April 30, 1957 J. W. F. HOLL 2,790,391

# TWO STAGE VARIABLE DELIVERY VANE-TYPE PUMP

Filed Nov. 19, 1954 3 Sheets-Sheet 1

FIG. 1.

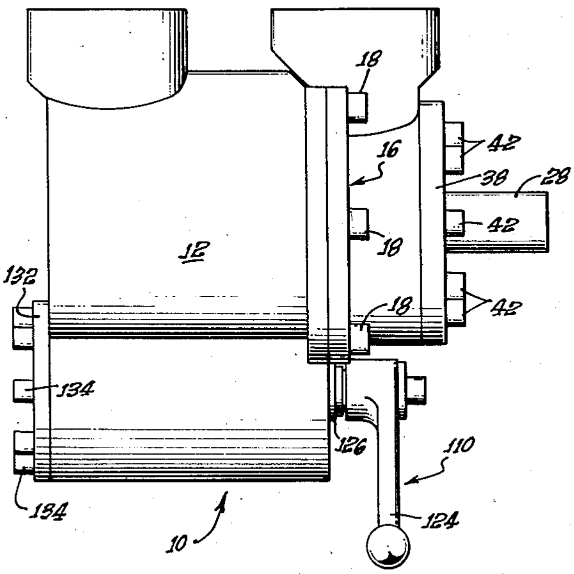



FIG. 2.

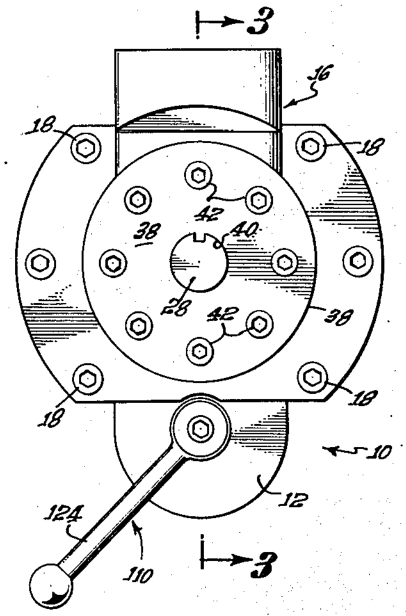



FIG. 8.

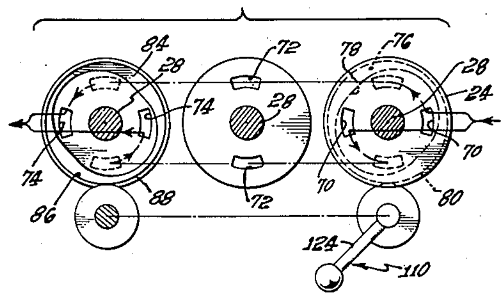
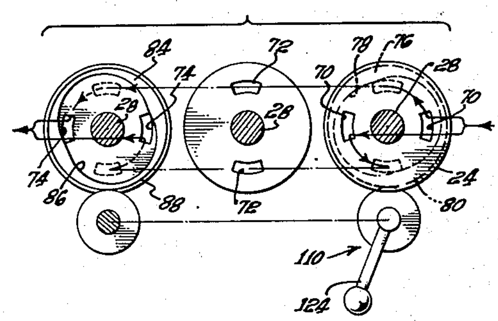



FIG. 9.



