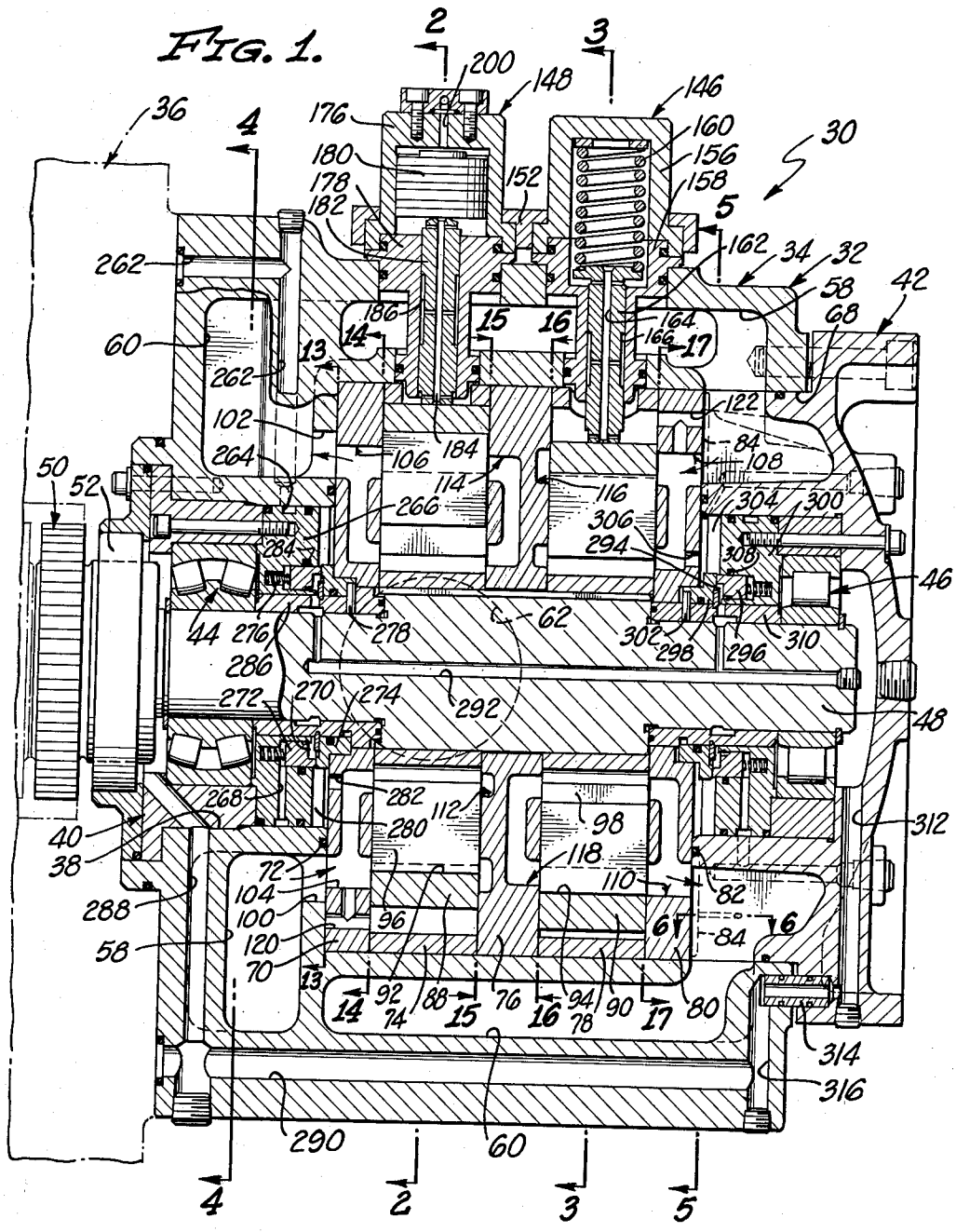
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VANE-TYPE VARIABLE DELIVERY PUMP

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FIG. 1.




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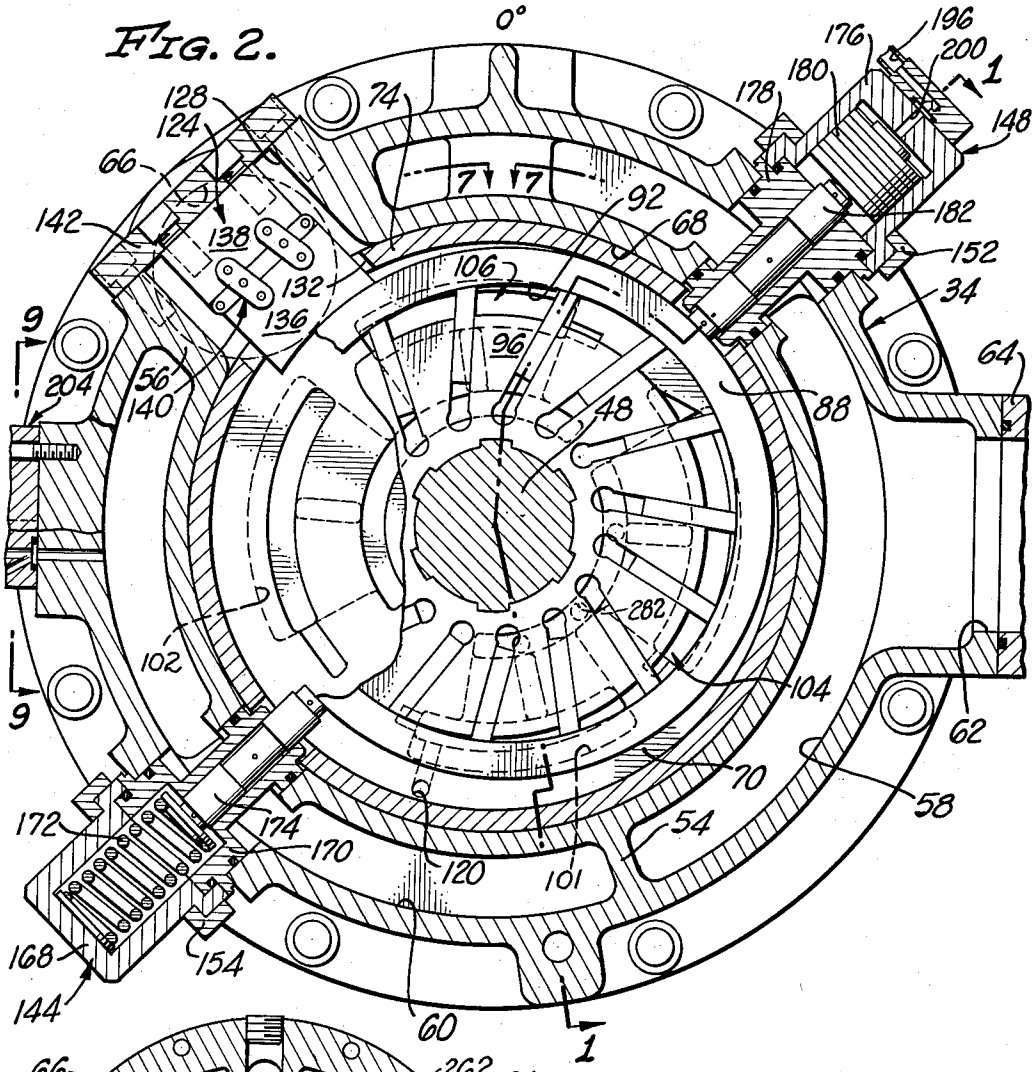
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FIG. 2.



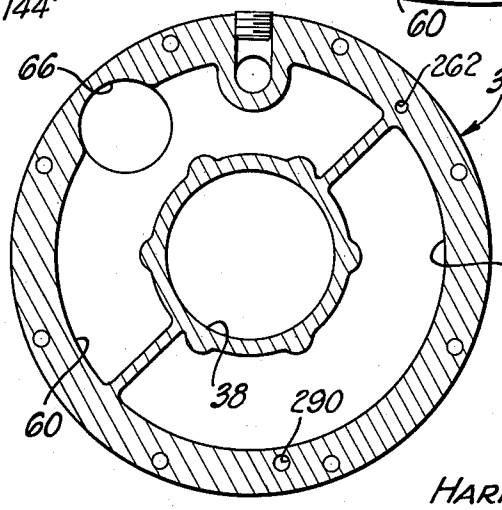




FIG. 4.

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FIG. 3.

