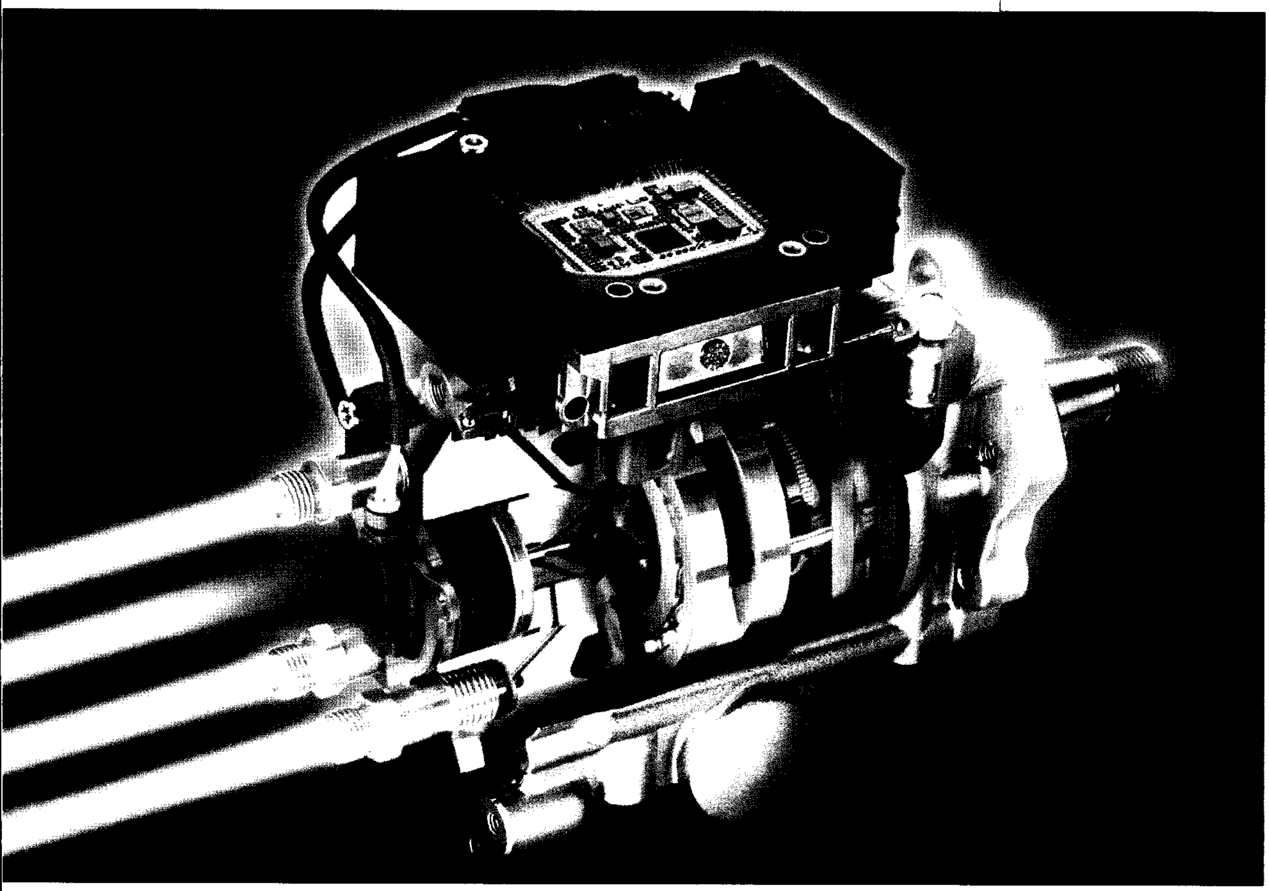
The Bosch Yellow Jackets **Edition 2003** | **Expert Know-How on Automotive Technology** | Diesel-Engine Management

Distributor-Type Diesel Fuel-Injection Pumps

BOSCH

Automotive Technology

- System Overview
- Helix-and-port-controlled distributor injection pumps
- Axial-Piston Pump (VP29, VP30)
- Radial-Piston Pumps (VP44)



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Distributor-Type Diesel Fuel-Injection Pumps

Robert Bosch GmbH

Contents

Properties and characteristic data of the most important fuel-injection systems for diesel engines

Fuel-injection system	Type of use	Injection parameters			Control method	Engine-related data			
Type	P Cars and light commercials N Trucks and buses O Off-road vehicles [1] S Ships/trains	Injected volume per stroke/ injection cycle mm³	Max. permissible pressure at jet bar (0.1 MPa)	PI Pre-injection PO Post-injection	h Hydraulic m Mechanical em Electromechanical Mv Solenoid valve	DI Direct injection IDI Indirect injection	Number of cylinders	Max. rated speed rpm	Max. power output per cylinder kW
In-line injection pumps									
M	P, O	60	550	–	m, em	IDI	4…6	5,000	20
A	O	120	750	–	m	DI/IDI	2…12	2,800	27
MW[8]	P, N, O	150	1,100	–	m	DI	4…8	2,600	36
P3000	N, O	250	950	–	m, em	DI	4…12	2,600	45
P7100	N, O	250	1,200	–	m, em	DI	4…12	2,500	55
P8000	N, O	250	1,300	–	m, em	DI	6…12	2,500	55
P8500	N, O	250	1,300	–	m, em	DI	4…12	2,500	55
H1	N	240	1,300	–	em	DI	6…8	2,400	55
H1000	N	250	1,350	–	em	DI	5…8	2,200	70
P10	S, O	800	1,200	–	m, em, h	DI/IDI	6…12	2,400	140
ZW (M)	S, O	900	950	–	m, em, h	DI/IDI	4…12	2,400	160
P9	S, O	1,200	1,200	–	m, em, h	DI/IDI	6…12	2,000	180
CW	S, O	1,500	1,000	–	m, em, h	DI/IDI	6…10	1,800	200
Axial-piston pumps									
VE..F	P	70	350	–	m	IDI	3…6	4,800	25
VE..F	P	70	1,250	–	m	DI	4…6	4,400	25
VE..F	N, O	125	800	–	m	DI	4, 6	3,800	30
VP37 (VE..EDC)	P	70	1,250	–	em[7]	DI	3…6	4,400	25
VP37 (VE..EDC)	O	125	800	–	em[7]	DI	4, 6	3,800	30
VP30 (VE..MV)	P	70	1,400	PI	Mv[7]	DI	4…6	4,500	25
VP30 (VE..MV)	O	125	800	PI	Mv[7]	DI	4, 6	2,600	30
Radial-piston pumps									
VP44 (VR)	P	85	1,900	PI	Mv[7]	DI	4, 6	4,500	25
VP44 (VR)	N	175	1,500	–	Mv[7]	DI	4, 6	3,300	45
Discrete/cylinder-pump systems									
PF(R)…	O	13… 120	450… 1,150	–	m, em	DI/IDI	Any	4,000	4… 30
PF(R)… large-scale diesel	P, N, O, S	150… 18,000	800… 1,500	–	m, em	DI/IDI	Any	300… 2,000	75… 1,000
UIS P1	P	60	2,050	PI	Mv	DI	5[2, 2a]	4,800	25
UIS 30	N	160	1,600	–	Mv	DI	8[2]	4,000	35
UIS 31	N	300	1,600	–	Mv	DI	8[2]	2,400	75
UIS 32	N	400	1,800	–	Mv	DI	8[2]	2,400	80
UPS 12	N	180	1,600	–	Mv	DI	8[2]	2,400	35
UPS 20	N	250	1,800	–	Mv	DI	8[2]	3,000	80
UPS (PF..MV)	S	3,000	1,600	–	Mv	DI	6…20	1,000	450
Common-rail injection systems									
CR 1st generation	P	100	1,350	PI, PO[3]	Mv	DI	3…8	4,800[4]	30
CR 2nd generation	P	100	1,600	PI, PO[5]	Mv	DI	3…8	5,200	30
CR	N, S	400	1,400	PI, PO[6]	Mv	DI	6…16	2,800	200

Overview of distributor fuel-injection pump systems

The combustion processes that take place inside a diesel engine are essentially dependent on the way in which the fuel is delivered by the fuel-injection system. The fuel-injection pump plays a decisive role in that connection. It generates the necessary fuel pressure for fuel injection. The fuel is delivered via high-pressure fuel lines to the nozzles, which in turn inject it into the combustion chamber. Small, fast-running diesel engines require a high-performance fuel-injection system capable of rapid injection sequences, and which is also light in weight and compact in dimensions. Distributor injection pumps meet those requirements. They consist of a small, compact unit comprising the fuel pump, high-pressure fuel-injection pump and control mechanism.

Areas of application

Since its introduction in 1962, the axial-piston distributor injection pump has become the most widely used fuel-injection pump for cars. The pump and its control system have been continually improved over that period. An increase in the fuel-injection pressure was required in order to achieve lower fuel consumption and exhaust-gas emissions on engines with direct injection. A total of more than 45 million distributor injection pumps were produced by Bosch between 1962 and 2001. The available designs and overall system configurations are accordingly varied.

Axial-piston distributor pumps for engines with indirect injection (IDI) generate pressures of as much as 350 bar (35 MPa) at the nozzle. For direct-injection (DI) engines, both axial-piston and radial-piston distributor injection pumps are used. They produce pressures of up to 900 bar (90 MPa) for slow-running engines, and up to 1,900 bar (190 MPa) for fast-running diesels.

The mechanical governors originally used on distributor injection pumps were succeeded by electronic control systems with electrical actuator mechanisms. Later on, pumps with high-pressure solenoid valves were developed.

Apart from their compact dimensions, the characteristic feature of distributor injection pumps is their versatility of application which allows them to be used on cars, light commercial vehicles, fixed-installation engines, and construction and agricultural machinery (off-road vehicles).

The rated speed, power output and design of the diesel engine determine the type and model of distributor injection pump chosen. They are used on engines with between 3 and 6 cylinders.

Axial-piston distributor pumps are used on engines with power outputs of up to 30 kW per cylinder, while radial-piston types are suitable for outputs of up to 45 kW per cylinder.

Distributor injection pumps are lubricated by the fuel and are therefore maintenance-free.

Designs

Three types of distributor injection pump are distinguished according to the method of fuel-quantity control, type of control system and method of high-pressure generation (Figure 1).

Method of fuel-quantity control

Port-controlled injection pumps
The injection duration is varied by means of control ports, channels and slide valves. A hydraulic timing device varies the start of injection.

Solenoid-valve-controlled injection pumps
A high-pressure solenoid valve opens and closes the high-pressure chamber outlet, thereby controlling start of injection and injection duration. Radial-piston distributor injection pumps are always controlled by solenoid valves.

Method of high-pressure generation

Type VE axial-piston distributor pumps
These compress the fuel by means of a piston which moves in an axial direction relative to the pump drive shaft.

Type VR radial-piston distributor pumps
These compress the fuel by means of several pistons arranged radially in relation to the pump drive shaft. Radial-piston pumps can produce higher pressures than axial-piston versions.

Type of control system

Mechanical governor
The fuel-injection pump is controlled by a governor linked to levers, springs, vacuum actuators, etc.

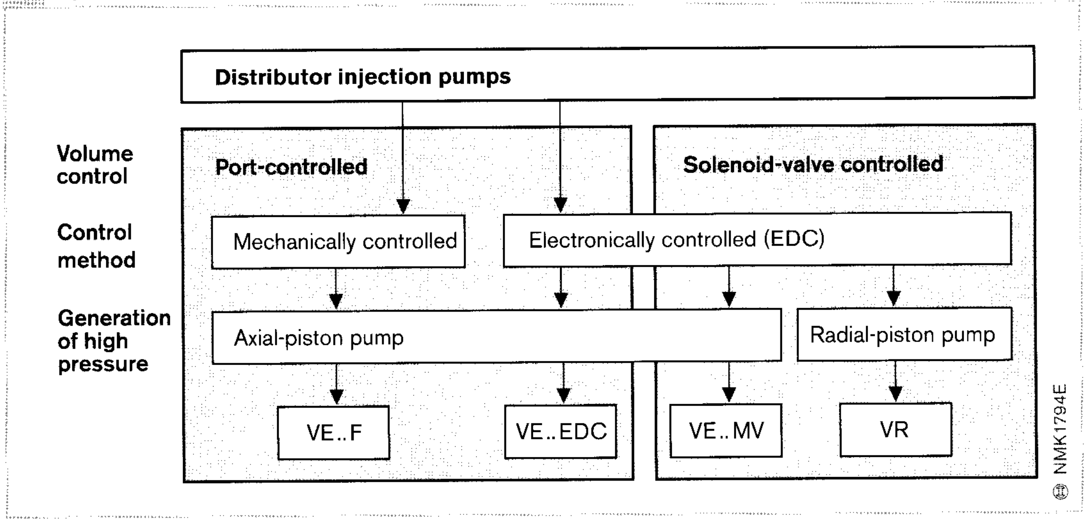
Electronic control system
The driver signals the desired torque output/engine speed by means of the accelerator pedal (sensor). Stored in the control unit are data maps for starting, idling, full load, accelerator characteristics, smoke limits and pump characteristics.

Using that stored information and the actual values from the sensors, specified settings for the fuel-injection pump actuators are calculated. The resulting settings take account of the current engine operating status and the ambient conditions (e.g. crankshaft position and speed, charge-air pressure, temperature of intake air, engine coolant and fuel, vehicle road speed, etc.). The control unit then operates the actuators or the solenoid valves in the fuel-injection pump according to the required settings.

The EDC (Electronic Diesel Control) system offers many advantages over a mechanical governor:

- Lower fuel consumption, lower emissions, higher power output and torque by virtue of more precise control of fuel quantity and start of injection.
- Lower idling speed and ability to adjust to auxiliary systems (e.g. air conditioning) by virtue of better control of engine speed.
- Greater sophistication (e.g. active surge damping, smooth-running control, cruise control).
- Improved diagnostic functions.
- Additional control functions (e.g. preheating function, exhaust-gas recirculation, charge-air pressure control, electronic engine immobilisation).
- Data exchange with other electronic control systems (e.g. traction control system, electronic transmission control) and therefore integration in the vehicle's overall control network.

Types of distributor injection pump



Helix and port-controlled systems

Mechanically controlled distributor injection pumps

Mechanical control is used only on axial-piston distributor pumps. This arrangement's assets consist of low manufacturing cost and relatively simple maintenance.

Mechanical rotational-speed control monitors the various operating conditions to ensure high quality in mixture formation. Supplementary control modules adapt start of delivery and injected-fuel quantity to various engine operating statuses and load factors:

- Engine speed
- Engine load
- Engine temperature
- Charge-air pressure and
- Barometric pressure

In addition to the fuel-injection pump (Fig. 1, 4), the diesel fuel-injection system includes the fuel tank (11), the fuel filter (10), the presupply pump (12), the nozzle-and-holder assembly (8) and the fuel lines (1, 6 and 7). The nozzles and their nozzle-and-holder assemblies are the vital elements of the fuel-injection system. Their design configuration has a major influence on the spray patterns and the rate-of-discharge curves. The solenoid-operated shutoff valve (5) (ELAB) interrupts the flow of fuel to the pump's plunger chamber [1]) when the "ignition" is switched off.

A Bowden cable or mechanical linkage (2) relays driver commands recorded by the accelerator pedal (3) to the fuel-injection pump's controller. Specialized control modules are available to regulate idle, intermediate and high-idle speed along.

The VE..F series designation stands for "Verteilereinspritzpumpe, fliehkraftgeregelt", which translates as flyweight-controlled distributor injection pump.

[1]) The operating concept is reversed on marine engines. On these powerplants, the ELAB shutoff closes under current.

Fuel-injection system with mechanically controlled axial-piston distributor pump, Type VE..F

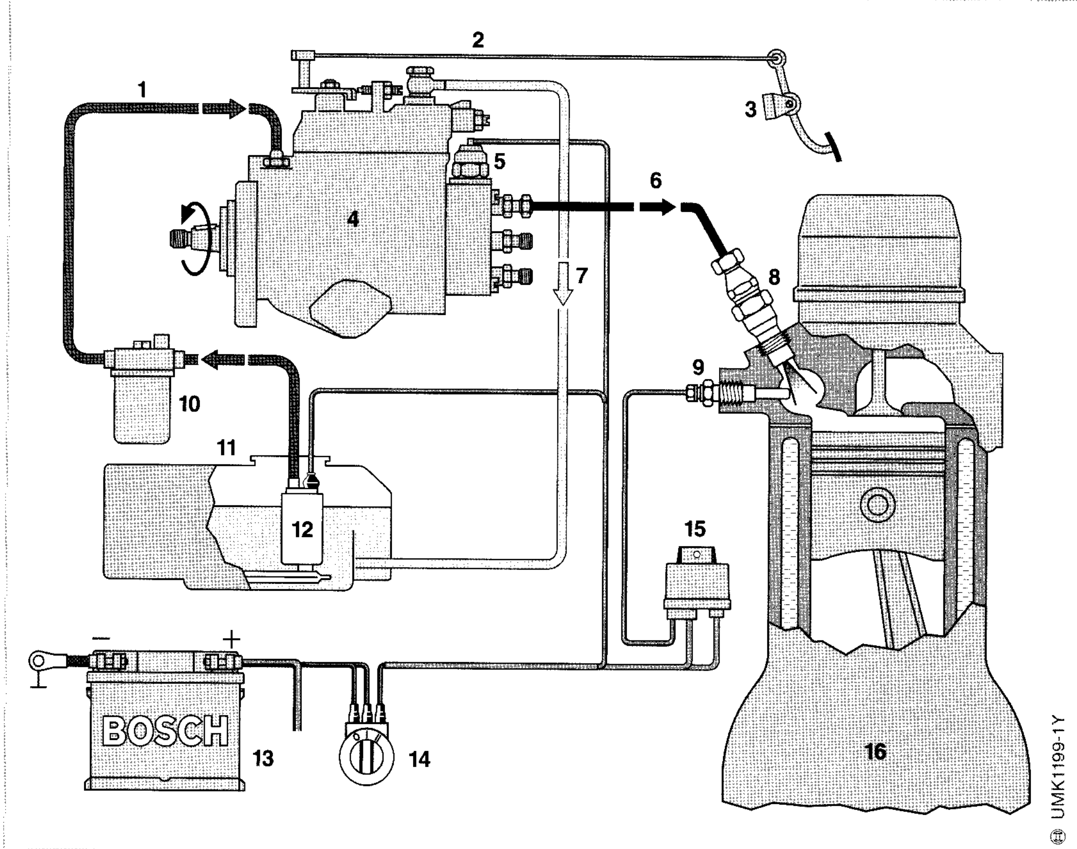


Fig. 1

1 Fuel supply line
2 Linkage
3 Accelerator pedal
4 Distributor injection pump
5 Solenoid-operated shutoff valve (ELAB)
6 High-pressure fuel line
7 Fuel-return line
8 Nozzle-and-holder assembly
9 Sheathed-element glow plug, Type GSK
10 Fuel filter
11 Fuel tank
12 Fuel presupply pump (installed only with extremely long supply lines or substantial differences in relative elevations of fuel tank and fuel-injection pump)
13 Battery
14 Glow-plug and starter switch ("ignition switch")
15 Glow control unit, Type GZS
16 Diesel engine (IDI)

Electronically controlled distributor injection pumps

Electronic Diesel Control (EDC) supports a higher level of functionality than that provided by mechanical control systems. Electric measuring combines with the flexibility contributed by electronic data processing and closed-loop control featuring electric actuators to embed additional operational parameters in the control process.

Figure 2 illustrates the components in a fully-equipped fuel-injection system featuring an electronically-controlled axial-piston distributor pump. Some individual components may not be present in certain applications or vehicle types. The system consists of four sectors:

- Fuel supply (low-pressure circuit)
- Fuel-injection pump
- Electronic Diesel Control (EDC) with system modules for sensors, control unit and final controlling elements (actuators), and
- Peripherals (e.g. turbocharger, exhaust-gas recirculation, glow-plug control, etc.)

The solenoid-controlled actuator mechanism in the distributor injection pump (rotary actuator) replaces the mechanical controller and its auxiliary modules. It employs a shaft to shift the control collar's position and regulate injected fuel quantity. As in the mechanical pump, control collar travel is employed to vary the points at which the port is opened and closed. The ECU uses the stored program map and instantaneous data from the sensors to define the default value for the solenoid actuator position in the fuel-injection pump.

An angle sensor (such as a semidifferential short-circuiting ring sensor) registers the actuator mechanism's angle. This serves as an indicator of control collar travel and this information is fed back to the ECU.

A pulse-controlled solenoid valve compensates for fluctuations in the pump's internal pressure arising from variations in engine speed by shifting the timing device to modify start of delivery.

2 Fuel-injection system with electronically controlled axial-piston distributor pump, Type VE..EDC

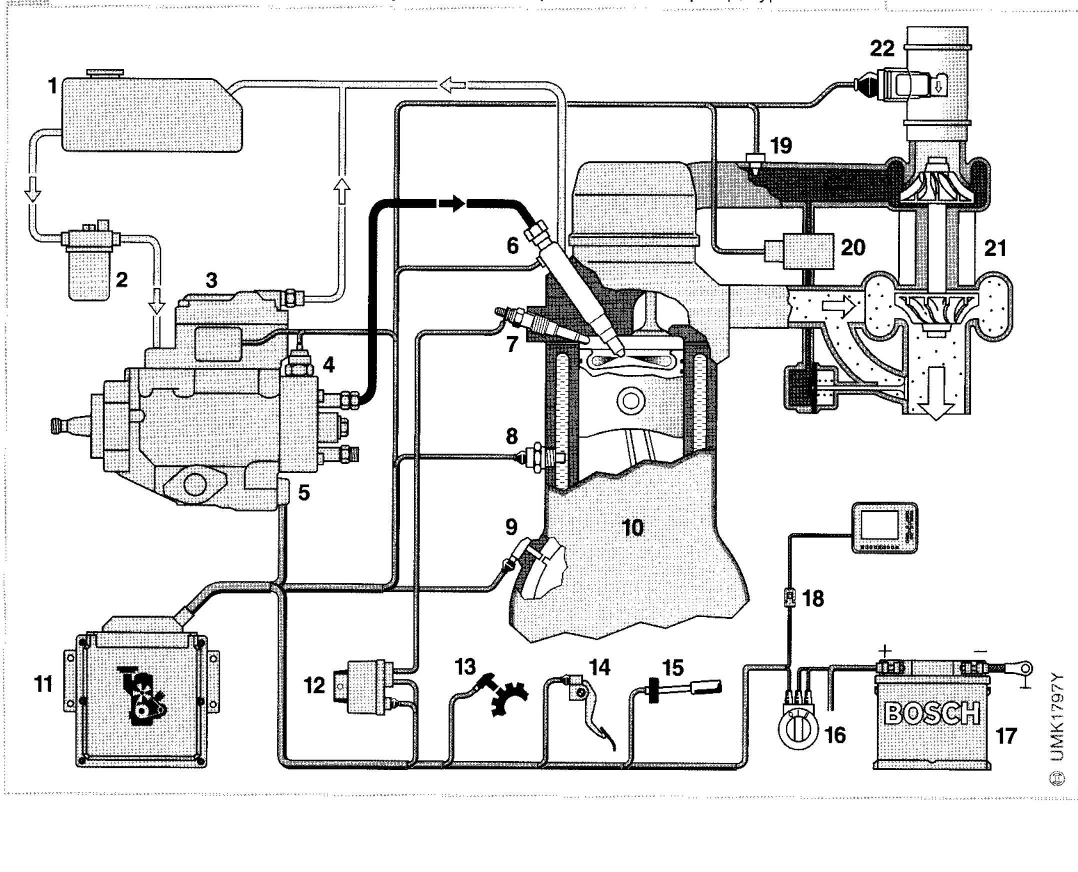


Fig. 2
1 Fuel tank
2 Fuel filter
3 Distributor injection pump with solenoid actuator, rack-travel sensor and fuel-temperature sensor
4 Solenoid-operated shutoff valve, Type ELAB
5 Timing-device solenoid valve
6 Nozzle-and-holder assembly with needle-motion sensor (usually on cylinder no. 1)
7 Glow plug, Type GSK
8 Engine-temperature sensor (in coolant system)
9 Crankshaft-speed sensor
10 Diesel engine (DI)
11 Electronic control unit (MSG
12 Glow control unit, Type GZS
13 Vehicle-speed sensor
14 Accelerator-pedal travel sensor
15 Operator level for cruise control
16 Glow-plug and starter switch ("ignition switch")
17 Battery
18 Diagnostic interface socket
19 Air-temperature sensor
20 Boost-pressure sensor
21 Turbocharger
22 Air-mass meter

Solenoid-valve-controlled systems

Solenoid-valve controlled fuel-injection systems allow a greater degree of flexibility with regard to fuel metering and variation of injection start than port-controlled systems. They also enable pre-injection, which helps to reduce engine noise, and individual adjustment of injection quantity for each cylinder.

Engine management systems that use solenoid-valve controlled distributor injection pumps consist of four stages (Figure 1):

- The fuel supply system (low-pressure stage)
- The high-pressure stage including all the fuel-injection components
- The Electronic Diesel Control (EDC) system made up of sensors, electronic control unit(s) and actuators and
- The air-intake and exhaust-gas systems (air supply, exhaust-gas treatment and exhaust-gas recirculation)

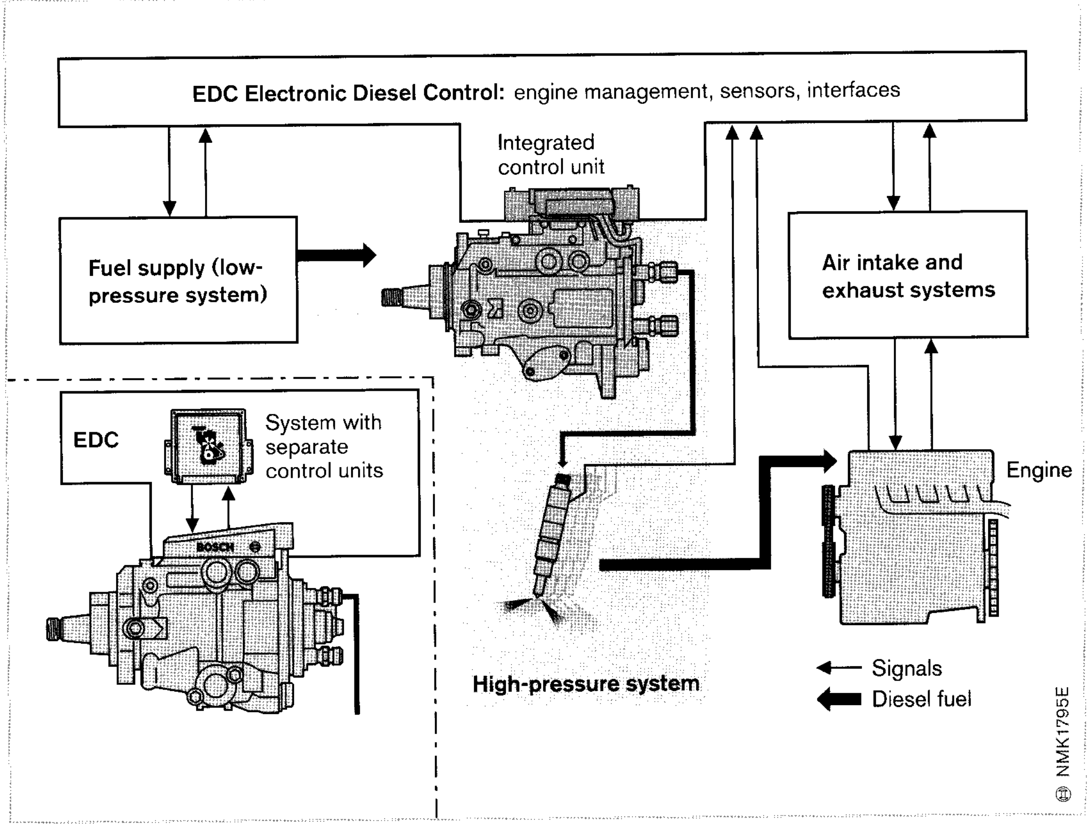
Control-unit configuration

Separate control units

First-generation diesel fuel-injection systems with solenoid-valve controlled distributor injection pumps (Type VE..MV [VP30], VR [VP44] for DI engines and VE..MV [VP29] for IDI engines) require two electronic control units (ECUs) – an engine ECU (Type MSG) and a pump ECU (Type PSG). There were two reasons for this separation of functions: Firstly, it was designed to prevent the overheating of certain electronic components by removing them from the immediate vicinity of pump and engine. Secondly, it allowed the use of short control leads for the solenoid valve. This eliminates interference signals that may occur as a result of very high currents (up to 20 A).

While the pump ECU detects and analyzes the pump's internal sensor signals for angle of rotation and fuel temperature in order to adjust start of injection, the engine ECU

Components of an engine ECU with solenoid-valve-controlled distributor injection pumps



processes all engine and ambient data signals from external sensors and uses them to calculate the actuator adjustments on the fuel-injection pump.

The two ECUs communicate over a CAN interface.

Integrated ECU
Heat-resistant printed-circuit boards designed using hybrid technology allow the integration of the engine ECU in the pump ECU on second-generation solenoid-valve-controlled distributor injection pumps. The use of integrated ECUs permits a space-saving system configuration.

Exhaust-gas treatment

There are various means employed for improving emissions and user-friendliness. They include such things as exhaust-gas recirculation, control of injection pattern (e.g. the use of pre-injection) and the use of higher injection pressures. However, in order to meet the increasingly stringent exhaust-gas regulations, some vehicles will require additional exhaut-gas treatment systems.

A number of exhaut-gas treatment systems are currently under development. It is not yet clear which of them will eventually become established. The most important are dealt with in a separate chapter.

Example of a diesel fuel-injection system with solenoid-valve-controlled radial-piston distributor injection pump and separate control units for engine and pump ECUs

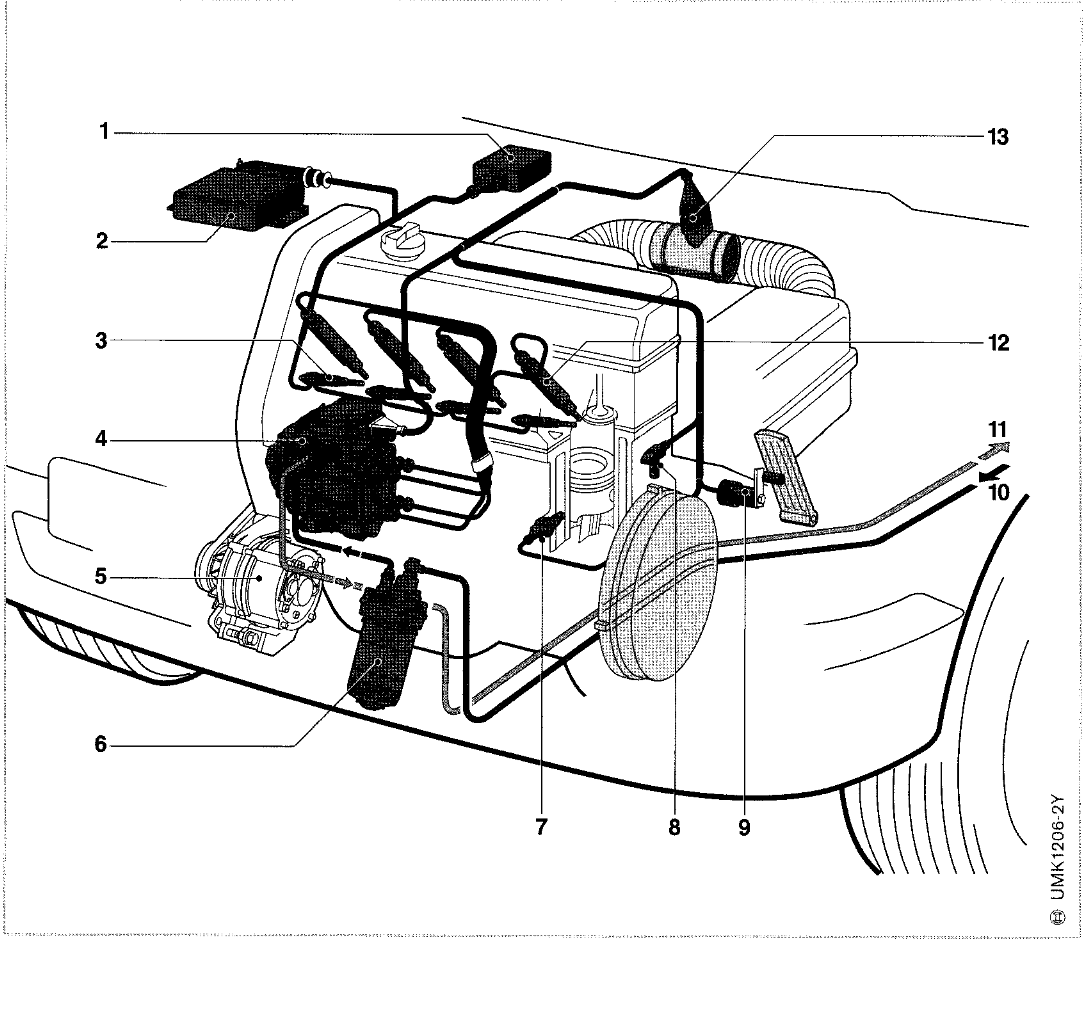


Fig. 2
1 Type GZS glow control unit
2 Type MSG engine ECU
3 Type GSK sheathed-element glow plug
4 Type VP44 radial-piston distributor injection pump with Type PSG5 pump ECU
5 Alternator
6 Fuel filter
7 Engine-temperature sensor (in cooling system)
8 Crankshaft speed sensor
9 Pedal travel sensor
10 Fuel inlet
11 Fuel return
12 Nozzle-and-holder assembly
13 Air-mass meter

System diagram

Figure 3 shows an example of a diesel fuel-injection system using a Type VR radial-piston distributor injection pump on a four-cylinder diesel engine (DI). That pump is fitted with an integrated engine and pump ECU. The diagram shows the full-configuration system. Depending on the nature of the application and the type of vehicle, certain components may not be used.

For the sake of clarity, the sensors and desired-value generators (A) are not shown in their fitted locations. One exception to this is the needle-motion sensor (21).

The CAN bus in the "Interfaces" section (B) provides the means for data exchange with a wide variety of systems and components such as

- The starter motor
- The alternator
- The electronic immobilizer
- The transmission-shift control system
- The traction control system (ASR) and
- The electronic stability program (ESP)

The instrument cluster (12) and the air conditioner (13) can also be connected to the CAN bus.

Fig. 3

Engine, engine ECU and high-pressure fuel-injection components

16 Fuel-injection pump drive
17 Type PSG16 integrated engine/pump ECU
18 Radial-piston distributor injection pump (VP44)
21 Nozzle-and-holder assembly with needle-motion sensor (cylinder no. 1)
22 Sheathed-element glow plug
23 Diesel engine (DI)
M Torque

A Sensors and desired-value generators

1 Pedal-travel sensor
2 Clutch switch
3 Brake contacts (2)
4 Vehicle-speed control operator unit
5 Glow-plug and starter switch ("ignition switch")
6 Vehicle-speed sensor
7 Crankshaft-speed sensor (inductive)
8 Engine-temperature sensor (in coolant system)
9 Intake-air temperature sensor
10 Boost-pressure sensor
11 Hot-film air mass-flow sensor (intake air)

B Interfaces

12 Instrument cluster with signal output for fuel consumption, rotational speed, etc.
13 Air-conditioner compressor and operator unit
14 Diagnosis interface
15 Glow control unit
CAN **C**ontroller **A**rea **N**etwork (onboard serial data bus)

C Fuel supply system (low-pressure stage)

19 Fuel filter with overflow valve
20 Fuel tank with preliminary filter and presupply pump (preliminary pump is only required with long fuel pipes or large height difference between fuel tank and fuel-injection pump)

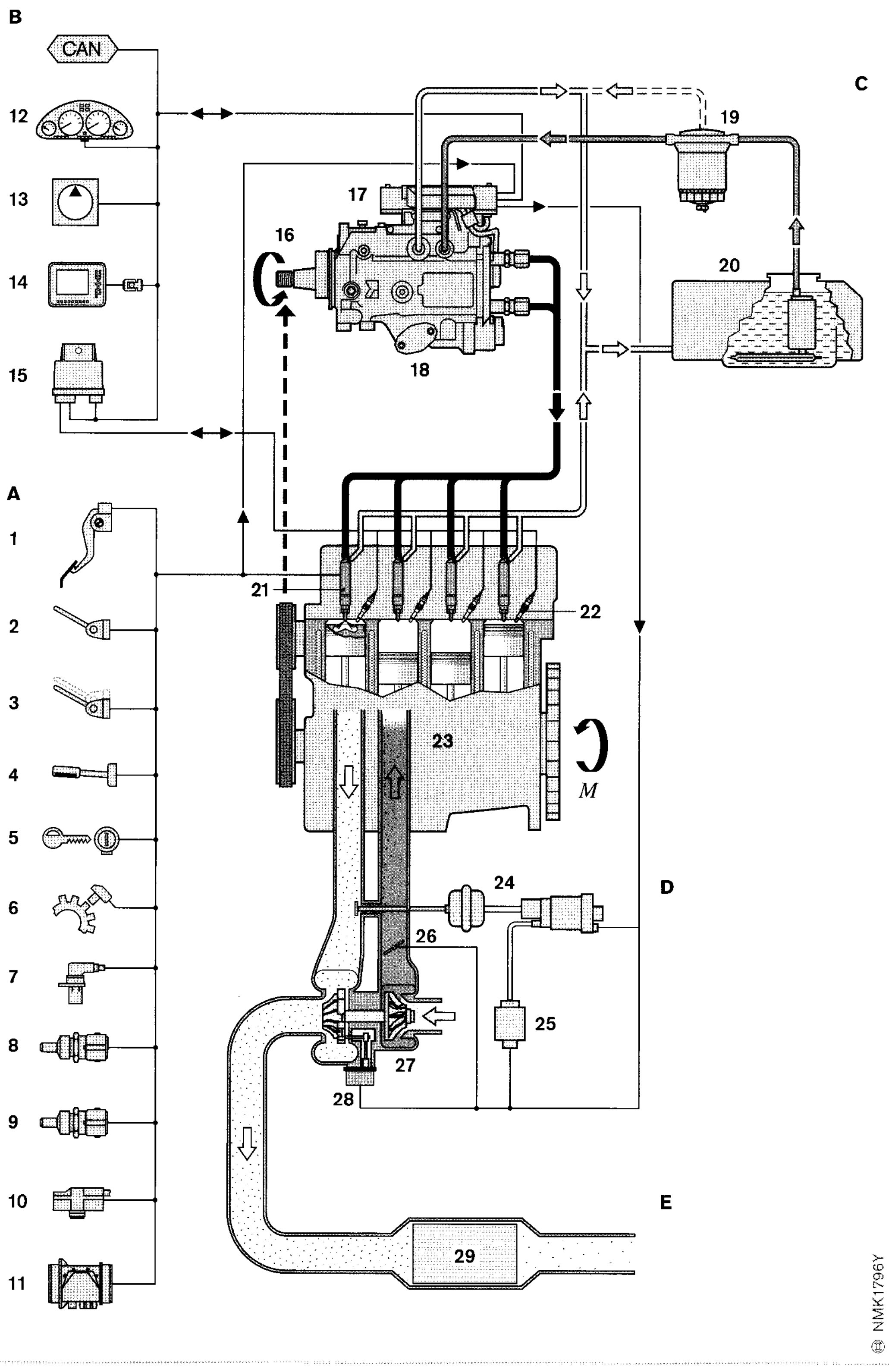
D Air supply system

24 Exhaust-gas recirculation positioner and valve
25 Vacuum pump
26 Control valve
27 Exhaust-gas turbocharger with VTG (variable turbine geometry)
28 Charge-pressure actuator

E Exhaust-gas treatment

29 Diesel-oxidation catalytic converter (DOC)

Diesel fuel-injection system with Type VP44 solenoid-valve-controlled radial-piston distributor injection pump and Type PSG16 integrated engine and pump ECU



Fuel supply (low-pressure stage)

It is the job of the fuel-supply stage to store the required fuel, filter it, and under all operating conditions supply it to the fuel-injection system at the stipulated pressure. For some applications the fuel is also cooled.

The fuel-supply stage comprises the following major components:
- Fuel tank (1)
- Preliminary filter (not on UIS for passenger cars) (2)
- ECU cooler (optional) (3)
- Presupply pump (optional, on passenger cars also in-tank pump) (4)
- Fuel filter (5)
- Fuel pump (low pressure) (6)
- Pressure-control valve (overflow valve) (7)
- Fuel cooler (optional) (9)
- Low-pressure fuel lines

Individual components can be combined to form modules (for instance the fuel pump and the pressure limiter). On the axial and radial-piston distributor pumps, as well as on the Common Rail System, the fuel pump is integrated in the high-pressure pump.

Fuel-supply components (low-pressure stage)

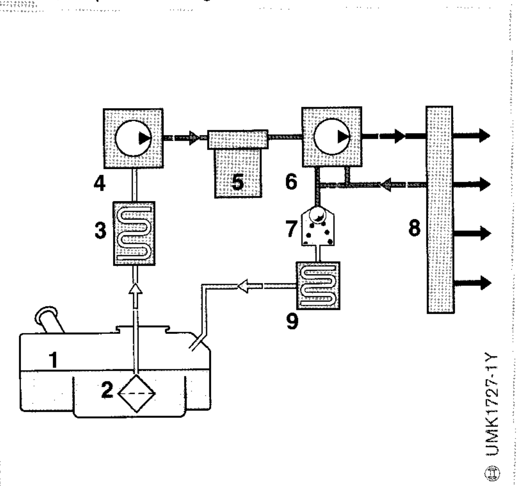


Fig. 1
1 Fuel tank
2 Preliminary filter
3 ECU cooler
4 Presupply pump with non-return valve
5 Fuel filter
6 Fuel pump
7 Pressure-control valve (UIS, UPS)
8 Distributor tube (passenger-car UIS)
9 Fuel cooler (UIS, UPS, CR)

Fuel tank

As its name implies, the fuel tank stores the fuel. It must be corrosion-resistant and not leak even at a pressure defined as double the normal operating pressure, or at least at 0.3 bar overpressure. Suitable openings, safety valves etc. must be provided to permit excess pressure to escape. Fuel must not escape past the filler cap, nor through the pressure-equalisation devices. This also applies in the case of road shocks, in curves, or when the vehicle is tilted. The fuel tank must be remote from the engine so that ignition of the fuel is not to be expected even in the event of an accident.

Fuel lines

Fuel lines for the low-pressure stage can be manufactured from seamless metal tubing, or flame and fuel-resistant synthetic hose. They must be protected against mechanical damage, and must be positioned so that the possibility of dripping or evaporating fuel accumulating on hot components where it can ignite is ruled out. Fuel lines are not to be impaired in their correct operation by vehicle twist, engine movement or similar motions. All fuel-carrying components must be protected against heat which could otherwise impair correct operation.

Diesel fuel filters

The fuel filter removes the solid particles from the fuel in order to reduce its level of contamination. By doing so, it ensures that the injection components which are subject to wear are supplied with fuel which has a minimum level of contamination. In order to ensure long service intervals, the filter must feature adequate particle-storage capacity. A blocked filter results in a reduction of fuel delivery and engine output power drops accordingly.

Extremely high precision applies in the manufacture of the components of the diesel fuel-injection systems, and these react drastically to even the most minute contamination. This means that in order to guarantee that reliability, fuel-consumption figures, and compliance with the emission limits are maintained throughout the vehicle's service life (for commercial vehicles this is taken to be approx. 1,000,000 km), very high demands are made upon the measures taken to protect against wear. The fuel filter must be precisely matched to the fuel-injection system in question.

For extended maintenance intervals or particularly high levels of protection against wear, filter systems are installed which feature a preliminary filter and a fine filter.

Versions

A variety of combinations are available which incorporate the following functions:

Preliminary filters for presupply pumps
The preliminary filter (Fig. 1, Pos. 2) is usually a strainer with 300 mm mesh size, and is installed in addition to the main fuel filter (Fig. 1, Pos. 5).

Main filter
Main filters in the form of easy-change filters (Fig. 2) with pleated-star or wound filter elements (Fig. 2, Pos. 3) are in widespread use, and are screwed onto a filter bracket in the vehicle. Two filters can also be fitted in parallel (higher retention capacity), or in series (multistage filter for increased filtration efficiency or fine filter with precisely matched preliminary filter). Filters in which only the filter element is replaced are becoming increasingly popular again.

Water separator
The diesel fuel can contain water in emulsified or free form (for instance, condensate as a result of temperature change). Such water must be prevented from entering the injection system.

Water droplets form on the filter medium due to the different surface tensions of water and fuel, and accumulate in the water separator (Fig. 2, Pos. 8). For free (non-emulsified) water, an additional water separator can be used which removes water droplets by means of centrifugal force. Conductivity sensors are used to monitor water level.

Fuel preheating
During low-temperature operation, this facility prevents the filter-element pores becoming blocked due to paraffin crystals in the fuel. The preheating components are usually incorporated in the filter and heat the fuel either electrically, by means of the engine coolant, or by using heat from the fuel-recirculation system.

Hand primer pump
Used to refill and vent the system following a filter change, and usually incorporated in the filter cover.

Diesel fuel filter with water separator

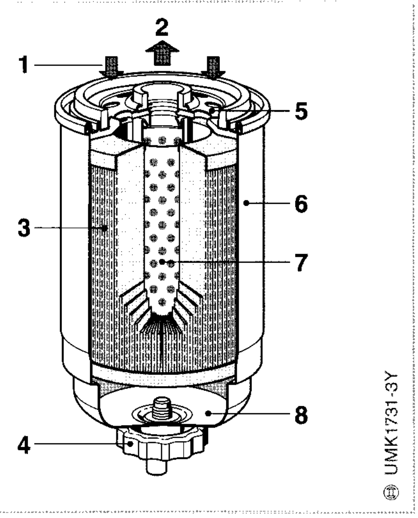


Fig. 2
1 Filter inlet
2 Filter outlet
3 Filter element
4 Water drain screw
5 Filter cover
6 Filter case
7 Support tube
8 Water reservoir

Helix and port-controlled distributor injection pumps

The helix-and-port distributor injection pump is always an axial-piston unit. As the design relies on a single high-pressure element to serve all of the engine's cylinders, units can be extremely compact. Helices, ports and collars modulate injected-fuel quantities. The point in the cycle at which fuel is discharged is determined by the hydraulic timing device. Mechanical control modules or an electric actuator mechanism (refer to section on "Auxiliary control modules for distributor injection pumps") provide flow control. The essential features of this injection-pump design are its maintenance friendliness, low weight and compact dimensions.

This type of pump makes up our VE series. This design replaced the EP/VA series pumps in 1975. In the intervening years it has undergone a range of engineering advances intended to adapt it to meet growing demands. The electric actuator mechanism's advent in 1986 (Fig. 2) started a major expansion in the VE distributor pump's performance potential. In the period up to mid-2002 roughly 42 million VE pumps were manufactured at Bosch. Every year well over a million of these ultra-reliable pumps emerge from assembly lines throughout the world.

The fuel-injection pump pressurizes the diesel fuel to prepare it for injection. The pump supplies the fuel along the high-pressure injection lines to the nozzle-and-holder assemblies that inject it into the combustion chambers.

The shape of the combustion process in the diesel engine depends on several factors, including injected-fuel quantity, the method used to compress and transport the fuel, and the way in which this fuel is injected in the combustion chamber. The critical criteria in this process are:

- The timing and duration of fuel injection
- The distribution pattern in the combustion chamber
- The point at which combustion starts
- The quantity of fuel injected for each degree of crankshaft travel and
- The total quantity of fuel supplied relative to the engine's load factor

Series VE...F mechanically-controlled distributor injection pump on a 4-cylinder diesel engine

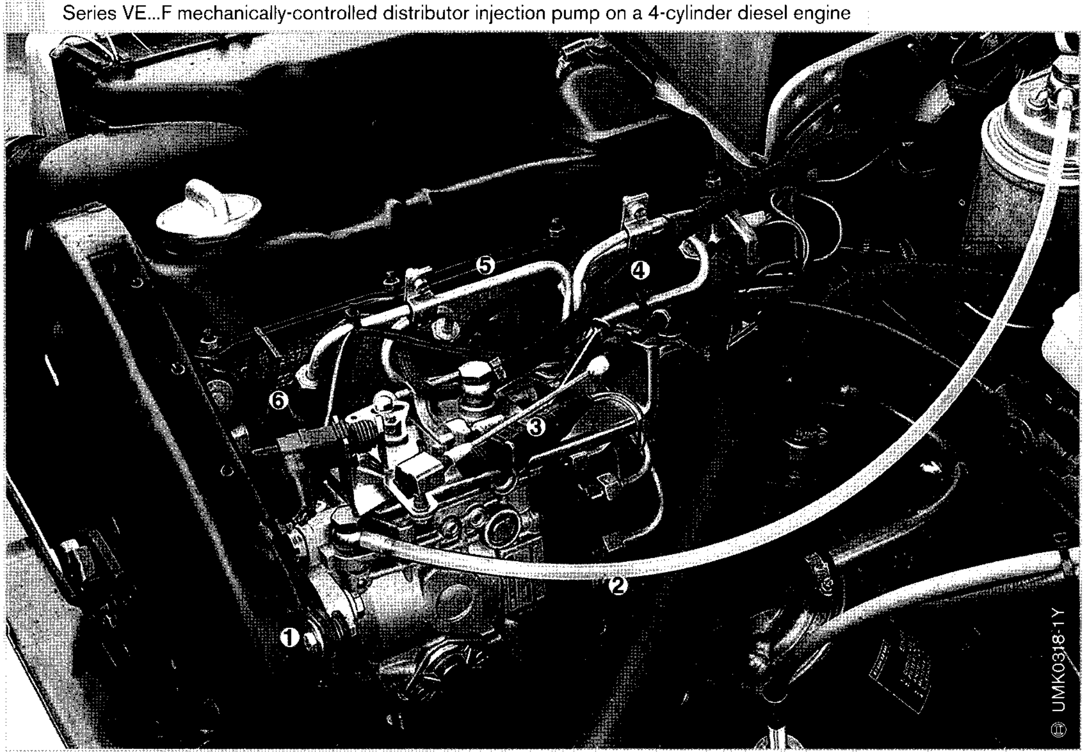


Fig. 1
1 Pump drive
2 Fuel inlet
3 Accelerator pedal linkage
4 Fuel return
5 High-pressure fuel line
6 Nozzle-and-holder assembly

Series VE...EDC axial-piston distributor injection pump with electric actuator mechanism

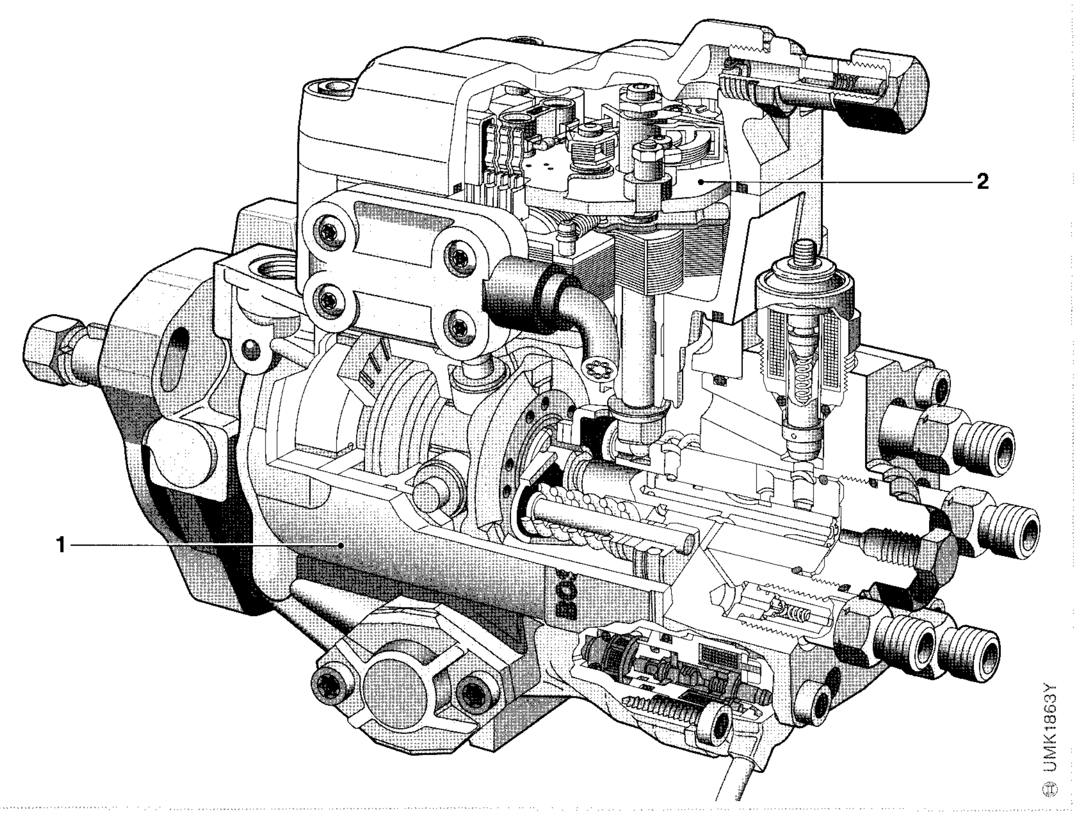


Fig. 2
1 Axial-piston distributor injection pump
2 Electric actuator mechanism

Applications and installation

Fast-turning diesels with limited displacement are one of the applications for helix and port-controlled distributor injection pumps. The pumps furnish fuel in both direct-injection (DI) and prechamber (IDI) powerplants.

The application and the fuel-injection pump's configuration are defined by such factors as nominal speed, power output and design of the individual diesel engine. Distributor injection pumps are fitted in passenger cars, commercial vehicles, construction and agricultural machinery, ships and stationary powerplants to produce power of up to 30 kW per cylinder.

These distributor injection pumps are available with high-pressure spill ports for engines of 3...6 cylinders. The maximum injected-fuel quantity is 125 mm^3 per stroke. Requirements for injection pressure vary according to the specific engine's individual demand (DI or IDI). These pressures reach levels of 350...1,250 bar.

The distributor injection pump is flange-mounted directly on the diesel engine (Fig. 1). Motive force from the crankshaft is transferred to the pump by toothed belt, pinion, ring gear, or a chain and sprocket. Regardless of the arrangement selected, it ensures that the pump remains synchronized with the movement of the pistons in the engine (positive coupling).

On the 4-stroke diesel engine, the rotational speed of the pump is half that of the crankshaft. Expressed another way: the pump's rotational speed is the same as that of the camshaft.

Distributor injection pumps are available for both clockwise and counterclockwise rotation [1]). While the injection sequences varies according to rotational direction, the injection sequence always matches the geometrical progression of the delivery ports.

[1]) Rotational direction as viewed from the pump drive side

In order to avoid confusion with the designations of the engine's cylinders (cylinder no. 1, 2, 3, etc.) the distributor pump's delivery ports carry the alphabetic designations A, B, C, etc. Example: on a four-stroke engine with the firing order 1–3–4–2, the correlation of delivery ports to cylinders is A-1, B-3, C-4 and D-2.

The high-pressure lines running from the fuel-injection pump to the nozzle-and-holder assemblies are kept as short as possible to ensure optimized hydraulic properties. This is why the distributor injection pump is mounted as close as possible to the diesel engine's cylinder head.

The distributor injection pump's lubricant is fuel. This makes the units maintenance-free.

The components and surfaces in the fuel-injection pump's high-pressure stage and the nozzles are both manufactured to tolerances of just a few thousandths of a millimeter. As a result, contamination in the fuel can have a negative impact on operation. This consideration renders the use of high-quality fuel essential, while a special fuel filter, custom-designed to meet the individual fuel-injection system's requirements, is another factor. These two elements combine to prevent damage to pump components, delivery valves and nozzles and ensure trouble-free operation throughout a long service life.

Diesel fuel can absorb 50...200 ppm water (by weight) in solution. Any additional water entering the fuel (such as moisture from condensation) will be present in unbound form. Should this water enter the fuel-injection pump, corrosion damage will be the result. This is why fuel filters equipped with a water trap are vital for the distributor injection pump. The water collected in the trap must be drained at the required intervals. The increasing popularity of diesel engines in passenger cars has resulted in the need for an automatic water level warning system. This system employs a warning lamp to signal that it is time to drain the collected water.

Both the fuel-injection system and the diesel engine in general rely on consistently optimized operating parameters to ensure ideal performance. This is why neither fuel lines nor nozzle-and-holder assemblies should be modified during service work on the vehicle.

Bosch type code for distributor injection pumps

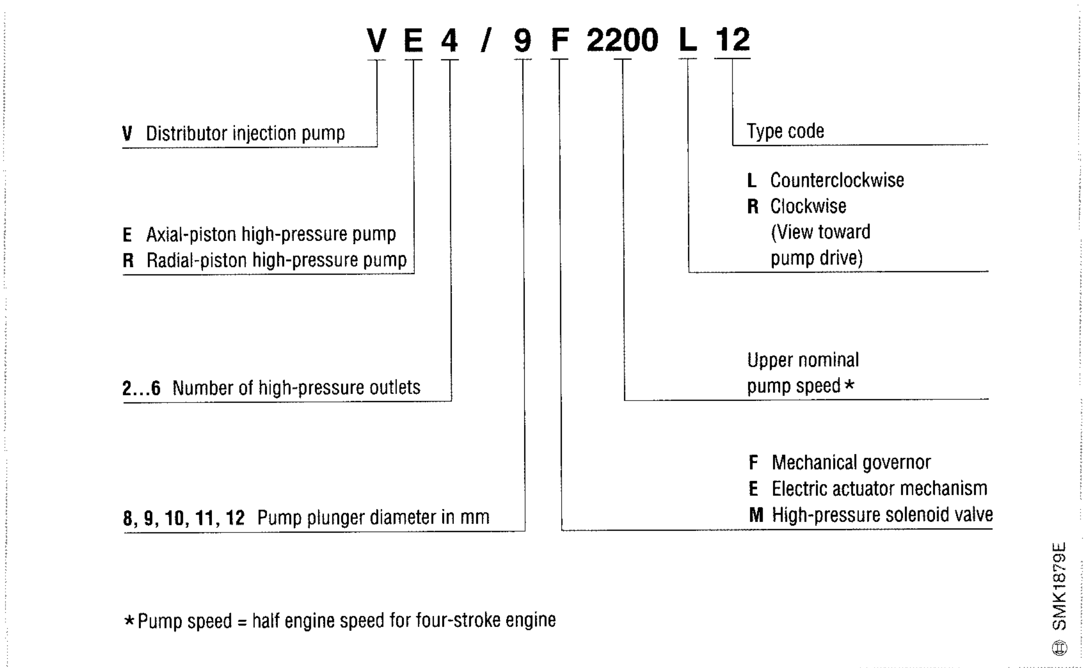


Fig. 1
This data code is affixed to the distributor injection pump housing for precise identification

Design

The distributor injection pump consists of the following main assembly groups (Fig. 2):

Low-pressure stage (7)

The vane-type supply pump takes in the diesel fuel and pressurizes the inner chamber of the pump. The pressure-control valve controls this internal pressure (3...4 bar at idle, 10...12 bar at maximum rpm). Air is discharged through the overflow valve. It also returns fuel in order to cool the pump.

High-pressure pump with distributor (8)

High pressure in the helix and port-controlled distributor injection pump is generated by an axial piston. A distributor slot in the pump's rotating plunger distributes the pressurized fuel to the delivery valves (9). The number of these valves is the same as the number of cylinders in the engine.

Control mechanism (2)

The control mechanism regulates the injection process. The configuration of this mechanism is the most distinctive feature of the helix-and-port distributor pump. Here the operative distinction is between:

- The mechanical governor assembly, with supplementary control modules and switches as needed and
- The electric actuator mechanism (VE...EDC), which is controlled by the engine ECU

Both can be equipped with a solenoid-operated shutoff valve (ELAB) (4). This solenoid device shuts off the fuel-injection system by isolating the high-pressure from the low-pressure side of the pump.

Pump versions equipped with a mechanical governor also include a mechanical shutoff device which is integrated in the governor cover.

Hydraulic timing device (10)

The timing device varies the point at which the pump starts to deliver fuel.

Component assemblies in axial-piston distributor injection pump (schematic)

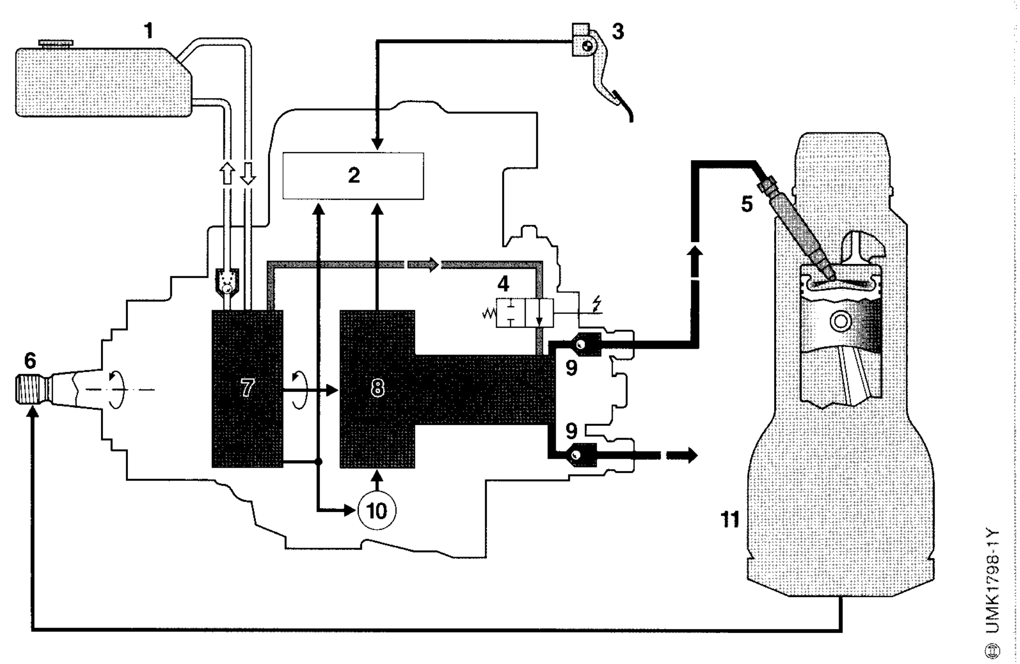


Fig. 2
1 Fuel supply (low-pressure)
2 Control mechanism
3 Accelerator pedal
4 Solenoid-operated shutoff valve (ELAB)
5 Nozzle-and-holder assembly
6 Pump drive
7 Low-pressure stage (vane-type supply pump with pressure-control valve and overflow throttle valve)
8 High-pressure pump with fuel rail
9 Delivery valve
10 Hydraulic timing device
11 Diesel engine

Fuel flow and control lever

The diesel engine powers an input shaft mounted in two plain bearings in the pump's housing (Fig. 4, Pos. 2). The input shaft supports the vane-type supply pump (3) and also drives the high-pressure pump's cam plate (6). The cam plate's travel pattern is both rotational and reciprocating; its motion is transferred to the plunger (11).

