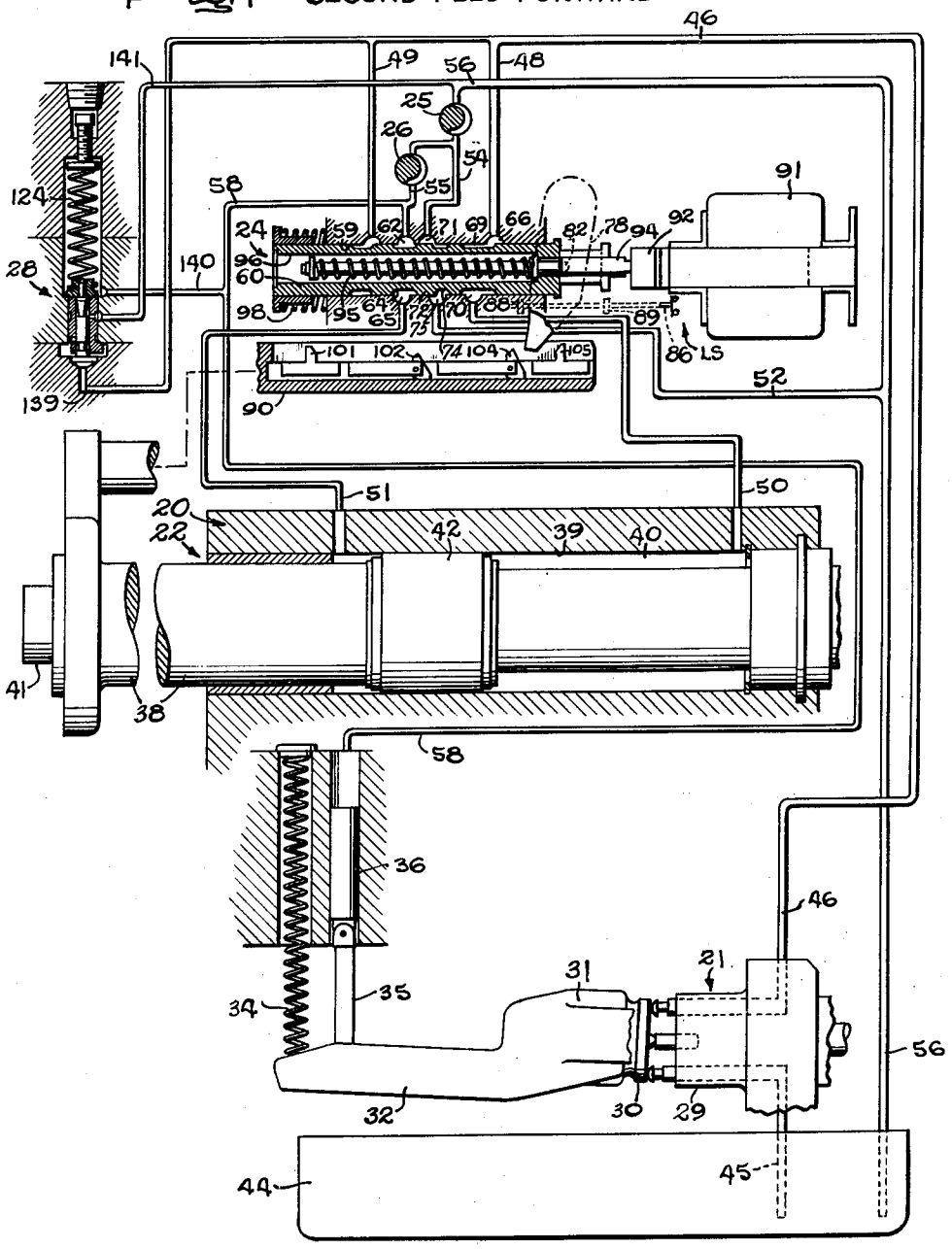
Oct. 17, 1950 M. A. MATHYS 2,526,034

SMALL HYDRAULIC UNIT

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Fig. 1 SECOND FEED FORWARD



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Fig. 2

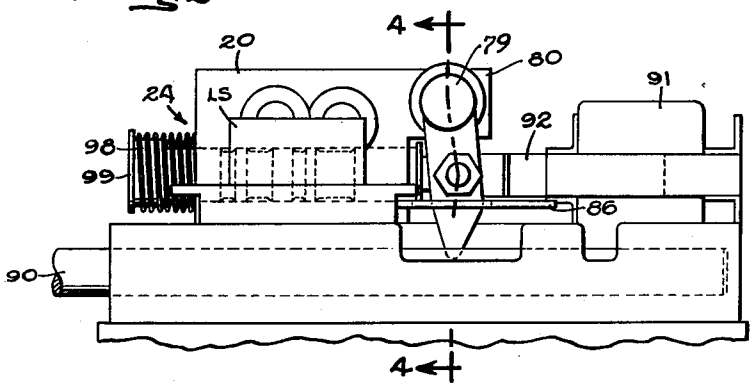
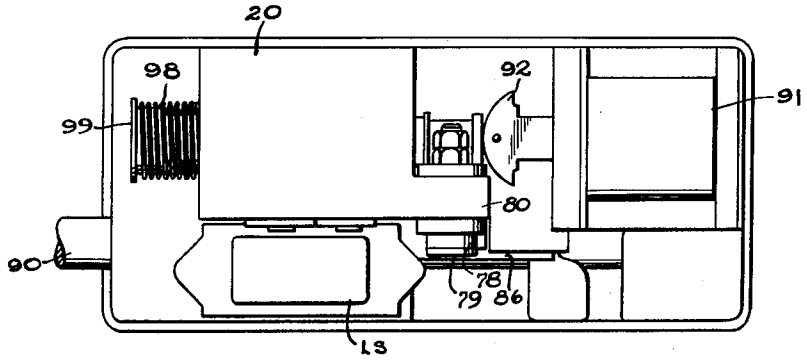


Fig. 3



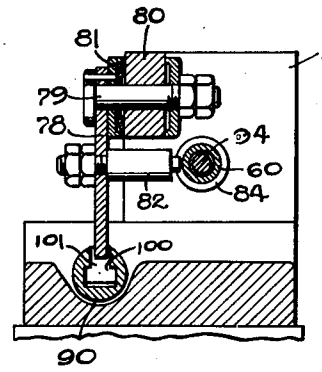


Fig. 4

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Fig. 5 RAPID TRAVERSE FORWARD

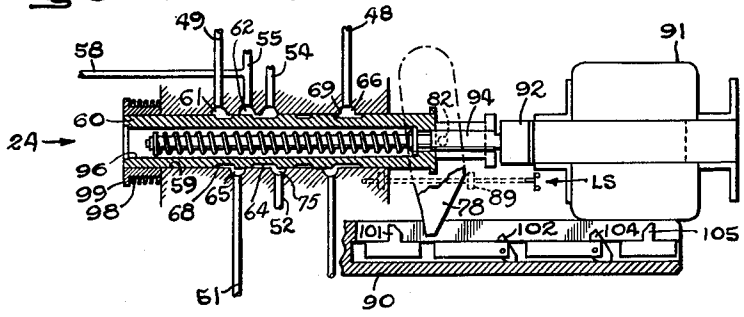


Fig. 6 FIRST FEED FORWARD

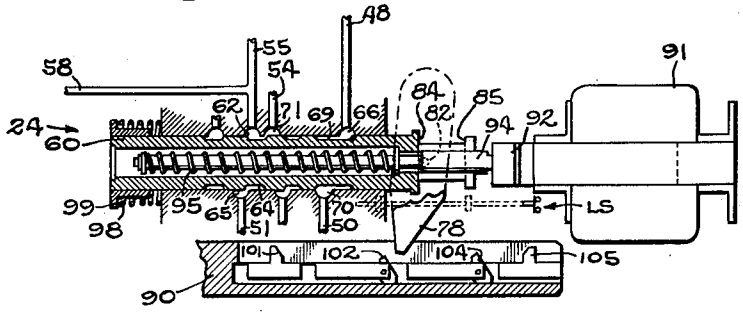
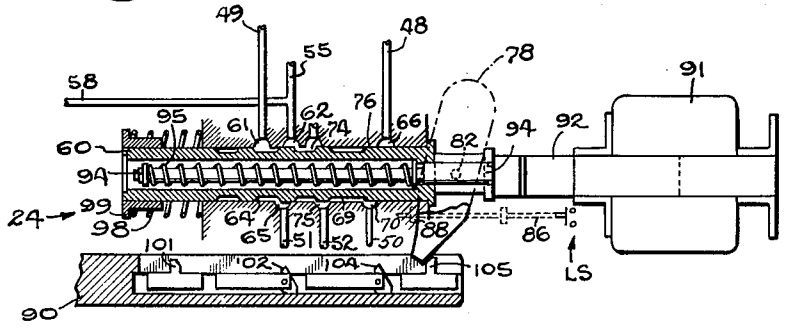


