comprises the brake pedal and all devices up to the point where influence is exerted on the control units.

Single-box air drier

1 Desiccant box, 2 From pressure regulator, 3 From air compressor, 4 Heating element, 5 Bleeder valve, 6 Exhaust, 7 To regeneration-air reservoir, 8 To supply-air reservoir.



Transmission device

The transmission device comprises the following:

- Circuit isolation (e.g., multiple-circuit protection valve),
- Energy storage (e.g., air reservoirs),
- Control units (e.g., brake valves),
- Load-sensitive braking-force metering device (e.g., automatic load-sensitive braking-force control), and
- Brake cylinders or servo cylinders.

These components work together as shown in the block diagram below of a power-braking system with dual-circuit service-braking system (see also the braking-system diagram on P. 676).

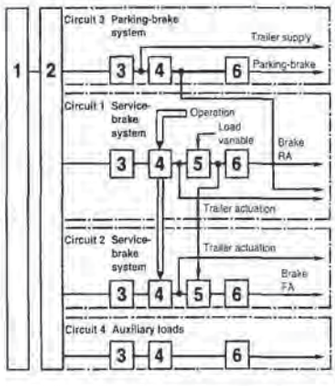
Functions and design of components:

Circuit isolation: Separation of the circuits with respect to one another in the event of damage in one circuit, as well as preservation of the operational integrity of the intact circuits.

Circuit isolation is primarily achieved by a combination of overflow valves grouped together as a unit; the operation of these valves is ensured at both low and high delivery rates.

Energy storage: Provision of the required volume of energy for all circuits in the braking system, including provision of energy in the event of failure of the energy source. Commercially available air reservoirs are used for this purpose; these have corresponding safety allowances for excess pressure and rust.

Transmission device in a power-braking system *(block diagram)*
1 Energy supply, 2 Circuit isolation, 3 Energy storage, 4 Control equipment, 5 Automatic load-sensitive braking-force metering, 6 Brake or servo cylinder, VA Front axle, HA Rear axle.



Control equipment

This equipment is used to control the metering of the required pressure to the corresponding parts of the system. Mechanically, hydraulically or pneumatically actuated or controlled reaction valves are used to control the pressure at the output of each valve as a function of the input variable. Due to the wide range of applications, a correspondingly large number of different components are in use. Dual-circuit control valves are also required for dual-circuit service-braking systems. Proper braking-system operation requires good control behavior, good pressure-metering capability, fast reaction times and low braking-system hysteresis.

Automatic load-sensitive braking-force metering (ALB)

Automatic braking-pressure control as a function of vehicle load. Load is often determined by spring compression (in the case of steel-spring suspension) and bellows pressure (in the case of pneumatic suspension). A control valve with a variable reaction-surface area reduces the output pressure in the valve in relation to the input pressure as a function of spring compression or bellows pressure.

Brake cylinders or servo cylinders

Convert the pressure applied to the braking system into braking force. Both plunger- and diaphragm-type cylinders are used. Diaphragm-type cylinders are primarily used for the service-braking system, while spring-brake actuators are used for the parking system. In the case of axles which are acted upon by both the service-braking system and the parking-brake system, combined single-chamber spring-brake actuators (so-called combination cylinders) are used in braking systems without hydraulic force transmission.

Wheel brakes (see P. 678)

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Service-brake valve

Two tandem-arranged control valves are actuated by a common device (brake pedal with transmission). The synchronized opening of both circuits is ensured by identical spring and valve sealing forces as well as by the mechanical overcoming of the opening forces in both control valves. In the braking position, the rocking piston between the control circuits is subjected at its two ends to the currently applied braking pressure and thus guarantees that the circuits are synchronized. The preloaded travel spring provides small response travels of the service-brake valve. The interaction of the force of the reaction piston with the travel spring enables the system to execute the necessary control travels independently. Dual-circuit sealing of the rocking piston ensures the required safety.

Parking-brake valve

Today's compact design of the parking-brake valve is a direct result of the lack of installation space at the instrument panel (where it used to be fitted). The parking-brake valve always actuates the brake cylinders via relay valves.

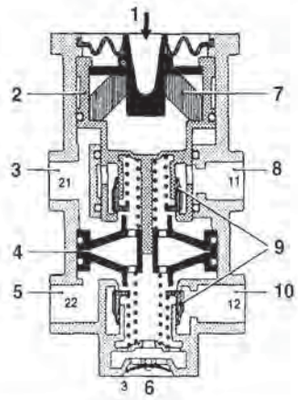
A hand lever (actuating lever) adjusts an internal valve seat by means of an eccentric and a linking strap. In doing so, it controls a double-seat valve in which compressed air from above and the force of compression springs from below act on a valve piston. In the brake position the actuating lever latches into position automatically, and the space above the valve piston is purged of air. As many intermediate settings between the drive and brake settings as desired are possible.

If the actuating lever is moved beyond the brake position, the auxiliary (test) valve is actuated and compressed air flows from the supply-air reservoir into the connecting line to the trailer-control valve. This leads to the towing-vehicle braking effect being retained, although it is cancelled for the trailer.

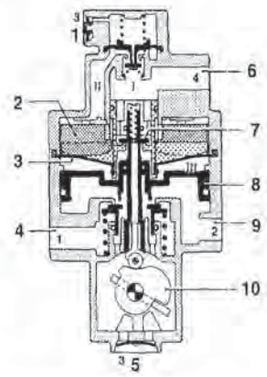
Automatic load-sensitive braking-force regulator

This device is connected between the service-brake valve and the brake cylinders. Depending on the vehicle load, it regulates the applied braking pressure. The device has a reaction diaphragm with variable active area. The diaphragm is held in two radially arranged and interlocking rakes. Depending on the position of the control-valve seat in the vertical direction, there is a large reaction area

Service-brake valve
1 Actuation, 2 Reaction piston, 3 Brake circuit 1, 4 Rocking piston, 5 Brake circuit 2, 6 Exhaust, 7 Travel spring, 8 Supply circuit 1, 9 Control valves, 10 Supply circuit 2.



Automatic load-sensitive braking force regulator
1 Exhaust, 2 Rake, 3 Reaction diaphragm, 4 From air reservoir, 5 Exhaust, 6 From service-brake valve, 7 Control valve, 8 Relay piston, 9 To brake cylinders, 10 Rotary cam.



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(valve position at bottom) or a small reaction area (valve position at top). Consequently, the brake cylinders are supplied via an integrated relay valve with a pressure which is lower than (unladen), or which is the same as (fully laden), that coming from the service-brake valve. The regulator is mounted on the vehicle frame and senses the compression position of the axle by a rotary lever via linkages. The rotary cam moves the valve rod accordingly in the vertical direction and thus determines the valve position. The pressure limiter which is integrated into the device at the top allows a small partial pressure to flow in on the top side of the diaphragm. Thus, up to this pressure there is no reduction in the brake-cylinder pressure. This results in the synchronous application of the brakes on all vehicle axles. If the rotary lever breaks, the applied pressure flows to the brake cylinders at a ratio of 2:1.

Combination brake cylinder for wedge brakes

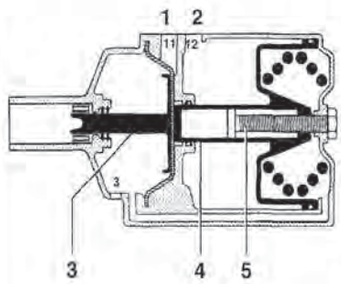
The combination brake cylinder consists of a single-chamber diaphragm cylinder for the service brakes and a spring-brake actuator for the parking brakes. The cylinder and spring-brake actuator are in tandem and act on a common pressure rod. They can be actuated independently of each other. Simultaneous actuation results in the addition of their forces. The central release screw permits tensioning the spring of the spring-brake actuator without compressed air having to be applied. This is the setting when installed in the vehicle. Following installation, the release screw is screwed into the spring-actuator cylinder, and the spring acts via the piston rod on the wedge mechanism. The entry of compressed air into the spring-actuator cylinder (when parking brake is released) forces the piston back against the action of the spring, loads the spring and releases the brake (position illustrated). When the service brakes are operated, compressed air flows into the diaphragm cylinder, actuating the wedge mechanism by way of the diaphragm and the thrust rod. A drop in air pressure releases the brake.

