Jan. 19, 1954 F. C. BIGGERT, JR 2,666,292

CONTROL SYSTEM FOR HYDRAULIC PRESSES

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Fig. 1.

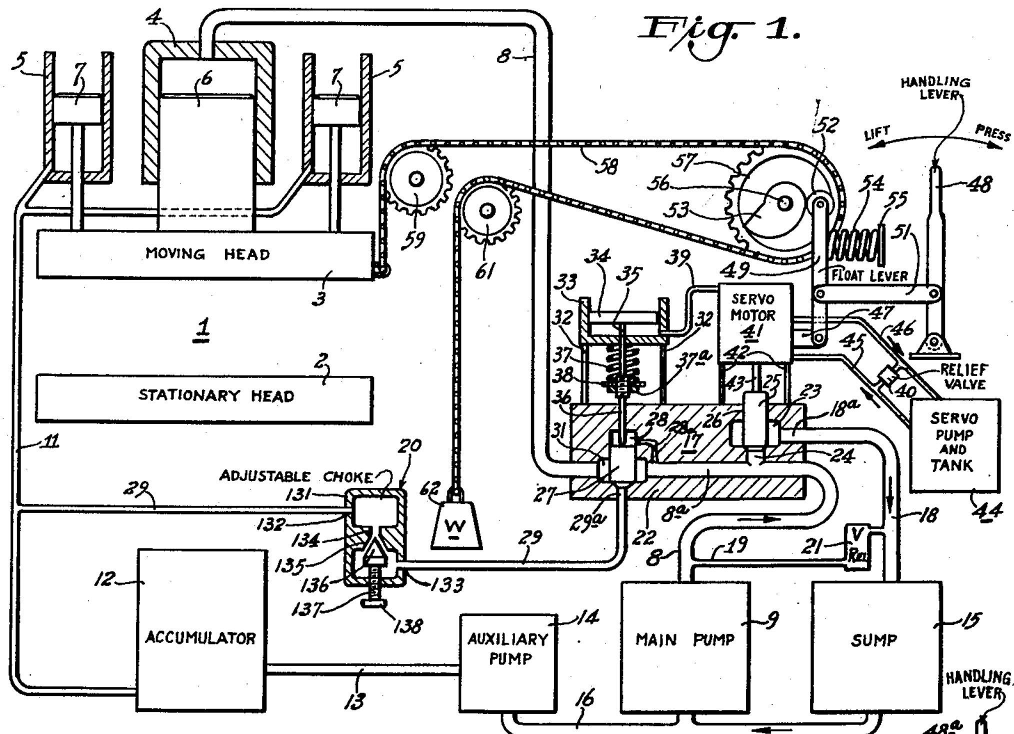


Fig. 7.

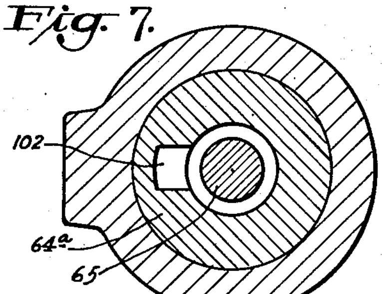


Fig. 9.

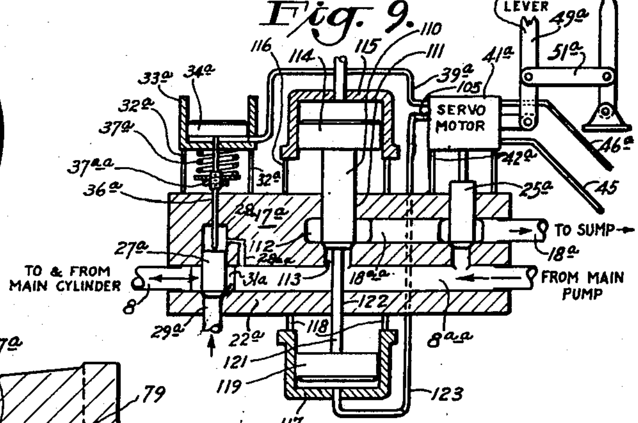
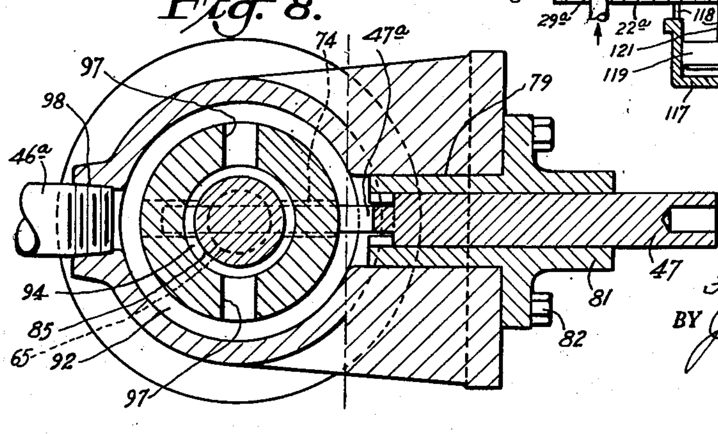


Fig. 8.



