

Dec. 29, 1953

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2,664,102

HYDRAULIC PUMPING SYSTEM AND BY-PASS VALVE THEREFOR

Filed March 19, 1948

3 Sheets-Sheet 1

Fig. 1.

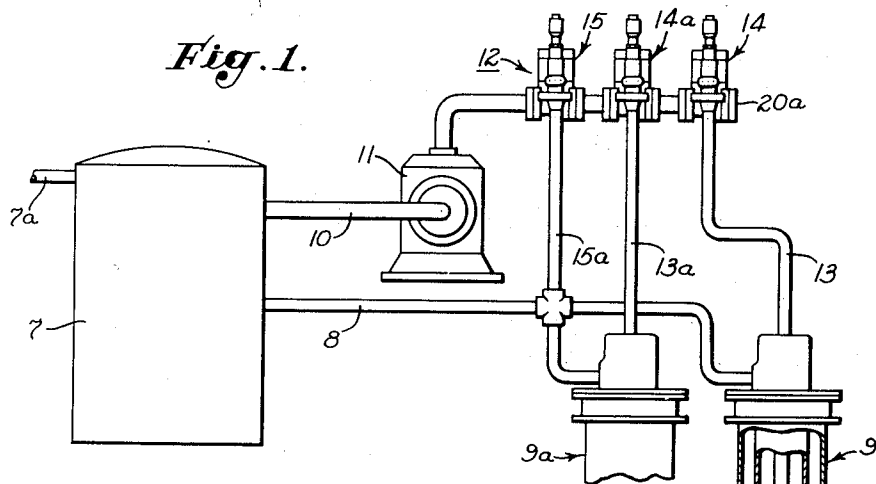
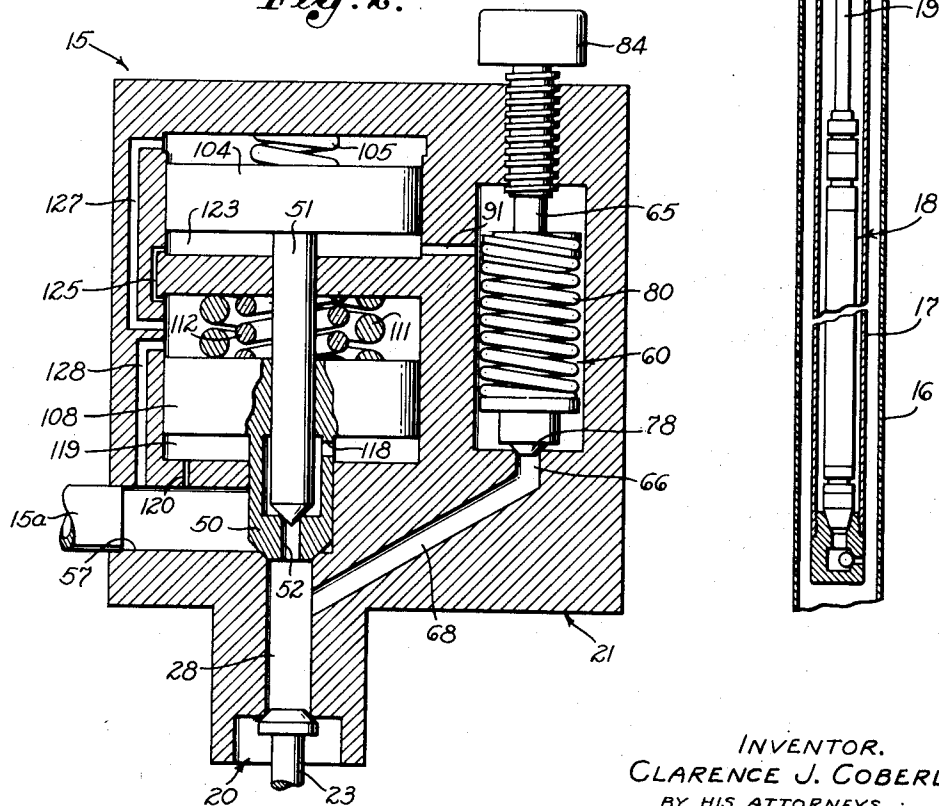


Fig. 2.



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3 Sheets-Sheet 2

Fig. 3.

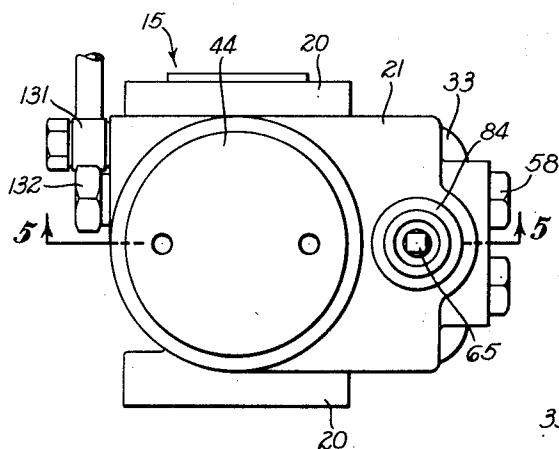


Fig. 4.

