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

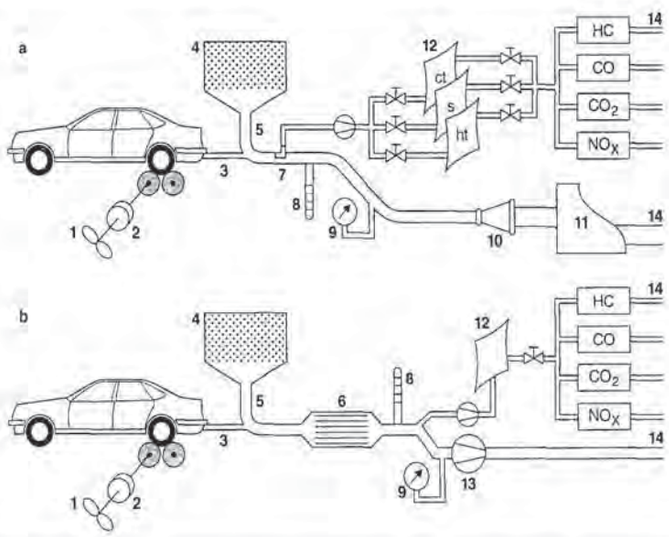
Advantages of this procedure: Condensation of the water vapor contained in the exhaust gas is avoided, which would otherwise cause a substantial reduction in the NO_X losses in the bag. In addition, dilution greatly inhibits the tendency of the exhaust components (especially hydrocarbons) to continue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

Dilution systems

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

Test setups

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



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Determining evaporative losses from the fuel system

(Evaporation tests)

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

SHED Test

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

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

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

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

ECE/EC test cycle and limits

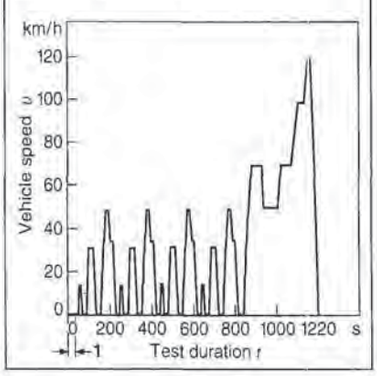
The ECE/EC test cycle incorporates a synthetically generated driving curve (see illustration) calculated to provide a reasonable approximation of driver behaviour in urban traffic. Since 1993, the test cycle has been supplemented by a suburban section with speeds of up to 120 km/h. This new ECE/EC test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

The exhaust emissions test is conducted as follows:

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

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

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



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

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

Test cycles in the USA

FTP 75 test cycle

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

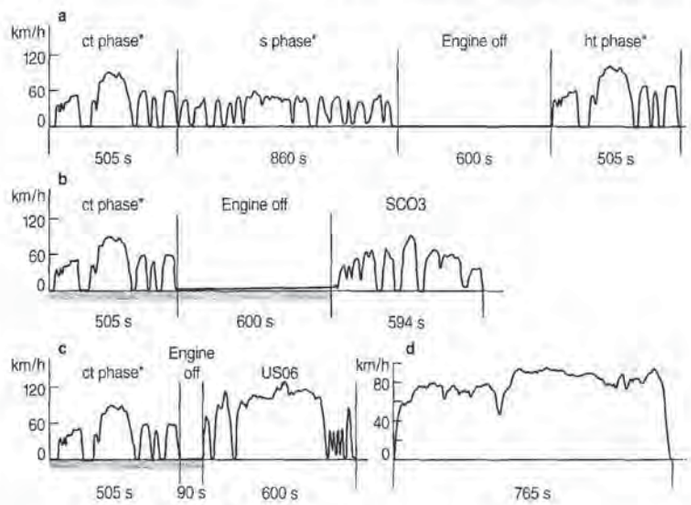
The vehicle to be tested is first conditioned (allowed to stand with engine off for 12 hours at a room temperature of 20...30 °C), then started and run through the prescribed test cycle:

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

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

USA test cycles

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



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

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

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

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

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

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

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

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

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

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

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

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

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

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

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However, a limit is prescribed for carbon monoxide only.

Japanese test cycle

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

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

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

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

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

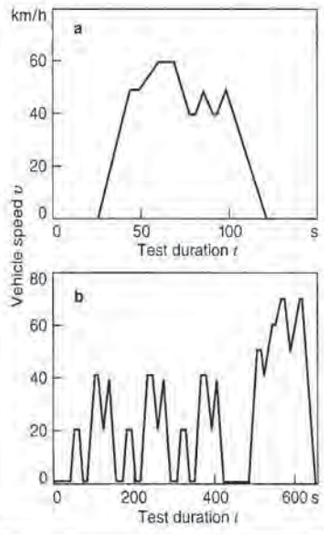
Japanese test cycles

a 11-mode cycle (cold test)

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

b 10·15-mode cycle (hot test)

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



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Exhaust-gas analyzers

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

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

Test procedure

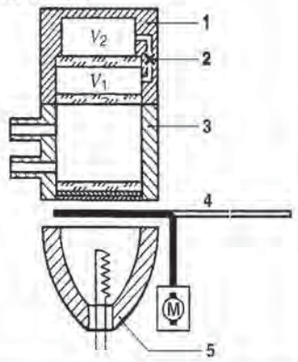
In automotive service operations, the infrared method has proved to be the only efficient and reliable means for testing exhaust gases.

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

Test chamber

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

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



Catalytic-converter testing

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

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Current (1998) emissions limits for gasoline engines

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

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

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

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

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

Table 3
US emissions limits. STFP test cycle

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

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

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

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

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

Table 5
Japanese emissions limits measured in Japanese test cycle

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

( ) planned figures

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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.

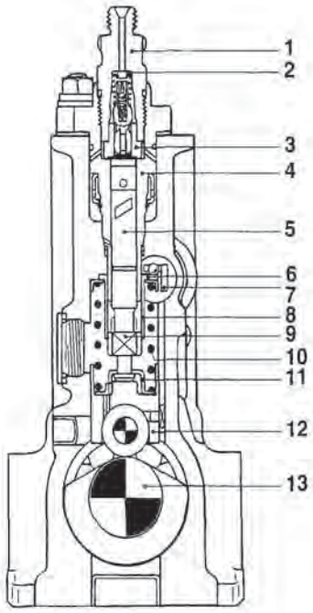
Common to all of these systems is the fact that the required injection pressure is generated at the moment each injection occurs. In the case of electronically controlled systems, however, the rate-of-discharge curve can be influenced not only by the cam contour but also by solenoid valve actuation.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

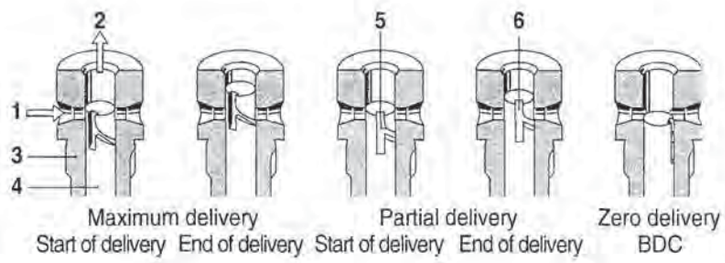
A piston pump delivers the fuel to the injection pump's fuel gallery at a pressure of 1...2.5 bar. The cam-driven supply-pump plunger travels to TDC on every stroke. It is not rigidly connected to the drive element; instead, a spring supplies the return pressure. The plunger spring responds to increases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

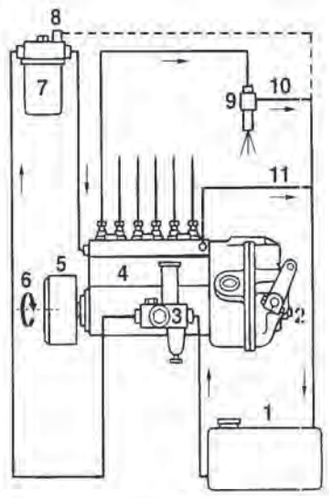
The delivery valve and pressure-relief characteristics must be specially designed for the specific application. Units incorporating a return-flow restriction or constant-pressure valve have an additional throttle element to damp the pressure waves reflected back from the injection nozzle, thus preventing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

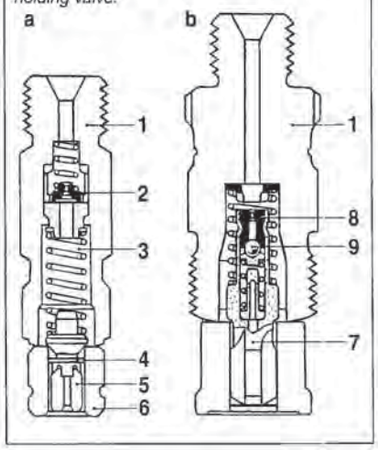
In pumps which generate injection pressures greater than approx. 600 bar, the plunger-and-barrel assembly, delivery valve and delivery-valve holder are screwed together to form a single unit, which means that the high sealing forces must no longer be accommodated by the pump housing (e.g. Sizes MW, P).

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.



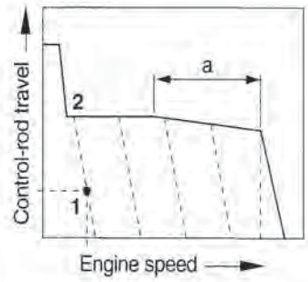
Delivery-valve holder with delivery valve
a) With constant-volume valve and return-flow restriction, b) With constant-pressure valve. 1 Delivery-valve holder, 2 Return-flow restriction, 3 Dead volume, 4 Retraction piston, 5 Valve ball, 6 Valve holder, 7 Supply valve, 8 Calibrated restriction, 9 Pressure-holding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



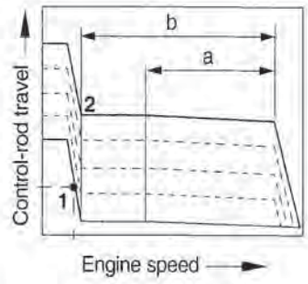
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Governor characteristic curves
a Positive torque control in upper speed range, b Unregulated range, c Negative torque control.
1 Idle-speed setpoint, 2 Full-load curve, 3 Full-load curve, turbocharged engine, 4 Full-load curve, naturally-aspirated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

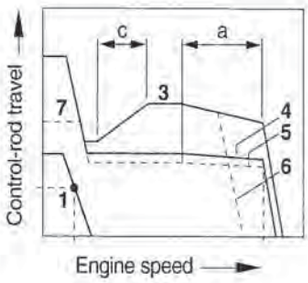
Variable-speed governor



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.

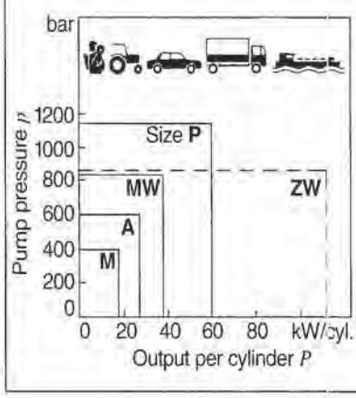
Speed droop

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

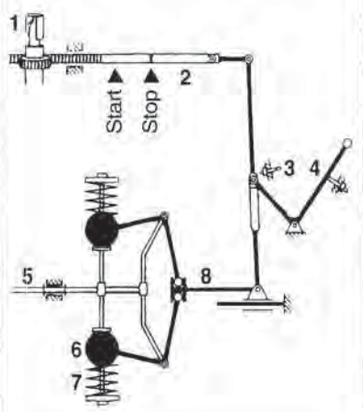
$$\delta = \frac{n_{LO} - n_{VO}}{n_{VO}} \cdot 100\,\%$$

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 %.

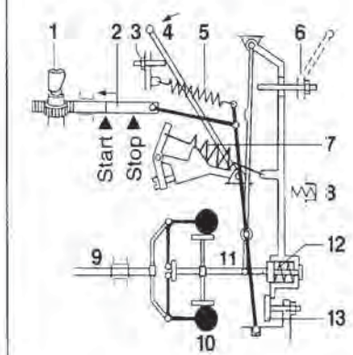
Applications for various types of in-line injection pump



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.



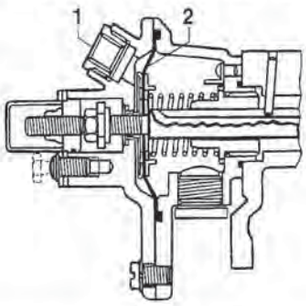
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Mechanical add-on equipment

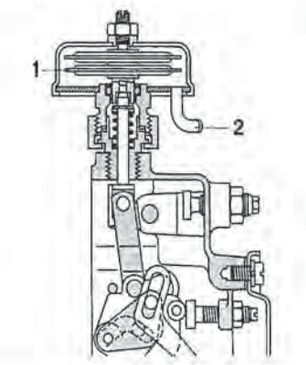
Torque control

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.

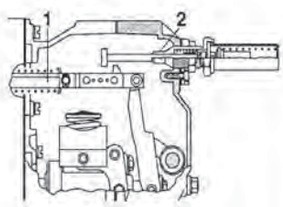
The ECU microprocessor compares accelerator position, min^{-1}, and a number of additional correction factors with the program maps stored in its memory in order to determine the correct injected fuel quantity, i.e. the correct rack position.

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.

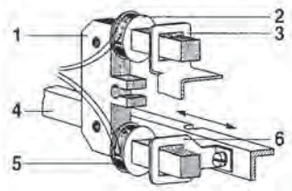
An inductive speed sensor monitors a camshaft-mounted pulse wheel; the ECU uses the pulse intervals to calculate engine speed.

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,

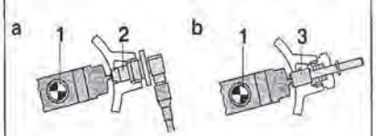
Temperature-dependent starting device
1 Control rack, 2 Start-quantity stop with expansion element.



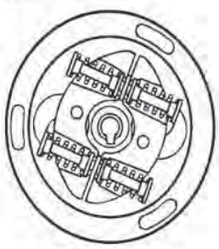
Rack-travel sensor (RWG)
1 Iron core, 2 Reference coil, 3 Fixed short-circuiting ring, 4 Control rack, 5 Measuring coil, 6 Moving short-circuiting ring



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

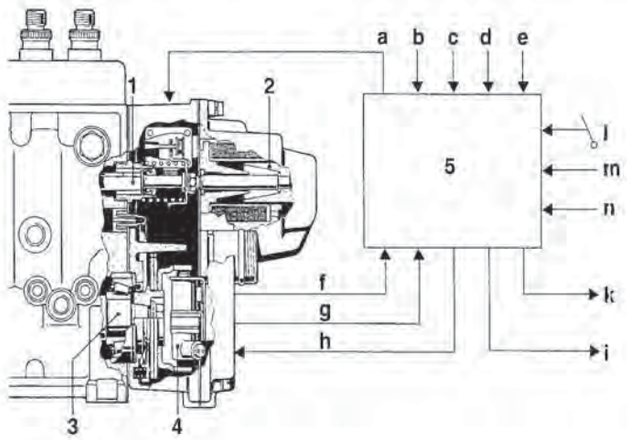
The in-line control-sleeve injection pump makes it possible to provide electronically-controlled adjustment of port closing (start of pump delivery). The spill port, which on conventional in-line pumps is in the housing and therefore immovable, is incorporated in the control sleeve which is a component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



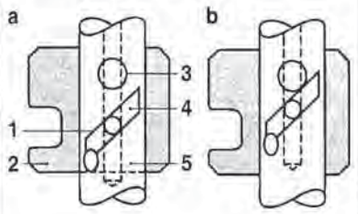
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Distributor-type fuel-injection pump (VE)

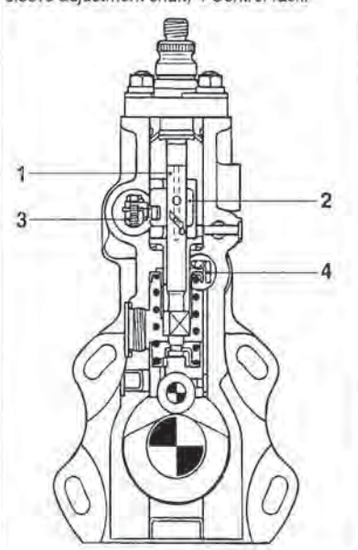
Distributor-type fuel-injection pumps are used in 3-, 4-, 5- and 6-cylinder diesel engines in passenger cars, tractors and light- and medium-duty commercial vehicles which generate up to 42 kW per cylinder, depending on engine speed and combustion system. Distributor-type fuel-injection pumps for direct-injection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

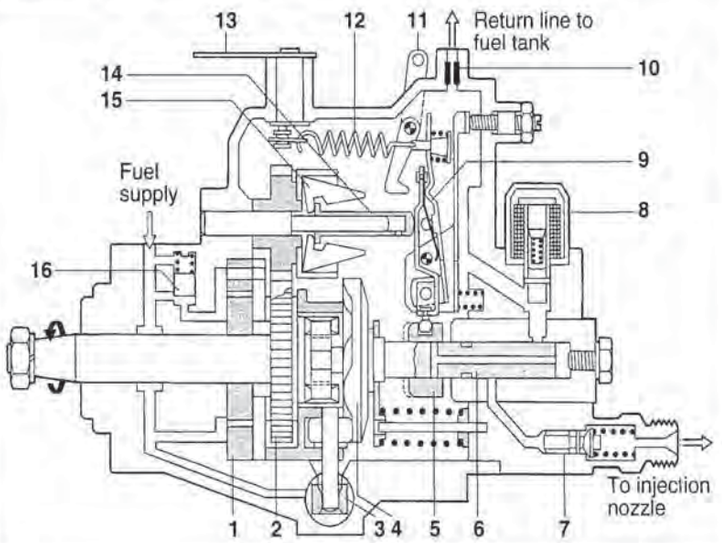
Add-on modules:
A number of add-on modules are available to process additional operating parameters for regulation of delivery quantity (such as manifold-pressure compensator (LDA), start quantity, hydraulic and mechanical full-load torque control) and for adjustment of the start of delivery – (e. g.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

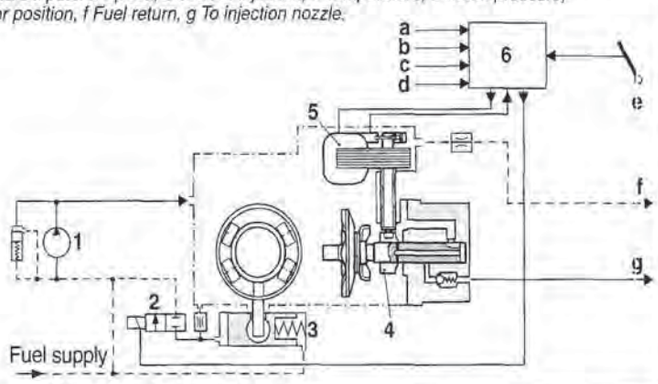
Electronic Diesel Control (EDC): Distributor-type fuel-injection pump with rotating-solenoid actuator
In contrast to the mechanically controlled

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

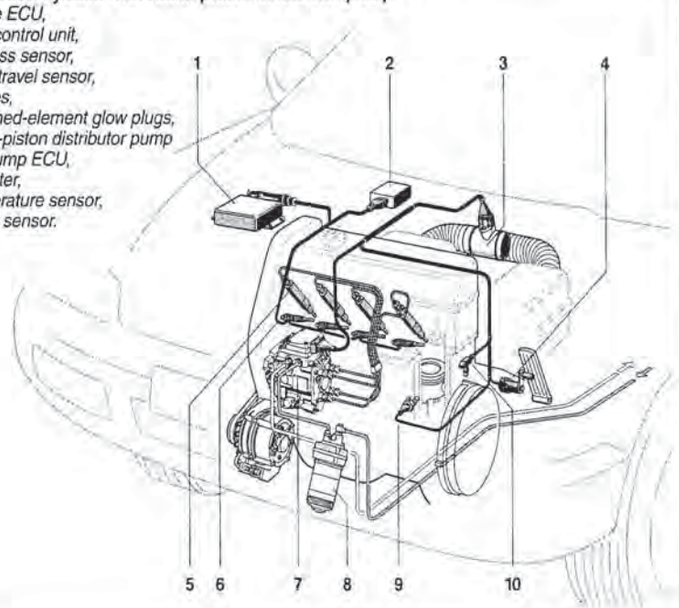
Axial-piston pumps use the same principle of pressure generation as EDC pumps with rotating-solenoid actuators. The injection pressure at the nozzle can be as high as 1500 bar, depending on the application.

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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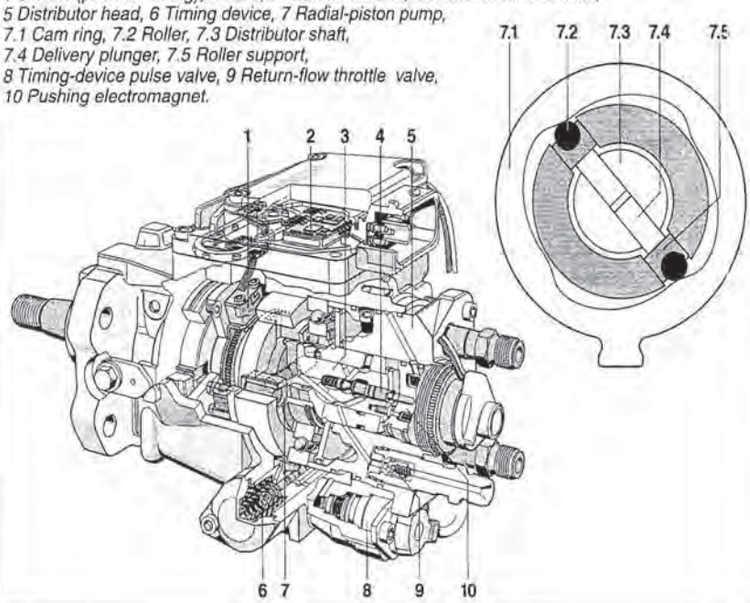
Fuel-injection system with radial-piston distributor pump

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

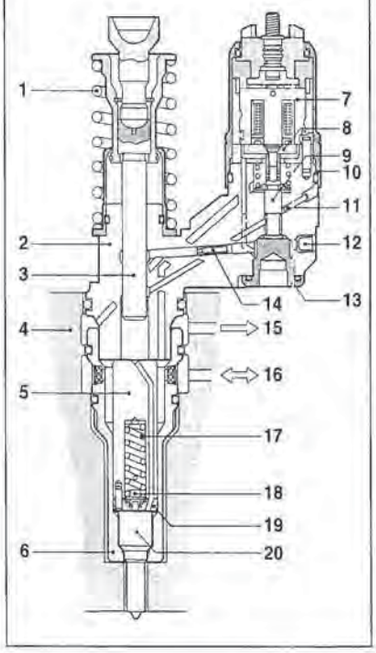
The new modular, time-controlled single-cylinder pump systems include the electronically controlled "Unit Injector System" (UIS) and the "Unit Pump System" (UPS) which are used on modern direct-injection engines for commercial vehicles and passenger cars.

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:

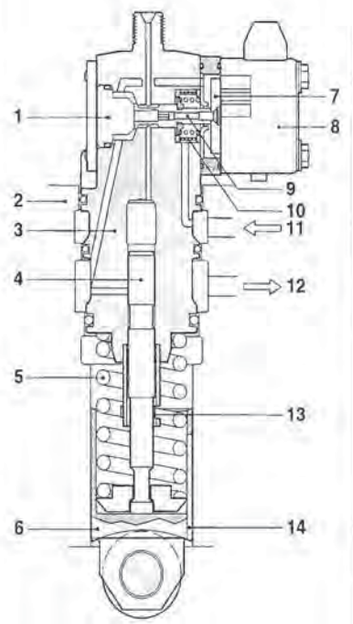
- high-pressure single-cylinder pump with integral high-speed solenoid valve,
- short high-pressure delivery line
- nozzle-holder assembly

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.

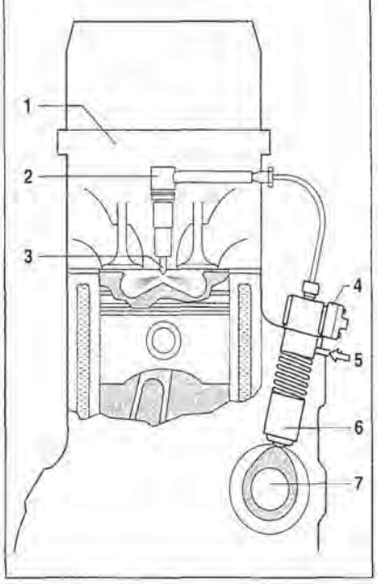
When the solenoid valve is open, the fuel can be drawn into the pump cylinder during the pump plunger's suction stroke, and return during the supply stroke. Only when the solenoid valve is energized, and thus closed, can pressure build up in the high-pressure system between the pump plunger and the nozzle 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 quantity to be selected as desired from hose stored in the program map. This eature, together with the high injection pressures, makes it possible to achieve very high power densities combined with very low emission levels and exceptionally low fuel consumption.

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

Electronic control unit (ECU)

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

Common-rail system (CRS)

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

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

System design

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

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

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

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

Hydraulic performance potential

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

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

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

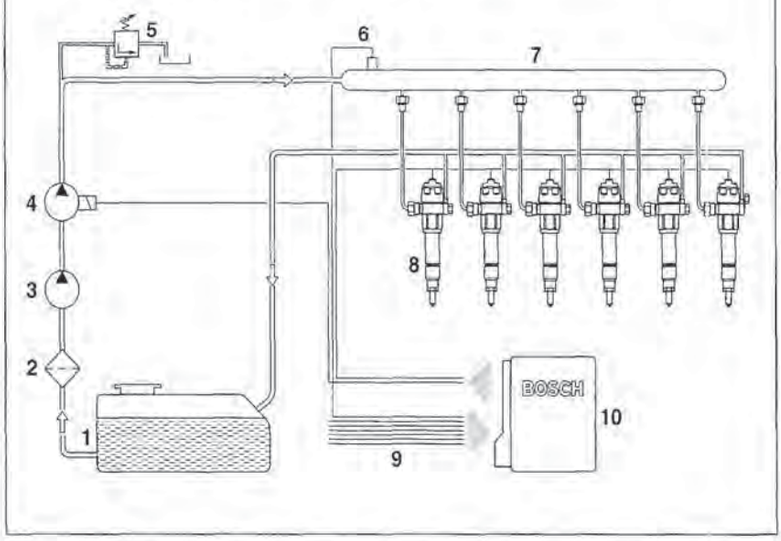
System application engineering on the engine

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

Injection-pump test benches

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

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



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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.

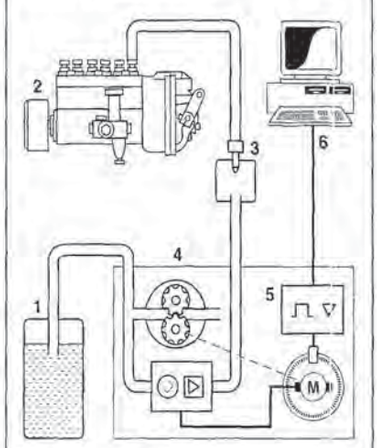
Continuous-flow delivery-quantity measurement

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).

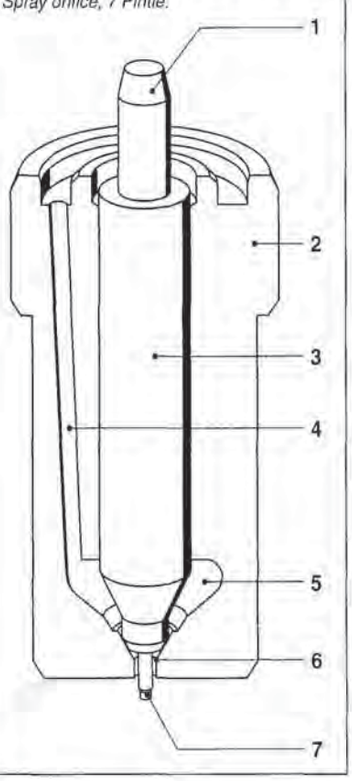
As a distinctive feature, throttling-pintle nozzles vary the discharge aperture – and thus the flow rate – as a function of needle stroke. The hole-type nozzle displays an immediate, sharp rise in aperture when the needle opens; in contrast, the throttling-pintle nozzle is characterized by an extremely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

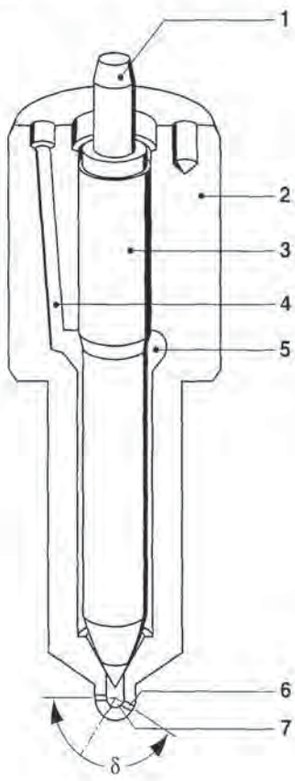
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Hole-type nozzles

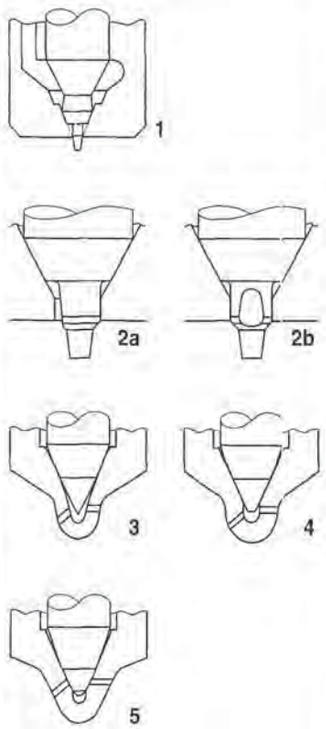
A wide range of nozzle-and-holder assemblies (DHK) is available for hole-type nozzles. In contrast to throttling pintle nozzles, hole-type nozzles must generally be installed in a specific position to ensure correct alignment between the orifices (which are at different angles in the nozzle body) and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

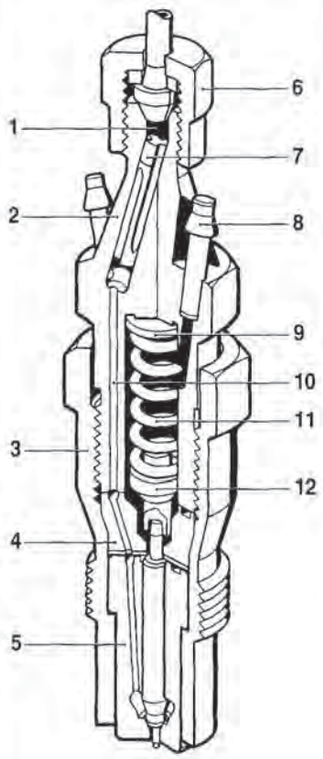
- opening pressure 1,
- opening pressure 2,
- prestroke, and
- overall stroke.

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.

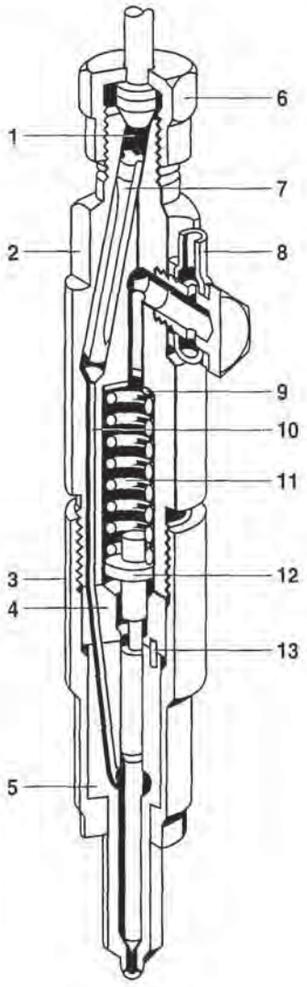
There are also dual-spring holders available for prechamber and whirl (turbulence) chamber engines. The setpoints are tailored to the respective injection system, with varying opening pressures of 130/180 bar and prestrokes of approx. 0.1 mm.

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.

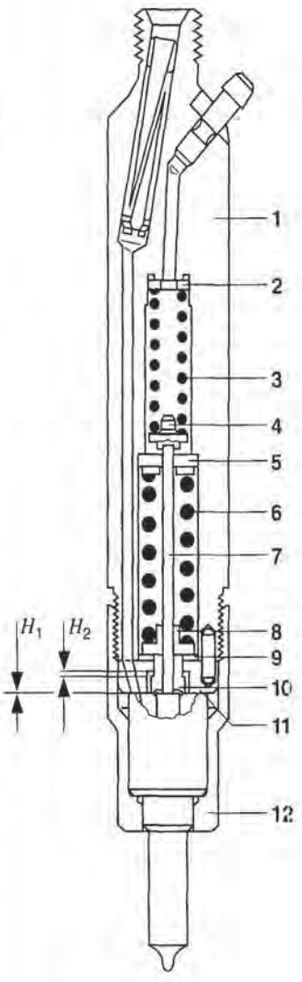


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Nozzle-holder assembly *with hole-type nozzle.*
1 Inlet , 2 Nozzle-holder body, 3 Nozzle-retaining nut, 4 Intermediate element, 5 Injection nozzle, 6 Union nut with high-pressure line, 7 Edge filter, 8 Leak-off connection, 9 Pressure-adjusting shims, 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 ($\lambda > 1$) 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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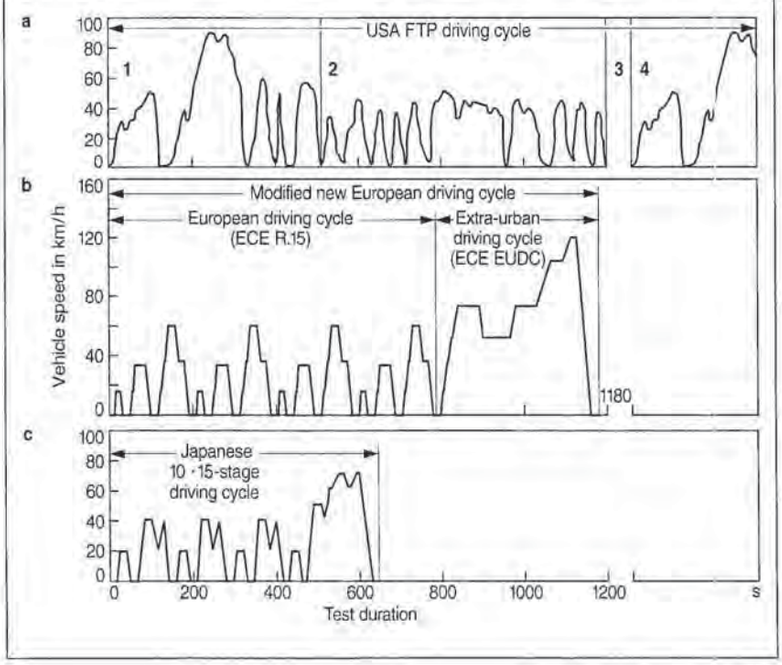
(max. 52 °C) must be maintained during the sampling procedure.

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

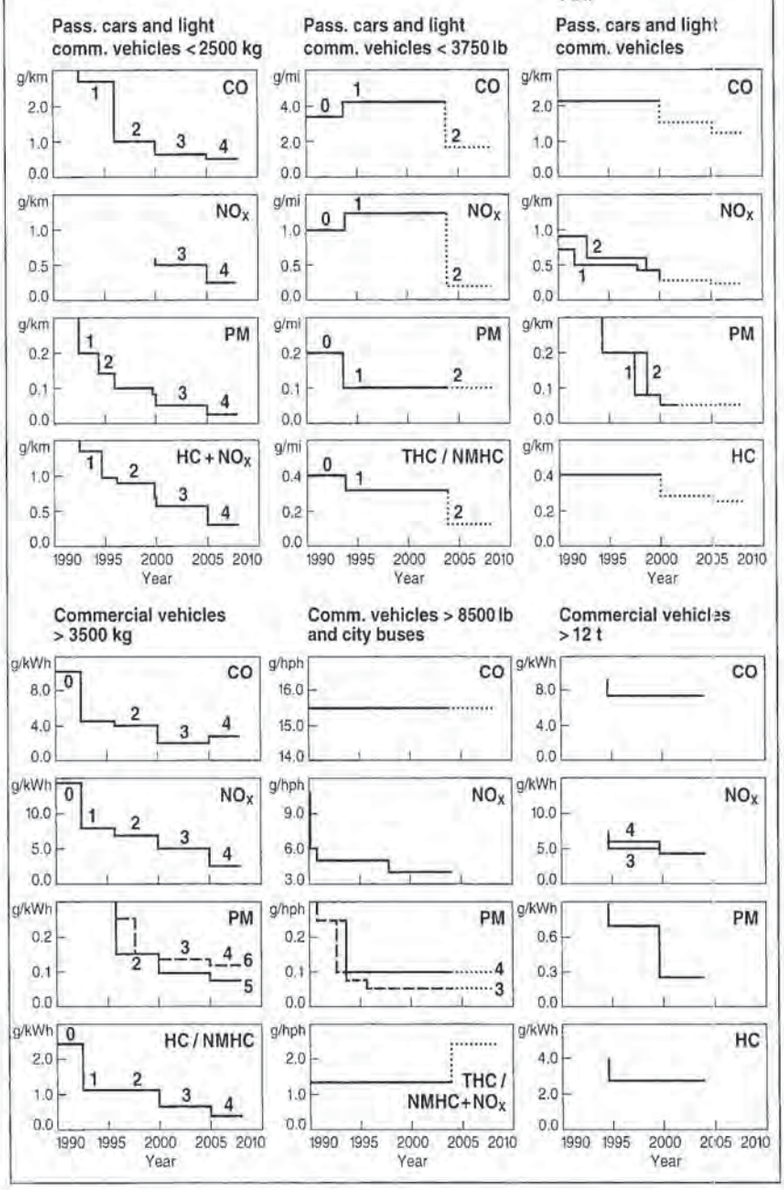
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Emission limits for diesel-powered passenger cars and commercial vehicles

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

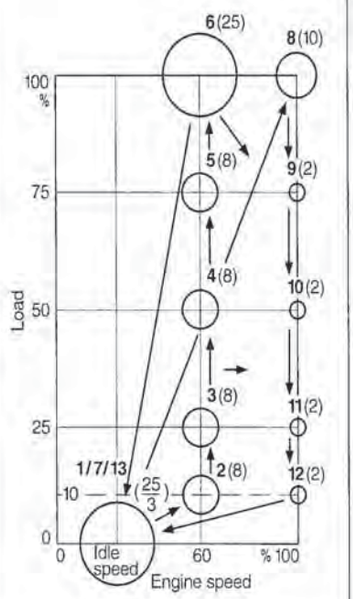
USA
0...2 Level 0...2,
3 HDV,
4 UB.

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

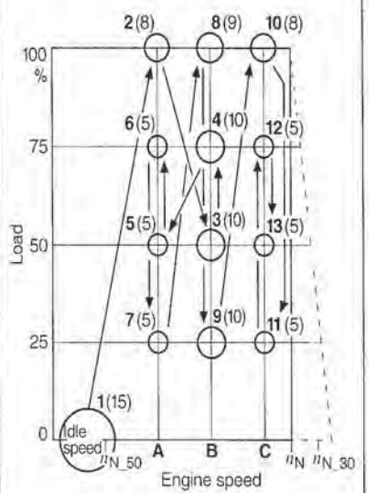


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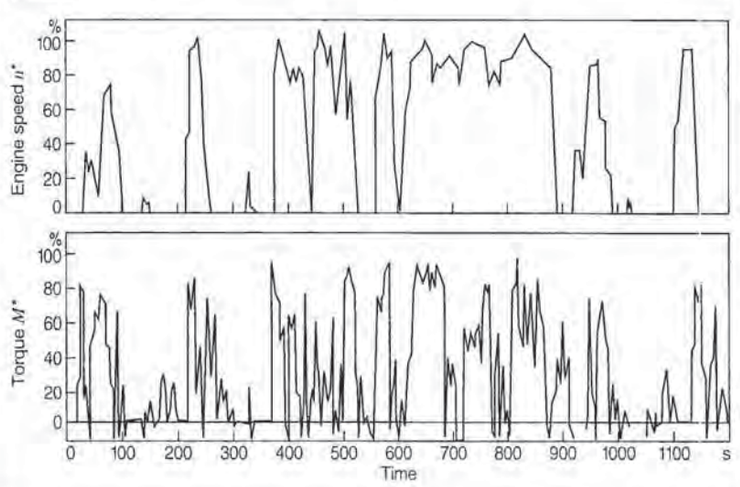
13-stage test ECE R.49 (Europe)
Order (bold) and Weighting
(% in parentheses) for test stages 1...13.



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.



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

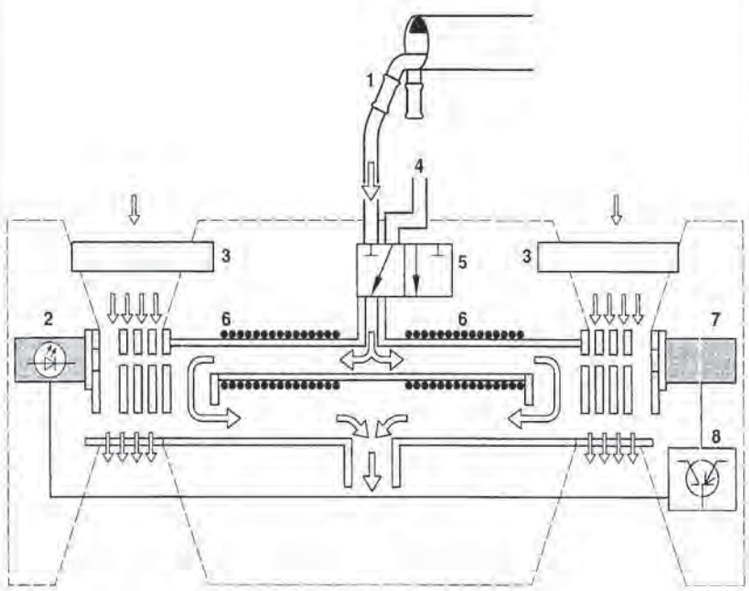
An important reason for testing diesel-engine exhaust emissions in the workshop, and for monitoring them while the vehicle is actually in operation on the road, is to determine the amount of soot (particulates) produced. This is indicated by the smoke number. Two standard procedures are used:

- 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.

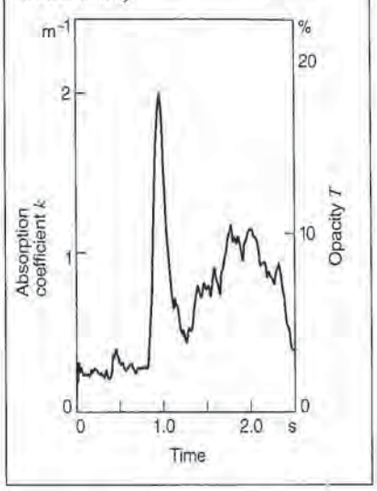
Inside the measuring chamber the diesel exhaust gas is penetrated by a beam of light. The attenuation of the light is measured photoelectrically, and is indicated as % opacity T or as a coefficient of absorption k. A high degree of accuracy and reproducibility of the test results requires that the length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

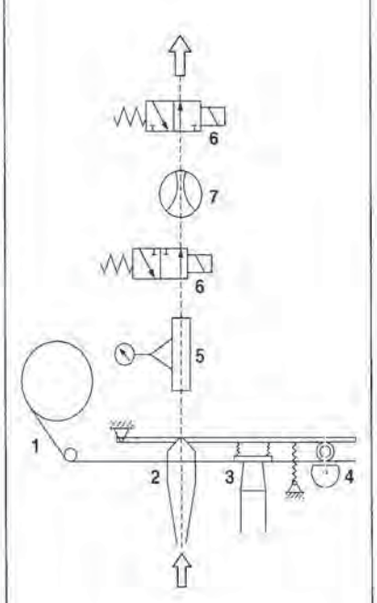
Smoke tester (filter method)
The smoke tester extracts a specified quantity (e.g. 0.1 or 1 *l*) of diesel exhaust gas through a strip of filter paper. Consistent, mutually comparable test results are achieved by recording the volume of gas processed in each test step; the device converts the results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

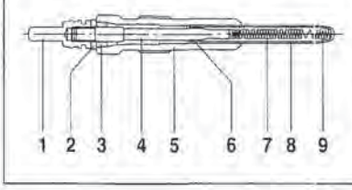
Operation

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.

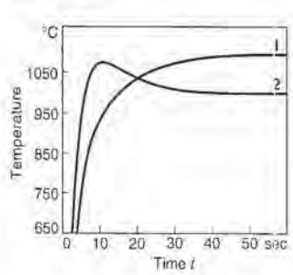
Start phase: Here the glow plug must heat to starting temperature (approx. 850 °C) as rapidly as possible. Plug locations within the combustion chamber are selected to ensure access to an ignitable mixture. Modern glow plugs heat to the required temperature in roughly 4 seconds.

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.

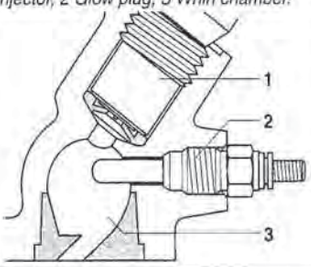


Temperature of sheathed-element glow plug in stationary air as function of time
1 S-RSK, 2 GSK2.



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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

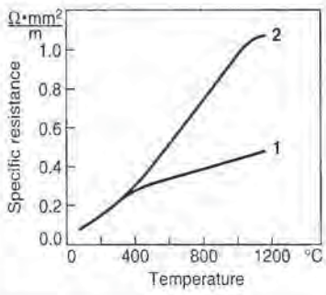
The complete glow-plug system incorporates not only the glow plugs themselves (only in rare cases are flame plugs used) but also a switching element for the high electrical glow-plug currents which is triggered by a special control unit. In addition, the system incorporates an indicator lamp (also controlled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

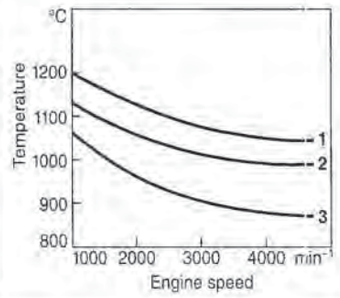
Design

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.

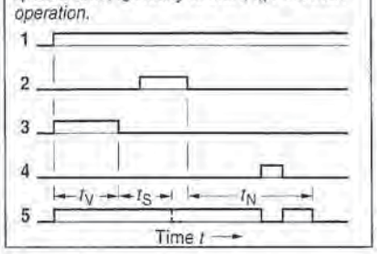


Glow-plug temperatures in DI engine
1 Full load, 2 50 % load, 3 No load.



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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

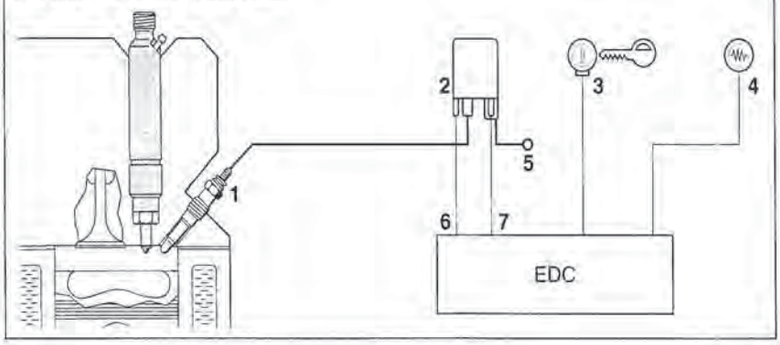
On basic units, a temperature sensor in the glow control unit controls the preheating period. This is designed to suit the specific requirements of the particular combination of engine and glow-plug so that the glow plug can reach the temperature necessary for efficient starting. At the end of the glow 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 informat on on when glow-plug operation is required, and when not, directly from the engine's central ECU. That unit provides (statically or by means of a serial data protocol) the information relating to the engine operating status (coolant temperature, engine speed and load) that is required for optimum control of the glow plugs. Similarly, the glow control unit also signals any faults it detects to the engine ECU via a diagnosis line or the serial interface. There, they are stored for servicing purposes or displayed if necessary for compliance with OBD requirements.

EDC-controlled glow-plug system on DI engine
1 Glow plug, 2 Glow control unit, 3 Glow-plug and starter switch, 4 Preheating indicator, 5 To battery, 6 Control, 7 Diagnosis.



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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.

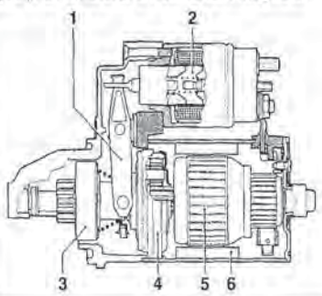
Design factors

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.

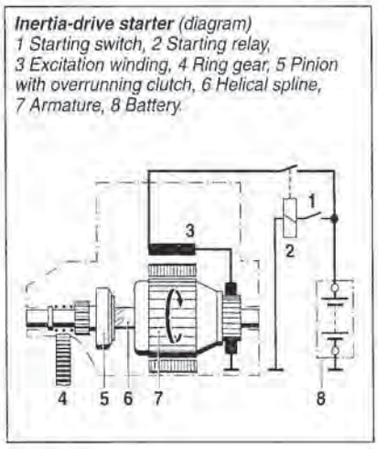
When the starting switch is closed, the hold-in winding H is energized, and current flows through the series circuit comprising pull-in winding E and electric motor. The starter motor solenoid picks up and moves the overrunning clutch and pinion gear assembly forwards by means of the thrust lever and 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.

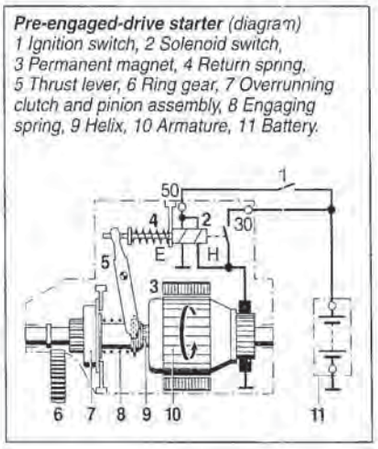
If the pinion teeth do not immediately mesh with the ring-gear gaps, the ring gear prevents the pinion from advancing any further. The engaging lever then compresses the meshing spring, and the main contact closes without the pinion having engaged.

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.

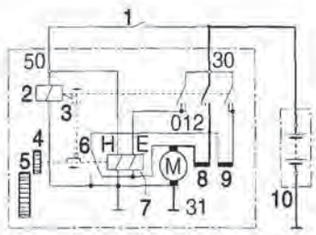
This disengaging operation is assisted by the helix when the engine speed overtakes the starter speed.

Sliding-gear starters

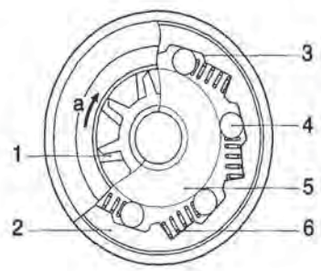
The sliding-gear drive switches the starter on in two stages.

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

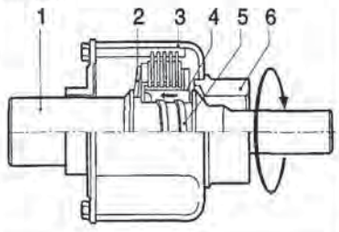
The multiplate overrunning clutch is used in large commercial-vehicle starters. The driver with the outer plates and the starter armature, and on the other side the driving shaft and the pinion are positively connected to one another. The inner plates fit in an inner clutch race which rides axially in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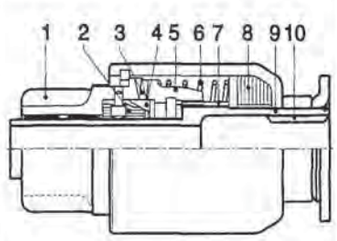
to transmit increasing torque as the starter load increases.

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.

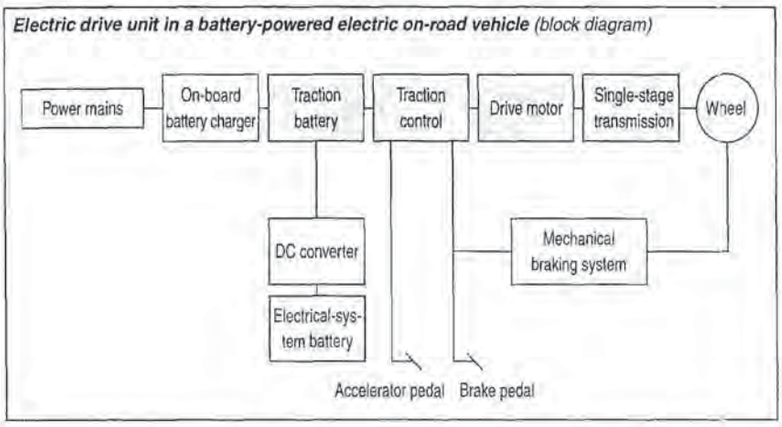
Depending on the intended application, battery-powered electric vehicles can be classified as either road vehicles or industrial trucks. Industrial trucks are used for transporting goods on company premises, and are generally not licensed for use on public roads. Their top speed is below 50 km/h. Due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

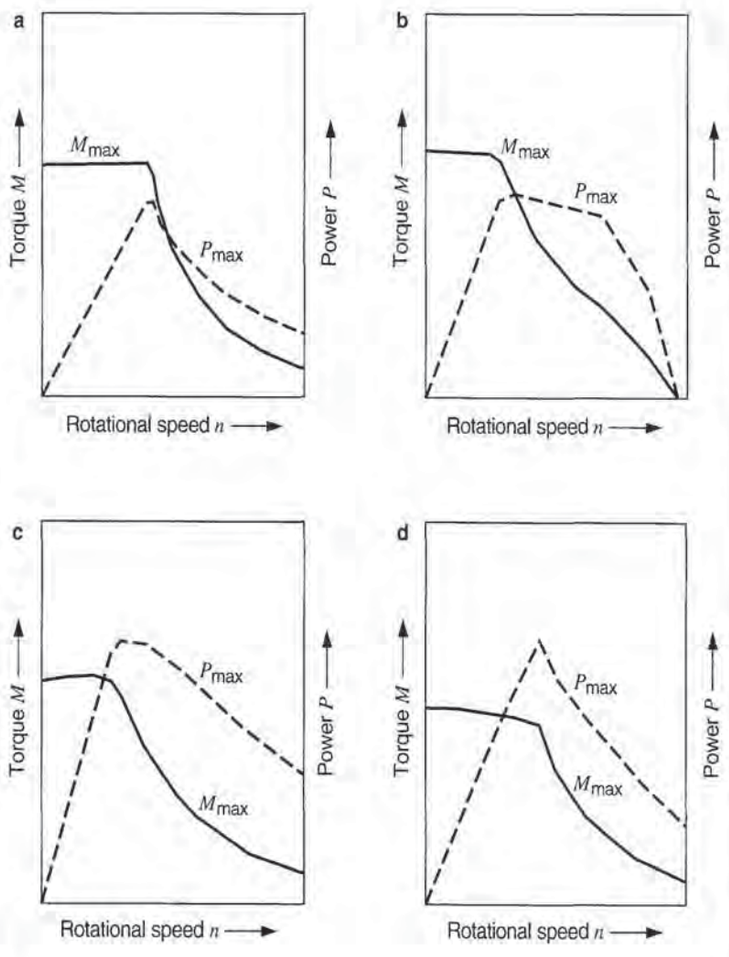
In this type of drive unit, the motor's magnetic excitation is provided by its own controller (field rheostat). Depending on the size of the motor, the field can be weakened in a ratio of up to approx. 1 4. Field strength starts to diminish at a nominal motor speed obtained with full motor voltage at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

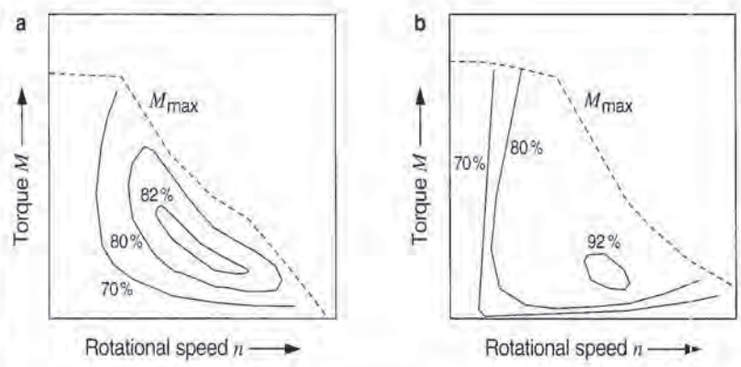
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Torque and power as a function of rotational speed for various types of drive
a) Series-wound DC drive,
b) Separately-excited DC drive,
c) Asynchronous drive,
d) Permanently-excited synchronous drive.
M_{max} *Maximum torque,*
P_{max} *Maximum power.*



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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.

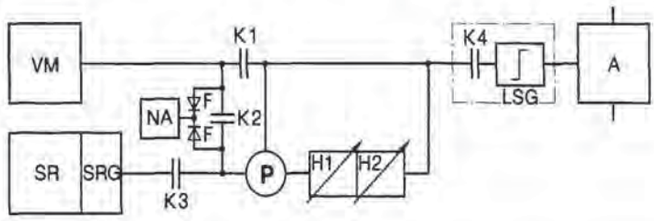
In addition, the internal-combustion engine can start the two kinetic-energy storage devices from standstill via the continuously variable transmission. This configuration also permits parallel operation with the two drive sources, diesel engine and flywheel. Here, only the power delivered by the flywheel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

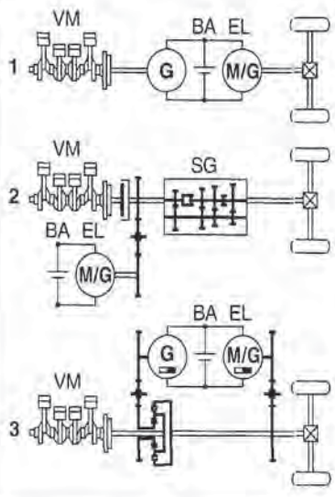
Combining a mechanical energy store with the battery prevents it being affected by power peaks, thus contributing to more efficient utilization of the battery energy. A hybrid system which uses a combination of two different electrochemical energy sources (battery and fuel cell) separates the energy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

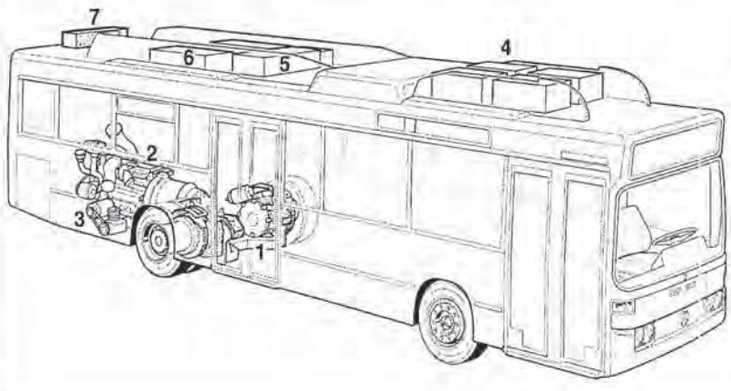
Because the diesel engine in the series configuration is mechanically decoupled from the vehicle drive, the diesel engine can be operated at a constant speed, i.e. at its optimum operating point in terms of efficiency and emissions. Despite the advantages of the series configuration, its disadvantage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



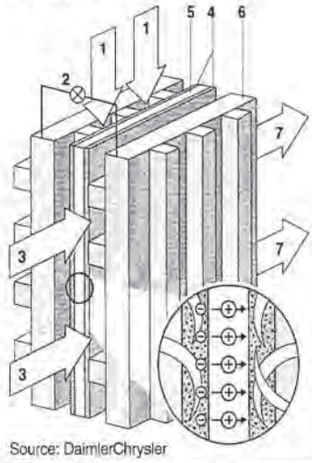
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Fuel Cells

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 Membrane	50...80	50...60	Mobile, stationary
Direct Methanol Fuel Cell	DMFC		110...130	30...40	Mobile
Phosphoric Acid Fuel Cell	PAFC	H_3PO_4	160...220	55	Stationary
Molten Carbonate Fuel Cell	MCFC	Alkaline carbonates	620...660	60...65	Stationary
Solid Oxide Fuel Cell	SOFC	ZrO_2	800...1000	55...65	Stationary

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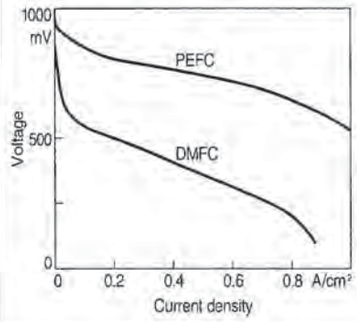
familiar electrolytic gas reaction, which in fuel cells is a controlled process ("cold combustion") rather than an explosive one. This reaction takes place in a controlled manner in fuel cells because the important reaction processes (H_2 oxidation and O_2 reduction) occur at physically separate electrodes and at a relatively low temperature (Figure and Table 1). It is therefore possible in fuel cells to directly and completely convert the chemical energy, which corresponds to the decrease in free enthalpy ΔG_R of the reaction, into electrical work A_e in accordance with Equation 1:

$$A_e = -\Delta G_R = n \cdot F \cdot E_0 \qquad \text{(Equation 1)}$$

(n Number of electrons converted per fuel molecule, F Faraday constant).

Because in this type of energy conversion the customary indirect path via the generation of heat is avoided, fuel-cell efficiency is not limited by the relatively poor efficiency of the Carnot cycle.

Typical current/voltage characteristics for low-temperature fuel cells



As shown in Table 2, the cell voltages E_0 for the two fuels are close to one another (approx. 1.2 V) and the thermodynamic efficiency η_{th} calculated according to Equation 2 is also < 1 for CH_3OH if liquid water is formed as the reaction product:

$$\eta_{th} = \Delta G_R / \Delta H_R \qquad \text{(Equation 2)}$$

At temperatures above 100 °C (if CH_3OH and H_2O are present in gaseous form), the CH_3OH/O_2 fuel cell shows that in principle fuel cells can also attain a thermodynamic efficiency of > 1.

In practice the theoretical fuel-cell figures are not reached with any of the fuels, even at high temperatures. The reasons for this are primarily to be found in the kinetic inhibition at the two electrodes (e.g. charge penetration and material transfer) as well as in the mixed potential formation at the positive electrode and the electrolyte impedance. The resulting polarization can be reduced but not completely eliminated by using noble metals (platinum, ruthenium) as catalysts, specially structured porous electrodes and small electrode gaps. This applies even in the absence of current flow when a steady-state voltage of only approx. 1 V is measured at an H_2/O_2 PEFC (see Table 1) instead of the expected 1.23 V. When a load is applied, fuel-cell voltage and efficiency drop by varying degrees, depending on the fuel, as current increases due to increasing polarization (see graph). Other variables which have a significant effect on the shape of the current/voltage characteristic of a fuel cell are temperature, gas pressure (1.5 ... 3 bar) and the noble-metal electrode coating, which is now as little as 0.1 ... 0.5 mg Pt/cm².

The typical working voltage of an H_2/air

Table 2. Reaction equations and calculated data for hydrogen (H_2) and methanol (CH_3OH) as energy sources for fuel cells

Reaction equations	State of H_2O/CH_3OH	Temperature °C	E_0 V	η_{th} %
$H_2 + {}^1/_2O_2 \rightarrow H_2O$	Liquid	25	1.23	83
	Liquid	127	1.15	81
	Gaseous	127	1.16	92
	Gaseous	227	1.11	87
$CH_3OH + {}^3/_2O_2 \rightarrow CO_2 + 2\,H_2O$	Liquid/liquid	25	1.21	97
	Liquid/liquid	127	1.20	99
	Gaseous/gaseous	127	1.20	103
	Gaseous/gaseous	227	1.21	104

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PEFC at rated continuous output (0.4...0.5 W/cm² of electrode surface area) is around 0.75...0.70 V (see graph) which, according to Equation 2, results in an efficiency of 51...48%. With CH_3OH as the PEFC energy source, despite the higher temperature (110...130 °C), lower power density (0.1...0.2 W/cm²) and greater amount of Pt on the electrodes, a lower voltage (0.50...0.35 V) and a correspondingly lower thermodynamic efficiency (41...30%) is produced than in the case of an H_2/air PEFC. However, this does not take into account the fact that, in the case of CH_3OH, its permeation through the electrolyte membrane results in a loss at the anode due to which the efficiency of a direct methanol fuel cell is even further reduced.

Fuel cells in motor vehicles

As clean, high-efficiency energy converters, from the point of view of environmental protection and conservation of resources, fuel cells represent an interesting alternative to the traditional methods of generating electrical power. Of the types of fuel cells described here, PEFCs in particular show the most promise in both fixed and mobile applications by virtue of their low operating temperature and compact and robust design. In the area of mobile applications, driven by the increasingly stringent exhaust-emission regulations in the USA and Europe, fuel-cell power units are clearly at the forefront of development efforts. In addition, however, the possibility of using PEFCs as the electrical power source in vehicles with conventional engines is also being investigated, and initial experiments have been carried out. A vehicle electrical system with a power supply independent of the engine which provides the motive power would enable implementation of desirable timer-controlled or remotely controlled auxiliary functions (e.g. preheating of the engine/catalytic converter, air-conditioner operation without the engine running).

All of the automobile manufacturers engaged in comprehensive fuel-cell development work have focused on two key problems, which in principle affect all fuel-cell applications: the high cost of PEFCs and the availability of pure H_2, which is essential for the operation of PEFCs and for which there is no infrastructure at present which could be used to supply vehicles powered by such fuel cells. An additional disadvantage of H_2 as a fuel for fuel-cell powered vehicles is the fact that its energy density is satisfactory only if it is stored in pressurized or liquefied form. For reasons of safety alone, there are serious concerns regarding the storage of H_2 in those forms in private passenger cars.

When using fuel cells as automotive drives, efficiency drops to around 30% from the H_2 storage tank to the wheels. This loss is attributable in part to the auxiliary systems required for monitoring and operation of the fuel cells (e.g. air compressor, coolant pumps, fan cooler control equipment and, where applicable, gas reformers). The electrical power required by the secondary loads can amount to as much as 25% of the fuel-cell output, depending on the nature of the peripherals and the size and operating point of the fuel cell. In addition to fuel-cell efficiency, the efficiency of the electric drive unit must also be taken into consideration. When considering the entire energy chain from the primary energy source to the vehicle's powered wheels, and given the technology available at present, the overall efficiency of modern diesel and fuel-cell-powered vehicles is comparable for vehicles of equal power-to-weight ratios.

No predictions can be made at this time regarding the service life of PEFCs under the dynamic conditions of motor-vehicle operation. As in all applications involving catalysts, fuel cells can also be expected to suffer from a reduction in the catalytic action of the electrodes over time. The result is a gradual decline in the voltage and efficiency of the fuel cell which is referred to as degradation. It is assumed, however, that over the long term the service life of fuel-cell drives will be similar to that of internal-combustion engines.

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Drivetrain

Units and symbols

Quantity		Unit
a	Acceleration	m/s²
c_w	Drag coefficient	–
e	Rotational inertia coefficient	–
f	Rolling resistance coefficient	–
g	Gravitational acceleration	m/s²
i	Transmission ratio	–
m	Vehicle mass	kg
n	Speed of rotation	min⁻¹
r	Dynamic tire radius	m
s	Wheel slip	–
v	Road speed	m/s
A	Frontal area	m²
D	System diameter	m
I	Overall conversion range	–
J	Moment of inertia	kg · m²
M	Torque	N · m
P	Power	kW
α	Angle of inclination	°
φ	Overdrive factor	–
η	Efficiency	–
λ	Performance index	–
μ	Conversion	–
ρ	Density	kg/m³
ω	Angular velocity	rad/s
ν	Speed ratio	–

Subscripts:			
eff	Effective	eng	Engine
tot	Total	o	Towards maximum output
hydr	Hydraulic	dt	Drivetrain
max	Maximum	G	Gearbox
min	Minimum	P	Pump
fd	Final drive	R	Roadwheel
		T	Turbine

Assignment

The function of the automotive drivetrain is to provide the thrust and tractive forces required to induce motion. Energy in chemical (fuels) or electrical (batteries, solar cells) form is converted into mechanical energy within the power unit, with spark-ignition and diesel internal-combustion engines representing the powerplants of choice. Every power unit operates within a specific revolution range as defined by two extremities: the idle speed and the maximum min⁻¹. Torque and power are not delivered at uniform rates throughout the operating range; the respective maxima are available only within specific bands. The drivetrain's conversion ratios adapt the available torque to the momentary requirement for tractive force.

Design

The dynamic condition of an automobile is described by the motion-resistance equation. It equates the forces generated by the drive train with the forces required at the driving wheels (resistance to motion).

From the motion-resistance equation it is possible to calculate acceleration, maximum speed, climbing ability and also the overall conversion range I of the transmission.

$$I = \frac{(i/r)_{max}}{(i/r)_{min}} = \frac{\tan \alpha_{max} \cdot v_o}{(P/(m \cdot g))_{eff.} \cdot \varphi}$$

Equilibrium relation between drive forces and tractive resistance

The equation defining the equilibrium between drive forces and resistance factors is applied to determine various quantities, such as acceleration, top speed, climbing ability, etc.

Available power = Tractive resistance at drive wheels (power requirement)

$$M_{eng} \cdot \frac{i_{tot}}{r} \cdot \eta_{tot} = m \cdot g \cdot f \cdot \cos \alpha + m \cdot g \cdot \sin \alpha + e \cdot m \cdot a + c_w \cdot A \cdot \frac{\rho}{2} \cdot v^2$$

Driving force applied to tyre footprint	Rolling resistance	Gravitational resistance	Acceleration	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

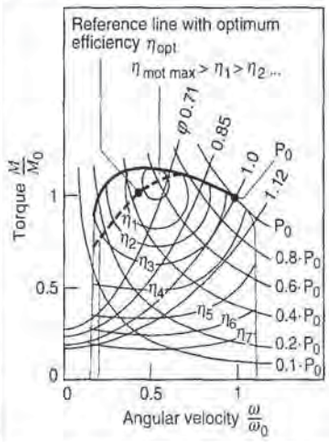
The layout of the automotive drivetrain varies according to the position of the engine and the drive axle:

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

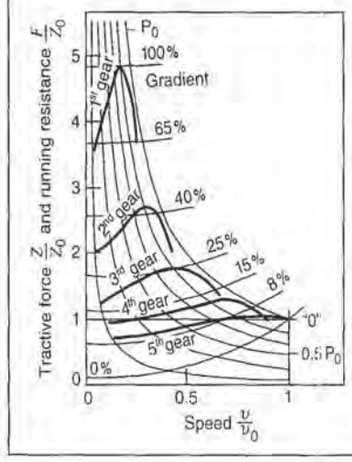
Drivetrain elements

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.

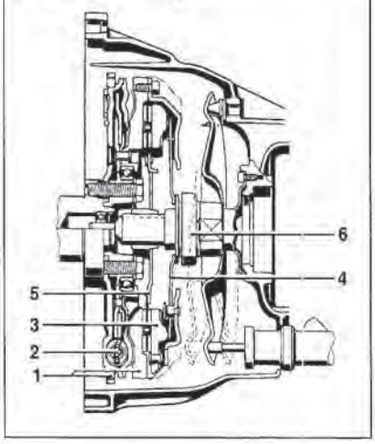
Power take-up elements

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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US PATENT NO. 6,581,565

A two-section (dual-mass) flywheel featuring a flexible intermediate element can be installed forward of the clutch for maximum insulation against vibrations. The resonant frequency of this sprung mass system is below the exciter frequency (ignition frequency) of the engine at idling speed and therefore outside the operating speed range. It acts as a vibration insulating element between the engine and the other drive-train components (low-pass filter).

When used together with electronic control units, the automatic clutch can provide either gradual engagement for starting off, or it can be applied in conjunction with a servo-operated shifting mechanism to form a fully-automatic transmission unit. Traction control and disengagement of power transmission during braking are also possible.

Wet-plate friction clutch

The wet-plate friction clutch has the advantage over the dry-plate version that its thermal performance is better as oil can be passed through to assist heat dissipation. However, its drag losses when disengaged are considerably higher than with a dry clutch. Use in combination with synchromesh gearboxes presents problems due to the increased synchronous load when changing gear. The wet clutch was introduced as a standard component on continuously variable car transmissions. It has space-saving advantages particularly when one or more friction-drive gear-shift components (multiplate clutch or clutch stop) that are present in any case can also be used for the power take-up process.

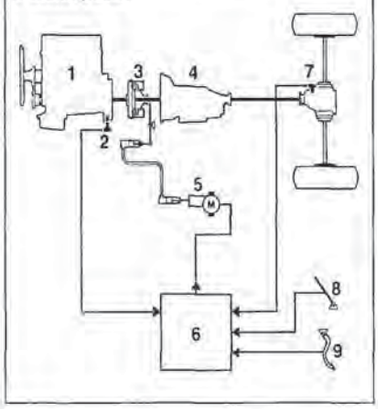
Hydrodynamic torque converter

The hydrodynamic torque converter consists of the impeller which is the driving component, the turbine which is the driven component and the stator which assists the torque converter function. The torque converter is filled with oil and transmits the engine torque by means of the viscosity of the oil. It compensates for the speed difference between the engine and the other drive-train components and is therefore ideally suited to a power take-up function. An impeller converts the mechanical energy into fluid energy; and a second transformation, back into mechanical energy, takes place at the blades within the turbine.

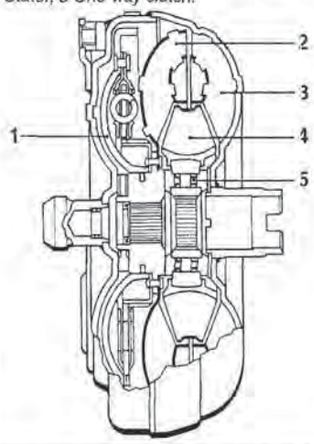
The impeller's input torque M_p and input power P_p are calculated as follows:

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

Automatic clutch, limited to clutch actuation
1 Engine, 2 Engine rpm sensor, 3 Clutch, 4 Transmission, 5 Servomotor, 6 ECU, 7 Speed sensor, 8 Accelerator pedal, 9 Clutch pedal.



Hydrodynamic converter with lockup clutch
1 Lockup clutch, 2 Turbine, 3 Impeller, 4 Stator, 5 One-way clutch.



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λ Performance index
ρ Density of medium
($\approx$ 870 kg/m³ for hydraulic fluid)
D Circuit diameter in m
ω_P Angular velocity of impeller

A stator located between impeller and turbine diverts the hydraulic oil back to the input side of the impeller. This raises the torque beyond the initial engine output as exerted at the impeller. Torque conversion is then

$$\mu = -M_T/M_P$$

The factor ν is defined as the ratio of turbine speed to impeller speed; it has a determining influence on both the performance index λ and the conversion factor μ:

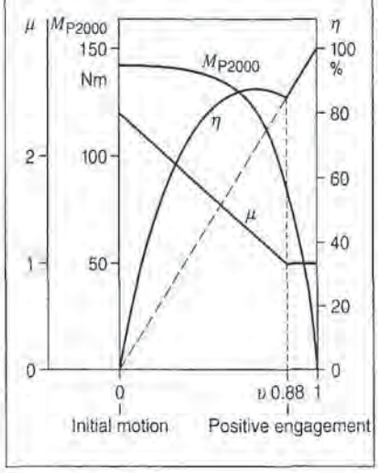
$$\nu = \omega_T / \omega_P.$$

The slip factor $s = 1 - \nu$ and the force conversion together determine the hydraulic efficiency:

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

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

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



In motor-vehicle applications, the two-phase Föttinger torque converter with centripetal flow through the turbine, the "Trilok converter", has become the established design. The geometrical configuration of this unit's blades is selected to provide torque multiplication in the range of 1.7...2.5 at stall speed ($\nu = 0$). The curve defining the hydraulic efficiency factor $\eta_{hydr} = \nu \cdot \mu$ in the conversion range is roughly parabolic. Above the coupling point, which is at 10...15% slip, the efficiency is equal to the speed ratio ν and reaches levels of around 97% at high speeds.

The hydrodynamic torque converter is a fully automatic infinitely variable transmission with virtually zero-wear characteristics; it eliminates vibration peaks and absorbs vibration highly effectively.

However, its conversion range and efficiency, particularly at high levels of slip, are not sufficient for motor-vehicle applications so that the torque converter can only be usefully employed in combination with multi-speed or continuously variable gearboxes.

Converter lockup clutch

In order to improve efficiency, the impeller and turbine can be locked together by a converter lockup clutch once power take-up is complete. The converter lockup clutch consists of a plunger with friction surface, which is connected to the turbine hub. The transmission's valve body regulates the direction in which the fluid flows through the converter to regulate coupling engagement.

The converter lockup clutch normally requires additional means of vibration absorption such as
- a torsion damper,
- controlled-slip operation of the converter lockup clutch at critical vibration levels or
- both of the above in combination.

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Multi-speed gearbox

Multi-speed gearboxes have become the established means of power transmission in motor vehicles. Good efficiency characteristics dependent upon the number of gears and engine torque characteristics, satisfactory to good adaptation to the traction hyperbola and easily mastered technology are the essential reasons for its success.

Gear shifting on multi-speed gearboxes is performed using either disengagement of power transmission (positively interlocking mechanism) or under load by a friction mechanism. The first group includes manual and semi-automatic gearboxes while the second group encompasses automatic transmissions.

The manually-shifted transmissions installed in passenger cars and in most heavy vehicles are dual-shaft units with main and countershaft (layshaft, idler gears). Transmissions in heavy commercial vehicles sometimes incorporate two or even three countershafts. In such cases, special design features are required in order to ensure that power is evenly distributed to all countershafts.

Automatic transmissions for cars and commercial vehicles are, in the majority of cases, planetary gear transmissions and only in rare cases are countershaft designs used. The planetary gears generally take the form of a planetary-gear link mechanism. They frequently involve the use of Ravigneaux or Simpson planetary gears.

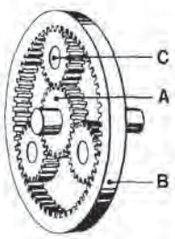
Planetary-gear sets

The basic planetary-gear set consists of the sun gear, internal ring gear and the planet gears with carrier. Each element can act as input or output gear, or may be held stationary. The coaxial layout of the three elements makes this type of unit ideal for use with friction clutches and brake bands, which are employed for selective engagement or fixing of individual elements. The engagement pattern can be changed – and a different conversion ratio selected – without interrupting torque flow; this capability is of particular significance in automatic transmissions.

As several gear wheels mesh under load simultaneously, planetary-gear transmissions are very compact. They have no free bearing forces, permit high torque levels, power splitting or power combination, and feature very good efficiency levels.

Planetary-gear set with various conversion ratios
A Sun gear, B Internal ring gear, C Planet gears with carrier, Z Number of teeth.

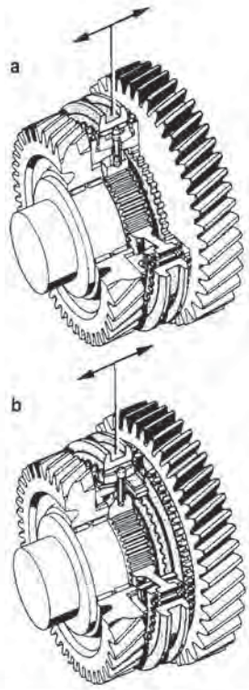
Basic equation for planetary-gear sets: $n_A + (Z_B / Z_A) \cdot n_B - [1 + Z_B / Z_A] \cdot n_C = 0$



Input	Output	Fixed	Transmission ratio	Remarks
A	C	B	$i = 1 + Z_B / Z_A$	$2.5 \le i \le 5$
B	C	A	$i = 1 + Z_A / Z_B$	$1.25 \le i \le 1.67$
C	A	B	$i = \frac{1}{1 + Z_B / Z_A}$	$0.2 \le i \le 0.4$ overdrive
C	B	A	$i = \frac{1}{1 + Z_A / Z_B}$	$0.6 \le i \le 0.8$ overdrive
A	B	C	$i = -Z_B / Z_A$	Non-automotive with reversible direction of rotation $-0.4 \le i \le -1.5$
B	A	C	$i = -Z_A / Z_B$	Non-automotive with reversible direction of rotation $-0.25 \le i \le -0.67$

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Manually-shifted transmission
a) Single-band synchromesh,
b) Dual-band synchromesh.



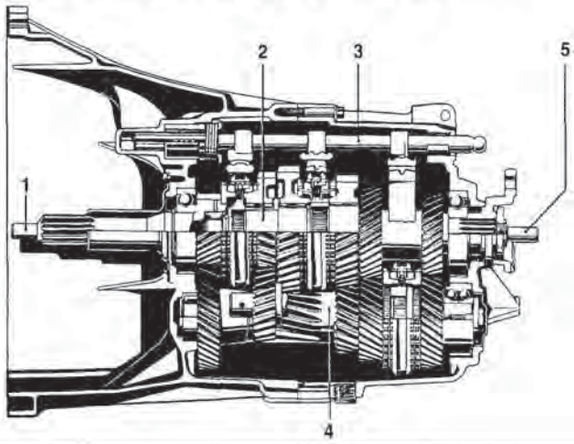
Manually-shifted transmissions

The basic elements of the manually-shifted transmission are:
- single or multiplate dry clutch for interrupting and engaging the power flow; actuation may be power-assisted to deal with high operating forces,
- variable-ratio gear transmission unit featuring permanent-mesh gears in one or several individual assemblies,
- shift mechanism with shift lever.

The force required for gear selection is transmitted via shift linkage rods or cable, while dog clutches or synchronizer assemblies lock the active gears to the shafts. Before a shift can take place, it is necessary to synchronize the rotating speeds of the transmission elements being joined. When the transmission incorporates dog clutches (of the type still sometimes used in transmissions for heavy commercial vehicles), the driver performs this task by double-clutching on both upshifts and downshifts, with the latter being accompanied by the application of throttle.

