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## (54) Variable-delivery pump

(57) A variable-delivery pump (1) has a first compression stage (18) and at least one second compression stage (23) with respective inlet passages (21) (28) communicating with an inlet (6) of the pump (1), in use, and respective exhaust openings (22) (29) for communicating with a delivery outlet (7) of the pump (1) itself; the pump (1) further being provided with a valved selector unit (30) interposed between the inlet passages (21) (28) and the exhaust passages (22) (29) which can be actuated to connect the passages (21) (22) (28) (29) together and to select different operative conditions for the stages (18) (23) according to the pressure of the fluid directed to the delivery outlet (7).

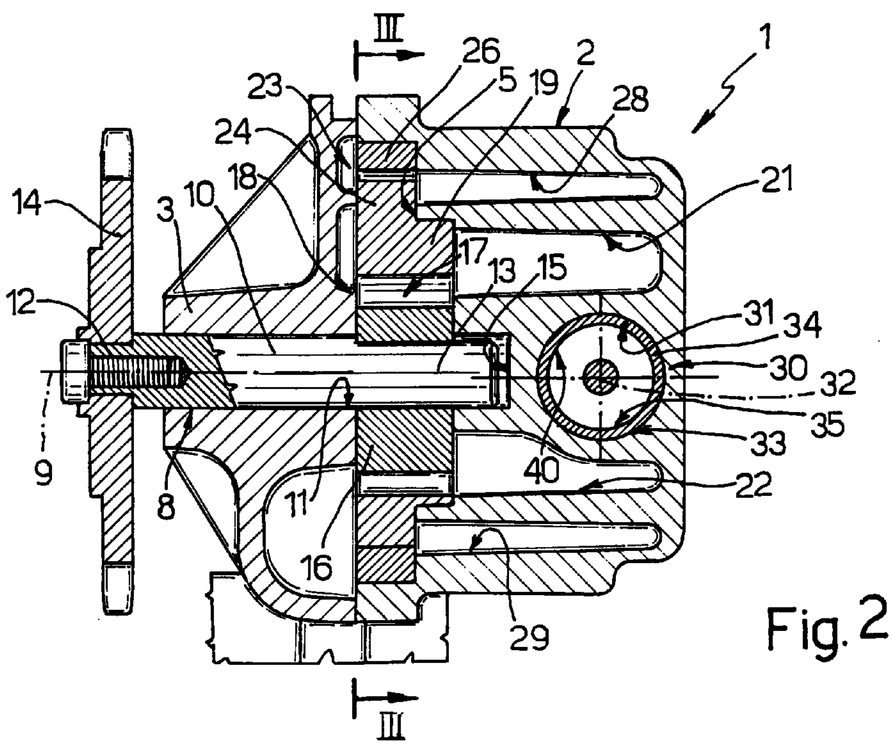


Fig. 2

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## Description

The present invention relates to a variable-delivery pump.

More particularly, the present invention relates to a fixed-displacement pump with a delivery which is variable in discrete increments.

In the field of fluid supply in general, and of engine fluids or lubricants in particular, the need is felt for the provision of large fluid flows under low running conditions, or low rates of rotation, while at the same time flow rates which are as small and as constant as possible need to be provided under high running conditions, or at high rates of rotation.

For this purpose pumps are currently used which have different constructions and which operate differently from each other and, in particular, fixed-displacement and variable-delivery rotary pumps are used to which the description below will make explicit reference without thereby losing the generality of the invention.

It is known that, in pumps of the first type specified, the flow rate and the supply pressure increase proportionally as the rate of rotation of the rotary members increases and the displacement is selected during the design stage so as to ensure that the flow rate required at low running conditions is supplied.

In order to limit the fluid flow supplied under high running conditions, the pump includes a by-pass valve system which is associated with the delivery outlet of the pump and is controlled so that, under low running conditions, it is inactive and the entire flow supplied by the pump is delivered to the utiliser while, when the rate of rotation exceeds a predetermined threshold, it is activated and causes part or, at the limit, the entire flow supplied by the pump to be discharged.

Even though the known pumps of the type described above are used, they suffer from various disadvantages all resulting from the fact that they do not perform with consistent efficiency and, in particular, they perform relatively inefficiently particularly at high rates of rotation when part of the flow supplied by the pump is not used by the utiliser but is discharged by the by-pass valve system.

Furthermore, due to the fact that high fluid flow rates must be ensured at low rates of rotation, the known pumps are usually very bulky and difficult to locate, not only in existing systems but, in particular, in newly designed systems.

The object of the present invention is to provide a variable-delivery pump which enables the problems explained above to be overcome in a simple manner and, in particular, allows a large flow to be provided under low running conditions while performing efficiently over its entire operating range and, at the same time, having a relatively small bulk.

According to the present invention there is provided a variable-delivery pump including an inlet and a delivery outlet and characterised in that it further includes a first compression stage and at least one second compression stage having respective inlet passages for communicating with the inlet and respective exhaust passages for communicating with the delivery outlet; selector means being associated with the inlet passages and exhaust passages for selecting different operative conditions for the two compression stages as the pressure of the fluid directed to the delivery outlet varies.

Preferably, in the pump defined above, the selector means comprise first valve means associated with the exhaust passages of the first and second stages and first switch means for switching the first valve means between a connecting position, in which the exhaust passages are connected together and an isolating position in which the exhaust passages are isolated from each other.

Conveniently the selector means further include second valve means associated with the first stage and second switch means for switching the second valve means between a connecting position in which they interconnect the inlet and exhaust passages of the first compression stage and an isolating position in which these inlet and exhaust passages are isolated from each other.

