United States Patent

[11] **3,575,000**

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[21] Appl. No. **868,964**
[22] Filed **Oct. 21, 1969**
[45] Patented **Apr. 13, 1971**
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Continuation-in-part of application Ser. No. 814,003, Apr. 7, 1969, now abandoned.

[54] **HIGH PRESSURE IMPLEMENT HYDRAULIC CIRCUIT**
17 Claims, 19 Drawing Figs.

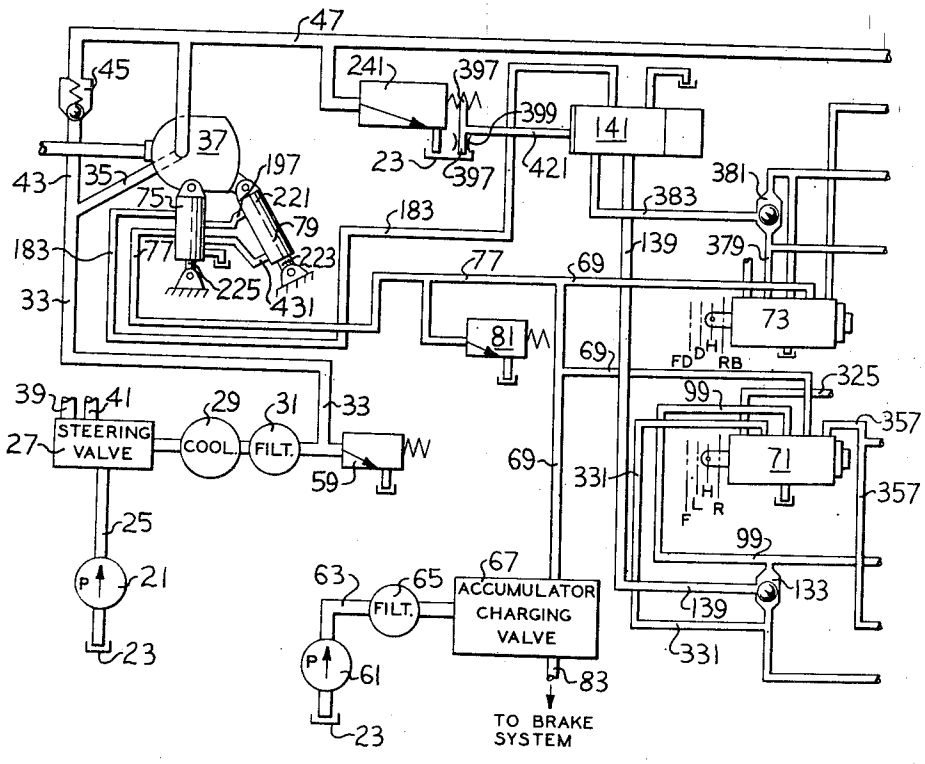
[52] **U.S. Cl.** **60/52**
[51] **Int. Cl.** **F15b 15/18**
[50] **Field of Search** ..... 137/(Inquired); 60/52 (S), 52 (HE), 52 (VSP)

[56] **References Cited**
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Primary Examiner—Edgar W. Geoghegan
Attorney—Fryer, Tjensvold, Feix, Phillips & Lempio

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.