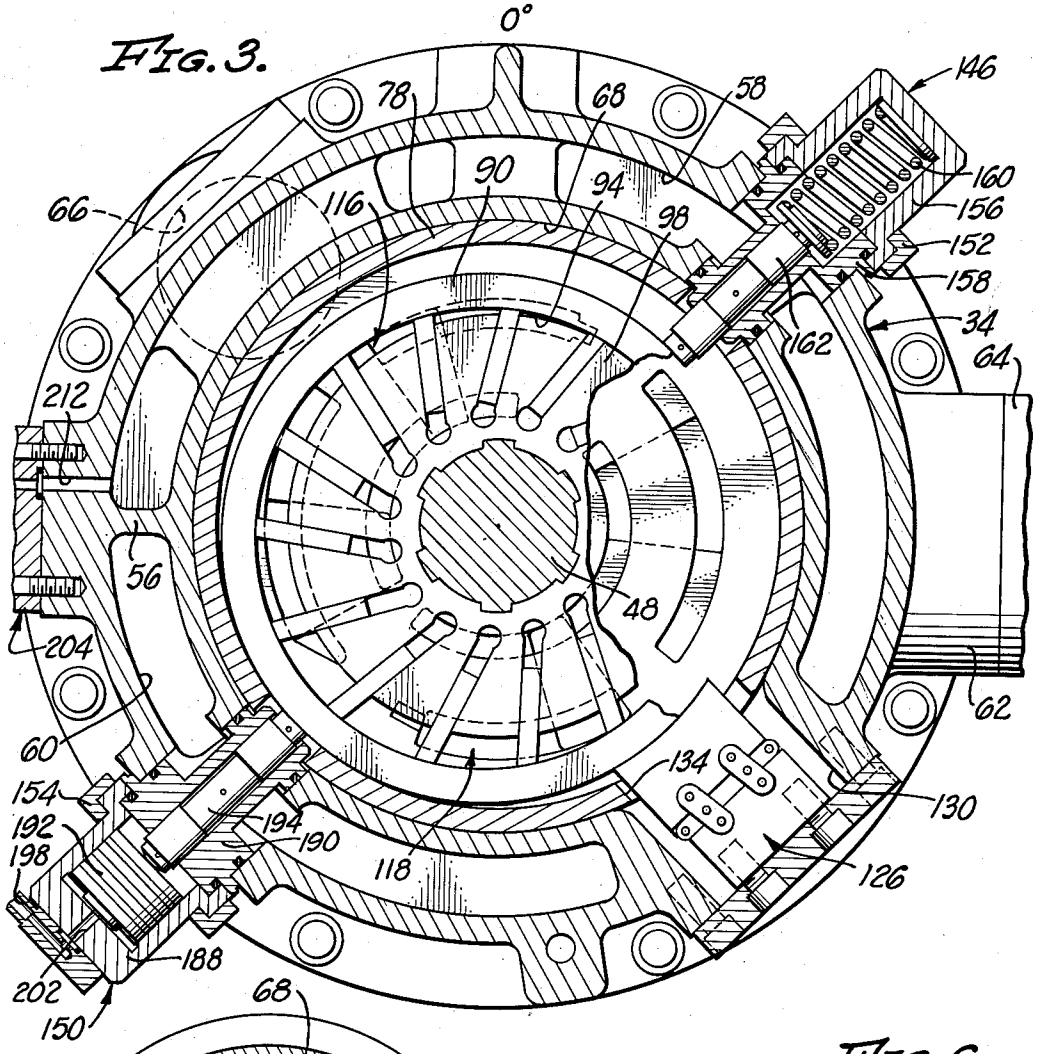



FIG. 5.

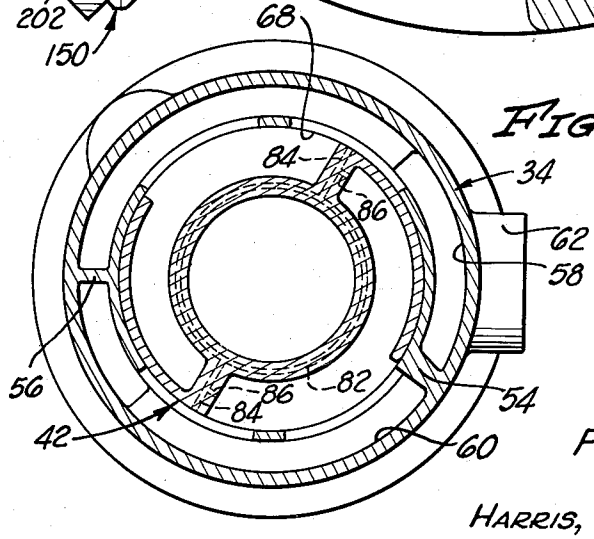
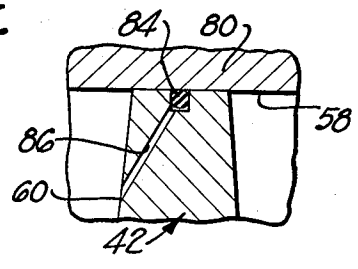



FIG. 6.




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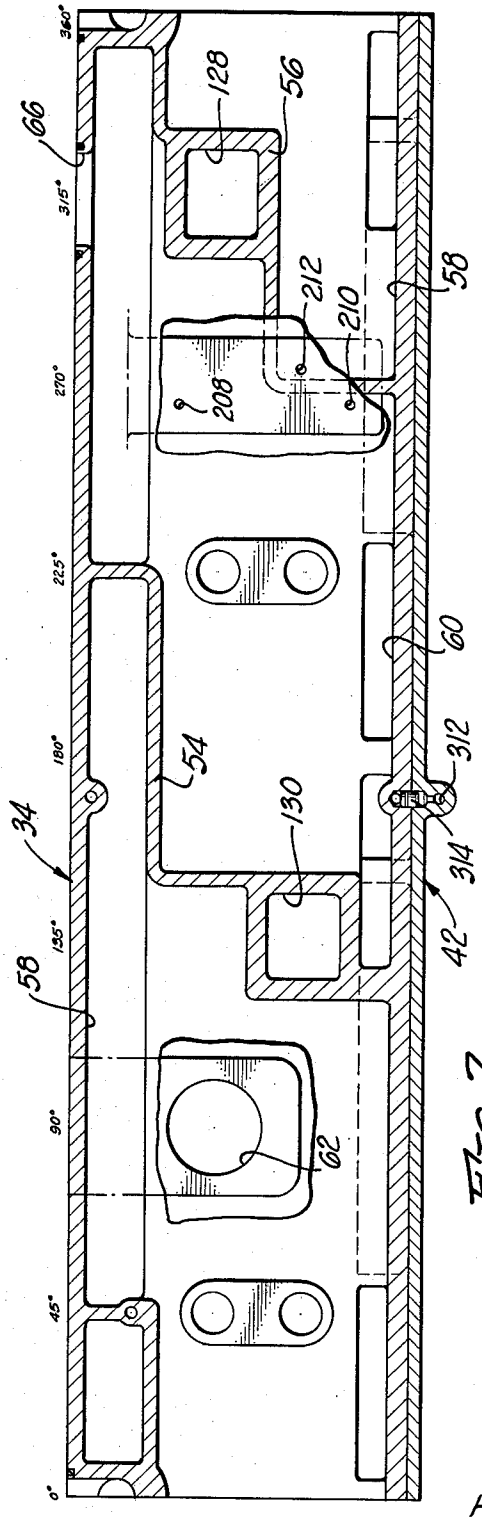
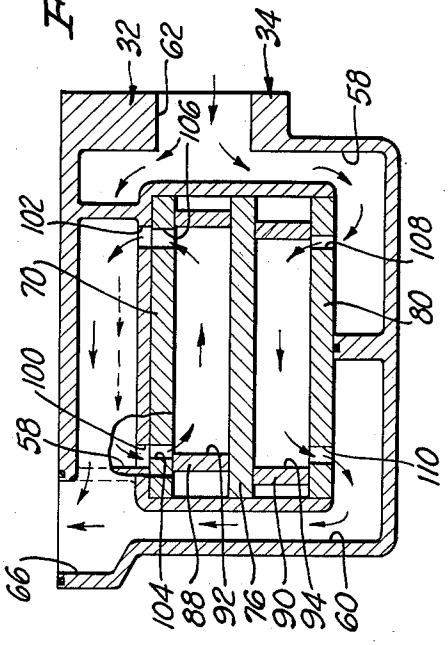



FIG. 7.

FIG. 8.




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FIG. 10.

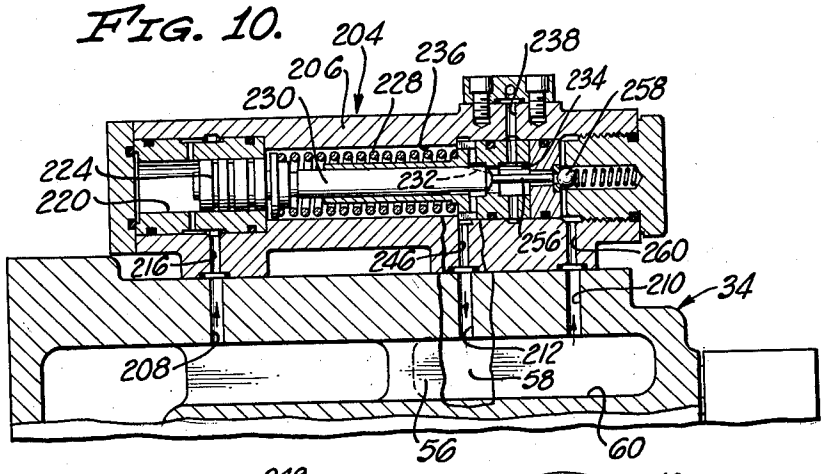



FIG. 9.

