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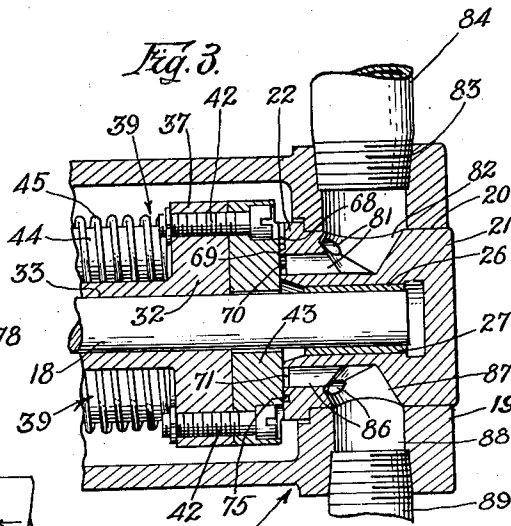
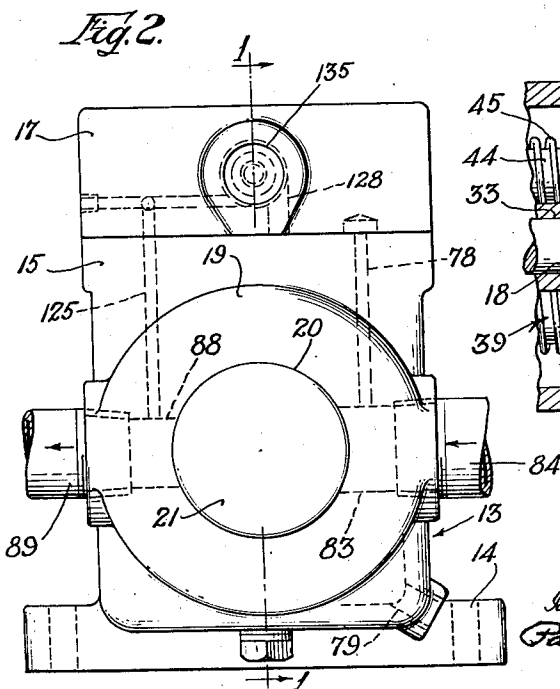
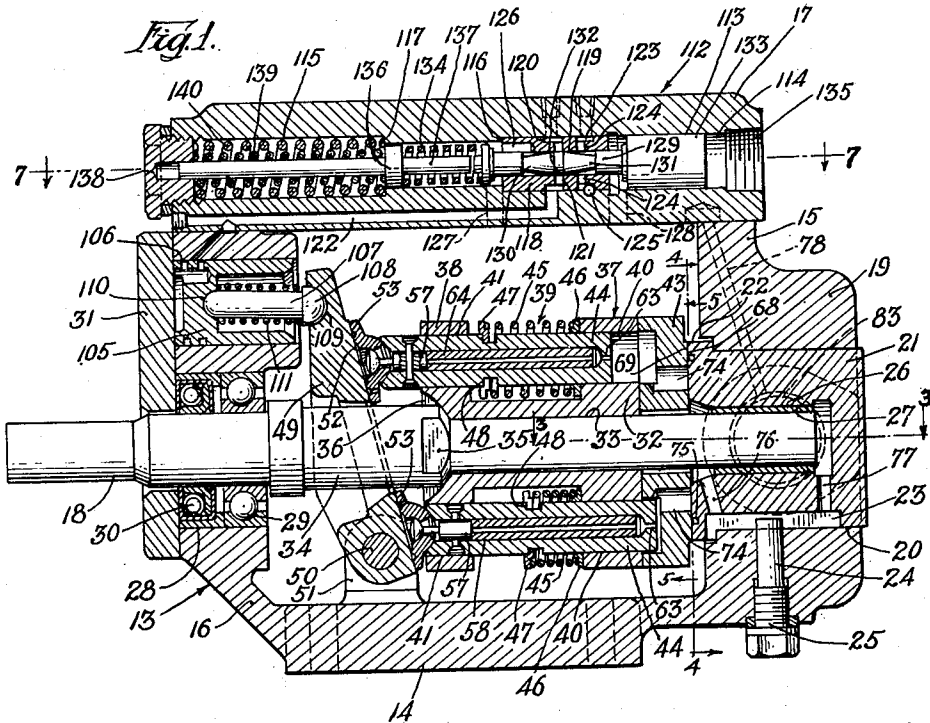
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2,299,234

HYDRAULIC PUMP AND CONTROL MEANS THEREFOR

Filed June 9, 1937

3 Sheets-Sheet 1



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3 Sheets-Sheet 2

Fig. 4.

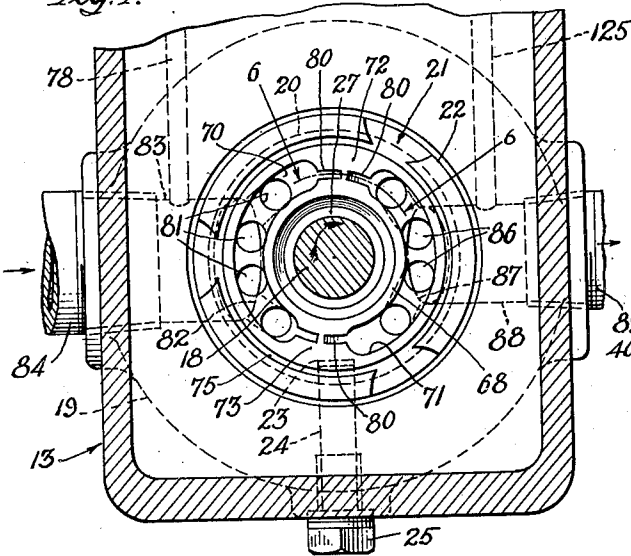


Fig. 5.

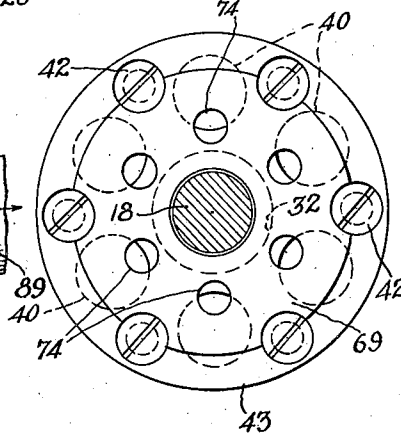


Fig. 6.

