

[54] SELF REGULATING FLUID PUMP
[75] Inventor: Carlo Cecchi, Turin, Italy
[73] Assignee: Fiat Societa per Azioni, Turin, Italy
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417/428
[51] Int. Cl. F04b 49/08
[58] Field of Search 417/310, 428, 288;
137/115

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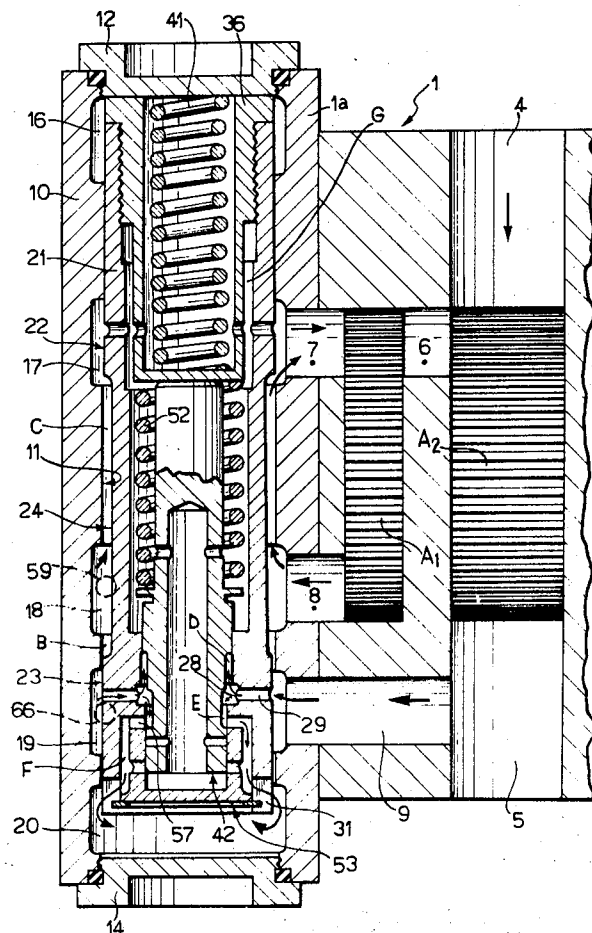
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Primary Examiner—Carlton R. Croyle
Assistant Examiner—Richard Sher
Attorney—Richard C. Sughrue et al.

[57] ABSTRACT

A fluid pressure pump of the type in which fluid is pumped by pairs of rotating gears, in which there are provided two pairs of gears and a control valve for by-passing one pair of gears if the delivery pressure of the pump exceeds a threshold value. This reduces the fluid flow through the pump and allows it to create a higher delivery pressure without requiring more power from the input shaft. The control valve is of the slide type and operates to short circuit the delivery side of one of the pairs of gears to its own induction side so that this pair of gears is inoperative when the valve is moved to its intervention position. The main slide of the valve has an auxiliary slide which enables the main slide to operate with a snap action to avoid hunting.

