

May 31, 1955

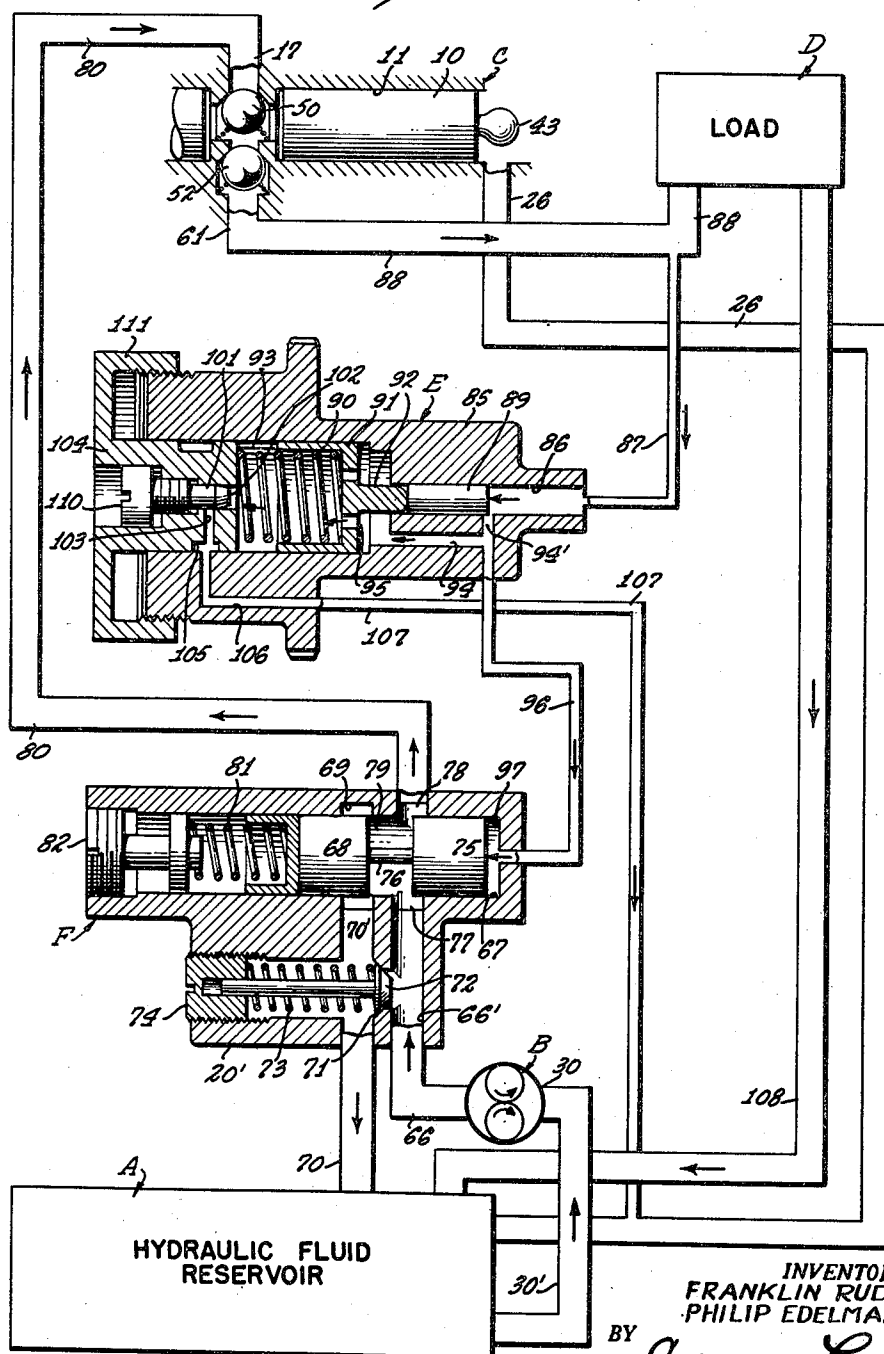
P. EDELMAN ET AL
TWO-STAGE PUMPING SYSTEM WITH AUTOMATIC
DEMAND RESPONSIVE CONTROL

2,709,339

Filed July 6, 1953

3 Sheets-Sheet 1

Fig. 1.



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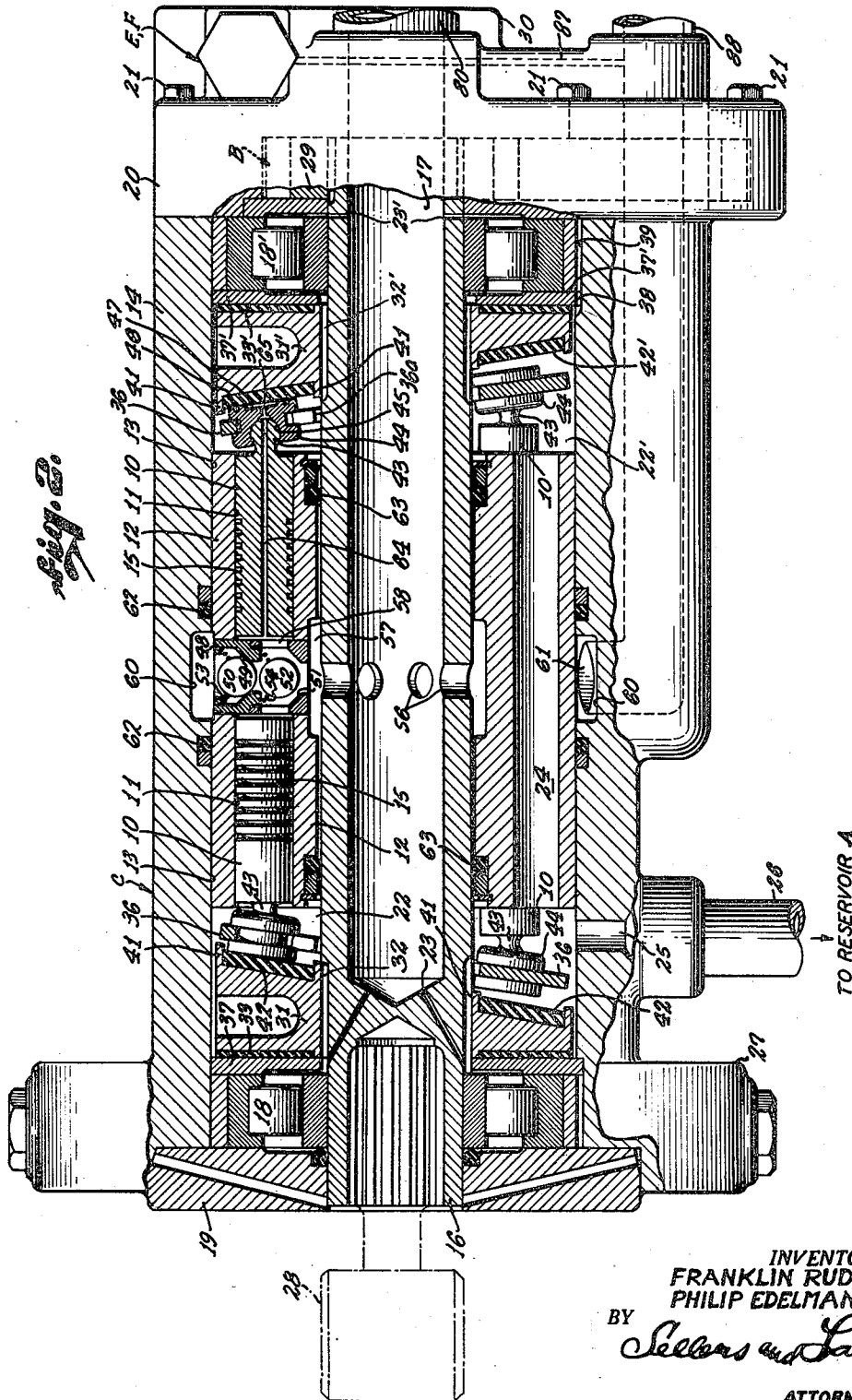
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Fig. 3.

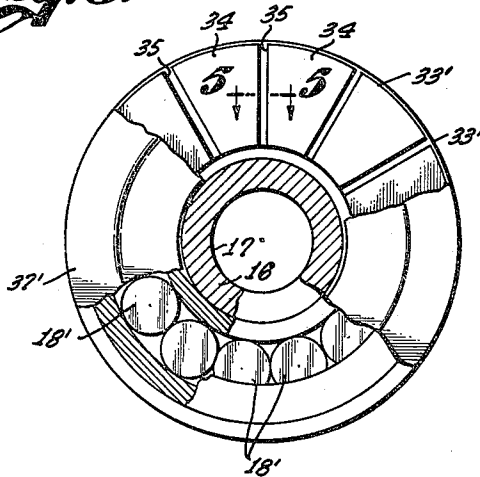


Fig. 5.

