United States Patent

Hufeld et al.

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[45] **Aug. 1, 1972**

[54] **HIGH PRESSURE IMPLEMENT HYDRAULIC CIRCUIT**

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[22] Filed: **Jan. 4, 1971**

[21] Appl. No.: **103,867**

Related U.S. Application Data

[60] Division of Ser. No. 868,964, Oct. 21, 1969, Pat. No. 3,575,000, Continuation-in-part of Ser. No. 814,003, April 7, 1969, abandoned.

[52] **U.S. Cl.** **137/596.18**

[51] **Int. Cl.** **F16k 11/10**

[58] **Field of Search** 137/594.14, 594.15, 594.18, 137/625.6, 625.66

[56] **References Cited**

UNITED STATES PATENTS

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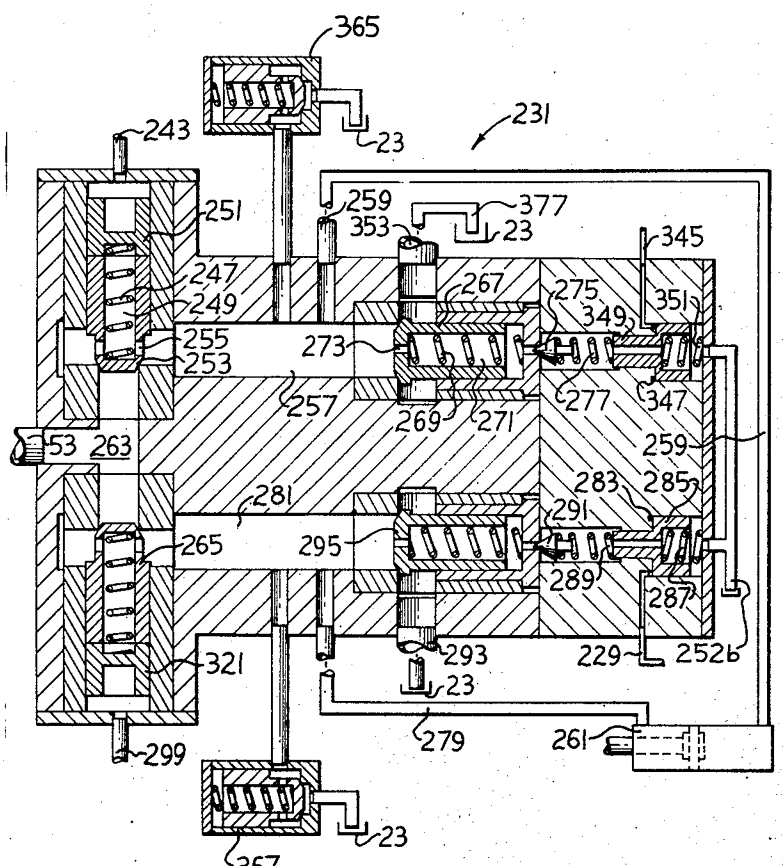
[57] **ABSTRACT**

A relatively low pressure hydraulic control system for a high pressure work output system in which a control valve is selectively actuated to position a variable displacement axial piston pump to control the flow in the work output system and to position a directional valve which controls the direction of work output. In such a system wherein two work outputs are provided, thereby requiring two control valves, a priority valve may be utilized in the low pressure system so that a signal from one of the control valves will always override a signal from the other control valve in controlling the axial piston pump.

3 Claims, 19 Drawing Figures



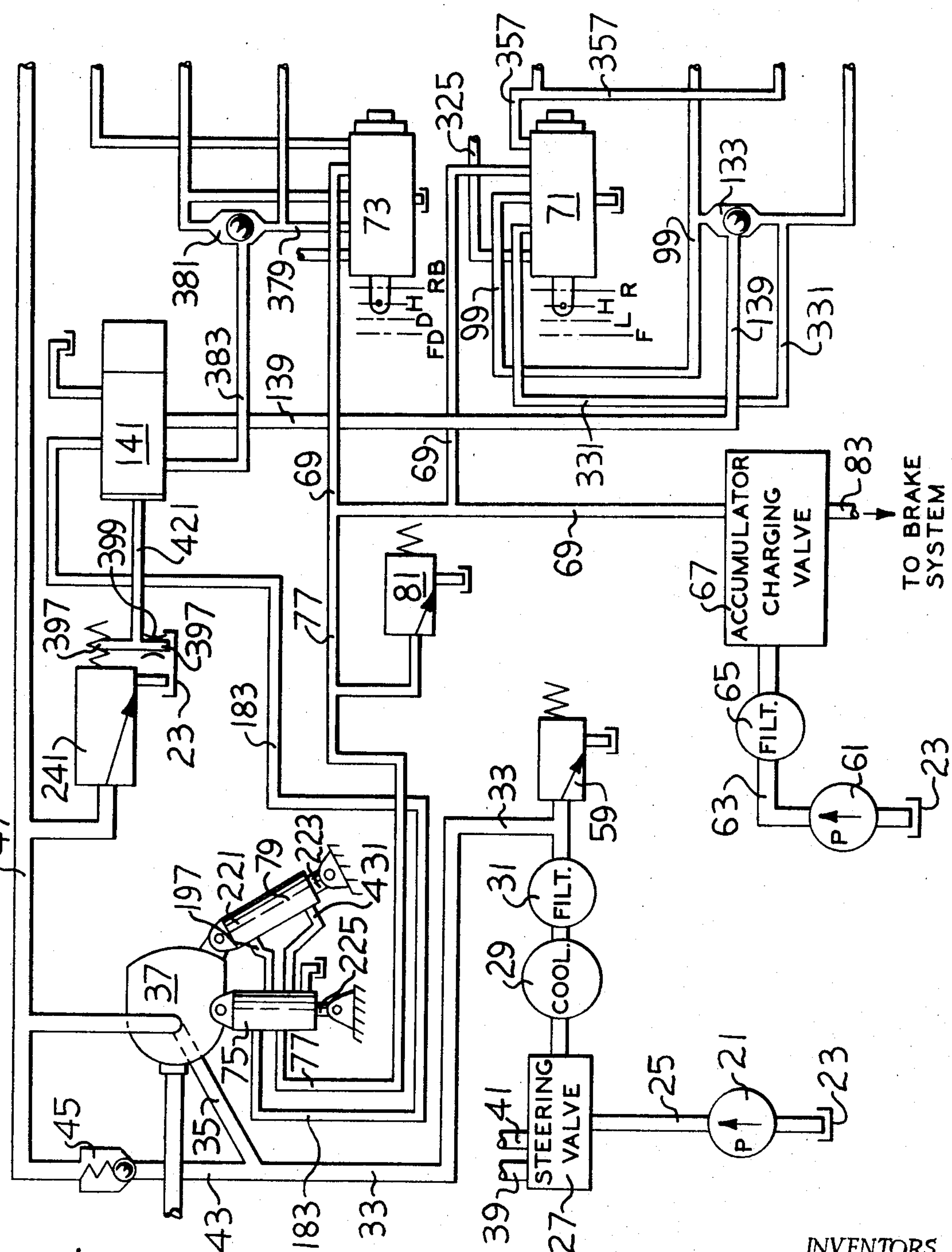


FIG. 1

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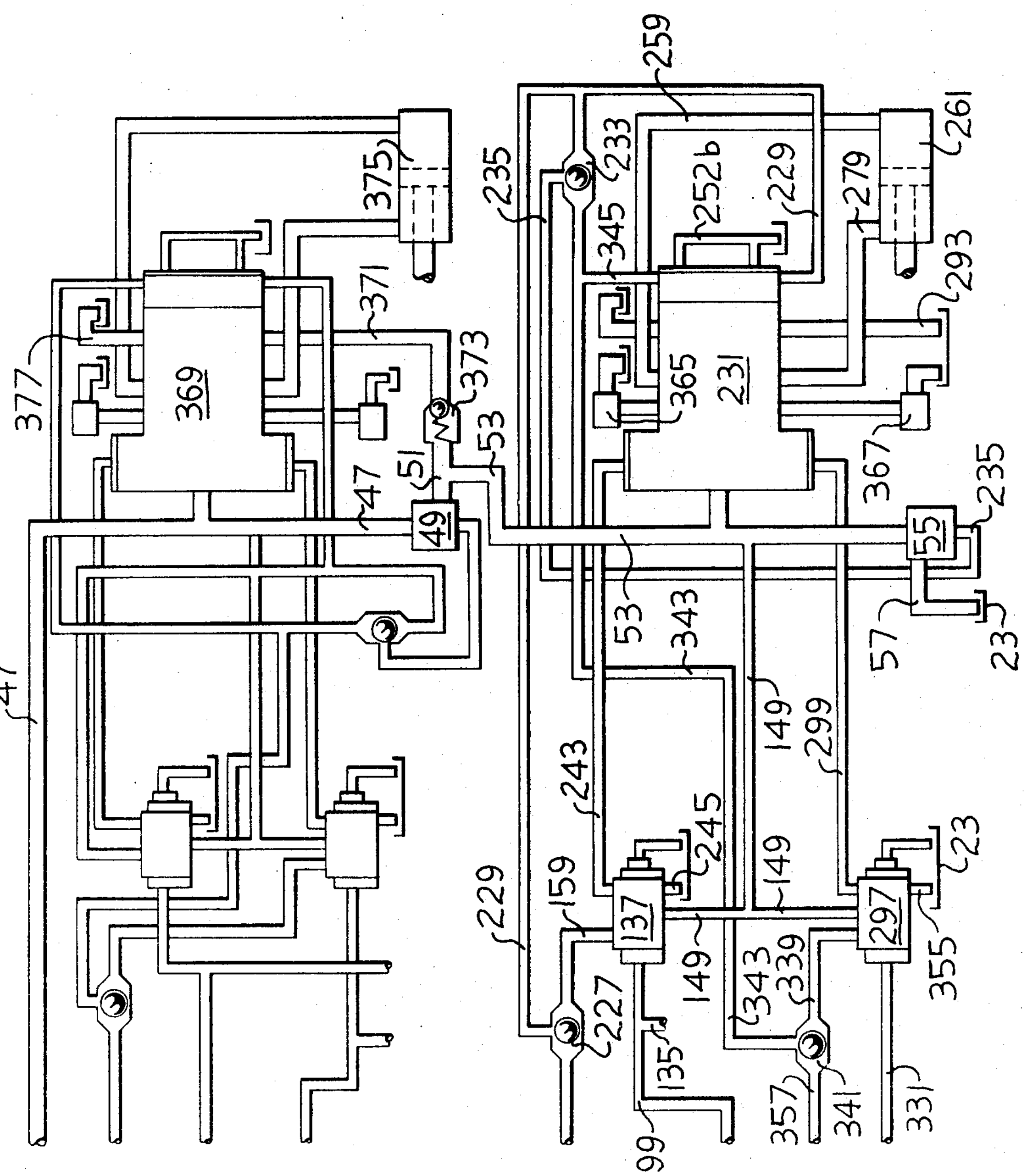

