Nov. 21, 1950 M. A. MATHYS 2,531,340

HYDRAULIC MECHANISM

Filed Jan. 8, 1944

6 Sheets-Sheet 1

Fig. 1

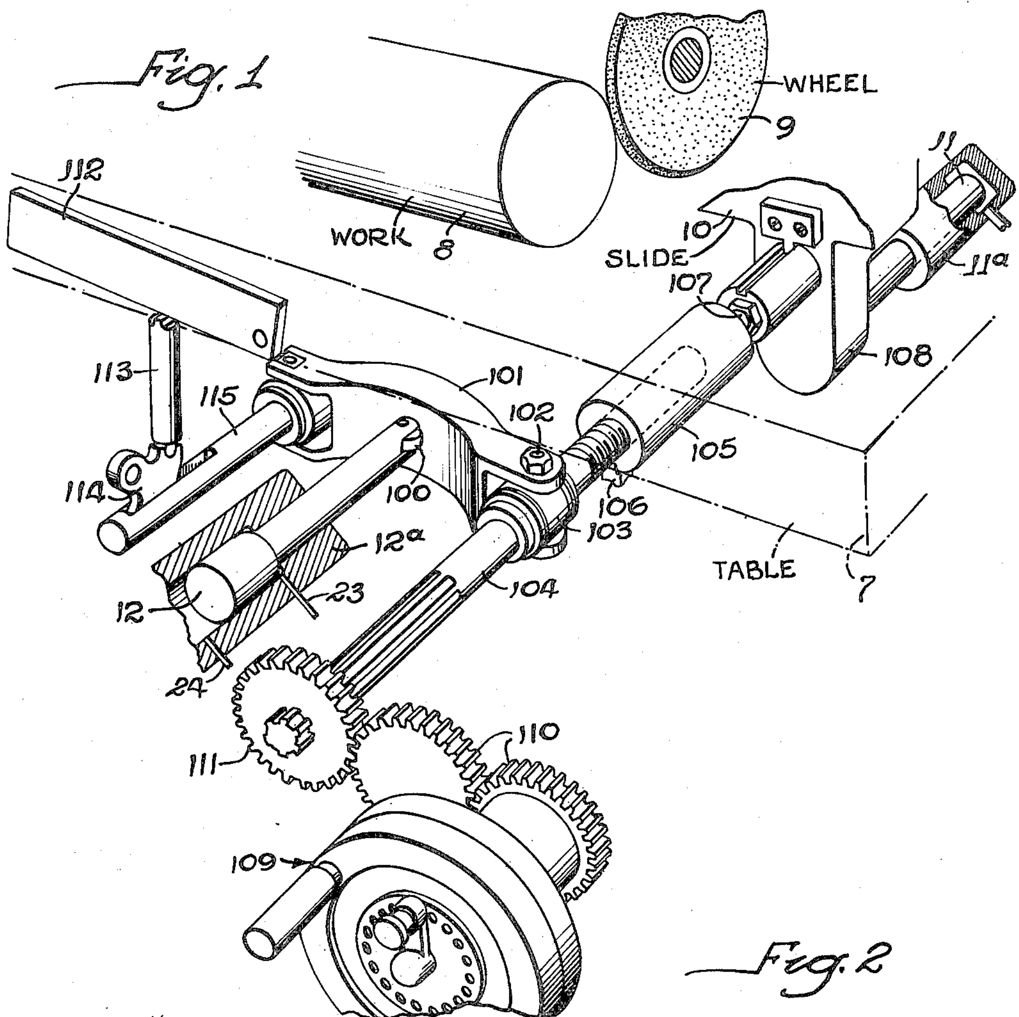
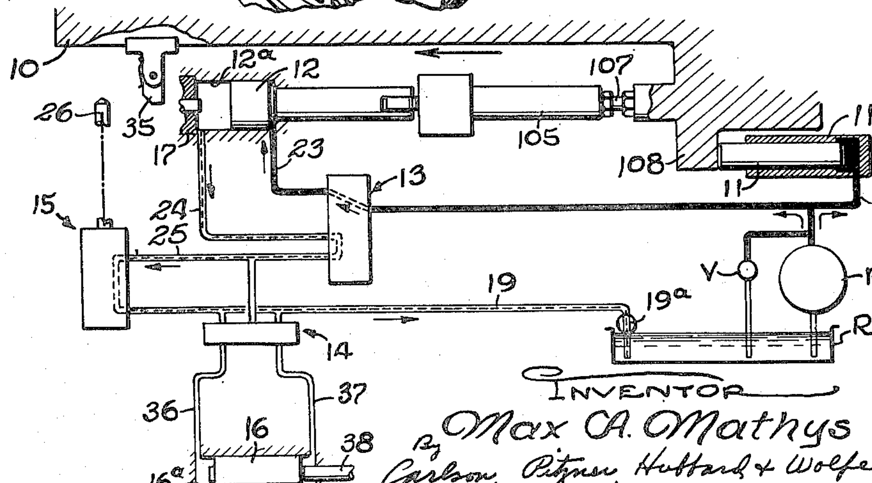


Fig. 2



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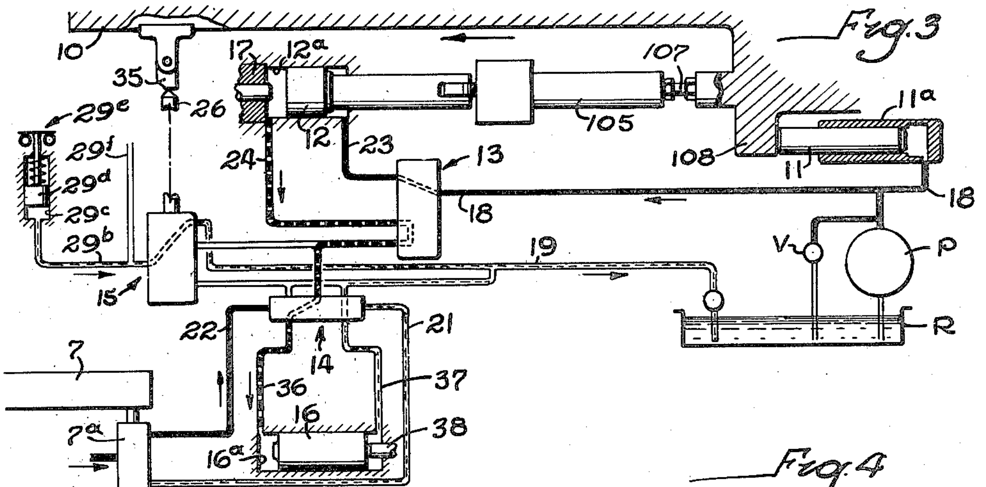

