United States Patent

[11] 3,613,730

[72] Inventor Robert H. Breeden
Metamora, Mich.
[21] Appl. No. 34,008
[22] Filed May 4, 1970
[45] Patented Oct. 19, 1971
[73] Assignee Sperry Rand Corporation
Troy, Mich.

[54] POWER TRANSMISSION
12 Claims, 10 Drawing Figs.

[52] U.S. Cl. 137/625.6, 267/151
[51] Int. Cl. F16k 11/00
[50] Field of Search 267/151; 137/625.6, 596.14

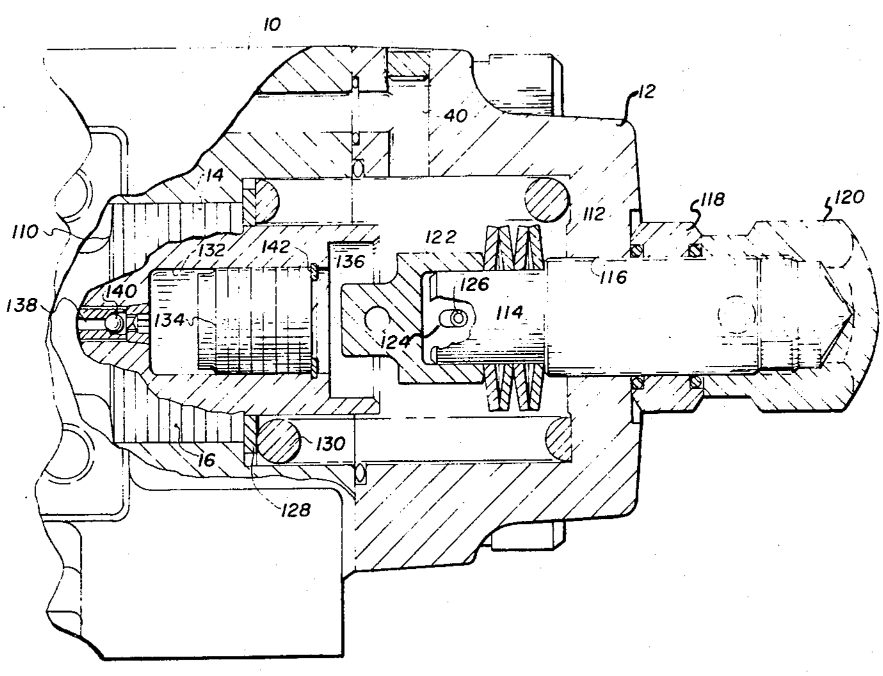
[56] References Cited

UNITED STATES PATENTS

2,922,440	1/1960	Hardy et al.	137/625.69 X
3,170,484	2/1965	Benz et al.	137/596.16
3,221,770	12/1965	Faisandier	137/625.68 X
3,279,484	10/1966	Brinkel	137/625.6 X

Primary Examiner—Henry T. Klinksiek
Attorney—Van Meter and George

ABSTRACT: Remote control of a directional control valve in a hydraulic power system is achieved through a pilot control circuit carrying a continuous flow of fluid through a fixed and a variable orifice in series. The variable pressure intermediate the orifices is applied in opposition to a spring system for achieving precise degree of opening of the directional valve and thus modulating the flow therethrough. The spring system achieves a fine and a coarse control of the output flow through the use of a high rate spring effective during initial opening travel of the valve. A flow rate limiter responsive to the pressure drop across an orifice in the main return flow line acts when this pressure drop becomes too high to override the remote control system and shift the main valve toward closed position. The control system is especially useful in hydraulic systems having a central power supply operating at a constant pressure to make fluid available selectively to any of a number of independent hydraulic motors, as for example for operating deck machinery on a ship.