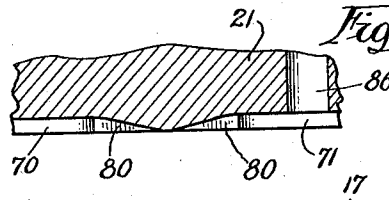
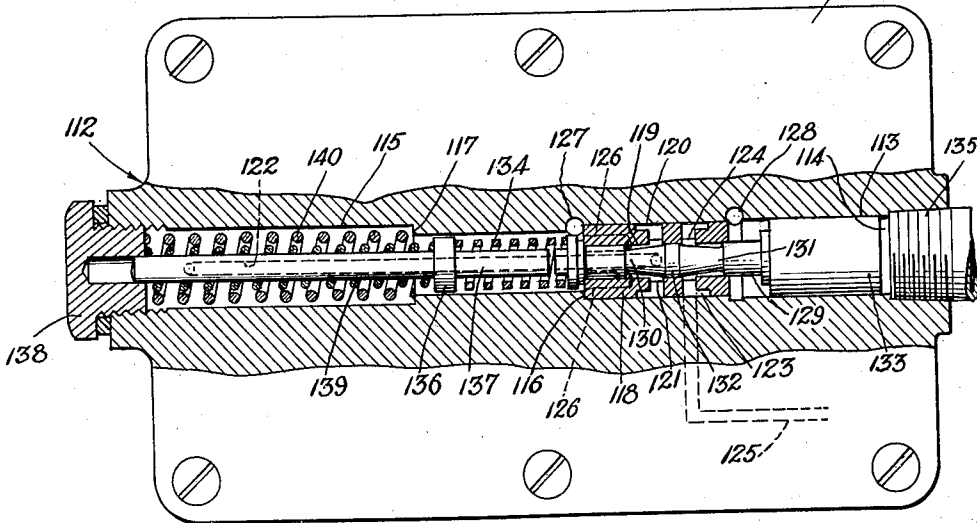


Fig. 7.



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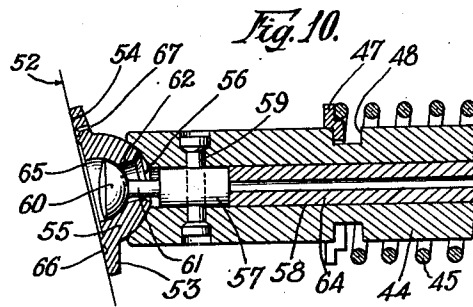
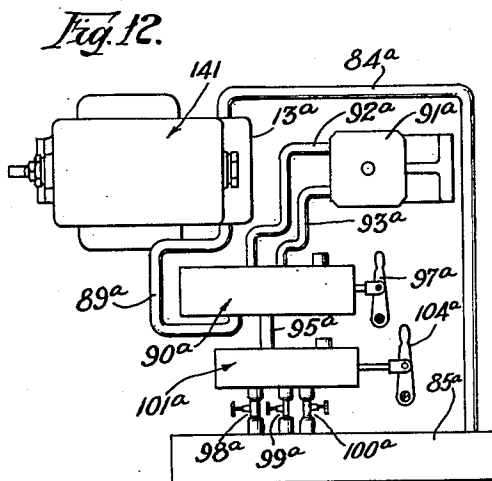
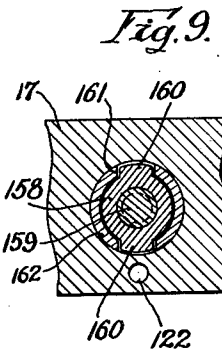
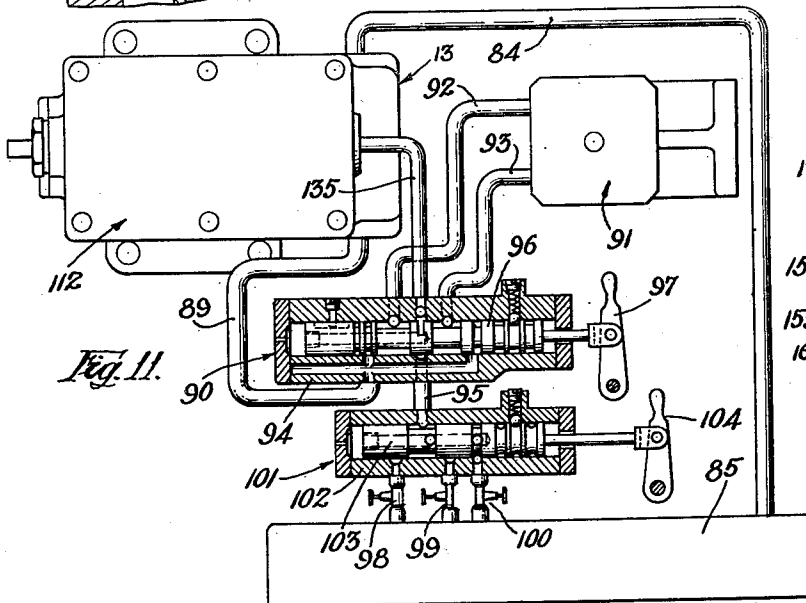
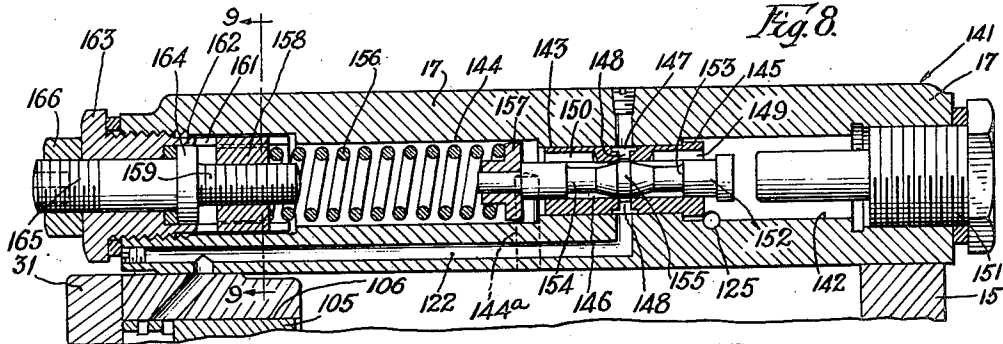
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HYDRAULIC PUMP AND CONTROL MEANS THEREFOR

Filed June 9, 1937

3 Sheets-Sheet 3



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UNITED STATES PATENT OFFICE

2,299,234

HYDRAULIC PUMP AND CONTROL MEANS
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Application June 9, 1937, Serial No. 147,178

9 Claims. (Cl. 103—162)

The present invention relates to improvements in hydraulic pumps and control means therefor, and particularly to a new and improved pump and control means therefor adapted for high speed and pressure operation.