INVENTOR.
Florence C. Biggert Jr
BY J. E. Dickmann
atty

CONTROL SYSTEM FOR HYDRAULIC PRESSES

Filed March 18, 1948 2 Sheets-Sheet 2

Fig. 2 Fig. 3 Fig. 4

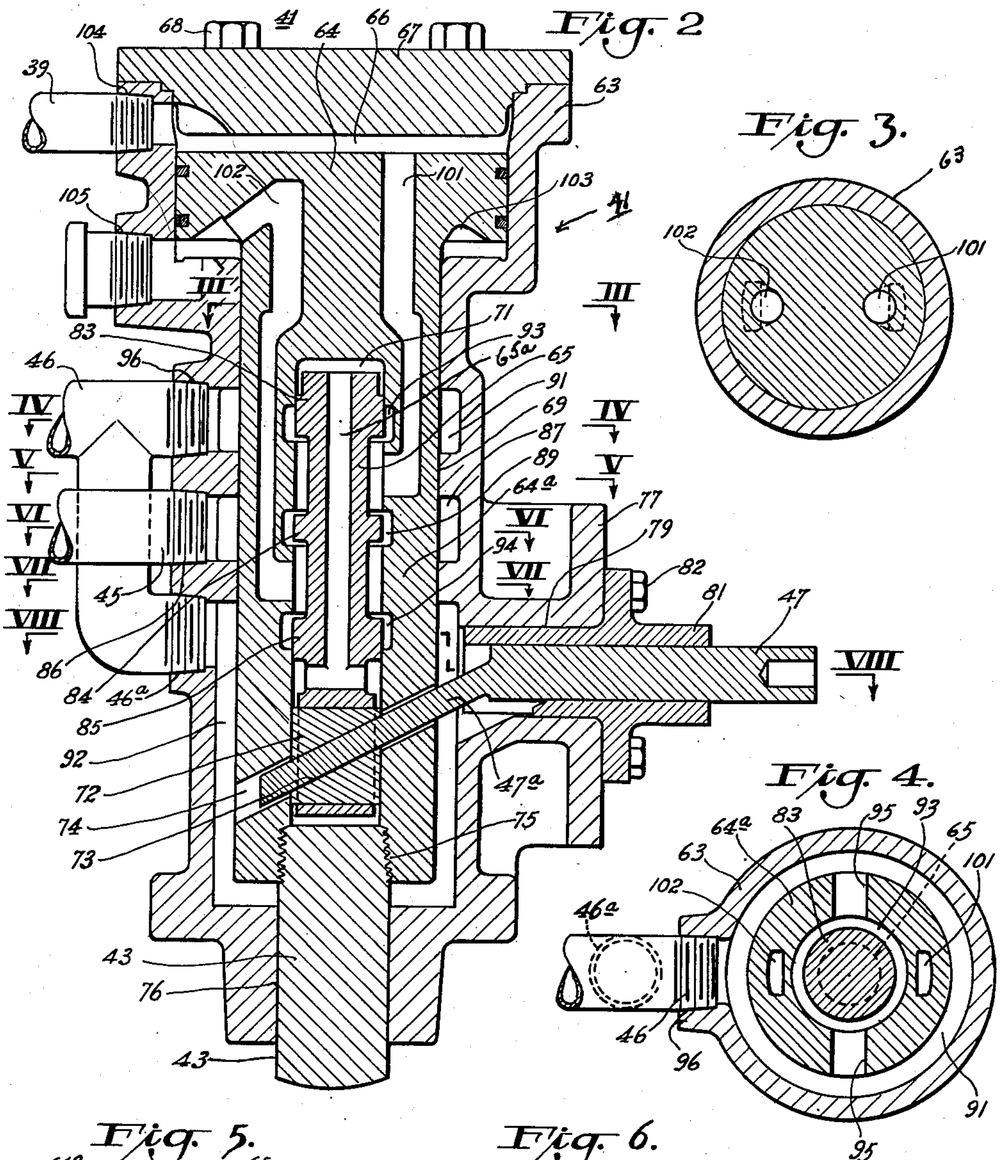


Fig. 5

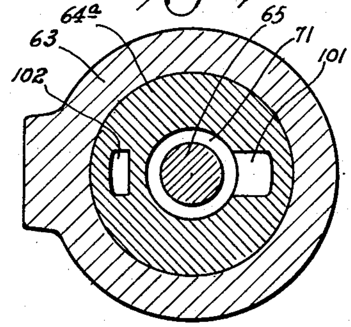
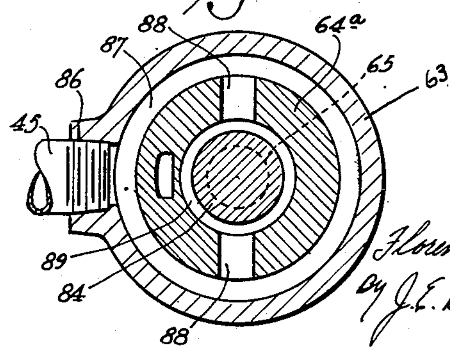


Fig. 6



INVENTOR.

Florence C. Biggert Jr

By J. E. Dickerson
atty

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CONTROL SYSTEM FOR HYDRAULIC PRESSES

Florence C. Biggert, Jr., Pittsburgh, Pa., assignor to United Engineering & Foundry Company, Pittsburgh, Pa., a corporation of Pennsylvania

Application March 18, 1948, Serial No. 15,613

13 Claims. (Cl. 60—51)

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This invention relates to an improvement in hydraulic controls, and while not necessarily so limited, pertains more particularly to a control system for forging presses and especially hydraulic presses of a type commonly referred to as "direct pumping presses." Presses of this type are presses in which the hydraulic pressure for effecting their pressing operation is supplied directly from power-driven pumps, the pressure delivered to the press by the pumps being determined, up to the extent of the pump capacity, by the resistance encountered by the movable press head.

One object of the invention is to provide a control which is capable of increasing the speed of operation of this type of press and which is of considerable advantage in heavy duty presses.

Another object of the invention is to improve the means available to the operator of presses of this character for holding a work piece at a selectively controlled pressure between the press heads, which is highly desirable for various reasons and especially when it is necessary to shift the handling equipment associated with the press.

A further object is to provide for easily and dependably immobilizing the moving press head at any selected point in its stroke.

It is also an object to provide a control of this character which lends itself to the utilization of improved features of design in the press itself.

These and various other objects, as well as numerous other features and advantages of the invention, will be readily apparent to those skilled in the art when the following description of the embodiments of the inventions therein illustrated is read in conjunction with the accompanying drawings.

Figure 1 is a schematic view partly in section of one embodiment of the invention.

Figure 2 is a sectional view of the servo-motor construction shown in Figure 1.

Figures 3, 4, 5, 6, 7 and 8 are horizontal sectional views taken on the lines III—III, IV—IV, V—V, VI—VI, VII—VII and VIII—VIII respectively of Figure 2 and Figure 9 is a sectional view of a modification of the valve assembly shown in Figure 1.

Referring to Figure 1 of the drawings, reference character 1 designates generally an hydraulic press of the direct pumping type which comprises a frame not shown, a stationary head 2, a moving head 3, a main cylinder 4, and balance cylinders 5. A plunger or piston 6 operating in the main cylinder 4 is connected to the center of the moving head to provide the working pressure therefor, and pistons 7 in the balance cylinders 5 are connected to opposite ends of the moving head 3 for retrieving the moving head.