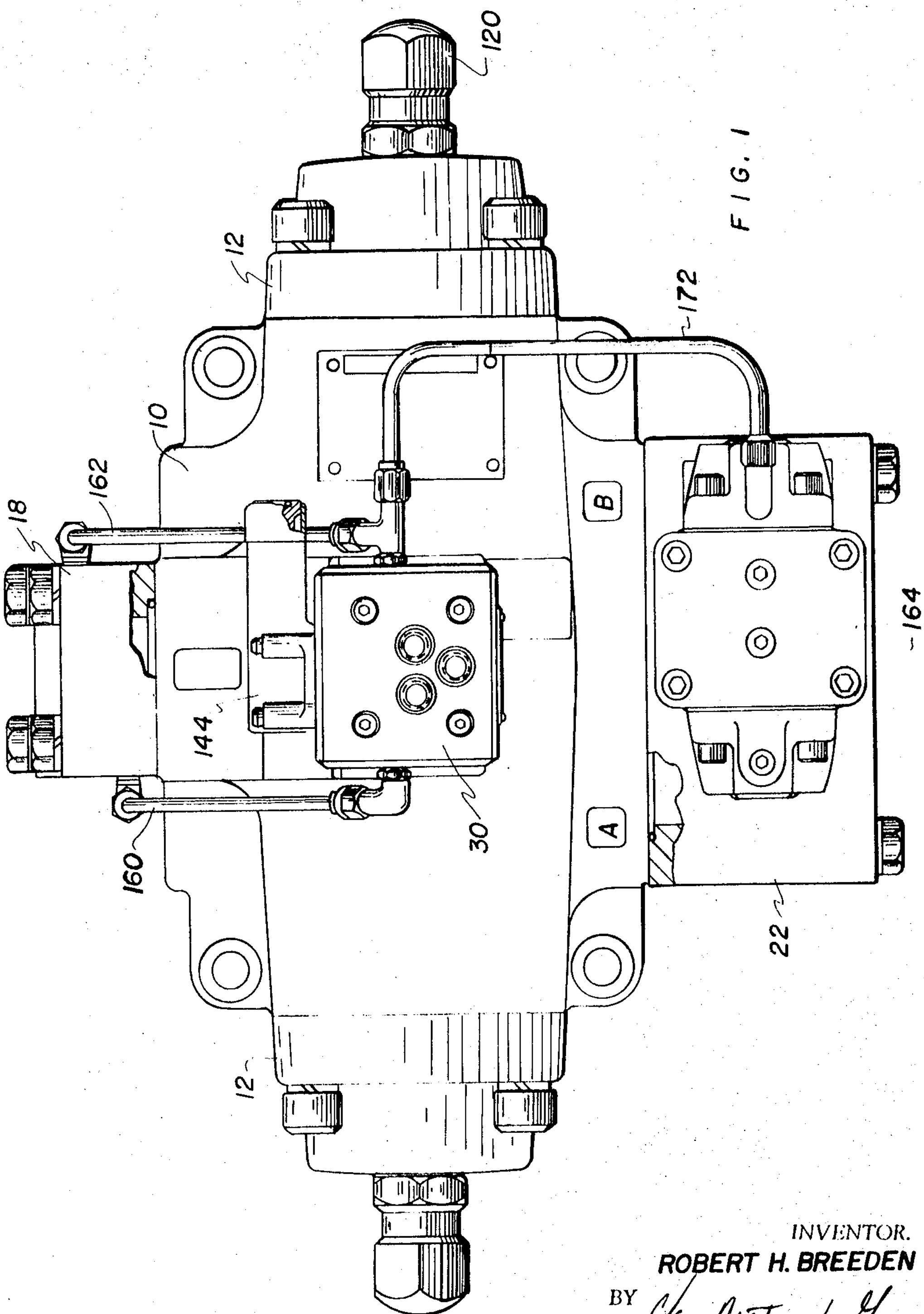


FIG. 1

INVENTOR.
ROBERT H. BREEDEN
BY Van Metre & George
ATTORNEYS

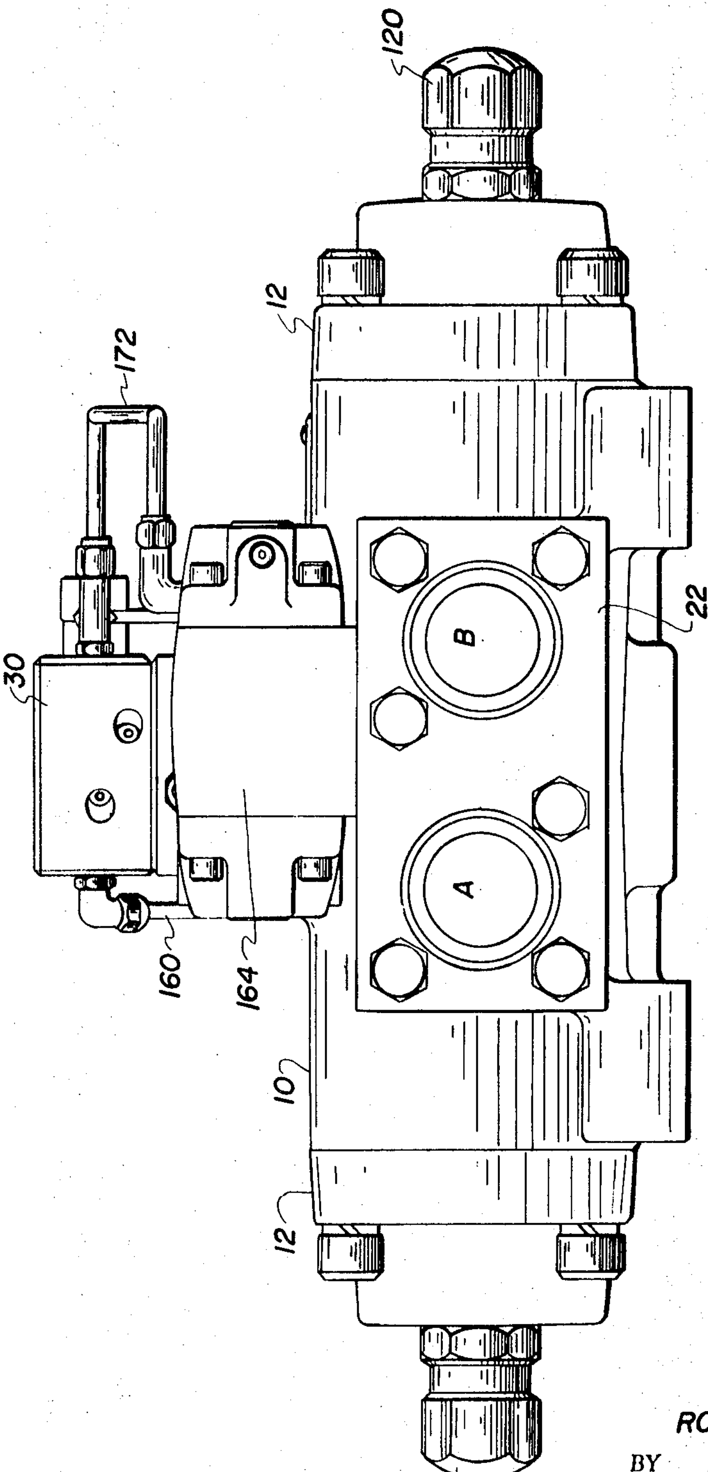


FIG. 2

INVENTOR.
ROBERT H. BREEDEN

