July 31, 1962 F. BERG ET AL 3,046,895

VARIABLE DISPLACEMENT PUMP CONTROL

Filed Feb. 27, 1958 3 Sheets-Sheet 1

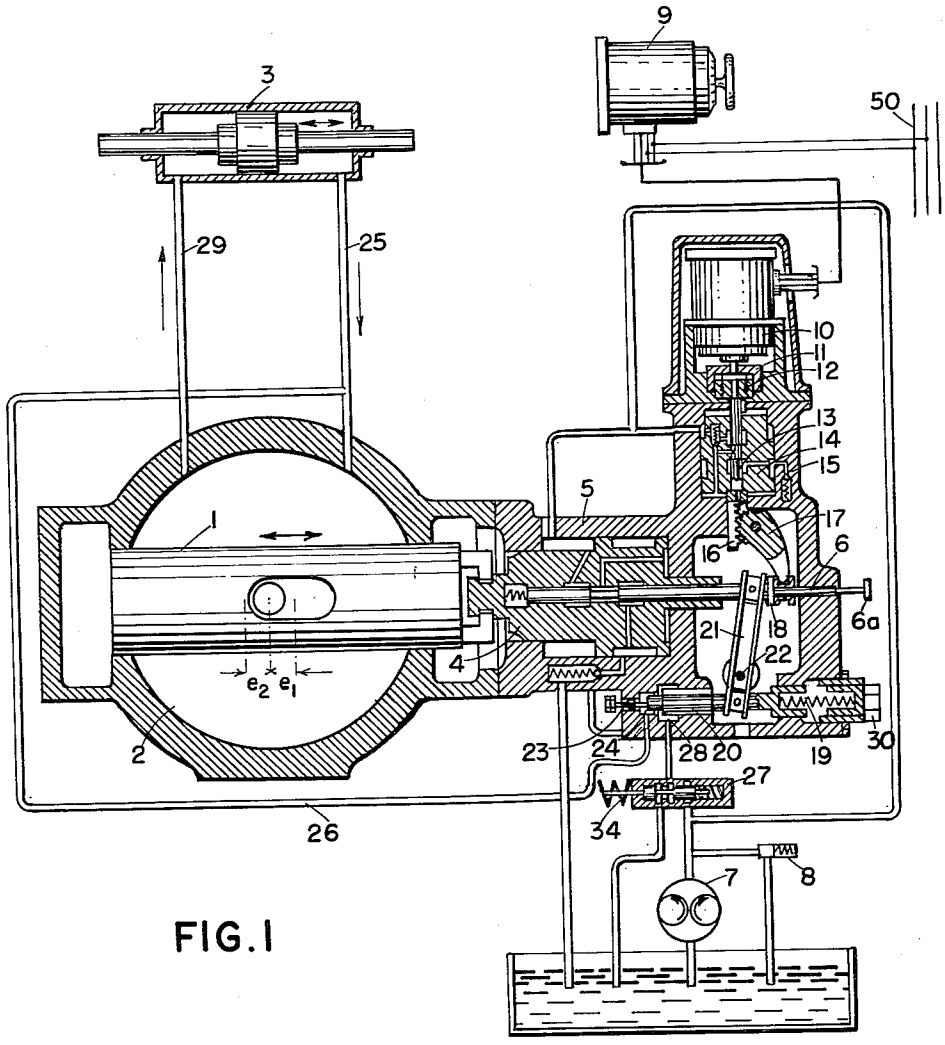


FIG. 1

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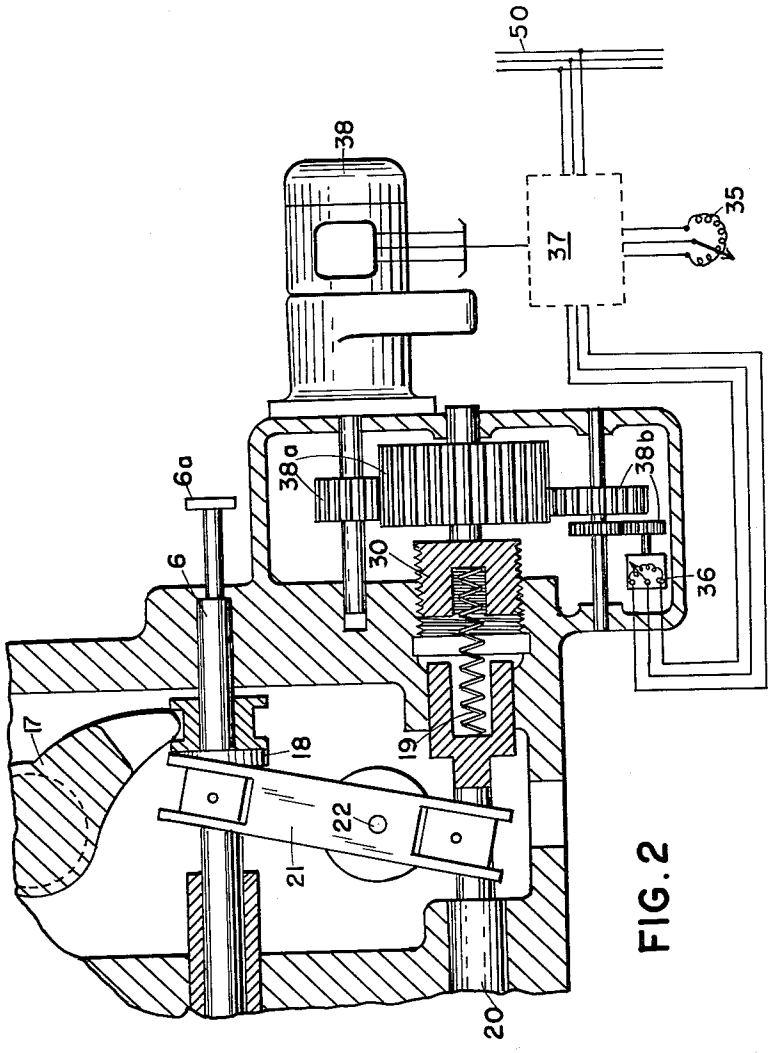


