**The combustion processes that take place inside a diesel engine are essentially dependent on the way in which the fuel is injected into the combustion chamber. The most important criteria are the timing and the duration of injection, the degree of atomization and the distribution of the fuel inside the combustion chamber, the timing of ignition, the mass of the fuel injected relative to crankshaft rotation, and the total amount of fuel injected relative to engine load. In order that a diesel engine and its fuel-injection system function properly, all of these variable factors must be carefully balanced.**

The design of the fuel-injection system must be precisely matched to the engine concerned and its application. As a variety of factors have to be taken into account, some of which are in conflict with one another, the final design can only ever be a compromise.

The composition and conditioning of the air/fuel mixture has a fundamental effect on an engine's specific fuel consumption, torque (and therefore power output), exhaust-gas composition and combustion noise. The quality and effectiveness of the mixture formation is largely attributable to the fuel-injection system.

A number of fuel-injection variables affect mixture formation and the course of combustion inside the combustion chamber and, therefore, the engine's emission levels and power output/efficiency. They are:

- Start of injection
- Injection characteristics (injection duration and rate-of-discharge curve)
- Injection pressure
- Injection direction and
- The number of injection jets

The injection mass and the engine speed are operating parameters that determine the engine power output.

## Mixture distribution

### Excess-air factor λ

The excess-air factor λ was devised in order to indicate the degree to which the actual air/fuel mixture achieved in reality diverges from the theoretical (stoichiometric [1])) mass ratio. It indicates the ratio of intake air mass to required air mass for stoichiometric combustion, thus:

$$\lambda = \frac{\textit{Air mass}}{\textit{Fuel mass} \cdot \textit{Stoichiometric ratio}}$$

$\lambda = 1$: The intake air mass is equal to the air mass theoretically required to burn all of the fuel injected.

$\lambda < 1$: The intake air mass is less than the amount required and therefore the mixture is rich.

$\lambda < 1$: The intake air mass is greater than the amount required and therefore the mixture is lean.

[1]) The stoichiometric ratio indicates the air mass in kg required to completely burn 1 kg of fuel ($m_L/m_K$). For diesel fuel, this is approx. 14.5.

1 Progress of combustion in a direct-injection test engine with a multihole nozzle

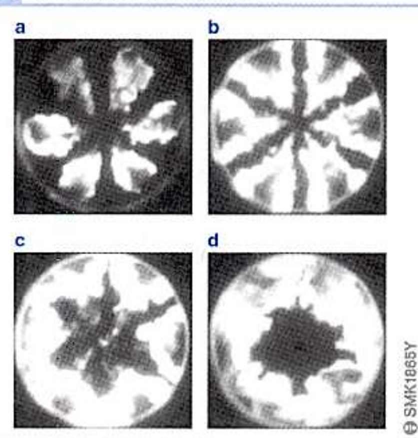



**Fig. 1**
Special engines with glass inserts and mirrors allow the fuel injection and combustion processes to be observed.

The times are measured from the start of spontaneous combustion.

a 200 µs
b 400 µs
c 522 µs
d 1,200 µs

## Lambda levels in diesel engines

Rich areas of mixture are responsible for sooty combustion. In order to prevent the formation of too many rich areas of mixture, diesel engines – in contrast to gasoline engines – have to be run with an overall excess of air.

The lambda levels for turbocharged diesel engines at full load are between $\lambda = 1.15$ and $\lambda = 2.0$. When idling and under no-load conditions, those figures rise to $\lambda > 10$.

Those excess-air factor figures represent the total masses of fuel and air in the cylinder. However, spontaneous ignition and pollutant formation are determined essentially by localized lambda levels.

Diesel engines operate with heterogeneous mixture formation and auto-ignition. It is not possible to achieve completely homogeneous mixing of the injected fuel with the air charge prior to or during combustion. Auto-ignition occurs a few degrees of crankshaft rotation after the point at which fuel injection starts (ignition lag).

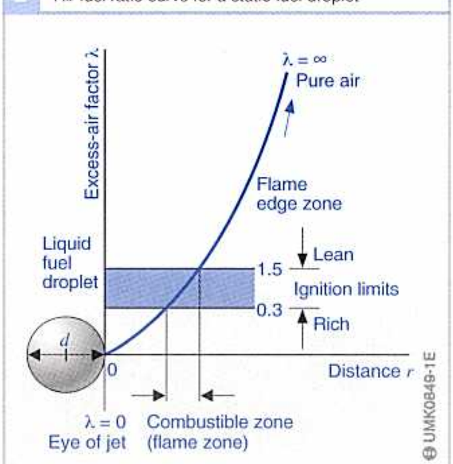
Within the heterogeneous mixture encountered in a diesel engine, the localized excess-air factors can cover the entire range from $\lambda = 0$ (pure fuel) in the eye of the jet close to the injector to $\lambda = \infty$ (pure air) at the outer extremities of the spray jet. Closer examination of a single droplet of liquid fuel reveals that around the outer zone of the droplet (vapor envelope), localized, combustible lambda levels of 0.3...1.5 occur (Figures 2 and 3). From this, it can be deduced that good atomization (large numbers of very small droplets), high levels of excess air and "moderate" motion of the air charge produce large numbers of localized zones with lean combustible lambda levels. The effect of this is that less soot and, in principle, less $NO_X$ is produced during combustion.

Good atomization is achieved by high injection pressures (the highest currently used is over 2,000 bar). This results is a high relative velocity between the jet of fuel and the air in the cylinder which has the effect of scattering the fuel jet.

With a view to reducing engine weight and cost, the aim is to obtain as much power as possible from a given engine capacity. To achieve that aim, the engine "must" be run with a "small" air excess at high loads. But small air excesses increase emission levels. Therefore, they have to be limited, i.e. the fuel volume delivered must be precisely proportioned to match the available amount of air and the speed of the engine.

Low atmospheric pressures (e.g. at high altitudes) also require the fuel volume to be adjusted to the smaller amount of available air.

2 Air-fuel ratio curve for a static fuel droplet




3 Air-fuel ratio patterns for a moving fuel droplet

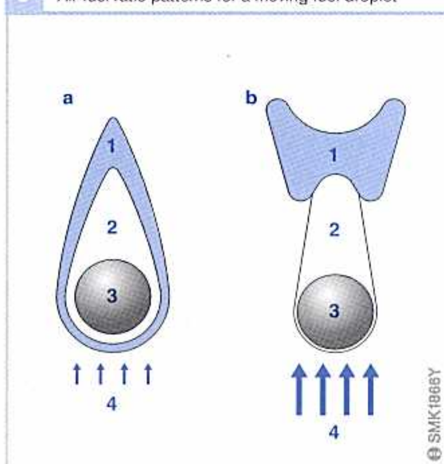



**Fig. 2**
*d* Droplet diameter (approx. 2...20 μm)

**Fig. 3**
a Low relative velocity
b High relative velocity

1 Flame zone
2 Vapor envelope
3 Fuel droplet
4 Air flow

## Start of injection and delivery

### Start of injection

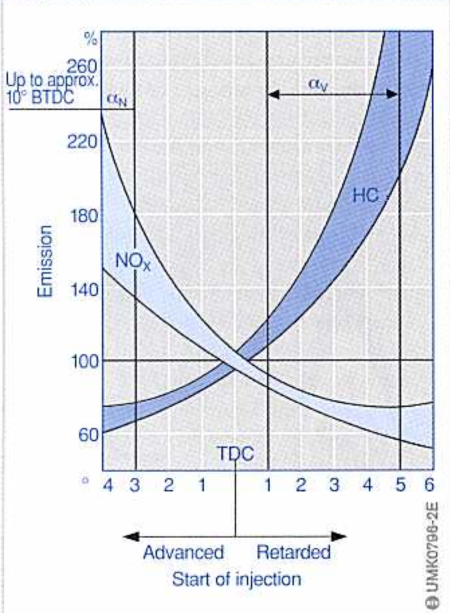
The point at which injection of fuel into the combustion chamber starts has a decisive effect on the point at which combustion of the air/fuel mixture starts, and therefore on emission levels, fuel consumption and combustion noise. Consequently, injection timing plays a major role in optimizing engine performance characteristics.

The point at which injection of fuel starts is the position stated in degrees of crankshaft rotation relative to crankshaft top dead center (TDC) at which the nozzle opens and fuel starts to enter the combustion chamber.

The position of the piston relative to top dead center at that moment (as well as the shape of the intake port), determines the nature of the air flow inside the combustion chamber, and the density and temperature of the air. Accordingly, the degree of mixing of air and fuel is also dependent on start of injection. Thus, start of injection affects emissions such as soot, a product of incomplete combustion, nitrogen oxides ($NO_X$), unburned hydrocarbons (HC) and carbon monoxide (CO).