BY

ATTORNEYS

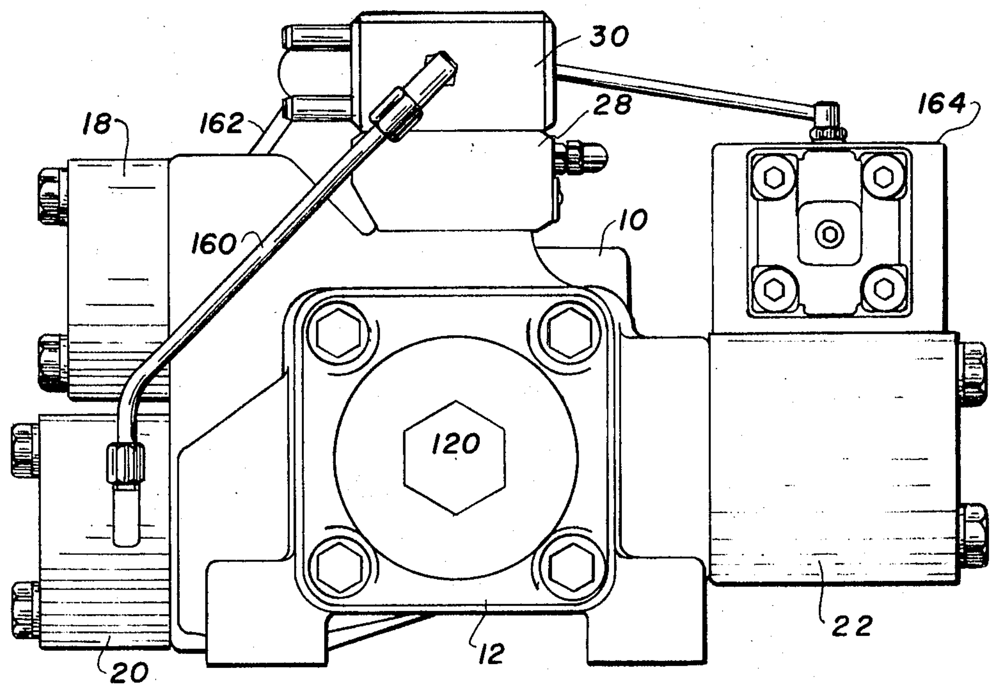


FIG. 3

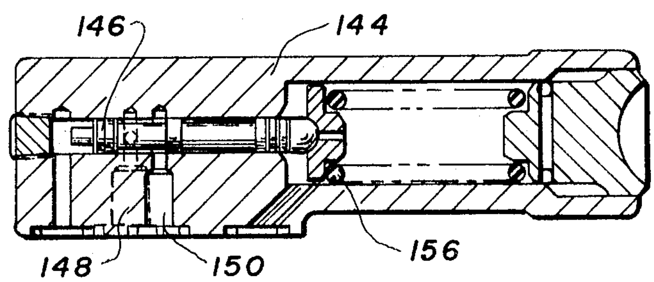


FIG. 5

INVENTOR.
ROBERT H. BREEDEN
BY Van Metre & Strange
ATTORNEYS