Fig. 3

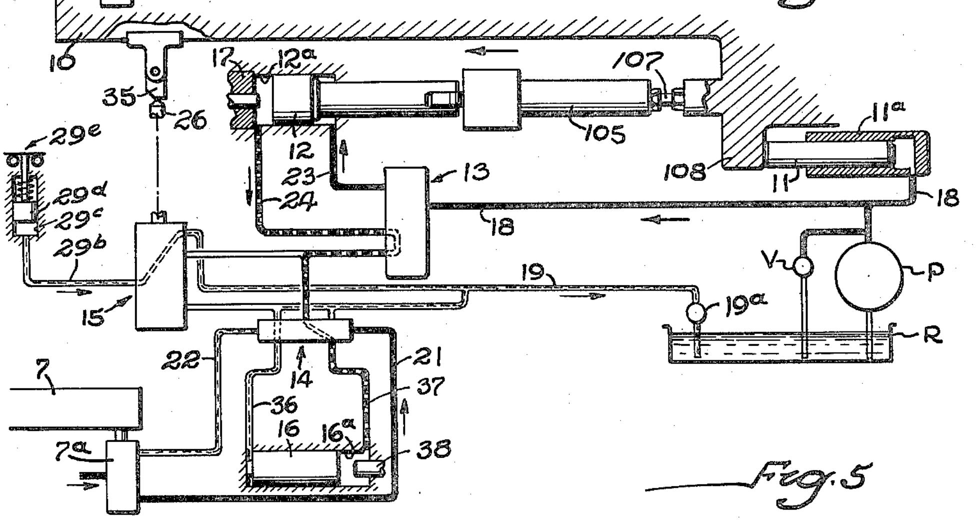

Fig. 4

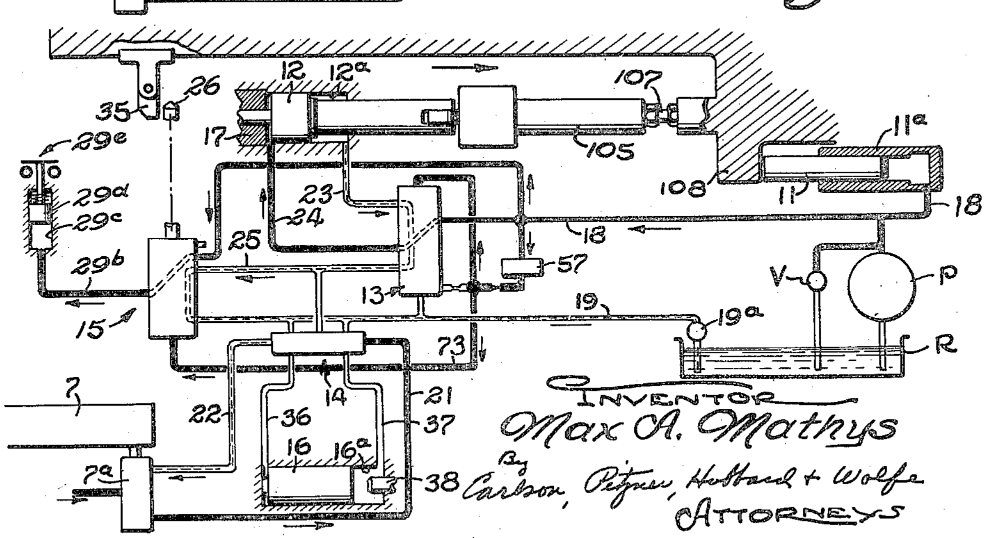

Fig. 5

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Fig. 6

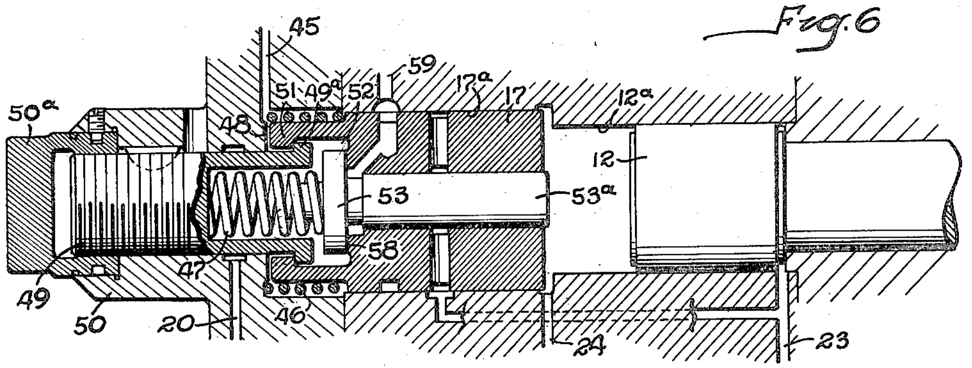


Fig. 7

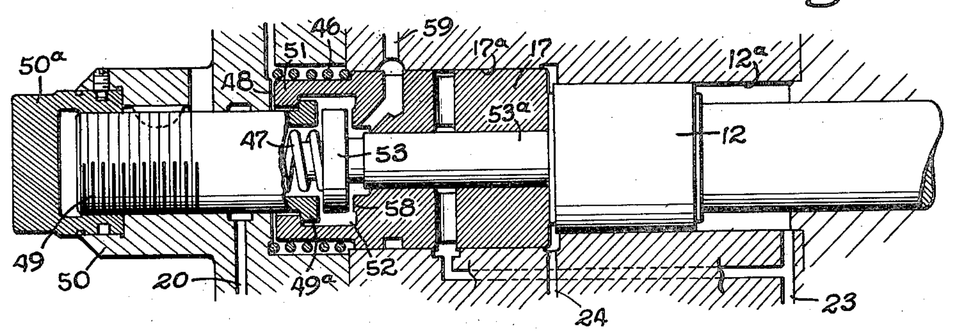
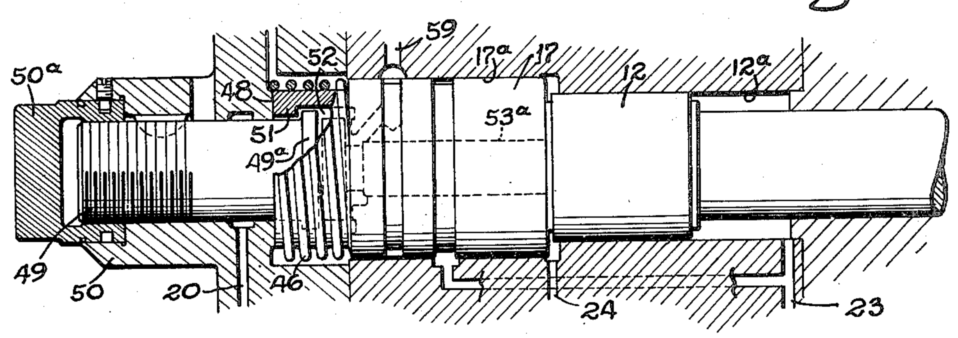


Fig. 8



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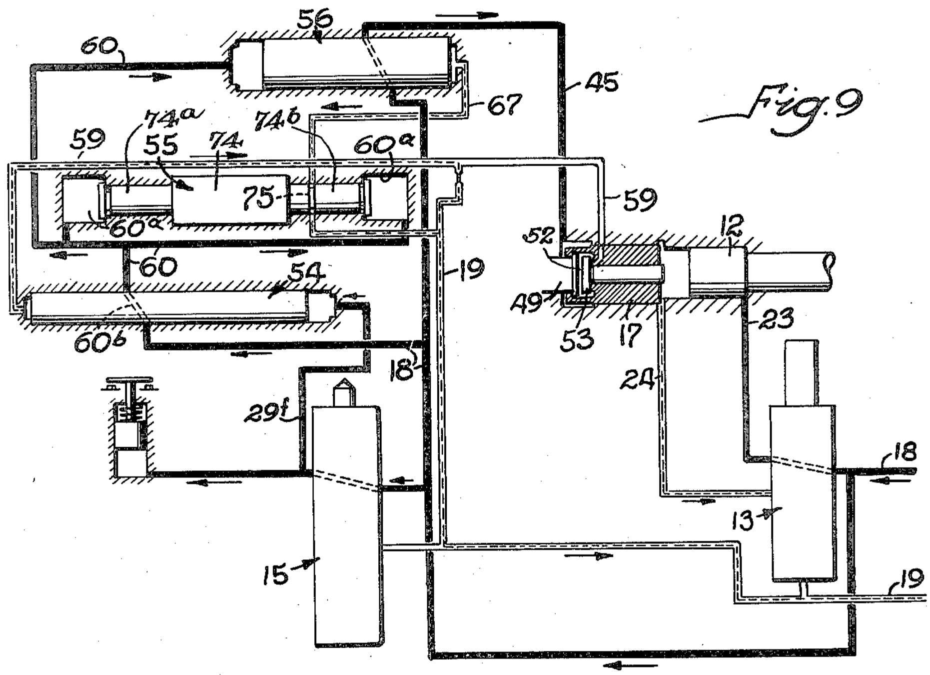

