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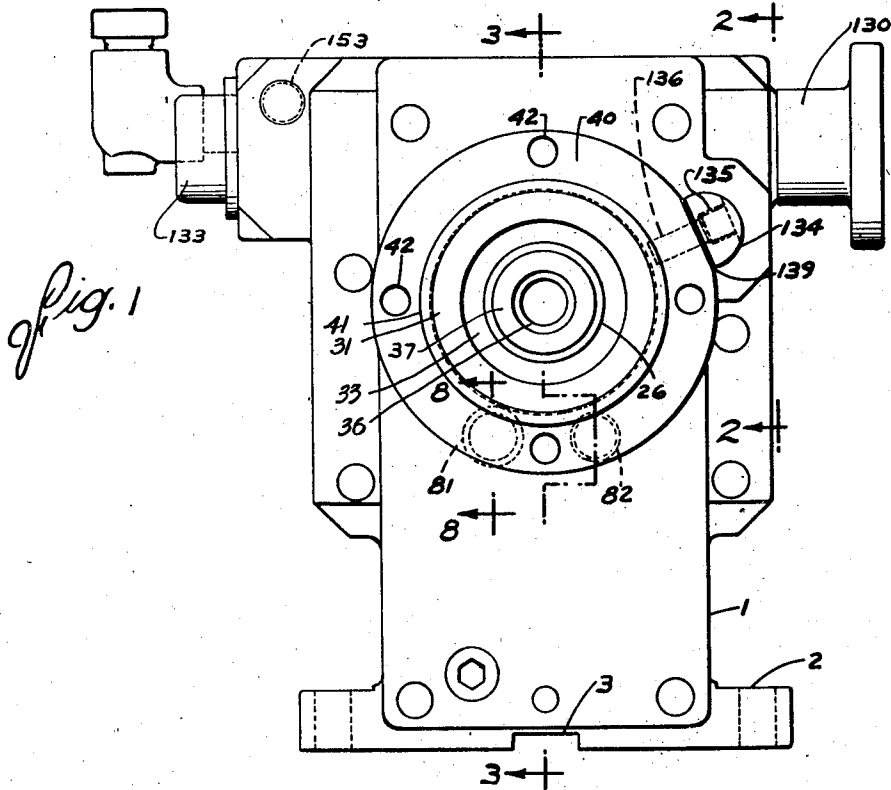
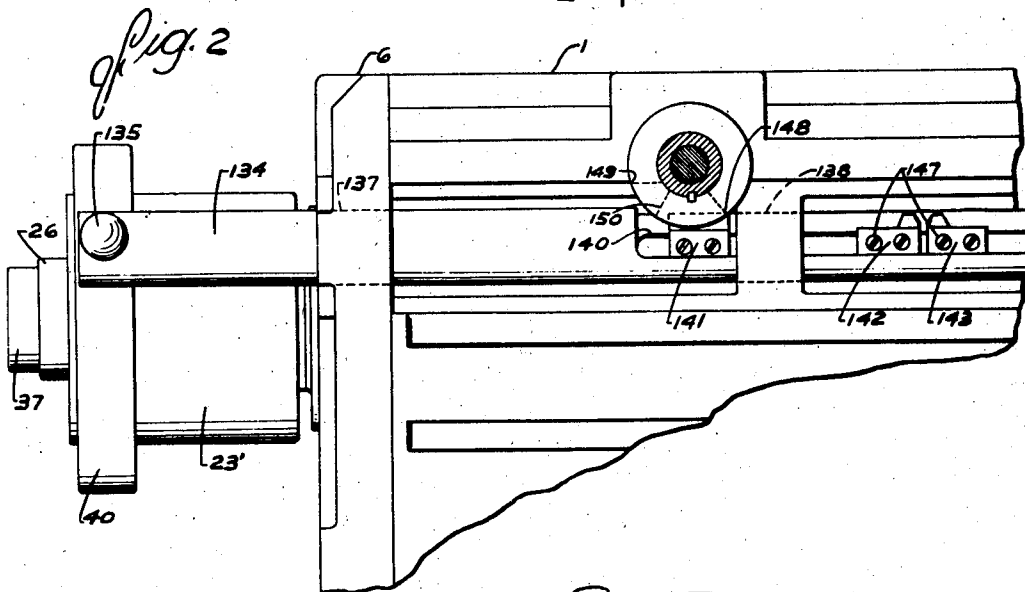
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2,416,339

HYDRAULIC POWER UNIT

Filed Feb. 17, 1941

6 Sheets-Sheet 1



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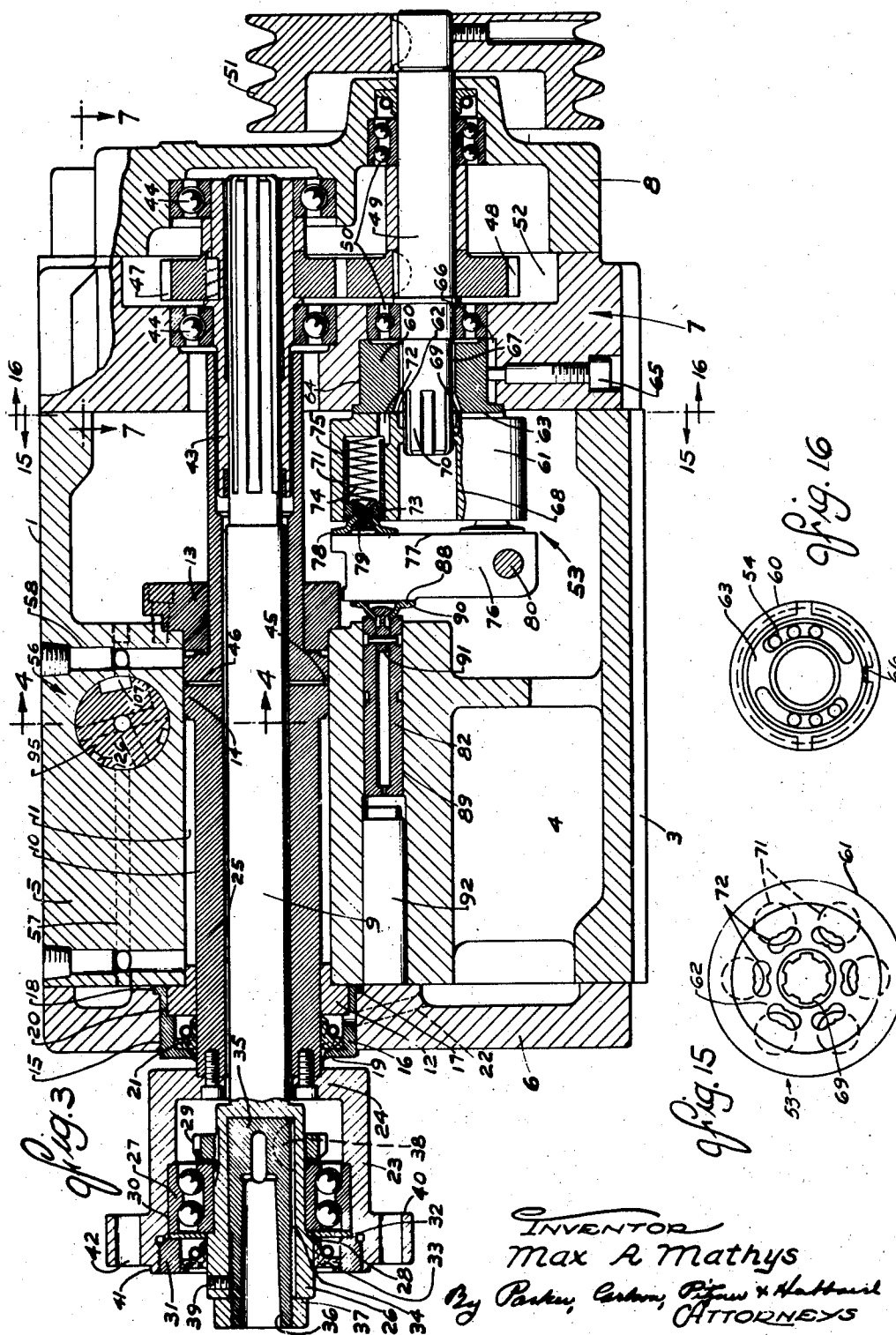
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HYDRAULIC POWER UNIT