FIG. 2

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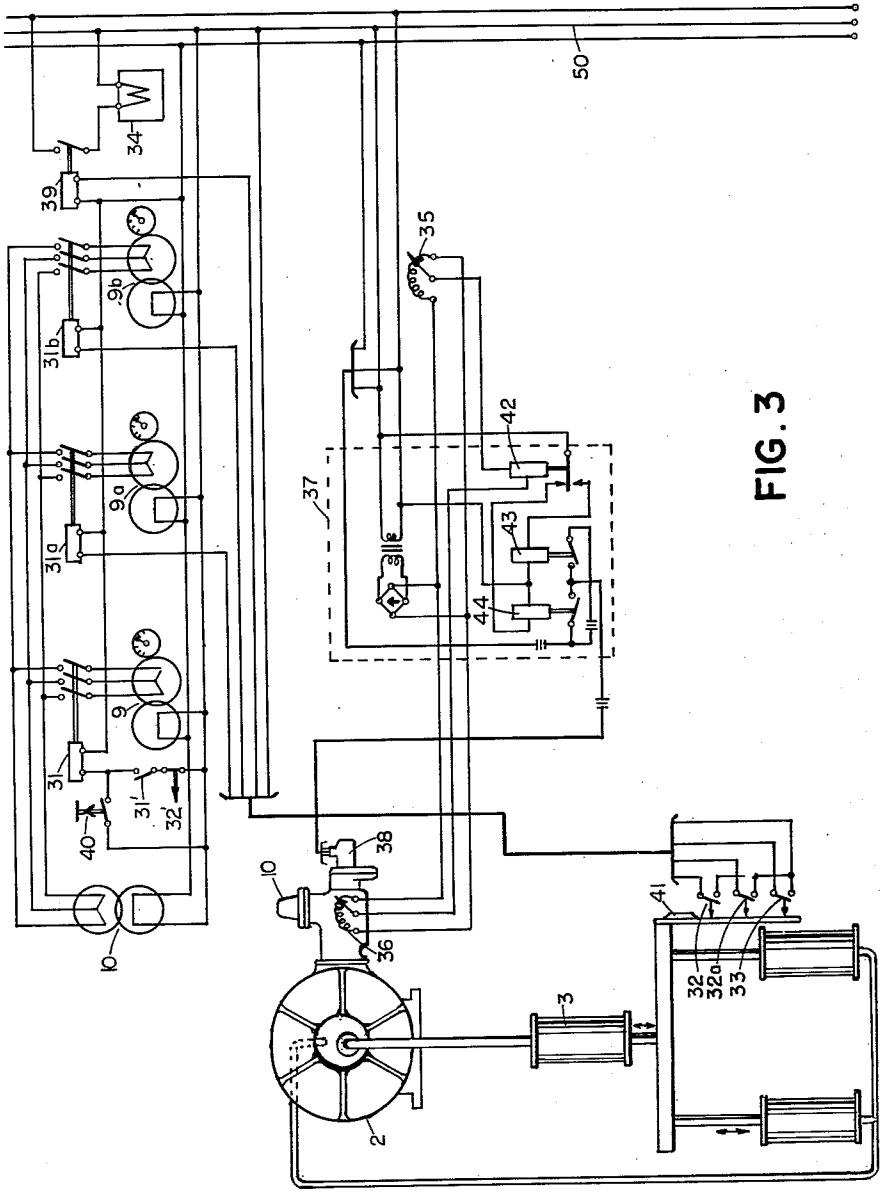


FIG. 3

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VARIABLE DISPLACEMENT PUMP CONTROL

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8 Claims. (Cl. 103—37)

The present invention relates to controls for variable-delivery pumps and more particularly for variable-and-reversible-delivery pumps, such as radial plunger pumps of the type described in the book "Oil Hydraulic Power and Its Industrial Application" by Walter Ernst (1st Ed., New York, 1949, pages 120–121), for use with hydraulic presses and the like.

The delivery of such pumps is varied by adjusting the eccentricity of a reactor ring or reaction rotor, arranged in a slide block which controls the stroke of the plungers and thereby permits variation of the delivery of the pump at constant motor speed. When the reactor ring or slide block is in a position concentric with the cylinder rotor in which the pistons are reciprocably supported, the pump is set for zero delivery. Shifting of the reaction rotor causes liquid to be pumped. A pump equipped for shifting of the reaction rotor into two eccentric positions on opposite sides of the zero-delivery position permits reversal of fluid flow without change in the speed or direction of the pump motor and without external reversing valves.

To bring about a change in pump output and direction of flow requires substantial forces to be applied to the slide block holding the reaction rotor. It has, therefore, formerly been proposed to actuate shifting of the slide block by means of hydraulic servo motors equipped with pilot valves which can be adjusted by relatively small forces. The hydraulic servo motor acts against the restraint of a spring which exerts pressure on the slide block and therefore prevents back lash and actuates the return movement of the slide block when the hydraulic servo motor is deactivated.