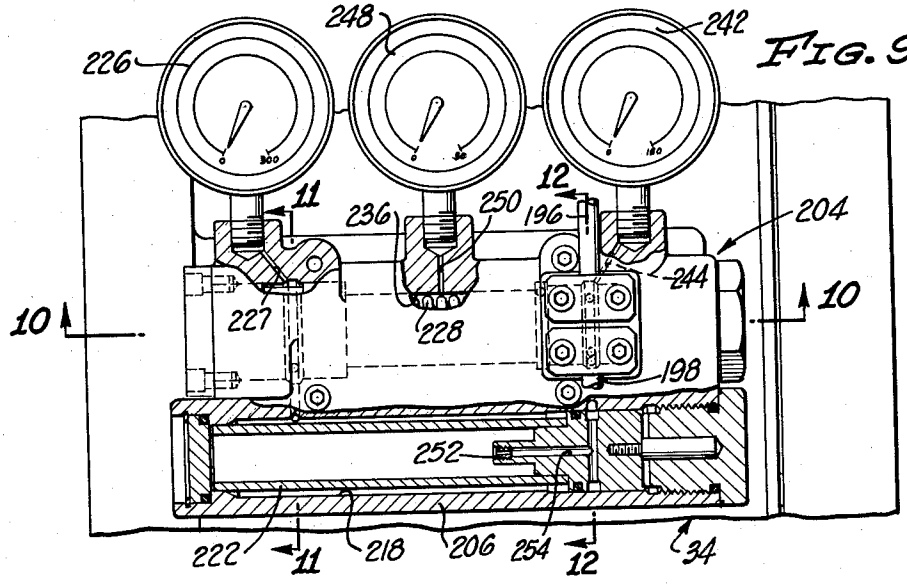



FIG. 11.

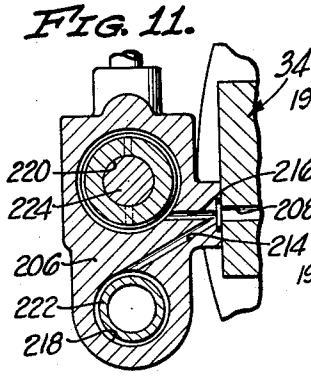
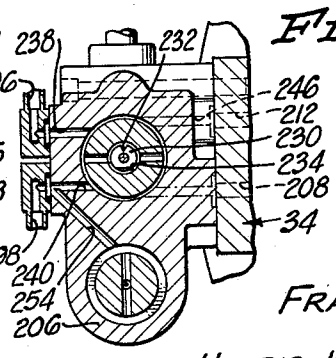



FIG. 12.

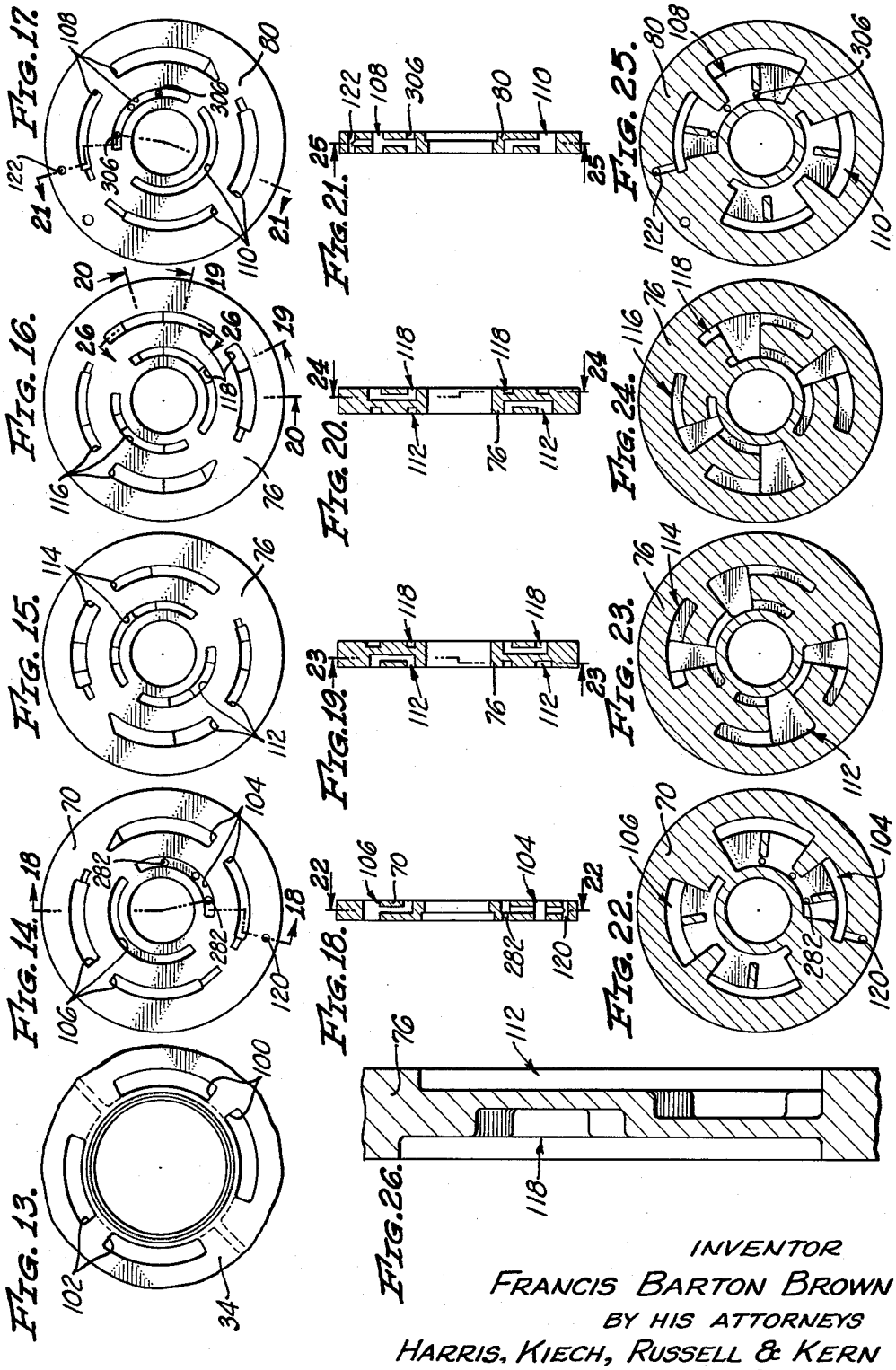



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VANE-TYPE VARIABLE DELIVERY PUMP

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United States Patent Office 3,137,235 Patented June 16, 1964



3,137,235

# VANE-TYPE VARIABLE DELIVERY PUMP

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Filed Sept. 26, 1960, Ser. No. 58,251
12 Claims. (Cl. 103—4)

The present invention relates in general to rotary motors or pumps and, more particularly, to variable volume rotary motors or pumps of the sliding vane type. Still more specifically, the invention is of particular utility as a variable delivery booster or charge pump for the high speed triplex pump disclosed in copending patent application Serial No. 5,840, filed February 1, 1960 by Clarence J. Coberly, Carter P. Williams and me, now Patent No. 3,077,836, granted February 19, 1963, and will be considered in such connection herein as a matter of convenience.

Triplex pumps are widely used in the oil industry to provide power oil, which is usually reasonably clean crude oil, to fluid operated well pumps. Such triplex pumps must be capable of meeting varying demands for power oil, must be capable of continuous operation for prolonged periods of time without shutdowns, must have lubricating systems which are not readily contaminated by the crude oil being pumped, must be capable of being serviced and/or repaired easily when necessary, and the like. A primary object of the present invention is to provide a compact variable delivery booster or charge pump which meets the foregoing requirements of triplex pumps and which may be mounted directly on a triplex pump and coupled directly to the crankshaft of the triplex pump.

In general, the invention contemplates a sliding vane pump the eccentricity of which is variable to vary the delivery of the pump as required to meet a varying demand on the part of the triplex. More particularly, the invention contemplates a pump having a sliding vane rotor disposed in a rotor chamber the eccentricity of which may be varied to vary the delivery of the pump by moving laterally a ring which defines the peripheral wall of the rotor chamber.

A primary object of the invention is to provide a variable delivery, sliding vane pump of the foregoing nature including two rotor chambers of variable eccentricity respectively having therein sliding vane rotors mounted on a common rotor shaft, the inlets and outlets of the rotor chambers being 180° out of phase so that lateral loads applied to the rotor shaft as the result of the application of lateral pressure forces to the two rotors balance out. This construction minimizes lateral or side loads on the rotor shaft bearings and thus permits the use of relatively small bearings, which is an important feature of the invention.

Another and important object of the invention is to provide a dual rotor pump of the foregoing character wherein the respective inlets and outlets are formed in two end plates defining outer end walls of the respective rotor chambers, and wherein dummy inlets and outlets are provided in a center plate separating and forming inner end walls of the respective rotor chambers. With this construction, axial pressure forces applied to the rotors are balanced out, which is an important feature of the invention.

Another object is to provide means for connecting the 180° out-of-phase inlets in parallel and the 180° out-of-phase outlets in parallel comprising a case which encircles the rotor chambers and which includes radially spaced inner and outer walls and partitions dividing the annular space between such inner and outer walls into



inlet and outlet chambers respectively communicating with the rotor-chamber inlets and outlets, the case having main inlet and outlet ports in communication with the inlet and outlet chambers.

Another object of the invention is to provide means for controlling the eccentricities of the rotor chamber rings comprising spring means respectively biasing the rings toward positions of maximum eccentricity and fluid operated control means respectively biasing the rings toward positions of minimum eccentricity.

Another object of the invention is to provide pilot valve means actuable by the pressure in the outlets of the rotor chambers for applying to the fluid operated control means a control pressure which is a function of the pressure in the rotor chamber outlets and which controls the output pressure of the pump.

A further object of the invention is to provide means for lubricating the rotor shaft bearings with a continuous flow of lubricating oil.

Still another object of the invention is to provide means for producing favorable pressure gradients on seals for the rotor shaft bearings so that the lubricating oil employed to lubricate the rotor shaft bearings tends to leak past the seals into the crude oil being pumped, rather than vice versa. A related object is to provide means for preventing flow of crude oil past the seals for the rotor shaft bearings into the lubricating oil system even when the pump is not operating.