Fig. 7 RAPID TRAVERSE RETURN



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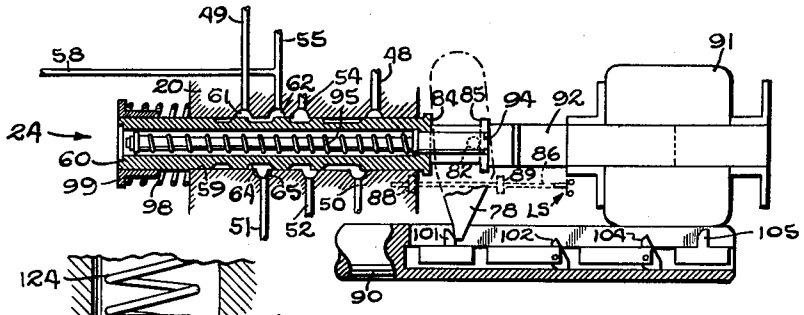
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Fig. 8 STOP POSITION



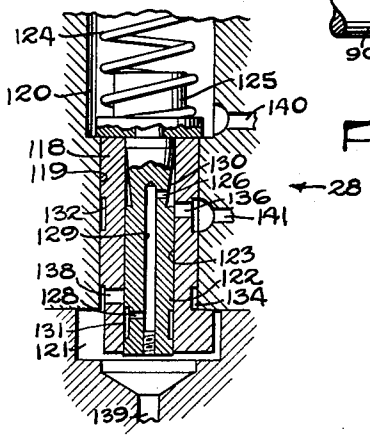
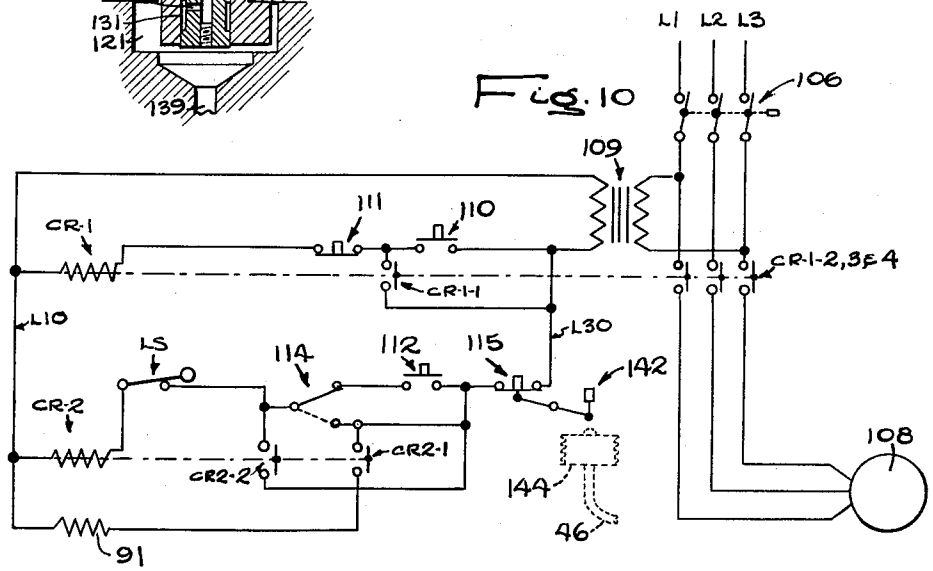


Fig. 9

Fig. 10



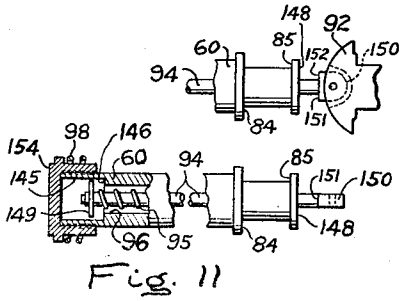


Fig. 12

Fig. 11

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2,526,034

SMALL HYDRAULIC UNIT

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Application June 27, 1947, Serial No. 757,562

21 Claims. (Cl. 60—52)

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The present invention pertains generally to an improvement in a hydraulic power system of the general type disclosed in applicant's prior Patent No. 2,416,339 and his copending application Serial No. 473,490, filed January 25, 1943, now Patent No. 2,427,970 issued Sept. 23, 1947. More specifically, the invention relates to improved actuating means for a hydraulic control valve, having particular, but by no means exclusive, utility in a small, self-contained, hydraulic power unit for a machine tool.

One of the objects of the invention is to provide a unit of the character set forth and having improved actuating means for the main control valve whereby the necessity for pilot valves, auxiliary actuating valves and complex mechanical linkages is completely eliminated.

Another object of the invention is to provide a valve member shiftable into a plurality of control positions as an incident to the shifting of an abutment member into a lesser number of positions. A related object is to provide actuating means for the valve member comprising a movable abutment effective to directly position the valve member in certain steps of a sequence and to indirectly position the valve member in the remainder of the steps of such sequence.

A further object is to provide valve actuating means of the above type including a plurality of means for applying forces to the valve member and also including a means for governing the sequential application of various force combinations to the valve member by the force-applying means.

Still another object is to provide a unit of the character set forth and having valve actuating means responsive to the actuation of a pressure-sensitive switch for automatically returning the valve to a stop position when the pressure in the system exceeds a predetermined value.

A further object is to provide a hydraulic unit of the foregoing type and which will be compact, simple, and suitable for high spindle speeds.

Other objects and advantages will become apparent as the following description proceeds, and from the accompanying drawings, in which:

Figure 1 is a diagrammatic view of an exemplary system embodying the present invention.

Fig. 2 is a fragmentary side elevational view of the upper portion of an illustrative unit employing the system of Fig. 1.

Fig. 3 is a fragmentary plan view of the unit shown in Fig. 2.

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Fig. 4 is a fragmentary transverse sectional view of the upper portion of the unit taken in the plane of line 4—4 in Fig. 2.

Figs. 5 through 8 inclusive are fragmentary vertical sectional views taken longitudinally through the main control valve with the latter respectively shown in the Rapid Traverse Forward, First Feed Forward, Rapid Traverse Return, and Stop positions.

Fig. 9 is an enlarged fragmentary vertical sectional view through the pump pressure limiting valve.

Fig. 10 is a wiring diagram showing the various electrical controls for the unit.

Figs. 11 and 12 are fragmentary views of a modified linkage for coupling the movable core of the solenoid to the valve plunger.

While the invention is susceptible of various modifications and alternative constructions, a preferred embodiment has been shown in the drawings and will be herein described in some detail, but it is to be understood that there is no intention to limit the invention to the specific form disclosed, the intention being, on the contrary, to cover all modifications and alternative constructions falling within the spirit and scope of the invention as expressed in the appended claims.

Turning now to Fig. 1, an illustrative system embodying the present invention is there shown comprising a hollow housing **20** which may be readily bolted to any appropriate support (not shown). Mounted within the housing **20** are the various sub-assemblies that make up the complete unit, such as a variable delivery motor driven pump **21**, a hydraulic actuator **22**, a main control valve member **24**, feed control orifices **25** and **26**, and a pressure limiting valve **28**.

The pump **21** is of the variable delivery class and preferably of the swash plate type, although other variable delivery pumps may be found satisfactory. Briefly, the pump includes a plurality of plunger devices carried by a rotatable cylinder body **29**, the plungers being arranged in an annular pattern with their axes parallel to the rotational axis of the cylinder body. A rockable swash plate **30** is pivotally mounted with its face in abutment with the outer ends of the pump plungers, the pivot point being located so as to balance the plunger reaction on both sides thereof. When the swash plate **30** is parallel with the end of the cylinder body **29**, the fluid delivery rate of the pump is at the minimum or zero value. However, as the swash plate is tilted with respect to the end of the cylinder body **29**,

the fluid delivery rate of the pump approaches its maximum value. The swash plate 30 is carried by a supporting block 31 having an adjusting arm 32 for altering the angular position of the plate. A tensile spring 34, which is connected between a fixed point in the housing 20 and the outward end of the arm 32, tends to urge the swash plate toward a position of maximum angularity and hence maximum fluid delivery. Also connected to the outward end of the arm 32 by means of a link 35 is a hydraulic plunger 36, the latter serving, when subjected to sufficient hydraulic pressure, to oppose the spring 34 and to rock the swash plate 30 into a parallel position with respect to the end of the cylinder body 29, thus reducing the fluid delivery rate of the pump.

