

A DEVICE FOR DIRECTION CONTROL OF A DOUBLE ACTION
HYDRAULIC CYLINDER

This invention relates to a device for direction control of a double action hydraulic cylinder.

According to the present invention there is provided a device for direction control of a double action hydraulic cylinder comprising a first fluid control valve adapted for connection to the extend power end of the hydraulic cylinder and a second fluid control valve adapted for connection to the retract power end of the hydraulic cylinder, the first and second valves being drivingly connected to one another by the cross-head of a scotch-yoke and arranged such that when one valve is being moved to its fully open position to permit flow of pressurised fluid to its respective power end of the hydraulic cylinder the other valve is being moved to its closed position such that when one valve is fully open the other valve is closed, and a rotary fluid motor connected to the scotch-yoke to reciprocate the cross-head.

Reference is now made to the accompanying drawings, in which:-

Figure 1 is a schematic view in section and elevation of one embodiment of a hydraulic well pump incorporating the features of the invention;

Figures 2A and 2B taken together form a schematic diagram of a hydraulic power and control system for operating the hydraulic well pump of Figure 1;

Figure 3 is a schematic side view in section and elevation of another form of hydraulic well pump embodying the features of the invention;

Figure 4 is a schematic diagram of components of a hydraulic power system which in combination with the components of the fluid system of Figure 2B may be used to operate the hydraulic pump of Figure 3;

Figure 5 is a view in section and elevation of the hydraulic cylinder linear motion control device;

Figure 6 is a right end view in section and elevation of the device of Figure 5;

5 Figure 7 is a view in section and elevation of the valve body of the device of Figure 5;

Figure 8 is a fragmentary top plan view of the central portion of the valve body of Figure 7;

10 Figure 9 is a bottom fragmentary view of the valve body of Figure 7;

Figure 10 is a front view in elevation of the cross head of the motion control device of Figure 5;

Figure 11 is a top plan view of the cross head of Figure 10;

15 Figure 12 is a front view in elevation of the cam crank of the motion control device of Figure 5;

Figure 13 is a side view in section of the cam crank along the line 14-14 of Figure 12;

20 Figure 14 is a side view in section and elevation of one of the valve seats of the motion control device of Figure 5;

Figure 15 is a side view in section and elevation of one of the valves of the motion control device of Figure 5;

25 Figure 16 is a longitudinal side view in section of one of the valve spools of the motion control device of Figure 5; and

Figure 17 is a view in section of the valve spool along the line 18-18 of Figure 16.

30 Referring to Figure 1, a hydraulic well pump A is shown including an air counterbalanced hydraulic cylinder assembly which operates a flexible tension member connected with a pump sucker rod raising and lowering the rod twice the distance of the lift of the hydraulic

5 cylinder. The hydraulic cylinder assembly includes a
stationary hydraulic piston 1 on the upper end of a
hollow piston rod 2 mounted in coaxial spaced relation
around a flow conductor 3. The piston rod has ports 4
below the piston 1 opening into the annular space between
the piston rod 2 and the flow conductor 3. A counter-
balance annular piston 5 is movable in sealed
relationship along the piston rod 2 within a stationary
cylinder 6 at the lower end of a hydraulic cylinder 7
10 which moves in sealed relationship along the stationary
hydraulic piston 1. An idler sheave platform 8 is
secured on the upper end of the movable cylinder 7. Idler
sheaves 9 and 10 are mounted in horizontal spaced
relation on the platform 8. A flexible tension member 11
15 is secured at one end 12 to the base or foundation for
the hydraulic pump, extends over the sheaves 9 and 10 and
downwardly connected at the other end with a well pump
sucker rod string 13. A counterbalance air receiver 14
supplied with air from a compressor 15 communicates
20 through a conduit with the stationary cylinder 6 below
the piston 5 for applying a pneumatic force upwardly on
the piston 5 substantially equal to the downward force
produced by the combined weights of the movable
components of the well pump including the polish rod
25 string and the fluid column in the well above the pump
plunger.

