

(19)



Europäisches Patentamt
European Patent Office
Office européen des brevets



(11)

EP 0 849 466 A1

(12)

EUROPEAN PATENT APPLICATION

(43) Date of publication:
24.06.1998 Bulletin 1998/26

(51) Int Cl.⁶: F02M 41/14

(21) Application number: 97309751.2

(22) Date of filing: 03.12.1997

(84) Designated Contracting States:
AT BE CH DE DK ES FI FR GB GR IE IT LI LU MC
NL PT SE
Designated Extension States:
AL LT LV MK RO SI

- Matsubara, Jun, c/o Zexel Corporation Higashimatsuyama, Saitama (JP)
- Narikiyo, Kazuaki, c/o Zexel Corporation Higashimatsuyama, Saitama (JP)
- Shiraishi, Tomohiko, c/o Zexel Corporation Higashimatsuyama, Saitama (JP)

(30) Priority: 19.12.1996 JP 354547/96
17.10.1997 JP 303493/97

(74) Representative: Matthews, Howard Nicholas et al
Page Hargrave,
Temple Gate House
Temple Gate, Bristol BS1 6PL (GB)

(71) Applicant: ZEXEL CORPORATION
Tokyo (JP)

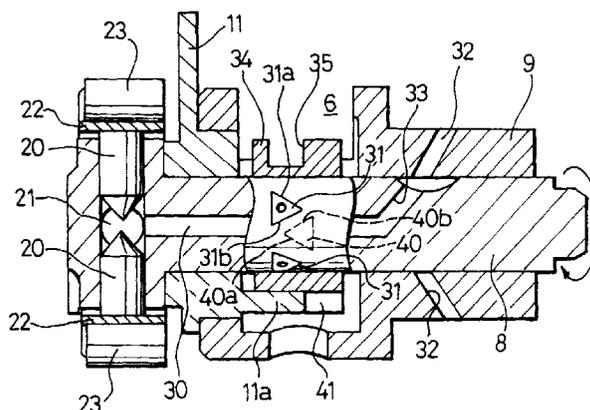
(72) Inventors:
• Ymaguchi, Susumu, c/o Zexel Corporation Higashimatsuyama, Saitama (JP)

(54) Distributor type fuel injection pump

(57) In a fuel injection pump employing an inner cam system, inflow / outflow ports (31) for taking in and cutting off fuel are formed at a rotor (8) that rotates in synchronization with an engine, a communicating hole (40) that is capable of coming into communication with these ports (31) is formed at a control sleeve (34) and two inclined edges (31a,31B,40a,40B) are formed inclining in opposite directions from a hypothetical line extending parallel to the axis at an opening end of each of the inflow / outflow ports (31) and the communicating hole (40). The inclined edges (31a,40a) of the inflow / outflow

ports (31) and of the communicating hole (40) that cross each other at the end of communication between them are formed parallel to each other and the inclined edges (31B,40B) that cross each other at the start of their communication are formed parallel to each other. This achieves a full advance angle at startup of the engine without having to rely on a timing device in the fuel injection pump employing an inner cam system. Even after startup, the oil feed rate at medium to low load can be raised while keeping the drive torque under the permissible value at high load.

FIG. 3



EP 0 849 466 A1

Description

The present invention relates to a distributor type fuel injection pump that force feeds fuel for injection into an engine via nozzles, and in particular, it relates to a distributor type fuel injection pump that varies the volumetric capacity of a compression space formed at a rotor that rotates in synchronization with the engine by causing plungers provided slidably in the direction of the radius at the rotor to make reciprocal movement with a cam ring.

In a distributor type fuel injection pump employing an inner cam system, an inner cam (cam ring) is provided concentrically around a fuel distribution rotating member (rotor) and force-feed plungers are provided at cam surfaces formed on the inner side of the inner cam via rolling elements or the like to cause the force-feed plungers to make reciprocal movement in the direction of the radius of the fuel distribution rotating member, as disclosed in, for instance, Japanese Unexamined Patent Publication No. 59-119056, Japanese Unexamined Patent Publication No. 60-79152, and Japanese Unexamined Patent Publication No. 3-175143. At the fuel distribution rotating member, a pump chamber (compression space) whose volumetric capacity is varied by the force-feed plungers, intake holes through which fuel is taken in to the pump chamber during an intake phase, a distribution port through which fuel that has been pressurized in the pump chamber is delivered during a force-feed phase and overflow ports for cutting off fuel delivery are formed. In addition, a ring-like member (control sleeve) is externally fitted on the fuel distribution rotating member, covering the overflow ports, and by moving this ring-like member in the axial direction, the cutoff timing during the force-feed phase is changed to vary the injection quantity.

In such a distributor type fuel injection pump in the known art, a cam ring is caused to rotate to control its advance angle by a hydraulic (fuel pressure type) timing device, as disclosed in, for instance, Japanese Unexamined Patent Publication No. 8-270521. This distributor type fuel injection pump has a basic structure in which a timer piston is provided at a right angle to the direction of the axis of the rotor, with the supply pressure of the feed pump applied to one end of the timer piston and a timer spring provided at the other end of the timer piston, the timer piston takes a position at the point where the supply pressure and the spring force imparted by the timer spring are in balance and the cam ring, which interlocks with the timer piston, is thereby caused to rotate so that an advance angle is achieved which corresponds to the supply pressure. In addition, it is further provided with a timer control valve to adjust the fuel pressure applied to the timer piston so that the advance angle can be freely changed.

Moreover, the first three publications mentioned above disclose that a groove portion is formed on the inside of the ring-like member (control sleeve), with one

end of the groove portion made to incline relative to the generating line of the ring-like member. The publications disclose that with this structure, while the stroke (pre-stroke) occurring after the cam lift starts until the force feed starts is set at a constant value, the force feed end stroke occurring until the force feed end, changes in correspondence to the position of the ring-like member in the axial direction, i.e., in correspondence to the load.

However, in the hydraulic timing device described above, since it basically utilizes the supply pressure of the feed pump, a considerable length of time elapses after the pump is started up until the supply pressure rises to full pressure. Thus, in an engine that requires that a large advance angle be achieved at startup, there is difficulty in achieving a sufficient advance angle with this timing device. In other words, at the time of startup, a state of conflict exists in that a large advance angle is required immediately, while the timing device is not yet ready to function fully.

In addition, in the prior art technology described above, since the fuel force-feed phase is started from a range over which the oil feed rate (cam speed X plunger diameter) is small, it is structurally difficult to fully assure a sufficient oil feed rate at the pump during normal operation after startup, in order to satisfy the need for improving engine performance during medium to low load operation. If it is the drive torque at medium to low load only that is to be improved, the overall oil feed rate characteristics simply need to be set higher by employing a cam profile with a high cam speed. However, this causes a problem in that, with operation at medium to low load given too much priority, the drive torque for the pump at high load exceeds the permissible value.

Accordingly, an object of the present invention is to provide a distributor type fuel injection pump that is capable of achieving a sufficient advance angle at the time of pump startup without having to wait for the supply pressure to rise at the pump so that the timing device can function. Another object of the present invention is to provide a distributor type fuel injection pump that is capable of assuring good injection performance over the entire load range during normal operation by increasing the oil feed rate at medium to low load and keeping the drive torque for the pump at a level equivalent to the level in the prior art without exceeding the permissible value at high load.

While an advance angle is achieved with the timing device by causing the cam ring to rotate in the direction opposite from the direction in which the rotor rotates, to hasten the injection timing, the same effect can be achieved by changing the timing with which the holes for intake and cut-off formed at the rotor cease to communicate with the hole in the control sleeve that is provided covering them (force feed start timing), by adjusting the control sleeve. If the force feed start timing can be varied through adjustment of the position of the control sleeve, it becomes possible to achieve a specific advance angle at startup independent of the timing device.

In addition, if the cam engagement range during a force feed period can be changed as appropriate in relation to the oil feed rate as the control sleeve becomes displaced in correspondence to the load, the required oil feed rate can be obtained in correspondence to the load. The inventor of the present invention has conducted focused research on a structure which would meet these requirements and the effort has culminated in the present invention.

Namely, the distributor type fuel detection pump according to the present invention comprises a rotor that rotates in synchronization with an engine, plungers that are slidably provided in the direction of the radius of the rotor to vary the volumetric capacity of a compression space formed at the rotor, a cam ring that is provided concentrically around the rotor and regulates the movement of the plungers, and a control sleeve that is externally fitted at the rotor in a fuel chamber at the downstream side of the feed pump and moves in the direction of the axis of the rotor so that its position relative to the rotor can be adjusted. At the rotor, first holes for taking in and cutting off fuel by coming into communication with the compression space are formed, whereas at the control sleeve, a second hole which is capable of coming into communication with the first holes is formed. At the first and second holes, each of the opening ends that are nearest each other have two inclined edges that incline relative to a hypothetical line that is parallel to the axis of the rotor toward the side farthest from each other are formed, with the inclined edges of the first holes and the inclined edge of the second hole that cross each other at the start of communication formed parallel to each other and the inclined edges of the first holes and the inclined edge of the second hole that cross each other at the end of communication formed parallel to each other.