The principle described above is modified for cam-operated brakes and for disc brakes.

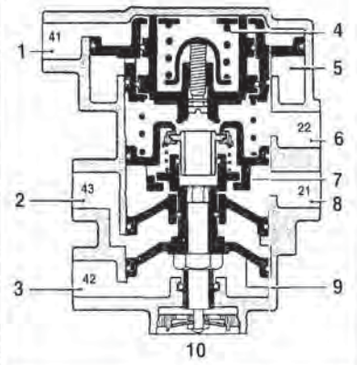
Trailer-control valve

In two-line braking systems, the trailer-control valve which is installed in the towing vehicle controls the trailer's service

Combination brake cylinder for wedge brakes
1 Single-chamber cylinder control line, 2 Spring-brake actuator control line, 3 Pressure rod, 4 Piston rod, 5 Release screw.



Trailer-control valve
1 Service-brake circuit 1, 2 Parking-brake circuit, 3 Service-brake circuit 2, 4 Pilot spring, 5 Control piston 1, 6 Control line to trailer, 7 Control-piston unit, 8 Supply line to trailer, 9 Control piston 2, 10 Exhaust.



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brakes. This multi-circuit relay valve is triggered by both service-brake circuits and by the parking-brake system.

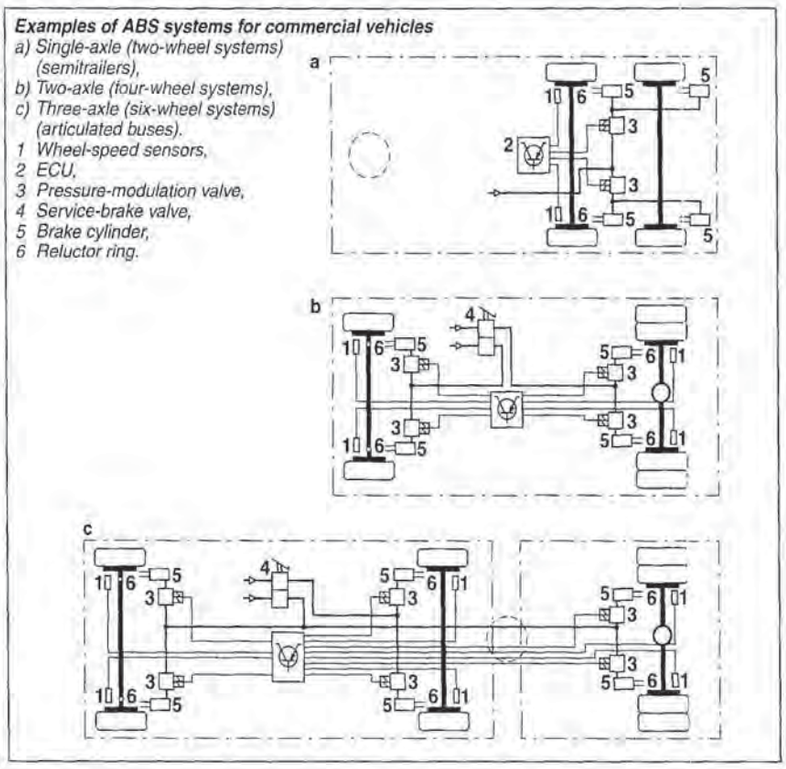
In the driving position, the supply chamber as well as the chamber of the parking-brake circuit are each subjected to equal pressure, and the trailer-control line is exhausted through the central exhaust port. A rise in pressure upstream of the control piston of brake circuit 1 (top) and/or of brake circuit 2 (bottom) leads to a corresponding rise in pressure in the trailer-control line. Brake circuit 1 is equipped with a larger control piston than brake circuit 2, which means that it has priority over the control piston of circuit 2. This priority ends when the pilot pressure is reached (pilot-spring force is exceeded). A reduction in pressure in the service-brake circuits leads to the same reduction in the trailer-control line. The exhausting of the parking-brake circuit (braking) increases the pressure in the chamber to the trailer-control line. The application of air to the parking-brake circuit (releasing) again exhausts the trailer-control line.

Mufflers

Current and future legislation stipulates the reduction of compressed-air valve exhaust noise. Absorption-principle mufflers are used whose size depends on the exhaust-air quantity, air pressure, exhaust duration, and required noise level.

The muffler is a cylinder with radial and axial slits, and is filled with insulating material. Noise damping is due to inflow geometry, the insulation cartridge, and the exhaust slits with spherical exhaust flow. Mufflers are attached to the compressed-air valves via threaded, snap-action, bayonet, or pipe connections. Some versions fulfill the "Low-Noise Vehicle" conditions [72 dB(A)].

Examples of ABS systems for commercial vehicles
a) Single-axle (two-wheel systems) (semitrailers),
b) Two-axle (four-wheel systems),
c) Three-axle (six-wheel systems) (articulated buses).
1 Wheel-speed sensors,
2 ECU,
3 Pressure-modulation valve,
4 Service-brake valve,
5 Brake cylinder,
6 Reluctor ring.



Antilock braking systems (ABS) for commercial vehicles

The ABS prevents the wheels from locking when the vehicle is overbraked. The vehicle therefore retains its directional stability and steerability even under emergency braking on a slippery road surface. The stopping distance is often shorter than with locked wheels. ABS prevents the danger of jackknifing in the case of vehicle combinations.

In contrast to passenger cars, commercial vehicles have pneumatic braking systems. Nevertheless, the functional description of the ABS control circuits for passenger cars (see P. 659) also applies in principle to commercial vehicles.

The antilock braking system as used in commercial vehicles consists of wheel-speed sensors, an electronic controller (ECU) and pressure-modulation valves. The ABS regulates the brake pressure in each brake cylinder by increasing the pressure, holding it constant or reducing it by exhausting to atmosphere.

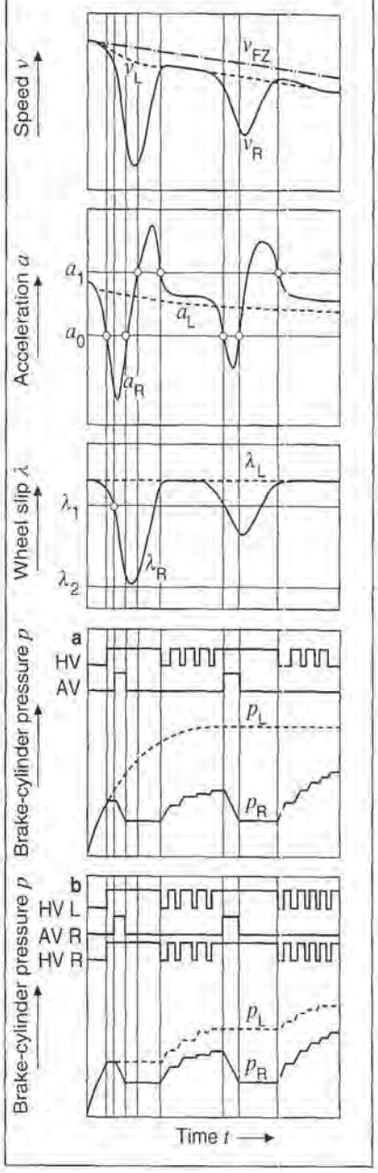
Individual control (IR)

This process, which sets and controls the optimum brake pressure individually for each wheel, produces the shortest braking distances. Under μ-split conditions (different friction coefficients between right-hand and left-hand wheels, e.g., black ice at edge of road surface, good grip at center of road surface) braking produces a high yawing moment about the vertical axis of the vehicle, thus making short-wheelbase vehicles difficult to control. In addition, this is coupled with a high steering moment as a result of the positive steering roll radii in commercial vehicles.

Select-low control (SL)

This process reduces the yawing and steering moments to zero. This is achieved by applying the same brake pressure to both wheels on the same axle. This requires only one pressure-control valve for each axle. The pressure applied is chosen to suit the wheel with less grip (select low), the wheel with more grip is braked less heavily compared to the individual control (IR) method. Under

ABS control processes
Example: Braking on μ-split.
a) IR individual control (rear axle), b) IRM individual control modified (steering axle).
HV = Holding valve, AV = Outlet valve.
Subscripts: FZ vehicle, R Right wheel, L Left wheel, 0, 1, 2 thresholds.



