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Tolbert

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[54] **PRESSURE COMPENSATING VALVE MECHANISM**

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[52] **U.S. Cl.** **137/117,** 91/446, 91/451, 137/596.13

[51] **Int. Cl.** **F15b 11/02**

[58] **Field of Search** 137/115, 116.3, 117, 137/522, 523, 596.13; 91/446, 451

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[57] **ABSTRACT**

A mechanism which, in one form, is useful as an unloading valve, and in another form as a pressure compensating valve. It comprises a valve plunger having a bypass position allowing all source fluid entering an inlet port to flow to an outlet port, and having a feed position compelling flow of source fluid to a feeder port in an amount depending upon the extent the plunger is displaced from its bypass position. The plunger is held in a feed position under substantially strong force exerted thereon by a primary spring and by pressure fluid from the feedback port. When the feedback port is vented, the plunger moves to its bypass position under force exerted thereon by pressure fluid from the inlet port, against the force of the primary spring diminished by the force of a secondary plunger spring.

13 Claims, 5 Drawing Figures

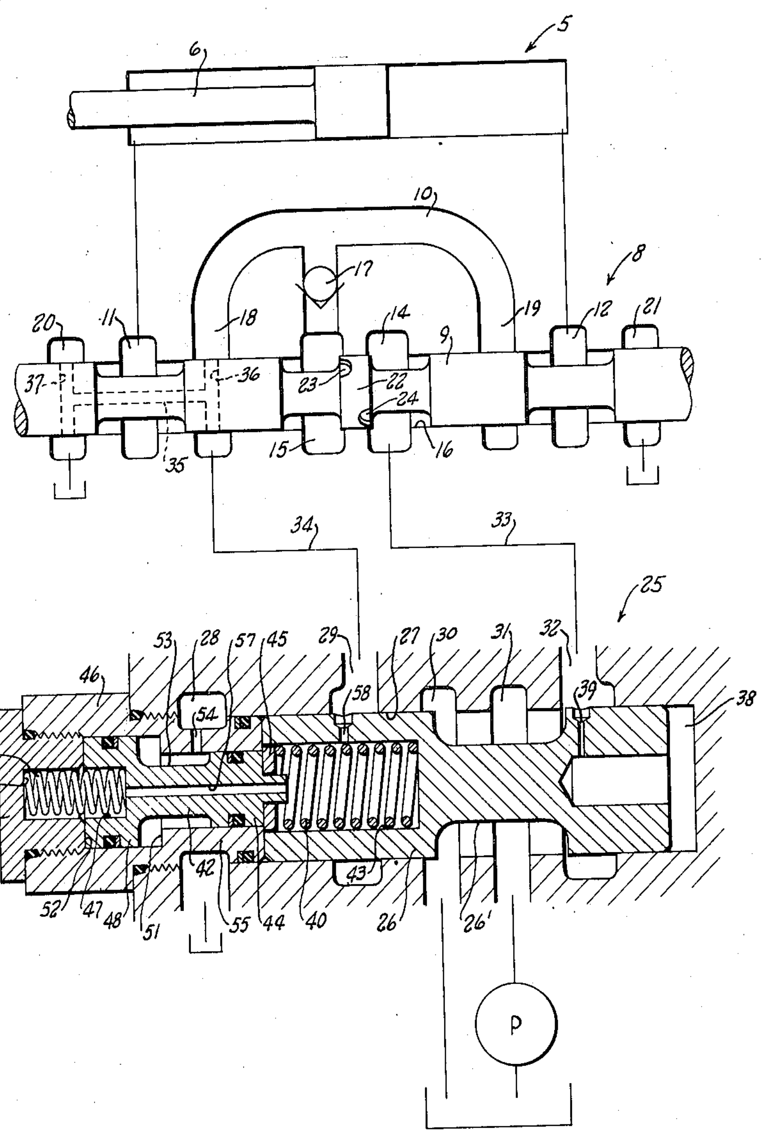

Fig. 1.

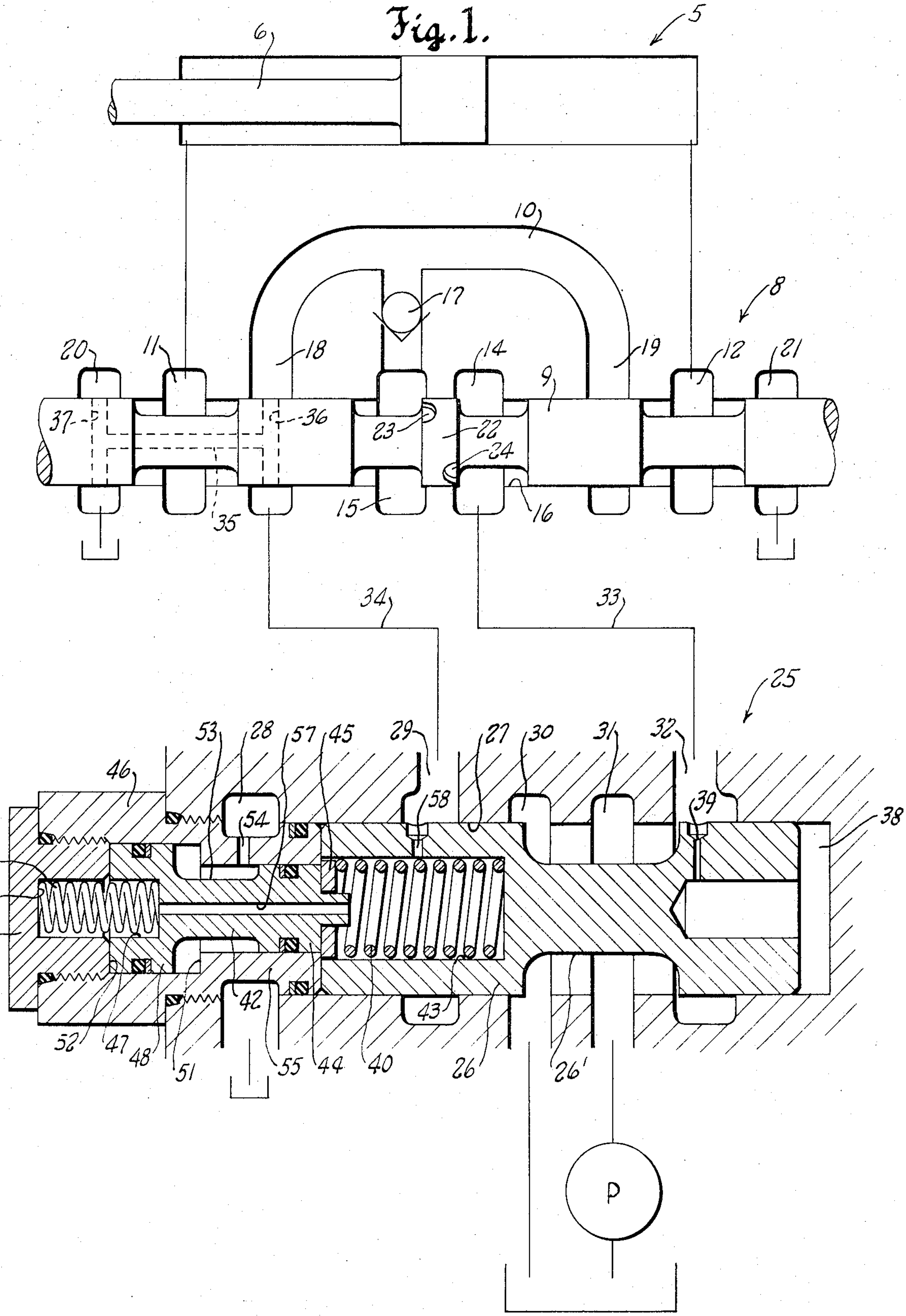
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Fig. 2.