Virtually all passenger-car transmissions, and the majority of those in commercial vehicles, employ locking synchronizer assemblies. These include a friction

5-speed transmission for passenger car with conventional drive layout *(ZF S5-31).*
1 Input shaft, 2 Main shaft, 3 Selector rail, 4 Idler shaft, 5 Output shaft



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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.

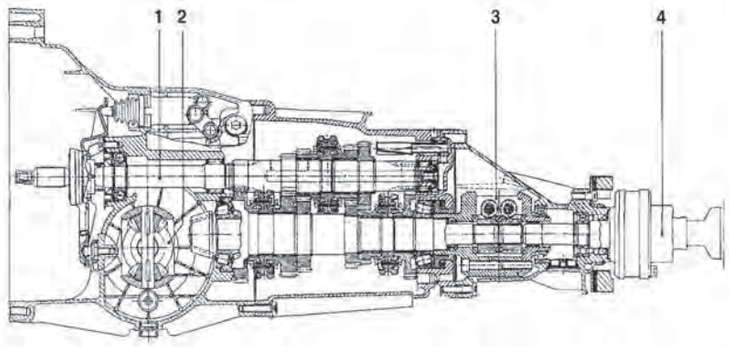
Power take-offs (auxiliary drives)

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

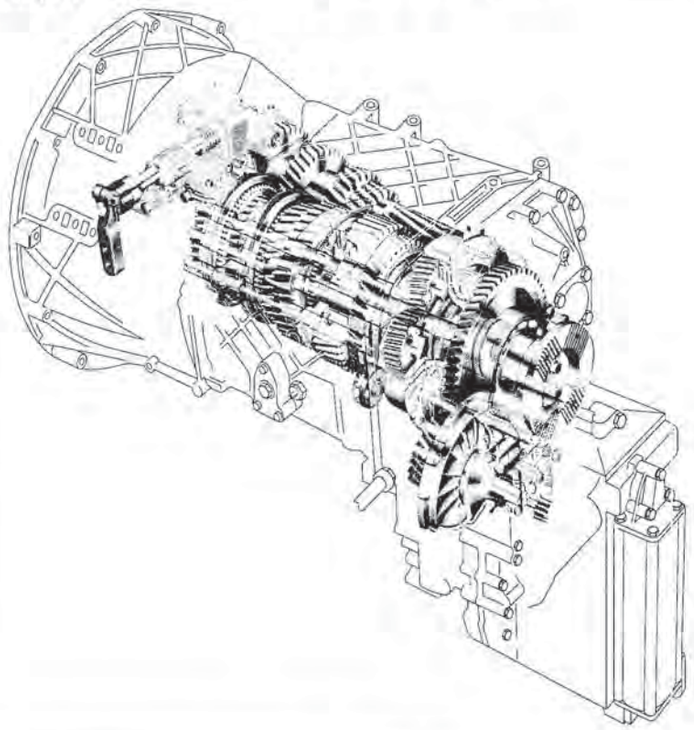
Hydrodynamic or electrodynamic retarders are non-wearing auxiliary brakes for reducing the thermal load on the road-wheel brakes under continuous braking. They can be fitted either on the drive input side (primary retarders) or the drive output side (secondary retarders) integral with the transmission,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

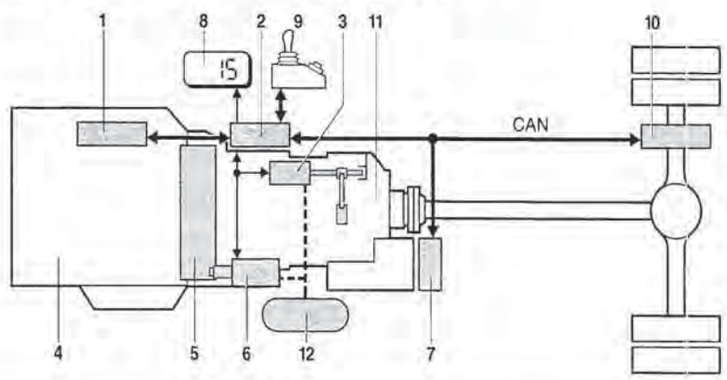


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16-speed multiple transmission with integral retarder for heavy trucks
(ZF Ecosplit 16 S221)



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.

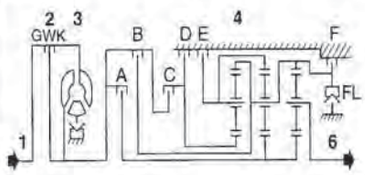
Electronic transmission control

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

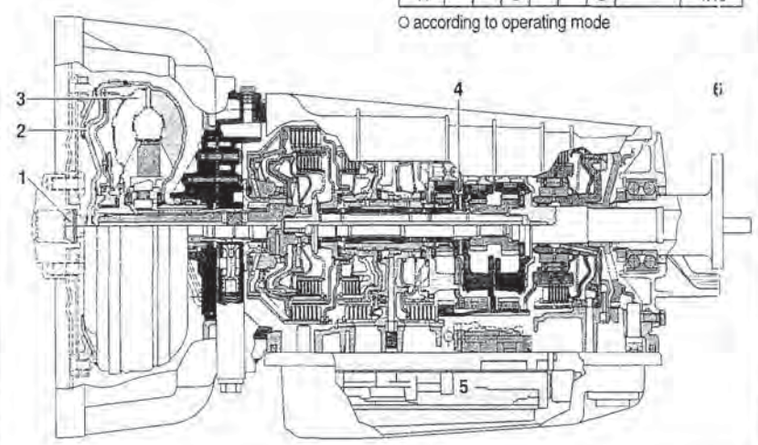
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5-speed passenger-car automatic gearbox *(ZF 5 HP24)*
1 Input shaft, 2 Controlled-slip converter lock up clutch (GWK), 3 Hydrodynamic torque converter, 4 5-speed planetary gears with three multiplate brakes, three multiplate clutches and one one-way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

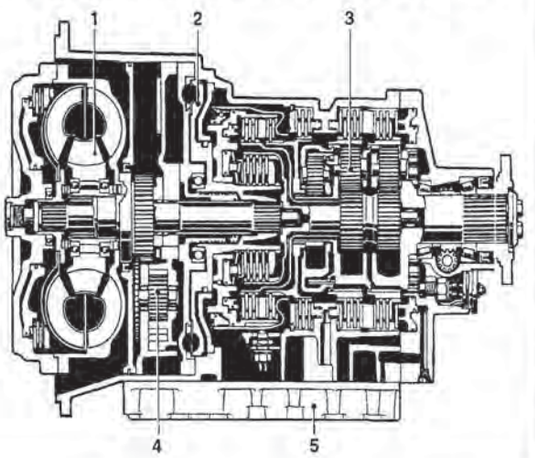


Gear	Clutch			Brake			Overrun	Ratio
	A	B	C	D	E	F	FL	i
1	●					○	●	3.57
2	●				●			2.20
3	●			●				1.51
4	●	●						1.00
5		●		●				0.80
R			●			●		-4.10

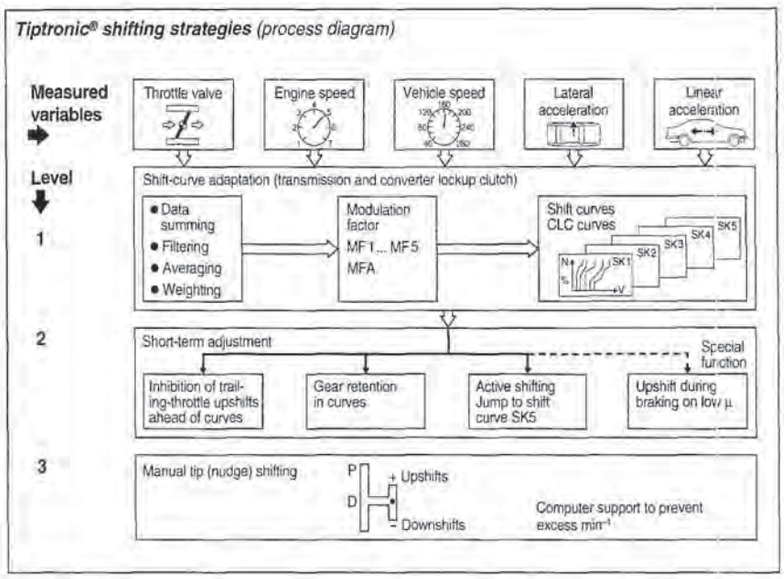
○ according to operating mode



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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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.

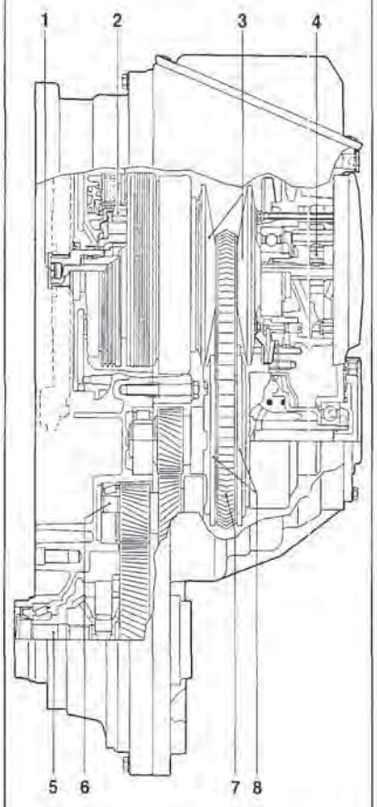
Continuously-variable transmissions

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).

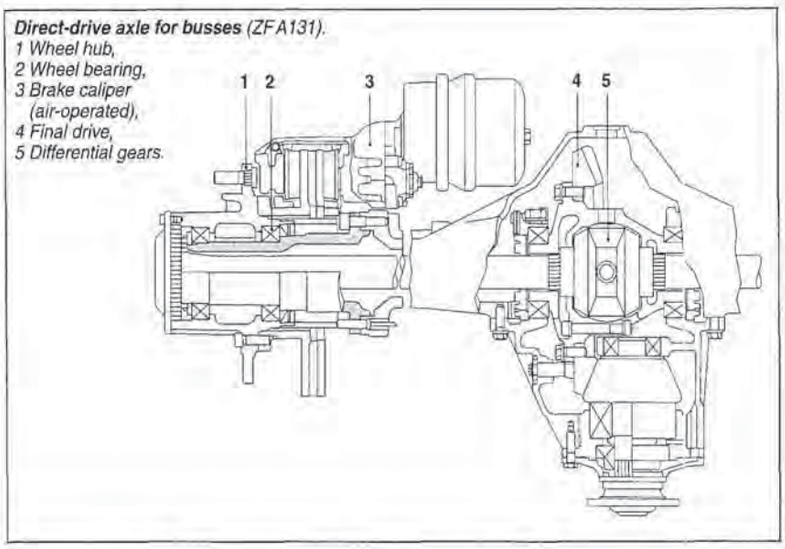
The continuously variable transmission (CVT) can operate mechanically (belt or friction-roller), hydraulically or electrically. The highest level of development has been achieved with mechanical continuously variable units employing steel belts (as installed in low-output series-production cars), where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

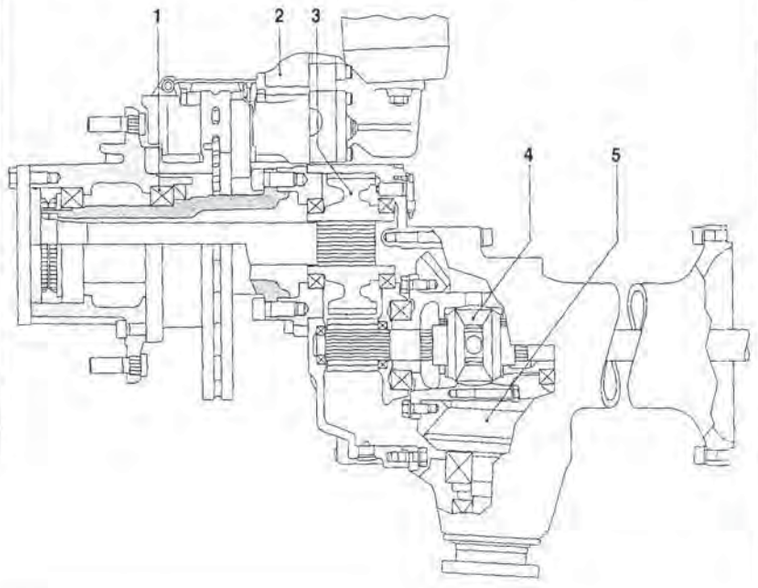
In addition to its torque transmission capabilities, mechanical efficiency and weight, the noise-producing characteristics of a final drive unit have become a decisive criterion in modern automobile development. In this regard, the crown wheel and pinion are of primary significance in terms of noise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

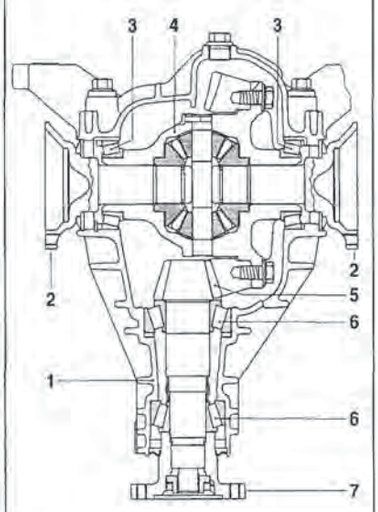
The differential unit compensates for discrepancies in the respective rotation rates of the drive wheels: between inside and outside wheels during cornering, and between different drive axles on AWD vehicles.

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.

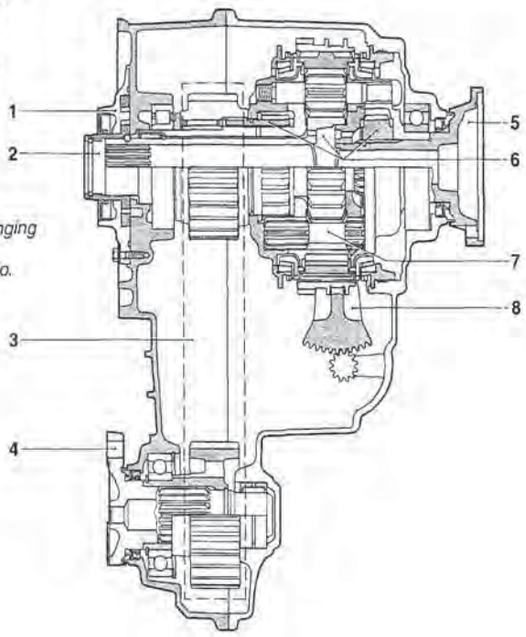
This undesirable effect can be eliminated by locking the differential either by a positively interlocking or friction mechanism. Positively interlocking differential locks are switched-in by the driver. Their disadvantage lies in the stress applied to the drivetrain which occurs during cornering. Friction-type differential locks operate automatically using friction plates, cones or a combination of worm and spur-gear drives and thus have a variable locking action depending on torque. The friction-type locking action can also be achieved with a viscous coupling and is then dependent on the differential speed.

The importance of self-locking differentials is declining. They are increasingly being displaced by electronic systems which slow down a spinning wheel by applying the brake and thereby transfer power transmission to the wheel with more grip (e.g. Traction Control System (TCS)).

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



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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

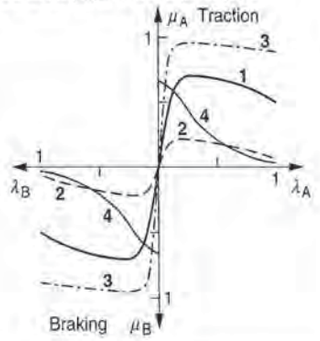
During standing-start and moving acceleration and under braking, the efficiency with which forces can be transferred to the road depends upon the traction available between tires and road surface. The adhesion/ slip curves for acceleration and braking display the same basic pattern as each other (see 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).

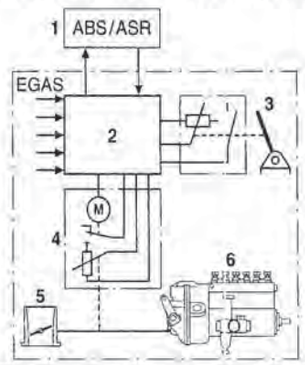
Traction-control systems for passenger cars

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.



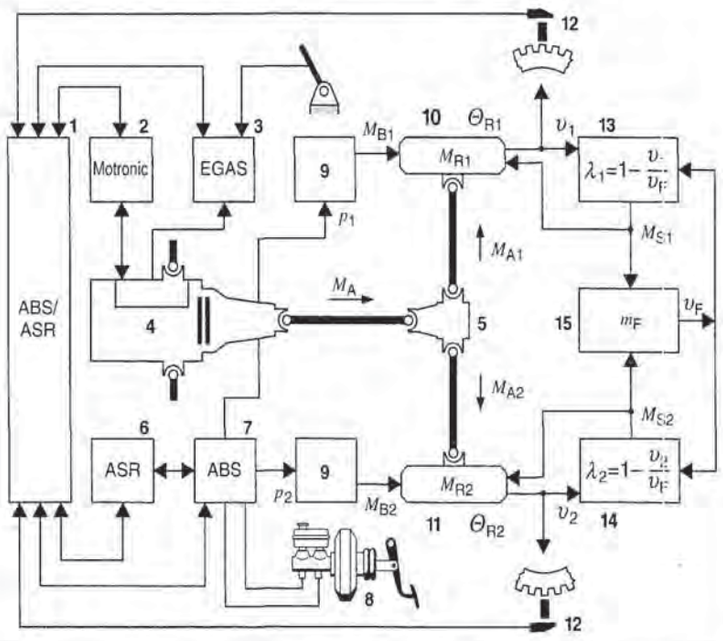
EMS electronic engine-power control for TCS
1 ABS/TCS control unit, 2 ETC control unit, 3 Accelerator pedal, 4 Servomotor, 5 Throttle valve, or 6 Diesel injection pump.



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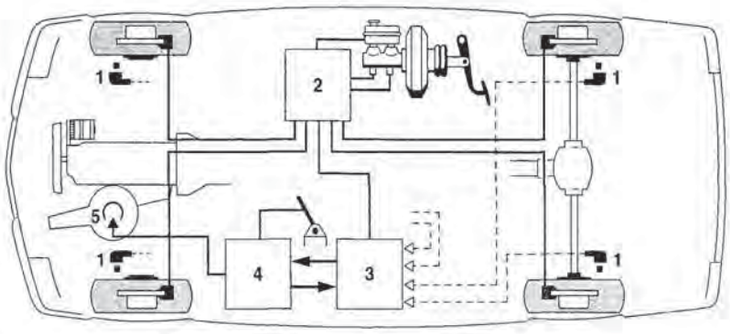
ABS/TCS schematic diagram for cars

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.

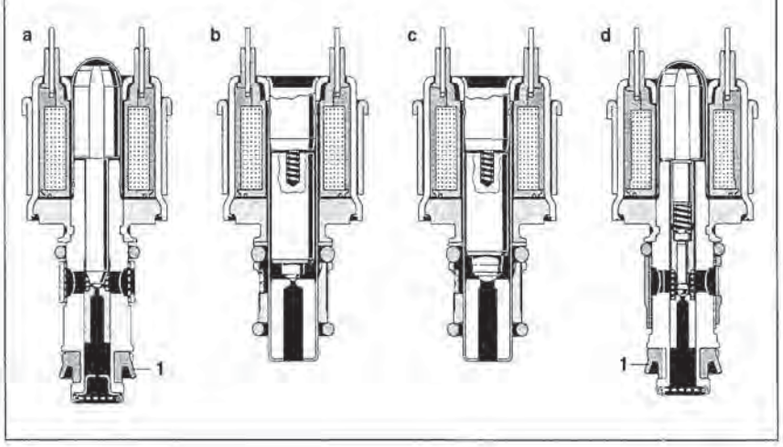
The engine-control functions remain largely independent of the hydraulic system, and the current design was adopted for ABS/TCS5 without modification.

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.

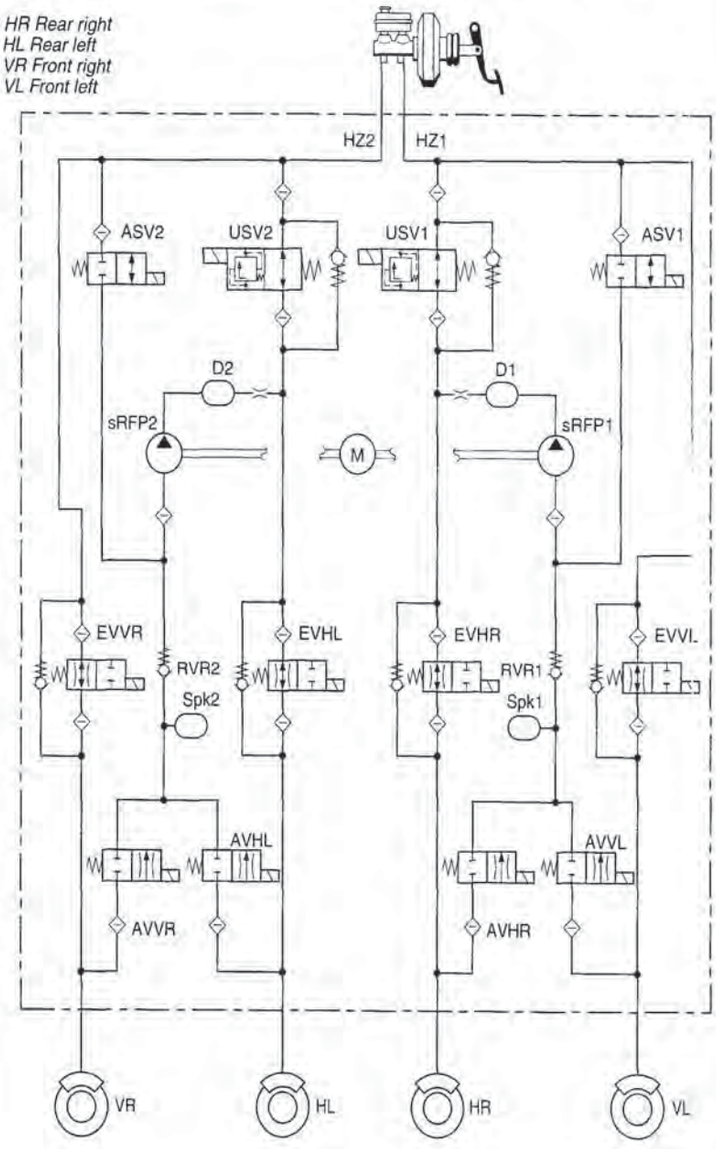


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Schematic diagram of ABS/TCS5 hydraulics for X brake-circuit configuration

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



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.

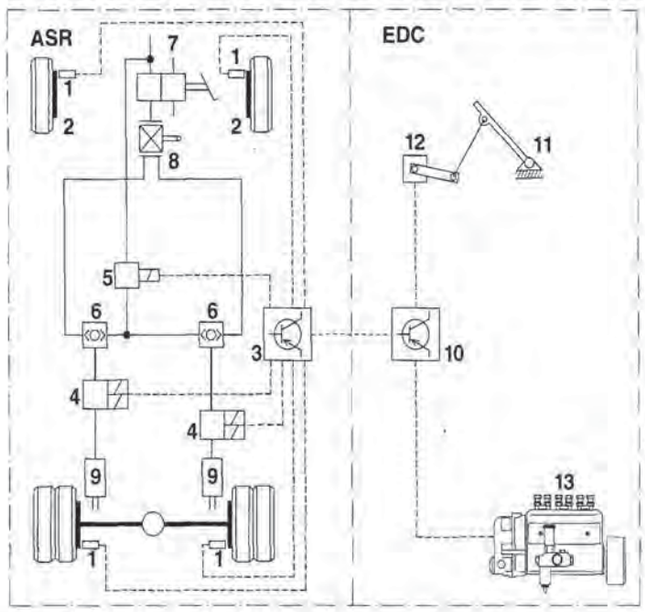
The following section illustrates the principles involved by describing two options for electric control:

- 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.

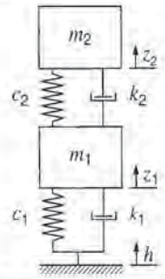


Table 1.
Effects of design parameters on the vehicle's vertical oscillations

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:

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

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

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

Both types of motion are characterized by specific frequency relationships in the form of amplitude ratios.

Table 1 enumerates the relative effects of parameter variations in a dual-mass model (also applicable to actual vehicles).

Body-mounted springs and dampers also affect pitch and roll of the vehicle body as well as its vertical vibration characteristics.

Pitch: Gyration around the vehicle's transverse axis, such as that encountered under acceleration from rest (front-suspension expansion and rear-suspension compression). Pitching movements under acceleration or braking are reduced by the chosen suspension kinematics (suspension-link configuration).

Roll: Gyration around a longitudinal axis which generally runs from the lower front to the upper rear of the vehicle; roll motion occurs in response to steering inputs (suspension compresses at the outside and expands on the inside of the curve). Stabilizers (anti-roll bars) at front and rear axles reduce this effect.

Suspension design elements

(See Table 2)

Controlled suspension systems

Level-control system

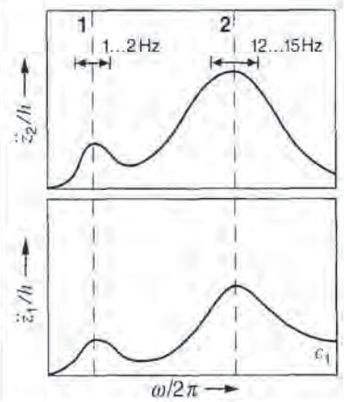
Partially loaded systems

The use of soft springs (comfort) results in long spring travel, such as that encountered with the vehicle loaded. In order to maintain vehicle-body height at an acceptable level, auxiliary air springs or hydropneumatic springs are employed.

The springing element is provided by the volume of the gas, while vehicle level is monitored mechanically directly at the suspension. Valves are used to control the input or output of the air or hydraulic fluid to/from the suspension element; the

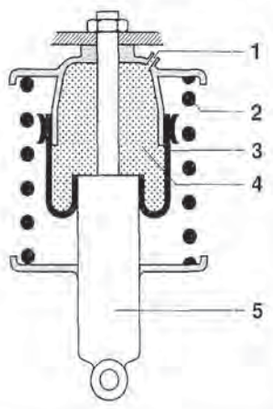
Effect of frequency on kinematic amplitudes

$\ddot{z}_1$ Amplitude of axle acceleration, $\ddot{z}_2$ Amplitude of body acceleration, h Excitation amplitude. 1 Natural body frequency, 2 Natural axle frequency.



Air-suspension level-control system (partially loaded system)

1 Air connection, 2 Steel spring, 3 Auxiliary air-spring element, 4 Gas volume, 5 Shock absorber.

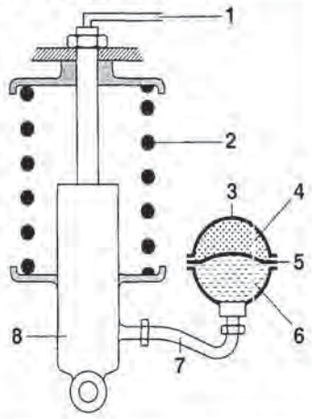


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system can also incorporate intermediate electronic level-control units acting upon solenoid valves.
Advantages of the electronic system:
- Reduced energy consumption achieved by avoiding control cycles during braking, acceleration and in curves,
-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

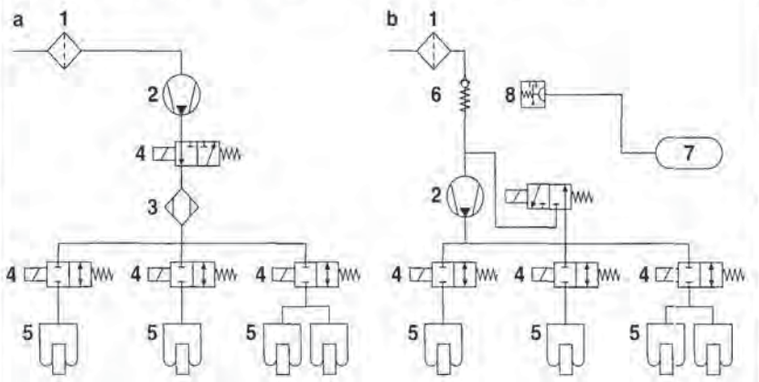
Additional advantages for commercial vehicles:
- Automatic limitation of suspension travel in order to accomodate interchangeable bodies and containers,
- Vehicle-body height can be adjusted as desired, e.g., to align load floor with loading ramps,
- Control of lifting axles (ride height raised automatically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

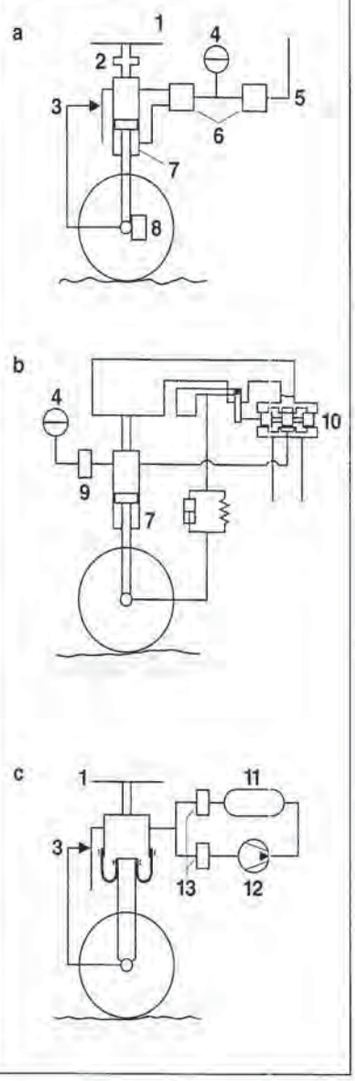
Active suspension controls both the "springing" and the "damping" functions. Various types of design concepts have been realized.

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.

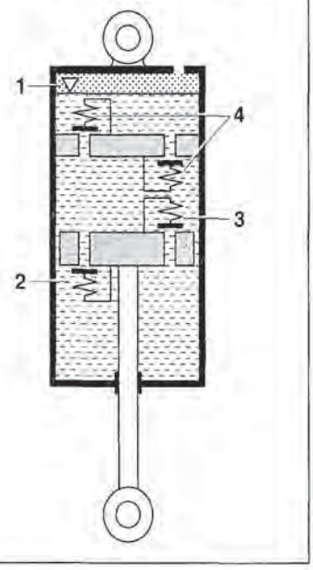
Single-tube shock absorbers

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

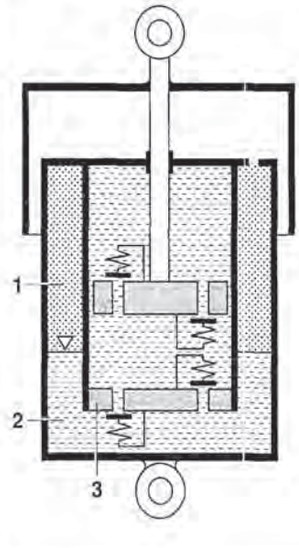
Advantages: Easy to tailor to specific applications, as the large piston diameter allows low working pressures. Sufficient room for valves and passages. Heat is dissipated directly via the outer tube. The shock absorber can be installed in any position.

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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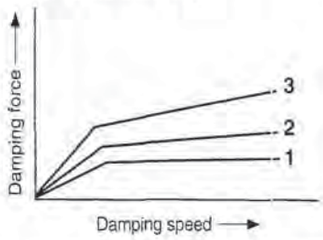
Twin-tube shock absorber

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.



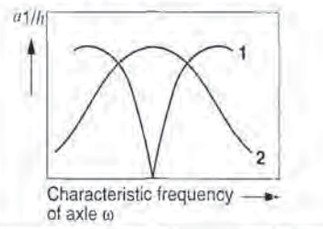
Damping characteristics

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.

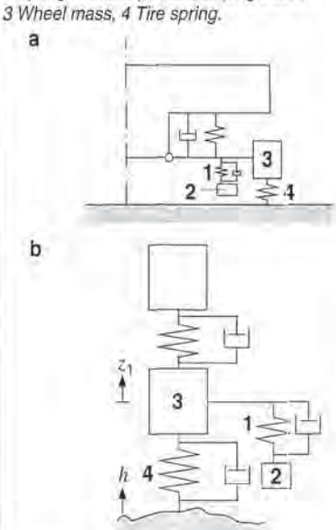


Vibration absorber

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	Element with constant gas volume. *1 Chassis frame, 2 Roll-gaiter air spring, 3 Piston, 4 Compressed air, 5 Mounting plate.*		$\frac{\nu_{loaded}}{\nu_{empty}} = 1$ Natural frequency remains constant when vehicle loaded. Characteristics are dependent on gas properties, piston shape, direction of plies in the gaiter.	Used as suspension strut or spring particularly on commercial vehicles and busses. Increasingly widespread use in passenger cars for level control at rear axle and for suspension on all wheels. Achieves high vertical compliance (increased comfort). Wheel travel must be defined by separate suspension components. Low pressure (<10 bar) demands large volume. Low vertical spring rate not achievable due to geometry of toroid air bellows.
Toroid bellows	Roll bellows	Toroid bellows		

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Hydropneumatic suspension Hydraulic diaphragm accumulator Piston accumulator	Hydropneumatic spring element with constant gas mass. *1 Gas, 2 Oil, 3 Diaphragm, 4 Steel spring.* Hydraulic diaphragm accumulator Piston accumulator 	$\frac{\nu_{loaded}}{\nu_{empty}} = \sqrt{\frac{m_{loaded}}{m_{empty}}}$ 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.

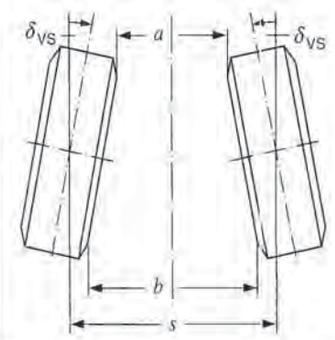
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Suspension linkage

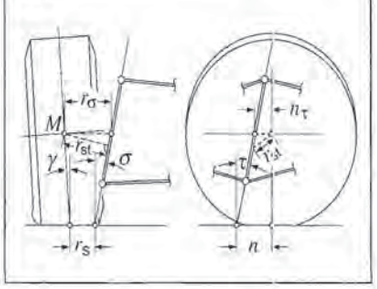
The suspension layout employed to connect the individual wheels to the vehicle structure defines the suspension-linkage design. The linkage enables the wheel for the most part to move vertically in order to compensate for irregularities in the road surface. In addition, the front wheels are steered, 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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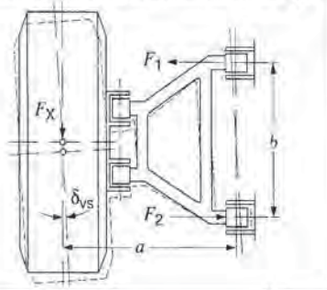
ences the camber change as a function of the steering angle, as well as the self-centering characteristics of the steering.

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.

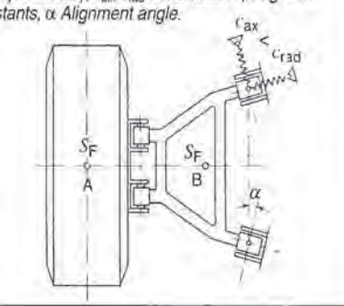
Camber angle γ
Camber angle is the inclination of the wheel axis towards the road surface in the vertical plane. The camber angle is negative when the top of the wheel is inclined toward the center of the vehicle. It influences lateral control (today's vehicles generally employ moderate cambers of –3...0°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

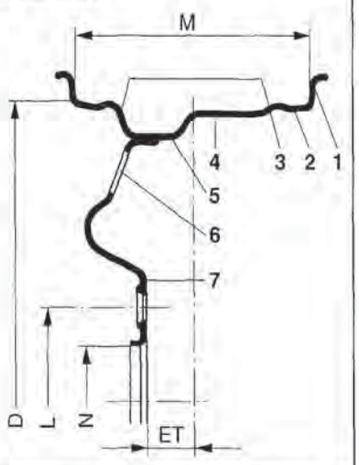
Rim designs
Rims differ from one another (depending upon the type of tire) in terms of number of parts and cross-sectional rim shape. The most important rim details are: rim flange, rim bead seat and rim base. Rims have one of the following cross-sectional shapes:
- Drop center,
- Flat base,
- 5° tapered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

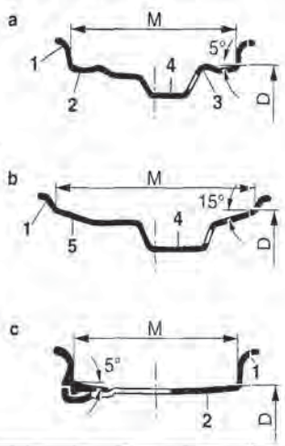
Passenger-car wheels

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 lug nuts whose collars have a special shape which conforms to the wheel which is usually centered. A high level of true-running is achieved by means of a central wheel mount at the wheel hub using an alignment shoulder. A central wheel mount designed as a Rudge hub or incorporating driving pins is only used with racing wheels.

Wheel trims are fixed onto the wheels with clips or (less frequently) by the wheel nuts in order to

– improve visual appearance,
– improve the drag coefficient,
– cool the wheels/and or the brakes.

Traditional materials for wheel covers are steel and aluminum. Plastics have re-

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cently been used much more frequently in order to reduce weight and cost.

Commercial-vehicle wheels

The primary requirements to be met by commercial-vehicle wheels are:
- high fatigue strength and long service life in order to maximize traffic safety,
- lowest possible wheel weight because the wheel, as an unsprung, rotating mass, influences the oscillation system "vehicle",
- high load capacity via appropriate wheel shape and use of optimum materials,
- reduction in wheel-disk unevenness,
- reduction of radial and lateral run-out and wheel imbalance,
- easy assembly of tire to rim during manufacture and in practical use.

15° tapered-bead seat rim
A modern technical design is the one-piece 15° tapered-bead seat rim for tubeless commercial-vehicle tyres.

Advantages:
- The one-piece wheel allows a reduction in wheel weight of up to 10% as compared to the two-piece rim, and exhibits improved true-running and less lateral run-out,
- increased rim diameter,
- sufficient free space,
- unified valve in a precise position and at a sufficient distance from the brake drum or caliper,
- unified balance-weight shapes,
- introduction of central centering and
- use of fully- or semi-automatic tire-fitting equipment.

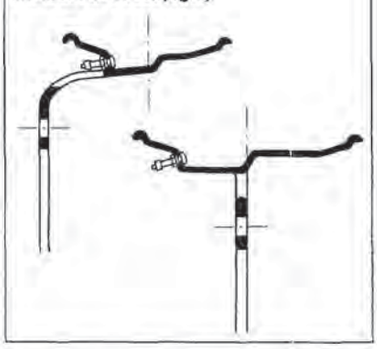
Radial and lateral run-out, imbalance
Radial run-out of the wheel is one of the main causes of vehicle vibrations. With today's commonly used 15° tapered-bead seat, the reduction of the permissible radial run-out to 1.25 mm (peak-to-peak) results in a marked improvement compared to the flat-bed rims. As opposed to radial run-out, lateral run-out is a less critical problem. Wheel imbalance is also not nearly as much of a problem as tire imbalance, which is of a greater magnitude. Maximum permissible static imbalance is limited to 2000 cmg.

Wheel centering
Lug centering via spherical washers or solely via ball-seat nuts has been replaced by central centering in order to reduce excessive radial run-out; this arrangement also permits different maximum and minimum play depending upon wheel size (especially with 22.5" wheels). This requires tolerances which are as close as possible.

Flatness of the contact surface
Any unevenness of the wheel contact surface (waviness, inclination, shielding, etc.) is transmitted to the brake drum when the wheel nuts are tightened, causing fluctuations in braking force as the wheel rotates. These fluctuations, in turn, cause vibrations in the steering system. Driving tests have led to the establishment of a maximum value for waviness of 0.15 mm and 0.2 mm for inclination from the center of the wheel outward.

Trilex wheel system
The Trilex wheel system with cast steel "spoked wheels" comprises a rim and spider. The removable rims are connected to the spider by means of clips and screws. In order to simplify mounting and dismounting of tube tires, the rim consists of three parts with transverse pitch. The same spider can be used in conjunction with the so-called Tublex rim for mounting tubeless tires.

MONOLEX wheel with rim offset (left) and without rim offset (right)



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A more recent development is the one-piece cast MONOLEX 15° tapered-bead seat wheel which in comparison with optimized-weight steel wheels is neutral in terms of weight effect.

Light-alloy wheels

One reason for the use of light-alloy wheels is the reduction in weight for 15° tapered-bead seat wheels. Light-alloy wheels are available with cast or forged rims. In spite of weight reduction and adequate strength characteristics, for reasons of cost, light-alloy wheels are used on commercial vehicles only in special cases.

Commercial-vehicle wheel loading

Prestress

Prestress occurs through the combination of stresses caused by assembling the wheel with those which result from inflating the tire.

Static nominal wheel stress

If the wheel is allowed to roll slowly on a perfectly flat road surface under static nominal wheel load, the stress in the wheel section in question varies periodically as the wheel rotates.

Additional dynamic stresses

These similar additional stresses are caused by dynamic wheel forces which result as the vehicle is driven straight ahead over an uneven road surface. Quasi-static wheel forces are then produced which occur during maneuvers such as cornering, steering while stationary, braking and accelerating.

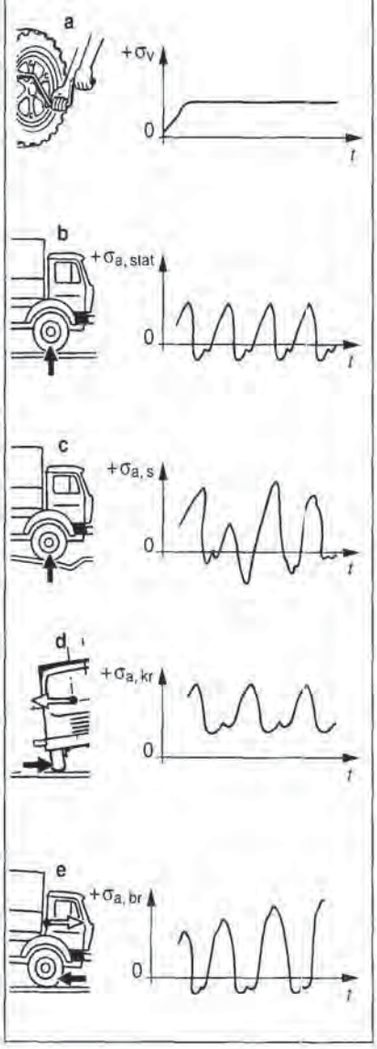
The resulting group of stresses caused by the above-mentioned types of wheel loading is used today as the basis for wheel dimensioning and testing.

Significant weak points

The highly stressed parts of the wheel which are thus most susceptible to cracking are the disc flange, the venting holes, the welded joint between disc and rim, and the rim-well radius.

The flange joint is particularly susceptible. Hub-centered wheels most often crack tangentially above the centre hole. With bolt-centered wheels, the cracks generally spread out radially from the bolt holes.

Types of wheel loading and stress distributions
a) Prestress σ_v
b) Static wheel load $\sigma_{a,stat}$
c) Straight-ahead travel over bumps $\sigma_{a,s}$
d) Cornering $\sigma_{a,kr}$
e) Braking $\sigma_{a,br}$



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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-

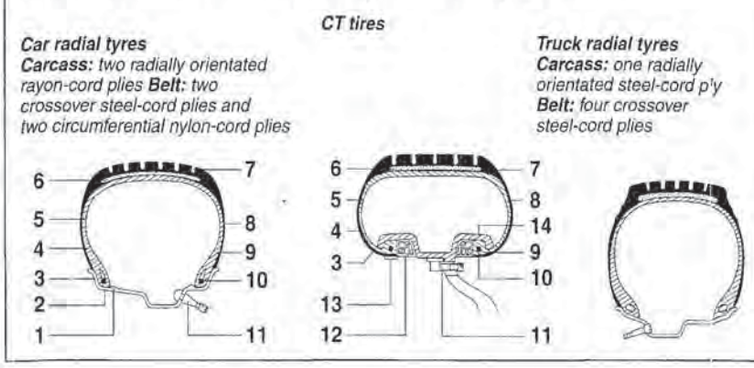
[1]) The corresponding European standards can be found in the "Data Book of Tires and Rims" of the ETRTO (European Tire and Rim Technical Organization, Brussels).
[2]) Guideline of the Economic Association of the German Rubber Industry, Frankfurt.

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 cm³ or attaining speeds in excess of 40 km/h).

Tires tested for compliance with ECE regulations are identified by the code molded into the sidewall adjacent to the bead. The code consists of a circle containing a large "E" followed by the code number of the responsible agency, which is in turn followed by an approval number.

Example: (E4) 020 427.

The small "e" which can be molded into the sidewall in accordance with the European guidelines 92/23 is of equal significance to the ECE marking

Example: (e4) 00321.

Appropriate European guidelines are being prepared for agricultural machinery.

The tire's width, its construction (R = Radial; "—" = Cross-ply; B = Bias-belted), and its rim diameter represent the minimum information requirement for tire designation. The diameter of the tire is also usually included on tires for industrial trucks. On tires for two-wheeled vehicles, passenger cars and heavy commercial vehicles, this information is frequently supplemented by the *H/W* aspect ratio in % which is appended directly behind the width information from which it is separated by a slash. ECE regulations require this information for all new tires. Although it is not stipulated by the ECE Directives, tires for passenger cars and two-wheeled vehicles may also feature a speed-rating code letter behind the *H/W* aspect ratio or tire width. On cross-ply tires the code letter replaces the horizontal line. On VR, VB, ZR and ZB tires, it is mandatory that the code letter forms an integral part of the tire-size designation.

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The PR (Ply Rating) follows the tire size, and is now employed as a code to distinguish the load ratings for various versions of the same size. Originally it indicated the number of plies in the tire carcass.

The service description is an additional suffix combining the load index (LI) and the speed-rating symbol (GSY). ECE Directives prescribe it as a replacement for the PR number or the speed-rating letters in the tire size. Specific values are assigned to the codes in the service description (Table 4).

For passenger-car tires:
Rated speed = Top speed. On vehicles whose design only allows top speeds of 60 km/h and below, the tire load ratings may be increased as shown in the table below so that higher loads can be carried.

Table 3. Load-rating increase

km/h	Load-rating increase in %	Air pressure increase in bar
60	10	0.1
50	15	0.2
40	25	0.3
30	35	0.4
25	42	0.5

Reductions in tire load can be traded for higher top speeds on most tires for motorscooters and heavy commercial vehicles. Conversely, both types of tire may be used to carry higher loads throughout virtually the entire speed range when operated below their rated (reference) speed, provided that the vehicle be specifically designed to operate at the lower maximum speed. Again, increased maximum load ratings are approved for tires on passenger-car trailers for speeds up to 100 km/h, and on certain commercial vehicles used for short-distance haulage and transportation.

The correct inflation pressure for a particular tire size and PR rating or service description are specified in the standards and/or the manuals provided by the tire's manufacturer (Table 1). The suffix containing the speed-rating symbol always indicates the actual rated speed.

Special passenger-car spare tires (low-weight and "space-saver" designs) are identified on the sidewall as intended for temporary use at limited speeds.

Correspondence between the speed rating for M+S tires on passenger cars, heavy commercial vehicles and motorcycles, and the respective maxima of the vehicles upon which they are mounted is not mandatory. However, a sticker indicating the lower speed for which the tires are approved must be affixed to the inside of the vehicle within the driver's field of vision. The following categories of tires may feature additional data prescribed by US highway safety legislation. These data, molded into the sidewall adjacent to the bead, are valid for Canada, and are also used in Israel:
- FMVSS 109 for passenger-car tires,
- FMVSS 119 for two-wheeled and heavy commercial vehicles.

These data are stamped adjacent to the letters "DOT", following which come the tire identification code and the date of manufacture, as well as further data on maximum load rating, maximum inflation pressure, and the cord plies which make up the casing and belts.

Australian Safety Regulation ADR 23, which applies to passenger-car tires, employs the identification codes from FMVSS 109 and ECE-R30.

Table 4. Service description codes (examples)

Load index

LI	50	51	88	89	112	113	145	149	157
kg	190	195	560	580	1120	1150	2900	3250	4125

Speed symbol

SS	F	G	J	K	L	M	N	P	Q	R	S	T	H	V	W	Y
km/h	80	90	100	110	120	130	140	150	160	170	180	190	210	240	270	300

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Table 5. Tire identification examples

Tire category		Identification example				Example includes details of			
		Tire designation symbols	PR[3] number	Usage code [4] LI	[5] GSY	Tyre dia. *A*	Tire width *B*	*H/W* %	Rim dia. *d*
	Suitability								
MC	Moped	2¼–16 Moped	–	–	–	–	Code	–	Code
	Small motor cycles	3–17 reinforced[2]	–	51	J	–	Code	–	Code
	Motorcycles	3.00–17 reinforced[2]	–	50	P	–	Code	–	Code
		110/80 R 18	–	58	H	–	mm	80	Code
		120/90 B 18	–	65	H	–	mm	90	Code
	Motor scooters	3.50–10	–	51	J	–	Code	–	Code
Passenger cars		165 R 14 M+S	–	84	Q	–	mm	–	Code
		195/65 R 15 reinforced[2]	–	95	T	–	mm	–	Code
		200/60 R 365	–	88	H	–	mm	60	mm
		205/60 ZR 15	–	91	W	–	mm	60	Code
		CT 235/40 ZR 475	–	–	–	–	mm	40	mm
CV	Delivery vans	185 R 14 C[1]	8 PR	102/100	M	–	mm	–	Code
	Light-duty trucks	245/70R 17.5	–	143/141	J	–	Code	–	Code
	Trucks	11/70 R 22.5	–	146/143	K	–	Code	70	Code
	Trailers	14/80 R 20	–	157	K	–	Code	80	Code
	Buses	295/80 R 22.5	–	149/145	M	–	mm	80	Code
MPV	Multi-purp. veh.	10.5 R 20 MPT[8]	14 PR	134	G	–	Code	–	Code
EM	Transporters	18.00–25 EM[9]	32 PR	–	–	–	Code	–	Code
	Loaders	29.5–29 EM[9]	28 PR	–	–	–	Code	–	Code
IT	Industrial trucks	6.50–10[6]	10 PR	–	–	–	Code	–	Code
	Carts	21 x 4[6]	4 PR	–	–	Code	Code	–	–
	Industrial trucks	28 x 9–15[7]	14 PR	–	–	Code	Code	–	Code
		300 x 15[7]	18 PR	–	–	–	mm	–	Code
AS	Tractors	480/70 R 34	–	143	A 8	–	mm	–	Code
		7.50–60 AS[10] Front	6 PR	–	–	–	Code	–	Code
	Equipment[11]	11.0/65–12 Impl.	6 PR	–	–	–	Code	65	Code

Example for passenger-car tires: **175/70R13 82S**
175 mm nominal tire width, **70** % H/W ratio, **R**adial tires, Code **13** Rim diameter (approx. 330 mm [13 inches]), Load index **82** (475 kg), Speed symbol **S** = 180 km/h.

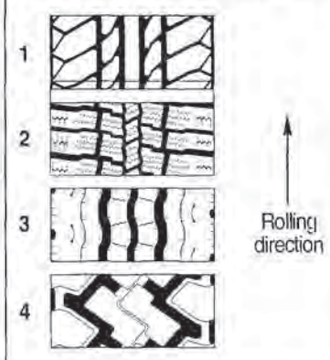
[1]) C = light-duty truck (delivery-van) tires (also for high-load-capacity motor-scooter tires). [2]) reinforced = additional designation for reinforced tires for two-wheeled motor vehicles and passenger cars. [3]) PR = load-range class. [4]) Load-range code for single/dual tires. [5]) Speed code for vehicle nominal (reference) speed. [6]) Pneumatic tires. [7]) Solid rubber tires. [8]) MPT = Multi-purpose tire. [9]) EM = Earthmoving machinery. [10]) AS = Agricultural tractor. [11]) Tires for equipment and trailers.

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Tire applications

Tire selection based upon the recommendations of the tire manufacturer is essential for obtaining satisfactory performance. Optimal operating characteristics can only be obtained when tires of one single design are mounted on all wheels (for instance, radial tires). This is obligatory for private cars. For commercial vehicles with a gross weight of over 2.8 t, tires on the same axle must be of identical design. During seasonal storage of the tires, inner tubes and bead bands tend to age and become brittle more rapidly when exposed to direct sunlight. Moving air promotes this process. Intact packaging is particularly important for ensuring that the tube remains in good condition. The storage area should thus be cool, dry and dark. Any contact with oil or grease is to be avoided. Particular care is required when mounting tires. They must never be mounted on anything other than undistorted, undamaged, rust-free wheel rims which show no signs of more than minimal wear. When a loose flange is used, that side should receive especially critical attention. New valves, and, where applicable, new tubes and bead bands are always to be installed with new tires. Caution is also advised when used inner tubes are refitted following repairs: Inner tubes expand during use, a condition which can result in dangerous folds forming when the tube is reinstalled. If there are any doubts at all, new tubes should be used.

Tire-tread examples
1 Passenger-car tire, 2 Passenger-car M+S tire, 3 Commercial-vehicle tire, 4 Commercial-vehicle tire, high-traction tread design.



Tire tread (see examples)
It is illegal to regroove the tread on tires for two-wheeled vehicles and passenger cars; manufacturer regulations must be complied with for other tire categories.
Tire rotation is recommended to deal with variations in tread wear between axles. The less tread there is on the tire, the thinner is the protective layer covering belts and casing. This aspect should be considered when planning operation under demanding operating conditions extending over a long period of time. In addition, reduced tread depth results in a more than proportional increase in braking distances on wet road surfaces. This applies in particular to passenger cars and fast commercial vehicles. For instance, for a light FWD passenger car and a heavy rear-wheel drive car, the figures apply in Table 6 below when the vehicle is braked from 100 km/h (highly dependent upon the road surface, tire profile, and the tire's rubber mixture).

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

Vehicle		Light passenger car with front-wheel drive					Heavy passenger car with rear-wheel drive (ABS)			
Tread depth	in mm	8	4	3	2	1	8	3	1.6	1
Braking distance	in m	76	99	110	129	166	59	63	80	97
	in %	100	130	145	170	218	100	107	135	165
Increased braking distance per mm of wear	in %	7	15	25	48		1.4	20	50	

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Tire traction

Units and symbols

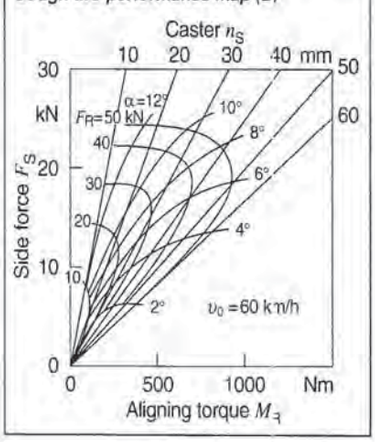
Symbol		Unit
f	Frequency	Hz
F_B	Braking force	kN
F_R	Wheel load	kN
F_S	Side force	kN
M_R	Aligning torque	N · m
n_S	Caster	mm
p_i	Tire pressure	bar
v_0	Test velocity	km/h
α	Slip angle	°
γ	Camber angle	°
λ	Slip	–

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.

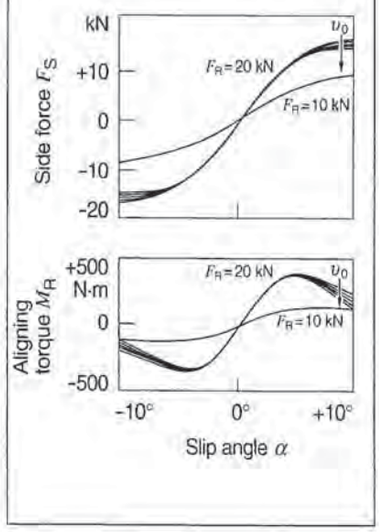
Representative tire-performance maps for passenger-cars and light commercial vehicles are already familiar and are available in various publications [1, 3, 4]. For this reason, the following will concentrate on tires for heavy-duty commercial vehicles with the dimensions 11 R 22.5. These tires are in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

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

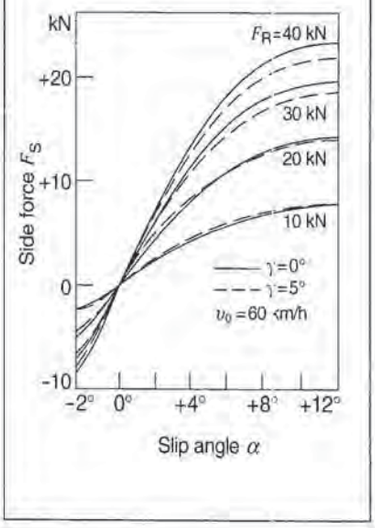
***Gough tire-performance map** (B)*



***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).

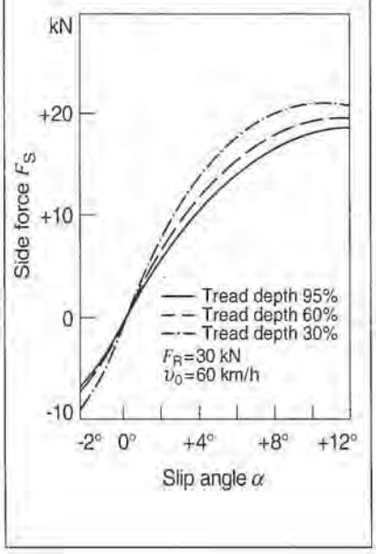
When dealing with tires for passenger-cars and light commercial vehicles, the situation is thus: If a wheel rotating at a given slip angle is subjected to camber, the camber and side forces cause a parallel displacement of the side-force/slip-angle curves. Tires on heavy-duty commercial vehicles also 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).

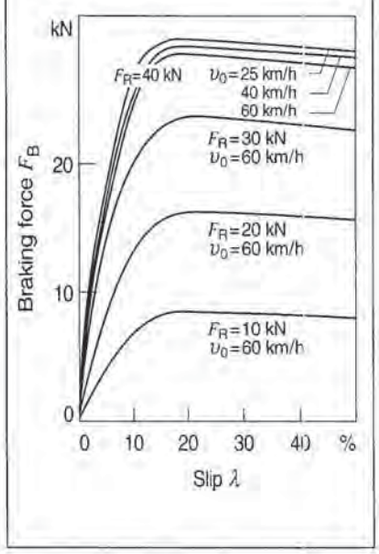
Tire performance under acceleration and braking with wheel rolling straight ahead

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.

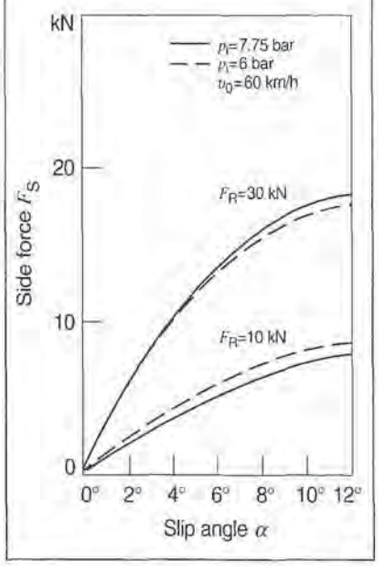
The tire-performance curves resulting from measurements actually taken, indicate the propagation of side force as a function of braking force within the slip-angle range of 0...10°. The parameters for wheel load, speed and tire pressure remain constant (Diagram K).

Dynamic tire-performance curves

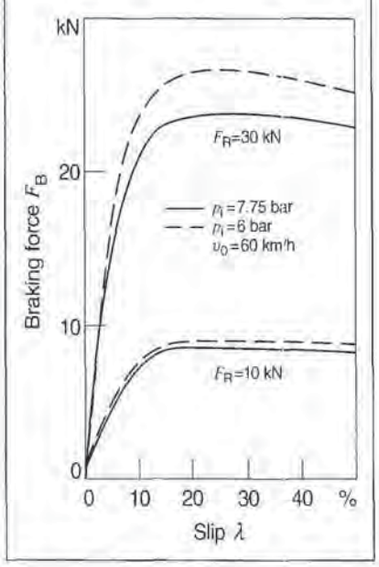
The illustrated tire-performance curves are based on parameters which are only subject to gradual change during the course of the measurements, i.e. quasi-static conditions. Actual operation, on the other hand, is characterized by dynamic processes. Increases in the speed at which the influencing parameters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

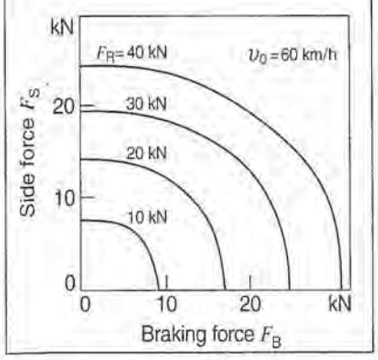


Braking force vs. slip on heavy-duty commercial-vehicle tires *(G)*
Parameters: Tire pressure and wheel load.

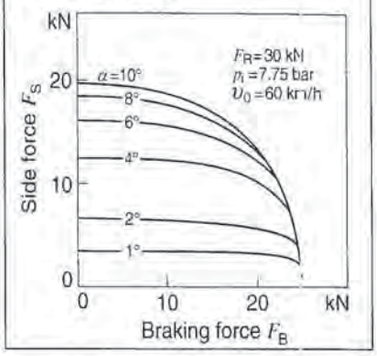


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Lateral force vs. braking force *(H)*
Maximum achievable adhesion at constant wheel load.



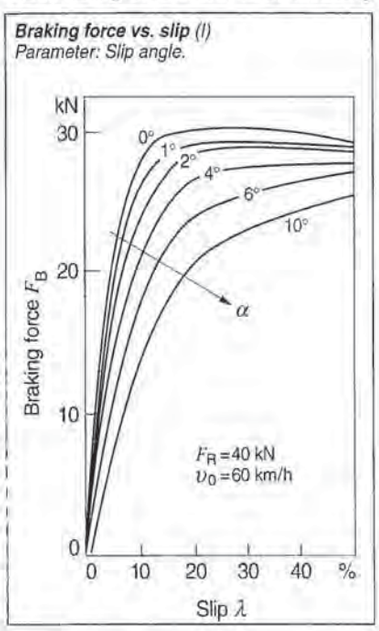
Tire-performance curves measured at a wheel load of 30 kN *(K)*
Side force as function of braking force, Parameter: Slip angle.



– 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

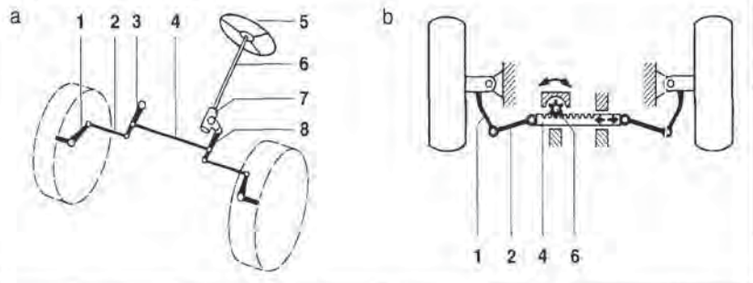
The requirements in terms of steering behavior can be summarized as follows:
1. Jolts from irregularities in the road surface must be damped as much as possible during transmission to the steering wheel. However, such damping must not cause the driver to lose contact with the road.
2. The basic design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

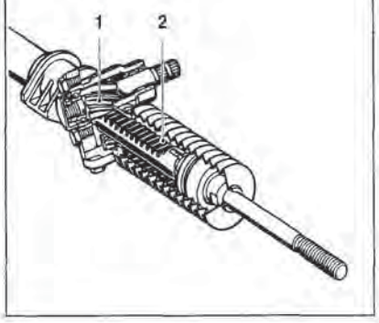
The steering-axis inclination causes the front section of the vehicle to lift when the wheels are at an angle. This leads to a steering-angle-dependent caster return torque.

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.

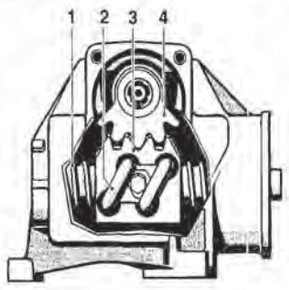
The caster produces a lever arm for lateral forces: i.e., a speed-dependent return torque.

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.

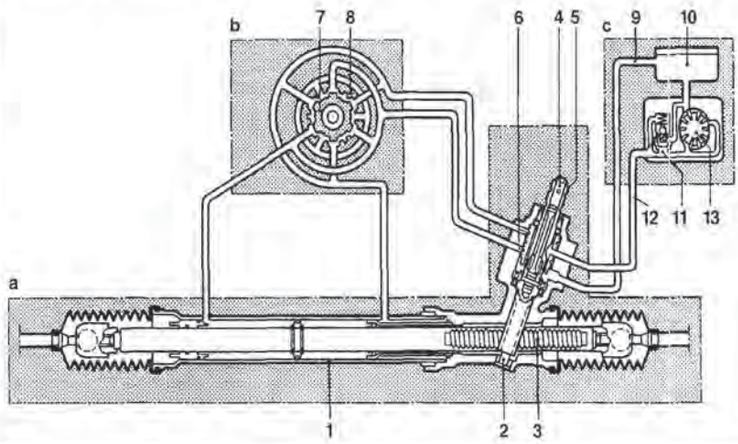
The pump and the system components must be designed such that the operating temperature of the hydraulic fluid does not rise to an excessive level (<100 °C) and such that no noise is generated and the oil does not foam.

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.

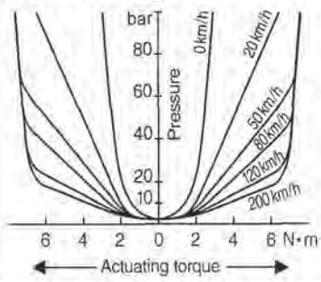
Steering cylinder

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.

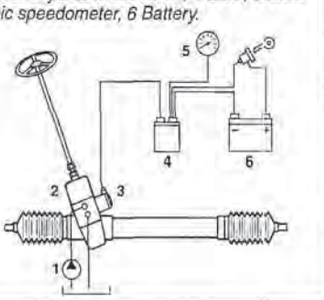
Parameterizable power-assisted steering *(characteristic curves)*
The characteristic curve form can be adjusted to the vehicle data.



Electric power-assisted steering

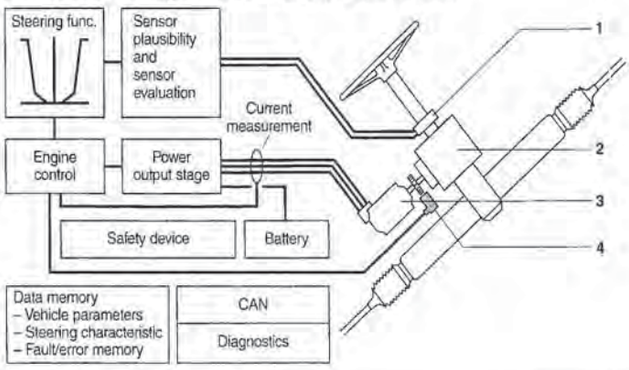
The SERVOLECTRIC (Bosch/ZF designation) is currently under development for medium-sized and smaller cars. This electric power-assisted steering system features an electric motor which is powered by the vehicle's electrical system and can be fitted in one of three possible arrangements either as a steering-column,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

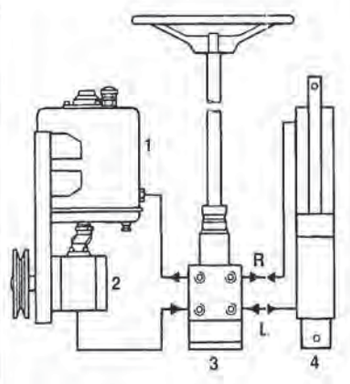
The maximum permissible speed is 25 km/h in many European countries; in Germany it is 50 km/h and, with dual-circuit design, could be increased to 62 km/h.

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



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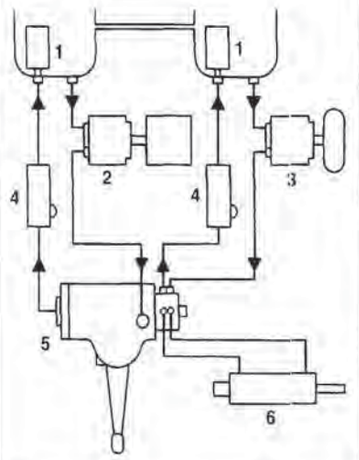
ing circuits of such a system are functionally tested by means of flow indicators. The steering-circuit supply pumps must be driven in differing ways (e.g., engine-dependent, vehicle-speed-dependent or electrically). In accordance with legislation, if the engine or one of the steering circuits fails,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

Single-circuit power-assisted steering system for commercial vehicles

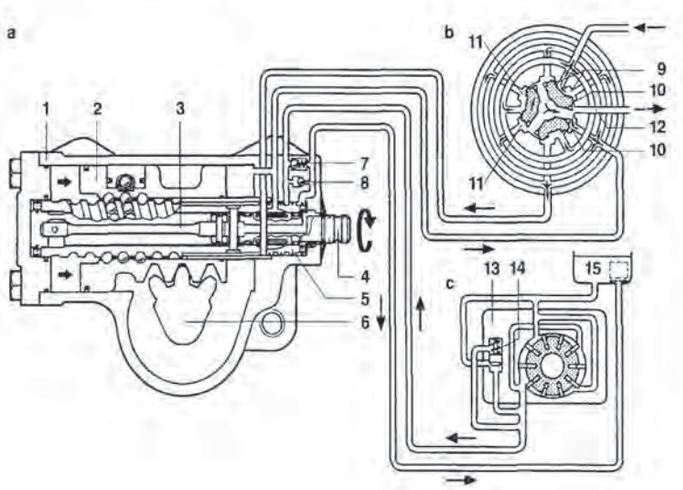
Commercial vehicles are usually fitted with recirculating-ball power steering. In modern systems the control valve is integrated into the steering worm, which leads to compact design and optimum weight.

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.

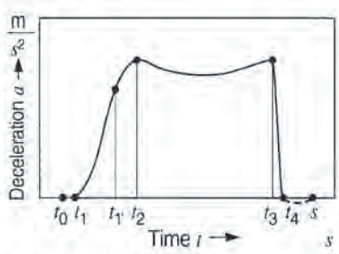
Application force F_s: In friction brakes, the total force applied to a brake lining, which causes the braking force by friction effect.

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_{mt}) over subdivision $t_3 - t_2$ of the fully developed deceleration.

$$a_{mf} = \frac{1}{t_3 - t_2} \cdot \int_{t_2}^{t_3} a \cdot dt$$

Braking factor z
Ratio between the total braking force, F_t, and the static weight, G_s, on the axle or the axles of the vehicle:

$$z = F_t/G_s$$

Legal regulations

The testing of braking equipment pursuant to the issuance of the type approval for a vehicle in Germany may take place in accordance with one of the following three groups of regulations as chosen by the vehicle manufacturer. They are:
- National regulations and statutory requirements for the testing of brakes applicable in the country of use,
- Council Directive of the European Communities: Directive 71/320/EEC and the Amending Directives and Annexes or
- ECE Regulation 13, 13H and 78 of the UN Economic Commission in Geneva.

In Europe, the national regulations of individual countries may specify introduction of the EU Directive or ECE Regulations. Essentially, they may be very similar. However, the ECE Regulations 13 and 13H are likely to be more recent and contain, for example, regulations for braking systems with electrical control systems (electronically controlled braking systems).

Braking systems conforming to EU Directives, and ECE Regulations 13 and 78 (for classification refer to P. 717).

Category L vehicles (less than 4 wheels)
Motorized two- and three-wheeled vehi-

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cles must be equipped with 2 mutually independent braking systems. In the case of heavy category L_5 3-wheel vehicles, the two braking systems must both act on two wheels. Additionally, these vehicles must be equipped with a parking braking system.

Category M and N vehicles

The categories M and N vehicles must meet the requirements which pertain to the service braking system, secondary braking system and parking braking system. These 3 braking systems may have common components. Such vehicles must have at least 2 mutually independent control devices.

The apportioning of braking force among the individual axles is prescribed. Vehicles in the categories M_2 and N_2 or higher must be fitted with an antilock braking system (ABS) (some exceptions may still apply for older models).

Additional retarding braking systems can be used in order to fulfill braking requirements on long inclines. Vehicles in category M_3 used for local or long-distance duty must be able to meet the requirements for braking on such "descents" using the continuous-operation braking system only.

Category O trailers

Category O_1 trailers are not required to have a braking system, however, there are requirements governing the safety connection to the towing vehicle. Beginning with category O_2, trailers must be fitted with a service braking system and a parking braking system which may have common components. It must be possible for the parking braking system to be operated by a person standing next to the vehicle. The apportioning of braking force among the individual axles is prescribed.

An antilock braking system (ABS) is prescribed for some trailers in Class O_3 and above.

Inertia braking systems are permissible for trailers up to category O_2.

The trailer must brake automatically if it becomes decoupled from the tractor vehicle while moving, or (for trailers < 1.5 t) it must be equipped with a safety connection to the towing vehicle.

Vehicles equipped with antilock braking systems (ABS)

ABS systems must conform to Appendix X of EU Directive 71/320/EEC or Annex 13 of ECE Regulation 13 (on Category 1 vehicles of classes M_2, M_3, N_2 and N_3). The essential requirements are:

- locking of the directly controllec wheels under braking on any road surface at speeds of over 15 km/h must be prevented,
- operating stability and steering control are to be maintained and
- grip on road surfaces offering the same or (in the case of Category 1) different levels of adhesion to wheels on different sides (μ split) must be utilized, and
- there must be a visual warning system for indicating electrical faults.

Only minimum standards are in force for trailer ABS. The ABS on tractor and trailer should be mutually compatible in order to ensure safety and prevent excessive tire wear.

Tractor vehicles and trailers with compressed-air braking systems

The compressed-air connections must be of the two- or multi-line design. When the service brakes on the tractor unit are operated, the service brakes on the trailer must also be operated with a var able degree of force. If any fault occurs on the service brakes of the tractor unit, that part of the system not affected by the fault must be cable of braking the trailer with a variable degree of force. If one of the lines between tractor unit and trailer is separated or develops a leak, it must still be possible to brake the trailer, or it must brake automatically.

The braking efficiency of the vehicle when fully loaded (and when empty for vehicles without ABS) is specified according to the pressure at the control-line coupling head. The service brakes of the trailer must only be capable of operation in conjunction with the service, secondary or parking brake system on the tractor unit.

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Requirements of StVZO (Germany), EU Directive 71/320 EEC and ECE Regulations

Vehicle class (For classification, refer to P. 717)		Passenger cars & motor coaches M_1	M_2	M_3	Commercial vehicles N_1	N_2	N_3	Trailers O_1	O_2	O_3	O_4
Service braking system		Acting on all wheels Prescribed force distribution to the axles						No braking system or as O_2	Inertia braking system or as O_3		
ABS as per EC Dir. or ECE[1]) ($v_{max} \geq 25$ km/h)		–	+	+	–	+	+	–	–	+	+
Type O test (drive disengaged)											
Test speed	km/h	80	60	60	80	60	60	–	60	60	60
Braking distance	≤m	50.7	36.7	36.7	61.2	36.7	36.7		$z \geq 0.30$, Semitrailer $z \geq 0.45$		
Formula for		$0.1v + \frac{v^2}{150}$	$0.15v + \frac{v^2}{130}$								
Mean fully developed deceleration	≥m/s²	5.8	5.0								
Actuating force	≤n	500	700						at ≤6.5 bar		
Type O test (drive engaged)		Behaviour of vehicle under braking from 30% – 80% v_{max} and braking efficiency									
Test velocity $v = 80\%\ v_{max}$, but	≤km/h	160	100	90	120	100	90	–	–	–	–
Braking distance	≤m	212.9	111.6	91.8	157.1	111.6	91.8				
Braking distance formula		$0.1v + \frac{v^2}{130}$	$0.15v + \frac{v^2}{103.5}$								
Mean fully developed deceleration	≥m/s²	5.0	4.0								
Actuating force	≤n	500	700								
Type I test		Repeated braking at 3 m/s² loaded, drive engaged							Continuous braking, loaded 40 km/h 7% downhill gradient 1.7 km $z \geq 0.36$ and $z \geq 60\%$ of figure measured in Type O test at 40 km/h		
$v_1 = 80\%\ v_{max}$, but $v_2 = {}^1/_2\ v_1$	≤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.

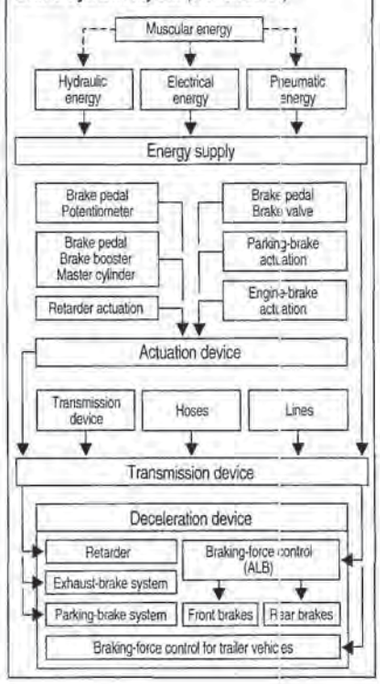
Different ranges of applications for different vehicle types result in highly variegated demands being placed on braking systems. The inescapable result is a multiplicity of highly diversified braking systems, differing from one another in both design and application.

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.

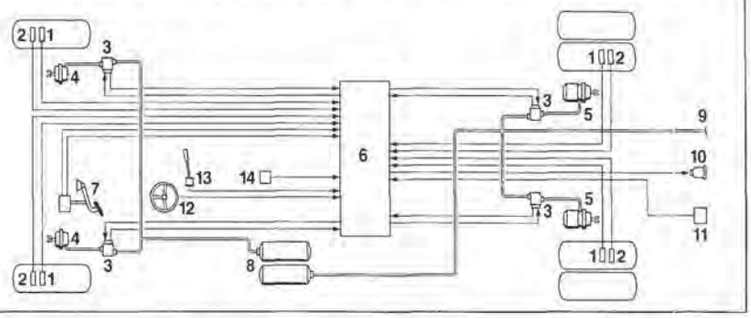
Energy-assisted and non-muscular-energy systems are distinguished not only by energy type, but also by the mediums used to transmit the energy. Pneumatic (vacuum, compressed air) and hydraulic energy are the most common, electrical energy is sometimes also employed.

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

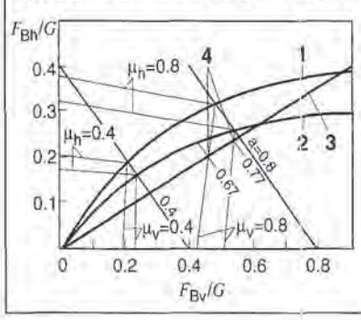
The braking system is designed with reference to both the requirements of the vehicle and the intrinsic imperatives of the system itself. In the case of the vehicle-oriented design, the vehicle's center of gravity and the specified distribution of braking force to the front and rear axles, determine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

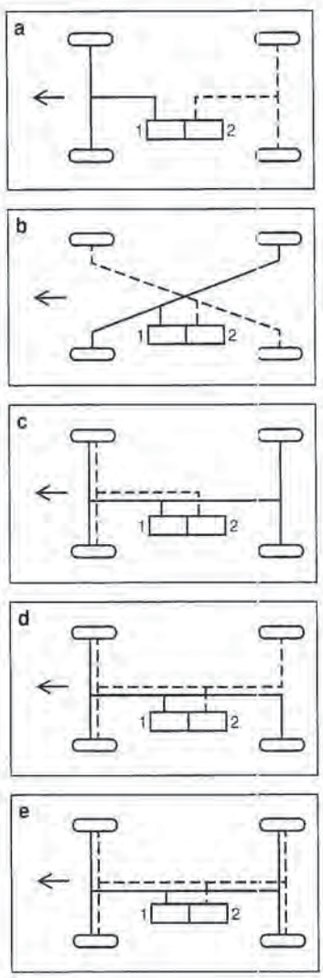
Brake-circuit configurations

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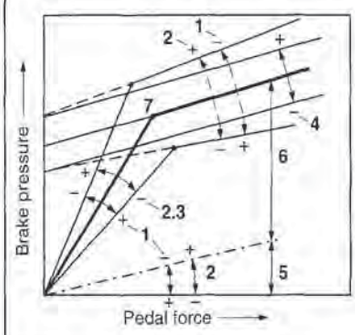
X distribution pattern
Diagonal distribution pattern. Each circuit brakes a given front wheel and the diagonally opposite rear wheel.

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.

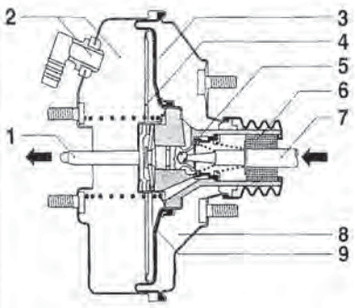
LL distribution pattern
Front-axle and rear-wheel/front-axle and rear-wheel split. Each circuit brakes the front axle and one rear wheel.

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

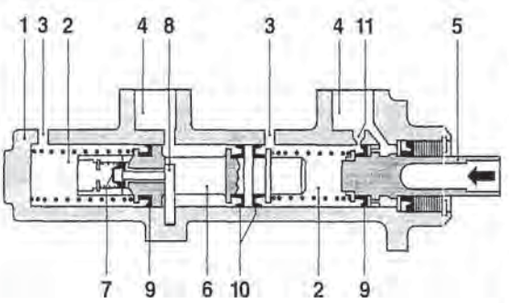
Vacuum-operated brake booster
Effect of design parameters.
1 Master-cylinder surface area, 2 Pedal leverage, 3 Boost factor, pedal leverage, 4 Diaphragm surface area, vacuum level, 5 Effect of pedal force, 6 Effect of servo assist, 7 Output point.



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.

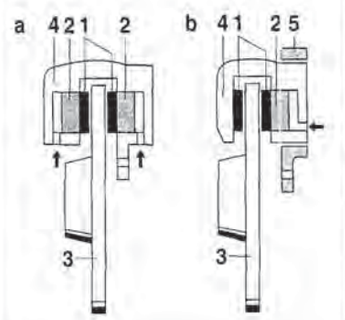
Wheel brakes

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.