The invention will now be described with reference to the appended drawings, which illustrate a non-limitative embodiment thereof, in which:

Figure 1 is a front elevational view of a preferred embodiment of a variable-delivery pump according to the present invention;
Figure 2 is a section taken on the line II-II of Figure 1;
Figure 3 is a section taken on the line III-III of Figure 2;
Figures 4 to 7 are sections of a central detail of Figures 1 and 3 in four different operating positions; and
Figure 8 is a graph showing characteristic curves of the pump of Figure 1.

In Figures 1 to 3, a variable-delivery, fixed-displacement rotary pump is indicated 1 which includes a rear body 2 for connection in known manner to a fixed framework (not illustrated) and a flanged front cover 3 fixed to the body 2 by a plurality of screws 4 (Figure 1) and, together with the body 2, defining a chamber 5.

The pump 1 has an inlet 6 and a delivery outlet 7 formed in the body 2 and includes a motor-driven input shaft 8 defining its own axis 9. The shaft 8 in turn comprises an intermediate portion 10 which is fixed axially but rotatably engaged in a through-hole 11 in the cover 3 and two end portions, indicated 12 and 13, of which the portion 12 projects from the cover 3 and has keyed thereto a gear 14 of a drive-transmission, not illustrated, for driving the shaft 8. The portion 13, on the other hand, extends through the chamber 5 to terminate in a seat 15 in the body 2 formed co-axially with the axis 9 and has keyed thereto a gear 16 with external teeth 17 (Figures 2 and 3).

The gear 16 constitutes the inner gear of a first geared compression stage 18 which also includes an outer gear defined by a second ring gear 19 with internal teeth 20 (Figure 2) in mesh with the teeth 17 and having one more tooth than the teeth 17. This stage 18 communicates, in use, with the inlet aperture 6 through its own intake passage 21 (Figures 2 and 3), which is substantially semi-cylindrical, and with the delivery outlet 7 of the pump 1 through its own exhaust passage 22 (Figures 2 and 3), also substantially semi-cylindrical, and has a linear characteristic represented by the straight line A in Figure 8.

The chamber 5 further houses a second geared compression stage 23 surrounding the first stage 18 and comprising a gear 24 integral with an outer peripheral portion of the gear 19 and having external teeth 25. The gear 24 constitutes the inner gear of the stage 23 which further includes an outer gear constituted by a ring gear 26. The ring gear 26 is coupled to the body 2 so as to be rotatable about an axis coincident with the axis 9 and has internal teeth 27 in mesh with the teeth 25 of the gear 24, with one more tooth than the teeth 25. The second stage 23 communicates permanently with the inlet 6 through its own inlet passage 28 (Figures 2 and 3) which is substantially semi-cylindrical and formed in the body 2 and partly in the cap 3 outside the passage 21. The second stage 23 also communicates with the delivery outlet 7 of the pump 1 through an exhaust passage 29 (Figures 2 and 3) also formed in the body 2 outside the passage 22 and has a characteristic which, in this particular case, coincides with the characteristic of the stage 18.

As illustrated in Figures 4 to 7, the pump 1 also includes a valved selector unit 30 associated with the passages 21, 22, 28 and 29 of the compression stages 18 and 23 for selecting different operative conditions for the compression stages 18 and 23 as the rate of rotation of the shaft 8 varies, and hence the pressure of the fluid supplied by the stages 18 and 23 to the delivery outlet 7 varies.

More particularly, at least part of the unit 30 is housed within a cylindrical blind seat 31 (Figure 2) formed in the body 2 between the passages 21, 22, 23 and 24 with its axis 32 perpendicular to the axis 9 and which houses a tubular cylindrical body 33 which extends co-axially with the axis 32 and in turn comprises a peripheral wall 34 bounded internally by a cylindrical surface 35. The body 33 has two portions 36 and 37 of which the portion 36 engages the seat 31 positively and is closed by an end wall 38 which bears against an end face of the seat 31 while the portion 37 projects from the body 2 (Figures 1, 2 and 4) and is closed by a cup-shaped cap 39 inserted in the portion 37 itself with its concavity facing into the body 33.

With reference to Figures 4 to 7, the body 33 defines, together with the wall 38 and the cap 39, a cylindrical chamber 40 which communicates with the inlet passages 21 and 29 of the stages 18 and 23 through two respective apertures 41 and 42 in the wall 34 of the portion 36 in adjacent positions and with the exhaust passages 22 and 29 of the stages 18 and 23 through two apertures 43 and 44 also formed in the wall 34 of the portion 36 in adjacent positions. The chamber 40 also communicates with the exhaust passage 29 of the second stage 23 through a further aperture 45, again formed in the wall 34 of the portion 36 adjacent the aperture 43 but alongside the wall 38.

The unit 30 further includes a slide 46 housed in the chamber 40 and slidable within the chamber 40 itself along the axis 32 to open and close the apertures 41, 42, 43, 44 and 45 and select different operating conditions for the stages 18 and 32.

More particularly, as illustrated in Figures 4 to 7, the slide 46, includes, starting from its end facing the wall 38, a first piston head 47 which is coupled with the surface 35 with a fluid-tight seal and has a length in a direction parallel to the axis 32 which is less than the maximum length of the aperture 42 measured in the same direction. The piston 47, together with the wall 38 and an end portion of the wall 34, defines a variable-volume chamber 48 which communicates permanently with the exhaust passage 29 of the stage 23 through the aperture 45 and is arranged to communicate, in use, with the inlet passage 28 through the aperture 42. The piston head 47 and the chamber 48 define the movable member and the internal chamber respectively of a linear actuator 49 for driving the slide 46.

The piston head 47 is connected by a rod 50 to an end wall 51 of a second piston head 52 which also includes a cylindrical peripheral wall 53 which is slidable in fluid-tight contact with the surface 35 and which extends from the wall 51 towards the piston head 47 surrounding an end portion of the rod 50.