Fig. 9

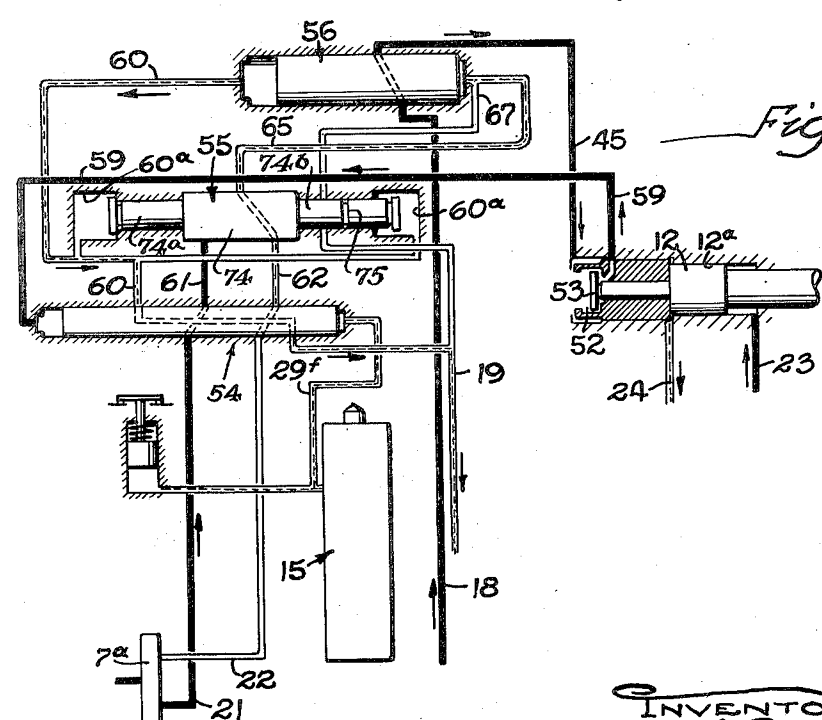

Fig. 10

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Fig. 11

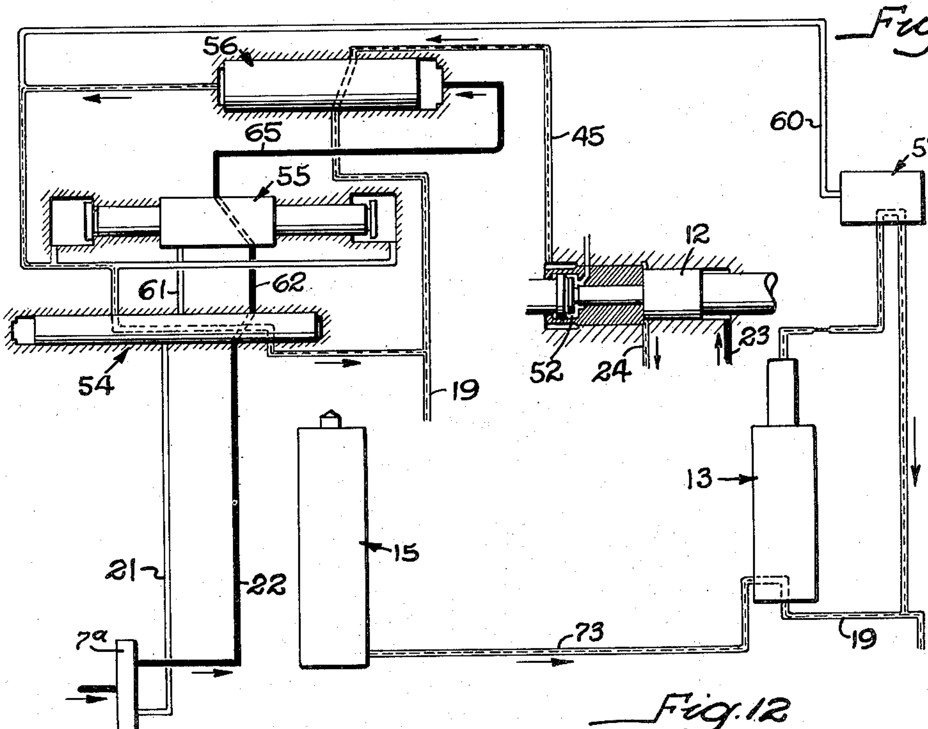
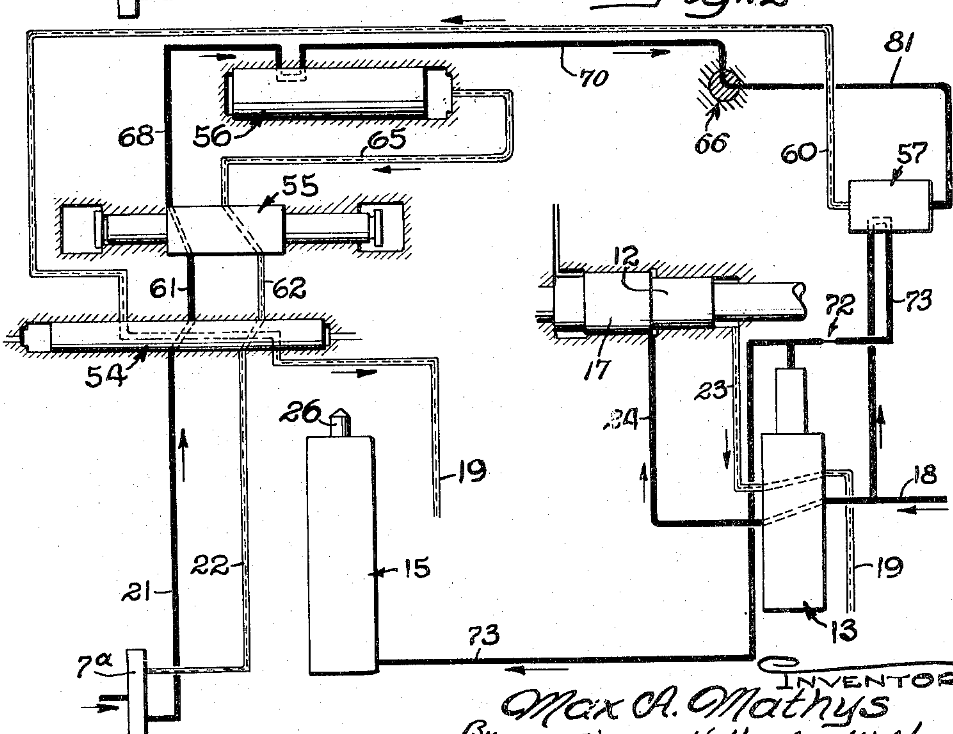


Fig. 12



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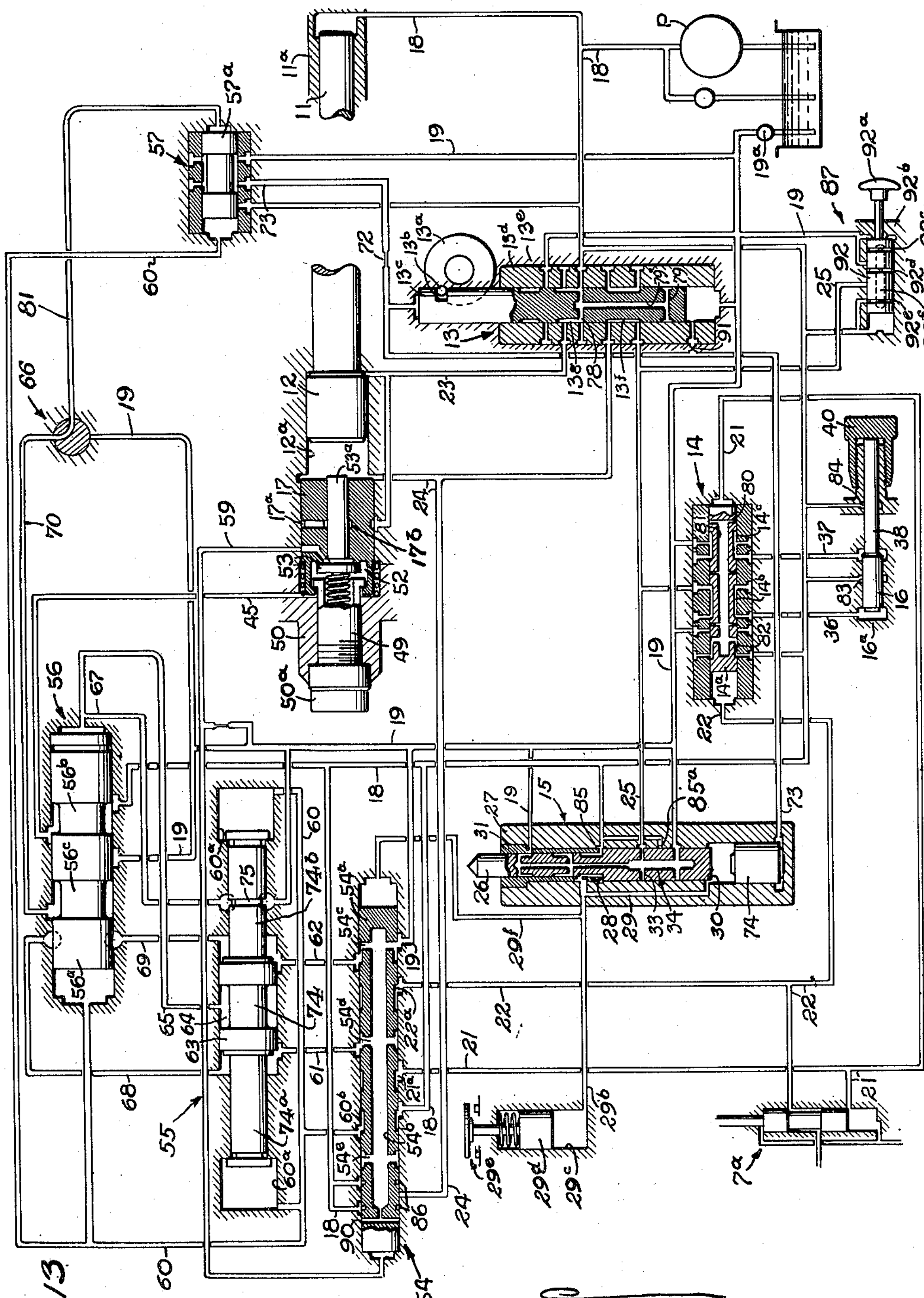

Fig. 13

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UNITED STATES PATENT OFFICE

2,531,340

HYDRAULIC MECHANISM

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Application January 8, 1944, Serial No. 517,541

18 Claims. (Cl. 121—45)

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The present invention pertains to a novel hydraulic mechanism, and more particularly to one which is especially suited for effecting, in the course of its cycle of operation, an accurately controlled step-by-step motion of a driven member, such as may be employed, for example, in accomplishing the infeed of the wheel slide of a traverse grinder.

The general aim of the present invention is to provide a hydraulic mechanism or actuating system of the general type indicated which is fully automatic and which is characterized not only by its precision of operation but also by its versatility of adjustment to accommodate different operating requirements.

More particularly, one object is to provide such a system in which a novel arrangement is provided for adjustably and precisely determining the limits of the last step in a step-by-step advance, thereby especially accommodating the same for use in a machine where such last step is used in making a finish pass or cut in bringing a work piece exactly to size.

Another object is to provide in a system employing withdrawal of metered increments of fluid from a piston and cylinder type actuator in effecting step-by-step advance, a novel arrangement for effecting an automatic reversal of the actuator after completion of the step-by-step advance.

Another object is to provide in a system employing withdrawal from an actuator of metered increments of fluid to effect the advance of such actuator, a novel arrangement for safeguarding the metering-out circuit against leakage.

Another object is to provide in a hydraulic system, a novel arrangement for insuring the precision stoppage of an actuator in a final preselectable position therefor and which arrangement operates with the same fidelity in duplicating the final position of the actuator at the termination of successive approaches therefor, irrespective of whether the actuator approaches such final position by an automatically controlled step-by-step motion or by a manually controlled operation.

Still another object is to provide in a system embodying means for effecting step-by-step advance by a fluid actuator, a novel arrangement for interrupting such advance and subsequently restoring the actuator to the position which it occupied at the time of interruption.

A further object is to provide a novel fluid actuator arrangement for reciprocating a member through a fixed stroke from an adjustably variable starting point, such arrangement being adapted for employment either with or without step-by-step advance of the member in at least a portion of a cycle of reciprocation therefor.

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Further objects and advantages of the invention will become apparent as the following description proceeds, taken in connection with the accompanying drawings in which:

Figure 1 is a generally schematic perspective view of the major elements of a traverse grinding machine having applied to its wheel slide a hydraulic actuating system embodying the present invention.