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6 Sheets-Sheet 2



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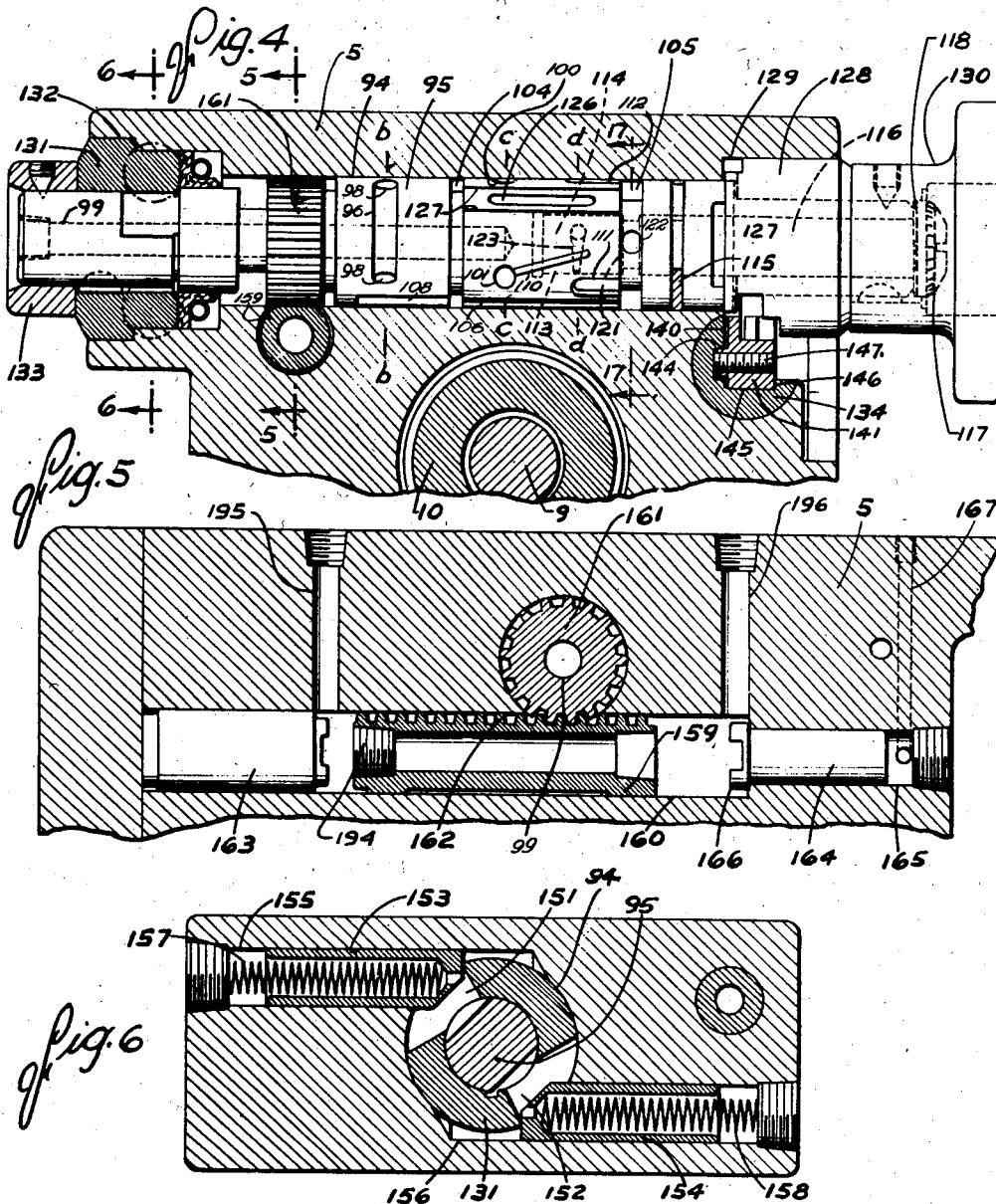
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HYDRAULIC POWER UNIT

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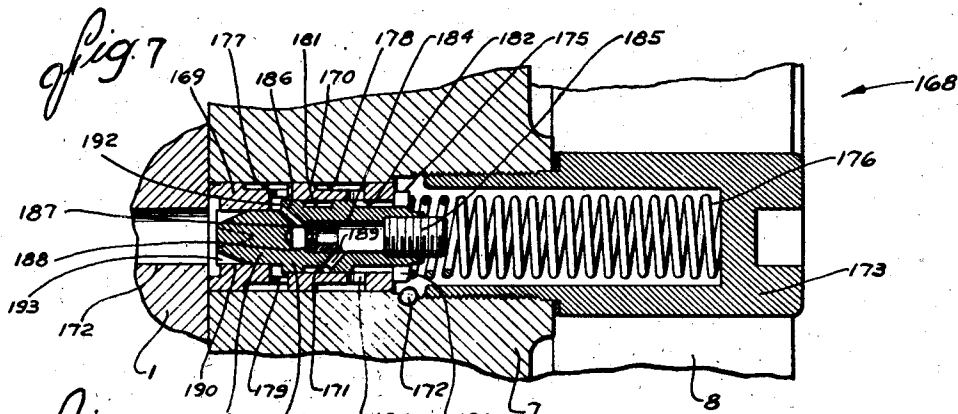
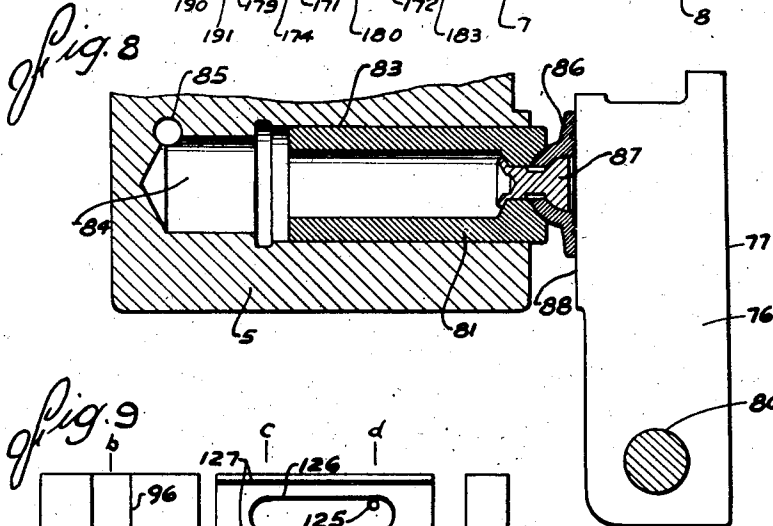
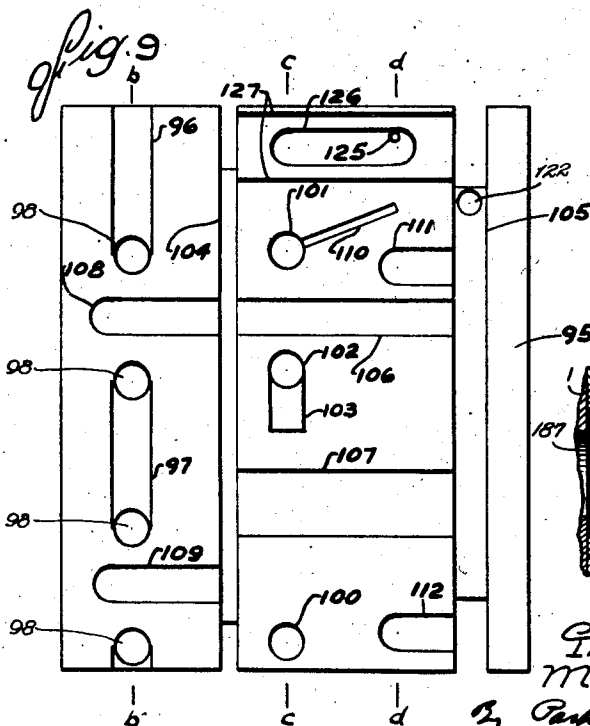
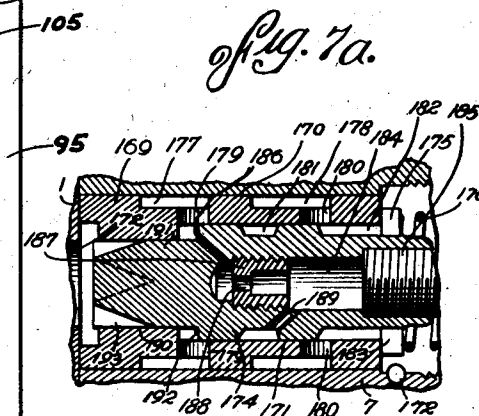
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HYDRAULIC POWER UNIT

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6 Sheets-Sheet 4



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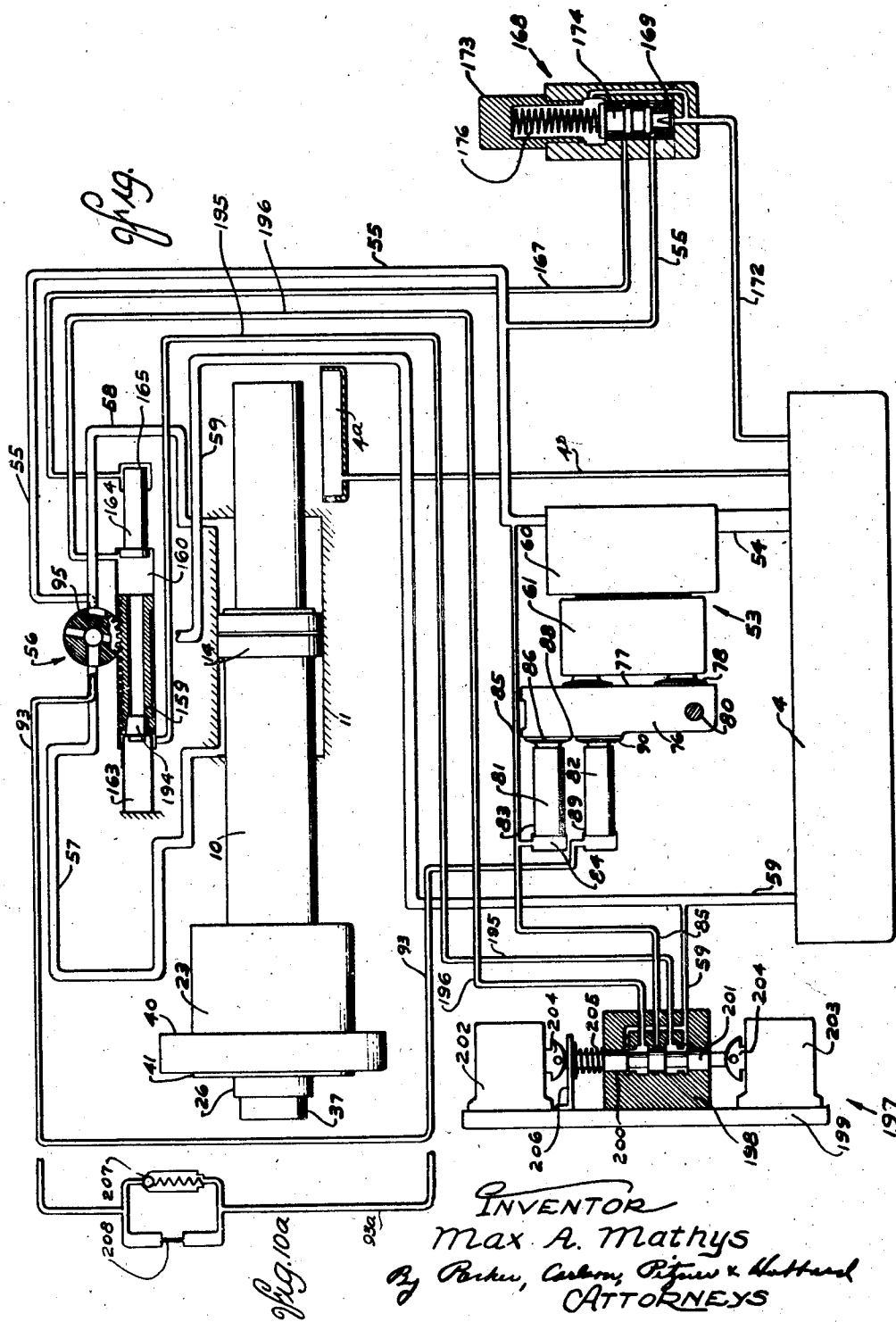
M. A. MATHYS