It has, furthermore, been formerly proposed to have the pilot valve of the hydraulic servo motor actuated against the pressure of a return spring by means of a hydraulic pressure regulator, so that the output of the pump, and thereby the working speed of the hydraulic press or other device driven by the pump can be adjusted in a stepless manner by an operator from the control panel of the press. In this known arrangement, the pilot valve of the speed adjustment is additionally acted upon by a lever system connected to means for automatically reducing the output of the pump in proportion to the pressure developed when there is an increase of pressure on the delivery side of the pump. It is apparent that with such an arrangement the delivery of the pump is reduced to a minimum at maximum design pressure, even if the machine is set for its highest working speed by the operator. Delivery will only be adequate to compensate for leakage in the hydraulic system and to maintain the set maximum pressure. It is inconvenient to control a pump, which is usually mounted on the crosshead of the press, by remote control from the operating stand of the press if a hydraulic connection between the control actuating means at the operating stand and the controls themselves has to be provided. Hydraulic control lines have a fixed length and cannot be readily changed without requiring major alterations in the entire system. Additional control elements are required if the pump control cycle is to be changed, for example, in coordination with a programming system.

It is an object of the invention to permit locating the control operating station independent of the location of the control device.

Another object is the provision of a control system which permits a plurality of successive control operations to be initiated by a single movement of the operator.

A further object of the invention is the provision of a device remotely controlled from the control station for regulating the pressure and the delivery of the fluid output of the pump.

In its more specific aspects, the present invention contemplates overcoming the disadvantages of hydraulic remote controls by replacing the remote control feature of a hydraulic control with an electric servo system, comprising at least two coordinated electric synchros, the transmitter synchro being arranged at the operating station of the press or other device actuated by the pump, which pump is controlled by the device of the invention, and the receiver synchro being arranged to actuate the pump controls.

Since the receiver of an electric servo system is capable of furnishing only a relatively weak torque, the receiving synchro of the control device of the invention is directly coupled to a first hydraulic servo motor acting on a lever which actuates the pilot valve of a second hydraulic servo motor, which in turn is in operative engagement with the slide block or reaction rotor of the variable-delivery pump. The second hydraulic servo motor is furthermore acted upon by a pressure compensating control adapted to adjust delivery of the pump, so as to maintain the predetermined fluid pressure. The device of the invention permits adjustment of this pressure compensating control from the operating stand for altering maximum fluid pressure and delivery.

Other objects and many of the attendant advantages of this invention will be readily appreciated as the same becomes better understood by reference to the following detailed description when considered in connection with the accompanying drawings, in which,

FIG. 1 is a diagrammatic view, partly in vertical section and partly in side elevation, of the control device according to the invention,

FIG. 2 is an enlarged vertical section of the pressure compensating control of FIG. 1, equipped with remote control mechanism, and

FIG. 3 is an electric wiring diagram of the control device as applied to a variable-displacement pump which actuates a hydraulic press.

Referring now to FIG. 1, there is shown a variable-and-reversible delivery pump 2 of the radial plunger type having a slide block 1 carrying the reaction rotor of the pump and shiftable between extreme eccentric positions e_1 and e_2 corresponding to maximum delivery for power stroke and return stroke, respectively, of a coordinated hydraulic machine, such as a hydraulic press actuated by a piston 3 connected to the pump 2 by a feed line 25 and a return line 29. The position of slide block 1 is shifted by means of a hydraulic servo motor 5 of conventional design and mode of operation, having a power piston 4 and a piston-type pilot valve 6 axially arranged in power piston 4 and equipped with a stop 6a limiting the stroke of valve 6 and thereby of piston 4 which follows the servo valve 6 in proportion to the servo valve movement.

A gear pump, schematically indicated at 7, provides auxiliary control fluid for the hydraulic control elements and make-up fluid for leakage losses in the main hydraulic system of pump 2 in a well-known manner. A spring-loaded relief valve 8 maintains constant pressure in the auxiliary hydraulic system.