Figs. 2 to 5 are simplified hydraulic diagrams of the actuating system, Fig. 2 indicating the flow of fluid during rapid approach of the slide, Figs. 3 and 4 indicating the flow of fluid during two successive steps in the step-by-step forward feed movement of the slide, and Fig. 5 indicating the flow of fluid during the rapid return motion of the slide.

Figs. 6, 7 and 8 are successive stop motion views of the stop mechanism in the end of the actuating cylinder which serves to determine adjustably the length of the last step of advance of the actuator, the parts being shown partially in longitudinal section.

Figs. 9 to 12 are simplified hydraulic diagrams of the control portion of the hydraulic circuit, illustrating the successive conditions of fluid flow in the same incident, respectively, to rapid approach, completion of the next to the last step and last step of the step-by-step advance of the actuator, and to rapid return movement of the same.

Fig. 13 is a complete hydraulic diagram of the system, illustrating in section the preferred form of the various valves.

While the invention is susceptible of various modifications and alternative constructions, I have shown in the drawings and will herein describe in detail the preferred embodiment, but it is to be understood that I do not thereby intend to limit the invention to the specific form disclosed, but intend to cover all modifications and alternative constructions falling within the spirit and scope of the invention as expressed in the appended claims.

Referring more particularly to the drawings, in Fig. 1 has been shown generally schematically the major elements of a traverse grinding machine, such a machine being simply representative of the type of installation in which hydraulic systems embodying the present invention are especially suited for use. Such major elements include a table 7 adapted to carry and reciprocate axially a revolving work piece 8 which is to be ground. On a wheel slide 10 is a revoluble grinding wheel 9 engageable with the work 8. The herein disclosed hydraulic system is adapted to effect the so-called infeed of the wheel slide, in a direction transaxially of the work.

In brief, the preferred controls for the wheel slide 10 are such that upon starting, an automatic cycle is executed which consists of a rapid approach of the slide 10 to working position, thereafter successive steps of advance for the slide alternated with passes of the work by the reciprocating table 7, and a rapid return of the slide to starting position after it has completed its step-by-step advance to a predetermined depth of cut.

In the illustrated embodiment, movement of the slide 10 is effected by the coaction of a first fluid type actuator, consisting of a cylinder 12*a* and differential piston 12, with a second fluid actuator, or more specifically, a fluid plunger device, the latter consisting of a cylinder 11*a* and plunger 11. Both the cylinders 11*a* and 12*a* are stationarily mounted, as, for example, in the base of the machine. The slide 10 is in effect forced constantly against the piston 12 by the plunger 11. To advance the slide 10, fluid is withdrawn from the cylinder 12*a* at the forward face of the piston so that the piston 12 acts as a retreating abutment for limiting the advance of the slide under the constant urge of the plunger 11. To cause the slide to return, the application of pressure fluid to the cylinder 12*a* is reversed, the plunger 11 being overpowered. The length of travel for the slide 10 is thus determined by the fixed stroke of the actuator piston 12, but the starting point can be adjustably varied by adjusting the length of the coupling between the actuator 12, 12*a* and slide, the plunger 11 serving throughout such range of adjustment to retain the slide and actuator engaged.

As to the particular coupling set-up shown in Fig. 1, the actuator piston 12 has at its outer end an abutment roller 100 bearing against the curved face of a transverse link 101. The ends of the link 101 are yoke-shaped, the righthand end being pivoted as indicated at 102 on a sleeve 103 which is journaled upon a shaft 104, the sleeve being restrained against axial movement with respect to the shaft. The shaft itself is threaded within a nut 105, the latter being held against rotation by a stationary spline key 106 entering a longitudinal slot in the nut. Bearing against the outer end face of the nut 105 is an adjustable abutment 107 fixed to the slide 10. By revolving the shaft 104 the same is screwed into or out of the nut 105, thus adjusting the position of the slide 10 relative to its actuator 12, 12*a*. In this way the location of the slide 10 relative to the work 8 may be adjusted at will even though its actuator 12, 12*a* has a stroke of fixed length.

The plunger 11 serves to urge the slide 10 constantly in a forward direction to retain the slide abutment 107 constantly in engagement with the end of the nut 105. For this purpose the plunger 11 is arranged to bear against the face of a bracket 108 rigid with the slide 10 and depending therefrom.

The usual size wheel 109 serves to adjust the shaft 104 within the nut 105. This wheel is connected through gears 110 with a gear 111 splined on the shaft 104 so that turning of the wheel 109 adjusts the slide 10 either forwardly or rearwardly and thereby determines the size of the finished work piece to be ground.

To accommodate the actuator system for grinding a tapered surface on the work 8, a sine bar 112 may be adjustably fixed to the front of the table 7. Riding along the inclined lower edge of the sine bar 112 is a vertically movable plunger 113 which operates through a bell crank 114 and axially shiftable rod 115 to swing the link 101, the rod 115 being pivoted to the lefthand end of the latter link. With such an arrangement, as the table advances the sine bar 112 shifts the member 113, thereby causing the link 101 to fulcrum about the roller 100 so that the slide 10 is advanced in timed relation with the axis of the work to produce a tapered surface on the latter.

Power circuits of the system

The preferred hydraulic circuit embodying the actuator 12, 12*a* and plunger 11 is shown in full in Fig. 13, but its intricacies will best be understood after a preliminary consideration of certain simplified or partial showings of the circuit in order that the general plan of operation may be clear. For that purpose the fluid flow for the actuator or power circuits proper has been diagrammed in Figs. 2 to 5 for successive cycle steps, and the fluid flow for the control circuits of the system has been similarly diagrammed in Figs. 9 to 12. Reference may be first made to Fig. 2. As there shown, fluid such as oil is supplied under pressure from a suitable constant pressure source such, for example, as a constant delivery pump P equipped with a conventional spring loaded working pressure relief valve V. The pump draws fluid from a reservoir R and delivers the same under pressure, the relief valve serving to return such excess fluid to the reservoir as may be required to maintain the pump delivery pressure substantially constant at a value determined by the adjustment of the relief valve. Pressure fluid from the pump P is delivered through pressure line 18 directly and constantly to the outer end of the plunger 11 and also to a starting and reversing valve 13. The latter is a two-position valve having forward and reverse positions and when in its forward position it dispatches fluid from the pressure line 18 through line 23 to the cylinder 12*a* for advancing the piston 12. In Figs. 2 to 5 and 9 to 12 the heavy black shading indicates fluid under pressure directly from the pump while a dotted line in the conduit indicates fluid at an exhaust pressure. In Figs. 3 and 4 the broken heavy lines in certain of the conduits indicate fluid trapped in the metering circuit and which is retained under pressure substantially equal to the pump discharge pressure for a purpose which will later appear.

To institute a cycle of operation, rapid advance of the slide 10 is initiated by shifting the valve 13 to its forward position (indicated in Fig. 2), whereupon it dispatches pressure fluid from line 18 through line 23 to the actuator cylinder 12*a* and causes fluid to be exhausted from the opposite end of such cylinder through line 24 and thence through line 25 and a feed control valve 15 back to the reservoir R through exhaust line 19. If desired, a conventional spring loaded back pressure valve 19*a* may be interposed in the exhaust line 19.

The feed control valve 15 is of the snap acting type (see Fig. 13 for detail). It has a plunger 26 slidable within a casing 27, the outer portion of the plunger being reduced in diameter to present a cross-section one-half that of the inner end. Pressure from the pressure line 18 is applied to the face 28 of the shoulder on the plunger between its large and small portion and through a line 29 to the inner end 30 of the plunger. Since the area at 30 is twice as large as that of the shoulder 28, the plunger is yieldably thrust outward against a bushing 31 fixed in the valve casing 27. In such position of the valve, the line 25 is connected through an internal bore 34 and cross passages 33 in the plunger to exhaust line 19 for the free exhaust of fluid from the line 25 to effect rapid approach of the slide as noted above. A separate plunger 74 in the lower end of the valve casing serves to thrust the main plunger 26 outward upon application of pressure fluid in a manner hereinafter described to set the plunger 26 in its projected position. Once the plunger 26 is so projected it will remain, due to the differential pressure action on it, until mechanically thrust in a short distance, whereupon fluid resistance to such thrust is relieved so that the plunger can continue its retraction without impediment.

At the end of the rapid advance of the slide 10 its motion is changed to a slow step-by-step advance or feed, timed with the reciprocations of the work table 7. Such change is instituted by engagement of a spring-urged dog 35 on the slide 10 with the projecting outer end of the plunger 26 of the feed valve 15 (Fig. 3). Shifting of this valve plunger 26 causes the valve 15 to interrupt the exhaust of fluid from the cylinder 12*a* through line 19 and to direct the fluid from such cylinder through an arrangement which meters the fluid out of the cylinder in predetermined small increments. Each increment of fluid so metered out permits the piston 12 to advance a corresponding fixed distance.

When the dog 35 thrusts the valve plunger 26 inward, the exhaust line 19 is connected through the passage 29 to the rear end 30 of the plunger so that the latter can complete its inward movement without impediment. Such inward shift of the plunger 26 cuts off the connection to exhaust from the line 25 and blocks the latter (see Fig. 13). A separate sliding plunger 74 limits the inward movement of the main valve plunger 26.

