

(19)



Europäisches Patentamt
European Patent Office
Office européen des brevets



(11)

EP 0 863 307 A1

(12)

EUROPEAN PATENT APPLICATION

(43) Date of publication:
09.09.1998 Bulletin 1998/37

(51) Int Cl.⁶: **F02M 41/14, F02M 59/16**

(21) Application number: **98301342.6**

(22) Date of filing: **24.02.1998**

(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

(30) Priority: **08.03.1997 GB 9704835**

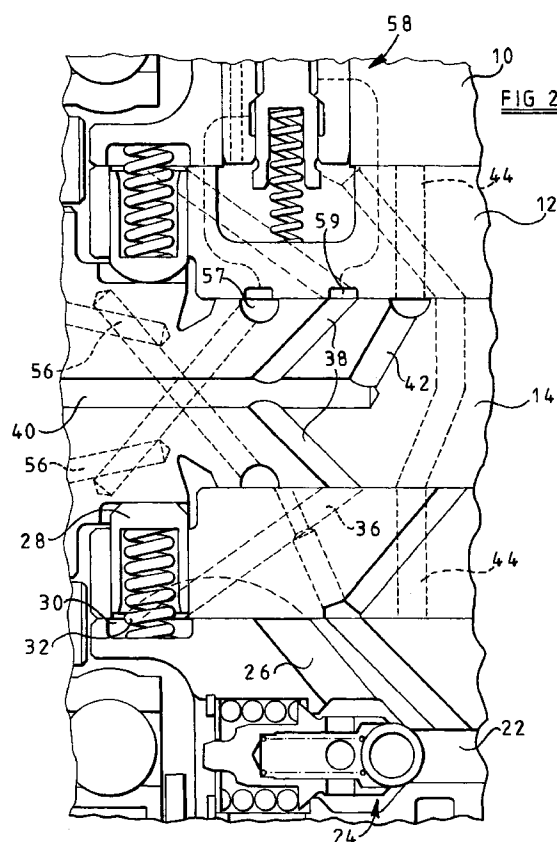
(71) Applicant: **LUCAS INDUSTRIES PUBLIC LIMITED
COMPANY**
Solihull B90 4LA (GB)

(72) Inventors:
• **Collingborn, Peter Alban George**
Gillingham, Kent, ME8 8RX (GB)
• **Thornthwaite, Ian Roy**
Gillingham, Kent, ME7 3PJ (GB)

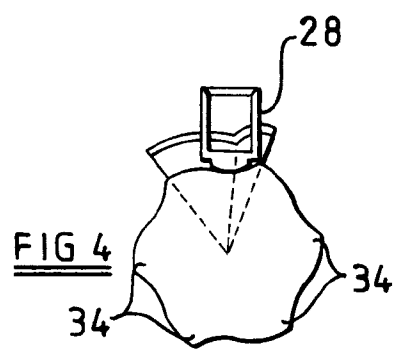
(74) Representative: **Bailey, Richard Alan et al**
Marks & Clerk,
Alpha Tower,
Suffolk Street Queensway
Birmingham B1 1TT (GB)

(54) Fuel pump

(57) A fuel pump comprises a high pressure pump, a low pressure pump (18) and an intermediate pump arrangement arranged to receive fuel from the low pressure pump (18) and to deliver fuel to the high pressure pump, the intermediate pump arrangement operating in a timed relationship with the high pressure pump. In one embodiment, the intermediate pump arrangement includes plungers (28) reciprocable under the influence of a cam surface (34) provided on a distributor member (14) forming part of the high pressure pump.



EP 0 863 307 A1



Description

This invention relates to a fuel pump for supplying fuel at high pressure to the cylinders of a compression ignition internal combustion engine.

In a known fuel pump a rotary vane pump is used to supply fuel at relatively low pressure to a high pressure pump. The high pressure pump is then used to pressurize the fuel and supply the fuel to the cylinders of an associated engine. Difficulties have been experienced in supplying a sufficiently large quantity of fuel to the high pressure pump in the time available, especially for pumps designed for use with engines having more than four cylinders. It is an object of the invention to provide a fuel pump in which this disadvantage is reduced.

According to the present invention there is provided a fuel pump comprising a high pressure pump, a low pressure pump, and an intermediate pump arrangement arranged to receive fuel from the low pressure pump, and supply fuel to the high pressure pump, the intermediate pump arrangement operating in a timed relationship with the high pressure pump.

