

July 22, 1952

B. BEAMAN ET AL

2,604,047

TWO-STAGE HYDRAULIC PRESSURE PUMP

Filed Sept. 19, 1945

4 Sheets-Sheet 1

Fig. 1.

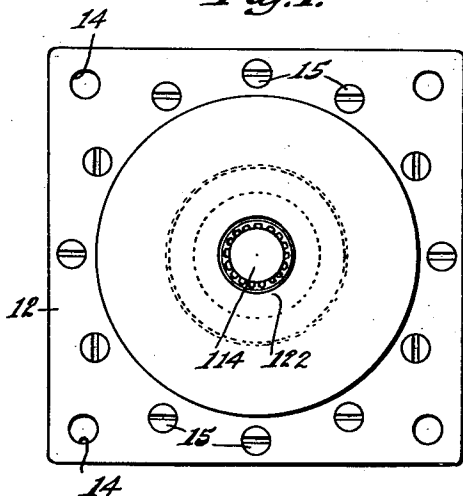


Fig. 2.

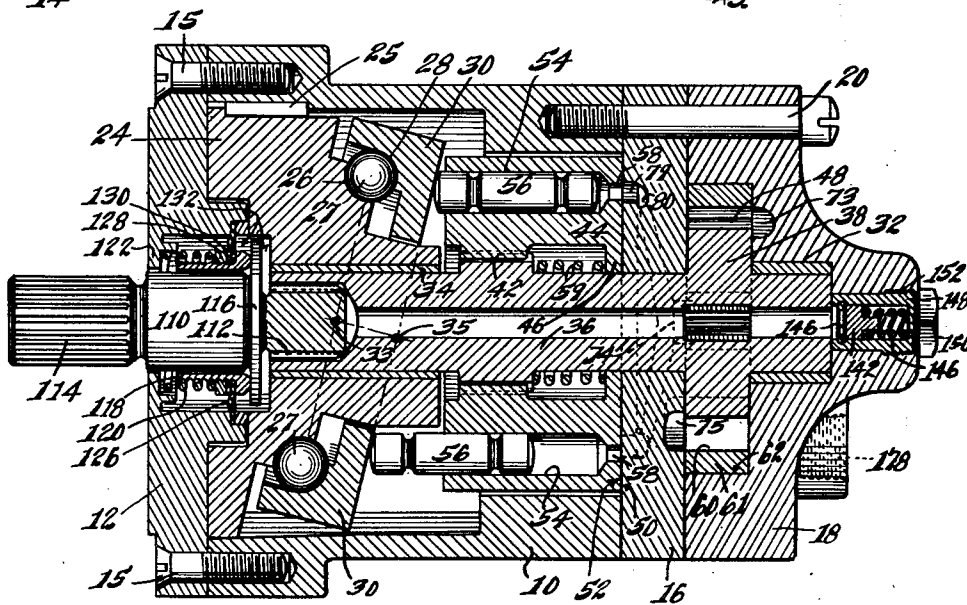
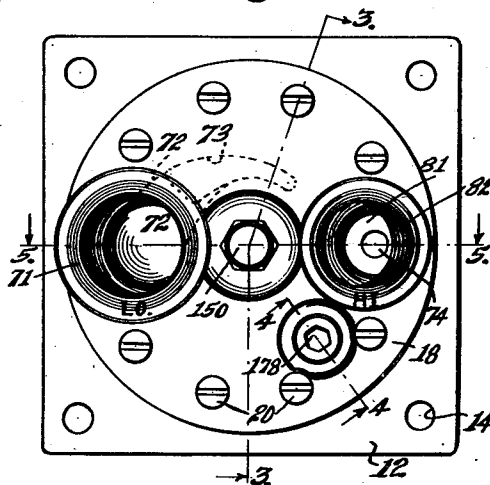
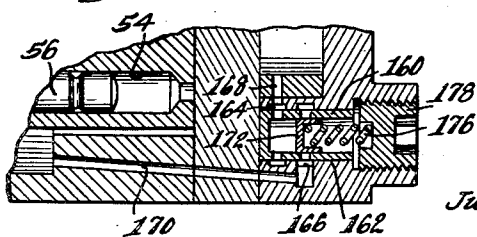


Fig. 3.

Fig. 4.



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4 Sheets-Sheet 2

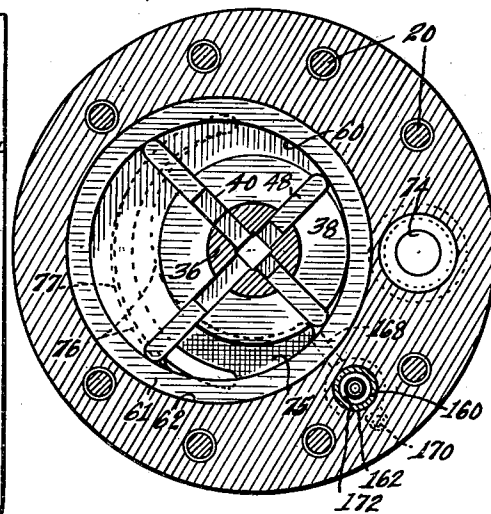
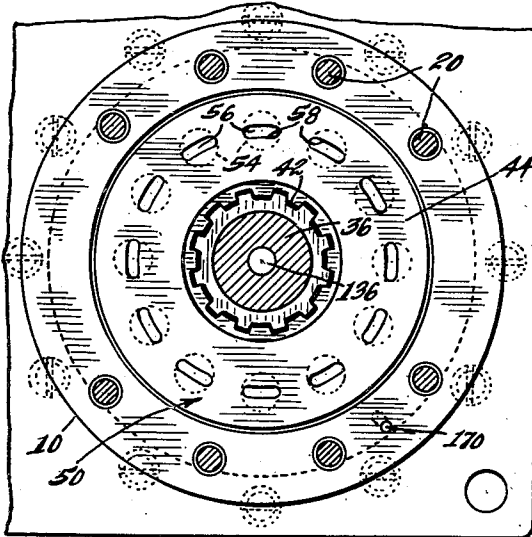
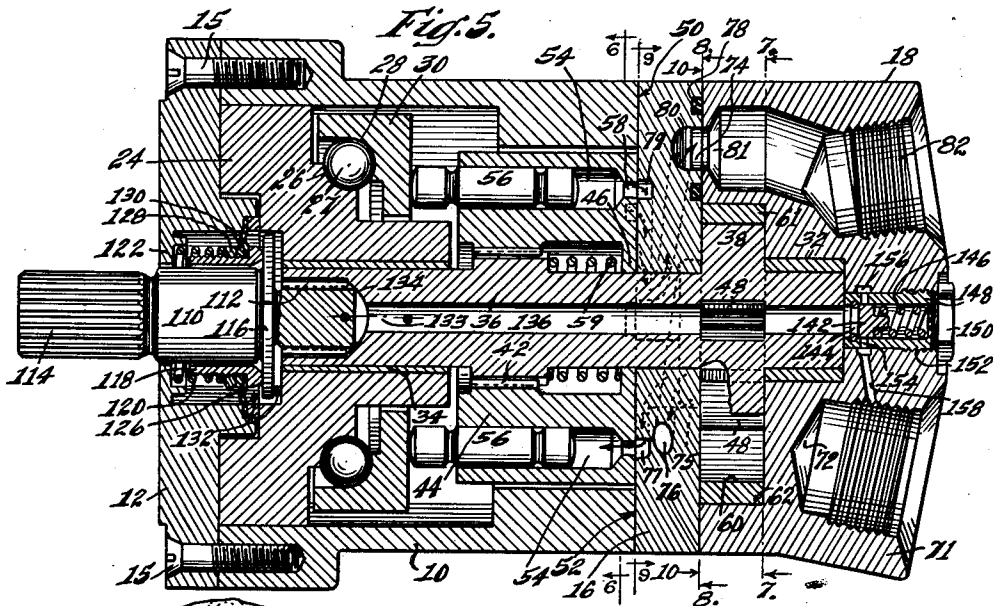