The wall 51 further defines the end wall of a third piston 54 having the same dimensions as the piston 52 and the cylindrical peripheral wall 55 of which extends from the wall 51 itself in the opposite direction from the cylindrical wall 53 and towards the cap 39. The cylindrical walls 53 and 55 have respective lengths, measured in a direction parallel to the axis 32, the sum of which is less than the maximum length of the aperture 43 measured in the same direction.

Still with reference to Figures 4 to 7, the wall 51 is rigidly connected by a rod 56 to an end wall 57 of a further cup-shaped piston head 58 which also includes a cylindrical peripheral wall 59 sealingly cooperating with the surface 35 and extending from the wall 57 towards the cap 39. The piston head 58 is releasably connected to the rod 56 by a screw connection and is adapted to be located, in use, against an axial shoulder defined by a resilient split ring 60 housed in a seat formed in the surface 35 between the aperture 44 and the cap 39.

The piston head 58, together with the cap 39 and the portion 37, defines a chamber 61 which houses a helical spring 62 compressed between the piston 58 and the cap 39 so as to exert a force, in use, opposing the force exerted by the linear actuator 49 in use.

The operation of the pump 1 will now be described with reference to Figures 4 to 8 and starting from a condition in which the slide 46 of the valve unit 30 is located in a first operative position, or initial operative position, illustrated in Figure 4, in which the piston head 58 is kept in abutment with the split ring 60 by the action of the spring 62, the piston heads 47 and 52 separate the inlet passages 21 and 28 from each other and the exhaust passages 21 and 28 from the respective exhaust passages 22 and 29, and the piston heads 54 and 58 together define a passage connecting the exhaust passage 22 of the stage 18 to the exhaust passage 29 of the stage 23.

Starting from this condition, immediately the shaft 8 is rotated by the transmission (not illustrated), the stage 23 starts to deliver fluid to the delivery outlet 7, causing an immediate rise in the pressure both in the exhaust passage 29 and in the chamber 48. As a result of the increase in pressure in the chamber 48, a force is exerted on the piston 47 which overcomes the action of the spring 62 and moves the slide 46 progressively towards the cap 39. As a result of the movement of the slide 46, the piston head 52 opens the aperture 41 progressively and puts the inlet passages 21 and 28 of the two stages 18 and 23 into communication, as illustrated in Figure 5.

At this point the stages 18 and 23, connected in parallel, supply respective flows which, in this particular case, are identical and which vary as the rate of rotation of the shaft 8 varies in accordance with a law represented by the straight line A in Figure 8. In this operative condition, the pump 1 thus supplies a flow through the delivery outlet 7 which, for any value of the rate of rotation, is exactly equal to the sum of the flows from the individual stages, as shown by the line OB in Figure 8.

This operative state lasts until the shaft 8 reaches a given rate of rotation n1 (Figure 8) and the pressure of the fluid in the exhaust passage 29, and hence in the chamber 48, and which passes through the delivery outlet 7, exceeds the threshold value p1 (Figure 8).

Immediately the pressure of the fluid supplied through the exhaust passage 29 exceeds the value p1, the slide 46 is moved further towards the cap 39. As a result of this movement, the piston head 52 progressively opens the passage 22 while the piston head 54 progressively prevents the movement of fluid away from and towards the aperture 44 by closing that portion of the aperture 43 adjacent the aperture 44 itself. Hence, during the movement of the slide 46, the exhaust passage 22 of the stage 18 is put progressively in communication with the inlet passage 21 of this same stage 18 and is simultaneously isolated from the exhaust passage 29 of the stage 23 whereby, while the stage 23 continues to deliver fluid to the delivery outlet 7, further increasing the pressure in the delivery outlet 7 itself, the stage 18 is progressively isolated and short-circuited and hence the delivery supplied by the stage 18 is kept first constant, up to a rate of rotation n2 (Figure 8), and then decreases progressively, to reach zero when the shaft 8 reaches the rate of rotation n3 and the slide 46 has reached an isolating operative position illustrated in Figure 6.

During the movement of the slide 46 to the said isolating position, the flow supplied by the pump 1 through the delivery outlet 7 varies progressively as shown by the lines BC and CD in Figure 8, which, as a result of the progressive reduction in the flow supplied by the stage 18, are obviously less inclined than the line OB.

At this point, as the rate of rotation increases yet more, only the stage 23 remains active and delivers its own variable flow to the delivery outlet 7 which flow varies with the rate of rotation in accordance with the portion DE of its characteristic curve A while the slide 46 continues to move towards the cap 39 as a result of the force exerted by pressure in the chamber 48.

This operative condition continues until a rate of rotation n4 is reached (Figure 8) and the piston 47 at this point puts the inlet passage 28 of the stage 23 in communication with the exhaust passage 29 of this same stage 23 through the apertures 42 and 45 and the chamber 48, as illustrated in Figure 7. In this way, as happened for the stage 18, the stage 23 is also progressively isolated or short-circuited and thus, notwithstanding the fact that the shaft 8 continues to rotate at a sustained rate, the flow supplied by the pump 1 through the outlet 7 first remains constant and then reduces progressively.

From the above it will be apparent that the characteristics of the pump 1 described and, in particular, the presence of the two compression stages 18 and 23 and a valve unit 30 for selecting different operative conditions for the stages 18 and 23 themselves, in comparison with known pumps first of all enables extremely high flow rates to be supplied even at low rates of rotation by the simple connection of the stages 18 and 23 in parallel, and then enables reduced flow rates to be supplied at high rates of rotation by the inhibition or isolation of one of the stages, as demonstrated by the characteristic of the pump 1 shown in Figure 8 by the chain line.