On injection pumps with mechanical control, the input shaft drives the control assembly (9) via a gear pair (4) with a rubber damper.

At the top of the control assembly is a control-lever shaft connected to the external control lever (1) on the governor cover. This control-lever shaft intervenes in pump operation based on the commands transmitted to it through the linkage leading to the accelerator pedal. The governor cover seals the top of the distributor injection pump.

Fuel supply

The fuel-injection pump depends on a continuous supply of pressurized bubble-free fuel in the high-pressure stage. On passenger cars and light trucks, the difference in the elevations of fuel-injection pump and fuel tank is usually minimal, while the supply lines are short with large diameters. As a result, the suction generated by the distributor injection pump's internal vane-type supply pump is usually sufficient.

Vehicles with substantial elevations differences and/or long fuel lines between fuel tank and fuel-injection pump need a presupply pump to overcome resistance in lines and filters. Gravity-feed fuel-tank operation is found primarily in stationary powerplants.

Component assemblies and their functions (cutaway)

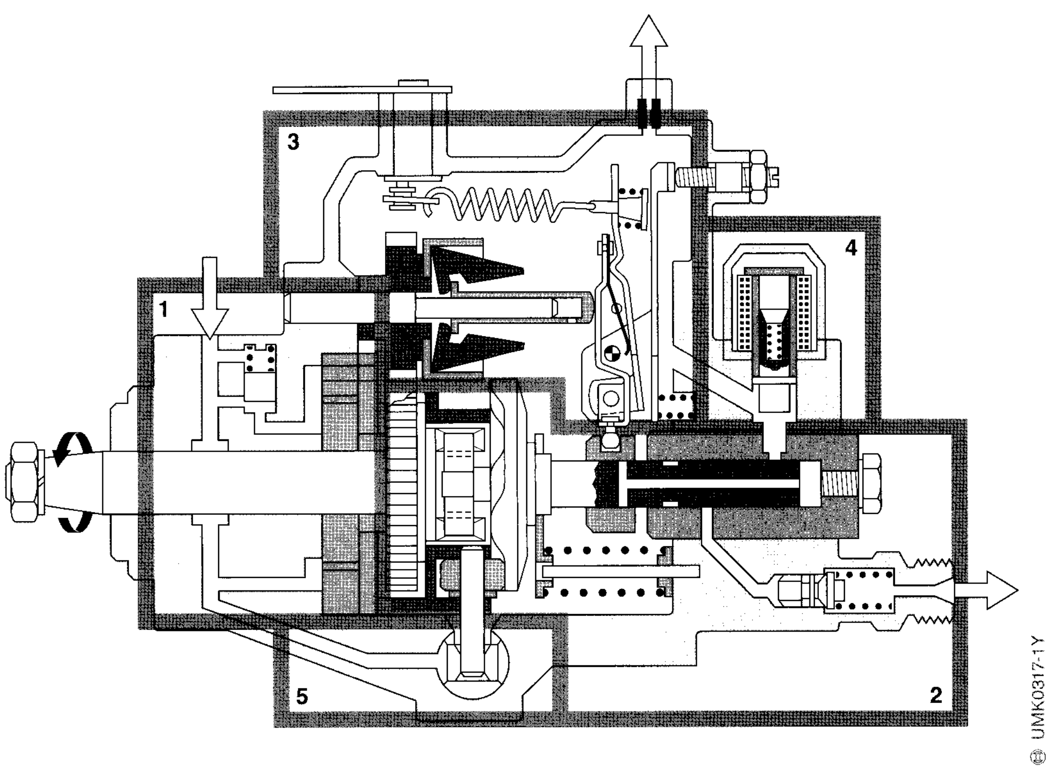


Fig. 3

1 **Vane supply pump with pressure-control valve:** Fuel induction and generation of internal pump pressure
2 **High-pressure pump with distributor head:** Generation of injection pressure, fuel delivery and distribution to individual engine cylinders
3 **Mechanical governor:** Controls rotational speed; control mechanisms vary delivery quantity in control range
4 **Solenoid-operated shutoff valve (ELAB):** Interrupts fuel supply to shut off engine
5 **Timing device:** Determines start of delivery as function of engine speed and (partially) engine load

Series VE...F axial-piston distributor injection pump with mechanical governor

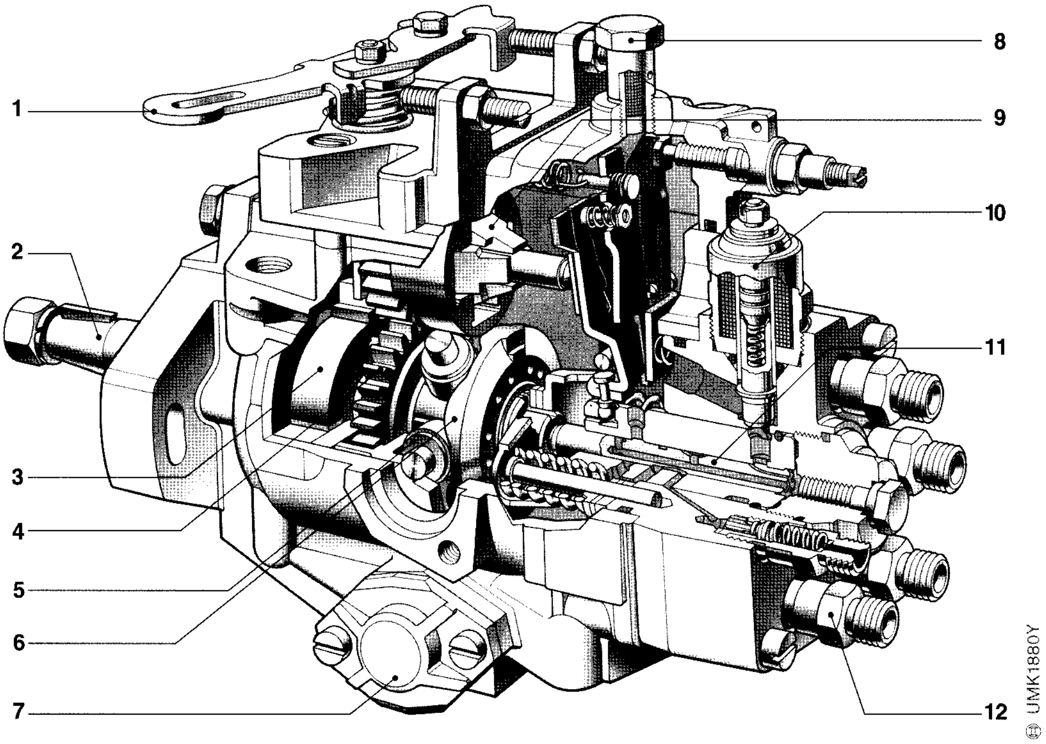


Fig. 4
1 Flow-control lever (linked to accelerator pedal)
2 Input shaft
3 Vane-type supply pump
4 Governor drive gear
5 Roller on roller ring
6 Cam plate
7 Hydraulic timing device
8 Overflow restriction
9 Governor assembly (mechanical governor)
10 Solenoid-operated shutoff valve (ELAB)
11 Distributor plunger
12 Delivery valve

Component assemblies and their locations

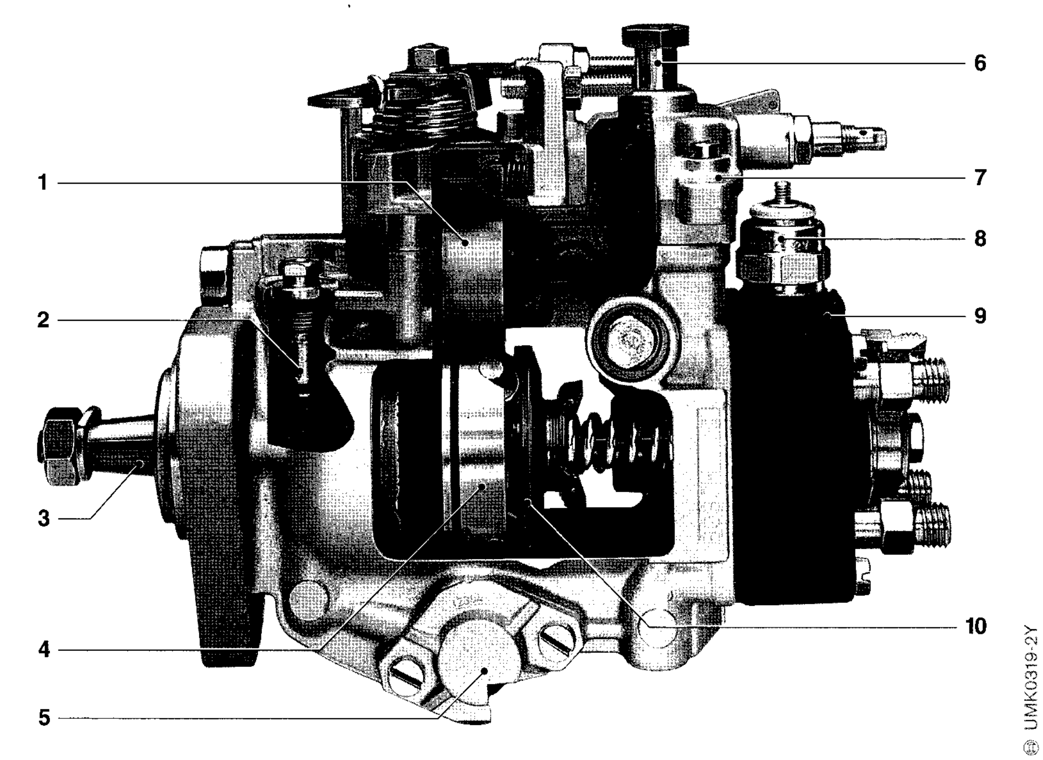


Fig. 5
1 Governor assembly
2 Pressure-control valve
3 Input shaft
4 Roller ring
5 Hydraulic timing device
6 Overflow restriction
7 Governor cover
8 Solenoid-operated shutoff valve (ELAB)
9 Distributor head with high-pressure pump
10 Cam plate

Low-pressure stage

The distributor injection pump's low-pressure stage comprises the following components (Fig. 1):

- The *vane-type supply pump* (4) supplies the fuel
- The *pressure-control valve* (3) maintains the specified fuel pressure in the system
- The *overflow restriction* (9) returns a defined amount of fuel to the pump to promote cooling

Vane-type supply pump

The vane-type supply pump extracts the fuel from the tank and conveys it through the supply lines and filters. As each rotation supplies an approximately constant amount of fuel to the inside of the fuel-injection pump, the supply volume increases as a function of engine speed. Thus the volume of fuel that the pump delivers reflects its own rotational speed, with progressively more fuel being supplied as pump speed increases. Pressurized fuel for the high-pressure side is available in the fuel-injection pump.

Design

The vane-type supply pump is mounted on the main pump unit's input shaft (Fig. 2). The impeller (10) is mounted concentrically on the input shaft (8) and powers the former through a disk spring (7). An eccentric ring (2) installed in the pump housing (5) surrounds the impeller.

Operating concept

As the impeller rotates, centrifugal force presses the four floating blades (9) outward against the eccentric ring. The fuel in the gap between the bottom of the blade and the impeller body supports the blade's outward motion.

Fuel travels through the fuel-injection pump housing's inlet passage and supply channel (4) to a chamber formed by the impeller, blades and eccentric ring, called the cell (3). The rotation presses the fuel from between

Interplay of vane-type supply pump, pressure-control valve and overflow restriction

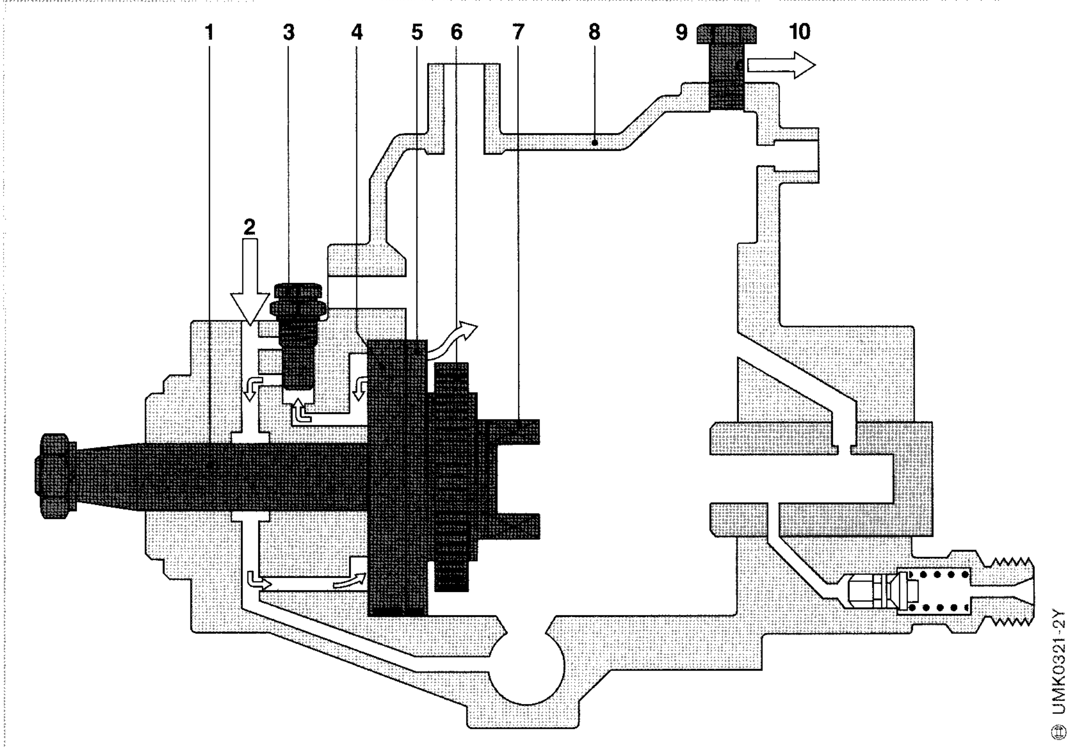


Fig. 1
1 Pump drive
2 Fuel supply
3 Pressure-control valve
4 Eccentric ring on vane supply pump
5 Support ring
6 Governor drive gear
7 High-pressure pump drive claw
8 Pump housing
9 Overflow restriction
10 Fuel return

the blades toward the spill port (6), from where it proceeds through a bore to the pump's inner chamber. The eccentric shape of the ring's inner surface decreases the volume of the cell as the vane-type supply pump rotates to compress the fuel. A portion of the fuel proceeds through a second bore to the pressure-control valve (see Fig. 1).

The inlet and discharge sides operate using suction and pressure cells and have the shape of kidneys.

Pressure-control valve

As fuel delivery from the vane-type supply pump increases as a function of pump speed, the pump's internal chamber pressure is proportional to the engine's rotational speed. The hydraulic timing device relies on these higher pressurization levels to operate (see section on "Auxiliary control modules for distributor injection pumps"). The pressure-control valve is needed to govern pressurization and ensure that pressures correspond to the levels required for optimized operation of both the timing device and the

Vane-type supply pump for low-pressure delivery

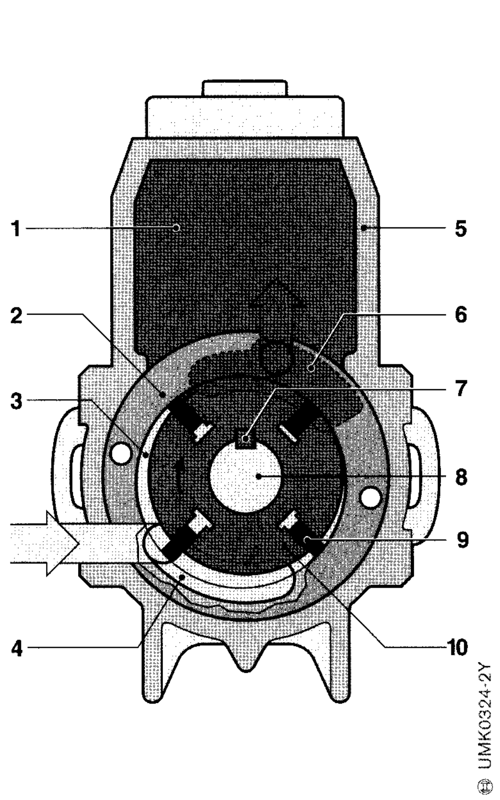


Fig. 2
1 Pump inner chamber
2 Eccentric ring
3 Crescent-shaped cell
4 Fuel inlet (suction cells)
5 Pump housing
6 Fuel discharge (pressure cells)
7 Woodruff key
8 Input shaft
9 Blade
10 Impeller

Components of vane-type supply pump on input shaft

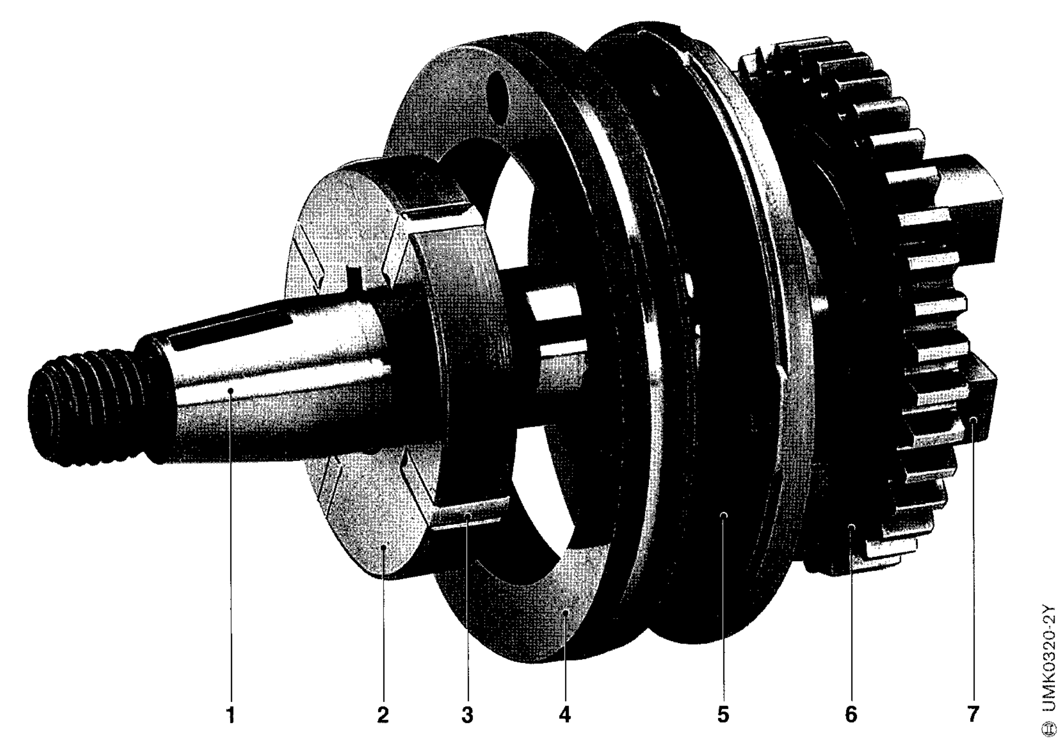
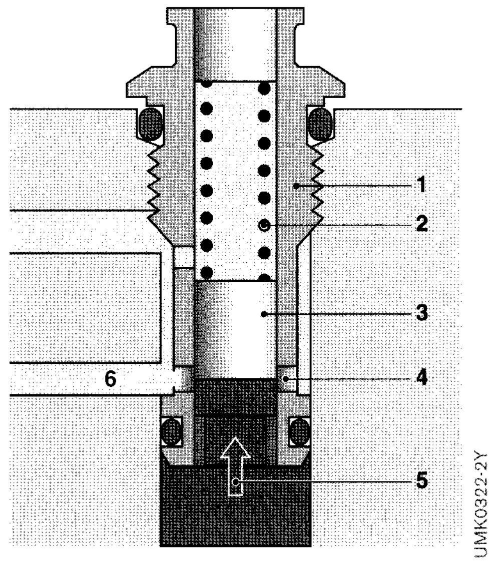


Fig. 3
1 Input shaft
2 Impeller
3 Blade
4 Eccentric ring
5 Support ring
6 Governor drive gear
7 High-pressure pump drive claw

Pressure-control valve



Overflow valve

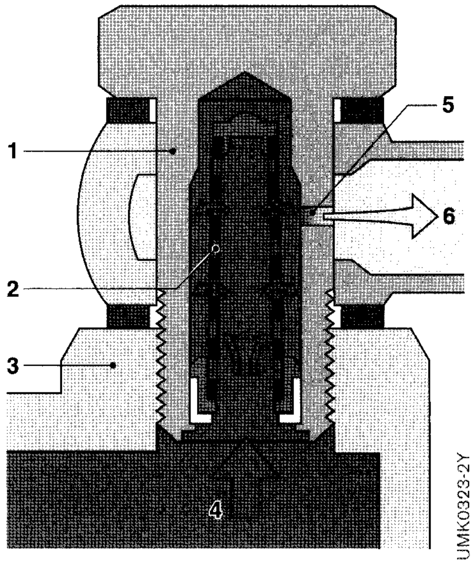


Fig. 4

1 Valve body
2 Compression spring
3 Valve plunger
4 Bore
5 Supply from vane-type supply pump
6 Return to vane-type supply pump

Fig. 5

1 Housing
2 Filter
3 Governor cover
4 Fuel supply
5 Throttle bore
6 Return line to fuel tank

engine itself. The regulator controls internal chamber pressures in the fuel-injection pump according to the quantity of fuel supplied by the vane-type supply pump. At a specific rotational speed, a specific internal pump pressure occurs, which then induces a defined shift in start of delivery.

A passage connects the pressure-control valve to the pressure cell (Fig. 2). It is directly adjacent to the vane-type supply pump.

The pressure-control valve is a spring-loaded slide valve (Fig. 4). When fuel pressure rises beyond a specified level, it pushes back the valve plunger (3), compressing the spring (2) and simultaneously exposing the return passage. The fuel can now proceed through the passage to the vane-type supply pump's suction side (6). When fuel pressure is low, the spring holds the return passage closed. The actual opening pressure is defined by the adjustable tension of the spring.

Overflow restriction

The distributor injection pump is cooled by fuel that flows back to the tank via an overflow restriction screwed to the governor cover (Fig. 5). The overflow restriction is located at the fuel-injection pump's highest point to bleed air automatically. In applications where higher internal pump pressures are required for low-speed operation, an overflow valve can be installed in place of the overflow restriction. This spring-loaded ball valve functions as a pressure-control valve.

The amount of fuel that the overflow restriction's throttle port (5) allows to return to the fuel tank varies as a function of pressure (6). The flow resistance furnished by the port maintains the pump's internal pressure. Because a precisely defined fuel pressure is required for each individual engine speed, the overflow restriction and the pressure-control valve must be matched.

High-pressure pump with fuel distributor

The fuel-injection pump's high-pressure stage pressurizes the fuel to the levels required for injection and then distributes this fuel to the cylinders at the specified delivery quantities. The fuel flows through the delivery valve and the high-pressure line to the nozzle-and-holder assembly, where the nozzle injects it into the engine's combustion chamber.

Distributor plunger drive

A power-transfer assembly transmits the rotational motion of the input shaft (Fig. 1, Pos. 1) to the cam plate (6), which is coupled to the distributor plunger (10). In this process, the claws from the input shaft and the cam plate engage in the intermediate yoke (3).

The cam plate transforms the input shaft's rotation into a motion pattern that combines rotation with reciprocation (total stroke of 2.2...3.5 mm, depending on pump version). The shaft's motion is translated by the motion of the cams on the cam plate (4) against the rollers on the roller ring (5). While the latter runs on bearing surfaces in the housing, there is no positive connection joining it to the input shaft. Because the profiles of the cams on the cam plate extend along the plane defined by the input shaft, they are sometimes referred to as "axial cams".

The distributor plunger's base (10) rests in the cam plate, where its position is maintained by a locating stud. Plunger diameters range from 8...12 mm, depending on the desired injected-fuel quantity.

High-pressure circuit in inner chamber of a helix and port-controlled distributor injection pump

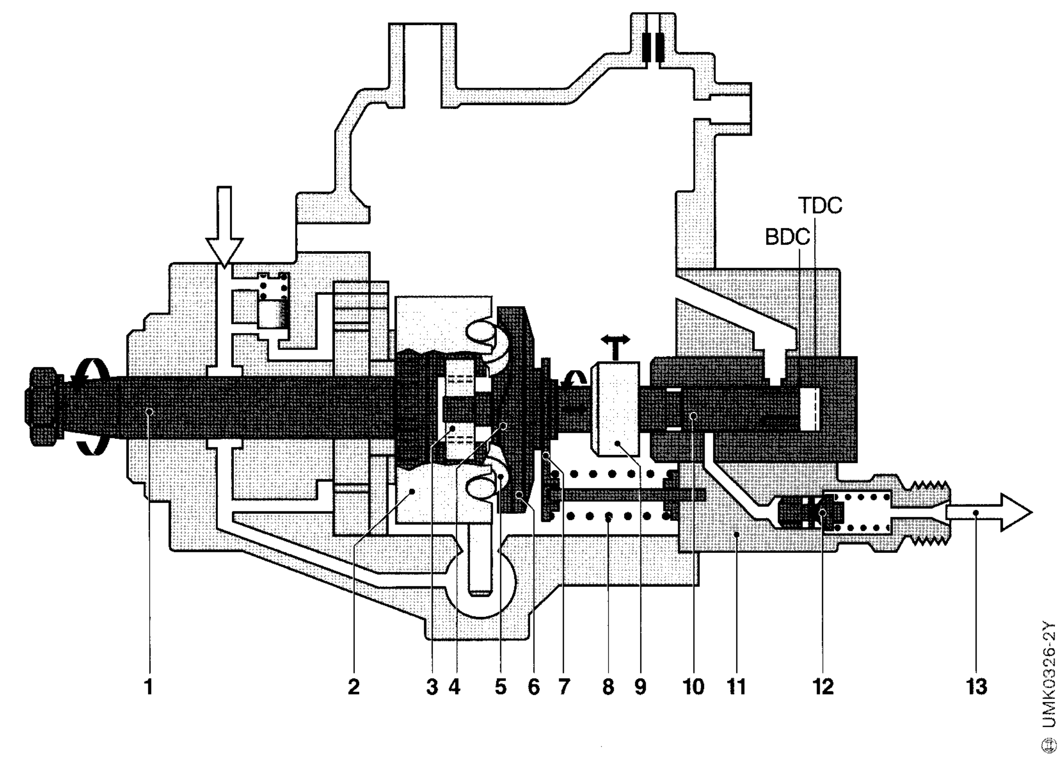


Fig. 1
1 Input shaft
2 Roller ring
3 Yoke
4 Cams
5 Roller
6 Cam plate
7 Spring-loaded cross brace
8 Plunger return spring
9 Control collar
10 Distributor plunger
11 Distributor head
12 Delivery valve
13 Discharge to high-pressure line

TDC Top Dead Center for pump plunger
BDC Bottom Dead Center for pump plunger

Pump assembly with distributor head

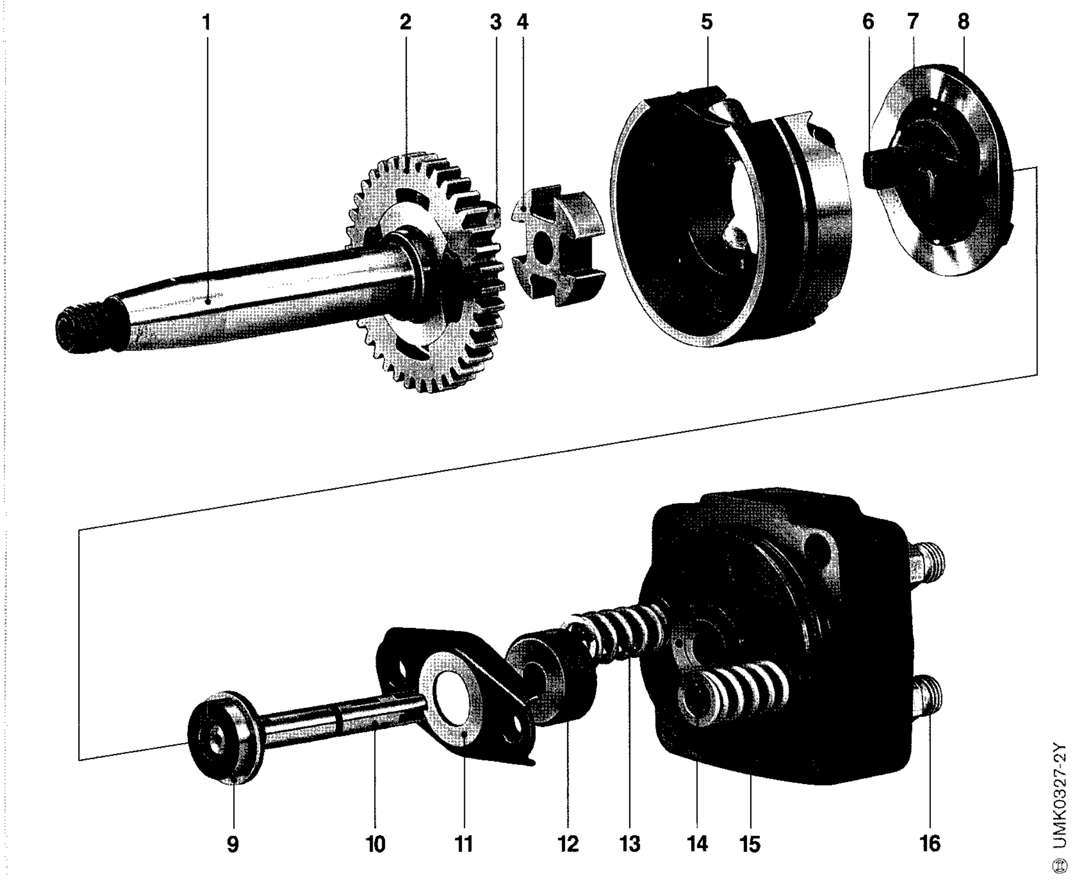


Fig. 2
1 Input shaft
2 Governor drive gear
3 Claw
4 Yoke
5 Roller ring
6 Claw
7 Cam plate
8 Cams
9 Distributor plunger base
10 Distributor plunger
11 Spring-loaded cross brace
12 Control collar
13 Plunger return spring
14 Guide pin
15 Distributor head
16 Delivery valve (discharge to high-pressure line)
9...16 Distributor head assembly

The cam plate's cams drive the plunger toward its **T**op **D**ead **C**enter (TDC) position. The two symmetrically arranged plunger return springs (Fig. 2, Pos. 13) push the plunger back to **B**ottom **D**ead **C**enter (BDC). At one end, these springs rest against the distributor body (15), while the force from the other end is transferred to the distributor plunger (10) through a spring coupling (11). The plunger return springs also prevent the cam plate (7) from slipping off the roller ring's rollers (5) in response to high rates of acceleration.

The heights of the plunger return springs are precisely matched to prevent the plunger from tilting in the distributor body.

Cam plates and cam profiles
The number of cams and rollers is determined by the number of cylinders in the engine and the required injection pressure (Fig. 3). The cam profile affects injection pressure as well as the maximum potential injection duration. Here the primary criteria are cam pitch and stroke velocity.

The conditions of injection must be matched to the combustion chamber's configuration and the engine's combustion process (DI or IDI). This is reflected in the cam profiles on the face of the cam plate, which are specially calculated for each engine type. The cam plate serves as a custom component in the specified pump type. This is why cam plates in different types of VE injection types are not mutually interchangeable.

Distributor body

The plunger (5) and the plunger barrel (2) are precisely matched (lapped assembly) in the distributor body (Fig. 4, Pos. 3) which is screwed to the pump housing. The control collar (1) is also part of a custom-matched assembly with the plunger. This allows the components to provide a reliable seal at extremely high pressures. At the same time, a slight pressure loss is not only unavoidable, it is also desirable as a source of lubrication for the plunger. Precise mutual tolerances in these assemblies mean that the entire distributor group must be replaced as a unit; no attempt should ever be made to replace the plunger, distributor body or control collar as individual components.

Also mounted in the distributor body are the solenoid-operated shutoff valve (ELAB) (not shown in this illustration) used to interrupt the fuel supply along with the screw cap (4) with vent screw (6) and delivery valves (7).

Various roller and cam combinations

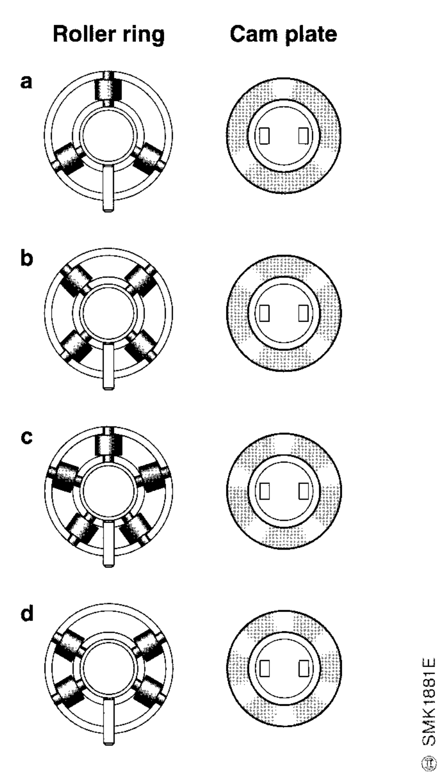


Fig. 3
- a **Three-cylinder engine** Six-cylinder version (d) is also available. In this version, every second spill port is rerouted to the inner chamber of the pump.
- b **Four-cylinder** On two-cylinder engines, every second spill port is rerouted to the inner chamber of the pump.
- c **Five-cylinder engine** The rollers in blue are not fitted to pumps for IDI engines as these operate at lower fuel-injection pressures and reduced physical loads.
- d **Six-cylinder engine** Only four rollers used in this application.

Distributor head component assembly

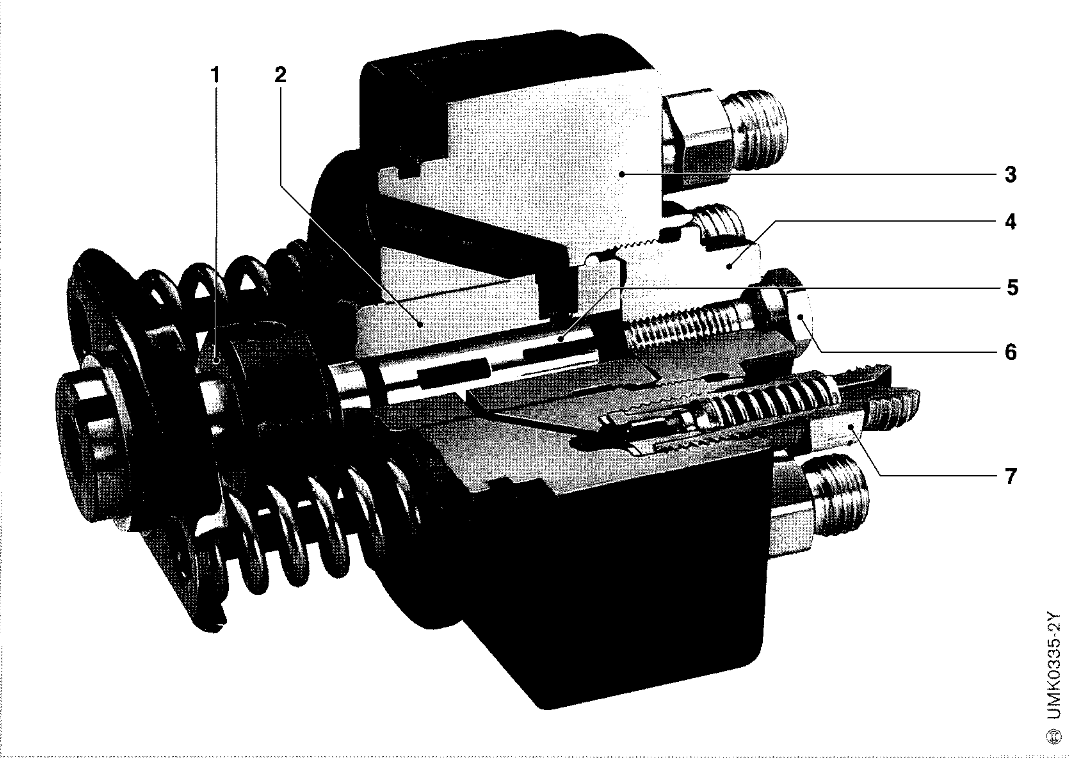


Fig. 4
1 Control collar
2 Plunger barrel
3 Distributor head
4 Screw cap
5 Distributor plunger
6 Vent screw
7 Delivery valve

Fuel metering

The distributor head assembly generates the pressure required for injection. It also distributes the fuel to the various engine cylinders. This dynamic process correlates with several different phases of the plunger stroke, called delivery phases.

The plunger's stroke phases as illustrated in Figure 6 show the process of metering fuel for a single cylinder. Although the plunger's motion is horizontal (as in the in-line fuel-injection pump), the extremes of its travel are still refereed to as top and bottom dead center (TDC and BDC). On four-cylinder engines the plunger rotates by one fourth of a turn during a delivery phase, while a sixth of a turn is available on six cylinder engines.

Suction (6a)
As the plunger travels from top to bottom dead center, fuel flows from the pump chamber and through the exposed inlet passage (2) into the plunger chamber (6) above the plunger. The plunger chamber is also called the element chamber.

Prestroke (option, 6b)
As the plunger rotates, it closes the inlet passage at the bottom dead center end of its travel range and opens the distributor slot (8) to provide a specific discharge.

After reaching bottom dead center, the plunger reverses direction and travels back toward TDC. The fuel flows back to the pump's inner chamber through a slot (7) at the front of the plunger.

This preliminary stroke delays fuel supply and delivery until the profile on the cam plate's cam reaches a point characterized by a more radical rise. The result is a more rapid rise in injection pressure for improved engine performance and lower emissions.

Fuel delivery (effective stroke, 6c)
The plunger continues moving toward TDC, closing the prestroke passage in the process. The enclosed fuel is now compressed. It travels through the distributor slot (8) to the delivery valve on one of the delivery ports (9). The port opens and the fuel is propelled through the high-pressure line to the nozzle-and-holder assembly.

Residual stroke (6d)
The effective stroke phase terminates when the lateral control passage (11) on the plunger reaches the control collar's helix (10). This allows the fuel to escape into the pump's inside chamber, collapsing the pressure in the element chamber. This terminates the delivery process. No further fuel is delivered to the nozzle (end of delivery). The delivery valve closes off the high-pressure line.

Fuel flows through the connection to the inside of the pump for as long as the plunger continues to travel toward TDC. The inlet passage is opened once again in this phase.

The governor or actuator mechanism can shift the control collar to vary the end of delivery and modulate the injected-fuel quantity.

Distributor plunger

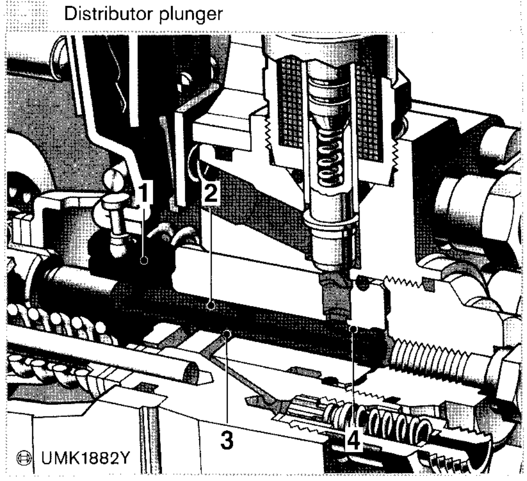


Fig. 5
1 Distributor plunger
2 Control collar
3 Distributor slot
4 Inlet metering slot (a metering slot is provided for each of engine cylinders)

Stroke phases on helix and port-controlled distributor injection pump

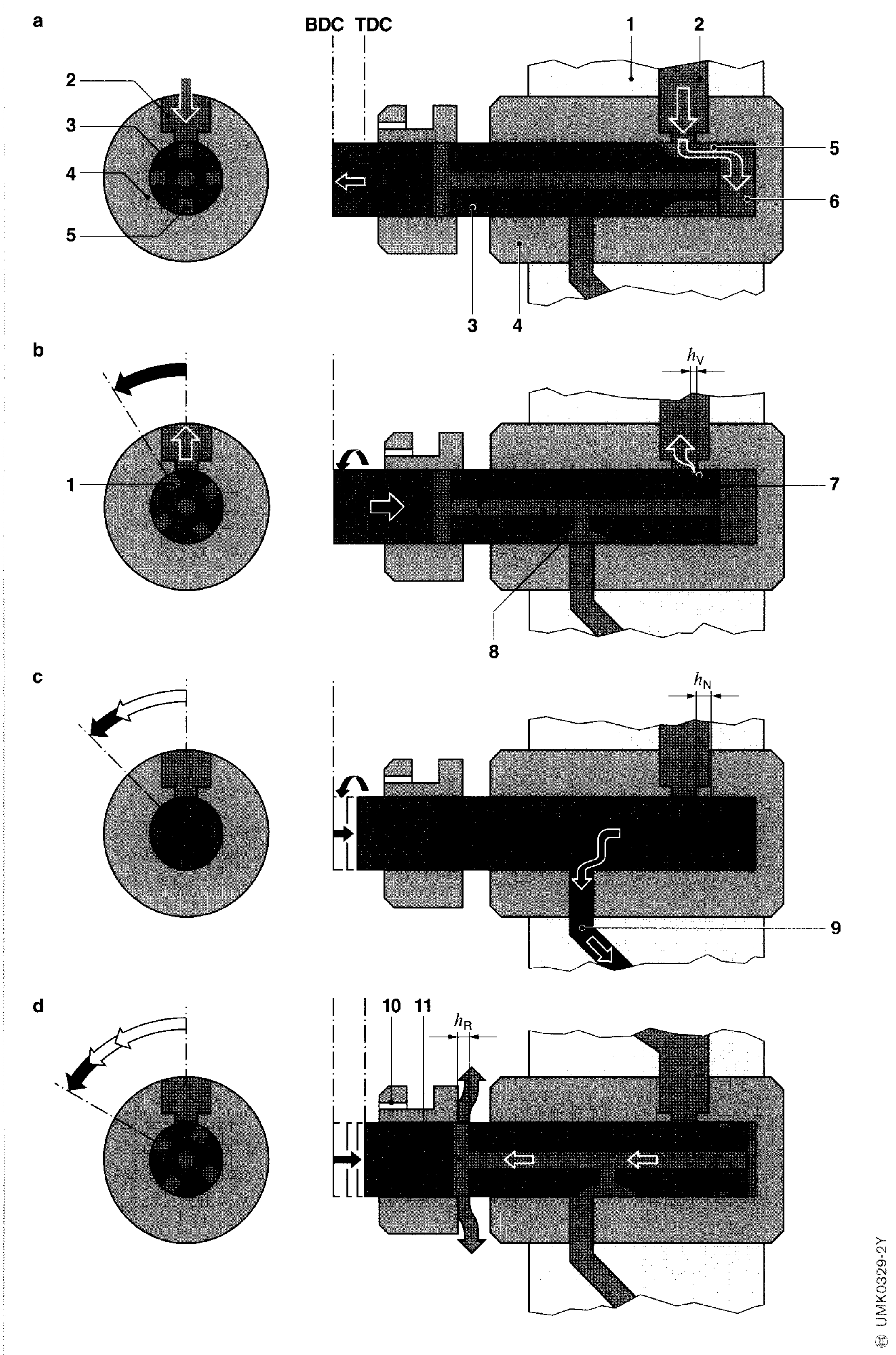


Fig. 6
- a Suction
- b Prestroke
- c Effective stroke
- d Residual stroke

1. Distributor head
2. Inlet passage (fuel supply)
3. Distributor plunger
4. Plunger barrel
5. Inlet metering slot
6. Plunger chamber (element chamber)
7. Prestroke groove
8. Distributor slot
9. Inlet passage to delivery valve
10. Control collar
11. Control bore

h_N Effective stroke
h_R Residual stroke
h_V Prestroke
TDC **T**op **D**ead **C**enter on pump plunger
BDC **B**ottom **D**ead **C**enter on pump plunger

Distributor plunger with precharge groove

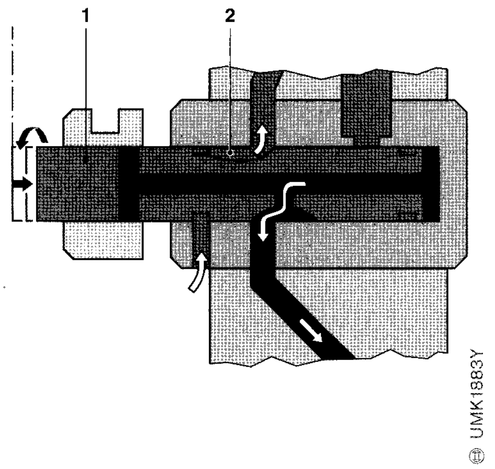


Fig. 7
1 Distributor plunger
2 Precharge groove

Precharge slot

During rapid depressurization at the end of delivery, the venturi effect extracts the remaining fuel from the area between the delivery valve and the plunger. At high rotational speeds with substantial delivery quantities, this area is closed off before enough fuel can flow back in. As a result, the pressure in this area is lower than that in the pump's inner chamber. The slot must thus be recharged before the next fuel injection. This reduces the delivery quantity.

The precharge slot (Fig. 7, Pos. 2) connects the pump's inner chamber to the area between the delivery valve and the plunger. The fuel always flows through the discharge opposite the discharge controlled for delivering fuel.

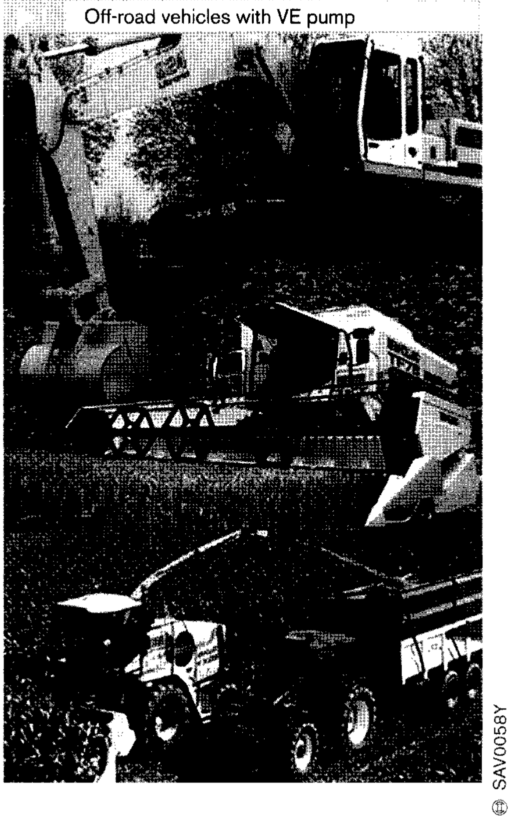
Off-road applications

In addition to their use in on-road vehicles, helix and port-controlled distributor injection pumps are also used in a multitude of off-road applications. This sector includes stationary powerplants as well as construction and agricultural machines. Here, the primary requirements are robust construction and ease of maintenance.

The conditions that fuel-injection systems meet in off-road use can be exceptionally challenging (for instance, when exposed engines are washed down with high-pressure steam cleaners, poor fuel quality, frequent refuelings from canisters, etc.). Special filtration systems operate in conjunction with water separators to protect fuel-injection pumps against damage from fuel of poor quality).

PTOs of the kind employed to drive pumps and cranes rely on constant engine speeds with minimum fluctuation in response to load shifts (low speed droop). Heavy flyweights in the governor assembly provide this performance.

Off-road vehicles with VE pump



Excavator with 112 kW (152 bhp)

Combine harvester with 125 kW (170 bhp)

Harvester with 85 kW (116 bhp)

Tractor with 98 kW (133 bhp)

Delivery valve

The delivery valve closes the high-pressure line to the pump in the period between fuel supply phases. It insulates the high-pressure line from the distributor head.

The delivery valve (Fig. 8) is a slide valve. The high pressure generated during delivery lifts the valve plunger (2) from its seat. The vertical grooves (8) terminating in the ring groove (6) carry the fuel through the delivery-valve holder (4), the high-pressure line and the nozzle holder to the nozzle.

At the end of delivery, the pressure in the plunger chamber above the plunger and in the high-pressure lines falls to the level present in the pump's inner chamber. The valve spring (3) and the static pressure in the high-pressure line push the valve plunger back against its seat.

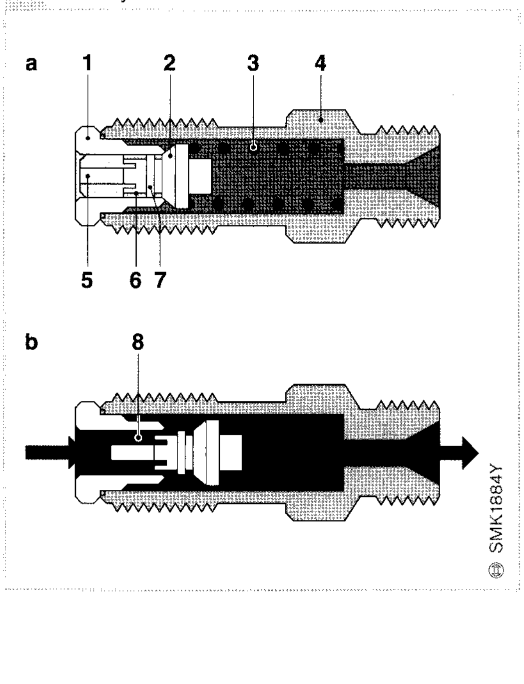
Yet another function of the delivery valve is to relieve injection pressure from the injection line after completion of the delivery phase by increasing a defined volume on the line side. This function is discharged by the retraction piston (7) that closes the valve before the valve plunger (2) reaches its seat. This pressure relief provides precisely calibrated termination of fuel discharge through the nozzle at the end of the injection process. It also stabilizes pressure in the high-pressure lines between injection processes to compensate for shifts in injected-fuel quantities.

Delivery valve with torque control

The dynamic response patterns associated with high-pressure delivery processes in the fuel-injection pump cause flow to increase as a function of rotational speed. However, the engine needs less fuel at high speeds. A positive torque control capable of reducing flow rates as rotational speed increases is thus required in many applications. This function is usually executed by the governor. Another option available on units designed for low injection pressures (IDI powerplants) is to use the pressure-control valve for this function.

Pressure-control valves with torque-control functions feature a torque-control collar (2) adjacent to the retraction piston (Fig. 9, Pos. 1) with either one or two polished recesses (3), according to specific requirements. The resulting restricted opening reduces delivery quantity at high rpm.

Delivery valve



Delivery-valve plunger with torque control

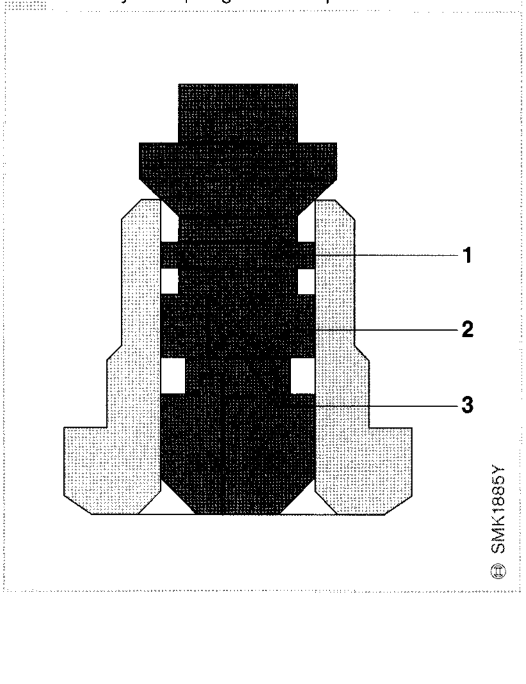


Fig. 8
- a Closed
- b Open

1. Valve holder
2. Valve plunger
3. Valve spring
4. Delivery-valve holder
5. Stem
6. Ring groove
7. Retraction piston
8. Vertical groove

Fig. 9
1. Retraction piston
2. Control-torque collar
3. Specially ground recess

Cutoff bores with flattened surfaces in the control collar, specially designed for the individual engine application, can also provide a limited degree of torque control.

Delivery valve with return-flow restriction

The precise pressure relief required at the end of the injection event generates pressure waves. These are reflected by the delivery valve. At high injection pressures, these pulses have the potential to reopen the nozzle needle or induce phases of negative pressure in the high-pressure line. These processes cause post-injection dribble, with negative consequences on emission properties and/or cavitation and wear in the high-pressure line or at the nozzle.

Harmful reflections are inhibited by a calibrated restriction mounted upstream of the delivery valve, where it affects return flow only. This calibrated restriction attenuates pressure waves but is still small enough to allow maintenance of static pressure between injection events.

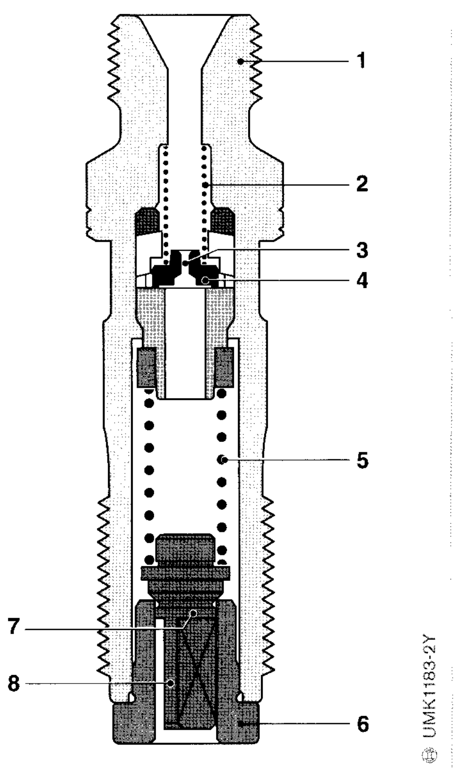
The return-flow restriction consists of a valve plate (Fig. 10, Pos. 4) with a throttle bore (3) and a spring (2). The valve plate lifts to prevent the throttle from exercising any effect on delivery quantity. During return flow, the valve plate closes to inhibit pulsation.

Constant-pressure valve

On high-speed diesel engines, the volumetric relief provided by the retraction piston and delivery valve is often not enough to prevent cavitation, dribble and blowback of combustion gases into the nozzle-and-holder assembly under all conditions.

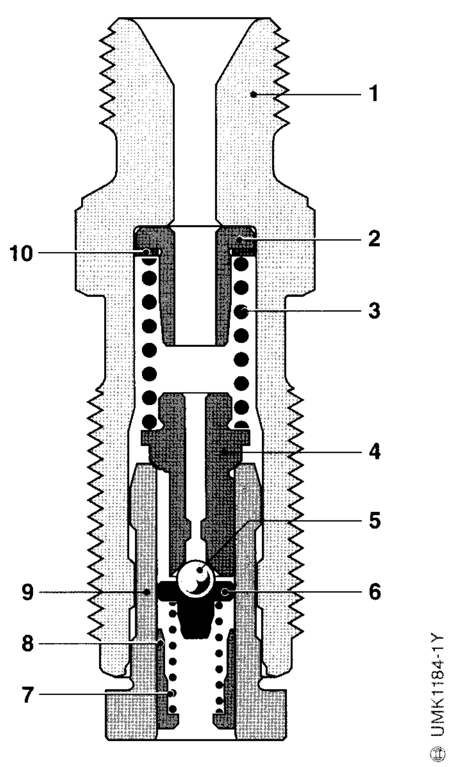
Under these conditions, constant-pressure valves (Fig. 11) are installed. These valves use a single-action non-return valve with adjustable pressure (such as 60 bar) to relieve pressure on the high-pressure system (line and nozzle-and-holder assembly).

Fig. 10 Delivery valve with return-flow restriction



1 Pressure valve holder
2 Valve spring (valve plate)
3 Return-flow restrictor
4 Valve plate
5 Valve spring (delivery valve)
6 Valve holder
7 Relief plunger
8 Plunger stem

Fig. 11 Constant-pressure valve

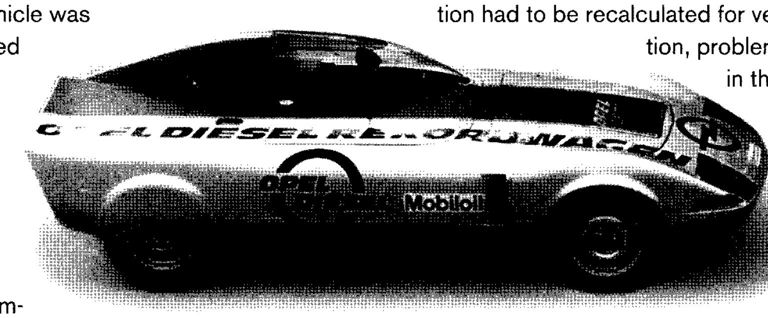


1 Delivery-valve holder
2 Spring-guided filler piece
3 Valve spring (delivery valve)
4 Delivery-valve plunger
5 Ball (constant-pressure valve)
6 Spring seat
7 Valve spring (constant-pressure valve)
8 Setting sleeve
9 Valve holder
10 Shims

Diesel records in 1972

In 1972 a modified Opel GT set a total of 20 international records for diesel-powered vehicles. The original roof structure was replaced to reduce aerodynamic resistance, while the 600-liter fuel tank was installed in place of the passenger seat.
This vehicle was propelled by a 2.1-liter 4-cylinder diesel engine fitted with combustion swirl chambers. A distinguishing factor relative to the standard factory diesel was the exhaust-gas turbocharger. This engine produced 95 DIN bhp (approx. 70 kW) at 4,400 rpm while consuming fuel at the rate of 13 liters per hundred kilometers. Optimized fuel injection was provided by an EP/VA CL 163 rotary axial-piston pump with mechanical injection from Bosch.

Unlike today's diesel engines, this unit was designed around an existing gasoline engine.

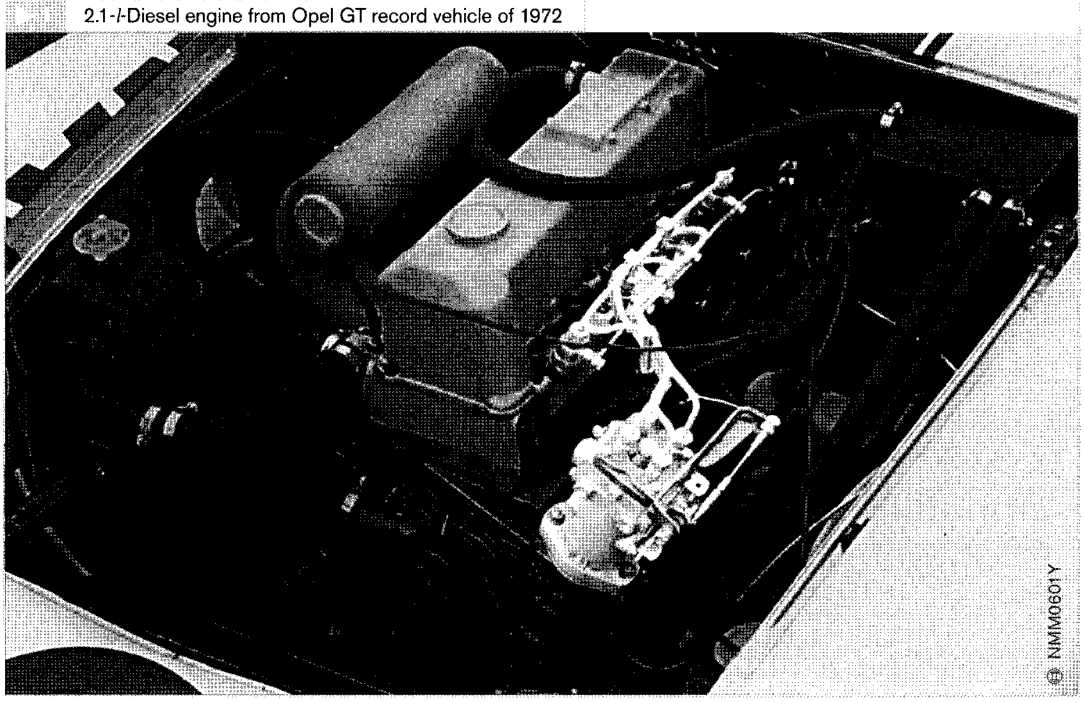


As a result, the distributor injection pump had to be installed in the location originally intended for the ignition distributor. This led to a virtually vertical installation in the engine compartment. Numerous issues awaited resolution: control variables based on horizontal installation had to be recalculated for vertical operation, problems with air in the fuel led to difficult starting, the pump tended to cavitate, etc.

Here is a sampling of the records posted for this vehicle:

- It covered 10,000 kilometers in 52 hours and 23 minutes. This translates into an average speed of 190.9 kph. The car was driven by six drivers relieving each other every three to four hours.
- Absolute world records for all vehicle classes were posted for 10 km (177.4 kph) and 10 miles (184.5 kph) from standing start.

2.1-*l*-Diesel engine from Opel GT record vehicle of 1972



Auxiliary control modules for distributor injection pumps

The auxiliary control modules installed on distributor injection pumps with axial pistons govern start of delivery and regulate the volume of fuel discharged into the combustion chamber during injection. These control modules are composed of mechanical control elements, or actuators. They respond to variations in operating conditions (load factor, rotational speed, charge-air pressure, etc.) with precise adjustments for ideal performance. On distributor injection pumps with Electronic Diesel Control (EDC), an electric actuator mechanism replaces the mechanical actuators.

Overview

Auxiliary control modules adapt the start of delivery and delivery period to reflect both driver demand and the engine's instantaneous operating conditions (Fig. 1).