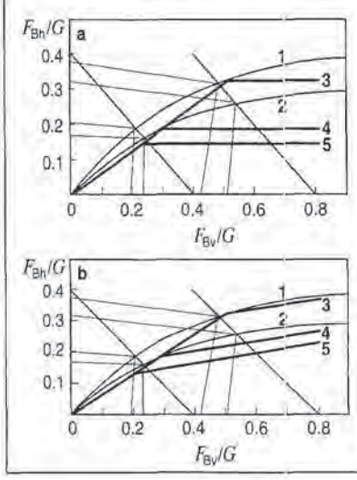
The braking force is apportioned between the front and rear axles in accordance with the dimensions of the particular brakes. It is the job of the metering device to adjust this braking-force apportionment in order to achieve a closer approximation to the ideal distribution, i.e., the parabolic curve.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

ABS antilock braking systems are closed-loop control devices within the braking system which prevent wheel lock-up during braking and, as a result, retain the vehicle's steerability and stability. The main ABS components are:

Hydraulic modulator, wheel-speed sensors, and the ECU for signal processing and control and triggering of the signal lamp and of the actuators in the hydraulic modulator.

Basic closed-loop control process

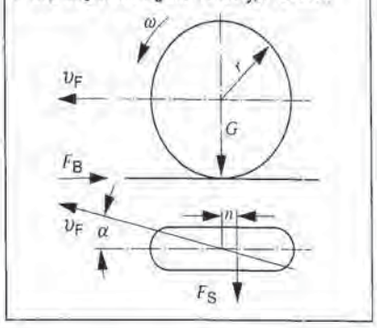
On initial braking, the brake pressure is increased; the brake slip λ rises and at the maximum point on the adhesion/slip curve, it reaches the limit between the stable and unstable ranges. From this point on, any further increase in brake pressure or braking torque does not cause any further increase in the braking force F_B. In the stable range, the brake slip is largely deformation slip, it increasingly tends towards skidding in the unstable range.

Brake slip $\lambda = (v_F - v_R)/v_F \cdot 100\,\%$
Wheel speed $v_R = r \cdot \omega$
Braking force $F_B = \mu_{HF} \cdot G$
Lateral force $F_S = \mu_S \cdot G$
μ_{HF} Coefficient of friction,
μ_S Lateral-force coefficient.

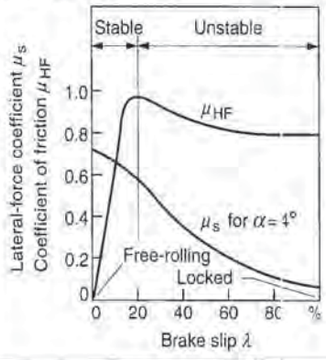
There is a more or less sharp drop in the coefficient of friction μ_{HF}, depending upon the shape of the slip curve. The resulting excess torque causes the wheel to lock-up very quickly (when braking without ABS); this is expressed as a sharp increase in wheel deceleration.

The wheel-speed sensors monitor the motion of the wheels. If one of the wheels shows signs of lock-up, there is a sharp rise in peripheral wheel deceleration and in wheel slip. If these exceed defined critical values, the controller sends commands to the solenoid-valve unit to stop or reduce the buildup of wheel-brake pressure until the danger of lock-up has passed. The brake pressure must then be built up again in order to ensure that the wheel is not underbraked. During automatic brake control, it is constantly necessary for the stability or instability of the wheel motion to be detected, and the

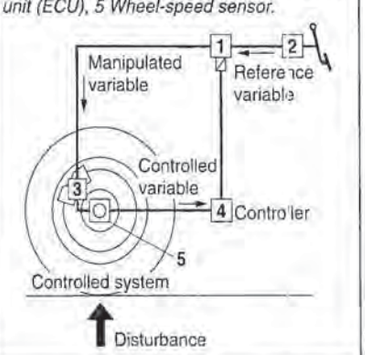
Forces at the braked wheel
G Force due to weight, F_B Braking force, F_S Lateral force, v_F Vehicle speed, α Slip angle, ω Angular velocity, n Caster.



Adhesion/slip curve
The curve shape differs greatly as a function of road surface and tire condition.



ABS control loop
1 Solenoid-valve unit, 2 Master cylinder, 3 Wheel-brake cylinder, 4 Electronic control unit (ECU), 5 Wheel-speed sensor.



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wheel must be kept in the slip range with maximum braking force by a succession of pressure-buildup, pressure-reduction and pressure-holding phases.

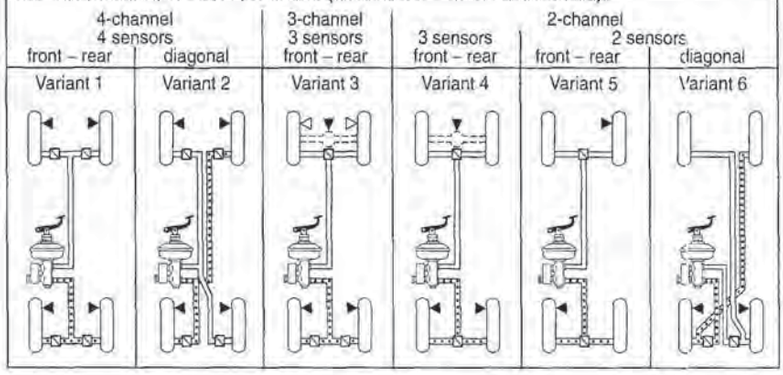
Disturbances in the closed control loop
The ABS system must take the following disturbances into account:
- Changes in the adhesion between the tires and the road surface caused by different types of road surface and changes in the wheel loadings, e.g. when cornering,
- Irregularities in the road surface causing the wheels and axles to vibrate,
- Out-of-roundness, brake hysteresis, brake fading,
- Variations in the pressure input to the master cylinder caused by the driver's brake-pedal actuation,
- Differences in wheel circumferences, for instance when the spare wheel is fitted.

Criteria of control quality
The following criteria for control quality must be fulfilled by efficient antilock braking systems:
- Maintenance of driving stability through provision of sufficient lateral guiding forces at the rear wheels.
- Maintenance of steerability through provision of adequate lateral guiding forces at the front wheels.
- Reduction in stopping distance as opposed to braking with locked-up wheels through optimum utilization of the adhesion between tires and road.
- Rapid matching of the braking force to different adhesion coefficients, for instance when driving through deep water or driving over patches of ice or hard snow.
- Guaranteeing low braking-torque control amplitudes to prevent vibrations in the running gear.
- High level of comfort due to silent actuators and low feedback through the brake pedal.

ABS system variants
A variety of versions are available depending upon the braking-force-distribution concept, the type of vehicle drive concerned, functional stipulations, and costs factor. The overview below shows six system variants, which are described in the following according to the number of channels and sensors.

4-channel systems (variants 1, 2)
These systems permit the individual control of the wheel-brake pressure of each wheel for the rear axle/front axle (II) and for the diagonal (X) braking-force-distribution concepts. When braking on split-coefficient road surfaces though, measures must be taken to ensure that the yaw moment (torque around the vertical axis) cannot adversely affect driving stability. The solution here is to control the wheels on the front axle individually and those on the rear axle in accordance with the "select-low" principle (that is, the rear wheel

ABS system variants
⊠ Control channel, ◀ Sensor, ◁ Sensor (alternative to sensor on differential).



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with the lowest coefficient of friction determines the braking pressure applied to both rear wheels).

3-channel system (variant 3)
The yaw moment when braking on split-coefficient road surfaces is reduced to such an extent (due to the system's operating principle) that passenger cars with a long wheelbase, and a high moment of mass inertia about the vertical axis, are well able to control this braking situation.

In the case of the 3- and 4-channel systems, passenger cars with a short wheelbase and a low moment of mass inertia, however, require electronic delay of the yaw-moment buildup. When braking on split-coefficient road surfaces, this causes a delayed buildup of the braking torque at the front wheel with the high coefficient of friction. As a result, the driver has enough time to correct the yaw by an appropriate steering adjustment.

2-channel systems (variants 4, 5, 6)
The 2-channel systems on the one hand need less components than the 3- and 4-channel versions, and this makes them less costly to manufacture. On the other hand though, this is accompanied by a number of functional limitations.

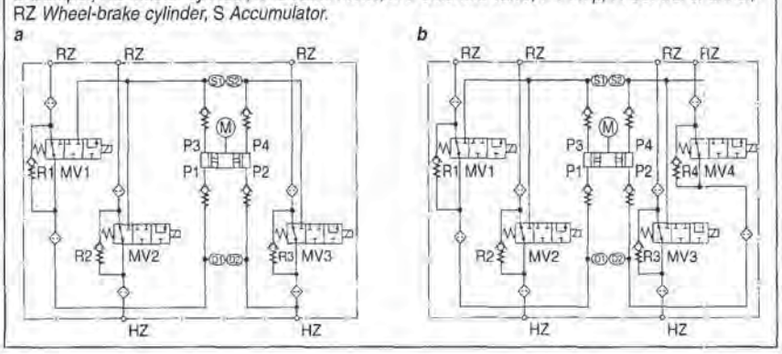
With variant 4 in the select-high mode (the front wheel with the higher coefficient of friction determines the brake pressure applied jointly to both front wheels), practically every time the driver hits the brakes hard (panic braking), one of the front wheels locks up. This is accompanied by a high level of tire wear and reduced steerability. With version 5, this always occurs when the "sensed" front wheel suddenly encounters a higher level of friction coefficient than the "non-sensed" wheel. Variant 6 can be used only with diagonal braking-force distribution (X). In this version, the brake pressures at the front wheels are controlled individually, while the brake pressures at the rear wheels are jointly controlled. Because the front-to-rear distribution of braking force bears the responsibility for ensuring that the rear wheels do not lock, this system provides somewhat lower deceleration rates than a 3- or 4-channel system.

Note: For some light-truck models with front-rear brake-circuit split in the American market there is a simple system consisting of a sensor on the rear differential and a control channel (without return pump) which prevents the rear wheels locking. The front wheels can lock up with sufficiently high braking pressures with the consequent loss of steering function. This system is not an ABS with the range of functions described at the beginning of this section and is used only for certain specialized types of vehicle.

ABS versions

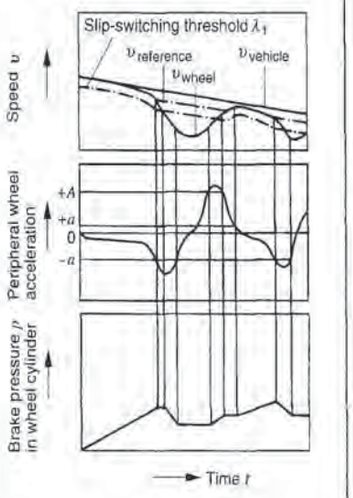
ABS 2S-3-channel/-4-channel systems (Bosch)
In this system, the ABS and the brake booster are separate units.

Functional diagrams
a) Three-channel hydraulic modulator, b) Four-channel hydraulic modulator.
D *Damper*, HZ *Master cylinder*, M *Electric motor*, MV *Solenoid valve*, P *Pump*, *R Control channel*, RZ *Wheel-brake cylinder*, S *Accumulator*.



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ABS control cycle for high friction coefficients



The 3-channel hydraulic modulator for the II braking-force-distribution version (front/rear) comprises three solenoid valves with three possible positions, and a return pump with electric drive motor.

In the first, de-energized position, there is an unhindered passage from the master cylinder to the wheel-brake cylinder, with the result that the wheel-brake pressure rises during initial braking and during automatic brake control. In the second, semi-energized position, the passage from the master cylinder to the wheel-brake cylinder is interrupted, with the result that the wheel-brake pressure is kept constant. In the third, fully energized position, the wheel-brake cylinder is connected to the return and the wheel-brake pressure drops.

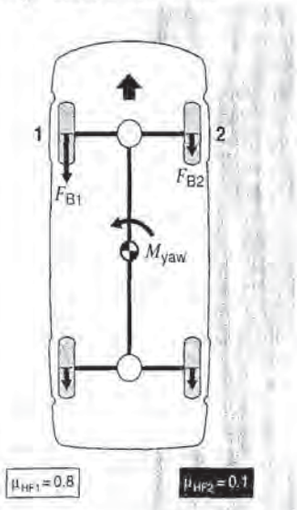
Pressure dissipation takes only about 20 ms so that the wheel does not lock up; pressure build-up, on the other hand, takes place in several stages with intervening holding phases and typically takes around 200 ms.

The 4-channel hydraulic-modulator valve for diagonal braking-force distribution calls for four solenoid valves, because the rear-wheel brakes belong to different brake circuits. However, both the rear-wheel valves are jointly energized, so that there is the same pressure in each of the rear-wheel-brakes, and the "select-low" control mode is easy to implement.

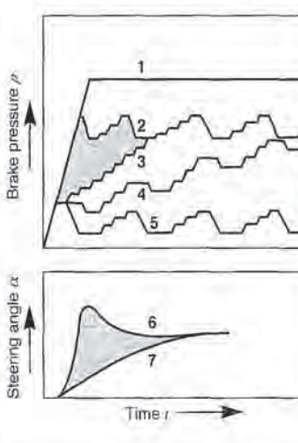
The depicted control cycle shows automatic brake control in the case of a high friction coefficient. The change in wheel speed (deceleration) is calculated in the ECU. After the value falls below the $(-a)$ threshold, the hydraulic modulator valve unit is switched to the pressure-holding mode. If the wheel speed then also drops below the slip-switching threshold λ_1, the valve unit is switched to pressure reduction; this is done as long as the $(-a)$ signal is present. During the subsequent pressure-holding phase, the peripheral wheel acceleration increases until the $(+a)$ threshold is exceeded; thereupon, the brake pressure continues to be kept constant. After the relatively high $(+A)$ threshold has been exceeded, the brake pressure is increased, so that the wheel is not accelerating excessively as it enters the stable range of the adhesion/slip curve. After the $(+a)$ signal has dropped out, the brake pressure is slowly raised until, when the wheel acceleration again falls below the $(-a)$ threshold, the second control cycle is initiated, this time with an immediate pressure reduction. In the first control cycle, a short pressure-holding phase was necessary initially for the filtering of disturbances. In the case of high wheel moments of inertia, low friction coefficient and slow pressure rise in the wheel-brake cylinder (cautious initial braking, e.g., on black ice), the wheel might lock-up without the deceleration switching threshold having responded. In this case, therefore, the wheel slip, too, is used in automatic brake control.

Under certain road-surface conditions, passenger cars with all-wheel drive and with differential locks engaged pose problems when the ABS system is in operation; this calls for special measures to support the reference speed, lower the wheel-deceleration thresholds, and reduce the engine drag torque.

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Buildup of yaw moment induced by large differences in friction coefficients
M_{yaw} Yaw moment, F_B Braking-force, μ_{HF} Coefficient of friction. 1 "High" wheel, 2 "Low" wheel.



***Curves for brake pressure/steering angle with delayed yaw-moment buildup** (GMA)*
1 Master-cylinder pressure p_{MC}, 2 Braking pressure p_{high} w/o GMA, 3 p_{high} with GMA 1, 4 p_{high} with GMA 2, 5 p_{low} at "Low" wheel, 6 Steering angle α w/o GMA, 7 Steering angle α with GMA.

Brake control with yaw-moment buildup delay

When the brakes are applied on an asymmetrical road surface (for instance, left wheels on dry asphalt, right wheels on ice), the result is vastly different braking forces at the front wheels. This difference induces a turning motion (yaw moment) around the vehicle's vertical axis.

When a heavy passenger car with ABS is braked, the resulting yaw sets in slowly and the driver has time for corrective steering maneuvers. On smaller cars, the ABS must be supplemented by an additional yaw-moment buildup-delay device (GMA) to ensure that control is maintained during panic stops on asymmetrical surfaces. GMA delays the pressure buildup in the wheel cylinder of the front wheel with the higher coefficient of braking force at the road surface ("high" wheel).

The GMA concept is demonstrated in the diagram: Curve 1 represents the master-cylinder pressure p_{MC}. Without GMA, the wheel on asphalt soon arrives at the pressure p_{high} (Curve 2), while the wheel which is running on ice goes to p_{low} (Curve 5); each wheel achieves the maximum retardation available under the given circumstances (individual control).

The GMA 1 System (Curve 3) is suited for use with vehicles with a less critical response pattern, while GMA 2 is designed for cars which display an especially marked tendency toward yaw-induced instability (Curve 4).

In all cases in which GMA comes into effect, the "High" wheel is under-braked at first. This means that the GMA must always be very carefully adapted to the vehicle in question in order to limit increases in stopping distances.

ABS 2E (Bosch)

This economical "full ABS" design (a 3-position solenoid valve with electronic control stage is replaced by a plunger/floating piston arrangement) offers the same safety features and functions as the ABS2S system with only minimal loss of comfort in terms of brake pedal feedback and noise.

Under normal braking, without active ABS, brake fluid flows to the right rear

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wheel through the rear-axle solenoid valve (4), while the plunger's central valve (6) supplies the left rear wheel.

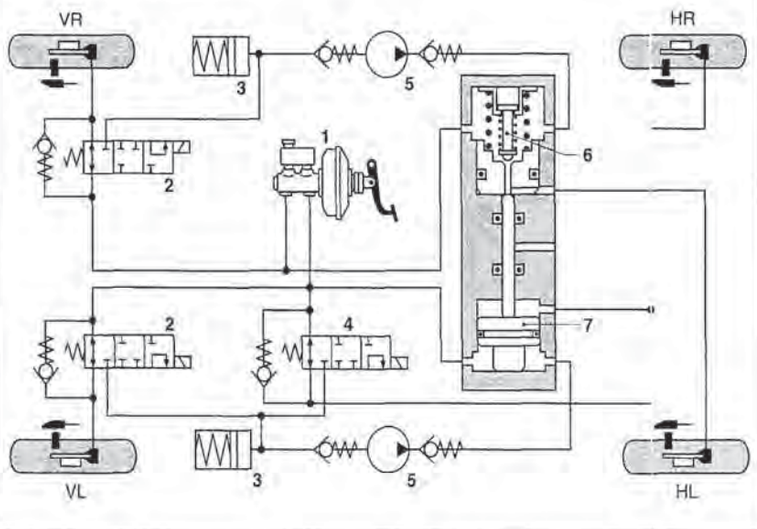
When the ABS is activated, each of the left-side solenoid valves (2) controls a front brake, while the rear-axle solenoid (4) directly controls the rear right wheel (HR). The rear-axle solenoid valve (4) switches to position 3 upon receiving the "pressure-reduce" command from the ECU. In the process, fluid is released from the rear right wheel (HR) into the accumulator chamber (3) via the valve (4) so that the pressure at the wheel brake (HR) is reduced. Since the upper side of the floating piston (7) is hydraulically connected to the wheel brake (HR), this pressure reduction is also applied above the floating piston (7). The underside of the piston (7) is subjected to pressure from the master cylinder (1), and the resulting force on the piston (7) causes the floating piston assembly to move upwards. The upper plunger now comes into contact with the central valve pin (6) so that the central valve closes and volume can be accepted from the rear brake (HL) when upward movement continues. At the same time, the pressure in the brake (LR) and on the underside of the top plunger also drops as required. The movement of the plunger then comes to a standstill when the resulting force on the plunger assembly has dropped to zero. This is the case when the pressure in the two brakes (HL) and (HR) is equal. The processes for "pressure hold" and "pressure buildup" are analogous to the pressure-reduction process and are controlled by the position of the solenoid valve (4).

ABS5 Family (Bosch)

As a result of the development of solenoid valves with two hydraulic positions it has been possible to thoroughly revolutionize the ABS system while retaining the range of functions of the ABS2S. The ABS5 is based upon the tried and proven return principle as used in the ABS2S. It contains the following components for each brake circuit (with X and II braking-force distribution):

- Return pump,
- Accumulator,

ABS2E hydraulic system *(X braking-force distribution)*
1 Brake booster, 2 Solenoid valves, 3 Accumulator chambers, 4 Rear-axle solenoid valve, 5 2-circuit return pump, 6 Central valve, 7 Plunger piston.
V front, H rear, R right, L left.



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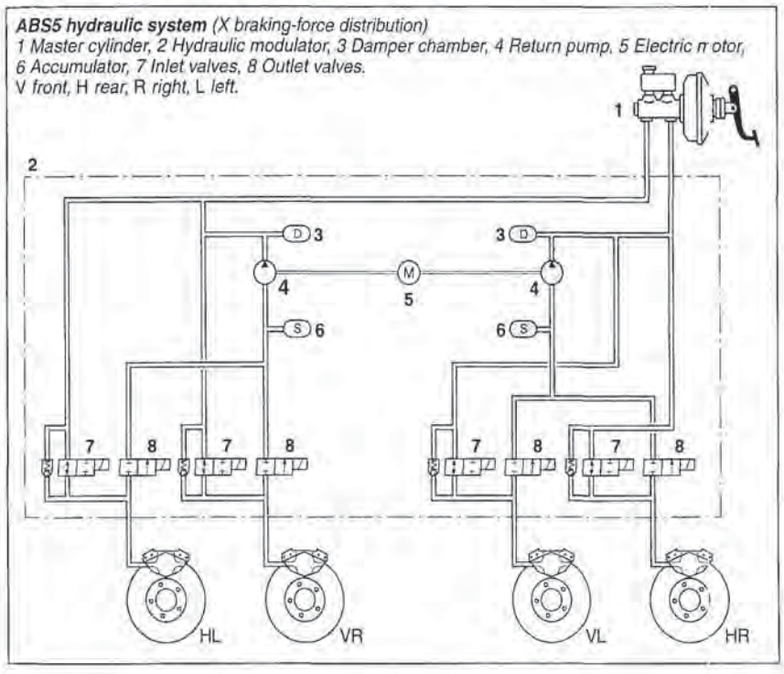
- Damper chamber, and
- 2/2 solenoid valves with two hydraulic positions and two hydraulic connections.

Each wheel (or the rear axle in the case of II braking-force distribution) is allocated a solenoid-valve pair, a solenoid valve (EV – open when de-energized) for the pressure increase (inlet) and a solenoid valve (AV – closed when de-energized) for pressure reduction (outlet). In order to achieve rapid release of pressure at the brakes when the pedal is released, the inlet valves each have a non-return valve which is integrated in the valve body (e.g. non-return valve sleeves or unsprung non-return valves).

Pressure increase and decrease are each allocated a solenoid valve which features only 1 active (energized) position. This leads to compact valve designs with only 2 hydraulic connections, smaller volume, lower weight, and low magnetic forces. The design means that only a single switching transistor is needed for the electrical triggering, with the attendant reduced electrical power loss in the solenoid coils and in the ECU. The valve block for mounting the 2/2 solenoid valves results in a smaller and lighter hydraulic modulator and means that the ECU can be attached directly to the hydraulic modulator. This has the advantage that the vehicle's wiring harness needs less lines. Because of their compact design, the 2/2 solenoid valves enable shorter electrical switching times and even pulse-width-modulated cyclic operation which substantially improves function (e.g. adaptation to changes in coefficient of friction) and control convenience (e.g. smaller delay fluctuations with the aid of pressure stages and lower levels of valve noise).

By coordinated development of mechanical valve design and pulse-width-modulated control (mechatronic optimization), it has been possible to produce an ABS system with "quiet" solenoic valves which represents a new milestone in terms of noise generation and pedal feedback.

Electric-motor variants, together with hydraulic throttling (restriction) of the flow



ABS5 hydraulic system *(X braking-force distribution)*
1 Master cylinder, 2 Hydraulic modulator, 3 Damper chamber, 4 Return pump, 5 Electric motor, 6 Accumulator, 7 Inlet valves, 8 Outlet valves.
V *front,* H *rear,* R *right,* L *left.*

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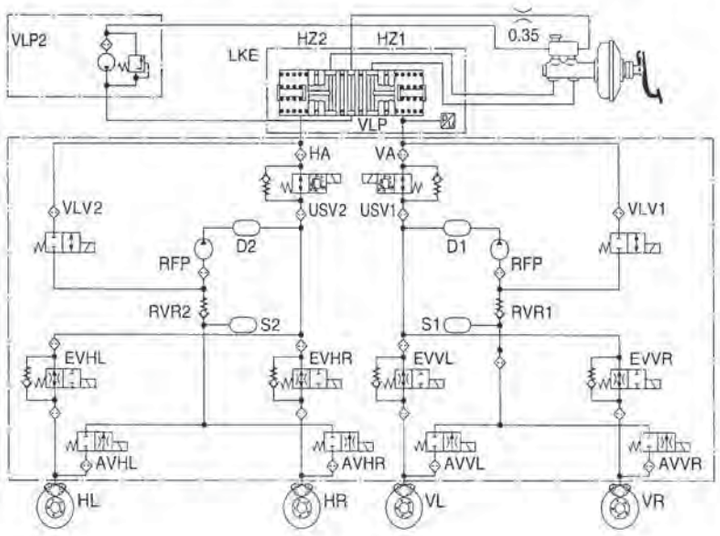
cross sections in the solenoid valves, and variations in the accumulator volumes, mean that the ABS5-family represents a new generation of ABS systems suitable for installation in all series-production passenger cars.

ABS with TCS (Bosch)
See chapter "TCS traction control system," P. 606

ABS and ESP hydraulics (Bosch)
Proper control of vehicle-handling dynamics by means of ESP Electronic Stability Program (previously known as VDC Vehicle Dynamics Control) requires sufficiently rapid build-up of braking force in the brakes. On the one hand, low temperatures and low coefficients of friction are the conditions under which the ESP is called upon most often. On the other though, at low temperatures the brake-fluid viscosity increases considerably, thus making it necessary to modify the hydraulic concept compared to that used with ABS/TCS. The ABS/TCS5 system as described in the Chapter "ABS with TCS (Bosch)" was taken as the basic system. Here, the pre-charge pump (VLP) was incorporated to ensure the return-pump delivery capacity when the ESP is needed under cold-weather conditions. The requirement for closed and separate brake circuits meant that this pre-charge pump could not be connected hydraulically directly upstream of the return pump. Instead it is applied to a charge-plunger unit (LKE) connected between the master cylinder and the ABS/TCS hydraulics. As soon as brake pressure is needed, the pre-charge pump switches on (together with the pre-charge valves (V_V) and the changeover valves (USV)) and delivers brake fluid to the charge-plunger unit (LKE) so that the two plungers in the unit move apart.

This mechanically closes the central valves of the LKE and the brake-fluid volume from the two LKE cylinders is forced through the opened pre-charge valves to the return pumps of the ABS/TCS hydraulics. The pre-charging of the return

Hydraulic system for ABS and ESP
VLP *Pre-charge pump,* VLV *Pre-charge valve,* RFP *Return pump,* RVR *Non-return valve,* LKE *Charge-plunger unit,* USV *Change-over valve,* D *Damper,* AV *Outlet valve,* EV *Inlet valve,* HZ *Master cylinder,* S *Accumulator,* VA *Front axle,* HA *Rear axle,* V *Front,* H *Rear,* R *Right,* L *Left.*



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pumps means that sufficiently rapid brake-pressure buildup is ensured at the wheel brakes even at low temperatures. A separate line with a small restriction is provided between the LKE and the reservoir in order that the brake fluid in the pre-charge circuit can be bled, and the LKE plungers return to their initial position when the pre-charge pump is switched off.

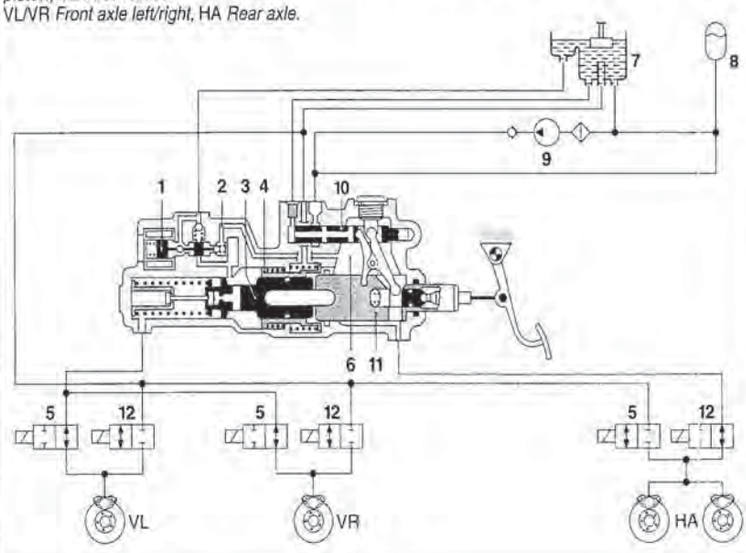
By further optimization of the master cylinder and the ABS/TCS5 induction valves with the aim of increasing the hydraulic flow cross-section, it has been possible to produce an ESP hydraulic modulator that enables adequate pressure generation dynamics even without a precharger pump at low temperatures and thus further reduces the number of system components.

MK 2 (Teves)
The hydraulic components – brake booster and ABS – form a compact integrated unit for installation on the firewall. Under normal braking, the booster piston impels brake fluid directly to the rear brakes while pushing the master cylinder piston to the left, supplying brake fluid to the front brakes.

When the ABS is activated, the main valve opens, connecting the booster chamber with the primary side of the master cylinder piston while closing off the connection between the primary side and the reservoir. Brake fluid flows from the booster chamber to the front brakes via the connecting line and seal of the master-cylinder piston. During ABS-controlled braking, booster pressure is exerted against the left side of the positioning sleeve, maintaining master cylinder and booster piston in a middle position. This ensures that sufficient piston travel remains available for front-wheel braking in the event of ABS failure.

The supply and discharge valves provide optimal regulation of the pressures in the wheel cylinders during ABS-controlled braking maneuvers, with the brake fluid which is ejected from the wheel cylinder flowing back to the reservoir.

Schematic diagram: ABS MK 2 *(Teves)*
Brakes not actuated.
1 Main valve, 2 Connecting line, 3 Master-cylinder piston, 4 Positioning sleeve, 5 Outlet valves, 6 Servo chamber, 7 Fluid reservoir, 8 Pressure accumulator, 9 Pump, 10 Brake valve, 11 Servo piston, 12 Inlet valves.
VL/VR *Front axle left/right,* HA *Rear axle.*



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The front brakes are controlled individually. The rear brakes are regulated together, whereby the wheel with the lower coefficient of traction determines the level of pressure (select low).

MK 4 with TCS (Teves)
This version is used with a conventional vacuum booster to provide "separate ABS". The system can also be extended to include TCS.

When the start of active ABS control makes it necessary to reduce the brake pressure at one of the wheels, the discharge valve is opened while the supply valve remains closed, allowing brake fluid to flow back to the reservoir from the brake. When the supply valve opens to increase the pressure, brake fluid flows from the master-cylinder chamber, and the pedal gives way somewhat. The hydraulic unit must supply new energy, as repeated cycles would otherwise cause the pedal to drop too far.

The ABS hydraulic energy-supply unit employs an electrically-driven, dual piston pump which is triggered when the system recognizes incipient wheel lock. The pump extracts brake fluid from the reservoir and pumps it through the supply-valve orifice and to the brake at a suitably increased pressure.

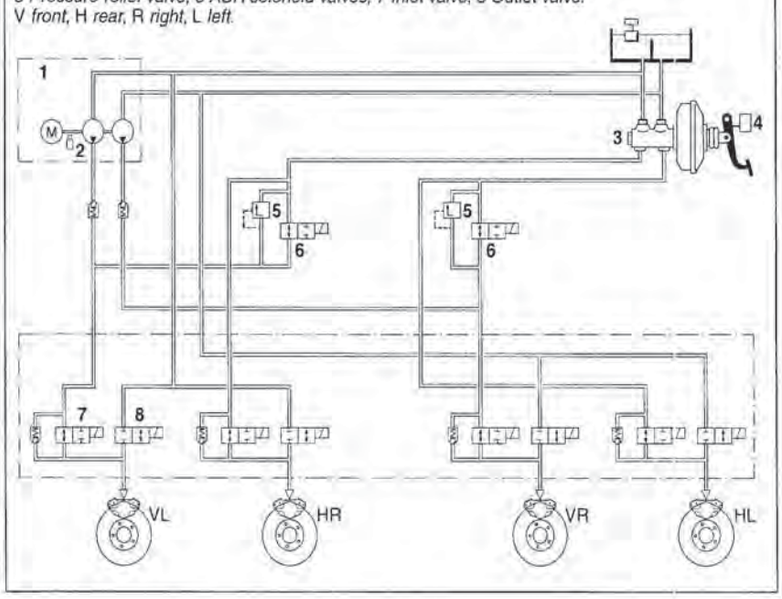
The excess flow quantity from the pump flows into the chambers of the master cylinder, where it presses back the master-cylinder piston and the brake pedal. The position sensor triggers and switches off the pump so as to ensure that the intermittent flow which it produces results in adequate feel and travel at the pedal.

The hydraulics can be expanded for traction control by adding 2 TCS solenoid valves, which serve to isolate the master cylinder from the TCS, and 2 pressure-relief valves, responsible for regulating pressure in the TCS system.

The rotation sensor is a security device for monitoring the operation of the pump motor.

MK 20 Family (Teves)
In contrast to the MK4, the MK 20 system operates according to the return principle using sealed brake circuits. Pressure

Schematic diagram: ABS/TCS MK 4 *(Teves)*
1 Energy supply, 2 Wheel-speed sensor, 3 Master-cylinder piston, 4 Position sensor, 5 Pressure-relief valve, 6 ASR solenoid valves, 7 Inlet valve, 8 Outlet valve.
V *front*, H *rear*, R *right*, L *left*.



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modulation is by means of a pair of 2/2 solenoid valves (inlet/outlet) for each wheel and the system can be upgraded to TCS/ESP by the addition of two changeover valves and two induction valves. The hydraulic components thus correspond exactly to the arrangement employed by the Bosch ABS5 Family.

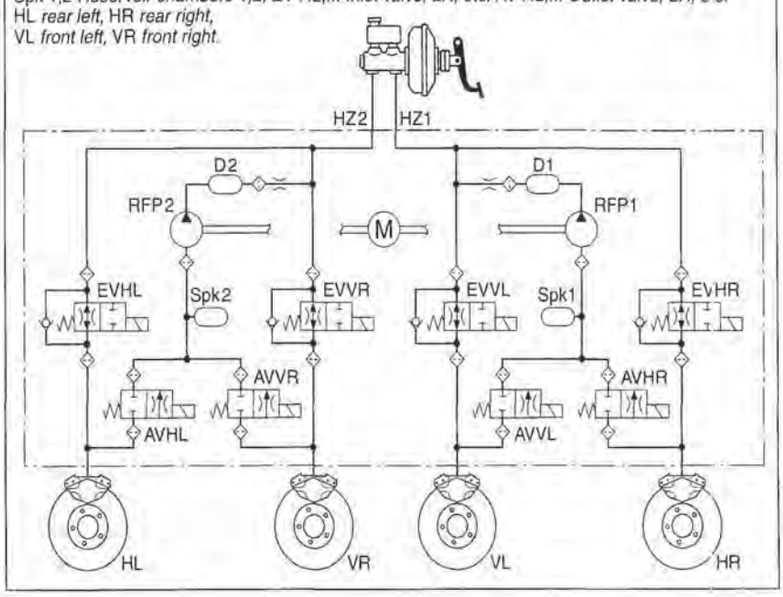
Pumpless ABS (Denso)

For a certain section of the Japanese market, a greatly simplified ABS system was developed which dispenses with damper chambers and return pumps together with their electric motors and control systems. It operates according to the following principle:

If the wheel-speed sensors detect that wheels are about to lock under braking, the appropriate inlet valves are closed in order to prevent further increase of pressure in the wheel cylinders. If this does not succeed in preventing the wheels locking, the corresponding pressure release valve is opened until the wheel stops slipping and is then closed again. The brake fluid that flows out of the wheel cylinders in the process returns to the reservoir chambers. Once the wheel speed becomes stable again, the inlet valves concerned are opened again. The process described repeats itself continually until the driver stops braking or the vehicle comes to a standstill. When the brake pedal is released by the driver, the brake fluid that has collected in the reservoir chambers flows back through the reservoir-chamber springs and the non-return valves to the master cylinder. The system is then ready for ABS braking again.

The smaller number of components compared with conventional systems offers considerable cost savings but the system has significant functional deficiencies. Among other things, under extended periods of ABS braking (e.g. from high speeds on road surfaces with poor adhesion or in situations where adhesion fluctuates dramatically), the reservoir chambers can reach their capacity while at the same time the brake-pedal travel increases considerably in an unfamiliar way. Apart from the loss in pedal-opera-

Schematic diagram: ABS MK 20 *(Teves)*
HZ 1,2 *Master cylinder 1,2,* D 1,2 *Damper chamber 1,2,* RFP 1,2 *Return pump 1,2,*
Spk 1,2 *Reservoir chambers 1,2,* EV HL,... *Inlet valve, LR, etc.* AV HL,... *Outlet valve, LR, etc.*
HL *rear left,* HR *rear right,*
VL *front left,* VR *front right.*



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tion comfort, this means that the pressure in the wheel cylinders can not be reduced any further with the result that the wheels could lock up. For that reason, this type of system is only used on certain vehicles with limited top speeds and has not as yet become established in the European or American automobile markets.

ABS components (Bosch)

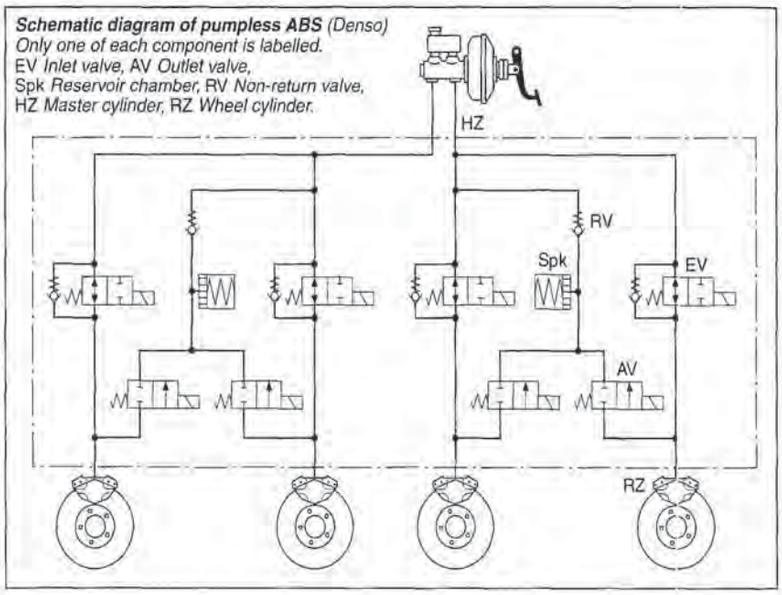
Wheel-speed sensor
The inductive wheel-speed sensor provides the control unit (ECU) with information on wheel speed.

ECU unit with vehicle-specific LSI circuits
The ECU shown in schematic form in a 4-channel system receives, filters and amplifies the speed-sensor signals and ascertains from them the degree of wheel slip and the acceleration of the individual wheels as well as the reference speed, which is the best possible calculation of vehicle road speed.

Input circuit:
The input circuit consists of a low-pass filter and input amplifier; the circuit suppresses interference and amplifies the signals from all wheel-speed sensors (Channels 1...4).

Digital controller:
The digital controller consists in the case of ABS2S of two identical but independent digital vehicle-specific LSI circuits which each process the information from two wheels (channels 1+2 or 3+4) simultaneously and perform the logical processes. Once processed, the wheel frequency data continues in the circuit to a serial arithmetic-logic unit. This logic unit, in turn, uses the data to calculate the values for "wheel slip" and "circumferential deceleration" or "circumferential acceleration" required for closed-loop control. A self-adapting, complex controller logic converts the control signals into position commands for the solenoid valves. A serial interface, connected to the input stage, arithmetic-logic unit and controller logic via data link, maintains data communications between the two digital LSI circuits.

Yet another function block contains the monitoring circuit for error recognition and analysis. Should the ECU malfunction, a warning lamp informs the vehicle operator that the ABS is no longer opera-



Schematic diagram of pumpless ABS *(Denso)*
Only one of each component is labelled.
EV *Inlet valve,* AV *Outlet valve,*
Spk *Reservoir chamber,* RV *Non-return valve,*
HZ *Master cylinder,* RZ *Wheel cylinder.*

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tional. However, the braking system retains full normal operating capability when the ABS is deactivated.

Output circuits:
Two output circuits function as current regulators for Channels 1 + 2 and 3 + 4 while receiving the position commands employed for solenoid excitation from the LSI circuits.

Output stage:
The output stage employs the input from the current regulators in the two output circuits in providing the excitation current for the solenoid valves.

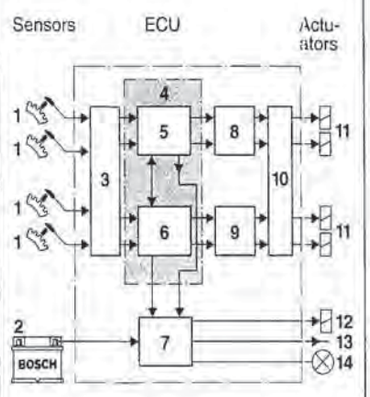
Voltage stabilizer, fault memory:
This function block stabilizes the supply voltage and monitors it to ensure that it remains within the tolerances required for reliable operation. The block also incorporates undervoltage recognition, which reacts to insufficient on-board voltage by shutting down the unit, as well as relays and the warning-lamp control circuit.

ECU unit with microcontrollers
Instead of the vehicle-specific LSI circuits, this control unit has two microcontrollers which take care of signal processing, "running" the controller program and self-monitoring on the part of the ABS system. The unit also carries out diagnosis in accordance with ISO standards, making it possible to track down defective ABS components with the aid of either the warning lamp or an "intelligent" tester.

The use of microcontrollers could achieve substantial optimization of controller algorithms incorporating customization to vehicle and driver requirements. For example, more precise calculation of wheel slip enables early detection of excessive rear-wheel braking which can then be corrected in time by changes in brake pressure with the result that, on some vehicles, the rear-wheel pressure reducer can be dispensed with. This achieves better overall vehicle braking (electronic brake balancing).

Further hybridization of the electronic components has succeeded in considerably reducing their number and thereby the size of the control unit as well.

***ECU** (Bosch, 4-channel layout).*
1 Wheel-speed sensors, 2 Battery, 3 Input circuit, 4 Digital controller, 5 LSI circuit 1, 6 LSI circuit 2, 7 Voltage stabilizer/fault memory, 8 Output circuit 1, 9 Output circuit 2, 10 Output stage, 11 Solenoid valves, 12 Safety relay, 13 Stabilized battery voltage, 14 Warning lamp.



Hydraulic modulator (for ABS2S and ABS5S with II and X braking-force distribution)
For each brake circuit, the hydraulic modulator consists of:
– Electric-motor-driven return pump P,
– Accumulator chamber S,
– Damper chamber D, and
– Various solenoid valves.

Return pump P:
The return pump returns the brake fluid from the wheel-brake cylinders to the master cylinder.

Accumulator S:
The accumulators provide temporary storage of the large quantity of brake fluid which accompanies pressure reduction.

Damper chamber D:
The dampers and their downstream throttling devices serve to smooth the high levels of pulsation which occur during the return of brake-fluid to the master cylinder.

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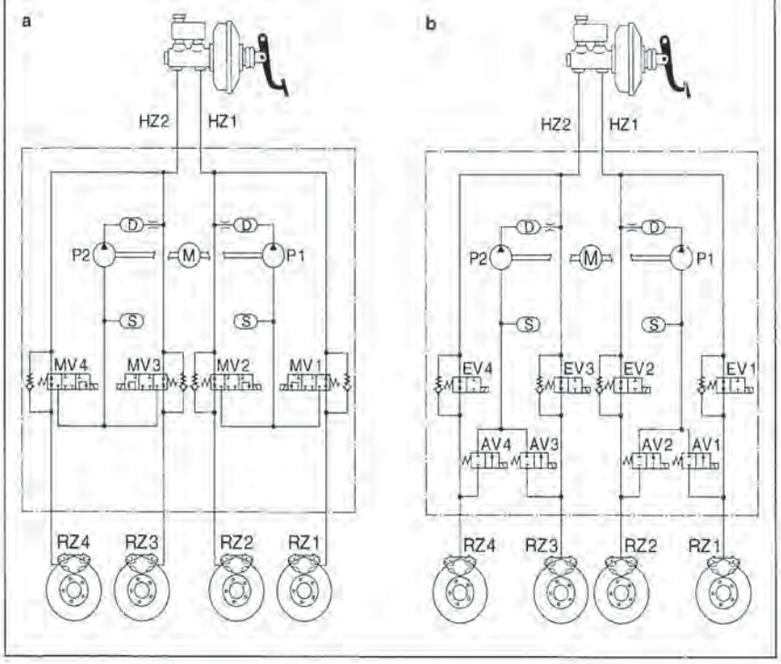
They ensure that the noise level is kept to a minimum.

3/3 solenoid valves on ABS2S:
Each wheel is allocated a 3/3 solenoid valve. The valve serves to modulate the pressure in the wheel cylinders during active ABS control. The modulation takes place in 3 modes (build-up, hold, and reduce).

2/2 solenoid valves on ABS5:
Each wheel is allocated an EV/AV valve pair. The same pressure-modulation modes as above can be achieved for each wheel brake by appropriate control of the valves.

In addition to the effect of the damper chambers, PWM control of the 2/2 solenoid valves allows the required pressure changes in the wheel cylinders to be achieved with a greater degree of driver comfort in terms of noise, initiation of suspension vibration and pedal feedback.

Schematic diagram of hydraulic modulator
a) ABS2S system, b) ABS5 system.
HZ *Master cylinder,* M *Electric motor,* P *Plunger pump,* D *Damper,* S *Volume accumulator,* MV *3/3 solenoid valve,* EV *2/2 inlet valves,* AV *2/2 outlet valve,* RZ *Wheel cylinder.*



Electrohydraulic Brakes (EHB)

In contrast to antilock braking systems (ABS), the traction control system (TCR) and electronic stability program (ESP) can generate pressure in the wheel cylinders independently of driver action. This capability forms an important basis for electrohydraulic brakes.

Function

In conventional car braking systems, the force applied by the driver is transmitted mechanically by the lever action of the brake pedal to the vacuum brake servo and from there in amplified form to the master cylinder. The pressure thus generated is used to achieve the desired braking effect with the individual brakes on each wheel. With electrohydraulic brakes, that purely mechanical-hydraulic sequence of actions is broken and replaced by sensors, an ECU and a hydraulic pressure supply. Under normal operating conditions, there is no mechanical link between the brake pedal and the wheel brake.

Design

An electrohydraulic braking system consists of the following components (see diagram):

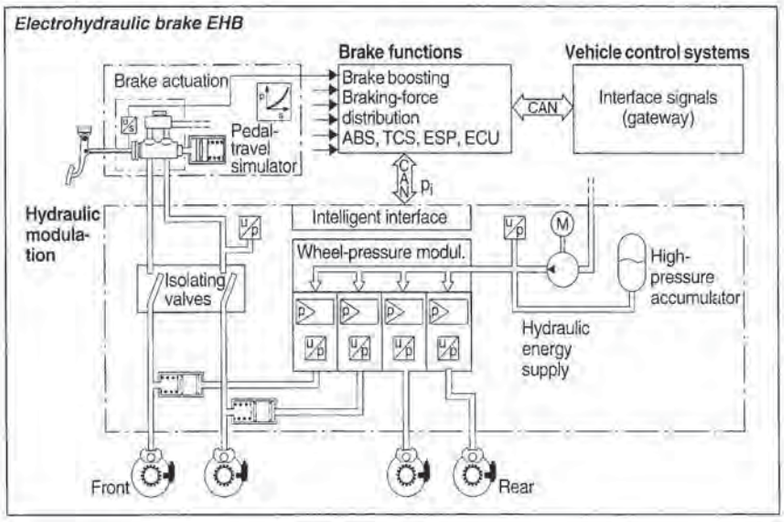
- Actuator unit,
- Hydraulic pressure modulator
- Sensors (e. g. wheel-speed sensor),
- Add-on ECU (at the hydraulic pressure modulator),
- or separate ECU,
- Control and pressure lines.

Mode of operation

For safety reasons, two separate sensors (one on the actuator unit for detecting the pedal travel and a pressure sensor on the hydraulic modulator) are used to detect a "braking request" and transmit it to the ECU. Using software, this ECU also incorporates the brake servo, ABS, TCS and ESP functions. The other sensors in the ABS, TCS and ESP systems provide the ECU with data relating to aspects of vehicle dynamics such as road speed, cornering, and the motion of the wheels. Using this information, the ECU calculates the signals to be sent to the hydraulic modulator which are converted by the wheel pressure modulators into the brake pressures for the individual wheels. An electrically driven hydraulic pump with a high-pressure accumulator and a pressure monitoring system provides the hydraulic pressure supply.

For safety reasons, in the event of a fault in the system it switches to an operating mode in which the vehicle can be braked without power assistance.

Electrohydraulic brake EHB



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Braking systems for commercial vehicles above 7.5 t laden weight

System and configuration

Power-braking systems of medium and heavy-duty commercial vehicles mostly operate with:

- Compressed air (normal pressure level) as a medium for energy supply and as a transmission device.
- Pneumatic-hydraulic power transmission in the service-braking system and pneumatic power transmission in the parking-brake system, or
- Pneumatic high pressure level (Figs. B & C).

Service-braking system for towing (tractor) vehicles

For practical operations, in medium and heavy commercial vehicles, the force exerted by the driver's foot alone is not sufficient to generate adequate braking deceleration. For this reason, compressed-air power-brake systems are principally used; these systems use compressed air as a stored form of energy to control or actuate the service-braking system. Diaphragm actuators create the brake application forces on the wheel brakes. "Air-over-hydraulic" brake systems are becoming less and less common, as the air pressure required to control the wheel brakes must be converted into hydraulic pressure by means of brake servo-unit cylinders, compact brake assemblies or actuating cylinders.

A modern dual-circuit compressed-air brake system with trailer-brake connection, spring-type brake actuator, secondary and parking-brake system consists of the following main components:

- Energy supply,
- Compressed-air reservoir,
- Brake valves,
- Braking-force controller,
- Wheel brakes, and
- Control and air supply for the trailer braking system (diagram D).

The energy supply comprises a compressor and pressure regulator. Where necessary, antifreeze pumps, automatic water drainage, air filters, air dryers and intermediate reservoir are added, providing clean and well-drained air.

As there may be a number of air-consuming installations apart from the braking system in the tractor-trailer combinations used in modern-day commercial-vehicle operations, it is advisable to install considerably more powerful compressors than those, for instance, required by EC/ECE brake regulations.

The four-circuit protection valve is located at the interface point between the energy supply and the compressed air reservoir. In the event of a fault, this protection valve ensures the security and continuing supply of the individual brake circuits and of priority secondary consumers.

The actuating mechanism for the service brakes starts with the brake pedal and ends at the mechanically operated components of the dual-circuit brake valve.

Sometimes, in commercial vehicles, the limited space available makes it necessary to fit the brake valve on the frame behind the cab instead of in the (tipping) cab itself. The problem of left-hand and right-hand drive trucks with the brake valve in the same position on the frame is solved by means of a dual-circuit hydraulic transmission device for the braking-circuit control. In this process an electrical warning system monitors the level of fluid in the compensating reservoirs of both circuits.

Additional installations in towing vehicles for pulling trailers/semitrailers with compressed-air brake systems (see "Basic components of a braking system") help to supply air to the trailer/semitrailer braking system, and provide a controlling function with regard to the braking effect (trailer control valve, coupling heads, etc.).

Two-line braking system for trailers

In the case of the two-line braking system, which is the standard European design version, one line (supply line) connects the energy reservoirs in the towing vehicle with those in the trailer/semitrailer; and this line is permanently under pressure.

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The second line (brake line) leads from the trailer control valve in the towing vehicle to the trailer brake valve in the trailer/semitrailer (diagram E). Braking occurs as a result of pressure increase. Automatic braking when the trailer/semitrailer accidentally comes loose from the towing vehicle is ensured in this system by the supply line. If the supply line comes loose or breaks, air flows out of it and the brake valve in the trailer/semitrailer activates the brakes. With the aid of a dual-circuit trailer control valve and the four-circuit protection valve, it is possible to continue the supply of air to the trailer/semitrailer and to control braking, even if one circuit of the dual-circuit braking system in the towing vehicle fails. The standardized coupling heads for "supply" and "brake" are fitted with an automatic shut-off element, which opens during the coupling process.

Automatic load-sensitive device for braking-force metering

The automatic device for correcting braking forces as a function of load (ALB) is a vital element of the braking-system transmission device. With the vehicle partially-loaded or empty, braking-force metering valves permit the braking forces to be adjusted to the reduced axle loading e.g., by sensing of the axle ride clearances) and thus permit a correction of braking-force metering to the vehicle's axles ("kinked load-sensitive braking-force metering"), or they permit a set level of braking (important for vehicles in road train or tractor-trailer operation). There are two kinds of load-sensitive braking-force metering valves (Fig. A):

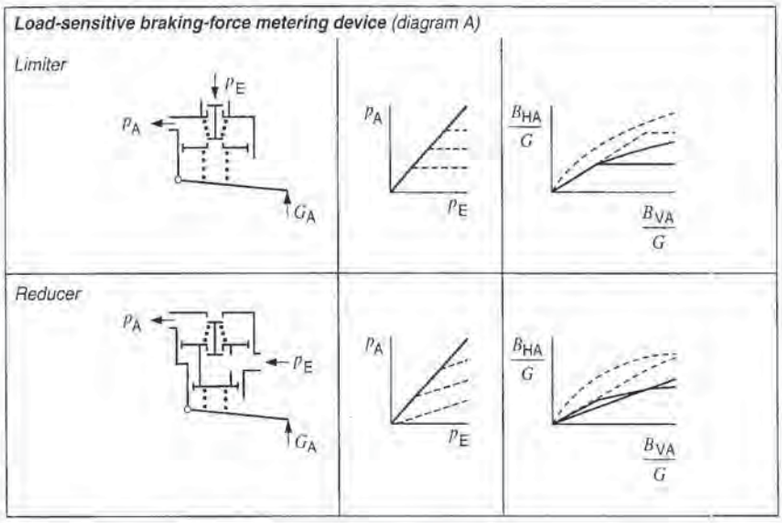
1. Braking-force limiters

Above a given "switchover point", the braking-force limiter restricts the increase of braking force (e.g., on the rear axle), that is to say there is a "kink" in the braking-force metering characteristic.

2. Braking-force reducers

Even under most unfavorable load conditions, load-sensitive braking-force reducers permit braking-force metering which approaches the parabola of the dynamic (ideal) braking-force metering characteristic (see "Design of a braking system"). In the area above the switchover point the braking forces on the particular axle are reduced with respect to the original braking-force metering level. Here, the installed braking-force metering is dependent on the transmission ratio and on the switchover pressure (which in turn is dependent on axle load) of the braking-force metering valve.

Load-sensitive braking-force metering device *(diagram A)*

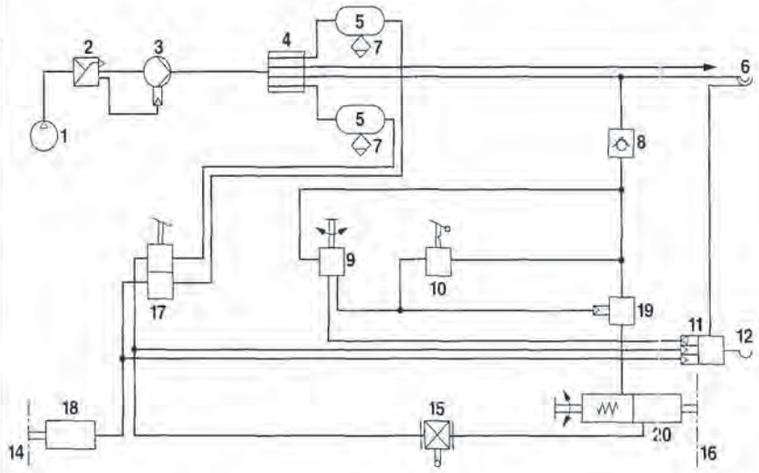


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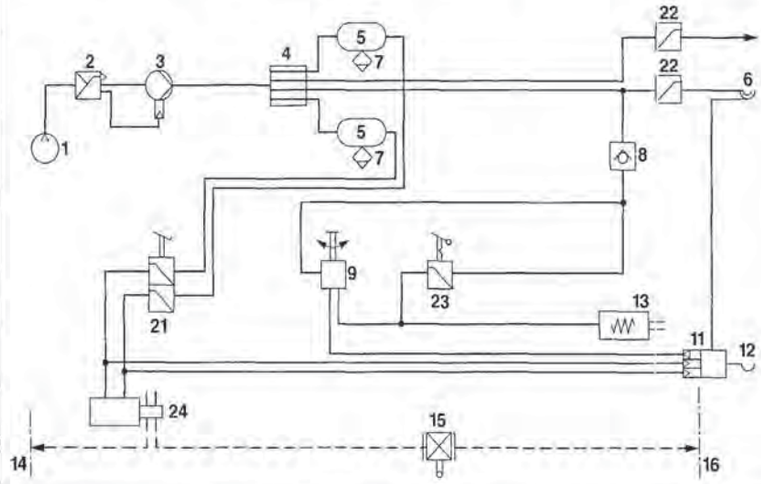
Compressed-air braking systems for trucks

1 Compressor,
2 Pressure regulator,
3 Anti-freeze pump,
4 Four-circuit safety valve,
5 Compressed-air cylinder,
6 Coupling head with automatic shutoff valve,
7 Water drain valve,
8 Non-return valve,
9 Test valve,
10 Parking-brake valve,
11 Trailer-control valve,
12 Coupling head without shutoff valve,
13 Spring-type brake cylinder,
14 Front wheels,
15 Automatic Load-sensitive Braking-force metering (ALB),
16 Rear wheels,
17 Service-brake valve,
18 Brake cylinder

Dual-circuit two-line power-brake system with pneumatic transmission device *(Fig. B)*

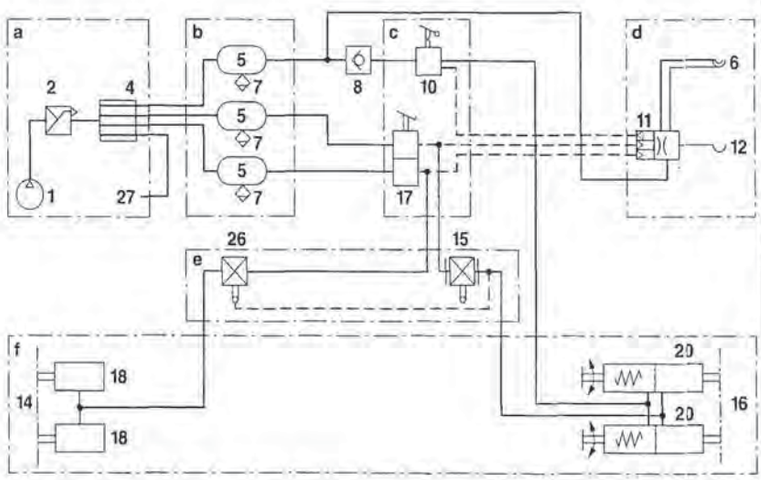


Dual-circuit two-line power-brake system ("Air-over-hydraulic" braking system) with hydraulic transmission device *(Fig. C)*

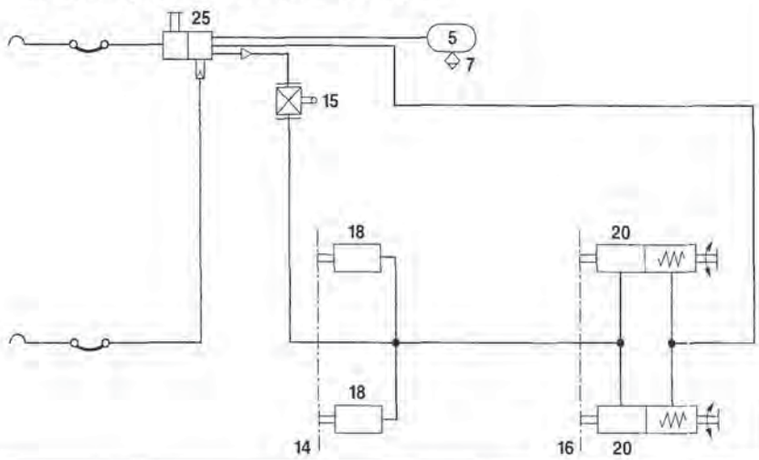


19 Relay valve,
20 Combination brake cylinder,
21 Service-brake valve with pressure limiter,
22 Pressure limiter valve,
23 Parking-brake valve with pressure limiter,
24 Precharger cylinder, dual-circuit,
25 Trailer-brake valve,
26 Loaded/empty valve,
27 Ancillary systems, (e.g. Engine braking system).

Main components of a modern compressed-air power-brake system *(Fig. D)*
a) Energy supply device, b) Reservoir, c) Brake valves, d) Trailer control and supply device, e) Braking-force control, f) Wheel brakes.



Two-line braking system for trailer/semitrailer *(Fig. E)*



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Basic comments on "kinked" braking-force metering

Load-sensitive braking-force metering devices adjust the installed braking-force metering to the dynamic (ideal) metering level and prevent the wheels of a given axle from locking prematurely. However, at low adhesion coefficients and low rear-axle load, it is possible for the wheels to lock. This is because the metering device's switchover point can enter the unstable range of braking-force metering if high engine braking torque (retarder braking torque), tolerance fluctuations in the metering device and/or high fluctuations at the wheel brakes occur.

A braking system which works precisely under all braking conditions is only possible if braking-force metering is optimized. Commercial vehicles with extreme differences between empty and fully laden conditions require braking systems with automatic load-sensitive braking-force control (ALB) on the rear axle together with empty/loaded valves (in order to increase the working range of the ALB, Fig. F).

Wheel brakes

In future, disc brakes will be fitted more and more on medium- to heavy-duty commercial vehicles (at least on the towing-vehicle's front axles). Currently, drum brakes are prevalent worldwide.

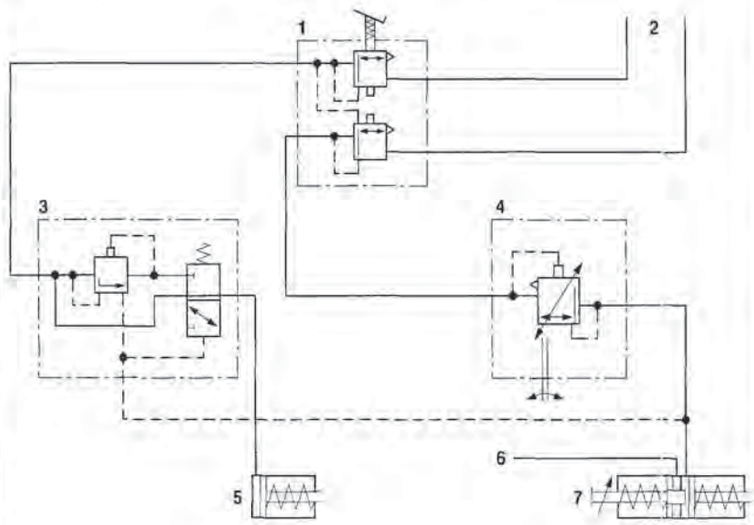
The brake factor C^* as an assessment criterion for brake performance, indicates the ratio of braking force to actuating force. This value takes into account the influence of the internal transmission ratio of the brake as well as the friction coefficient, which in turn is mainly dependent on the parameters speed, brake pressure and temperature (Fig. G).

Drum brakes

The actual construction of the drum brake depends upon the requirements imposed by brake-shoe actuation, anchorage, and adjustment.

Simplex drum brakes (Fig. H): These differ in particular according to their type of application (floating, fixed) and type of support or anchorage (rotating shoes,

ALB function diagram for four-wheel vehicles *(Fig. F)*
1 Service-brake valve, 2 From pressure supply, 3 Load/empty valve (FA), 4 ALB valve (RA), 5 Diaphragm actuator (FA), 6 From parking brake, 7 Combination brake cylinder (RA). RA = Rear axle, FA = Front axle.



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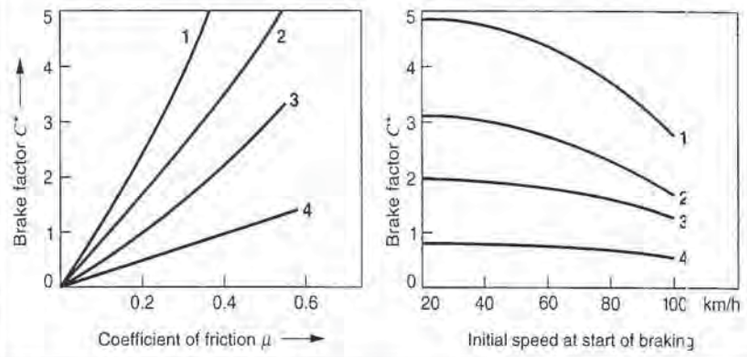
sliding shoes). Wheel brakes with floating-brake application and rotating-shoe support are common. In the case of hydraulic braking-force actuation, for instance, the brakes are applied by means of floating pressure pistons whose travel is not fixed, and which develop actuating forces which are equal in both directions. One of the shoes is the leading shoe and the other the trailing shoe. In the former, the frictional forces between the brake lining and the brake drum support the actuating force, whereas in the latter the frictional forces oppose it.

In the case of the Simplex drum brake, C^* is the sum of the values for the individual shoes, and is = 2.0 (referred to a coefficient of friction of $\mu = 0.38$; it always appears in the following C^* observations as the basis value). A disadvantage of this design is the considerable difference in the braking effect between the two brake shoes, and the resulting greatly increased wear on the leading shoe as compared to the trailing shoe.

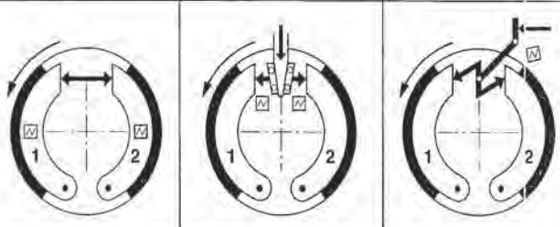
For this reason, the trailing shoe often has a much thinner lining than the leading shoe.

The simplex drum brake can also be actuated by means of a wedge unit (with integrated adjusting mechanism). This has become more and more prevalent,

Brake factor C* as a function of the coefficient of friction and initial road speed *(Fig. G)*
1 Duo-servo drum brake, 2 Duo-duplex drum brake, 3 Simplex drum brake, 4 Disc brake.



Simplex drum brakes *(Fig. H)*

Design	Rotating shoe	Wedge	S-cam
Operating principle			
Brake factor	$C^* = C_1 + C_2$		$C^* = 4/(1/C_1 + 1/C_2)$
Brake shoes	1 Leading shoe, 2 Trailing shoe		

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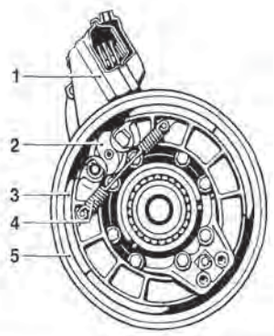
particularly in light and medium commercial vehicles with compressed-air braking systems (Figs. H, K).

The type of wheel brake used most often in heavy-duty commercial vehicles is the pneumatic S-cam simplex drum brake with fixed application (Fig. J).

Advantages:
- Uniform lining wear on leading and trailing shoes as a result of fixed application,
- Long lining life,
- An application mechanism which is simple, reliable and insensitive to temperature. It comprises diaphragm cylinder, automatic slack adjuster, brake shaft and S-cams,
- Little change in brake factor C^*,
- Simple operation of the parking-brake system via spring-brake actuators,
- Precise adjustment by means of automatic slack adjusters.

Disadvantages:
- High internal forces and thus relatively heavy brake construction, as unequal cam forces occur and lead to high free bearing forces,
- Relatively low brake factor C^*, which means considerable application work when braking,
- Due to the roughly equal application travel of leading and trailing shoes, compared to individual shoes, the application forces behave in the opposite manner.
- For the same coefficient of friction, the brake factor C^* is somewhat lower than that of simplex brakes with hydraulic or pneumatic application.

Simplex drum brake with S-cam *(Fig. J)*
1 Diaphragm actuator, 2 S-cam, 3 Brake shoes, 4 Return spring, 5 Brake drum.



Duo-duplex drum brakes: The Duo-duplex brake with two leading shoes (for instance with wedge-unit control) (Fig. K) is rarely used nowadays.

This brake features floating application and the resulting sliding shoe anchorage. An advantage of this brake type is the practically equal brake-lining wear on both shoes and the significantly higher internal transmission ratio in comparison to simplex drum brakes. Twin leading shoes achieve characteristics of $C^* \approx 3.0$ which, however, can not be sustained constantly for long periods due to the tendency of this type of drum brake to fade.

Duo-servo drum brakes: In the past, these were widely used in light commercial vehicles (particularly on the rear axle). This brake's primary characteristic is the fact that both when driving forwards and when reversing, the support force of the primary shoe is used as the actuation force for the secondary shoe. The brake factor is $C^* \approx 5.0$.

The popularity of the duo-servo brake lies in the fact that the high brake factors achieved permit even relatively heavy vans and light-duty trucks up to a weight of approx. 7.5 t to be equipped with vacuum-assisted braking systems. At the same time, the manually operated parking brake incorporated in such systems generates a braking torque of considerable magnitude. However, under conditions of high thermal stress, significant brake-factor fluctuations occur. This fact limits the range of application of this type of brake and necessitates a braking-force metering system which is precisely adapted to the vehicle in question. In future wheel-braking systems, duo-servo drum brakes will for the most part no longer be used for the service-braking system.

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Disc brakes

Apart from their use in express coaches, disc brakes are currently fitted to medium and heavy commercial vehicles.

The advantages of disc brakes in comparison to drum brakes are:
- Brakes can be applied with far more sensitivity,
- Equal wear of the inboard and outboard brake pads if the appropriate degree of heat dissipation is provided,
- Less tendency to brake noise,
- Relatively constant characteristics with minimal fade tendency.

Disadvantages:
- Shorter brake-lining life,
- Usually higher acquisition and operating costs (in comparison to drum brakes).

The high degree of adaptive braking required at high highway speeds is handled better by disc brakes. The brake discs are less susceptible to cracking than the drums of the drum brakes, apart from which disc brakes are less subject to fading. The brake factor of the disc brake is $C^* \approx 0.76$, referred to the basis value of $\mu = 0.38$.

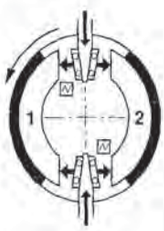
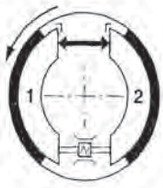
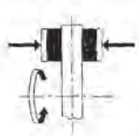
Floating-caliper disc brakes are currently replacing the fixed-caliper brakes used in the past. This development is the result of efforts to design lighter and cheaper brake assemblies which are more temperature-resistant. Floating calipers have a positive effect on variability and consistency of braking effect.

Automatic adjustment of wheel brakes

Brake-lining wear increases the clearance between brake lining and brake drum, and thus increases the braking distance. If the clearance is not adjusted correctly, in extreme cases the piston stroke in the brake cylinder may increase to such an extent that there is no braking effect. Automatic adjustment to the correct clearance is effected when the wheel brake is released.

When the vehicle is braked, the piston stroke in the brake cylinder necessary to bridge the total clearance can be divided into three sections:
- Preset constructive clearance between brake lining and brake drum/disc,
- Clearance resulting from lining wear,
- Clearance dependent on the elasticity of the brake drum/disc and of the brake linings, as well as on the transmission of force between brake cylinder and wheel brake ("elasticity clearance").

Duo-drum brakes and disc brake *(Fig. K)*

Design	***Duo-Duplex*** *Wedge*	***Duo-Servo*** *Positive-action adjustment*	***Disc brake***
Operating principle			
Brake factor	$C^* = C_1 + C_2$	$C^* = C_1 + C_2(k_1 + k_2 \cdot C_1)$	$C^* = 2 \cdot \mu$
Brake shoes	1 Leading shoe, 2 Trailing shoe		–

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A slack adjuster automatically ensures the correct adjustment (Fig. L).

Parking-brake system

The braking systems with spring-type brake actuators which are usual in commercial vehicles above 7.5 t are a convenient form of parking-brake and service-braking system. In these, in a purely compressed-air braking system, spring-type brake cylinders from the parking-brake system and diaphragm brake cylinders from the service-braking system are combined.

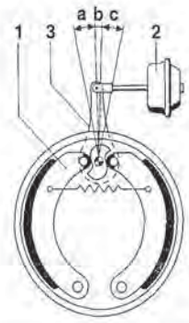
In the disengaged position the four-circuit protection valve and the handbrake valve connect the reservoirs of the service-braking system with the spring compression chamber, and maintain the spring under tension. In vehicles with a connection for the trailer/semitrailer braking system a buffer reservoir is also located in this line. When the handbrake valve is actuated, the pressure in the compression chamber is reduced. Consequently partial braking occurs at first and, with the handbrake valve still actuated, a pressure reduction down to the "surrounding conditions" takes place, and thus full braking of the spring-brake actuator (secondary braking), occurs. Further actuation defines a "Parking setting". With the aid of a further lever setting, in vehicle combinations only the tractor vehicle is braked, and not the entire tractor-trailer rig. As well as this test setting for checking the efficiency of the mechanical parking brake with a trailer attached, EU/ECE legislation for instance stipulates a safeguard emergency-air supply, as well as a nine-fold actuation and release using the energy reserve, a warning device indicating that the spring-brake is beginning to function, and an auxiliary release device.

Retarder braking systems (additional retarding braking systems)

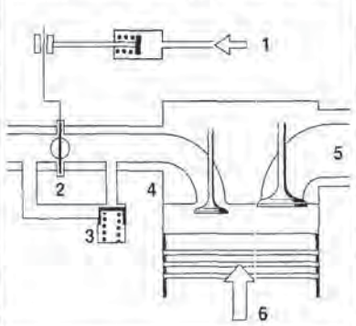
The wheel brakes used in passenger cars and commercial vehicles are not designed for continued retarding operation. In a prolonged period of braking (e.g., when driving downhill) the brakes can be thermally over-stressed, causing the braking effect to be reduced ("fading"). In extreme cases (particularly if the service-braking system has been badly maintained) this may even lead to complete braking-system failure. In order to permit continuous downhill braking, therefore, particularly vehicles with a high laden weight are frequently fitted with a wear-free supplementary retarding braking system in addition to their normal wheel brakes. This system is independent of the wheel brakes, and is also used for braking the vehicle so as to comply with speed limits.

This reduces brake wear as well as increasing braking comfort for the driver. In commercial vehicles two basic types of

Brake clearances *(Fig. L)*
a Due to elasticity. b Due to wear. c Constructive. 1 Brake shoe, 2 Diaphragm cylinder, 3 Automatic slack adjuster.



Exhaust brake with additional pressure-control valve *(Fig. M)*
1 Exhaust-flap actuation (compressed air), 2 Bypass, 3 Pressure-control valve, 4 Discharge, 5 Intake, 6 Piston (4th exhaust stroke).



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additional retarding braking system are used:

1. Exhaust (engine) braking system

The exhaust braking power is comprised of the drag power together with the braking power (effected by the throttling of exhaust-gas flow in the exhaust stroke). The maximum drag power of standard engines is 5...7 kW/l depending upon engine swept volume. In contrast, standard engines with conventional exhaust brakes ("exhaust-flap brakes") achieve braking powers of 14...20 kW/l. A further increase in exhaust braking power is only possible by means of additional design modification. Decompression exhaust brakes (e.g., "C-brake", "Jake brake", "Dynatard" or "Powertard") as well as the constant-throttle exhaust brake can significantly improve braking power (Fig. P).

Exhaust brake system with exhaust flap:

The exhaust-flap brake is still the most common system in use today. In this system the driver is able to close a butterfly valve in the exhaust tract by means of compressed air (relatively little constructional outlay is involved). As a result, a counterpressure is generated in the exhaust-gas system which must be overcome by each piston during its exhaust stroke (Fig. M).

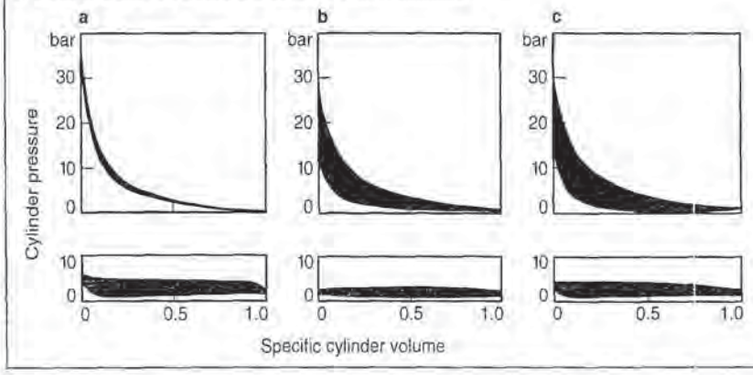
Through the use of a pressure-regulator valve in the bypass, in conjunction with an exhaust flap, the braking power can be increased in the engine's lower and medium speed ranges. At higher revs, the pressure-regulator valve prevents pressure increases beyond the limits which could lead to valve or valve-gear damage.

Exhaust brake system with constant throttle:

The conventional exhaust brake with exhaust flap utilizes only the energy available in the engine's gas-exchange process, i.e. during the 4th (exhaust) and 1st (intake) working strokes. Specific decompression during the 2nd and 3rd working strokes releases part of the compression energy. The "pressure-volume" diagrams below show the pressure curves in a cylinder for the "exhaust flap" and "constant throttle" exhaust braking systems as well as for the combination of both systems (Fig. N). The braking power when using the constant throttle system, as opposed to the flap system, is obtained principally in the engine's high-pressure cycle.

The installation of a small restriction valve in the bypass to the exhaust valve permits an increase in braking power. This valve is actuated by compressed air, in the same way as in the servo-cylinder of the exhaust-brake flap. During exhaust-brake operation the valve can remain open, thus providing a constant throttle cross-section (Fig. O).

Principle of operation of exhaust (engine) braking systems as illustrated by "pressure-volume" diagram (n_M = 1700 min^{-1}) *(Fig. N)*
a) Exhaust-brake flap closed, b) Constant throttle actuated,
c) Exhaust-brake flap closed and constant throttle actuated.



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2. Retarders

Retarders are increasingly being used as wear-free additional retarding braking systems in trucks and buses. In this way, the demands of legislation are met, active vehicle safety is increased due to the reduction of the stress placed on the service-braking system, and vehicle economy is increased as a result of higher average speeds and the reduction of brake-lining wear.

Retarders can be fitted between the engine and the transmission (primary retarders) or between the transmission and the driven axle(s) (secondary retarders). The disadvantage of primary retarders lies in the unavoidable interruption of power transmission, and thus braking effect, which occurs during gear changing with manual transmissions. Primary retarders can be employed in conjunction with power-shift transmissions, and in the case of steep downhill gradients which are negotiated at low speeds, this provides for a slight advantage in comparison to the secondary-retarder system. (Fig. R).

Currently, two basic design concepts represent the state-of-the-art:

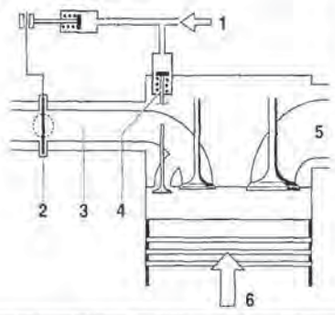
Hydrodynamic retarders: This retarder works in the same way as the Foettinger clutch (Fig. S). The rotor converts the mechanical energy supplied by the drive shaft into kinetic energy of a fluid. This kinetic energy is in turn converted into heat at the stator, which means that the fluid used must be cooled.

A hand lever or the brake pedal (in the case of an integrated retarder) transmits the driver's braking-power requirements. In conjunction with an electronic control circuitry, a defined control air pressure is set. By means of a corresponding quantity of air, this control air pressure then forces a quantity of oil into the retarder's working area between rotor and stator. The flow energy absorbed by the oil as a result of the vehicle speed and the associated rotor motion is braked by the stator's fixed blades. This in turn effects braking of the rotor and thus of the entire vehicle. Characteristics are:

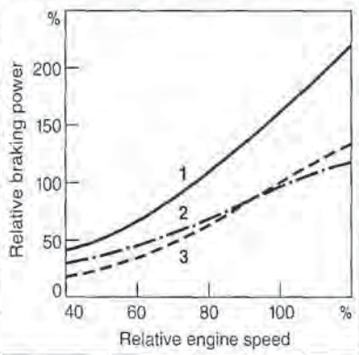
- Adequate cooling-circuit dimensioning is necessary in order to dissipate to the engine cooling circuit the heat generated by braking. An oil/water heat exchanger is used,
- Relatively complex design,
- Low weight of the hydrodynamic retarder which is integrated directly into the transmission,
- High specific braking powers,
- Very sensitive control of applied braking torque,
- Fan losses occurring when the retarder is not running must also be taken into consideration in retarder design.

In the hydrodynamic secondary retarder, an almost constant braking torque is available over a broad propshaft-

Exhaust brake with exhaust flap and constant throttle *(Mercedes-Benz) (Fig. O)*
1 Compressed air, 2 Exhaust flap, 3 Exhaust, 4 Constant throttle, 5 Intake, 6 Piston (2nd working stroke).



Braking-power curves *(V8 engine) (Fig. P)*
1 Exhaust flap and constant throttle, 2 Constant throttle, 3 Exhaust flap.



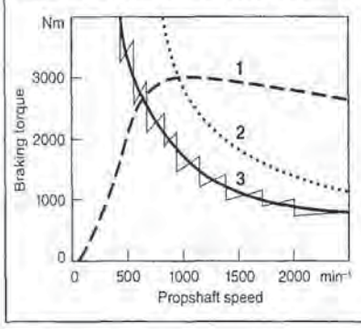
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speed range (Fig. T). Below approx. 1,000 min^{-1} the braking torque drops steeply. As a result of this characteristic, conventionally designed hydrodynamic retarders are particularly suited to high-speed transport vehicles (overland transport).