FIG. 2

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Fig. 3

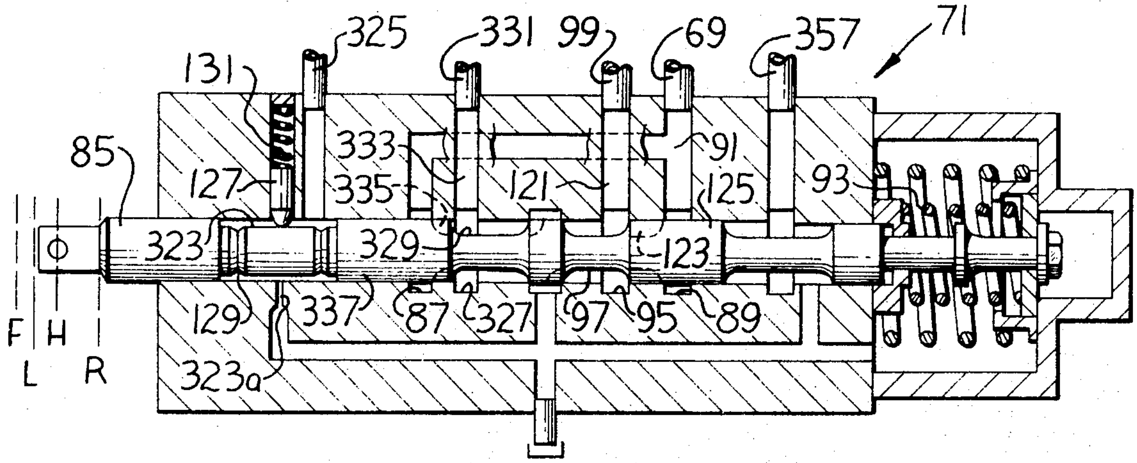

Fig. 4

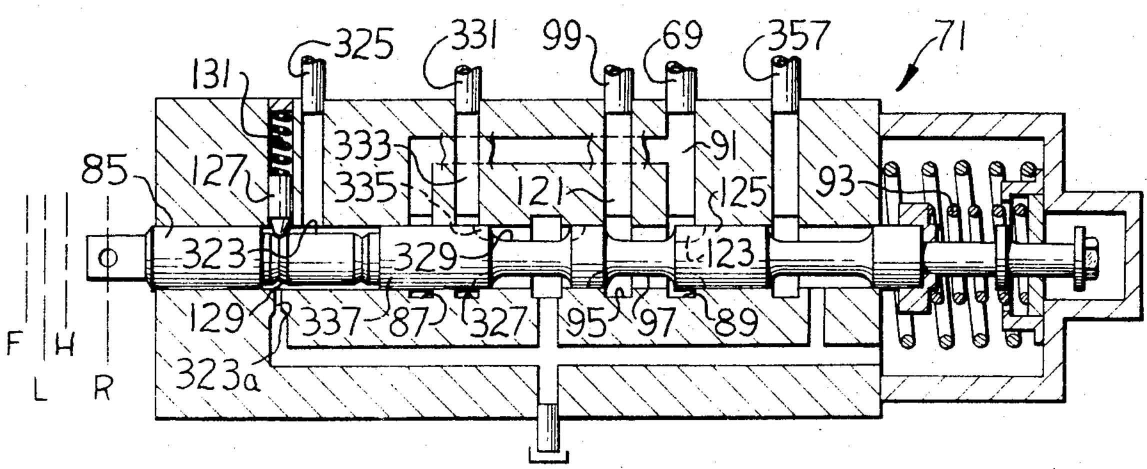

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FIG. 5

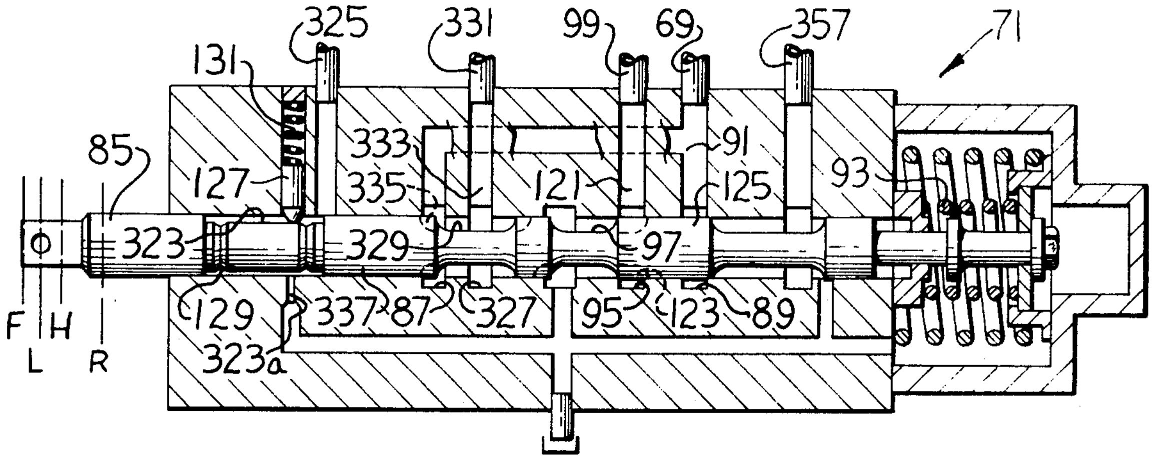

FIG. 6

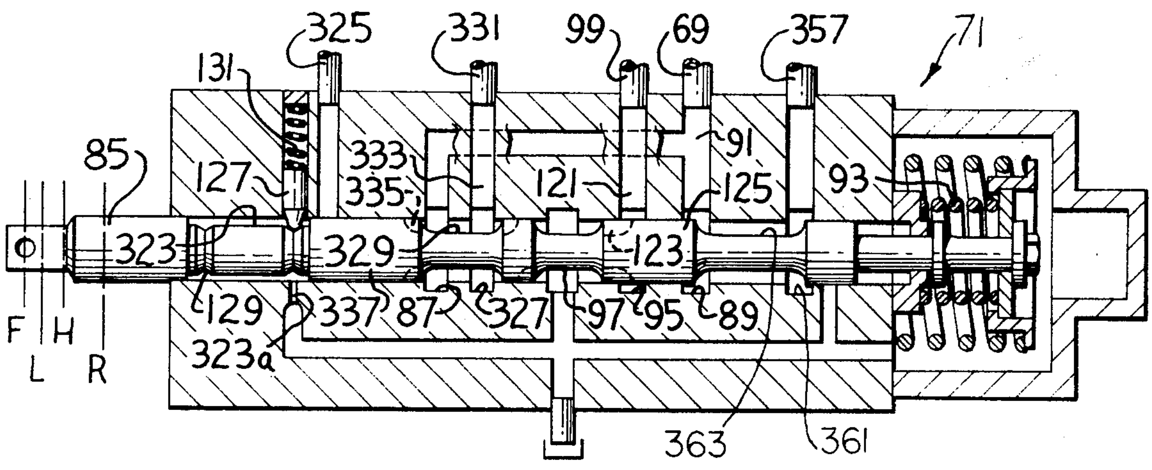

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Fig. 7

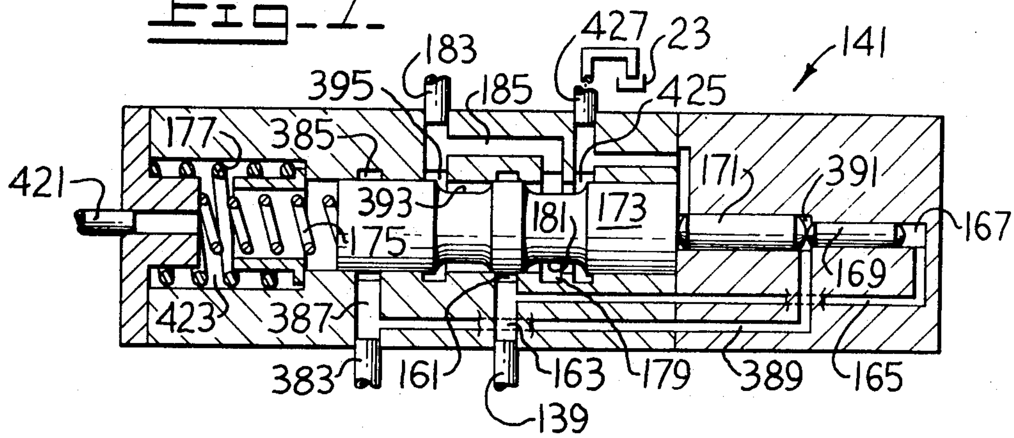

Fig. 8