The hydraulic actuator 22 comprises a tubular quill 38 which is mounted for reciprocation in a cylindrical bore 39, the annular space between these members serving as a piston chamber 40. The quill 38 rotatably supports a tool spindle 41, the details of which are not important here, other than the fact that the spindle is supported for reciprocation as well as rotation. Intermediate the ends of the quill, a sleeve-like piston 42 is formed unitary therewith, and in the present instance the piston is of the differential type with its rearward area approximately equal to twice its forward area.

The main control valve 24 is provided for the purpose of governing the movements of the actuator 22. This is accomplished by interposing the valve between the pump 21 and the actuator 22 and connecting these members by means of suitable hydraulic conduits. Thus fluid is drawn from a reservoir or sump 44 via a suction line 45, is delivered by the pump 21 to the main control valve 24 via a pressure conduit 46 and branch lines 48, 49, thence passing to one or both ends of the piston chamber 40 via lines 50, 51. Fluid may be exhausted from either end of the piston chamber 40 via the line 50 or 51, leaving the main control valve 24 via branch line 52, 54, or 55 and entering an exhaust conduit 56 which discharges into the sump 44. The adjustable feed orifices 25 and 26 are connected respectively into branches 54 and 55, the former serving to meter into the exhaust conduit fluid discharged from the control valve 24, thereby varying the back pressure in a pump control line 58 and thus regulating the rate of delivery of the pump 21.

The main control valve 24 is of the reciprocable as distinguished from the rotary type, being slidably disposed and axially shiftable within a ported cylindrical bore 59 in the housing 20 and adapted to define an operating cycle for the actuator 22. Basically, the valve comprises an axially shiftable plunger or spool 60 having a plurality of annular shoulders and grooves thereon which, in cooperation with the ports in the cylindrical bore, permit the valve to be employed in a series of successive steps or control positions, including Stop, Rapid Traverse Forward, First Feed Forward, Second Feed Forward, and Rapid Traverse Return. These will be discussed below in the foregoing order.

Referring to Fig. 8, the main control valve 24 is shown in the Stop position, from which it may be shifted rightwardly to begin an operating cycle for the actuator 22. When the valve 24 is in this position, the actuator is in its fully retracted position within the cylinder 39. The pressure conduit 46, via its branch 49 is then directly connected to the pump control line 58 in order to maintain the swash plate 30 substantially parallel to the end of the cylinder block 29 and thus keep the pump 21 in a condition of minimum fluid delivery. This arrangement is achieved by means of an inlet port 61 in the valve cylinder 59 connecting the branch line 49 to the latter, an outlet port 62 in the valve cylinder connecting the latter with the branch discharge line 55 which in turn communicates with the pump control line 58, and an annular groove 64 in the valve spool 60, the groove partially overlying both of the ports 61, 62. The circuit may readily be traced by reference to Fig. 8, taken in connection with the system illustrated in Fig. 1. Its course is as follows: pressure conduit 46, branch line 49, inlet port 61, groove 64 of the valve spool, outlet port 62, branch discharge line 55, and pump control line 58. As a natural consequence of this position of the valve spool 60, some pressure fluid will pass from the groove 64 through a port 65 and thence to the forward end of the actuator piston chamber 40 via the conduit 51. Because of the circuit just described any pressure in such fluid will merely produce a gentle restraining force upon the forward face of the piston 42 tending to hold the actuator 22 in the retracted position.

Upon the initiation of the operating cycle, the spool 60 of the valve 24 is shifted to its extreme right-hand or Rapid Traverse Forward position, as represented in Fig. 5, and which causes the actuator 22 to move in a forward or outward direction. When the valve is in this position, fluid under pressure is conducted to both ends of the actuator piston chamber 40 via the lines 50 and 51, subjecting both sides of the piston 42 to pressure. Due to the difference in area between the forward and rearward sides of the piston 42, the latter will move forward at a rate proportional to such area difference and also proportional to the pump discharge pressure. Communication between the pressure conduit 46 and the valve cylinder 59 now occurs via the branch line 48 and an inlet port 66, as well as through the branch 49 and its inlet port 61. Within the cylinder 59, an annular groove 68 in the valve spool connects the inlet port 61 with the port 65, while another annular groove 69 in the spool 60 connects the port 66 with a port 70. These grooves permit fluid under pressure to enter the actuator lines 50 and 51 through the ports 70 and 65, passing from the former to their respective ends of the piston chamber 40. Although pressure is applied to both sides of the piston 42, it will be noted that there is no return flow of fluid from the actuator 22 to the exhaust line 56, the lands of the valve spool 60 completely isolating the discharge lines 52, 54, 55 from the actuator lines 50, 51. Due also to such isolation, the pump control line 58 is allowed to communicate with the exhaust conduit 56 and the pump 21 therefore operates at maximum fluid delivery.

Turning to Fig. 6, the First Feed Forward position of the valve 24 is there shown, the valve spool 60 having been shifted one step to the left. Under this condition, the valve is arranged to admit fluid under pressure to the rearward end of the piston chamber 40 and exhaust it from the forward end at a predetermined rate governed by the setting of the first feed orifice 25. The application of pressure to the large rearward area of the piston 42 produces the substantial force necessary to move the actuator 22 and the tool spindle 41 against the resistance offered by the workpiece, while the use of the orifice 25

in the exhaust circuit from the piston chamber 40 insures that the actuator will be advanced at the proper feed rate. For the purpose of obtaining a pump delivery rate commensurate with the feed rate determined by the setting of the first feed orifice, the valve 24 connects the discharge line from the piston chamber 40 to the pump control line 58. The foregoing circuits may be followed by reference to Figs. 1 and 6. Fluid under pressure passes from the branch 48 of the pressure conduit 46 and into the valve cylinder 59 via the inlet port 66. The annular groove 69 in the valve spool 60 connects the port 66 with the port 70 and the actuator conduit 50, the latter admitting the pressure fluid to the rearward end of the piston chamber 40. Fluid discharged from the forward end of the chamber 40 is conducted by the actuator line 51 to the port 65, from which it passes along the annular groove 64 in the valve spool. The groove 64 is positioned to overlie the exhaust port 62 of the exhaust branch line 55, and to overlie partially a port 71 communicating with the exhaust branch line 54. The latter conducts the discharged fluid to the first feed orifice 25 which meters it into the exhaust conduit 56. Fluid entering the port 62 is directed by the branch line 55 into the pump control line 58 for the purpose already explained. The second feed orifice 26, being by-passed in this arrangement, is thereby rendered ineffective.

The valve 24 in its Second Feed Forward position is illustrated in Fig. 1, the spool 60 having been moved another step to the left. The valve 24 now admits fluid under pressure to the rearward end of the piston chamber 40 and permits it to be exhausted therefrom at a predetermined rate governed by the settings of both the feed orifices 25 and 26. Pressure fluid is applied to the rearward area of the piston 42 in the same manner as that obtaining in the First Feed Forward position, namely, from the pressure conduit 46 into the cylinder 59 via the branch line 48 and the inlet port 66, along the annular groove 69 in the valve spool 60, out through the port 70 and into the rearward end of the piston chamber 40 via the actuator line 50. Fluid discharged from the forward end of the chamber 40 passes into the cylinder 59 via the actuator line 51 and the port 65. However, the groove 64 now overlies only the ports 65 and 62, the port 71 of the branch line 54 being isolated from the port 62 by an annular land 72 on the valve spool. Thus the discharged fluid is conducted from the port 65 to the port 62, via the groove 64, thence passing along the branch line 55 through the second feed orifice 26 and also through the first feed orifice 25, finally entering the exhaust conduit 56 which empties into the sump 44. Since the orifices are connected in series under the above conditions, the piston 42 and consequently the actuator 22 will move outwardly at a rate determined principally by the sum of the restrictive effects of the orifices. In the event that the latter when in series provide more resistance to fluid flow than is needed to achieve a given rate of actuator movement, such rate may be increased through the expedient of by-passing a relatively small quantity of the fluid discharged from the orifice 26, via the branch line 54, the port 71, an annular groove 74 in the valve spool which barely communicates with the port 71, an exhaust port 75, and the branch 52, to the exhaust conduit 56. As in the case of the First Feed Forward position, fluid from the branch line 55 is admitted to the pump control line 58 for the purpose of securing the proper fluid delivery rate.