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HYDRAULIC POWER UNIT

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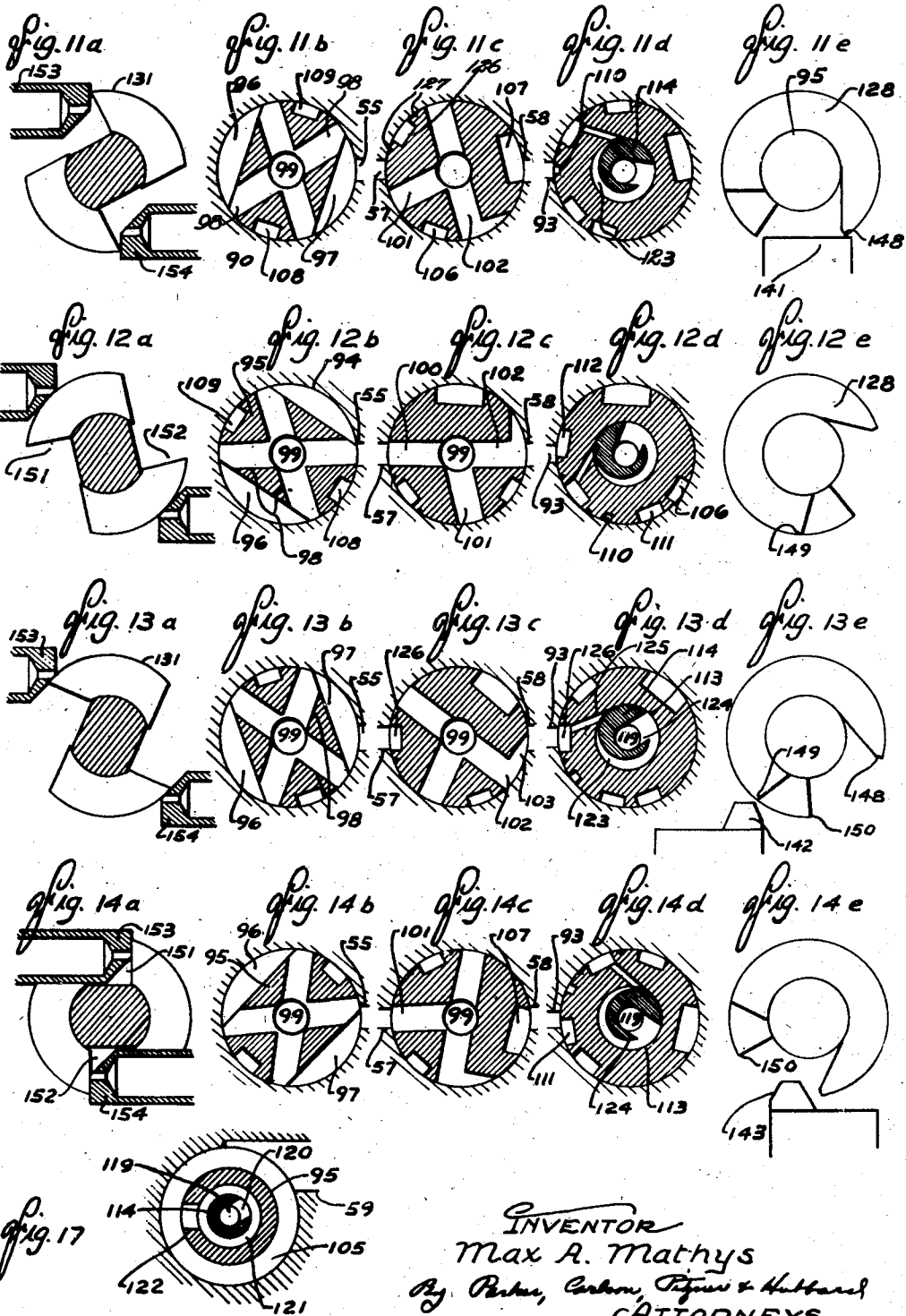
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HYDRAULIC POWER UNIT

Filed Feb. 17, 1941

6 Sheets-Sheet 6



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UNITED STATES PATENT OFFICE

2,416,339

HYDRAULIC POWER UNIT

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Application February 17, 1941, Serial No. 379,211

22 Claims. (Cl. 60—52)

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The present invention relates generally to improvements in hydraulic power units, and has particular reference to such unit having a spindle mounted for rotation and axial translation, and having self-contained means for driving the spindle and for translating the spindle automatically through a predetermined program of movements, such for example, as a cycle comprising a rapid approach, a forward feed, and a rapid return.

One of the objects of the invention is to provide a novel hydraulic power unit which is simple, compact and sturdy in construction and which is extremely powerful, rapid, efficient and silent in operation.

Another object is to provide a new and improved power unit embodying a hydraulic actuating system capable of utilizing extremely high pressures without resulting in excessive heat.

A further object is to provide a novel hydraulic power unit which is adapted to maintain a uniform adjusted rate of feed regardless of variations in resistance.

A more specific object is to provide a new and improved power unit embodying a variable delivery pump, and an orifice control for effecting a uniform rate of feed uninfluenced by changes in the pump delivery pressure.

Still another object is to provide a power unit embodying a novel hydraulic system for translating the spindle, which system utilizes relatively high working pressures so as to minimize irregularities and inaccuracies in the feed movement automatically caused by the presence of entrained air or vapor in the motive fluid, and consequently to obtain a uniform rate of feed regardless of varying resistance to the movement of the actuator.

A further object is to provide a new and improved hydraulic power unit in which the component elements, such as the pump, the valves and the actuator are so correlated that leakage is automatically compensated for and not reflected in the feed movement of the actuator under varying load conditions.

Another object is to provide a power unit having novel means for controlling the operation of the spindle actuator automatically, and having a single manual control operable within the limits of the automatic range without likelihood of injury to the structure.

Further objects and advantages will become apparent as the description proceeds.

In the accompanying drawings,

Figure 1 is a forward end elevational view of

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a power unit embodying the features of my invention.

Fig. 2 is a fragmentary front side elevational view of the power unit, partially sectioned along line 2—2 of Fig. 1.

Fig. 3 is a longitudinal vertical sectional view of the power unit taken substantially along line 3—3 of Fig. 1.

Fig. 4 is a fragmentary transverse vertical sectional view taken substantially along line 4—4 of Fig. 3, and illustrating the control valve.

Fig. 5 is a fragmentary sectional view taken substantially along line 5—5 of Fig. 4.

Fig. 6 is a detail sectional view taken substantially along line 6—6 of Fig. 4.

Fig. 7 is a fragmentary detail sectional view taken substantially along line 7—7 of Fig. 3, and illustrating the relief valve and dwell return.

Fig. 7a is a fragmentary enlarged view of a portion of the relief valve and dwell return shown in Fig. 7.

Fig. 8 is a fragmentary detail sectional view taken along line 8—8 of Fig. 1, and illustrating part of the pump delivery control.

Fig. 9 is a plan development of the periphery of the control valve.

Fig. 10 is a diagrammatic representation of the hydraulic control circuit of the power unit.

Fig. 10a is a fragmentary view of a modified form of pump control line adapted for use in the circuit.

Figs. 11a to 11e are a set of fragmentary transverse sectional views illustrating the adjustment of different axially spaced portions of the control valve when the latter is in stop position.

Figs. 12a to 12e are corresponding views of the control valve in rapid approach position.

Figs. 13a to 13e are corresponding views of the control valve in forward feed position.

Figs. 14a to 14e are corresponding views of the control valve in reverse or rapid return position.

Figs. 15 and 16 are views of the pump valve taken respectively in the directions of the arrows 15—15 and 16—16 of Fig. 3.

Fig. 17 is a fragmentary sectional view of the control valve taken along line 17—17 of Fig. 4.

Referring more particularly to the drawings, the power unit constituting the exemplary embodiment of the invention comprises a hollow housing 1 adapted to be mounted in fixed position on a suitable support (not shown). The housing 1 preferably is generally rectangular in shape, longitudinally and transversely. A mounting base or plate 2 is integral with the bottom of the

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housing 1, and is formed with a central longitudinal locating groove 3.

The housing 1 has an internal space, the lower portion of which constitutes an enclosed liquid reservoir 4, and is formed in the forward end over the reservoir with an internal solid body structure 5. A cover plate 6 is removably secured to the forward end of the housing 1, and serves to close one end of the interior. Similarly, the other end of the housing 1 is closed by a removable gear case 7 having a cover plate 8.

A longitudinal spindle 9 is mounted for rotation and axial reciprocation in the housing 1, and projects from the forward end thereof for attachment of a suitable tool or workpiece. To support the spindle 9, a tubular quill 10 extends through, and is mounted for reciprocation in, a cylinder 11 formed in the body structure 5. Two flanged heads or bushings 12 and 13 are rigidly secured to opposite ends of the body structure 5 to close the cylinder 11, and serve as guides for opposite end portions of the quill 10. The quill 10 is formed intermediate its ends with a piston 14 slidably disposed in the cylinder 11, and the opposite end portions of the quill therefore constitute piston rods. In the present instance, the piston 14 is of the differential type, with the rear area approximately equal to twice the forward area.

