United States Patent

Wilke

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[54] **FLUID FLOW CONTROLLING DEVICE FOR REVERSIBLE FLUID MOTORS**

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[52] **U.S. Cl.** **137/596.13**
[51] **Int. Cl.** **F16k 11/07**
[58] **Field of Search** 137/625.66, 625.6, 625.63, 137/596.12, 596.2, 625.26, 596.14, 596.66, 87, 100, 106; 91/420, 461

[56] **References Cited**

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[57] **ABSTRACT**

A fluid flow controlling device having independent service passages extending therethrough which provide portions of the service lines leading to a reversible fluid motor from the control valve governing the direction of motor operation. A single fluid pressure actuatable valve element in the device functions as a counterbalance valve and acts to restrict return flow of motor exhaust fluid through either service passage in consequence of decrease in the pressure of source fluid flowing through the other service passage to the motor. Check valves can be located in portions of the service passages so as to be bypassed by return flow of motor exhaust fluid.

9 Claims, 6 Drawing Figures

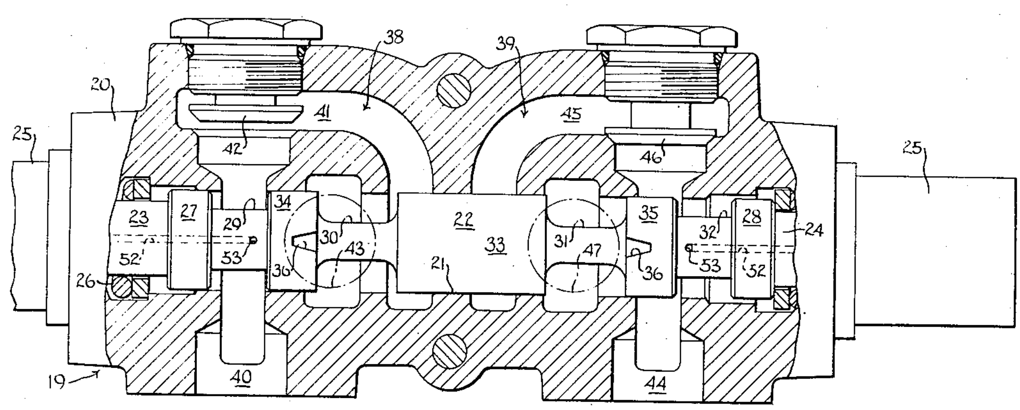

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FIG.1.

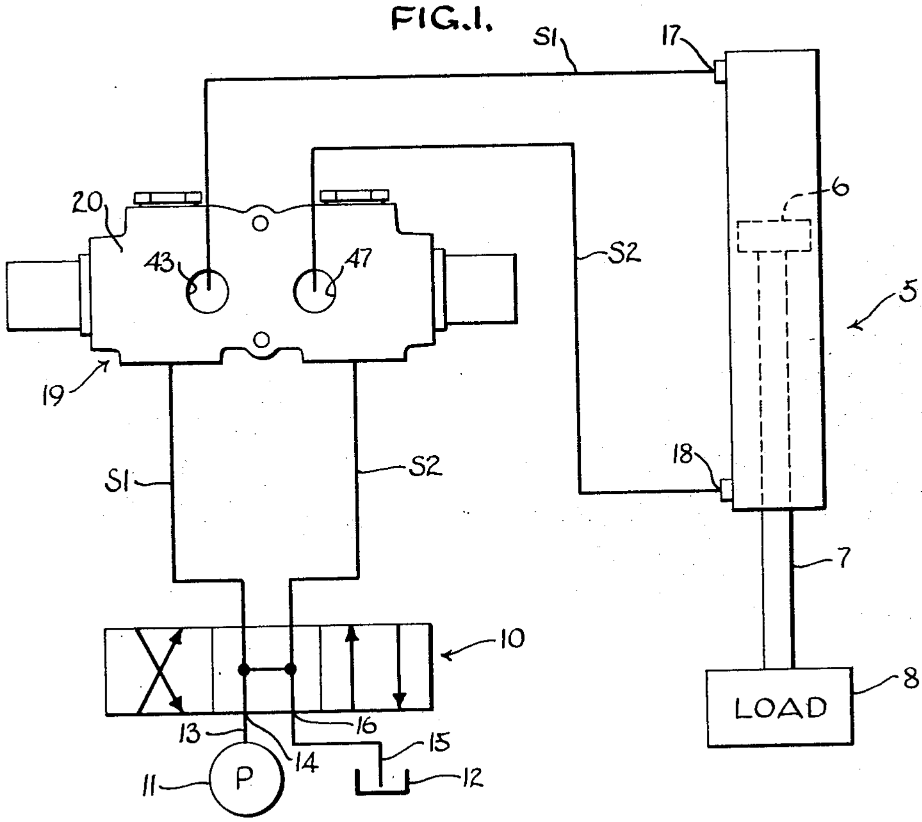
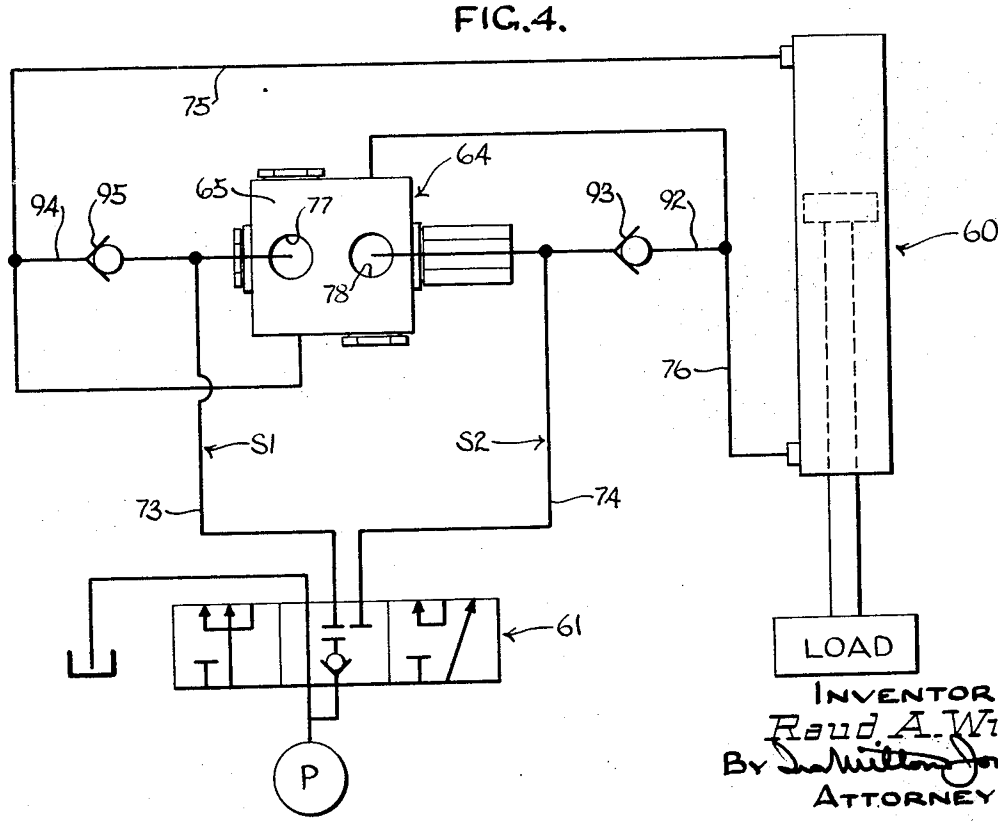


