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-

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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.

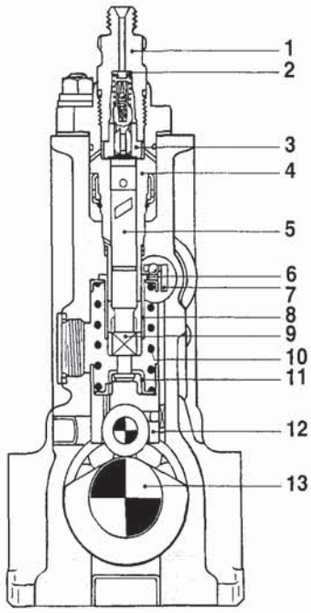
Common to all of these systems is the fact that the required injection pressure is generated at the moment each injection occurs. In the case of electronically controlled systems, however, the rate-of-discharge curve can be influenced not only by the cam contour but also by solenoid valve actuation. 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

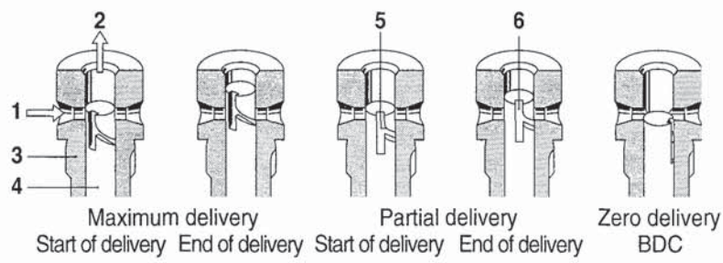
A piston pump delivers the fuel to the injection pump's fuel gallery at a pressure of 1...2.5 bar. The cam-driven supply-pump plunger travels to TDC on every stroke. It is not rigidly connected to the drive element; instead, a spring supplies the return pressure. The plunger spring responds to increases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

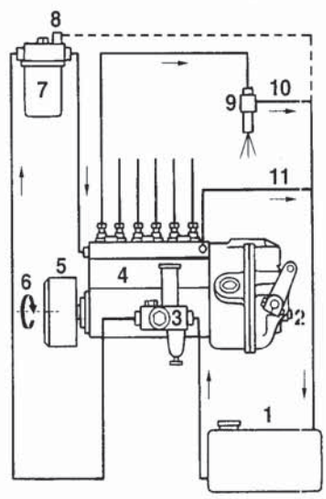
The delivery valve and pressure-relief characteristics must be specially designed for the specific application. Units incorporating a return-flow restriction or constant-pressure valve have an additional throttle element to damp the pressure waves reflected back from the injection nozzle, thus preventing 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.

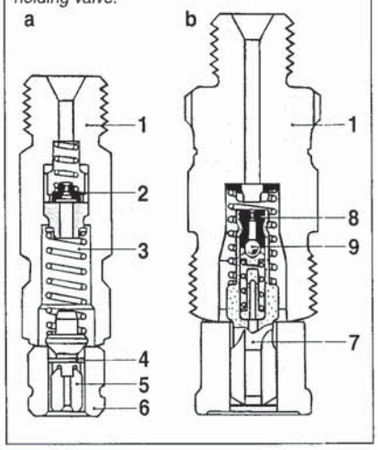
In pumps which generate injection pressures greater than approx. 600 bar, the plunger-and-barrel assembly, delivery valve and delivery-valve holder are screwed together to form a single unit, which means that the high sealing forces must no longer be accommodated by the pump housing (e.g. Sizes MW, P).

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.



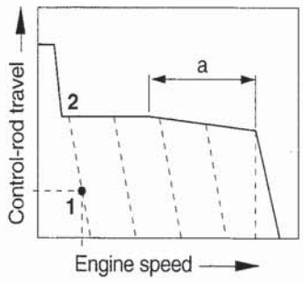
Delivery-valve holder with delivery valve
a) With constant-volume valve and return-flow restriction, b) With constant-pressure valve. 1 Delivery-valve holder, 2 Return-flow restriction, 3 Dead volume, 4 Retraction piston, 5 Valve ball, 6 Valve holder, 7 Supply valve, 8 Calibrated restriction, 9 Pressure-holding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



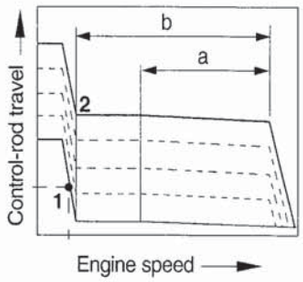
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Governor characteristic curves
a Positive torque control in upper speed range, b Unregulated range, c Negative torque control.
1 Idle-speed setpoint, 2 Full-load curve, 3 Full-load curve, turbocharged engine, 4 Full-load curve, naturally-aspirated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

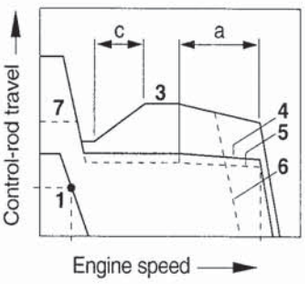
Variable-speed governor



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.

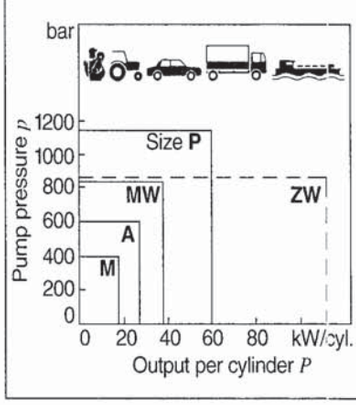
Speed droop

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

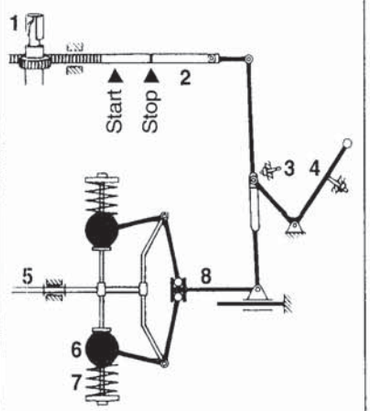
$$\delta = \frac{n_{LO} - n_{VO}}{n_{VO}} \cdot 100\,\%$$

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 %.

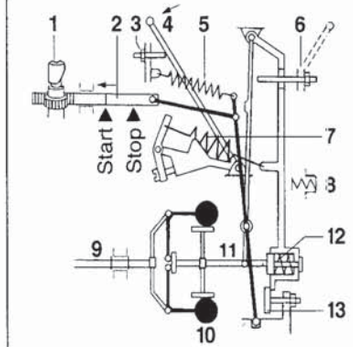
Applications for various types of in-line injection pump



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.



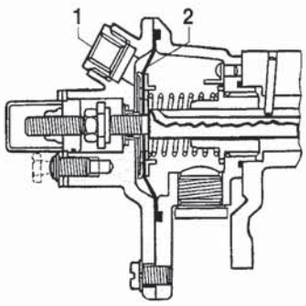
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Mechanical add-on equipment

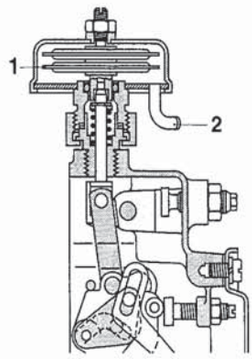
Torque control

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.

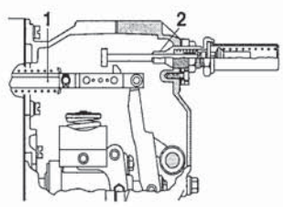
The ECU microprocessor compares accelerator position, min^{-1}, and a number of additional correction factors with the program maps stored in its memory in order to determine the correct injected fuel quantity, i.e. the correct rack position.

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.

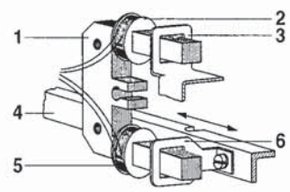
An inductive speed sensor monitors a camshaft-mounted pulse wheel; the ECU uses the pulse intervals to calculate engine speed.

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,

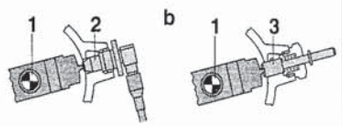
Temperature-dependent starting device
1 Control rack, 2 Start-quantity stop with expansion element.



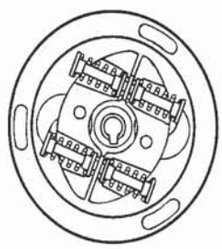
Rack-travel sensor (RWG)
1 Iron core, 2 Reference coil, 3 Fixed short-circuiting ring, 4 Control rack, 5 Measuring coil, 6 Moving short-circuiting ring



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



Timing device
Off position.



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- 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

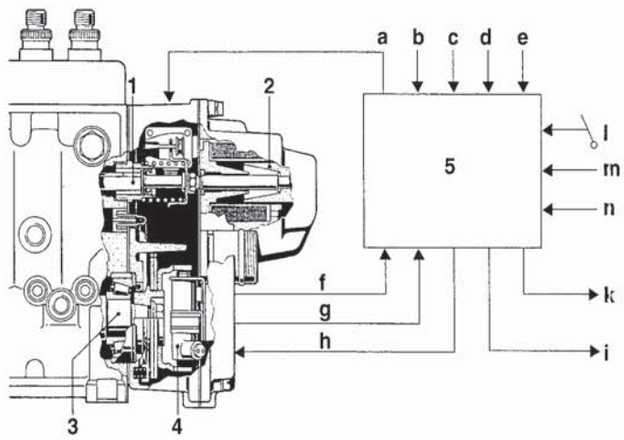
The in-line control-sleeve injection pump makes it possible to provide electronically-controlled adjustment of port closing (start of pump delivery). The spill port, which on conventional in-line pumps is in the housing and therefore immovable, is incorporated in the control sleeve which is a component 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.



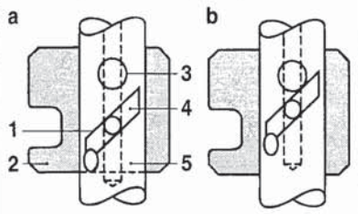
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Distributor-type fuel-injection pump (VE)

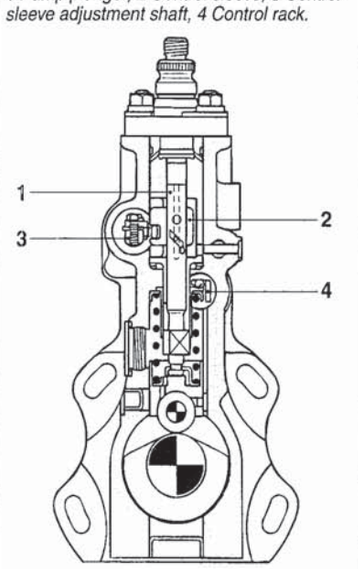
Distributor-type fuel-injection pumps are used in 3-, 4-, 5- and 6-cylinder diesel engines in passenger cars, tractors and light- and medium-duty commercial vehicles which generate up to 42 kW per cylinder, depending on engine speed and combustion system. Distributor-type fuel-injection pumps for direct-injection (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.

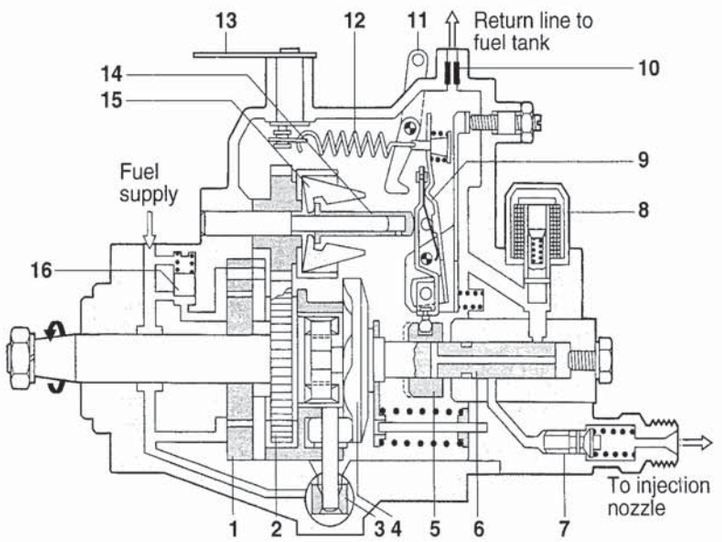
Add-on modules:
A number of add-on modules are available to process additional operating parameters for regulation of delivery quantity (such as manifold-pressure compensator (LDA), start quantity, hydraulic and mechanical full-load torque control) and for adjustment of the start of delivery – (e.g.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

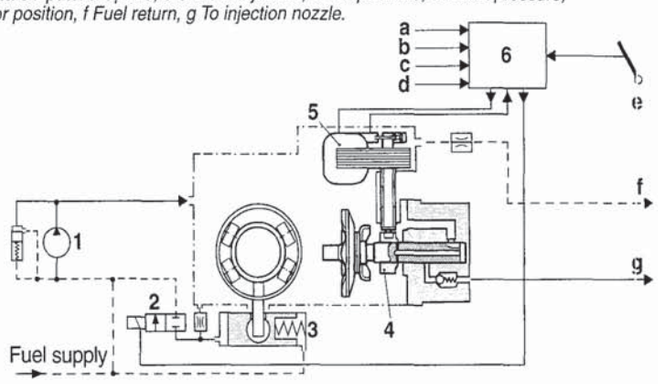
Electronic Diesel Control (EDC): Distributor-type fuel-injection pump with rotating-solenoid actuator
In contrast to the mechanically controlled

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.



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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

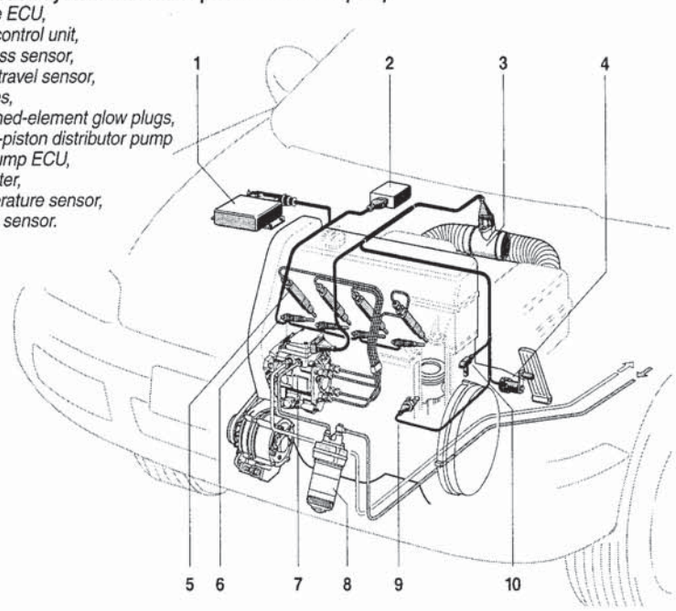
Axial-piston pumps use the same principle of pressure generation as EDC pumps with rotating-solenoid actuators. The injection pressure at the nozzle can be as high as 1500 bar, depending on the application.