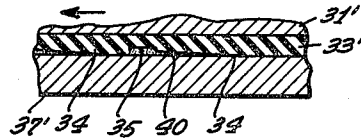


Fig. 6.

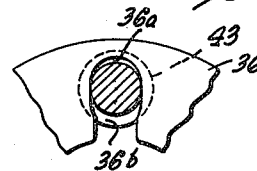


Fig. 4.

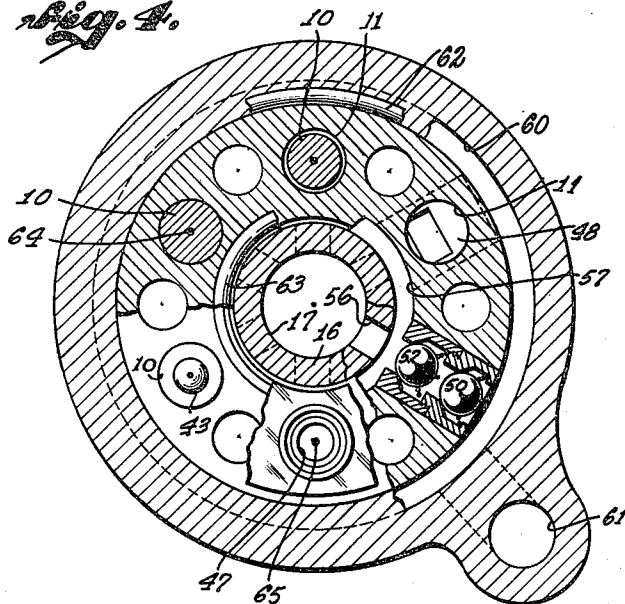
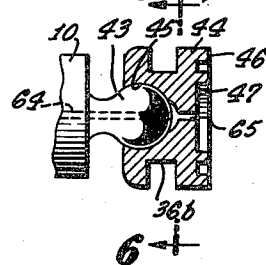


Fig. 7.



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TWO-STAGE PUMPING SYSTEM WITH AUTOMATIC DEMAND RESPONSIVE CONTROL

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17 Claims. (Cl. 60—52)

This invention relates to pumping systems of the type utilizing a high pressure piston pump for delivering hydraulic fluid to a hydraulic motor or other device requiring a supply of high pressure operating fluid. The invention is directed particularly to pumping systems operating with a two-stage pumping action derived from a super-charger or low pressure feed pump, feeding working fluid to the inlet of a high pressure, piston pump of a feathering type (for adjustment to varying load demand), the output of which is delivered to the device requiring the high pressure fluid. Systems of this kind are commonly in use in various types of apparatus wherein a servo unit (such as a hydraulic servo-motor for operating a gun turret or hydraulic mechanism for actuating the landing gear or other accessories of an aircraft) is actuated by working fluid supplied under pressure by the pumping mechanism, which may be located remotely from the servo unit. In many of such installations there are a number of the servo units, all supplied from a common pumping source. Some of the requirements of such a system, as used for example in aircraft, include the ability to develop high operating pressures and to withstand elevated temperatures. A primary requirement for aircraft is compactness and minimum weight. Accordingly, a general object of the invention is to afford a compact, low weight pump, capable of developing high flow rates at high operating pressures and of withstanding the accompanying elevated temperatures, so as to be especially suitable for airborne and missile service. Heretofore, it has been possible to obtain such performance only in heavy industrial pumps. The present invention contemplates a pump capable of developing pressures as high as 5000 pounds per square inch, while operating (with a suitable fluid) at temperatures up to 450° Fahrenheit.

Existing pumping systems of the two-stage type with automatic feathering are characterized by the fact that the system, when operating, is constantly subjected to the maximum output load of the supercharger. In our system, the supercharger is designed to deliver an output in excess of the maximum requirement of the high pressure pump, yet the full output load of the supercharger is not imposed on the system except at maximum demand conditions. One of the important objects of the invention is to provide a two-stage pumping system embodying means to automatically unload the system at all times to whatever extent is dictated by reduction in load demands. Thus when the high pressure pump is fully feathered, the pump will be substantially unloaded, with the supercharger output reduced in pressure and bypassed back to reservoir. Also when the high pressure pump is operating at partial capacity, the supercharger output pressure will be substantially reduced below its maximum value.

A particular object of the invention is to provide an improved feathering piston pump of the multiple piston type. A further object is to provide such a pump, wherein the piston stroke is controlled in accordance with the

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rate at which working fluid is fed to the inlet of the pump.

Another object is to provide such a pump in combination with a controller adapted to vary the rate of feed of the working fluid to the high pressure pump (and to thereby vary the feathering of the high pressure pump) in accordance with load demand.

Another object is to provide a feathering pump of the axial piston and wobble cam type in which variation in piston stroke is attained without the necessity for varying the angle of inclination of the wobble cam to its axis. Accordingly, the invention contemplates such a pump in which a conventional adjusting mechanism for varying the inclination of the wobble cams, is completely eliminated.

The pump of our invention, in its operation, is particularly characterized by an intermittent engagement of the wobble cams with the bearing ends of plural pump pistons, with means for retaining control over the movements of the pistons when they are out of contact with the cams.

Another object of the invention is to provide a piston pump capable of operating at high temperatures and having improved piston expansion compensation characteristics, such as to reduce to a minimum, loss of volumetric efficiency over a wide range of temperature change. More specifically, the invention aims to provide a pump having the ability to compensate for temperature-induced viscosity changes in the working fluid, in a manner to maintain the efficiency factor at as nearly a constant high value as possible.

A further object is to extend upwardly the practical range of operating temperatures of a piston type pump.

Reference is made to the accompanying drawings, in which:

Fig. 1 is a schematic illustration of a pumping system embodying the invention;

Fig. 2 is an axial sectional view of a suitable high pressure pump of our invention;

Fig. 3 is a transverse sectional view through an end portion of the piston pump;

Fig. 4 is a transverse sectional view through the body of the piston pump;

Fig. 5 is a detail sectional view taken on the line 5—5 of Fig. 3;

Fig. 6 is a detail sectional view taken on the line 6—6 of Fig. 7; and

Fig. 7 is a detail sectional view of one of the piston shoes.

The system in general

Referring now to Fig. 1 of the drawings, we have shown, as an example of a system utilizing the invention, a hydraulic drive system in which a working fluid, maintained in a reservoir A, is withdrawn from the reservoir by a super-charger B which pumps it to the inlet of the high pressure pump C; the output of high pressure pump C being delivered to a servomotor or other fluid utilizing device, indicated at D and designated "LOAD"; the output of high pressure pump C being also connected to the pilot unit E of a controller which includes both the pilot unit E and an unloading and volume-control valve unit F. The pilot unit E is pressure sensitive, responding to the output pressure of pump C, which will tend to vary with changes in the load demand. For example, assuming that the load comprises the hydraulic actuating cylinders of an aircraft landing gear, connected to the pumping system by a suitable control valve (not shown) which is opened when the pilot desires to actuate the landing gear, the normal condition will be one in which the valve is closed, causing the pressure to build up to its set maximum value at the outlet of high pressure pump C, and this pressure will also be applied to the

pilot unit E of the control mechanism in a manner to reduce the flow of working fluid to the inlet of the high pressure pump C, thereby effecting the feathering of the pump C so as to correspondingly reduce its output in accordance with the reduced demand.

The present invention makes no effort to improve upon the construction of the supercharger B per se. Any conventional low pressure pump, such as the gear pump which is indicated schematically in Fig. 1, may be utilized. The main requirement for pump B is that it shall be capable of sufficiently high volume output to exceed the maximum demand for high rate of feed of working fluid to the pressure pump C. Obviously, it must be capable of matching the maximum pumping rate of pump C under all operating conditions.

To assure a pressure head of fluid delivered from super-charger B to the inlet of high pressure pump C at all times, the pump B is designed to deliver at an output rate in excess of the rate at which pump C can accept working fluid when operating at maximum capacity. Thus under all conditions, including operation of pump C "wide open," there will be available inlet pressure to drive the pistons of pump C to their fully extended positions.