FIG.4.



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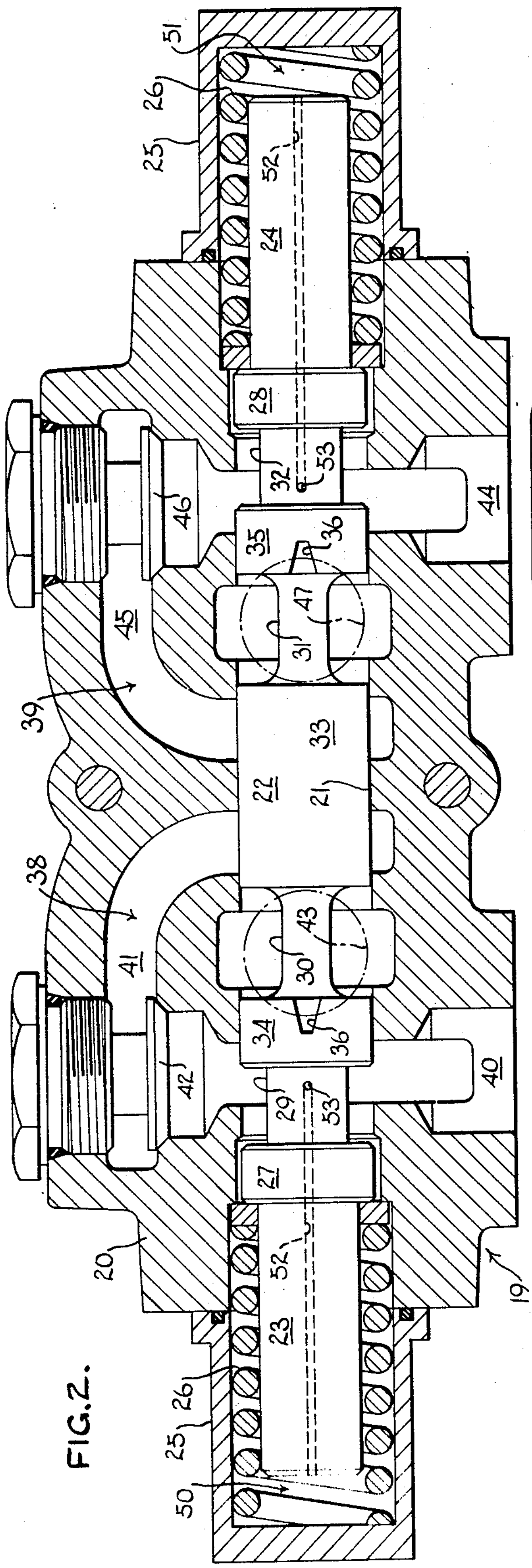

FIG. 2.

