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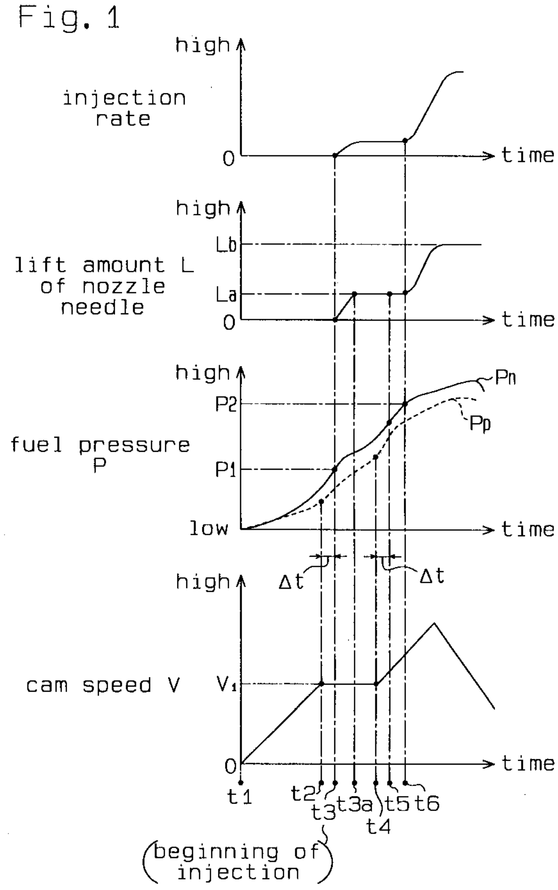
(54) **Fuel injection apparatus.**

(57) A low-speed area where a cam speed is low is provided in the cam characteristic of non-uniform speed cams (8a) of an injection pump (1) for moving a plunger (12) at a low speed, and a high-speed area for moving the plunger (12) at a high speed is provided in the back portion of the cam characteristic. The nozzle needle (37) of an injection valve (2) is opened at a predetermined first injection-valve opening pressure (P1), and is held at a predetermined valve-open position until fuel pressure reaches a second injection-valve opening pressure (P2) set higher than the first injection-valve opening pressure (P1). The injection valve (2) is provided with a steady flow-rate portion for keeping the area of an injection port (45) nearly constant even when the shift amount of the nozzle needle (37) varies, and the nozzle needle (37) is held at the predetermined valve-open position in this steady flow-rate area. The period in which the plunger moves in the low-speed area of the cam characteristic and the period in which the nozzle needle (37) is held at the predetermined valve-open position in the steady flow-rate area overlap each other at least partially. Accordingly, the low injection-rate period is secured at the beginning of fuel injection, thus ensuring high-pressure injection.

Fig. 1



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Technical Field

The present invention relates to a fuel injection apparatus which is used to supply fuel to a diesel engine in, for example, an automobile.

Related Background Art

A conventional fuel injection apparatus for supplying fuel to a diesel engine in an automobile is disclosed in Japanese Unexamined Patent Publication No. Sho 62-288364. The appratus uses a distributer type fuel injection pump which distributes compressed fuel to injection valves provided with the respective cylinders while rotating a single plunger. A non-uniform speed cam is used to reciprocatingly move the plunger. The cam has its characteristic representing a ratio of a moving speed to a rotation angle of the cam. as shown in Fig. 10. According to the characteristic, the cam speed determines the moving speed of the plunger. The begining part of the characteristic constitutes a low-speed area in accordance with low and constant speed, which indicates a slow mevement of the plunger. The end part of the characteristic constitues a high-speed area in accordance with the relatively high and chagable speed, indicating the quick movement of the plunger.

Angleich mechanism is provided at a delivery valve in the injection pump for changing the areas concerning the movement of the plunger according to the rotational speed of the diesel engine. The Angleich mechanism includes a notch provided with the periphery of a retraction collar of the delivery valve. The Angleich mechanism adjusts the residual pressure in the injection pipe that connects the injection pump to the injection valve. More specifically, with the engine running at a low speed, the fuel retracted by the delivery valve escapes through the notch toward the injection pipe, reducing the amount of retracted fuel, so that the residual pressure rises. With the engine running at a high speed, the fuel retracted by the delivery valve is difficult to escape through the notch toward the injection pipe, so that the residual pressure becomes as low as the one in the case where no notch is provided.

When the engine speed is as the the residual pressure in the injection pipe is high, the fuel pressure in the injection pipe becomes greater than the pressure for opening the injection valve even when the stroke of the plunger is small. Consequently, the plunger moves according to the low-speed area of the cam characteristic shown in Fig. 10, lowering the injection rate (rate of a change in the amount of fuel supply to time). When the engine speed is as the residual pressure in the injection pipe is low, the fuel pressure in the injection pipe will not become greater than the injection-valve opening pressure unless the stroke of the plunger becomes larger. Accordingly, the plunger moves according to the high-speed area in the cam characteristic, resulting in increasing the injection rate.

In the apparatus, the injection valve in use is designed to be opened in two stages in accordance with two pressures. This type of injection valve opens by a predetermined amount when the pressure of fuel from the injection pump exceeds a predetermined first injection-valve opening pressure. When the fuel pressure becomes a second injection-valve opening pressure set higher than the first injection-valve opening pressure, the injection valve further opens by a predetermined amount. In this manner, the pressure of the fuel from the injection pump when the plunger moves in the low-speed area of the cam characteristic is set to come between the first injection-valve opening pressure and the second injection-valve opening pressure. When the plunger is moved at a low speed, therefore, fuel injection is carried out with the nozzle needle held at the position corresponding to the first injection-valve opening pressure although the pressure of fuel from the injection pump is low.

To cope with the recent severer restrictions on the exhaust gas of diesel engines, particularly, the reduction in nitrogen oxides (NO_x), a so-called EGR, which extracts part of the exhaust gas from the exhaustion system and recirculates it in the intake system, is considered as effective means. The EGR, if carried out too much, however reduces the oxygen concentration in the combustion chamber, thus lowing the burning speed of the fuel mixture. To compensate for this reduction, fuel should be injected under a relatively high pressure so that the fuel would be atomized.

If fuel is simply injected under high pressure to help fuel atomization, the fuel would be sprayed out in short time. Consequently, the swirl causes the injection rate to become too high with respect to the speed of mixing air with fuel in the combustion chamber, causing incomplete combustion that increases the amount of smoke. If the injection-valve opening pressure is set low or the fuel supply rate of the injection pump is set low to reduce the injection rate, the atomization of fuel becomes difficult, thus increasing the amount of hydrocarbon (HC) in the exhaust gas.

To conduct EGR, therefore, it is important to accomplish both fuel atomization through high-pressure injection and suppression of the injection rate.

The non-uniform speed cam used in the prior art is so designed that the low-speed area comes at the front portion of the cam characteristic, and the injection valve in use has two injection-valve opening pressures. However, since the low-speed area and high-speed area based on the characteristic of the non-uniform speed cam are simply utilized separately in accordance with the rotational speed of the engine as discussed above, no consideration has been given to the flow-rate characteristic of the injection valve. Therefore, it is difficult to meet the aforementioned demand, i.e., execution of EGR while suppressing the injection rate at the initial stage of injection, with simple application of the prior art to a diesel engine. To fulfill the demand, a global measure including the control on the flow-rate characteristic of the injection valve should be taken. In short, the prior art is not sufficient to reduce the nitrogen oxides in the exhaust gas.

Disclosure of the Invention

Accordingly, it is a primary object of the present invention to provide a fuel injection apparatus which will prevent the burning speed of fuel mixture from decreasing at the time EGR is applied, and prevent an increase in the amount of hydrocarbon and the amount of smoke.

To achieve the foregoing and other objects and in accordance with the purpose of the present invention,

an improved fuel injection apparatus is provided. The apparatus has an injection pump having non-uniform speed cams. The cams rotates for reciprocatingly moving a plunger to press fuel to be supplied to injection valves. The valve has a nozzle needle moved by fuel pressure to selectively close and open an injection hole and an injection port injecting the fuel when the injection hole is open. The plunger is moved at a variable speed in accordance with a cam characteristic which represents a ratio of a cam speed to a rotation angle of the cam. The cam characteristic has a low-speed area where the cam speed is low, so as to adjust supply speed of the pressed fuel. The nozzle needle is moved to alter a area of fuel passage of the injection port to adjust an injection amount and pressure of the pressed fuel. Each injection valve includes means for holding said nozzle needle against the fuel pressure from said injection pump at a position in the injection port where a fuel passage becomes constricted for reducing a fuel injection amount at least during a part of movement of the plunger according to the low-speed area of the cam characteristic.