Yet another object of the invention is to provide a pump having a construction such that its various parts subject to wear can be removed and replaced readily in the field.

The foregoing objects, advantages, features and results of the present invention, together with various other objects, advantages, features and results thereof which will be evident to those skilled in the art to which the invention relates in the light of this disclosure, may be achieved with the exemplary embodiment of the invention described in detail hereinafter and illustrated in the accompanying drawings, wherein:

FIG. 1 is a longitudinal sectional view of a variable delivery vane pump of the invention mounted on and directly coupled to the triplex pump of the aforementioned copending application, FIG. 1 being taken along the irregular arrowed line 1—1 of FIG. 2;

FIGS. 2, 3, 4 and 5 are transverse sectional views respectively taken along the arrowed lines 2—2, 3—3, 4—4 and 5—5 of FIG. 1, FIGS. 4 and 5 being on reduced scales;

FIG. 6 is an enlarged, fragmentary sectional view taken along the arrowed line 6—6 of FIG. 1;

FIG. 7 is a semidiagrammatic, developed sectional view of a case of the pump of the invention and is taken as indicated by the arrowed line 7—7 of FIG. 2, some parts of the case being shown in external elevation;

FIG. 8 is a semidiagrammatic sectional view illustrating the flow of pumped fluid through the rotary pump of the invention;

FIG. 9 is a sectional view of a pilot valve means of the pump of the invention and is taken along the arrowed line 9—9 of FIG. 2;

FIGS. 10, 11 and 12 are fragmentary sectional views respectively taken along the arrowed lines 10—10, 11—11 and 12—12 of FIG. 9;

FIG. 13 is a fragmentary elevational view on a reduced scale showing porting of one end of the case of the pump of the invention and is taken along the arrowed line 13—13 of FIG. 1;

FIGS. 14, 15, 16 and 17 are elevational views on a reduced scale showing porting of the rotor chambers of the pump of the invention and are taken along the ar-

rowed lines 14—14, 15—15, 16—16, and 17—17, respectively, of FIG. 1;

FIGS. 18, 19, 20 and 21 are longitudinal sectional views respectively taken along the irregular arrowed lines 18—18, 19—19, 20—20 and 21—21 of FIGS. 14, 16 and 17;

FIGS. 22, 23, 24 and 25 are transverse sectional views respectively taken along the arrowed lines 22—22, 23—23, 24—24 and 25—25 of FIGS. 18, 19, 20 and 21, respectively; and

FIG. 26 is an enlarged, fragmentary longitudinal sectional view taken along the irregular arrowed line 26—26 of FIG. 16.

Referring initially to FIG. 1 of the drawings, the rotary pump of the invention is designated generally by the numeral 30 and includes a housing 32 comprising a case 34 adapted to be bolted, or otherwise secured, to the housing of a triplex pump 36. The inner end of the case 34, i.e., the end thereof adjacent the triplex 36, is provided with an axial opening 38 which is closed by an end closure or closure assembly 40 forming part of the housing 32 and bolted, or otherwise secured, to the case 34. The outer end of the case 34 is open and is closed by an end closure or closure assembly 42 also forming part of the housing 32 and bolted, or otherwise secured, to the case 34.

The inner and outer end closures 40 and 42 respectively carry inner and outer bearings 44 and 46 for a rotor shaft 48. The inner end of the rotor shaft 48 is coupled directly to the crankshaft, not shown, of the triplex 36 by coupling means 50, as more fully disclosed in the aforementioned copending application, whereby the pump 30 is driven at the same speed as the triplex. An annular shaft seal 52 is carried by the end closure 40 between the inner bearing 44 and the coupling means 50. The inner bearing 44 is preferably a thrust bearing, shown as a roller-type thrust bearing, while the outer bearing 46 is a simple journal bearing, preferably of the roller type, for reasons which will be discussed hereinafter.

As shown in FIGS. 1 to 3, 5, 7 and 8 of the drawings, the case 34 includes radially spaced inner and outer walls forming therebetween an annular space which, as best shown in FIG. 7 of the drawings, is divided by partitions 54 and 56 into irregular inlet and outlet chambers 58 and 60. These inlet and outlet chambers have varying angular extents in the direction of the axis of the pump 30 and extend radially inwardly into both the inner end wall of the case 34 and into the outer end closure 42, as best shown in FIGS. 1, 4 and 5 of the drawings. The case 34 is provided intermediate its ends with a lateral inlet port 62, best shown in FIG. 2 of the drawings, which communicates with the inlet chamber 58 and to which an inlet line 64 may be connected by bolting it, or otherwise securing it, to the case. The inlet line 64 leads to a suitable source of supply of the crude oil, or other fluid, to be pumped. As shown in FIGS. 2, 3, 4 and 7, the inner end wall of the case 34 is provided therein with an outlet port 66 which faces the triplex 36 and which communicates with the inlet thereof. Thus, in the environment illustrated, the pump 30 serves as a booster or charge pump for delivering crude oil, or other fluid, to the inlet of the triplex 36 at an elevated pressure, which may be of the order of 250 p.s.i.

The open outer end of the case 34 is formed by the outer end of a main bore 68 in the case which is symmetrical about the axis of the rotor shaft 48, this axis sometimes being referred to herein as the rotor axis. Going from the inner end of the main bore 68 toward the outer end thereof, i.e., going from the left end of the main bore toward the right end thereof as viewed in FIG. 1 of the drawings, the main bore contains an end plate 70 seated against the inner end thereof and sealed relative to the inner end wall of the case 34 around the opening 38 therein by an O-ring 72, an annular



spacer 74 seated against the end plate 70, a center plate 76 seated against the annular spacer 74, an annular spacer 78 seated against the center plate, and an end plate 80 seated against the annular spacer 78. The assembly of internal components comprising the end plate 70, the annular spacer 74, the center plate 76, the annular spacer 78 and the end plate 80 is retained within the main bore 68 by the end closure 42. The latter is seated against the end plate 80 and is sealed relative thereto by a sealing element 82 having radial portions 84, FIG. 5, which separate from each other the portions of the inlet and outlet chambers 58 and 60 that are formed within the end closure 42. As shown in FIGS. 5 and 6, passages 86 within the portions of the end closure 42 which separate the corresponding portions of the inlet and outlet chambers 58 and 60 carry the pressure in the outlet chamber 60 behind the radial portions 84 of the sealing element 82 to urge same into fluid tight engagement with the end plate 80.

As shown in FIGS. 1 and 2 of the drawings, a rotor chamber ring 88 is located between the end plate 70 and the center plate 76 and within the annular spacer 74. Similarly, a rotor chamber ring 90 is located within the annular spacer 78 and between the center plate 76 and the end plate 80. The ring 88 cooperates with the end and center plates 70 and 76 to form a rotor chamber 92, and the ring 90 cooperates with the center and end plates 76 and 80 to form a rotor chamber 94. As will be apparent from FIGS. 1, 2 and 3 of the drawings, the axes of the rotor chambers 92 and 94 are parallel to the axis of the rotor shaft 48, but are offset laterally therefrom in opposite directions. In other words, the rotor chambers 92 and 94 are oppositely eccentric relative to the rotor axis. The rings 88 and 90 are adapted to be moved laterally, in a manner to be described, in directions such as to move the axes of the rotor chambers 92 and 94 toward and away from the rotor axis to vary the eccentricities of the rotor chambers between minimum and maximum values, it being understood that the minimum eccentricity of the rotor chambers may be zero.

Rotors 96 and 98 of the generally-radially-slidable vane type are disposed within the respective rotor chambers 92 and 94 and are splined, or otherwise keyed, to the rotor shaft 48 so as to rotate therewith about the rotor axis, the direction of rotation being counterclockwise as viewed in FIGS. 2 and 3 of the drawings. The rotors 96 and 98 are of more or less conventional construction so that a detailed description is not necessary.

As best shown in FIGS. 1, 2 and 13 of the drawings, the inner end of the main bore 68 in the case 34 is provided with two diametrically opposed groups of ports respectively forming an inlet 100 and an outlet 102, the former communicating with the inlet chamber 58 and the latter with the outlet chamber 60. As shown in FIGS. 1, 2, 14, 18 and 22 of the drawings, the end plate 70 is provided therethrough with two diametrically opposed groups of ports respectively forming an inlet 104 and an outlet 106 respectively connecting the inlet 100 and the outlet 102 to the rotor chamber 92 on opposite sides of the rotor axis.

Similarly, the end plate 80 is provided therethrough with two diametrically opposed groups of ports respectively forming an inlet 108 and an outlet 110, as best shown in FIGS. 17, 21 and 25 of the drawings. The inlet 108 and the outlet 110 respectively communicate with the portions of the inlet and outlet chambers 58 and 60 which are formed in the end closure 42, and respectively communicate with the rotor chamber 94 on opposite sides of the rotor axis.

An important feature of the present invention resides in the fact that the group of ports forming the inlet 104 in the end plate 70 and the group of ports forming the inlet 108 in the end plate 80 are diametrically opposite each other, i.e., are 180° out of phase. Similarly, the group of ports forming the outlet 106 in the end plate

70 and the group of ports forming the outlet 110 in the end plate 80 are diametrically opposite each other, i.e., are 180° out of phase. This will be apparent from a comparison of FIGS. 14 and 17, keeping in mind that these figures are views taken in opposite axial directions. As the result of the foregoing 180° out-of-phase relation between the inlets 104 and 108 and between the outlets 106 and 110, the outlet or discharge pressure is developed on diametrically opposite sides of the rotors 96 and 98. Consequently, the lateral loads resulting from the application of the discharge pressure laterally to the rotors 96 and 98 act in opposite directions so that no net lateral load is applied to the rotor shaft 48, although a small couple is applied thereto because of the axial spacing of the two rotors. Thus, the rotor shaft bearing loads are minimized and the rotor shaft bearings 44 and 46 may be relatively small, as shown, which is an important feature of the invention.