μ-split conditions, braking distances are longer but steering control/directional stability of the vehicle is better. If grip conditions (coefficients of friction) are the same on both sides, braking distances, steering control and directional stability are virtually identical with systems which control each wheel individually (IR).

In the case of similar control principles with higher pressure levels than that in the SL method, the wheel running with the lower friction coefficient is forced to run with higher slip until it eventually locks up. Under μ-split conditions, braking distances are shorter compared with pure SL control systems. Steering control and directional stability of the vehicle may be very slightly worse. Normally, due to the lower frictional coefficient, there is no tire damage at the wheel which locks up.

Individual control modified (IRM)
This process necessitates a pressure-modulation valve at each wheel of the axle. It reduces yawing and steering moments only as far as necessary, and limits brake-pressure difference between left and right sides to a permissible level. This leads to the wheel which is running at the high friction coefficient being braked less strongly. This compromise results in a braking distance which is only a little longer than that for individual control (IR), but it does ensure the safe control of vehicles which tend to handle critically.

Commercial-vehicle ABS versions
Since 1 October 1991, within the area covered by the EU member states, ABS has been prescribed by law for the initial registration of commercial vehicles intended for trailer operation, and of semitrailer tractors (> 16t), as well as trailers (> 10t), and busses (> 12t). This regulation has been extended so that since 1/10/1998 all busses and since 1/10/1999 all trucks and trailers (> 3.5t) have had to be fitted with ABS.

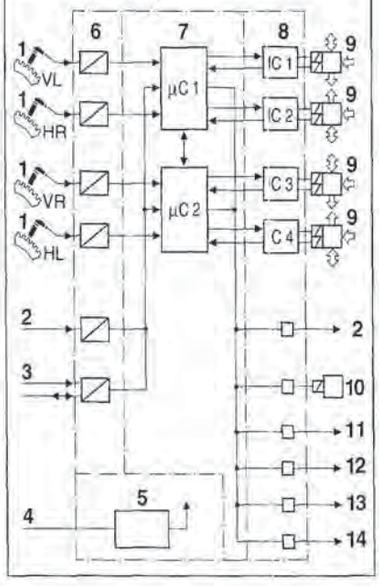
The law stipulates 3 categories of ABS which differ from each other with respect to their specifications regarding retardation and the wheel and vehicle behavior on μ-split road surfaces. Most European vehicle manufacturers only use ABS systems in category 1. Only those have to conform to all specifications of Directive 71/320/EEC.

All ABS installations must be equipped with a warning lamp which must light up for at least 2 secs after switching on the driving switch. This lamp is to be referred to by the driver as a form of visual inspection. If it lights up while the vehicle is being driven, this indicates that the installation's continuous self-testing facility has detected a fault. This can result in the ABS being switched off completely.

Towing vehicles and trailers with ABS systems from different manufacturers may be combined as desired provided there is an ABS plug and socket connection between the two vehicles in accordance with ISO 7638.

On all vehicle combinations (semitrailer tractors, truck tractor/trailer combinations), optimum control of braking at the physical limits is only guaranteed when both tractor vehicle and trailer are equipped with ABS. However, even with-

ABS/TCS control unit (ECU)
1 Wheel-speed sensor, 2 TCS engine-management interface, 3 Self-diagnosis, 4 Vehicle power supply, 5 Power supply, safety unit, 6 Input stages, 7 Microcomputers 1 and 2, 8 Output stages, 9 Pressure-control valve, 10 TCS solenoid valve, 11 Warning lamp, 12 TCS Indicator lamp, 13 Retarder relay, 14 Valve relay.



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out ABS throughout (only on tractor unit or only on trailer) there are significant improvements compared to a vehicle combination which has no ABS at all.

In order to be able to provide the optimum and cost-effective equipment for the very wide range of commercial vehicles on today's roads, single-axle, 2-axle, and 3-axle ABS systems are available in accordance with the number of axles which are to be controlled. The single-axle systems as used primarily on semitrailers, individually control the wheels on a given axle. The remaining axles are supplied with the same brake pressures as the controlled axle.

2-axle systems are mainly fitted to two-axle buses, trucks, and trailers. Three-axle vehicles can also be equipped with a 2-axle ABS system insofar as two of the axles are in close vicinity to each other and can be provided with the same brake pressure (as for the single-axle ABS systems). The 3-axle systems are mainly intended for installation in long-wheelbase vehicles (e.g. articulated buses). In the case of 2-axle and 3-axle ABS installations, the steered axle uses either the IRM control method or, if this axle is only provided with a single pressure-control valve, the SL control method. Individual control (IR) is always used on commercial-vehicle rear axles.

The range of available ECU's permits further control combinations (not described here in detail). Example: If both axles of a semitrailer have wheel-speed sensors, but each side of the vehicle is only equipped with a single pressure-modulation valve, the wheels of one side of the vehicle are SL-controlled. In this case, one axle may be a lifting axle, which is automatically excluded from the control process when lifted.

All ABS systems can be equipped with single-channel pressure-modulation valves. ABS trailer systems can also be fitted with relay-type pressure-modulation valves (see Fig.).

In light commercial vehicles with pneumatic/hydraulic converters, ABS intervenes in the pneumatic brake circuit via single-channel pressure-modulation valves and defines the hydraulic brake pressure. In other versions, an ABS pressure modulator with integral solenoid valves is connected in parallel to the pneumatic/hydraulic converter. The modulators are controlled by the same ECUs as the single-channel pressure-modulation valves.

When the vehicle is running on a low-friction-coefficient road surface, the operation of an additional retarding brake (engine brake or retarder) can lead to excessive slip at the driven wheels. This would impair vehicle stability. ABS therefore monitors the brake slip and controls it to permissible levels by switching the additional retarding brake on and off.

ABS components

Wheel-speed sensor

For most applications nowadays, an application-engineered inductive sensor attached to the axle tube is used (for description of function, refer to P. 103). In contrast with car applications (rigidly attached sensors), on commercial vehicles the sensor is held in a spring sleeve. When the vehicle is in motion, the action of the wheel-bearing play and the flexing of the axle moves the sensor along its axis, thereby automatically adjusting the clearance between the sensor and the reluctor ring. If, under exceptional circumstances, the clearance gap becomes excessive, the ECU switches off the control for the wheel in question.

In the future, inductive sensors integrated in the wheel bearings and semiconductor speed sensors will be used more and more frequently due to economic considerations.

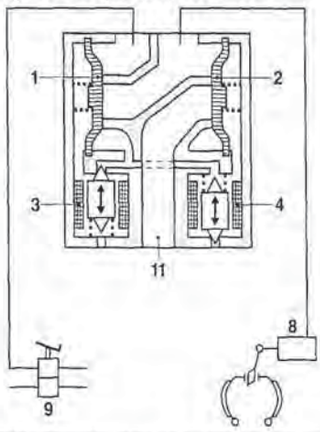
Electronic control unit (ECU)

The ECU's input stages convert the incoming sinusoidal signals from the wheel-speed sensors into square-wave signals. The wheel speeds are calculated from the frequency of the square-wave signals by (redundant) microcomputers. These speeds are used to estimate a vehicle reference speed. The brake slip for each wheel is calculated using this reference speed and the individual wheel speed. If a wheel has a tendency to lock, this is determined from the "wheel-acceleration" and "wheel-slip signals". In such a case,

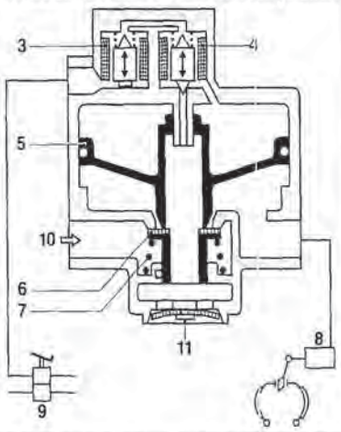
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Pressure-modulation valve (schematic)
1 Holding valve, 2 Outlet valve, 3 Solenoid valve for "Maintain pressure", 4 Solenoid valve for "Reduce pressure", 5 Control piston, 6 Valve plate, 7 Compression spring, 8 Brake cylinder, 9 Service-brake valve, 10 Supply air, 11 Exhaust.

Single-channel pressure-modulation valve



Pressure-modulation valve with relay action



the microcomputer energizes (via the ECU output stages) the solenoids of the pressure-modulation valves which control the brake pressure in the individual wheel-brake cylinders.