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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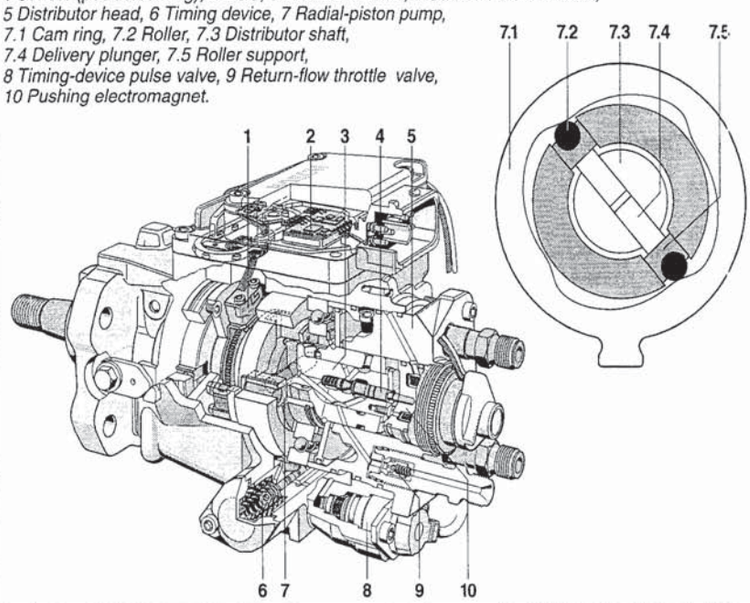
Fuel-injection system with radial-piston distributor pump

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.



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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

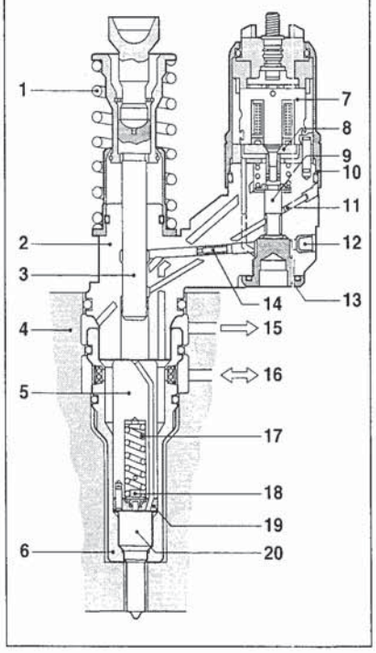
The new modular, time-controlled single-cylinder pump systems include the electronically controlled "Unit Injector System" (UIS) and the "Unit Pump System" (UPS) which are used on modern direct-injection engines for commercial vehicles and passenger cars.

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:

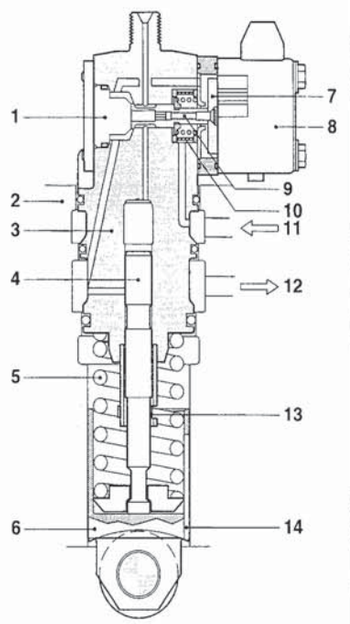
- high-pressure single-cylinder pump with integral high-speed solenoid valve,
- short high-pressure delivery line
- nozzle-holder assembly

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.

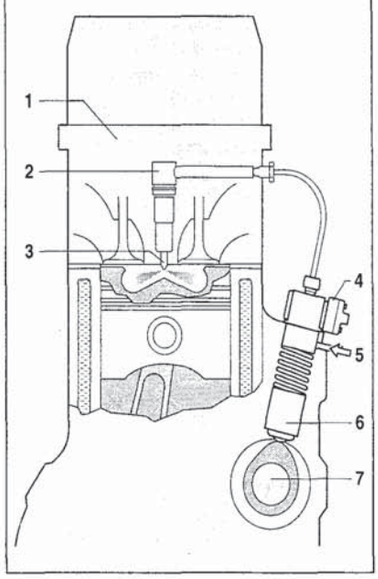
When the solenoid valve is open, the fuel can be drawn into the pump cylinder during the pump plunger's suction stroke, and return during the supply stroke. Only when the solenoid valve is energized, and thus closed, can pressure build up in the high-pressure system between the pump plunger and the nozzle 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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ınd 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 he main-injection phase.

Electronic control allows the values for he start of injection and the injected fuel ıuantity to be selected as desired from hose stored in the program map. This eature, together with the high injection ›ressures, makes it possible to achieve ⁄ery high power densities combined with ⁄ery low emission levels and exception-ılly low fuel consumption.

The further reduction in size of the unit njectors will allow them to be used on 1-valve-per-cylinder engines in the future, hereby making it possible to reduce ɜmissions even more. Its ability to further ɾeduce emissions, coupled with further ›ptimization of injection characteristics, nakes the unit injector system capable of neeting 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 sta-:us parameters in the system relative to :he engine and its environment, and de-ines the exact start of injection and in-ected fuel quantity for the operating state ›f 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.

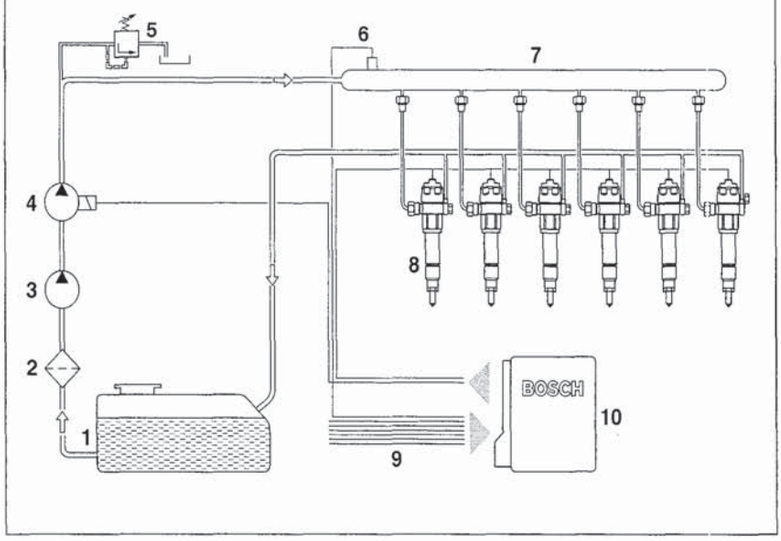
System application engineering on the engine

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.



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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.

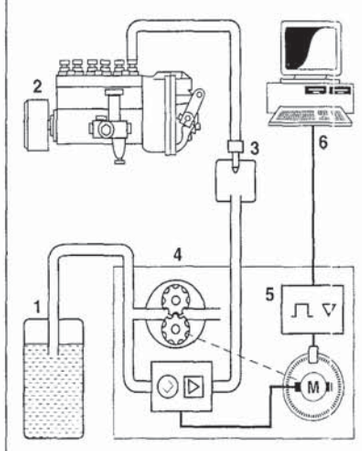
Continuous-flow delivery-quantity measurement

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).

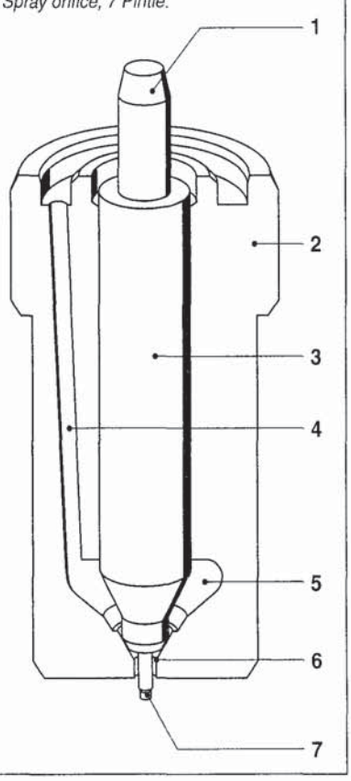
As a distinctive feature, throttling-pintle nozzles vary the discharge aperture – and thus the flow rate – as a function of needle stroke. The hole-type nozzle displays an immediate, sharp rise in aperture when the needle opens; in contrast, the throttling-pintle nozzle is characterized by an extremely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

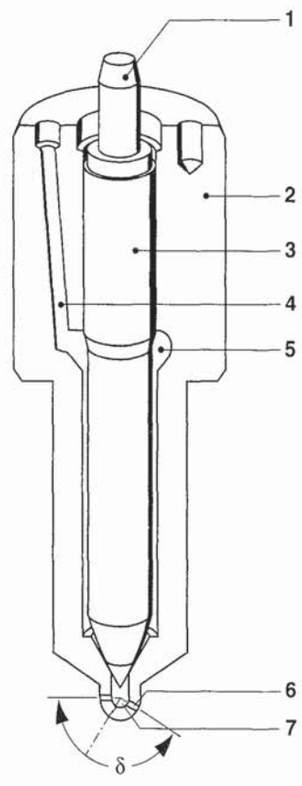
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Hole-type nozzles

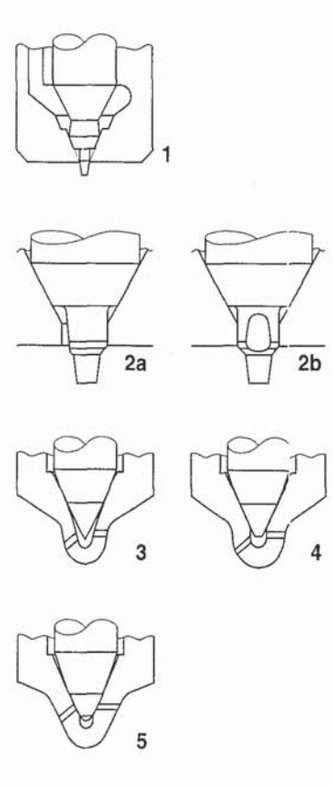
A wide range of nozzle-and-holder assemblies (DHK) is available for hole-type nozzles. In contrast to throttling pintle nozzles, hole-type nozzles must generally be installed in a specific position to ensure correct alignment between the orifices (which are at different angles in the nozzle body) and 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.



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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

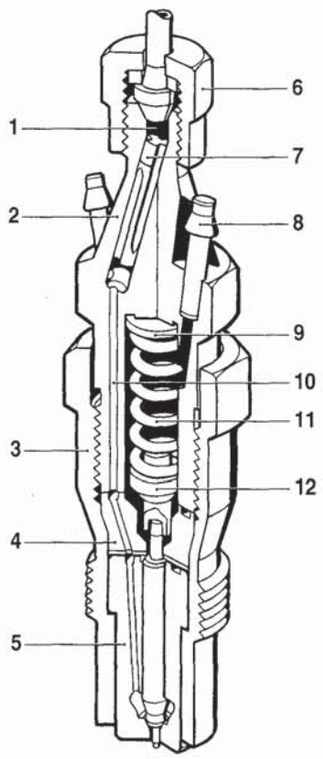
- opening pressure 1,
- opening pressure 2,
- prestroke, and
- overall stroke.

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.

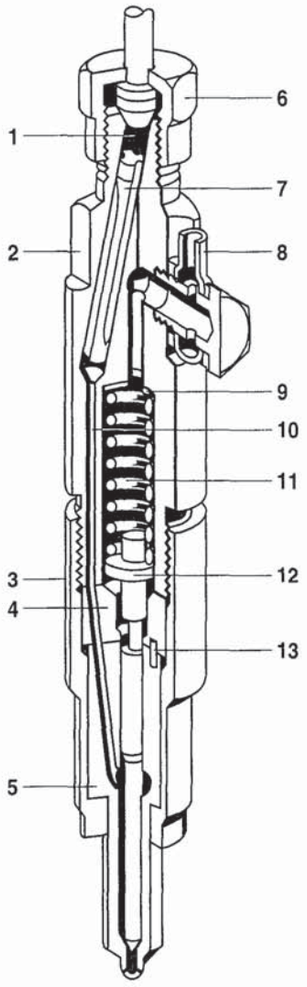
There are also dual-spring holders available for prechamber and whirl (turbulence) chamber engines. The setpoints are tailored to the respective injection system, with varying opening pressures of 130/180 bar and prestrokes of approx. 0.1 mm.

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.

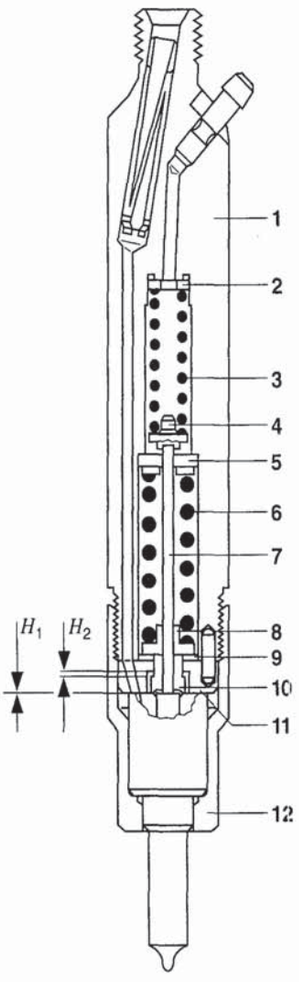


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Nozzle-holder assembly *with hole-type nozzle.*
1 Inlet , 2 Nozzle-holder body, 3 Nozzle-retaining nut, 4 Intermediate element, 5 Injection nozzle, 6 Union nut with high-pressure line, 7 Edge filter, 8 Leak-off connection, 9 Pressure-adjusting shims,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



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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.

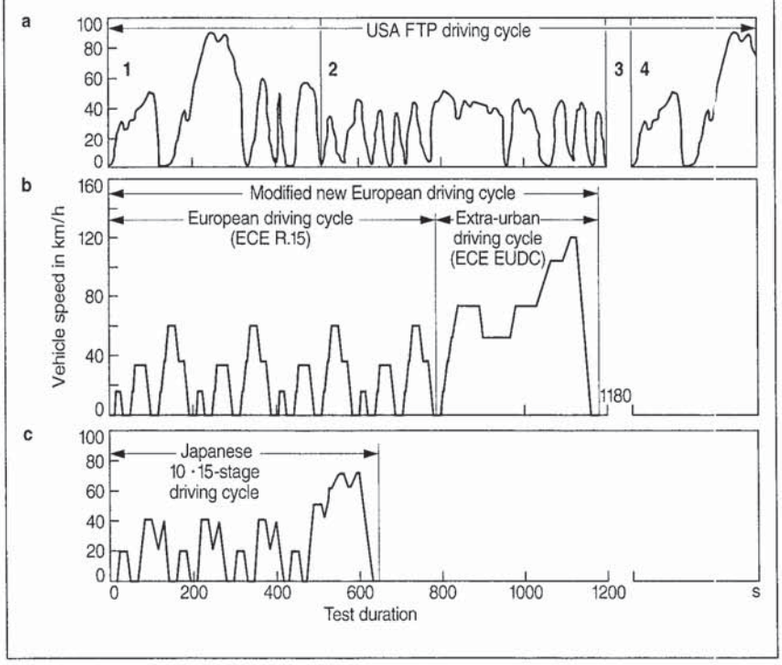
A sample of the diluted exhaust gas is passed through special filters whose increase in mass (determined by weighing on a microbalance before and after sampling) is used to calculate the level of particulate emissions.

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).