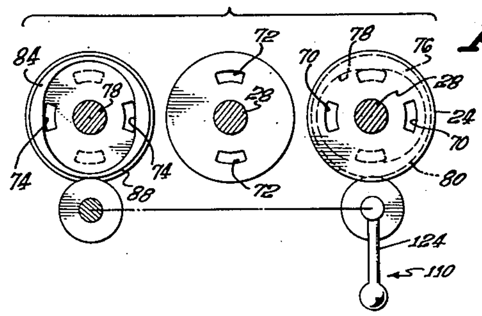



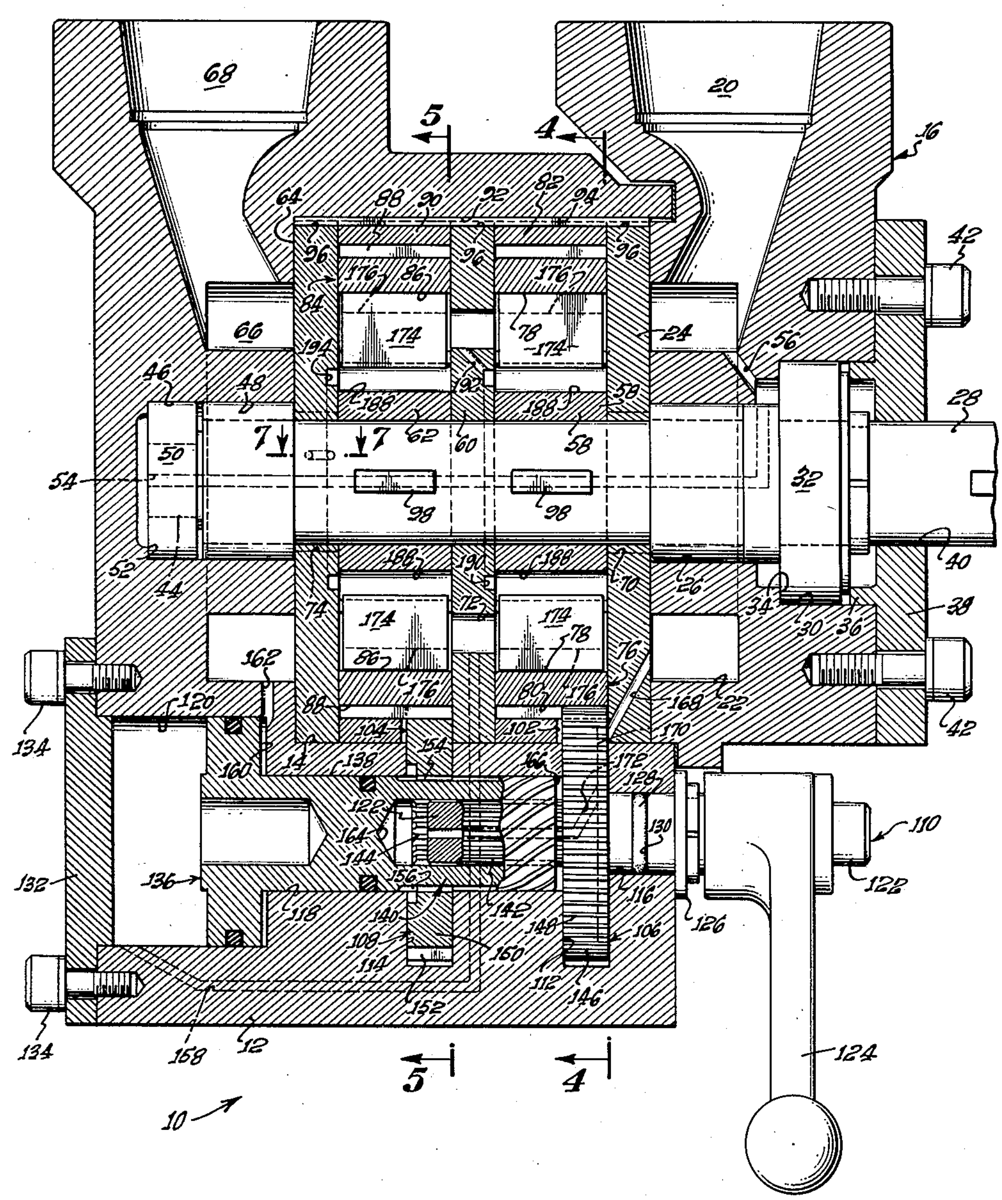

FIG. 10.

JAMES W. F. HOLL,
INVENTOR.

BY HIS ATTORNEYS.

HARRIS, KIECH, FOSTER & HARRIS.

FIG. 3.



JAMES W. F. HOLL,
INVENTOR.

BY HIS ATTORNEYS.

HARRIS, KIECH, FOSTER & HARRIS.

TWO STAGE VARIABLE DELIVERY VANE-TYPE PUMP

Filed Nov. 19, 1954 3 Sheets-Sheet 3

FIG. 5.

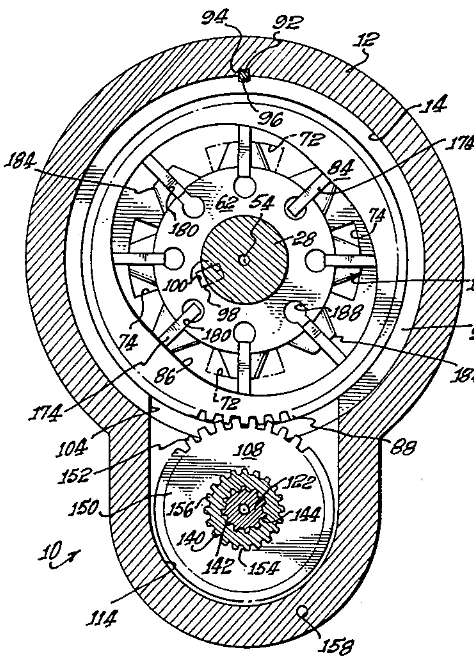



FIG. 4.