The Rapid Traverse Return position of the valve 24, and which represents the extreme leftward position of the valve spool 60, is shown in Fig. 7. The main branch 48 of the pressure conduit 46 is now completely blocked off by a land 76 on the valve spool, and consequently no fluid under pressure is supplied to the rearward end of the piston chamber 40 or the rearward face of the piston 42. Pressure fluid is, however, admitted to the forward end of the chamber 40, exerting a force on the piston 42 tending to return the actuator to its retracted position. This circuit includes the branch 49, the intake port 61, the annular groove 64, the port 65, and the actuator line 51 which communicates with the chamber 40. Fluid discharged from the rearward end of the chamber 40 passes through the actuator line 50, and port 70, the annular groove 69, the exhaust port 75, and thence to the branch 52 of the exhaust line 56. The application of fluid pressure to the forward face of the piston 42, together with the unrestricted exhaust path for the fluid discharged from the rearward end of the piston chamber 40, results in a high speed return of the actuator 22 to its retracted position, thus completing the operating cycle. Another contributing factor to such speed is the fact that the pump control line 58 is connected directly to the exhaust line 56 via the ports 62, 75, and the groove 74 in the valve spool, producing the maximum fluid delivery rate for the pump.

Actuation of control valve

For the purpose of moving the plunger of the valve 24 through a series of successive steps in timed relation to the axial movements of the actuator 22 and the tool spindle 41, thereby defining an operating cycle, means are provided for the application of four external forces to the plunger 60. These forces are applied sequentially in combinations governed by the position of still another means in the form of a movable abutment member, which, although shiftable into a lesser number of releasable positions than the valve plunger, actually causes the plunger to occupy a greater number of positions than such means. In the illustrative embodiment (Fig. 1 to 8 inclusive), the movable abutment comprises a rocker arm 78 depending from a pivotal support 79 carried by an extension 80 of the unit housing 20. An adjustable frictional device such as a disk spring 81 is associated with the support 80 for enabling the arm to withstand a predetermined force without moving.

The rocker arm 78 is coupled to two other members of the valve assembly, namely, the valve plunger 60 and a limit switch LS, by means of a pair of lost-motion connections. For engaging the valve plunger, the arm 78 is provided with a rigidly attached transverse pin 82 disposed between and projecting into the path of a pair of axially spaced apart shoulders 84, 85, formed in the valve plunger adjacent its rearward or inner end. For engaging the limit switch LS, the arm 78 has associated therewith an operating bar 86 having abutments interposed in the path of the lever near each extreme of its travel. Preferably, the bar 86 is formed as a flat plate with an elongated slot for the lever, as indicated in Fig. 2, but for purposes of clearer illustration the bar has been represented in Figs. 1 and 5 to 8 as a rod carrying abutments in the form of a pair

of spaced apart nuts 88, 89 which are engageable with the arm 78.

The force-applying means earlier referred to include a longitudinally movable dog bar or tripping bar 90 rigidly coupled at its outer end to the bearing housing of the tool spindle 41, as indicated schematically in Fig. 1, and arranged to move the rocker arm 78 through a series of successive angular positions with respect to the axis of the valve plunger 60; electromagnetic means such as a solenoid 91 having a movable core 92 and acting through a slidable tension rod 94 and an internal compression spring 95 which are housed within an axial bore 96 in the valve spool 60, as shown in Figs. 1, and 5 to 8; the disk spring friction means 81 associated with the pivotal mounting of the arm 78, illustrated in Fig. 4; and a relatively weak second compression spring 98 encircling the forward or outer end of the valve spool 60, being disposed between an adjustable collar 99 threaded on the valve spool and the unit housing 20 so as to exert a yieldable thrust or bias tending to move the spool 60 to the extreme left.

Referring more particularly to the drawings just mentioned, the dog bar 90 has a T-shaped slot 100 milled therein for receiving a plurality of axially adjustable, spaced-apart tripping dogs. In the embodiment illustrated, going from left to right, these include a stop dog 101, a one-directional first feed dog 102, a one-directional second feed dog 104, and a reverse dog 105. The end of the rocker arm 78 is slightly narrower than the width of the upper portion of the slot 100, enabling it to be projected into the slot and consequently into the path of movement of the dogs. Thus each dog is arranged to apply a positive, intermittent force to the valve spool 60 at a different point in the cycle, assuming the slack has been taken up in the lost-motion connection between the rocker arm and the valve spool.

As already indicated to some extent, one of the means for taking up the slack in the lost-motion connection between the arm 78 and the valve spool 60 comprises the solenoid 91. When the latter is energized, a yieldable thrust is exerted upon the valve spool 60 through the movable core 92 of the solenoid, the tension rod 94, and the internal compression spring 95, tending to move the valve spool to the right. This thrust or force is transmitted to the rocker arm through the shoulder 84 and the transverse pin 82.

After a dog has passed out of contact with the lower end of the rocker arm 78, the friction means 81 associated with the pivot thereof becomes effective to hold the arm in the position determined by such dog until the arm is acted upon by the next dog, the means 81 offering a force in opposition to any thrust produced by the solenoid 91 as well as to the biasing force of the external compression spring 98.

The small biasing force of the external compression spring 98 is constantly exerted upon the forward or outer end of the valve spool 60, tending to urge the former to the extreme left or Rapid Traverse Return position shown in Fig. 7. This force is rendered effective only after the solenoid 91 has been deenergized and the rocker arm has been moved clockwise from the position it occupies when the valve spool is in the Stop position.

The relationship of the various positioning means discussed above will be better understood following a brief description of the manner in which they are coordinated to actuate the valve

24. Considering first the Stop position, illustrated in Fig. 8, it will be observed that the rocker arm 78 occupies its extreme counterclockwise position, having been moved there by the stop dog 101 on the dog bar 90. The solenoid 91, being deenergized, exerts no force on the valve spool 60 and, as a consequence, the latter is shifted leftwardly by the external compression spring 98 so as to bring the shoulder 85 into abutment with the pin 82 of the rocker arm 78. With the spool 60 in the position shown, the fluid delivery rate from the pump 21 is held at a minimum and the actuator 22 remains in its fully retracted condition. Assume now that the solenoid 91 becomes energized by the depression of an appropriately connected starting switch. The valve spool 60 will be immediately shifted to the right due to the thrust exerted thereon through the movable core 92, the tension rod 94, and the internal compression spring 95, bringing the shoulder 84 into abutment with the pin 82 of the rocker arm 78, and thereby taking up in one direction the slack in the lost-motion connection between the rocker arm and the valve plunger or spool. Notwithstanding the fact that such slack or lost-motion is fully taken up by the thrust of the solenoid, the friction means 81 serves to prevent the rocker arm from being rotated in either direction except in response to the considerable force exerted by the dog bar through the tripping dogs. The valve spool 60 is now in the Rapid Traverse Forward position, shown in Fig. 5, causing the actuator 22 to move leftwardly or forwardly and to pull with it the dog bar 90. This continues until the first feed dog 102 is cammed against the lower end of the rocker arm 78, rotating the latter clockwise through a predetermined angle. Since the solenoid remains energized, the pin 82 will remain in abutment with the shoulder 84 and the valve spool 60 will therefore be positively positioned in response to the positioning of the rocker arm, which further compresses the spring 95. The valve spool 60 is now in the First Feed Forward position, as indicated in Fig. 6, and the actuator 22 will continue its leftward or forward movement at a feed rate predetermined by the setting of the first feed orifice 25. This motion continues until the second feed dog 104, which projects higher than the dog 102, engages the lower end of the rocker arm 78, shifting the latter clockwise through another angle. The abutting relationship between the shoulder 84 and the pin 82 persists due to the fact that the solenoid 91 is still energized, enabling the rocker arm 78 to move the valve spool 60 positively and directly into the Second Feed Forward position, illustrated in Fig. 1. The leftward or forward motion of the actuator 22 will continue but at a feed rate determined principally by the combined settings of both feed orifices 25 and 26. The dog bar 90 of course moves with the actuator 22, eventually bringing the reverse dog 105 to a point where it cams the rocker arm 78 clockwise to the extreme position of the latter. As an incident to its clockwise travel, the arm 78 is brought into engagement with the left abutment 88 of the limit switch operating bar 86, moving the latter to the left and opening the limit switch LS. Due to the fact that the switch LS is connected in series with the solenoid 91, the opening of the former will deenergize the solenoid and thereby permit the valve spool 60 to be moved to its extreme left position under the biasing force of the external compression spring 98, at the same time