The axial piston pump

Referring now to Fig. 2, the axial high pressure pump of our invention is particularly characterized by utilizing a balanced arrangement of plural pistons 10 in axially opposed aligned pairs, mounted in cylinder bores 11 in a cylinder block 12 of sleeve form, which is mounted within a cylindrical bore 13 in a housing 14. Pistons 10 have annular labyrinth oil grooves 15 for minimizing the escape of working fluid past the pistons. Such fluid control also effects lubrication of the cylinder bores 11, insures a uniform pressure gradient along the length of the piston engagement in the cylinder and hydraulically centers the piston in the cylinder. Extending through the central passage of cylinder block 12 is a tubular drive shaft 16 which provides an inlet passageway 17. Shaft 16 is journaled in roller bearings 18, 18' the outer races of which are mounted in the end portions of the cylindrical bore 13 of housing 14, and have end thrust abutting engagement with an end plate 19 closing one end of the housing and a cap 20 at the other end of the housing. Cap 20 is secured to the housing in any suitable manner as by means of cap screws 21. Annular chambers 22, 22' are defined between the ends of cylinder block 12 and end plate 19 and cap 20 respectively.

Bearings 18, 18' and other parts, hereinafter mentioned, are lubricated by diversion of a small portion of the supercharged liquid flowing in passageway 17, through lubricant ports 23, 23' in shaft 16. As will be more fully described hereinafter, the lubricant passing through ports 23, 23' will find its way into annular chambers 22, 22'. Fluid from chamber 22' will pass through cooling ducts 24 in cylinder block 12, for cooling the latter, and will join the fluid in chamber 22 to leave the pump through an outlet 25, whence it will return to reservoir A through a return line 26.

When installed in a pumping system, housing 14 is fixed in a stationary position, utilizing mounting feet 27; and when in operation, shaft 16 is rotated by any suitable drive connection, indicated at 28.

As indicated in dotted lines in Fig. 2, supercharger pump B is mounted within end cap 20 at one end of the high pressure pump A, cap 20 providing a casing for pump B. One of the gears 29 of super-charger B is splined upon shaft 16 as indicated, whereby both pumps are driven by the same drive shaft. The controller units E, F may also be mounted within end cap 20 as indicated in Fig. 2. A supercharger pump suction inlet 30 also is provided in end cap 20. A suction line 30' leads to pump B from reservoir A to suction inlet 30 (Fig. 1).

Rotating with shaft 16 are a pair of wobble cams 31, 31', which may be splined on the shaft as indicated at 32,

32'. Cams 31, 31' are backed up by respective end thrust bearings in the form of relatively thin washers 33, 33' of compressible semi-elastic, absorbent, wear resistant material such as plastic impregnated fabric or glass fiber, or soft metal fabric; each of the bearing washers 33 having a series of segmental flat bearing faces or lands 34 separated by radial grooves 35 (see Fig. 3). Washers 33, 33' in their normal, unstressed state are flat and of uniform thickness (except at grooves 35) and lands 34 are perfectly flat and co-planar. Lands 34 bear against rigid thrust washers 37, 37' which back up the bearing washers 33, 33'. Washers 37, 37' are slidably keyed in housing 14 as by means of a key 38 on the periphery of each washer, engaged in a keyway 39 in the housing. Washers 33, 33' rotate with cams 31, 31'.

Grooves 35 function for distributing lubricant between lands 34 and the coating faces of thrust washers 37, 37'. Lubricant flowing through passages 23, 23' will flow into clearance spaces between the inner races of bearings 18, 18' and end plate 19 and caps 21 respectively, thence axially through the bearings, thence radially inwardly through clearance spaces between the inner bearing races and thrust washers 37, 37' respectively, thence axially through the clearance spaces between the inner margins of the thrust washers and shaft 16, thence radially outwardly through slots 35 in bearing washers 33, 33', thence axially in the clearance spaces between the peripheries of the wobble cams 31, 31' and bore 13, into the annular chambers 22, 22'. From chambers 22, 22', the flow returns to reservoir A, as previously explained. As the fluid travels radially outwardly in grooves 35, it comes into contact with the bearing faces of thrust washers 37, 37'. As they rotate relative to the thrust washers 37, 37', the washers 33, 33' will be compressed to form wedge shaped pockets 40 of minute depth, loaded with hydraulic fluid which supports the end thrust load. This is accomplished in such a manner that the load is completely supported by the lubricant film between these bearing surfaces, rather than by actual physical contact of the mating bearing surfaces, in spite of the fact that they are subjected to intensely high axial pressures. As indicated in Fig. 5, these wedge shaped pockets are complete on each bearing segment 34, and exist by virtue of the semi-elastic, compressible, absorbent characteristics of the bearing material and not due to any pre-shaping of the washers. This results in a novel bearing that will operate equally well regardless of the direction of movement and will properly adjust itself to meet various speed and load conditions. A suitable material for element 33 is commercial fabric phenolic sheet, prepared with lapped flat segments 34.

The opposite or forward faces of wobble cams 31, 31' (Fig. 2) are inclined so as to define dihedral angles relative to the back bearing faces referred to above. These forward faces are recessed, being defined between inner and outer marginal flanges 41. In the annular face grooves thus formed are mounted annular cam facing washers 42, 42', which may be of a wear resistant bearing material similar to that of washers 33, 33'. On the ends of pistons 10 are ball heads 43. Bearing shoes 44 (Fig. 7) are provided with sockets 45 and are coupled to pistons 10 by the reception of ballheads 43 in sockets 45. Opposite sockets 45, shoes 44 are provided with annular bearing lands 46 which are flat and are adapted to bear against the cam face members 42. Defined within the lands 46, which are annular in form, are pockets 47 adapted to be supplied with oil under pressure for hydraulically balancing the shoes against the cam face members 42. Lands 46 may be provided with annular concentric oil seal grooves as shown.

As shown in Fig. 6 the shoes 44 are tied together by aligning coupler rings 36, having notches or elongated holes 36a for receiving reduced medial waist portions of the respective shoes, defined by medial annular grooves 36b in which the coupler rings 36 are loosely fitted (suffi-

ciently to allow tilting of a shoe 44 to a slight angle of inclination of the face plane of its lands 46 relative to the plane of coupler ring 36 when the latter is tilted away from cam face washer 42 or 42' as shown in Fig. 2; thus allowing the lands 46 to seal flatly against the cam face washer). Alternatively, the coupler ring 36 may be of a sufficiently flexible material (e. g. a lightly plasticized synthetic resin thermo-plastic material hardened synthetic rubber) so that the ring itself may flex in the area of the shoe that is making full contact with the cam facing. Coupler rings 36 function to maintain their respective sets of shoes 44 at all times in general planar alignment, and thereby dictate the relative positions of pistons 10 which are coupled to the shoes 44.

Mounted in radial bores in cylinder block 12, directly between the ends of cylinders 11, are ball seat cages 48 each having an outer seat 49 for an outlet valve 50 and an inner seat 51 for an inlet valve 52. Valves 50 and 52, as shown, may be of the ball-check type, urged toward seating positions by conical coil springs 53, 54 respectively. Inlet ports 56, extending radially through the tubular wall of shaft 16, communicate with an annular inlet passage 57 in cylinder block 12 surrounding shaft 16. Annular inlet passage 57 in turn communicates with the inlet ports defined within valve seats 51. Lateral ports 58 in cages 48 provide communication between inlet valve seats 51 and piston cylinders 11, and also provide communication between the piston cylinders and outlet ports which are defined within outlet valve seats 49. The open outer ends of cages 48 communicate with an annular outlet passage or manifold 60 in housing 14. Passage 60 in turn communicates with a delivery passageway 61, in housing 14, which is coupled with pilot valve E through a duct 87, and is also (as best shown in Fig. 1) coupled to a fluid pressure line 88 leading to the servo unit to be supplied by the pump. On either side of annular outlet passage 60, cylinder block 12 is sealed to housing 14 by annular packing glands 62. Similarly, the interior of cylinder block 12, at the respective ends thereof, is sealed to the outer face of shaft 16 by packing glands 63 disposed in counterbores in the end of cylinder block 12.

Oil under pressure is supplied to the hydraulic balancing pockets 47 (best shown in Fig. 7) by capillary passages 64 extending axially through pistons 10 and through ball heads 43 and communicating with small ports 65 in the centers of shoes 44.