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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

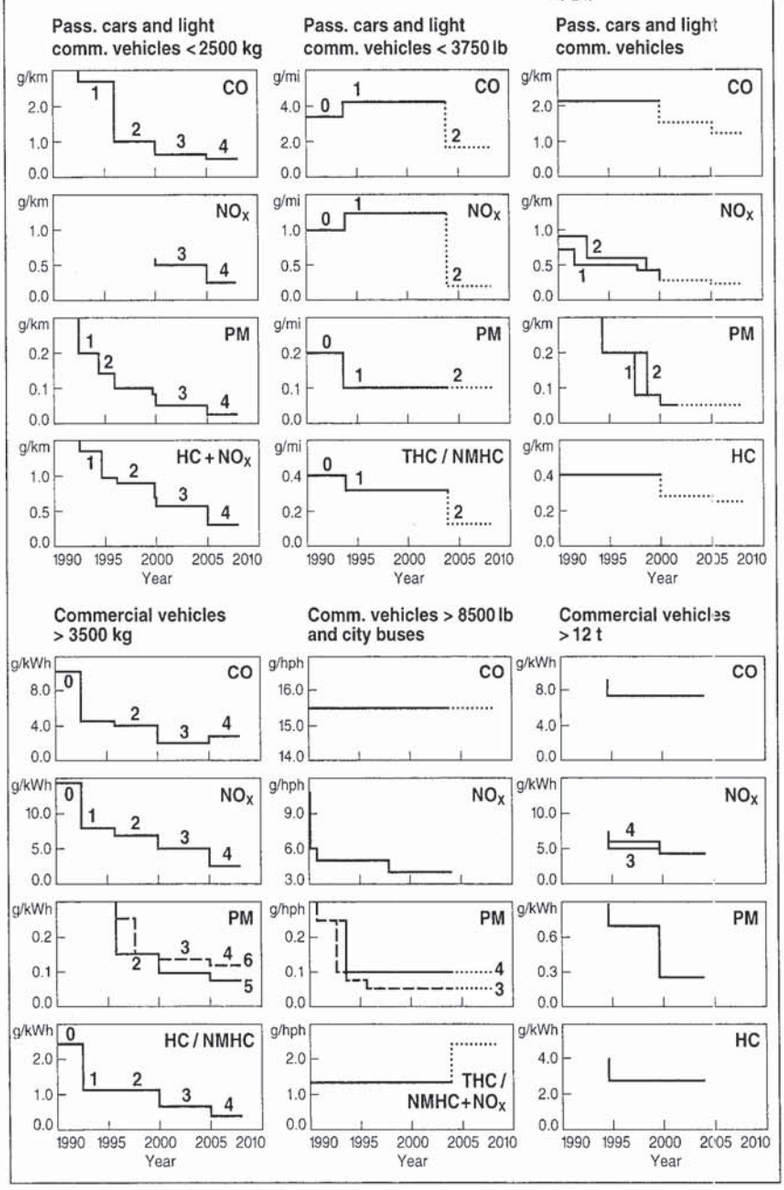
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Emission limits for diesel-powered passenger cars and commercial vehicles

Europe
0...**4** Euro 0...4,
5 >85 kW,
6 <85 kW.

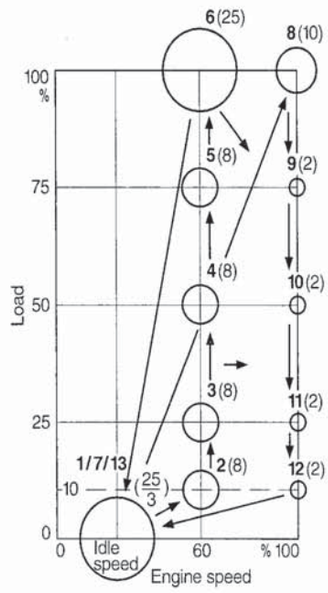
USA
0...**2** Level 0...2,
3 HDV,
4 UB.

Japan
1 EIW >1265 kg,
2 EIW <1265 kg,
3 IDI,
4 DI.

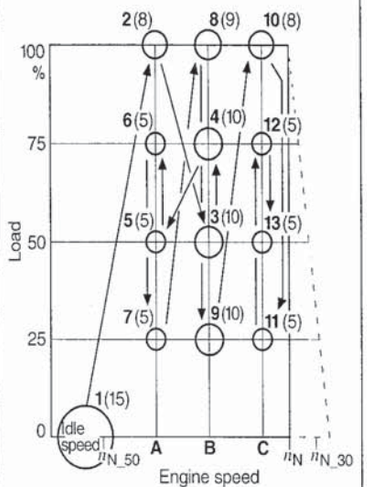


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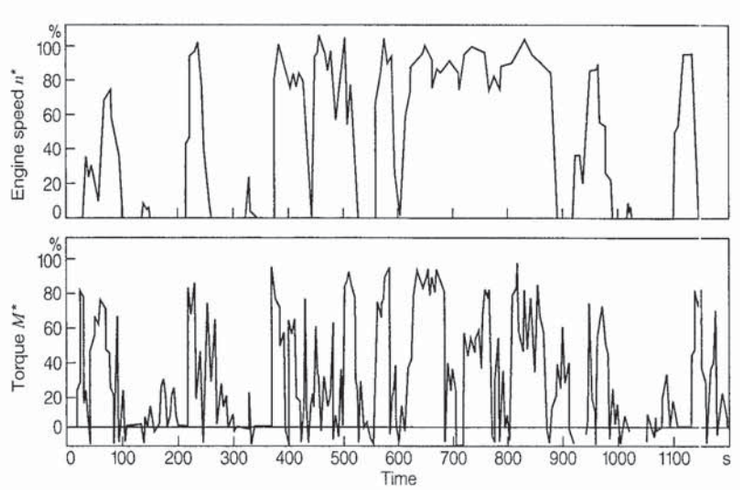
13-stage test ECE R.49 (Europe)
Order (bold) and Weighting
(% in parentheses) for test stages 1...13.



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.



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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

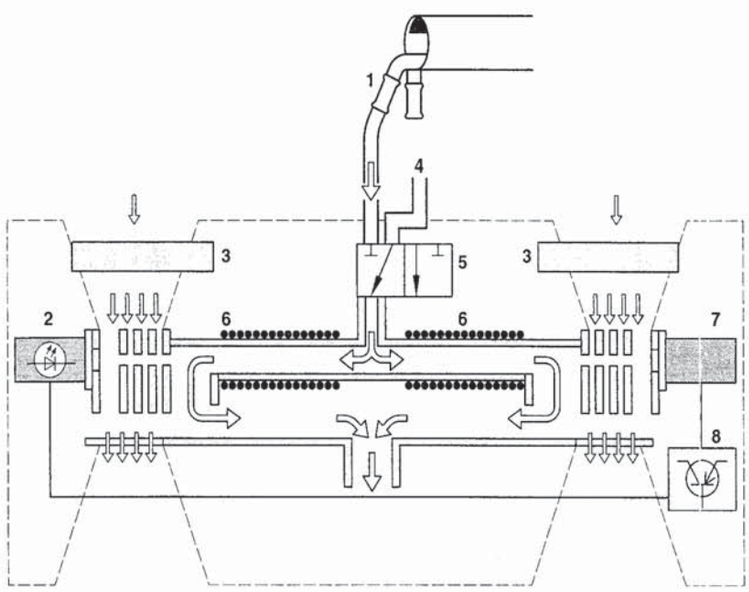
An important reason for testing diesel-engine exhaust emissions in the workshop, and for monitoring them while the vehicle is actually in operation on the road, is to determine the amount of soot (particulates) produced. This is indicated by the smoke number. Two standard procedures are used:

- 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



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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.

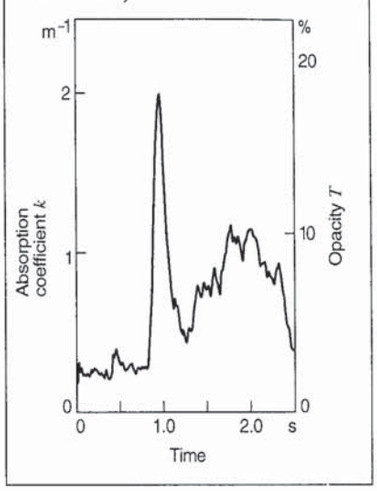
Inside the measuring chamber the diesel exhaust gas is penetrated by a beam of light. The attenuation of the light is measured photoelectrically, and is indicated as % opacity T or as a coefficient of absorption k. A high degree of accuracy and reproducibility of the test results requires that the length 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.

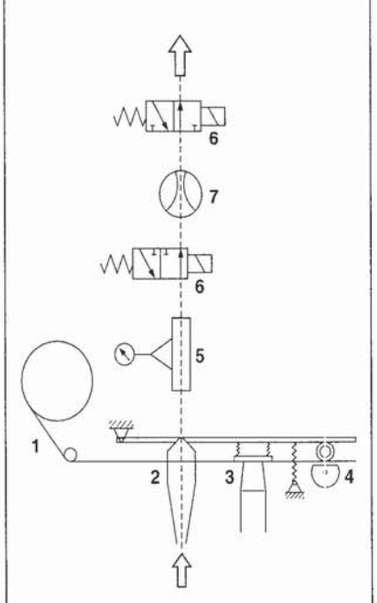
Smoke tester (filter method)
The smoke tester extracts a specified quantity (e.g. 0.1 or 1 *l*) of diesel exhaust gas through a strip of filter paper. Consistent, mutually comparable test results are achieved by recording the volume of gas processed in each test step; the device converts the results 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.



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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.

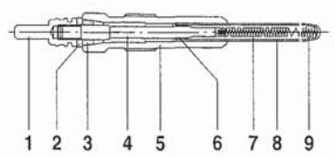
Operation

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.

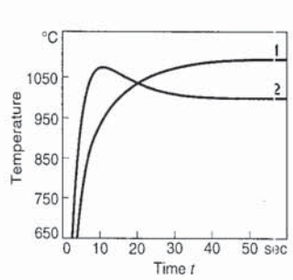
Start phase: Here the glow plug must heat to starting temperature (approx. 850 °C) as rapidly as possible. Plug locations within the combustion chamber are selected to ensure access to an ignitable mixture. Modern glow plugs heat to the required temperature in roughly 4 seconds.

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.

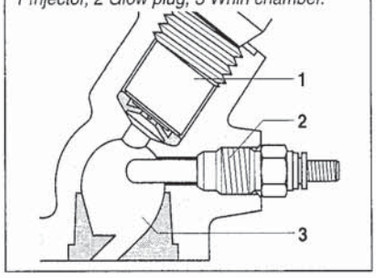


Temperature of sheathed-element glow plug in stationary air as function of time
1 S-RSK, 2 GSK2.



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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

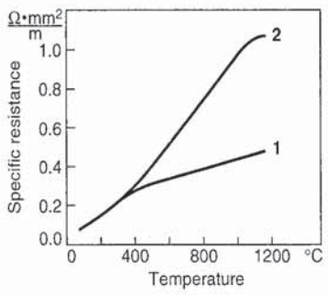
The complete glow-plug system incorporates not only the glow plugs themselves (only in rare cases are flame plugs used) but also a switching element for the high electrical glow-plug currents which is triggered by a special control unit. In addition, the system incorporates an indicator lamp (also controlled 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.

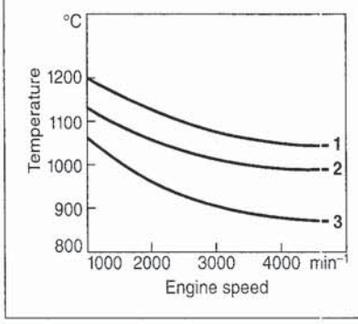
Design

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.

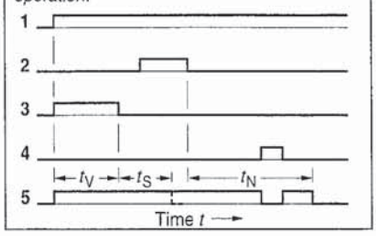


Glow-plug temperatures in DI engine
1 Full load, 2 50 % load, 3 No load.



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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

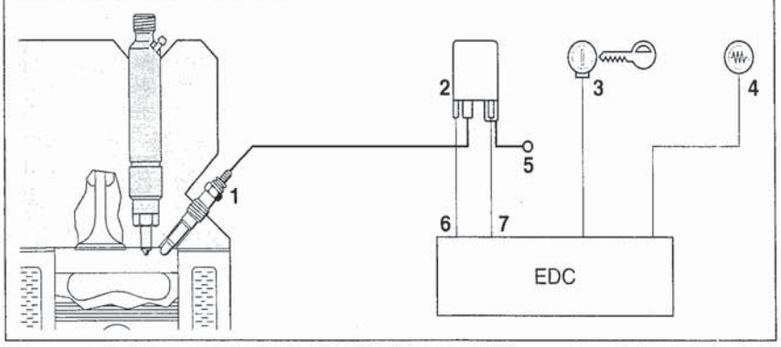
On basic units, a temperature sensor in the glow control unit controls the preheating period. This is designed to suit the specific requirements of the particular combination of engine and glow-plug so that the glow plug can reach the temperature necessary for efficient starting. At the end of the glow 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 information 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.



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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.

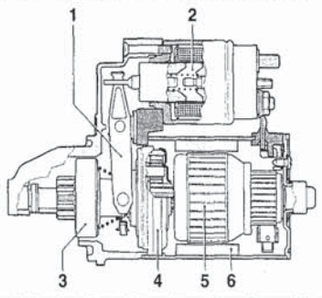
Design factors

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.

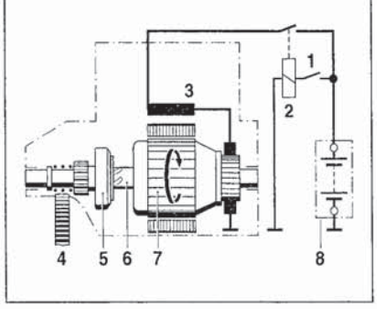
If the pinion and ring gear happen to find themselves in mutually optimal positions, then the pinion will immediately mesh with a gap in the ring gear's teeth. In this case the pinion teeth's engagement depth in the ring gear increases until the travel limit is reached and the contact bridge in the solenoid 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.

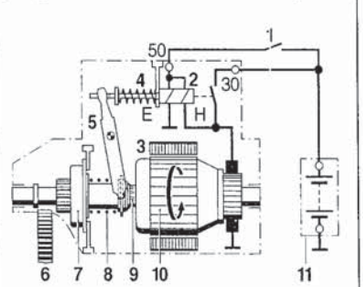
The electric motor then turns the pinion which is in contact with the face of the ring gear until a pinion tooth finds a ring-gear gap and the meshing spring pushes the pinion and overrunning clutch forward.

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.

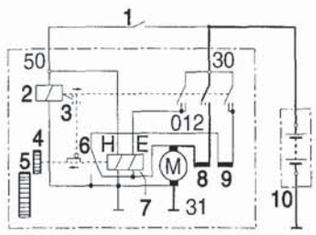
This disengaging operation is assisted by the helix when the engine speed overtakes the starter speed.

Sliding-gear starters

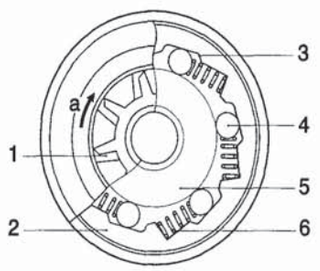
The sliding-gear drive switches the starter on in two stages.

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

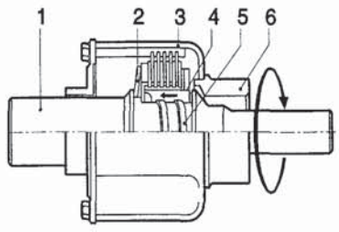
The multiplate overrunning clutch is used in large commercial-vehicle starters. The driver with the outer plates and the starter armature, and on the other side the driving shaft and the pinion are positively connected to one another. The inner plates fit in an inner clutch race which rides axially in 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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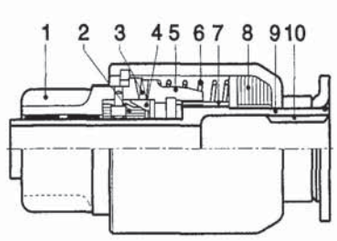
to transmit increasing torque as the starter load increases.