An important object of the invention is to provide a pump with a novel hydraulic control for automatically varying the delivery as required within the range of volumetric capacity to maintain a predetermined delivery pressure.

A more specific object is to provide a new and improved variable delivery pump having a volume control element adjustable by a motor under the control of a metering valve responsive to a control pressure.

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

In the accompanying drawings,

Figure 1 is an axial sectional view, taken along line 1—1 of Fig. 2, of a pump with control means embodying the features of our invention.

Fig. 2 is an end elevational view.

Fig. 3 is a fragmentary sectional view of the pump valve taken along line 3—3 of Fig. 1.

Fig. 4 is a fragmentary transverse sectional view taken substantially along line 4—4 of Fig. 1, and showing the face of the stationary valve member.

Fig. 5 is a fragmentary transverse sectional view taken substantially along line 5—5 of Fig. 1, and showing the face of the movable valve member.

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

Fig. 7 is a horizontal sectional view taken substantially along line 7—7 of Fig. 1, but with the volume control valve in a different position of adjustment.

Fig. 8 is a longitudinal sectional view of a modified form of volume control valve.

Fig. 9 is a transverse sectional view taken along line 9—9 of Fig. 8.

Fig. 10 is an enlarged fragmentary sectional view of one of the pump pistons.

Fig. 11 is a diagrammatic view of one form of hydraulic circuit including the pump illustrated in Fig. 1.

Fig. 12 is a diagrammatic view of a modified form of hydraulic circuit including the pump with the volume control valve of Fig. 8.

While the invention is susceptible of various modifications and alternative constructions, we have shown in the drawings and will herein describe in detail the preferred embodiment, but it is to be understood that we 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, the hydraulic pump, constituting the exemplary embodiment of the invention, comprises a closed pump housing 13 which may be of any suitable form or construction, and which in the present instance is generally rectangular in shape, and has a bottom wall 14 adapted to be bolted or otherwise secured to a suitable support (not shown). The housing 13 also has opposite end walls 15 and 16, and is closed at the top by a removable cover plate 17.

A suitable drive shaft 18 extends longitudinally through the housing 13, and is journaled at opposite ends in the walls 15 and 16. Integral with the wall 15 is a large external circular boss 19 having an axial bore 20 counterbored at the inner end. A stationary valve plug 21, in the form of a single cylindrical block, is tightly pressed into the bore 20, and is located axially by a peripheral end flange 22 seated in the counterbore. To locate the plug 21 angularly, it is formed with a longitudinal groove 23 engaging a pin 24 on the inner end of a screw 25 threaded radially into the boss 19. Formed in the plug 21 is an axial bore 26 which is closed at the outer end, and which is open and flared at the inner end. A bushing 27 is fitted in the bore 26, and supports one end of the shaft 18.

The opposite end wall 16 is formed with a bore 28 in axial alignment with the bushing 27. Preferably, an antifriction bearing 29 is mounted in the bore 28 to support the shaft 18 which extends therefrom externally of the housing 13 for connection to a suitable source of power (not shown). To prevent the entry of air, dust and other foreign matter through the bearing 29 into the housing 13, an oil seal 30 is mounted in the outer end of the bore 28, and is held in place by an end plate 31 bolted to the outside of the wall 16.

Secured to the shaft 18 for rotation therewith is a cylinder block or body 32. In the present instance, the body 32 is formed with an axial bore 33 slidably receiving the shaft 18, and is located axially thereon by abutment against a shaft enlargement 34. The end of the enlargement 34 is flattened on opposite sides to define a key 35 which interfits separably with a diametrical notch 36 in the adjacent end of the body 32. Prefer-

ably, the root of the notch 36 and the end face of the key 35 are arcuate in shape.

The body 32 is in the form of a spool having a peripheral cylinder flange 37 and an axially spaced guide flange 38, and defining an intermediate spring space 39. A plurality of parallel bores 40, six in the present instance, open through the flange 37, and constitute pump cylinders annularly arranged in uniformly spaced relation about the shaft 18. Similarly, the flange 38 is formed with guide bores 41 in axial alignment with the bores or cylinders 40. Rigidly secured as by means of bolts 42 to the outer end face of the flange 37 and coextensive in diameter therewith is a circular valve plate 43 which closes the pressure ends of the cylinders 40, and which preferably is formed with circular recesses defining slightly enlarged closed end extensions of the cylinders.

A plurality of pump plungers or pistons 44 are reciprocally disposed respectively in the open ends of the cylinders 40 and are slidably guided at their outer ends in the bores 41. The pistons 44 are urged outwardly on their suction or intake strokes by individual compression springs 45 located within the annular space 39. Each of the springs 45 seats at one end against a ring retainer 46 positioned against the flange 37, and at the other end against a key retainer 47 engaging a peripheral groove 48 in the associated piston 44.

The pistons 44 are operable in uniformly timed sequence, with a variable stroke as hereinafter described, by a swash or cam plate 49 which is journaled on a pin 50 for angular or rocking adjustment into different inclined positions. The pin 50 is pressed in spaced lugs 51 in the housing 13 and is perpendicular to the axis of the shaft 18. The plate 49 has a flat annular cam face 52 which freely encircles the shaft enlargement 34 and which is engaged by floating shoes 53 operatively connected to the outer ends of the pistons 44.

Each of the shoes 53 is in the form of a disk having a flat annular bearing face 54 of ample size in sliding engagement with the cam face 52, and having a hollow central spherical projection 55 seating for universal movement in a complementary socket 56 in the outer end of the associated piston 44. A retainer 57 is loosely fitted in the outer end of an axial bore 58 in the piston 44, and is secured therein by a transverse lock pin or rivet 59. The retainer 57 has a semispherical head 60 which engages the interior of the projection 55 to confine the shoe 53 adjustably in the socket 56, and which is connected by a reduced stem 61 extending freely through a central opening 62 in the projection.