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For actuating the plunger 6 of the main cylinder 4 it is connected by a pipe 8 to a main pump 9 which is driven by a motor not shown. In a similar fashion the balance cylinders 5 are connected by a pipe 11 to an accumulator 12 which is in turn connected by a pipe 13 to an auxiliary pump 14 also equipped with a driving motor not shown. The auxiliary pump 14 supplies the desired pressurized fluid to the accumulator 12 so that from the accumulator, fluid under constant pressure will be supplied to the pipe 11 and thence to the two pull-back cylinders 5. By a proper regulation of the controls of the pump 14 desired pressure may be maintained within the accumulator 12 so that the combination of the two serve as a constant pressure source.

While other means may be employed for such purpose, in the system shown, a single sump 15 is provided for supplying and storing the hydraulic fluid used in the system. This sump, as indicated, is connected by a header 16 to both the main pump 9 and the auxiliary pump 14, and may be either elevated or provided with a means not shown to establish in it a desired static pressure.

A valve assembly 17 is interposed in the supply line 8 between the main pump 9 and the main cylinder 4 to control the flow of water from the main pump to the main cylinder, and back to the sump 15. A pipe 18 is connected between the valve assembly 17 and the sump 15 to control, in a manner which will be presently explained, the flow of water to and from the valve assembly and the sump.

To prevent over-loading of the main pump 9, which is preferably of a constant discharge type, a by-pass 19 having in it a relief valve 21 is connected between the main line 8 and the return line 18 to the sump 15. While this relief valve which determines the maximum output pressure of the main pump may take various forms, the construction disclosed in the present inventor's United States patent application Serial No. 765,125, filed July 31, 1947, and issued as Patent No. 2,571,007 on October 9, 1951, is recommended for this purpose.

Referring in detail to the valve assembly 17, a main housing 22 is provided in which there is a suitably formed passage or duct 8a comprising that part of the main line 8 extending through the valve assembly. Also formed in the housing 22 is a duct 18a which is connected to the sump return line 18 and which terminates in the housing 22 in an enlarged opening 23. This latter opening 23 is connected by a duct 24 with the main line duct 8a to provide a flow connection between the main cylinder 4, the main pump 9, and the sump 15. In order to regulate the flow between main line duct 8a and the return flow duct 18a a shut-off and throttling valve 25 is positioned in a bore 26 communicating with the enlarged opening 23 at the inner end of the return flow duct 18a, and is so arranged as to seat on the upper end of the duct 24 which connects the main line duct 8a with the return flow duct 18a.

For controlling the flow between the balance cylinders and the main cylinder through the main line duct 8a of the valve assembly 17, a valve 27 is arranged to operate in a bore 28 which intersects the main line duct 8a and to seat when closed on the upper end of a smaller duct 29a. This latter duct 29a has connected to it a pipe 29 which is in turn connected to the pipe 11 connecting the balance cylinders 5 of the press with the accumulator 12 which supplies their operating pressure. At the point where the bore 28 for the main line valve 27 intersects the main line duct 8a the main line duct is enlarged as indicated at 31 to permit a flow through the main line duct around the main line valve 27.

Mounted on supports 32 on the valve housing 22 in line with the bore 28 for the main line valve 27 there is arranged a cylinder 33 for controlling the movement of the main line valve 27. The piston 34 of this cylinder is equipped with a piston rod 35 which is fitted in a bore 36 communicating with the bore 28 for the main line valve 27, and has its free end disposed to rest on the top of the main line valve 27. Underneath this cylinder 33 a coil spring 37 is fitted on the piston rod 35 and is adapted to normally bias the main line valve 27 to its closed position. To perform this function the upper end of this spring 37 is rested against the bottom of the cylinder 33 and its lower end against a fitting 38 secured to the piston rod 35. This spring is provided with sufficient tension to hold the valve 27 closed when the throttling or return-flow valve 25 is open and there is little or not any pressure in 8a. To permit adjusting the tension in the spring 37, the fitting is arranged to rest on an adjustable nut 37a threadably secured to the rod 35, and between the top of the bore 28 and the duct 8a a bleed duct 28a is provided.

The object of the cylinder 33 more specifically is to provide, as may be needed in the functioning of the system, for relieving the pressure imposed on the main line valve 27 by the rod 35 through the influence of the spring 37. To accomplish this, a liquid supply line 39 is connected to this cylinder 33 and to a suitably controlled pressure supply which in this instance, although it could be otherwise arranged, is furnished by the servo-motor 41.

In light presses the return-flow throttling valve 25 may be operated directly from an operating lever, or by other means not shown. For heavy duty presses, however, a servo-motor such as the servo-motor 41 or the like is desirable for this purpose.

The primary duty of this servo-motor 41 is to control in the main cylinder exhaust circuit the operation of the return-flow and throttling valve 25 contained therein and located in the sump return line duct 18a. To so function it is mounted on supports 42 secured to the valve housing 22 with its piston connected by a connecting rod 43 to the throttling valve 25. The operating pressure for this servo-motor is supplied by a pump 44 actuated by a pump motor not shown and connected thereto by inlet and exhaust lines 45 and 46 respectively. For controlling said servo-motor a control element 47 is connected to a pivoted operating lever 48 by means of a floating lever connection consisting of a lever 49 and a link 51. The lower end of the lever 49 is pivotally connected to the outer end of the servo-motor control element 47, and the ends of the link 51 are pivotally connected to the lever 49 and to the control lever 48.

The free end of the lever 49 is equipped with a roller 52 that is arranged to bear on a cam 53 which is actuated by the movement of the movable press head 3, the roller 52 being held in contact with the cam by a coil spring 54 mounted between it and a support 55. The curvature of cam 53 is such that in accordance with the movement of the press head the roller 52 in contact with the cam moves a proportional amount and the diameters of the sprockets 57 and 59 are so related that the cam roller will be moved from its minimum to its maximum position when the press head moves from its top to its lowermost position of its stroke.