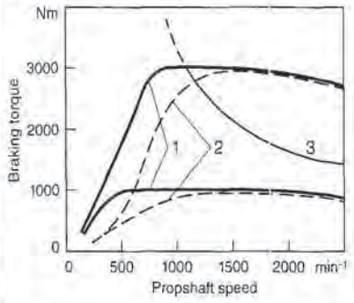
Modern retarder designs rectify the unfavorable braking torque characteristic of the secondary retarder design described above by providing high braking torques even at still low propshaft speeds. A spur-gear stage with a transmission ratio of approx. 1 : 2 drives such "boost retarders", which are fitted to the side of the transmission (Fig. S). Even in the propshaft lower speed range, a microprocessor control ensures acceptable braking-torque levels via a proportional valve.

Hydrodynamic retarders can only be used as continuous-operation braking systems in certain situations for limited

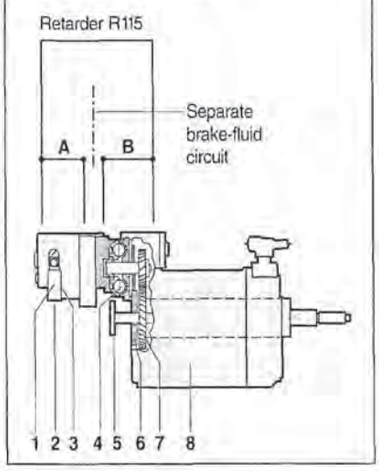
Braking-torque characteristics of primary and secondary retarders *(Fig. R)*
1 Secondary retarder, 2 Limit of cooling effect under continuous load (300 kW), 3 Primary retarder on input side of 16-speed gearbox.



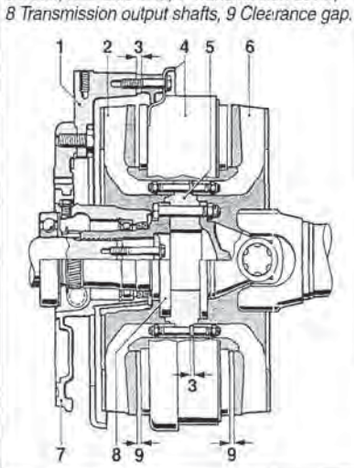
Hydrodynamic retarder performance characteristics *(Fig. T)*
1 Boost retarder, 2 Conventional retarder, 3 Cooling-power limit at continuous load (300 kW).



Boost retarder *(Fig. S)*
1 Transmission, 2 Spur gears (transmission ratio), 3 Gearbox flange, 4 Rotor, 5 Stator, 6 Compressed-air cylinder, 7 Proportional valve, 8 Wiring connector.



Electrodynamic retarder *(Fig. U)*
1 Star-shaped bracket, 2 Rotor, transmission side, 3 Spacers (for adjusting clearance), 4 Stator with coils, 5 Intermediate flange, 6 Rotor, rear-axle side, 7 Transmission cover, 8 Transmission output shafts, 9 Clearance gap.



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periods. The maximum cooling power of modern diesel engines is approx. 300 kW. Thus, as a result of the coupling of the engine and retarder cooling circuits, there exists a risk for both engine and retarder if additional safety measures are not implemented. For this reason, thermo switches are used to restrict the retarder's braking power so that thermal equilibrium is ensured.

Electrodynamic retarders: Currently common electrodynamic retarders have a mounting in the form of a stator, on which field coils are mounted (Fig. U). The rotors mounted on both sides of the drive shaft are ribbed for better heat dissipation. In order to brake the vehicle, voltage is applied to the field coils (from the battery or alternator) which thus generate a magnetic field which induces eddy currents in the rotors as they pass through the field. This generates a braking-torque level which is dependent on the stator excitation and on the air gap between rotor and stator.

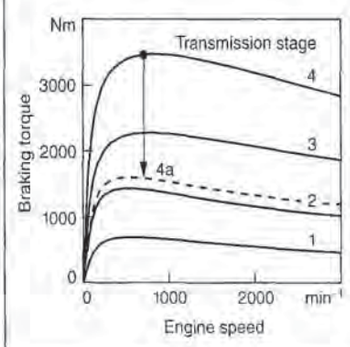
Characteristics are:
- Dissipation to the atmosphere of the heat produced,
- Relatively simple design,
- Relatively heavy construction,
- Trouble-free operation only with adequate current supply,
- Heating of the retarder leads to a reduction in braking torque,
- High braking powers even at low vehicle speeds,
- Braking power influenced by rotor blading, air-flow conditions around the eddy-current brake, and by ambient temperature.

In contrast to the conventional hydrodynamic secondary retarders, electrodynamic retarders provide relatively high braking torques at low driveshaft speeds (Fig. V).

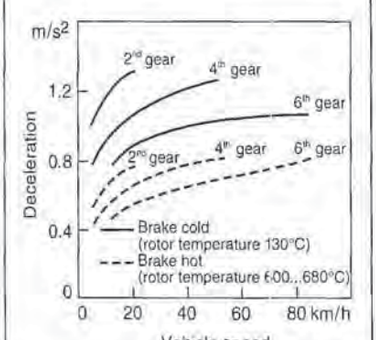
The significant reduction in braking torques of the electrodynamic retarder as the rotor temperature increases result from the thermal safeguard (Fig. W). Vehicle deceleration is reduced as thermal stress on the electrodynamic retarder increases.

In order to prevent temperature-related destruction of the retarder when the vehicle is being braked, a bi-metal switch restricts the current supply to half of the eight coils when the stator temperature reaches approx. 250 °C (Fig. W).

Braking-torque characteristic of an electrodynamic retarder *(Fig. V)*
4a Braking power when the cooling power limit has been reached (switching stage 4).

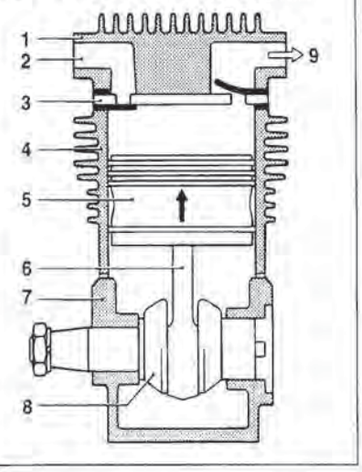


Influence of transmission ratio and rotor temperature on the performance of electrodynamic retarders *(17 t commercial vehicle, laden) (Fig. W).*

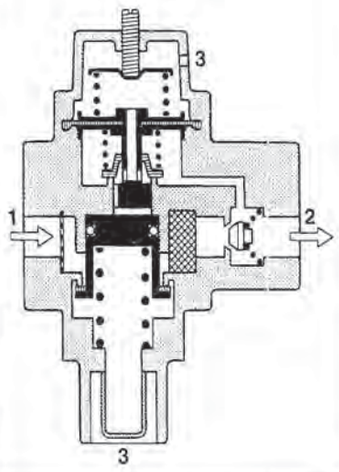


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Air compressor (compression and discharge)
1 Cylinder head, 2 Air intake, 3 Intermediate plate (with intake and outlet valve), 4 Cylinder, 5 Piston, 6 Connecting rod, 7 Crankcase, 8 Crankshaft, 9 To compressed-air system.



Pressure regulator
1 From air compressor, 2 To air reservoirs, 3 Blow-off.



Components for compressed-air brakes

Energy-supplying device

The energy-supplying device comprises the following:

- Energy source
- Pressure regulator
- Air conditioning

Energy source: The energy source is a continuously running air compressor which is driven by the engine via V-belts or toothed gears. It consists of:

- Crankcase with crankshaft (driver for power-steering pump at the free end of the shaft), bearing assembly and connections for circulating engine lubrication;
- Cylinder with piston and connecting rod;
- Intermediate plate with intake and outlet valves;
- Cylinder cover with suction and pressure connections for air and, if appropriate, the fittings required for liquid-cooled versions.

Energy-economy controls are increasingly being used in order to reduce the losses during idle operation (opening and flow resistances in the valves, lines, and pressure regulator). These are integrated in the valve plate and actuated pneumatically via an actuator. At idle, either a bypass from the compressor's pressure chamber to the input side is opened, or the intake bore is opened by rotating/shifting the intake valve.

The air compressor is usually mounted on the vehicle engine by means of base or flange attachment. In some cases it is integrated in the engine block.

During its downward stroke, the piston draws in air after the intake valve has automatically opened as a result of the vacuum. The intake valve closes at the beginning of the return movement of the piston. The air is now compressed and, after reaching a set pressure, it is conveyed via the outlet valve, which also opens automatically, into the downstream compressed-air system.

In terms of delivery rate, the aim is a volumetric efficiency of 70 % and in terms of oil consumption a maximum of 0.5 g/h is desirable.

Pressure regulation: The pressure regulator ensures that the desired pressure level is maintained. Two principal types of regulation are used:

1. Regulation in which the pressure regu-

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lator has no influence on the energy source (in compressors with maximum speeds >2,500 min^{-1}).

The pressure regulator switches off when the desired maximum operating pressure is reached, and returns the air supplied by the compressor to atmosphere during the subsequent no-load period. If the pressure in the air reservoirs reaches the lower operating-pressure limit the pressure regulator switches on again and supplies the air delivered by the compressor to the air reservoirs.

2. Regulation in which the pressure regulator influences the energy source (in compressors with maximum speeds <2,500 min^{-1}).

When the desired maximum operating pressure is reached, the regulator applies pressure to a plunger in the compressor and opens its intake valve. Without being supplied to the air reservoirs, the intake air is expelled through the intake fitting. When the pressure in the air reservoirs reaches the lower operating-pressure limit, the regulator switches and the intake valve can again automatically open and close, and the air reservoirs are filled.

Pressure level: In towing vehicles, values between 7 and 12 bar (low pressure) and between 14 and 20 bar (high pressure) are used. In the two-line braking system, the pressure in the lines connecting towing vehicle and trailer is between 6 and 8 bar.

Compressed-air conditioning: The air must be conditioned in order to ensure proper operation of the downstream braking-system components. Impurities in the air can cause leaks in the control valves, and water in the compressed air leads to corrosion or icing-up during frost. To combat these problems, an air drier is connected downstream of the air compressor. Such a configuration makes it unnecessary to add antifreeze to the system.

Single-box air drier

Basically, an air drier consists of a desiccant box and a housing. The housing incorporates the air passage, a bleeder valve, and a control element for granulate regeneration. Normally, the granulate is regenerated by switching in a regeneration air reservoir with integral regenerating throttle.

Function: When the bleeder valve is closed, the compressed air from the air compressor flows through the desiccant box and from there to the supply-air reservoirs. When flowing through the desiccant box, water is removed from the moist compressed air by means of condensation and adsorption. At the same time a regeneration-air reservoir with a volume of approx. 4...6 liters is filled with dry compressed air.

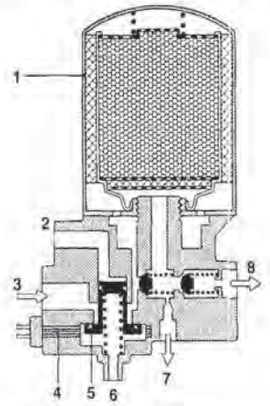
The granulate in the desiccant box has a limited water absorption capacity and must therefore be regenerated at regular intervals. In the reverse process, dry compressed air from the regeneration-air reservoir is reduced to atmospheric pressure via the regenerating throttle, flows back through the moist granulate from which it draws off the moisture, and flows as moist air via the opened bleeder valve into the atmosphere. On air driers with an integrated pressure regulator, its control element is fitted to port 4 of the air-drier valve body.

Control device

Normally, the control device comprises the brake pedal and all devices up to the point where influence is exerted on the control units.

Single-box air drier

1 Desiccant box, 2 From pressure regulator, 3 From air compressor, 4 Heating element, 5 Bleeder valve, 6 Exhaust, 7 To regeneration-air reservoir, 8 To supply-air reservoir.



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Transmission device

The transmission device comprises the following:

- Circuit isolation (e.g., multiple-circuit protection valve),
- Energy storage (e.g., air reservoirs),
- Control units (e.g., brake valves),
- Load-sensitive braking-force metering device (e.g., automatic load-sensitive braking-force control), and
- Brake cylinders or servo cylinders.

These components work together as shown in the block diagram below of a power-braking system with dual-circuit service-braking system (see also the braking-system diagram on P. 676).

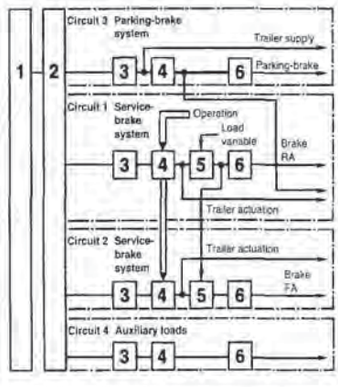
Functions and design of components:

Circuit isolation: Separation of the circuits with respect to one another in the event of damage in one circuit, as well as preservation of the operational integrity of the intact circuits.

Circuit isolation is primarily achieved by a combination of overflow valves grouped together as a unit; the operation of these valves is ensured at both low and high delivery rates.

Energy storage: Provision of the required volume of energy for all circuits in the braking system, including provision of energy in the event of failure of the energy source. Commercially available air reservoirs are used for this purpose; these have corresponding safety allowances for excess pressure and rust.

Transmission device in a power-braking system *(block diagram)*
1 Energy supply, 2 Circuit isolation, 3 Energy storage, 4 Control equipment, 5 Automatic load-sensitive braking-force metering, 6 Brake or servo cylinder, VA Front axle, HA Rear axle.



Control equipment

This equipment is used to control the metering of the required pressure to the corresponding parts of the system. Mechanically, hydraulically or pneumatically actuated or controlled reaction valves are used to control the pressure at the output of each valve as a function of the input variable. Due to the wide range of applications, a correspondingly large number of different components are in use. Dual-circuit control valves are also required for dual-circuit service-braking systems. Proper braking-system operation requires good control behavior, good pressure-metering capability, fast reaction times and low braking-system hysteresis.

Automatic load-sensitive braking-force metering (ALB)

Automatic braking-pressure control as a function of vehicle load. Load is often determined by spring compression (in the case of steel-spring suspension) and bellows pressure (in the case of pneumatic suspension). A control valve with a variable reaction-surface area reduces the output pressure in the valve in relation to the input pressure as a function of spring compression or bellows pressure.

Brake cylinders or servo cylinders

Convert the pressure applied to the braking system into braking force. Both plunger- and diaphragm-type cylinders are used. Diaphragm-type cylinders are primarily used for the service-braking system, while spring-brake actuators are used for the parking system. In the case of axles which are acted upon by both the service-braking system and the parking-brake system, combined single-chamber spring-brake actuators (so-called combination cylinders) are used in braking systems without hydraulic force transmission.

Wheel brakes (see P. 678)

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Service-brake valve

Two tandem-arranged control valves are actuated by a common device (brake pedal with transmission). The synchronized opening of both circuits is ensured by identical spring and valve sealing forces as well as by the mechanical overcoming of the opening forces in both control valves. In the braking position, the rocking piston between the control circuits is subjected at its two ends to the currently applied braking pressure and thus guarantees that the circuits are synchronized. The preloaded travel spring provides small response travels of the service-brake valve. The interaction of the force of the reaction piston with the travel spring enables the system to execute the necessary control travels independently. Dual-circuit sealing of the rocking piston ensures the required safety.

Parking-brake valve

Today's compact design of the parking-brake valve is a direct result of the lack of installation space at the instrument panel (where it used to be fitted). The parking-brake valve always actuates the brake cylinders via relay valves.

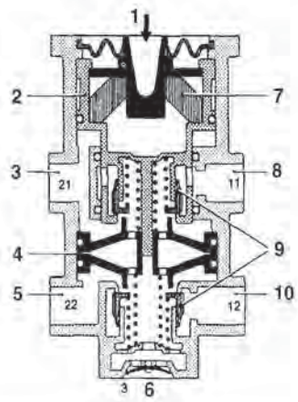
A hand lever (actuating lever) adjusts an internal valve seat by means of an eccentric and a linking strap. In doing so, it controls a double-seat valve in which compressed air from above and the force of compression springs from below act on a valve piston. In the brake position the actuating lever latches into position automatically, and the space above the valve piston is purged of air. As many intermediate settings between the drive and brake settings as desired are possible.

If the actuating lever is moved beyond the brake position, the auxiliary (test) valve is actuated and compressed air flows from the supply-air reservoir into the connecting line to the trailer-control valve. This leads to the towing-vehicle braking effect being retained, although it is cancelled for the trailer.

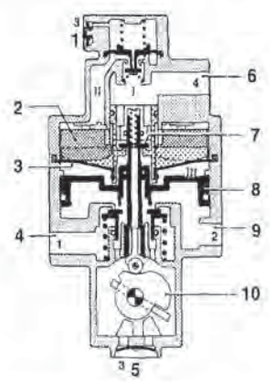
Automatic load-sensitive braking-force regulator

This device is connected between the service-brake valve and the brake cylinders. Depending on the vehicle load, it regulates the applied braking pressure. The device has a reaction diaphragm with variable active area. The diaphragm is held in two radially arranged and interlocking rakes. Depending on the position of the control-valve seat in the vertical direction, there is a large reaction area

Service-brake valve
1 Actuation, 2 Reaction piston, 3 Brake circuit 1, 4 Rocking piston, 5 Brake circuit 2, 6 Exhaust, 7 Travel spring, 8 Supply circuit 1, 9 Control valves, 10 Supply circuit 2.



Automatic load-sensitive braking force regulator
1 Exhaust, 2 Rake, 3 Reaction diaphragm, 4 From air reservoir, 5 Exhaust, 6 From service-brake valve, 7 Control valve, 8 Relay piston, 9 To brake cylinders, 10 Rotary cam.



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(valve position at bottom) or a small reaction area (valve position at top). Consequently, the brake cylinders are supplied via an integrated relay valve with a pressure which is lower than (unladen), or which is the same as (fully laden), that coming from the service-brake valve. The regulator is mounted on the vehicle frame and senses the compression position of the axle by a rotary lever via linkages. The rotary cam moves the valve rod accordingly in the vertical direction and thus determines the valve position. The pressure limiter which is integrated into the device at the top allows a small partial pressure to flow in on the top side of the diaphragm. Thus, up to this pressure there is no reduction in the brake-cylinder pressure. This results in the synchronous application of the brakes on all vehicle axles. If the rotary lever breaks, the applied pressure flows to the brake cylinders at a ratio of 2:1.

Combination brake cylinder for wedge brakes

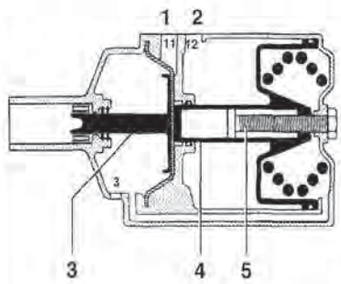
The combination brake cylinder consists of a single-chamber diaphragm cylinder for the service brakes and a spring-brake actuator for the parking brakes. The cylinder and spring-brake actuator are in tandem and act on a common pressure rod. They can be actuated independently of each other. Simultaneous actuation results in the addition of their forces. The central release screw permits tensioning the spring of the spring-brake actuator without compressed air having to be applied. This is the setting when installed in the vehicle. Following installation, the release screw is screwed into the spring-actuator cylinder, and the spring acts via the piston rod on the wedge mechanism. The entry of compressed air into the spring-actuator cylinder (when parking brake is released) forces the piston back against the action of the spring, loads the spring and releases the brake (position illustrated). When the service brakes are operated, compressed air flows into the diaphragm cylinder, actuating the wedge mechanism by way of the diaphragm and the thrust rod. A drop in air pressure releases the brake.

The principle described above is modified for cam-operated brakes and for disc brakes.

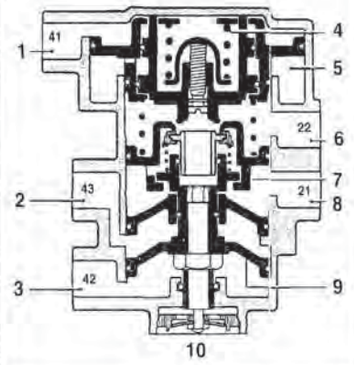
Trailer-control valve

In two-line braking systems, the trailer-control valve which is installed in the towing vehicle controls the trailer's service

Combination brake cylinder for wedge brakes
1 Single-chamber cylinder control line, 2 Spring-brake actuator control line, 3 Pressure rod, 4 Piston rod, 5 Release screw.



Trailer-control valve
1 Service-brake circuit 1, 2 Parking-brake circuit, 3 Service-brake circuit 2, 4 Pilot spring, 5 Control piston 1, 6 Control line to trailer, 7 Control-piston unit, 8 Supply line to trailer, 9 Control piston 2, 10 Exhaust.



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brakes. This multi-circuit relay valve is triggered by both service-brake circuits and by the parking-brake system.

In the driving position, the supply chamber as well as the chamber of the parking-brake circuit are each subjected to equal pressure, and the trailer-control line is exhausted through the central exhaust port. A rise in pressure upstream of the control piston of brake circuit 1 (top) and/or of brake circuit 2 (bottom) leads to a corresponding rise in pressure in the trailer-control line. Brake circuit 1 is equipped with a larger control piston than brake circuit 2, which means that it has priority over the control piston of circuit 2. This priority ends when the pilot pressure is reached (pilot-spring force is exceeded). A reduction in pressure in the service-brake circuits leads to the same reduction in the trailer-control line. The exhausting of the parking-brake circuit (braking) increases the pressure in the chamber to the trailer-control line. The application of air to the parking-brake circuit (releasing) again exhausts the trailer-control line.

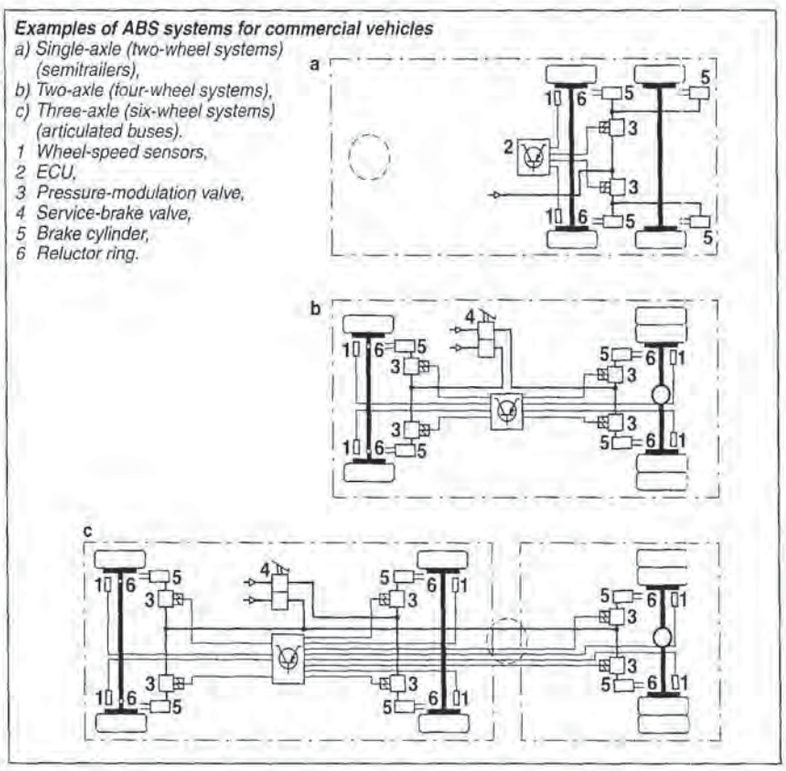
Mufflers

Current and future legislation stipulates the reduction of compressed-air valve exhaust noise. Absorption-principle mufflers are used whose size depends on the exhaust-air quantity, air pressure, exhaust duration, and required noise level.

The muffler is a cylinder with radial and axial slits, and is filled with insulating material. Noise damping is due to inflow geometry, the insulation cartridge, and the exhaust slits with spherical exhaust flow. Mufflers are attached to the compressed-air valves via threaded, snap-action, bayonet, or pipe connections. Some versions fulfill the "Low-Noise Vehicle" conditions [72 dB(A)].

Examples of ABS systems for commercial vehicles

a) Single-axle (two-wheel systems) (semitrailers),
b) Two-axle (four-wheel systems),
c) Three-axle (six-wheel systems) (articulated buses).
1 Wheel-speed sensors,
2 ECU,
3 Pressure-modulation valve,
4 Service-brake valve,
5 Brake cylinder,
6 Reluctor ring.



Antilock braking systems (ABS) for commercial vehicles

The ABS prevents the wheels from locking when the vehicle is overbraked. The vehicle therefore retains its directional stability and steerability even under emergency braking on a slippery road surface. The stopping distance is often shorter than with locked wheels. ABS prevents the danger of jackknifing in the case of vehicle combinations.

In contrast to passenger cars, commercial vehicles have pneumatic braking systems. Nevertheless, the functional description of the ABS control circuits for passenger cars (see P. 659) also applies in principle to commercial vehicles.

The antilock braking system as used in commercial vehicles consists of wheel-speed sensors, an electronic controller (ECU) and pressure-modulation valves. The ABS regulates the brake pressure in each brake cylinder by increasing the pressure, holding it constant or reducing it by exhausting to atmosphere.

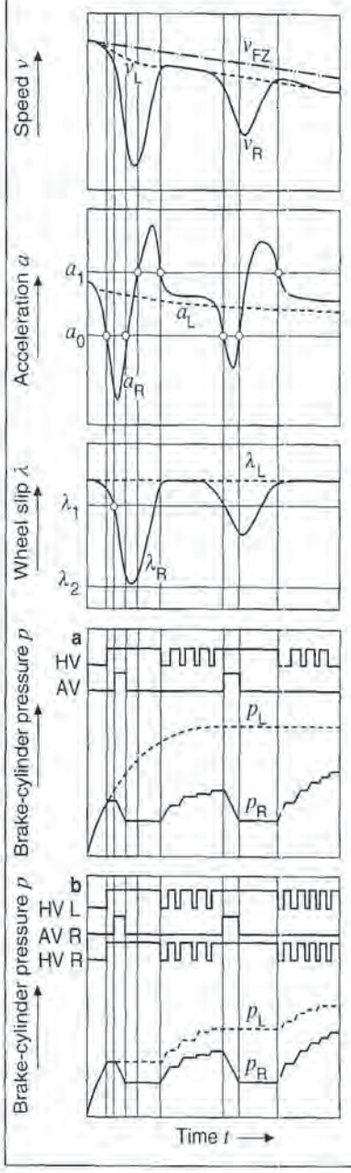
Individual control (IR)

This process, which sets and controls the optimum brake pressure individually for each wheel, produces the shortest braking distances. Under μ-split conditions (different friction coefficients between right-hand and left-hand wheels, e.g., black ice at edge of road surface, good grip at center of road surface) braking produces a high yawing moment about the vertical axis of the vehicle, thus making short-wheelbase vehicles difficult to control. In addition, this is coupled with a high steering moment as a result of the positive steering roll radii in commercial vehicles.

Select-low control (SL)

This process reduces the yawing and steering moments to zero. This is achieved by applying the same brake pressure to both wheels on the same axle. This requires only one pressure-control valve for each axle. The pressure applied is chosen to suit the wheel with less grip (select low), the wheel with more grip is braked less heavily compared to the individual control (IR) method. Under

ABS control processes
Example: Braking on μ-split.
a) IR individual control (rear axle), b) IRM individual control modified (steering axle).
HV = Holding valve, AV = Outlet valve.
Subscripts: FZ vehicle, R Right wheel, L Left wheel, 0, 1, 2 thresholds.



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μ-split conditions, braking distances are longer but steering control/directional stability of the vehicle is better. If grip conditions (coefficients of friction) are the same on both sides, braking distances, steering control and directional stability are virtually identical with systems which control each wheel individually (IR).

In the case of similar control principles with higher pressure levels than that in the SL method, the wheel running with the lower friction coefficient is forced to run with higher slip until it eventually locks up. Under μ-split conditions, braking distances are shorter compared with pure SL control systems. Steering control and directional stability of the vehicle may be very slightly worse. Normally, due to the lower frictional coefficient, there is no tire damage at the wheel which locks up.

Individual control modified (IRM)
This process necessitates a pressure-modulation valve at each wheel of the axle. It reduces yawing and steering moments only as far as necessary, and limits brake-pressure difference between left and right sides to a permissible level. This leads to the wheel which is running at the high friction coefficient being braked less strongly. This compromise results in a braking distance which is only a little longer than that for individual control (IR), but it does ensure the safe control of vehicles which tend to handle critically.

Commercial-vehicle ABS versions
Since 1 October 1991, within the area covered by the EU member states, ABS has been prescribed by law for the initial registration of commercial vehicles intended for trailer operation, and of semitrailer tractors (> 16t), as well as trailers (> 10t), and busses (> 12t). This regulation has been extended so that since 1/10/1998 all busses and since 1/10/1999 all trucks and trailers (> 3.5t) have had to be fitted with ABS.

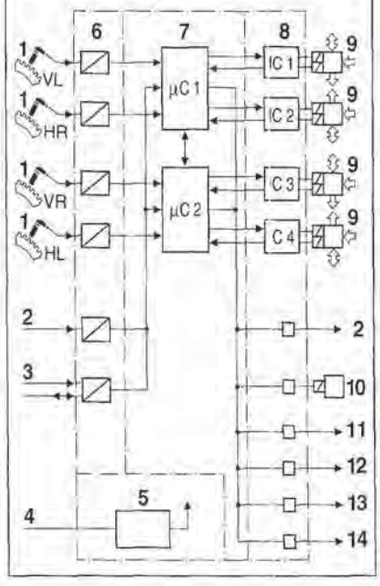
The law stipulates 3 categories of ABS which differ from each other with respect to their specifications regarding retardation and the wheel and vehicle behavior on μ-split road surfaces. Most European vehicle manufacturers only use ABS systems in category 1. Only those have to conform to all specifications of Directive 71/320/EEC.

All ABS installations must be equipped with a warning lamp which must light up for at least 2 secs after switching on the driving switch. This lamp is to be referred to by the driver as a form of visual inspection. If it lights up while the vehicle is being driven, this indicates that the installation's continuous self-testing facility has detected a fault. This can result in the ABS being switched off completely.

Towing vehicles and trailers with ABS systems from different manufacturers may be combined as desired provided there is an ABS plug and socket connection between the two vehicles in accordance with ISO 7638.

On all vehicle combinations (semitrailer tractors, truck tractor/trailer combinations), optimum control of braking at the physical limits is only guaranteed when both tractor vehicle and trailer are equipped with ABS. However, even with-

ABS/TCS control unit (ECU)
1 Wheel-speed sensor, 2 TCS engine-management interface, 3 Self-diagnosis, 4 Vehicle power supply, 5 Power supply, safety unit, 6 Input stages, 7 Microcomputers 1 and 2, 8 Output stages, 9 Pressure-control valve, 10 TCS solenoid valve, 11 Warning lamp, 12 TCS Indicator lamp, 13 Retarder relay, 14 Valve relay.



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out ABS throughout (only on tractor unit or only on trailer) there are significant improvements compared to a vehicle combination which has no ABS at all.

In order to be able to provide the optimum and cost-effective equipment for the very wide range of commercial vehicles on today's roads, single-axle, 2-axle, and 3-axle ABS systems are available in accordance with the number of axles which are to be controlled. The single-axle systems as used primarily on semitrailers, individually control the wheels on a given axle. The remaining axles are supplied with the same brake pressures as the controlled axle.

2-axle systems are mainly fitted to two-axle buses, trucks, and trailers. Three-axle vehicles can also be equipped with a 2-axle ABS system insofar as two of the axles are in close vicinity to each other and can be provided with the same brake pressure (as for the single-axle ABS systems). The 3-axle systems are mainly intended for installation in long-wheelbase vehicles (e.g. articulated buses). In the case of 2-axle and 3-axle ABS installations, the steered axle uses either the IRM control method or, if this axle is only provided with a single pressure-control valve, the SL control method. Individual control (IR) is always used on commercial-vehicle rear axles.

The range of available ECU's permits further control combinations (not described here in detail). Example: If both axles of a semitrailer have wheel-speed sensors, but each side of the vehicle is only equipped with a single pressure-modulation valve, the wheels of one side of the vehicle are SL-controlled. In this case, one axle may be a lifting axle, which is automatically excluded from the control process when lifted.

All ABS systems can be equipped with single-channel pressure-modulation valves. ABS trailer systems can also be fitted with relay-type pressure-modulation valves (see Fig.).

In light commercial vehicles with pneumatic/hydraulic converters, ABS intervenes in the pneumatic brake circuit via single-channel pressure-modulation valves and defines the hydraulic brake pressure. In other versions, an ABS pressure modulator with integral solenoid valves is connected in parallel to the pneumatic/hydraulic converter. The modulators are controlled by the same ECUs as the single-channel pressure-modulation valves.

When the vehicle is running on a low-friction-coefficient road surface, the operation of an additional retarding brake (engine brake or retarder) can lead to excessive slip at the driven wheels. This would impair vehicle stability. ABS therefore monitors the brake slip and controls it to permissible levels by switching the additional retarding brake on and off.

ABS components

Wheel-speed sensor

For most applications nowadays, an application-engineered inductive sensor attached to the axle tube is used (for description of function, refer to P. 103). In contrast with car applications (rigidly attached sensors), on commercial vehicles the sensor is held in a spring sleeve. When the vehicle is in motion, the action of the wheel-bearing play and the flexing of the axle moves the sensor along its axis, thereby automatically adjusting the clearance between the sensor and the reluctor ring. If, under exceptional circumstances, the clearance gap becomes excessive, the ECU switches off the control for the wheel in question.

In the future, inductive sensors integrated in the wheel bearings and semiconductor speed sensors will be used more and more frequently due to economic considerations.

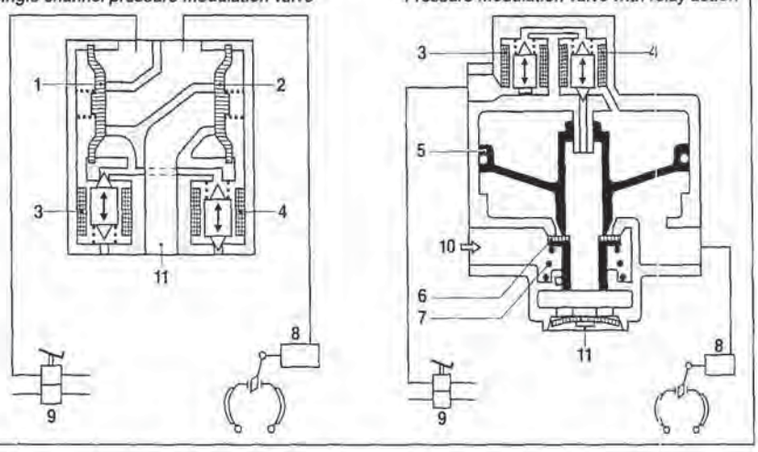
Electronic control unit (ECU)

The ECU's input stages convert the incoming sinusoidal signals from the wheel-speed sensors into square-wave signals. The wheel speeds are calculated from the frequency of the square-wave signals by (redundant) microcomputers. These speeds are used to estimate a vehicle reference speed. The brake slip for each wheel is calculated using this reference speed and the individual wheel speed. If a wheel has a tendency to lock, this is determined from the "wheel-acceleration" and "wheel-slip signals". In such a case,

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Pressure-modulation valve (schematic)
1 Holding valve, 2 Outlet valve, 3 Solenoid valve for "Maintain pressure", 4 Solenoid valve for "Reduce pressure", 5 Control piston, 6 Valve plate, 7 Compression spring, 8 Brake cylinder, 9 Service-brake valve, 10 Supply air, 11 Exhaust.

Single-channel pressure-modulation valve ***Pressure-modulation valve with relay action***



the microcomputer energizes (via the ECU output stages) the solenoids of the pressure-modulation valves which control the brake pressure in the individual wheel-brake cylinders.

The ECU contains a comprehensive program for the detection of faults throughout the entire antilock system (wheel-speed sensors, ECU, pressure-modulation valves, wiring harness). If a fault is detected, the ECU switches off the defective part of the system and stores a code detailing the faulty signal path. That code can be retrieved by a service technician with the aid of the warning lamp (flashing signal) or using an intelligent testing device (e.g. personal computer) via a standardized serial interface.

The ECU's of some European ABS manufacturers include not only the ABS function but also functions for TCS traction control and in some cases for cruise control (see also page 611). The most important factor is that depending upon the model, the ECU automatically configures itself to the required function. In other words, if the vehicle concerned is only ABS-equipped, the ECU only carries out the ABS function; if the vehicle has TCS components the ECU also controls wheel slip automatically.

Pressure-modulation valve

Single-channel pressure-modulation valves are available with and without relay action. The relay-action valves are installed in semitrailers and drawbar trailers. The standard trailer-braking system often includes relay valves which can then be replaced by ABS relay-action valves. Non-relay-action ABS valves are used in all other vehicles, i.e. in buses, trucks and tractor-trailer rigs, as well as in trailers and special vehicles. Both types of valve have 3/2 pilot valves. The non-relay-action valves thereby control 2/2 diaphragm valves, which have a sufficiently large cross-section for almost all applications. In the case of the relay-action valves, the 3/2 solenoid valves affect the pressure in the pilot chamber of a relay valve. The electronics control the solenoid valves in the appropriate combination so that the required function is performed (pressure retention or pressure release). If no pilot-valve actuation takes place, "pressure build-up" is the result.

When braking normally (that is, without ABS response = no locking tendency of a wheel), the air flows through the pressure-modulation valves unhindered in both directions when pressure is applied to or removed from the brake cylinders. This ensures fault-free functioning of the service-braking system.

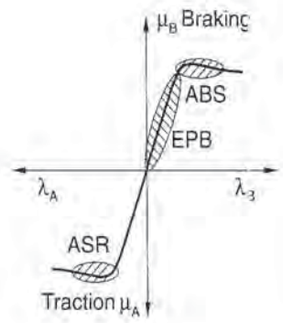
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Electronically controlled braking system (ELB) for commercial vehicles

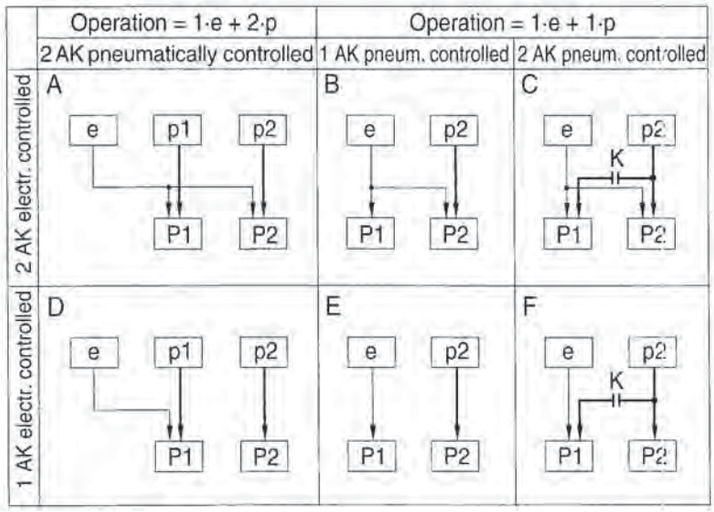
Assignment

The aim of the electronically controlled braking system (ELB) for commercial vehicles is to optimize the braking process and, in certain cases, the power transmission process. The ELB system also supports the driver and the service technician during monitoring and servicing of the braking system, as well as contributing to an increase in economic efficiency. ELB comprises the areas of ABS (Antilock Braking System), EPB (Electropneumatic Brake), and TCS (Traction Control System). Within an electronic vehicle-management system, the ELB utilizes the information available from the data bus network to produce easy-to-control and

Adhesion-slip diagram, scope of ELB
λ Slip, μ Adhesion coefficient.
Indices: A Traction, B Brake.



Structure of the operating and working circuits
e/p Electrical/pneumatic operation, P Pneumatic working circuit (AK),
1, 2 Compressed air from reservoir 1, 2,
K Valve for circuit isolation (decoupling),
A...F variants.



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directionally stable braking and power transmission, and to provide information for other systems. Compared with conventional braking systems, an ELB system can be simpler in design. In the ELB, complex functions are processed electronically.

Function blocks

An ELB encompasses at least the following function groups:
- Electropneumatic braking system (EPB),
- Antilock braking system (ABS)
- Traction control system (TCS).

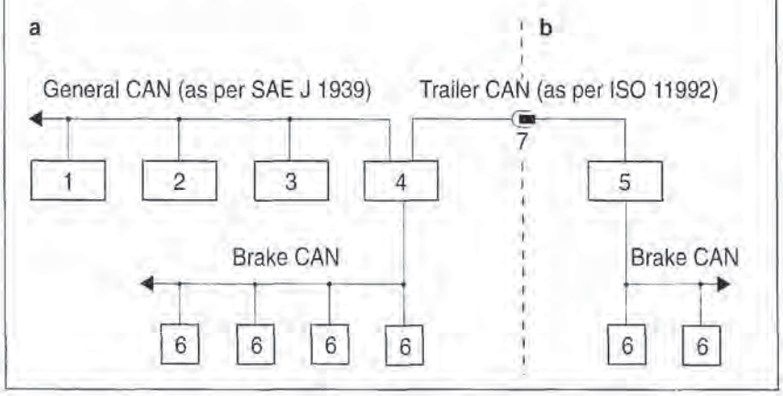
ABS and TCS are already in use on commercial vehicles (see PP. 693 and 610) and in the ELB are up-graded by the availability of additional information such as brake pressures and axle loads. Under stable adhesion/slip range (see Adhesion-Slip Curve) the action of the brakes is controlled electronically by the EPB. In this process, the effect of pressure control on the wheels, axles and the tractor and trailer units over the EPB range as a whole, is designed to comply with a variety of different targets. Thus, for light braking operations, the control-system objective may be even brake-lining wear, for medium to heavy braking it may be precise balancing of braking forces, and for emergency braking the aim may be maximum possible braking effect.

Design

Compressed-air supply, circuit isolation, energy storage and brake cylinders are the same as in conventional braking systems. The additional components include: pressure-control modules in the vicinity of the wheels for adjusting the brake pressure in the wheel cylinders as required, and an ECU. The system may also include a trailer control module the main function of which is to brake a conventional trailer.

Today, ELB systems with pneumatic retention circuits are available in the form of back-up systems at the safety level. Tomorrow's vehicle generations will be able to provide adequate safety while featuring electrically controlled multiple circuits without retention circuits. Normally, the pneumatic working circuits for operation of the brake cylinder in the tractor vehicle are of the dual-circuit design (II braking-force distribution, see P. 654) and in the trailer they are single circuit. The Figure

CAN networking of the brake components and their networking to other systems in the vehicle
a) Tractor vehicle, b) Trailer.
1, 2, 3 CAN stations (e.g. information, chassis, drivetrain),
4, 5 CAN stations (ELB control unit),
6 CAN stations (ELB pressure-control module),
7 Plug connection ISO 7638 (7-pole).



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"Structure of the operating and working circuits" shows combinations of electrical and pneumatic control of two working circuits. For example, electrical operation of all axles is possible with Variant A which also features separate pneumatic retention circuits. Trailers are controlled via a standardized CAN data bus (ISO 11 992) (see P. 884). The pneumatic control circuit is retained in order to cope with tractor/trailer rigs using conventional vehicles and the stipulated control for all braking circuits. However, this function too can be performed by an electronically controlled trailer control module.

Apart from the CAN data bus connecting it to the trailer, an ELB has at least two more: One bus links it to the ECUs for the drivetrain (e.g. EDC and/or retarder ECUs, transmission ECU, etc.) and one internal bus to the system for communicating with the pressure-control modules and the trailer control module (see diagram "CAN Networking"). The ELB is thus able to exchange information with other systems in order to optimize vehicle management functions.

Functional sequence EPB

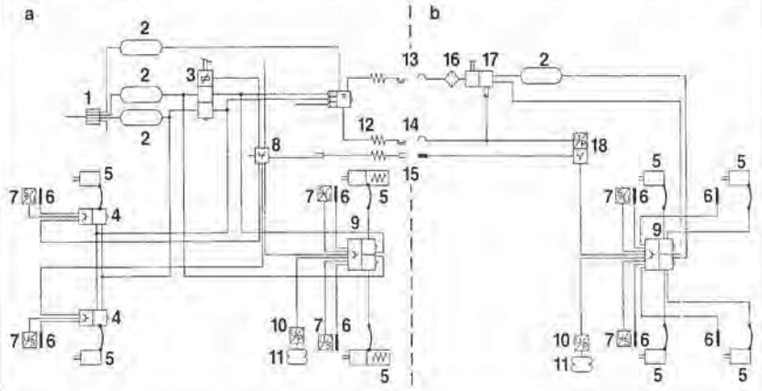
A single-circuit ELB with 2-circuit retention circuit (1e + 2p) is shown in the Fig. "Service-brake installation of an ELB". In addition to a conventional pneumatic trailer control, this ELB also features an electrical control.

After the driving switch is turned, the ELB is initialized and subjected to a self-test. If no faults are detected on the ELB, the warning lamps go out and electrically controlled braking can now take place. Redundant principles are applied in the registration of the brake-pedal position. This is used by the ELB control unit to calculate the optimum pressure level for each control channel. The data is transmitted to the pressure-control modules (located near to the wheels) via the braking-system CAN. At the same time, the brake values are sent to the trailer via CAN. The pressure-control modules (DRM) on the tractor unit and trailer automatically adjust the brake pressure to the required level. Sensor signals from the wheels for variables such as wheel speed, brake lining thickness, etc. are sent by the pressure control modules via the CAN to the ELB control unit. The system's rapid reaction with simultaneous pressure regulation on all axles produces a direct braking feel similar to that in a passenger car.

Service-brake installation of an ELB

a) Tractor vehicle, b) Trailer.

1 Four-circuit protection valve, 2 Air reservoir, 3 Service-brake valve with braking-value sensor (1e+2p), 4 Single-channel pressure-control module (DRM), 5 Wheel-brake cylinder, 6 Wheel-speed sensor, 7 Lining-wear sensor, 8 ELB control unit in the tractor vehicle, 9 Two-channel pressure-control module (2K-DRM), 10 Pressure sensor, 11 Air-suspension bellows, 12 Trailer control valve, 13 Supply coupling head (glad-hand), 14 Brake coupling head (glad-hand), 15 Plug-in connection ISO 7638 (7-pole), 16 Line filter, 17 Trailer brake valve with release device, 18 ELB control unit in the trailer.



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Trailers equipped with ELB which are towed by tractor vehicles having a 5-pole ABS plug-in connection generate their electrical brake value from the pressure in the pneumatic trailer brake line.

When the driving switch is at "Off", or if pressure control is deactivated in the event of a fault, the solenoid valves in the pressure-control modules revert to the mechanically held normal position. In that position, the pneumatic retention circuits from the service brake valve apply the brake pressure to the wheel cylinders unmodified (back up braking).

Extended functions

ELB incorporates EPB, ABS and TCS as standard functions. Other functions for automatic brake balancing and functions which require interaction with other systems are already available fur use on production models or are in the course of development.

A few examples are given below:

Brake management

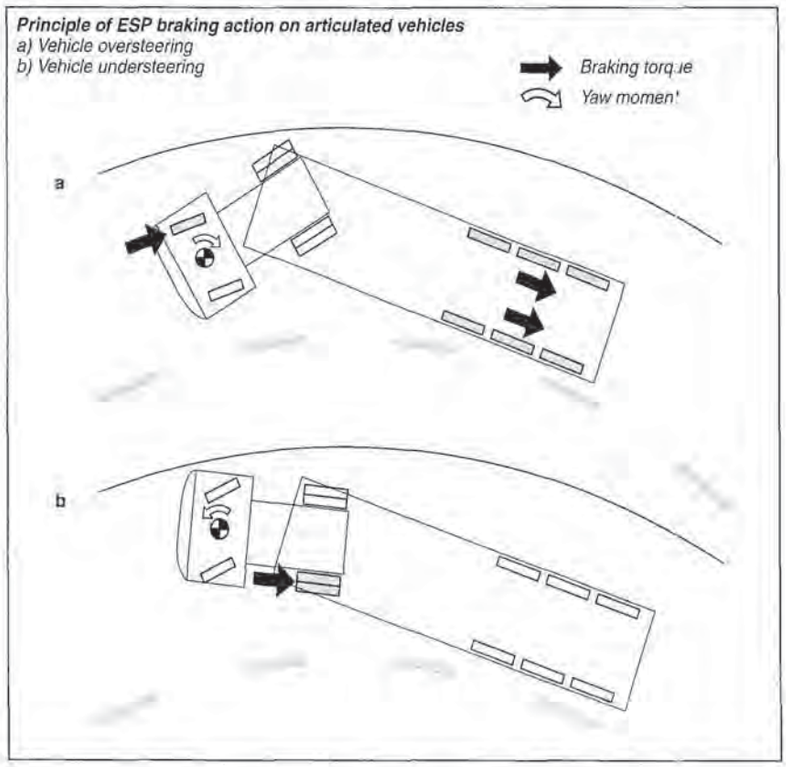
The braking effect desired by the driver can be achieved by means of either the retarder and/or the engine brake and/or the service brakes. The ELB control unit can perform the task of allocating the braking function to the appropriate braking system or systems.

Braking regulation

Vehicle deceleration is performed according to the driver's command, i.e. the position of the footbrake valve, regardless of whether the vehicle is loaded or empty. An automatic brake monitoring system prevents overload.

Brake-lining wear control

The brake-lining thickness is detected by sensors. Under normal braking with low levels of deceleration, the brake-lining



Principle of ESP braking action on articulated vehicles
a) Vehicle oversteering
b) Vehicle understeering

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wear between the wheels on the same axle is equalized by slight adjustment of the brake balance.

Brake balancing between tractor unit and trailer

The aim of automatic balancing of the braking force of the tractor unit and the trailer is to generate optimum trailer-coupling forces between the two units when braking. For articulated vehicles, the optimum trailer-coupling force is equal to the coupling load; for truck-and-trailer combinations it is zero. Each unit is then braked individually according to its own mass. Assuming appropriate braking-system design and choice of brake linings, this then generally leads to equalization of temperatures and therefore of wear between the brake linings on the tractor unit and the trailer.

Brake monitoring

The energy generated in a brake is dependent on the vehicle's speed and the braking torque, while the temperature of the brakes is also dependent on the dissipation of the heat produced by friction. Using those variables, the ELB can estimate the temperature of the brakes and then take action to prevent overloading.

Inappropriate reduction of braking torque on one or more wheels can be detected by ECU analysis of the vehicle behaviour under braking.

Rollback limiter/"Bus-stop brake"

The use of information that is normally available from other electronic systems via the SAE J1939 CAN enables automatic application of the brakes shortly after the vehicle starts to roll with the drive disengaged, e.g. when stopping at traffic lights. In the same way, the brake is automatically released when the vehicle starts to pull away. On public transport busses, this function can replace the so-called "bus-stop brake".

Electronic stability program (ESP)

ESP is an extension of the ABS system and controls not only longitudinal dynamics (like ABS) but also actively responds to the vehicle's transverse dynamics at the limits of adhesion (e.g. in the event of a skid). This requires a yaw-rate sensor and a lateral-acceleration sensor for determining the actual vehicle behaviour, as well as a steering-angle sensor for detecting the driver's intentions. If there are inconsistencies between the driver's intentions and the vehicle's behaviour, e.g. if the tractor unit starts to understeer or oversteer, the ESP modifies the vehicle behaviour by automatic braking of individual wheels to reflect the intentions of the driver. The physical processes involved are explained in the description of ESP for cars (see P. 704). The same basic principles apply to commercial-vehicle ESP systems with the exception of the braking strategy. This is modified to suit commercial vehicles in their various forms. For example, as well as braking of individual wheels on the tractor unit, correction of oversteer can be assisted by braking the trailer to tension the tractor-and-trailer combination (see diagram "Principle of ESP braking action on articulated vehicles"). The trailers do not normally have sensor systems for detecting handling dynamics and are therefore not involved to any greater degree in the control of vehicle directional stability.

It is clear that the risk of overturning is considerably greater with commercial vehicles than with cars. For that reason, the commercial-vehicle ESP also incorporates functions for reducing the likelihood of vehicle overturn. This applies to ELB systems both on the tractor unit and on the trailer. From the available signals, critical vehicle conditions are detected and the vehicle or vehicle combination automatically braked as required by the particular situation. The behaviour of trailers without ELB is estimated by the tractor unit's ECU. It then instigates appropriate braking operations as required to avert hazardous situations.

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Brake test stands

The brakes are of decisive importance in maintaining high technical standards of vehicle safety; it is therefore imperative that the brake system be inspected on a regular basis. The German motor vehicle inspection prescribed by Paragraph 29 of the StVZO Road Licensing Regulations (FMVSS/CUR) employs brake test stands (roller dynamometers) for this purpose, as do automotive service centers for inspections and repairs. The braking forces monitored at the wheel's circumference provide the basis for evaluating the operation and effectiveness of the brake system. Brake stands employed for the inspections stipulated in Paragraph 29 must conform to the "Regulations governing use, design and testing of brake test stands" as administered by the Federal Transport Minister.

Design

The brake test stand's main components are two roller sets, for the left and right sides of the vehicle respectively. The vehicle is driven onto the test stand so that the wheels of the axle being tested rest upon the rollers.

A stable frame supports the roller sets, which assume the form of a drive roller and a secondary roller in a parallel layout, while a drive chain provides the positive dynamic connection between the two rollers. An AC motor powers the drive roller through a gear set with an upward conversion ratio, while the drive unit itself is suspended on an extension of the drive roller's shaft. Pressure exerted against a torque lever flanged onto the gear-drive unit is transferred through the load sensor, with the frame providing positive support for the entire assembly. The braking force F_{Br} is actually measured by monitoring the reaction torque M_R. The electric motors set the rollers in motion and then maintain an almost constant rotational speed against the opposing forces that occur when the vehicle's brakes are applied. The suspended drive unit with torque lever transmits the braking forces to the load-sensing device. The load-sensing device is designed as a flexural sensor with wire strain gauge.

The system's computer employs digital technology in the subsequent evaluation of the various data derived from braking-force testing, such as fluctuations or differences in braking force. Once processed, the information is presented in either analog or digital form, depending upon the specific system, while a printer can be connected to provide a hardcopy test protocol.

Operation

The drive motors for the roller sets may be activated in one of two ways, either by remote control or with an integral automatic on/off switch. An automatic-activation test stand is distinguished by the pressure-sensitive rollers located between the main test rollers on each roller set. When the vehicle is driven onto the test stand, it pushes down the pressure-sensitive rollers and activates the stand. When the vehicle leaves the stand the pressure-sensitive rollers are released and the unit is switched off. Should the applied braking force start to exceed the available traction between tires and test rollers, the wheel will respond by starting to slip and will then lock. Tire slip, however, makes it impossible to perform useful measurements of braking force. Under these conditions, it is the slip resistance between tire and roller (as a function of wheel load) that is measured and this is of no use. Here, assistance is provided by an automatic override device which recognizes this kind of slip by monitoring the test-roller speed, and which responds to it by switching off the test unit when a given maximum figure is exceeded. This avoids both false measurements and possible tire damage. The display, meanwhile, continues to show the maximum braking force achieved before the override device was activated. An electronic memory circuit ensures that the final display remains in place long enough to be recorded by the operator.

In addition, the vehicle and/or axle weight can also be measured at the test stand or entered remotely; the test unit can then use this information to calculate the effective retardation.

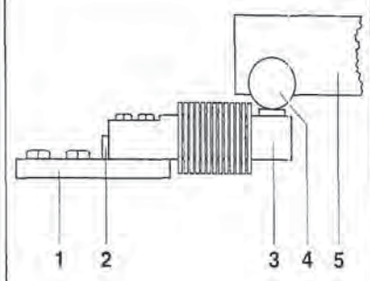
The brake test stand employs specific test sequences to provide extremely rationalized examination procedures, allowing

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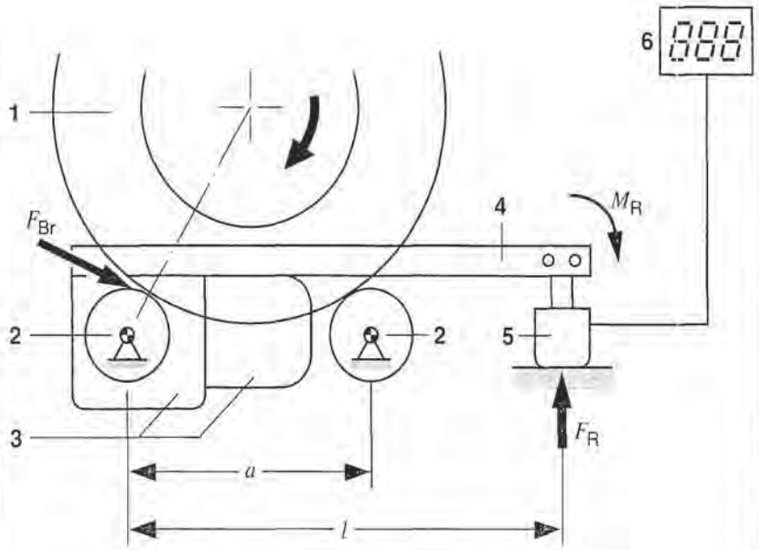
the operator to carry out complete testing of both front and rear brakes without leaving the vehicle.

Vehicles with permanent 4-wheel drive and variable torque distribution are tested on special stands, which are constructed to prevent the forces generated at the test axle from being transferred to that axle which is at rest.

Brake test stand: Measurement sensor
1 Adjustment plate, 2 Alignment pin,
3 Flexural sensor with wire strain gauge,
4 Thrust block, 5 Torque lever.



Determining braking force F_{Br} via measurement of reaction torque M_R
1 Vehicle tire,
2 Roller set with spacing a,
3 Motor with gear set,
4 Torque lever with length l,
5 Measurement sensor,
6 Display.



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Electronic Stability Program (ESP) for passenger cars

Quantities and terms

a	Distance of front wheels from center of vehicle mass
a_x	Estimated longitudinal acceleration of vehicle
a_y	Measured lateral acceleration of vehicle
A_0, A_1, A_2	Parameters for calculation of tire operating point
c	Distance of rear wheels from center of vehicle mass
C_p	Brake-torque ratio
C_λ	Longitudinal tire stiffness
D_T	Sampling time of the controller
D_λ	Tolerance band of drive-slip difference
F_B	Brake force on the tire
F_{BF}	Steady-state (filtered) brake force on the tire
F_N	Normal force on the tire
F_R	Resultant force on the tire
F_S	Lateral force on the tire
J_{Mot}	Moment of inertia of the engine
J_{Whl}	Moment of inertia of the wheel
K_p, K_d, K_i	Controller gains for the P-, D- and I-parts
M_{Dif}	Local brake-torque difference between the driven wheels
M_{DR}	Engine torque requested by driver
M_{YwNo}	Nominal yaw moment on the vehicle
ΔM_{YwNo}	Small change in the nominal yaw moment on the vehicle
M_{CaHalf}	Half of the torque on the cardan axle
M_{Ca}	Torque on the cardan axle
M_{Mot}	Actual engine torque
M_{WhlNo}	Nominal brake torque
M_{NoMot}	Nominal engine torque
M_{NoLock}	Nominal brake-torque difference between the driven wheels
M_{NoSPR}	Nominal engine-torque reduction by spark retard
p_{Circ}	Brake circuit pressure induced by the driver
p_{Whl}	Pressure in wheel brake cylinder
p_{WhlPre}	Nominal pressure in wheel brake cylinder
R	Wheel radius
T_{fOFF}	Time during which the fuel injection is shut off
$\ddot{U}_{Tr}$	Transmission ratio
U_{val}	Actuation mode of solenoid valve
υ_{CH}	Characteristic velocity of the vehicle
υ_{Dif}	Difference between driven-wheel velocities
υ_V	Vehicle velocity
υ_{Ca}	Cardan-axle speed
υ_{Whl}	Measured wheel speed
υ_{Whl3}	Measured wheel speed, rear left
υ_{Whl4}	Measured wheel speed, rear right
υ_{WhlFre}	Wheel speed (free-rolling)
υ_{NoDif}	Nominal difference between wheel velocities
υ_{NoCa}	Nominal velocity of the cardan axle
υ_x	Vehicle speed in the longitudinal direction
υ_y	Vehicle speed in lateral direction
x_1, x_2	Parameters of the inverse hydraulic model
α	Slip angle of the tire
α_0, λ_0	Arbitrary operating point of the tire
β	Slip angle of the vehicle
β_{No}	Nominal slip angle of the vehicle
δ	Steering-wheel angle
δ_W	Front-wheel steering angle
λ	Tire slip
λ_{No}	Nominal tire slip
λ_{Ma}	Nominal average drive slip of the driven wheels
μ_{HF}	Coefficient of friction between tires and road
μ_{Res}	Resultant coefficient of friction between tire and road
$\dot{\psi}$	Yaw rate
$\dot{\psi}_{No}$	Nominal yaw rate
m	Minimum value of the nominal driven-wheel brake torque
MIN	Minimum-value operator
SUM	Event-driven integration
ZWV	Spark retard (SPR)

Assignment

The Electronic Stability Program ESP (vehicle dynamics control) system is a closed-loop control system which prevents lateral instability of the vehicle. It is integrated within the vehicle's brake system and dri-

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vetrain. While ABS prevents wheel lockup when braking and TCS prevents spin of the driven wheels, ESP prevents the vehicle from "pushing out" of the turn or spinning out of the turn when it is steered.

Further to the advantages inherent in ABS and TCS, ESP improves the active driving safety in the following points:

- Provides the driver with active support, even in laterally critical dynamic situations.
- Enhances vehicle stability and tracking performance even in limit situations in all operating modes such as full braking, partial braking, coasting, accelerating, engine drag, and load shift.
- Enhances directional stability even during extreme steering maneuvers (panic reactions), resulting in a drastic reduction in the danger of skidding.
- Improved handling behavior also in limit situations. For the driver, this behavior becomes predictable as a function of his (or her) experience. The vehicle remains fully under control even in critical traffic situations.
- Depending upon the situation, even better utilization of the friction potential between the tires and the road when ABS and TCS intervene, and therefore improved traction and stopping distances in addition to improved steerability and stability.

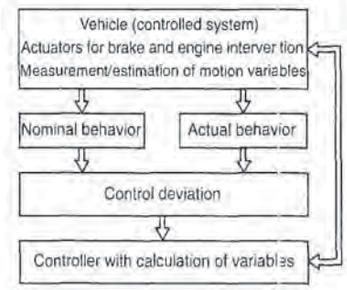
Vehicle handling

The vehicle's steering properties are defined in the description of the vehicle's lateral dynamics (P. 350). This description is derived from steady-state cornering and shows the dependence of the tire slip angle α on the lateral vehicle acceleration a_y and therefore on the lateral tire forces. Furthermore, the description of ABS (P. 659) and TCS (P. 606) also underlines the dependence of the lateral tire forces on the tire slip. This means that the vehicle's steering properties can be influenced by means of the tire slip. ESP utilizes this tire characteristic in order to effect servo-control for vehicle handling.

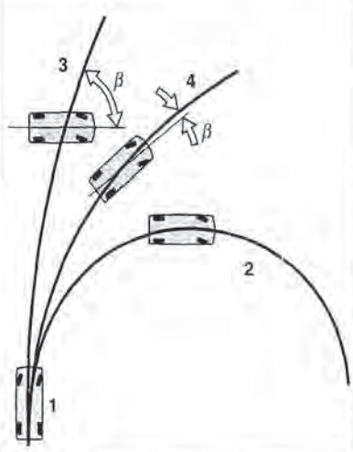
It is important for good vehicle handling that the vehicle remains safely on a track which corresponds as far as possible to the course of the steering-angle (see Fig. "Lateral dynamics of a vehicle", Curve 2). During steering maneuvers, this is the case if the lateral tire forces remain considerably below the friction potential between the tire and the road surface. The yaw-rate curve then corresponds to that of the steering angle.

For ESP though, the mere control of the yaw rate according to the steering angle proves to be inadequate and the vehicle

Block diagram of ESP controller



Lateral dynamics of a vehicle
1 Step input at steering wheel, steering-wheel angle fixed, 2 Track on high-μ road, 3 Track on low-μ road with "open-loop" steering correction and yaw-rate cor trol, 4 Track on low-μ road when the slip angle is also controlled (ESP).



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can become instable (see Fig., Curve 3). The ESP therefore controls not only the yaw rate but also the vehicle's sideslip angle.

The ESP vehicle-handling control is not restricted to ABS and TCS/MSR operations, but also extends to cover the area in which the vehicle is rolling freely, and also comes into action during partial braking if the vehicle is moving at the physical driving limits.

ESP control systems

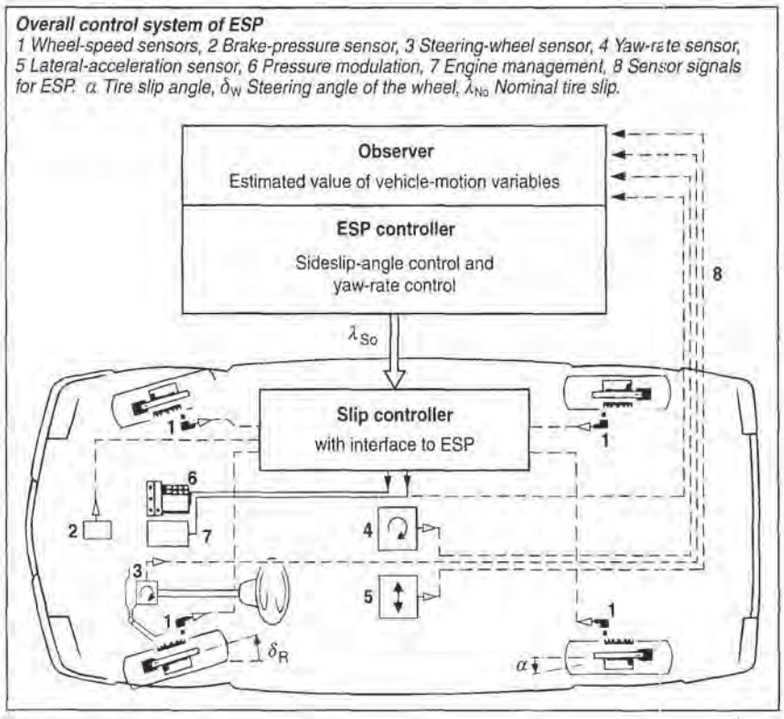
Control

The control of the vehicle's behavior at the physical driving limit must influence the vehicle's three degrees of freedom in the plane of the road (longitudinal and lateral velocities, and yaw moment about the vertical axis) so that vehicle handling is in line with the driver's input and with the prevailing road conditions. In this respect, as shown in the block diagram, it must first of all be defined how the vehicle is to behave at the physical driving limit in accordance with the driver's input (nominal behavior), and how it actually behaves (actual behavior). In order to minimize the difference between nominal and actual behavior (deviation), the tire forces must in some way be controlled by actuators.

The overall system (Fig. "Overall control system of ESP" below), shows the vehicle as the controlled system, with sensors (1...5) for defining the controller input variables, and actuators (6 and 7) for influencing the tractive force and the braking force. Also shown are the hierarchically structured controller comprising the higher-level vehicle dynamics controller, and the lower-level slip controllers. The higher-level controllers input the nominal values to the lower-level controllers in the form of a nominal slip. The "observer" determines the controlled state variable (vehicle slip angle β).

In order to determine the desired performance, the signals are eva uated which

Overall control system of ESP
1 Wheel-speed sensors, 2 Brake-pressure sensor, 3 Steering-wheel sensor, 4 Yaw-rate sensor, 5 Lateral-acceleration sensor, 6 Pressure modulation, 7 Engine management, 8 Sensor signals for ESP. α Tire slip angle, δ_W Steering angle of the wheel, λ_{No} Nominal tire slip.



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define the driver's input. These comprise the signals from the steering-wheel sensor (3, driver's steering input), the brake-pressure sensor (2, desired deceleration input), and the engine management (7, desired drive torque). Apart from the vehicle speed, the calculation of the desired performance also takes the coefficients of friction between the tires and the road into account. These are calculated from the signals outputted by the wheel-speed sensors (1), the lateral-acceleration sensor (5), the yaw-rate sensor (4), and the brake-pressure sensor (2). Depending upon the control deviation, the yaw moment which is necessary to make the actual-state variables approach the desired-state variables is then calculated.

In order to generate the required yaw moment, it is necessary for the changes in desired slip at the wheels to be determined by the vehicle dynamics controller. These are then set by means of the lower brake-slip controller and traction controller together with the "brake-hydraulics" actuator (6), and the "engine-management actuator" (7).

The system relies on tried and proven ABS and TCS components. The TCS hydraulic modulator (6), which is described elsewhere, permits high levels of dynamic braking of all wheels throughout the complete temperature range encountered.

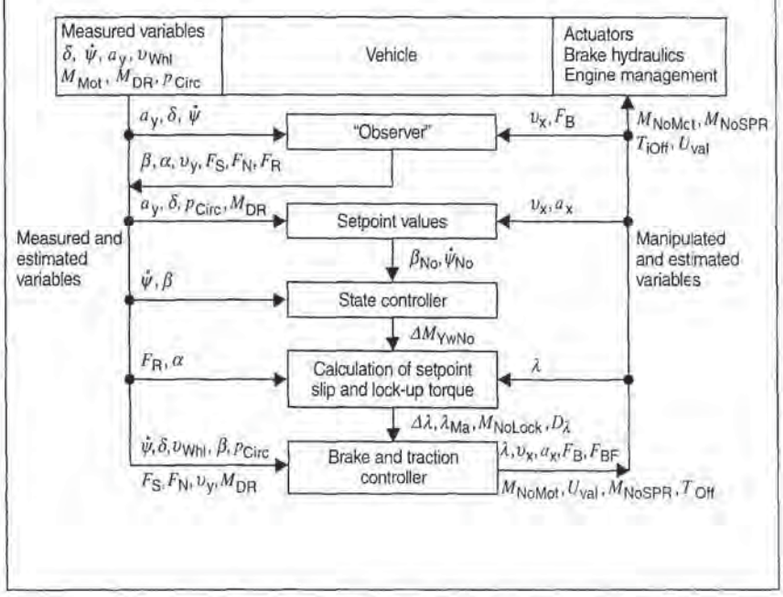
The necessary engine torque can be set by means of the engine management (7) and the CAN interface, so that the traction-slip values at the wheels can be adjusted accordingly.

System components

Vehicle dynamics controller

The relationship between steering angle, vehicle speed, and yaw rate during steady-state driving on a circular course (P. 355), is taken as the basis for the desired vehicle motion during steady-state driving, and when braking and/or accelerating. By applying the "Single-track (bicycle) model": the nominal yaw rate is cal-

Block diagram of ESP vehicle dynamics controller with input and output variables



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culated from the vehicle speed and the steering angle:

$$\dot{\psi}_{No} = \frac{v_x \cdot \delta_W}{(a+c)\left(1+\frac{v_x^2}{v_{CH}^2}\right)}$$

For the sideslip-angle control, this value is first of all limited according to the road-surface coefficient of friction:

$$|\dot{\psi}_{No}| \leq \mu_{HF} \cdot g / v_V$$

whereby g is the acceleration due to gravity, and the coefficient of friction μ_{HF} and the vehicle speed v_V are estimated values.

No attempt is made to directly influence the lateral speed, and with it the sideslip angle, by means of changes in the lateral forces. It is better to adjust the lateral motion indirectly by means of changes in the slip angle, whereby the vehicle's rotation is initiated by the generation of yaw moments.

A simplified block diagram is used to demonstrate the structure of the vehicle dynamics controller. Using the measured variables (nominal yaw rate $\dot{\psi}_{No}$, steering-wheel angle δ, and the vehicle lateral acceleration a_y), together with the estimated variables (vehicle longitudinal speed v_x, and braking forces F_B), a so-called "Observer", using a simple vehicle model, determines the slip angle β, the sideslip angle α, the vehicle lateral speed v_y, the wheel forces in the lateral and vertical directions F_S and F_N, and the resulting wheel forces F_R in the plane of the road.

The nominal sideslip-angle β_{No}, and the nominal yaw rate $\dot{\psi}_{No}$, are determined from the following driver inputs: steering-wheel angle δ, engine-torque input M_{DR} (accelerator-pedal position), and the brake pressure p_{Circ}, together with the estimated vehicle longitudinal speed v_x, and the coefficient of friction μ_{HF} which is determined using the estimated longitudinal acceleration a_x and the measured lateral acceleration. In the process, the vehicle's dynamic response, and special situations such as inclinations, banks, and/or μ-split road surfaces (different coefficients of friction between tires and road on the left and right side of the vehicle) are taken into account.

The vehicle dynamics controller is designed as a state controller. The controlled-state variables are the sideslip angle and the yaw rate. The more the sideslip angle increases, the more it is taken into account by the vehicle dynamics controller. The state controller's output variable corresponds to a yaw moment M_{YwNo}.

Together with the actual slip values λ, the resultant wheel forces F_R, and the slip angles α, the linearized vehicle model is now also applied to convert this yaw moment into changes of the nominal slip at the appropriate wheels. For instance, presuming that the vehicle is rolling freely and oversteers in a right-hand bend so that the nominal yaw-rate is exceeded, among other things a nominal brake slip is requested at the front left wheel. This subjects the vehicle to a counterclockwise change in yaw moment which reduces the excessive yaw rate. The nominal slip is changed by the lower-level ABS or TCS wheel controller. If the brakes have not been applied, or the driver's brake-pressure input does not suffice to set the required nominal slip (partial-braking range), the pressure in the brake circuits is actively increased.

When TCS is operative, in order to set the vehicle's nominal yaw moment M_{YwNo}, the vehicle dynamics controller outputs a mean absolute drive slip $\lambda_{Mε}$, a drive-slip tolerance band D_λ, and a nominal brake locking torque M_{NoLock}. In order to achieve distinct improvements for the ABS and TCS basic functions with regard to their utilization of the adhesion potential in line with the driving situation, all available measurement quantities and estimated values are also applied to the full in the slip controllers.

Lower-level brake-slip controller (ABS) and engine drag-torque controller (MSR)

In order to be able to control the wheel slip to a definite level, it must be possible to evaluate it with an adequate degree of accuracy. The vehicle's longitudinal speed is not measured, but is determined from the wheel speeds v_{Whl}. To do so, individual wheels are "under-braked" during ABS application. In other words, slip control is interrupted, and the instantaneous wheel

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brake torque is reduced by a defined amount and held constant for a given period of time. Assuming that the wheel speed is stable toward the end of this period, the zero-slip (free-rolling) wheel speed v_{WhlFre} can be calculated from the momentary braking force F_B and the tire stiffness C_λ.

$$v_{WhlFre} = v_{Whl} \cdot \frac{C_\lambda}{C_\lambda - \frac{F_B}{F_N}}$$

Using the yaw rate $\dot{\psi}$, the steering angle δ_W, and the lateral velocity v_y, together with the vehicle geometry, the (free-rolling) wheel speed is transformed into the center of gravity in order to generate the center-of-gravity velocity v_x in the vehicle's longitudinal direction. Finally, v_x is transformed back to the four wheel positions in order to evaluate the free-rolling speed of all four wheels. This enables the actual slip λ to be calculated for the control of the three remaining wheels.

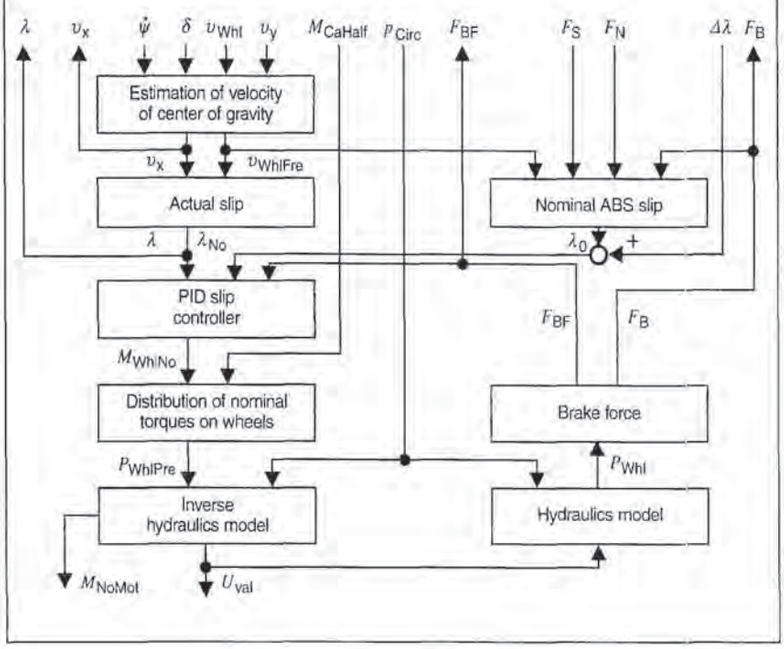
$$\lambda = 1 - \frac{v_{Whl}}{v_{WhlFre}}$$

Starting from the stationary brake force F_{BF}, using a PID control law the nominal torque at the wheel can be calculated as a function of the slip deviation.

$$M_{WhlNo} = F_{BF} \cdot R + K_p(\lambda_{No} - \lambda)R + K_d\left(\frac{d}{dt}v_{Whl} - \frac{d}{dt}v_{WhlFre}\right)\frac{J_{Whl}}{R} + K_i \cdot C_p \cdot \text{SUM}\{(\lambda_{No} - \lambda) \cdot D_T\}$$

In the case of the driven wheels, in order to implement an engine drag torque control (MSR) the nominal braking torque M_{WhlNo} can be partially set by the engine (or completely when the brakes are not applied). The driven wheel with the lower nominal wheel torque is controlled within the permissible limits by engine intervention.

Block diagram of the brake-slip controller for ABS and MSR with input and output quantities



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The following applies for rear-wheel drive:

$$M_{NoMot} = -\frac{2m}{\ddot{U}_{Tr}} + \frac{J_{Mot} \cdot \ddot{U}_{Tr}}{R} \cdot \frac{d}{dt} v_x$$

$m = MIN (M_{Whl3}, M_{Whl4})$

For negative values, the nominal engine torque M_{NoMot} is limited by the maximum engine drag torque, and in the driven case (positive values) it is limited to the maximum dynamic drive torque as permitted by the manufacturer. In case of a positive nominal brake torque M_{WhlNo}, the remaining braking torque must be adjusted through the brake pressure.

$$p_{WhlPre} = \frac{M_{WhlNo} + M_{CaHalf}}{C_p}$$

The pressure p_{WhlPre} applied to the wheel-brake cylinders by the controller, is adjusted by the braking hydraulics and relevant valve-triggering mode U_{val}. The required valve time is calculated using an inverse "hydraulic model", whose parameters x_1, x_2 have been defined beforehand and stored in the controller. Essentially, the model uses the Bernoulli theorem for incompressible media, and a pressure/volume characteristic.

$$U_{val} = \frac{p_{WhlPre} - p_{Whl}}{(x_1 + x_2 \cdot p_{Whl}) \sqrt{|p_{Circ} - p_{Whl}|}}$$

$U_{val} > 0$ Build up pressure,
$U_{val} = 0$ Hold pressure,
$U_{val} < 0$ Reduce pressure.

Since the valve-control mode U_{val} is limited by the sampling time and quantified, the adjusted pressure p_{Whl} that has actually been set must be calculated using the hydraulic model. By means of the dynamic wheel equations, taking the known (estimated) wheel brake pressure and the measured wheel speeds, the actual tire braking force F_B and the steady-state braking force F_{BF} can now be calculated:

$$F_B = C_p \cdot \frac{p_{Whl}}{R} - \frac{M_{CaHalf}}{R} + \frac{J_{Whl}}{R^2} \cdot \frac{d}{dt} v_{Whl}$$

$$F_B = T_1 \cdot \frac{d}{dt} F_{BF} + F_{BF}$$

The steady-state (filtered) braking force F_{BF} now serves as the reference variable for the PID controller. Using the calculated operating point λ_0, and the change in slip stipulated by the vehicle dynamics controller, the ABS controller calculates the nominal tire slip λ_{No}.

$$\lambda_0 = A_0 \cdot \mu_{Res} + \frac{A_1}{v_{WhlFre}} + A_2$$

$$\mu_{Res} = \frac{\sqrt{F_B + F_S}}{F_N}$$

Traction controller (TCS)

On rear-drive vehicles, the traction controller is used for slip control of the driven wheels only. Active intervention at the front wheels is through the brake-slip controller. Unlike ABS, the traction controller receives from the vehicle dynamics controller the nominal mean traction-slip value λ_{Ma} of both driven wheels and a nominal brake-locking torque value M_{NoLock} to be used as reference variables for directly influencing the yaw moment. The nominal value for the differential speed of the two driven wheels v_{NoDif} is the difference in their (free-rolling) speeds, whereby the vehicle dynamics controller also stipulates a tolerance band D_λ for the difference between the two drive-slip values. This difference represents a dead zone for the control deviation in order that a nominal brake-locking torque M_{NoLock} can be generated.

The TCS module calculates the nominal braking torque M_{WhlNo} for both driven wheels, the nominal engine torque M_{NoMot} to be set by throttle-valve intervention, the nominal value M_{NoSPR} for engine-torque reduction using ignition timing (SPR = spark retard) and, as options, the length of time T_{iOFF} during which the EFI is to be cut off and the number of cylinders this cutoff is to be applied to.