Brief Description of the Drawings

The features of the present invention that are believed to be novel are set forth with particularity in the appended claims. The invention, together with objects and advantages thereof, may best be understood by reference to the following description of the presently preferred embodiments together with the accompanying drawings in which:

Fig. 1 is a timing chart showing the relation among the injection rate, the lift amount of a nozzle needle, the fuel pressure and the cam speed;

Fig. 2 is a cross-sectional view of a fuel injection pump;

Fig. 3 is a cross-sectional view of a fuel injection valve;

Fig. 4 is a partially enlarged cross-sectional view showing the closed state of the nozzle needle of the injection valve;

Fig. 5 is a partially enlarged cross-sectional view showing the open state of the nozzle needle of the injection valve;

Fig. 6 is a characteristic diagram showing the relation between the cam angle and cam speed for the injection pump;

Fig. 7 is a characteristic diagram showing the relation between the lift amount of the nozzle needle of the injection valve and the flow rate of pressurized air;

Fig. 8 is a characteristic diagram showing a modification of the relation between the cam angle and the cam speed;

Fig. 9 is a characteristic diagram showing another modification of the relation between the cam angle and the cam speed; and

Fig. 10 is a timing chart showing the relation between the cam speed and the fuel pressure according to the conventional art.

Descroption of Special Embodiment

A fuel injection apparatus for use in a diesel engine for an automobile according to one embodiment of the present invention will now be described referring to Figs. 1 through 7.

As shown in Fig. 2, the fuel injection apparatus comprises a fuel injection pump 1, and fuel injection valves 2 attached to the respective cylinders of the diesel engine. The injection pump 1 of this embodiment is a type which distributes fuel under pressure to the individual injection valves 2 while rotating a single plunger 12. The injection valve 2 is of a type which will be opened in two stages depending on two pressures due to the action of two springs arranged in series. The injection pump 1 and each injection valve 2 are coupled by way of an injection pipe 4 of a given length (about 600 mm).

A drive shaft 5 is rotatably supported in the injection pump 1, with a drive pulley 3 attached to one end (left-hand end in Fig. 2) thereof. The pulley 3 is drivably coupled to the crankshaft of the diesel engine through a belt or the like. A vane type fuel feed pump 6 (developed 90 degrees in Fig. 2) is mounted on the drive shaft 5. The drive shaft 5 is coupled to a cam plate 8 via a coupling (not shown). Formed on that side of the cam plate 8 which faces the drive shaft 5 are non-uniform speed cams 8a equal in number to the cylinders of the diesel engine. A roller ring 9 is attached rotatable to that end of the drive shaft 5 which faces the cam plate 8, with a plurality of cam rollers 10 attached inside the roller ring 9 so as to face the non-uniform speed cams 8a. The cam plate 8 is urged by a spring 11 to be always in engagement with the cam rollers 10.

The plunger 12 for compressing the fuel is attached to the cam plate 8 to be rotatable together therewith. As the torque of the drive shaft 5 is transmitted via the coupling to the cam plate 8, the cam plate 8 and the plunger 12 reciprocate in the right and left directions in the diagram while rotating together therewith. The plunger 12 is inserted in a cylinder 14 formed in a pump housing 13, defining a high-pressure chamber 15 between the distal end face of the plunger 12 (right-hand end face in Fig. 2) and the bottom of the cylinder 14.

Intake grooves 16 and distribution ports 17, which are equal in number to the cylinders of the diesel engine, are formed on the outer surface of the plunger 12 at its distal end portion. In association with those intake grooves 16 and distribution ports 17, intake ports 19 and distribution passages 18, equal in number to the cylinders, are formed in the pump housing 13. A delivery valve 27 is disposed in a midway of each distribution passage 18. The delivery valve 27 serves to perform a so-called sharp-cut, which retracts fuel when fuel supply under pressure is completed to rapidly reduce the fuel pressure (residual pressure) in the injection pipe 4, thereby promptly terminating the injection, or to keep the residual pressure, or to prevent the counterflow of the fuel in the injection pipe 4. This delivery valve 27 is not provided with an Angleich mechanism which serves to control the residual pressure in the injection pipe 4.

When the feed pump 6 is driven with the rotation of the drive shaft 5, fuel from a fuel tank (not shown) is supplied via a fuel supply port 20 into a fuel chamber 21. In the induction stroke where the plunger 12 moves leftward (backward) in the diagram to reduce the pressure in the high-pressure chamber 15, one of the intake grooves 16 is connected to the associated intake port 19, supplying the fuel from the fuel chamber 21 to the high-pressure chamber 15. In the compression stroke where the plunger 12 moves rightward (forward) in the diagram to compress the high-pressure chamber 15, the compressed fuel is supplied under pressure to the injection valve 2 via the associated distribution passage 18, delivery valve 27 and injection pipe 4.

A fuel-overflow spill passage 22, which connects the high-pressure chamber 15 to the fuel chamber 21, is formed in the pump housing 13, with an electromagnetic spill valve 23 disposed in a midway of the passage 22. The electromagnetic spill valve 23 has a valve piece 25 open when a coil 24 is not excited, allowing the fuel in the high-pressure chamber 15 to flow into the fuel chamber 21. When the coil 24 is excited or supplied with a current, the valve piece 25 is closed to block the fuel flow from the high-pressure chamber 15 to the fuel chamber 21.

The electromagnetic spill valve 23 is therefore selectively closed and opened by controlling the time for energizing the electromagnetic spill valve 23. When the electromagnetic spill valve 23 is open during the compression stroke of the plunger 12, the pressure on the fuel in the high-pressure chamber 15 is reduced, stopping the fuel injection from the injection valve 2. Even if the plunger 12 reciprocates, therefore, the pressure of the fuel in the high-pressure chamber 15 will not rise during the period in which the electromagnetic spill valve 23 is open, disabling the fuel injection from the injection valve 2. That is, the amount of fuel injection from the injection valve 2 is adjusted by controlling the timings of closing and opening of the electromagnetic spill valve 23 during the reciprocal movement of the plunger 12.

A timer device 26 (developed 90 degrees in the diagram) for controlling the timing of fuel injection is provided below the pump housing 13. The timer device 26 controls the timing at which the non-uniform speed cam 8a engages with the cam roller 10, i.e., the reciprocation timing for the cam plate 8 and plunger 12, by adjusting the position of the roller ring 9 with respect to the rotational direction of the drive shaft 5.

In addition to the above-described basic structure of the injection pump 1, a characteristic C of the cam 8a representing a ratio of the cam speed V to the rotation angle θ is set as follows in this embodiment. This characteristic C is a factor which determines the fuel supply rate of the injection pump 1. As shown in Fig. 6, in the period where the cam angle θ lies between $\theta 0$ and $\theta 1$, as the cam 8a rotates, the cam speed V increases at a given ratio, and the cam speed V becomes constant ($V = V1$) between $\theta 1$ and $\theta 2$. In the period where the cam angle θ lies between $\theta 2$ and $\theta 3$, as the cam 8a rotates, the cam speed V increases at a given ratio, and reaches the peak at $\theta 3$. In the period where the cam angle θ lies between $\theta 3$ and $\theta 4$, as the cam 8a rotates, the cam speed V decreases at a given ratio. In this embodiment, the area in the cam characteristic C where the cam angle θ lies between $\theta1$ and $\theta2$ and the cam speed V is constant is a low-speed area L, and the area where the cam speed V is greater than V1 (the cam angle θ lying between $\theta2$ and $\theta3a$) is a high-speed area H.

A description will now be given of the injection valve 2 which atomizes fuel, supplied under high pressure from the injection pump 1, and injects that fuel in each high-combustion chamber.

As shown in Figs. 3 to 5, the injection valve 2 has a thin nozzle holder 28 extending long in the vertical direction. A spacer 29 and a nozzle body 30 are disposed, former on the latter, at the bottom portion of the nozzle holder 28. The spacer 29 and the nozzle body 30 are attached to the nozzle holder 28 by a retaining nut 31 that is fastened around the bottom portion of the nozzle holder 28.

An oil passage 32 is formed in the nozzle holder 28, the spacer 29 and the nozzle body 30. The upper end of the oil passage 32 is open to the upper end face of the nozzle holder 28. The lower end of the oil passage 32 is open to an oil retainer 33 at the bottom portion of the nozzle body 30. The oil retainer 33 is connected to the bottom of the nozzle body 30 via an injection hole 35. The injection hole 35 has a tapered surface 35a which becomes narrower in the downward direction. When high-pressure fuel is supplied to the injection valve 2 from the injection pump 1, the fuel passes the oil passage 32 and then the oil retainer 33 so that it can be injected outward through the injection hole 35.