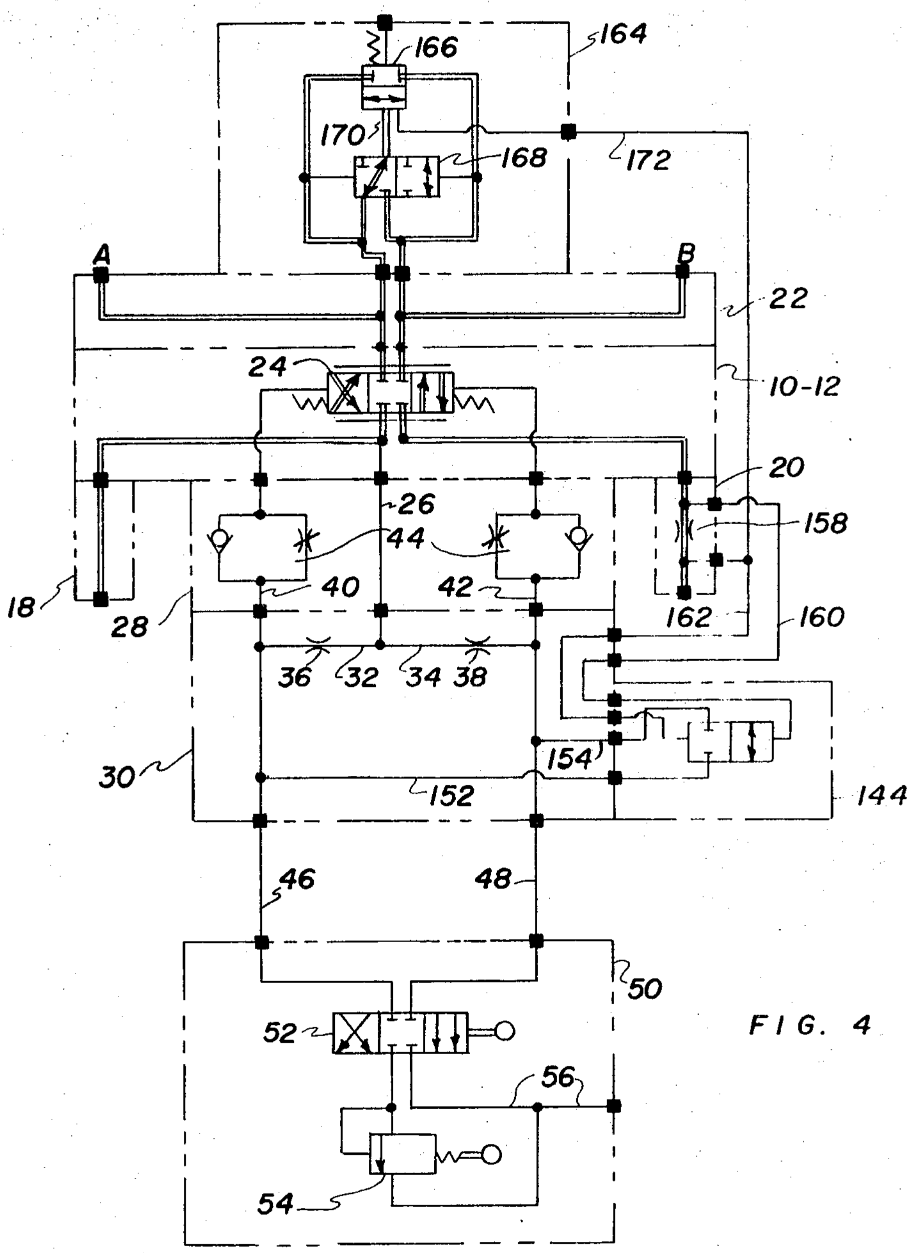


FIG. 4

INVENTOR.
ROBERT H. BREEDEN
BY [signature]
ATTORNEYS

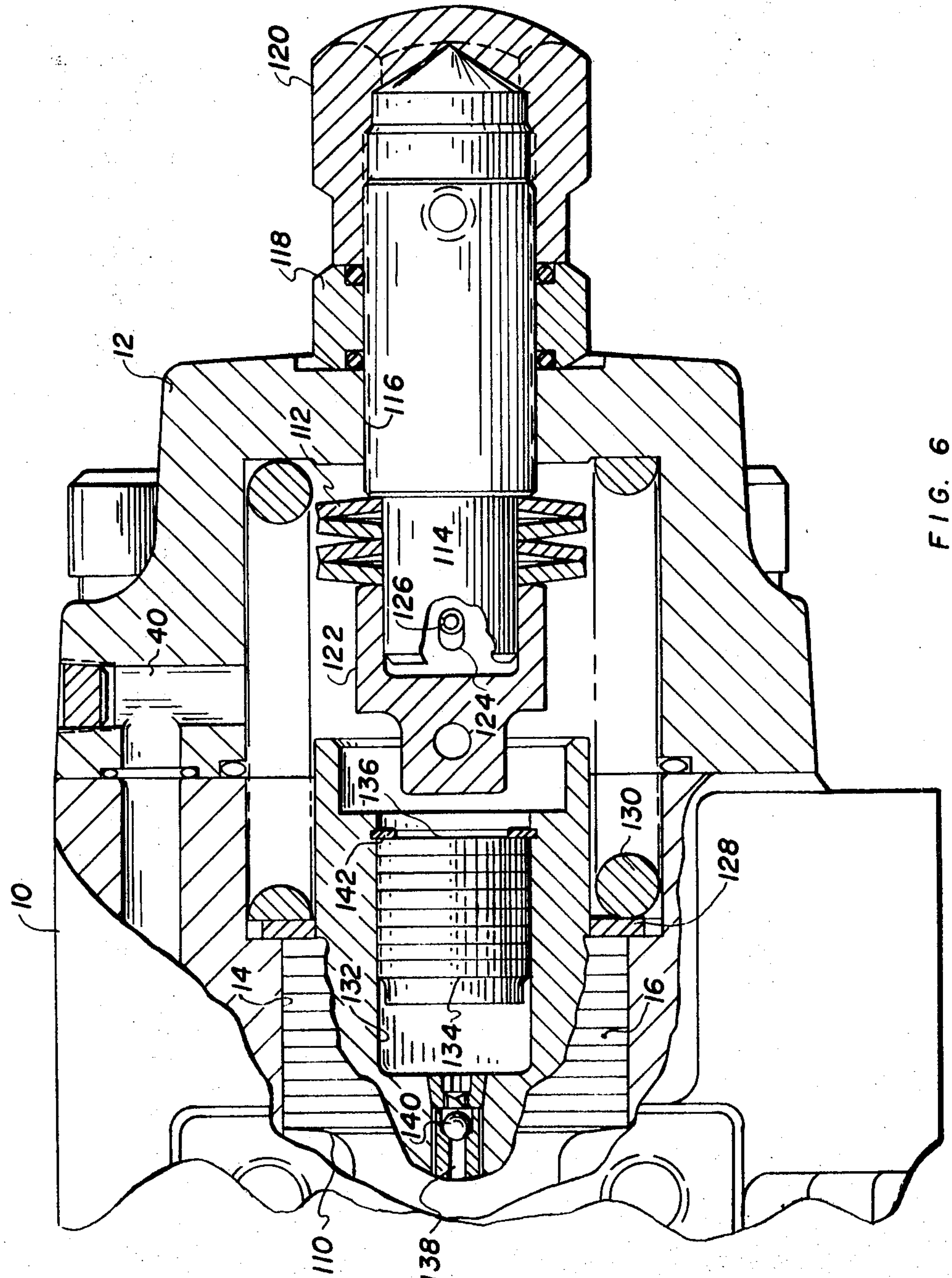
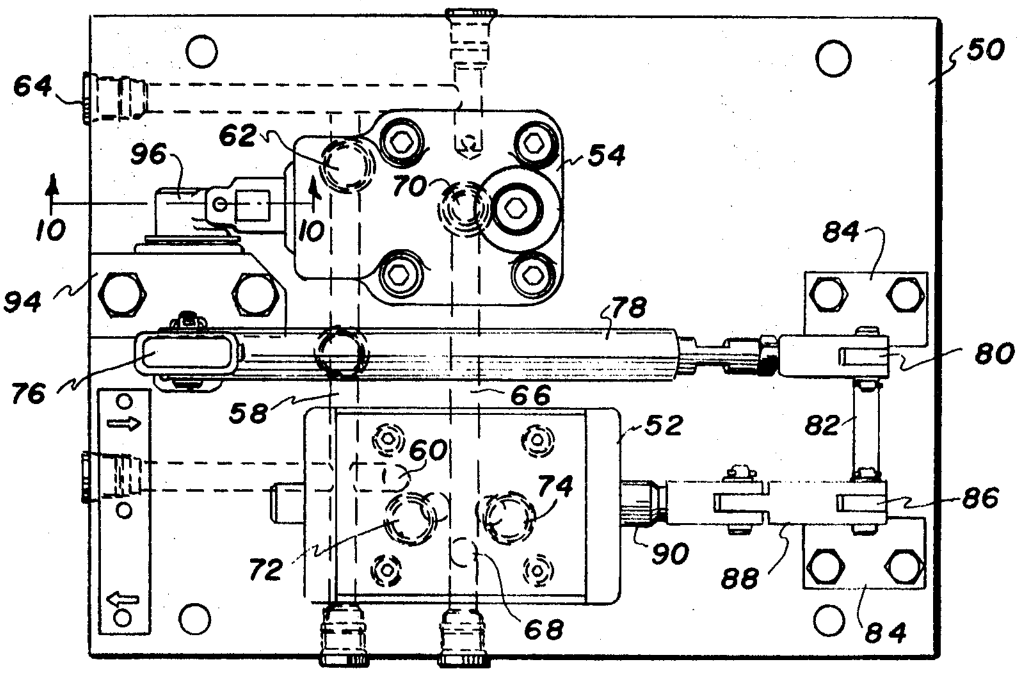