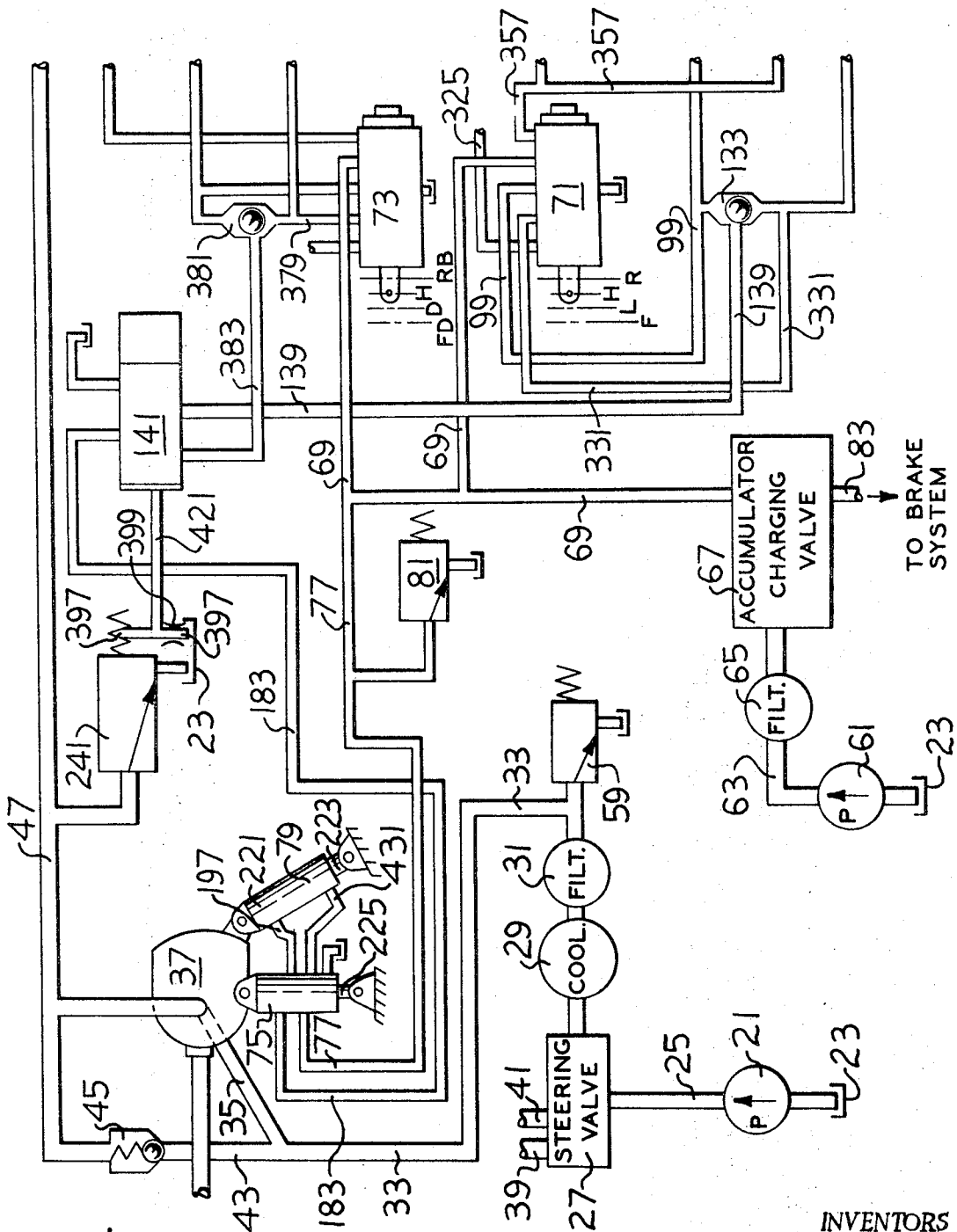


Fig. 1

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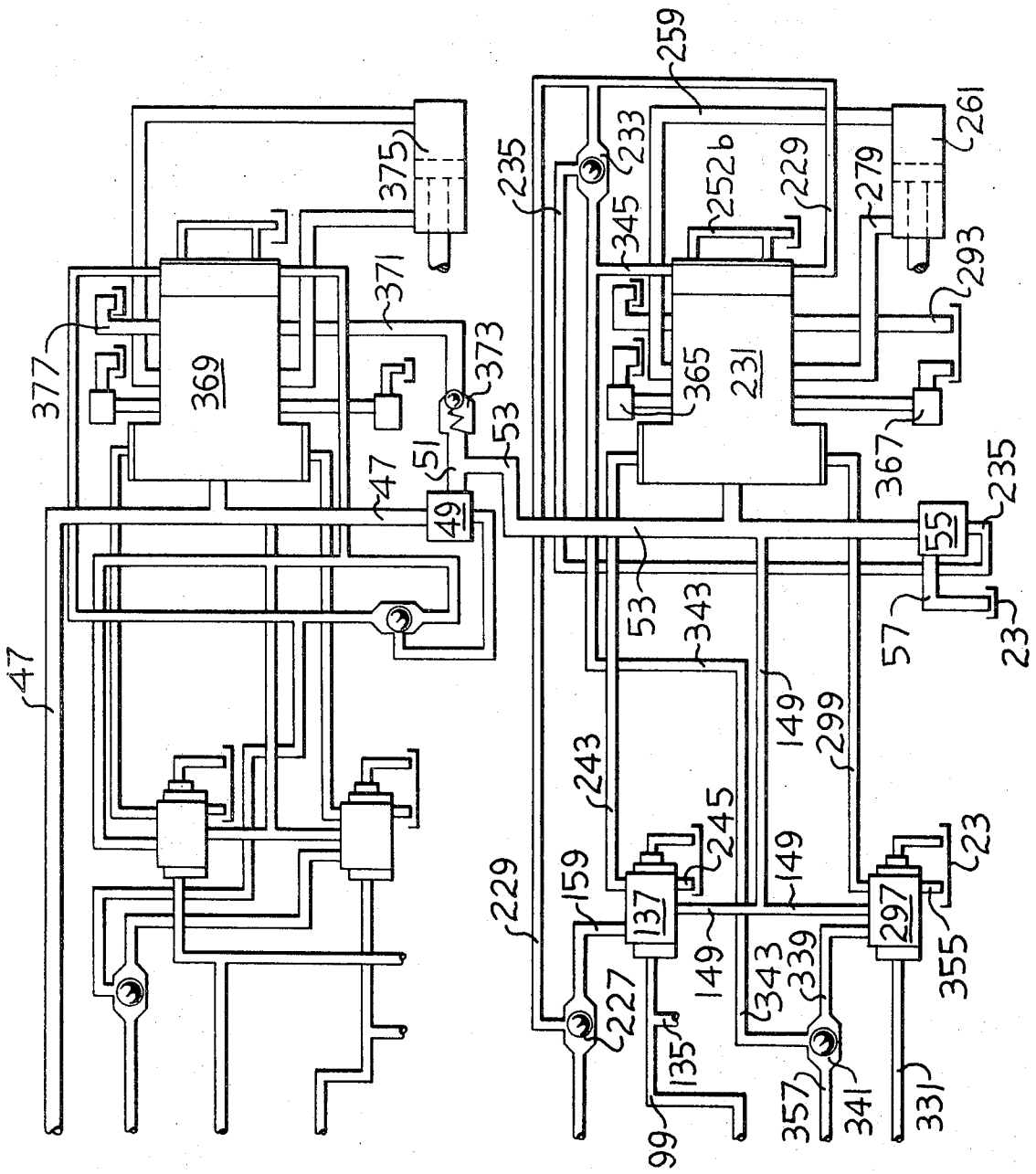


FIG-2-

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Fig. 3.

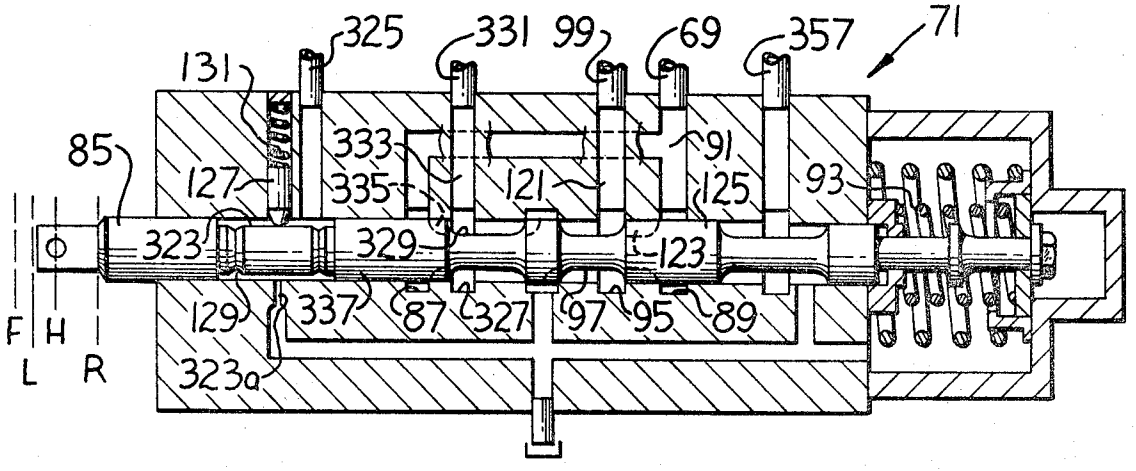
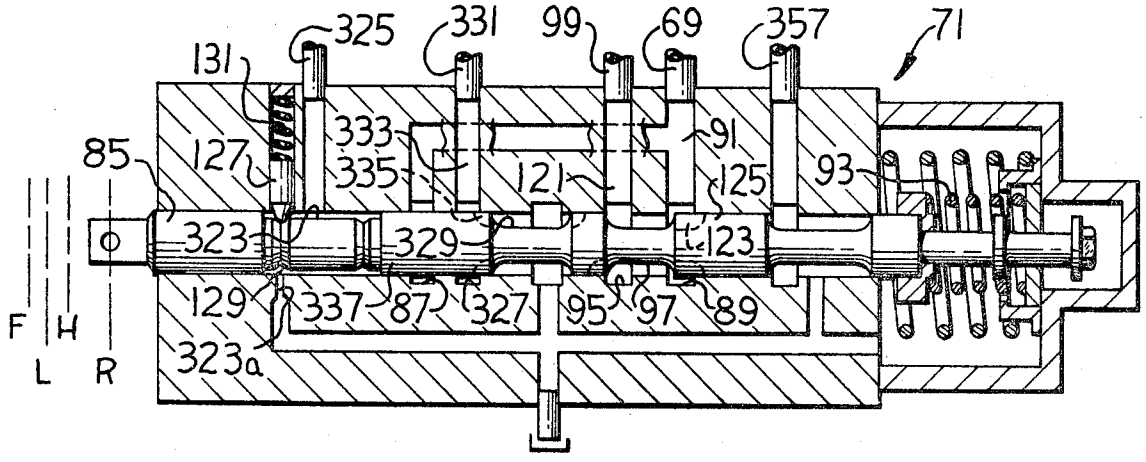