Since the first holes and the second hole are each provided with two inclined edges inclining from a hypothetical line extending in the direction of the axis of the rotor toward the sides farthest from each other, the opening end of each hole can be machined in a triangular shape easily, and the first holes and the second hole may be formed so that the sides at which the intersecting points of their inclined edges are present (sides corresponding to the apexes of the triangles) are opposite from each other in the axial direction.

As a result, an intake phase, in which fuel is taken in, is effected during the period of time over which the rotor rotates and a first hole and the second hole are in communication with each other, and a force-feed phase is effected during the period of time elapsing after the first hole becomes disengaged from the second hole until the next first hole comes into communication with the second hole. This force-feed phase is in synchronization with the period over which the plungers are lifted by the cam ring. In fact, however, fuel force feed starts after the communication between the first hole and the second hole is cut off following the start of the plunger lift.

The period elapsing after the start of plunger lift until the communication between the first hole and the second hole is cut off constitutes a pre-stroke period, and the shorter this period the larger the advance angle.

5 While it is difficult to achieve a specific advance angle at startup with a timing device that utilizes the supply pressure of the feed pump, since the timing with which a first hole and the second hole which are provided with the inclined edges cease to communicate with each other can be adjusted by moving the control sleeve in the axial direction, the pre-stroke period can be reduced to achieve a specific advance angle without depending upon the timing device. Thus, the structure described above is particularly effective when adopted in an injection pump provided with a timing device for adjusting the advance angle through rotation of the cam ring. In addition, the pump structure described above is employed in such a manner that the range over which the cam ring is engaged to force feed fuel increases toward the advance angle side and the retard angle side to contain the middle range of the rise area of the cam surfaces of the cam ring as the load increases. Thus, if the control sleeve is moved so as to reduce the pre-stroke period, the timing with which a first hole and the second hole come into communication with each other, i.e., the timing with which fuel is cut off, too, is retarded, resulting in a longer period of time for fuel force feed, to increase the injection quantity. Consequently, an advance angle is achieved and, at the same time, by increasing the injection quantity, startup torque can be increased, at the time of startup, thereby achieving the injection state that ideally suits the requirements for startup.

Since, after the engine is started up, the supply pressure of the feed pump increases and the timing device starts to function, the advance angle control during normal operation can be performed through the timing device, and the injection quantity control is implemented through the operation of the control sleeve, as with an injection device in the prior art. While, in the structure according to the present invention, the pre-stroke quantity will be changed if the control sleeve is moved in order to change the injection quantity even during normal operation, this change can be compensated for by the timing device to achieve problem-free operation.

45 Furthermore, for the cam ring engagement range for force feeding fuel, it is desirable to use only the cam speed range over which the cam speed is at or higher than a specific speed when the load is at or lower than a specific load and to use a range that includes the cam speed range over which the cam speed is lower than the specific speed when the load is greater than the specific load. With such a structure, since the range over which the cam speed is high (the range over which the oil feed rate is high) can be used at medium to low load even during normal control after startup, the oil feed rate can be improved compared to that in a fixed pre-stroke type pump in the prior art, in which the cam ring must be used starting from the range over which the cam

speed is low. At the same time, at high load, the cam can be engaged over to the low cam speed range, thereby making it possible to keep the drive torque down to a level equivalent to that in the pump in the prior art.

The above and other features of the invention and the concomitant advantages will be better understood and appreciated by persons skilled in the field to which the invention pertains in view of the following description given in conjunction with the accompanying drawings which illustrate a preferred embodiment. In the drawings:

FIG. 1 is a cross section of the distributor type fuel injection pump according to the present invention; FIG. 2 is an enlarged cross section of the cam ring shown in FIG. 1 and members provided on its inside, viewed from the direction of the axis of the rotor;

FIG. 3 is an enlarged cross section showing the rotor and the members around the rotor;

FIG. 4 illustrates the changes that occur in the positional relationship between the inflow / outflow ports 31 and the communicating hole 40 at the control sleeve as the rotor rotates, with the communicating hole presented in the broken line indicating its position during idling and the communicating hole presented in the solid line indicating its position at startup;

FIG. 5 presents a graph of the force feed period, the advance angle state and the pre-stroke relative to the cam angle during idling and at startup;

FIG. 6 is a graph of the force feed period (force feed angle) in the present invention and that in a prior art product at low load on an oil feed rate characteristics curve;

FIG. 7 is a graph of the force feed period (force feed angle) in the present invention and that in a prior art product at medium load on an oil feed rate characteristics curve;

FIG. 8 is a graph of the force feed period (force feed angle) in the present invention and that in a prior art product at high load on an oil feed rate characteristics curve;

FIG. 9 is a double graph illustrating the relationships among the force feed angle, the pre-stroke, the force feed end stroke and the effective stroke at different loads; and

FIG. 10 is a graph of the relationship between the change in the load and the drive torque.

The following is an explanation of an embodiment of the present invention in reference to the drawings. In FIG. 1, which illustrates a distributor type fuel injection pump employing an inner cam system, a drive shaft 3 is inserted in a pump housing 2 in a distributor type fuel injection pump 1, with one end of the drive shaft 3 projecting out of the pump housing 2 to receive drive torque from an engine (not shown) so that it rotates in synchro-

nization with the engine (at a rotation rate half the rotation rate of the engine). The other end of the drive shaft 3 extends into the pump housing 2, and a feed pump 4 is linked to the drive shaft 3 so that fuel supplied via a low pressure fuel region 5a, which is to be detailed later, is then supplied to a fuel chamber 6 by the feed pump 4.

The pump housing 2 comprises a housing member 2a, in which the drive shaft 3 is inserted, a housing member 2b that is mounted at the housing member 2a and is provided with delivery valves 7 and a housing member 2c that is provided on an extended line of a rotor 8 blocking off the opening portion of the housing member 2b. The fuel chamber 6 is constituted of the space enclosed by a rotor supporting member 9 provided inside the pump housing, a wall member 10 that holds the rotor supporting member 9 and an adapter 11, which is to be detailed later. The fuel chamber 6 communicates with a governor storage chamber 13, the extent of which is defined by a governor housing 12. In addition, the rotor supporting member 9 is fitted in an insertion hole 14 formed at the housing member 2b, which is provided with the delivery valves 7.

The rotor 8 is rotatably supported at an insertion hole 15 formed at the rotor supporting member 9 with a high degree of oil-tightness, having its base end portion linked to the drive shaft 3 via a coupling 16 so that only rotation that corresponds with the rotation of the drive shaft 3 is allowed. In addition, a spring 19 is provided between a spring receptacle 18 provided at the front end portion of the rotor 8 via a thrust bearing 17, and the housing member 2c, to apply a force to the rotor 8 toward the coupling to eliminate play in the axial direction.

At the base end portion of the rotor 8 that links with the drive shaft, plungers 20 are inserted slidably in the direction of the radius (radial direction), as shown in FIG. 2. In this structural example, four plungers 20 are provided on the same plane over 90° intervals. The front end of each of the plungers 20 blocks off and faces a compression space 21 which is provided at the center of the base end portion of the rotor 8. The base end of each plunger 20 slides in contact with the internal surface of a ring-like cam ring 24 via a shoe 22 and a roller 23. This cam ring 24 is provided concentrically around the rotor 8, with cam surfaces 24a, the number of which corresponds to the number of cylinders in the engine, formed on the inside so that when the rotor 8 rotates, each plunger 20 makes reciprocal movement in the direction of the radius of the rotor 8 (radial direction) to vary the volumetric capacity of the compression space 21.

In other words, the cam ring 24, which is formed to support four cylinders, is provided with projecting surfaces on the inside of the cam ring 24 over 90° intervals. As a result, the four plungers 20 move together to perform compression by clamping the compression space 21 and withdraw together from the center of the cam ring 24.

The circular adapter 11 is externally fitted at the ro-

tor 8 rotatably, with a portion of the circumferential edge of the adapter 11 retained by the cam ring 24 to restrict its rotation so that it can only rotate together with the cam ring 24. In addition, the adapter 11 is fitted at the rotor supporting member 9 rotatably.

At the upper portion of the housing member 2b, a fuel inflow port 25 for inducing fuel from a fuel tank (not shown) is provided, and fuel flowing in through the fuel inflow port 25 is induced from a fuel supply passage 26 formed at the housing member 2b through the space formed around the wall member 10 and the adapter 11, the space formed between the cam ring 24 and the rotor 8 and the area surrounding the coupling 16 toward the intake side of the feed pump 4. These areas constitute the low pressure fuel region 5a ranging from the fuel inflow port 25 to the feed pump 4.

In addition, the fuel that has been compressed by the feed pump 4 travels via a passage 27 formed at the upper portion of the pump housing and a gap 28 formed between the pump housing 2 and the governor housing 12 mounted at the pump housing 2 to be induced toward the fuel chamber 6, and fuel is also induced via the governor storage chamber 13 to an overflow valve 29, with these communicating areas constituting a high pressure fuel region 5b.

As shown in FIG. 3, a longitudinal hole 30 that is formed in the axial direction and communicates with the compression space 21, inflow / outflow ports 31 that communicate with the longitudinal hole 30 and open at the circumferential surface of the rotor 8 and a distribution port 33 that enables communication between fuel delivery passages 32 formed at the rotor supporting member 9 and the housing member 2b and the longitudinal hole 30 are formed at the rotor 8. The number of inflow / outflow ports 31 provided corresponds to the number of cylinders, and they are formed with their phases offset over equal intervals. Each of the inflow / outflow ports 31 is formed to have a roughly triangular shape. To be more specific, in each of the inflow / outflow ports 31, an edge 31a, which determines the timing with which the communication with the inflow / outflow port 31 ends, is formed inclining at a specific angle relative to the direction of the axis of the rotor 8 and an edge 31b, which determines the timing with which the communication with the inflow / outflow port 31 starts, is formed inclining in the opposite direction from the edge 31a relative to the direction of the axis of the rotor 8.