The hydraulic well pump A is operated by pumping
hydraulic fluid through the conduit 3 into the movable
hydraulic cylinder 7 above the piston 1 raising the
30 platform 8 with the sheaves 9 and 10. Because the first
end of the tension member 11 is secured at 12 the second
end of the tension member connected with the sucker rod
13 is lifted twice the distance that the sheaves are
raised. The free running end of the tension member moves
35 at twice the rate of extension of the cylinder 7 with the

platform 8 and the sheaves 9 and 10. The weight supported by the hydraulic cylinder assembly is equal to twice the weight supported by the sucker rod 13. That weight is also equal to the sum of the vertical force provided by the piston 5 and the vertical force provided by the upper or capend of the cylinder 7. When the hydraulic cylinder is extended to the upper end of the stroke, the well pump is reversed by pumping hydraulic fluid into the movable cylinder 7 below the piston 1 through the annulus between the conduit 3 and the piston rod 2 and outwardly through the ports 4 below the piston 1. Thus the hydraulic well pump is reciprocated by alternately pumping the hydraulic cylinder assembly upwardly and downwardly. The pneumatic counterbalancing of the hydraulic well pump reduces the force required to reciprocate the pump to the sum of the force necessary to overcome mechanical and fluid friction in the pumping system and column of fluid being lifted and accelerate the mass of the fluid column and the components of the pump and sucker rod being moved. Of course as the pump moves downwardly, the total forces are reduced by the value attributable to the column of fluid above the plunger pump which of course is not lowered during the downward stroke. The counterbalancing substantially reduces the forces required to operate the hydraulic well pump and the employment of the particular arrangement of the idler sheaves and flexible tension member provides a pump plunger and sucker rod stroke twice the length of the travel of the hydraulic piston assembly thereby cutting the height of the required structure to half of the conventional walking beam type pumping jack.

A hydraulic fluid power system which may be used to operate the well pump A of Figure 1 is illustrated in Figures 2A and 2B. Referring to Figures 2A and 2B, only the hydraulic power and control circuitry is illustrated,

it being understood that the reciprocating cylinder 7 which moves relative to the stationary piston 1 is connected at a lower end with the piston 5 operating in the outer cylinder 6 in response to the air counter-balance system schematically represented in Figure 1. The same reference numerals are used in Figure 2A to designate the corresponding parts of the hydraulic cylinder system as are used with such parts in Figure 1, such for example, as the numeral 2 designates the hollow piston rod 2 with the flow conductor connected with the piston rod for supplying the hydraulic fluid which drives the movable cylinder 7 and thus the pump A downwardly. The power circuit for delivering hydraulic fluid to the hydraulic cylinder assembly includes two fixed volume pumps 20a and 20b each capable of delivering a desired volume and pressure for the particular function of the well pump. The pump 20a is associated with the cap end of the cylinder 7 while the pump 20b is associated with the piston rod end of the cylinder. Thus the pump 20a drives the well pump during the lift cycle and the pump 20b drives the well pump during the retract or lowering cycle. The two pumps 20a and 20b are coupled to and driven by a common drive shaft 21 and a single power source, not shown, which may be an electric motor or internal combustion engine. A hydraulic fluid reservoir 22 indicated schematically with respect to several returns in the system provides the source of hydraulic fluid for both of the pumps. The outlet of the pump 20a is connected to the cap end of the cylinder 7 by a line 23a including a check valve 24a permitting flow only in the direction into the cap end of the cylinder. A pressure relief valve 25a is connected in a line 26a leading from the line 23a and dumping into the tank 22. The relief valve 25a responds to pressure in the line 23a and opens to dump fluid to the tank when the maximum

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selected pressure of that line is reached. The valve 25a thus limits the maximum fluid pressure available to the cap end of the cylinder 7. The outlet of the pump 20b is connected with the rod end of the cylinder by the line 23b including the check valve 24b and the line and piston rod 2 defining the flow path into the rod end of the cylinder. Because the effective area of the piston rod in the cylinder is less than that of the cap end, the pump 20b may have different operating parameters from those of pump 20a. A pressure relief valve 25b is connected by the line 26b into the line 23b and to the tank 22 for dumping fluid back to the tank when a selected maximum pressure in the line 23b is reached.