The start of injection requirements differ according to engine load (Figure 1). This fact demands load-dependent adjustment of the start of injection. The characteristic operating data of each engine is thus determined and stored electronically in the form of an engine data map. The engine data map plots the required start of injection points against engine load, speed and temperature. It also takes account of fuel-consumption considerations, pollutant-emission requirements and noise levels at any given power output (Figure 2).

1 Distribution patterns for $NO_X$ and HC emissions plotted against start of injection for a commercial vehicle without exhaust-gas recirculation




2 Start of injection versus engine speed and load for a car engine started from cold and at normal operating temperature (example)

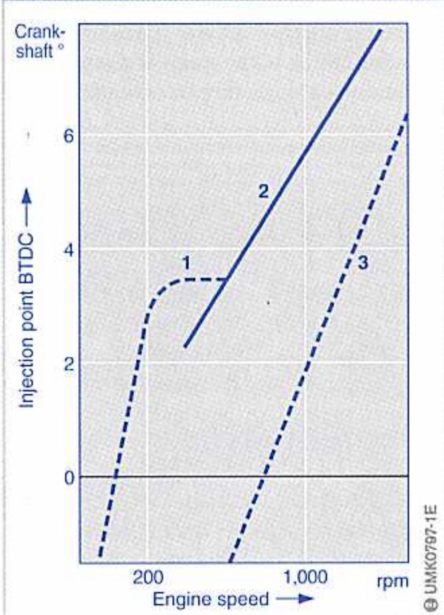



**Fig. 1**
Example of an application:
$\alpha_N$ Optimum start of injection for emissions at no load, as $NO_X$ emissions are lower under those conditions
$\alpha_V$ Optimum start of injection for emissions at full load, as HC emissions are lower under those conditions

**Fig. 2**
1 Cold start (<0 °C)
2 Full load
3 Medium load

### Guide figures (EURO III)

On a diesel engine's data map, the optimum points of combustion start for low fuel consumption are in the range of 0...8° crankshaft before TDC. On that basis and based on the statutory exhaust-gas emission limits, the start of injection points are as follows:

Direct-injection car engines:

- No load: 2° crankshaft before TDC to 4° crankshaft after TDC
- Part load: 6° crankshaft before TDC to 4° crankshaft after TDC
- Full load: 6...15° crankshaft before TDC

Direct-injection commercial-vehicle engines (without exhaust-gas recirculation):

- No load: 4...12° crankshaft before TDC
- Full load: 3...6° crankshaft before TDC to 2° crankshaft after TDC

When the engine is cold, the start of injection for car and commercial-vehicle engines is 3...10° earlier. The duration of combustion at full load is 40...60° of crankshaft rotation.

### Advanced start of injection

The highest final compression temperature is reached at TDC. If combustion is initiated a long way before TDC, the combustion pressure rises steeply and acts as a retarding force against the movement of the piston. The heat lost in the process diminishes the efficiency of the engine and therefore increases its fuel consumption. The steep increase in compression pressure also makes combustion much noisier.

An advanced start of injection increases the temperature in the combustion chamber. As a result, the $NO_X$ emission levels rise while HC emissions are lower (Figure 1).

### Retarded start of injection

A retarded start of injection under no-load conditions can result in incomplete combustion and therefore in the emission of unburned hydrocarbons (HC) since combustion takes place at a time when the temperature in the combustion chamber is dropping (Fig. 1).

The partially conflicting interdependence of specific fuel consumption and hydrocarbon emission levels on the one hand, and soot (black smoke) and $NO_X$ emissions on the other, demand a trade-off combined with very tight tolerances when modifying the start of injection to suit a particular engine.

Minimizing blue and white smoke levels requires advanced start of injection and/or pre-injection when the engine is cold.

In order to keep noise and pollutant emissions at acceptable levels, a different start of injection is frequently necessary when the engine is running at part load than when it is at full power. The start-of-injection map (Figure 2) shows the inter relationship between the start of injection and engine temperature, load and speed for a car engine.

## Start of delivery

In addition to start of injection, start of delivery is another aspect that is often considered. It relates to the point at which the fuel injection pump starts to deliver fuel to the injector. Since, on older fuel-injection systems and when the engine is not running, the start of delivery is easier to determine than the actual injection point, synchronization of the start of injection with the engine (particularly in the case of in-line and distributor injection pumps) is performed on the basis of the start of delivery. This is possible because there is a definite relationship between the start of delivery and the start of injection (injection lag [1]).

The time it takes for the pressure wave to travel from the high-pressure pump to the nozzle depends on the length of the pipe and produces an injection lag stated in degrees of crankshaft rotation that varies according to engine speed. The engine also has a longer ignition lag (in terms of crankshaft rotation) at higher speeds [2]). Both these effects must be compensated for – which is the reason why a fuel-injection system must be able to adjust the start of delivery/start of injection in response to engine speed, load and temperature.

1) Time from start of fuel delivery to start of injection

2) Time from start of injection to start of ignition

## Injected-fuel quantity

The required fuel mass, $m_e$, in mg for an engine cylinder per power stroke is calculated using the following equation:

$$m_e = \frac{P \cdot b_e \cdot 33.33}{n \cdot z} \text{ [mg/stroke]}$$

where
- $P$ is the engine's power output in kW
- $b_e$ is the engine's specific fuel consumption in g/kWh
- $n$ is the engine speed in rpm and
- $z$ is the number of cylinders in the engine

The corresponding fuel volume (injected fuel quantity), $Q_H$, in mm³/stroke or mm³/injection cycle is then:

$$Q_H = \frac{P \cdot b_e \cdot 1{,}000}{30 \cdot n \cdot z \cdot \rho} \text{ [mm}^3\text{/stroke]}$$

Fuel density, $\rho$, in mg/mm³ is temperature-dependent.

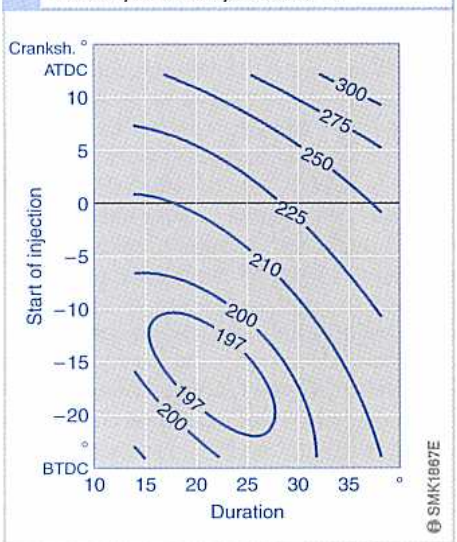
It is evident from this equation that the engine's power output at a constant level of efficiency ($\eta \sim 1/b_e$) is directly proportional to the injected fuel quantity.

The mass of fuel injected by the fuel-injection system depends on the following variables:
- The fuel-metering cross-section of the nozzle
- The injection duration
- The variation over time of the pressure difference between the injection pressure and the pressure in the combustion chamber and
- The density of the fuel

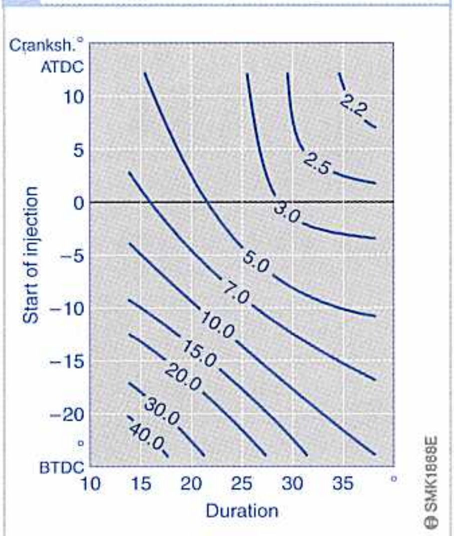
At high pressures, the diesel fuel is compressible, i.e. it is, in fact, compressed. This affects the injected fuel quantity and must therefore be taken into account by the injection control system.

Variations in the injected-fuel quantity lead to fluctuations in the level of pollutant emissions and in the engine's power output. By the use of high-precision fuel-injection systems controlled by an electronic governor, the required injected fuel quantity can be delivered with a high degree of accuracy.

**1** Specific fuel consumption $b_e$ in g/kWh versus start of injection and injection duration




**2** Specific nitrogen oxide ($NO_x$) emission in g/kWh versus start of injection and injection duration




**Figures 1 to 4**
Engine:
Six-cylinder diesel commercial-vehicle engine with common-rail fuel injection

Operating conditions:
$n$ = 1,400 rpm,
50 % power

In this example, the injection duration is varied by variation of the injection pressure

## Injection characteristics

An engine's emission and fuel-consumption characteristics are very important considerations. For that reason, the following demands are placed on the fuel-injection system:

- Fuel injection must be precisely timed. Even small discrepancies have a substantial effect on fuel consumption, emission levels and combustion noise (Figures 1 to 4).
- It should be possible to vary the injection pressure as independently as possible to suit the demands of all engine - operating conditions (e.g. load, speed).
- The injection must be reliably terminated. Uncontrolled "post-injection" leads to higher emission levels.