The ECU contains a comprehensive program for the detection of faults throughout the entire antilock system (wheel-speed sensors, ECU, pressure-modulation valves, wiring harness). If a fault is detected, the ECU switches off the defective part of the system and stores a code detailing the faulty signal path. That code can be retrieved by a service technician with the aid of the warning lamp (flashing signal) or using an intelligent testing device (e.g. personal computer) via a standardized serial interface.

The ECU's of some European ABS manufacturers include not only the ABS function but also functions for TCS traction control and in some cases for cruise control (see also page 611). The most important factor is that depending upon the model, the ECU automatically configures itself to the required function. In other words, if the vehicle concerned is only ABS-equipped, the ECU only carries out the ABS function; if the vehicle has TCS components the ECU also controls wheel slip automatically.

Pressure-modulation valve

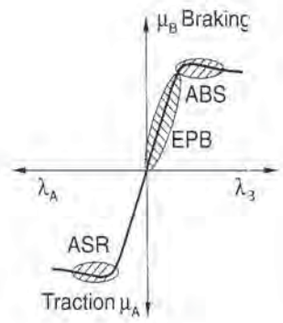
Single-channel pressure-modulation valves are available with and without relay action. The relay-action valves are installed in semitrailers and drawbar trailers. The standard trailer-braking system often includes relay valves which can then be replaced by ABS relay-action valves. Non-relay-action ABS valves are used in all other vehicles, i.e. in buses, trucks and tractor-trailer rigs, as well as in trailers and special vehicles. Both types of valve have 3/2 pilot valves. The non-relay-action valves thereby control 2/2 diaphragm valves, which have a sufficiently large cross-section for almost all applications. In the case of the relay-action valves, the 3/2 solenoid valves affect the pressure in the pilot chamber of a relay valve. The electronics control the solenoid valves in the appropriate combination so that the required function is performed (pressure retention or pressure release). If no pilot-valve actuation takes place, "pressure build-up" is the result.

When braking normally (that is, without ABS response = no locking tendency of a wheel), the air flows through the pressure-modulation valves unhindered in both directions when pressure is applied to or removed from the brake cylinders. This ensures fault-free functioning of the service-braking system.

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Electronically controlled braking system (ELB) for commercial vehicles

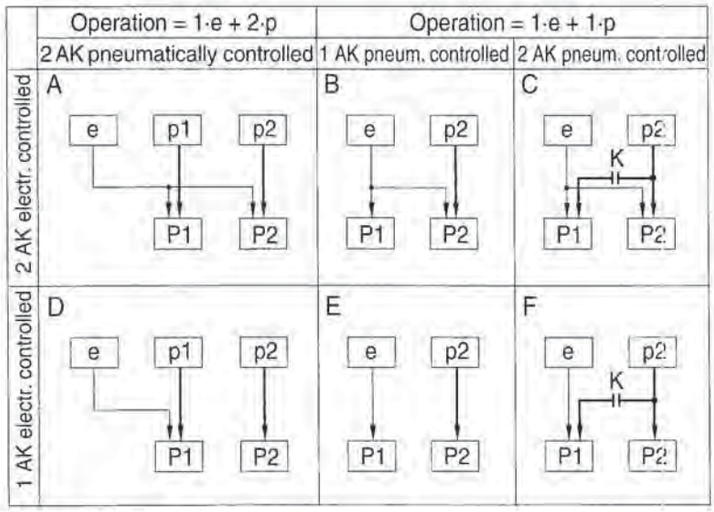
Adhesion-slip diagram, scope of ELB
λ Slip, μ Adhesion coefficient.
Indices: A Traction, B Brake.



Assignment

The aim of the electronically controlled braking system (ELB) for commercial vehicles is to optimize the braking process and, in certain cases, the power transmission process. The ELB system also supports the driver and the service technician during monitoring and servicing of the braking system, as well as contributing to an increase in economic efficiency. ELB comprises the areas of ABS (Antilock Braking System), EPB (Electropneumatic Brake), and TCS (Traction Control System). Within an electronic vehicle-management system, the ELB utilizes the information available from the data bus network to produce easy-to-control and

Structure of the operating and working circuits
e/p Electrical/pneumatic operation, P Pneumatic working circuit (AK),
1, 2 Compressed air from reservoir 1, 2,
K Valve for circuit isolation (decoupling),
A...F variants.



directionally stable braking and power transmission, and to provide information for other systems. Compared with conventional braking systems, an ELB system can be simpler in design. In the ELB, complex functions are processed electronically.

Function blocks

An ELB encompasses at least the following function groups:
- Electropneumatic braking system (EPB),
- Antilock braking system (ABS)
- Traction control system (TCS).

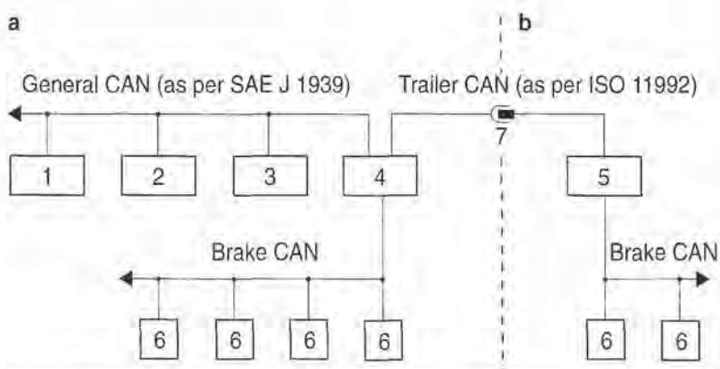
ABS and TCS are already in use on commercial vehicles (see PP. 693 and 610) and in the ELB are up-graded by the availability of additional information such as brake pressures and axle loads. Under stable adhesion/slip range (see Adhesion-Slip Curve) the action of the brakes is controlled electronically by the EPB. In this process, the effect of pressure control on the wheels, axles and the tractor and trailer units over the EPB range as a whole, is designed to comply with a variety of different targets. Thus, for light braking operations, the control-system objective may be even brake-lining wear, for medium to heavy braking it may be precise balancing of braking forces, and for emergency braking the aim may be maximum possible braking effect.

Design

Compressed-air supply, circuit isolation, energy storage and brake cylinders are the same as in conventional braking systems. The additional components include: pressure-control modules in the vicinity of the wheels for adjusting the brake pressure in the wheel cylinders as required, and an ECU. The system may also include a trailer control module the main function of which is to brake a conventional trailer.

Today, ELB systems with pneumatic retention circuits are available in the form of back-up systems at the safety level. Tomorrow's vehicle generations will be able to provide adequate safety while featuring electrically controlled multiple circuits without retention circuits. Normally, the pneumatic working circuits for operation of the brake cylinder in the tractor vehicle are of the dual-circuit design (II braking-force distribution, see P. 654) and in the trailer they are single circuit. The Figure

CAN networking of the brake components and their networking to other systems in the vehicle
a) Tractor vehicle, b) Trailer.
1, 2, 3 CAN stations (e.g. information, chassis, drivetrain),
4, 5 CAN stations (ELB control unit),
6 CAN stations (ELB pressure-control module),
7 Plug connection ISO 7638 (7-pole).



"Structure of the operating and working circuits" shows combinations of electrical and pneumatic control of two working circuits. For example, electrical operation of all axles is possible with Variant A which also features separate pneumatic retention circuits. Trailers are controlled via a standardized CAN data bus (ISO 11 992) (see P. 884). The pneumatic control circuit is retained in order to cope with tractor/trailer rigs using conventional vehicles and the stipulated control for all braking circuits. However, this function too can be performed by an electronically controlled trailer control module.

Apart from the CAN data bus connecting it to the trailer, an ELB has at least two more: One bus links it to the ECUs for the drivetrain (e.g. EDC and/or retarder ECUs, transmission ECU, etc.) and one internal bus to the system for communicating with the pressure-control modules and the trailer control module (see diagram "CAN Networking"). The ELB is thus able to exchange information with other systems in order to optimize vehicle management functions.