As seen in Figure 2A, the control system for the delivery of hydraulic power fluid to the pump cylinder includes sequence valves 31a and 31b associated respectively with the cap and rod ends of the cylinder 7. The sequence valves are connected as cross-piloted valves to prevent the overrun of the reciprocating cylinder in the event resistance to movement should reverse for some reason. The sequence valve 31a is connected between the line 23a and the tank 22 by a line 32a. The valve 31b is connected between the line 23b and the tank 22 by a line 32b. The valve 31a is connected to the line 23b by a pilot line 33a so that the valve 31a is opened in response to pressure in the rod end of the cylinder. The valve 31b is connected to the line 23a by a pilot line 33b so that the valve 31b operates in response to pressure in the cap end of the cylinder 7. It will be apparent that as the cylinder 7 reciprocates in each direction, the pressure within the cylinder on the opposite end must be relieved for the cylinder to move. Thus, the sequence valve 31a relieves the pressure in the cap end of the cylinder 7 as the cylinder retracts or moves down downwardly; the sequence valve 31b relieves

the pressure in the rod end of the cylinder as the cylinder extends or moves upwardly.

Direction control of the cylinder 7 is effected by control valve 40 connected between the outlets of the pumps 20a and 20b and the tank 22. The valve 40 is a three-position valve having an intermediate dump position in which the outlets of both of the pumps are communicated with the tank 22 effectively unloading both pumps. The valve 40 has extend and retract positions for fluid flow from each of the pumps to its respective end of the cylinder 7. For controlling flow to the cap end of the cylinder 7, the outlet of the pump 20a is connected through the line 41a to the valve 40. When the valve 40 is in the dump position for the cap end of the cylinder, the outlet of the pump 20a is dumped to the tank 22. The outlet of the pump 20b is connected with the valve 40 through the line 41b. When the valve 40 is in the dump position for the rod end of the cylinder 7, the outlet of the pump 20b is dumped through the valve 40 to the tank 22. To extend the cylinder 7 to the right as seen in Figure 2A, lift the cylinder as viewed in Figure 1, the valve 40 is shifted to the left as seen in Figure 2A blocking the line 41a at the valve 40 while the line 41b remains open to dump fluid from the piston end of the cylinder back to the tank 22. The output from the pump 20a necessarily flows through the line 23a, the check valve 24a, and the line 3 into the cap end of the cylinder 7 thereby moving the cylinder 7 relative to the fixed piston 1. Similarly, to retract the cylinder 7 downwardly, to the left in Figure 2A, the valve 40 is shifted to the right to block the line 41b while the line 41a is opened to the tank 22. Fluid from the pump 20b flows to the rod end of the cylinder 7 through the line 23b, the check valve 24b, and the flow passage 2 extending into and through the piston rod 2. The pressure

in the line 23b acts through the pilot line 33a to the valve 31a opening the valve permitting flow from the cap end of the cylinder 7 through the line 32a and the valve 31a back to the tank 22. The acceleration or deceleration of the reciprocating cylinder 7 will be directly related to the manner in which the direction control valve 40 is shifted. With appropriate manipulation of the direction control valve, it is possible to cause the cylinder to emulate simple harmonic motion in the pattern of acceleration and deceleration. For both directions of the cylinder movement, the speed of movement will be proportional to the discharge rate of the particular pump driving the piston and the maximum force applied to the piston will be limited by the setting of the respective pressure relief valves.

The hydraulic system of Figures 2A and 2B includes mechanism provided to reciprocate the direction control valve 40 including a small rotary hydraulic motor 43 having an output shaft 44 driving a crank arm 45 non-rotatably fixed to the output shaft, Figure 2B. A pitman arm or link 46 is connected between the crank arm and the reciprocating valve member of the valve 40 to shift the direction control valve. A rotary cam 48 having diametrically opposed external lobes 49 is also non-rotatably fixed to the motor shaft 44. A low pressure hydraulic pump 50 with an associated relief valve 51 provides pressurized fluid for driving the motor 43 and provides pilot fluid for operating certain pilot operated valves included in the hydraulic logic circuit and system of Figures 2A and 2B. The pump 50 may be driven by the drive shaft 21 also driving the pumps 20a and 20b. The pump 50 discharges to the motor 43 through a fluid line 52 which includes branches 52a and 52b including pilot operated, two-way stop and start valves 53 and 54. The valves 53 and 54 control intermittent flow of fluid from