1 Claim, 7 Drawing Figures



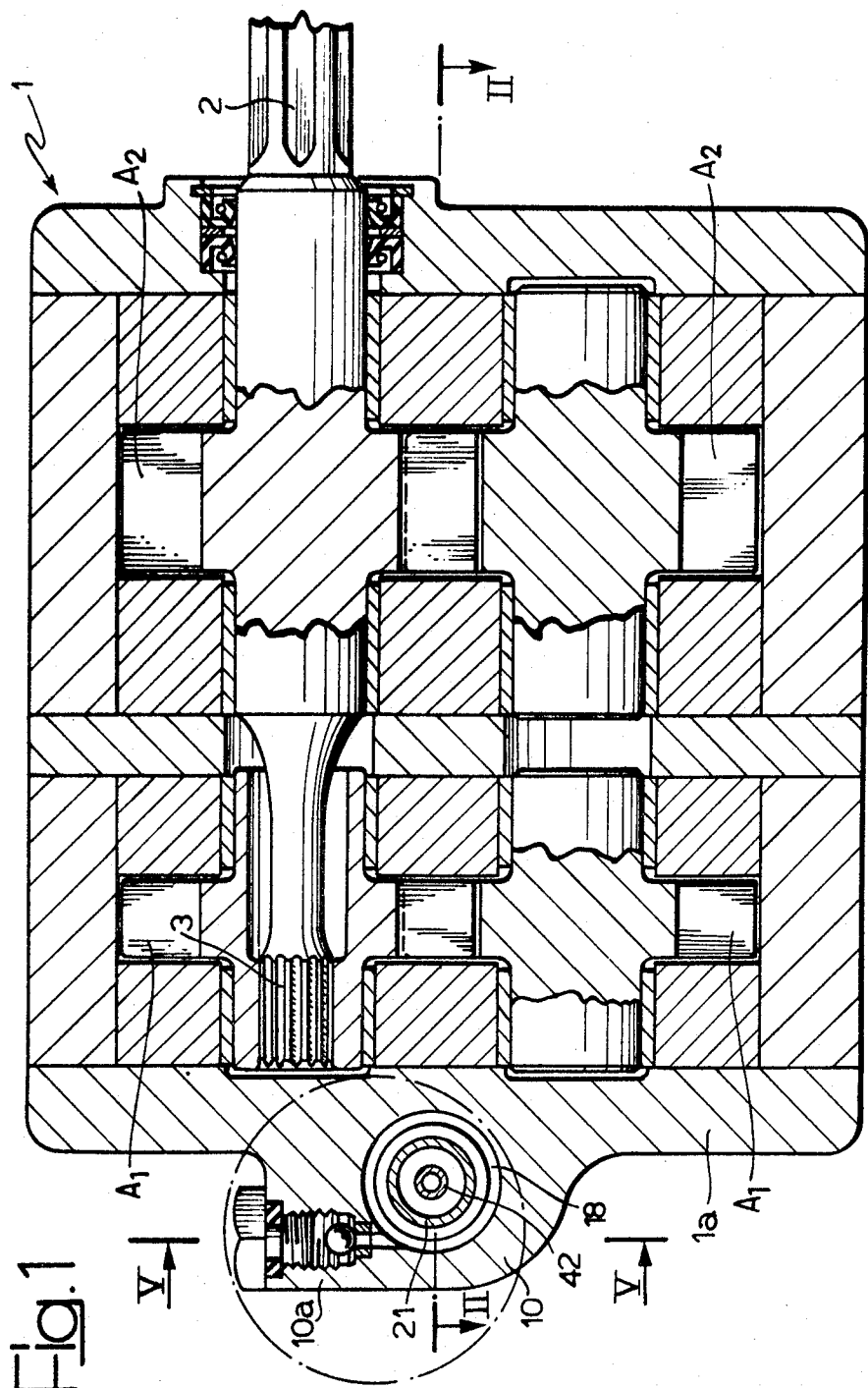


Fig. 3

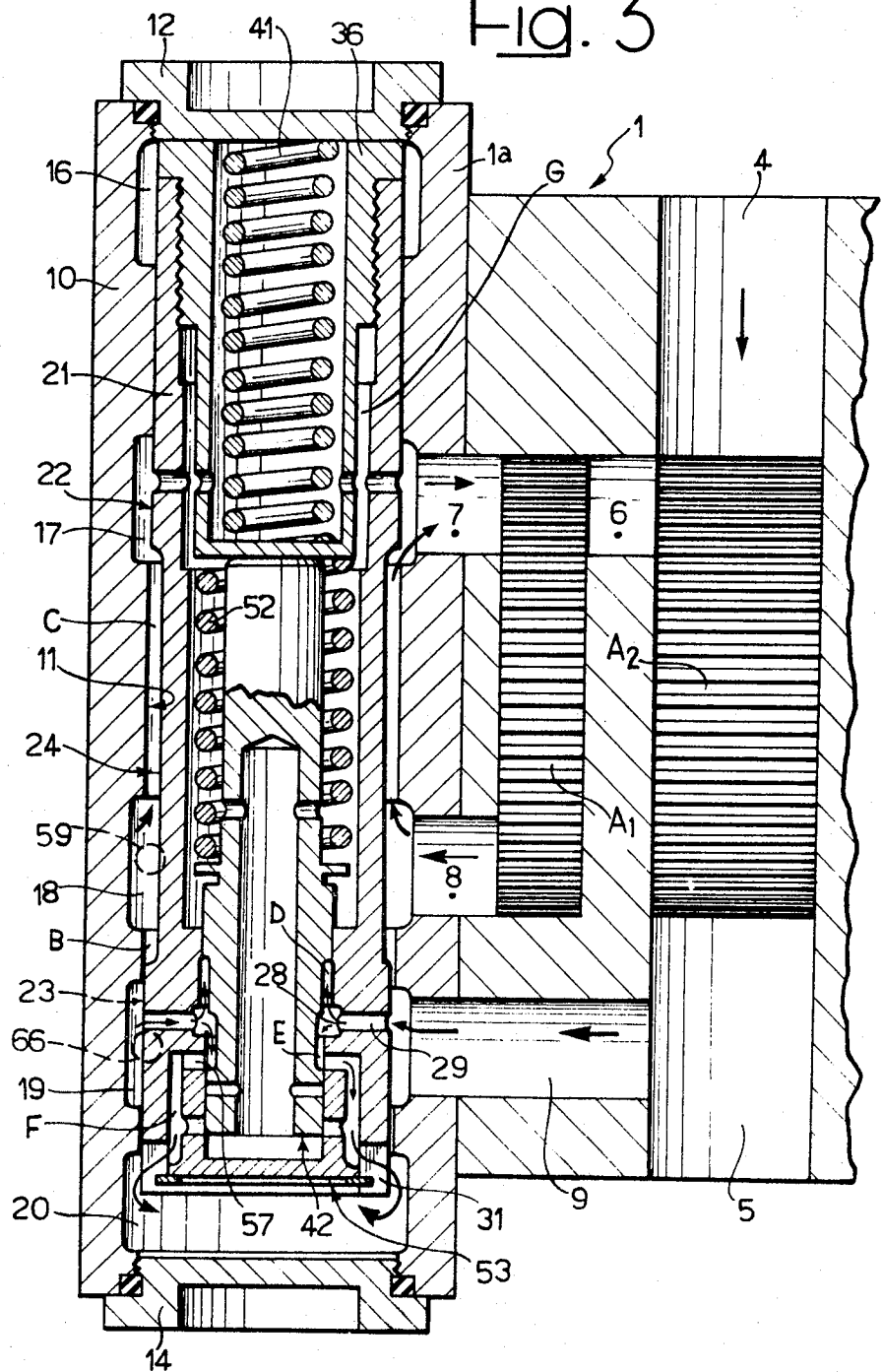