As shown, although the equipment for this purpose may take other forms, the cam 53 is secured to a shaft 56 or to a sprocket wheel 57 mounted on the shaft 56. Around this sprocket wheel 57 there is mounted a chain 58 which is extended in one direction over an idler 59 and connected to the moving press head 3 and in the other direction over another idler 61 and connected to a counter-weight 62. With this arrangement when the press head is moved downward, the cam 53 and the sprocket 57 are caused to move in a counter-clockwise direction by the press head 3, and when the press head is raised, they are caused to move in a clockwise direction by the counterweight 62. Thus irrespective of the position of the servo-motor control lever 48, the position of the servo-motor control element 47 is influenced by the position of the moving press head. For reasons which will presently be made more apparent, the cam 53 is generated on a curve which imparts to the roller 52 on the lever 49 as the cam is rotated a displacement relative to the cam center which is substantially directly proportionate to the distance which the movable press head 3 moves relative to the fixed press head 2 when the former is actuated.

While other designs of servo-motors may be used, the servo-motor 41 is a constant-flow type of motor similar in general principles to that disclosed in the present inventor's Patent No. 2,370,137, but widely varying in construction. As shown in Figures 2 to 8, this motor comprises principally a combined cylinder and housing 63, a piston 64, an internal multiple valve 65, and a valve-controlling element 47. At its upper end the housing 63 has an enlarged main cylinder bore 66 and is provided with a detachable head 67 held in place by bolts 68. In this main cylinder bore 66 the main piston 64 of the motor is arranged for reciprocal movement. Depending from the main piston is an elongate reduced portion 64a which moves in a reduced bore 69. This elongate portion 64a of the piston 64 is provided with a centrally extending axial bore 71 in which the multiple valve 65 is arranged for movement axially of and relative to the piston portion 64a.

In the lower end of the multiple valve 65 there is arranged a fitting 72 which is provided with a diagonal bore 73. In axial alignment within this bore 73 a somewhat larger bore 74 is extended through the piston extension 64a. The purpose of the bore 73 is to receive an angularly displaced extension 47a secured to the inner end of the valve-actuating element 47. Consequently as the valve-actuating element 47 is displaced axially by the actuating lever 48 or the cam 53, the multiple valve 65 is correspondingly moved with respect to the piston extension 64a, such relative movement being permitted due to the bores 74 in the extension 64a being larger than the bore 73.

To permit the assembly of the multiple valve 65 in the piston extension 64a, the bore 71 is extended through the bottom of the piston extension 64a. The lower end of this latter bore 71 is provided with internal threads 75 for receiving the externally threaded end of the rod 43 which passes through a bore 76 in the lower end of the housing 63 and is connected to the throttling valve 25 in the return-flow duct 18a of the valve assembly housing 22.

For housing and supporting the valve-actuating element 47, a laterally extending projection 77 is formed on the housing 63. This projection 77 has in it an axially extending bore 79 which communicates with the vertical bore 69 in the housing 63, and in it there is inserted a centrally bored fitting 81 in which the valve-actuating element is mounted, being secured to the outer end of the projection 77 by bolts 82.

The multiple valve 65 is equipped with three axially spaced valve sections 83, 84, and 85, which have a diameter permitting them to fit snugly in the internal bore 71 of the piston extension 64a, the balance of the valve element 65 having a lesser diameter to allow for a flow of the motor-actuating fluid around the valve element 65 between and below the valve sections. A central bore 65a is provided in the body of the multiple valve to equalize pressure on the ends of the valve.

The centrally disposed valve section 84 is employed to control the flow of the actuating fluid from the servo-pump 44 into the servomotor. To accomplish this the inlet pipe 45 from the servo-pump is connected to a port 86 which communicates with an annular groove 87 formed in the inner wall of the housing 63 opposite the valve section 84. This groove 87 is in turn connected by radial ducts 88 in the piston extension 64a with an annular groove 89 formed in the bore 71 opposite the valve section 84. To function as required the axial length of the valve section 84 is made less than the axial width of the groove 89. Thus when the valve section is centered with respect to the groove 89, as it is when the servomotor is in its neutral position, the inlet fluid is permitted to flow both upwardly and downwardly of the body of the valve element 65 in the bore 71.

The valve sections 83 and 85 are employed to control the flow of the actuating fluid from the servomotor back to the servo-pump 44 by way of the return-flow pipe 46. Toward this end an annular groove 91 is formed in the inner wall of the housing 63 opposite the valve section 83, and an enlargement 92 is provided in the housing 63 which extends from a point above the valve section 85 to the bottom of the bore 69. Also adjacent valve sections 83 and 85 respectively, internal annular grooves 93 and 94 are provided in the bore 71. These grooves 93 and 94 and the valve sections 83 and 85 are so proportioned and arranged that when the valve section 84 is centered, an opening exists between the lower edge of groove 93 and the bore 71, and also an opening between the upper edge of groove 94 and the bore 71. This relationship, however, is such that when valve section 84 is moved to close off the lower edge of the adjacent groove 89 from bore 71, valve 83 will close off its adjacent groove 93 from bore 71 and valve 85 will increase the opening between its adjacent groove 94 and bore 71. Conversely when valve section 84 is moved upwardly to shut off the upper edge of groove 89 from the bore 71, valve section 85 will close off groove 94 from bore 71 and increase the opening between groove 93 and bore 71.

Between the internal grooves 93 in the bore 71 and 91 in the bore 69 adjacent valve section 83 there are formed radially disposed ducts 95 which permit a flow of fluid from the groove 93 into the groove 91. This latter groove 91 in turn communicates with a port 96 to which the exhaust or return line 46 from the servomotor to the servo-pump is connected. Similarly, between the enlargement 92 in the bore 69 and the groove 94 adjacent the valve section 85 there are formed radially disposed ducts 97 which permit a flow of fluid from the groove 94 into the enlargement 92 of the bore 69, which in turn communicates with a port 98 that is connected by a pipe 46a with the servo-pump return-flow pipe 46.

For allowing the actuating fluid to flow from the servo-pump into the cylinder bore 66 at the top of the piston 64 of the servomotor, a duct 101 is formed in the piston 64 with its upper end terminating at the top of the piston and its lower end communicating with the internal bore 71 substantially midway between the annular grooves 93 and 89 opposite the valve sections 83 and 84 respectively. Similarly, for allowing the actuating fluid to flow from the servo-pump to the cylinder bore 66 at the bottom of the piston 64, a duct 102 is formed in the piston 64 with its upper end terminating in a circular groove 103 formed in the lower face of the piston 64, and its lower end communicating with the internal bore 71 substantially midway between the grooves 89 and 94 in the bore 71 adjacent the valve sections 84 and 85 respectively.