the pump 50 to the motor 43. The stop and start valves 53 and 54 are controlled by limit valves 55 and 56 and a cam operated stop pilot valve 57 operated by the rotary stop cam 48. A variable orifice 60 is provided in the line 52
5 between the stop and start valves to function as a speed control for the hydraulic motor 43. The limit valves 55 and 56 are actuated by a cam C secured and movable with the cylinder 7. The limit valve 55 is operated by the cam when the cylinder 7 reaches the limit of its retract
10 stroke; the limit valve 56 is operated by the cam C when the cylinder 7 approaches the limit of its extend stroke. The stop pilot valve 57 is actuated by the lobes 49 on the stop cam 48 for blocking the fluid line 52 to stop the motor 43. Fluid under pressure is supplied to the
15 limit valves 55 and 56 from the pump through the line 58 and the branch lines 58a and 58b. Pressurized fluid is also supplied to the pilot valve 57 from the pump 50 through the lines 52 and 59. Pilot fluid is conducted from the limit valves 55 and 56 to the start valve 54
20 through the pilot lines 61a, 61b, the shuttle valve 62, and the pilot line 61c. Pilot fluid is conducted from the stop pilot valve 57 to the stop valve 53 through a pilot line 65. The stop valve 53 is normally opened passing fluid to the pump 43 and is closed by pilot fluid from
25 the pilot valve 57 when actuated by one of the lobes 49 on the stop cam 48. The pilot valve 57 is normally closed communicating the pilot line 65 with the tank 22 allowing the stop valve 53 to shift to the normal open position. When the pilot valve 57 is engaged by one of the cam
30 lobes 49 on the cam 48, the valve 57 is opened passing pilot fluid from the line 59 to the stop valve 53 closing the stop valve.

The pilot operated start valve 54 connected in the branch line 52b is normally closed and is opened by pilot
35 fluid from either one of the limit valves 55 and 56

conducted through the pilot line 61c leading from the shuttle valve 62. The limit valves 55 and 56 are identical in structure and function. Fluid is supplied to the limit valves 55 and 56 from the pump 50 through the lines 52 and 58 and through the dump valves 81 and 82 associated, respectively, with limit valves 55 and 56. The dump valve 81 supplies fluid to the limit valve 55 through the line 58a; the dump valve 82 supplies fluid to the limit valve 56 through the line 58b. The dump valves are bi-staple pilot operated valves opened and closed by pilot fluid. When open the dump valves pass fluid from the line 58 to the lines 58a and 58b associated with the limit valves 55 and 56, respectively. When closed, the dump valves communicate the lines 58a and 58b with the tank 22 allowing the dumping of fluid from the lines 58a and 58b. Each of the dump valves 81 and 82 is opened to enable the limit valve associated with the dump valve to pass fluid when the limit valve is opened by the cam C. Referring to Figure 2A, when the cam C is moving to the left, the dump valve 81 has been previously opened when the cam C engage the limit valve 56 allowing flow of fluid through the line 61b to the shuttle valve 62. At this time pilot fluid is passed through the line 61e to open the dump valve 81 so that the valve 81 is pre-conditioned to allow fluids to pass through the limit valve 55 when that valve is engaged by the cam C at the retract stroke limit. Similarly, at that point, the dump valve 82 is preconditioned by fluid passing through the lines 61d to allow fluid to pass through the limit valve 56 at the extend stroke limit.

As illustrated in Figure 2B, the system and logic circuitry for closing the dump valves includes a rotary release cam 83 and cam operated release valves 84 and 85. The cam 83 is mounted on and rotated by the shaft 44 of the motor 43 and driven in timed relation with the crank

45 and the stop cam 48. The valves 84 and 85 are spring biased two-way normally closed valves opened by the operator lobe on the cam 83. Pilot fluid is supplied to the valves 84 and 85 through the lines 52, 59 and 59a.

5 The valve 84 is associated with the dump valve 81 through pilot fluid line 86. Similarly, the valve 85 is associated with the dump valve 82 through pilot fluid line 87.