The nominal values for the crankshaft and wheel differential speeds (v_{NoCa} and v_{NoDif}) are calculated from nominal slip values together with the (free-rolling) wheel speeds v_{WhlFre}. The controlled variables v_{Ca} and v_{Dif} are calculated from wheel speeds v_{Whl3} and v_{Whl4}:

$$v_{Ca} = \frac{1}{2}(v_{Whl3} + v_{Whl4})$$

$$v_{Dif} = v_{Whl3} - v_{Whl4}$$

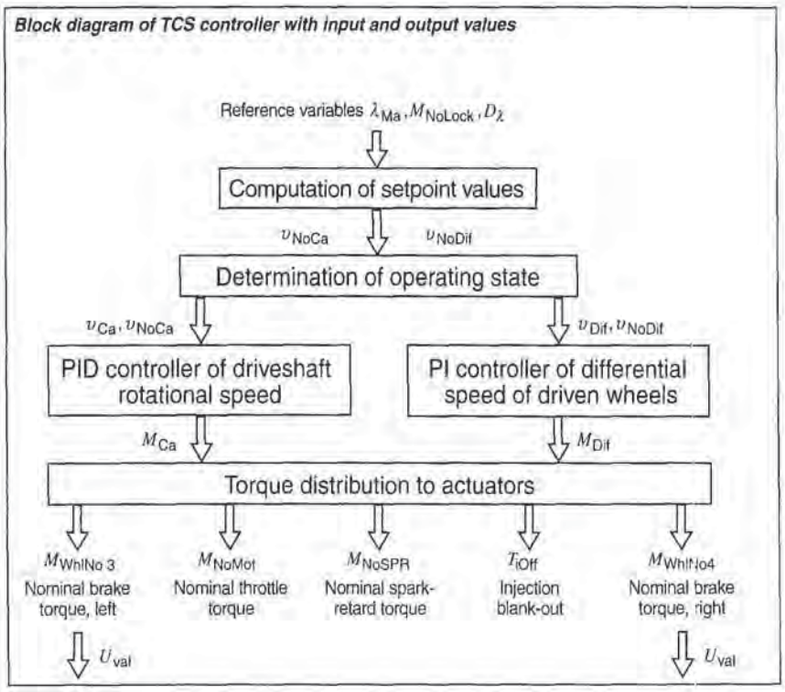
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The dynamic response depends on the highly differing operating modes of the controlled system. It is therefore necessary to determine the operating mode in order to be able to adapt the controller parameters to the controlled system's dynamic response and to the non-linearities. The large moment of inertia of the complete drivetrain (engine, gearbox, cardan axle, and driven wheels) has an influence on the cardan-axle speed υ_{Ca} which is therefore characterized by a relatively large time constant (low dynamic response). On the other hand, the time constant of the wheel differential speed υ_{Dif} is relatively small since the dynamic response of υ_{Dif} is determined almost completely by the small moments of inertia of the two wheels. Apart from this, υ_{Dif} in contrast to υ_{Ca} is not directly influenced by the engine. υ_{Dif} and υ_{Ca} are used therefore as controlled variables because they permit the separation of the drivetrain system (measured wheel speed, rear left and rear right, υ_{Whl3} and υ_{Whl4} respectively) into two subsystems with distinct dynamic response and distinct engine influence. The engine intervention and the "symmetrical" part of the brake intervention represent the controlled variables of the cardan-axle speed controller υ_{Ca}. The "asymmetrical" portion of the brake intervention is the actuating signal for the differential wheel-speed controller υ_{Dif}.

The cardan-axle speed is controlled by means of a non-linear PID controller, whereby in particular the gain of the I-component (dependent upon operating mode) can vary over a wide range. At steady state, the I-component is a measure for the torque which can be transferred to the road surface. The cardan-axle torque M_{Ca} is the controller output.

A non-linear PI controller is used for controlling the wheel differential speed υ_{Dif}, the controller parameters being dependent upon the engaged gear and upon influences from the engine. From the tolerance band D_λ for the differential slip of the driven wheels, which is output by the vehicle dynamics controller, a dead zone is calculated for the control deviation from υ_{Dif}. For

Block diagram of TCS controller with input and output values



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"μ-split", in order to ensure adequate traction, the vehicle dynamics controller specifies a relatively narrow dead zone. In doing so it increases the sensitivity of the controller for the wheel differential speed v_{Dif}. If the nominal brake-locking torque M_{NoLock} is to be reduced, or in case of the optional Select-Low control, the vehicle dynamics controller defines a wider tolerance band. In these cases, the controller for the differential wheel speed v_{Dif} permits larger differences between the rotational speeds of the rear wheels. The controller output is the differential nominal torque M_{Dif}.

The cardan-axle torque M_{Ca} and the nominal torque differential M_{Dif} are distributed between the actuators. A corresponding valve-triggering mode U_{val} in the hydraulic modulator adjusts the differential nominal torque M_{Dif} via the braking-torque difference between the left and right driven wheels. The nominal cardan-axle torque M_{Ca} is achieved by engine intervention and by means of symmetrical brake intervention. There is a relatively long delay (dead time and transient response of the engine) before throttle-valve intervention comes into effect. For rapid-acting engine intervention, if optionally available, the ignition is retarded, and the fuel injection is cut off. The symmetrical brake intervention provides short-term support for the reduction of engine torque.

By means of this module, the traction controller can be relatively easily adapted to the various engine-intervention types.

System realization

The hydraulic modulator and the wheel-speed sensors are suitable for operation in underhood conditions. The yaw-rate sensor, the lateral-acceleration sensor, the steering-wheel-angle sensor, and the ECU are designed for installation in the passenger compartment or in the trunk. An example of the in-vehicle installation of the components, together with their electrical and mechanical connections, is given on the opposite page.

Sensors

Interface monitoring is a necessity, and can be implemented efficiently and at low cost in modern-day ECUs. It has a decisive effect upon the architecture of the sensor interfaces (for sensors, see P. 94).

The demands reflected in the sensor requirements were ascertained by evaluating simulation studies together with an extensive program of road tests. The consequences of side-effects on ESP operation were also investigated (influence of positioning tolerances, cross-couplings, and other sensor malfunctions). The result was a multi-level, and thus very dense monitoring network which incorporates the aspects of analytical redundancy. In order to be able to master with the necessary reliability those complex systems which are relevant from the safety viewpoint, the aspects of analytical redundancy must be incorporated if reasons of cost forbid the application of redundant sensors.

ECU

The ECU is constructed in conventional (4-layer) pcb technology, and in addition to two partially redundant computers, it incorporates all drivers for valve and lamp triggering, as well as the semiconductor relays for valve and pump triggering, the interface circuits for sensor-signal conditioning, and the appropriate switch inputs for auxiliary signals such as those from the brake-light switch. Furthermore, a CAN interface is integrated in order to facilitate communication with other systems in the vehicle (e.g. engine management and transmission control). The large number of additional signal inputs requires the use of a special plug and socket connection in order that the size of the ECU housing can be kept to a minimum.

Monitoring system

A comprehensive safety-monitoring system is of fundamental importance for the ESP's reliable functioning. The system used encompasses the complete system together with all components and all their functional interactions. The safety system is based on safety methods such as e.g. FMEA, ETA and error-simulation studies.

From these, measures are derived for avoiding errors which could have safety-related consequences. Extensive monitoring programs guarantee the reliable detec-

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tion of all sensor errors which cannot be prevented completely. These programs are based upon the well-proven safety software from the ABS and ABS/TCS systems which monitor all the components connected to the ECU together with their electrical connections, signals, and functions. The safety software was further improved by fully utilizing the possibilities offered by the additional sensors, and by adapting them to the special ESP components and functions.

The sensors are monitored at a number of stages:

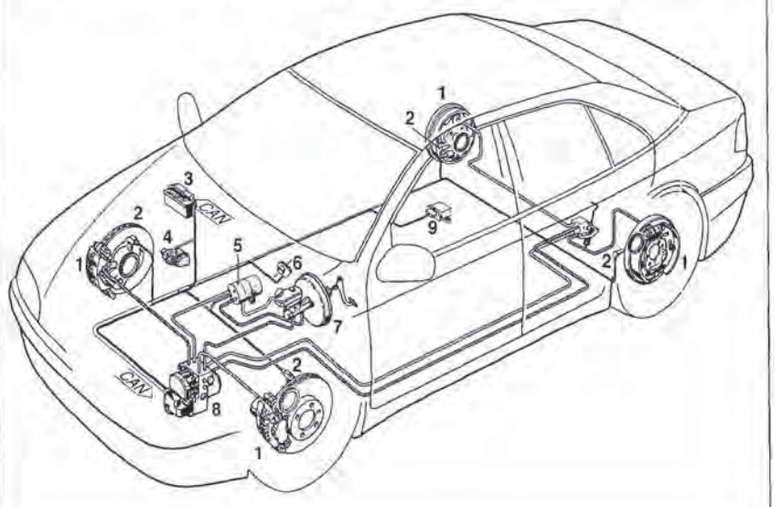
In the first stage, the sensors are continuously monitored for cable open-circuit, and signal implausibility (out-of-range check, detection of interference, physical plausibility).

In a second stage, the most important sensors are tested individually. The yaw-rate sensor is tested by intentionally detuning the sensor element and then evaluating the signal response. Even the acceleration sensor has internal background monitoring. When activated, the pressure-sensor signal must show a predefined characteristic, and the offset and amplification are compensated for internally. The steering-wheel angle sensor is provided with "local intelligence" and has its own monitoring functions which directly deliver any error message to the ECU. In addition, the digital signal transmission to the ECU is permanently monitored.

In a third stage, analytical redundancy is applied to monitor the sensors during the steady-state operation of the vehicle. Here, a vehicle model is used to check that the relationships between the sensor signals, as determined by the vehicle motion, are not violated. These models are also frequently applied to calculate and compensate for the sensor offsets as long as they stay within the sensor specifications.

In case of error, the system is switched off either partially or completely depending upon the type of error concerned. The system's response to errors also depends upon whether the control is activated or not.

The ESP system with its electrical connections in the vehicle
1 Wheel brakes, 2 Wheel-speed sensors, 3 Engine-management ECU with CAN interface, 4 Throttle-valve actuator, 5 Precharging pump with admission-pressure sensor, 6 Steering-wheel angle sensor, 7 Brake booster with master cylinder, 8 Hydraulic system with brake-pressure sensor and attached ECU, 9 Yaw sensor with integrated lateral-acceleration sensor.



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Adaptive Cruise Control (ACC)

Operation

The basic function of Adaptive Cruise Control relies on the conventional cruise-control system (vehicle-speed controller), which maintains a desired speed specified by the driver. In addition, ACC can adapt vehicle speed to changing traffic conditions by means of automatic acceleration, deceleration or braking. This system thus maintains the vehicle's distance to the vehicle driving in front as a function of road speed.

Distance sensor

The most important component in an ACC system is a sensor which measures the distance, the relative speed and the relative position of the preceding vehicles. Maximum performance is achieved – even in poor weather conditions – with a radar sensor.

The radar sensor (P. 101) operates at a frequency of 76 ... 77 GHz which was specially allocated for ACC. Three beams are emitted simultaneously for measurement purposes. The beams reflected by the preceding vehicles are analyzed regarding their propagation time, Doppler shift and amplitude ratio, and from these factors the distance, relative speed, and relative position are calculated (for details, refer to PP. 101 ... 103).

Course setting

To ensure reliable ACC operation, no matter what the situation – e.g. also on curves/bends – it is essential that the preceding vehicles are allocated to the correct lane(s). For this purpose, the information from the ESP sensor system (yaw rate, steering-wheel angle, wheel speeds and lateral acceleration) is evaluated with regard to the ACC-equipped vehicle's actual curve status. Further information on the traffic flow is obtained from the radar signals.

Video imaging and navigation systems are also being considered for future assistance in defining the courses taken by the vehicles.

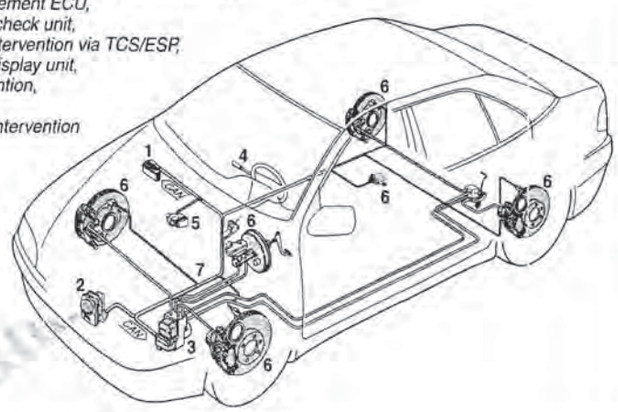
Engine intervention

Speed control requires an electronic engine-power control system (ETC or EDC). Such a system enables the vehicle to be accelerated to the desired speed or, if an obstacle appears, to be decelerated by means of automatic throttle closing.

Active brake intervention

Experience has shown that deceleration by means of throttle closing is not sufficient for ACC operation. Only the inclusion of brake intervention makes it possible for longer follow-on control operations

Bosch Adaptive Cruise Control *(components)*
1 Engine-management ECU,
2 Radar-sensor check unit,
3 Active brake intervention via TCS/ESP,
4 Controls and display unit,
5 Engine intervention,
6 Sensors,
7 Transmission intervention (optional).



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with ACC without the need for frequent driver interventions. ESP provides the possibility of braking without driver intervention.

ACC permits only "soft" brake interventions. Emergency braking due to the sudden appearance of obstacles (e.g. the slow-moving vehicle in front suddenly changing its lane) is therefore not possible.

Setting options
The driver inputs the desired speed and the desired time gap; values of 1 ... 2 s are offered for the latter.

Display elements
The driver must be provided with at least the following information:
- indication of the desired speed,
- indication of the switch-on status,
- indication of the desired time gap selected by the driver,
- indication of the follow-on mode, which informs the driver as to whether the system is controlling the distance to a detected target object or not.

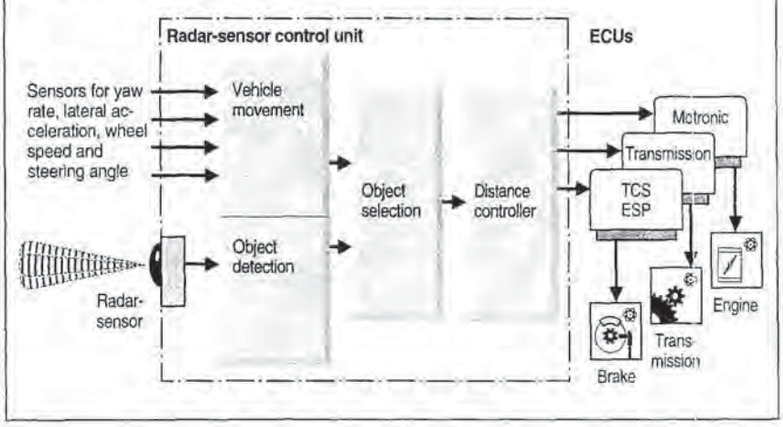
Aim of ACC
The aim of ACC is to relieve the driver of the stress associated with "mindless" driving tasks such as maintaining speeds and driving behind other vehicles in heavy traffic. This system helps to improve road safety as well as driver comfort.

System limits
Even with this form of driver support, the driver remains fully responsible for controlling the vehicle. He/she continues to remain responsible for complex decisions relating to straight-ahead driving and control of the vehicle – and obviously for steering as well. The alignment of the functions to emphasize the comfort aspect for the driver, means that a clear boundary has been defined between the tasks which are the responsibility of ACC and those which are the responsibility of the driver. Safety functions such as emergency braking and forced spacing are thus not featured in this system. These functions, together with selecting the desired speed and the desired time gap, are the sole responsibility of the driver.

ACC does not yet permit control operations in urban environments. This system can only be activated at speeds in excess of 30 km/h.

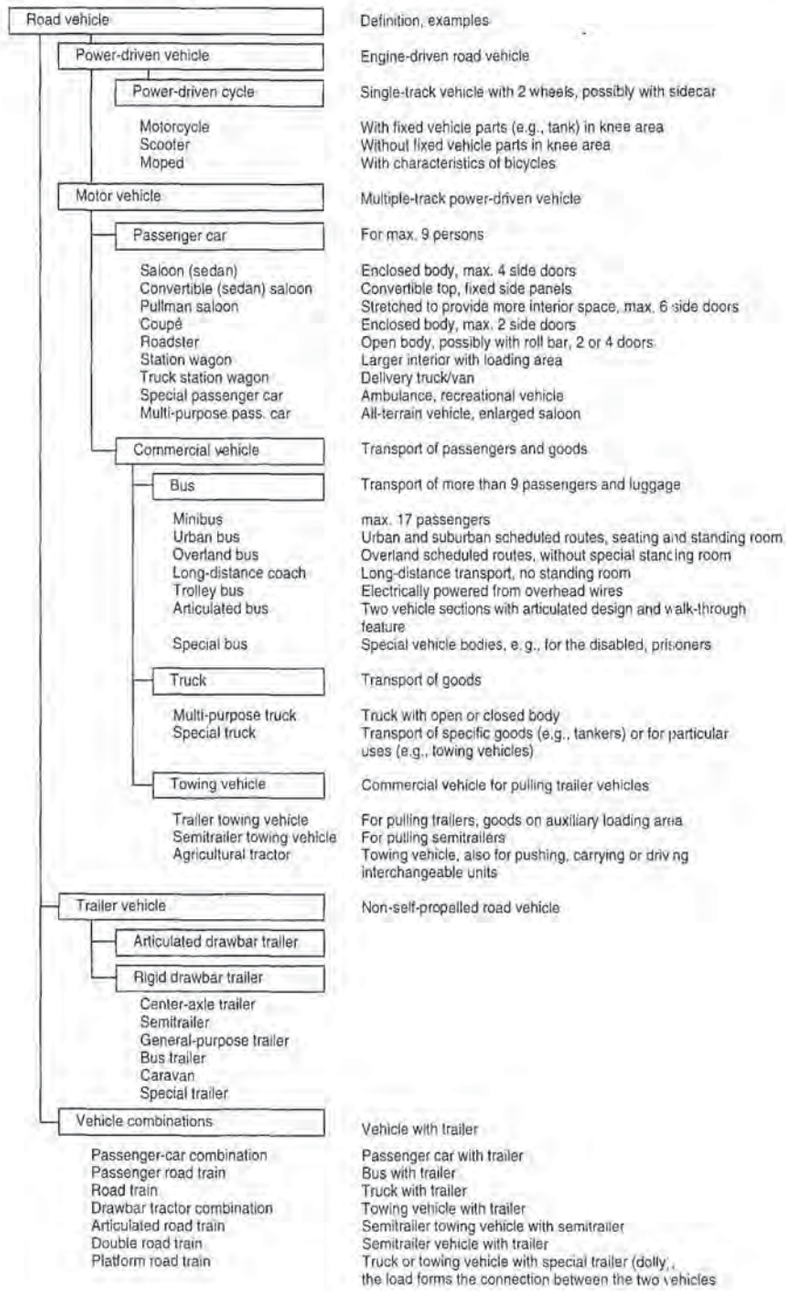
Expanding functions for operation in urban areas would require considerably higher performance on the part of the sensors responsible for monitoring the environment. Such performance cannot be achieved by the 76.5 GHz radar alone.

Basic structure of ACC control



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Road-vehicle systematics



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Classification[1])

Category L
Motor vehicles with fewer than 4 wheels, motorized two-wheeled and three-wheeled vehicles.

Category	Design	Piston displacement	Top speed
L_1	two-wheeled	≤ 50 cm³	≤ 50 km/h
L_2	three-wheeled	≤ 50 cm³	≤ 50 km/h
L_3	two-wheeled	> 50 cm³	> 50 km/h
L_4	three-wheeled asymmetrical to longitudinal vehicle axis	> 50 cm³	> 50 km/h
L_5	three-wheeled symmetrical to longitudinal vehicle axis	> 50 cm³ ≤ 1 t Total weight	> 50 km/h

Category M
Passenger vehicles with at least 4 wheels or with 3 wheels and an overall weight > 1 t.

Category	Driver's seat + passenger seats	Total weight
M_1	1 ≤ 9	
M_2	> 9	< 5 t
M_3	> 9	> 5t

Category N
Goods transport vehicles with at least 4 wheels or with 3 wheels and a laden weight > 1 t.

Category	Total weight
N_1	≤ 3.5 t
N_2	> 3.5 t ≤ 12 t
N_3	> 12 t

Category O
Trailers and semitrailers

Category	Total weight
O_1 only single-axle trailers	≤ 0,75 t
O_2	> 0.75 t ≤ 3.5 t
O_3	> 3.5 t ≤ 10 t
O_4	> 10 t

[1]) Vehicle classification in accordance with the Directives of the European Community (71/320/EEC) and ECE Regulation No. 13 with regard to brake equipment.

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Vehicle bodies, passenger cars

Main dimensions

Interior-dimensions

Dimensional layout depends upon body shape, type of drive, scope of aggregate equipment, desired interior size, luggage compartment volume and other considerations such as driving comfort, driving safety and operating safety. Seating positions are designed in accordance with ergonomic findings and with the aid of templates or 3D CAD dummy models (DIN, SAE, RAMSIS): Body template as per DIN 33 408 for men (5th, 50th, and 95th percentile) and women (1st, 5th and 95th percentile). For example, the 5th percentile template represents "small" body size, i.e., only 5 % of the population have smaller bodies, and 95 % have larger body dimensions.

SAE H-point template in accordance with SEA J826 (May 1987): 10th, 50th and 95th percentile thigh segments and lower leg segments. For legal reasons, motor-vehicle manufacturers in the USA and in Canada must use the SAE H-point template for determining the seating reference point. Body templates in accordance with DIN 33408 in accelerated form are particularly well suited to the dimensional design of seats and passenger compartments. Most motor-vehicle manufacturers round the world use the 3D CAD dummy model "RAMSIS" in development (RAMSIS: computer-aided anthropological-mathematical system for passenger simulation).

The hip point (H-point) is the pivot center of torso and thigh, and roughly corresponds to the hip joint location. The seating reference point (in accordance with ISO 6549 and US legislation) or R-point (ISO 6549 and EEC Directives/ECE Regulations) indicates the position of the design H-point in the rearmost normal driver seating position in the case of adjustable seats. In determining the design H-point position, many vehicle manufacturers use the 95th percentile adult-male position or, if this position is not reached, the position with the seat adjusted to its rearmost setting. In order to check the position of the measured H-point relative to the vehicle, a three-dimensional, adjustable SAE H-point machine weighing 75 kg is used. The seating reference point, heel point, vertical and horizontal distance between these two points, and body angles specified by the vehicle manufacturer form the basis for determining the dimensions of the driver's seating position.

The seating reference point is used

- to define the positions of the eye ellipse (SAE J941) and the eye points (RREG 77/649) as a basis for determining the driver's direct field of view;
- to define hand reach envelopes in order to correctly position controls and actuators;
- to determine the accelerator heel point (AHP) as a reference point for positioning the pedals.

The space required by the rear axle as well as location and shape of the fuel tank primarily determine the rear seating arrangement (height of the seating reference point, rear seating room, head-room) and thus the shape of the roof rear portion. Depending on the type of vehicle being developed, the projected main dimensions and the required passenger sizes, there are different body angles for the 2D templates or body postures (RAMSIS) and different distances between the seating reference points of the driver's and rear seats. The longitudinal dimensions are greatly influenced by the height of the seating reference point above the heel point. Lower seats require a more stretched passenger seating position, and thus greater interior length.

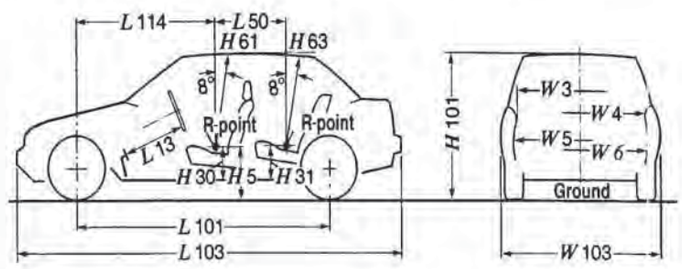
The passenger-compartment width is dependent on the projected exterior width, the shape of the sides (curvature) and the space required for door mechanisms, passive restraint systems and various assemblies (propellor-shaft tunnel, exhaust system etc.).

Luggage-compartment dimensions

The size and shape of the luggage compartment are dependent on the design of the rear of the vehicle, the position of the fuel tank and its volume, the position of the spare wheel and the location of the main muffler.

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Typical internal and external dimensions *(as per DIN 70020, Part 1)*



Dimension			Subcompact car mm	Long car mm
H	5	R-point to ground, front	460	510
H	30	R-point to accelerator heel point, front	240	300
H	31	R-point to accelerator heel point, rear	300	310
H	61	Effective headroom, front	940	980
H	63	Effective headroom, rear	920	950
H	101	Vehicle height	1360	1400
L	13	Steering wheel to brake pedal	480	630
L	50	R-point distance (front to rear seat)	710	830
L	101	Wheelbase	2430	2880
L	103	Vehicle length overall	3840	4930
L	114	Front wheel center to R-point	1250	1590
W	3	Shoulder room, front	1310	1430
W	4	Shoulder room, rear	1290	1420
W	5	Hip room, front	1260	1430
W	6	Hip room, rear	1240	1470
W	103	Vehicle width overall	1620	1820

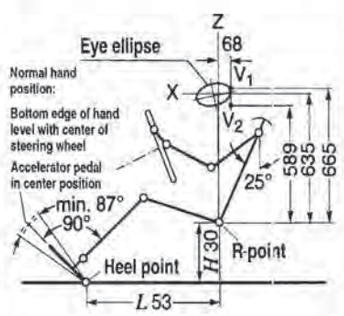
Luggage-compartment capacity is determined in accordance with DIN-ISO 3832 or, more commonly, with the VDA method using the VDA unit module (a right parallelepiped with dimensions of 200 x 100 x 50 mm – corresponds to a volume of 1 dm^3).

Exterior dimensions

The following factors must be taken into consideration:

- Seating arrangement and luggage compartment,
- Engine, transmission, radiator,
- Auxiliary and special equipment,
- Space required by sprung and turned wheels (allowance for snow chains),
- Type and size of drive axle,
- Position and volume of fuel tank,
- Front and rear bumpers,

Parameters which determine passenger-car driver's seat location



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- Aerodynamic considerations,
- Ground clearance (approx. 100...180 mm),
- Effect of body structure width on windshield wiper system (ADR 16, FMVSS 104).

Vision

A compromise must be found between optimum vision and the functional layout of components which tend to obstruct the field of vision.

The following influencing factors must be taken into consideration:

- Obscuration by pillars, the roof, the hood and the luggage-compartment lid. The eye ellipse in accordance with SAE J941 and the "eye points" in accordance with RREG 77/649[1]) are the basis for assessing the field of vision,
- Location, size and shape of the rear-view mirror in accordance with RREG 79/795.
- Windshield wiper pattern in accordance with FMVSS 104[2]) and RREG 78/318,
- Windshield curvature,
- View of instruments (obstruction of vision by steering wheel, ADR 18 A[3])).

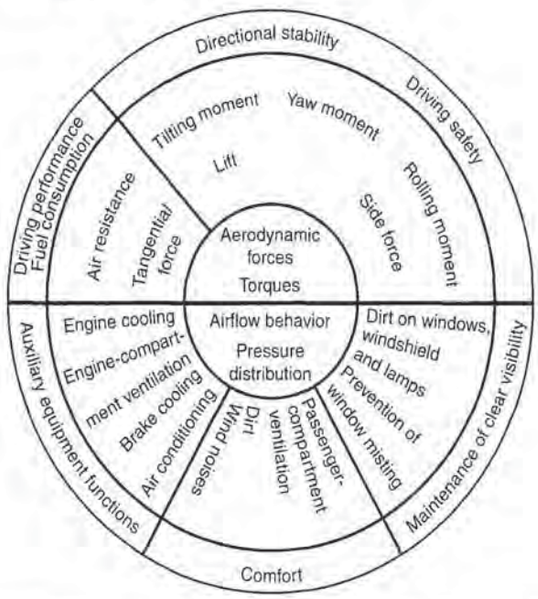
Body design

The following technical requirements must be met in interior and exterior body design:

- Mechanical functions (lowering of side windows, opening of the hood, luggage-compartment lid and sunroof, positions of lamps),
- Manufacturability and ease of repair (gap widths, bodywork assembly, window shape, protective molding rails, paint feature lines),
- Safety (position and shape of bumpers, no sharp edges or points),
- Aerodynamics (air resistance, dirt on the vehicle body, wind noises, ventilation openings, windshield-wiper operation),
- Optics (distortion caused by window type and slope, glare due to reflection),
- Legal requirements (position and size of lamps, rear-view mirror, license plates),

[1]) Directives of the Council of the European Union.
[2]) Federal Motor Vehicle Safety Standard (USA).
[3]) Australian Design Rule.

Aerodynamic effects on vehicle functions



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– Design and layout of controls (positions, shapes and surface contours),
– Clear layout (parking).

Aerodynamics

Aerodynamics deals with all processes which are observed as air flows through and around a vehicle.
Aerodynamic drag $W = c_W \cdot A \cdot v^2 \cdot \frac{\varrho}{2}$

where c_W = drag coefficient,
A = cross-sectional area,
v = driving speed,
ϱ = air density.

Table 1.
c_w values for various vehicles

Vehicle (Examples)	c_W –	A m²
Audi A8	0.29	2.25
Porsche 911	0.29	1.95
Mercedes C 200 D	0.30	2.05

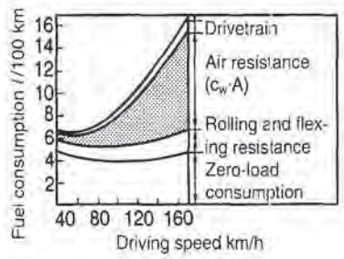
The factors to be influenced by the vehicle manufacturer are the drag coefficient c_W as a measure of the aerodynamic quality of the vehicle shape, and the projected vehicle cross-sectional area A.
Attempts to reduce the c_W value by adding spoilers, underbody panels, etc. are not practical, since their effect depends upon the aerodynamic characteristics of the basic vehicle body (see Table 2 for effects of individual modifications on drag coefficient). These types of modifications in optimized form are standard features of modern vehicle design. This means that the fitting of supplementary devices to further improve the c_W value and the axle lifting-force coefficient can only result in very minor improvements. Often, such modifications result in the impairment of the vehicle's day-to-day capabilities due to reduced overhang angle or reduced ground clearance. In other words, such add-on parts serve more for improving the vehicle's appearance than for improving its performance.

The c_W value can be influenced by individual aerodynamic or design measures. Airflow through the vehicle as well as roof-mounted fixtures will always increase the c_W value. Examples (– = better, + = worse). Examples are given in Table 2.

Table 2.
c_w values for vehicle modifications

Effect of	Δc_W in %
Lowering vehicle height by 30mm	approx. –5
Smooth wheel covers	–1...–3
Wide tires	+2...+4
Windows flush with exterior	approx. –1
Sealing body gaps	–2...–5
Underbody panels	–1...–7
Concealed headlamps	+3...+10
Outside rear-view mirrors	+2...+5
Airflow through radiator and engine compartment	+4...+14
Brake cooling devices	+2...+5
Interior ventilation	approx. +1
Open windows	approx. +5
Open sunroof	approx. +2
Roof-mounted surfboard rack	approx. +40

Effect of $c_W \cdot A$ on fuel consumption (mid-sized vehicle)



Effect of windshield slope α on the c_W value
(– = better, + = worse)



α	c_W	Δc_W in %
50°	0.345	–
55°	0.342	–0.8
65°	0.340	–1.4
40°	0.349	+1.1
30°	0.349	+1.1
0°	0.369	+7.0

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Body structure

Unitized (self-supporting) body
(Standard design)
The conventional unitized (self-supporting) body is built up of hollow sheet-steel components onto which body panels are welded by welding robots or in multi-spot welding units. Individual parts can also be glued in position.

Depending upon vehicle type, roughly 5000 spot welds must be made along a total flange length of 120...200 m. The flange widths are 10...18 mm. Other parts (front fenders, doors, hood and luggage-compartment lid) are bolted to the supporting structure of the body. Other types of body construction include frame and sandwich designs.
General requirements:

Rigidity
Torsional and bending rigidity should be as high as possible in order to minimize elastic deformation of the apertures for the doors, hood and luggage-compartment lid. The effect of body rigidity on the vibrational characteristics of the vehicle must be taken into consideration.

Vibrational characteristics
Body vibrations as well as vibrations of individual structural components as a result of excitation by the wheels, suspension system, engine and drivetrain can severely impair driving comfort if resonance occurs.

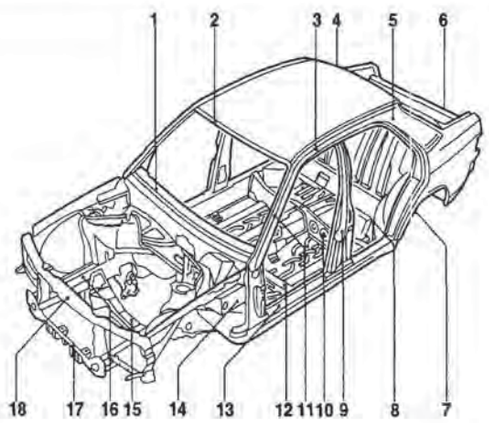
The natural frequency of the body, and of those body components which are capable of vibration, must be detuned by means of creasing and changing the wall thicknesses and cross sections, such that resonance and its consequences are minimized.

Operational integrity
Alternating stresses which can affect the body as the vehicle is driven can lead to incipient structural cracks or weld failure. Areas which are particularly susceptible are the bearing points of the running gear, the steering system and engine units.

Body stresses due to accidents
In the event of a collision, the body must be capable of transforming as much kinetic energy as possible into deformation work while minimizing deformation of the vehicle interior.

Ease of repair
Those components which are most susceptible to damage as a result of minor accidents ("fender-benders") must be easily replaceable or repairable (access

Body structure
1 Cross member under windshield,
2 Roof frame, front,
3 Roof frame, side,
4 Roof frame, rear,
5 C-pillar,
6 Rear-facing panel,
7 Rear floor and spare-wheel pan,
8 Side member, rear,
9 B-pillar,
10 Cross member under rear seat,
11 A-pillar,
12 Cross member under driver's seat,
13 Side member,
14 Wheel well,
15 Engine-support cross member,
16 Side member, front,
17 Cross member, front,
18 Radiator cross member.



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to exterior body panels from inside, access to bolts, favorable location of joints, feature lines for the repainting of individual components).

Body materials

Sheet steel

Sheet steel of various grades (see table on P. 197) is customarily used for the vehicle body structure.

Sheet thicknesses range from 0.6 to 3.0 mm, with most pieces being between 0.75 and 1.0 mm thick. Due to the mechanical properties of steel with regard to stiffness, strength, economy and ductility, alternative materials for the vehicle body structure are not yet available.

High-strength, low-alloy (HSLA) sheet steel is used for highly-stressed structural components. The resulting high strength of these components allows their thickness to be reduced.

Aluminum

In order to reduce weight, aluminum can be used for separate body components such as the hood, luggage-compartment lid, etc.

Since 1994, an aluminum body has been in use for one of the German luxury-class cars. The vehicle's frame is constructed from aluminum extruded sections, and the sheet components are self-supporting in integrated form (ASF Audi Space Frame). The realization of this principle necessitated the employment of suitable aluminum alloys, as well as new production processes and special repair facilities. According to the manufacturers, the rigidity and deformation characteristics are identical to those of steel or are even superior.

Plastics

Plastics as materials for separate body components are used in a limited number of cases in place of steel (see Table below).

Table 3. Examples of alternative materials

Examples of application	Material	Abbreviation	Processing method
Load-bearing components, e.g. bumper beam	Thermoplastics reinforced by glass-fiber mats	PP-GMT	Injection pressure
Moldings/covers, e.g. front apron, spoiler, front section, radiator grill, wheel-well liners, wheel covers Separate vehicle-body components, e.g. hood, fender, luggage-compartment lid, sunroof	Thermoplastics reinforced by glass-fiber mats	PP-GMT	
	Polyurethane	PUR	RIM (Reaction-Injection-Molding) RRIM (Reinforced-Reaction-Injection-Molding)
	Polyamide Polypropylene Polyethylene Acrylonitrile-butadiene-styrene copolymeres Polycarbonate (with Polybutadiene theraphthalate mod.)	PA PP PE ABS PC-PBT	Injection-molding, glass-fiber constituent determines elasticity
Protective molding rails	Polyvinylchloride Ethylene-Propylene-Terpolymers Elastomer-modified polypropylene	PVC EPDM PP-EPDM	Injection-molding/extrusion
Energy-absorbing foam	Polyurethane Polypropylene	PUR PP	Reaction foaming

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Body surface

Corrosion protection
Allowance must be made for corrosion protection as early as during the body design phase ("Anti-Corrosion Code", Canada).
Corrosion protection measures:
- Minimize flanged joints, sharp edges and corners,
- Avoid areas where dirt and humidity can accumulate,
- Provide holes for pretreatment and electrophoretic enameling,
- Provide good accessibility for the application of corrosion inhibitor,
- Allow for ventilation of hollow spaces,
- Prevent the penetration of dirt and water to the greatest extent possible; provide water drain openings,
- Minimize the area of the body exposed to stone-chip throw,
- Prevent contact corrosion.

Precoated sheet steel (inorganic zinc, electrolytically galvanized, hot-dip galvanized) is often used for those components which are particularly endangered, such as doors and load-bearing members at the front of the vehicle. Particularly inaccessible structural areas are coated with spot-welding paste (PVC or epoxy adhesive, approx. 10 ... 15 m seam length per vehicle) prior to assembly.

Painting (P. 228)
Measures subsequent to electrophoretic enameling:
- Covering the spot-welded seams (up to 90 ... 110 m), folds and joints with PVC sealing compound,
- Coating the underbody with PVC underseal (0.3 ... 1.4 mm thick, 10 ... 18 kg per vehicle) to protect against damage due to stone chips. Plastic trim panels can be used as an alternative.
- Preservation of hollow parts with penetrating, non-aging water-based wax,
- Use of corrosion-resistant, separate plastic components in highly susceptible areas such as the front wheel wells (PVC coating is not suitable here).
- Preservation of underbody and engine compartment after final assembly.

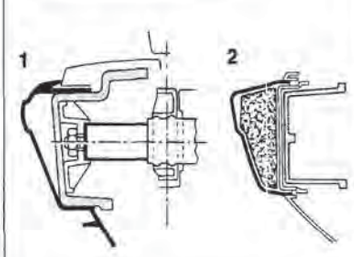
Table 4. Paint/coating thicknesses

Coating system, overall thickness	≈ 120 µm
Zinc-phosphate coating	≈ 2 µm
Electrophoretic enameling (cathodic)	13 ... 18 µm
Filler	≈ 40 µm
Top-paint coat	35 ... 45 µm
Varnish coat (only for metallic paint)	40 ... 45 µm

Body finishing components

Bumpers
The front and rear of the vehicle should be protected in such a manner that low-speed collisions will only damage the vehicle slightly, or not at all. Prescribed bumper evaluation tests (US Part 581, Canada CMVSS 215, and ECE-R 42) specify minimum requirements in terms of energy absorption and installed bumper height. Bumper evaluation tests in accordance with US Part 581 (4 km/h barrier collision, 4 km/h pendulum tests) must be passed by a bumper system whose energy absorber is of the no-damage absorber type. The requirements of the ECE standard are satisfied by plastically deformable retaining elements located between the bumper and the vehicle body structure. In addition to sheet steel, many bumpers are manufactured using fiber-reinforced plastics and aluminum sections.

Bumpers
1 Shock-absorber system, 2 Energy-absorbing PUR-foam system



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Exterior trim, impact strips

Plastics have become the preferred materials for external impact strips, trim, skirts and spoilers, and particularly for those components whose purpose is to improve the aerodynamic characteristics of the vehicle. Criteria used in the selection of the proper material are flexibility, high-temperature shape retention, coefficient of linear expansion, notched-bar toughness, resistance to scratches, resistance to chemicals, surface quality and paintability.

Glazing

The windshield and rear window are usually held in rubber strips and sealed or bonded in place.

The total weight of the windows in a vehicle ranges from 25 to 35 kg. Due to a number of inherent disadvantages, weight considerations have not yet led to plastics (PC, PMMA) being introduced as the substitute for glass. Due to its heat-insulation and noise-damping properties, 2-layer insulating glass has come into use for door windows.

Door latches

Door latches are of great importance with regard to passive accident safety (pertinent regulations: ECE-R 11 and FMVSS 206, among others):

- Fully latched and secondary latched positions,
- Complete integrity under a longitudinal load of up to 12 kN and a transverse load of up to 10 kN (fully latched position),
- Complete integrity under a longitudinal or transverse inertia load of up to $30 \cdot g$.

Individual manufacturers have varying solutions with regard to ease of operation, anti-theft protection and child-proof operation.

Seats

The strength requirements which must be met by the seats in a collision pertain to the seat cushion and backrest, the head restraints, the seat adjustment mechanism and the seat anchors (pertinent regulations: FMVSS 207, 202; ECE-R 17, 25; RREG 74/408, 78/932 and others). One component of active safety is seating comfort. Seats must be designed such that vehicle occupants with different body dimensions do not suffer from driving fatigue.

Parameters:

- Support of individual body areas (distribution of pressure),
- Lateral support when cornering,
- Seating ambience,
- Freedom of movement so that an occupant can change his/her sitting position without having to readjust the seat,
- Vibrational and damping characteristics (matching of the natural frequency with the excitation frequency band),
- Adjustability of seat cushion, backrest and head restraint.

The above parameters are affected by the following:

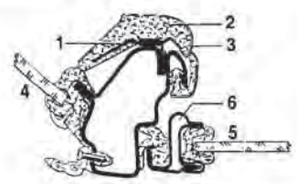
- Dimensions and shapes of the upholstery in the seat cushion and backrest,
- Distribution of the spring rates of individual cushioned zones,
- Overall spring rate and damping capacity, of the seat cushions in particular,
- Thermal conductivity and moisture-absorption capacity of the covers and upholstery,
- Operation and range of the seat adjustment mechanisms.

Interior trim

A section of trim consists of a dimensionally stable core (sheet steel, sheet aluminum or plastic) with mounting hardware, and energy-absorbing cushion made of foam material (e.g. PUR) and a flexible surface layer. One-piece plastic trim sections made of injection-molded thermoplastic material are also used.

The headliner is made either as a stretched liner or finished liner. The ma-

Section through an A-pillar with trim *(Principle)*
1 Core, 2 Foam, 3 Film, 4 Windshield, 5 Side window, 6 Door frame.



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terials used must be flame-retardant and slow burning (FMVSS 302).

Safety

Active safety:
Prevention of accidents
Passive safety:
Reduction of accident consequences

Active safety

Driving safety is the result of a harmonious chassis and suspension design with regard to wheel suspension, springing, steering and braking, and is reflected in optimum dynamic vehicle behavior.

Conditional safety results from keeping the physiological stress that the vehicle occupants are subjected to by vibration, noise, and climatic conditions down to as low a level as possible. It is a significant factor in reducing the possibility of misactions in traffic.

Vibrations within a frequency range of 1 to 25 Hz (stuttering, shaking, etc.) induced by wheels and drive components reach the occupants of the vehicle via the body, seats and steering wheel. The effect of these vibrations is more or less pronounced, depending upon their direction, amplitude and duration.

Noises as acoustical disturbances in and around the vehicle can come from internal sources (engine, transmission, propshafts, axles) or external sources (tire/road noises, wind noises), and are transmitted through the air or the vehicle body. The sound pressure level is measured in dB (A) (see also PP. 52).

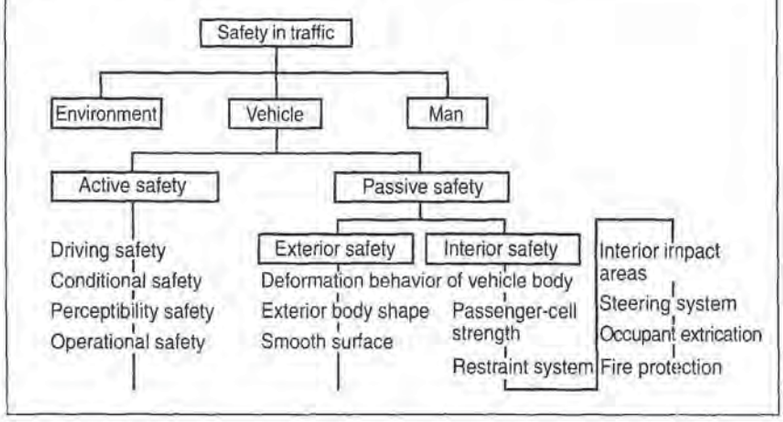
Noise reduction measures are concerned on the one hand with the development of quiet-running components and the insulation of noise sources (e.g., engine encapsulation), and on the other hand with noise damping by means of insulating or anti-noise materials.

Climatic conditions inside the vehicle are primarily influenced by air temperature, air humidity, rate of air flow through the passenger compartment and air pressure (see PP. 364 for additional information).

Perceptibility safety
Measures which increase perceptibility safety are concentrated on
- Lighting equipment (see P. 738),
- Acoustic warning devices (see P. 766),
- Direct and indirect view (see P. 720) (Driver's view: The angle of obscuration caused by the A-pillars for both of the driver's eyes – binocular – must not be more than 6 degrees).

Operating safety
Low driver stress, and thus a high degree of driving safety, requires optimum design

Safety in traffic. *Terms and influencing factors*



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of the driver's surroundings with regard to ease of operation of the vehicle controls.

Passive safety

Exterior safety

The term "exterior safety" covers all vehicle-related measures which are designed to minimize the severity of injury to pedestrians and bicycle and motorcycle riders struck by the vehicle in an accident. Those factors which determine exterior safety are:
- Vehicle-body deformation behavior,
- Exterior vehicle-body shape.

The primary objective is to design the vehicle such that its exterior design minimizes the consequences of a primary collision (a collision involving persons outside the vehicle and the vehicle itself).

The most severe injuries are sustained by passengers who are hit by the front of the vehicle, whereby the course of the accident greatly depends upon body size. The consequences of collisions involving two-wheeled vehicles and passenger cars can only be slightly ameliorated by passenger-car design due to the two-wheeled vehicle's often considerable inherent energy component, its high seat position and the wide dispersion of contact points. Those design features which can be incorporated into the passenger car are, for example:
- Movable front lamps,
- Recessed windshields wipers,
- Recessed drip rails,
- Recessed door handles.

See also ECE-R26, RREG 74/483

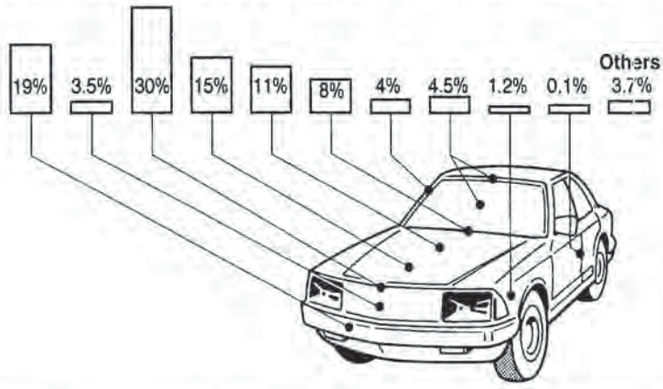
Interior safety

The term "interior safety" covers vehicle measures whose purpose is to minimize the accelerations and forces acting on the vehicle occupants in the event of an accident, to provide sufficient survival space, and to ensure the operability of those vehicle components critical to the removal of passengers from the vehicle after the accident has occurred.

The determining factors for passenger safety are:
- Deformation behavior (vehicle body),
- Passenger-compartment strength, size of the survival space during and after impact,
- Restraint systems,
- Impact areas (vehicle interior), (FMVSS 201),
- Steering system,
- Occupant extrication,
- Fire protection.

Laws which regulate interior safety (frontal impact) are:
- Protection of vehicle occupants in the event of an accident, in particular restraint systems (FMVSS 208, ECE R94, injury criteria),
- Windshield mounting (FMVSS 212),

Risk to pedestrians in event of collisions with passenger cars
as a function of impact frequency and seriousness of injury (based on 246 collisions)



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- Penetration of the windshield by vehicle body components (FMVSS 219),
- Parcel-shelf and compartment lids (FMVSS 201).

Rating-Tests:
- New-Car Assessment Program (NCAP, USA, Europe, Japan, Australia),
- IIHS (USA, insurance test),
- ADAC, ams, AUTO-BILD.

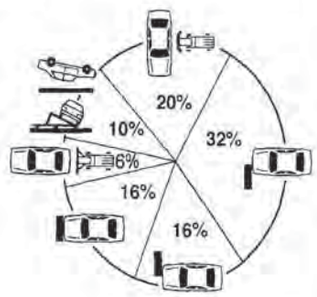
Deformation behavior of vehicle body

Due to the frequency of frontal collisions, an important role is played by the legally stipulated frontal impact test in which a vehicle is driven at a speed of 48.3 km/h (30 mph) into a rigid barrier which is either perpendicular or inclined at an angle of up to 30° relative to the longitudinal axis of the car.

Because 50 % of all frontal collisions in right-hand traffic primarily involve the left-hand half of the front of the vehicle, manufacturers worldwide conduct left asymmetrical front impact tests on LHD vehicles covering 30...50 % of the vehicle width.

In a frontal collision, kinetic energy is absorbed through deformation of the bumper, the front of the vehicle, and in severe cases the forward section of the passenger compartment (dash area). Axles, wheels (rims) and the engine limit the deformable length. Adequate deformation lengths and displaceable vehicle aggregates are necessary, however, in order to minimize passenger-compartment acceleration. Depending upon vehicle design (body shape, type of drive and engine position), vehicle mass and size, a frontal impact with a barrier at approx. 50 km/h results in permanent deformation in the forward area of 0.4...0.7 m. Damage to the passenger compartment should be minimized. This concerns primarily
- dash area (displacement of steering system, instrument panel, pedals, toe-panel intrusion),
- underbody (lowering or tilting of seats),
- the side structure (ability to open the doors after an accident).

Distribution of accidents by type of collision
Symbolized by test methods yielding equal results



Acceleration measurements and evaluations of high-speed films enable deformation behavior to be analyzed precisely. Dummies of various sizes are used to simulate vehicle occupants and provide acceleration figures for head and chest as well as forces acting on thighs.

Head acceleration values are used to determine the head injury criterion (HIC). The comparison of measured values supplied by the dummies with the permissible limit values as per FMVSS 208 (HIC: 1000, chest acceleration: 60 *g*/3 ms, upper leg force: 10 kN) are only limited in their applicability to the human being.

The side impact, as the next most frequent type of accident, places a high risk of injury on the vehicle occupants due to the limited energy absorbing capability of trim and structural components, and the resulting high degree of vehicle interior deformation.

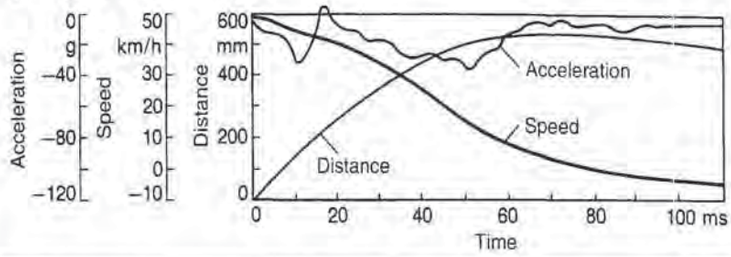
The risk of injury is largely influenced by the structural strength of the side of the vehicle (pillar/door joints, top/bottom pillar points), load-carrying capacity of floor cross-members and seats, as well as the design of inside door panels (FMVSS 214, ECE R95, Euro-NCAP, US-SINCAP).

In the rear impact test, deformation of the vehicle interior must be minor at most. It should still be possible to open the doors, the edge of the trunk lid should not penetrate the rear window and enter the vehicle interior, and fuel-system integrity must be preserved (FMVSS 301).

Roof structures are investigated by means of rollover tests and quasi-static car-roof crush tests (FMVSS 216).

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Acceleration, speed and distance traveled, of a passenger compartment when impacting a barrier at 50 km/h



In addition, at least one manufacturer subjects his vehicles to the inverted vehicle drop test in order to test the dimensional stability of the roof structure (survival space) under extreme conditions (the vehicle falls from a height of 0.5 m onto the left front corner of its roof).

Steering system

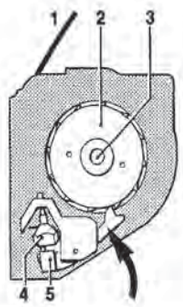
Legal requirements (FMVSS 203 and 204) regulate the maximum displacement of the top end of the steering column toward the driver (max. 127 mm, frontal impact at 48.3 km/h) and the limit of the impact on the steering system of a test piece (maximal 1111 daN at an impact speed of 24.1 km/h). Slotted tubes, corrugated tubes and breakaway universal joints (among others) are used in the design of the lower section of the steering column spindle so that it can be deformed both longitudinally and transversely.

Passenger restraint systems (P. 803)

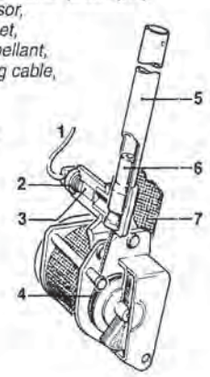
Automatic seat belt (manual systems)
The most frequently installed three-point seat belt with retractor mechanism ("automatic seat belt") represents a good compromise between effective safety, ease of buckling, comfort and cost. When a specific vehicle-deceleration value is reached, a built-in, quick-response interlock inhibits the seat-belt roller.

Seat-belt tightener systems
Seat-belt tightener systems represent a further development and improvement of the three-point automatic seat-belt systems. By reducing seat-belt slack, they eliminate excessive forward passenger movement in serious accidents. This in turn reduces the differential speed between the vehicle and passengers, and thus also reduces the corresponding

Inertia-reel seat-belt system
1 Seat belt, 2 Ratchet wheel, 3 Inertia-reel shaft, 4 Pendulum, 5 Pawl (in locked position).



Seat-belt tightener (Example)
1 From sensor,
2 Firing pellet,
3 Solid propellant,
4 Tensioning cable,
5 Cylinder,
6 Piston,
7 Seat belt.



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forces acting on the passengers.

Integrated belt-force limiters ensure that controlled give of the belt takes place after it has been tightened, so as to prevent potential overloading in the chest area.

Airbag systems

Airbags (frontal airbags, side bags, window bags) serve to prevent or reduce the impact of the occupant against interior vehicle components (steering wheel, instrument panel, doors, windows, roof pillars).

Calculations

Finite-Element Method

The Finite-Element Method (FEM) can be used to calculate static, dynamic and acoustic characteristics of components and complete bodies. In finite-element analysis, a support structure of any degree of complexity is broken down into simple structural elements (beams, shells and solid elements etc.), of which the elastic behavior is known and can be easily defined. These elements are then assembled to form the overall structure, taking into account compatibility conditions.

This enables a mathematical model to be constructed in such a manner that it sufficiently corresponds to the actual body in terms of its elastic characteristics.

A number of systems are available for practical FE calculation, such as PERMAS, NASTRAN, ABAQUS and DYNA3D.

Advantages of FEM:

- Characteristics of structures of any level of complexity can be calculated,
- Anisotropic and non-linear material properties can be taken into account,
- Variations can be rapidly examined,
- Tried and tested program systems are available (easy to incorporate into the CAD/CAM chain.

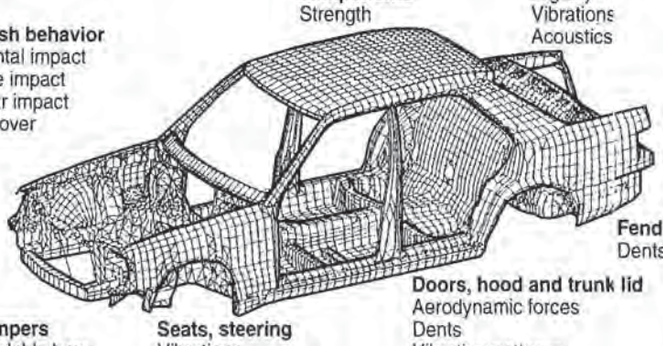
Limits of FEM:

- Accuracy is dependent on element type and precision of element distribution within the structure,
- Changes in sheet-metal thicknesses and material characteristic values resulting from the deep-drawing process are not taken into account,
- Welding joints cannot be precisely replicated in the model.

Complete body calculation

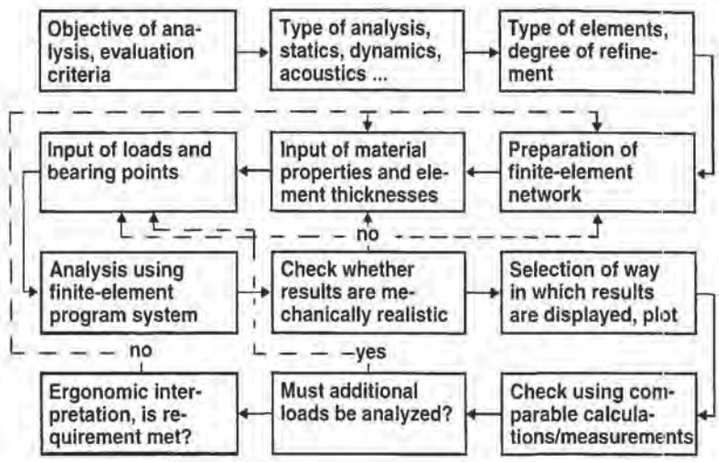
For the calculation, the bodywork structure is broken down into elements with the required level of precision depending on the problem to be solved (as at 1999: up to approx. 300,000 elements with over 1.8 million unknowns). Results under static

Calculation of a vehicle body structure



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The structural analysis process using the finite-element method



load are supplied in the form of deformations, stresses and deformation work.

Strength analysis
Detailed analyses are carried out for individual parts and body sections which are exposed to special stresses caused by factors such as e.g. restraint systems or towed loads. The aim of these analyses is to furnish proof of adequate strength or to reduce unacceptably high stresses by modifying the design.

Analysis of dynamic behavior
Dynamic analyses are carried out both for the complete body and for the individual components. In these analyses, the natural vibration (frequencies, forms of vibration) and the system's response to periodic or generally time-dependent excitation is determined. In this way, critical resonances can be identified and the loads which determine operational integrity can be defined.

For the examination of ride comfort and acoustical behavior in the vehicle interior, FE models of the chassis, engine, doors etc., as well as a model of the interior, are added to the body model.

Analysis of crash behavior
Accident tests carried out by automobile manufacturers (frontal, rear and side impacts, rollover and drop tests) as well as traffic accidents, are dynamic, to a large degree non-linear processes which cannot be described using the current FE programs. Special FE program systems (e.g., DYNA3D, PAM-CRASH) have been developed for numerical simulation of these processes, and are proving successful in the field. These systems include analysis of severe plastic deformations as well as recording of the contact areas arising between various vehicle parts during crash processes.

Analysis of occupant-protection systems
The following main tasks are carried out with the aid of computer simulation:
- Design/optimization of restraint systems with regard to front- and rear-end impacts by means of crash-test sled simulations in the cases in which the interaction between structure and dummy is negligible. The tool for this purpose is a multibody system (e.g. Madymo 3D) in conjunction with an FE

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program system (e.g. LS-DYNA-3D).
- Design of protective components with regard to side impacts in integrated simulations, i.e. complete-vehicle simulations with dummy, seat and moldings. Calculation with FE program system (LS-DYNA-3D).
- Design and optimization of cushioning measures (in accordance with FMVSS 201) with regard to satisfying protective criteria. Calculation with FE program system (LS-DYNA-3D).

Recycling, environmental protection

The automobile manufacturers are making progress in their endeavors to maintain clean water and air, prevent noise pollution and recycle raw materials:
- The recycling quota for metallic materials is 95%.
- The recycling quota for plastics is approximately 14%. In cooperation with the chemical industry and the suppliers of plastic components, intensive efforts are being made to further improve this figure.
- Exhaust-gas purification and recovery of high-value catalyst materials.
- Recycling of battery materials.
- Chlorofluorocarbons (CFCs) are no longer used as refrigerants or as expanding agents in the manufacture of plastics.
- Solvents for body degreasing are almost entirely free of chlorinated hydrocarbons; water-based paints are used in dip-priming.
- Water-based paints for top coating are in use.
- Reprocessing of manufacturing agents, e.g. oil, coolant, antifreeze.
- Recycling of some 60% of residual materials (e.g. scrap metal, waste paper, leather, textile and wood waste).

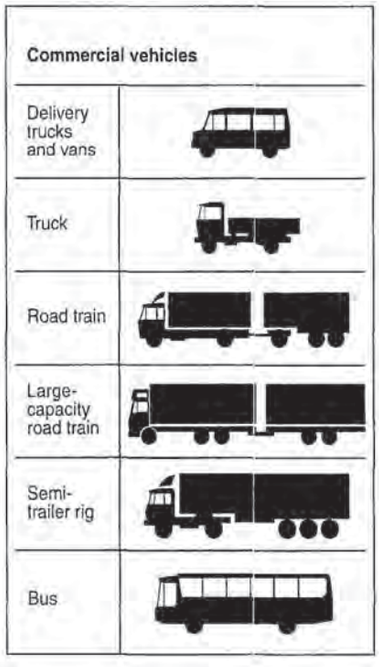
Vehicle bodies, commercial vehicles

Commercial vehicles

Commercial vehicles are used for the safe and efficient transportation of persons and goods. In this respect, the degree of economic efficiency is determined by the ratio of usable space to overall vehicle volume, and of useful load to laden vehicle weight. Dimensions and weights are limited by legal regulations.

From the design-concept viewpoint, a distinction must be made between cab-over-engine (COE) and cab-behind-engine (CBE) vehicles.

A wide variety of vehicle types meet the demands of local and long-distance transportation, as well as the demands encountered on building sites and in special applications.



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Delivery trucks and vans

These are light-duty trucks (2...7 t) used in the transportation of persons and in local goods distribution. In fulfilling this function stringent demands are made on the vehicle in terms of mobility, maneuverability, performance, operating comfort and safety. The design concepts are based on front-mounted engine, front or rear-wheel drive, independent suspension or rigid axle and, from 3.5 t laden weight, twin tires on the rear axle.

The product range includes enclosed-body multi-purpose vehicles and vans, as well as low-bed and high-bed platform-body vehicles with special superstructures and double cabs.

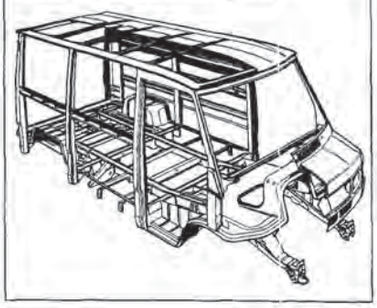
In small delivery trucks and vans, the bodies form an integral load-bearing unit together with the chassis.

The body and chassis frameworks consist of sheet-metal pressed elements and flanged profiles. Platform-body vehicles have a ladder-type frame with open or closed side members and cross members as the primary load-bearing structure.

Delivery trucks and vans

Van	
Double-cab low-bed truck	
Platform-type high-bed truck	
Chassis	

Delivery-truck load-bearing unit



Large delivery trucks and vans mostly have their own framework and a separate body.

Medium- and heavy-duty trucks and tractor vehicles

The vehicles in this sector have either a load-bearing chassis or partially load-bearing body. In most cases the engine is at the front. It is seldom fitted as an underfloor engine between the axles. The vehicle is driven via the twin-tire axle(s). In individual cases, the rear axle is fitted with single tires. For building-site (off-road) use with high traction requirements, all-wheel drive with longitudinal and cross-lock technology is applied.

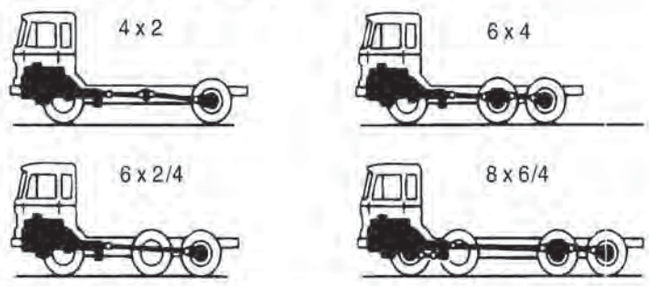
Type of truck undercarriage (see Figure below):
N x Z/L
N = Number of wheels
Z = Number of driven wheels
L = Number of steered wheels

(twin wheels count as one wheel)

Normal chassis have leaf- or pneumatically-sprung rigid front and rear axles. Pneumatic suspension facilitates the simple mounting and removal of interchangeable bodies and unhitching of semi-trailers. Three-axle vehicles (6x2) are fitted with either a leading or a trailing axle (in front of or behind the driven axle) to increase the useful load. High-traction 6x4 vehicles for use on building sites have a twin-axle configura-

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Types of truck undercarriage (examples)



tion with axle-load compensation and center bearing point or air-sprung single axles.

Chassis frames

The chassis frame is the commercial vehicle's actual load-bearing element. It is designed as a ladder-type frame, consisting of side and cross members. The choice of profiles decides the level of torsional stiffness. Torsionally flexible frames are preferred in medium- and heavy-duty trucks because they enable the suspension to comply better with uneven terrain. Torsionally stiff frames are more suitable for smaller delivery vehicles and vans.

Apart from the force introduction points, critical points in the chassis-frame design are the side-member and cross-member junctions. Special gusset plates or pressed cross-member sections form a broad connection basis. The junctions are riveted, bolted and welded. U- or L-shaped side-member inserts provide increased framework flexural strength and reinforcement at specific points.

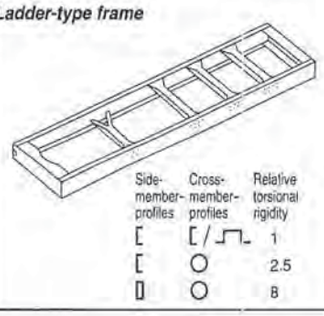
Driver's cab

There are a variety of cab designs available depending on the vehicle concept. In delivery vehicles and vans, low, convenient entrances are an advantage, whereas in long-distance transport space and comfort are more important. Modular design concepts allow for short, medium and long cab versions while retaining the same front, rear and doors.

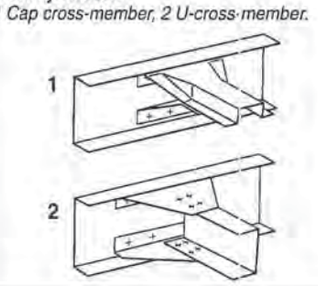
In the case of cab-over-engine (COE) vehicles, the steering system is positioned right at the front of the vehicle. The engine is located under the cab or under an engine tunnel between the driver and the co-driver. The entrance is positioned in front of or above the front axle. A mechanical (pretensioned torsion bar) or hydraulic cab-tipping mechanism ensures good access to the engine.

In cab-behind-engine (CBE) vehicles, the engine/transmission assembly is mounted ahead of the cab firewall beneath a steel or plastic hood (which is usually tiltable for reasons of accessibility). The driver enters the cab behind the front axle.

Ladder-type frame



Frame junctions
1 Cap cross-member, 2 U-cross-member.



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Bodies

Specific body structures such as flatbeds, standard vans, box vans, dump-truck deep-beds, tankers, concrete mixers etc. permit the economical and efficient transportation of a wide variety of goods and materials. Connection between body and load-bearing chassis frame is effected in part by means of auxiliary frames with non-positive or positive attachments.

Road trains and semitrailer rigs are used in long-distance transport. As the size of the transportation unit increases, the costs relative to the freight volume decrease.

Load volume is increased by reducing the empty spaces between cab, cargo area and trailer (high-capacity road train). Advantages of semitrailer rigs lie in the greater uninterrupted loading length of the cargo area and the shorter inoperative times of the tractor units. Measures to improve aerodynamics, such as front and side trim on the vehicle and specially adapted air deflectors from the cab to the body, are applied to minimize fuel consumption.

Buses

The bus market offers a specific vehicle for practically every application. This has resulted in a wide range of bus types, which differ in their overall dimensions (length, height, width) and appointments (depending on the application).

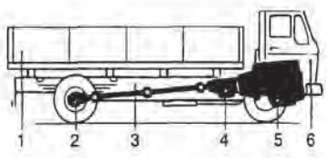
Microbuses

Microbuses carry up to approx. 19 passengers. These vehicles have been developed from delivery trucks or vans weighing up to approx. 4.5 t.

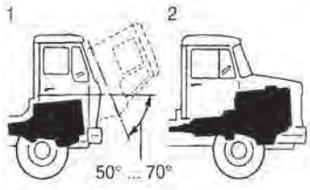
Minibuses

Minibuses carry up to approx. 25 passengers. These vehicles have been developed from delivery trucks or vans weighing up to approx. 7.5 t. They are occasionally built on the ladder-type chassis of light-duty trucks. A modified suspension design and special measures carried out on the body result in optimum ride comfort and noise levels.

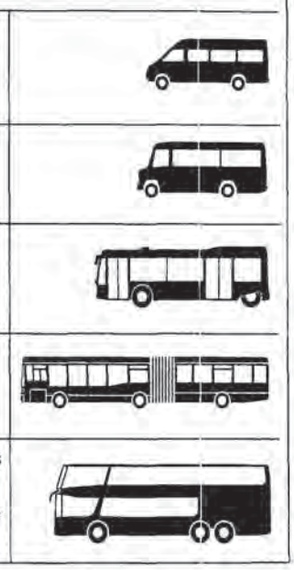
Truck assemblies
1 Body, 2 Axle, 3 Chassis frame, 4 Transmission, 5 Engine, 6 Cab.



Driver's cab
1 Cab over engine (COE), 2 Cab behind engine (CBE).



Bus types



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Midibuses
Depending on the application, midibuses can carry up to approx. 35 passengers (in tour buses) and up to approx. 65 passengers (in city buses). These vehicles weigh up to approx. 12.5 t and are predominantly built on the ladder-type chassis of light-duty trucks. There are also integral-framework designs in operation. A modified suspension design and special measures carried out on the body result in optimum ride comfort and noise levels.

City buses
City buses are equipped with seating and standing room for scheduled routes. The short intervals between stops in suburban passenger transportation operations necessitate a rapid passenger turnover. This is achieved by wide doors that open and close swiftly, low boarding heights (approx. 320 mm) and low floor heights (approx. 370 mm).

Main specifications for a 12 m standard public-service bus:
Vehicle length approx. 12 m,
Total weight 18.0 t,
Number of seats 32...44,
Total passenger capacity approx. 105 persons.

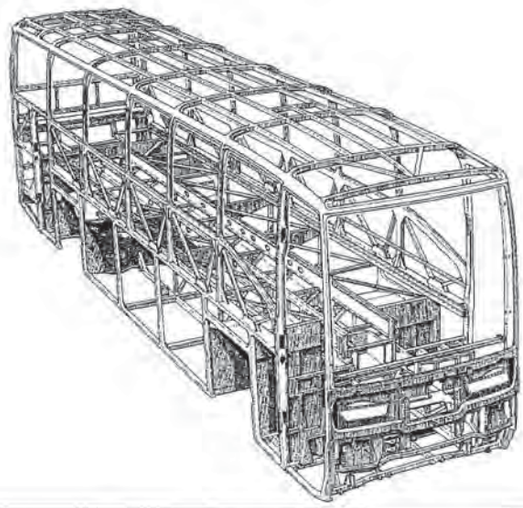
The use of double-decker buses (length 12 m carrying up to approx. 130 passengers), three-axle rigid buses (length up to 15 m carrying up to approx. 135 passengers) and articulated buses (approx. 160 passengers) provides increased transport capacity.

Intercity buses or touring coaches
Depending on the application (standing passengers are not permitted at $v > 60$ km/h), either low-floor designs featuring the low boarding and floor heights of public-service buses are used, or buses are employed which, with their higher floors and small luggage compartments, are already very much like tour buses. Overland buses come in lengths of 11...15 m as rigid vehicles or in lengths of 18 m as articulated vehicles.

Tour buses (long-distance coaches)
Tour buses are designed to provide comfortable travel over medium and long distances. They range from the low, two-axle standard bus through to the double-decker luxury bus and come in lengths of approx. 10...15 m.

Body structure
Light design based on an integral body.

Integral bus body



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The body and base frames, which are firmly welded together, consist of pressed grid-type support elements and rectangular tubes.

Undercarriage
The horizontally or vertically mounted engine drives the rear axle. Pneumatic suspension at all axles permits ride-level stabilization and a high degree of ride comfort. Intercity buses and touring coaches are mainly equipped with independent wheel suspension on their front axles. Disk brakes, frequently supported by retarders, are used on all the axles.

Passive safety in commercial vehicles

Passive safety is intended to limit the consequences of accidents and to protect other road users. Systematic recording of accidents, accident tests with complete commercial vehicles and intensive computer optimization help to devise safety measures.

In the event of a collision, the driver's cab and the passenger compartment must maintain the amount of room necessary for occupant survival, while at the same time deceleration must not be excessive. Depending upon vehicle design, there are a variety of solutions to this problem.

In delivery trucks and vans, front-section design is energy-absorbing as in passenger cars. In spite of shorter deformation routes and higher levels of released energy, the physiologically permissible limits are not exceeded in the case of virtually all passenger-car crash-test standards (legal requirements and rating tests).

In the case of trucks, the side members extend up to the front bumper and can absorb high longitudinal forces. Such passive-safety measures are based on accident analyses and are intended to improve the structural design of the cab. Static and dynamic stress and impact tests at the front and rear of the cab, as well as on its roof, simulate the stresses involved in a frontal impact and in accidents in which the vehicle overturns or rolls over, as well as in which the load shifts.

Statistical analyses have proved that the bus is one of the safest means of passenger transportation. Static roof-load tests and dynamic overturning tests provide evidence of body strength. The use of flame-retardant and self-extinguishing materials for the interior of the vehicle minimizes the risk of fire.

Because road traffic involves many different kinds of vehicles, collisions between light and heavy vehicles are unavoidable. As a result of the differences in vehicle weight, and incompatibility in terms of vehicle geometry and structural stiffness, the risk of injury in the lighter vehicle is greater. The formulas below define the change in speed during a normal (non-oblique) plastic impact for frontal or rear collisions between two vehicles:

$$\text{Vehicle 1 } \Delta c_1 = \frac{\mu \cdot \Delta v}{1+\mu}$$

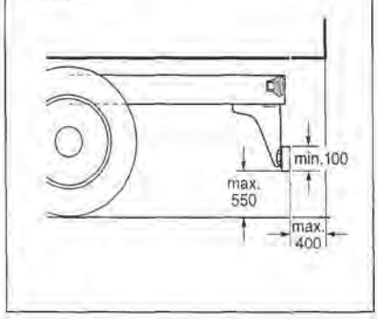
$$\text{Vehicle 2 } \Delta c_2 = \frac{\Delta v}{1+\mu}$$

where $\mu = m_2/m_1$

m_1, m_2 = masses of the vehicles involved,
v relative speed prior to impact.

Side, front and rear underride guards help to reduce the danger of the lighter vehicle driving under the heavier vehicle in the event of a collision. In other words, they serve to protect other road users.

Rear underride guard for commercial vehicles



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Lighting

Functions

Vehicle front end
The primary function of the headlamps at the vehicle front end is to illuminate the roadway so that the driver can register the traffic conditions and recognize any obstacles and hazards in good time. They also serve to identify and mark out the vehicle to oncoming traffic. The turn-signal lamps serve to show the driver's intention to change direction or to indicate a hazardous situation. The headlamps and lights at the front end include the following:
- Low-/high-beam headlamps,
- Fog lamps,
- Auxiliary driving lamps,
- Turn-signal lamps,
- Parking lamps,
- Side-marker/clearance lamps (wide vehicles) and
- Daytime running lamps (if required by law in individual countries).

Vehicle rear end
Lights are turned on at the vehicle's rear end in accordance with the weather conditions and indicate the vehicle's position. They also indicate how the vehicle is moving and in which direction, e.g. whether it is traveling unbraked straight ahead, or whether the brakes are applied or the driver is intending to change direction, or whether a hazardous situation exists. The backup lamps illuminate the roadway while the vehicle is backing up/reversing. The lamps/lights at the rear end include the following:
- Stop lamps,
- Tail lamps,
- Fog warning lamps,
- Turn-signal lamps,
- Parking lamps,
- Clearance lamps (wide vehicles),
- Backup lamps and
- License-plate lamps.

Vehicle interior
In the vehicle interior, priority over all other functions is given to the ease and reliability with which the switch elements can be reached and operated and to provide the driver with sufficient information on the vehicle's operating states (while distracting him/her from driving as little as possible). A well-lit instrument panel (P. 783) and discreet lighting of the various functional groups, such as the radio or navigation system, are absolutely essential to ensure relaxed and safe driving. Optical and acoustic signals must be prioritized according to their urgency and then relayed to the driver.

Regulations

Approval codes and symbols
Automotive lighting equipment is governed by national and international design and operating regulations according to which the equipment in question must be manufactured and tested. Each class of illumination device is assigned a specific approval code. The symbol for this code must be legible e.g. on all lenses for headlights and other vehicle lamps. This also applies to approved replacement headlamps and lights.

If a device has one of these approval symbols, this indicates that it has been tested by a technical service body and approved by an official approval authority (in Germany: Federal Motor-Vehicle Agency). All units which are series-produced and assigned an approval symbol must conform in all respects to the approved unit. Examples of approval symbols:

∩∪ K National test symbol D,

(E1) ECE test symbol,

e 1 EU test symbol.

The number 1 following each letter for example indicates that approval has been granted in Germany.

In Europe, installation of all automotive lighting and visual signaling equipment is governed not only by national guidelines but also by the higher European directives (ECE: whole of Europe, EU and Japan). In the course of the ongoing union of Europe, the implementation regulations are being increasingly simplified by the harmonization of directives and legislation.

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Bosch headlamps conform to the applicable ECE and EU directives and can therefore be used in all ECE and EU countries regardless of the country in which they were acquired.

In the USA, lighting equipment is governed by regulations that are very different from those in Europe. The principle of self-certification compels each manufacturer as an importer of lighting equipment to ensure and in an emergency to furnish proof that his products conform 100% to the regulations of FMVSS 108 laid down in the Federal Register. There is therefore no type approval in the USA. The regulations of FMVSS 108 are partly based on the SAE industry standard.

Motor vehicles which for instance are reimported into Europe must be refitted to comply with the European directives.

Photometric terms and definitions

Headlamp range
The distance at which the headlamp beam continues to supply a specified luminous intensity: mostly the 1 lux line at the right side of the road (RH traffic).

Headlamp geometrical range
This is the distance to the horizontal portion of the light/dark cutoff line on the road surface. A low-beam inclination of 1% or 10 cm per 10 meters results in a geometric range equal to 100 times the headlamp's installation height (as measured between the center of the reflector and the road surface).

Visual range
Distance at which an object (vehicle, object etc.) within the luminous distribution field is still visible.

The visual range is influenced by the shape, size and reflectance of the objects, road-surface type, headlamp design and cleanliness, and the physiological condition of the driver's eyes. Because of this large number of influencing factors, it is not possible to quantify this range using precise numerical definitions. Under extremely unfavorable conditions (with RH traffic, on the left side of a wet road surface) the visual range can fall to below 20 m. Under optimal conditions it can extend outward to more than 100 m (with RH traffic, on the right side of the road).

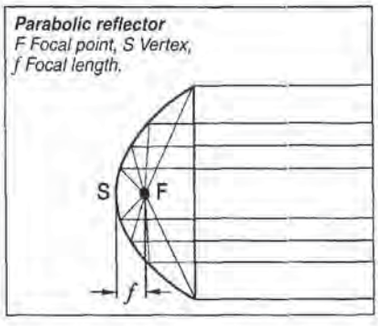
Signal identification distance
Maximum distance at which a visual signal (e.g. fog warning lamp) remains just visible in fog or other inclement weather conditions.

Glare, physiological
This is the quantifiable reduction in visual performance that occurs in response to light sources emitting glare. An example would be the reduction in visual range that occurs as two vehicles approach one another.

Glare, psychological
(discomfort glare)
This condition occurs when a glare source induces discomfort without, however, causing an actual reduction in visual performance. Psychological glare is assessed according to a scale defining different levels of comfort/discomfort.

Reflector focal length
Conventional reflectors for headlamps and other automotive lamps are usually parabolic in shape. The focal length f (distance between the vertex of the parabola and the focal point) is between 15 and 40 mm.

Free-form reflectors
The geometrical configurations of modern reflectors are generated using complex mathematical calculations (HNS, Homogeneous Numerically Calculated Surface). Here, the low mean focal length $\bar{f}$ is defined relative to the distance between



Parabolic reflector
F Focal point, S Vertex, f Focal length.

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the reflector's vertex and the center of the filament. Typical values are between 15 and 25 mm.

In the case of reflectors partitioned with steps or facets (PD2 Partition Design 2), each partition can be created with its own mean focal length *f*.

Reflector illuminated area
Parallel projection of the entire reflector opening onto a transverse plane. The standard reference plane is vertically perpendicular to the vehicle's direction of travel.

Effective luminous flux, headlamp efficiency ratings
The first of the above is that portion of the light source's luminous flux able to supply effective illumination via its reflective or refractive components (for instance, as projected onto the road surface via the headlamp reflector). A reflector with a short (or limited mean) focal length makes efficient use of the incandescent filament because such a reflector extends outward to encompass the bulb, allowing it to convert a large proportion of the luminous flux into a useful beam of light.

Angles of geometric visibility
Those angles, defined relative to the axis of the lighting device, at which the illuminated surface must be visible.

Definitions of device design

Grouped design
A single housing, but with different lenses and bulbs.
Example: Multichamber rear-lamp assemblies containing different individual light units.

Combined design
A single housing and bulb assembly with more than one lens.
Example: Combined rear lamp and license-plate lamp.

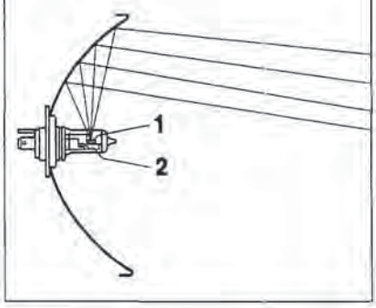
Nested design
Common housing and lens, but with individual bulbs.
Example: Headlamp assembly with built-in side-marker lamp.

Main headlamps, European system

Low beam (dipped beam)
The high traffic density on modern roads severely restricts the use of high-beam headlamps. Under most standard conditions, the low beams are the actual driving lamps. Basic design modifications have allowed substantial improvements in low-beam performance. These include:
- Introduction of the asymmetrical low-beam pattern, characterized (RH traffic) by an extended visual range along the right side of the road,
- Official approval for various types of halogen lamps, making it possible to enhance the luminous intensity at the road surface by 50...80 %.
- Introduction of innovative headlamp systems featuring complex geometrical configurations (PES, HNS, PD2) designed to improve efficiency levels by up to 50 %.
- The "Litronic" gaseous-discharge headlamp (with luminous arc) supplies more than twice the light generated by comparable halogen units.