moving the shoulder 84 out of contact with the pin 82 on the rocker arm. This locates the valve spool 60 in the Rapid Traverse Return position, shown in Fig. 7. The forward or outward motion of the actuator 22 and the dog bar 90 is thus arrested and reversed, the actuator 22 now starting to move toward its retracted position. This motion continues without interruption by the one-directional feed dogs 102 and 104, which when traveling in the reverse direction, merely tilt out of the way in the event of contact with the lower end of the rocker arm 78. The latter, however, remains in its extreme clockwise position until engaged by the stop dog 101, whereupon it is moved counterclockwise to the position indicated in Fig. 8. As an incident to the counterclockwise motion of the rocker arm, the pin 82 moves across the lost-motion space and into abutment with the shoulder 85, rendering the rocker arm effective to move the valve spool 60 positively and directly to the right until it occupies the Stop position shown in Fig. 8. The rocker arm in the course of its counterclockwise movement also closes the limit switch LS due to engagement with the right-hand abutment 89 of the switch operating bar 86. Depending upon the arrangement of the electrical controls, the closing of the limit switch may or may not be effective to energize the solenoid 91 and initiate another operating cycle.

It will be appreciated from the foregoing that the rocker arm 78, by reason of being shifted through a series of four successive angular positions, is utilized to define accurately the five successive steps or control positions assigned to the valve spool or plunger 60. Due to the use of the lost-motion connections between the rocker arm, the valve plunger, and the limit switch, the rocker arm is rendered effective when the solenoid is energized to directly position the valve plunger in certain of the successive steps or control positions of the latter, the arm also being rendered effective when the solenoid is deenergized to indirectly position the plunger in the remainder of the control positions thereof. Thus the plunger 60 is directly positioned by the rocker arm 78 in the steps of Stop, Rapid Traverse Forward, First Feed Forward, and Second Feed Forward; the plunger is indirectly positioned by the rocker arm in the step of Rapid Traverse Return.

Electrical control circuit

Turning to the schematic wiring diagram of Fig. 10 it will be noted that three phase A. C. power is supplied to the unit through the lines L1, L2, L3 and main line switch 106. An electrical motor 108, which serves as the driving means for the entire unit, obtains power from this supply through contacts CR1–2, 3 and 4 of a motor control contactor CR–1. Power for a single phase control circuit L10, L30 is derived from phase L1, L3 through a control transformer 109. This circuit includes the motor control contactor CR–1, the solenoid 91, and a contactor CR–2 for the solenoid, together with various control switches. The motor 108 is controlled by means of the contactor CR–1 which, when energized in response to the closing of a started switch 110, completes a circuit through the line contacts CR1–2, 3 and 4, also simultaneously closing a pair of sealing-in contacts CR1–1, the latter being connected in parallel with the starter switch. The motor is stopped by depressing the plunger of a stop switch 111 which interrupts the circuit of the contactor CR–1, thereby opening the contacts CR1–1 and CR1–2, 3 and 4. The contactor CR–2 is arranged to energize the solenoid 91 through the closure of contacts CR2–1. To this end, there is connected in series with the contactor the limit switch LS, a cycle starter switch 112, a selector switch 114, and an emergency cycle stop switch 115. A pair of sealing-in contacts CR2–2 are connected in parallel with the switches 112 and 114 for holding in the contactor following the momentary closure of the switch 112. Assuming that the plunger 60 of the valve 24 is in the Stop position and that the limit switch LS is therefore closed, if it be desired to initiate only a single complete operating cycle the selector switch 114 would be raised to its upper position. The emergency cycle stop switch 115 being normally closed, it then becomes possible to energize the contactor CR–2 and hence the solenoid 91 by momentarily depressing the plunger of the cycle starter switch 112, the sealing-in contacts CR2–2 serving thereafter to keep the circuit closed. The energizing of the solenoid 91 serves to shift the valve spool 60 into the Rapid Traverse Forward position and the actuator 22 is consequently put through the previously described steps of the operating cycle. When the actuator reaches the forward extreme of its travel, the reverse dog 105 engages the rocker arm 78 causing the latter to open the limit switch LS, deenergizing the contactor CR–2 and consequently the solenoid 91. The actuator 22 is thereupon returned automatically to the Stop position, thereby completing the cycle. Note that the closing of the limit switch upon the return of actuator to the Stop position is ineffective to initiate another cycle with the arrangement just described. If it be desired that the cycle should be automatically repeated, it is merely necessary to move the selector switch 114 to its lower position (indicated in broken lines in Fig. 10), thus rendering the limit switch effective to initiate the next cycle automatically upon the completion of the preceding one.

Safeguards

Various safeguards, both hydraulic and electrical, are incorporated into the unit to make it more reliable and foolproof than prior units. Referring to Figs. 1 and 9, one hydraulic safeguard resides in the use of the overload valve 28. This valve comprises a stationary bushing 118 mounted within a suitable bore 119 in the unit housing. Formed at either end of the bore and in communication therewith are an upper pressure chamber 120 and a lower pressure chamber 121. A hollow plunger 122 is slidably disposed within an axial bore 123 in the bushing 113 and is held in place by means of a compression spring 124 acting upon its upper end through a collar 125. Radial passages 126 and 128 in the plunger respectively connect a small diameter axial bore 129, which is closed at both ends, with annular peripheral grooves 130 and 131, the groove 130 being tapered and located in the upper portion of the plunger. Intermediate its ends, the bushing 118 is also provided with a pair of annular grooves 132, 134 in its external periphery and which communicate with the bore 123 therein via radial passages 136 and 138, respectively. The lowermost pressure chamber 121 of the valve is connected to the pressure conduit 46 via a branch line 139 while the upper pressure chamber 120 communicates with the pump control line 58 via a branch line 140. The uppermost annular grove 132 in the bushing 118 is connected directly to the exhaust conduit 56 through the branch 141. In operation, the plunger 122 occu-

pies the position shown in the drawings as long as the pressure in the conduit 46 remains below a predetermined value, and there is no flow of fluid through the restricted bore 129 since the lower peripheral groove 131 is completely blocked off. If the pressure in the conduit 46 exceeds the predetermined maximum value, the plunger 122 is forced upwardly, bringing its lower annular groove 131 into registry with the passage 138 in the bushing and connecting its upper annular groove 130 with the upper pressure chamber 120. Thus high pressure fluid is conducted from the chamber 121, through the passages of the plunger 122 and into the upper chamber 120, flowing therefrom to the branch line 140 and finally to the pressure control line 58. The suddenly increased pressure in the line 58 immediately causes the swash plate 30 of the pump to assume its parallel position, minimizing the fluid delivery rate of the pump.

Another hydraulic safeguard resides in the fact that in the event of a failure of electrical power, the constant biasing force of the external compression spring 98 will immediately become effective to shift the valve spool 60 to either the Rapid Traverse Return or the Stop position, depending upon the angular position of the rocker arm at the time of the failure.

One of the electrical safeguards already mentioned is the emergency cycle stop switch 115. The opening of this switch deenergizes both the contactor CR–2 and the solenoid 91, and the switch is susceptible of actuation at any point in the operating cycle of the unit. Once having been opened, the switch 115 does not automatically return to a closed position, but must be deliberately closed through the use of a manual reset 142. Ordinarily, manual means alone will be found satisfactory for actuating the switch 115. However, if desired a pressure controlled actuator 144, shown schematically as a bellows, may be mounted adjacent the operating arm of the switch 115 for opening the latter in the event that the pressure in the system becomes excessive. Such a condition might arise, for example, upon the breaking of a tool in the spindle 41 during one of the feeds of the cycle. Should something of this nature occur, the opening of the switch 115 immediately deenergizes the solenoid 91, permitting the valve plunger 60 to be shifted immediately through Stop and into the Rapid Traverse Return position. In lieu of the arrangement described, an equally good alternative would be the use of a separate pressure-actuated switch in series with the switch 115 but having its own manual reset.