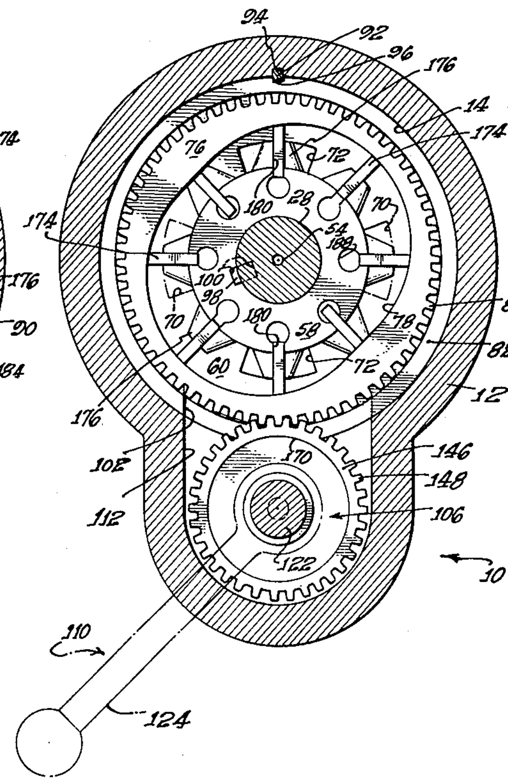



FIG. 6.

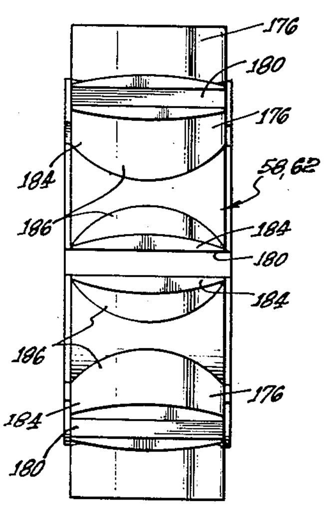
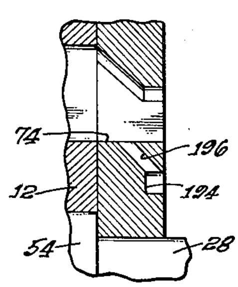



FIG. 7.




JAMES W. F. HOLL,
INVENTOR.

BY HIS ATTORNEYS.

HARRIS, KIECH, FOSTER & HARRIS.

United States Patent Office

2,790,391

Patented Apr. 30, 1957



2,790,391

# TWO STAGE VARIABLE DELIVERY VANE-TYPE PUMP

James W. F. Holl, Temple City, Calif.

Application November 19, 1954, Serial No. 469,933

7 Claims. (Cl. 103—5)

The present invention relates to a two stage variable volume vane-type pump in which the work performed by each of the pump stages is equal regardless of the volume of liquid being delivered by the pump.

It is a broad object of the present invention to provide a relatively simple, yet exceedingly effectual, two stage vane-type pump which can be used to deliver varying volumes of liquid at a variety of pressures. A further object of the present invention is to provide an efficient pump of the class described in which means are provided so that the work performed by each of the pumping stages is equal regardless of the output of the pump or the pressure head against which it is working.

It will be realized by those skilled in the art that a pump having the characteristics indicated in the foregoing description represents a decided improvement over the presently available two stage vane-type pumps. The only available pumps of this category are designed to produce a constant volume, and the first pumping stage within them is of generally larger displacement than the second stage because of the volume lost between the intake and outlet ports of each stage under various pressure conditions. This size differential between the two stages can be worked out fairly satisfactorily for a fixed speed and output pressure, but, when a pump is working at other than the designed conditions, the size differential between the two pumping stages is out of proportion. One pumping stage will be large and will be overworking at a power loss in addition to fluid loss by way of the pressure relief valve between the two stages. Therefore, these prior two stage constant volume vane-type pumps are satisfactory only at one working pressure. It will be realized from a description of the pump of the present invention that these disadvantages are not present within the structure herein described.