Fig. 4

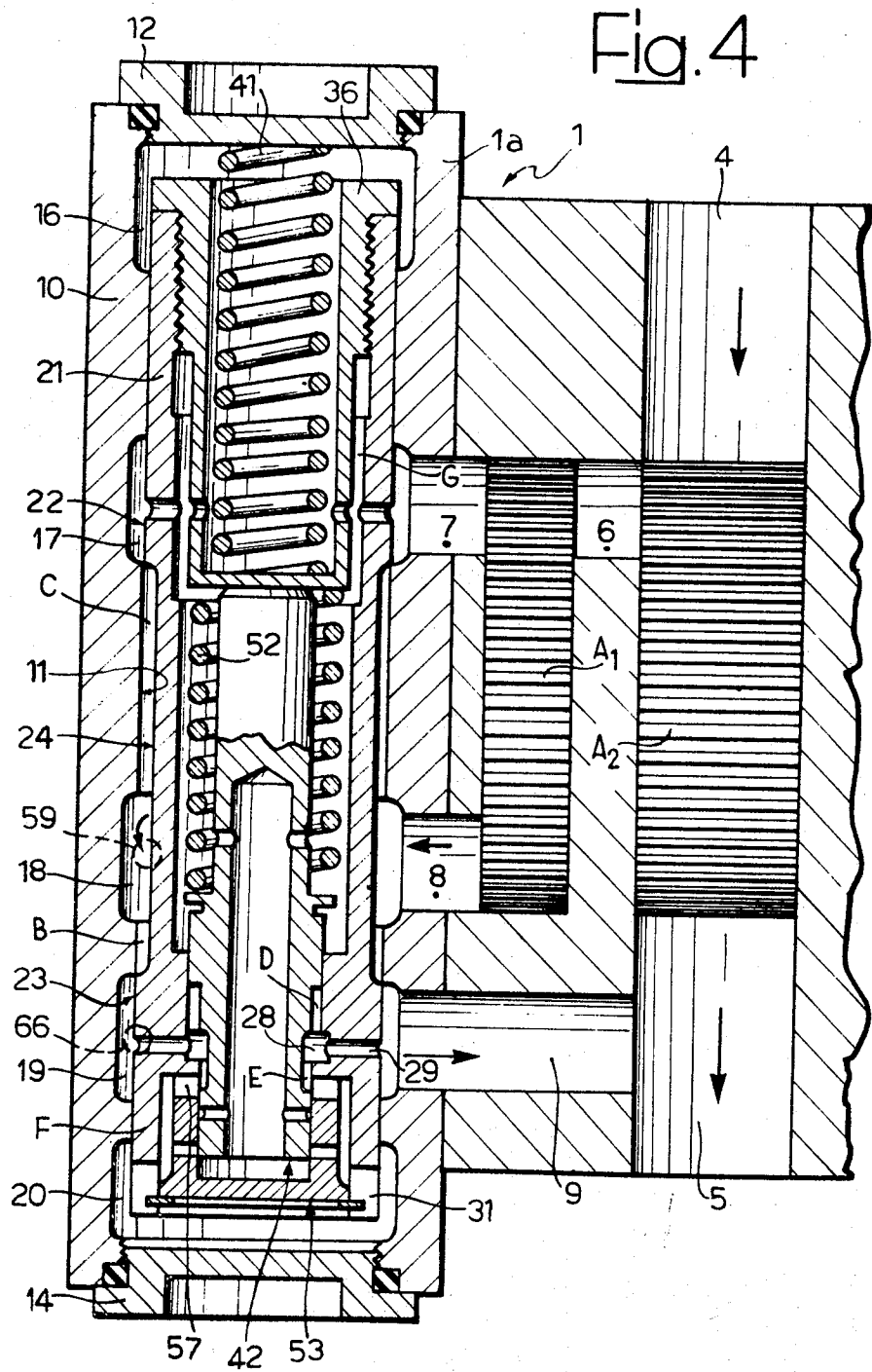


Fig. 5

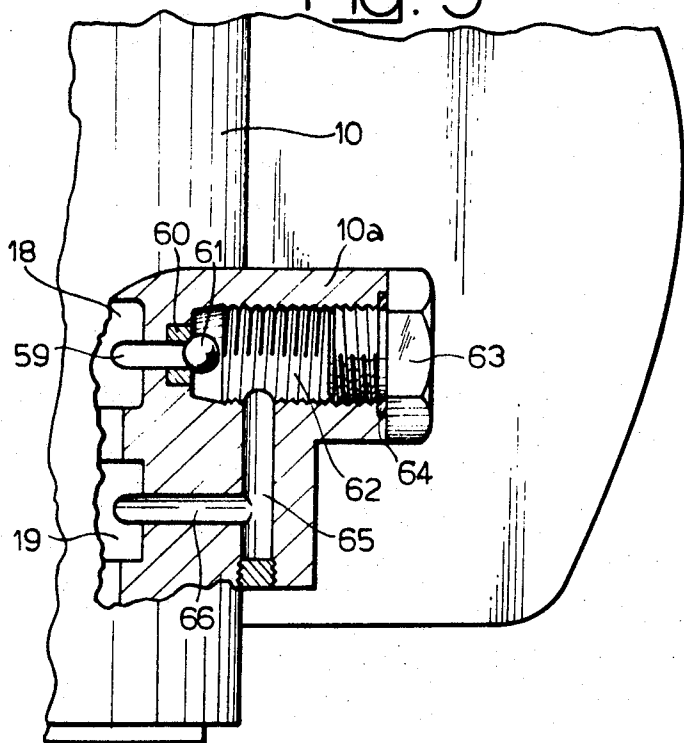