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.

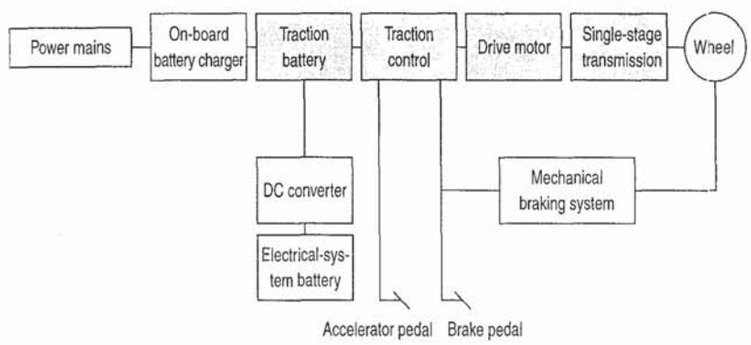
Depending on the intended application, battery-powered electric vehicles can be classified as either road vehicles or industrial trucks. Industrial trucks are used for transporting goods on company premises, and are generally not licensed for use on public roads. Their top speed is below 50 km/h. Due 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)*



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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

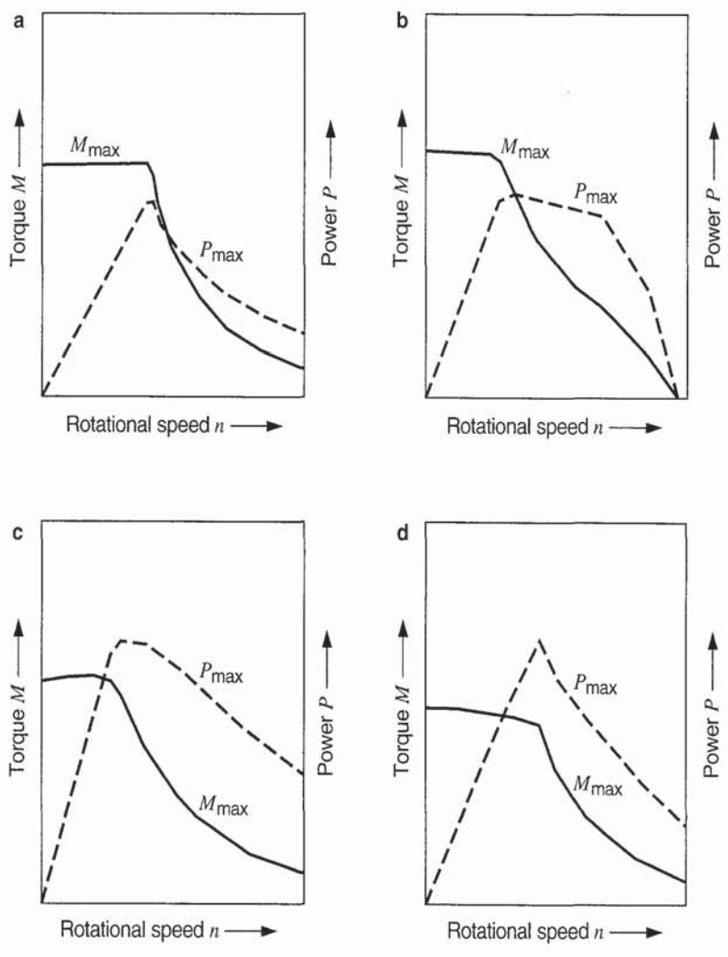
In this type of drive unit, the motor's magnetic excitation is provided by its own controller (field rheostat). Depending on the size of the motor, the field can be weakened in a ratio of up to approx. 1 4. Field strength starts to diminish at a nominal motor speed obtained with full motor voltage at 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.

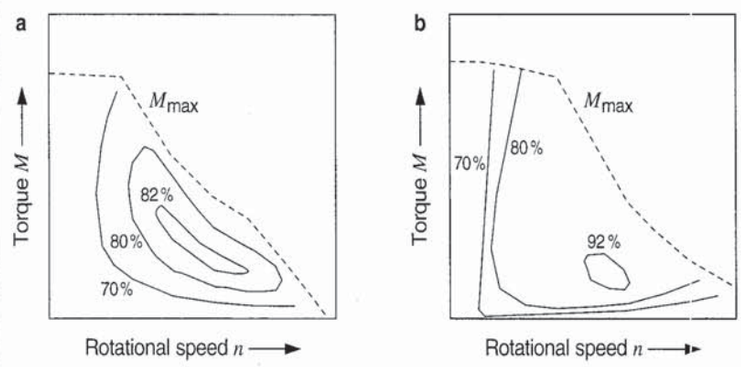
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Torque and power as a function of rotational speed for various types of drive
a) Series-wound DC drive,
b) Separately-excited DC drive,
c) Asynchronous drive,
d) Permanently-excited synchronous drive.
M_{max} *Maximum torque,*
P_{max} *Maximum power.*



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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.

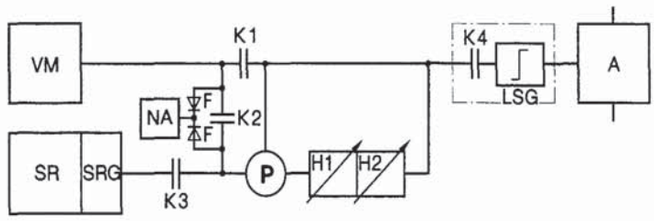
In addition, the internal-combustion engine can start the two kinetic-energy storage devices from standstill via the continuously variable transmission. This configuration also permits parallel operation with the two drive sources, diesel engine and flywheel. Here, only the power delivered by the flywheel 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.

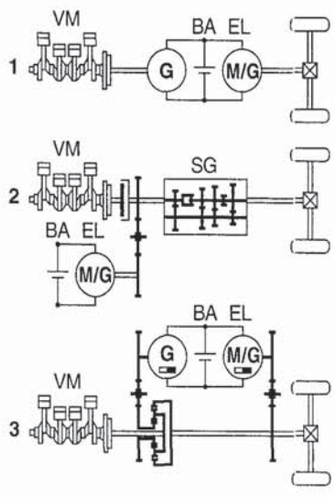
Combining a mechanical energy store with the battery prevents it being affected by power peaks, thus contributing to more efficient utilization of the battery energy. A hybrid system which uses a combination of two different electrochemical energy sources (battery and fuel cell) separates the energy 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.

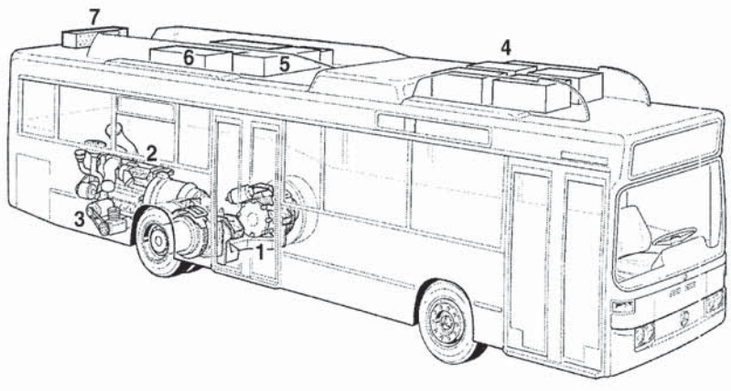
Because the diesel engine in the series configuration is mechanically decoupled from the vehicle drive, the diesel engine can be operated at a constant speed, i.e. at its optimum operating point in terms of efficiency and emissions. Despite the advantages of the series configuration, its disadvantage 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.



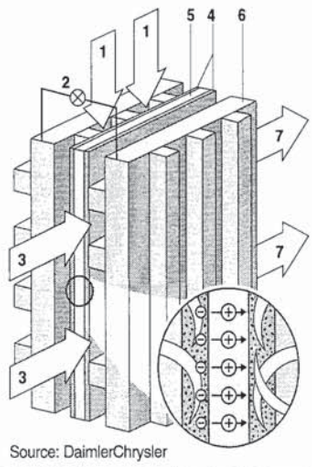
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Fuel Cells

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

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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	50...80	50...60	Mobile, stationary
Direct Methanol Fuel Cell	DMFC	Membrane	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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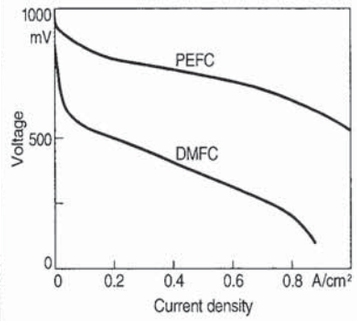
familiar electrolytic gas reaction, which in fuel cells is a controlled process ("cold combustion") rather than an explosive one. This reaction takes place in a controlled manner in fuel cells because the important reaction processes (H_2 oxidation and O_2 reduction) occur at physically separate electrodes 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.

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Karl Kordesch, Günther Simander, Fuel Cells and Their Applications, VCH Weinheim 1996.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,557,540

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 $J = J_R + i_{fd}^2 \cdot J_{dt} + i_{fd}^2 \cdot i_G^2 \cdot J_{eng}$

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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

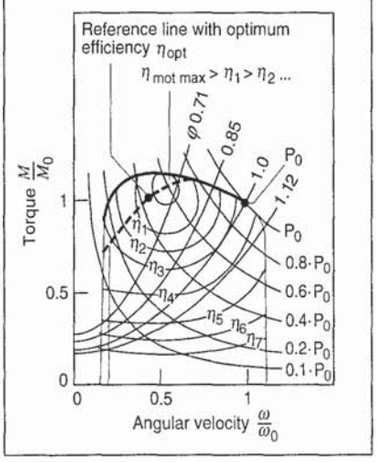
The layout of the automotive drivetrain varies according to the position of the engine and the drive axle:

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

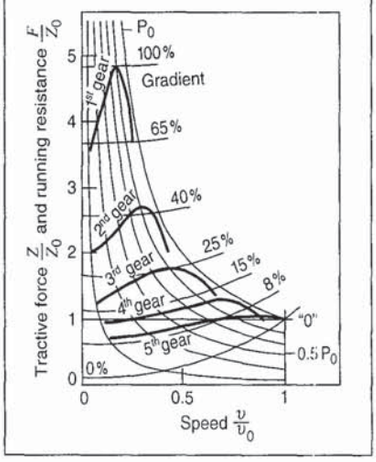
Drivetrain elements

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



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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.

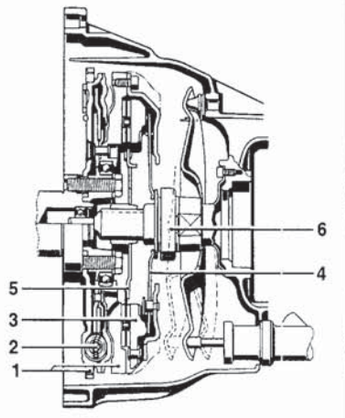
Power take-up elements

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.

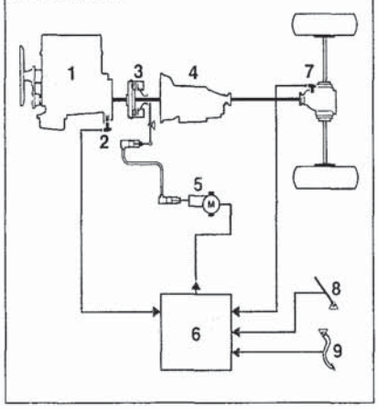
Hydrodynamic torque converter

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.

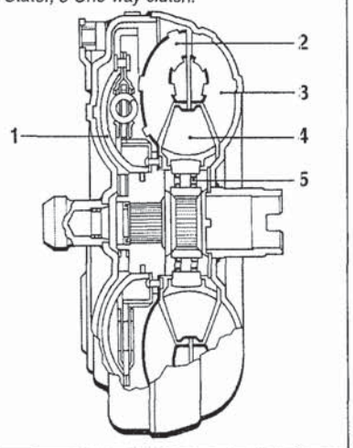
The impeller's input torque M_p and input power P_p are calculated as follows:

$$M_p = \lambda \cdot \rho \cdot D^5 \cdot \omega^2_p$$
$$P_p = \lambda \cdot \rho \cdot D^5 \cdot \omega^3_p$$

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:

$$\eta_{hydr} = \mu\,(1 - s) = \mu \cdot \nu.$$

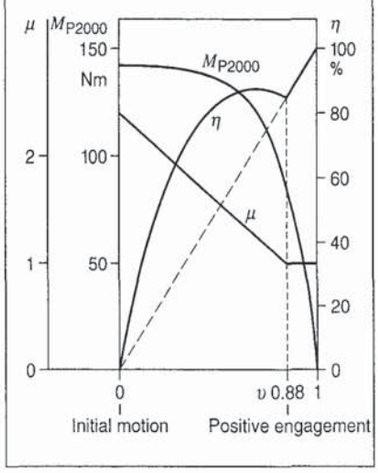
Maximum torque multiplication is achieved at $\nu = 0$, i.e., with the turbine at stall speed. Further increases in turbine speed are accompanied by a virtually linear drop in multiplication until a torque ratio of 1:1 is reached at the coupling point. Above this point the stator, which is housing-mounted with a one-way clutch, freewheels in the flow.

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.

Trilok converter
(typical passenger-car performance curve)
$M_{P2000} = M_P$ at n = 2000 min⁻¹



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.

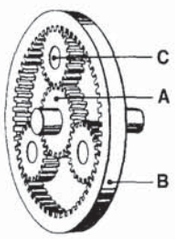
Planetary-gear sets

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.

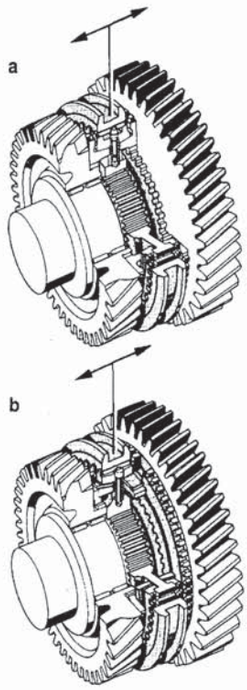
Basic equation for planetary-gear sets: $n_A + (Z_B / Z_A) \cdot n_B - [1 + Z_B / Z_A] \cdot n_C = 0$



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.



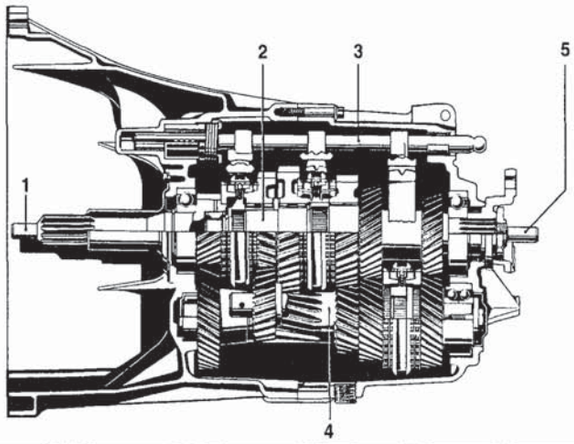
Manually-shifted transmissions

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



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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.