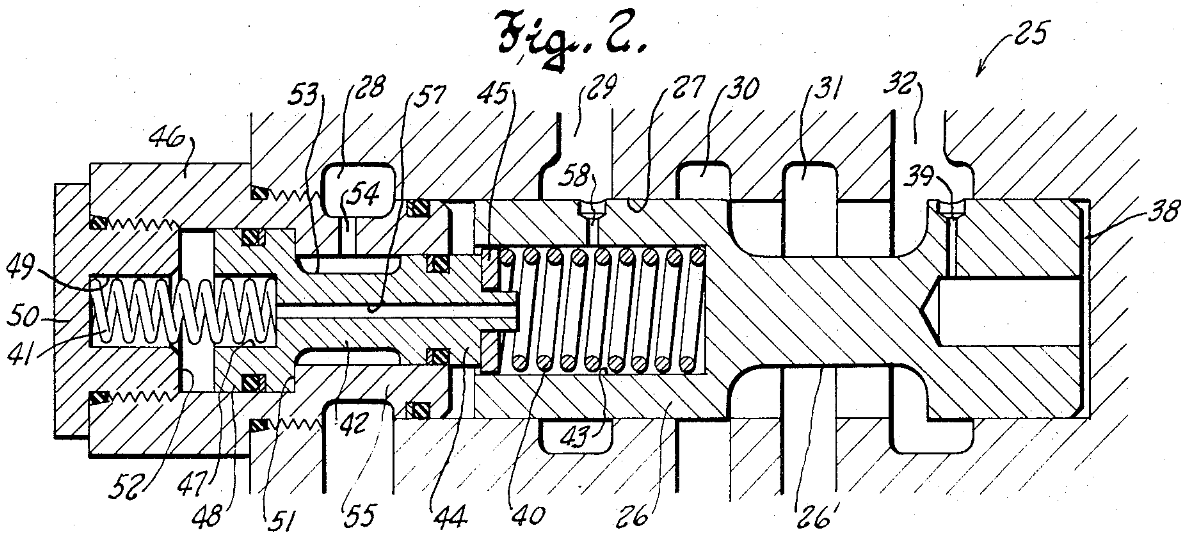
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Fig. 3.

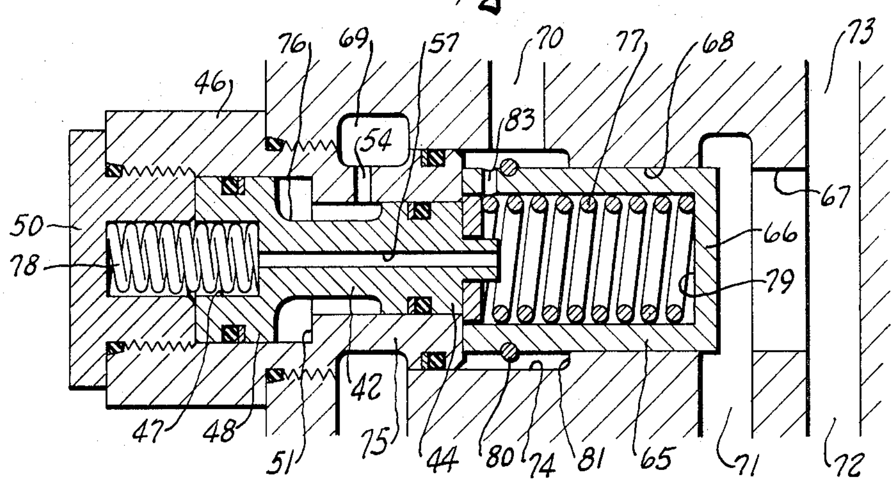
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Fig. 4.