As a further feature of this construction ports 104 and 105 are provided in the cylinder wall 63 adjacent the upper and lower ends respectively of the cylinder bore 66 so that the fluid pressures existing at these points can be utilized, in addition to actuating the piston 64, for other control purposes as will be presently explained. As illustrated in Figure 1, port 104 is connected by pipe 39 with the cylinder 33 utilized to counteract the influence of the coil spring 37 which normally urges the main line valve 27 toward its closed position. In this particular construction port 105 is plugged as it has no useful purpose but in the modified form of valve assembly this port is employed and serves a definite purpose.

According to the construction shown in Figures 2 to 8 inclusive, when the multiple valve 65 is in its central or neutral position, that is, when the inlet valve section 84 is centered with respect to the adjacent groove 89, the actuating fluid from the servo-pump flows through pipe 45 into the annular groove 87 in bore 69 by way of port 86, thence through radial ducts 88 into groove 89. From the groove 89 part of the fluid flows over the top of the valve 84 into bore 71 from whence part thereof passes through duct 101 into the top of the cylinder bore 66, and the balance around valve 83 into groove 93, and out radial ducts 95 to groove 91, and by way of port 96 into return line 46 back to the servo-pump. In a like fashion part of the inlet fluid passes around valve 84 into bore 71 below the valve section 84. This portion of the fluid flows in part through duct 102 to the lower end of cylinder bore 66, whereby subjecting piston 64 to substantially equal forces on its opposite sides, varying, of course, in accordance with the difference in their effective areas; the balance of the fluid passing out the bottom of groove 89 flows through bore 71 around valve section 85 into groove 94 from whence by way of radial ducts 97 is passes into the enlargement 92 of bore 69 and out port 98 to pipe 46*a* and into the return-flow pipe 46.

To raise the connecting rod 43 the control element 47 is moved inwardly and to force it downwardly the control element 47 is moved outward. By these different movements of the control element 47 the multiple valve 65 is moved up or down as the case may be with respect to the piston extensions 64*a*. Hence the valve sections 83, 84, and 85 are correspondingly moved with respect to the annular grooves in bore 71 adjacent to them.

If the control element 47 is moved inwardly and thus the multiple valve 65 is moved upwardly, valve section 84 will tend to cause the inlet pressure to flow through duct 102 to the bottom of cylinder bore 66 beneath piston 64, and when moved sufficienly will completely shut off the inlet flow to duct 101. In the same way valve 85 will tend to prevent the inlet pressure escaping through port 98 until moved sufficiently when it will stop it completely, while valve 83 will facilitate the escape of pressure from the top of piston 64 through duct 101 out through port 96 to the return-flow pipe 46. As a result pressure will build up on the lower face of piston 64 and cause the latter to rise, raising the connecting rod 43 with it. As the piston raises, however, the valves 83, 84, and 85 will be returned toward their neutral positions with respect to the adjacent grooves 93, 89, and 94 respectively, and as soon as this is accomplished no further movement of piston 64 will take place. When the control element 47 is moved outward, exactly the same operation of the piston occurs, but in the opposite direction and, of course, with attendant reverse flow into and out of the main cylinder bore 66 and the inlet and exhaust passages.

In such a servo-motor the fluid pressure acting on the piston thereof will vary according to the resistance to movement which the piston encounters. When closing throttling valve 25, very little pressure will develop above the piston 64 until such valve is completely closed, but immediately upon its being closed, this pressure will rise to the pressure which the servo-motor pump 44 will develop. To prevent overloading of this pump a relief valve 40 is connected between the supply and return lines 45 and 46 respectively connected to this pump, and the setting of which will determine the maximum pressure the pump will deliver to the servo-motor. While with such an arrangement a somewhat wasteful pump action is encountered

when the servo-motor is in stalled condition, this is not particularly objectionable for ordinary press operation because the duration of stalled periods is short. The constant flow pump is recommended because it is simple and cheap. When operations requiring long periods during which the servo-motor is stalled are involved, a radial piston or equivalent variable flow pump with zero discharge at maximum pressure may be used.

In accordance with the construction illustrated in Figure 1, the pressure which develops in the top of the servo-motor cylinder 66 is communicated by the pipe 39 to the cylinder 33 employed to influence the operation of the main line valve 27. As previously explained, the rod 35 bearing against the top of this valve is not connected to it. It is arranged to rest only on the top of this valve. Under normal conditions when there is no pressure in the cylinder 33, the main line valve 27 is urged to its closed position by the spring 37, and this spring is provided with a sufficient tension to hold the valve 27 closed against the pressure in the line 29 connected to the accumulator 12 which supplies pressure for the balance cylinders 5.

When the throttling valve 25 in the sump return line 18 is fully open, due to lack of pressure in the top of the servo-motor cylinder 66, there will also be a lack of pressure in cylinder 33. On the other hand, when the throttling valve 25 is fully or practically closed, and under which circumstances a pressure must be established in the top of the servo-motor cylinder 66 which is sufficient to overcome the pressure in main pump line 8, a similar pressure will be present in the control cylinder 33 tending to overcome the pressure of the spring 37. If the throttling valve 25 is fully closed, the pressure in the control cylinder 33 will be sufficient to cause the piston 34 which is attached to the rod 35 to compress the spring 37 and thus render the main valve 27 free to act as a simple check valve, but it will only act as such under these conditions. It is to be noted that only part of the pressure in spring 37 need be relieved in order that the main valve 27 may open, since the pressure in line 29 strongly urges this valve toward its open position. This is important to the operator's control of the main valve because it makes it possible through the use of the servo-motor which controls the throttling valve 25 for the operator to exercise control over the opening of the main line interconnecting valve 27.

When the throttle valve 25 is partially closed, only sufficient pressure is required at the top of piston 64 to offset the force produced on the bottom of the valve 25 by the pressure in the main pump line 8 and which will vary with the extent the valve is open.