Further objects and advantages of the pump of the present invention will be apparent from the remainder of this specification and the appended claims. For convenience, the two stage vane-type pump of the present invention may be briefly summarized as required by the Patent Office rules as including a housing within which there is located a pumping chamber, which pumping chamber in turn contains means defining a first and a second pumping stage; means for turning said first and second pumping stages so as to operate the same; and control means located within said housing for proportioning the work performed by each of said pumping stages so that such work is equally divided between these stages regardless of the output of the pump or the pressures against which the pump is working, said control means including means for varying the output of the pump. The invention is, of course, more fully defined and summarized in detail by the appended claims forming a part of this description. It is best further described directly by reference to the accompanying drawings, in which:

Fig. 1 is a side view of a pump of the invention;

Fig. 2 is an end view of the pump shown in Fig. 1



showing the end of this pump located at the right of Fig. 1;

Fig. 3 is a cross-sectional view taken at line 3—3 of Fig. 2 of the drawings;

Fig. 4 is a cross-sectional view taken at line 4—4 of Fig. 3 of the drawings;

Fig. 5 is a cross-sectional view taken at line 5—5 of Fig. 3 of the drawings;

Fig. 6 is a side view of a pump rotor employed with the invention;

Fig. 7 is a fragmentary sectional view taken at line 7—7 of Fig. 3; and

Figs. 8, 9 and 10 are diagrammatic views illustrating the operation of the pump of the present invention.

It is readily seen from an examination of Figs. 1, 2 and 3 of the drawings that the pump **10** of the invention includes a housing **12** within which there is located a cylindrical pump chamber **14**, one end of which is normally closed by means of a cover plate **16** which is attached to the housing **12** by means of bolts **18**. The cover plate **16** includes an inlet **20** leading to an annular groove **22** facing an inlet port plate **24** held within the pump chamber **14** immediately adjacent the cover plate **16**. The cover plate **16** also includes a central cylindrical bore **26** which is adapted to hold a shaft **28** used in driving the pump **10**. This shaft projects from the cover plate **16** out of the pump **10** through a cylindrical chamber **30** where a cylindrical bearing **32** attached to the shaft **28** is secured against a shoulder **34** upon the cover plate **16** by means of an annular shoulder **36** formed upon a retaining plate **38** and engaging the side of the bearing **32** remote from the pump chamber **14**. This retaining plate **38** includes an opening **40** through which the shaft **28** projects and is held against the cover plate **16** by means of bolts **42**.

The shaft **28** projects entirely through the pump chamber **14** in such a position that the axis of this shaft coincides with the axis of the pump chamber **14**. An enlarged end **44** of the shaft remote from the cover plate **16** is held against an annular shoulder **52** within a cylindrical bore **46** by means of a sleeve **48** carrying the shaft **28** and fitting within this bore **46**. Thus, the end **44** of the shaft **28** is spaced slightly from the actual end of the bore **46**. With this construction any fluid which may escape by leakage from the pump chamber **14** around the shaft **28** to within the bore **46** can be conveyed through a passage **54** formed within the shaft **28** and leading to the chamber **30**. This chamber **30** is in fluid communication by means of another passage **56** with the inlet **20**. Thus, any undesired fluid entering the chamber **30** as by leakage is exhausted during the operation of the pump from this chamber **30** so as to be commingled with fluid entering the pump **10** through the inlet **20**.

Within the pump chamber **14** there is located a first pump rotor **58**, a center port plate **60**, a second pump rotor **62**, and an outlet port plate **64**, in addition to the inlet port plate **24** previously described. As is most easily seen in Fig. 3 of the drawings, the outlet port plate **64** is located within the portion of the pump chamber **14** furthest removed from the cover plate **16** immediately adjacent an annular groove **66** communicating with a pump outlet **68**. Between the port plates **24** and **64**, the first and second pump rotors are held in what may be loosely termed a "sandwich-type" construction with the center port plate **60** disposed between these two rotors.

The inlet port plate **24** is provided with two ports **70** which are located symmetrically with respect to the shaft **28** on opposite sides of this shaft. The center port plate **60** is provided with ports **72** which are also located symmetrically with respect to the shaft **28** on opposite sides of this shaft. These ports **72** are located in a 90° relationship with respect to the ports **70** when the position of these ports is considered with respect to the axis of the shaft 28. The outlet port plate 64 is provided with two ports 74. These last ports are aligned with the ports 70 in the inlet port plate 24 when the position of these ports is considered with respect to the axis of the shaft 28. Thus, the ports 74 are located at an angle of 90° with respect to the shaft 28 from the ports 72 in the center port plate 60. All of the ports 70, 72 and 74 are preferably formed of substantially the shape of a sector of a ring and are located the same distance from the axis of the shaft 28. In effect, the port plates 24, 60 and 64 may be termed "port discs" inasmuch as they are essentially of disc-like shape.