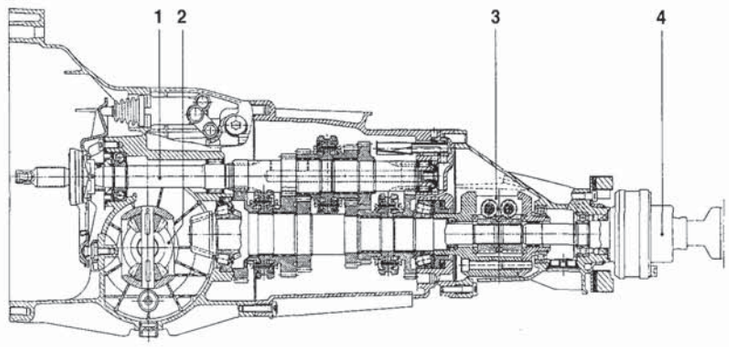
Power take-offs (auxiliary drives)

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

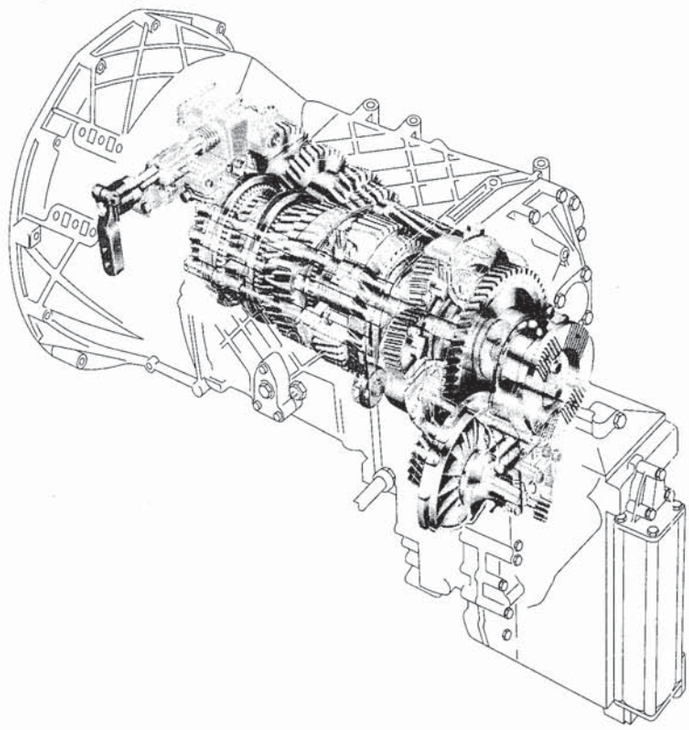
Hydrodynamic or electrodynamic retarders are non-wearing auxiliary brakes for reducing the thermal load on the road-wheel brakes under continuous braking. They can be fitted either on the drive input side (primary retarders) or the drive output side (secondary retarders) integral with the transmission, 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

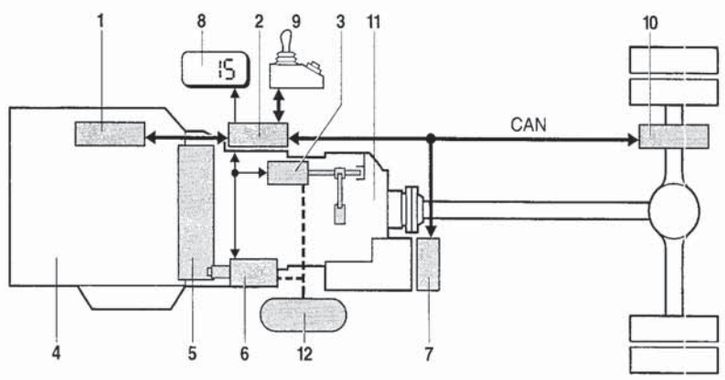


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16-speed multiple transmission with integral retarder for heavy trucks
(ZF Ecosplit 16 S221)



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.

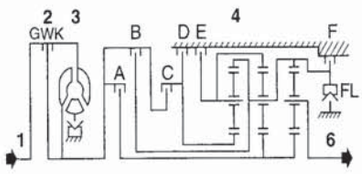
Electronic transmission control

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

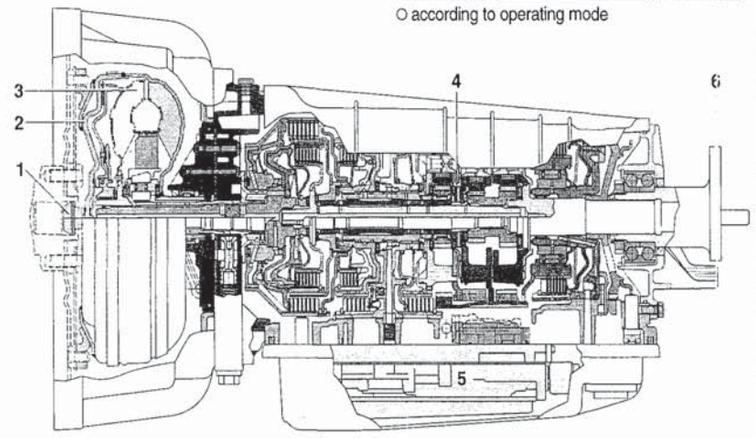
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5-speed passenger-car automatic gearbox *(ZF 5 HP24)*
1 Input shaft, 2 Controlled-slip converter lock up clutch (GWK), 3 Hydrodynamic torque converter, 4 5-speed planetary gears with three multiplate brakes, three multiplate clutches and one one-way clutch, 5 Electronic-hydraulic transmission-shift control, 6 Output shaft

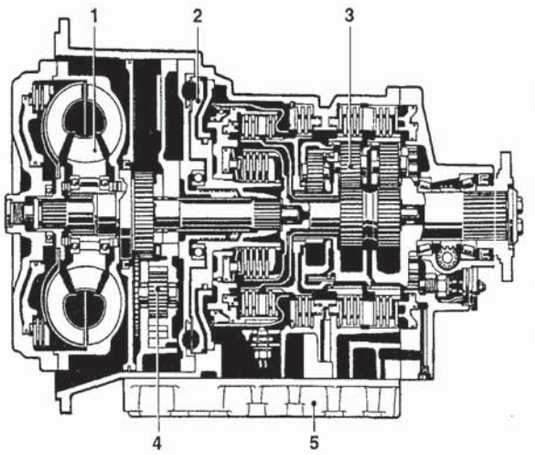


Gear	Clutch			Brake			Overrun	Ratio
	A	B	C	D	E	F	FL	i
1	●					○	●	3.57
2	●				●			2.20
3	●			●				1.51
4	●	●						1.00
5		●		●				0.80
R			●			●		-4.10

○ according to operating mode

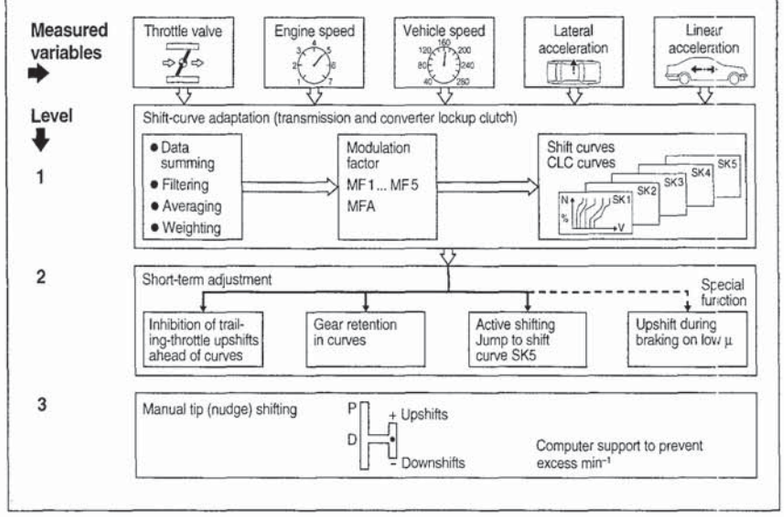


Automatic transmission with integral primary retarder for busses, trucks and special vehicles
(ZF Ecomat 5HP 500).
1 Hydrodynamic torque converter with lockup clutch,
2 Hydrodynamic retarders,
3 5-speed planetary gears,
4 Oil pump,
5 Transmission-shift control.



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Tiptronic® shifting strategies *(process diagram)*



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

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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.

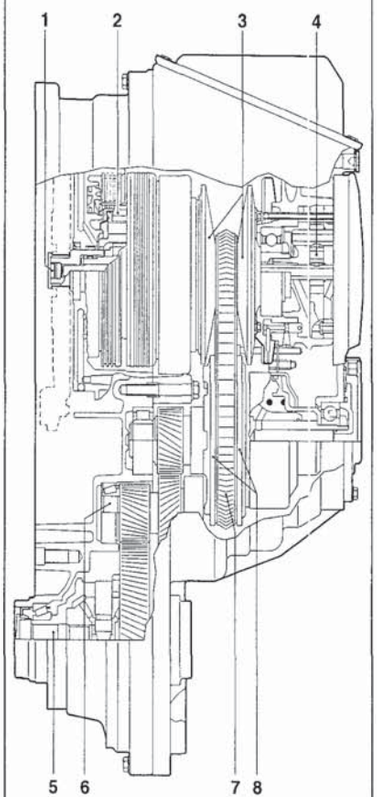
Continuously-variable transmissions

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).

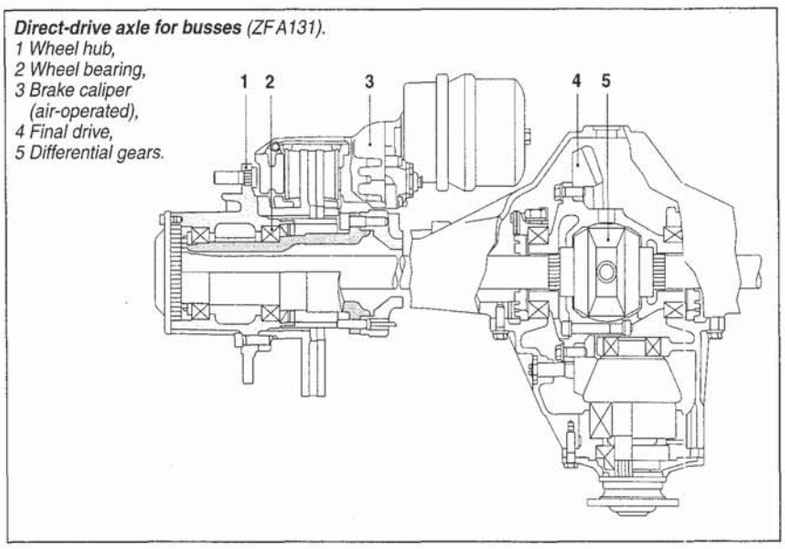
The continuously variable transmission (CVT) can operate mechanically (belt or friction-roller), hydraulically or electrically. The highest level of development has been achieved with mechanical continuously variable units employing steel belts (as installed in low-output series-production cars), where 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.

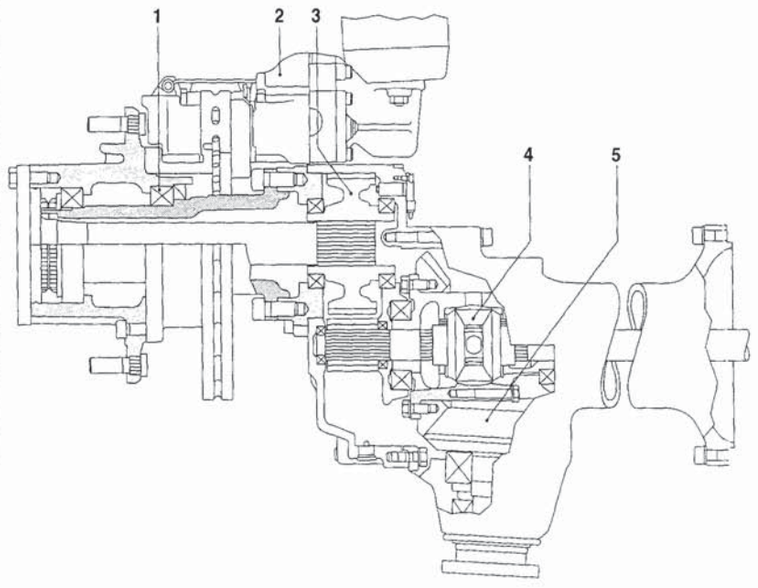
In addition to its torque transmission capabilities, mechanical efficiency and weight, the noise-producing characteristics of a final drive unit have become a decisive criterion in modern automobile development. In this regard, the crown wheel and pinion are of primary significance in terms of noise 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.



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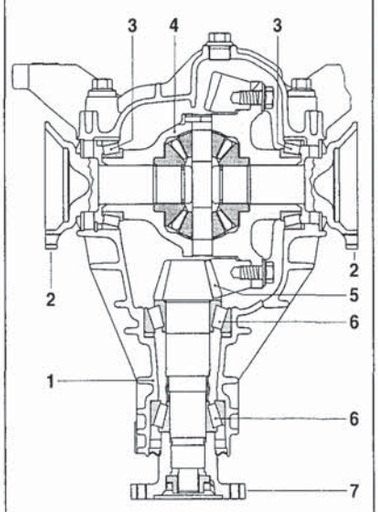
cles), planetary axles are used. Propeller shaft and halfshafts can be made smaller by splitting the power transmission, and the available space is increased.

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.

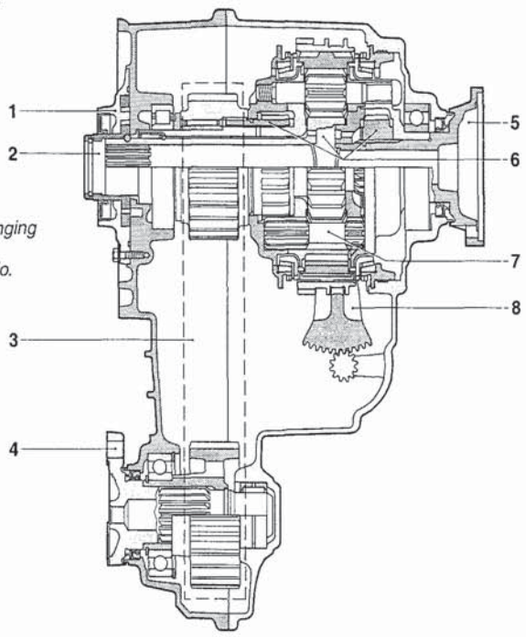
Passenger-car rear-axle final-drive unit (ZF HAG210)
1 Housing (two-piece),
2 Halfshaft flange,
3 Differential-case bearings,
4 Differential gears,
5 Crownwheel and pinion,
6 Pinion bearing,
7 Propeller-shaft flange.



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)).

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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.

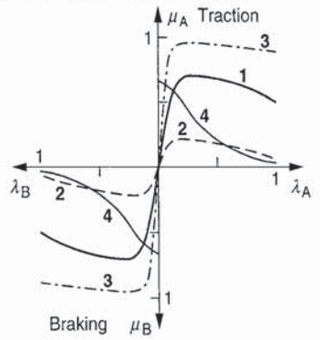
ABS responds to the first case (braking) by inhibiting wheel lock. ASR reacts to the second scenario by holding acceleration slip within acceptable levels to prevent wheel spin. The system actually performs two functions:
– enhancing traction
– maintaining vehicle stability (true tracking).

Traction-control systems for passenger cars

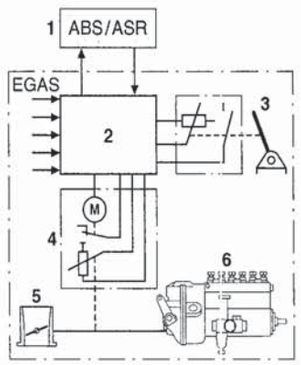
ABS/TCS 2I (Bosch)

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.