In the period since the distributor injection pump's introduction in 1962, an extensive range of controllers has evolved to suit a wide range of application environments. Numerous component configurations are also available to provide ideal designs for a variety of engine versions. Any attempt to list all possible versions of auxiliary control modules would break the bounds of this section. Instead, we will concentrate on the most important control modules. The individual units are the:

- Speed governor
- Timing device
- Adjustment and torque control devices
- Switches and sensors
- Shutoff devices
- Electric actuator mechanisms and
- Diesel immobilizers (component in the electronic vehicle immobilizer)

Component assemblies in the axial-piston distributor pump (schematic)

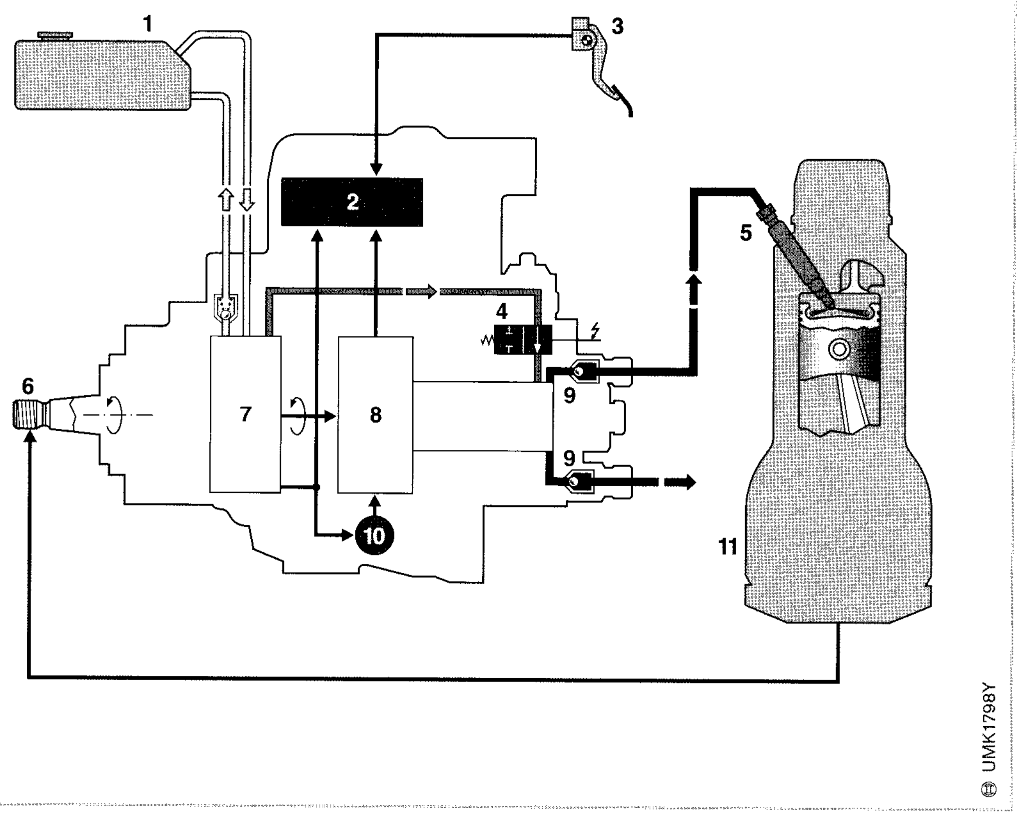
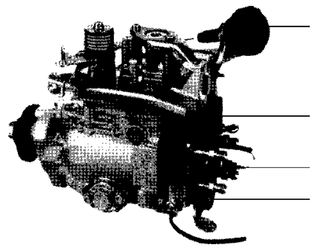


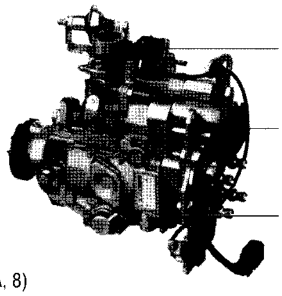
Fig. 1
1 Fuel supply (low-pressure)
2 Controlling system
3 Accelerator pedal
4 ELAB Electric shutoff device
5 Nozzle-and-holder assembly
6 Pump drive assembly
7 Low-pressure stage (vane-type supply pump with pressure-control valve and overflow throttle valve)
8 High-pressure pump with fuel rail
9 Delivery valve
10 Hydraulic timing device
11 Diesel engine

History of the mechanically controlled distributor injection pump from Bosch

1995 Diesel antitheft protection (DDS, 10)

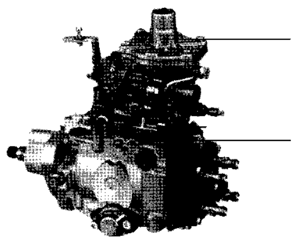


1993 Delivery signal sensor (FSS, 9)



1988 Potentiometer (EGR)

1987 Pneumatic idle-speed increase (PLA, 8) and microswitch (7)

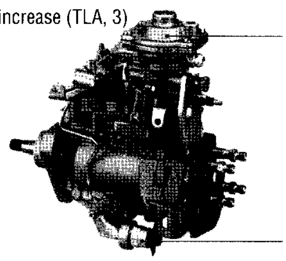


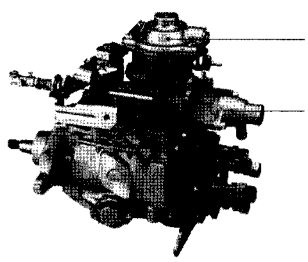
1986 Pneumatic switch

1984 Atmospheric-pressure dependent full-load stop

1983 Manifold-pressure compensator (LDA, 6) and hydraulically controlled torque control (HBA, 5)

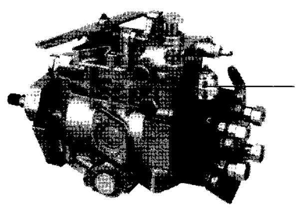
1982 Timing-device solenoid valve (4)
Temperature-controlled idle-speed increase (TLA, 3)

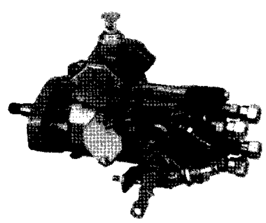




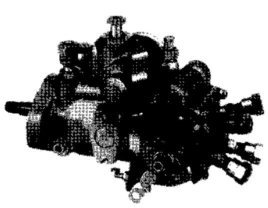
1978 Load-dependent start of delivery with deactivation feature (ALFB, 2)

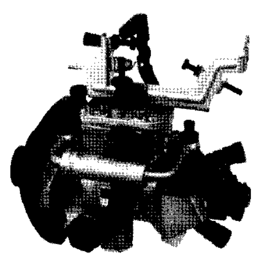
1975 VE (d) with cold-start accelerator (KSB) and solenoid-operated shutoff valve (ELAB, 1)





1968 EP/VA...C (c)





1963 EP/VA...B (b)

1962 EP/VM (a)

SMK1800E

Governors

Function

Engines should not stall when exposed to progressive increases in load during acceleration when starting off. Vehicles should accelerate or decelerate smoothly and without any surge in response to changes in accelerator pedal position. Speed should not vary on gradients of a constant angle when the accelerator pedal does not move. Releasing the pedal should result in engine braking.

The distributor injection pump governor offers active control to help cope with these operating conditions.

The basic function of every governor is to limit the engine's high idle speed. Depending on the individual unit's configuration, other functions can involve maintaining constant engine speeds, which may include specific selected rotational speeds or the entire rev band as well as idle. The different governor designs arise from the various assignments (Fig. 1):

Idle controller
The governor in the fuel-injection pump controls the idle speed of the diesel engine.

Maximum-speed governor
When the load is removed from a diesel engine operated at WOT, its rotational speed should not rise above the maximum permitted idle speed. The governor discharges this function by retracting the control collar toward "stop" at a certain speed. This reduces the flow of fuel into the engine.

Variable-speed governor
This type of governor regulates intermediate engine speeds, holding rotational speed constant within specified limits between idle and high idle. With this setup, fluctuations in the rotational speed n of an engine operating under load at any point in the power band are limited to a range between a speed on the full-load curve n_{VT} and an unloaded engine speed n_{LT}.

Other requirements
In addition to its primary functions, the governor also controls the following:

- Releases or blocks supplementary fuel flow for starting
- Modulates full-throttle delivery quantity as a function of engine speed (torque control)

Special torque-control mechanisms are required for some of these operations. They are described in the text below.

Control precision

The index defining the precision with which the governor regulates rotational speed when load is removed from the engine is the droop-speed control, or droop-speed control. This is the relative increase in engine

Speed governor response curve

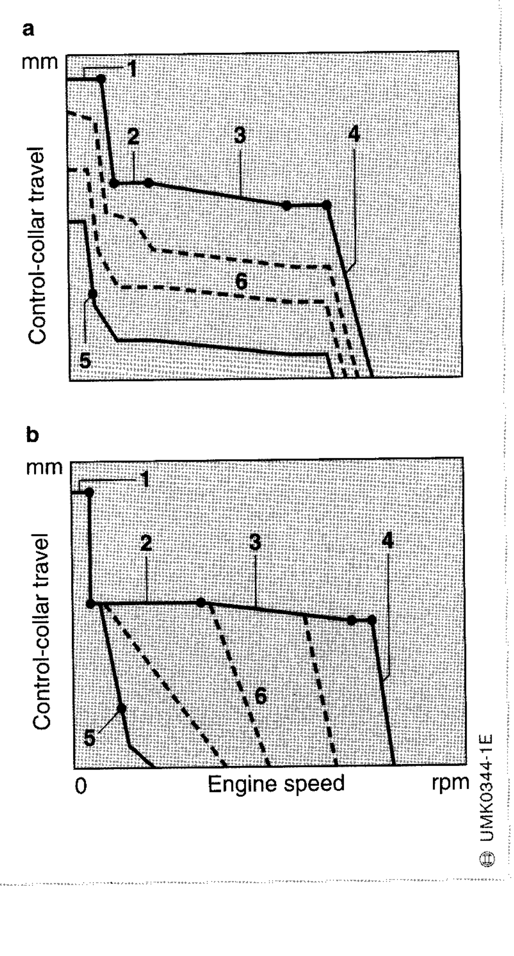


Fig. 1
a Idle/maximum-speed governor
b Variable-speed governor

1 Start quantity
2 Full-load delivery quantity
3 Torque control (positive)
4 Full-load speed regulation
5 Idle
6 Intermediate speed

speed that occurs when load is removed from the diesel engine while control lever position (accelerator-pedal travel) remains constant. The resulting rise in engine speed should not exceed a certain level in the controlled range. The maximum rpm specified for an unloaded engine represents the high-idle speed. This figure is encountered when the load factor of a diesel engine operating at wide-open throttle (WOT) decreases from 100% to 0%. The rise in rotational speed is proportional to the variation in load factor. Larger load shifts produce progressively larger increases in rotational speed.

$$\delta = \frac{n_{no} - n_{vo}}{n_{vo}}$$

or in %:

$$\delta = \frac{n_{no} - n_{vo}}{n_{vo}} \cdot 100\,\%$$

where: δ droop-speed control, n_{no} upper no-load speed, n_{vo} upper full-load speed (some sources refer to the upper no-load speed as the high-idle speed).

The desired droop-speed control is defined by the diesel engine's intended application environment. Thus low degrees of proportionality (on the order of 4%) are preferred for electric power generators, as they respond to fluctuations in load factor by holding changes and engine speed and the resultant frequency shifts to minimal levels. Larger degrees of proportionality are better in motor vehicles because they furnish more consistent control for improved driveability under exposure to minor variations in load (vehicle acceleration and deceleration). In vehicular applications, limited droop-speed control would lead to excessively abrupt response when load factors change.

Design structure

The governor assembly (Fig. 2) consists of the mechanical governor (2) and the lever assembly (3). Operating with extreme precision, it controls the position of the control collar (4) to define the effective stroke and with it the injected-fuel quantity. Different versions of the control lever assembly can be employed to vary response patterns for various applications.

Axial-piston distributor pump with governor assembly (section)

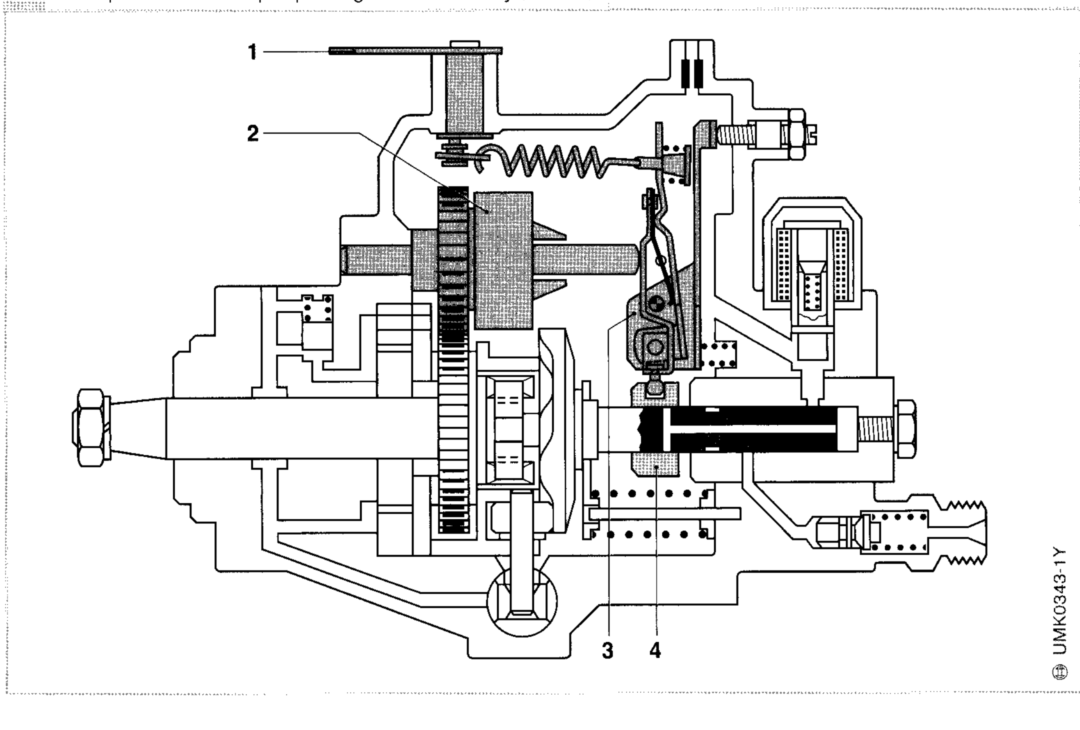


Fig. 2

1 Rotational-speed control lever (accelerator pedal)
2 Mechanical governor
3 Lever assembly
4 Control collar

Variable-speed governor

The variable-speed governor regulates all engine speeds from start or high idle. In addition to these two extremes, it also controls operation in the intermediate range. The speed selected at the rotational-speed control lever (accelerator pedal or supplementary lever) is held relatively constant, with actual consistency varying according to the droop-speed control (Fig. 4).

This sort of adjustment is required when ancillary equipment (winch, extinguisher water pump, crane, etc.) is used on commercial vehicles or in stand-alone operation. This function is also frequently used in commercial vehicles and with agricultural machinery (tractors, combine harvesters).

Design

The governor assembly, consisting of flyweight housing and flyweights, is powered by the input shaft (Fig. 2). A governor shaft mounted to allow rotation in the pump housing supports the assembly (Fig. 3, Pos. 12). The radial travel of the flyweights (11) is translated into axial movement at the sliding sleeve (10). The force and travel of the sliding sleeve modify the position of the governor mechanism. This mechanism consists of the control lever (3), tensioning lever (2) and starting lever (4).

The control lever's mount allows it to rotate in the pump housing, where its position can be adjusted using the WOT adjusting screw (1). The starting and tensioning levers are also mounted in bearings allowing them to rotate in the control lever. At the lower end of the starting lever is a ball pin that engages with the control collar (7), while a starting spring (6) is attached to its top. Attached to a retaining stud (18) on the upper side of the tensioning lever is the idle-speed spring (19). The governor spring (17) is also mounted in the retaining stud. A lever (13) combines with the control-lever shaft (16) to form the link with the rotational-speed control lever (14).

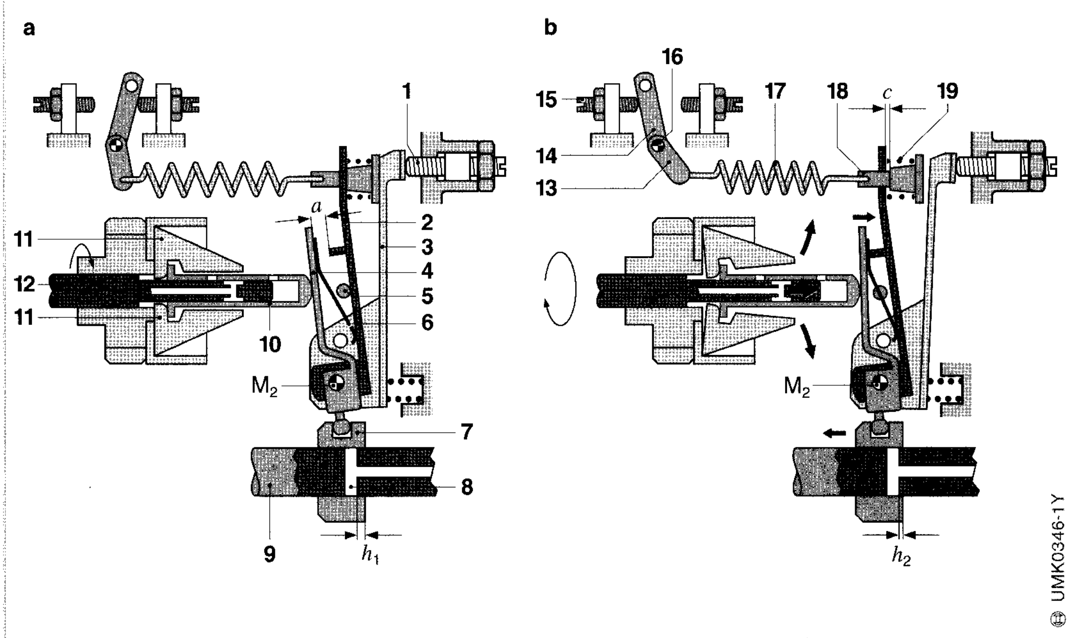
The spring tension operates together with the sliding sleeve's force to define the position of the governor mechanism. The adjustment travel is transferred to the control collar to determine the delivery quantity h_1 and h_2, etc.).

Fig. 3
- a Start position (rotational-speed control lever can be at WOT or idle position for starting)
- b Idle position

1. Full-load screw
2. Tensioning lever
3. Control lever
4. Starting lever
5. Stop pin in housing
6. Starting spring
7. Sliding sleeve
8. Distributor plunger spill port
9. Distributor plunger
10. Sliding sleeve
11. Flyweight
12. Controller base
13. Lever
14. Rotational-speed control lever
15. Idle-speed adjusting screw
16. Control-lever shaft
17. Governor spring
18. Retaining pin
19. Idle-speed spring

- a Starting-spring travel
- c Idle-speed spring travel
- h_1 Max. effective stroke (start)
- h_2 Min. effective stroke (idle)
- M_2 Pivot point for 4 and 5

Variable-speed governor during starting and idle



Starting response

When the distributor injection pump is stationary, the flyweights and the sliding sleeve are at their base positions (Fig. 3a). The starting spring pushes the starting lever into the position for starting by rotating it about its axis M_2. The starting lever simultaneously transfers force through the ball pin onto the distributor plunger to shift the control collar to its starting position. This causes the distributor plunger (9) to execute substantial effective stroke. This provides maximum fuel start quantity.

Low rotational speeds are enough to overcome the compliant starting spring's tension and shift the sliding sleeve by the distance *a*. The starting lever again rotates about its pivot axis M_2, and the initial enhanced start quantity is automatically reduced to idle delivery quantity.

Idle-speed control

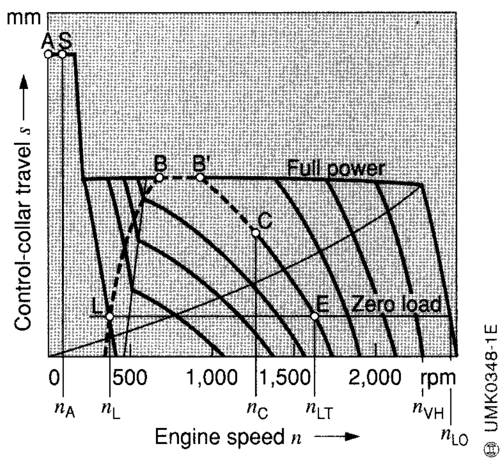
After the diesel engine starts and the accelerator pedal is released, the control lever returns to its idle position (Fig. 3b) as defined by the full-load stop on the idle adjusting screw (15). The selected idle speed ensures that the engine consistently continues to turn over smoothly when unloaded or under minimal loads. The control mechanism regulates the idle-speed spring mounted on the retaining stud. It maintains a state of equilibrium with the force generated by the flyweights.

By defining the position of the control collar relative to the control passage in the plunger, this equilibrium of forces determines the effective stroke. At rotational speeds above idle, it traverses the spring travel *c* to compress the idle-speed spring. At this point, the effective spring travel *c* is zero.

The housing-mounted idle-speed spring allows idle adjustments independent of accelerator pedal position while accommodating increases to compensate for temperature and load factor shifts (Fig. 5).

Characteristic curves of variable-speed governor

- **A:** Control-collar start position.
- **S:** Engine start quantity.
- **S-L:** Reduction of start quantity to idle quantity.
- **L:** Idle speed n_{LN} after engine start (no load).
- **L-B:** Engine acceleration phase after rotational-speed control lever is moved from idle to set speed n_C.
- **B-B':** The control collar remains briefly at full-load position to initiate rapid rise in engine speed.
- **B'-C:** Control-collar retraction phase (less quantity, higher engine speed), droop-speed control defines extent to which vehicle now maintains desired speed or engine speed n_C in part-load range.
- **E:** Engine speed n_{LT}, occurs when rotational-speed control lever remains stationary after load is removed from engine.



Idle-speed spring attached to governor housing (LFG)

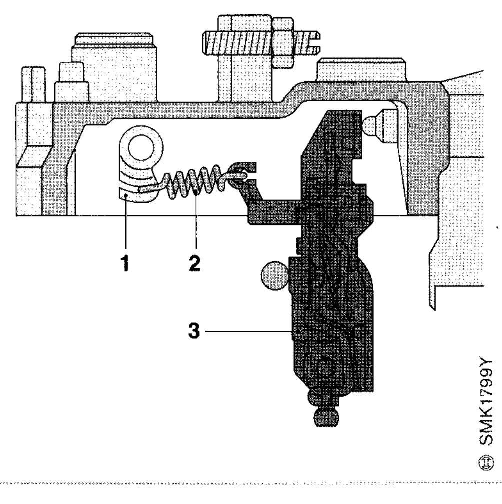


Fig. 5
1 Rocker (fixed to pump housing)
2 Idle-speed spring
3 Lever assembly

Operation under load
During normal operation, the rotational-speed control lever assumes a position in its overall travel range that corresponds to the desired rotational or speed. The driver dictates this position by depressing the accelerator pedal to the desired angle. Because the starting and idle-speed springs are both fully compressed at rotational speeds above idle, they have no influence on control in this range. Control is exercised by the governor spring.

Example (Fig.6):
The driver uses the accelerator pedal or auxiliary control lever to move the flow-control lever (2) to a position corresponding to a specific (higher) speed. This adjustment motion compresses the governor spring (4) by a given increment. At this point, the governor spring's force exceeds the centrifugal force exerted by the flyweights (1). The spring's force rotates the starting lever (7) and tensioning lever (6) around their pivot axis M_2 to shift the control collar toward its WOT position by the increment defined by the design's ratio of conversion. This increases delivery quantity to raise the engine speed. The flyweights generate additional force, which the sliding sleeve (11) then transfers to oppose the spring force.

The control collar remains in its WOT position until the opposed forces achieve equilibrium. From this point onward, any additional increase in engine speed will propel the flyweights further outwards and the force exerted by the sliding sleeve will predominate. The starting and tensioning levers turn about their shared pivot axis (M_2) to slide the control collar toward "stop" to expose the cutoff bore earlier. The mechanism ensures effective limitation of rotational speeds with its ability to reduce delivery capacity all the way to zero. Provided that the engine is not overloaded, each position of the flow-control lever corresponds to a specific rotational range between WOT and no-load. As a result, the governor maintains the selected engine speed with the degree of intervention defined by the system's droop-speed control (Fig. 4).

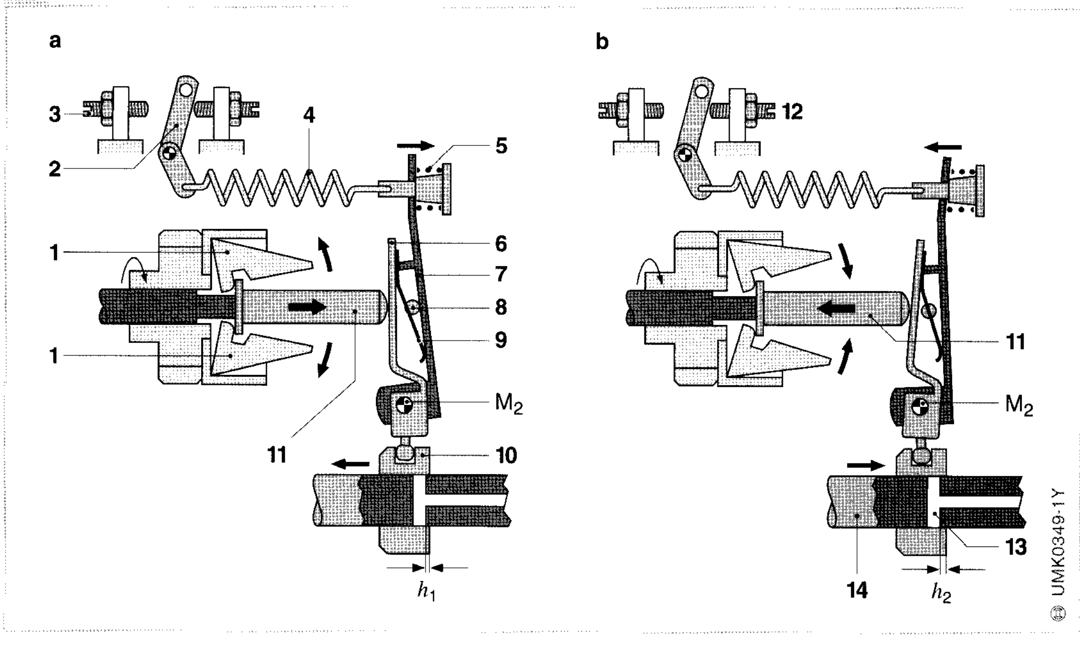
Once the imposed loads have reached such a high order of magnitude as to propel the control collar all the way to its WOT position (hill gradients, etc.), no additional increases in fuel quantity will be available,

Fig. 6
a Operation with rising speed
b Operation with falling speed

1 Flyweights
2 Rotational-speed control lever
3 Idle adjusting screw
4 Governor spring
5 Idle-speed spring
6 Tensioning lever
7 Starting lever
8 Tensioning lever full-load stop
9 Starting spring
10 Control collar
11 Sliding sleeve
12 High-idle speed adjusting screw
13 Distributor plunger spill port
14 Distributor plunger

h_1 Effective stroke at idle
h_2 Effective stroke at wide-open throttle
M_2 Pivot point for 6 and 7

Variable-speed governor under load (shown without control lever)



even in response to lower rotational speed. The engine is overloaded, and the driver should respond by downshifting to a lower gear.

Overrun
When the vehicle descends a steep gradient, or when the accelerator pedal is released at high speeds (overrun), the engine is driven by the vehicle's inertia. The sliding sleeve responds by pressing against the starting and tensioning levers. Both levers move to shift the control collar to decrease delivery quantity; this process continues until the fuel-delivery quantity reflects the requirements of the "new" load factor, or zero in extreme cases. The response pattern of the variable-speed governor described here is valid at all flow-control lever positions, and occurs whenever any factor causes load or rpm to vary so substantially as to shift the control collar all the way to its WOT or "stop" end position.

Minimum-maximum speed governors

This governor regulates the idle and high-idle engine speeds only. Response in the intermediate range is regulated exclusively by the accelerator pedal (Fig. 6). This reduces surge, but is not suitable for use in light commercial vehicles equipped with ancillaries.

Governor response map for idle/maximum-speed governor with idle-speed spring and intermediate spring

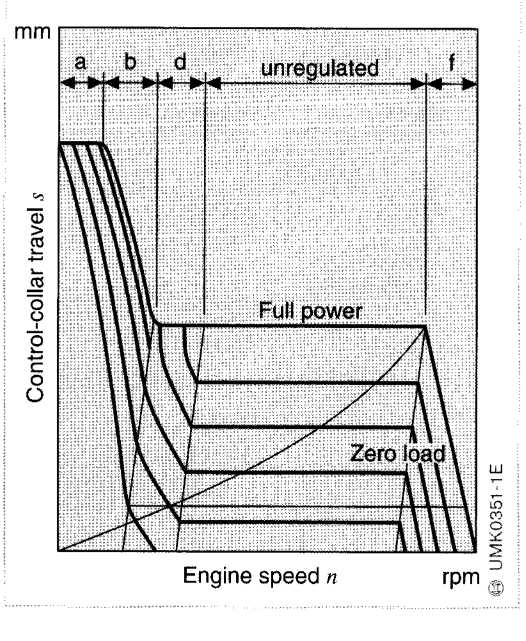


Fig. 7
- a Starting spring range
- b Starting and idle-speed spring range
- d Intermediate spring range
- f Governor spring range

Design
The governor assembly with its flyweights and assembly of control levers is essentially comparable to the variable-speed governor described above. One difference in the idle and maximum-speed governor's design is the governor spring (Figure 7 on next page, Pos. 4) and its installation. The compression spring applies pressure from its position in a guide sleeve (5). A retaining stud (7) provides the link between the tensioning lever and the governor spring.

Starting response
Because the flyweights (1) are at rest, the sliding sleeve (15) is in its base position. This allows the starting spring (12) to transfer force through the starting lever (9) and the sliding sleeve to push the flyweights to the inside. The control collar (13) on the distributor plunger is positioned to provide the delivery quantity prescribed for starting.

Idle-speed control
Once the engine starts and the accelerator pedal is released, the rotational-speed control lever (2) responds to the force exerted by the return spring against the pump housing by returning to the idle position. As engine speed climbs, the centrifugal force exerted by the flyweights (Fig. 8a, next page) rises, and their slots press the sliding sleeve back against the starting lever. The control process relies on the idle-speed spring (8) mounted on the tensioning lever (10). The starting lever's rotation moves the control collar to reduce the delivery quantity. The control collar's position is regulated by the combined effects of centrifugal and spring force.

Operation under load
When the driver varies the position of the accelerator pedal the rotational-speed control lever rotates by a given angle. This action negates the effect of the starting and idle-

speed spring and engages the intermediate spring (6), which smoothes the transition to the unregulated range on engines with idle/maximum-speed governors. Rotating the rotational-speed control lever further toward WOT allows the intermediate spring to expand until the retainer's shoulder is against the tensioning lever (Fig. 8b). The assembly moves outside the intermediate spring's effective control sector to enter the uncontrolled range. The uncontrolled range is defined by the tension on the governor spring, which can be considered as rigid in this speed range. The driver's adjustments to the rotational-speed control lever (accelerator pedal) can now be transferred directly through the control mechanism to the control collar. Delivery quantity now responds directly to movement at the accelerator pedal.

The driver must depress the pedal by an additional increment in order to raise the vehicle speed or ascend a gradient. If the object is to reduce engine output, the pressure on the accelerator pedal is reduced.

If the load on the engine decreases while the control lever position remains unchanged, the delivery quantity will remain constant, and rotational speed will increase. This leads to greater centrifugal force, and the flyweights push the sliding sleeve against the starting and tensioning levers with increased force. The full-load speed governor assumes active control only once the sleeve's force overcomes the spring's tension.

When the load is completely removed from the engine, it accelerates to its high-idle speed, where the system protects it against further increases in speed.

Part-load governor

Passenger vehicles are usually equipped with a combination of variable-speed and idle/maximum-speed governors. This type of part-load governor features supplementary governor springs allowing it to function as a variable-speed governor at the low end of the rev band while also operating as an idle/maximum-speed governor when engine speed approaches the specified maximum. This arrangement provides a stable rotational speed up to roughly 2,000 rpm, after which no further constraints are imposed on further rises if the engine is not under load.

Idle/maximum-speed governor (shown without control lever)

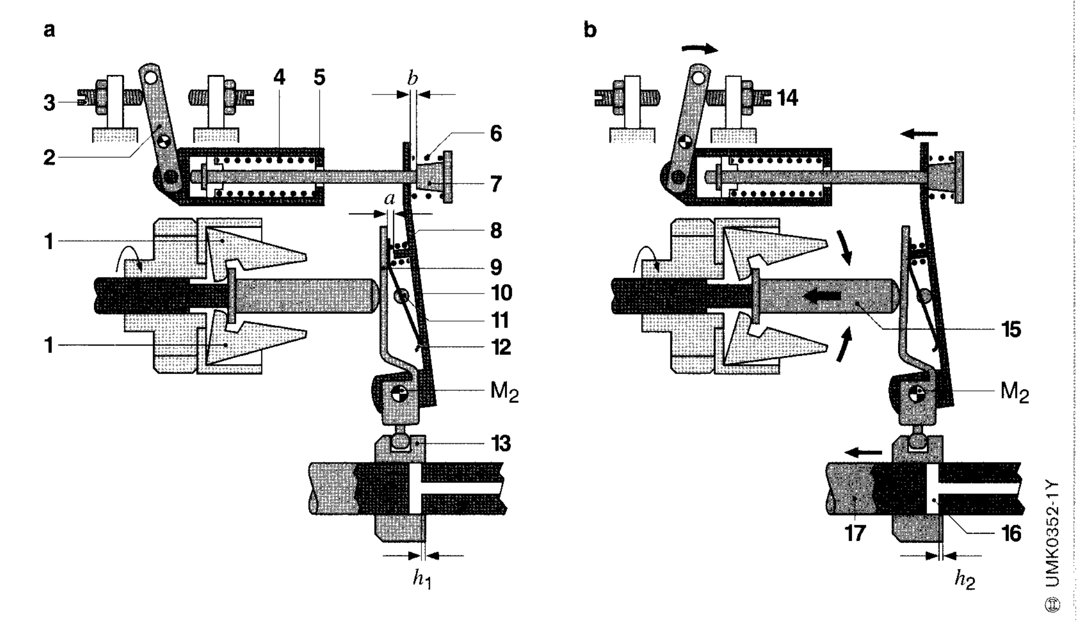


Fig. 8
- a Idle position
- b Full-load position

1 Flyweights
2 Rotational-speed control lever
3 Idle-speed adjusting screw
4 Governor spring
5 Guide
6 Intermediate spring
7 Retaining pin
8 Idle-speed spring
9 Starting lever
10 Tensioning lever
11 Tensioning lever full-load stop
12 Starting spring
13 Control collar
14 Full-load speed regulation adjusting screw
15 Sliding sleeve
16 Distributor plunger spill port
17 Distributor plunger

a Free travel of starting and idle-speed springs
b Intermediate spring travel
h_1 Effective stroke at idle
h_2 Effective stroke at wide-open throttle
M_2 Pivot point for 8 and 9

Timing device

Function

The injection event must be initiated at a specific crankshaft angle (piston position) to ensure efficient combustion and optimal power generation. The pump's start of delivery and the resultant start of delivery must thus vary as rotational speed changes in order to compensate for two primary factors:

Injection lag
Fuel delivery to the nozzles starts when the plunger in the distributor head closes the inlet passage (start of delivery). The resulting pressure wave initiates the injection event when it reaches the injector (start of delivery). The pulse propagates through the injection line at the speed of sound. The travel duration remains essentially unaffected by rotational speed. The pressure wave's propagation period is defined by the length of the injection line and the speed of sound, which is approximately 1500 m/s in diesel fuel. The period that elapses between the start of delivery and the start of injection is the injection lag.

Thus the actual start of injection always occurs with a certain time offset relative to the start of delivery. As a result of this phenomenon, the nozzle opens later (relative to engine piston position) at high than at low rotational speeds. To compensate for this, start of delivery must be advanced, and this forward shift is dependent on pump and engine speed.

Ignition lag
Following injection, the diesel fuel requires a certain period to evaporate into a gaseous state and form a combustible mixture with the air. Engine speed does not affect the time required for mixture formation. The time required between the start of injection and the start of combustion in diesel engines is the ignition lag.

Progress of a compression and ignition stroke at wide-open throttle and low speed (not to scale)

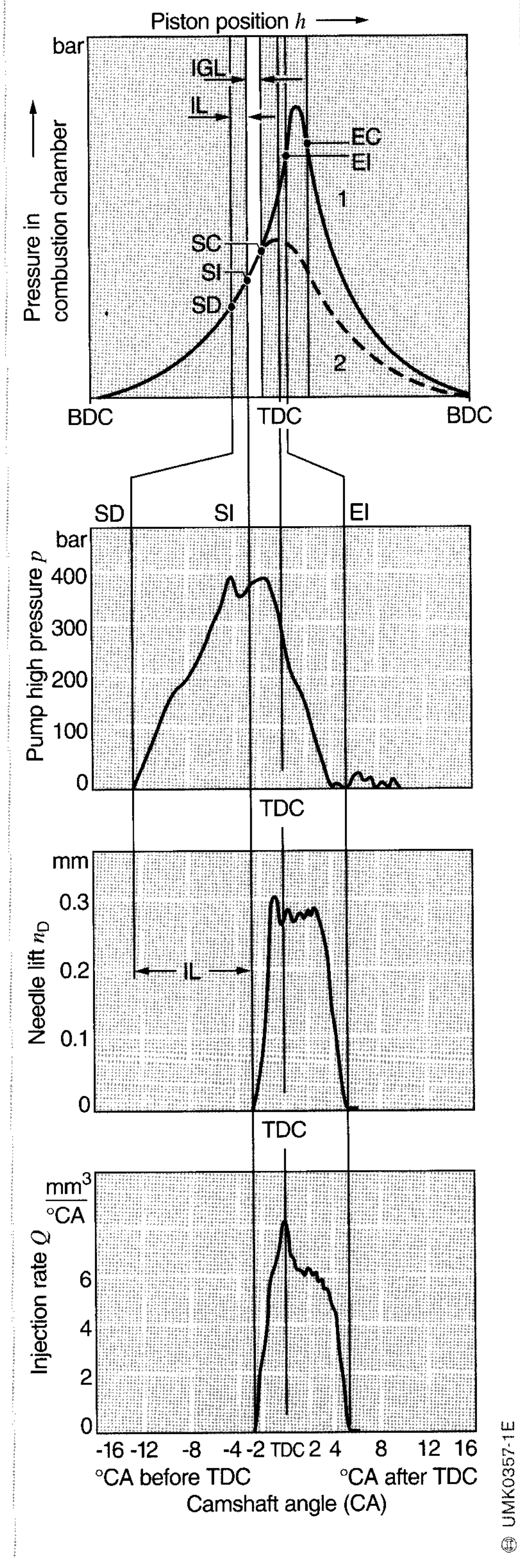


Fig. 1
1 Combustion pressure
2 Compression

SD Start of delivery
SI Start of injection
IL Injection lag
SC Start of combustion
IGL Ignition lag
EI End of injection
EC End of combustion
BDC Engine piston at Bottom Dead Center
TDC Engine piston at Top Dead Center

Factors influencing ignition lag include
- The ignitability of the diesel fuel (as defined by the cetane number)
- The compression ratio
- The air temperature, and
- The fuel discharge process

The usual ignition lag is in the order of one millisecond.

If the start of injection remains constant, the crankshaft angle traversed between the start of injection and the start of combustion will increase as rotational speed rises. This, in turn, prevents the start of combustion from occurring at the optimal moment relative to piston position.

The hydraulic timing device compensates for lag in the start of injection and ignition by advancing the distributor injection pump's start of delivery by varying numbers of degrees on the diesel engine's crankshaft. This promotes the best-possible combustion and power generation from the diesel engine at all rotational speeds.

Spill and end of combustion
Opening the cutoff bore initiates a pressure drop in the pump's high-pressure system (spill), causing the nozzle to close (end of injection). This is followed by the end of combustion. Because spill depends on the start of delivery and the position of the control collar, it is indirectly adjusted by the timing device.

Axial-piston distributor injection pump with hydraulic timing control

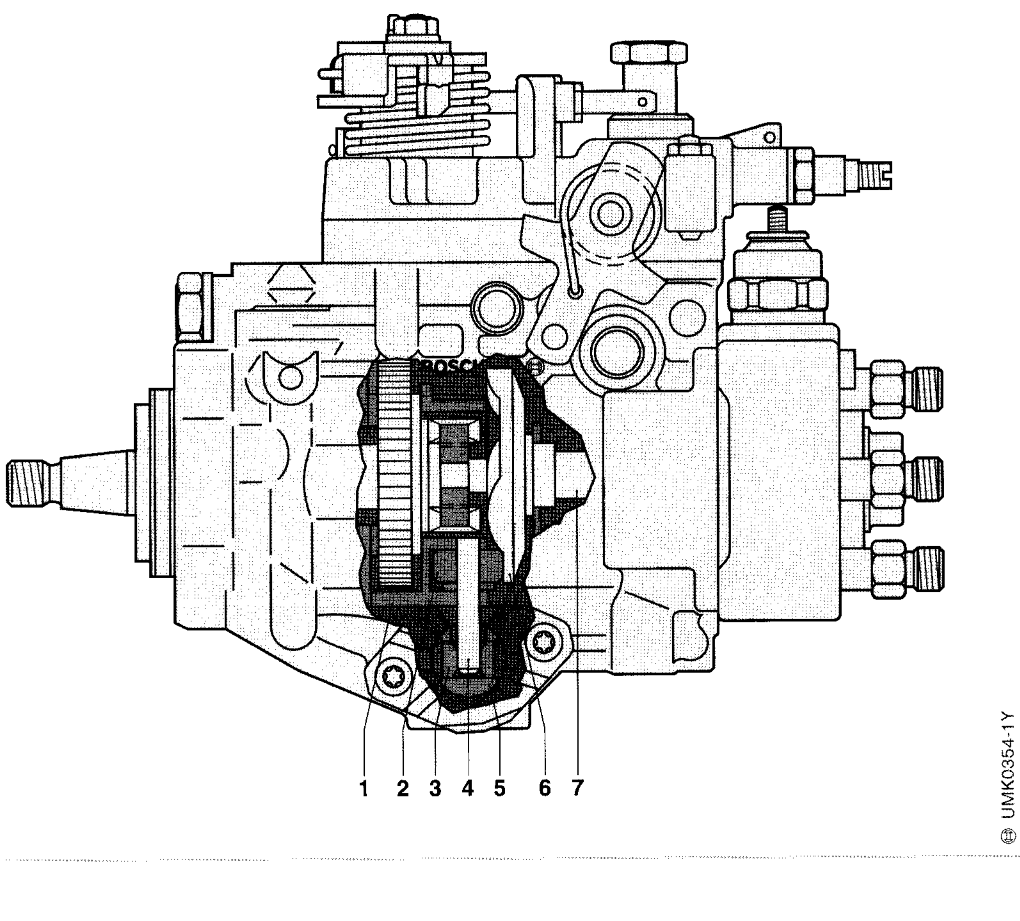


Fig. 2
1 Roller ring
2 Rollers on roller ring
3 Sliding block
4 Pin
5 Timing plunger
6 Cam plate
7 Distributor plunger

Design

The hydraulic timing device is located in the distributor injection pump's underside at a right angle to the pump's longitudinal axis (Fig. 2 and 3).

The timing plunger (Fig.3, Pos.7) is guided in the pump housing (1). Both ends of this housing are sealed by covers (6). A bore (5) in the timing plunger allows fuel inlet, and a spring (9) is installed on the opposite side.

A sliding block (8) and pin (4) connect the timing plunger to the roller ring (2).

Operating concept

The timing plunger in the distributor injection pump is held in its base position by the pre-loaded tension of the compressed timing spring (Fig. 3a). As the pump operates, the pressure-control valve regulates the pressure of the fuel in its inner chamber in proportion to rotational speed. The fuel thus acts on the side of the plunger opposite the timing spring at a pressure proportional to the pump's rotational speed.

The pump must reach a specific rotational speed, such as 300 rpm, before the fuel pressure (pressure in inner chamber) overcomes the spring's tension to compress it and shift the timing plunger (to the left in Fig, 3b). The sliding block and pin transfer the plunger's axial motion to the rotating roller ring. This modifies the relative orientation of the cam disk and roller ring, and the rollers in the roller ring lift the turning cam disk earlier. Thus the rollers and roller ring rotate relative to the cam disk and plunger by a specific angular increment (a) determined by rotational speed. (α). The maximum potential angle is usually twelve camshaft degrees (24 degrees crankshaft).

Operation of the hydraulic timing device

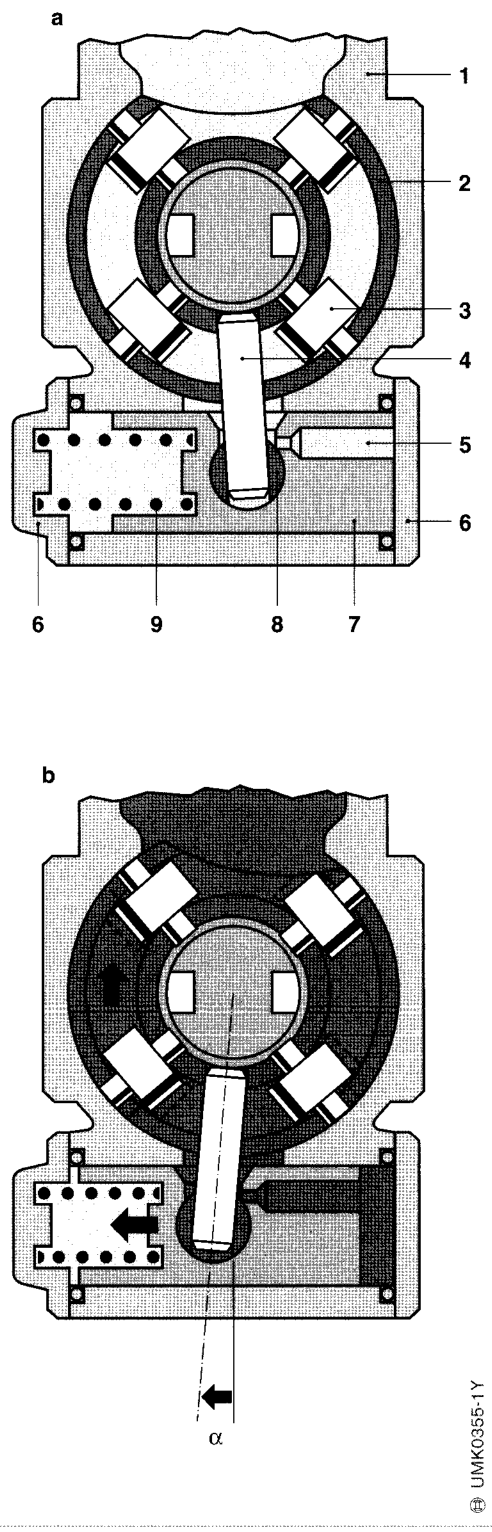


Fig. 3
a Passive state
b In operation

1 Pump housing
2 Roller ring
3 Rollers on roller ring
4 Pin
5 Bore in timing plunger
6 Cover plate
7 Timing plunger
8 Sliding block
9 Timing spring

α Roller ring pivot angle

Mechanical torque-control modules

Application

Distributor injection pumps are designed around modular building blocks, with various supplementary control devices available to satisfy individual engine requirements (as in Fig. 1). This concept affords an extended range of torque-control options allowing pumps to provide maximum torque, power and fuel economy along with low emissions. This overview serves as a compilation of the various torque-control modules and their effects on the diesel engine. The schematic diagram illustrates the interrelationships between the basic distributor injection pump and the different torque-control modules (Fig. 2).

Torque control

The torque-control process is the operation in which delivery quantity is adjusted to the engine's full-load requirement curve in response to changes in rotational speed.

This adaptive strategy may be indicated when special demands on full-load characteristics (improved exhaust-gas composition, torque generation and fuel economy) are encountered.

Axial-piston distributor injection pump (VE..F) with mechanical adjusters, potentiometer and timing-device solenoid valve (illustration)

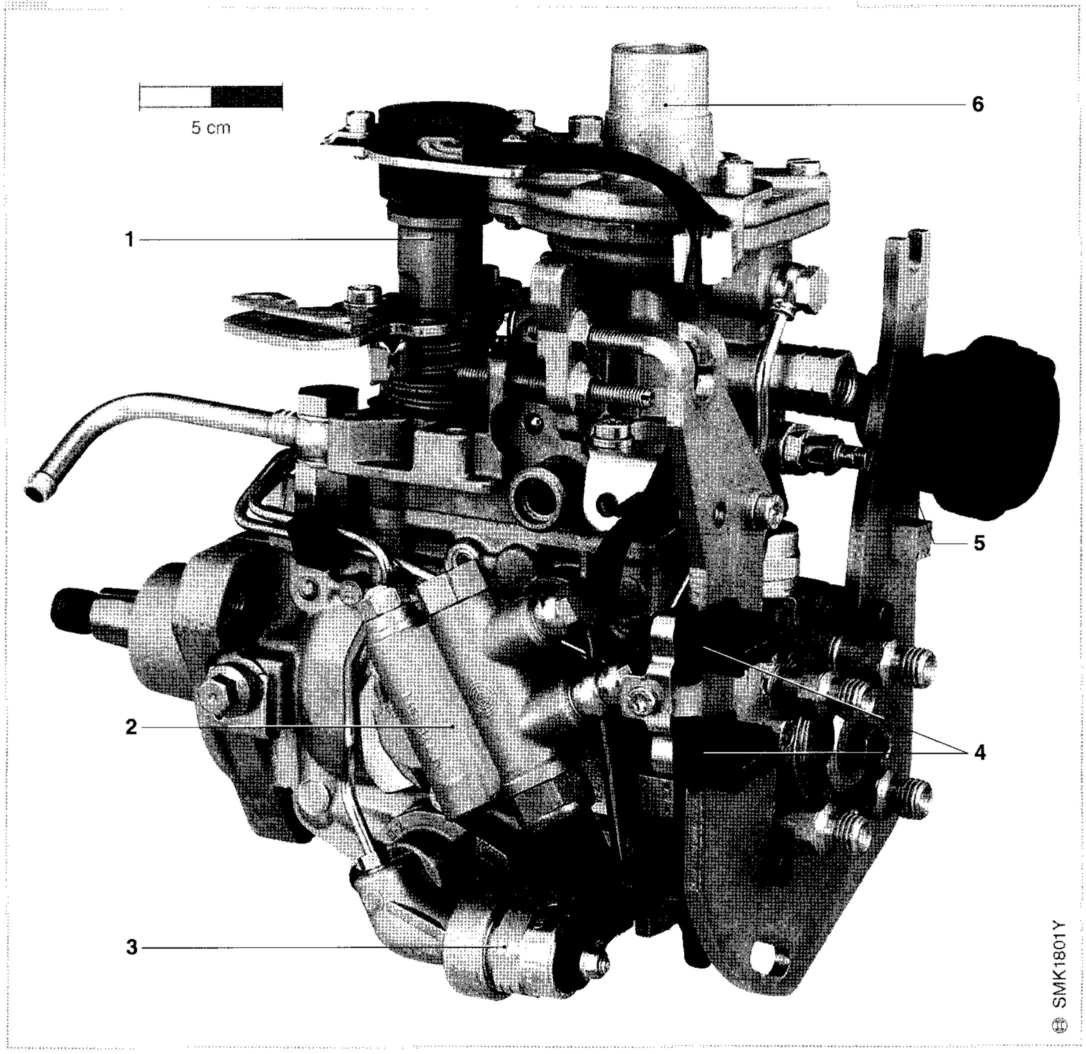


Fig. 1
1 Potentiometer
2 Hydraulic cold-start accelerator KSB
3 Load-dependent start of delivery with deactivation feature-ALFB
4 Connector
5 Pneumatic idle-speedincrease PLA
6 Hydraulically controlled torque-control HBA

Block diagram of VE distributor injection pump with mechanical/hydraulic full-load torque control

Charge-air pressure-dependent manifold-pressure compensator LDA
Delivery-quantity control relative to charge-air pressure (engine with turbocharger).

Hydraulically controlled torque control device HBA
Control of delivery quantity based on engine speed (not on turbocharged engines with LDA).

Load-dependent start of delivery LFB
Adjust start of delivery to reflect load to reduce noise emissions and, as primary aim, to reduce exhaust-gas emissions.

Atmospheric-pressure dependent full-load stop ADA
Control of delivery quantity based on atmospheric pressure.

Cold-start accelerator KSB
Adjusts start of delivery for improved performance during cold starts.

Graduated (or adjustable) start quantity GST
Avoid start enrichment during hot starts.

Temperature-controlled idle-speed increase TLA
Improved post-start warm-up and smoother operation with idle-speed increase on cold engine.

Solenoid-operated shutoff valve ELAB
Makes it possible to shut down engine with "ignition key".

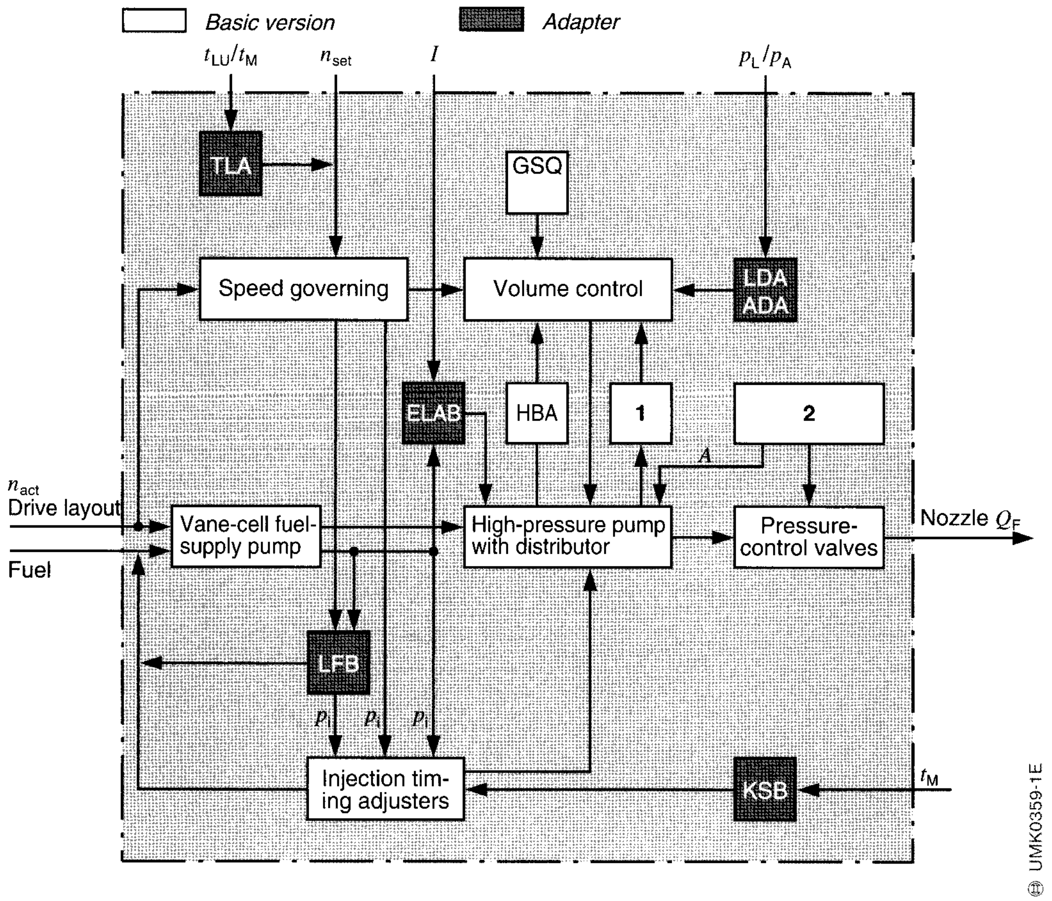


Fig. 2
1 Full-load torque control with control-lever assembly
2 Hydraulic full-load torque control

A Cutoff bore
n_{act} Actual speed (control parameter)
n_{set} Set speed (reference parameter)
Q_F Delivery quantity
t_M Engine temperature
t_{LU} Ambient air temperature
p_L Charge-air pressure
p_A Atmospheric pressure
p_i Pump inner chamber pressure
I Current at ELAB (PWM signal)

The object is to inject precisely the amount of fuel required by the engine. Following an initial rise, the engine's fuel requirement falls slightly as its speed increases. Fig. 3 shows the curve for fuel delivery on an fuel-injection pump without torque control (1).

As the curve indicates, the distributor injection pump delivers somewhat more fuel at high rotational speeds, while the control collar remains stationary relative to the plunger. This increase in the pump's delivery quantity is traceable to the venturi effect at the cutoff bore on the plunger.

Permanently defining the fuel-injection pump's delivery quantity for maximum torque generation at low rotational speeds results in an excessive rate of high-speed fuel injection, and the engine is unable to burn the fuel without producing smoke. The results of excessive fuel injection include engine overheating, particulate emissions and higher fuel consumption.

The contrasting case occurs when maximum fuel delivery is defined to reflect the engine's requirements in full-load operation at maximum rotational speed; this strategy fails to exploit the engine's low-speed power-production potential. Again, delivery quantity decreases as engine speed increases. In this case, power generation is the factor that is less than optimal. Conclusion: A means is required to adjust injected fuel quantities to reflect the engine's actual instantaneous requirements.

On distributor injection pumps, this torque-control can be executed by the delivery valve, the cutoff bore, an extended control lever assembly or hydraulically controlled torque control (HBA). The control lever assembly is employed when negative full-load torque control is required.

Positive torque control

Positive full-load torque control is required on fuel-injection pumps that would otherwise deliver too much fuel at the top of the speed range. This type of system is employed to avoid this issue by reducing the fuel-injection pump's high-speed delivery quantity.

Positive torque control with the pressure-control valve

in certain limits, pressure-control valves can provide positive torque control, for instance, when equipped with more compliant springs. This strategy limits the magnitude of the high-speed rise in the pump's internal pressure.

Positive torque control using cutoff bore

Selected shapes and dimensions for the plunger's cutoff bore can be selected to reduce delivery quantities delivered at high speed.

Delivery quantity curves with and without full-load torque control

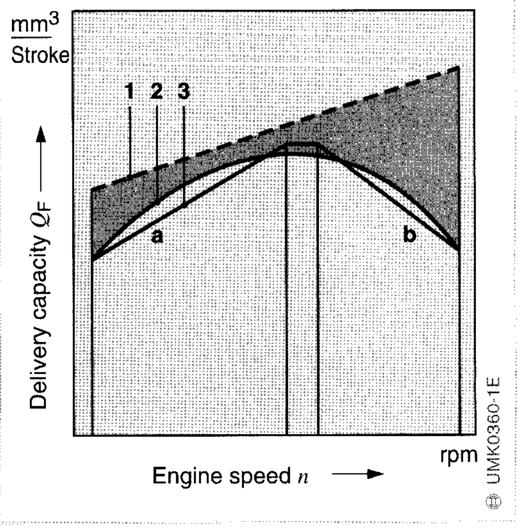


Fig. 3

- a Negative torque control
- b Positive torque control

1. Full-load delivery quantity with no torque control
2. Engine fuel requirement
3. Full-load delivery quantity with torque control

Shaded area:
Excessive fuel injected

Negative torque control

Negative full-load torque control may be indicated with engines that suffer from a low-speed tendency to generate black smoke or need special torque-rise characteristics. Turbocharged engines often require negative torque control to take over from the manifold-pressure compensator (LDA). The response to these scenarios is to increase delivery quantity as engine speed rises (Fig. 3, Sector a).

Negative torque control with control lever assembly (Fig. 4)

The starting spring (4) compresses and the torque-control lever (9) presses against the tensioning lever (2) via the stop pin (3). The torque-control shaft (11) is also pressed against the tensioning lever. When speed rises, increasing the sleeve force F_M, the torque-control lever presses against the torque-control spring. If the sleeve force exceeds the torque-control spring's force, the torque-control lever (9) is pressed toward the pin shoulder (5). This shifts the pivot axis M_4 shared by the starting lever and the torque-control lever. The starting lever simultaneously rotates about M_2 to slide the control collar (8) for increased delivery quantity. The torque-control process terminates with the torque-control lever coming to rest against the pin shoulder.

Negative torque control with hydraulically controlled torque control (HBA)

A torque control mechanism similar to the manifold-pressure compensator (LDA) can be used to define the full-load delivery characteristics as a function of engine speed (Fig. 5). As engine speed increases, rising pressure in the pump's internal chamber p_i is transferred to the control plunger (6). This system differs from spring-based adjustment by allowing use of a cam profile on the control pin to define full-throttle curves (to a limited degree).

Negative torque control with control-lever assembly

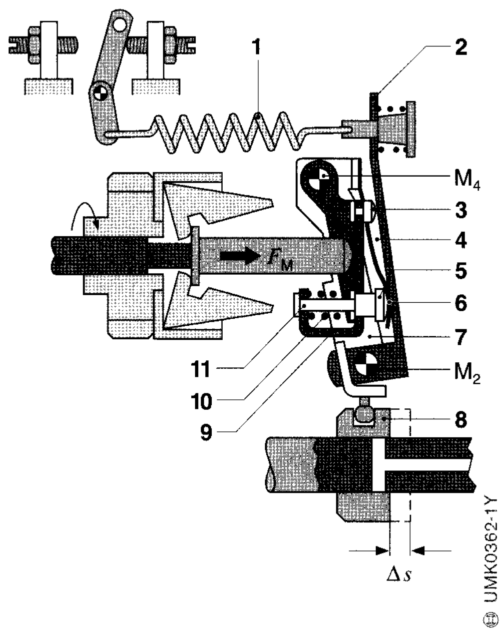


Fig. 4
1 Governor spring
2 Tensioning lever
3 Full-load stop pin
4 Starting spring
5 Pin shoulder
6 Impact point
7 Starting lever
8 Control collar
9 Torque-control lever
10 Torque-control spring
11 Torque-control shaft

M_2 Pivot point for 2 and 7
M_4 Pivot axis for 7 and 9
F_M Sleeve force
s Control-collar travel

Hydraulically controlled torque control (HBA)

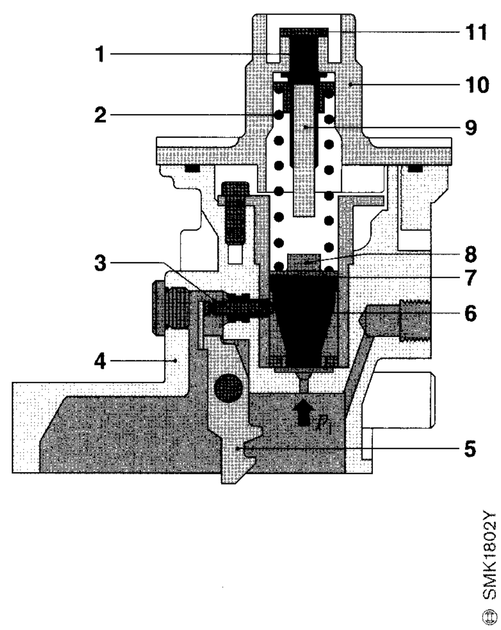


Fig. 5
1 Adjusting screw
2 Spring
3 Pin
4 Pump cover
5 Reverse-transfer lever with full-load stop
6 Adjustment piston
7 Shim
8 Base disk
9 Stop pin
10 Cover plate
11 Locknut

p_i Pump inner chamber pressure

Charge-air pressure torque control

During artificial induction a (turbo)supercharger forces pressurized fresh air into the intake tract. This charge-air pressure allows a diesel engine of any given displacement to generate more power and torque than its atmospheric-induction counterpart in any given speed band. The rise in effective power corresponds to the increase in air mass (Fig. 6). In many cases, it proves possible to reduce specific fuel consumption at the same time. A standard means of generating charge-air pressure for diesel engines is the exhaust-gas turbocharger.