Hydraulic servo motor 5 is controlled by an electric servo system consisting of a transmitting electric servo motor or synchro 9, the angular position of which is set

by the operator, or by an automatic programming device and a receiving synchro **10** which follows the angular movement of transmitter **9** in a well known manner, and which displaces the axial pilot valve **13** in a power piston **14** of a hydraulic servo motor **15**. The rotary movement of receiving servo motor **10** is transformed into translatory movement of the valve **13** by means of an internally threaded bushing **11**, fixedly mounted on the shaft of motor **10**, cooperating with a threaded plug **12** mounted on valve **13**. Hydraulic servo motor **15** actuates a rack **16** which engages a gear segment on one end of a lever **17**. The other end of lever **17** is forked and is adapted to move the stem of pilot valve **6** of hydraulic servo motor **5** towards piston **4** by abutting against collar **18**.

A shifting movement of valve **6** in response to an upward movement of power piston **14** in hydraulic servo motor **15** is opposed by a compression spring **19**. Spring **19**, by means of a linkage consisting of one end of a piston **20** and a lever **21** pivotally supported in the case of the control device at **22** urges collar **18** towards the forked end of lever **17** with a fixed pressure. Piston **20** is integral with a differential piston **23**, one surface of which is exposed in cylinder space **24** to the pressure in feed line **25** of the main hydraulic system with which it is connected by conduit **26**. Cylinder space **28**, in which the larger surface of differential piston **20** is exposed, is connected selectively with the pressure system of gear pump **7** or with the sump of pump **7** by a slide valve **27** actuated by a solenoid **34**.

The initial stress of compression spring **19** can be adjustably fixed by changing the position of a threaded spring cover **30**. Electrical means for remote control of the adjustment of spring **19** is illustrated in FIG. 2 which shows on an enlarged scale the forked end of lever **17** engaging the collar **18**, supported on the stem of valve **6** which is equipped with stop **6***a*. Lever **21**, which is supported on pivot **22**, mounted in the housing of the control device, slidably engages rings pivotally mounted on valve **6** and piston **20**, respectively, piston **20** being acted upon by compression spring **19**, the initial stress of which is adjustable by means of a threaded cover **30**. Cover **30** is adapted to be rotated by the gear train **38***a* of a gear motor **38**, which is furthermore connected to a potentiometer **36** by means of reduction gears **38***b*.

Operation of the device of FIGS. 1 and 2 will now be described, together with FIG. 3 which illustrates the electric circuit of a pump control device of the invention as applied to a hydraulic system operating a hydraulic press. In order to provide for several different working speeds of piston **3** of the press, there are provided in addition to transmitting synchro **9**, shown in FIG. 1, additional synchros **9***a* and **9***b*.

The operating cycle of the press is started by pushing starter button **40**, which energizes relay **31**, thus closing self-holding contact **31'** and connecting the stator windings of synchros **9** and **10**. Compensating currents will flow between the rotors of synchros **9** and **10**, until the rotor of receiving synchro **10** will assume the position corresponding to that for which the rotor of transmitting synchro **9** is set on the dial. Power for operation of the servo system is derived from a three-wire main line, generally indicated by reference numeral **50**.

The rotary movement of synchro **10**, induced by transmitting synchro **9**, causes translatory movement of pilot valve **13** and of the coordinated power piston **14**, which by means of rack **16** and lever **17** pushes collar **18** and therefore the pilot valve **6** towards the right (FIG. 1). This causes slide block **1** to shift towards the eccentric position e_1.