The mechanism for metering out increments of fluid from the cylinder 12*a* consists of a plunger 16 slidable within a cylinder 16*a* provided with an adjustable stop pin 38 at one end thereof for limiting the path of travel of the plunger. A micrometer head 40 (see Fig. 13) on the stop pin 38 serves to adjust its position. Cooperating with the metering plunger 16 and connected thereto by lines 36, 37 is a reversing valve 14 which serves to direct fluid to alternate ends of the plunger 16 from the outlet of the actuator cylinder 12*a* and in each case to exhaust the opposite end of cylinder 16*a* to line 19. The valve 14 is fluid operated in timed relation with the table 7 in such manner that the valve 14 is actuated upon each stroke of table movement. For this purpose a reversing pilot valve 7*a* (Fig. 3) is actuated by the table 7 to direct pressure fluid through alternate ones of pilot lines or conductors 21, 22 and exhaust the other, thereby alternating the position of the plunger of valve 14.

Since the shifting of the valve 15 to institute feed blocks the line 25, as heretofore described, so that fluid from the cylinder 12*a* can no longer go to the exhaust line 19, it is, instead, forced through the valve 14 into one end or the other of the metering cylinder 16*a* as determined by the position of the valve 14. Under the conditions shown in Fig. 3, fluid is forced out of the actuator cylinder 12*a* through the line 24 and is directed by the valve 14 into the lefthand end of the metering cylinder 16*a*, thus forcing the metering plunger 16 to the right to the limit position determined by the adjustable stop 38. Simultaneously, valve 14 exhausts the opposite end of the metering cylinder 16*a* by connecting it to exhaust line 19. An increment of fluid is thus displaced from the actuator cylinder 12*a* which is equal to the volume of the space vacated by the plunger 16 in its shift, and the actuator piston 12 together with the slide 10 is advanced a corresponding fixed distance. Upon the completion of the shift of the metering plunger 16 to the right, the slide 10 comes to rest, since no further fluid can be displaced from the actuator cylinder 12*a*.

Upon the completion of the next stroke of the work supporting table 7 the pilot valve 7*a* is actuated thereby to reverse the pressure conditions in the pilot lines 21, 22 (Fig. 4) so that pressure is applied through line 21 and fluid exhausted through line 22, thus shifting the reversing valve 14. Such shift of the valve 14 directs fluid into the righthand end of the metering cylinder 16*a* and connects the lefthand end to exhaust through line 19. Thereupon the metering plunger 16 is displaced to the left, a corresponding amount of fluid being received from the actuator cylinder 12*a* so that the slide 10 can advance another step. In this same general manner the metering plunger 16 is shuttled back and forth with one stroke of the plunger upon the completion of each stroke of the work supporting table 7, so that the slide 10 is advanced one step after each stroke of the table.

Provision is also made for instituting drive of the work spindle as an incident to shift of the valve 15 at the completion of rapid approach. For that purpose the valve 15 connects line 29*b* to exhaust line 19, thereby exhausting fluid from a cylinder 29*c* so that a spring-urged plunger 29*d* moves downward to close its associated switch contacts 29*e*. Closure of these contacts may be used to complete the energizing circuit for the electric drive motor (not shown) which revolves the work piece 8.

Upon completion of the step-by-step feeding advance of the slide 10, its motion is reversed through the medium of an automatic control arrangement hereinafter described, and it is automatically returned to starting position. During such rapid return movement of the slide (see Fig. 5) the valve 13 is in its reverse position, while the valve 15 is restored to the position which it occupied during the rapid approach movement illustrated in Fig. 2. The control circuits for operating the valves 13 and 15 to effect reversal are hereinafter detailed in connection with Figs. 9 to 13. Incidentally, it will be noted that the spring-urged dog 35 rides freely over the plunger 26 during its return movement and consequently does not actuate the latter during such return. For the rapid return movement in question, pressure fluid from the line 18 (Fig. 5) is dispatched by the valve 13 to the outer or lefthand side of the piston 12 through line 24 and fluid is exhausted from the righthand end of the actuator cylinder through line 23, valve 13, line 25, valve 15, and thence through exhaust line 19 back to the reservoir. Although the application of pressure fluid to the plunger 11 is continued during such return movement, the greater area of the lefthand face of the differential piston 12, as compared to the area of the plunger 11, causes the piston to overpower the plunger so that the slide 10 is returned rapidly to its initial position. By way of analysis it should be noted that during both forward feed and reverse motion, the plunger 11 and larger face of the differential piston 12 and subjected to substantially equal pressures, and the direction of piston motion is determined by either applying pressure fluid to, or exhausting fluid from, the smaller face of the piston.

At the end of the return movement of the slide it comes to rest upon abutment of the piston 12 against the righthand end of the cylinder 12*a*. During the return of the slide the spindle control switch contacts 29*e* are opened by pressure fluid supplied from the pressure line 18 through the valve 15 and line 29*b*.

Readily adjustable step-by-step motion of the slide 10 is afforded in the feed portion of the cycle, as will be plain from the foregoing. Each step of such advance is positively controlled by the amount of fluid metered out of the actuator cylinder 12*a* by the plunger 16. As will be evident from an inspection of Figs. 3 and 4, the outlet from the cylinder 12*a* is completely blocked except for such fluid as is permitted to enter the metering cylinder 16*a* by shift of the plunger 16. The piston 12 and plunger 16 are in effect joined by a column of incompressible fluid so that the step of advance permitted to the actuator piston 12 is limited to an amount proportionate to the shift of the plunger 16. Such plunger shift can be readily adjusted by changing its stroke through the use of the stop pin 38.

Control of last feed step

Special provision is made for adjustably controlling the length of the last step in the step-by-step advance of the slide 10. Such adjustment is quite independent of the length of the preceding steps which, as heretofore explained, are determined by the setting of the adjustable stop pin 38. Precise and accurate control of the length of the last step of advance is especially desirable in that the length of such step commonly determines the amount of metal which is to be removed during the last or finishing pass of the work past the tool. It is requisite not only that the length of this step should be accurately controlled, but also that the final position of the slide be determined with extreme certainty as well as exactitude since that position determines the final size of the work.

In accomplishing such control of the last step of advance of the slide, a movable stop in the form of a bushing 17 is interposed in the path of advance of the piston 12 (see Figs. 2, and 6 to 8). This stop 17 intercepts the advance of the piston at the end of the next to the last step, and for the final step the stop 17 is permitted to retreat an accurately determined distance. The stop bushing 17 is slidable axially within a bore 17*a* coaxial with and opening into the end of the cylinder 12*a* opposite the large face of the piston 12. Outward movement of the stop bushing 17 is limited by the seating of its outer end against an abutment surface 48 on the end wall of the bore 17*a*, while movement of the bushing in an opposite or inward direction toward the piston 12 is limited by engagement of projections 51 with mating projections 49*a* on a plug 49 keyed to slide endwise within a stationary housing 50. A micrometer adjusting knob 50*a* is threaded on the plug 49 and held against endwise displacement in the housing 50. By turning the adjusting knob 50*a* the plug 49 is moved endwise, thereby altering the spacing between the abutment surfaces 48 and projections 49*a* so that the length of permitted stroke for the stop bushing 17 is correspondingly adjusted.

At the outer end of the stop bushing 17 a chamber 52 is provided to which pressure fluid is supplied in a manner hereinafter described so that the pressure of such fluid, augmented by a spring 46, normally urges the stop bushing toward the piston 12 whose motion it intercepts (Fig. 6). Slidable axially within the bushing 17 is the stem 53*a* of a disk-shaped valve element 53 which seats on an annular seat 58 formed on the outer face of the bushing. The valve element is urged into seated position by a compression spring 47 interposed between the bottom of a bore in the plug 49 and the outer face of the valve disk 53. When the valve element 53 is unseated, pressure fluid passes by the same and out a passage 59 from the chamber 52 to condition the control circuits hereinafter described so that upon the next stroke of the work table 7 the chamber 52 will be exhausted, thereby permitting the actuator piston 12 to force the bushing 17 to the left and against the abutment face 48 for the last step of advance of the actuator piston.

As the actuator piston approaches the stop bushing 17 the parts occupy the relative positions shown in Fig. 6, the bushing being yieldably urged toward the actuator piston by the spring 46 and the pressure fluid within the chamber 52. In the course of the next to the last step of advance of the actuator piston 12, the latter engages and forces inward the valve plunger 53*a*, thereby unseating the valve element 53. To accommodate such action the end of the plunger 53*a* is dimensioned to extend a few thousandths of an inch beyond the face of the bushing 17. Accordingly, at the end of the next to the last step of advance of the actuator piston the parts come to rest in the position shown in Fig. 7. In the latter position the stop bushing 17 is still thrust to the right, intercepting the piston 12 in the desired position for institution of the last step for the latter. Even though the valve 53 is open so that pressure fluid is permitted to pass into the line 59 for conditioning the circuit for the last step of advance, still the pressure has not been relieved in chamber 52 so the stop bushing is still held by fluid pressure against mating projection 49*a*. As hereinafter detailed, the completion of the next stroke of the work table 7 results in the exhaust of fluid from the chamber 52, whereupon the actuator piston 12 is freed to thrust the stop bushing 17 to the left against the abutment faces 48 for the last step of advance of the actuator piston. In this way the rigid abutment 48 positively determines the final position of the actuator piston 12, and a simple change in the setting of the adjustment knob 50*a* determines the length of such last step of piston advance.

Automatic reversal

Opening of the valve 53 as described above in the course of the next to the last step of advance for the actuator piston not only conditions the circuit for retreat of the stop bushing 17 as indicated, but in addition sets up the circuits for institution of rapid return of the slide after the last stroke of the work table has been completed. It will be evident that one factor which complicates the problem is that at least one stroke of the work table 7 must follow each step of infeed of the wheel slide 10. Consequently, the circuits must be arranged so that a predetermined advance of the slide 10 does not merely result in some change in the slide movement, as, for example, in its reversal, but must instead preliminarily condition the circuits for such change in movement in response to completion of a succeeding stroke of the work table.