Manifold-pressure compensator (LDA)

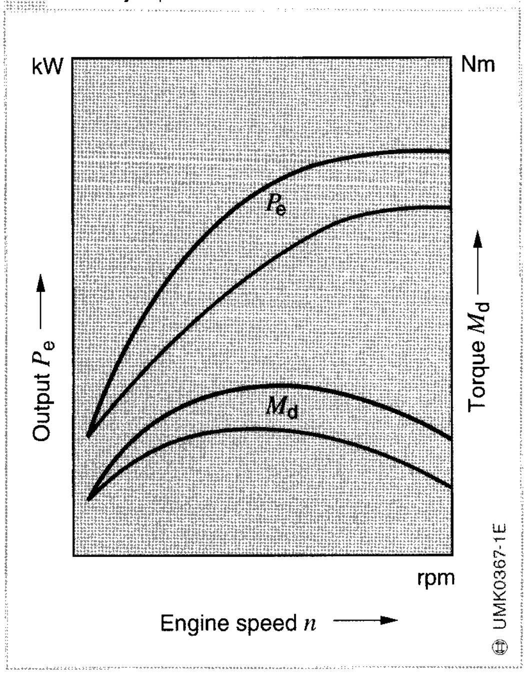
Function

Fuel delivery in the turbocharged diesel engine is adapted to suit the increased density of the air charges produced in charging mode. When the turbocharged diesel operates with cylinder charges of relatively low density (low induction pressure), fuel delivery must be adjusted to reflect the lower air mass. The manifold-pressure compensator performs this function by reducing fuel-throttle fuel flow below a specific (selected) charge-air pressure (Fig. 7).

Design

The manifold-pressure compensator is mounted on top of the distributor injection pump (Fig. 8 and 9). On top of this mechanism are the charge-air-pressure connection (7) and the vent port (10). A diaphragm (8) separates the inside chamber into two airtight and mutually isolated sections. A compression spring (9) acts against the diaphragm, while the adjusting screw (5) holds the other side. The adjusting screw is used to adjust the spring's tension. This process adapts the manifold-pressure compensator to the charge-air pressure generated by the turbocharger. The diaphragm is connected to the sliding bolt (11). The sliding bolt features a control cone (12) whose position is monitored by a probe (4). The probe transfers the sliding bolt's motion through the reverse-transfer lever (3) to vary the full-load stop. The adjusting screw (6) on top of the LDA defines the initial positions of the diaphragm and sliding bolt.

Power and torque comparison: charge-air vs. naturally aspirated



Charge-air pressure range

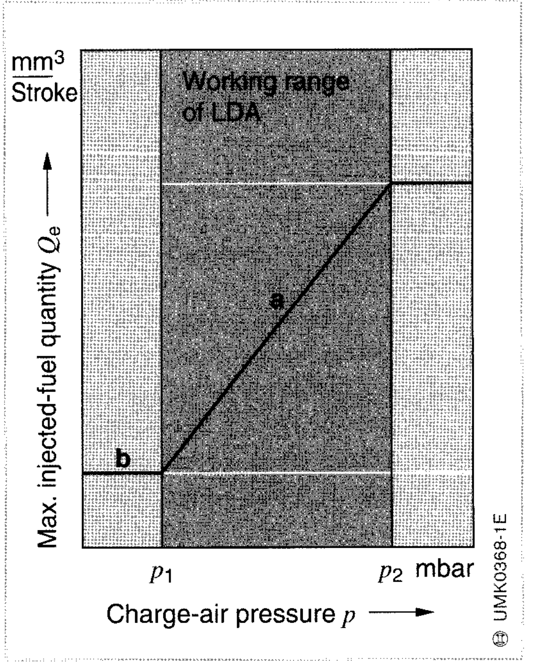


Fig. 6
— Naturally aspirated engine
— Turbocharged engine

Fig. 7
a with turbocharger
b naturally aspirated

p_1 Lower charge-air pressure
p_2 Upper charge-air pressure

Operating concept
At the lower end of the speed range, the charge-air pressure generated by the turbocharger is not powerful enough to compress the spring. The diaphragm remains in its initial position. Once the rising charge-air pressure p_L starts to deflect it, the diaphragm pushes the sliding bolt and control cone down against the pressure of the spring.

This vertical movement in the sliding bolt shifts the position of the probe, causing the reverse-transfer lever to rotate about its pivotal point M_1. The tensile force exerted by the governor spring creates a positive connection between tensioning lever, reverse-transfer lever, probe and control cone.

The tensioning lever thus mimics the reverse-transfer lever's rotation, and the starting and tensioning levers execute a turning motion around their shared pivot axis to shift the control lever and raise delivery quantity. This process adapts fuel flow to meet the demands of the greater air mass with the engine's combustion chambers.

Axial-piston distributor injection pump with manifold-pressure compensator

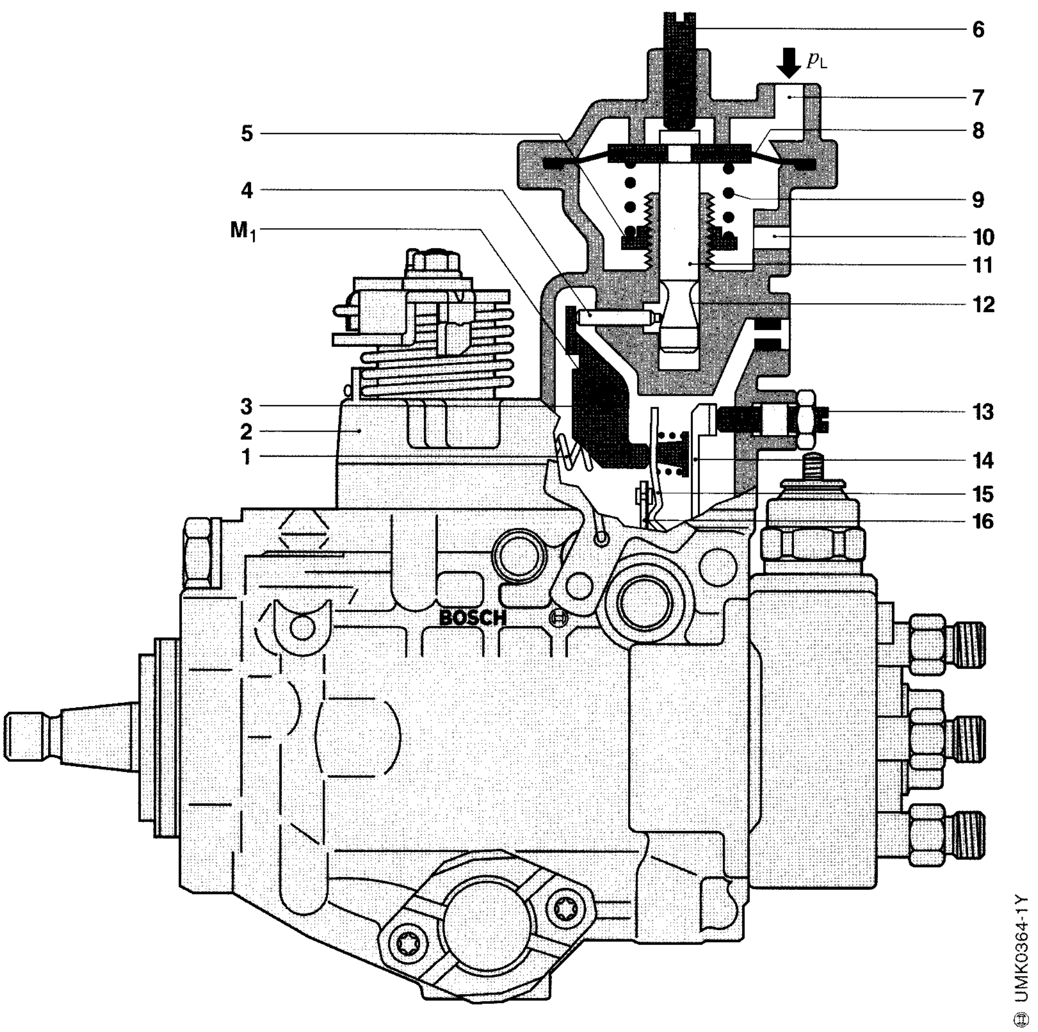


Fig. 8
1 Governor spring
2 Governor cover
3 Reverse-transfer lever
4 Sensor pin
5 Adjustment nut
6 Adjustment pin
7 Charge-air pressure connection
8 Diaphragm
9 Spring
10 Vent
11 Sliding bolt
12 Control cone
13 Adjusting screw for full-load delivery
14 Control lever
15 Tensioning lever
16 Starting lever

p_L Charge-air pressure
M_1 Pivot axis for 3

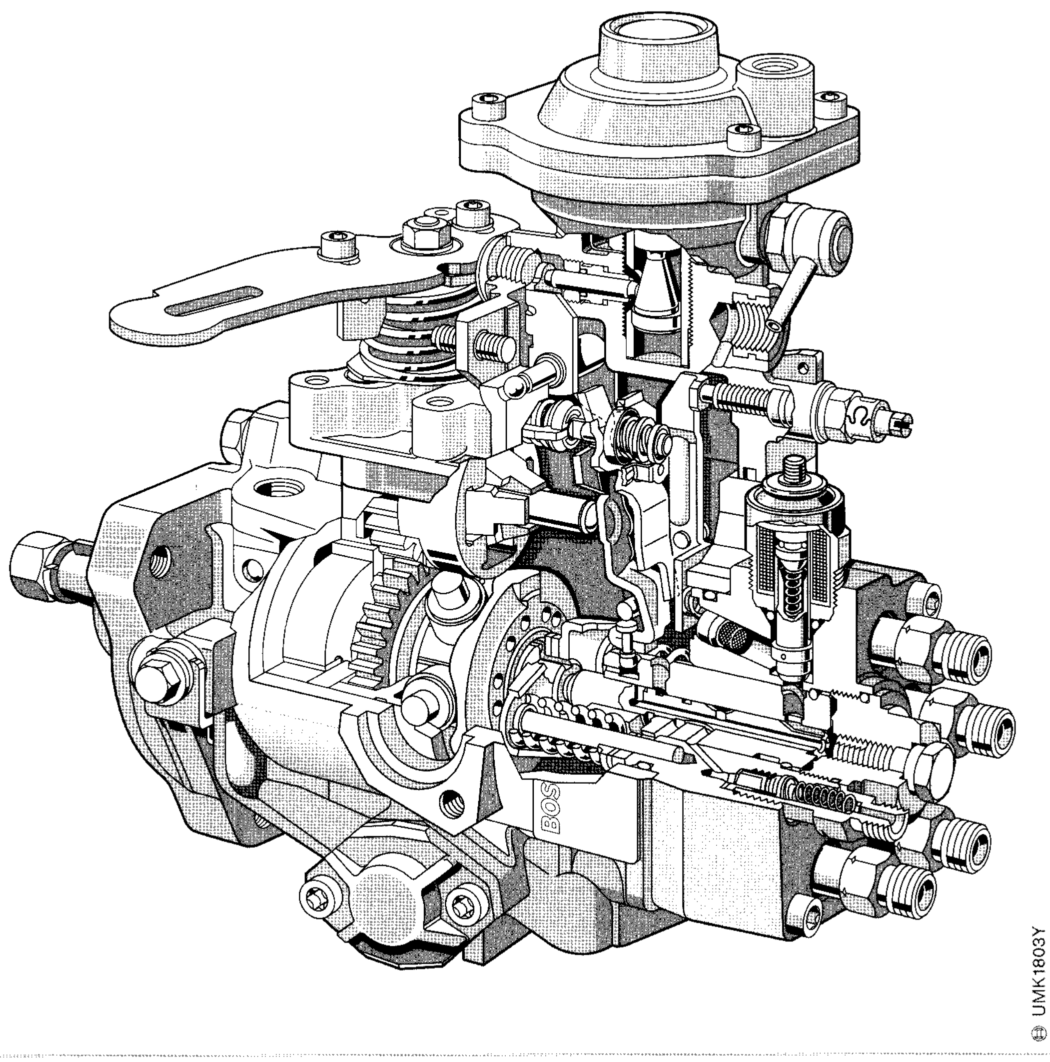
As charge-air pressure falls, the spring beneath the diaphragm presses the sliding bolt back up. The control mechanism now acts in the opposite direction, and fuel quantity is reduced to reflect the needs of the lower charge-air pressure.

The manifold-pressure compensator responds to turbocharger failure by reverting to its initial position, and limits full-throttle fuel delivery to ensure smoke-free combustion. Full-load fuel quantity with charge-air pressure is adjusted by the full-throttle stop screw in the governor cover.

Atmospheric pressure-sensitive torque control

Owing to the lower air density, the mass of inducted air decreases at high altitudes. If the standard fuel quantity prescribed for full-load operation is injected, there will not be enough air to support full combustion. The immediate results are smoke generation and rising engine temperatures. The atmospheric pressure-sensitive full-load stop can help prevent this condition. It varies full-load fuel delivery in response to changes in barometric pressure.

3 Axial-piston distributor injection pump with manifold-pressure compensator (section)



Atmospheric-pressure sensitive full-load stop (ADA)

Design

The basic structure of the atmospheric pressure-sensitive full-load stop is identical to that of the charge-air pressure-sensitive full-load stop. In this application, it is supplemented by a vacuum unit connected to a vacuum-operated pneumatic device (such as the power brake system). The vacuum unit provides a constant reference pressure of 700 mbar (absolute).

Operating concept

Atmospheric pressure acts on the upper side of the ADA's internal diaphragm. On the other side is the constant reference pressure supplied by the vacuum unit.

Reductions in atmospheric pressure (as encountered in high-altitude operation) cause the adjustment piston to rise away from the lower full-load stop. As with the LDA, a reverse-transfer lever then reduces the injected fuel quantity.

Load-sensitive torque control

Load-sensitive start of delivery (LFB)

Function

As the diesel engine's load factor changes, the start of injection – and thus the start of delivery – must be advanced or retarded accordingly.

The load-sensitive start of delivery is designed to react to declining loads (from full-load to part throttle, etc.) at constant control lever positions by retarding start of delivery. It responds to rising load factors by shifting the start of delivery forward. This adaptive process provides smoother engine operation along with cleaner emissions at part throttle and idle.

Fuel-injection pumps with load-sensitive start of delivery can be recognized by the press-fit ball plug inserted during manufacture (Fig. 10, Pos. 10).

10 Governor assembly with load-dependent start of delivery (design)

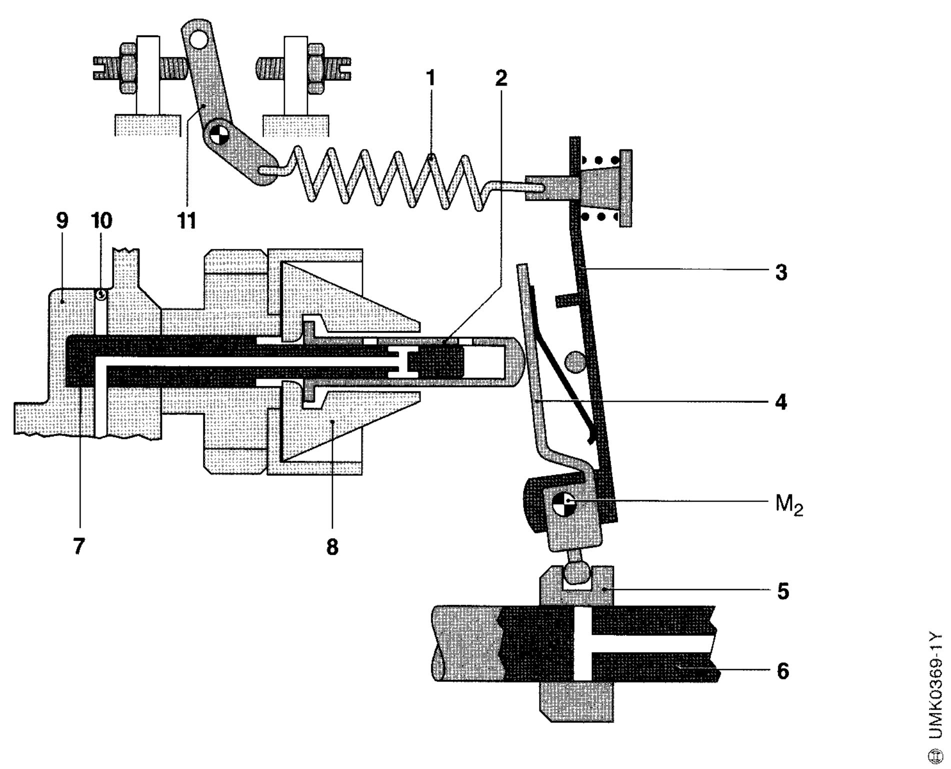


Fig. 10
1 Governor spring
2 Sliding sleeve
3 Tensioning lever
4 Starting lever
5 Control collar
6 Distributor plunger
7 Controller base
8 Flyweight
9 Pump housing
10 Cone
11 Rotational-speed control lever

M_2 Pivot for 3 and 4

Design
Load-sensitive start of delivery torque control (Fig. 10) relies on modifications to the sliding sleeve (2), governor base (7) and pump housing (9). In this configuration, the sliding sleeve is equipped with a supplementary control port while the base includes a ring groove, one longitudinal and two transverse passages. The pump housing contains an additional bore allowing this assembly to link the pump's inner chamber with the suction-side of the vane-type supply pump.

Operating concept
Rises in engine speed and the accompanying increases in the pump's internal pressure advance the start of delivery. The reduction in the pump's inner chamber supplied by the LFB provide a (relative) retardation of the timing. The control function is regulated by the ring groove on the base and the control port on the sliding sleeve. Individual full-load speeds can be specified by the rotational-speed control lever (11).

If the engine attains this speed without reaching wide-open throttle, the rotational speed can continue to rise. The flyweights (8) respond by spinning outward to shift the position of the sliding sleeve. Under normal control conditions, this reduces delivery quantity, while the sliding sleeve's control port is exposed by the ring groove in the base (open, Fig. 11). At this point, a portion of the fuel flows through the base's longitudinal and transverse passages to the suction side, reducing the pressure in the pump's inner chamber.

This pressure reduction moves the timing plunger to a new position. This forces the roller ring to rotate in the pump's rotational direction, retarding the start of delivery.

The engine speed will drop if load factor continues to rise while the control lever's position remains constant. This retracts the flyweights to shift the sliding sleeve and reclose its control port. The flow of fuel to the pump's inside chamber is blocked, and pressure in the chamber starts to rise. The timing plunger acts against the spring pressure to turn the roller ring against the pump's direction of rotation and the start of delivery is advanced.

Load-dependent start of delivery with deactivation feature (LFB)
The LFB can be deactivated to reduce HC emissions generated by the diesel engine when it is cold (< 60° C). This process employs a solenoid valve (8) to block the fuel flow. This solenoid open when de-energized.

Load-dependent start of delivery (operating concept)

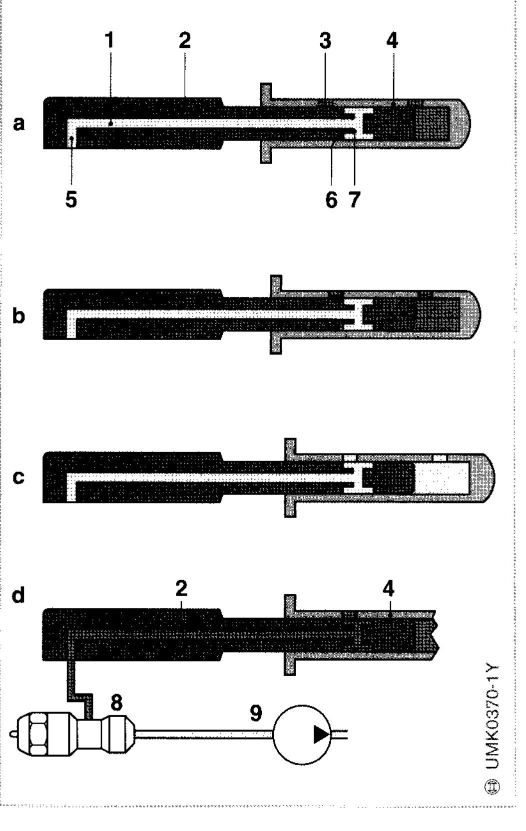


Fig. 11
- a Start (base) position
- b Full-load position just before activation
- c Activation (pressure reduction in inner chamber)
- d Load-dependent start of delivery with deactivation feature ALFB

1. Longitudinal passage in plunger
2. Controller axis
3. Sliding sleeve control port
4. Sliding sleeve
5. Transverse passage in controller axis
6. Control helix on ring groove on controller axis
7. Transverse passage in controller axis
8. Solenoid valve
9. Vane-type supply pump

Cold-start compensation

The cold-start compensation device improves the diesel engine's cold-start response by advancing the start of delivery. This feature is controlled by a driver-operated cable or by an automatic temperature-sensitive control device (Fig. 13).

Mechanical cold-start accelerator (KSB) on roller ring

Design

The KSB is mounted on the pump's housing. In this assembly, a shaft (Fig. 12, Pos. 12) connects the stop lever (Fig. 13, Pos. 3) with an inner lever featuring an eccentrically mounted ball head (3). This lever engages with the roller ring. The stop lever's initial position is defined by the full-load stop and the leg spring (13). The control cable attached to the upper end of the stop lever serves as the link to the adjustment mechanism which is either manual or automatic. The automatic adjuster is installed in a bracket on the distributor injection pump (Fig. 13), while the manual adjustment cable terminates in the passenger compartment.

There also exists a version in which the adjuster intervenes through the timing plunger.

Operating concept

The only difference between the manual and automatic versions of the cold-start accelerator is the external control mechanisms. The key process is always the same. When the control cable is not tensioned, the leg spring presses the stop lever against its full-load stop. Both ball head and roller ring (6) remain in their initial positions. Tension on the control cable causes the stop lever, shaft and inside lever to turn with the ball head.

This rotation changes the position of the roller ring to advance the start of delivery. The ball head engages a vertical groove in the roller ring. This prevents the timing plunger from advancing the start of delivery further before a specified engine speed is reached.

When the driver activates the cold-start acceleration device (KSB timing device), a position shift of approximately 2.5° camshaft (b) remains, regardless of the adjustment called for by the timing device (Fig. 14a).

Mechanical cold-start accelerator (section)

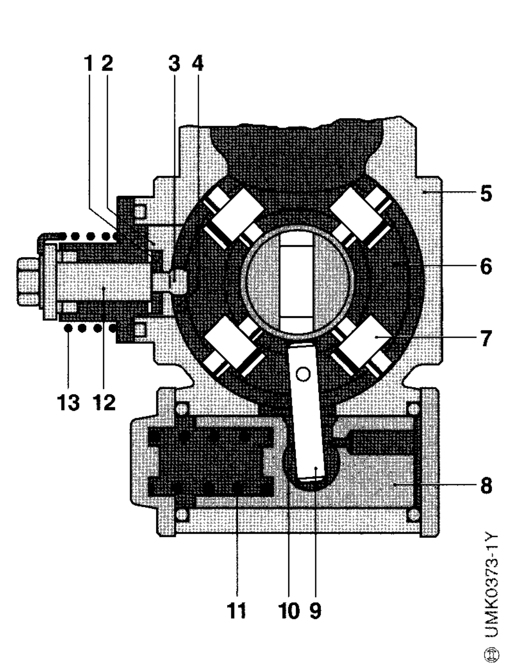


Fig. 12
1 Lever
2 Adjustment window
3 Ball head
4 Vertical groove
5 Pump housing
6 Roller ring
7 Rollers in roller ring
8 Timing plunger
9 Pin
10 Sliding block
11 Timing spring
12 Shaft
13 Leg spring

Mechanical cold-start accelerator, adjuster with automatic control (cold position)

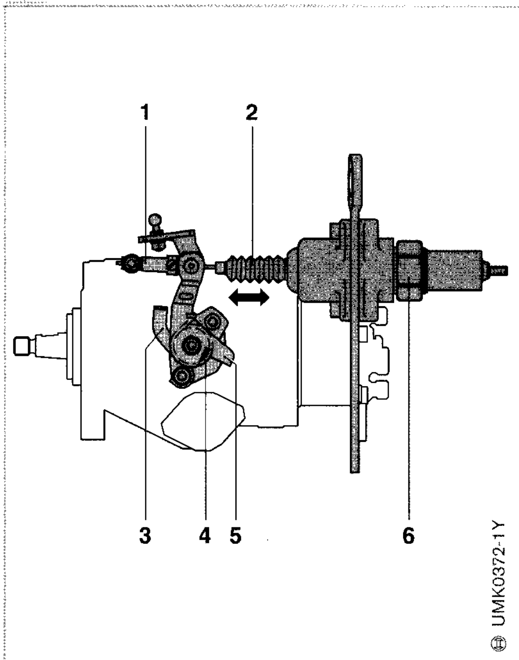


Fig. 13
1 Retainer
2 Bowden cable
3 Stop lever
4 Leg spring
5 Control lever KSB
6 Adjuster dependent on coolant and ambient temperatures

In cold-start systems with automatic control, the actual increment depends on engine temperature and/or ambient temperature.

Automatic adjustment relies on a control mechanism in which a temperature-sensitive expansion element translates variations in engine temperature into linear motion.

This arrangement's special asset is that it provides the optimum start of delivery and start of injection for each individual temperature.

Various lever layouts and actuation mechanisms are available according to the mounting side and rotational direction encountered in individual installation environments.

Temperature-controlled idle-speed increase (TLA)

The temperature-controlled idle-speed increase, which is combined with the automatic KSB, is also operated by the control mechanism (Fig. 15). The ball pin in the extended KSB control lever presses against the rotational-speed control lever to lift it from the idle-speed stop screw when the engine is cold. This raises the idle speed to promote smoother engine operation. The KSB control lever rests against its full-load stop when the engine is warm. This allows the rotational-speed control lever to return to its own full-load stop, at which point the temperature-controlled idle-speed increase system is no longer active.

Hydraulic cold-start accelerator

There are inherent limits on the use of strategies that shift the timing plunger to advance start of injection. Hydraulic start of injection advance applies speed-controlled pressure in the pump's inside chamber to the timing plunger. The system employs a bypass valve in the pressure-control valve to modify the inner chamber's automatic pressure control, automatically increasing internal pressure to obtain additional advance extending beyond the standard advance curve.

Design

The hydraulic cold-start accelerator comprises a modified pressure-control valve (Fig. 17, Pos. 1), a KSB ball valve (7), an electrically heated expansion element (6) and a KSB control valve (9).

Effects of mechanical cold-start accelerator KSB

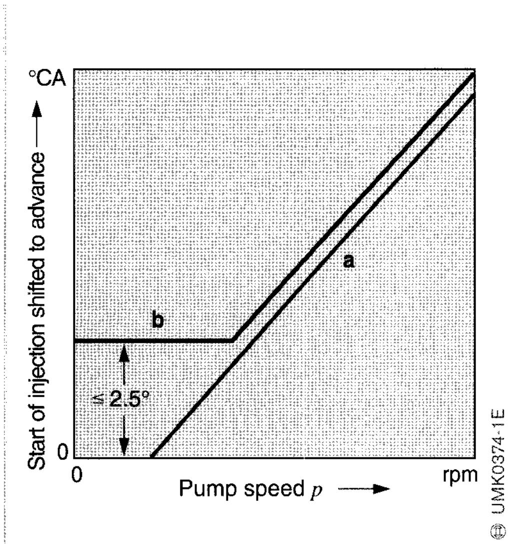


Fig. 14
- a Injection adjusted by timing device
- b Minimum adjustment (approximately 2.5° camshaft)

Mechanical cold-start accelerator (automatic) with temperature-controlled idle-speed increase

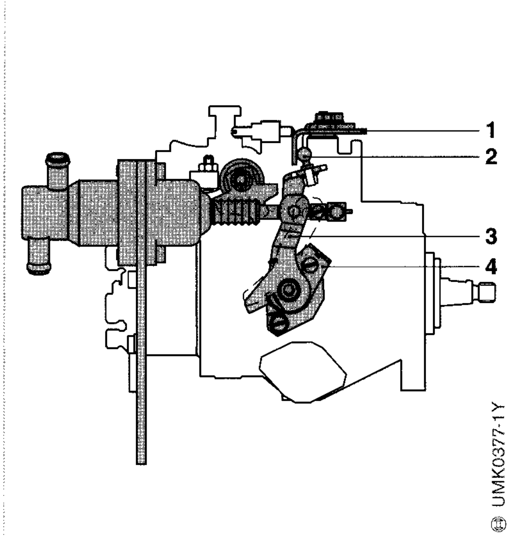


Fig. 15
1. Rotational-speed control lever
2. Ball pin
3. Control lever KSB
4. Full-load stop

Operating concept
The fuel supplied by the supply pump (5) flows through the distributor injection pump's inner chamber and to the end of the timing plunger (11), which compresses the return spring (12) and shifts position in response to the inner chamber pressure to vary the start of injection. The pressure-control valve controls the pressure in the inner chamber, raising pressure for greater delivery quantity as engine speed increases (Fig. 16).

The throttle port in the pressure-control valve's plunger (Fig. 17, Pos. 3) supplies the added pressure that allows the KSB to provide the forward offset in start of injection advance (Fig. 16, blue curve). This conveys an equal pressure to the spring side of the pressure-control valve. The KSB ball valve, with its higher pressure setting, regulates activation and deactivation (with the thermal element) while also serving as a safety release. An adjusting screw on the integrated KSB control valve is available for adjusting KSB operation to a specific engine speed. Pressure from the supply pump presses the KSB control-valve plunger (10) against a spring. A damping throttle inhibits pulsation against the control plunger. The KSB pressure curve is controlled by the timing edge on the control plunger and the opening on the valve holder. The spring rate on the control valve and the control port configuration can be modified to match KSB functionality to the individual application. The ambient temperature will act on the expansion element to open the cold-start accelerator ball valve before starting the hot engine.

Hydraulic cold-start accelerator (operation)

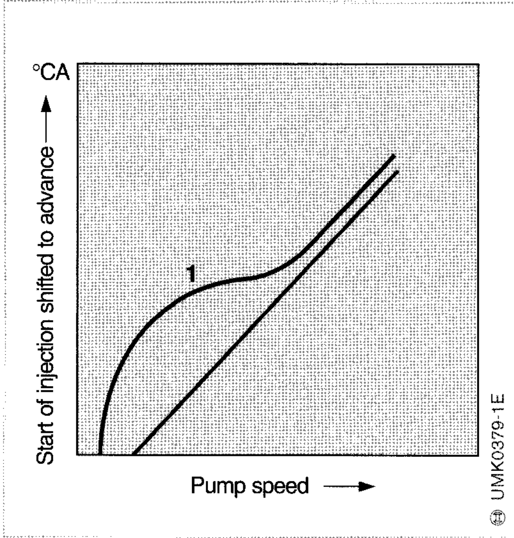


Fig. 16
1 Start of delivery advanced

Hydraulic cold-start accelerator

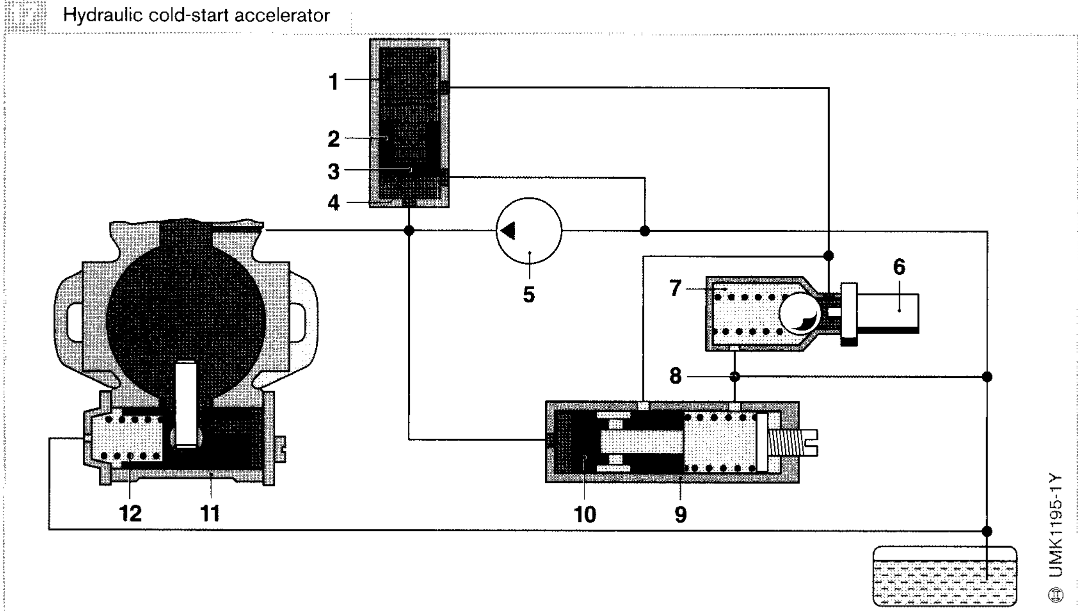


Fig. 17
1 Pressure-control valve
2 Valve plunger
3 Throttled bore
4 Inner chamber pressure
5 Vane-type supply pump
6 Electrically heated expansion element
7 Ball valve KSB
8 Fuel drains without pressure
9 Adjustable KSB control valve
10 Control plunger
11 Timing device
12 Return spring

Soft-running device

To achieve the desired emission properties, the system injects the fuel charge into the engine's combustion chamber in the briefest possible time span; it operates at high fuel-delivery rates.

Depending on system configuration, these high fuel-delivery rates can have major consequences, especially in the form of diesel knock at idle. Remedial action is available through the use of extended injection periods to achieve smoother combustion processes at and near idle.

Design and operating concept

In distributor injection pumps featuring an integral soft-running device, the plunger is equipped with two longitudinal passages (Fig. 19, Pos. 3 and 5) connected by a ring groove (6). The longitudinal passage 3 is connected to a cutoff bore (7) with a restrictor in the area adjacent to the control collar (1).

The plunger travels through the stroke h_1 on its path toward TDC. The cutoff bore (7) connected to passage 3 emerges from the control collar earlier than the cutoff bore (2) on passage 5.

Because the ring groove (6) links passages 3 and 5, this causes a portion of the fuel to seep from the plunger chamber back to the pump's inner chamber. This reduces the fuel-delivery rate (less fuel discharged for each degree of camshaft travel). The travel that the camshaft executes for any given quantity of fuel injected is roughly doubled (Fig. 18).

Under extreme loads, the control collar is closer to the distributor head. This makes the distance h_2 smaller than h_1. When the plunger now moves toward TDC, the ring groove (6) is covered before the cutoff bore (7) emerges from the control collar. This cancels the link joining passages 3 and 5, deactivating the soft-running mechanism in the high-load range.

Quiet running device (operation)

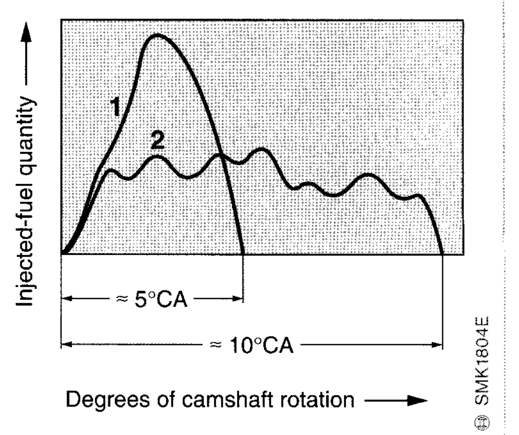


Fig. 18
1 without quiet running
2 with quiet running

Quiet running device (section)

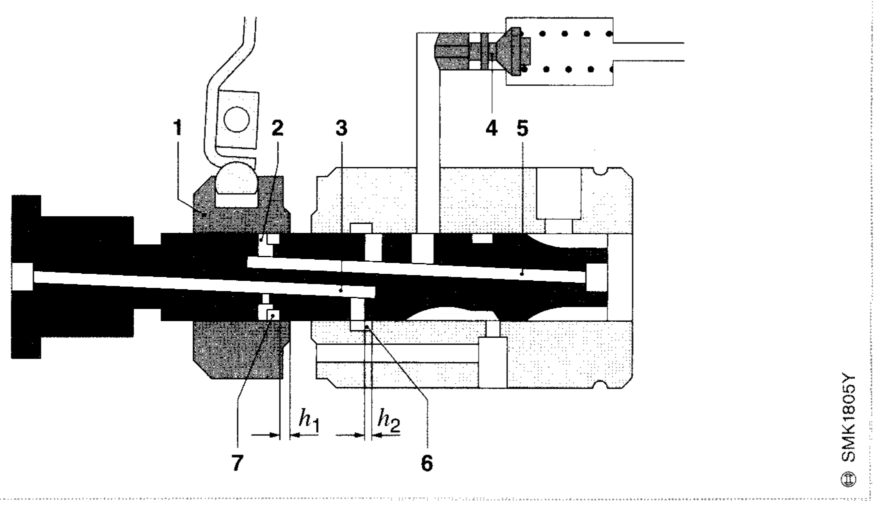


Fig. 19
1 Control collar
2 Cutoff bore
3 Port 3
4 1-way non-return valve for nozzle
5 Port 5
6 Ring groove
7 Cutoff bore

h_1 Lift 1
h_2 Lift 2

TDC **T**op **D**ead **C**enter on distributor plunger

Load switch

The load switches are installed at the distributor injection pump's main rotational-speed control lever. These switches control operation of assemblies outside of the fuel-injection pump. They rely on microswitches for electronic control, or valves for pneumatic switching of these external units. Their primary application is in triggering the exhaust-gas recirculation valve. In this application, the switch opens and closes the valve in response to variations in the main control lever's position.

Two different elements are available to control microswitches and pneumatic valves:
- Angular stop plate with control recess and
- Cast aluminum control cams

The control plate or cam attached to the main control lever initiates activation or deactivation based on the lever's travel. The switching point corresponds to a specific coordinate on the pump's response curve (rotational speed vs. delivery quantity).

Either two microswitches or one pneumatic valve can be installed on the fuel-injection pump due to differences in the respective layouts.

Microswitch

The electric microswitch is a bipolar make-and-break switch. It consists of leaf spring elements with a rocker arm. A control pin presses on the springs. A supplementary control element (Fig. 1, Pos. 6) limits mechanical wear and provides a unified control stroke for all applications on distributor injection pumps.

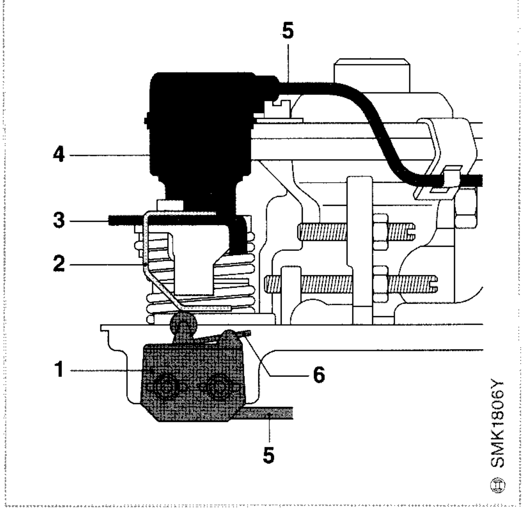
Pneumatic valve

The pneumatic valve (Fig. 2) blocks air flow in a vacuum line.

Potentiometer

The potentiometer is an option available for cases in which the object is to control several different points on the curve for rotational speed vs. load factor. It is mounted on top of the rotational-speed control lever, to which it is attached by clamps (Fig. 1). When energized, the potentiometer transmits a continuous electrical voltage signal that reflects the control lever's position with a linear response curve. A suitable governor design then provides useful information in the defined range.

Potentiometers and microswitches



Pneumatic valve

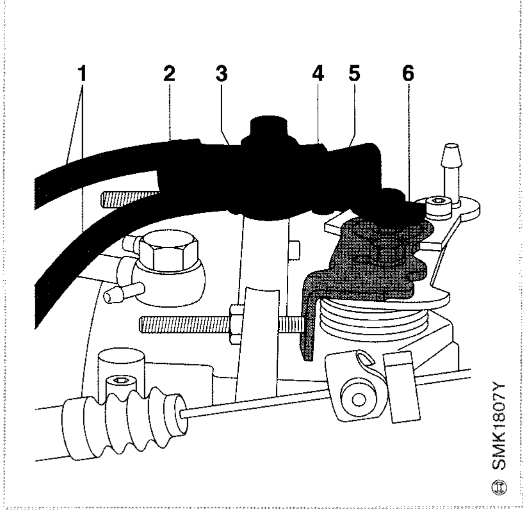


Fig. 1
1 Microswitch
2 Angular stop bracket
3 Rotational-speed control lever
4 Potentiometer
5 Electrical connection
6 Auxiliary controller

Fig. 2
1 Pneumatic connections
2 Vacuum adjusting screw
3 Pneumatic valve
4 Control lever
5 Control roller
6 Rotational-speed control lever

Delivery-signal sensor

Application

The FSS delivery-signal sensor is a dynamic pressure sensor installed in the threaded center plug of the diesel fuel-injection pump in place of the vent screw (Fig. 2). It monitors pressure in the element chamber. The sensor signal can be employed to register the start of delivery, the delivery period and the pump speed *n*.

The pressure-signal measurement range extends from 0...40 MPa, or 0...400 bar, at which it is suitable for use with IDI fuel-injection pumps. This device is a "dynamic sensor". It registers pressure variations instead of monitoring static pressure levels.

Design and operating concept

The delivery-signal sensor relies on piezoelectric principles for operation. The pressure in the element chamber acts on a sensor measurement cell (Fig. 2, Pos. 13). This contains the piezoelectric ceramic layer. Pressure variations modify electrical charge states in this layer. These charge shifts generate minute electric voltages, which the integrated circuit (11) in the sensor converts to a square-wave signal (Fig. 1).

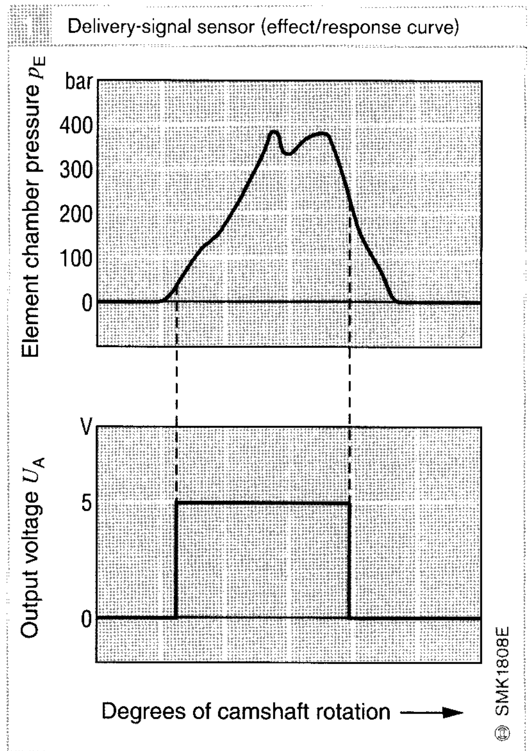


Fig. 1
a Pressure curve for element chamber
b Signal from delivery-signal sensor

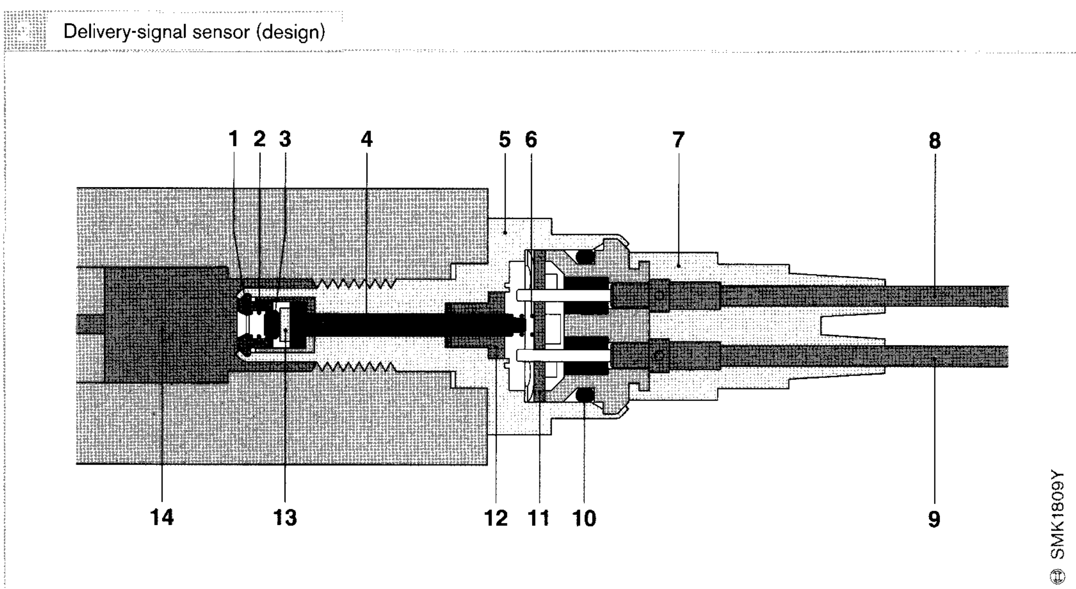


Fig. 2
1 Disk
2 Spring
3 Seal
4 Contact pin
5 Housing
6 Contact spring
7 Cable housing
8 Power supply connection (yellow/white)
9 Signal connection (gray/white)
10 O-ring
11 Circuit board with integrated circuit
12 Insulator
13 Sensor measurement cell (piezoelectric ceramic material)
14 Fuel-injection pump element chamber

Shutoff devices

Shutoff

One inherent property of the diesel engine's auto-ignition concept is that interrupting the fuel supply represents the only way to switch off the engine.

The mechanically controlled distributor injection pump is usually deactivated by an electric shutoff valve (ELAB). It is only in rare special application environments that these pumps are equipped with mechanical shutoff devices.

Solenoid-operated shutoff valve (ELAB)
The electric shutoff device is a solenoid valve. Operation is controlled by the "ignition" key to offer drivers a high level of convenience.

The solenoid valve employed to interrupt fuel delivery in the distributor injection pump is mounted on top of the distributor head (Fig. 1). When the diesel engine is running, the solenoid (4) is energized, and the armature retracts the sealing cone (6) to hold the inlet passage to the plunger chamber open. Switching off the engine with the key interrupts the current to the solenoid coil. The magnetic field collapses and the spring (5) presses the armature and sealing cone back against the seat. By blocking fuel flow through the plunger chamber's inlet passage, the seal prevents delivery from the plunger. The solenoid can be designed as either pulling or pushing electromagnet.

The ELABs used on marine engines open when de-energized. This allows the vessel to continue in the event of electrical system failure. Minimizing the number of electrical consumers is a priority on ships, as the electromagnetic fields generated by continuous current flow promote salt-water corrosion.

Mechanical shutoff device
The mechanical shutoff devices installed on distributor injection pumps consist of lever assemblies (Fig. 2). This type of assembly consists of an inside and outside stop lever mounted on the governor cover (1, 5). The driver operates the outer stop lever from the

Solenoid-operated shutoff valve (ELAB)

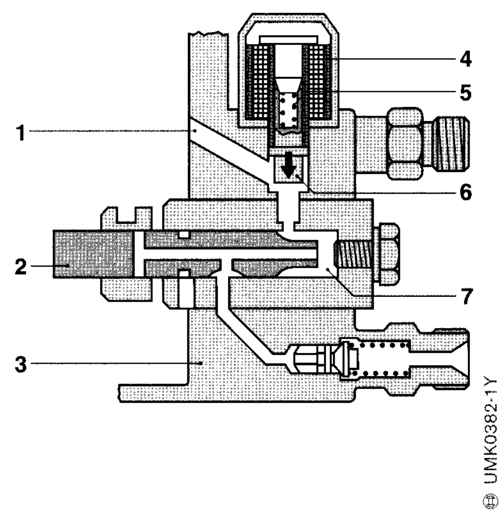


Fig. 1
1 Inlet passage
2 Distributor plunger
3 Distributor head
4 Solenoid (pulling electromagnet in this instance)
5 Spring
6 Armature with sealing cone
7 Plunger chamber

Mechanical shutoff device

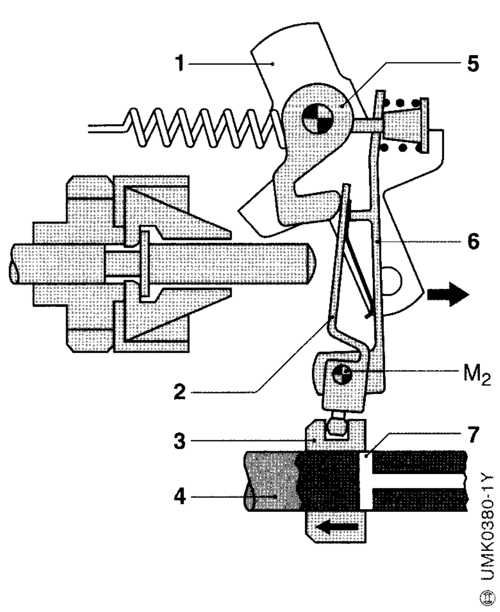


Fig. 2
1 External stop lever
2 Starting lever
3 Control collar
4 Distributor plunger
5 Inner stop lever
6 Tensioning lever
7 Control port

M_2 Pivot axis for 2 and 6

vehicle's interior via Bowden cable, etc. Cable operation causes the two stop levers to rotate about their pivot axis, with the inside stop lever pressing against the control mechanism's (2) starting lever. The starting lever pivots about its axis M_2 to push the control collar (3) into its stop position. This acts on the control passage (7) in the plunger to prevent fuel delivery.

Electric actuator mechanism
Systems with electronic diesel control employ the fuel-delivery actuator to switch off the engine (control parameter from electronic control unit (ECU): injected-fuel quantity = zero; refer to following section). The separate solenoid-operated shutoff valve (ELAB) serves only as a backup system in the event of a defect in the actuator mechanism.

Electronic Diesel Control

Systems with mechanical rotational-speed control respond to variations in operating conditions to reliably guarantee high-quality mixture formation.

EDC (Electronic Diesel Control) simultaneously satisfies an additional range of performance demands. Electric monitoring, flexible electronic data processing and closed-loop control circuits featuring electric actuators extend performance potential to include parameters where mechanical systems have no possibility of supply regulation. At the core of EDC is the electronic control unit that governs distributor injection pump operation.

As Electronic Diesel Control supports data communications with other electronic systems (traction control, electronic transmission-shift control, etc.) it can be integrated in an all-encompassing vehicle system environment.

Helix and port-controlled axial-piston distributor injection pump with electric control mechanism (VP37)

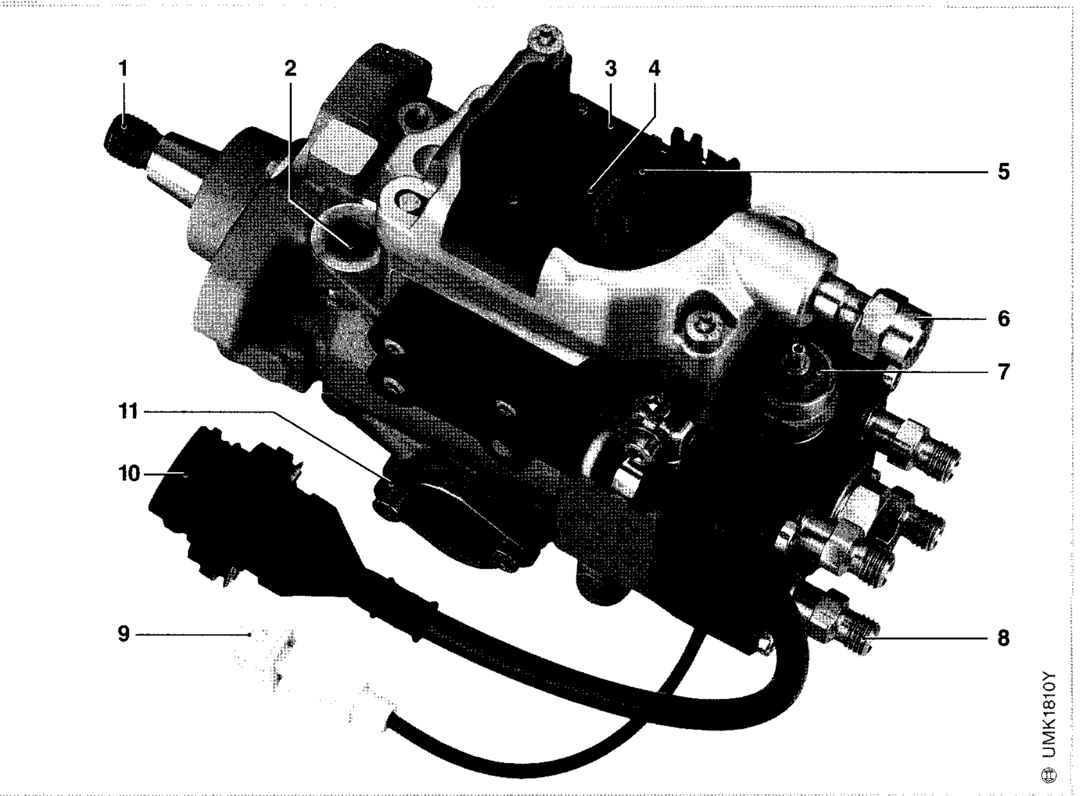


Fig. 1
1 Pump drive
2 Fuel inlet
3 Fuel-injection pump delivery positioner
4 Fuel temperature sensor
5 Angle sensor
6 Fuel return
7 Type ELAB electric shutoff valve
8 Delivery valve (discharge to nozzle)
9 Connection for start of injection solenoid valve
10 Connection for fuel-injection pump delivery positioner
11 Timing device

Helix and port-controlled electronic distributor injection pumps use a fuel-delivery actuator mechanism to regulate injected-fuel quantities and a solenoid valve to control start of injection.

Solenoid actuator for fuel-delivery control

The solenoid actuator (rotary actuator, Fig. 2, Pos. 2) acts on the control collar via shaft. On mechanically controlled fuel-injection pumps, the position determines the point at which the cutoff bores are exposed.

Injected fuel quantity can be progressively varied continuously from zero to the system's maximum capacity (for instance, during cold starts). This valve is triggered by a pulse-width modulated (PWM) signal. When current flow is interrupted, the rotary-actuator return springs reduce the fuel-delivery quantity to zero.

A sensor with a semidifferential short-circuit ring (1) relays the actuator-mechanism rotation angle, and thus the control collar position, back to the control unit [1]). This action determines the injected-fuel quantities for the instantaneous engine speed.

Solenoid valve for start of injection timing

As with mechanical timing devices, the timing plunger reacts to pressure in the pump's inner chamber, which rises proportionally to rotational speed. This force against the timing device's pressure-side is controlled by the timing-device solenoid valve (5). This valve is triggered by a PWM signal.

When the solenoid valve remains constantly open, pressure falls to retard start of injection, while closing the valve advances the timing. The electronic control unit can continuously vary the PWM signal's pulse-duty factor (ratio of open to closed periods at the valve) in the intermediate range.

[1]) On 1st generation pumps, a potentiometer was employed to register rotation angle.

Helix and port-controlled axial-piston distributor injection pumps with electric control mechanisms (VE..EDC)

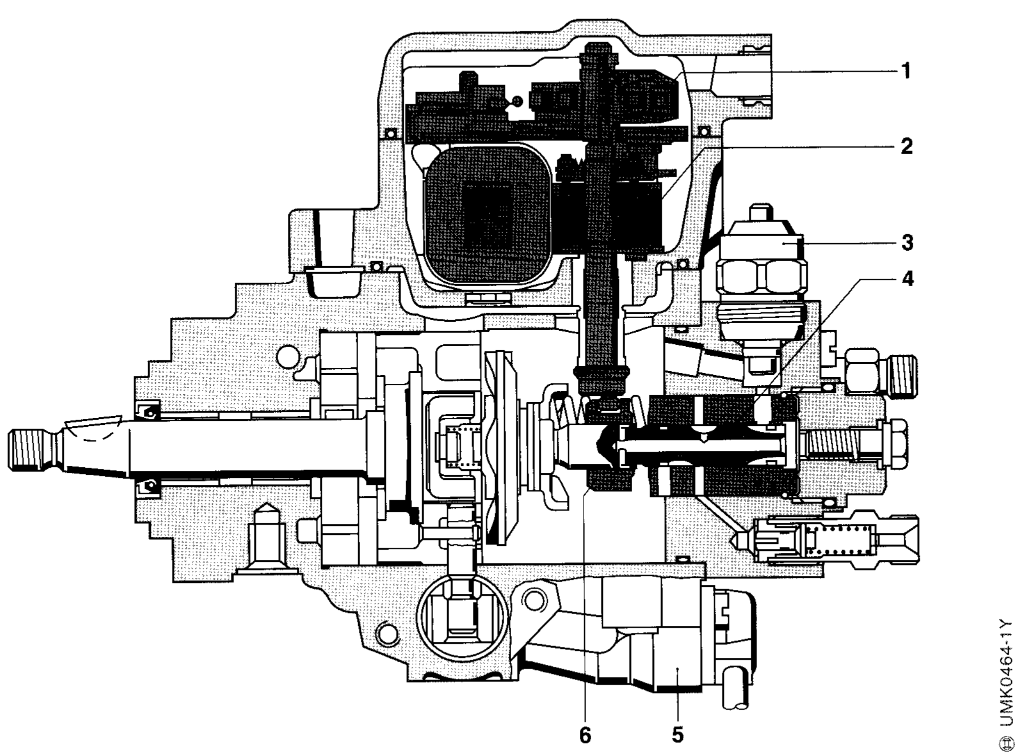


Fig. 2
1 Semi-differential short-circuiting ring sensor
2 Solenoid control mechanism
3 Electric shutoff valve ELAB
4 Distributor plunger
5 Timing-device solenoid valve
6 Control collar

Sensor with semidifferential short-circuiting ring

Applications

Semidifferential short-circuit ring sensors are position sensors for monitoring travel and angles. These sensors are extremely precise and robust. They are used as:

- Control-rack travel sensors for monitoring control-rack position on in-line diesel fuel-injection pumps and
- Angle sensors for fuel-delivery actuator assemblies in diesel distributor injection pumps

Design and operating concept

The sensors (Figures 3 and 4) are built around a laminated soft-iron core. Attached to each leg of the soft-iron core are a sensor coil and a reference coil.

As the AC current transmitted by the control unit permeates the coils, it creates alternating magnetic fields. The copper short-circuiting rings surrounding the legs of the soft-iron core serve as a shield against these alternating magnetic fields. While the short-circuiting ring for reference remains stationary, the measuring ring is attached to the control rack or control-collar shaft (control-rack travel s or advance angle φ).

Changes in the measuring ring's position change the magnetic flux to vary the coil's voltage, while the control unit maintains voltage at a constant level (impressed current).

Evaluation circuitry calculates the ratio of output voltage U_A to reference voltage U_{Ref} (Fig. 5), which is proportional to the deflection angle of the measurement short-circuiting ring. This curve's gradient can be adjusted by bending the reference short-circuiting ring while the base point is repositioned by shifting the position of the measuring ring.

Design of the HDK semidifferential short-circuit-ring sensor as used in distributor injection pumps for diesels

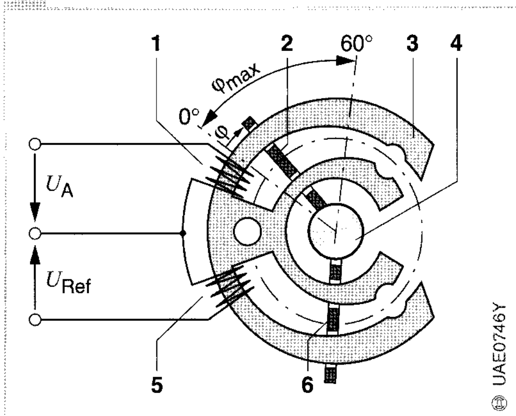


Fig. 3
1 Sensor coil
2 Sensor short-circuiting ring
3 Soft iron core
4 Control collar shaft
5 Reference coil
6 Reference short-circuiting ring
φ_{max} Adjustment range of control-collar shaft
φ Registered angle

Design of RWG Control-rack travel sensor for in-line diesel fuel-injection pumps

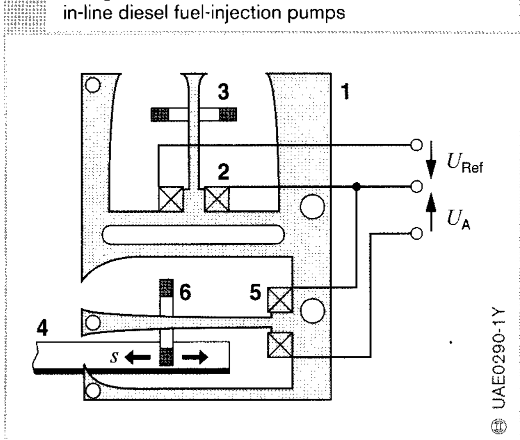


Fig. 4
1 Soft-iron core
2 Reference coil
3 Reference short-circuiting ring
4 Control rack
5 Sensor coil
6 Sensor short-circuiting ring
s Control-rack travel

Voltage ratio as a function of control-rack travel

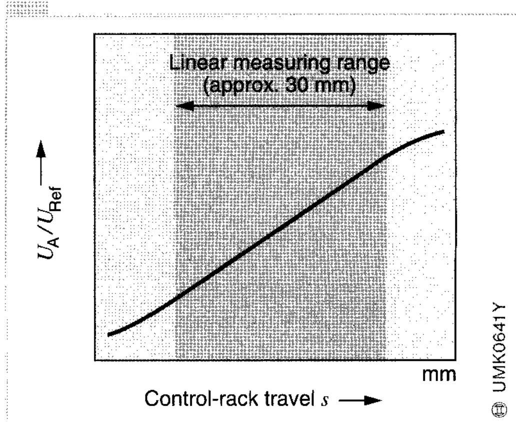


Fig. 5
U_A Voltage output
U_{Ref} Reference voltage

Diesel-engine immobilizers

Diesel immobilizers is a component in the electronic vehicle immobilizer installed on vehicles with a mechanically-controlled distributor injection pump. The diesel immobilizer is mounted above the solenoid-operated shutoff valve (ELAB) of the distributor injection pump. It responds to signals from the immobilizer control unit by switching current flow to the ELAB off and on to allow or block the flow of diesel fuel to the engine. The diesel immobilizer unit and ELAB are mounted behind a metal plate which must be destroyed for removal.

The diesel-engine immobilizer's input wiring usually comprises three terminals, consisting of power supply, data line and ground wire. The single exit wire carries the current to the ELAB.

Control units

The DDS 1.1. and DDS 3.1 control units are available for diesel-engine immobilizers.

DDS1.1

DDS 1.1 is a PCB-equipped ECU featuring an integrated protective housing made of plastic with injected sheet metal. On its own, DDS 1.1 provides only the lowest degree of protection. The security rating can be raised by adding various external protection features to the pump.