Fig. 4.



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Fig. 5.

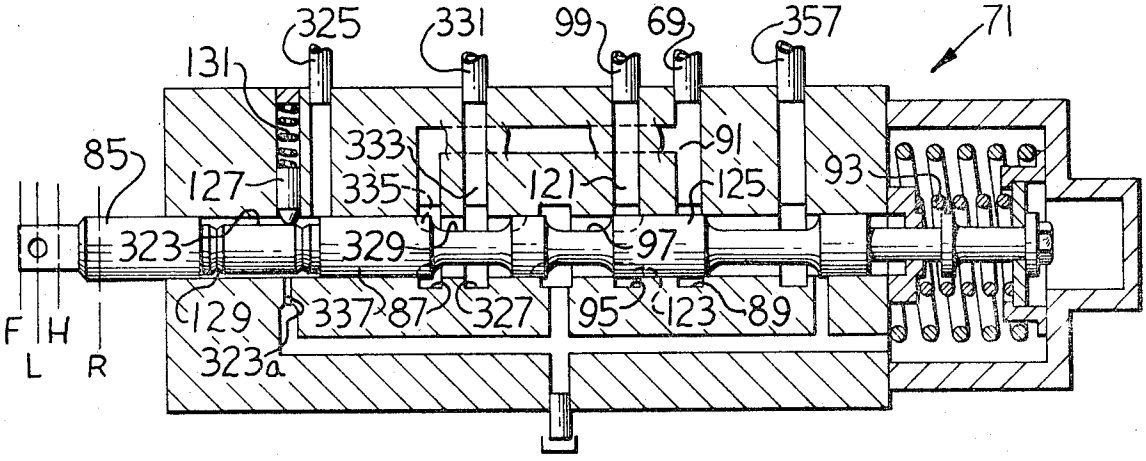
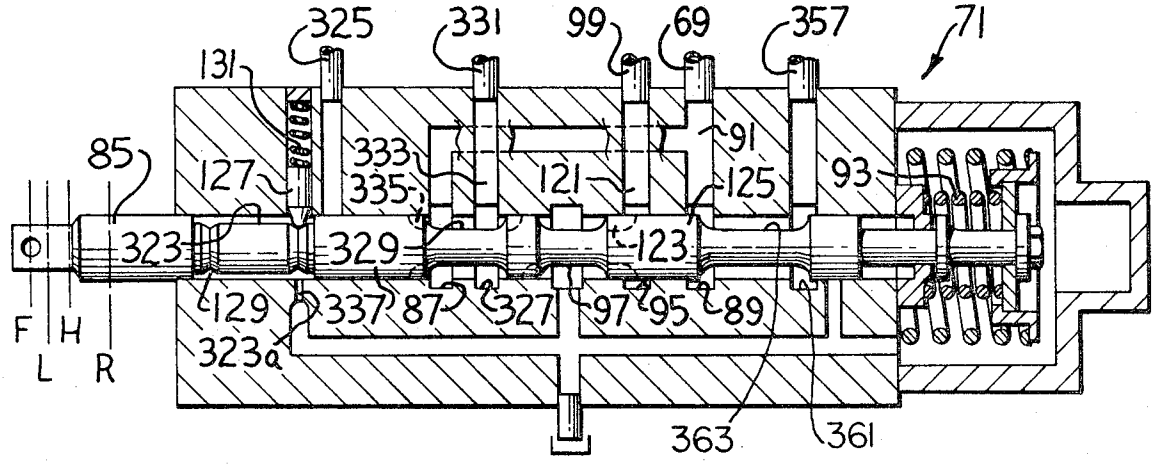


Fig. 6.