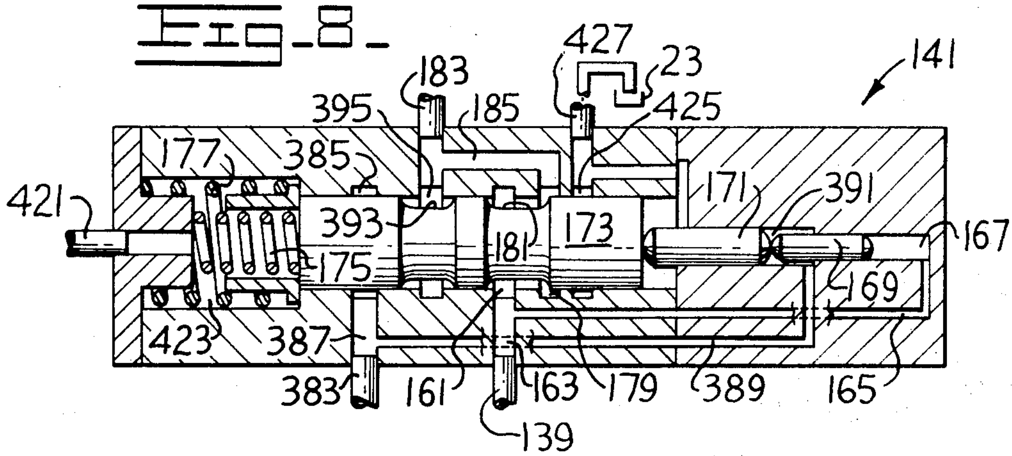

Fig. 9

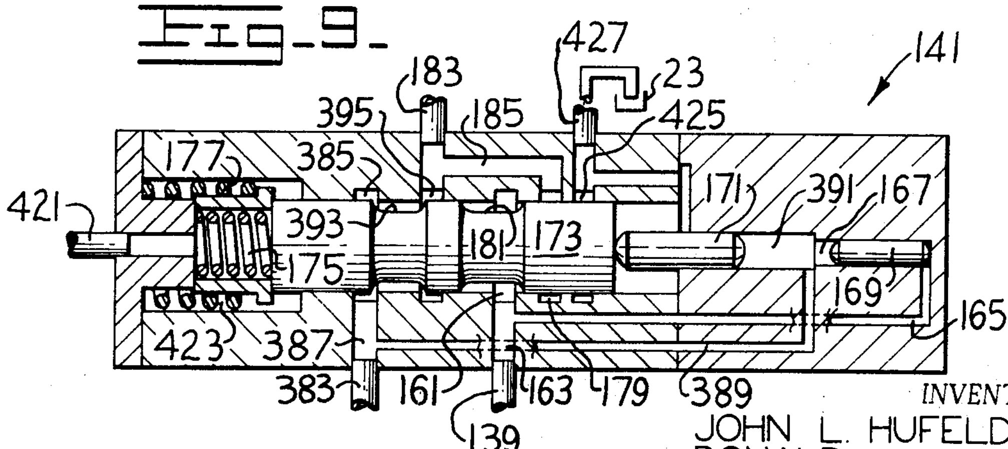

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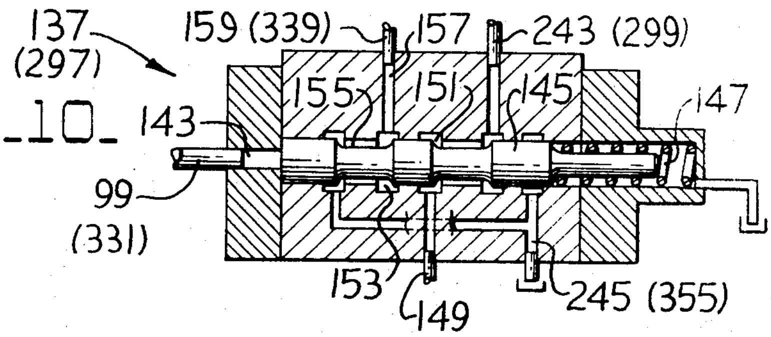
Fig_10_



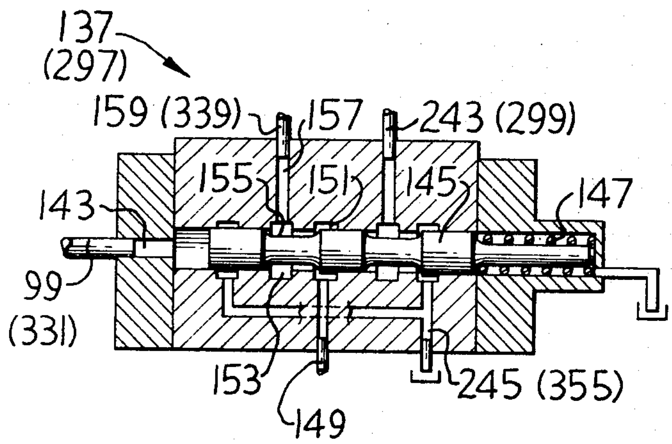
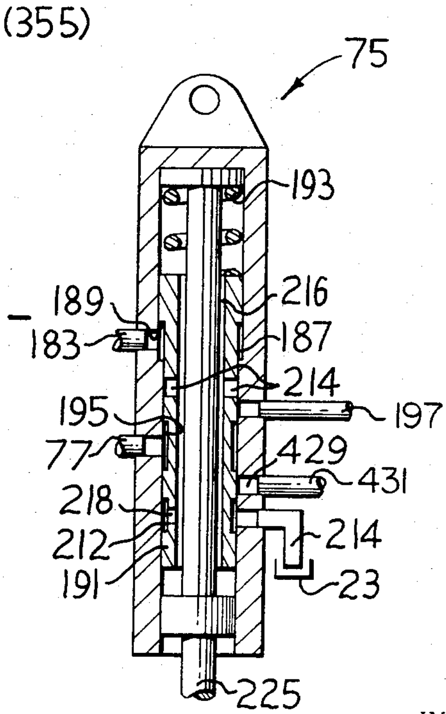


Fig_11_

Fig_12_



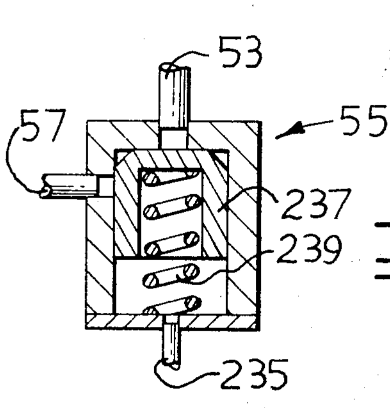


Fig_13_

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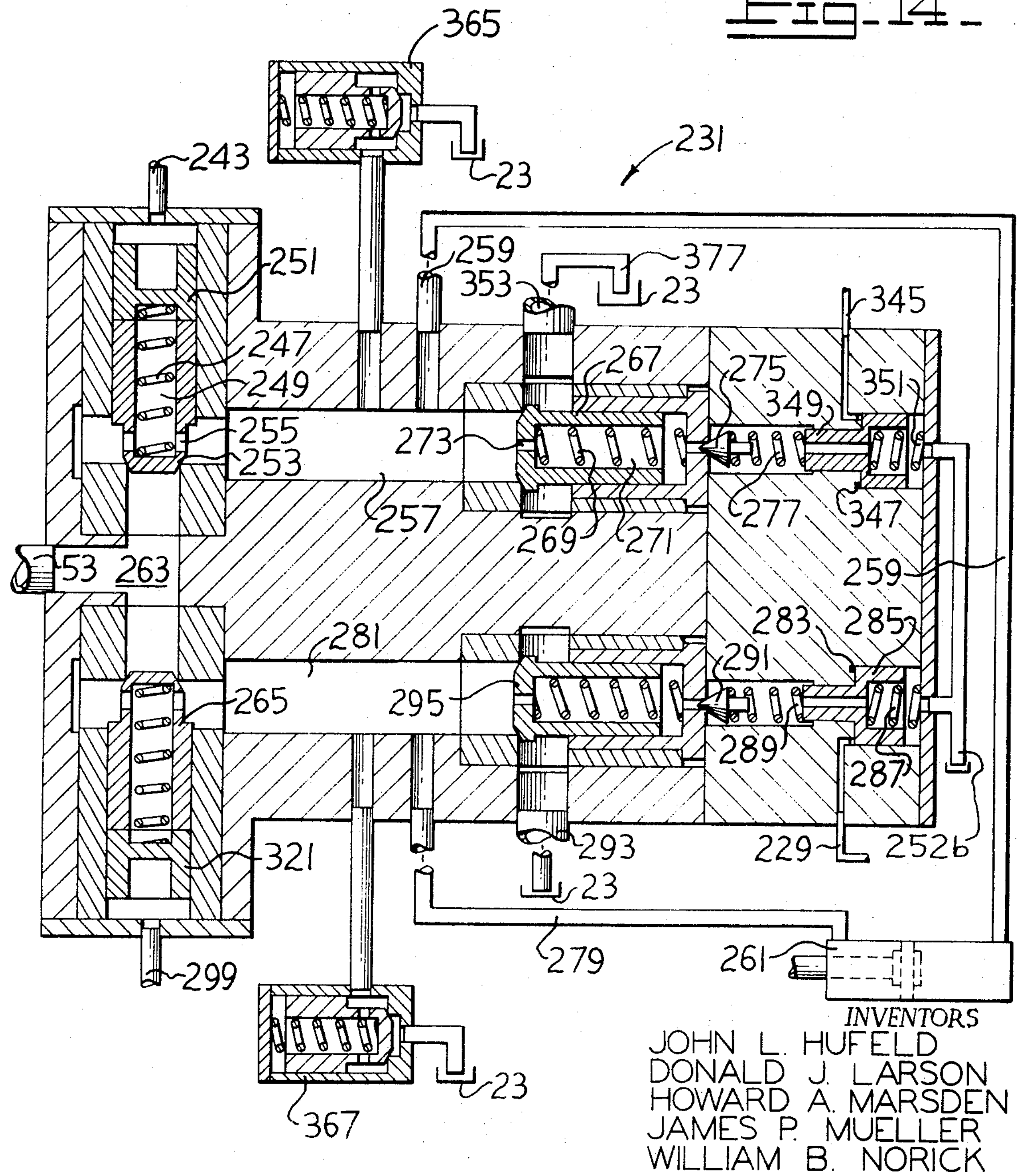
Fig. 14