To lift the movable press head 3, the pressure operated throttling valve 25 is opened and the interconnecting pressure operated valve 27 closed. This allows the water in the main press cylinder 4 and main pump 9 to exhaust through the throttling valve 25 into the sump 15, and the pressure in the balance cylinders 5 effected by the accumulator to move the movable press head upwardly to the top of its stroke.

To move the movable press head 3 down at a relatively slow speed, through proper actuation of the servo-motor 41 the main line pressure operated valve 27 is completely closed and the pressure operated throttling valve 25 is only partially closed. This causes substantially all, but not all of the water delivered by the main pump 9 to flow into the main press cylinder 4 and develop a pressure therein which is sufficient to overcome the pressure in the balance cylinders 5 and move the press head downwardly.

When a rapid lowering motion is desired, through proper actuation of the servo-motor 41 the throttling valve 25 is closed entirely and the interconnecting valve 27 is opened. This allows the balance cylinders to exhaust into the main press cylinder along with all of the water from the main pump, with the result that the motion of the movable head is speeded up because of the larger area of the main press cylinder. Since downward movement of the press head is opposed by the fluid in the balance cylinders, obviously the rate at which the fluid escapes therefrom will be one of the factors determining the speed at which the head descends.

When the movable press head engages the work on the fixed press head, the interconnecting valve 27 acting as a simple check valve is closed by dropping due to its own weight into its closed position inasmuch as the high fluid pressure which normally exists in the line 29 during downward movement of the press is now relieved by reason of interruption of rapid movement of the press head as it engages with a workpiece so that the movable press head will thereafter move down at normal pressing speed.

To immobilize the movable press head, the throttling valve 25 in the sump line 18a is so adjusted, in accordance with the operation thereof hereinafter set forth, that a pressure will be established in the main press cylinder 4 which is just sufficient to counteract the pressure in the balance cylinders. Likewise when it is desired to hold the work between the press heads with a light pressure, as is desirable when shifting the handling equipment associated with the press, the throttling valve 25 is adjusted, in accordance with the operation thereof hereinafter set forth, to maintain in the main press cylinder whatever pressure is required for this purpose.

In accordance with this invention when it is desired to raise the movable press head 3, the control or handling lever 48 is moved to the left, as shown in the drawings, the distance necessary to adjust the throttling valve 25 to its open position and allow the water in the main press cylinder to flow into the sump 15. This movement of the handling lever 48 lifts the valve 65 of the servo-motor upward and produces as previously explained a resulting upward movement of the piston 64 and the throttling valve 25 which is connected to it by the connecting rod 43. With this operation of the servo-motor, the pressure in the servo-motor cylinder 66 is substantially zero and consequently the pressure within the cylinder 33, with which it communicates, is also substantially zero so that there is no force tending to compress the spring 37. Consequently the main line valve is held closed by the spring 37.

When the press head starts its upward travel, the counter-weighted chain 58 passing around the sprocket 57 carrying the cam 53, against which the end of the control lever 49 bears, starts to rotate in a clockwise direction. As this takes place, the spring 54 bearing against the lever 49 moves the upper end of lever 49 to the left as permitted by the cam 53 which results in lever 49 pivoting about the point where the link 51 is connected to it. This tends to move the control rod 47 of the servo-motor in a direction to cause the servo-motor piston 64 to close the throttling valve 25. To overcome this the operator must continue to move the control lever 48 to the left sufficiently to keep the throttling valve in the required position to cause the press head to move upward. By the time the press head 3 reaches the top of its stroke, the operator will have moved the control lever 48 to its farthest left position.

To move the movable press head 3 downward at a reduced speed, the operator moves the control lever 48 to the right sufficiently to cause the servo-motor piston to move the throttling valve into a position with respect to its seat such that enough pressure will be supplied by the main pump 9 to move the press head downwardly against the pressure of the balance cylinders 5. In this operation of the servo-motor the high pressure in the servo-motor cylinder 66 is at the top of the piston 64, and although the pressure may balance or even be sufficient to draw the piston 34 upwards and compress the spring 37, nevertheless upward movement of the valve 27 is prevented due to the high pressure imposed upon the upper portion of the valve through the duct 28a connected to the main line duct 8a carrying the full pumping pressure, which is sufficient to oppose that resulting from the balance cylinders themselves. Consequently in this downward movement the press head is resisted by the full effect of the balance cylinder pressure, and the down speed is determined by the pressure delivered to the main cylinder 4 as controlled by the throttling valve 25.

As the press head 3 starts down the cam 53 actuated by the chain 58 attached to the press head is rotated in a counterclockwise direction. This results in the cam 53 tending to readjust the valve element 65 to a position which, without adjustment of the hand lever 48, would cause the press head to be stopped. To avoid this the operator must continue to move the control lever 48 sufficiently to overcome the compensating effect of the cam 53 to keep the press head moving.

When it is desired to have the press head move downward at a rapid speed, the handling lever 48 is moved to the right sufficiently to completely close the throttling valve 25, and thereafter this movement of the handling lever is continued sufficiently to compensate for the offsetting action of the cam 53 so long as such downward movement is desired. Under these conditions the full effective high pressure in the servo-motor is at the top of the piston 64 and this pressure, which is communicated to the control cylinder 33 for the main press valve 27, is sufficiently high to cause the piston 34 in that cylinder to so compress the spring 37 which urges that valve closed, and due to the difference in the area between the main press cylinder and the two balance cylinders, the main press cylinder being the greater, the main press plunger will move downward and, as the intensity of the pressure from the balance cylinders imposed upon the lower portion of the valve 27 is so much greater than that which is imposed by the pump on the top portion of the valve 27 tending to keep it closed, the valve 27 consequently will open and permit the fluid being expelled from the balance cylinders to enter the line leading to the top of the main press cylinder. When this occurs the discharge from the balance cylinders is fed into the main press cylinder by way of the main line 8, and due to the area of the main piston 6 greatly exceeding that of the balance pistons 7, the downward movement of the press head is thereby speeded up. When the press head meets a resistance, as by striking a work piece on the stationary head, the discharge from the balance cylinder ceases and valve 27, acting as a check valve, falls closed by its own weight. Thereafter as the pressing action takes place, the main pump 9 delivers its entire output into the main press cylinder 4 which causes its pressure to build up rapidly. This pressing action is the same, as will be apparent, when the press head is lowered into engagement with the work at a reduced speed, and in each instance to continue the press action the control lever 48 must be moved sufficiently to always overcome the compensating action of the cam 53.