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Fig-7-

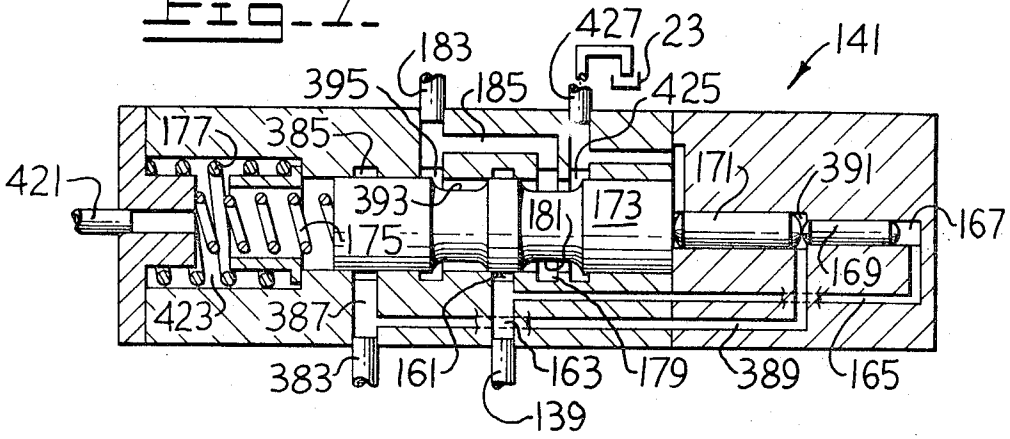


Fig-8-

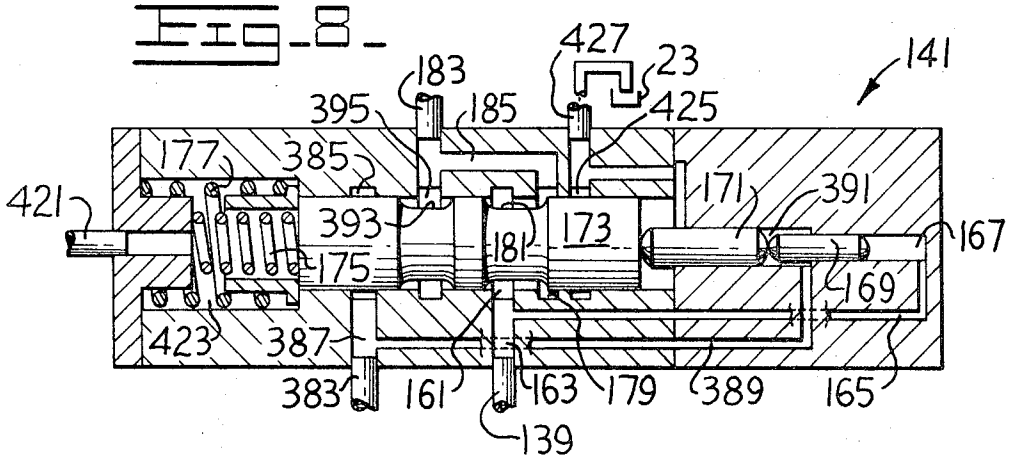
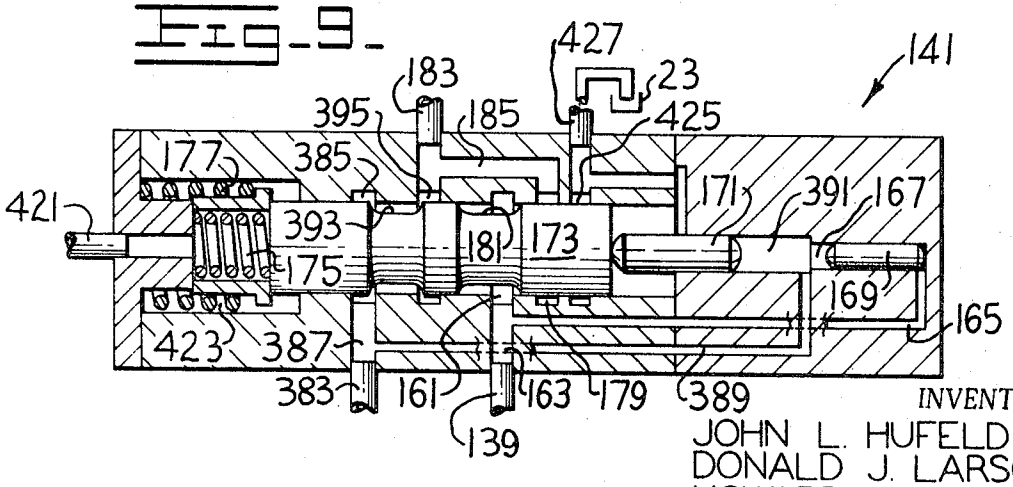


Fig-9-



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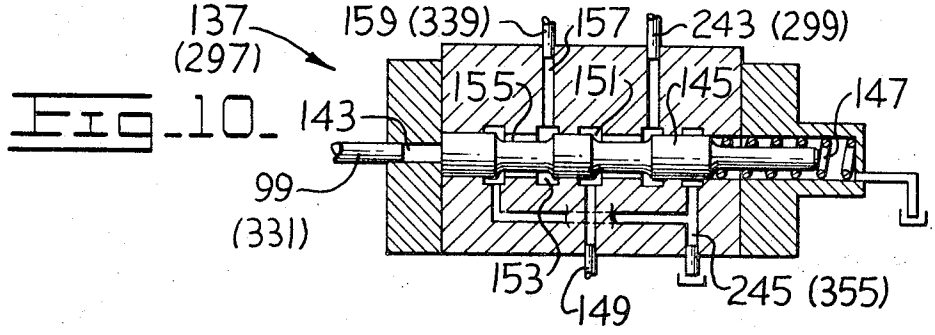


