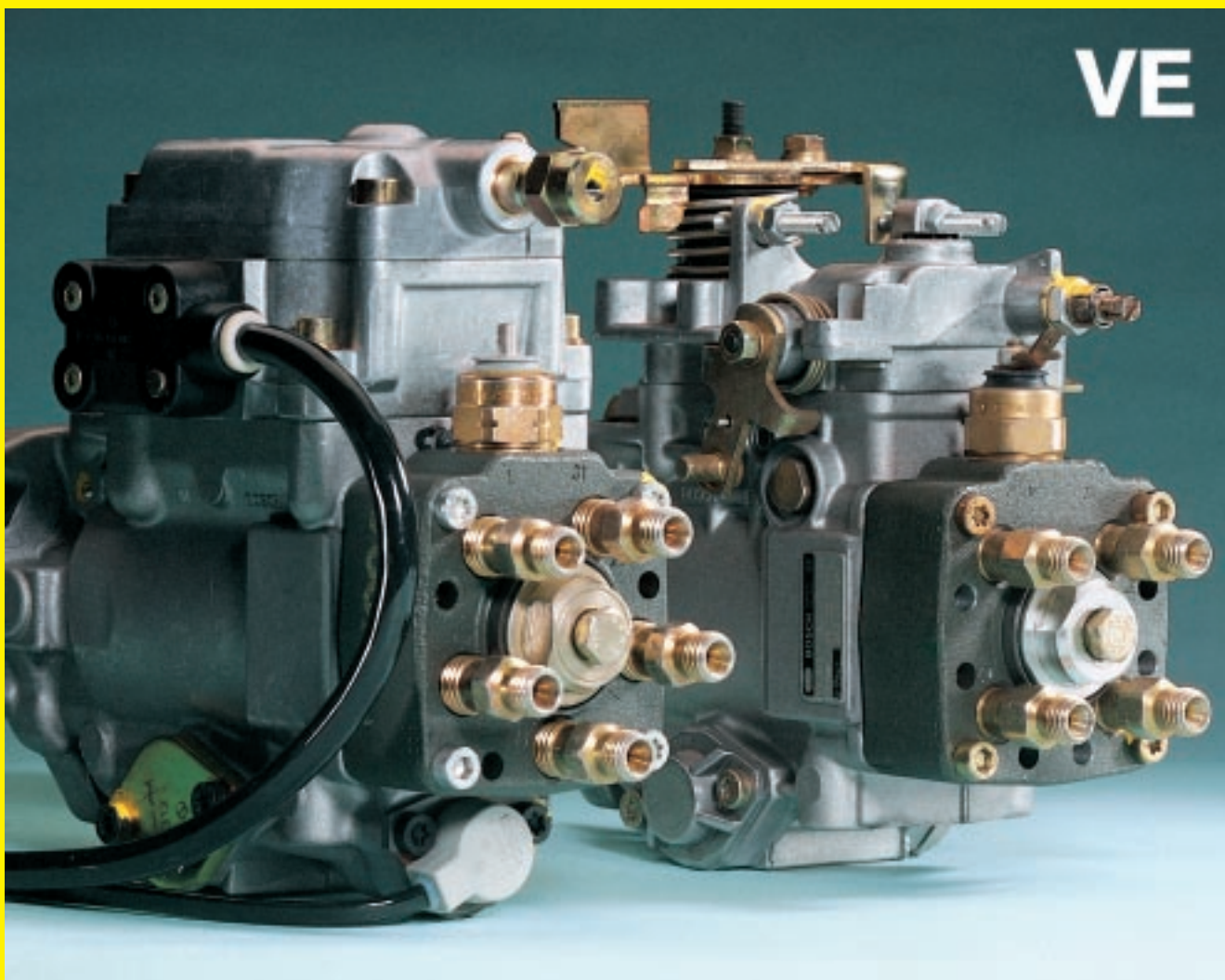


Diesel-engine management

Diesel distributor fuel-injection pumps



Technical Instruction



BOSCH

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Diesel distributor fuel-injection pumps VE

The reasons behind the diesel-powered vehicle's continuing success can be reduced to one common denominator: Diesels use considerably less fuel than their gasoline-powered counterparts. And in the meantime the diesel has practically caught up with the gasoline engine when it comes to starting and running refinement. Regarding exhaust-gas emissions, the diesel engine is just as good as a gasoline engine with catalytic converter. In some cases, it is even better. The diesel engine's emissions of CO₂, which is responsible for the "green-house effect", are also lower than for the gasoline engine, although this is a direct result of the diesel engine's better fuel economy. It was also possible during the past few years to considerably lower the particulate emissions which are typical for the diesel engine.

The popularity of the high-speed diesel engine in the passenger car though, would have been impossible without the diesel fuel-injection systems from Bosch. The very high level of precision inherent in the distributor pump means that it is possible to precisely meter extremely small injection quantities to the engine. And thanks to the special governor installed with the VE-pump in passenger-car applications, the engine responds immediately to even the finest change in accelerator-pedal setting. All points which contribute to the sophisticated handling qualities of a modern-day automobile.

The Electronic Diesel Control (EDC) also plays a decisive role in the overall improvement of the diesel-engined passenger car.

The following pages will deal with the design and construction of the VE distributor pump, and how it adapts injected fuel quantity, start-of-injection, and duration of injection to the different engine operating conditions.

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Combustion in the diesel engine

The diesel engine

Diesel combustion principle

The diesel engine is a compression-ignition (CI) engine which draws in air and compresses it to a very high level. With its overall efficiency figure, the diesel engine rates as the most efficient combustion engine (CE). Large, slow-running models can have efficiency figures of as much as 50% or even more.

The resulting low fuel consumption, coupled with the low level of pollutants in the exhaust gas, all serve to underline the diesel engine's significance.

The diesel engine can utilise either the 4- or 2-stroke principle. In automotive applications though, diesels are practically always of the 4-stroke type (Figs. 1 and 2).

Working cycle (4-stroke)

In the case of 4-stroke diesel engines, gas-exchange valves are used to control the gas exchange process by opening and closing the inlet and exhaust ports.

Induction stroke

During the first stroke, the downward movement of the piston draws in unthrottled air through the open intake valve.

Compression stroke

During the second stroke, the so-called compression stroke, the air trapped in the cylinder is compressed by the piston which is now moving upwards. Compression ratios are between 14:1 and 24:1. In the process, the air heats up to temperatures around 900°C. At the end of the compression stroke the nozzle injects fuel into the heated air at pressures of up to 2,000 bar.

Power stroke

Following the ignition delay, at the beginning of the third stroke the finely atomized fuel ignites as a result of auto-ignition and burns almost completely. The cylinder charge heats up even further and the cylinder pressure increases again. The energy released by the ignition is applied to the piston.

The piston is forced downwards and the combustion energy is transformed into mechanical energy.

Exhaust stroke

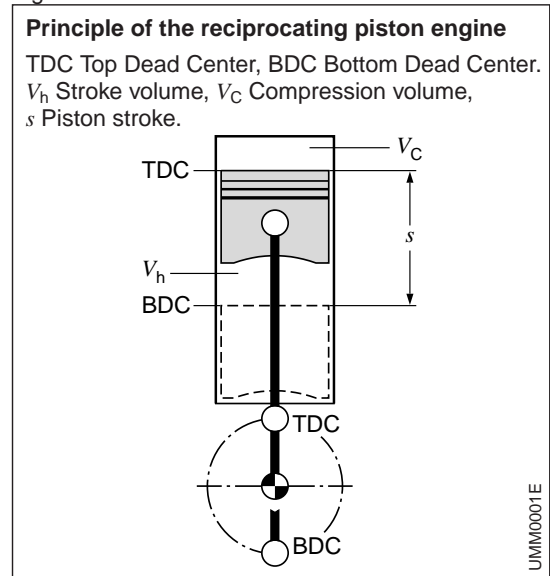
In the fourth stroke, the piston moves up again and drives out the burnt gases through the open exhaust valve.

A fresh charge of air is then drawn in again and the working cycle repeated.

Combustion chambers, turbocharging and supercharging

Both divided and undivided combustion chambers are used in diesel engines

Fig. 1



(prechamber engines and direct-injection engines respectively).

Direct-injection (DI) engines are more efficient and more economical than their prechamber counterparts. For this reason, DI engines are used in all commercial-vehicles and trucks. On the other hand, due to their lower noise level, prechamber engines are fitted in passenger cars where comfort plays a more important role than it does in the commercial-vehicle sector. In addition, the prechamber diesel engine features considerably lower toxic emissions (HC and NO_x), and is less costly to produce than the DI engine. The fact though that the prechamber engine uses slightly more fuel than the DI engine (10...15%) is leading to the DI engine coming more and more to the forefront. Compared to the gasoline engine, both diesel versions are more economical especially in the part-load range.

Diesel engines are particularly suitable for use with exhaust-gas turbochargers or mechanical superchargers. Using an exhaust-gas turbocharger with the diesel engine increases not only the power yield, and with it the efficiency, but also reduces the combustion noise and the toxic content of the exhaust gas.

Diesel-engine exhaust emissions

A variety of different combustion deposits are formed when diesel fuel is burnt.

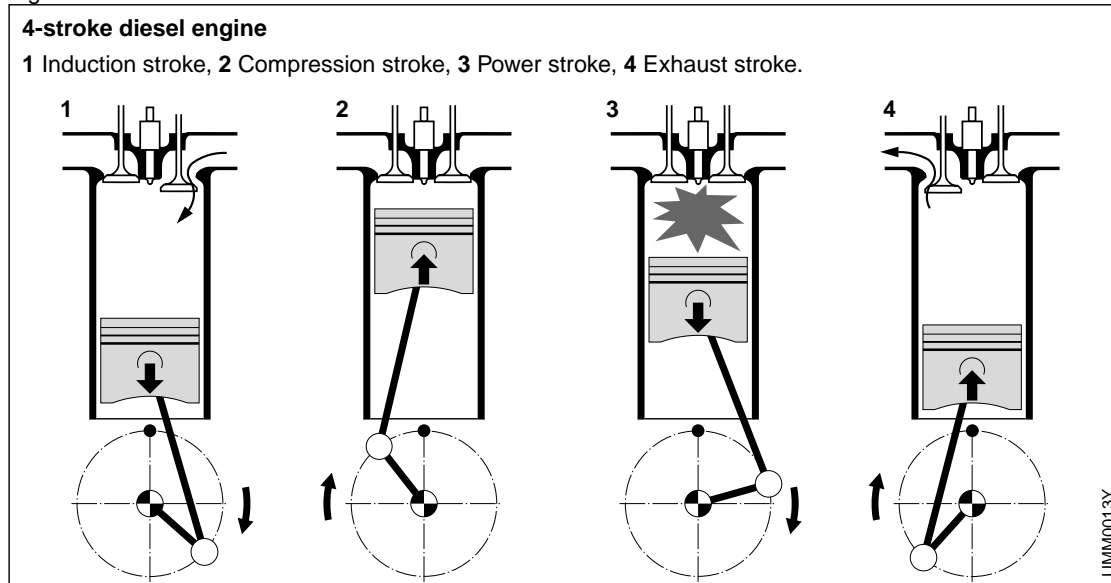
These reaction products are dependent upon engine design, engine power output, and working load.

The complete combustion of the fuel leads to major reductions in the formation of toxic substances. Complete combustion is supported by the careful matching of the air-fuel mixture, absolute precision in the injection process, and optimum air-fuel mixture turbulence. In the first place, water (H_2O) and carbon dioxide (CO_2) are generated. And in relatively low concentrations, the following substances are also produced:

- Carbon monoxide (CO),
- Unburnt hydrocarbons (HC),
- Nitrogen oxides (NO_x),
- Sulphur dioxide (SO_2) and sulphuric acid (H_2SO_4), as well as
- Soot particles.

When the engine is cold, the exhaust-gas constituents which are immediately noticeable are the non-oxidized or only partly oxidized hydrocarbons which are directly visible in the form of white or blue smoke, and the strongly smelling aldehydes.

Fig. 2



Diesel fuel-injection systems: An overview

Fields of application

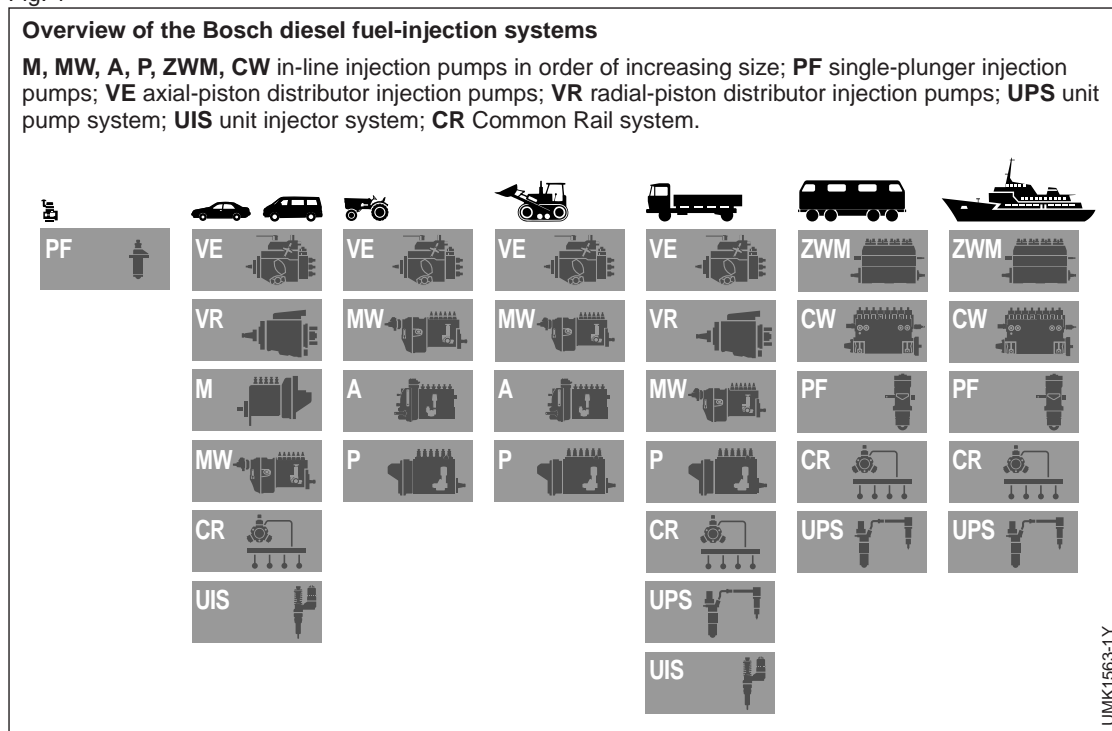
Diesel engines are characterized by their high levels of economic efficiency. This is of particular importance in commercial applications. Diesel engines are employed in a wide range of different versions (Fig. 1 and Table 1), for example as:

- The drive for mobile electric generators (up to approx. 10 kW/cylinder),
- High-speed engines for passenger cars and light commercial vehicles (up to approx. 50 kW/cylinder),
- Engines for construction, agricultural, and forestry machinery (up to approx. 50 kW/cylinder),
- Engines for heavy trucks, buses, and tractors (up to approx. 80 kW/cylinder),
- Stationary engines, for instance as used in emergency generating sets (up to approx. 160 kW/cylinder),
- Engines for locomotives and ships (up to approx. 1,000 kW/cylinder).

Technical requirements

More and more demands are being made on the diesel engine's injection system as a result of the severe regulations governing exhaust and noise emissions, and the demand for lower fuel-consumption. Basically speaking, depending on the particular diesel combustion process (direct or indirect injection), in order to ensure efficient air/fuel mixture formation, the injection system must inject the fuel into the combustion chamber at a pressure between 350 and 2,050 bar, and the injected fuel quantity must be metered with extreme accuracy. With the diesel engine, load and speed control must take place using the injected fuel quantity without intake-air throttling taking place. The mechanical (flyweight) governing principle for diesel injection systems is in-

Fig. 1



creasingly being superseded by the Electronic Diesel Control (EDC). In the passenger-car and commercial-vehicle sector, new diesel fuel-injection systems are all EDC-controlled.

According to the latest state-of-the-art, it is mainly the high-pressure injection systems listed below which are used for motor-vehicle diesel engines.

Fields of application, Technical requirements

Table 1
Diesel fuel-injection systems: Properties and characteristic data

Fuel-injection system Type	Injection				Engine-related data			
	Injected fuel quantity per stroke mm ³	Max. nozzle pressure bar	Mechanical Electronic Electromechanical Solenoid valve m e em MV	Direct injection Indirect injection DI IDI	Pilot injection Post injection VE NE	No. of cylinders	Max. speed min ⁻¹	Max. power per cylinder kW
In-line injection pumps								
M	60	550	m, e	IDI	–	4...6	5,000	20
A	120	750	m	DI / IDI	–	2...12	2,800	27
MW	150	1,100	m	DI	–	4...8	2,600	36
P 3000	250	950	m, e	DI	–	4...12	2,600	45
P 7100	250	1,200	m, e	DI	–	4...12	2,500	55
P 8000	250	1,300	m, e	DI	–	6...12	2,500	55
P 8500	250	1,300	m, e	DI	–	4...12	2,500	55
H 1	240	1,300	e	DI	–	6...8	2,400	55
H 1000	250	1,350	e	DI	–	5...8	2,200	70
Axial-piston distributor injection pumps								
VE	120	1,200/350	m	DI / IDI	–	4...6	4,500	25
VE...EDC ¹⁾	70	1,200/350	e, em	DI / IDI	–	3...6	4,200	25
VE...MV	70	1,400/350	e, MV	DI / IDI	–	3...6	4,500	25
Radial-piston distributor injection pump								
VR...MV	135	1,700	e, MV	DI	–	4.6	4,500	50
Single-plunger injection pumps								
PF(R)...	150... 18,000	800... 1,500	m, em	DI / IDI	–	arbitrary	300... 2,000	75... 1,000
UIS 30 ²⁾	160	1,600	e, MV	DI	VE	8 ^{3a)}	3,000	45
UIS 31 ²⁾	300	1,600	e, MV	DI	VE	8 ^{3a)}	3,000	75
UIS 32 ²⁾	400	1,800	e, MV	DI	VE	8 ^{3a)}	3,000	80
UIS-P1 ³⁾	62	2,050	e, MV	DI	VE	6 ^{3a)}	5,000	25
UPS 12 ⁴⁾	150	1,600	e, MV	DI	VE	8 ^{3a)}	2,600	35
UPS 20 ⁴⁾	400	1,800	e, MV	DI	VE	8 ^{3a)}	2,600	80
UPS (PF[R])	3,000	1,400	e, MV	DI	–	6...20	1,500	500
Common Rail accumulator injection system								
CR ⁵⁾	100	1,350	e, MV	DI	VE ^{5a)} /NE	3...8	5,000 ^{5b)}	30
CR ⁶⁾	400	1,400	e, MV	DI	VE ^{6a)} /NE	6...16	2,800	200

¹⁾ EDC Electronic Diesel Control; ²⁾ UIS unit injector system for comm. vehs. ³⁾ UIS unit injector system for pass. cars; ^{3a)} With two ECU's large numbers of cylinders are possible; ⁴⁾ UPS unit pump system for comm. vehs. and buses; ⁵⁾ CR 1st generation for pass. cars and light comm. vehs.; ^{5a)} Up to 90° crankshaft BTDC, freely selectable; ^{5b)} Up to 5,500 min⁻¹ during overrun; ⁶⁾ CR for comm. vehs., buses, and diesel-powered locomotives; ^{6a)} Up to 30° crankshaft BTDC.

Injection-pump designs

In-line fuel-injection pumps

All in-line fuel-injection pumps have a plunger-and-barrel assembly for each cylinder. As the name implies, this comprises the pump barrel and the corresponding plunger. The pump camshaft integrated in the pump and driven by the engine, forces the pump plunger in the delivery direction. The plunger is returned by its spring.

The plunger-and-barrel assemblies are arranged in-line, and plunger lift cannot be varied. In order to permit changes in the delivery quantity, slots have been machined into the plunger, the diagonal edges of which are known as helixes. When the plunger is rotated by the movable control rack, the helixes permit the selection of the required effective stroke. Depending upon the fuel-injection conditions, delivery valves are installed between the pump's pressure chamber and the fuel-injection lines. These not only precisely terminate the injection process and prevent secondary injection (dribble) at the nozzle, but also ensure a family of uniform pump characteristic curves (pump map).

PE standard in-line fuel-injection pump

Start of fuel delivery is defined by an inlet port which is closed by the plunger's top edge. The delivery quantity is determined by the second inlet port being opened by the helix which is diagonally machined into the plunger.

The control rack's setting is determined by a mechanical (flyweight) governor or by an electric actuator (EDC).

Control-sleeve in-line fuel-injection pump

The control-sleeve in-line fuel-injection pump differs from a conventional in-line injection pump by having a "control sleeve" which slides up and down the pump plunger. By way of an actuator shaft, this can vary the plunger lift to port closing,

and with it the start of delivery and the start of injection. The control sleeve's position is varied as a function of a variety of different influencing variables. Compared to the standard PE in-line injection pump therefore, the control-sleeve version features an additional degree of freedom.

Distributor fuel-injection pumps

Distributor pumps have a mechanical (flyweight) governor, or an electronic control with integrated timing device. The distributor pump has only one plunger-and-barrel assembly for all the engine's cylinders.

Axial-piston distributor pump

In the case of the axial-piston distributor pump, fuel is supplied by a vane-type pump. Pressure generation, and distribution to the individual engine cylinders, is the job of a central piston which runs on a cam plate. For one revolution of the driveshaft, the piston performs as many strokes as there are engine cylinders. The rotating-reciprocating movement is imparted to the plunger by the cams on the underside of the cam plate which ride on the rollers of the roller ring.

On the conventional VE axial-piston distributor pump with mechanical (flyweight) governor, or electronically controlled actuator, a control collar defines the effective stroke and with it the injected fuel quantity. The pump's start of delivery can be adjusted by the roller ring (timing device). On the conventional solenoid-valve-controlled axial-piston distributor pump, instead of a control collar an electronically controlled high-pressure solenoid valve controls the injected fuel quantity. The open and closed-loop control signals are processed in two ECU's. Speed is controlled by appropriate triggering of the actuator.

Radial-piston distributor pump

In the case of the radial-piston distributor pump, fuel is supplied by a vane-type pump. A radial-piston pump with cam ring and two to four radial pistons is responsible

for generation of the high pressure and for fuel delivery. The injected fuel quantity is metered by a high-pressure solenoid valve. The timing device rotates the cam ring in order to adjust the start of delivery. As is the case with the solenoid-valve-controlled axial-piston pump, all open and closed-loop control signals are processed in two ECU's. Speed is controlled by appropriate triggering of the actuator.

Single-plunger fuel-injection pumps

PF single-plunger pumps

PF single-plunger injection pumps are used for small engines, diesel locomotives, marine engines, and construction machinery. They have no camshaft of their own, although they correspond to the PE in-line injection pumps regarding their method of operation. In the case of large engines, the mechanical-hydraulic governor or electronic controller is attached directly to the engine block. The fuel-quantity adjustment as defined by the governor (or controller) is transferred by a rack integrated in the engine.

The actuating cams for the individual PF single-plunger pumps are located on the engine camshaft. This means that injection timing cannot be implemented by rotating the camshaft. Here, by adjusting an intermediate element (for instance, a rocker between camshaft and roller tappet) an advance angle of several angular degrees can be obtained.

Single-plunger injection pumps are also suitable for operation with viscous heavy oils.

Unit-injector system (UIS)

With the unit-injector system, injection pump and injection nozzle form a unit. One of these units is installed in the engine's cylinder head for each engine cylinder, and driven directly by a tappet or indirectly from the engine's camshaft through a valve lifter.

Compared with in-line and distributor injection pumps, considerably higher injection pressures (up to 2050 bar) have be-

come possible due to the omission of the high-pressure lines. Such high injection pressures coupled with the electronic map-based control of duration of injection (or injected fuel quantity), mean that a considerable reduction of the diesel engine's toxic emissions has become possible together with good shaping of the rate-of-discharge curve.

Electronic control concepts permit a variety of additional functions.

Unit-pump system (UPS)

The principle of the UPS unit-pump system is the same as that of the UIS unit injector. It is a modular high-pressure injection system. Similar to the UIS, the UPS system features one UPS single-plunger injection pump for each engine cylinder. Each UP pump is driven by the engine's camshaft. Connection to the nozzle-and-holder assembly is through a short high-pressure delivery line precisely matched to the pump-system components.

Electronic map-based control of the start of injection and injection duration (in other words, of injected fuel quantity) leads to a pronounced reduction in the diesel engine's toxic emissions. The use of a high-speed electronically triggered solenoid valve enables the characteristic of the individual injection process, the so-called rate-of-discharge curve, to be precisely defined.

Accumulator injection system

Common-Rail system (CR)

Pressure generation and the actual injection process have been decoupled from each other in the Common Rail accumulator injection system. The injection pressure is generated independent of engine speed and injected fuel quantity, and is stored, ready for each injection process, in the rail (fuel accumulator). The start of injection and the injected fuel quantity are calculated in the ECU and, via the injection unit, implemented at each cylinder through a triggered solenoid valve.

Mechanically-controlled (governed) axial-piston distributor fuel-injection pumps VE

Fuel-injection systems

Assignments

The fuel-injection system is responsible for supplying the diesel engine with fuel. To do so, the injection pump generates the pressure required for fuel injection. The fuel under pressure is forced through the high-pressure fuel-injection tubing to the injection nozzle which then injects it into the combustion chamber.

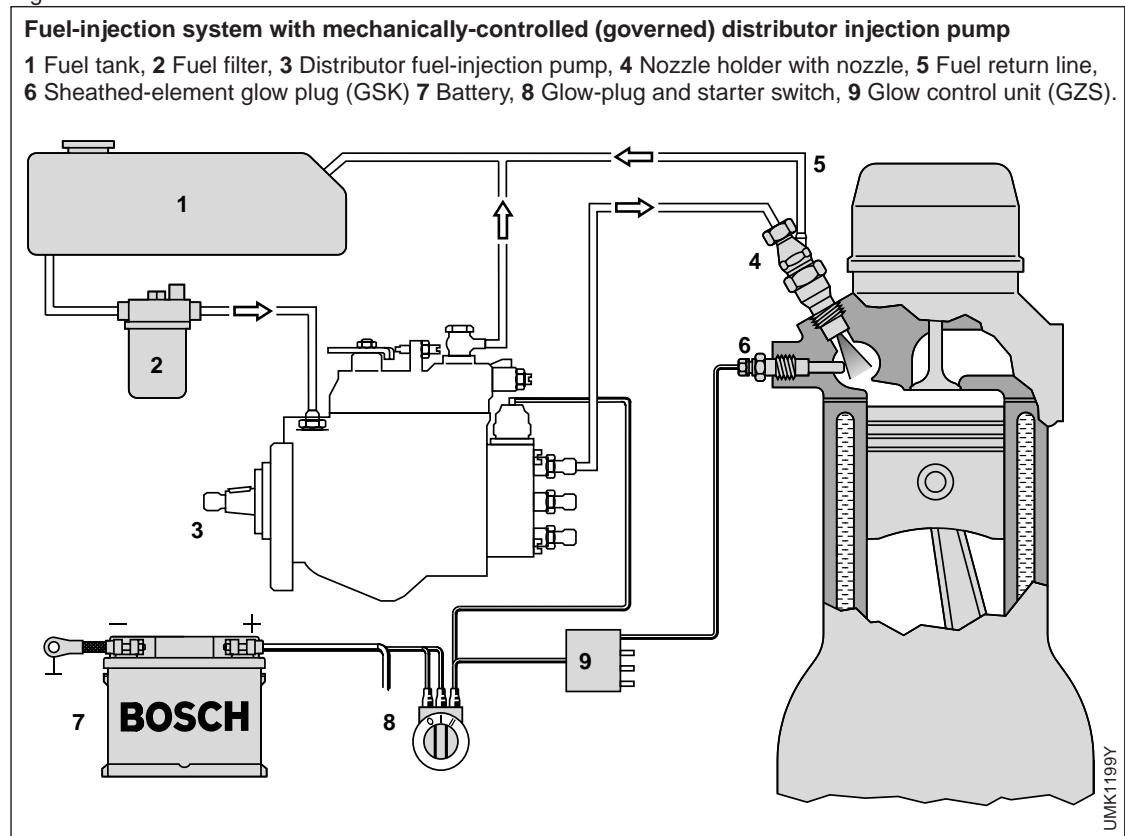
The fuel-injection system (Fig. 1) includes the following components and assemblies: The fuel tank, the fuel filter, the fuel-supply pump, the injection nozzles, the high-pressure injection

tubing, the governor, and the timing device (if required).

The combustion processes in the diesel engine depend to a large degree upon the quantity of fuel which is injected and upon the method of introducing this fuel to the combustion chamber.

The most important criteria in this respect are the fuel-injection timing and the duration of injection, the fuel's distribution in the combustion chamber, the moment in time when combustion starts, the amount of fuel metered to the engine per degree crankshaft, and the total injected fuel quantity in accordance with the engine loading. The optimum interplay of all these parameters is decisive for the faultless functioning of the diesel engine and of the fuel-injection system.

Fig. 1



Types

The increasing demands placed upon the diesel fuel-injection system made it necessary to continually develop and improve the fuel-injection pump.

Following systems comply with the present state-of-the-art:

- In-line fuel-injection pump (PE) with mechanical (flyweight) governor or Electronic Diesel Control (EDC) and, if required, attached timing device,
- Control-sleeve in-line fuel-injection pump (PE), with Electronic Diesel Control (EDC) and infinitely variable start of delivery (without attached timing device),
- Single-plunger fuel-injection pump (PF),
- Distributor fuel-injection pump (VE) with mechanical (flyweight) governor or Electronic Diesel Control (EDC). With integral timing device,
- Radial-piston distributor injection pump (VR),
- Common Rail accumulator injection system (CRS),
- Unit-injector system (UIS),
- Unit-pump system (UPS).

Fuel-injection techniques

Fields of application

Small high-speed diesel engines demand a lightweight and compact fuel-injection installation. The VE distributor fuel-injection pump (Fig. 2) fulfills these stipulations by combining

- Fuel-supply pump,
- High-pressure pump,
- Governor, and
- Timing device,

in a small, compact unit. The diesel engine's rated speed, its power output, and its configuration determine the parameters for the particular distributor pump.

Distributor pumps are used in passenger cars, commercial vehicles, agricultural tractors and stationary engines.