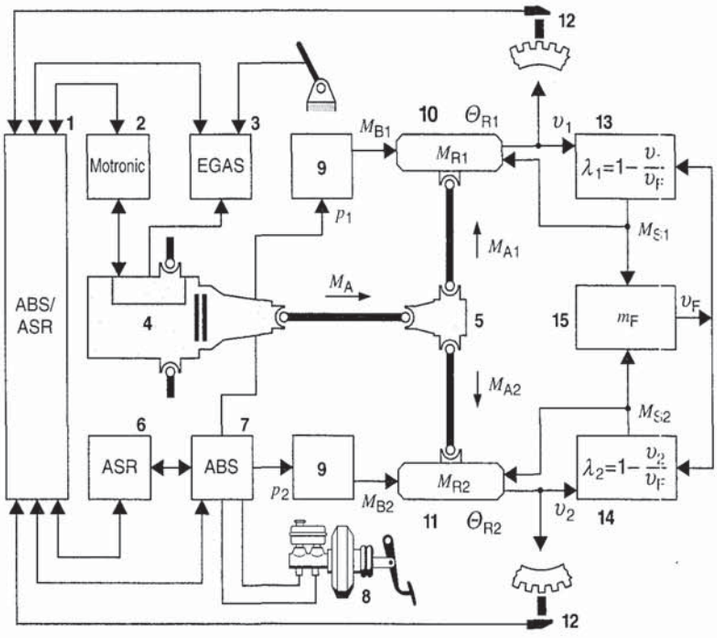
EMS electronic engine-power control for TCS
1 ABS/TCS control unit, 2 ETC control unit, 3 Accelerator pedal, 4 Servomotor, 5 Throttle valve, or 6 Diesel injection pump.



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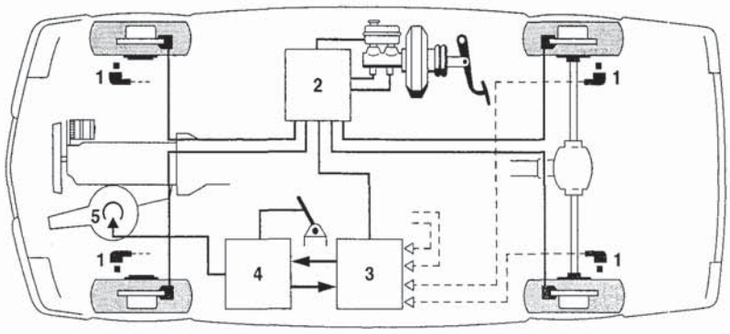
ABS/TCS schematic diagram for cars

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.



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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.

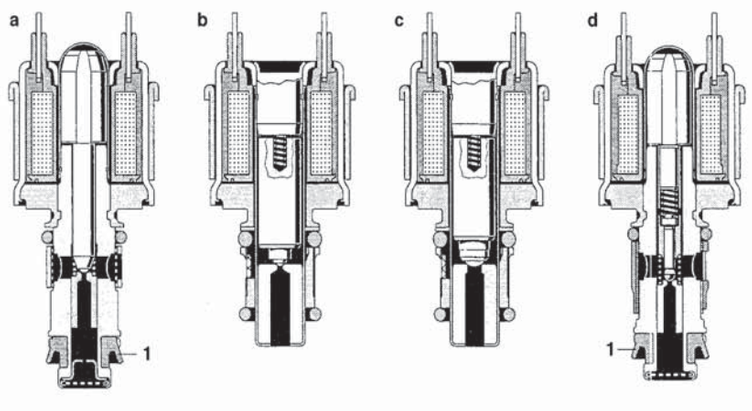
The engine-control functions remain largely independent of the hydraulic system, and the current design was adopted for ABS/TCS5 without modification.

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.

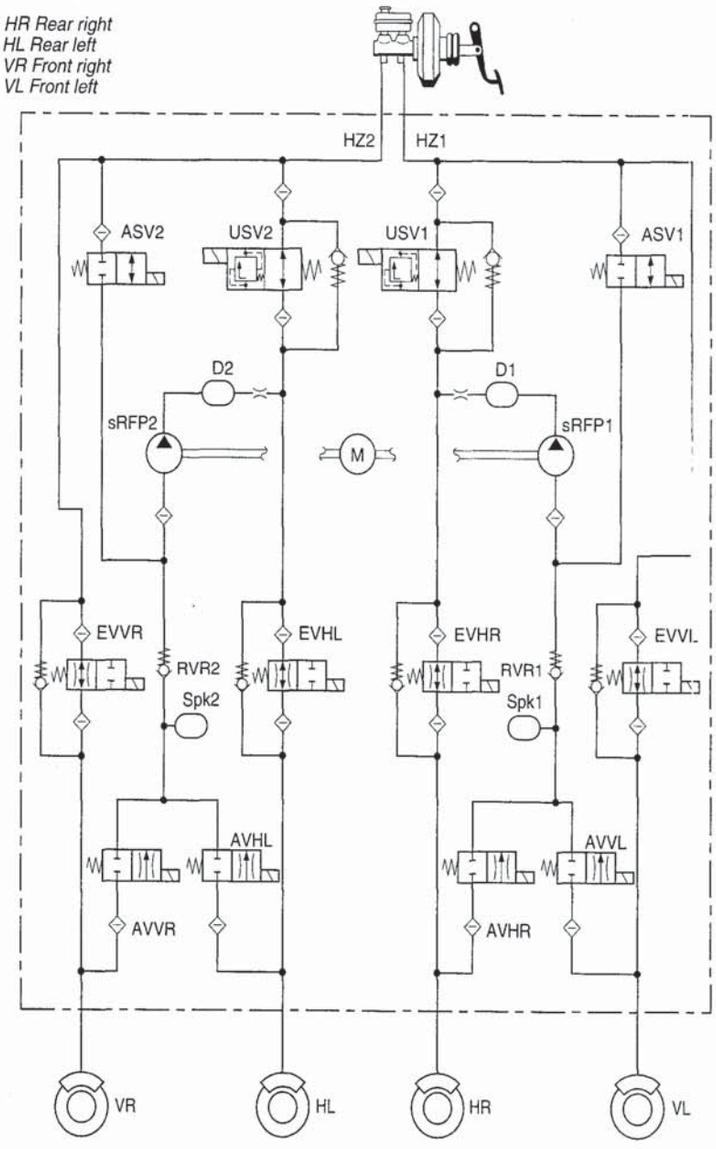


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Schematic diagram of ABS/TCS5 hydraulics for X brake-circuit configuration

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



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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.

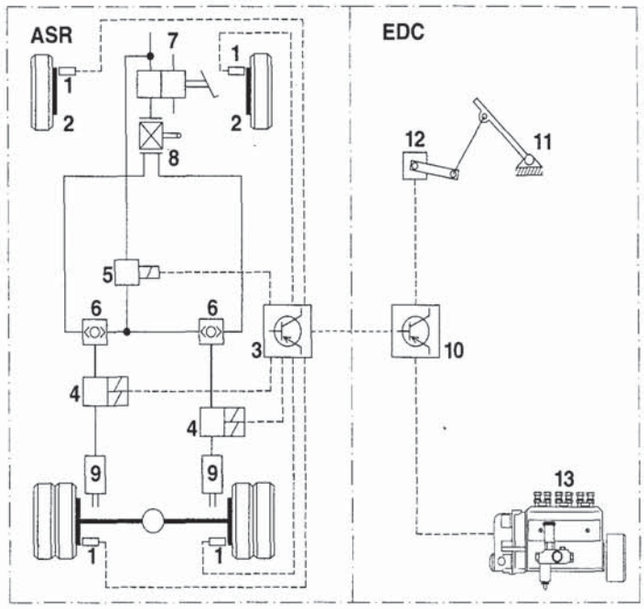
The following section illustrates the principles involved by describing two options for electric control:

- 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.

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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.

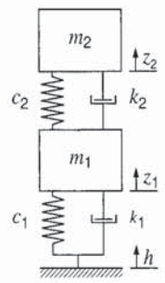


Table 1.
Effects of design parameters on the vehicle's vertical oscillations

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

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the degree of body oscillation. Root mean square of vertical body acceleration:

$$\vec{\ddot{z}}_2 / \vec{h}$$

Axle oscillations (and thus, indirectly, fluctuations in wheel load) are the salient factor in determining driving safety. Root mean square of vertical axle oscillations:

$$\vec{\ddot{z}}_1 / \vec{h}$$

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

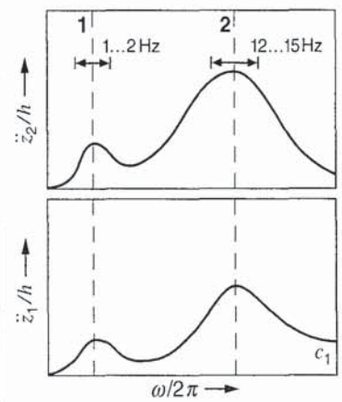
Partially loaded systems

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

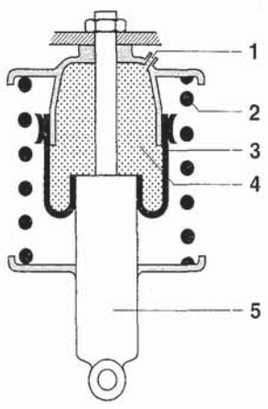
Effect of frequency on kinematic amplitudes

$\ddot{z}_1$ 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.

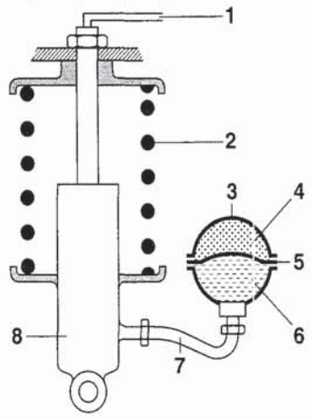


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system can also incorporate intermediate electronic level-control units acting upon solenoid valves.
Advantages of the electronic system:
- Reduced energy consumption achieved by avoiding control cycles during braking, acceleration and in curves,
- 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.

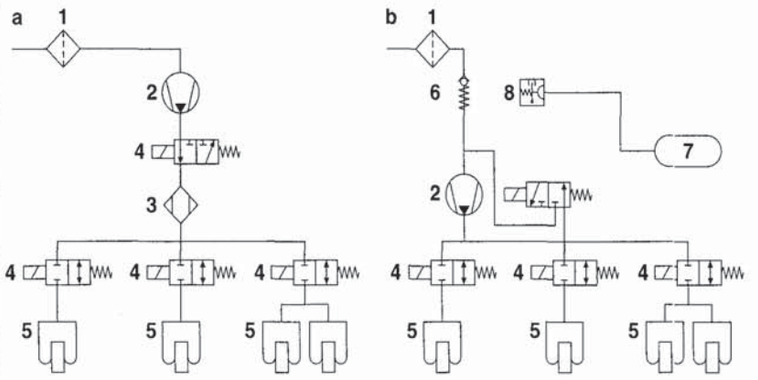
Additional advantages for commercial vehicles:
- Automatic limitation of suspension travel in order to accomodate interchangeable bodies and containers,
- Vehicle-body height can be adjusted as desired, e.g., to align load floor with loading ramps,
- Control of lifting axles (ride height raised automatically 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

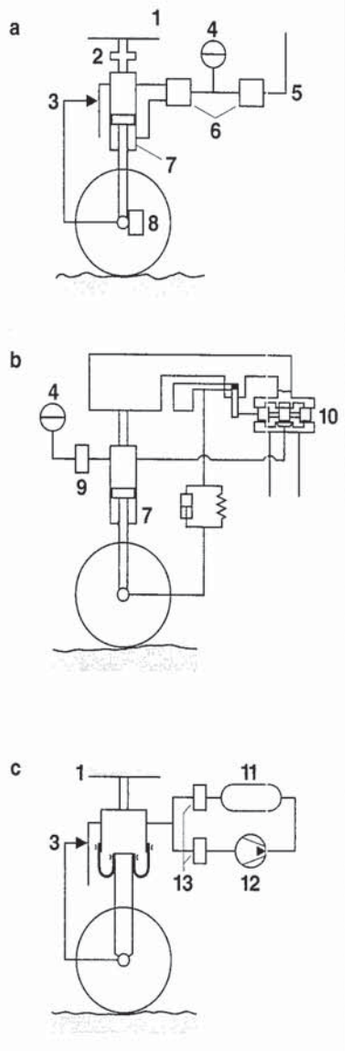
Active suspension controls both the "springing" and the "damping" functions. Various types of design concepts have been realized.

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.

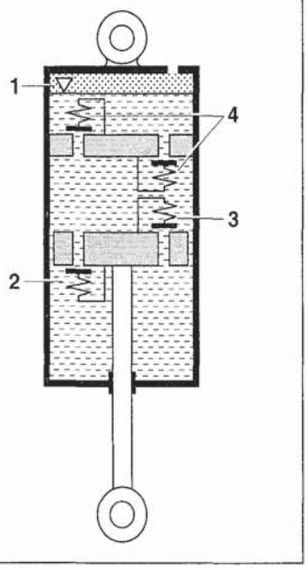
Single-tube shock absorbers

A sliding separating-piston and gas cushion form the gas-pressure damper in the single-tube shock absorber.

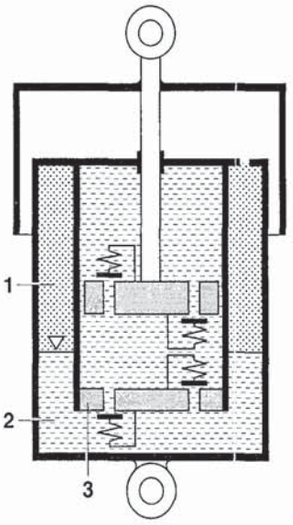
Advantages: Easy to tailor to specific applications, as the large piston diameter allows low working pressures. Sufficient room for valves and passages. Heat is dissipated directly via the outer tube. The shock absorber can be installed in any position.

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.



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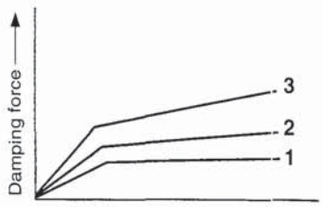
Twin-tube shock absorber

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.



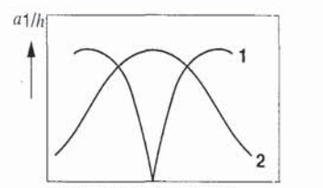
Damping characteristics

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.

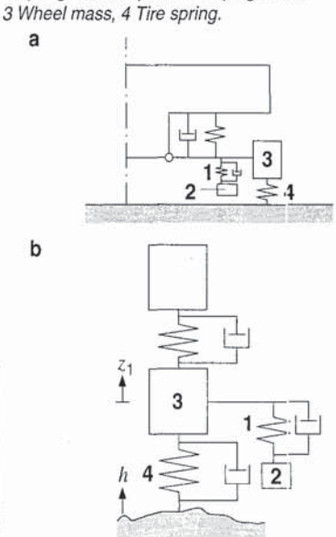


Vibration absorber

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 Toroid bellows	Element with constant gas volume. *1 Chassis frame, 2 Roll-gaiter air spring, 3 Piston, 4 Compressed air, 5 Mounting plate.* Roll bellows Toroid bellows 	$\frac{\nu_{loaded}}{\nu_{empty}} = 1$ 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.

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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.

