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(54) Fuel injection pump for diesel engine

Brennstoffeinspritzpumpe für einen Dieselmotor

Pompe d'injection de combustible d'un moteur diesel

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Description

TECHNICAL FIELD

The present invention relates generally to a fuel injection pump for injecting a fuel supplied to a diesel engine. More particularly, the present invention relates to a distributor type fuel injection pump in which the fuel injection rate varies depending on the operational state of the engine.

RELATED BACKGROUND ART

Conventionally, diesel engines have injection nozzles for injecting fuel into combustion chambers and a fuel injection pump for pumping the fuel to the injection nozzles. The injection nozzles are fixed to cylinder heads, and the fuel injection pump is driven under the power of the engine.

Japanese Unexamined Patent Publication No. Hei 2-298635 discloses an example of fuel injection unit for automotive diesel engines. This fuel injection unit has a distributor type fuel injection pump and fuel injection nozzles. The injection pump contains a plunger disposed in a cylinder. The fuel dwelling in a fuel chamber of the pump is sucked into a pressure chamber when the plunger reciprocates within the cylinder with the operation of the engine, and the thus sucked fuel is pressurized and distributed to a plurality of fuel lines through a plurality of distribution ports and the like. The thus distributed fuel is injected into combustion chambers through the injection nozzles contained in the engine. The injection pump has a drive mechanism containing a cam plate and a roller ring. This mechanism reciprocates the plunger. The cam plate has a cam face provided with cam peaks having nonuniform speed characteristics. The cam plate and the plunger rotate integrally, and the cam plate is urged by a spring toward the roller ring. The number of cam peaks on the cam face corresponds to the number of cylinders or the number of injection nozzles.

The graph in Fig. 19 shows cam characteristics of the cam face. This graph shows change in the cam lift vs. cam rotation angle and a relationship between cam speed and cam rotation angle. As the graph demonstrates, the greater the cam rotation angle, the greater the cam lift. However, since the cam face has nonuniform speed cam characteristics, the cam speed also changes with the change in the cam rotation angle. In other words, the cam speed changes depending on the using zone on the cam face.

The cam face has nonuniform speed cam characteristics such that the rate of injection in one injection from an injection nozzle (time rate of fuel to be injected from one injection nozzle) can be changed. Accordingly, on this cam face, the cam speed is increased at a constant rate within the cam rotation angle range of t_0 to t_1 (initial stage) and at a constant rate lower than that of

the initial stage within the cam angle range of t_1 to t_2 (first stage), as shown in Fig. 19. Within the cam rotation angle range of t_2 to t_3 (middle stage), the cam speed is increased at a constant rate to the maximum value, and within the cam rotation angle range of t_3 and thereafter (latter stage), the cam speed is reduced from the maximum value at a constant rate. In each stage, the cam speed changes linearly. The cam speed is a factor decisive of the speed of moving the plunger.

The engine in which such type of injection pump is employed generally has 4 or 6 cylinders. However, it is also possible to employ the injection pump in an engine having more than 6 cylinders. In this case, since the space on the cam face is limited, the portion on the cam face corresponding to one injection per cylinder in accordance with the cam characteristics is made narrower as the number of cylinders increases. Accordingly, it is not easy to secure the nonuniform speed cam characteristics on the cam face. Since the cam speed changes linearly with respect to the cam rotation angle in the respective stages expressing the cam characteristics, the cam speed is substantially discontinuous at each boundary between the stages. Since the cam face has recesses, it is difficult for the cam face to follow the roller ring and it is likely to jump, causing wear of the cam face. Further, in order to increase injection pressure, the cam face must have cam characteristics such that the cam speed may be linearly changed to the maximum value in the middle stage. However, in such a construction, the fuel pressure in the high pressure chamber of the injection pump increases to apply reactive high pressure hydraulic force to the cam face, and thus durability and reliability deteriorate.

In other words the reference JP 2-298 635 discloses a fuel injection pump having a cam plate with a predetermined characteristic curve including a first region representing the speed of the plunger increasing linearly at a predetermined rate from zero, a second region representing the speed of the plunger increasingly accelerated up to a maximum value along two respective linear curves, and a third region representing the speed of the plunger decreasing from the maximum value.

Especially the second region is represented by two intersecting linear curves, wherein the intersecting point is mathematically defined as a sharply bent point.

DISCLOSURE OF THE INVENTION

Accordingly, it is a primary objective of the present invention to provide a fuel injection pump for a diesel engine,

which increases fuel injection pressure in accordance with the cam characteristics of a cam plate with the lowest possible maximum cam speed, and which reduces the pressure applied to the cam face.

Further, it is another objective of the present invention to provide a fuel injection pump for a diesel engine which has cam characteristics for performing one effec-

tive fuel injection per cylinder even when the pump is employed in an engine having a number of cylinders, e. g., six cylinders, and which can inject fuel at a low injection pressure when the amount of fuel to be injected is small and at a high injection pressure when the amount of fuel to be injected is large.

To achieve the foregoing and other objects and in accordance with the purpose of the present invention, a fuel injection pump used in a Diesel engine is provided. The pump is arranged to distribute fuel to a plurality of fuel injection nozzles provided in said engine. Each of the injection nozzles are arranged to open by the fuel pressurized up to a predetermined value to inject the fuel therefrom. The injection pump includes a cam plate which has a plurality of projections arranged along a circumferential direction of the cam plate to reciprocally drive a roller ring along an axis of the cam plate. A plunger integrally rotatable with the cam plate and integrally reciprocable with the roller ring, wherein each single rotation of said plunger causes the reciprocating movements of the plunger in a number of the projections to draw the fuel into a pressure chamber in which the drawn fuel is pressurized and supplied to each of the injection nozzles within a predetermined time period. The plunger is driven at a speed changing in accordance with a characteristic curve representing each angular position of the cam plate so as to adjust a supplying speed of the fuel to the injection nozzle, wherein the characteristic curve includes a first region representing the speed of the plunger increasing at a predetermined rate from zero, a second region representing the speed of the plunger increasingly accelerated up to a maximum value in a nonlinear curve manner and a third region representing the speed of the plunger decreasing from the maximum value. Each of the projections has a first portion to drive the plunger at the speed based on the first region of the characteristic curve, a second portion formed continuously to the first portion to drive the plunger at the speed based on the second region of the characteristic curve and a third portion formed continuously to the second portion to drive the plunger at the speed base on the third region of the characteristic curve. According to the invention, the non-linear curve includes a quadratic curve represented by a predetermined bivariate quadratic equation.

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 the objects and advantages thereof, may best be understood by reference to the following description of the presently preferred embodiment together with the accompanying drawings in which:

Fig. 1 is a graph showing relationships between cam speed and cam rotation angle and between

cam acceleration and cam lift with respect to a cam face having nonuniform speed cam characteristics in a fuel injection pump;

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

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

Fig. 4 is a partial enlarged cross-sectional view of the fuel injection nozzle of Fig.3 in a state where a needle is in a valve closing position;

Fig. 5 is also a partial enlarged cross-sectional view of the fuel injection nozzle of Fig.3 in a state where the needle is in a valve opening position;

Fig. 6 is a graph showing a relationship between lift of the needle and flow rate of compressed air;

Fig. 7 is a timing chart comparing the wave profile of the fuel rate of injection when the engine is operated at a medium speed and under a high load to that of a comparative example;

Fig. 8 is also a timing chart comparing the wave profile of the fuel rate of injection when the engine is operated at a medium speed and under a low load to that of the comparative example;

Fig. 9 is also a timing chart comparing the wave profile of fuel rate of injections between comparative examples;

Fig. 10 is a graph showing static fuel delivery rate when the engine is operated at the medium speed and under the high load according to an example;

Fig. 11 is also a graph showing static fuel delivery rate when the engine is operated at the medium speed and under the medium load according to an example;

Fig. 12 is also a graph showing static fuel delivery rate;

Fig. 13 is a graph showing static fuel delivery rate when the engine is idling under the minimum load according to an example;

Fig. 14 is also a graph showing static fuel delivery rate when the engine is idling under the full load according to an example;

Figs. 15 (a) to (c) are graphs each showing a relationship between cam speed and cam lift vs. cam rotation angle of a cam plate according to one ex-

ample; wherein (a) shows a pumping period when the engine is operated under the minimum load, (b) under the medium load and (c) under the full load, respectively;

Figs. 16 (a) to (c) are graphs each showing a relationship between cam speed and cam lift vs. cam rotation angle of a cam plate according to another example; wherein (a) shows a pumping period when the engine is operated under the minimum load, (b) under the medium load and (c) under the full load, respectively;

Fig. 17 (a) to (c) are graphs each showing a relationship between cam speed and cam lift vs. cam rotation angle of a cam plate according to a comparative prior art example; wherein (a) shows a pumping period when the engine is operated under the minimum load, (b) under the medium load and (c) under the full load, respectively;

Fig. 18 (a) to (c) are graphs each showing a relationship between cam speed and cam lift vs. cam rotation angle of a cam plate according to another comparative prior art example; wherein (a) shows a pumping period when the engine is operated under the minimum load, (b) under the medium load and (c) under the full load, respectively; and

Fig. 19 is a graph showing a relationship between cam speed and cam lift vs. cam rotation angle of a cam plate having nonuniform cam characteristics according to the prior art example.

DESCRIPTION OF SPECIAL EMBODIMENT

One embodiment of the fuel injection pump according to the present invention embodied in an automotive diesel engine will be described in detail below referring to the attached drawings.

Fig. 2 shows a fuel injection unit according to the present embodiment. This unit contains a distributor type fuel injection pump 1 and fuel injection nozzles 21 attached to respective cylinders of a diesel engine 3. The injection pump 1 drives a plunger 12 to reciprocate with rotation and to pump and distribute fuel to each injection nozzle 21. The injection pump 1 has a housing 2 and a drive shaft 4 supported rotatably in the housing 2. The shaft 4 is connected to the crank shaft (not shown) of the engine 3 via a pulley (not shown) and a belt (not shown). A vane type feed pump 5 is shown in Fig. 2, in which the major portion is depicted in 90° developed view. In the housing 2, a rotor 5a of this feed pump 5 is fitted on the shaft 4 to be rotatable therewith. The pump 5 draws the fuel from a fuel tank 6 via a sedimenter (water separator) 7 and a filter 8 and discharges it into a fuel chamber 9.

In the housing 2, a cylinder 11 is located in align-

ment with the shaft 4, and the plunger 12 is slidably fitted in the cylinder 11. The space defined between the distal end of the plunger 12 and the cylinder 11 constitutes a pressure chamber 13 for pressurizing the fuel to a high pressure level. The shaft 4 and the plunger 12 are connected to each other with a coupling 14. This connection between the plunger 12 and the shaft 4 enables their integral rotation and relative movement in the axial direction (horizontal direction in Fig. 2).