The bore 58 opens at the inner end through a small aperture 63 to the cylinder 40, and hence provides for a longitudinal pressure passage permitting the flow of lubricant to the bearing surfaces of the shoe 53. As one factor affecting quietness in pump operation, the lubricant passage should be small. Since it is impracticable to drill the long bore 58 sufficiently small in hard metal, such as the piston material, a seamless tube 64 is tightly fitted therein between the retainer 57 and the aperture 63, and defines the actual pressure passage. Fluid or oil from the closed end of the cylinder 40 passes through the aperture 63, the tube 64 and about the retainer 57 to the bearing surfaces of the socket 56 and ball projection 55. Fluid or oil also passes through the opening 62 to the bearing surfaces between the plate 49 and the shoe 53.

The bearing surfaces between the plate 49 and the shoe 53 and between the projection 55 and the socket 56 are closely lapped to provide a hydraulic seal, and are hydraulically balanced against the mechanical reaction on the piston to the hydraulic pressure in the cylinder 40, so as to maintain at all times good bearing engagement with a proper lubricating film. The bearing face 54 of the shoe 53 is relieved at the inner edge by an annular notch 65, and is formed with a concentric drain groove 66 relieved through a plurality of drain apertures 67. The central portion of the bearing face 54, circumscribed by the inner edge of the groove 66, is under pressure, and is substantially equal in area to the pressure end of the piston 44 so as to counterbalance the outward thrust of the hydraulic pressure in the cylinder 40. The pressure area of the face 54 may be equal to the piston end area, but preferably is slightly larger to balance part of the outward pressure of the spring 45. Since the groove 66 prevents outward spreading of the pressure film, the full wearing surface of the face 54 can be much larger than the pressure balanced area without interfering at any time with good bearing contact. The effective pressure area of the socket 56 is somewhat less by reason of the opening 62 than that of the piston 44, but is sufficiently large to obtain substantially a hydraulic balance, the lubricating film being maintained under a slight excess pressure to assure good bearing contact.

The valve stator 21 and the plate 43 have annular end faces 68 and 69 in bearing engagement, and constitute a rotary end face valve for controlling the admission and discharge of fluid respectively to and from the pump units. Since the stator 21 is in the form of a heavy plug mounted in the large boss 19, the stationary valve face 68 is not subject to deflection under heavy pressure.

The valve stator 21 (see Fig. 4) is provided with two arcuate suction and delivery ports 70 and 71 separated by diametrically spaced land areas 72 and 73 and arranged in concentric relation to the shaft 18. These ports are located at opposite sides of an axial plane intersecting the land areas and perpendicular to the pivotal axis of the swash plate 49. Formed in the valve face 69 of the plate 43 are a plurality of ports 74 which open respectively to the closed ends of the pump cylinders 40, and which are annularly arranged in a circle of the same curvature as the ports 70 and 71 for movement successively across the latter upon rotation of the cylinder body 32. The ports 70, 71 and 74 are offset from the centers of the cylinders 40, and preferably are located at the innermost side areas thereof adjacent the shaft 18. As a result, the contacting areas of the valve faces 68 and 69 can be made small to lessen the likelihood of manufacturing errors or irregularities and the chance of valve leakage, and are located closely to the center of rotation to reduce the peripheral speed. Also, any air collecting in the cylinders 40 will be separated from the liquid by centrifugal force, and pass out with the pump delivery.

The valve faces 68 and 69 are hydraulically balanced against the pressure in the minimum number of cylinders 40 on the pressure side of the pump to maintain a proper oil film and good contact. Since the cylinders under pressure are alternately two and three, the area of the valve face 68 on the pressure side, including the port 71, is made substantially equal to the aggregate area of the ends of two pistons 44. To obtain a bear-

ing surface larger than the balancing pressure area, an annular concentric drain groove 75 is formed in the face 68 about the ports 70 and 71 to prevent spreading of the pressure film, and is relieved at spaced points to the periphery of the plug 21. The inner peripheral edge of the valve face 68 is relieved to the bore 26, and the liquid draining from the valve faces into the bore serves to lubricate the bushing 27. Liquid collecting in the bore 26 at opposite ends of the bushing 27 is free to pass through drain holes 76 and 77 to the locating spline groove 23, and from there into the housing 13. Liquid collecting in the housing 13 is free to return through a passage 78 to the pump inlet. If the pump is mounted above the level of the source of liquid supply, the passage 78 may be closed, and the housing 13 may be drained through a bottom opening 79 connected to the source.

It will be evident that when the swash plate 49 is angularly disposed, rotation of the cylinder body 32 will cause the pistons 44 to be reciprocated in timed sequence and with a simple harmonic motion, and that each piston will be moving through its suction and pressure strokes while the associated cylinder 40 is in communication respectively with the inlet and delivery ports 70 and 71. As the pistons 44 of each set of diametrically opposed pistons reach top and bottom dead centers respectively, the associated cylinders 40 will be closed by the land areas 72 and 73, and at these times only two cylinders will be open to either of the valve ports 70 and 71. At all other times, three cylinders 40 will be open to each of the valve ports 70 and 71. Consequently, the pump will discharge continuously, and the rate of discharge is substantially constant for any given setting of the swash plate 49. If the angular position of the swash plate 49 is changed, the length of the piston stroke will be adjusted to adjust the rate and volume of pump delivery. When the plate 49 is adjusted into a plane perpendicular to the shaft 18, no reciprocation will be imparted to the pistons 44.

Preferably, the suction and delivery ports 70 and 71 are angularly advanced in relation to the top and bottom dead center positions of the pistons 44 so as to effect a pressure equalization in each cylinder 40 just before the start of both the pressure and suction strokes. Each of the cylinder ports 74 will be closed by the land area 72 during the initial part of the pressure stroke to cause the pressure therein to be built up approximately to the pump delivery pressure before free communication with the port 71. Similarly, each port 74 will be closed by the land area 73 during the initial part of the suction stroke so that the pressure will be expanded approximately to the suction pressure effective in the port 70 before free communication therewith.