The inflow / outflow ports 31 open at the surface of the rotor 8 at positions corresponding to the fuel chamber 6, and these opening portions are covered by a control sleeve 34 which is externally fitted at the rotor 8 with a high degree of oil tightness. A connecting groove 35 is formed over a specific angular range in the direction of the circumference at the upper surface of the control sleeve 34, and a lug 38 formed at the front end of a shaft 37 of an electric governor 36 is engaged in the connecting groove 35. The lug 38 is provided decentered from the shaft 37, and when the shaft 37 is caused to rotate

by an external signal, the control sleeve 34 is caused to move in the direction of the axis of the rotor 8.

In addition, a communicating hole 40 that is capable of sequentially coming into communication with the individual inflow / outflow ports 31 is formed as indicated with the broken line in FIG. 3 at the control sleeve 34. This communicating hole 40 has a roughly triangular shape formed in symmetrical relation to the inflow / outflow ports 31, with an edge 40a that determines the timing with which its communication with an inflow / outflow port 31 ends, formed inclining relative to the direction of the axis of the rotor 8 and extending parallel to the edge 31a of the inflow / outflow port 31 and an edge 40b that determines the timing with which its communication with an inflow / outflow port 31 starts, formed inclining toward the opposite direction from the edge 31a relative to the direction of the axis of the rotor 8 and extending parallel to the edge 31b of the inflow / outflow port 31.

Since the range over which the control sleeve 34 can move in the direction of the axis of the rotor is determined in relation with the governor and the quantities of changes in the timing with which the force feed starts and the timing with which cut-off occurs per unit movement of the control sleeve 34 are determined accordingly, the angles of inclination of the edges 31a and 40a that are formed at the inflow / outflow ports 31 and the communicating hole 40 to determine the timing of communication end and the angles of inclination of the edges 31b and 40b that determine the timing of communication start are individually determined based upon these quantities of change and they do not necessarily have to be set at the same angle.

In addition, below the control sleeve 34, a retaining groove 39 is formed extending in the axial direction, and a projecting portion 11a of the adapter 11 is retained in the retaining groove 39 to maintain a constant phase relationship between the adapter 11 and the control sleeve 34 at all times.

A timing device 140 is constituted by storing a timer piston 41 slidably in a cylinder provided in the lower portion of the pump housing 2 and linking the timer piston 41 to the cam ring 24 via a lever 42 to adjust the injection timing by converting the movement of the timer piston 41 into rotation of the cam ring 24.

At one end of the timer piston 41, a high pressure chamber into which high pressure fuel in the high pressure fuel region 5b is induced is formed, whereas at the other end, a low pressure chamber that communicates with the low pressure fuel region 5a is formed. Furthermore, a timer spring is provided in the low pressure chamber and this timer spring applies a constant force to the timer piston 41 toward the high pressure chamber at all times. Consequently, the timer piston 41 stops at the position at which the spring pressure imparted by the timer spring and the pressure of the fuel in the high pressure chamber are in balance, and when the pressure in the high pressure chamber increases, the timer piston 41 moves toward the low pressure chamber

against the force imparted by the timer spring to cause the cam ring 24 to rotate in the direction in which the injection timing is hastened, resulting in the injection timing being advanced. In addition, when the pressure in the high pressure chamber becomes lower, the timer piston 41 moves toward the high pressure chamber causing the cam ring 24 to rotate in the direction in which the injection timing is delayed to retard the injection timing. It is to be noted that the pressure in the high pressure chamber at the timer is adjusted by a timing control valve (TCV) 43 to achieve the required timer advance angle.

In the structure described above, when the rotor 8 rotates, the inflow / outflow ports 31, the number of which corresponds to the number of cylinders, come into communication with the communicating hole 40 of the control sleeve 34 sequentially and in an intake phase, during which the plungers 20 move away from the center of the cam ring 24, an inflow / outflow port 31 and the communicating hole 40 of the control sleeve 34 become aligned to allow the fuel to be taken into the compression space 21 from the chamber 6.

Then, when the operation enters the force-feed phase, during which the plungers 20 move toward the center of the cam ring 24, the communication between the inflow / outflow port 31 and the communicating hole 40 of the control sleeve 34 is cut off, the distribution port 33 and one of the fuel delivery passages 32 become aligned and compressed fuel is delivered to a delivery valve 7 via this fuel delivery passage 32. After this, the fuel delivered from the delivery valve 7 is sent to an injection nozzle via an injection pipe (not shown) to be injected into a cylinder of the engine from the injection nozzle.

When, during the force-feed phase, an inflow / outflow port 31 and the communicating hole 40 of the control sleeve 34 come into communication with each other again, the compressed fuel flows out into the chamber 6 and the fuel delivery to the injection nozzle is stopped to end the injection. This process is sequentially repeated, resulting in four cycles completed for each rotation of the rotor.

Since the inflow / outflow ports 31 and the communicating hole 40 of the control sleeve 34 are formed in triangular shapes, as explained above, the timing with which an inflow / outflow port 31 and the communicating hole 40 end their communication and the timing with which they start to communicate with each other can be varied with the control sleeve 34. In other words, the force feed start timing and the force feed end timing (cut-off timing) can be adjusted through the positional adjustment of the control sleeve 34, and as the control sleeve 34 is moved to the left in FIG. 3 (toward the base end portion of the rotor 8), the force feed start timing is delayed and the force feed end is hastened, whereas, as it is moved to the right in the figure (toward the front end portion of the rotor 8), the force feed start timing is hastened and the force feed end is delayed.

The following is a specific explanation of this relationship at startup and during idling of the engine in reference to FIGS. 4 and 5. In FIG. 4, the inflow / outflow ports 31 and the communicating hole 40 of the control sleeve are shown in a sequence of positions on the same plane to illustrate how the inflow / outflow ports 31 gradually become offset in the downward direction in the figure relative to the communicating hole 40 as the rotor rotates. The communicating hole 40 is represented with the solid line indicating its position at startup and the communicating hole represented with the broken line indicates its position during idling. In addition, the cam angles (I) ~ (IV) in FIG. 4 are achieved within a period over which the cam lift gradually increases from zero, and they respectively correspond to the positions (I) ~ (IV) in FIG. 5, for instance.

Since the edges of the inflow / outflow port 31 and the communicating hole 40 that determine the timing of communication end and the timing of communication start are made to incline as described above, the following injection characteristics are achieved during idling. Namely, when the cam angle is at (I), an inflow / outflow port 31 and the communicating hole 40 are still in communication with each other although the cam lift has started, and the fuel is leaking into the fuel chamber. Then, when the cam angle is at (II), the communication between the inflow / outflow port 31 and the communicating hole is cut off, the compression of the fuel that has been taken in starts and compressed fuel is forced into a fuel delivery passage. This force feed state is sustained until the cam angle is at (III). When the cam angle is at (III) the next inflow / outflow port 31 and the communicating hole 40 start to communicate with each other and the fuel is cut off. When the cam angle is at (IV), while the volumetric capacity of the compression space is further reduced, since the inflow / outflow port 31 and the communicating hole are in communication with each other, the fuel leaks into the fuel chamber.

As a result, as shown in FIG. 5, force feed is performed in the intermediate range of the cam angle from (II) to (III), thereby reducing the length of the force feed period, which in turn, results in a reduced injection quantity and an increase in the quantity of pre-stroke occurring after the start of cam lift until force feed starts.

In contrast, at startup of the engine, the communication between an inflow / outflow port 31 and the communicating hole is cut off when the cam angle is at (I), and the compression of the fuel that has been taken in starts, at this point, to force feed compressed fuel into the fuel delivery passages. This force feed state is sustained until the cam angle reaches (IV), and when the cam angle is at (IV) the next inflow / outflow port 31 and the communicating hole start to communicate with each other, thereby cutting off the fuel to end the force feed.

Thus, at startup, since a wider range of cam angle from (I) to (IV) is utilized as the force feed period, as shown in FIG. 5, the length of the force feed period is increased, thereby increasing the injection quantity. In

addition, the quantity of the pre-stroke occurring after the cam lift starts until the force feed starts becomes reduced.

In an injection apparatus in the prior art, the cam ring is made to rotate by a desired quantity by a timing device to change the injection timing (to perform advance angle control) and with this, the phase of the cam relative to the rotor 8 (or the drive shaft 3) is changed by offsetting the cam characteristics in FIG. 5 themselves to the left and to the right. However, in such advance angle control, it is a prerequisite that the timing device 140 function fully, and since the supply pressure of the feed pump is low at startup, as explained earlier, it is not possible to implement the advance angle control with the timing device 140. With this structural example, on the other hand, the injection timing can be hastened by operating the control sleeve 34 to offset the communicating hole 40 in the axial direction so that the communicating hole 40 departs from the inflow / outflow port 31, thereby making it possible to essentially achieve the required advance angle without having to offset the cam ring 24 at startup.