Viscosity compensation

One of the problems in high temperature, high pressure piston pumps—one that has created difficulty for engineers throughout many years of effort to overcome it, is the problem of attempting to maintain high volumetric efficiency of the pump pistons at all times without causing the pistons to freeze in their cylinders, to become scored or galled, or to exercise an excessive wearing action on the cylinder bores. Modern finishing techniques have made it possible to observe very close tolerances in the fitting of pistons to their cylinders. This, however, does not solve the problem, which is complicated by two variable factors, namely (a) the change in viscosity of the working fluid in response to change in temperatures; and (b) the effect of expansion of piston and cylinder in response to temperature changes.

The problem is made especially difficult by the fact that the escape of fluid through the clearance spaces between pistons and cylinders increases directly in proportion to the increase in the length of the engaging piston surfaces and as the cube of the clearance. At extremely high temperatures, the working fluids now considered suitable for use in hydraulic drive systems, (usually a hydrocarbon oil) approach the viscosity of water, although most of them are fairly viscous at low temperatures.

The present invention deals with this problem by utilizing, as the material for the pistons, a metal alloy which expands at a faster rate than the metal of the

cylinder block, whereby the clearance between pistons and cylinders will decrease as temperature increases.

More specifically, the invention provides for selecting a ratio between the coefficient of expansion of the piston and that of the cylinder, such that the decrease in clearance will be so proportioned to temperature increase as to compensate for the thinning of the working fluid under such temperature increase. The optimum result aimed at by the invention is to proportion the value of clearance decrease to the value of viscosity decrease so as to approximate a constant ratio between these variables, and at the same time to cause the clearance between pistons and cylinders to decrease as temperature increases, but without binding.

More specifically, the invention provides for selecting a ratio between the coefficient of expansion of the pistons and that of the cylinders, such that the decrease in clearance will be so proportioned to temperature increase as to compensate for the thinning of the working fluid under such temperature increase. The optimum result aimed at by the invention is to so proportion the curve of clearance decrease to the curve of viscosity decrease so as to approximate a constant ratio between these variables, and thereby to compensate for the temperature-induced change in the fluid escape factor referred to above, in a manner to maintain the pump's volumetric efficiency at a constant high value.

As an example of a reduction to practice of this concept which we have found to be highly successful in approaching this optimum goal, the cylinder block may be constructed of steel with cylinder walls hardened by nitriding, and the pistons may be constructed from heat treated, hardened, beryllium-copper. The respective coefficients of expansion of these materials are such as to achieve the proportionate decrease in piston clearance in step with temperature increase, as outlined above. As another example, the pistons could be constructed of hardened steel and the cylinder block of another alloy having a lower coefficient of expansion than hardened steel, such as "Invar" metal.

The primary advantage obtained by this improvement is in providing for a substantial increase in volumetric efficiency at elevated operating temperatures. In accordance with the results indicated by tests that we have made on the preferred example given above, the invention makes it possible to obtain a 50% increase in operating temperatures over the maximum temperatures at which it has been possible in the past to operate piston pumps without reducing efficiency below minimum requirements.

In addition to the use of metals having the novel expansion characteristics indicated above, the invention is further characterized by fitting the pistons to their cylinders with minimum clearance at the maximum operating temperature, after having determined in advance the minimum clearance which can be tolerated without encountering the danger of freezing or galling of the pistons in the cylinders. More clearance is satisfactory at lower temperature because the working fluid is then more viscous. Accordingly, at minimum temperatures (normal atmospheric temperatures) the pistons have maximum clearance in the cylinders, corresponding to maximum viscosity of the working fluid. As temperature increases, piston clearance will decrease in step with decrease in viscosity of the working fluid.

The sealing effectiveness of the labyrinth sealing grooves 15 will increase as the clearance decreases. We find that by determining the minimum clearance at maximum temperature, thereby avoiding an interference fit at any temperature, there is a complete avoidance, throughout the range of operation of the pump, of any galling of the pistons or cylinders, and the surfaces of these parts simply become polished and burnished to a high degree, which extends the high efficiency of operation of the pump indefinitely.

A particular advantage arising from the novel use of

beryllium-copper as the material for the pistons, is the high heat conductivity of that material, which avoids any localized hot spots on the piston surfaces or cylinder walls and thereby promotes uniformity of clearance throughout the engaging areas of the pistons and cylinders.

Volume control and bypass valve F

As best shown in Fig. 1, supercharged fluid is delivered from the outlet of supercharger B through a fluid line 66 to a passage 66' in the casing 20' (a portion of end cap 20) of valve unit F. Valve unit F provides two separate paths, both branching from passages 66', for bypassing excess working fluid back to reservoir A. Normal by-passing, to dispose of supercharged fluid rejected by the pressure pump C when operating at less than full capacity, is through a spool valve bore 67 in casing 20', past a piston 68 of a double piston spool valve which is slidable in bore 67, into an annular port 69 communicating with bore 67, thence into a return passage 70' leading directly back to the reservoir through a return line 70. Such bypassing occurs at pressures below maximum supercharger outlet pressure.

When the supercharger outlet pressure tends to exceed its maximum value (when pump C is operating at full capacity) excess fluid will be relieved through a pressure relief port 71 past a pressure relief valve 72, into return passage 70'. Relief valve 72 may be of the poppet type as shown. A loading spring 73, adjustably backed up by an abutment screw 74, determines the pressure at which valve 72 will open port 71. The relief poppet 72 will establish the normal supercharger operating pressure. It is intended to operate only when the main piston pump C is in a maximum displacement, or non-feathered position. Since the supercharger pump is designed to have a capacity slightly in excess of the piston pump C, only this excess fluid will be bypassed through relief valve 72.

A second piston 75 is connected to piston 68 by a stem 76 to form the complete spool valve element. The inner edge of piston 75 controls two semi-circular ports 77, 78 respectively in the casing 20'. Port 77 provides for passage of fluid from the passageway 66' into a central chamber 79 defined between the pistons 68 and 75, within the spool valve bore 67. The operation of piston 75 in throttling port 78 controls the rate of flow of fluid from chamber 79 through a connecting line 80 to the inlet passage 17 of the high pressure pump C. Ports 77, 78 are slightly offset; port 77 remaining open when piston 75 has closed off port 78.

Piston 68 controls the escape of fluid from chamber 79 through annular port 69, into return passage 70'. As the two pistons move leftwardly as viewed in Fig. 1, the aperture of port 78 will be decreased while that of port 69 is increased, thus bypassing an increasing amount of fluid through the reservoir and decreasing the flow through line 80 to pump C.

The movement of spool valve 68, 75 is governed by the opposed forces of fluid pressure acting against the outer end of piston 75 and a spring 81 acting against piston 68. Spring 81 is adjustably backed up by an abutment screw 82, the axial position of which may be varied so as to vary the positions occupied by spool valve 68, 75 in response to changes in the discharge pressure of pump C. By advancing screw 82, the discharge pressure of pump C required to shut off flow through line 80 will be increased. Screw 82 may be pre-set and locked in place (as by staking).

The valve unit F actually comprises two separate valves, i. e., the volume control valve 67-69, 75-82, and the bypass valve 71-74, the latter being located between the supercharger B and the volume control valve. Although, for compactness, these valves are illustrated as being combined in a single unit, they could be physically separated, it being only necessary to extend ports 70' and 66' into passages of adequate length to connect the

two valves. Similarly, the valve units E and F, though shown in Fig. 2 as being built into a common casing, could be physically separated as suggested in Fig. 1.

Obviously, in order to function as intended, the bypass valve 71-74 must open at a pressure lower than the maximum rated output pressure of supercharger B.

The latter pressure need only be high enough to maintain flow to conduit 80 when port 78 has been throttled down to such restriction as will result in minimum flow to pump C, and to effect opening of inlet valves 50. This pressure may be quite low as compared to the pressures developed by pump C, and thus the unloading is effective to minimize working losses.

Volume control pilot E

The volume control pilot E embodies a casing 85 having a bore 86 which communicates, through a fluid line 87, with the delivery line 88 through which working fluid is conveyed from the outlet of pressure pump C to the servomotor or other device constituting the load D. The piston 89, slidable in bore 86, is subjected at its outer end to the discharge pressure of pump C through line 87, its inner end transmitting this pressure to a spring 90 through a spring retainer cup 91 having a stem 92 against which the inner end of piston 89 engages. Retainer cup 91 is slidably mounted in a differential pressure chamber 93 in casing 85, into which pressure fluid from line 87 is introduced through a passage 94 in casing 85, to reach the rear side of cup 91, thence through ports 95 in cup 91 as indicated by the arrows. A transfer line 96, branching from passage 94, leads to a fluid chamber 97 in the rear end of the volume control valve F, and acts upon the volume control piston 75 of valve F in accordance with the pressure in passages 94 and 96.