Functional sequence EPB

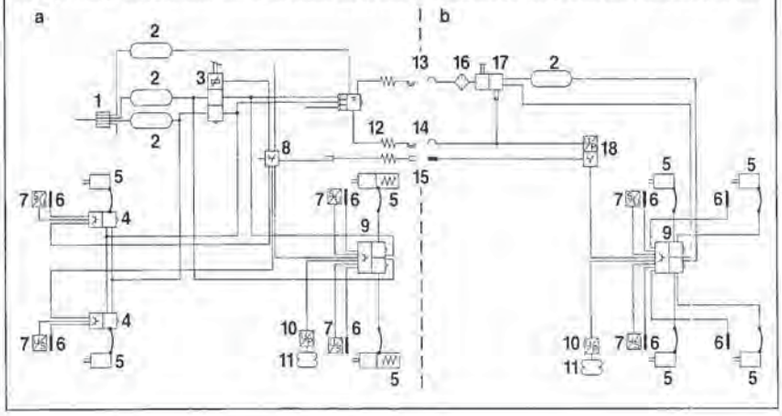
A single-circuit ELB with 2-circuit retention circuit (1e + 2p) is shown in the Fig. "Service-brake installation of an ELB". In addition to a conventional pneumatic trailer control, this ELB also features an electrical control.

After the driving switch is turned, the ELB is initialized and subjected to a self-test. If no faults are detected on the ELB, the warning lamps go out and electrically controlled braking can now take place. Redundant principles are applied in the registration of the brake-pedal position. This is used by the ELB control unit to calculate the optimum pressure level for each control channel. The data is transmitted to the pressure-control modules (located near to the wheels) via the braking-system CAN. At the same time, the brake values are sent to the trailer via CAN. The pressure-control modules (DRM) on the tractor unit and trailer automatically adjust the brake pressure to the required level. Sensor signals from the wheels for variables such as wheel speed, brake lining thickness, etc. are sent by the pressure control modules via the CAN to the ELB control unit. The system's rapid reaction with simultaneous pressure regulation on all axles produces a direct braking feel similar to that in a passenger car.

Service-brake installation of an ELB

a) Tractor vehicle, b) Trailer.

1 Four-circuit protection valve, 2 Air reservoir, 3 Service-brake valve with braking-value sensor (1e+2p), 4 Single-channel pressure-control module (DRM), 5 Wheel-brake cylinder, 6 Wheel-speed sensor, 7 Lining-wear sensor, 8 ELB control unit in the tractor vehicle, 9 Two-channel pressure-control module (2K-DRM), 10 Pressure sensor, 11 Air-suspension bellows, 12 Trailer control valve, 13 Supply coupling head (glad-hand), 14 Brake coupling head (glad-hand), 15 Plug-in connection ISO 7638 (7-pole), 16 Line filter, 17 Trailer brake valve with release device, 18 ELB control unit in the trailer.



Trailers equipped with ELB which are towed by tractor vehicles having a 5-pole ABS plug-in connection generate their electrical brake value from the pressure in the pneumatic trailer brake line.

When the driving switch is at "Off", or if pressure control is deactivated in the event of a fault, the solenoid valves in the pressure-control modules revert to the mechanically held normal position. In that position, the pneumatic retention circuits from the service brake valve apply the brake pressure to the wheel cylinders unmodified (back up braking).

Extended functions

ELB incorporates EPB, ABS and TCS as standard functions. Other functions for automatic brake balancing and functions which require interaction with other systems are already available fur use on production models or are in the course of development.

A few examples are given below:

Brake management

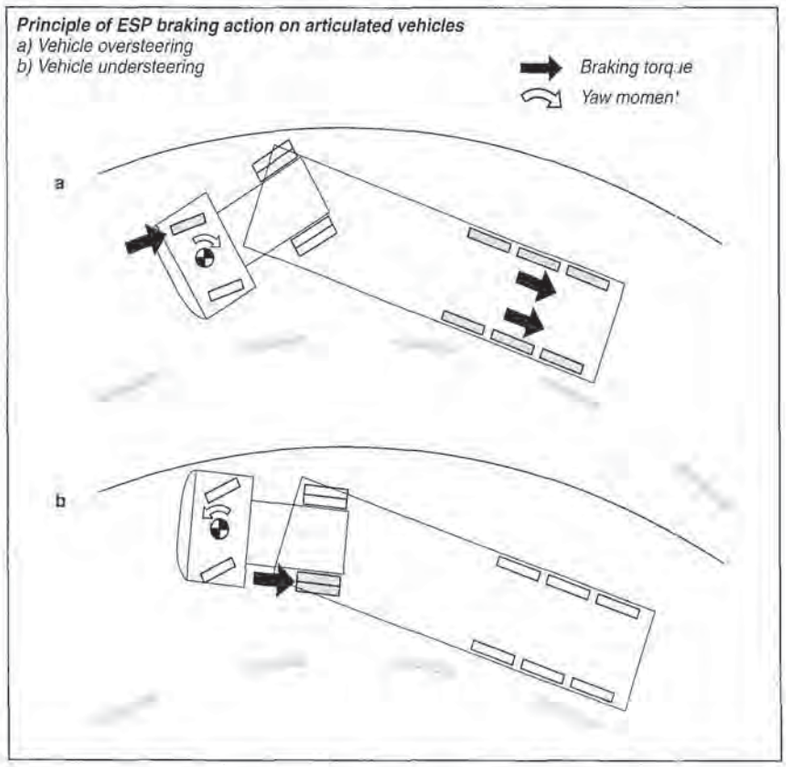
The braking effect desired by the driver can be achieved by means of either the retarder and/or the engine brake and/or the service brakes. The ELB control unit can perform the task of allocating the braking function to the appropriate braking system or systems.

Braking regulation

Vehicle deceleration is performed according to the driver's command, i.e. the position of the footbrake valve, regardless of whether the vehicle is loaded or empty. An automatic brake monitoring system prevents overload.

Brake-lining wear control

The brake-lining thickness is detected by sensors. Under normal braking with low levels of deceleration, the brake-lining



wear between the wheels on the same axle is equalized by slight adjustment of the brake balance.

Brake balancing between tractor unit and trailer

The aim of automatic balancing of the braking force of the tractor unit and the trailer is to generate optimum trailer-coupling forces between the two units when braking. For articulated vehicles, the optimum trailer-coupling force is equal to the coupling load; for truck-and-trailer combinations it is zero. Each unit is then braked individually according to its own mass. Assuming appropriate braking-system design and choice of brake linings, this then generally leads to equalization of temperatures and therefore of wear between the brake linings on the tractor unit and the trailer.

Brake monitoring

The energy generated in a brake is dependent on the vehicle's speed and the braking torque, while the temperature of the brakes is also dependent on the dissipation of the heat produced by friction. Using those variables, the ELB can estimate the temperature of the brakes and then take action to prevent overloading.

Inappropriate reduction of braking torque on one or more wheels can be detected by ECU analysis of the vehicle behaviour under braking.

Rollback limiter/"Bus-stop brake"

The use of information that is normally available from other electronic systems via the SAE J1939 CAN enables automatic application of the brakes shortly after the vehicle starts to roll with the drive disengaged, e.g. when stopping at traffic lights. In the same way, the brake is automatically released when the vehicle starts to pull away. On public transport busses, this function can replace the so-called "bus-stop brake".

Electronic stability program (ESP)

ESP is an extension of the ABS system and controls not only longitudinal dynamics (like ABS) but also actively responds to the vehicle's transverse dynamics at the limits of adhesion (e.g. in the event of a skid). This requires a yaw-rate sensor and a lateral-acceleration sensor for determining the actual vehicle behaviour, as well as a steering-angle sensor for detecting the driver's intentions. If there are inconsistencies between the driver's intentions and the vehicle's behaviour, e.g. if the tractor unit starts to understeer or oversteer, the ESP modifies the vehicle behaviour by automatic braking of individual wheels to reflect the intentions of the driver. The physical processes involved are explained in the description of ESP for cars (see P. 704). The same basic principles apply to commercial-vehicle ESP systems with the exception of the braking strategy. This is modified to suit commercial vehicles in their various forms. For example, as well as braking of individual wheels on the tractor unit, correction of oversteer can be assisted by braking the trailer to tension the tractor-and-trailer combination (see diagram "Principle of ESP braking action on articulated vehicles"). The trailers do not normally have sensor systems for detecting handling dynamics and are therefore not involved to any greater degree in the control of vehicle directional stability.