While an advance angle that cannot be achieved through the timing device 140 is obtained through control with the control sleeve 34 at startup in this manner, once the pump starts up to raise the supply pressure of the feed pump 4 (i.e., the chamber pressure) and the operation reaches a stage at which the advance angle adjustment can be fully implemented with the timing device 140, the advance angle control is performed by the timing device 140. In addition, while the control sleeve 34 is operated to change the injection quantity during normal operation, there is a problem with the structural example above in that when the control sleeve is moved to achieve a desired injection quantity, the injection timing is also changed. However, since the timing device functions in a normal manner during normal operation, the change in the injection timing can be corrected by causing the cam ring 24 to rotate with the timing device.

Moreover, from the viewpoint of controlling the injection rate during normal operation, changing the range over which the cam is engaged with the control sleeve without moving the cam ring (without offsetting the cam characteristics) leaves room for possible injection rate control implemented through the operation of the control sleeve. While it is true that, in the structure described above, the change in the range over which the cam is engaged, along with the change in the length of the force feed period, is the result of having moved the control sleeve in order to perform injection quantity control, and unlike in an injection pump in the prior art, the range over which the cam is engaged is not changed while maintaining consistency in the length of the force feed period, control similar to the injection rate control in the prior art is possible depending on how the cam surfaces are formed.

Furthermore, through the structure described above, the following advantages are achieved during

normal operation after startup. The following is a specific explanation of this point in reference to FIGS. 6 through 10. In the explanation, the characteristics curves indicated with the solid lines in FIGS. 6 through 8 schematically show the oil feed rate (cam speed X plunger diameter) characteristics and these are the characteristics of a conventional injection pump in which the oil feed rate is at its maximum in the intermediate range during the rise of the cam.

when the load is low (①) including the idling period, since the intermediate range of the cam lift is used as indicated with the broken line in FIG. 5 and, as explained earlier, the fuel is force fed during a short period of time over the range over which the oil feed rate is at its maximum, as can be ascertained from FIGS. 6 and 9, resulting in a large pre-stroke quantity and a small effective stroke quantity, i.e., a small quantity of cam lift achieved after the force feed start until the force feed end (Amm). In the figures, Uaist indicates the voltage applied to the actuator that drives the control sleeve, and since the movement of the control sleeve interlocks with the load, this Uaist is used as a parameter that indicates the load. The larger this value is, the larger the load.

In addition, at a medium load (②) since the intermediate range of the cam lift in effect during the injection period is wide, injection is performed over a wide range over which the oil feed rate is at its maximum as can be ascertained from FIGS. 7 and 9, resulting in a somewhat reduced prestroke quantity and a somewhat increased effective stroke quantity (Bmm).

At high load (③), since the range of the cam lift in effect during the injection period becomes even wider, as indicated by the startup force feed period in FIG. 5, injection is performed over an even wider period including the range over which the oil feed rate is at its maximum and a range over which the oil feed rate is low, thereby reducing the pre-stroke quantity and increasing the effective stroke quantity to its maximum (Cmm).

Now, the injection characteristics of the present invention described above are compared against those of prior art products disclosed in Japanese Unexamined Patent Publication No. 59-119056, Japanese Unexamined Patent Publication No. 60-79152, Japanese Unexamined Patent Publication No. 3-175143 and the like. With the prior art products, unless the cam ring is caused to rotate, the pre-stroke remains at a constant low level regardless of the load. when the load increases and the control sleeve becomes displaced, only the force feed end stroke increases. As a result, regardless of what the state of the load is, the fuel starts to be force fed from the low oil feed rate range.

The effective stroke of the prior art product is matched with that of the product according to the present invention to examine its range of utilization in the oil feed rate characteristics curve. As shown in FIGS. 6 through 8, when the load is small, the force feed angle (force feed period) greatly encroaches onto the range over which the oil feed rate is low, and as the load in-

creases, the force feed angle encroaches on the high oil feed rate range to a greater degree. At high load, the range covered is approximately equal to the range covered according to the present invention.

Due to such differences between the characteristics, and since the high oil feed rate range is utilized during a force feed at medium to low load according to the present invention, the oil feed rate during a force feed period can be increased compared to the prior art product, whereas an oil feed rate approximately equal to that of the prior art product is achieved during a force feed period at high load. Since the drive torque is determined in correspondence to the oil feed rate, the relationship as indicated with the solid line in FIG. 10 is achieved between the drive torque and the load, and when these characteristics are compared with those of the prior art product indicated with the one-point chain line, it is clear that while the drive torque increases at medium to low load as a result of improving the oil feed rate, the drive torque at high load is kept down to a degree equal to that in the prior art product.

In order to raise the oil feed rate at medium to low load in the structure of the prior art, the overall oil feed rate characteristics may be raised by forming a cam lobe which achieves a high cam speed to offset the characteristics curve of the drive torque indicated with the broken line in FIG. 10 upward. However, when designing an injection pump, the drive torque at maximum load (maximum drive torque) is often set very close to the permissible value. Thus, if the characteristics are changed to those indicated with the broken line by changing the shape of the cam, while the oil feed rate at medium to low load improves, the drive torque exceeds its permissible value at high load (the hatched area in FIG. 10), thereby necessitating reinforcement of the structure of the pump drive system (drive shaft, bearings and the like). Furthermore, the size of the pump itself may sometimes have to be increased in order to reinforce the structure of the pump drive system. In contrast, according to the present invention, good injection performance can be assured over the entire load range during normal operation without having to fortify the structure of the pump drive system by increasing the oil feed rate at medium to low load and keeping down the drive torque under the permissible value at high load.

As has been explained, according to the present invention, since two inclined edges that incline toward opposite directions from a hypothetical line which is parallel to the axis of the rotor are formed in each of the first holes formed at the rotor and in the second hole which is formed at the control sleeve and is capable of communicating with the first holes with the inclined edges of the first holes and the second hole that cross each other at the start of communication running parallel to each other and the inclined edges that cross each other at the end of communication parallel to each other, the timing with which a first hole and the second hole come into communication with each other and the timing with

which they cease to communicate with each other can be adjusted by moving the control sleeve in the axial direction.

As a result, the quantity of advance angle can be varied by changing the length of the pre-stroke period elapsing after plunger lift starts until communication between a first hole and the second hole is cut off, which makes it possible to obtain the required startup advance angle only through the control with the control sleeve even at an initial phase of startup when the supply pressure of the feed pump has not yet risen.

While, since the advance angle is varied by the control sleeve, the force feed start timing is changed because of the change in the pre-stroke quantity when the control sleeve is adjusted in order to change the injection quantity after startup, the timing device is fully capable of performing advance angle control by this point in time with the supply pressure of the feed pump having risen. Consequently, control of the injection quantity can be achieved without any problems by correcting the injection timing with the timing device.

In addition, the structure according to the present invention through which the pre-stroke is changed by interlocking with the load is effective for normal control as well as for control at startup, and good injection performance can be assured over the entire load range by keeping the drive torque at high load down to a level equal to that in the prior art while satisfying the requirement that the oil feed rate at medium to low load be fully assured.

Claims

1. A distributor type fuel injection pump (1) comprising:
 - a rotor (8) rotating in synchronization with an engine;
 - plungers (20) slidably provided in a direction of a radius of said rotor (8) and varying the volumetric capacity of a compression space (21) formed at said rotor (8);
 - a cam ring (24) concentrically provided around said rotor (8) to regulate the movement of said plungers (20);
 - a control sleeve (34) externally fitted at said rotor (8) in a fuel chamber (6) at a downstream side of a feed pump (4) with a position thereof relative to said rotor (8) adjusted by movement thereof in an axial direction of said rotor (8);
 - first holes (31) formed at said rotor (8) to take in and cut off fuel by coming into communication with said compression space (21); and
 - a second hole (40) formed at said control sleeve (34) and communicatable with said first holes (31);
 - two inclined edges (31a, 31b) formed at an opening end of each of said first holes (31) and

inclining in opposite directions from a hypothetical line extending parallel to said axis of said rotor (8); further provided with:

two inclined edges (40a, 40b) formed at an opening end of said second hole (40) at a side close to said first holes (31), and inclining in opposite directions from a hypothetical line extending parallel to an axis of said control sleeve (34); characterized in that:

inclined edges (31b, 40b) of said first holes (31) and said second hole (40) that cross each other at a start of communication therebetween are formed parallel to each other and inclined edges (31a, 40a) of said first holes (31) and said second hole (40) that cross each other at an end of communication therebetween are formed parallel to each other.

2. A distributor type fuel injection pump (1) according to claim 1, further comprising:

a timing device (140) adjusting an advance angle through rotation of said cam ring (24).

3. A distributor type fuel injection pump (1) according to claim 1 or claim 2, characterized in that:

said first holes and said second hole (31, 40) are each formed in a polygonal shape with a bottom side thereof running perpendicular to said hypothetical line; and

said inclined edges (31a, 31b, 40a, 40b) are provided at sides of said polygonal shape other than at said bottom side thereof.

4. A distributor type fuel injection pump (1) according to claim 3, characterized in that:

said first holes and said second hole (31, 40) are each formed in a triangular shape with a bottom side thereof running perpendicular to said hypothetical line; and

said inclined edges (31a, 31b, 40a, 40b) are provided at leg lines of said triangular shape.