The various groups of ports forming the inlets 104 and 108 and the outlets 106 and 110 are designed to provide communication with the rotor chambers 92 and 94 both adjacent the roots and the tips of the sliding vanes of the rotors 96 and 98, are designed to minimize hydraulic shock as communication with successive intervane spaces is initiated and terminated, and the like. Such details of the inlets 104 and 108 and the outlets 106 and 110 per se form no part of the present invention so that a further description thereof is unnecessary.

The center plate 76 is provided on the side thereof facing the end plate 70 with two groups of dummy ports respectively forming a dummy inlet 112 and a dummy outlet 114 respectively facing, axially opposite and identical to the inlet 104 and the outlet 106 in the end plate 70. Similarly, the side of the center plate 76 which faces the end plate 80 is provided therein with two groups of ports respectively forming a dummy inlet 116 and a dummy outlet 118 respectively facing, axially opposite and identical to the inlet 108 and the outlet 110 in the end plate 80. The foregoing will be apparent from a comparison of FIGS. 14 and 15, keeping in mind that these are views taken in opposite directions, and from a comparison of FIGS. 16 and 17, again keeping in mind that these are views taken in opposite directions.

Without the dummy inlets 112 and 116 and the dummy outlets 114 and 118, axial pressure forces tending to cock the rotors 96 and 98 would be produced. For example, the difference between the inlet pressure in the inlet 104 and the discharge pressure in the outlet 106 would produce a pressure force couple tending to cock the rotor 96 in the clockwise direction, as viewed in FIG. 1. The provision of the dummy inlets 112 and 116 and the dummy outlets 114 and 118 results in counterbalancing pressure forces which eliminate any net pressure force couples tending to cock the rotors 96 and 98. Also, the entire assembly of the rotor shaft 48 and the rotors 96 and 98 is subjected to no net axial pressure force as the result of the provision of the dummy inlets 112 and 116 and the dummy outlets 114 and 118. Consequently, friction and wear are minimized, which is an important feature.

The end plates 70 and 80 are respectively provided therein with ports 120 and 122 which connect the inlet chamber 58 to the normally eccentric clearances between the ring 88 and the spacer 74 and between the ring 90 and the spacer 78. Thus, the entire external peripheries of the rings 88 and 90 are exposed to the inlet pressure.

Referring to FIGS. 2 and 3 of the drawings, the rings 88 and 90 are laterally shiftable to vary the eccentricities of the rotor chambers 92 and 94 along a path which is shown as extending in the 45°–225° direction, regarding the tops of FIGS. 2 and 3 as the 0° position. The inlet 104 and the outlet 106 lie on opposite sides of this ring path, the same being true of the inlet 108 and the outlet 110. Consequently, the discharge pressure in the outlets 106 and 110 tends to displace the rings 88 and 90 in directions perpendicular to the ring path, i.e., along a 135°–315° line. More particularly, the location of the outlet 106 is such that the discharge pressure tends to shift the ring 88 toward the 315° position, i.e., toward the upper left as viewed in FIG. 2, and the location of the outlet 110 is such that the discharge pressure tends to shift the ring 90 toward the 135° position, i.e., toward the lower right as viewed in FIG. 3.

To prevent lateral movement of the rings 88 and 90 along the 135°–315° line, while permitting lateral shifting of the rings along the 45°–225° ring path, thrust bearing means 124 and 126 carried by the case 34 respectively engage the rings 88 and 90 at the 315° and the 135° positions. The thrust bearing means 124 and 126 are respectively disposed in radial wells 128 and 130 in the periphery of the case 34 and respectively extend through openings 132 and 134 in the annular spacers 74 and 78.

Referring particularly to FIG. 2 of the drawings for a more detailed consideration of the thrust bearing means 124, it includes inner and outer thrust blocks 136 and 138 separated by a needle bearing assembly 140. The inner end of the inner thrust block 136 is concave to fit the external periphery of the ring 88, and the outer thrust block 138 is bolted, or otherwise secured, to a closure 142 for the corresponding well 128, this closure being bolted, or otherwise secured, to the case 34. With this construction, the thrust bearing means 124 may readily be removed when disassembly of the pump 30 is required for replacement of such components as the end and center plates 70, 80 and 76, the rings 88 and 90, and the rotors 96 and 98.

The thrust bearing means 126 for the ring 90 is structurally identical to the thrust bearing means 124 for the ring 88. Consequently, a detailed description is unnecessary.

As best shown in FIGS. 2 and 3 of the drawings, the rings 88 and 90 are respectively biased along the 45°–225° ring path toward positions of maximum eccentricity by spring means 144 and 146, and are respectively biased toward positions of minimum eccentricity by fluid operated control means 148 and 150. As best shown in FIG. 1 of the drawings, the spring means 146 and the control means 148 are axially aligned and are secured to the case 34 by a common clamp 152 which, in turn, is bolted, or otherwise secured, to the case. Thus, the spring means 146 and the control means 148 may be removed readily when disassembly of the pump 30 is required. Similarly, the spring means 144 and the control means 150 are axially aligned and are secured to the case 34 by a common clamp 154.

Continuing to refer to FIG. 1 of the drawings, the spring means 146 includes a housing 156 in the form of a flanged cup secured in place by the clamp 152, and includes a plunger guide 158 which extends through and is disposed in suitable radial openings in the outer and inner walls of the case 34 and the annular spacer 78, the plunger guide 158 being sealed relative to the outer and inner walls of the case by O-rings, or the like. The housing 156 is seated on the plunger guide 158 and retains it in position when the housing is secured in place by the clamp 152. Within the housing 156 is a compression coil spring 160 which is seated at one end against the outer end of such housing and which is seated at its other end against the outer end of a plunger 162 radially reciprocable in the plunger guide 158. The inner end of the plunger 162 engages the outer periphery of the ring 90 so that the spring 160 biases this ring toward its maximum-eccentricity position. The plunger 162 is provided axially therethrough with a radial passage 164 communicating at its inner end with the clearance around the ring 90 and communicating at its outer end with the interior of the housing 156. Thus, both ends of the plunger 162 are exposed to the inlet pressure to hydraulically balance the plunger. The plunger 162 is provided intermediate

its ends with an external annular groove 166 which communicates with the passage 164 to convey the crude oil at the inlet pressure to the interface between the plunger 162 and its guide 158 for lubricating purposes.

As shown in FIG. 2, the spring means 144 acting on the ring 88 is identical to the spring means 146 acting on the ring 90 so that a detailed descripton is not required. Briefly, the spring means 144 includes a housing 168, a plunger guide 170, a compression coil spring 172 and a plunger 174 respectively corresponding to the housing 156, the plunger guide 158, the spring 160 and the plunger 162 of the spring means 146.

Returning to FIG. 1 of the drawings, the fluid operated control means 148 includes a radial cylinder 176 in the form of a flanged cup held in place by the clamp 152. The cylinder 176 is seated on and retains a plunger guide 178 which extends through and is disposed in radial openings in the outer and inner walls of the case 34 and the annular spacer 74, the plunger guide being suitably sealed with respect to the outer and inner walls of the case. Within the cylinder 176 is a control piston 180 which is seated against the outer end of a plunger 182 radially reciprocable in the plunger guide 178, the inner end of the plunger 182 being seated against the outer periphery of the corresponding ring 88. The plunger 182 is provided axially therethrough with a radial passage 184 which communicates the inlet pressure in the clearance around the ring 88 to the inner end of the control piston 180, the plunger 182 also being provided therein with an external lubricating groove 186 which communicates with the passage 184.

The fluid operated control means 150 for the ring 90 is identical to the fluid operated control means 148 for the ring 88 so that a full description is not necessary. Briefly, and referring to FIG. 3 of the drawings, the fluid operated control means 150 includes a cylinder 188, a plunger guide 190, a control piston 192 and a plunger 194 respectively corresponding to the cylinder 176, the plunger guide 178, the control piston 180 and the plunger 182 of the control means 148.

A control pressure is applied to the outer ends of the control pistons 180 and 192 through control lines 196 and 198 communicating with control passages 200 and 202 in the outer ends of the cylinders 176 and 188. This control pressure, which is higher than the inlet pressure acting on the inner ends of the control pistons 180 and 192, is provided by a fluid operated pilot valve means 204, FIGS. 2, 3 and 9 to 12, which will be described in detail hereinafter.

As will be apparent, the control pressure applied to the control pistons 180 and 192 tends to shift the rings 88 and 90 toward positions of minimum eccentricity in opposition to the action of the respective springs 172 and 160. The springs 160 and 172 have high spring rates so that the respective rings 90 and 88 seek symmetrical positions the eccentricities of which depend on the magnitude of the control pressure applied to the outer ends of the respective control pistons 180 and 192. Therefore, no mechanical interconnection between the control systems for the two rings 88 and 90 is necessary.