Fig. 6.

Fig. 7.

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Fig. 8.

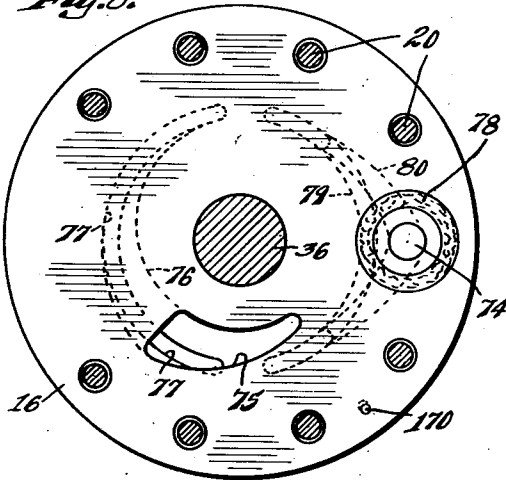


Fig. 9.

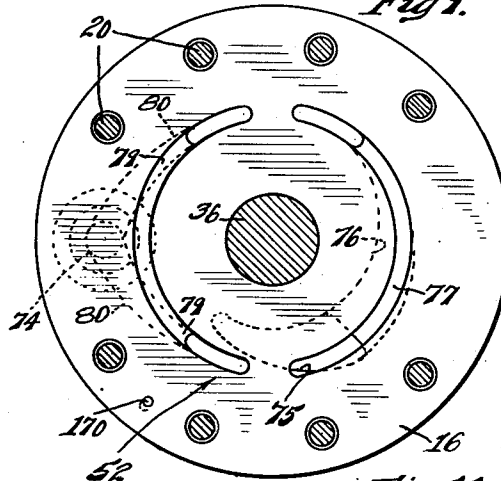


Fig. 10.

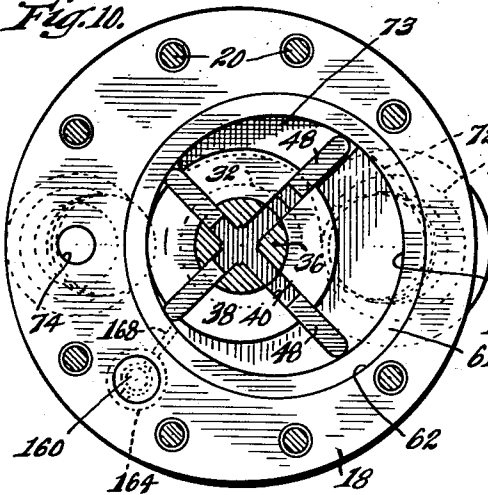


Fig. 11.

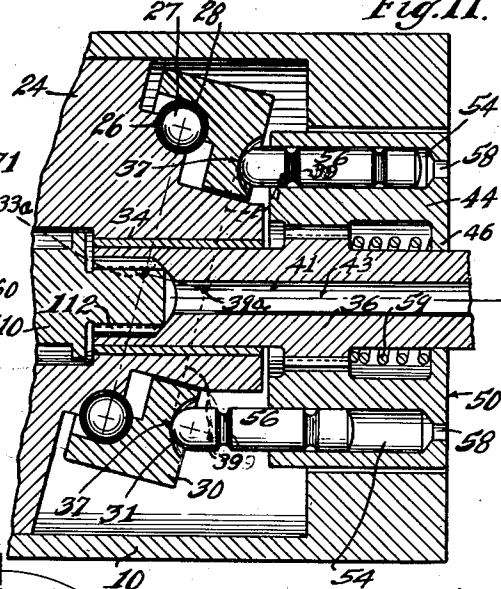


Fig. 11a.

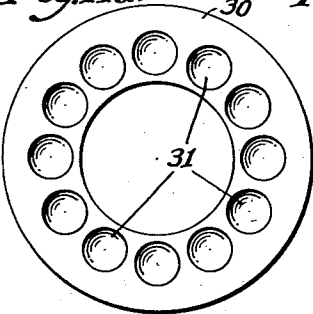


Fig. 11b.

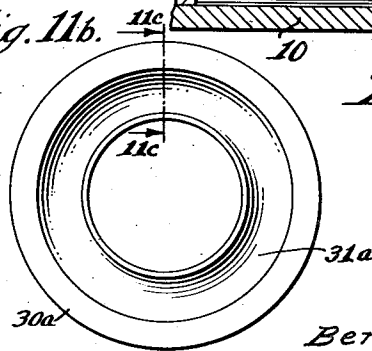
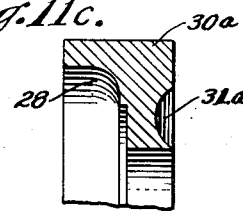


Fig. 11c.