DDS3.1

DDS 3.1 is a hybrid control unit with a supplementary protective housing made of manganese steel. DDS 3.1 can provide optimum security without any supplements.

The DDS system must be released on the pump for operation on the test bench.

Helix and port-controlled axial-piston distributor injection pump with diesel antitheft protection

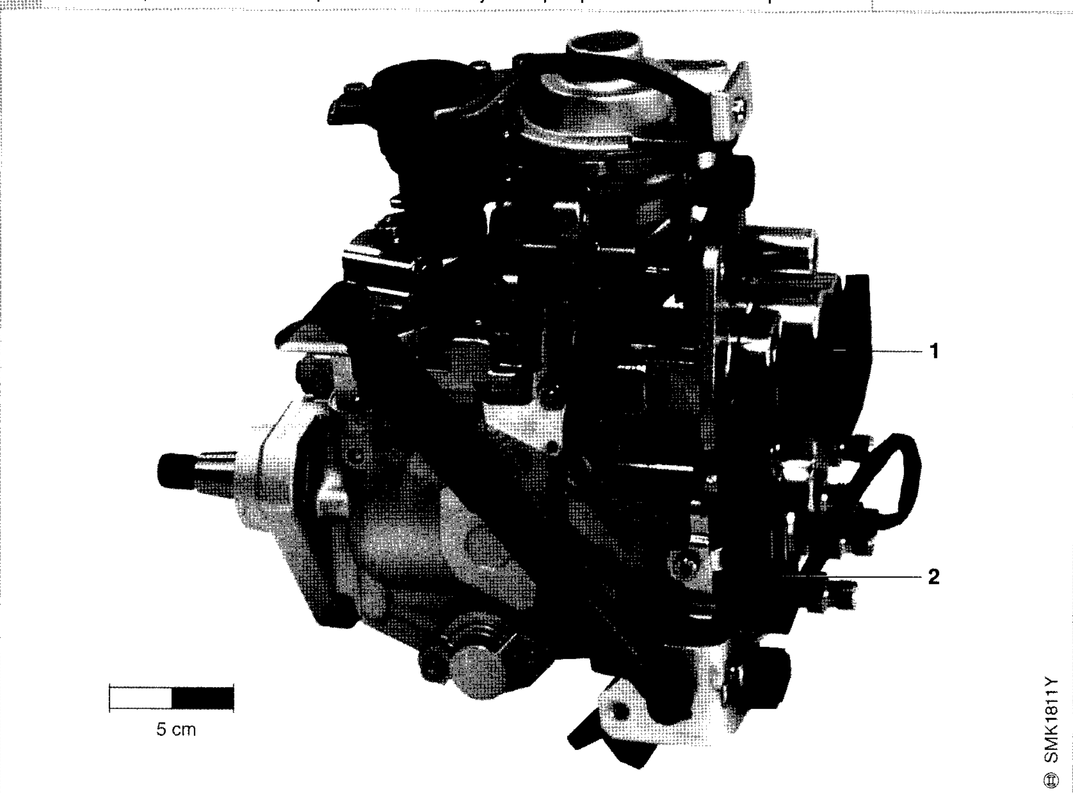


Fig. 1
1 DDS 1 (via ELAB)
2 Plug connection

Solenoid-valve controlled distributor injection pumps

Ever stricter emission limits for diesel engines and the demand for further reductions in fuel consumption have resulted in the continuing refinement of electronically controlled distributor injection pump. High-pressure control using a solenoid valve permits greater flexibility in the variation of start and end of delivery and even greater accuracy in the metering of the injected-fuel quantity than with port-controlled fuel-injection pumps. In addition, it permits pre-injection and correction of injected-fuel quantity for each cylinder.

The essential differences from port-controlled distributor injection pumps are the following:
- A control unit mounted on the pump
- Control of fuel injection by a high-pressure solenoid valve and
- Timing of the high-pressure solenoid valve by an angle-of-rotation sensor integrated in the pump

As well as the traditional benefits of the distributor injection pump such as light weight and compact dimensions, these characteristics provide the following additional benefits:
- A high degree of fuel-metering accuracy within the program map
- Start of injection and injection duration can be varied independently of factors such as engine speed or pump delivery quantity
- Injected-fuel quantity can be corrected for each cylinder, even at high engine speeds
- A high dynamic volume capability
- Independence of the injection timing device range from the engine speed and
- The capability of pre-injection

Areas of application

Solenoid-valve controlled distributor injection pumps are used on small and medium-sized diesel engines in cars, commercial vehicles and agricultural tractors. They are fitted both on direct-injection (DI) and indirect-injection (IDI) engines.

The nominal speed, power output and design of the diesel engine determine the type and size of the fuel-injection pump. Distributor injection pumps are used on car, commercial-vehicle, agricultural tractor and fixed-installation engines with power outputs of up to 45 kW per cylinder. Depending on the type of control unit and solenoid valve, they can be run off either a 12-volt or a 24-volt electrical system.

Solenoid-valve controlled distributor injection pumps are available with high-pressure outlets for either four or six cylinders. The maximum injected-fuel capacity per stroke is in the range of 70...175 mm^3. The required maximum injection pressures depend on the requirements of the engine (DI or IDI). They range from 800...1,950 bar.

All distributor injection pumps are lubricated by the fuel. Consequently, they are maintenance-free.

Designs

There are two basic designs:
- Axial-piston distributor pumps (Type VE..MV or VP29/VP30) and
- Radial-piston distributor pumps (Type VR or VP44)

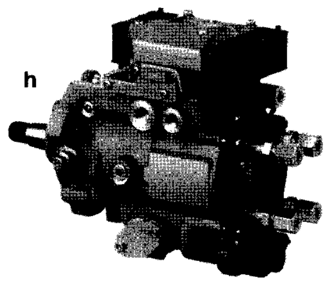
There are a number of design variations (e.g. number of outlets, pump drive by ring gear) according to the various types of application and engine.

The hydraulic performance capacity of the axial-piston pump with injection pressures up to 1,400 bar at the nozzle will remain sufficient for many direct-injection and indirect-injection engines in the future.

Where higher injection pressures are required for direct-injection engines, the radial-piston distributor pump introduced in 1996 is the more suitable choice. It has the capability of delivering injection pressures of up to 1950 bar at the nozzle.

Family tree of Bosch electronically controlled distributor injection pumps

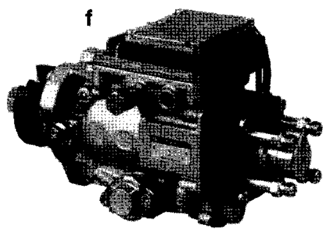
IDI-engines | **DI-engines**

h


2000 VR (VP44, h) with integral engine-management control unit (PSG16)

1999 VE..MV (VP30, g)

1998 VE..MV (VP 29, f)

f


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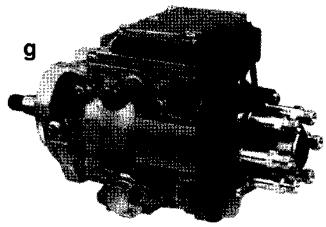
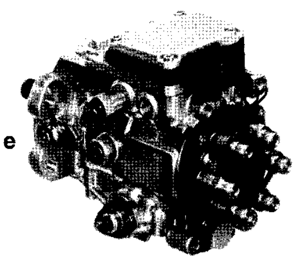
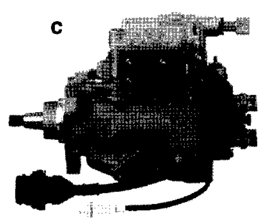


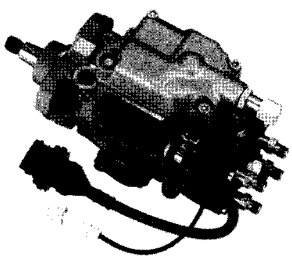
1996 VR (VP44, e)

e


Solenoid-valve controlled

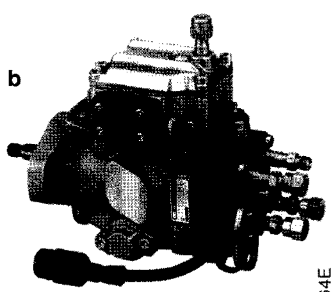
Port-controlled

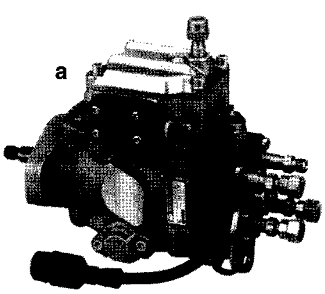
c


d


1990 VE..EDC (VP36, c)
VE..EDC (VP37, d)

1989 VE..EDC (VP34, b)

b


a


1986 VE..EDC (VP15, a)

SMK1864E

Fitting and drive system

Distributor injection pumps are flange-mounted directly on the diesel engine. The engine's crankshaft drives the fuel-injection pump's heavy-duty drive shaft by means of a toothed-belt drive, a pinion, a gear wheel or a chain. In axial-piston pumps, the drive shaft runs on two plain bearings in the pump housing. In radial-piston pumps, there is a plain bearing at the flange end and a deep-groove ball bearing at the opposite end. The distributor injection pump is fully synchronous with the engine crankshaft and pistons (positive mechanical link).

On four-stroke engines, the pump speed is half that of the diesel-engine crankshaft speed. In other words, it is the same as the camshaft speed.

There are distributor injection pumps for clockwise and counterclockwise rotation [1]). Consequently, the injection sequence differs according to the direction of rotation – but is always consecutive in terms of the geometrical arrangement of the outlets (e.g. A-B-C or C-B-A).

In order to prevent confusion with the numbering of the engine cylinders (cylinder no. 1, 2, 3, etc.), the distributor-pump outlets are identified by letters (A, B, C, etc., Fig. 1). For example, the assignment of pump outlets to engine cylinders on a four-cylinder engine with the firing sequence 1-3-4-2 is A-1, B-3, C-4 and D-2.

1) Direction of rotation as viewed from drive-shaft end of the pump

In order to achieve good hydraulic characteristics on the part of the fuel-injection system, the high-pressure fuel lines between the fuel-injection pump and the nozzle-and-holder assemblies must be as short as possible. For this reason, the distributor injection pump is fitted as close as possible to the diesel-engine cylinder head.

The interaction of all components has to be optimized in order to insure that the diesel engine and the fuel-injection system function correctly. The fuel lines and the nozzle-and-holder assemblies must therefore not be interchanged when carrying out servicing work.

In the high-pressure system between the fuel-injection pump and the nozzles, there are degrees of free play manufactured to an accuracy of a few thousandths of a millimeter. That means that dirt particles in the fuel can impair the function of the system. Consequently, high fuel quality and a fuel filter are required specifically designed to meet the requirements of the fuel-injection system. They prevent damage to pump components, delivery valves and nozzles and guarantee trouble-free operation and long service life.

Diesel fuel can absorb water in solution in quantities ranging between 50...200 ppm (by weight), depending on temperature. This dissolved water does not damage the fuel-injection system. However, if greater quantities of water find their way into the fuel (e.g. condensate resulting from changes in temperature), they remain present in undissolved form. If undissolved water accesses the fuel-injection pump, it may result in corrosion damage. Distributor injection pumps therefore need fuel filters with water chambers (see chapter "Fuel supply system"). The collected water has to be drained off at appropriate intervals. The increased use of diesel engines in cars has resulted in the demand for an automatic water warning device. It indicates whenever water needs to be drained off by means of a warning light.

Outlet designation of a six-cylinder distributor injection pump

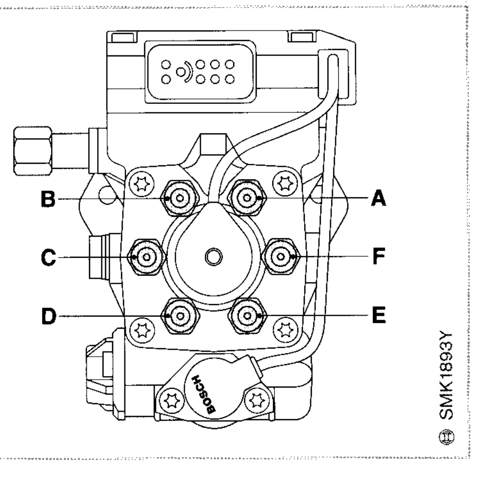


Fig. 1
Outlets are always numbered counterclockwise. Numbering starts at top right.

1998 Diesel Records

The 24-hour race on Germany's famous "Nürburgring" race track is not just about speed but about the durability of automotive technology. On 14th June 1998 the race was won for the first time by a diesel-engined car. The BMW 320 d left its gasoline-engined rivals trailing.

The average speed achieved during practice was approximately 160 kph. The maximum speed was around 250 kph. The 1,040-kg vehicle accelerated from 0 to 100 kph in 4.5 seconds.

The racing car was powered by a four-cylinder direct-injection diesel engine with a capacity of 1950 cc and a maximum power output of more than 180 kW (245 bhp) at 4,200...4,600 rpm. The engine produced its maximum torque of 430 N·m at 2,500...3,500 rpm.

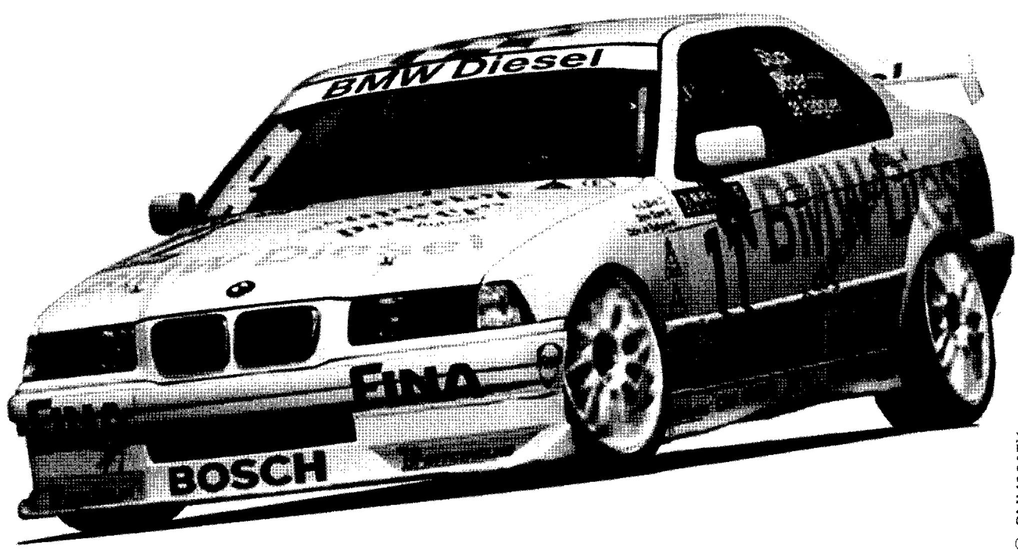
The car's performance was due in no small part to its high-performance fuel-injection system designed by Bosch. The central component of the system is the Type VP44 radial-piston distributor injection pump that has been in volume production since 1996.

Compared with normal production engines, the racing diesel is an "enhanced-performance" model achieved by larger injected-fuel quantities and even higher injection pressures and engine speeds. For this purpose, a new high-pressure stage with 4 delivery plungers instead of 2 was fitted in combination with new nozzles. The software for the control units was also modified.

These design modifications and the fact that the engines are run continuously at full power obviously reduce the life of the components (designed for 24...48 hours of service), and increase the concentration levels of the noxious constituents and fuel consumption. The car used around 23 *l*/100 km (a comparable gasoline racing engine uses almost twice as much).

That impressive victory demonstrates once again that the diesel engine is no longer the "lame duck" that it used to be.

BMW 320 d: Winner of the 1998 24-hour race



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Design and method of operation

Assemblies

Solenoid-valve controlled distributor injection pumps are modular in design. This section explains the interaction between the individual modular components. They are described in more detail later on. The modular components referred to are the following (Figs. 1 and 2):

- *The low-pressure stage* (7), consisting of the vane-type supply pump, pressure-control valve and overflow throttle valve
- *The high-pressure stage* (8)
- *The delivery valves* (11)
- *The high-pressure solenoid valve* (10)
- *The timing device* (9) with timing-device solenoid valve and angle-of-rotation sensor and
- *The pump ECU* (4)

The combination of these modular components in a compact unit allows the interaction of individual functional units to be very precisely coordinated. This allows compliance with strict tolerances and fully meets demands regarding performance characteristics.

The high pressures and the associated physical stresses require a wide variety of fine-tuning adjustments in the design of the components. Lubrication is carried out by the fuel itself. In some cases, lubricant-film thickness is less than 0.1 µm and is much less than the minimum achievable surface roughness. For this reason, the use of special materials and manufacturing methods is necessary in addition to special design features.

Fuel supply and delivery

Low-pressure stage

The vane-type supply pump in the low-pressure stage (7) pumps fuel from the fuel tank and produces a pressure of 8...22 bar inside the fuel-injection pump depending on pump type and speed.

The vane-type supply pump delivers more fuel than is required for fuel injection. The excess fuel flows back to the fuel tank.

Assemblies and their functions (schematic diagram)

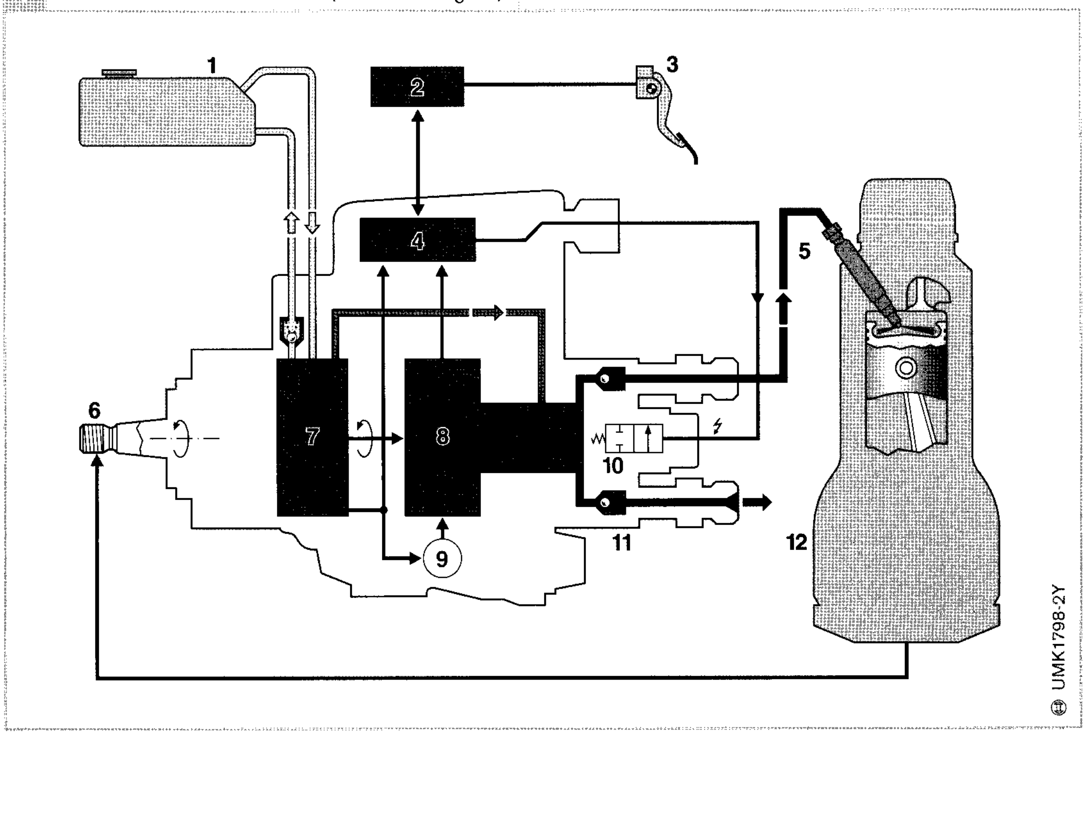


Fig. 1
1 Fuel supply system (low-pressure stage)
2 Type MSG engine ECU
3 Accelerator pedal
4 Type PSG pump ECU
5 Nozzle-holder assembly
6 Pump drive shaft
7 Low-pressure stage (vane-type supply pump with pressure-control valve and overflow throttle valve)
8 High-pressure stage (axial or radial-piston high-pressure pump with fuel rail)
9 Timing device with timing-device solenoid valve and angle-of-rotation sensor
10 High-pressure solenoid valve
11 Delivery valve
12 Diesel engine

High-pressure stage
The high-pressure pump (8) generates the high pressure required for fuel injection and controls the injected-fuel quantity as required. At the same time the fuel rail opens the outlet for the appropriate engine cylinder so that the fuel is delivered via the delivery valve (11) to the nozzle-and-holder assembly (5). The assembly then injects the fuel into the combustion chamber (12) of the engine. It is in the design of the high-pressure pump and fuel distributor that the axial-piston and radial-piston distributor pumps differ the most.

The high-pressure pump is able to deliver fuel as long the high-pressure solenoid valve (10) keeps the pump's plunger chamber sealed. In other words, it is the high-pressure solenoid valve that determines the delivery period and therefore, in conjunction with pump speed, the injected-fuel quantity and injection duration.

The injection pressure increases during the period of fuel injection. The maximum pressure depends on the pump speed and the injection duration.

Timing
As with port-controlled distributor injection pumps, the timing device (9) varies the start of injection. It alters the cam position in the high-pressure pump.

An injection-timing solenoid valve controls the supply-pump pressure acting on the spring-loaded timing-device piston. This controls the start of injection independently of engine speed.

The integrated angle-of-rotation sensor detect and, as necessary, correct the pump speed and, together with the speed sensor on the engine crankshaft, the position of the timing device.

Electronic control unit
The pump ECU (4) calculates the triggering signals for the high-pressure solenoid valve and the timing-device solenoid valve on the basis of a stored map.

2 Assemblies and their functions (schematic section diagram of a radial-piston distributor injection pump)

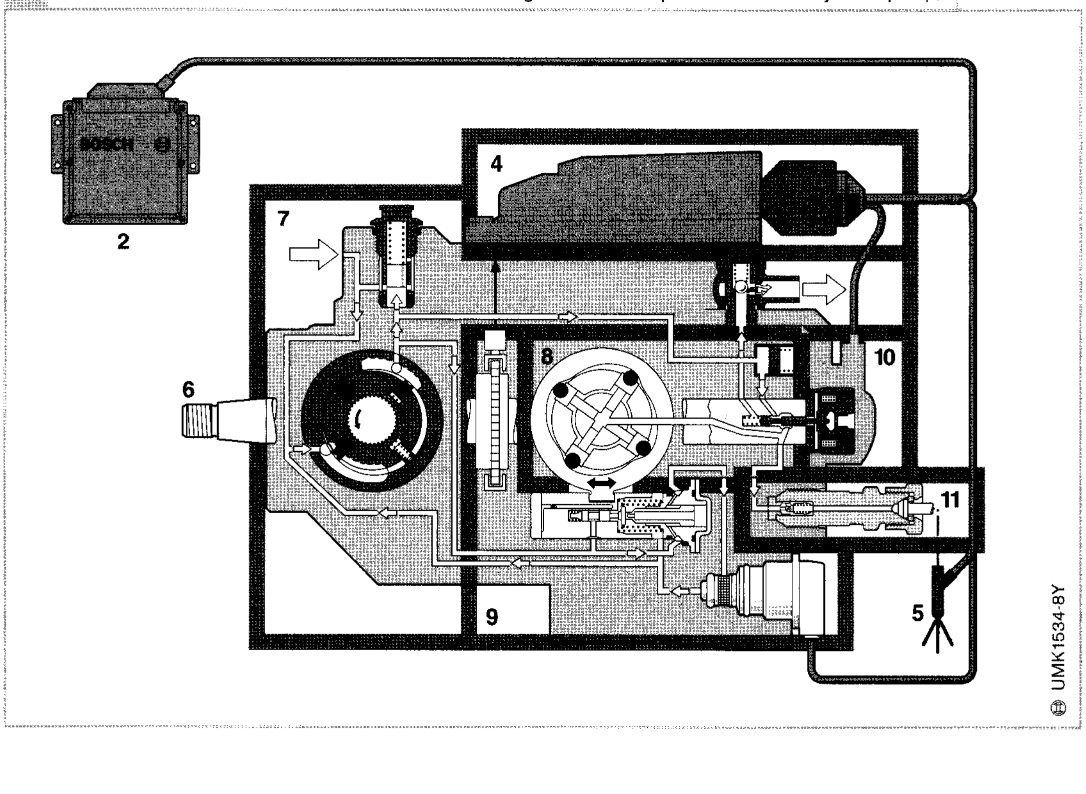


Fig. 2
For the sake of clarity, various components are shown end-on rather than side-on.
The index figures are the same as for Fig. 1.

2 Engine ECU
4 Pump ECU
5 Nozzle-holder assembly
6 Pump drive shaft
7 Low-pressure stage (vane-type supply pump with pressure-control valve and overflow throttle valve)
8 High-pressure pump with fuel rail
9 Timing device with timing-device solenoid valve and angle-of-rotation sensor
10 High-pressure solenoid valve
11 Delivery valve

Low-pressure stage

The low-pressure stage delivers sufficient fuel for the high-pressure stage and generates the pressure for the high-pressure pump and the timing device (8...25 bar depending on pump type). Its basic components include the vane-type supply pump, the pressure-control valve and the overflow valve.

Vane-type supply pump

The purpose of the vane-type supply pump (Fig. 1) is to draw in a sufficient quantity of fuel and to generate the required internal pressure.

The vane-type supply pump is positioned around the drive shaft (4) in the distributor injection pump. Between the inner surface of the pump housing and a support ring acting as the end plate is the retaining ring (2) which forms the inner surface of the vane-pump stator. On the inner surface of the pump housing, there are two machined recesses which form the pump inlet (5) and outlet (6). Inside the retaining ring is the impeller (3) which is driven by an interlocking gear (Type VP44) or a Woodruff key (VP29/30) on the drive shaft. Guide slots in the impeller hold the vanes (8) which are forced outwards against the inside of the retaining ring by centrifugal force. Due to the higher delivery pressures involved in radial-piston distributor injection pumps, the vanes have integrated springs which also help to force the vanes outwards. The compression-chamber "cells" (7) are formed by the following components:

- The inner surface of the pump housing ("base")
- The support ring ("cap")
- The shaped inner surface of the retaining ring
- The outer surface of the impeller and
- Two adjacent vanes

The fuel that enters through the inlet passage in the pump housing and the internal passages in the compression-chamber cell is conveyed by the rotation of the impeller to the compression-chamber outlet. Due to the eccentricity (VP29/30) or profile (VP44) of the inner surface of the retaining ring, the cell volume reduces as the impeller rotates. This reduction in volume causes the fuel pressure to rise sharply until the fuel escapes through the compression-chamber outlet – in other words, the fuel is compressed. From

Fig. 1
1 Pressure-control valve
2 Eccentric retaining ring
3 Impeller
4 Pump drive shaft
5 Fuel inlet
6 Fuel outlet to pump intake chamber
7 Compression-chamber cell
8 Vane

Vane-type supply pump

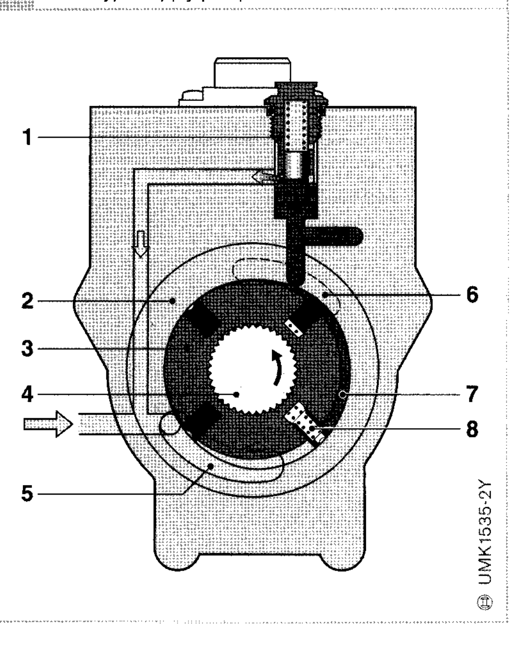
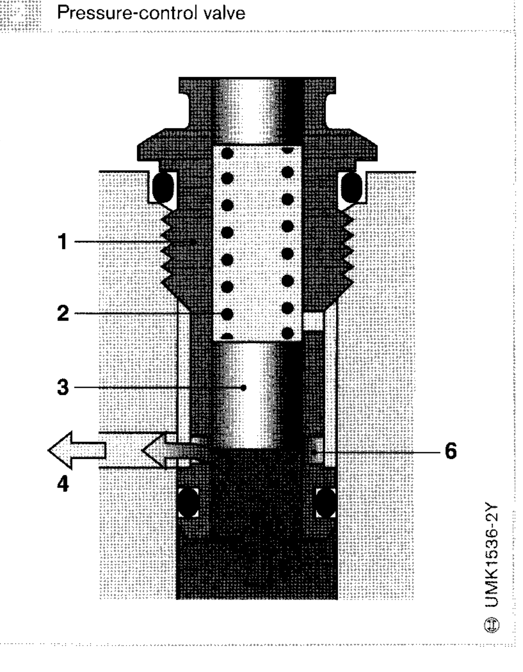


Fig. 2
1 Valve body
2 Compression spring
3 Valve plunger
4 Outlet to pump intake
5 Inlet from pump outlet
6 Bore

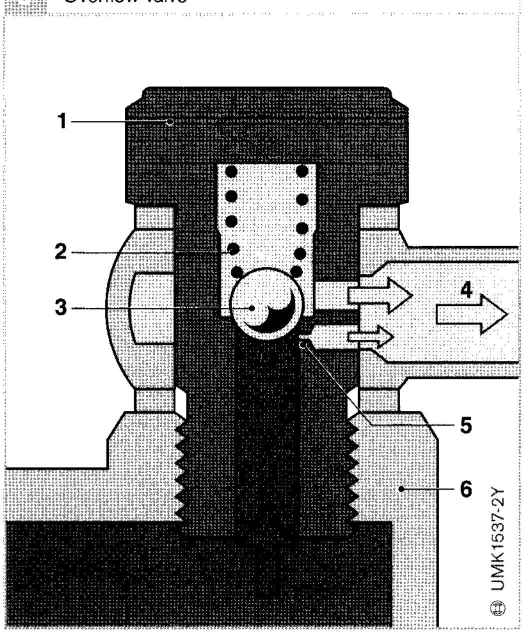
Pressure-control valve



the compression-chamber outlet, the various components are supplied with pressurized fuel via internal passages in the pump housing.

The pressure level required in a radial-piston distributor injection pump is relatively high compared with other types of distributor injection pump. Due to high pressure, the vanes have a bore in the center of the end face so that only one of the end-face edges is in contact with the inner surface of the retaining ring at one time. This prevents the entire end face of the vane from being subjected to pressure, which would result in an undesirable radial movement. At the point of changeover from one edge to the other (e.g. when changing over from inlet to outlet), the pressure acting on the end face of the vane can transfer to the other side of the vane through the bore. The opposing pressures balance each other out to a large extent and the vanes are pressed against the inner surface of the retaining ring by centrifugal force and the action of the springs as described above.

Overflow valve



Pressure-control valve

The fuel pressure created at the pressure outlet of the vane-type supply pump depends on the pump speed. To prevent the pressure from reaching undesirably high levels at high pump speeds, there is a pressure-control valve in the immediate vicinity of the vane-type supply pump which is connected to the pressure outlet by a bore (Fig. 1, Pos. 1). This spring-loaded slide valve varies the delivery pressure of the vane-type supply pump according to the fuel quantity delivered. If the fuel pressure rises beyond a certain level, the valve plunger (Fig. 2, Pos. 3) opens radially positioned bores (6) through which the fuel can flow via an outlet (4) back to the intake port of the vane-type supply pump.
If the fuel pressure is too low, the pressure-control valve remains closed and the entire fuel quantity is pumped into the distributor-pump intake chamber. The adjustable tension on the compression spring (2) determines the valve opening pressure.

Overflow valve

In order to vent and, in particular, cool the distributor injection pump, excess fuel flows back to the fuel tank through the overflow valve (Fig. 3) screwed to the pump housing.

The overflow valve is connected to the overflow valve (4). Inside the valve body (1), there is a spring-loaded ball valve (3) which allows fuel to escape when the pressure exceeds a preset opening pressure.

In the overflow channel to the ball valve, there is a bore that is connected to the pump overflow via a very small throttle bore (5). Since the overflow valve is mounted on top of the pump housing, the throttle bore facilitates automatic venting of the fuel-injection pump.

The entire low-pressure stage of the fuel-injection pump is precisely coordinated to allow a defined quantity of fuel to escape through the overflow valve and return to the fuel tank.

Fig. 3
1 Valve body
2 Compression spring
3 Valve ball
4 Fuel overflow
5 Throttle bore
6 Pump housing

High-pressure stage of the axial-piston distributor injection pump

Solenoid-valve controlled (Fig. 1) and port-controlled distributor injection pumps have essentially the same dimensions, fitting requirements and drive system including cam drive.

Design and method of operation

A clutch unit transmits the rotation of the drive shaft (Fig. 2 overleaf, Pos. 1) to the cam plate (5). The claws on the drive shaft and the cam plate engage in the yoke (3) positioned between them.

The cam plate converts the purely rotational movement of the drive shaft into a combined rotating-reciprocating movement. This is achieved by the fact that the cams on the cam plate rotate over rollers held in the roller ring (2). The roller ring is mounted inside the pump housing but has no connection to the drive shaft.

The cam plate is rigidly connected to the distributor plunger (8). Consequently, the distributor plunger also performs the rotating-reciprocating movement described by the cam plate. The reciprocating movement of the distributor plunger is aligned axially with the drive shaft (hence the name axial-piston pump).

The movement of the plunger back to the roller ring takes place the symmetrically arranged plunger return springs (7). They are braced against the distributor body (9) and act against the distributor plunger by means of a thrust plate (6). The piston return springs also prevent the cam plate from jumping away from the rollers in the roller ring when subjected to high acceleration forces.

Solenoid-valve controlled axial-piston distributor pump (cutaway view)

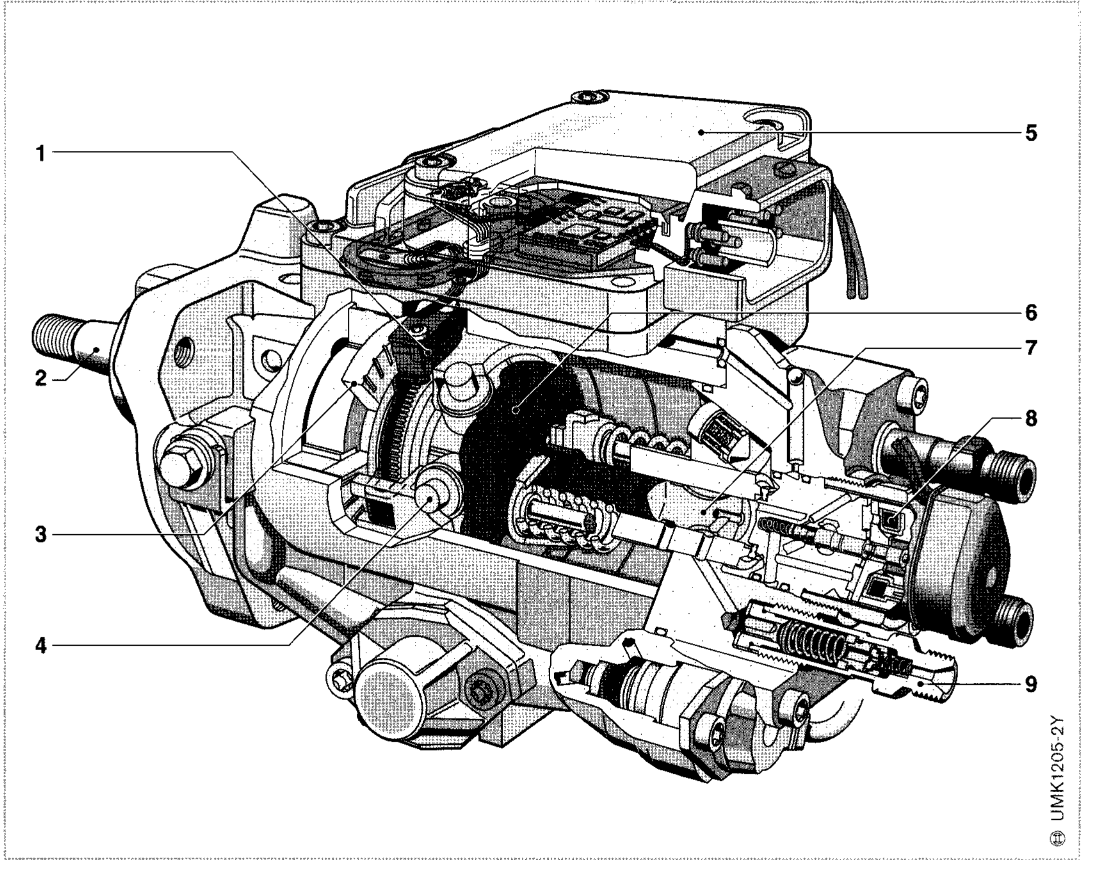


Fig. 1
1 Angle-of-rotation sensor
2 Pump drive shaft
3 Support ring of vane-type supply pump
4 Roller ring
5 Pump control unit
6 Cam plate
7 Distributor plunger
8 High-pressure solenoid valve
9 High-pressure outlet

The lengths of the plunger return springs are precisely matched to one another in order to prevent lateral forces from acting on the distributor plunger.

Although the distributor plunger moves horizontally, the limits of its travel are still referred to as top dead center (TDC) and bottom dead center (BDC). The length of the plunger stroke between bottom and top dead center is application-specific. It can be up to 3.5 mm.

The number of cams and rollers is determined by the number of cylinders in the engine. The cam shape affects injection pressure (injection pattern and maximum injection pressure) and the maximum possible injection duration. The factors determining this connection are cam lift and the speed of movement.

When designing the fuel-injection pump, the injection parameters must be individually adapted to suit the design of the combustion chamber and the nature of the combustion process (DI or IDI) employed by the engine on which the pump is to be used. For this reason, a specific cam profile is calculated for each type of engine and is then machined on the end face of the cam plate. The cam plate produced in this way is an application-specific component of the distributor injection pump. Cam plates are not interchangeable between different types of pump.

2 High-pressure stage inside a solenoid-valve controlled axial-piston distributor injection pump

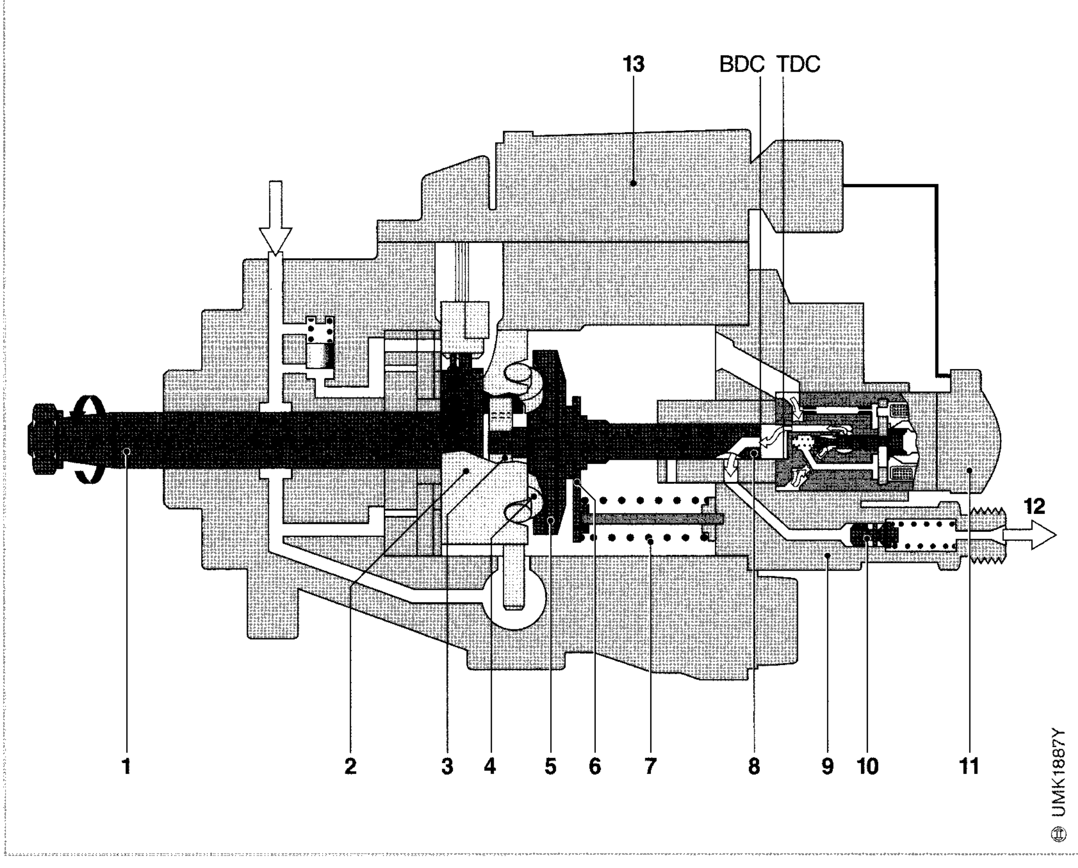


Fig. 2
1 Drive shaft
2 Roller ring
3 Yoke
4 Roller
5 Cam plate
6 Spring plate
7 Piston return spring (only one shown)
8 Distributor plunger
9 Distributor body (also called distributor head or distributor-head flange)
10 Delivery valve
11 High-pressure solenoid valve
12 Outlet to high-pressure delivery line
13 Pump ECU

TDC Distributor-plunger top dead center
BDC Distributor-plunger bottom dead center

Delivery phases

Induction (Fig. 4a)
As the distributor plunger (4) moves towards bottom dead center (BDC), it draws fuel into the plunger chamber (6) through the fuel inlet (1) in the distributor body (3) and the open high-pressure solenoid valve (7).

Effective stroke (Fig. 4b)
At bottom dead center, before the cam lobes are in contact with the rollers, the pump ECU sends a control signal to the high-pressure solenoid valve. The valve needle (9) is pressed against the valve seat (7). The high-pressure solenoid valve is then closed.

When the distributor plunger then starts to move towards top dead center (TDC), the fuel cannot escape. It passes through channels and passages in the distributor plunger to the high-pressure outlet (10) for the appropriate cylinder. The rapidly developed high pressure opens the orifice check valve (DI) or delivery valve (IDI), as the case may be, and forces fuel along the high-pressure fuel line to the nozzle integrated in the nozzle holder (start of delivery). The maximum injection pressure at the nozzle is around 1,400 bar.

The rotation of the distributor plunger directs the fuel to the next outlet on the next effective stroke.

The rapid release of pressure when the high-pressure solenoid valve opens can cause the space between the delivery valve and the distributor plunger to be over-depressurized. The filler channel (5) simultaneously fills the space for the outlet opposite to the outlet which is currently being supplied by the delivery stroke.

Residual stroke (Figure 4c).
Once the desired injected-fuel quantity has been delivered, the ECU cuts off the power supply to the solenoid coil. The high-pressure solenoid valve opens again and pressure in the high-pressure stage collapses (end of delivery). As the pressure drops, the nozzle and the valve in the high-pressure outlet close again and the injection sequence comes to an end. The point of closure and the open duration of the high-pressure solenoid valve, the cam pitch during the delivery stroke and the pump speed determine the injected-fuel quantity.

The remaining travel of the pump plunger to top dead center forces the fuel out of the plunger chamber and back into the pump intake chamber.

As there are no other inlets, failure of the high-pressure solenoid valve prevents fuel injection altogether. If the valve remains open, the required high pressure cannot be generated. If it remains closed, no fuel can enter the plunger chamber. This prevents uncontrolled over-revving of the engine and no other shutoff devices are required.

Axial-piston high-pressure pump

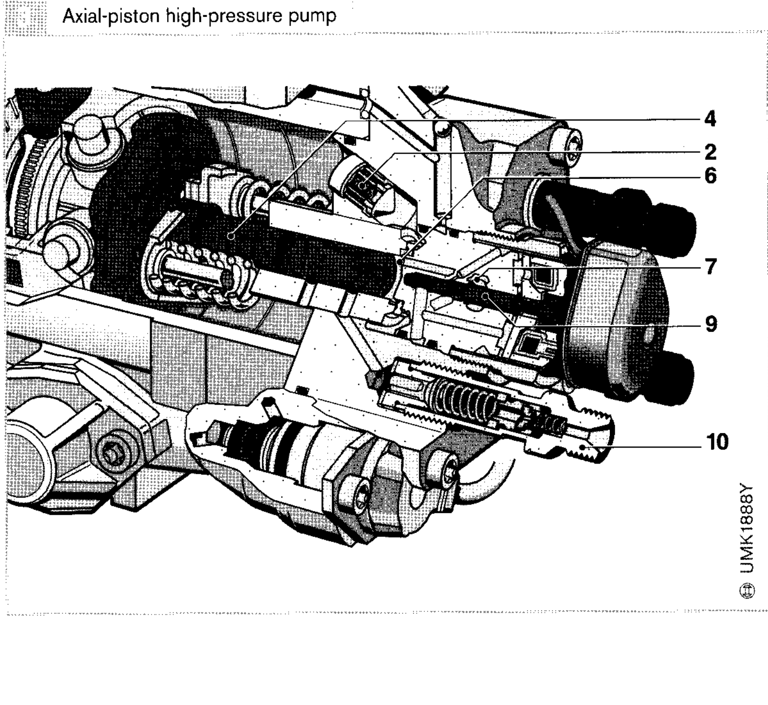


Fig. 3
The index figures are the same as for Fig. 4.

2 Filter
4 Distributor plunger
6 Plunger chamber
7 Valve seat
9 Solenoid-valve needle
10 Outlet to high-pressure delivery line

Delivery phases of solenoid-valve controlled axial-piston distributor injection pumps

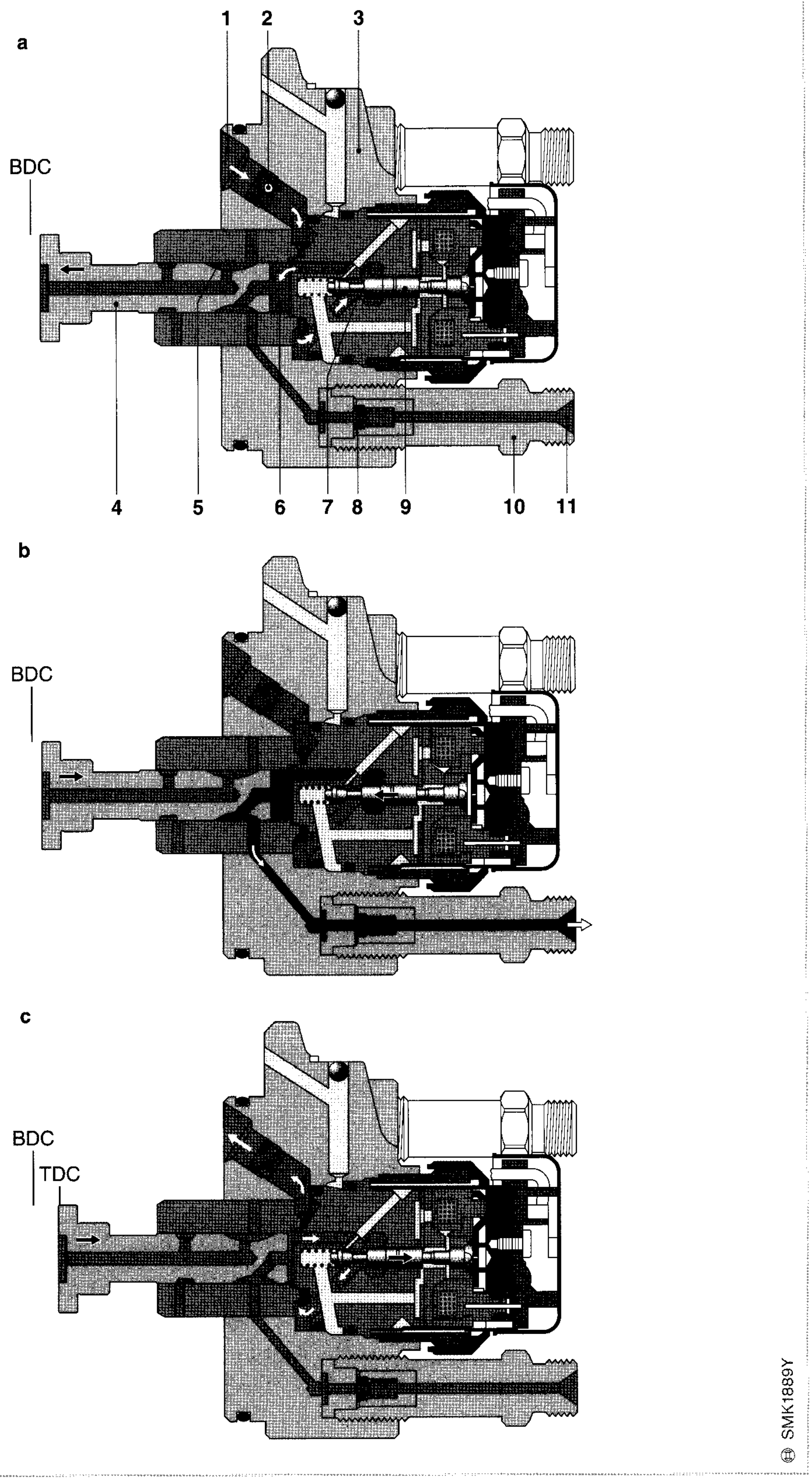


Fig. 4

a Induction
b Effective stroke
c Residual stroke

1 Inlet passage (fuel inlet)
2 Filter
3 Distributor body
4 Distributor plunger
5 Filler groove
6 Plunger chamber
7 Valve seat
8 Delivery valve
9 Solenoid-valve needle
10 High-pressure outlet
11 Outlet to high-pressure delivery line

TDC Pump-plunger **t**op **d**ead **c**enter
BDC Pump-plunger **b**ottom **d**ead **c**enter

High-pressure stage of the radial-piston distributor injection pump

Radial-piston high-pressure pumps (Fig. 1) produce higher injection pressures than axial-piston high-pressure pumps. Consequently, they also require more power to drive them (as much as 3.5...4.5 kW compared with 3 kW for axial-piston pumps).

Design

The radial-piston high-pressure pump (Fig. 2 overleaf) is driven directly by the distributor-pump drive shaft. The main pump components are

- The cam ring (1)
- The roller supports (4) and rollers (2)
- The delivery plungers (5)
- The drive plate and
- The front section (head) of the distributor shaft (6)

The drive shaft drives the drive plate by means of radially positioned guide slots. The guide slots simultaneously act as the locating slots for the roller supports. The roller supports and the rollers held by them run around the inner cam profile of the cam ring that surrounds the drive shaft. The number of cams corresponds to the number of cylinders in the engine.

The drive plate drives the distributor shaft. The head of the distributor shaft holds the delivery plungers which are aligned radially to the drive-shaft axis (hence the name "radial-piston high-pressure pump").

The delivery plungers rest against the roller supports. As the roller supports are forced outwards by centrifugal force, the delivery plungers follow the profile of the cam ring and describe a cyclical-reciprocating motion (plunger lift 3.5...4.15 mm).

Solenoid-valve-controlled radial-piston distributor pump (cutaway view)

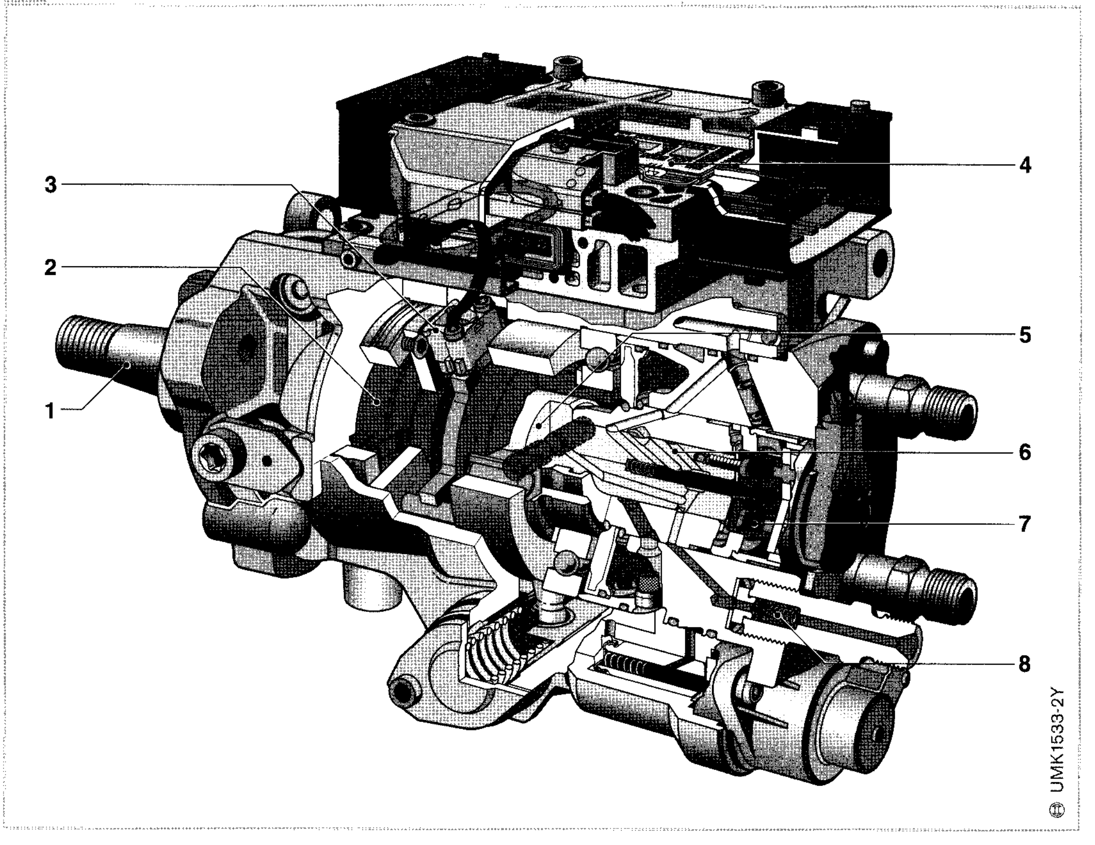


Fig. 1
1 Pump drive shaft
2 Vane-type supply pump
3 Angle-of-rotation sensor
4 Pump ECU
5 Radial-piston high-pressure pump
6 Distributor shaft
7 High-pressure solenoid valve
8 Delivery valve

When the delivery plungers are pushed inwards by the cams, the volume in the central plunger chamber between the delivery plungers is reduced. This compresses and pumps the fuel. Pressures of up to 1,200 bar are achievable at the pump.

Through passages in the distributor shaft, the fuel is directed at defined times to the appropriate outlet delivery valves (Fig. 1, Pos. 8 and Fig. 3, Pos. 5).

There may be 2, 3 or 4 delivery plungers depending on the number of cylinders in the engine and the type of application (Fig. 2). Sharing the delivery work between at least two plungers reduces the forces acting on the mechanical components and permits the use of steep cam profiles with good delivery rates. As a result, the radial-piston pump achieves a high level of hydraulic efficiency.

The direct transmission of force within the cam-ring drive gear minimizes the amount of "give", which also improves the hydraulic performance of the pump.

Delivery-plunger configurations for radial-piston high-pressure pumps (examples)

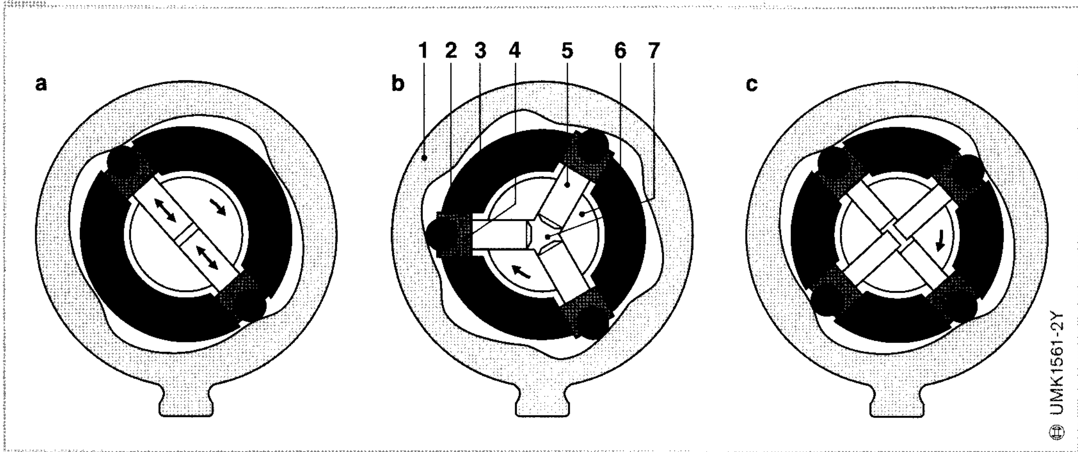


Fig. 2
- a For 4- or 6-cylinder engines
- b For 6-cylinder engines
- c For 4-cylinder engines

1. Cam ring
2. Roller
3. Guide slot in drive shaft
4. Roller support
5. Delivery plunger
6. Distributor shaft
7. Plunger chamber

High-pressure stage within a solenoid-valve controlled radial-piston distributor injection pump

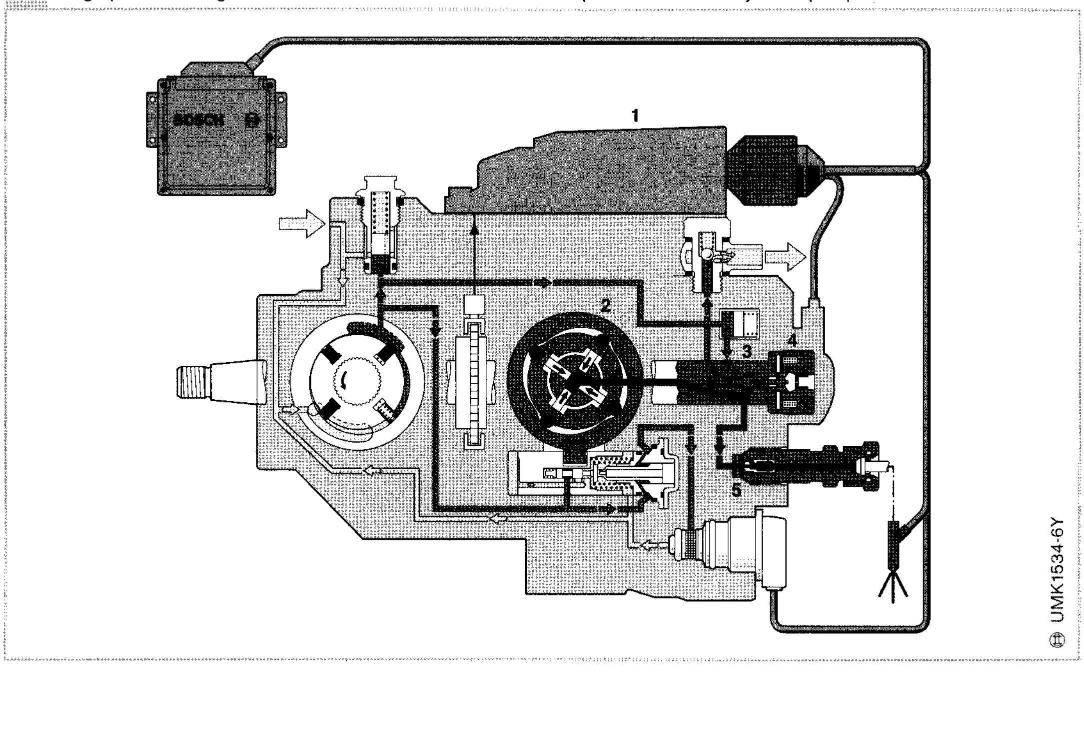


Fig. 3
For the sake of clarity, various components are shown end-on rather than side-on.

1. Pump ECU
2. Radial-piston high-pressure pump (end-on view)
3. Distributor shaft
4. High-pressure solenoid valve
5. Delivery valve

Distributor-body assembly

The distributor-body assembly (Fig. 4) consists of the following:

- The distributor body (2)
- The control sleeve (5) which is shrink-fitted in the distributor body
- The rear section of the distributor shaft (4) which runs in the control sleeve
- The valve needle (6) of the high-pressure solenoid valve
- The accumulator diaphragm (1) and
- The delivery valve (7) with orifice check valve

In contrast with the axial-piston distributor pump, the intake chamber that is pressurized with the delivery pressure of the vane-type supply pump consists only of the diaphragm chamber enclosed by an accumulator diaphragm (1). This produces higher pressures for supplying the high-pressure pump.

Delivery phases (method of operation)

Induction

During the induction phase (Fig. 5a), the delivery plungers (1) are forced outwards by the supply-pump pressure and centrifugal force. The high-pressure solenoid valve is open. Fuel flows from the diaphragm chamber (12) past the solenoid-valve needle (4) and through the low-pressure inlet (13) and the annular groove (10) to the plunger chamber (8). Excess fuel escapes via the return passage (5).

Effective stroke

The high-pressure solenoid valve (Fig. 5b, Pos. 7) is closed by a control pulse from the pump ECU when the cam profile is at bottom dead center. The plunger chamber is now sealed and fuel delivery starts as soon as the cams start to move the pistons inwards (start of delivery).

