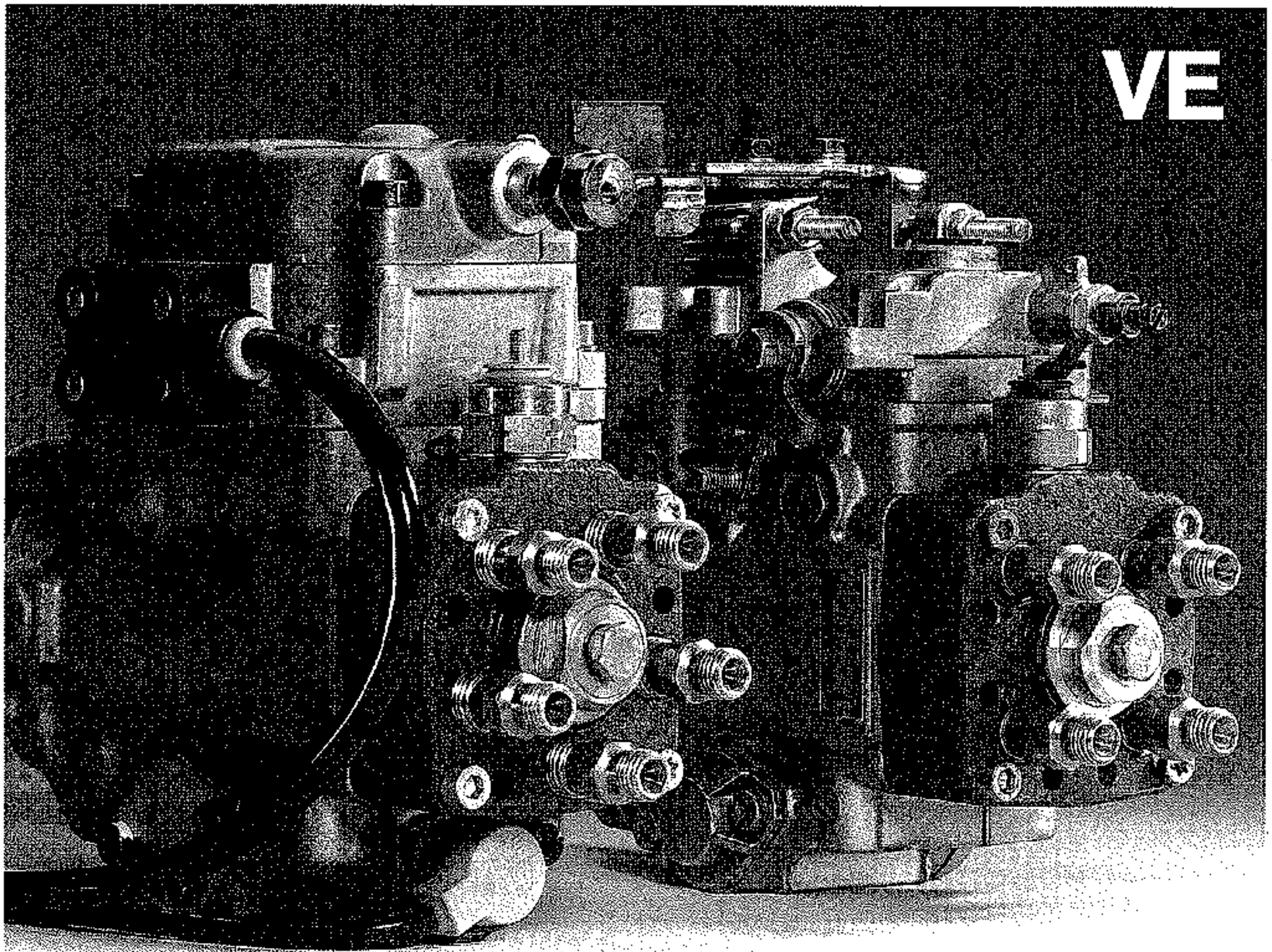
Diesel engine management

Diesel
Distributor Fuel-Injection Pumps

Edition 94/95



Technical Instruction

BOSCH

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 CO_2, 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.

Combustion in the diesel engine

The diesel engine

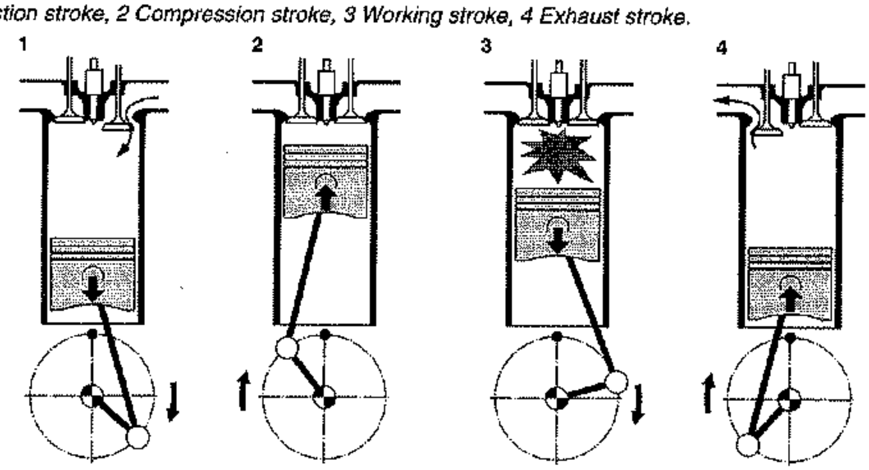
Diesel combustion principle

The diesel engine is a compression-ignition (CI) engine. Being as CI engines only draw in air, they are able to compress this to a level which is considerably higher than that in the spark-ignition (SI) engine using an air-fuel mixture. In addition, the SI engine is also sensitive to knock. 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 and the considerably reduced level of noise, 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 (Fig. 1).

Working cycle (4-stroke)

During the first stroke, the downward movement of the piston draws in unthrottled air through the open intake valve.
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 800°C. At the end of the compression stroke the nozzle injects fuel into the heated air at pressures of up to 1500 bar.
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. In the fourth stroke,

Fig. 1: 4-stroke diesel engine
1 Induction stroke, 2 Compression stroke, 3 Working stroke, 4 Exhaust 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 (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. Taking these features into account, the fact that the prechamber engine uses slightly more fuel than the DI engine (10...15%) is generally accepted as a compromise solution. Compared to the spark-ignition (SI) 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 toxic content of the exhaust gas. A further feature of the diesel engine is its suitability for running on alternative fuels (e.g., alcohol or rape-seed oil) whereby it may be necessary to adapt the fuel-injection equipment.

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 harmless 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.

The following contribute to the reduction of fuel consumption and exhaust-gas emissions: Accurate start-of-injection timing, precision-manufactured injection nozzles, fuel-injection pumps with precise fuel metering, modified combustion chambers, precisely-defined fuel-spray geometry, and further increases in injection pressure.

Distributor 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 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 process in the diesel engine depends 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.

Types

To keep pace with the ever-increasing demands placed upon the diesel fuel-injection system, it has been necessary to continually develop and improve the fuel-injection pump. The result is that today we have available an extensive range of in-line pumps, distributor pumps, and single-plunger pumps in a wide variety of sizes and types.
The following fuel-injection systems are in line with the present state-of-the-art:
– In-line injection pump (PE) with mechanical (flyweight) or electronic governor and, if required, timing device
– Control-sleeve injection pump (PE) with electronic governor and infinitely variable port closing (start of delivery). Without fitted timing device.
– Single-plunger injection pump (PF).
– Distributor injection pump (VE) with mechanical or electronic governor and integral timing device.
– Unit injector (PDE), in the form of a compact system, and
– Unit pump (PLD), a modular fuel-injection system.

Overview

Characteristics	Diesel fuel-injection pumps			
	VE	PE	PF	PDE/PLD
Injection pressure in bar (pump-side)	up to 700	up to 1150	up to 1500	up to 1500
Application	High-speed passenger-car and commercial-vehicle engines	Commercial vehicles, special vehicles, stationary engines	Marine engines, construction machinery	Commercial vehicles, passenger cars
Output per cylinder in kW/cylinder	up to 25	up to 70	up to 1000	up to 70

Fuel-injection technology

Applications

Today's small, high-speed diesel engine demands a lightweight and compact fuel-injection installation. The VE distributor pump 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.

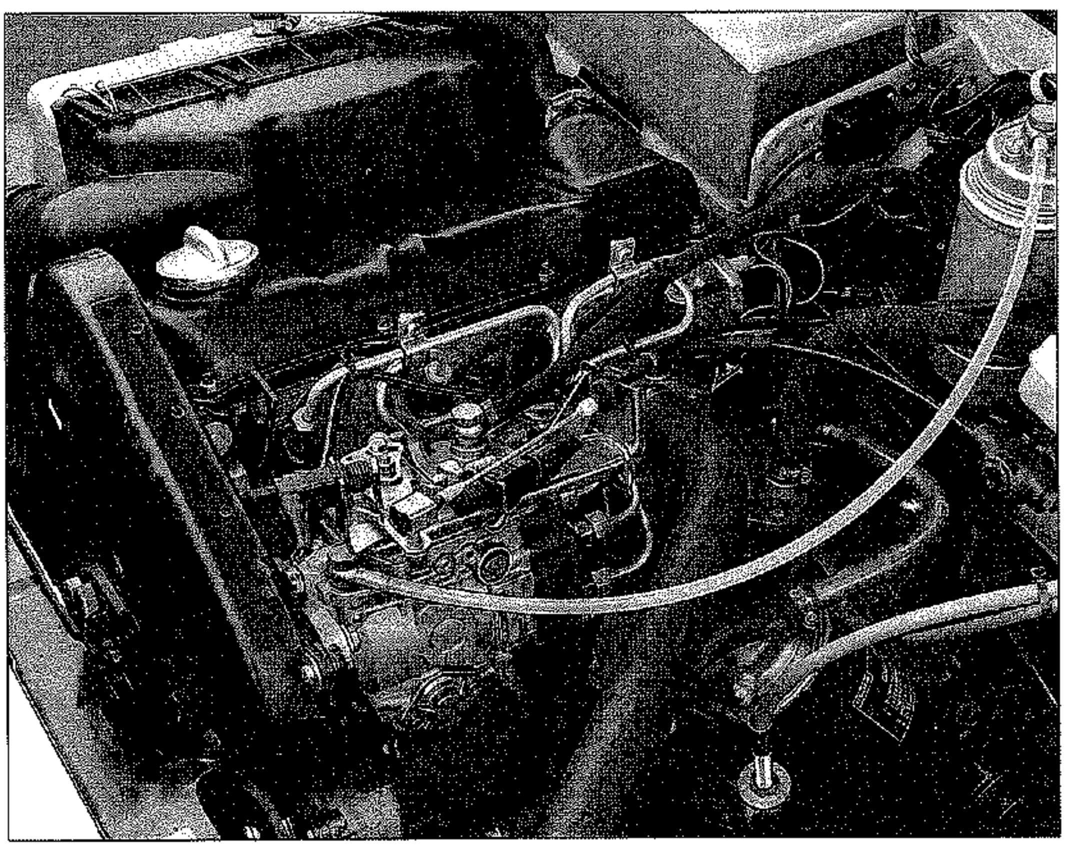
Subassemblies

In contrast to the in-line injection pump, the VE distributor pump has only one pump cylinder and one plunger regardless of the number of cylinders in the engine (Fig. 1). 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. 2 shows the functional groups and their assignments. The add-on modules facilitate adaptation to the specific requirements of the diesel engine in question.

Fig. 1: VE distributor pump fitted to a 4-cylinder diesel engine.

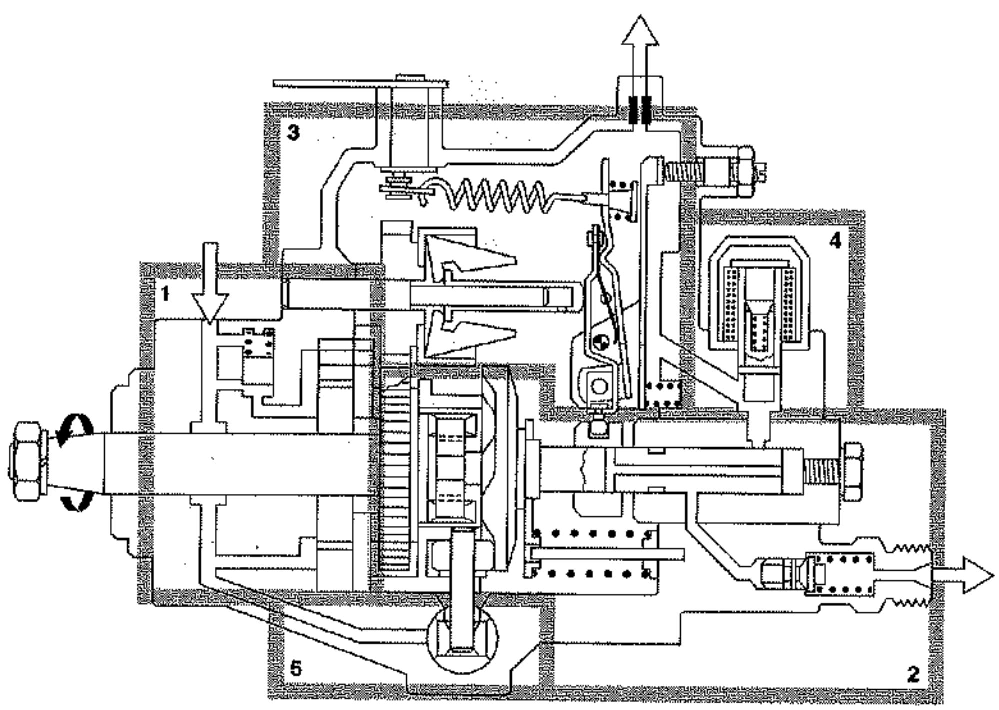


Design and construction

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 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

Fig. 2: The subassemblies and their functions
1 Vane-type fuel-supply pump with pressure regulating valve: Draws in fuel and generates pressure inside the pump.
2 High-pressure pump with distributor: Generates injection pressure, delivers and distributes fuel.
3 Mechanical (flyweight) governor: Controls the pump speed and varies the delivery quantity within the control range.
4 Electromagnetic fuel shutoff valve: Interrupts the fuel supply.
5 Timing device: Adjusts the start of delivery (port closing) as a function of the pump speed and in part as a function of the load.

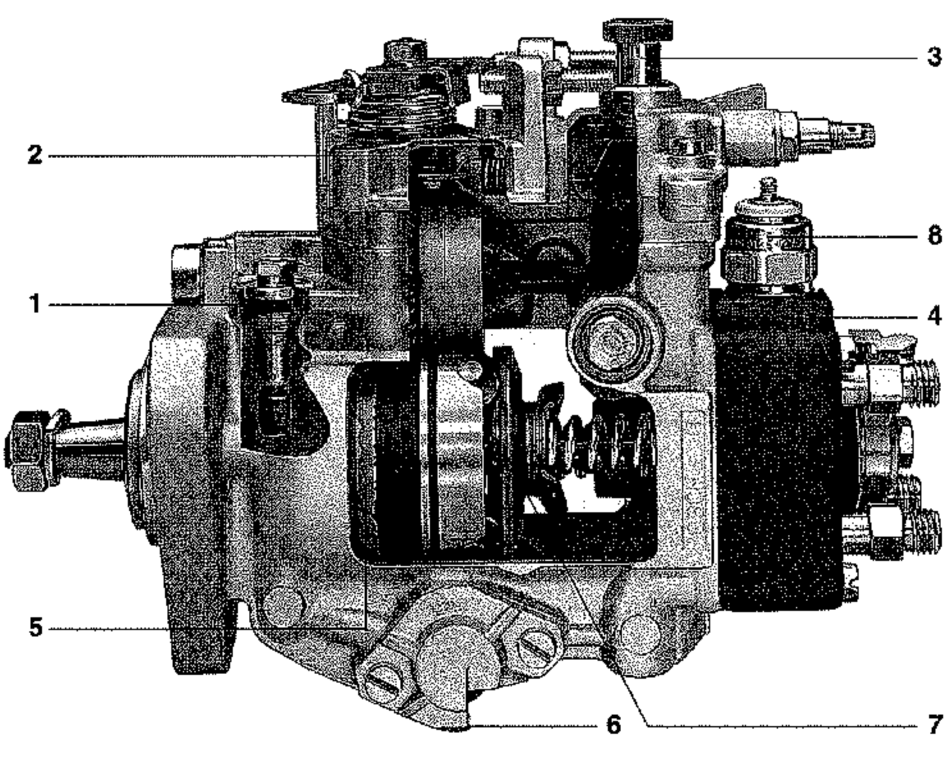


the pump at right angles to the pump's longitudinal axis. Its operation is influenced by the pump's internal pressure which in turn is defined by the vane-type fuel-supply pump and by the pressure-regulating valve. The timing device is closed off by a cover on each side of the pump (Figs. 2 and 3).

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.