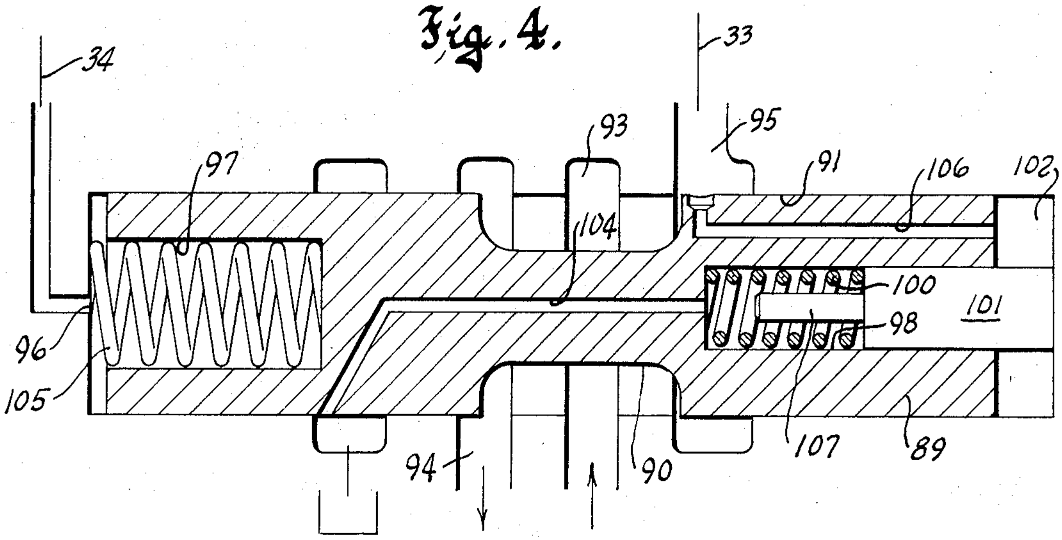
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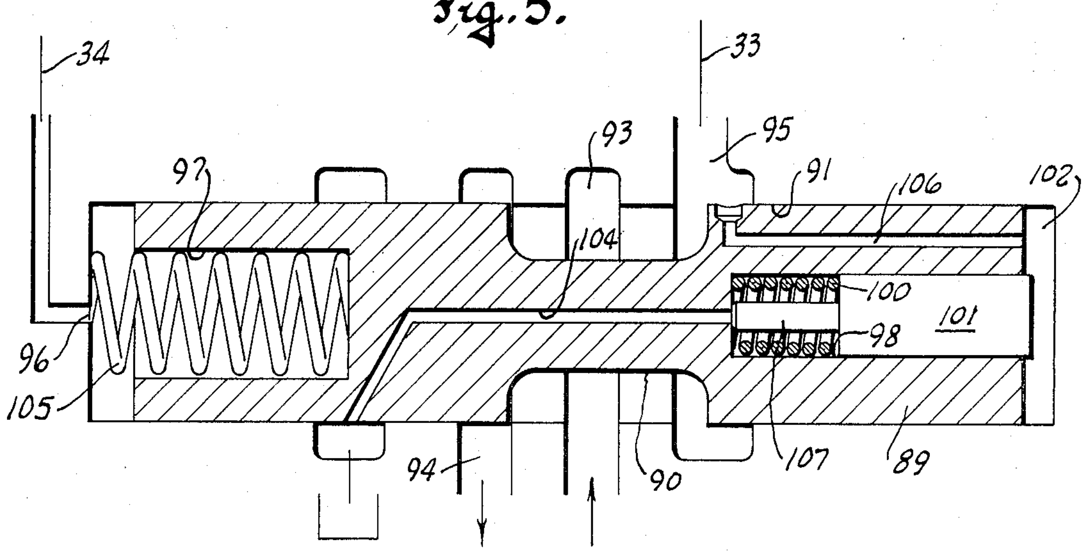
Fig. 5.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PRESSURE COMPENSATING VALVE MECHANISM

This invention relates to controls for fluid motors, and has more particular reference to bypass type pressure compensating valve mechanisms such as are used conjunctionally with a closed center control valve to govern the speed of a fluid motor.

In general, such pressure compensating valve mechanisms comprise a fluid pressure actuatable valve plunger or spool which cooperates with the valve element or main spool of the control valve to regulate flow of pump fluid to the inlet of the control valve in accordance with variations in the pressure differential between pump output fluid and that motor port of the control valve through which pump fluid is being supplied to the motor.

Ordinarily, when the spool of a closed center control valve is in a neutral or hold position closing off the motor ports from both supply and return passages, the pressure of pump output fluid cannot be dissipated through the control valve. At that time, pump output fluid acts upon one end of the plunger of the associated pressure compensating valve, against an opposing spring force acting upon the other end of the plunger, to hold the plunger in a bypass position at which all of the pump output fluid can be bypassed to an outlet port. However, when the main spool is in an operating position directing source fluid to one of the motor ports, fluid pressure corresponding to that which exists at said motor port is fed back to the pressure compensating valve mechanism and is imposed upon the other end of its plunger to act with the spring force thereon in holding the plunger in a feed position at which it maintains the drop or differential between pump output and motor port pressures at a constant value. That value, of course, is determined by the size of the variable flow restriction or orifice provided by the valve spool and across which source fluid flows to the selected motor port. Thus, bypass flow is reduced by the volume of pressure fluid allowed to flow across the orifice of the valve spool to the motor port in any given operating position of the spool.

It is customary to provide such pressure compensating valve plungers with a spring to strongly resist plunger movement out of its feed position during operation. As is well known in the art, precise control over the speed of the governed fluid motor is impossible unless this spring force is great enough to assure the desired quick response of the compensating plunger to the smallest variations in pump output and/or feedback pressure under all operating conditions.

While the importance of strongly biasing the pressure compensating plunger toward its feed position was widely recognized, this prerequisite nevertheless gave rise to a serious problem which until now has defied solution. As stated, the compensating plunger must be held in a bypass position at times when the control valve element is in its neutral or hold position. The purpose of this, of course, is to unload the pump just as an open center control valve does when its valve element is in a neutral position.