Fig-10-

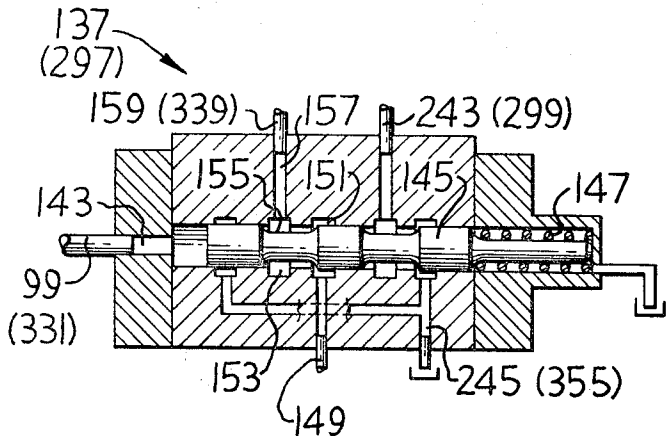


Fig-11-

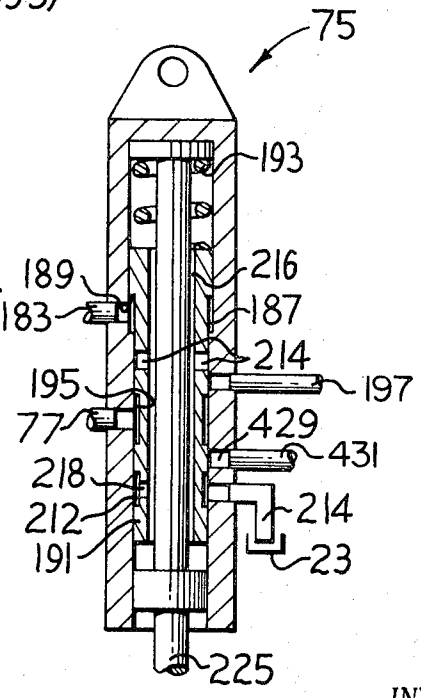


Fig-12-

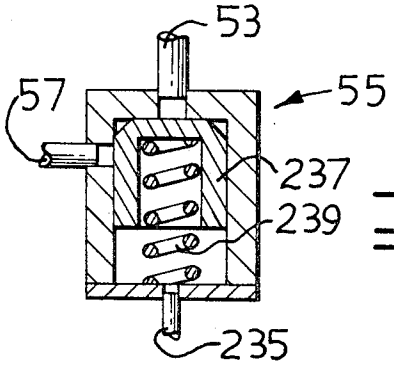
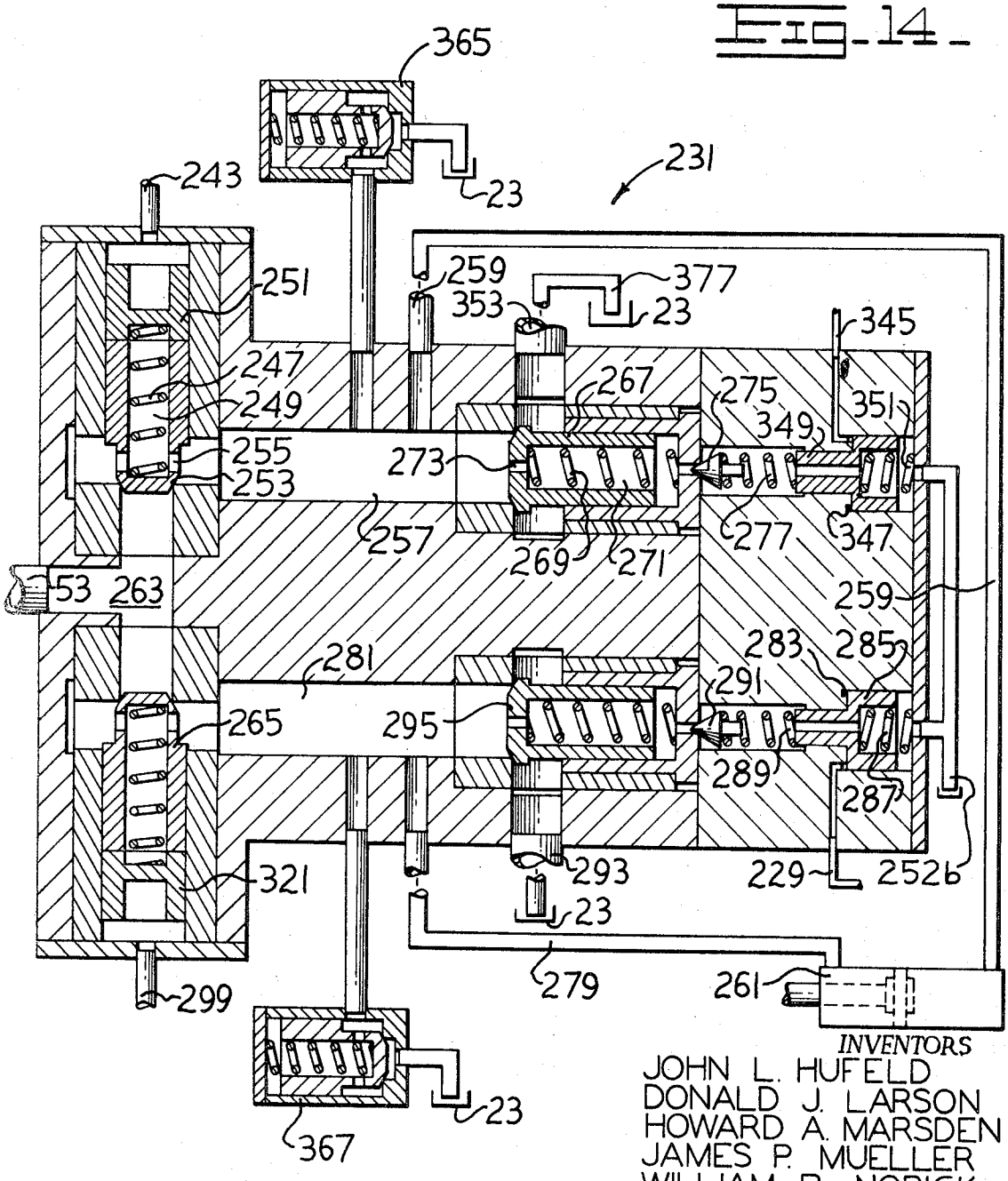