Fig. 7

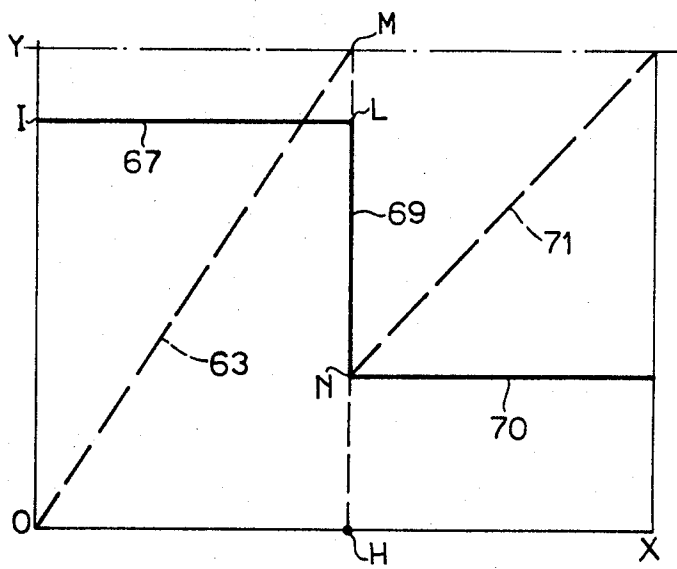
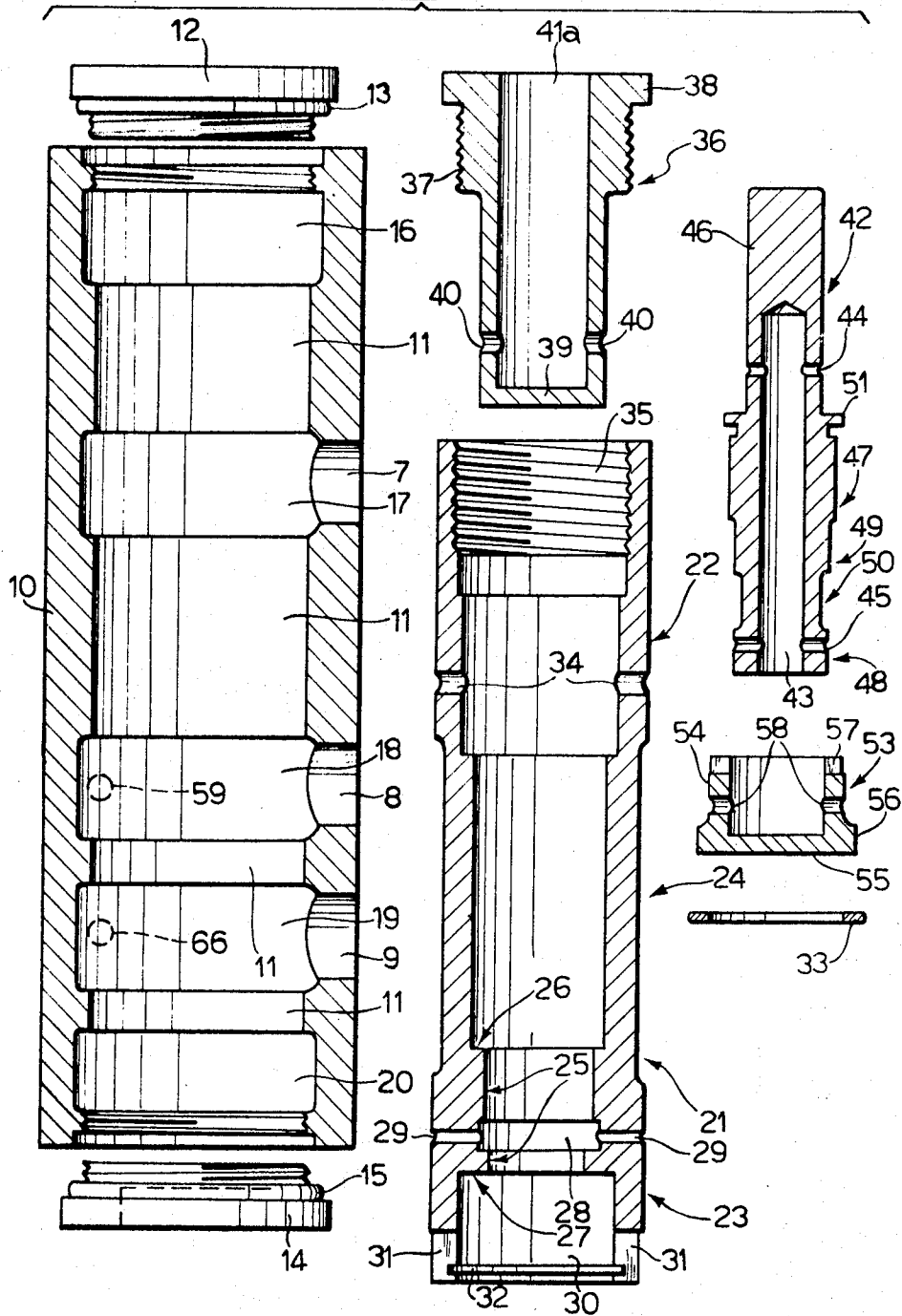