10 Referring to Figure 2B, the stop cam 48 and release cam 83 are timed so that when the motor 43 is started by the engagement of the cam C with the limit valve 55, the stop valve 53 is opened by disengagement of the cam lobe 49 from the stop pilot valve 57. The release valve 84 is opened by the cam 83 passing pilot fluid to the dump valve 81 which occurs before the opposite lobe 49 of the cam 48 re-engages the stop pilot valve 57. The passing of the pilot fluid in the line 86 to the valve 81 closes the valve 81 allowing the dumping of fluid from the line 58a to tank 22 permitting the start valve 54 to close. The start valve 54 is closed even though the limit valve 55 is still engaged by the cam C so that the motor 43 is stopped when the opposite lobe 49 of the stop cam 48 re-engages the stop pilot valve 57. The dump valve 81 will remain closed in the dump condition until pre-conditioned by the limit valve 56 at the extend stroke limit. With the operation of the motor 43 initiated by the limit valve 56 at the extend stroke limit, a similar operating cycle occurs with the lobe of release cam 83 operating the release valve 85 to deliver pilot fluid through the line 87 to the dump valve 82.

30 Briefly, the operation of the hydraulic systems of Figures 2A and 2B is as follows. With the cylinder 7 moving to the left extending the cam C toward the limit valve 55, the dump valve 81 has been previously opened to supply fluid to the limit valve 55. When the cam C

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approaches the limit of the stroke engaging the limit valve 55, pilot fluid is passed to the start valve 54 opening that valve and starting the motor 43. Simultaneously, pilot fluid is passed to the dump valve 82 through the line 61d opening that valve for a subsequent operation. The motor 43 first disengages the stop cam lobe 49 from the stop pilot valve 57 closing the stop pilot valve 57 removing pilot pressure from the stop valve 53 which is then opened by the stop valve spring. Shortly thereafter the lobe of the release cam 83 engages the release valve 84 closing the dump valve 81. Fluid is dumped from the line 58a and the connecting lines allowing the start valve 54 to close. The motor 43 continues to operate until the opposite stop cam lobe 49 engages the stop pilot valve 57 opening the valve 57 thereby closing the valve 53 stopping the motor 43. At the end of the stroke to the right in Figure 2A the cam C engages the limit valve 56. First, the start valve 54 is opened to start the motor 43 and simultaneously the dump valve 81 is opened for a succeeding operation. Again, the motor 43 first disengages a stop cam lobe from the stop pilot valve 57 followed by the engagement of the release valve 85 by the lobe of release cam 83. This closes the dump valve 82 to close the start valve 54 even though the limit valve 56 is held open by the cam 16. When the cams 48 and 83 again reach the condition illustrated in Figure 2B, the stop valve 53 is closed to stop the motor 43.

It will be recognized that the hydraulic power and logic system of Figure 2A and 2B as used to operate the well pump A of Figure 1 functions independently of the counterbalance system including the air receiver 14 and the compressor 15 which supply air into the outer cylinder 6 below the base piston 5. As the well pump reciprocates to raise and lower the sucker rod string 13, the weight of the rod string and the reciprocating parts

of the well pump is supported by the air supplied into the system beneath the piston 5. Thus, the hydraulic power system is relieved of this weight of such movable components, the sucker or polish rod string, and the fluid column in the well above the pump plunger during the upstroke. Thus the hydraulic system is primarily concerned with overcoming friction and accelerating and decelerating the movable masses involved in operating the pump A.

Referring to Figure 3, the hydraulic well pump B illustrated schematically is a variation of the pump A shown in Figure 1 wherein the only force supporting the pump load is contributed by the hydraulic cylinder. Counterbalancing is achieved by hydraulically supercharging the hydraulic pump supplying the pressure for lifting the sucker rod string. In Figure 3 those parts corresponding with similar parts of the pump A in Figure 1 will be referred to by the same reference numerals as used in Figure 1. The well pump B primarily differs from the well pump A by the elimination of the counterbalance piston 5 because the counterbalancing is obtained by supercharging the pump supplying the hydraulic pressure for the lift stroke. The lower end of the movable cylinder 7 is closed in sliding sealed relationship with the fixed piston 2 by the annular closure cap 7a. In the well pump B the hydraulic pump 20a is supercharged by a gas charged hydraulic accumulator N1 or a dead weight activated hydraulic accumulator W either of which is connected with the intake side of the pump 20a as shown in Figure 3. The hydraulic power and logic circuitry of Figures 4 and 2B taken together may be used to operate the well pump B. The portion of the system shown in Figure 2B is exactly the same as that portion of the stem described in connection with the operation of the well pump A. The portion of the circuitry shown in Figure 4