The use of a pressure controlled device with the switch 115, or the use of a separate pressure-actuated switch in series therewith, and may be selectively employed to provide a dwell control feature in the unit. This is helpful where it is necessary to hold the actuator 22 at its forward extreme of travel long enough to complete a facing-off or similar operation. In such event, a rigid mechanical abutment is placed near the forward end of the path of the actuator 22, and when encountered by the latter a large increase in pressure results. By interposing a time delay device between the pressure actuator and the switch it becomes possible to delay the automatic return of the valve plunger and hence the actuator for the desired period of dwell. If such an arrangement is utilized, the closing of any manual reset which might be used is made ineffective to initiate another cycle until the closing of the cycle starting switch 112. When operating on a cycle including dwell return, the pressure-actuated switch does not serve as an overload protective device, and consequently it becomes necessary for the operator to observe the action of the tool to avoid jamming due to breakage or other causes.

Synopsis of operation

While various phases of operation have been separately discussed above, it will be helpful to present at this stage an overall picture of the manner in which the unit works. Assume first that the actuator 22 is in the fully retracted position and that the plunger of the control valve 24 is occupying the Stop position as indicated in Fig. 8. Under the conditions illustrated, the branch 48 of the pressure conduit 46 is completely blocked off and the fluid from the branch 49 is short circuited into the pump control line 58 via the inlet port 61, the groove 64, the port 62 and the branch line 55. This forces the pump control plunger 36 downwardly and causes the swash plate 30 to assume the minimum fluid discharge position. The operating cycle is then initiated by the momentary depression of the plunger of the cycle starter switch 112, energizing the contactor CR–2, which, in turn, energizes the solenoid 91 through the closure of the contacts CR2–1. The movable core 92 of the solenoid, together with the tension rod 94 is thereby drawn to the right, exerting an axial thrust in that direction upon the valve spool or plunger 60 through the internal compression spring 95. Consequently, the spool 60 is moved to the right until its shoulder 84 is brought into abutment with the pin 82 of the rocker arm 78. The frictional means adjacent the pivot of the rocker arm enables the latter to withstand the thrust of the solenoid, thereby accurately locating the valve spool 60 in the Rapid Traverse Forward position, shown in Fig. 5.

With the valve in the Rapid Traverse Forward position, pressure fluid is applied to both ends of the piston chamber 40, causing the piston 42 of the actuator 22 to advance forwardly or outwardly at a rate proportional to the ratio between the areas of the rearward and forward faces of the piston. Briefly, the rearward pressure circuit includes the branch 48, the inlet port 66, the groove 69, the port 70, and the actuator line 50; the forward pressure circuit comprises the branch 49, the inlet port 61, the groove 68, the port 65, and the actuator line 51. Maximum fluid delivery from the pump is obtained by connecting the pump control line 58 to the exhaust conduit 56 via a circuit including the branch 55, the port 62, the groove 64, the port 75, and the exhaust branch 52. With the advance of the actuator 22, the dog bar 90 is also drawn leftwardly, bringing the first feed dog 102 into engagement with the rocker arm 78 and rotating the latter clockwise through a small angle until it occupies the position shown in Fig. 6. Since the solenoid 91 is still energized, the shoulder 84 remains in abutment with the pin 82, and the rocker arm in its new position therefore defines a new position for the valve spool 60. This is the First Feed Forward position, whereby pressure fluid is applied to the rearward face of the piston 42 via the branch 48, the inlet port 66, the groove 69, the port 70, and the actuator line 50. Fluid discharged from the forward end of the piston chamber 40 is metered through the first feed orifice 25, causing the actuator to advance the tool

spindle at a feed rate predetermined by the setting of that orifice. The course of this discharged fluid includes the actuator line 51, the port 65, the groove 64, the port 71, and the branch line 54. Since the groove 64 overlies both ports 62 and 71, some of the fluid is admitted to the pump control line to maintain a fluid delivery rate also determined by the setting of the orifice 25. Under these conditions the actuator 22 and the dog bar 90 will continue to advance, bringing the second feed dog 104 into engagement with the rocker arm 78 and rocking the latter clockwise through another angle to the position indicated in Fig. 1. Since the solenoid 91 remains energized, it continues to take up the lost motion in the connection between the rocker arm 78 and the valve spool 60, thereby enabling the arm to position the valve spool once more, this time in the Second Speed Forward position. The hydraulic conditions obtaining under these circumstances are rather similar to those existing for First Feed Forward, except for the fact that the combined restrictive effect of the orifices in series is now utilized to govern the rate of feed. Thus pressure is applied to the rearward end of the piston 42 via the branch 48, the inlet port 66, the groove 69, the port 70, and the actuator line 50. Fluid is discharged from the forward end of the piston chamber 40 through the actuator line 51, the port 65, the groove 64, the port 62, the branch line 55, the orifice 26, the orifice 25, and the exhaust line 56. The port 71 is isolated by the land 72 from the port 62, although a small portion of the fluid discharged from the orifice 26 might be diverted directly into the exhaust line through the branch 54, the port 71, the groove 74, the port 75, and the exhaust branch 52. Because of the direct connection between the pump control line 58 and the branch 55, the back pressure in the latter, which is determined by the orifice settings, will govern the fluid delivery rate of the pump. The actuator and the dog bar 90 will continue to move to the left at the second feed rate until the reverse dog 105 engages the rocker arm 78. The arm is thereby moved to its extreme clockwise position, and as an incident to such motion it opens the limit switch LS, deenergizing the solenoid 91. This releases the thrust on the internal compression spring 95, permitting the external compression spring 98 to take charge, the latter moving the valve spool 60 to its extreme leftward or Rapid Traverse Return position. Note, however, that the shoulder 84 now no longer abuts the pin 82 of the rocker arm. The valve in this position admits pressure fluid to the forward end of the piston chamber 40 via the branch conduit 49, the inlet port 61, the groove 64, the port 65, and the actuator line 51. Fluid is discharged from the rearward end of the piston chamber via the actuator line 50, the port 70, the groove 69, the exhaust port 75, and the branch 52 of the exhaust conduit 56. The pump control line 58 is also directly connected to the exhaust line via the branch 55, the port 62, the groove 74, the exhaust port 75, and the branch 52, causing the pump to be operated at the maximum fluid delivery rate and therefore producing a rapid retraction of the actuator. This rearward travel of the actuator continues until the stop dog 101 engages the rocker arm 78, moving it together with the valve spool 60 to the Stop position and closing the limit switch LS. If the selector switch 114 is in its upper or solid line position, as shown in Fig. 10, the cycle will not repeat automatically. If, however, the switch 114 happens to be in its lower or broken line position, the closing of the limit switch LS will immediately be effective to repeat the cycle automatically without first closing the cycle starting switch 112.

Modification

For the purpose of insuring still further the positive positioning of the valve plunger 60 under the most adverse conditions, a slightly modified linkage for coupling the latter and the movable core 92 of the solenoid 91 is provided. In a valve of the type described, the plunger may occasionally become sticky in its operation due, for example, to continued idleness or to a phenomenon known in the art as hydraulic pressure lock. Stickiness due to the former condition is most apt to occur at the start of an operating cycle when the valve plunger is in the Stop position. Stickiness due to the latter or pressure lock condition may occur during a cycle when the valve plunger is in the Second Feed Forward position immediately prior to moving to Rapid Traverse Return.

In Figs. 11 and 12, an illustrative embodiment of the modified coupling linkage is shown. Thus the valve plunger 60 is formed adjacent its left hand end with a relatively short axial bore 145 of somewhat greater diameter than the bore 96, the ends of the bores meeting to define an annular abutment or shoulder 146. In the region of its extremities, the tension rod 94 is provided with suitable abutting surfaces for respectively engaging the shoulder 146 at one end of the plunger 60 and an external annular face 148 at the opposite end thereof. Accordingly, the left hand end of the rod 94 carries rigidly fixed thereto a large washer 149 of smaller diameter than the bore 145 but of sufficient size to have substantial overlap with the shoulder 146. Unitary with the right hand end of the rod 94 is a head portion 150 adapted for pivotal attachment to the solenoid core 92 and having a pair of coplanar shoulders 151, 152 adapted to abuttingly engage the face 148 on the right hand end of the plunger 60. In addition to the foregoing, the open-ended collar 99 may be replaced by an adjustable cap 154 threaded on the plunger and blocking off the ends of the bore 145 to keep out foreign material.