The intermediate pump arrangement conveniently comprises a plunger reciprocable within a bore, the plunger being moveable under the influence of a cam surface provided upon the distributor member of the high pressure pump.

The invention will further be described, by way of example, with reference to the accompanying drawings, in which:-

Figure 1 is a sectional view of a fuel pump in accordance with an embodiment of the invention;

Figure 2 is an enlarged view of part of Figure 1;

Figure 3 is an enlargement of another part of Figure 1; and

Figure 4 is a view illustrating the shape of part of the distributor member of the pump of Figure 1.

The fuel pump illustrated in Figure 1 is intended for supplying fuel to an engine having six cylinders, the pump comprising a body 10 having a substantially cylindrical sleeve 12 located therein. A distributor member 14 is rotatable within the sleeve 12, the distributor member 14 being keyed to a drive shaft 16 arranged to transmit rotary motion to the distributor member 14 so that the distributor member 14 rotates at a speed related to the operating speed of an associated engine. The rotary part of a rotary vane pump 18 is connected to the distributor member 14, and a fuel pressure regulator 20 is connected between the inlet and outlet of the rotary vane pump 18. Fuel is drawn from a suitable fuel reservoir to the vane pump 18, the fuel being drawn through passages provided in the pump body 10 for the purposes of cooling prior to reaching the inlet of the rotary vane

pump 18. The outlet of the vane pump communicates with a passage 22 provided in the body 10, an inlet non-return valve 24 controlling communication between the passage 22 and a supply passage 26.

The sleeve 12 is provided with six equiangularly spaced, radially extending bores within which hollow plungers 28 of relatively large diameter are reciprocable, the plungers 28 defining, with the bores, low pressure pumping chambers which communicate with one another through an annular groove 30 which is provided in the body 10 aligned with the radially extending bores provided in the sleeve 12, the groove 30 providing communication between each of the low pressure pumping chambers and the supply passage 26. Each plunger 28 has a spring 32 associated therewith to bias the plunger member 28 into engagement with the distributor member 14, the springs 32 assisting the low pressure pump 18 in filling the low pressure pumping chambers. The part of the distributor member 14 with which the plungers 28 engage is shaped to define a series of cam lobes 34 as illustrated in Figure 4. As shown in Figure 4, the ends of the plungers 28 are of part spherical form. As a result, when the plungers ride over the crests of the lobes 34, the radius of curvature of the path followed by each plunger is greater than that of the lobe 34. The rate of change of fuel supply is low, thus the force applied to the lobes 34 is reduced. The reduced load results in a reduction in wear. It will be appreciated that as the distributor member 14 is driven by the drive shaft 16, the movement of the distributor member 14 causes the plungers 28 to reciprocate within their bores, thus changing the volume of the low pressure pumping chambers in time with the operation of the associated engine.

Filling of the low pressure pumping chambers can occur over a relatively long time, thus the rate at which fuel is supplied thereto can be low. It is envisaged that fuel will be supplied to the low pressure pumping chambers over approximately 40° of rotation of the distributor member 14 between each filling cycle of the high pressure pumping chambers, or longer for a pump used to supply fuel to an engine having fewer cylinders.

As indicated hereinbefore, the ends of the plungers are of part spherical form. In order to increase the area of contact between the plungers 28 and the cam surface, the use of cylindrically ended plungers may be preferred. Alternatively, spherical ended plungers may be used in conjunction with a cam profile of concave cross-section.

It will be appreciated that by providing the groove 30 in the body 10 rather than in the sleeve 12, the maximum length of engagement between the sleeve 12 and the plungers 28 is increased, thus the risk of fuel leakage therebetween is reduced. Also, the side load resulting from the cam surface being angled with respect to the axis of the plungers 28 is spread over a relatively large area.

The low pressure pumping chambers further com-

municate with supply ports 36 provided in the sleeve 12, the supply ports 36 being arranged to register, upon rotation of the distributor member 14, with inlet passages 38 provided in the distributor member 14. The inlet passages 38 communicate with an axially extending passage 40 provided in the distributor member 14. The axially extending passage 40 further communicates with a delivery passage 42 which is arranged to register, in turn, upon rotation of the distributor member 14 with six equiangularly spaced delivery ports 44 provided in the sleeve 12, the delivery ports 44 communicating with respective outlets of the pump.