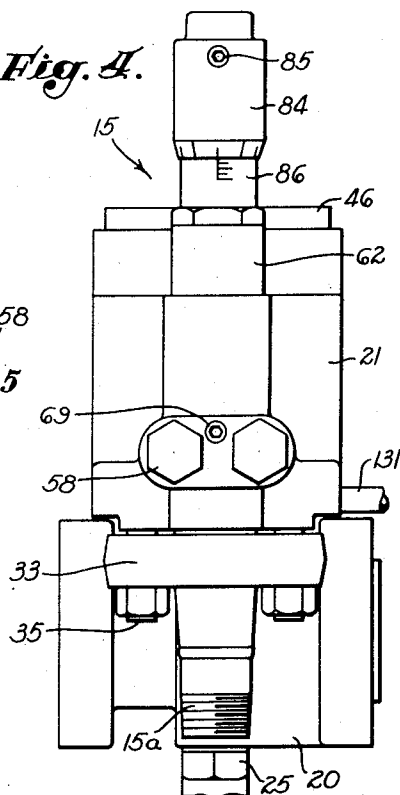
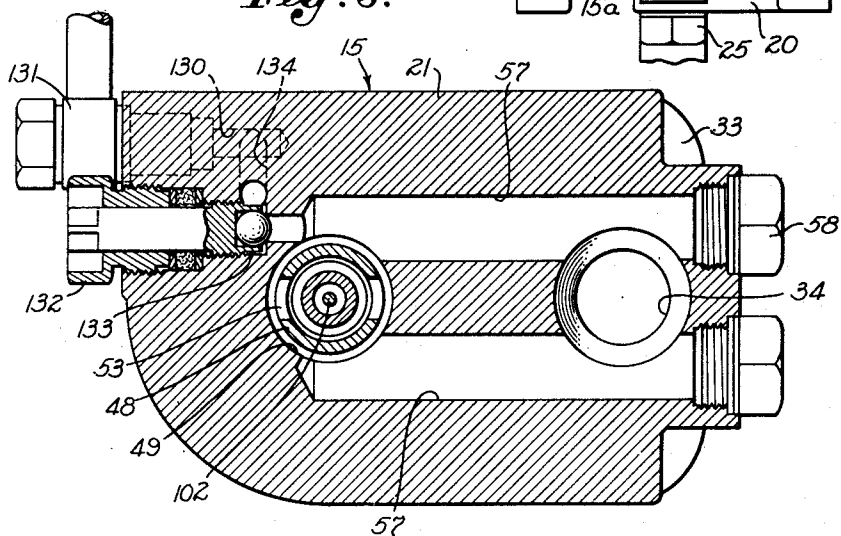


Fig. 6.



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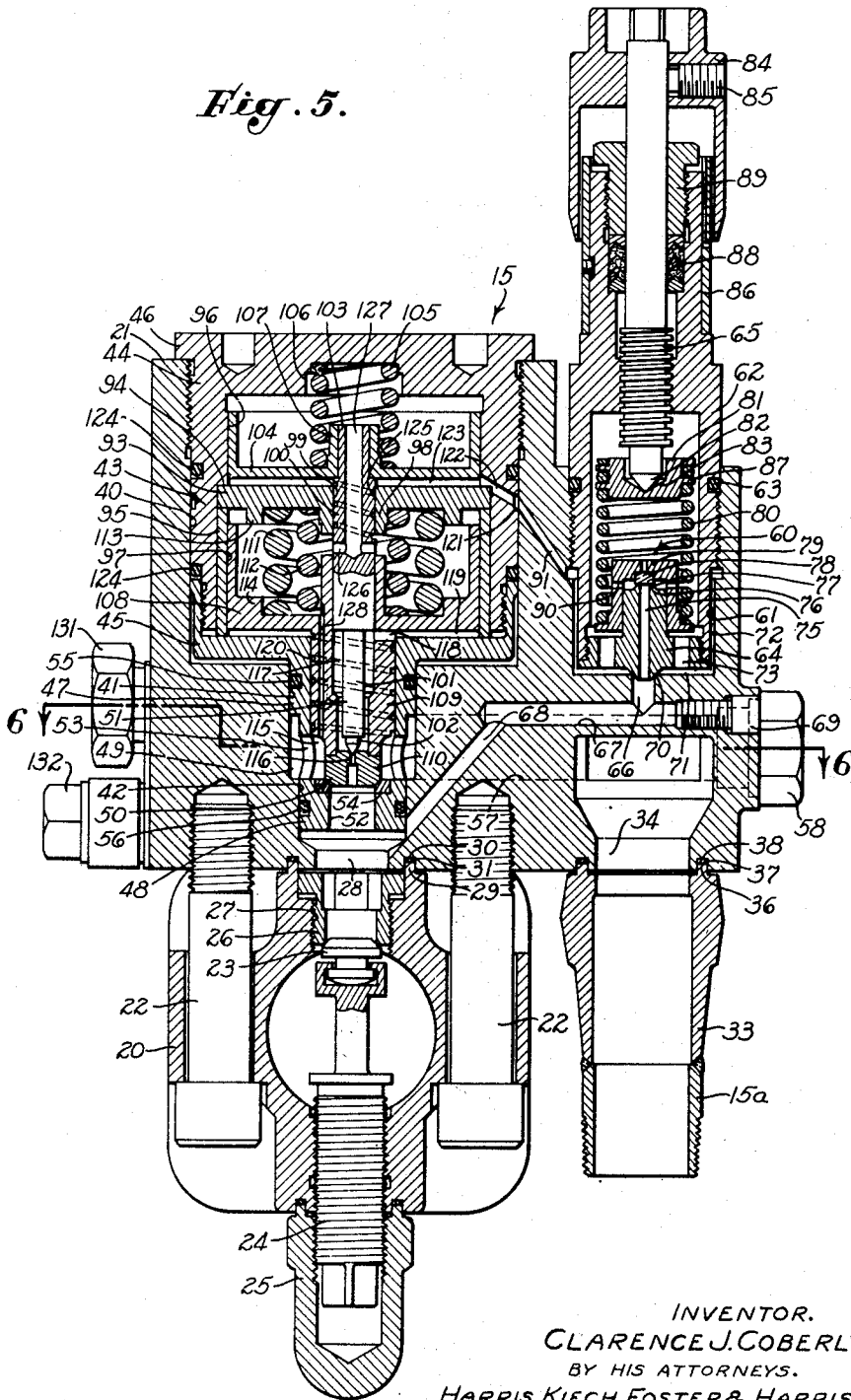
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3 Sheets-Sheet 3

Fig. 5.