In operation, assume that the valve plunger is in the Stop position of Fig. 8. Upon the energizing of the solenoid 91 at the start of a cycle, the movable core 92 shifts to the right, exerting through the tension rod 94 and its washer 149 a force in excess of the pressure of the spring 95, and capable of fully compressing the same. During this time, if the valve plunger does not move in response to the reaction of the spring 95 to the compression force thereon, the plunger 60 will receive a hammer blow upon the engagement of the washer 149 with the shoulder 146. As a result of such impact, the valve plunger 60 will be jarred free, whereupon it will be shifted into the Rapid Traverse Forward position by the action of the spring 95 and the continued rightward movement of the solenoid core 92.

Assume further that the cycle has progressed to a point at which the plunger 60 is in the Second Feed Forward position. As the reverse dog 105 cams the rocker arm 78 into engagement with the abutment 88 of the limit switch bar 86, the solenoid becomes deenergized. The spring 95 which heretofore had been under compression, is now released suddenly, accelerating the tension rod 94 and the movable core 92 to the left and producing

a sharp impact between the shoulders 151, 152 of the tension arm and the external face 148 of the valve plunger. The force of this impact tends to break the plunger 60 loose from the Second Feed Forward position, rendering the spring 98 effective to shift the former through Stop and into the Rapid Traverse Return position.

I claim as my invention:

1. A hydraulic control system for machine tools and the like comprising, in combination, an actuator, a variable delivery pump, a sump, pressure and exhaust conduits leading from the pump and the sump respectively, conduits leading from said actuator, a main control valve connected with said pressure and exhaust conduits and said conduits leading from said actuator for delivering thereto fluid under pressure and receiving fluid discharged therefrom, a reciprocable valve spool slidably disposed within a ported cylindrical bore in said main control valve for defining a stepped sequence of actuator movements including the steps of rapid traverse forward, first feed forward, second feed forward, rapid traverse return, and stop, a rocker arm movable into a number of positions less than the number of said steps and having a lost-motion connection with said valve spool, electromagnetic means resiliently connected to said valve spool and arranged when energized to exert a force tending to take up the slack in the lost-motion connection thereby permitting said rocker arm to positively position said valve spool, spring means also connected to said valve spool so as to exert an opposing but weaker force than that of said electromagnetic means, a dog bar rigidly attached to said actuator and having a plurality of tripping dogs for positively positioning said rocker arm, and switch means operable by said rocker arm to deenergize said electromagnetic means and thereby permit said spring means to position said valve spool independently of said rocker arm.

2. A hydraulic control system for machine tools and the like comprising, in combination, an actuator, a variable delivery pump, a sump, fluid conduits leading from the actuator, the pump and the sump respectively, a main control valve interposed in said conduits for delivering fluid under pressure to said actuator and receiving fluid discharged therefrom, a reciprocable valve spool slidably received within a ported cylindrical bore in said main control valve for defining by means of a predetermined series of axial positions a stepped sequence of actuator movements including the steps of rapid traverse forward, first feed forward, second feed forward, rapid traverse return, and stop, a pair of axially spaced-apart shoulders in said valve spool adjacent one end thereof, a rocker arm movable into a number of positions less than the number of said steps and having a lost-motion connection with said shoulders of said valve spool, a solenoid having a movable core resiliently connected to said valve spool through a plunger and a compression spring housed within the latter, said solenoid arranged when energized to exert a force tending to take up the slack in the lost-motion connection causing said valve spool to move positively with said rocker arm, an external compression spring connected to said valve spool so as to exert an opposing but weaker force than that of said solenoid, a dog bar rigidly attached to said actuator and having a plurality of spaced-apart tripping dogs for positively positioning said rocker arm and also said valve spool when said solenoid is energized, and a limit switch operable by said rocker arm in one of its positions to deenergize said solenoid and thereby permit said external compression spring to position said valve spool independently of said rocker arm.

3. In a hydraulic power unit of the character set forth, the combination comprising a valve spool axially shiftable into a plurality of positions, and means including a rockable abutment member having a lost-motion connection with said valve spool, means for shifting said rockable abutment member into a lesser number of positions than said valve spool, and biasing means for taking up the slack of said lost-motion connection in either direction, thereby permitting said rockable abutment member to define all the positions of said valve spool.

4. In a hydraulic power unit of the character set forth and including a main control valve having a ported bore, the combination comprising a valve spool slidably disposed within the bore and shiftable into a plurality of positions, a movable abutment member having a lost-motion connection with said valve spool for shifting the same into said plurality of positions as an incident to being shifted itself into a lesser number of positions, and opposed but coacting biasing means for taking up the slack of said lost-motion connection in either direction.

5. In a hydraulic power unit of the character set forth, the combination comprising a main control valve having a ported cylindrical bore, a valve spool slidably disposed within the bore and axially shiftable into a plurality of control positions, a rocker arm associated with said valve spool and having a lost-motion connection therewith, releasable means for taking up in one direction the slack in the lost-motion connection between said rocker arm and said spool, and resilient means arranged to act upon said spool in opposition to said releasable means for taking up the slack in the opposite direction, said latter means becoming effective to take up such slack upon the release of said releasable means in response to the position of said rocker arm.

6. In a hydraulic power unit of the character set forth, the combination comprising a main control valve having a ported cylindrical bore, a valve spool slidably disposed within the bore and axially shiftable into a plurality of control positions, spaced-apart abutments in said valve spool adjacent one end thereof, a rocker arm associated with said valve spool and having a lost-motion connection with the latter between said abutments, releasable means including a solenoid for taking up in one direction the slack in said lost-motion connection, and a compression spring mounted for engagement with said spool and arranged to act thereupon in opposition to said releasable means for taking up the slack in the opposite direction, said compression spring becoming effective to take up such slack upon the release of said releasable means in response to the position of said rocker arm.

7. In a hydraulic power unit for a machine tool, the combination comprising an axially slidable valve plunger, a spring yieldably urging said plunger toward one extremity of its path of axial movement, means including a movable abutment member having a lost-motion connection with said plunger for positioning the latter axially in either direction, means for shifting said abutment to a series of successive positions axially of said plunger and releasably holding the same in each of such positions, and means including a solenoid for compressing said spring

and retaining said plunger against said abutment at the extreme of lost motion opposite that toward which said spring tends to urge said valve plunger.

8. In a hydraulic power unit for a machine tool, the combination comprising a main control valve having an axially shiftable valve plunger, axially spaced-apart shoulders in said valve plunger adjacent one end thereof, a rocker arm associated with said valve plunger and having an abutment projecting between said shoulders to produce a lost-motion connection with the latter, a dog bar for moving said rocker arm through a series of successive angular positions with respect to the axis of said valve plunger, a solenoid having a movable core connected to said valve plunger by means of a first compression spring housed therein and adapted when energized to exert a resilent thrust upon said lost-motion connection eliminating the slack in one direction, a second compression spring mounted for engagement with said valve plunger and arranged to act thereupon in opposition to the thrust of said solenoid for taking up the slack in said lost-motion connection in the opposite direction, and switch means operable as an incident to the positioning of said rocker arm to deenergize said solenoid, said second compression spring becoming effective to take up slack upon the deenergizing of said solenoid by the positioning of said rocker arm.

9. In a hydraulic power unit for a machine tool, the combination comprising a main control valve having an axially shiftable valve plunger, axially spaced-apart shoulders in said valve plunger adjacent one end thereof, a rocker arm associated with said valve plunger and having a transverse pin projecting between said shoulders to produce a lost-motion connection with the latter, a movable bar having a plurality of spaced-apart tripping dogs arranged for moving said rocker arm through a series of successive angular positions with respect to the axis of said valve plunger, friction means associated with said rocker arm for maintaining the latter in any given one of said series of successive angular positions until moved into the next one of said series by any one of said tripping dogs, a solenoid having a movable core connected to said valve plunger by means of a tension rod and a first compression spring housed therein, said solenoid being adapted when energized to exert a resilient thrust in one direction upon said lost-motion connection thereby bringing one of said shoulders into abutment with said transverse pin, a second compression spring mounted for engagement with said valve plunger for exerting a yieldable thrust tending to urge the opposite one of said shoulders into abutment with said transverse pin, and switch means mounted adjacent said rocker arm and operable as an incident to the positioning of the latter to deenergize said solenoid and thereby render effective said second compression spring.

10. In a hydraulic control system of the type described, an actuator, a control valve for defining an operating cycle for said actuator in response to being moved through a series of successive steps, a movable abutment having a lost-motion connection with said valve and shiftable into a plurality of axial positions with respect to said valve, electromagnetic means for rendering said movable abutment effective to directly position said valve in certain of said successive steps, and means responsive to the positioning of said movable abutment for rendering said abutment effective to indirectly position said valve in the remaining of said successive steps.