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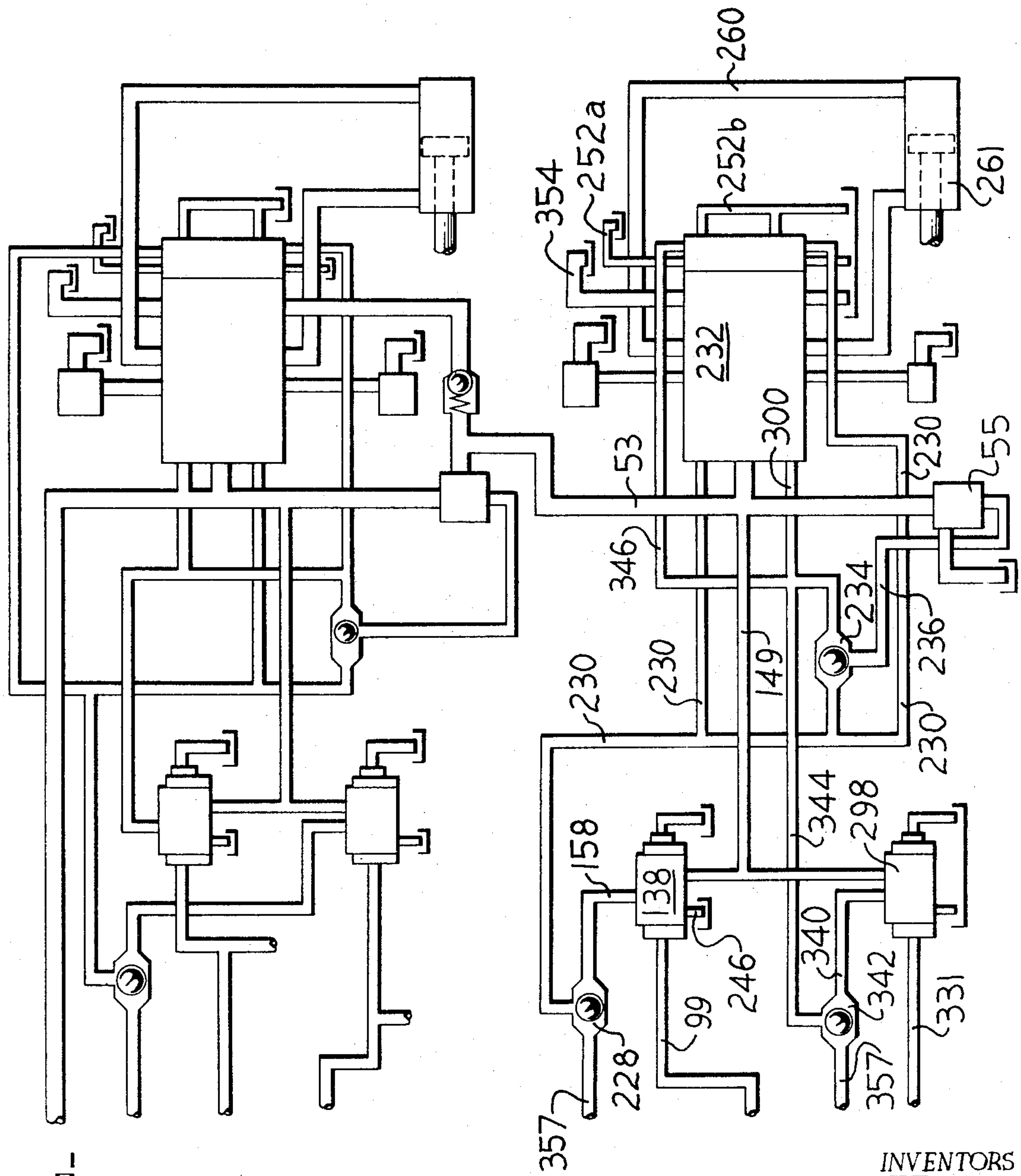

FIG. 15

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Fig. 16.