In the housing 2, a roller ring 15 is rotatably fitted on the shaft 4. A plurality of rollers 16 are supported at equiangular intervals on the surface of the roller ring 15 opposed to the plunger 12 along a circular orbit centering around the shaft 4. Meanwhile, a cam plate 17 is fixed to the proximal end of the plunger 12 to oppose the roller ring 15 such that the plunger 12 and the cam plate 17 are integrally rotatable. The surface of the cam plate 17 opposing the roller ring 15 has a cam face 17b. The cam face 17b has a number of cam peaks 17a corresponding to the number of cylinders of the engine 3. These cam peaks 17a have nonuniform speed cam characteristics such that the rate of fuel injected from one injection nozzle 21 varies in one injection. A spring 18, which is located between the housing 2 and the cam plate 17, urges the plunger 12 and cam plate 17 to bring the cam face 17b of the cam plate 17 into contact with the rollers 16. The torque of the shaft 4 is transmitted by the coupling 14 to the cam plate 17, and thus the cam plate 17 and the plunger 12 are reciprocated in the horizontal direction in Fig. 2 when they are rotated. This reciprocating motion of the cam plate 17 and plunger 12 causes the fuel to be drawn into the pressure chamber 13 where it is compressed by the plunger 12.

In order to introduce the fuel in the fuel chamber 9 into the pressure chamber 13, a suction passage 19 is defined through the housing 2 and the cylinder 11. This suction passage 19 provides communication between the pressure chamber 13 and the fuel chamber 9. A plurality of suction grooves 20 are defined on the circumference of the plunger 12 at its distal end. These suction grooves 20 communicate with the pressure chamber 13. The number of suction grooves 20 corresponds to the number of cylinders in the engine 3. In the suction stroke where the plunger 12 moves to the left in Fig. 2 to reduce the pressure in the pressure chamber 13, one of the suction grooves 20 is brought into communication with the suction passage 19, and thus the fuel in the fuel chamber 9 is drawn through the suction passage 19 and the suction groove 20 and into the pressure chamber 13.

The plunger 12 contains a fuel passage 22 and a plurality of distribution ports 23, which allow distribution and pumping of the fuel in the pressure chamber 13 to the respective cylinders of the engine 3. A plurality of distribution passages 24 are defined in the cylinder 11 and housing 2 to communicate with the distribution ports 23, respectively. Each distribution passage 24 contains a delivery valve 25. Each valve 25 is connected to its corresponding injection nozzle 21 via a fuel line 26.

In the compression stroke, subsequent to the suction stroke, where the plunger 12 moves to the right in Fig. 2, the suction groove 20 is caused to deviate from the suction passage 19 as the plunger 12 is rotated. When the plunger 12 rotates further to allow the cam peaks 17a on the cam face 17b to ride on the rollers 16, the plunger 12 is moved to the right in Fig. 2 to compress the fuel in the pressure chamber 13. The pressurized fuel is pumped to the corresponding injection nozzle 21 through the distribution port 23, distribution passage 24, delivery valve 25 and fuel line 26, when the port 23 is brought into communication with the passage 24, and is then injected into the cylinder.

The plunger 12 contains a spill port 27 for terminating fuel injection in the course of the compression stroke, and a spill ring 28 is slidably fitted on the plunger 12. This port 27 communicates at its middle with the compression chamber 13 via the fuel passage 22, and each extremity of the port 27 is open to the fuel chamber 9. The spill ring 28 is located with respect to the extremities or openings of this port 27. The openings of the spill port 27 closed so far by the spill ring 28 are opened when the plunger 12 is moved in the compression stroke to communicate with the fuel chamber 9. In this process, the fuel, having been compressed by the plunger 12 in the pressure chamber 13, spills through the spill port 27 into the fuel chamber 9, and thus the pressure of the fuel in the pressure chamber 13 drops suddenly. This pressure drop terminates pumping of the fuel from the injection pump 1 to the injection nozzle 21 to stop injection of the fuel from the injection nozzle 21.

The amount of fuel injected through the injection nozzle 21 is determined by the effective stroke of the plunger 12, which is the distance that the plunger 12 moves after compression of the fuel in the pressure chamber 13 is started until it is terminated. The fuel injection quantity is adjusted by changing this effective stroke. It should be noted here that the actual stroke of the plunger 12 is fixed, so that the injection quantity is adjusted by sliding the spill ring 28 on the plunger 12 in the axial direction. A governor mechanism 29, which is housed in the housing 2, adjusts the position of the spill ring 28 depending on the operational state of the engine 1 (depending on the parameters such as rotation speed, load, etc.). This mechanism 29 is of a type utilizing centrifugal force.

An overflow valve 31 containing an orifice is attached to the top of the housing 2. The valve 31 and the fuel tank 6 are connected to each other via an overflow line 32. The fuel discharged from the pump 5 to the fuel chamber 9 in excess is returned through the valve 31 and line 32 to the fuel tank 6.

A timer 33, which is disposed on the bottom of the housing 2, is operated by the fuel pressure in the fuel chamber 9. This timer 33 adjusts the rotational position of the roller ring 15 in the direction that the shaft 4 is rotated. Thus, the timing relationship between the cam peaks 17a of the cam face 17b and the rollers 16, i.e.,

the timing relationship between the cam plate 17 and the plunger 12, is adjusted, and thus the timing of injecting the fuel can be changed.

In addition to the basic structure of the injection pump 1, the cam peaks 17a are designed to have non-uniform cam characteristics (cam speed V relative to the cam rotation angle θ) in the following manner. The cam characteristics are factors which determine the rate that the injection pump 1 pumps out the fuel (fuel delivery rate).

As shown in Fig. 1, when the cam rotation angle θ is changed over the range of θ_0 to θ_1 with respect to one cam peak 17a, the cam speed V is increased from 0 at a constant rate. The portion of the cam peak 17a corresponding to this range is referred to as a first cam portion a. When the cam rotation angle θ of the cam peak 17a is further changed over the range of θ_1 to θ_2 , the cam speed V is increased at a quadratic rate (parabolically in this embodiment) as a nonlinear curve such that the cam acceleration α may increase. The portion of the cam peak 17a corresponding to this range is referred to as a second cam portion b for obtaining the maximum value V_{max} . When the cam rotation angle θ of the cam peak 17a is further changed over the range of θ_2 to θ_3 , the cam speed V is decreased gradually. The portion of the cam peak 17a corresponding to this range is referred to as a third cam portion c. The third cam portion c is continuous from the second cam portion b. The third cam portion c has an inflection point P in the middle as clearly shown by the change in the cam acceleration α .

Next, the injection nozzles 21 for atomizing and injecting the fuel pumped from the injection pump 1 under high pressure into the respective combustion chambers of the engine will be described.

As shown in Figs. 3 to 5, each injection nozzle 21 has an elongated nozzle holder 35, with a spacer 36 and a nozzle body 37 being situated under the holder 35 to abut with one another. The spacer 36 and the body 37 are attached to the holder 35 by a retaining nut 38 screwed onto the lower portion of the holder 35.

A fuel passage 39 is defined through the holder 35, spacer 36 and body 37. The fuel passage 39 opens at the upper extremity to the upper end of the holder 35. The lower extremity of the fuel passage 39 opens to a fuel reservoir 40 defined at the lower portion of the body 37. The fuel reservoir 40 communicates with the lower surface of the body 37 via a plurality of injection holes 41 (see Figs. 4 and 5). The fuel reservoir 40 has a tapered face 41a converging downward between the fuel passage 39 and the injection holes 41. When the pressurized fuel pumped from the injection pump 1 is supplied to the injection nozzle 21, the fuel is injected out through the fuel passage 39 and fuel reservoir 40 successively.

A needle 42 for opening and closing the injection holes 41 is incorporated into the body 37 and spacer 36. The needle 42 consists of a main body 43 and a pair of

shafts 44,45 extending upward and downward from the main body 43, respectively. The main body 43 has a rod-like shape and is axially movable in the body 37. The lower end face of the main body 43 faces the fuel reservoir 40, and the pressure of the fuel in the fuel reservoir 40 acts upon the lower end face to urge the needle 42 upward.

One shaft 44 is slidably incorporated into the spacer 36. A seat face 45a is formed on the lower end portion of the other shaft 45. The seat face 45a is selectively brought into or out of contact with the tapered face 41a.

A pressure pin 47 is attached to the shaft 44, and a sleeve 48 is incorporated into the holder 35 substantially at its middle. A first spring 49 is compressed between the pressure pin 47 and the sleeve 48. The needle 42 is normally urged by the spring 49 against the tapered face 41a (in the valve closing direction, i.e., downward in Fig. 3). Thus, the seat face 45a is brought into contact with the tapered face 41a, as shown in Fig. 4, to interrupt communication between the fuel passage 39 and the injection holes 41 and to stop injection of the fuel from the injection nozzle 21. In this state, the main body 43 is spaced downward from the spacer 36 by the length Lb.

Meanwhile, when the fuel is to be injected, the needle 42 moves upward to allow the seat face 45a to be spaced from the tapered face 41a, and thus the main body 43 abuts against the spacer 36.

A push rod 50, which is within the sleeve 48 movable in the axial direction, is aligned with the needle 42. A second spring 51 is compressed within the sleeve 48, and the rod 50 is normally urged toward the pressure pin 47 by this spring 51.

When the needle 42 is in its valve closing state, the pin 47 is spaced downward from the rod 50 by the length La ($La < Lb$). Accordingly, the pin 47 and the needle 42 are not subject to the urging force of the spring 51. When the needle 42 is moved upward by the length La, the pin 47 abuts abutted against the rod 50. If the needle 42 is moved upward beyond the length La, the needle 42 is subject to the urging force of the spring 51. In this embodiment, the pin 47, sleeve 48, spring 49, rod 50 and spring 51 constitute a valve opening pressure adjust mechanism A.

In this embodiment, the fuel pressure required for starting the upward movement of the needle 42 from the valve closing state is referred to as a first valve opening pressure level P1; while the fuel pressure required for resuming upward movement of the needle 42 when it is abutted against the rod 50 via the pin 47 is referred to as a second valve opening pressure level P2. The first valve opening pressure level P1 is preferably, about 200 kg/cm². The second valve opening pressure level P2 is desirably set depending on the increased level of the fuel pressure corresponding to the portion of the cam peak 17a used for obtaining the desired injection timing.