Fig. 3: 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.



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 delivery

Low-pressure stage

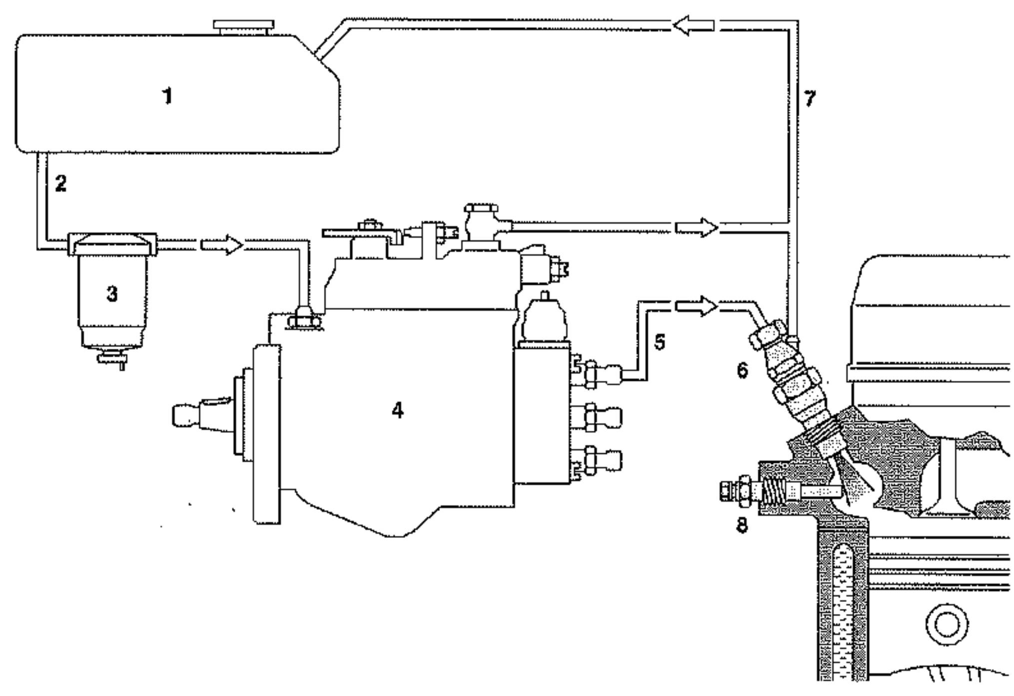
The low-pressure stage of a distributor-pump fuel-injection installation comprises the fuel tank, fuel lines, fuel filter, vane-type fuel-supply pump, pressure-control valve, and overflow restriction.

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 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

Fig. 1: Fuel supply and delivery in a distributor-pump fuel-injection system
1 Fuel tank, 2 Fuel line (suction pressure), 3 Fuel filter, 4 Distributor injection pump, 5 High-pressure fuel-injection line, 6 Injection nozzle, 7 Fuel-return line (pressureless), 8 Sheathed-element glow plug.



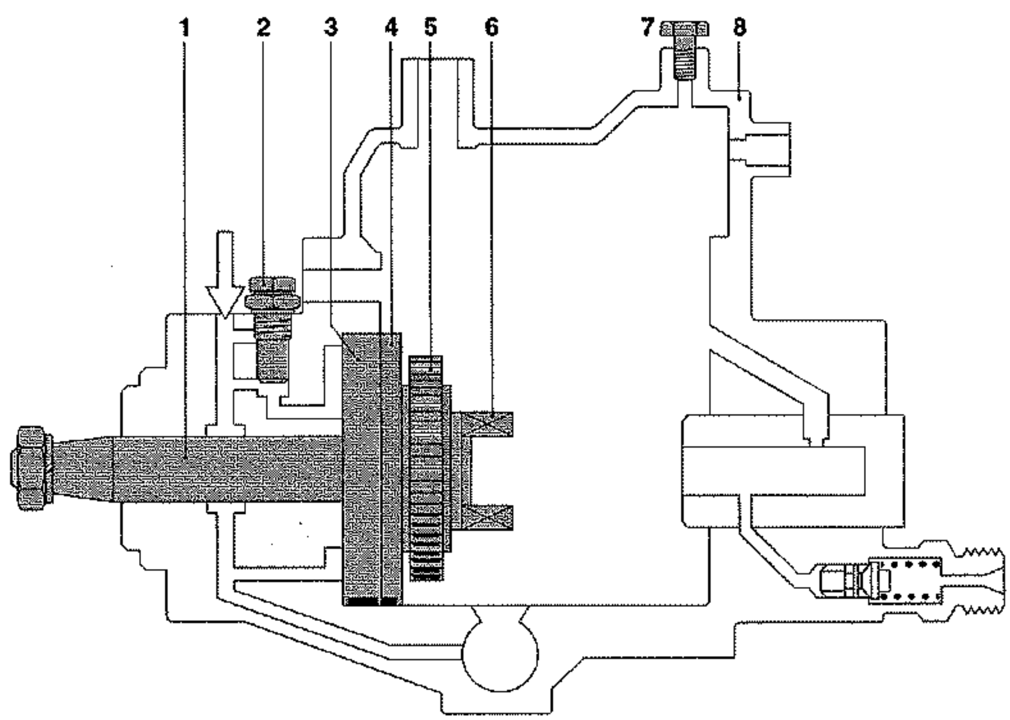
result, the vane-type supply pump in the injection pump is powerful enough to draw the fuel out of the fuel tank and to build up sufficient pressure in the interior of the injection pump.
In those cases in which the difference in height between fuel tank and injection pump is excessive and (or) the fuel 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 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 only several thousandths of a millimeter. This means

Fig. 2: Interaction of the fuel-supply pump, pressure-control valve, and overflow restriction
1 Drive shaft, 2 Pressure-control valve, 3 Eccentric ring, 4 Support ring, 5 Governor drive, 6 Drive-shaft dogs, 7 Overflow restriction, 8 Pump housing.



that contaminants in the fuel can lead to malfunctions. 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.

Fig. 3: Vane-type fuel-supply pump for low-pressure delivery
1 Inlet, 2 Outlet.

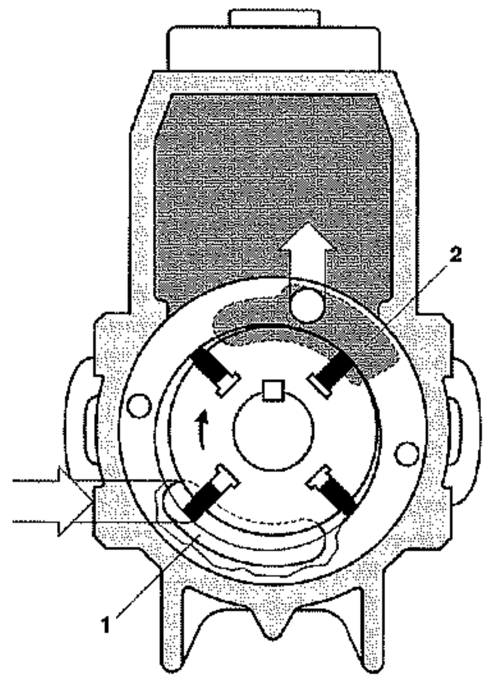
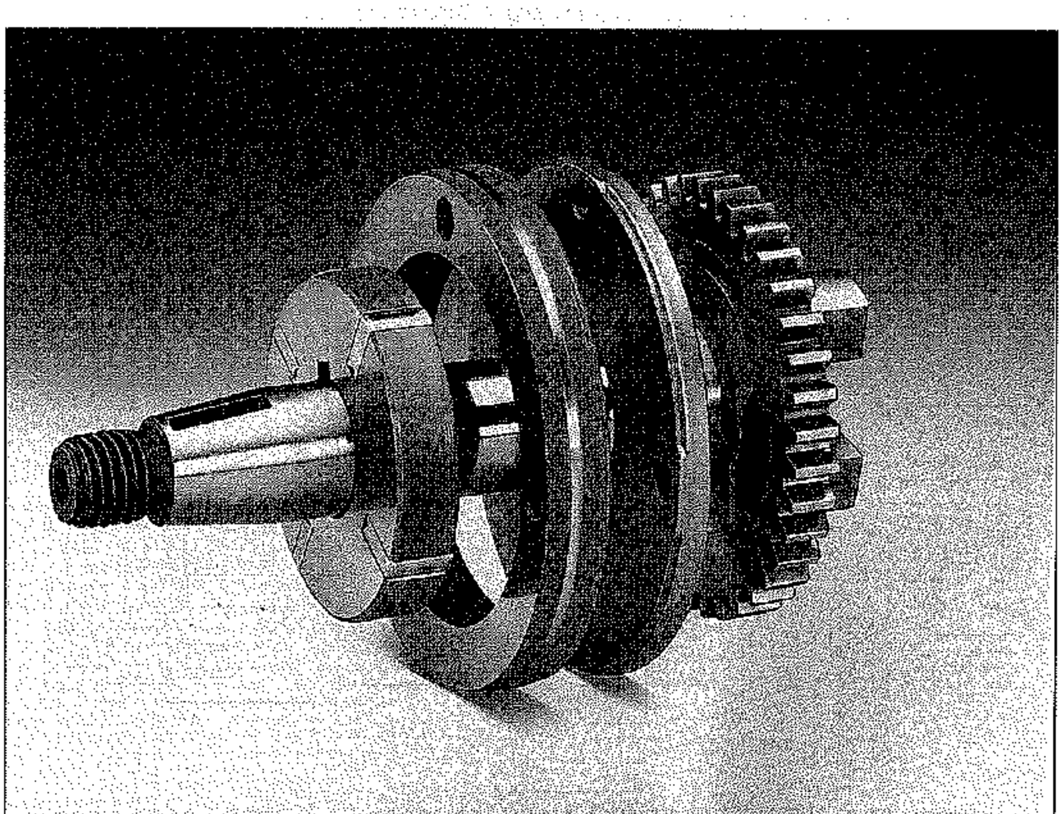


Fig. 4: Vane-type fuel-supply pump with impeller on the drive shaft.



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

Overflow restriction

The overflow restriction (Fig. 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 pressure-control valve are precisely matched to each other.

Fig. 5: Pressure-control valve

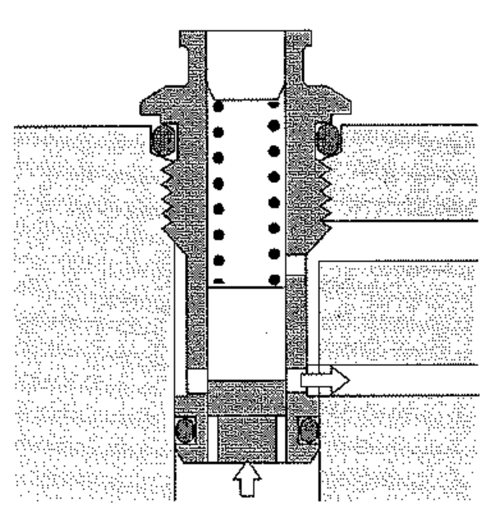
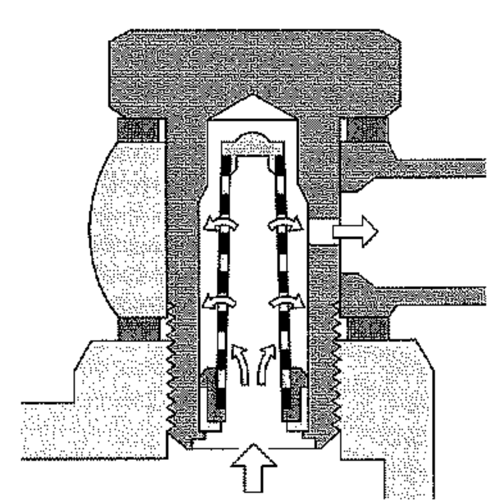


Fig. 6: Overflow restriction



High-pressure delivery

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 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. 7: Pump assembly for generation and delivery of high pressure in the distributor-pump interior

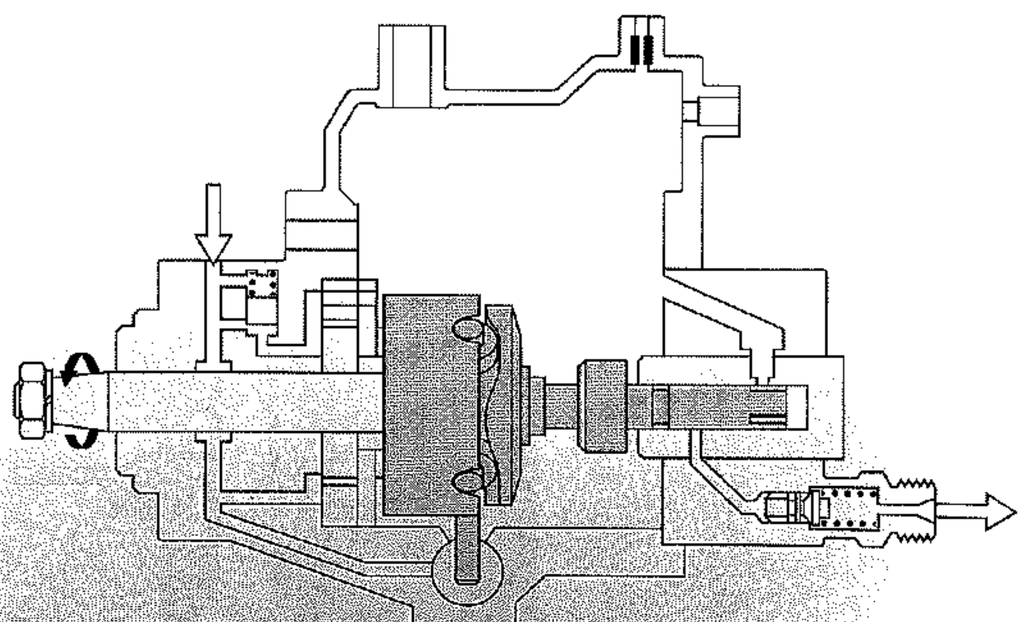
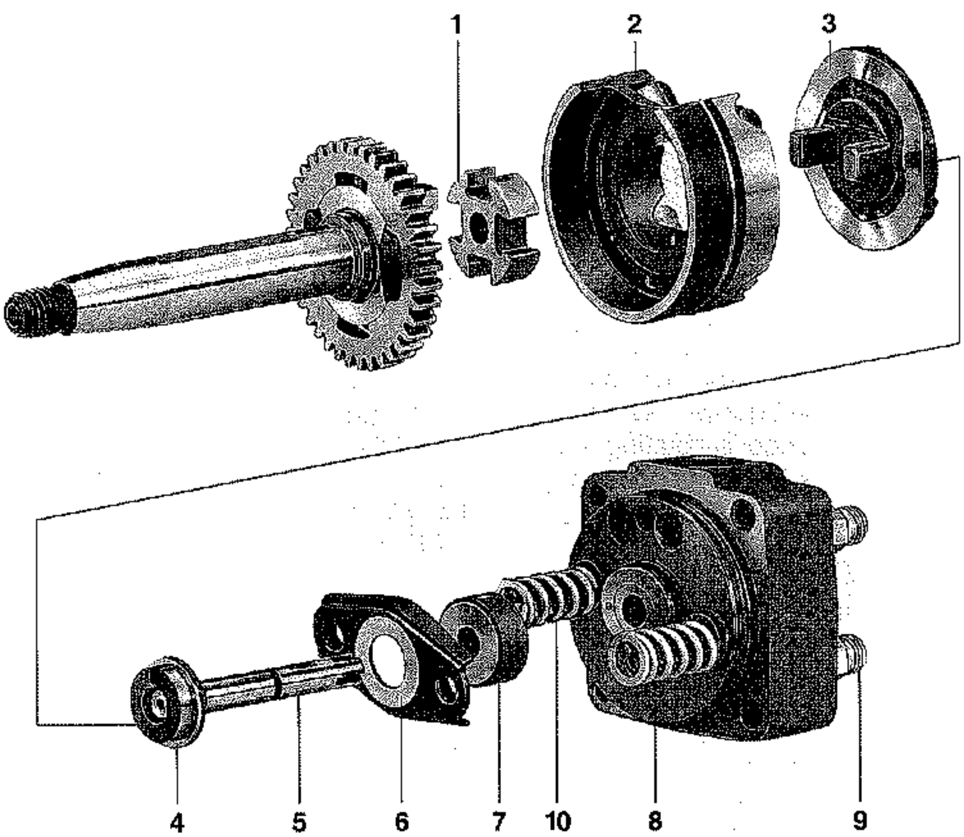


Fig. 8: Pump assembly with distributor head
Generates the high pressure and distributes the fuel to the respective fuel injector.
1 Yoke, 2 Roller ring, 3 Cam plate, 4 Distributor-plunger foot, 5 Distributor plunger, 6 Link element, 7 Control collar, 8 Distributor-head flange, 9 Delivery-valve holder, 10 Plunger-return spring.
4...8 Distributor head.