The distributor member 14 is provided with three equiangularly spaced, diametrically extending bores within which high pressure plunger members 46 are reciprocable. The bores communicate with the axially extending passage 40. The outer end of each plunger 46 carries a shoe and roller arrangement, the roller 48 of which is arranged to engage the cam surface of a cam ring 50. The cam ring 50 is angularly adjustable in order to adjust the timing of fuel delivery by the high pressure pump using a conventional advance arrangement 53.

The end of the axially extending passage 40 defines a seating with which a spill valve member 52 is engageable to control communication between the passage 40 and a low pressure volume. The spill valve member 52 is carried by a piston member 54 which is slidable within a cylinder. The piston member 54 and cylinder together define a chamber 55 which is in constant communication through passages 56 with recesses 57. The recesses 57 are registrable, in turn, with the outlet of an electromagnetically controlled spill trigger valve 58, the inlet of which communicates with a groove 59 provided in the sleeve 12 which is registerable with the inlet passages 38, upon rotation of the distributor member 14.

The piston member 54 is provided with drillings 74 which are closed by a plate valve 72 so that once the pressure applied to the piston member 54 exceeds a predetermined pressure, sufficient to move the spill valve member 52 away from its seating, fuel is able to flow through the drillings 74 past the plate valve member 72 to the low pressure drain. In addition, the piston member 54 is provided with recesses 76 which communicate with the drillings 74 and are arranged such that after movement of the piston member 54 beyond a predetermined position, fuel is able to escape at a high rate through the recesses 76 to the low pressure drain. Conveniently, all spilt fuel escapes either through the recesses 76 or past the plate valve 72, virtually no fuel being returned past the spill valve member 52 to the high pressure pumping chamber.

The recesses 76 conveniently take the form of annular grooves provided on the piston member 54 and in the cylinder within which the piston member 54 is slidable, and flats provided at an end part of the cylinder, the parts of the piston member 54 between the flats acting to guide sliding movement of the piston member 54.

Figure 3 illustrates the inlet non-return valve 24 in

detail. As illustrated in Figure 3, the inlet non-return valve 24 comprises a hollow spherical valve member 60 which is engageable with a seating defined around an end of the passage 22. A spring 62 biases the valve member 60 into engagement with its seating, the spring 62 engaging a second valve member 64 which is biased by means of spring 66 into engagement with a frusto-conical seating. The second valve member 64 acts as a guide, guiding movement of the spherical valve member 60 away from its seating. In the illustrated, rest position of the valve 24, the valve member 60 is biased into engagement with its seating, and the second valve member 64 is biased into engagement with its seating. When the pressure within the passage 22 is greater than that within the supply passage 26 by more than a predetermined amount, the spherical valve member 60 is lifted away from its seating against the action of the spring 62 thus permitting fuel to flow to the supply passage 26. Should the pressure within the supply passage 26 become excessively large, fuel is prevented from returning to the passage 22 by the engagement of the spherical valve member 60 with its seating, the fuel pressure acting against the second valve member 64 being sufficient to move the valve member 64 against the action of the spring 66 permitting fuel to flow to an outlet 68, thus relieving the excess pressure within the supply passage 26.

In use, starting from the position in which the plungers 46 occupy their innermost position, the rollers 48 having ridden over the crest of the cam lobes provided on the cam ring 50 rotation of the drive shaft 16 and distributor member 14 causes the inlet passages 38 to register with the supply ports 36. The plungers 26 which are spring biased into engagement with the cam surface of the distributor member 14 occupy positions in which the volume of the low pressure pumping chamber is relatively large, this volume having been charged with fuel by the low pressure feed pump 18 through the inlet non-return valve 24. Continued rotation of the distributor member 14 causes the plungers 28 to be forced radially outward under the influence of the lobes 34 provided on the distributor member 14. The outward movement of the plungers 28 compresses the fuel in the low pressure pumping chambers, displacing the fuel therefrom through the supply port 36 to the inlet passages 38, the fuel being displaced to the through bores provided in the distributor member 14 pushing the plungers 46 radially outwardly. The non-return valve 24 prevents fuel from the low pressure pumping chambers being displaced back to the passage 22.

The volume of the low pressure pumping chambers is selected to be substantially equal to the maximum volume of the high pressure pumping chamber defined by the through bores of the distributor member 14 and the plungers 46, thus when the plungers 28 occupy their radially outer positions, the through bores are charged with fuel and the plungers 46 occupy their outer positions. Figure 1 illustrates the pump in this position.