Fig. 6



SELF REGULATING FLUID PUMP

BACKGROUND OF THE INVENTION

This invention relates to gear pumps and particularly to pumps used in hydraulic systems. Pumps of this type find particular utility for use in the hydraulic system of tractors adapted for moving earth or similar.

In such vehicles the power delivered by the engine is partly used for the operation of the hydraulic tools such as a mechanical shovel, a scoop or a grab, and partly used in the locomotion of the tractor. Thus it is necessary to distribute the available power between the drive transmission of the tractor and the hydraulic system of the tool depending on the instantaneous requirements of the work in progress.

In known systems this distribution is obtained by using two separate and independent pumps in the hydraulic circuit, which are controlled by a regulating and control valve. Such a system, although satisfactory in other respects, is cumbersome and costly as each pump must be able to provide the whole of the available power from the engine, in cases where all the power of the engine is to be delivered to one or other part of the system and each pump must therefore have a considerable size and bulk; this makes the use of such systems on small tractors difficult due to the lack of available space.

OBJECTS OF THE INVENTION

One object of this invention is to avoid the above mentioned disadvantages by providing a double gear pump having an hydraulic device capable of controlling the pump output of the pump to maintain the output pressure within predetermined limits by dividing the load between two parts of the pump automatically in dependence on the delivery pressure in the output of the pump.

A further object of this invention is to provide a pump of the above type, which is smaller and less cumbersome than known similar pumps and which results in being compact and simple to manufacture, and which has a precise and safe operation.

SUMMARY OF THE INVENTION

According to the present invention there is provided a fluid pump for a hydraulic system comprising:

- a housing for said pump,
- two pairs of cooperating gears in said housing, said two pairs of gears being of the same diameter and driven by a single shaft,
- an induction conduit in said housing, said induction conduit leading to one of said pairs of gears,
- a first passageway in said housing, said first passageway connecting said induction conduit to said other of said pairs of gears,
- a delivery conduit in said housing, said delivery conduit leading from said one of said pairs of gears,
- a second passageway in said housing, said second passageway communicating with said delivery conduit,
- a control valve in said housing, said control valve being interposed between said other of said pairs of gears and said second passageway in said housing leading to said delivery conduit, said control valve responding to the pressure in said delivery conduit to provide a connection between said other of said pairs of gears and said second passageway leading

to said delivery conduit, when said pressure in said delivery conduit is below a predetermined threshold value, and to short circuit said other of said pairs of gears when said pressure in said delivery conduit is above said threshold value whereby at pressures above said threshold value only said one of said pairs of gears of said pump operates to create said delivery pressure such that the fluid flow through said pump is reduced.

The pump can thus operate in one of two different states depending on the required delivery pressure; in one state both pairs of gears are in operation causing a higher rate of flow of the hydraulic fluid and requiring a large part of the available power from the engine, which is transmitted through the input shaft to the pump; in the other state, to which the pump is automatically switched when the required delivery pressure exceeds the threshold value of the control valve one of the pairs of gears is short circuited effectively reducing the rate of fluid flow through the pump thereby allowing higher output pressures to be achieved from the same input power from the engine with, of course, a slower movement of the hydraulically actuated members due to the reduced rate of flow.

During operation of a tractor equipped with hydraulic earth moving apparatus or similar hydraulic equipment one of the most critical situations from the point of view of power distribution arises when it is necessary to move the tractor vehicle at the same time as the hydraulic equipment is applying a considerable force. In this case the device which is the subject of the present invention operates to distribute the power available from the engine so that there is sufficient for the operation of the hydraulic equipment under load and also for the locomotion of the tractor, and this operation is independent of vehicle's motion.

In particular, during operation of an industrial tractor having hydraulic equipment there are two operations in particular which require the maximum available power to be appropriately distributed, these are when the hydraulic equipment is applying a digging or tearing action which is assisted by movement of the tractor in the direction in which the force is applied, and when the hydraulic equipment is raising a maximum load against the force of gravity.