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. And because the cam-plate surface is specific to a given engine type, the cam plate is not interchangeable with those of other engine types.

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 (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 (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.

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 (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 (10d).

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).

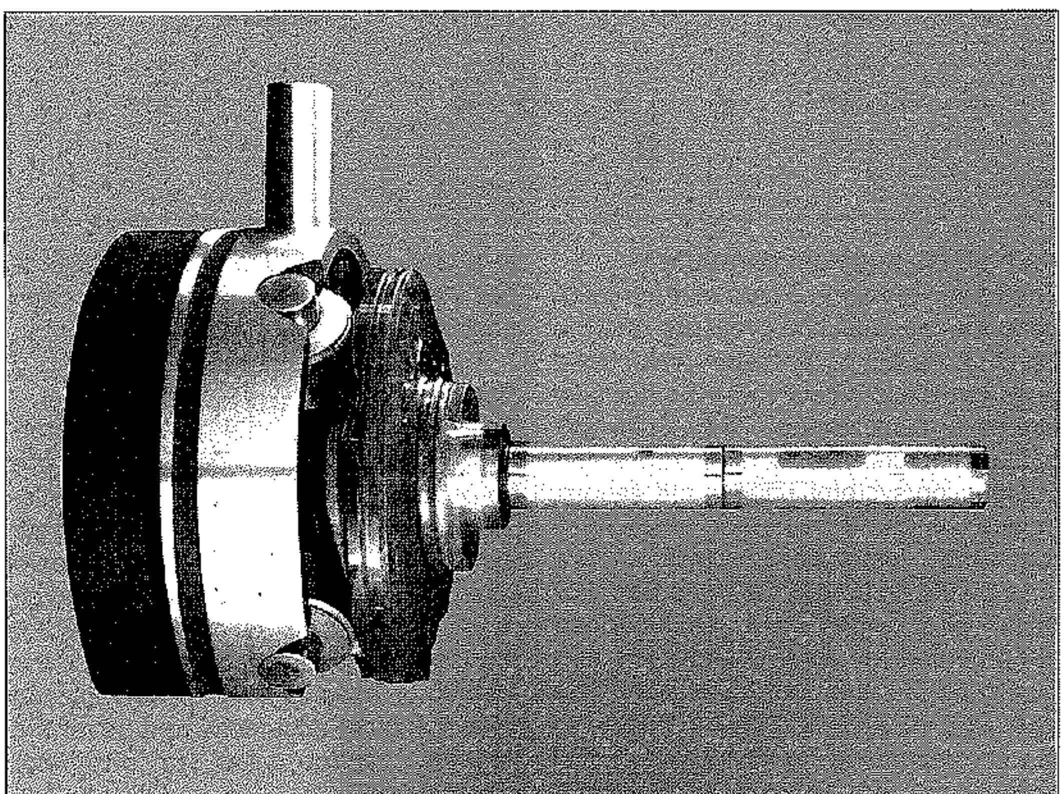
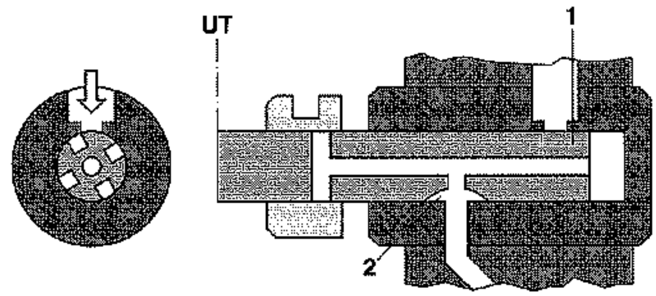
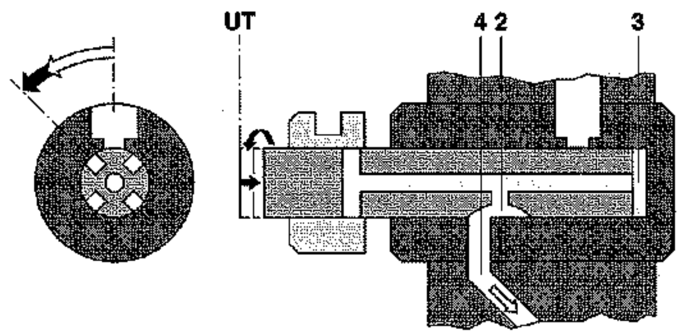


Fig. 10: Distributor plunger with stroke and delivery phases

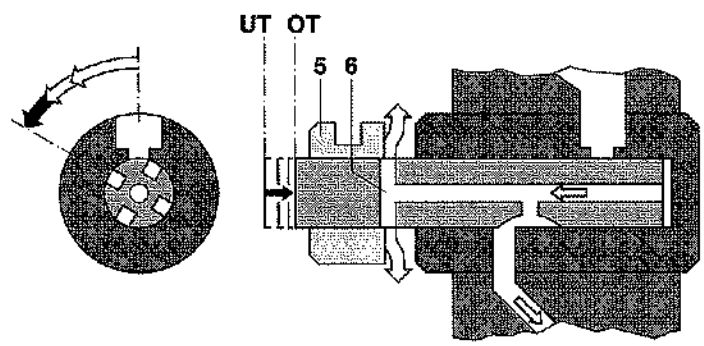
a)
Inlet passage closes.
At BDC, the metering slot (1) closes the inlet passage, and the distributor slot (2) opens the outlet port.



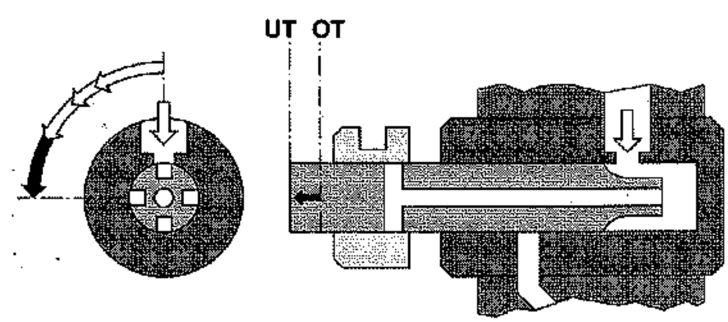
b)
Fuel delivery.
During the plunger stroke towards TDC (working stroke), the plunger pressurizes the fuel in the high-pressure chamber (3). The fuel travels through the outlet-port passage (4) to the injection nozzle.



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.

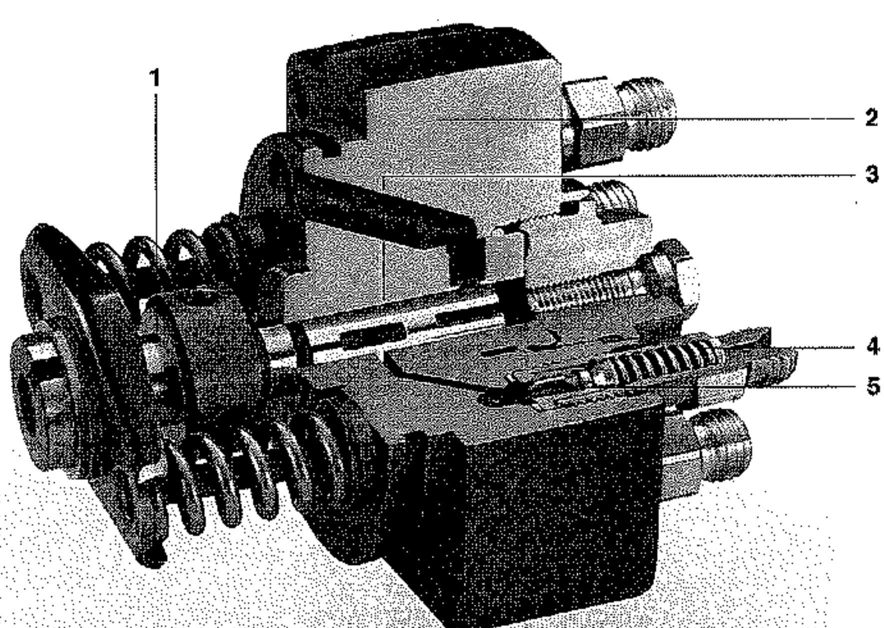


OT = TDC
UT = BDC

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 a given 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: Delivery valve with return-flow restriction
1 Delivery-valve holder, 2 Return-flow restriction, 3 Delivery-valve spring, 4 Valve holder, 5 Valve stem, 6 Retraction piston.

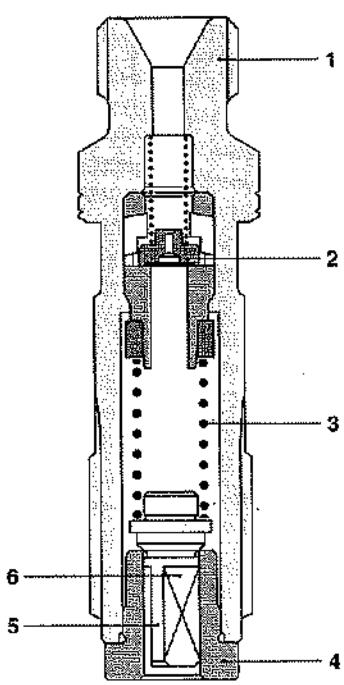
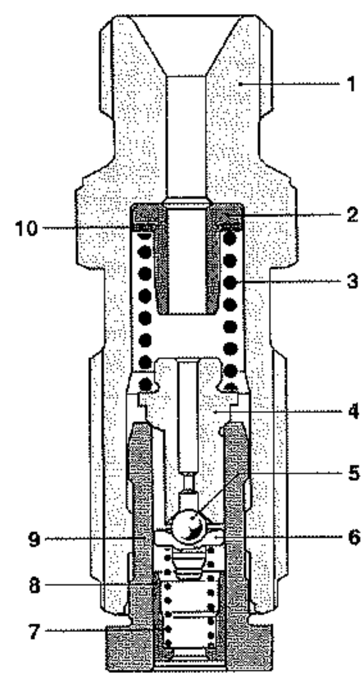


Fig. 13: Constant-pressure valve
1 Delivery-valve holder, 2 Filler piece with spring locator, 3 Delivery-valve spring, 4 Delivery-valve plunger, 5 Constant-pressure valve, 6 Spring seat, 7 Valve spring (constant-pressure valve), 8 Setting sleeve, 9 Valve holder, 10 Shims.



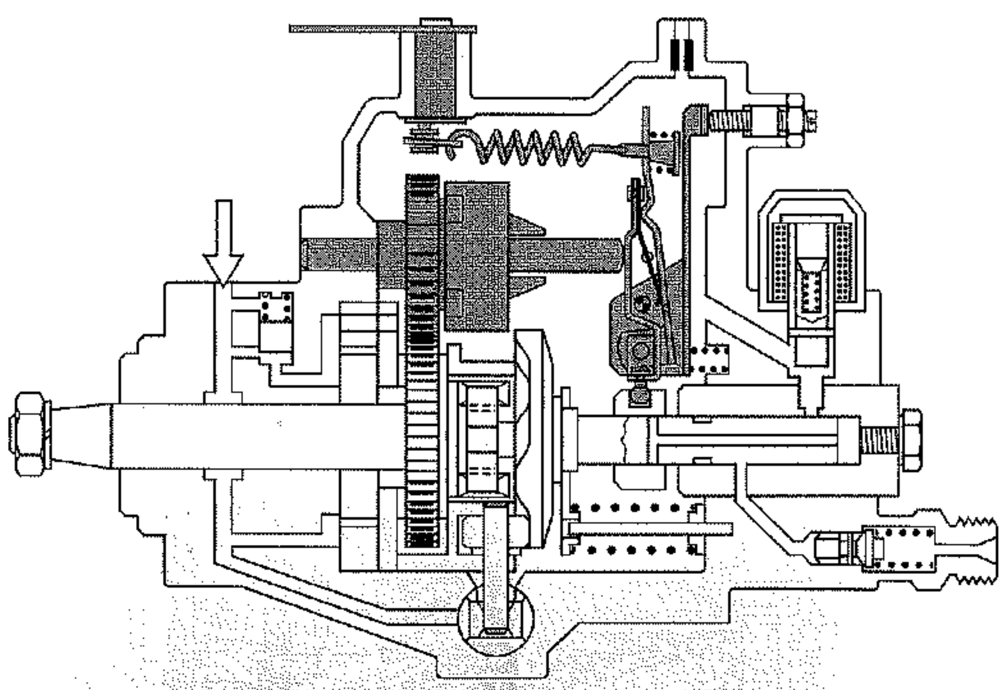
Mechanical engine-speed control (governing)

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).

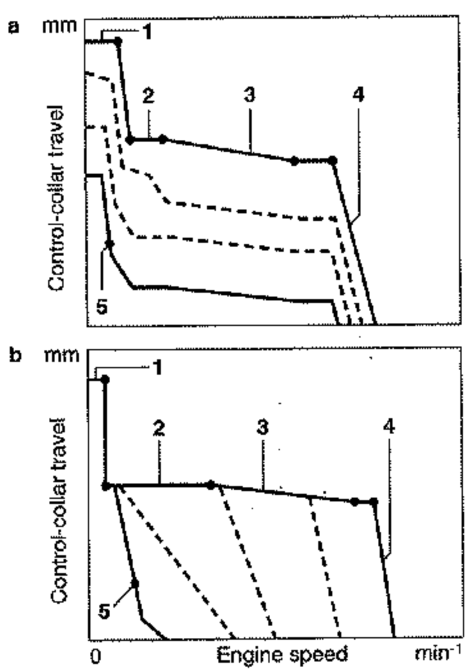
Governor functions
The basic function of all governors is the limitation of the engine's maximum speed. Depending upon type, the governor is also responsible for keeping certain engine speeds constant, such as idle speed, or the minimum and maximum engine speeds of a stipulated engine-speed range, or 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: Distributor injection pump with governor assembly, comprising flyweight governor and lever assembly



– 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.

Fig. 2: Governer characteristics
a) Minimum-maximum-speed governor,
b) Variable-speed governor.
1 Start quantity, 2 Full-load delivery,
3 Torque control (positive),
4 Full-load speed regulation, 5 Idle.



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 P-degree
n_{lo} High idle (maximum) speed
n_{vo} Maximum full-load speed

The required speed droop depends on engine application. For instance, for 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.

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).

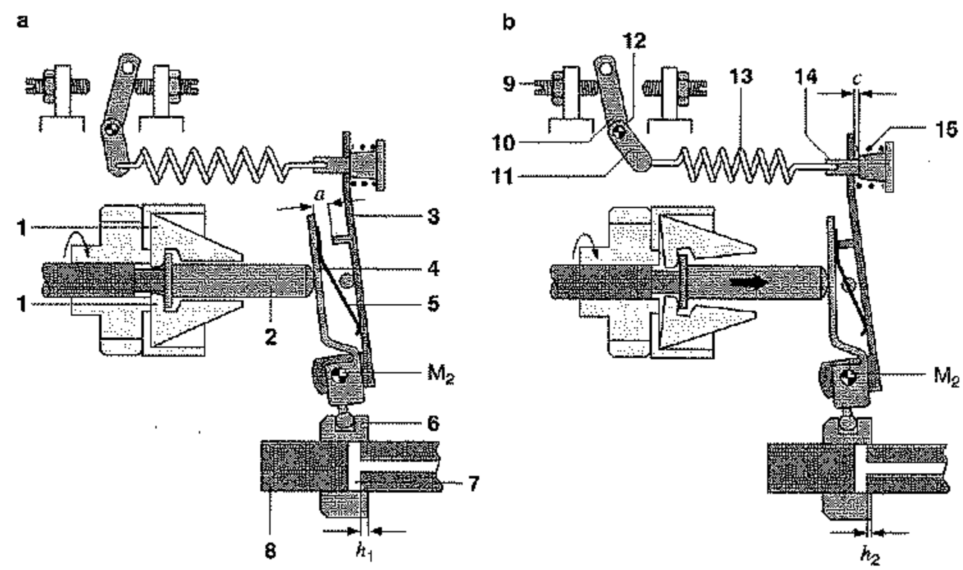
Design and construction

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: Variable-speed governor. Start and idle positions
a) Start position, b) Idle position.
1 Flyweights, 2 Sliding sleeve, 3 Tensioning lever, 4 Starting lever, 5 Starting spring, 6 Control collar, 7 Distributor-plunger cutoff port, 8 Distributor plunger, 9 Idle-speed adjusting screw, 10 Engine-speed control lever, 11 Control lever, 12 Control-lever shaft, 13 Governor spring, 14 Retaining pin, 15 Idle spring.
a Starting-spring travel, c Idle-spring travel, h_1 max. working stroke (start); h_2 min. working stroke (idle); M_2 fulcrum for 4 and 5.