To permit such pressure equalization, the land areas 72 and 73 are somewhat wider than the ports 74 so that each cylinder 40 is closed over a predetermined degree of revolution. For fixed operating conditions and upon making proper allowance for leakage past the valve faces and pistons, the angular advance of the ports 70 and 71 may be accurately calculated to obtain a quietly operating pump. However, the operating conditions are subject to variations, such, for example, as changes in delivery pressure, speed, displacement, temperature influences and the kind and condition of the fluid being handled. These changes influence the pump leakage and the pressure gradients. The valve ports 70 and

71 are therefore advanced to time the opening of the cylinder ports 74 for average or mean operating conditions.

Means may be provided for graduating even the small pressure changes which will be present when the variable pressures in the effective range deviate from the mean operating pressure. This means consists of short restricted bleed grooves 80 in the valve face 68 (see Figs. 4 and 6) opening to and constituting extensions of the ends of the arcuate ports 70 and 71. The grooves are so graduated that the cylinder ports 74 will not open abruptly to the ports 70 and 71, thereby avoiding sudden shock even if the cylinder pressure is not equalized with that of the stationary port with which it is about to communicate.

The gradual pressure changes in passing from either to the other of the ports 70 and 71 also have the important function of preventing the piston shoes 53 from clicking. During movement through the intake zone, each shoe 53 is held against the cam plate 49 with a light initial pressure determined by the spring 45. In the pressure zone, the spring pressure is supplemented by the hydraulic pressure acting through the piston 44 so that the oil film between the cam plate 49 and the shoe 53 and between the latter and the piston is compressed under a heavy pressure. By reason of the advance of the ports 70 and 71 and/or the provision of the bleed grooves 80, the pressure on the film is alternately gradually relieved and applied, thereby placing the shoe 53 in substantial balance before each port opening. Sudden pressure surges tending to effect separation of the shoe 53 from the cam plate 49 at transition points in the operation are consequently avoided.

As noted, one of the features of the present pump is quietness in operation. The hydraulic balance of the piston shoes 53 and the valve 68, 69 against the mechanical reaction of the hydraulic pressure contributes to this result. The elimination of pressure surges or impulses, by reason of the angular advance of the ports 70 and 71 and the provision of the bleed grooves 78, also reduces noise in operation. As another important factor in avoiding noise, the volume of pressure liquid in the closed ends of the cylinders 40 at the end of the pressure stroke is held to a minimum. To this end, the piston clearance in the valve plate 43 is made as small as is practically convenient, and the pressure passages through the pistons 44 are reduced in size by means of the tubes 64. Also, the pivot pin 50 is located at a point adjacent the axis of each piston 44 as the latter moves into top dead center position at the end of the pressure stroke. As a result, the inward limit of movement of the pistons 44 is substantially constant and close to the inner ends of the cylinders 40, while the outward limit of movement is variable in accordance with changes in the angular position of the swash plate 49.

The inlet port 70 may be connected in any suitable manner to the source of liquid supply, and in the present instance is in communication at the bottom or root with a plurality of longitudinal bores 81 in the valve plug 21. The outer ends of the bores 81 are intersected by a recess 82 formed in the periphery of the plug, and registering with a passage 83 opening through the peripheral wall of the boss 19. A line 84 connects the passage 83 with a source of supply, such as a sump or reservoir 85. The delivery port 71 is in communication at the bottom or

root with a plurality of longitudinal bores 86 formed in the plug 21. A recess 87 in the periphery of the plug 21 intersects the bores 86, and is in registration with an outlet passage 88 opening through the boss 19 and connected to a pressure line 89.

The pump is adapted for various purposes, and is particularly useful as a source of power in a hydraulic transmission system or operating circuit. In Fig. 11, illustrating a hydraulic machine tool circuit, the pressure line 89 is connected to the inlet of a reversing valve 90 for controlling the operation of a hydraulic motor 91. The motor may be of any suitable character, and is herein shown as of the reversible rotary type having two liquid supply and discharge lines 92 and 93. The valve 90 also may be of any suitable type, and for purposes of illustration is herein disclosed as comprising an outer casing 94 connected to the lines 89, 92 and 93, and to an exhaust line 95 discharging to the sump 85. A valve member 96, of the spool type, is reciprocable in the casing 94 by means of a hand lever 97, and is adapted in one end position to connect the line 89 to the line 92 and the line 93 to the line 95, in the other end position to reverse these connections, and in an intermediate position to block the lines 89, 92 and 93.

An adjustable flow restriction means is interposed in the line 95 to build up a low pressure controlling the operation of the motor 91. More particularly, the speed of the motor 91 is subject to variation by adjusting the exhaust flow restriction. This restriction may be of any suitable character, and in the present instance consists of three parallel orifices 98, 99 and 100 selectively available under the control of a valve 101. Assuming the motor 91 to be connected to drive a reciprocatory machine tool carriage (not shown), the largest orifice 98 is adapted to effect rapid traverse, the smallest orifice 99 is adapted to effect a slow feed, and the intermediate orifice 100 is adapted to effect a relatively fast feed. It will be understood, that each orifice may also be subject to individual adjustment. The selector valve 100 may be of any suitable character, and is herein shown as comprising a casing 102 and a valve member 103 adjustable by a hand actuator 104 to connect the line 95 to the sump 85 through any one of the orifices 98, 99 and 100.

Provision is made for automatically controlling the position of the swivel plate 49 to maintain a predetermined control pressure substantially constant regardless of volume requirements. Thus, the plate 49 is automatically adjusted to reduce the pump delivery upon an increase in the control pressure, and to increase the delivery upon a decrease in the pressure. As applied to the hydraulic circuit of Fig. 11, the pump is controlled by the fluid pressure in the line 95 between the motor 91 and the selective orifices 98, 99 and 100. If the orifice restriction is adjusted, the pressure in the line 95 will effect a corresponding variation in the pump delivery. For any selected orifice restriction, tending pressure variations in the line 95 will effect compensatory adjustments in the pump delivery so that the pressure drop across the restriction regardless of motor torque will remain practically uniform. The speed of the motor 91 is therefore entirely under the control of the orifice restriction.