The piston **3** of the hydraulic press, which is set by synchro **9** for rapid forward feed, will thus move downward until cam **41** actuates switch **32** which has a normally closed contact **32'** arranged in the holding circuit of relay **31**. Relay **31** is de-energized and the circuit connecting synchros **9** and **10** is interrupted. A normally open contact of switch **32** energizes relay **31***a*, which connects the stator winding of receiving synchro **10** to that of transmitting synchro **9***a*, the rotor of which is set at an angular position corresponding to slower advance of piston **3**. When, upon further downward movement of piston **3**, cam **41** actuates switch **32***a*, relay **31***a* is de-energized and relay **31***b* actuated, and receiving synchro **10** is disconnected from transmitting synchro **9***a* and connected to synchro **9***b*, which is assumed to be set for the actual working speed. The pressure in lines **25** and **26** is increased because of the resistance of the work, which is transmitted to piston **3**. As long as the working pressure of the main hydraulic system which is transmitted to space **24** is too low to overcome the pressure of spring **19**, piston **20** remains in a position determined by the abutment of collar **18** against the forked end of lever **17**. The position of lever **17**, which is set by synchro **10**, therefore determines the eccentricity of slide block **1** which controls the delivery of pump **2** and thereby sets the traveling speed of piston **3**. As the pressure in lines **25** and **26**, and therefore in cylinder space **24**, increases further, it becomes sufficiently high to overcome the opposing force of spring **19**. Lever **21** then shifts the pilot valve **6** and slide block **1** to the left to maintain the pressure in cylinder space **28** necessary to balance the force of spring **19**. If the pressure in line **25** reaches the maximum rated value, the slide block **1** is set for approximately zero eccentricity and therefore zero pressing fluid delivery.

When the pressure in the main hydraulic system drops, lever **21** is again urged into abutment against collar **18** by spring **19** and the stem of pilot valve **6** is moved away from power piston **4** as far as the pressure in line **26** and cylinder space **24** will permit. If this pressure is low enough, the collar **18** will abut against the forked head of the lever **17** and the speed of main piston **3** will again be determined by the setting of servo motors **9** and **10**.

The speed of the working stroke of a press equipped with the control device of the invention can therefore be controlled at will by setting of transmitting servo motors **9**, **9***a*, **9***b*, etc. When a pressure set by adjustment of the stress of spring **19** is exceeded, the traveling speed of the press piston **3** is reduced to that value which is compatible with the maximum pressure set. It is apparent that a complex working cycle may be automatically controlled by an arrangement comprising the necessary number of cam-actuated switches and transmitting servo motors.

After completion of the work stroke, switch **33** is actuated by cam **41**. Closing of the normally open contact of switch **33** initiates the rapid return stroke of piston **3** by energizing solenoid **34** of slide valve **27** (FIG. 1). The slide valve is shifted towards the right and pressing fluid from gear pump **7** is admitted to the cylinder space **28**, exerting sufficient force on piston **20** to overcome the pressure of spring **19**. The hydraulic servo motor **5** therefore moves the slide block **1** into the extreme left position e_2 for reverse flow of fluid in the main hydraulic system. The speed of the return stroke of piston **3** is determined by the position of stop **6***a* which limits the leftward movement of pilot valve **6**, and thereby of slide block **1**. The return stroke is not controlled by the electric servo motor system. It is performed at maximum speed, that is at maximum delivery of the pump **2**. The position of the lever **17** and the setting of servo motors **9**, **9***a*, **9***b*, and **10** remains unchanged during the return stroke.

The position of the spring cap **30** can be changed, by means of the gear motor **38**, for adjusting the initial tension of spring **19**, and thereby the maximum working pressure of the main hydraulic system. It is understood that the cap **30** may also be adjusted by a system including an electric servo motor actuating a hydraulic servo motor, whereby a visual indication of the position of the

spring cap 30 may be had on the control board from the position of the rotor of the transmitting electric servo motor.

The wiring diagram of FIG. 3 shows the circuitry used in the device of the invention for electrical remote control of the initial tension of spring 19 by means of gear motor 38 and potentiometer 36. A bridge circuit 37 of a type known per se is employed between a master potentiometer 35 and a slave potentiometer 36 connected with spring cap 30.