Turning now to FIGS. 9 to 12 of the drawings, the pilot valve means 204 includes a housing 206 which is bolted, or otherwise secured, to the external periphery of the case 34 in a position straddling a portion of the partition 56 between the inner and outer walls of the case. Thus, the housing 206 overlies ports 208 and 210 in the case 34 which communicate with the outlet chamber 60 and overlies a port 212 in the case which communicates with the inlet chamber 58, the locations of these ports in the case relative to the partition 56 also being shown in FIG. 7 of the drawings. As best shown in FIG. 11, the outer end of the discharge pressure port 208 communicates with passages 214 and 216 in the housing 206 and respectively leading to a filter chamber 218 and a cylinder 220 in this housing. The filter chamber 218 contains a filter 222 into the interior of which the discharge pressure fluid may flow from the port 208 and the passage 214. Within the cylinder 220 is a pilot valve piston 224 to one end of which the discharge pressure is applied by way of the port 208 and the passage 216. Thus, the discharge pressure biases the pilot valve piston 224 in one direction, i.e., toward the right as viewed in FIG. 10, the discharge pressure being registered on a pressure gauge 226 in communication with the cylinder 220 through a passage 227.

The pilot valve piston 224 is biased in the opposite direction, i.e., toward the left as viewed in FIG. 10, by a compression coil spring 228 encircling a pilot valve 230 in the form of a stem connected to the piston 224. The pilot valve 230 is provided at the end thereof farthest from the piston 224 with an axially inclined V-notch 232, FIGS. 10 and 12, which controls flow from a chamber 234, FIG. 10, into a chamber 236 surrounding the pilot valve 230 and the spring 228. As will become apparent, the pressure in the chamber 234 is the aforementioned control pressure, and the pressure in the chamber 234 is communicated to the respective control lines 196 and 198 through passages 238 and 240 in the housing 206. The control pressure in the chamber 234 is registered on a pressure gauge 242 which communicates with the chamber in question through a passage 244 in the housing 206.

The chamber 236, with which the control chamber 234 communicates through the V-notch 232 in the pilot valve 230, communicates in turn with the inlet pressure port 212, leading to the inlet chamber 58 in the case 34, through a passage 246 in the housing 206. Thus, the pressure in the chamber 236 is substantially, although slightly higher than, the inlet pressure to the pump 30, the pressure in the chamber 236 being registered on a pressure gauge 248 which is connected to the chamber 236 by a passage 250, FIG. 9, in the housing 206.

As will be apparent, the pressure in the control chamber 234 is thus partially determined by the pilot valve 230. More particularly, as the discharge pressure applied to the pilot valve piston 224 increases, the pilot valve 230 is moved toward the right, as viewed in FIG. 10, to increase the resistance provided by the V-notch 232 to flow from the control chamber 234 into the low pressure chamber 236, thereby tending to increase the pressure in the control chamber 234. The reverse occurs if the discharge pressure applied to the pilot valve piston 224 decreases.

The pressure in the control chamber 234, which is partially determined by the pilot valve 230 as hereinbefore explained, is also determined by a restrictor 252, FIG. 9, which receives filtered crude oil, or other fluid, from the interior of the filter 222 and reduces the pressure thereof to a value below the discharge pressure of the pump 30. The downstream side of the restrictor 252 is connected to the control chamber 234 by a passage means 254 which, as shown in FIG. 12, includes the passage 240 as part thereof.

Thus, it will be apparent that the control pressure developed in the control chamber 234 and applied to the control pistons 180 and 192 is determined by both the restrictor 252 and the pilot valve 230. The restrictor 252, of course, provides a constant resistance to flow of fluid at the discharge pressure into the control chamber 234, while the V-notch 232 in the pilot valve 230 provides a variable resistance to flow out of the control chamber 234 and into the chamber 236 leading to the inlet chamber 58, the resistance offered by the pilot valve 230 depending on the magnitude of the discharge pressure applied to the pilot valve piston 224. Thus, the control pressure developed in the control chamber 234 and applied to the control pistons 180 and 192 to determine the eccentricities of the rings 88 and 90, varies as a function of the discharge pressure of the pump 30 throughout a range intermediate the inlet and discharge pressures of the pump. As the control pressure increases

toward the upper end of its range, the eccentricities of the rings 88 and 90 are reduced to reduce the discharge pressure. Conversely, as the control pressure decreases toward the lower end of its range, the springs 160 and 172 increase the eccentricities of the rings 88 and 90 to increase the discharge pressure. Thus, the discharge pressure is maintained within finite limits despite variations in demand by the triplex 36.

Referring to FIG. 10 of the drawings, the stem forming the pilot valve 230 terminates in a secondary stem 256 which unseats a check valve 258 in the event that the discharge pressure applied to the pilot valve piston 224 rises sufficiently to displace the pilot valve 230 beyond the corresponding end of its normal range of travel. When this occurs, fluid at the discharge pressure may flow from the outlet chamber 60 in the case 34 directly into the control chamber 234 by way of the port 210, a passage 260 in the housing 206, and the check valve 258. Consequently, full discharge pressure is applied to the control pistons 180 and 192 to prevent the pump 30 from developing an excessive pressure.

Turning to FIG. 1 of the drawings, the manner in which the bearings 44 and 46 are lubricated and the manner in which the oil for lubricating same is prevented from becoming contaminated by the crude oil, or other fluid, being pumped will now be considered, primarily in connection with the bearing 44.

Lubricating oil is delivered to a supply passage 262 in the case 34 by a source, not shown, within the triplex 36, attention in this connection being directed to the aforementioned copending application. The supply passage communicates with an external annular groove 264 in an annular seal retainer 266 which forms part of the end closure 40 for the opening 38 in the inner end of the case 34. From the annular groove 264, a passage 268 leads radially inwardly to an inner annular chamber 270 the outer periphery of which is bounded by interengageable annular sealing surfaces of nonrotary and rotary annular sealing elements 272 and 274 constituting an annular seal means surrounding the rotor shaft 48 between the bearing 44 and the rotor chamber 92. Communication between the radial passage 268 and the annular chamber 270 is established by an annular clearance, not shown, between the seal retainer 266 and the inner periphery of the annular sealing element 272, it being noted there is no O-ring, or equivalent, in this region. The nonrotary annular sealing element 272 is carried by the seal retainer 266 and is biased into engagement with the rotary annular sealing element 274 by compression springs 276. The rotary annular sealing element 274 is suitably keyed to the rotor shaft 48 in a manner permitting axial removal of this annular sealing element, as by means of a pin 278 fitting into a notch in the rotary annular sealing element.

With the foregoing construction, the lubricating oil is present under substantial pressure, e.g., 60 p.s.i., throughout the entire region around the nonrotary annular sealing element 272 radially inwardly of the interengageable annular sealing surfaces of the annular sealing elements 272 and 274, i.e., throughout the entire inner annular chamber 270. Radially outwardly of the interengageable annular sealing surfaces of the annular sealing elements 272 and 274 is an outer annular chamber 280 which is connected to inlet pressure by ports 282, FIGS. 1 and 14, connecting the outer annular chamber 280 to the inlet 104 in the end plate 70. During operation of the pump 30 and the triplex 36, the inlet pressure to the pump 30 is considerably less than the lubricating oil pressure so that the lubricating oil pressure in the inner annular chamber 270 is considerably higher than the pressure in the outer annular chamber 280. Consequently, there is a favorable pressure gradient across the interengageable annular sealing surfaces of the annular sealing elements 272 and 274 which tends to cause lubricating oil leakage into the crude oil being pumped, thus preventing any crude oil from entering the lubricating system of the pump 30 and of the triplex 36, which is an important feature.

When the pump 30 and the triplex 36 are not running, crude oil is prevented from entering the lubricating oil system because the interengageable annular sealing surfaces of the annular sealing elements 272 and 274 are maintained in engagement by the springs 276 and because the nonrotary annular sealing element 272 is sealed relative to the seal retainer 266 by an O-ring 284 having the same sealing diameter as the outer diameter of the interengageable annular sealing surfaces on the annular sealing elements 272 and 274. Thus, as long as the interengageable annular sealing surfaces of the annular sealing elements 272 and 274 are in contact, there is no net area on which the crude oil, or other fluid, being pumped can act to tend to separate the annular sealing elements 272 and 274 even when the pump 30 and the triplex 36 are not in operation. It will be noted that when the pump 30 and the triplex 36 are in operation, the lubricating oil pressure assists the springs 276 in maintaining the interengageable annular sealing surfaces of the annular sealing elements 272 and 274 in contact. Under these conditions also there is no net area on the nonrotary annular sealing element 272 on which the inlet pressure of the crude oil can act to tend to separate the annular sealing elements 272 and 274. Thus, so locating the O-ring 284 that it seals on the same diameter as the outside diameter of the interengageable annular sealing surfaces of the annular sealing elements 272 and 274 represents an important feature.

From the inner annular chamber 270 inwardly of the interengageable annular surfaces of the annular sealing elements 272 and 274, the lubricating oil flows to the bearing 44 through a small annular clearance between the seal retainer 266 and a sleeve 286 on a reduced portion of the rotor shaft 48. The annular clearance between the seal retainer 266 and the sleeve 286 has a radial dimension of, for example, 0.003 inch and forms a restriction for leaking lubricating oil from the inner annular chamber 270 to the bearing 44 at a greatly reduced pressure, such pressure being substantially atmospheric, and at a slow rate suitable for proper lubrication of the bearing 44. For example, lubricating oil may leak to the bearing 44 from the inner annular chamber 270 at a rate of about 0.1 gallon per minute. From the bearing 44, the excess lubricating oil flows through a branch drain passage 288 into a main drain passage 290 which communicates with the crankcase of the triplex 36. Thus, excess lubricating oil is returned to the triplex crankcase in a substantially uncontaminated condition.