The first pump rotor is surrounded by a first eccentric ring 76 having an eccentric or cam-shaped internal surface 78 and an outer periphery 80 formed in the shape of a gear having conventional gear teeth. This first eccentric ring is spaced within the pump chamber 14 about the first rotor 58 by means of a first spacer ring 82 fitting within the walls of the pump chamber 14. The second pump rotor 62 is similarly surrounded by a second eccentric ring 84 which also has a cam or eccentric shaped internal surface 86 and an outer periphery 88 which is formed in the shape of a common gear. The second eccentric ring 84 is held about the second pump rotor 62 by means of a second spacer ring 90 fitting within the pump chamber 14.

The first spacer ring 82, the second spacer ring 90, and the port plates 24, 60 and 64 are all secured against rotation by means of a key 92 which is fitted within a groove 94 within the wall of the pump chamber 14, and within appropriate grooves 96 formed in these five members. The pump rotors 58 and 62 are also similarly secured to the shaft 28 by means of keys 98 fitting within grooves 100 in the shaft 28 and the rotors 58 and 62 (Figs. 4 and 5). If desired, the spacer rings 82 and 90 and the port plates 24, 60 and 64 can be secured together by means of bolts (not shown) before these members and the various other members normally held in position between the spacer rings 82 and 90 and the port plates 24 and 64 are placed within the housing 12. This construction is particularly advantageous where it is desired to assemble the complete pump 10 by sections. Thus, all of the members indicated in this discussion may be secured together as a subassembly before being placed within the housing 12.

For the spacer rings 82 and 90 to operate satisfactorily, they must be very carefully constructed so that their dimensions are slightly larger than the thickness of the rotors 58 and 62. Thus, with this type of construction, it is impossible to secure the spacer rings 82 and 90 so close together as to impede rotation of the pump rotors when the complete pump 10 is assembled as shown in Figs. 1, 2 and 3 of the drawings, or when the pump 10 is manufactured using the subassembly technique, forming the spacer rings 82 and 90, the port plates 24, 60 and 64, and the various members secured within and between these members as a complete subassembly unit.

Both the eccentric rings 76 and 84 are provided with small segmental openings 102 and 104, respectively, which are designed to be traversed by gears 106 and 108 forming a part of control means 110 employed with the invention. These two gears 106 and 108 are located almost wholly within flat slots 112 and 114, respectively, formed within the pump housing 12. These slots are substantially identical in shape and are positioned at right angles to the axis of the shaft 28. Within the housing 12 there is located a small bore 116 which is axially aligned with an intermediate bore 118 which is in turn axially aligned with an enlarged cylindrical piston chamber 120. The bores 116 and 118 and the chamber 120 are all aligned with the axis of the shaft 28. The bore 116 is, as is best seen in Fig. 3, open to the side of the pump housing 12 adjacent the cover plate 16 and is designed to carry a shaft 122 which projects from the outside of the housing 12 through the bore 116 into the intermediate bore 118. An appropriate conventional handle 124 is secured to this shaft adjacent the outside of the housing 12. A collar 126 is provided on the shaft 122 so as to limit the movement of this shaft towards the housing 12. Also, a conventional seal 128 is provided within a groove 130 in the shaft 122 within the bore 116 so as to prevent fluid leakage out of the housing 12.



The piston chamber 120 is normally sealed by means of a cap plate 132 secured to the housing 12 by bolts 134, and is designed to carry, as the name of this chamber implies, a piston 136 which slides within this chamber 120. A center portion 138 of the piston 136 projects from the piston chamber 120 into the intermediate bore 118 where a generally cylindrical gear member 140 is attached to or formed upon the portion 138 so as to project about conventional gear teeth 142 formed on the portion of the shaft 122 within the intermediate bore 118. Gear teeth corresponding to the gear teeth 142 are formed upon the inner surface of this cylindrical gear member 140 so as to engage the gear teeth 142. Thus, with this construction the gear member 140 can slide with respect to the shaft 122 with the two sets of gear teeth 142 and 144 engaging one another at all times. Such sliding, of course, occurs as the piston 136 moves within the piston chamber 120.

A first control gear 146 is secured about the shaft 122 within the slot 112 so as to project through the opening 102 within the first spacer ring 82 so that the teeth 148 on this first control gear 146 coact with the teeth on the outer periphery 80 of the first eccentric ring 76. Positioned within the second slot 114 is a second control gear 150 which is located about the gear member 140 so as to project from the slot 114 past the opening 104 so that the teeth 152 formed on the outer surface of this second control gear 150 coact with the teeth on the outer periphery 88 of the second eccentric ring 84. The interior surface of the second control gear 150 is provided with spiral gear teeth 154 which are adapted to engage similar spiral gear teeth 156 formed upon the outer surface of the gear member 140.