FIG. 6

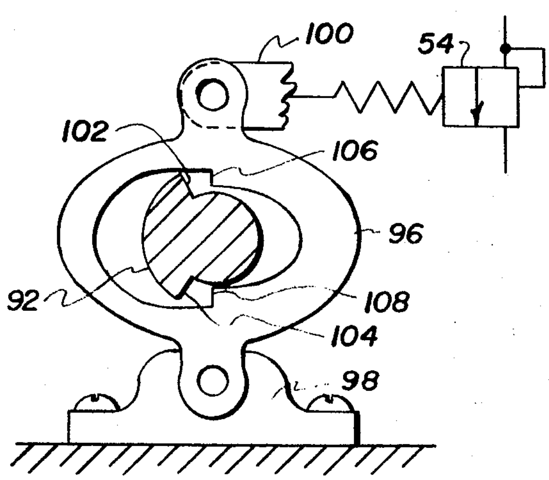
INVENTOR.
ROBERT H. BREEDEN

BY

ATTORNEYS



F I G. 7



F I G. 10

INVENTOR.
ROBERT H. BREEDEN

BY *[signature]*

ATTORNEYS

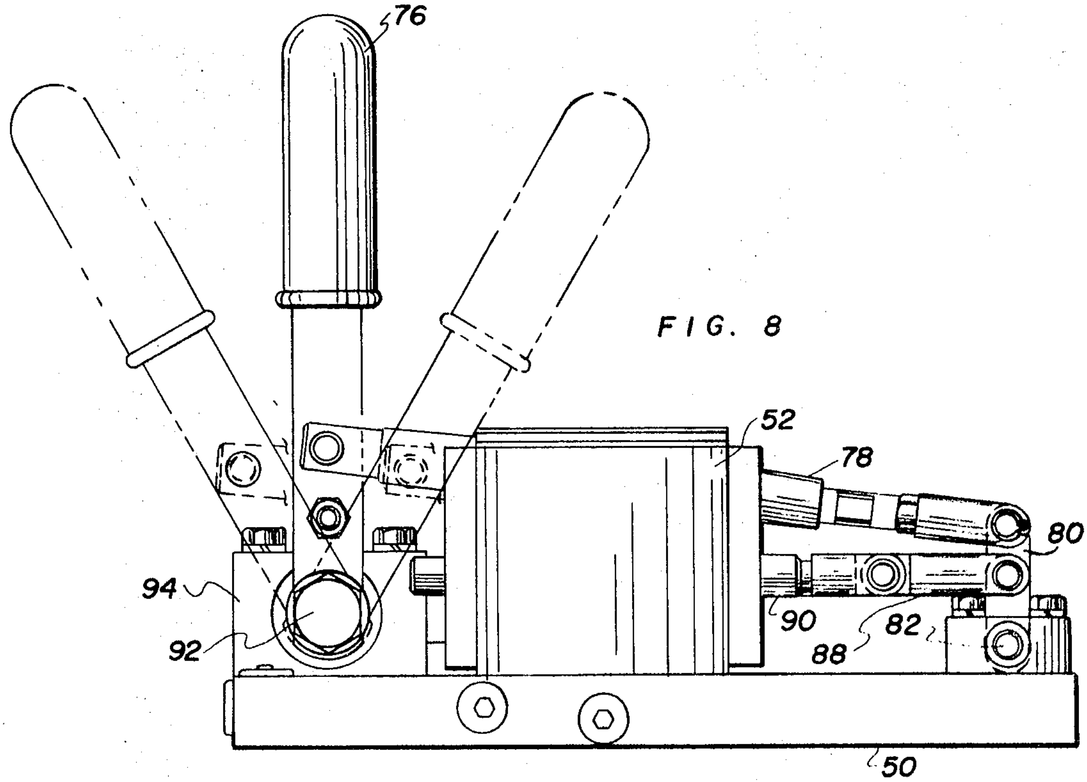


FIG. 8

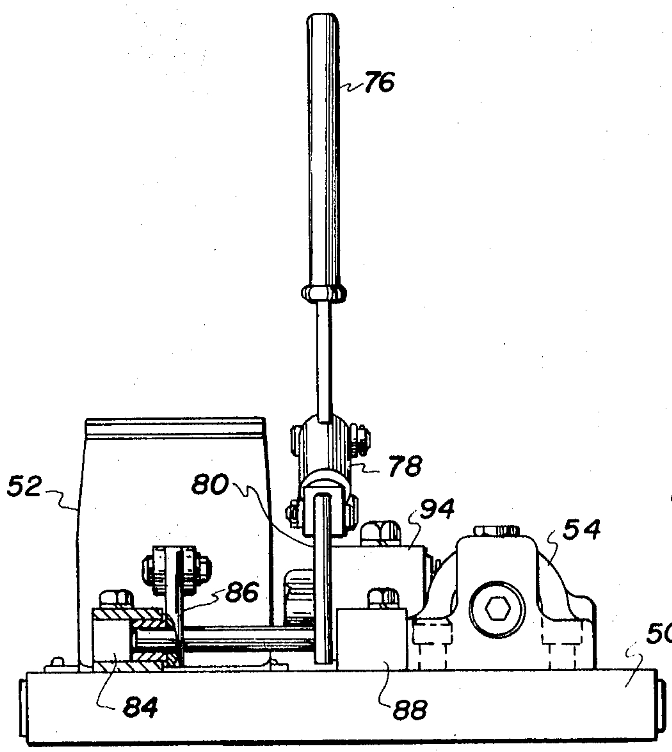


FIG. 9

INVENTOR.
ROBERT H. BREEDEN
BY Van Meter & Strange
ATTORNEYS

POWER TRANSMISSION

Hydraulic power transmission systems of the type having a central pressure supply operating at a nearly constant pressure for driving selectively any one or more of a plurality of hydraulic motors are widely used in many applications such as aircraft, farm tractors and the like, especially where a simple on-off control of the hydraulic motors is all that is necessary. Where a more precise modulation of the flow to the hydraulic motor is required, such systems have heretofore not been well suited except by using costly precision servocontrols. This is particularly true in systems where the operator's control handle must be at a distance from the directional valve which supplies fluid to a motor and where it is difficult or impossible to provide direct mechanical connection between the operator and the directional valve.

It is an object of the invention to provide a directional valve and a remote control system therefor which dispenses with mechanical connections between the remote controlling station and the valve, and which at the same time provides for precise modulation of the flow through the valve conveniently from the remote control station.

The sliding spool type of directional valve which is almost universally used in hydraulic systems does not lend itself readily to precise flow modulation because the rate of flow increase with each increment of spool travel is very high at small valve openings and falls off rapidly after the first few increments of spool travel. This characteristic increases the difficulty of achieving precise modulation when the spool travel is controlled from a distance.

It is an object also of the invention to provide a remote control system capable of application to a standard sliding spool directional valve with only minor modifications and which through the use of only pipe connections between the valve and the remote control station can provide a precise control of spool travel during its initial opening as well as throughout its full travel.

A further object is to provide in a valve of this type a simple flow limiting control which may be adjusted to hold the maximum flow through the valve at any desired level.

The invention consists in a valve for controlling fluid flow comprising a body having a laterally ported bore, a spool slidable in the bore to control flow between ports of the bore, pressure actuated pilot means for shifting the spool, a high-rate spring means positioned to oppose the pilot means during a portion of the spool travel, a low-rate spring means positioned to oppose the pilot means during at least the remainder of the spool travel, a yieldable abutment between the spool and the high-rate spring, fluid-operated means for controlling the yieldable abutment and means for applying and modulating a supply of pressure fluid to the pilot means.

FIG. 1 is a top view of a valve incorporating a preferred form of the present invention.

FIG. 2 is a side view of the valve of FIG. 1.

FIG. 3 is an end view of the valve of FIG. 1.

FIG. 4 is a circuit diagram of the valve and its controls.

FIG. 5 is a sectional view of a bypass valve incorporated in the controls.

FIG. 6 is a fragmentary view on a larger scale showing a portion of the valve of FIG. 1, partly in section.