It is clear that the risk of overturning is considerably greater with commercial vehicles than with cars. For that reason, the commercial-vehicle ESP also incorporates functions for reducing the likelihood of vehicle overturn. This applies to ELB systems both on the tractor unit and on the trailer. From the available signals, critical vehicle conditions are detected and the vehicle or vehicle combination automatically braked as required by the particular situation. The behaviour of trailers without ELB is estimated by the tractor unit's ECU. It then instigates appropriate braking operations as required to avert hazardous situations.

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Brake test stands

The brakes are of decisive importance in maintaining high technical standards of vehicle safety; it is therefore imperative that the brake system be inspected on a regular basis. The German motor vehicle inspection prescribed by Paragraph 29 of the StVZO Road Licensing Regulations (FMVSS/CUR) employs brake test stands (roller dynamometers) for this purpose, as do automotive service centers for inspections and repairs. The braking forces monitored at the wheel's circumference provide the basis for evaluating the operation and effectiveness of the brake system. Brake stands employed for the inspections stipulated in Paragraph 29 must conform to the "Regulations governing use, design and testing of brake test stands" as administered by the Federal Transport Minister.

Design

The brake test stand's main components are two roller sets, for the left and right sides of the vehicle respectively. The vehicle is driven onto the test stand so that the wheels of the axle being tested rest upon the rollers.

A stable frame supports the roller sets, which assume the form of a drive roller and a secondary roller in a parallel layout, while a drive chain provides the positive dynamic connection between the two rollers. An AC motor powers the drive roller through a gear set with an upward conversion ratio, while the drive unit itself is suspended on an extension of the drive roller's shaft. Pressure exerted against a torque lever flanged onto the gear-drive unit is transferred through the load sensor, with the frame providing positive support for the entire assembly. The braking force F_{Br} is actually measured by monitoring the reaction torque M_R. The electric motors set the rollers in motion and then maintain an almost constant rotational speed against the opposing forces that occur when the vehicle's brakes are applied. The suspended drive unit with torque lever transmits the braking forces to the load-sensing device. The load-sensing device is designed as a flexural sensor with wire strain gauge.

The system's computer employs digital technology in the subsequent evaluation of the various data derived from braking-force testing, such as fluctuations or differences in braking force. Once processed, the information is presented in either analog or digital form, depending upon the specific system, while a printer can be connected to provide a hardcopy test protocol.

Operation

The drive motors for the roller sets may be activated in one of two ways, either by remote control or with an integral automatic on/off switch. An automatic-activation test stand is distinguished by the pressure-sensitive rollers located between the main test rollers on each roller set. When the vehicle is driven onto the test stand, it pushes down the pressure-sensitive rollers and activates the stand. When the vehicle leaves the stand the pressure-sensitive rollers are released and the unit is switched off. Should the applied braking force start to exceed the available traction between tires and test rollers, the wheel will respond by starting to slip and will then lock. Tire slip, however, makes it impossible to perform useful measurements of braking force. Under these conditions, it is the slip resistance between tire and roller (as a function of wheel load) that is measured and this is of no use. Here, assistance is provided by an automatic override device which recognizes this kind of slip by monitoring the test-roller speed, and which responds to it by switching off the test unit when a given maximum figure is exceeded. This avoids both false measurements and possible tire damage. The display, meanwhile, continues to show the maximum braking force achieved before the override device was activated. An electronic memory circuit ensures that the final display remains in place long enough to be recorded by the operator.

In addition, the vehicle and/or axle weight can also be measured at the test stand or entered remotely; the test unit can then use this information to calculate the effective retardation.

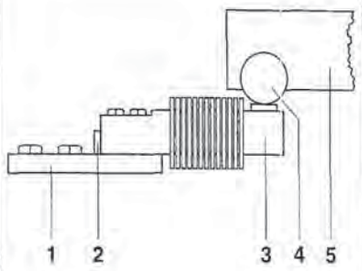
The brake test stand employs specific test sequences to provide extremely rationalized examination procedures, allowing

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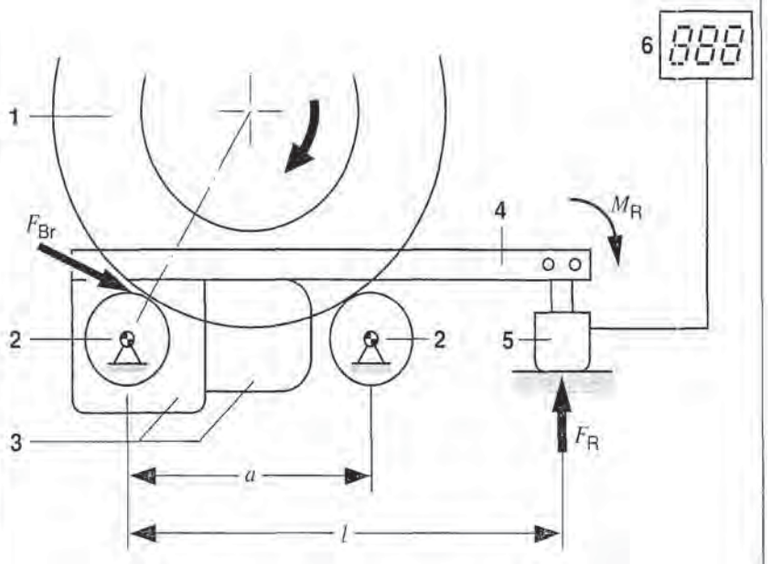
the operator to carry out complete testing of both front and rear brakes without leaving the vehicle.

Vehicles with permanent 4-wheel drive and variable torque distribution are tested on special stands, which are constructed to prevent the forces generated at the test axle from being transferred to that axle which is at rest.

Brake test stand: Measurement sensor
1 Adjustment plate, 2 Alignment pin,
3 Flexural sensor with wire strain gauge,
4 Thrust block, 5 Torque lever.



Determining braking force F_{Br} via measurement of reaction torque M_R
1 Vehicle tire,
2 Roller set with spacing a,
3 Motor with gear set,
4 Torque lever with length l,
5 Measurement sensor,
6 Display.



Electronic Stability Program (ESP) for passenger cars

Quantities and terms

a	Distance of front wheels from center of vehicle mass
a_x	Estimated longitudinal acceleration of vehicle
a_y	Measured lateral acceleration of vehicle
A_0, A_1, A_2	Parameters for calculation of tire operating point
c	Distance of rear wheels from center of vehicle mass
C_p	Brake-torque ratio
C_λ	Longitudinal tire stiffness
D_T	Sampling time of the controller
D_λ	Tolerance band of drive-slip difference
F_B	Brake force on the tire
F_{BF}	Steady-state (filtered) brake force on the tire
F_N	Normal force on the tire
F_R	Resultant force on the tire
F_S	Lateral force on the tire
J_{Mot}	Moment of inertia of the engine
J_{Whl}	Moment of inertia of the wheel
K_p, K_d, K_i	Controller gains for the P-, D- and I-parts
M_{Dif}	Local brake-torque difference between the driven wheels
M_{DR}	Engine torque requested by driver
M_{YwNo}	Nominal yaw moment on the vehicle
ΔM_{YwNo}	Small change in the nominal yaw moment on the vehicle
M_{CaHalf}	Half of the torque on the cardan axle
M_{Ca}	Torque on the cardan axle
M_{Mot}	Actual engine torque
M_{WhlNo}	Nominal brake torque
M_{NoMot}	Nominal engine torque
M_{NoLock}	Nominal brake-torque difference between the driven wheels
M_{NoSPR}	Nominal engine-torque reduction by spark retard
p_{Circ}	Brake circuit pressure induced by the driver
p_{Whl}	Pressure in wheel brake cylinder
p_{WhlPre}	Nominal pressure in wheel brake cylinder
R	Wheel radius
T_{fOFF}	Time during which the fuel injection is shut off
$\ddot{U}_{Tr}$	Transmission ratio
U_{val}	Actuation mode of solenoid valve
υ_{CH}	Characteristic velocity of the vehicle
υ_{Dif}	Difference between driven-wheel velocities
υ_V	Vehicle velocity
υ_{Ca}	Cardan-axle speed
υ_{Whl}	Measured wheel speed
υ_{Whl3}	Measured wheel speed, rear left
υ_{Whl4}	Measured wheel speed, rear right
υ_{WhlFre}	Wheel speed (free-rolling)
υ_{NoDif}	Nominal difference between wheel velocities
υ_{NoCa}	Nominal velocity of the cardan axle
υ_x	Vehicle speed in the longitudinal direction
υ_y	Vehicle speed in lateral direction
x_1, x_2	Parameters of the inverse hydraulic model
α	Slip angle of the tire
α_0, λ_0	Arbitrary operating point of the tire
β	Slip angle of the vehicle
β_{No}	Nominal slip angle of the vehicle
δ	Steering-wheel angle
δ_W	Front-wheel steering angle
λ	Tire slip
λ_{No}	Nominal tire slip
λ_{Ma}	Nominal average drive slip of the driven wheels
μ_{HF}	Coefficient of friction between tires and road
μ_{Res}	Resultant coefficient of friction between tire and road
$\dot{\psi}$	Yaw rate
$\dot{\psi}_{No}$	Nominal yaw rate
m	Minimum value of the nominal driven-wheel brake torque
MIN	Minimum-value operator
SUM	Event-driven integration
ZWV	Spark retard (SPR)