A nozzle needle 36 for opening and closing the injection hole 35 is mounted in the nozzle body 30 and the spacer 29. The nozzle needle 36 comprises a body portion 37 and upper and lower shaft portions 38 and 39. The body portion 37 is a rod inserted slidable in the nozzle body 30. The lower end face of the body portion 37 faces the oil retainer 33, so that the pressure of the fuel in the oil retainer 33 acts on this lower end face in the direction to push the nozzle needle 36 upward.

The shaft portion 38 protruding upward from the body portion 37 is inserted slidable in the spacer 29. A sheet surface 39a, which comes in contact with or away from the tapered surface 35a, is formed in a middle portion of the shaft portion 39 protruding downward from the body portion 37. The shaft portion 39 has a pin portion 39b formed below the sheet surface 39a and slightly narrower than the injection hole 35 of the nozzle body 30. A tubular injection port 45, which allows fuel to pass through, is formed between the pin portion 39b and the nozzle body 30. In this embodiment, the pin portion 39b of the nozzle needle 36 is designed relatively long, and it constitutes a steady flow-rate portion to keep the area of the injection port 45 nearly constant even when the shift amount of the nozzle needle 36 changes.

The upper end portion of the shaft portion 38 protrudes upward through the spacer 29, and a pressure pin 40 is attached to this protruding portion. A first spring 42 is disposed, compressed, between this pressure pin 40 and a guide sleeve 41, which is incorporated at nearly the center of the nozzle holder 28. This first spring 42 always urges the nozzle needle 36 in the valve closing direction (downward in the diagram). When the sheet surface 39a comes in contact with the tapered surface 35a due to this urging force as shown in Fig. 4, the communication of the oil passage 32 with the injection port 45 is blocked, stopping fuel injection. At this time, the lower end of the pin portion 39b slightly sticks out downward from the injection hole 35, and the body portion 37 comes away downward from the spacer 29 by a length Lb.

At the time of fuel injection, the nozzle needle 36 rises to separate the sheet surface 39a upward from the tapered surface 35a. When the nozzle needle 36 shifts upward by the length Lb from the close state, the body portion 37 abuts on the spacer 29. At this time, the pin portion 39b comes inside the injection hole 35 as shown in Fig. 5.

A push rod 43 is inserted in the guide sleeve 41 in such a manner as to be movable up and down. The push rod 43 is located coaxial with respect to the nozzle needle 36. A second spring 44 is disposed, compressed, in the guide sleeve 41, so that the spring 44 always urges the push rod 43 downward.

With the nozzle needle 36 closed, the pressure pin 40 is spaced away downward from the push rod 43 by a length La (< Lb). Accordingly, the urging force of the second spring 44 will not be applied to the pressure pin 40 and the nozzle needle 36. When the nozzle needle 36 moves upward by the length La, the pressure pin 40 contacts the push rod 43. When the shift amount of the nozzle needle 36 exceeds the length La, the urging force of the second spring 44 is applied to the nozzle needle 36. In this embodiment, the pressure pin 40, guide sleeve 41, first spring 42, push rod 43 and second spring 44 constitute an injection-valve opening pressure adjusting mechanism.

In addition, in this embodiment, the fuel pressure necessary to lift the nozzle needle 36 in the closed state is a first injection-valve opening pressure P1, and the fuel pressure necessary to lift again the nozzle needle 36 which is in contact with the push rod 43 via the pressure pin 40, is a second injection-valve opening pressure P2. It is desirable that the first injection-valve opening pressure P1 be set to about 200 Kg/cm^2, for example. It is desirable that in order to obtain the necessary fuel injection period, the second injection-valve opening pressure P2 be set in accordance with the degree of the rise in fuel pressure which is associated with the portion of the cam characteristic where the cam speed is constant.

For the injection valve 2 with the above-described structure, the shift amount (lift amount L) of the nozzle needle 36 is determined in accordance with the pressure P of the fuel supplied from the injection pump 1. When the fuel pressure Pn which acts on the nozzle needle 36 in the oil retainer 33 is lower than the first injection-valve opening pressure P1, the sheet surface 39a is pressed against the tapered surface 35a. When the fuel pressure Pn becomes higher than the first injection-valve opening pressure P1, the nozzle needle 36 starts lifting, causing the sheet surface 39a to move away from the tapered surface 35a. This lift continues until the pressure pin 40 abuts on the push rod 43. After the contact, the lifting of the nozzle needle 36 stops in the period where the fuel pressure Pn is lower than the second injection-valve opening pressure P2. When the fuel pressure Pn further rises and comes higher than the second injection-valve opening pressure P2, the nozzle needle 36 moves upward again. This lift continues until the body portion 37 of the nozzle needle 36 contacts the spacer 29.

The injection valve 2 has a flow-rate characteristic as shown in Fig. 7. This flow-rate characteristic is about the same as the typical flow-rate characteristic of a throttle type injection valve belonging to pin type injection valves. The vertical axis in Fig. 7 indicates the flow rate Q of compressed air that is injected through the injection port 45 when compressed air is supplied, in place of fuel, to the oil passage 32 of the injection valve 2.

It is apparent from Fig. 7 that in the period where the lift amount L lies between 0 and L1, the flow rate Q increases nearly in proportion to an increase in lift amount L. In the period where the lift amount L lies between L1 and L2, the flow rate Q becomes a nearly constant, low flow rate (Q1) irrespective of the lift amount L. In the period where the lift amount L is greater than L2, the flow rate Q increases nearly in proportion to an increase in lift amount L. The period between L1 and L2 is when the relatively long pin portion 39b of the nozzle needle 36 is passing the injection hole 35. During this period, the area of the injection port 45 is kept nearly constant and the flow rate Q becomes almost constant. As is apparent from the above, the flow rate Q is suppressed to a low value (Q1) due to the action of the throttle in the period between L1 and L2.

In this embodiment, the length La is set in such a way that the nozzle needle 36 is kept at the aforementioned valve-open position (where the pressure pin 40 abuts on the push rod 43) in the steady flow-rate area (lift amount L lying between L1 and L2) of the flow-rate characteristic where the flow rate Q becomes a constant, low flow rate Q1, i.e., that $L1 \leqq La \leqq L2$ is satisfied. This setting is employed to prevent a variation in the flow rate Q even when the lift amount L of the nozzle needle 36 fluctuates in the vicinity of La.

In addition, the area of the cam characteristic which concerns with the movement of the plunger 12 at the time of fuel injection is set as follows in this embodiment. This setting reflects such a consideration that the injection rate is determined mainly by the cam speed V of the injection pump 1, the diameter of the plunger 12 and the flow-rate characteristic of the injection valve 2. The period in which the plunger 12 is moved in the low-speed area of the cam characteristic is set to at least partially overlap the period in which the nozzle needle 36 is kept at the predetermined valve-open position in the steady flow-rate area of the flow-rate characteristic. More specifically, the pressure pin 40 abuts on the push rod 43 and the flow rate Q of the fuel from the injection port 45 is small and is kept at a constant value (Q1) in at least some portion of the period ($\theta 1$ to $\theta 2$) of the cam angle θ where the cam speed V becomes constant (V1) in Fig. 6.

Further, the first injection-valve opening pressure P1 of the injection valve 2 is set to meet the following condition according to this embodiment. The condition is such that in the fuel compression stroke, when the engaging portion of the non-uniform speed cam 8a with the cam roller 10 comes to where the cam speed V becomes constant from the increasing state (where the cam angle θ is $\theta 1$ in Fig. 6), the fuel pressure Pn applied to the nozzle needle 36 exceeds the first injection-valve opening pressure P1, thus opening the nozzle needle 36.

The function and advantages of the thus constituted embodiment will now be discussed with reference to the timing chart given in Fig. 1. In the characteristic of the fuel pressure in Fig. 1, the broken line indicates the fuel pressure Pp at the outlet portion of the injection pump 1 (directly downstream of the delivery valve 27), while the solid line indicates the fuel pressure Pn at the oil retainer 33 of the injection valve 2. Here, the injection pipe 4 about 600 mm long is disposed between the injection pump 1 and the injection valve 2. Accordingly, the pressure of fuel supplied under pressure from the injection pump 1 reaches the oil retainer 33 with a slight delay or a slight phase lag in actual measurement. Therefore, there is a time lag between both pressures Pp and Pn as shown in Fig. 1.