differs only in the inclusion of the hydraulic accumulators. Referring back to Figures 3 and 4, the notations V1 and V2 as used in Figure 3 designate the right and left sides, respectively of the reversing valve 40 shown in Figure 4. Referring to Figure 3, during the lift stroke of the well pump B, hydraulic fluid pressure is supplied by the pump 20a through the line 23a into the conduit 3 raising the pressure in the chamber 104 above the piston 1 lifting the movable cylinder 7. The reversing valve side V1 is closed forcing the pressure in the accumulator W or N2, which ever is being used, to supply supercharging pressure into the intake of the pump 20a thereby enhancing the lift of the pump. Hydraulic fluid below the piston 1 returns as the cylinder 7 is raised through the flow channel 2 along the line 23b and through the open reversing cylinder side V2 back to the tank 22. During the downstroke of the well pump B the reversing valve side V2 is closed whereby the output pressure from the pump 20b must flow through the line 23b into the flow passages 2 and outwardly through the ports 4 into the cylinder 7 below the piston 1 forcing the movable cylinder 7 downwardly. During the downward stroke the reversing valve side V1 is open permitting counterbalancing hydraulic fluid pressure to be effective from either the accumulator W or in the accumulator N2 along the line 41c, the line 23a, and the flow conductor 3 into the cylinder chamber 104 above the piston 1 which opposes the downward movement of the cylinder 7. The hydraulic power fluid and logic circuitry of Figures 4 and 2A taken together operate the hydraulic well pump B in exactly the same manner as previously described with respect to the system of Figures 2A and 2B in operating the well pump A. Looking at Figure 4, when the reversing valve 40 is shifted to the left communication between lines 41a and 41c is closed while the line 41b is opened to the tank

22. In this mode of operation the flow from the accumulators W or N2 can only pass to the intake of the pump 20a which discharges into the line 23a flowing to the cap end of the cylinder 7 for operating the pump in the upstroke. The pump 20a is thus supercharged from one of the hydraulic accumulators. During the downstroke the valve 40 is shifted to the right closing off flow in the line 41b to the tank. The pump 20b discharges into the line 23b pumping the cylinder 7 downwardly while the hydraulic accumulator W or N2 is communicated to the line 41a applying the pressure from the accumulator from the line 23a into the cap end of the cylinder 7. With the exception of the hydraulic accumulators, the remainder of the power and logic circuitry for the hydraulic well pump B as illustrated in Figures 4 and 2B taken together operates exactly as previously described in connection with the well pump A.

The control valve device or mechanism 151 illustrated in Figures 5 - 17 provides the operation requirements of the following components of the hydraulic power and logic system shown in Figure 2A and 2B: valve 40; coupling 46; crank arm 45; valve 85; cam 83; valve 84; valve 57; cam 48 with the cam lobes 49; and the logic drive motor 43. Utilizing the control valve mechanism 151, it is to be understood that the other components of the system of Figures 2A and 2B are connected with the control valve mechanism.

Referring to Figures 5 and 6, the valve control mechanism 151 includes a body 200 mounted on a bracket 201. The hydraulic drive motor 43 is secured to the back of the body. The valves 57 and 84 are mounted on top of the body. The valve 85 is supported from the bottom of the body. Identical poppet valve assemblies 202 are mounted on opposite sides of the body providing valve functions corresponding with the opposite side or end