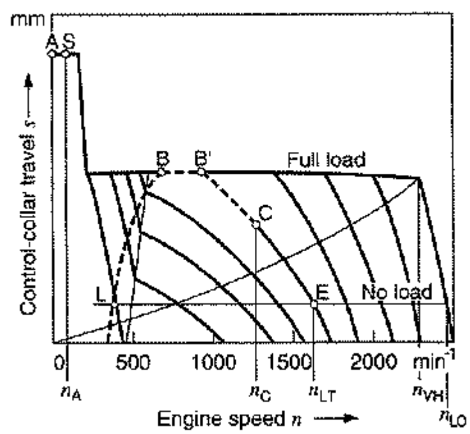
start-quantity position by the ball pin on the starting lever. This means that when the engine is cranked the distributor plunger must travel through a complete working stroke (= maximum delivery quantity) before the cutoff bore is opened and delivery ceases. Thus the start quantity (= maximum delivery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

Fig. 4: 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.



Low-idle-speed control

With the engine running, and the accelerator pedal released, the engine-speed control lever shifts to the idle position (Fig. 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.

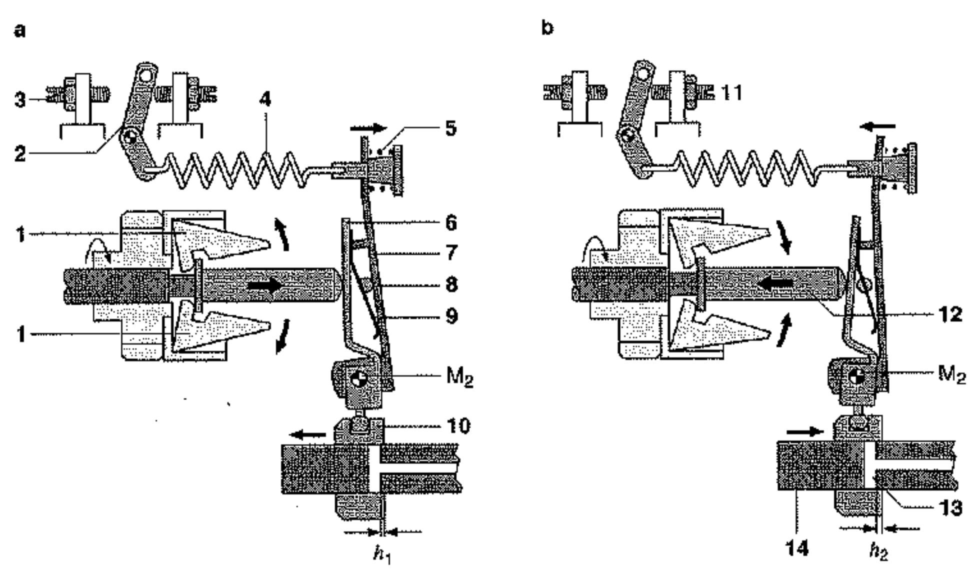
This balance of forces determines the sliding-sleeve's position relative to the distributor plunger's cutoff bore, and with it the working stroke. At speeds above idle, the spring has been compressed by the amount c and is no longer effective. Using the special idle spring attached to the governor housing,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

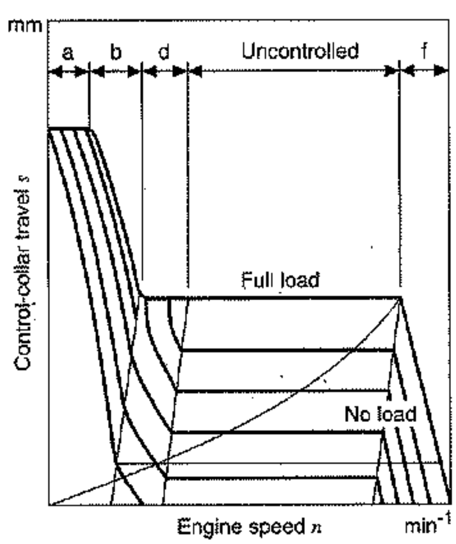
tion. This is taken over by the governor spring.
Example (Fig. 5):
Using the accelerator pedal, the driver sets the engine-speed control lever to a specific position corresponding to a desired (higher) speed. As a result of this adjustment of the control-lever position, the governor spring is tensioned 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 betwen 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. The engine is overloaded and the driver must change down to a lower gear.

Fig. 5: Variable-speed governor, operation under load
a) Governor function with increasing engine speed, b) with falling engine speed.
1 Flyweights, 2 Engine-speed control lever, 3 Idle-speed adjusting screw, 4 Governor spring, 5 Idle spring, 6 Start lever, 7 Tensioning lever, 8 Tensioning-lever stop, 9 Starting spring, 10 Control collar, 11 Adjusting screw for high idle (maximum) speed, 12 Sliding sleeve, 13 Distributor-plunger cutoff bore, 14 Distributor plunger.
h_1 Working stroke, idle, h_2 Working stroke, full-load, M_2 fulcrum for 6 and 7.



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.

Fig. 6: Characteristic curves of the minimum-maximum-speed governor with idle spring and intermediate spring
a: Starting-spring range,
b: Range of starting and idle spring ,
d: Intermediate-spring range,
f: Governor-spring range.



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.

Operation under load

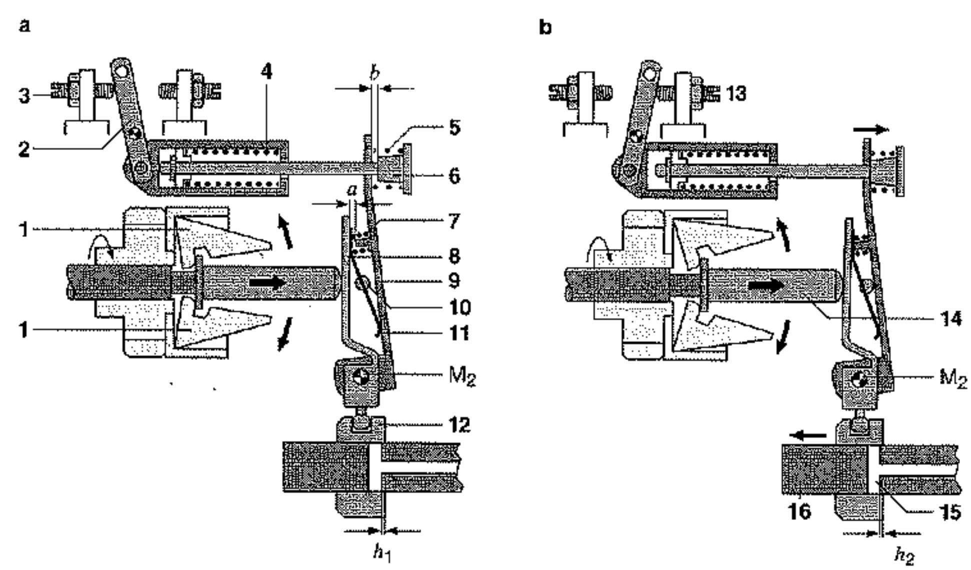
If the driver depresses the accelerator pedal, the engine-speed control lever is pivoted through a given angle. The starting and idle springs are no longer effective and the intermediate spring comes into effect. The intermediate spring on the minimum-maximum-speed governor provides a "soft" transition 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 pre-tension, 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, its speed increases to the high idle speed, and the engine is thus protected against overrevving.

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

Fig. 7: Minimum-maximum-speed governor
a) Idle setting, b) Full-load setting.
1 Flyweights, 2 Engine-speed control lever, 3 Idle-speed adjusting screw, 4 Governor spring, 5 Intermediate spring, 6 Retaining pin, 7 Idle spring, 8 Start lever, 9 Tensioning lever, 10 Tensioning-lever stop, 11 Starting spring, 12 Control collar, 13 Full-load speed control, 14 Sliding sleeve, 15 Distributor-plunger cutoff bore, 16 Distributor plunger.
a Start and idle-spring travel, b Intermediate-spring travel, h_1 Idle working stroke, h_2 Full-load working stroke, M_2 fulcrum for 8 and 9.



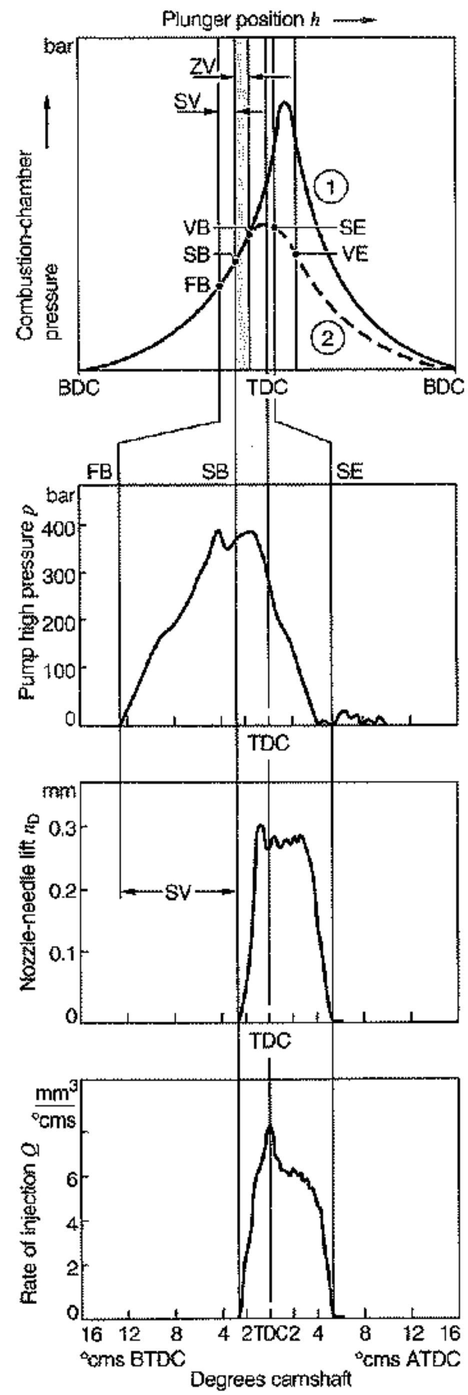
Injection timing

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. 1500 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: Curve of a working stroke at full load and at low speed *(not drawn to scale).*
FB *Start of delivery,* SB *Start of injection,*
SV *Injection delay,* VB *Start of combustion,*
ZV *Ignition lag,* SE *End of injection,* VE *End of combustion,*
① *Combustion pressure,*
② *Compression pressure,*
UT *BDC,*
OT *TDC.*



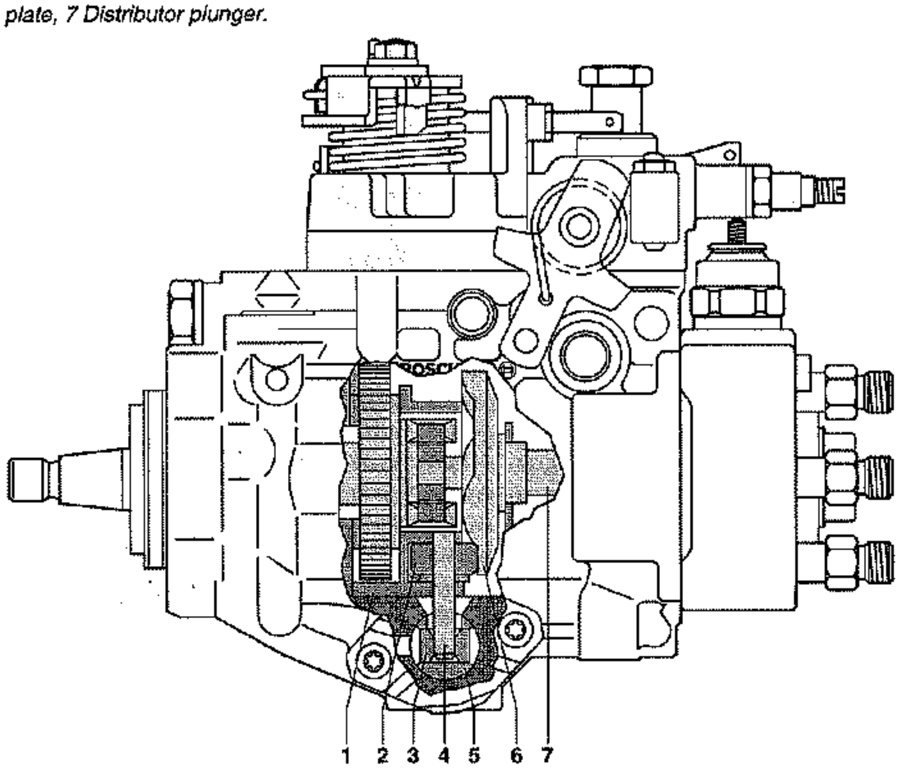
order to atomize and mix with the air to form an ignitable mixture.
This is termed the air-fuel mixture preparation time and is independent of engine speed. In a diesel engine, the time required between start of injection and start of combustion is termed the ignition lag.
The ignition lag is influenced 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 through a sliding block and a pin so that piston movement can be converted to rotational movement of 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.

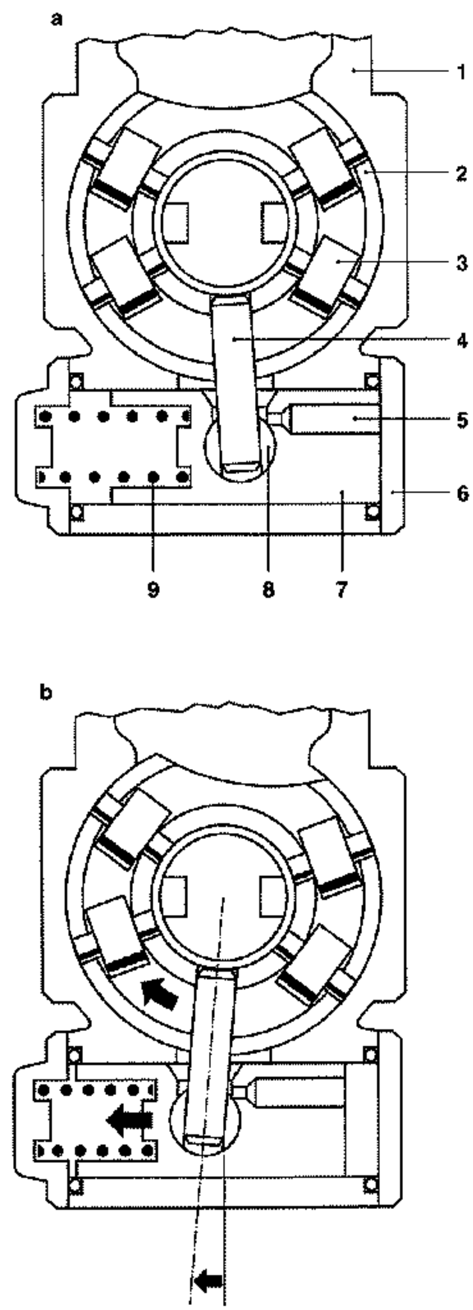


Method of operation

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: Timing device, method of operation
a) Initial position,
b) Operating position.
1 Pump housing, 2 Roller ring,
3 Roller-ring rollers, 4 Pin,
5 Passage in timing-device piston,
6 Cover, 7 Timing-device piston,
8 Sliding block, 9 Timing-device spring.



Injection timing

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

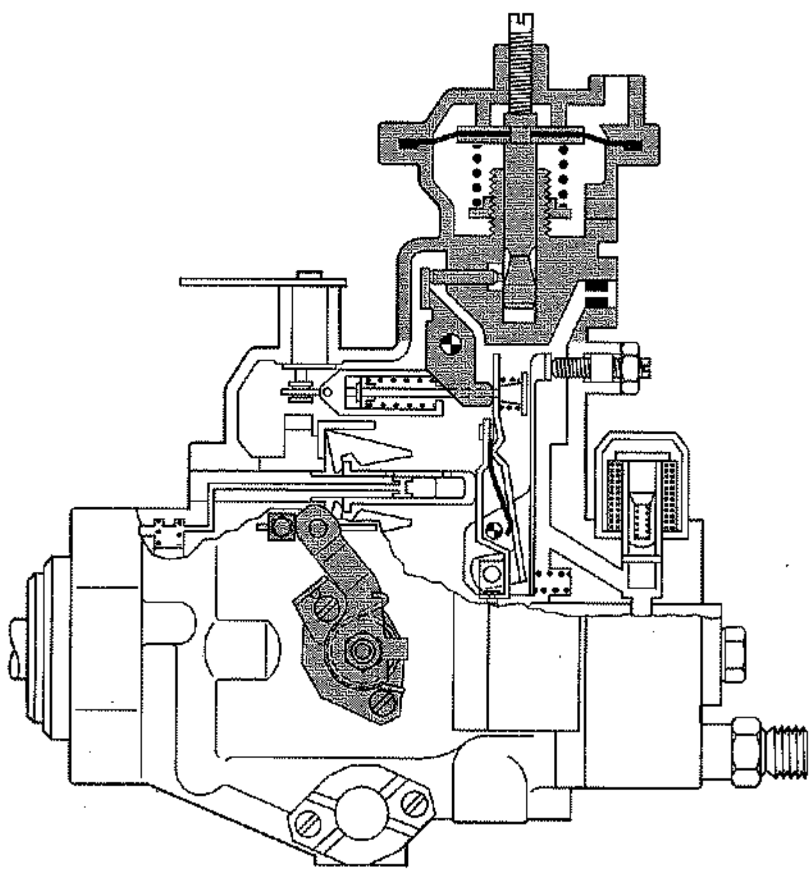


Fig. 2: Schematic of the VE distributor pump with mechanical/hydraulic full-load torque control

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.