FIG. 7 is a top view of a remote control station forming part of the controls.

FIG. 8 is a side view of the control station of FIG. 7.

FIG. 9 is an end view of the control station of FIG. 7.

FIG. 10 is a fragmentary sectional view taken on line 10—10 of FIG. 7.

The main directional control valve illustrated in FIGS. 1, 2 and 3 comprises a main body 10 having a pair of end caps 12 which are identical and which has an internal bore provided with the usual series of lateral ports for pressure for exhaust and for supply to either of the motor connections A and B. The central bore 14 in FIG. 6 carries a spool 16 for controlling the flow between the various lateral ports of the bore, as is well understood in the hydraulic art. External connection flanges are provided at 18 and 20, FIG. 3, for pressure and exhaust and a common flange at 22 is provided for connections to the motor passages A and B.

Valves of this type have not heretofore been well suited for use in central hydraulic systems wherein flow from a substantially constant pressure source must be modulated to control the operation of a fluid motor, since the degree of valve opening required for a given motor speed varies widely depending upon the load which the motor has to overcome. At very light loads, only a slight opening through the valve is required since a high degree of throttling at this point is necessary in order to control the operating speed of the motor. On the other hand, when the load is high, very little throttling is required and much wider openings of the valve are necessary to produce a given motor speed. These problems have been overcome to some extent by providing long tapers or tapered grooves on the spool heads, but this greatly increases the required size of the valve and introduces problems when the valve is to be pilot operated by fluid pressure.

Referring now to FIG. 4, the main directional control valve as a whole is there designated as 24, being illustrated in accordance with the international standard diagrammatic conventions, and as contained within the block designated 10–12. For the purpose of providing pilot pressure control, a circuit is established utilizing as a source of pressure the main pressure supply entering through the flange connection 18 and tapped off from the same by the line 26. This line may be contained in a block or sandwich plate 28 which can be bolted to the top of the body 10 as shown in FIG. 3. Line 26 leads to a second block or sandwich plate 30 within which the line 26 divides into branches 32 and 34 containing restrictions 36 and 38 respectively. The branch 36 has an arm 40 extending to one of the end caps 12 of the valve 24 and the branch 34 similarly has an arm 42 extending to the opposite end cap of the valve 24. Each of the arms 40 and 42 may have a variable restrictor with a check valve bypass as indicated at 44 for the purpose of regulating the rate of travel of the spool 16. The branches 32 and 34 also have arms 46 and 48 respectively which serve as the remote control connectors between the main directional valve and the remote control station indicated by the block 50. The arms 46 and 48 connect to a four-way directional valve 52 which is manually controlled at the remote location as will later be described. The directional valve 52 serves to reverse the connections of the two arms 46 and 48 to an adjustable pressure relief valve 54 and a direct exhaust line 56 respectively. That is to say, when one of the arms is connected to the relief valve 54, the other arm is connected directly to the exhaust line 56 and vice versa.