The term "injection characteristics" refers to the pattern of the fuel quantity injected into the combustion chamber as a function of time.

### Injection duration

One of the main parameters of the injection pattern is the injection duration. This refers to the period of time that the nozzle is open and allows fuel to flow into the combustion chamber. It is specified in degrees of crankshaft or camshaft rotation, or in milliseconds. Different diesel combustion processes demand different injection durations as illustrated by the following examples (approximate figures at rated power):

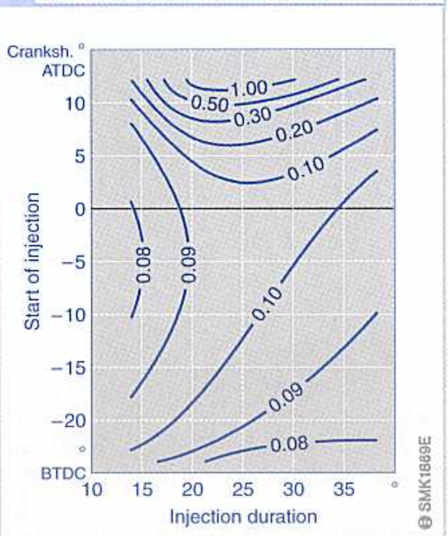
- Direct-injection car engines: 32...38° of crankshaft rotation
- Indirect-injection car engines: 35...40° of crankshaft rotation and
- Direct-injection commercial-vehicle engines: 25...36° of crankshaft rotation

An injection duration of 30° of crankshaft rotation corresponds to 15° of camshaft rotation. In terms of time at an injection pump speed [1]) of 2,000 rpm, that is equal to an injection duration of 1.25 ms.

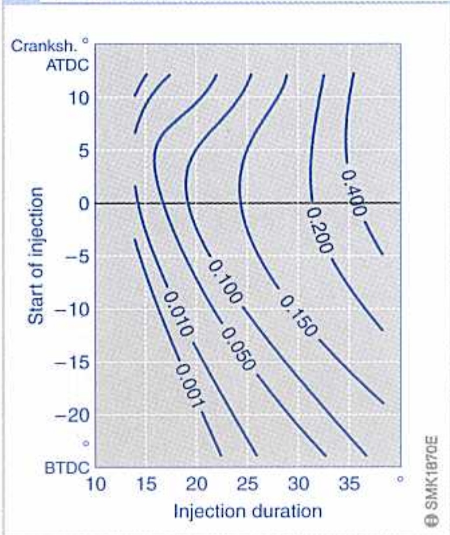
In order to minimize fuel consumption and soot emission, the injection duration must be defined on the basis of the engine operating conditions and the start of injection (Figures 1 and 4).

[1]) Equal to half the engine speed on four-stroke engines

3 Specific emission of unburned hydrocarbons (HC) in g/kWh versus start of injection and injection duration




4 Specific soot emission in g/kWh versus start of injection and injection duration



# Overview of governor types

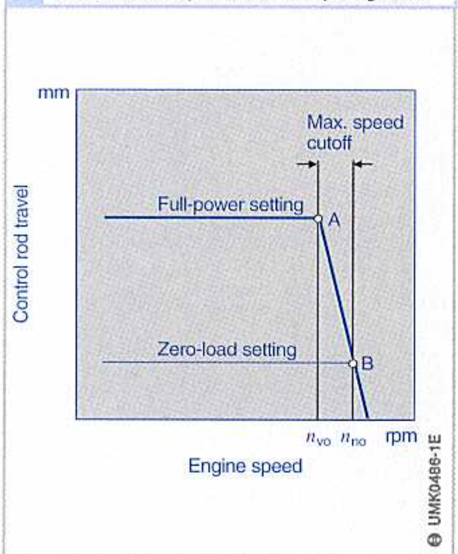
## Governor type designations

The governor type designation is shown on the identification plate. It indicates the essential features of the governor (e.g. design type, governed speed range, etc.). Figure 3 details the individual components of the governor type designation.

## Maximum-speed governors

Maximum-speed governors are intended for diesel engines that drive machinery at their nominal speed. For such applications, the governor's job is merely to hold the engine at its maximum speed; control of idle speed and start quantity are not required. If the engine speed rises above the nominal speed, $n_{vo}$, because the load decreases, the governor shifts the control rack towards the stop setting, i.e. the control rack travel is shortened and the delivery quantity reduced (Figure 1). Engine speed increase and control rack travel decrease follow the gradient A – B. The maximum no-load speed, $n_{no}$, is reached when the engine load is removed entirely. The difference between $n_{no}$ and $n_{vo}$ is determined by the proportional response of the governor.

1 Characteristic map for a maximum-speed governor



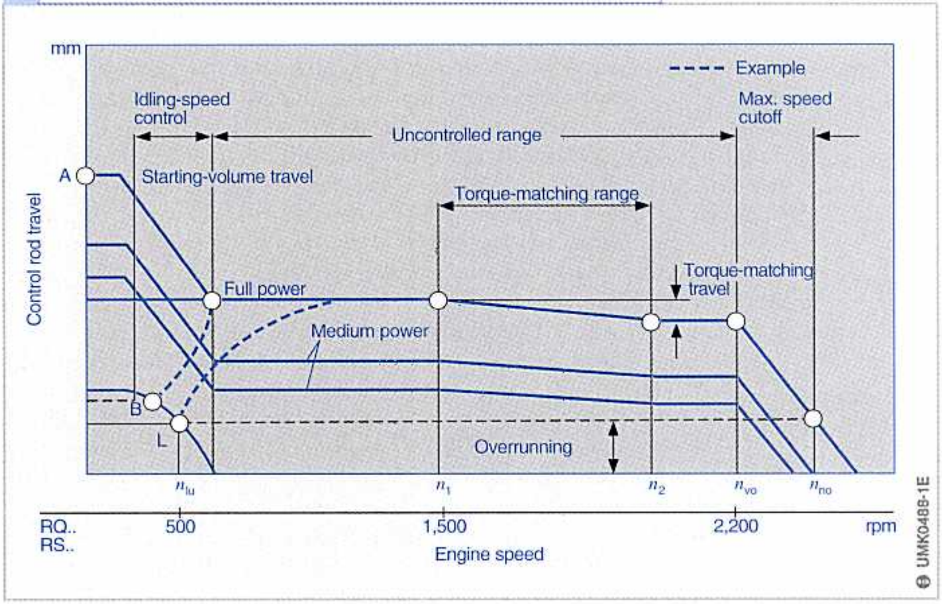

## Minimum/maximum-speed governors

Diesel engines for motor vehicles frequently do not require engine speeds between idling and maximum speed to be governed. Within this range, the fuel-injection pump's control rack is directly operated by the accelerator pedal under the control of the driver so as to obtain the required engine torque. At idle speed, the governor ensures that the engine does not cut out; it also limits the engine's maximum speed. The governor's characteristic map (Figure 2) shows the following: When the engine is cold, it is started using the start quantity (A). At this point, the driver has fully depressed the accelerator pedal.

If the driver releases the accelerator, the control rack returns to the idle speed setting (B). While the engine is warming up, the idle speed fluctuates along the idle speed curve and finally comes to rest at the point L. Once the engine has warmed up, the maximum start quantity is not generally required when the engine is restarted. Some engines can even start with the control rack actuating lever (accelerator pedal) in the idling position.

An additional device, the temperature-dependent start quantity limiter, can be used to limit the start quantity when the engine is warm even if the accelerator pedal is fully depressed. If the driver fully depresses the accelerator pedal when the engine is running, the control rack is moved to the full-load setting. The engine speed increases as a result and when it reaches $n_1$, the torque control function comes into effect, i.e. the full-load delivery quantity is slightly reduced. If the engine speed continues to increase, the torque control function ceases to be effective at $n_2$. With the accelerator pedal fully depressed, the full-load volume continues to be injected until the maximum full-load speed, $n_{vo}$, is reached. Upwards of $n_{vo}$, the maximum speed limiting function comes into effect in accordance with the proportional response characteristics so that a further small increase in engine speed results in the control rack travel backing off so as to reduce the delivery quantity. The maximum no-load speed, $n_{no}$, is reached when the engine load is entirely removed.