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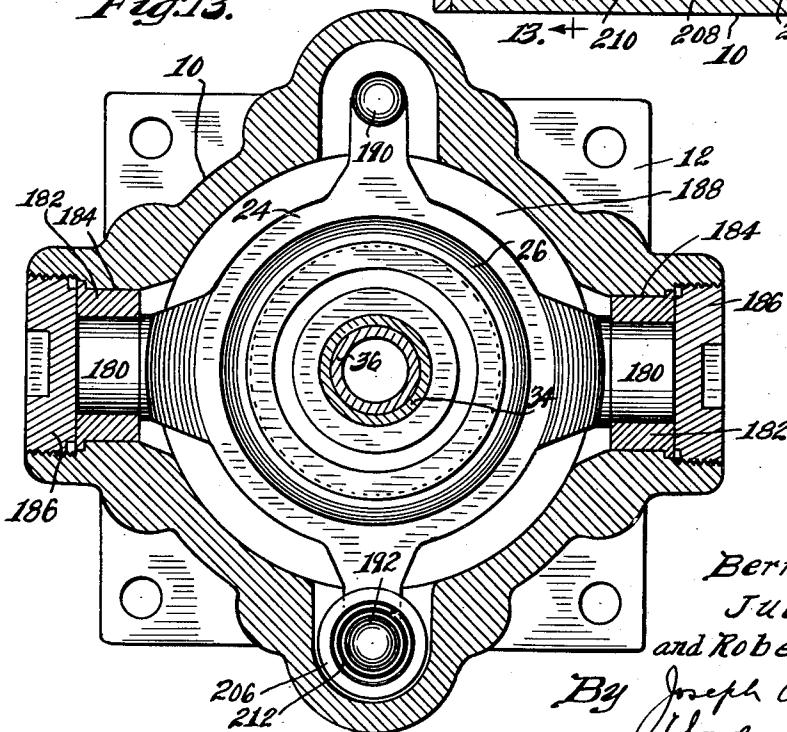
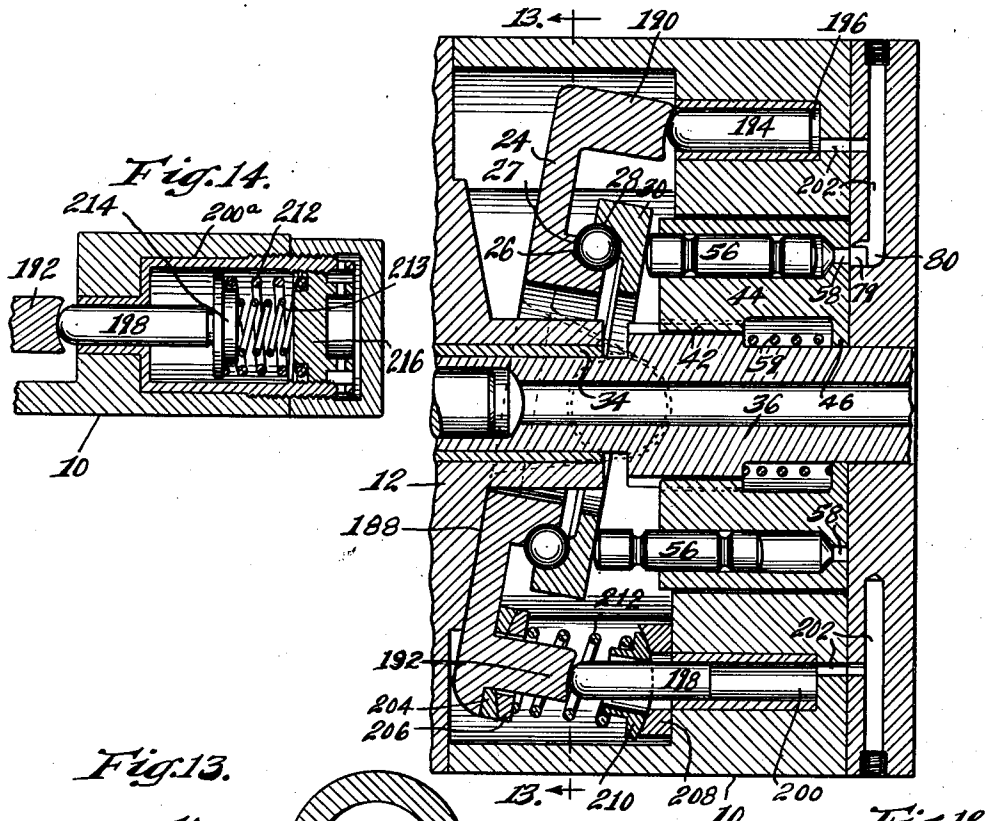
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TWO-STAGE HYDRAULIC PRESSURE PUMP

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4 Sheets-Sheet 4



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UNITED STATES PATENT OFFICE

2,604,047

TWO-STAGE HYDRAULIC PRESSURE PUMP

Bernard Beaman, Julius Hulman, and Robert A. Stein, Dayton, Ohio; said Beaman and said Stein assignors to said Hulman

Application September 19, 1945, Serial No. 617,424

13 Claims. (Cl. 103—5)

(Granted under the act of March 3, 1883, as amended April 30, 1928; 370 O. G. 757)

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The invention described herein may be manufactured and used by or for the Government for governmental purposes without the payment to us of any royalty thereon.

Our present invention is related in subject matter to our co-pending applications, Serial No. 582,376 filed March 12, 1945, now Patent No. 2,459,786 of January 25, 1949, and Serial No. 598,356 filed June 8, 1945, now Patent No. 2,495,685 of January 31, 1950, and in general is a hydraulic pressure pump of two-stage character which is designed for efficient operation at extremely high pressures in the second stage.

One object of the invention is to provide a two-stage pump which is compact in design and simple of construction, involving but few parts which can be readily fabricated and assembled.

Another object is to provide a pump wherein loaded bearings are minimized, the thrust forces being taken by stationary thrust surfaces and a ball bearing arrangement being provided between a cam face and the pistons of the high pressure pump.

Still another object is to provide a pump wherein effective seals against fluid loss are provided in a simple manner.

A further object is to provide a first-stage pump of sliding vane type and a second-stage pump comprising a rotatable barrel having cylinder bores therein with pistons provided in the bores and slidable under the action of a cam, the porting arrangement from the inlet of the pump to the low pressure pump section and then to the high pressure pump section and back to the outlet being comparatively direct and covering but a short distance, thus minimizing turbulence and conduit friction.

Still a further object is to floatingly mount the barrel of the high pressure pump so that it effectively seats against a valve plate of the pump under the force of internal pressure within the housing and acting on partially closed cylinder bore ends of the pump, and to provide a relief valve for such pressure to maintain it at a desired value somewhat lower than the pressure in the outlet of the pump without exceeding such desired pressure.

An additional object is to provide a two-stage pump wherein the mechanism of two pumps in

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series is built into one housing to form a compact, self-contained unit.

Another additional object is to provide a high pressure piston pump which is pressurized by a low pressure vane pump, the vane pump over-feeding the piston pump and having a pressure relief means from the outlet of the vane pump to the inlet thereof to thus maintain a predetermined charging pressure for the piston pump which forces the pistons against the cam without the necessity of providing return springs, linkage or some other mechanical means for this purpose.

Still another additional object is to provide a structure which can be rotated in one direction for pumping liquid, and can be readily adjusted for rotation in the opposite direction by turning an end plate and a valve plate through 180° of movement.

A further additional object is to bleed the excess capacity of the vane pump into the body of the piston or main pump through the relief valve whereby the by-passed liquid provides a circulating cooling fluid for reducing the temperature rise of the pump and to provide a lowered operating temperature within the pump.

Still a further additional object is to provide a novel ball bearing arrangement between a creep plate and a cam which is designed to reduce contact pressure appreciably and to provide extremely high thrust forces as found necessary when pumping liquid at extremely high pressure, such as three thousand pounds per square inch.