However, it was heretofore possible to only partly unload the pump in the neutral position of the control valve element. This was due to the fact that at that time the pump had to generate an output pressure of a magnitude which, when imposed upon the compensating

plunger, was capable of holding it in its bypass position against the force of the strong plunger spring. As a result, not only was considerable power wasted, but undesirable heating was an unavoidable consequence of such unproductive effort on the part of the pump.

With the foregoing in mind, it is a purpose of the present invention to provide a compensating valve mechanism having a fluid pressure actuatable plunger movable between feed and bypass positions, and in the latter of which positions the plunger can be held by an exceptionally low pump output pressure, while in the feed position of the plunger, a desirably strong spring force is exerted thereon to strongly resist return movement thereof toward its bypass position.

More specifically, it is an object of the invention to provide a pressure compensating valve mechanism wherein a substantially strong primary spring bearing upon the plunger urges it toward its feed position, and a weaker secondary spring connected with the plunger normally acts to diminish the force of the primary spring and thus assure a low pressure drop across the inlet and outlet ports of the mechanism in the bypass position of the plunger.

In this respect, it is a further object of the invention to provide a valve mechanism such as described above where the primary and secondary springs normally act in series upon the plunger to only lightly resist motion thereof toward its bypass position under force of inlet fluid thereon, and wherein the force of feedback fluid is utilized to render the secondary spring ineffective so that the full force of the primary spring plus that of feedback fluid acting on the plunger is then made available to resist movement of the plunger out of its feed position.

Another object of the invention is to adapt the series-connected spring concept mentioned in the preceding paragraph to bypass valve mechanisms per se, as distinguished from pressure compensating valve mechanisms.

In a preferred embodiment of the invention, a piston is confined between the primary and secondary springs and is actuatable in a direction to relax the secondary plunger spring in consequence of subjection of the piston to feedback pressure from the governed motor.

With these observations and objectives in mind, the manner in which the invention achieves its purpose will be appreciated from the following description and the accompanying drawings, which exemplify the invention, it being understood that changes may be made in the specific apparatus disclosed herein without departing from the essentials of the invention set forth in the appended claims.

The accompanying drawings illustrate several complete examples of embodiments of the invention constructed according to the best modes so far devised for the practical applications of the principles thereof, and in which:

FIG. 1 is a diagrammatic view illustrating a fluid pressure operated system embodying a pressure compensating valve mechanism of this invention, showing the plunger thereof in its bypass position;

FIG. 2 is a view of the pressure compensating valve mechanism of FIG. 1, but showing the plunger thereof in a feed position;

FIG. 3 is a diagrammatic view of a pump unloading valve embodying this invention; and

FIGS. 4 and 5 are diagrammatic views of a pressure compensating valve mechanism of modified construction.

Referring now to the accompanying drawings, and particularly to FIGS. 1 and 2 thereof, the numeral 5 designates a reversible fluid motor, here shown as a double acting hydraulic cylinder having a piston rod 6 by which the cylinder can be operatively connected to a load (not shown). The cylinder is supplied with pressure fluid from a pump P, at the dictate of a control valve 8. The control valve has been shown by way of example as a single spool valve of a conventional type. Its valve spool 9 is shiftable axially to working positions at opposite sides of a neutral position (shown), to direct output fluid from the pump through a supply passage 10 to one or the other of a pair of motor ports 11, 12 connecting with the opposite sides of the cylinder.

The control valve is of the closed center type having feeder passage means comprising an upstream branch 14 that can be considered as the inlet of the valve, and a downstream branch 15 which is communicable with the upstream branch 14 through the bore 16 in which the valve spool operates. The downstream branch 15 of the feeder passage is communicated with the supply passage 10 through a load holding check valve 17.

The supply passage is of inverted U-shape to provide branches 18 and 19 which intersect the bore at locations inwardly adjacent to the junctions of the bore with the service passages 11 and 12, respectively. The valve spool is grooved to provide lands which control communication of the motor ports 11, 12 with either the supply passage branches 18, 19 or with exhaust passages 20, 21 which intersect the bore 16 at locations outwardly adjacent to the junctions of the bore with the motor ports 11 and 12, respectively.

A central land 22 on the spool is situated to close off communication between the feeder passage branches 15, 16 in the neutral or hold position of the valve spool shown. Throttle notches 23 and 24 in the left and right hand ends respectively, of the land 22 provide for adjusting the rate at which pressure fluid flows from the upstream feeder branch 14 to the downstream feeder branch 15, upon shifting of the valve spool in either direction out of neutral to a flow metering position short of a full operating position of the spool.

In a right hand flow metering position of the valve spool 9, pressure fluid entering the inlet 14 flows through throttle groove 23 and to the rod end of cylinder 5 via the feeder passage branch 15, check valve 17 and branch 18 of the supply passage 10 then in communication with service passage 11. In a left hand flow metering position of the spool, pressure fluid entering the inlet 14 flows through throttle notch 24 and to the head end of the cylinder via feeder branch 15, check valve 17 and supply branch 19 then in communication with the service passage 12. In each case, the non-selected service passage is communicated with its adjacent exhaust passage branch 20 or 21 to conduct to the reservoir of the system pressure fluid which is expelled from the cylinder 5.

The pressure compensating valve mechanism 25 of this invention is connected in the system between the pump P and the control valve 8, and its purpose is to maintain the flow of pressure fluid to the cylinder at a constant rate determined by the metering position or setting of the control spool 9. As is customary, the pressure compensating valve is provided with a fluid pressure actuatable plunger 26 which is sensitive to the pressure drop across the orifice provided by either throttle notch 23, 24, and which regulates the flow of pump fluid to the inlet 14 in accordance with variations in said pressure drop from that value thereof which exists when fluid flows to the motor at the desired rate.

The plunger 26 of the pressure compensating valve mechanism has hollow opposite end portions, and it is slidably received in a bore 27 which is closed at each end. Five different passages open to the bore at axially spaced zones. Reading from left to right, these passages comprise a low pressure or return port 28, a feedback port 29, an outlet port 30, an inlet port 31, and a feeder port 32.