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UNITED STATES PATENT OFFICE

2,664,102

HYDRAULIC PUMPING SYSTEM AND
BY-PASS VALVE THEREFOR

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Application March 19, 1948, Serial No. 15,848

10 Claims. (Cl. 137—490)

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This invention relates to valves for the relief of pressure and more particularly to valves for by-passing fluid in systems of hydraulic pumping wherein a large quantity of high pressure operating fluid must be handled at an accurately controlled pressure.

The invention is of particular utility in the oil industry, and is described in connection with such use, although its utility is not limited to that field.

In the operation of fluid-operated pumps in oil wells, fluid is pumped at high pressure to the pumps located within the wells, and this operating fluid, which enters the pumps and causes movement of their mechanism, is usually crude oil which has been filtered and cleaned of foreign materials. The wells are frequently of great depth and the pressure in the supply lines extending down the wells to the pumps is the pressure of the hydrostatic head in the tubing multiplied by the area ratio of the engine to pump piston in the bottom hole unit. The individual supply lines and pumps are therefore usually provided with protective devices by which the pressure applicable to them may be limited.

It is common practice in the oil industry to operate a number of fluid-operated well pumps from a central pumping station, the pumping station including one or more conventional triplex surface pumps connected so as to supply power oil through a supply manifold and piping to all of the well pumps in the system. In such a system, the well pumps are normally operated at different depths and at different rates, the speed of operation of each well pump being controlled and governed by a flow governor, such as shown in my Patent No. 2,119,737, for example, which is commonly installed in the power oil tubing leading to each of the well pumps. The maximum fluid pressure developed in the supply manifold by the central pumping station as a practical matter must be maintained at a value at least slightly greater than that required for the well pump in the system having the greatest pumping head and requiring power oil of the highest pressure. In addition, the total volume of power oil delivered by the central pumping station to the supply manifold must be slightly greater than enough to operate all of the well pumps in the system when they are operating at their maximum production capacities, so as to provide a slight excess of pressure to the supply manifold. Consequently, there is also an excess volume of power oil delivered by the central pumping station to the supply manifold, and

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this excess fluid must be continuously by-passed from the supply manifold to the low pressure reservoir from which the triplex pumps draw their supply. Also, in the operation of such a system, frequently the operation of one or more of the well pumps is stopped for repairs or for other purposes which, of course, increases substantially the amount of power oil that must be by-passed from the supply manifold to the reservoir. If all of the well pumps are stopped at the same time, of course, all of the power oil from the central pumping station must be by-passed. It is important that the fluid pressure of the power oil be maintained with a variation of not to exceed 2% and, consequently, some means is desired which will automatically by-pass the power oil from the supply manifold ranging from the full capacity of the system down to zero capacity, without materially varying the pressure of the power oil remaining in the system.

The triplex surface pumps in such a system are usually provided with conventional pressure relief valves of the spring-held poppet type, but such valves are not intended for use as by-pass valves, and are usually incapable of by-passing a wide range of capacities without wide pressure fluctuations in the manifold pressure. At the normally high fluid pressures involved in such a system, normally in excess of 2000 pounds per square inch, conventional pressure relief valves are wholly inadequate.

It is therefore a primary object of my invention to provide a pressure controlled by-pass valve for a hydraulic pumping system having incorporated therewith means for overcoming the above discussed undesirable conditions and difficulties.

It is a further object of my invention to provide a pressure by-pass valve that will discharge a large volume of fluid at a pressure closely limited to a predetermined value.

Another object of my invention is to provide a pressure controlled by-pass valve which will maintain pressure at a definite value while discharging a widely variable volume of fluid.

A further object of my invention is to provide a pressure controlled by-pass valve operable without chattering at a pressure so closely limited as to be substantially constant, and capable of discharging a large volume of fluid.

Yet another object of my invention is to provide a pressure by-pass valve adapted to discharge into a closed system but so constructed that the effect of back pressure in the closed system upon the operation of the valve is minimized

and the pressure at which the valve operates is not materially varied thereby.

A still further object of my invention is to provide a pressure controlled by-pass valve in which a main valve is operated in response to an actuating valve assemblage discharging into the same system as the main valve, which system may be closed and under pressure.

Another object of the invention is to provide a by-pass valve which may be used in connection with a manifold flow control valve, such as shown in my copending application Serial Number 748,921, filed May 19, 1947, to form units of a manifold assembly.

Other objects and advantages will appear in the following specification and in the drawings, in which:

Fig. 1 is a utility view, diagrammatically showing a fluid-operated pumping system embodying my invention;

Fig. 2 is a simplified diagrammatic view illustrating the principal parts and passages of my invention;

Fig. 3 is a plan view of my invention shown as installed upon a section of a manifold;

Fig. 4 is an elevational view taken from the right of Fig. 3;

Fig. 5 is a vertical sectional view, on an enlarged scale, taken on the line 5—5 of Fig. 3; and

Fig. 6 is a horizontal sectional view, taken on the line 6—6 of Fig. 5.