5. A distributor type fuel injection pump (1) according to claim 1 or claim 2, characterized in that:

the timing with which communication between said first holes (31) and said second hole (40) ends is hastened and the timing with which communication therebetween starts is delayed as load increases.

6. A distributor type fuel injection pump (1) according to claim 1 or claim 2, characterized in that:

at startup of said engine, the length of time elapsing after start of cam lift until said communication between said first communicating hole (31) and said second communicating hole (40) ends is re-

duced by adjusting the relative position of said control sleeve (34) to obtain an advance angle.

7. A distributor type fuel injection pump (1) according to claim 1 or claim 2, characterized in that:

during idling of said engine, the length of time elapsing after start of cam lift until said communication between said first communicating hole (31) and said second communicating hole (40) ends is increased by adjusting the relative position of said control sleeve (34).

8. A distributor type fuel injection pump (1) according to claim 1 or claim 2, characterized in that:

a range over which said cam ring (24) is engaged in order to force feed fuel increases toward an advance angle side and toward a retard angle side to include an intermediate range over a rise range of cam surfaces (24a) of said cam ring (24) as said load increases.

9. A distributor type fuel injection pump (1) according to claim 1, characterized in that:

a range over which said cam ring (24) is engaged in order to force feed fuel only uses a range of cam speed over which said cam speed is at or over a specific speed when said load is at or less than a specific load, and encroaches over to use a range of cam speed over which said cam speed is lower than said specific speed as well when said load exceeds said specific load.

10. A distributor type fuel injection pump (1) according to claim 9, characterized in that:

said range over which said cam ring (24) is engaged in order to force feed fuel only covers a range of cam speed over which said cam speed is at maximum when said load is low, and includes from said range of cam speed over which said cam speed is at maximum and another range of cam speed as well when said load is high.