The return port 28 can comprise part of the exhaust passage 20; the feeder port 32 is communicated with the upstream feeder branch 14 by means of a duct 33; and the feedback port 29 can be communicated with either motor port 11, 12 through a shuttle valve in a conventional way. In the single spool control valve shown, however, the feedback port 29 is communicated with the branch 18 of the supply passage 10 by means of a duct 34; or it can be connected to feeder branch 15 with the same results. The feedback port 29 is vented to the reservoir of the system when the control valve spool 9 is in its neutral position by means of an axial passage 35 in the valve spool which communicates at one end with the supply passage branch 18 through a radial passage 36 in the spool, and with the exhaust branch 20 at its other end through a radial passage 37 in the spool.

In the neutral position of the valve spool 9, seen in FIG. 1, the land 22 on the spool closes off communication between the feeder branches 14 and 15. Pump output fluid then entering the inlet port 31 of the compensating valve acts upon the right hand end of the plunger 26 to hold the same in a bypass position at which all of the incoming pump fluid flows to the outlet port via a circumferential groove 26′ in the plunger communicating ports 30 and 31. For this purpose, the pressure chamber 38 provided by the right hand end of the bore 27 is communicated with the inlet port 31, as by a radial hole 39 in the hollow end portion of the plunger. The hole 39 is at all times in direct communication with the feeder port 32, and consequently also with the inlet port 31.

When the spool 9 of the control valve is moved to a full operating position at either side of neutral, it provides unrestricted communication between the feeder branches 14 and 15, and the pressure of fluid in the supply passage 10 will be at substantially the same valve as that of pressure fluid in whichever motor port 11 or 12 is then in communication with it. As is customary, this pressure is fed back to the compensating valve mechanism and imposed upon the left hand end of the compensating plunger 26 in opposition to the force which pump output fluid in pressure chamber 38 exerts upon the right hand end of the plunger. A spring also customarily urges the plunger toward its feed position.

According to this invention, the spring bias for the plunger is provided by a pair of compression springs, namely, a substantially strong primary spring 40 and a weaker secondary or auxiliary spring 41. Both of these springs are located at the left hand end of the plunger, at opposite axial ends of a stepped piston 42. The primary spring is confined in the well 43 provided by the hollow left hand end portion of the plunger, between the bottom of the well and the adjacent inner end portion 44 of the piston, which is smaller in diameter than the well. A washer 45 is preferably interposed between the piston and the primary spring. The piston is axially slidable in a stepped cylinder 46 which is coaxial with the bore 27, and the small diameter inner end portion of which cylinder opens to the bore.

One end portion of the secondary spring extends into a well 47 in the larger diameter outer end portion 48 of the piston, and the other end of the spring extends into a well 49 in a cap 50 which closes the large diameter outer end portion of the cylinder. The spring 41, of course, is confined between the bottoms of the wells 47 and 49.

The large diameter end portion of the cylinder 46 has greater axial length than that of the piston portion therein, so as to provide a stop 51 which cooperates with the inner end 52 of the cap 50 to define the limits of axial sliding motion of the piston.

It should here be observed that the piston is circumferentially reduced between its large and small diameter ends, as at 53. The space thus provided between the large diameter end of the piston and the inner stop 51 is vented through a hole 54 in the wall of the cartridge 55 in which the piston is housed. The inner end of this cartridge also provides a stop to define the neutral or full bypass position of the compensating plunger 26.

It is important to note that the piston is formed with a passage 57 which extends axially therethrough, so that the fluid pressures in the spring chambers at the opposite ends of the piston 42 can be equalized.

In the neutral or hold position of the control valve spool 9 seen in FIG. 1, the compensating plunger occupies a bypass position at which all of the pump output fluid entering its inlet port 31 flows to the outlet port 30, in bypass relation to the feeder port 32. This condition is brought about by reason of the fact that the primary and secondary spring chambers 43 and 47 are vented to the exhaust passage 20 in the control valve through a hole 58 in the side wall of the chamber 43, which hole opens to the feedback port 29 and is always in communication therewith. In this connection, it should be recalled that the feedback port 29 is communicated with the exhaust passage 20 via the duct 34, branch 18 of the supply passage, and the passages 35, 36 and 37 in the control spool 9 in the neutral position of the latter.

When the spring chambers are vented in this fashion, the pressure of pump output fluid at the inlet port 31, acting upon the right hand end of the compensating plunger 26 in chamber 38, is easily able to hold the plunger in its bypass position against the action of the primary and secondary plunger springs 40 and 41, respectively. It should be appreciated at this point that the primary and secondary springs can be said to act in opposition to one another. The total force which they exert upon the plunger tending to move it out of its bypass position is actually far less than the force which the primary spring alone would be capable of exerting upon the plunger. In fact, because the primary and secondary springs 40 and 41 are connected in series with the plunger, the total force which they exert thereon is computed in the same way as the resistance of electrical resistors in parallel. Thus, if the ratings of the primary and secondary springs are represented by R1 and R2, respectively, the total force RT can be found using the formula

$$RT = (R1 \times R2/R1 + R2)$$

Merely by way of example, if $R1$ equals 100 psi and $R2$ equals 50 psi, the total force RT which can be exerted on the plunger is 5000 psi divided by 150 psi, or only slightly over 33 psi. Thus it will be seen that the pressure drop across the inlet and outlet ports 31 and 30 will be held to a very desirable low value whenever the control valve spool is in its neutral position and the compensating plunger 26 is in a corresponding neutral or bypass position such as seen in FIG. 1.

The spring bias acting upon the compensating plunger becomes much greater, as is essential, whenever the control spool 9 is actuated to either working position at which fluid at high pressure flows from the supply passage 10 to the selected motor port 11 or 12. Assuming that the control spool is moved to the left, pump fluid will then flow from the feeder passage 14 to motor port 12 via throttle groove 24, feeder branch 15, check valve 17, and branch 19 of the supply passage. A metered amount of pump fluid then flows into the head end of cylinder 5 to effect extension of its piston rod 6.