The succession of changes in position of the interrelated control valves which takes place preceding and as an incident to reversal of the slide 10 is shown generally diagrammatically in Figs. 9 to 12. Fig. 9 exemplifies the conditions prevailing during rapid approach movement of the slide. Fig. 10 exemplifies the circuit conditions at the completion of the next to the last step of advance, thus matching Fig. 7; while Fig. 11 shows the circuit conditions prevailing after the last step of advance of the slide, but during the succeeding stroke of the work table, thus matching Fig. 8. Finally, Fig. 12 shows the fluid flow in the control circuit after the last stroke of the work table.

Included in the portion of the circuit to be considered in connection with reversal and the delayed action control thereof are, in addition to the reversing valve 13 and feed valve 15 already identified, four pilot valves 54, 55, 56 (Figs. 9 to 12) and 57 (Figs. 11 and 12). Each of the additional valves noted is fluid operated, all being of the sliding plunger type, and of them the valves 54, 56 and 57 are two-position valves, while the valve 55 is of the three-position type, having a normally centered plunger.

In brief, the arrangement is such that the pilot valve 54 is shifted in response to completion of the next to the last infeed step of the actuator piston 12. Thereafter the pilot valves 55, 56 coact to delay withdrawal of the stop bushing 17 until after one further stroke of the work table 7 and to delay operation of the valve 57 (which controls the reversing valve 13) until after two further strokes of the work table. The pilot valves are coordinated so that the action shall be the same irrespective of which of the intermittently reversed lines 21, 22 happens to be connected to pressure at the time when the next to the last step of infeed is completed.

Attention may now be given to the successive conditions of fluid flow diagrammed in Figs. 9 to 12. During rapid approach of the actuator piston 12 (Fig. 9) the feed valve 15 dispatches fluid from the pressure line 18 through a branch conduit 29*f* to the valve 54 so that the latter is shifted to the left. With the valve 54 in the latter position, pressure fluid from the line 18 passes through an annular peripheral groove 60*b* in valve 54 (see Fig. 13) and thence through the branched conduits 60 to the chambers 60*a* at opposite ends of the valve 55, thereby centering the latter. Pressure fluid in the conduit 60 also shifts the valve 56 to the right, the right end of the latter valve being connected to exhaust line 19 through a conduit 67 and a transverse passage 75 in the valve 55. When valve 56 is thus shifted to the right it directs pressure fluid from the line 18 through line 45 to the chamber 53 located at the outer face of the stop bushing 17 as heretofore described. The pressure fluid thus supplied to the chamber 52 urges the stop bushing 17 toward the actuator piston 12 in position to intercept the latter at the completion of the next to the last step of the step-by-step infeed which is to follow. By connections which will later appear (Fig. 13) pressure fluid in line 60 also shifts a valve 57 to the right.

Upon the dog operation of the feed valve 15 to change from rapid approach to step-by-step infeed movement in the manner heretofore described, such shift of the valve 15 connects the line 29*f* to the exhaust line 19 and disconnects it from the pressure line 18. At that point in the cycle both ends of the valve 54 are thus connected to exhaust so that it remains in the position to which it was shifted by the previous application of pressure shown in Fig. 9.

At the end of the next to the last step of infeed for the actuator piston 12, the valve 53 is, as heretofore explained, opened by contact of the piston 12 with the end of the valve stem 53*a*. Such opening of the valve 53 directs pressure fluid from the line 45 and chamber 52 into a line 59 leading to one end of the pilot valve 54. The opposite end of such valve 54 having been previously connected through line 29*f* to exhaust, the valve 54 is shifted to the right. The shift in valve 54 results in connecting line 60 to exhaust which allows shifting of valves 55, 56 and 57. Pressure fluid is dispatched from one of the table-controlled pair of lines 21, 22, which happens at that moment to be connected to pressure, to one of the ends of the central portion of the valve 55 through the corresponding one of lines 61, 62 and connection of another part of such central portion of the valve 55 to exhaust through the other one of lines 61, 62 and the other of the pair of lines 21, 22. The central portion of the valve 55 is constructed in a manner such that application of pressure through the line 21 and exhaust through line 22 causes it to shift to the right and, conversely, application of pressure through line 22 and exhaust through line 21 shifts the valve 55 to the left. But in either case, whether the valve 55 is shifted right or left, it connects to the exhausted one of the pair of lines 21, 22, a line 65 leading from the right end of the final pilot valve 56.

To accommodate the desired action the valve 55 (Fig. 13) comprises a central valve plunger 74 and two shoulder type side plungers 74*a* and 74*b* slidable endwise within a suitable casing. The central valve plunger 74 has two spaced collars or lands 63 and is disposed within a central chamber 64 in the valve housing. The lines 61, 62 open into the central chamber 64 at points spaced apart slightly more than the spacing of the collars 63, whereas the line 65 to the right end of the valve 56 opens into the center of the chamber of valve 55 and lines 68, 69 lead from spaced points in such chamber.

The cooperating pilot valve 54 has an axial bore 54*b* in its plunger 54*a*, such bore being always connected to exhaust line 19 by transverse passage 54*c* in the valve plunger. Such transverse passage 54*c* also registers with line 62, as does a second transverse passage 54*d* with line 61, so that both lines 61, 62 are exhausted. Upon shifting of the valve 54 to the right, such connection of the lines 61, 62 to exhaust line 19 is cut off, and lines 21, 22 are connected, respectively, to lines 61, 62 through peripheral annular grooves 21*a*, 22*a*.

With the valves 54, 55 so constructed, when the valve 54 shifts to the right as described, lines 21, 22 are respectively connected to lines 61, 62, and chambers 60*a* at the ends of valve 55 are connected to exhaust line 19 through line 60, a transverse bore 54*e* in the plunger of valve

54, and the axial bore 54b in the latter. Accordingly, should pressure be available in line 21 the plunger of valve 55 is shifted to the right, while if pressure is available in line 22 such plunger is shifted to the left. In either case the one of the lines having pressure in it is connected to the corresponding one of lines 68, 69 while the other or exhausted one of lines 21, 22 is connected to line 65.

The net effect of shifting the pilot valves 54 and 55 as just described (see Fig. 10) is, as noted, to connect the right end of the third pilot valve 56, through line 65, to the one of the lines 21, 22 which is at the moment exhausted. That does not result in any shift of the valve 56, for its right end was previously connected to exhaust through conduit 67 and an annular peripheral groove 75 (see also Fig. 13) in the plunger 74b. The latter connection is, incidentally, interrupted by the shift of the valve 55. But the newly established connection to the valve 56 does insure its actuation upon the next succeeding reversal of pressure in the lines 21, 22. Accordingly, when the table 7 completes its next stroke (i. e., for the last roughing cut in the cycle of operation of the grinding machine) the consequent reversal in pressure in the lines 21, 22 causes the valve 56 to be shifted to the left (Fig. 11).

Building up of undue pressure in the line 24 from the actuator cylinder is prevented after contact of the piston 12 with valve stem 53a by arranging the pilot valve 54, when shifted to the right, to connect line 24 to exhaust (Fig. 13). Thus a cross passage 90 in the plunger of valve 54 connects line 24 to exhaust line 19 through the central bore 54b.

The first result of shifting the pilot valve 56 to the left is to establish a connection through it from the chamber 52 to exhaust so that the stop 17 is withdrawn to permit the final infeed step of the actuator piston 12 (Fig. 11). For that purpose the line 45 leading from the chamber 52 is connected to the exhaust line 19 through an annular peripheral groove 56b in the plunger 56a of the valve 56 (see Fig. 13).

A secondary result of shifting the pilot valve 56 is to establish a connection for actuation of the reversing valve 57 upon completion of the next or final stroke of the work table 7 (Fig. 12). At the completion of such final table stroke (i. e., for the final finishing cut on the work in the cycle of the grinding machine) pressure conditions in the lines 21, 22 are again reversed (compare Figs. 11 and 12). Accordingly, if line 21 is the one of the pair put under pressure at that time, as is the condition indicated in Fig. 12, pressure fluid is supplied from line 21, through valve 54, line 61, valve 55 and line 68 and valve 56 to a line 70 leading through a valve 66 and line 81 to the righthand end of the valve 57, lines 68 and 70 being interconnected through an annular peripheral groove 56c in the plunger of the valve 56 (see Fig. 13). This shifts the plunger 57a of the valve 57 to the left, its left end being connected to exhaust through line 60 and valve 54. Similarly, if pressure happens to be applied to line 22 at such time, pressure fluid will also be supplied to line 70 with like result, but this time through valve 54, line 62, valve 55, and thence through line 69 and the annular groove 56c in valve 56 (Fig. 13).

Shifting of the valve 57 to the left as described causes the valve 13 to be shifted to its reversing position and the feed valve 15 to be restored to its initial or rapid position (Fig. 12). Thus valve 57 applies pressure from the pressure line 18 through the line 73 and a choke 72 to a chamber at the upper end of the plunger of the valve 13 to force the same downward. Pressure fluid is also supplied to a chamber at the lower end of the actuating piston 74 of the valve 15 to thrust the plunger 26 of this valve outward, such pressure fluid being applied through line 73 and a choke 72 as well as from the pressure passage 79 and a choke 91 after the valve 13 has shifted. The purpose of the chokes 72 and 91 is to make it possible to supply line 73 via either the valve 57 or 13. The valves 13 and 15 are thus conditioned for rapid return movement of the slide in the manner heretofore described in connection with Fig. 5.