Residual stroke

Once the desired injected-fuel quantity has been delivered, the ECU cuts off power supply to the solenoid coil. The high-pressure

Distributor-body module of a radial-piston distributor injection pump

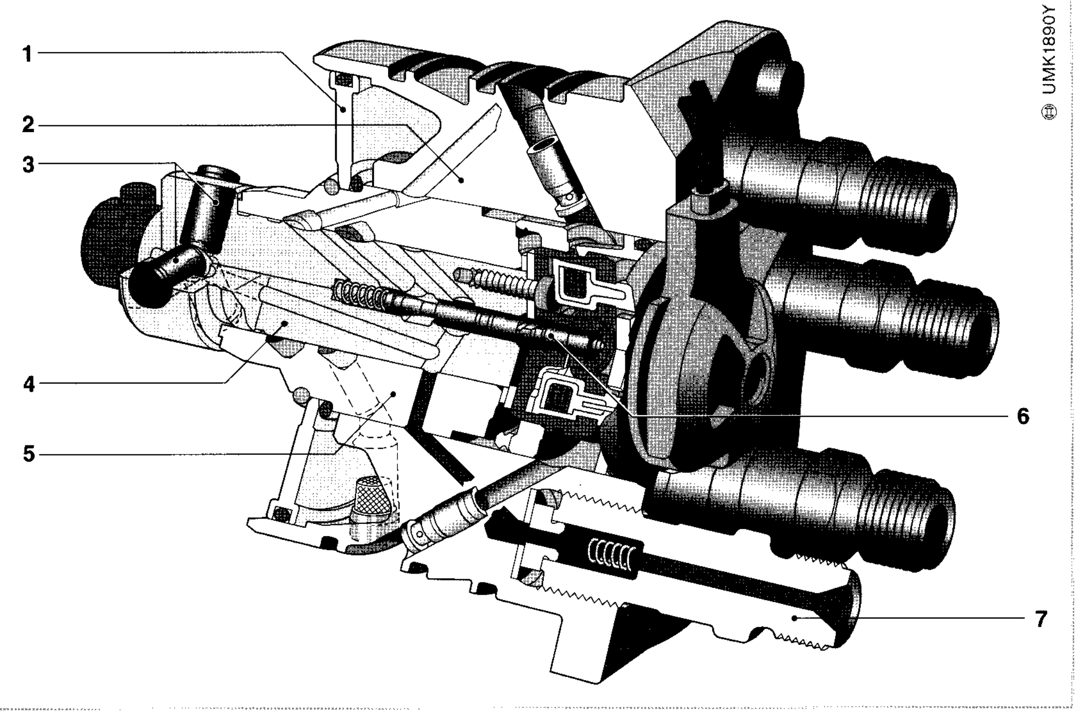


Fig. 4
1 Accumulator diaphragm
2 Distributor body
3 Delivery plunger
4 Distributor shaft
5 Control sleeve
6 Valve needle
7 Delivery valve

Induction and effective stroke of distributor head of radial-piston distributor injection pump

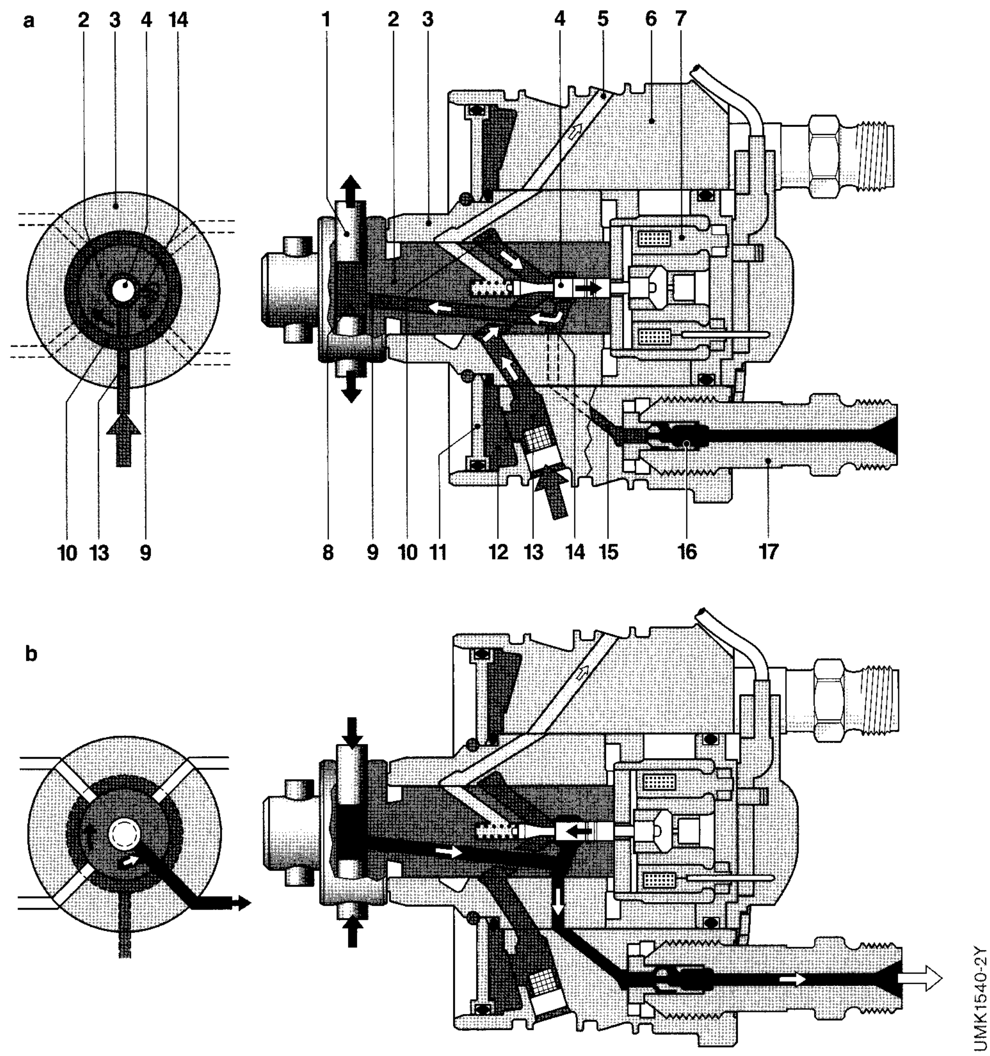


Fig. 5
For the sake of clarity, the pump plungers are shown end-on rather than side-on.

a Induction
b Effective stroke

1 Delivery plunger
2 Distributor shaft
3 Control sleeve
4 Valve needle
5 Fuel return
6 Distributor body
7 Solenoid coil
8 Plunger chamber
9 High-pressure fuel
10 Annular groove
11 Accumulator diaphragm
12 Diaphragm chamber
13 Low-pressure inlet
14 Distributor slot
15 High-pressure outlet
16 Orifice check valve
17 Delivery-valve body

solenoid valve opens again and pressure in the high-pressure stage collapses (end of delivery). The pressure drop closes the nozzle and the delivery valve again and the injection sequence comes to an end.

The excess fuel that is delivered by the pump while the pistons continue to move toward the cam top dead center is diverted back to the diaphragm chamber (12). The high pressure peaks that are thus produced in the low-pressure stage are damped by the accumulator diaphragm (11). In addition, the fuel stored in the diaphragm chamber helps to fill the plunger chamber for the next injection cycle.

Fuel metering takes place between the start of the cam lift and opening of the high-pressure solenoid valve. This phase is referred to as the delivery period. It determines the injected-fuel quantity in conjunction with the pump speed.

The high-pressure solenoid valve can completely shut off high-pressure fuel delivery in order to stop the engine. For this reason, an additional shutoff valve as used with port-controlled distributor injection pumps is not necessary.

Delivery valves

Between injection cycles, the delivery valve shuts off the high-pressure delivery line from the pump. This isolates the high-pressure delivery line from the outlet port in the distributor head. The residual pressure retained in the high-pressure delivery line ensures rapid and precise opening of the nozzle during the next injection cycle.

Integrated orifice check valve

The delivery valve with integral Type RSD orifice check valve (Figure 1) is a piston valve. At the start of the delivery sequence, the fuel pressure lifts the valve cone (3) away from the valve seat. The fuel then passes through the delivery-valve holder (5) to the high-pressure delivery line to the nozzle-and-holder assembly. At the end of the delivery sequence, the fuel pressure drops abruptly. The valve spring (4) and the pressure in the delivery line forces the valve cone back against the valve seat (1).

With the high pressures used for direct-injection engines, reflected pressure waves can occur at the end of the delivery lines. They can cause the nozzle to reopen, resulting in undesired dribble which adversely affects pollutant levels in the exhaust gas. In addition, areas of low pressure can be created and this can lead to cavitation and component damage.

A throttle bore (2) in the valve cone dampens the reflected pressure waves to a level at which they are no longer harmful. The throttle bore is designed so that the static pressure in the high-pressure delivery line is retained between injection cycles. Since the throttle bore means that the space between the fuel-injection pump and the nozzle is no longer hermetically sealed, this type of arrangement is referred to as an open system.

Separate Type RDV orifice check valve

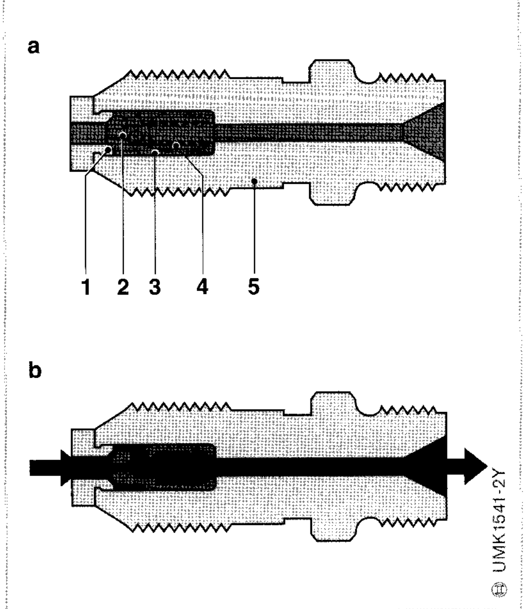
On axial-piston pumps and some types of radial-piston pump, a delivery valve with a separate orifice check valve is used (Fig. 2). This valve is also referred to as a Type GDV constant-pressure valve. It creates dynamic pressure in the delivery line. During the injection sequence, the valve plate (5) opens so that the throttle bore (6) has no effect. When fuel flows in the opposite direction, the valve plate closes and the throttle comes into action.

Fig. 1
a Valve closed
b Valve open

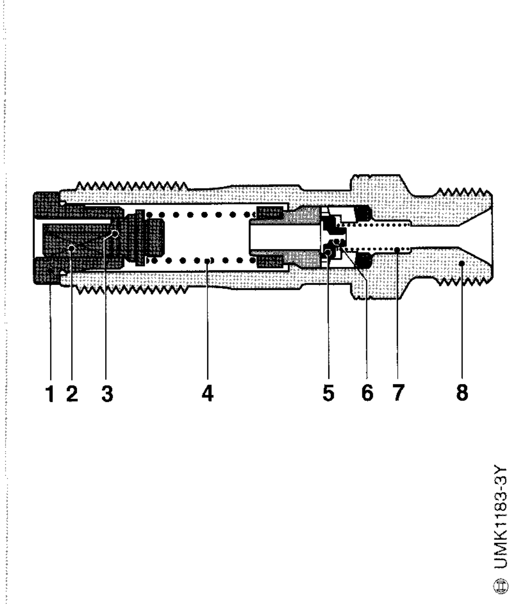
1 Valve seat
2 Throttle bore
3 Valve cone
4 Valve spring
5 Delivery-valve holder

Fig. 2
1 Valve holder
2 Pressure-valve stem
3 Retraction piston
4 Valve spring (delivery valve)
5 Valve plate
6 Throttle bore
7 Valve spring (valve plate)
8 Delivery-valve holder

Delivery valve with integrated orifice check valve



Delivery valve with separate orifice check valve



High-pressure solenoid valve

The high-pressure solenoid valve (Fig. 3) is a 2/2-way valve, i.e. it has two hydraulic ports and two possible settings. It is built into the distributor-body assembly. The valve needle (3) protrudes into the distributor shaft (5) and rotates in synchronization with it.

Arranging the solenoid coil (7) concentrically with the valve needle creates a compact combination of high-pressure solenoid valve and distributor body.

The high-pressure solenoid valve opens and closes in response to control signals from the pump ECU (valve-needle stroke 0.3...0.4 mm). The length of time it remains closed determines the delivery period of the high-pressure pump. This means that the fuel quantity can be very precisely metered for each individual cylinder.

The high-pressure solenoid valve has to meet the following criteria:

- Large valve cross-section for complete filling the plunger chamber, even at high speeds
- Light weight (small mass movements) in order to minimize component stress
- Rapid switching times for precise fuel metering and
- Generation of high magnetic forces in keeping with the loads encountered at high pressures

The high-pressure solenoid valve consists of:

- The valve body consisting of the housing (4), and attachments
- The valve needle (3) and solenoid armature (6)
- The solenoid plate and
- The electromagnet (7) with its electrical connection to the pump ECU (8)

The high-pressure solenoid valve is controlled by regulating the current. Steep current-signal edges must be used to achieve a high level of reproducibility on the part of the injected-fuel quantity. In addition, control must be designed in such a way as to minimize power loss in the ECU and the solenoid valve. This is achieved by keeping the control currents as low as possible, for example. Therefore, the solenoid valve reduces the current to the holding current (approx. 10 A) after the pickup current (approx. 18 A) has been applied.

The pump control unit can detect when the solenoid-valve needle meets the valve seat by means of the current pattern (BIP signal; Beginning of Injection Period). This allows the exact point at which fuel delivery starts to be calculated and the start of injection to be controlled very precisely.

High-pressure solenoid valve

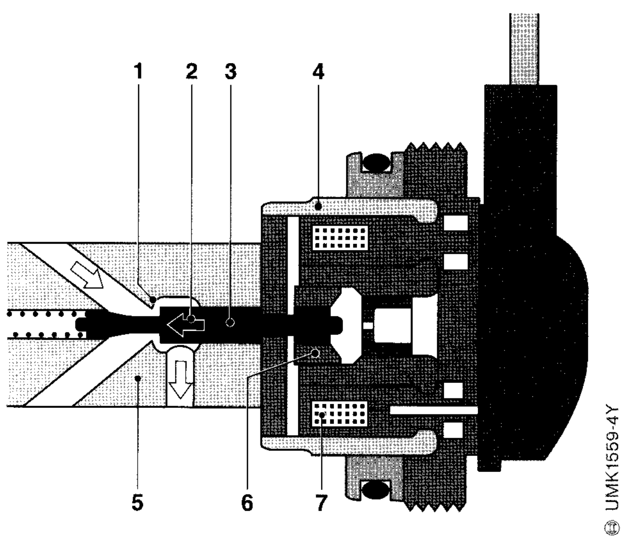


Fig. 3
1 Valve seat
2 Closing direction
3 Valve needle
4 Housing
5 Distributor shaft
6 Solenoid armature
7 Solenoid coil
8 Electrical connection

Injection timing adjustment

Purpose

The point at which combustion commences relative to the position of the piston/crankshaft has a decisive impact on engine performance, exhaust emission levels and noise output. If the start of delivery remains constant without adjusting the injection timing as engine speed increases, the amount of crankshaft rotation between the start of delivery and the start of combustion would increase to such an extent that combustion would no longer take place at the correct time.

As with port-controlled distributor injection pumps, the timing device rotates the roller ring/cam ring so that the start of delivery occurs earlier or later relative to the position of the engine crankshaft. The interaction between the high-pressure solenoid valve and the timing device thus varies the start of injection and the injection pattern to suit the operating status of the engine.

Explanation of terms

For a proper understanding of injection timing adjustment, a number of basic terms require explanation.

Pressure graph of operating cycle at full power and high speed (not to scale)

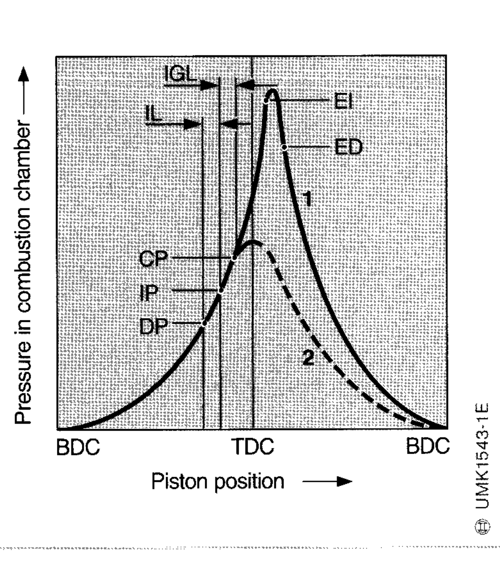


Fig. 1
1 Combustion pressure
2 Compression pressure

SD Start of delivery
TDC Engine-piston top dead center
SI Start of injection
EI End of injection
IL Injection lag
BDC Engine-piston bottom dead center
SC Start of combustion
EC End of combustion
IGL Ignition lag

Injection lag

The start of delivery (SD, Fig. 1) occurs after the point at which the high-pressure solenoid valve closes. High pressure is generated inside the fuel line. The point at which that pressure reaches the nozzle opening pressure and opens the nozzle is the start of injection (SI). The time between the start of delivery and the start of injection is called the injection lag (IL).

The injection lag is largely independent of pump/engine speed. It is essentially determined by the propagation of the pressure wave along the high-pressure delivery line. The propagation time of the pressure wave is determined by the length of the delivery line and the speed of sound. In diesel fuel, the speed of sound is approx. 1,500 m/s.

If the engine speed increases, the amount of degrees of crankshaft rotation during the injection lag also increases. As a consequence, the nozzle opens later (relative to the position of the engine piston). This is undesirable. For this reason, the start of delivery must be advanced as engine speed increases.

Ignition lag

The diesel fuel requires a certain amount of time after the start of injection to form a combustible mixture with the air and to ignite. The length of time required from the start of injection to the combustion start (SC) is the ignition lag (IGL). This, too, is independent of engine speed and is affected by the following variables:

- The ignition quality of the diesel fuel (indicated by the cetane number)
- The compression ratio of the engine
- The temperature in the combustion chamber
- The degree of fuel atomization and
- The exhaust-gas recirculation rate

The ignition lag is in the range of 2...9° of crankshaft rotation.

End of injection
When the high-pressure solenoid valve opens again, the high fuel pressure is released (end of injection, EI) and the nozzle closes. This is followed by the end of combustion (EC).

Other impacts
In order to limit pollutant emissions, the start of injection also has to be varied in response to engine load and temperature. The engine ECU calculates the required start of injection for each set of circumstances.

Design and method of operation

The timing device "advances" the position of the cams in the high-pressure pump relative to the diesel-engine crankshaft position as engine speed increases. This advances the start of injection. This compensates for the timing shift resulting from the injection lag and ignition lag. The impacts of engine load and temperature are also taken into account.

The injection timing adjustment is made up of the timing device itself, a timing-device solenoid valve and an angle-of-rotation sensor. Two types are used:

- The hydraulic timing device for axial-piston pumps and
- The hydraulically assisted timing device for radial-piston pumps

Hydraulic timing device
The hydraulic timing device (Fig. 2) is used on Type VP29 and VP30 axial-piston distributor injection pumps. Its design is the same as the version used on the electronically modulated Type VE..EDC port-controlled distributor injection pump.

The hydraulic timing device with its timing-device solenoid valve (5) and timing-device piston (3), which is located transversely to the pump axis, is positioned on the underside of the fuel-injection pump. The timing-device piston rotates the roller ring (1) according to load conditions and speed, so as to adjust the position of the rollers according to the required start of delivery.

As with the mechanical timing device, the pump intake-chamber pressure, which is proportional to pump speed, acts on the timing-device piston. That pressure on the

Timing device of an axial-piston distributor pump

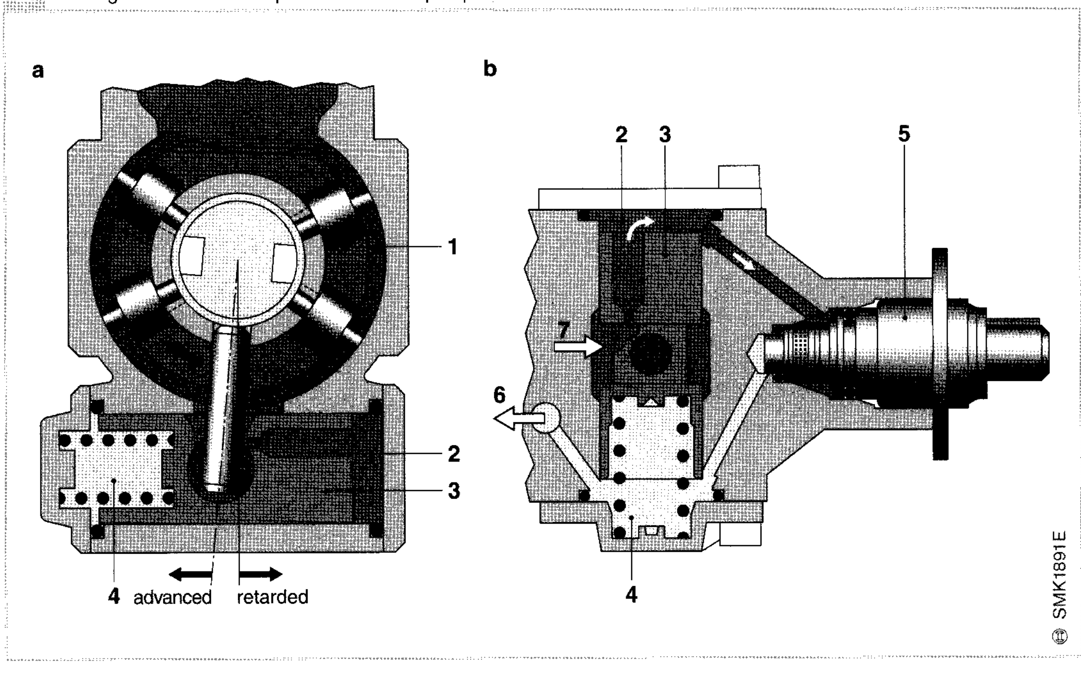


Fig. 2
a Front view
b Top view

1 Roller ring
2 Pump intake-chamber pressure
3 Timing-device piston
4 Pressure controlled by solenoid valve
5 Timing-device solenoid valve
6 Fuel return
7 Fuel inlet from pump intake chamber

pressure side of the timing device is regulated by the timing-device solenoid valve.

When the solenoid valve is open (reducing the pressure), the roller ring moves in the "retard" direction; when the valve is fully closed (increasing the pressure), it moves in the "advance" direction.

In between those two extremes, the solenoid valve can be "cycled", i.e. opened and closed in rapid succession by a pulse-width modulation signal (PWM signal) from the pump control unit. This is a signal with a constant voltage and frequency in which the ratio of "on" time to "off" time is varied. The ratio between the "on" time and the "off" time determines the pressure acting on the adjuster piston so that it can be held in any position.

Hydraulically assisted timing device

The hydraulically assisted timing device is used for radial-piston distributor injection pumps. It can produce greater adjustment forces. This is necessary to securely brace the cam ring with the greater drive power of the radial-piston pump. This type of timing device responds very quickly and regardless of the friction acting on the cam ring and the adjuster piston.

Hydraulically assisted timing device

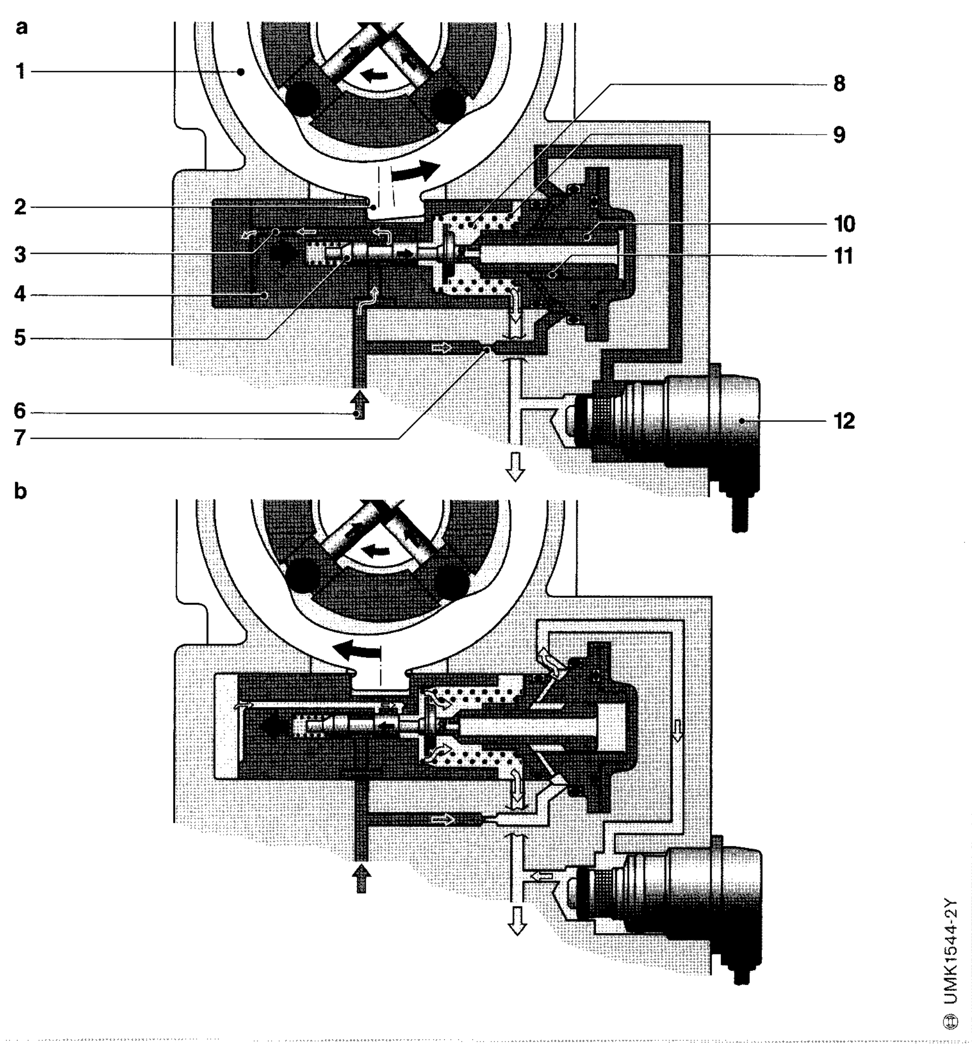


Fig. 3
a Advance setting
b Retard setting

1 Cam ring
2 Ball pivot
3 Inlet channel/ outlet channel
4 Timing-device piston
5 Control collar
6 Inlet from vane-type supply pump
7 Throttle
8 Control-plunger spring
9 Return spring
10 Control plunger
11 Annular chamber of hydraulic stop
12 Timing-device solenoid valve

The hydraulically assisted timing device, like the hydraulic timing device, is positioned on the underside of the injection pump (Fig. 3). It is also referred to by the Bosch type designation NLK.

The cam ring (1) engages in a cross-slot in the adjuster piston (4) by means of an adjuster lug (2) so that the axial movement of the adjuster piston causes the cam ring to rotate. In the center of the adjuster piston is a control sleeve (5) which opens and closes the control ports in the adjuster piston. In axial alignment with the adjuster piston is a spring-loaded hydraulic control piston (10) which defines the required position for the control sleeve.

At right-angles to the adjuster piston is the injection-timing solenoid valve (Pos. 12, shown schematically in Fig. 3 in the same plane as the timing device). Under the control of the pump control unit, the solenoid valve modulates the pressure acting on the control piston.

Advancing injection
When at rest, the adjuster piston (4) is held in the "retarded" position by a return spring (9). When in operation, the fuel supply pump pressure is regulated according to pump speed by means of the pressure control valve. That fuel pressure acts as the control pressure on the annular chamber (11) of the hydraulic stop via a restrictor bore (7) and when the solenoid valve (12) is closed moves the control piston (10) against the force of the control-piston spring (8) towards an "advanced" position (to the right in Fig. 3). As a result, the control sleeve (5) also moves in the "advance" direction so that the inlet channel (3) opens the way to the space behind the adjuster piston. Fuel can then flow through that channel and force the adjuster piston to the right in the "advance" direction.

The rotation of the cam ring relative to the resulting pump drive shaft causes the rollers to meet the cams sooner in the advanced position, thus advancing the start of injection. The degree of advance possible can be as much as 20° of camshaft rotation. On four-stroke engines, this corresponds to 40° of crankshaft rotation.

Retarding injection
The timing-device solenoid valve (12) opens when it receives the relevant PWM signal from the pump ECU. As a result, the control pressure in the annular chamber of the hydraulic stop (11) drops. The control plunger (10) moves in the "retard" direction (to the left in Figure 3) by the action of the control-plunger spring (8).

The timing-device piston (4) remains stationary to begin with. Only when the control collar (5) opens the control bore to the outlet channel can the fuel escape from the space behind the timing-device piston. The force of the return spring (9) and the reactive torque on the cam ring then force the timing-device piston back in the "retard" direction and to its initial position.

Regulating the control pressure
The timing-device solenoid valve acts as a variable throttle. It can vary continuously the control pressure so that the control plunger can assume any position between the fully advanced and fully retarded positions. The hydraulically assisted timing device is more precise in this regard than the straightforward hydraulic timing device.

If, for example, the control plunger is to move more in the "advance" direction, the on/off ratio of the PWM signal from the pump ECU is altered so that the valve closes more (low ratio of "on" time to "off" time). Less fuel escapes through the timing-device solenoid valve and the control plunger moves to a more "advanced" position.

Timing-device solenoid valve

The timing-device solenoid valve (Fig. 4) is the same in terms of fitting and method of control as the one used on the port-controlled type VE..EDC electronically regulated distributor injection pump. It is also referred to as the Type DMV10 diesel solenoid valve.

The pump ECU controls the solenoid coil (6) by means of a PWM signal. The solenoid armature (5) is drawn back against the force of the valve spring (7) while the solenoid valve is switched on. The valve needle (3) connected to the solenoid armature opens the valve. The longer the "on" times of the PWM signal are, the longer the solenoid valve remains open. The on/off ratio of the signal thus determines the flow rate through the valve.

In order to avoid problems caused by resonance effects, the otherwise fixed timing frequency of the PWM signal (Fig. 5) does not remain constant over the entire speed range. It changes over to a different frequency (30...70 Hz) in specific speed bands.

Incremental angle-time system with angle-of-rotation sensor

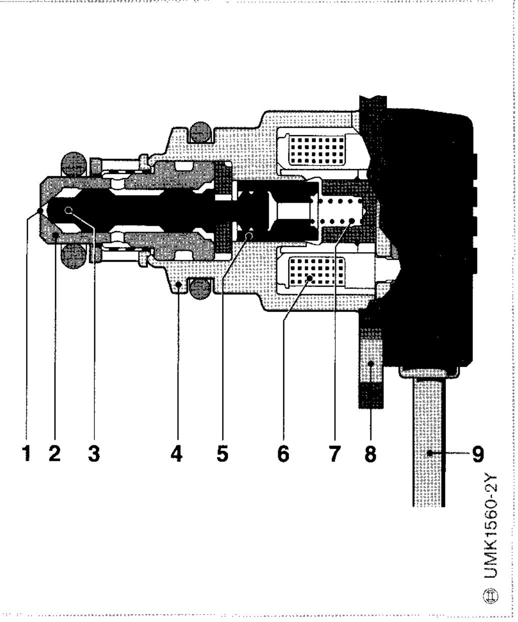
The closed-loop position control of the timing device uses as its input variables the signal from the crankshaft speed sensor and the pump's internal incremental angle-time system signal from the angle-of-rotation sensor.

The incremental angle-time system is located in the fuel-injection pump between the vane-type supply pump and the roller ring/cam ring (Fig. 6). Its purpose is to measure the angular position of the engine camshaft and the roller ring/cam ring relative to one another. This information is used to calculate the current timing device setting. In addition, the angle-of-rotation sensor (2) supplies an accurate speed signal.

Design and method of operation

Attached to the pump drive shaft is the increment ring (4). This has a fixed number of 120 teeth (i.e. one tooth for every 3°). In addition, there are reference tooth spaces (3) according to the number of cylinders in the engine. The increment ring is also called the angle-sensor ring or the sensor ring.

Timing-device solenoid valve



PWM signals

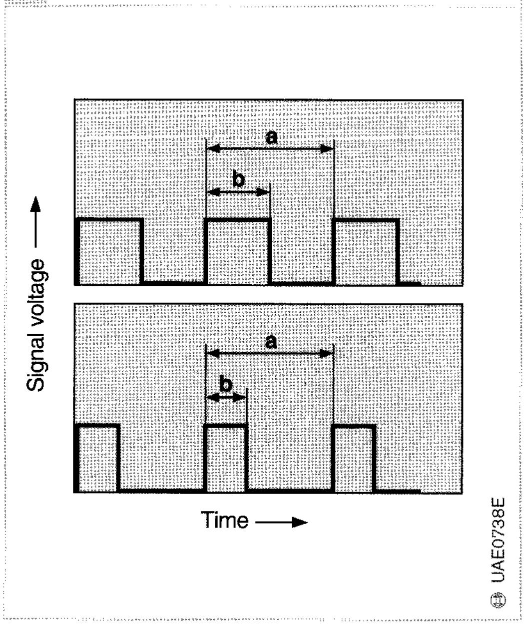


Fig. 4
1 Throttle bore
2 Valve body
3 Valve needle
4 Valve housing
5 Solenoid armature
6 Solenoid coil
7 Valve spring
8 Mounting flange
9 Electrical connection

Fig. 5
a Fixed timing frequency
b Variable switch-on time

The speed sensor is connected to the bearing ring (5) of the timing device. As the increment ring rotates, the sensor produces an electrical signal by means of magnetically controllable semiconductor resistors and relative to the number of teeth that pass by the pickup. If the position of the timing device changes, the sensor moves along with the roller ring/cam ring. Consequently, the positions of the reference tooth spaces in the increment ring alter relative to the TDC signal from the crankshaft speed sensor.

The angular separation between the reference tooth spaces (or the synchronization signal produced by the tooth spaces) and the TDC signal is continuously detected by the pump ECU and compared with the stored reference figure. The difference between the two signals represents the actual position of the timing device.

If even greater accuracy in determining the start of injection is required, the start-of-delivery control system can be supplemented by a start-of-injection control system using a needle-motion sensor.

Start-of-injection control system

The start of delivery and start of injection are directly related to one another. This relationship is stored in the "wave-propagation time map" in the engine ECU. The engine ECU uses this data to calculate a start-of-injection setpoint according to the operating status of the engine (load, speed, temperature). It then sends the information to the pump ECU. The pump ECU calculates the necessary control signals for the high-pressure solenoid valve and the setpoint position for the timing-device piston.

The timing-device controller in the pump ECU continuously compares the actual position of the timing-device piston with the setpoint specified by the engine ECU and, if there is a difference, alters the on/off ratio of the signal which controls the timing-device solenoid valve. Information as to the actual start-of-injection setting is provided by the signal from an angle-of-rotation sensor or alternatively, from a needle-motion sensor in the nozzle. This variable control method is referred to as "electronic" injection timing adjustment.

The benefits of a start-of-delivery control system are its rapid response characteristics, since all cylinders are taken into account. Another benefit is that it also functions when the engine is overrunning, i.e. when no fuel is injected. It means that the timing device can be preset for the next injection sequence.

Fitted position of angle-of-rotation sensor

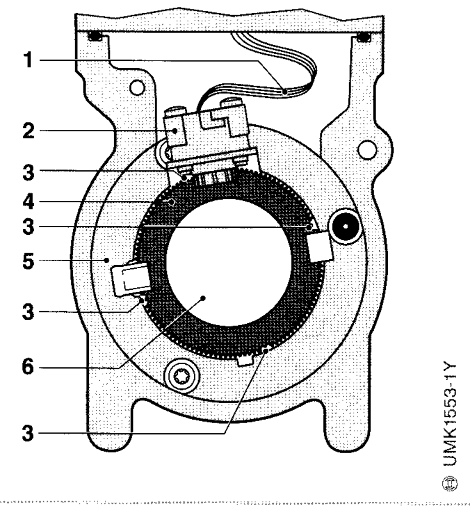


Fig. 6
1 Flexible conductive foil to ECU
2 Type DWS angle-of-rotation sensor
3 Tooth space
4 Increment ring
5 Bearing ring (connected to roller ring/cam ring)
6 Pump drive shaft

Control by increment angle-time system signal

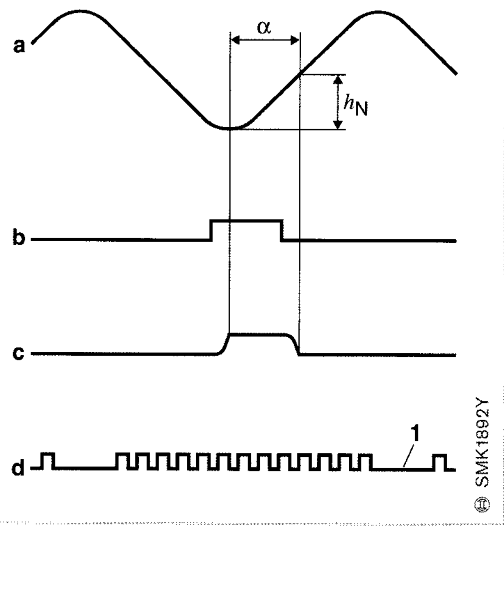


Fig. 7
a Cam pitch
b Control pulse for high-pressure solenoid valve
c Valve lift of high-pressure solenoid valve
d Signal of angle-of-rotation sensor

1 Tooth space

h_N Effective stroke
α Delivery angle

Electronic control unit

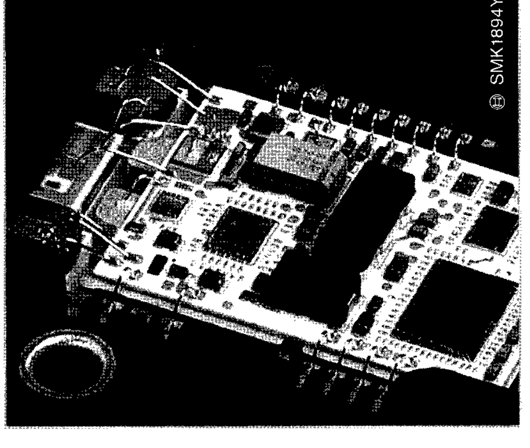
Requirements

Screwed to the top of the fuel-injection pump and identifiable by its cooling fins is the pump ECU or combined pump/engine ECU. It is an LTCC (Low Temperature Cofired Ceramic) hybrid design. This gives it the capability to withstand temperatures of up to 125°C and vibration levels up to 100 g (acceleration due to gravity). These ECUs are available in 12-volt or 24-volt versions.

In addition to withstanding the external conditions in the engine compartment, the pump ECU has to perform the following tasks:

- Exchange data with the separate engine ECU via a serial bus system
- Analyze the increment angle-time system signals
- Control the high-pressure solenoid valve
- Control the timing-device solenoid valve and
- Detect fuel temperature with the aid of an integrated temperature sensor in order to take fuel density into account when calculating injected-fuel quantity

LTCC circuit board of a pump ECU



The control of fuel injection must operate very precisely so that the fuel-injection pump delivers precisely and consistently the required quantity of fuel at the required time in all engine operating conditions. Even minute discrepancies in start of injection and injection duration have a negative effect on the smooth running and noise levels in the diesel engine as well as its pollutant emissions.

The fuel-injection system also has to respond very quickly to changes. For this reason, the calculations taking place in the microcontroller and conversion of the control signals in the output stages are performed in real time (approx. 50 µs). In the case of a fuel-injection pump for a six-cylinder engine, the fuel-injection data is calculated up to 13,000 times a minute.

A pre-injection (PI) phase controlled by the high-pressure solenoid valve is also a viable option. This involves the injection of 1...2 mm^3 of fuel before the main injection (MI) phase. This produces a more gradual increase in combustion pressure and therefore reduces combustion noise.

During the pre-injection phase, the fuel-quantity solenoid valve is operated ballistically, i.e. it is only partially opened. Consequently, it can be closed again more quickly. This keeps the injection gap as short as possible so that even at high speeds, there is sufficient cam lift remaining for the main injection phase. The entire injection sequence lasts approx. 1...2 ms.

The timing of the individual control phases is calculated by the microcontroller in the pump ECU. It also makes use of stored data maps. The maps contain settings for the specific vehicle application and certain engine characteristics along with data for checking the plausibility of the signals received. They also form the basis for determining various calculated variables.

In order to achieve specific and rapid opening of the solenoid valve, "fast extinction" of the energy stored in the solenoid valve takes place combined with a high extinction potential.

Two-ECU concept
Separate ECUs are used in diesel fuel-injection systems with solenoid-valve controlled axial-piston distributor injection pumps and first-generation radial-piston distributor injection pumps. These systems have a Type MSG engine ECU in the engine compartment and a Type PSG pump ECU mounted directly on the fuel-injection pump. There are two reasons for this division of functions: Firstly, it prevents the overheating of certain electronic components by removing them from the immediate vicinity of pump and engine. Secondly, it allows the use of short control leads for the high-pressure solenoid valve, so eliminating interference that may occur as a result of the very high currents (up to 20 A).

The pump ECU detects and analyzes the pump's internal sensor signals for angle of rotation and fuel temperature in order to adjust the start of injection. On the other hand, the engine ECU processes all engine and ambient data signals from external sensors and interfaces and uses them to calculate actuator adjustments on the fuel-injection pump.

The two control units communicate via a CAN interface.

Integrated engine and pump ECU on the fuel-injection pump
Increasing levels of integration using hybrid technology have made it possible to combine the engine-management control unit with the pump control unit on second-generation solenoid-valve controlled distributor injection pumps. The use of integrated ECUs allows a space-saving configuration. Other advantages are simpler installation and lower system costs due to fewer electrical interfaces.

The integrated engine/pump ECU is only used with radial-piston distributor injection pumps.

Summary

The overall system and the many modular assemblies are very similar on different types of solenoid-valve controlled distributor injection pump. Nevertheless, there are a number of differences. The main distinguishing features are detailed in Table 1.

Essential distinguishing features of solenoid-valve controlled distributor injection pumps

Type	VP29 (VE..MV)	VP30 (VE..MV)	VP44 (VRV)
Application	IDI engines	DI engines	DI engines
Maximum injection pressure at nozzle	800 bar	1,400 bar	1,950 bar
High-pressure pump	Axial-piston	Axial-piston	Radial-piston
Delivery valve	Separate orifice check valve	Integrated orifice check valve	Integrated orifice check valve
Timing device	Hydraulic	Hydraulic	Hydraulically assisted
Vane-type delivery pump	Circular retaining ring Vanes without springs	Circular retaining ring Vanes without springs	Profiled retaining ring Vanes with springs
Engine ECU and pump ECU	Separate	Separate	Integrated or separate

Table 1

Nozzles

The nozzle injects the fuel into the combustion chamber of the diesel engine. It is a determining factor in the efficiency of mixture formation and combustion and therefore has a fundamental effect on engine performance, exhaust-gas behavior and noise. In order that nozzles can perform their function as effectively as possible, they have to be designed to match the fuel-injection system and engine in which they are used.

The nozzle is a central component of any fuel-injection system. It requires highly specialized technical knowledge on the part of its designers. The nozzle plays a major role in

- shaping the rate-of-discharge curve (precise progression of pressure and fuel distribution relative to crankshaft rotation)
- optimum atomization and distribution of fuel in the combustion chamber and
- sealing off the fuel-injection system from the combustion chamber

Because of its exposed position in the combustion chamber, the nozzle is subjected to constant pulsating mechanical and thermal stresses from the engine and the fuel-injection system. The fuel flowing through the nozzle must also cool it. When the engine is overrunning, when no fuel is being injected, the nozzle temperature increases steeply. Therefore, it must have sufficient high-temperature resistance to cope with these conditions.

In fuel-injection systems based on in-line injection pumps (Type PE) and distributor injection pumps (Type VE/VR), and in unit pump systems (UPS), the nozzle is combined with the nozzle holder to form the nozzle-and-holder assembly (Figure 1) and installed in the engine. In high-pressure fuel-injection systems such as the common rail (CR) and unit injector systems (UIS) the nozzle is a single integrated unit so that the nozzle holder is not required.

Indirect-injection (IDI) engines use pintle nozzles, while direct-injection engines have hole-type nozzles.

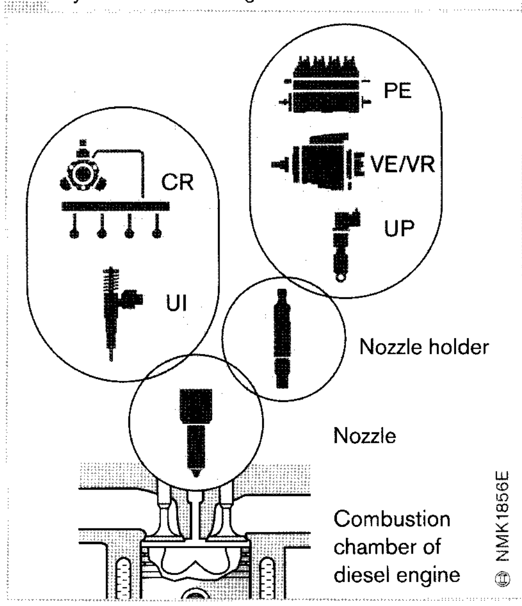
The nozzles are opened by the fuel pressure. The nozzle opening, injection duration and rate-of-discharge curve (injection pattern) are the essential determinants of injected fuel quantity. The nozzles must close rapidly and reliably when the fuel pressure drops. The closing pressure is at least 40 bar above the maximum combustion pressure in order to prevent unwanted post-injection or intrusion of combustion gases into the nozzle.

The nozzle must be designed specifically for the type of engine in which it is used as determined by

- the injection method (direct or indirect)
- the geometry of the combustion chamber
- the required injection-jet shape and direction
- the required penetration and atomization of the fuel jet
- the required injection duration and
- the required injected fuel quantity relative to crankshaft rotation

Standardized dimensions and combinations provide the required degree of adaptability combined with the minimum of component diversity. Because of the superior performance combined with lower fuel consumption that it offers, all new engine designs use direct injection (and therefore hole-type nozzles).

The nozzle as the interface between fuel-injection system and diesel engine



Dimensions of diesel fuel-injection technology

The world of diesel fuel injection is a world of superlatives.

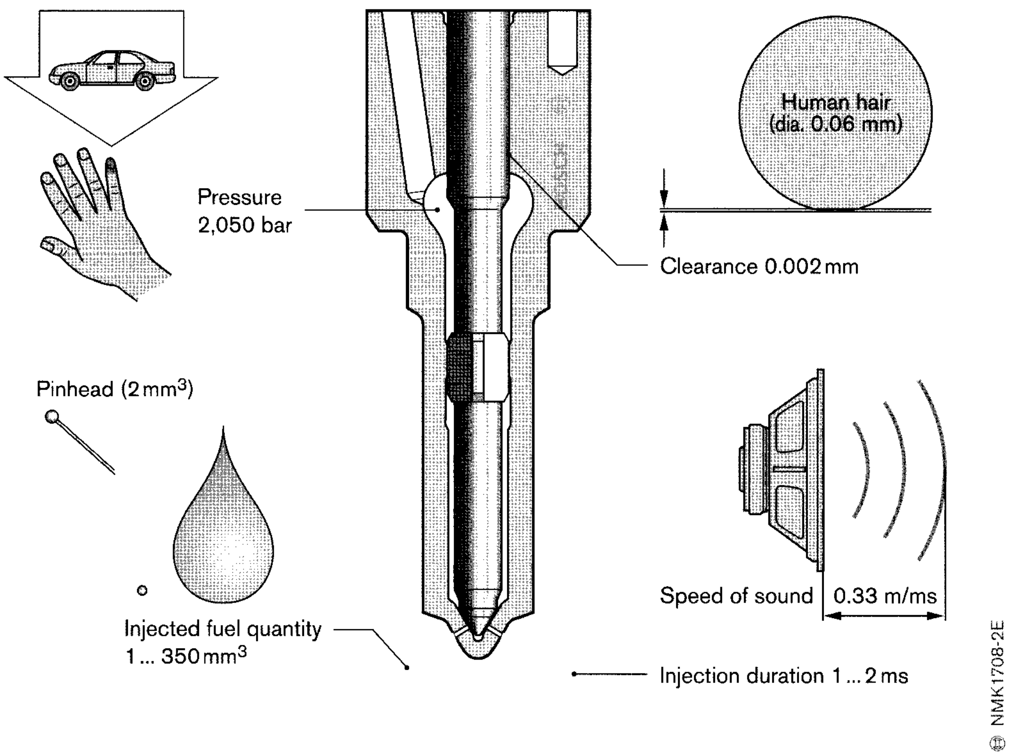
The valve needle of a commercial-vehicle nozzle will open and close the nozzle more than a billion times in the course of its service life. It provides a reliable seal at pressures as high as 2,050 bar as well as having to withstand many other stresses such as

- the shocks caused by rapid opening and closing (on cars this can take place as frequently as 10,000 times a minute if there are pre- and post-injection phases)
- the high flow-related stresses during fuel injection and
- the pressure and temperature of the combustion chamber

The facts and figures below illustrate what modern nozzles are capable of.

- The pressure in the fuel-injection chamber can be as high as 2,050 bar. That is equivalent to the pressure produced by the weight of a large executive car acting on an area the size of a fingernail.
- The injection duration is 1...2 milliseconds (ms). In one millisecond, the sound wave from a loudspeaker only travels about 33 cm.
- The injection durations on a car engine vary between 1 mm^3 (pre-injection) and 50 mm^3 (full-load delivery); on a commercial vehicle between 3 mm^3 (pre-injection) and 350 mm^3 (full-load delivery). 1 mm^3 is equivalent to half the size of a pinhead. 350 mm^3 is about the same as 12 large raindrops (30 mm^3 per raindrop). That amount of fuel is forced at a velocity of 2,000 km/h through an opening of less than 0.25 mm^2 in the space of only 2 ms.
- The valve-needle clearance is 0.002 mm (2 µm). A human hair is 30 times as thick (0.06 mm).

Such high-precision technology demands an enormous amount of expertise in development, materials, production and measurement techniques.



Pintle nozzles

Usage

Pintle nozzles are used on indirect injection (IDI) engines, i.e. engines that have prechambers or whirl chambers. In this type of engine, the mixing of fuel and air is achieved primarily by the whirl effects created inside the cylinder. The shape of the injection jet can also assist the process. Pintle nozzles are not suitable for direct-injection engines as the peak pressures inside the combustion chamber would open the nozzle. The following types of pintle nozzle are available:

- Standard pintle nozzles
- Throttling pintle nozzles and
- Flatted-pintle nozzles

Design and method of operation

The fundamental design of all pintle nozzles is virtually identical. The differences between them are to be found in the geometry of the pintle t(Figure 1, Item 7). Inside the nozzle body is the nozzle needle (3) It is pressed downwards by the force F_F exerted by the spring and the pressure pin in the nozzle holder so that it seals off the nozzle from the combustion chamber. As the pressure of the fuel in the pressure chamber (5) increases, it acts on the pressure shoulder (6) and forces the nozzle needle upwards (force F_D). The pintle lifts away from the injector orifice (8) and opens the way for fuel to pass through into the combustion chamber (the nozzle "opens"; opening pressure 110...170 bar). When the pressure drops, the nozzle closes again. Opening and closing of the nozzle is thus controlled by the pressure inside the nozzle.

Design variations

Standard pintle nozzle

The nozzle needle of (Figure 1, Item 3) of a standard pintle nozzle has a pintle (7) that fits into the injector orifice (8) of the nozzle with a small degree of play. By varying the dimensions and geometry of the of the pintle, the characteristics of the injection jet produced can be modified to suit the requirements of different engines.

Throttling pintle nozzle

One of the variations of the pintle nozzle is the throttling pintle nozzle. The profile of the pintle allows a specific rate-of-discharge curve to be produced. As the nozzle needle opens, at first only a very narrow annular orifice is provided which allows only a small amount of fuel to pass through (throttling effect).

As the pintle draws further back with increasing fuel pressure, the size of the gap through which fuel can flow increases. The greater proportion of the injected fuel quantity is only injected as the pintle approaches the limit of its upward travel. By modifying the rate-of-discharge curve in this way, "softer" combustion is produced because the pressure in the combustion chamber does not rise so quickly. As a result, combustion noise is reduced in the part-load range. This means that the shape of the pintle in combination with the throttling gap and the characteristic of the compression spring in the nozzle holder produces the desired rate-of-discharge curve.

Standard pintle nozzle

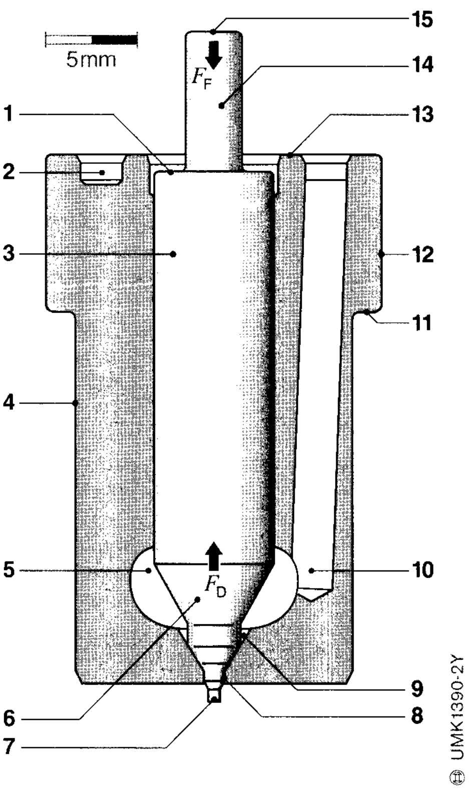


Fig. 1
1 Stroke-limiting shoulder
2 Ring groove
3 Nozzle needle
4 Nozzle body
5 Pressure chamber
6 Pintle shoulder
7 Pintle
8 Injection orifice
9 Seat lead-in
10 Inlet port
11 Nozzle-body shoulder
12 Nozzle-body collar
13 Sealing face
14 Pressure pin
15 Pressure-pin contact face

F_F Spring force
F_D Force acting on pressure shoulder due to fuel pressure

Flatted-pintle nozzle

The flatted-pintle nozzle (Figure 3) has a pintle with a flatted face on its tip which, as the nozzle opens (at the beginning of needle lift travel) produces a wider passage within the annular orifice. This helps to prevent deposits at that point by increasing the volumetric flow rate. As a result, flatted-pintle nozzles "coke" to a lesser degree and more evenly. The annular orifice between the jet orifice and the pintle is very narrow (< 10 µm). The flatted face is frequently parallel to the axis of the nozzle needle. By setting the flatted face at an angle, the volumetric flow rate, Q, can be increased in the flatter section of the rate-of-discharge curve (Figure 4). In this way, a smoother transition between the initial phase and the fully-open phase of the rate-of-discharge curve can be obtained. Specially designed variations in pintle geometry allow the flow-rate pattern to be modified to suit particular engine requirements. As a result, engine noise in the part-load range is reduced and engine smoothness improved.

Flatted-pintle nozzle

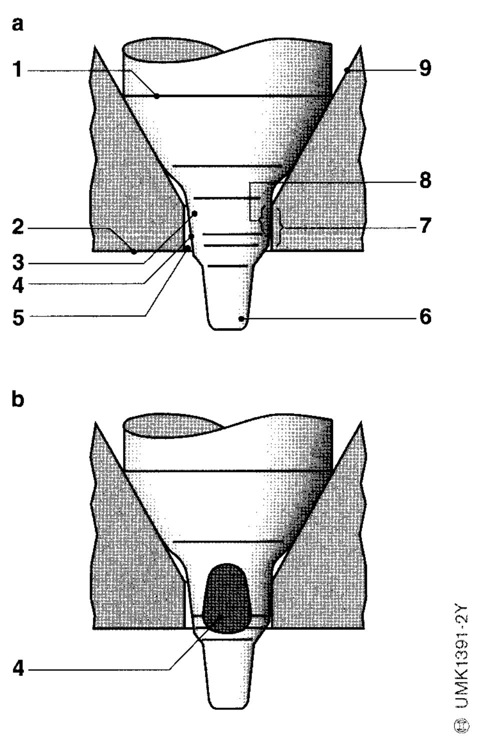


Fig. 3
a Side view
b Front view (rotation of 90° relative to side view)

1 Pintle seat face
2 Nozzle-body base
3 Throttling pintle
4 Flatted face
5 Injection orifice
6 Profiled pintle
7 Total contact ratio
8 Cylindrical overlap
9 Nozzle-body seat face

Heat shielding

Temperatures above 220 °C also promote nozzle coking. Thermal-protection plates or sleeves (Figure 2) help to overcome this problem by conducting heat from the combustion chamber into the cylinder head.

Thermal-protection sleeve

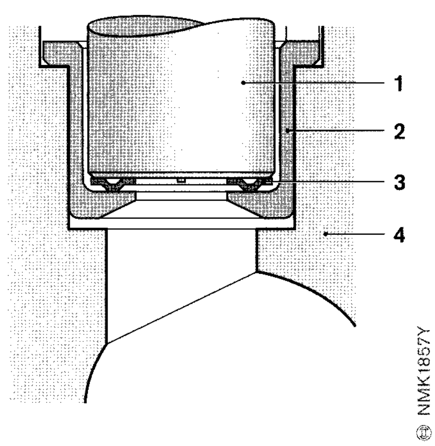


Fig. 2
1 Pintle nozzle
2 Thermal-protection sleeve
3 Protective disc
4 Cylinder head

Volumetric flow rate as a function of pintle travel and nozzle design

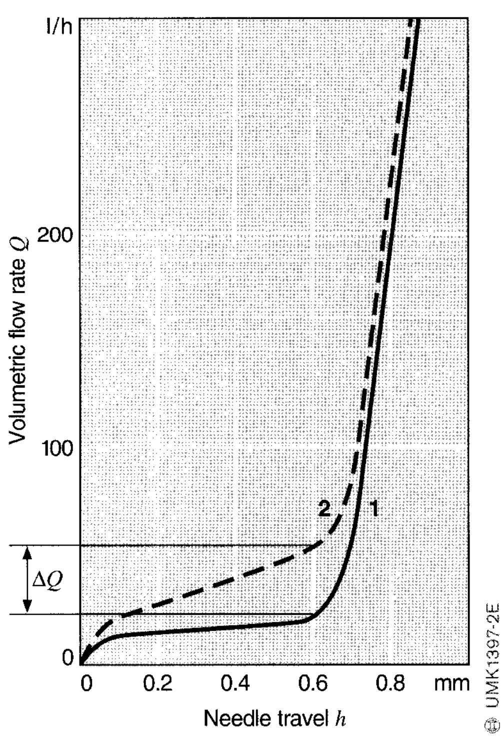


Fig. 4
1 Throttling pintle nozzle
2 Flatted-pintle nozzle (throttling pintle nozzle with flatted face)

ΔQ Difference in volumetric flow rate due to flatted face

Hole-type nozzles

Usage

Hole-type nozzles are used on direct-injection (DI) engines. The position in which the nozzles are fitted is generally determined by the engine design. The injector orifices are set at a variety of angles according to the requirements of the combustion chamber (Figure 1). Hole-type nozzles are subdivided into

- blind-hole nozzles and
- sac-less (vco) nozzles.

Hole-type nozzles are also divided according to size into

- *Type P* which have a needle diameter of 4 mm (blind-hole and sac-less (vco) nozzles), and
- *Type S* which have a needle diameter of 5 or 6 mm (blind-hole nozzles for large engines)

In the common rail (CR) and unit injector (UI) fuel-injection systems, the hole-type nozzle is a single integrated unit. It therefore combines the functions of nozzle and nozzle holder.

The opening pressure of hole-type nozzles is in the range 150...350 bar.

Position of hole-type nozzle in combustion chamber

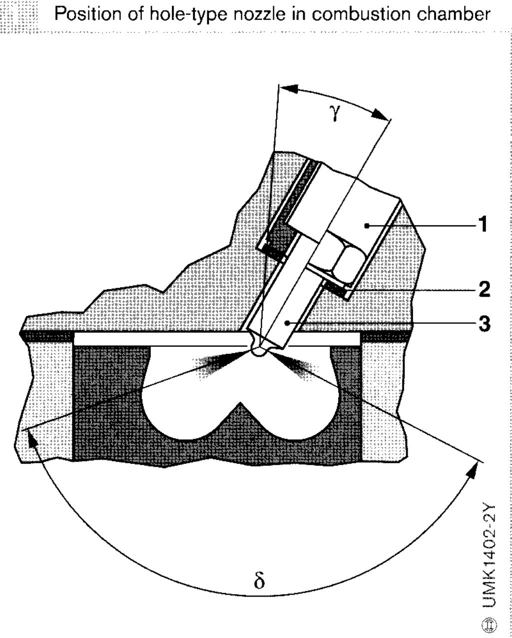


Fig. 1
1 Nozzle
2 Sealing washer
3 Hole-type nozzle

γ Inclination
δ Jet cone angle

Design

The injection orifices (Figure 2, Item 6) are positioned around the cladding of the nozzle cone (7). The number and size are dependent on

- the required injected fuel quantity
- the shape of the combustion chamber, and
- the air vortex (whirl) inside the combustion chamber

The bore of the injection orifices is slightly larger at the inner end than at the outer end. This difference is defined by the port taper factor. The leading edges of the injection orifices may be rounded by using the hydro-erosion (HE) process. This involves the use of an HE fluid that contains abrasive particles which smooth off the edges at points where high flow velocities occur (leading edges of injection orifices). Hydro-erosion can be used both on blind-hole and sac-less (vco) nozzles. Its purpose is to

- optimize the flow resistance coefficient
- pre-empt erosion of edges caused by particles in the fuel and/or
- tighten flow-rate tolerances

Nozzles have to be carefully designed to match the engine in which they are used. Nozzle design plays a decisive role in

- precise metering of injected fuel (injection duration and injected fuel quantity relative to degrees of crankshaft rotation)
- fuel conditioning (number of jets, spray shape and atomization of fuel)
- fuel dispersal inside the combustion chamber and
- sealing the fuel-injection system against the combustion chamber

The pressure chamber (10) is created by electrochemical machining (ECM). An electrode through which an electrolyte solution is passed is introduced into the pre-bored nozzle body. Material is then removed from the positively charged nozzle body (anodic dissolution).

Design variations

The fuel in the space below the seat of the nozzle needle evaporates after combustion and, therefore, contributes significantly to the hydrocarbon (HC) emissions produced by the engine. For this reason, it is important to keep that dead volume or "detrimental" volume as small as possible.

In addition, the geometry of the needle seat and the shape of the nozzle cone have a decisive influence on the opening and closing characteristics of the nozzle. This in turn affects the soot and NO_X emissions produced by the engine.

The consideration of these various factors in combination with the demands of the engine and the fuel-injection system has led to a variety of nozzle designs.

There are two basic types of injector:
- Blind-hole nozzles and
- Sac-less (vco) nozzles

Among the blind-hole nozzles, there are a number of variations.

Blind-hole nozzles

On a blind-hole nozzle (Figure 2, Item 6) the injection orifices exit from a blind hole in the tip of the nozzle.

If the nozzle has a *cone*, the injection orifices are drilled either mechanically or by electro-erosion depending on design.

In blind-hole nozzles with a *conical tip*, the injection orifices are generally created by electro-erosion.

Blind-hole nozzles may have a cylindrical or conical blind hole of varying dimensions.

Blind-hole nozzles with a cylindrical blind hole and conical tip (Figure 3), which consists of a cylindrical and a hemispherical section, offer a large amount of scope with regard to the number of holes, length of injection orifices and orifice taper angle. The nozzle cone is hemispherical in shape, which – in combination with the shape of the blind hole – ensures that all the spray holes are of equal length.