The lubricating oil also flows from the inner annular chamber 270 through a passage 292 in the rotor shaft 48 to an inner annular chamber 294 adjacent and radially inwardly of interengageable annular sealing surfaces of nonrotary and rotary annular sealing elements 296 and 298 constituting an annular seal means interposed between the rotor chamber 94 and the bearing 46. The nonrotary annular sealing element 296 is carried by a seal retainer 300 in the same manner as the nonrotary annular sealing element 272 is carried by the seal retainer 266. The rotary annular sealing element 298 is removably secured to the rotor shaft 48 by a pin 302 which permits axial removal of this annular sealing element. Radially outwardly of the interengageable annular sealing surfaces of the annular sealing elements 296 and 298 is an outer annular chamber 304 to which the pumped fluid at the inlet pressure is delivered through ports 306, FIGS. 1 and 17, in the end plate 80. The nonrotary annular sealing element 296 is sealed with respect to the seal retainer 300 by an O-ring 308 acting on the same diameter as the outside diameter of the interengageable annular sealing surfaces of the two annular sealing elements 296 and 298. Thus, the annular seal means represented by the annular sealing elements 296 and 298 has the same

structure and mode of operation as the annular seal means constituted by the annular sealing elements 272 and 274. Consequently, a further description is not required.

The seal retainer 300 cooperates with a sleeve 310 on the rotor shaft 48 to provide an annular labyrinth seal means for leaking lubricating oil at reduced pressure and a slow rate to the bearing 46 in the same manner as lubricating oil is leaked to the bearing 44, no further description being necessary, therefore. The excess lubricating oil leaked to the bearing 46 in this manner flows through a drain passage 312 in the end closure 42, a nipple 314, and a branch drain passage 316 in the case 34, the branch drain passage 316 communicating with the main drain passage 290 so that excess lubricating oil from the bearing 46 is also returned to the crankcase of the triplex 36.

The pump 30 is so designed that the components thereof subject to the most rapid wear, such as the end and center plates 70, 80 and 76, the rings 88 and 90 and the rotors 86 and 98, can be removed readily in the field and without detaching the housing 32 of the pump from the triplex 36, or disconnecting the inlet line 64. Considering how this is accomplished, it will be noted that the bearing 46 is a simple roller bearing the inner race of which has an outside diameter less than the inside diameters of components to be removed thereover, viz., the seal retainer 300, the annular sealing elements 296 and 298, the end and center plates 70, 80 and 76, the rotors 96 and 98, the rings 88 and 90 and the annular spacers 74 and 78. The outer race of the bearing 46 is carried by the end closure 42, as by clamping it between such end closure and the seal retainer 300, the latter being bolted, or otherwise secured, to the end closure 42. The roller bearing elements of the bearing 46 are carried by the outer race of the bearing 46 and are disengageable from the inner race thereof. Thus, it is unnecessary to remove the inner race of the bearing 46 from the rotor shaft 48 in removing the components hereinbefore discussed.

Considering how the foregoing disassembly of the pump 30 is accomplished, it is first necessary to remove the spring means 144 and 146 and the control means 148 and 150, which may be accomplished readily by removing the clamps 152 and 154. Also, the thrust block assemblies 124 and 126 must be removed. Then, the end closure 42 is detached from the case 34, the end closure carrying with it the outer race of the bearing 46, the roller bearing elements of this bearing, the seal retainer 300, and the nonrotary annular sealing element 296. The rotary annular sealing element 298 may then be removed, whereupon the end plate 80, the rotor 94, the ring 90, the annular spacer 78, the center plate 76, the rotor 96, the ring 88, the annular spacer 74 and the end plate 70 may be removed, all without removing the inner race of the bearing 46. Thus, access to the components of the pump 30 which are most susceptible to wear is provided, and such access is provided without disconnecting the case 34 and the rotor shaft 48 from the triplex.

If removal of such components as the rotor shaft 48, the annular sealing elements 272 and 274, the seal retainer 266, and/or the bearing 44 is required, then, of course, the entire pump 30 must be disconnected from the triplex 36 and from the inlet line 64. However, access to such components is required only very infrequently, access to the components subject to the most rapid wear being readily accomplished, as hereinbefore outlined, without disconnecting the pump 30 from the triplex 36 and without disconnecting the inlet line 64.

Although an exemplary embodiment of the invention has been disclosed herein for purposes of illustration, it will be understood that various changes, modifications and substitutions may be incorporated in such embodiment without departing from the spirit of the invention as defined by the following claims.

I claim:

1. In combination: a case providing a rotor axis; a rotor shaft mounted in said case for rotation about said rotor axis; two end plates and a center plate mounted in said case and spaced apart along said rotor axis; two rings respectively disposed between said center plate and said end plates and respectively having ring axes parallel to and spaced laterally from said rotor axis and angularly spaced apart relative to said rotor axis by 180°, said rings and said end and center plates cooperating to provide two rotor chambers separated by said center plate, said rings being movable laterally relative to said case in directions to move said ring axes toward and away from said rotor axis; two rotors of the sliding vane type mounted on said rotor shaft and respectively occupying said rotor chambers, said end plates respectively being provided therein with inlets respectively communicating with said rotor chambers and respectively being provided therein with outlets respectively communicating with said rotor chambers, said inlets being angularly spaced apart relative to said rotor axis by 180° and said outlets being angularly spaced apart relative to said rotor axis by 180°, whereby lateral forces respectively applied to said rotor shaft by said rotors are 180° out of phase relative to said rotor axis and thus balance each other, said center plate being provided therein with dummy inlets respectively complementary to and facing said inlets in said end plates and being provided therein with dummy outlets respectively complementary to and facing said outlets in said end plates, said dummy inlets communicating with said inlets and said dummy outlets communicating with said outlets only through the spaces between said vanes of said rotors, whereby axial pressure forces applied to said rotors are balanced out; two spring means respectively engaging said rings for biasing said rings in directions to move said ring axes away from said rotor axis; two separate fluid operated control means respectively engaging said rings for moving said rings in directions to move said ring axes toward said rotor axis; and means interconnecting said outlets and said control means for applying a control pressure to said control pistons.

2. In combination: a case providing a rotor axis; a rotor shaft mounted in said case for rotation about said rotor axis; two end plates and a center plate mounted in said case and spaced apart along said rotor axis; two rings respectively disposed between said center plate and said end plates and respectively having ring axes parallel to and spaced laterally from said rotor axis and angularly spaced apart relative to said rotor axis by 180°, said rings and said end and center plates cooperating to provide two rotor chambers separated by said center plate, said rings being movable laterally relative to said case in directions to move said ring axes toward and away from said rotor axis; two rotors of the sliding vane type mounted on said rotor shaft and respectively occupying said rotor chambers, said end plates respectively being provided therein with inlets respectively communicating with said rotor chambers and respectively being provided therein with outlets respectively communicating with said rotor chambers; two spring means respectively engaging said rings for biasing said rings in directions to move said ring axes away from said rotor axis; two separate fluid operated control means respectively engaging said rings for moving said rings in directions to move said ring axes toward said rotor axis; and means interconnecting said outlets and said control means for applying a control pressure to said control pistons.

3. In combination: a case providing a rotor axis; a rotor shaft mounted in said case for rotation about said rotor axis; two end plates and a center plate mounted in said case and spaced apart along said rotor axis; two rings respectively disposed between said center plate and said end plates and respectively having ring axes parallel to and spaced laterally from said rotor axis and angularly spaced apart relative to said rotor axis by 180°, said rings and said end and center plates cooperating to provide two rotor chambers separated by said center plate, said

rings being movable laterally relative to said case in directions to move said ring axes toward and away from said rotor axis; two rotors of the sliding vane type mounted on said rotor shaft and respectively occupying said rotor chambers, said end plates respectively being provided therein with inlets respectively communicating with said rotor chambers and respectively being provided therein with outlets respectively communicating with said rotor chambers; two spring means respectively engaging said rings for biasing said rings in directions to move said ring axes away from said rotor axis; two separate fluid operated control means respectively engaging said rings for moving said rings in directions to move said ring axes toward said rotor axis; means interconnecting said outlets and said control means for applying a control pressure to said control pistons; and two thrust bearing means interposed between said case and said rings, respectively, said thrust bearing means being located on the same sides of said rotor chambers as and being generally radially aligned with said outlets, respectively.

4. In combination: a housing having a rotor axis and provided therein with two rotor chambers spaced apart along said rotor axis, said rotor chambers respectively having chamber axes parallel to and spaced laterally from said rotor axis, said chamber axes being angularly spaced apart relative to said rotor axis by 180°; a rotor shaft mounted in said housing and rotatable about said rotor axis; and two vaned rotors mounted on said rotor shaft and respectively occupying said rotor chambers, said housing being provided therein with inlets respectively communicating with said rotor chambers and being provided therein with outlets respectively communicating with said rotor chambers, said inlets being angularly spaced apart relative to said rotor axis by 180° and said outlets being angularly spaced apart relative to said rotor axis by 180°, whereby lateral forces respectively applied to said rotor shaft by said rotors are 180° out of phase relative to said rotor axis and thus balance each other, said housing including a case which encircles said rotor chambers, said case including radially spaced inner and outer walls providing an annular space therebetween and including partitions dividing said annular space into inlet and outlet chambers respectively communicating with said inlets and said outlets, said case having main inlet and outlet ports in communication with said inlet and outlet chambers.