Fig. 2: VE distributor pump fitted to a 4-cylinder diesel engine



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Subassemblies

In contrast to the in-line injection pump, the VE distributor pump has only one pump cylinder and one plunger, even for multi-cylinder engines. The fuel delivered by the pump plunger is apportioned by a distributor groove to the outlet ports as determined by the engine's number of cylinders. The distributor pump's closed housing contains the following functional groups:

- High-pressure pump with distributor,
- Mechanical (flyweight) governor,
- Hydraulic timing device,
- Vane-type fuel-supply pump,
- Shutoff device, and
- Engine-specific add-on modules.

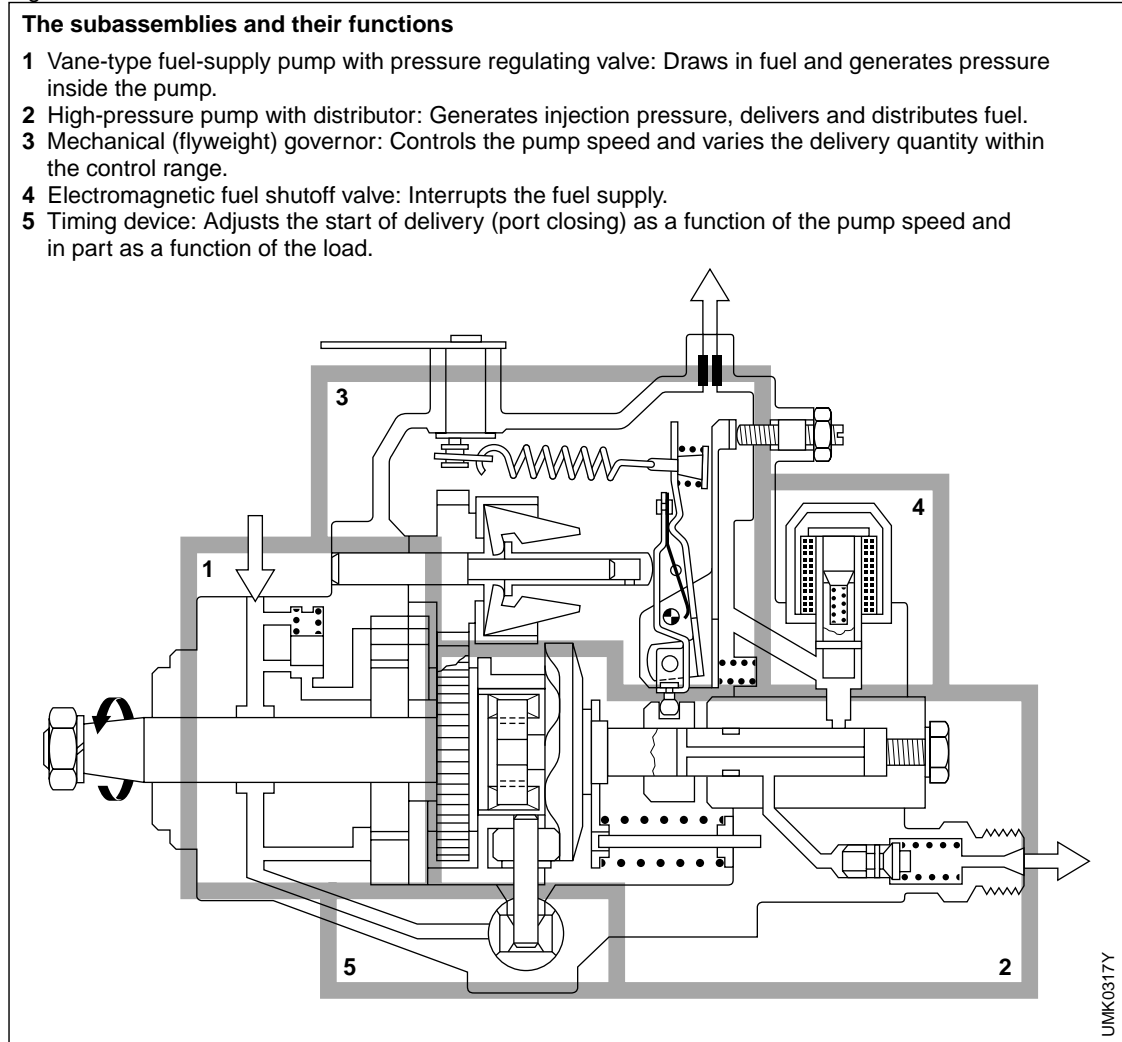
Fig. 3 shows the functional groups and their assignments. The add-on modules

facilitate adaptation to the specific requirements of the diesel engine in question.

Design and construction

The distributor pump's drive shaft runs in bearings in the pump housing and drives the vane-type fuel-supply pump. The roller ring is located inside the pump at the end of the drive shaft although it is not connected to it. A rotating-reciprocating movement is imparted to the distributor plunger by way of the cam plate which is driven by the input shaft and rides on the rollers of the roller ring. The plunger moves inside the distributor head which is bolted to the pump housing. Installed in the distributor head are the electrical fuel shutoff device, the screw plug with vent screw, and the delivery valves with their

Fig. 3



holders. If the distributor pump is also equipped with a mechanical fuel shutoff device this is mounted in the governor cover.

The governor assembly comprising the flyweights and the control sleeve is driven by the drive shaft (gear with rubber damper) via a gear pair. The governor linkage mechanism which consists of the control, starting, and tensioning levers, can pivot in the housing.

The governor shifts the position of the control collar on the pump plunger. On the governor mechanism's top side is the governor spring which engages with the external control lever through the control-lever shaft which is held in bearings in the governor cover.

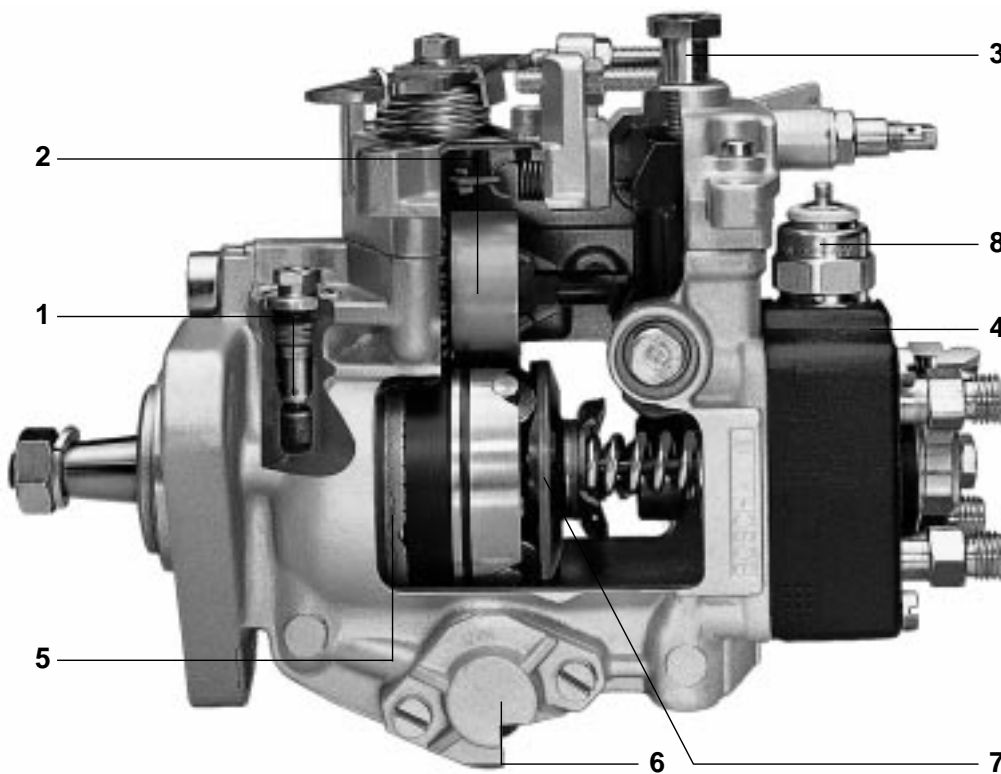
The control lever is used to control pump function. The governor cover forms the top of the distributor pump, and

also contains the full-load adjusting screw, the overflow restriction or the overflow valve, and the engine-speed adjusting screw. The hydraulic injection timing device is located at the bottom of the pump at right angles to the pump's longitudinal axis. Its operation is influenced by the pump's internal pressure which in turn is defined by the vane-type fuel-supply pump and by the pressure-regulating valve. The timing device is closed off by a cover on each side of the pump (Fig. 4).

Fig. 4

The subassemblies and their configuration

- 1 Pressure-control valve, 2 Governor assembly, 3 Overflow restriction,
- 4 Distributor head with high-pressure pump, 5 Vane-type fuel-supply pump, 6 Timing device,
- 7 Cam plate, 8 Electromagnetic shutoff valve.



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Pump drive

The distributor injection pump is driven by the diesel engine through a special drive unit. For 4-stroke engines, the pump is driven at exactly half the engine crankshaft speed, in other words at camshaft speed. The VE pump must be positively driven so that its drive shaft is synchronized to the engine's piston movement.

This positive drive is implemented by means of either toothed belts, pinion, gear wheel or chain. Distributor pumps are available for clockwise and for counter-clockwise rotation, whereby the injection sequence differs depending upon the direction of rotation.

The fuel outlets though are always supplied with fuel in their geometric sequence, and are identified with the letters A, B, C etc. to avoid confusion with the engine-cylinder numbering. Distributor pumps are suitable for engines with up to max. 6 cylinders.

Fuel supply and delivery

Considering an injection system with distributor injection pump, fuel supply and delivery is divided into low-pressure and high-pressure delivery (Fig. 1).

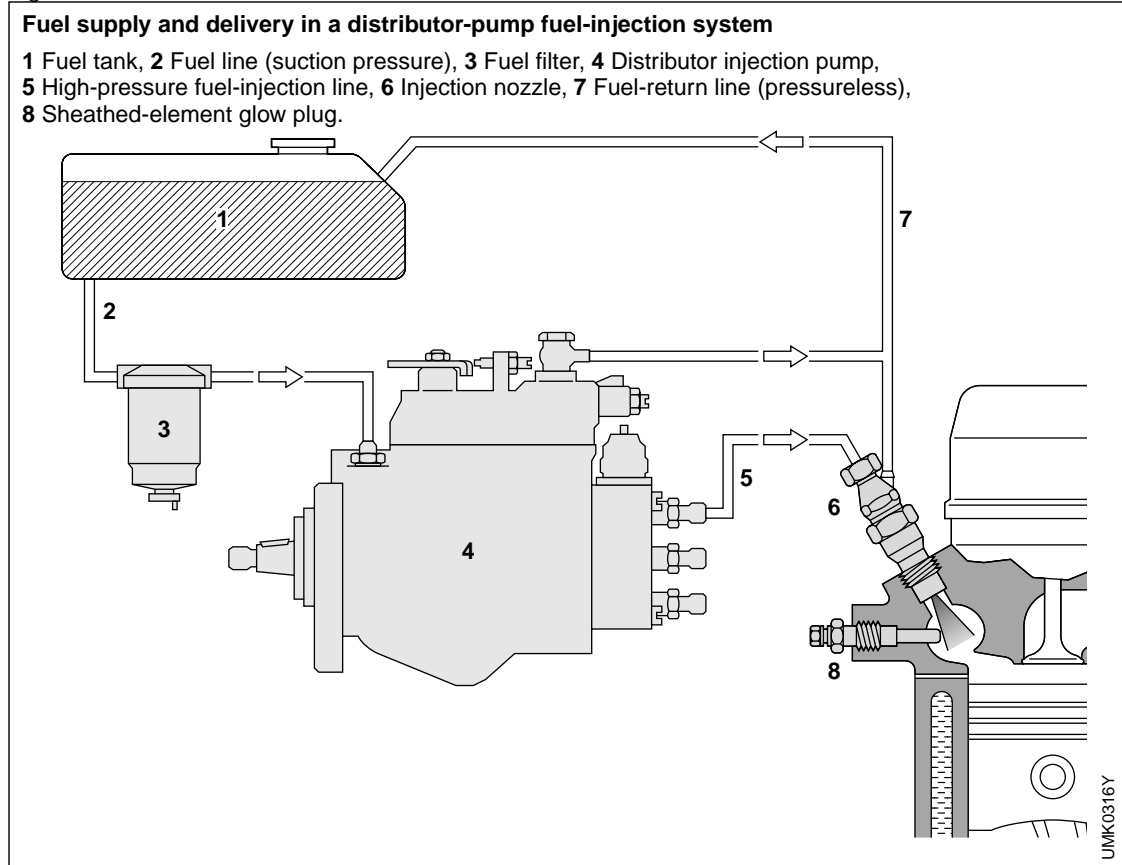
Low-pressure stage

Low-pressure delivery

The low-pressure stage of a distributor-pump fuel-injection installation comprises the fuel tank, fuel lines, fuel filter, vane-type fuel-supply pump, pressure-control valve, and overflow restriction.

The vane-type fuel-supply pump draws fuel from the fuel tank. It delivers a virtually constant flow of fuel per revolution to the interior of the injection pump. A pressure-control valve is fitted to ensure that a defined injection-pump interior pressure is maintained as a function of supply-pump speed. Using this valve, it is possible to set a defined pressure for a given speed. The pump's

Fig. 1



interior pressure then increases in proportion to the speed (in other words, the higher the pump speed the higher the pump interior pressure). Some of the fuel flows through the pressure-regulating valve and returns to the suction side. Some fuel also flows through the overflow restriction and back to the fuel tank in order to provide cooling and self-venting for the injection pump (Fig. 2). An overflow valve can be fitted instead of the overflow restriction.

Fuel-line configuration

For the injection pump to function efficiently it is necessary that its high-pressure stage is continually provided with pressurized fuel which is free of vapor bubbles. Normally, in the case of passenger cars and light commercial vehicles, the difference in height between the fuel tank and the fuel-injection equipment is negligible. Furthermore, the fuel lines are not too long and they have adequate internal diameters. As a result, the vane-type supply pump in the

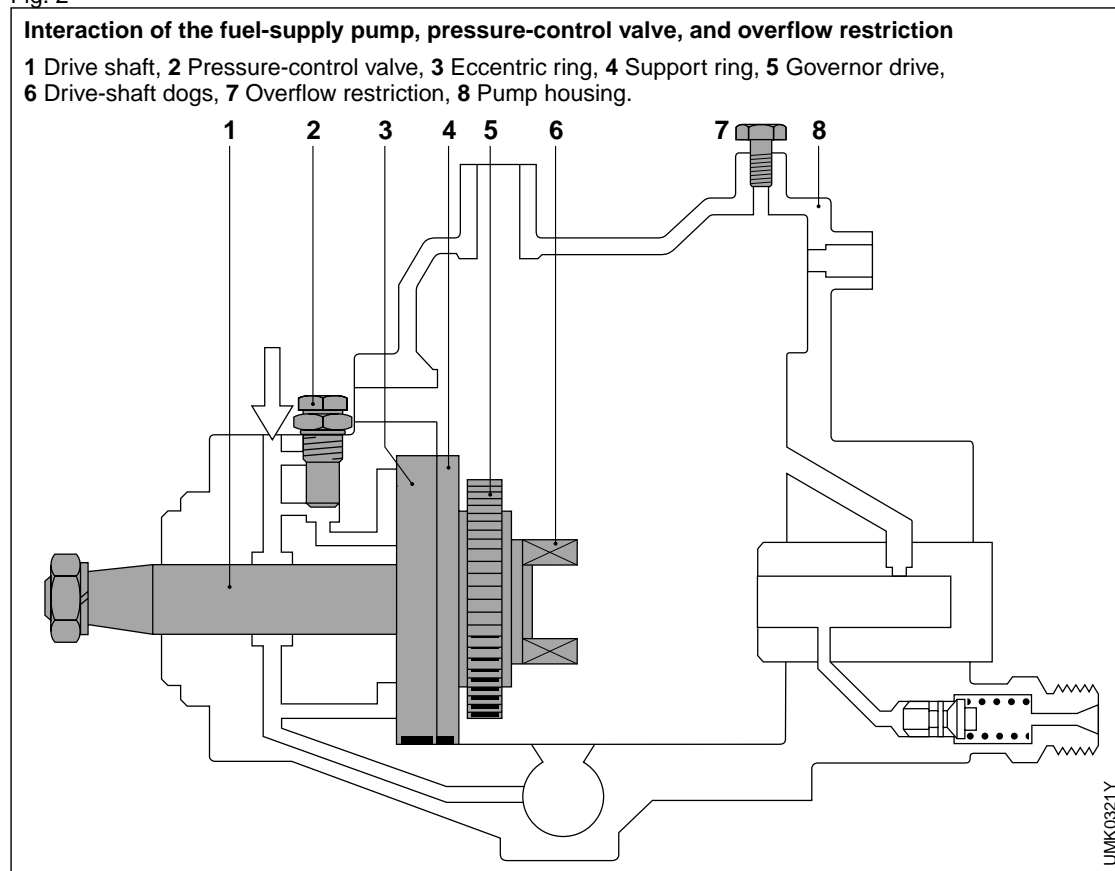
injection pump is powerful enough to draw the fuel out of the fuel tank and to build up sufficient pressure in the interior of the injection pump.

In those cases in which the difference in height between fuel tank and injection pump is excessive and (or) the fuel line between tank and pump is too long, a pre-supply pump must be installed. This overcomes the resistances in the fuel line and the fuel filter. Gravity-feed tanks are mainly used on stationary engines.

Fuel tank

The fuel tank must be of noncorroding material, and must remain free of leaks at double the operating pressure and in any case at 0.3 bar. Suitable openings or safety valves must be provided, or similar measures taken, in order to permit excess pressure to escape of its own accord. Fuel must not leak past the filler cap or through pressure-compensation devices. This applies when the vehicle is subjected to minor mechanical shocks, as well as when

Fig. 2



Axial-piston distributor pumps

cornering, and when standing or driving on an incline. The fuel tank and the engine must be so far apart from each other that in case of an accident there is no danger of fire. In addition, special regulations concerning the height of the fuel tank and its protective shielding apply to vehicles with open cabins, as well as to tractors and buses

Fuel lines

As an alternative to steel pipes, flame-inhibiting, steel-braid-armored flexible fuel lines can be used for the low-pressure stage. These must be routed to ensure that they cannot be damaged mechanically, and fuel which has dripped or evaporated must not be able to accumulate nor must it be able to ignite.

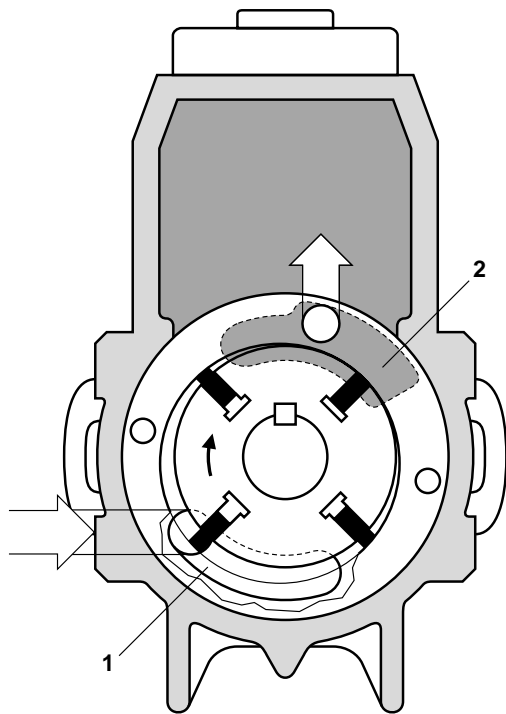
Fuel filter

The injection pump's high-pressure stage and the injection nozzle are manufactured with accuracies of several thousandths of a millimeter. As a result,

Fig. 3: Vane-type fuel-supply pump with impeller on the drive shaft

Vane-type fuel-supply pump for low-pressure delivery

1 Inlet, 2 Outlet.



UMK0324Y

Fig. 4



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contaminants in the fuel can lead to malfunctions, and inefficient filtering can cause damage to the pump components, delivery valves, and injector nozzles. This means that a fuel filter specifically aligned to the requirements of the fuel-injection system is absolutely imperative if trouble-free operation and a long service life are to be achieved. Fuel can contain water in bound form (emulsion) or unbound form (e.g., condensation due to temperature changes). If this water gets into the injection pump, corrosion damage can be the result. Distributor pumps must therefore be equipped with a fuel filter incorporating a water accumulator from which the water must be drained off at regular intervals. The increasing popularity of the diesel engine in the passenger car has led to the development of an automatic water-warning device which indicates by means of a warning lamp when water must be drained.

Vane-type fuel supply pump

The vane-type pump (Figs. 3 and 4) is located around the injection pump's drive shaft. Its impeller is concentric with the shaft and connected to it with a Woodruff key and runs inside an eccentric ring mounted in the pump housing. When the drive shaft rotates, centrifugal

force pushes the impeller's four vanes outward against the inside of the eccentric ring. The fuel between the vanes' undersides and the impeller serves to support the outward movement of the vanes. The fuel enters through the inlet passage and a kidney-shaped recess in the pump's housing, and fills the space formed by the impeller, the vane, and the inside of the eccentric ring. The rotary motion causes the fuel between adjacent vanes to be forced into the upper (outlet) kidney-shaped recess and through a passage into the interior of the pump. At the same time, some of the fuel flows through a second passage to the pressure-control valve.

Pressure-control valve

The pressure-control valve (Fig. 5) is connected through a passage to the upper (outlet) kidney-shaped recess, and is mounted in the immediate vicinity of the fuel-supply pump. It is a spring-loaded spool-type valve with which the pump's internal pressure can be varied as a function of the quantity of fuel being delivered. If fuel pressure increases beyond a given value, the valve spool opens the return passage so that the fuel can flow back to the supply pump's suction side. If the fuel pressure is too low, the return passage is closed by the spring.

Fig. 5

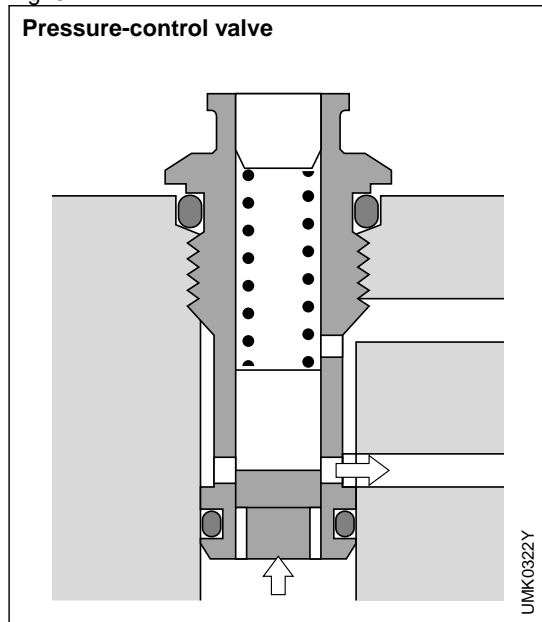
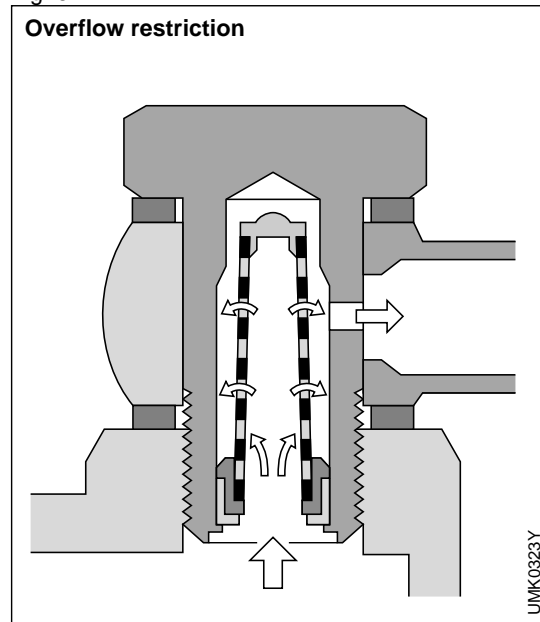


Fig. 6



The spring's initial tension can be adjusted to set the valve opening pressure.

Overflow restriction

The overflow restriction (Figure 6) is screwed into the injection pump's governor cover and connected to the pump's interior. It permits a variable amount of fuel to return to the fuel tank through a narrow passage. For this fuel, the restriction represents a flow resistance that assists in maintaining the pressure inside the injection pump. Being as inside the pump a precisely defined pressure is required as a function of pump speed, the overflow restriction and the flow-control valve are precisely matched to each other.

High-pressure stage

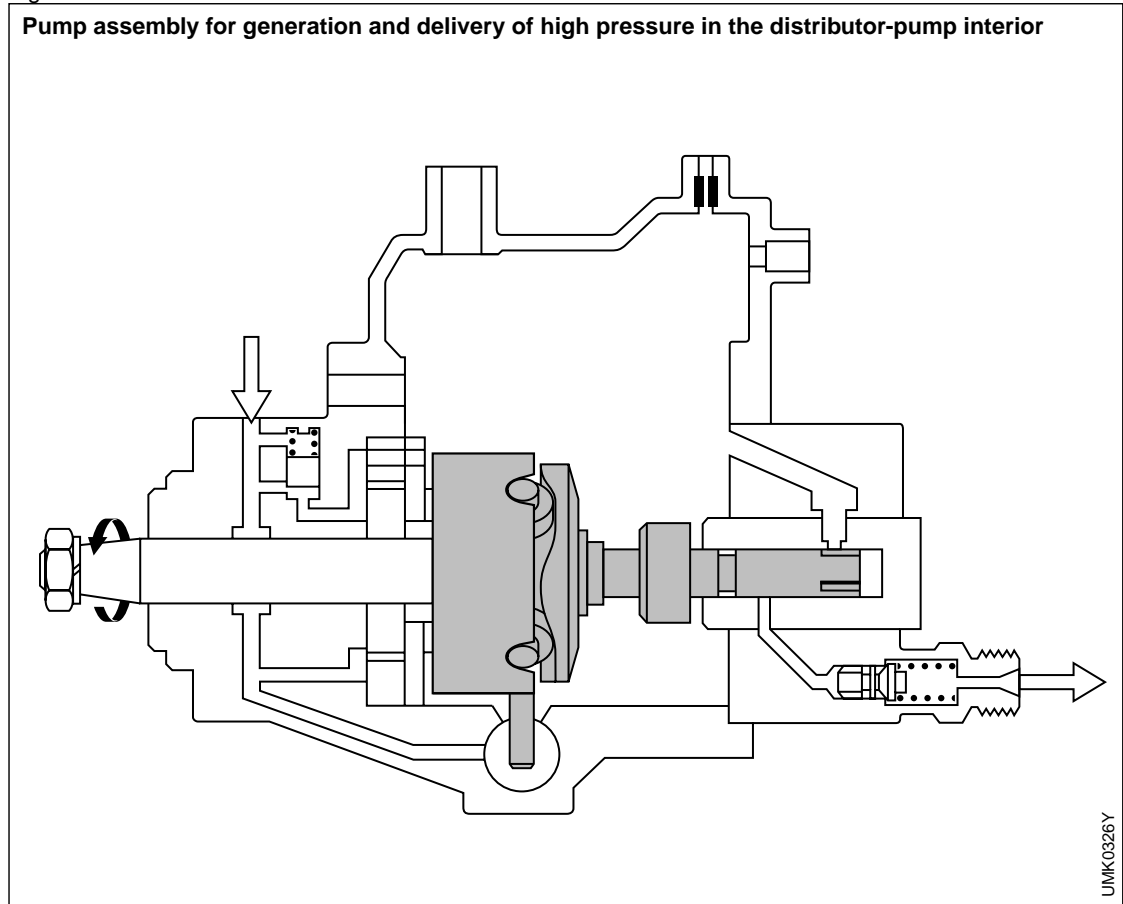
The fuel pressure needed for fuel injection is generated in the injection pump's high-pressure stage. The pressurized fuel then travels to the injection nozzles through the delivery valves and the fuel-injection tubing.

Distributor-plunger drive

The rotary movement of the drive shaft is transferred to the distributor plunger via a coupling unit (Fig. 7), whereby the dogs on cam plate and drive shaft engage with the recesses in the yoke, which is located between the end of the drive shaft and the cam plate. The cam plate is forced against the roller ring by a spring, and when it rotates the cam lobes riding on the ring's rollers convert the purely rotational movement of the drive shaft into a rotating-reciprocating movement of the cam plate.

The distributor plunger is held in the cam plate by its cylindrical fitting piece and is locked into position relative to the cam

Fig. 7



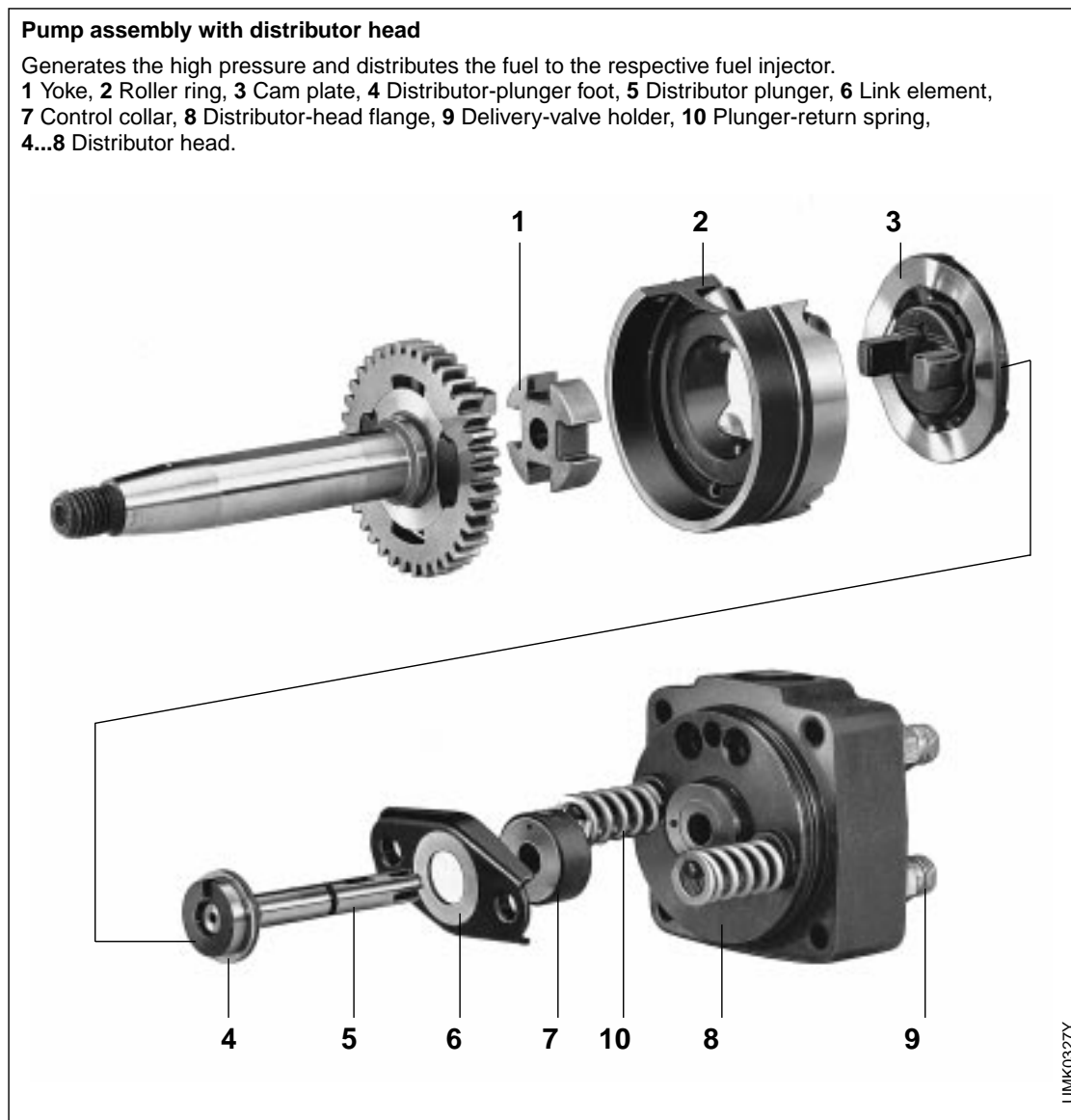


Fig. 8

plate by a pin. The distributor plunger is forced upwards to its TDC position by the cams on the cam plate, and the two symmetrically arranged plunger-return springs force it back down again to its BDC position.