Fig-13-

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Fig. 14



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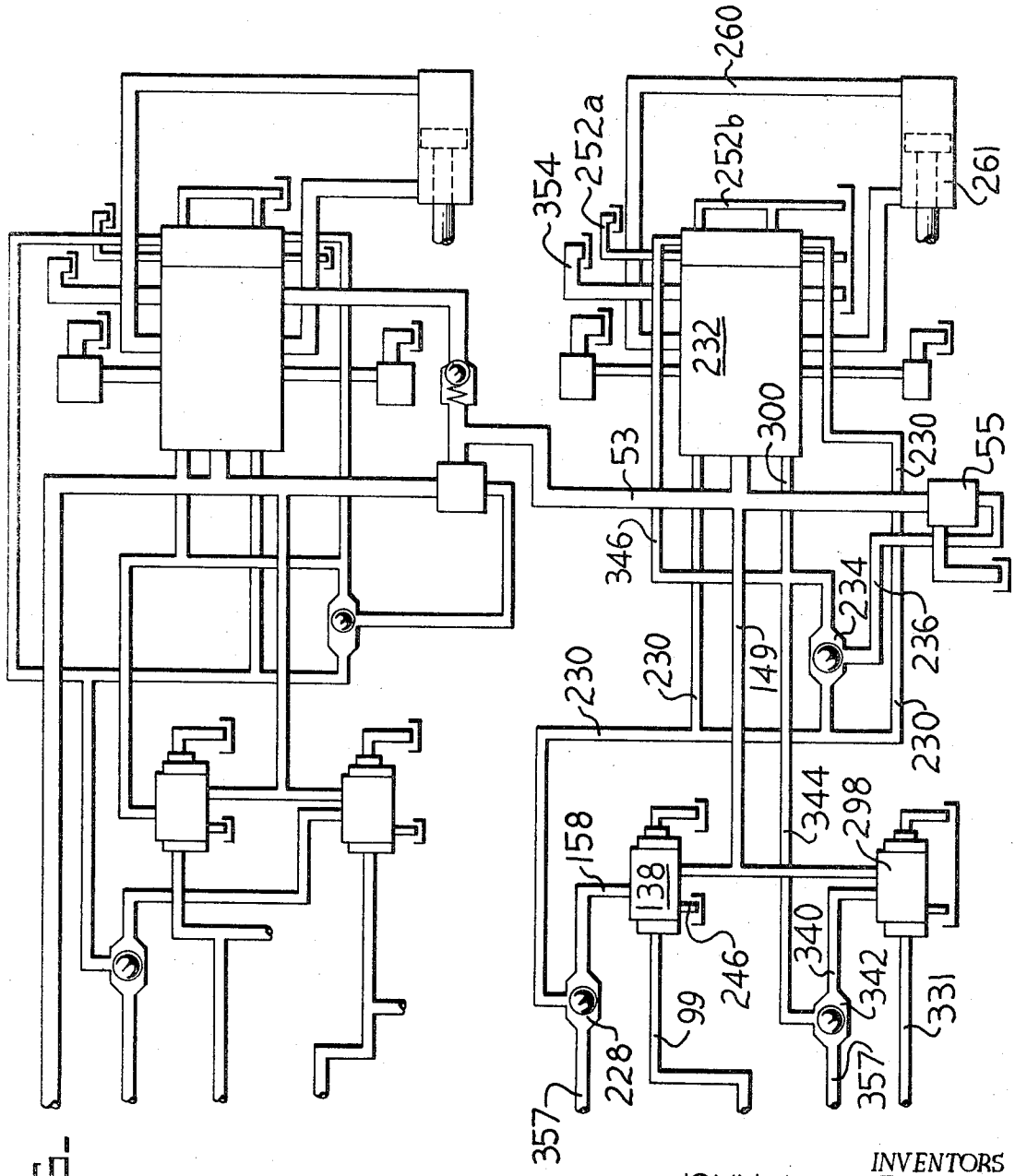
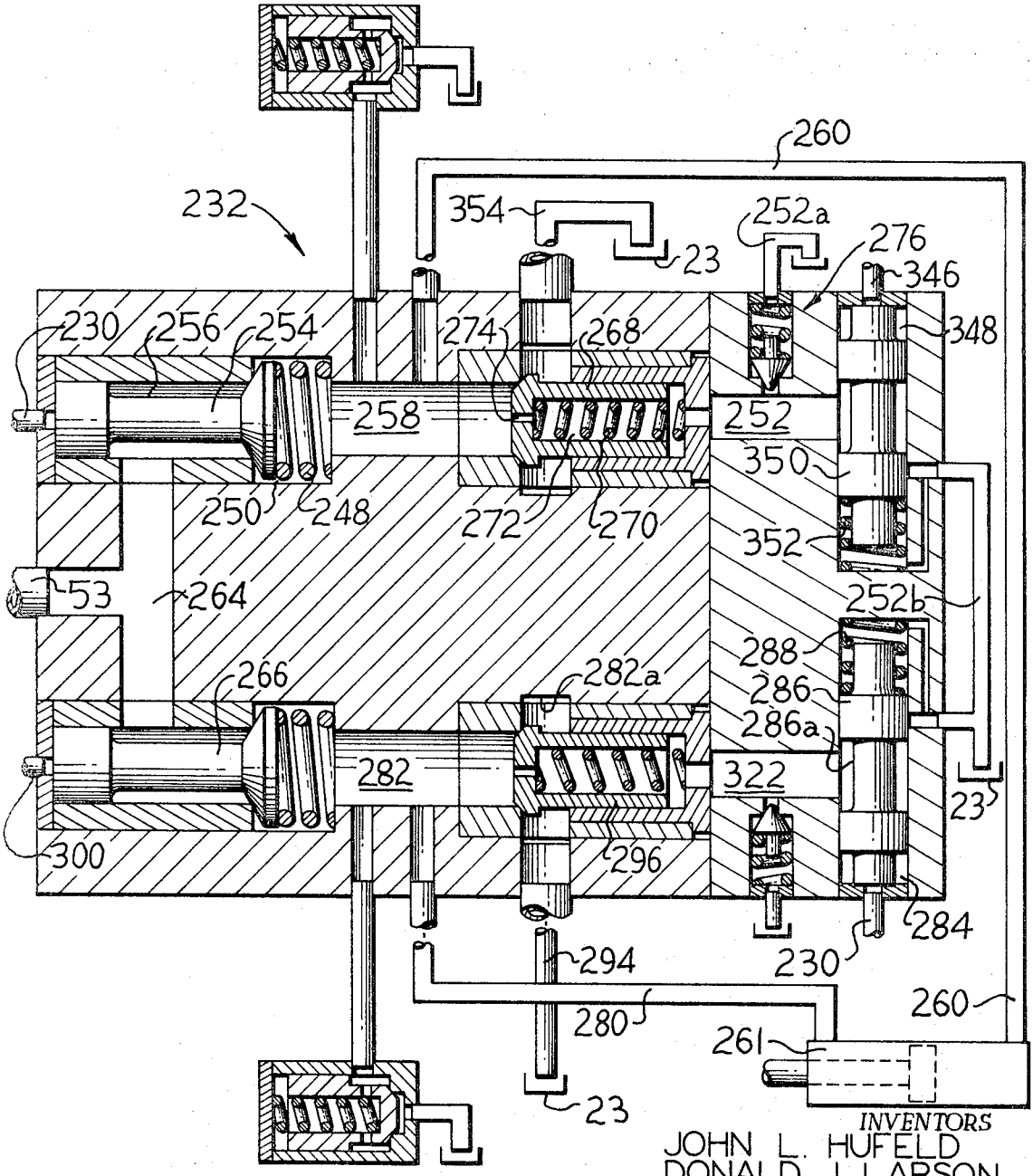


FIG. 15.

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Fig. 16.



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Fig. 17.