When the drive shaft 5 of the injection pump 1 rotates on the driving power from the diesel engine, the torque is transmitted via the coupling to the cam plate 8. The torque transmission causes the cam plate 8 and plunger 12 to reciprocate horizontally in Fig. 2 while rotating.

In the induction stroke where the plunger 12 moves leftward in Fig. 2, one of the intake grooves 16 faces the associated intake port 19, allowing the fuel from the fuel chamber 21 to be taken into the high-pressure chamber 15 via that intake groove 16. Thereafter, the connection between the intake port 19 and the intake groove 16 is blocked, and the distribution port 17 faces one of the distribution passages 18.

As the plunger 12 further rotates, the non-uniform speed cam 8a rides over the cam roller 10 to move the plunger 12 rightward in Fig. 2 and the compression stroke starts (timing t1 in Fig. 1). When the engaging portion of the non-uniform speed cam 8a with the cam roller 10 comes to where the cam angle θ lies between $\theta 0$ and $\theta 1$, the moving speed of the plunger 12 increases at a constant ratio as time elapses. This speed increase gradually reduces the volume of the high-pressure chamber 15, compressing the fuel in the high-pressure chamber 15. The compressed fuel is supplied to the injection valve 2 from the distribution port 17 via the associated distribution passage 18, delivery valve 27 and injection pipe 4 (between timings t1 and t2). In this period, therefore, the fuel pressures Pp and Pn both rise as the time elapses.

At this time, since the fuel pressure Pn in the oil retainer 33 is lower than the first injection-valve opening pressure P1, the sheet surface 39a is pressed against the tapered surface 35a, closing the nozzle needle 36. As a result, fuel will not be injected from the injection valve 2. And the lift amount L and the injection rate α are both "0".

When the engaging portion of the non-uniform speed cam 8a with the cam roller 10 comes to where the cam angle θ is $\theta 1$ (timing t2) and a time Δt corresponding to a phase lag from that timing, the fuel pressure Pn in the oil retainer 33 reaches the first injection-valve opening pressure P1 (timing t3). This lifts up the nozzle needle 36 in the injection valve 2 against the force of the first spring 42 to separate the sheet surface 39a from the tapered surface 35a, causing fuel injection to start. Consequently, the lift amount L and injection rate α start increasing.

During a period from the start of the fuel injection to the point where the engaging portion of the non-uniform speed cam 8a with the cam roller 10 comes to where the cam angle θ becomes $\theta 2$, the cam speed V is kept constant (V1). That is, the moving speed of the plunger 12 becomes constant and the fuel supply speed under pressure becomes constant. At this time, however, the injection port 45 of the injection valve 2 is narrowed by the nozzle needle 36. Even though a constant amount of fuel is supplied from the injection pump 1 to the injection valve 2 under pressure, therefore, only a less amount of fuel than what has been supplied will be injected. Consequently, the fuel pressure Pn in the oil retainer 33 gradually rises (timings t3 to t5).

Suppose that the cam characteristic in Fig. 6 has neither an area where the cam speed V is kept constant (V1) nor an area where the cam speed V varies gently. Then, after exceeding the first injection-valve opening pressure P1, the fuel pressure Pn in the oil retainer 33 swiftly rises to become the second injection-valve opening pressure P2. In this case, therefore, while the fuel can be atomized by the high-pressure injection, the period of a low injection rate α (t3 to t6) is shortened, eventually increasing the amount of nitrogen oxides and smoke.

As the injection valve 2 is openable in two stages depending on two pressures (injection-valve opening pressures P1 and P2), the nozzle needle 36 moves up in the period from the point when the fuel pressure Pn in the oil retainer 33 rises above the first injection-valve opening pressure P1 to the point when the pressure pin 40 abuts on the push rod 43 (when the pin 40 reaches the predetermined injection-valve opening pressure). When the nozzle needle 36 rises by the length La and the pressure pin 40 abuts on the push rod 43 (timing t3a), the urging force of the second spring 44 will act on the nozzle needle 36 thereafter. In the period where the fuel pressure Pn is lower than the second injection-valve opening pressure P2, therefore, the lift amount L of the nozzle needle 36 is kept constant (La).

According to this embodiment, the length La is set in the period (L1 to L2) where the flow rate Q becomes constant (Q1) as shown in Fig. 7. What is more, the flow rate Q1 at this time is low. In addition, the cam speed V is a low constant value (V1) at this time, so that the speed of fuel supplying from the injection pump 1 to the injection valve 2 under pressure is suppressed. In the period between the timings t3a to t5, therefore, high-pressure injection is carried out with the injection rate α suppressed. At this time, even if the lift amount L of the nozzle needle 36 varies around La, a variation of the flow rate Q is small, thus ensuring a stable flow rate Q.

When the engaging portion of the non-uniform speed cam 8a with the cam roller 10 comes to where the cam angle θ is $\theta 2$ (timing t4) and a time Δt corresponding to a phase lag from that timing passes (timing t5), the cam speed V is greater than the low value V1. The fuel pressure Pn in the oil retainer 33 therefore keeps rising. At this time, the lift amount L of the nozzle needle 36 is La and the injection rate α is kept low.

When the fuel pressure Pn in the oil retainer 33 further rises and becomes the second injection-valve opening pressure P2 (timing t6), the nozzle needle 36 rises again against the forces of both springs 42 and 44. When the lift amount L increases above La, the flow rate Q rises nearly in proportion to that increase. Accordingly, the injection rate α rises rapidly. The lifting of the nozzle needle 36 stops when the body portion 37 contacts the spacer 29.

When the electromagnetic spill valve 23 is opened during the compression stroke of the plunger 12 thereafter, the pressure of the fuel in the high-pressure chamber 15 is reduced, stopping the fuel injection from the injection valve 2. In other words, even if the plunger 12 reciprocates, the fuel pressure in the high-pressure chamber 15 does not rise and no fuel will be injected from the injection valve 2 while the electromagnetic spill valve 23 is open.

If the duration that the period in which the plunger 12 is moving in the low-speed area of the non-uniform speed cam 8a overlaps the period in which the nozzle needle 36 is held at the predetermined injection-valve opening pressure to ensure a constant flow rate Q is set longer, the injection rate a can be kept low for a longer period of time.

According to this embodiment, as described above, the cam speed V is kept low (V1) and the fuel supply speed by the injection pump 1 is suppressed during the overlapping period. And the period in which the injection valve 2 keeps injecting fuel at a low flow rate becomes longer. The injection rate α thus becomes low in this overlapping period. The fuel pressure Pn applied on the nozzle needle 36 however is equal to or higher than the first injection-valve opening pressure P1 and is equal to or lower than the second injection-valve opening pressure P2. If those first and second injection-valve opening pressures P1 and P2 are both set to high values, it is possible to suppress a rapid rise of the nozzle needle 36 after the fuel injection starts, and to accomplish high-pressure fuel injection while securing the low injection-rate period.

This high-pressure injection will prevent the burning speed of fuel mixture from decreasing at the time EGR is performed. The provision of the low injection-rate period will prevent an increase in the amount of hydrocarbon (HC) or the amount of smoke, which would inevitably occur when fuel is simply injected under high pressure. In other words, under the operational conditions requiring the atomization of fuel, such as at the time of performing EGR, the generation of smoke can be suppressed while sufficiently reducing the amount of EGR-originated nitrogen oxides NO_x.

Further, since the plunger 12 moves in the most part of the high-speed area (high fuel-supply rate portion) of the cam characteristic, fuel can be injected under high pressure and at a high injection rate when a large amount of fuel should be injected, such as at the time of high-load running of the engine. Such fuel injection cancels out an excessive extension of the injection period that is resulted from widening the low injection-rate period at the beginning of injection, so that a drop in output power can be suppressed.

Further, the nozzle needle 36 is held at the predetermined injection-valve opening pressure in the steady flow-rate area by the shaft portion 39 of the injection valve 2. Even if the nozzle needle 36 changes its position around the predetermined injection-valve opening pressure, therefore, a variation in the rate of fuel injection can be suppressed to secure a steady flow rate Q.

In addition, as the injection valve 2 used in this embodiment has relatively high injection-valve opening pressures P1 and P2, the general fuel pressure Pn applied to the nozzle needle 36 becomes high, increasing the speed of the fuel passing the injection port 45. This increase in flow rate will prevent the accumulation of deposit which is one of the problems of the prior art.

When the low injection-rate period is short and the injection rate at the beginning of injection rises (when a large amount of fuel is injected at the beginning of injection), the fuel is burned at a time, rapidly increasing the pressure in the combustion chamber so that the burning sound increases. As the low injection-rate period is sufficient secured in this embodiment, however, a small amount of fuel injected at the beginning of injection is burned. Even when the low injection-rate period ends and the injection rate rises, the pressure in the combustion chamber rises only slowly, reducing the burning sound.