When it is desired to immobilize the press head at any selected position between the two extremities of its stroke, the control lever 48 is held at or moved to a selected position coinciding with the position at which it is wished to stop the head. Thereafter, depending on the direction of movement of the head, as the head moves toward such position, the cam 53 will so adjust the valve element 65 as to set up the proper balance in the servo-motor 41 to maintain the throttle valve 25 in the position required to hold the press head in such position. Should the press head tend to pass such position, the cam 53 readjusts the valve element 65 to make the necessary correction.

Referring to Figure 9, a modified form of valve assembly 17*a* is shown which has for its principal objects to provide for making it possible to raise the press head at a faster speed, to reduce the forces required of the servo-motor, and to afford a more sensitive throttling control. In this figure for purposes of description the parts of the assembly which are similar to the parts of the assembly 17 illustrated in Figure 1 are identified by the same reference characters, but have the suffix "*a*" added to them.

The principal difference between the assembly 17*a* and the assembly 17 is that an extra and larger throttling valve 110 is provided in the sump return line 18. As shown, this additional valve 110 is arranged in a bore 111 which communicates with a chamber 112 at the end of an extension 18*aa* of the bore 18*a*, and rests on the upper end of bore 113 which connects the chamber 112 with the main line duct 8*aa*.

To normally hold the additional throttling valve 110 on its seat, it is provided at its upper end with a piston 114 which is located in a cylinder 115 mounted on the housing 22*a* by supports 116. To supply the piston 114 with the necessary force to perform its functions, the cylinder 114 is connected to any suitable fluid pressure supply such as an auxiliary accumulator such as, for example, the accumulator 12 illustrated in Figure 1. As will be obvious, since this valve functions quite similarly to the main line valve 27*a*, a spring arrangement such as used in the main line valve and designated by numerals 37*a* and 37*aa* may be used instead of the piston and cylinder arrangement for normally urging the valve 110 toward its closed position.

For opening the valve 110 when it is desired to impart rapid lifting movement to the press head, a cylinder 117 is secured to the housing 22*a* by supports 118. In this cylinder a piston 119 is located which is connected to the lower end of the valve 110 by a rod 121 which extends through a bore 122 in the housing in alignment with the axis of the valve 110. This latter cylinder 117 is connected by a pipe 123 to the port 105 communicating with the lower end of the cylinder bore 66 in the servo-motor. With this arrangement whatever pressure is supplied to the lower side of the servo-motor piston 66 is communicated to piston 119 which is employed to open the extra throttling valve 110 the pressure above piston 114 tending to keep it closed.

To function as intended the pistons 114 and 119 connected to the additional throttling valve 110 are so proportioned, and the pressure applied to the piston 114 is so determined that the valve 110 will be opened only when the piston 119 is exposed to that pressure which is created in the servo-motor underneath the piston 64 when the servo-motor is actuated to move and hold the throttling valve 25*a* in its fully open position. This set of conditions exists when the control lever 48 is moved to open the first throttle valve 25*a*, and such motion of the control lever is continued in the same direction at a speed which exceeds the compensating action of the cam 53. With this arrangement the additional throttling valve 110 is opened only when the pressure is sufficient to maintain the first throttling valve 25*a* fully opened and causes the press head 3 to rise at a higher speed because the resistance to the flow of water from the main press cylinder into the sump 15 is thereby reduced.

With this valve control assembly, slow lifting of the press head is obtained by opening only the first throttling valve 25*a*. By using two such valves, the one actuated directly by the servo-motor 41*a* may be made smaller than is generally necessary where only one such valve is used. Also the force needed in the servo-motor 41*a* for operating this smaller valve is correspondingly smaller. This makes for more sensitive control of the press.

Another feature of the invention which is particularly desirable for use in presses having heavy moving parts is the provision of a throttling valve 20 in the line 29 which connects the pressure supply line 11 feeding the balance cylinders with the main press line 8 through the main press line valve 27. This throttling valve 20 has for one of its objectives to provide for obtaining approximately equal speeds in the upward and downward strokes of the movable press head when idling. Another objective is to provide a suitably high balance pressure for the balance cylinders so that these cylinders may be of reasonable size.

An analysis of the conditions governing these desiderata shows that, given a system in which the moving weights and pressure losses in pipes and valves are established, there is only one size of balance cylinder and one pressure which gives the required equality of lifting and lowering press head speeds, and frequently the cylinders are excessively large and the pressure inconveniently low. The analysis also shows that by restricting the flow between the balance cylinders and the main cylinder, the balance cylinder size and balance pressures may be adjusted to convenient design dimensions; i. e., by adjusting the degree of throttling the desired equality of speeds may be attained for any combination of cylinder size and balance pressure; in other words, small balancing cylinders using relatively high balancing pressure may be arbitrarily designed to give the desired lifting speed and lowering speed of the press head by suitably throttling the flow between balance cylinders and main cylinder.

To those skilled in the art it will be obvious that if check valve 27 and port 29*a* are of suitable size, these parts may provide the required throttling, and in some cases it is preferable to use this construction. There are other cases in which the adjustable choke 20 is preferable. This improvement is principally for use in presses of the type in which rapid lowering of the press head is accomplished by interconnecting the balance cylinders and main cylinder, which consists in choking or throttling the flow through this connection to allow freedom of design as regards size of balance cylinders and hydraulic pressure in the balance system while maintaining suitable speeds of raising and lowering movements.

Although other means for such purpose may be utilized, the valve **20**, as shown, comprises a housing **131** provided adjacent its opposite ends with an inlet port **132** and outlet port **133** and a centrally disposed partition **134** in which a bore **135** is formed for the reception of an adjustable valve element **136**. This valve element **136** is mounted on a stem **137** which is threadably fitted in the base of the housing and equipped with a hand wheel **138** for adjusting the valve element.