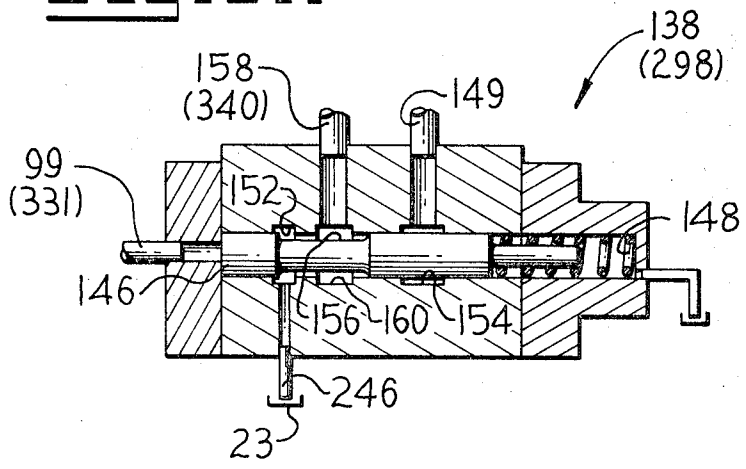
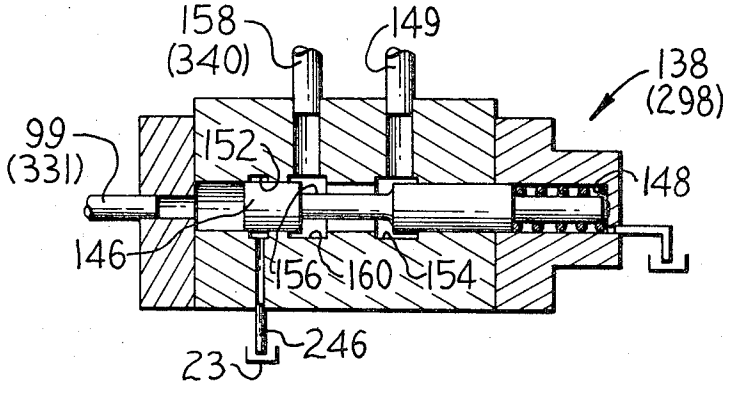


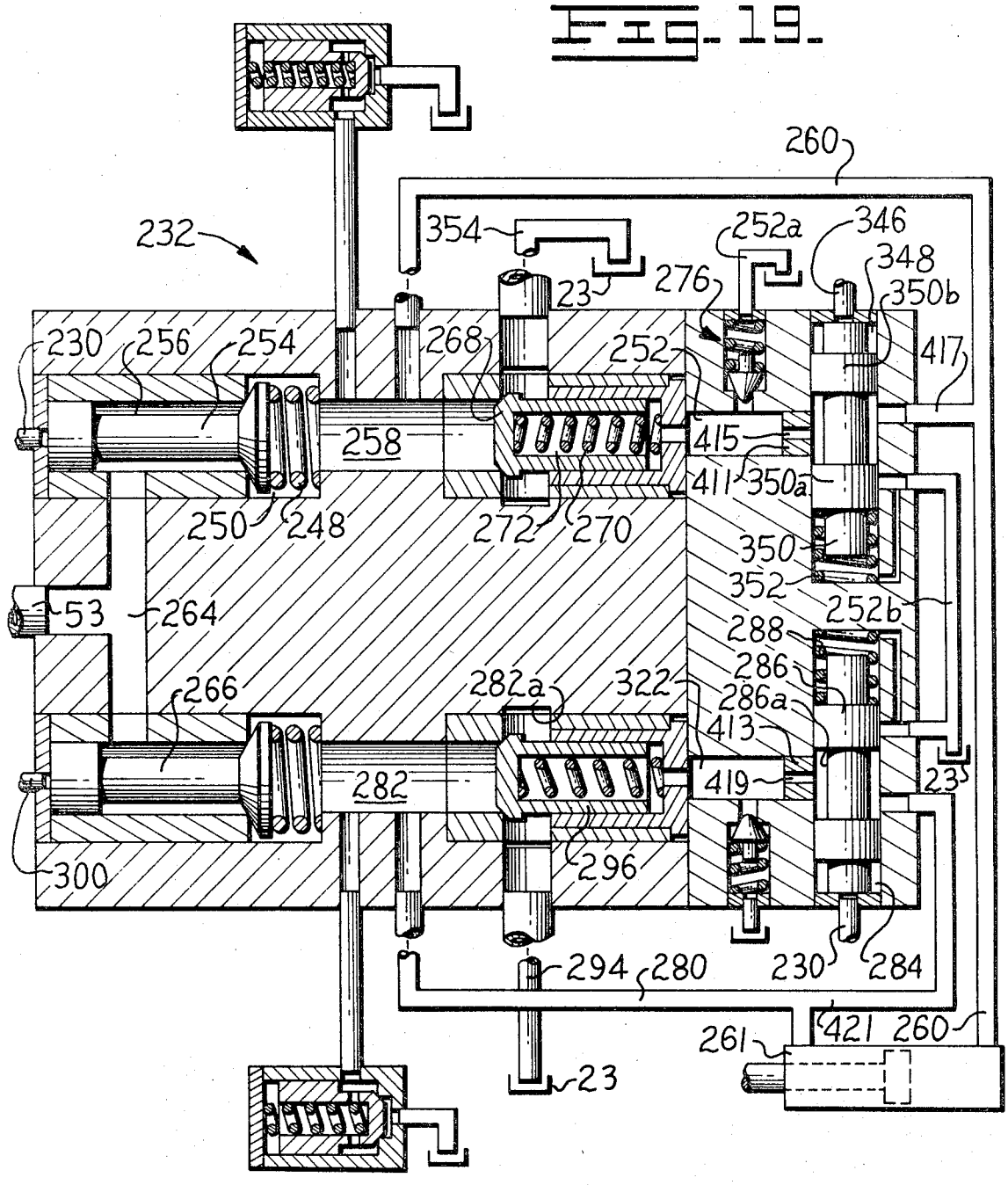
Fig. 18.



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Fig. 19.



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HIGH PRESSURE IMPLEMENT HYDRAULIC CIRCUIT

This application is a continuation-in-part of our 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 tolerances 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 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 a 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 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 servovalve 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 p.s.i. Many modifications of the circuit, including deletion and duplication of elements, will be obvious to those skilled in the art when using 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 **77** and is then directed to an actuating cylinder **79**, when the servovalve is shifted in either one of two operating directions. A relief valve **81** maintains a maximum pressure of about **625** p.s.i. 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 p.s.i., 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** p.s.i. 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 p.s.i.

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 p.s.i. When it is between 300 and 625 p.s.i., it is used to stroke the linear servovalve **75**, thereby creating an implement pump flow proportional to the control pressure, i.e., 300—625 p.s.i.