This embodiment is advantageous over the prior art in the following points.

First, the pressure of fuel from the injection pump with the plunger moving in the low-speed area of the cam characteristic is set to lie between the first and second injection-valve opening pressures according to the prior art. The ranges for the first and second injection-valve opening pressures are therefore greatly restricted. This will be explained with reference to Fig. 10. Suppose that the fuel pressure in the low-speed area varies between Pa and Pb. Then, the first injection-valve opening pressure P1 should be set lower than the pressure Pa, while the second injection-valve opening pressure P2 should be set higher than the pressure Pb.

Whereas in this embodiment, the period in which the plunger 12 moves in the low-speed area of the cam characteristic and the steady flow-rate area of the flow-rate characteristic of the injection valve 2 have only to overlap each other partially. In other words, it is sufficient in this embodiment that the first injection-valve opening pressure P1 is lower than Pn (t5) which reflects a phase lag Δt with respect to the fuel pressure Pn at the end of the constant cam speed period (Pn (t4) in Fig. 1). Further, the second injection-valve opening pressure P2 has only to be set higher than Pn (t3) which reflects a phase lag Δt with respect to the fuel pressure Pn at the beginning of the constant cam speed period (Pn (t2) in Fig. 1). Therefore, the ranges for the first injection-valve opening pressure P1 and second injection-valve opening pressure P2 are larger than those in the prior art. It is therefore possible to better cope with various variations in the injection pump 1 and injection valve 2 at the time of their production, such as a variation in the shape of the non-uniform speed cam 8a, variations in both injection-valve opening pressures P1 and P2 of the injection valve 2, and a variation in flow rate Q.

Secondly, the amount of fuel leak in the high-pressure chamber 15 of the injection pump 1 in a typical diesel engine differs according to the engine speed. More specifically, the amount of fuel leak is large at a low engine speed and this amount decreases as the engine speed increases. At a low engine speed, therefore, injection will not start unless in the back portion of the cam characteristic. At a high engine speed, injection will start in the front portion of the cam characteristic.

To satisfy the aforementioned pressure setting condition in the prior art, the fuel pressure should reach the first injection-valve opening pressure before the steady cam-speed period starts, and the fuel pressure should reach the second injection-valve opening pressure after the steady cam-speed period ends. This requires a separate mechanism to adjust the fuel pressure. The prior art therefore has an Angleich mechanism provided in the delivery valve to retract extra fuel from the injection pipe 4 when the engine speed is high, thereby reducing the residual pressure.

According to this embodiment, however, the plunger 12 has only to be movable at the portion in the cam characteristic where the cam speed is constant, during the period from the point at which the fuel pressure Pn reaches the first injection-valve opening pressure P1 to the point at which the fuel pressure Pn reaches the second injection-valve opening pressure P2. This embodiment will not therefore be subjected to the aforementioned restriction, and can eliminate the need for a separate mechanism for adjusting the pressure, such as the Angleich mechanism.

Although only one embodiment of the present invention has been described herein, it should be apparent to those skilled in the art that the present invention may be embodied in many other specific forms without departing from the spirit or scope of the invention. Particularly, it should be understood that this invention may be worked out in the following manners.

(1) Although the fuel pressure Pn applied to the nozzle needle 36 is set to become the first injection-valve opening pressure P1 at the transitional point where the increasing cam speed V becomes constant (where the cam angle θ is $\theta 1$ in Fig. 6) in this embodiment, the pressure Pn should not necessarily match with the pressure P1. The period in which the plunger 12 moves in the low-speed area of the cam characteristic and the steady flow-rate area of the flow-rate characteristic of the injection valve 2 have only to overlap each other partially.

(2) Although the steady cam-speed area ($\theta 0$ to $\theta 2$) is set in the cam characteristic in the above-described embodiment, this area may be so set that the cam speed V increases at a given ratio as shown in Fig. 8 or that the cam speed V decreases at a given ratio as shown in Fig. 9. The former setting is suitable when the injection-valve opening pressures P1 and P2 are set to relatively low values, while the latter setting is suitable when the injection-valve opening pressures P1 and P2 are set to relatively high values so that the fuel pressure Pn slowly rises after the pressure pin 40 abuts on the push rod 43.

Therefore, the present examples and embodiment are to be considered as illustrative and not restrictive and the invention is not to be limited to the details given herein, but may be modified within the scope of the appended claims.

A low-speed area where a cam speed is low is provided in the cam characteristic of non-uniform speed cams of an injection pump for moving a plunger at a low speed, and a high-speed area for moving the plunger at a high speed is provided in the back portion of the cam characteristic. The nozzle needle of an injection valve is opened at a predetermined first injection-valve opening pressure, and is held at a predetermined valve-open position until fuel pressure reaches a second injection-valve opening pressure set higher than the first injection-valve opening pressure. The injection valve is provided with a steady flow-rate portion for keeping the area of an injection port nearly constant even when the shift amount of the nozzle needle varies, and the nozzle needle is held at the predetermined valve-open position in this steady flow-rate area. The period in which the plunger moves in the low-speed area of the cam characteristic and the period in which the nozzle needle is held at the predetermined valve-open position in the steady flow-rate area overlap each other at least partially. Accordingly, the low injection-rate period is secured at the beginning of fuel injection, thus ensuring high-pressure injection.

Claims

1. A fuel injection apparatus comprising:

an injection pump (1) having non-uniform speed cams (8a), said cams (8a) rotataing for reciprocatingly moving a plunger (12) to press fuel to be supplied to injection valves (2), said valve (2) accommodating a nozzle needle (36) moved by fuel pressure to selectively close and open an injection hole (35), an injection port (45) injecting the fuel when the injection hole (35) is open;

said plunger (12) being moved at a variable speed in accordance with a cam characteristic (C) representing a ratio of a cam speed (V) to a rotation angle (θ) of the cam (8a), said cam characteristic (C) having a low-speed area (L) where the cam speed (V) is low, so as to adjust supply speed of the pressed fuel; and

said nozzle needle (36) being moved to alter a size of a fuel passage of said injection port (45) to adjust an injection amount and pressure of the pressed fuel; characterized in that

each injection valve (2) includes means for holding said nozzle needle (36) against the fuel pressure from the injection pump (1) at a position in the injection port (45) where a fuel passage becomes constricted for reducing a fuel injection amount at least during a part of movement of the plunger (12) according to the low-speed area of the cam characteristic.

2. A fuel injection apparatus according to Claim 1, wherein said injection port (45) is formed between an outer surface of the nozzle needle (36) inserted in the injection hole (35) and an inner wall of the injection hole (35); and wherein the size of the fuel passage in the injection port (45) varies in accordance with movement of the nozzle needle (36).

3. A fuel injection apparatus according to Claim 1, wherein said mechanism includes:

a first spring (42) for urging said nozzle needle (36) to a closed position, said first spring (42) holding the nozzle needle (36) at the closed position when the fuel pressure from said injection pump (1) is smaller than a first predetermined pressure (P1), and allowing the nozzle needle (36) to move to opened position when the fuel pressure is greater than the first predetermined pressure (P1);

a push rod (43) abutting on the nozzle needle (36) when the nozzle needle (36) is open at the predetermined position; and

a second spring (44) for urging the push rod (43) to the closed position of the nozzle needle (36), holding the nozzle needle (36) open at the predetermined position when the nozzle needle (36) is open at the predetermined position and the fuel pressure is smaller than a second predetermined pressure (P2) which is greater than the first predetermined pressure (P1), and allowing the nozzle needle (36) to move further to the open position when the fuel pressure is greater than the second predermined pressure (P2); and

wherein the nozzle needle (36) is held open at a predetermined position during a period for restricting the fuel injection amount.

4. A fuel injection apparatus according to Claim 2 or 3, wherein each injection valve (2) includes means (39a) for keeping the fuel passage of the injection port (45) substantially constant even when the nozzle needle (36) moves, whereby said portion controls the fuel injection amount to be substantially constant during the fuel injection amount restricting period.

5. A fuel injection apparatus according to Claim 4, wherein said keeping means includes an elongated pin (39b) formed in the nozzle needle (36) and having a uniform cross-sectional shape, and the size of the fuel passage of the injection port (45) is kept substantially constant when the pin (39b) is positioned in the injection hole (35).