Other additional objects are to provide a modified structure wherein a cam plate may be adjusted as to inclination for changing the degree of reciprocation of pump plungers and to provide a means for automatically adjusting the position of the cam plate in response to the pressure delivered by the pump.

With these and other objects in view, our invention consists in the construction, arrangement and combination of the various parts of our device whereby the objects contemplated are attained, as hereinafter more fully set forth, pointed out in our claims and illustrated in the accompanying drawings, wherein:

Fig. 1 is a front elevation of our pump;

Fig. 2 is a rear elevation thereof;

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Fig. 3 is vertical sectional view on the line 3—3 of Fig. 2;

Fig. 4 is a sectional view on the line 4—4 of Fig. 2 showing a relief valve between the outlet and the inlet of the low pressure pump;

Fig. 5 is a sectional view on the line 5—5 of Fig. 2;

Fig. 6 is a sectional view on the line 6—6 of Fig. 5 showing the end of the pump housing and the ported end of the high pressure cylinder barrel therein;

Fig. 7 is a sectional view on the line 7—7 of Fig. 5 showing the low pressure pump and a portion of the porting arrangement;

Fig. 8 is a sectional view on the line 8—8 of Fig. 5 showing a rear elevation of the valve plate between the low pressure pump and the high pressure pump and the porting arrangement therethrough;

Fig. 9 is a sectional view on the line 9—9 of Fig. 5 showing the front view of the valve plate;

Fig. 10 is a sectional view on the line 10—10 of Fig. 5 showing a front view of the rear end plate of the pump and a front view of the stator and rotor of the low pressure pump;

Fig. 11 is a sectional view similar to a portion of Fig. 3 showing a modified bearing construction for the cam and creep plate;

Fig. 11a is a face view of a creep plate shown in Fig. 11.

Fig. 11b is a face view of a modified form of creep plate.

Fig. 11c is an enlarged sectional view on the line 11c—11c of Fig. 11b;

Fig. 12 is a sectional view similar to a portion of Fig. 3 showing a modified construction wherein a cam plate, instead of being stationary as in the previous figures of the drawing, is adjustable as to inclination for thereby varying the reciprocatory strokes of the pistons in the high pressure pump;

Fig. 13 is a sectional view on the line 13—13 of Fig. 12 showing details of construction of the modified form of our invention shown in Fig. 12; and

Fig. 14 is a sectional view of a modified form of pressure responsive regulating means for the cam plate of Fig. 12.

On the accompanying drawings we have used the reference numeral 10 to indicate a cylindrical pump housing. The housing 10 has an end plate 12 at its front end which constitutes a mounting flange for mounting the housing, for instance, on an aircraft engine. For mounting purposes, bolts or screws may extend through perforations 14 of the flange 12 and thread into the supporting surface.

A valve plate 16 is mounted on the back end of the housing and a rear end plate 18 is mounted against the valve plate, as by cap screws 20, with a suitable gasket or sealing compound between the parts 10, 16 and 18. A cam 24 is mounted in the housing 10 and is retained against rotation relative thereto by a key 25 (see Fig. 3). The cam is positioned next to the end plate 12, and the end plate itself is retained in position as by cap screws 15. The cam 24 has a bearing race 26 receiving ball bearings 27, and the balls interfit with a second race 28 of a creep plate 30 of washer-like character. The groove 26 follows the path of a truncated cylinder end so as to operate the pistons of a high pressure pump, as will hereinafter appear.

Rotatable in bearing sleeves 32 and 34 is a barrel drive shaft 36. The sleeve 32 is retained

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in the end plate 18 and the sleeve 34 is retained in the cam 24. Loosely splined at 42 on the barrel drive shaft 36 is a barrel 44 having a flat sealing face 50 to contact with a flat sealing face 52 of the valve plate 16. The loose splining of the barrel 44 to the shaft 36 permits ready conformity of the face 50 to the face 52.

Circumferentially spaced cylinder bores 54 are provided in the barrel 44 having an annular shoulder 46 engaging the shaft 36 for guide purposes, and in each cylinder bore a plunger-like piston 56 is reciprocable. Each cylinder has a cylinder port 58 of the shape shown in Fig. 6 and of somewhat less cross-sectional area than the area of the cylinder bore itself, in order to provide the necessary thrust force between the barrel 44 and the valve plate 16 by internal pressure acting on the cylinder bore ends surrounding the ports 58. The cylinder bores 54 and the pistons 56 constitute a high pressure pump. During starting of the pump, when pressure has not yet built up, it is desirable to make sure that the faces 50 and 52 are in contact. This may be accomplished by a light spring 59 interposed between the shoulder on the shaft 36 formed by the splining 42 and the barrel shoulder 46, as shown in Figs. 3 and 5.

A low pressure pump is provided in the form of a rotor 38 located within a stator bore 60 of a ring-like stator 61. The stator fits in a bore 62 of the end plate 18. The stator and its bore are eccentric with relation to the axis of rotation, and the rotor is formed on or secured to the barrel drive shaft 36. The rotor 38 has substantially radial slots 40 in which vanes 48 are slidably mounted, as shown in Fig. 7. The eccentricity of the stator bore 60 effects inward and outward movement of the vanes 48 upon rotation of the rotor 38.

A low pressure inlet boss 71 is provided on the end plate 18, as shown in Figs. 2 and 5. This boss communicates by means of a passageway 72 with an inlet port 73 of the low pressure pump. The inlet port is provided in the end plate 18. A low pressure outlet port 75, of similar shape, is provided in the rear face of the valve plate 16, and the front face of the valve plate has an arcuate high pressure inlet port 77 for the high pressure pump. A passageway 76 leads from the port 75 to the port 77.

The front face of the valve plate 16 also has a high pressure outlet port 79 of arcuate shape, and this communicates by means of a pair of passageways 80 with a passageway 81 in the end plate 18 and a high pressure outlet boss 82 leading from the pump. The passageway 81, at its intake end, is reduced, as indicated at 74, and a sealing ring 78 surrounds the joint to effect a seal from the passageway 80 to the passageway 81 between the contacting faces of the valve plate 16 and the end plate 18.

From the description just made of the passageways and ports it will be obvious that the oil or other liquid being pumped can be traced in a circuit through the following elements: 71, 72, 73, low pressure or vane pump, 75, 76, 77, high pressure or piston pump, 79, 80, 74, 81 and 82. This arrangement produces a direct passage of the oil from the rear of the pump through the end plate 18 and the valve plate 16, and then into the barrel 44, with a direct return from the barrel through the valve plate 16 and the end plate 18. Thus turbulence is minimized, and, likewise, the conduit friction on the liquid being pumped is minimized.

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The cylinder ports 58 of the high pressure pump successively register with the arcuate ports 77 and 79 to draw liquid into the barrel 44 during one-half the revolution thereof, and expel oil therefrom during the other half revolution. The ports 71 and 82 may be considered respectively as the low pressure and high pressure ports of the pump, and are indicated as such by the legends "HI" and "LO" in Fig. 2.