The pressure in passage 94 is a function of the pressure in line 87 and is also a function of the rate at which fluid is bled out of chamber 93 of pilot unit E past a needle type bleed valve 101, the latter controlling the bleeding of fluid from chamber 93 through a port 102 to a radial passage 103 in a plug 104 (constituting a casing for the needle valve 101). From an annular peripheral groove 105 in plug 104, with which passage 103 communicates, the fluid which has bled past needle valve 101 escapes through a passage 106 in pilot casing 85 and thence passes through a return line 107 which may join the return line 108 leading from the servomotor D to the reservoir, as shown in Fig. 2, or the return line 26, as shown in Fig. 1, or may join the suction line 30' to pump B.

The apertures between needle valve 101 and port 102 may be varied by operating an adjusting screw 110, threaded into sleeve 104, the needle valve 101 constituting an end portion of screw 110. Further adjustment of the pilot unit E is provided for by an adjusting knob 111 having a threaded connection with casing 85 and formed as an integral head on the outer end of sleeve 104. By rotating knob 111, sleeve 104 is axially shifted in chamber 93, and since one end of spring 90 abuts against the inner end of sleeve 104, the compression loading of the spring is thereby varied. Correspondingly, the spring load which is applied to the inner end of piston 89 by stem 92 will be varied, so that the hydraulic pressure required to operate the pilot piston 89 (to adjust the pump feathering pressure) may be externally adjusted. It will now be apparent that piston 89 operates as a slide valve to vary the effective area of port 94', thereby varying the relation between the pressure in line 87 and the pressure in line 96. The said lines may be wholly or partly in the form of drilled passageways in the pump body, as indicated in Fig. 2.

Referring to the pilot E as shown in Fig. 1, the area of port 94' will vary in accordance with fluctuations in the discharge pressure of pump C. The pressure in differential pressure chamber 93, passage 94, transfer line 96 and chamber 97 of volume control valve F is gov-

erned by the ratio between the rate of passage of fluid into passage 94 through port 94' and the escape of fluid past needle valve 101 into return line 107, and is therefore a function of the discharge pressure of pump C which governs the area of port 94'. This function can be adjusted, both as to pressure and rate of feathering, by actuating adjusting knob 111 and adjusting screw 110 respectively.

The pressure pump C of our invention is particularly characterized by its ability to achieve feathering or piston stroke adjustment in response to changes in its own discharge pressure, without utilizing conventional tiltable mountings of the wobble cams and pressure responsive means for adjusting the inclination of the cams. The cams of our pump are fixed to shaft 16 with fixed angles of inclination of their bearing faces 42, and, except at maximum piston displacement, the shoes 44 will contact the cam faces only during a portion of their paths of rotary movement. The contacts will be made of course in the "high" regions of the cam faces, and in the low areas thereof the shoes will draw away from the cam faces.

The factor which governs the piston stroke is the rate of delivery of fluid into cylinders 11 through inlet ports 58. This delivery rate, in turn, is controlled by the operation of volume control valve F in response to changes in the discharge pressure of pump C, varying the proportion of the output of super-charger B which is delivered into line 89 and the proportion that is bypassed through return line 70.

Planar substantially circular paths of travel of shoes 44 are dictated by coupler rings 36, best shown in Fig. 6. The retracted position of a piston 10 is always the same, being determined by engagement of a shoe 44 against the high point of the cam face, as best shown in Fig. 2. As the shoe 44 passes the high point, fluid pressure in inlet port 58 will act against piston 10 to push it outwardly, so that the shoe 44 tends to follow the cam face down the incline toward its low point (point of maximum separation of shoes 44 from the cam face).

Similar action takes place sequentially in the other cylinders. However, the outward movement of pistons 10, directly governed by the rate of inlet flow, will not be fast enough to maintain the shoes in contact with the cams except during maximum volume operating conditions. Consequently, the shoes, after having been pushed inwardly to the fully retracted positions by engagement with the cams, will separate from the cam faces and will follow paths which are inclined at a lower angle of inclination than that of the cam faces, the divergence between these angles increasing with a decrease in the rate of inlet flow.

Inlet and outlet valves 52 and 50, being spring controlled check valves, will move in accordance with the conditions of fluid pressure affecting them. As a piston is pushed inwardly by cam operation, the pressure of fluid at the inner end of the piston will quickly build up to a value sufficiently near that of pressure in inlet passage 17, so as to cause inlet valve 52 to be closed by its operating spring. This build up in pressure will increase the pressure tending to unseat outlet valve 50 until the spring 54 yields and allows valve 50 to open, whereupon the fluid pushed ahead of pistons 10 will be expelled through the outlet ports into the outlet manifold 60, thence to the trunk outlet 61 from which it is delivered to the delivery line 88.

As the shoes 44 pass the high points of the cams, the cam pressure against the pistons will be relieved and fluid pressure at the ends of the pistons will quickly drop to the point where outlet valves 50 will close and inlet valves 52 will open in response to the pressure in inlet passage 17. This stage of operation is shown in Fig. 2. Because of the pressure drop across inlet valves 52, the pressure of fluid passing the inlet valves and entering inlet ports 58 will be insufficient to unseat outlet valves 50. It will however, be sufficient to return pistons

10 outwardly at a rate corresponding to the rate of inlet flow. Thus the piston stroke is fully controlled by the rate of inflow, and a corresponding rate of discharge from the outlet 61 will be observed.

The pump will operate with any suitable hydraulic fluid. No means for extending the pistons against the wobble cams, other than the supercharger pressurized fluid, is required in the pump of our invention. Since the "suction" or return stroke is not mechanically actuated, but rather is affected by the low pressure fluid from the supercharger pushing the pistons against the wobble cams and filling the cylinder with fluid, it is impossible for the piston pump to cavitate. Since the supercharger pump is working only under relatively light loads it is of such a design as to be capable of operating under very low negative pressure heads (i. e. high suction) on the suction port; and could resemble a conventional vacuum pump. An cavitation that may take place would then occur (harmlessly) at the supercharger pump. Since the supercharger pump does not require as exact machining tolerances etc., the phenomenon of cavitation may more readily be tolerated at this point in the circuit. Because of the aforementioned feature, efficient abnormally high altitude operation, required for modern aircraft usage, is attained by the invention.

Since the shoes 44 are out of contact with the wobble cams during a majority of their operating time, wear on the shoes is reduced to a minimum.

The bleeding of pressure fluid through piston passages 64, which are of capillary dimensions, serves to produce fine high velocity jets through ports 65, which jets will impinge upon cam facings 42 as the shoes 44 approach the same, thereby cushioning the impact of the shoes against the facing. As the peripheral lands of a shoe approach contact with the cam facing sufficiently closely to restrict the escape out of chamber 47 of the fluid coming from this jet, the cushioning action is built up in chamber 47 so that the actual contact of the shoe with the facing 42 is completely devoid of "slap" or jar, and there can be no rebound or chattering action under these conditions. Accordingly, the intermittent cam action is smooth, gradual, and lacking in vibration or chatter. As the shoe makes contact, it will adjust its position to one of flat face to face contact between its lands and the cam facing, tilting slightly, if necessary, in order to do so.

The operation of volume control valve F may be made extremely sensitive because of the pressure reduction which occurs across port 94' of pilot unit E. The metering action of piston 89 in pilot unit E, in cooperation with the bleed past valve 101, reduces the extremely high pressure in line 87 to a low value of operating pressure in line 96 which may be adjusted to exactly the right value for optimum performance of valve F, while maintaining a fixed ratio between the high pressure fluctuations in line 87 and the induced low pressure fluctuations in line 96.

An important aspect of the invention is the provision for unloading the system. Although the supercharger B provides a constant output in excess of maximum demand of pump C, its maximum output pressure will be attained only when pump C is operating at full capacity. Pressure in lines 87, 96 will then be at a minimum value, and spool valve 68, 75 will be moved to an extreme rightward position, completely shutting off bypass 70' and fully opening passage 78. At such times, relief poppet 72 comes into operation to effect high-pressure relief bypassing of excess fluid. When load demand drops off (e. g., as by restricting passage 88) pressure in lines 87 and 96 will increase, spool valve 68, 75 will be driven leftward, opening bypass 70' and dropping the pressure in lines 66 and 80 so as to reduce the loading between the outlet of the supercharger and the inlet of pump C. In contrast, conventional two-stage pumping systems maintain maximum supercharger outlet pressure imposed on the pressure pump inlet at all times. Our improvement ac-

cordingly reduces the power consumed at drive element 28 of the pump because the pump combination is not required to dissipate a large amount of power in the form of heat while its high pressure pistons operate at minimized strokes. Such economical operation is possible during a major portion of the cruise of a modern airplane but the pump is nevertheless always in readiness to serve whenever and as much hydraulic power as is necessary, up to its maximum capacity or any fraction of this amount that may be required.