11. In a hydraulic control system of the type described, an actuator, a control valve having a cylindrical bore, an axially shiftable plunger slidably disposed within said bore for defining an operating cycle for said actuator in response to being moved through a series of successive steps including the steps of rapid traverse forward, first feed forward, second feed forward, rapid traverse return, and stop, a rocker arm having a lost-motion connection with said plunger and shiftable into a plurality of axial positions with respect thereto, a solenoid adapted when energized to yieldably urge said plunger into abutment with said rocker arm rendering the latter effective to directly position said plunger in the steps of rapid traverse forward, first feed forward, and second feed forward, spring means associated with said plunger and adapted when said solenoid is deenergized to render said rocker arm effective to directly position said plunger in the stop step, and switch means responsive to the positioning of said rocker arm for rendering said rocker arm effective to indirectly position said plunger in the rapid traverse return step.

12. In a hydraulic power system of the class set forth, the combination comprising an actuator, a control valve for defining an operating cycle for said actuator and having a plunger axially shiftable into a plurality of positions including a rapid traverse return position, spring means associated with said plunger and adapted to bias said plunger toward said rapid traverse return position, electromagnetic means associated with said plunger and adapted when energized to overcome the bias of said spring means, and a pressure-actuated switch in series with said electromagnetic means, said switch being adapted when the pressure in said system exceeds a predetermined value to interrupt the circuit of said electromagnetic means thereby permitting the bias of said spring means to move said valve plunger into said rapid traverse return position.

13. In a hydraulic power system of the class set forth, the combination comprising an actuator, a control valve for defining an operating cycle for said actuator and having a plunger axially shiftable into a plurality of positions including a rapid traverse return position, a compression spring associated with said plunger and adapted to bias the latter toward said rapid traverse return position, a solenoid associated with said plunger and adapted when energized to overcome the bias of said compression spring, a control circuit for said solenoid and a pressure-actuated switch in series with said control circuit and adapted when the pressure in said system exceeds a predetermined value to interrupt said control circuit, thereby deenergizing said solenoid and permitting the bias of said spring means to move said plunger into said rapid traverse return position.

14. In a hydraulic power unit of the class described, the combination comprising a main control valve having an axially shiftable valve plunger, electromagnetic means for yieldably applying a force to said plunger, spring means for yieldably applying a force to said plunger in opposition to the force of said electromagnetic means, a reciprocatory tripping bar, a rockable abutment member movable intermittently by said reciprocatory tripping bar and disposed between said bar and said valve plunger, the position of said abutment member serving to govern the sequen-

tial application to said valve plunger of various combinations of forces directly by said electromagnetic and said spring means and indirectly by said tripping bar.

15. In a hydraulic power unit of the class described, the combination comprising a main control valve having an axially shiftable valve spool, solenoid means for yieldably applying a force to said valve spool, spring means for yieldably applying a force to said spool in opposition to the force of said solenoid means, a slidable dog bar, a rocker arm movable only by said dog bar for intermittently applying a positive force to said spool to position the latter, said rocker arm having a frictional mounting adapted to exert a neutralizing force on said valve spool in opposition to the resultant of the forces of said solenoid and said spring means, the position of said rocker arm serving to govern the sequential application to said valve spool of various combinations of forces directly by said solenoid means and said spring means and indirectly by said tripping bar.

16. In a hydraulic power system having a main control valve of the character set forth, the combination comprising a housing having a ported cylindrical bore, a valve plunger slidably disposed within the bore and shiftable into a plurality of control positions, a movable abutment having a lost-motion connection with said valve plunger for shifting the same into said control positions, and releasable means for moving said plunger to take up the slack in said lost-motion connection, said means when actuated to take up such slack being adapted to strike said plunger a hammer blow to free the latter if stuck.

17. In a hydraulic power system having a main control valve of the character set forth, the combination comprising a housing having a ported cylindrical bore, a valve plunger slidably disposed within the bore and shiftable into a plurality of control positions, a movable abutment having a lost-motion connection with said valve plunger for shifting the same into said control positions, releasable means including a solenoid for imparting motion to said plunger through a limited distance and in either of two directions in order to take up the slack in said lost-motion connection, and abutment means carried by said releasable means for striking said plunger a hammer blow to free the latter if stuck.

18. In a hydraulic power system having a main control valve including a valve plunger slidably housed within a ported cylindrical bore and shiftable axially into a plurality of control positions, the combination comprising a tension rod slidably disposed within an axial bore in said valve plunger, an abutment rigidly attached to said tension rod adjacent one end thereof, a compression spring mounted on said tension rod between said abutment and one end of the bore in said plunger, an annular abutment within said plunger adjacent the opposite end of the bore thereof and engageable by said abutment on said tension rod, a head portion unitary with said tension rod and defining coplanar shoulders thereon, an external annular face at the extremity of said valve plunger away from said annular abutment and adapted for engagement with said coplanar shoulders on said tension rod, and a solenoid including a movable core coupled to said tension arm and adapted upon being energized to fully compress said compression spring thereby bringing said abutment on said tension arm sharply into engagement with said annular abutment of said valve plunger so as to free the latter for movement in one direction if stuck, said compression spring being adapted upon the deenergizing of said solenoid to produce an impact between said coplanar shoulders and said external annular face of said valve plunger so as to free the latter for movement in the opposite direction if stuck.

19. A hydraulic control system for machine tools and the like comprising, in combination, an actuator, a pump, a sump, fluid conduits leading from the actuator, the pump and the sump respectively, a main control valve connected with said fluid conduits for delivering fluid to and receiving fluid from said actuator, a valve spool slidably disposed within a ported bore in said main control valve for directing the movements of said actuator in response to movement of said valve spool through a series of predetermined axial positions, a rocker arm having a lost-motion connection with said valve spool, said rocker arm being movable into a lesser number of positions than said valve spool, a tripping bar movable with said actuator and adapted to position said rocker arm, a first means for taking up the slack of said lost-motion connection in one direction to permit the definition by said rocker arm of certain ones of said axial positions of said valve spool, said first means being rendered effective or ineffective in response to the position of said rocker arm, and a second means adapted, upon the rendering ineffective of said first means, to take up the slack of said lost-motion connection in the opposite direction and thus permit the definition by said rocker arm of the remainder of said axial positions of said valve spool.

20. A hydraulic control system for machine tools and the like comprising, in combination, an actuator, a pump, a sump, pressure and exhaust conduits leading from the pump and the sump respectively, conduits leading from said actuator, a main control valve connected with said pressure and exhaust conduits and said conduits leading from said actuator for delivering fluid to and receiving fluid from said actuator, a valve spool slidably disposed within a ported bore in said main control valve for effecting a sequence of movements of said actuator in response to movement of said spool through a predetermined sequence of axial positions, a rocker arm having a lost-motion connection with said valve spool, said rocker arm being movable into a lesser number of positions than said valve spool, a tripping bar rigidly attached to said actuator and adapted to position said rocker arm, releasable means for taking up the slack of said lost-motion connection in one direction to permit the definition by said rocker arm of certain ones of said axial positions of said valve spool, said releasable means being susceptible of actuation by said rocker arm, and resilient means for taking up the slack of said lost-motion connection in the opposite direction to permit the definition by said rocker arm of the remainder of said axial positions of said valve spool.

21. A hydraulic control system for machine tools and the like comprising, in combination, an actuator, a variable delivery pump, a sump, pressure and exhaust conduits leading from the pump and the sump respectively, conduits leading from said actuator, a main control valve connected with said pressure and exhaust conduits and said conduits leading from said actuator for delivering fluid to and receiving fluid from said actuator, a reciprocable valve spool disposed within a ported bore in said main control valve, said valve spool

being shiftable into a plurality of positions for defining a stepped sequence of actuator movements including the steps of rapid traverse forward, first feed forward, second feed forward, rapid traverse return, and stop, a rocker arm adapted to define said valve spool positions but movable itself into a lesser number of positions, a tripping bar rigidly attached to said actuator for moving said rocker arm, a lost-motion connection between said rocker arm and said valve spool, electromagnetic means adapted to exert a force in one direction tending to take up the slack of said lost-motion connection, and spring means effective upon deenergizing of said electromagnetic means to exert a force tending to take up said slack in the opposite direction.

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