In these two cases the maximum power required from the engine can be about the same, but with different distributional requirements of the output pressure and pumping capacity. The double gear pump with self-regulating device of this invention resolves in a satisfactory manner the problem of the best utilization and distribution of the available power. For example, in the case in which the force applied by the hydraulic equipment is assisted by a force applied by the wheels of the tractor in attempted locomotion it is necessary to apply a considerable force to overcome the resistance of the material being torn or dug so that the hydraulic equipment requires the maximum amount of power while still leaving a margin sufficient for the locomotive force of the tractor. The speed of movement of the components of the hydraulic system during such an operation is negligibly small and since it is necessary to apply a large force the pressure at the delivery of the pump is higher than the threshold value of the control valve. The pumping capacity of the pump is therefore automatically reduced and this lowers the operating speed of movement (which is acceptable) while containing

the power requirements of the system within reasonable limits.

In the case where the operation of the hydraulic system is not accompanied by locomotive force such as during a lifting operation it is preferable to move the apparatus at a high speed in order to shorten the working time and it is only necessary to apply a relatively low force for the operation of the hydraulic system. The pressure at the delivery of the pump is therefore lower than the threshold value of the control valve and the full pumping capacity of the pump can be used. This causes the hydraulic system to move at its maximum speed.

Further characteristics and advantages of this invention will become evident in the course of the following detailed description with reference to the accompanying drawings which is provided solely by way of non-restrictive example.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a cross sectional plan view of a double gear pump with a control valve constructed as an embodiment of the present invention;

FIG. 2 is a longitudinal section, taken on the line II—II of the embodiment of FIG. 1 illustrating the control valve in a first operative position, corresponding to the maximum pumping capacity of the pump;

FIG. 3 is a view similar to FIG. 2, and illustrates the control valve in a second operational position in which one of the pairs of gears is prevented from pumping;

FIG. 4 is a view similar to FIGS. 2 and 3, and illustrates the control valve in an intermediate position between the two positions illustrated in FIGS. 2 and 3;

FIG. 5 is a partial transverse section taken on the line V—V of FIG. 1;

FIG. 6 is an axial section of the components of the control valve, showing the valve in a dismantled condition; and

FIG. 7 is a diagram illustrating the variation and intercalation of pumping capacity, input power and output pressure from the pump.

DESCRIPTION OF THE PREFERRED EMBODIMENT

In this description the pump and control valve are described for convenience in the orientation in which they are shown in the drawings; thus the part of the pump and control valve nearest the top of the page will be referred to as the top of the pump and movement towards the top of the pump as upward movement, although it will be realized that, in practice, the pump could be mounted in any orientation.

Referring now to FIG. 1 there is shown a double gear pump having a body 1 within which there are mounted two pairs of pumping gears A1 and A2 which operate in the normal fashion of pump gears.

The gears A1 and A2 have the same diameter but different axial width, and are driven by the same shaft 2 which in turn is driven by the engine (not shown). The shaft is joined to one of the gears of the pair A2 and has a splined end 3 which transmits drive to one of the gears of the other pair A1.

The two pairs of gears are parallel to each other and have a single induction conduit 4 and a single delivery conduit 5, both formed in the body 1. One of the pairs of gears A2 is directly connected between the two conduits 4 and 5 mentioned above, while the other pair A1

is connected to the induction conduit 4 by a conduit 6 and connected to the common delivery conduit 5 via two transverse conduits 8 and 9 the coupling between which is controlled by a valve sliding in a part 10 of one end 1a of the casing 1.

As illustrated in FIG. 6, the control valve comprises a cylindrical bore 11, extending axially through the part 10 of the end 1a and closed at each end by a respective plug 12, 14 provided with sealing rings 13, 15 and threaded so as to screw into the respective ends. The bore 11 is formed with a number of enlarged chambers 16, 17, 18, 19, 20, axially spaced one from the other; the intermediate chamber 17, 18, 19 are respectively connected to transverse openings 7, 8, 9 which head respectively to the induction conduit 7 of the pair of gears A1, to the delivery conduit of said pair of gears A1 and to the common delivery conduit 5.

Within the bore 11 there is mounted a main slide 21 which operates to control the interconnection of the delivery conduit 8 of the pair of gears A1 and the two discharge openings 7 and 9. The main slide 21 is in its turn provided with a further inner slide 42 which is slidable with respect to the main slide and which is moved by the effect of the pressure in the delivery conduit 5 to determine the shifting of the main slide 21.

The main slide 21 comprises a tubular member having two terminal parts 22, 23 of such a diameter that they can slide sealed within the bore 11 and an intermediate zone 24 of a reduced diameter.

Inside the slide 21 there is a calibrated chamber 25 extending between two transverse shoulders 26 and 27 and having an enlarged intermediate annular chamber 28 which communicates with the outside through radial openings 29. Adjacent and below the calibrated chamber 25 there is an enlarged zone 30 which communicates with the outside through side openings 31 and which has an annular groove 32 for receiving a circlip 33.