2 Characteristic map for a minimum/maximum-speed governor with torque control




3 Bosch governor type designations

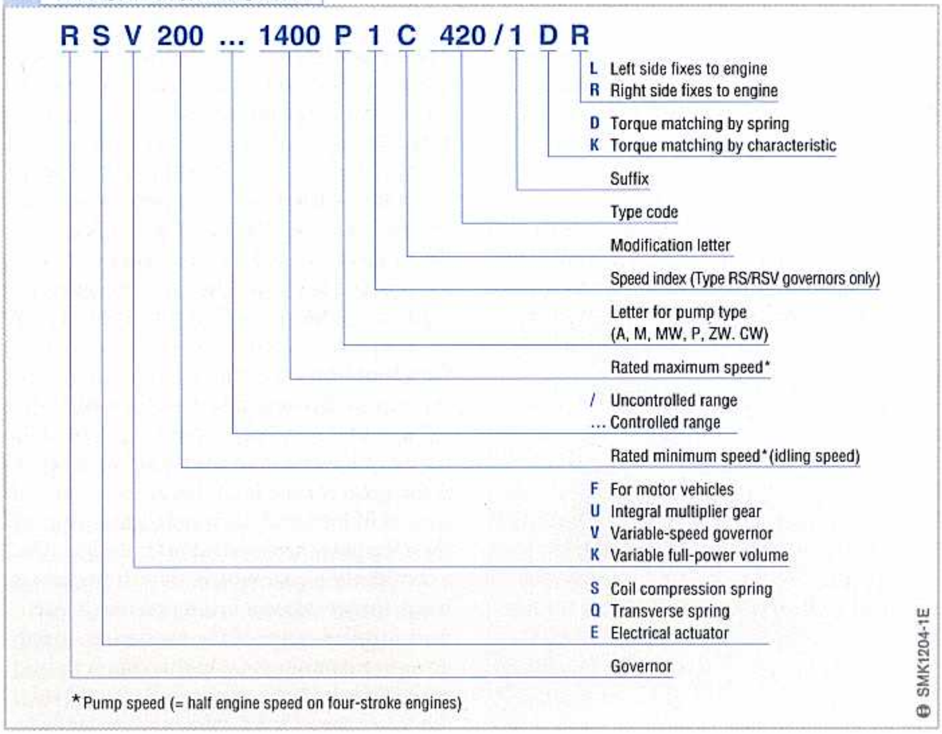



Fig. 3
With combination governors, multiple speeds are specified (e.g. 300/900...1,200).

ally the entire output range is taken into consideration.

The largest proportion of particulate emissions is made up of soot particles (black smoke). As a large part of the air/fuel mixing process only takes place in the course of combustion, localized over-enrichment occurs and this in some cases leads to an increase in black smoke emissions even at moderate levels of excess air. The air-fuel ratio usable at the statutory full-load smoke limit is a measure of the efficiency of air utilization.

### Combustion pressure limits

During the ignition process, the partially vaporized fuel mixed with the air burns under high compression at a rapid rate and with a high initial thermal-release peak (without pre-injection). This is referred to as "hard" combustion. High combustion pressure peaks are produced and this requires a relatively heavy engine. The forces generated during combustion place periodic alternating stresses on the engine components. The dimensioning and durability of the engine and drivetrain components therefore limit the permissible maximum compression pressure and consequently the amount of fuel injected.

### Exhaust-gas temperature limits

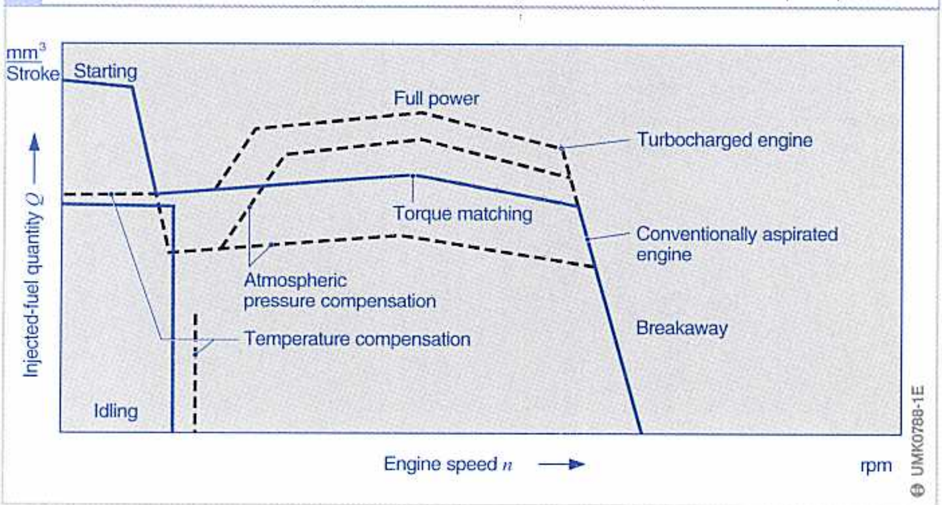
The high thermal stresses placed on the engine components surrounding the hot combustion chamber, the heat resistance of the exhaust valves and of the exhaust system and cylinder head determine the maximum exhaust temperature of a diesel engine.

### Engine speed limits

The fact that diesel engines operate on the basis of excess air with regulation of the injected-fuel quantity means that the power output at a constant engine speed is basically dependent solely on the amount of fuel injected. If the amount of fuel supplied to a diesel engine is increased without a corresponding increase in the load that it is working against, then the engine speed will rise. If the fuel supply is not reduced before the engine reaches a critical speed, the engine may rev itself to the point of destruction. Consequently, an engine speed limiter or governor is absolutely essential on a diesel engine.

Diesel engines that drive machinery are expected to maintain a constant speed or to keep their speed within certain upper and lower limits regardless of the load applied. For such requirements, there are variable-speed or intermediate-speed governors.

2 Fuel-injection volume relative to engine speed and load with adjustment for temperature and atmospheric pressure



### Method of operation of plunger-and-barrel assembly (stroke phase sequence)

The rotation of the camshaft is converted directly into a reciprocating motion on the part of the roller tappet and consequently into a similar reciprocating action on the part of the pump plunger.

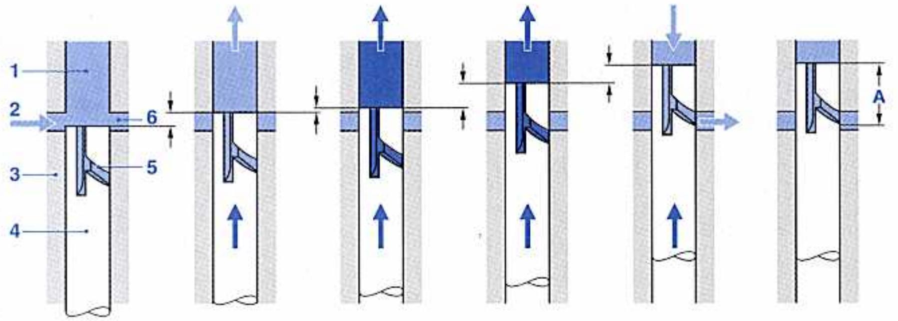
The delivery stroke, whereby the piston moves towards its "top dead center" (TDC), is assumed by the action of the cam. A compression spring performs the task of returning the plunger to "bottom dead center" (BDC). It is dimensioned to keep the roller in contact with the cam even at maximum speed, as loss of contact between roller and cam, and the consequent impact of the two surfaces coming back into contact, would inevitably cause damage to both components in the course of continuous operation.

The plunger-and-barrel assembly operates according to the overflow principle with helix control (Figure 6). This is the principle adopted on Type PE in-line fuel-injection pumps and Type PF single-plunger fuel-injection pumps.

When the pump plunger is at bottom dead center (BDC) the cylinder inlet passages are open. Under pressure from the presupply pump, fuel is able to flow through those passages from the fuel gallery to the plunger chamber. During the delivery stroke, the pump plunger closes off the inlet passages. This phase of the plunger lift is referred to as

6 Stroke phases

| 1 Bottom dead center | 2 Preliminary phase | 3 Pressure-relief phase | 4 Effective stroke | 5 Residual stroke | 6 Top dead center |
|---|---|---|---|---|---|




| | | | | | |
|---|---|---|---|---|---|
| Fuel flows from the injection pump's fuel gallery into the plunger chamber of the pump element | The pump plunger moves from bottom dead center to the point where its top edge closes off the inlet passages (variable depending on pump element) | Pump plunger moves from point marking the end of the plunger lift to port closing to the point at which the delivery valve opens (units with constant-volume valve only) | Pump plunger moves from the point at which the delivery valve opens to the point at which the helix opens the inlet passage (overflow) | Pump plunger travels from the point at which the inlet passage is opened to top dead center | Point at which plunger movement reverses |

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Fig. 6
1 Plunger chamber
2 Fuel inlet
3 Pump barrel
4 Pump plunger
5 Helix
6 Fuel return