In the injection nozzle 21, the stroke (lift L) of the needle 42 is determined depending on the pressure P

of the fuel supplied from the injection pump 1. When the fuel pressure acting upon the needle 42 is lower than the first valve opening pressure level P1 in the fuel reservoir 40, the seat face 45a is pressed against the tapered face 41a. When the fuel pressure becomes higher than the first valve opening pressure level P1, the needle 42 starts moving upward to allow the seat face 45a to be spaced from the tapered face 41a. This upward movement of the needle 42 continues until the pin 47 abuts against the rod 50. While the fuel pressure is lower than the second valve opening pressure level P2 after the pin 47 is abutted against the rod 50, the needle 42 ceases to move upward. Further, when the fuel pressure is increased to exceed the second valve opening pressure level P2, the needle 42 moves upward again. This upward movement of the needle 42 continues until the main body 43 of the needle 42 abuts against the spacer 36.

Fig. 6 shows flow rate characteristics of the injection nozzle 21. The flow rate characteristics are substantially the same as the general characteristics of a hole injection nozzle. In Fig. 6, the ordinate represents flow rate Q, and the abscissa represents lift of the needle 42. Fig. 6 shows flow rate Q of air, which is supplied in place of fuel to the fuel passage 39 of the injection nozzle 21 and is injected through the injection holes 41. As clearly shown in Fig. 6, in the range where the lift L is between 0 to La, the flow rate Q increases substantially proportionally to the lift L. When the lift L exceeds the prelift La, the flow rate Q ceases to increase proportionally, and stabilizes.

Next, operation of the fuel injection unit will be described. It should be noted here that, in an actual measurement, change in the pressure of the fuel to be fed from the injection pump 1 is slightly retarded by the presence of the fuel line 26 between the injection pump 1 and the injection nozzle 21. In other words, the change in the fuel pressure reaches the fuel reservoir 40 with a predetermined phase delay. Accordingly, there is a time lag between the change in the fuel pressure Pp at the outlet (immediately downstream of the delivery valve 25) of the injection pump 1 and the change in the fuel pressure Pn at the fuel reservoir 40.

When the shaft 4 of the injection pump 1 is rotated under the driving force of the engine, the torque is transmitted via the coupling 14 to the cam plate 17. Thus, the cam plate 17 and the plunger 12 are reciprocated with rotation in the horizontal direction in Fig. 2.

During the suction stroke, where the plunger 12 moves leftward in Fig. 2, when one of the suction grooves 20 is brought into communication with the suction port 19, the fuel in the fuel chamber 9 is drawn through the suction groove 20 and suction port 19 into the pressure chamber 13. Then, the communication between the port 19 and the groove 20 is interrupted, and the corresponding distribution port 23 is brought into communication with one of the distribution passages 24.

When the plunger 12 is rotated further, the cam

peaks 17a ride on the rollers 16 to move the plunger 12 rightward in Fig. 2 to start the step of pumping the fuel. If the rollers 16 engage with the first cam portions a of the cam peaks 17, respectively, as shown in Fig. 1, the speed of the plunger 12 is increased at a fixed rate with time. With the increase in the speed, the volume of the pressure chamber 13 is reduced gradually to compress the fuel in the pressure chamber 13.

Subsequently, the compressed fuel is pumped through the distribution passage 24, valve 25 and fuel line 26 to the injection nozzle 21. Accordingly, both of the fuel pressures P_p and P_n increase with time at the first cam portion a.

In this stage, since the fuel pressure P_n in the fuel reservoir 40 is lower than the first valve opening pressure level P_1 , the seat face 45a remains pressed against the tapered face 41a, and thus the needle 42 is in its valve closing state. Accordingly, no fuel is injected from the injection nozzle 21. In this case, both the lift L of the needle 42 and the fuel rate of injection at the injection nozzle 21 are zero.

When the rollers 16 engage with the cam peaks 17a at the boundary between the first cam portion a and the second cam portion b in Fig. 1, and when time Δt corresponding to the phase delay passes after such timing, the fuel pressure P_n in the fuel reservoir 40 reaches the first valve opening pressure level P_1 . The needle 42 is then pushed upward against the urging force of the spring 49 to allow the seat face 45a to be spaced from the tapered face 41a and to start injection of the fuel. Concomitantly, the lift L of the needle 42 and the fuel rate of injection at the injection nozzle 21 start increasing.

When the rollers 16 engage with the second cam portions b of the cam peaks 17a after fuel injection is started, the cam speed V is increased such that the cam acceleration α increases as the cam rotation angle θ increases. Accordingly, the speed of moving the plunger 12 is accelerated along with the speed of pumping the fuel. In this case, the fuel pressure in the pressure chamber 13 is not increased very much but the internal fuel pressure P_n of the fuel reservoir 40 is increased by the inertia of the fuel pumped thereto at the accelerating speed.

The internal fuel pressure P_n in the fuel reservoir 40 further increases, after it exceeds the first valve opening pressure level P_1 , to rapidly reach the second valve opening pressure level P_2 . Consequently, the fuel is injected under high pressure from the injection nozzle 21, and the injected fuel is atomized into very fine particles.

The injection nozzle 21 is in the valve opening position between two valve opening pressure levels P_1 and P_2 . Accordingly, the needle 42 ascends after the internal fuel pressure P_n of the fuel reservoir 40 exceeds the first valve opening pressure level P_1 and until the pin 47 abuts against the rod 50 (until the pin 47 reaches the predetermined valve opening position). However,

when the needle 40 ascends by the length L_a to bring the pin 47 into contact with the rod 50, the urging force of the spring 51 is then applied to the needle 42. Accordingly, when the fuel pressure P_n is lower than the second valve opening pressure level P_2 , the lift L of the needle 42 is maintained constantly at the prelift value L_a .

When the internal fuel pressure P_n of the fuel reservoir 40 is further increased to reach the second valve opening pressure level P_2 , the needle 42 ascends again against the urging forces of the two springs 49, 51. When the lift L increases to exceed the prelift value L_a , the fuel flow rate Q increases with the increase in the lift L and then assumes a constant level. Concomitantly, the fuel rate of injection of the injection nozzle 21 is increased. The ascending of the needle 42 is stopped when the main body 43 is abuts against the spacer 36.

Subsequently, when a spill port 28 is opened during the compression stroke of the plunger 12, the fuel pressure in the pressure chamber 13 is reduced to stop injection of the fuel from the injection nozzle. In other words, even if the plunger 12 is moved in the compression stroke, the fuel pressure in the pressure chamber 13 is not increased so long as the spill port 28 is open, so that no fuel is injected from the injection nozzle 21.

In the above description, the time when pumping of the fuel by the injection pump 1 is started is designed to correspond to the boundary between the cam portion a and the cam portion b of each cam peak 17a based on the supposition that the engine 1 is rotated at a medium speed or high speed.

Next, the static fuel delivery rate was measured in the above-described fuel injection unit and in a comparative example, which has different nonuniform speed cam characteristics. The results are shown in Figs. 10 to 14.

The static fuel delivery rate was measured when the engine 1 was operated at low speed, at medium speed and at high speed with a different fuel pumping period (fuel injection quantity), i.e., with different engine loads, respectively.

Example 1 (E1) shown in Figs. 10 to 14 is of a cam plate 17 having cam peaks 17a with cam characteristics as shown in Figs. 1 and 15 (a) to (c). Example 2 (E2) is of a cam plate 17 having cam peaks 17a with cam characteristics as shown in Figs. 16 (a) to (c). The cam peaks in Example 2 (E2) have characteristics corresponding to the first cam portion a, second cam portion b and third cam portion c in Figs. 16 (a) to (c). What is different from Example 1 (E1) is that the second cam portion b is of a profile taken along a hyperbola, and the cam acceleration increases as the cam speed approaches the maximum value V_{max} at the second cam portion b.

Figs. 17 (a) to (c) show cam characteristics of the cam peaks according to a prior art Comparative Example 1 (S1). The cam peaks of the prior art have, as shown in Figs. 17 (a) to (c), a zone corresponding to the first cam portion d where the cam speed is linearly increased at a constant rate and another zone, continuous to the

former cam portion d, corresponding to the latter cam portion e, where the cam speed exceeding the maximum value V_{max} is reduced at a constant rate.

Figs. 18 (a) to (c) show nonuniform speed cam characteristics of the cam peaks disclosed in Japanese Unexamined Patent Publication No. Hei 2-298635 as Comparative Example 2 (S2). The cam peaks in the prior art has, as shown in Figs. 18 (a) to (c), a zone corresponding to the former cam portion f, where the cam speed is linearly increased at a constant rate; a zone corresponding to the intermediate cam portion g, formed continuous to the cam portion f, where the cam speed is linearly increased at a rate lower than in the cam portion f; and another zone, formed continuous to the cam portion f, corresponding to the latter cam portion h where the cam speed exceeded the maximum value V_{max} is reduced at a constant rate.

In Figs. 15 to 18, pumping periods equal (fuel injection quantity) are affixed with the same codes, respectively. A first pumping period H1 ($5 \text{ mm}^3/\text{st}$) is where the engine is idling under the minimum load; the second pumping period H2 ($15 \text{ mm}^3/\text{st}$), the third pumping period H3 ($35 \text{ mm}^3/\text{st}$) and the fourth pumping period H4 ($55 \text{ mm}^3/\text{st}$) are where the engine is operated at the medium speed and under the low load, at the medium speed and under the medium load, and at the medium speed and under the high load, respectively. The fifth pumping period H5 ($60 \text{ mm}^3/\text{st}$) is where the engine is operated at the high speed and under the full load. Incidentally, mm^3/st is a unit of fuel quantity per stroke of the plunger 12.

Fig. 10 is a graph showing static fuel delivery rate under the high load, i.e., in the fourth pumping period H4, in each Example and each Comparative Example. Fig. 11 is a graph showing static fuel delivery rate under medium load, i.e., in the third pumping period H3, in each Example and each Comparative Example. Fig. 12 is a graph showing static fuel delivery rate under low load, i.e., in the second pumping period H2, in each Example and each Comparative Example. Fig. 13 is a graph showing static fuel delivery rate under full load, i.e., in the fifth pumping period H5, in each Example and each Comparative Example. Fig. 14 is a graph showing static fuel delivery rate under minimum load, i.e., in the first pumping period H1, in each Example and each Comparative Example.

As clearly shown in Figs. 10 to 14, it can be understood that the static fuel delivery rates of the injection pump 1 in Example 1 (E1) and Example 2 (E2) are lower than those in Comparative Examples 1 and 2 (S1 and S2).

Figs. 7 to 9 each show fuel rate of injection (unit: mm^3/deg) on the abscissa vs. time on the ordinate. Fig. 7 shows a wave profile of fuel rate of injection with respect to Example 1 (E1: indicated by the dotted line) when the engine is rotated at medium speed in the fourth pumping period H4 and that of Comparative Example 1 (S1: indicated by the solid line), which are superimposed

to be registered at the injection starting time. Fig. 8 shows a wave profile of fuel rate of injection with respect to Example 1 (E1: indicated by the dotted line) when the engine is rotated at medium speed in the pumping period where fuel injection quantity is $10 \text{ mm}^3/\text{st}$ and that of Comparative Example 1 (S1: indicated by the solid line), which are superimposed to be registered at the injection starting time.