Referring to Fig. 1, I show a settling tank 7 having a discharge pipe 7a, a production pipe 8 adapted to convey well production from a plurality of wells 9 and 9a, and a supply pipe 10. The supply pipe 10 connects with a triplex surface pump 11 adapted to draw well fluid from the settling tank 7, which acts as a reservoir, and is adapted to supply it as power oil under relatively high pressure to a supply manifold 12 and thence through suitable supply tubings 13 and 13a to each of the wells 9 and 9a, respectively, through metering valves 14 and 14a, respectively. The metering valves 14 and 14a are preferably of the type shown and described in my copending application Serial Number 748,921, filed May 19, 1947, and are each adapted to supply power oil at a selected volume and pressure to their respective wells, which volumes and pressure are normally different. Although I have shown only the two wells 9 and 9a, it is to be understood that normally a large number of such wells may be supplied with power oil from the triplex pump 11, or a number of such pumps. It is also to be understood that although only one triplex pump 11 is shown, for simplicity, a multiple well installation will normally utilize a number of such pumps operating to deliver power oil from the settling tank 7 to the supply manifold 12, and such triplex pumps, whether one or a number, may be installed at a central pumping station which may be remote from the wells supplied thereby.

Also connected to the supply manifold 12 is a by-pass valve 15, which constitutes a principal element of the present invention, the same being adapted to provide fluid communication between the supply manifold 12 and the production pipe 8 through a discharge pipe 15a, as will be described in detail hereinafter.

The well 9 is a conventional installation, including a perforated casing 16 having a production tubing 17 set therein and a fluid-operated well pump 18 set in the production tubing. The well pump 18 is supplied with power oil to actuate it through power oil tubing 19 which connects

with the supply tubing 13. As will be understood, the production pipe 8, or a branch thereof, connects with the production tubing 17 and is adapted to convey the well production from the fluid-operated well pump 18 to the settling tank 7. The well 9a is provided with a similar bottom-hole installation (not shown), and is similarly connected to the supply tubing 13a and the production pipe 8. As will also be understood by those skilled in the art, the well pumps, such as 18, may and probably will, be set at different levels in their respective wells, pump against different heads, and otherwise operate under different bottom-hole conditions, requiring different volumes of power oil to be delivered thereto at different pressures.

As best shown in Figs. 3-6, the by-pass valve 15 includes a manifold section, comprising a part of the supply manifold 12, to which a housing 21 is secured by bolts 22. The manifold section 20 forms part of the supply manifold 12, the metering valves 14 and 14a being provided with identical manifold sections, all of which are adapted to be bolted together in longitudinal alignment with a suitable cap 20a closing the end section. As will be apparent, such a manifold construction permits any desired number of metering valves, such as the valves 14 and 14a, to be readily connected or disconnected from the pumping system, which readily permits the addition of additional well pumps to the system, or the withdrawal thereof from the system, and this is a further valuable feature of the invention.

The manifold section 20 is provided with a manually-operated valve 23, which is mounted on a valve stem 24, protected and secured by a locking cap 25, and which seats upon a valve seat 26 threadedly secured in a port 27 aligned with an inlet port 28 in the housing 21. The manifold section 20 and the housing 21 are sealed to each other by an annular ridge 29 on the manifold section which fits within an annular groove 30 in the housing and is there held against a gasket 31. The discharge pipe 15a is secured to a flange member 33, which is secured to the housing 21 in alignment with a discharge port 34 by studs 35, and is sealed to the housing in the same manner as the manifold section 20 by an annular ridge 36 held against a gasket 37 in an annular groove 38.

The housing 21 is provided with a cylindrical chamber 40 in axial alignment with the inlet port 28 and connected thereto by a coaxial bore 41 and a second coaxial bore 42 of less diameter than the bore 41. A valve casing 43, comprising an upper casing member 44 and lower casing member 45 threaded thereto, is threadedly secured in the chamber 40, the upper casing member 44 having a flange 46 limiting the entry of the valve casing 43 into the chamber 40 so as to cause registering of connecting passages hereinafter described. The lower casing member 45 has an extension 47 fitting closely within the bore 41 and extending partly therethrough, and a further extension 48 fitting closely within the bore 42. The reduced diameter of the extension 48 leaves an annular space 49 around that part of the extension 48 contained in the bore 41.

A main valve 50 and a valve pilot 51 are housed within the valve casing 43, with the main valve 50 in communication with the inlet port 28 by means of an axial passage 52 in the lower end of the extension 48 and in communication with the annular space 49 by means of radial ports 53 in the extension 48. A valve seat 54 is provided for

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the main valve 50 at the upper end of the axial passage 52 adjacent the radial ports 53. Sealing rings 55 on the extension 47, and 56 on the extension 48, seal the lower casing member 45 relative to the housing 21 above and below the radial ports 53. Main flow passages 57 (best shown in Fig. 4) extend into the housing 21 from the end thereof adjacent the discharge port 34, and are spaced apart so as to communicate tangentially with the discharge port 34 and with the annular space 49 around the main valve 50, providing a flow channel through the by-pass valve 15 when the main valve 50 is open. Plugs 58 close the outer ends of the passages 57.

A master control valve 60 is mounted in a chamber 61 formed in the housing 21 between the main flow passages 57. The master control valve 60 has a valve body 62 threadedly secured within the chamber 61 and sealed therein by a sealing ring 63, and a nozzle 64 threadedly secured in the lower end of the valve body 62. An adjusting screw 65 is movably threaded in the valve body 62 to regulate a spring loaded pressure-release mechanism mounted between the adjusting screw 65 and the nozzle 64. The chamber 61 has a coaxial bore 66 at its lower end, communicating through a horizontal passage 67 with an inclined passage 68 which leads from the inlet port 28 on the upstream side of the main valve 50. The horizontal passage 67 is closed at its outer end by a plug 69. A chamfered boss 70 on the lower face of the nozzle 64 is threaded into the lower end of the valve body 62 so as to leave a space 71 therebelow in the lower portion of the chamber 61. The lower portion of the valve body 62 is proportioned to leave an annular space 72 between the valve body 62 and the wall of the chamber 61. Circumferentially disposed longitudinal passages 73 through the nozzle 64 lead from the interior of the valve body 62 to the space 71 and thence to the annular space 72.