It is readily realized from a consideration of Fig. 3 of the drawings that both of these sets of spiral gear teeth are located within the inner bore 118 in such a manner that as the piston 136 is moved within the piston chamber 120 the attached gear member 140 will be moved within the intermediate bore 118, causing the second control gear 150 to rotate. Rotation of the second control gear 150 may also be caused by the handle 124 being turned so as to rotate the shaft 122. Since such rotation transmitted through the gear teeth 142 and 144 to the gear member 140, and thence to the second control gear 150, it is also obvious that when the handle 124 is turned the first control gear 146 attached to the shaft 122 will also be turned. During such rotation of the handle, the location of the piston 136 within the piston chamber 120 will be regulated by virtue of the angle of the spiral gear teeth 154 and 156 forcing this piston to move.

The end of the piston chamber 120 immediately adjacent the cap plate 132 is connected by means of a passage 158 to one of the ports 72 within the center port plate 60. Thus, with this construction the end of the piston 136 adjacent the cap plate 132 is at all times during the operation of the pump 10 subjected to the pressure of the hydraulic fluid passing from a first pumping stage which is constituted by means of the first pump rotor 58 and the adjacent parts. A surface 160 of the piston 136 surrounding the base of the center portion 138 is connected by means of another passage 162 to the annular groove 66. Thus, this surface 160 is, during the operation of the pump 10, subjected to the pressure of hydraulic fluid within the annular groove 66 coming from a second pumping stage which is constituted by means of the second pump rotor 62 and the adjacent parts.

The end 164 of the center portion 138 of the piston 136, and the end 166 of the gear member 140 are both subjected during the operation of the pump 10 to the pressure of the inlet fluid coming into this pump by means of a passage 168 in the inlet port plate 24 leading to an annular groove 170 within the first control gear 146, which annular groove is connected by means of a passage 172 to the end of the shaft 122. With this construction fluid within the annular groove 22 is in communication with the interior of the gear member 140 and the space between this gear member 140 and the first control gear 146. The gear teeth 142 and 144 are with the construction of the invention preferably manufactured so as not to be an absolute fit in order to permit fluid flow between these gear teeth.

It will be seen from a consideration of the above description that the piston 136 effectively has three areas, the largest of which is subjected to the pressure of fluid passing from the first pumping stage within the pump 10, and the other two of which are subjected to the inlet and outlet pressures of the pump 10.

By making these three areas of the proper dimensions, it is possible to counterbalance the forces acting upon the two smaller areas against the force acting upon the larger area in order to balance the work load performed by each of the pumping stages either equally or in any desired proportion. In order that both of these pumping stages perform an equal amount of work, it is necessary that the forces acting upon the surface 160 in a direction parallel to the axis of the piston chamber 120, plus the forces acting upon the piston 136 in the same direction resulting from the pressure of the fluid within the inlet to the pump 10, be equal to the forces acting upon the enlarged end of the piston in a direction parallel to the axis of the piston chamber 120. It is further necessary that the areas which are subjected to pressure of the inlet and outlet fluids within the pump 10 during the operation of this pump be so proportioned that the effective components of these pressures parallel to the axis of the piston chamber exerted by these inlet and outlet fluids upon the piston 136 be each equal to one-half of the effective area of the surface 160 which is subjected to forces tending to move the piston 136 within the piston chamber 120. By virtue of this construction, the piston 136 will move within the piston chamber whenever the work done by either of the two pump rotors 58 or 62 is greater than the work done by the other of the pump rotors in order to adjust the operation of the pump 10 so that both of the pump rotors will perform an equal proportion of the work done. This adjustment will take place automatically whenever the handle 124 is turned so as to adjust the volumetric output of the pump 10.

The actual pumping operation performed within the pump 10 is carried out in an essentially conventional manner utilizing vanes 174 projecting from vane guides 176 located upon the external surfaces of the first and second pump rotors 58 and 62. It is readily seen from an examination of Figs. 4, 5 and 6 that the vane guides employed are arranged in pairs about the exterior of the pump rotors 58 and 62 so that straight sides of these vanes are parallel to one another defining slots 180 forming continuations of slots (designated by the same number) formed within the interior of the pump rotors 58 and 62.

The exterior portions of these vane guides 176 remote from their straight sides are of generally curved configuration so as to have thin, tapered, side edges 184 and enlarged base portions 186. This form of construction of the vane guides 176 is quite advantageous inasmuch as with it the vane guides 176 provide adequate support for the vanes 174, and are not apt to catch upon the ports 70, 72 and 74 as the pump rotors 58 and 62 are turned, and do not obstruct the entry and exit of fluid. Regardless of the positions of the eccentric rings 76 and 84, these ports are located within the cam surfaces 78 and 86, and beyond the peripheries of the rotors 58 and 62. With this construction, the area of the ports are not obstructed when the rings 76 and 84 are turned.