From Figs. 7 and 8, it was identified that the injection period is longer in Example 1 (E1) than in Comparative Example 1 (S1) when the fuel injection quantity is $10 \text{ mm}^3/\text{st}$, although the peak value of the fuel rate of injection is lower. It was also identified that, in the case where the engine is operated at high load in the fourth pumping period H4 at the fuel injection quantity of $55 \text{ mm}^3/\text{st}$, the peak value of the fuel rate of injection is the same in Example (E1) and in Comparative Example 1 (S1), and that the initial fuel rate of injection is lower in Example 1 (E1).

Fig. 9 shows a wave profile of fuel rate of injection with respect to Comparative Example 1 (S1: indicated by the solid line) when the engine is rotated at medium speed in the pumping period where fuel injection quantity is $10 \text{ mm}^3/\text{st}$ and that of Comparative Example 2 (S2: indicated by the dotted line), which are superimposed to be registered at the injection starting time. Comparative Example 2 demonstrated a higher initial fuel rate of injection than in Comparative Example 1, which is of no satisfactory rate of injection. While Comparative Example 2 corresponds to the technique disclosed in Japanese Unexamined Patent Publication No. Hei 2-298635, when this technique is to be embodied in a six-cylinder engine, the angle to be allotted for each of the cam peaks on the cam face of the cam plate corresponding to the number of cylinders will be 60° . In the nonuniform speed cam characteristics shown in Fig. 19, the degree that the cam speed in the zone ZT is increased, where the cam speed is sharply increased compared with the zone XT is small. Accordingly, in the case of a direct injection type diesel engine, it is necessary to effect a high cam speed in the zone XT, where increase in the cam speed is mild, because some degree of static fuel delivery rate is required, depending on the increased stress, when the engine is operated at high speed. However, due to the limitation on the pressure to be applied to the cam face, the portion where the cam speed is increased cannot be effected in the zone ZT, where the cam speed shifts from the lower side to the higher side. Meanwhile, Examples 1 and 2 are free from this problem even when they are used in a six-cylinder engine.

To summarize the above description, it was identified that the static fuel delivery rate is low in Example 1, but the actual fuel rate of injection during the high load operation of the engine is comparable to that of the prior art Comparative Example 1, and that the fuel rate of injection is low like the static fuel delivery rate during low load operation of the engine.

Ideally, the fuel rate of injection should be low during

the low load operation of the engine and high during high load operation of the engine. Further, the wave showing the characteristics of the fuel rate of injection should satisfy a so-called sharp-cut profile where the initial fuel rate of injection is low and its peak value is great under the high load operation.

Accordingly, as described above, the sharp-cut profile can be satisfied with respect to the cam characteristics in Example 1 by allowing the second cam portion b shown in Fig. 1 to have a shape where the acceleration is increased such that the cam speed V increases along a parabolic curve. Further, it was identified from Fig. 7 that the tendency of increasing pressure is based on the inertia of the fuel, even when the cam speed V is reduced after the injection pressure of the injection pump 1 is increased toward the end point of injection. Therefore, the pressure at the portion close to the injection nozzle 21 connected to the injection pump 1 can be increased without increasing the fuel pressure in the pressure chamber 13 by much. Thus, a high fuel injection pressure can be obtained without increasing the pressure to be applied to the cam face 17a. As a result, the maximum value of the cam speed can be reduced as compared to the prior art to reduce the pressure to be applied to the cam face 17a as compared the prior art. Accordingly, even if the pressure to be applied to the cam face 17a happens to be increased, the cam face 17a is prevented from undergoing undesired pitching and seizure, leading to improvement in reliability.

In Example 1, the cam face 17a is employed in combination with fuel injection nozzles 21 which are operated to be in the valve opening position between two valve opening pressure levels P1, P2. Accordingly, the initial rate of injection pattern can be changed so that an optimum rate of injection pattern can be obtained. Thus, the optimum rate of injection pattern can reduce noise and improve the quality of the exhaust gas (low NOx and low smoke).

Particularly, it should be understood that the present invention may be embodied in the following manners.

While injection nozzles 21 of the type each containing two springs are employed in the above embodiment, the nozzles may be single-hole nozzles, multihole nozzles, pintle nozzles, throttle nozzles, etc.

The term "nonlinear curve" referred to in the description above means a segment excluding a straight line and includes, a quadratic curve. The term "quadratic curve" referred to here means a conical curve to be obtained by the intersection of a plane and a right circular cone (section) and includes an ellipse, a circle, a hyperbola and a parabola formed as a function of the angle of intersection. Namely, the quadratic curve can be expressed by the following quadratic equation with two unknowns:

$$ax^2 + by^2 + 2cxy + dx + ey + f = 0,$$

provided that $a = b = c \neq 0$.

Claims

1. A fuel injection pump used in a Diesel engine (3), said pump being arranged to distribute fuel to a plurality of fuel injection nozzles (21) provided in said engine (3), each of said injection nozzles (21) being arranged to open by the fuel pressurized up to a predetermined value to inject the fuel therefrom, said injection pump including a cam disk (17) which has a plurality of projections (17a) arranged along a circumferential direction of the cam disk (17) to reciprocally drive a roller ring (15) along an axis of the cam disk (17), a plunger (12) integrally rotatable with the cam disk (17) and integrally reciprocable with the roller ring (15), wherein each single rotation of said plunger (12) causes the reciprocating movements of the plunger (12) in a number of the projections (17a) to draw the fuel into a pressure chamber (13) in which the drawn fuel is pressurized and supplied to each of said injection nozzles (21) within a predetermined time period, and wherein said plunger (12) is driven at a speed changing in accordance with a characteristic curve representing each angular position of the cam disk (17) so as to adjust a supplying speed of the fuel to the injection nozzle (21), wherein said characteristic curve includes a first region representing the speed of the plunger increasing at a predetermined rate from zero, a second region representing the speed of the plunger increasingly accelerated up to a maximum value in a nonlinear curve manner and a third region representing the speed of the plunger decreasing from the maximum value, wherein each of said projections (17a) has a first portion (a) to drive the plunger (12) at the speed based on the first region of the characteristic curve, a second portion (b) formed continuously to the first portion (a) to drive the plunger (12) at the speed based on the second region of the characteristic curve and a third portion (c) formed continuously to the second portion (b) to drive the plunger (12) at the speed base on the third region of the characteristic curve characterized by that said nonlinear curve includes a quadratic curve represented by a predetermined bivariate quadratic equation.
2. The pump as set forth in Claim 1, characterized in that the engine (3) rotates at a high speed, an intermediate speed or a low speed, wherein the fuel supply begins when the plunger (12) is driven by a boundary between the first portion (a) and the second portion (b) during the rotation of the engine (3) at the high speed or the intermediate speed.
3. The pump as set forth in Claims 1 or 2, character-

ized in that said third portion (c) includes a turning point (P) for changing the reciprocating speed of the plunger (12).

4. The pump as set forth in any one other preceding claims, characterized in that the pressure of the fuel supplied to each of said injection nozzles (21) increases up to a predetermined magnitude through an intermediate magnitude, wherein the intermediate magnitude of the pressure causes the injection nozzle (21) to be open and the predetermined magnitude of the pressure causes the wider opening of the injection nozzle (21).
5. The pump as set forth in Claim 4, characterized in that said intermediate magnitude includes a first intermediate magnitude and a second intermediate magnitude.

Patentansprüche

1. Kraftstoffeinspritzpumpe zum Verwenden in einem Dieselmotor (3), wobei die Pumpe zum Verteilen von Kraftstoff auf eine Vielzahl von in dem Motor (3) vorgesehenen Einspritzdüsen (21) angeordnet ist, wobei jede der Einspritzdüsen (21) angeordnet ist, um durch den auf einen vorgegebenen Wert mit Druck beaufschlagten Kraftstoff zu öffnen, um den Kraftstoff von dieser einzuspritzen, wobei die Einspritzpumpe eine Nockenscheibe (17), die eine Vielzahl von entlang einer Umfangsrichtung der Nockenscheibe (17) angeordneten Vorsprüngen (17a) hat, um einen Walzenring (15) entlang einer Achse der Nockenscheibe (17) hin- und hergehend anzutreiben, und einen Kolben (12) umfaßt, der einstückig mit der Nockenscheibe (17) drehbar ist und einstückig mit dem Walzenring (15) hin- und herbewegbar ist, wobei jede einzelne Drehung des Kolbens (12) die Hin- und Herbewegungen des Kolbens (12) in einer Anzahl der Vorsprünge (17a) veranlaßt, um den Kraftstoff in eine Druckkammer (13) zu saugen, in der der angesaugte Kraftstoff mit Druck beaufschlagt wird und jeder der Einspritzdüsen (21) innerhalb einer vorgegebenen Zeitperiode zugeführt wird, und wobei der Kolben (12) mit einer Geschwindigkeit angetrieben wird, die sich in Übereinstimmung mit einer kennzeichnenden Kurve ändert, die jede Winkelposition der Nockenscheibe (17) repräsentiert, um eine Zuführgeschwindigkeit des Kraftstoffs zu der Einspritzdüse (21) einzustellen, wobei die kennzeichnende Kurve einen ersten Bereich, der die mit einer vorgegebenen Rate von Null aus ansteigende Geschwindigkeit des Kolbens repräsentiert, einen zweiten Bereich, der die auf einen Maximalwert in einer nichtlinearen Verlaufsweise in zunehmendem Maße beschleunigte Geschwindigkeit des Kolbens repräsentiert, und einen

dritten Bereich umfaßt, der die von dem Maximalwert aus abnehmende Geschwindigkeit des Kolbens repräsentiert, wobei jeder der Vorsprünge (17a) einen ersten Abschnitt (a), um den Kolben (12) mit der Geschwindigkeit auf der Grundlage des ersten Bereichs der kennzeichnenden Kurve anzutreiben, einen zweiten Abschnitt (b), der kontinuierlich zu dem ersten Abschnitt (a) ausgebildet ist, um den Kolben (12) mit der Geschwindigkeit auf der Grundlage des zweiten Bereichs der kennzeichnenden Kurve anzutreiben, und einen dritten Abschnitt (c) hat, der kontinuierlich zu dem zweiten Abschnitt (b) ausgebildet ist, um den Kolben (12) mit der Geschwindigkeit auf der Grundlage des dritten Bereichs der kennzeichnenden Kurve anzutreiben, **dadurch gekennzeichnet, daß** die nichtlineare Kurve eine quadratische Kurve umfaßt, die durch eine vorgegebene quadratische Gleichung mit zwei Merkmalsvariablen repräsentiert wird.