An axial passage 75 extends through the nozzle 64 and through an axial upward extension 76 which is chambered to terminate in a knife-edge valve seat 77 annularly surrounding the axial passage 75. An optically flat valve head 78, of hard metal or sapphire, is held by a yoke 79 which has a sliding fit on the axial extension 76. The valve head 78 is pressed against the valve seat 77 by a spring 81 bearing against the yoke 79 and reacting against a spring follower 81 which is engaged by the adjusting screw 65 and so positioned as to give the spring 80 any desired initial loading. To allow the valve head 78 to rotate and thereby to wear evenly on the valve seat 77, the adjusting screw 65 has a conical point 82 engaged by a more obtuse conical recess 83 in the spring follower 81. A micrometer thimble 84 is secured to the upper end of the adjusting screw 65 by a set screw 85, to register with a calibrated skirt 86. The thimble 84 and skirt 86 may be calibrated in pounds per square inch or other convenient units indicative of the pressure against the valve head 78. To permit adjusting the spring 80 to an exact spring rate to which the thimble 84 and skirt 86 may be calibrated, the spring is threaded upon a round bottom helix 87 formed on the outside of the spring follower 81, and is soldered thereto when correctly adjusted. Packing 88, compressed by a packing nut 89, is provided to prevent leakage along the stem of the adjusting screw 65.

The yoke 79 has radial ports 90 at the level of the chamfered portion of the nozzle extension 76 admitting fluid which has passed between the

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valve seat 77 and the valve head 78 to the interior of the valve body 62. Such fluid may then pass through the passages 73 in the nozzle 64 to the space 71 below the nozzle and to the annular space 72 surrounding the valve body 62. An inclined passage 91 through the housing 21 leads from the annular space 72 to the interior of the chamber 40, to convey fluid from the control valve 60 to the following described mechanism for operating the main valve 50.

The interior of the valve casing 43 is divided by a transverse partition 93 held against a shoulder 94 in the upper casing member 44 by a lining sleeve 95, so as to form an upper cylinder 96 and a lower cylinder 97. The partition 93 has a downwardly extending central boss 98 perforated with an axial bore 90 which forms a guide or bushing for the stem 100 of the pilot valve 51. The stem 100 has a lower contracted portion 101 terminating in a needle 102, and an upper contracted portion 103 tightly pressed into a pilot valve piston 104, which is reciprocable in the upper cylinder 96. A spring 105, guided in a recess 106 in the upper casing member 44 and by a boss 107 on the pilot valve piston 104 urges the piston 104 and the attached needle 102 downwardly.

A main valve piston 108 is reciprocably mounted in the lower cylinder 97, with a sliding fit on both the lining sleeve 95 and the pilot valve stem 100, and has a downwardly extending tubular piston rod 109 terminating in a main valve head 110 designed to seat in the valve seat 54. Springs 111 and 112, reacting against the partition 93 and guided by shoulders 113 on the latter and 114 on the piston 108, urge the piston downwardly to normally maintain the main valve 50 closed. The lower portion of the tubular piston rod 109 is contracted in external diameter to provide an annular space 115 within the extension 47 of the lower casing member 45 which is always in registry with the radial ports 53 at any position within the scope of movement of the tubular piston rod 109. The main valve head 110 is perforated with a passage 116 into which the needle 102 seats to form a closure.

When the needle 102 is lifted from the passage 116, fluid may flow from the inlet port 28 through the passage 116 to an annular space 117 between the lower contracted portion 101 of the pilot valve stem 100 and the wall of the tubular piston rod 109. A radial port 118 in the tubular piston rod 109 permits this fluid to enter a space 119 below the main valve piston 108 where the fluid pressure tends to compress the springs 111 and 112 and to open the main valve 50. A helical groove 120 on the outer surface of the tubular piston rod 109 then permits the fluid to flow to the annular space 115 and thence through the radial ports 53 and the main flow passages 57 to the discharge port 34, but forms a resistance which preserve the fluid pressure in the space 119 at a higher level than the pressure in the main flow passages 57.

When the control valve 60 opens, as will be described hereinafter, power oil under high pressure may flow from the inlet port 28 through the inclined passage 68 and the horizontal passage 67, the conical bore 66, the axial passage 75, the radial ports 90, the passages 73, space 71, and annular space 72 into the inclined passage 91 from which the fluid passes to an annular groove 121, on the outer surface of the upper casing member 44, and thence through a passage 122 in the upper casing member 44 to a space 123 between the partition 93 and the pilot valve pis-