In referring to the paths of travel of shoes 44 as being substantially circular, it is to be understood that these paths are almost circular, or slightly elliptical.

The term "constant output" as used herein to refer to the supercharger, is to be understood to mean constant output for any given speed of operation of the supercharger. Stated somewhat differently the term indicates that the supercharger is a constant displacement pump.

We claim:

1. In a two-stage hydraulic pumping system, in combination: a constant output, low pressure supercharger; a variable output, automatically feathering high pressure pump; a combined volume control and bypass valve unit having an inlet port receiving fluid from the outlet of said supercharger, and having a delivery outlet and a bypass outlet, and having a bypass leading from said inlet port to said bypass outlet; a connecting fluid line extending from said delivery outlet to the inlet of said high pressure pump; means defining fluid passages for delivering the output of said high pressure pump to a load and for returning spent fluid from the load to the inlet of said supercharger; said volume control and bypass valve unit including a pressure responsive valve for metering the flow from its said inlet to said connecting line and bypassing the excess to said bypass outlet, and a pressure responsive bypass valve in said bypass, said bypass valve being adapted to open in response to supercharger outlet pressure at a selected maximum value thereof, and to limit at said maximum value, the outlet load of the supercharger by bypassing back toward the supercharger inlet, that portion of the supercharger output which is in excess of the amount accepted by the high pressure pump; and a pilot valve responsive to fluctuations in the output pressure of said high pressure pump, for supplying to said volume control valve, pressure fluid derived from said high pressure pump output, for actuating said volume control valve in a manner to throttle the flow from the supercharger to the high pressure pump so as to furnish to the latter only that quantity of working fluid which is required to meet the load demand.

2. In a two-stage hydraulic pumping system, in combination: a constant output, low pressure supercharger; a variable output, automatically feathering high pressure pump; a volume control unit having an inlet port receiving fluid from the outlet of said supercharger and having a delivery outlet; a connecting fluid line extending from said delivery outlet to the inlet of said high pressure pump; means defining fluid passages for delivering the output of said high pressure pump to a load and for returning spent fluid from the load to the inlet of said supercharger; said volume control unit including a pressure responsive valve for metering the flow from its said inlet to said connecting line and bypassing the excess to said bypass outlet, said valve controlling inlet pressure of the high pressure pump when the latter is operating in a feathered condition; means defining a bypass from the outlet to the inlet of said supercharger, bypassing said volume control unit, high pressure pump and load; a pressure responsive bypass valve in said bypass, said bypass valve being in response to supercharger outlet pressure at a selected maximum value thereof, and to limit at said maximum value, the outlet load of the supercharger by bypassing back toward the supercharger inlet, that portion of the supercharger output which is in excess of the amount accepted by said high pressure pump; and a pilot

valve responsive to fluctuations in the output pressure of said high pressure pump, for applying to said volume control unit, pressure fluid derived from said high pressure pump output, for actuating said volume control unit in a manner to throttle the flow from the supercharger to the high pressure pump so as to furnish to the latter only that quantity of working fluid which is required to meet the load demand.

3. In a two-stage hydraulic pumping system, in combination: a constant output, low pressure supercharger; a variable output, automatically feathering high pressure pump; a volume control unit having an inlet port, a delivery outlet and a bypass outlet; a conduit for conducting fluid from the outlet of said supercharger to said inlet port; a connecting fluid line extending from said delivery outlet to the inlet of said high pressure pump; means defining fluid passages for delivering the output of said high pressure pump to a load and for returning spent fluid from the load to the inlet of said supercharger; said volume control unit including a pressure responsive valve for metering the flow from its said inlet to said connecting line while bypassing the excess to said bypass outlet, said valve controlling inlet pressure of the high pressure pump when the latter is operating in a feathered condition; means defining a bypass flow path from said bypass outlet to said supercharger inlet; a bypass leading from said supercharger outlet conduit to said bypass path, bypassing said volume control unit; a pressure responsive bypass valve in said bypass, adapted to open in response to supercharger outlet pressure at a selected maximum value thereof, and to limit at said maximum value, the outlet load of the supercharger by bypassing back toward the supercharger inlet, that portion of the supercharger output which is in excess of the amount accepted by said high pressure pump; and a pilot valve responsive to fluctuations in the output pressure of said high pressure pump, for supplying to said volume control unit, pressure fluid derived from said high pressure pump output, for actuating said volume control unit in a manner to throttle the flow from the supercharger to the high pressure pump so as to furnish to the latter only that quantity of working fluid which is required to meet the load demand.

4. In a two-stage hydraulic pumping system, in combination: a constant output, low pressure supercharger; a variable output, automatically feathering high pressure pump; a volume control unit comprising a casing having a bore, an end member closing one end of said bore and an axially adjustable abutment member closing the other end of said bore, said casing further having an inlet and a delivery outlet both spaced from said end member, and a bypass outlet spaced from said inlet in the direction away from said end member, said volume control unit further including a valve comprising a control piston, a bypass piston and a stem connecting said pistons in axially spaced relation, said valve being slidable in said bore with said control piston disposed between said end member and said inlet and delivery outlets and with said bypass piston disposed between said bypass outlet and said adjustable abutment member, and a spring interposed between said bypass piston and said adjustable abutment member and loading said valve, one end of said control piston cooperating with said end member to define in said bore a control chamber, in which fluid pressure may act against said control piston to balance the load of said spring and to shift the valve in response to fluid pressure changes in said control chamber, the other end of said control piston being disposed for throttling said delivery outlet in response to pressure increase in said control chamber, said bypass piston being disposed for throttling said bypass outlet as said delivery outlet is opened, the space between said pistons providing for fluid flow from said inlet to said delivery outlet and to said bypass outlet; a conduit for conducting fluid from the outlet of said supercharger to said inlet port; a connecting fluid line extending from said delivery outlet to

the inlet of said high pressure pump; means defining fluid passages for delivering the output of said high pressure pump to a load and for returning spent fluid from the load to the inlet of said supercharger; said volume control unit functioning to meter the flow from its said inlet to said connecting line while bypassing the excess to said bypass outlet; means defining a bypass flow path from said bypass outlet to said supercharger inlet; a bypass leading from said supercharger outlet conduit to said bypass path, bypassing said volume control unit; a pressure responsive bypass valve in said bypass and adapted to open so as to unload the supercharger at a selected maximum output pressure thereof by bypassing back toward the supercharger inlet, that portion of the supercharger output which is in excess of the amount accepted by said high pressure pump; and a pilot valve responsive to fluctuations in the output pressure of said high pressure pump, for supplying to said volume control unit, pressure fluid derived from said high pressure pump output, for actuating said volume control unit in a manner to throttle the flow from the supercharger to the high pressure pump so as to furnish to the latter only that quantity of working fluid which is required to meet the load demand.