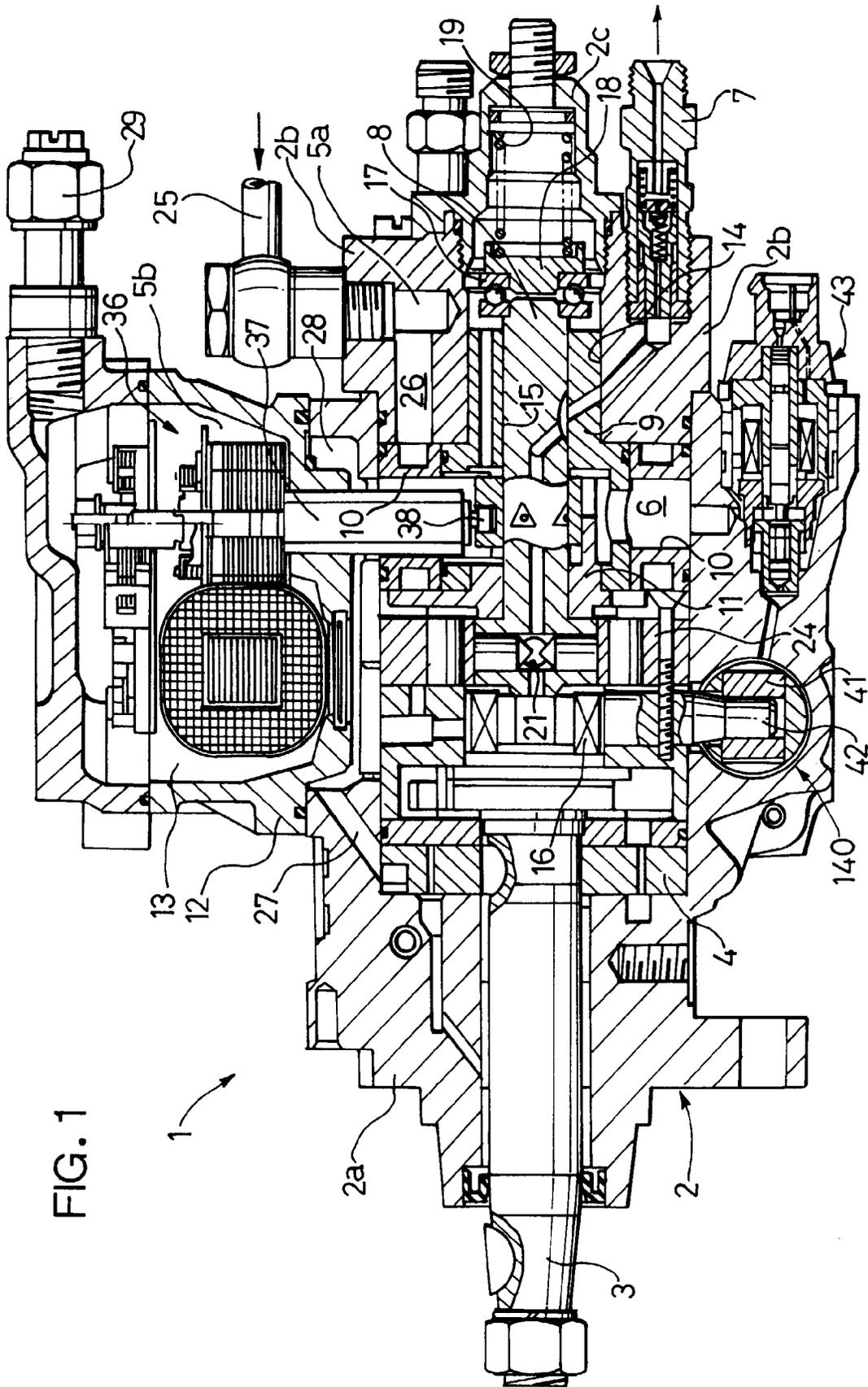


FIG. 2

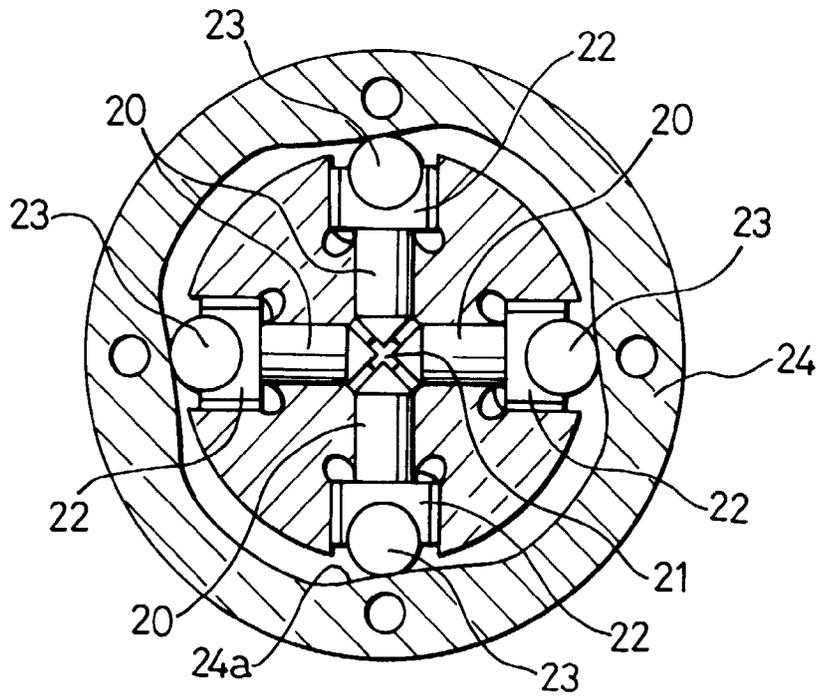


FIG. 3

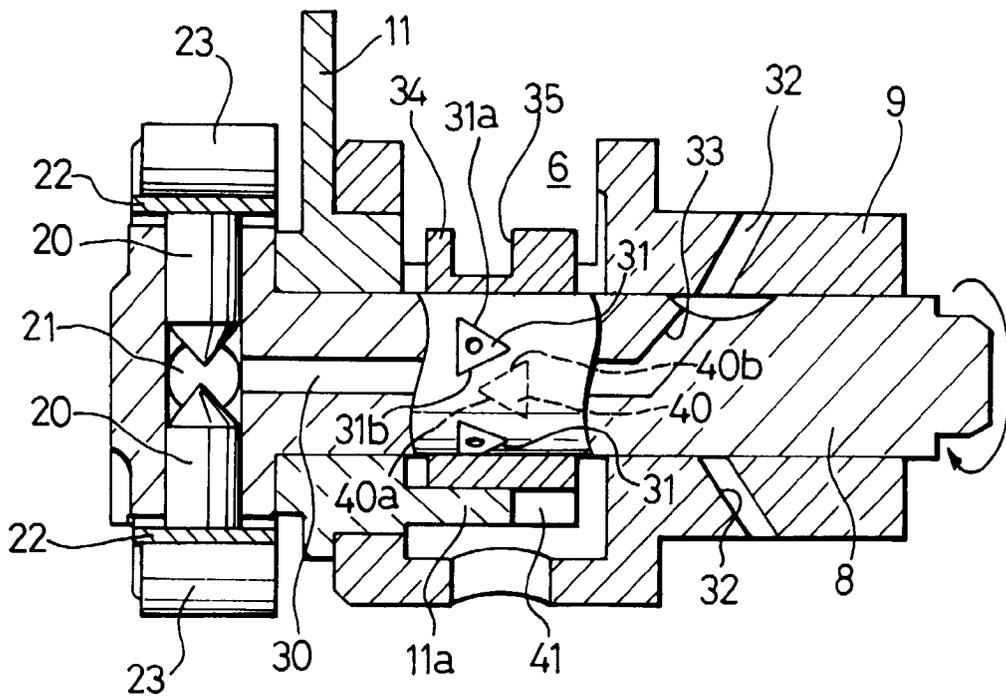


FIG. 4

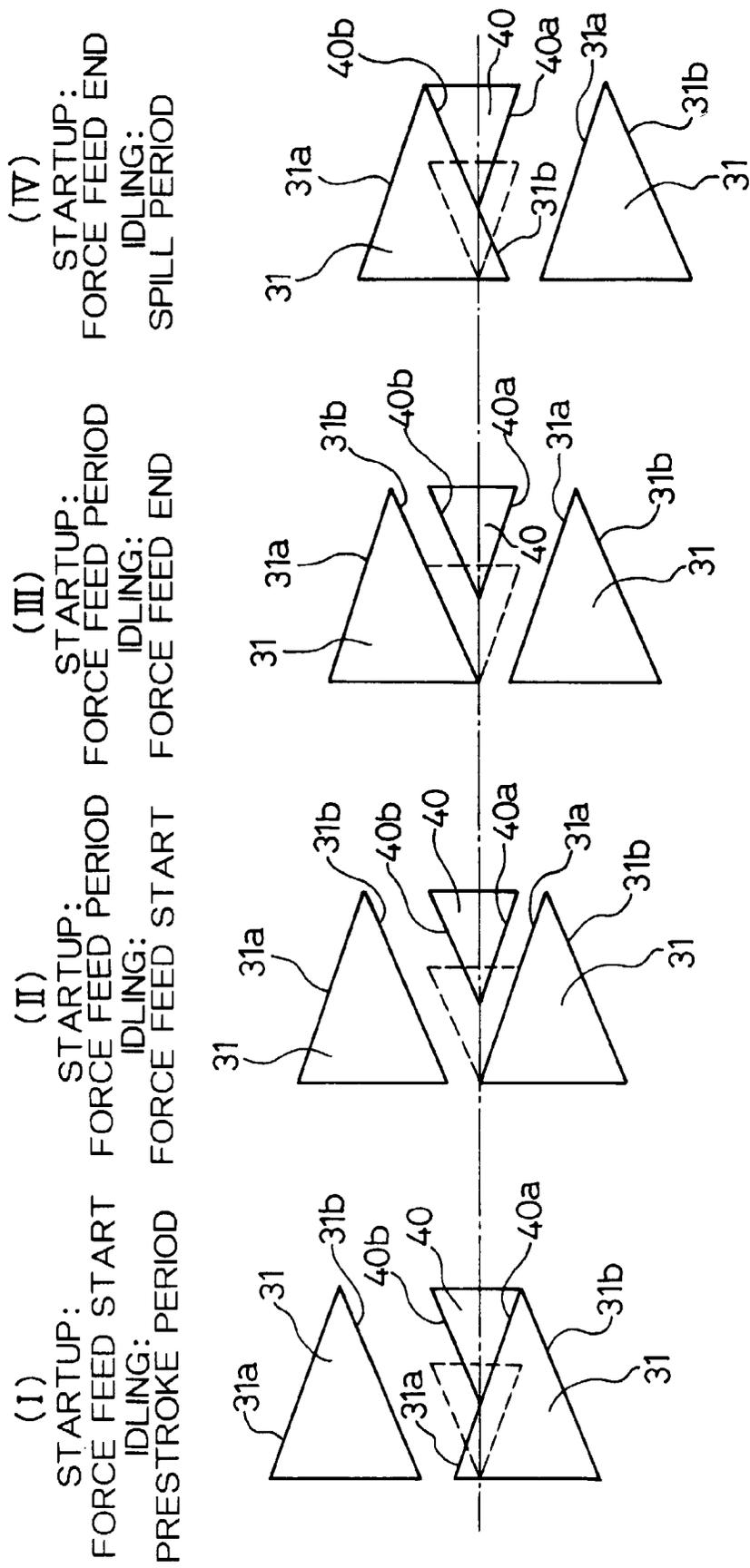


FIG. 5

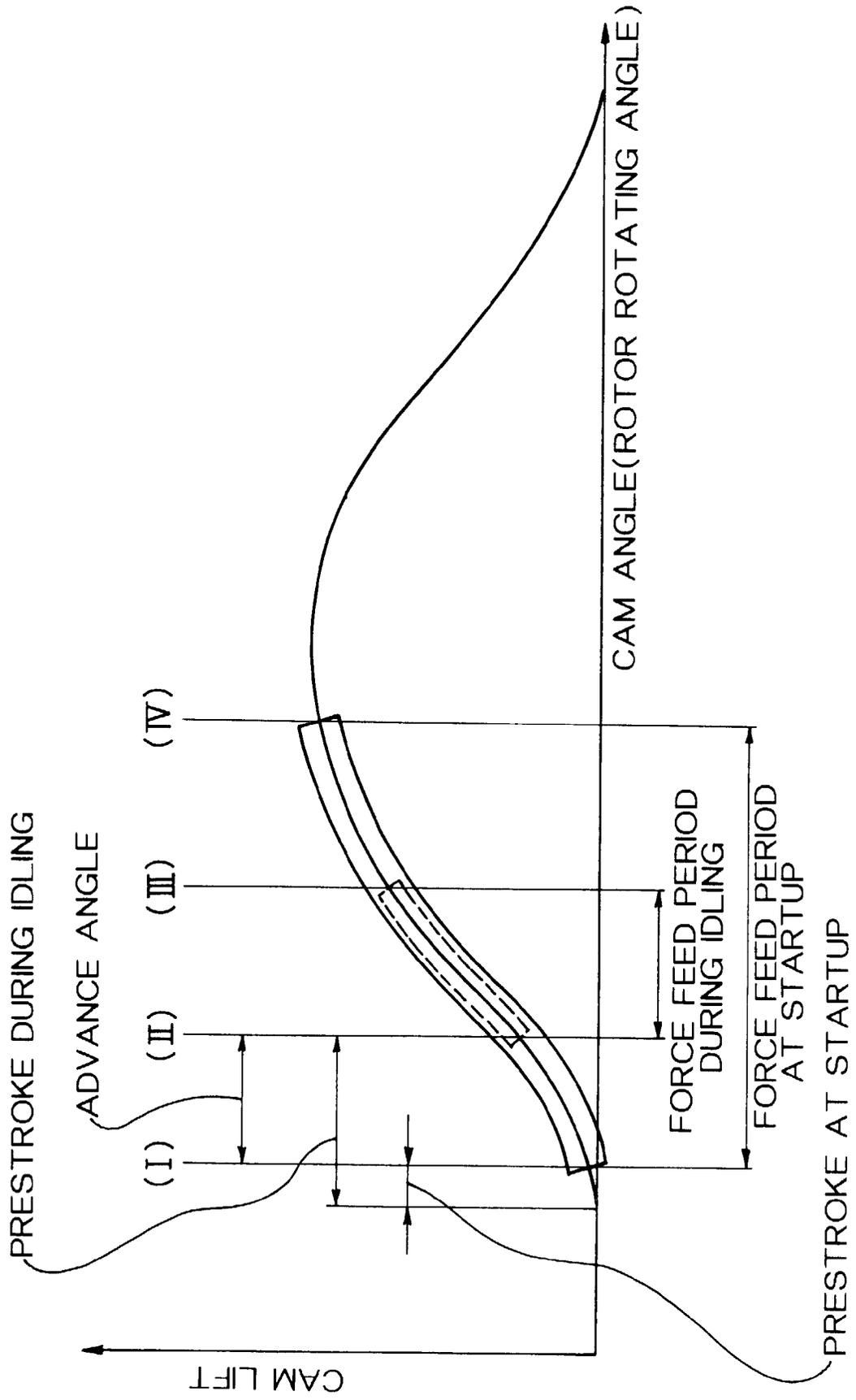


FIG. 6

AT LOW LOAD: $U_{\alpha ist} = 1$
 EFFECTIVE STROKE = A_{mm}

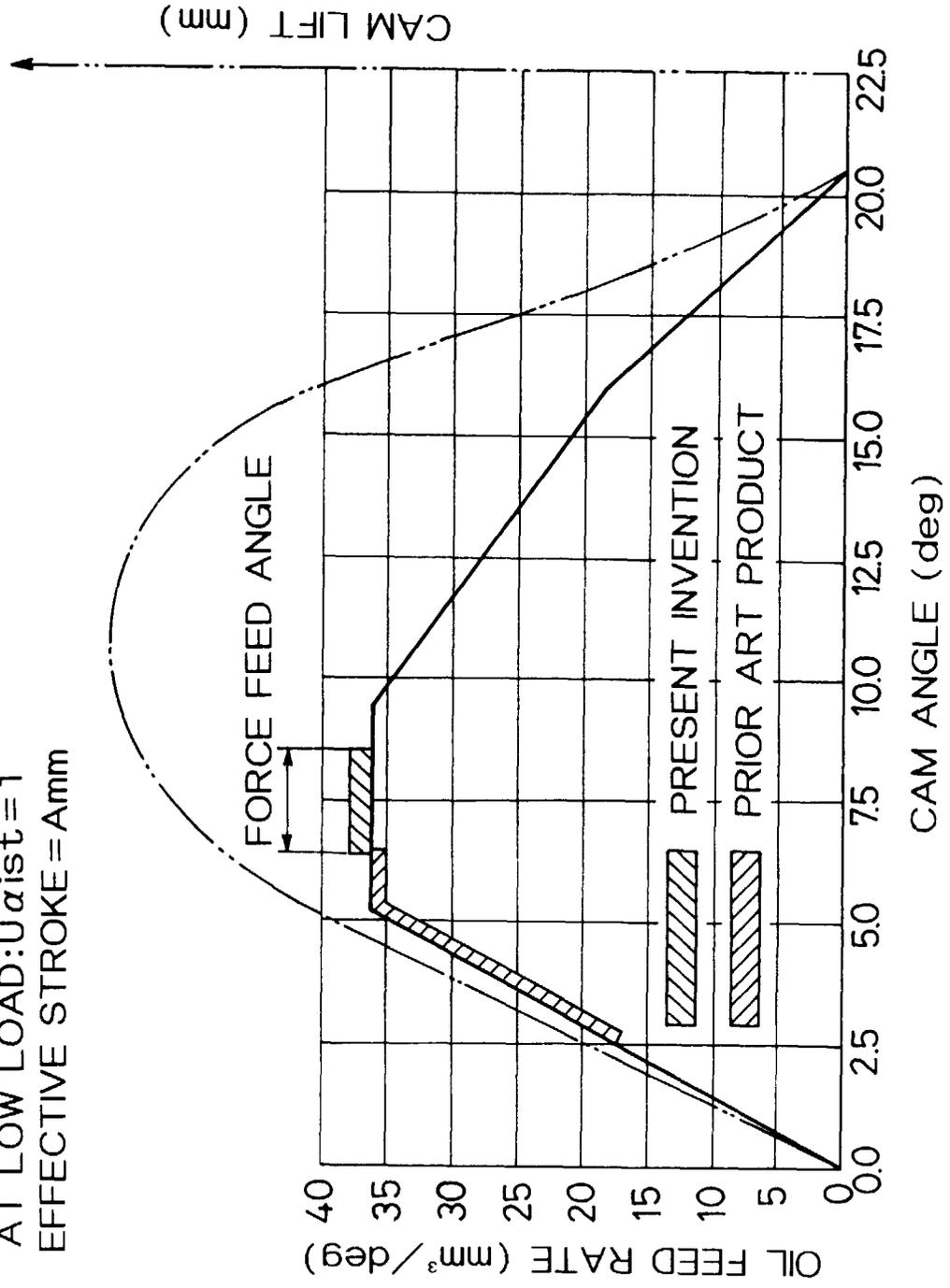


FIG. 7

AT MEDIUM LOAD: $U_{\alpha ist} = 2$
 EFFECTIVE STROKE = 8mm

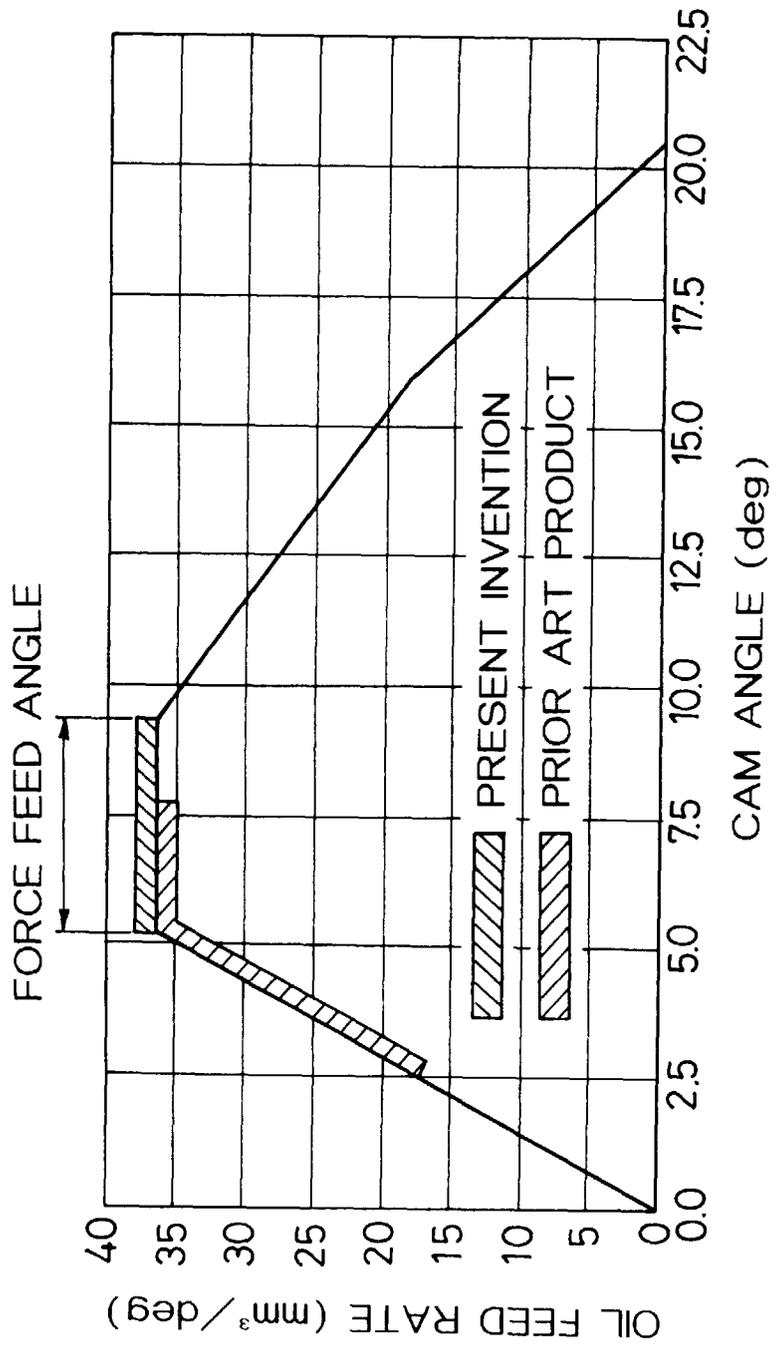


FIG. 8

AT HIGH LOAD: $U_{\alpha ist} = 3$
EFFECTIVE STROKE = 6mm

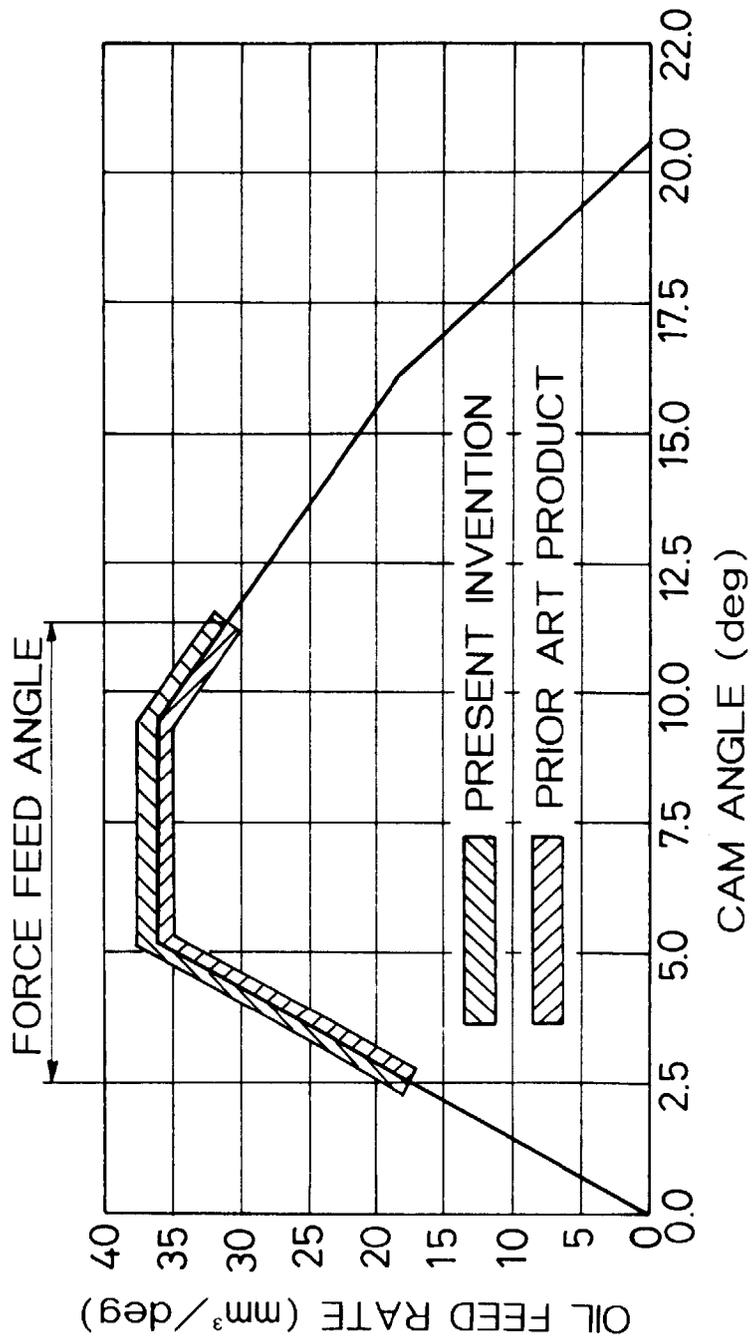


FIG. 9

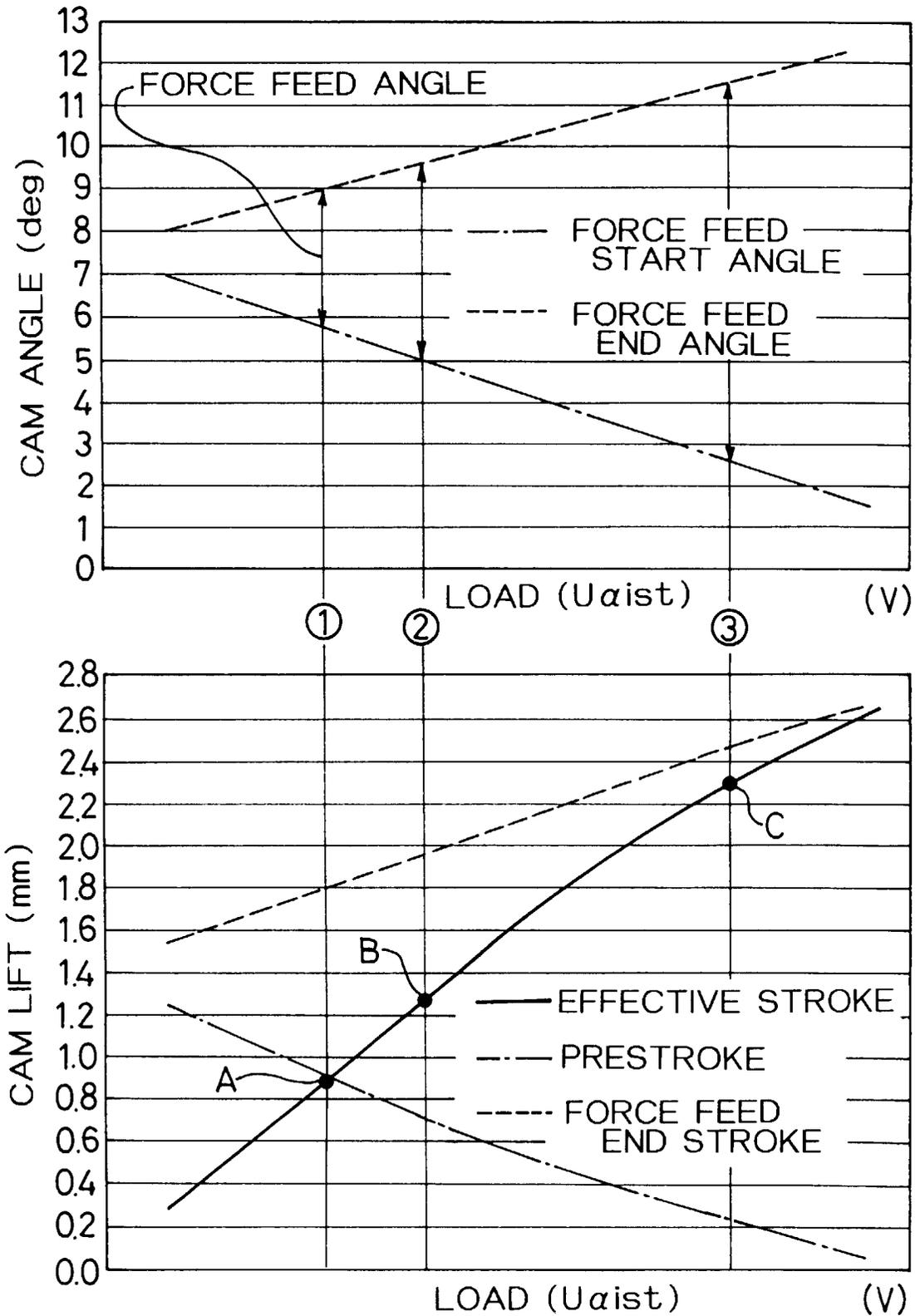