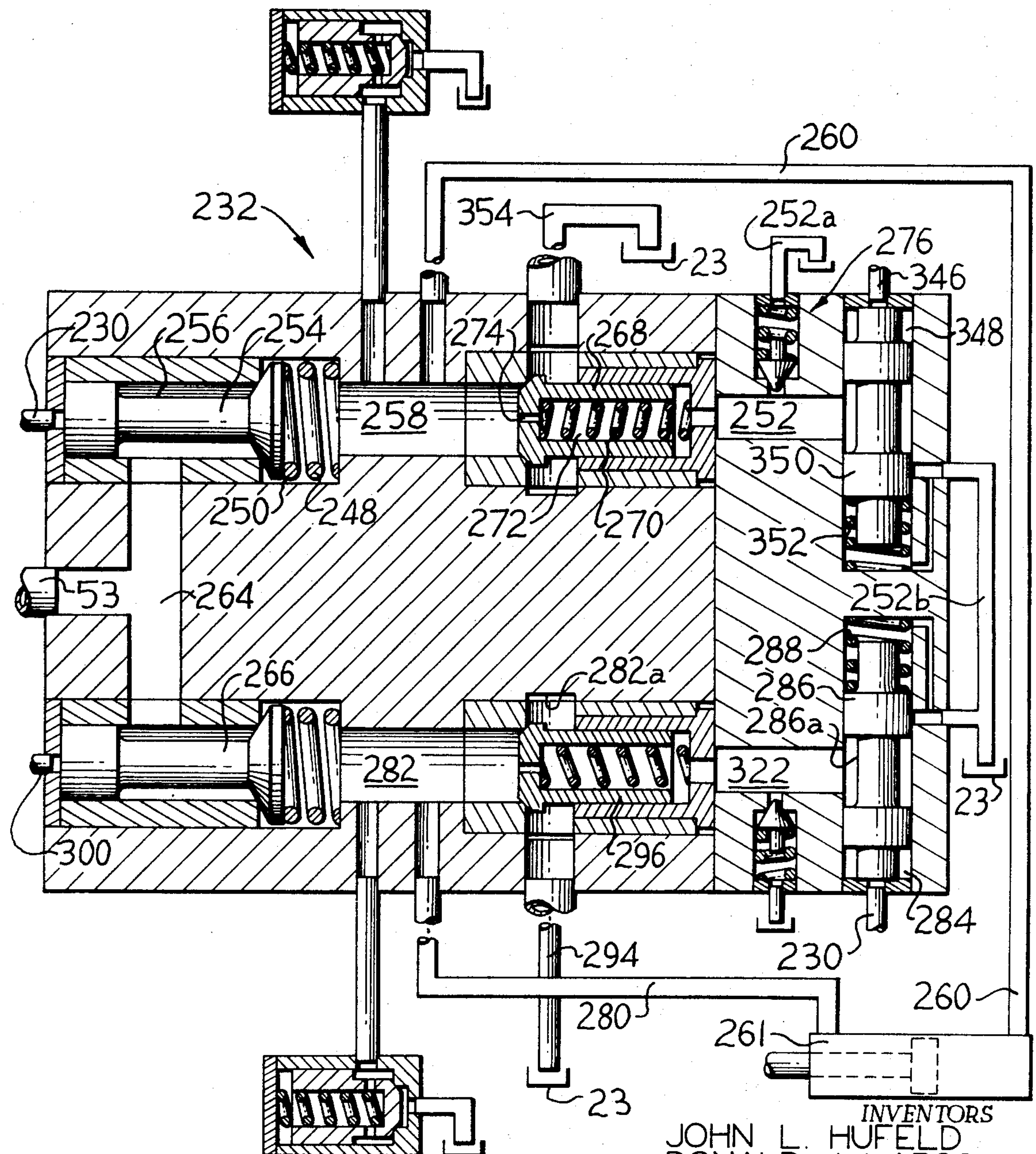

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FIG. 17

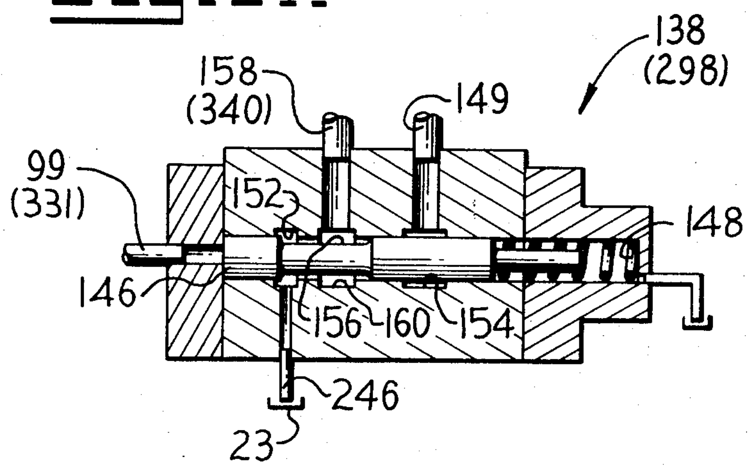

FIG. 18

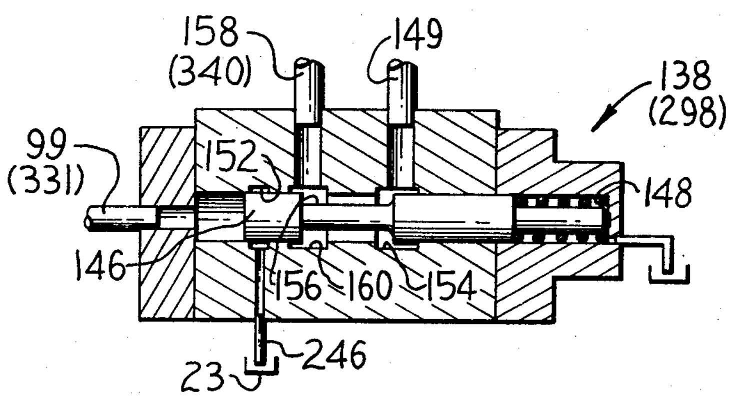

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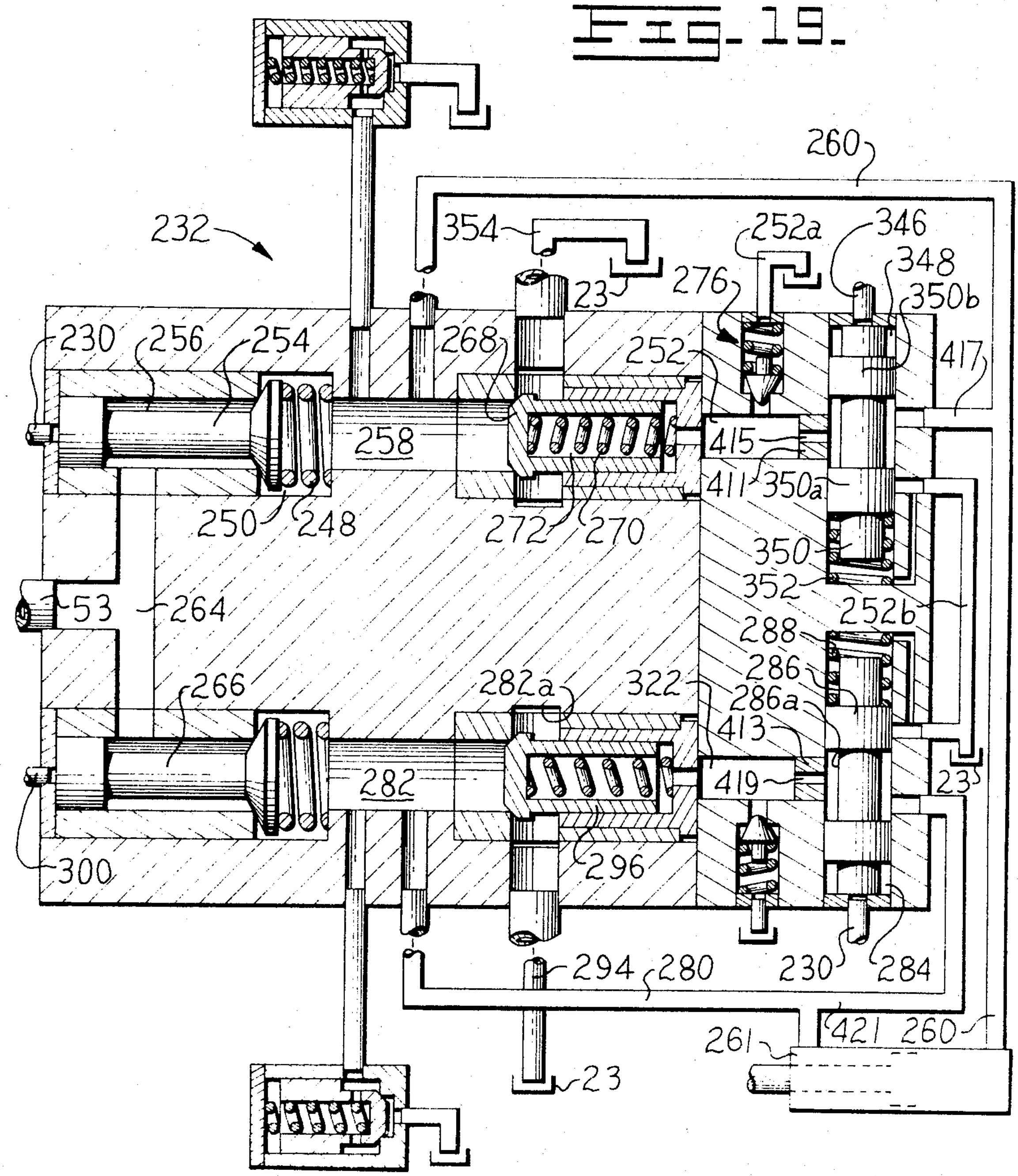
Fig. 19



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HIGH PRESSURE IMPLEMENT HYDRAULIC CIRCUIT

This application is a division of application Ser. No. 868,964, filed Oct. 21, 1969, now U. S. Pat. No. 3,575,000, which in turn is a continuation-in-part of application Ser. No. 814,003, filed Apr. 7, 1969 and now abandoned.

BACKGROUND AND SUMMARY OF THE INVENTION

As hydraulically controlled machinery becomes larger and more complex, the component parts used in controlling such machinery, such as cylinders, connecting lines, and valves, must be increased in size so as to withstand the high pressure forces which are developed both by the control system itself and by reaction forces from the work. With increased flow and higher pressures, leakage within directional valves which utilize cylindrical spools has become a problem and the costs of manufacture of such valves has greatly increased due to the close tolarances which must be maintained.

If the control systems are simply increased in size and the supply of fluid comparably increased, the excess flow must be throttled back to the tank or reservoir during times when all the flow is not desired to perform a particular function. The excess flow is converted to heat and wasted in the throttling process. Similarly, when the machinery is stalled, much energy is again wasted.

In machinery having articulated movement, it is quite desirable to reduce the number of large high pressure lines that are required to pass over the swivel joints. For this reason, directly operated control valves are not suitable for use on the large articulated machines since the valves would have to be mounted close to the operator's station, requiring a considerable number of high pressure hoses to pass over swivel joints.

The high pressure implement hydraulic control system of the present invention utilizes a variable displacement, axial piston-type pump that is capable of producing the increased flows and pressures that are necessary to operate large modern machinery. Although any desired machinery may be so operated, the system is especially adaptable to articulated machinery since it obviates the problem of high pressure lines passing over swivel joints.

Essentially, the invention involves the use of a high pressure machinery actuating circuit which is operated and controlled by a low pressure control circuit.

With this system, the operator can limit the flow necessary to perform a particular function reducing heat and energy losses — thereby making the system more efficient. Additionally, the control of pump displacement allows the utilization of directional valves having poppet-type spools, since the throttling is not done by the spools. Use of the poppet-type spools provides positive leakage control and decreases overall manufacturing costs of the directional valve packages. Precise control of the directional valve is achieved by the use of pilot-operated, vented check valve arrangements to control the actions of the poppet-type spools, and is synchronized with the rest of the pilot system so that the directional valves will function at the proper times.

Should the machinery become stalled, the pump is automatically adjusted to a low delivery rate while maintaining the same pressure or cylinder force, causing a reduction in heat and energy loss. In such a case, the engine power demand is also substantially reduced.

To reduce the number of high pressure lines crossing the swivel joints, the main control valves will be mounted close to the components they control and will be actuated by the low pressure, pilot system. The displacement of the variable displacement pump is also controlled by the pilot system and the control of the displacement and directional valve is synchronized so that the proper amount of supply fluid and pressure is available at all times.

It is therefore an object of this invention to provide a hydraulic system wherein the pressure and direction of flow in a high pressure circuit are controlled by a low pressure circuit.

It is also an object of this invention to provide a hydraulic circuit which may be used to control an articulated machine while not requiring a large number of high pressure lines to pass over the machine's swivel joints.

It is also an object of this invention to provide a high pressure machinery actuating circuit which is operated and controlled by a low pressure control circuit.

It is a further object of this invention to provide a hydraulic system capable of producing the high pressures required by today's machinery while maintaining a high degree of efficiency by constantly controlling the fluid flow in the system.

It is also an object of this invention to control the high pressure fluid flow in such a system by means of a low pressure actuated, variable displacement, axial piston-type pump.

It is also an object of this invention to provide a hydraulic circuit having high efficiency due to the reduction of heat and energy losses through the control of hydraulic flow necessary to perform a particular function.

It is a further object of this invention to provide a hydraulic circuit utilizing directional valves having poppet-type spools.

It is a further object of this invention to provide in such a circuit a system which produces positive leakage control while decreasing overall manufacturing costs of the directional valves.

It is a still further object of this invention to provide, in such a circuit, precise directional valve control through the use of pilot-operated, vented check valve arrangement to control the actions of poppet-type spools.

It is also an object of this invention to provide such a circuit wherein, when the machine is stalled, automatic adjustment to the circuit occurs to allow a low fluid delivery rate while maintaining a constant pressure.

Other objects of the invention will become apparent to those skilled in the art upon perusal of the following description in light of the accompanying drawings which illustrate preferred embodiments of the invention. Embodiments using similar or equivalent structure, which are included in the scope of the appended claims, will become obvious to those skilled in the art without departing from the present invention as defined in the appended claims.

BRIEF DESCRIPTION OF THE DRAWINGS

FIGS. 1 and 2, when combined by placing FIG. 2 to the right of FIG. 1, illustrate a complete schematic of the high pressure implement hydraulic circuit of the instant invention;

FIGS. 3–6 illustrate schematic sectional views of an operator-actuated control valve utilized in the subject hydraulic circuit, with the valve spool in various positions of actuation;

FIGS. 7–9 illustrate schematic sectional views of a priority valve which may be utilized in the subject circuit with the valve spool subjected to varying degrees of actuation;

FIGS. 10 and 11 illustrate schematic sectional views of a pilot valve in the subject hydraulic circuit, with the valve spool in unactuated and fully actuated positions respectively;

FIG. 12 illustrates a schematic sectional view of a servo valve in the subject circuit;

FIG. 13 illustrates a schematic sectional view of a sequence valve used in the control circuit; and

FIG. 14 is a sectional schematic illustration of a directional valve in the subject circuit;

FIG. 15 is a schematic view of a modified embodiment of that portion of the invention shown in FIG. 2;

FIG. 16 is a sectional schematic illustration of a modified embodiment of the directional valve shown in FIG. 14, utilized in the circuit of FIG. 15;

FIGS. 17 and 18 are modified embodiments of the valves illustrated in FIGS. 10 and 11, which may be utilized in the circuit shown in FIG. 15; and

FIG. 19 is a sectional schematic illustration of a modified embodiment of the directional valve shown in FIG. 16 which may be utilized in the circuit of FIG. 15 with slight modification thereof.

DETAILED DESCRIPTION

The following description relates to the use of a high pressure implement hydraulic circuit in any desired application. For ease in illustration, however, the description will refer to the use of the circuit on large loaders wherein lift and tilt circuits will be designed to utilize maximum pressures up to about 6,000 psi. Many modifications of the circuit, including deletion and duplication of elements, will be obvious to those skilled in the art when using the circuit for other applications.

It should be noted that the lift and tilt circuits in the system are, in many respects, duplicate circuits. Therefore, the following description will be essentially directed toward the lift circuit upon the understanding that similar actions cause similar results in the tilt circuit. Of course, where distinctions do exist between the circuits, they will be fully discussed.

Now referring to the schematic illustration of the circuit shown in FIGS. 1 and 2 together, a charging pump **21** drawing fluid from a reservoir or tank **23** passes the fluid through a line **25**, a steering valve **27**, a cooler **29**, a filter **31**, and lines **33** and **35** to a variable displacement, axial piston-type pump **37**.

When steering control valve **27** is actuated by the operator, fluid is directed from pump **21** to the steering cylinders (not shown) and back again through lines **39** and **41** so that the flow to pump **37** is not interrupted when the vehicle is being steered.

When pump **37** is in a neutral position, it requires very little flow or pressure through line **35**. In such a condition of operation, the fluid in line **33** passes into line **43** and through a bypass check valve **45** which allows the flow to bypass the pump and pass into the main supply manifold **47**. With the lift and tilt circuits, which will be more fully described below, in "hold" positions, fluid in line **47** will pass through a sequence valve **49**, lines **51** and **53**, sequence valve **55**, line **57**, and back to the tank **23**.

Although the pressure of the fluid delivered by pump **21** to line **33** is normally governed by a relief valve **59**, when pump **37** is in the neutral condition previously described, the pressure in line **33** is dictated only by line losses, thus reducing horsepower loss while maintaining full supply lines.

When either the lift or the tilt circuit is actuated, causing pump **37** to be stroked, pressure in line **47** will close check valve **45**, and that portion of the charging flow not needed to supply pump **37** will be returned to tank **23** through relief valve **59**.