ton 104, where the pressure of the fluid tends to move the pilot valve piston upwardly and, consequently, to open the pilot valve 51. The upper casing member 44 is sealed to the wall of the chamber 40 above and below the annular groove 121 by sealing rings 124, to prevent loss of pressure. A helical groove 125 on the outer surface of the pilot valve stem 100 permits the fluid to flow from the space 123 to radial ports 126 in the pilot valve stem 100, the groove being of small cross-sectional area so as to resist loss of pressure from the space 123. The helical groove 125 communicates with the upper side of the main valve piston 108, entering the lower cylinder 97 between the main valve piston 108 and the central boss 98 on the partition 93. The radial ports 126 communicate with the upper side of the pilot valve piston 104, through an axial passage 127 in the pilot valve stem 100. The pressure residual in the fluid after passing through the helical groove 125 is therefore available to supplement the spring 105 in tending to close the pilot valve 51, and to supplement the springs 111 and 112 in tending to close the main valve 50. Fluid entering the upper cylinder 96 above the pilot valve piston 104 is returned to the radial ports 126 through the axial passage 127 when the piston rises, and so enters the lower cylinder 97. Fluid entering the lower cylinder 97 on the upper side of the main valve piston 108 escapes therefrom by a longitudinal escape vent 128, provided in the external surface of the tubular piston rod 109 and which communicates with the annular space 115 and thence through the main flow passages 57 to the discharge port 34. The escape vent 128 is of such cross-sectional area that it does not act as a restriction to the flow of the fluid therethrough, but permits the fluid pressure on the upper, or low-pressure, sides of the pistons 104 and 108 to be approximately the same as the pressure in the production pipe 8, thereby obtaining higher pressure differentials across the pistons 104 and 108 when fluid is admitted to the lower, or high-pressure, sides of those members so as to more effectively move them.

As best shown in Fig. 6, to allow for drainage when it is desired to remove the valve casing 43 or other operating parts, a drainage bore 130 is provided in the housing 21, and is fitted with a drain pipe 131 secured to its outer end by any suitable fittings and leading to any convenient pit or receptacle. A valve 132, seated in a bore 133 of stepped diameter in continuation of one of the main flow passages 57, opens and closes communication between that main flow passage and the drainage bore 130 through a connecting passage 134.

In the operation of the by-pass valve 15 with particular reference to Figs. 1 and 2, it may be cut in or out of the supply manifold 12 of which the manifold section 20 is a part by manually opening or closing the manual valve 23, respectively.

When the manual valve 23 is opened power oil under high pressure flows from the supply manifold 12 and the manifold section 20 into the inlet port 28 of the housing 21. The main valve springs 111 and 112 are selected so as to exert a closing force on the main valve piston 108 and, consequently, on the main valve 50 equal to the maximum fluid pressure desired or to be allowed in the supply manifold 12. Thus, for example, if the maximum fluid pressure desired or allowable in the supply manifold 12 is 5000 pounds

per square inch, the springs 111 and 112 are selected so as to maintain the main valve 50 closed until the fluid pressure in the inlet port 28 exceeds this value. When the fluid pressure in the inlet port 28 exceeds such value for which the main valve springs 111 and 112 are selected, they are overbalanced by such excessive fluid pressure and the main valve pops open, permitting power oil to flow from the supply manifold 12 past the main valve 50, through the radial ports 53 and the annular space 49 into the main flow passages 57, from which such fluid flows into the discharge pipe 15a and thence through the production pipe 8 into the settling tank 7. It will therefore be appreciated that the main valve 50 acts as a pressure relief valve to limit the fluid pressure in the manifold section 20, and, consequently, in the supply tubing 13 and 13a, to a value below a predetermined maximum. This pressure-relief action of the by-pass valve 15 is an important feature of the invention, but is secondary to the by-pass function of the valve which will now be described.

When the manual valve 23 is opened, power oil under high pressure also flows through the inclined passage 68, the horizontal passage 67, the coaxial bore 66, and into the axial passage 75, to exert an upward force on the bottom of the valve head 78 tending to move the valve head up off its seat 77 to open it, against the action of the spring 80. The downward force exerted by the spring 80 on the valve head 78 tending to maintain the valve head closed may be varied as desired by rotating the thimble 84, to increase or decrease the compression on the spring 80. The compression of the spring 80 may thus be varied from zero to 100%. As will be understood, however, if the compression of the spring 80 is increased to a value such that the closing force exerted by it on the head 78 more than balances the fluid pressure on the upstream side of the main valve 50 for which the main valve springs 111 and 112 have been selected, the main valve 50 will simply open to relieve excess pressure. However, if the compression of the spring 80 is adjusted to a value overbalanced by the fluid pressure in the inlet port 28 and axial passage 75, but lower than the fluid pressure for which the main valve springs 111 and 112 have been selected, e. g., lower than 5000 pounds per square inch, then such fluid pressure will move the valve head 78 upwardly, permitting fluid to flow therepast, through the radial ports 90, the interior of the valve body 62, the passages 73, the space 71, and into the annular space 72 around the lower end of the valve body 62. From there, such fluid flows through the inclined passage 91, the annular groove 121, and the passage 122, into the space 123 below the pilot valve piston 104.

When high pressure fluid flows into the space 123 below the pilot valve piston 104, it exerts an upward force on the pilot valve piston tending to move it upwardly against the action of the pilot valve spring 105. Although the space 123 is in communication with the low pressure existing in the discharge pipe 32, as described above, such communication is restricted by the relatively small cross-sectional area of the helical groove 125 and, consequently, there is a substantial fluid pressure drop through the groove. The spring 105 is selected so that only a small fluid pressure drop through the helical groove 125 will be sufficient to build up enough fluid pressure beneath the pilot valve piston 104 to overbalance the spring and raise the piston. As soon as the fluid

pressure in the space 123 rises to a value at which it overbalances the force exerted by the spring 105 and the weight of the pilot valve piston assembly, the pilot valve piston 104 moves upwardly, carrying with it the pilot valve 51, to open the latter. Of course, fluid in the space 123 bleeds therefrom through the helical groove 125 and the escape vent 128 so long as the master control valve 60 is open.