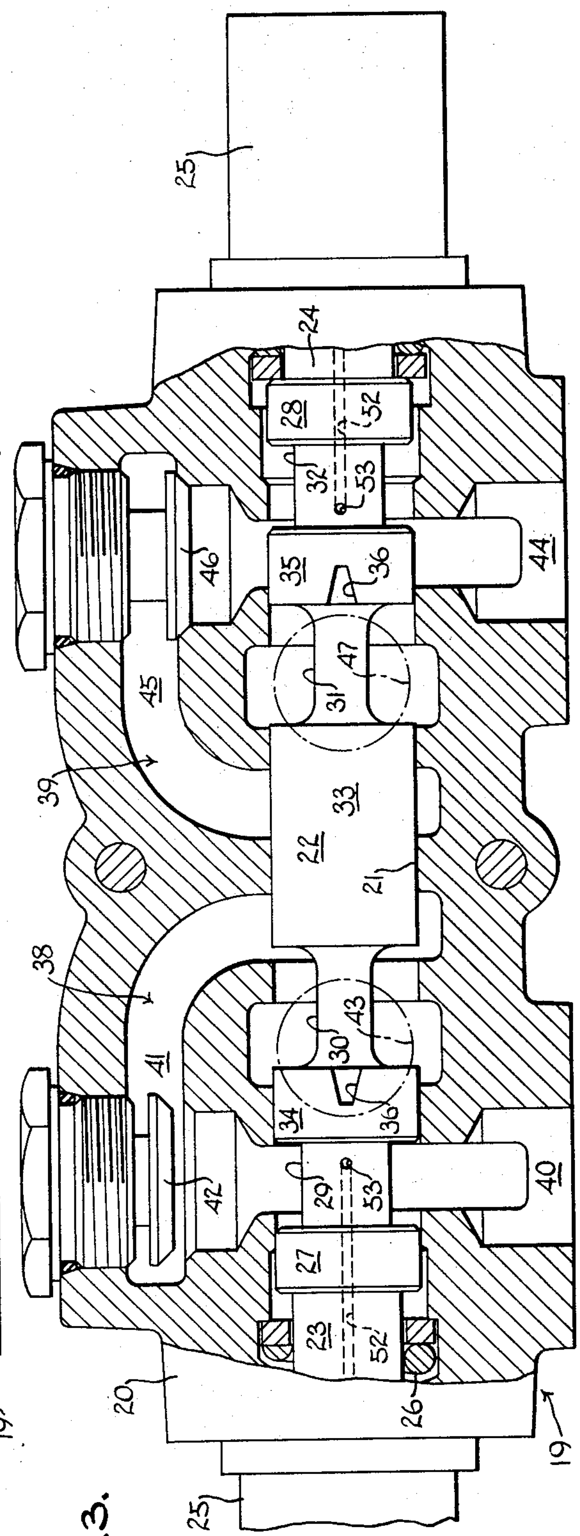

FIG. 3.

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FIG. 5.

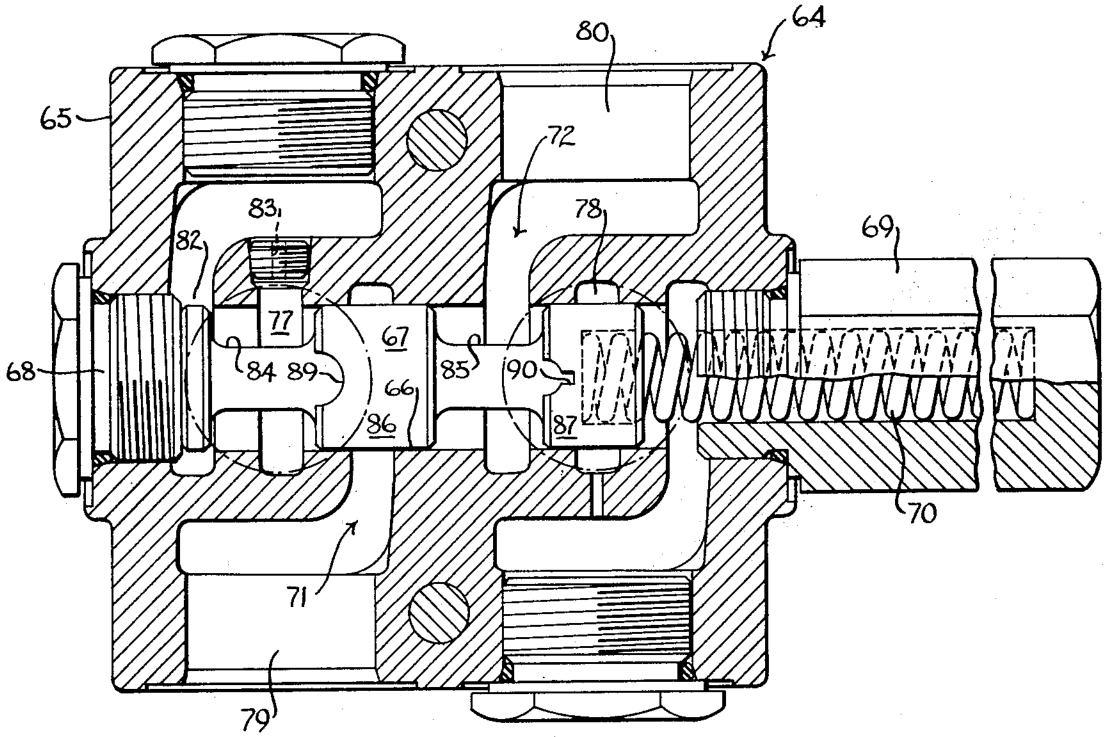
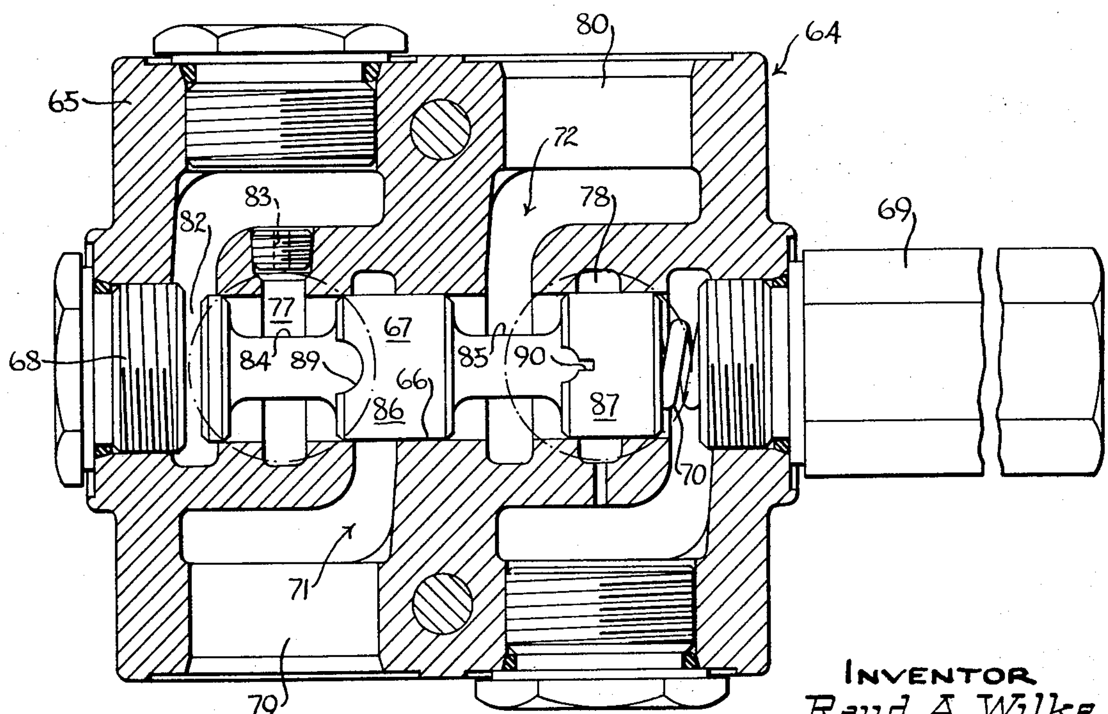


FIG. 6.