A drive shaft 110 is provided having a splined end 114 for connection to a power take-off shaft of an aircraft engine, or any other suitable means to drive the pump. The power take-off shaft would include a means to align the outer end of the drive shaft 110 for proper operation without binding. The inner end of the drive shaft 110 has a loosely splined connection 112 with the barrel drive shaft 36.

The drive shaft 110 has a flange 116 surrounding it, with which a seal plate 118 in the form of a ring is engageable. A spring 120 is seated against an intumed flange 122 of the end plate 12. The seal plate 118 is sealed relative to the housing 10 by means of a diaphragm 126 having its inner edge clamped to the seal plate by a spanner nut 128 and its outer edge clamped between a washer 130 and the cam 24 by the end plate 12. This provides a pressure chamber, indicated at 132, inside of the flange 16.

The inner end of the drive shaft 110 is seated against a shoulder 134 of the barrel drive shaft 36 which serves merely as a thrust connection, but, normally, when the pump is operating, the two are spaced from each other. A passageway 136 is provided through the center of the barrel drive shaft through which liquid is adapted to flow after passing through the splined connections 42 and 112 and the bearing sleeve 34.

The passageway 136 communicates with a pressure relief valve in the form of a valve plunger 142 seated against a ring 144 by a spring 146. The valve plug acts as a piston in a sleeve 148 which serves to guide its sliding movement. The sleeve 148 has a head 150 for screwing the sleeve into the end plate 18 at a screw-threaded connection 152. The sleeve 148 has ports 154 communicating with an annular groove 156 which, in turn, communicates by means of a passageway 158 with the low pressure boss 71.

The low pressure pump comprising the rotor 38 and the vanes 48 is higher capacity than the high pressure pump comprising the cylinder bores 54 and the pistons 56. This is for the purpose of assuring full charging of the high pressure cylinders for reducing voids in the liquid and for keeping the pistons in contact with the creep plate 30.

To avoid excessive build-up of pressure and keep a substantially even charging pressure from the low pressure pump to the high pressure pump a relief valve from the outlet of the low pressure pump to the interior of the housing 10 is provided. This is shown in Figs. 4 and 7 and consists of a sleeve 160 in a bore 162 of the end plate 18. There is a pair of annular grooves 164 and 166 surrounding the sleeve 160, and passageways 168 and 170 serve to communicate these grooves with the high pressure side of the low pressure pump and the interior of the housing 10, respectively. Within the sleeve 160 is a plunger 172 which is moved toward the right in Fig. 4 against the action of a spring 176, by liquid under pressure from the passageway 168. The spring 176 is retained in position by a closure plug 178, and the entire relief valve assembly is held in posi-

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tion by this plug. The operation of this valve will be referred to hereinafter.

One of the major problems in a small, compact, highly stressed pump of the character herein disclosed is a high contact pressure which exists between the ends of the pistons 56 and the flat creep plate 30. This, heretofore, has largely been alleviated by the use of complicated shoes or piston rods, or by enlarging the piston ends. However, a cheaper and more compact method is to have the pistons work in a conforming spherical seat in the creep plate. This is shown in Fig. 11 and 11a, wherein the creep plate 30 has a spherical seat 31 for each piston. Consequent wear is materially reduced by this arrangement. A modified form of creep plate 30a is shown in Figs. 11b and 11c and will be referred to specifically later herein.

The peculiarities of motion due to the inclination of the axis of rotation of the barrel 44 and the inclined creep plate 30 make it essential that the spherical indentations 31 be properly located. It is quite usual in pumps of this kind to locate the center line through the bearings 27 for the creep plate centered on the axis of rotation of the barrel 44 (indicated at 33 in Fig. 3). This causes the contact face of the creep plate to rotate about a center (see point 35) displaced from the axis of the barrel. The consequent relative motion of the piston contact end on the flat face is thereby increased. As long as the plate is flat and sufficiently large, there is little objection except for the excessive motion and wear.

When providing individual spherical seats 31, as in Fig. 11, means must be taken to have the seat sufficiently large to insure piston engagement throughout 360° of rotation, and it must be made with a small radius to insure sufficient conformity to reduce contact pressure appreciably. This is accomplished by having the center of rotation of the contact surfaces indicated at 37 within the indentations 31 and with a line 39 connecting the contact faces 37 having its center indicated by the point 39a falling on an axially extending center line 41 located at, or, at least, near the axis of rotation 43 of the shaft 36. It will be noted that the center line 41 is displaced only slightly from the center line 43 and the center of rotation 33a for the balls 27 is somewhat further displaced from 43. The creep plate motion is thereby reduced considerably by its axis of rotation being positioned at an optimum position.

The creep plate seats or indentations 31 serve as a driving means for the creep plate and insure that it moves at the same rotative speed as the cylinder barrel. The indentations 31 (see Figs. 11 and 11a), however, may be replaced with a continuous annular groove 31a as shown in Figs. 11b and 11c of the cross-section shown for the indentations 31 if it is not desirable to cause rotation of the creep plate simultaneously with the barrel. Such an arrangement as shown distributes the wear around the entire circumference of such groove instead of localizing it in the individual indentations.

In Figs. 12 and 13 we show a modified construction wherein some of the parts correspond to those already described and, accordingly, bear the same reference numerals. The modified construction, however, provides a means whereby the displacement or volume output of the high pressure pump per revolution can be varied from almost zero up to the rated capacity of the pump in infinitesimal increments.

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Constructionally, the cam plate 24 has been modified by providing it with trunnion bearings 180 journaled in bearing sleeves 182. The bearing sleeves 182 are retained in lateral bores 184 of the housing 10 by plugs 186. The cam plate is thereby mounted for oscillation on the axis of the trunnions 180 from the maximum tilted position shown in Fig. 12 to a vertical or untilted position where reciprocation of the plungers 56 by the creep plate 30 is reduced to zero. At any intermediate angle the displacement will be a proportional intermediate value.

Inasmuch as the displacement of the pistons 54 will be a function of the angular position of the creep plate 30, the displacement will vary from maximum, in the position shown in Fig. 12, to minimum when the creep plate is perpendicular to the axis of the pump shaft. The limit of movement of the creep plate is determined by a face 188 of the end plate 12.

In our modified construction being now described, control of the angle of the creep plate may be responsive to a mechanical motion or to differential pressure by utilizing linkage of appropriate nature. By way of illustration we show the creep plate controlled by the pump discharge pressure. It is desirable to maintain full displacement up to a predetermined pressure, and then with infinitesimal increases in pressure to have the displacement decrease until, at some pressure slightly higher than that at which full displacement occurs, the displacement is zero.

For this purpose a pair of projections 190 and 192 is formed on the cam plate 24 at diametrically opposite positions. The projection 190 has a plunger 194 engaging therewith and sliding in a cylinder 196. The projection 192 has a plunger 198 engaged therewith and sliding in a cylinder 200. The cylinders 196 and 200 are connected by means of passageways 202 with the high pressure outlet 81 of the pump so that the plungers 194 and 198 are responsive to the outlet pressure thereof as a unit.