The plunger-return springs abut at one end against the distributor head and at the other their force is directed to the plunger through a link element. These springs also prevent the cam plate jumping off the rollers during harsh acceleration. The lengths of the return springs are carefully matched to each other so that the plunger is not displaced from its centered position (Fig. 8).

Cam plates and cam contours

The cam plate and its cam contour influence the fuel-injection pressure and the injection duration, whereby cam stroke and plunger-lift velocity are the decisive criteria. Considering the different combustion-chamber configurations and combustion systems used in the various engine types, it becomes imperative that the fuel-injection factors are individually tailored to each other. For this reason, a special cam-plate surface is generated for each engine type and machined into the cam-plate face. This defined cam plate is then assembled in the corresponding distributor pump. Since the cam-plate surface is specific to a given engine type, the cam plates are not interchangeable between the different VE-pump variants.

Distributor head

The distributor plunger, the distributor-head bushing and the control collar are so precisely fitted (lapped) into the distributor head (Fig. 8), that they seal even at very high pressures. Small leakage losses are nevertheless unavoidable, as well as being desirable for plunger lubrication. For this reason, the distributor head is only to be replaced as a complete assembly, and never the plunger, control collar, or distributor flange alone.

Fuel metering

The fuel delivery from a fuel-injection pump is a dynamic process comprising several stroke phases (Fig. 9). The pressure required for the actual fuel injection is generated by the high-pressure pump. The distributor plunger's stroke and delivery phases (Fig. 10) show the metering of fuel to an engine cylinder. For a 4-cylinder engine the distributor plunger rotates through 90° for a stroke from BDC to TDC and back again. In the case of a 6-cylinder engine, the plunger must have completed

these movements within 60° of plunger rotation.

As the distributor plunger moves from TDC to BDC, fuel flows through the open inlet passage and into the high-pressure chamber above the plunger. At BDC, the plunger's rotating movement then closes the inlet passage and opens the distributor slot for a given outlet port (Fig. 10a). The plunger now reverses its direction of movement and moves upwards, the working stroke begins. The pressure that builds up in the high-pressure chamber above the plunger and in the outlet-port passage suffices to open the delivery valve in question and the fuel is forced through the high-pressure line to the injector nozzle (Fig. 10b). The working stroke is completed as soon as the plunger's transverse cutoff bore reaches the control edge of the control collar and pressure collapses. From this point on, no more fuel is delivered to the injector and the delivery valve closes the high-pressure line.

Fig. 9: The cam plate rotates against the roller ring, whereby its cam track follows the rollers causing it to lift (for TDC) and drop back again (for BDC)

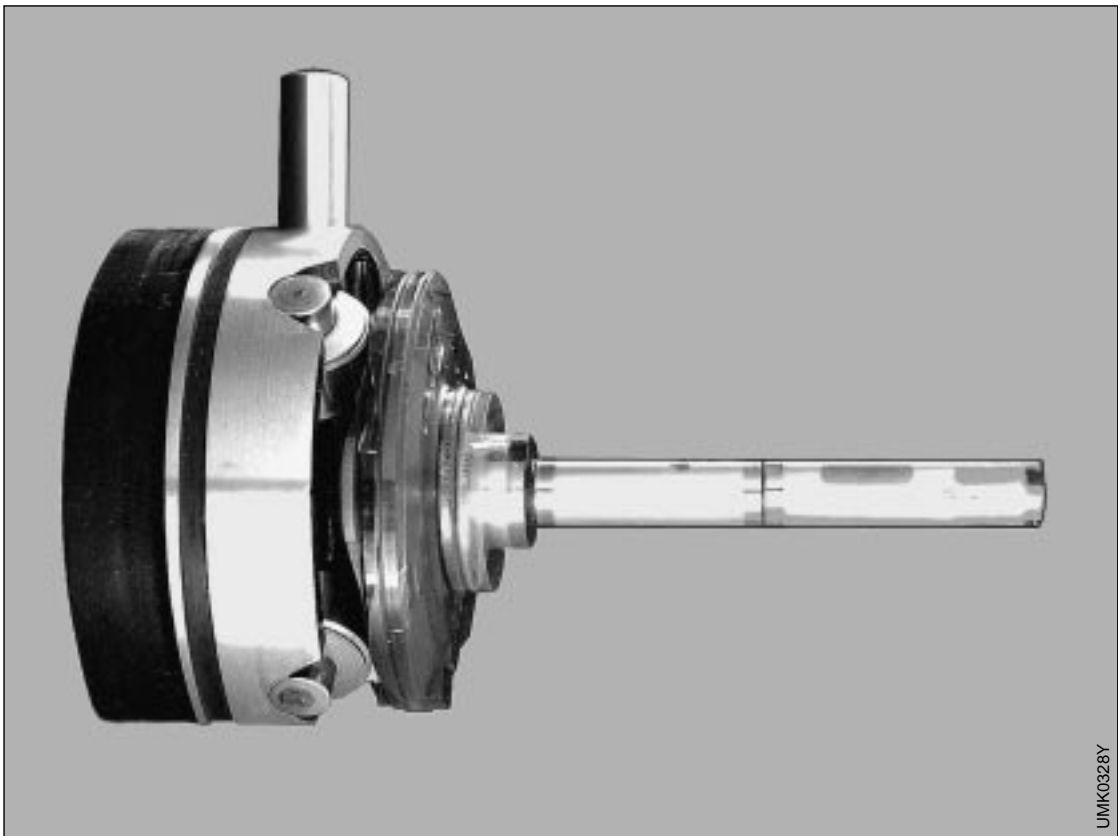
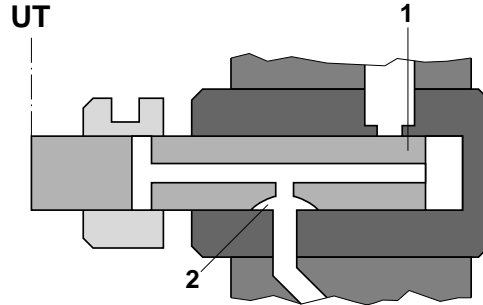
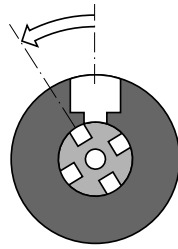


Fig. 10

Distributor plunger with stroke and delivery phases

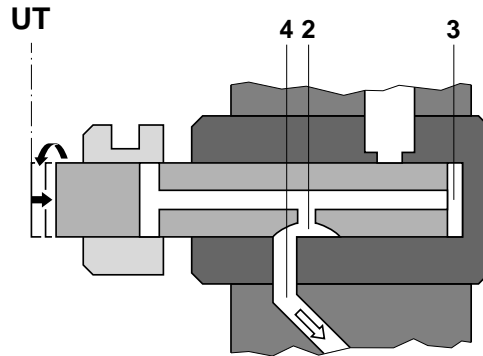
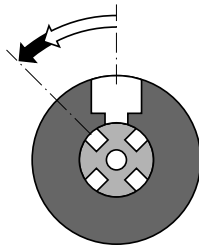
a Inlet passage closes.

At BDC, the metering slot (1) closes the inlet passage, and the distributor slot (2) opens the outlet port.



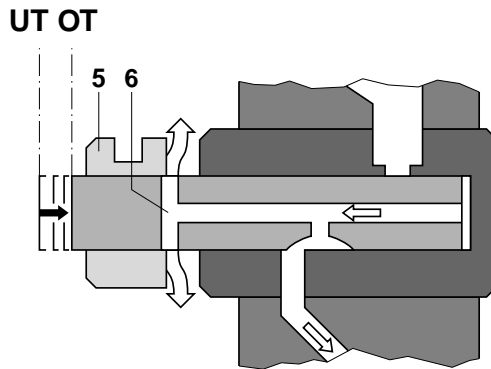
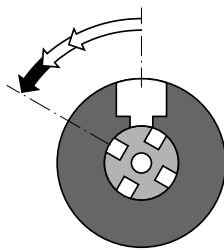
b Fuel delivery.

During the plunger stroke towards TDC (working stroke), the plunger pressurizes the fuel in the high-pressure chamber (3). The fuel travels through the outlet-port passage (4) to the injection nozzle.



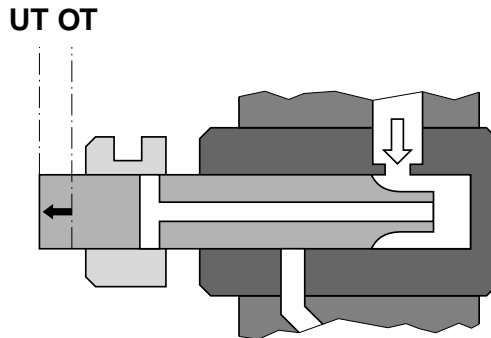
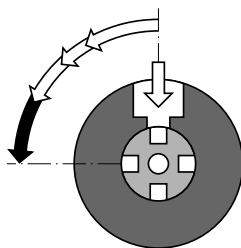
c End of delivery.

Fuel delivery ceases as soon as the control collar (5) opens the transverse cutoff bore (6).



d Entry of fuel.

Shortly before TDC, the inlet passage is opened. During the plunger's return stroke to BDC, the high-pressure chamber is filled with fuel and the transverse cutoff bore is closed again. The outlet-port passage is also closed at this point.



OT = TDC
UT = BDC

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During the plunger's continued movement to TDC, fuel returns through the cutoff bore to the pump interior. During this phase, the inlet passage is opened again for the plunger's next working cycle (Fig. 10c).

During the plunger's return stroke, its transverse cutoff bore is closed by the plunger's rotating stroke movement, and the high-pressure chamber above the plunger is again filled with fuel through the open inlet passage (Fig. 10d).

Delivery valve

The delivery valve closes off the high-pressure line from the pump. It has the job of relieving the pressure in the line by removing a defined volume of fuel upon completion of the delivery phase. This ensures precise closing of the injection nozzle at the end of the injection process. At the same time, stable pressure conditions between injection pulses are created in the high-pressure lines, regardless of the quantity of fuel being injected at a particular time.

The delivery valve is a plunger-type valve. It is opened by the injection pressure and closed by its return spring.

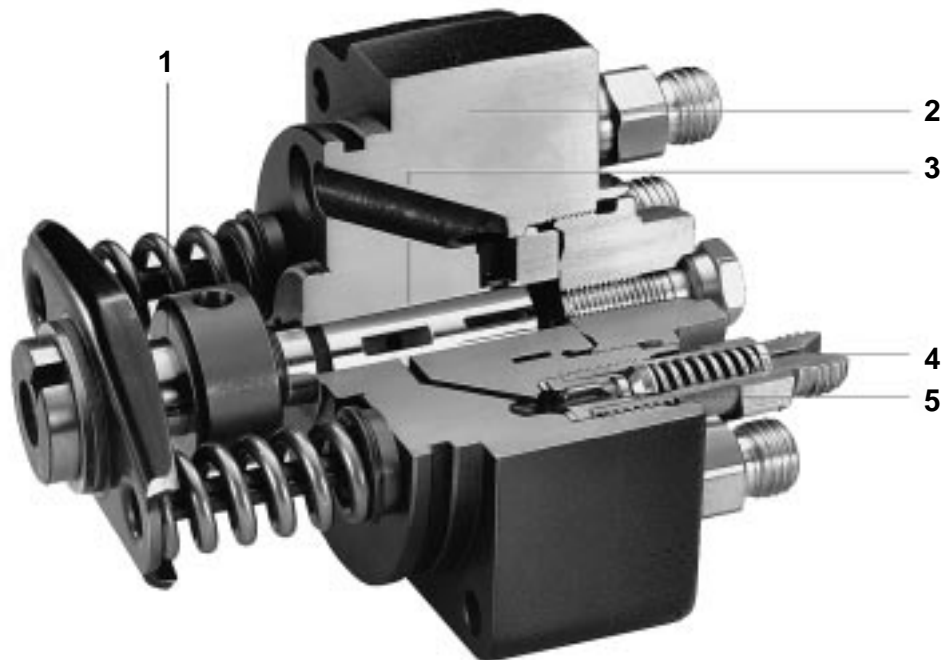
Between the plunger's individual delivery strokes for a given cylinder, the delivery valve in question remains closed. This separates the high-pressure line and the distributor head's outlet-port passage. During delivery, the pressure generated in the high-pressure chamber above the plunger causes the delivery valve to open. Fuel then flows via longitudinal slots, into a ring-shaped groove and through the delivery-valve holder, the high-pressure line and the nozzle holder to the injection nozzle.

As soon as delivery ceases (transverse cutoff bore opened), the pressure in the high-pressure chamber above the plunger and in the high-pressure lines drops to that of the pump interior, and the delivery-valve spring together with the static pressure in the line force the delivery-valve plunger back onto its seat again (Fig. 11).

Fig. 11

Distributor head with high-pressure chamber

1 Control collar, 2 Distributor head, 3 Distributor plunger, 4 Delivery-valve holder, 5 Delivery-valve.



Delivery valve with return-flow restriction

Precise pressure relief in the lines is necessary at the end of injection. This though generates pressure waves which are reflected at the delivery valve. These cause the delivery valve to open again, or cause vacuum phases in the high-pressure line. These processes result in post-injection of fuel with attendant increases in exhaust emissions or cavitation and wear in the injection line or at the nozzle. To prevent such harmful reflections, the delivery valve is provided with a restriction bore which is only effective in the direction of return flow. This return-flow restriction comprises a valve plate and a pressure spring so arranged that the restriction is ineffective in the delivery direction, whereas in the return direction damping comes into effect (Fig. 12).

Constant-pressure valve

With high-speed direct-injection (DI) engines, it is often the case that the

“retraction volume” resulting from the retraction piston on the delivery-valve plunger does not suffice to reliably prevent cavitation, secondary injection, and combustion-gas blowback into the nozzle-and-holder assembly. Here, constant-pressure valves are fitted which relieve the high-pressure system (injection line and nozzle-and-holder assembly) by means of a single-acting non-return valve which can be set to a given pressure, e.g., 60 bar (Fig. 13).

High-pressure lines

The pressure lines installed in the fuel-injection system have been matched precisely to the rate-of-discharge curve and must not be tampered with during service and repair work. The high-pressure lines connect the injection pump to the injection nozzles and are routed so that they have no sharp bends. In automotive applications, the high-pressure lines are normally secured with special clamps at specific intervals, and are made of seamless steel tubing.

Fig. 12

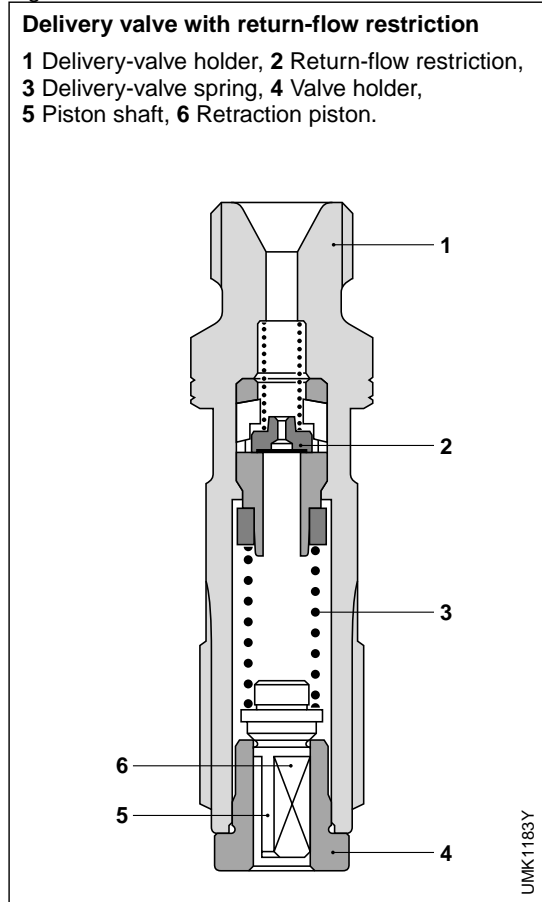
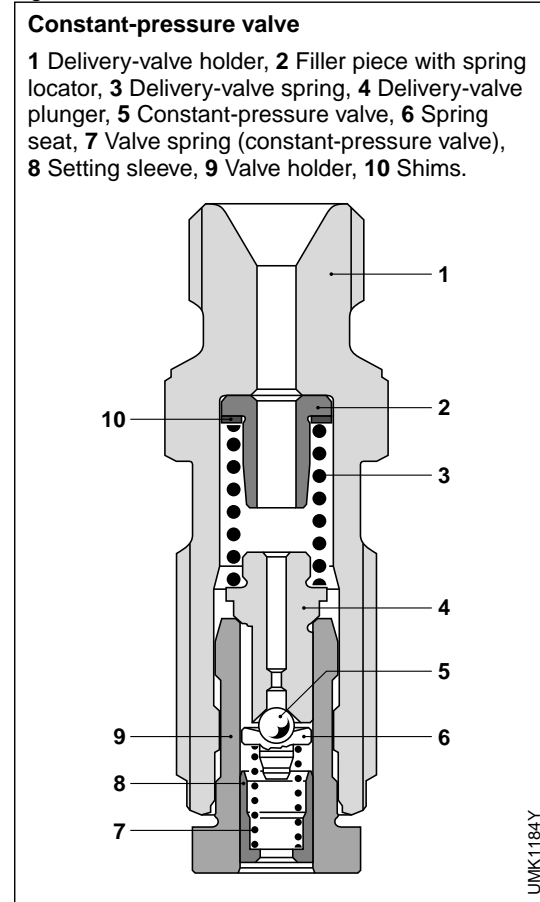


Fig. 13



Mechanical engine- speed control (governing)

Application

The driveability of a diesel-powered vehicle can be said to be satisfactory when its engine immediately responds to driver inputs from the accelerator pedal. Apart from this, upon driving off the engine must not tend to stall. The engine must respond to accelerator-pedal changes by accelerating or decelerating smoothly and without hesitation. On the flat, or on a constant gradient, with the accelerator pedal held in a given position, the vehicle speed should also remain constant. When the pedal is released the engine must brake the vehicle. On the diesel engine, it is the injection pump's governor that ensures that these stipulations are complied with. The governor assembly comprises the

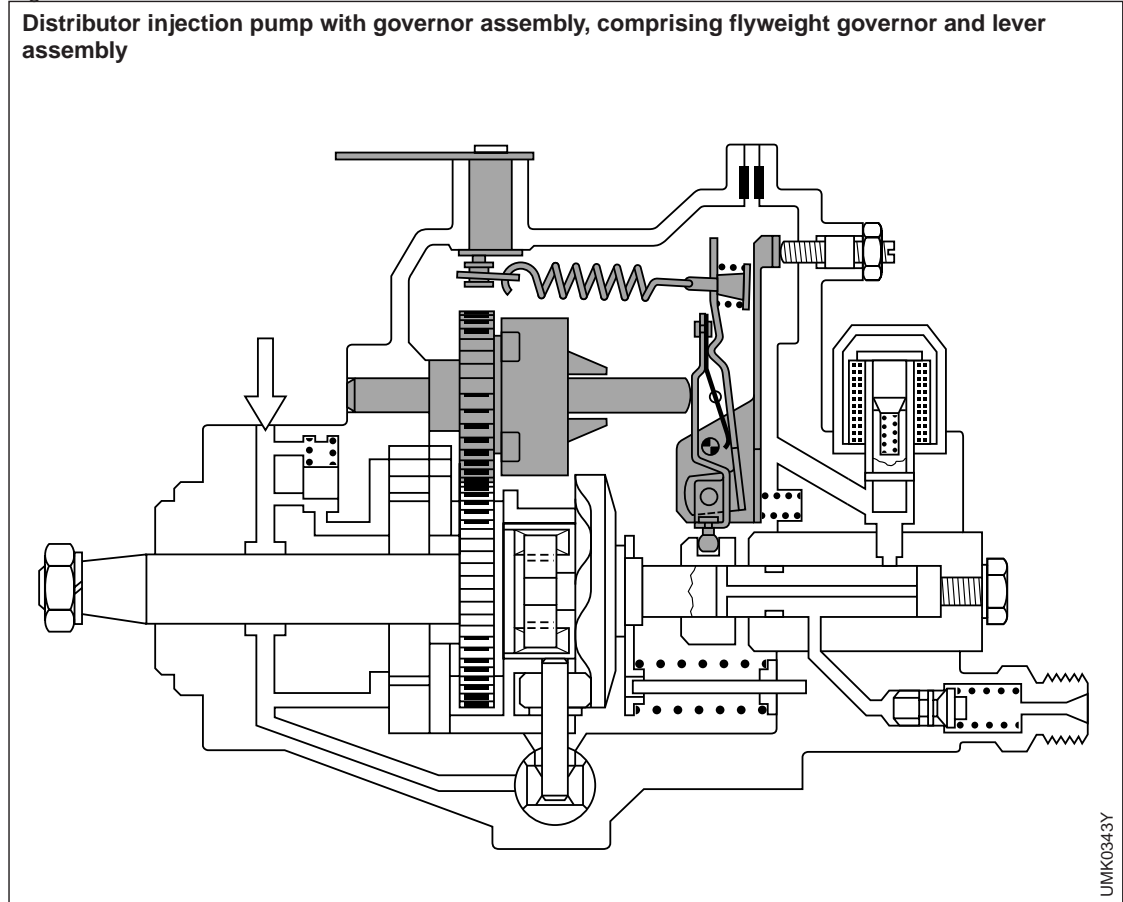
mechanical (flyweight) governor and the lever assembly. It is a sensitive control device which determines the position of the control collar, thereby defining the delivery stroke and with it the injected fuel quantity. It is possible to adapt the governor's response to setpoint changes by varying the design of the lever assembly (Fig. 1).

Governor functions

The basic function of all governors is the limitation of the engine's maximum speed. Depending upon type, the governor is also responsible for keeping certain engine speeds constant, such as idle speed, or the minimum and maximum engine speeds of a stipulated engine-speed range, or of the complete speed range, between idle and maximum speed. The different governor types are a direct result of the variety of governor assignments (Fig. 2):

- Low-idle-speed governing: The diesel engine's low-idle speed is controlled by the injection-pump governor.

Fig. 1



- Maximum-speed governing: With the accelerator pedal fully depressed, the maximum full-load speed must not increase to more than high idle speed (maximum speed) when the load is removed. Here, the governor responds by shifting the control collar back towards the “Stop” position, and the supply of fuel to the engine is reduced.
- Intermediate-speed governing: Variable-speed governors incorporate intermediate-speed governing. Within certain limits, these governors can also maintain the engine speeds between idle and maximum constant. This means that depending upon load, the engine speed n varies inside the engine’s power range only between n_{VT} (a given speed on the full-load curve) and n_{LT} (with no load on the engine). Other control functions are performed by the governor in addition to its governing responsibilities:
 - Releasing or blocking of the extra fuel required for starting,
 - Changing the full-load delivery as a

function of engine speed (torque control). In some cases, add-on modules are necessary for these extra assignments.

Speed-control (governing) accuracy

The parameter used as the measure for the governor’s accuracy in controlling engine speed when load is removed is the so-called speed droop (P-degree). This is the engine-speed increase, expressed as a percentage, that occurs when the diesel engine’s load is removed with the control-lever (accelerator) position unchanged. Within the speed-control range, the increase in engine speed is not to exceed a given figure. This is stipulated as the high idle speed. This is the engine speed which results when the diesel engine, starting at its maximum speed under full load, is relieved of all load. The speed increase is proportional to the change in load, and increases along with it.

$$\delta = \frac{n_{lo} - n_{vo}}{n_{vo}}$$

or expressed in %:

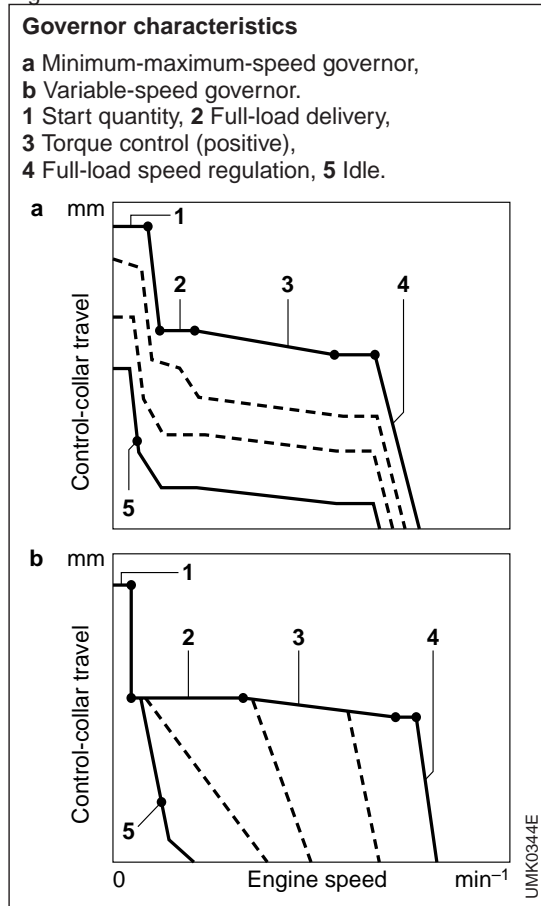
$$\delta = \frac{n_{lo} - n_{vo}}{n_{vo}} \cdot 100\%$$

where

- δ = Speed droop
- n_{lo} = High idle (maximum) speed
- n_{vo} = Maximum full-load speed

The required speed droop depends on engine application. For instance, on an engine used to power an electrical generator set, a small speed droop is required so that load changes result in only minor speed changes and therefore minimal frequency changes. On the other hand, for automotive applications large speed droops are preferable because these result in more stable control in case of only slight load changes (acceleration or deceleration) and lead to better driveability. A low-value speed droop would lead to rough, jerking operation when the load changes.

Fig. 2



Variable-speed governor

The variable-speed governor controls all engine speeds between start and high idle (maximum). The variable-speed governor also controls the idle speed and the maximum full-load speed, as well as the engine-speed range in between. Here, any engine speed can be selected by the accelerator pedal and, depending upon the speed droop, maintained practically constant (Fig. 4).

This is necessary for instance when ancillary units (winches, fire-fighting pumps, cranes etc.) are mounted on the vehicle. The variable-speed governor is also often fitted in commercial and agricultural vehicles (tractors and combine harvesters).

Design and construction

The governor assembly is driven by the drive shaft and comprises the flyweight housing complete with flyweights.

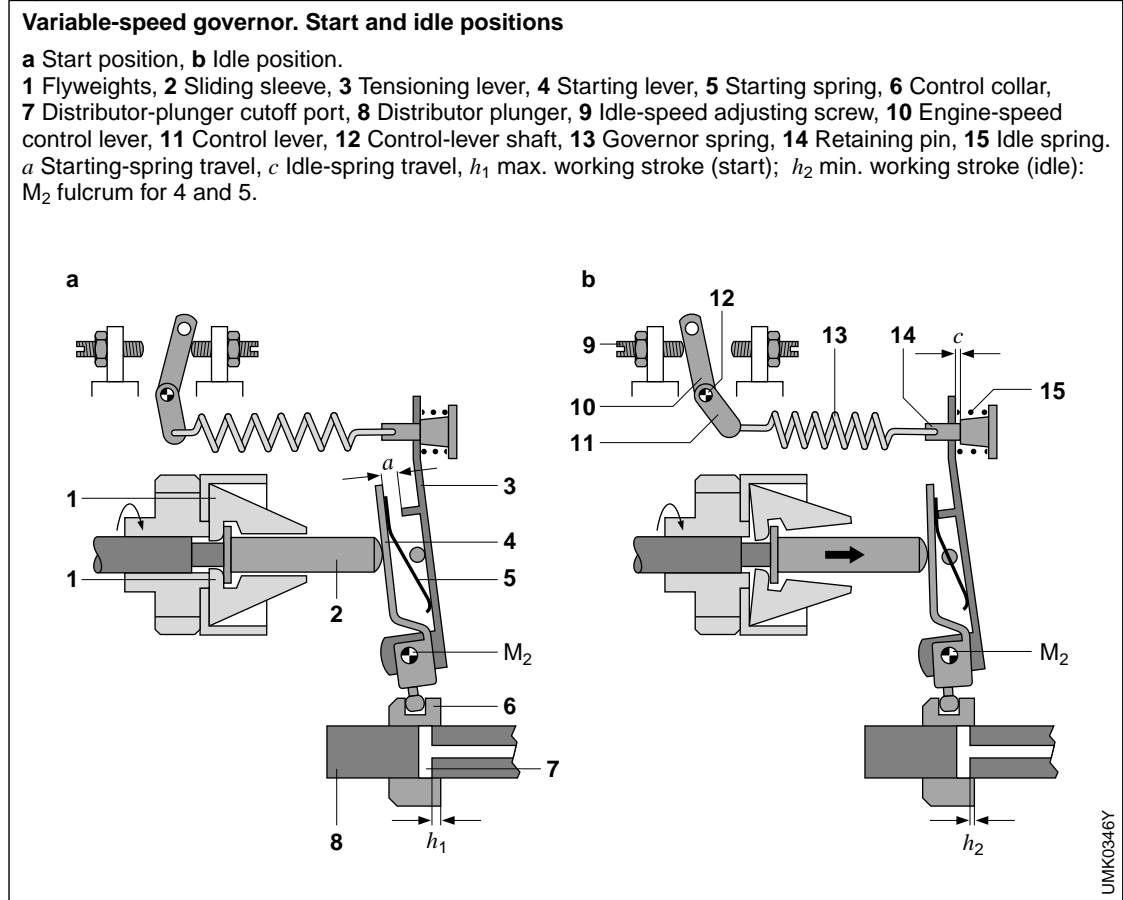
The governor assembly is attached to the governor shaft which is fixed in the

governor housing, and is free to rotate around it. When the flyweights rotate they pivot outwards due to centrifugal force and their radial movement is converted to an axial movement of the sliding sleeve. The sliding-sleeve travel and the force developed by the sleeve influence the governor lever assembly. This comprises the starting lever, tensioning lever, and adjusting lever (not shown). The interaction of spring forces and sliding-sleeve force defines the setting of the governor lever assembly, variations of which are transferred to the control collar and result in adjustments to the injected fuel quantity.