2. Pumpe nach Anspruch 1, **dadurch gekennzeichnet, daß** der Motor (3) mit einer hohen Drehzahl, einer mittleren Drehzahl oder einer niedrigen Drehzahl dreht, wobei während dem Drehen des Motors (3) mit der hohen Drehzahl oder der mittleren Drehzahl die Kraftstoffzufuhr dann beginnt, wenn der Kolben (12) durch eine Grenze zwischen dem ersten Abschnitt (a) und dem zweiten Abschnitt (b) angetrieben wird.
3. Pumpe nach Anspruch 1 oder 2, **dadurch gekennzeichnet, daß** der dritte Abschnitt (c) einen Wendepunkt (P) zum Ändern der hin- und hergehenden Geschwindigkeit des Kolbens (12) umfaßt.
4. Pumpe nach einem der vorangegangenen Ansprüche, **dadurch gekennzeichnet, daß** der Druck des Kraftstoffs, der jeder der Einspritzdüsen (21) zugeführt wird, über einen Zwischenbetrag zu einem vorgegebenen Betrag ansteigt, wobei der Zwischenbetrag des Drucks den offenen Zustand der Einspritzdüse (21) veranlaßt, und der vorgegebene Betrag des Drucks das breitere Öffnen der Einspritzdüse (21) veranlaßt.
5. Pumpe nach Anspruch 4, **dadurch gekennzeichnet, daß** der Zwischenbetrag einen ersten Zwischenbetrag und einen zweiten Zwischenbetrag umfaßt.

Revendications

1. Une pompe d'injection de combustible utilisée dans un moteur Diesel (3). ladite pompe étant agencée pour distribuer du combustible à une pluralité d'orifices ou buses d'injection de combustible (21) prévus dans ledit moteur (3), chacun desdits orifices

d'injection (21) étant agencé pour être ouvert par le combustible pressurisé jusqu'à une valeur prédéterminée afin d'injecter le combustible à partir de l'orifice, ladite pompe d'injection comprenant un disque de came (17) qui présente une pluralité de parties en saillie (17a) agencées le long d'une direction circonférentielle du disque de came (17) pour entraîner réciproquement une bague à rouleaux (15) le long d'un axe du disque de came (17), un piston plongeur (12) susceptible de tourner monobloc avec le disque de came (17) et susceptible de se déplacer alternativement de façon monobloc avec la bague à rouleaux (15), dans laquelle chaque tour complet dudit plongeur (12) provoque les déplacements alternatifs du piston plongeur (12) selon le même nombre que celui des parties en saillie (17a) pour aspirer le combustible dans une chambre de pressurisation (13) dans laquelle le fluide aspiré est pressurisé et refoulé à chacun desdits orifices d'injection (21) pendant une période de temps prédéterminée, et dans laquelle ledit piston plongeur (12) est entraîné à une vitesse se modifiant en correspondance avec une courbe caractéristique représentant chaque position angulaire du disque de came (17), de façon à régler la vitesse de refoulement du combustible vers l'orifice d'injection (21), dans laquelle ladite courbe caractéristique comporte une première région représentant la vitesse du plongeur augmentant à un taux prédéterminé à partir de zéro, une deuxième région représentant la vitesse du plongeur augmentant de façon accélérée jusqu'à une valeur maximale selon une courbe non-linéaire, et une troisième région représentant la vitesse du plongeur diminuant depuis la valeur maximale, dans laquelle chacune desdites parties en saillie (17a) comporte une première partie (a) pour entraîner le piston plongeur (12) à la vitesse basée sur la première région de la courbe caractéristique, une deuxième partie (b) formée de manière continue vers la première partie (a) pour entraîner le piston plongeur (12) à la vitesse basée sur la deuxième région de la courbe caractéristique et une troisième partie (c) formée de manière continue vers la deuxième partie (b), pour entraîner le piston plongeur (12) à la vitesse basée sur la troisième région de la courbe caractéristique, caractérisée en ce que ladite courbe non-linéaire comporte une courbe quadratique représentée par une équation quadratique à deux variables.

2. La pompe selon la revendication 1, caractérisée en ce que lorsque le moteur (3) tourne à vitesse élevée, à vitesse intermédiaire ou à vitesse faible, l'alimentation en combustible commence lorsque le piston plongeur (12) est entraîné par une zone frontière entre la première partie (a) et la deuxième partie (b) pendant la rotation du moteur (3) à vitesse élevée ou à vitesse intermédiaire.

3. La pompe selon les revendications 1 ou 2, caractérisée en ce que ladite troisième partie (c) présente un point d'inflexion (P) où est modifiée la vitesse de déplacement alternatif du piston plongeur (12).

4. La pompe selon l'une quelconque des revendications précédentes, caractérisée en ce que la pression du combustible fournie à chacun desdits orifices d'injection (21) augmente jusqu'à une amplitude prédéterminée en passant par une amplitude intermédiaire, en ce que l'amplitude intermédiaire de la pression oblige l'orifice d'injection (21) à être ouvert et l'amplitude prédéterminée de la pression oblige l'orifice d'injection (21) à être ouvert sur une section plus grande.

5. La pompe selon la revendication 4, caractérisée en ce que ladite amplitude intermédiaire comporte une première amplitude intermédiaire et une deuxième amplitude intermédiaire.