The pressure of fluid then in the cylinder head will be transmitted to spring chambers 43 and 47 via supply passage 10, duct 34, feedback port 29, hole 58 in the compensating plunger, and axial passage 57 in piston 42.

Feedback fluid then flows into chamber 47 and moves the piston to the right to its limit of motion seen in FIG. 2, so as to then relax the secondary spring 41. When that occurs, the secondary spring 41 becomes ineffective, and the full force of the primary spring is then exerted on the compensating plunger, without opposition from the secondary spring. The position of the compensating plunger, in all operating positions of the control valve spool, will then depend upon pump output pressure as imposed upon the right hand end of the plunger, and upon the combined forces of feedback fluid and of the primary spring 40 acting upon the left hand end of the plunger.

At any given setting of the main valve spool, the position of the compensating plunger will be automatically adjusted to effect regulation of fluid pressure at the feeder port 32 in accordance with variations in pump output pressure at port 31 and/or in response to variations in feedback pressure; and the plunger will be moved in response to such variations in pressure in the direction to compensate for the changed condition and thereby maintain the pressure at port 32 at a constant value. That is to say, for example, that the plunger will be caused to move to the right to increase fluid pressure at the feeder port 32 in consequence of a rise in pressure at the feedback port 29; and it will be caused to move to the left to decrease the feeder port pressure as a consequence of a decrease in pressure at the feedback port 29.

It will be seen, therefore, that in the single spool control valve illustrated, the compensating plunger functions to maintain constant pressure at the feeder port 32 by varying the degree of communication between the pump port 31 and the outlet port 30. In a plural spool valve, however, the pressure compensating plunger will function to maintain feeder port pressure constant by varying the degree of communication between the feeder port and the pump port 31 at times when port 30 is in use as a high pressure carry-over port for a downstream control valve and the spool of the latter is operating a fluid motor at a greater pressure than that being operated by the upstream spool.

Actuation of the control spool 9 to a full flow position, of course, will effect complete close-off of the outlet port **30** from the inlet port **31,** so that all pump fluid will then be compelled to flow to the work cylinder.

As soon as the control spool **9** is returned to its neutral position, of course, the feeder passage **14** is blocked and the feedback port **29** is vented. As a result, pump output fluid flows into chamber **38** at the right hand end of the plunger and promptly moves the same to its bypass position at which the secondary spring **41** is again effective to oppose the action of the primary spring **40.**

From the above, it will be seen that the spring force tending to move the compensating plunger out of its bypass position is desirably low, while a much greater spring force resists movement of the plunger out of its feed position at times when pressure fluid is being directed to one end or the other of the cylinder **5.** This last assures the desired precise control over the speed at which the cylinder operates.

FIG. 3 illustrates how the series spring concept described above can be used to advantage in a pump unloading valve. The unloading valve is similar in most respects to the pressure compensating valve described earlier, although it can be made somewhat simpler. Thus, its plunger **65** comprises a cup shaped member comparable to the large diameter end portion of the compensating plunger **26** at the left hand end thereof, and having its closed end **66** movable into and out of engagement with a seat **67** provided by a short portion of the bore **68** in which the valve plunger is axially slidably received. Reading from left to right, the unloading valve is also provided with a reservoir port **69,** a feedback port **70,** an outlet port **71,** an inlet port **72,** and a feeder port **73.** While the inlet and outlet ports open to the bore **68** at axially opposite sides of the bore portion **67,** the feeder port **73** can be directly communicated with the inlet port **72,** to form a part thereof, as shown.

The reservoir port **69** opens to a counterbore **74** in which is secured the cartridge **75** containing the stepped piston **76.** The piston, of course, is slidable axially in a stepped cylinder provided in the interior of the cartridge, as before; and the primary and secondary plunger springs **77** and **78,** respectively, are again located at axially opposite ends of the piston, with the primary spring extending into the hollow interior **79** of the valve plunger **65.**

In the present case, the plunger **65** is shown as extending outwardly of the bore **68** into the counterbore **74,** to engage the inner end of the cartridge **75,** which thus defines the fully open or bypass position of the plunger. A snap ring **80** confined in a groove in the exterior of the plunger is engageable with the bottom **81** of the counterbore to define the closed position of the plunger at which its right hand end is received within the seat defining bore portion **67** to block communication between the inlet port **72** and the outlet port **71.**

As before, a hole **83** in the side wall of the plunger **65** communicates with the feedback port **70** and provides for entry of feedback fluid into the interior of the plunger to effect pressurization of the chambers at the opposite ends of the piston **76.**

The pump unloading valve operates in substantially the same way as the pressure compensating valve. When the spool of the associated control valve is in neutral position, the feedback port **70** is vented and the feeder passage is closed off. Accordingly, pump output fluid entering the inlet port **72** acts upon the closed end **66** of the plunger and moves the same to its fully open position shown, against the substantially light resistance of the serially connected primary and secondary springs **77** and **78.** All of the pump fluid entering port **72** then flows through outlet port **71** and back to the reservoir, with but slight pressure drop between ports **71** and **72.**

As soon as the spool of the associated control valve is shifted to an operating position allowing pump output fluid to flow through the feeder passage to the selected motor port, the pressure of fluid at the latter port is transmitted to the port **70** of the unloading valve to effect pressurization of both spring chambers, at the opposite ends of the piston **76.** Pressure fluid then flowing into the secondary spring chamber moves piston **76** toward the plunger and away from the secondary spring **78** so as to relax the secondary spring **78,** while pressure fluid in the primary spring chamber acts directly upon the plunger in the direction to move it toward its closed position. The plunger may then move into the bore portion **67** under this combination of forces acting thereon, to close off the outlet port **71** from the inlet port **72** and thereby constrain pump output fluid entering the inlet port to flow through the feeder port for passage to the controlled cylinder.