Should excess fuel be supplied by the plungers 28, the pressure within the supply passage 26 increases by a sufficient amount to cause the second valve member 64 of the inlet non-return valve 24 to open, thus the excess fuel is permitted to escape without significantly increasing the pressure within the low pressure, intermediate pumping chambers.

Continued rotation of the distributor member 14 breaks the communication between the inlet passages 38 and supply ports 36, and subsequently the delivery passage 42 moves into register with one of the delivery ports 44. Also, one of the inlet passages 38 registers with the groove 59 which communicates with the inlet of the solenoid actuated trigger valve 58, and one of the recesses 57 registers with the outlet of the trigger valve 58. Shortly after such a position has been achieved, the rollers 48 move into engagement with the cam lobes provided on the cam ring 50. Such engagement results in the plungers 46 commencing inward movement pressurizing the fuel within the high pressure pumping chamber, and displacing fuel to the delivery port 44 which is registered with the delivery passage 42.

In order to terminate the delivery of fuel, the solenoid actuated trigger valve 58 is opened to permit communication between the inlet passage 38 which registers with the groove 59, and the chamber 55. Such communication results in an increase in the pressure applied to the chamber 55 causing the piston member 54 to move lifting the spill valve member 52 away from the end of the passage 40. Such movement permits fuel to flow from the high pressure pumping chamber to the low pressure drain thus relieving the pressure within the high pressure pumping chamber. Continued inward movement of the plungers 46 continues to displace fuel past the spill valve member 52 to the low pressure drain. Conveniently, substantially all of the pressurized, and hence hot, fuel is displaced past the spill valve in order to minimise heating of the distributor member 14.

After completion of inward movement of the plungers 46, the pressure within the chamber 55 is insufficient to maintain the spill valve member 52 in its lifted position, and the piston 54 moves under the action of a spring to return the spill valve member 52 into engagement with its seating. From this position, continued rotation of the distributor member 14 breaks the communication between the delivery passage 42 and one of the delivery ports 44 and between one of the inlet passages 38 and the groove 59, and subsequently the inlet passages 38 will register with the supply ports 36 ready for another filling cycle to occur.

It is convenient to wipe each of the delivery ports 44 other than the one which communicates with the delivery passage 42 with fuel at transfer pressure. In the arrangement illustrated in Figure 1, this is achieved by means of passages 78 which receive fuel at low, transfer pressure from the low pressure pump 18, and a passage 80 provided in the distributor member 14 registrable, upon rotation of the distributor member, with both the pas-

sages 78 and the delivery ports 44. In addition to wiping the outlet ports 80, the passages 56 are also wiped through a passage 82, thus ensuring that the fuel pressure within the chamber 55 is reduced to an acceptable level prior to commencement of fuel delivery from the high pressure pumping chamber.

The distributor member 14 is of increased axial length in order to house the cam surface with which the plungers 28 engage, and in order to accommodate the passages 56, the passages 56 are each composed of three separate drillings. The central drilling of each passage 56 extends to the outer periphery of the distributor member 14 and is closed by an annular ring 70 which is mounted upon the distributor member 14 in a known manner.

In an alternative embodiment, the inlet non-return valve 24 may be replaced by a simple non-return valve, a separate pressure relief valve being provided. In one arrangement, the bores within which the plungers 28 are reciprocable extend to the exterior of the body 10, and a flexible annular member covers the open ends of the bores. Upon the pressure within the bores exceeding a predetermined level, the member flexes to permit venting of fuel from the bores.

Although the pump described hereinbefore is intended for use with an engine having six cylinders or combustion spaces, it will be appreciated that the invention is suitable for use with engines having fewer or a greater number of cylinders.

Claims

1. A fuel pump comprising a high pressure pump, a low pressure pump (18), and an intermediate pump arrangement arranged to receive fuel from the low pressure pump (18), and supply fuel to the high pressure pump, the intermediate pump arrangement operating in a timed relationship with the high pressure pump.
2. A fuel pump as claimed in Claim 1, wherein the intermediate pump arrangement comprises a plurality of pumping plungers (28) reciprocable under the influence of a cam surface (34).
3. A fuel pump as claimed in Claim 2, wherein the high pressure pump includes a distributor member (14), the cam surface (34) being provided on the distributor member (14).
4. A fuel pump as claimed in Claim 2 or Claim 3, wherein the pumping plungers (28) include part spherical end regions engageable with the cam surface (34).
5. A fuel pump as claimed in any one of the preceding claims, further comprising a non-return valve (24)

located between the low pressure pump (18) and the intermediate pump arrangement to prevent fuel from returning from the intermediate pump arrangement to the low pressure pump (18).