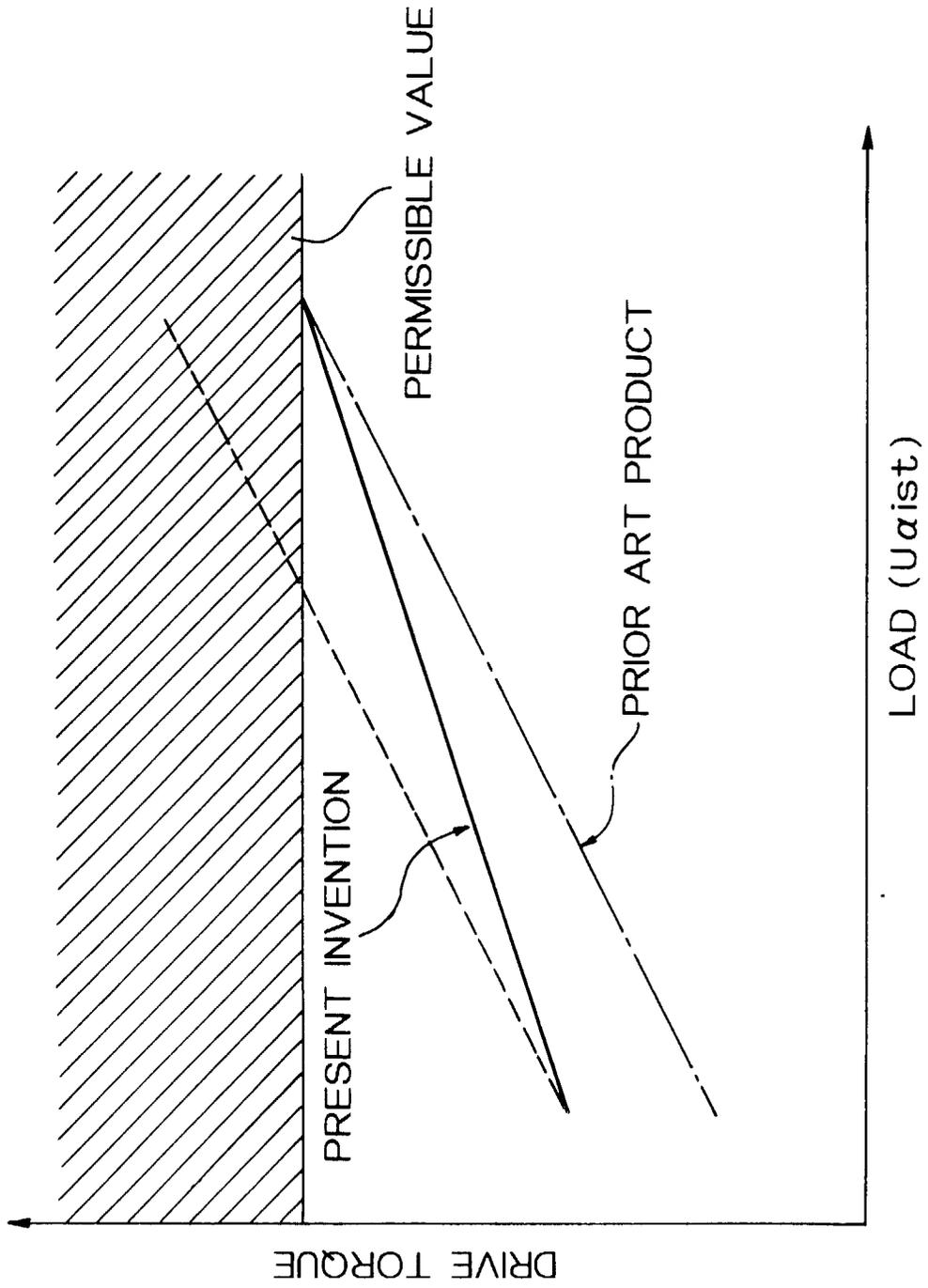


FIG. 10





European Patent Office

EUROPEAN SEARCH REPORT

Application Number
EP 97 30 9751

DOCUMENTS CONSIDERED TO BE RELEVANT		
Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim
A	EP 0 743 446 A (ZEXEL CORP) * column 3, line 49 - column 9, line 22; figures 1-7 * ---	1-10
A	EP 0 692 622 A (ZEXEL CORP) * column 7, line 3 - column 12, line 31; figures 1-5 * -----	1-4
The present search report has been drawn up for all claims		
Place of search		Date of completion of the search
THE HAGUE		2 April 1998
		Examiner
		Hakhverdi, M
CATEGORY OF CITED DOCUMENTS X : particularly relevant if taken alone Y : particularly relevant if combined with another document of the same category A : technological background O : non-written disclosure P : intermediate document		T : theory or principle underlying the invention E : earlier patent document, but published on, or after the filing date D : document cited in the application L : document cited for other reasons & : member of the same patent family, corresponding document
CLASSIFICATION OF THE APPLICATION (Int.CI.6) F02M41/14		
TECHNICAL FIELDS SEARCHED (Int.CI.6) F02M		

EPO FORM 1503 03/82 (P/AC001)