Moreover the pump 1 differs from known pumps in that it performs efficiently and substantially consistently whatever the rate of rotation of the shaft 8. In fact, while in known fixed-delivery pumps the excess flow supplied by the pumps at high rates of rotation is discharged in such a way that a considerable quantity of the energy used in compressing the fluid is wasted, in the pump 1 described, the reduction in the flow rate at high rates of rotation is achieved simply by the inhibition or short-circuiting of one of the two stages, and, in particular, the stage 18, that is by the interconnection or short-circuiting of the inlet and exhaust passages of this stage 18, the stage 23 being left to function normally. Thus, not only is the flow of fluid to the delivery outlet 7 reduced, but this reduction occurs without a substantial waste of energy.

Moreover, by means of the valve unit 30, it is possible to short-circuit, in addition to a predetermined delivery pressure, the stage 23 as well and hence to vary or regulate the flow supplied through the delivery outlet 7 even when only one of the two stages is operating. It is thus clear that, in these conditions, while an effect similar to that achieved by known pumps with the aid of the by-pass system is achieved, the efficiency of the pump 1 continues to be acceptable, no additional energy being expended in compressing some of the fluid which is immediately exhausted without serving the utiliser.

From the above it is also clear that the arrangement of the stages 18 and 23 within each other, as well as the arrangement and compactness of the valve unit 30, enables a pump to be obtained which is of extremely small bulk and can thus easily be adapted for any installation.

Finally, from the above it is clear that variations and modifications may be made to the pump 1 described above which do not fall outside the field of protection of the present invention. In particular, the two geared stages 18 and 23 may be replaced by two or more compression stages of a different type and are not necessarily the same as each other with regard, for example, to their displacements.

Moreover, the valve unit 30 may be replaced by a valve unit having different characteristics and, in particular, including, for example, several valves which are not necessarily slide valves located separately from each other and switched by one or more actuators which need not necessarily be pneumatic. In particular, the valve unit 30 could be arranged so that the inlet passages 21 and 28 are permanently connected to the inlet 6.

## Claims

1. A variable-delivery pump (1) including an inlet (6) and a delivery outlet (7) and characterised in that it further includes a first compression stage (18) and at least one second compression stage (23) having respective inlet passages (21) (28) for communicating with the inlet (6) and respective exhaust passages (22) (29) for communicating with the delivery outlet (7); selector means (30) being associated with the inlet passages and exhaust passages (21) (22) (28) (29) for selecting different operative conditions for the two compression stages (18) (23) as the pressure of the fluid directed to the delivery outlet (7) varies.

2. A pump according to Claim 1, characterised in that the selector means (30) comprise first valve means (34, 35, 58) associated with the exhaust passages (22) (29) of the first stage (18) and the second stage (23) and first switch means (49, 62) for switching the first valve means (34, 54, 58) between a connecting position in which the exhaust passages (22) (29) are interconnected and an isolating position in which the exhaust passages (22) (29) are isolated from each other.

3. A pump according to Claim 1 or Claim 2, characterised in that the selector means (30) further include second valve means (34, 47, 52) associated with the first stage (18) and second switch means (49, 62) for switching the second valve means (34, 47, 52) between a connecting position in which the inlet and exhaust passages (21 and 22) of the first compression stage (18) are connected together and an isolating position in which these inlet and exhaust passages (21 and 22) are isolated from each other.

4. A pump according to any one of the preceding Claims, characterised in that the selector means (30) further include third valve means (34, 47, 48) associated with the second stage (23) and third switch means (49, 62) for switching the third valve means (34, 47, 48) between a connecting position in which the inlet and exhaust passages (28 and 29) of the second compression stage (28) are connected together and an isolating position in which these inlet and exhaust passages (28 and 29) are isolated from each other.

5. A pump according to any one of the preceding Claims, characterised in that the selector means (30) include fourth valve means (34, 52) for connecting the inlet passages (21) (28) in succession with the inlet (6).

6. A pump according to Claims 2 to 5, characterised in that the first (34, 54, 58), second (34, 47, 52), third (34, 47, 48) and fourth (34, 52) valve means are aligned with each other on a single axis (32).

7. A pump according to Claim 6, characterised in that the switch means (49, 62) comprise a single, common linear actuator (49).

8. A pump according to Claim 7, characterised in that the linear actuator (49) includes a variable-volume chamber (48) and in that connection means (29, 45) are provided for connecting the variable-volume chamber (48) to the delivery outlet (7).

9. A pump according to Claim 7 or Claim 8, characterised in that the switch means (49, 62) further include resilient biasing means (62) for exerting a force opposing that exerted by the linear actuator (49), in use.

10. A pump according to Claim 9, characterised in that it includes a cylindrical chamber (40) extending coaxially with the axis (32) and communicating with the inlet (21) (28) and exhaust passages (22) (29) of the first (18) and second (23) compression stages through respective apertures (41) (42) (43)

(44) (45); the first (34, 54, 58), second (34, 47, 52), third (34, 47, 48) and fourth valve means comprising respective pistons (47) (52) (54) (58) sealingly slidable in the chamber (35) to open and close the respective apertures (41) (42) (43) (44) (45); rigid connecting means (50) (56) being provided for connecting the pistons (47) (52) (54) (58) rigidly together.

**11.** A pump according to Claim 10, characterised in that the resilient means (62) are housed in the cylindrical chamber (40) and the linear actuator (49) includes a piston (47) slidable in the cylindrical chamber (35).

**12.** A pump according to any one of the preceding Claims, characterised in that the first compression stage (18) extends completely within the second compression stage (23).

**13.** A pump according to Claim 12, characterised in that the first compression stage (18) and the second compression stage (23) are geared stages and each includes a respective external gear (16) (24) and a respective external gear (19) (26) in mesh with the respective internal gear (16) (24).