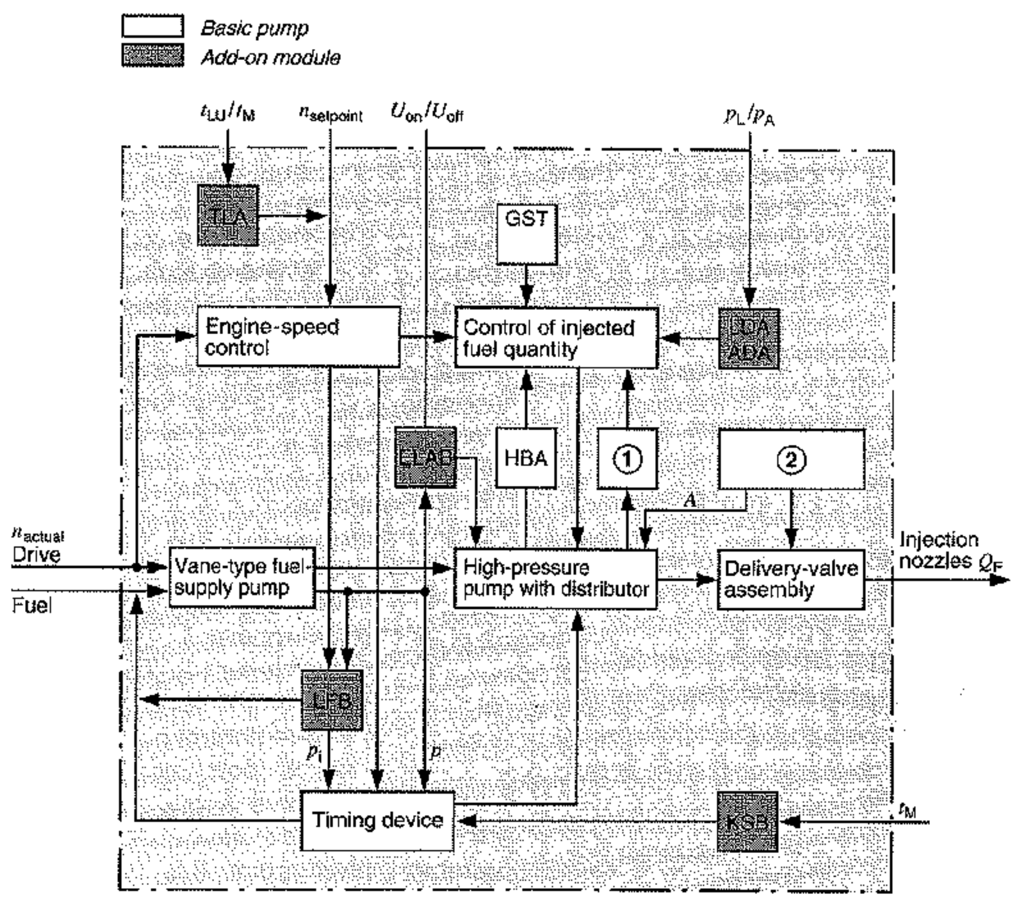
KSB Cold-start accelerator.
Improves cold-start behavior by changing the start of delivery.

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.

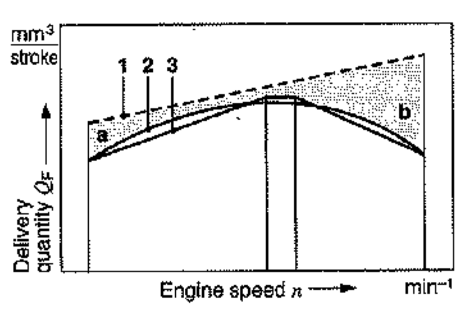
ELAB Electrical shutoff device.

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.



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 delivery-quantity 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.

Fig. 3: Fuel-delivery characteristics, with and without torque control
a) Negative, b) Positive torque control.
1 Excess injected fuel,
2 Engine fuel requirement,
3 Full-load delivery with torque control,
Shaded area:
Full-load delivery without torque control.



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.

<u>Positive torque control using the delivery valve</u>

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

<u>Positive torque control using the cutoff port</u>

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.

<u>Positive torque control using the governor lever assembly</u> (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, 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.

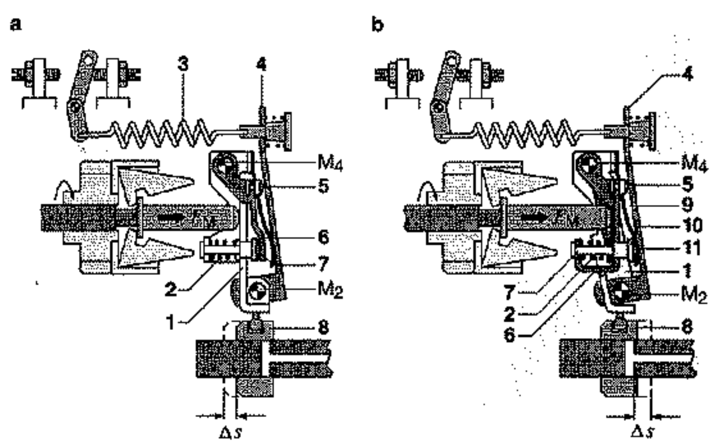
Negative torque control using hydraulically controlled torque control (HBA)

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: Torque control using the governor-lever assembly

a) Positive torque control,
b) Negative torque control.
1 Starting lever,
2 Torque-control spring,
3 Governor spring,
4 Tensioning lever,
5 Stop lug,
6 Torque-control lever,
7 Torque-control pin,
8 Control collar,
9 Starting spring,
10 Pin collar,
11 Stop point,
M_2 *Pivot point for 1 and 4,*
M_4 *Pivot point for 1 and 6,*
F_M *Sliding-sleeve force,*
Δs *Control-collar travel.*



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 (Fig. 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 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

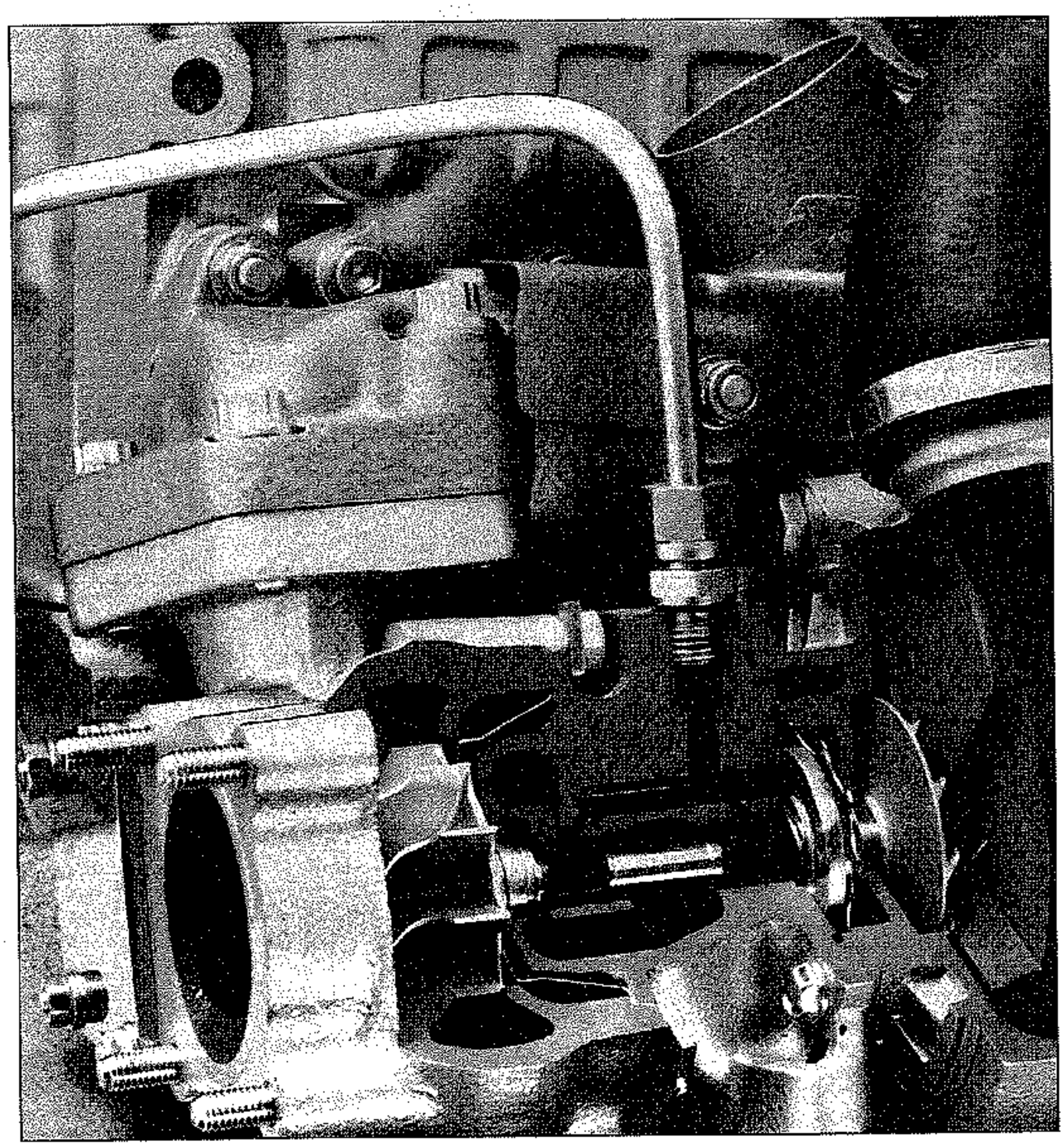
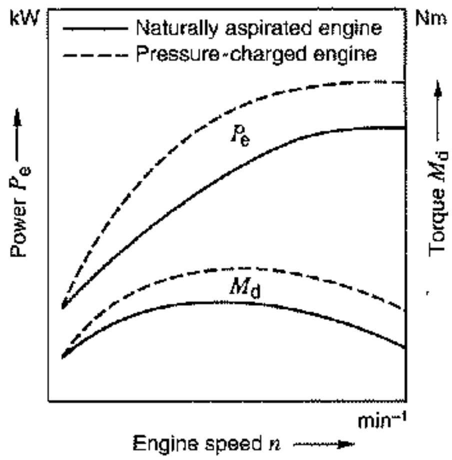


Fig. 6: Power and torque comparison, naturally aspirated and pressure-charged engines



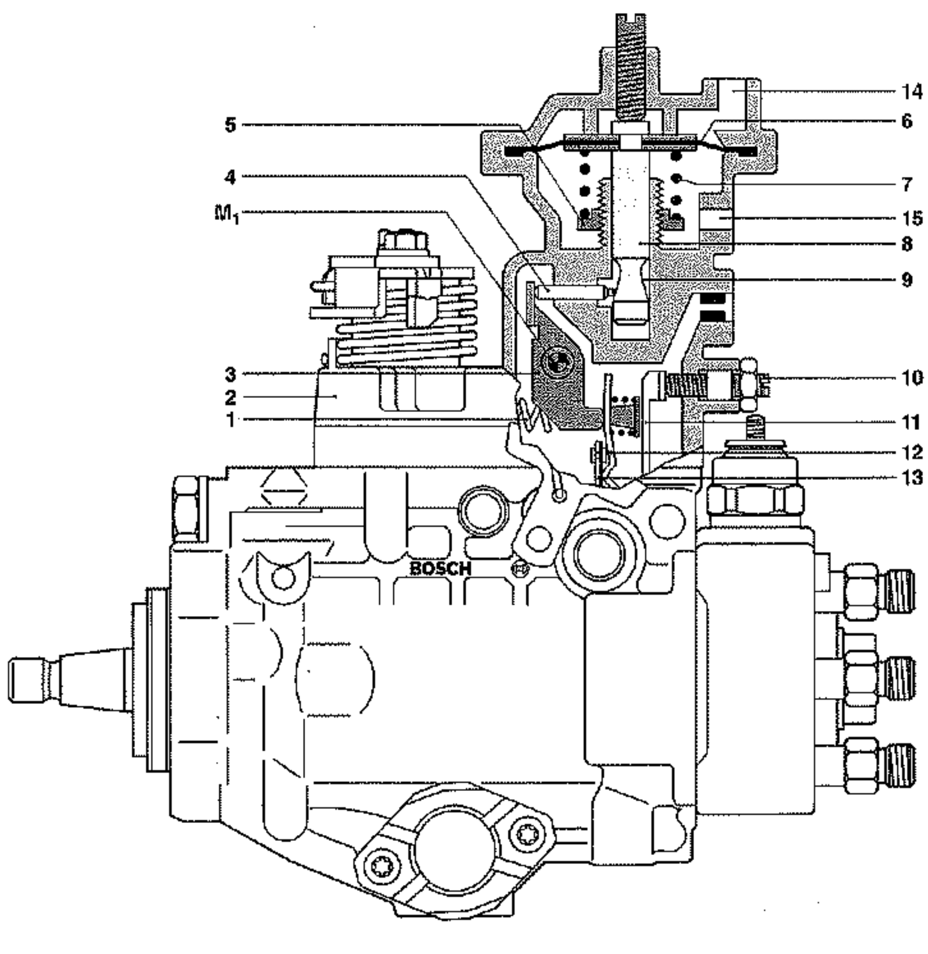
Manifold-pressure compensator (LDA)

The manifold-pressure compensator (LDA) reacts to the charge-air pressure generated by the exhaust-gas turbocharger, or the (mechanical) supercharger, 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-

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.*



charged diesel engine operates with a reduced cylinder air charge, the injected 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.

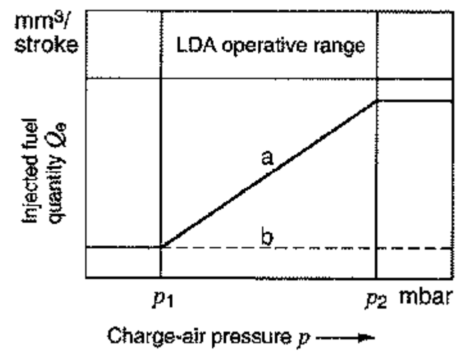
Design and construction

The LDA is mounted on the top of the distributor pump. 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 diaphragn 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. 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

Fig. 8: Charge-air pressure – operative range
a Turbocharger operation, b Normally aspirated operation.
p_1 Lower charge-air pressure,
p_2 Upper charge-air pressure.



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.

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.

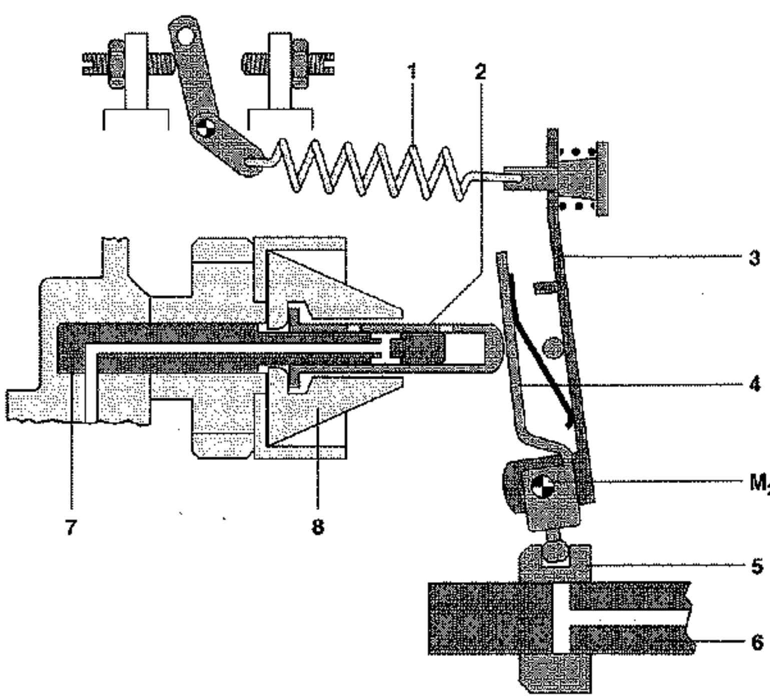
Design and construction

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

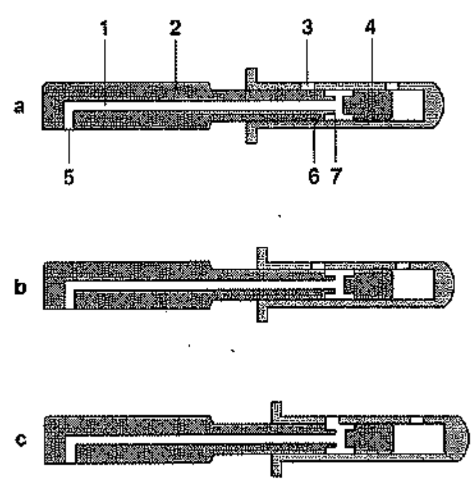
As a result of the rise in the supply-pump pressure when the engine speed increases, the timing device adjusts the start of delivery in the "advance" direction. On the other hand, with the drop in the pump's interior pressure caused by the LFB it is possible to implement a (relative) shift in the "retard"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

Fig. 9: Design and construction of load-dependent start of delivery (LFB)
1 Governor spring, 2 Sliding sleeve, 3 Tensioning lever, 4 Start lever, 5 Control collar, 6 Distributor plunger, 7 Governor shaft, 8 Flyweights.
M_2 *Pivot point for 3 and 4.*



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).