A control pump **61** also draws hydraulic fluid from tank **23** for actuation of the various control valves in the implement hydraulic circuit. Fluid from the control pump is delivered through a line **63** and a filter **65** to an accumulator charging valve **67**. The fluid is then directed through the charging valve into a line **69** which delivers the fluid to a pair of closed-center control valves **71** and **73**. The fluid in line **69** is also delivered to a linear servo valve **75** through a line **77** and is then directed to an actuating cylinder **79**, when the servo valve is shifted in either one of two operating directions. A relief valve **81** maintains a maximum pressure of about **625** psi in the lines **69** and **77**.

Pump **61** also supplies fluid to the accumulator and brake circuit, when needed, through a line **83**.

As previously stated, it will be assumed that the invention is being applied to a loader having a lift circuit which is controlled by valve **71** and a tilt circuit which is controlled by valve **73**.

In FIGS. 3–6, there are illustrated schematic sectional views of a control valve which may be either lift control valve **71** or tilt control valve **73**, with the discussion directed to it as the former.

FIG. 3 illustrates the valve in its closed-center, unactuated, or hold condition wherein fluid entering the valve through line **69** is blocked from passage through the valve by suitable lands on the valve spool **85**.

RAISE

When the operator wishes to raise the loader bucket, he actuates the spool **85** to the position shown in FIG. 4 and labeled "R" in FIGS. 1 and 3 by means of any suitable linkage (not shown). Control fluid, maintained at 625 psi, is available to valve **71** through line **69**, wherein it is directed to annuli **87** and **89** by a passage **91**. Movement of the spool **85** to the "R" raise position overcomes the force exerted by a centering spring **93**. This causes pressurized fluid in annulus **89** to be communicated to an annulus **95**, via an annular groove **97**, and then into a line **99** via passage **121**. The pressure in line **99** can be controlled by the operator between the limits of 0 and 625 psi by means of variable area throttling slots **123** between land **125** and annular groove **97** on the spool **85**. The pressure in line **99** increases as the spool is moved to the right, and when spool **85** is in the position shown in FIG. **4**, i.e., its fully shifted position, control pressure in line **99** is 625 psi.

As will be explained later, the pressure in line **99** will be used to properly sequence the other control valves when it is in a range from 0 to 300 psi. When it is between 300 and 625 psi, it is used to stroke the linear servo valve **75**, thereby creating an implement pump flow proportional to the control pressure, i.e., 300–625 psi.

Thus the operator has the ability to actuate the lift cylinder at any velocity he desires. Since valves **71** and **73** are essentially identical, it can be seen that the operator also has the ability to actuate the tilt cylinder at any desired velocity.

When the spool **85** is in the extreme raise position, as shown in FIG. **4**, a plunger **127** is forced into a detent groove **129** by a spring **131**. This action will cause the spool to be maintained in that position either until a signal is received by valve **71** in a manner to be explained later, or until the operator manually actuates the spool **85**, causing the plunger to ride up the sides and out of the groove. Thus the operator is relived of the necessity of holding the control lever in the raise position during that portion of the work cycle.

Referring again to FIGS. **1** and **2**, line **99** directs the pressurized fluid from valve **71** to a shuttle valve **133** which is shifted by the fluid so that the fluid is directed through a line **139** to a priority valve **141** which will be described later.

The fluid in line **99** is also directed to a line **135** leading to a kick-out valve (not shown), mounted on the lift cylinder, and to a pilot valve **137**.

Referring to FIGS. **10** and **11**, fluid from line **99** enters a chamber **143** in valve **137** and the force of the pressurized fluid works against the end of a spool **145** to move the spool to the right, against a force developed by a spring **147**, to a maximum position as shown in FIG. **11**. The pre-load on spring **147** is such that the spool **145** will attain the position shown in FIG. **11** when the pressure in chamber **143** reaches **150** psi.

As spool **145** moves to the right, pressure in a line **149**, which is the same as the pressure in line **53**, (see FIG. **2**) will be directed from an annulus **151** to an annulus **153**, via an annular groove **155**, into a passage **157**, and then to line **159** for a purpose to be described later.

With respect to priority valve **141**, FIGS. **7–9**, the line **139** directs the pressurized fluid to an annulus **161** via a passage **163** to which a passage **165** is interconnected. As the manipulation of spool **85** in valve **71** provides an increase in pressure in line **139**, that pressure is also sensed in a chamber **167** in the valve **141**, via passage **165**. The force in chamber **167** acts upon a piston **169** which, in turn, acts on a piston **171** and a spool **173**, to move the pistons and the spool against a force exerted by a spring **175**, to the position shown in FIG. **8**. When control pressure in line **139** reaches approximately 200 psi, the position of the pistons and spool as shown in FIG. **8** will be attained, and even through the signal pressure increases to 625 psi, the spool **173** will not move further to the left because of the small reaction area of piston **169** as opposed to the additional force of a spring **177** against which the slidable assembly now abuts. At that time, pressurized fluid in annulus **161** will be in communication with an annulus **179** via annular groove **181**. This fluid is then directed to a line **183** via a passage **185**.

The line **183** directs the fluid to the linear servo valve **75** as shown in FIGS. **1** and **12**. The fluid is directed into a chamber **187** wherein it works against a piston area **189** on a reciprocable spool **191** located in the valve **75**, to move the spool against a force developed by a spring **193**.

As this occurs the previously mentioned pressurized fluid in line **77** enters an annular groove **195** and is communicated to a line **197** by the annular groove. As shown in FIG. **1**, the line **197** communicates with the actuating cylinder **79**, and introduction of pressurized fluid into the head end of the cylinder will cause the cylinder housing **221** to move upwardly since the rod **223** is pivotably secured to the vehicle frame. Since the housing **221** is pinned to the body of pump **37**, an upward movement of he housing imparts an upward movement to the pump. As the fluid enters the head end of cylinder **79** through line **197**, fluid is expelled from the rod end of the cylinder through line **431**. When the fluid enters valve **75** through line **431**, it enters an annular groove **212** around the sleeve **191** which connects with a drain line **214** so as to return the fluid to tank **23**.

Since the body of valve **75** is fastened to pump **37** and a rod **225** of the valve is fastened to the vehicle frame, the resultant upward movement of the body of valve **75**, relative to the spool **191** therein, will close off the supply of pressurized fluid from line **77** to line **197**, thereby cutting off flow of fluid to cylinder **79** and stopping the rotation of the pump **37** at its desired displacement position. Thus, the pump delivers oil flow to the line **47** in proportion to the pressure of the fluid in line **183**.

When priority valve **141** is returned to a neutral position so that pressure is released from line **183**, the force of spring **193** will move sleeve **191** downwardly so that line **197** is connected to passages **214** which are in communication with an enlarged inner diameter **216** in sleeve **191**. The enlarged inner diameter is in communication with the annular groove **212** by means of a passage **218**. This allows fluid in the head end of the actuating cylinder **79** to be expelled to tank **23** since pressure in line **77** will be directed to the rod end of the actuating cylinder via annular groove **195**, passage **429**, and line **431**. This will effectively swivel the pump **37** back to its minimum displacement position.

Referring once again to the pilot actuated valve **137** (FIGS. **10** and **11**), when the valve is actuated to the position shown in FIG. **11**, the pressure in line **159** will set a shuttle valve **227** (FIG. **2**) to communicate the fluid from line **149** to a line **229** which, in turn, communicates with a main directional control valve **231**, the operation of which will be described later. The fluid in line **229** also connects with a shuttle valve **233** which is shifted so that the fluid from line **229** is communicated to a line **235** which is connected to the sequence valve **55**.

Referring now to FIG. **13**, it can be seen that fluid entering the sequence valve through line **235** acts against a poppet **237** and the combined pressure of the fluid and the force of a biasing spring **239** serve to set the poppet so as to block the flow of fluid from line **53** to return line **57**. This allows pressure to build up in line **53**, as required, up to a maximum of approximately 6,000 psi as determined by the setting of a relief valve **241** (FIG. **1**).

As shown in FIGS. **2** and **11**, movement of the spool **145** of valve **137** to the position shown in FIG. **11** will also open a line **243** to tank **23** via a line **245**. The opening of line **243** to tank will aid in the conditioning of valve **231** for operation as described below.

A sectional schematic view of the main control valve **231** is shown in FIG. **14**. Since the valve is a directional-type valve, all of the components that are in the lower half of a figure are identical to and function in a manner similar to the components shown in the upper half.

As described above, the line **243** will be fully open to tank when the spool **145** in pilot valve **137** is fully shifted. When this occurs, a combination of force from a spring **247** and hydraulic force from fluid in chamber **249** will move a piston **251** away from a poppet valve **253** and the poppet will function as a load check valve. The hydraulic fluid in chamber **249** enters the chamber via orifices **255** in the poppet and is communicated to the orifices via a chamber **257** and line **259** connecting the chamber to the head end of a lift cylinder **261**. The hydraulic pressure exerted by this fluid is created by the normal load reaction forces exerted by the bucket on the lift cylinder **261**. Thus, the pressure in chamber **249** forces piston **251** away from the poppet **253**, but also serves to hold the poppet against its seat.

As the operator strokes the control valve **71** so that a control pressure of at least 300 psi is reached, the pump **37** provides a flow through lines **47** and **53**. As a result, a build-up of fluid pressure in line **53** and passage **263** occures. When this pressure raises to a level slightly higher than the load reaction pressure in passage **257**, poppet **253** will move upwardly and allow a flow of fluid from passage **263** to passage **257**, which fluid is then directed to the head end of the lift cylinder **261** via line **259**.

During this operation, no fluid in passage **263** can move past poppet **265** for a reason which will become obvious later.

Also during this time, a check valve spool **267** will be maintained against its seat by the combined force of a spring **269** and fluid under pressure in chamber **271**. The fluid pressure in chamber **271** is gained by fluid entering the chamber from passage **257** via an orifice **273** in the head of the spool. Since the fluid in chamber **271** will act upon an area of the spool equal to the area against which the fluid in chamber **257** will act, the spring **269** will normally cause the check valve spool to be retained against its seat. The pressure in chambers **257** and **271** will be blocked from tank by a poppet-type relief valve **275**, which will be maintained against its seat by the force exerted by a spring **277**, unless the pressure in chamber **271** exceeds the pressure setting predetermined by the spring **277**.