The slots 180 formed within the pump rotors 58 and 62 are connected within these rotors with cylindrical chambers 188 which are designed to convey hydraulic fluid to the slots 180 and 182 so as to constantly force the vanes 174 from these slots against the cam surfaces 78 and 86 of the first and second eccentric rings 76 and 84, respectively. The hydraulic fluid used in this manner during the operation of the pump to force the vanes 174 from the first pump rotor 58 is conveyed to the cylindrical chambers 188 from one of the ports 72 by means of an annular groove 190 formed within the center port plate 60 immediately adjacent the first pump rotor 58 by means of a short passage 192. Hydraulic fluid used for the same purpose in conjunction with the vanes 174 employed with the second pump rotor 62 is conveyed to the cylindrical chambers 188 within this second pump rotor by means of a similar annular groove 194 (Fig. 7) which in turn is connected through another passage 196 to one of the ports 74 within the outlet port plate 64.

During operation of the pump 10 as the shaft 28 is rotated fluid is introduced into this pump through the inlet 20, and thence flows through the ports 70 within the inlet port plate 24 to the space surrounding the first pump rotor 58. The same fluid is then forced by virtue of the operation of the vanes 174 and the first eccentric ring 76 in connection with the pump rotor 58 through the ports 72 within the center port plate 60 from which it flows into the space between the second pump rotor 62 and the second eccentric ring 84. Here, by operation of the vanes 174 in connection with the other parts described, this fluid is forced out through the ports 74 in the outlet port plate 64, and thence into the annular groove 66 and out through the outlet 68.

The position of the first and second eccentric rings during this operation is exceedingly important with respect to the quantity of liquid passed through the pump 10. When the first eccentric ring 76 is located with respect to the ports 70 and 72, as indicated in Fig. 8 of the drawings, the amount of pumping done by the first pump rotor 58 will be the maximum possible. When the first eccentric ring is in this location, the second eccentric ring 84 is located as is indicated in Fig. 8. When it is desired to decrease the output of the pump 10, the handle 124 is moved to a position such as is shown in Fig. 9 of the drawings, turning the eccentric rings 76 and 84 through the operation of the various gear means previously described. As this occurs, the output of the two pumping stages will temporarily be unequal until the piston 136 moves, as previously described, so as to cause the second eccentric ring to turn in order to adjust the output of the second pumping stage to a point where both of the pumping stages perform the same amount of work. If it is desired to keep the pump 10 running without pumping any fluid, the handle 124 may be turned to the position shown in Fig. 10 of the drawings, rotating the first and second eccentric rings 76 and 84 to the positions indicated in this figure.

It is considered obvious from the foregoing description that certain aspects of the present invention may be incorporated within a single stage pumping unit. As an example of this, the precise spacer ring construction shown using the spacer rings 82 and 90 may be incorporated into a single stage pump in which only one spacer ring is used to surround a single pump rotor. Further, the feature of the invention of varying the output of a pumping stage by the use of a handle which serves to turn an eccentric surrounding the pump rotor may also be incorporated into a single stage. This type of construction is exceedingly advantageous because of its efficiency of operation.

The complete pump 10 of the invention includes, as indicated in the preceding discussion, a pump rotor which is turned within an eccentric ring which is of symmetrical construction as indicated in the drawings, this construction being such that an equal amount of pumping is done on both sides of the shaft 28. Because of this, the load upon the shaft 28 during the pumping operation is substantially balanced. This has the effect of reducing the pressures upon the bearing surfaces upon which the shaft 28 rests. This type of construction is in and of itself quite advantageous, particularly in combination with the rotating eccentric ring structure described above. In a two stage pump as herein described, the eccentric rotors are set at approximately a 90° angle with respect to one another during normal operation, as indicated in the drawings. By virtue of the fact that they are so positioned, the bearing pressures are further equalized to a larger extent than is possible in constructions in which a balanced rotor of the type indicated is used in a single stage pump.

Those skilled in the art will realize that an extremely wide variety of modifications may be made within the construction shown without departing from certain essential teachings of the invention herein described. All such modifications are to be considered as part of the inventive concept insofar as they are defined by the appended claims. The construction herein set forth is particularly advantageous in that it is comparatively easily and cheaply manufactured, is extremely efficient for the purpose intended, and is virtually foolproof in operation.