Fig.1

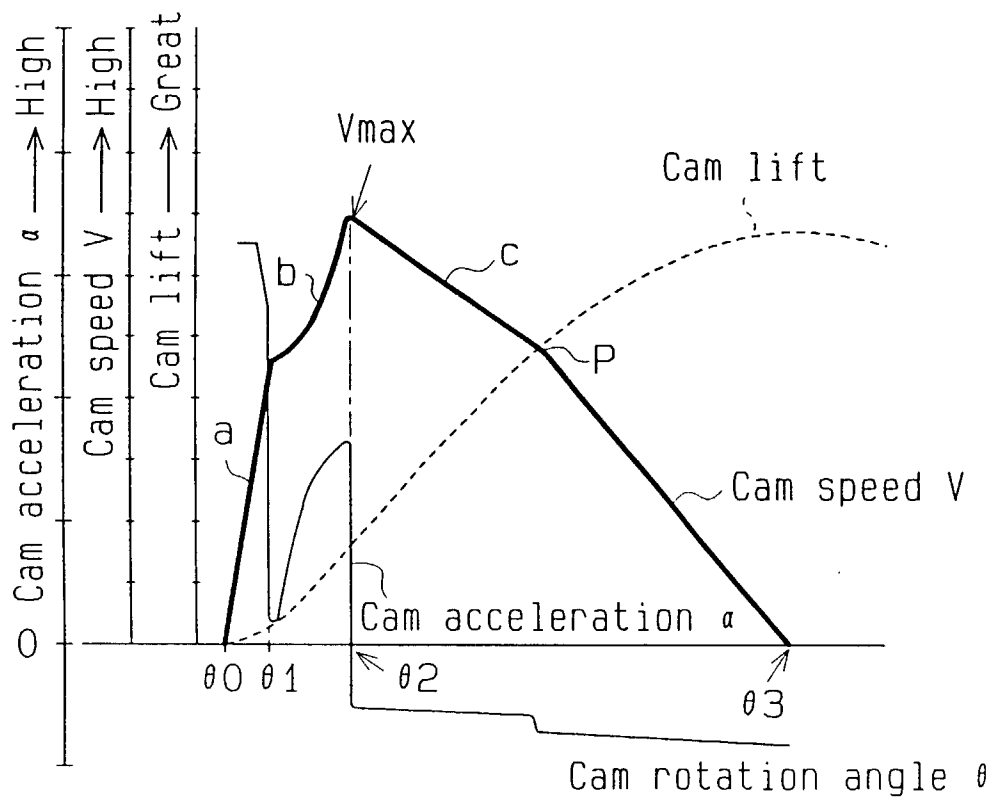


Fig. 2

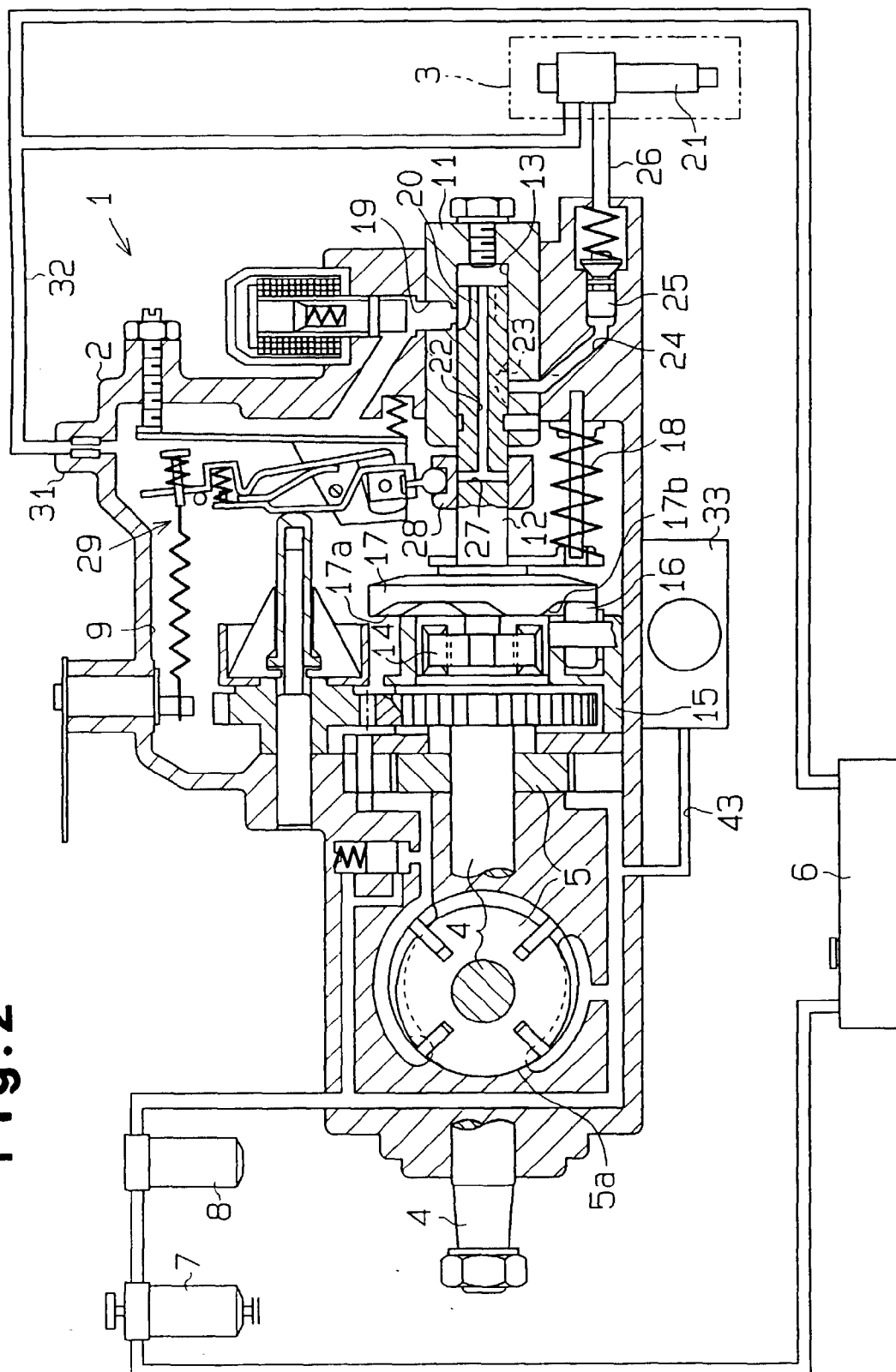


Fig. 3

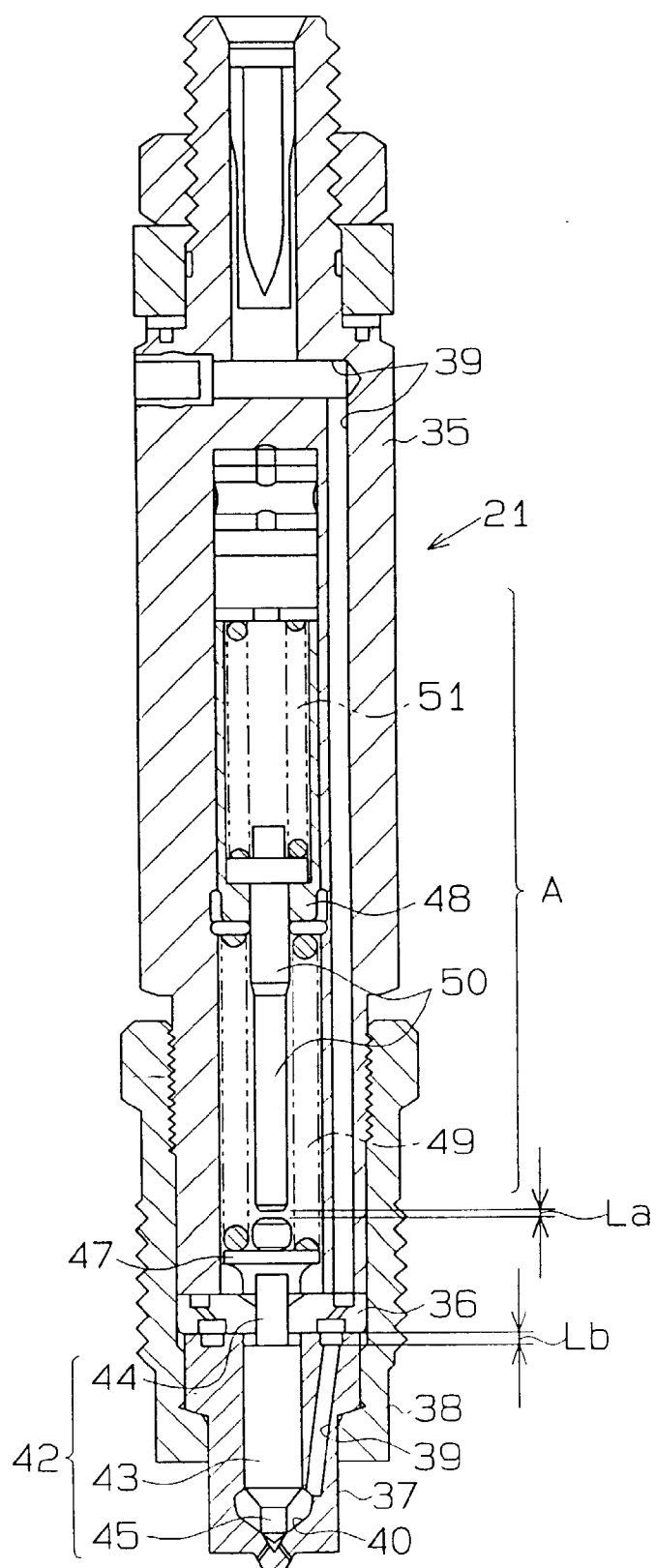


Fig.4

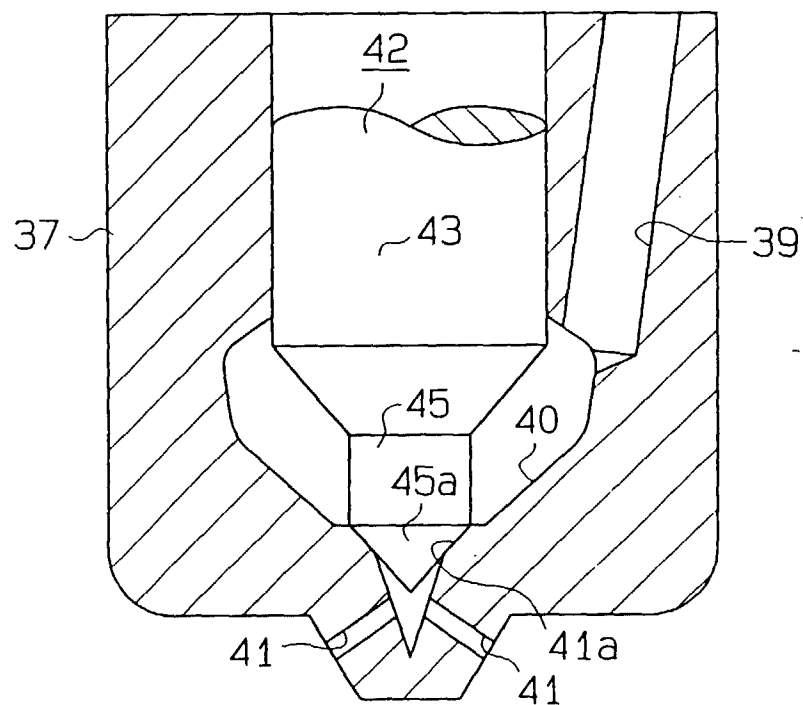


Fig.5

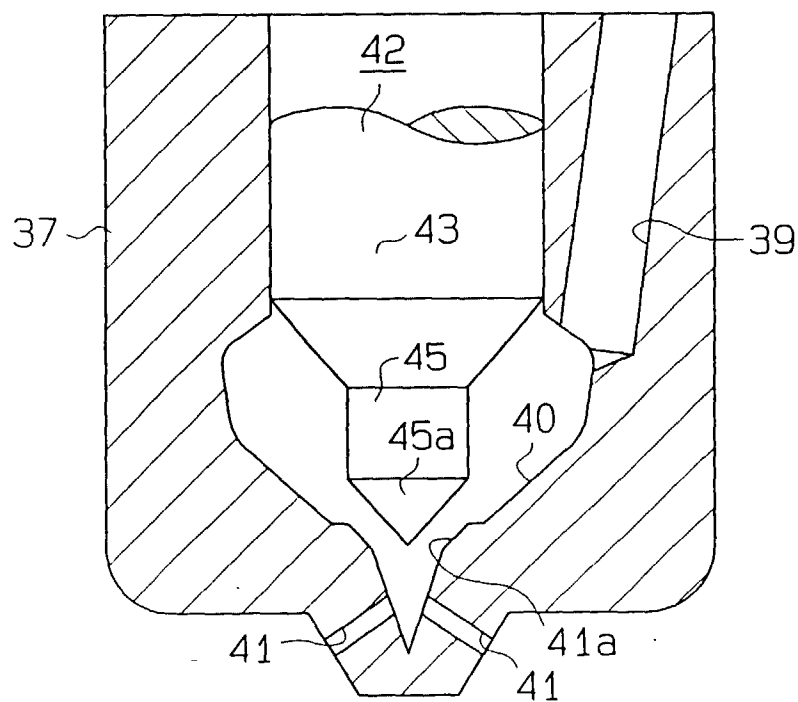


Fig.6

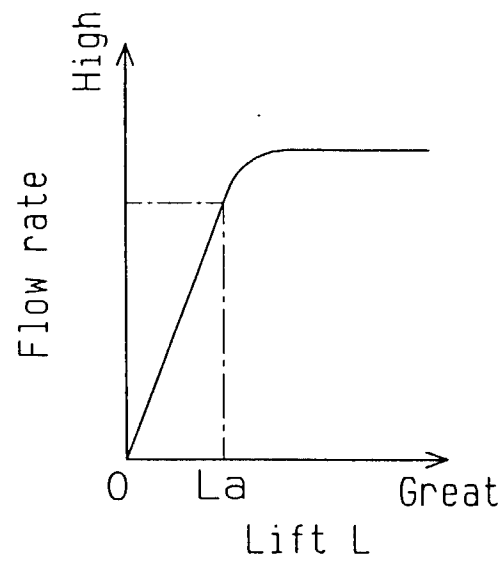


Fig.7