In its preferred form, the pump adjusting means comprises a control piston 105 acting on the free end of the swash plate 49 in a direction to level off the latter against the hydraulic re-

action in the cylinders 40 and the action of the piston return springs 45. The control piston 105 is reciprocable in a cylinder 106 opening through the wall 16, and closed at the outer end by the cover or stop plate 31. The cylinder 106 extends parallel to the shaft 18, and preferably is located diametrically opposite the pin 50 and outside the circle of revolution of the cylinders 40 so that the thrust of the piston 105 against the plate 49 will have a moment arm considerably longer than that of the aggregate opposing forces resulting from the hydraulic reaction through the pistons 44. The diameter of the piston 105 and the moment arm of the force exerted thereby determine the pressure required in the cylinder 106 in relation to the pump delivery pressure to balance the cam or swash plate 49. By providing a comparatively large diameter and a relatively long moment arm as shown, the balancing pressure required in the cylinder 106 will be much lower than the corresponding pump delivery pressure.

The control piston 105 is operatively connected to the swash plate 49 by a pin 107. One end of the pin 107 is formed with a spherical head 108 seated for universal pivotal movement in a complementary socket 109 in the free end of the plate 49, and the other end is rounded and seated in a complementary socket 110 in the end wall of the piston 105. A coiled compression spring 111 acts to maintain the head 108 within the socket 109.

The piston 105 is controlled by a metering valve 112 which is automatically responsive to the pressure in the line 95 in advance of the selected orifice restriction. The valve 112 is operable to admit fluid under pressure to the closed end of the cylinder 106 or to exhaust fluid therefrom respectively upon an increase or decrease in the pressure in the line 95, whereby the speed of the hydraulic motor 91 is maintained constant regardless of varying load conditions. For example, if the load is increased, which results in increased leakage, the motor 91 will tend to slow down and hence exhaust less fluid, but the resulting drop in the pressure built up by the orifice restriction will immediately effect a compensatory increase in the pump delivery.

Preferably, the valve 112 is built into the cover 17, and comprises a bore 113 opening through and enlarged at opposite ends to define a pressure chamber 114, a spring chamber 115 and oppositely facing shoulders 116 and 117. A valve sleeve 118 is fixed in the bore 113 against the shoulder 116, and is formed with an axial valve passage 119. A peripheral groove 120 opens through radial ports 121 to the passage 119, and is in registration with a passage 122 connecting with the closed end of the cylinder 106. The sleeve 118 is also formed at one side of the groove 120 with a second peripheral groove 123 opening through a plurality of radial inlet ports 124 to the valve passage 119, and registering with a pressure supply passage 125 in the cover 17 and wall 15 connecting with the pump outlet passage 88. Formed in the end of the sleeve 118 at the opposite side of the groove 120 and longitudinally intersecting the valve passage 119 to define exhaust ports are a plurality of bores 126 opening to the reduced intermediate section of the bore 113. Two relief passages 127 and 128 open from the bore 113 at opposite ends of the sleeve 118 through the cover 17 to the interior of the pump housing 13 from where spent fluid may be discharged through the passage 78 to the pump inlet passage 83 as described.

Reciprocally disposed in the sleeve 118 is a

valve member 129 of the spool type having two spaced peripheral grooves 130 and 131 opening respectively to the inlet and exhaust ports 124 and 126, and separated by a collar 132 adapted in intermediate position to block the control ports 121. To obtain accurate metering, the roots of the grooves 130 and 131 are tapered conically in opposite directions to restrict gradually the flow connection with the ports 121 as the valve member 129 is moved toward neutral position.

The valve member 129 is operable by a pressure responsive element 133 and resilient pressure means 134 acting in opposition. In the present instance, the element 133 consists of a piston in abutment with one end of the valve member 129, and reciprocates in the pressure chamber 114. A control line 135 opens from the outer end of the chamber 114 to the interior of the reversing valve casing 94 for connection with the exhaust line 95 when the motor 91 is in operation, and with the pump discharge line 89 when the valve member 96 is in stop position.

The resilient pressure means 134 consists of a square wire coil compression spring engaging the other end of the valve member 129 and seating against a collar 136 slidably guided in the intermediate section of the bore 113. The collar 136 is integral with a pin 137, one end of which extends through the spring 134 into closely spaced relation to the valve member 129, and the other end of which is guided in a screw plug 138 closing the outer end of the spring chamber 115. The spring 134 is maintained under normal compression by a second spring 139 encircling the pin 137 and seating under compression at opposite ends against the collar 136 and the plug 138. The two springs 134 and 139 have different natural periods of frequency, so that any vibrations tending to result therein from pump pulsations are out of synchronism and neutralize or offset each other. Hence, the valve member 129 is maintained in sensitive balance without fluttering. A heavy coil compression shock spring 140 encircles the spring 139 and normally abuts at opposite ends against the shoulder 117 and the plug 138. The inner end convolution of the spring 140 projects inwardly for engagement by the collar 136 in the event of sudden and extended movement of the valve member 129 by the piston 133.

The valve 112 is of the graduated metering type, and when in control is normally either closed or slightly open to permit only a restricted flow. The flow restriction in the valve 112 results in a large pressure drop in the passage 122 when connected to the passage 125 so that the control pressure on the piston 105 is reduced and always less than the pump delivery pressure. Only small valve movements are required and only small pressure changes are necessary to move the valve member 129. The control is sensitive and does not reflect or develop pulsations at critical motor speeds.

In operation, when the orifice restriction in the line 95 is set for maximum opening, as by connecting the orifice 98, a relatively low pressure drop is maintained, and hence the liquid or oil exhausting from the motor 91 is practically free to return without restriction to the sump 85. Since no effective pressure will prevail in the line 135, the valve member 129 will be adjusted by the springs 134 and 139 into its extreme right hand position as shown in Fig. 7, forcing the piston 133 to the right, thereby fully exhausting the control cylinder 106 and causing the pump to deliver at maximum capacity.

If the orifice restriction is now gradually increased, as by connecting the orifice 100 for fast feed, a pressure will build up in the control line 135, and act through the piston 133 to force the valve member 129 and pin 137 to the left until the collar 136 engages the shock spring 140 as shown in Fig. 1. The valve member 129 will continue movement against the action of the spring 134 to connect the passages 122 and 125, whereupon the swash plate 49 will be adjusted to reduce the delivery of the pump to a point where the resultant pressure in the line 135 will maintain the valve 112 in closed position.