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FLUID FLOW CONTROLLING DEVICE FOR REVERSIBLE FLUID MOTORS

BACKGROUND OF THE INVENTION

This invention relates to flow control instrumentalities for use in fluid pressure operated systems incorporating a reversible fluid motor such as a double acting hydraulic cylinder; and it has more particular reference to improvements in flow correlating or counterbalance valve mechanisms such as have been used adjunctively with directional control valves in an effort to provide more positive control over the motion of the load carried by the cylinder and to also minimize void formation therein.

The type of hydraulically driven loads referred to are those commonly found in earth handling apparatus such as bulldozers, front end loaders, and power shovels. All of these are characterized by a heavy boom structure that constitutes a load which is movable up and down about a horizontal axis. The boom of a power shovel, however, must also be movable from side to side about a vertical axis. The boom is raised, for example, as a consequence of flow of pump fluid into one end of its lift cylinder, and it is lowered as a result of flow of pump fluid into the opposite end of said cylinder. In each instance, fluid expelled from the contracting end of the cylinder is returned to the reservoir of the system through the directional control valve.

Because of its great weight, the boom structure of earth handling apparatus such as mentioned above is easily started downwardly when pump fluid is directed into the "load lowering" end of its lift cylinder, and its descent is accelerated by the force of gravity. As a result, the boom tends to fall at an uncontrolled rate and to thus effect expulsion of pressure fluid from the contracting end of its cylinder faster than the pump can supply fluid to its expanding end.

Not only is such uncontrolled descent of the boom undesirable, but it has the further objection of creating a void in the expanding end of its cylinder. When that occurs, operation of the apparatus is delayed for the period of time it takes for the pump to fill the void in the cylinder.

Similarly, the boom swing cylinder of a power shovel, which moves the boom about a vertical axis, also tends at times to be driven in one direction or the other by the load, especially when the shovel is in operation on an inclined surface. The dipstick of a backhoe is an example of an overcenter load which is movable about a horizontal axis, and which can be hydraulically driven toward and beyond a vertically pendant position from either side of said pendant position. In that case also, both ends of the dipstick cylinder can have voids drawn therein, and the motion of the stick is difficult to control, especially if the bucket on its outer end is loaded.

It has been proposed to relieve voids in the expanding ends of the work cylinders of apparatus such as described by a control valve which functions to augment the flow of pump fluid to the starving end of the governed cylinder with fluid expelled from its contracting end. This is known as low pressure regeneration.

Not only is such a void relieving expedient unreliable at times when the pump is supplying pressure fluid to several hydraulic cylinders for concurrent performance of a number of work operations, but it is then impossible to cure a void if the work cylinder is of the type

wherein pressure fluid is expelled from its rod end by the rapidly moving load. In other words, there is just not enough fluid available to fill the expanding larger end of the cylinder. Even if there were, the load would still descend at an objectionably fast, or uncontrolled rate, in the absence of some restriction to exhaust flow from the cylinder.

One such scheme for restricting exhaust flow is disclosed in Tennis U.S. Pat. No. 3,513,877 dated May 26, 1970. Other schemes are disclosed in the Stephens U. S. Pat. No. 2,477,669 dated Aug. 2, 1949; the Kirkham U.S. Pat. No. 2,608,824 dated Sept. 2, 1952; Tennis U.S. Pat. No. 3,250,185 of May 10, 1966; and the U.S. Pat. to Falendysz et al, No. 2,926,634 dated Mar. 1, 1960.

SUMMARY OF THE INVENTION

This invention has as its purpose the provision of a flow coordinating or counterbalance valve mechanism for use with reversible fluid motors such as double acting hydraulic cylinders and their control valves, wherein the likelihood of malfunctioning is much less than in any such mechanism proposed heretofore.

In this respect it is an important and related objective of this invention to provide a counterbalance valve mechanism which features a single fluid pressure actuatable valve element for regulating exhaust fluid flow from either end of a double acting cylinder in accordance with the pressure of fluid supplied to its other end. This is in sharp contrast to past practice, where counterbalancing was achieved through the provision of separate fluid pressure actuatable valve mechanisms, one for each motor port, along with a common piston actuator between them.

Another object of the invention resides in the provision of a flow correlating valve mechanism embodying a single counterbalancing valve element and service passages so arranged as to enable load drop check valves to be incorporated therein without interfering with return flow of fluid through said passages.

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 two complete examples of the physical embodiments of the invention constructed according to the best modes so far devised for the practical application of the principles thereof, and in which:

FIG. 1 is a diagrammatic view illustrating a fluid pressure operated system including a double acting hydraulic cylinder, a control valve for governing the direction of motor operation, and a fluid flow controlling valve of this invention connected between the control valve and cylinder;

FIG. 2 is a longitudinal section through the flow controlling valve seen in FIG. 1, showing the same in a non-operating or hold position;

FIG. 3 is a sectional view similar to FIG. 2 but showing the flow restricting action of the device;

FIG. 4 is a diagrammatic view of a fluid pressure operated system like that seen in FIG. 1, but illustrating a modified form of flow controlling device; and

FIGS. 5 and 6 are sectional views of the flow controlling device seen in FIG. 4, showing different operating positions thereof.

Referring now to the accompanying drawings, and particularly to the fluid pressure operated system seen in the FIG. 1 diagram, the numeral 5 designates a reversible fluid motor here shown as a double acting hydraulic cylinder. The cylinder is shown in a vertical position merely by way of example, and it has a piston 6 fixed to the inner end of a piston rod 7, that projects from the bottom of the cylinder to be operatively connected with a load 8.