Fig. 10: Sliding-sleeve positions in the load-dependent injection timing (LFB)
a) Start position (initial position),
b) Full-load position shortly before the control port is opened,
c) Control port opened, pressure reduction in pump interior.
1 Longitudinal bore in the governor shaft, 2 Governor shaft, 3 Sliding-sleeve control port, 4 Sliding sleeve, 5 Governor-shaft transverse passage, 6 Control edge of the groove in the governor shaft, 7 Governor-shaft transverse passage.



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 700 mbar (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. 7).

Cold-start compensation

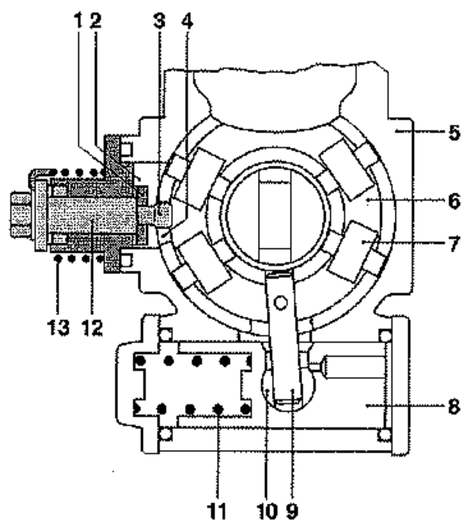
The diesel engine's cold-start characteristics are improved by fitting a cold-start compensation module which shifts the start of injection in the "advance" direction. Operation is triggered either by the driver using a bowden cable in the cab, or automatically by means of a temperature-sensitive advance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

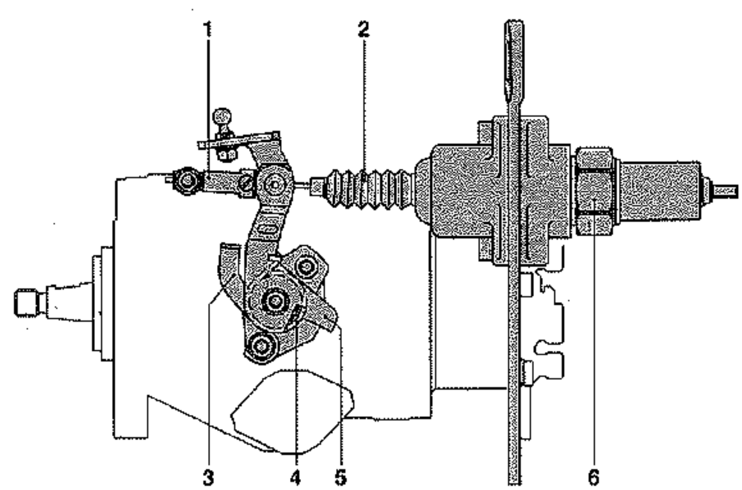
Fig. 12: Mechanical cold-start accelerator (KSB) engaging in roller ring (cold-start position)
1 Lever, 2 Access window, 3 Ball pin, 4 Longitudinal slot, 5 Pump housing, 6 Roller ring, 7 Roller in the roller ring, 8 Timing-device piston, 9 Torque-control pin, 10 Sliding block, 11 Timing-device spring, 12 Shaft, 13 Coil spring.



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

Fig. 11: Mechanical cold-start accelerator (KSB), advance mechanism with automatic operation (cold-start position)
1 Clamp,
2 Bowden cable,
3 Stop lever,
4 Coil spring,
5 KSB advance lever,
6 Control device sensitive to coolant temperature.



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: Effect of the mechanical cold-start accelerator (KSB)
a Timing-device advance,
b Minimum advance (approx. 2.5 °camshaft).

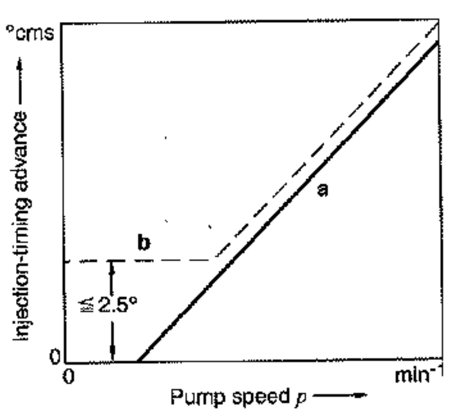
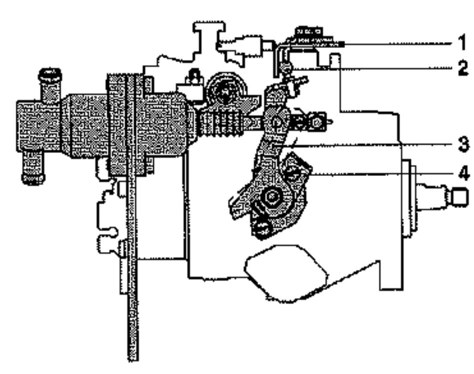


Fig. 14: Mechanical cold-start accelerator (automatically controlled) with temperature-dependent idle-speed increase
1 Engine-speed control lever, 2 Ball pin, 3 KSB advance lever, 4 Stop.



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 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. 16: Effect of the hydraulic cold-start accelerator (KSB)
1 Injection-timing advance.

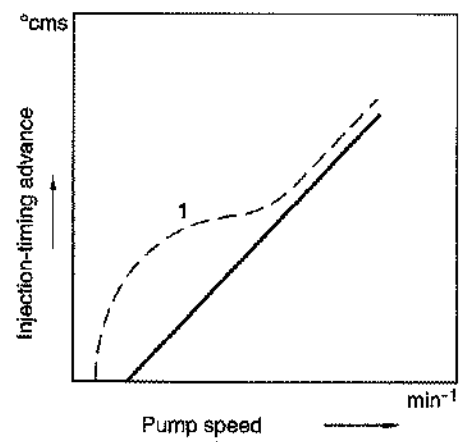
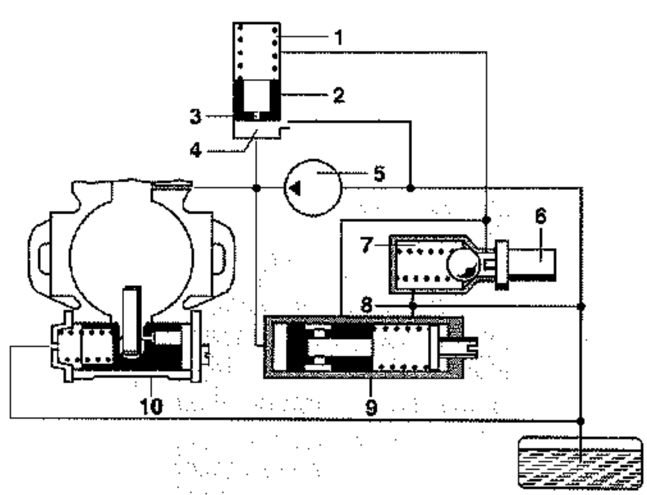


Fig. 15: Hydraulic cold-start accelerator (KSB)
1 Pressure-control valve,
2 Valve plunger,
3 Restriction passage,
4 Internal pressure
5 Fuel-supply pump,
6 Electrically heated expansion element,
7 KSB ball valve,
8 Pressureless fuel return,
9 KSB control valve, adjustable,
10 Timing device.



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 injector 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: Electrical shutoff device (pull solenoid)
1 Inlet passage, 2 Distributor plunger, 3 Distributor head, 4 Push or pull solenoid, 5 High-pressure chamber.

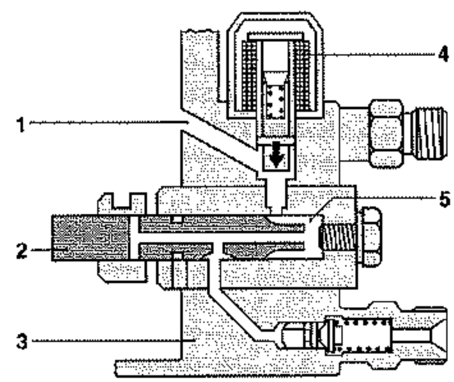
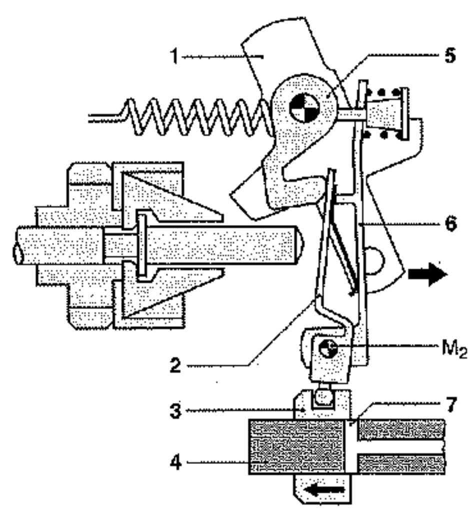


Fig. 18: Mechanical shutoff device
1 Outer stop lever, 2 Start lever, 3 Control collar, 4 Distributor plunger, 5 Inner stop lever, 6 Tensioning lever, 7 Cutoff port, M_2 Pivot point for 2 and 6.

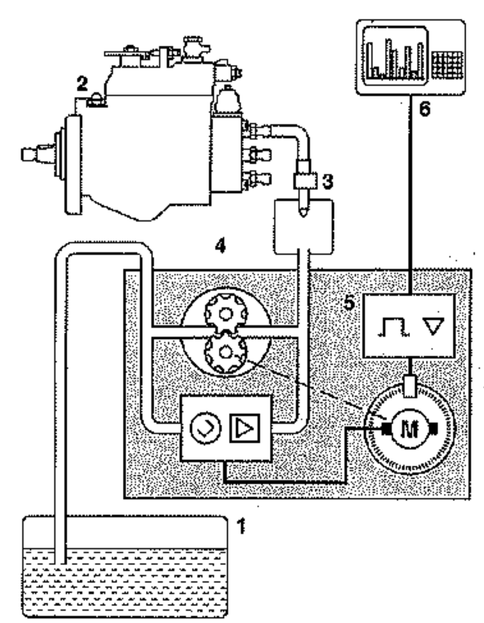


Testing and calibration

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. 1: Continuous injected fuel-quantity measuring system
1 Calibrating-oil tank, 2 Injection pump,
3 Calibrating nozzle, 4 Measuring cell,
5 Pulse counter, 6 Display monitor.



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.

Electronic Diesel Control (EDC)

Application

The development of the automotive diesel engine is governed primarily by requirements for clean exhaust, improved fuel economy, and the optimization of driveability. These stipulations are placing increasingly stringent demands upon the fuel-injection system, namely:
– sensitive controls,
– ability to process additional parameters,
– tighter tolerances and increased accuracy even over very long periods of operation.

These demands are fulfilled by the Electronic Diesel Control (EDC). This system provides for electronic measurement, as well as flexible data processing, and closed control loops with electrical actuators. In comparison to the conventional mechanical governor therefore, EDC implements new and improved control functions.

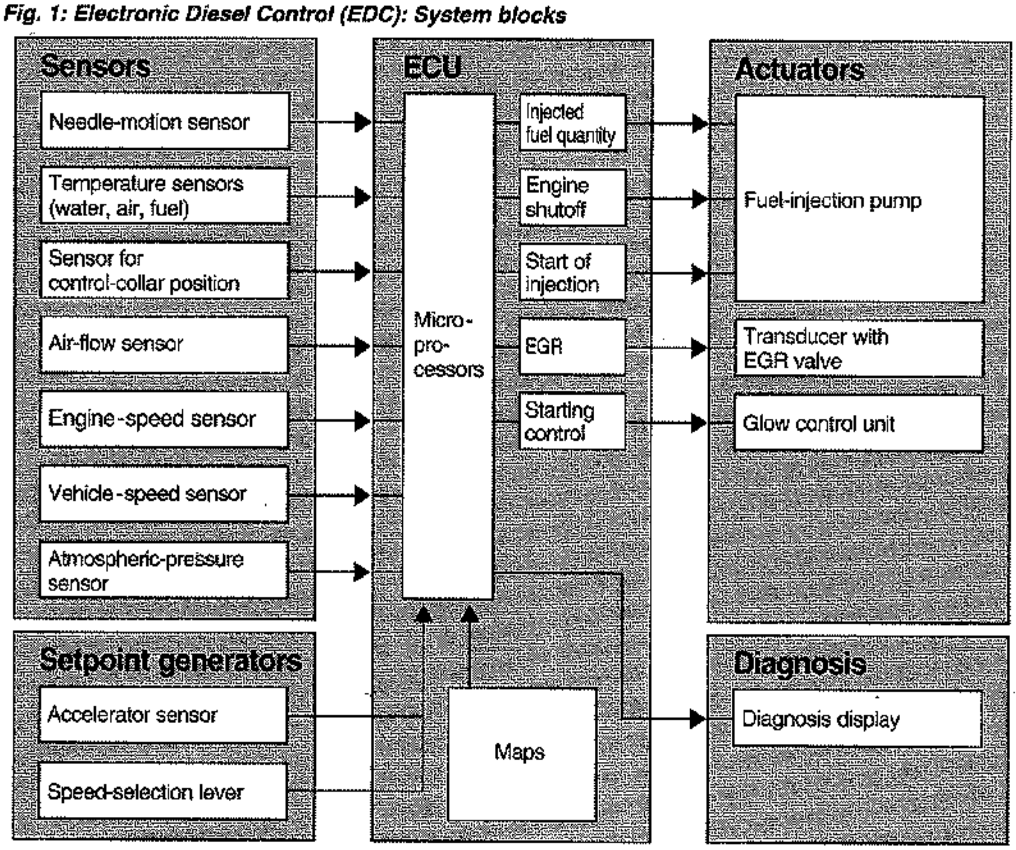
In the diesel engine, operating characteristics and combustion are influenced by:
– Injected fuel quantity,
– Start of injection,
– Exhaust-gas recirculation (EGR)
– Charge-air pressure.

These controlled variables must be optimally adjusted for every working mode in order ensure efficient diesel engine operation. To this end, EDC incorporates automatic control loops for the main parameters.

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.

Fig. 1: Electronic Diesel Control (EDC): System blocks



2. Electronic control unit (ECU) with microprocessors which processes the information 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 either contacting or non-contacting methods. 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.

Fig. 2: Sensor signals
1 Untreated signal from the needle-movement sensor (NBF),
2 Signal derived from the NBF signal,
3 Untreated signal from the engine-speed signal,
4 Signal derived from untreated engine-speed signal,
5 Evaluated start-of-injection signal.

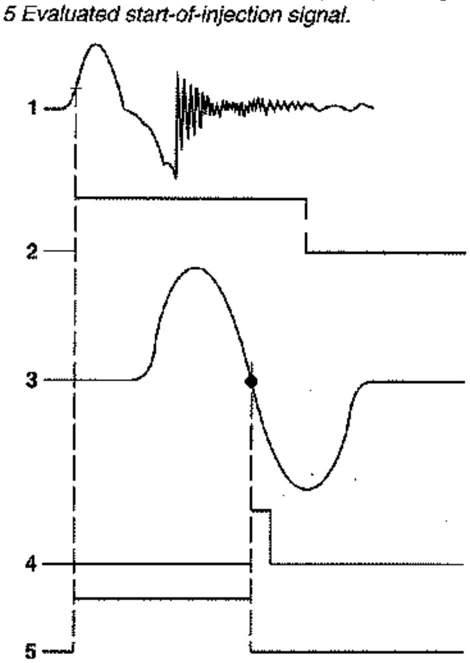
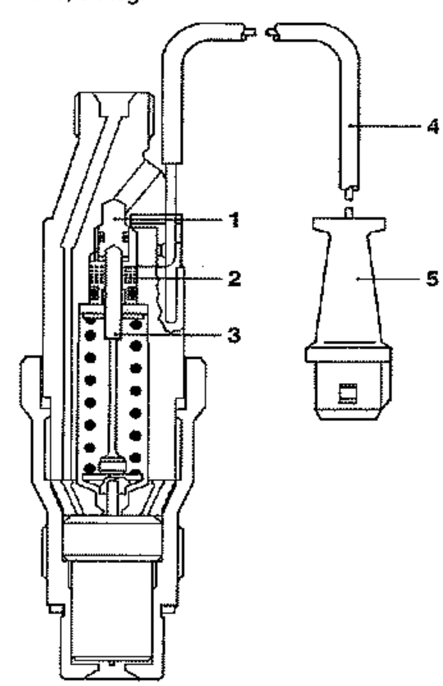


***Fig. 3: Nozzle-and-holder assembly with needle-movement sensor** (NBF)*
1 Setting pin,
2 Sensor winding, 3 Pressure pin,
4 Cable, 5 Plug.