**14.** A pump according to Claim 13, characterised in that it includes an input drive shaft (8) and in that only one of the internal gears (16) (24) is keyed directly to the input drive shaft (8).

**15.** A pump according to Claim 14, characterised in that the internal gear (16) of the first stage (18) is keyed to the input drive shaft (8) and the internal gear (24) of the second stage (23) is rigidly fixed to the external gear (19) of the first compression stage (18).

**16.** A pump according to any one of Claims 13 to 15, characterised in that the outer gear (19) (26) of each compression stage (18) (23) includes respective internal teeth (20) (27) with one tooth more than the external teeth (17) (25) of the respective internal gear (16) (24).

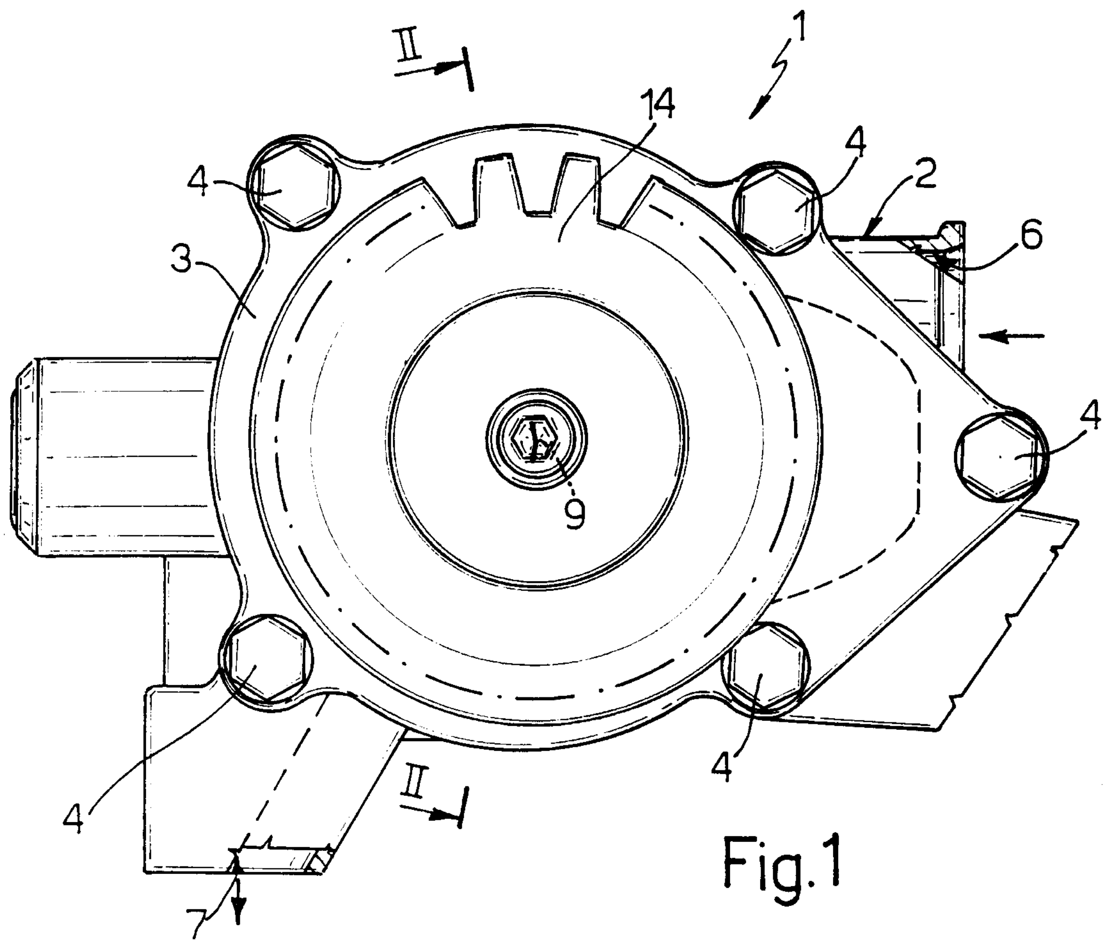
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Fig.1

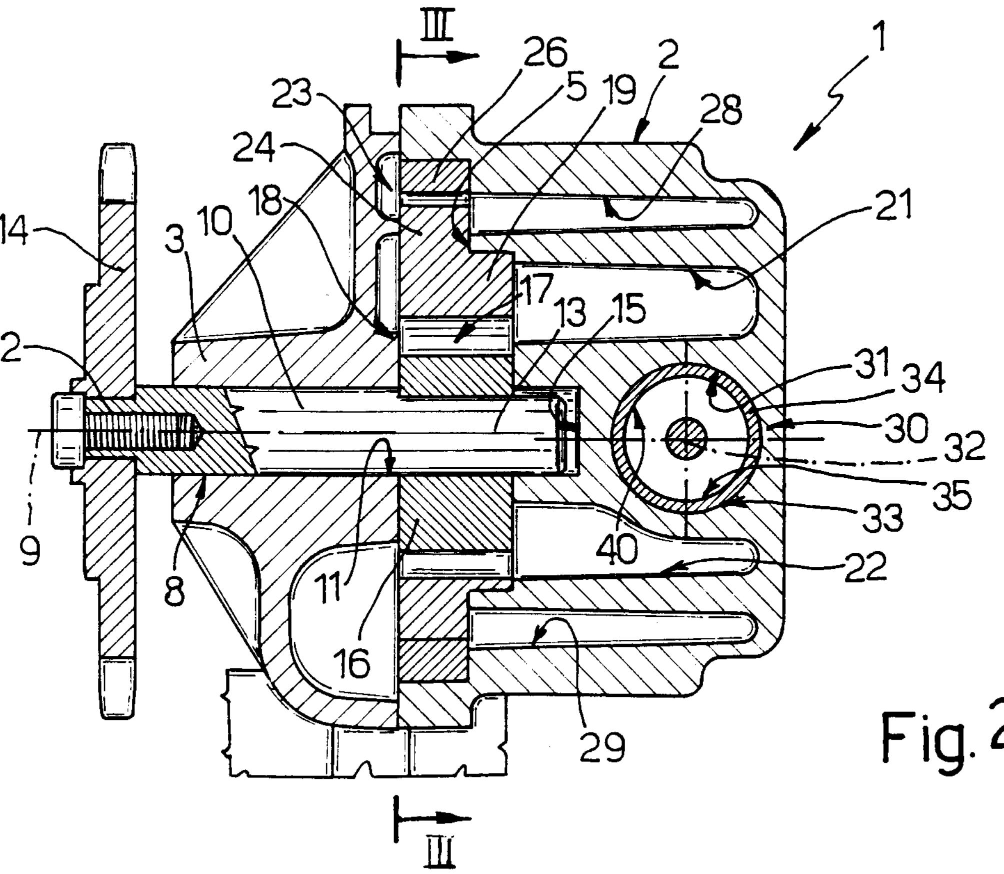
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Fig. 2