Spring seats 204 and 206 surround the projection 192, and spring seats 208 and 210 surround the plunger 198. A spring 212 is interposed between the seats 206 and 210 so that the position which the cam plate 24 assumes for any given pressure condition will be a function of the force of the springs 212, the force of oil pressure on the plunger 198, and the force of oil pressure on the plunger 194. The plunger 194 is slightly larger than the plunger 198 so that the force exerted to tilt the cam plate 24 will be proportional to the pump discharge pressure and the relative areas of the plungers 194 and 198. The force of the spring 212 will be almost constant, but will vary slightly with the degree of compression.

In Fig. 14 we show a modified arrangement for the spring 212 wherein the spring is enclosed within the cylinder 200^a. This cylinder corresponds to the cylinder 200 but is enlarged to include the spring 212 and an additional spring 213, if desired. The springs are enclosed within the cylinder, and both springs are interposed between a head 214 on the plunger 198 and an adjusting plug 216. The plug may be screwed inwardly or outwardly for increasing or decreasing the pressure of the springs 212 and 213 for thus varying the predetermined regulated outlet pressure for the pump.

Practical operation

The shaft 110 is driven by any suitable power means to thereby drive the barrel-driving shaft 36

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and the rotor 38 of the low pressure or first-stage pump. Rotation of the rotor 38 causes liquid to be drawn in from the intake 71 and pass through the passageway 72 to the intake port 73 of the low pressure pump, this liquid being discharged through the outlet port 75 of the low pressure pump into the passageway 76. The liquid is then picked up from the intake port 77 of the high pressure pump and discharged into its outlet port 79, from which it passes through the passageways 80, the opening 74 and the passageway 81 to the pump outlet 82. The creep plate 30 is positioned so that the cylinder ports 58 communicate with the port 77 on the intake strokes of the pistons 56 and with the port 79 on the outlet strokes thereof. The liquid is thus pumped at high pressure from the pump.

As to the low pressure or sliding vane pump, it pumps liquid at low pressure to charge the high pressure pump, and is designed to overcharge the high pressure pump. Pressure by the low pressure pump, however, is regulated by the relief valve 172, the excess liquid passing through the passageway 168 to the interior of the sleeve 160, and there forcing the plunger 172 upwardly until the liquid pressure overcomes the spring 176, whereupon the excess liquid is bled through the annular passageway 166 and the passageway 170 into the interior of the housing 10. The interior of the housing is maintained at a predetermined pressure between that for which the valve 172 is set and atmospheric pressure by the relief valve 142. The by-passed oil thus circulates through the housing 10, lubricating the working surfaces and finding its way into the passageway 136, and, finally, back through the relief valve 142 and the passageway 158 to the pump inlet 71. The liquid is sealed off from atmosphere around the drive shaft 110 by the sealing arrangement 116—118.

By the arrangement just described a substantially even pressure of oil is supplied from the low pressure pump to the high pressure pump to assure full charging of the high pressure pump even after the pump becomes worn to some extent. During starting operation, the spring 59 keeps the barrel 44 in engagement with the valve plate 16, and the spring 120 keeps the seal plate 118 in engagement with the flange 116 so that there is a sealed chamber within the housing 10 all the way from the end plate 18 to the diaphragm 126. These springs, particularly, the spring 59, maintain the seal necessary for priming the pump. There is some leakage of liquid past the faces 50 and 52, especially before pressure is built up in the housing 10, and this liquid, together with the small amount which creeps past the pistons 56, and any which may leak from the low pressure pump, will charge the interior of the housing 10, in addition to the by-passed oil from the passageway 170, thereby lubricating the bearing surfaces 32, 34 and 106, and producing pressure within the housing. The barrel 44 will be seated against the valve plate 16 under the action of the charging pressure for the housing plus the internal pressure of the cylinder bores acting on the partially closed cylinder ends.

The housing pressure also acts on the flange 116 to compress the spring 120 somewhat and thereafter maintain the flange against the inner face of the seal plate 118 under such pressure. The pressure is limited, however, by the relief valve 142, depending upon the strength of the spring 146. The liquid pressure will move the relief valve to a relief position, permitting the ex-

cess liquid to flow through the ports 154, 156 and 158 back to the low pressure boss 71. A selected spring 146 may be used as required to keep a desired pressure within the housing. The valve 142 therefore acts as a pressure regulating valve.

Our two-stage pump herein disclosed can be readily adjusted for a reversal in the direction of rotation by turning the end plate 18, and the valve plate 16, 180°. This is accomplished by removing the screws 20 and reinserting them after the adjustment has been made.

In the operation of the form of invention shown in Figs. 12 and 13, when the pump is at rest the spring 212 acts to tilt the cam plate 24 to the maximum displaced position of Fig. 12. As the pump discharge pressure is built up the plungers 194 and 198 begin to exert force on the projections 190 and 192, with the plunger 194 exerting a larger force than the plunger 198 due to their difference in area.

At the predetermined pressure for which the pump has been designed the force exerted by the plunger 194 is exactly equal to the force exerted by the plunger 198 plus the force of the spring 212. At any pressure less than this the force of the plunger 194 is less than the additive forces of the plunger 198 and the spring 212. As the pressure increases beyond the value at which these forces are in equilibrium, and the cam plate is in maximum displaced position, the force exerted by the plunger 194 will be larger than the forces exerted by the spring and the plunger 198, thus causing the cam plate to be tilted toward a vertical position. Further increases in pressure will finally cause the cam plate to assume a position where the forces are again in equilibrium, since the spring force increases slightly as the spring is compressed. In such a position the displacement of the pump will be proportionally lower than the maximum displacement.

If the flow through the outlet of the pump is reduced to zero the cam plate will assume an angle slightly less than vertical, and the displacement of the pump will be only enough to overcome the slip or volumetric inefficiency of the pump and whatever other leakage of high pressure oil occurs. The pressure will be only that much higher than the maximum at which full displacement occurs as occasioned by the difference in spring force at the minimum compression and at the maximum compression positions.

With the alternate method (Fig. 14) of installing spring means for the regulating function just described the action is substantially the same, with the spring exerting its force through the plunger 198 instead of directly on the cam plate 24. This arrangement has the additional advantage that the spring force can be adjusted and, by so doing, the cut-off pressure can be adjusted.

From the foregoing specification, we believe it will be obvious that we have provided a structure which compactly includes both a low pressure pump and a high pressure pump, whereby a complete hydraulic pump suitable for high pressure operation is had. The design is such that there are few parts, and they are simple to form, machine and assemble. The necessary thrust of the rotating barrel to effect pressure seal is had without the necessity of providing the spring 59 other than for the purpose of the initial thrust before pressure is built up within the housing. The spring can therefore be comparatively light, yet will insure proper priming

of the pump. The seal at 116—118 remains effective even if the shaft 110 is out of alignment or eccentric, or if the spline 112 should happen to stick.