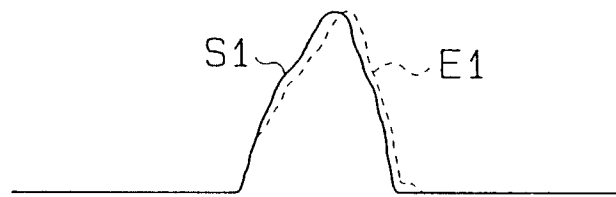


Fig.8



Fig.9



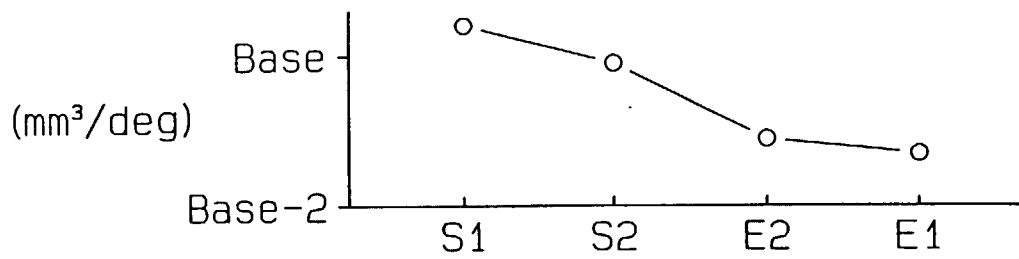
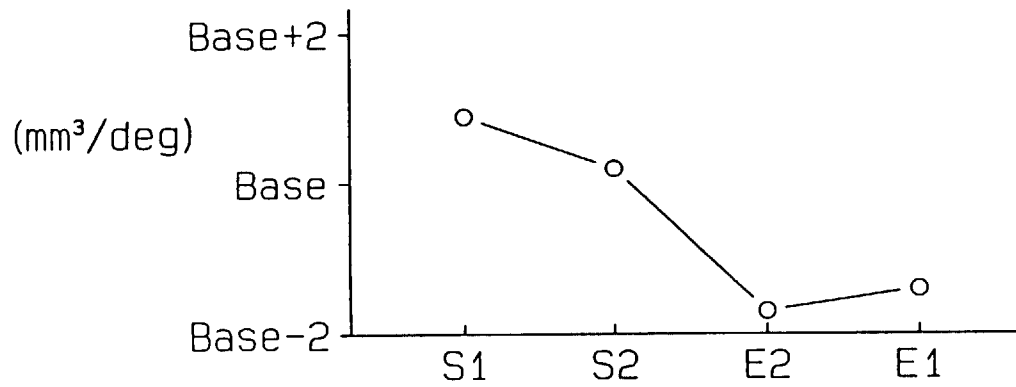
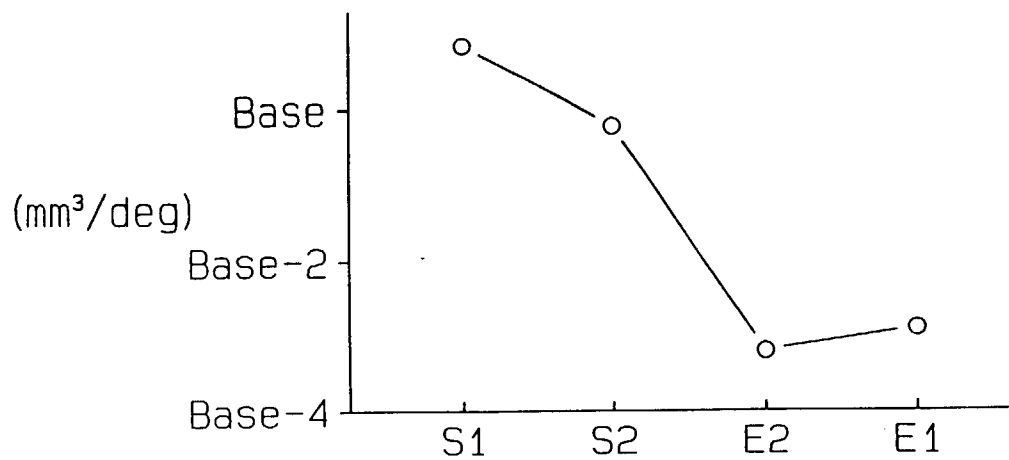
Fig.10**Fig.11****Fig.12**

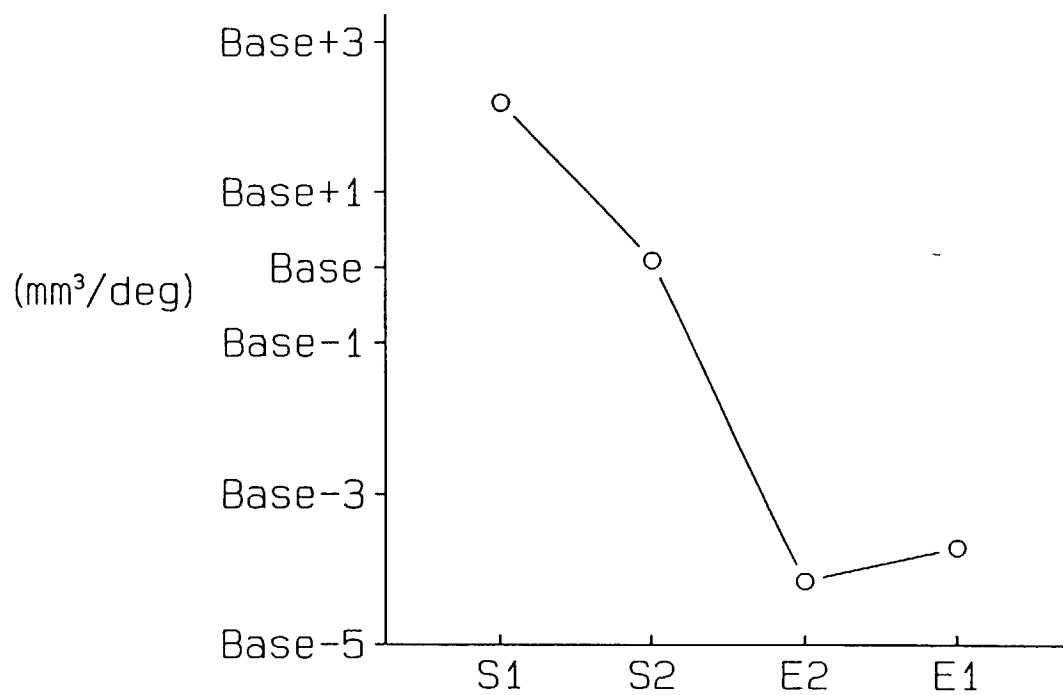
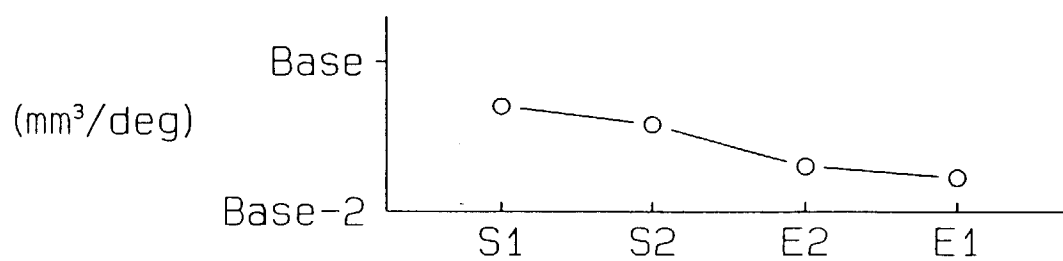
Fig.13**Fig.14**