sections of the valve 40 for controlling the extend and retract functions of the hydraulic cylinder assembly. The details of the body 200 are shown in Figures 7 - 9. The body has a central rectangular portion 203 provided with an internal rectangular cavity 204. Cylindrical valve body portions 205 extend from the opposite sides of the body for housing the poppet valve assemblies 202. The valve bores of the body portions 205 communicate through cylindrical bores 210 to the central cavity 204 of the body. As seen in Figure 8, the top of the body is provided with a forward opening 211 for the valve 57 and a rearward opening 212 for the valve 84. Similarly the bottom of the body, Figure 9, is provided with an opening 213 for the valve 85. The poppet valve body portions 205 each has a poppet valve inlet 214 and a poppet valve outlet 215. The valves 57, 84, 85, and the poppet valves 202 are operated by the hydraulic motor 43 through cam and cross head structure mounted on the motor shaft. Referring to Figures 5 and 6, a cam crank 220 is held on the motor shaft 221 of the motor 43 by a retainer 222 secured by a bolt 223. A key 224 is positioned in aligned slots of the shaft and crank for driving the crank as the shaft rotates. As shown in Figures 12 and 13, the cam crank 220 has an integral cam 225 for operating the valves 84 and 85 as the cam crank is turned by the motor. The cam also has an integral cross head shaft 230 for driving the cross head of the valve control mechanism. A cross head 231, Figures 10 and 11, is coupled with the cross head shaft 230. The cross head has a vertical slot 232 through which the cross head shaft extends. A bushing 233 is fitted on the cross head shaft within the slot 232. The cross head and bushing are held on the cross head shaft by a thrust washer secured on the cross head shaft by a lock ring 235. The cross head shaft has horizontally spaced sidewardly opening slots 240 for

coupling the poppet valve assemblies 202 with the cross head. The cams 49 are secured in horizontal spaced relation in the top portion of the cross head in an upwardly opening recess 241 by cap screws 242. As the cross head reciprocates horizontally the cams 49 operate the valve 57.

The poppet valve assemblies 202 of the valve control mechanism 151, Figures 5 and 14 to 17, each includes a valve seat 250, a valve 251, and a valve operator spool 252. As shown in Figure 14, the valve seat has a cylindrical externally threaded outer portion 253 which secures the valve seat in the body portion 205. The valve seat also has a tubular inner portion 254 provided with circumferentially spaced elongated flow ports 255. The valve seat portions 254 has a seat surface 260. The bore of the body portion 205 is enlarged along the valve seat providing an annular poppet valve discharge chamber 261 which communicates with the discharge opening 215 in the body portion 205. A ring seal 212 around the valve seat portion 254 seals between the valve seat and the poppet valve body portion 205 inward from the discharge chamber 261. Referring to Figure 15, the valve 251 has a tubular portion 270 which telescopes into the valve seat tubular portion 254. The tubular portion 270 of the valve is provided with four circumferentially spaced elongated discharge ports 271 which are circumferentially aligned with the discharge port 255 of the valve C so that fluid within the valve portion 270 flows outwardly through the ports 271 of the valve and through the ports 255 of the valve seat into the discharge chamber 261. The valve has an enlarged body portion 272 and an external annular tapered valve seat 273 between the tubular portion 270 and the body portion. The valve seat 273 on the valve is engageable with the valve seat 260 on the valve seat. The tubular portion 270 of the valve fits in close sliding

relationship within the tubular portion 254 of the valve seat so that as the valve is moved relative to the valve seat in an axial direction, a linear relationship exists between the valve discharge ports 271 and the valve seat so that the flow rate through the valve is directly proportional to the distance travelled by the valve. For example if the valve is moved 25% of its total travel, the flow rate therethrough is changed 25 percent. The body portion of the valve is secured with the valve spool 252 by a retainer screw 274. As seen in Figure 16 the valve spool 252 has an endwardly opening internally threaded blind bore 275 for engagement of the retainer screws 274 in the spool. The bore of the body portion 205 along the valve and spool is enlarged to provide an annular inlet chamber 280 which communicates with the poppet valve inlet port 214. To provide for a tight shut-off between the valve seat and the poppet valve, an area differential between the poppet seal area and the area of the spool is provided so that the shut in pressure within the chamber 280 biases the poppet valve toward the seat. The inward end of the valve spool has upwardly and downwardly opening recesses 281 and flange portions 282 for coupling the valve spools with the cross head in the slots 240 of the cross head. The front of the body 203 of the valve control mechanism is closed by the plate 283 so that the chamber 204 in which the cams and cross head operate is sealed. Such chamber is communicated with the fluid reservoir of the system when the valve control mechanism is connected into the power and logic system such as shown in Figures 2A and 2B. The

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spool 252, and the retainer screws 274 have a longitudinal axial bore 284 which communicates the chamber 204 with the chamber 261 both of which are at reservoir pressure so that there is no pressure differential across the spool.