The forward end of the quill 10 extends from the cylinder head 12 through an aligned opening or bore 15 in the cover plate 6. A sleeve 16 is fitted in the bore 15, and is formed at its inner end with an external peripheral flange 17 seated in a counterbore 18, and at its outer end with an internal peripheral flange 19 in wiping engagement with the quill 10 to prevent the entry of chips and other foreign matter. The cylinder head 12 is seated in a counterbore 20 in the inner end of the sleeve 16. Consequently, the cover plate 6 serves to maintain the cylinder head 12 and the sleeve 16 both tightly in position. An oil seal 21 is confined within the sleeve 16 in wiping engagement with the quill 10. Any fluid reaching the sleeve 16 is returned through a passage 22 to the reservoir 4 within the housing 1.

A tubular bearing housing 23 has an internal flange 24 at one end secured coaxially to the quill 10. The spindle 9 extends in peripherally spaced relation through the quill 10, thereby defining an intervening clearance space 25, and projects through the flange 24. The forward end of the spindle 9 is formed with an enlarged nose 26 which projects from the bearing housing 23, and which is suitably supported therein by a radial and end thrust anti-friction bearing 27. More particularly, the inner raceway of the bearing 27 is confined on the spindle nose 26 against an inwardly facing annular shoulder 28 by an annular nut 29, and the outer raceway is confined in a counterbore 30 in the housing 23 by an annular nut 31. The bearing 27 is closed from the exterior by a washer 32 and oil seal 33 secured in position by the nut 31. Consequently, the spindle 9 is freely rotatable within the quill 10, but is connected for axial movement therewith.

Formed in, and opening to, the outer end of the spindle nose 26 is an axial bore 34 adapted to receive a removable socket member 35. In the present instance, the member 35 is formed with a conventional taper socket 36 adapted to receive the shank of a tool (not shown). The socket member 35 may be adjusted axially within the spindle nose 26 by means of a nut 37 to compensate for tool wear, a spline key 38 serving to con-

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fine the member against rotation, and a set screw 39 being available to lock the member in selected position of adjustment.

Integral with the forward end of the bearing housing 23 is an external peripheral mounting flange 40, provided with a locating pilot 41 and with suitable apertures 42 for the attachment of a flange-type multiple spindle head (not shown). Such head would have an operative drive connection with the spindle 9 through the bore 34.

The rear or inner end of the spindle 9 is longitudinally splined for independent axial movement relative to a rotary drive sleeve 43 journaled in the gear case 7. More particularly, this sleeve 43 is mounted in spaced anti-friction ball bearings 44 supported respectively in the gear case 7 proper and the cover 8. The inner end of the quill 10 in its rearward position telescopes freely with the contiguous end of the spline sleeve 43, and is open to the interior of the housing 1. It may be mentioned at this time that any liquid tending to leak past the piston 14 in the cylinder 11, from one side to the other, will be intercepted by a peripheral drain groove 45 opening through radial passages 46 in the piston to the clearance space 25, and from there will drain through the inner end portion of the quill 10 to the reservoir 4. The path of drainage to the reservoir 4 is represented schematically in Fig. 10 by the catch pan 4^a and the line 4^b.

The rotary drive for the spindle 9 comprises a gear 47 removably keyed to the spline sleeve 43 between the bearings 44. Meshing with the gear 47 is a gear 48 removably keyed to a drive shaft 49 journaled in spaced anti-friction bearings 50 mounted respectively in the gear case 7 proper and the cover 8. The shaft 49 extends through the gear case to the exterior of the housing 1, and a pulley 51 adapted to be connected to a suitable source of power (not shown) is fixed on the outer end. The gears 47 and 48 are located within a recess 52 in the outer face of the case 7 proper, and may be changed, upon removal of the cover 8, to obtain different selected speed ratios in the spindle drive from the power input shaft 49.

A hydraulic system (Fig. 10) is incorporated in the power unit for translating the spindle 9 axially through a predetermined program of movements. While different cycles may be employed, an illustrative cycle comprises rapid approach from the stop position, forward feed, automatic reverse, and rapid return to the stop position. Optionally, a dwell may be provided at the end of the forward feed and before reverse. In the present system, the rate of feed and the period of dwell are subject to accurate adjustment and control.

The hydraulic system, in its preferred form, comprises generally a variable delivery pump 53 driven at a constant speed by the shaft 49. The pump 53 has an inlet line 54 taking fluid from the reservoir or sump 4, and a discharge line 55 for supplying fluid under pressure. A control valve 56 is operable to connect the pressure line 55 selectively to two supply lines 57 and 58 leading to opposite ends of the cylinder 11, and also on occasion to direct fluid from the cylinder to an exhaust line 59 discharging to the sump 4. A selected uniform rate of movement is obtained during forward feed by supplying fluid to the cylinder 11 at a propelling pressure, and by metering the exhaust fluid through an orifice control and utilizing the back pressure thus created to control the displacement of the pump 53.

The pump 53 (Figs. 3 and 8) shown in the drawings, is of the type having parallel annularly-arranged piston and cylinder units, and is mounted on the inside of the gear case 7 within the housing 1. In its preferred form, the pump 53 comprises a stationary valve member 60 and a rotary cylinder body 61, having flat annular end valve faces 62 and 63 in rotary bearing engagement. The valve member 60 has the form of a cylindrical plug tightly pressed into a bore 64 in the gear case 7, and is located radially by means of a spline screw 65 projecting from the case into a groove 66. The inner end of the drive shaft 49 extends freely through an axial bore 67 in the valve plug 60 into an axial bore 68 in the cylinder body 61. Formed in the bore 68 are internal splines 69 which interfit slidably with longitudinal splines 70 on the inner end of the shaft 49 to establish the drive connection for the pump 53, and which are relatively short so as to permit a limited amount of peripheral float of the cylinder body 61 and thereby insure accurate seating of the valve faces 62 and 63.

Formed in the cylinder body 61 in parallel uniformly spaced relation about the bore 68 are a plurality of pump cylinders 71 having cylinder ports 72 opening from the inner ends to the valve face 63. Suitable pump pistons or plungers 73 are respectively reciprocable in the cylinders 71. The plungers 73 are urged outwardly by coiled compression springs 74. These springs engage the inner ends of the cylinders 71, and extend into bores 75 in the inner ends of the plungers 73 so that the latter may be comparatively short in length.

A swash plate 76 having a flat bearing face 77 is in operative association with the outer ends of the plungers 73, and when inclined is adapted to cause reciprocation of the plungers in uniformly timed sequence upon rotation of the cylinder body 61. Bearing shoes 78 are connected by means of suitable inserts 79, for universal pivotal adjustment, to the outer ends of the plungers 73, and have flat faces in sliding engagement with the bearing face 77 of the swash plate 76. The swash plate 76 is mounted in the housing 1 on a pivot pin 80 at one end for angular adjustment about an axis extending transversely of the axis of rotation of the cylinder body 61. Preferably, the pin 80 is located at a point adjacent the axis of each plunger 73 as the latter reaches top dead center position at the end of the pressure stroke. As a result, the inward limit of movement of the plungers 73 is substantially constant and close to the inner ends of the cylinders 71.

In operation, the pump 53 will discharge continuously, and the rate of discharge will be substantially constant for any given setting of the swash plate 76. If the angular position of the swash plate 76 is changed, the length of the plunger stroke will be adjusted to vary the rate and volume of pump displacement. When the plate 76 is adjusted into a plane perpendicular to the shaft 49, no reciprocation will be imparted to the plungers 73 and the pump displacement will be zero. In the reciprocation of the plungers 73, those on the suction stroke take in fluid from the line 54, and those on the pressure stroke discharge fluid to the line 55. For further details of the pump valve porting, reference may be had to a copending application, Serial No. 348,904, filed July 31, 1940, by Ira J. Snader and Max A. Mathys (now Patent No. 2,299,235, issued October 20, 1942).

The swash plate 76 is angularly adjustable about the pin 80 to vary the displacement of the pump. Within the broad aspects of the present invention, such adjustment may be effected by any suitable means. In the present instance, the swash plate 76 is automatically controlled in response to a predetermined control pressure. Thus, the plate 76 is automatically adjusted to reduce the pump delivery upon an increase in control pressure, and to increase the delivery upon a decrease in the pressure. The particular means for this purpose comprises a balancing plunger 81 and a control plunger 82 acting on the free end of the swash plate 76 in a direction to level off the latter. The balancing plunger 81 establishes a condition of balance against the hydraulic reaction in the cylinders 71, and in the absence of pressure upon the control plunger 82, the return springs 74 tend to swing the swash plate into its maximum stroke position.

The pressure balancing plunger 81 (Fig. 8) is reciprocable in a bore 83 formed in the body structure 5 of the housing and extending transversely of the pin 80. One end of the bore 83 opens toward the free end of swash plate 76, and the other end is closed to define a pressure chamber 84 and is connected through a branch line 85 to the pressure line 55 of the pump 53. A shoe 86 is connected by means of an insert 87 for universal pivotal adjustment to the outer end of the plunger 81, and bears slidably against a flat face 88 on the rear of the swash plate 76. The diameter of the plunger 81 and the moment arm of force exerted thereby on the swash plate 76 are such that this force will substantially counterbalance the hydraulic reaction in the pump cylinders 71. Since the delivery pressure of the pump 53 is always connected through the lines 55 and 85 to the pressure chamber 84 at the inner end of the plunger 81, the swash plate 76 will be hydraulically balanced throughout the range of pump delivery.

The control plunger 82 similarly is mounted for reciprocation in a bore 89 formed in the body structure 5 of the housing 1 and opening transversely of the pin 80 to the rear face 88 of the swash plate 76. A shoe 90 is connected by means of a suitable insert 91 (Fig. 3) for universal pivotal adjustment to the outer end of the plunger 82, and slidably engages the face 88. A positive stop is provided to limit the position of maximum inclination of the swash plate 76 which determines the maximum pump delivery. In the present instance, this stop consists of a plug 92 positioned in the inner end of the bore 89 for engagement by the plunger 82. Opening from the inner end of the bore 89, between the plunger 82 and the plug 92, is a pump control line 93. This line may be connected to any suitable source of control pressure, and in the present instance is adapted to be connected by the control valve 56 to the exhaust side of the cylinder 11 so as to utilize the back pressure under orifice control for controlling the delivery of the pump 53.