A Total stroke

7 Fuel-delivery control

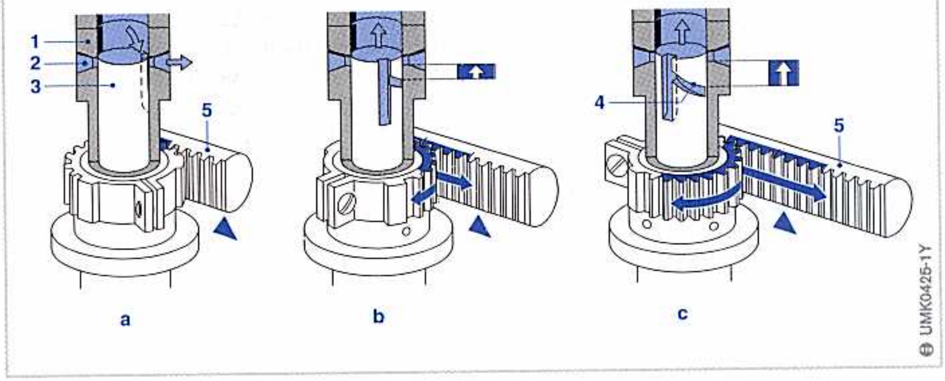



Fig. 7
a Zero delivery
b Partial delivery
c Maximum delivery

1 Pump barrel
2 Inlet passage
3 Pump plunger
4 Helix
5 Geared control rack

the preliminary phase. As the delivery stroke continues, fuel pressure increases and causes the delivery valve at the top of the plunger-and-barrel assembly to open. If a constant-volume valve is used (see section "Delivery valves") the delivery stroke also includes a retraction-lift phase. Once the delivery valve has opened, fuel flows along the high-pressure line to the nozzle for the duration of the effective stroke. Finally, the nozzle injects a precisely metered quantity of fuel into the combustion chamber of the engine.

Once the pump plunger's helix releases the inlet passage again, the effective stroke is complete. From this point on, no more fuel is delivered to the nozzle as, during the residual stroke, the fuel can escape through the vertical groove from the plunger chamber back into the fuel gallery so that pressure in the plunger-and-barrel assembly breaks down.

After the piston reaches top dead center (TDC) and starts to move back in the opposite direction, fuel flows through the vertical groove from the fuel gallery to the plunger chamber until the helix closes off the inlet passage again. As the plunger continues its return stroke, a vacuum is created inside the pump barrel. When the inlet passage is opened again, fuel then immediately flows into the plunger chamber. At this point, the cycle starts again from the beginning.

### Fuel-delivery control

Fuel delivery can be controlled by varying the effective stroke (Figure 7). This is achieved by means of a control rack (5) which twists the pump plunger (3) so that the pump plunger helix (4) alters the point at which the effective delivery stroke ends and therefore the quantity of fuel delivered.

In the final zero-delivery position (a), the vertical groove is directly in line with the inlet passage. With the plunger in this position, the pressure chamber is connected to the fuel gallery through the pump plunger for the entire delivery stroke. Consequently, no fuel is delivered. The pump plungers are placed in this position when the engine is switched off.

For partial delivery (b), fuel delivery is terminated depending on the position of the pump plunger.

For maximum delivery (c), fuel delivery is not terminated until the maximum effective stroke is reached, i.e. when the greatest possible delivery quantity has been reached.

The force transfer between the control rack and the pump plunger, see Figure 7, takes place by means of a geared control rack (PE..A and PF pumps) or via a ball joint with a suspension arm and control sleeve (Type PE..M, MW, P, R, ZW(M) and CW pumps).

## Size M fuel-injection pumps

The size M in-line fuel-injection pump (Figures 3 and 4) is the smallest of the Series PE pumps. It has a light-metal (aluminum) body that is attached to the engine by means of a flange.

The size M pump is an open-type in-line fuel-injection pump which has a cover plate on the side and the base. On size M pumps, the peak injection pressure is limited by the pump to 400 bar.

After removal of the side cover plate, the delivery quantities of the plunger-and-barrel assemblies can be adjusted and matched to one another. Individual adjustment is effected by moving the position of the clamp blocks (Figure 4, Item 5) on the control rack (4). When the fuel-injection pump is running, the control rack is used to adjust the position of the pump plungers and, as a result, the delivery quantity within design limits. On the size M pump, the control rack consists of a round steel rod that is flatted on one side. Fitted over the control rack are the slotted clamp blocks. Together with its control sleeve, the lever (3), which is rigidly attached to the control sleeve, forms the mechanical link with the corresponding clamp block. This arrangement is referred to as a rod-and-lever control linkage.

The pump plungers sit directly on top of the roller tappets (6). LPC adjustment is achieved by selecting tappet rollers of different diameters.

The size M pump is available in 4, 5 and 6 cylinder versions, and is suitable for use with diesel fuel only.

3 Type M in-line fuel-injection pump (external view)


4 Type M in-line fuel-injection pump (sectional view)

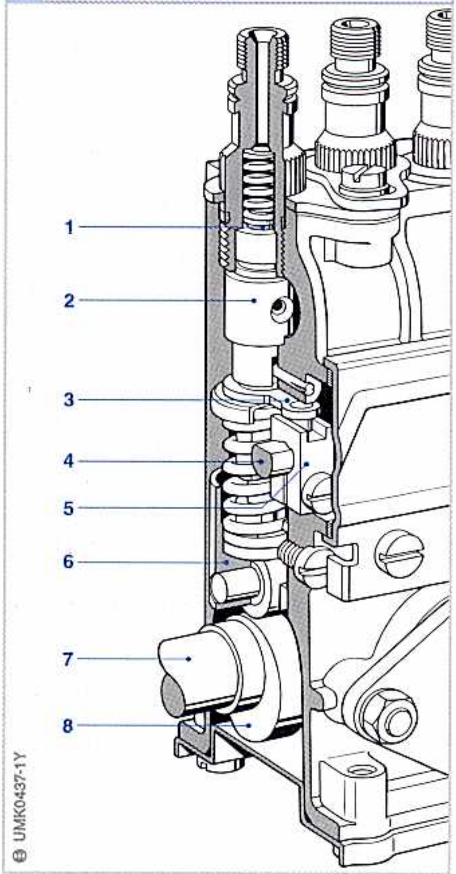



Fig. 4
1 Delivery valve
2 Pump barrel
3 Control-sleeve lever arm
4 Control rack
5 Clamp block
6 Roller tappet
7 Camshaft
8 Cam

## Size MW fuel-injection pumps

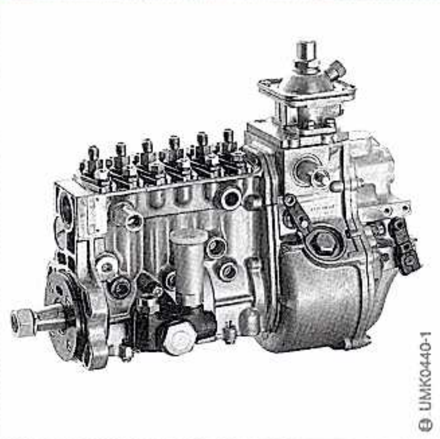
For higher pump outputs, the size MW in-line fuel-injection pump was developed (Figures 7 and 8).

The MW pump is a closed-type in-line fuel-injection pump which has a peak pressure limited to 900 bar, it is a lightweight metal design similar to the smaller models, and is attached to the engine by a baseplate, flange or cradle mounting.

Its design differs significantly from that of the Series M and A pumps. The main distinguishing feature of the MW pump is the barrel-and-valve assembly that is inserted into the pump housing from above. The barrel-and-valve assembly is assembled outside the housing and consists of the pump barrel (Figure 8, Item 3), the delivery valve (2) and the pressure-valve holder. On the MW pump, the pressure-valve holder is screwed directly into the top of the longer pump barrel. Shims or spacers of varying thicknesses are fitted between the pump housing and the barrel-and-valve assembly to achieve LPC adjustment. The uniformity of fuel delivery between the barrel-and-valve assemblies is adjusted by rotating the barrel-and-valve assembly from the outside. To achieve this, the flange (1) is provided with slots. The position of the pump plunger is not altered by this adjustment.

The MW pump is available with the various mounting options in versions for up to 8 cylinders. It is suitable for diesel fuel only. MW pumps are no longer used for new engine designs.