Low-beam headlamps require a light/dark cutoff line in the light distribution. This is generated in halogen headlamps with H4 bulbs and in Litronic headlamps with a D2R bulb by reflecting the cap (H4) or the light shield (D2R). In headlamps with all-round use (H1, H7, HB4 bulbs), the light/dark cutoff line is created by

***Low beam** (projection pattern, H4 lamp)*
1 Low-beam filament, 2 Shield



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specifically reflecting the filaments.

The "dark above/bright below" distribution pattern resulting from the light/dark cutoff line furnishes acceptable visual ranges under all driving conditions. This configuration holds the glare to which approaching traffic is exposed within reasonable limits while at the same time supplying relatively high levels of illumination in the area below the light/dark cutoff line.

The light distribution pattern must combine maximum visual ranges with minimum glare. These demands are supplemented by other requirements affecting the area directly forward of the vehicle. For instance, the headlamps must provide assistance when cornering, i.e., the light distribution pattern must extend beyond the left and right-side extremities of the road surface.

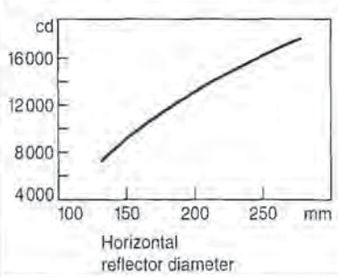
Automotive headlamp performance is subject to technical assessment and verification. Among the requirements are minimum illuminance levels, to ensure adequate road-surface visibility, and maximum intensity levels, to prevent glare (points for measuring illuminance, P. 742).

Headlamp versions

Conventional headlamps

With conventional headlamp systems, the quality of the low beam increases directly in proportion to the size of the reflector. At the same time, the geometrical range increases as a function of installation height. These factors must be balanced against the aerodynamic dictates according to which the vehicle's front-end profile must be kept as low as possible.

Luminous intensity of a headlamp for illuminating right edge of road as function of horizontal reflector diameter



These mutually antagonistic requirements are reconciled by using wider headlamps to accommodate larger reflectors.

Reflectors of a given size, but with different focal lengths, also perform differently. Shorter focal lengths produce wider light beams with better close-range and side illumination. This is of particular advantage during cornering.

Stepped reflectors

Stepped, or graduated reflectors are segmented reflectors consisting of paraboloid and/or parelliptical (combined parabola and ellipse) sections designed to provide various focal lengths. These units retain the advantages of a deep reflector in a shallow unit suitable for compact installation.

Homofocal reflector

The homofocal reflector's supplemental sectors share a single focal point to achieve a shorter focal length than is provided by a basic reflector, thus making a particularly useful contribution to the effective luminous flux. While the light from the supplemental reflectors improves close-range and side illumination, there is no improvement in range.

Multifocal reflector

The principle behind the multifocal reflector is similar to that of its homofocal counterpart. Based on mathematical definitions, the introduction of parelliptical reflector sections, which disperse the light horizontally, produces a large number of focal points.

Variable-focus (stepless) reflectors

Specially designed programs (CAL, Computer Aided Lighting) assist in designing the transitionless VF (Variable Focus Reflector) with its non-parabolic sectors. The focal points of the various reflector zones can change position relative to the light source. This principle can be applied to exploit the entire reflector surface.

Headlamps with clear lenses

Developments in the area of HNS reflector technology (Homogeneous Numerically Calculated Surface) have now made it possible for headlamps to operate at ef-

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Perspective illustration with points for measuring illuminance in accordance with ECE Directive 8 and/or 20

Specifications in lx for headlamp with halogen lamp at distance of 25 meters.

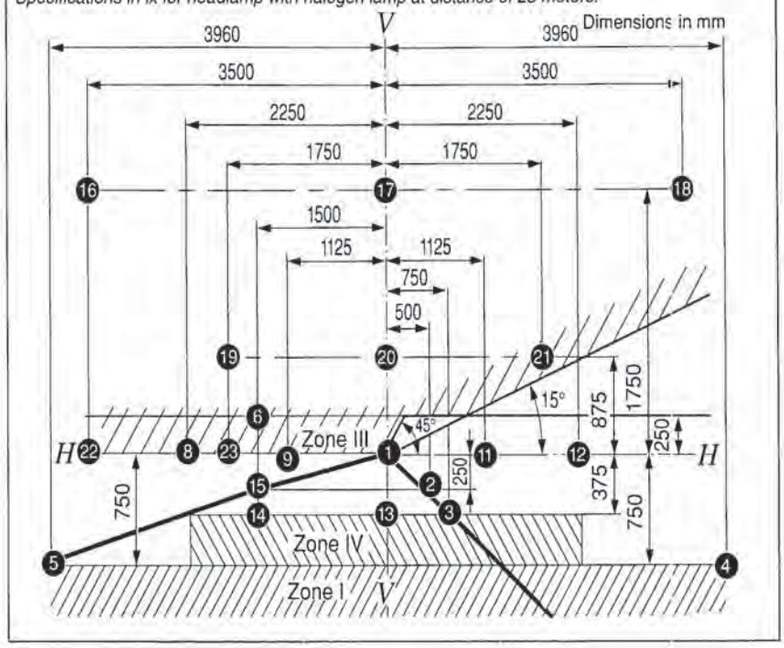


Table 1. Points for measuring low and high beams

Low beam				High beam	
Test points	Specifications in lx	Test points	Specifications in lx	Test points	Specifications in lx
1. E_{HV}	≤ 0.7	14. E_{50L}	≥ 2	7. E_{max}	> 48
2. E_{75R}	≥ 12	15. E_{75L}	≥ 12		< 16 · E_{75R}
3. E_{50R}	≥ 12	16. $E_{8L/4U}$	≥ 0.1; ≤ 0.7		(< 240)
4. E_{25R}	≥ 2	17. $E_{V/4U}$	≥ 0.1; ≤ 0.7	8. $E_{H-5.15°}$	≥ 6
5. E_{25L}	≥ 2	18. $E_{8R/4U}$	≥ 0.1; ≤ 0.7	9. $E_{H-2.55°}$	≥ 24
6. E_{B50L}	≤ 0.4	19. $E_{4L/2U}$	≥ 0.2; ≤ 0.7	10. E_{HV}	≥ 0.8 E_{max}
Max. value in Zone I	≤ E_{50R}	20. $E_{V/2U}$	≥ 0.2; ≤ 0.7	11. $E_{H+2.55°}$	≥ 24
Max. value in Zone III	≤ 0.7	21. $E_{4R/2U}$	≥ 0.2; ≤ 0.7	12. $E_{H+5.15°}$	≥ 6
Min. value in Zone IV	≥ 3	22. $E_{8L/H}$	≥ 0.1; ≤ 0.7		
13. E_{50V}	≥ 6	23. $E_{4L/H}$	≥ 0.2; ≤ 0.7		

Test figures for test lumens of relevant lamp at approx. 12 V.

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ficiency levels of up to 50%. The entire light-distribution pattern is generated on the lens from the reflector surface alone, without additional optical elements. This permits a new, unconventional headlamp design.

Headlamps with facet-type reflectors

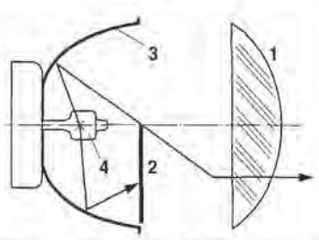
The reflector surface is partitioned with facets. The PD2 program provides the CAL and HNS modules for individual optimization of each individual segment. The important feature of surfaces developed with PD2 is that discontinuity and steps are permitted at all boundary surfaces of the partition. This results in freely shaped reflector surfaces with maximum homogeneity and side illumination.

PES headlamps

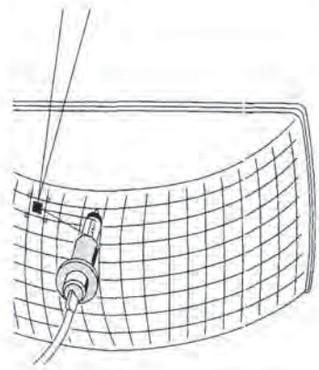
The PES headlamp system (Poly-Ellipsoid-System) employs imaging optics to improve on the technical performance provided by conventional headlamps, making it possible to obtain the light-distribution patterns associated with large-surface conventional lamps from effective projection surfaces as small as 28 cm². This result is obtained using an elliptical (CAL designed) reflector in combination with optical projection technology. A screen reflected with the objective projects precisely defined light/dark cutoffs. Depending upon specific individual requirements, these transitions can be defined as sudden or gradual intensity shifts, making it possible to obtain any desired geometry.

Together with conventional high beams, side-marker lamps and PES fog lamps, PES headlamps can be incorporated in

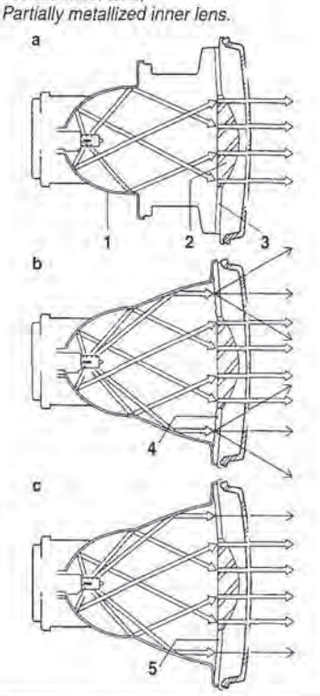
PES reflector *(optical principles)*
1 Objective, 2 Screen, 3 Reflector, 4 Lamp.



HNS or facet-type reflector
Filament pattern reflection with mirror optics.



PES reflectors
(reflection and signal patterns)
a) PES, b) PES with ring parabola and lens,
c) PES with ring parabola and partially metallized tube body.
1 Reflector, 2 Objective, 3 Screen
4 Profiled inner lens,
5 Partially metallized inner lens.



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light-strip assemblies to obtain a unit no higher than approximately 80 mm from top to bottom.

In PES headlamps, the beam path can be configured in such a way that the surroundings of the objective are also used in the signal image. This enlargement of the signal image is used above all with small objective diameters to reduce the psychological glare for the oncoming traffic. The annular reflector uses that portion of the light which is not detected by the PES reflector and directs it forward past the objective. An inner screen, which prevents a direct view of the lamp, can be designed as a lens or a partially metallized screen.

Litronic

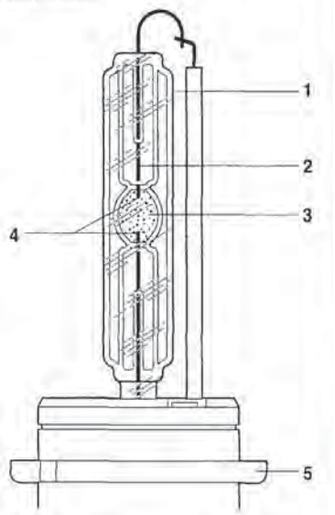
The Litronic (Light-Electronic) headlamp system with a xenon gaseous-discharge lamp as its central element generates high illumination-intensity levels with minimal front-end surface requirements. This makes it ideal for installation in vehicles with aerodynamic front-end styling exhibiting optimum c_w values.

The 35 W "D2S" bulb's arc generates a luminous flux twice as intense as that produced by the H1 bulb. At 4200 K, the color temperature[1]) is higher, which means it contains – similar to sunlight – larger components of green and blue. Maximum illumination, corresponding to approx. 90 lm/W, is available as soon as the quartz element reaches its normal operating temperature of more than 900 °C. Brief high-power operation at currents of up to 2.6 A (continuous operation: approx. 0.4 A) can be used to obtain "immediate light". The life expectancy of 1500 hours corresponds to the operating time to be anticipated during the vehicle's service life. Failure is not sudden, as with filament bulbs. Instead, gradual dimming facilitates early diagnosis and replacement.

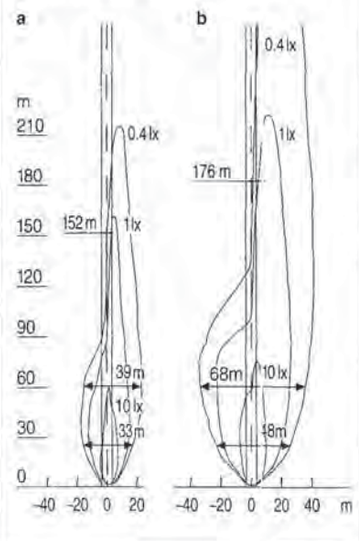
The second generation of these gaseous-discharge lamps for automotive applications features high-voltage-proof sockets and UV glass shielding elements:
– the D2S lamps for PES-design headlamps,

[1]) Temperature of the "black beam" which appears to have the same or a similar color when viewed by an observer.

Gaseous-discharge lamp D2S
1 UV inert-gas bulb, 2 Lead-in insulator, 3 Discharge chamber, 4 Electrodes, 5 Lamp holder.

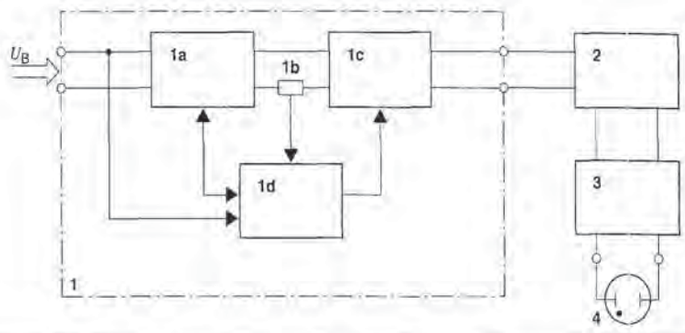


Light-distribution patterns *(road traffic)*
a) PES halogen lamp,
b) Litronic PES D2S lamp.

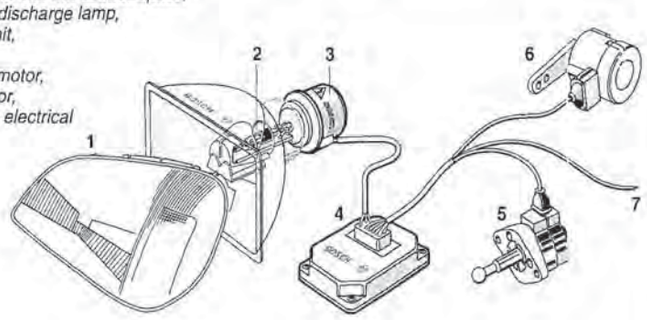


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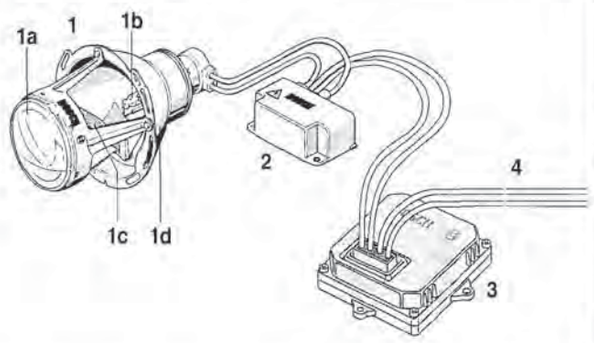
Electronic ballast unit (EVG) for 400 Hz alternating current and pulse ignition of the bulb
1 ECU (1a DC/DC converter, 1b Shunt, 1c DC/AC Converter, 1d Microprocessor),
2 Ignition unit, 3 Lamp socket, 4 D2S lamp, U_B Battery voltage.



System components for reflection-type headlamp with integrated dynamic headlight vertical-aim control
1 Lens with or without scatter optics,
2 Gaseous-discharge lamp,
3 Ignition unit,
4 ECU,
5 Stepping motor,
6 Axis sensor,
7 To vehicle electrical system.

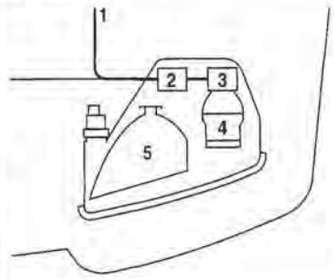


System components for PES-design headlamps
1 Headlamp
(1a Lens,
1b D2S lamp,
1c Screen,
1d Reflector),
2 Ignition unit,
3 ECU,
4 To vehicle electrical system.

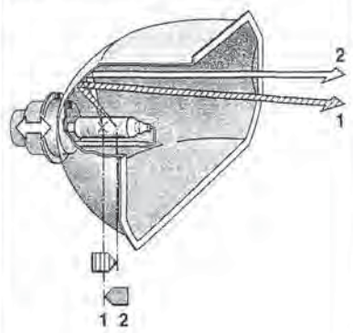


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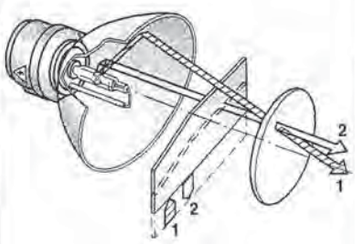
Quad-headlamp system with Litronic
1 Vehicle electrical system, 2 ECU, 3 Ignition unit with lamp connection, 4 Headlamp optics with gaseous-discharge lamp, 5 Halogen high beam.



Bi-Litronic "Reflection"
1 Low beam, 2 High beam.



Bi-Litronic "Projection"
1 Low beam, 2 High beam.



– D2R lamps for reflection headlamps; an integral light shield, comparable to the bulb cover used for H4 low beams, generates the light/dark cutoff line.

An integral part of the headlamp is the electronic ballast unit (EVG) responsible for activating and monitoring the lamp. Its functions include:
– Ignition of the gaseous discharge (voltage 10...20 kV)
– Regulated power supply in the cold lamp's warm-up phase and
– Demand-oriented supply in continuous operation.

The system furnishes largely consistent levels of illumination by compensating for fluctuations in the vehicle's electrical system. Should the lamp go out (for instance, due to a momentary lapse in the voltage supply), reignition is spontaneous and automatic.

The electronic ballast unit responds to defects (such as a damaged lamp) by interrupting the power supply to help avoid injury in the event of contact.

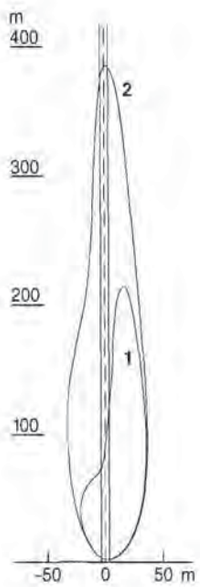
Litronic's initial area of application has been as a high-performance low-beam unit in quad headlamp assemblies, where it is designed to operate simultaneously with specially designed halogen high-beam units. This has made it possible to achieve substantially wider road-illumination patterns for lighting the edges of curves and of wide roads as effectively as a halogen unit illuminates straight stretches of road. The driver enjoys substantial improvements in both visibility and orientation in difficult operating conditions and inclement weather. Compliance with ECE-R regulations is achieved by equipping Litronic headlamps with vertical aim control and washer systems to consistently ensure optimal utilization of the extended range while at the same time maintaining excellent beam projection.

Bi-Litronic "Reflection"

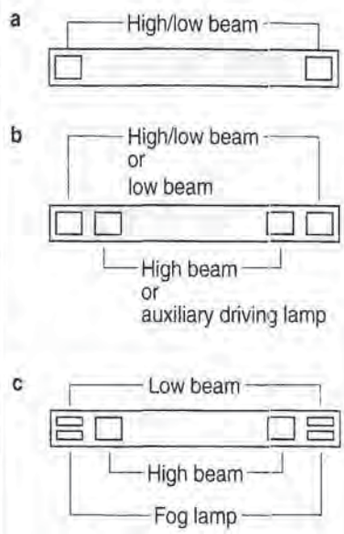
Bi-Litronic is a special system. It permits both the low and high beams to be generated by a dual-headlamp system using only one gaseous-discharge lamp. For this purpose, when the high-/low-beam switch is operated, an electromechanical

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Light-distribution patterns of Bi-Litronic
1 Low beam, 2 High beam.



Headlamp systems
a) Dual headlamps,
b) Quad headlamps,
c) Six headlamps.



actuator moves the gaseous-discharge lamp in the reflector into the appropriate position for defining the escape of the light cone for the main and low beams. The essential advantages of Bi-Litronic "Reflection" are:

- Xenon light for high-beam operation,
- Visual guidance through continuous shifting of light distribution from close to long range,
- Clear reduction of required space compared with four-compartment systems,
- Lower cost through use of only one lamp and a ballast unit.

Bi-Litronic "Projection"

Bi-Litronic "Projection" system is based on a PES Litronic headlamp. Xenon light is made available for high-beam operation by shifting a light shield for the light/dark cutoff. With lens diameters of 60 and 70 mm, Bi-Litronic "Projection" enables the presently most compact headlamp design with combined high/low beam and at the same time outstanding luminous efficiency.

High beam

The high beam is usually generated by a light source located at the focal point of the reflector (see Fig. on P. 739). This causes the light to be reflected outward in the direction of the reflector's axis. The intensity of the high beam projected along the axial plane is largely a function of the reflector's luminous surface area. In four and six-headlamp systems, in particular, purely paraboloid high-beam reflectors can be replaced by units with complex geometrical configurations designed to supply a "superimposed" high-beam pattern.

The calculations employed to design these units seek to achieve a high-beam distribution that harmonizes with the low-beam pattern (simultaneous activation). The pure high beam is virtually "superimposed" on the low-beam projection to do away with the annoying overlap normally

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found in the transitional area at the front of the light-distribution pattern.

Designs
Global regulations mandate two headlamps for low beam and at least two (or the option of four) high-beam units for all dual-track vehicles.

Dual headlamp system
This design utilizes bulbs comprising two separate light sources (Bilux, Duplo) to project both low and high beams via a single reflector.

Quad headlamp system
Two of the headlamps provide both high and low beam or low beam only, while the second pair only provides high-beam illumination.

Six-headlamp system
This configuration differs from the quad design by incorporating a supplementary fog lamp in the main headlamp.

Component groups:

Lens (adjustable in bodywork)
The lens and reflector are combined in a single headlamp unit. The beam is adjusted by pivoting the complete assembly. Under unfavorable circumstances, this can lead to the lens being at a slight angle relative to the car's bodywork. The headlamp assemblies are generally equipped with seals in the area adjacent to the bulb, and also feature special ventilation systems.

Lens (fixed to the bodywork)
There is no connection between the lens and the reflector which is mounted in a housing designed to move relative to the lens for adjustment (housing type). Because the reflector has no connection to the lens it remains stationary and can be fully integrated into the vehicle's bodywork. The complete headlamp is sealed or provided with ventilation elements.

Components

Reflector
Reflectors are made of sheet metal or plastic.
Sheet-metal reflectors are manufactured as follows:
- Deep-draw processing to obtain a paraboloid or more complex geometrical configuration,
- Galvanization or powder-coating to protect against corrosion,
- Paint application to obtain a smooth surface,
- Evaporative application of aluminum to form the reflective layer, and
- Evaporative application of a special protective layer onto the aluminum substrate.

This process hermetically seals the sheet metal while providing an extremely smooth reflective surface featuring a maximum residual ripple of 1/10,000 mm.

Plastic reflectors are produced by injection molding (duroplastic, thermoplastic); no special anti-corrosion treatment is required. Plastic can be shaped with much greater precision and can also be used to manufacture stepped reflectors.

Lens
A large proportion of lenses is manufactured using high-purity glass (free of bubbles and streaks). During the lens molding process high priority is assigned to surface quality in order to prevent undesirable upward light refraction in the final product – this would tend to blind oncoming traffic. The type and configuration of the lens prisms depend upon reflector size and focal depth as well as the desired light-distribution pattern. With the improvement in calculation methods, it is also possible to develop headlamps where complete light distribution is determined by the reflector. This eliminates all the diffuser elements on the lens, thus producing a "clear lens". The first plastic lens entered series production in 1992. Aside from reduced weight, plastic lenses provide other advantages for automotive applications including greater latitude in headlamp and vehicle design.

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Main headlamps, American system

Low beam

The demands placed on light distribution in American motoring differ from those in Europe. Since 1.5.1997, however, headlamps with light/dark cutoffs have also been authorized in the USA. These though must be visually adjusted. It is thus now possible to develop headlamps which conform to both legal requirements.

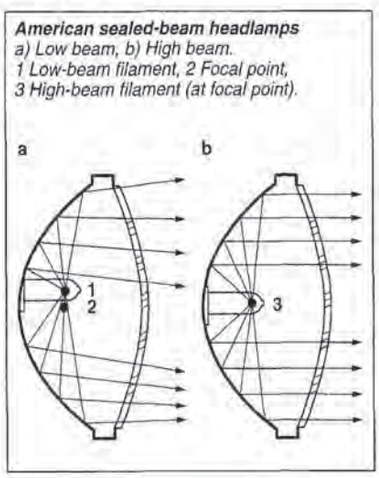
High beam

As with the European system, the light source is usually located at the focal point of the parabolic reflector. Dual, quad and six-headlamp configurations are employed.

Designs

Sealed beam

In this design which is no longer used, the aluminized glass reflector must be sealed gas-tight with the lens on account of the unencapsulated light sources. The whole unit is sealed and filled with an inert gas. Should a filament burn through, the entire unit must be replaced. Units with halogen light sources are also available. The limited range of available sealed-beam headlamps on the market severely restricted the designers' latitude for front-end styling variations.



American sealed-beam headlamps
a) Low beam, b) High beam.
1 Low-beam filament, 2 Focal point,
3 High-beam filament (at focal point).

Replaceable Bulb Headlamp (RBH)

European developments based on replaceable bulb technology also had an effect on the American system from 1983 onwards. The headlamps' size and shape can be adapted for improved vehicle design (styling considerations). The RBH design usually features plastic reflectors and lenses.

Vehicle Headlamp Aiming Device (VHAD)

This design involves RBH headlamps which are mechanically adjusted vertically with the aid of a level gauge integrated in each headlamp and horizontally by means of a system comprising needle and dial: This is in fact "on-board aiming".

Headlamp systems

As in Europe, dual-, quad- and six-headlamp configurations are used in North America.

Main headlamps, European regulations

Regulations and directives for main headlamps

76/761/EEC and ECE-R1 and R2:
Headlamps for high and low beam including their bulbs.
ECE-R8: Headlamps with H1, H2, H3, H7, H8, H9, H11, HIR1, HIR2, HB3 and HB4 bulbs.
ECE-R20: Headlamps with H4 bulbs.
StVZO §50: Headlamps for high and low beam.
76/756/EEC and ECE-R48: for installation and application.
ECE-R 98/99: Headlamps with gaseous-discharge lamps.

Low beam, installation

Regulations prescribe 2 white-light low-beam headlamps for multiple-track vehicles.

Low beam, illumination technology

The relevant ordinances governing symmetrical low-beam illumination in Germany are contained in the Technical Requirements (TA) of the StVZO (FMVSS/

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CUR; Vehicle Homologation and Licensing Regulations). The only applicable regulations for asymmetrical low beams are international regulations and guidelines containing precise stipulations governing photometric testing of the different low-beam units (with standard or halogen bulb).

Homologation testing is carried out using test lamps manufactured to more precise tolerances than those installed in production vehicles.

Headlamp glare is evaluated based on StVZO §50 (6). Glare is considered to be eliminated when the illuminance at a height equal to that of the center of the headlamp does not exceed 1 lx at a distance of 25 m. This test is carried out with the engine running at moderate speed.

Low beam, switching

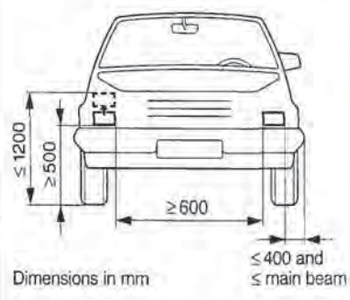
All high-beam lamps must extinguish simultaneously when the low beams are switched on. Dimming (gradual deactivation) is permitted, with a maximum dimming period of 5 seconds. A 2-second response delay is required to prevent the dimming feature from activating when the high-beam flashers are used. When the high beams are switched on, the low-beam units may continue to operate (simultaneous operation). H4 bulbs are generally designed for limited use with both filaments in operation.

High beams, installation

A minimum of two and a maximum of four headlamps are prescribed for the high-beam mode.

Prescribed instrument-cluster high-beam indicator lamp: blue or yellow.

***European headlamp system** (low beam)*



Dimensions in mm

High beams, illumination technology

High-beam light distribution is defined in the regulations and guidelines together with the stipulations governing the low beams.

The most important specifications are: symmetrical distribution relative to the central vertical plane, maximum light along the headlamp's center axis.

The maximum approved luminous intensity, a composite of the intensity ratings for all high-beam headlamps installed on the vehicle, is 225,000 cd. The respective ratings are indicated by reference codes located adjacent to the homologation code on each headlamp. 225,000 cd corresponds to the number 75. The luminous intensity of the high beam could be indicated, e.g., by the number 20 stamped next to the round ECE test symbol.

If these are the only headlamps on the vehicle (no auxiliary driving lamps) then the composite luminous intensity is to be in the range of 40/75 of 225,000 cd, viz., 120,000 cd.

High- and low-beam adjustments

Conditions for aiming

- Tire pressures must conform to specifications.
- Vehicle loaded, in accordance with model type:

For passenger cars: one person or 75 kg on the driver's seat.
For trucks: unladen;
For single-track vehicles and single-axle tractors: one person or 75 kg on the driver's seat.

- The vehicle should be rolled several meters to allow the suspension to adapt to the added load.
- The vehicle must be parked on a flat, level surface.

If adjustment is carried out without a headlamp aiming device, a test surface must be set up at a distance of 10 meters from the vehicle so that the center mark is located in the direction of travel in front of the headlamp to be adjusted (see Figs.).

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– The headlamps must be adjusted individually, with all other headlamps covered.
– If the vehicle is equipped with a manual headlamp range adjuster, the switch should be set to the position prescribed by the manufacturer.

Table 2. Setting dimension "e" for headlamp adjustment, Europe[1])

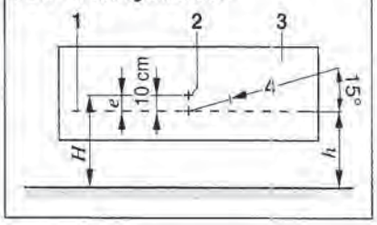
Vehicle type	Headlamp cm	Fog lamp cm
1 Motor vehicles on which the upper edge of the reflector surface is not more than 140 cm above the standing surface.		
1a Pass. cars (also station wagons/estates)	12	20
1b Vehicles with self-leveling suspension or automatic beam adjusters.[2]) 1c Multi-axle tractors and special-purpose machinery. 1d Single-track vehicles. 1e Trucks with forward loading area.	10	20
1f Trucks with rear loading area.[3]) 1g Semitrailer tractors.[3]) 1h Buses.[3])	30	40
2 Motor vehicles on which the upper edge of the reflector surface is not more than 140 cm above the standing surface.[4])	H/3	H/3)+7
3 Single-axle tractors or special-purpose machinery with permanently dimmed headlamps on which the required light beam center inclination is indicated.[5])	2 N	20
4 Vehicle with homologation in acc. with Directive 76/756/EEC or ECE R48	Reference dimension on vehicle	see above

[1]) The results obtained with the aid of headlamp aiming equipment must correspond to the figures in the table.
[2]) Special attention must be paid to compliance with the manufacturer's instructions.
[3]) Except motor vehicles as defined in 1b.
[4]) H Height of center of headlamp above support surface in cm.
[5]) N Specified figure for inclination of beam center at distance of 5 m.

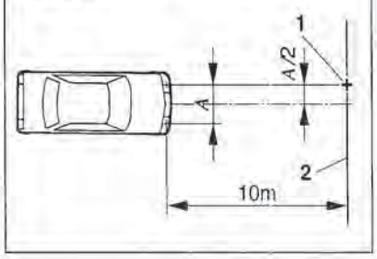
Adjustment information:
The adjustment marks and lines are arranged to accommodate main headlamps at a normal installation height, with the center mark set to height H, corresponding to the middle of the headlamp. The adjustment dimension e = 10 cm represents the vertical distance between the center mark and the boundary line.

On headlamps with e > 10 cm the boundary line is lowered to the required level. Obviously, the center mark will then no longer be level with the center of the headlamp. It can, however, be used to check the position of the high beam (where present).

Test surface for headlamp beam
1 Boundary line, 2 Center mark,
3 Test surface, 4 Break point.
H Height of headlamp center above standing surface in cm.
h Height of test-surface boundary line above standing surface in cm.
$e = H - h$ Setting dimension.



Relative positions of test surface and vehicle's longitudinal axis
1 Center mark
2 Test surface
A Distance between headlamp centers.



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Table 3. Geometric range of the horizontal component of the low beam's light/dark cutoff line. Headlamp installed at height of 65 cm

Inclination of light/dark cutoff line in % (1% = 10 cm/10 m)	1%	1.5%	2%	2.5%	3%
Reference dimension *e*	10 cm	15 cm	20 cm	25 cm	30 cm
Geometric range in m for the horizontal component of the light/dark cutoff line	65 m	43.3 m	32.5 m	26 m	21.7 m

The left-side (horizontal) section of the light/dark cutoff line must be adjusted to the limit line.
Refer to Table 2 for dimension *e*.
The headlamps must be readjusted following any operations that could affect their height (suspension work, etc.). The procedure is also recommended after bulb replacement.

If the high beam is incorporated in a single unit along with the asymmetrical low beam (Bilux and Duplo bulbs) then it will be aimed automatically when the low beam is adjusted.

Separate individual high-beam units are aimed horizontally and symmetrically using the center of the headlamp and the center mark as references.

Headlamp aiming devices

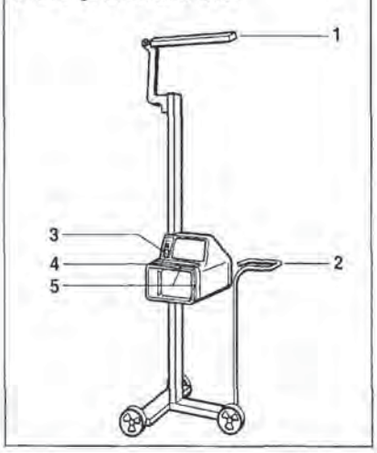
Function

Correct adjustment of motor-vehicle headlamps should ensure the best possible illumination of the roadway by the low beam while at the same time minimizing dazzle for oncoming traffic. For this purpose, the inclination of the headlamp beams with respect to a level base surface, and their direction to the vertical longitudinal center plane of the vehicle, must satisfy the official requirements.

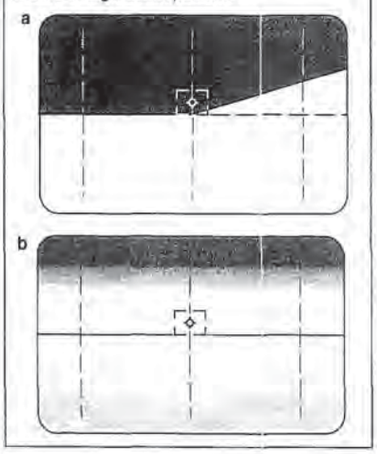
Equipment design

Headlamp aiming devices are portable imaging chambers which comprise a single lens and a collector screen located in the focal plane of the lens and rigidly connected to it. The collector screen has

Headlamp test and adjustment device
1 Orientation mirror, 2 Handle, 3 Photometer, 4 Refraction mirror, 5 Markings for center of lens.



Viewing window in aiming device
a) Boundary line for light/dark cutoff on asymmetrical low beam, b) Center mark for middle of high-beam pattern.



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markings to facilitate correct headlamp adjustment, and can be viewed by the equipment operator using suitable appendages such as windows and adjustable refraction mirrors.

The prescribed headlamp adjustment, i.e. the inclination relative to the center line of the headlamp in cm at a fixed distance of 10 m, is set by turning a knob to move the collector screen.

The aiming device is aligned with the vehicle axis using a sighting device such as a mirror with an orientation line. It is turned and aligned so that so that the orientation line uniformly touches two external vehicle-reference marks. The imaging chamber can be moved vertically and clamped at the level of the vehicle headlamp.

Headlamp testing

The headlamp can be tested once the equipment has been correctly positioned at the front of the lens. An image of the light distribution pattern as emitted by the headlamp appears on the collector screen. Some test devices are also equipped with photodiodes and a display to allow measurement of luminous intensity.

On headlamps with asymmetrical low-beam patterns the light/dark cutoff line should contact the marked limits; the point of intersection for the horizontal and vertical components must lie on the center mark of the vertical line. Following adjustment of the low beam's light/dark cutoff line in accordance with the regulations, the center of the high beam (assuming that high and low beam are adjusted together) should be within the corner limits indicated around the center mark.

Main headlamps, North American regulations

Regulations and guidelines

Federal Motor Vehicle Safety Standard (FMVSS) No. 108 and SAE Ground Vehicle Lighting Standards Manual (Standards and Recommended Practices).

The regulations governing installation and control circuits for headlamps are comparable to those in Europe. Since 1. 5. 1997, headlamps with light/dark cutoffs have likewise been authorized in the USA.

The main differences were previously in the headlamp systems. Until 1983 the available sealed-beam types in the USA were restricted to the following dimensions:

Dual-headlamp systems:
- 178 mm diameter (round),
- 200 x 142 mm (rectangular).

Quad-headlamp systems:
- 146 mm diameter (round),
- 165 x 100 mm (rectangular).

Since 1983 a supplement to FMVSS 108, has made it possible to use RBH (Replaceable Bulb Headlamps) headlamp inserts of various shapes and sizes.

Headlamps conforming to ECE regulations or EEC directives are not approved for use on dual-track vehicles in the USA, even if they satisfy the American requirements relating to lighting and aimability (visual). They are however approved for use without restriction on motorcycles.

Conditions for aiming

In Europe, headlamps are always adjusted based on visual assessment of the light beam, but the use of mechanical aiming devices has become the most common method of aiming headlamps in the USA. The headlamp units are equipped with three pads on the lens – one for each of the three adjustment planes. A calibrating unit is placed against these pads. The calibration is carried out with spirit levels.

The VHAD (Vehicle Headlamp Aiming Device) method is employed to adjust the headlamps relative to a fixed reference axis defined by the vehicle. This procedure is carried out using a spirit level firmly attached to the headlamp. The three lens pads are no longer required.

Since mid-1997, however, visual (only vertical) aiming authorizec as from 1. 5. 1997 has also gained increasing acceptance in the USA. There is no horizontal aiming here.

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Headlight leveling control (vertical-aim control)

Table 3 indicates the geometrical ranges at various inclination angles for headlamps installed at a height of 65 cm. The inspection tolerance extends to include inclination angles of up to 2.5% (1.5% below standard setting). According to European Union regulations, the basic setting based on dimension *e* is (10...15 cm)/10 m with the weight of one person on the driver's seat. The specifications for this setting are provided by the vehicle manufacturer.

Since 1.1.1998, an automatic or manually-operated headlight leveling control (range adjustment) device has been mandatory in Germany for all vehicles registered for the first time, except in those cases where other equipment (e. g., hydraulic suspension control) ensures that the light beam's inclination will remain within the prescribed tolerances. Although this equipment is not mandatory in other countries, its use is permitted.

Automatic leveling
must be designed to compensate for load variations by lowering or raising the low beam's projection by between 5 cm/10 m (0.5%) and 25 cm/10 m (2.5%).

Manual leveling
This device is operated from the driver's seat and must incorporate a detent at the base setting; beam adjustment is also performed at this position. Units with infinitely variable and graduated control, must both feature visible markings in the vicinity of the hand switch which correspond to the vehicle load conditions necessitating vertical-aim adjustment. All design variations employ an adjustment mechanism to provide vertical adjustment of either the headlamp reflector (housing design) or the entire headlamp unit.

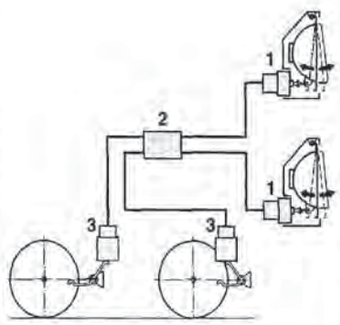
Manually operated units employ a switch at the driver's seat to control the setting, while automatic units rely on level sensors on the vehicle axles to monitor suspension compression and relay the corresponding signals to the adjustment actuators.

Hydromechanical systems
This type of system operates by transmitting a fluid through the connecting hoses between the manual switch (or level sensor) and the adjustment elements. The degree of adjustment corresponds to the quantity of pumped fluid.

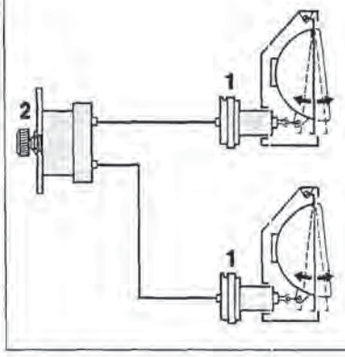
Vacuum systems
The manual switch (or level sensor) in this type of system modulates vacuum from the intake manifold and transmits it to the adjustment actuators to achieve varying degrees of adjustment.

Electrical systems
These employ electric gear motors as ad-

Automatic vertical-aim control *(principle)*
1 Adjustment actuator, 2 Processing unit, 3 Level sensors.



Manual vertical-aim control *(principle)*
1 Adjustment actuator, 2 Switch.



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justment actuators. These are switched either by the relevant switches in the vehicle or by axle sensors.

Headlamp cleaning systems

Headlamp cleaning systems remove dirt from the lenses of the main headlamps. This ensures unimpaired illumination of the roadway and prevents oncoming traffic from being dazzled. There are two types of cleaning system:

Wipe/wash system

The use of these systems is restricted to glass lenses because plastic lenses, despite having high scratch-resistant coatings, are too sensitive to mechanical cleaning.

High-pressure washer system

High-pressure washer systems (see Fig.) are gaining increasing acceptance as they can be used on both glass and plastic lenses. The cleaning effect is chiefly determined by the cleaning pulse of the water droplets. The decisive factors are:
- the distance between jet nozzle(s) and lens,
- the size, contact angle and contact velocity of the water droplets and
- the amount of water.

As well as fixed nozzle holders on the bumpers, telescopically extending nozzle holders are also used. The telescope significantly improves the cleaning effect because it always assumes an optimal spray position. Furthermore, when not in use, the nozzle holder can be concealed e.g. inside the bumper.

High-pressure washer systems consist of
- water reservoir, pump, hose and non-return valve and
- nozzle holder, which is extended by means of a telescope, with one or more jet nozzles.

Fog lamps

Fog lamps are intended to improve road-surface illumination in fog, snow, heavy rain and dust.

Optical concept

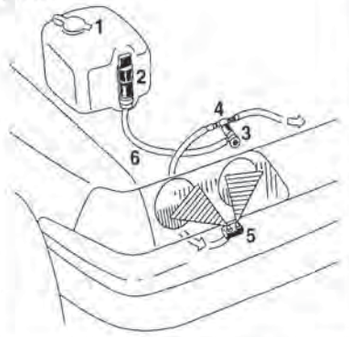
Paraboloid

A parabolic reflector featuring a light source located at the focal point reflects light along a parallel axis (as with the high-beam lamp) and the lens extends this beam to form a horizontal band. A special screen prevents the beam from being projected upward.

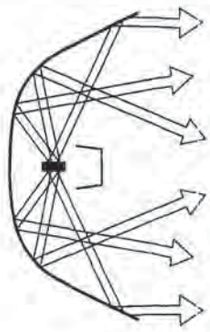
Free-form technology

New calculation methods (CAL Computer Aided Lighting) can be used to shape reflectors in such a way that they scatter

High-pressure washer system for headlamps
1 Water reservoir, 2 Pump, 3 Non-return valve, 4 T-connection, 5 Nozzle holder, 6 Hose.



Fog lamp with free-form reflector



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light directly (i.e. without optical lens profiling) and simultaneously generate (without separate shading) a sharp light/dark cutoff line.

The fact that the lamp features pronounced envelopment of the bulb leads to an extremely high volume of light combined with maximum dispersion width.

PES fog lamps

In fog, this technology minimizes the glare reflected back to the driver. The screen, the image of which is projected onto the road surface by the lens, furnishes maximum contrast at the light/dark cutoff line.

Designs

DIY and dealer-installed fog lamps are designed as individual projection units in their own housings. These are installed either upright on the bumper, or suspended beneath it.

Stylistic and aerodynamic considerations have led to increased use of integrated fog lamps, designed either for installation within body openings or included as a component within a larger light assembly (with adjustable reflectors when the fog lamps are combined with the main headlamps).

Most fog lamps are designed to project a white beam. There is no substantive evidence that yellow lamps provide any physiological benefits. A fog lamp's effectiveness depends upon the size of the illuminated area and the focal length of the reflector. Assuming the same illuminated area and focal length, from the lighting-engineering viewpoint the differences between round and rectangular fog lamps are insignificant.

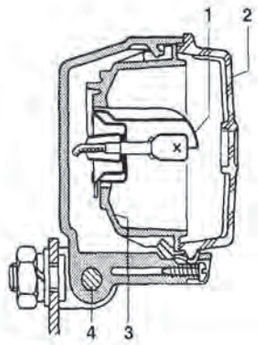
Regulations

Design is governed by ECE R19, installation by EEC 76/756, ECE R48, StVZO (FMVSS/CUR) § 52. Two white- or yellow-light fog lamps are permitted. The control circuit for switching the fog lamps must be independent of that used for high and low beams. The (German) StVZO allows fog-lamp installation in positions more than 400 mm from the widest point of the vehicle's periphery provided that the switching circuit is designed to ensure that they operate only in conjunction with the low beams. The adjustment procedures for fog lamps are basically the same as for headlamps. Adjustment dimensions *e* are provided in Table 2.

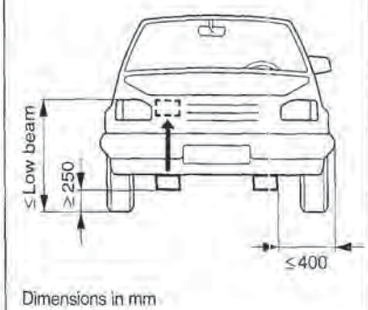
Auxiliary driving lamps

Auxiliary driving lamps enhance the high-beam visibility provided by dual, quad and six-headlamp systems. The basic optical principles are similar to those used for fog lamps. The only difference being the lens, which is specifically designed to furnish an extended beam. Auxiliary driving lamps are often identical to fog lamps in

Fog lamp *(parabolic reflector, upright installation)*
1 Mask, 2 Lens, 3 Reflector, 4 Vertical adjustment shaft.



Fog lamps *(positioning)*



Dimensions in mm

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shape and size.

Auxiliary driving lamps are mounted and aimed in the same way as standard headlamps, and the underlying lighting concepts are the same. Auxiliary driving lamps are also subject to the cited regulations governing maximum luminous intensity in vehicular lighting systems, according to which the sum of all reference numbers is not to exceed 75. For older lamps without approval number, the number 10 is used for general assessment purposes.

Lights and lamps

Design specifications are contained in ECE R6, R7 and R38, installation is governed by EEC 76/756, ECE R48 and StVZO § 49a (FMVSS/CUR).

These types of lamp are intended to facilitate recognition of the vehicle and to alert other road users to any intended or current changes in direction or motion. Either of two basic concepts can be employed to satisfy the corresponding technical requirements.

Lamps with reflector optics

A reflector in one of various forms (often parabolic) redirects the light emitted by the bulb to obtain a roughly axial projection angle. A lens with optical dispersion elements then determines the beam pattern.

Lamps with Fresnel optics

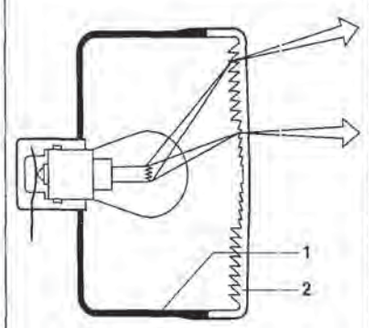
The light is projected directly against the lens, with no refraction at a reflector. A lens utilizing Fresnel technology then directs the beam in the desired direction.

Fresnel optics devices are usually less efficient than reflector-optics lamps.

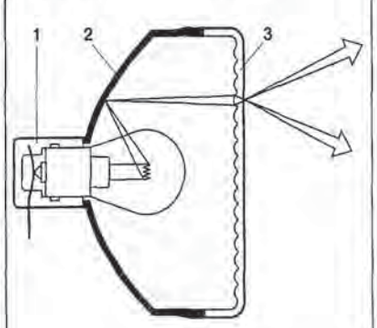
Lamps combining reflector and Fresnel optics

Designs combining both the above principles have been used with success. The GP reflector (rotated parabola) is employed to derive undiminished luminous flux from more compact units featuring

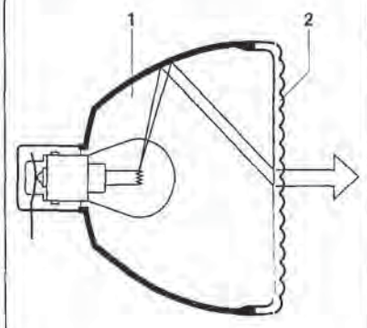
Lamp with Fresnel optics *(principle)*
1 Housing, 2 Lens with Fresnel optics.



Lamp with reflector optics *(principle)*
1 Housing, 2 Reflector,
3 Lens with dispersion elements.

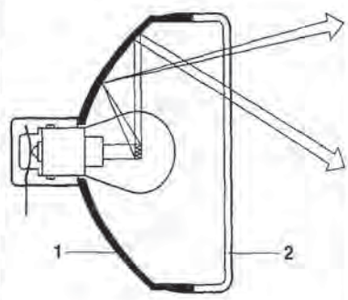


Reflector with rotated parabola
1 GP reflector, 2 Fresnel lens.

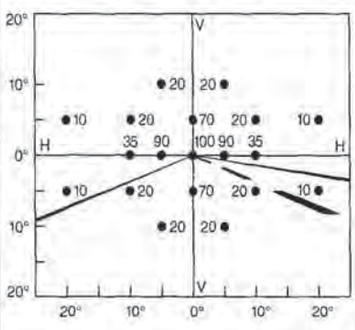


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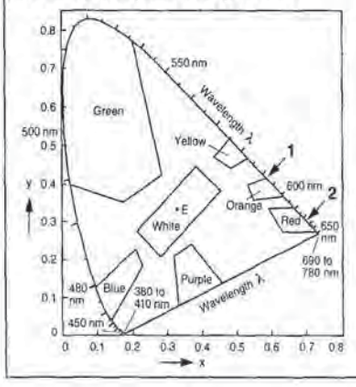
Free-form lamp with clear lens
1 Reflector, 2 Clear lens.



ECE measuring screen "lights and lamps" *(values in %)*



Standardized color scale *(color location)*
1, 2 Ranges for yellow/orange and red.



both shallower reflectors and smaller lenses. These designs employ a specially shaped reflector (rotated parabola) to capture the light beam emanating from the bulb at the largest possible peripheral angle. The Fresnel optics then homogenize the light beam for projection along the desired path. A free-form lamp with Fresnel cap combines excellent luminous efficiency with a variety of stylistic conversion options. The light from the lamp is deflected by the reflector. In the process, the required expansion of light distribution is generated completely or partially by the reflector. The outer lens can thus be designed as a clear lens or supplemented with cylindrical lenses in the horizontal or vertical direction. The Fresnel lens improves the luminous efficiency by deflecting in the desired direction a further part of the light which would not otherwise contribute to the function of the lamp. Both versions are mainly used in the front turn-signal lamps. Which version is actually used depends on the body shape and thus on the available structural space, on the stylistic requirements and on the luminous values required.

Illumination technology, luminous intensities

As projected along the reference axis, minimum and maximum luminous intensities for all lamps must remain within a range calculated to ensure signal recognition without, however, inducing annoying glare for other road users.

Luminous intensity levels to the sides, and above and below the reference axis, may be lower than along the axis itself. This deviation has been quantified on a percentual basis ("unified spatial light distribution").

Color filters

Depending on their application (e.g. stop, turn-signal, fog warning lamps), motor-vehicle lamps must display uniform, distinctive colors in the red or yellow color range. These colors are defined in specific ranges of a standardized color scale (color location). Since white light is composed of various colors, filters can be used to weaken or filter out completely the radiation of undesired spectral regions

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(colors). The color-filter functions may be performed by either the tinted lenses of the lamp or a colored coating on the glass bulb of the lamp (e.g. yellow lamp in turn-signal lamps with clear lens).

Filter technology can also be used to adapt lamp lenses in such a way that when the lamp is switched off the color is matched to the vehicle's paintwork and nevertheless the existing homologation regulations are complied with when the lamp is on. In the EU region, color locations have been laid down which e.g. correspond to a wavelength of 592 nm for turn-signal lamps that are colored "Yellow/Orange" and to a wavelength of 625 nm for stop and tail lamps that are colored "Red" (see Fig.).

Turn-signal indicators

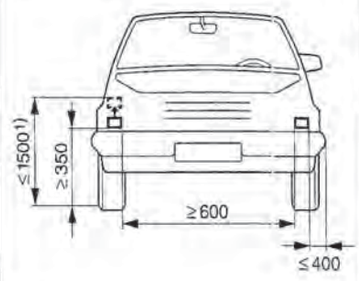
EEC 76/759, ECE R6, StVZO § 54 (FMVSS/CUR).
Group 1 (front-mounted), Group 2 (rear-mounted) and Group 5 (side) turn-signal lamps are required on dual-track vehicles. Group 2 turn-signal lamps are sufficient for motorcycles and mopeds. The instrument cluster indicator lamp may be in any desired color. The flash frequency is defined as 90 ± 30 cycles per minute.

Front turn-signal lamps
Two yellow turn-signal lamps are stipulated. An indicator is required inside the vehicle.

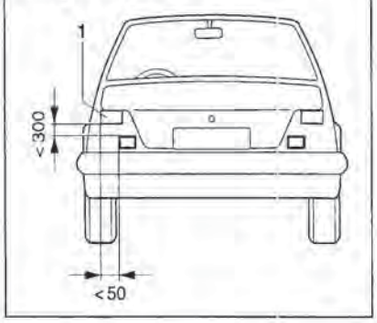
Rear turn-signal lamps
Two yellow turn-signal lamps are stipulated.

Side-mounted turn-signal lamps
Two yellow turn-signal lamps are stipulated.

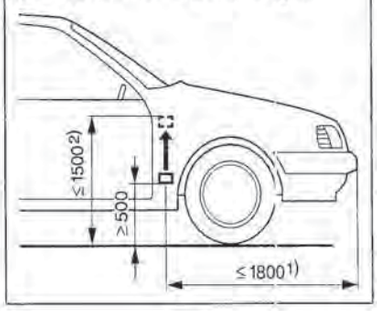
Front turn-signal lamps
(position, dimensions in mm)
1) less then 2100 mm if vehicle body type prohibits compliance with regulations on maximum height.



Rear turn-signal lamps
(position, dimensions in mm)
1 Tail lamp. Height and width as for front turn-signal lamps.



Side-mounted turn-signal lamps
(position, dimensions in mm)
1) or 2500 mm
2) or 2300 mm
if vehicle body type prohibits compliance with regulations on maximum height.



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Hazard-warning and turn-signal devices

Regulations

Both the StVZO (FMVSS/CUR) and EU Directive EEC 76/756 stipulate that the standard lighting will be supplemented by visual signaling devices for indicating turns and for use as hazard-warning flashers on all vehicles with a top speed exceeding 25 km/h.

Flashing signals: Signals with a frequency of 60...120 pulses per minute and a relative illumination period of 30...80 %. Light must be emitted within 1.5 s of initial activation. Should one lamp fail, the remaining lamps must continue to generate visible light.

Turn-signal flashing: Synchronous signal generated by all turn-signal lamps on one side of the vehicle. The lamps are electrically monitored. Malfunctions are indicated.

Hazard-warning flashing: Synchronous flashing of all turn-signal lamps, also operational with the ignition off. An operation indicator is mandatory.

Hazard-warning and turn-signal devices for vehicles without trailer

The electronic hazard-warning and turn-signal flasher includes a pulse generator, designed to switch on the lamps via relay, and a current-controlled monitoring circuit to modify the flashing frequency in response to bulb failure. The turn-signal stalk controls the turn signals, while the hazard flashers are operated using a separate switch.

Hazard-warning and turn-signal devices for vehicles with/without trailers

This type of hazard-warning and turn-signal flasher differs from that employed on vehicles without trailers in the way that turn-signal operation is monitored.

Single-circuit monitoring

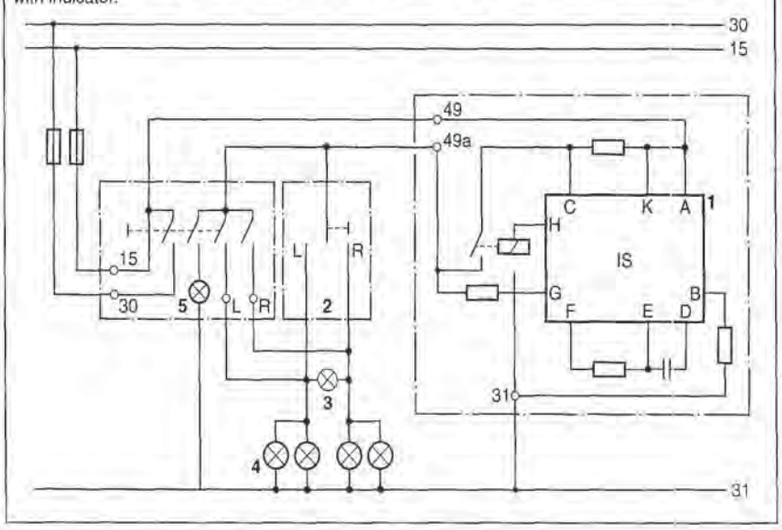
The tractor and trailer share a single monitoring circuit designed to activate the two indicator lamps at the flashing frequency. This type of unit cannot be used to localize lamp malfunctions. The flashing frequency remains constant.

Dual-circuit monitoring

Tractor and trailer are equipped with separate monitoring circuits. Localization of malfunction is by means of the indicator

Hazard-warning and turn-signal flasher for passenger cars

1 Hazard-warning and turn-signal flasher unit with integrated circuit (IS) or pulse generator G and control circuit H, 2 Signal stalk, 3 Indicator lamp, 4 Turn-signal lamps, 5 Hazard-warning switch with indicator.



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lamp which fails to light up. The flashing frequency remains constant.

Side-marker, clearance and tail lamps

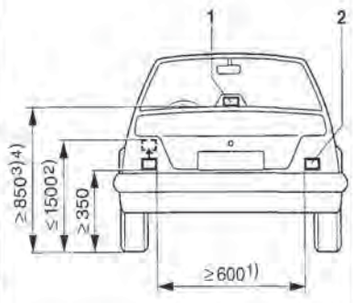
EEC 76/758, ECE R7, StVZO §§ 51 and 53 (FMVSS/CUR). Side-marker lamps (visible from the front) are stipulated on vehicles and trailers with a width exceeding 1600 mm. Tail lamps (at rear) are mandatory equipment on vehicles of all widths. Vehicles wider than 2100 mm (e.g., trucks) must also be equipped with clearance lamps visible from the front and rear.

Side-marker lamps
Two white-light marker lamps are stipulated.

Tail lamps
Two red tail lamps are mandatory equipment.

When the tail and stop lamps are joined in a nested assembly the luminous-intensity ratio for the individual functions must be at least 1:5. Tail lamps are to operate together with the side-marker lamps.

Stop lamps *(positions, dimensions in mm)*
1 Central high-mounted stop lamp (cat. S3),
2 Stop lamps (cat. S1/S2).
[1]) ≥ 400 mm (if width < 1300 mm),
[2]) ≤ 2100 mm (if compliance with max. height regulations not possible) or
[3]) ≤ 150 mm below lower edge of rear window,
[4]) however, the lower edge of the central high-mounted stop lamp must be higher than the upper edge of the (main) stop lamps.



Clearance lamps
Two white-light lamps to be visible from the front, and two red lamps from the rear. Position: As far outward and as high as possible.

Parking lamps

EEC 77/540, ECE R77, StVZO § 51 (FMVSS/CUR).
The vehicle may be equipped with either two parking lamps front and rear or one parking lamp on each side. White light is prescribed at the front and red at the rear. Yellow may also be used at the rear if the parking lamps have been designed as single units with the side turn-signal lamps.

The parking lamps must be designed to operate even when no other vehicle lights (headlamps) are on. The parking-lamp function is usually assumed by the tail and side-marker lamps.

License-plate lamps

EEC 76/760, ECE R4, StVZO § 60 (FMVSS/CUR).
The rear license-plate lamp(s) must be designed to ensure that the rear license plate remains legible up to a distance of at least 25 m by night. Approved are:

The minimum luminance at al points on the surface of the license plate is 2.5 cd/m². The luminance gradient of 2x B_{min}/cm should not be exceeded between any of the test points distributed across the surface of the test plate. B_{min} is defined as the smallest luminance measured at the test point.

Stop lamps

EEC 76/58, ECE R7, StVZO § 53 (FMVSS/CUR).
For each passenger car, two category S1 or S2 stop lamps and one category S3 stop lamp are prescribed as mandatory.

When a nested design with the stop and tail lamps is used, the luminous-intensity ratio between individual functions must be at least 5:1.

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The category S3 stop lamp (central high-mounted stop lamp) must not be incorporated in a nested design with another lamp.

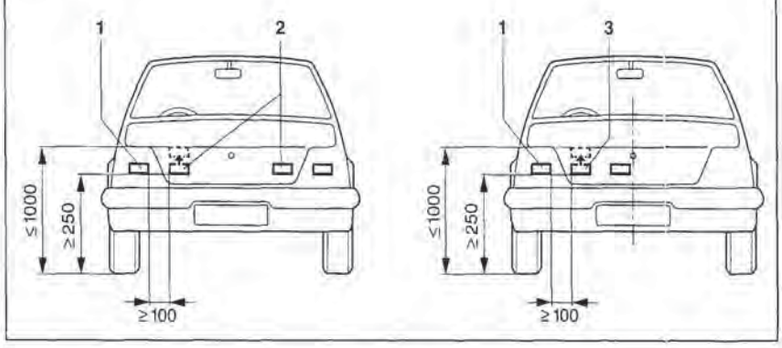
Fog warning lamps (rear fog lamps)

EEC 77/538, ECE R38, StVZO § 53d (FMVSS/CUR).
The European Union nations have mandated that one or two red-light fog warning lamps (rear fog lamps) will be installed on all new vehicles.

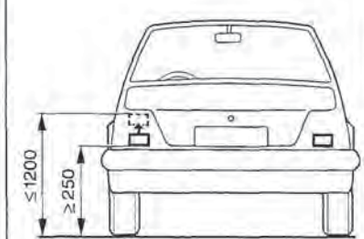
They must be distanced at least 100 mm from the stop lamp.

The visible illuminated area along the reference axis is not to exceed 140 cm². The electrical switching must be designed to ensure that the fog warning lamp operates only in conjunction with the low beam, high beam and/or front fog lamp. It must also be possible to switch off the fog warning lamps independently of the front fog lamps.

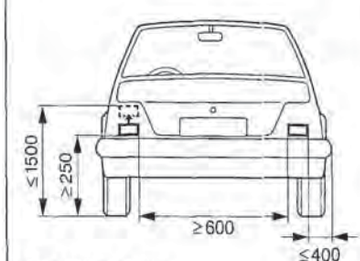
Fog warning lamps (rear fog lamps)
(position, dimensions in mm)
1 Stop lamp,
2 Fog warning lamps (2 units),
3 Fog warning lamp (1 unit).
Position for 1 unit shown for right-hand traffic.



Backup (reversing) lamps
(position, dimensions in mm)
Qty: 1 or 2.



Daytime running lamps
(position, dimensions in mm)



Dimensions in mm

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Prescribed indicator lamp: Yellow (green also approved on vehicles initially registered prior to January 1981).

Backup (reversing) lamps

EEC 77/539, ECE R23, StVZO § 52 (FMVSS/CUR).
One or a maximum of two white-light lamps are approved.

The switching circuit must be designed to ensure that the backup lamps operate only with reverse gear engaged and the ignition on.

Daytime running lamps

ECE R87.
ECE R87 authorizes the installation of daytime running lamps in Europe. Their use or the use of low-beam headlamps to generate daytime running light is prescribed as mandatory in Denmark, Norway, Finland and Sweden. "Low beam on" during daytime hours is required by law in out-of-town areas in Poland and Hungary.

Other lighting devices

Identification lamps StVZO § 52 (FMVSS/CUR), ECE R65.
Identification lamps must be visible through 360 degrees from the vehicle, and should be perceived as flashing when viewed from any particular location. The flashing frequency f lies between 2 and 5 Hz. Blue identification lamps are intended for installation on official vehicles. Yellow identification lamps are designed to warn of dangers or dangerous transport.

Table 4. Specified minimum flash intensities for identification lamps

Measurement range		Flash intensity cd blue	yellow
Parallel to road surface		> 20	> 40
Within light beam Angle to road surface	± 4°	> 10	
	± 8°		> 30

Spot lamps
Spot lamps generate a narrow beam of light of high luminous intensity, making it possible to illuminate a small area from a substantial distance.

Floodlamps
On a moving vehicle, floodlamps may be switched on only when the vehicle's motion represents an integral part of the operation being performed, for instance, when tractors are used in agriculture and forestry, on self-propelled machinery, on rescue vehicles, etc.

Motor-vehicle bulbs

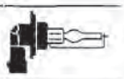
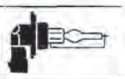
Bulbs for automotive lighting in accordance with ECE R37 are available in 6 V, 12 V and 24 V versions (see table on following pages). To help avoid mix-ups, different bulb types are identified by different base configurations. Bulbs of differing operating voltages are labeled with this voltage in order to avoid mix-ups in the case of identical configurations. The bulb type suitable in each case must be indicated on the equipment.

The luminous efficiency (lumens per watt) represents the bulb's photometric performance level relative to its power input. The luminous efficiency of vacuum bulbs is 10...18 lm/W. The higher luminous efficiency of halogen bulbs of roughly 22...26 lm/W is primarly a result of the increased filament temperature. D2S and D2R gaseous-discharge bulbs (Litronic) provide a luminous efficiency level in the order of 85 lm/W for substantial improvements in low-beam performance.

Replaceable bulbs must be homologated in accordance with ECE R37. Other light sources which do not conform to this regulation (LED, neon, light bulbs) can only be installed as a fixed component part of a lamp. In the event of repair work, the entire unit must be replaced.

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Table 5. Specifications for motor-vehicle bulbs (2-wheeled vehicles not included)

Application	Category	Voltage rating V	Power rating W	Luminous flux Lumen	IEC Base type	Illustration
High beam/ Low beam	R2	6 12 24	45/40[1]) 45/40 55/50	600 min/ 400-550[1])	P 45 t-41	
Fog lamp, High, low beam in 4-HL systems	H1	6 12 24	55 55 70	1350[2]) 1550 1900	P14.5 e	
Fog lamp, High beam	H3	6 12 24	55 55 70	1050[2]) 1450 1750	PK 22s	
High beam/ Low beam	H4	12 24	60/55 75/70	1650/ 1000[1]), [2]) 1900/1200	P 43 t – 38	
High beam, Low beam in 4-HL systems, Fog lamp	H7	12	55	1500[2])	PX 26 d	
Fog lamp	H10	12	55	1250(?)	PY20d	
Low beam in 4-HL systems	HB4	12	55	1100	P 22 d	
High beam/ Low beam	HIR2	12	55	1875	PX22d	
High beam in 4-HL systems	HB3	12	60	1900	P 20 d	
High beam/ Low beam	HIR1	12	65	2500	PX20d	
Stop, flasher, rear fog, reversing lamp	P 21 W PY 21 W[6])	6 12 24	21	460[3])	BA 15 s	
Stop lamp	P 21/5 W PY 21 W[7])	6 12 24	21/5[4]) 21/5 21/5	440/35[3]), [4]) 440/35 440/40[3])	BAY 15d	
Side-marker lamp, tail lamp	R 5 W	6 12 24	5	50[3])	BA 15 s	

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Table 5. Continued

Application	Category	Voltage rating V	Power rating W	Luminous flux Lumen	IEC Base type	Illustration
Tail lamp	R 10 W	6 12 24	10	125[3])	BA 15 s	
Stop, flasher, rear fog, reversing lamp	P 27 W	12	27	475[3])	W2.5x16d	
Stop lamp/ tail lamp	P 27/7 W	12	27/7	475/36[3])	W2.5x16q	
License-plate lamp, tail lamp	C 5 W	6 12 24	5	45[3])	SV 8.5	
Reversing lamp	C 21 W	12	21	460[3])	SV 8.5	
Side-marker lamp	T 4 W	6 12 24	4	35[3])	BA 9 s	
Side-marker lamp, License-plate lamp	W 5 W	6 12 24	5	50[3])	W 2.1 x 9.5 d	
Side-marker lamp, License-plate lamp	W 3 W	6 12 24	3	22[3])	W 2.1 x 9.5 d	
Low beam in 4-HL systems	D1S	85 12[5])	35	3200 approx. 40[5])	PK 32 d-2	
Low beam in 4-HL systems	D2S	85 12[5])	35	3200 approx. 40[5])	P 32 d-2	
Low beam in 4-HL systems	D2R	85 12[5])	35	2800 approx. 40[5])	P 32 d-3	

[1]) High/low beam. [2]) Specifications at test voltage of 6.3; 13.2 or 28.0 V.
[3]) Specifications at test voltage of 6.75; 13.5 or 28.0 V
[4]) Main/secondary filament. [5]) With ballast unit. [6]) Yellow-light version.

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Acoustic signaling devices

Regulations

The internationally applicable ECE Regulation No. 28 specifies that the acoustic signals produced by motor vehicles will maintain a uniform sound quality with no perceptible frequency fluctuations during operation. The use of sirens, bells and the like is not permitted, as well as the playing of melodies by means of sound generators operating in a given sequence.

Installation

Elastic couplings must be used to decouple electric horns from the vehicle's body, as the horn would otherwise induce sympathetic oscillations in the adjacent bodywork. The resulting feedback would diminish both volume and tone quality. Both electric and electropneumatic horns are sensitive to resistance in their control circuits. When horns are installed in pairs, they should be triggered by relays.

Horns

Standard horns

The mass of the armature together with the flexible diaphragm forms an oscillating system within the horn. When voltage is applied to the contacts controlling the solenoid coil, the armature impacts against the magnetic core at the horn's fundamental frequency. A fixed tone disk, directly attached to the armature, responds to these intense periodic impacts by radiating harmonic waves. Legal regulations stipulate that the maximum sound energy must lie within a band of 1.8...3.55 kHz. This frequency explains the horn's relatively piercing note. This sound, which is essentially emitted along the horn's axis and out toward the front, can be heard above the background noise in traffic, even over longer distances.

The horn's size is one of the factors which determine the fundamental frequency and volume.

Supertone horns

As well as increased diameters, supertone horns have a higher electrical power input. Their warning signals can therefore still be heard under extreme conditions (truck driver's cab with high noise levels).

Fanfare horns

The electropneumatic fanfare horn employs the same basic actuating system as its standard counterpart, the salient difference being that the armature is allowed to vibrate freely in front of the coil with no impact occurring. The oscillating diaphragm induces vibration of an air column within a tube. The resonant frequency of the diaphragm and the air column are tuned to one another and they determine the pitch of the signal. The tube is in the shape of a funnel, with a wide opening to enhance the efficiency of sound propagation. The funnel tube is generally coiled to minimize the unit's size.

The presence of upper harmonics in the lower range of the frequency spectrum endows the fanfare horn with a rich, melodious sound. The tone is less penetrating than that of a standard horn due to the more consistent distribution of sound energy over a wide spectrum.

Selecting the correct horn

The warning produced by the impact horn is more conspicuous; it thus represents the better choice for vehicles which are frequently used for long hauls on roads with a large amount of truck traffic. Alternatively, the fanfare horn is superior for urban driving, as pedestrians often find the standard horn excessively loud and unpleasant. Both sets of requirements can be fulfilled by installing the two horn types together in a single system with a selection switch for urban or rural traffic. Standard and fanfare horns both operate at standardized frequencies. High and low tones can be combined to produce a harmonious dual-tone sound.

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Theft-deterrent systems

Regulations

Theft-deterrent systems (car alarms) installed in motor vehicles must conform to the regulations defined in Paragraph 38 b of the StVZO Road Licensing Regulations (FMVSS/CUR) as well as ECE Directive R18. A primed alarm system responds to attempts at unauthorized vehicle entry by emitting warning signals.

Approved warning signals

- Intermittent audible signals of max. 30 s in length emitted by either the vehicle's own horn or an auxiliary unit.
- Flashing signals of max. 5 min in length using the hazard warning flashers (StVZO), or 30 s flashing of the low beams (ECE).

If the vehicle is repeatedly tampered with, the system may only produce a further acoustic alarm once the previous alarm has expired. The driver or authorized person must be able to deprime the alarm immediately. It must be impossible for alarm units to become active when the engine is running. The alarm is not to be triggered by vehicle movements (such as tremors etc.) not caused by attempted unauthorized entry.

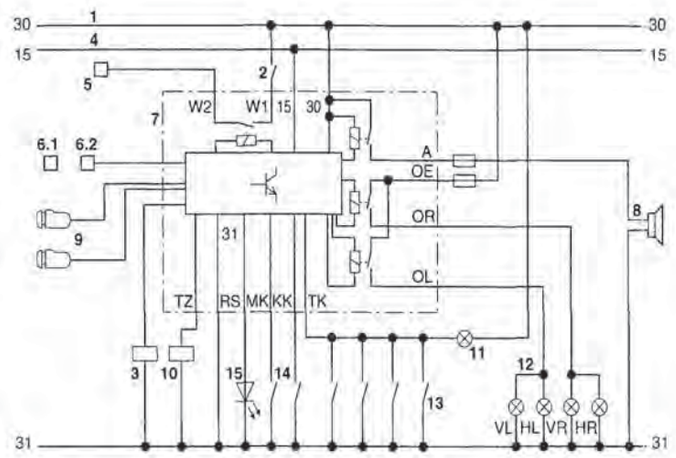
Automotive alarm systems

Basic system

The schematic diagram below is of an actual unit available on the market. The central ECU evaluates the signals at its input terminals, activates the system and switches on the warning signals (horn and visual signals) via its outputs.

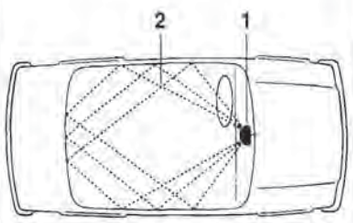
The system is primed and deprimed with an infrared or radio remote-control unit which uses individual codes. This secures the vehicle against unauthorized

Theft-deterrent system (car alarm)
1 Battery, 2 Driving switch, 3 Radio, 4 Ignition, 5 Starting motor, 6 Remote control system (6.1 Transmitter, 6.2 Receiver), 7 Alarm system ECU, 8 Alarm horn, 9 Ultrasonic passenger-compartment protection, 10 Tilt alarm, 11 Interior lighting, 12 Direction indicator, 13 Door contact switch, 14 Contact switches for hood and trunk, 15 Status display.



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Ultrasonic protection for passenger compartment
1 Ultrasonic detector, 2 Ultrasonic field.



use. The ignition must be off (i.e. engine off) before the system can be primed.

The input terminals TK, MK, KK serve as connections for the door and hood switches. Opening one of these will immediately trigger the alarm. This will also occur when the radio-monitoring loop is open-circuited or the ignition is turned on with the alarm primed. Output Z activates the external ancillaries which trigger the alarm via input TZ.

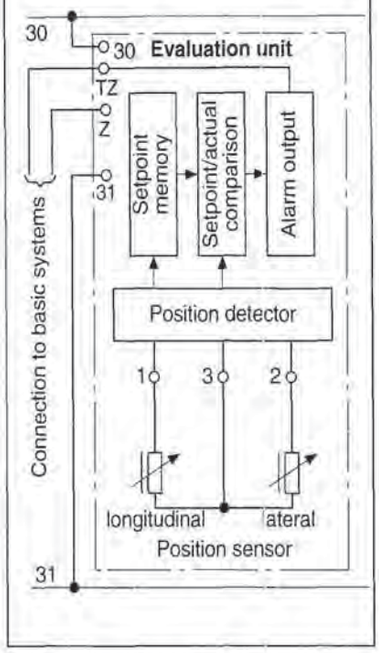
All of the mutually independent inputs are able to trigger consecutive alarms. The starter-motor disable facility prevents unauthorized starting of the engine.

An LED at output RS blinks when the unit is primed and in doing so signals the unit's operating status (primed/deprimed).

Ultrasonic passenger-compartment protection
Such installations generate an ultrasonic field inside the vehicle. An ultrasonic detector registers motion or pressure variations that cause the field to fluctuate. Such variations occur, for instance, when a hand reaches into the vehicle or when a window is broken). The electronic evaluation unit responds with an immediate alarm. The response threshold can be adjusted to optimize the system's protective efficiency.

The following principles are applied for the evaluation of the ultrasonic field: The amplitude method, the pulse-echo method, or a method based on the Doppler effect.

Electronic wheel and tow-away protection



Optional extras

Electronic wheel and tow-away protection
This unit consists of a position sensor and evaluation electronics. Upon priming the alarm unit the inclination or tilt angle at which the vehicle has been parked (on a level surface or on a hill) is stored as the zero reference angle. An alarm is triggered when either the rate of variation in the angle or the angle itself exceeds the programmed limits. Normal changes in the vehicle's position (for instance due to air loss in the tires, deliberate rocking of the vehicle, or settling in soft ground) are recognized as such by the electronics, and do not trigger an alarm.

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Vehicle immobilizers

In addition to the ignition and steering-wheel lock, electronic vehicle immobilizers are employed to protect the vehicle against unauthorized use.

After switching off the ignition, and at the latest when locking the doors after leaving the vehicle, the vehicle immobilizer is automatically activated and puts one or more of those major units out of action which are essential to vehicle operation. Systems incorporating interrupter circuits use a relay to immobilize the units. Normally, the following three major circuits are interrupted:

- Starter motor,
- Fuel supply,
- Ignition or diesel fuel-injection pump.

In the case of units featuring coded access, one or more units are blocked or switched into circuit again by entering a code.

The following systems are used for deactivating the immobilizer:

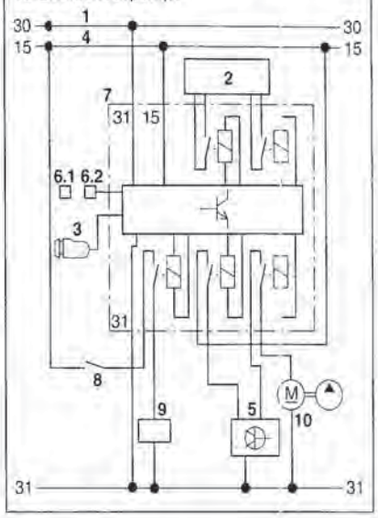
Radio/infrared system
By means of a hand-held transmitter, a coded signal is sent to the immobilizer. The fact that the central-locking system can also be operated with this system represents a further increase in operative comfort for the driver.

Transponder systems
The key is fitted with a transponder which from a short distance transmits a coded signal to a receiver coil (wound around the ignition lock for instance). The system deactivates the immobilizer as soon as the transponder enters the coil's receive area.

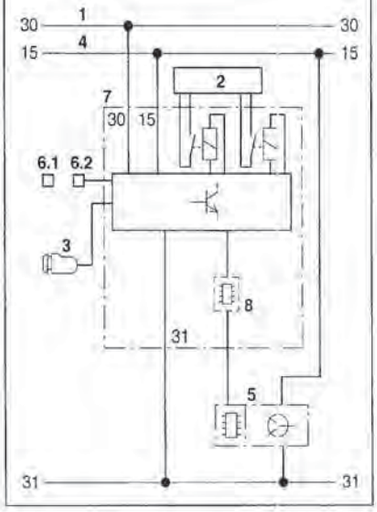
Electronic key
The immobilizer is deactivated when conductive coupling takes place between the electronic key and the immobilizer system.

Code keypad
The secret code number to deactivate the system is entered using a keypad.

System with interrupter circuits
1 Battery, 2 Central locking system, 3 Display unit, 4 Ignition, 5 Engine ECU, 6 Remote-control system (6.1 Transmitter, 6.2 Receiver), 7 Immobilizer ECU, 8 Starting switch, 9 Starting motor, 10 Electric fuel pump.



System with coded access
1 Battery, 2 Central locking system, 3 Display unit, 4 Ignition, 5 Engine ECU, 6 Remote-control system (6.1 Transmitter, 6.2 Receiver), 7 Immobilizer ECU, 8 Code unit.



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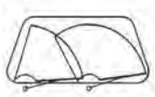
Windshield and rear-window cleaning

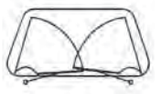
Systems for cleaning the vehicle's windshield, headlamps, and rear window are needed in order to comply with legal stipulations for good visibility at all times. Such systems can be subdivided as follows:
- Windshield wiper systems,
- Rear-window wiper systems,
- Headlamp wiper systems
- Headlamp washer systems
- Combination wiper-washer (wipe/wash) systems.

Windshield wiper systems

The following are the most important windshield-cleaning systems, using the passenger car as an example. These systems are based on the legally prescribed areas of vision (Europe, USA and Australia). The wiped areas may be changed by additional controls acting on the wiper blades (parallelogram, general four-bar mechanism).

Windshield cleaning systems


Tandem system


Opposed system


Tandem system, with aerodynamic characteristics


Single-arm wiper system, not controlled


Single-arm wiper system controlled

The windshield wiper system must meet the following requirements:
- Removal of water and snow,
- Removal of dirt (mineral, organic or biological),
- Operation at high temperature (+ 80 °C) and low temperature (– 30 °C),
- Corrosion resistance against acids, alkalis, salts (240 h), ozone (72 h),
- Service life: passenger cars $1.5 \cdot 10^6$ wipe cycles, commercial vehicles $3 \cdot 10^6$ wipe cycles,
- Stall test.

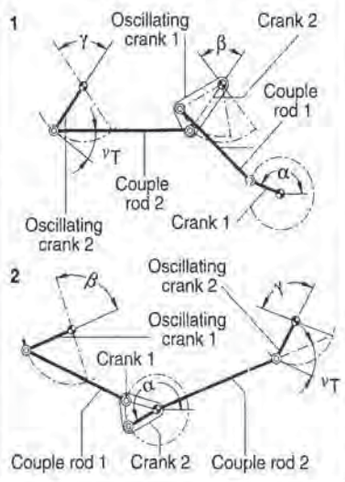
Mechanical wiper mechanism

Series- or parallel-coupled four-bar mechanisms are used, with transversely jointed linkage or additional controlled four-bar linkages also being used for large wipe angles or difficult mechanical transmission configurations.