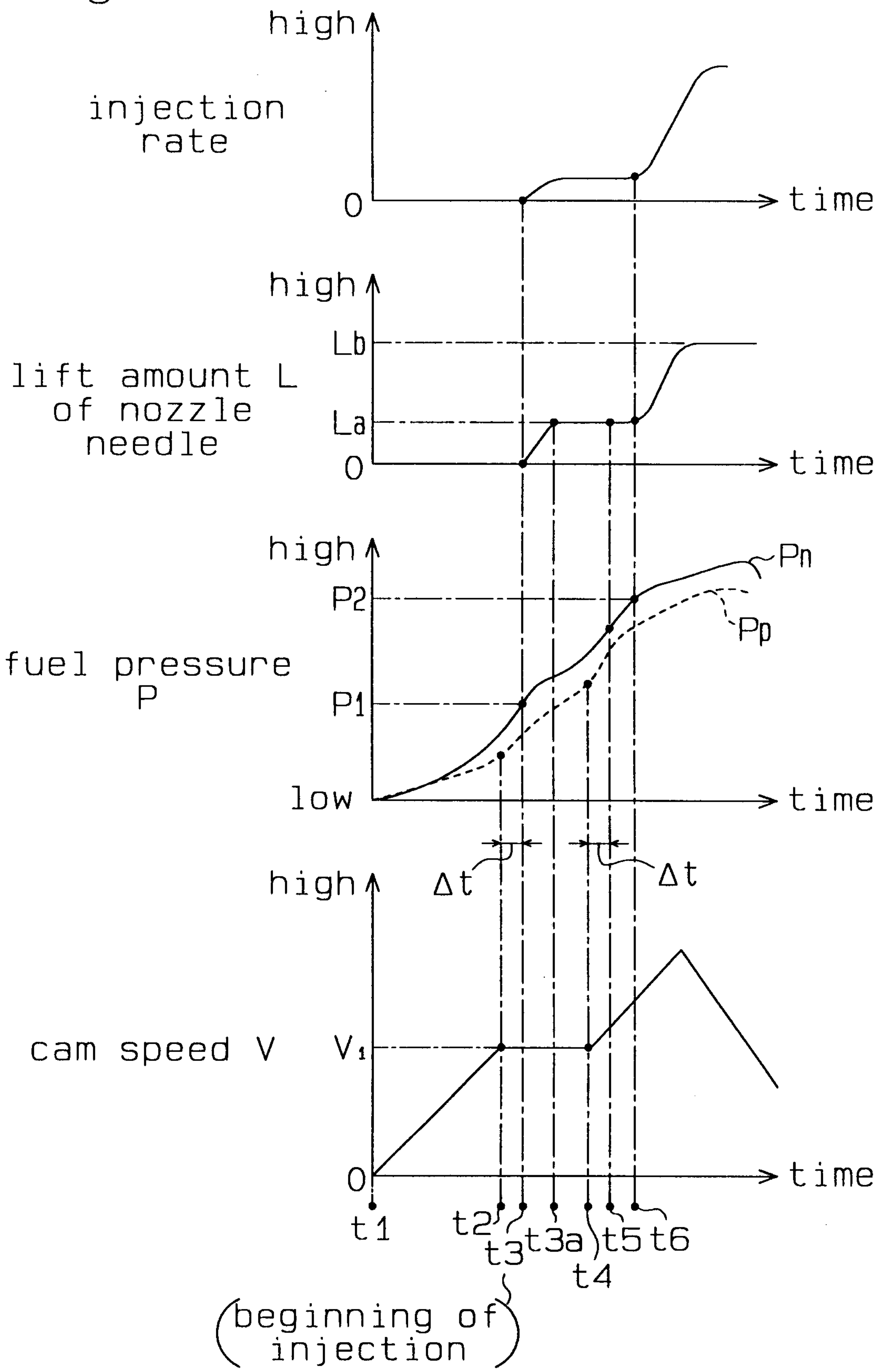
6. A fuel injection apparatus according to Claim 1, wherein the cam speed (V) in the low-speed area is constant regardless of the angle (θ) of the cams (8a).

7. A fuel injection apparatus according to Claim 1, wherein the cam speed (V) in the low-speed area slowly changes in accordance with the rotation of the cams (8a).

8. A fuel injection apparatus according to Claim 1, wherein a high-speed area with a higher cam speed (V) than that of said low-speed area is provided in a back portion of said cam characteristic, and said high-speed area includes two portions where said cam speed (V) increases and decreases at a given ratio in accordace with the rotation of the cams (8a), respectively.

9. A fuel injection apparatus according to Claim 1, wherein said low speed area follws an area where said cam speed (V) increases at a given ratio in accordance with the rotation of the cams (8a); and wherein the nozzle needle (36) is arraged to open at a transitional point between the cam speed increasing area to the low-speed area by a fuel pressure applied to the nozzle needle (36) set to exceed said first predetermined pressure (P1).

Fig. 1


high
injection
rate
0
time
high
Lb
lift amount L
of nozzle
needle
La
0
time
high
Pn
P2
Pp
fuel pressure
P
P1
low
time
high
Δt
Δt
cam speed V
V₁
0
time
t1
t2
t3
t3a
t4
t5
t6
(beginning of injection)