Assignment

The Electronic Stability Program ESP (vehicle dynamics control) system is a closed-loop control system which prevents lateral instability of the vehicle. It is integrated within the vehicle's brake system and dri-

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vetrain. While ABS prevents wheel lockup when braking and TCS prevents spin of the driven wheels, ESP prevents the vehicle from "pushing out" of the turn or spinning out of the turn when it is steered.

Further to the advantages inherent in ABS and TCS, ESP improves the active driving safety in the following points:

- Provides the driver with active support, even in laterally critical dynamic situations.
- Enhances vehicle stability and tracking performance even in limit situations in all operating modes such as full braking, partial braking, coasting, accelerating, engine drag, and load shift.
- Enhances directional stability even during extreme steering maneuvers (panic reactions), resulting in a drastic reduction in the danger of skidding.
- Improved handling behavior also in limit situations. For the driver, this behavior becomes predictable as a function of his (or her) experience. The vehicle remains fully under control even in critical traffic situations.
- Depending upon the situation, even better utilization of the friction potential between the tires and the road when ABS and TCS intervene, and therefore improved traction and stopping distances in addition to improved steerability and stability.

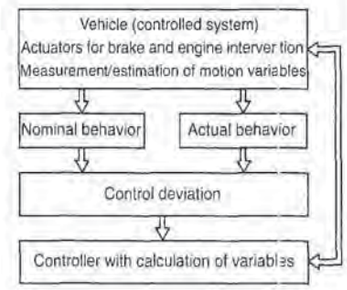
Vehicle handling

The vehicle's steering properties are defined in the description of the vehicle's lateral dynamics (P. 350). This description is derived from steady-state cornering and shows the dependence of the tire slip angle α on the lateral vehicle acceleration a_y and therefore on the lateral tire forces. Furthermore, the description of ABS (P. 659) and TCS (P. 606) also underlines the dependence of the lateral tire forces on the tire slip. This means that the vehicle's steering properties can be influenced by means of the tire slip. ESP utilizes this tire characteristic in order to effect servo-control for vehicle handling.

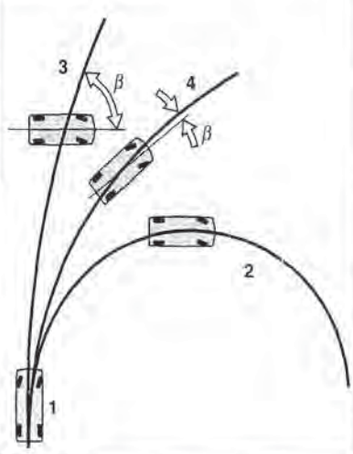
It is important for good vehicle handling that the vehicle remains safely on a track which corresponds as far as possible to the course of the steering-angle (see Fig. "Lateral dynamics of a vehicle", Curve 2). During steering maneuvers, this is the case if the lateral tire forces remain considerably below the friction potential between the tire and the road surface. The yaw-rate curve then corresponds to that of the steering angle.

For ESP though, the mere control of the yaw rate according to the steering angle proves to be inadequate and the vehicle

Block diagram of ESP controller



Lateral dynamics of a vehicle
1 Step input at steering wheel, steering-wheel angle fixed, 2 Track on high-μ road, 3 Track on low-μ road with "open-loop" steering correction and yaw-rate cor trol, 4 Track on low-μ road when the slip angle is also controlled (ESP).



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can become instable (see Fig., Curve 3). The ESP therefore controls not only the yaw rate but also the vehicle's sideslip angle.

The ESP vehicle-handling control is not restricted to ABS and TCS/MSR operations, but also extends to cover the area in which the vehicle is rolling freely, and also comes into action during partial braking if the vehicle is moving at the physical driving limits.

ESP control systems

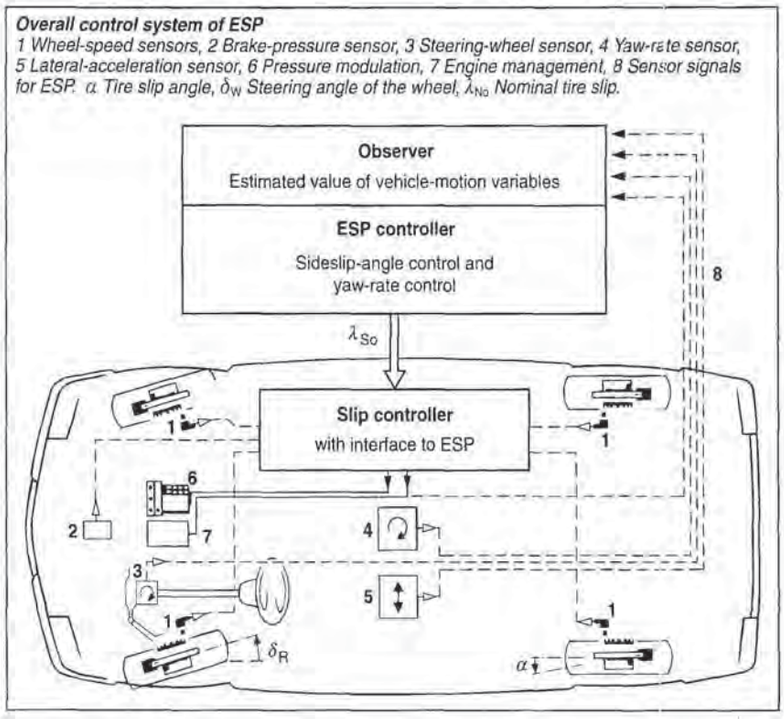
Control

The control of the vehicle's behavior at the physical driving limit must influence the vehicle's three degrees of freedom in the plane of the road (longitudinal and lateral velocities, and yaw moment about the vertical axis) so that vehicle handling is in line with the driver's input and with the prevailing road conditions. In this respect, as shown in the block diagram, it must first of all be defined how the vehicle is to behave at the physical driving limit in accordance with the driver's input (nominal behavior), and how it actually behaves (actual behavior). In order to minimize the difference between nominal and actual behavior (deviation), the tire forces must in some way be controlled by actuators.

The overall system (Fig. "Overall control system of ESP" below), shows the vehicle as the controlled system, with sensors (1...5) for defining the controller input variables, and actuators (6 and 7) for influencing the tractive force and the braking force. Also shown are the hierarchically structured controller comprising the higher-level vehicle dynamics controller, and the lower-level slip controllers. The higher-level controllers input the nominal values to the lower-level controllers in the form of a nominal slip. The "observer" determines the controlled state variable (vehicle slip angle β).

In order to determine the desired performance, the signals are eva uated which

Overall control system of ESP
1 Wheel-speed sensors, 2 Brake-pressure sensor, 3 Steering-wheel sensor, 4 Yaw-rate sensor, 5 Lateral-acceleration sensor, 6 Pressure modulation, 7 Engine management, 8 Sensor signals for ESP. α Tire slip angle, δ_W Steering angle of the wheel, λ_{No} Nominal tire slip.



define the driver's input. These comprise the signals from the steering-wheel sensor (3, driver's steering input), the brake-pressure sensor (2, desired deceleration input), and the engine management (7, desired drive torque). Apart from the vehicle speed, the calculation of the desired performance also takes the coefficients of friction between the tires and the road into account. These are calculated from the signals outputted by the wheel-speed sensors (1), the lateral-acceleration sensor (5), the yaw-rate sensor (4), and the brake-pressure sensor (2). Depending upon the control deviation, the yaw moment which is necessary to make the actual-state variables approach the desired-state variables is then calculated.