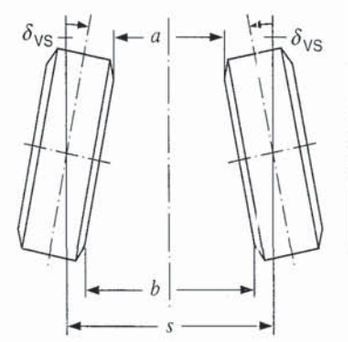
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Suspension linkage

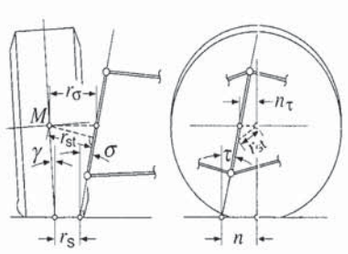
The suspension layout employed to connect the individual wheels to the vehicle structure defines the suspension-linkage design. The linkage enables the wheel for the most part to move vertically in order to compensate for irregularities in the road surface. In addition, the front wheels are steered, 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.

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.



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.

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-

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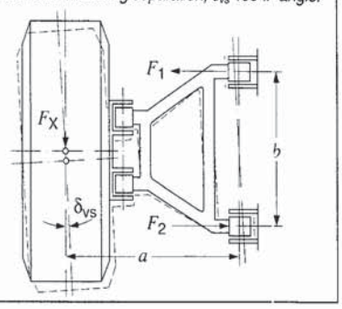
ences the camber change as a function of the steering angle, as well as the self-centering characteristics of the steering.

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.

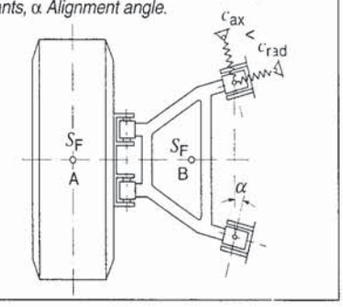
Camber angle γ
Camber angle is the inclination of the wheel axis towards the road surface in the vertical plane. The camber angle is negative when the top of the wheel is inclined toward the center of the vehicle. It influences lateral control (today's vehicles generally employ moderate cambers of –3...0° 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	

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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

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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.

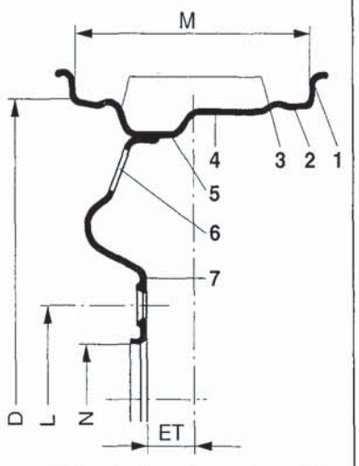
Rim designs
Rims differ from one another (depending upon the type of tire) in terms of number of parts and cross-sectional rim shape. The most important rim details are: rim flange, rim bead seat and rim base. Rims have one of the following cross-sectional shapes:
- Drop center,
- Flat base,
- 5° tapered 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.

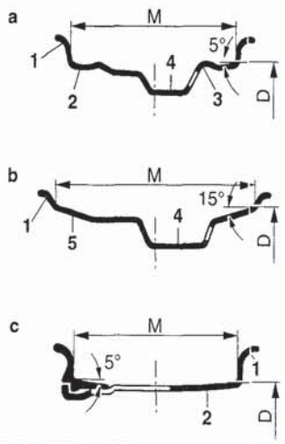
Passenger-car wheels

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 tug 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.

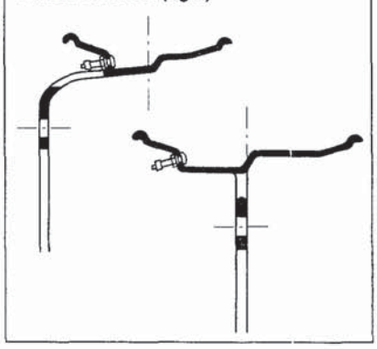
Radial and lateral run-out, imbalance
Radial run-out of the wheel is one of the main causes of vehicle vibrations. With today's commonly used 15° tapered-bead seat, the reduction of the permissible radial run-out to 1.25 mm (peak-to-peak) results in a marked improvement compared to the flat-bed rims. 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.

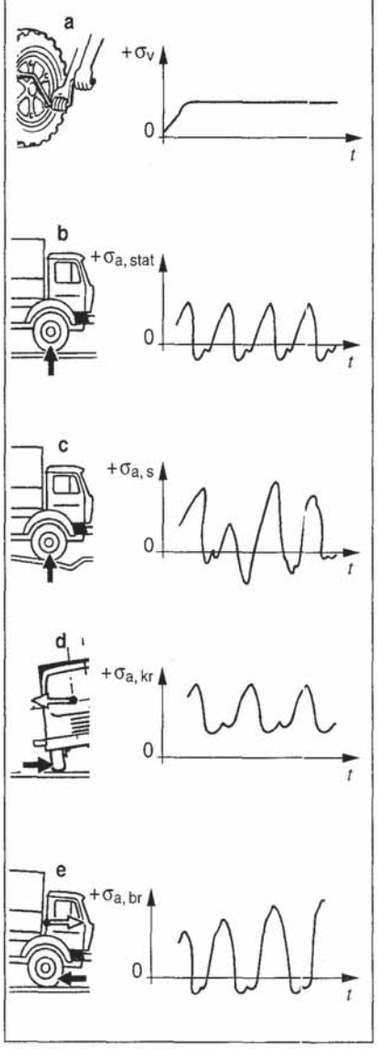
The resulting group of stresses caused by the above-mentioned types of wheel loading is used today as the basis for wheel dimensioning and testing.

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 $\sigma_{a,br}$



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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-

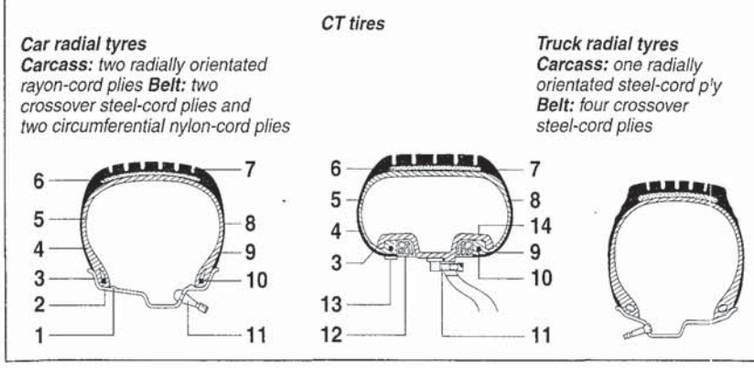
[1]) The corresponding European standards can be found in the "Data Book of Tires and Rims" of the ETRTO (European Tire and Rim Technical Organization, Brussels).
[2]) Guideline of the Economic Association of the German Rubber Industry, Frankfurt.

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.



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 cm^3 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	2 1/4–16 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 (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).

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.

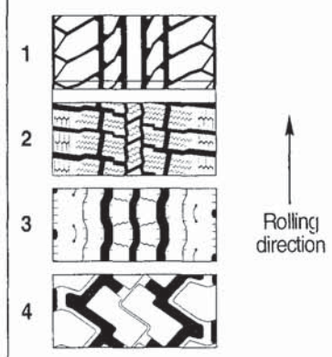


Table 6. Tread depth and braking distances (braking from 100 km/h)

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	

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Tire traction

Units and symbols

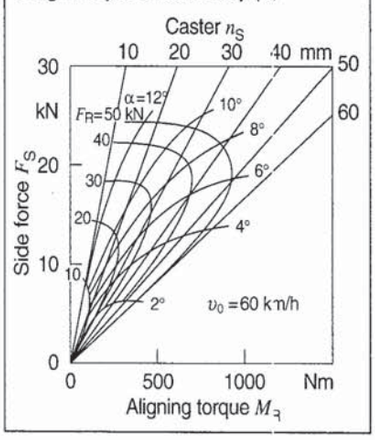
Symbol		Unit
f	Frequency	Hz
F_B	Braking force	kN
F_R	Wheel load	kN
F_S	Side force	kN
M_R	Aligning torque	N · m
n_S	Caster	mm
p_i	Tire pressure	bar
v_0	Test velocity	km/h
α	Slip angle	°
γ	Camber angle	°
λ	Slip	–

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.

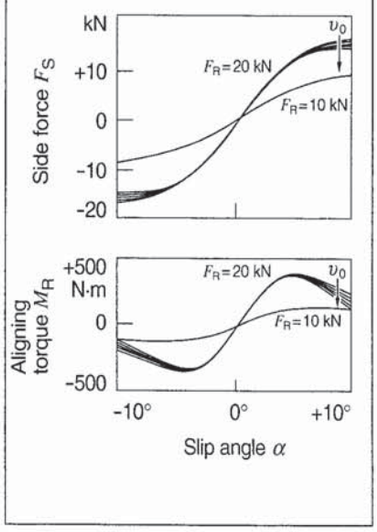
Representative tire-performance maps for passenger-cars and light commercial vehicles are already familiar and are available in various publications [1, 3, 4]. For this reason, the following will concentrate on tires for heavy-duty commercial vehicles with the dimensions 11 R 22.5. These tires are in widespread use [2].

All data in the performance maps refer to the Michelin XZA 11 R 22.5 tire.

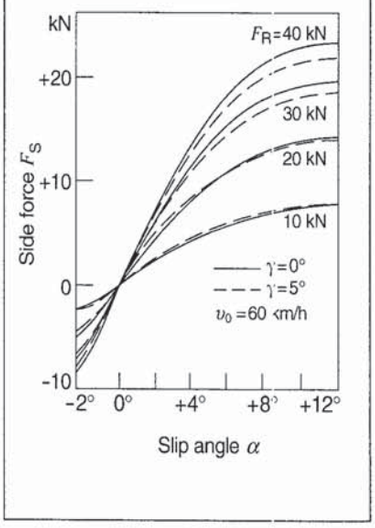
***Gough tire-performance map** (B)*



***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.



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Tire performance with wheel rotating freely at a given slip angle

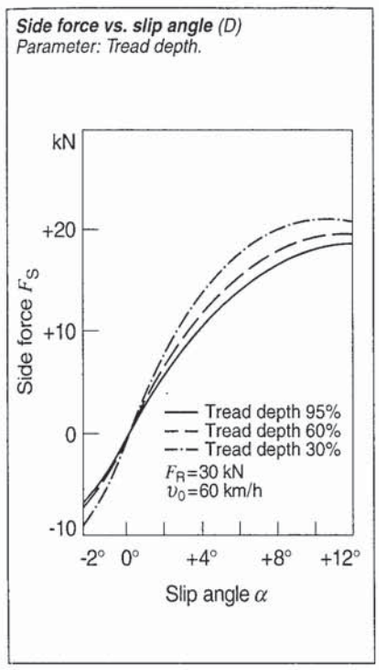
When the tire rotates at a slip angle, side forces are generated as a function of the slip angle. These side forces are accompanied by an aligning torque (Diagram B). The Gough diagram [3] is frequently employed to illustrate this phenomenon. The side force is dependent on the slip angle and increases 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).

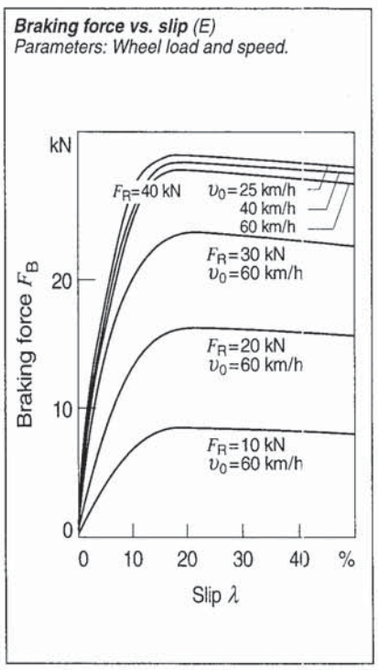
On dry roads, reduced tread depth results in steeper side-force/slip-angle curves, with an accompanying increase in the maximum side forces which can be transferred (Diagram D).

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.

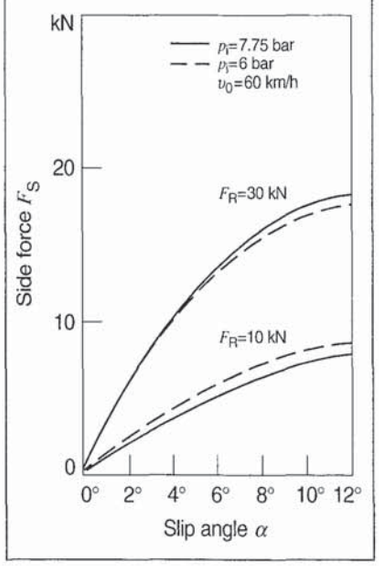
The tire-performance curves resulting from measurements actually taken, indicate the propagation of side force as a function of braking force within the slip-angle range of 0...10°. The parameters for wheel load, speed and tire pressure remain constant (Diagram K).

Dynamic tire-performance curves

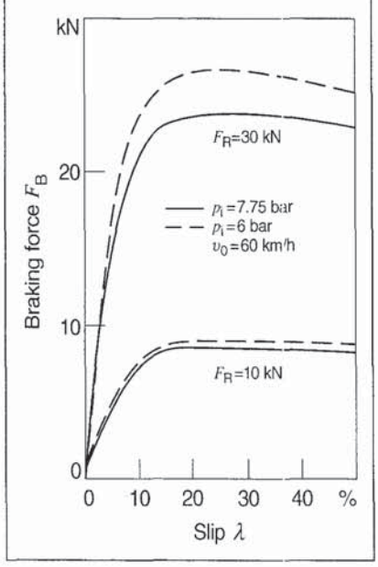
The illustrated tire-performance curves are based on parameters which are only subject to gradual change during the course of the measurements, i.e. quasi-static conditions. Actual operation, on the other hand, is characterized by dynamic processes. Increases in the speed at which the influencing parameters 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.

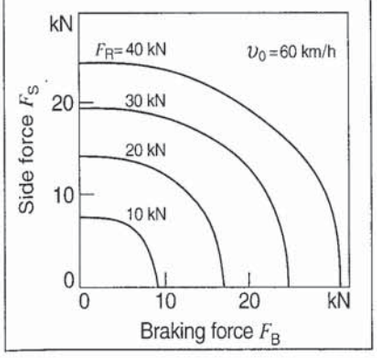


***Braking force vs. slip on heavy-duty commercial-vehicle tires** (G)*
Parameters: Tire pressure and wheel load.

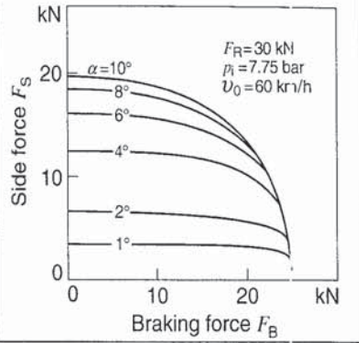


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Lateral force vs. braking force *(H)*
Maximum achievable adhesion at constant wheel load.



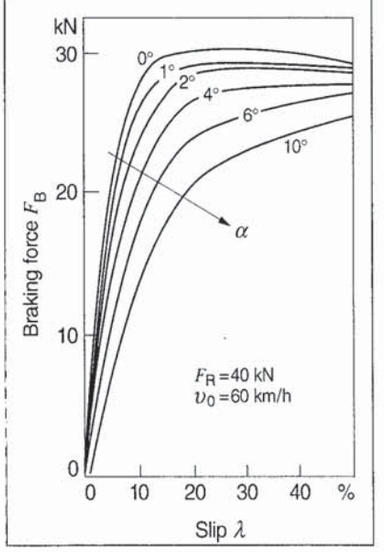
Tire-performance curves measured at a wheel load of 30 kN *(K)*
Side force as function of braking force, Parameter: Slip angle.