Optimization of the mechanism is important. Smooth operation can be achieved by matching the maximum values of angular acceleration and/or the force-transmission angles near to the wiper reversal points.

The trend is no longer to allow the crank to rotate fully but to allow it to complete

Principle of the mechanical wiper mechanism
1 Series-coupled, 2 Parallel-coupled, α Crank angle, β, γ Wipe angle, ν_T Tangential force-transmission angle



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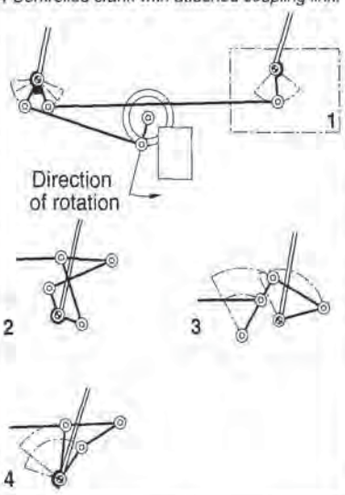
only (barely) a half rotation. This significantly reduces the space needed by the linkage for movement. For this purpose, the drive motor is electronically controlled and its direction of rotation reversed in line with the wiping motion.

It is thus also possible with only minor additional expenditure to achieve an "extended parking position", "protection against snow overload" and, despite varying operating conditions, a relatively constant wiped area.

If problems are experienced in locating the linkage, the above-mentioned technology permits a large linkage to be replaced by 2 significantly smaller units, each with its own (smaller) drive motor.

A second step towards optimization of the wiper mechanism involves the working position of the wiper-blade lip relative to the surface of the windshield or rear-window. By positioning the wiper bearings at the proper angle to the windshield, and by applying additional torsion to the wiper arms, the position of the wiper blades is defined so that at their reversal points they are inclined laterally towards the wiped-area bisector, thus assisting the wiper-blade elements to swivel into their new working position when reversal takes place.

Tandem wiper system *(series-coupled)*
Right-hand part of wiper mechanism designed as:
1 Oscillating crank, 2 Transversely jointed linkage, 3 Additional controlled four-bar linkage, 4 Controlled crank with attached coupling link.



Wiper blades

Wiper blades are used in lengths of 260...1000 mm. Their dimensions for mounting (e. g., slide-in or hook mounting) are standardized. Minimum-wear operation is achieved by eliminating play in their mountings and joints. The tops of the center brackets are perforated to prevent blade lift-off at high speeds. In special cases aerodynamic deflectors are integrated in the wiper arms or blades to press the blades against the windshield.

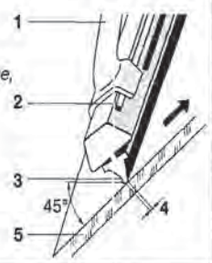
Rubber wiper-blade elements

The most important component of a wiper system is the rubber element. It is loaded by the claws of the bracket system, and supported by spring strips. Its double microedge is pressed against the windshield, and at its point of contact has a width of only 0.01...0.015 mm. When moving across the windshield, the wiper-blade element must overcome coefficients of dry friction of 0.8...2.5 (depending upon air humidity), and coefficients of wet friction of 0.6...0.1 (depending upon frictional velocity). The correct combination of wiper-element profile and rubber properties must be chosen so that the wiper lip can wipe the complete wiped area of the windshield surface at an angle of approx. 45°.

Rear-window cleaning systems

These systems perform similar to windshield-cleaning systems. However, service life is limited to $0.5 \cdot 10^6$ wipe cycles.

Rubber wiper-blade element in working position
1 Claw bracket,
2 Spring strip,
3 Lip,
4 Double microedge,
5 Windshield.



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In right-hand traffic vehicles, the wiped area is preferred with parking position on the right-hand side (as viewed in the direction of travel). The 180° system is used when rear-window dimensions permit.

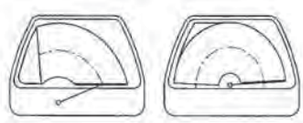
Headlamp cleaning systems

Two systems have been adopted for cleaning headlamps: the wipe/wash system and the wash-only system (P. 773). For the wipe/wash system a wiper arm is driven directly by a motor with a step-down gear unit; the water required to clean the headlamps is taken from the windshield-washer reservoir.

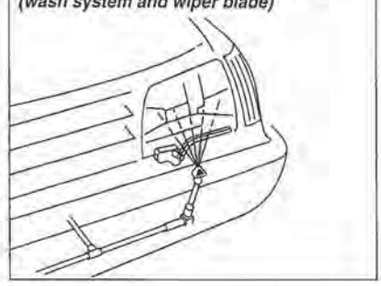
The advantages of the headlamp-wash system lie in its simplicity and often in its easier adaptation to the vehicle styling concept. It is important that the nozzles are positioned correctly so that water jets properly cover the headlamps at all driving speeds.

The law requires that a dirty headlamp whose luminosity has been reduced to 20% must regain 80% luminosity within a period of 8 seconds. The system must be able to complete at least 50 cycles on one charge of cleaning fluid.

Rear-window wiped areas
The shaded areas (here, LHD vehicle) represent impaired visibility with regard to vehicles which are overtaking.



Location of headlamp cleaning nozzle (wash system and wiper blade)



Wiper motors

Permanent-magnet DC motors are used as wiper motors. For normal use in windshield-wiper systems, they incorporate a worm-gear unit, but when used in rear-window and headlamp cleaning systems, they often incorporate an additional gear unit for translating rotary motion into oscillating motion (four-bar linkage, rack-and-pinion mechanism or crank-wheel mechanism).

The output of wiper motors for windshield and window cleaning systems is rated differently from that of common electric drives. Legal regulations and application requirements usually specify that the operating speed of the first wipe frequency be $n_{B1} = 45\ min^{-1}$ and for the second wipe frequency $n_{B2} = 65\ min^{-1}$. Taking the maximum friction between rubber element and window, this results in a torque for each wiper arm which the wiper motor must produce at a minimum speed of $n_A = 5\ min^{-1}$. The starting torque M_{An} in Nm for operation of one wiper arm is calculated as follows:

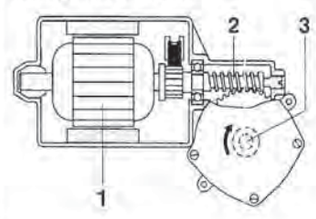
$$M_{An} = F_{WFN} \cdot \mu_{max} \cdot f_S \cdot f_T \cdot L_A \cdot (\omega_{H\,max}/\omega_{Mot}) \cdot (1/\eta_{Getr.}) \cdot (R_{Aw}/R_{Ak})$$

F_{WFN} Downward nominal load of wiper arm in N (arm movement on windshield or rear window, approx. 15 N per m wiper-blade length),

μ_{max} Max. coefficient of dry friction of rubber element (2.5 at φ = 93% relative air humidity),

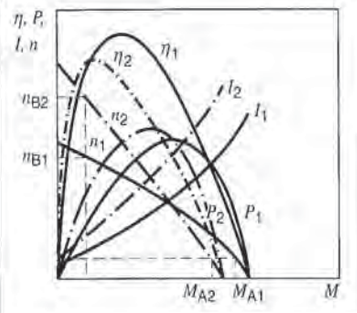
f_S Multiplier to account for joint fric-

Wiper motor with worm gear
1 Permanent-magnet DC motor, 2 Worm gear, 3 Shaft end.

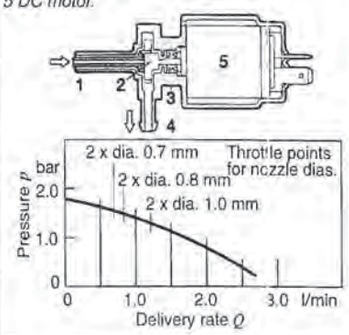


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Characteristic curve of a wiper motor
n Speed, I Current, P Power output, η Efficiency, M Torque. The subscripts refer to wiper speeds 1 and 2.



Electric washing-system pump with characteristic curve
1 Inlet connection with inlet fitting, 2 Impeller, 3 Pump housing, 4 Delivery connection, 5 DC motor.



tion of wiper arm (usually 1.15),

f_T Tolerance factor to account for wiper-arm load tolerance (usually 1.12),

L_A Wiper-arm length in m,

ω_{Hmax} Max. angular velocity of the wiper arm,

ω_{Mot} Mean angular velocity of the wiper-motor crank,

$\omega_{H\ max}/\omega_{Mot} \approx 0.15 \cdot (0.01 \cdot \omega_W)^2 + \sin(\omega_W/2)$

ω_W = Wiper angle β or γ

η_{Getr} Efficiency of gear unit, usually assumed to be 0.8; must be measured separately when using special mechanism components (e.g. transversely jointed linkage, additional controlled four-bar linkage, multiple O-ring seals),

R_{Aw} Electrical resistance of armature winding heated by nominal operation,

R_{Ak} Electrical resistance of cold armature winding.

(R_{Aw}/R_{Ak}) usually 1.25

The motor's short-circuit strength is another important factor in its design. This is defined as the length of time during which the stalled motor must withstand the full test voltage without short-circuiting its windings (usually specified as t_K = 15 min).

Together, the frictional load and the gear ratio result in an operating torque on wet glass which is 0...20 % of the tightening torque. In the case of very large wiper systems operating on nearly vertical windshields (e.g. on buses), the torque of the wiper arm resulting from its own weight must also be taken into consideration (separately calculated allowance).

In the case of drive motors with pendular mechanisms (rear window and headlamp cleaning systems), the required torque is determined at the oscillating shaft.

It is calculated as:

$M_H = F_{WFN} \cdot \mu_{max} \cdot f_S \cdot f_T \cdot L_A \cdot (R_{Aw}/R_{Ak})$

Washing systems

To ensure good visibility in the wiped areas, it is imperative that the wiper system is backed by a washing system. Electrical centrifugal pumps of simple design are used (characteristic pump curve) to pump the water through 2 to 4 nozzles and onto the windshield in a narrow spray pattern. The water, to which a cleaning additive is added, is contained in a reservoir of 1.5...2 l capacity. If the same water is used also to clean the headlamps, a larger volume of up to 7 l is necessary. A separate reservoir can be provided for the rear-window cleaning system. The washing system is often coupled to the corresponding wiper system by means of an electronic control so that water is sprayed onto the window or windshield while a pushbutton is pressed, with the wiper system continuing to operate for several additional cycles after the pushbutton is released.

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Automotive windshield and window glass

Silica glass is used for automotive windshields and windows. Approximate composition:
72 % SiO_2 as vitrifier
14 % Na_2O as flux
10 % CaO } as stabilizer
4 % MgO }

Windows are made of flat glass which is poured using the float-glass process. Fusion occurs at a temperature of 1550 °C; the melt then passes through a refining zone of 1500 °C to 1100 °C before being poured onto the so-called float bath (liquid tin) and cooled to a temperature of 600 °C. It is at this stage that the two plane-parallel surfaces of exceptionally high quality are produced (tin-bath surface below, fire-finished surface above). After passing through a cooling zone, the glass is then cut to pieces of 6.10 m × 3.35 m.

The glass can now be further processed in one of two ways:
- as single-pane toughened safety glass (TSG)
  or
- as laminated safety glass (LSG), which is primarily used for windshields.

TSG windows differ from LSG windows in that they exhibit higher mechanical and thermal strength as well as different breakage and shattering behavior. TSG panes pass through a toughening process which greatly prestresses the surface of the glass. In case of breakage, these windows break into many small dull-edged pieces.

The LSG window, on the other hand, exhibits a normal shattering pattern. When an LSG window is cracked, it is

Material properties and physical data of automotive glass (finished windshields and windows)

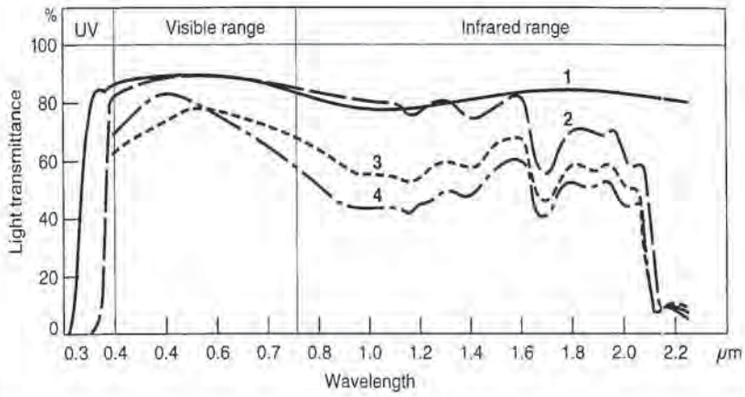
Property	Dimension	TSG	LSG
Density	kg/m³	2500	2500
Surface hardness	Mohs	5...6	5...6
Compressive strength	MN/m²	700...900	700...900
Modulus of elasticity	MN/m²	68 000	70 000
Bending strength			
before prestressing	MN/m²	30[2])	30[1])
after prestressing	MN/m²	50[2])	
Impact-ball strength (DIN 52 306)	Nm	> 7	> 90[1])
(at room temperature)		(227 g ball)	(2.26 kg ball)
Arrow-drop strength			> 18[1])
DIN 52 307 (at room temperature)			
Specific heat	kJ/kgK	0.75...0.84	0.75...0.84
Coefficient of thermal conductivity	W/mK	0.70...0.87	0.70...0.87
Coefficient of thermal expansion	m/mK	$9.0 \cdot 10^{-6}$	$9.0 \cdot 10^{-6}$
Dielectric constant		7...8	7...8
Light transmittance (DIN 52 306), clear	%	≈ 90	≈ 90[1])
Index of refraction		1.52	1.52[1])
Deviation angle of wedging	Angle	< 1.0 flat	≤ 1.0 flat[1])
	minute	< 1.5 bent	≤ 1.5 bent[1])
Dioptric faults (DIN 52 305)	Dptr.	< 0.03	≤ 0.03[1])
Temperature resistance	°C	200	90[1]) (max. 30')
Resistance to temperature shocks	K	200	

[1]) Properties of finished laminated safety glass (LSG). In calculating the permissible bending stress, the coupling effect of the PVB film is to be disregarded.

[2]) Calculated values; these values already contain the necessary safety margins.

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Light transmittance of automotive windshield and glass
1 Float glass and TSG windows, thickness: 4 mm,
Tinted LSG windows of different colors, total thickness: 5.5 mm,
2 Clear, 3 Bronze-colored, 4 Green



easier to see through than a TSG window. Laminated glass is made of two panes of glass bonded to each other by means of an intermediate film of plastic (polyvinyl butyral).

The formation of many small, blunt fragments of glass which are held in position by the plastic film reduces the danger of injury. In the case of the Sekuriflex window (a refinement of the LSG window), an additional plastic film with excellent optical and mechanical properties (e.g., abrasive and scratch resistance) is applied to the inside surface of the window.

Green or bronze-colored windows are used for heat absorption because they block the transmission of infrared light more strongly than light of shorter wave lengths. On the other hand, light transmission in the visual spectrum is reduced from 90% to 80%.

Optical properties

The optical properties of TSG and LSG windows are roughly the same, because the optical properties of the PVB film are very closely aligned to those of glass.

Heatable safety glass

In the case of heatable safety glass (TSG), heating conductors are applied to the window (not yet prestressed) using the silk screen process, and are sintered in place during the prestressing process. Subsequent galvanization increases the strength of the heating conductors, tempers them and protects them against environmental influences. Window areas treated in this way can be kept free of mist by heating, and even of ice if the heating power is high enough.

Heated window areas must never be cleaned using caustic or scouring cleansers (e.g. chlorine, ammonia, sand or acids).

In the case of LSG windows, very thin, nearly invisible heating wires are applied in wavy lines to the plastic adhesive film. They are connected in series and/or parallel in order to achieve the intended electrical resistance. The thin wires provide even better heating coverage. The heating power of passenger-car rear window heating systems is aproximately 3...5 W/dm².

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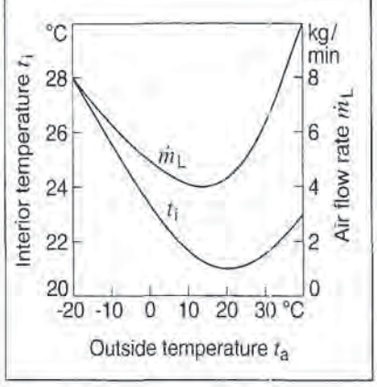
Passenger-compartment heating, ventilation and air conditioning (HVAC)

Function

The vehicle's climate-control system provides for the following:

- a comfortable climate for all passengers,
- an environment calculated to minimize driver stress and fatigue,
- more recent units use filters to remove particulate matter (pollen, dust) and even odors from the air,
- good visibility through all windows, and windshield.

Comfortable interior temperature and air flow rate as a function of outside temperature



In many countries the performance of the heater unit is governed by legal requirements, with emphasis on the defroster's ability to maintain clear windows and windshields (such as EEC Directive 78/317 within the European Community, and MVSS 103 in the USA).

Systems deriving heat from the engine

On vehicles with liquid-cooled engines, the engine heat (by-product of the combustion process) contained in the coolant is used to warm the passenger compartment. With air-cooled engines, engine heat is taken from the exhaust or, in some cases, from the engine's lubrication circuit. The heater core consists of tubes and fins, and has the same basic design as the engine radiator. Coolant flows through the core's tubes while air flows through its fins. Two design concepts are available for regulating the heater's thermal output.

Coolant-side heater control

In these systems, the entire air flow is usually directed through the heater core while a valve controls the heating output by regulating the flow of coolant through the unit. Extreme precision is required from the valves, which must be capable of providing consistent, stable settings for accurate control at the minimal flow rates necessary for maintaining low heat levels (important between seasons). A disadvantage lies in the fact that the heating output varies according to the coolant's pressure and temperature, meaning that heater performance is affected by engine speed and load.

Air-side heater control

In this type of system the flow of coolant through the heater core is unrestricted. The heat is regulated by dividing the air flow before it reaches the core: A portion of the air flows through the core, while the rest is directed around it. The two currents are subsequently reunited in the plenum chamber. An air flap can be used to regulate the distribution of the two currents, thereby determining the amount of heat taken from the coolant. This type of control arrangement is less sensitive to fluctuations in engine load, and air-temperature adjustments take effect immediately. Water and/or air-side blocks prevent undesirable residual warmth from emerging from the hot core when the heater is not in operation.

One disadvantage of the air-side control layout lies in the larger installation volume required for housing the bypass duct and the plenum chamber.

The air-ventilation current is provided by a constant-speed or adjustable-speed electric blower, whereby aerodynamic

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pressure can also become a significant factor as vehicle speed increases. The minimum hourly air-flow rate should be 30 m³ per person (figure for reference purposes only). Factors such as passenger-compartment temperature, outside temperature, air flow and (to a certain degree) heat radiation, all affect the process of achieving a comfortable climate (see illustration). Because the precise data on these variables vary substantially from one vehicle to the next, actual figures must be derived empirically.

As motor vehicles have relatively small passenger compartments, drafts and sunlight entering through windshield and windows confront the heating, ventilation and air-conditioning (HVAC) systems with extreme challenges. In order to enhance passenger comfort, an attempt should be made to maintain the temperature in the footwell 4...8 °C above that of the air around the upper body.

Electronic heater control

Variations in outside-air temperature and vehicle speed cause fluctuations in the temperature of the passenger compartment. On standard systems, the heater controls must be constantly readjusted by hand in order to maintain an agreeable temperature. Electronic heater control dispenses with this requirement by maintaining the interior temperature at the desired level automatically.

On heater units featuring water-side regulation, sensors monitor the temperatures of the vehicle's interior and of the emerging air; the control unit processes this information and compares it with the preselected temperature. Meanwhile, a solenoid valve installed in the cooling circuit opens and closes at a given frequency in response to the signals which it receives from the control unit. The adjustments in open/close ratio in the cycle periods regulate the flow rate from the closed position up to the maximum. A servo-actuated adjustment flap is usually employed to provide infinitely-variable temperature regulation in air-side systems (pneumatic linear adjusters are also occasionally used). Sophisticated systems allowing separate adjustment for the left and right sides of the vehicle are also available.

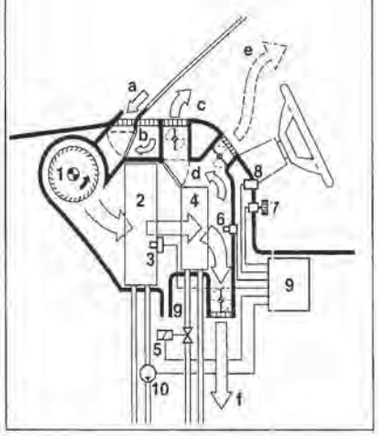
Air conditioners

The heater unit alone is not capable of providing a comfortable environment at all times. When the outside temperature climbs beyond 20 °C, the air must be cooled to achieve the required interior temperatures. Here, compressor-driven refrigeration units with R 134a refrigerant are in use (until 1992, R12 refrigerant).

The engine-driven compressor compresses the vaporous refrigerant, which heats up in the process and is then directed to the condenser where it cools and liquefies. The energy supplied in the compressor and the heat absorbed in the evaporator are dissipated to the environment here.

An expansion valve sprays the cooled liquid into the evaporator where the evaporation process extracts the required evaporation heat from the incoming stream of fresh air, thereby cooling the air.

Air conditioner with electronic water-side control (schematic)
1 Blower, 2 Evaporator, 3 Evaporator temperature sensor, 4 Heater core, 5 Solenoid valve, 6 Air exit-temperature sensor, 7 Setpoint control, 8 Interior sensor (ventilated), 9 Control unit, 10 Compressor; a) Fresh air, b) Recirculated air, c) Defrost, d) AC bypass, e) Ventilation, f) Footwell, g) Condensation drain.



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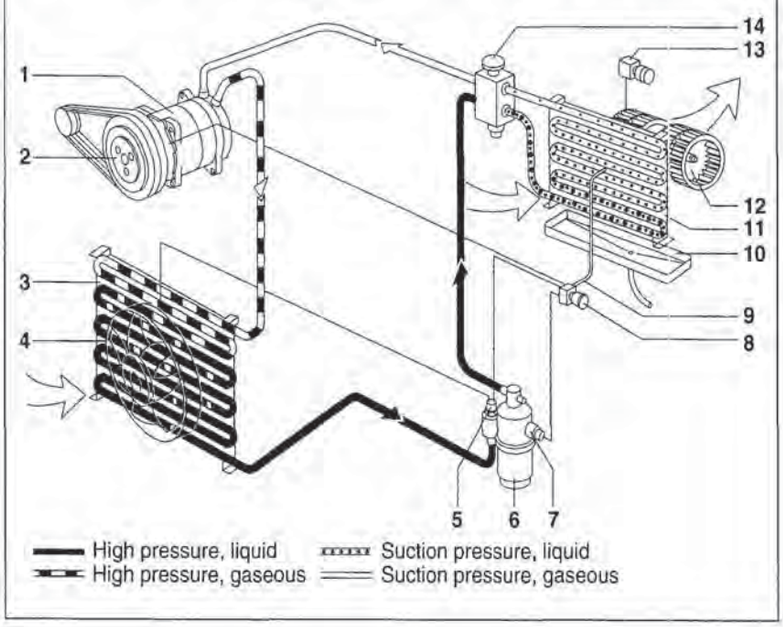
Moisture is extracted from the cooled air as condensation, and the air's humidity is reduced to the desired level. Evaporators and condensers are generally designed as tube-and-fin heat exchangers. The evaporator is located before the heater core in the fresh-air stream and cools the air to approx. 3...5 °C. The dehumidified air is reheated in the heater core to the desired temperature.

Automatic climate control

Automatic climate control is particularly useful for vehicles in which both air conditioner and heater are installed, because the constant monitoring and adjustment required to maintain a temperate climate presents the occupants with a complicated task. This rule applies to bus drivers in particular, as they are exposed only to the conditions at the front of the vehicle. An automatic climate control system incorporating a preselection feature can automatically maintain the correct temperature, air flow and air distribution in the passenger compartment. These parameters are mutually interdependent, and changes to one will affect the others. At the center of the system is a temperature-control circuit for interior temperature. The control unit continuously monitors both the preselected temperature and all essential variables which affect the system, using this information to calculate a setpoint t_i. The setpoint is compared with the actual temperature, and the control unit uses the difference between the two as the basis for determining the required heating, refrigeration, and air-flow rate. Another function controls the position of the air-distribution flaps with reference to the program which the occupants have selected. Meanwhile, all control circuits continue to respond to manual inputs.

Coolant circuit of an air-conditioning system

1 Compressor, 2 Electric clutch (for compressor on/off function), 3 Condenser, 4 Auxiliary fan, 5 High-pressure switch, 6 Fluid reservoir with desiccant insert, 7 Low-pressure switch, 8 Temperature switch or on/off control (for compressor on/off function), 9 Temperature sensor, 10 Condensate drip pan, 11 Evaporator, 12 Evaporator fan, 13 Fan switch, 14 Expansion valve.



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The setpoint temperature determined by the control unit is achieved by means of water or air-side adjustments (as described in "Electronic heater control").

Infinitely-variable or graduated blower control is used to adjust the air flow to the specified level. There is generally no setpoint processing involved in this operation. This type of arrangement is inadequate for dealing with the increases in flow rate caused by aerodynamic pressure at high speeds. Here, a special control function can compensate by responding to increasing vehicle speeds, initially by reducing the blower speed to zero, and then, should the flow continue to rise, by using a restriction flap to throttle the stream of incoming air.

Control of air distribution to the three levels – defroster, upper compartment and footwell – is effected manually, with preselection, or with a fully-automatic program. Especially popular are units featuring program control buttons in which each button provides a specific air-distribution pattern for the three levels.

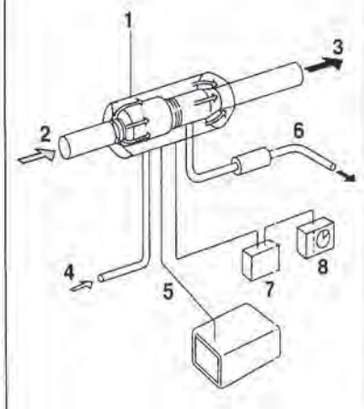
The defroster represents a special case. In order to clear the windows as rapidly as possible, the temperature control must revert to maximum heat and maximum blower speed while directing the air flow through the upper defroster outlets. On systems with program switches and fully-automatic units, this operating mode is selected with a single button; at temperatures above 0 °C the refrigeration unit is also activated to extract humidity from the air. To prevent the still unheated air giving rise to drafts, the blower is switched off electronically after cold starts in winter, except when "DEF" and cooling are in operation.

The variations described above are used in both passenger cars and trucks. Buses require more complicated layouts. The passenger compartment can be divided into several control zones, in which the temperature is controlled by electronically regulating the speed of the zone's individual water pump.

Auxiliary heater installations

The fuel for units which produce heat without the aid of the engine is supplied either by the standard vehicle tank, or, on large vehicles, by a separate tank. An electric pump supplies the fuel to an injector, which sprays a fuel mist into the combustion chamber; the mist mixes with the air and burns. The hot exhaust gases are then directed to a heat exchanger. The heat exchanger can function in one of two ways, either warming the interior air directly, or by transferring heat to the engine's coolant circuit. In the latter case, circulation is maintained by a separate electric pump, making it possible to use the standard heating unit to warm the passenger compartment. This type of auxiliary heating for the engine coolant also improves cold-start response in the winter.

Auxiliary air heater (independent of engine)
1 Air heater with blower for combustion and heating air, combustion chamber and heat exchanger, 2 Hot-air intake, 3 Air outlet to vehicle interior, 4 Combustion-air intake, 5 Fuel supply, 6 Exhaust system, 7 Electronic control unit, 8 Thermostat and timer for preselecting switch-on time.



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Instrumentation

Drivers have to process a permanently increasing stream of information originating from their own and other vehicles, from roads and from telecommunications equipment. All this information must be conveyed to drivers in the information and communications areas of their vehicles with suitable display and indicating equipment and in compliance with ergonomic requirements. In future, it will increasingly become the norm for cellular phones, navigation systems and ranging systems to join radios and vehicle monitoring systems as standard features in motor vehicles.

Information and communications areas

In any vehicle, there are four information and communications areas, which must satisfy different requirements in terms of their display features:
- the instrument cluster,
- the windshield,
- the center console and
- the vehicle rear compartment.

The display features of these areas are determined by the available range of information and the necessary, useful or desirable information for the occupants concerned.

Dynamic information and monitoring information (e.g. fuel level) to which the driver should respond is indicated in the instrument cluster (i.e. as near as possible to the primary field of vision).

A head-up display (HUD), which reflects the information in the windshield, is especially suited to engaging the driver's attention (e.g. in the case of warnings from a ranging warning radar (ACC) or route directions). The display is supplemented acoustically by voice output.

Status information or dialog prompts are mainly displayed in the vicinity of the operator unit in the center console.

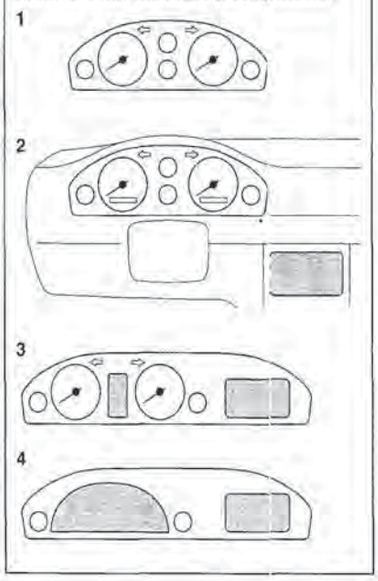
Information of an entertainment nature is featured in the vehicle rear compartment, away from the primary field of vision. This is also the ideal location for the mobile office. The backrest of the front passenger seat is a suitable installation location for the monitor and operator terminal of a laptop computer.

Driver information systems

The driver information area in the vehicle cockpit and the display technologies used have gone through the following stages of development:

Individual and combined instruments
Conventional individual instruments for the optical output of information were initially superseded by more cost-effective instrument clusters (combination of several information units in a single housing) with good illumination and antireflection qualities. The passage of time, with the continual increase in information, saw the creation in the existing space available of the modern instrument cluster with several needle instruments and numerous tell-tale lamps (Fig., no. 1).

Driver information area *(development)*
1 Needle instrument, 2 Needle instrument with TN LCD and separate AMLCD in center console, 3 Needle instrument with (D)STN and integrated AMLCD, 4 Programmable instrument with two AMLCD components.



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Digital displays

Digital instruments

The first digital instruments used on a large, worldwide scale featuring vacuum fluorescent displays (VFDs) and later liquid-crystal displays (LCDs) can now mainly be found in Japan and the USA in luxury-class cars. These instruments found little favor in Europe and have now virtually disappeared from use. Here the trend towards needle instruments has continued uninterrupted but with a radical technical change in the way in which they are activated (Fig., no. 2).

Central display and operator unit in center console

The introduction of navigation and driver information systems (PP. 796/798) has seen the incorporation of a display monitor and a keypad in the center console. Such systems combine all the additional information from functional units and information components (e.g. cellular phone, car radio/CD, controls for heating/air conditioning (HVAC) and – important for Japan – the "TV" function) in a central display and operator unit. The components are interconnected in a network and are capable of interactive communication.

Positioning this terminal, which is of universal use to driver and passenger, in the center console is effective and necessary from both ergonomic and technical standpoints. The optical information appears in a graphics display. The demands placed by TV reproduction and the navigation system on the image/map display determine the resolution and color reproduction.

Graphics modules

Fitting vehicles with a driver's airbag and power steering as standard has resulted in a reduction in the view through the top half of the steering wheel. At the same time, the amount of information that has to be displayed in the installation space available has increased. This creates the need for additional display modules with graphics capabilities whose display areas are able to show any information flexibly and in prioritized form.

This tendency results in instrumentation featuring a classical needle instrument but supplemented by a graphics display. Even the central display monitor is located at the height of the instrument cluster. What is important to all optical displays is that they can be easily read inside the driver's primary field of vision or in its immediate vicinity without the driver having to divert his eyes from the road for long periods. This is the case for instance when the displays are positioned in the lower area of the center console (Fig., no. 3).

The graphics modules in the instrument cluster permit mainly the display of driver- and vehicle-related functions such as e.g. service intervals, check functions covering the vehicle's operating state and also vehicle diagnostics as needed for the workshop. They can also show route-direction information from the navigation system (no digitized map excerpts, only route-direction symbols such as arrows as turn-off instructions or intersection symbols). It is to be expected that the initially monochrome modules will be succeeded in a second stage by color displays whose reading speed and reliability are increased by color reproduction.

For the central display monitor with an integrated information system, the tendency is now to switch from a 4:3 aspect ratio to a wider format with a 16:9 aspect ratio (cinema-film format), which enables additional route-direction symbols to be displayed as well as the map. More recent developments see the display being split into two separate screens: one screen is only read by the driver and displays driver-related information while the other screen can only be seen by the passenger and shows e.g. a cinema film.

Individual module with computer monitor

By roughly 2003, the remaining mechanical measuring instruments will have disappeared and the entire area of the instrument cluster will have been replaced by a freely addressable graphics monitor (provided the price of LCD computer monitors develops economically). Such a unit module would offer an array of information display options. It is conceivable to have an individually selectable form of display which is geared to the specific needs of individual user groups (e.g. frequent, older or sports car users) (Fig., no. 4).

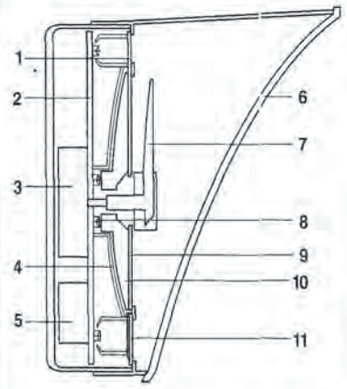
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Instrument clusters

Design

Microcontroller technology and the ongoing networking of motor vehicles have in the meantime transformed instrument clusters from precision-mechanical instruments to electronically dominated devices. A typical instrument cluster (LED-illuminated, with TN-type conductive-rubber-contacted segment LCDs, see Fig.) is a very flat component (electronics, flat stepping motors) and virtually all the components (mainly SMT) are directly contacted on a printed-circuit board.

***Instrument cluster** (design)*
1 Tell-tale lamp, 2 Printed-circuit board, 3 Stepping motor, 4 Reflector, 5 Plug, 6 View cover, 7 Needle, 8 LED, 9 Dial face, 10 Optical waveguide, 11 LCD.



Operating concept

While the basic functions are the same in most instrument clusters (see typical block diagram), the partitioning of the function blocks into (partly application-specific) microcontrollers, ASICs and standard peripherals sometimes differs significantly (product range, display scope, display types).

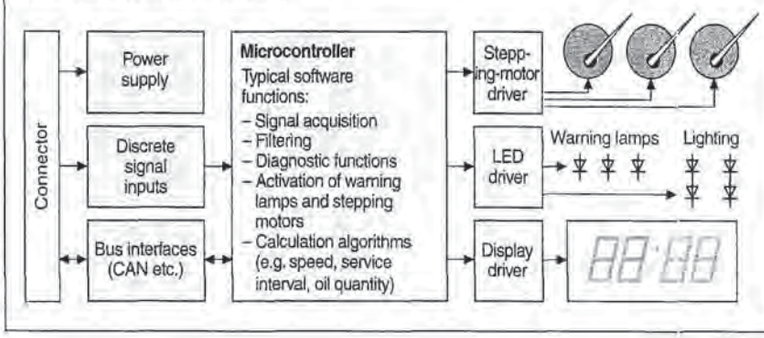
Electronic instrument clusters indicate measured variables with high accuracy thanks to stepping-motor technology, and also take over "intelligent" functions such as engine-speed-dependent oil-pressure warning, prioritized fault indication in matrix displays, or service-interval indicator. Even online diagnostic functions are standard and take up a significant part of the program memory.

Because instrument clusters are standard features on all vehicle types and all the bus systems merge here anyway, the former are becoming increasingly established as gateways, i.e. bridges between the different bus systems in the motor vehicle (e.g. engine CAN, body CAN and diagnostic bus).

Measuring instruments

In Europe, manufacturers still tend towards the classical instrument with mechanical needle and dial face. Here, initially the compact, electronically triggered moving-magnet quotient measuring instrument replaced the bulky eddy-current speedometer. Geared stepping motors with very low

***Instrument cluster** (block diagram)*



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overall depth and still higher precision are currently used. Thanks to a compact magnetic circuit and (mostly) 2-stage gearing with a power output of only approx. 100 mW, these motors permit swift and highly accurate needle positioning.

Lighting

Instrument clusters were originally lit by frontlighting technology in the form of electric bulbs. Backlighting technology has in the meantime gained acceptance on account of its attractive appearance. Bulbs have been replaced by long-lasting light-emitting diodes (LEDs, see P. 143). LEDs are also suitable for warning lamps and for the backlighting of scales, displays and (via plastic optical waveguides) needles (refer to table headed "Overview of lighting sources").

Highly efficient InAlGaP-technology LEDs are meanwhile available in the colors yellow, orange and red. The newer InGaN technology is currently producing significant efficiency improvements in the colors green, blue and white. Here the color white is obtained through the combination of a blue LED chip with an orange-emitting luminescent material (yttrium-aluminum granulate).

However, new technologies are also being used for special configurations:

- CCFLs (Cold Cathode Fluorescent Lamps): mainly for "black screen" instruments, which appear black when they are deactivated. The combination of a tinted view cover (e.g. 25% transmission) with these very bright lamps (high luminance, high voltage) produces a brilliant appearance with outstanding contrast. Since color LCDs have very low transmission (typically approx. 6%) it is imperative that CCFLs are used to backlight them in order to obtain good contrast even in daylight.
- EL (electroluminescent) film: AC is applied to this flat film in order for it to light up. It features very uniform light distribution, and has only recently become suitable for automotive applications. It offers extensive freedom of design for color combinations and/or for superimposing dial areas on display surfaces.

Overview of lighting sources

Lighting source	Possible colors	Typical data[1]	Technically suitable for	Conventional ICs	Black-screen instrum.	Service life[2] in h	Activation
Bulb	White (every color possible with filter)	2 lm/W 65 mA 14 V	Dial face Needle Display	+ o +	– – o	$B_3 \approx 4{,}500$	No special activation required
SMD LED luminescent diode	Red, orange, yellow (InAlGaP)	8 lm/W 25 mA 2 V	Dial face Needle Display	o + +	– + –	$B_3 \gg 10{,}000$	Series resistors or control required
	Blue, green (InGaN), white (with converter)	2...5 lm/W 20 mA 3.6 V	Dial face Needle Display	o o o	– o –	$B_3 > 10{,}000$	
EL film electroluminescent	Blue, violet, yellow, green, orange, white	2 lm/W 100 V AC 400 Hz	Dial face Needle Display	+ – –	– – –	approx. 10,000	High voltage required
CCFL cold-cathode lamp	White (every color possible depending on luminescent material)	25 lm/W 2 kV AC 50...100 kHz	Dial face Needle Display	+ – +	+ o +	$B_3 \approx 10{,}000$	High voltage required

[1]) Efficiency in lm/W (lumen per watt), current in mA, voltage in V or kV, activation frequency in kHz.
[2]) B_3 Point with 3% failure probability. Suitability: + preferred, o conditional, – no application.

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Display types

TN LCD

With its high status of development, TN-LCD technology ("Twisted Nematic Liquid-Crystal Display", see P. 143) is the most commonly used form of display. The term stems from the twisted arrangement of the elongated liquid-crystal molecules between the locating glass plates with transparent electrodes. A layer of this type forms a "light valve", which blocks or passes polarized light depending on whether voltage is applied to it or not. It can be used in the temperature range of –40 °C...+85 °C. The switching times are relatively long at low temperatures on account of the high viscosity of the liquid-crystal material.

TN LCDs can be operated in positive contrast (dark characters on a light background) or negative contrast (light characters on a dark background). Positive-contrast cells are suitable for front- and backlighting while negative-contrast cells can only be read with satisfactory reading contrast when strongly lit from the rear. TN technology is suitable not only for smaller display modules but also for larger display areas in modular or even full-size LCD instrument clusters.

Graphics displays for instrument clusters

Dot-matrix displays with graphics capabilities are needed to display infinitely variable information. They are activated by line scanning and therefore require multiplex characteristics. Under the conditions prevailing in a motor vehicle, conventional TN LCDs can today produce multiplex rates of up to 1:4 with good contrast and up to 1:8 with moderate contrast. Other LCD display technologies are needed to achieve higher multiplex rates. STN and DSTN technologies are now being used for modules with moderate resolution. DSTN technology can be implemented to provide monochrome or color displays.

STN LCD and DSTN LCD

The molecule structure of an STN (Super Twisted Nematic) display is more heavily twisted inside the cell than in a conventional TN display. STN LCDs permit only monochrome displays; usually in blue-yellow contrast. Neutral color can be obtained by applying "retarder film", but this is not effective throughout the entire temperature range encountered in the vehicle.

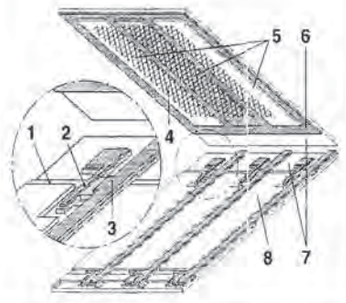
DSTN LCDs (Double-layer STN) feature considerably improved characteristics, which permit neutral black-and-white reproduction over wide temperature ranges with negative and positive contrast. Color is created by backlighting with colored LEDs. Multicolor reproduction is created by incorporating red, green and blue thin-film color filters on one of the two glass substrates. Under automotive conditions, shades of gray are only possible to a very limited extent, the result of which is that the range of colors is limited to black, white, the primary colors red, green and blue and their secondary colors yellow, cyan and magenta.

AMLCD

The task of the visually sophisticated and rapidly changing display of complex information in the area of the instrument cluster and the center console with high-resolution liquid-crystal monitors with video capabilities can only effectively be performed by an AMLCD (Active-Matrix Liquid-Crystal Display).

The best developed and mostly widely used are the TFT LCDs (Thin-Film Transistor LCDs) addressed with thin-film

Thin-film transistor LCD (TFT LCD)
1 Row circuit, 2 Thin-film transistor, 3 Column circuit, 4 Front-plane electrode, 5 Color layers, 6 Black matrix, 7 Glass substrate, 8 Pixel electrode.



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transistors. Display monitors with diagonals of 4″...7″ in the center-console area and an extended temperature range (–25 °C...+85 °C) are available for motor vehicles. Formats of 10″...14″ with a still wider temperature range (–40 °C... +95 °C) are planned for programmable instrument clusters.

TFT LCDs consist of the "active" glass substrate and the opposing plate with the color-filter structures. The active substrate accommodates the pixel electrodes made from tin-indium oxide, the metallic row and column circuits and the semiconductor structures. At each intersecting point of the row and column circuits, there is a field-effect transistor which is etched in several masking steps from a previously applied sequence of layers. A capacitor is likewise generated at each pixel.

The opposite glass plate accommodates the color filters and a "black-matrix" structure, which improves the contrast of the display. These structures are applied to the glass in a sequence of photolithographic processes. A continuous counter-electrode is applied on top of them for all the pixels. The color filters are applied either in the form of continuous strips (good reproduction of graphics information) or as mosaic filters (especially suitable for video pictures).

Head-up display (HUD)

Conventional instrument clusters have a viewing distance of 0.8...1.2 m. In order to read information in the area of the instrument cluster, the driver must adjust his vision from infinity (observing the road ahead) to the short viewing distance for the instrument. This process of adjustment usually takes 0.3...0.5 s. Older drivers find this process strenuous and in some cases, depending on their constitution, even impossible. HUD, a technology involving projection, can eliminate this problem. Its optical system generates a virtual image at such a viewing distance that the human eye can remain adjusted to infinity. This distance begins at approx. 2 m, and the driver can read the information with very little distraction, and without having to divert his eyes from the road to the instrument cluster.

Design

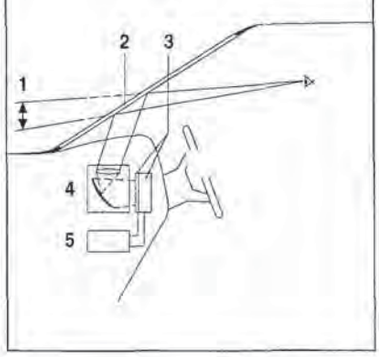
The HUD features an activated display for generating the image, a lighting facility, an optical imaging system and a "combiner", with which the image is reflected into the driver's eyes. The untreated windshield can also take the place of the combiner. Active displays (light emitters) such as CRTs or VFDs and passive displays (light modulators) are used for HUD. PDLC or DSTN displays are suitable but they do need very bright lighting because they absorb approx. 70 % of the transmitted light on account of the required polarizers.

Indication of HUD information

The virtual image should not cover the road ahead so that the driver is not distracted from the traffic or road conditions. It is therefore displayed in a region with a low road or traffic-information content.

In order to prevent the driver from being overwhelmed with stimuli in his primary field of vision, the HUD should not be overloaded with information, and is therefore not a substitute for the conventional instrument cluster. It is however particularly well-suited to displaying safety-related information such as warnings, safety distance and route directions.

Head-up display HUD *(principle)*
1 Virtual image, 2 Reflection in windshield, 3 LCD and lighting (or CRT, VFD), 4 Optical system, 5 Electronics.



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Automotive sound systems

Reception conditions

Critical FM reception conditions occur e.g. when the transmitter signal is briefly interrupted by a mountain or building (resulting in hiss) or with the delayed reception of broadcast waves which are reflected by these obstructions (resulting in distortion). Other sources of interference: Cross-talk from adjoining high-power transmitters, interference at one frequency emanating from intermodulation products of two other – more powerful – transmitters.

Technical terms

ARI: Autofahrer-Rundfunk-Information (radio information for the driver). This traffic-information system is the product of a cooperative development project involving Blaupunkt, the ADAC (the largest German motorists' association) and the German ARD radio network. It is in active use in Germany, Switzerland, Austria, and Luxembourg:

Those FM broadcasters participating in the traffic-information network constantly transmit a special identification signal (SK), which is recognized by all ARI-equipped radios. If the ARI button has been activated, the channel scanner will pause only at traffic-information stations.

The stations also transmit a supplementary regional identification (BK) code to identify the specific area for which traffic information is being broadcast. A code letter (A through F) appears in the display on radios which are capable of regional recognition (letters correspond to those found on the blue signs on German highways).

Yet another signal is transmitted for the actual traffic announcements: DK, the announcement code. This signal ensures that the driver always hears the traffic information at an appropriate volume (adjustable), quite aside from whether the radio is turned down or a cassette or CD is playing (playback mechanism pauses for the duration of the announcement).

Antenna-diversity recognition: This is a receiver system for vehicles which are equipped with several – usually two – antennas (e.g., window antennas). It's chief function is to alleviate the effects of the multipath distortion induced by reflected radio waves. As multipath interference is characterized by substantial local fluctuations in field strength (within a range of 50...100 cm), at least one of the vehicle's antennas can usually be counted on to be positioned for clear reception, provided that there is sufficient distance between them. The diversity system automatically switches over to the antenna with the best reception. Different systems are characterized by varying degrees of technical complexity.

a) Conventional scanner (1-tuner diversity): Should the reception quality at antenna "A" fall below a specified level, the unit responds by tentatively switching to antenna "B". If "B" is even worse, then the unit reverts to "A". If "B" is better, the unit remains at "B".

b) Selective process (multi-tuner diversity): Each antenna feeds to its own tuner. The system compares the reception quality at the respective antennas on a continuous, uninterrupted basis, using this information to select the antenna providing the best signal.

ADA (Auto-Directional Antenna): This is a multi-antenna reception system developed by Blaupunkt which uses a separate tuner with each antenna in a layout similar to that of the "selective process" described above. The salient difference lies in the fact that there is no alternation between antennas with ADA. Instead, the system processes all – up to four – antenna/tuner signals, modifying amplitudes and phase lengths as required for subsequent summation. A special control algorithm automatically processes amplitude and phase to ensure that minimal interference is reflected in the cumulative signal. The system can thus selectively deactivate reception from individual directions, electronically simulating the response of a directional antenna.

ADA has been designed for application

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with several window antennas and/or simple wire and foil antennas in bumpers.
ASU: Automatic interference suppression for car radios, suppresses ignition noise in the FM band.

Output power: Amplified output which the car radio or auxiliary amplifier (booster) transmits to the speakers. DIN 45324 distinguishes between sinusoidal output (continuous power with pure tone) and music output (continuous power with complex sample signal) in watts. When units on the market are compared, max. power (brief maximum output power) is generally specified. The relationship between "volume" and "output" is exponential. The increase in volume produced by doubling the output is just barely audible (3 dB). More important is the speakers' efficiency factor (volume in dB/1W/1m). While "output variations in watts" play a secondary role in practice, technical justification for the installation of ultra-high-output systems may well be present in the form of reduced low-frequency distortion (→ Automotive speakers → Subwoofers).

CD (Compact Disc): 12- or 8-centimeter sound-reproduction medium featuring signals stored in digital form and playing time of more than 60 minutes. A laser beam is directed toward a reflective information layer for contactless optical scanning. Variations in this layer (pits) dampen the reflected beam received by the prism. A photo diode monitors this information for binary transmission to a digital-analog converter. A new 16-bit amplitude value is processed 44,100 times per second, providing the D/A converter with the data to reconstitute the original sound signal "bit by bit".

DigiCeiver: New technology developed by Blaupunkt in which the radio receiver digitally processes the signal through to the prestage output. The results in significant advantages in terms of reception performance and sound quality. A constituent part of the DigiCeiver concept is Sharx, a digitally variable filter for optimizing the selectivity. Radio stations which are very close together can thus be crisply picked out from each other.

DPE (Digital Parametric Equalizing): Steep-edged two-stage equalizer with freely selectable operating frequencies and filter levels with which unwanted resonances are effectively suppressed and frequency-response incursions of speakers are linearized. The sound becomes much more homogeneous and coherent.

DSA (Digital Signal Adaptation) comprises a digital 4-channel 9-band equalizer (self-calibrating or manually adjustable) and a DNC component (Dynamic Noise Covering). DNC permanently analyzes the driving noise with an external microphone and adapts the volume and sound of the radio accordingly on a frequency-selective basis. Its operation is adjustable in three stages.

RDS (Radio Data System): Digital data-transmission system for FM broadcasters, who transmit the RDS "data telegram" in a single standardized format throughout Europe. The data, which can be interpreted by car (or home) radios equipped with RDS decoders, include the following:
- The call letters or identity of the station being received (e.g., "BAYERN 3" or "BBC Radio 1") appear on the radio's display (PS Code).
- Also transmitted is a list of alternative frequencies on which the current broadcast can also be received (AF Code). The RDS receiver responds to reductions in the quality of reception from the station to which the radio is tuned by searching through the AF list for alternative frequencies for the same station. If the first alternative provides better reception, then the unit locks onto the new frequency. If it is worse, then RDS continues to the next substitute frequency. The process is virtually inaudible under ideal conditions (depending on individual unit design).
- Each station also has an identification number (PI Code) for backup verification. The PI code must remain constant when the system selects a substitute frequency, otherwise the system registers an impermissible station change.
- Traffic-information and announcement codes (TP and TA Codes). The RDS receiver evaluates TP and TA in a process

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comparable to that employed for ARI-SK and DK (→ ARI). As most European countries are still without an ARI traffic-information system, implementation with RDS was a logical step. The present ARI system is expected to remain in place though in A, CH, D, and LUX, operating parallel to RDS, until 2005. In all North, South, and West-European countries, the stations of the national networks and most independent broadcasters can be received with the RDS services listed above.

The RDS format incorporates further options, e.g.:
- PTY Code (Program Type): Classifies up to 16 different program types according to their content, i.e., news, classical music, etc.
- "PTY 31": Control signal for superimposition of civil defense messages for the general population (similar to TA). The PTY 31 function cannot be deactivated.
- EON Code (Enhanced Other Networks): "EON" informs the receiver when a traffic announcement is being transmitted on a parallel program of a chain of stations, and can switch to this parallel broadcast for the announcement's duration.
- TMC Code (Traffic Message Channel): For transmitting standardized information bits, e.g., on traffic congestion. The information bits can be triggered for reproduction – in any language, at any time – via speech synthesizer. Navigation systems can monitor warnings on obstructions for immediate calculation of the best route for the given conditions.
- CT Code (Clock/Time): Time, date.
- RT Code (Radio Text): Text transmission for receiver with suitable display, e.g. display of music track currently being heard.

Auxiliary equipment

- **Amplifier, booster:** Separate auxiliary amplifier (→ Output power).
- **Equalizer:** Equalizers are employed to improve bass and treble response through graduated amplification and/or suppression of individual frequency ranges. The specific acoustical properties of the motor vehicle's interior result in distortions of emphasis (non-linear propagation); an equalizer can compensate for these distortions. Today, it is usually installed as a digital chip in top-line car radios → DPE, DSA.
- **CD changer:** CD player for installation in the luggage compartment/trunk. The CD changer is loaded with a disk magazine which is usually capable of holding up to 10 disks. The CDs and the individual tracks are selected using either a suitably-equipped radio or a separate remote-control unit.

Automotive speakers

The acoustical properties of the vehicle's interior, the speakers' locations and the speakers themselves all influence the quality of reproduction. The larger the speakers, the better the bass response (e.g., four x 160 mm system). Separate subwoofers are becoming popular in sound systems at the higher end of the price range; they can be hidden, e.g. below the rear tray or on the panel between passenger compartment and trunk. For optimal reproduction, subwoofer systems generally require powerful → Amplifiers.

Antennas

The effectiveness of a vehicle's antenna is directly proportional to its distance from the vehicle's major ground masses. Good reception is ensured by the telescoping rod antenna, also available as a motor-driven "automatic antenna". An alternative is the integral window or windshield-mounted antenna which is also available as an aftermarket unit for attaching to the inner surface of windshield or windows. Although it is more sensitive to direction, which puts it at a slight disadvantage relative to the rod-type unit, among its main advantages are the fact that it needs no maintenance and is unaffected by wear: In contrast to the rod antenna, the window unit causes no aerodynamic drag, and it requires no retraction space. A third alternative is a short, flexible whip antenna which is technically a compromise between the other two extremes.

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Digital Audio Broadcasting DAB

Features

The new digital broadcasting system DAB is the most significant development in broadcast engineering within the framework of the European EUREKA 147 project since the introduction of FM. It offers the following features:

- safe and interference-free reception for both mobile and portable applications,
- CD-comparable sound quality,
- economical in frequencies due to sound-data reduction and mutual broadcasting networks,
- economical in power due to suitable transmission procedures and coding,
- suitable for international, national, regional and local radio and
- assured future on account of multimedia compatibility.

The system developed by the Eureka 147 DAB project and standardized by the European Telecommunications Standards Institute (ETSI) is up to now the only system which satisfies these requirements and for which the International Telecommunications Union (ITU) has voiced a recommendation.

Typically, 5 ... 7 radio stations and several data channels are combined into an ensemble. A station ensemble, e.g. all the stations of a German ARD broadcasting network (e.g. BR, WDR), is broadcast throughout the country on a mutual network so that the stations can be received everywhere on the same frequency. Band III (175 ... 239 MHz) is designated for national broadcasting while local radio is planned in the L-band (1453 ... 1491 MHz).

Through flexible division of an ensemble, it is possible to transmit other data such as video/television or Internet pages instead of the radio stations. These DAB-system applications have been introduced under the term DMB (Digital Multimedia Broadcasting). Its highly favorable mobile reception characteristics make it possible to provide drivers with intelligent telematics services (DAB/ITS). Under the title MEMO (Multimedia Environment for Mobiles), a return channel via GSM is on offer for the DAB system. This enables the transmission of the Internet or other data services with high data rates and interactive user operation.

Principles of Eureka 147 DAB system

The DAB system is essentially based on three components:

- the sound-data reduction procedure in accordance with MPEG-1 (ISO 11172-3) or MPEG-2 (ISO 13818-3), Layer II in each case,
- the transmission procedure Coded Orthogonal Frequency Division Multiplexing (COFDM) and
- a flexible division of the transmission capacity into a variety of subchannels which, independently of each other, can transmit sound and data programs with differing data rates and with differing error-protection levels (DAB multiplex).

Audio coders are used for the data reduction of the individual radio stations. Channel coding is then employed to add the redundancy required for error correction in the receiver. Several data services can be combined in a packet multiplexer which is then also submitted to channel coding. The subchannels' channel-coded data are combined with the Fast Information Channel (FIC) data, which contain the multiplex and program-information structure, and are then COFDM-modulated. This generally takes place digitally in the form of a Fast-Fourier Transformation (FFT). The signal is converted from digital to analog, mixed to the corresponding transmit frequency, amplified, filtered and transmitted.

The antenna signal is converted to the intermediate frequency (IF) in the high-frequency stage and then filtered and digitized. The information is then recovered from the individual COFDM carriers (generally by means of an FFT again). Here, in principle only that part of the data which contains the desired programs needs to be evaluated.

After error correction (by Viterbi decoding), the data are either directed to the audio decoder, which recovers the sound signal, or made available for further use at

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an appropriate data interface. The FIC data are used to control and operate the receiver.

For sound-data reduction, Eureka 147 has settled on the procedure complying with MPEG-1 Layer II and several supplements from the follow-up standard MPEG-2 Layer II, which is sometimes also (although incorrectly) referred to as the "Musicam procedure". The advantage of this procedure is that it permits high compression factors while maintaining manageable complexity and relatively short delays in coding and decoding. With data rates of 160...256 kBit/s, for stereo transmission it provides quality which is comparable to that of a CD.

The COFDM procedure is particularly well-suited to providing reliable multichannel reception and shared-channel broadcasting by networks. The essential concept of the procedure is to divide the entire data stream into many data substreams (192...1536). Each of these substreams is transmitted on a subcarrier with a low data rate and thus a long symbol duration (magnitude 100 µs...1 ms). The symbol duration for COFDM is therefore significantly greater than the typical propagation-time differences of the received signal elements (magnitude 10...100 µs). In this way, and due to the additional protection interval, it is even possible to avoid intersymbol interference completely. Error protection coding is used to eliminate remaining bit errors.

A bandwidth of 1.536 MHz has been selected for DAB. There are four parameter records for the COFDM procedure, which facilitate the implementation of both terrestrial common-frequency networks in Band III and the L-band, as well as satellite broadcasting up to 3 GHz. The number of COFDM subcarriers amounts here to 1536, 768, 384 or 192 and the carrier spacing is accordingly 1, 2, 4 or 8 kHz.

On account of the choice of bandwidth, the DAB system offers a considerably higher data rate than is needed to transmit a radio program. For this reason, several programs and data services are combined to form a multiplex which is then transmitted. All the stations transmitted on a DAB multiplex are jointly termed an "ensemble". The ensemble multiplex is flexible, both in terms of the number and data rate of the individual subchannels and also in terms of its error protection, i.e. the redundancy added for transmission. With minimum redundancy, the bit rate of the DAB system can extend to 1.84 MBit/s while it is possible to start out from approx. 1.2 MBit/s for mobile reception.

Because the multiplex is flexible, the receivers must be informed as to how the multiplex of the DAB ensemble that has just been received is configured or when and how it will be reconfigured. This process takes place in the Fast Information Channel (FIC). All other data which is needed to operate and control the receiver, e.g. station names and alternative frequencies, are transmitted on this channel.

The combination described above of sound-data reduction, COFDM, and flexible multiplex, makes DAB an extremely versatile system whose applications can extend far beyond traditional radio broadcasting. In the end, these will lead to digital multimedia broadcasting, which can also be received in cars and with portable devices.

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Parking systems

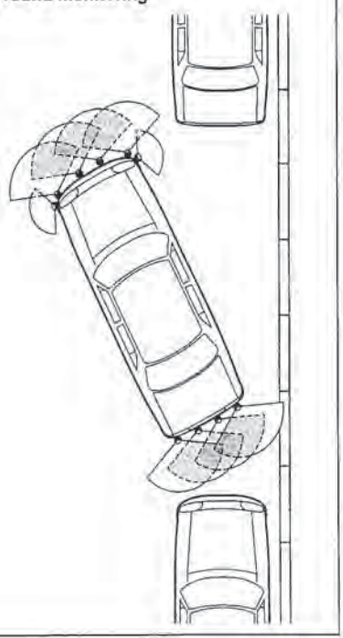
Parking aid with ultrasonic sensors

Introduction

On virtually all motor vehicles, in order to reduce fuel consumption the bodies have been designed and developed in such a way as to achieve the lowest possible drag-coefficient values. Generally speaking, this trend has resulted in a gentle wedge shape which greatly restricts the driver's view when maneuvering. Obstacles can only be poorly discerned – if at all.

Parking aids with ultrasonic sensors provide drivers with effective support when parking. They monitor an area of approx. 30 cm to 150 cm behind or in front of the vehicle. Obstacles are detected and brought to the driver's attention by optical and/or acoustic means.

Detection range of parking systems with all-round monitoring



System

The system comprises the following components: ECU, warning element and ultrasonic sensors.

On vehicles with only rear-end protection, usually 4 ultrasonic sensors are installed in the rear bumper.

Additional front-end protection is provided by installing a further 4 to 6 ultrasonic sensors in the front bumper.

The system is activated automatically when reverse gear is engaged or, in a system with additional front-end protection, when the vehicle drops below a speed threshold of approx. 15 km/h. The system's self-test function ensures that all the system components are permanently monitored during operation.

Ultrasonic sensor

Following a principle that is the same as echo depth sounding, the sensors transmit ultrasonic pulses with a frequency of approx. 40 kHz and detect the time taken for the echo pulses to be reflected back from obstacles. The distance of the vehicle to the nearest obstacle is calculated from the propagation time of the first echo pulse to be received back according to the formula:

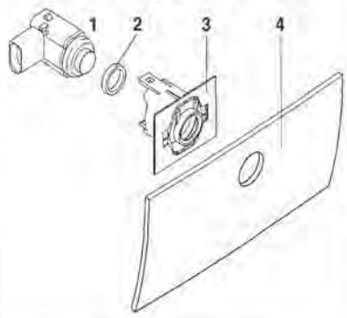
$$a = 0.5 \cdot t_e \cdot c$$

t_e Propagation time of ultrasonic signal(s)
c Speed of sound in air (approx. 340 m/s).

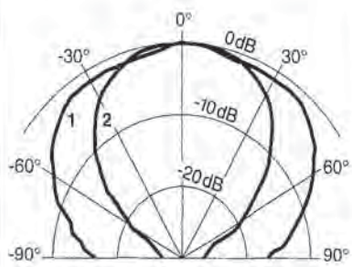
The sensors themselves consist of a plastic housing with integrated plug connection, an aluminum diaphragm on the inner side of which a piezoceramic water is attached, and a printed-circuit board with transmit and evaluation electronics. They are electrically connected to the ECU by three leads, two of which serve to supply power. The third, bidirectiona signal line is responsible for activating the transmit function and returning the evaluated received signal to the ECU. When the sensor receives a digital transmit pulse from the ECU, the electronic circuit excites the aluminum diaphragm with square-wave pulses at the resonant frequency so that it vibrates, and ultrasound is emitted. The diaphragm, which in the meantime has returned to rest, is made to vibrate again by

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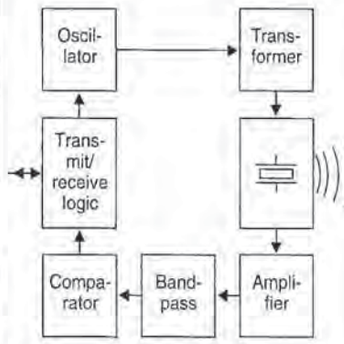
Mounting principle of ultrasonic sensor in bumper
1 Sensor, 2 Decoupling ring, 3 Installation housing, 4 Bumper.



Antenna emitting diagram of an ultrasonic sensor
1 Vertical, 2 Horizontal.



Block diagram of sensor



the sound reflected back from the obstacle. These vibrations are converted by the piezoceramic wafer into an analog electrical signal which is then amplified and converted into a digital signal by the sensor electronics.

In order to be able to cover as extensive a range as possible, the detection characteristic must fulfill special requirements. In the horizontal range, a wide detection angle is desirable. In the vertical range, however, it is necessary to have a smaller angle in order to avoid interference from ground reflections. A compromise is needed here so that obstacles can be reliably detected.

Specifically adapted mounting brackets secure the sensors in their respective positions in the bumper.

ECU

The ECU contains a voltage stabilizer for the sensors, an integrated microprocessor and all the interface circuits needed to adapt the different input and output signals. The software assumes the following functions:

- sensor activation and echo reception,
- propagation-time evaluation and calculation of obstacle distance,
- activation of the warning elements,
- evaluation of the input signals from the vehicle,
- monitoring of the system components including fault storage and
- provision of the diagnostic function.

Warning elements

The warning elements identify the distance to an obstacle. Their design is vehicle-specific, and they usually provide for a combination of acoustic signal and optical display. Both LEDs and LCDs are currently used for optical displays.

In the example of a warning element shown here, the indication of the distance to the obstacle is divided into 4 main ranges (see table).

Protection area

The protection area is determined by the range and number of sensors and by their emission characteristic.

Experience has shown that 4 sensors are sufficient for rear-end protection, and

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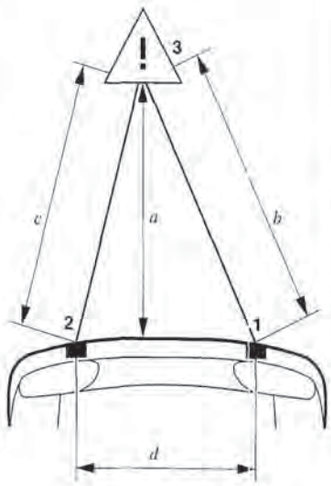
Range	Distance s	Opt. display LED	Acoustic signal
I	<1.5 m	green	Intermittent
II	<1.0 m	green + yellow	Intermittent
III	<0.5 m	green + yellow + red	Continuous
IV	<0.3 m	All LEDs flash	Continuous

4 to 6 for front-end protection. The sensors are integrated in the bumpers, which means that the distance to the ground is already fixed.

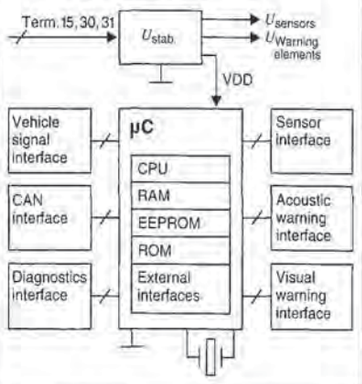
The installation angle of, and the gaps between the sensors are vehicle-specific. These data are taken into account in the ECU's calculation algorithms. At the time of going to press, application engineering had already been carried out on more than 150 different vehicle types. Thus, even older vehicles can be retrofitted.

Calculating distance to an individual obstacle *(example)*
a Distance, bumper/obstacle,
b Distance, sensor 1/obstacle,
c Distance, sensor 2/obstacle,
d Distance, sensor 1/Sensor 2.
1 Transmit and receive sensor,
2 Receive sensor, 3 Obstacle.

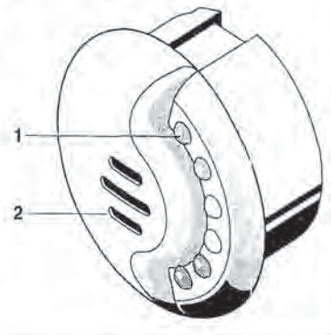
$$a = \sqrt{c^2 - \frac{(d^2 + c^2 - b^2)^2}{4d^2}}$$



Block diagram of ECU



Example of a warning element
1 LED warning lights,
2 Opening for a acoustic signal.



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Trip recorders

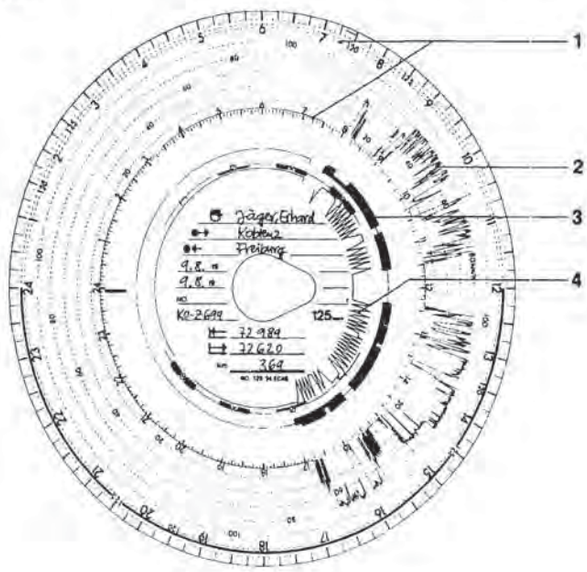
These devices record the vehicle speed, the distance covered (odometer), and the time. They also incorporate a warning lamp which is triggered when vehicles exceed a preset speed (such as the legally permitted limit, or the maximum speed commensurate with economical operation). The following data is also recorded on the tachograph chart along with the corresponding clock time: Road-speed curve, wheel time and pauses, distance covered. The trip recorder, which is designed specifically so that it can be calibrated, is mandatory for certain vehicle categories within Germany (Paragraph 57b, StVZO (FMVSS/CUR)). Various other countries also have specific national regulations governing the use of trip recorders.

Mode of operation

The EC tachograph is a special form of trip recorder which incorporates an auxiliary feature allowing differentiated recording of wheel time and test periods (time-group monitor); different versions are available, for either one driver or two. The EC tachograph meets all the requirements set forth in EEC Directive 3821/85, i.e., the EC tachograph charts serve as the driver's official daily record. Within the countries of the European Union, the EC tachograph is obligatory equipment on specific types of vehicle, where it monitors compliance with the time behind the wheel and the rest periods prescribed in EEC Directive 3820/85. The supplementary features on the EC tachograph include a clock-function display and an LED which confirms to the driver that the charts are installed and that all styluses are functioning properly.

Some tachographs also display and record engine speed. Operation control and further parameters can also be recorded when the appropriate options

EC tachograph chart
1 Clock time scales, 2 Speed recordings, 3 Driving and stationary times (time group recording), 4 Distance traveled.



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are installed. These include the two-stage auxiliary stylus (e.g., for recording fuel consumption) and contacts which trigger other warning and control devices in response to vehicle and engine speed.

Trip recorders and EC tachographs must maintain the strict operating tolerances defined in the German Calibration Regulations, in Paragraph 57b of the StVZO (FMVSS/ CUR) and in EEC Directive 3821/ 85. These regulations also extend to govern periodic inspection of tachograph and related equipment by authorized facilities.

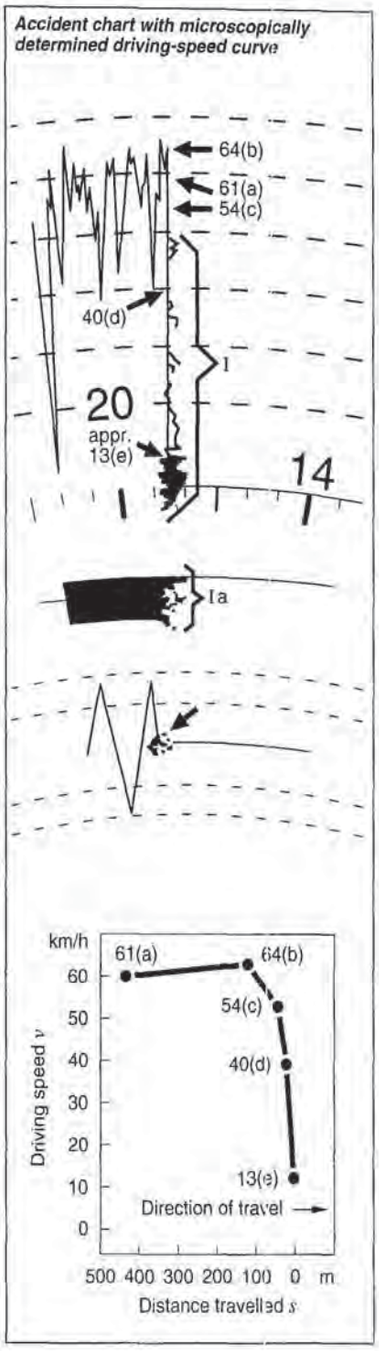
The tachograph chart can be interpreted using any of several methods, including visual, electronic and microscopic evaluation. Visual evaluation is the simplest method, as the chart disk makes it possible to appraise and check an entire day's operation at a glance. Systematic visual evaluation embraces the following individual checks: Written entries, working time, breaks, rest periods, assessment of driving style, fuel consumption and engine speed, as well as manipulated or incorrect recordings. Microscopic examination uses a special microscope to analyze the recording down to a precision range measured in seconds and meters. The data gathered can be entered in a time/distance graph (see illustration) for precise reconstruction, e.g., of the events preceding an accident.

Within the framework of a fleet-management system, tachograph charts can also be subjected to semi- or fully automatic evaluation and the data processed in a computer.

Pulse drive is employed on tachographs featuring electronic monitoring systems. It runs on signals produced by a pulse generator, installed either within the vehicle or at the transmission's speedometer-drive gear, which converts the mechanical rotation of the speedometer pinion into electronic pulses.

The most modern electronic trip recorders and EC tachographs incorporate an integral conversion feature to adapt the unit for variations in the number of pulses per kilometer traveled. No extraneous devices are required. On vehicles with variable-ratio rear axles, a conversion-gear unit modifies the ratio of the tachograph's drive signal to maintain synchronization with vehicle conditions.

Accident chart with microscopically determined driving-speed curve



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Navigation systems

Navigation systems have become extremely popular in recent years. Initially, the available systems were mainly intended for retrofitting. Since then, navigation systems are mainly fitted as optional extras on new cars and are no longer the sole preserve of a few luxury-sedan models. This development has enabled such systems to be better integrated in the vehicle. Sensors can be jointly used by various vehicle systems. Displays in the instrument cluster place important route-guidance information in the driver's primary field of vision.

Some vehicle manufacturers have even incorporated navigation as part of a complete driver information system with audio and telephone functions. This development will continue.

Simple systems which offer the driver no more than a mere indication of his/her position on a card and a straight-line direction to destination, have for the most part disappeared from the market.

Common to all systems is the combination of the basic functions of "positioning", "destination selection", "route computation" and "route guidance". Systems in the upper performance spectrum also offer a color map display. All the functions require a digital map of the road network and this is generally stored on a CD-ROM.

Orientation

Position location

The position is located by means of composite positioning whereby the route elements are added cyclically according to amount and angle. This process involves the accumulation of errors but these are compensated for by permanent comparison of the position with the road in the digital map (map matching). Thanks to the GPS satellite positioning system, the navigation system can still be operated without any problems after short trips outside the digitized road network or after the vehicle has been transported by sea or rail.

Sensors

For positioning purposes, it was frequently the case that two wheel sensors were used to determine the travel and changes of direction, and a geomagnetic sensor was used to determine the absolute direction of travel. Essentially, the GPS positioning system served to correct heavy sensor interference or to locate the correct point of reentry into the stored road network after trips outside the range of the digital map. More modern systems only need a simple travel signal, as is already often available for speed-dependent volume control of car radios. The change of direction is determined by a yaw-rate sensor (gyrometer). The geomagnetic sensor has become superfluous as it has now become possible to determine the absolute direction of travel via the Doppler effect from the GPS signals.

Destination entry

Directories

The digital CD map contains directories so that a destination can be entered as an address. Lists of all the locations (towns, villages) available on the CD are required for this purpose. In turn, all locations are allocated lists containing the names of the stored roads/streets. For further precise pinpointing of destinations, it is also possible to select road/street intersections or street numbers. However, the availability of house numbers is still very limited due to the prohibitive expenditure involved in ascertaining such data. Drivers rarely know the addresses of destinations such as airports, rail stations, service stations, car parks and many more. For this reason, there are classified directories listing these destinations, which are also frequently called points of interest (POIs). With these directories, it is even possible for instance to find a filling station in the vicinity of one's vehicle. Marking a destination in the map display or calling up destinations which were previously stored in a destination memory are further selection options.

Travel guides

As a logical development of selecting POIs, CDs are now available with travel

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guides as the result of joint cooperation between publishing companies and the manufacturers of the digital maps. For instance, these CDs can be used to look up hotels in the vicinity of the destination. Information on size, prices and facilities of the POIs are also contained on the CDs.

Route computation

Destination selection
It must be possible to adapt the route computation to the driver's wishes. Settings for optimizing the route in accordance with driving time or mileage are available in many systems. Route computation should take care of such assignments as avoiding freeways/motorways, ferry connections and toll roads. Initial computation of the route once the destination has been selected must be completed inside half a minute. If the driver departs from the recommended route, it is critical for the system to recompute the route still more quickly. The option to request an alternative route at the touch of a button is helpful in that it enables the driver to bypass traffic congestion ahead.

Dynamic routing
Automatic bypassing of traffic congestion is already possible through evaluation of RDS-TMC-coded traffic messages. These can be received by way of RDS or GSM.

Route guidance

Determination of route
Route guidance is performed by comparing the present position with the computed route. From the sequence of the road section just covered and the roads still to be covered on the route, the system can decide whether the driver must make a turn-off maneuver or simply continue along the present road.

Route and direction recommendation
The route and direction recommendations are first and foremost issued acoustically. This enables the driver to follow the directions without being distracted from his/her job of driving. Simple graphics, if at all possible in the primary field of vision (instrument cluster), support the driver in understanding the directions. The conciseness of these acoustic and graphic directions is the decisive factor in the quality of route guidance. Identifying the direction to be taken from a map in the display is not considered a primary option as this distracts the driver.

Dynamic routing
Traffic congestion is automatically bypassed by evaluating the coded traffic messages which are transmitted inaudibly over the Traffic-Message Channel of the Radio Data System (RDS-TMC) parallel to the radio program. They are already available in GSM services.

Map display

Depending on the system, the map can be displayed on a color monitor with a scale ranging between approx. 1:8000 and 1:16 million. It is helpful depending on the scale, in maintaining an overview of the route in the vicinity or on a more far-ranging scale. Driver orientation is supported by the provision of topographical background information showing such things as lakes, rivers and canals, built-up areas, railroads and forests.

Road-map memory

The CD-ROM has become the primary medium for storing road maps. The structures of the data stored on them represent manufacturer-specific know-how and significantly influence system performance. For this reason, the CDs presently on the market for systems of different manufacturers are not compatible with each other. Efforts are being made however to standardize navigation CDs.

With the size of the digitized areas, the completeness of recording the road network and the increasing wealth of additional information, the capacity of the CD-ROM (which initially seemed so great) will be exhausted in the foreseeable future. The Digital Versatile Disc (DVD), with many times the capacity of the CD-ROM, has already been marked out as its successor.

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Traffic telematics

The word "telematics" is derived from the contraction of the words "telecommunications" and "informatics". Accordingly, the transmission of traffic-related information from and to motor vehicles, and its subsequent, mostly automatic evaluation is combined under the term "traffic telematics".

Transmission paths

Today, the main transmission paths for telecommunications are provided by radio and mobile radiocommunications networks.

Radio only allows a path into the vehicle and does not serve to communicate individual messages. With GSM, information can be exchanged in both directions between vehicles and control centers run by service providers. The amount of information exchanged is restricted in each case by the bandwidth of the available transmission channels. It is therefore essential that the coding of standardized message content is as free as possible of redundant information.

Standardization

In the case of reports of traffic holdups, the standardization of content refers for instance to the type of holdup (such as "jam", "road block"), to the causes (such as "accident", "black ice") and to its extent and expected duration. Coding of the location of a traffic holdup is also necessary.

While a person can interpret spoken messages with his/her geographical experience and e.g. the signposts in his/her surroundings, this option is not available for computer-controlled evaluation of information.

The coding of the designations for geographical regions, longer freeway/motorway sections (segments), and individual road points (locations) such as freeway/motorway junctions and intersections, has been standardized to permit transmission over the Traffic Message Channel of the Radio Data System (RDS-TMC).

For traffic-telematics applications using the Short Message Service (SMS) in GSM networks, service providers have drawn up the Global Automotive Telematics Standard (GATS), which is used for breakdown and emergency calls, information services and traffic-data acquisition in accordance with the Floating Car Data principle described below.

Selection

One of the strengths of traffic telematics is its ability to select from a frequently very large stream of information those messages which are of relevance to the driver. For instance, by evaluating the vehicle's position, microcomputers in the vehicle can determine which traffic message could affect the vehicle's route.

Decoding of traffic messages

Decoding and selection of TMC-coded traffic messages are today already integrated features in 1-block car radios, such as the Blaupunkt "VIKING TMC 148". The driver can enter up to 5 freeways/motorways, national highways or regions for which a selection of messages is to be performed. These are then converted by a voice-output chip into audible messages. Because the messages conform to standardized information groups, there is also no problem in converting them into different languages.

Telematics services

Devices such as the Blaupunkt "Gemini GPS 148" have a GSM module and a receiver for the Global Positioning System (GPS). As well as normal telephone functions, the GSM module also facilitates the bidirectional transmission of messages via SMS. The GPS module enables the vehicle's position to be pinpointed to an accuracy of approx. 100 m. Such devices can make use of telematics services such as "traffic information", "breakdown calls"

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and "calls for assistance".

Traffic messages can be selected as a function of the vehicle's actual location and direction of travel, and also through selection of freeways/motorways and national highways. In the event of a breakdown, the driver can establish a telephone link to a control center at the touch of a button. At the same time, the vehicle's position is transmitted via SMS.

For future GSM services, the functions of the devices can be extended by loading new software.

Dynamic route guidance

The most extensively automated evaluation of traffic information is performed for the purposes of dynamic route guidance. Thanks to standardized coding of locations (see "Standardization"), disturbances and their extent and expected duration, computers in route-guidance systems can assess the effect of a traffic holdup on the driver's route and then assess whether there is a better alternative route.

In this case, the driver is instructed that the route has been recalculated on account of traffic messages. Subsequent route directions follow in accordance with the new route.

For dynamic route guidance, it is necessary for the CDs, which usually carry the digital map for route-guidance systems, to include a reference table covering the locations of the traffic messages.

The traffic messages are received either through an FM receiver with RDS decoder or through SMS from GSM networks.

Systems with dynamic route guidance have been available since 1998, and have rapidly become popular because of their comfort and convenience for the driver.