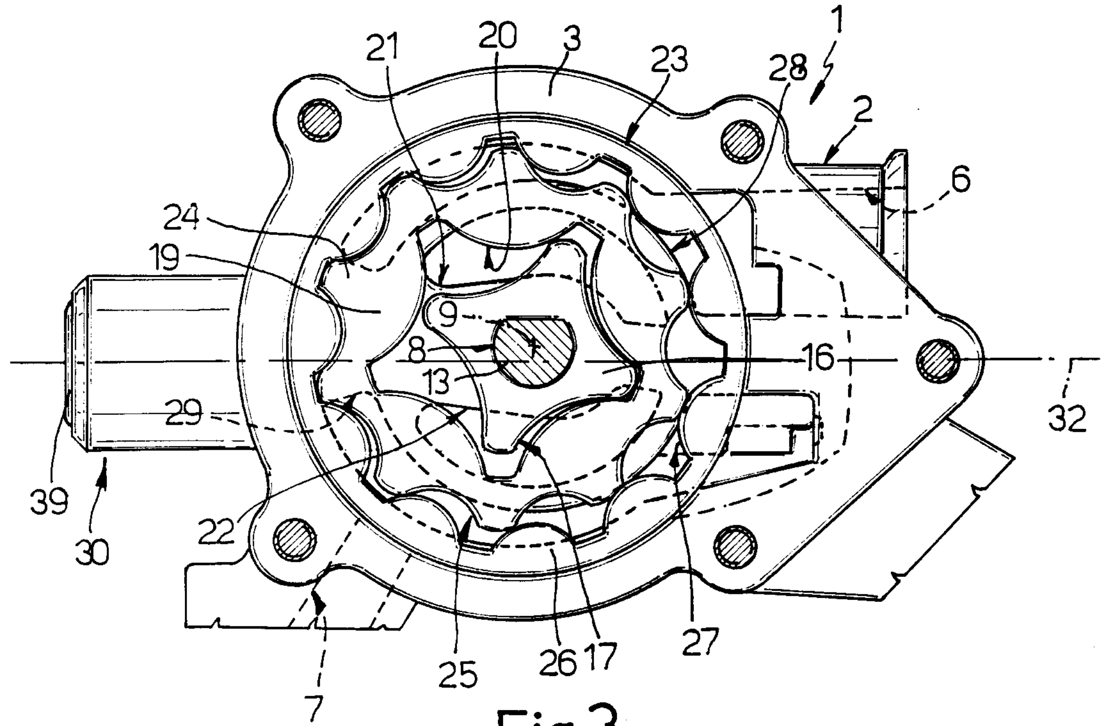
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Fig.3

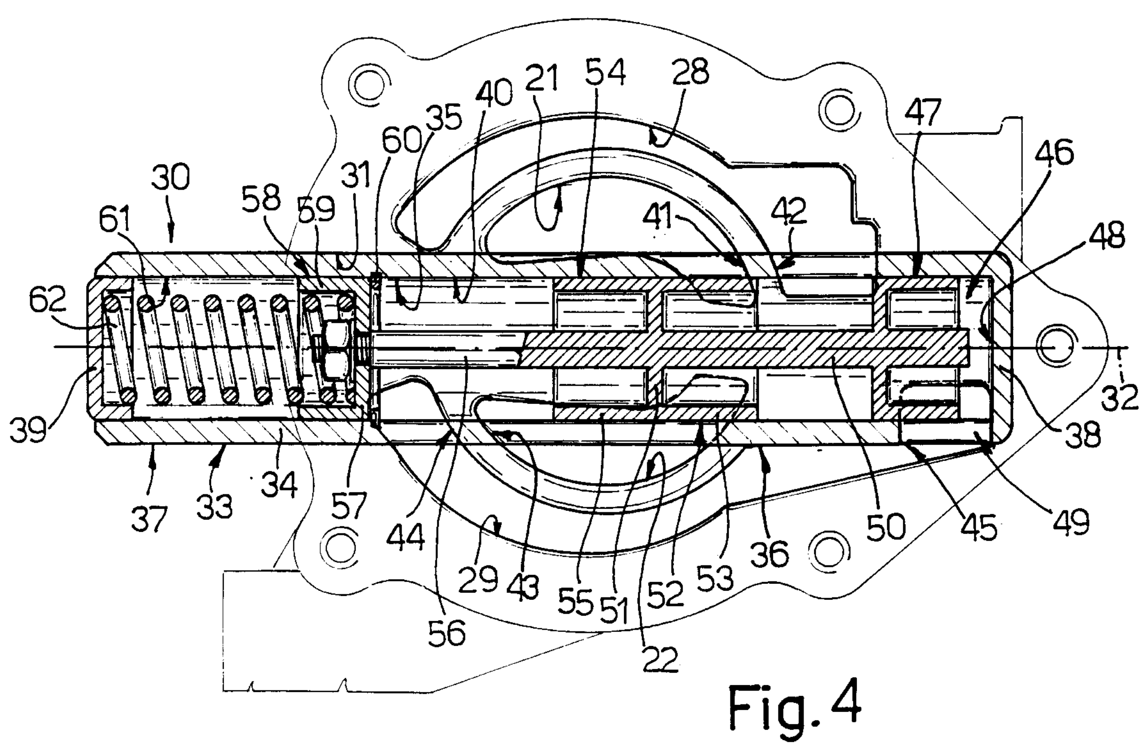
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Fig. 4