I claim as my invention:

1. A new and improved two stage vane-type pump, which comprises: a housing; means defining a pump chamber located within said housing; a shaft projecting from the exterior of said housing into said housing and through said pump chamber; means defining a first pump rotor secured to said shaft within said pump chamber; means defining a second pump rotor secured to said shaft adjacent said first pump rotor within said pump chamber; means including ports defining an inlet to said pump chamber, said inlet being located adjacent one side of said first pump rotor; means including ports defining an outlet from said pump chamber, said outlet being located on the side of said second pump rotor remote from said first pump rotor; means secured to said housing between said first and said second pump rotors, said means being disposed within said pump chamber and having ports formed therein providing fluid communication between said rotors; first modulator means defining a first cam surface disposed within said pump chamber around said first pump rotor with said first cam surface facing said first pump rotor; second modulator means defining a second cam surface disposed within said pump chamber around said second pump rotor with said second cam surface facing said second pump rotor; means actuated externally of said housing for rotating both said modulator means about said pump rotors; and means actuated internally of said housing for rotating one of said modulator means relative to the other of said modulator means.

2. A vane-type pump as defined in claim 1, wherein said internally actuated means for rotating are responsive to the pressures developed by each of said pump rotors during the operation thereof.

3. A vane-type pump as defined in claim 2, wherein said internally actuated means for rotating include a piston, and wherein said piston is movable in response to the pressures developed by said first and said second pump rotors.

4. A vane-type pump as defined in claim 1, wherein said internally actuated means for rotating include: means defining a piston chamber; a piston movably located within said piston chamber; passage means connecting different portions of said piston chamber with different portions of said pump chamber; and gear means connecting said piston with said first and second modulator means, whereby as said piston is moved within said piston chamber, said gear means cause said cam surfaces to rotate relative to each other.

5. A pump as defined in claim 1, wherein said internally and externally actuated means for rotating include: means defining a piston chamber; a piston located within said piston chamber; a shaft projecting into said piston chamber; means attached to said shaft, whereby when said shaft is turned said piston is rotated; first gear means connected to said shaft and to one of said modulator means, whereby as said shaft is rotated one of said cam surfaces is rotated; second gear means connecting said piston with the other of said modulator means, whereby as said piston is rotated said other cam surface is rotated; and passage means connecting said piston chamber with different portions of said pump, whereby said piston is moved within said piston chamber in accordance with the pressures developed within said pump chamber by the operation of said first and said second pump rotors.

6. In a two stage vane-type pump, the combination of: means defining a pump chamber; a shaft located within said pump chamber; a first pump rotor mounted upon said shaft within said pump chamber; a second pump rotor mounted upon said shaft within said pump chamber; an input chamber including ports communicating with said pump chamber adjacent said first pump rotor; an output chamber including ports communicating with said pump chamber adjacent said second pump rotor; interstage means positioned within said pump chamber around said shaft between said first and said second pump rotors, said interstage means including ports communicating between portions of said pump chamber adjacent said first pump rotor and adjacent said second pump rotor; first and second eccentric rings located about said first and said second pump rotors respectively; power means; means coupling said power means to said first eccentric ring and to said second eccentric ring for rotating said eccentric rings relative to each other; and conduit means coupling said power means to said input chamber, said output chamber and said ports in said interstage means in fluid communicating relationship.

7. A two stage vane-type pump as defined in claim 6, in which said power means includes a piston chamber and a piston slidable therein, said piston and said piston chamber defining three pressure working zones, said zones communicating with said input means, said output means and said interstage means respectively.

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

UNITED STATES PATENTS

| | | |
|---|---|---|
| 1,077,724 | Kneedler | Nov. 4, 1913 |
| 1,488,170 | Sandoz | Mar. 25, 1924 |
| 1,651,336 | Wissler | Nov. 29, 1927 |
| 2,166,423 | Clark | July 18, 1939 |
| 2,371,922 | Saito | Mar. 20, 1945 |
| 2,426,491 | Dillon | Aug. 26, 1947 |
| 2,433,484 | Roth | Dec. 30, 1947 |
| 2,570,411 | Vickers | Oct. 9, 1951 |
| 2,612,110 | Delegard | Sept. 30, 1952 |
| 2,649,739 | Hufferd et al. | Aug. 25, 1953 |
| 2,658,456 | Wahlmark | Nov. 10, 1953 |
| 2,660,123 | Vlachos | Nov. 24, 1953 |
| 2,685,842 | Hufferd | Aug. 10, 1954 |

FOREIGN PATENTS

| | | |
|---|---|---|
| 415,425 | Germany | June 19, 1925 |
| 419,421 | Great Britain | Nov. 12, 1934 |
| 988,511 | France | May 9, 1951 |