The control valve 56 (Figs. 4, 9, 10 and 11a to 14e) may be of any suitable type having the desired operating characteristics. In the present instance, the valve 56 is shown as of the rotary type, comprising a valve bore 94 opening transversely through the body structure 5 of the housing 1, and a valve plug 95 mounted for rotary adjustment therein. It will be understood that the various lines or passages 55, 57, 58, 59 and 93 have appropriate port openings to the periphery of the valve bore 94. The valve 56 may be

constructed to provide any desired program of axial movements of the spindle 9, and in the present instance is of the four-position type, defining a cycle comprising rapid approach, forward feed, reverse, rapid return and stop.

In its preferred construction, the valve plug 95 is formed in one transverse plane *b-b* (Figs. 4, 9, 11b, 12b, 13b and 14b) with two diametrically opposite transverse notches 96 and 97 opening at their ends through radial ports 98 to an axial bore 99. The pressure line 55 opens to the bore 94 in this plane for communication with the notch 97 so that the bore 99 is adapted to be under pressure, the notch 96 being provided to obtain a pressure balance on the plug 95. Formed in another transverse plane *c-c* (Figs. 4, 9, 11c, 12c, 13c and 14c) of the plug 95 are three peripherally spaced radial ports 100, 101 and 102 opening to the pressure bore 99, and hence constituting pressure ports. The outer end of the port 102 is peripherally elongated as indicated at 103. The cylinder lines 57 and 58 open at diametrically opposite points to the bore 94 in the plane of the ports 100, 101 and 102. Also formed in the plug 95 are two axially spaced peripheral exhaust grooves 104 and 105 which are interconnected by two peripherally spaced longitudinal grooves 106 and 107 located respectively intermediate the pressure ports 101 and 102, and 102 and 100. The exhaust line 59 opens to the bore 94 in the plane of the groove 105. Two additional longitudinal exhaust grooves 108 and 109 are formed in the plug 95 respectively between the notches 96 and 97, and open to the peripheral groove 104.

The pump control line 93 opens to the periphery of the valve bore 94 in a transverse plane *d-d* (Figs. 4, 9, 11d, 12d, 13d and 14d). Formed in the periphery of the plug 95 is a longitudinally inclined pressure groove 110 which extends from the pressure port 101 to the plane *d-d*, and two short longitudinal exhaust grooves 111 and 112 which extend from the peripheral exhaust groove 105 through the plane *d-d*, and which are approximately in alignment with the ports 101 and 100.

Formed in one end of the valve plug 95 is an axial orifice valve bore 113. An orifice valve plug 114 is secured by means of a flat key 115 for rotary adjustment in the bore 113, and has an actuating shaft 116 formed with an adjusting dial 117 disposed against the outer end of the control valve plug 95. If desired, a spring washer 118 may be interposed between the plug 95 and the dial 117 to take up lost motion and to provide a frictional retaining means. The dial 117 is slotted to receive a suitable hand tool for rotatably adjusting the plug 114, and is appropriately graduated to indicate the selected feed adjustment.

The orifice valve plug 114 is formed with an axial bore 119 which opens through a radial port 120 to a peripheral groove 121 in turn connected through a radial port 122 in the plug 95 to the peripheral exhaust groove 105. Also formed in the plug 114 is a tapered peripheral notch 123 which is connected at its large end through a radial port 124 to the bore 119, and which is rotatably adjustable across an orifice 125 opening to the bore 113. The outer end of the orifice 125 opens to a longitudinal groove 126 formed in the periphery of the main valve plug 95, and projecting at opposite ends through the transverse planes *c-c* and *d-d*. Suitable small drain V-grooves 127 are formed in the plug 95 along op-

posite sides of the grooves 126 and open to the peripheral grooves 104 and 105. The grooves 127 prevent leakage from the pressure ports 100 and 101 and the groove 110 to the groove 126, so that variations in the pump pressure will not be reflected in the rate of feed as controlled by the orifice 125. It will be understood that the effective flow area of the orifice 125 is controlled by the position of the tapered notch 123, and is progressively reduced upon rotary adjustment of the plug 114 to move in a direction of decreasing lateral area across the orifice.

Assuming that the spindle 9 is in retracted stop position, the control valve 56 will occupy the position shown in Figs. 11b, 11c and 11d. In this position of the valve, the line 57 to the forward end of the cylinder 11 is blocked, the line 58 from the other end of the cylinder is connected through 107-105 to the exhaust line 59, and the pump control line 93 is connected through 110-101-99-98-97 to the pressure line 55 of the pump 53. The pressure in the line 93 causes the swash plate 76 to level off into no-delivery position so that the pump serves only to maintain a constant low pressure in the line 55 equal to the pressure in the line 93.

Upon adjustment of the valve 56 in a counterclockwise direction (Figs. 12b, 12c, and 12d) into rapid approach position, both cylinder lines 57 and 58 are connected through 100-102-99-98 to the pressure line 55, and the pump control line 93 is connected through 112-105 to the exhaust line 59. Consequently, the swash plate 76 is adjusted by the springs 74 into maximum delivery position as determined by the stop 92. Because of the differential area of the piston 14, fluid displaced from the front end of the cylinder 11 is directed to the large end of the cylinder. The rapid approach, therefore, occurs at a maximum rate of speed.

Upon subsequent adjustment of the valve 56 in a clockwise direction into forward feed position (Figs. 13b, 13c and 13d), the line 58 is connected through 103-102-99-98-97 to the pressure line 55, the line 57 is connected through 126-the orifice 125-123-124-119-120-121-122-105 to the exhaust line 59, and the pump control line 93 likewise is connected to the groove 126 at the inlet side of the orifice. Consequently, the orifice 123 meters the fluid discharging from the forward end of the cylinder 11, and the back pressure thus created is utilized to reduce and control the discharge of the pump 53 to the desired uniform feed rate.

At the end of the forward feed, the control valve 56 is reversed (Figs. 14b, 14c and 14d) to effect a rapid return of the spindle 9 into initial stop position. In this condition of adjustment of the valve 56, the line 57 to the forward end of the cylinder 11 is connected through 101-99-98-97 to the pressure line 55, the line 58 from the rear end of the cylinder is connected through 107-105 directly to the exhaust line 59, and the pump control line 93 is connected through 111-105 to the exhaust line 59. Consequently, the pump 53 is again adjusted for maximum displacement. Fig. 10 shows the valve 56 in the rapid return position but for convenience of inclusion of both plungers 81 and 82 in the diagrammatic disclosure, the pump 53 has been shown in the position of zero displacement. In the actual construction, the plungers 81 and 82 are disposed side by side at the same distance from the pivotal axis of the swash plate 76 and for maximum displacement the swash plate

would occupy a position of maximum inclination.

Since the rear area of the piston 11 is twice as large as the forward area, and since in rapid approach the opposite ends of the cylinder are interconnected, the rapid return and the rapid approach will have the same rate of movement.

The orifice control in the exhaust side of the hydraulic system sets up a back pressure which is utilized to control the displacement of the variable delivery pump during feed operation. This control, since it is not required to operate against the full delivery pressure of the pump, is accurate and sensitive in operation. More particularly, the control pressure acting through the plunger 82 tends to adjust the pump 53 in a direction to decrease the displacement, and the springs 74 tend to adjust the pump 53 in a direction to increase the displacement. During rapid traverse the pump 53 is adjusted for maximum displacement. In the stop position, the pump 53 is substantially at zero displacement, but does supply fluid to maintain the system under low pressure. During feed condition, the control pressure and the spring pressure are in sensitive balance. The foregoing arrangement serves ideally to dampen any pressure pulsation tending to occur. Thus, the opening pressure imparted to the swash plate 76 by the springs 74 is comparatively light, and a momentary closing pressure impulse is comparatively high. Consequently, one side of the harmonic pulsation curve is slowed down, although the action is so rapid that it is not discernible to the eye, thereby resulting in effective dampening.

The pump delivery pressure will vary over a wide range, depending on the work resistance, and on occasion may be extremely high. However, the pressure for regulating the pump 53 during feed operation, being independent of the pump delivery pressure, is relatively constant, and may be comparatively low and much less than the maximum pump pressure. The low control pressure is advantageous since less heat will be generated in the passage of fluid through the orifice than if a high pressure were employed. Also, pulsations are more effectively eliminated with the use of a low control pressure. While a comparatively low control pressure is desirable, it should be sufficiently high to avoid substantial changes in the compressibility of the fluid. I have found that in the present installation, a sensitive control is obtained when the back pressure created by the orifice is maintained above 110 pounds per square inch. The most suitable orifice pressure and hence the pump control pressure can be determined from the compressibility curve of the motive fluid for each installation. In a typical curve, the compressibility initially increases rapidly with an increase in pressure, and then levels off and increases only slightly as the pressure continues to increase over a substantial range. If the pressure is selected in this range where the compressibility of the fluid is fairly uniform, the control of the pump will have a low elasticity regardless of load variations, changes in fluid viscosity, the presence of entrained vapors and pressure vibrations in the system.

The valve plug 95 is confined in the bore 94 against endwise movement and for rotary adjustment. To this end, a collar 128 is removably keyed on one end of the valve plug 95 for rotation therewith, and is rotatable within a counterbore 129 in one end of the valve bore 94. A hand knob

130 is fixed to the extreme end of the valve plug 95 at the front of the housing 1 and serves to confine the collar 128 in position, and as a means for adjusting the valve 56 manually. A collar 131 is keyed to the other end of the valve plug 95 and is rotatably confined in a counterbore 132. The collar 131 is held in position by an abutting collar 133 fixed on the extreme end of the plug 95. It will be seen that the valve plug 95 and the collars 128 and 131 are confined in assembled relation within the bore 94 by the hand knob 130 and the collar 133. The extreme ends of the valve plug 95 are of the same diameter so that the hand knob 130 and the collar 133 may be interchanged if it should be desired to adjust the valve manually from the other side of the housing 1.