In order to generate the required yaw moment, it is necessary for the changes in desired slip at the wheels to be determined by the vehicle dynamics controller. These are then set by means of the lower brake-slip controller and traction controller together with the "brake-hydraulics" actuator (6), and the "engine-management actuator" (7).

The system relies on tried and proven ABS and TCS components. The TCS hydraulic modulator (6), which is described elsewhere, permits high levels of dynamic braking of all wheels throughout the complete temperature range encountered.

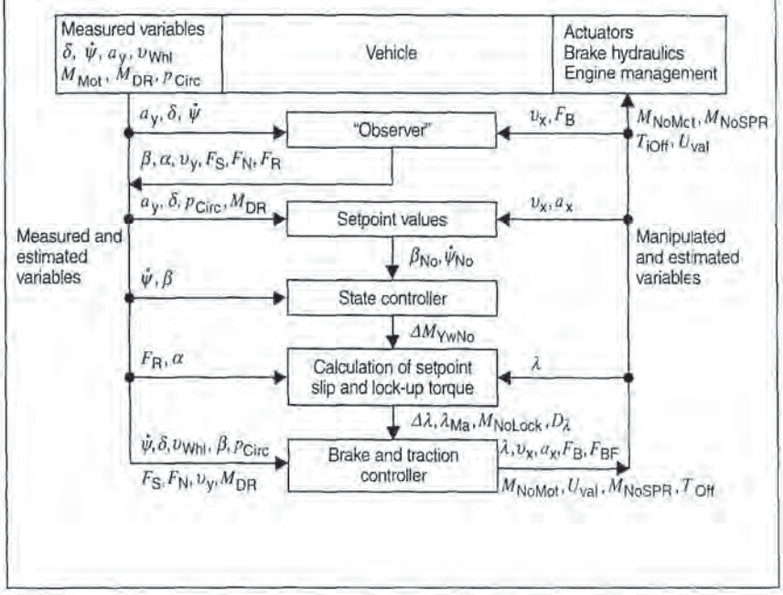
The necessary engine torque can be set by means of the engine management (7) and the CAN interface, so that the traction-slip values at the wheels can be adjusted accordingly.

System components

Vehicle dynamics controller

The relationship between steering angle, vehicle speed, and yaw rate during steady-state driving on a circular course (P. 355), is taken as the basis for the desired vehicle motion during steady-state driving, and when braking and/or accelerating. By applying the "Single-track (bicycle) model": the nominal yaw rate is cal-

Block diagram of ESP vehicle dynamics controller with input and output variables



culated from the vehicle speed and the steering angle:

$$\dot{\psi}_{No} = \frac{v_x \cdot \delta_W}{(a+c)\left(1+\frac{v_x^2}{v_{CH}^2}\right)}$$

For the sideslip-angle control, this value is first of all limited according to the road-surface coefficient of friction:

$$|\dot{\psi}_{No}| \leq \mu_{HF} \cdot g / v_V$$

whereby g is the acceleration due to gravity, and the coefficient of friction μ_{HF} and the vehicle speed v_V are estimated values.

No attempt is made to directly influence the lateral speed, and with it the sideslip angle, by means of changes in the lateral forces. It is better to adjust the lateral motion indirectly by means of changes in the slip angle, whereby the vehicle's rotation is initiated by the generation of yaw moments.

A simplified block diagram is used to demonstrate the structure of the vehicle dynamics controller. Using the measured variables (nominal yaw rate $\dot{\psi}_{No}$, steering-wheel angle δ, and the vehicle lateral acceleration a_y), together with the estimated variables (vehicle longitudinal speed v_x, and braking forces F_B), a so-called "Observer", using a simple vehicle model, determines the slip angle β, the sideslip angle α, the vehicle lateral speed v_y, the wheel forces in the lateral and vertical directions F_S and F_N, and the resulting wheel forces F_R in the plane of the road.

The nominal sideslip-angle β_{No}, and the nominal yaw rate $\dot{\psi}_{No}$, are determined from the following driver inputs: steering-wheel angle δ, engine-torque input M_{DR} (accelerator-pedal position), and the brake pressure p_{Circ}, together with the estimated vehicle longitudinal speed v_x, and the coefficient of friction μ_{HF} which is determined using the estimated longitudinal acceleration a_x and the measured lateral acceleration. In the process, the vehicle's dynamic response, and special situations such as inclinations, banks, and/or μ-split road surfaces (different coefficients of friction between tires and road on the left and right side of the vehicle) are taken into account.

The vehicle dynamics controller is designed as a state controller. The controlled-state variables are the sideslip angle and the yaw rate. The more the sideslip angle increases, the more it is taken into account by the vehicle dynamics controller. The state controller's output variable corresponds to a yaw moment M_{YwNo}.

Together with the actual slip values λ, the resultant wheel forces F_R, and the slip angles α, the linearized vehicle model is now also applied to convert this yaw moment into changes of the nominal slip at the appropriate wheels. For instance, presuming that the vehicle is rolling freely and oversteers in a right-hand bend so that the nominal yaw-rate is exceeded, among other things a nominal brake slip is requested at the front left wheel. This subjects the vehicle to a counterclockwise change in yaw moment which reduces the excessive yaw rate. The nominal slip is changed by the lower-level ABS or TCS wheel controller. If the brakes have not been applied, or the driver's brake-pressure input does not suffice to set the required nominal slip (partial-braking range), the pressure in the brake circuits is actively increased.

When TCS is operative, in order to set the vehicle's nominal yaw moment M_{YwNo}, the vehicle dynamics controller outputs a mean absolute drive slip $\lambda_{Mε}$, a drive-slip tolerance band D_λ, and a nominal brake locking torque M_{NoLock}. In order to achieve distinct improvements for the ABS and TCS basic functions with regard to their utilization of the adhesion potential in line with the driving situation, all available measurement quantities and estimated values are also applied to the full in the slip controllers.

Lower-level brake-slip controller (ABS) and engine drag-torque controller (MSR)

In order to be able to control the wheel slip to a definite level, it must be possible to evaluate it with an adequate degree of accuracy. The vehicle's longitudinal speed is not measured, but is determined from the wheel speeds v_{Whl}. To do so, individual wheels are "under-braked" during ABS application. In other words, slip control is interrupted, and the instantaneous wheel

brake torque is reduced by a defined amount and held constant for a given period of time. Assuming that the wheel speed is stable toward the end of this period, the zero-slip (free-rolling) wheel speed v_{WhlFre} can be calculated from the momentary braking force F_B and the tire stiffness C_λ.

$$v_{WhlFre} = v_{Whl} \cdot \frac{C_\lambda}{C_\lambda - \frac{F_B}{F_N}}$$

Using the yaw rate $\dot{\psi}$, the steering angle δ_W, and the lateral velocity v_y, together with the vehicle geometry, the (free-rolling) wheel speed is transformed into the center of gravity in order to generate the center-of-gravity velocity v_x in the vehicle's longitudinal direction. Finally, v_x is transformed back to the four wheel positions in order to evaluate the free-rolling speed of all four wheels. This enables the actual slip λ to be calculated for the control of the three remaining wheels.

$$\lambda = 1 - \frac{v_{Whl}}{v_{WhlFre}}$$

Starting from the stationary brake force F_{BF}, using a PID control law the nominal torque at the wheel can be calculated as a function of the slip deviation.

$$M_{WhlNo} = F_{BF} \cdot R + K_p(\lambda_{No} - \lambda)R + K_d\left(\frac{d}{dt}v_{Whl} - \frac{d}{dt}v_{WhlFre}\right)\frac{J_{Whl}}{R} + K_i \cdot C_p \cdot \text{SUM}\{(\lambda_{No} - \lambda) \cdot D_T\}$$

In the case of the driven wheels, in order to implement an engine drag torque control (MSR) the nominal braking torque M_{WhlNo} can be partially set by the engine (or completely when the brakes are not applied). The driven wheel with the lower nominal wheel torque is controlled within the permissible limits by engine intervention.

Block diagram of the brake-slip controller for ABS and MSR with input and output quantities