When the pilot valve 51 opens, power oil under high pressure can then flow directly from the inlet port 28, the axial passage 52, the passage 116, past the pilot valve 51, through the annular space 117 and the radial port 118 into the space 119 below the main valve piston 108, to exert an upward force on the main valve piston tending to move it upwardly against the action of the main valve springs 111 and 112. Since the area of the bottom of the main valve piston 108 is substantially greater than the area of the main valve head 109, the fluid pressure in the space 119 required to overbalance the springs 111 and 112 is substantially lower than the pressure required in the inlet port 28 to accomplish the same effect. Although the space 119 is in constant communication with the discharge pipe 32 through the helical groove 120, the cross-section area of the groove is small enough to create a substantial fluid pressure drop therethrough and, thus, fluid pressure quickly builds up in the space 119. When the fluid pressure in the space 119 overbalances the action of the main valve springs 111 and 112, the main valve piston 108 rises, to open the main valve 50. Power oil can then flow directly from the inlet port 28 to the discharge pipe 15a, through the main flow passages 57, as described hereinabove, until the fluid pressure in the inlet port and, consequently, the upward fluid pressure on the valve head 78, is overbalanced by the force exerted by the spring 80, at which point the valve head 78 seats on its seat 77 to close the master control valve 60. Upon such closure of the master control valve 60, the fluid pressures on the upper and lower faces of the pilot valve piston 104 quickly equalize, to permit the spring 105 to close the pilot valve 51. As soon as the pilot valve 51 closes, the fluid pressures on the upper and lower faces of the main valve piston 108 similarly equalize, permitting the main valve springs 111 and 112 to close the main valve 50.

As will be apparent from the foregoing description, adjustment of the tension on the spring 80 of the master control valve 60 will permit the valve to be set to open at any desired predetermined maximum fluid pressure existing in the inlet port 28 below the value of the springs 111 and 112. This will permit ready and close regulation of the maximum fluid pressure in the manifold 20 and in the supply manifold 12, to suit well pumping conditions. As pointed out above, the thimble 84 and skirt 86 may be calibrated directly in pounds per square inch, to indicate the maximum pressure to which the master control valve 60 is set.

The pilot valve 51 and its actuating piston 104 may be regarded as a hydraulic relay means which receives a signal from the control valve 60 and which transforms this signal into an actuating force applied to the piston 108 of the main valve means to open, or close, the main valve 50.

Because the seat of the pilot valve 51 is in the head 110 of the main valve 50, the main valve tends, in opening, to close the pilot valve and so to exert a dampening influence on its own movement by reducing the force which causes it to

open. The effect of this influence is transmitted along the chain of equilibrium to the control valve 60. Because the pilot valve 51 is never quite as far open as it would be if it were not seated in the main valve head, the control valve 60 is required to open a little more in order to produce the extent of opening of the pilot valve which will result in an equilibrating extent of opening of the main valve. To produce the additional opening of the control valve requires somewhat more pressure at the control valve head, and this results in extending the range of pressure through which the valves must hunt to find an equilibrating level. With even a slightly extended hunting range, chattering of any of the valves as they seek equilibrium is practically eliminated. The degree to which the dampening influence extends the hunting range may be controlled by suitable selection of the cross-area and length of the helical groove 120. By increasing the resistance of this groove, the effect of a small opening of the pilot valve 51 upon the opening of the main valve 50 may be made greater, and the requisite opening of the control valve, and therefore its hunting range, may be reduced.

When the main valve 50 acts as a safety valve independently of the functioning of the control valve 60, it pushes upon the needle 102 of the pilot valve 51, keeping the pilot valve closed, and the main valve piston 108 has a dash-pot action, drawing fluid into the space 119 below the piston through the resistance of the helical groove 120. When the excess pressure has been relieved and the springs 111 and 112 move the main valve 50 again towards closed position, the fluid just drawn into the space 119 is again forced through the helical groove 120, and sudden movement or chattering of the main valve is prevented. Fluid is, of course, also drawn in and forced out through the helical groove 125 to the space 123 below the pilot valve piston 104, assuming that the control valve 60 is closed, increasing the dash-pot effect on opening movement of the main valve and not decreasing it on the closing movement so long as the closing movement of the pilot valve keeps pace with that of the main valve.

As has been seen, the resistance afforded by the helical groove 120 affects the hunting range of the valves. This is also true of the resistance afforded by the helical groove 125 on the pilot valve stem 100, because the responsiveness of the pilot valve piston 104 is directly affected by the pressure drop across the helical groove 125. The resistance of both of the helical grooves 120 and 125 are variable automatically in a manner tending to increase the hunting range of the valves as the volume of fluid passing from the supply manifold 12 to the production pipe 8 increases. When the pistons respective to these two grooves are raised to open the valves respective thereto, the upper portions of the helical grooves become uncovered and, of course, ineffective as resistances. This facilitates the escape of pressure from the high pressure sides of the respective piston, and therefore requires additional pressure to be supplied by the control valve 60 to achieve an equilibrating extent of opening of the main valve 50. The equilibrating level of pressure is therefore raised slightly and the hunting range of the valves is increased. The extent of the increase in the hunting range varies with the extent of opening of the main valve requisite to maintain an equilibrating level of pressure, or, in other words, is automatically variable with respect to the volume of fluid which must be re-

leased in order not to exceed the desired pressure level.