To provide means for adjusting the valve 56 in timed relation to the axial movements of the spindle 9, a dog bar 134 is connected at its forward end by a pin 135 to the flange or face plate 40 of the housing 23. The pin 135 is rigidly threaded into the dog bar 134 and has an accurate sliding fit with a radial bore 136 in the face plate 40. The bar 134 extends slidably through a hole 137 in the front cover 6 and an aligned hole 138 formed in the side of the housing 1, and is therein supported to provide a slight clearance 139 between the bar and the face plate 40. The loose connection between the dog bar 134 and the face plate 40 serves to maintain the quill 10 securely against angular displacement without requiring close accuracy in alinement.

Formed in the bar 134 is a longitudinal groove 140 for adjustably supporting a plurality of valve control dogs. In the present instance, three dogs are shown, namely, a stop dog 141, a feed dog 142, and a reversing dog 143. To support the dogs, the longitudinal groove 140 is formed with three accurate faces 144, 145 and 146. Suitable clamping screws 147 are threaded through each dog into engagement with the bar 134 opposite the face 146. The clamping pressure is mainly absorbed by the face 146 and comparatively small pressures are absorbed by the faces 144 and 145. Consequently, the main pressure is absorbed longitudinally of the screws 147 so that even with a large clamping pressure, no substantial bending force is imparted to the dog bar. Although more than half of the dog bar is cut away, the bar affords a rigid support for the dogs and retains a good sliding fit in the bores 137 and 138.

The dogs 141, 142 and 143 are arranged for selective engagement with dog abutments 148, 149 and 150 on the collar 128. These abutments are spaced axially of the collar 128 to avoid interference, and are movable successively into position for engagement by the respective dogs.

Coacting with the dog control is a spring reversing mechanism. In its preferred form, this mechanism comprises the collar 131 which is formed with diametrically opposite peripheral notches 151 and 152 having perpendicular sides. To provide clearance, the valve plug 95 is flattened at opposite sides in the transverse plane of the notches. Two spring plungers 153 and 154 are slidably disposed respectively in bores 155 and 156, opening tangentially to opposite sides of the valve bore 94, and are adapted for engagement with the reverse collar 131. Suitable coiled compression springs 157 and 158 are disposed in the bores 155 and 156 and tend to urge the plungers 153 and 154 toward valve 56.

In operation, the plungers 153 and 154 engage the edges of the notches 151 and 152 (Fig. 11a)

to hold the abutment 148 against the stop dog 141 (Fig. 11e) in stop position.

To start the cycle, the valve 56 is rotated in a counter-clockwise direction, for example by means of the hand knob 130, thereby forcing the plungers 153 and 154 (Fig. 12a), and moving the abutment 149 into the path of the feed dog 142 (Fig. 12e). In this position, the plungers 153 and 154 engage the periphery of the reverse collar 131, and hence are ineffective. The valve 56 is now in position to effect the rapid approach movement of the spindle 9.

At the end of the forward approach movement, the feed dog 142, which travels with the quill 10, engages the abutment 149 to rotate the valve 56 clockwise through a predetermined angle into the forward feed position. As a result, the abutment 150 is moved into the path of the reverse dog 143 (Fig. 13e), but the plungers 153 and 154 still remain in engagement with the periphery of the collar 131 (Fig. 13a). The spindle 9 now moves as a feed rate determined by the setting of the adjustable groove 123.

At the end of the feed movement, the reverse dog 143 engages the abutment 150 to rotate the valve 56 still further in a clockwise direction into the stop position. This causes the plungers 153 and 154 to ride off the periphery of the collar 131 into engagement with the notches 151 and 152, and therefore to complete the reversal of the valve 56 with a snap action independently of the dog 143 (Figs. 14a and 14e). The abutment 148 is thereby returned into the path of the stop dog 141, and the valve 56 is positioned to effect the rapid return movement of the spindle 9 into the initial stop position.

At the end of the rapid return, the stop dog 141 engages the abutment to rotate the valve 56 counter-clockwise against the spring action of the plungers 153 and 154 into the stop position (Figs. 11a and 11e).

Instead of reversing the valve 56 immediately at the end of the forward feed, a dwell may be interposed before reversal. Such dwell is obtained by means of a valve reversing mechanism having a time delay. To render such dwell mechanisms functionally operative, the reverse dog 143 is removed, and the quill 10 is caused to engage a positive stop at the end of the forward movements. In the present instance, the forward end of the cylinder 11, upon engagement with the piston 14, serves as the stop.

The dwell mechanism (Figs. 4, 5, 7 and 10) comprises a tubular or hollow floating plunger 159 reciprocable in a bore 160 formed in the body structure 5 of the housing 1 and transversely intersecting the main valve bore 94. A gear 161 rigid with the valve plug 95 meshes with a longitudinal gear rack 162 formed in the side of the plunger 159. The range of the reciprocation of the plunger 159 is suitably limited by a stop plug 163 removably inserted in one end of the bore 160, and an actuating piston 164 slidable in a cylinder bore 165 opening to the other end of the bore 160. The actuating piston 164 has a head 166 defining its retracted position, and is movable inwardly against the contiguous end of the plunger 159 to shift the latter, together with the plungers 153 and 154, into one end position in engagement with the stop plug 163, and thereby to locate the control valve 56 in reverse position. Movement of the plunger 159 into the other end position against the piston 164 when the latter is relieved serves to locate the valve 56 in rapid approach position.

The closed end of the cylinder bore 165 is connected through a line 167 to a dwell and relief valve 168 (Figs. 7 and 10). This valve comprises a sleeve 169 fixed in a bore 170 in the gear case 7, and having an axial bore 171 open at one end to a drain line 172 leading to the reservoir 4. Threaded into the other end of the bore 170 is a closed tubular cap 173 connected to a branch of the drain line 172. A valve plunger 174 is reciprocable in the bore 171, and has a stop flange 175 normally urged into engagement with the inner end of the sleeve 169 by a coil compression spring 176 confined within the cap 173.

Two spaced annular grooves 177 and 178 are formed in the outer periphery of the sleeve 169, and open respectively through radial ports 179 and 180 to the bore 171. These grooves are connected respectively to the pressure line 55 and the line 167. Formed in the periphery of the valve plunger 174 are two axially spaced grooves 181 and 182 movable selectively into registration with the ports 180. Suitable openings 183 are formed in the flange 175 connecting the groove 182 to the interior of the cap 173.

The valve plunger 174 is formed in one end portion with an axial bore 184. The outer end of this bore is closed by a screw plug 185, and the inner end is connected through a passage 186 open to the port 179. An orifice plug 187, with a restricted axial orifice 188 in one end wall, is removably threaded into the inner end of the bore 184 and is spaced from the plug 185 so that the middle portion of the bore defines a confined space open to the interior of the plug and hence to the orifice. The intermediate portion of the bore 184, at the downstream side of the orifice 188 is connected through a passage 189 in the plunger 174 to the groove 181.

The inner end of the bore 171 opening to the line 172 is reduced in diameter to define a relief valve passage 190, and the corresponding end of the valve plunger 174 is reduced in size to constitute a relief valve member 191 slidable in the passage. Hence, the plunger 174 is formed with a pressure face 192 open to the ports 179. Suitable tapered grooves 193 are formed in the periphery of the valve member 191 for movement into and out of communication with the ports 179.

In operation, the valve member 174 normally occupies the position shown in Fig. 7, in which the groove 181 is closed, and the tapered grooves 193 are out of communication with the port 179. At this time, no fluid passes through the valve 168, and the cylinder 165 is relieved through 167—178—180—182—183—173—172 to the reservoir 4. Upon feed movement of the quill 10 into the forward end position against the positive stop, the spindle 9 will of course come to a stop, and upon the cessation of motion, the pressure in the pump control line 93 will drop, thereby causing an increase in the pump displacement. Since the delivery fluid has no place to go, the pump pressure will rise. The rising pressure, acting on the pressure face 192, will ultimately shift the valve plunger 174 to move the groove 182 out of communication with the ports 180, and the groove 181 into communication therewith. Thereupon, a flow connection is established to the cylinder 165 from the pump 53 through 55—177—179—186—188—184—189—181—180—178 and 167. The transmitted pressure acts on the piston 164 to move the latter at a graduated rate depending on the flow rate through the orifice 188. It will be understood that the elevated pressure,

which acts on the plunger 81, serves to reduce the pump displacement, and hence the orifice 188 sets up a pressure differential influencing the volume output of the pump and the supply of fluid to the cylinder 165. The piston 164 in the cylinder 165 in turn acts on the plunger 159 to move the control valve 56 slowly toward the stop position, the elapsed time defining a dwell in the reversal of the spindle translation at the forward end of the cutting feed movement. As soon as the valve 56 is near the stop position, the spring-actuated plungers 153 and 154 will act to complete rotation thereof quickly into reverse position.

The pressure rise during the dwell period is limited to a predetermined maximum. Should the pump pressure rise to this maximum, which is determined by the pressure of the spring 176, it will move the valve plunger 174 sufficiently to open the tapered grooves 193 to the ports 179, and thereby cause excess fluid to be relieved through the relief valve passage 190 to the drain line 172. In this condition, the valve 168 will act both as a dwell return and a relief valve. Thus, for the longer dwell periods, the relief of excess fluid serves to maintain the pump delivery pressure substantially constant, and the orifice 188 passes fluid entirely under pressure control to meter the supply to the cylinder 165 at a rate determined by the orifice flow characteristics.

Optional means is provided for actuating the control valve 56 by remote control. When the remote control is not utilized, the hollow plunger 159 is open at both ends so as to be pressure balanced for free manual operation of the valve 56 by means of the hand knob 130. When the remote control is desired, the plunger 159 is closed by threading a plug 194 (Fig. 5) into one end. Two fluid lines 195 and 196 open to opposite ends of the bore 160, and are adapted to be connected by a remote control pilot valve 197 to exhaust or selectively to pressure.