Information collection

The benefit to drivers presented by traffic telematics can only be as good as the information used. In particular, the up-to-dateness of the messages has not been without problems.

Information collection by way of road infrastructure

For many years now, information about the traffic flow on important stretches of road has been recorded by induction loops inserted in the roads themselves. Using these loops, the number and speed of the vehicles passing over the loops can be ascertained, and the figures used to calculate the traffic density (vehicles per km) and traffic intensity (vehicles per hour). There has been a marked step-up in the installation of such loops in recent years, but this is a time-consuming and expensive process.

Alternatively, GSM service providers install sensors on freeway/motorway bridges. These are powered by solar cells and transmit their information without electric cables. They are simple and cheap to install, but due to their system of measurement they are limited to counting vehicles and roughly classifying their speed.

Floating Car Data

A further option for collecting traffic information involves the Floating Car Data principle.

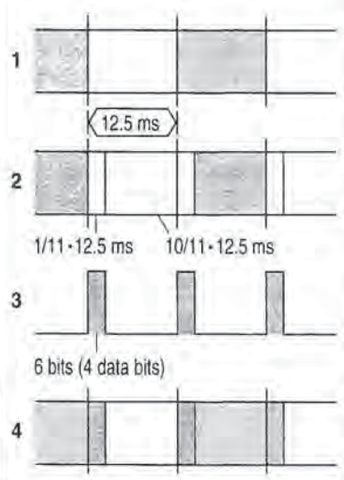
A car "floats" along in the flow of traffic and transmits its position and speed cyclically to a control center. In the control center, statistical analysis of the individually collected data is employed to generate current messages on the traffic situation. For this method, all vehicles have a bidirectional connection (such as SMS). However, the precondition for the collection of statistically reliable data is that there is a high percentage of appropriately equipped vehicles on the road. Because this is not yet the case, this procedure has up to now only been used in field tests. Since the vehicle percentage was still not high enough in the field tests, final proof of effectiveness is still to come.

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Mobile radio

Mobile radiocommunications systems can be divided into two categories: Public radio networks (car phones, etc.) and private mobile radio (or PMR). The public radio networks are – like their land-line counterparts – universal-access systems. In contrast, access to a PMR system is restricted to specific groups. Mobile radio networks consist of mobile transmitter/receiver units (mobile stations) and the requisite infrastructure (system administration, stationary units and base stations). Attempts to economize on the number of necessary frequencies have led to the establishment of cellular networks; these make it possible to use the same frequency repeatedly in numerous cells, provided that the distances between the cells are sufficient. The precise number of stationary transmitters employed in a single network depends upon the size of the area being served and/or the number of subscribers. Mobile radio units fall into three categories: Vehicle units, portable radios and hand-held devices.

C-Net communications control
1 Speech, 2 Compressed speech, 3 Digital telegram, 4 Compressed speech & digital telegram.



Cellular telephone networks

Communications within the mobile telephone network are always established via a stationary base station; the network is not restricted to providing communications between mobile units, and access to and from the standard land-line telephone system is also possible. At present, there are (technically different) analog cellular telephone networks and digital systems (D-Net: GSM, E-Net) in operation in Europe.

C-Net

This radio-telephone network operates in Germany in the 450 MHz band.

The speech transmission utilizes analog modulation. The communications path between two units is established and monitored with digital telegrams relayed on special organization channels. The system administration facility continually monitors existing communications links on the oral-communications channels by slotting digital telegrams into gaps in the speech transmission. The speech transmission is subdivided into blocks of 12.5 ms. Before transmission, these blocks are compressed into time units of 10/11 x 12.5 ms. The digital telegrams are inserted into the resulting gaps of 1/11 x 12.5 ms. Upon arrival at the receiver, the telegram is filtered from the transmission signal and the speech is re-expanded to its original value of 12.5 ms. In recent years, there has been a drop-off in the number of users in favor of the digital networks.

D- and E-Net

The Europe-wide digitized cellular mobile-telephone system GSM (Global System for Mobile Communications) in the 900 MHz range (E-Nets in the 1.86 GHz range) is intended to overcome the compatibility problems of the various European analog telephone systems. In contrast to the analog telephone networks, in the D- and E-Net the speech is transmitted in digital form. Here, 8 speech channels each with 16 kbit/s, or 16 speech channels with 8 kbit/s, are superimposed upon a high-frequency channel with a bandwidth of 200 kHz. Speech-channel

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access inside the high-frequency channel utilizes the TDMA (Time-Division Multiple Access) method.

Private mobile radio systems (PMR)

These are mobile-radio networks designed specifically to meet the individual operational requirements of given customer groups. The essential object is to provide total radio coverage within a specific, well-defined area, thus ensuring consistent access to the mobile stations. For analog PMR systems, the typical times for establishing a link are below 1 second, whereas in special operating modes ("open channel") no time at all is needed to establish a link.
The most important subscriber groups are:
- Electric utilities,
- Public-transport authorities,
- Official agencies and security organizations,
- Common-frequency subscribers,
- Taxi and rental-car agencies.

Various analog and digital signaling procedures are employed to administer the communications links. Selective-call networks allow the user to call a specific individual station. The most important signaling procedures are the 5-tone sequence and the digital calling methods. The five digits of the 5-tone sequence contain the code for the station address of the party being called, with each digit being assigned a value of 0...9. Each number represents a frequency. The transmission time for each number is 70 ms.

Upon receiving and processing the 5-tone sequence, the target station transmits an acknowledge signal in the form of its own 5-tone sequence, while a call tone is also generated to alert the caller. This establishes the communications link.

Systems employing a digital calling system represent a further development of the tone-sequence method. The data in the "call" and "acknowledge" telegrams are contained in binary codes. Frequencies of 1,800 Hz and 1,200 Hz are assigned to the bits "0" and "1" respectively. The advantage over the tone-sequence method lies in the fact that the digital telegram can relay additional status information over and beyond the target address.

Trunked radio networks

In order to eliminate the increasing frequency-crowding which was plaguing the PMR networks, the German Post Office's TELEKOM division set up trunked radio networks in the 400 MHz band. The object is to increase utilization efficiency through common use and administration of several radio channels. Under the same conditions, the system accommodates more users on each channel than a conventional network. It provides good access within the trunked radio network and exclusive communications links. The possibility of dialing from the mobile station into the public land-line telephone system should be mentioned.

The link is administered through a special radio frequency within the frequency group (organization channel), as defined by Specification MPT 1327 (MPT, Ministry of Post and Telecommunications). The digital trunked radio network TETRA-25 (Trans-European Trunked Radio) has been specified by the European Telecommunications Standards Institute (ETSI). This TDMA (Time Division Multiplex Access) procedure uses a channel grid of 25 kHz each with four time slots in the 400 MHz range. The bit rate is 36 kbit/s (overall) per carrier frequency

Another digital trunked radio network – TETRAPOL – uses the FDMA (Frequency Division Multiplex Access) procedure with a narrow-band channel grid of 10 or 12.5 kHz. The bit rate here is 8 kbit/s (overall) per carrier frequency

The European railways are planning a radio-transmission procedure based on GSM technology (GSM 2+, GSM-Railway) for their radiocommunications operations.

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Vehicle information system

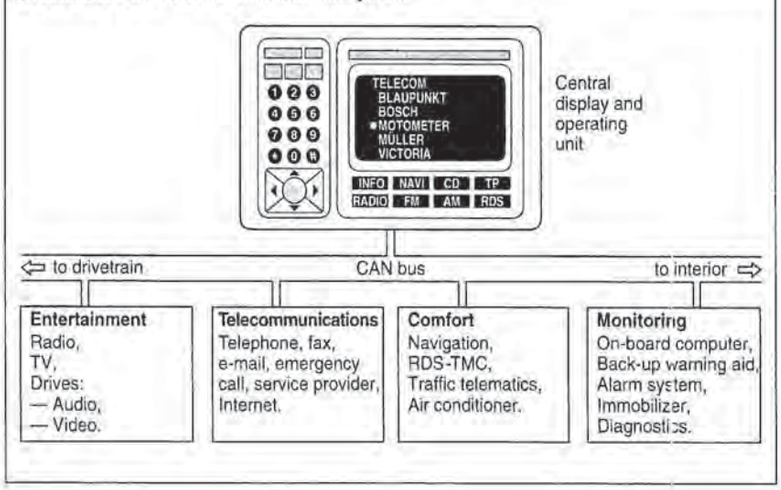
In addition to the display and control elements for the predominantly vehicle-related functions, there is an increasing array of information, communication and comfort applications in a modern-day motor vehicle. The radio is virtually a standard feature, and telephone, navigation system etc. are following this trend. Each of these further applications would normally require its own display, specially configured controls and a different procedure for its operation. This agglomeration of controls and displays would place additional burdens on the driver and, under certain circumstances, would no longer satisfy future requirements for comfort and road safety. The motor-vehicle information system provides the driver with a standardized "user interface" for an assortment of different applications.

This system combines the display and operating elements from several applications within a single display and operating unit. A substantial reduction in the number of input and output elements facilitates implementation of an ergonomically satisfactory control layout. In the face of increasing complexity, the vehicle information system furnishes an opportunity to design clear, convenient vehicle instrumentation, an endeavor in which traffic safety is one of the chief beneficiaries. The display and operating unit maintains mutual (bidirectional) communication with the connected components via a bus system (e.g. CAN) to control and indicate information in the display.

The most important control functions are mainly input via input elements which can be located by the driver "without looking" within his/her immediate reach and also via control elements on the steering wheel. More extensive programming tasks (such as storing automatic-dialing codes for telephone numbers) can be performed by remote control; safety considerations dictate that this type of programming is only carried out with the vehicle stationary.

The central display serves to indicate the most varied of image information such as texts, pictures and videos. Information that is important to the driver while driving (e.g. name of the traffic program station being received, or a direction arrow as a navigation instruction) can be shown in a display in the instrument cluster. A voice-output facility can provide the optical display with additional support. In the future, a voice-input facility will assist the driver in any number of system functions.

Structure of a motor-vehicle information system



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Safety systems

Safety in motor vehicles

Active and passive safety

Active safety systems help prevent accidents and thus make a preventive contribution to safety in road traffic. One example of an active driving safety system is the Antilock Braking System (ABS) with Electronic Stability Program (ESP) from Bosch, which stabilizes the vehicle even in critical braking situations and maintains steerability in the process.

Passive safety systems serve to protect the occupants against serious or even fatal injuries. An example of passive safety are the airbags, which protect the occupants following an unavoidable impact.

Occupant safety systems

Seat belts and seat-belt tighteners

Function

The function of seat belts is to restrain the occupants of a vehicle in their seats when the vehicle hits an obstacle.

Seat-belt tighteners improve the restraining characteristics of a three-point inertia-reel belt and increase the protection against injury. In the event of a frontal impact, they pull the seat belts tighter against the body and thus hold the upper body as closely as possible against the seat backrest. This prevents excessive forward displacement of the occupants caused by mass inertia (see Fig.).

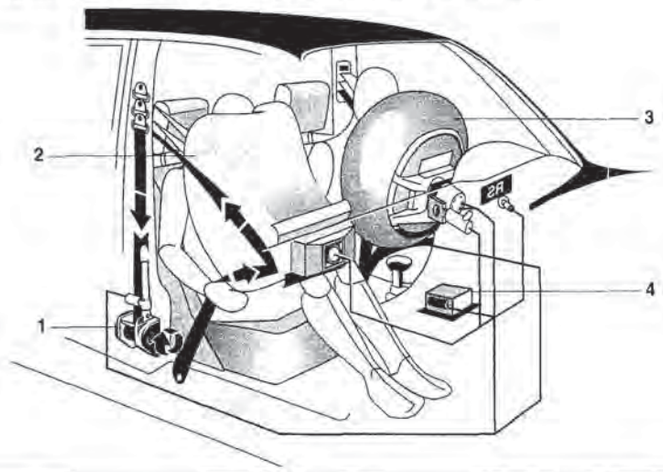
Operating concept

In a frontal impact with a solid obstacle at a speed of 50 km/h, the seat belts must absorb a level of energy comparable to the kinetic energy of a person in free fall from the 4th floor of a building.

Because of the belt slack, the belt stretch and the delayed effect of the belt retractor ("film-reel effect"), three-point inertia-reel belts provide only limited protection in frontal impacts with solid obstacles at speeds of over 40 km/h because they can no longer safely prevent the head and body from impacting against the steering wheel or the instrument panel. An occupant experiences extensive forward displacement without restraint systems (see Fig. on next page).

In an impact, the shoulder belt tightener compensates for the belt slack and the "film-reel effect" by retracting and tighten-

Occupant protection systems with belt tighteners and front airbags
1 Belt tightener, 2 Front airbag for passenger, 3 Front airbag for driver, 4 ECU.



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ing the belt strap. At an impact speed of 50 km/h, this system achieves its full effect within the first 20 ms of the impact; and thus supports the airbag which needs approx. 40 ms to inflate completely. The occupant continues to move forward slightly until making contact with the deflating airbag and in this manner is protected from injury.

A prerequisite for optimum protection is that the occupants' forward movement away from their seats remain minimal as they decelerate along with the vehicle. This is achieved by triggering the belt tighteners immediately upon initial impact to ensure that safe restraint of the occupants in the front seats starts as soon as possible. The maximum forward displacement with tightened seat belts is approx. 1 cm and the duration of mechanical tightening is 5...10 ms.

On activation, a pyrotechnical propellant charge is electrically fired. The explosive pressure acts on a piston, which turns the belt reel via a steel cable in such a way that the belt rests tightly against the body (see Fig.).

Variants

In addition to the above-mentioned shoulder-belt tighteners for retracting the belt reel, there are variants which pull the belt buckle back (buckle tighteners) and thus simultaneously tighten the shoulder and lap belts. The restraining effect and the protection afforded against occupants sliding forward beneath the lap belt ("submarining effect") are improved still further by buckle tighteners. The tightening process in these two systems takes place in the same period of time as for shoulder-belt tighteners.

Mechanical belt tighteners are also available in addition to the pyrotechnically triggered versions. In the case of a mechanical tightener, a mechanical or electrical sensor releases a pretensioned spring, which pulls the belt buckle back. The sole advantage of these systems is that they are cheaper.

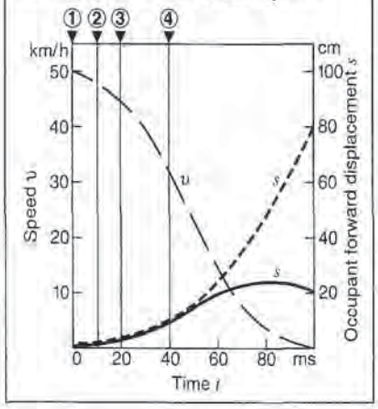
Further developments

The tightening performance of pyrotechnical seat-belt tighteners is continually being improved: "high-performance tighteners" are able to pull back an extended belt approx. 18 cm in length in roughly 5 ms.

Deceleration to standstill and forward displacement of an occupant at an impact speed of 50 km/h

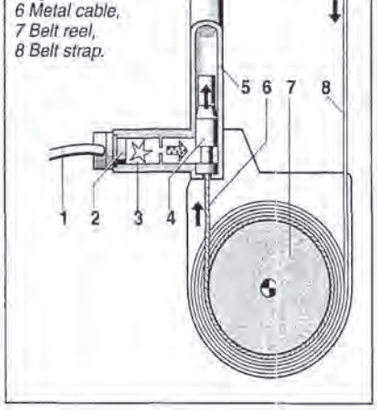
① Impact, ② Firing of belt tightener/airbag, ③ Belt tightened, ④ Airbag inflated.

– – – without/ —— with restraint systems.



Shoulder-belt tightener

1 Ignition cable,
2 Firing element,
3 Propellant charge,
4 Piston,
5 Cylinder,
6 Metal cable,
7 Belt reel,
8 Belt strap.



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Front airbag

Function

The function of front airbags is to protect the driver and the front passenger against head and chest injuries in a vehicle impact with a solid obstacle at speeds of up to 60 km/h. In a frontal impact between two vehicles, the front airbags afford protection at relative speeds of up to 100 km/h. A belt tightener alone cannot prevent the head from hitting the steering wheel in response to severe impact. In order to fulfill this function, depending on the installation location, vehicle type and structure-deformation response, airbags have different filling capacities and pressure build-up sequences adapted to the specific vehicle conditions.

In a few vehicle types, front airbags also operate in conjunction with "inflatable knee pads", which safeguard the "ride down benefit", i.e. the speed decrease of the occupants together with the speed decrease of the passenger cell. This ensures the rotational forward motion of the upper body and head which is actually needed for optimal airbag protection, and is of particular benefit in countries where seat-belt usage is not mandatory.

Operating concept

To protect driver and front passenger, pyrotechnical gas inflators inflate the driver and passenger airbags in pyrotechnical, highly dynamic fashion after a vehicle impact detected by sensors. In order for the affected occupant to enjoy maximum protection, the airbag must be fully inflated before the occupant comes into contact with it. The airbag then responds to upper-body contact with partial deflation in a response pattern calculated to combine "gentle" impact-energy absorption with non-critical (in terms of injury) surface pressures and decelerative forces for the occupant. This concept significantly reduces or even prevents head and chest injuries.

The maximum permissible forward displacement before the driver's airbag is fully inflated is approx. 12.5 cm, corresponding to a period of approx. 10 ms + 30 ms = 40 ms after the initial impact (at 50 km/h with a solid obstacle) (see Fig. "Deceleration to standstill"). It needs 10 ms for electronic firing to take place and 30 ms for the airbag to inflate.

In a 50 km/h crash, the airbag takes approx. 40 ms to inflate fully and a further 80...100 ms to deflate through the deflation holes. The entire process thus takes little more than a tenth of a second, i.e. the batting of an eyelid.

Impact detection

Optimal occupant protection against the effects of frontal, offset, oblique or pole impact is obtained through the precisely coordinated interplay of electrically fired pyrotechnical front airbags and seat-belt tighteners. To maximize the effect of both protective devices, they are activated with optimized time response by a common ECU (triggering unit) installed in the passenger cell. The ECU's deceleration calculations are based on data from one or two electronic acceleration sensors used to monitor the decelerative forces that accompany an impact. The impact must also be analyzed. A hammer blow in the workshop, gentle pushing, driving over a curbstone or a pothole should not trigger the airbag. With this end in mind, the sensor signals are processed in digital analysis algorithms whose sensitivity parameters have been optimized with the aid of crash-data simulations. Depending on the impact type, the first trigger threshold is reached within 5...60 ms. The acceleration characteristics, which are influenced for instance by the vehicle equipment and the body's deformation performance, are different for each vehicle. They determine the setting parameters which are of crucial importance for the sensitivity in the analysis algorithm (computing process) and, in the end, for airbag and belt-tightener firing. Depending on the vehicle-manufacturer's production concept, the trigger parameters and the extent of vehicle equipment can also be programmed into the ECU at the end of the assembly line ("end-of-line programming" or "EoL programming").

In order to prevent injuries caused by airbags or fatalities to "out-of-position" occupants or to small children in Reboard child seats, it is essential that the front airbags are triggered and inflated in ac-

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cordance with the particular situations. The following improvement measures are available for this purpose:

1. Deactivation switches. These switches can be used to deactivate the driver or passenger airbag. The airbag function states are indicated by special lamps.

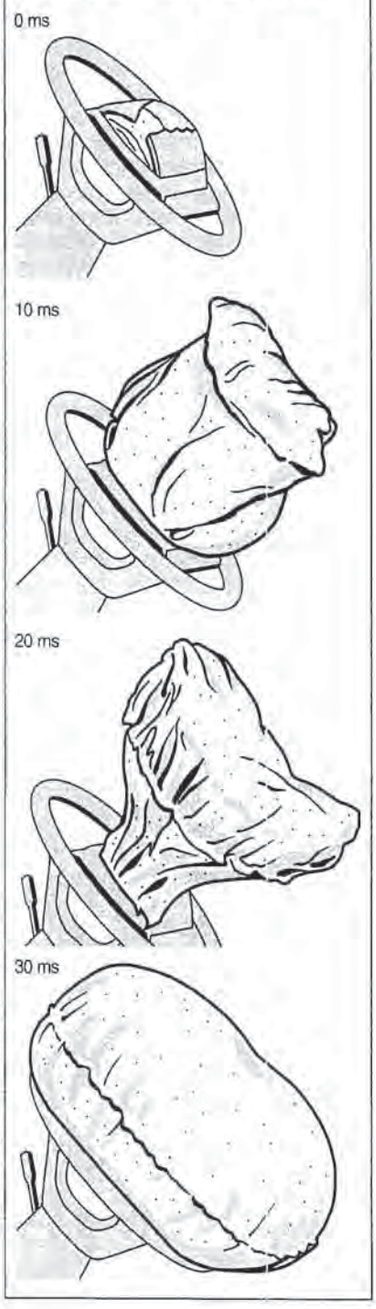
2. In the USA, where there have been approx. 130 fatalities caused by airbags, attempts are being made to reduce aggressive inflation by introducing "depowered airbags". These are airbags whose gas-inflator power has been reduced by 20...30%, which itself reduces the inflation speed, the inflation severity and the risk of injury to "out-of-position" occupants. "Depowered airbags" can thus be depressed more easily by large and heavy occupants, i.e. they have a reduced energy-absorption capacity. It is therefore essential – above all with regard to the possibility of severe frontal impacts – for the occupants to fasten their seat belts.

3. "Intelligent airbag systems". The introduction of improved sensing functions and control options for the airbag inflation process, with the accompanying improvement of the protective effect, is intended to result in a step-by-step reduction in the risk of injury. Such improvements of function are:

- Impact-severity detection through improvement of the triggering algorithm or through the use of one or two upfront sensors installed in the vehicle's crumple zone (e.g. on the radiator cross-member). These are acceleration sensors which facilitate early detection of impacts that are difficult to sense centrally, e.g. ODB (Offset Deformable Barrier crashes, offset against soft crash barriers), pole or underride impacts. It is also possible to detect the impact energy with these sensors:
- Selt-belt usage detection.
- Occupant presence, position and weight detection.
- Seat-position and backrest-inclination detection.
- Introduction of up to ten and more different triggering thresholds (seat-belt tightener

  stage 1 and stage 2; driver's airbag
  stage 1, unbelted; driver's airbag
  stage 2, unbelted; driver's airbag

"Dynamic" inflation of a driver's airbag



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stage 1, belted; driver's airbag
stage 2, belted; passenger airbag
stage 1, unbelted; passenger airbag
stage 2, unbelted; passenger airbag
stage 1, belted; passenger airbag
stage 2, belted; rear-end impact stage 1 and stage 2).

- Use of airbags with two-stage gas inflators or with a single-stage gas inflator and pyrotechnically activated gas-discharge valve.
- Use of seat-belt tighteners with occupant-weight-dependent belt-force limitation.
- Including the occupant protection system in a CAN bus network for the communication of data (diagnostic information, warning-lamp activation etc.), and for the synergetic use of sensor data from other systems (driving-speed, brake-actuation, buckle-switch and door-contact information).

The "crash output" is used for emergency calls following a crash and to activate "secondary safety systems" (hazard-warning system, opening of central locking system, shutdown of fuel pump etc.) (see Fig. "Combined control unit").

Side airbag

Function

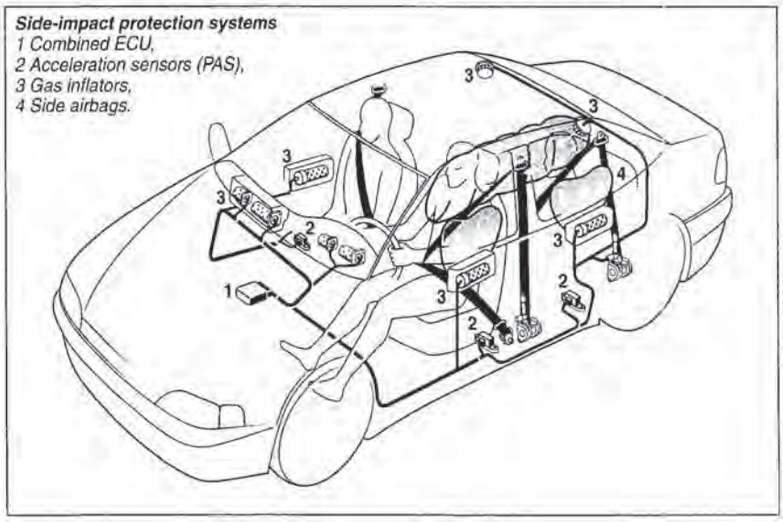
Side impacts make up approx. 20 % of all accidents. This makes the side collision the second most common type of impact after the frontal impact. An increasing number of vehicles are therefore being fitted with side airbags in addition to seat-belt tighteners and front airbags. Side airbags, which in order to protect the head and upper body inflate across the length of the roof lining (e.g. inflatable tubular systems, window bags, inflatable curtains) and from the door or the seat backrest (thorax bags), are designed to cushion the occupants gently and thus protect them against injury in the event of a side impact (see Fig.).

Operating concept

Due to the lack of a crumple zone, and the minimum distance between the occupants and the vehicle's side structural components, it is particularly difficult for side airbags to inflate in time. In the case of severe impacts, therefore, the time needed for impact detection and activation of the side airbags must be approx. 3...5 ms and the time needed to inflate the approx. 12 l thorax bags must not exceed 10 ms.

Bosch offers the following option for satisfying the above requirements: an in-

Side-impact protection systems
1 Combined ECU,
2 Acceleration sensors (PAS),
3 Gas inflators,
4 Side airbags.



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strument-cluster ECU which processes the input signals of peripheral (mounted at suitable points on the body), side-sensing acceleration sensors, and which can trigger side airbags as well as the belt tighteners and the front airbags.

Components

Acceleration sensors

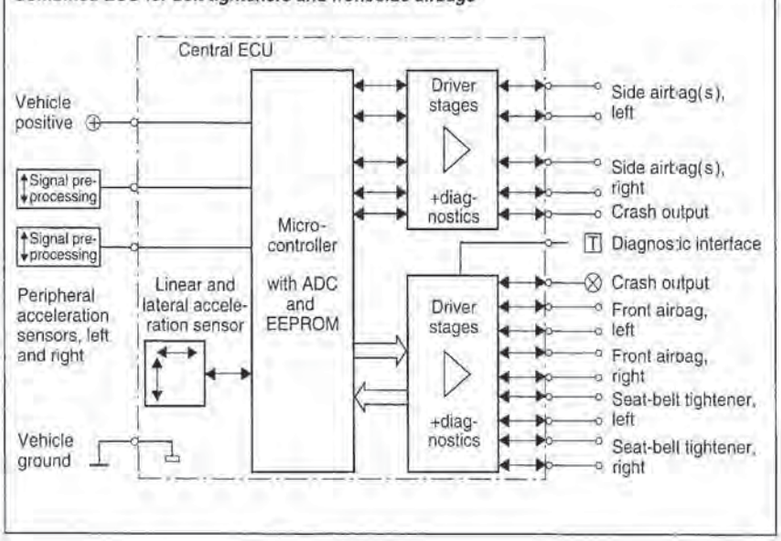
Acceleration sensors for impact detection are integrated directly in the ECU (belt tightener, front airbag) and mounted at selected points on the left and right body sides (side airbag) or in the vehicle's front-end deformation area (upfront sensors for "intelligent airbag systems"). The precision of these sensors is crucial in saving lives. They are generally surface-micromechanical sensors consisting of fixed and moving finger structures and spring pins. A special process is used to incorporate the "spring/mass system" on the surface of a silicon wafer. Since the sensors only have low working capacitance (≈ 1 pF), it is necessary to accommodate the evaluation electronics in the same housing so as to avoid stray-capacitance and other forms of interference.

Combined ECUs for belt tighteners and front/side airbags

The central ECU, also called the triggering unit, incorporates the following functions (current status):

- Impact detection by acceleration sensor and safety switch, or by two acceleration sensors with no safety switch (redundant, fully electronic sensing).
- Prompt activation of front airbags and belt tighteners in response to different types of impact in the vehicle longitudinal direction (e.g. frontal, oblique, offset, pole, rear-end). Here the acceleration is recorded at a central point in the passenger compartment and evaluated by the triggering algorithm.
- For the side airbags, the ECU operates in conjunction with a central lateral sensor and two or four peripheral acceleration sensors. These are attached to the seat cross-members, to the B-pillars, or to the B- and C-pillars. The peripheral acceleration sensors (PAS) transmit the triggering command to the central ECU via a digital interface. The central ECU triggers the side airbags provided the internal lateral sensor has confirmed a side impact by means of a plausibility check.

Combined ECU for belt tighteners and front/side airbags



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- Voltage transformer and energy accumulator in case the supply of power from the vehicle battery should fail.
- Selective triggering of belt tighteners according to monitored belt-buckle status: firing takes place only when buckle is engaged.
- Setting of up to more than ten triggering thresholds for two-stage belt tighteners and two-stage front airbags depending on the status of belt use and seat occupation.
- Adaptation to the different vehicle features (energy absorption and vibrational behavior of the vehicle structure).
- Diagnosis of internal and external functions and of system components.
- Storage of fault types and durations with crash recorder; readout via the diagnostic or CAN-bus interface.
- Warning-lamp activation.

Gas inflators

The pyrotechnical propellant charges of the gas inflators for generating the airbag inflation gas (mainly nitrogen) and for actuating belt tighteners are activated by an electrically operated firing element.

The gas inflator in question inflates the airbag with nitrogen. The driver's airbag integrated in the steering-wheel hub (volume 35...67 l) or the passenger airbag installed in the glovebox (70...150 l) is inflated approx. 30 ms after firing.

AC firing

In order to prevent inadvertent triggering through contact between the firing element and the system voltage (e.g. faulty insulation in the wiring harness), AC firing involves firing by alternating-current pulses at approx. 80 kHz. A capacitor in the firing-element plug incorporated in the firing circuit isolates the firing element from the DC current. This isolation from the system voltage prevents unintentional triggering, even following an accident where the airbag remains untriggered and the occupants have to be freed from the deformed passenger cell by emergency services, whereby it may be necessary to cut through the (permanent +) ignition cables in the steering-column wiring harness.

Rollover protection systems

Function

In the event of an accident where the vehicle rolls over, open-top vehicles such as convertibles, off-road vehicles etc. lack the protecting and supporting roof structure of closed-top vehicles. Initially, therefore, rollover sensing and protection systems were only installed in convertibles and roadsters without fixed rollover bars.

Today, engineers are developing rollover sensing for use in closed passenger cars where there is the danger of non-belted occupants being thrown through side windows in the event of a crash and crushed by their own vehicle. There is also the danger that body parts of belted occupants, such as arms, head and upper body, can project out of the vehicle and be seriously injured.

To protect against this danger, already existing restraint systems such as belt tighteners and head airbags are triggered in any case, and in convertibles, the extendable rollover bars or the extendable head restraints, are triggered.

Operating concept

The earlier sensing concepts (from mid-1989) started out with an omnidirectional sensing function, i.e. the intention was to detect rollovers in all directions. For this purpose, manufacturers used all-round-sensing acceleration sensors and tilt sensors (AND-wired) with rear-axle switches or level-gauge (spirit-level principle) and gravitation sensors (sensor closes a reed contact with spring support when contact with the ground is lost).

Today's sensing concepts no longer trigger the system at a fixed threshold but rather at one that conforms to a situation and only for the most common rollover situations, i.e. overturning about the longitudinal axis.

The Bosch sensing concept involves a surface-micromechanical yaw sensor (see Fig.) and high-resolution acceleration sensors in the vehicle transversal and vertical directions (y- and z-axes).

The yaw sensor is the primary sensor; the y- and z-acceleration sensors serve primarily to check the plausibility of identi-

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fied rollover situations. These sensors are incorporated in the airbag triggering unit.

Depending on the rollover situation (rollover on an embankment or slope, or due to an obstacle underneath the vehicle), and on the yaw rate and lateral acceleration, the occupant protection devices are adapted to the situation. In other words, using automatic selection and application of the algorithm module appropriate to the type of rollover, triggering takes place after 30...3000 ms.

Passenger-compartment sensing

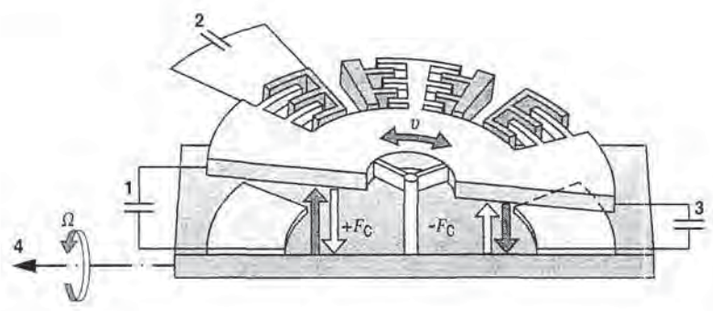
Occupant-classification mats, which measure the pressure profile on the seat, are used to identify whether the seat is being occupied by a person or by an object. In addition, the pressure distribution and the pelvic-bone spacing are used to indicate the occupant's size and thus indirectly the occupant's weight. The mats consist of individually addressable force-sensing points which reduce their resistance according to the FSR principle (Force Sensing Resistor) as the pressure increases.

An absolute-weight measurement is also carried out with two to four piezoresistive sensors or wire strain gauges on the seat frame.

Active infrared or ultrasonic sensor arrays are used for "out-of-position" detection with triangulation.

There is as yet no unified standard for interior-compartment sensing. For this reason, occupant classification mats are sometimes also combined with ultrasonic sensors.

Operating principle of the yaw sensor
1 Detection capacitance, 2 Drive capacitance, 3 Detection capacitance, 4 Sensing axis
F_c Coriolis force,
υ Drive speed,
Ω Yaw rate.



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Outlook

As well as front-airbag switchoff using deactivation switches, in the near future child seats with standardized anchorage, so-called ISOFIX child seats, will come onto the market. Switches integrated in the anchoring locks initiate an automatic passenger-airbag switchoff, which must be indicated by a special lamp.

For further improvement of the triggering function and better advance detection of the type of impact ("precrash" detection), relative speed, distance, and impact angle in a frontal impact, will be detected with microwave radar sensors.

In connection with precrash sensing, reversible seat-belt pretensioners are being developed. They are electromechanically actuated, i.e. they take longer to tighten, and must be triggered earlier, i.e. before initial impact, by precrash sensing alone.

A further improvement in the restraining effect will be provided by airbags integrated in the thorax section of the seat belt ("air belts" or "inflatable tubular torso restraints"); these airbags will reduce the risk of broken ribs in older occupants.

The same path for improving protective functions is being pursued by engineers developing "inflatable headrests" (to prevent whiplash trauma and cervical-vertebra injuries), "inflatable carpets" (prevention of foot and ankle injuries) and two-stage belt tighteners.

In order to reduce wiring-harness thickness and complexity, the firing-circuit network is being developed to include an actuator bus or Firing bus. In this respect, Bosch and Temic are jointly developing "BOTE-ACT", the Bosch-Temic actuator-bus concept. This is a "daisy-chain" principle which e.g. in the event of a short circuit between the two bus lines enables the line section in question to be completely disconnected and all the bus users to function fully thanks to the bus-ring structure. Signals from slow sensors or switches, e.g. the belt-buckle or ISOFIX switches, can also be transmitted with the BOTE-ACT bus.

Efforts are currently being made in the USA to standardize a Firing-bus concept from a range of products from five different manufacturers. Standardization is imperative in order to ensure market penetration and because of the potential usability of standardized firing elements with standardized bus-user electronics. Efforts are underway to integrate the receiver electronics in the firing elements, without increasing diameter and while maintaining a maximum cap extension of 5 mm. This would increase the usability of standard gas inflators.

The most promising and economical development is the use of "semiconductor-based bridgewires" (SCBs), i.e. firing resistors incorporated in silicon which are monolithically integrated with the bus-user electronics on a single chip.

In addition to the Firing bus, a "sensor bus" will be introduced – in the longer term – for networking the signals of fast sensors.

Engineers are also developing CMOS-type stereo video cameras for passenger-compartment sensing. This technology will provide an adequate amount of information for ensuring reliable monitoring of the passenger compartment possibly in conjunction with an additional "auxiliary sensor", the type of which is not yet certain. A video camera also provides for the implementation of numerous comfort functions (this is a question of computer capacity) and thus the required allocation of costs to the various systems.

Because pedestrian protection might possibly be required by law in a few years, work is also being conducted on pedestrian detection and protection devices. Tests are being carried out on a variety of sensing concepts, e.g. foil pressure sensors on the bumpers. Bumper airbags and liftable engine hoods which will cushion the impact of a pedestrian are being considered as protective measures.

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Comfort and convenience systems

Power windows

Power windows have mechanisms that are driven by electric motors. There are two types of system in use. The available installation space assumes a prominent place among the criteria applied in determining which system to install.

Regulator mechanism (now declining in popularity):
An electric-motor driven spur pinion transmits the force to a conventional window regulator.

Cable-operated mechanism:
The electric motor transfers the force through a Bowden cable.

Power-window motors

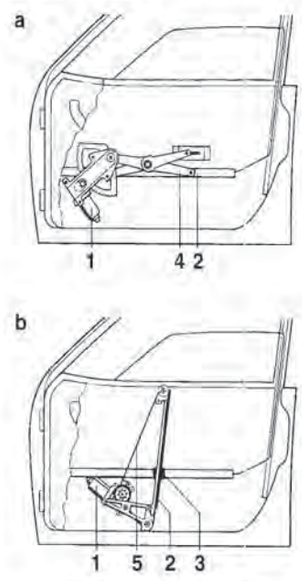
Space limitations inside the door make narrow drive-units imperative (flat motors). The reduction gearing is a self-locking worm-gear design. This prevents the window opening inadvertently of its own accord or being opened by force. Dampers integrated in the gearing mechanism provide for good damping characteristics at the limits of movement.

Power-window control

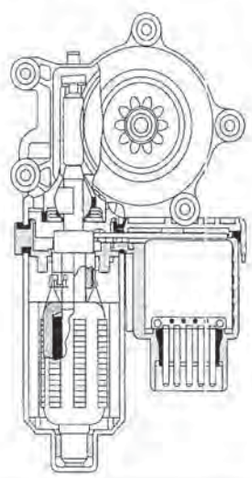
Window operation is controlled manually by means of a rocker switch. For greater convenience, power windows can be linked to a central or decentralized closing system which, when the vehicle is locked, closes the windows automatically or sets them to a predefined partially-open position for ventilation purposes.

During closing, a force-limitation device (anti-squeeze or finger protection) is in operation. The device serves to prevent human appendages from being caught by the closing window. In Germany, paragraph 30 of the StVZO Road Licensing Regulations stipulates that this protection mechanism remain effective while the window moves upward through a 200... 4 mm travel range (as measured from the upper edge of the window opening).

Power-window drives
a) System 1 with regulator,
b) System 2 with flexible Bowden cable.
1 Electric gear motor, 2 Guide rail,
3 Control rod, 4 Lift mechanism,
5 Flexible drive cable.



Power-window motor with integrated electronics (force limiter)

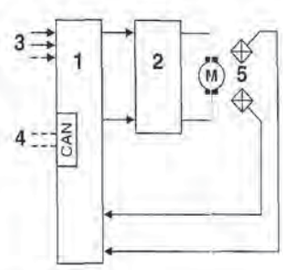


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The window drive units include integral Hall sensors to monitor motor speed during operation. If a reduction in speed is detected, the motor's direction of rotation is immediately reversed. The window closing force must not exceed 100 N at a spring rate of 10 N/mm. The unit automatically overrides the anti-squeeze protection immediately before the window enters the door seal, allowing the motor to run to its end position and permitting complete closure of the window. At the same time, the final window position is signalled back to the control unit.

Electronic control may be concentrated in a central control unit, or the control elements may be dispersed among the individual window motors in order to reduce the complexity of the wiring. Those positioners allow multiple use of the wiring (multiplex systems).

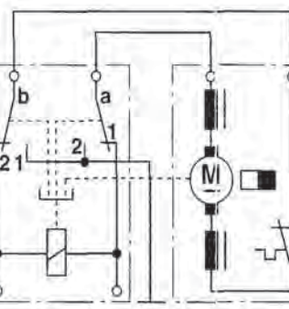
Power-window control unit featuring electronic force limitation
1 Microcomputer, 2 Relay output stage, 3 Position commands, 4 Multiplex bus, 5 Hall sensors.



Power sunroof

Power sunroofs can combine lifting and tilting-sunroof functions. This requires the use of special control systems that may be electronic or electromechanical in design. With electromechanical control (see illustration), mechanical interlocks on the limit switches a) and b) ensure that the roof can be either slid or tilted open from the closed position, depending on the polarity applied at Terminals I and II. Once the sunroof has been tilted or opened, a polarity shift will initiate the corresponding lowering or closing process. An electronic control unit featuring integral force limitation provides benefits if the sunroof is incorporated in a central-locking system. The electronic control system includes a microcomputer responsible for evaluating incoming signals and monitoring sunroof position. The closed and maximum-opening positions for the sliding action of the sunroof are monitored with the aid of microswitches or Hall-effect sensors. Supplementary functions such as
– preset position control,
– closing via rain sensor,
– motor-speed control, and
– electronic motor protection
can all be included in the package at relatively modest expense.

Sunroof drive unit with electromechanical control of opening, closing and tilt functions



Drive for the roof is provided by Bowden cables or other torsion- and pressure-resistant control cables. The drive motor is usually installed in the roof or at the rear of the vehicle (e.g., in the trunk). Permanently energized worm-drive motors with outputs of approx. 30 W are used to drive the sunroof mechanism. They are protected against thermal overload by a thermostatic switch (declining in popularity) or (most frequently) by a software thermoswitch.

Provision must also be made for ensuring that the roof can be closed using the on-board tools (e.g. wheel-nut wrench or similar) in the event of a failure in the electrical system.

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Seat and steering-column adjustment

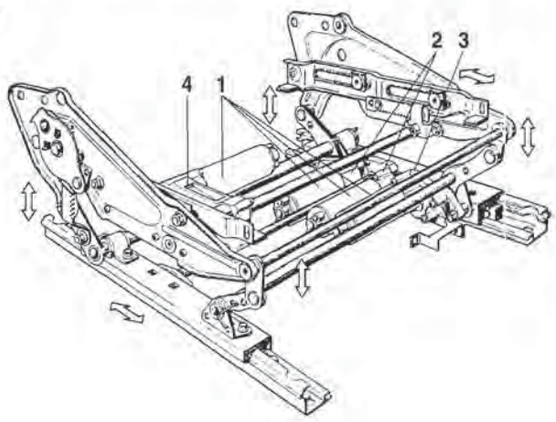
Electric power seats are still essentially restricted to passenger cars of the medium and higher price classes. The primary function of these systems is to enhance comfort, but multiple adjustment options combine with space restrictions and possibly difficult control access to limit the practicality of manually-operated mechanisms. Up to seven motors control the following functions:
- seat-cushion height, front/rear,
- longitudinal travel,
- seat-cushion tilt angle,
- backrest tilt adjustment,
- lumbar support, height/curvature,
- shoulder-support tilt angle,
- head-restraint height adjustment.

A common seat-adjustment system includes four motors driving two gearsets. One of the gearsets governs height adjustment, while the other is designed for combined longitudinal and height adjustment. The unit for adjusting the seat height is not included on simpler seat designs. Yet another concept features three identical gear motors with four height and two linear-adjustment gearsets. The gear motors drive the gearsets via flexible shafts. This is a highly universal system and can be installed on any seat design. Modern seats (especially for sporting vehicles) do not merely affix the lap belt to the seat frame, they also attach the shoulder strap – together with its height adjuster, inertia reel and tensioner mechanism – to the backrest. This type of seat design ensures optimal belt positioning for a wide range of different-sized passengers at all available seat positions, and thus makes an important contribution to occupant safety. The seat frame must be reinforced for this type of design, while both the gearset components and their connections to the frame must be strengthened.

The optional programmable electric seat adjuster ("memory seat") can recall several previously selected seat positions. Potentiometers or Hall-effect sensors are employed to signal the current seat position. On two-door vehicles a feature designed to slide the front seat to its extreme forward position can be incorporated to ease ingress for rear passengers.

Electrically-adjustable steering columns are also seeing increased use as yet another means of enhancing driver comfort. The adjustment mechanism, consisting of a single electric motor and self-arresting gearset for each adjustment plane, forms an integral part of the steering column.

Electromechanical seat-adjustment system *(source: Brose Vehicle Components)*
1 Electric motors,
2 Longitudinal/height-adjustment gearset,
3 Height-adjustment gearset,
4 Gearset for seat-cushion tilt control.



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The gearset for telescopic adjustment must be capable of absorbing any and all impact forces (crash forces) which might be applied to the steering column. The adjustment can be triggered in either of two ways, using the manual position switch or with the programmable seat adjustment. Also available is a provision for tilting the column upward to facilitate driver entry and egress.

Central locking system

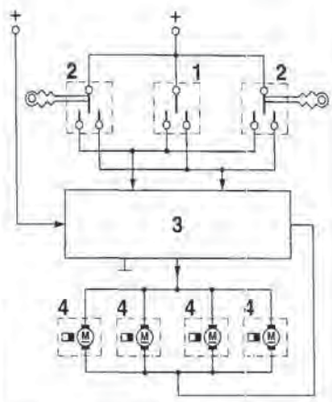
Either pneumatic or electric actuators can be used to power central locking systems for vehicle doors, luggage compartments and fuel-filler flaps.

In pneumatic systems, an electric motor drives the reversible dual-pressure pump which provides the required system pressure (positive or vacuum). The system can be switched on and off by a central position switch inside the vehicle and by the ignition switch. As an optional feature, the system can be operated from a number of points (driver door, front-seat passenger door, and trunk lid).

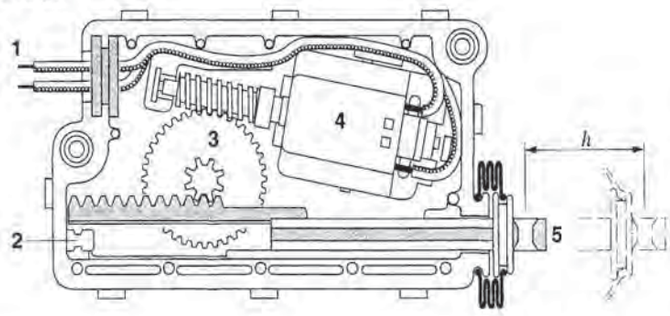
More widespread than the pneumatic systems are those which depend on electric motors for central locking. Although various technologies are used, according to function range and lock type, the basic principle remains constant: a small electric motor featuring a reduction-gear drive unit powers the actuating lever responsible for opening and closing the lock. Provision must be made to ensure that the door can always be unlocked with the key and the interior handle in the event of a power failure. Central locking systems incorporating special theft-deterrence features must be designed to preclude deactivation of the security system using any means other than the vehicle key.

Ultrasonic or infrared remote control provide increased convenience. Such systems permit remote operation of the central locking system when the driver is still some distance away from the vehicle.

Central locking with electric motor
1 Central switch, 2 Contacts in door-lock mechanisms, 3 Control unit, 4 Servomotors.



Central locking servo unit
1 Wiring connection, 2 Flexible end-position coupling, 3 Gear unit, 4 Electric motor, 5 Actuating lever, h Travel range.



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Automotive hydraulics

Quantities and units[1])

Symbols		Units
A	Area of flow cross section	cm^2
A_D	Cross-sect. area of restriction	cm^2
A_K	Piston area	cm^2
A_R	Cross-sectional area of line	cm^2
b	Gap width	mm
d	Line diameter	mm, cm
E_{Fl}	Modulus of elasticity, fluid	$N \cdot mm^{-2}$
E_{Oil}	Bulk modulus, oil	$N \cdot mm^{-2}$
e	Eccentricity offset	mm
F	Force	N
g	Gravitational acceleration	$m \cdot s^{-2}$
H	Cylinder stroke length	cm, mm
h	Gap height	µm, mm
l	Gap length, line length	mm, m
l_b	Output length	cm
M_1	Input torque	N · m
M_2	Output torque	N · m
M_{th}	Theoretical input or output torque	N · m
M_{verl}	Torque loss	N · m
n	Rotational speed	min^{-1}
P_{an}	Input power	kW
P_{ab}	Output power	kW
p_z	Cylinder pressure	bar
Δp	Pressure difference	bar
Q	Delivery rate of hydraulic pump, or intake rate of hydraulic motor or hydraulic cylinder	$l \cdot min^{-1}$
Q_1	Delivery rate	$l \cdot min^{-1}$
Q_2	Intake rate	$l \cdot min^{-1}$
Q_L	Leakage rate	$l \cdot min^{-1}$
Q_{th}	Theoretical delivery or intake rate	$l \cdot min^{-1}$
Re	Reynolds number	—
r	Line radius	mm
t	Stroke time of hydraulic cyl.	s
U	Perimeter of flow cross section	cm
V_{Fl}	Fluid volume	cm^3
V_H	Displacement of hydraulic cyl.	cm^3
V_o	Output volume	cm^3
V_{th}	Theoretical delivery displacement /intake displacement	cm^3
v	Line flow rate	$m \cdot s^{-1}$
v_1	Stroke rate	$m \cdot s^{-1}$
α_D	Flow coefficient of restrictors, orifices etc.	—
η	Dynamic viscosity	$Ns \cdot m^{-2}$
η_{hm}	Hydromechanical efficiency	—
η_{vol}	Volumetric efficiency	
λ	Coefficient of flow resistance	—
ν	Kinematic viscosity	$m^2 \cdot s^{-1}$
ϱ	Density	$kg \cdot dm^{-3}$
ω	Angular velocity	s^{-1}

Calculation of coefficient of flow resistance λ

$\lambda = \frac{64}{Re}$ for laminar flow and isothermal change of state

$\lambda = \frac{75}{Re}$ for laminar flow and adiabatic change of state

$\lambda = \frac{0.316}{Re^{0.25}}$ for turbulent flow up to $Re = 80{,}000$ and smooth lines

$Re = v \cdot D_{H/V}$ where $D_H = A/U$

Terms and formulas
(quantity equations)

Hydraulic pump

Delivery rate

$$Q_1 = V_{th} \cdot n \cdot \eta_{vol}$$

Output power

$$P_{ab} = Q_1 \cdot \Delta p$$

Input torque

$$M_1 = \frac{V_{th} \cdot \Delta p}{2\pi} \cdot \frac{1}{\eta_{hm}}$$

Input power

$$P_{an} = M_1 \cdot \omega$$

Volumetric efficiency

$$\eta_{vol} = \frac{Q_1}{Q_{th}} = \frac{Q_{th} - Q_L}{Q_{th}} = 1 - \frac{Q_L}{Q_{th}}$$

Hydromechanical efficiency

$$\eta_{hm} = \frac{Mth}{M_1} = \frac{M_{th}}{M_{th} + M_{verl}}$$

Hydraulic motor

Intake rate

$$Q_2 = V_{th} \cdot n/\eta_{vol}$$

Output power

$$P_{ab} = M_2 \cdot \omega$$

Output torque

$$M_2 = \frac{V_{th} \cdot \Delta p}{2\pi} \cdot \eta_{hm}$$

Volumetric efficiency

$$\eta_{vol} = \frac{Q_{th}}{Q_2} = \frac{Q_{th}}{Q_{th} + Q_L}$$

Hydromechanical efficiency

$$\eta_{hm} = \frac{M_2}{Mth} = \frac{M_{th} - M_{verl}}{Mth}$$

Hydraulic cylinder

Cylinder pressure $p_z = F/(A_K \cdot \eta_{hm})$

Swept volume $V_H = A_K \cdot H$

Stroke time $t = V_H/Q_1$

Stroke rate $v_1 = Q_1/A_K$

[1]) These are commonly used units, not SI-units.

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Flow rates in lines and gaps

Required line cross section

$$A_R = Q_1/\upsilon$$

Pressure loss in straight lines

$$\Delta p = \lambda \cdot \frac{l}{d} \cdot \frac{\varrho}{2} \cdot \upsilon^2$$

Flow through a pipe (according to Hagen-Poiseuille)

$$Q = \frac{\pi \cdot r^4}{8 \cdot \eta \cdot l} \cdot \Delta p$$

Flow (laminar) through a smooth gap

$$Q = \frac{b \cdot h^3}{12 \cdot \eta \cdot l} \cdot \Delta p$$

Flow (laminar) through an eccentric sealing gap

$$Q = \frac{d \cdot \pi \cdot \Delta r^3}{12 \cdot \eta \cdot l} \cdot \left[1 + 1.5 \cdot \left(\frac{e}{\Delta r}\right)^2\right] \cdot \Delta p$$

($2 \cdot \Delta r$ = clearance between piston and bore)

Flow through restrictors and orifices

$$Q = \alpha_D \cdot A_D \cdot \sqrt{2 \Delta p/\varrho}$$

(α_D at control slide valves: 0.6 to 0.8)

Compressibility of a hydraulic fluid

$$\Delta V_{Fl} = A_K \cdot \Delta l = V_o \cdot \Delta p/E_{Fl}$$

where the initial volume is

$$V_o = A_K \cdot l_o$$

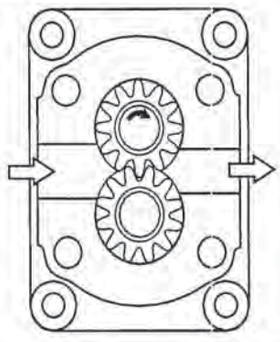
and the bulk modulus for oil is

$$E_{Oil} \approx 1.6 \cdot 10^9 \text{ N} \cdot \text{m}^{-2}$$

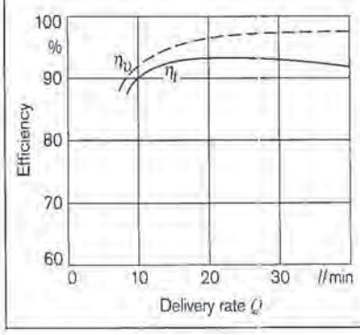
Gear pumps

Gear pumps are designed either with one external-toothed and one internal-toothed gear or with two external-toothed gears. External-toothed pumps are cheaper to manufacture and are therefore in much more common use. The displacement per revolution is constant, and is determined by the gear diameter, the center-to-center distance and the width of the teeth. The rotating (meshed) gears transport the hydraulic fluid in the spaces between the teeth from the low-pressure to the high-pressure side, and the teeth immersed in the fluid force it into the delivery line. There is almost no clearance between the housing and the tips of the teeth; thus the pump exhibits good radial sealing properties. The pump chamber is axially sealed by plates or bushings which are hydraulically pressed against the gears. These plates or bushings also act as bearings for the gears. This design achieves the high efficiency characteristic of high-pressure pumps. Drive speeds of up to 4000 min^{-1}, maximum pressures of up to approx. 300 bar and high power densities (6 kW/kg) make gear pumps particularly well suited for use in automotive hydraulic systems. The span of delivery rates required, which ranges from 0.5...300 l/min, is covered by a range of 4 to 5 pump sizes.

Gear pumps and motors *(Direction of fluid flow and gear rotation shown for pump operation)*



High-pressure gear pump
Volumetric efficiency η_v and overall efficiency η_t according to delivery rate at Δp = 210 bar.



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Gear motors

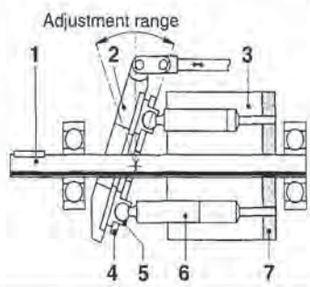
Like gear pumps, the simplest gear motors are designed to operate in only one direction of rotation. In gear motors, oil flows from the high-pressure to the low-pressure side, and the gears rotate in a direction opposite to the direction in which they would rotate if the device were a pump. Motors suitable for use as vehicle drives, i.e. motors which can be operated in both directions of rotation and which can be loaded in the reverse direction, have also been derived from gear pumps through appropriate design of the axial pressure field and the leakage fluid passages. The advantages of external toothed pumps such as high power density, small installation-space requirements and low manufacturing costs also apply to gear motors. They are therefore chosen for road vehicles, construction equipment and agricultural machinery for driving cooling and cleaning fans, screw conveyors, sweepers, spreading plates, vibrators, etc. The excellent starting behavior of gear motors is utilized in driving pumps and compressors, as well as in motive drives.

Piston pumps and motors

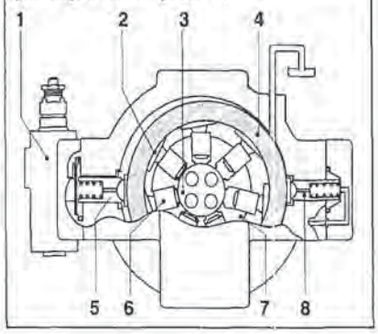
The drive mechanism of hydraulic piston pumps and motors differs significantly from the classical piston-machine design. The high pressure level (standard pressure for hydrostatic drives 350...400 bar) results in high piston forces which necessitate sturdy and rigid mechanical drive systems. The drive mechanisms of modern machines are nevertheless highly compact, particularly due to the hydrostatic transmission of force between the two primary reciprocating components. The good lubricating and cooling qualities of the hydraulic fluid promote such space-saving designs. Hydraulic piston machines are thus able to achieve a maximum power density of more than 5 kW/kg.

In order to achieve a uniform volumetric flow rate, hydraulic piston machines are designed with an odd number of piston elements. A differentiation is made between radial- and axial-piston machines, depending upon the configuration of the drive mechanism. Both types are available as either pumps or motors with constant or variable displacement, suitable for use in open and closed circuits. Since the phase control of hydraulic machines did not prove itself in practice, continuous control is only possible by varying the piston stroke. Rotating, stroke-generating mechanisms, such as eccentric shaft or crankshaft (radial), and swashplate (axial), are unsuitable for stroke adjustment and are only used in some fixed-capacity machines. All variable-displacement units and many constant-displacement units use other drive mechanisms specific to the hydraulic system. These types incor-

Axial-piston machine
(swash-plate unit)
1 Drive shaft, 2 Swash plate, 3 Cylinder barrel (rotates), 4 Retaining plate, 5 Slippers, 6 Piston, 7 Control plate



Radial-piston machine
1 Open- or closed-loop control device, 2 Slippers, 3 Control journal, 4 Stroke ring, 5 Control piston 1, 6 Piston, 7 Cylinder block (rotates), 8 Control piston 2.



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porate a rotating cylinder assembly. Together with a stationary control plate or control journal, the cylinder assembly forms a rotary slide valve which alternately exhausts and fills the cylinders. Axial-piston machines are designed as swashplate units.

The centrally-ported machine has been adopted as the standard radial design. The cylinder block rotates on the control journal. The bearing forces are supported by hydrostatic pressure fields on the journal. Similar pressure fields transmit the forces between the rotating piston slippers and the stationary cam ring. The ring's variable eccentric position relative to the control journal generates the piston stroke. In the case of pumps in which the delivery direction can be reversed, the stroke ring can be moved in both directions via its adjustment piston. All control elements are located in the stator, allowing rapid and precise adjustments in flow volume using hydraulic or electronic servo elements and controllers. The hydraulic adjustable piston unit can be integrated within the electronic control circuit using either proportional valves in the control circuit or an electrohydraulic adjustment system.

Electrohydraulic pumps and small units

An electrohydraulic pump is a combination of a hydraulic pump and an electric motor. A DC motor is used in most mobile hydraulics applications, however AC and three-phase motors are also used in stationary operation.

Gear pumps are characterized by a low degree of pulsation and quiet operation. As such, these hydraulic components have proven their worth for generating pressure in electrohydraulic pumps. Sizes B and F with capacities of 1...22.5 cm^3 per revolution are used, and generate operating pressures of up to approx. 280 bar. Together with size I to T electric motors which generate a maximum nominal output of 8 kW, they are used in a number of different mobile-hydraulics applications.

Electrohydraulic pumps are used to supply the hydraulic energy for the "raising/lifting" and "steering" functions in all kinds of vehicles, in particular in industrial trucks (fork-lift trucks and pallet trucks); mobile lifting platforms; trucks; special construction, transport and rescue vehicles; and passenger cars. In passenger cars, electrohydraulic pumps, including increasing numbers of miniaturized versions, are being applied in functions such as ride-level control, and power-assists for steering and parking. Safety-related braking and acceleration control represent a special area of application. A hydraulic unit is at the heart of the ABS and ASR systems, and generates the oil pressure needed when these come into operation.

In addition to electrohydraulic pumps, various valves or valve assemblies must be installed in motor vehicles in order to implement a wide variety of control functions. This has led to the development of small, compact units with outputs of up to 4 kW. In these units, the electric motor and hydraulic pump are supplemented by a valve block, a hydraulic-fluid tank, and fluid and air filters. The design concept accommodates individual modifications for specific control functions. Sleeve and seat valves can be combined in a compact control block or at unions in the system. Small and miniaturized units are used in those applications where high power is required despite limited space being available. Examples include municipal vehicles (street sweepers, rotary snow plows, utility tractors, industrial trucks, special-purpose vehicles, and passenger-transport vehicles with lifts and pivoting equipment for carrying handicapped persons).

Novel applications are found in passenger-vehicle functions. The most advanced convertibles utilize hydraulic mechanisms to operate the folding top; these devices fold the top, stow it in a specified manner, and lock it into place.

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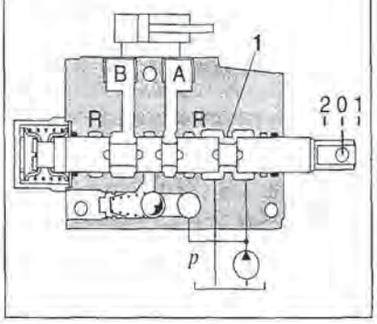
Valves

Directional-control valves

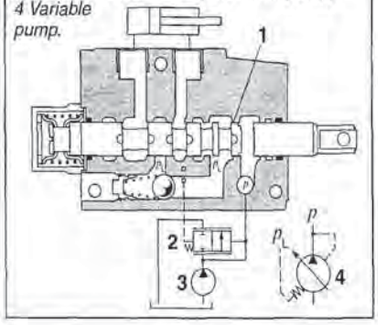
OC valves (open-center)
When the valve unit is in the neutral position, the flow from the pump is directed through up to 10 valves (neutral circulation pattern). The fluid flow is restricted when a valve is actuated prior to the opening of the line to the servo unit.
Disadvantages:
- High pressure loss = Energy loss in neutral position,
- Control precision affected by load pressure.

LS Valves (load-sensing)
Used in systems with variable-capacity pumps or constant-capacity pumps and auxiliary pressure compensator. In the neutral position, the LS control line removes the pressure from the pump and pressure compensator. When the valve unit is actuated, the pump controller/pressure regulator maintains the pressure differential at the valve spool at a constant level. The result: The flow of hydraulic fluid to the servo unit is not affected by the load pressure.
Advantages:
- Minimal neutral-position loss,
- Improved precision control independent of load pressure.

OC directional-control valve
1 Neutral flow.



LS directional-control valve
1 Measuring diaphragm, 2 Pressure compensator, 3 Constant-displacement pump, 4 Variable pump.

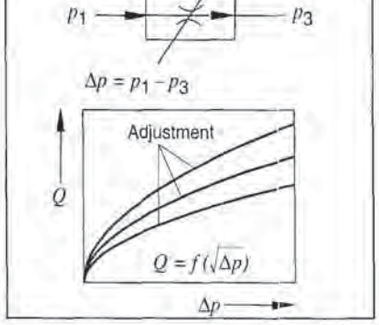


Flow-control valves

Throttle valves
Throttle valves are used to adjust the fluid flow rate by altering the cross-sectional area of the flow. According to the laws of rheological science, this flow-limiting function is pressure-dependent; it is therefore used only for simple flow adjustment. Pressure-independent adjustment requires the use of control valves.

Flow-pressure-compensated control valves
In order to be able to set the fluid flow Q independently of the load pressure p_3 at 2-way flow-control valves, the pressure difference at the metering orifice ($p_1 - p_2$) is held constant by a variable restrictor (pressure compensator). The pressure difference $p_1 - p_2$ corresponds to the spring force acting on the pressure compensator. In this type of control, the sur-

Throttle valve



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plus fluid flows via the pressure-relief valve in the system.

Losses can be reduced by using 3-way flow-control valves which have an additional drain through which the surplus fluid flows back to the tank or to other hydraulic devices.

Pressure-control valves

Pressure-relief valve

Hydraulic circuits incorporate a pressure-relief valve in order to protect the components as well as to ensure the operational safety of the system. If the pressure acting on the seat diameter exerts a force equal to the force exerted by the precompressed spring, the valve cone lifts off its seat and the fluid flows to the tank. Pilot-operated pressure-relief valves are used for greater flow volumes and to achieve valve characteristics which are independent of the flow rate. The pilot valve relieves the spring chamber of the main valve which controls return flow to the tank.

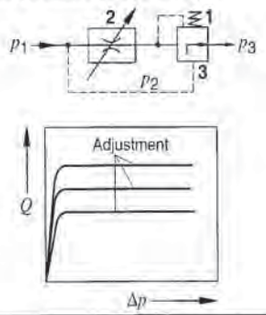
Electric proportional directional-control valves

Nowadays, agricultural and industrial vehicles such as tractors and fork-lift trucks increasingly use electrically operated directional-control valves. The reasons for this are higher productivity, lower demands on the driver and more economical installation costs as a result of the greater scope in the choice of valve location on the vehicle. Electronic circuitry provides for easy-to-use, safe and reliable control systems which in some cases extend up to automatic movement sequences.

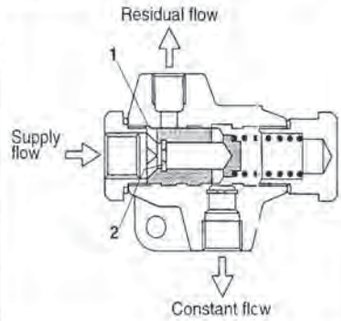
Direct solenoid actuation

In applications involving small to medium

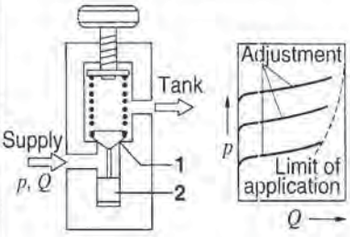
2-way flow-control valve
1 Spring, 2 Measuring orifice, 3 Regulating throttle (pressure compensator).



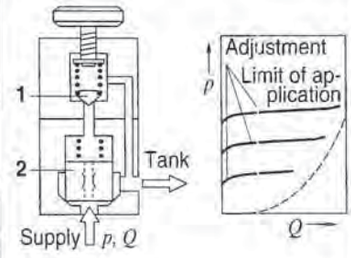
3-way flow-control valve
1 Pressure compensator, 2 Measuring diaphragm.



Directly controlled pressure-relief valve
1 Valve seat, 2 Cushioning pistor..



Pilot-operated pressure-relief valve
1 Pilot valve, 2 Main valve.



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hydraulic powers (e.g. positioning movements on combine harvesters, tilting function on fork-lift trucks), the directional-control-valve spool is displaced by a solenoid against the action of a spring by an amount proportional to the excitation current. The application limit is determined by the available solenoid force, e.g., 30 l/min, 200 bar. The short stroke of the solenoid limits application of this principle to 3-position valves.

Electrohydraulic actuation

The new "electrohydraulic positioner unit" (EHSe) generates a high actuation force for high-performance control such as is required for the front lifter on agricultural tractors or the jib on a truck loading lift. A positioning piston which moves the valve armature to the position specified by the setpoint generator is used to amplify the force. A position control loop compensates for interference from flow forces thereby providing low hysteresis. The 4/3 pilot-operated directional-control valve with double solenoid is supplied with a control pressure of approx. 20 bar. An inductive travel sensor detects the spool position for the purposes of position control. It also performs the same function for diagnosis and safety purposes.

Electronic control circuitry

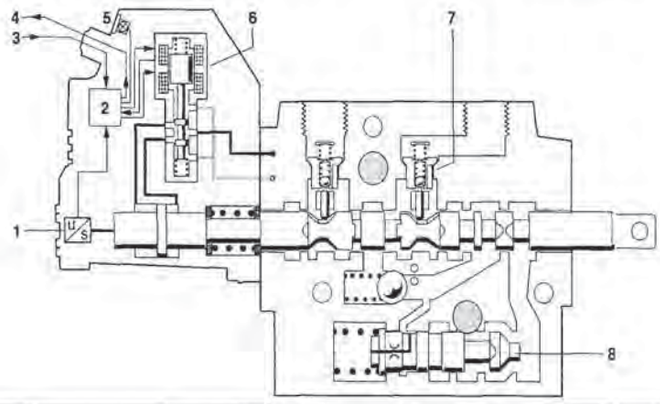
The electronic analyzer circuitry of the travel sensor (ASIC) and the digital control electronics with serial CAN bus interface are housed inside the positioner unit casing. The use of a microcontroller with an EEPROM means that the valve characteristics are programmable. The valve and timing characteristics can be chosen and modified according to application. Static characteristics which define the relationship between input signal and flow volume (precision control), and time gradients for defined acceleration and deceleration of hydraulic drive units are stored.

In addition to the primary control option using the CAN bus, control by means of analog or pulse-width-modulated (PWM) voltage signal is also possible. In the event of malfunctions on the part of the directional-control valve, e.g. sticking valve armature, loss of control pressure, etc., depending on the seriousness of the fault it is indicated (coded flashing signal, CAN message) and/or the actuator unit is switched off.

Cylinders

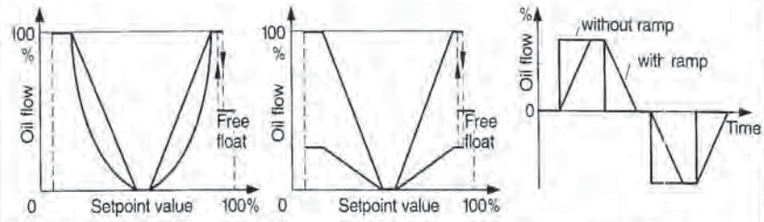
Cylinders convert hydraulic power (pressure, oil flow) into rectilinear motion (force, velocity). They are characterized

SB23LS directional-control valve with electrohydraulic positioning unit EHSe
1 Inductive travel sensor, 2 Digital electronics module, 3 Control input: CAN, PWM or potentiometer signal, 4 Diagnostics output: fault signal, 5 Diagnostics output: visual indicator, 6 Pilot valve, 7 Check valve, 8 Pressure compensator.



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ESHe electrohydraulic positioner unit for mobile automotive control valves
Programmable characteristics

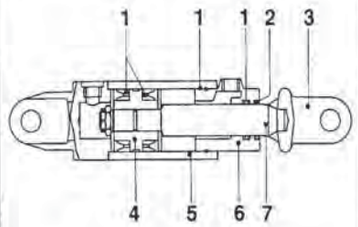


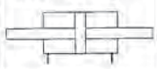
Programmable characteristic-curve shape
- *Linear (x) to progressive (x^3)*
- *Individual for each direction*
- *For adjustment of precision control characteristics*

Programmable characteristic-curve gradient
- *0...100 %*
- *Individual for each direction*
- *For adjusting to specific hydraulic motor/cylinder*

Programmable timing characteristics
- *Timing gradients from 0.07...4 s*
- *Individual for each direction*
- *For adjustment of acceleration and deceleration to suit different load parameters*

Cylinder
1 Sealing element, 2 Wiper, 3 Pivoting head, 4 Piston, 5 Cylinder barrel, 6 Guide sleeve, 7 Piston rod.



Design	Remarks
Single-acting 	Drive possible in only one direction.
Double-acting 	Acts in two directions, pressure acts on different surface area on piston and rod sides.
Double-acting 	Acts in two directions, both surface areas equal, continuous piston rod.

by high power density and relatively simple design. Cylinder efficiency is determined by the seals, the operating pressure and the piston surface quality. In addition to force and speed, resistance to buckling is also an important design criterion. It determines the dimensions and the extension of the cylinder piston. Types of cylinder mounting at the head and bottom include holes, clevises, pivoting bearings and threads.

Tractor hydraulics

A hydraulic system transforms the tractor into a universal mobile device for agricultural and forestry applications. With the aid of hydraulics, the wide variety of implements used can be quickly mounted to the tractor at the front, rear and between the axles, and moved into their appropriate operating positions by means of open- or closed-loop controls. Quick-disconnect hydraulic couplings are used to control additional linear and rotary motors on the implement itself. Operation of the tractor is facilitated by means of hydraulic power-assisted steering (see also the section on steering systems), brake systems, clutches and gear-shifting systems. Pressure relief valves prevent the tractor from being overloaded. Trailers pulled by the tractor and matched to its brake system

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can be hydraulically braked.

High power density and flexibility account for the wide use of hydraulics in tractors. The wide variety of applications ranging from small vineyard tractors rated at approximately 20 kW to large center-pivot-steered tractors rated at roughly 300 kW, and including tool carriers, standard tractors, forestry tractors and construction-work tractors, represent a number of different requirements in terms of hydraulic power, hydraulic systems and system operation.

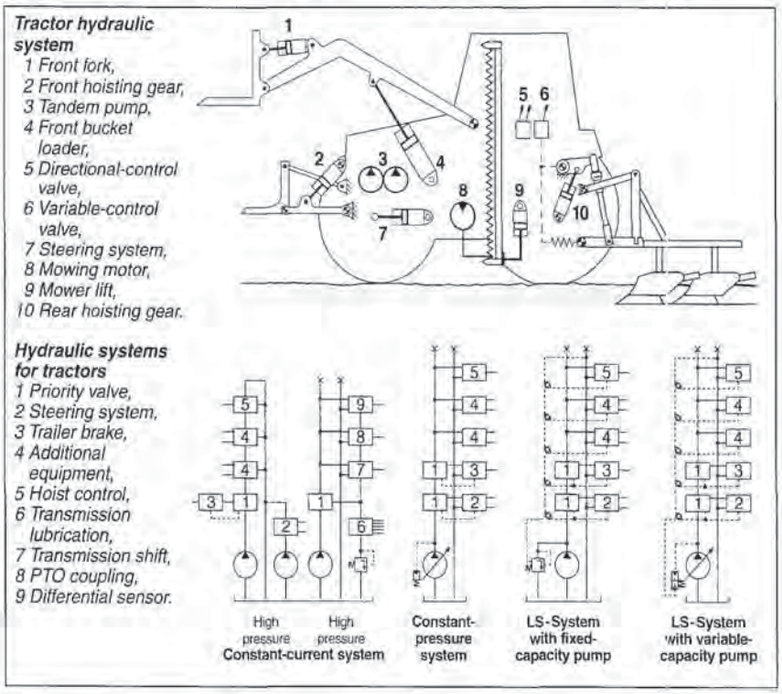
Hydraulic systems for tractors

Tractor hydraulic systems generally have three pressure circuits:

- the high-pressure or work circuit with pressures of up to 250 bar and flow rates of up to 120 *l*/min for steering, trailer brake, control of lifting equipment and other loads,
- the control circuit which has a pressure of approx. 20 bar and a flow rate of around 30 *l*/min for power-shift transmission, PTO, locking differential, etc.
- the lubrication circuit with a pressure of approx. 3...5 bar.

There are considerable differences in the high-pressure circuit as described below. The essential criteria for choice of design are function, cost, energy losses, complexity and ease of operation.

Constant flow system/Open center (OC) system (*Q* constant, *p* variable)
This system is the most common owing to its favorable price-performance ratio. The pressure is most frequently provided by gear pumps. With all valves in their neutral positions, the hydraulic fluid circulates around the neutral circuit to the tank, and its volumetric flow rate is variable only as a function of pump speed. In order to operate one or more loads, the neutral-flow channel in the valve is constricted, and hydraulic fluid is supplied to the working ports according to the valve-spool displacement. Priority valves are used to ensure that precedence is given to the pressure supply for safety-related functions



Tractor hydraulic system
1 Front fork,
2 Front hoisting gear,
3 Tandem pump,
4 Front bucket loader,
5 Directional-control valve,
6 Variable-control valve,
7 Steering system,
8 Mowing motor,
9 Mower lift,
10 Rear hoisting gear.

Hydraulic systems for tractors
1 Priority valve,
2 Steering system,
3 Trailer brake,
4 Additional equipment,
5 Hoist control,
6 Transmission lubrication,
7 Transmission shift,
8 PTO coupling,
9 Differential sensor.

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such as steering and trailer-brake operation. The inherent high levels of power loss in the neutral circulation system can be counteracted by appropriate splitting of the work circuit and the use of two or more pumps to provide the pressure.

Constant pressure system/Closed center (CC) system (Q variable, p constant)
A pump with variable volumetric flow rate operates at a regulated constant pressure. When the valves are in their neutral positions, the pump reduces its delivery volume to the rate of leakage loss. When a valve is operated, the pump automatically adjusts its flow volume to that required by the load. Priority valves are used to give precedence to safety-related functions. This type of system is insignificant in terms of automotive hydraulics.

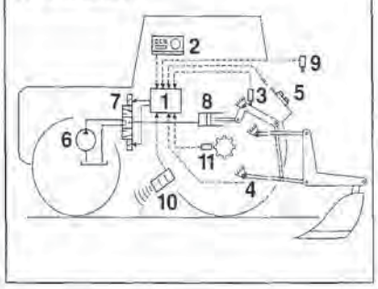
Load-compensating system with constant-displacement pump/Open center load sensing (OCLS) system (Q constant, p variable)
In this type of system, the differential pressure is kept constant with the aid of a control element (pressure compensator) using the variable valve cross-section (measuring diaphragm). Thus, the flow of hydraulic fluid directed to the consumer unit is proportional to the valve opening and unaffected by the load pressure. The highest load pressure within the control system is selected and directed to the pressure compensator by shuttle valves and control lines. The excess pump flow is returned to the tank via the pressure compensator. If load compensation is also to be maintained for all consumer units in parallel operation, a second 2-way flow regulator (individual pressure compensator) is used for each consumer unit.

This system is technically very involved but is becoming increasingly popular in mobile hydraulics (materials flow/agricultural applications) due to its ease of operation. The energy losses are only negligibly lower than with the constant-flow system.

Load-compensating system with variable-displacement pump/Closed center load sensing (CCLS) system (Q variable, p variable)
This system is similar to the OCLS system. The difference, however, is that the pressure is supplied by a variable-displacement pump with regulated pressure and flow, instead of by a constant-displacement pump with pressure compensator. In addition to the gains in convenience, this system also offers substantially reduced energy losses at partial load. This type of system is being increasingly used in applications involving high-power hydraulics (construction machinery, large tractors).

Rear hoisting-gear control
The rear hoisting gear with its standardized 3-point coupling is the most frequently used type of mount for implements. The attached implements can be raised, lowered and held in position. In addition, the tractive force in the hitch linkage can be held constant, or the position of the implement with respect to the tractor can be held constant. Tractive-force regulation is primarily used in working the soil, e.g., plowing (a constant tractive force produces a constant working depth in homogeneous soil). High control quality, i.e. small fluctuations in tractive force, is required for full utilization of engine characteristics and small fluctuations in depth. Because the implements are

Electronic hoist-control mechanism (EHR–D)
1 Electronics, 2 Control panel, 3 Position sensor, 4 Force-sensor pins, 5 Rear actuation, 6 Pump, 7 Control valve, 8 Cylinder, 9 External sensor, 10 Radar sensor, 11 RPM sensor.



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guided and thus largely supported by the hoisting-gear control, the resistance associated with the implement's movement through the soil generates additional downward force at the drive wheels. This reduces wheel slip, and thus energy losses. Position control is used primarily for implements which do not penetrate the ground. In addition, in the case of highly variable soil resistance, a certain percentage of the position control can be mixed with the tractive-force control (mixed control) in order to limit depth fluctuations.

Mechanical hoisting-gear control (MHR)
The sensor signals are monitored and processed as mechanical travel. The tractive force is measured in the form of the spring travel on the upper or lower control arms. The actual position can be read from a cam disc on the lifting shaft. A control rod relays the signals to the control valve according to the selected ratio. At the control valve, the actual values are compared with those selected by the operator. The hoisting mechanism is then raised or lowered to compensate for any control deviations.

Electronic hoisting-gear control (EHR)
The salient feature of EHR lies in the fact that the monitored and control signals are derived, transmitted and processed electronically. The tractive force is measured directly by force-sensor pins at the hitch coupling point. It is possible to supplement position and force control with other functions by expanding the electronic controller and the number of sensors. Rear activation eases attachment of implements. An external sensor can be installed to monitor the travel height of an attached implement (e.g., beet lifter). A speed sensor (radar) and a wheel-speed sensor make it possible to determine slip, providing the basis for a slip-control feature. Active suspension control is useful for enhancing safety and comfort when heavy attachments are being towed.

Hoisting-gear control with hydraulic signal transmission
The basis of this hydraulic signal-transmission system is a hydraulic bridge. The setpoint and actual values of the controlled variable are applied via throttle elements to the bridge's arms and, if the setpoint and actual values do not coincide, this causes the control valve to be shifted against the force of a spring by the diagonally tapped-off differential pressure.

Directional-control valves for tractors
Depending upon the type of hydraulic system, directional-control valves for the high-pressure circuit have either open or closed neutral positions with load sensing. In order to hold heavy loads over a long period of time, as well as for reasons of safety, poppet valves or slide valves (see the section on directional-control valves) are used with downstream mechanically or hydraulically pilot-operated poppet valves in the cylinder port. In addition to the three positions for load extension, retraction and holding, the valves often have a fourth position (free-floating position) in order to permit the implement to be guided on the soil, for example by means of supporting wheels. Detent mechanisms with hydraulic maximum-pressure release automatically reset a valve which has moved past its neutral position in response to overload, or if a cylinder reaches its limit position (operating convenience). Integrated flow regulators allow pressure-independent parallel connections and constant speed of linear and rotary motors. Built-in shock absorbers protect the tractor from overloads if the cylinder port is closed.

Solenoid valves are used to actuate a number of hydraulic functions on the implement, and are controlled from the tractor by means of cables (e.g., beet lifters).

Hydraulic fluid is supplied via hydraulic quick-disconnect couplings or by a separate pump driven by the tractor's power take-off shaft. Electromagnetic switching valves for gear-shifting purposes, and for actuating the various clutches in the tractor, are increasingly being incorporated in the low-pressure circuit.

For trailer braking, a braking valve is included in the high-pressure circuit. It is actuated by the tractor brake, and supplies a correspondingly controlled brake pressure to the trailer.

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