The barrel 44 is fully floating on the barrel shaft so that it can properly seat against the end plate 18 without any binding whatever. The vanes in the low pressure pump are forced outwardly by centrifugal force, and the volume trapped between two adjacent blades alternately increases and decreases as the rotor turns, thereby giving the desired pumping action. The inlet and outlet ports 73 and 75 of the low pressure pump are preferably overlapped slightly at their large ends with respect to the adjacent vanes 58 to insure good filling of the low pressure pump, especially at high rotative speeds.

The high pressure pump is preferably provided with a considerable number of pistons to reduce pressure pulsations, and this pump is pressurized by the low pressure pump to eliminate cavitation therein. The low pressure pump also eliminates the necessity of springs or linkage normally used for returning the pistons of the high pressure pump.

The vane pump ports 73 and 75, it will be noted, are axial, to avoid pressure drop experienced with ports of the radial entry type. The passageway arrangement from the inlet 71 to the outlet 82 is reduced to a minimum, thus conducting the liquid in the most direct path possible to minimize both turbulence and conduit wall friction on the flowing liquid. All of the pumping power transmitted to the rotating barrel 44 is carried through concentric shafts which are not subjected to any appreciable thrust. Ball bearing arrangements disclosed also contribute to the efficiency of the pump, and the series arrangement of the low and high pressure sections of the pump is such that an extremely compact structure results. At the same time, the parts are designed rugged enough for containing the high pressures generated.

The valve plate 16 carries the outlet port for the low pressure pump, and both the inlet and outlet port for the high pressure pump. Since this plate is flat on both surfaces it can easily be fabricated, and ground and lapped on its surfaces to insure proper working fit.

With the modified arrangement shown in Figs. 12, 13 and 14 automatic regulation of the output of the pump is had. The trunnion mounting arrangement permits oscillation of the cam plate to the desired control position, and it will be noted that the center line of the trunnion bearings is located coincident with a plane through the flat surface of the creep plate 30 against which the high pressure pistons engage. For such engagement, see Fig. 3. With the arrangement of two opposed plungers 194 and 198, stability of operation is had due to the outlet pressure operating on both plungers. Proper selection of plunger diameters will allow the use of a spring with a low rate of regulation, so that zero flow pressure is only slightly higher than maximum displacement pressure. This design also is readily adapted for mechanical or any other type of control, such as the hydraulic control disclosed.

A simple means of adjusting the cut-off pressure is included in the alternate arrangement of the spring as shown in Fig. 14. The trunnions 180 are installed in the sleeves 182 in such manner that the necessity of employing split construction of the sleeves is obviated.

Some changes may be made in the construction and arrangement of the parts of our pump structure without departing from the real spirit and purpose of our invention, and it is our intention to cover by our claims any modified forms of structure or use of mechanical equivalents which may be reasonably included within their scope.

We claim as our invention:

1. In a hydraulic pressure pump of the character disclosed, a housing, a barrel driving shaft journaled in said housing, a barrel mounted on said driving shaft slidably and non-rotatably relative thereto, said barrel having cylindrical bores substantially parallel to the axis of rotation of said barrel and driving shaft, a piston in each of said cylindrical bores, a valve plate at one end of said barrel, said barrel having one of its ends seated thereagainst, a cam plate having a cam surface surrounding said driving shaft and in the shape of a truncated cylinder end, a creep plate interposed between said cam and said pistons, bearings interposed between said cam plate and said creep plate, an end plate beyond said valve plate and having intake and discharge openings therein, a low pressure vane pump recessed in said end plate and having one face against said valve plate, passageway connections through said end plate from said intake opening to said low pressure pump, said valve plate having inlet and outlet ports for said bores of said barrel, and passageways through said valve plate only from the outlet of said low pressure pump to said inlet port and from said outlet port to the discharge opening of said end plate.

2. In a hydraulic pressure pump of the character disclosed, a housing, a barrel driving shaft journaled in said housing, a barrel mounted on said driving shaft and operatively connected thereto for rotation therewith and for sliding movement relative thereto, said barrel having cylindrical bores substantially parallel to the axis of rotation of said driving shaft, a piston in each of said cylindrical bores, a valve plate at one end of said barrel, said barrel having one of its ends seated thereagainst, a cam having a cam surface surrounding said driving shaft and in the shape of a truncated cylinder end, an end plate beyond said valve plate and having intake and discharge openings therein, a low pressure vane pump located in said end plate and against said valve plate, passageway connections through said end plate from said intake opening to said low pressure pump, said valve plate having inlet and outlet ports for said bores of said barrel, and passageways through said valve plate from the outlet of said low pressure pump to said inlet port and from said outlet port to the discharge opening of said end plate.

3. In a hydraulic pressure pump, a housing, a barrel driving shaft journaled in said housing, a barrel slidably mounted on said driving shaft and operatively connected therewith, said barrel having cylindrical bores, a piston in each of said cylindrical bores, a valve plate at one end of said barrel, said barrel having one of its ends seated thereagainst, ports in said valve plate and communicating with said cylinder bores, a cam having a cam surface surrounding said driving shaft and in the shape of a truncated cylinder end, an end plate beyond said valve plate and having intake and discharge openings therein, and a low pressure vane pump located in said end plate and against said valve plate for pumping fluid under low pressure to said cylindrical bores.

4. In a hydraulic pressure pump, a housing, a barrel driving shaft journaled in said housing, a barrel mounted on said driving shaft and operatively connected therewith, said housing providing a space surrounding said barrel, said barrel having cylindrical bores, a piston in each of said cylindrical bores, a valve plate at one end of said barrel, said barrel having one of its ends seated thereagainst, ports in said valve plate and communicating with said cylinder bores, a cam having a cam surface surrounding said driving shaft and in the shape of a truncated cylinder end, an end plate beyond said valve plate and having intake and discharge openings therein, a low pressure vane pump located in said end plate and a passageway through said valve plate for pumping fluid under low pressure to said cylindrical bores, and a pressure regulating by-pass from said low pressure pump to said space in said housing surrounding said barrel.

5. In a hydraulic pressure pump, a housing, a barrel driving shaft journaled in said housing, a barrel mounted on said driving shaft and operatively connected therewith, said housing providing a primary space surrounding said barrel, there being a secondary space between said barrel and said driving shaft, said barrel having cylindrical bores, a piston in each of said cylindrical bores, a valve plate at one end of said barrel, said barrel having one of its ends seated thereagainst, intake and exhaust ports in said valve plate and communicating with said cylinder bores, a cam having a cam surface surrounding said driving shaft and in the shape of a truncated cylinder end, an end plate beyond said valve plate and having intake and discharge openings therein, a low pressure vane pump located in said end plate and pumping fluid under low pressure to said intake ports, said primary space and said secondary space receiving any fluid that leaks past said pistons and/or from between said barrel and said valve plate, and a pressure regulating by-pass from said secondary space to the intake opening of said end plate.