When the master potentiometer 35 is set for a desired working pressure, a zero voltage relay 42, which is an element of the bridge circuit 37 between the slave potentiometer 36 and the master potentiometer 35 energizes contactor 44, which starts gear motor 38 to rotate in one direction. Let it be assumed, for example, that this be clockwise rotation and that it cause reduction of the initial stress of compression spring 19 which causes a decrease of the working pressure of the main hydraulic system. Movement of motor 38 continues until the slave potentiometer 36, which is moved by means of the gears 38*b*, reaches a position corresponding to that of transmitting potentiometer 35. The bridge circuit reaches equilibrium, and zero voltage relay 42 returns to its neutral center position. Contactor 44 is deenergized and the gear motor 38 stops. When it is desired to increase the working pressure, it is necessary to increase the stress of spring 19. This is done by adjusting the master potentiometer 35 in the opposite direction, whereby the zero voltage relay energizes the contactor 43 for counterclockwise rotation of gear motor 38 until the positions of the two potentiometers 35 and 36 are again brought into agreement and the bridge circuit is balanced.

While the control device of the invention has been illustrated by an embodiment controlling a variable-and-reversible displacement pump of the radial plunger type having a reaction rotor contained within a slide block, it will be understood by those skilled in the art that the device of the invention is equally applicable to variable displacement pumps of different types which require substantial forces for the shifting of their reaction rotors, whereby the pump delivery is adjusted.

Various modifications are contemplated and may obviously be resorted to by those skilled in the art without departing from the spirit and scope of the invention as hereinafter defined by the appended claims, as only a preferred embodiment thereof has been disclosed.

What is claimed is:

1. A variable delivery pump comprising, in combination, a coordinated fluid motor, conduit means between said pump and said motor for a fluid circulated by said pump, reactor ring means shiftable for the adjustment of delivery and direction of flow of said fluid in said conduit means, a control device including first hydraulic servo motor means having a first movable portion connected to said reactor ring means for shifting the position thereof, movable control means connected to said first hydraulic servo motor means and effective to move the latter and said reactor ring means an amount in proportion to movement of said movable control means, second hydraulic servo motor means having a second movable portion connected to said movable control means for movement of the latter in response to movement of said second movable portion, and supplemental control means acting at the connection of said second servo motor means and said movable control means, and including a sliding member, a predetermined biasing force means biased against one side of said member to urge said member in one direction, and means communicating the other side of said member with said conduit means to bias this side in the opposite direction by means of said fluid delivered by said pump.

2. A variable delivery pump according to claim 1 wherein said sliding member has at least two differential piston area portions at said other side, one portion being exposed to discharge pressure from said conduit means, further comprising constant pressure control fluid means, the other one of said differential piston area portions being adapted to be selectively exposed to the positive and negative pressure sides of said control fluid means.

3. A variable delivery pump according to claim 2 wherein said one differential piston area portion is smaller than said other portion, said discharge pressure applied to said one portion normally balancing the action of said biasing force means, while upon connecting said positive pressure side to said other portion said biasing force means is overcome, said sliding member is shifted in the direction of said one side and said supplemental control means may act by shifting said reactor ring means to reverse the direction of flow of said fluid delivered by said pump.

4. A variable delivery pump according to claim 1, further including remote control means for adjusting said predetermined biasing force means.

5. A variable delivery pump according to claim 1 wherein said predetermined biasing force means is a spring.

6. A variable delivery pump according to claim 5 wherein said remote control means includes electric motor means for adjusting the tension of said spring, master potentiometer means settable for a predetermined pressure of said spring, slave potentiometer means operatively coupled to said electric motor means for coordinated setting movement therewith, and bridge circuit means responsive to the relative settings of said master and said slave potentiometer means for controlling the movement of said electric motor means.

7. A variable delivery pump according to claim 4 wherein said remote control means further includes an electric servo system having transmitter means and receiver means, and a hydraulic servo motor connected to said predetermined biasing force means, said receiver means being adapted to actuate said hydraulic servo motor for adjusting said predetermined biasing force means.

8. A variable delivery pump according to claim 7 wherein said electric servo system includes a plurality of transmitter means, and control means operatively connected to said fluid motor for selectively and individually connecting said transmitter means to said receiver means.

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