Solenoid actuator for injected fuel quantity control

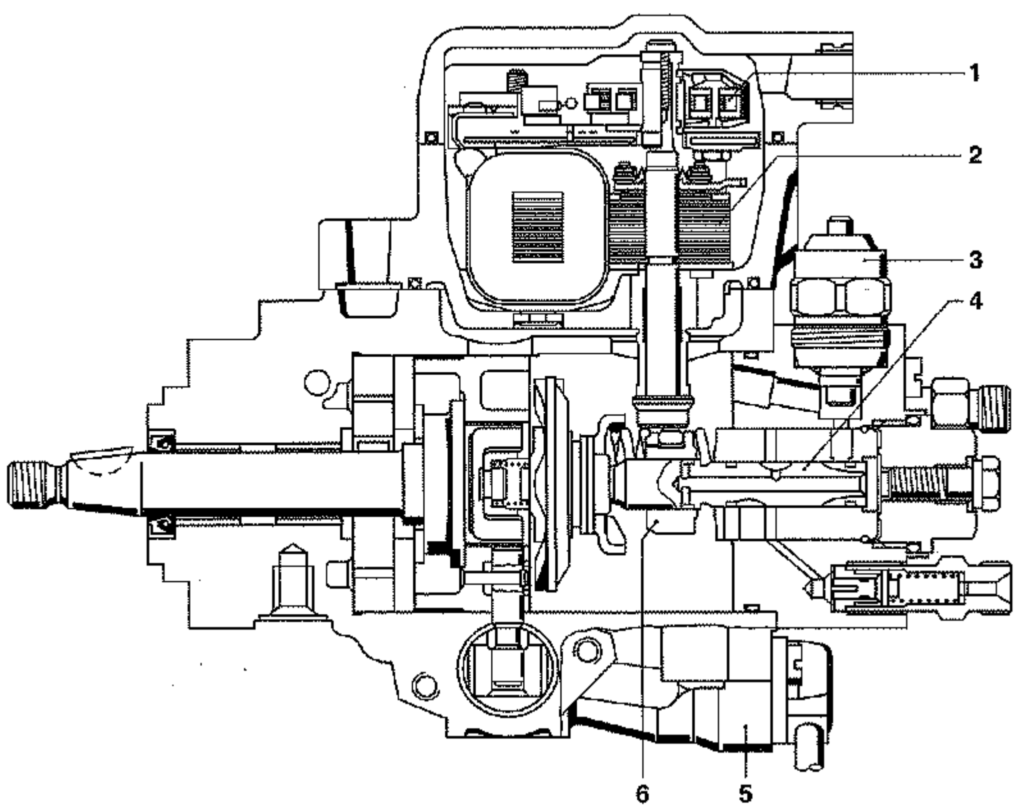
The solenoid actuator (rotary actuator) engages with the control collar through a shaft (Fig. 4). Similar to the mechanically governed fuel-injection pump, the cutoff ports are opened or closed depending upon the control collar's position. The injected fuel quantity can be infinitely varied between zero 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.

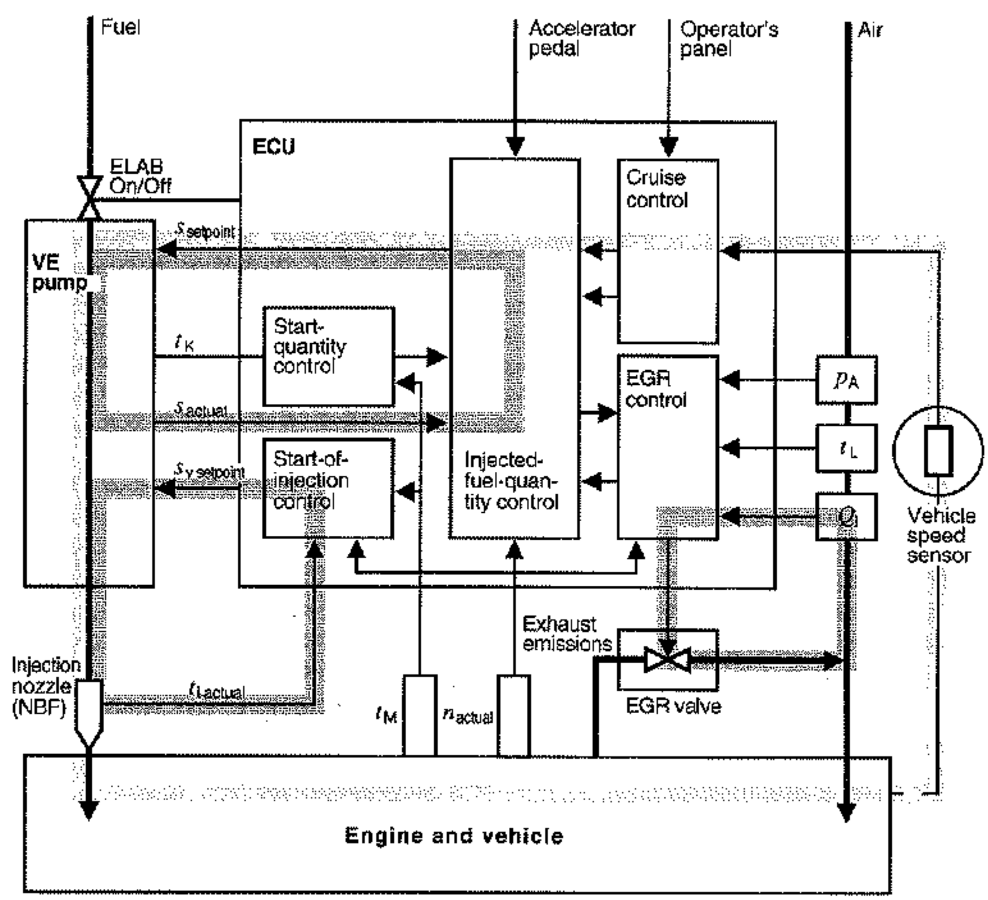
Taking into account the stored map data, and the actual input values from the sensors, a setpoint is calculated for the setting of the rotary actuator in the pump. This rotary actuator is equipped with a check-back signalling unit and ensures that the control collar is correctly set.

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.

Fig. 5: Closed control loop of the electronic diesel control (EDC)

Q Air-flow quantity, n_{act} Engine speed (actual), p_A Atmospheric pressure, s_{set} Control-collar signal (setpoint), s_{act} Control-collar position (actual), $s_{v\ set}$ Timing-device signal (setpoint), t_K Fuel temperature, t_L Intake-air temperature, t_M Engine temperature, $t_{i\ act}$ Start of injection (actual).



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 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.

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	●	●
Injected fuel quantity actuator	Permanent control deviation	Engine switchoff	●	●

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 accomodated, 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.

Peripheral equipment for diesel fuel-injection systems

Nozzles and nozzle holders

Assignments

In the diesel-engine's fuel-injection system, the nozzles in their respective nozzle holders are an important link between injection pump and engine. Their assignments are:

– meter the injected fuel,
– manage and prepare the fuel spray ,
– define the rate-of-discharge curve, and
– seal-off the injection system from the combustion chamber.

Diesel fuel is injected at high pressures, with peak pressures as high as 1200 bar, which in future will be even higher. At such high pressures, the diesel fuel no longer behaves like a rigid liquid but becomes compressible. During the brief delivery period (approx. 1ms), the high pressure causes the injection system to "expand" at certain points, whereby the nozzle cross-section defines the quantity of fuel which is injected into the combustion chamber.
The nozzle's spray-hole length and diameter, and (to a limited extent) its spray-orifice shape, have an influence upon fuel-spray management and, as a result, upon the engine's output power, its fuel consumption and its exhaust emissions.
Within certain limits, the rate-of-discharge curve can be tailored to requirements by "correct" control of the nozzle fuel-flow section (as a function of needle lift) and by controlling the nozzle-needle motion. And finally, the nozzle must be able to seal off the fuel-injection system against the hot, highly compressed gases from the combustion chamber (up to approx. 1000 °C). In order to avoid blow-back of these gases when the injection nozzle opens, the pressure in the nozzle's pressure chamber must always be higher than the combustion pressure. This requirement is particularly difficult to comply with at the end of the injection process (when injection pressure has already dropped while combustion pressure is increasing rapidly), and it requires careful matching of the injection pump, the nozzle, and the pressure spring.

Designs

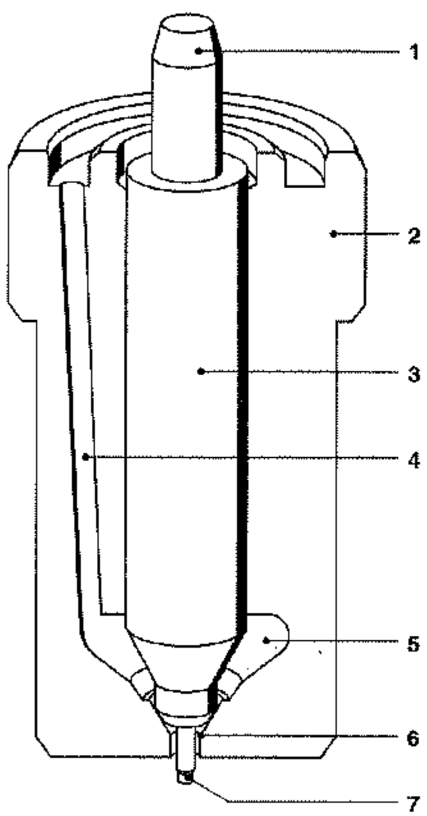
According to whether they have a divided combustion chamber (prechamber, turbulence, or swirl-chamber engine) or a non-divided combustion chamber (direct-injection (DI)), each design requires its own special nozzle.
The throttling pintle nozzle is used on prechamber, turbulence or swirl-chamber engines with divided combustion chamber. This nozzle injects a coaxially shaped jet of fuel and the needle normally opens inwards. On the other hand, hole-type nozzles are used for direct-injection (DI) engines with non-divided combustion chambers.

Throttling pintle nozzles
The standard nozzle-and-holder assembly for prechamber and turbulence (swirl) chamber engines is composed of the injection nozzle (Type DN..SD..) together with the nozzle holder (Type KCA with screw-in thread). The normal version of this nozzle holder has an M24x2 screw-in thread and an A/F dimension of 27 mm. Usually, DN O SD.. injection nozzles are used which have a needle diameter of 6 mm and a spray angle of 0° (pencil

spray). It is much rarer to find an injector nozzle with a defined spray angle (e.g., 12° in the DN 12 SD..). For restricted cylinder-head space, more compact nozzle-holder versions (e.g., KCE) are available.

One of the characteristic features of the throttling pintle nozzle is the control of its discharge cross-section, in other words the throughflow quantity, as a direct function of the needle lift. Whereas in the case of the hole-type nozzle the cross-section increases sharply as soon as the needle opens, the throttling pintle nozzle features a very flat cross-section characteristic in the range of small needle strokes. In this range, the throttling pintle, a pin-shaped extension of the nozzle needle, remains inside the spray hole and only the small ring-shaped area between the spray hole and the pintle remains available as the flow cross-section. When large needle strokes take place, the pintle lifts out of the spray hole completely and the flow cross section increases rapidly (Fig. 1).

Fig. 1: Throttling pintle nozzle
1 Pressure pin, 2 Nozzle body,
3 Nozzle needle, 4 Inlet passage,
5 Pressure chamber, 6 Spray hole,
7 Pintle



To a certain degree, this change in cross section as a function of needle stroke controls the rate-of-injection curve, in other words the injected fuel quantity per unit of time. At the start of injection only a small quantity of fuel can leave the nozzle, while a large quantity emerges at the end of the injection process. Above all, this characteristic has a positive effect on engine combustion noise.

It must be noted that if the cross-section values are too small there is insufficient needle lift, the injection pump accelerates the needle in the "open" direction more quickly than would otherwise be the case, and the pintle leaves the spray hole sooner thus terminating the throttling action more quickly. The injected fuel quantity per unit of time climbs rapidly as a result and combustion noise increases. Excessively small cross sections at the end of injection have a similar negative effect because when the needle closes again the displaced fuel has difficulty in leaving through the restricted cross section, with the attendant end-of-injection delay. It is therefore important to match the cross-section characteristic to the the rate-of-discharge curve and to the particular combustion process.

For the spray holes, appropriate manufacturing processes must be applied in order to comply with the close-tolerance dimensions concerned.

During operation, the throttling gap cokes up rather heavily and very unevenly. The degree of coking is determined by the fuel quality and by the engine's operating mode. Only about 30% of the original fuel-flow section remains free of coke.

The so-called flat-pintle injection nozzle, is a special version of the throttling pintle nozzle. The ring gap between the nozzle's spray hole and its throttling pintle is practically zero and apart from coking up less than the throttling pintle nozzles, coking is more evenly distributed (Fig. 2). The pintle of this injection nozzle is provided with a ground surface which opens the fuel-flow section when the needle is lifted (Fig. 2; 2a and 2b). A flow channel is then generated whose total surface, referred to the fuel-flow section, is less whereby the self-cleaning effect is greater. The pintle's ground surface is often parallel to the nozzle needle's axis. If the ground angle is increased, the flat section of the throughflow curve rises faster and this results in a gentler transition to the fully open state. This has a positive effect on the vehicle's part-load noise and upon its driveability (Fig. 2). Because temperatures at the nozzles above 220°C also result in pronounced coking, thermal protection plates and caps are available which dissipate the combustion-chamber heat away from the nozzles and into the cylinder head.

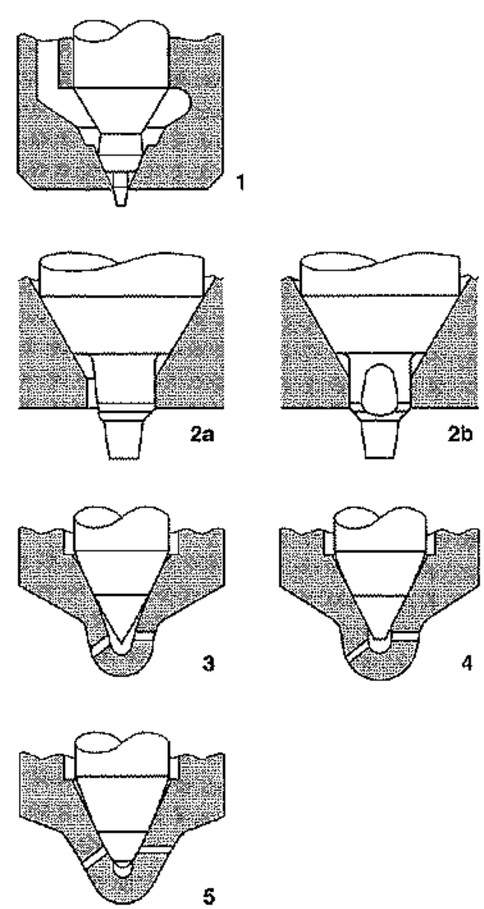
Hole-type nozzles

There are a very wide variety of different nozzle-and-holder assemblies for hole-type nozzles on the market. In contrast to the throttling-pintle nozzles, the hole-type nozzles must be installed in a given position. The spray holes are at different angles in the nozzle body and must be correctly aligned with regard to the combustion chamber. The nozzle and holder assembly is therefore fastened to the cylinder head with hollow screws or claws. A special mount is used to lock the nozzle in the correct position.

The hole-type nozzles (Figs. 2 and 3) have needle diameters of 4 mm (Size P) and from between 5 and 6 mm (Size S). The seat-hole nozzle is only available as a Size P version. The nozzle pressure springs must be matched to the needle diameters and to the high opening pressures which are usually above 180 bar. The nozzle-sealing function is particularly important at the end of injection because there is a risk of the combustion gases blowing back into the nozzle and in the long run destroying it and causing hydraulic instability. Precision matching of the pressure spring and the needle diameter ensures efficient sealing. In certain cases, it may be necessary to take into account the oscillations of the pressure spring. There are three designs for the arrangement of the spray holes in the nozzle cone (Fig. 2). These three designs also differ from each other with respect to the amount of fuel which remains inside the injector and which can evaporate into the combustion chamber when injection has finished. Versions with cylindrical blind hole, conical blind hole, and seat hole, have decreasing fuel quantities in this order.

Furthermore, the less fuel that can evaporate from the nozzle, the lower are the engine's hydrocarbom emissions. The levels of these emissions therefore also correspond to the (nozzle) order given above.