6. In a hydraulic pressure pump, a housing, a barrel driving shaft journaled in said housing, a barrel mounted on said driving shaft and operatively connected therewith said housing providing a primary space surrounding said barrel, there being a secondary space between said barrel and said driving shaft, said barrel having cylindrical bores, a piston in each of said cylindrical bores, a valve plate at one end of said barrel, said barrel having one of its ends seated thereagainst, intake and exhaust ports in said valve plate and communicating with said cylinder bores, a cam having a cam surface surrounding said driving shaft and in the shape of a truncated cylinder end, an end plate beyond said valve plate and having intake and discharge openings therein, a low pressure vane pump located in said end plate and a passageway through said valve plate for pumping fluid under low pressure to said intake ports, said primary space and said secondary space receiving any fluid that leaks past said pistons and/or from between said barrel and said valve plate to said cylindrical bores, and a pressure regulating by-pass from said secondary space to said intake opening of said end plate.

7. In a two-stage hydraulic pressure pump, a housing, a shaft journaled in said housing, a barrel mounted on said shaft and operatively connected thereto, said barrel having cylindrical bores, a piston in each of said cylindrical bores,

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a valve plate at one end of said barrel, said barrel having one of its ends seated thereagainst, ports in said valve plate and communicating with said cylinder bores, a cam having a cam surface surrounding said driving shaft and in the shape of a truncated cylinder end, a creep plate interposed between said cam and said pistons, an end plate beyond said valve plate and having intake and discharge openings therein, a low pressure vane pump located in said end plate and faced on one side against said valve plate, and a passageway connection through said end plate from said intake opening to said low-pressure pump, and passageway connections through said valve plate from said low-pressure pump to said ports in said valve plate and from said ports in said valve plate to said discharge opening of said end plate.

8. In a two-stage hydraulic pressure pump, a housing, a valve plate against one end of said housing, an end plate against said valve plate, a low pressure pump located in and faced against said end plate, said end plate having an inlet port therefor, said valve plate having an outlet port for said low pressure pump, a high pressure pump comprising a barrel rotatable in said housing and provided with cylinder bores, pistons in said cylinder bores, said valve plate having inlet and outlet ports from said cylinder bores and passageways from the outlet port for said low pressure pump to the inlet port for said cylinder bores and from the outlet port for said cylinder bores to said end plate, said end plate having intake and discharge openings in the outer face thereof, said openings communicating respectively with said inlet port for said low pressure pump and the valve plate passageway from said outlet port for said cylinder bores.

9. A two-stage hydraulic pressure pump comprising a housing, a valve plate against one end of said housing, an end plate against said valve plate, a low pressure pump recessed in said end plate and having one face thereof coincident with a face of said valve plate, said end plate having an inlet port therefor, said valve plate having an outlet port for said low pressure pump, a high pressure pump comprising a barrel rotatable in said housing and provided with cylinder bores, pistons in said cylinder bores, cam means to reciprocate said pistons, said valve plate having inlet and outlet ports from said cylinder bores and passageways directly from the outlet port for said low pressure pump to the inlet port for said cylinder bores and from the outlet port for said cylinder bores to said end plate, said end plate having in its outer face intake and discharge openings communicating respectively with said inlet port for said low pressure pump and the valve plate passageway from said outlet port for said cylinder bores.

10. In a pump structure, a housing, a valve plate at one end thereof, an end plate against said valve plate, a shaft journaled in said housing and said valve and end plates, a low pressure pump rotor on said shaft within said end plate, one face of said rotor being against said valve plate, a high pressure pump barrel on said shaft within said housing, said barrel having a plurality of high pressure cylinders, a piston in each of said cylinders, an annular cam for reciprocating said pistons upon rotation of said shaft, said barrel having an outlet port for each of said cylinder bores, said end plate having an intake opening and a discharge opening, said valve plate having an outlet port for said low-pressure

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pump and inlet and outlet ports for said high-pressure pump, said end plate having an inlet port for said low-pressure pump, and passageway connections in said valve plate and said end plate from said intake opening to said inlet port for said low-pressure pump, from the outlet port for said low-pressure pump to the inlet port for said high-pressure pump, and from the outlet port for said high-pressure pump to said discharge opening.

11. In a pump structure, a housing, a recessed end plate at one end thereof, the bottom of said recess constituting a face of said end plate, a valve plate interposed between said housing and said end plate, a shaft, a low pressure pump rotor on said shaft within the recess of said end plate, said rotor being substantially so thick as the depth of said recess, a high pressure pump barrel on said shaft within said housing, said barrel having a plurality of high pressure cylinders, a piston in each of said cylinders, an annular cam for reciprocating said pistons upon rotation of said shaft, said barrel having an outlet port for each of said cylinder bores, said end plate having an intake opening and a discharge opening, said valve plate having an outlet port for said low-pressure pump and inlet and outlet ports for said high-pressure pump, said end plate having an inlet port for said low-pressure pump and passageway connections from face to face of said end plate from said intake opening to said inlet port for said low-pressure pump, from face to face of said valve plate from said outlet port for said low-pressure pump to said inlet port for said high-pressure pump, and from face to face through said valve plate and said end plate in series from said outlet port for said high pressure pump to said discharge opening.

12. In a pump structure, a housing, a valve plate at one end thereof, a recessed end plate against said valve plate, a shaft journaled in said housing and said valve and end plates, a low pressure pump rotor on said shaft within the recess of said end plate and faced against said valve plate, a high pressure pump on said shaft within said housing, said end plate having an intake opening and a discharge opening, said valve plate having an outlet port on one side for said low-pressure pump and inlet and outlet ports on the other side for said high-pressure pump, said end plate having an inlet port for said low-pressure pump, passageway connections in said valve plate and said end plate from face to face thereof and leading from said intake opening to said inlet port for said low-pressure pump, from said outlet port for said low-pressure pump to said inlet port for said high-pressure pump and from said outlet port for said high-pressure pump to said discharge opening, a pressure regulating valve between said low and high pressure pumps, and a pressure relief valve between said low pressure pump and said intake opening.

13. In a pump structure, a housing, a valve plate at one end thereof, a recessed end plate against said valve plate, a shaft journaled in said housing and said valve and end plates, a low pressure pump rotor on said shaft within the recess of said end plate, said rotor having one face thereof contacting said valve plate, a high pressure pump on said shaft within said housing, said end plate having an intake opening and a discharge opening, said valve plate having an outlet port for said low-pressure pump on one side and inlet and outlet ports on the other side for said high-pressure pump, said end plate hav-

ing an inlet port for said low-pressure pump, passageway connections from face to face of said valve plate and said end plate leading from said intake opening to said inlet port for said low-pressure pump, from said outlet port for said low-pressure pump to said inlet port for said high-pressure pump, and from said outlet port for said high-pressure pump to said discharge opening.

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