5

6. A fuel pump as claimed in Claim 5, further comprising a pressure relief valve (64, 66) arranged to open when the fuel pressure between the non-return valve (24) and the intermediate pump arrangement exceeds a predetermined level.

10

15

20

25

30

35

40

45

50

55

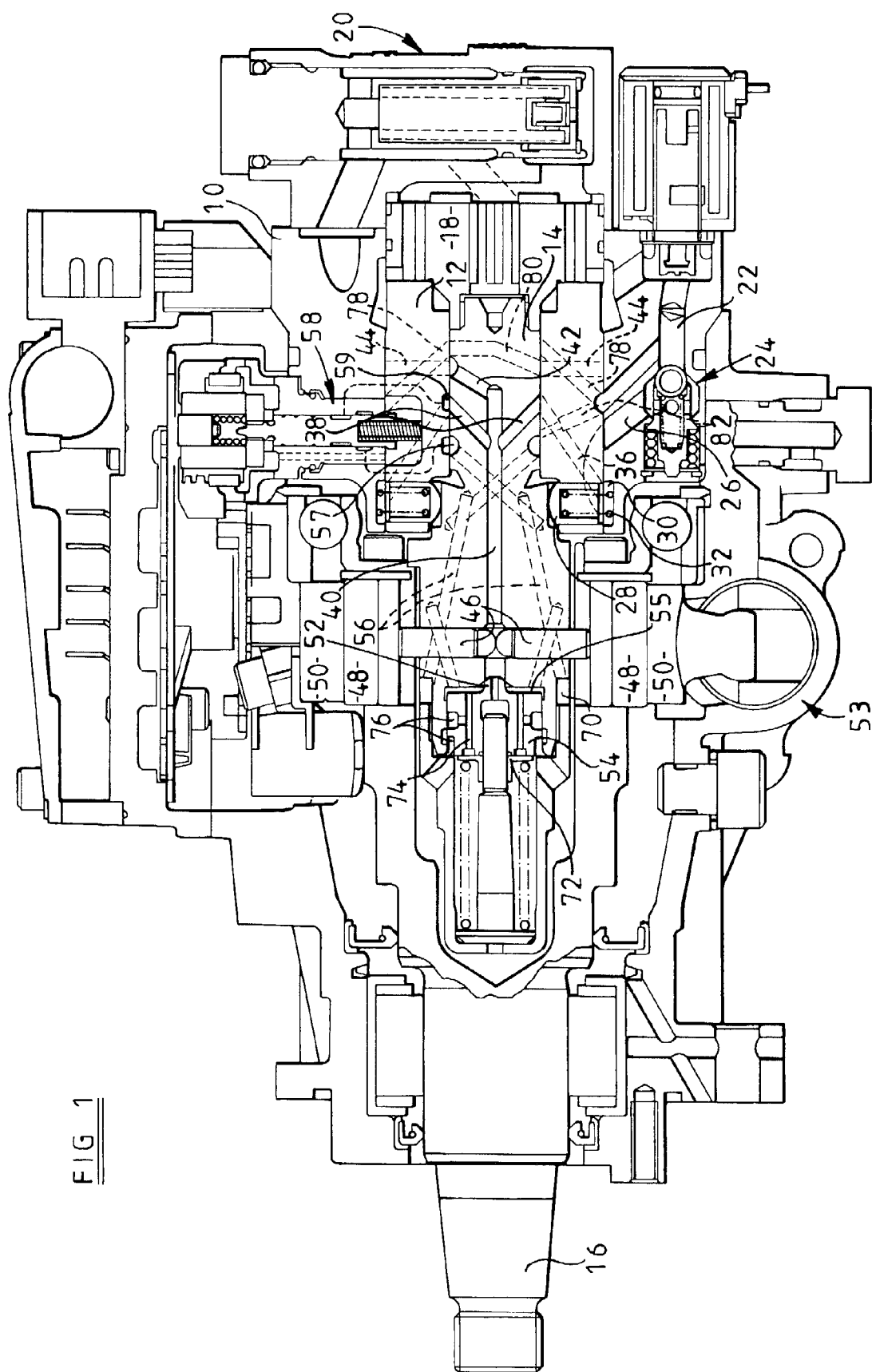


FIG 1

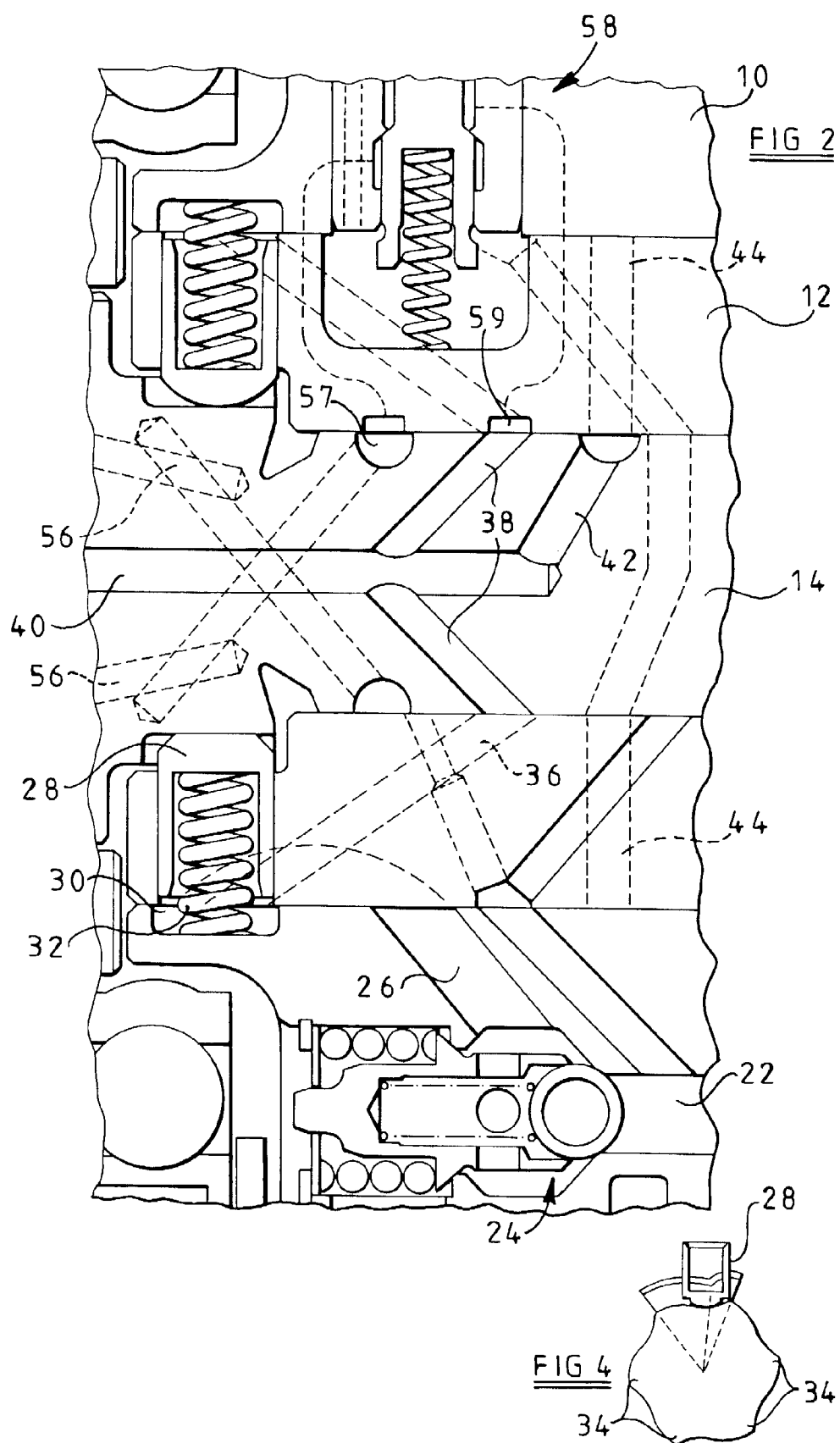