5. In a two-stage hydraulic pumping system: an axial piston type automatically feathering, variable output, high pressure fluid pump comprising a drive shaft; a casing having therein a plurality of cylinders arranged in axially opposed groups comprising pairs of axially opposed cylinders, in circumferentially spaced annular array, coaxial with and surrounding said shaft; pistons in said cylinders arranged in groups corresponding to the cylinder groups; a pair of wobble cams surrounding and secured to said shaft and disposed at opposite ends of said cylinders, said cams having annular piston-operating faces disposed in planes that are inclined in opposite directions with reference to the axis of said shaft, with fixed angles of inclination, and with their respective high points disposed in axially opposed alignment; bearing shoes attached to the outer ends of the respective pistons and engageable against said cam faces with only intermittent contact with the high spots thereof when the pump is operating at less than full capacity; said casing having an inlet and a delivery outlet; a plurality of valve units each interposed between adjacent ends of the cylinders of a respective pair and embodying an inlet valve disposed for controlling flow from said inlet to respective cylinders and an outlet valve disposed for controlling flow from respective cylinders to said outlet, said valves being spring biased toward their closed positions; a low pressure supercharger for delivering to said inlet a variable flow of fluid which is operable to successively move apart the pistons of succeeding pairs of cylinders, with the stroke of the pistons varying directly in response to the rate of delivery of the fluid to said inlet; means for effecting relative rotation between said cylinders and cams, whereby to cause each pair of pistons to be successively returned into its cylinders to effect a high pressure pumping action wherein the fluid is forced through the respective outlet valve to said casing outlet; a volume control unit having an inlet port receiving fluid from the outlet of said supercharger and having a delivery outlet; a connecting fluid line extending from said delivery outlet to the inlet of said high pressure pump; means defining fluid passages for delivering the output of said high pressure pump to a load and for returning spent fluid from the load to the inlet of said supercharger; said volume control unit including a pressure responsive valve for metering the flow from its said inlet to said connecting line; means defining a bypass from the outlet to the inlet of said supercharger, bypassing said volume control unit, high pressure pump and load; a pressure responsive bypass valve in said bypass and adapted to open so as to unload the supercharger at a selected maximum output pressure thereof by bypassing back toward the super-

charger inlet, that portion of the supercharger output which is in excess of the total amount that is accepted for handling by said high pressure pump; and a pilot valve responsive to fluctuations in the output pressure of said high pressure pump, for supplying to said volume control unit, pressure fluid derived from said high pressure pump output, for actuating said volume control unit in a manner to throttle the flow from the supercharger to the high pressure pump so as to furnish to the latter only that quantity of working fluid which is required to meet the load demand.

6. An axial piston type automatically feathering variable output fluid pump comprising: a drive shaft; a casing having therein a plurality of cylinders arranged in axially opposed groups comprising pairs of axially opposed cylinders, in circumferentially spaced annular array, coaxial with and surrounding said shaft; pistons in said cylinders arranged in groups corresponding to the cylinder groups; a pair of wobble cams surrounding and secured to said shaft and disposed at opposite ends of said cylinders, said cams having annular piston-operating faces disposed in planes that are inclined in opposite directions with reference to the axis of said shaft, with fixed angles of inclination, and with their respective high points disposed in axially opposed alignment; bearing shoes attached to the outer ends of the respective pistons and engageable against said cam faces with only intermittent contact with the high spots thereof when the pump is operating at less than full capacity; a pair of coupling means each coupling together the shoes of a respective group, for constraining the group of shoes to remain in a common plane while tilting to varying angles of inclination; said casing having an inlet and a delivery outlet; a plurality of valve units each interposed between adjacent ends of the cylinders of a respective pair and embodying an inlet valve disposed for controlling flow from said inlet to respective cylinders and an outlet valve disposed for controlling flow from respective cylinders to said outlet, said valves being spring biased toward their closed positions; means for delivering to said inlet a variable flow of fluid which is operable to successively move apart the pistons of succeeding pairs of cylinders, with the stroke of the pistons varying directly in response to the rate of delivery of the fluid to said inlet; and means for effecting relative rotation between said cylinders and cams, whereby to cause each pair of pistons to be successively returned into its cylinders to effect a high pressure pumping action wherein the fluid is forced through the respective outlet valve to said casing outlet.

7. An axial piston type automatically feathering variable output fluid pump comprising a drive shaft; a casing having therein a plurality of cylinders arranged in axially opposed groups comprising pairs of axially opposed cylinders, in circumferentially spaced annular array, coaxial with and surrounding said shaft; pistons in said cylinders arranged in groups corresponding to the cylinder groups; a pair of wobble cams surrounding and secured to said shaft and disposed at opposite ends of said cylinders, said cams having annular piston-operating faces disposed in planes that are inclined in opposite directions with reference to the axis of said shaft, with fixed angles of inclination, and with their respective high points disposed in axially opposed alignment; the respective pistons having at their outer ends, means engageable against said cam faces with only intermittent contact with the high spots thereof when the pump is operating at less than full capacity; a pair of coupling means each coupling together the pistons of a respective group, for constraining the ends of the pistons of the group to remain in a common plane while tilting to varying angles; said casing having an inlet and a delivery outlet; a plurality of valve units each interposed between adjacent ends of the cylinders of a respective pair and embodying an inlet valve disposed for controlling flow from said inlet

to respective cylinders and an outlet valve disposed for controlling flow from respective cylinders to said outlet, said valves being spring biased toward their closed positions; means for delivering to said inlet a variable flow of fluid which is operable to successively move apart the pistons of succeeding pairs of cylinders, with the stroke of the pistons varying directly in response to the rate of delivery of the fluid to said inlet; and means for effecting relative rotation between said cylinders and cams, whereby to cause each pair of pistons to be successively returned into its cylinders to effect a high pressure pumping action wherein the fluid is forced through the respective outlet valve to said casing outlet.

8. An axial piston type automatically feathering variable output fluid pump comprising a drive shaft; a casing having therein a plurality of cylinders arranged in axially opposed groups comprising pairs of axially opposed cylinders, in circumferentially spaced annular array, coaxial with and surrounding said shaft; pistons in said cylinders arranged in groups corresponding to the cylinder groups; a pair of wobble cams encircling and secured to said shaft and disposed at opposite ends of said cylinders, said cams having annular piston-operating faces disposed in planes that are inclined in opposite directions with reference to the axis of said shaft, with fixed angles of inclination, and with their respective high points disposed in axially opposed alignment; the respective pistons having at their outer ends, means engageable against said cam faces with only intermittent contact with the high spots thereof when the pump is operating at less than full capacity; said casing having an inlet and a delivery outlet; a plurality of valve units each interposed between adjacent ends of the cylinders of a respective pair and embodying in inlet valve disposed for controlling flow from said inlet to respective cylinders and an outlet valve disposed for controlling flow from respective cylinders to said outlet, said valves being spring biased toward their closed positions; means for delivering to said inlet a variable flow of fluid which is operable to successively move apart the pistons of succeeding pairs of cylinders, with the stroke of the pistons varying directly in response to the rate of delivery of the fluid to said inlet; and means for effecting relative rotation between said cylinders and cams, whereby to cause each pair of pistons to be successively returned into its cylinders to effect a high pressure pumping action wherein the fluid is forced through the respective outlet valve to said casing outlet.

9. A pump as defined in claim 8, including a cylinder block of sleeve form having a plurality of axial bores constituting said cylinders, and having a plurality of radial recesses disposed between adjacent ends of paired cylinders; said valve units each including a cage having respective inlet and outlet ports and valve seats encircling the same, said ports aligned on a common radial axis substantially coaxial with the respective recess, and said cages each having a pair of diametrically opposed lateral ports communicating with respective cylinder bores and with the space within the cage between the respective valve seats, said recesses being open at the periphery of said cylinder block and said cylinder block being removable endwise from said casing to provide access to said valve units.

10. A pump as defined in claim 9, wherein said casing has, intermediate its ends, an internal annular groove cooperating with the outer wall of said cylinder block to define an annular discharge port which communicates with all of said valve units to provide a common outlet therefor, and wherein said cylinder block has an internal annular recess cooperating with said shaft to define an annular inlet port communicating with the inlet ports of said valve units to provide a common inlet therefor.

11. A pump as defined in claim 10, wherein said shaft is tubular, providing therein an inlet passage, is connected to said casing inlet at one end, and has a radial

port for fluid flow from said inlet passage to said annular inlet port.

12. An axial piston type automatically feathering variable output fluid pump comprising a drive shaft; a casing having therein a plurality of cylinders arranged in axially opposed groups comprising pairs of axially opposed cylinders, in circumferentially spaced annular array, coaxial with and surrounding said shaft; pistons in said cylinders arranged in groups corresponding to the cylinder groups; a pair of wobble cams encircling and secured to said shaft and disposed at opposite ends of said cylinders, said cams having annular piston-operating faces disposed in planes that are inclined in opposite directions with reference to the axis of said shaft, with fixed angles of inclination, and with their respective high points disposed in axially opposed alignment; the respective pistons having at their outer ends, means engageable against said cam faces with only intermittent contact with the high spots thereof when the pump is operating at less than full capacity; said casing having an inlet and a delivery outlet; a plurality of valve units each interposed between adjacent ends of the cylinders of a respective pair and embodying an inlet valve disposed for controlling flow from said inlet to respective cylinders and an outlet valve disposed for controlling flow from respective cylinders to said outlet, said valves being spring biased toward their closed positions; means for delivering to said inlet a variable flow of fluid which is operable to successively move apart the pistons of succeeding pairs of cylinders, with the stroke of the pistons varying directly in response to the rate of delivery of the fluid to said inlet; said shaft being journaled in said casing for rotating said cams, whereby to cause each pair of pistons to be successively returned into its cylinders to effect a high pressure pumping action wherein the fluid is forced through the respective outlet valve to said casing outlet.