The direction in which the cylinder operates, or more particularly, the direction in which the piston 6 is driven, can be governed by a substantially conventional control or main valve 10. The control valve, which has been diagrammatically illustrated, is of the manually operated type having a valve spool movable from a neutral position shown to each of a pair of operating positions at opposite sides of neutral, to direct pressure fluid from the output of pump 11 to either end of the cylinder and to direct fluid expelled from the other end of the cylinder to the reservoir 12 of the system. The control valve has been shown as provided with a so-called motor spool, which vents the cylinder ports of the valve to the reservoir in the neutral position of the spool shown.

A delivery line 13 connects the outlet of the pump with the inlet 14 of the main valve 10; a return line 15 normally connects the exhaust port 16 as well as the cylinder ports of the control valve 10 with the reservoir 12; and a pair of service lines S1 and S2 connect the cylinder ports of the control valve with motor ports 17 and 18 in the top and bottom ends, respectively, of the cylinder 5.

The flow controlling or counterbalance valve 19 of this invention can be considered as a throttling device that can meter the rate at which fluid is expelled from the cylinder 5 under the influence of the load thereon. For that purpose, the counterbalance valve is connected in the service lines S1-S2 at a location between the cylinder 5 and the main valve 10. It comprises a body 20 having a bore 21 extending therethrough to receive an axially slidable fluid pressure actuatable spool type valve element 22. The opposite end portions 23 and 24 of the spool are reduced in diameter and they project into the interiors of cup-like covers 25 which not only serve to close the opposite ends of the bore 21 but to also house the spool centering springs 26. The springs encircle the opposite end portions 23 and 24 of the valve spool, and they bear against lands 27 and 28 thereon to normally hold the spool in the neutral position seen in FIG. 2.

Four axially spaced circumferential grooves 29, 30, 31 and 32 in the spool, located intermediate the lands 27 and 28 cooperate therewith and with one another to define an axially elongated central land 33, and flanking the latter, a pair of axially shorter lands 34 and 35. The last two can be termed metering lands since each has one or more metering notches 36 in its face, opening to the groove between it and the central land 33.

The counterbalance valve has a pair of service passages 38 and 39 which extend therethrough near the ends of the bore 21 which provide intermediate sections of the service lines S1 and S2, respectively. Pump fluid flowing to the cylinder through service line S1, from the main valve 10, enters an inlet branch 40 of service passage 38, which inlet branch communicates with the bore 21 and is at all times communicable with an intermediate branch 41 of the service passage 38 through a check valve 42. The intermediate branch 41 also communicates with the bore 21 at a location spaced axially inwardly from the junction between the bore and the inlet branch 40. A third branch 43 of the service passage provides a cylinder port which is communicated with the motor port 17 of cylinder 5 by that section of the supply line S1 which extends downstream from the counterbalance valve. This third branch of service passage 38 leads from the bore 21 at a zone situated between the junctions of the bore and the first two branches 40–41 of the service passage, and it is communicable with the intermediate branch 41 through the bore 21 under the control of the counterbalance valve spool therein.

Similarly, pressure fluid flowing out of the control valve through service line S2 connecting with the other motor port 18 of the cylinder, enters the inlet branch 44 of service passage 39 in the counterbalance valve, for flow to the intermediate branch 45 of the second service passage. The inlet branch 44 is communicable with the intermediate branch 45 through a check valve 46; and the intermediate branch is also communicable with the outlet branch or second cylinder port 47 through the bore 21 under the control of the valve spool 22.

In the neutral or hold position of the valve spool, its lands 34 and 35 block direct communication through the bore 21 of the inlet branches 40 and 44 of the service passages with their respective outlet branches 43 and 47; and the elongated central land 33 on the spool blocks indirect communication between those inlet and outlet branches through their respective intermediate branches 41 and 45.

From the description thus far, it will be apparent that the inlet branch 40 of service passage 38 will be communicated with the cylinder port provided by the outlet branch 43 thereof through the intermediate branch 41 whenever the valve spool is moved to the right of its neutral position seen in FIG. 2 to a first operating position such as seen in FIG. 3. In that operating position of the spool, pressure fluid directed to the inlet branch 40 by the main valve 10 can unseat the check valve 42 and flow into the intermediate branch 41 of the service passage, and thence through the bore 21 and spool groove 30 to the cylinder port 43, for flow to the upper port 17 of the cylinder 5. At the same time, fluid expelled from the lower port 18 of the cylinder by the downwardly driven piston 6 will return to service passage 39, for flow from the cylinder port provided by the outlet branch 47, through the bore 21 and spool groove 31 to the inlet branch 44, from whence it can flow to the reservoir through the main control valve 10. It should be noted, however, that such return flow of fluid bypasses the check valve 46.

The flow of pressure fluid to and from the cylinder is reversed, of course, when the counterbalance valve spool 22 is shifted to a working position to the left of neutral, and the main valve 10 directs pressure fluid to the inlet branch 44 of service passage 39, and communicates its other service passage 38 with the reservoir. The valve spool 22 is shifted automatically by means of hydraulic actuators **50** and **51**, capable of imposing fluid pressure actuating force upon one end or the other of the spool. In this respect, it will be observed that each of the hydraulic actuators comprises a piston provided by one end portion of the spool, and a pressure chamber or cylinder provided by the interior of the adjacent spring cover **25**, in which the pistons operate. Axial passages **52** in the end portions **23–24** of the valve spool open outwardly into the interiors of the cylinders of actuators **50** and **51** and communicate the latter with the inlet branches **40** and **44** of service passages **38** and **39**, respectively, through radial holes **53** in the spool at each of its grooves **29** and **32**.

The holes **53** and passages **52** in the end portions of the spool provide for pilot flow of pump fluid under pressure entering either service passage to the adjacent hydraulic actuator for actuation of the valve spool out of its neutral position to one or the other a pair of working positions at opposite sides of neutral. For example, if the main control valve **10** is actuated to communicate service passage **38** with the pump and to communicate service passage **39** with the reservoir, some of the pump fluid entering the inlet branch **40** of service passage **38** will flow to the actuator **50** and exert force on the left-hand end of the spool to shift it to the right of neutral, to its operating position seen in FIG. 3. At the same time, fluid displaced from the cylinder of actuator **51** by the valve spool will be vented to the inlet branch **44** of service passage **39** through the axial passage **52** and hole **53** in the right-hand end portion of the spool.