The valve 197 comprises a valve housing 198 mounted on a suitable support, such as a plate 199 attached to some part of the machine frame (not shown) on which the power unit is used. Formed in the housing 198 is a valve bore 200 having longitudinally spaced port connection with the pump branch pressure line 85, the exhaust line 59 and the remote control lines 195 and 196. A valve plunger 201 of the spool type is slidably adjustable in the bore 200.

The valve plunger 201 may be adjustable by any suitable means, and in the present instance two solenoids 202 and 203 controlled by electric push buttons (not shown) are mounted on the support 199 and have cores or armatures 204 connected respectively to opposite ends of the plunger.

Upon energization of the solenoid 202, the plunger 201 will be shifted in a direction (Fig. 10) to connect the remote start line 195 to pressure and the line 196 to exhaust. Thereupon, the pressure fluid acting on the remote control plunger 159 will shift the control valve 56 from the stop position into the rapid approach or starting position. As soon as this adjustment is effected, the push button is released to deenergize the solenoid. A coiled compression spring 205, interposed between the valve plunger 201 and a bracket 206 on the support, then returns the plunger to intermediate neutral position. Expansion of the spring 205 is limited by abutment with the housing 198 to prevent over travel past neutral of the

plunger 201. In the neutral position, the valve plunger 201 connects both lines 195 and 196 to the exhaust line 59 and blocks the pressure line 85 so that the remote control plunger 159 is pressure balanced to permit free adjustment of the valve 56 by the dogs 141, 142 and 143 or by operating of the hand knob 130.

Upon energization of the solenoid 203, for example at any time desired or in case of emergency during the forward translation of the quill 10, the plunger will be shifted to connect the remote reverse line 196 to pressure and the line 195 to exhaust. As a result the control valve 56 will be shifted into reverse position. Thereupon, the spring 205 again returns the valve member 201 to neutral. At the end of the return movement of the quill 10, the stop dog 141 moves the valve 56 into stop position and maintains it there until the next cycle is initiated by again energizing the solenoid 202.

It will be noted that the hydraulic power unit is subject to manual control, either by means of the hand knob 130 or the remote control at any point within the limit of the automatic range without likelihood of injury to the structure. The unit is compact and sturdy, and can be operated at a relatively high fluid pressure without creating objectionable heating since no excess fluid required to be discharged through a relief valve is normally delivered by the pump. Any leakage in the hydraulic system is automatically compensated for since the orifice control will reflect such leakage and adjust the pump displacement accordingly. By using a suitable pump control pressure in the low-compressibility range of the fluid and by compensating for any leakage, a uniform rate of feed movement is obtained regardless of load variations.

It is to be understood that while the invention has been shown as incorporated in a hydraulic power unit of the quill type, various features thereof are also applicable to hydraulic power units of the sliding head type.

In instances where a comparatively high control pressure is desired, it may be advantageous to provide a modified pump control line 93a having interposed therein a check valve 207 permitting free flow to the control plunger 82, and a parallel flow restriction 208 for dampening the flow of fluid from the plunger. Such optional construction is indicated in Fig. 10a, the modified line 93a being used in lieu of the line 93 in Fig. 10.

The pump 53 per se is covered in my copending divisional application, Serial No. 710,153, filed November 15, 1946.

I claim as my invention:

1. A hydraulic power unit comprising, in combination, a hydraulic motor, a variable delivery pump having an adjustable member for controlling the pump displacement, spring means in said pump acting on said member in a direction to increase the displacement of said pump, means for driving said pump, an exhaust line, hydraulic means for counterbalancing said member against the pump reaction, hydraulic means acting on said member in a direction to decrease the displacement of said pump, and a control valve adapted in one position to connect said pump and exhaust line to opposite sides of said motor and being adjustable into a stop position to block the forward side of said motor, to connect the other side of said motor to said exhaust line, and to connect the outlet of said pump to the last

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mentioned hydraulic means whereby to adjust said pump into position of zero displacement.

2. A hydraulic power unit comprising, in combination, a piston and cylinder motor, a variable delivery pump, means for driving said pump, said pump having an actuating member adjustable to vary the displacement from zero to maximum, means for counterbalancing said member against the pressure reaction in said pump, valve means for connecting said pump reversibly to opposite ends of said motor, and for varying the resistance to flow of the fluid discharging from said cylinder for rapid traverse and feed operation, and hydraulic means responsive to the pressure of the discharge end of said motor for automatically adjusting the displacement of said pump to obtain maximum displacement during rapid traverse, and substantially constant reduced displacement during feed operation.

3. A hydraulic power unit comprising, in combination, a cylinder, a piston slidably disposed in said cylinder, a variable delivery pump having a pressure discharge line, means for driving said pump, an exhaust line, adjustable valve means for connecting said discharge line and said exhaust line selectively to opposite ends of said cylinder to translate said piston at rapid traverse and feed rates, orifice means connected to receive fluid discharging from said cylinder during feed operation, and means responsive to the pressure at the inlet side of said orifice means for controlling the displacement of said pump to maintain a substantially constant rate of fluid flow through said orifice under varying load conditions during feed operation.

4. A hydraulic power unit comprising, in combination, a cylinder, a piston slidably disposed in said cylinder, a variable delivery pump having a pressure discharge line, means for driving said pump, an exhaust line, adjustable valve means for connecting said discharge line and said exhaust line selectively to opposite ends of said cylinder to translate said piston at rapid traverse and feed rates, orifice means selectively available under the control of said valve means for setting up a back pressure in said exhaust line during feed operation, and means responsive to the pressure at the discharge side of said cylinder for controlling the displacement of said pump.

5. A hydraulic power unit comprising, in combination, a cylinder, a piston reciprocable in said cylinder, a translatable element connected with said piston, a drive shaft, a variable delivery pump driven by said shaft and having an adjustable element for controlling the pump displacement, supply lines opening to opposite ends of said cylinder, an exhaust line having a restricted adjustable orifice control passage and a parallel less restricted passage, pressure responsive means operable to shift said adjustable element respectively in response to an increase or decrease in pressure in said exhaust line to decrease or increase said pump displacement, a rotary control valve for connecting said supply lines selectively to said pump and to said passages and said pressure responsive means selectively to said passages whereby to translate said first-mentioned element selectively at a rapid approach, forward feed or rapid return, and means controlled by the translation of said first-mentioned element for automatically adjusting said valve.

6. A hydraulic power unit comprising, in combination with a translatable element, a hydraulic motor for translating said element, a variable delivery pump having an adjustable member for

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controlling the pump displacement, spring means in said pump acting on said member in a direction to increase the displacement of said pump, means for driving said pump, an exhaust line, hydraulic means for counterbalancing said member against the pump reaction, hydraulic means acting on said member in a direction to decrease the displacement of said pump and responsive to the pressure in said exhaust line exerted in opposition to said spring means, and a control valve adapted in one position to connect said pump and exhaust line respectively to opposite sides of said motor and to interpose a flow restriction in said exhaust line, and being adjustable into another position to by-pass said restriction, whereby to effect adjustment of said member into position of maximum pump displacement.

7. A hydraulic power unit comprising, in combination with a translatable element, a differential hydraulic motor for translating said element, a variable delivery pump having an adjustable member for controlling the pump displacement, spring means in said pump acting on said member in a direction to increase the displacement of said pump, means for driving said pump, an exhaust line, hydraulic means for counterbalancing said member against the pump reaction, hydraulic means acting on said member in a direction to decrease the displacement of said pump and responsive to the pressure in said exhaust line exerted in opposition to said spring means, and a control valve adapted in one position to connect said pump in parallel to opposite sides of said motor and to connect second mentioned hydraulic means to said exhaust line to effect adjustment of said member into position for maximum pump displacement, and being adjustable into another position to connect one end of said motor to said pump and the other end of said motor to said exhaust line, and to interpose a flow restriction in said exhaust line to adjust said member into position for reduced pump displacement.

8. A hydraulic transmission for a reversibly movable machine tool element comprising, in combination, a hydraulic motor for translating said element, supply lines opening to opposite sides of said motor, a variable delivery pump having a pressure discharge line and an adjustable control member movable to vary the pump displacement through a range from zero to maximum, spring means acting on said member in a direction to increase said pump displacement, means responsive to the discharge pressure of said pump for substantially counterbalancing said member against the pumping reaction, an exhaust line, a control valve adjustable into position to connect said supply lines respectively to said discharge and exhaust lines to effect movement of said element in one direction, stop means for positively limiting the movement of said element in said one direction, a restricted flow orifice functionally interposed to receive fluid discharging from said motor through said exhaust line when said element is moving in said one direction, hydraulic means acting in opposition to said spring means for controlling said member to adjust the pump displacement inversely in response to variations in the pressure set up by said orifice, and hydraulic means automatically operable in response to a predetermined pressure rise in said discharge line upon engagement of said member with said stop means to adjust said valve out of said position.

9. A hydraulic transmission for a reversibly movable machine tool element comprising, in

combination, a hydraulic motor for translating said element, supply lines opening to opposite sides of said motor, a variable delivery pump having a pressure discharge line and an adjustable control member movable to vary the pump displacement through a range from zero to maximum, spring means acting on said member in a direction to increase said pump displacement, means responsive to the discharge pressure of said pump for substantially counter balancing said member against the pumping reaction, an exhaust line, a control valve adjustable into position to connect said supply lines respectively to said discharge and exhaust lines to effect movement of said element in one direction, stop means for positively limiting the movement of said element in said one direction, a restricted flow orifice functionally interposed to receive fluid discharging from said motor through said exhaust line when said element is moving in said one direction, means including a hydraulic actuator acting on said member in opposition to said spring means and responsive through a control line to adjust said member to decrease the pump displacement upon an increase in the pressure at the inlet side of said orifice and to increase said displacement upon a decrease in the pressure at the inlet side of said orifice, said valve being adjustable into a second position to reverse the connections of said supply lines to said discharge and exhaust lines and thereby to effect movement of said element in the reverse direction, and means including a hydraulic actuator automatically operable in response to a predetermined pressure rise in said discharge line upon engagement of said element with said stop means to adjust said valve into said second position after a predetermined dwell period of said element.