13. A pump as defined in claim 12, including anti-friction bearings interposed between said cams and the respective ends of the casing, said bearings having outer races mounted in the casing, inner races in which the shaft is mounted, and rolling bearing elements interposed between said races; said shaft being tubular to provide a fluid inlet passage, having one end communicating with said casing inlet, and having at its respective ends, fluid ducts extending from said inlet passage into communication with said anti-friction bearings and said cams, for lubricating the same.

14. In a two-stage hydraulic pumping system: an axial piston type automatically feathering, variable output, high pressure fluid pump comprising a casing, a drive shaft journaled therein for rotation; said casing having therein a plurality of cylinders arranged in axially opposed groups comprising pairs of axially opposed cylinders in circumferentially spaced annular array, coaxial with and surrounding said shaft; pistons in said cylinders arranged in groups corresponding to the cylinder groups; a pair of wobble cams encircling and secured to said shaft and disposed at opposite ends of said cylinders, said cams having annular piston-operating faces disposed in planes that are inclined in opposite directions with reference to the axis of said shaft, with fixed angles of inclination, and with their respective high points disposed in axially opposed alignment; the respective pistons having at their outer ends means engageable against said cam faces for actuation of the pistons as the cams rotate; said casing having an inlet and a delivery outlet for hydraulic fluid; a plurality of valve units each interposed between adjacent ends of the cylinders of a respective pair and embodying an inlet valve disposed for controlling flow from said inlet to respective cylinders and an outlet valve disposed for controlling flow from respective cylinders to said outlet, said valves being spring biased toward their closed positions; means for delivering to said inlet a variable flow of fluid which is operable to successively move apart the pistons of succeeding pair of cylinders, with the stroke

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of the pistons varying directly in response to the rate of delivery of the fluid to said inlet; means for effecting relative rotation between said cylinders and cams, whereby to cause each pair of pistons to be successively returned into its cylinders to effect a high pressure pumping action wherein the fluid is forced through the respective outlet valve to said casing outlet; a pair of bearing washers of synthetic resin plastic sheet material having a resiliency and wear resistance equivalent to that of the phenolic aldehyde resins, said washers having radial bearing faces provided with face grooves extending radially from their inner to their outer margins and separating said bearing faces into radial bearing lands, and said bearing washers having their opposite radial faces mounted against the respective cams; metal thrust washers fixed in the respective ends of said casing and having radial bearing faces in bearing engagement with said bearing lands; and means providing a path for flow of hydraulic fluid to the inner extremities of said grooves, said lands, as they rotate against said thrust washers, being slightly compressed by the lubricant entering between their leading edges and the thrust washer bearing faces, whereby there is formed a plurality of wedge-shaped spaces of minute depth, loaded with hydraulic fluid which supports the end thrust loads developed between said washers by pump operation.

15. An axial piston type fluid pump comprising: a drive shaft; a casing having therein a plurality of cylinders arranged in axially opposed groups comprising pairs of axially opposed cylinders in circumferentially spaced annular array, coaxial with and surrounding said shaft; pistons in said cylinders arranged in groups corresponding to the cylinder groups; a pair of wobble cams surrounding and secured to said shaft and disposed at opposite ends of said cylinders, said cams having annular piston-operating faces disposed in planes that are inclined in opposite directions with reference to the axis of said shaft, with their respective high points disposed in axially opposed alignment; the respective pistons having at their outer ends means cooperating with said cam faces for effecting reciprocation of the pistons; said casing having an inlet and a delivery outlet for hydraulic fluid; valve means interposed between the respective groups of cylinders and embodying an inlet valve disposed for controlling flow from said inlet to respective cylinders and an outlet valve disposed for controlling flow from respective cylinders to said outlet; a pair of bearing washers of synthetic resin plastic sheet material having a resiliency and wear resistance equivalent to that of the phenolic aldehyde resins, said washers having radial bearing faces provided with face grooves extending radially from their inner to their outer margins and separating said bearing faces into radial bearing lands, and said bearing washers having their opposite radial faces mounted against the respective cams; metal thrust washers fixed in the respective ends of said casing and having radial bearing faces in bearing engagement with said bearing lands; and means providing a path for flow of hydraulic fluid to the inner extremities of said grooves, said lands, as they rotate against said thrust washers, being slightly compressed by the lubricant entering between their leading edges and the thrust washer bearing faces, whereby there is formed a plurality of wedge-shaped spaces of minute depth, loaded with hydraulic fluid which supports the end thrust loads developed between said washers by pump operation.

16. An axial piston type fluid pump comprising: a drive shaft; a casing having therein a plurality of cylinders arranged in axially opposed groups comprising pairs of axially opposed cylinders in circumferentially spaced annular array, coaxial with and surrounding said shaft; pistons in said cylinders arranged in groups corresponding to the cylinder groups; a pair of wobble cams surrounding and secured to said shaft and disposed at opposite ends of said cylinders, said cams having annular piston-operating faces disposed in planes that are inclined

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in opposite directions with reference to the axis of said shaft, with their respective high points disposed in axially opposed alignment; the respective pistons having at their outer ends means cooperating with said cam faces for effecting reciprocation of the pistons; said casing having an inlet and a delivery outlet for hydraulic fluid, valve means interposed between the respective groups of cylinders and embodying an inlet valve disposed for controlling flow from said inlet to respective cylinders and an outlet valve disposed for controlling flow from respective cylinders to said outlet; said pistons being fitted to the cylinders so as to have minimum clearance therein at the maximum temperature of the range of temperatures within which the pump is to operate, and so as to have maximum clearance in the cylinders at the minimum temperature of said range, said cylinders being of a material having a lower coefficient of expansion than the material of said piston, whereby, with an increase in temperature, the pistons will expand at a faster rate than the cylinders to compensate for the decrease in viscosity of the working fluid resulting from such temperature increase, said coefficients of expansion being so proportioned that the change in clearance will bear approximately a relation to the change in viscosity such that the fluid sealing efficiency of the piston in the cylinder, as translated into terms of volumetric efficiency of the pump in high pressure operation, will be maintained approximately a constant factor.

17. Control mechanism for a two-stage hydraulic pumping system, utilizing a constant output, low pressure supercharger, a variable output, automatically feathering high pressure pump, a connecting fluid line for delivery of fluid to the inlet of said high pressure pump, and means defining fluid passages for delivering the output of said high pressure pump to a load and for returning spent fluid from the load to the inlet of said supercharger, said control mechanism comprising: a combined volume control and bypass valve unit including a casing having a bore, an end member closing one end of said bore and an axially adjustable abutment member closing the other end of said bore, said casing further having an inlet and a delivery outlet both spaced from said end member, said delivery outlet communicating with said connecting line, and a bypass outlet spaced from said inlet in the direction away from said end member; a volume control valve comprising a control piston, a bypass piston and a stem connecting said pistons in axially spaced relation, said valve being slidable in said bore with said control piston disposed between said end member and said inlet and delivery outlet and with said bypass piston disposed between said bypass outlet and said adjustable abutment member; a spring interposed between said bypass piston and said adjustable abutment member and loading said valve; one end of said control piston cooperating with said end member to define in said bore a control chamber, in which fluid pressure may act against said control piston to balance the load of said spring and to shift the valve in response to fluid pressure changes in said control chamber, the other end of said control piston being disposed for throttling said delivery outlet in response to pressure increase in said control chamber; an inlet receiving fluid from the outlet of said supercharger, said volume control valve functioning for metering the flow from said inlet to said connecting line and bypassing the excess to said bypass outlet; said casing further having a bypass leading from said inlet to said bypass outlet, a pressure responsive bypass valve in said casing, normally closing said bypass and adapted to open so as to unload the supercharger by bypassing back toward the supercharger inlet, that portion of the supercharger output which is in excess of the amount accepted by said volume control valve; and a pilot valve responsive to fluctuations in the output pressure of said high pressure pump, for supplying to said volume control valve, pressure fluid derived from said high pressure pump output, for actuating said volume control valve in a manner to throttle the flow from

the supercharger to the high pressure pump so as to furnish to the latter only that quantity of working fluid which is required by said high pressure pump to meet the load demand.

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