By means of a pilot control circuit of this kind, the spool of the main valve 24 may be given a modulating type of control from a remote location. There is a continual small flow from the high pressure supply line into the line 26 and out through the branches 32 and 34, through the restrictions 36 and 38, and through the arms 46 and 48 to the four-way directional valve 52, and ultimately from thence to the exhaust line 56. Depending upon the direction in which flow takes place through the valve 52, one or the other of the arms 46 and 48 will be exhausted directly to the line 56 and the other will be exhausted thereto through the adjustable pressure relief valve 54. Thus, by first shifting the four-way directional valve 52 and then by increasing the pressure on the spring of the relief valve 54, pressure may be built up in one or the other of the arms 46 and 48 and this pressure is reflected in one or the other of the arms 40 or 42 respectively. It is this control pressure which is applied within an end cap 12 and which opposes the spring means in the opposite end cap and thus enables an operator to exert a modulating type of control on the main valve 24.

In order to provide a convenient single lever control for the valves 52 and 54, the remote control station may be constructed as illustrated in FIGS. 7 through 10. Thus, the block 50 may be provided with internal passages such as 58 which connect with four-way valve 52 at the port 62 and with the external exhaust connection 64. Likewise, passage 66 connects with valve **52** at port **68** and with valve **54** at port **70**. Ports **72** and **74** in the top of the four-way valve **52** are provided for connection with the remote control arms **46** and **48** respectively of the control circuit.

The valves **52** and **54** may be manually actuated from a single lever **76** through a suitable mechanical linkage such as provided by the double acting collapsible link **78** which has a pair of internal springs preloaded to yield only under a force significantly greater than that required to shift the four-way directional valve **52**. When the later has reached its extreme position, one or the other of these springs will compress permitting further movement of the lever **76**. The output end of the link **78** is connected to an arm **80** rigidly secured to a rotatable shaft **82** journaled in bearings **84** on the base **50**. A second arm **86** also rigidly secured to the shaft **82** is connected by a link **88** with the operating spool **90** of the valve **52**. Thus, movement of the lever **76** through a small angle to one side or the other of its central position will correspondingly shift the spool **90** of valve **52** to its left or right-hand limit of travel. Thereafter, further movement of the lever **76** compresses one or the other of the springs within the collapsible link **78**.

Lever **76** is rigidly secured to a shaft **92** journaled in a bearing block **94** and projecting from the opposite side thereof to engage within the central bore of an operating arm **96**. The relation between the shaft **92** and the arm **96** is illustrated in FIG. **10**. The arm **96** is pivoted to the base in a block **98** and has a link **100** pivoted at its upper end for exerting varying degrees of compression upon the spring of the adjustable relief valve **54**. The shaft **92** has abutment surfaces **102** and **104** which contact abutments **106** and **108** respectively in the arm **96** after a predetermined degree of angular motion of the shaft **92**. Thus, when the spool **90** of four-way valve **52** has been shifted to one extreme or the other, one pair of these abutments will come in contact and further angular motion of the shaft **92** will shift the link **100** to increase the compression of the spring in the relief valve **54**.

Referring now to FIG. **6**, there is illustrated the spring mechanism which offers differing force-displacement rates to oppose the manually varied pilot pressure at the opposite end of the main valve spool. FIG. **6** illustrates a construction which is duplicated at each end cap **12**. With a spool **16** which has square-end heads along its length, such as that illustrated at **110**, it is necessary under conditions where the motor is lightly loaded that the opening movements of the spool be opposed by a very high-rate spring. Such a spring is illustrated in the assembly of discs **112** which are positioned upon a pin **114** threaded at **116** in the end cap **12** and provided with a lock nut **118** and a closure cap **120**. The disc springs **112** are retained on the pin **114** by a stop member **122** having a slot **124** which receives a pin **126** pressed into the pin **114**.

The end of the spool **16** abuts a washer **128** which contacts a helical spring **130**, the opposite end of which rests within the end of the cap **12**. This helical spring and its counterpart at the opposite end cap act as conventional preloaded centering springs. The spool **16** has a recess **132** in its end which receives a piston **134** having an abutment face **136** on its outer end for the purpose of contacting the stop member **122** to compress the disc springs **112**. The chamber **132** is connected by a central passage **138** in the spool **16** with that groove or grooves on the spool which are in communication with the high-pressure supply and it thus serves to maintain high pressure within the recess **132** at all times. A check valve **140** prevents displacement of fluid out of the chamber **132** through this path and the only escape from the chamber is that which occurs through leakage past the piston **134**, the fit of which within the cylinder **132** is selected for this purpose. A snapring **142** limits the outward movement of the piston **134**.

The force-displacement rate of the spring **130** will be much lower than that of the spring assembly **112**. Depending upon the distance which the spool **16** must travel from its centered position to the point where flow starts to occur, the pin **114** will be adjusted so that stop **122** is just contacted by the abutment surface **136** at this point. During this travel, only the low rate spring **130** opposes the pressure applied by the pilot control circuit in the opposite end cap. This motion will, of course, have been caused by the operator shifting the handle at the remote control station, first shifting the four-way directional valve so as to exhaust the end cap **12** in which the spring action is occurring and then to apply a progressively increasing pressure to the opposite end cap through progressively increasing the compression of the spring of the relief valve **54**. When the abutment **136** and stop **122** come in contact, the disc springs **112** commence to oppose the pilot pressure in the opposite end cap and from this point on, both the low-rate and the high-rate springs are opposing the pilot pressure. Thus, for a given increment of pilot pressure, only a much smaller movement of the spool **16** occurs. In this way, precise modulation of the flow at low motor loads may be achieved and, of course, at high motor loads as well. The chamber **132** is maintained at full system pressure by admission of fluid through passage **138** and check valve **140**. The area of the piston **134** is so chosen in relation to the system pressure and the maximum force of the fully compressed disc springs **112** that piston **134** is maintained fully projected under these conditions.