The flow of high pressure fluid through line **259** and into the head end of lift cylinder **261** will cause fluid to be expelled from the rod end of the cylinder into a line **279**. The line **279** communicates with a chamber **281** in the bottom half of valve **231**.

As previously stated, line **229** is connected to line **53** through the pilot valve **137** connection of lines **159** and **149**. Therefore, pressure developed by the pump **37** will be exerted in a chamber **283** in valve **231** and will work against a piston **285**, moving it to the right against force developed by a spring **287**. This relieves the preload on a spring **289** and thereby relieves a poppet valve **291**, allowing the fluid flowing into chamber **281** from line **279** to pass into tank **23** via a line **293**. This occurs since the relief of the poppet-type relief valve allows the unseating of a check valve **295**, similar to valve **267**, in the lower half of the directional valve **231**.

Should cylinder **261** attempt to run faster than that velocity dictated by the pressure flow of pump **37**, the pressure in line **53** decreases since the pump is not actuated to an amount required for that movement. When this pressure decreases to about 200 psi, the force of spring **287** overcomes the hydraulic force in chamber **283**, and piston **285** moves to the left, setting the poppet-type valve **291**, so that the lower check valve **295** reseats and restricts the flow of fluid exiting through line **293**. Pump pressure will then build up to a level required to position piston **285** so that the check valve **295** meters a flow rate out of cylinder **261** equal to the pump input of the cylinder.

Throughout the above sequence of events, pressure in line **149** is provided through a spool in a pilot valve **297** to a line **299**. Valve **297** is similar to valve **137** and, during this time, the spool in valve **297** is in the position shown in FIG. **10**. This causes piston **321** to set poppet-type valve **265** (FIG. **14**) so as to prohibit flow of fluid from passage **263** into passage **281** since the area of piston **321** is large enough that the pressure in passage **263** cannot overcome it due to the smaller area of valve **265**. Thus, fluid in passage **263** cannot move into chamber **281**.

As mentioned previously, fluid under pressure in line **99** is communicated to a line **135** which delivers the fluid to a kick-out valve of conventional design mounted on the lift cylinder. When the lift cylinder reaches its maximum stroke, a cam will open a spool in the kick-out valve so that flow is provided to an annular chamber **323** (FIG. **4**) around spool **85** of valve **71** from a line **325**. Pressure will build in chamber **323** due to the orifice **323a** in the drain passage of the valve **71**, to counteract the force from spring **131** and retract plunger **127**, allowing the force of spring **93** to return the spool **85** to the neutral position, thereby stopping the raise cycle.

ALTERNATE EMBODIMENT

The total hydraulic circuitry for the alternate embodiment can be seen by separating FIGS. **1** and **2** and placing FIGS. **1** and **15** in a similar end-for-end position. The valves illustrated in FIGS. **16–18** are those utilized in the alternate embodiment.

When it is desired to put the machine of the alternate embodiment in the raised mode, valve **71** is actuated in the manner previously described, with identical results relative to line pressurization, priority valve actuation, etc. Pilot flow from line **69** is directed to line **99** which connects with shuttle valve **133** to direct pilot fluid to the valve **141** and to a second stage pilot valve **138** which replaces the previously described valve **137**. Valve **138** is shown in cross-section in FIGS. **17** and **18** in the unactuated and fully actuated positions respectively. Pilot pressure in line **99** will shift the valve **138** so that pump pressure in line **149** will be directed to a line **158** via an annulus **154**, an annular groove **156** around spool **146**, and annulus **160**, since spool **146** is shifted to the right against force from spring **148**, as shown in FIG. **18**.

With line **158** at pump pressure from line **149**, the shuttle valve **228** will be shifted so that line **158** is communicated with a line **230**. The line **230** communicates with the modified directional control valve **232** and a shuttle valve **234**. When line **230** is pressurized, shuttle valve **234** is shifted so that line **230** will be connected to a line **236** to set the sequence valve **55** for proper operation of the directional valve **232**.

The modified directional control valve **232** is shown in detail in FIG. **16**, and pressurization of line **230** will condition the valve for actuation of lift cylinder **261** as follows:

A check valve spool **254** is moved to the right against a force from a spring **248** so that pump fluid under pressure from line **53** and passage **264** can get into passage **258**, via an annular groove **256** around spool **254** and a chamber **250** containing the spring **248**. Fluid under pressure in chamber **258** will be directed to a chamber **272**, through an orifice **274**, so that pressure can aid the force of a spring **270** in blocking a check valve **268** in the closed position. Thus, fluid in passage **258** will be directed to lift cylinder **261** via a line **260**.

The poppet-type relief valve, shown generally at **276**, will protect the circuit from any over-pressure conditions in that when a predetermined pressure is reached, it will open and vent the spring chamber **272** to tank through passage **252** and line **252**a.

Since spool **350** will not be under the influence of a pilot pressure, the spring **352** will keep it in the position shown in FIG. **16** to prevent communication between passage **252** and a line **252***b*.

Pressurization in line **230** is also directed to a chamber **284** and will move a spool **286** upwardly against force from a spring **288**, Upward movement of the spool **286** will connect a passage **322** to tank line **252***b* via an annular groove **286***a* around spool **286**. This will cause the chamber behind the load check valve **296** to be at tank pressure so that the load check valve can be opened, thereby connecting line **280** to the tank line **294** via a passage **282** and an annulus **282***a*. Thus, fluid can be expelled from the rod-end of the lift cylinder **261** during the raise mode when pump fluid is directed to the head end of the cylinder as described above.

LOWER

When the operator wishes to lower the loader bucket, he actuates the spool **85** to the position shown in FIG. **5** and labeled "L" in FIGS. **1** and **3**, overcoming the centering force of spring **93**.

The 625 psi control fluid available to valve **71** through line **69** is again directed to annuli **87** and **89** via passage **91**. The pressurized fluid in annulus **89** is blocked by land **125** on the spool **85**; on the other hand, the fluid in annulus **87** is delivered to an annulus **327** via an annular groove **329**. The fluid is then passed into a line **331** through passage **333**. The pressure in line **331** can be controlled by the operator between the limits of 0 and 625 psi by means of variable area throttling slots **335** between land **337** and annular groove **329** on the spool. The pressure in line **331** increases as the spool is moved to the left, and when spool **85** is in the position shown in FIG. **5**, control pressure in line 331 is 625 psi.

When the control pressure in line **331** is in a range from 0 to 300 psi, it is used to properly sequence the other control valves and when it is between 300 and 625 psi, it is used to stroke the linear servo valve **75**, thereby creating an implement pump flow proportional to the controlled pressure, i.e., 300–625 psi.

Thus the operator has the ability to lower the loader bucket at any velocity he desires.

Referring again to FIGS. **1** and **2**, line **331** directs the pressurized fluid from valve **71** to shuttle valve **133** which is shifted by the fluid so that the fluid is directed to line **139** and priority valve **141**. The priority valve is actuated by the control fluid in the same manner as previously described for the raise mode, relative to FIG. **8**.

The fluid in line **331** is also directed to a pilot valve **297** which is identical in construction to the pilot valve **137** which was previously described. As shown in FIGS. **10** and **11**, the line reference numerals in parentheses refer to those lines leading to and from valve **297** which are distinct from the lines leading to and from valve **137**. Since the interior structure of the valves are identical, however, the same reference numerals are used in the following discussion wherever possible. Similar use of parentheses will be used relative to FIGS. **17** and **18** when describing the alternate embodiment.

Fluid from line **331** enters chamber **143**, in valve **297**, and the force of the pressurized fluid works against the end of spool **145**, moving the spool to the right against the force developed by spring **147**, to the position shown in FIG. **11**. The preload on spring **147** is such that the spool **145** will attain the position shown in FIG. **11** when the pressure in chamber **143** reaches 150 psi.

As spool **145** of valve **297** moves to the right, pressure in line **149** (which is the same as pressure in line **53**) will be directed from an annulus **151** to an annulus **153** via an annular groove **155**, into a passage **157**, and then to a line **339**.

The fluid in line **339** sets a shuttle valve **341** which then directs the fluid to a line **343**. Fluid passing through line **343** then sets shuttle valve **233** and, passing through line **235**, sets sequence valve **55** in the manner previously described relative to FIG. **13**. Additionally, the fluid in line **343** passes through line **345** to a chamber **347** in control valve **231** (FIG. **14**) where it serves to actuate a piston **349** against the biasing force of a spring **351**, thereby relieving poppet valve **275**. This allows fluid flowing into chamber **257** from line **259** and the head end of lift cylinder **261** to pass into tank **23** via a line **353**. This occurs since the relief of the poppet-type relief valve **275** allows the unseating of check valve **267**.

As shown in FIGS. **2** and **11**, movement of the spool **145** of valve **297** to the position shown in FIG. **11** will open line **299** to tank **23** via a line **355**. Since valve **137** is in he unactuated position shown in FIG. **10**, line **149** provides pressure to a line **243** through the valve so that load check valve **253** in the main directional control valve **231**, is maintained in the closed position. On the other hand, since line **299** is vented to the tank through valve **297**, which is in the position shown in FIG. **11**, load check valve **265** of the directional valve allows fluid in chamber **263** to pass into chamber **281** and the rod end of lift cylinder **261** via line **279**.

In other words, when the operator places spool **85** of control valve **71** in the lower position, fluid passes through the main directional control valve **231** in the opposite direction to that which occurs when spool **85** is placed in the raise position.

ALTERNATE EMBODIMENT

With reference to FIGS. **1** and **15–18**, when the valve **71** is shifted to the "L" or lower position, the second stage pilot valve **298** will be activated, causing pressure in line **149** to be transmitted to a line **340** to shift a shuttle valve **342**. This connects line **340** with a line **344** which, in turn, connects with a line **300**, a line **346**, and shuttle valve **234**.

As shown in FIG. **16**, the line **300** will direct pressure so as to shift a check valve spool **266** which is identical to the spool **254**. Pressure in line **346** will be directed to a chamber **348** to move spool **350** downwardly to vent the chamber behind the load check valve **268**. Due to these events, pump flow from line **53** will be directed to passage **282** and since that passage will be blocked from the line **294** by load check valve **296** in a manner similar to the way that load check valve **268** blocked line **354** in the raise mode, pump fluid will be directed to the lift cylinder **261** via line **280**.