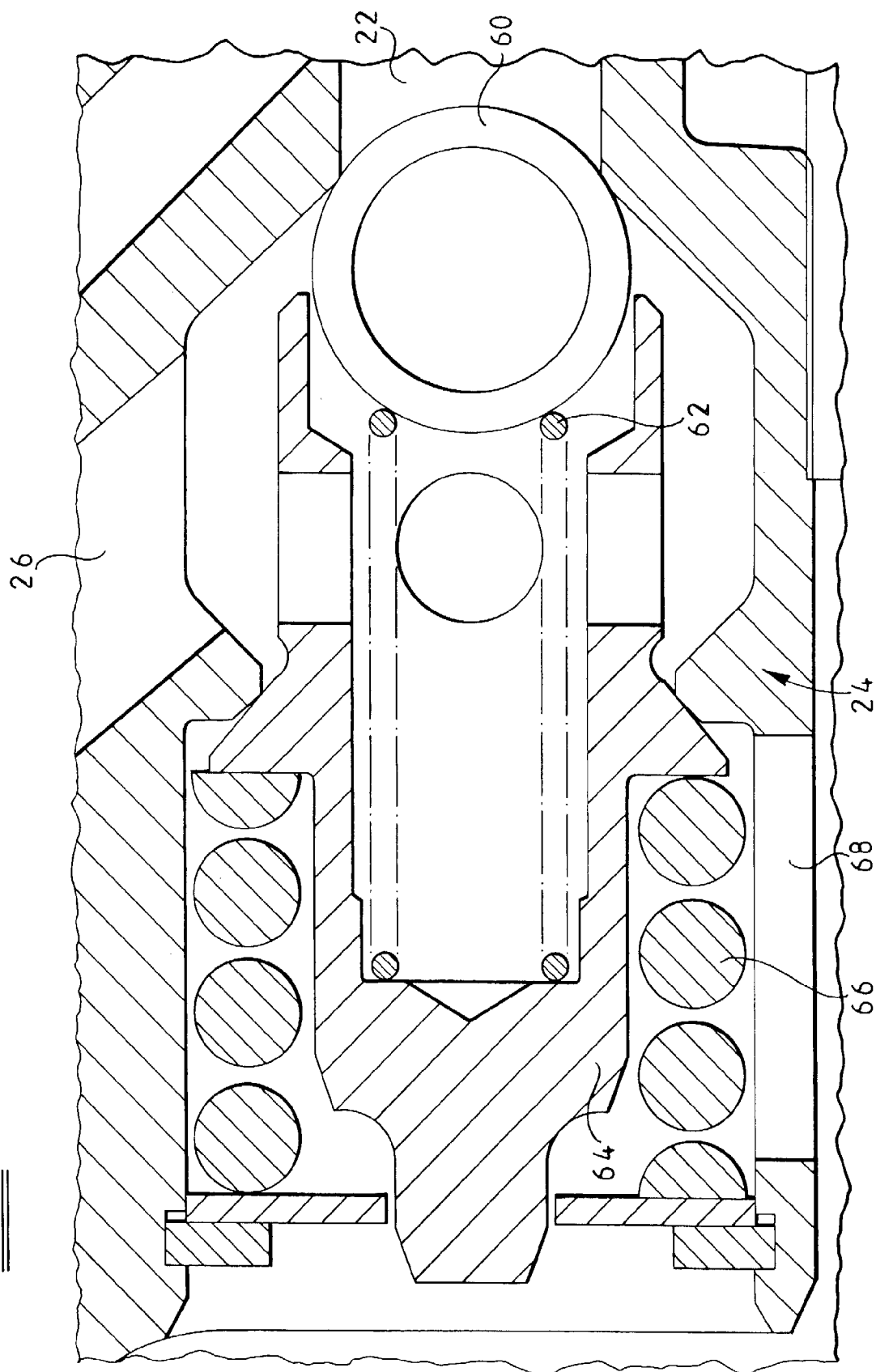


FIG 3





European Patent
Office

EUROPEAN SEARCH REPORT

Application Number
EP 98 30 1342

DOCUMENTS CONSIDERED TO BE RELEVANT			
Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (Int.Cl.6)
X	GB 2 173 923 A (RICARDO CONSULTING ENG) 22 October 1986 * page 3, line 113 - page 4, line 33; figure 1 *	1	F02M41/14 F02M59/16
A	US 4 530 324 A (TANAKA TOSHIAKI ET AL) 23 July 1985 * abstract; figure 1 *	2	
A	EP 0 611 117 A (LUCAS IND PLC) 17 August 1994		
			TECHNICAL FIELDS SEARCHED (Int.Cl.6)
			F02M
The present search report has been drawn up for all claims			
Place of search THE HAGUE		Date of completion of the search 28 May 1998	Examiner Sideris, 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			

EPO FORM 1503 03/82 (P04C01)