It is to be appreciated that the construction herein disclosed may be widely modified for purposes of fabrication or for other reasons without departing from the spirit of this invention. In this connection, for example, in order to afford the operator a better feel of the assistance provided by the servo-motor, in place of having the valve sections **83** and **85** operated with a sliding fit in the bore **71** of the piston extension **64a**, they may be designed to effect a seat in the bore **71**. This, of course, would involve providing a bore **71** with different inside diameters and the extension **64a**, as well as the valve element **65**, being made in section in order to effect the assembly. This modification is not illustrated as it is deemed to require, in the light of this disclosure and the teaching of Patent No. 2,370,137, nothing more than ordinary mechanical skill.

According to the provisions of the patent statutes, the construction and operation of the invention has been illustrated and described in connection with a recommended embodiment. It is desired that it be understood, however, that within the scope of the appended claims the invention may be practiced otherwise than as specifically illustrated and described.

I claim:

1. A control system for hydraulic presses equipped with a main piston-cylinder assembly and a balancing piston-cylinder assembly for the movable head of the press, the main cylinder having an area greater than that of the balance cylinder, comprising a source for supplying a constant volume of fluid at varying pressures connected to the main cylinder, a constant pressure source of fluid for the balancing cylinder, a fluid pressure operable valve interconnecting the balancing cylinder and the main cylinder to effect rapid lowering of the movable press head, yieldable pressure means disposed to normally urge said interconnecting valve towards its closed position, and other pressure means which is selectively operable for offsetting the effect on said interconnecting valve of said first-mentioned pressure means.

2. A control system according to claim 1 which is equipped with a by-pass valve for diverting fluid supplied to the main cylinder, and means for rendering the operation of the valve interconnecting the balance cylinder with the main cylinder dependent upon the pressure acting upon said by-pass valve and so operative as to prevent opening of the interconnecting valve when the proper functioning of the press requires it to be closed.

3. A control system according to claim 1 in which the interconnecting valve is arranged to be opened by the pressure of the fluid supplied to the balance cylinder, yieldable means normally urging said interconnecting valve toward its closed position, a by-pass valve for diverting fluid supplied to the main cylinder, and means coordinated with the movement of the by-pass valve for offsetting the effect upon the interconnecting valve of the yieldable means normally urging the interconnecting valve toward its closed position.

4. A control system according to claim 1 in which the interconnecting valve is arranged to be opened by the pressure of the fluid supplied to the balance cylinder, yieldable means normally urging said interconnecting valve towards its closed position, a by-pass valve for diverting fluid supplied to the main cylinder, a servo-motor for operating said by-pass valve, and means for utilizing the servo-motor to offset the effect of said yieldable means upon said interconnecting valve and so disposed that the operation of said interconnecting valve and said by-pass valve is coordinated by said servo-motor.

5. A control system according to claim 1 which is equipped with a by-pass valve for diverting fluid supplied to the main cylinder, a lever for operating said by-pass valve, and means for modifying the action of said lever to immobilize the movable press head at points within its stroke approximately in proportion to the position of said lever within the latter's stroke.

6. A control system according to claim 1 which is equipped with a by-pass valve for diverting fluid supplied to the main cylinder, a lever for operating said by-pass valve, and means for automatically modifying the effect of said by-pass valve lever upon said by-pass valve to produce a pressure in said main cylinder sufficient to counteract the force of the balance cylinder to poise the movable press head at all positions of said lever.

7. A control system according to claim 1 which is equipped with a choke arranged to regulate the flow of the fluid discharged from the balancing cylinder into the main cylinder to control the lowering speed of the movable press head.

8. A control system according to claim 1 which is equipped with a pair of fluid circuits for by-passing the fluid pressure supplied to the main cylinder, a valve in each of said by-pass circuits for controlling the flow through them, a servo-motor for opening and closing one of said by-pass valves, a power unit for opening the other of said by-pass valves, and a connection between said servo-motor and said power unit for rendering the operation of said power unit dependent upon the operation of said servo-motor.

9. A control system for hydraulic presses equipped with a main piston-cylinder assembly and a balancing piston-cylinder assembly for the movable head of the press, the main cylinder having an area greater than that of the balance cylinder, comprising a source for supplying a constant volume of fluid at varying pressures connected to the main cylinder, a constant pressure source of fluid for the balancing cylinder, a valve for interconnecting said main cylinder and said balancing cylinder to effect rapid lowering of the movable press head, yieldable pressure means for normally urging said interconnecting valve toward its closed position, a by-pass valve for by-

passing the fluid normally delivered to the main cylinder, an hydraulic servo-motor for actuating said by-pass valve, a control element for controlling the operation of said servo-motor, a pressure-responsive element for rendering ineffective the pressure means provided for urging said interconnecting valve to its closed position, and means connecting said servo-motor to said pressure-responsive element for utilizing the pressure in the servo-motor to actuate said pressure-responsive element to adjust the yieldable pressure means for the interconnecting valve when the servo-motor is operated to close the throttling valve and thereby facilitate the opening of said interconnecting valve.

10. A control system according to claim 9 which is equipped with a displacing element for modifying the action of the control element for the servo-motor, and a drive for said displacing element actuated by the movable press head of the press.

11. A control system according to claim 9 which is equipped with a cam for modifying the action of the control element of the servo-motor, and which has an operating surface so formed as to bear a direct relation with the various positions of the movable head of the press.

12. A control system according to claim 9 which is equipped with an hydraulic motor for rendering inoperable the pressure means provided for urging the interconnecting valve toward its closed position, and a fluid connection between said hydraulic motor and said servo-motor arranged to employ the hydraulic pressure in the servo-motor to energize said hydraulic motor.

13. A control system for hydraulic presses equipped with a main piston-cylinder assembly and a balancing piston-cylinder assembly for the movable head of the press, comprising a source of fluid pressure for the main cylinder, a source of fluid pressure for the balancing cylinder, an interconnecting valve for connecting the pressure supply for the balancing cylinder with the pressure supply for the main cylinder, a spring mounted on said interconnecting valve and arranged to normally urge said valve to its closed position, a power unit connected to said spring to render the spring's action on the interconnecting valve ineffective when the power unit is energized, an exhaust circuit for the main cylinder of the press, a by-pass valve disposed in said exhaust circuit, a servo-motor for actuating said by-pass valve and for energizing said power unit, and control means for regulating the operation of said servo-motor.

FLORENCE C. BIGGERT, Jr.

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