It is also possible for the valve plunger **65** to occupy a partially closed position allowing some of the pump output fluid to flow to the outlet port and constraining the remainder to flow to the feeder port. Such a situation can arise from placement of the control valve spool in a metering position on the order of that described earlier, so that the plunger **65** would then respond to variations in the pressure drop across the orifice provided by the throttle groove through which the work cylinder was supplied with pump output fluid.

It is also believed to be obvious to those skilled in the art that the feedback port **70** of the unloading valve could be communicated with the motor ports of the associated control valve through a shuttle valve arrangement in a more or less conventional way.

FIG. 4 illustrates another version of the pressure compensating valve mechanism of this invention. Its plunger **89,** shown in bypass position, is again provided with a circumferential groove **90,** and it is reciprocable endwise in a bore **91** whose opposite ends are closed.

In the pump unloading position shown, the plunger groove communicates a pump inlet port **93** with an outlet port **94** to cause pump output fluid to bypass a feeder port **95.** The plunger occupies this unloading or bypass position when the spool of the associated control valve is in its neutral position closing off flow therethrough of pump output fluid from the feeder port and venting the left hand end of the bore **91** through a feedback port **96** therein.

In the feed position of the plunger **89,** its left hand land can limit or even close off flow of pump output fluid to the outlet port from the inlet port to thereby divert a regulatable volume of pump output fluid to the feeder port **95** through the groove **90.** The plunger moves to its feed position in consequence of actuation of the spool of the associated control valve to an operating position allowing for flow of pump output fluid therethrough from the feeder port and pressurizing the feedback port **96** in the manner described earlier.

Coaxial wells **97** and **98** provide spring chambers in the opposite ends of the plunger. A secondary spring **100** situated in the well **98** acts upon a piston **101** in said well to urge the piston toward one limit of motion at which its outer end abuts the adjacent end of the bore **91.** In that limit of motion, the piston may hold the secondary spring lightly loaded. The piston **101** can be moved inwardly into the well **98** by the pressure of fluid in the chamber **102** provided by the right hand end portion of the bore to further compress the secondary spring. It is for this reason that the well **98** is at all times vented to a reservoir port **103,** through a passageway **104** in the compensating plunger.

The secondary spring **100** tends to move the compensating plunger **89** to the left, toward its bypass or pump unloading position. It acts upon the plunger in opposition to a primary spring **105** which is situated in the well **97** in the outer end of the compensating plunger **89** and tends to move the same toward its feed position. While the secondary spring thus opposes the primary spring, the latter is slightly stronger and tends to normally move the plunger toward the right hand end of the bore **91,** to its feed position.

The pressure chamber **102** in the right hand end of the bore **91** is at all times communicated with the feeder port **95,** and hence with the inlet port **93,** via a passageway **106** in the compensating plunger.

The piston **101** is provided with a stem **107** which projects coaxially into the secondary spring and toward the bottom of the well **98,** with which it can engage to define the inner limit of motion of the piston relative to the compensating plunger.

In operation, the compensating plunger will be normally held in its bypass or pump unloading position as long as the spool of the associated control valve remains in its neutral position venting the feedback port **96** and blocking flow of pump output fluid through the feeder port **95.** At that time, pump output fluid flows directly to the outlet port **94** in bypass relation to the feeder port, and its pressure is manifested in the chamber **102** at the right hand end of the compensating plunger. The substantially small force which pump output fluid is able to exert upon the right hand end of the plunger, when added to the greater force of the secondary spring **100,** produces a total force in excess of that exerted upon the plunger by the primary spring **105.** Hence, the plunger is held in its bypass position mainly by reason of the force of the secondary spring, which greatly counteracts the force of the primary spring.

Thus it will be seen that again the force of the primary spring is greatly diminished in the bypass position of the plunger, and that the pressure drop between the inlet port **93** and outlet port **94** will be held at a desirably low value.

As soon as the spool of the associated control valve is actuated to an operating position, feedback fluid from the supply passage flows into the port **96** and the compensating plunger is moved to the right thereby to limit or even close off fluid flow to the outlet port **94,** and to open up the feeder port **95** for the flow of pump fluid therethrough to the selected motor port of the associated control valve.

As the pressure rises in feeder passage **95,** as it must in order to overcome the load on the governed work cylinder, the pressure in chamber **102** rises accordingly and forces piston **101** to its inner extreme of motion at which its stem engages the bottom of the well **98.** The piston then can be said to become part of the plunger, and the effect of the primary spring is nullified. In the feed position of the compensating plunger, seen in FIG. **5,** therefore, the primary spring acts upon the plunger without opposition from the secondary spring and provides the desirably strong bias essential to precise control over the rate at which pump output fluid flows to the feeder port **95.**

As before, of course, the compensating plunger will respond to variations in the pressure differential between fluid at the inlet port **93** and feedback fluid at the feedback port **96,** and effect regulation of fluid flow to the feeder port in accordance with such variations.

From the foregoing description, together with the accompanying drawings, it will be appreciated that this invention makes possible the provision of a pressure compensating valve having a plunger which can be maintained in a bypass position under relatively light force, and in which a substantially strong spring force is made available to resist movement of the plunger out of its feed position.

Those skilled in the art will appreciate that the invention can be embodied in forms other than as herein disclosed for purposes of illustration.

The invention is defined by the following claims:

1. A fluid flow controlling valve mechanism having a body with an inlet and an outlet and feeder and feedback ports all opening to a bore containing a fluid pressure responsive valve plunger, means to impose the pressure of fluid at the inlet upon a first surface of the plunger to urge it in one axial direction toward a normal position at which pressure fluid entering the bore from the inlet flows to the outlet in bypass relation to the feeder port, and means by which the pressure of fluid at the feedback port can be imposed upon a second surface of the plunger to move it in the opposite axial direction toward a feed position compelling inlet fluid to flow to the feeder port, characterized by:

A. a strong primary spring connected with the plunger to resist movement thereof in said one axial direction, out of its feed position;

B. a weaker secondary spring for the plunger;

C. and means connected with the secondary spring and rendered operative by the plunger in the normal position of the latter for effecting imposition of secondary spring force upon the plunger in opposition to the force of the primary spring to thus enable the plunger to be held in its normal position by inlet fluid at substantially low pressure.