Fig.15

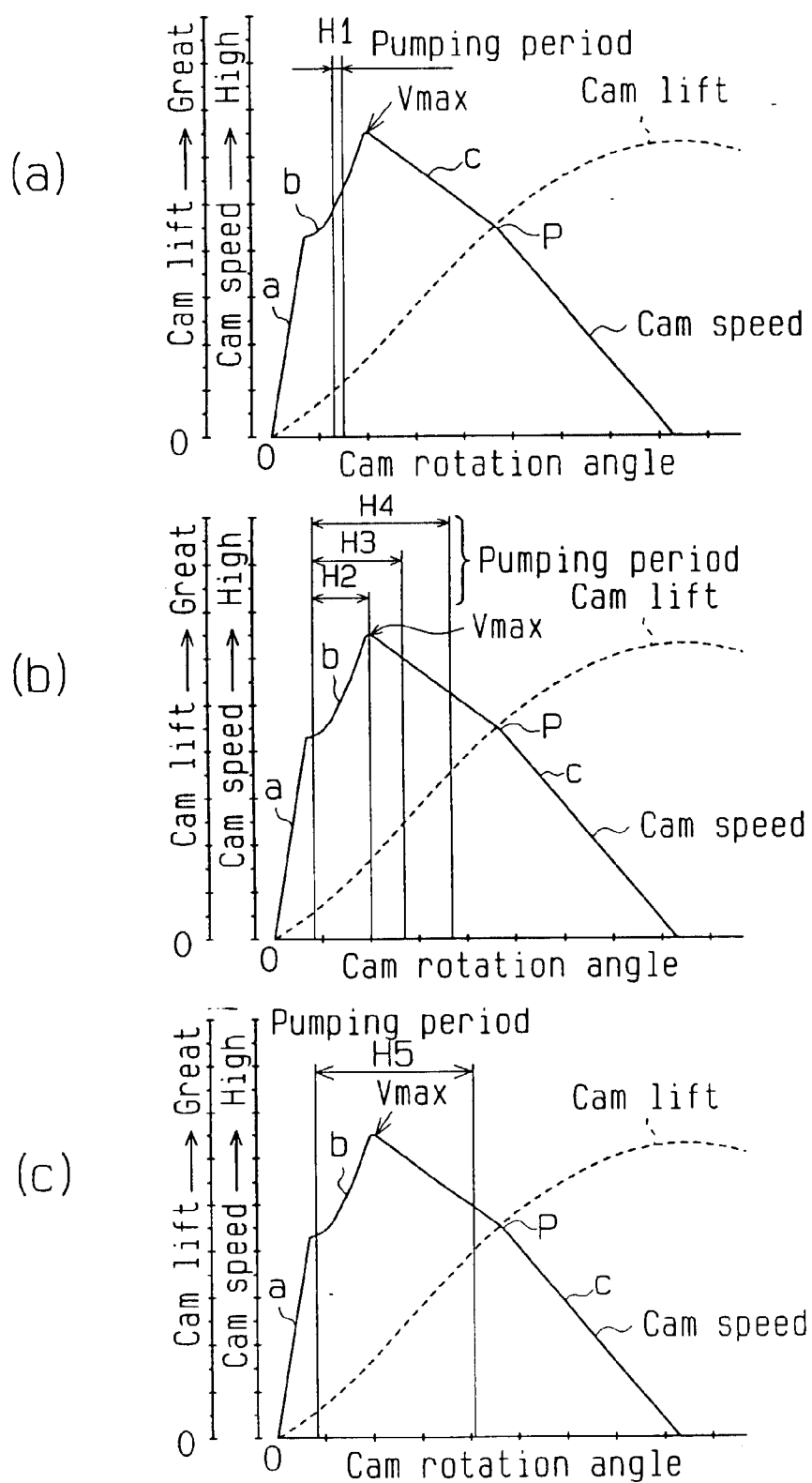


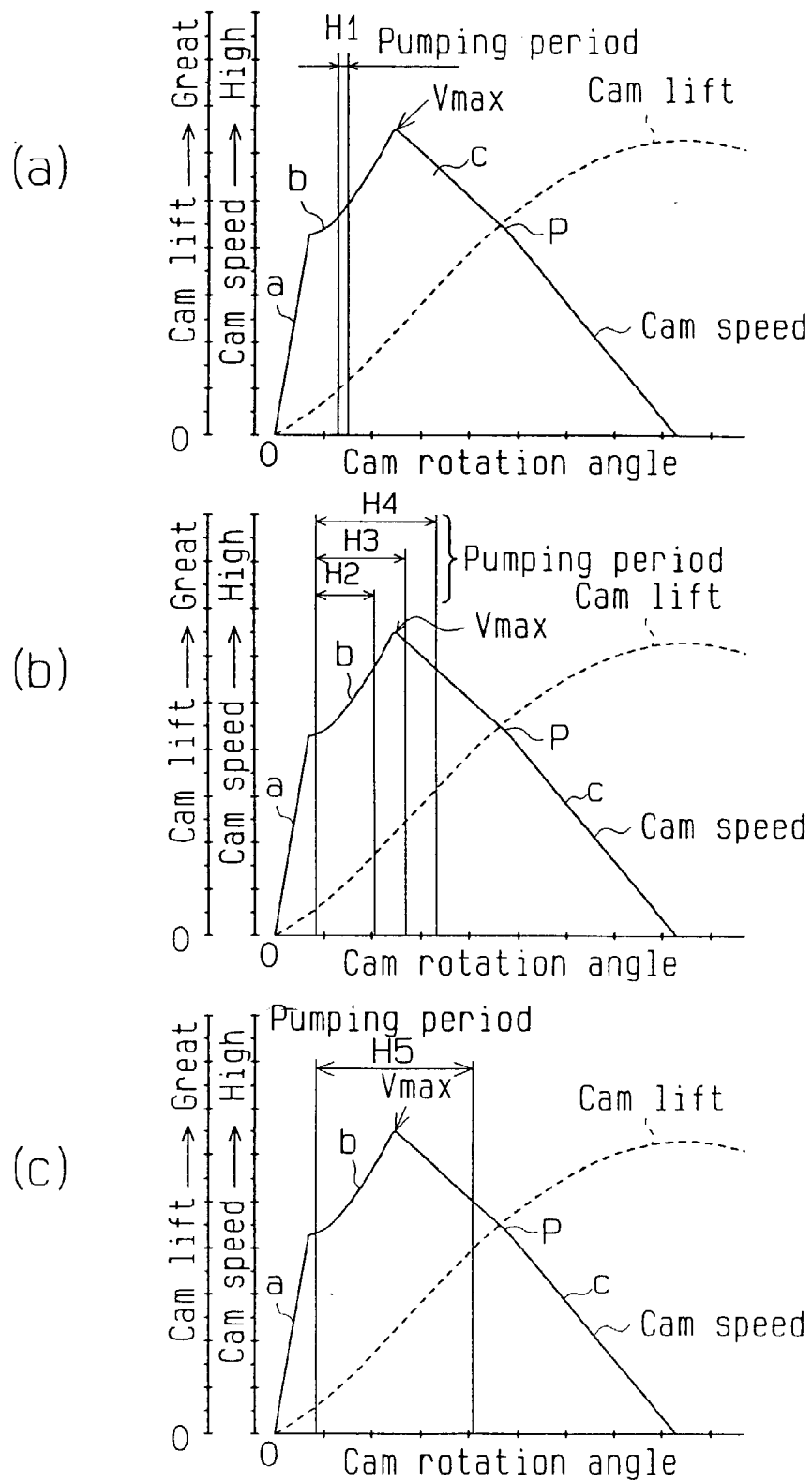
Fig.16

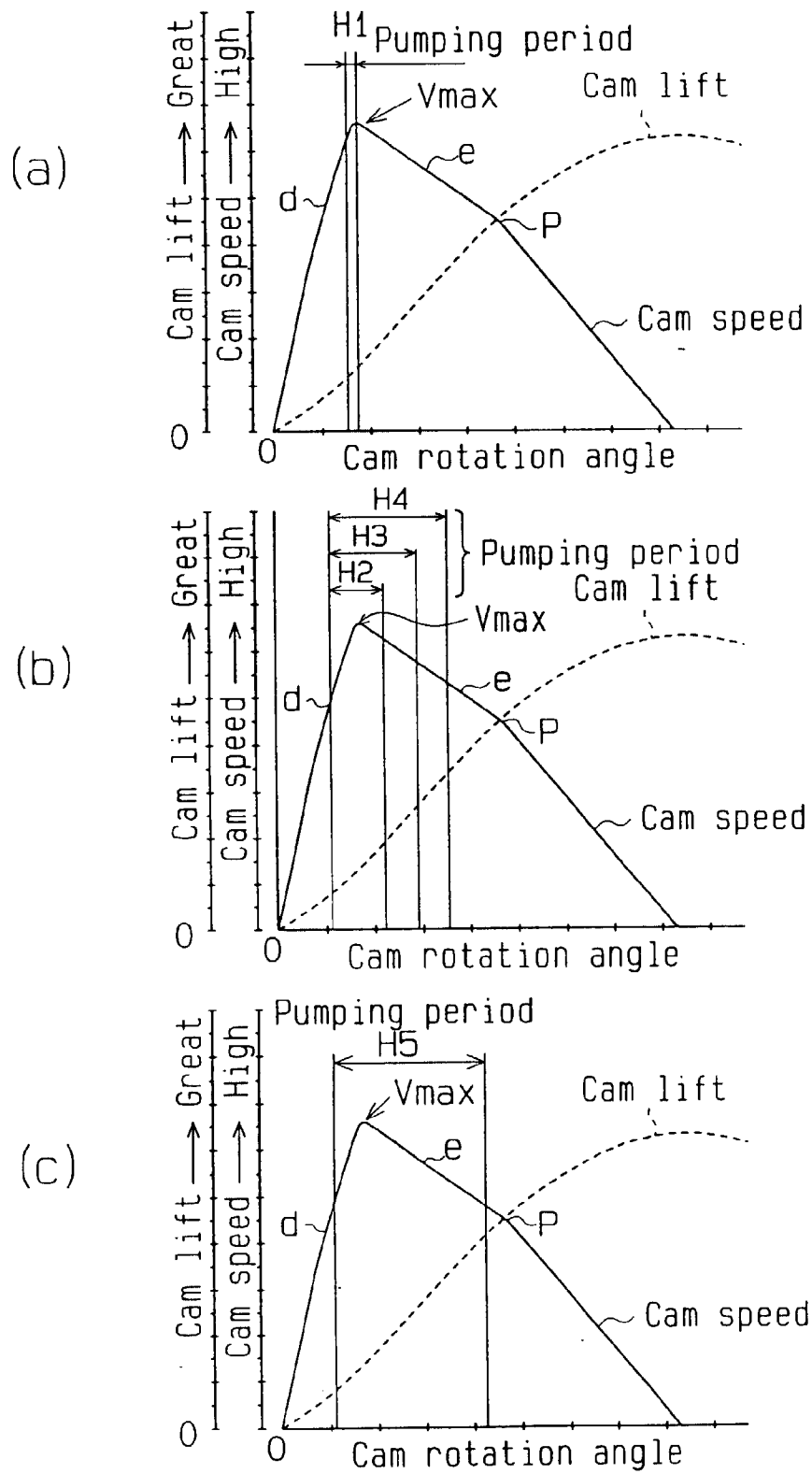
Fig.17

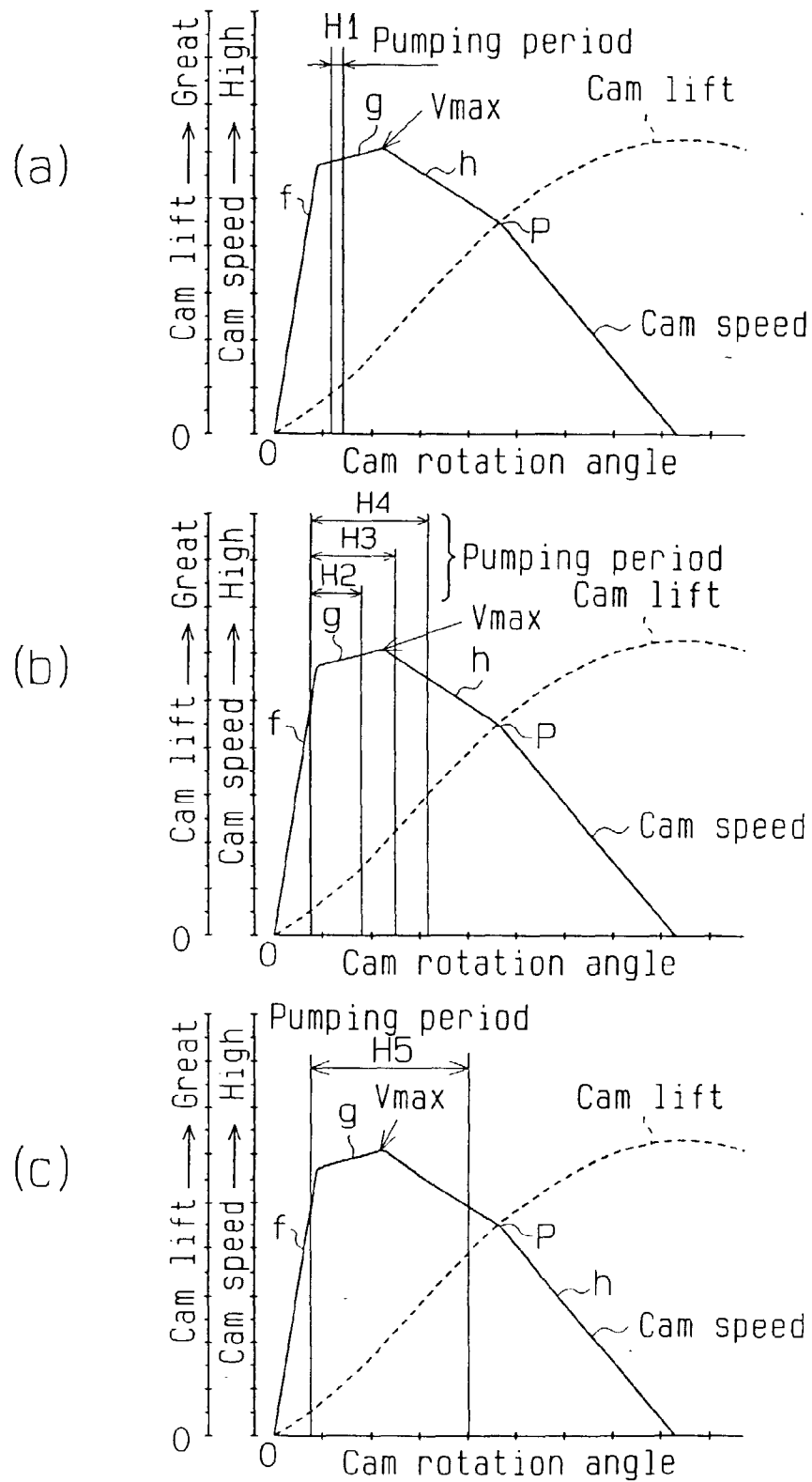
Fig.18

Fig.19