Starting

With the engine at standstill, the flyweights and the sliding sleeve are in their initial position (Fig. 3a). The starting lever has been pushed to the start position by the starting spring and has pivoted around its fulcrum M_2 . At the same time the control collar on the distributor plunger has been shifted to its

Fig. 3



start-quantity position by the ball pin on the starting lever. This means that when the engine is cranked the distributor plunger must travel through a complete working stroke (= maximum delivery quantity) before the cutoff bore is opened and delivery ceases. Thus the start quantity (= maximum delivery quantity) is automatically made available when the engine is cranked.

The adjusting lever is held in the pump housing so that it can rotate. It can be shifted by the fuel-delivery adjusting screw (not shown in Figure 3). Similarly, the start lever and tensioning lever are also able to rotate in the adjusting lever. A ball pin which engages in the control collar is attached to the underside of the start lever, and the start spring to its upper section. The idle spring is attached to a retaining pin at the top end of the tensioning lever. Also attached to this pin is the governor spring. The connection to the engine-speed control lever is through a lever and the control-lever shaft.

It only needs a very low speed for the sliding sleeve to shift against the soft start spring by the amount a . In the process, the start lever pivots around fulcrum M_2 and the start quantity is automatically reduced to the idle quantity.

Low-idle-speed control

With the engine running, and the accelerator pedal released, the engine-speed control lever shifts to the idle position (Figure 3b) up against the idle-speed adjusting screw. The idle speed is selected so that the engine still runs reliably and smoothly when unloaded or only slightly loaded. The actual control is by means of the idle spring on the retaining pin which counteracts the force generated by the flyweights.

This balance of forces determines the sliding-sleeve's position relative to the distributor plunger's cutoff bore, and with it the working stroke. At speeds above idle, the spring has been compressed by the amount c and is no longer effective. Using the special idle spring attached to the governor housing,

Characteristic curves of the variable-speed governor

- A: Start position of the control collar,
- S: Engine starts with start quantity,
- S-L: Start quantity reduces to idle quantity,
- L: Idle speed n_{LN} following engine start-up (no-load),
- L-B: Engine acceleration phase after shifting the engine-speed control lever from idle to a given required speed n_C ,
- B-B': The control collar remains briefly in the full-load position and causes a rapid increase in engine speed,
- B'-C: Control collar moves back (less injected fuel quantity, higher engine speed). In accordance with the speed droop, the vehicle maintains the required speed or speed n_C in the part-load range,
- E: Engine speed n_{LT} , after removal of load from the engine with the position of the engine-speed control-lever remaining unchanged.

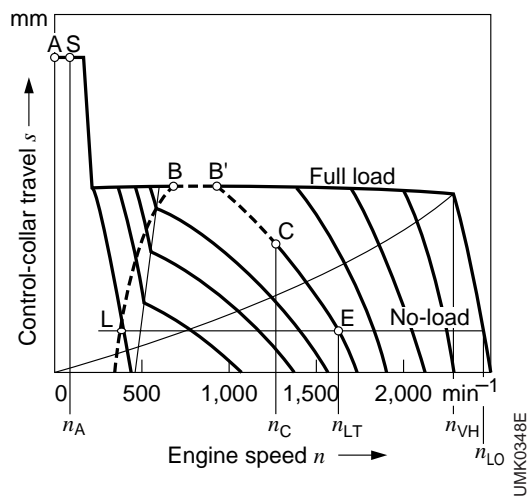


Fig. 4

this means that idle speed can be adjusted independent of the accelerator-pedal setting, and can be increased or decreased as a function of temperature or load.

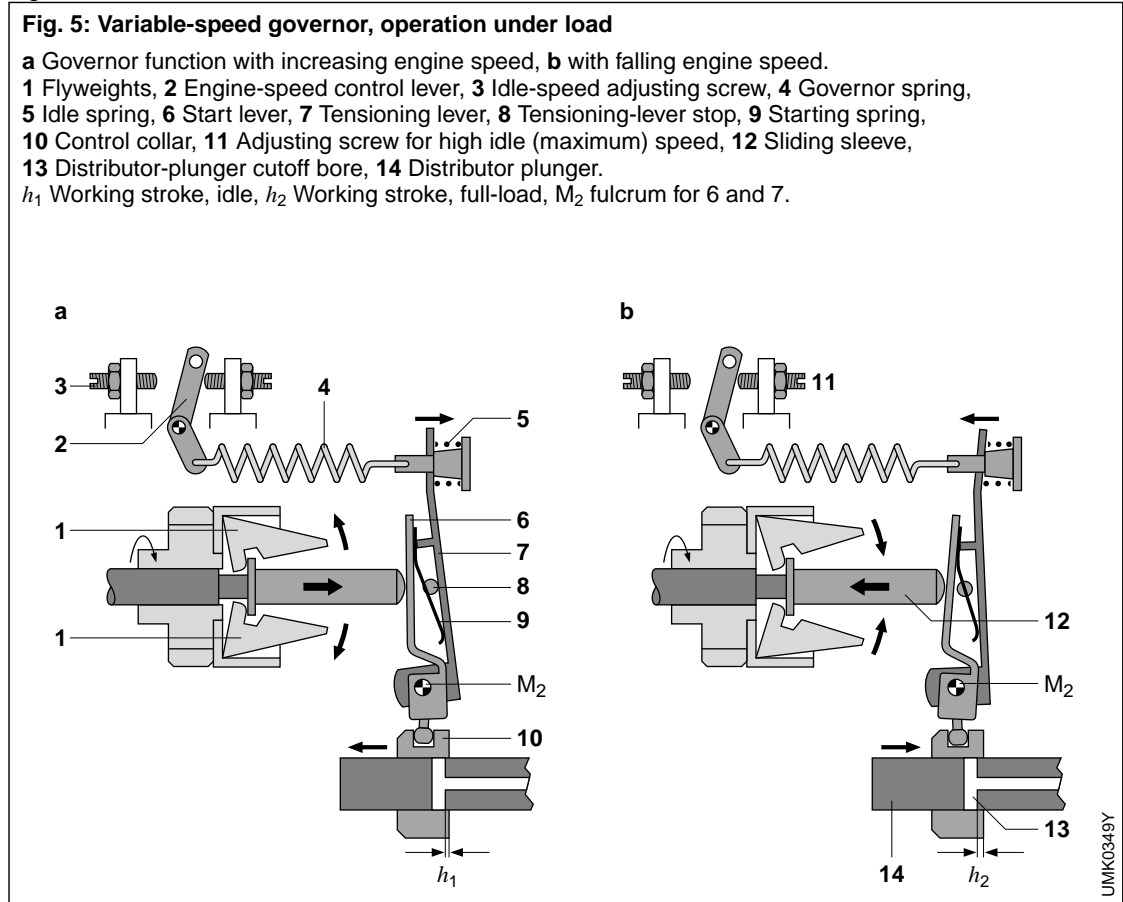
Operation under load

During actual operation, depending upon the required engine speed or vehicle speed, the engine-speed control lever is in a given position within its pivot range. This is stipulated by the driver through a given setting of the accelerator pedal. At engine speeds above idle, start spring and idle spring have been compressed completely and have no further effect on governor action. This is taken over by the governor spring.

Example (Fig. 5):
Using the accelerator pedal, the driver sets the engine-speed control lever to a specific position corresponding to a desired (higher) speed. As a result of this adjustment of the control-lever position, the governor spring is tensioned by a given amount, with the result that the governor-spring force exceeds the centrifugal force of the flyweights and causes the start lever and the tensioning lever to pivot around fulcrum M_2 . Due to the mechanical transmission ratio designed into the system, the control collar shifts in the "Full-load" direction. As a result, the delivery quantity is increased and the engine speed rises. This causes the flyweights to generate more force which, through the sliding sleeve, opposes the governor-spring force. The control collar remains in the "Full-load" position until a torque balance occurs. If the engine speed continues to increase, the flyweights separate even further, the sliding-sleeve force prevails,

and as a result the start and tensioning levers pivot around M_2 and push the control collar in the "Stop" direction so that the control port is opened sooner. It is possible to reduce the delivery quantity to "zero" which ensures that engine-speed limitation takes place. This means that during operation, and as long as the engine is not overloaded, every position of the engine-speed control lever is allocated to a specific speed range between full-load and zero. The result is that within the limits set by its speed droop, the governor maintains the desired speed (Fig. 4).
If the load increases to such an extent (for instance on a gradient) that even though the control collar is in the full-load position the engine speed continues to drop, this indicates that it is impossible to increase fuel delivery any further. This means that the engine is overloaded and the driver must change down to a lower gear.

Fig. 5



Overrun (engine braking)

During downhill operation the engine is “driven” by the vehicle, and engine speed tends to increase. This causes the flyweights to move outwards so that the sliding sleeve presses against the tensioning and start levers. Both levers change their position and push the control collar in the direction of less fuel delivery until a reduced fuel-delivery figure is reached which corresponds to the new loading level. At the extreme, the delivery figure is zero. Basically, with the variable-speed governor, this process applies for all settings of the engine-speed control lever, when the engine load or engine speed changes to such an extent that the control collar shifts to either its full-load or stop position.

Minimum-maximum-speed governor

The minimum-maximum-speed governor controls (governs) only the idle (minimum) speed and the maximum speed. The speed range between these points is directly controlled by the accelerator pedal (Fig. 6).

Design and construction

The governor assembly with flyweights, and the lever configuration, are comparable with those of the variable-speed governor already dealt with. The main difference lies in the governor spring and its installation. It is in the form of a compression spring and is held in a guide element. Tensioning lever and governor spring are connected by a retaining pin.

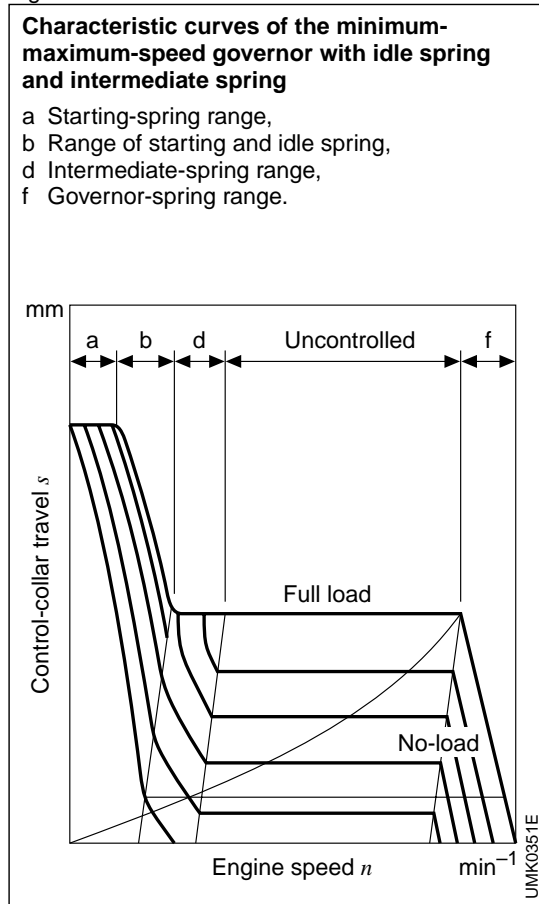
Starting

With the engine at standstill, the flyweights are also stationary and the sliding sleeve is in its initial position. This enables the starting spring to push the flyweights to their inner position through the starting lever and the sliding sleeve. On the distributor plunger, the control collar is in the start-quantity position.

Idle control

Once the engine is running and the accelerator pedal has been released, the engine-speed control lever is pulled back to the idle position by its return spring. The centrifugal force generated by the flyweights increases along with engine speed (Fig. 7a) and the inner flyweight legs push the sliding sleeve up against the start lever. The idle spring on the tensioning lever is responsible for the controlling action. The control collar is shifted in the direction of “less delivery” by the pivoting action of the start lever, its position being determined by interaction between centrifugal force and spring force.

Fig. 6



Operation under load

If the driver depresses the accelerator pedal, the engine-speed control lever is pivoted through a given angle. The starting and idle springs are no longer effective and the intermediate spring comes into effect. The intermediate spring on the minimum-maximum-speed governor provides a "soft" transition to the uncontrolled range. If the engine-speed control lever is pressed even further in the full-load direction, the intermediate spring is compressed until the tensioning lever abuts against the retaining pin (Fig. 7b). The intermediate spring is now ineffective and the uncontrolled range has been entered. This uncontrolled range is a function of the governor-spring pretension, and in this range the spring can be regarded as a solid element. The accelerator-pedal position (engine-speed control lever) is now transferred directly through the governor lever mechanism to the control collar, which means that the injected

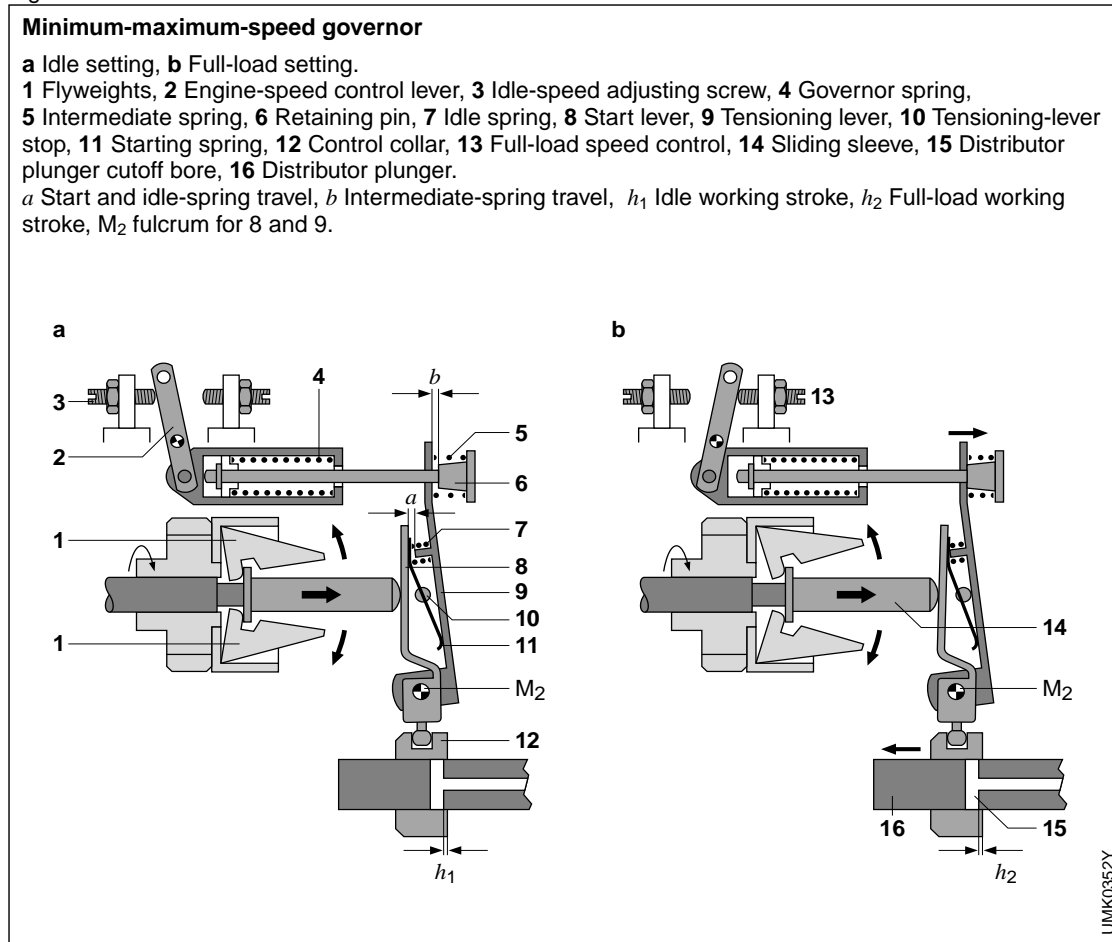
fuel quantity is directly determined by the accelerator pedal. To accelerate, or climb a hill, the driver must "give gas", or ease off on the accelerator if less engine power is needed.

If engine load is now reduced, with the engine-speed control lever position unchanged, engine speed increases without an increase in fuel delivery. The flyweights' centrifugal force also increases and pushes the sliding sleeve even harder against the start and tensioning levers. Full-load speed control does not set in, at or near the engine's rated speed, until the governor-spring pre-tension has been overcome by the effect of the sliding-sleeve force.

If the engine is relieved of all load, speed increases to the high idle speed, and the engine is thus protected against over-revving.

Passenger cars are usually equipped with a combination of variable-speed governor and minimum-maximum-speed governor.

Fig. 7



Injection timing

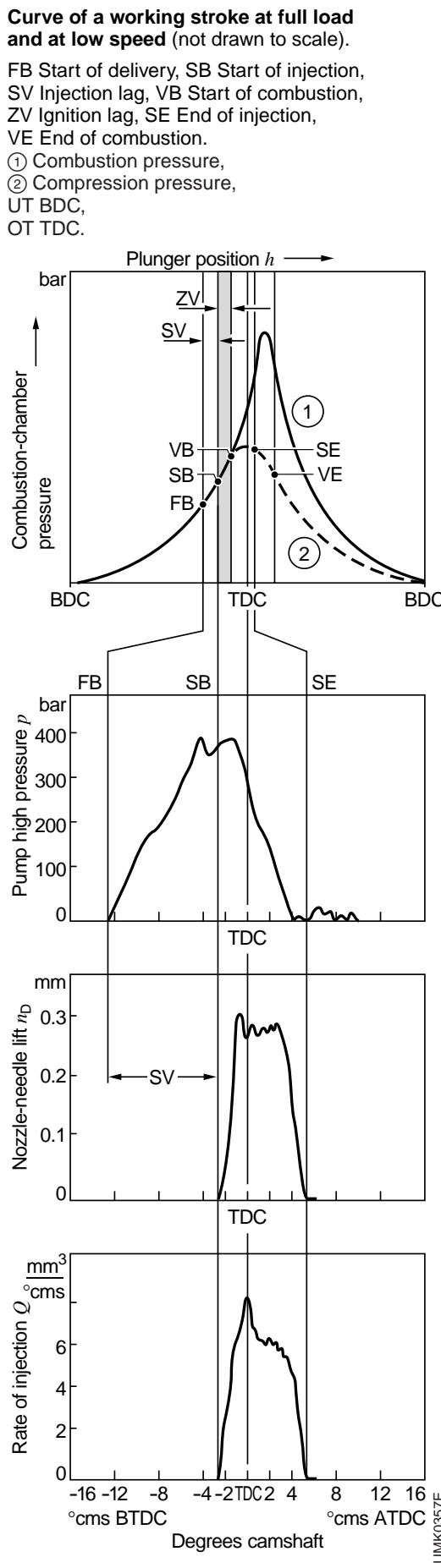
In order to compensate for the injection lag and the ignition lag, as engine speed increases the timing device advances the distributor pump's start of delivery referred to the engine's crankshaft. Example (Fig. 1):

Start of delivery (FB) takes place after the inlet port is closed. The high pressure then builds up in the pump which, as soon as the nozzle-opening pressure has been reached leads to the start of injection (SB). The period between FB and SB is referred to as the injection lag (SV). The increasing compression of the air-fuel mixture in the combustion chamber then initiates the ignition (VB). The period between SB and VB is the ignition lag (ZV). As soon as the cutoff port is opened again the pump pressure collapses (end of pump delivery), and the nozzle needle closes again (end of injection, SE). This is followed by the end of combustion (VE).

Assignment

During the fuel-delivery process, the injection nozzle is opened by a pressure wave which propagates in the high-pressure line at the speed of sound. Basically speaking, the time required for this process is independent of engine speed, although with increasing engine speed the crankshaft angle between start of delivery and start of injection also increases. This must be compensated for by advancing the start of delivery. The pressure wave's propagation time is determined by the length of the high-pressure line and the speed of sound which is approx. 1,500 m/s in diesel fuel. The interval represented by this propagation time is termed the injection lag. In other words, the start of injection lags behind the start of delivery. This phenomena is the reason for the injector opening later (referred to the engine's piston position) at higher engine speeds than at low engine speeds. Following injection, the injected fuel needs a certain time in

Fig. 1



order to atomize and mix with the air to form an ignitable mixture.

This is termed the air-fuel mixture preparation time and is independent of engine speed. In a diesel engine, the time required between start of injection and start of combustion is termed the ignition lag.

The ignition lag is influenced by the diesel fuel's ignition quality (defined by the Cetane Number), the compression ratio, the intake-air temperature, and the quality of fuel atomization. As a rule, the ignition lag is in the order of 1 millisecond. This means that presuming a constant start of injection, the crankshaft angle between start of injection and start of combustion increases along with increasing engine speed. The result is that combustion can no longer start at the correct point (referred to the engine-piston position). Being as the diesel engine's most efficient combustion and power can only be developed at a given crankshaft or

piston position, this means that the injection pump's start of delivery must be advanced along with increasing engine speed in order to compensate for the overall delay caused by ignition lag and injection lag. This start-of-delivery advance is carried out by the engine-speed-dependent timing device.

Timing device

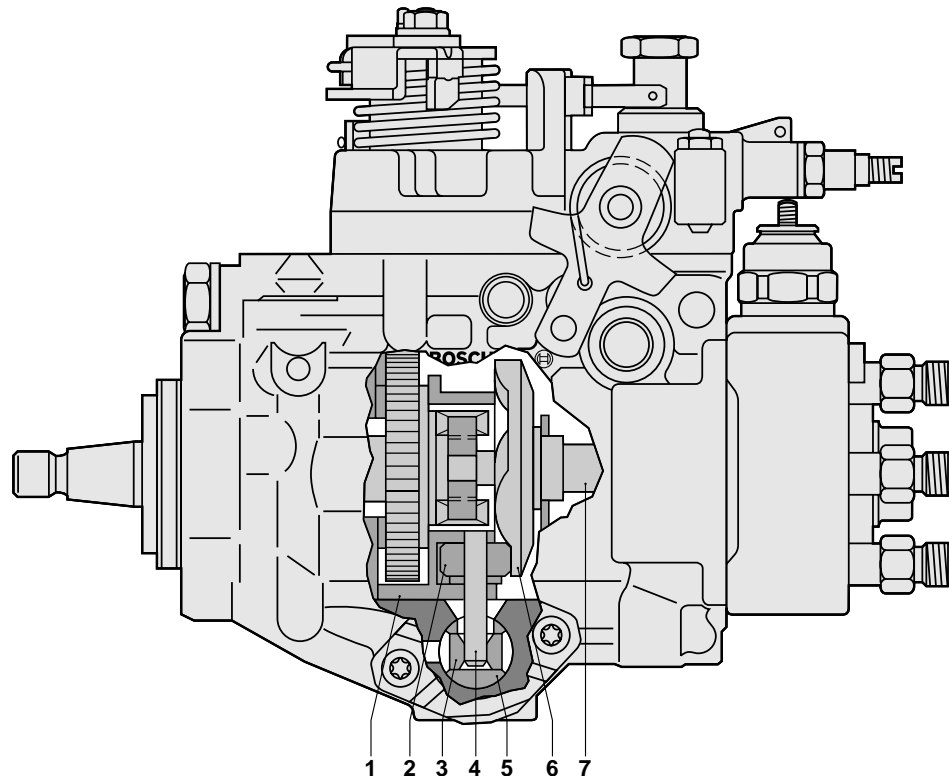
Design and construction

The hydraulically controlled timing device is located in the bottom of the distributor pump's housing, at right angles to the pump's longitudinal axis (Fig. 2), whereby its piston is free to move in the pump housing. The housing is closed with a cover on each side. There is a passage in one end of the timing device plunger through which the fuel can enter, while at the other end the plunger is held by a compression spring. The piston is connected to the roller ring

Fig. 2

Distributor injection pump with timing device

- 1 Roller ring, 2 Roller-ring rollers, 3 Sliding block, 4 Pin, 5 Timing-device piston, 6 Cam plate, 7 Distributor plunger.



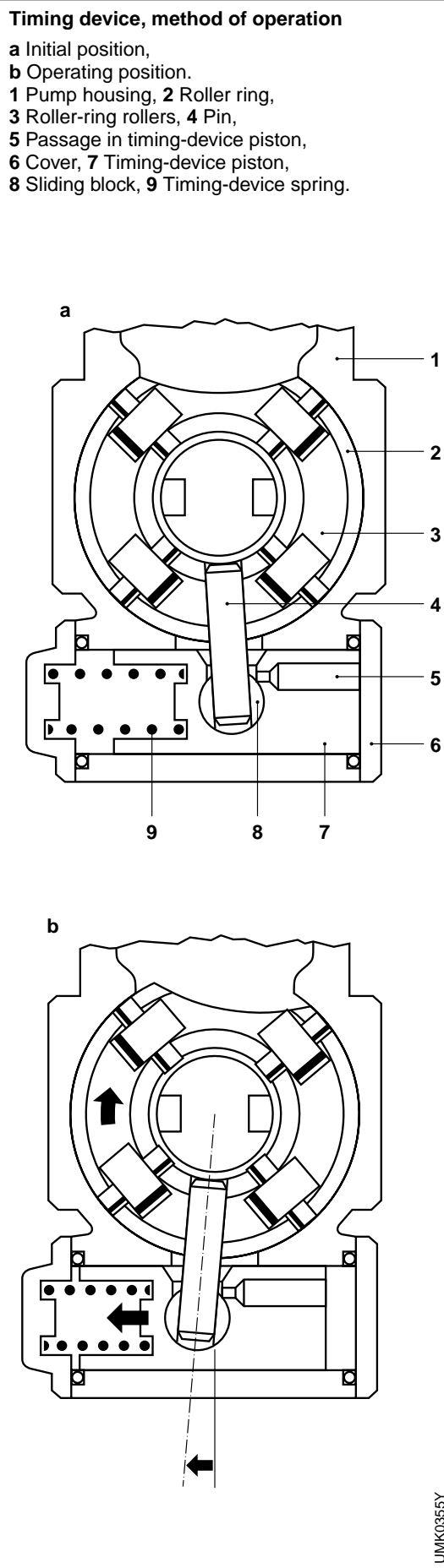
through a sliding block and a pin so that piston movement can be converted to rotational movement of the roller ring.

Method of operation

The timing-device piston is held in its initial position by the timing-device spring (Fig. 3a). During operation, the pressure-control valve regulates the fuel pressure inside the pump so that it is proportional to engine speed. As a result, the engine-speed-dependent fuel pressure is applied to the end of the timing-device piston opposite to the spring.

As from about 300 min^{-1} , the fuel pressure inside the pump overcomes the spring preload and shifts the timing-device piston to the left and with it the sliding block and the pin which engages in the roller ring (Fig. 3b). The roller ring is rotated by movement of the pin, and the relative position of the roller ring to the cam plate changes with the result that the rollers lift the rotating cam plate at an earlier moment in time. In other words, the roller ring has been rotated through a defined angle with respect to the cam plate and the distributor plunger. Normally, the maximum angle is 12 degrees camshaft (24 degrees crankshaft).

Fig. 3



Add-on modules and shutoff devices

Application

The distributor injection pump is built according to modular construction principles, and can be equipped with a variety of supplementary (add-on) units (Fig. 1). These enable the implementation of a wide range of adaptation possibilities with regard to optimization of engine torque, power output, fuel economy, and exhaust-gas composition. The overview provides a summary of

the add-on modules and their effects upon the diesel engine. The schematic (Fig. 2) shows the interaction of the basic distributor pump and the various add-on modules.

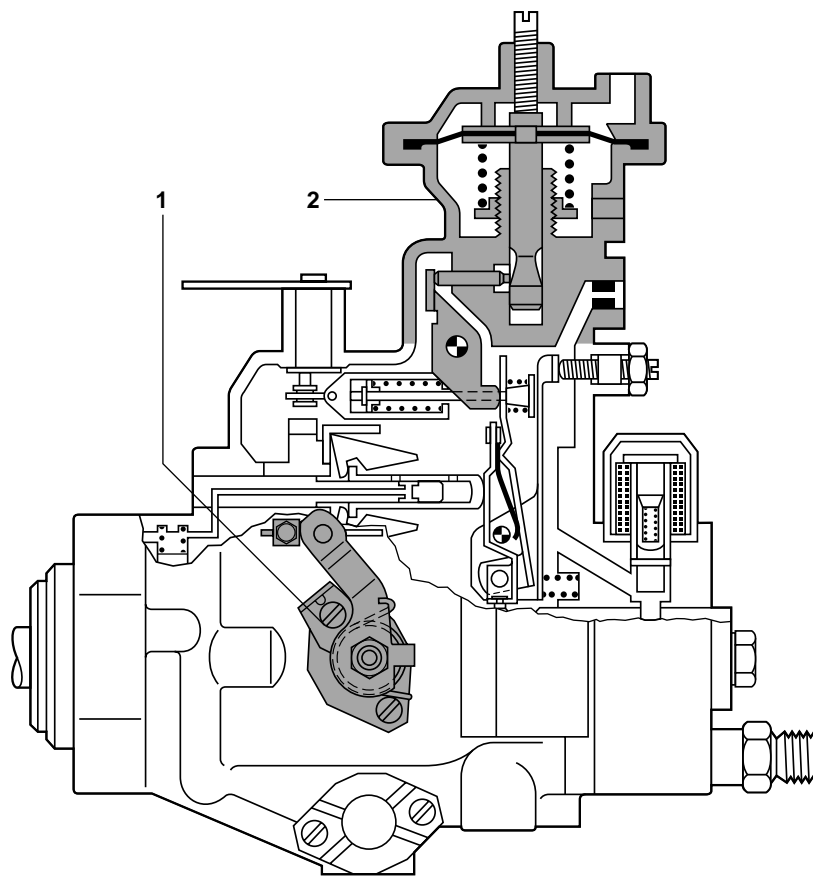
Torque control

Torque control is defined as varying fuel delivery as a function of engine speed in order to match it to the engine's fuel-requirement characteristic. If there are special stipulations with regard to the full-load characteristic (optimization of exhaust-gas composition, of torque characteristic curve, and of fuel economy), it may be necessary

Fig. 1

Distributor injection pump with add-on modules

- 1 Cold-start accelerator,
- 2 Manifold-pressure compensator.