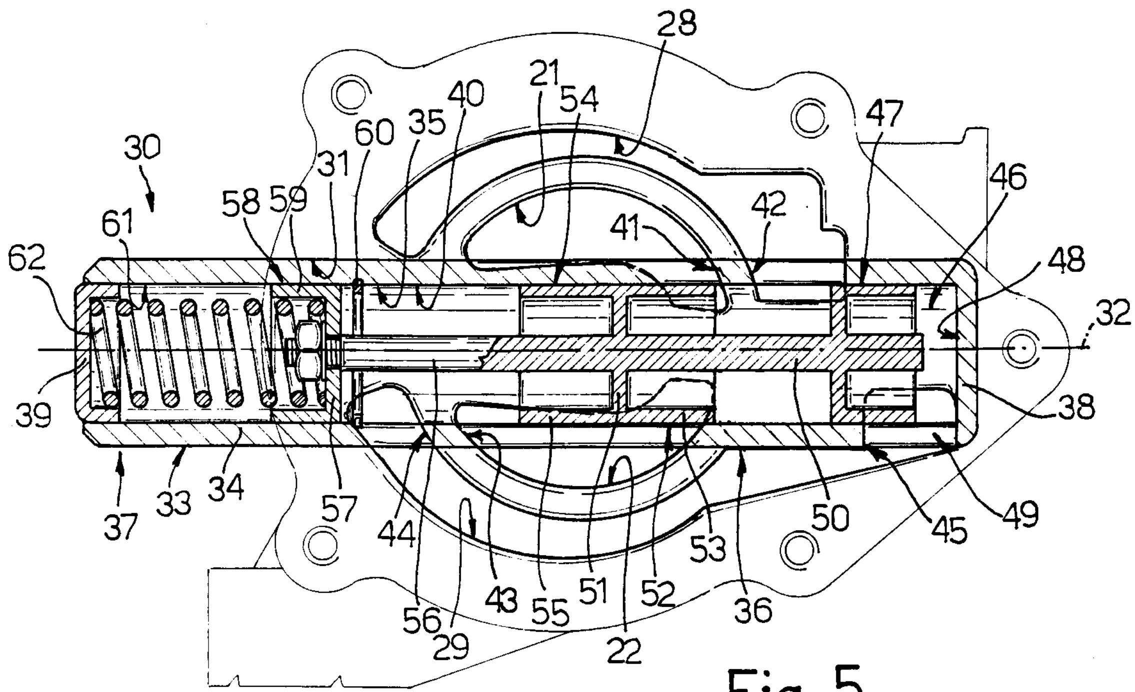
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Fig. 5

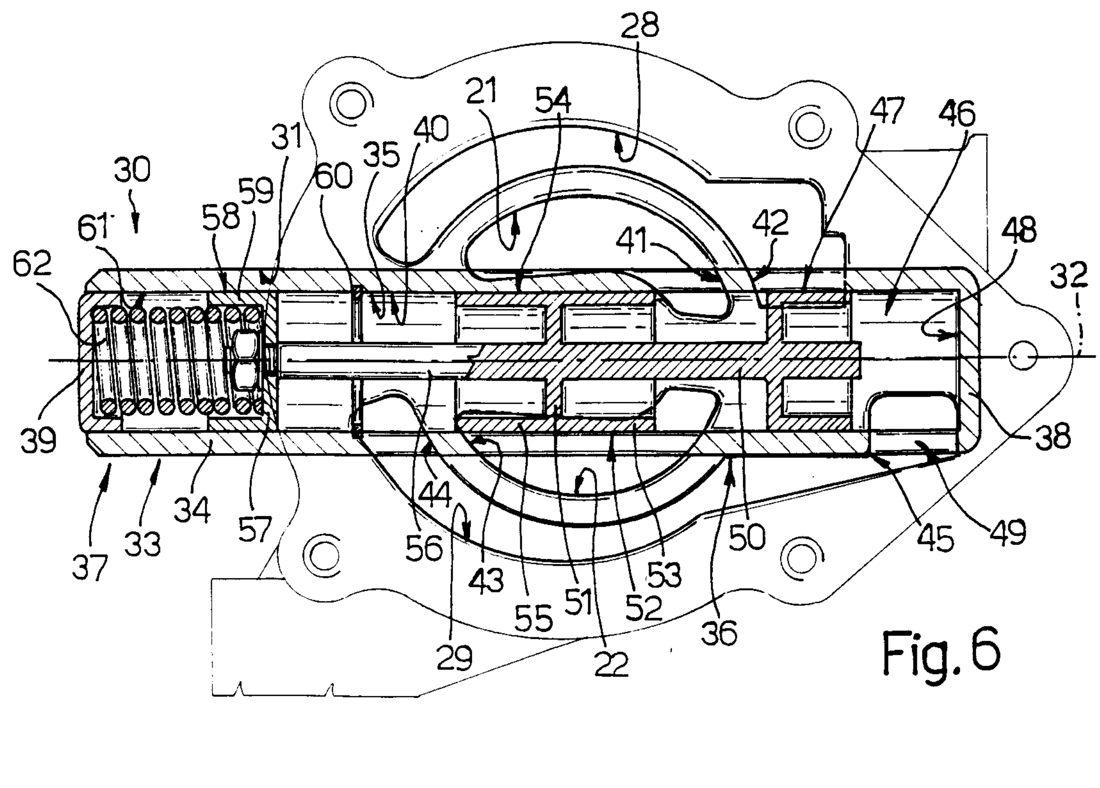
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Fig. 6