10. A hydraulic transmission for a reversibly movable machine tool element comprising, in combination, a hydraulic motor for translating said element, supply lines opening to opposite sides of said motor, a variable delivery pump having a pressure discharge line and an adjustable control member movable to vary the pump displacement through a range from zero to maximum, spring means acting on said member in a direction to increase said pump displacement, means responsive to the discharge pressure of said pump for substantially counter balancing said member against the pumping reaction, an exhaust line, a control valve adjustable into position to connect said supply lines respectively to said discharge and exhaust lines to effect movement of said element in one direction, stop means for positively limiting the movement of said element in said one direction, means including a hydraulic actuator acting on said member in opposition to said spring means and operable in response to the pressure in said exhaust line to effect an increase in pump displacement upon a decrease in the exhaust pressure and a decrease in pump displacement upon an increase in the exhaust pressure, said valve being further adjustable to vary the back pressure in said exhaust line whereby to effect adjustment of said pump control member.

11. A hydraulic transmission for a reversibly movable machine tool element comprising, in combination, a hydraulic motor for translating said element, supply lines opening to opposite sides of said motor, a pump having a delivery line, an exhaust line, a rotary control valve adjustable into opposite limit positions to connect

said supply lines reversibly to said delivery and exhaust lines, a hydraulic valve motor having a reciprocatory plunger extending transversely of said valve and geared thereto, automatic means for adjusting said valve in timed relation to the movement of said element, a remote control pilot valve normally in intermediate neutral position to connect opposite ends of said valve motor to said exhaust line, and being adjustable oppositely out of said neutral position to connect said delivery and exhaust lines reversibly to opposite ends of said valve motor, and manually controlled means for selectively adjusting said pilot valve at any time within the range of control of said automatic means.

12. A hydraulic transmission for a reversibly movable machine tool element comprising, in combination, a hydraulic motor for translating said element, supply lines opening to opposite sides of said motor, a pump having a delivery line, an exhaust line, a rotary control valve adjustable into opposite limit positions to connect said supply lines reversibly to said delivery and exhaust lines, a hollow plunger reciprocable transversely of said valve and operatively geared to said valve, means for automatically adjusting said valve in timed relation to the movement of said element, and a hydraulic actuator for shifting said plunger.

13. A hydraulic transmission for a reversibly movable machine tool element comprising, in combination, a hydraulic motor for translating said element, supply lines opening to opposite sides of said motor, a pump having a delivery line, an exhaust line, a rotary control valve adjustable into opposite limit positions to connect said supply lines reversibly to said delivery and exhaust lines, a hydraulic valve motor having a reciprocatory hollow plunger extending transversely of said valve and geared thereto, automatic means for adjusting said valve in timed relation to the movement of said element, a remote control pilot valve normally in intermediate neutral position to connect opposite ends of said valve motor to said exhaust line, and being adjustable oppositely out of said neutral position to connect said delivery and exhaust lines reversibly to opposite ends of said valve motor, manually controlled means for selectively adjusting said pilot valve at any time within the range of control of said automatic means, and hydraulic dwell means for shifting said plunger when said pilot valve is in neutral position.

14. A hydraulic transmission for a reversibly movable machine tool element comprising, in combination, a hydraulic motor for translating said element, supply lines opening to opposite sides of said motor, a variable delivery pump having a pressure discharge line and an adjustable control member movable to vary the pump displacement through a range from zero to maximum, spring means acting on said member in a direction to increase said pump displacement, means responsive to the discharge pressure of said pump for substantially counter balancing said member against the pumping reaction, an exhaust line, a control valve adjustable into position to connect said supply lines respectively to said discharge and exhaust lines to effect movement of said element in one direction, stop means for positively limiting the movement of said element in said one direction, hydraulic means including a plunger acting on said member in opposition to said spring means in a direction to decrease the pump displacement and having a pres-

sure control line adapted for connection to said exhaust line, a one-way check valve interposed in said control line and permitting free flow of fluid to said hydraulic means, and a flow restriction interposed in said control line in parallel to said check valve.

15. A hydraulic power unit, comprising, in combination with a translatable element, a hydraulic motor for reversibly translating said element, a delivery pump having a pressure discharge line, a direction valve for connecting said line reversibly to opposite sides of said motor, means for positively limiting the movement of said element in one direction, a hydraulic actuator for reversing said valve to effect movement of said element in the other direction and having a pressure control line, and a dwell and relief valve operable by a predetermined rise in pressure in said discharge line upon engagement of said element with said means to relieve said discharge line for diversion of excess fluid and to connect said discharge line through an orifice to said control line whereby to effect a delayed reversal of said direction valve by said actuator.

16. A hydraulic power unit, comprising, in combination with a translatable element, a hydraulic motor for reversibly translating said element, a delivery pump having a pressure discharge line, a direction valve for connecting said line reversibly to opposite sides of said motor, means for positively limiting the movement of said element in one direction, a hydraulic actuator for reversing said valve to effect movement of said element in the other direction, and having a pressure control line, a dwell valve normally conditioned to connect said control line to exhaust and being operable by a predetermined rise in pressure in said discharge line upon engagement of said element with said means to connect said discharge line through a flow restriction to said control line, and means normally closed during the translation of said element and operative to relieve excess fluid from said discharge line to limit said pressure rise to a substantially constant maximum, whereby to effect a delayed reversal of said direction valve by said actuator.

17. A hydraulic power unit, comprising, in combination with a translatable element, a hydraulic motor for reversibly translating said element, a delivery pump having a pressure discharge line, a direction valve for connecting said line reversibly to opposite sides of said motor, means for positively limiting the movement of said element in one direction, a hydraulic actuator for reversing said valve to effect movement of said element in the other direction, and having a pressure control line, a dwell valve normally conditioned to connect said control line to exhaust and being operable by a predetermined rise in pressure in said discharge line upon engagement of said element with said means to connect said discharge line through a flow restriction to said control line, means normally closed during the translation of said element and operative to relieve excess fluid from said discharge line to limit said pressure rise to a substantially constant maximum, whereby to effect a delayed reversal of said direction valve by said actuator, and a pilot valve operable independently of said actuator to effect reversal of said direction valve.

18. A hydraulic power unit comprising, in combination, a hydraulic motor, a variable delivery pump having an adjustable member for controlling the pump displacement, spring means in said pump acting on said member in a direction to

increase the displacement of said pump, means for driving said pump, hydraulic means for counterbalancing said member against the hydraulic reaction of said pump, hydraulic means acting on said member in a direction to decrease the displacement of said pump, and control valve means selectively adjustable to direct fluid from said pump to said motor to effect operation of said motor, or to connect said pump to said last-mentioned hydraulic means whereby to adjust said pump into position of substantially zero displacement.

19. A hydraulic power unit comprising, in combination, a reversible hydraulic motor, a variable delivery pump, means for driving said pump, said pump having an actuating member adjustable to vary the displacement from zero to maximum, means for counterbalancing said member against the hydraulic pressure reaction in said pump, an exhaust line including a passage with a flow restriction and a less restricted passage about said pump reversibly to opposite ends of said motor and for directing fluid discharging from said motor to said exhaust line selectively through said first named passage or through said less restricted passage about said restriction, respectively for feed and rapid traverse operation, and pressure responsive means selectively connectible by said valve means to said exhaust line ahead of said restriction for automatically adjusting said member to adjust the displacement of said pump to obtain respectively maximum displacement during rapid traverse, and substantially constant reduced displacement during feed operation.

20. A hydraulic transmission for a reversibly movable machine tool element comprising, in combination, a hydraulic motor for translating said element, supply lines opening to opposite sides of said motor, a pump having a delivery line, an exhaust line, a rotary control valve adjustable into opposite limit positions to connect said supply lines reversibly to said delivery and exhaust lines, a plunger reciprocable transversely of said valve and operatively geared to said valve, means for automatically adjusting said valve in timed relation to the movement of said element, a hydraulic actuator having a control line and a piston movable in response to the supply of pressure fluid through said control line out of idle position into end engagement with said plunger to move said plunger from the adjacent end position into an intermediate position, load and fire mechanism operable upon movement of said plunger into said intermediate position to continue movement thereof into the opposite end position, and means automatically operable upon a predetermined rise in pressure in said delivery line to supply fluid under pressure at a graduated rate to said control line.

21. A hydraulic transmission for a reversibly movable machine tool element comprising, in combination, a hydraulic motor for translating said element, supply lines opening to opposite sides of said motor, a pump having a delivery line, an exhaust line, a rotary control valve adjustable into opposite limit positions to connect said supply lines reversibly to said delivery and exhaust lines, a plunger reciprocable transversely of said valve and operatively geared to said valve, means for automatically adjusting said valve in timed relation to the movement of said element, a hydraulic actuator having a control line and a piston movable in response to the supply of pressure fluid through said control line out of idle

position into end engagement with said plunger to shift the latter, and means automatically operable upon a predetermined rise in pressure in said delivery line to supply fluid under pressure at a graduated rate to said control line.

22. A hydraulic power unit comprising, in combination, a differential hydraulic motor having unequal oppositely acting pressure areas, a variable delivery pump having an adjustable member for controlling the pump displacement, spring means in said pump acting on said member in a direction to increase the displacement of said pump, means for driving said pump, hydraulic means for counterbalancing said member against the hydraulic reaction of said pump, second hydraulic means acting on said member in a direction to decrease the displacement of said pump, a flow restriction adapted to be interposed in an exhaust line, and control valve means having different selective conditions of adjustment, in one of which said pump is connected to opposite sides of said motor and said second hydraulic means is connected to said exhaust line, in another of which said pump is connected to one side of reduced pressure area of said motor and the other side of said motor and said second hydraulic means are connected to said exhaust line, in another of which said pump is connected to said other side of said motor and said one side of said motor and said second hydraulic means are connected to said exhaust line in advance of said flow restriction, and in another of which said one side of said motor is blocked and said other side

of said motor is connected to said exhaust line and said pump is connected to said second hydraulic means.

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