7 Type MW in-line fuel-injection pump (external view)




8 Type MW in-line fuel-injection pump (sectional view)

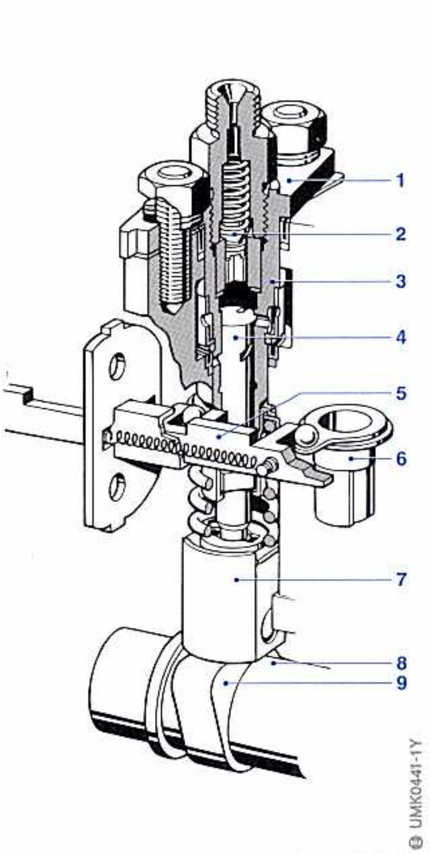



Fig. 8
1 Pump unit mounting flange
2 Delivery valve
3 Pump barrel
4 Pump plunger
5 Control rack
6 Control sleeve
7 Roller tappet
8 Camshaft
9 Cam

## Type RSF minimum/maximum-speed governor

### Design

The Type RSF centrifugal mechanical governor was designed specifically as a minimum/ maximum-speed governor for motor-vehicle engines with Type M in-line diesel fuel-injection pumps. It is suitable for road-going vehicles (cars and commercials) which only require limitation of the minimum and maximum speeds. Within the uncontrolled intermediate-speed range, the fuel-injection pump's control rack is directly operated by the accelerator pedal under the control of the driver so as to obtain the required engine torque (Figure 43).

The Type RSF governor meets demanding requirements in respect of governor characteristics, ease of operation and driver convenience. It is intended primarily for fast-revving diesel engines in cars. In addition, it offers the facility for combination with compensating mechanisms and is easily adjustable. The governor design can be divided into two sections: the governor movement and the actuator mechanism (Figure 44).

*Governor movement stage 1* (idle speed)
The force originates from the flyweights (22) and is transmitted via the sliding sleeve (20) and the guide lever (9) to the idle-speed spring (12) and the auxiliary idle-speed spring (14) – both are leaf springs.

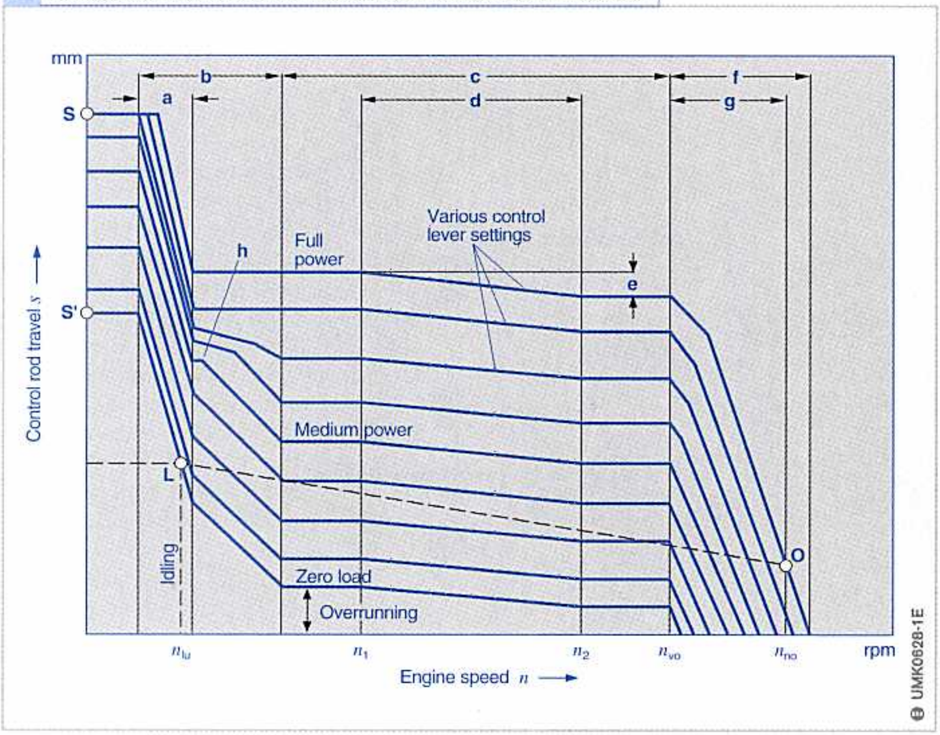
*Governor movement stage 2*
(up to full-load speed regulation)
After the idle-speed travel has been completed, the force is transmitted from the sliding sleeve (20) via the torque-control spring retainer (18) and the tensioning lever (16) to the governor spring (17).

**Fig. 43**

- a Idle-speed range (effective range of the idle-speed spring)
- b Extended idle-speed range at no load and part load (combined effective range of idle-speed and auxiliary idle-speed springs)
- c Uncontrolled range
- d Torque-control range (effective range of torque-control spring)
- e Torque-control travel
- f Breakaway range (effective range of governor spring)
- g Full-load speed regulation at high no-load speed
- h Start of auxiliary idle-speed spring shutoff
- S Starting position with accelerator fully depressed (cold starting)
- S' Starting position with accelerator not depressed (hot starting)
- L Minimum no-load position
- O Maximum no-load position

- $n_{lu}$ Minimum no-load speed (idling)
- $n_{no}$ Maximum no-load speed
- $n_{vo}$ Maximum full-load speed (limit speed)
- $n_1$ Speed at start of torque control
- $n_2$ Speed at end of torque control

**43** Characteristic map for Type RSF minimum/maximum-speed governor (example)



When the flyweights move outwards, the sliding sleeve is moved along its central axis. Apart from in the idle-speed, full-load torque-control and breakaway range, the sliding sleeve is stationary and the injected fuel quantity required to obtain the necessary engine output is set by moving the control lever of the actuator mechanism.

At the pivot B, the guide lever (9) is attached to the sliding sleeve. In addition, the guide lever and the tensioning lever (16) pivot around point A.

*Actuator mechanism*
The desired engine speed is set by means of the control lever (6) which acts via the linkage lever (5) and the reverse-transfer lever (11) on the variable-fulcrum lever (13) which in turn transmits the movement to the sprung link (2) and the control rack (4) of the fuel-injection pump.

The sprung link compensates for excess travel on the part of the variable-fulcrum lever. Like the guide lever, the reverse-transfer lever is also attached by a pivoting joint at point B to the sliding sleeve, and is also attached by another connecting pin to the variable-fulcrum lever (13).

The lower anchor point of the variable-fulcrum lever can be moved by means of the full-load adjusting screw (19) in order to vary the full-load injected fuel quantity. It also acts as a spring buffer for the variable-fulcrum lever so that the excess sliding-sleeve travel can be accommodated if the engine is overrevving.

The pivot shaft of the stop lever (3) passes through the governor housing and is attached on the outside to a shutoff lever (1) that can be used to stop the engine. In that case, the stop lever moves the control rack to the stop setting.

44 Type RSF minimum/maximum-speed governor

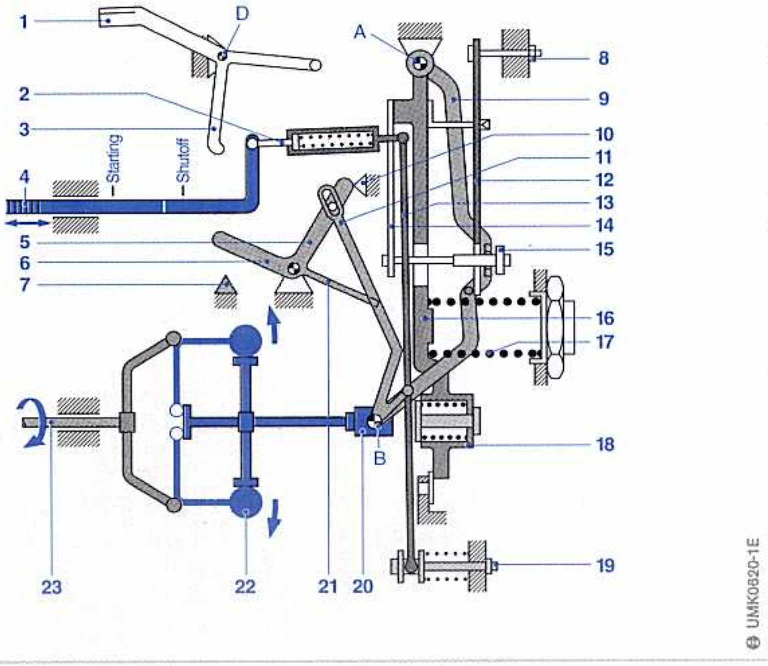



Fig. 44
1 Shutoff lever
2 Sprung link
3 Stop lever
4 Control rack
5 Linkage lever (internal)
6 Control lever (external)
7 Full-load stop
8 Adjusting screw for idle speed
9 Guide lever
10 Low-idle stop
11 Reverse-transfer lever
12 Idle-speed spring
13 Variable-fulcrum lever
14 Auxiliary idle-speed spring
15 Adjusting screw for auxiliary idle-speed spring
16 Tensioning lever
17 Governor spring
18 Spring retainer (torque control)
19 Full-load adjusting screw
20 Sliding sleeve
21 Auxiliary idle-speed spring shutoff
22 Flyweight
23 Fuel-injection pump camshaft

## Starting the engine

The required setting for starting the engine is specified by the engine manufacturer. As a rule, the engine can be started without pressing the accelerator pedal. Only when there is a combination of cold weather and cold engine is the control lever (6) set to the full-load stop (7) – a fixed stop on the governor housing corresponding to a fully depressed accelerator (Figure 45). The reverse-transfer lever (11) pivots around point B, thereby moving the variable-fulcrum lever (13) towards the start-quantity position. As a result, the control rack (4) moves to the start-quantity position so that the engine receives the necessary fuel quantity for starting. Rapid speed-regulation breakaway from the start-quantity position is made possible by the fact that when the control lever is in full-load position, the auxiliary idle-speed spring (14) is lifted away from the guide lever (9) by a shutoff arm (21).