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Fig. 2

Schematic of the VE distributor pump with mechanical/hydraulic full-load torque control

LDA Manifold-pressure compensator.

Controls the delivery quantity as a function of the charge-air pressure.

HBA Hydraulically controlled torque control.

Controls the delivery quantity as a function of the engine speed (not for pressure-charged engines with LDA).

LFB Load-dependent start of delivery.

Adaptation of pump delivery to load. For reduction of noise and exhaust-gas emissions.

ADA Altitude-pressure compensator.

Controls the delivery quantity as a function of atmospheric pressure.

KSB Cold-start accelerator.

Improves cold-start behavior by changing the start of delivery.

GST Graded (or variable) start quantity.

Prevents excessive start quantity during warm start.

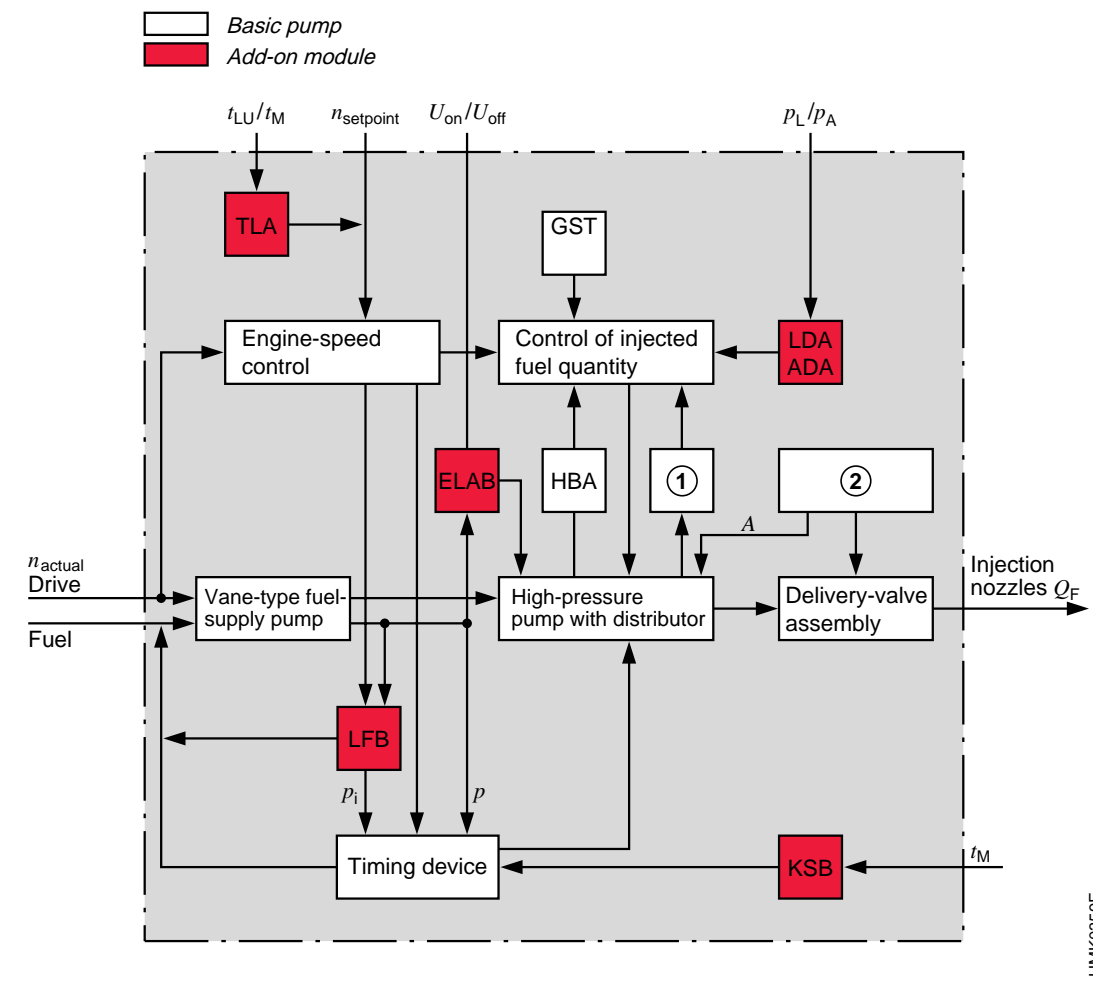
TLA Temperature-controlled idle-speed increase.

Improves engine warm-up and smooth running when the engine is cold.

ELAB Electrical shutoff device.

A Cutoff port, n_{actual} Actual engine speed (controlled variable), $n_{setpoint}$ Desired engine speed (reference variable), 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 interior pressure.

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



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Add-on modules and shutoff devices

to install torque control. In other words, the engine should receive precisely the amount of fuel it needs. The engine's fuel requirement first of all climbs as a function of engine speed and then levels off somewhat at higher speeds. The fuel-delivery curve of an injection pump without torque control is shown in Fig. 3. As can be seen, with the same setting of the control collar on the distributor plunger, the injection pump delivers slightly more fuel at high speeds than it does at lower speeds. This is due to the throttling effect at the distributor plunger's cutoff port. This means that if the injection pump's delivery quantity is specified so that maximum-possible torque is developed at low engine speeds, this would lead to the engine being unable to completely combust the excess fuel injected at higher speeds and smoke would be the result together with engine overheat. On the other hand, if the maximum delivery quantity is specified so that it corresponds to the engine's requirements at maximum speed and full-load, the engine will not be able to develop full power at low engine speeds due to the delivery quantity dropping along with reductions in engine speed. Performance would be below optimum. The injected fuel quantity must therefore be adjusted to the engine's

actual fuel requirements. This is known as "torque control", and in the case of the distributor injection pump can be implemented using the delivery valve, the cutoff port, or an extended governor-lever assembly, or the hydraulically controlled torque control (HBA). Full-load torque control using the governor lever assembly is applied in those cases in which the positive full-load torque control with the delivery valve no longer suffices, or a negative full-load torque control has become necessary.

Positive torque control

Positive torque control is required on those injection pumps which deliver too much fuel at higher engine revs. The delivery quantity must be reduced as engine speed increases.

Positive torque control using the delivery valve

Within certain limits, positive torque control can be achieved by means of the delivery valve, for instance by fitting a softer delivery-valve spring.

Positive torque control using the cutoff port

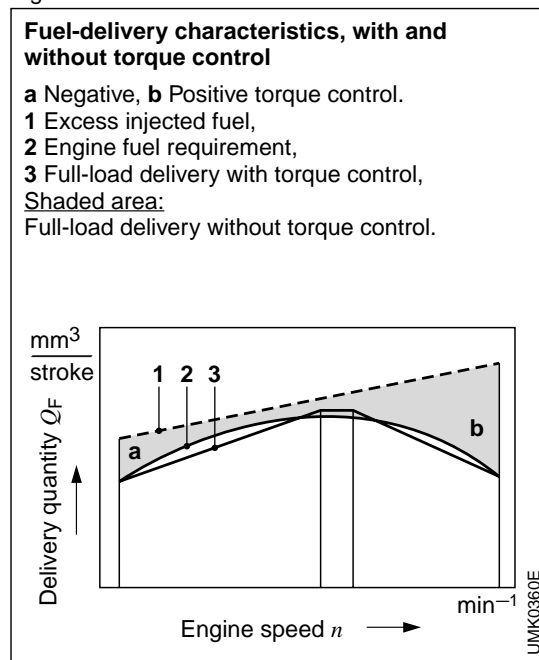
Optimization of the cutoff port's dimensions and shape permit its throttling effect to be utilized for reducing the delivery quantity at higher engine speeds.

Positive torque control using the governor lever assembly (Fig. 4a)

The decisive engine speed for start of torque control is set by preloading the torque-control springs. When this speed is reached, the sliding-sleeve force (F_M) and the spring preload must be in equilibrium, whereby the torque-control lever (6) abuts against the stop lug (5) of the tensioning lever (4). The free end of the torque-control lever (6) abuts against the torque-control pin (7).

If engine speed now increases, the sliding-sleeve force acting against the starting lever (1) increases and the common pivot point (M_4) of starting lever and torque-control lever (6) changes its position. At the same time,

Fig. 3



the torque-control lever tilts around the stop pin (5) and forces the torque-control pin (7) in the direction of the stop, while the starting lever (1) swivels around the pivot point (M_2) and forces the control collar (8) in the direction of reduced fuel delivery. Torque control ceases as soon as the torque-control-pin collar (10) abuts against the starting lever (1).

Negative torque control

Negative torque control may be necessary in the case of engines which have black-smoke problems in the lower speed range, or which must generate specific torque characteristics. Similarly, turbocharged engines also need negative torque control when the manifold-pressure compensator (LDA) has ceased to be effective. In this case, the fuel delivery is increased along with engine speed (Fig. 3).

Negative torque control using the governor lever assembly (Fig. 4b)

Once the starting spring (9) has been compressed, the torque-control lever (6) applies pressure to the tensioning lever (4) through the stop lug (5). The torque-control pin (7) also abuts against the tensioning lever (4). If the sliding-sleeve force (F_M) increases due to rising engine speed, the torque-control lever

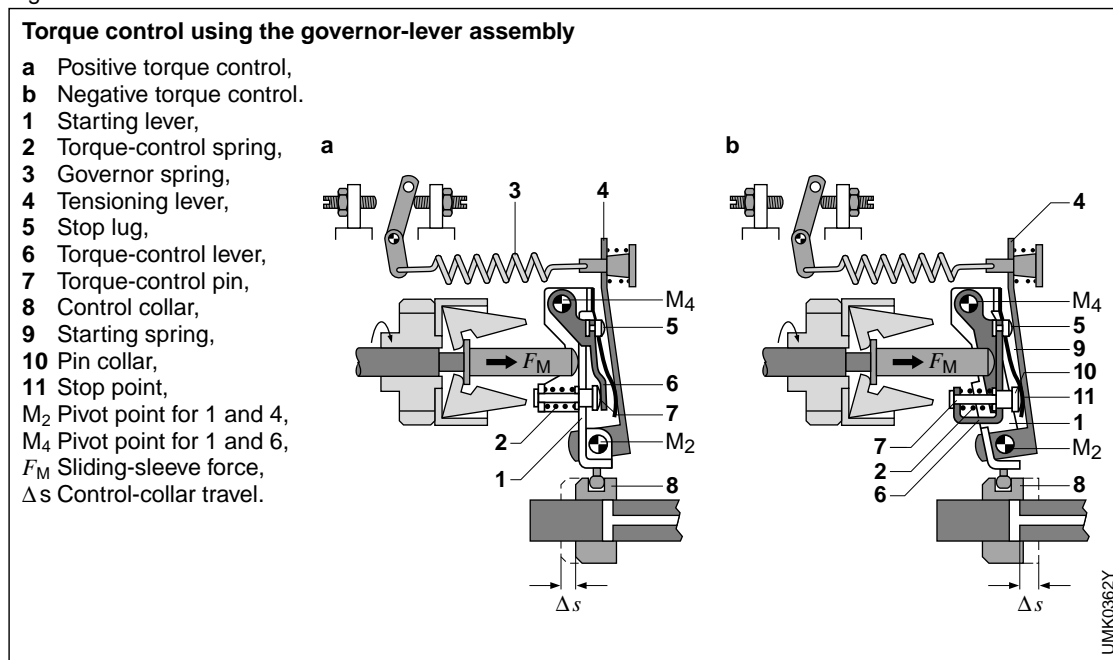
presses against the preloaded torque-control spring. As soon as the sliding-sleeve force exceeds the torque-control spring force, the torque-control lever (6) is forced in the direction of the torque-control-pin collar. As a result, the common pivot point (M_4) of the starting lever and torque-control lever changes its position. At the same time the starting lever swivels around its pivot point (M_2) and pushes the control collar (8) in the direction of increased delivery. Torque control ceases as soon as the torque-control lever abuts against the pin collar.

Negative torque control using hydraulically controlled torque control HBA

In the case of naturally aspirated diesel engines, in order to give a special shape to the full-load delivery characteristic as a function of engine speed, a form of torque control can be applied which is similar to the LDA (manifold-pressure compensator).

Here, the shift force developed by the hydraulic piston is generated by the pressure in the pump interior, which in turn depends upon pump speed. In contrast to spring-type torque control, within limits the shape of the full-load characteristic can be determined by a cam on a sliding pin.

Fig. 4



Manifold-pressure compensation

Exhaust-gas turbocharging

Because it increases the mass of air inducted by the engine, exhaust turbocharging boosts a diesel engine's power output considerably over that of a naturally aspirated diesel engine, with little increase in dimensions and engine speeds. This means that the brake horsepower can be increased corresponding to the increase in air mass (Figure 6). In addition, it is often possible to also reduce the specific fuel consumption. An exhaust-gas turbocharger is used to pressure-charge the diesel engine (Fig. 5).

With an exhaust turbocharger, the engine's exhaust gas, instead of simply being discharged into the atmosphere, is used to drive the turbocharger's turbine at speeds which can exceed $100,000 \text{ min}^{-1}$. Turbine and turbocharger compressor are connected through a shaft. The compressor draws in air, compresses it, and supplies it to the engine's combustion chambers under pressure, whereby not only the air pressure rises but also the air temperature. If temperatures become excessive, some form of air cooling (intercooling) is needed between the turbocharger and the engine intake.

Fig. 5: Diesel engine with exhaust-gas turbocharger



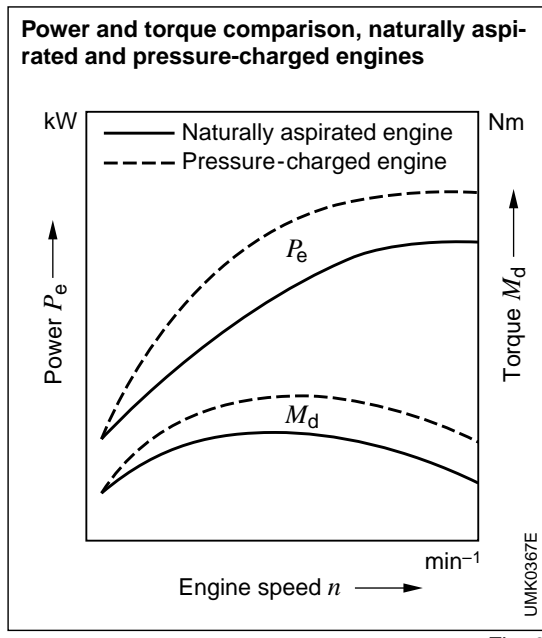


Fig. 6

Manifold-pressure compensator (LDA)

The manifold-pressure compensator (LDA) reacts to the charge-air pressure generated by the exhaust-gas turbo-charger, or the (mechanical) super-charger, and adapts the full-load delivery to the charge-air pressure (Figs. 6 and 7).

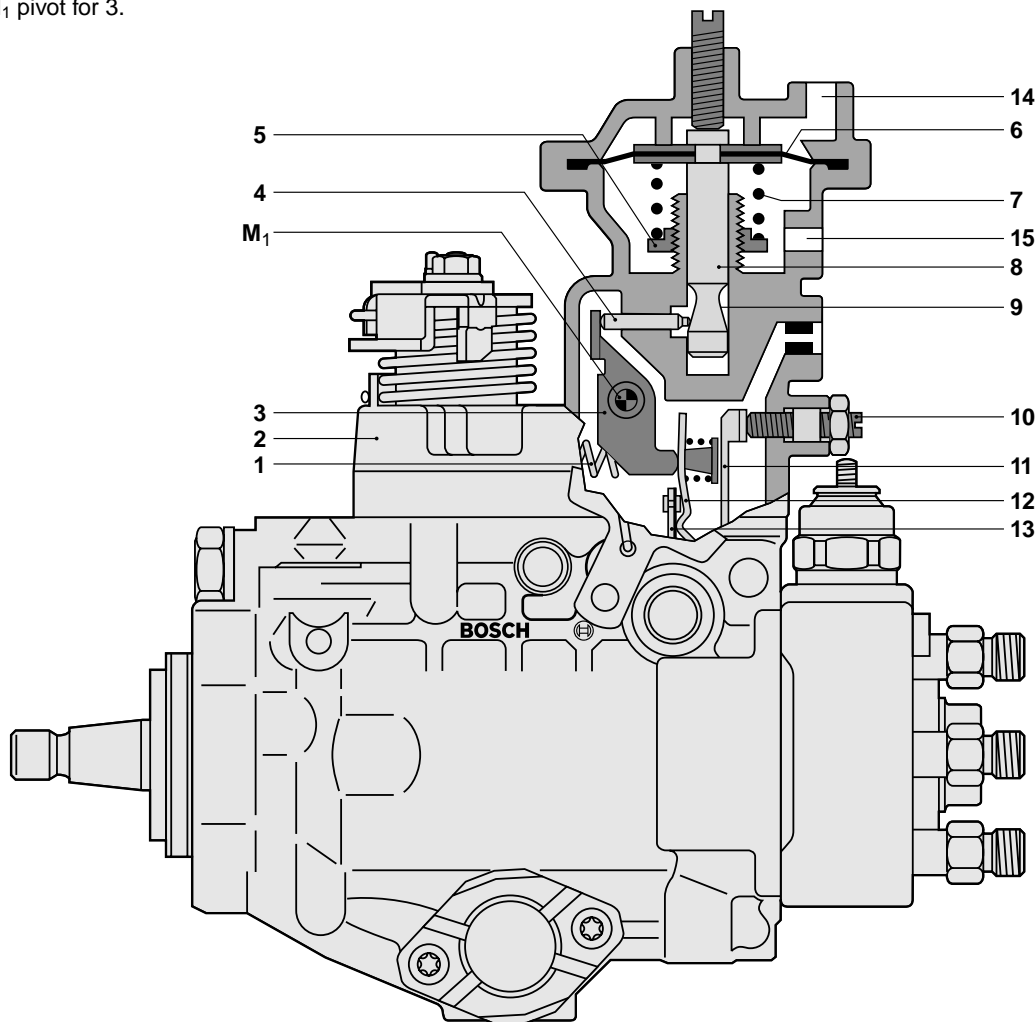
Assignment

The manifold-pressure compensator (LDA) is used on pressure-charged diesel engines. On these engines the injected fuel quantity is adapted to the engine's increased air charge (due to pressure-charging). If the pressure-charged diesel engine operates with a reduced cylinder air charge, the in-

Fig. 7

Distributor injection pump with manifold-pressure compensator (LDA)

- 1 Governor spring, 2 Governor cover, 3 Reverse lever, 4 Guide pin, 5 Adjusting nut, 6 Diaphragm,
 - 7 Compression spring, 8 Sliding pin, 9 Control cone, 10 Full-load adjusting screw, 11 Adjusting lever,
 - 12 Tensioning lever, 13 Starting lever, 14 Connection for the charge-air, 15 Vent bore.
- M_1 pivot for 3.



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jected fuel quantity must be adapted to the lower air mass. This is performed by the manifold-pressure compensator which, below a given (selectable) charge-air pressure, reduces the full-load quantity.

Design and construction

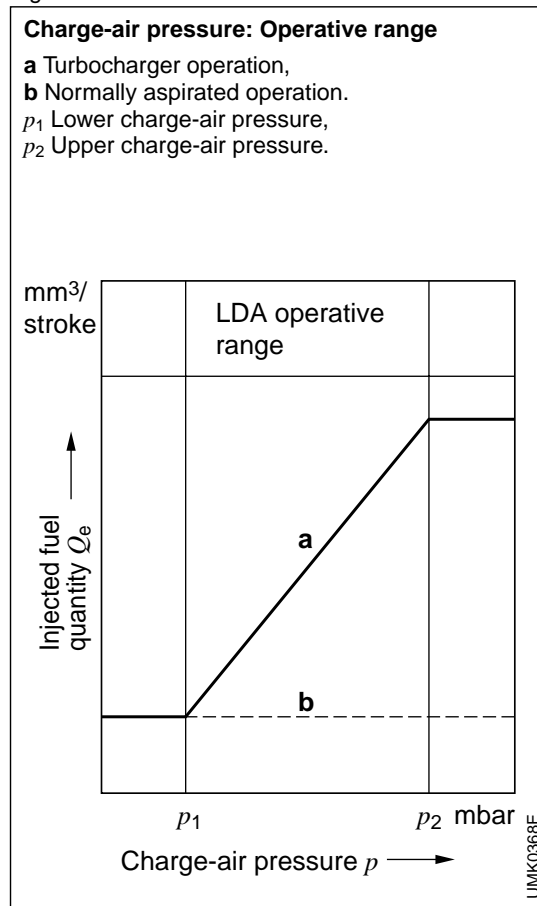
The LDA is mounted on the top of the distributor pump (Fig. 7). In turn, the top of the LDA incorporates the connection for the charge-air and the vent bore. The interior of the LDA is divided into two separate airtight chambers by a diaphragm to which pressure is applied by a spring. At its opposite end, the spring is held by an adjusting nut with which the spring's preload is set. This serves to match the LDA's response point to the charge pressure of the exhaust turbocharger. The diaphragm is connected to the LDA's sliding pin which has a taper in the form of a control cone. This is contacted by a guide pin which transfers the sliding-pin movements to the reverse lever which in turn changes the setting of the full-load stop. The initial setting of the diaphragm and the sliding pin is set by the adjusting screw in the top of the LDA.

Method of operation

In the lower engine-speed range the charge-air pressure generated by the exhaust turbocharger and applied to the diaphragm is insufficient to overcome the pressure of the spring. The diaphragm remains in its initial position. As soon as the charge-air pressure applied to the diaphragm becomes effective, the diaphragm, and with it the sliding pin and control cone, shift against the force of the spring. The guide pin changes its position as a result of the control cone's vertical movement and causes the reverse lever to swivel around its pivot point M_1 (Fig. 7). Due to the force exerted by the governor spring, there is a non-positive connection between tensioning lever, reverse lever, guide pin, and sliding-pin control cone. As a result, the tensioning lever follows the reverse lever's swivelling movement, causing the

starting lever and tensioning lever to swivel around their common pivot point thus shifting the control collar in the direction of increased fuel delivery. Fuel delivery is adapted in response to the increased air mass in the combustion chamber (Fig. 8). On the other hand, when the charge-air pressure drops, the spring underneath the diaphragm pushes the diaphragm upwards, and with it the sliding pin. The compensation action of the governor lever mechanism now takes place in the reverse direction and the injected fuel quantity is adapted to the change in charge pressure. Should the turbocharger fail, the LDA reverts to its initial position and the engine operates normally without developing smoke. The full-load delivery with charge-air pressure is adjusted by the full-load stop screw fitted in the governor cover.

Fig. 8



Load-dependent compensation

Depending upon the diesel engine's load, the injection timing (start of delivery) must be adjusted either in the "advance" or "retard" direction.

Load-dependent start of delivery (LFB)

Assignment

Load-dependent start of delivery is designed so that with decreasing load (e.g., change from full-load to part-load), with the control-lever position unchanged, the start of delivery is shifted in the "retard" direction. And when engine load increases, the start of delivery (or start of injection) is shifted in the "advance" direction. These adjustments lead to "softer" engine operation, and cleaner exhaust gas at part- and full-load.

Design and construction

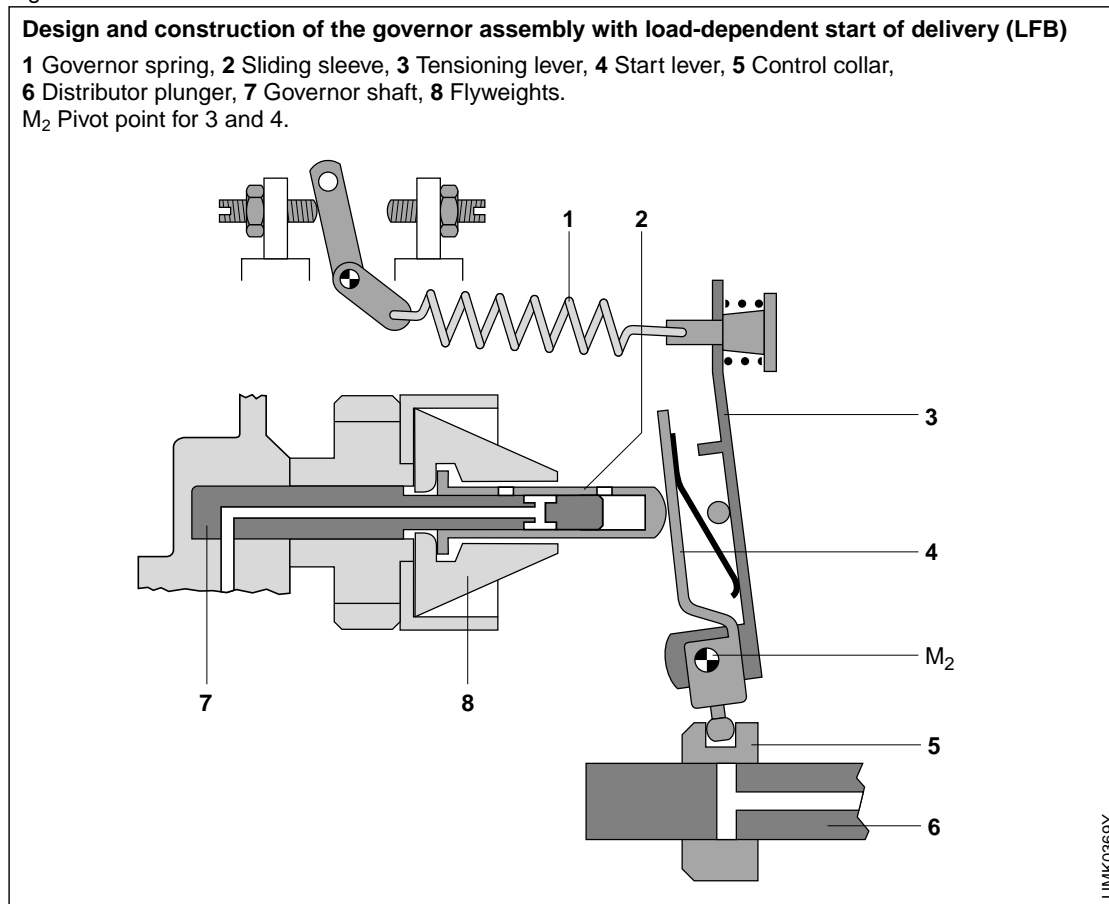
For load-dependent injection timing, modifications must be made to the governor shaft, sliding sleeve, and pump housing. The sliding sleeve is provided with an additional cutoff port, and the governor shaft with a ring-shaped groove, a longitudinal passage and two transverse passages (Fig. 9). The pump housing is provided with a bore so that a connection is established from the interior of the pump to the suction side of the vane-type supply pump.

Method of operation

As a result of the rise in the supply-pump pressure when the engine speed increases, the timing device adjusts the start of delivery in the "advance" direction. On the other hand, with the drop in the pump's interior pressure caused by the LFB it is possible to implement a (relative) shift in the "retard" direction. This is controlled by the ring-shaped groove in the governor shaft and the sliding-sleeve's control port. The control

Add-on modules and shutoff devices

Fig. 9



lever is used to input a given full-load speed. If this speed is reached and the load is less than full load, the speed increases even further, because with a rise in speed the flyweights swivel outwards and shift the sliding sleeve. On the one hand, this reduces the delivery quantity in line with the conventional governing process. On the other, the sliding sleeve's control port is opened by the control edge of the governor-shaft groove. The result is that a portion of the fuel now flows to the suction side through the governor shaft's longitudinal and transverse passages and causes a pressure drop in the pump's interior. This pressure drop results in the timing-device piston moving to a new position. This leads to the roller ring being turned in the direction of pump rotation so that start of delivery is shifted in the "retard" direction. If the position of the control lever remains unchanged and the load increases again, the engine speed drops. The flyweights move inwards and the sliding sleeve is shifted so that its control

port is closed again. The fuel in the pump interior can now no longer flow through the governor shaft to the suction side, and the pump interior pressure increases again. The timing-device piston shifts against the force of the timing-device spring and adjusts the roller ring so that start of delivery is shifted in the "advance" direction (Fig. 10).

Atmospheric-pressure compensation

At high altitudes, the lower air density reduces the mass of the inducted air, and the injected full-load fuel quantity cannot burn completely. Smoke results and engine temperature rises. To prevent this, an altitude-pressure compensator is used to adjust the full-load quantity as a function of atmospheric pressure.

Altitude-pressure compensator (ADA)

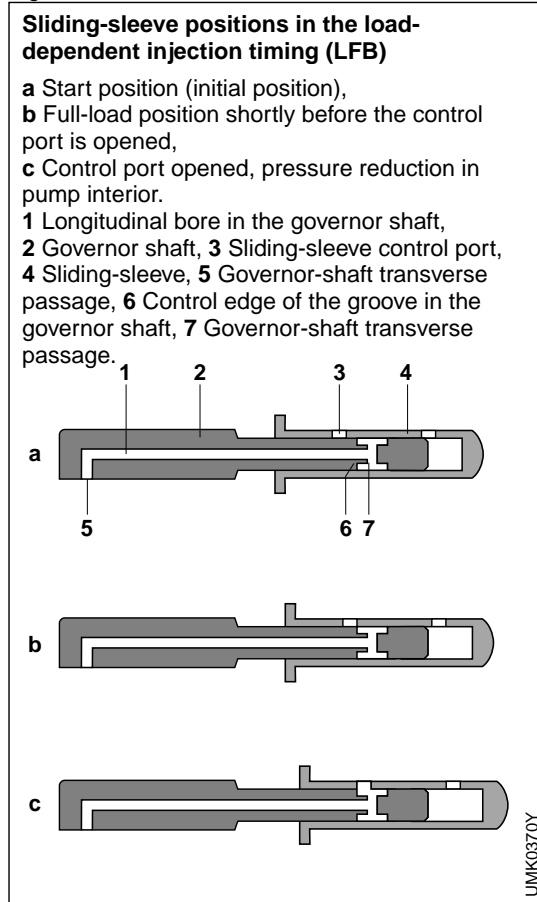
Design and construction

The construction of the ADA is identical to that of the LDA. The only difference being that the ADA is equipped with an aneroid capsule which is connected to a vacuum system somewhere in the vehicle (e.g., the power-assisted brake system). The aneroid provides a constant reference pressure of 700mbar (absolute).