Upon further elevation of the pilot control pressure in the opposite end cap, however, the force applied over the full end area of the spool **16** is sufficient to overcome system pressure applied over the area of piston **134** and thus further travel of the spool **16** is permitted to occur. This travel is opposed only by the low-rate spring **130** and by the hydraulic resistance to the exhausting of fluid from chamber **132** past the clearance space around piston **134**. Thus, the piston **134** acts as a dash pot to control the further movement of spool **16**. The rate of travel of the spool **16** is under the additional control of one or the other of the adjustable restrictors **44**. In this way, reliable modulating control of the main directional valve **24** is achieved for both conditions of light and motor loading. Under light loads, almost the full range of flows must be modulated through only a small percentage of the total range of spool travel after opening. Under heavy loads, however, when very little throttling through the directional valve is required, the range of modulation extends throughout the full travel of the spool after opening. The net effect of this type of dual-rate spring mechanism is to give the operator a natural feel at the control handle. When the load is light, modulation of the flow rate or motor speed from zero to full speed is achieved through only about half the range of control lever travel. At heavier loads, a similar modulation of flow rates requires substantially full travel of the control lever to achieve full speed, and in addition there is a "softer" initial control during which the acceleration of the loaded motor occurs.

The pilot control circuit may be overridden for the purpose of establishing a maximum limit upon flow through the main valve. For this purpose, a bypass valve **144** is mounted on the side of the block **30**, its internal construction being illustrated in FIG. **5** and diagrammed within the block **144** in FIG. **4**. The valve in FIG. **5** is illustrated in its open position in which the spool **146** connects the two ports **148** and **150** which are connected by lines **152** and **154** with the arms **46** and **48** respectively of the pilot control circuit. The spool **146** is biased to the left in FIG. **5** by a spring **156** and the ends of the spool are connected so as to be responsive to the pressure drop across an orifice in the main exhaust flow from the valve **24**. This orifice is illustrated in FIG. **4** at **158** and may be conveniently located within the exhaust connection flange **20**. Suitable conduits such as **160** and **162** serve to sense the pressure in the flange **20** ahead of and beyond the orifice **158** and transmit these pressures through the block **30** to the left and the right ends of the spool **146** in FIG. **5**.

In many applications, it is desirable to provide overload protection for the motor which is connected to the flange **22** and for this purpose a relief valve block **164** may be mounted atop the flange **22**. This carries internally a single main relief valve **166** connected across the main motor connection A and B. A shuttle valve **168** is arranged to connect the sensing port **170** of relief valve **166** with which ever of the main ports A and B is at the moment under the higher pressure. Conveniently, a drain conduit **172** may lead from the relief valve **166** to join with the conduit **162** for communication with the exhaust line.

The invention thus enables a standard sliding spool directional valve with a slight modification to be used for a modulating control in a central hydraulic system and to permit pilot control from a remote distance through manipulation of a simple pressure relief valve. At the same time, fully effective modulation is achieved, both under conditions requiring a high degree of throttling at the directional valve as well as under conditions which require very little throttling.

I claim:

1. A valve for controlling fluid flow comprising a body having a laterally ported bore, a spool slidable in the bore to control flow between the ports of the bore, pressure actuated pilot means for shifting the spool, a high-rate spring means positioned to oppose the pilot means during a portion of the spool travel, a low-rate spring means positioned so as to oppose the pilot means during at least the remainder of the spool travel, a yieldable abutment contacting one end of the high-rate spring, fluid-operated means for controlling the yieldable abutment, and means for applying and modulating a supply of pressure fluid to the pilot means.

2. A valve as defined in claim **1** wherein the means for controlling the abutment comprises a piston slidable in the spool and supporting the yieldable abutment.

3. A valve as defined in claim **2** wherein means is provided for supplying the pressure fluid to the piston to resist yielding of the abutment during compression of the high-rate spring.

4. A valve as defined in claim **2** wherein means is provided for allowing escape of pressure fluid from the piston after the high-rate spring is fully compressed.

5. A valve as defined in claim **1** which includes means for holding the abutment against yielding during the compression of the high-rate spring and allowing yielding after the high-rate spring is fully compressed.

6. A directional control valve comprising a body having a laterally ported bore, a spool slidable in the bore to control flow between the ports of the bore, pressure actuated pilot means for shifting the spool, there being one pilot means at each end of the spool, spring means at each end of the spool to oppose with a predetermined gradient the pilot means at the opposite end of the spool, a pilot fluid pressure circuit comprising a high pressure source, a pair of branches, each containing a fixed restriction and a first arm connected to one of the pilot means and a second arm, a pilot four-way directional valve connected to each of the second arms, a variable pressure relief valve, and an open exhaust path selectively connectable to either of the second arms by shifting the four-way valve, and a single manual lever for shifting the four-way valve in one direction or the other and connected to operate the relief valve during the latter portion of its travel in either direction.

7. A valve as defined in claim **6** wherein each branch includes a variable restrictor and a check valve in parallel in the first arm to limit the maximum rate of travel of the spool.

8. A valve as defined in claim **6** wherein there is provided a bypass valve which when open connects the two branches together, and means responsive to the flow rate controlled by the spool for actuating the bypass valve.

9. A directional control valve comprising a body having a laterally ported bore, a spool slidable in the bore to control flow between the ports of the bore, pressure actuated pilot means for shifting the spool, there being one pilot means at each end of the spool, means for applying and modulating pressure fluid at any desired level to one or the other of the pilot means, and spring means for opposing each pilot means and having a force-displacement rate which is high during initial flow across the spool and decreases thereafter.

10. A valve as defined in claim **9** wherein the spring means includes a series of disc springs for imposing a high force-displacement rate and a helical spring for imposing a lower rate.

11. A valve as defined in claim **10** wherein fluid pressure operated means is provided to allow travel of the spool while the disc springs are fully compressed.

12. A valve as defined in claim **11** wherein the fluid pressure operated means includes a piston in the valve spool and means for applying system pressure to the piston.