## Operating characteristics

*Idle speed* (Figure 46)

When the accelerator is released after the engine has started, a compression spring (not illustrated) moves the control lever (6) back to the idle-speed position. This means that the linkage lever (5) is in contact with the low-idle stop screw (10).

While the engine is warming up, the idle speed fluctuates along the idle speed curve and finally comes to rest at the point L (Figure 43). As the engine speed increases, the flyweights (22) move outwards and push the sliding sleeve (20) to the right. During the idle-speed stage, the control rack (4) is moved by the action of the sliding sleeve transmitted via the reverse-transfer lever (11) and the variable-fulcrum lever (13) towards the stop setting. At the same time, the movement of the sliding sleeve causes the guide lever (9) to pivot around point A and press against the idle-speed spring (12), the tension of which (and therefore the idle speed) can be preset by means of the adjusting screw (8). At a certain speed, the guide lever also comes into contact with the adjusting nut for the auxiliary idle-speed spring (14).

**Fig. 45**
(Only those components involved in the governing function are illustrated)
4 Control rack
6 Control lever (external)
7 Full-load stop
9 Guide lever
11 Reverse-transfer lever
13 Variable-fulcrum lever
14 Auxiliary idle-speed spring
21 Auxiliary idle-speed spring shutoff

**45** Type RSF minimum/maximum-speed governor in cold-starting position

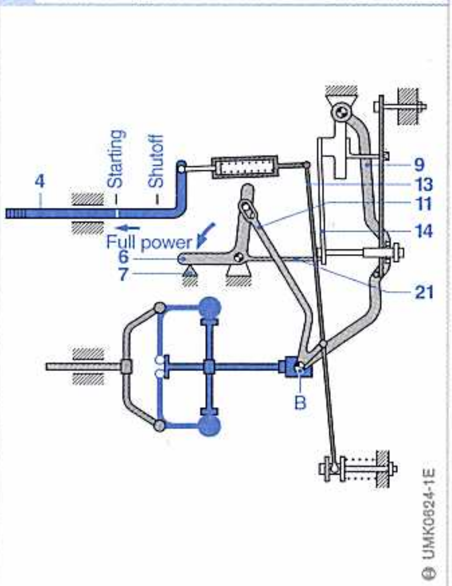
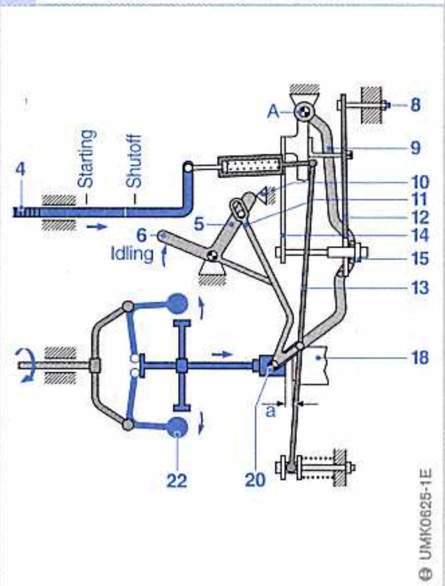



**Fig. 46**
(Only those components involved in the governing function are illustrated)
4 Control rack
5 Linkage lever (internal)
6 Control lever
8 Adjusting screw for idle speed
9 Guide lever
10 Low-idle stop
11 Reverse-transfer lever
12 Idle-speed spring
13 Variable-fulcrum lever
14 Auxiliary idle-speed spring
15 Adjusting screw for auxiliary idle-speed spring
18 Spring retainer (torque control)
20 Sliding sleeve
22 Flyweight

**46** Type RSF minimum/maximum-speed governor in idle-speed position



*Intermediate speeds*
After passing beyond the idle-speed travel (a), the sliding sleeve (20) comes into contact with the torque-control spring retainer (18). In the uncontrolled range between idle speed and maximum speed, the flyweights (22) do not change their position, apart from the small amount of travel for torque control, until the maximum speed is reached. The control-rod position, and therefore the injected fuel quantity, is set directly by moving the control lever (6), i.e. the driver varies the delivery quantity (e.g. in order to increase vehicle speed or to negotiate an uphill gradient) by means of the accelerator pedal (control lever position is between the idle-speed and maximum-speed stops. If the accelerator pedal is fully depressed, the control rack moves to the full-load position.

*Torque control*
When the torque-control function is active, the full-load delivery quantity is reduced if the engine speed exceeds $n_1$ because the force of the flyweights acting on the sliding sleeve (20) is greater than the force of the torque-control spring in the spring retainer (18). The torque-control spring "gives" so that the control rack (4) shifts by the torque-control travel if the speed continues to increase. At the speed $n_2$ the torque-control phase comes to an end. The Type RSF governor may also incorporate a mechanism for negative as well as positive torque control. In this case, the control rack position is controlled by combination of springs.

*High-idle speed* (Figure 47)
With the accelerator pedal fully depressed, the full-load volume continues to be injected until the maximum full-load speed, $n_{vo}$, (breakaway speed) is reached. If the engine speed continues to increase beyond the maximum full-load speed, the force of the flyweights (22) is enough to overcome the force of the governor spring (17). Full-load speed regulation then comes into effect. The engine speed then increases a little further, the control rack is pushed back towards the stop setting and as a result the fuel delivery quantity is reduced. The point at which start of speed regulation takes effect depends on the tension of the governor spring. The maximum no-load speed, $n_{no}$, is reached when the engine load is entirely removed. When the engine is overrunning, e.g. if the vehicle is traveling downhill, the engine is accelerated by the road wheels. Under such conditions, no fuel is injected (overrun fuel cutoff).

## Stopping the engine

Manual operation of the shutoff lever (1) moves the control rack (4) by means of the stop lever (3) to the stop setting. Fuel delivery is shut off and, therefore, the engine stopped. The engine can also be stopped by means of a pneumatically operated shutoff valve (refer to the section "Calibration devices").

47 Type RSF minimum/maximum-speed governor in full-load position

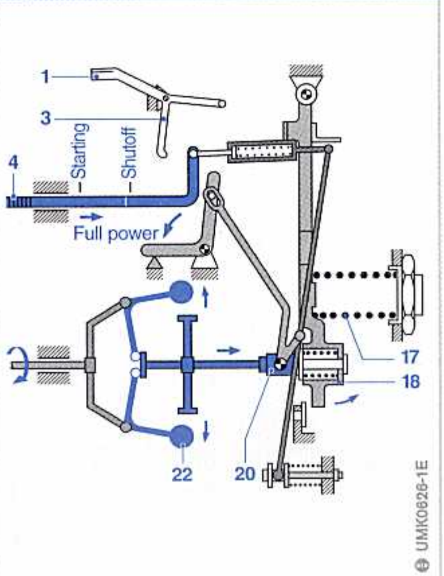



Fig. 47
(Only those components involved in the governing function are illustrated)
1 Shutoff lever
3 Stop lever
4 Control rack
17 Governor spring
18 Spring retainer (torque control)
20 Sliding sleeve
22 Flyweight

## Delivery valve

The delivery valve is fitted between the plunger-and-barrel assembly and the high-pressure delivery line. Its purpose is to isolate the high-pressure delivery line from the plunger-and-barrel assembly. It also reduces the pressure in the high-pressure delivery line and the nozzle chamber following fuel injection to a set static pressure. Pressure reduction causes rapid and precise closure of the nozzle and prevents undesirable fuel dribble into the combustion chamber.

In the course of the delivery stroke, the increasing pressure in the plunger chamber lifts the delivery-valve cone (Figure 11, Item 3) from the valve seat (4) in the delivery-valve body (5). Fuel then passes through the delivery-valve holder (1) and into the high-pressure delivery line to the nozzle. As soon as the helix of the pump plunger brings the injection process to an end, the pressure in the plunger chamber drops. The delivery-valve cone is then pressed back against the valve seat by the valve spring (2). This isolates the space above the pump plunger and the high-pressure side of the system from one another until the next delivery stroke.