As suggested hereinabove, the pressure of the power oil in the supply manifold 12 is normally maintained slightly above the maximum pressure required by either of the well pumps in the wells 9 and 9a. Consequently, during normal operation the by-pass valve 15 will remain slightly open to relieve this pressure and by-pass power oil from the supply manifold 12 back to the settling tank 7. If the well 9a is shut out of the system, by closing the metering valve 14a, the volume of power oil normally required to operate the well pump in well 9a will obviously be backed up into the supply system in the supply manifold 12, and this will tend to increase the fluid pressure in the supply manifold which, in turn, will cause the by-pass valve 15 to open sufficiently to by-pass the increased unused volume of power oil to the settling tank 7 without appreciably varying the pressure of the power oil in the supply manifold. It will thus be appreciated that any or all of the well pumps in the system may be shut-off or reduced as to their power oil requirements, and all of the excess power oil remaining in the supply system is by-passed by the by-pass valve 15 to the settling tank. In addition, and more important, any such excess power oil is by-passed by the by-pass valve 15 without varying the fluid pressure in the supply manifold 12 more than 2%, regardless of the volume of power oil by-passed, from zero to the full capacity of the supply system.

Changes in the strength of springs, the areas of pistons and unbalanced valves, and the resistance to flow afforded by the various passages, will be readily apparent to one skilled in the art, and such changes do not depart from the spirit of my invention, nor do such alterations of arrangement as fall within the scope of the following claims.

I claim as my invention:

1. In a bypass valve connectible between a high pressure line and a low pressure line, the combination of: main passage means having an inlet connectible to the high pressure line and an outlet connectible to the low pressure line; pressure-actuable main valve means including a main valve in said main passage means and including main piston means connected to said main valve for regulating flow through said main passage means from said inlet to said outlet, said main valve being movable toward its open position by pressure applied to one side of said main piston means; pilot passage means communicating with said main passage means between said inlet and said main valve and communicating with said one side of said main piston means; pressure-actuable pilot valve means including a pilot valve in said pilot passage means and including pilot piston means connected to said pilot valve for regulating the pressure applied to said one side of said main piston means, said pilot valve being movable toward its open position by pressure applied to one side of said pilot piston means; control passage means communicating with said main passage means between said inlet and said main valve and communicating with said one side of said pilot piston means; and pressure-actuable control valve means including a control valve in said control passage means for regulating the pressure applied to said one side of said pilot piston means, said control valve being movable toward its open position by a predetermined

maximum pressure in said main passage means between said inlet and said main valve.

2. In a bypass valve connectible between a high pressure line and a low pressure line, the combination of: main passage means having an inlet connectible to the high pressure line and an outlet connectible to the low pressure line; pressure-actuable main valve means including a main valve in said main passage means and including main piston means connected to said main valve for regulating flow through said main passage means from said inlet to said outlet, said main valve being movable toward its open position by pressure applied to one side of said main piston means; pilot passage means communicating with said main passage means between said inlet and said main valve and communicating with said one side of said main piston means; pressure-actuable pilot valve means including a pilot valve in said pilot passage means and including pilot piston means connected to said pilot valve for regulating the pressure applied to said one side of said main piston means, said pilot valve being movable toward its open position by pressure applied to one side of said pilot piston means; and means spaced from said pilot valve means and communicating with said main passage means between said inlet and said main valve and communicating with one side of said pilot piston means and actuable by a predetermined maximum pressure in said main passage means between said inlet and said main valve for applying pressure to said one side of said pilot piston means.

3. A bypass valve according to claim 1 wherein said pilot passage means also communicates between said one side of said main piston means and said main passage means intermediate said main valve and said outlet, and wherein said control passage means also communicates between said one side of said pilot piston means and said main passage means intermediate said main valve and said outlet.

4. A bypass valve as defined in claim 3 wherein said pilot passage means is provided with a restricted portion between said one side of said main piston means and where it communicates with said main passage means intermediate said main valve and said outlet, and wherein said control passage means is provided with a restricted portion between said one side of said pilot piston means and where it communicates with said main passage means intermediate said main valve and said outlet.

5. A bypass valve according to claim 4 wherein the other side of said main piston means and the other side of said pilot piston means communicate with said main passage means intermediate said main valve and said outlet.

6. A bypass valve according to claim 1 wherein said control valve means includes resilient means engaging said control valve and biasing it toward its closed position, and includes means engaging said resilient means for adjusting the biasing force applied to said control valve by said resilient means, whereby to vary the predetermined maximum pressure in said main passage means between said inlet and said main valve at which said control valve opens.

7. A bypass valve according to claim 1 wherein said pilot passage means extends through said main valve means and wherein said pilot valve is engageable with a seat in said main valve means, said main and pilot valves being mounted for movement toward their respective open posi-

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tions in the same direction so that opening movement of said main valve tends to close said pilot valve.

8. A bypass valve according to claim 1 wherein said control valve includes a nozzle and a plane member engageable with the end of said nozzle.

9. A bypass valve as defined in claim 1 wherein said main valve is provided with an area exposed to pressure in said main passage means between said main valve and said inlet, said main valve being mounted for movement toward its open position by a predetermined maximum pressure acting on said area.

10. In a bypass valve connectible between a high pressure line and a low pressure line, the combination of: main passage means having an inlet connectible to the high pressure line and an outlet connectible to the low pressure line; pressure-actuable main valve means including a main valve in said main passage means for regulating flow through said main passage means from said inlet to said outlet; control passage means communicating with said main passage means between said inlet and said main valve; pressure-actuable control valve means including a control valve in said control passage means, said control valve being movable toward its open position by a predetermined maximum pressure in said main passage means between said inlet

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and said valve; and hydraulic relay means communicating between said control valve means and said main valve means and controlled by said control valve means for opening said main valve whenever said control valve is moved toward its open position by said predetermined maximum pressure in said main passage means between said inlet and said main valve, including a piston on said main valve and including pilot valve means controlled by said control valve means for applying pressure to said piston in a direction to open said main valve.

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