After the piston 105 is away from the stop plate 31, the pressure in the line 135 will be maintained substantially constant regardless of gradual changes in the size of the orifice restriction. For example, if the orifice restriction is still further increased, as by connecting the smallest orifice 99, the same pressure in the line 135 will be maintained by a corresponding reduction in the volume of pump delivery to obtain a slow feed. In general, the pressure drop across the orifice restriction will be constant for all sizes of orifices, and the volume of liquid flow there-through will be constant for any one size but will vary with different sizes. Hence, the speed of the motor 91 for any given orifice setting will be constant regardless of varying load conditions and resulting variations in leakage in the pressure side of the system, and will be determined entirely by the size of orifice employed. The delivery of the pump is fully utilized at all times in doing useful work, and is varied as required without the use of relief valves and other accessories commonly employed for by-passing excess fluid.

The leveling adjusting of the swash plate 49 is gradual by reason of the metering characteristics of the valve 112. However, if the orifice restriction should be changed or closed suddenly, a pressure impulse would be transmitted through the line 135 and cause the valve member 129 acting through the pin 137 to compress the shock spring 140, thereby permitting the pressure fluid to level off the plate 49 immediately. Conversely, if there should be a sudden pressure drop in the line 135, the spring 139 would urge the valve member 129 far to the right to effect an immediate and substantial increase in pump delivery. Upon movement of the reversing valve 90 into neutral position, the motor lines 92, 93 will be blocked, and the control line 135 will be disconnected from the exhaust line 95 and connected to the pump discharge line 89. Since the line 89 is also blocked, the delivery pressure of the pump will rise and effect movement of the swash plate 49 into level position to stop delivery.

A modified form of metering valve 141 for controlling the volume of pump delivery in response to the pump delivery pressure is illustrated in Fig. 8. The valve 141 is built into the pump cover 17, and comprises a bore opening longitudinally therethrough and consisting of three sections 142, 143 and 144. A valve sleeve 145 is fixed in the intermediate bore section 143, and is formed with an axial passage 146, an external peripheral groove 147 opening through radial ports 148 to the passage and communicating with the passage 122 leading to the control cylinder 106, and a plurality of longitudinal bores 149 and 150 in opposite ends longitudinally intersecting the valve passage 146 to define inlet and outlet ports opening respectively to the bore sections 142 and 144. A passage 144^a connects the bore section

144 to the interior of the casing 13. The bore section 142 is closed by a plug 151, and is connected to the pressure passage 125 leading from the pump outlet passage 88.

A valve member 152 is reciprocable in the passage 146, and is formed with spaced external peripheral grooves 153 and 154 defining an intermediate land 155 adapted in neutral position to close the ports 148. The roots of the grooves 153 and 154 are conically tapered to effect gradual valve restriction as the valve member 152 is moved toward closed position.

Fluid pressure in the bore section 142 tends to move the valve member 152 in a direction to connect the ports 148 and 149 to effect a decrease in the pump delivery. A coiled compression spring 156 opposes the pump delivery pressure, and tends to move the valve member 152 in a direction to connect the ports 148 and 150 to effect an increase in the pump delivery. The spring 156 is disposed in the bore section 144 and bears at one end against a shoe or retainer 157 fixed on the valve member 152, and at the other end against an adjustable stop 158. Preferably, the stop 158 consists of a nonrotatable nut threaded on an adjusting screw 159 and having longitudinal keys 160 slidable in slots 161 in a guide sleeve 162. The sleeve 162 is integral with a plug 163 threaded into and closing the outer end of the bore section 144. The screw 159 has a shoulder 164 disposed rotatably against the inner face of the plug 163, and has a shaft 165 extending axially through the latter for external adjustment. A lock nut 166 on the shaft 165 is available to clamp the screw 159 in adjusted position.

The pump with the modified valve 141 is included as the source of pressure fluid in the hydraulic machine tool circuit shown in Fig. 12. Since the pump is controlled directly from the delivery pressure, this circuit does not include a control line, such as the line 135, to the exhaust side of the system, but in all other respects it is like the circuit in Fig. 11, and hence like parts are identified by the same reference numerals plus the subscript *a*.

In operation, when the motor torque is low and the largest orifice 98^a is connected, the spring 156 will hold the valve member 152 to the right against the stop 151, so that the pump will deliver at maximum capacity. Upon an increase in pressure due to a load increase or a greater orifice restriction, as by connecting the orifice 99^a or orifice 100^a, the valve member 152 will be forced to the left against the action of the spring 156 to supply fluid under pressure to the control cylinder 106. Thereafter, the valve 141 will maintain a constant maximum pressure, subject to adjustment by varying the initial tension of the spring 156, and will cause the pump to supply whatever fluid the system is able to take at this pressure.

The pump per se is disclosed and claimed in our co-pending application Serial No. 348,904 filed July 31, 1940.

The hydraulic transmissions are disclosed and claimed in our co-pending application Serial No. 348,905 filed July 31, 1940.

We claim as our invention:

1. In a variable flow pump having a pressure delivery outlet and an adjustable volume control element, in combination, a pressure chamber having an inlet passage, a pressure responsive member in said chamber operatively connected to said element and movable to adjust said element, a valve casing having a control port open to said

passage, a pressure port connected to said outlet, and an exhaust port, a single valve member reciprocable in said casing and having a land area adapted to block said control port when in neutral position and having recesses adapted respectively to connect said control port to said pressure port and exhaust port upon adjustment in opposite directions, each recess having a tapered surface for gradually restricting the port connection as said valve member moves into neutral position, resilient means aligned with and acting directly on one end of said valve member to urge said valve member in a direction to connect said control and exhaust ports to increase the pump delivery, and means acting on the other end of said valve member for adjusting said valve member against said resilient means in response to the pressure in said outlet to connect said control and pressure ports to decrease the pump delivery.

2. In a variable flow pump having a pressure delivery outlet and an adjustable volume control element, in combination, a pressure chamber having an inlet passage, a pressure responsive member operatively connected to said element and movable in said chamber to adjust said element, a valve casing having a control port open to said passage, a pressure port connected to said outlet and an exhaust port, a valve member reciprocable in said valve casing and having a land area adapted to block said control port when said valve member is in neutral position and having metering recesses at opposite sides of said land area adapted respectively to connect the control port to said pressure port and exhaust port upon adjustment of said valve member in opposite directions out of neutral position, and resilient means tending to urge said valve member in one direction, said valve member being movable in the reverse direction in response to the pressure in said outlet.