Method of operation

Atmospheric pressure is applied to the upper side of the ADA diaphragm. The reference pressure (held constant by the aneroid capsule) is applied to the diaphragm's underside. If the atmospheric pressure drops (for instance when the vehicle is driven in the mountains), the sliding bolt shifts vertically away from the lower stop and, similar to the LDA, the reverse lever causes the injected fuel quantity to be reduced.

Fig. 10



Cold-start compensation

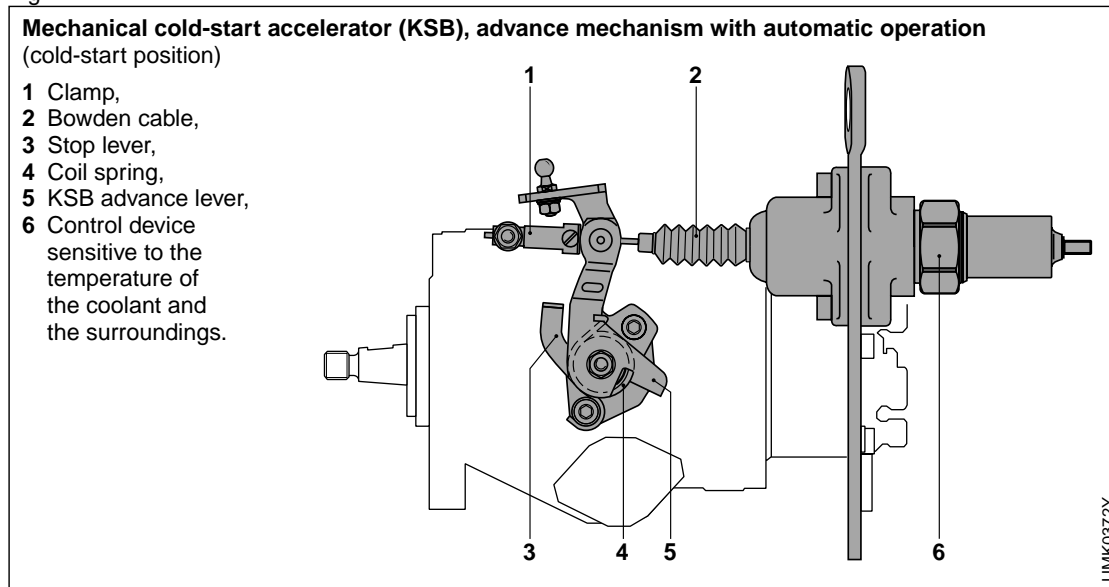
The diesel engine's cold-start characteristics are improved by fitting a cold-start compensation module which shifts the start of injection in the "advance" direction. Operation is triggered either by the driver using a bowden cable in the cab, or automatically by means of a temperature-sensitive advance mechanism (Fig. 11).

Mechanical cold-start accelerator (KSB) on the roller ring

Design and construction

The KSB is attached to the pump housing, the stop lever being connected through a shaft to the inner lever on which a ball pin is eccentrically mounted. The ball pin's head extends into the roller ring (a version is available in which the advance mechanism engages in the timing-device piston). The stop lever's initial position is defined by the stop itself and by the helical coiled spring. Attached to the top of the stop lever is a bowden cable which serves as the connection to the manual or to the automatic advance mechanism. The automatic advance mechanism is mounted on the distributor pump, whereas the manual operating mechanism is in the driver's cab (Fig. 12).

Fig. 11



Mechanical cold-start accelerator (KSB) engaging in roller ring (cold-start position)

- 1 Lever,
- 2 Access window,
- 3 Ball pin,
- 4 Longitudinal slot,
- 5 Pump housing,
- 6 Roller ring,
- 7 Roller in the roller ring,
- 8 Timing-device piston,
- 9 Torque-control pin,
- 10 Sliding block,
- 11 Timing-device spring,
- 12 Shaft,
- 13 Coil spring.

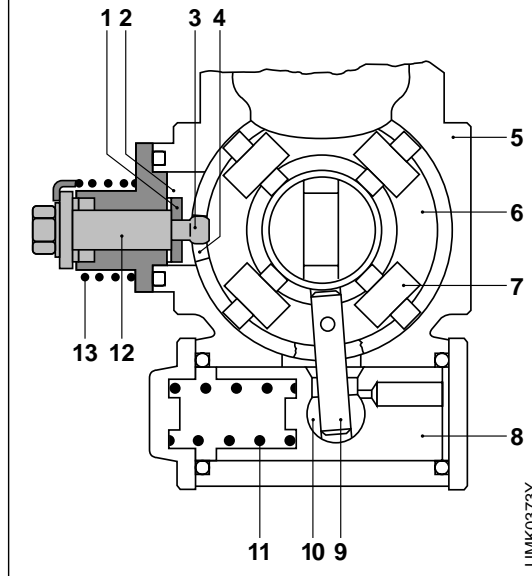


Fig. 12

Method of operation

Automatically and manually operated cold-start accelerators (KSB) differ only with regard to their external advance mechanisms. The method of operation is identical. With the bowden cable not pulled, the coil spring pushes the stop lever up against the stop. Ball pin and roller ring are in their initial position. The force applied by the bowden cable

*Add-on
modules
and shutoff
devices*

causes the stop lever, the shaft, the inner lever and the ball pin, to swivel and change the roller ring's setting so that the start of delivery is advanced. The ball pin engages in a slot in the roller ring, which means that the timing-device piston cannot rotate the roller ring any further in the "advance" direction until a given engine speed has been exceeded.

In those cases in which the KSB is triggered by the driver from the cab (timing-device KSB), independent of the advance defined by the timing device (a), an advance of approx. 2.5° camshaft is maintained (b), as shown in Fig. 13. With the automatically operated KSB, this advance depends upon the engine temperature or ambient temperature.

The automatic advance mechanism uses a control device in which a temperature-sensitive expansion element converts the engine temperature into a stroke movement. The advantage of this method is that for a given temperature, the optimum start of delivery (or start of injection) is always selected.

There are a number of different lever configurations and operating mechanisms in use depending upon the direction of rotation, and on which side the KSB is mounted.

Temperature-controlled idle-speed increase (TLA)

The TLA is also operated by the control device and is combined with the KSB. Here, when the engine is cold, the ball pin at the end of the elongated KSB advance lever presses against the engine-speed control lever and lifts it away from the idle-speed stop screw. The idle speed increases as a result, and rough running is avoided. When the engine has warmed up, the KSB advance lever abuts against its stop and, as a result, the engine-speed control lever is also up against its stop and the TLA is no longer effective (Fig. 14).

Hydraulic cold-start accelerator

Advancing the start of injection by shifting the timing-device piston has only limited applications. In the case of the hydraulic start-of-injection advance, the speed-dependent pump interior pressure is applied to the timing-device piston. In order to implement a start-of-injection advance, referred to the conventional timing-device curve, the pump interior pressure is increased automatically. To do so, the automatic control of pump interior pressure is modified through a bypass in the pressure-holding valve.

Fig. 13

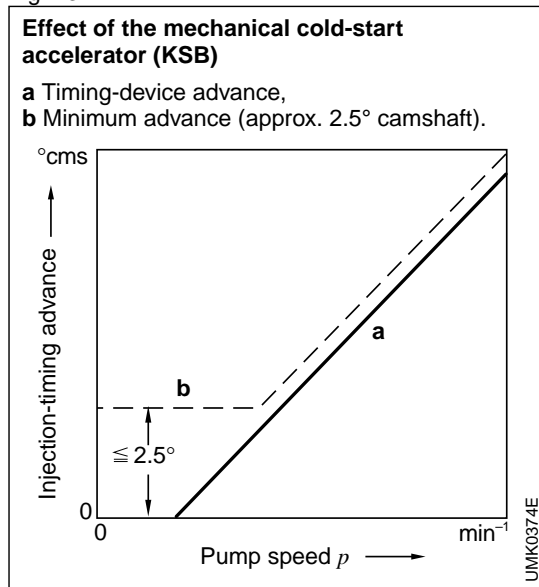
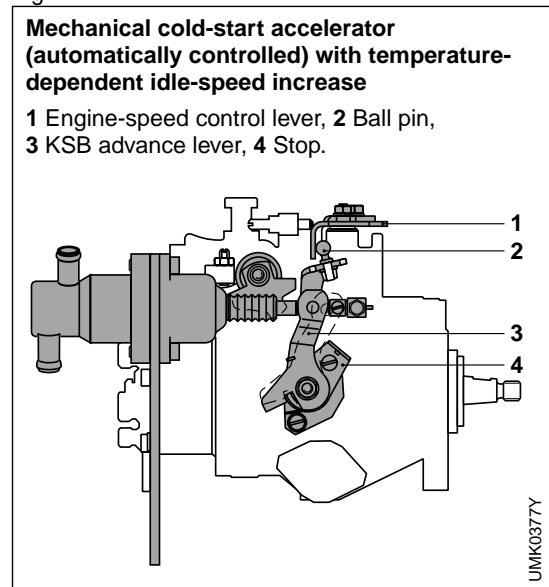


Fig. 14



Design and construction

The hydraulic cold-start accelerator comprises a modified pressure-control valve, a KSB ball valve, a KSB control valve, and an electrically heated expansion element.

Method of operation

The fuel delivered by the fuel-supply pump is applied to one of the timing device piston's end faces via the injection pump's interior. In accordance with the injection pump's interior pressure, the piston is shifted against the force of its spring and changes the start-of-injection timing. Pump interior pressure is determined by a pressure-control valve which increases pump interior pressure along with increasing pump speed and the resulting rise in pump delivery (Fig. 15).

There is a restriction passage in the pressure-control valve's plunger in order to achieve the pressure increase needed for the KSB function, and the resulting advance curve shown as a dotted line in Fig. 16. This ensures that the same pressure is effective at the spring side of the pressure-control valve. The KSB ball-type valve has a correspondingly higher pressure level and is used in conjunction with the thermo-element both for switching-on and switching-off the KSB function, as well as for safety switchoff. Using an

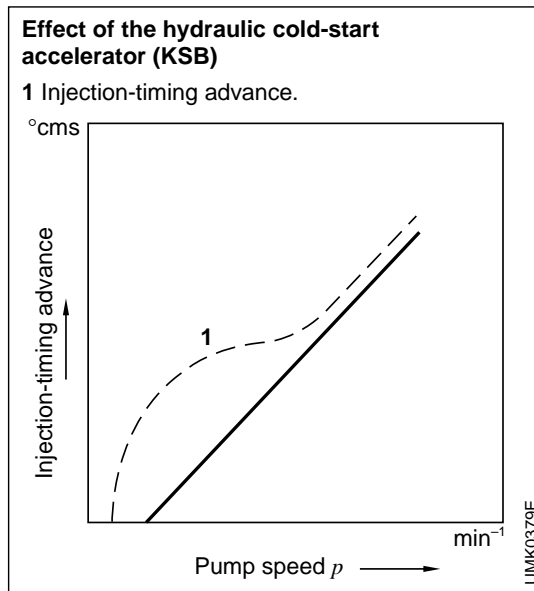
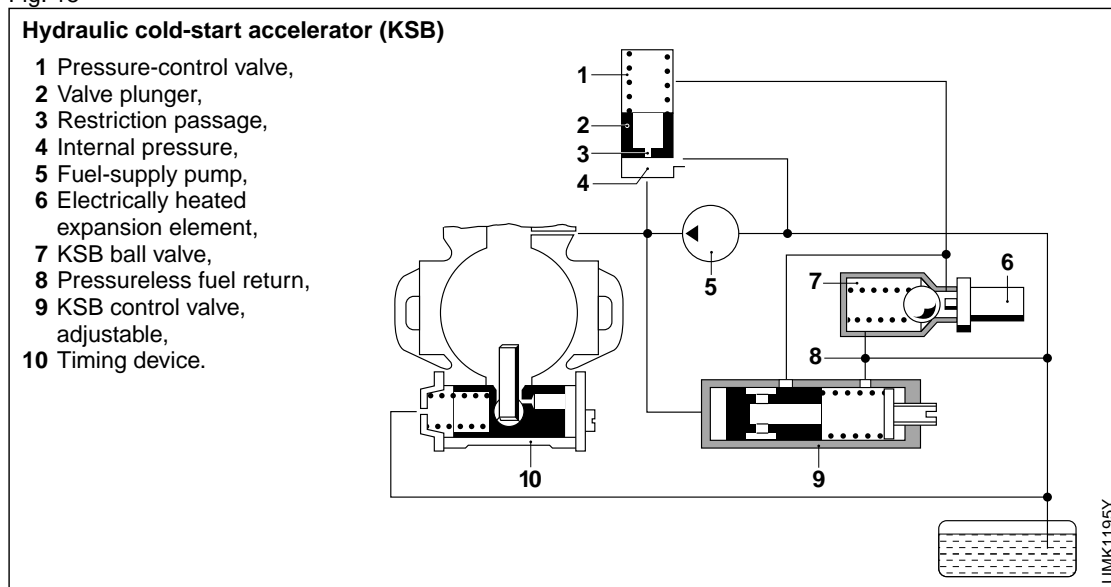


Fig. 16

adjusting screw in the integrated KSB control valve, the KSB function can be set to a given engine speed. The fuel supply pump pressure shifts the KSB control valve's plunger against the force of a spring. A damping restriction is used to reduce the pressure fluctuations at the control plunger. The KSB pressure characteristic is controlled by its plunger's control edge and the section at the valve holder. The KSB function is adapted by correct selection of the KSB control valve's spring rate and its control section. When the warm engine is started, the expansion element has already opened the ball valve due to the prevailing temperature.

Fig. 15



Add-on modules and shutoff devices

Engine shutoff

Assignment

The principle of auto-ignition as applied to the diesel engine means that the engine can only be switched off by interrupting its supply of fuel.

Normally, the mechanically governed distributor pump is switched off by a solenoid-operated shutoff (ELAB). Only in special cases is it equipped with a mechanical shutoff device.

Electrical shutoff device (ELAB)

The electrical shutoff (Fig. 17) using the vehicle's key-operated starting switch is coming more and more to the forefront due to its convenience for the driver.

On the distributor pump, the solenoid valve for interrupting the fuel supply is installed in the top of the distributor head. When the engine is running, the solenoid is energized and the valve keeps the passage into the injection pump's high-pressure chamber open (armature with sealing cone has pulled in). When the driving switch is turned to "OFF", the current to the solenoid winding is also cut, the magnetic field collapses, and the spring forces the armature and sealing cone back onto the valve seat again. This closes the inlet passage to the high-pressure chamber, the distributor-pump plunger ceases to deliver fuel, and the engine stops. From the circuitry point of view, there are a variety of different possibilities for implementing the electrical shutoff (pull or push solenoid).

Mechanical shutoff device

On the injection pump, the mechanical shutoff device is in the form of a lever assembly (Fig. 18). This is located in the governor cover and comprises an outer and an inner stop lever. The outer lever is operated by the driver from inside the vehicle (for instance by means of bowden cable). When the cable is pulled, both levers swivel around their common pivot point, whereby the inner

stop lever pushes against the start lever of the governor-lever mechanism. This swivels around its pivot point M_2 and shifts the control collar to the shutoff position. The distributor plunger's cutoff port remains open and the plunger delivers no fuel.

Fig. 17

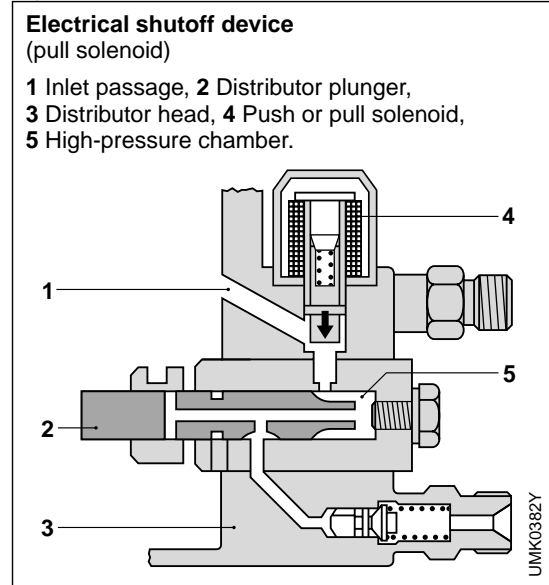
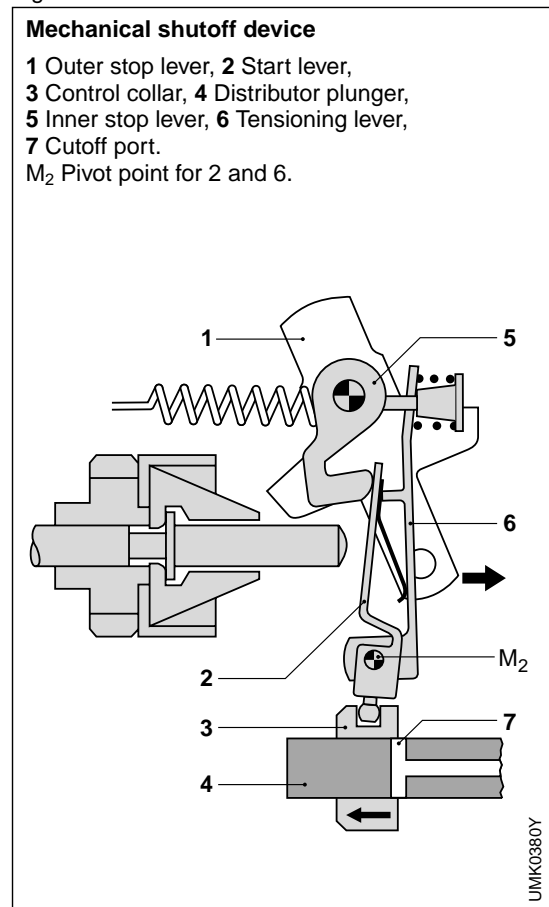


Fig. 18



Testing and calibration

Injection-pump test benches

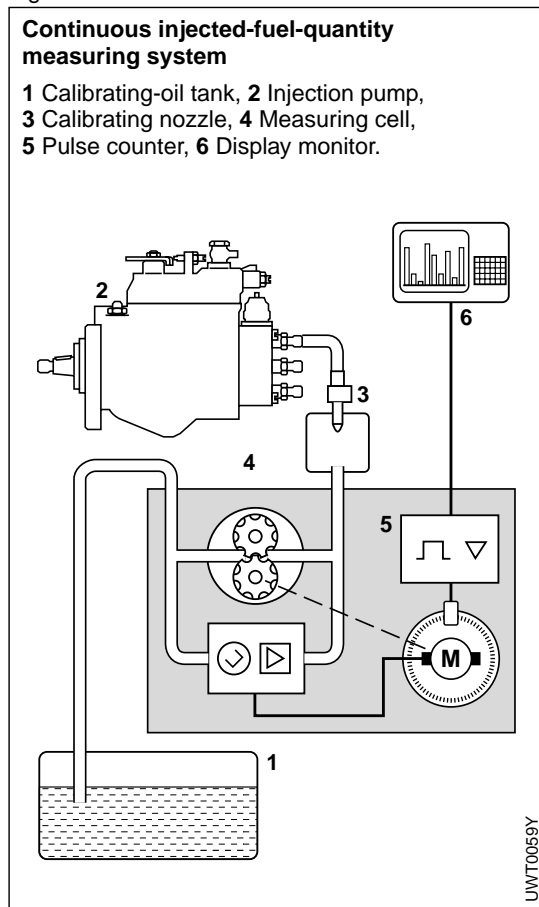
Precisely tested and calibrated injection pumps and governors are the prerequisite for achieving the optimum fuel-consumption/performance ratio and compliance with the increasingly stringent exhaust-gas legislation. And it is at this point that the injection-pump test bench becomes imperative. The most important framework conditions for the test bench and for the testing itself are defined in ISO-Standards which, in particular, place very high demands upon the rigidity and uniformity of the pump drive.

The injection pump under test is clamped to the test-bench bed and connected at its drive end to the test-bench coupling. Drive is through an electric motor (via hydrostatic or manually-switched transmission to flywheel and

coupling, or with direct frequency control). The pump is connected to the bench's calibrating-oil supply via oil inlet and outlet, and to its delivery measuring device via high-pressure lines. The measuring device comprises calibrating nozzles with precisely set opening pressures which inject into the bench's measuring system via spray dampers. Oil temperature and pressure is adjusted in accordance with test specifications. There are two methods for fuel-delivery measurement. One is the so-called continuous method. Here, a precision gear pump delivers per cylinder and unit of time, the same quantity of calibrating-oil as the quantity of injected fuel. The gear pump's delivery is therefore a measure of delivery quantity per unit of time. A computer then evaluates the measurement results and displays them as a bar chart on the screen. This measuring method is very accurate, and features good reproducibility (Fig. 1).

The other method for fuel-delivery measurement uses glass measuring graduates. The fuel to be measured is at first directed past the graduates and back to the tank with a slide. When the specified number of strokes has been set on the stroke-counting mechanism the measurement starts, and the slide opens and the graduates fill with oil. When the set number of strokes has been completed, the slider cuts off the flow of oil again. The injected quantity can be read off directly from the graduates.

Fig. 18



Engine tester for diesel engines

The diesel-engine tester is necessary for the precise timing of the injection pump to the engine. Without opening the high-pressure lines, this tester measures the start of pump delivery, injection timing, and engine speeds. A sensor is clamped over the high-pressure line to cylinder 1, and with the stroboscopic timing light or the TDC sensor for detecting crankshaft position, the tester calculates start of delivery and injection timing.

Nozzles and nozzle holders

The injection nozzles and their respective nozzle holders are vitally important components situated between the in-line injection pump and the diesel engine.

Their assignments are as follows:

- Metering the injection of fuel,
- Management of the fuel,
- Defining the rate-of-discharge curve,
- Sealing-off against the combustion chamber.

Considering the wide variety of combustion processes and the different forms of combustion chamber, it is necessary that the shape, “penetration force”, and atomization of the fuel spray injected by the nozzle are adapted to the prevailing conditions. This also applies to the injection time, and the injected fuel quantity per degree camshaft.

Since the design of the nozzle-holder combination makes maximum use of standardized components and assemblies, this means that the required flexibility can be achieved with a minimum of components. The following nozzles and nozzle holders are used with in-line injection pumps:

- Pintle nozzles (DN..) for indirect-injection (IDI) engines, and
- Hole-type nozzles (DLL../DLSA..) for direct-injection (DI) engines,
- Standard nozzle holders (single-spring nozzle holders), with and without needle-motion sensor, and
- Two-spring nozzle holders, with and without needle-motion sensor.

Pintle nozzles

Application

Pintle nozzles are used with in-line injection pumps on indirect-injection engines (pre-chamber and whirl-chamber engines).

In this type of diesel engine, the air/fuel mixture is for the most part formed by the air’s vortex work. The injected fuel spray serves to support this mixture-formation process.

The following types of pintle nozzle are available:

- Standard pintle nozzles (Fig. 1),
- Throttling pintle nozzles, and
- Flat-cut pintle nozzles (Fig. 2).

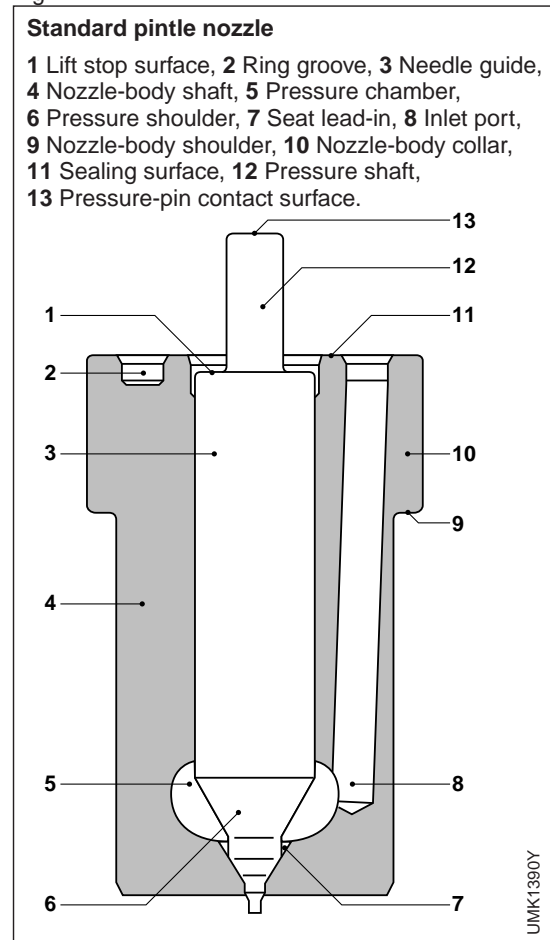
Design and construction

All pintle nozzles are of practically identical design, the only difference being in the pintle’s geometry:

Standard pintle nozzle

On the standard pintle nozzle, the nozzle needle is provided with a pintle which extends into the injection orifice of the nozzle body in which it is free to move with a minimum of play. The injection spray can be matched to the engine’s requirements by appropriate choice of dimensions and pintle designs.

Fig. 1



UMK1390Y

Throttling pintle nozzle

The throttling pintle nozzle is a pintle nozzle with special pintle dimensions. The special pintle design serves to define the shape of the rate-of-discharge curve. When the nozzle needle lifts it first of all opens a small annular gap so that only a small amount of fuel is injected (throttling effect).

As needle lift increases (due to pressure rise), the spray orifice is opened increasingly until the major portion of the injection (main injection) takes place towards the end of needle lift. Since the pressure in the combustion chamber rises less sharply, this shaping of the rate-of-injection curve leads to “softer” combustion. This results in quieter combustion in the part-load range. In other words, it is possible to shape the required rate-of-discharge curve by means of the pintle shape, the characteristic of the nozzle needle’s spring, and the throttling gap.

Flat-cut pintle nozzle

This nozzle’s pintle has a ground surface which opens a flow cross-section in addition to the annular gap when the pintle opens (only slight needle lift). The resulting increased flow volume prevents deposits forming in this flow channel. This is the reason why flat-cut pintle nozzles coke-up far less, and any coking which does take place is more uniform. The annular gap between spray orifice and throttling pintle is very small (less than $10\ \mu\text{m}$). Very often, the flat-cut pintle surface is parallel to the nozzle-needle axis. Referring to Fig. 3, with an additional inclined cut on the pintle, the gradient of the injected-fuel-quantity curve’s flat portion can be increased so that the transition to full nozzle opening is less abrupt. Specially shaped pintles, such as the “radius” or “profile surface” types, can be applied to match the flow curve to engine-specific requirements. Part-load noise and vehicle driveability are both improved as a result.

Fig. 2

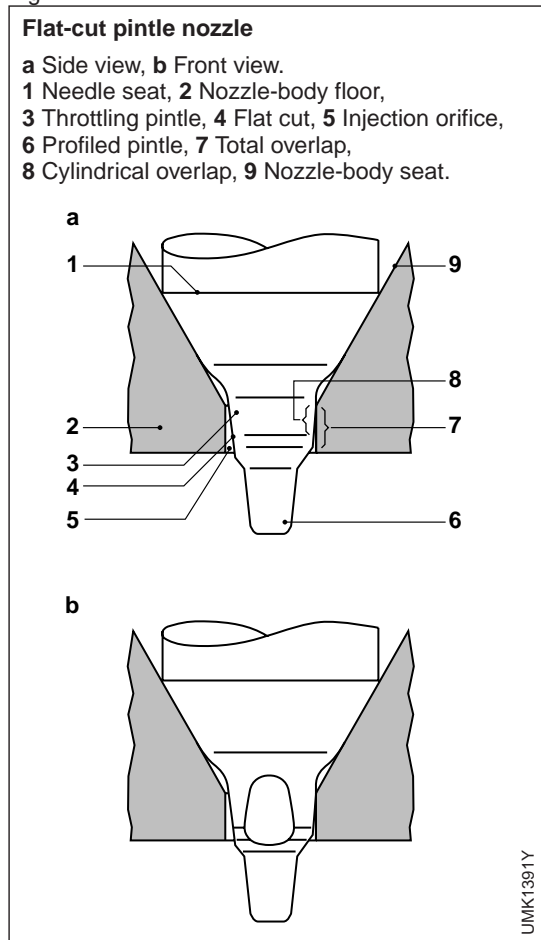
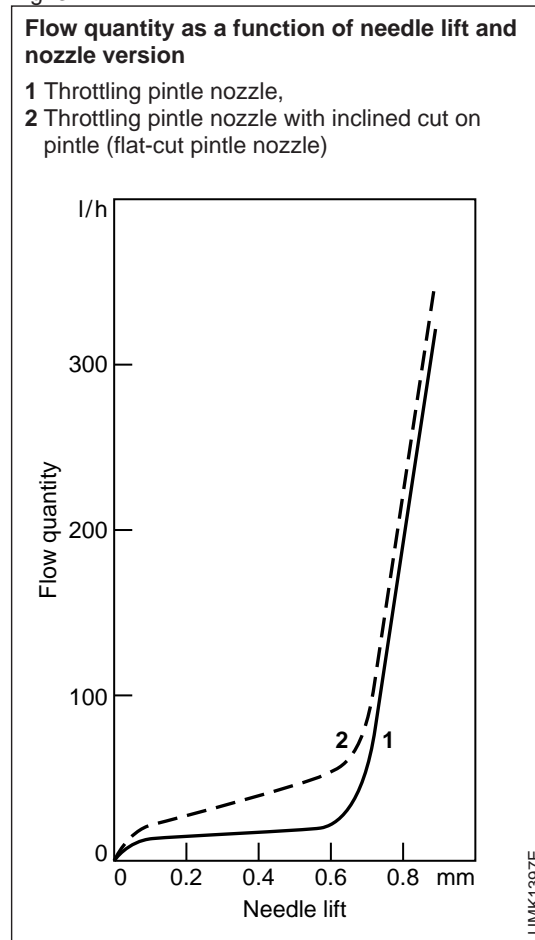


Fig. 3



Hole-type nozzles

Application

Hole-type nozzles are used with in-line injection pumps on direct-injection engines.

One differentiates between:

- Sac-hole, and
- Seat-hole nozzles.

The hole-type nozzles also vary according to their size:

- Type P with 4 mm needle diameter, and
- Type S with 5 and 6 mm needle diameters.

Design and construction

The spray holes are located on the envelope of a spray cone (Fig. 4). The number of spray holes and their diameter depend upon:

- The injected fuel quantity,
- The combustion-chamber shape, and
- The air swirl in the combustion chamber.