If the main control valve **10** is actuated to reverse the direction of flow to the work cylinder, pump fluid flows to inlet branch **44** of service passage **39**, and a pilot flow of such pump fluid will enter the cylinder of actuator **51** to effect actuation of spool **22** to a working position to the left of neutral. Fluid then displaced from the cylinder of actuator **50** is returned to the reservoir via the inlet branch **40** of service passage **38** and the passages in the main control valve **10**.

It will be seen, therefore, that the main control valve **10** acts in the nature of a pilot valve for the hydraulic actuators **50** and **51** for the counterbalance valve spool **22**, since it controls the supply of pilot fluid to the cylinder of either actuator while simultaneously venting the cylinder of the other actuator.

During operation, if it is assumed that the control valve **10** has been actuated to effect descent of the load on the cylinder **5**, pressure fluid from the pump **11** will flow to the upper port **17** of the cylinder via the service passage **38** in the flow controlling device of this invention; and fluid expelled from the bottom motor port **18** will be returned to the reservoir via part **47**, notch **36**, part **44** and control valve **10**.

Upon actuation of control valve **10** to start the load downwardly, pressure fluid flows through service line **S1** to the inlet branch **40** of service passage **38** and to the interior of the cylinder for actuator **50** to effect movement of the valve spool to the right, to a working position such as seen in FIG. 3. As therein seen, supply fluid entering the inlet branch **40** can unseat the check valve **42** and flow into the intermediate branch **41** of service passage **38**. It can then flow to the cylinder port **43** through the bore **21** and groove **30** in spool **22**. Exhaust fluid from the cylinder **5** flows directly from cylinder port **47** to inlet branch **44** via the bore **21** and spool groove **31**.

As soon as the load **8** on the cylinder **5** begins to drive the cylinder, that is, to effect extension of the piston rod faster than pump fluid can be supplied to the expanding upper end of the cylinder, there will be a corresponding drop in pressure in the upper end of the cylinder. This same drop in pressure will be manifested in the service passage **38** and also in the cylinder of actuator **50** for the valve spool **22**. As a result, the hydraulic force with which the valve spool **22** is held in its right hand operating position will be decreased in proportion to the pressure drop in the expanding end of the cylinder **5**. When this hydraulic actuating force decreases to the point at which it is overcome by the force of the return spring acting on the right hand end of the valve spool **22**, the spool will begin to move to the left under spring bias, toward neutral, to restrict flow of cylinder exhaust fluid through the spool groove **31** to the inlet branch **44** of service passage **39**. Thus, if the load on the cylinder **5** descends rapidly enough, it will effect return motion of the valve spool **22** an extent such as to cause land **35** to enter that portion of bore **21** which lies between the cylinder port **47** and the inlet branch **44** of service passage **39** and constrain exhaust fluid from the cylinder to flow through the metering notch or notches **36** in land **35**.

At the same time, the land **33** will have moved toward a position restricting the flow of supply fluid to cylinder port **43** from the intermediate branch **41** of the service passage **38**. This has the effect of raising the pressure in the inlet branch **40** of service passage **38**, and in the left hand actuator **50** for the valve spool **22**.

Return motion of the valve spool will cease, of course, when the fluid pressure force imposed upon the left hand end of the valve spool by actuator **50** balances the return force of the spring **26** acting on the right hand end of the spool. As a result, the spool **22** can float back and forth in its right hand working position, in accordance with changes in pressure in the expanding end of the cylinder as detected in the left hand hydraulic actuator **50** for the valve spool, to meter the exhaust of fluid from the cylinder **5** to whatever extent is necessary for controlled descent of the load and to minimize void formation in the upper end of the cylinder.

It will be seen, therefore, that exceptionally effective counterbalance and flow coordination is achieved by the flow restricting action of the lands **33** and **35** at times when the load **8** begins to drive the work cylinder **5**.

The counterbalance spool will be returned to neutral promptly in response to actuation of the main control spool to its neutral position. Assurance of such response of the counterbalance spool is had by reason of the fact that with the main valve spool in neutral, the inlet branches **40** and **44** of the service passages in the counterbalance valve are vented to the reservoir. Otherwise, fluid might become trapped in the inlet branches **40**, **44** of the service lines, in which event such trapped fluid could interfere with proper return of the counterbalance spool to its neutral or hold position.

If the hydraulic motor **5** should comprise the swing cylinder of a backhoe, by which its boom is swung back and forth about a vertical axis, or even the crowd cylinder of the backhoe, it can at times be driven in either direction by the load operatively connected thereto. The counterbalance valve of this invention, however, will function to restrict exhaust flow from either contracting end of the work cylinder in accordance with the drop in pressure occurring in its expanding end. This is because the hydraulic actuators **50** and **51** are duplicated at opposite ends of the valve spool **22**, as are the notches **36** in lands **34** and **35**, which are effective to meter exhaust flow regardless of which end of the work cylinder the fluid is exhausting from.

It is of considerable importance to note that as long as pump output fluid is flowing to one or the other inlet branch **40** or **44** of the counterbalance valve, the spool **22** thereof can never be returned so far toward neutral as to completely block flow to the corresponding cylinder port **43** or **47**. For example, with pump output fluid flowing to port **43** via the intermediate branch **41** of service passage **38**, a severe drop in pressure in port **43** and in actuator **50** can result in spring biased return motion of the counterbalance spool **22** toward neutral; but the extent of such return motion is limited by the tendency of the pressure in the inlet branch **40** and chamber **50** to increase in accordance with the decrease in flow of pump output fluid permitted from the intermediate branch **41**, past the left hand face of land **33** to the port **43**, via the bore **21** and spool groove **30**.