It will thus be seen that the stop 17 is withdrawn an the slide actuator 12, 12a reversed in proper sequence, and all in carefully coordinated timed relation with the intermittent reversals of pressure and exhaust in lines 21, 22 occasioned by reciprocations of the work table 7.

Manual control

If desired, the machine may be operated by manual control of the advance of the slide 10 rather than with an automatic cycle of successive steps as described. Such operation is especially required for obtaining the initial setting on the size wheel 109, while setting the machine to a new work piece. In such manually controlled infeed of the slide the size control wheel 109 (Fig. 1) is turned a desired distance for each successive step. To condition the machine for such manually controlled operation, two valves are utilized in addition to certain of those heretofore noted; namely, an automatic feed disconnect valve 66 and a size pick-up valve 87 (Fig. 13).

The disconnect valve 66 is a simple manually operable two position rotary valve. When in its automatic feed position shown in Fig. 13, it connects line 70 to line 81 for supply of pressure fluid for shifting the valve 57 at the appropriate point in the cycle, as heretofore described. Turning the valve 66 to its alternate position, however, connects the line 81 to the exhaust line 19 and interrupts the connection to line 70 so that shifting of the valve 57 leftward is disabled.

The pickup valve 87 is a two position plunger type valve having a plunger 92 equipped with an operating handle 92a and slidable in a casing 92b. Pressure fluid from the line 18 constantly urges the plunger 92 outward. Annular peripheral grooves 92c, 92d and 92e in the plunger are interconnected by an axial bore 92f in the plunger, grooves 92c and 92d having the same spacing as ports from the lines 25 and 19.

Assuming the slide 10 is withdrawn its full distance, the machine is conditioned for manual operation by turning the valve 66 to connect line 81 to exhaust. The valve 13 is then shifted manually to its forward position, whereupon the slide advances rapidly, just as in the automatic cycle, until the valve 15 is tripped. Then the operator pushes in valve 87, connecting line 25 to exhaust so that the advance of the piston 12 continues without interruption until it comes up against the stop bushing 17. The direction of travel for the table 7 is reversed manually. This will exhaust chamber 52 causing piston 12 to be shifted to the maximum forward stop position. Either desired subsequent advance of the wheel slide 10 for grinding or location of the wheel slide at a desired final stopped position may be obtained by turning the size wheel 109. If it is desired to grind automatically, after the size setting is thus established, valves 66 and 13 are shifted, dog 35, knob 50a and pin 38 are adjusted, and the machine is ready for the automatic cycle.

The valve 87 may also be used in effecting rapid restoration of the wheel slide 10 to a previous operating position after interruption of the automatic cycle. Should it be necessary to withdraw the grinding wheel at any time during the automatic infeed, the operator need only shift the valve 13 to its reverse position by means of the rotary operating device 13a (Fig. 13) having thereon an eccentric pin 13b received in a slot in the side of the plunger 13c. Then to restore the wheel to its previous position, the slide need not be advanced step-by-step in alternation with strokes of the table. Instead, the valve 13 is shifted manually to its advance position, and when the rapid advance of the slide has been terminated by tripping of the valve 15, the operator pushes in the valve 87. As previously noted, this connects line 25 to exhaust so that the slide piston 12 advances rapidly. When the operator observes that the grinding wheel is approached close to its previous position, he releases the valve 87. The latter is thrust back out by the biasing pressure from line 18, thereby interrupting the connection of the line 25 to exhaust, and the system resumes its normal automatic step-by-step advance of the wheel slide.

Leakage prevention in metering circuit

Step-by-step advance of the slide 10 is, as heretofore described in connection with Figs. 3 and 4, accomplished by metering out successive increments of fluid from the actuator cylinder 12a via the lines 24, 25 and metering plunger 16. In the event that each step of advance is but a few thousandths of an inch, as is the case in a grinding machine, even a small volume of fluid leaked from the metering line or circuit would result in a large percentage error in the length of the step. Even in instances where longer steps are used, precision results depend upon the prevention of loss of fluid from the metering line.

Examination of the detail of the path of fluid through the metering circuit (see Fig. 13) reveals sliding surfaces, along which leakage might take place, at a number of points, viz.: along the piston 12, the stop bushing 17, the plunger 13c of the valve 13, the plunger 26 of the valve 15, plunger 14a of the valve 14, the plunger 54a of the pilot valve 54, and along the metering plunger 16 itself. Even with precision fitting which should be used for these sliding parts, some leakage would tend to occur.

The general plan followed in preventing leakage from the metering circuit along the various sliding parts noted is to maintain the metering circuit pressure substantially equal to that in some available source of pressure fluid, here the pressure line 18, and apply pressure from the latter at the sliding parts in question in a manner to counteract any tendency of fluid to flow out of the metering circuit between the sliding surfaces and thus to prevent leakage therealong.

To maintain the pressure in the metering circuit equal to that in the pressure line 18, the active faces of the piston 12 and active face of the plunger 11 are suitably dimensioned. This is relatively simple in a grinding machine, diamond boring machine, or the like since the load or resistance to the tool offered by the work is very uniform as well as being substantially negligible in comparison with the positioning pressure from the plunger 11. Accordingly, the area of the plunger 11 is but slightly less than the differential in area between the two faces of the piston 12, the plunger having been shown as exaggeratedly small in the drawings for the sake of clarity. In machines having heavy work resistance, the same can be compensated by dimensioning piston 12 and plunger 11 or by making some exterior provision for fluid pressure control. It will be understood, of course, that the reference herein to maintenance of pressure in line 24 equal to that in line 18 has to do only with the feed portion of the cycle (Figs. 2 and 3), for in rapid approach the line 24 is exhausted (Fig. 2).

With the pressure in lines 18 and 24 equalized as described, the fluid pressures, that is, the forces per unit area, on opposite faces of the piston 12 are balanced so there is no tendency for fluid to leak past it from either and to the other (see Fig. 13). Pressure from line 18, via line 23, is also applied to annular interconnected grooves 17a and 17b in the stop bushing 17 so that leakage along it is likewise prevented. In this way leakage of fluid trapped in the cylinder 12a on the advancing side of the piston 12 is prevented and all of it forced to emerge from the cylinder into the line 24.

Following down the line 24 (Fig. 13) it will be seen that in the valve 13 the valve ports are formed in a stationary sleeve 13d fixed in the casing 13e and receiving the sliding plunger 13c. When the plunger 13c is in its upper or advance position shown, a peripheral groove 13f therein connects lines 24 and 25. In the plunger 13c on opposite sides of the annular passage 13f are transverse passages, interconnected by a longitudinal bore 79a and to which pressure fluid is supplied from the line 18 through the peripheral annular groove 13g which also supplies fluid to the line 23. The length of the groove 13g insures registry of the same with the port of line 18 in either position of the valve plunger so that pressure is always applied in 78, 79 from the line 18. Such provision of fluid at 78, 79 equalized in pressure with that in lines 24, 25 effectually prevents leakage from the latter along the valve plunger 13c.

Similar precautions against leakage are taken in the valves 14, 15 and 54 (see Fig. 13). Thus in the valve 14 the valve plunger 14a is provided with two peripheral annular grooves 14b and 14c which serve to connect the lines 36, 37 leading from opposite ends of the metering plunger cylinder 16a, respectively, to lines 25 and 19 in alternate positions of the valve plunger. To guard against the leakage of fluid from the line 25 along the valve plunger 14a from either of the passages 14b or 14c pressure fluid from the line 18 is supplied to transverse passages 80 in the valve plunger through a longitudinal bore 81 and an annular peripheral passage 82 in the plunger. The latter passage 82 is long enough to register with the port from the line 18 in either of the alternate positions of the plunger 14a; and the transverse passages 80, leading to annular peripheral grooves, are located on respective opposite sides of each of the annular passages 14b and 14c.

In the valve 15 pressure fluid from the line 18 is supplied to opposite sides of the port from line 25 to annular grooves 85 and 85a surrounding the valve plunger on opposite sides of the port from line 25. In the pilot valve 54 pressure fluid from the line 18 is directed into ports which, when the valve plunger is in its left position shown, register with annular grooves 86 located in the valve plunger on opposite sides of the port leading from the line 24.

In the case of the metering plunger 16 and its associated adjustable stop pin 38, pressure is applied from the line 18 to annular grooves 83 and 84 surrounding respective ones of the same to prevent leakage along them.

I claim as my invention:

1. In a hydraulic system for effecting a step-by-step relative movement of a pair of members, the combination of a piston and cylinder type fluid actuator, means for withdrawing metered increments of fluid from the cylinder to effect successive steps of advance of the piston within the cylinder, means including a movable stop for intercepting the advance of the piston, means for adjustably limiting movement of said stop both toward and away from the piston in the direction of piston travel, means for urging said stop to its limit of movement toward the piston, and means responsive to completion of the advance of the piston forward into juxtaposition with said stop for freeing the stop for movement to its opposite limit position under the thrust of the piston in a subsequent and final step of advance of the latter, whereby the setting of the limit positions of the stop by said adjustable limit means determines the length of said final step of advance for the piston.

2. The combination with a piston and cylinder type actuator, and means for withdrawing determinate metered increments of fluid from the cylinder to effect successive steps of relative advance for the piston within the cylinder, of stop means positionable in response to fluid pressure for adjustably predetermining the length of only the last step in the series.