The input edges of the spray holes can be rounded by hydro-erosive (HE) machining.

At those points where high flow rates occur (spray-hole entrance), the abrasive particles in the hydro-erosive (HE) medium cause material loss.

This so-called HE-rounding process can be applied to both sac-hole and seat-hole nozzles, whereby the target is:

- Prevent in advance the edge wear caused by abrasive particles in the fuel and/or
- Reduce the flow tolerance.

For low hydrocarbon emissions, it is highly important that the volume filled with fuel (residual volume) below the edge of the nozzle-needle seat is kept to a minimum. Seat-hole nozzles are therefore used.

Designs

Sac-hole nozzle

The spray holes of the sac-hole nozzle (Fig. 5) are arranged in the sac hole.

In the case of a round nozzle tip (Fig. 6a), depending upon design the spray holes are drilled mechanically or by means of electrochemical machining (e.c.m.).

Sac-hole nozzles with conical tip (Figs. 6b and 6c) are always drilled using e.c.m. Sac-hole nozzles are available

- With cylindrical, and
 - Conical sac holes
- in a variety of different dimensions.

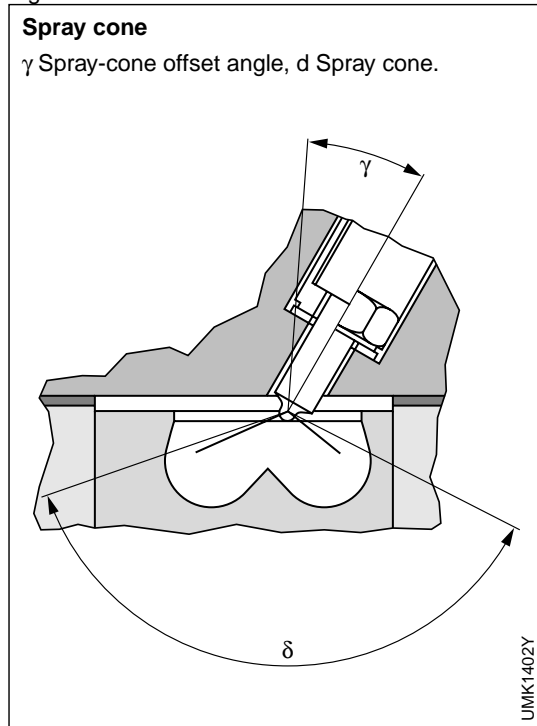
Sac-hole nozzle with cylindrical sac hole and round tip (Fig. 6a):

This nozzle's sac hole has a cylindrical and a semispherical portion, and permits a high level of design freedom with respect to

- Number of spray holes,
- Spray-hole length, and
- Injection angle.

The nozzle tip is semispherical, and together with the shape of the sac hole, ensures that the spray holes are of identical length.

Fig. 4



Sac-hole nozzle with cylindrical sac hole and conical tip (6b):

This type of nozzle is used exclusively with spray-hole lengths of 0.6 mm. The tip's conical shape enables the wall thickness to be increased between the throat radius and the nozzle-body seat with an attending improvement of nozzle-tip strength.

Sac-hole nozzle with conical sac hole and conical tip (Fig. 6c):

Due to the conical shape of this nozzle's sac hole, its volume is less than that of a nozzle with cylindrical sac hole. The volume is between that for a seat-hole nozzle and a sac-hole nozzle with cylindrical sac hole. In order to achieve uniform tip-wall thickness, the tip's conical design corresponds to that of the sac hole.

Fig. 5

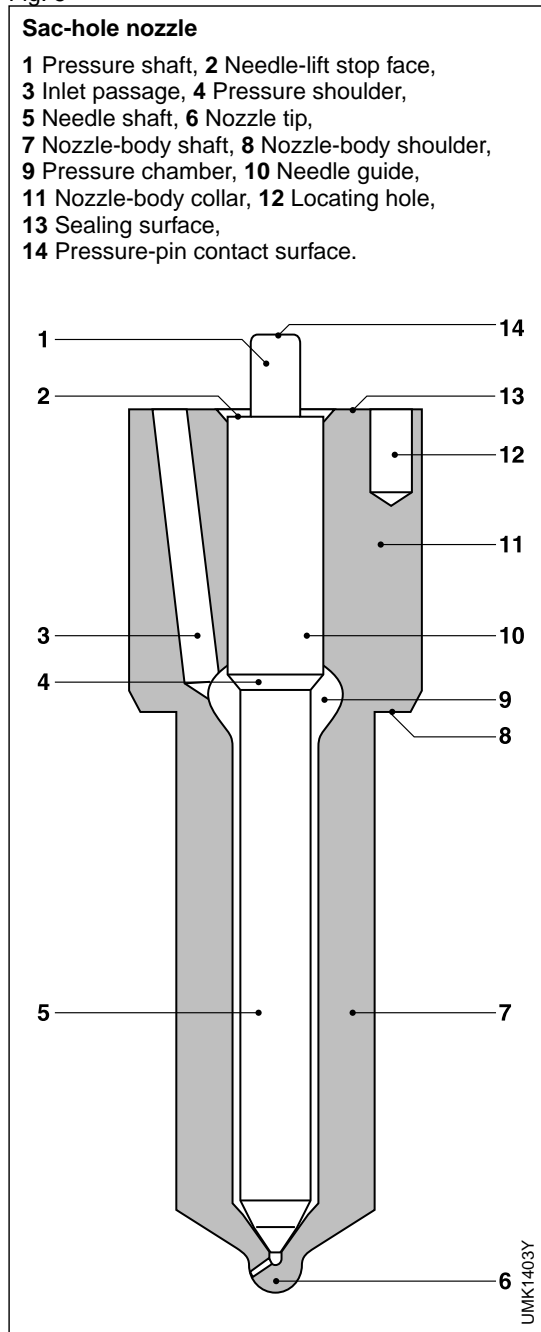
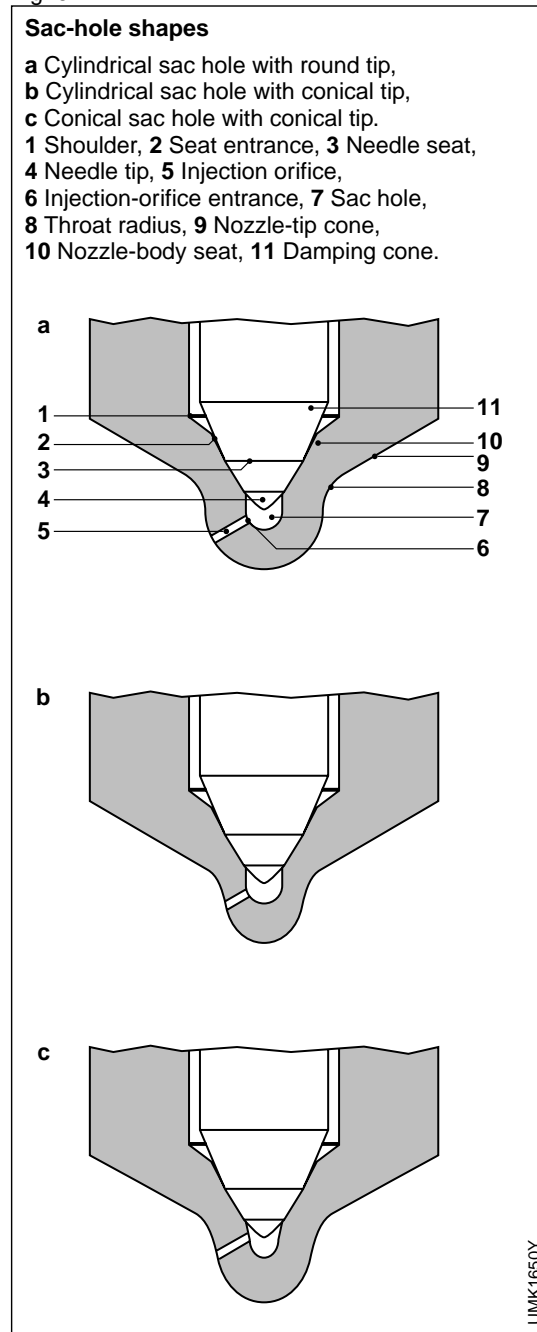


Fig. 6



Seat-hole nozzle

In order to minimise the residual volume – and therefore the HC emissions – the start of the spray hole is located in the seat taper, and with the nozzle closed it is covered almost completely by the nozzle needle. This means that there is no direct connection between the sac hole and the combustion chamber (Figs. 7 and 8). The sac-hole volume here is much lower than that of the sac-hole nozzle. Compared to sac-hole nozzles, seat-hole nozzles have a much lower loading limit and are therefore only manufactured as Size P with a spray-hole length of 1 mm.

For reasons of strength, the nozzle tip is conically shaped. The spray holes are always formed using e.c.m. methods.

Fig. 7

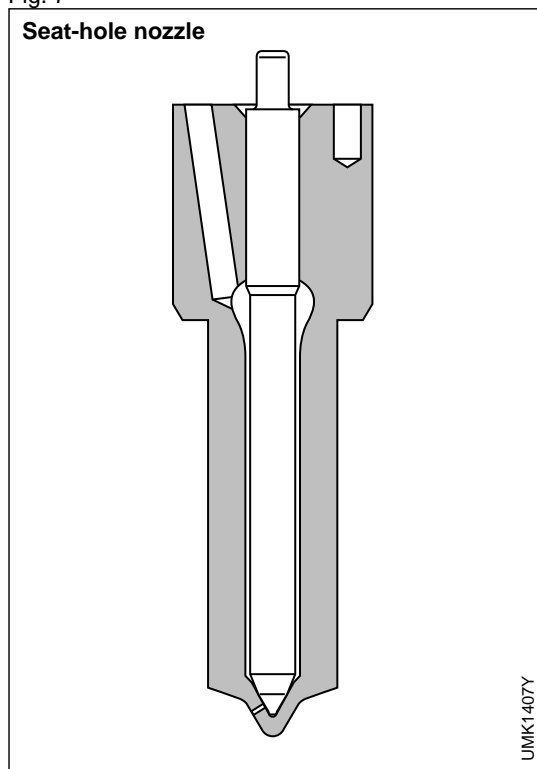
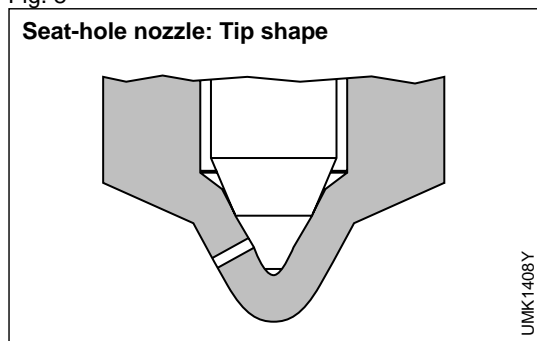


Fig. 8



Standard nozzle holders

Assignments and designs

Nozzle holders with hole-type nozzles in combination with a radial-piston distributor injection pump are used on DI engines.

With regard to the nozzle holders, one differentiates between

- Standard nozzle holders (single-spring nozzle holders) with and without needle-motion sensor, and
- Two-spring nozzle holders, with and without needle-motion sensor.

Application

The nozzle holders described here have the following characteristics:

- Cylindrical external shape with diameters between 17 and 21 mm,
- Bottom-mounted springs (leads to low moving masses),
- Pin-located nozzles for direct-injection engines, and
- Standardised components (springs, pressure pin, nozzle-retaining nut) make combinations an easy matter.

Design

The nozzle-and-holder assembly is composed of the injection nozzle and the nozzle holder.

The nozzle holder comprises the following components (Fig. 9):

- Nozzle-holder body,
- Intermediate element,
- Nozzle-retaining nut,
- Pressure pin,
- Spring,
- Shim, and
- Locating pins.

The nozzle is centered in the nozzle body and fastened using the nozzle-retaining nut. When nozzle body and retaining nut are screwed together, the intermediate element is forced up against the sealing surfaces of nozzle body and retaining nut. The intermediate element serves as the needle-lift stop and with its locating pins centers the nozzle in the nozzle-holder body.

The nozzle-holder body contains the

- Pressure pin,
- Spring, and
- Shim.

The spring is centered in position by the pressure pin, whereby the pressure pin is guided by the nozzle-needle's pressure shaft.

The nozzle is connected to the injection pump's high-pressure line via the nozzle-holder feed passage, the intermediate element, and the nozzle-body feed passage. If required, an edge-type filter can be installed in the nozzle holder.

Method of operation

The nozzle-holder spring applies pressure to the nozzle needle through the pressure pin. The spring's initial tension defines the nozzle's opening pressure which can be adjusted using a shim.

On its way to the nozzle seat, the fuel passes through the nozzle-holder inlet passage, the intermediate element, and the nozzle body. When injection takes place, the nozzle needle is lifted by the injection pressure and fuel is injected through the injection orifices into the combustion chamber. Injection terminates as soon as the injection pressure drops far enough for the nozzle spring to force the nozzle needle back onto its seat.

Two-spring nozzle holders

Application

The two-spring nozzle holder is a further development of the standard nozzle holder, and serves to reduce combustion noise particularly in the idle and part-load ranges.

Design

The two-spring nozzle holder features two springs located one behind the other. At first, only one of these springs has an influence on the nozzle needle and as such defines the initial opening pressure. The second spring is in contact with a stop sleeve which limits the needle's initial stroke.

Fig. 9

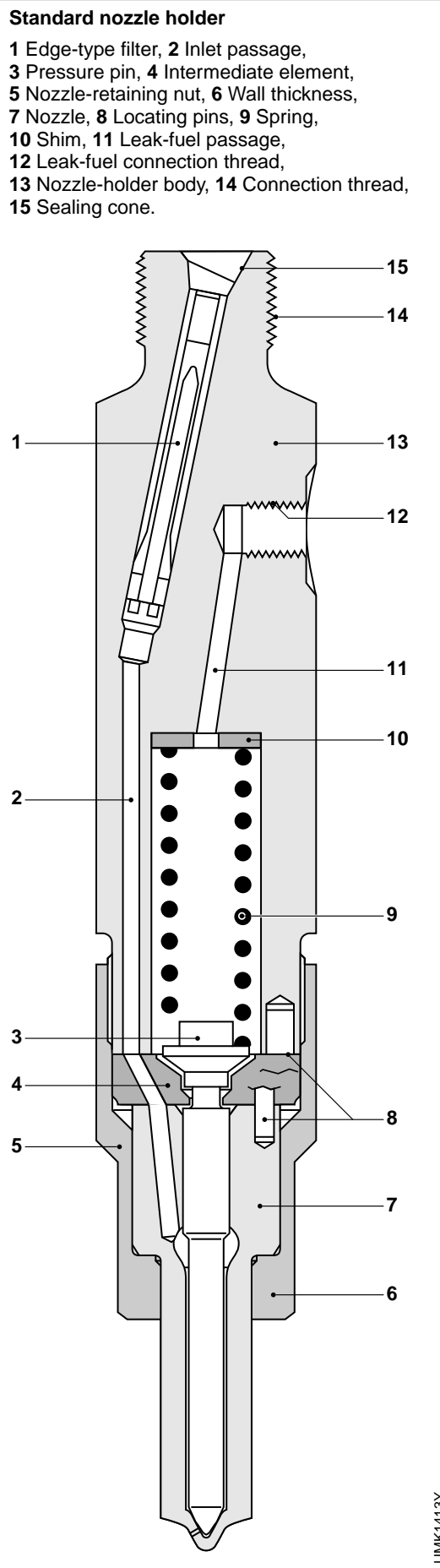
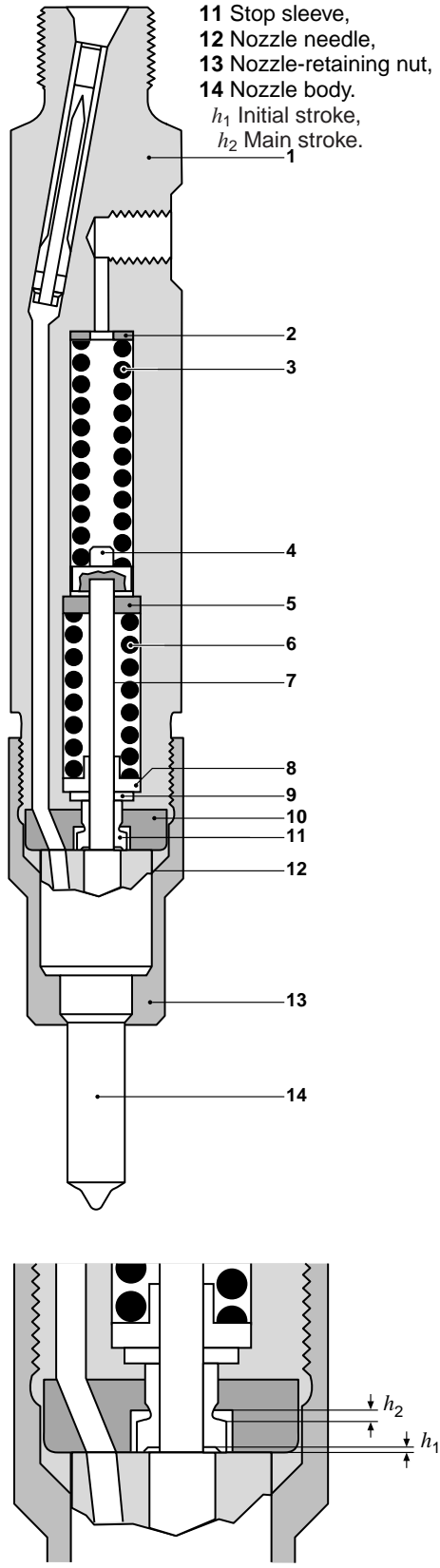


Fig. 10

Two-spring nozzle holder for direct-injection (DI) engines

- 1 Nozzle-holder body, 2 Shim,
 - 3 Spring 1, 4 Pressure pin,
 - 5 Guide element, 6 Spring 2,
 - 7 Pressure pin, 8 Spring seat,
 - 9 Shim, 10 Intermediate element,
 - 11 Stop sleeve,
 - 12 Nozzle needle,
 - 13 Nozzle-retaining nut,
 - 14 Nozzle body.
- h_1 Initial stroke,
 h_2 Main stroke.



When strokes take place in excess of the initial stroke, the stop sleeve lifts and both springs have an effect upon the nozzle needle (Fig. 10).

Method of operation

During the actual injection process, the nozzle needle first of all opens an initial amount so that only a small volume of fuel is injected into the combustion chamber.

Along with increasing injection pressure in the nozzle holder though, the nozzle needle opens completely and the main quantity is injected (Fig. 11). This 2-stage rate-of-discharge curve leads to "softer" combustion and to a reduction in noise.

Nozzle holders with needle-motion sensor

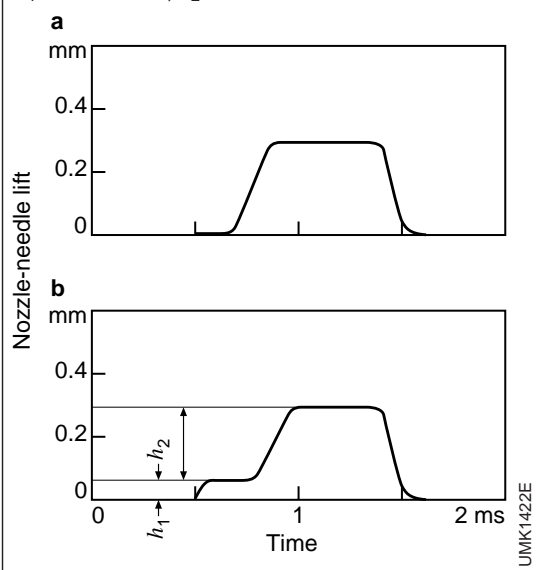
Application

The start-of-injection point is an important parameter for optimum diesel-engine operation. For instance, its evaluation permits load and speed-dependent injection timing, and/or control of the exhaust-gas recirculation (EGR) rate.

Fig. 11

Comparison of needle-lift curves

- a Standard nozzle holder (single-spring nozzle holder),
 - b Two-spring nozzle holder.
- h_1 Initial stroke, h_2 Main stroke.



This necessitates a nozzle holder with needle-motion sensor (Fig. 13) which outputs a signal as soon as the nozzle needle opens.

Design

When it moves, the extended pressure pin enters the current coil. The degree to which it enters the coil (overlap length "X" in Fig. 14) determines the strength of the magnetic flux.

Method of operation

The magnetic flux in the coil changes as a result of nozzle-needle movement and induces a signal voltage which is proportional to the needle's speed of movement but not to the distance it has travelled. This signal is processed directly in an evaluation circuit (Fig. 12). When a given threshold voltage is exceeded, this serves as the signal to the evaluation circuit for the start of injection.

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

- 1 Nozzle-holder body, 2 Needle-motion sensor,
- 3 Spring 1, 4 Guide element, 5 Spring 2,
- 6 Pressure pin, 7 Nozzle-retaining nut.

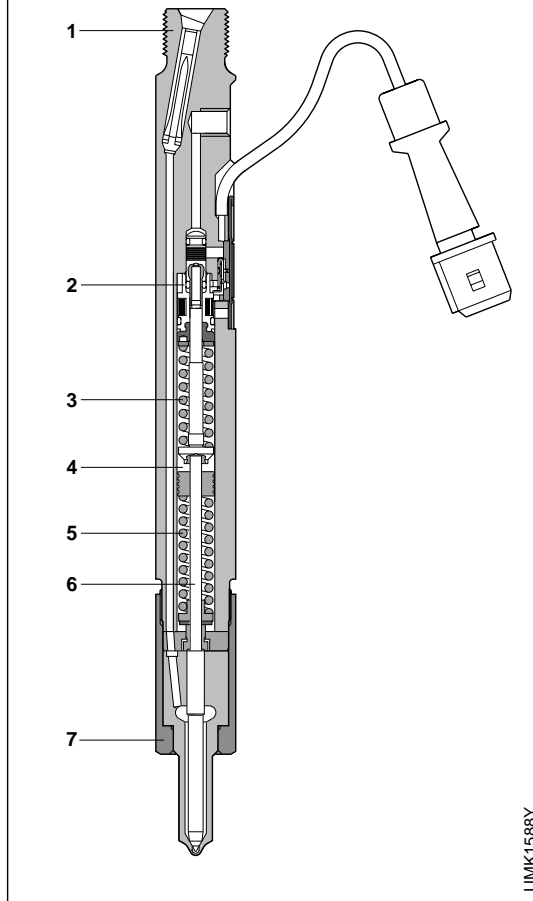


Fig. 13

Fig. 12

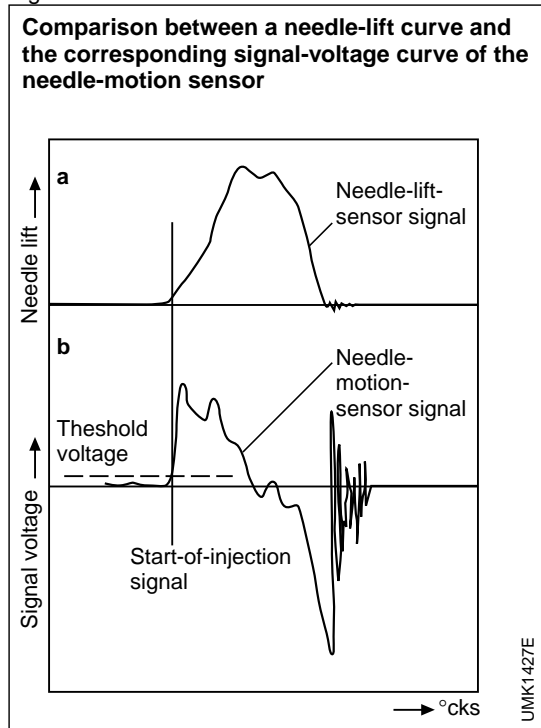
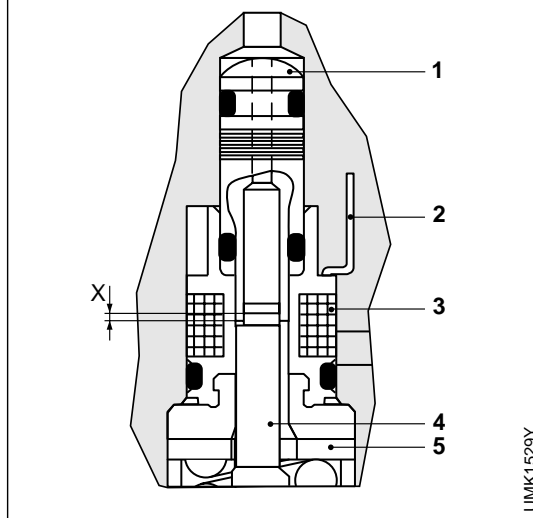


Fig. 14

Needle-motion sensor in a two-spring nozzle holder for direct-injection (DI) engines

- 1 Adjusting pin, 2 Terminal,
 - 3 Current coil, 4 Pressure pin,
 - 5 Spring seat.
- X Overlap length.



Electronically-controlled axial-piston distributor fuel-injection pumps VE-EDC

Mechanical diesel-engine speed control (mechanical governing) registers a wide variety of different operating statuses and permits high-quality A/F mixture formation.

The Electronic Diesel Control (EDC) takes additional requirements into account. By applying electronic measurement, highly-flexible electronic data processing, and closed control loops with electric actuators, it is able to process mechanical influencing variables which it was impossible to take into account with the previous purely mechanical control (governing) system.

The EDC permits data to be exchanged with other electronic systems in the

vehicle (for instance, traction control system (TCS), and electronic transmission-shift control). In other words, it can be integrated completely into the overall vehicle system.

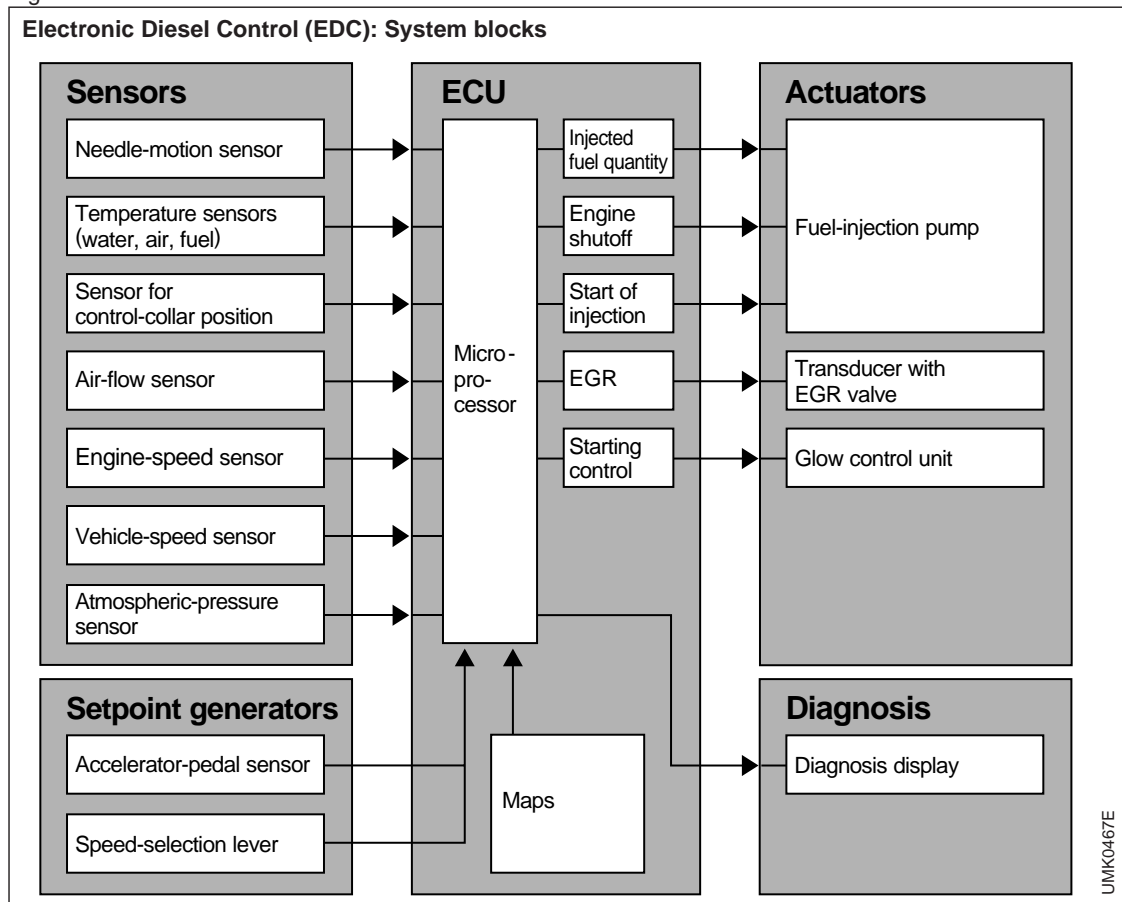
System blocks

The electronic control is divided into three system blocks (Fig. 1):

1. Sensors for registering operating conditions. A wide variety of physical quantities are converted into electrical signals.

2. Electronic control unit (ECU) with microprocessors which processes the in-

Fig. 1



formation in accordance with specific control algorithms, and outputs corresponding electrical signals.

3. Actuators which convert the ECU's electrical output signals into mechanical quantities.

Components

Sensors

The positions of the accelerator and the control collar in the injection pump are registered by the angle sensors. These use contacting and non-contacting methods respectively. Engine speed and TDC are registered by inductive sensors. Sensors with high measuring accuracy and long-term stability are used for pressure and temperature measurements. The start of injection is registered by a sensor which is directly integrated in the nozzle holder and which detects the start of injection by sensing the needle movement (Figs. 2 and 3).

Electronic control unit (ECU)

The ECU employs digital technology. The microprocessors with their input and output interface circuits form the heart of the ECU. The circuitry is completed by the memory units and devices for the conversion of the sensor signals into computer-compatible quantities. The ECU is installed in the passenger compartment to protect it from external influences.

There are a number of different maps stored in the ECU, and these come into effect as a function of such parameters as: Load, engine speed, coolant temperature, air quantity etc. Exacting demands are made upon interference immunity. Inputs and outputs are short-circuit-proof and protected against spurious pulses from the vehicle electrical system. Protective circuitry and mechanical shielding provide a high level of EMC (Electro-Magnetic Compatibility) against outside interference.