Blind-hole nozzle

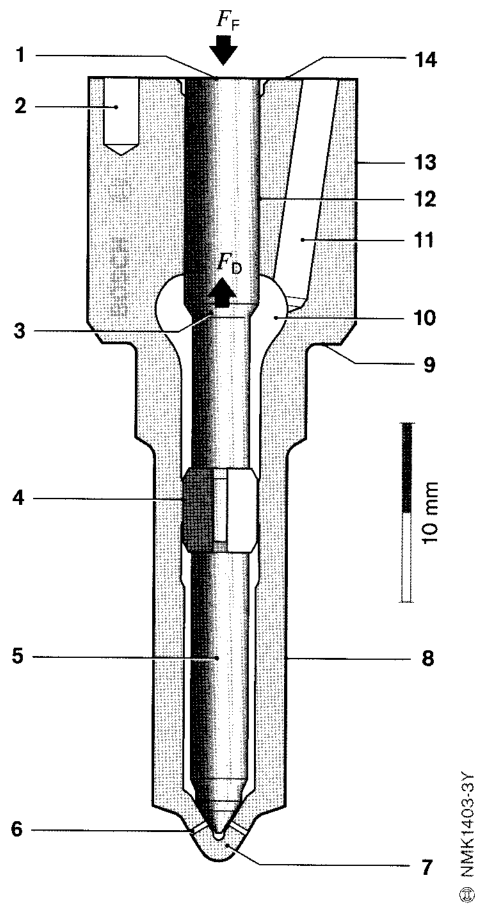


Fig. 2
1 Stroke-limiting shoulder
2 Fixing hole
3 Pressure shoulder
4 Secondary needle guide
5 Needle shaft
6 Injection orifice
7 Nozzle cone
8 Nozzle body
9 Nozzle-body shoulder
10 Pressure chamber
11 Inlet passage
12 Needle guide
13 Nozzle-body collar
14 Sealing face

F_F Spring force
F_D Force acting on pressure shoulder due to fuel pressure

Features of a nozzle with cylindrical blind hole and conical tip

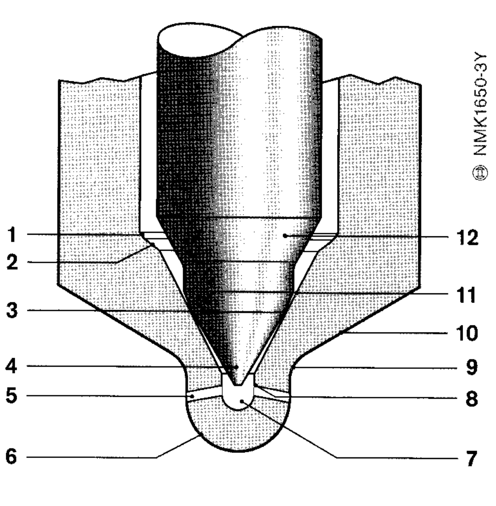


Fig. 3
1 Shoulder
2 Seat lead-in
3 Needle-seat face
4 Needle tip
5 Injection orifice
6 Conical nozzle tip
7 Cylindrical blind hole (dead volume)
8 Injection orifice leading edge
9 Neck radius
10 Nozzle-cone taper
11 Nozzle-body seat face
12 Damping taper

Blind-hole nozzles with cylindrical blind holes and conical tip (Figure 4a) are produced only with a spray-hole length of 0.6 mm. The conical-shaped tip increases the strength of the cone by virtue of the greater wall thickness between the neck radius (3) and the nozzle-body seat (4).

Nozzle cones

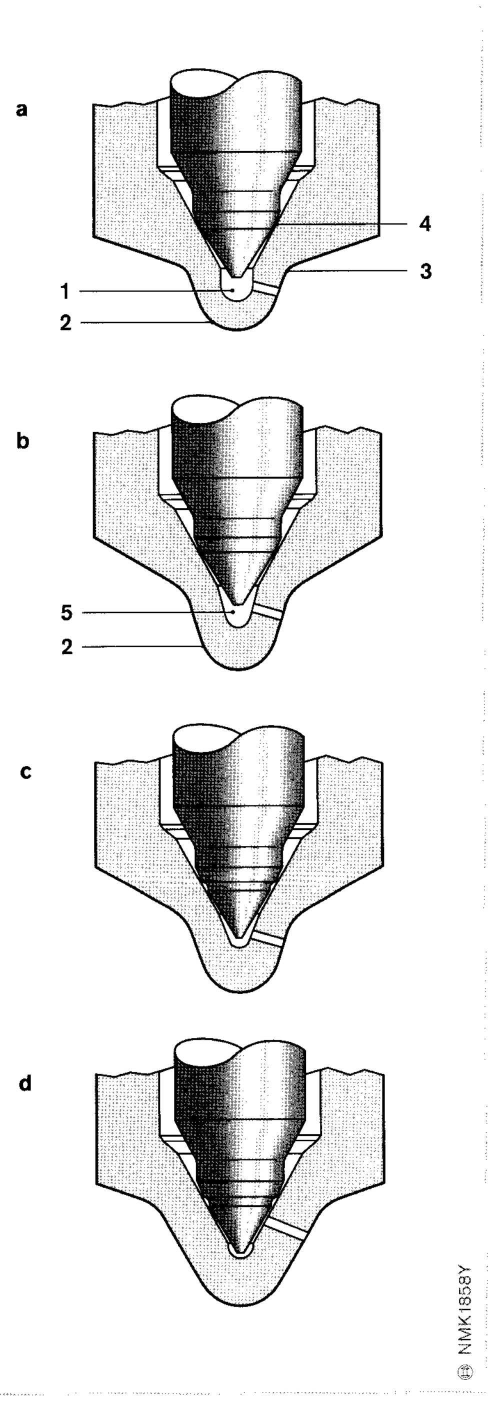


Fig. 4
- a Cylindrical blind hole and conical tip
- a Conical blind hole and conical tip
- c Micro-blind-hole
- d Sac-less (vco) nozzle

1. Cylindrical blind hole
2. Conical nozzle tip
3. Neck radius
4. Nozzle-body seat face
5. Conical blind hole

Blind-hole nozzles with conical blind holes and conical tip (Figure 4b) have a smaller dead volume than nozzles with a cylindrical blind hole. The volume of the blind hole is between that of a sac-less (vco) nozzle and a blind-hole nozzle with a cylindrical blind hole. In order to obtain an even wall thickness throughout the cone, it is shaped conically to match the shape of the blind hole.

A further refinement of the blind-hole nozzle is the *micro-blind-hole nozzle* (Figure 4c). Its blind-hole volume is around 30 % smaller than that of a conventional blind-hole nozzle. This type of nozzle is particularly suited to use in common-rail fuel-injection systems, which operate with a relatively slow needle lift and consequently a comparatively long nozzle-seat restriction. The micro-blind-hole nozzle currently represents the best compromise between minimizing dead volume and even spray dispersal when the nozzle opens for common-rail systems.

Sac-less (vco) nozzles

In order to minimize the dead volume – and therefore the HC emissions – the injection orifice exits from the nozzle-body seat face. When the nozzle is closed, the nozzle needle more or less covers the injection orifice so that there is no direct connection between the blind hole and the combustion chamber (Figure 4d). The blind-hole volume is considerably smaller than that of a blind-hole nozzle. Sac-less (vco) nozzles have a significantly lower stress capacity than blind-hole nozzles and can therefore only be produced with a spray-hole length of 1 mm. The nozzle tip has a conical shape. The injection orifices are generally produced by electro-erosion.

Special spray-hole geometries, secondary needle guides and complex needle-tip geometries are used to further improve spray dispersal, and consequently mixture formation, on both blind-hole and sac-less (vco) nozzles.

Heat shielding

The maximum temperature capacity of hole-type nozzles is around 300 °C (heat resistance of material). Thermal-protection sleeves are available for operation in especially difficult conditions, and there are even cooled nozzles for large-scale engines.

Effect on emissions

Nozzle geometry has a direct effect on the engine's exhaust-gas emission characteristics.

- The *spray-hole geometry* (Figure 5, Pos. 1) affects particulate and NO_X emissions.
- The *needle-seat geometry* (2) affects engine noise due to its effect on the pilot volume, i.e. the volume injected at the beginning of the injection process. The aim of optimizing spray-hole and seat geometry is to produce a durable nozzle capable of mass production to very tight dimensional tolerances.
- *Blind-hole geometry* (3) affects HC emissions, as previously mentioned. The designer can select and combine the various nozzle characteristics to obtain the optimum design for a particular engine and vehicle concerned.

For this reason, it is important that the nozzles are designed specifically for the vehicle, engine and fuel-injection system in which they are to be used. When servicing is required, it is equally important that genuine OEM parts are used in order to ensure that engine performance is not impaired and exhaust-gas emissions are not increased.

Spray shapes

Basically, the shape of the injection jet for car engines is long and narrow because these engines produce a large degree of swirl inside the combustion chamber. There is no swirl effect in commercial-vehicle engines. Therefore, the injection jet tends to be wider and shorter. Even where there is a large amount of swirl, the individual injection jets must not intermingle otherwise fuel would be injected into areas where combustion has already taken place and therefore where there is a lack of air. This would result in the production of large amounts of soot.

Hole-type nozzles have up to six injection orifices in cars and up to ten in commercials. The aim of future development will be to further increase the number of injection orifices and to reduce their bore size (< 0.12 mm) in order to obtain even finer dispersal of fuel.

Decisive areas of nozzle geometry

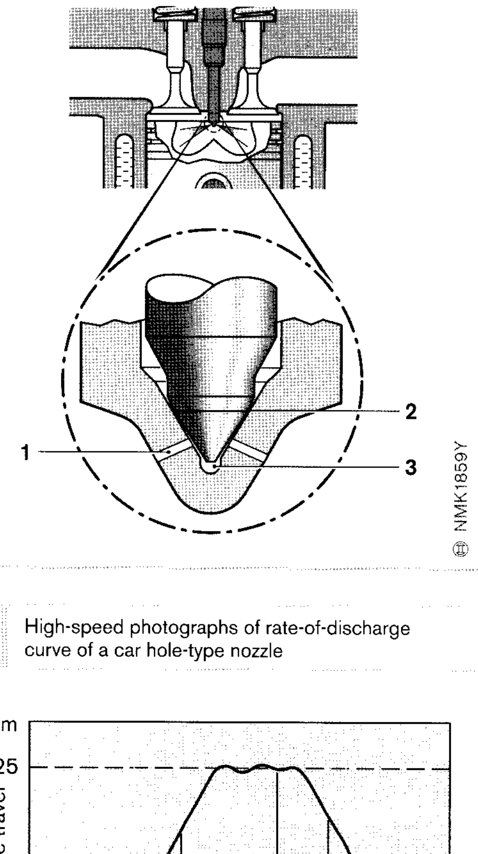
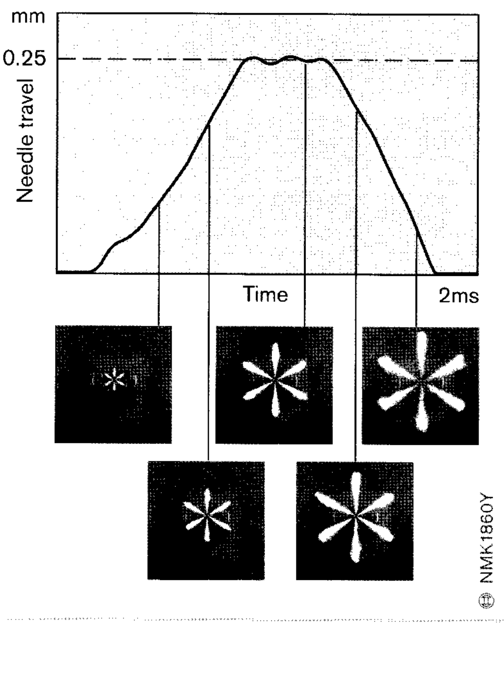


Fig. 5
1 Injection-orifice geometry
2 Seat geometry
3 Blind-hole geometry

High-speed photographs of rate-of-discharge curve of a car hole-type nozzle



Future development of the nozzle

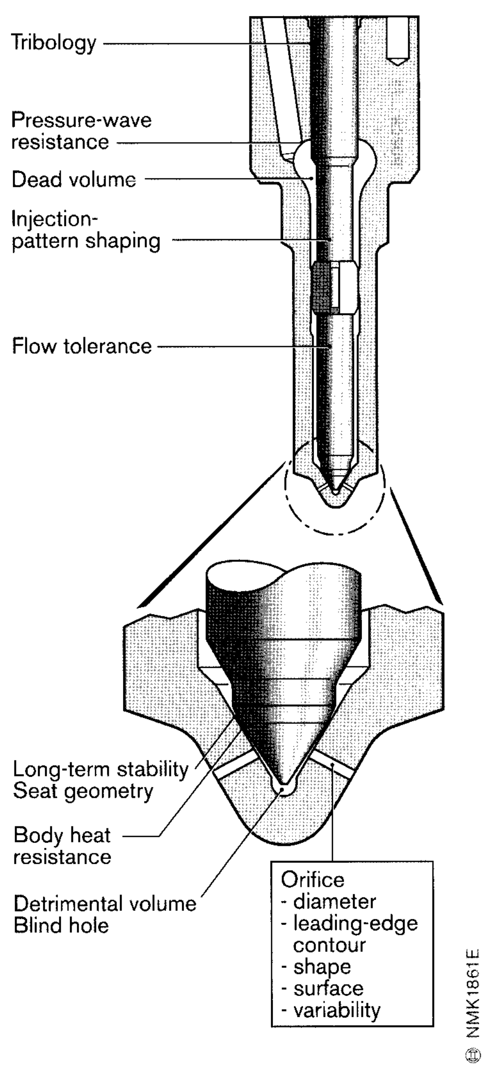
In view of the rapid development of new, high-performance engines and fuel-injection systems with sophisticated functionality (e.g. multiple injection phases), continuous development of the nozzle is a necessity. In addition, there are number of aspects of nozzle design which offer scope for innovation and further improvement of diesel engine performance in the future. The most important aims are:

- Minimizing untreated emissions in order to reduce or even eliminate the expense of costly exhaust-gas treatment equipment that also presents difficulties with regard to waste disposal (e.g. soot filters)
- Minimizing fuel consumption
- Optimizing engine noise

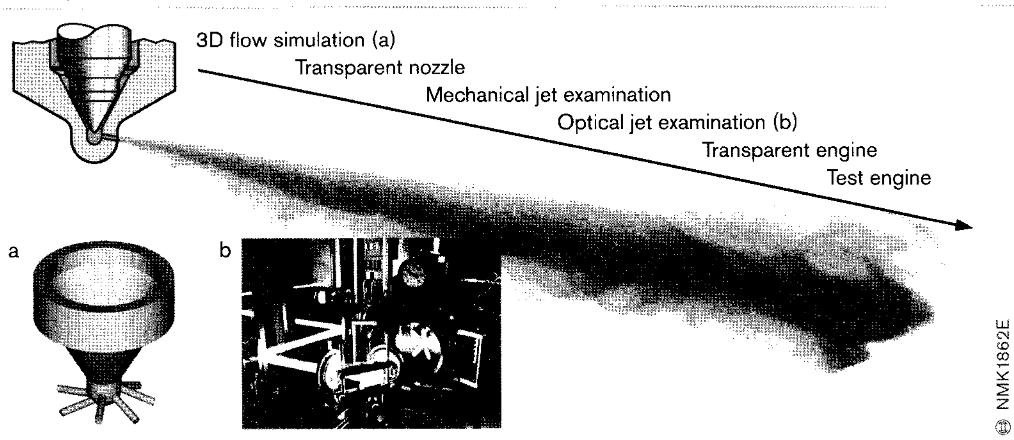
There various different areas on which attention can be focused in the future development of the nozzle (Figure 1) and a corresponding variety of development tools (Figure 2). New materials are also constantly being developed which offer improvements in durability. The use of multiple injection phases also has consequences for the design of the nozzle.

If different types of fuel (e.g. designer fuels) are used, this also affects nozzle design because of the differences in viscosity or flow characteristics. Such changes will in some cases also demand new production processes such as laser drilling for the injection orifices.

Main points of focus of nozzle development



Development tools for nozzles



Nozzle holders

A nozzle holder combines with the matching nozzle to form the nozzle-and-holder assembly. There is a nozzle-and-holder assembly fitted in the cylinder head for each engine cylinder (Figure 1). These components form an important part of the fuel-injection system and help to shape engine performance, exhaust emissions and noise characteristics. In order that they are able to perform their function properly, they must be designed to suit the engine in which they are used.

The nozzle (4) in the nozzle holder sprays fuel into the diesel-engine combustion chamber (6). The nozzle holder contains the following essential components:

- *Valve spring(s)* (9) which act(s) against the nozzle needle so as to close the nozzle
- *Nozzle-retaining nut* (8) which retains and centers the nozzle
- *Filter* (11) for keeping dirt out of the nozzle
- *Connections* for the fuel supply and return lines which are linked via the *pressure channel* (10)

Depending on design, the nozzle holder may also contain seals and spacers. Standardized dimensions and combinations provide the required degree of adaptability combined with the minimum of component diversity.

Schematic diagram of a nozzle-and-holder assembly on a direct-injection engine

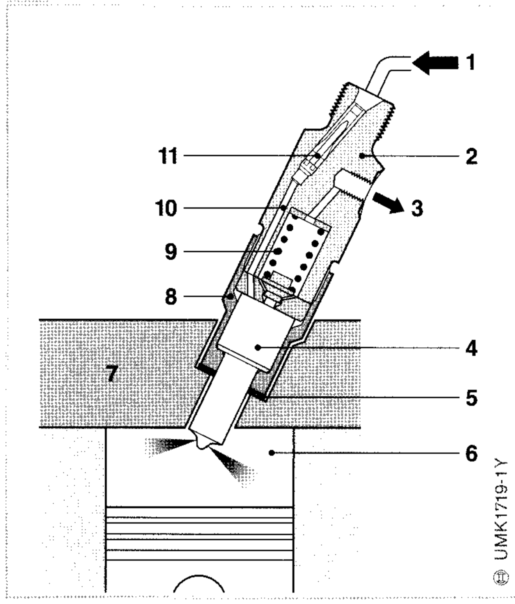


Fig. 1
1 Fuel supply
2 Holder body
3 Fuel return
4 Nozzle
5 Sealing gasket
6 Combustion chamber of diesel engine
7 Cylinder head
8 Nozzle-retaining nut
9 Valve spring
10 Pressure channel
11 Filter

Bosch type designation codes for nozzle holders

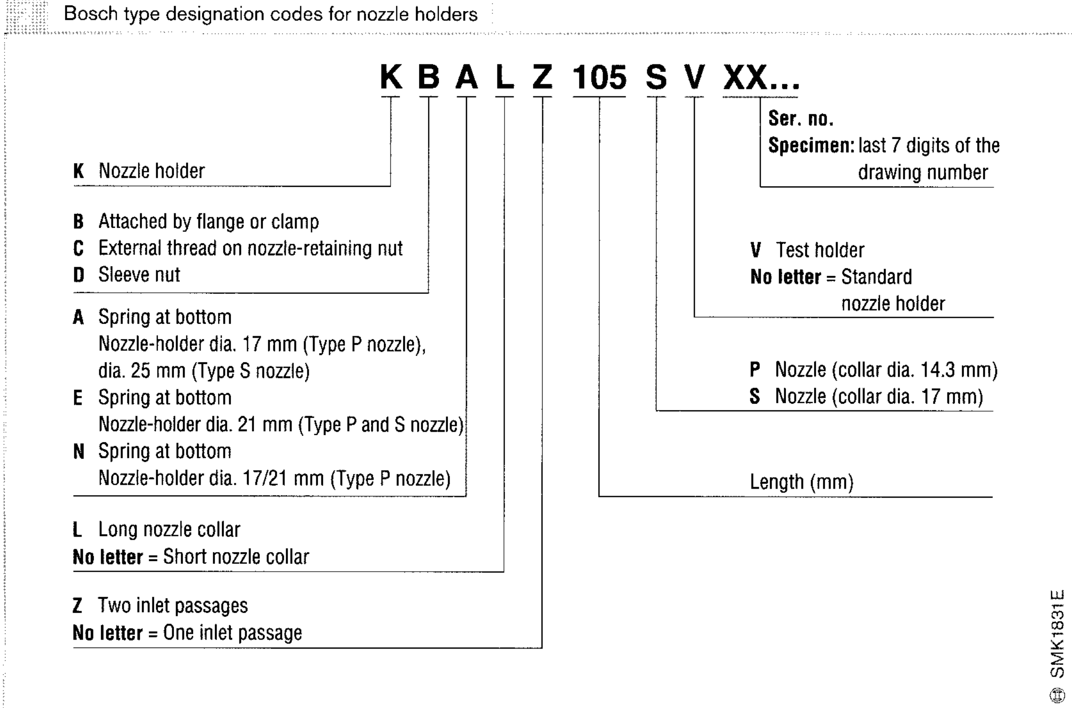


Fig. 2
This number is stamped on the nozzle holder and enables precise identification.

High-precision technology

The image associated with diesel engines in many people's minds is more one of heavy-duty machinery than high-precision engineering. But modern diesel fuel-injection systems are made up of components that are manufactured to the highest degrees of accuracy and required to withstand enormous stresses.

The nozzle is the interface between the fuel-injection system and the engine. It has to open and close precisely and reliably for the entire life of the engine. When it is closed, it must not leak. This would increase fuel consumption, adversely affect exhaust-gas emissions and might even cause engine damage.

To ensure that the nozzles seal reliably at the high pressures generated in modern fuel-injection systems such as the VR (VP44), CR, UPS and UIS designs (up to 2,050 bar), they have to be specially designed and very precisely manufactured. By way of illustration, here are some examples:

- To ensure that the sealing face of the nozzle body (1) provides a reliable seal, its has a dimensional tolerance of 0.001 mm (1 μm). That means it must be accurate to within approximately 4,000 metal atom layers!
- The nozzle-needle guide clearance (2) is 0.002...0.004 mm (2...4 μm). The dimensional tolerances are similarly less than 0.001 mm (1 μm).

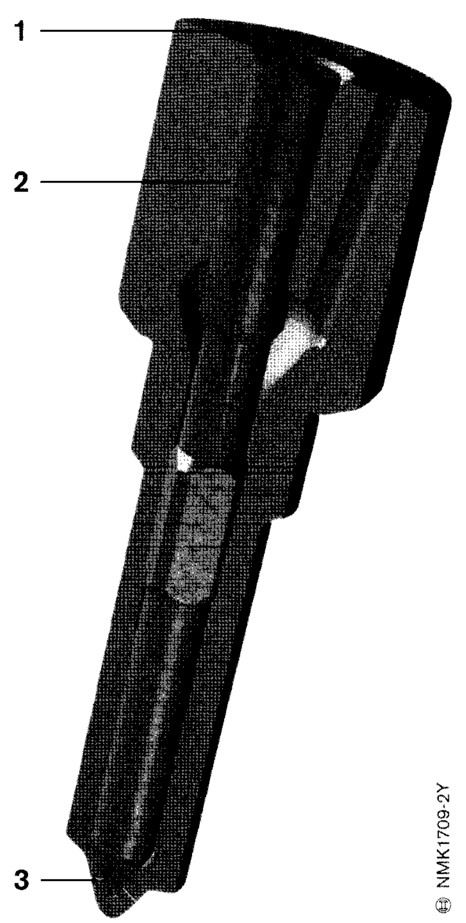
The injection orifices (3) in the nozzles are created by an electro-erosion machining process. This process erodes the metal by vaporization caused by the high temperature generated by the spark discharge between an electrode and the workpiece. Using high-precision electrodes and accurately configured parameters, extremely precise injection orifices with diameters of 0.12 mm can be produced. This means that the smallest injection orifice diameter is only twice the thickness of a human hair (0.06 mm). In order to obtain better injection characteristics, the leading edges of the nozzle injection orifices are rounded off by special abrasive fluids (hydro-erosion machining).

The minute tolerances demand the use of highly specialized and ultra-accurate measuring equipment such as

- optical 3-D coordinate measuring machine for measuring the injection orifices, or
- laser interferometers for checking the smoothness of the nozzle sealing faces.

The manufacture of diesel fuel-injection components is thus "high-volume, high-technology".

A matter of high-precision



1 Nozzle body sealing face
2 Nozzle-needle guide clearance
3 Injection orifice

Standard nozzle-and-holder assembly for direct-injection (DI) engines

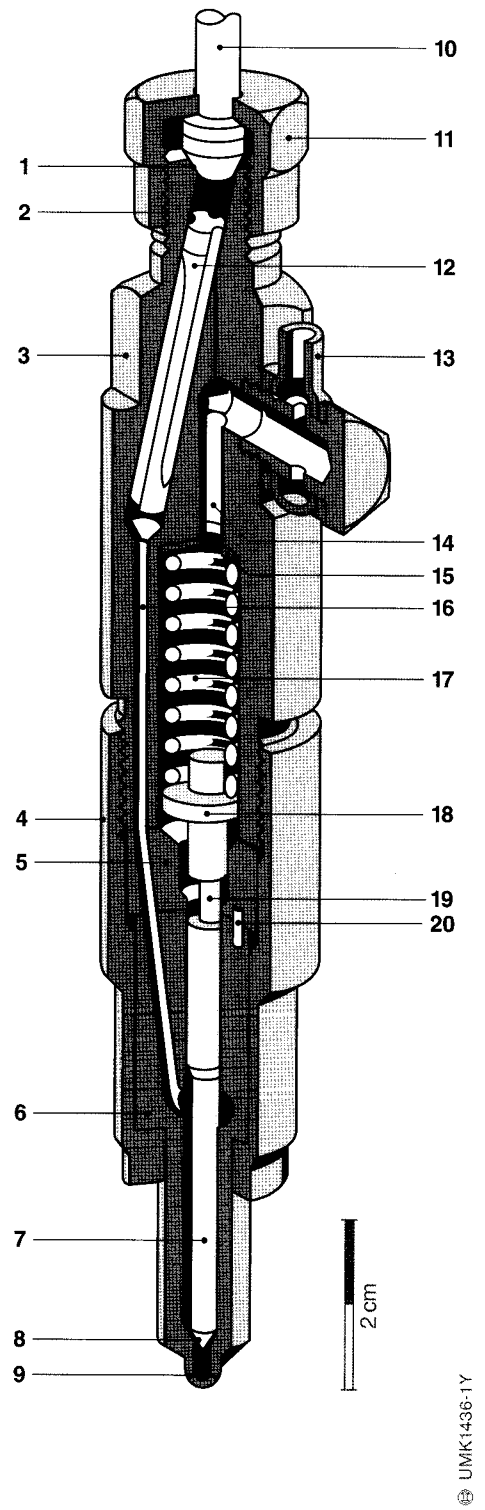


Fig. 1

1 Sealing cone
2 Screw thread for central pressure connection
3 Holder body
4 Nozzle-retaining nut
5 Intermediate disk
6 Nozzle body
7 Nozzle needle
8 Nozzle-body seat face
9 Injection orifice
10 Fuel inlet
11 Sleeve nut
12 Edge-type filter
13 Leak fuel connection
14 Leak fuel port
15 Shim
16 Pressure passage
17 Compression spring
18 Pressure pin
19 Pressure pin
20 Locating pin

Standard nozzle holders

Design and usage

The key features of standard nozzle holders are as follows:

- Cylindrical exterior with diameters of 17, 21, 25 and 26 mm
- Non-twist hole-type nozzles for engines with direct injection and
- Standardized individual components (springs, pressure pins, nozzle retaining nuts) that permit different combinations

The nozzle-and-holder assembly is made up of nozzle holder and nozzle (Figure 1, with hole-type nozzle). The nozzle holder consists of the following components:

- Holder body (3)
- Intermediate disk (5)
- Nozzle-retaining nut (4)
- Pressure pin (18)
- Compression spring (17)
- Shim (15), and
- Locating pin (20)

The nozzle is attached centrally to the holder by the nozzle-retaining nut. When the retaining nut and holder body are screwed together, the intermediate disk is pressed against the sealing faces of the holder and nozzle body. The intermediate disk acts as a limiting stop for the needle lift and also centers the nozzle relative to the nozzle holder by means of the locating pins.

The pressure pin centers the compression spring and is guided by the nozzle-needle pressure pin (19).

The pressure passage (16) inside the nozzle holder body connects through the channel in the intermediate disk to the inlet passage of the nozzle, thus connecting the nozzle to the high-pressure line of the fuel-injection pump. If required, an edge-type filter (12) may be fitted inside the nozzle holder. This keeps out any dirt that may be contained in the fuel.

Standard nozzle-and-holder assembly for direct-injection (DI) engines

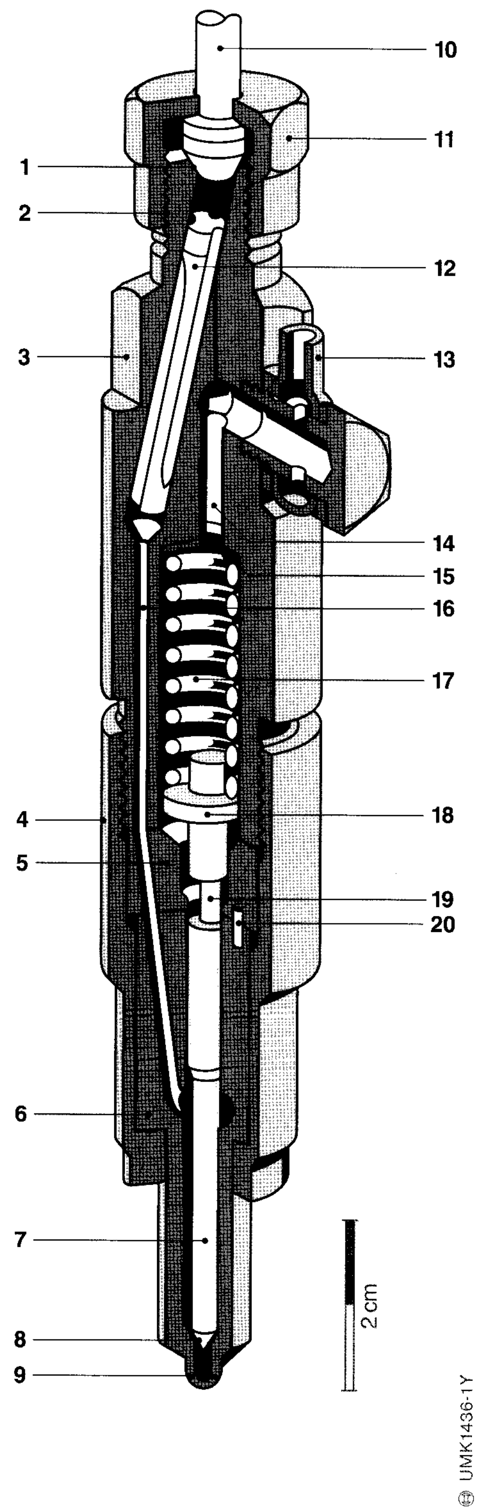


Fig. 1
1 Sealing cone
2 Screw thread for central pressure connection
3 Holder body
4 Nozzle-retaining nut
5 Intermediate disk
6 Nozzle body
7 Nozzle needle
8 Nozzle-body seat face
9 Injection orifice
10 Fuel inlet
11 Sleeve nut
12 Edge-type filter
13 Leak fuel connection
14 Leak fuel port
15 Shim
16 Pressure passage
17 Compression spring
18 Pressure pin
19 Pressure pin
20 Locating pin

Standard nozzle holders

Design and usage

The key features of standard nozzle holders are as follows:

- Cylindrical exterior with diameters of 17, 21, 25 and 26 mm
- Non-twist hole-type nozzles for engines with direct injection and
- Standardized individual components (springs, pressure pins, nozzle retaining nuts) that permit different combinations

The nozzle-and-holder assembly is made up of nozzle holder and nozzle (Figure 1, with hole-type nozzle). The nozzle holder consists of the following components:

- Holder body (3)
- Intermediate disk (5)
- Nozzle-retaining nut (4)
- Pressure pin (18)
- Compression spring (17)
- Shim (15), and
- Locating pin (20)

The nozzle is attached centrally to the holder by the nozzle-retaining nut. When the retaining nut and holder body are screwed together, the intermediate disk is pressed against the sealing faces of the holder and nozzle body. The intermediate disk acts as a limiting stop for the needle lift and also centers the nozzle relative to the nozzle holder by means of the locating pins.

The pressure pin centers the compression spring and is guided by the nozzle-needle pressure pin (19).

The pressure passage (16) inside the nozzle holder body connects through the channel in the intermediate disk to the inlet passage of the nozzle, thus connecting the nozzle to the high-pressure line of the fuel-injection pump. If required, an edge-type filter (12) may be fitted inside the nozzle holder. This keeps out any dirt that may be contained in the fuel.

Method of operation

The compression spring inside the nozzle holder acts on the nozzle needle via the pressure pin. The spring tension is set by means of a shim. The force of the spring thus determines the opening pressure of the nozzle.

The fuel passes through the edge-type filter (12) to the pressure passage (16) in the holder body (3), through the intermediate disk (5) and finally through the nozzle body (6) to the space (8) surrounding the nozzle needle. During the injection process, the nozzle needle (7) is lifted upwards by the pressure of the fuel (110...170 bar for pintle nozzles and 150...350 bar for hole-type nozzles). The fuel passes through the injection orifices (9) into the combustion chamber. The injection process comes to an end when the fuel pressure drops to a point where the compression spring (17) is able to push the nozzle needle back against its seat. Start of injection is thus controlled by fuel pressure. The injected fuel quantity depends essentially on how long the nozzle remains open.

In order to limit needle lift for pre-injection, some designs have a nozzle-needle damper (Figure 2).

Stepped nozzle holders

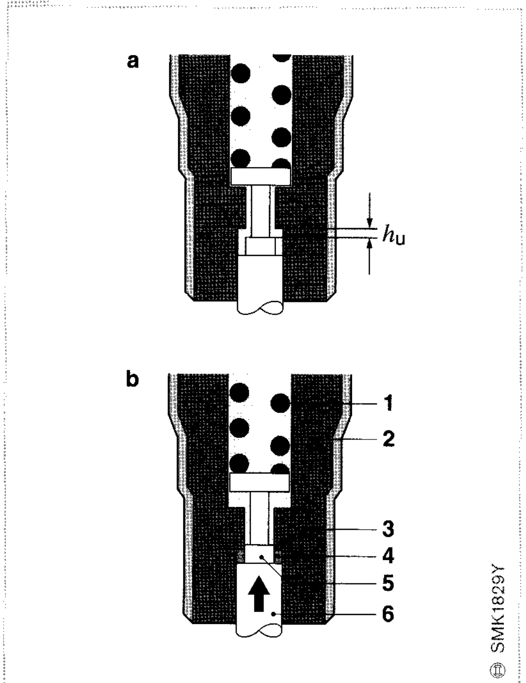
Design and usage

On multi-valve commercial-vehicle engines in particular, where the nozzle-and-holder assembly has to be fitted vertically because of space constraints, stepped nozzle-and-holder assemblies are used (Figure 3). The reason for the name can be found in the graduated dimensions (1).

The design and method of operation are the same as for standard nozzle holders. The essential difference lies in the way in which the fuel line is connected. Whereas on a standard nozzle holder it is screwed centrally to the top end of the nozzle holder, on a stepped holder it is connected to the holder body (11) by means of a delivery connection (10). This type of arrangement is normally used to achieve very short injection fuel lines, and has a beneficial effect on the injection pressure because of the smaller dead volume in the fuel lines.

Stepped nozzle holders are produced with or without a leak fuel connection (9).

Nozzle-needle damper



Stepped nozzle-and-holder assembly

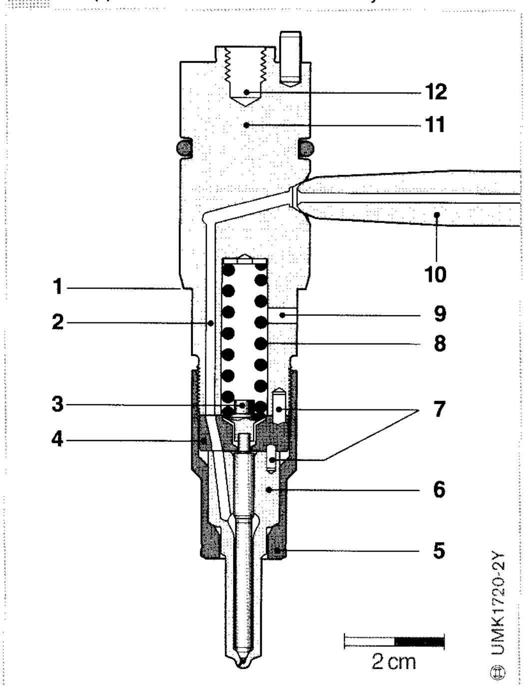


Fig. 2
a Closed nozzle
b Damped lift

1 Compression spring
2 Holder body
3 Leak gap
4 Hydraulic cushion
5 Damper piston
6 Nozzle needle

h_u Undamped lift (approx. 1/3 of full lift)

Fig. 3
1 Step
2 Pressure passage
3 Pressure pin
4 Intermediate disk
5 Nozzle-retaining nut
6 Nozzle body
7 Locating pin
8 Compression spring
9 Leak fuel port
10 Delivery connection
11 Holder body
12 Thread for extractor bolt

Two-spring injector assembly

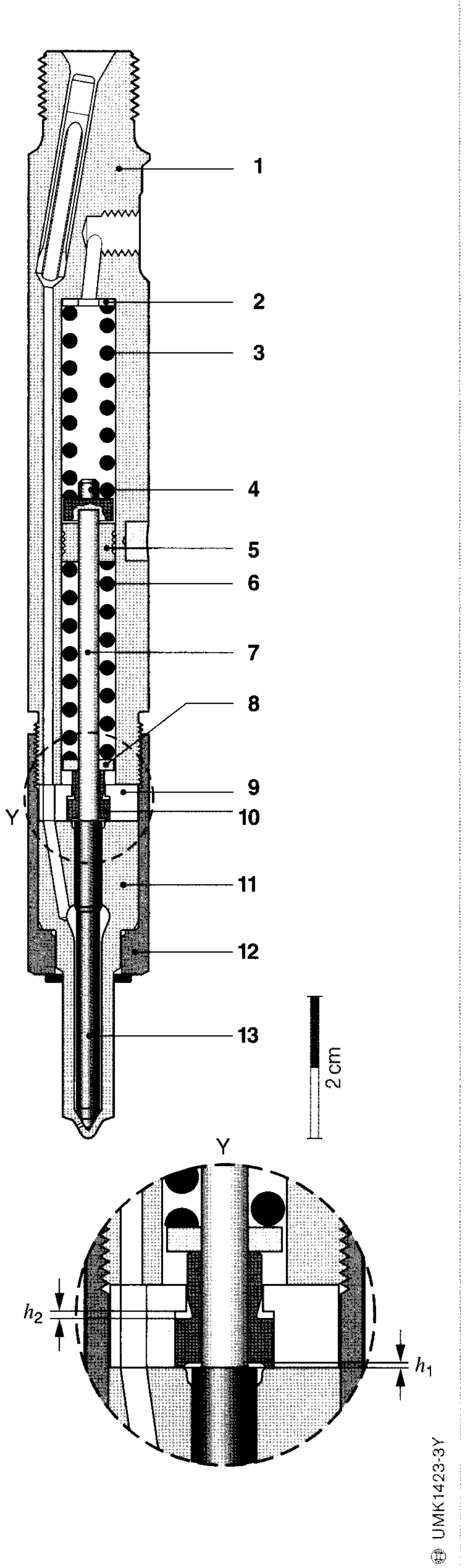


Fig. 1
1 Holder body
2 Shim
3 Compression spring 1
4 Pressure pin
5 Guide washer
6 Compression spring 2
7 Pressure pin
8 Spring seat
9 Intermediate disk
10 Stop sleeve
11 Nozzle body
12 Nozzle-retaining nut
13 Nozzle needle

h_1 Plunger lift to port closing
h_2 Main lift

Two-spring nozzle holders

Usage

The two-spring nozzle holder is a refinement of the standard nozzle holder. It has the same external dimensions. Its graduated rate-of-discharge curve (Figure 2) produces "softer" combustion and therefore a quieter engine, particularly at idle speed and part load. It is used primarily on direct-injection (DI) engines.

Design and method of operation

The two-spring nozzle holder (Figure 1) has two compression springs positioned one behind the other. Initially, only one of the compression springs (3) is acting on the nozzle needle (13) and thus determines the opening pressure. The second compression spring (6) rests against a stop sleeve (10) which limits the plunger lift to port closing. During the injection process, the nozzle needle initially moves towards the plunger lift to port closing, h_1 (0.03...0.06 mm for DI engines, 0.1 mm for IDI engines). This allows only a small amount of fuel into the combustion chamber.

As the pressure inside the nozzle holder continues to increase, the stop sleeve overcomes the force of both compression springs (3 and 6). The nozzle needle then completes the main lift (h_1 + h_2, 0.2...0.4 mm) so that the main injected fuel quantity is injected.

Comparison of needle lift curve

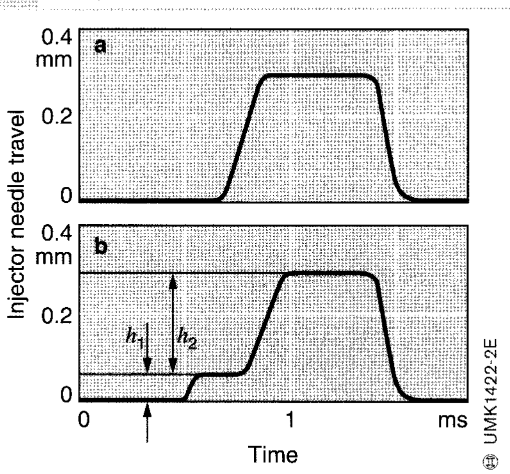


Fig. 2
a Standard nozzle holder (single-spring)
b Two-spring nozzle holder

h_1 Plunger lift to port closing
h_2 Main lift

Nozzle holders with needle-motion sensors

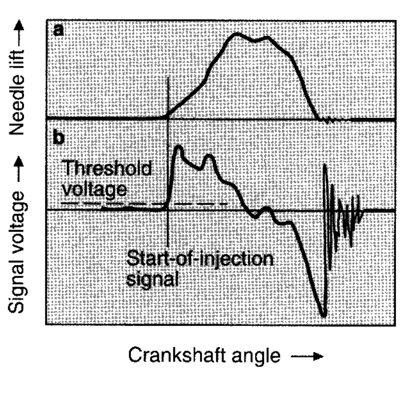
Usage

Start of delivery is a key variable for optimizing diesel-engine performance. Detection of this variable allows the adjustment of start of delivery according to engine load and speed within a closed control loop. In systems with distributor and in-line fuel-injection pumps, this is achieved by means of a nozzle with a needle-motion sensor (Figure 2) which transmits a signal when the nozzle needle starts to move upwards. It is sometimes also called a needle-motion sensor.

Design and method of operation

A current of approximately 30 mA is passed through the detector coil (Figure 2, Item 11). This produces a magnetic field. The extended pressure pin (12) slides inside the guide pin (9). The penetration depth X determines the magnetic flux in the detector coil. By virtue of the change in magnetic flux in the coil, movement of the nozzle needle induces a velocity-dependent voltage signal (Figure 1) in the coil which is processed by an analyzer circuit in the electronic control unit. When the signal level exceeds a threshold voltage, it is interpreted by the analyzer circuit to indicate the start of injection.

Needle-motion sensor signal



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Two-spring nozzle holder with needle-motion sensor for direct-injection engines

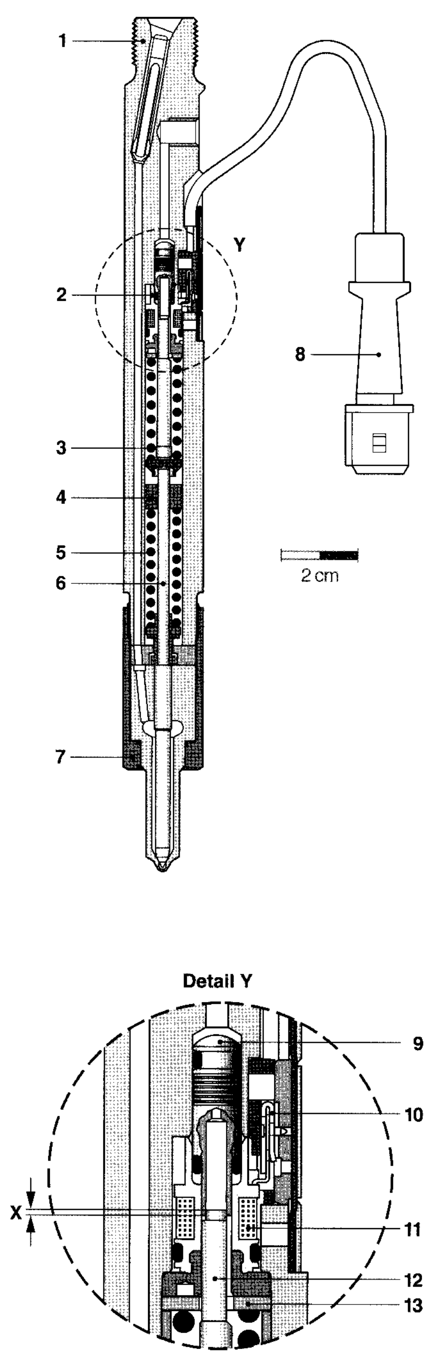


Fig. 1
a Needle-lift curve
b Corresponding coil signal voltage curve

Fig. 2
1 Holder body
2 Needle-motion sensor
3 Compression spring
4 Guide washer
Compression spring
Pressure pin
7 Nozzle-retaining nut
8 Connection to analyzer circuit
9 Guide pin
10 Contact tab
11 Detector coil
12 Pressure pin
13 Spring seat

X Penetration depth

High-pressure lines

Regardless of the basic system concept – in-line fuel-injection pump, distributor injection pump or unit pump systems – it is the high-pressure delivery lines and their connection fittings that furnish the links between the fuel-injection pump(s) and the nozzle-and-holder assemblies at the individual cylinders. In common-rail systems, they serve as the connection between the high-pressure pump and the rail as well as between rail and nozzles. No high-pressure delivery lines are required in the unit-injector system.

High-pressure connection fittings

The high-pressure connection fittings must supply secure sealing against leakage from fuel under the maximum primary pressure. The following types of fittings are used:

- Sealing cone and union nut
- Heavy-duty insert fittings and
- Perpendicular connection fittings

Sealing cone with union nut

All of the fuel-injection systems described above use sealing cones with union nuts (Fig. 1). The advantages of this connection layout are:

- Easy adaptation to individual fuel-injection systems
- Fitting can be disconnected and reconnected numerous times
- The sealing cone can be shaped from the base material

At the end of the high-pressure line is the compressed pipe-sealing cone (3). The union nut (2) presses the cone into the high-pressure connection fitting (4) to form a seal. Some versions are equipped with a supplementary thrust washer (1). This provides a more consistent distribution of forces from the union nut to the sealing cone. The cone's open diameter should not be restricted, as this would obstruct fuel flow. Compressed sealing cones are generally manufactured in conformity with DIN 73 365 (Fig. 2).

Heavy-duty insert fittings

Heavy-duty insert fittings (Fig. 3) are used in unit-pump and common-rail systems as installed in heavy-duty commercial vehicles. With the insert fitting, it is not necessary to route the fuel line around the cylinder head to bring it to the nozzle holder or nozzle. This allows shorter fuel lines with associated benefits when it comes to space savings and ease of assembly.

The screw connection (8) presses the line insert (3) directly into the nozzle holder (1) or nozzle. The assembly also includes a mainte-

Fig. 1
1 Thrust washer
2 Union nut
3 Pipe sealing cone on high-pressure delivery line
4 Pressure connection on fuel-injection pump or nozzle holder

High-pressure connection with sealing cone and union nut

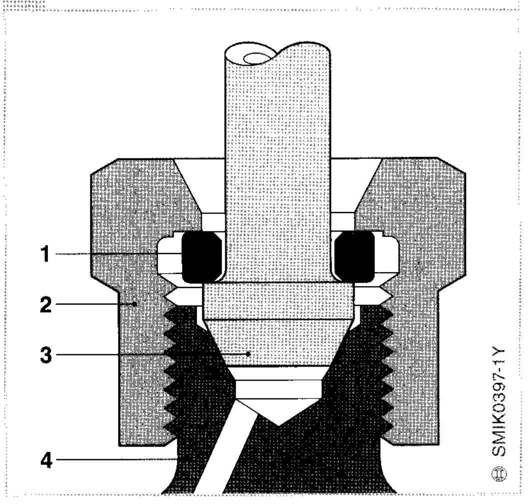
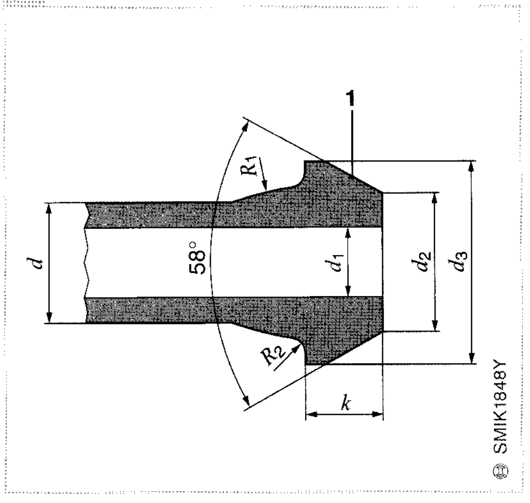


Fig. 2
1 Sealing surface

d Outer line diameter
d_1 Inner line diameter
d_2 Inner cone diameter
d_3 Outer cone diameter
k Length of cone
R_1, R_2 Radii

Compressed sealing cone (main dimensions)



nance-free edge-type filter (5) to remove coarse contamination from the fuel. At its other end, the line is attached to the high-pressure delivery line (7) with a sealing cone and union nut (6).

Perpendicular connection fittings

Perpendicular connection fittings (Fig. 4) are used in some passenger-car applications. They are suitable for installations in which there are severe space constraints. The fitting contains passages for fuel inlet and return (7, 9). A bolt (1) presses the perpendicular fitting onto the nozzle holder (5) to form a sealed connection.

High-pressure delivery lines

The high-pressure fuel lines must withstand the system's maximum pressure as well as pressure variations that can attain very high fluctuations. The lines are seamless precision-made steel tubing in killed cast steel which has a particularly consistent microstructure. Dimensions vary according to pump size (Table 1, next page).

All high-pressure delivery lines are routed to avoid sharp bends. The bend radius should not be less than 50 mm.

Sample or a high-pressure fitting

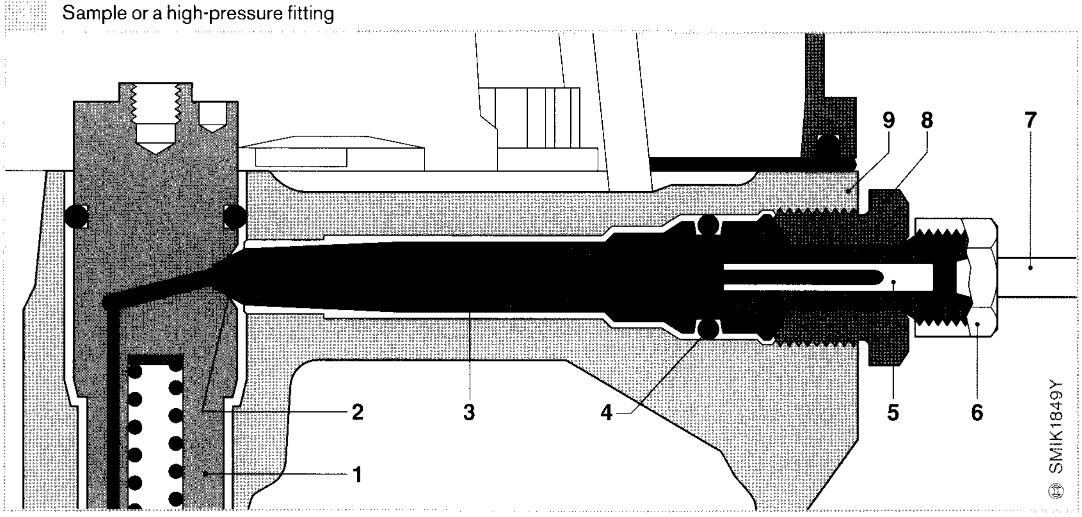


Fig. 3
1 Nozzle holder
2 Sealing cone
3 High-pressure fitting
4 Seal
5 Edge-type filter
6 Union nut
7 High-pressure delivery line
8 Screw connections
9 Cylinder head

Sample of a perpendicular fitting

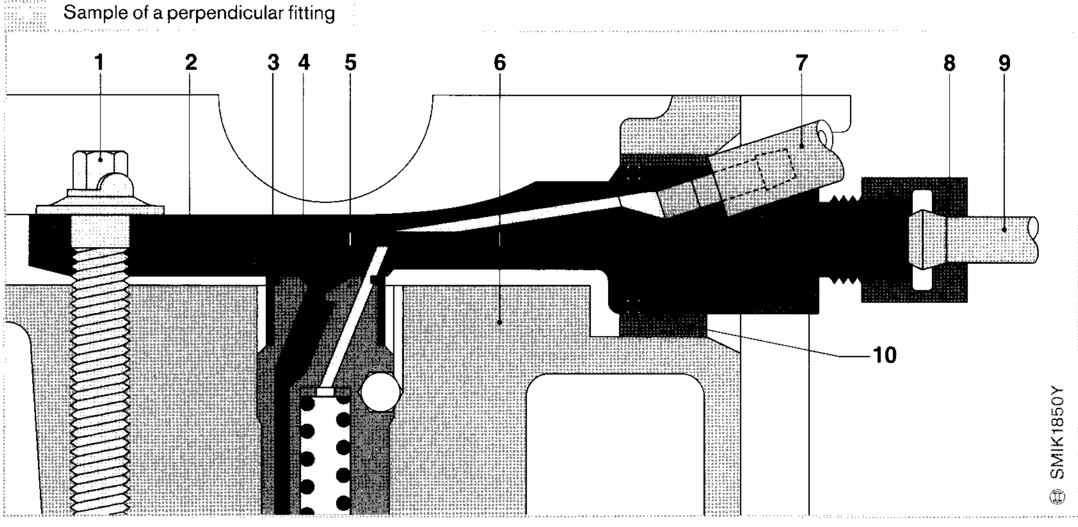


Fig. 4
1 Expansion bolt
2 Perpendicular fitting
3 Molded seal
4 Edge-type filter
5 Nozzle holder
6 Cylinder head
7 Fuel return line (leakage-fuel line)
8 Union nut
9 High-pressure delivery line
10 Clamp

Length, diameter and wall depth of the high-pressure lines all affect the injection process. To cite some examples: Line length influences speed-sensitive the rate of discharge, while internal diameter is related to throttling loss and compression effects, which will be reflected in the injected-fuel quantity. These considerations lead to prescribed line dimensions that must be strictly observed. Tubing of other dimensions should never be installed during service and repairs. Defective high-pressure tubing should always be replaced by OEM lines. During servicing or maintenance, it is also important to observe precautions against fouling entering the system. This applies in any case to all service work on fuel-injection systems.

A general priority in the development of fuel-injection systems is to minimize the length of high-pressure lines. Shorter lines produce better injection-system performance.

Injection is accompanied by the formation of pressure waves. These are pulses that propagate at the speed of sound before finally being reflected on impact at the ends. This phenomenon increases in intensity as engine speed rises. Engineers exploit it to raise injection pressure. The engineering process entails defining line lengths that are precisely matched to the engine and the fuel-injection system.

All cylinders are fed by high-pressure delivery lines of a single, uniform length. More or less angled bends in the lines compensate for the different distances between the outlets from the fuel-injection pump or rail, and the individual engine cylinders.

The primary factor determining the high-pressure line's compression-pulsating fatigue strength is the surface quality of the inner walls of the lines, as defined by material and peak-to-valley height. Especially demanding performance requirements are satisfied by prestressed high-pressure delivery lines (for applications of 1,400 bar and over). Before installation on the engine, these customized lines are subjected to extremely high pressures (up to 3,800 bar). Then pressure is suddenly relieved. The process compresses the material on the inner walls of the lines to provide increased internal strength.

The high-pressure delivery lines for vehicle engines are normally mounted with clamp brackets located at specific intervals. This means that transfer of external vibration to the lines is either minimal or nonexistent.

The dimensions of high-pressure lines for test benches are subject to more precise tolerance specifications.

Table 1

d Outer line diameter

d_1 Inner line diameter

Wall thicknesses indicated in **bold** should be selected when possible.

Dimensions for high-pressure lines are usually indicated as follows:

d x *s* x *l*

l Line length

Main dimensions of major high-pressure delivery lines in mm

d \ d_1	1.4	1.5	1.6	1.8	2.0	2.2	2.5	2.8	3.0	3.6	4.0	4.5	5.0	6.0	7.0	8.0	9.0
	Wall thickness *s*																
4	1.3	1.25	**1.2**														
5	1.8	1.75	1.7	1.6													
6		**2.25**	**2.2**	**2.1**	**2**	1.9	**1.75**	**1.6**	**1.5**								
8					**3**	2.9	**2.75**	**2.6**	**2.5**	2.2	**2**						
10							**3.75**	**3.6**	**3.5**	**3.2**	**3**	2.75	2.5				
12									**4.5**	4.2	**4**	3.75	**3.5**				
14											**5**	4.75	**4.5**	**4**		3	
17													6	**5.5**	**5**	**4.5**	
19																	**5**
22																**7**	

Cavitation in the high-pressure system

Cavitation can damage fuel-injection systems (Fig. 1). The process takes place as follows:

Local pressure variations occur at restrictions and in bends when a fluid enter an enclosed area at extremely high speeds (for instance, in a pump housing or in a high-pressure line). If the flow characteristics are less than optimum, low-pressure sectors can form at these locations for limited periods of time, in turn promoting the formation of vapor bubbles.

These gas bubbles implode in the subsequent high-pressure phase. If a wall is located immediately adjacent to the affected sector, the concentrated high energy can create a cavity in the surface over time (erosion effect). This is called cavitation damage.

As the vapor bubbles are transported by the fluid's flow, cavitation damage will not necessarily occur at the location where the bubble forms. Indeed, cavitation damage is frequently found in eddy zones.

The causes behind these temporary localized low-pressure areas are numerous and varied. Typical factors include:

- Discharge processes
- Closing valves
- Pumping between moving gaps, and
- Vacuum waves in passages and lines

Attempts to deal with cavitation problems by improving material quality and surface-hardening processes cannot produce anything other than very modest gains. The ultimate objective is and remains to prevent the vapor bubbles from forming, and, should complete prevention prove impossible, to improve flow behavior to limit the negative impacts of the bubbles.

Cavitation damage in the distributor head of a VE pump

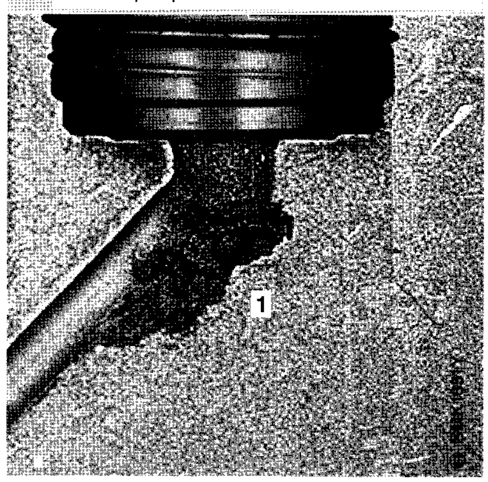


Fig. 1
1 Cavitation

Implosion of a cavitation bubble

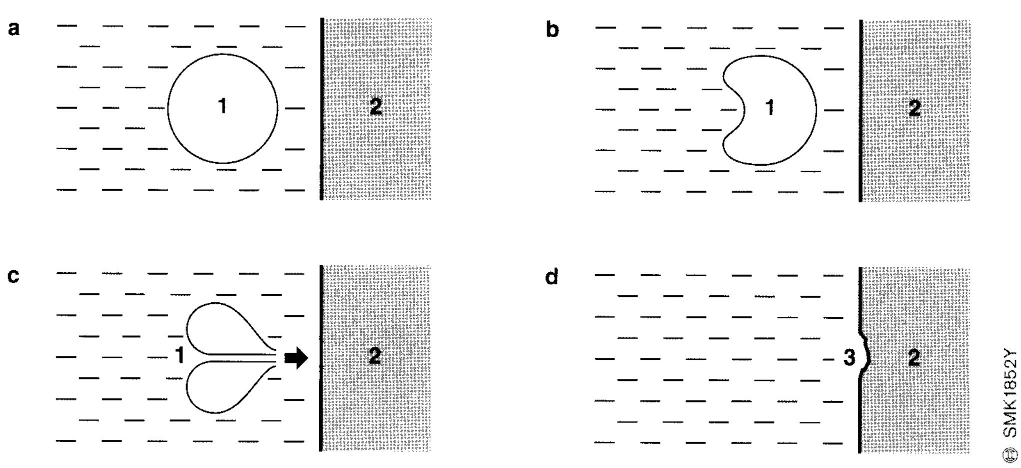


Fig. 2
a A vapor bubble is formed
b The vapor bubble collapses
c The collapsed sections form a sharp edge with extremely high energy
d The imploding vapor bubble leaves a recess on the surface

1 Vapor bubble
2 Wall
3 Recess

Electronic diesel control (EDC)

Modern electronic diesel-engine control permits the precise and highly flexible definition of the fuel-injection parameters. This is the only way to comply with the wide range of technical demands made on a modern diesel engine. The Electronic Diesel Control (EDC) is subdivided into the three system blocks "Sensors and desired-value generators", "ECU", and "Actuators".

Technical requirements

The calls for reduced fuel consumption and emissions, together with increased power output and torque, are the decisive factors behind present-day developments in the diesel fuel-injection field.

In the past years this has led to an increase in the use of direct-injection (DI) diesel engines. Compared to prechamber or whirl-chamber engines, the so-called indirect-injection (IDI) engines, the DI engine operates with far higher injection pressures. This leads to improved A/F mixture formation, combustion of the more finely atomized fuel droplets is more complete, and there are less unburnt hydrocarbons (HC) in the exhaust gas. In the DI engine, the improved mixture formation and the fact that there are no overflow losses between pre-chamber/whirl chamber and the main combustion chamber results in fuel-consumption savings of between 10...15% compared to the IDI engine.

In addition, the increasing requirements regarding vehicle driveability have a marked effect on the demands made on modern engines, and these are subject to increasingly more severe requirements with regard to exhaust-gas and noise emissions (NO_X, CO, HC, particulates).

This has led to higher demands being made on the injection system and its control with respect to:

- High injection pressures
- Structured rate-of-discharge curve
- Pilot injection and possibly post injection
- Adaptation of injected fuel quantity, boost pressure, and injected fuel quantity to the given operating state
- Temperature-dependent start quantity
- Load-independent idle-speed control
- Cruise Control
- Closed-loop-controlled exhaust-gas recirculation (EGR)
- Tighter tolerances for injected fuel quantity and injection point, together with high accuracy to be maintained throughout the vehicle's useful life

Conventional mechanical (flyweight) governors use a number of add-on devices to register the various operating conditions, and ensure that mixture formation is of high standard. Such governors, though, are restricted to simple open-loop control operations at the engine, and there are many important actuating variables which they cannot register at all or not quickly enough.

The increasingly severe demands it was subjected to, meant that the EDC developed from a simple system with electrically triggered actuator shaft to become a complex engine-management unit capable of carrying out real-time processing of a wide variety of data.