3. The combination with a piston and cylinder type actuator, and means for withdrawing metered increments of fluid from the cylinder to effect successive steps of relative advance for the piston within the cylinder, of adjustable stop means positionable in response to fluid pressure for predetermining the length of the last step in the series and positively arresting further advance of the piston upon the completion of said last step.

4. The combination of a piston and cylinder type actuator, means for supplying pressure fluid to one side of said piston and for intermittently withdrawing determinate metered increments of fluid from the cylinder at the opposite side of said piston to effect successive steps of advance for the piston relative to the cylinder, adjustable stop means positionable in response to fluid pressure for predetermining the length of the last step in the series and for positively arresting further advance of the piston upon completion of said last step, and means rendered operable upon completion of such last step of advance for automatically initiating a change in the pressure fluid connections to effect an uninterrupted supply of pressure fluid to said other side of the piston and an uninterrupted exhaust of fluid from said cylinder at said one side of the piston to thereby effect a substantially continuous relative movement of said piston and cylinder in a direction opposite to said stepped advance.

5. In a delayed action hydraulic control, the combination with a fluid operable valve, a pair of fluid conductors, and means for intermittently connecting alternate ones of said conductors to a source of pressure fluid and the remaining one to exhaust, of means including a second fluid operable valve shiftable alternatively in opposite directions from an initial mid-position upon application thereto of pressure fluid from corresponding ones of said conductors for connecting the other or exhausted one of said conductors in each case to said first valve.

6. In a delayed action hydraulic control, the combination with a fluid operable valve, a pair of fluid conductors, and means for intermittently connecting alternate ones of said conductors to a source of pressure fluid and the remaining one to exhaust, of means operable upon actuation thereof for establishing a connection to said valve from the one of said conductors which is at the moment of such actuation exhausted and for maintaining such connection throughout at least the next succeeding alternation in pressure and exhaust conditions in said conductors.

7. In a delayed action hydraulic control, the combination of a pair of fluid conductors, means for intermittently connecting alternate ones of said conductors to a source of pressure fluid and the remaining one to exhaust, a first fluid operable valve, means including a second fluid operable valve shiftable alternately in opposite directions from an initial mid-position upon application thereto of pressure fluid from corresponding ones of said conductors for connecting the other or exhausted one of said conductors in each case to said first valve, means for shifting said first valve from a first position thereof to a second position upon application thereto of pressure fluid at the next succeeding alternation of pressure and exhaust in said conductors following a shift of said second valve, and means operable in response to shift of said first valve from said first to said second position thereof for establishing a connection to said third valve from the one of said conductors which is exhausted in said next succeeding alternation of pressure and exhaust conditions, whereby pressure fluid will be supplied through the last-mentioned connection to said third valve upon the second succeeding alternation in pressure and exhaust conditions following the shift of said second valve.

8. The combination of a pair of fluid conductors, means for intermittently connecting alternate ones of said conductors to a source of pressure fluid and the remaining one to exhaust, a fluid operable valve, and control means operable upon actuation thereof for establishing a connection to said valve for the supply of pressure fluid thereto for actuating the same from one of said conductors only after two alternations in pressure and exhaust conditions in said conductors have followed such actuation.

9. In a delayed action hydraulic control, the combination of a member shiftable between alternate limit positions, means defining a fluid pressure chamber for moving said member from one of said limit positions to the other upon application of fluid pressure to such chamber, means defining a relief passage for relieving the pressure in said chamber, means including a first fluid operable valve shiftable between alternate positions for opening and closing said relief passage, a pair of fluid conductors, means for intermittently connecting alternate ones of said conductors to a source of pressure of fluid and the other to exhaust, means for shifting said first valve to its relief-closing position, and means including a second fluid operable valve shiftable in respective opposite directions upon application

thereto of pressure fluid from corresponding ones of said conductors for connecting the other of said conductors to said first valve to apply pressure fluid to the latter in a direction to shift the same to its relief-opening position upon the next succeeding reversal in exhaust and pressure conditions in said conductors.

10. In a delayed action hydraulic control, the combination of a member shiftable between alternate limit positions, means defining a fluid pressure chamber for moving said member from one limit position to the other upon application of fluid pressure to such chamber, a pair of fluid conductors, means for intermittently connecting alternate ones of said conductors to a source of pressure fluid and the other to exhaust, and means operable upon actuation thereof for relieving the pressure within said chamber in response to the next succeeding alternation in pressure and exhaust conditions in said conductors following such actuation.

11. In a delayed action hydraulic control, the combination of a fluid operable valve, a pair of fluid conductors, means for intermittently connecting alternate ones of said conductors to a source of pressure fluid and the remaining one to exhaust, means including a second fluid operable valve shiftable alternately in opposite directions from an initial mid-position upon application thereto of pressure fluid from corresponding ones of said conductors for connecting the other or exhausted one of said conductors in each case to said first valve, and a third valve shiftable between alternate positions in which it respectively connects and disconnects both of said conductors to said second valve.

12. The combination of a movable member, means including a reversible hydraulic actuator for traversing said member reversely to and fro along a predetermined path, a movable mechanical stop positionable in response to fluid pressure and interposed in said path to yieldably intercept the motion of said member in one direction along the same, means for automatically initiating a reversal of said actuator in response to interception of said member by said stop, and means manually operable at will to disable said reversing means, whereby said actuator continuously urges said member against the stop after interception of the member by the stop, and manually operable means for adjustably varying the stroke of said yieldable stop along said path.

13. In a hydraulic system, the combination of a piston and cylinder type actuator, a stop member located at one end of said cylinder in position to intercept the relative advance of the piston toward said one end of the cylinder, means including a pair of rigid abutments for defining respective limit positions of movement of said stop member axially of the cylinder, means for applying pressure fluid to the outer side of said stop member to urge the same into its limit position toward said piston, means including a valve for initiating relief of such pressure on said stop member, and means positioned for contact by said piston upon approach thereof to said stop member for actuating said valve.

14. In a hydraulic system, the combination of a piston and cylinder type actuator, a stop member located at one end of said cylinder in position to intercept the relative advance of the piston toward said one end of the cylinder, means including a pair of rigid abutments for defining respective limit positions of movement of said stop member axially of the cylinder, means for applying pressure fluid to the outer side of said stop member to urge the same into its limit position toward said piston, a valve, means positioned for contact thereof by said piston upon close approach of the piston to said stop member for opening said valve, a pair of fluid conductors, means for intermittently connecting alternate ones of said conductors to a source of fluid pressure and the other to exhaust, and means for relieving said fluid pressure on the stop member in response to the next succeeding alternation in pressure and exhaust conditions in said conductors following the opening of said valve by said piston.

15. In combination, an actuator cylinder having a piston slidable therein with a piston rod projecting from but one face of the piston, a stop bushing slidably mounted in the end of said cylinder adjacent the other face of said piston, means including a pair of rigid abutments engageable with said bushing for limiting its sliding motion axially of the cylinder, means for applying pressure fluid to the outer end of said bushing to urge the same toward said piston, a valve element having a stem slidable in said bushing and projecting from the inner end thereof in position to be contacted by said piston as the latter approaches the stop bushing, and means for initiating the relief of said fluid pressure on the stop bushing in response to actuation of said valve element by contact of the piston with said stem.

16. In a hydraulic system, the combination of a piston and cylinder actuator, means for withdrawing metered increments of fluid from the cylinder to effect a step-by-step relative advance of the piston within the cylinder, and means for maintaining during a portion of said step-by-step advance a substantial equalization of pressure on opposite sides of said piston to prevent the leakage of fluid past the same.

17. In a hydraulic system, the combination of an actuator including a piston and cylinder, a metering device including a plunger and cylinder, means defining an outlet passage from said actuator cylinder to said metering cylinder for withdrawal from the actuator cylinder of a metered increment of fluid corresponding to the displacement shift of said metering plunger upon each stroke of the latter, and means for maintaining during withdrawal of said metered increment of fluid a substantial equalization of pressures on opposite sides of said piston as well as for applying fluid to the periphery of the plunger intermediate its ends at a pressure substantially equal to that of the fluid entering said metering cylinder from said actuator cylinder.

18. In a hydraulic system, the combination of an actuator including a piston and cylinder, means defining an outlet passage from the cylinder for withdrawal of fluid therefrom on the advancing side of the piston, means for applying pressure fluid to the opposite side of said piston, metering means operable upon successive actuations thereof to effect the displacement from said cylinder and through said passage of determinate successive increments of fluid, a movable valve element interposed in said passage, and means for applying to said valve element fluid under a pressure substantially equal to that in said passage and such application being at plural

points located on opposite sides of the point of contact of fluid from said passage with said valve element.

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REFERENCES CITED

The following references are of record in the file of this patent:

UNITED STATES PATENTS

Number	Name	Date
448,277	Taylor	Mar. 17, 1891
526,930	Maxon	Oct. 2, 1894
637,461	Hartness	Nov. 21, 1899
639,744	Leavitt	Dec. 26, 1899
651,502	Fitzgerald	June 12, 1900
882,887	Hoxie	Mar. 4, 1908
1,582,468	Heald	Apr. 27, 1926
1,583,296	Laussucq	May 4, 1926
1,820,653	Ernst	Aug. 25, 1931
1,877,701	Speck	Sept. 12, 1932
1,998,003	Ernst	Apr. 16, 1935
2,127,877	Maglott	Aug. 23, 1938
2,157,707	Keel	May 9, 1939
2,361,460	Daugherty	Oct. 31, 1944
2,367,009	Davis	Jan. 9, 1945
2,368,791	Waldie	Feb. 6, 1945