3. In a variable flow pump having a pressure delivery outlet and an adjustable volume control element, in combination, a pressure chamber having an inlet passage, a pressure responsive member operatively connected to said element and movable in said passage to adjust said element, a valve casing having a control port open to said passage, a pressure port connected to said outlet, and an exhaust port, a valve member reciprocable in said valve casing and having a land area adapted to block said control port when said valve member is in neutral position and having recesses at opposite sides of said land area adapted respectively to connect the control port to said pressure port and exhaust port upon adjustment of said valve member in opposite directions out of neutral position, a pin reciprocable in said valve casing and having a peripheral collar, a compression spring acting against said collar to urge said pin toward said valve member, a second compression spring interposed between said valve member and said collar and tending to urge said valve member in a direction to connect said control and exhaust ports, a plunger operable in response to fluid pressure to move said valve member in a direction to connect said control and pressure ports, and a shock spring disposed for engagement by said collar and adapted to compress upon sudden movement of said valve member by said plunger.

4. In a variable flow pump having a pressure delivery outlet and an adjustable volume control element, in combination, a pressure chamber having an inlet passage, a pressure responsive

member operatively connected to said element and movable in said passage to adjust said element, a valve casing having a control port open to said passage, a pressure port connected to said outlet and an exhaust port, a valve member reciprocable in said valve casing and having a land area adapted to block said control port when said valve member is in neutral position and having metering recesses at opposite sides of said land area adapted respectively to connect the control port to said pressure port and exhaust port upon adjustment of said valve member in opposite directions out of neutral position, a compression spring acting on one end of said valve member tending to urge said valve member in one direction, and means for adjusting the normal compression of said spring, the other end of said valve member having a pressure area exposed to the pressure of the fluid from said outlet, whereby said valve member is movable against the resistance of said spring in response to pressure increases in said outlet.

5. In a variable flow pump having a pressure delivery outlet and an adjustable volume control element, in combination, a pressure chamber, a pressure responsive member in said chamber and operatively connected to said element to move the latter, a valve chamber having a fluid pressure inlet and an exhaust outlet, a valve member reversibly adjustable in said valve chamber to connect said pressure chamber either to said inlet or to said outlet or to block said pressure chamber, two springs having different natural periods of vibration operative in series on said valve member and tending to adjust said valve member in a direction to connect said outlet to said pressure chamber, and means for directing fluid under pressure to act on said valve member in opposition to said spring means.

6. In a variable flow pump having a pressure delivery outlet and an adjustable volume control element, in combination, a pressure chamber, a pressure responsive member in said chamber and operatively connected to said element to move the latter, a valve chamber having a fluid pressure inlet and an exhaust outlet, a valve member reversibly adjustable in said valve chamber to connect said pressure chamber either to said inlet or to said outlet or to block said pressure chamber, spring means operative on said valve member in a direction to connect said outlet to said pressure chamber, means for directing fluid under pressure to act on said valve member in opposition to said spring means, and a shock spring disposed in position for engagement by said valve member upon movement through a predetermined distance in response to the action of said fluid under pressure.

7. In a variable flow pump having a pressure delivery outlet and an adjustable volume control element, in combination, a pressure chamber having an inlet passage, a pressure responsive member in said chamber operatively connected to said element and movable to adjust said element, a valve casing having a pressure port for supplying fluid under pressure and an exhaust port, a valve member reciprocable in said casing and having a land area adapted to block said pressure chamber when in neutral position and having recesses adapted upon movement of said valve member respectively in opposite directions to connect said pressure chamber either to said pressure port or to said exhaust port, each recess having a tapered surface for gradually restricting the associated port connection as said valve

member moves toward and into neutral position, resilient means tending to urge said valve member in a direction to connect said pressure chamber to said exhaust port, and means for directing fluid under pressure to act on said valve member in opposition to said spring means and tending to urge said valve member in a direction to connect said pressure chamber to said pressure port.

8. In a variable flow pump having a pressure delivery outlet and an adjustable volume control element, in combination, a pressure chamber having an inlet passage, a pressure responsive member operatively connected to said element and movable in said chamber to adjust said element, a valve casing having a bore, a valve sleeve fitted in an intermediate portion of said bore and having a valve bore opening therethrough to opposite ends of said first mentioned bore, said sleeve having a control port opening to said passage, a pressure port adapted to receive fluid under pressure and an exhaust port, a valve member reciprocable in said valve bore and having an intermediate peripheral land area adapted to block said control port when said valve member is in neutral position, and having recesses at opposite sides of said land area adapted respectively to connect said control port to said pressure port or said exhaust port upon adjustment of said valve member in opposite directions out of neutral position, resilient means in one end of said first mentioned bore acting on said valve member to urge said valve member in one direction, and fluid pressure responsive means coacting with said valve member and slidable in the other end of said first mentioned bore, said other end of said first mentioned bore being adapted to receive fluid under pressure tending to urge said valve member in the reverse direction against the action of said resilient means.

9. In a variable flow pump having a pressure delivery outlet and an adjustable volume control element, in combination, a pressure chamber having an inlet passage, a pressure responsive member operatively connected to said element and movable in said chamber to adjust said element, a valve casing having a bore, a valve sleeve fitted in an intermediate portion of said bore and having a valve bore opening therethrough to opposite ends of said first mentioned bore, said sleeve having a control port open to said passage, a pressure port and an exhaust port, a valve member reciprocable in said valve bore and having an intermediate peripheral land area adapted to block said control port when said valve member is in neutral position, and having recesses at opposite sides of said land area adapted respectively to connect said control port to said pressure port or said exhaust port upon adjustment of said valve member in opposite directions out of neutral position, said pressure and exhaust ports opening respectively to opposite ends of said first mentioned bore and one end of the latter being adapted to receive fluid under pressure to act on one end of said valve member to urge said valve member in one direction, and adjustable resilient means in the other end of said first mentioned bore and acting on said valve member to urge said valve member in the reverse direction.

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