System overview

In the past years, the marked increase in the computing power of the microcontrollers available on the market has made it possible for the EDC (Electronic Diesel Control) to comply with the above-named stipulations.

In contrast to diesel-engine vehicles with conventional in-line or distributor injection pumps, the driver of an EDC-controlled vehicle has no direct influence, for instance through the accelerator pedal and Bowden cable, upon the injected fuel quantity.

On the contrary, the injected fuel quantity is defined by a variety of actuating variables, for instance:

- Driver input (accelerator-pedal setting)
- Operating state
- Engine temperature
- Intervention from other systems (e.g. TCS)
- Effects on toxic emissions etc.

Using these influencing variables, the ECU not only calculates the injected fuel quantity, but can also vary the instant of injection. This of course means that an extensive safety concept must be implemented that detects deviations and, depending upon their severity, initiates appropriate countermeasures (e.g. limitation of torque, or emergency (limp-home) running in the idle-speed range). EDC therefore incorporates a number of closed control loops.

EDC also permits the exchange of data with other electronic systems in the vehicle (e.g. with the traction control system (TCS), the electronic transmission-shift control, or with the electronic stability program (ESP). This means that engine management can be integrated in the overall vehicle system (e.g. for engine-torque reduction when shifting gear with an automatic gearbox, adaptation of engine torque to wheel slip, release signal for fuel injection from the vehicle immobilizer, etc.).

The EDC system is fully integrated in the vehicle's diagnostics system. It complies with all OBD (On-Board-Diagnosis) and EOBD (European On-Board Diagnosis) stipulations.

System blocks

The EDC system comprises three system blocks: (Fig. 1):

1. *Sensors and desired-value generators* (1) for the detection of operating conditions (e.g. engine rpm) and of desired values (e.g. switch position). These convert the various physical quantities into electrical signals

2. *Electronic control unit (ECU)* (2) processes the information from the sensors and the desired-value generators in accordance with given computational processes (control algorithms). The ECU triggers the actuators with its electrical output signals and also sets up the interfaces to other systems in the vehicle (4) and to the vehicle diagnosis facility (5).

3. *Solenoid actuators* (3) convert the ECU's electrical output signals into mechanical quantities (e.g. for the solenoid valve which controls the injection, or for the solenoid of the actuator mechanism).

Major EDC components

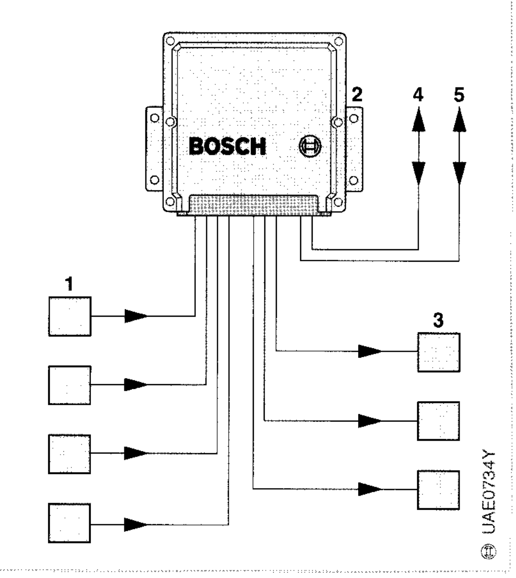
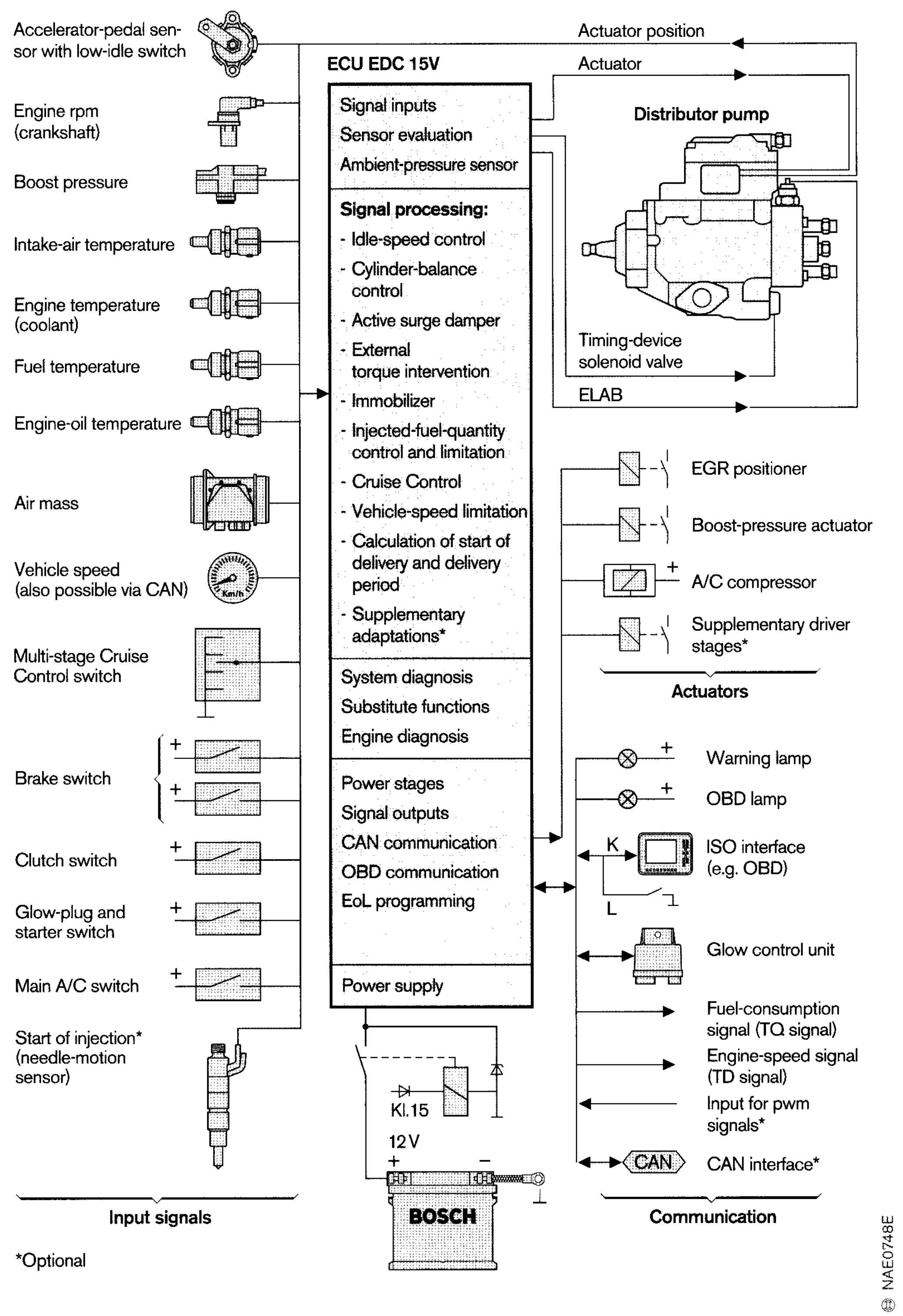


Figure 1
1 Sensors and desired-value generators (input signals)
2 ECU
3 Actuators
4 Interface to other systems
5 Diagnosis interface

Helix and-Port-controlled axial-piston distributor pumps

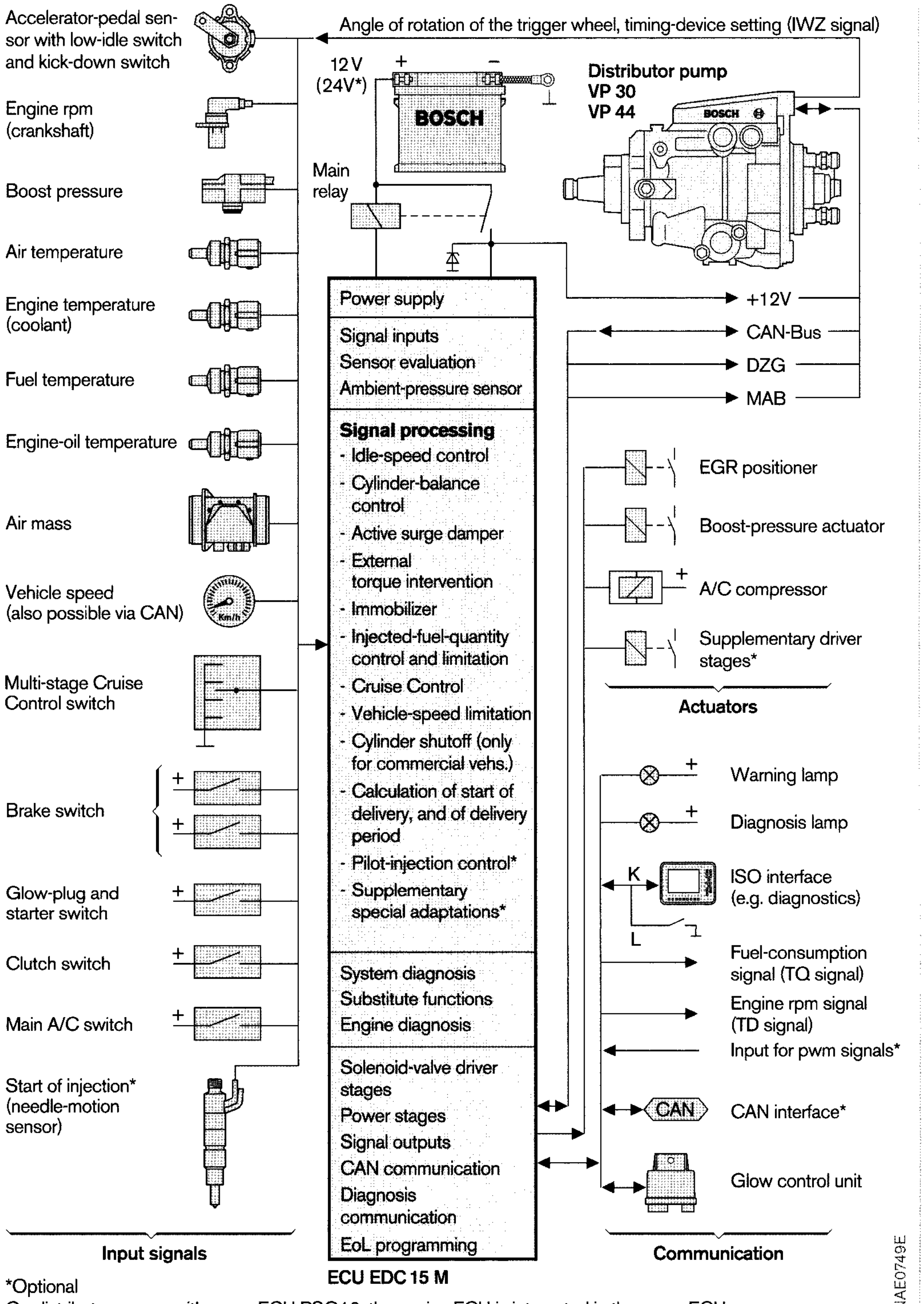
Overview of the EDC components for VE-EDC port-and-helix-controlled distributor pumps



*Optional

Solenoid-valve-controlled axial-piston and radial-piston distributor pumps

Overview of the EDC components for solenoid-valve-controlled distributor pumps VE-MV, VR



*Optional
On distributor pumps with pump ECU PSG16, the engine ECU is integrated in the pump ECU

Service technology

Important
This chapter provides general descriptions of service technology, and is *not* intended to replace repair and instruction manuals. Repairs should always be performed by qualified professional technicians.

When car drivers need help, they can count on more than 10,000 Bosch Service centers located in 132 countries. As these centers are not associated with any specific automotive manufacturer, they can provide neutral, impartial assistance. Fast assistance is always available, even in the sparsely populated regions of South America and Africa. A single set of quality standards applies everywhere. It is no wonder, therefore, that the Bosch service warranty is valid throughout the world.

Overview

The specifications and performance data of Bosch components and systems are precisely matched to the requirements of each individual vehicle. Bosch also develops and designs the test equipment, special tools and diagnosis technology needed for tests and inspections.

Bosch universal testers – ranging from the basic battery tester to the complete vehicle test stand – are being used in automotive repair shops and by inspection agencies all over the world.

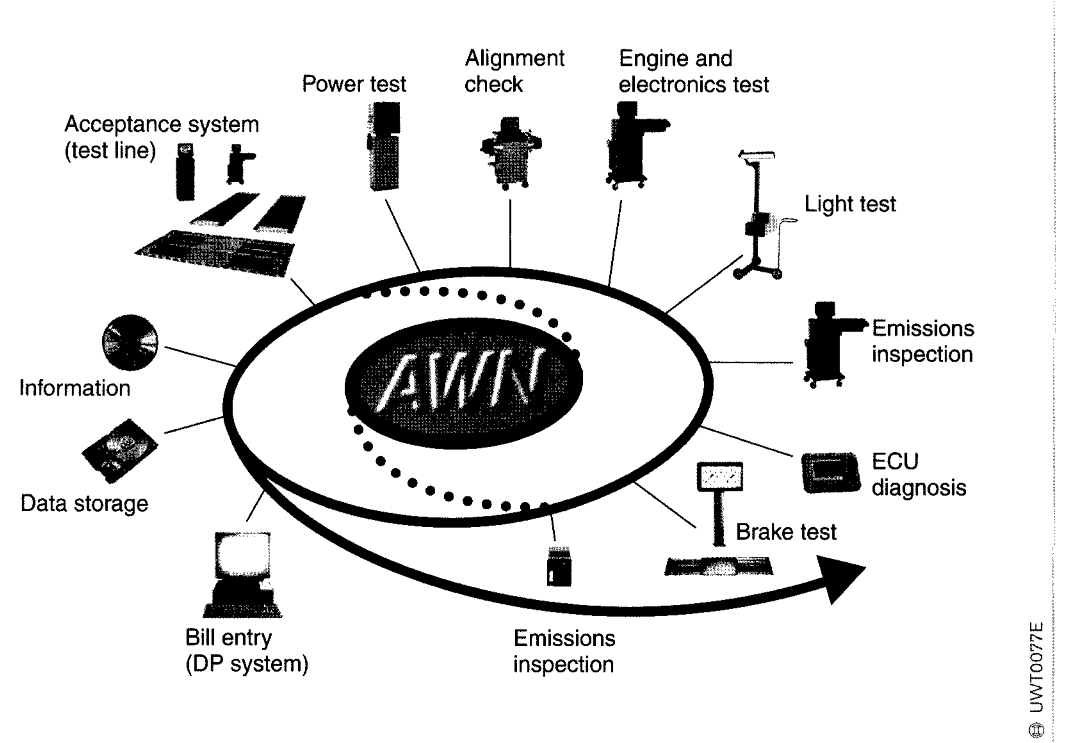
Service personnel receive training in the efficient use of this test technology as well as information on a range of automotive systems. Meanwhile, feedback from our customers constantly flows back to the development of new products.

AWN service network

Test technology
It is still possible to test mechanical systems in motor vehicles using relatively basic equipment. But mastering the increasingly complex electronic systems found in modern

[1] Bosch service technology stems from development activities carried out by the Bosch AWN service network.
The "asanetwork GmbH" is responsible for advanced development and marketing under the "AWN" name.

The AWN service network [1]



vehicles means using new test methods that rely on electronic data processing. The future belongs to a technology that links every IT system in every service center in a single, unified network, the AWN Asanet WorkshopNetwork (Fig.1). In 1998 Bosch received the Automechanika Innovation Prize in the Shop and Service category for this innovation.

Test process

When a vehicle arrives for a service inspection, the job-order processing system database provides immediate access to all the available information on the vehicle. The moment the vehicle enters the shop, the system offers access to the vehicle's entire service history, including all service and repairs that it has received in the past.

Individual diagnostic testers provide the data needed for direct comparisons of setpoint values and actual measured values, with no need for supplementary entries. All service procedures and replacement components are recorded to support the billing process. After the final road test, the bill is produced simply by striking a few keys. The system also provides a clear and concise printout with the results of the vehicle diagnosis. This offers the customer a full report detailing all of the service operations and materials that went into the vehicle's repair.

Electronic Service Information (ESI[tronic])

Even in the past the wide variety of vehicle makes and models made the use of IT systems essential (for part numbers, test specifications, etc.) Large data records, such as those containing information on spare parts, are contained on microfiche cards. Microfiche readers provide access to these microfiche libraries and are still standard equipment in every automotive service facility.

In 1991 ESI[tronic] (Electronic Service Information), intended for use with a standard PC, was introduced to furnish data on CDs. As ESI[tronic] can store much more data than a conventional microfiche system, it accommodates a larger range of potential applications. It can also be incorporated in electronic data processing networks.

Application

The ESI[tronic] software package supports service personnel throughout the entire vehicle-repair process by providing the following information:

- Spare component identification (correlating spare part numbers with specific vehicles, etc.)
- Flat rates
- Repair instructions
- Circuit diagrams
- Test specifications
- Test data from vehicle diagnosis

Service technicians can select from various options for diagnosis problems and malfunctions: The KTS500 is a high-performance portable system tester, or the KTS500C, which is designed to run on the PCs used in service shops (diagnostic stations). The KTS500C consists of a PC adapter card, a plugin card (KTS) and a test module for measuring voltage, current and resistance. An interface allows ESI[tronic] to communicate with the electronic systems in the vehicle, such as the engine control unit. Working at the PC, the user starts by selecting the SIS (Service Information System) utility to initiate diagnosis of on-board control units and access the engine control unit's fault storage. ESI[tronic] uses the results of the diagnosis as the basis for generating specific repair instructions. The system also provides displays with other information, such as component locations, exploded views of assemblies, diagrams showing the layouts of electrical, pneumatic and hydraulic systems, etc. Working at the PC, users can then proceed directly from the exploded view to the parts list with part numbers to order the required replacement components.

Testing EDC systems

Every EDC system has a self-diagnosis capability which allows comprehensive testing of the entire fuel-injection system.

Testing equipment

Effective testing of the system requires the use of special testing equipment. Whereas in the past an electronic fuel-injection system could be tested using simple electronic testers (e.g. a multimeter), the continuous advancement of EDC systems means that complex testing devices are essential today.

The system testers in the KTS series are widely used by vehicle repairers. The KTS500 (Fig. 1) offers a wide range of capabilities for use in the workshop, enhanced in particular by its graphical display of data, such as test results. In the descriptions that follow, these system testers are sometimes also referred to as "engine testers".

Functions of KTS500

The KTS500 offers a wide variety of functions which are selected by means of buttons and menus on the large display screen. The list below details the most important functions offered by the KTS500.

Reading the fault memory: The KTS500 can read out faults stored in the fault memory during system operation by the self-diagnosis function and display them on screen in plain text.
Reading actual data: Actual data that is used in calculations by the engine ECU can be read and displayed as a physical quantity (e.g. engine speed in rpm).
Actuator diagnosis: The function of the electrical actuators can be tested.
Engine test: The system tester initiates programmed test sequences for checking the engine ECU or the engine (e.g. compression test).
Multimeter function: Electrical current, voltage and resistance can be tested in the same way as with a conventional multimeter.
Time graph display: The measured data recorded over time is displayed as a signal graph as on an oscilloscope.
Additional information: Specific additional information relevant to the faults/components displayed can also be shown (e.g. location, component test data, electrical circuit diagrams).
Printout: All data (e. g. list of actual data) can be printed out on standard PC printers.
Programming: The software on the engine ECU can be reprogrammed with the KTS500 (software update).

System tester KTS500

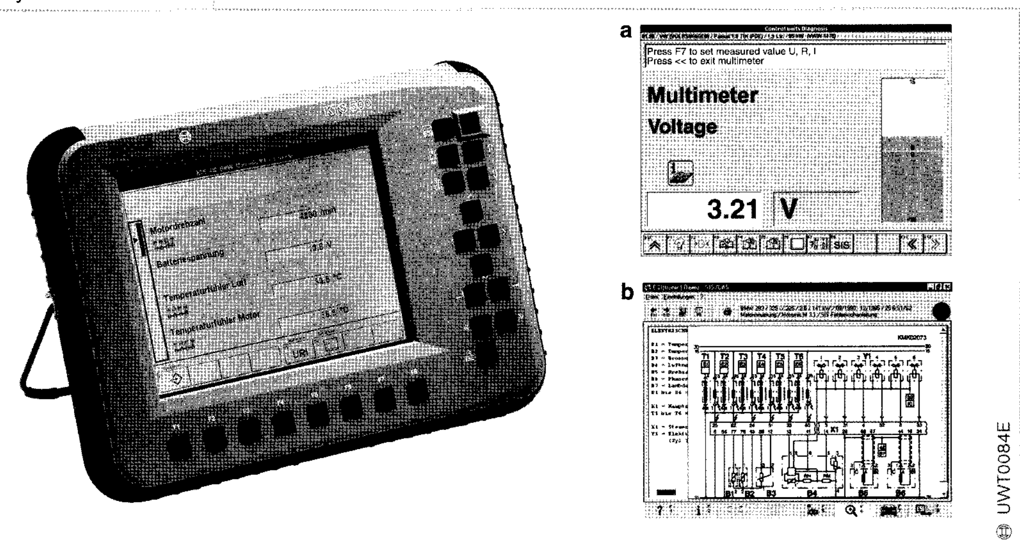


Fig. 1
a Graphical display of multimeter function
b Graphical display of an electrical terminal diagram

The extent to which the capabilities of the KTS500 can be utilized depends on the system under test. Not all engine ECUs support its full range of functions.

Workshop procedures (standard procedures)

The diagnosis procedure is the same for all systems with an electronic diesel-engine control (EDC) system. The most important tool is the engine tester which is connected to the engine ECU via the diagnosis interface.

Vehicle identification

First of all, the vehicle must be identified. The engine tester has to know which model of vehicle is under test so that it can refer to the correct data.

Reading the fault memory

The self-diagnosis function of the EDC system checks the electrical components for malfunctions. If a fault is detected, a record is permanently stored in the fault memory detailing

- the fault source (e.g. engine temperature sensor)
- the failure mode (e.g. short circuit to ground, implausible signal)
- the fault status (e.g. constantly present, sporadically present)
- the ambient conditions (readings at the time the fault was recorded, e.g. engine speed, temperature, etc.)

Using the menu option "Fault Memory", the fault data stored on the engine ECU can be transferred to the engine tester. The fault data is displayed on the tester in plain text giving details of the fault source, location, status, etc.

Fault localization

For some types of engine fault, the self-diagnosis function is unable to establish the cause. Maintenance technicians must be able to identify and rectify these types of fault quickly and reliably.

For both types of fault – those recorded in the fault memory and those that are not – the Electronic Service Information ESI[tronic] provides assistance with fault localization. It offers instructions for fault localization for every conceivable problem (e.g. engine stutter) or malfunction (e.g. engine-temperature sensor short-circuit).

Fault rectification

Once the cause of the fault has been localized with the aid of the ESI[tronic] information, the fault can be rectified.

Clearing the fault memory

When the fault has been rectified, the record of the fault must be deleted from the fault memory. This function can be performed using the menu option "Clear Fault Memory" on the engine tester.

Road test

In order to ensure that the fault has really been eliminated, a road test is carried out. During the road test, the self-diagnosis function checks the system and records any faults that are still present.

Rechecking the fault memory

After the road test, the contents of the fault memory are checked again. If the problem has been solved, no faults should have been recorded. Assuming that is the case, the repair procedure is then successfully completed.

Other testing methods

The KTS 500 system tester offers other EDC diagnosis facilities in addition to the standard diagnosis functions. The additional functions are initiated by the engine tester and then carried out by the engine ECU.

Actuator diagnosis

Many ECU functions (e.g. exhaust-gas recirculation) only operate under certain conditions when the vehicle is being driven. For this reason, it is not possible to check the function of the relevant actuators (e.g. the exhaust-gas recirculation valve) in the workshop without a means of operating them selectively.

The engine tester can be used to initiate actuator diagnosis in order to check the function of actuators in the workshop. The correct functioning of the components concerned is then indicated by audible or visual feedback.

The actuator diagnosis function tests the entire electrical path from the engine ECU via the wiring harness to the actuator. However, even that is insufficient to make a definitive statement about the complete functional capability of the component.

Actuator diagnosis is generally carried out with the vehicle stationary. It extends only to those actuators that do not affect the function of important components. This means that actuators that could be damaged or could cause engine damage if the actuator diagnosis were to be incorrectly carried out (e.g. nozzle solenoid valves) are excluded from the test.

Signal testing

Where there is a component malfunction, an oscilloscope function allows the progression of an actuator control signal to be checked over time. This applies in particular to actuators that cannot be included in the actuator diagnosis function (e.g. nozzles or injectors).

Engine test function

Faults that the self-diagnosis function cannot identify can be localized by means of supplementary functions (engine-testing functions). There are testing routines stored on the engine ECU that can be initiated by the engine tester (e.g. compression test). These tests last for a specific period of time. The start and finish of the test are indicated by the engine tester. The results are transmitted to the engine tester by the engine ECU in the form of a list.

In the case of the compression test, the fuel-injection system is deactivated while the engine is turned over by the starter motor. The engine ECU records the crankshaft speed pattern. From the fluctuations in crankshaft speed, i.e. the difference between the highest and lowest speeds, conclusions can be drawn as to the compression in individual cylinders and, therefore, about the condition of the engine.

Engine-speed and injected-fuel quantity comparison tests

Different injected-fuel quantities in different cylinders produce varying, cylinder-specific torque levels and, therefore, uneven engine running. The engine ECU measures the momentary speeds and transmits the readings to the engine tester. The engine tester then shows the speeds for each cylinder. Large divergences in speed and injected-fuel quantity between individual cylinders indicate fuel-metering problems.

The engine-smoothness function on the engine ECU evens out engine-speed fluctuations by adjustments to the injected-fuel quantity for each cylinder. For this reason, the engine-smoothness function is deactivated while the speed-comparison test is in progress.

The quantity-comparison test is carried out with the engine-smoothness function active. The engine tester displays the injected-fuel quantities for each cylinder required to produce smooth engine running.

The assessment is based on a comparison of the cylinder-specific speeds and injected-fuel quantities.

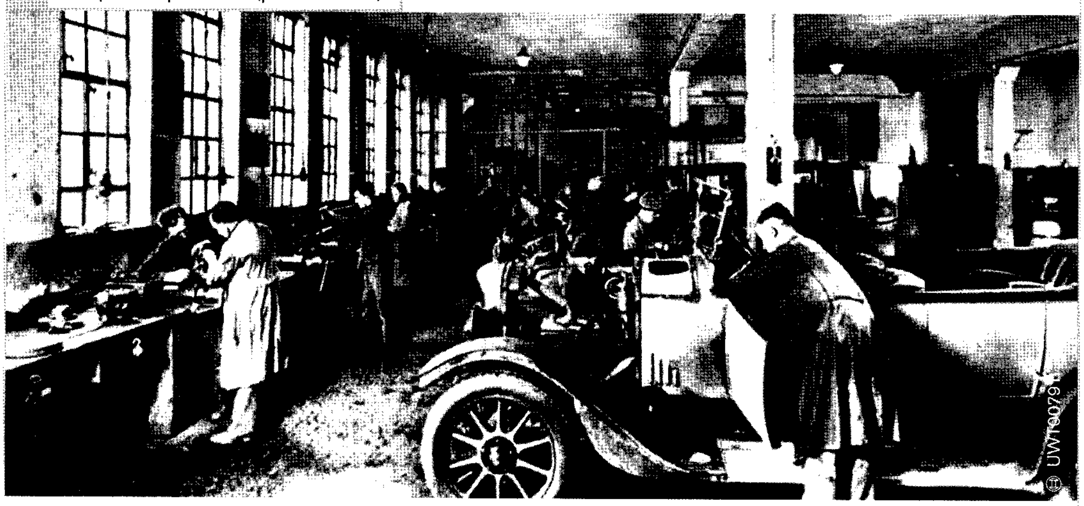
Global service

"Once you have driven an automobile, you will soon realize that there is something unbelievably tiresome about horses (...). But you do require a conscientious mechanic for the automobile (...)".

Robert Bosch wrote these words to his friend Paul Reusch in 1906. In those days, it was indeed the case that breakdowns could be repaired on the road or at home by an employed chauffeur or mechanic. However, with the growing number of motorists driving their own cars after the First World War, the need for workshops offering repair services increased rapidly. In the 1920s Robert Bosch started to systematically create a nationwide customer-service organization. In 1926 all the repair centers were uniformly named "Bosch Service" and the name was registered as a trademark.

Today's Bosch Service agencies retain the same name. They are equipped with the latest electronic equipment in order to meet the demands of 21st-century automotive technology and the quality expectations of the customers.

A repair shop in 1925 (photo: Bosch)



A Bosch service in 2001, carried out with the very latest electronic testing equipment

Fuel-injection pump test benches

Accurately tested and precisely adjusted fuel-injection pumps and governor mechanisms are key components for obtaining optimized performance and fuel economy from diesel engines. They are also crucial in ensuring compliance with increasingly strict exhaust-gas emission regulations. The fuel-injection pump test bench (Fig. 1) is a vital tool for meeting these requirements.

The main specifications governing both test bench and test procedures are defined by ISO standards; particularly demanding are the specifications for rigidity and geometrical consistency in the drive unit (5).

As time progresses, so do the levels of peak pressure that fuel-injection pumps are expected to generate. This development is reflected in higher performance demands and power requirements for pump test benches. Powerful electric drive units, a large flyweight and precise control of rotational speed guarantee stability at all engine speeds. This stability is an essential requirement for repeatable, mutually comparable measurements and test results.

Flow measurement methods

An important test procedure is to measure the fuel pumped each time the plunger moves through its stroke. For this test, the fuel-injection pump is clamped on the test bench support (1), with its drive side connected to the test bench drive coupling. Testing proceeds with a standardized calibrating oil at a precisely monitored and controlled temperature. A special, precision-calibrated nozzle-and-holder assembly (3) is connected to each pump barrel. This strategy ensures mutually comparable measurements for each test. Two test methods are available.

Glass gauge method (MGT)

The test bench features an assembly with two glass gauges (Fig. 2, Pos. 5). A range of gages with various capacities are available for each cylinder. This layout can be used to test fuel-injection pumps for engines of up to 12 cylinders.

Bosch fuel-injection pump test bench with electronic test system (KMA)

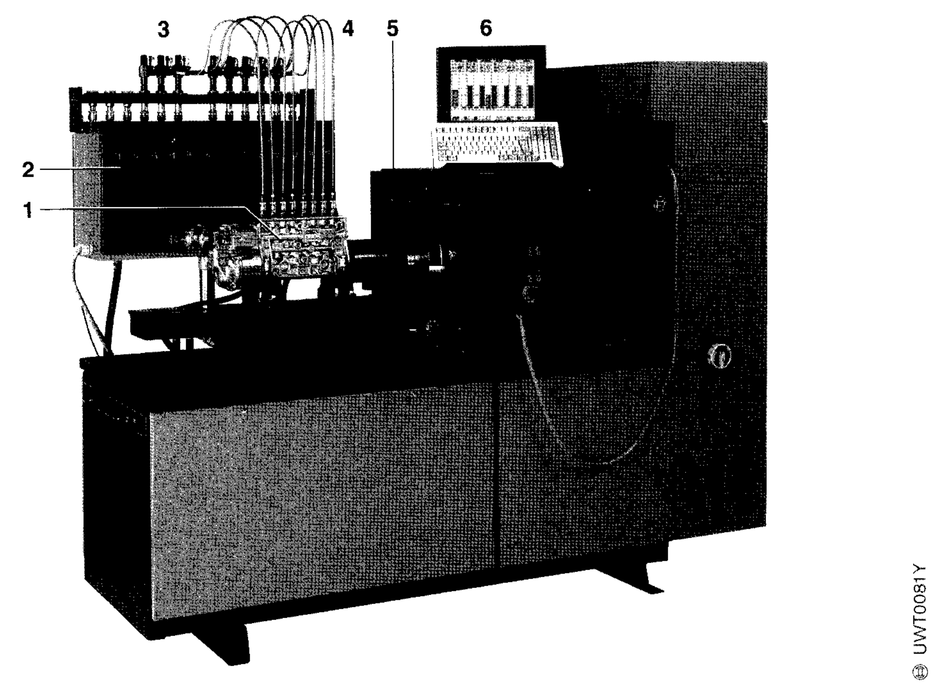
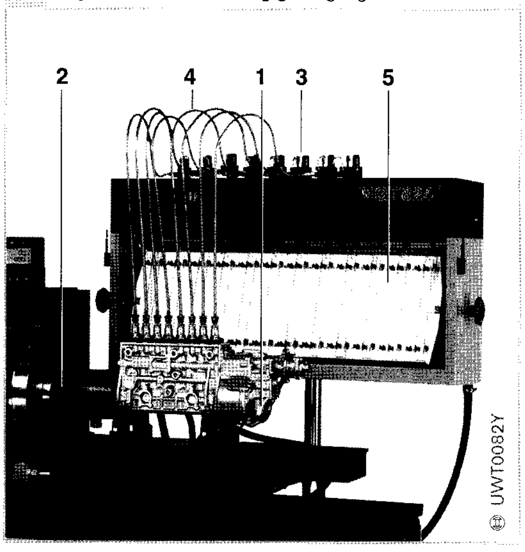


Fig. 1
1 Fuel-injection pump on test bench
2 Quantity test system (KMW)
3 Test nozzle-and-holder assembly
4 High-pressure test line
5 Electric drive unit
6 Control, display and processing unit

Layout of test stand using glass-gauge methods (MGT)



Measurement cell concept (KMA)

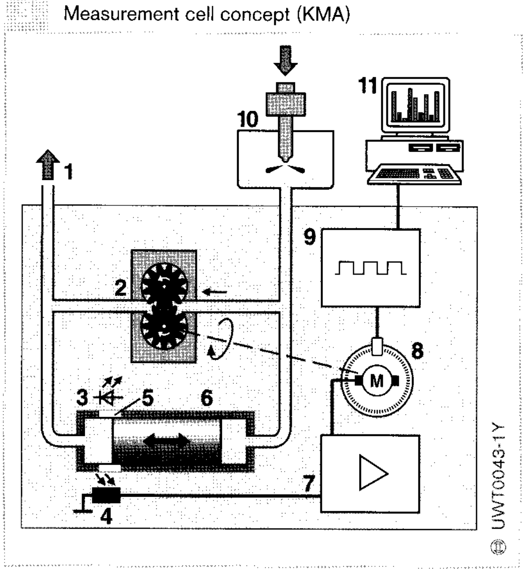


Fig. 2
1 Fuel-injection pump
2 Electric drive unit
3 Test nozzle-and-holder assembly
4 High-pressure test line
5 Glass gages

Fig. 3
1 Return line to calibrating oil tank
2 Gear pump
3 LED
4 Photocell
5 Window
6 Plunger
7 Amplifier with electronic control circuitry
8 Electric motor
9 Pulse counter
10 Test nozzle-and-holder assembly
11 Monitor (PC)

In the first stage, the discharged calibrating flows past the glass gages to return directly to the oil tank. As soon as the fuel-injection pump reaches the rotational speed indicated in the test specifications, a slide valve opens, allowing the calibrating oil from the fuel-injection pump to flow to the glass gages. Supply to the glass containers is then interrupted when the pump has executed the preset number of strokes.

The fuel quantity delivered to each cylinder in cm^3 can now be read from each of the glass gages. The standard test period is 1,000 strokes, making it easy to interpret the numerical result in mm^3 per stroke of delivered fuel. The test results are compared with the setpoint values and entered in the test record.

Electronic flow measurement system (KMA)
This system replaces the glass gauges with a control, display and processor unit (Fig. 1, Pos. 6). While this unit is usually mounted on the test bench, it can also be installed on a cart next to the test bench.

This test relies on continuous measuring the delivery capacity (Fig. 3). A control plunger (6) is installed in parallel with the input and output sides of a gear pump (2). When the pump's delivery quantity equals the quantity of calibrating oil emerging from the test nozzle (10), the plunger remains in its center position. If the flow of calibrating oil is greater, the plunger moves to the left – if the flow of calibrating oil is lower, the plunger moves to the right. This plunger motion controls the amount of light traveling from an LED (3) to a photocell (4). The electronic control circuitry (7) records this deviation and responds by varying the pump's rotational speed until its delivery rate again corresponds to the quantity of fluid emerging from the test nozzle. The control plunger then returns to its center position. The pump speed can be varied to measure delivery quantity with extreme precision.

Two of these measurement cells are present on the test bench. The computer connects all of the test cylinders to the two measurement cells in groups of two, proceeding sequentially from one group to the next (multiplex operation). The main features of this test method are:

- Highly precise and reproducible test results
- Clear test results with digital display and graphic presentation in the form of bar graphs
- Test record for documentation
- Supports adjustments to compensate for variations in cooling and/or temperature

Testing helix and port-controlled distributor injection pumps

Good engine performance, high fuel economy and low emissions depend on correct adjustment of the helix and port-controlled distributor injection pump. This is why compliance with official specifications is absolutely essential during testing and adjustment operations on fuel-injection pumps.

One important parameter is the start of delivery (in service bay), which is checked with the pump installed. Other tests are conducted on the test bench (in test area). In this case, the pump must be removed from the vehicle and mounted on the test bench. Before the pump is removed, the engine crankshaft should be rotated until the reference cylinder is at TDC. The reference cylinder is usually cylinder No. 1. This step eases subsequent assembly procedures.

Test bench measurements

The test procedures described here are suitable for use on helix and port-controlled axial-piston distributor pumps with electronic and mechanical control, but not with solenoid-controlled distributor injection pumps.

Test bench operations fall into two categories:

- Basic adjustment and
- Testing

The results obtained from the pump test are entered in the test record, which also lists all the individual test procedures. This document also lists all specified minimum and maximum results. The test readings must lie within the range defined by these two extremes.

A number of supplementary, special-purpose test steps are needed to assess all the different helix and port-controlled axial-piston distributor pumps; detailed descriptions of every contingency, however, extend beyond the bounds of this chapter.

Equipment for testing the distributor injection pump (on test bench)

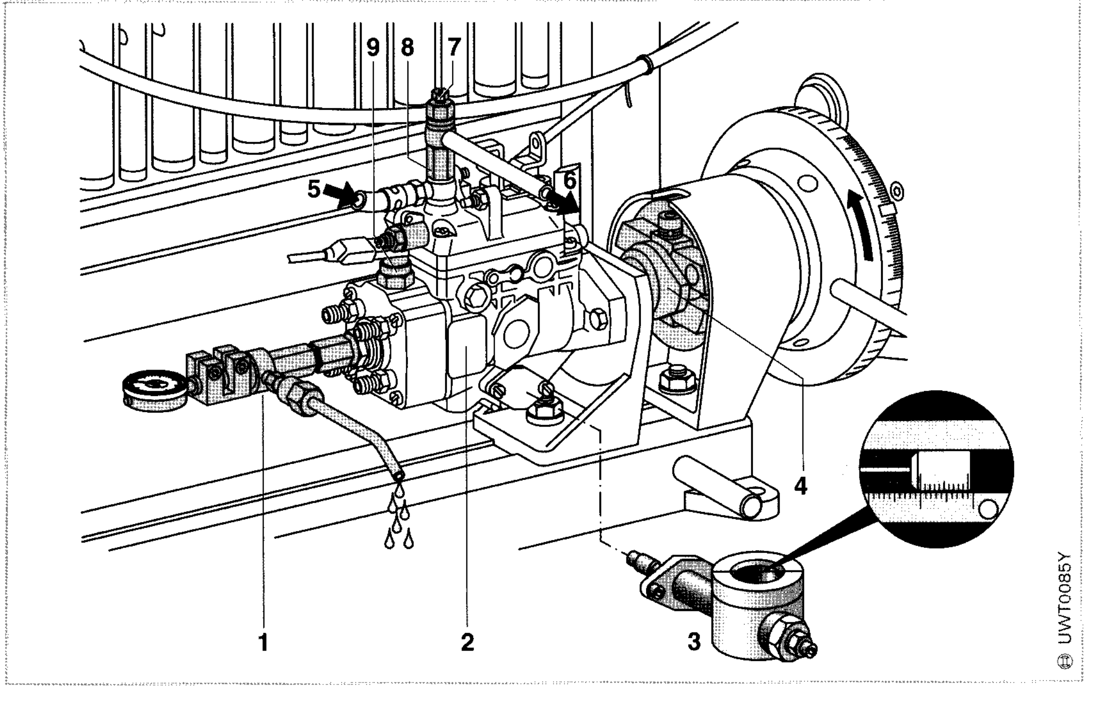


Fig. 1

1 Test layout with drain hose and dial gauge
2 Distributor injection pump
3 Timing device travel tester with vernier scale
4 Pump drive
5 Calibrating oil inlet
6 Return line
7 Overflow restrictor
8 Adapter with connection for pressure gauge
9 Electric shutoff valve (ELAB) (energized)

Basic adjustments

The first step is to adjust the distributor injection pump to the correct basic settings. This entails measuring the following parameters under defined operating conditions.

LPC adjustment

This procedure assesses the distributor plunger lift between bottom dead center (BDC) and the start of delivery. The pump must be connected to the test-bench fuel supply line for this test. The technician unscrews the 6-point bolt from the central plug fitting and then installs a test assembly with drain tube and gauge in its place (Fig. 1, Pos. 1).

The gauge probe rests against the distributor plunger, allowing it to measure lift. Now the technician turns the pump's input shaft (4) by hand until the needle on the gauge stops moving. The control plunger is now at top dead center (TDC).

A supply pressure of roughly 0.5 bar propels the calibrating oil into the plunger chamber behind the distributor plunger (5). For this test, the solenoid-operated shutoff valve (ELAB) (9) is kept energized to maintain it in its open position. The calibrating oil thus flows from the plunger chamber to the test assembly before emerging from the drain hose.

Now the technician manually rotates the input shaft in its normal direction of rotation. The calibrating oil ceases to flow into the plunger chamber once its inlet passage closes. The oil remaining in the chamber continues to emerge from the drain hose. This point in the distributor plunger's travel marks the start of delivery.

The lift travel between bottom dead center (BDC) and the start of delivery indicated by the gauge can now be compared with the setpoint value. If the reading is outside the tolerance range, it will be necessary to dismantle the pump and replace the cam mechanism between cam disk and plunger.

Supply-pump pressure

As it affects the timing device, the pressure of the supply pump (internal pressure) must also be tested. For this procedure, the overflow restrictor (7) is unscrewed and an adapter with a connection to the pressure gauge (8) is installed. Now the overflow restrictor is installed in an adapter provided in the test assembly. This makes it possible to test the pump's internal chamber pressure upstream of the restrictor.

A plug pressed into the pressure-control valve controls the tension on its spring to determine the pump's internal pressure. Now the technician continues pressing the plug into the valve until the pressure reading corresponds to the setpoint value.

Timing device travel

The technician removes the cover from the timing device to gain access for installing a travel tester with a vernier scale (3). This scale makes it possible to record travel in the timing device as a function of rotational speed; the results can then be compared with the setpoint values. If the measured timing device travel does not correspond with the setpoint values, shims must be installed under the timing spring to correct its initial spring tension.

Adjusting the basic delivery quantity

During this procedure the fuel-injection pump's delivery quantity is adjusted at a constant rotational speed for each of the following four conditions:

- Idle (no-load)
- Full-load
- Full-load governor regulation and
- Starting

Delivery quantities are monitored using the MGT or KMA attachment on the fuel-injection pump test bench (refer to section on "Fuel-injection pump test benches").

First, with the control lever's full-load stop adjusted to the correct position, the full-load governor screw in the pump cover is adjusted to obtain the correct full-load delivery quantity at a defined engine speed. Here, the governor adjusting screw must be turned back to prevent the full-load stop from reducing delivery quantity.

The next step is to measure the delivery quantity with the control lever against the idle-speed stop screw. The idle-speed stop screw must be adjusted to ensure that the monitored delivery quantity is as specified.

The governor screw is adjusted at high rotational speed. The measured delivery quantity must correspond to the specified full-load delivery quantity.

The governor test also allows verification of the governor's intervention speed. The governor should respond to the specified rpm threshold by first reducing and then finally interrupting the fuel flow. The breakaway speed is set using the governor speed screw.

There are no simple ways to adjust the delivery quantity for starting. The test conditions are a rotational speed of 100 rpm and the control lever against its full-load shutoff stop. If the measured delivery quantity is below a specified level, reliable starting cannot be guaranteed.

Testing

Once the basic adjustment settings have been completed, the technician can proceed to assess the pump's operation under various conditions. As during the basic adjustment procedure, testing focuses on

- Supply-pump pressure
- Timing device travel
- Delivery quantity curve

The pump operates under various specific conditions for this test series, which also includes a supplementary procedure.

Overflow quantity

The vane-type supply pump delivers more fuel than the nozzles can inject. The excess calibrating oil must flow through the overflow restriction valve and back to the oil tank. It is the volume of this return flow that is measured in this procedure. A hose is connected to the overflow restriction valve; depending on the selected test procedure. The other end is then placed in a glass gauge in the MGT assembly, or installed on a special connection on the KMA unit. The overflow quantity from a 10-second test period is then converted to a delivery quantity in liters per hour.

If the test results fail to reach the setpoint values, this indicates wear in the vane-type supply pump, an incorrect overflow valve or internal leakage.

Dynamic testing of start of delivery

A diesel engine tester (such as the Bosch ETD 019.00) allows precise adjustment of the distributor injection pump's delivery timing on the engine. This unit registers the start of delivery along with the timing adjustments that occur at various engine speeds with no need to disconnect any high-pressure delivery lines.

Testing with piezo-electric sensor and stroboscopic timing light

The piezo-electric sensor (Fig. 2, Pos. 4) is clamped onto the high-pressure delivery line leading to the reference cylinder. Here, it is important to ensure that the sensor is mounted on a straight and clean section of tubing with no bends; the sensor should also be positioned as close as possible to the fuel-injection pump.

The start of delivery triggers pulses in the fuel-injection line. These generate an electric signal in the piezo-electric sensor. The signal controls the light pulses generated by the timing light (5). The timing light is now aimed at the engine's flywheel. Each time the pump starts delivery to the reference cylinder, the timing light flashes, lighting up the TDC mark on the flywheel. This allows correlation of timing to flywheel position. The flashes occur only when delivery to the reference cylinder starts, producing a static image. The degree markings (6) on the crankshaft or flywheel show the crankshaft position relative to the start of delivery.

Engine speed is also indicated on the diesel engine tester.

Setting start of delivery

If the results of this start-of-delivery test deviate from the test specifications, it will be necessary to change the fuel-injection pump's angle relative to the engine.

The first step is to switch off the engine. Then the technician rotates the crankshaft until the reference cylinder's piston is at the point at which delivery should start. The crankshaft features a reference mark for this operation; the mark should be aligned with the corresponding mark on the bellhousing. The technician now unscrews the 6-point screw from the central plug screw. As for basic adjustment process on the test bench, the technician now installs a dial-gauge assembly in the opening. This is used to observe distributor plunger travel while the crankshaft is being turned. As the crankshaft is turned counter to its normal direction of rotation (or in the normal direction on some engines), the plunger retracts in the pump. The technician should stop turning the crankshaft once the needle on the gauge stops moving. The plunger is now at bottom dead center. Now the dial gauge is reset to zero. The crankshaft is then rotated in its normal direction of rotation as far as the TDC mark. The dial gauge now indicates the travel executed by the distributor plunger on its way from its bottom dead center position to the TDC mark on the reference cylinder. It is vital to comply with the precise specification figure for this travel contained in the fuel-injection pump's datasheet. If the dial gauge reading is not within the specification, it will be necessary to loosen the attachment bolt on the pump flange, turn the pump housing and repeat the test. It is important to ensure that the cold-start accelerator is not active during this procedure.

Measuring the idle speed

The idle speed is monitored with the engine heated to its normal operating temperature, and in a no-load state, using the engine tester. The idle speed can be adjusted using the idle-speed stop screw.

Checking start of delivery with piezo-electric sensor and timing light

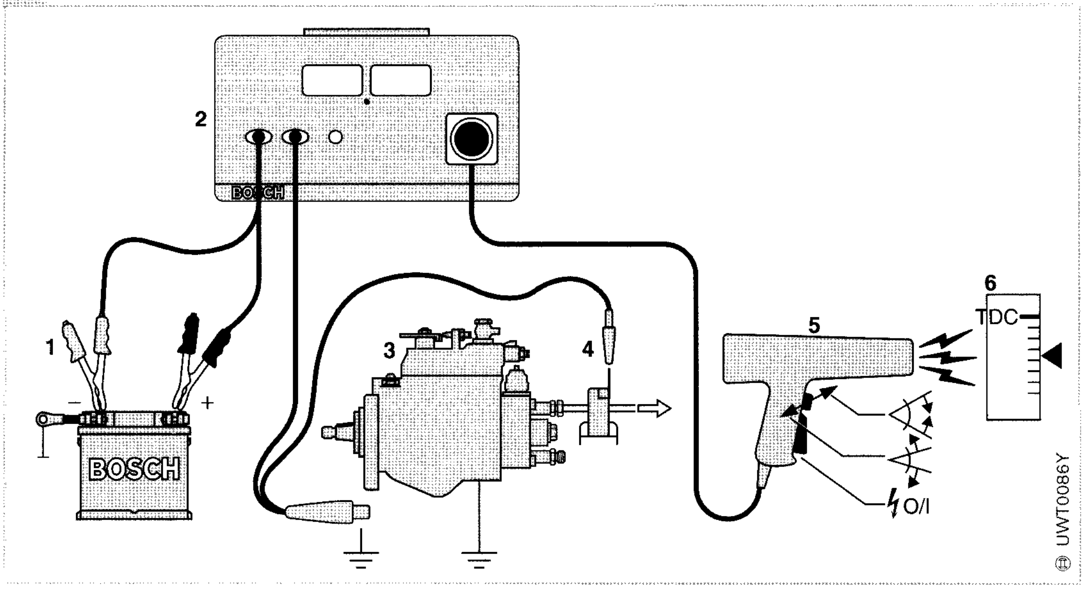


Fig. 2
1 Battery
2 Diesel tester
3 Distributor injection pump
4 Piezo-electric sensor
5 Stroboscopic timing light
6 Angle and TDC marks

Nozzle tests

The nozzle-and-holder assembly consists of the nozzle and the holder. The holder includes all of the required filters, springs and connections.

The nozzle affects the diesel engine's output, fuel economy, exhaust-gas composition and operating refinement. This is why the nozzle test is so important.

An important tool for assessing nozzle performance is the nozzle tester.

Nozzle tester

Keep your hands away from the nozzle jet. Spray from the nozzle stings and penetrates the skin. *There is a risk of blood poisoning.*

Wear safety goggles.

The nozzle tester is basically a manually operated fuel-injection pump (Fig. 1). For testing, a high-pressure delivery line (4) is used to connect the nozzle-and-holder assembly (3) to the tester. The calibrating oil is contained in a tank (5). The required pressure is generated using the hand lever (8). The pressure gage (6) indicates the pressure of the calibrating oil; a valve (7) can be used to disconnect it from the high-pressure circuit for specific test procedures.

Nozzle tester with nozzle-and-holder assembly

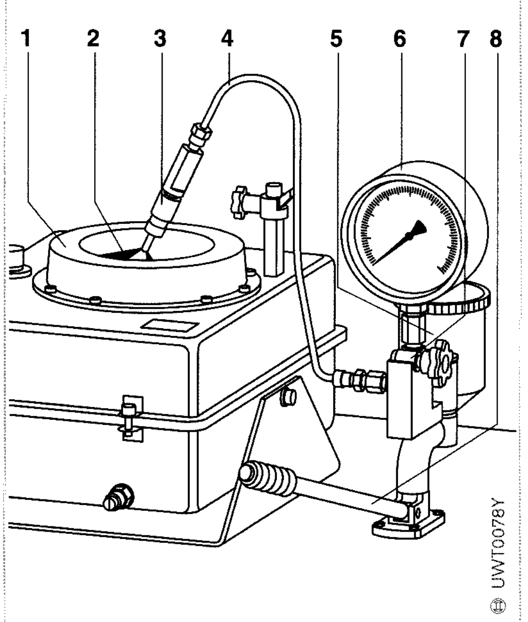


Fig. 1
1 Suction equipment
2 Injection jet
3 Nozzle-and-holder assembly
4 High-pressure test line
5 Calibrating oil tank with filter
6 Pressure gage
7 Valve
8 Hand lever

The EPS100 (0684200704) nozzle tester is specified for testing nozzles of Sizes P, R, S and T. It conforms to the standards defined in ISO 8984. The prescribed calibrating oil is defined in ISO standard 4113. A calibration case containing all the components is required to calibrate inspect the nozzle tester.

This equipment provides the basic conditions for reproducible, mutually compatible test results.

Test methods

Ultrasonic cleaning is recommended for the complete nozzle-and-holder assemblies once they have been removed from the engine. Cleaning is mandatory on nozzles when they are submitted for warranty claims.

Important: Nozzles are high-precision components. Careful attention to cleanliness is vital for ensuring correct operation.

The next step is to inspect the assembly to determine whether any parts of the nozzle or holder show signs of mechanical or thermal wear. If signs or wear are present, it will be necessary to replace the nozzle or nozzle-and-holder assembly.

The assessment of the nozzle's condition proceeds in four test steps, with some variation depending on whether the nozzles are pintle or hole-type units.

Chatter test

The chatter test provides information on the smoothness of action of the needle. During injection, the needle oscillates back and forth to generate a typical chatter. This motion ensures efficient dispersion of the fuel particles.

The pressure gage should be disconnected for this test (close valve).

Pintle nozzle

The lever on the nozzle tester is operated at a rate of one to two strokes per second. The pressure of the calibrating oil rises, ultimately climbing beyond the nozzle's opening pressure. During the subsequent discharge, the nozzle should produce an audible chatter; if it fails to do so, it should be replaced.

When installing a new nozzle in its holder, always observe the official torque specifications, even on hole-type nozzles.

Hole-type nozzle
The hand lever is pumped at high speed. This produces a hum or whistling sound, depending on the nozzle type. No chatter will be present in some ranges. Evaluation of chatter is difficult with hole-type nozzles. This is why the chatter test is no longer assigned any particular significance as an assessment tool for hole-type nozzles.

Spray pattern test
High pressures are generated during this test. Always wear safety goggles.

The hand lever is subjected to slow and even pressure to produce a consistent discharge plume. The spray pattern can now be evaluated. It provides information on the condition of the injection orifices. The prescribed response to an unsatisfactory spray pattern is to replace the nozzle or nozzle-and-holder assembly.

The pressure gage should also be switched off for this test.

Pintle nozzle
The spray should emerge from the entire periphery of the injection orifice as even tapered plume. There should be no concentration on one side (except with flatted pintle nozzles).

Hole-type nozzle
An even tapered plume should emerge from each injection orifice. The number of individual plumes should correspond to the number of orifices in the nozzle.

Checking the opening pressure
Once the line pressure rises above the opening pressure, the valve needle lifts from its seat to expose the injection orifice(s). The specified opening pressure is vital for correct operation of the overall fuel-injection system.

The pressure gage must be switched back on for this test (valve open).

Pintle nozzle and hole-type nozzle with single-spring nozzle holder
The operator slowly presses the lever downward, continuing until the gage needle indicates the highest available pressure. At this point, the valve opens and the nozzle starts to discharge fuel. Pressure specifications can be found in the “nozzles and nozzle-holder components” catalog.

Opening pressures can be corrected by replacing the adjustment shim installed against the compression spring in the nozzle holder. This entails extracting the nozzle from the nozzle holder. If the opening pressure is too low, a thicker shim should be installed; the response to excessive opening pressures is to install a thinner shim.

Hole-type nozzle with two-spring nozzle holder
This test method can only be used to determine the initial opening pressure on two-spring nozzle-and-holder assemblies.

The is no provision for shim replacement on some nozzle-and-holder assemblies. The only available response with these units is to replace the entire assembly.

Leak test
The pressure is set to 20 bar above the opening pressure. After 10 seconds, formation of a droplet at the injection orifice is acceptable, provided that the droplet does not fall.

The prescribed response to an unsuccessful leak test is to replace the nozzle or nozzle-and-holder assembly.

Index of technical terms

Technical Terms

Abbreviations

A

A pump: In-line fuel-injection pump size **A**

ADA: Atmospheric-pressure sensitive full-load stop (German: **A**tmosphären**d**ruck**a**bhängiger Volllastanschlag)

ALDA: Manifold-pressure compensator (German: **L**ade**d**ruck**a**bhängiger Volllastanschlag, **a**bsolut messend)

ALFB: Load-dependent start of delivery with deactivation feature (German: **A**bschaltbarer, **l**astabhängiger **F**örder**b**eginn)

ASIC: **A**pplication-**S**pecific **I**ntegrated **C**ircuit

ASR: Traction control system (German: **A**ntriebs**s**chlupf**r**egelung)

AWN: Bosch workshop network

B

BDC: **B**ottom **D**ead **C**enter (piston/crankshaft)

bhp: **B**rake **h**orse **p**ower

BIP-Signal: **B**egin of **I**njection **P**eriod-**Signal**

C

CA: Camshaft

CAN: Controller **A**rea **N**etwork

cks: Cran**ks**haft

CR System: Common-**R**ail **System**

CRS: Common-**R**ail **S**ystem

CW pump: In-line fuel-injection pump size **CW**

D

DDS: Diesel-engine immobilizers (German: **D**iesel-**D**iebstahl-**S**chutz)

DHK: Nozzle-and-holder assembly (German: **D**üsen**h**alter**k**ombination)

DI: Direct **I**njection

DMV: Diesel solenoid valve (German: **D**iesel-**M**agnet**v**entil)

DWS: Angle-of-rotation sensor (German: **D**reh**w**inkel**s**ensor)

DZG: Speed sensor (German: **D**reh**z**ahl**g**eber (Drehzahlsensor))

E

EAB: Solenoid-operated shutoff valve (ELAB)

EC: End of **c**ombustion

ECM: Electro**c**hemical **M**achining (Hole-type nozzles)

ECU: Electronic **C**ontrol **U**nit

EDC: Electronic **D**iesel **C**ontrol

EDR: Maximum-speed governor (German: **E**nd**d**rehzahl**r**egelung)

EGS: Electronic transmission control (German: **E**lektronische **G**etriebe**s**teuerung)

EGR: Exhaust-**g**as **r**ecirculation

EI: End of **i**njection

ELAB: Solenoid-operated shutoff valve (German: **El**ektrisches **Ab**stellventil)

EoL programming: End **o**f **L**ine **programming**

ESI[tronic]: Electronic **S**ervice **I**nformation

ESP: Electronic **S**tability **P**rogram

F

FD: Delivery period (German: **F**örder**d**auer)

FGB: Vehicle-speed limitation (German: **F**ahr**g**eschwindigkeits**b**egrenzung)

FGR: Cruise control (German: **F**ahr**g**eschwindigkeits**r**egelung)

FSS: Delivery-signal sensor (German: **F**örder**s**ignal**s**ensor)

G

GDV: Constant-pressure valve (German: **G**leich**d**ruck**v**entil)

GSK: Glow plug (German: **G**lüh**s**tift**k**erze)

GST: Graduated (or adjustable) **st**art quantity

GZS: Glow plug control unit (German: **G**lüh**z**eit**s**teuergerät)

H

H pump: In-line control-sleeve injection pump (German: Hubschieber-Reiheneinspritzpumpe)

HBA: Hydraulically controlled torque control device (German: Hydraulisch betätigte Angleichung)

HDK: Sensor with semidifferential short-circuiting ring (German: Halb-Differenzial-Kurzschlussring)

HE process: Hydro-erosion process (Hole-type nozzles)

HGB: Maximum-speed limiter (German: Höchstgeschwindig-keitsbegrenzung)

I

IDI: Indirect Injection

IGL: Ignition lag

IL: Injection lag

ISO: International Organization for Standardization

IWZ system: Incremental angle-time system (German: Inkrementales-Winkel-Zeit-System)

K

KMA: Electronic flow measurement system (German: Kontinuierliche Mengenanalyse)

KSB: Cold-start accelerator (German: Kaltstartbeschleuniger)

KTS: Small tester series (German: Klein-Tester-Serie)

kW: Kilowatt (= 1.3596 bhp)

L

LDA: Manifold-pressure compensator (German: Ladedruckabhängiger Volllastanschlag)

LED: Light-Emitting Diode

LFB: Load-sensitive start of delivery (German: Lastabhängiger Förderbeginn)

LFG: Idle-speed spring attached to governor housing (German: Leerlauffeder – gehäusefest)

LTCC: Low Temperature Cofired Ceramic

M

M pump: In-line fuel-injection pump size **M**

MAB: Fuel cutout (German: Mengenabschaltung)

MBEG: Fuel limitation (German: Mengenbegrenzung)

MGT: Glass gauge method (German: Messglas-Technik)

MPC: Manifold-pressure compensator

MI: Main Injection

MSG: Engine ECU (German: Motorsteuergerät, Motor-ECU)

MV: Solenoid valve (German: Magnetventil)

MW pump: In-line fuel-injection pump size **MW**

N

NBS: Needle-motion sensor (German: Nadelbewegungssensor)

NLK: Hydraulically assisted timing device (German: Nachlaufkolben-Spritzversteller)

P

P pump: In-line fuel-injection pump size **P**

PE pump: In-line fuel-injection pump (German: Reiheneinspritzpumpe mit eigener Nockenwelle)

PF pump: Discrete fuel-injection pump (German: Einzeleinspritzpumpe mit Fremdantrieb)

PI: Pre-Injection

PLA: Pneumatic idle-speed increase (German: Pneumatische Leerlaufanhebung)

PO: Post-Injection

ppm: Parts per million (1,000 ppm = 0,1 %)

PSG: Pump ECU (German: Pumpensteuergerät)

PWM: Pulse-Width modulation signal

R

R pump: In-line fuel-injection pump size **R**

RDV: Orifice check valve (German: Rückströmdrosselventil)

RSD: Orifice check valve (German: Rückströmdrosselventil)

RWG: Control-rack travel sensor (German: Regelweggeber)

S

SC: Start of combustion

SD: Start of delivery

SI: Start of injection

T

TD signal: Engine-speed signal

TLA: Temperature-controlled idle-speed increase (German: Temperaturabhängige Leerlaufanhebung)

TDC: Top Dead Center (piston/crankshaft)

TQ signal: Fuel-consumption signal

U

UIS: Unit Injector System

UPS: Unit Pump System

V

VE..MV: Solenoid-valve-controlled injection pump

VE pump: Axial-piston distributor pump (German: Axialkolben-Verteilereinspritzpumpe)

VP15, VP34, VP36, VP37: Port-controlled injection pump

VP29, VP30: Solenoid-valve-controlled injection pump

VP44: Radial-piston distributor pump

VR pump: Radial-piston distributor pump (German: Radialkolben-Verteilereinspritzpumpe)

VTG: Variable Turbine Geometry

W

WOT: Wide-open throttle

Z

ZDR: Intermediate-speed control (German: Zwischendrehzahl-regelung)

ZW pump: In-line fuel-injection pump size **ZW**

ZW(M) pump: In-line fuel-injection pump size **ZW** for multifuel operation

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