Towards the other (upper) end of the slide 21 there are at least two radial discharge openings 34; the upper end of the slide 21 is threaded on the inside over a portion 35, which cooperates with a threaded part 37 of a closing element 36 which is screwed into the upper end of the slide 21. The element 36 has a terminal radial flange 38, and is provided with an axial cavity 41a which is open at the top and closed at the bottom by a transverse wall 39 adjacent which there are two radial openings 40 for communication with the outside. In the cavity 41a of the element 36 there is an helical spring 41, acting between the upper plug 12 and the bottom wall 39 of the element 36 and operating to urge the slide 21 downwards to the position shown in FIG. 2.

Within the calibrated chamber 25 of the slide 21 there is an auxiliary slide comprising a body 42 which has an internal bore 43 which is open at the bottom and which has radial openings 44 and 45 at the two ends thereof. The auxiliary slide 42 has an upper part 46 with a reduced diameter, a centre part 47 able to slide sealed within the chamber 25 of the main slide 21, and two lower zones 49 and 48 of reduced diameter, between which is interposed a third zone 50 having a further reduced diameter; above the intermediate part 47 there is a flange 51 against which acts one end of an helical spring 52 the other end of which acts against the lower end of the upper closure element 36 to urge the auxiliary slide 42 downwards with respect to the main slide 21.

In the lower chamber 30 of the main slide there is a further closure element 53, held in place by a circlip 33 in the groove 32 and having a cylindrical part 54 closed below by a transverse wall 55 having an annular radial protruding flange 56. The tubular part 54 is provided with upper radial opening 57 and lower radial openings 58; the inside of the closure element 53 cooperates with the lower zones 48, 50 of the auxiliary slide 42.

Two of the intermediate chambers 18, 19 within the bore 11 communicate with inclined openings 59, 66 which are parallel to one another and connected by an axial passage 65 and by a chamber 62 closed by a plug 63 having a sealing ring 64, arranged in an appendix 10a of the part 1a. The opening 59 is controlled by a ball valve having a ball 61 cooperating with a valve seat 60 the operation of which will be explained in detail below.

OPERATION OF THE PREFERRED EMBODIMENT

The operation of the pump and of the control device illustrated is as follows:

When the delivery pressure is lower than the threshold value of the control valve, the position adopted by the device is that shown in FIG. 2 in which the springs 41 and 52 keep the main slide 21 and the auxiliary slide 42 at the end of their axial travel. The oil which enters the induction conduit 4 to the gears A2 also passes through the conduit 6 and is sucked by the two pairs of gears of the pump; the pair of gears A2 forces the oil directly into the delivery conduit 5, while the pair of gears A1 forces the oil into the delivery conduit 5 via the conduit 8, the chamber 18, the passage B, the chamber 19 and the conduit 9; therefore the two stages of the pump operate in parallel with the maximum pumping capacity. The oil in the chamber 19 also extends into the radial perforations 29 and exerts a pressure on the auxiliary slide 42 opposite to that of the spring 52, without, however, overcoming the resistance of this spring.

When the delivery pressure becomes greater than the threshold or calibration pressure of the control valve, the various components of the device move to the position shown in FIG. 3; that is, if the pressure which is exerted in the perforations 29 overcomes the resistance of the spring 52 the auxiliary slide 42 is shifted upwards. In this new position the oil which passes through the openings 29 in the chamber 28, acts both in the chamber D thereby continuing to act to push the auxiliary slide upwards, and in the lower chamber E, from where it passes through the radial openings 57, to a chamber F formed by the lower closure member 53 and the lower part of the main slide, from whence it passes through the openings 31 and reaches the lower chamber 20 of the valve where it exerts on the wall 55 of the lower closure member 53 an upwards force sufficient to overcome the resistance of the spring 41 and to cause upward shifting of the whole of the main slide assembly.

In this new position the pair of gears A2 continue to pump the oil directly to the delivery conduit 5, whereas the pair of gears A1 is no longer in communication with the chamber 19 and therefore with the delivery conduit 5, because the passage B leading to the chamber 19 is now obstructed by the slide 21. The pump A1 is only in connection with the chamber 18 and the cavity C via the delivery conduit 8, and the chamber 17 via the induction conduit 7, and therefore will be effectively

short circuited so that the only pumping action will be that provided by the gears A2.

As the slide 21 shifts from the position illustrated in FIG. 2 to the position illustrated in FIG. 3 it must pass through a position in which both of the passages B and C are simultaneously closed. This position is illustrated in FIG. 4. In this position the fluid pressure created by the pump A1, not finding an outlet, would create a potentially damaging and dangerous pressure. To overcome this problem a by-pass route, shown in FIG. 5, is provided to allow communication between the chambers 18 and 19 by means of the ball valve 61; therefore, for the short period during which this situation exists, until the sliding parts reach the position illustrated in FIG. 4, the output pressure of the gears A1 is discharged into the delivery conduit 5 by following the passages 18, 59, 62, 65, 66 and 19.

The ball valve 61 is normally held in a closed position on the seat 60 because the delivery pressure in the conduit 5 is applied thereto since the chamber 62 is in communication with the chamber 19 and therefore the delivery conduit 5.