Thus, it can be said that as long as pump fluid is issuing from the control valve **10**, the counterbalance spool will be held in an operating position displaced from neutral a distance depending upon the pressure of fluid at that cylinder port thereof through which pressure fluid is flowing to the work cylinder **5**. In that respect, it can be said that the throttling or counterbalance valve of this invention is flow responsive, since its valve spool will never close off flow from the motor as long as pump fluid is flowing to one or the other of its service passages.

It is also significant to note that if the flow of pump fluid to either inlet branch **40** or **44** of the counterbalance valve ceases for any reason, as for instance because of stalling of the internal combustion engine power source on the governed machine, the counterbalance spool will immediately return to neutral, not because of any influence which the work cylinder has thereon, but because of lack of pump pressure in the actuator **50** or **51** that had been holding the spool in an operative position. The same thing would occur if the service line supplying pump fluid to the work cylinder were to rupture at a point between the control and counterbalance valves. In both cases, of course, one or the other of the check valves **42** or **46** would close to prevent load drop; and the main valve **10** does not need to be burdened with load drop check valves as was customary heretofore.

It should now be apparent also, that the exhaust throttling action of the counterbalance spool **22** will not only prevent uncontrolled motion of the load on a motor such as the work cylinder **5**, but that it will also minimize cavitation in the expanding end of the cylinder at times when the load tends to drive the cylinder. Obviously, if control over the load is achieved by slowing its motion, the pump will have less difficulty in supplying pressure fluid to the expanding end of the cylinder.

While the counterbalance valve device described above is operable to control exhaust flow from either side of a reversible fluid motor, FIG. **4** illustrates a simpler version **64** thereof which can be used to the same advantage when control over the exhaust flow of fluid from one end only of the motor is needed, as when it merely drives its load up and down. A double acting lift cylinder **60** has been shown as such a motor; and its direction of operation is governed by a control valve **61** which differs from the valve **10** hereinbefore described in that it has a hold-in-neutral spool and a conventional load drop check valve arrangement.

The counterbalance valve device **64** comprises a body **65** having a bore **66** to receive an endwise slidable counterbalance spool **67**. Plugs **68** and **69** close the opposite ends of the bore, and the plug **68** provides a stop which defines the left hand limit of motion of the spool. It is at all times urged to that position under force imposed thereon by a spring **70** reacting between the spool and the plug **69** closing the right hand end of the bore **66**.

The service lines **S1** and **S2**, in this case, also lead to the cylinder **60** through service passages **71** and **72** in the body of the counterbalance valve **64**. These service lines can be said to comprise upstream branches **73** and **74**, respectively, which extend between the control valve **61** and the counterbalance valve, and downstream branches **75** and **76**, respectively, which extend between the counterbalance valve and the work cylinder **60**.

The inlet branches **77** and **78** of service passages **71** and **72**, respectively, open to the bore **66** at axially spaced zones located a short distance inwardly of the plugs **68–69** and a greater distance from one another. The outlet ports **79** and **80** of service passages **71** and **72**, respectively, open to the bore at locations between the zones at which the inlet ports join with the bore, and spaced from one another and from those zones.

The left-hand end of the counterbalance spool **67** projects into a hydraulic actuator chamber **82** which is communicated with the inlet port **77** through a substantially small diameter passageway **83**. Hence, whenever pump output fluid is directed into the inlet branch **77** from the main valve **61**, the pressure of such fluid will be manifested in the chamber **82**, and hydraulic force will thus be imposed upon the left-hand end of the counterbalance spool to move it to an operating position.

The spool **67** has two circumferential grooves **84** and **85** therein, which grooves define axially spaced lands **86** and **87**. The groove **84** normally registers with the inlet branch **77** of service passage **71**, while the groove **85** normally registers with the outlet port **80** of service passage **72**. The land **86** normally blocks communication through the bore **66** between the inlet branch **77** and outlet port **79** of service passage **71**; while the land **87** similarly normally blocks communication through the bore between the inlet branch **78** and outlet port **80** of service passage **72**. A metering notch (or notches) **89** is formed in that face of the land **86** which is defined by the groove **84**; and a similar notch (or notches) **90** is formed in that face of the land **87** which is defined by the groove **85**.

It will thus be apparent that when the main valve spool is shifted to the right of its position shown, it will direct pump fluid to the inlet branch **77** of service passage **71** of the counterbalance valve **64**. Because the actuator chamber **82** is communicated with the inlet branch **77**, pressure fluid thus entering the latter will activate the actuator **82** and cause the counterbalance spool to shift to the right, against the return bias of its spring **70**, to an operating position at which the two ports **77** and **79** comprising service passage **71** are communicated with one another through the spool groove **84**, and the two ports **78** and **80** comprising service passage **72** are communicated with one another through the spool groove **85**. This, of course, opens up service lines **S1** and **S2** for flow of pump fluid to the upper port of cylinder **60** through service line **S1**, and for return to the reservoir of fluid expelled from the lower port of the cylinder, through service line **S2**.

As before, if the load on the cylinder begins to descend at a rate such as to effect a decrease in the pressure of fluid in its expanding upper end, such decreased pressure will cause a corresponding decrease in pressure in the hydraulic actuator **82**.

If the pressure in the upper end of the work cylinder and in actuator **82** drops to a value such that the hydraulic actuating force on the counterbalance spool is exceeded by the force of its return spring **70**, the spool will be shifted toward the left, to a position at which motor exhaust fluid can only flow to the inlet branch **78** of service passage **72** via the metering notch (or notches) **90** in the spool land **87** (see FIG. **6**). At the same time, the flow of pump fluid from the inlet port **77** to the outlet port **79** of service passage **71** will be restricted because of constraint of such flow through the metering notch (or notches) **89**.

As before, the counterbalance spool **67** will seek a stable metering position somewhere between its left hand limit of motion and a full flow operating position, depending upon the pressure in the upper end of the work cylinder, and at which position the spring return force on the spool will be in balance with the hydraulic actuating force thereon. The spool, of course, will be returned to its normal position, at its left hand limit of motion, under the influence of the return spring **70** as soon as the spool of the main control valve is returned to neutral.