Movement of spool **350** downwardly, due to the flow in line **346**, causes passage **252** to be vented to line ***252b***, and thereby to tank **23**, to unload the check valve **268** and permit fluid being expelled from lift cylinder **261** to be returned to tank via line **260**, passage **258**, and a line **354**.

FLOAT

When the operator moves the control spool **85** of valve **71** to the float position shown in FIG. **6** and labeled "F" in FIGS. **1** and **3**, the bucket is allowed to move up or down as external forces dictate and flow from lift cylinder **261** can pass to and from tank **23** as needed.

With the control spool in the position shown in FIG. **6**, line **69** is connected to a line **357** via annuli **89** and **361** and annular groove **363**. Thus, a control signal of **625** psi is sent to shuttle valves **341** and **227**, setting the shuttle valves and passing the fluid to lines **229** and **343**, thereby causing relief of poppet-type relief valves **275** and **291** and setting of sequence valve **55**, all in the previously described manner.

Line **69** is also connected, via passage **91**, annulus **87**, groove **329**, annulus **327**, and passage **333** to line **331**. As previously described, the fluid in line **331** actuates priority valve **141** to create a flow in line **53** and also actuates pilot valve **297** to connect line **299** to tank, thereby releasing piston **321** so as to allow valve **265** (in valve **321**) to act as a check valve.

Since load check valve **253** is held closed because pilot valve **137** is not actuated and since sequence valve **55** is closed and poppet valve **291** is relieved as described, the flow in line **53** will pass load check valve **265** and valve **295** and return to tank via line **293**. Since the flow in line **53** is returning to tank, the fluid passing from line **149** to **339** through valve **297** will not have great enough pressure to effect the setting of shuttle valve **341**.

With the hydraulic system set in this manner, it can readily be seen that the system will allow a true "float" of the bucket but is in a condition to allow immediate force application to lift cylinder **261** when the operator actuates spool **85** of the control valve to the lower position. Thus, for example, the operator can allow the bucket to fall by means of gravity and immediately move the bucket to a digging position at the bottom of the fall with no time delay.

During any mode of operation, make-up valves **365** and **367** are used in a conventional manner to replenish the fluid supply, if necessary, when the pressure of fluid in either chamber **257** or chamber **281** falls below the pressure in tank **23**.

ALTERNATE EMBODIMENT

Once again with respect to the alternate embodiment, in FIG. **17**, with the valve **138** in its neutral position, the annulus **160** is connected to a tank line **246** via an annulus **152** and the annular groove **156**. Therefore, line **158** will be at tank pressure so that the shuttle valve **228** (FIG. **15**) can be shifted to condition the circuitry for the float mode. When valve **71** is shifted to the float position, pilot pressure from line **69** is directed to line **357** and will shift shuttle valves **228** and **342** so that lines **230** and **346** will be pressurized. This has the effect of unloading both the valves **268** and **296** to permit free flow of fluid within the directional control valve and allow the lift cylinder **261** to let the bucket drop to the ground by gravitational force alone.

TILT CIRCUIT

With respect to the tilt circuit, it should be noted that the dump and rack back modes, labeled "D" and "RB" (FIG. **1**), as controlled by valve **73**, are identical to and operate in the same manner as the raise and lower modes as controlled by valve **71**, with the exception that flow through the sequence valve **49** does not return to tank but instead communicates to line **53** as shown in FIG. **2**. Also, fluid flow from the lower half of directional valve **369**, in the tilt circuit, is directed to line **53** via a line **371** and a check valve **373**. The checkvalve **373** prevents high pressure fluid from flowing back into the low pressure side of the directional valve **369**. This allows activation of the lift system when the vehicle bucket is being dumped. When the bucket is being racked back, the upper half of valve **369** will be the load pressure side of the valve and fluid returning from the tilt cylinder **375** will go directly to tank via line **377** and will not be available to the lift system. Consequently activation of the lift system is interrupted during the rack back portion of the work cycle.

In priority valve **141**, during a tilt operation, pressure from the tilt control valve **73** is directed through a line **379**, a shuttle valve **381** and a line **383** to an annulus **385** via a passage **387** in valve **141**. Control pressure in passage **387** is also directed through a passage **389** to a chamber **391** wherein it acts against piston **171** to drive spool **173** to the extreme left position shown in FIG. **9** against the forces of springs **175** and **177**. The piston **171** is sized such that, at a control pressure of **300** psi in line **383**, the spool **173** is shifted to the FIG. **9** position. The control signal is then supplied through annulus

385, annular groove **393**, annulus **395**, passage **185**, and line **183** to the linear servo valve **75** for control of the displacement of pump **37**.

Thus it can be seen that the tilt signal will override the lift signal. For example, if the lift control valve is calling for maximum pressure in line **139**, **625** psi will be urging piston **169** to the left. However, the force available against the smaller piston is inadequate to move the spool **173** past the position shown in FIG. **8**. If the tilt control valve is also actuated, the tilt signal causes the spool to be moved to the position shown in FIG. **9** by the time it reaches 300 psi, causing an interruption of the lift signal from line **139**. The tilt control signal is now supplied to the linear servo valve **75** and the displacement of the pump can be further increased to supply both the lift and tilt circuits adequately.

If the maximum pressure output of pump **37** should ever exceed 6,000 psi, a pilot-operated relief valve **241** will automatically activate to relieve that pressure. In so doing, some of the relieved flow will pass through the pilot portion of the valve and through a line **397** back to tank **23**. However, a small orifice **399** restricts the flow through line **397** enough to produce a pressure in a line **421** which communicates with the priority valve **141**. Fluid under pressure in line **421** is communicated to chamber **423** in the priority valve and the combined force resulting from this pressure in chamber **423** and the biasing force of springs **175** and **177** is large enough to overcome the force being developed in either chamber **167** or **391**, so that the spool **173** will be moved back toward the neutral position.

In that event, pressure in line **183** is communicated to tank via passage **185**, annulus **179**, annular groove **181**, annulus **425**, and a line **427**. This will cause the pressure in chamber **187** of servo valve **75** to reduce to tank pressure and allow force from the spring **193** to move spool **191** downwardly, communicating annuli **195** and **429** so as to transfer fluid from line **77** to line **431** and force pump **37** to return to the zero displacement position. This assures that parts and components will not be damaged during the overpressure condition and that over-heating will not occur.

ALTERNATE EMBODIMENT

It is felt that the description of the original embodiment, together with the description of the alternate embodiment for the three operational modes, will cause the operation of the machine utilizing the alternate embodiment to be obvious.

MODIFIED ALTERNATE EMBODIMENT

The modified alternate embodiment of FIG. **19** is essentially identical with that illustrated in FIG. **16**; therefore, wherever possible, identical identification labels have been utilized. Except as explained below, the operation of the two valves is the same. The distinctions between the two directional valves occur primarily with respect to the structure and operation of the load check valves **268** and **296**.

Under certain conditions of operation, it may prove to be desirable to provide greater stability in the operation of the check valves **268** and **296**. This can effectively be accomplished by eliminating the apertures **274** (FIG. **16**) from the valves, and installing inserts **411** and **413** (FIG. **19**) in the compartments **252** and **322**, respectively. Insert **411** may be provided with an aperture **415** which communicates with passage **258** via a line **417** which is connected to line **260** in the manner illustrated. Similarly, an aperture **419** and insert **413** communicates with passage **282** via a line **421** which is connected to line **280**.

Referring to the upper half of the valve for example, the pressure in passage **258** is communicated to chamber **272** via line **260**, line **417**, aperture **415**, and passage **252**. When the pressure in passage **258**, and therefore passage **252**, reaches a predetermined level, the poppet relief valve **276** will be opened so that the passage **252** and the chamber **272** will be connected to tank. This will allow pressure in passage **258** to work against the face of the check valve **268** and move it from its seat so that the passage **258** can be opened to tank **23** via line **354**.

When the head end of the cylinder **261** is thus relieved, and it is desired to maintain a controlled flow of fluid from the passage **258** to the line **354**, the valve spool **350** will be moved downwardly against the force of spring **352** by pressure exerted through line **346** in response to manipulation of the main control valve. When the spool **350** is shifted, fluid flows from line **417** around an annular groove on the spool, past a land **350*a***, and into the line **252*b***. A pressure drop in passage **272** will occur as a result of this connection, allowing the valve **268** to unseat. This pressure drop is modulatable by adjustment of the position of land **350*a***. Further movement of the spool **350** will cause the effect of the pressure drop on check valve **268** to change from the control of land **350*a*** to land **350*b***, which influences flow from line **417** to the annular groove.

Flow check valve **268** will open only so when there is pump pressure available to maintain spool **350** in the shifted position. The flow from line **417** to line **252*b*** can thus be modulated in relation to the pressure on the head end of cylinder **261** and the position of spool **350**. This flow will be considerably greater than that which can flow through a small orifice, such as **274** of the FIG. **16** embodiment, while maintaining the regulation and response of the load check valve **268** at a very satisfactory level. The placement of the insert **411** within the passage **252** allows the system to function satisfactorily as a relief valve by means of poppet **276**.

Thus, the Applicants have provided a vastly-improved control system wherein a rather low pressure hydraulic system is utilized to operate and control a high-pressure work output system. The invention has been illustrated and described relative to two possible embodiments and it is realized that many other embodiments, modifications, etc., will be obvious to those skilled in the art which do not exceed the purview of the following claims.

What is claimed is:

1. In a directional valve having a passage therein connected to a source of fluid pressure and two controlled ports for connection to a fluid motor, a pilot operated control including at least one check valve means controlling fluid communication between one of said controlled ports and said passage, said check valve means being responsive to pilot fluid pressure changes communicated thereto to connect and disconnect said controlled port and said passage.

2. The directional valve defined in claim 1 wherein the pilot operated control includes a second check valve means controlling fluid communication between the passage and the other controlled port, said second check valve means being responsive to pilot fluid pressure changes communicated thereto to connect and disconnect said other controlled port with said passage whereby proper pilot pressure changes will connect one or the other of said controlled ports with said passage.

3. The directional valve defined in claim 1 wherein each controlled port is also connected to drain through a separate controlled relief valve means, each of said controlled relief valve means being responsive to fluid pilot pressure to change its relief pressure whereby one of said controlled ports can be connected to the passage in communication with the source of fluid pressure and the other controlled port can be relieved to drain at a lower relief pressure for controlling the direction of a fluid motor connected to said controlled ports by appropriate selection of fluid pilot pressures.

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