2. The fluid flow controlling valve mechanism of claim **1** further characterized by:

A. said means which is connected with the secondary spring comprising a piston which is movable in one direction from one position to another position at which the secondary spring is ineffective to oppose the primary spring;

B. and means for effecting motion of the piston to said other position thereof in consequence of rise in fluid pressure at one of said ports.

3. The fluid flow controlling valve mechanism of claim **1,** wherein the piston is confined between the primary and secondary springs and can be held in said one position thereof by the primary spring.

4. The fluid flow controlling valve mechanism of claim 3, wherein pressure of fluid at the feedback port acts upon the piston to move the same to said other position thereof at which it effects relaxation of the secondary spring.

5. The fluid flow controlling valve mechanism of claim 1, further characterized by:

A. said means which is connected with the secondary spring comprising a piston confined between said springs and movable axially toward said second surface on the plunger to relax the secondary spring;

B. and means for effecting movement of the piston axially toward said second surface on the plunger in consequence of rise in pressure at the feedback port.

6. The fluid flow controlling valve mechanism of claim 5, further characterized by:

A. the secondary spring being confined in the bottom portion of a cylinder in which said piston operates;

B. and means providing a passageway communicating the bottom portion of said cylinder with the feedback port.

7. The fluid flow controlling valve mechanism of claim 6, further characterized by:

A. the plunger having a well in its end adjacent to said piston, in which the primary spring is received;

B. the feedback port opening to said well;

C. and the piston having an axial passage therethrough communicating said well with the bottom of the cylinder.

8. The fluid flow controlling valve mechanism of claim 7, further characterized by:

A. the opposite ends of the piston being subjected to the pressure of feedback fluid in said well and in the bottom portion of the cylinder;

B. and the cylinder and piston having correspondingly larger diameter portions at the end of the piston remote from the plunger.

9. The fluid flow controlling valve mechanism of claim 1, further characterized by:

A. said springs being connected in series with the plunger whereby the force which they exert thereon in its normal position is equal to the product of their ratings divided by the sum thereof;

B. and said means which is connected with the secondary spring being operable in the feed position of the plunger, in response to pressure of fluid at said feed back port, for substantially nullifying the effect of said secondary spring upon the plunger.

10. A pressure compensating valve mechanism having inlet, outlet, feeder and feedback ports all opening to a bore containing a fluid pressure responsive valve plunger, said mechanism being of the type having provision for delivery of inlet fluid to one end of the bore where it can exert force on the plunger and normally hold it in a neutral position at which inlet fluid can flow to the outlet in bypass relation to the feeder port, and wherein the plunger is movable to an operating position limiting said bypass flow and diverting inlet fluid to the feeder port, characterized by:

A. means opening to the other end of the bore providing a cylinder coaxial with the bore;

B. a piston movable axially in the cylinder;

C. a primary compression spring reacting between the piston and the plunger and operable in the neutral position of the latter to hold the piston in engagement with an abutment in the outer end of the cylinder;

D. a secondary compression spring confined in said outer end of the cylinder and normally held in a loaded condition by the piston under the force which the primary spring exerts thereon in said neutral position of the plunger, whereby the force which the primary spring exerts upon the plunger tending to move it toward its operating position is substantially diminished except at times when the secondary spring is relaxed;

E. and means communicating said outer end of the cylinder with the feedback port so that the force which feedback fluid exerts upon the piston will move the same in the direction to relax the secondary spring.

11. A pressure compensating valve mechanism having a plunger movable axially in a bore in the body of the mechanism and having means to impose the pressure of fluid at an inlet upon a first surface of the plunger to urge it axially in one direction in the bore toward a normal position at which fluid entering the bore from the inlet flows to an outlet in bypass relation to a feeder port, and having a feedback port by which the pressure of feedback fluid can be imposed upon a second surface of the plunger to move it in the opposite axial direction toward a feed position compelling inlet fluid to flow to the feeder port, characterized by:

A. a primary plunger spring connected with the plunger to substantially strongly resist movement thereof in said one direction out of its feed position;

B. a secondary plunger spring which is weaker than the primary plunger spring;

C. means defining a cylinder;

D. a piston in the cylinder connected with the secondary spring and operable in said normal position of the plunger to render the secondary spring effective to impose force upon the plunger in opposition to the force of the primary spring providing the piston is in one axial position at which it engages an abutment on the body, said piston being movable away from said abutment to a second position at which it renders the secondary spring ineffective;

E. and means for effecting movement of the piston to its said second position in consequence of rise in fluid pressure at one of said ports.

12. The pressure compensating valve mechanism of claim 11 further characterized by:

A. said primary and secondary springs being connected with the plunger at opposite ends thereof;

B. and said piston being movable to its said second position in consequence of rise in fluid pressure at the feeder port.

13. A pressure compensating valve mechanism having a plunger operable in a feed position to regulate the rate at which pump fluid flows from an inlet to a feeder port in accordance with variations in the pressure differential between the feeder port and a feedback port, and wherein the plunger is held in a normal position displaced from its feed position under force which inlet fluid exerts upon one surface thereof to allow inlet fluid to flow to an outlet in bypass relation to the feeder port, characterized by:

A. a strong primary plunger spring;

B. a weaker secondary plunger spring;

C. means connecting the primary spring with the plunger so that the primary spring will substantially strongly resist movement of the plunger out of its feed position;

D. means connecting the secondary spring with the plunger;

E. said last named means being rendered operative by the plunger, in said normal position thereof, to impose the force of the secondary spring upon the plunger in offsetting relation to the force of the primary spring;

F. and said last named means being rendered operative in consequence of fluid pressure at one of said ports, in the feed position of the plunger, to nullify said force offsetting effect of the secondary spring upon the plunger.

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