5. In combination: a case providing a rotor axis; a rotor shaft mounted in said case for rotation about said rotor axis; two end plates and a center plate mounted in said case and spaced apart along said rotor axis; two rings respectively disposed between said center plate and said end plates and respectively having ring axes parallel to and spaced laterally from said rotor axis and angularly spaced apart relative to said rotor axis by 180°, said rings and said end and center plates cooperating to provide two rotor chambers separated by said center plate, said rings being movable laterally relative to said case in directions to move said ring axes toward and away from said rotor axis; two rotors of the sliding vane type mounted on said rotor shaft and respectively occupying said rotor chambers, said end plates respectively being provided therein with inlets respectively communicating with said rotor chambers and respectively being provided therein with outlets respectively communicating with said rotor chambers; two spring means respectively including spring housings having therein springs engaging said rings for biasing said rings in directions to move said ring axes away from said rotor axis; two separate fluid operated control means respectively including control cylinders having therein control pistons engaging said rings for moving said rings in directions to move said ring axes toward said rotor axis; said case having means therein providing external clearances respectively extending around said rings and respectively located between said end plates and said center plate, said end plates having passages therein respectively connecting said inlets to said clearances; two passages means respectively connecting said clearances to said spring housings; two passage means respectively connecting said clearances to corresponding ends of said control cylinders; and means for delivering a control pressure to the other ends of said control cylinders.

6. In combination: a case providing a rotor axis; a rotor shaft mounted in said case for rotation about said rotor axis; two end plates and a center plate mounted in said case and spaced apart along said rotor axis; two rings respectively disposed between said center plate and said end plates and respectively having ring axes parallel to and spaced laterally from said rotor axis and angularly spaced apart relative to said rotor axis by 180°, said rings and said end and center plates cooperating to provide two rotor chambers separated by said center plate, said rings being movable laterally relative to said case in directions to move said ring axes toward and away from said rotor axis; two rotors of the sliding vane type mounted on said rotor shaft and respectively occupying said rotor chambers, said end plates respectively being provided therein with inlets respectively communicating with said rotor chambers and respectively being provided therein with outlets respectively communicating with said rotor chambers; two spring means respectively engaging said rings for biasing said rings in directions to move said ring axes away from said rotor axis; two separate fluid operated control means respectively including control pistons engaging said rings for moving said rings in directions to move said ring axes toward said rotor axis; and means interconnecting said outlets and said control means for applying a control pressure to said control pistons.

7. In combination: a case providing a rotor axis; a rotor shaft mounted in said case for rotation about said rotor axis; two end plates and a center plate mounted in said case and spaced apart along said rotor axis; two rings respectively disposed between said center plate and said end plates and respectively having ring axes parallel to and spaced laterally from said rotor axis and angularly spaced apart relative to said rotor axis by 180°, said rings and said end and center plates cooperating to provide two rotor chambers separated by said center plate, said rings being movable laterally relative to said case in directions to move said ring axes toward and away from said rotor axis; two rotors of the sliding vane type mounted on said rotor shaft and respectively occupying said rotor chambers, said end plates respectively being provided therein with inlets respectively communicating with said rotor chambers and respectively being provided therein with outlets respectively communicating with said rotor chambers; two spring means respectively engaging said rings for biasing said rings in directions to move said ring axes away from said rotor axis; two separate fluid operated control means respectively including control pistons engaging said rings for moving said rings in directions to move said ring axes toward said rotor axis; and pilot valve means actuable by the pressure in said outlets for applying to said control pistons a control pressure which is a function of, but less than, the pressure in said outlets.

8. In combination: a case providing a rotor axis; a rotor shaft mounted in said case for rotation about said rotor axis; two end plates and a center plate mounted in said case and spaced apart along said rotor axis; two rings respectively disposed between said center plate and said end plates and respectively having ring axes parallel to and spaced laterally from said rotor axis and angularly spaced apart relative to said rotor axis by 180°, said rings and said end and center plates cooperating to provide two rotor chambers separated by said center plate, said rings being movable laterally relative to said case in directions to move said ring axes toward and away from said rotor axis; two rotors of the sliding vane type mounted on said rotor shaft and respectively occupying said rotor chambers, said end plates respectively being

provided therein with inlets respectively communicating with said rotor chambers and respectively being provided therein with outlets respectively communicating with said rotor chambers; two spring means angularly spaced apart relative to said rotor axis by 180° and respectively engaging said rings for biasing said rings in directions to move said ring axes away from said rotor axis; two fluid operated control means angularly spaced apart relative to said rotor axis by 180° and respectively axially aligned with said spring means and respectively engaging said rings for moving said rings in directions to move said ring axes toward said rotor axis; and two clamping means each securing one of said spring means and the adjacent one of said control means to said case.

9. In combination: a case providing a rotor axis and open at one end; a rotor shaft rotatable in said case about said rotor axis; two end plates and a center plate mounted in said case and spaced apart along said rotor axis; two rings respectively disposed between said center plate and said end plates and respectively having ring axes parallel to and spaced laterally from said rotor axis and angularly spaced apart relative to said rotor axis by 180°, said rings and said end and center plates cooperating to provide two rotor chambers separated by said center plate; two rotors of the sliding vane type mounted on said rotor shaft and respectively occupying said rotor chambers, said end plates respectively being provided therein with inlets respectively communicating with said rotor chambers and respectively being provided therein with outlets respectively communicating with said rotor chambers, said inlets being angularly spaced apart relative to said rotor axis by 180° and said outlets being angularly spaced apart relative to said rotor axis by 180°, whereby lateral forces respectively applied to said rotor shaft by said rotors are 180° out of phase relative to said rotor axis and thus balance each other; an end closure connected to said case and closing said open end thereof; and a rotor shaft bearing interposed between said rotor shaft and said end closure and including an inner race mounted on said shaft and including an outer race and rotatable bearing elements mounted on said end closure and separable from said inner race, the outside diameter of said inner race being less than the inside diameters of said end and center plates, said rings and said rotors, whereby said end and center plates, said rings and said rotors may be withdrawn from said case, through said open end thereof, over said inner race, after removal of said end closure, said outer race and said rotatable bearing elements, without removing said inner race from shaft.

10. In combination: a case providing a rotor axis; a rotor shaft mounted in said case for rotation about said rotor axis; two end plates and a center plate mounted in said case and spaced apart along said rotor axis; two rings respectively disposed between said center plate and said end plates and respectively having ring axes parallel to and spaced laterally from said rotor axis and angularly spaced apart relative to said rotor axis by 180°, said rings and said end and center plates cooperating to provide two rotor chambers separated by said center plate, said rings being movable laterally relative to said case in directions to move said ring axes toward and away from said rotor axis; two rotors of the sliding vane type mounted on said rotor shaft and respectively occupying said rotor chambers, said end plates respectively being provided therein with inlets respectively communicating with said rotor chambers and respectively being provided therein with outlets respectively communicating with said rotor chambers; said case including radially spaced inner and outer walls providing therebetween an annular space which encircles said rotor chambers, said case including partitions dividing said annular space into inlet and outlet chambers respectively communicating with said inlets and said outlets, said case having main inlet and outlet ports in communication with said inlet and outlet chambers; two spring means mounted on said case and extending radially through said annular space and respectively engaging said rings for biasing said rings in directions to move said ring axes away from said rotor axis; and two fluid operated control means mounted on said case and extending radially through said annular space and respectively engaging said rings for moving said rings in directions to move said ring axes toward said rotor axis.

11. The combination set forth in claim 10: wherein said spring means and said control means are movable radially outwardly out of engagement with said case; wherein said case has an open end and a removable end closure for closing said open end thereof; and wherein said end and center plates, said rings and said rotors are movable axially out of said case through said open end thereof, after removal of said end closure and after radial disengagement of said spring means and said control means.

12. The combination set forth in claim 10: wherein said case has means therein providing external clearances respectively extending around said rings and respectively located between said end plates and said center plate, said end plates having passages therein respectively connecting said inlets to said clearances; wherein said spring means respectively include spring housings having springs therein and respectively include tubular members bridging said annular space and seated against said rings and said springs, respectively, and providing fluid communication between said clearances and said spring housings, respectively; and wherein said control means respectively include control cylinders having control pistons therein and respectively include tubular members bridging said annular space and seated against said rings and said control pistons, respectively, and providing fluid communication between said clearances and said control cylinders, respectively.

**References Cited** in the file of this patent

UNITED STATES PATENTS

| | | |
|---|---|---|
| 1,460,487 | Hawkins | July 3, 1923 |
| 1,943,929 | Rayburn | Jan. 16, 1934 |
| 2,031,749 | Vincent | Feb. 25, 1936 |
| 2,064,421 | Erskine | Dec. 15, 1936 |
| 2,192,660 | Johnson | Mar. 5, 1940 |
| 2,313,246 | Kendrick et al. | Mar. 9, 1943 |
| 2,469,097 | Wrenn | May 3, 1949 |
| 2,558,970 | Lipfert | July 3, 1951 |
| 2,710,581 | Rosaen | June 14, 1955 |
| 2,720,171 | Harrington | Oct. 11, 1955 |
| 2,764,941 | Miller et al. | Oct. 2, 1956 |
| 2,799,995 | Herman | July 23, 1957 |
| 2,804,016 | Moore | Aug. 27, 1957 |
| 2,809,595 | Adams et al. | Oct. 15, 1957 |
| 2,845,873 | Lapsley | Aug. 5, 1958 |
| 2,854,927 | Berg | Oct. 7, 1958 |
| 2,921,439 | Krafft et al. | Jan. 19, 1960 |
| 2,975,717 | Rynders et al. | Mar. 21, 1961 |