When the valve control device 151 is connected in a hydraulic power and logic system such as that shown in Figures 2A and 2B, the driving of the hydraulic motor 43 turns the cam crank 220 rotating the cam lobe 225 and the cross head shaft 230 which causes the cross head 231 to reciprocate horizontally. As the cam lobe 225 rotates the valves 84 and 85 are operated. As the cross head reciprocates the cam lobes 49 connected with the cross head operate the valve 57. Since the cross head is coupled with the poppet valve spools 284 reciprocation of the spools opens and closes the poppet valves performing the valving function of both sides of the valve 40. At midposition of the cross head both of the poppet valves are open and thus the chambers 202 of both poppet valves communicate with the chambers 261 of the poppet valves so that the pumps 20a and 20b both communicate with the reservoir and thus are not operating the hydraulic well pump cylinder 7. At each extreme side position of the cross head, the poppet valve on the side to which the cross head is nearest is closed while the opposite poppet valve is fully open. The relationship between the ports in the valve and the ports in the valve seat of the poppet valve provides linear opening of each of the poppet valves so that the valves flow in direct proportion to the extent to which the valve is open. This arrangement provides for direct control of the acceleration and deceleration of the hydraulic well pump which is dependent upon the rates of opening and closing the poppet valves. In other words, the rate at which the hydraulic well pump accelerates or decelerates is

directly proportional to the rate at which the poppet valves are opened and closed. That rate is controllable by the rate at which the motor 43 is operated which in turn may be controlled by a manual control of the
5 metering valve 60 in the line 52 supplying hydraulic drive fluid to the motor 43. One of the poppet valves controls the cylinder extension in the hydraulic well pump while the other of the valves controls the cylinder retraction. The hydraulic motor driven cross head or
10 "scotch yoke" mechanism when operating uniformly causes the two poppet valves to alternately open and close in a velocity pattern of harmonic motion. The cam lobes 49 on the cross head operate the limit valve 57 at the extreme right and left positions of the cross head.

15 Reference is made to our U.K. patent specification No. 2131890 from which the present specification has been divided.

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CLAIMS

1. A device for direction control of a double action hydraulic cylinder comprising a first fluid control valve adapted for connection to the extend power end of the hydraulic cylinder and a second fluid control valve adapted for connection to the retract power end of the hydraulic cylinder, the first and second valves being drivingly connected to one another by the cross-head of a scotch-yoke and arranged such that when one valve is being moved to its fully open position to permit flow of pressurised fluid to its respective power end of the hydraulic cylinder the other valve is being moved to its closed position such that when said one valve is fully open the other valve is closed, and a rotary fluid motor connected to the scotch-yoke to reciprocate the cross-head.
2. A device according to Claim 1 wherein a flow control valve is provided for controlling flow of pressurised fluid to the fluid motor such that the speed of rotation of the output of the fluid motor can be controlled by said flow control valve.
3. A device according to Claim 1 or 2 wherein the first and second valves are poppet valves each having a valve member connected with the scotch-yoke cross-head.
4. A device according to Claim 3 wherein said poppet valves each include port means providing linearly progressive opening of the valve from its closed to its fully open position.
5. A device according to any preceding claim including hydraulic control means for stopping the fluid motor when the scotch-yoke cross-head reaches its opposed limits of reciprocal movement, the hydraulic control means including a pilot valve arranged to operate a main valve to prevent flow to the motor and cam means on the cross-head for operating the pilot valve to close said



main valve when said opposed limits are reached.

6. A device according to Claim 5 wherein the hydraulic control means further comprises a pilot valve which is arranged to operate a main valve to supply pressurised fluid to the fluid motor on completion of the extend or retract stroke of the hydraulic cylinder, limit valves being provided adapted for co-operation with the stroke of the hydraulic cylinder, each limit valve causing operation of the pilot valve when sensing the termination of said stroke.

7. A device for direction control of a double action hydraulic cylinder substantially as described in Figures 2A, 2B, 4 to 17.

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