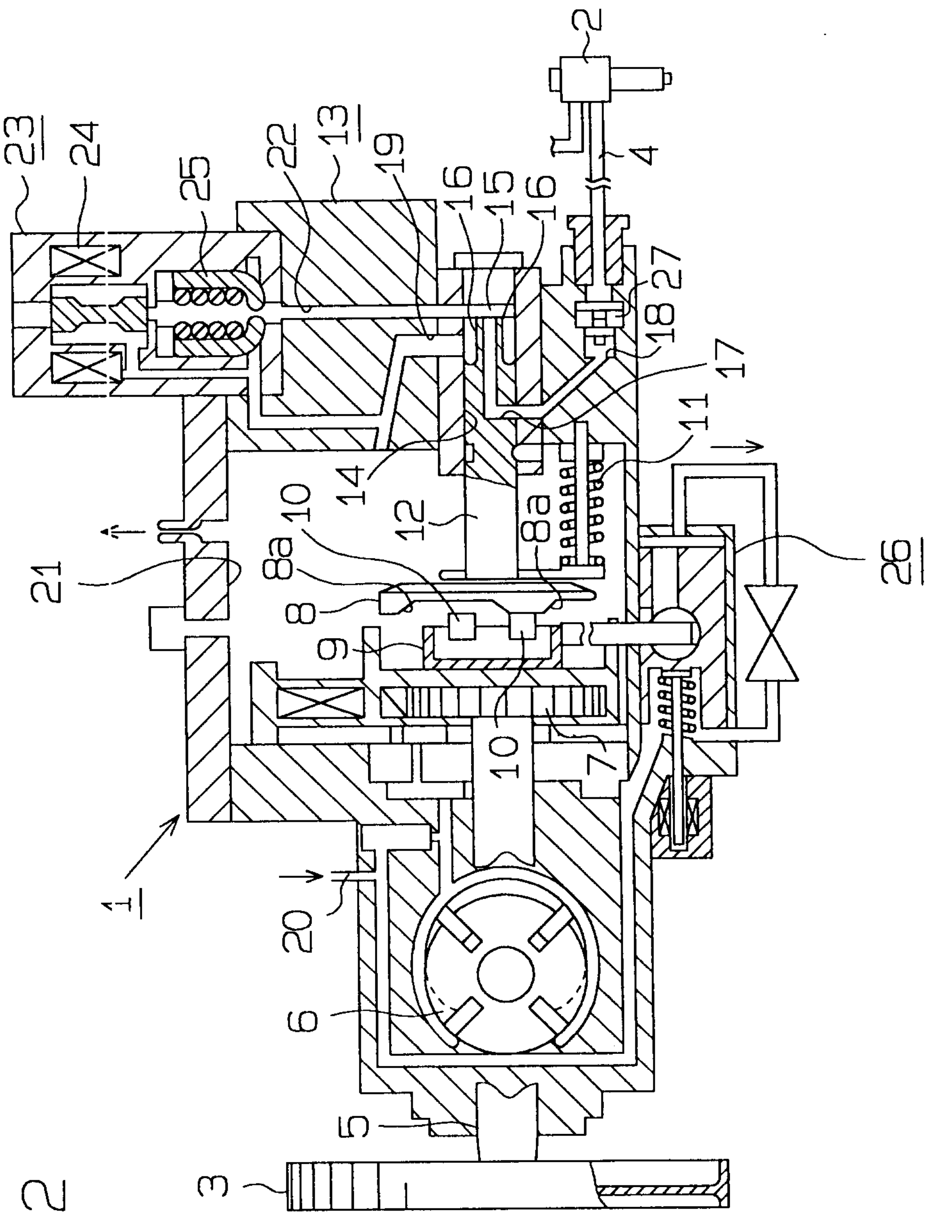


Fig. 2

Fig. 3

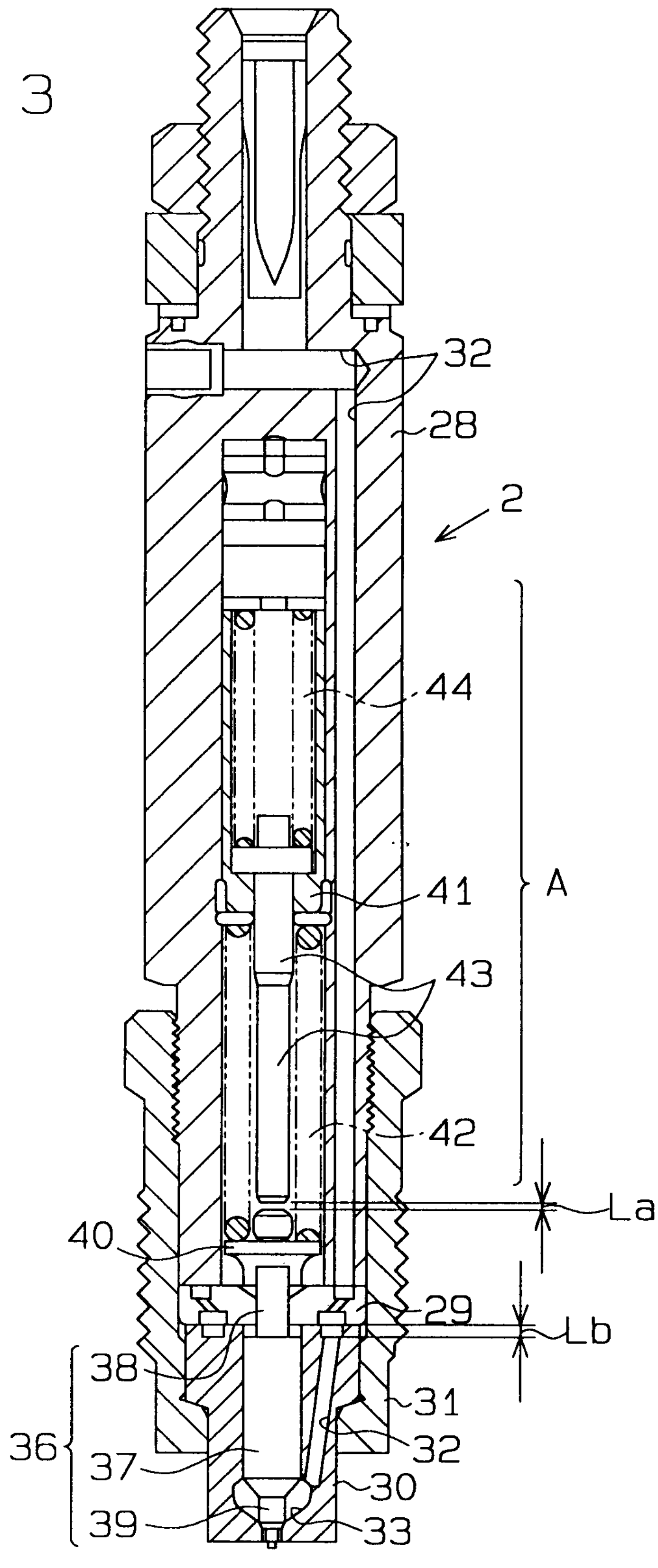


Fig.4

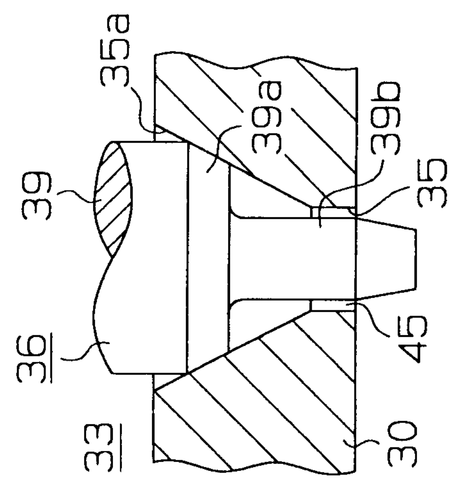


Fig.5

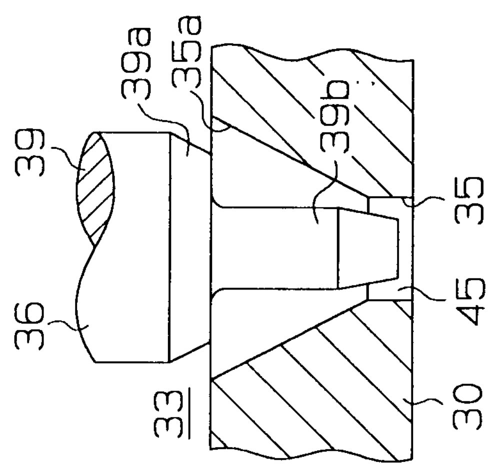


Fig.6

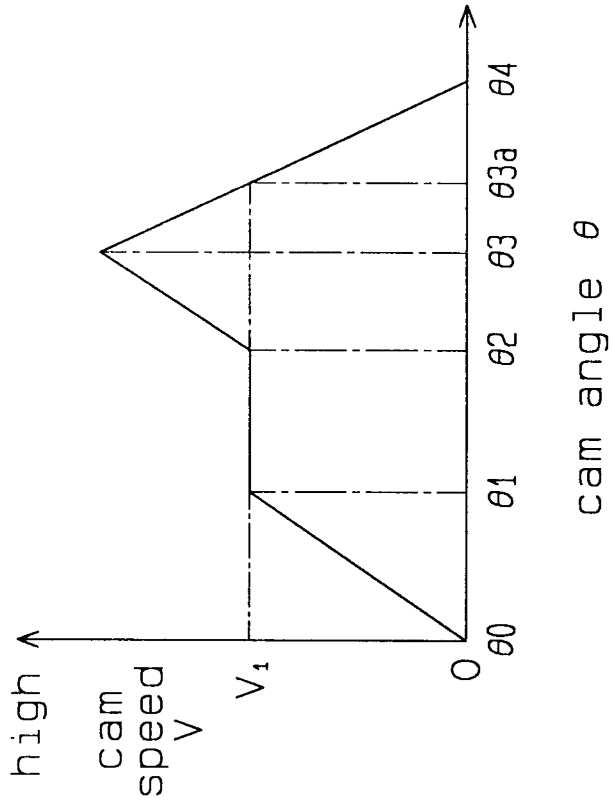


Fig. 7

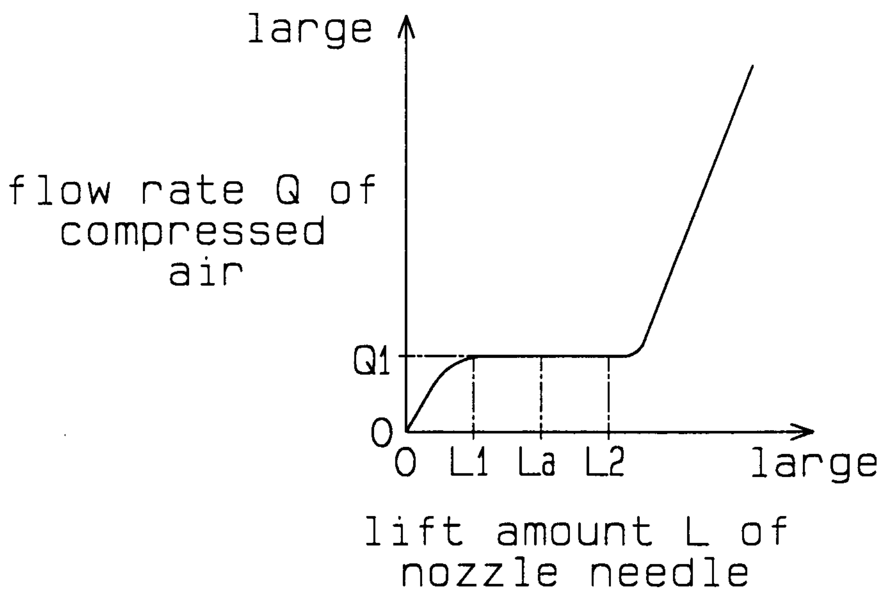
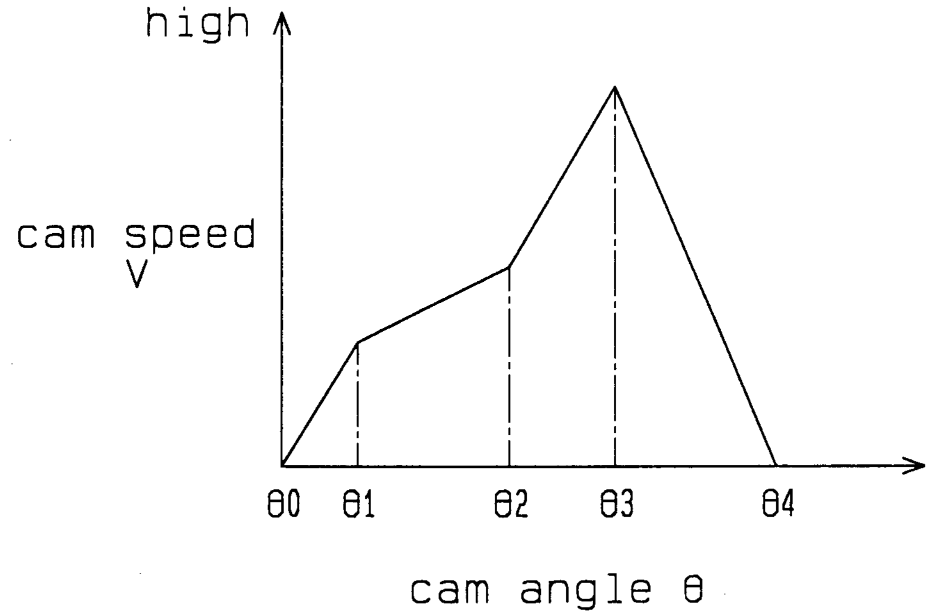