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 relieved 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 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 kickout 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 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 p.s.i.

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 p.s.i., the position of the pistons and spool as shown in FIG. **8** will be attained, and even though the signal pressure increases to 625 p.s.i., **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 servovalve 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 the 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 form line 183, the force of spring 193 will move sleeve 191 downwardly so that line 197 is connected to passages 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 buildup in line 53, as required, up to a maximum of approximately 6,000 p.s.i. 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 p.s.i. is reached, the pump 37 provides a flow through lines 47 and 53. As a result, a buildup of fluid pressure in line 53 and passage 263 occurs. 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 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 left 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 p.s.i., 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 kickout 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 kickout 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 **323***a* 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 raise 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 **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 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 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 overpressure 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 p.s.i. 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 p.s.i. 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 p.s.i.

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

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 **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 p.s.i.

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 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 the 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 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 252*b*, 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 p.s.i. 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 231) 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 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, makeup 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 circuit, is directed to line 53 via a line 371 and a check valve 373. The check valve 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, 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 p.s.i. 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 servovalve 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 p.s.i. 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 p.s.i., causing an interruption of the lift signal from line 139. The tilt control signal is now supplied to the linear servovalve 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 p.s.i., 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 servovalve 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 overheating 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 e provided with an aperture 415 which communicates with passage 258 via a line 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 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.

We claim:

1. A low-pressure hydraulic control circuit for controlling at least two high-pressure work output systems including at least two selectively actuatable control valves, conduit means connecting each of said control valves to a variable displacement pump whereby flow in either work output system may be controlled by said two selectively actuatable control valves, a separate directional valve in each of said high-pressure work output systems, and means separately connecting each of said selectively actuatable control valves with said directional valve in its associated work output system whereby flow in either of said work output systems can be controlled in direction and amount.

2. The circuit of claim 1 including a priority valve in the means connecting the control valves to the variable displacement pump, and means in the priority valve for providing that a control signal from one of the control valves shall always take priority over the control signal of the other control valve in controlling the amount of work output flow.

3. The circuit of claim 2 wherein the priority valve has a spool means therein, means biasing the spool to a first position, means responsive to an actuation signal from the one control valve for actuating the spool to a second position, and means responsive to an actuation signal from the other control valve for actuating the spool to a third position.

4. The circuit of claim 3 wherein the actuating means comprise pistons of unequal areas whereby equal actuation signals from the control valves cause the larger area piston to override any force exerted by the smaller area piston against the spool means.

5. The circuit of claim 3 wherein the actuating means comprise axially aligned pistons of unequal surface area, the larger piston acting against one end of the spool means and the smaller piston acting against the larger piston so as to exert force on the same one end of the spool means.

6. The circuit of claim 5 wherein means deliver fluid to the surface areas of the pistons in response to the actuation signals of the control valves, and the relative sizes of the piston surface areas are such that equal fluid pressure delivered to both pistons causes the larger piston to move the spool means past the second position and on to the third position.

7. The circuit of claim 1 including means associated with one of the directional valves denying work output flow to the other directional valve when the one directional valve is actuated by its control valve to provide a work output in a first direction.

8. The circuit of claim 8 including means associated with the one directional valve providing work output flow to the other directional valve when the one directional valve is actuated by its control valve to provide a work output in a second direction.

9. The circuit of claim 1 including valve means in the means connecting each control valve to each directional valve, said valve means selectively actuated by the control valve, means connecting said output flow of the variable displacement pump to each valve means, and actuatable means in the directional valve selectively positioned to produce the required work direction in response to a resultant output flow through the valve means.

10. A hydraulic circuit wherein a low-pressure system is employed for controlling a high-pressure fluid work output system including a selectively actuatable control valve connected to a source of low-pressure fluid, a variable displacement pump having a high-pressure variable flow output and a control servo for adjusting said high-pressure variable flow output, conduit means connecting said control valve and said control servo operable to control said variable displacement pump as a function of said control valve by fluid pressure in said conduit means, a directional valve having its output connected to said work output system and its inlet connected to said high-pressure variable flow output of said variable displacement pump, and fluid actuated means connected between said control valve and directional valve whereby both the high-pressure flow to and direction of said work output is controlled.

11. The circuit of claim 10 including a first valve means and a second valve means in the means connecting the control valve to the directional valve, the first valve means actuated when the control valve is operated in a first direction and the second valve means actuated when the control valve is operated in a second direction, the first and second valve means operatively connected to the directional valve so that actuation of the first valve means causes the directional valve to provide a work output flow in a first direction and actuation of the second valve means causes the directional valve to provide a work output flow in a second direction.

12. The circuit of claim 11 including at least one check valve means in the directional valve, the check valve means being selectively relieved by actuation of either the first or second valve means so as to allow the work output flow of the variable displacement pump to flow through the directional valve in either the first or second direction.

13. The circuit of claim 11 including actuatable means, in the means connecting the control valve to the directional valve, actuated when the control valve is operated to a selected position, allowing the directional valve to be so conditioned as to allow external forces to motivate the machinery controlled by the work output system.

14. The circuit of claim 10 wherein the fluid actuated means includes valve means whereby the directional valve will be positioned so that work output is not hydraulically locked when the control valve is placed in a float position.

15. The circuit of claim 10 wherein the directional valve includes modulating means to reduce pressure surges in the work output system.

16. The circuit of claim 10 wherein the directional valve includes poppet means in its inlet preventing coupling the high-pressure variable displacement output of the variable displacement output of the variable displacement pump to the work output system until the pressure exceeds that in said work output system.

17. The circuit of claim 16 wherein the poppet means in the directional valve will close if the flow in the work output system exceeds that of the variable displacement pump.

UNITED STATES PATENT OFFICE
CERTIFICATE OF CORRECTION

Patent No. 3,575,000 Dated April 13, 1971

Inventor(s) JOHN L. HUFELD et al

It is certified that error appears in the above-identified patent and that said Letters Patent are hereby corrected as shown below:

IN THE SPECIFICATION:

Col. 10, line 43, after "valve" insert --141--.

IN THE CLAIMS:

Col. 12, claim 8, line 1, "8" should be --7--.

Signed and sealed this 16th day of May 1972.

(SEAL)
Attest:

EDWARD M.FLETCHER, JR.
Attesting Officer

ROBERT GOTTSCHALK
Commissioner of Patents