### Constant-volume valve without return-flow restriction

In a constant-volume valve (Bosch designation GRV), part of the valve stem takes the form of a "retraction piston" (Figure 12, Item 2). It fits into the valve guide with a minimum degree of play. At the end of fuel delivery, the retraction piston slides into the valve guide and shuts off the plunger chamber from the high-pressure delivery line. This increases the space available to the fuel in the high-pressure delivery line by the charge volume of the retraction piston. The retraction volume is dimensioned precisely to suit the length of the high-pressure delivery line, which means that the latter must not be altered.

In order to achieve the desired fuel-delivery characteristics, torque-control valves are used in some special cases. They have a retraction piston with a specially ground pintle (6) on one side.

### Constant-volume valve with return-flow restriction

A return-flow restriction (Bosch designation RDV or RSD) may also be used in addition to the constant-volume valve. Its purpose is to dampen and render harmless returning pressure waves that are produced when the nozzle

11 Delivery valve

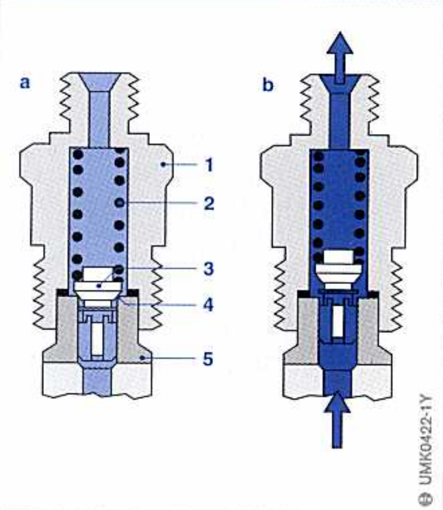



**Fig. 11**
a Closed
b During fuel delivery

1 Pressure-valve holder
2 Pressure-valve spring
3 Delivery-valve cone
4 Valve seat
5 Delivery-valve support

12 Constant-volume delivery-valve cone

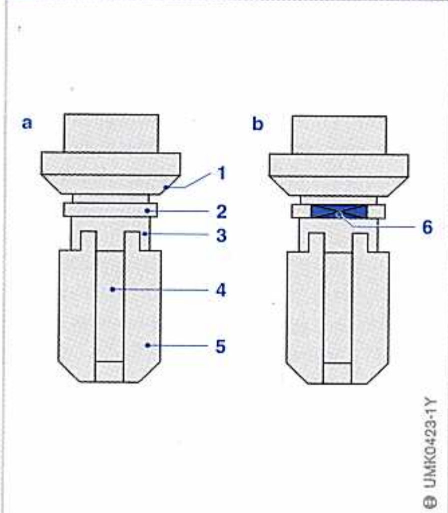



**Fig. 12**
a Normal
b With specially ground pintle pressure matching

1 Valve seat
2 Retraction piston
3 Ring groove
4 Delivery-valve stem
5 Vertical groove
6 Specially ground pintle

closes. This reduces or entirely eliminates wear effects and cavitation in the plunger chamber. It also prevents undesirable secondary injection.

The return-flow restriction is integrated in the upper part of the delivery-valve holder (Figure 13), in other words between the constant-volume valve and the nozzle. The valve body (4) has a small bore (3) the size of which is dimensioned to suit the application so as to achieve, firstly, the desired flow restriction and, secondly, to prevent reflection of pressure waves as much as possible. The valve opens when fuel is flowing in delivery direction. The delivery flow is therefore not restricted. For pressures up to approx. 800 bar, the valve body shaped like a disk. For higher pressures it is a guided cone.

Pumps with return-flow throttle valves are "open systems", i.e. during the plunger lift to port closing and retraction lift, the static pressure in the high-pressure delivery line is the same as the internal pump pressure. Consequently, this pressure must be at least 3 bar.

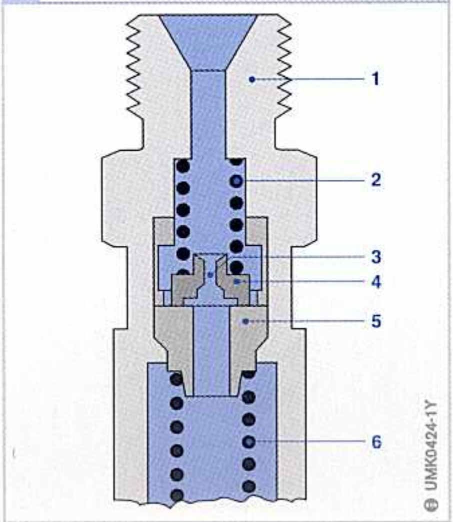
### Constant-pressure valve

The constant-pressure valve (Bosch designation GDV) is used on fuel-injection pumps with high injection pressures (Figure 14). It consists of forward-delivery valve (consisting of delivery valve, 1, 2, 3) and a pressure-holding valve for the return-flow direction (consisting of 2, 5, 6, 7 and 8) which is integrated in the delivery-valve cone (2). The pressure-holding valve maintains a virtually constant static pressure in the high-pressure delivery line between fuel-injection phases under all operating conditions. The advantages of the constant-pressure valve are the prevention of cavitation and improved hydraulic stability which means more precise fuel injection.

During the delivery stroke, the valve acts as a conventional delivery valve. At the end of the delivery stroke, the ball valve (7) is initially open and the valve acts like a valve with a return-flow restriction. Once the closing pressure is reached, the compression spring (5) closes the return-flow valve, thereby maintaining a constant pressure in the fuel line.

However, correct functioning of the constant-pressure valve demands greater accuracy of adjustment and modifications to the governor. It is used for high-pressure fuel-injection pumps (upwards of approx. 800 bar) and for small, fast-revving direct-injection engines.

13 Constant-volume delivery valve with return-flow restriction




14 Constant-pressure valve

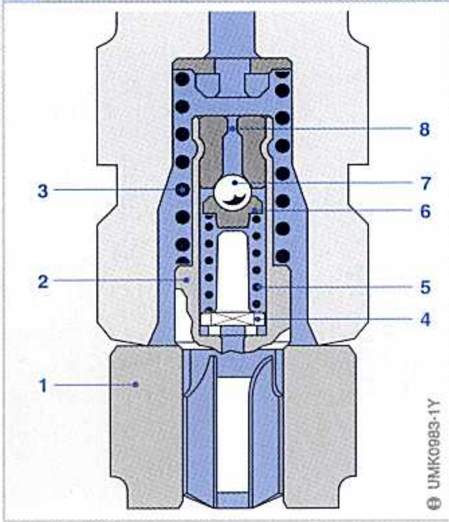



**Fig. 13**
1 Pressure-valve holder
2 Valve spring
3 Flow throttle
4 Valve body (disk in this case)
5 Valve holder
6 Pressure-valve spring

**Fig. 14**
1 Delivery-valve support
2 Delivery-valve cone
3 Pressure-valve spring
4 Filler piece
5 Compression spring (pressure-holding valve)
6 Spring seat
7 Ball
8 Flow throttle

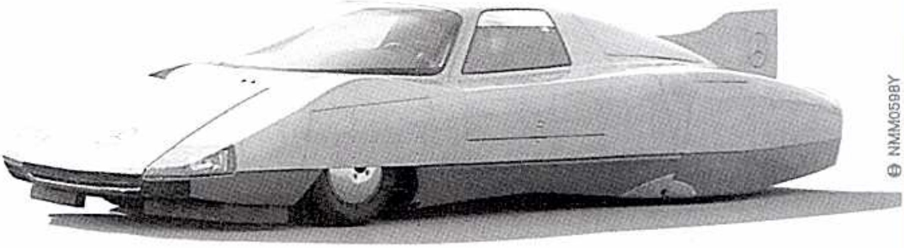
## 1978 diesel speed records

In April 1978 the experimental Mercedes-Benz C111-III set nine world speed records, some of which still stand today, and eleven international class records. Some of those records had previously been held by gasoline-engine cars.

The average speed of the record attempts was approximately 325 kph. The highest speed reached was measured at 338 kph. The average fuel consumption was only 16 *l*/100 km.

These considerable achievements were made possible primarily by the highly streamlined plastic body. Its aerodynamic drag coefficient of 0.195 was sensationally low for the time.

The car was powered by a 3-liter, five-cylinder in-line diesel engine with a maximum power output of 170 kW (230 bhp). That meant that it was twice as powerful as its standard production counterpart. The maximum torque of 401 Nm was produced at 3,600 rpm. This performance was made possible by a turbocharger and an intercooler.



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## Engine compartment of the Mercedes-Benz C111-III

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At the engine's nominal speed, the turbocharger was rotating at 150,000 rpm.

Precise fuel delivery and metering was provided by a Bosch Type PE...M in-line fuel-injection pump