large
flow rate Q of compressed air
Q1
O
O
L1
La
L2
large
lift amount L of nozzle needle

Fig. 8


high
cam speed V
θ0
θ1
θ2
θ3
θ4
cam angle θ

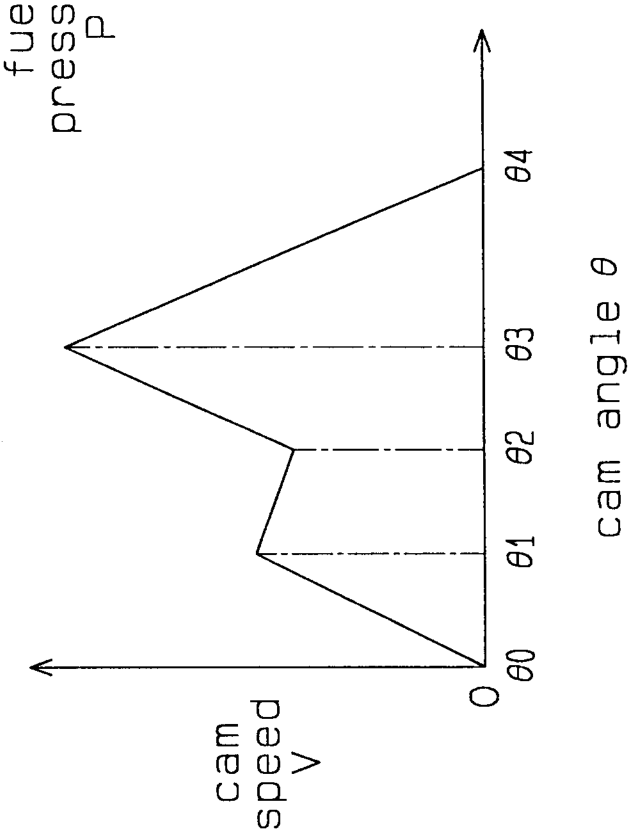

cam speed V
0
θ0
θ1
θ2
θ3
θ4
cam angle θ

Fig. 9

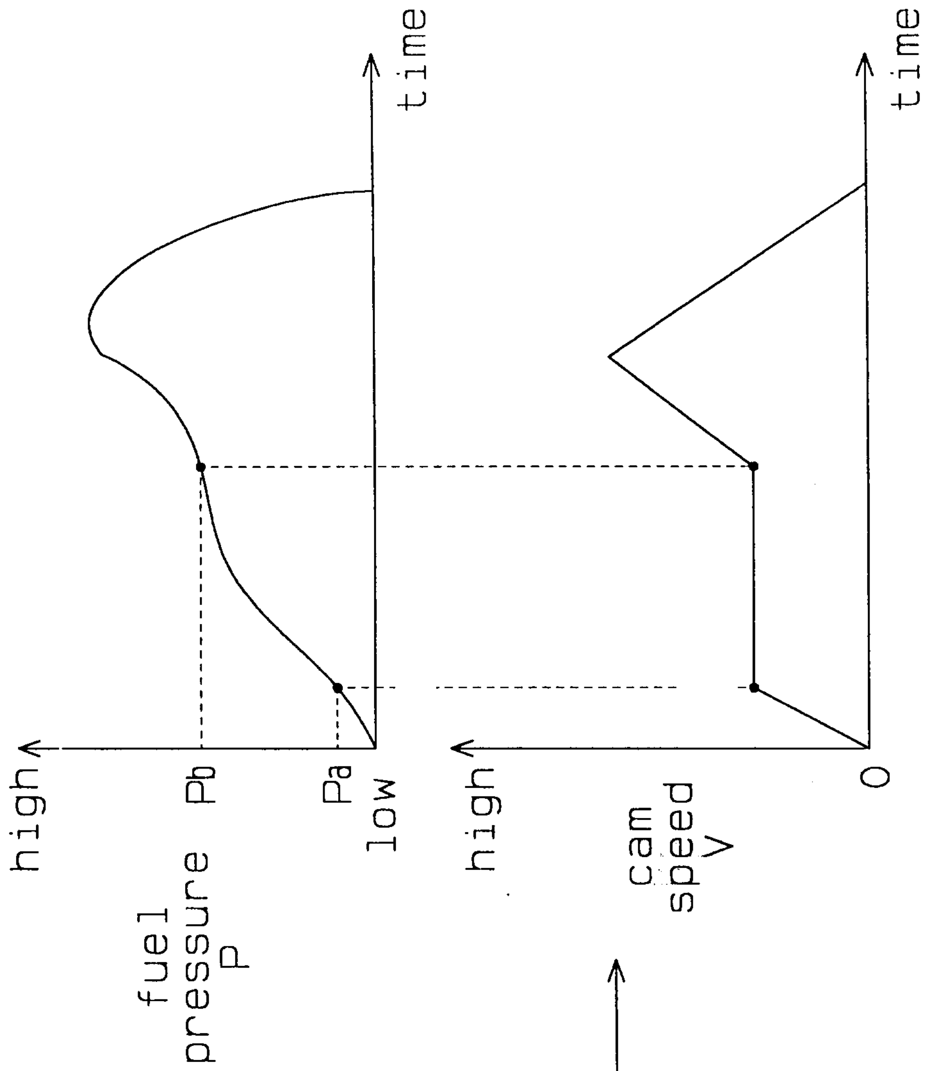

high
fuel pressure P
Pb
Pa
low
time
high
cam speed V
0
time

Fig. 10

European Patent Office

EUROPEAN SEARCH REPORT

Application Number
EP 94 10 3689

DOCUMENTS CONSIDERED TO BE RELEVANT

Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (Int.Cl.5)
A	PATENT ABSTRACTS OF JAPAN vol. 9, no. 25 (M-355) (1748) 2 February 1985 & JP-A-59 170 463 (NIPPON DENSO) 26 September 1984 * abstract * ---	1,3	F02M61/06 F02M59/10 F02M45/12
D,A	PATENT ABSTRACTS OF JAPAN vol. 12, no. 179 (M-701) 26 May 1988 & JP-A-62 288 364 (DIESEL KIKI) 15 December 1987 * abstract * ---	1	
A	CH-A-350 835 (FRIEDMANN & MAIER) * page 2, line 75 - page 4, line 16; figures * ---	1-5	
A	PATENT ABSTRACTS OF JAPAN vol. 11, no. 247 (M-615) 12 August 1987 & JP-A-62 055 454 (DIESEL KIKI) 11 March 1987 * abstract * ---	1	TECHNICAL FIELDS SEARCHED (Int.Cl.5)
A	FR-A-887 679 (BOSCH) -----		F02M

The present search report has been drawn up for all claims

Place of search	Date of completion of the search	Examiner
THE HAGUE	16 June 1994	Sideris, M

CATEGORY OF CITED DOCUMENTS

X : particularly relevant if taken alone
Y : particularly relevant if combined with another document of the same category
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