Electronic control for distributor pumps

Fig. 2

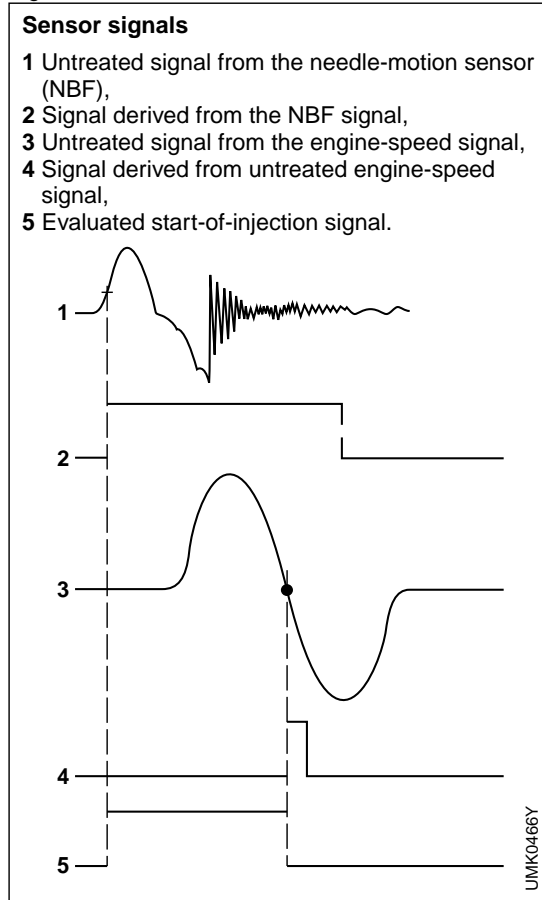
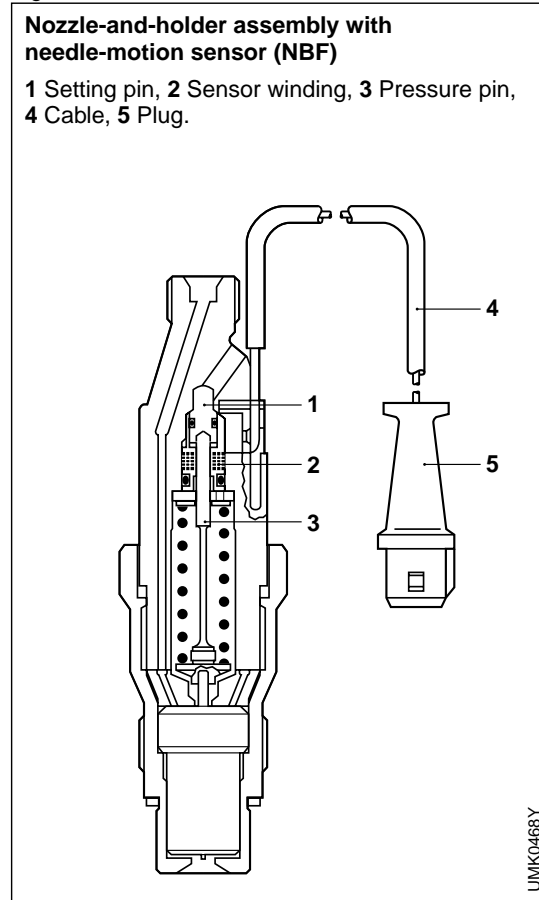


Fig. 3



Solenoid actuator for injected-fuel quantity control

The solenoid actuator (rotary actuator) engages with the control collar through a shaft (Fig. 4). Similar to the mechanically governed fuel-injection pump, the cutoff ports are opened or closed depending upon the control collar's position. The injected fuel quantity can be infinitely varied between zero and maximum (e.g., for cold starting). Using an angle sensor (e.g., potentiometer), the rotary actuator's angle of rotation, and thus the position of the control collar, are reported back to the ECU and used to determine the injected fuel quantity as a function of engine speed. When no voltage is applied to the actuator, its return springs reduce the injected fuel quantity to zero.

Solenoid valve for start-of-injection control

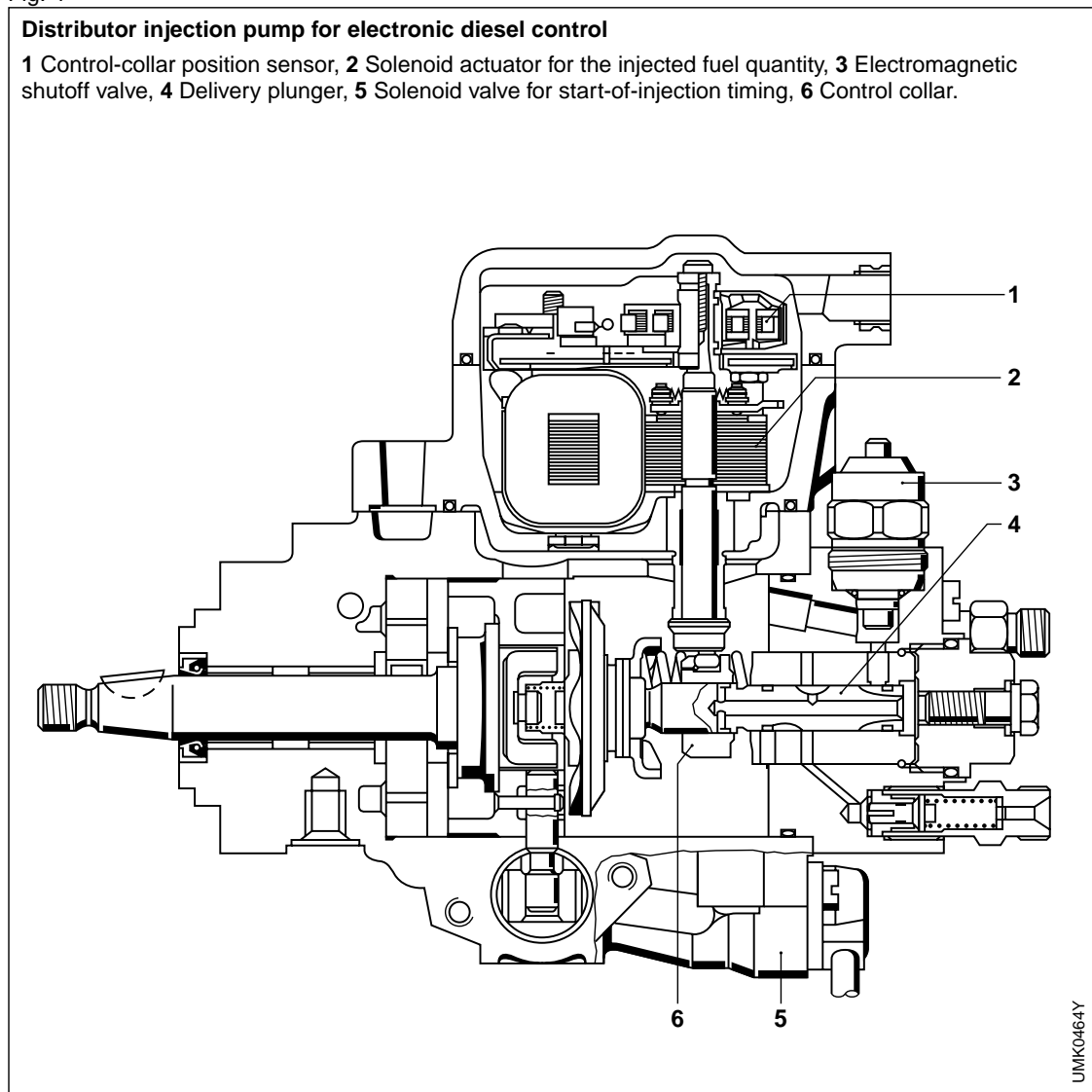
The pump interior pressure is dependent upon pump speed. Similar to the mechanical timing device, this pressure is applied to the timing-device piston (Fig. 4). This pressure on the timing-device pressure side is modulated by a clocked solenoid valve.

With the solenoid valve permanently opened (pressure reduction), start of injection is retarded, and with it fully closed (pressure increase), start of injection is advanced. In the intermediate range, the on/off ratio (the ratio of solenoid valve open to solenoid valve closed) can be infinitely varied by the ECU.

Fig. 4

Distributor injection pump for electronic diesel control

1 Control-collar position sensor, 2 Solenoid actuator for the injected fuel quantity, 3 Electromagnetic shutoff valve, 4 Delivery plunger, 5 Solenoid valve for start-of-injection timing, 6 Control collar.



Closed control loops (Fig. 5)

Injected fuel quantity

The injected fuel quantity has a decisive influence upon the vehicle's starting, idling, power output and driveability characteristics, as well as upon its particulate emissions. For this reason, the corresponding maps for start quantity, idle, full load, accelerator-pedal characteristic, smoke limitation, and pump characteristic, are programmed into the ECU. The driver inputs his or her requirements regarding torque or engine speed through the accelerator sensor. Taking into account the stored map data, and the actual input values from the sensors, a setpoint is calculated for the setting of the rotary actuator in the pump. This rotary actuator is equipped

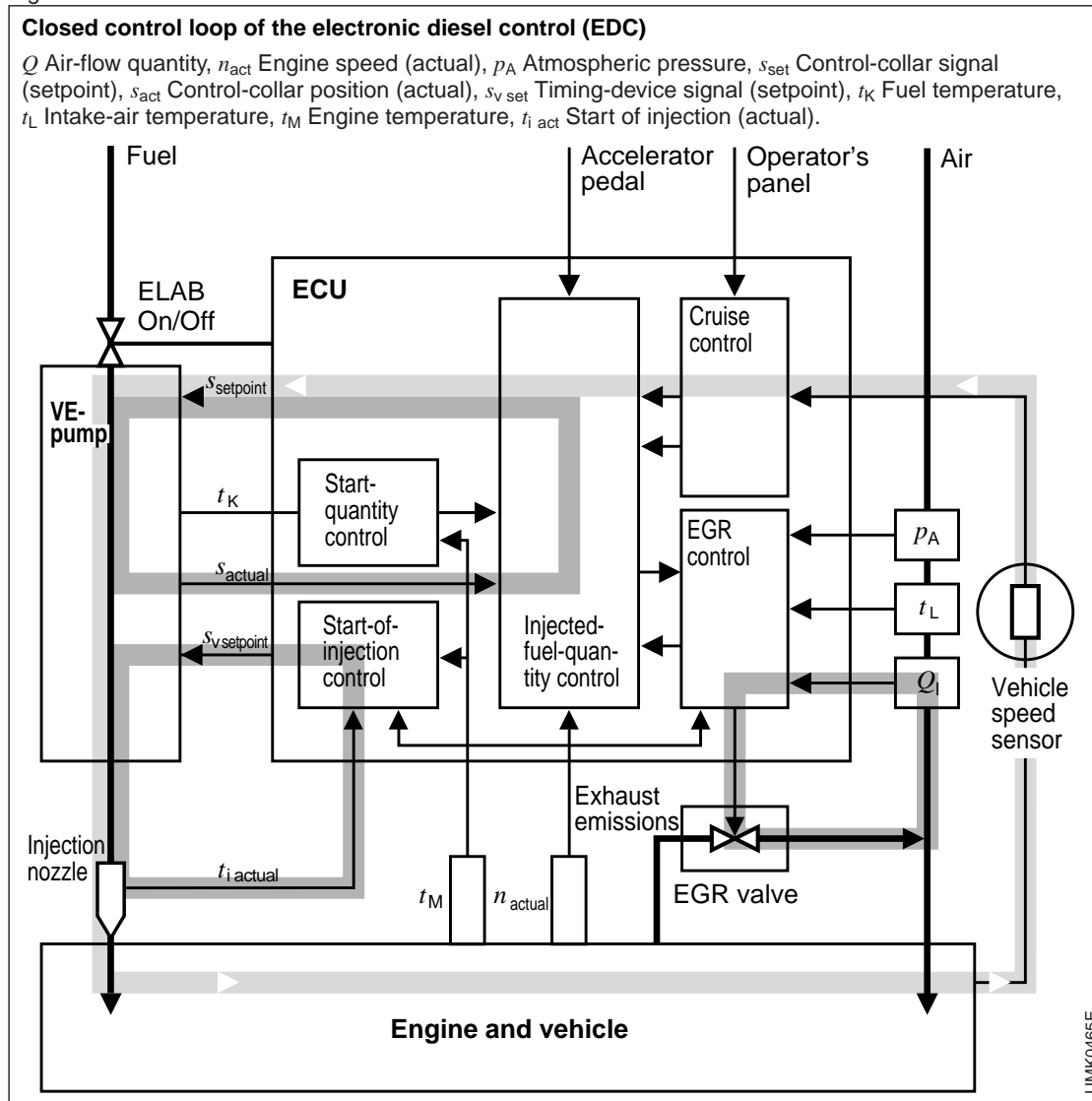
with a check-back signalling unit and ensures that the control collar is correctly set.

Start of injection

The start of injection has a decisive influence upon starting, noise, fuel consumption, and exhaust emissions. Start-of-injection maps programmed into the ECU take these interdependencies into account. A closed control loop is used to guarantee the high accuracy of the start-of-injection point. A needle-motion sensor (NBF) registers the actual start of injection directly at the nozzle and compares it with the programmed start of injection (Figs. 2 and 3). Deviations result in a change to the on/off ratio of the timing-device solenoid valve, which continues until deviation reaches zero.

Electronic control for distributor pumps

Fig. 5



This clocked solenoid valve is used to modulate the positioning pressure at the timing-device piston, and this results in the dynamic behavior being comparable to that obtained with the mechanical start-of-injection timing.

Because during engine overrun (with injection suppressed) and engine starting there are either no start-of-injection signals available, or they are inadequate, the controller is switched off and an open-loop-control mode is selected. The on/off ratio for controlling the solenoid valve is then taken from a control map in the ECU.

Exhaust-gas recirculation (EGR)

EGR is applied to reduce the engine's toxic emissions. A defined portion of the exhaust gas is tapped-off and mixed with the fresh intake air. The engine's intake-air quantity (which is proportional to the EGR rate) is measured by an air-flow sensor and compared in the ECU with the programmed value for the EGR map, whereby additional engine and injection data for every operating point are taken into account.

In case of deviation, the ECU modifies the triggering signal applied to an electropneumatic transducer. This then adjusts the EGR valve to the correct EGR rate.

Cruise control

An evaluated vehicle-speed signal is compared with the setpoint signal inputted by the driver at the cruise-control panel. The injected fuel quantity is then adjusted to maintain the speed selected by the driver.

Supplementary functions

The electronic diesel control (EDC) provides for supplementary functions which considerably improve the vehicle's driveability compared to the mechanically governed injection pump.

Active anti-buck damping

With the active anti-buck damping (ARD) facility, the vehicle's unpleasant longitudinal oscillations can be avoided.

Idle-speed control

The idle-speed control avoids engine "shake" at idle by metering the appropriate amount of fuel to each individual cylinder.

Safety measures

Self-monitoring

The safety concept comprises the ECU's monitoring of sensors, actuators, and microprocessors, as well as of the limp-home and emergency functions provided in case a component fails. If malfunctions occur on important components, the diagnostic system not only warns the driver by means of a lamp in the instrument panel but also provides a facility for detailed trouble-shooting in the workshop.

Limp-home and emergency functions

There are a large number of sophisticated limp-home and emergency functions integrated in the system. For instance if the engine-speed sensor fails, a substitute engine-speed signal is generated using the interval between the start-of-injection signals from the needle-motion sensor (NBF). And if the injected-fuel quantity actuator fails, a separate electrical shutoff device (ELAB) switches off the engine. The warning lamp only lights up if important sensors fail. The Table below shows the ECU's reaction should certain faults occur.

Diagnostic output

A diagnostic output can be made by means of diagnostic equipment, which can be used on all Bosch electronic automotive systems. By applying a special test sequence, it is possible to systematically check all the sensors and their connectors, as well as the correct functioning of the ECU's.

Table 1. ECU reactions

Failure	Monitoring of	Reaction	Warning lamp	Diagnostic output
Correction sensors	Signal range	Reduce injected fuel quantity		●
System-sensors	Signal range	Limp-home or emergency function (graded)	●	●
Computer	Program runtime (self-test)	Limp-home or emergency function	●	●
Fuel-quantity actuator	Permanent deviation	Engine shutoff	●	●

Advantages

- Flexible adaptation enables optimization of engine behavior and emission control.
- Clear-cut delineation of individual functions: The curve of full-load injected fuel quantity is independent of governor characteristic and hydraulic configuration.
- Processing of parameters which previously could not be performed mechanically (e.g., temperature-correction of the injected fuel quantity characteristic, load-independent idle control).
- High degree of accuracy throughout complete service life due to closed control loops which reduce the effects of tolerances.
- Improved driveability: Map storage enables ideal control characteristics and control parameters to be established independent of hydraulic effects. These are then precisely adjusted during the optimisation of the complete engine/vehicle system. Bucking and idle shake no longer occur.
- Interlinking with other electronic systems in the vehicle leads the way towards making the vehicle safer, more comfortable, and more economical, as well as increasing its level of environmental compatibility (e.g., glow systems or electronic transmission-shift control). The fact that mechanical add-on units no longer need to be accommodated, leads to marked reductions in the amount of space required for the fuel-injection pump.

Engine shutoff

As already stated on Page 40, the principle of auto-ignition as applied to the diesel engine means that the engine can only be switched off by interrupting its supply of fuel.

When equipped with Electronic Diesel Control (EDC), the engine is switched off by the injected-fuel quantity actuator (Input from the ECU: Injected fuel quantity = Zero). As already dealt with, the separate electrical engine shutoff device serves as a standby shutoff in case the actuator should fail.

Electrical shutoff device

The electrical shutoff device is operated with the “ignition key” and is above all used to provide the driver with a higher level of sophistication and comfort.

On the distributor fuel-injection pump, the solenoid valve for interrupting the supply of fuel is fitted in the top of the distributor head. With the diesel engine running, the inlet opening to the high-pressure chamber is held open by the energized solenoid valve (the armature with sealing cone is pulled in). When the “ignition switch” is turned to “Off”, the power supply to the solenoid is interrupted and the solenoid de-energized. The spring can now push the armature with sealing cone onto the valve seat and close off the inlet opening to the high-pressure chamber so that the distributor plunger can no longer deliver fuel.

Solenoid-valve-controlled axial-piston distributor fuel-injection pumps VE-MV

Prospects

On the electronically-controlled distributor pumps of the future, the electrical actuator mechanism with control collar for fuel metering will be superseded by a high-pressure solenoid valve. This will permit an even higher degree of flexibility in the fuel metering and in the variability of the start of injection.

Design and construction

This pump is of modular design. The field-proven distributor injection pump can thus be combined with a new electronically controlled fuel-metering system (Fig. 1). Basically speaking, the solenoid-valve-controlled distributor pump's dimensions, installation conditions, and drivetrain including the pump's cam drive, are identical to those of the conventional distributor pump. The most important new components are:

- Angle-of-rotation sensor (in the form of an incremental angle/time system [IWZ]) which is located in the injection pump on the driveshaft between the vane-type supply pump and the roller ring,
- Electronic pump ECU, which is mounted as a compact unit on the top side of the pump and connected to the engine ECU,
- High-pressure solenoid valve, installed in the center of the distributor head.

With regard to its installation and hydraulic control, the timing device with pulse valve is identical to the one in the previous electronically-controlled distributor pump.

Components

Angle-of-rotation sensor

Angle-of-rotation detection uses the following components: Sensor, sensor retaining ring on the driveshaft, and the trigger wheel with a given tooth pitch. Detection is based upon the signals generated by the sensor.

The pulses generated by the sensor are inputted to the ECU where they are processed by an evaluation circuit. The fact that the sensor is coupled to the pump's roller ring ensures the correct assignment of the angular increment to the position of the cam when the roller ring is rotated by the timing device.

Pump ECU

The pump ECU is mounted on the upper side of the pump and uses hybrid techniques. In addition to the mechanical loading with which it is confronted in the vehicle's under-hood environment, the pump must also fulfill the following assignments:

- Data exchange with the separately mounted engine ECU via the serial bus system,
- Evaluation of the signal from the angle-of-rotation sensor (IWZ),
- Triggering of the high-pressure solenoid valve,
- Triggering of the timing device.

Maps are stored in the pump ECU which not only take into account the setting points for the particular vehicle application and certain engine characteristics, but also permit the plausibility of the received signals to be checked. In addition, they form the basis for defining a number of different computational values.

High-pressure solenoid valve

The high-pressure solenoid valve must fulfill the following assignments:

- Large valve cross-section for efficient filling of the high-pressure chamber, even at very high rotational speeds,
- Low weight (low moving masses), to keep the loading of the parts to a minimum,
- Short switching times to guarantee high-precision fuel metering, and
- Magnetic forces which are powerful enough to cope with the high pressures.

The high-pressure solenoid valve is comprised of:

- The valve body,
- The valve needle, and
- The electromagnet with electrical connection to the pump ECU.

The magnetic circuit is concentric to the valve. This fact permits a compact assembly comprising high-pressure solenoid valve and distributor head.

Method of operation

Principle

Pressure generation in the solenoid-valve-controlled distributor injection pump is based on the same principle as that in the conventional electronically-controlled VE pump.

Fuel supply and delivery

Via the distributor head and the opened high-pressure solenoid valve, the vane-type supply pump delivers fuel to the high-pressure chamber at a pressure of approx. 12 bar.

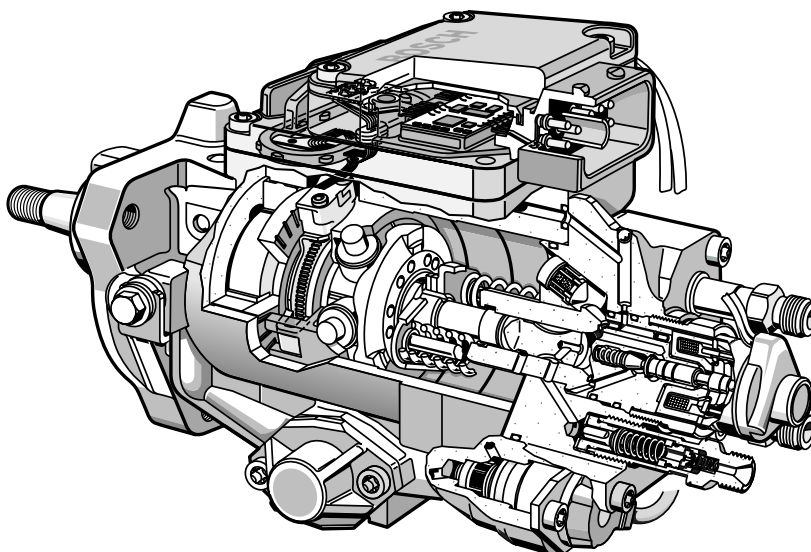
No fuel is delivered when the high-pressure solenoid valve is de-energized (open). The valve's instant of closing defines the injection pump's start of delivery. This can be located at the bottom dead center (BDC) of the cam or on the rise portion of the cam slope. Similarly, the valve's instant of opening defines the pump's end of delivery. The length of time the valve is closed determines the injected fuel quantity.

The high pressure generated in the high-pressure chamber (the fuel from the supply pump is compressed by the axial piston when this is forced up by the cam plate riding over the rollers of the roller ring) opens the delivery valve and the fuel is forced through the pressure line to the injection nozzle in the nozzle holder. Injection pressure at the nozzle is 1400 bar. Excess fuel is directed back to the tank through return lines.

Since there are no additional intake ports available, if the high-pressure solenoid valve should fail, fuel injection stops. This prevents uncontrolled "racing" of the engine.

Fig. 1

Solenoid-valve-controlled axial-piston distributor fuel-injection pump (section)



UMIK1205Y

Start-assist systems

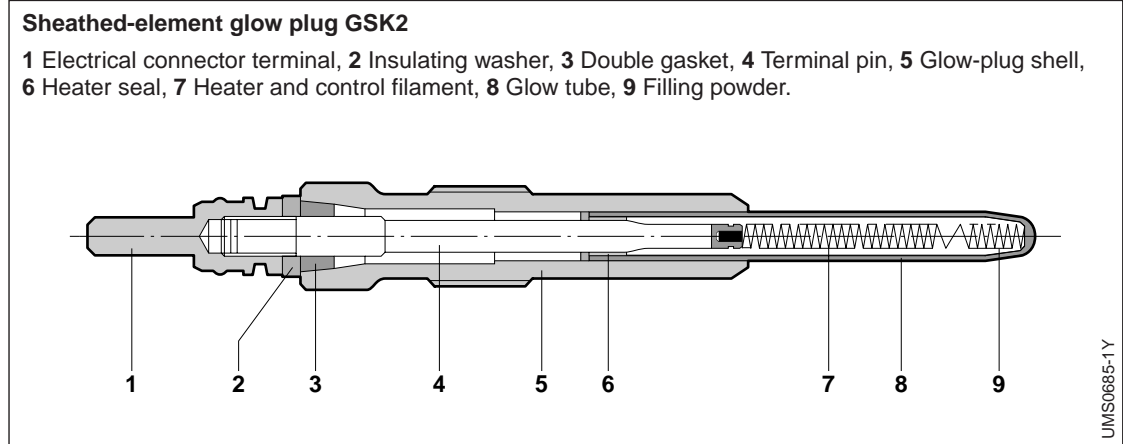
Since leakage and heat losses reduce the pressure and the temperature of the A/F mixture at the end of the compression stroke, the cold diesel engine is more difficult to start and the mixture more difficult to ignite than it is when hot. These facts make it particularly important that start-assist systems are used. The minimum starting temperature depends upon the engine type. Pre-chamber and swirl-chamber engines are equipped with a sheathed-element glow plug (GSK) in the auxiliary combustion chamber which functions as a "hot spot". On small direct-injection (DI) engines, this "hot spot" is located on the combustion chamber's periphery. Large DI truck engines on the other hand have the alternative of using air preheating in the intake manifold (flame start) or special, easily ignitable fuel (Start Pilot) which is sprayed into the intake air. Today, the start-assist systems use sheathed-element glow plugs practically without exception.

Sheathed-element glow plug

The sheathed-element glow plug's tubular heating element is so firmly pressed into the glow-plug shell that a gas-tight seal is

formed. The element is a metal tube which is resistant to both corrosion and hot gases, and which contains a heater (glow) element embedded in magnesium-oxide powder (Fig. 1). This heater element comprises two series-connected resistors: the heater filament in the glow-tube tip, and the control filament. Whereas the heater filament maintains virtually constant electrical resistance regardless of temperature, the control filament is made of material with a positive temperature coefficient (PTC). On newer-generation glow plugs (GSK2), its resistance increases even more rapidly with rising temperature than was the case with the conventional S-RSK glow plug. This means that the newer GSK2 glow plugs are characterized by reaching the temperature needed for ignition far more quickly (850 °C in 4s). They also feature a lower steady-state temperature (Fig. 2) which means that the glow plug's temperature is limited to a non-critical level. The result is that the GSK2 glow plug can remain on for up to 3 minutes following engine start. This post-glow feature improves both the warm-up and run-up phases with considerable improvements in noise and exhaust-gas emissions.

Fig. 1



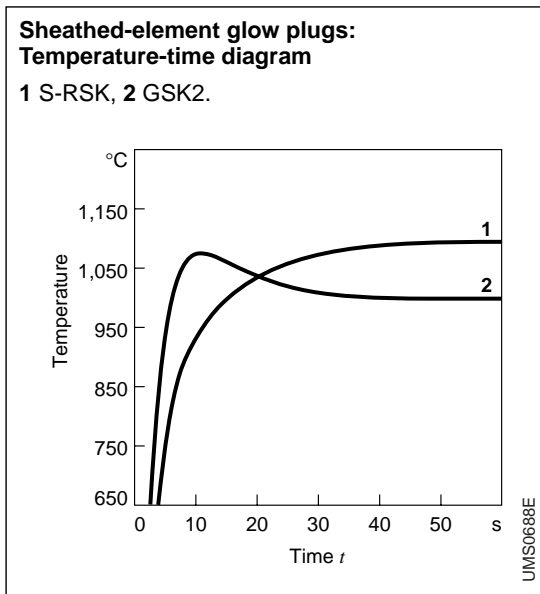


Fig. 2

Flame glow plug

The flame glow plug burns fuel to heat the intake air. Normally, the injection system's supply pump delivers fuel to the flame plug through a solenoid valve. The flame plug's connection fitting is provided with a filter, and a metering device which permits passage of precisely the correct amount of fuel appropriate to the particular engine. This fuel then evaporates in an evaporator tube surrounding the tubular heating element and mixes with the intake air. The resulting mixture ignites on the 1,000 °C heating element at the flame-plug tip.

Glow control unit

For triggering the glow plugs, the glow control unit (GZS) is provided with a power relay and a number of electronic switching blocks. These, for instance, control the glow duration of the glow plugs, or have safety and monitoring functions. Using their diagnosis functions, more sophisticated glow control units are also able to recognise the failure of individual glow plugs and inform the driver accordingly. Multiple plugs are used as the control inputs to the ECU. In order to avoid voltage drops, the power supply to the glow plugs is through suitable threaded pins or plugs.

Functional sequence

The diesel engine's glow plug and starter switch, which controls the preheat and starting sequence, functions in a similar manner to the ignition and starting switch on the spark-ignition (SI) engine. Switching to the "Ignition on" position starts the preheating process and the glow-plug indicator lamp lights up. This extinguishes to indicate that the glow plugs are hot enough for the engine to start, and cranking can begin. In the following starting phase, the droplets of injected fuel ignite in the hot, compressed air. The heat released as a result leads to the initiation of the combustion process (Fig. 3).

In the warm-up phase following a successful start, post-heating contributes to faultless engine running (no misfiring) and therefore to practically smokeless engine run-up and idle. At the same time, when the engine is cold, pre-heating reduces combustion noise. A glow-plug safety switchoff prevents battery discharge in case the engine cannot be started.

The glow-control unit can be coupled to the ECU of the Electronic Diesel Control (EDC) so that information available in the EDC control unit can be applied for optimum control of the glow plugs in accordance with the particular operating conditions. This is yet another possibility for reducing the levels of blue smoke and noise.

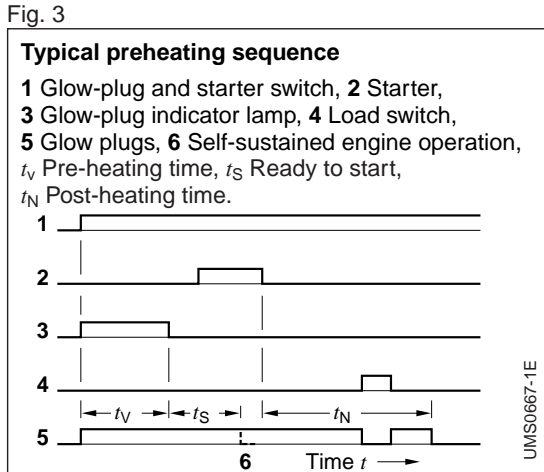


Fig. 3

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