When the delivery pressure falls to a value lower than that at which the control valve changes position the component parts of the valve return to the positions illustrated in FIG. 2. As the pressure passes the threshold value mentioned above, the auxiliary slide 42 urged by the spring 52 moves with respect to the main slide 21 thereby breaking the communication between the passages 29 and 57 first established by the annular recesses D, E and aligning the conduits 45 and 58. At this point the fluid contained in the lower chamber 20, which had been kept under pressure through the passages 9, 29, D, 57, F, and 31, cannot oppose the action of the spring 41, which therefore causes the shifting of the main slide 21 downwards until it engages with the lower plug 14. While this movement is taking place the oil which remained closed in the passages 20, 31, 58, F, and 57, can escape through the passages 58, 45, 43, 44, G, 34, 17, 7, thus returning to the induction conduit.

The variation of the throughput capacity of the pump, and the required input power in dependence on the required output pressure is qualitatively shown in a theoretical diagram illustrated in FIG. 7 of the accompanying drawings, given a constant rate of rotation of the pump. The axis O-X shows the value of the required delivery pressure from the pump, and on the axis O-Y there are plotted the values both of the throughput capacity of the pump and the input power required to provide the given output pressure. As the required output pressure increases from zero to the value H which is the discriminative threshold pressure of the control valve, the throughput capacity remains constant and is represented by a straight line 67 parallel to the axis O-X, which extends from the point I to the point L; since the throughput capacity remains constant the power required increases linearly from zero to a maximum value M as shown by the inclined straight line 68.

When the delivery pressure reaches the threshold value H of the control valve, the throughput capacity of the pump is rapidly decreased to a reduced value as is shown by line 69 which extends from the point L to the point N; the delivery pressure remains constant and the input power required decreases sharply as shown by the broken line which extends from the point M to the point N. If the output pressure increases subsequently,

the input power required increases from the value to which it dropped at the threshold pressure (shown at N) in a linear fashion as shown by the line 71. The pumping capacity remains constant at the reduced value, as indicated by the straight line 70, parallel to the axis O-X. The power continues to rise until it reaches about the maximum value which was reached at the threshold value of the output pressure when the pump was operating at the higher pumping capacity.

It will be appreciated from the above description how the device described above automatically regulates the delivery of the double gear pump in dependence on the load pressure in the output of the pump; the shifting of the movable parts of the control device taking place in a totally automatic way and without any manual intervention whatsoever.

Naturally still keeping to the principle of this invention, the forms of realisation and the manufacturing details can be amply varied, in respect to what has been described and illustrated, without departing from the scope of this invention.

I claim:

1. A fluid pump for a hydraulic system comprising:
 - a housing for said pump,
 - two pairs of cooperating gears in said housing, said two pairs of gears being of the same diameter and driven by a single shaft,
 - an induction conduit in said housing leading to one of said pairs of gears,
 - a first passageway in said housing connecting said induction conduit to the other of said pairs of gears,
 - a delivery conduit in said housing leading from said one of said pairs of gears,
 - a second passageway in said housing communicating with said delivery conduit,
 - a bore in said housing, said bore being closed at each end thereof,
 - a number of annular chambers spaced along said bore, a first one of said chambers communicating with said first passageway and a second one of said chambers communicating with said second passageway.

- a third passageway in said housing communicating with the delivery side of said other pair of gears and a third one of said chambers communicating with said third passageway,
- a main slide movable in said bore between first and second positions; said main slide being shaped to allow communication between said first one of said chambers and said second one of said chambers in said first position of the main slide and to allow communication between said first one of said chambers and said third one of said chambers in said second position of the main slide,
- means resiliently biasing said main slide to said first position,
- a fluid flow path extending between said first and second chambers and communicating with the opposite ends of the main slide,
- an auxiliary slide movably mounted in said main slide between a first and second position,
- means resiliently biasing said auxiliary slide to its first position against fluid pressure in said delivery conduit to close said fluid path when the pressure in said delivery conduit is below a predetermined threshold pressure,
- said auxiliary slide being movable to its second position under pressure in said delivery conduit above said threshold pressure to open said path and to subject one of said ends of the main slide to the delivery conduit pressure to move said main slide to its second position,
- said main slide, in moving from its said first to its said second position, occupying a third position in which said communication between said first and third chambers and said communication between said second and third chambers are interrupted,
- a fourth passageway connecting said second and third annular chambers in said bore and a pressure relief valve disposed in said fourth passageway to permit fluid flow from said third chamber to said second chamber when said main slide is in said third position.

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UNITED STATES PATENT OFFICE
CERTIFICATE OF CORRECTION

Patent No. 3,751,190

Dated August 7, 1973

Inventor(s) Carlo Cecchi

It is certified that error appears in the above-identified patent and that said Letters Patent are hereby corrected as shown below:

After Item [21] insert-- Foreign Application Priority Date
November 17, 1970 Italy.....70831-A/70--

Signed and sealed this 20th day of August 1974.

(SEAL)
Attest:

McCOY M. GIBSON, JR.
Attesting Officer

C. MARSHALL DANN
Commissioner of Patents