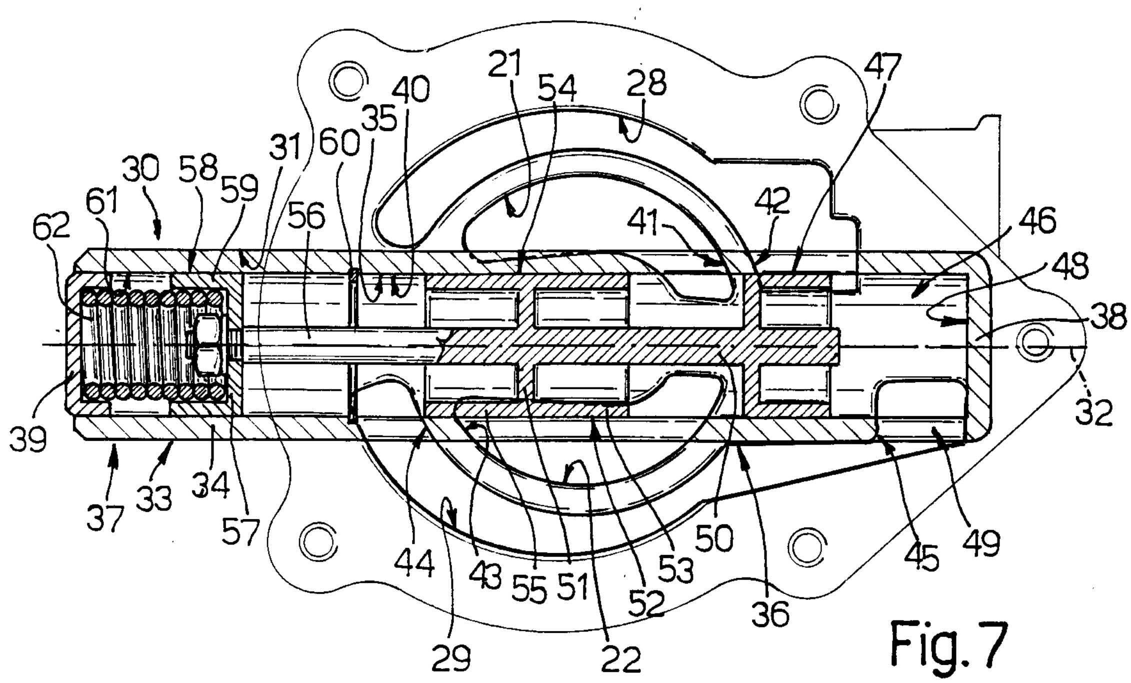
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Fig. 7

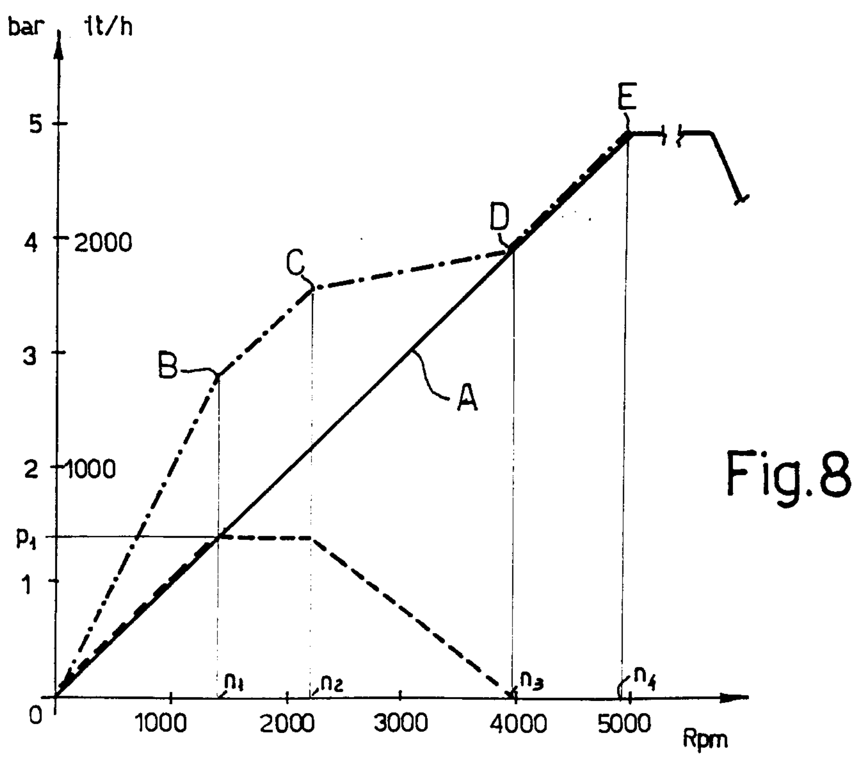
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Fig. 8