– 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.



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[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

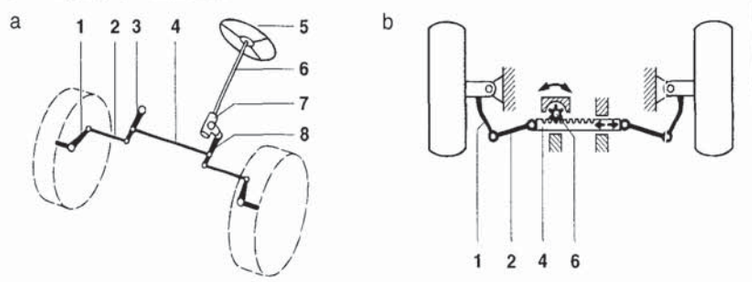
The requirements in terms of steering behavior can be summarized as follows:
1. Jolts from irregularities in the road surface must be damped as much as possible during transmission to the steering wheel. However, such damping must not cause the driver to lose contact with the road.
2. The basic design 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).

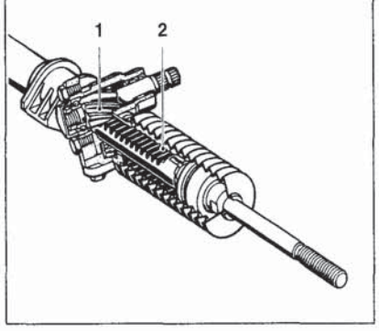
The steering-axis inclination causes the front section of the vehicle to lift when the wheels are at an angle. This leads to a steering-angle-dependent caster return torque.

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.

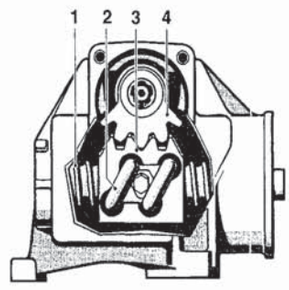
The caster produces a lever arm for lateral forces: i.e., a speed-dependent return torque.

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.

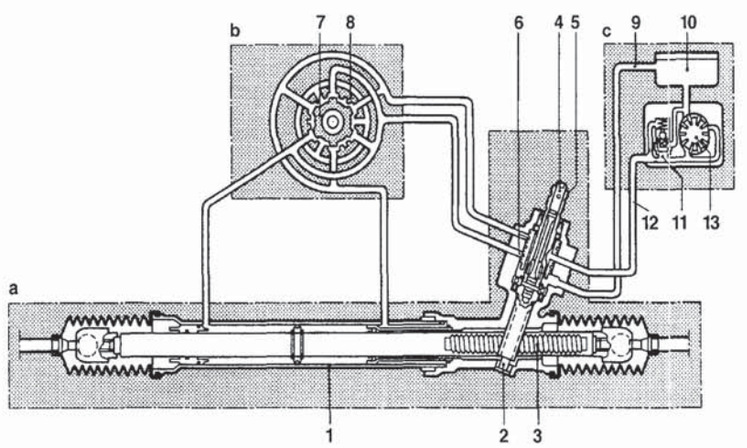
The pump and the system components must be designed such that the operating temperature of the hydraulic fluid does not rise to an excessive level (<100 °C) and such that no noise is generated and the oil does not foam.

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.



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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.

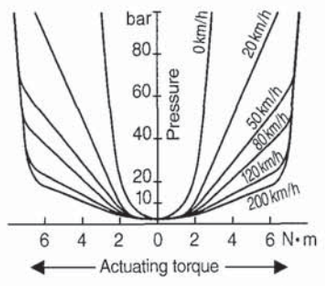
Steering cylinder

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.

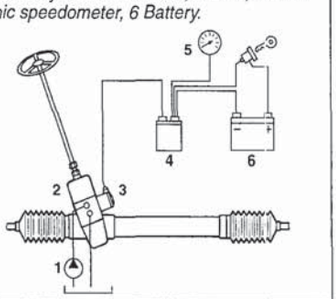
Parameterizable power-assisted steering *(characteristic curves)*
The characteristic curve form can be adjusted to the vehicle data.



Electric power-assisted steering

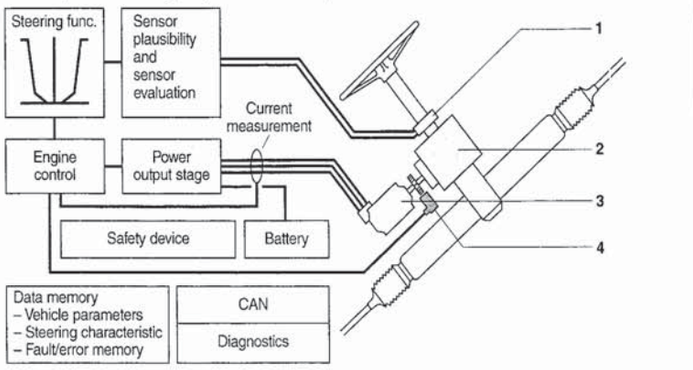
The SERVOLECTRIC (Bosch/ZF designation) is currently under development for medium-sized and smaller cars. This electric power-assisted steering system features an electric motor which is powered by the vehicle's electrical system and can be fitted in one of three possible arrangements either as a steering-column, 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.

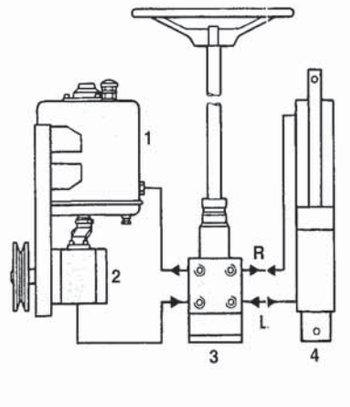
The maximum permissible speed is 25 km/h in many European countries; in Germany it is 50 km/h and, with dual-circuit design, could be increased to 62 km/h.

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 cylinder. Connections: L Oil flow for left lock, R Oil flow for right lock



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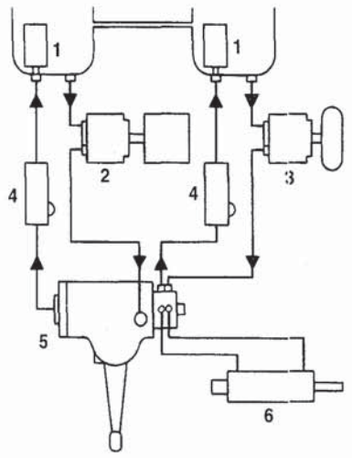
ing circuits of such a system are functionally tested by means of flow indicators. The steering-circuit supply pumps must be driven in differing ways (e.g., engine-dependent, vehicle-speed-dependent or electrically). In accordance with legislation, if the engine or one of the steering circuits fails, the vehicle can be steered with the still functioning circuit.

Single-circuit power-assisted steering system for commercial vehicles

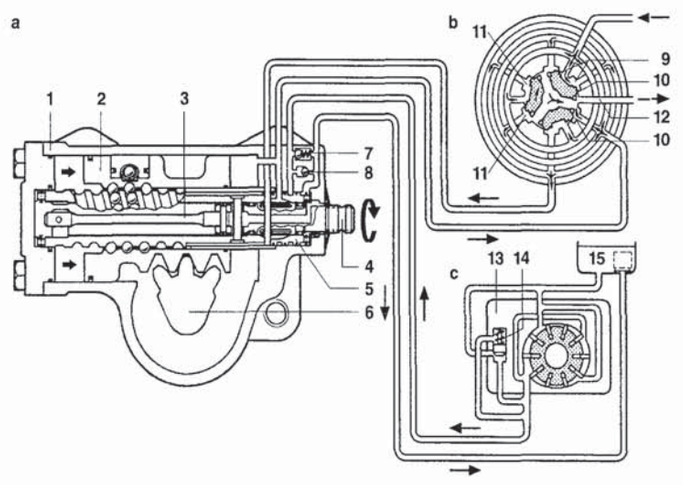
Commercial vehicles are usually fitted with recirculating-ball power steering. In modern systems the control valve is integrated into the steering worm, which leads to compact design and optimum weight.

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.



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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.

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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.

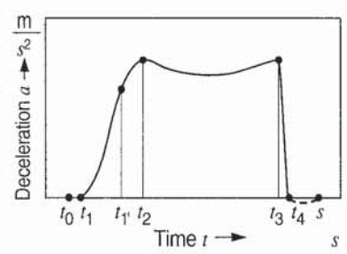
Application force F_s: In friction brakes, the total force applied to a brake lining, which causes the braking force by friction effect.

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_{mf}) 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		$z \geq 0.50$, 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 $v_2 = {}^1/_2\, v_1$	≤km/h	120	100	60	120	60	60				
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.

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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.

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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.

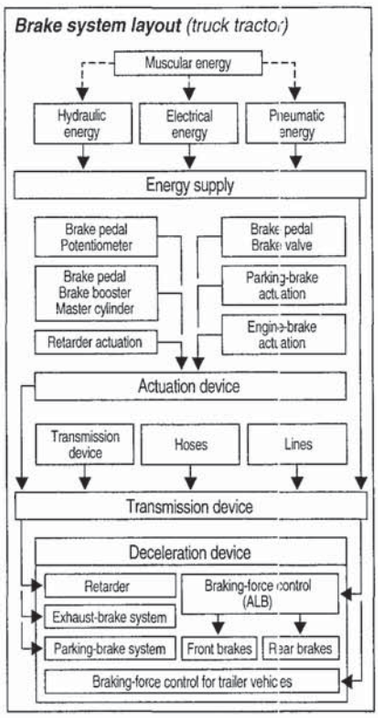
Different ranges of applications for different vehicle types result in highly variegated demands being placed on braking systems. The inescapable result is a multiplicity of highly diversified braking systems, differing from one another in both design and application.

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.

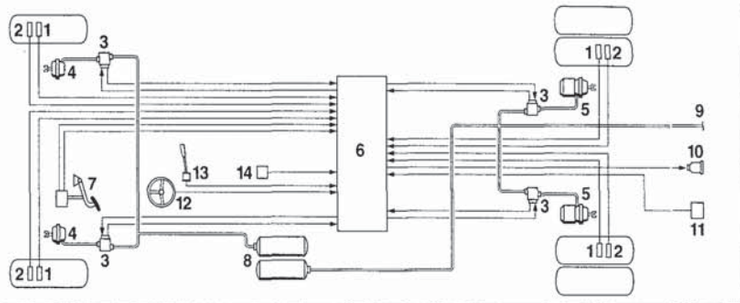
Energy-assisted and non-muscular-energy systems are distinguished not only by energy type, but also by the mediums used to transmit the energy. Pneumatic (vacuum, compressed air) and hydraulic energy are the most common, electrical energy is sometimes also employed.

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

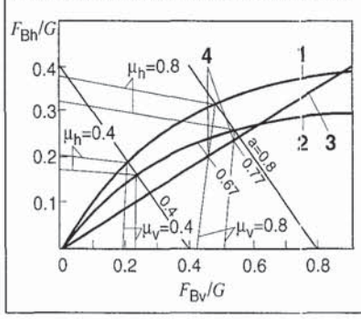
The braking system is designed with reference to both the requirements of the vehicle and the intrinsic imperatives of the system itself. In the case of the vehicle-oriented design, the vehicle's center of gravity and the specified distribution of braking force to the front and rear axles, determine 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}.



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- 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.

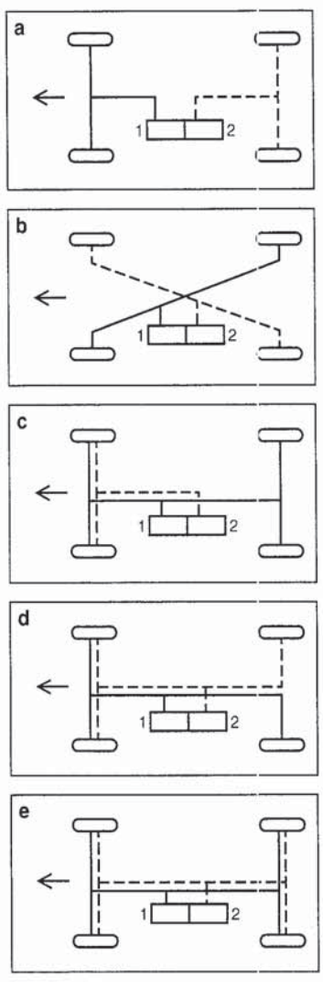
Brake-circuit configurations

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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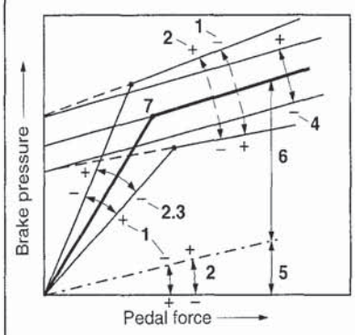
X distribution pattern
Diagonal distribution pattern. Each circuit brakes a given front wheel and the diagonally opposite rear wheel.

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.

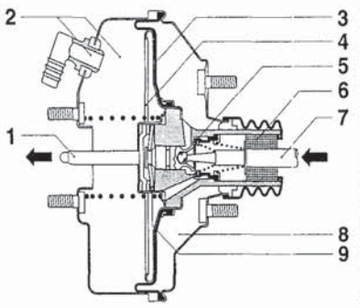
LL distribution pattern
Front-axle and rear-wheel/front-axle and rear-wheel split. Each circuit brakes the front axle and one rear wheel.

HH distribution pattern
Front-axle and rear-axle/front-axle and rear-axle split. Each circuit brakes the front axle and the rear axle.

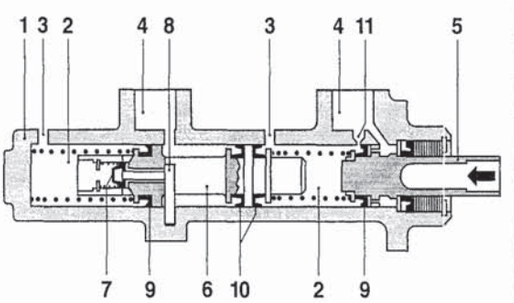
Vacuum-operated brake booster
Effect of design parameters.
1 Master-cylinder surface area, 2 Pedal leverage, 3 Boost factor, pedal leverage, 4 Diaphragm surface area, vacuum level, 5 Effect of pedal force, 6 Effect of servo assist, 7 Output point.



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.

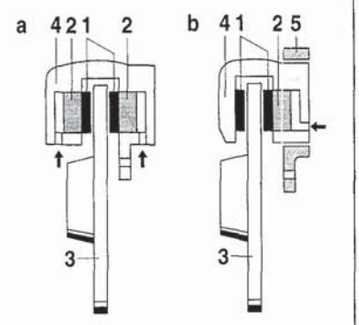
Wheel brakes

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.

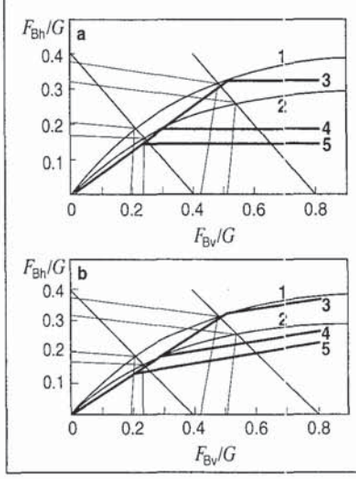
The braking force is apportioned between the front and rear axles in accordance with the dimensions of the particular brakes. It is the job of the metering device to adjust this braking-force apportionment in order to achieve a closer approximation to the ideal distribution, i.e., the parabolic curve. 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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