Fig. 2: Nozzle shapes
1 Throttling pintle nozzle,
2 Throttling pintle nozzle with flat-cut pintle,
2a Side view, 2b Front view,
3 Hole-type nozzle with conical blind hole,
4 Hole-type nozzle with cylindrical blind hole,
5 Seat-hole nozzle.



The nozzle cone's mechanical integrity is the limiting factor in the length of the spray hole. At present, spray-hole length is 0.6 ... 0.8 mm in the case of the cylindrical and conical blind holes. With the seat-hole nozzle, the minimum spray-hole length is 1 mm, whereby special hole-making techniques must be applied.

Developments are proceeding towards shorter hole length, because as a rule the shorter the hole, the better the engine's smoke values. In the case of hole-type nozzles, when the spray hole is bored this results in throughflow tolerances of ±3.5%. If rounding is also carried out (hydro-erosive machining), the throughflow tolerances can be reduced to ±2%. Due to the thermal stability of the materials used in the hole-type nozzle, its upper temperature limit is in the area of 270°C. Particularly difficult applications can necessitate the use of thermal protection sleeves, and cooled injection nozzles are available for large engines

Standard nozzle holders

Figs. 4 and 5 show the basic design of two nozzle-and-holder assemblies each consisting of an injection nozzle and nozzle holder. The nozzle itself comprises the nozzle body and the nozzle needle which moves freely within the nozzle body's guide bore, while at the

Fig. 3: Hole-type nozzle
1 Pressure pin, 2 Nozzle body, 3 Nozzle needle, 4 Inlet passage, 5 Pressure chamber, 6 Spray hole, 7 Blind hole, δ Spray-hole cone angle.

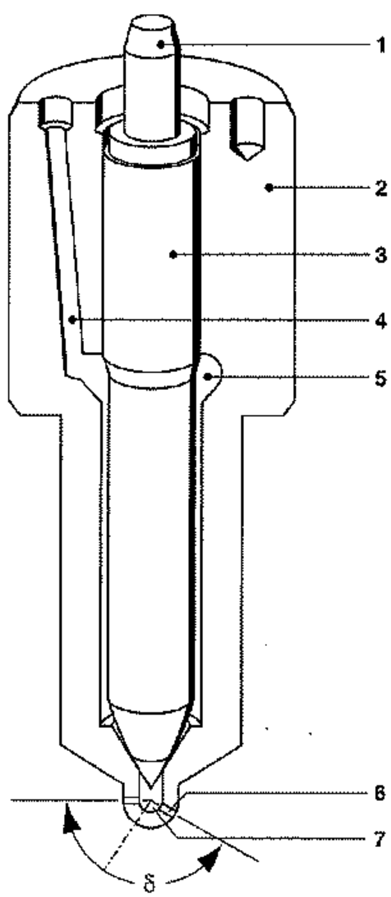
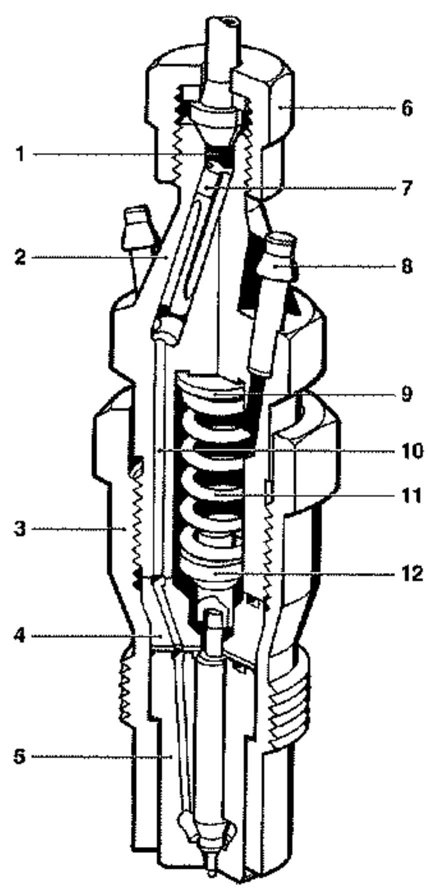


Fig. 4: Nozzle-and-holder assembly
With throttling pintle nozzle.
1 Inlet, 2 Nozzle-holder body, 3 Nozzle-retaining nut, 4 Intermediate element, 5 Injection nozzle, 6 Union nut with fuel-injection tubing, 7 Edge filter, 8 Leak-off connection, 9 Pressure-adjusting shims, 10 Pressure passage, 11 Pressure spring, 12 Pressure pin.



same time sealing against high injection pressures. At the combustion-chamber end, the nozzle needle has a sealing cone which is forced by the pressure spring against the nozzle body's conical sealing surface when the nozzle is closed. The sealing cone's opening angle is slightly different to that of the nozzle body's sealing cone, with the result that linear contact with high compression and good sealing properties is formed between them.

The needle-guide diameter is larger than the seat diameter. The injection pump's hydraulic pressure is applied to the differential area between the needle cross section and the area covered by the seat. As soon as the product of sealing surface and pressure exceeds the force of the pressure spring, the nozzle opens. Since the area of the nozzle which is subjected to pressure increases abruptly by the area of the seat as soon as the needle starts to lift, the injection nozzle "snaps" open very quickly provided the injection pump's delivery rate is sufficient. The nozzle closes again only when the pressure falls below the closing pressure (which is lower than the opening pressure). In the design of injection systems, this hysteresis effect is particularly important with regard to hydraulic stability.

The opening pressure of a nozzle-and-holder assembly (approx. 110 ... 140 bar for the throttling pintle nozzle, and 150 ... 250 bar for the hole-type nozzle) is adjusted by inserting shims underneath the pressure spring.

The closing pressure then results from the injection-nozzle geometry, that is, the ratio of needle-guide diameter to seat diameter, the so-called pressure stage.

2-spring nozzle-holder

These are mainly used with direct-injection (DI) engines, in which pilot injection is the most important measure for minimizing the combustion noise.

Since pilot injection ensures a relatively gentle pressure rise, engine idle is both quiet and stable, and combustion noises are reduced.

The 2-spring nozzle holder (Fig. 6) achieves this effect by improving the shape of the rate-of-discharge curve.

The improvement is the result of the adjustment and matching of
- opening pressure 1,
- opening pressure 2,
- prestroke, and
- total needle lift.

Opening-pressure adjustment is the same as for the single-spring nozzle holder. Opening pressure 2 is given by the pretension of spring 1 together with that of spring 2 which is supported on a stop sleeve into which the prestroke dimension has been machined. When injection takes place, the nozzle needle opens at first in the prestroke range, whereby common lifts in this range are 0.03 ... 0.06 mm. The further rise of the pressure in the nozzle causes the stop sleeve to lift and the nozzle needle can open fully.

There are also special nozzles available for the 2-spring nozzle holder in which the nozzle needle has no pintle and the needle shoulder is level with the end of the nozzle body.

To put it more simply: The springs of the 2-spring nozzle holder are so calibrated that first of all only a small amount of fuel is injected into the combustion chamber with a resulting slight rise in combustion pressure. This extends the duration of injection and together with the following increased residual fuel quantity leads to the desired softer combustion.

There are also 2-spring nozzle holders available for prechamber and turbulence-chamber engines. Their settings are matched to the injection system in question. The various opening pressures are around 130/180 bar and the prestrokes approx. 0.1 mm.

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

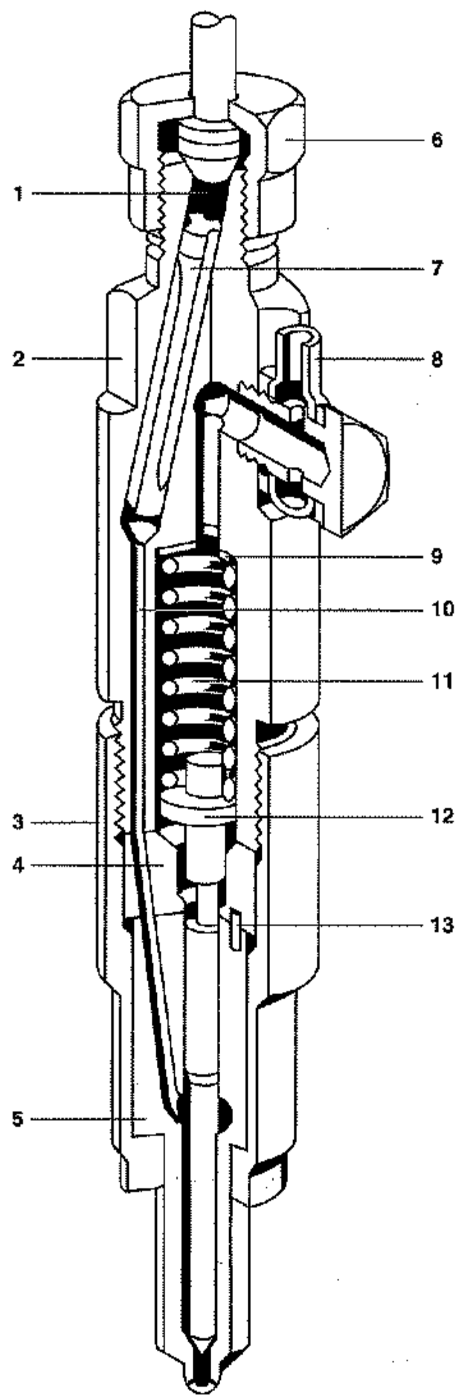
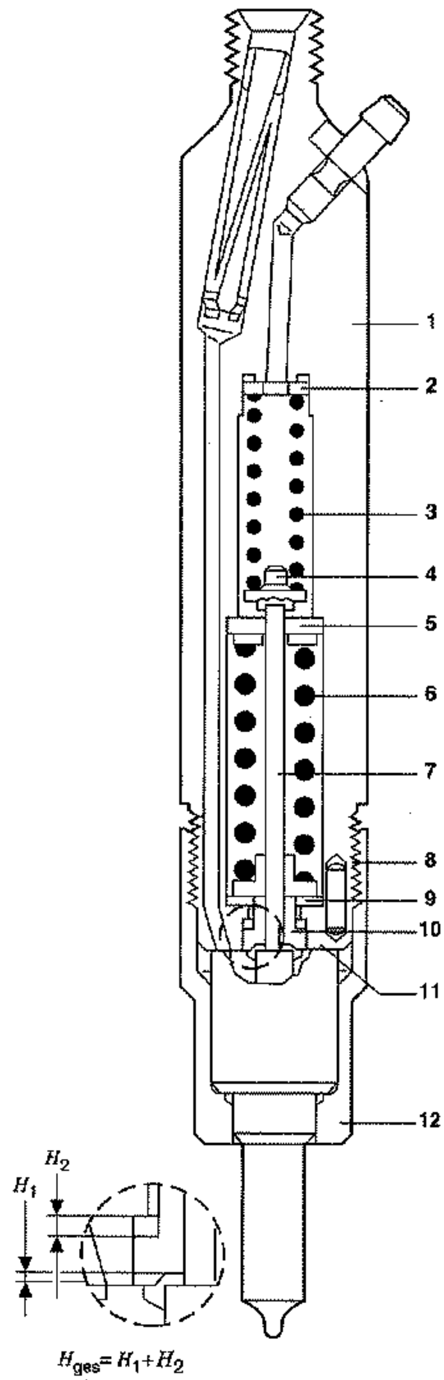


Fig. 6: 2-spring nozzle holder KBEL..P...
H_1 Prestroke, H_2 Main needle stroke, $H_{tot} = H_1 + H_2$ Total stroke.
1 Nozzle-holder body, 2 Shim, 3 Pressure spring 1, 4 Pressure pin, 5 Guide element, 6 Pressure spring 2, 7 Pressure pin, 8 Spring seat, 9 Shim, 10 Stop sleeve, 11 Intermediate element,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



Peripheral equipment for diesel fuel-injection systems

Start-assist systems for diesel engines

Being as leakage and heat losses reduce the pressure and temperature of the air-fuel mixture at the end of the compression stroke, the colder the diesel engine, the more difficult it is to start. These facts make it all the more important that starting aids are used. The starting limit temperature depends upon the engine type. Prechamber and turbulence-chamber engines are equipped with a glow plug in their secondary combustion chambers which acts as a "hot spot" (Fig. 1). In the case of direct-injection (DI) engines, this hot spot is shifted to the combustion-chamber periphery. The large DI engines fitted in trucks operate with either air preheating in the intake manifold (flame starting), or with a special, easily ignited fuel (Startpilot) which is injected into the intake air. Today glow plugs are used almost without exception.

Sheathed-element glow plug

The sheathed-element glow plug is in the form of a tubular heating element. This comprises a non-corroding glow tube inside which the glow filament is embedded in magnesium-oxide powder so as to be vibration-proof (Fig. 2). Both in the conventional glow plug (Type S-RSK), and in the newer-generation version (Type GSK 2) this glow filament has a heating filament at its tip (Fig. 2). Compared to the conventional glow plug, this newer version reaches the ignition temperature more quickly as well as having a lower steady-state temperature (Fig. 3). This means that even after the engine has started, this glow plug can be left switched on for up to 3 minutes, thus contributing to lower exhaust emissions and reduced noise. A control filament with PTC characteristic is connected in series with the

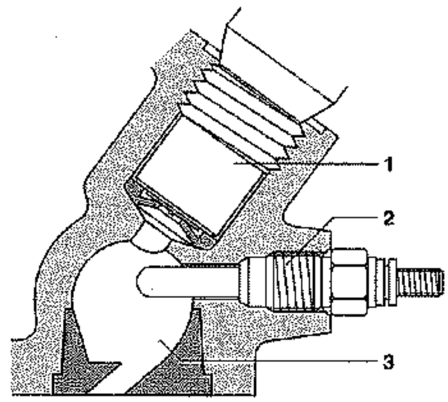


Fig. 1: Sheathed-element glow plug
Configuration in the turbulence chamber.
1 Injection nozzle, 2 Sheathed-element glow plug, 3 Turbulence chamber.

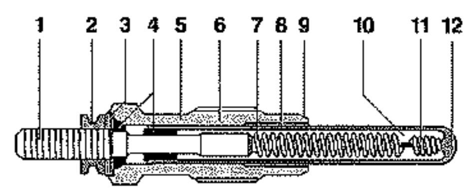


Fig. 2: Sheathed-element glow plug
1 Terminal stud, 2 Round nut, 3 Insulating washer, 4 Seal, 5 Glow-plug shell, 6 Screw-in thread, 7 Control filament, 8 Ring gap, 9 Conical seat, 10 Insulating powder, 11 Heating filament, 12 Glow tube.

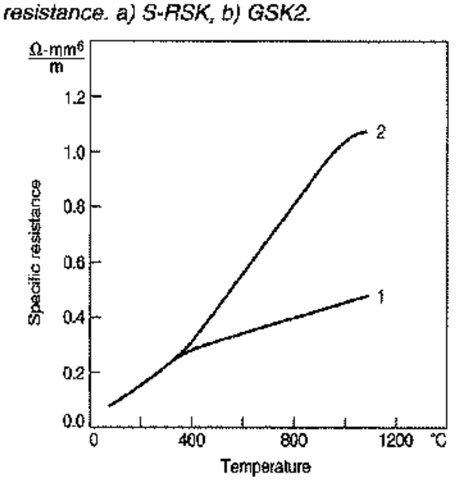


Fig. 3: Sheathed-element glow plugs
Diagram of temperature as a function of specific resistance. a) S-RSK, b) GSK2.

heating filament. The PTC characteristic means that the control filament's electrical resistance increases along with temperature and limits the glow plug's temperature to a value which is uncritical for the glow-tube material. Appropriate matching of control-filament and heating-filament resistances enables the starting temperature to be reached within 3 ... 10 secs. The heating element (control filament + heating filament) is pressed into the glow-plug housing so as to be gas-tight. The electrical connection for the single-pole glow-plug version is in parallel.

Flame glow plug

The flame plug heats the intake air by burning fuel. Normally, the injection system's fuel-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 amount of fuel appropriate to the particular engine.
This fuel then evaporates in an evaporator tube surrounding the glow plug and mixes with the intake air. The resulting mixture ignites on the 1000 °C glow element at the flame-plug tip.

Glow control unit

For the triggering of the glow plugs, the glow control unit 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. The control inputs to the ECU are in the form of multiple plugs, and in order to avoid voltage drops the glow plugs are provided with current through appropriate threaded bolts or plugs.

Functional sequence

The diesel engine's glow plug and starter switch functions in a similar manner to the ignition and starting switch used in the spark ignition engine. Switching to the "Ignition on" position starts the preheating process and the glow-plug indicator lamp lights up. As soon as this goes out again, this indicates that the glow plugs are hot enough to start the engine and cranking can begin. In the subsequent 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. 4).
In the warm-up phase following a successful engine 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, preheating 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. 4: Typical preheating sequence
1 Glow-plug and starter switch, 2 Starter, 3 Glow-plug indicator lamp, 4 Load switch, 5 Glow plugs.
t_V Preheating time, t_S Ready to start, t_N Post-heating time.