As diagrammatically shown, the flow of pressure fluid to and from the work cylinder **60** will bypass the counterbalance valve device **64** when the main valve spool is actuated to the left, to a "load raising" position. This results from the fact that branches **74** and **76** of supply line **S2** are connected with one another by a bypass duct **92** containing a check valve **93**. The check valve opens to allow pump fluid flowing through branch **74** to flow directly to branch **76** in bypass relation to the counterbalance valve **64**. Similarly, the branches **73** and **75** of supply line **S2** are connected by a bypass duct **94** containing a check valve **95**. Check valve **95** opens in the direction to allow return fluid from the upper end of the cylinder to flow to the supply line branch **73** in bypass relation to the counterbalance valve.

From the foregoing description, together with the accompanying drawings, it will be apparent to those skilled in the art that this invention provides an exceptionally efficacious counterbalance and flow coordinating valve, because of its ability to remain in a stable flow metering position as long as pump fluid is flowing to the valve.

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 control device for regulating flow of fluid in the service lines connecting the opposite sides of a reversible fluid motor with the control valve governing the direction of motor operation, characterized by the following:

A. a body having a bore, and independent first and second service passages intersecting the bore and respectively defining individual flow paths leading through the body from separate source fluid inlets to provide for flow of pressure fluid from a source to one side of a reversible fluid motor along one of said paths and for return flow of exhaust fluid from the other side of the motor along the other of said paths;

B. a valve element in the bore movable between a working position providing for such flow through the service passages and a hold position blocking said flow therethrough;

C. spring means acting on the valve element to yieldingly resist movement thereof to its said working position;

D. a hydraulic actuator activated by pressure of source fluid in said first service passage for shifting the valve element toward its said working position a distance determined by the pressure of said activating fluid, whereby decrease in the pressure of the latter causes return motion of the valve element under the force of said spring means;

E. and throttling means on the valve element operable upon such return motion thereof to increasingly restrict said second service passage.

2. The control device of claim **1**, further characterized by:

A. said hydraulic actuator comprising a cylinder;

B. the valve element having an end portion projecting into said cylinder and providing a piston therein;

C. and means providing a pilot passageway communicating the cylinder with the inlet portion of said first service passage.

3. The control device of claim **1**, further characterized by:

A. said first service passage comprising inlet and outlet branches communicating with the bore at axially spaced locations;

B. and a check valve arranged to prevent back flow of fluid from the outlet to the inlet branch.

4. The control valve of claim **1**, further characterized by:

A. said first service passage comprising communicating inlet and intermediate branches which open to the bore at axially spaced first and second zones, and an outlet branch which opens to the bore at a location between said zones;

B. and a check valve in said first service passage at a location between said first and second zones.

5. The control valve of claim **1**, further characterized by:

A. said first service passage comprising communicating inlet and intermediate branches which open to the bore at axially spaced first and second zones, and an outlet branch which opens to the bore at a location between said first and second zones;

B. and the valve element having means thereon to effect communication of said outlet branch with the inlet branch via the intermediate branch in said working position of the valve element and to effect communication of said outlet branch with the inlet branch in bypass relation to the intermediate branch in another working position of the valve element.

6. The control device of claim 1, further characterized by:

A. another hydraulic actuator for shifting the valve element toward a second working position at the side of its hold position remote from said first working position thereof, and at which second working position source fluid can flow to the motor through said second service passage and fluid expelled from the motor can be exhausted through said first service passage;

B. other spring means yieldingly resisting movement of the valve element to said second working position;

C. means for activating said other hydraulic actuator in response to pressure of source fluid in said second service passage, whereby the extent of motion of the valve element toward its second working position is determined by the pressure of said last named activating fluid and decrease in the pressure thereof likewise causes return motion to be imparted to the valve element by said other spring means;

D. and other throttling means on the valve element operable to increasingly restrict said first service passage during such return motion of the valve element from its second working position.

7. The control device of claim 6, further characterized by:

A. a check valve for each of said service passages, arranged to be opened by flow of source fluid through either passage to a fluid motor;

B. and said service passages having branches controlled by the valve element and through which fluid exhausting from the motor can flow in bypass relation to the check valves thereof in either working position of the valve element.

8. The control device of claim 6, further characterized by:

A. each of said service passages comprising

1. a passageway having a portion extending longitudinally along the bore at one side thereof to have its ends communicate with the bore at axially spaced first and second zones,

2. one end of said longitudinal passage portion providing an inlet passage,

3. and a cylinder port which opens to the bore at a third zone between said two zones;

B. a load drop check valve in each of said service passages between its said inlet passage and its longitudinal passage portion;

C. and means on the valve element operable in each working position thereof to communicate a different one of the cylinder ports with the associated inlet passage through its respective longitudinal passage portion and to communicate the other cylinder port with the associated inlet passage in bypass relation to its longitudinal passage portion and the check valve therein.

9. A control valve for governing operation of a reversible fluid motor, characterized by the following:

A. a valve body having a bore and a valve element movable therein from a neutral to an operating position;

B. means defining a pair of individual service lines rendered effective by the valve element in said operating position thereof to provide for flow of pressure fluid from a source through one of said service lines to one side of a reversible fluid motor and for return flow of motor exhaust fluid through the other service line, each of said service lines having a portion downstream from the bore which is connectible with one side of a reversible fluid motor and having a portion upstream of the bore communicable therethrough with its said downstream portion by the valve element upon movement thereof out of its neutral position toward said operating position thereof;

C. spring means acting on the valve element to yieldingly resist movement thereof toward said operating position thereof;

D. a hydraulic actuator activated by pressure of fluid from said source in the upstream portion of said one service line for shifting the valve element out of its neutral position toward said operating position thereof a distance determined by the pressure of said activating fluid, whereby decrease in the pressure of the latter causes return motion of the valve element toward neutral under the force of said spring means;

E. and throttle means on the valve element operable upon such return motion thereof to increasingly restrict flow through said other service line.

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