

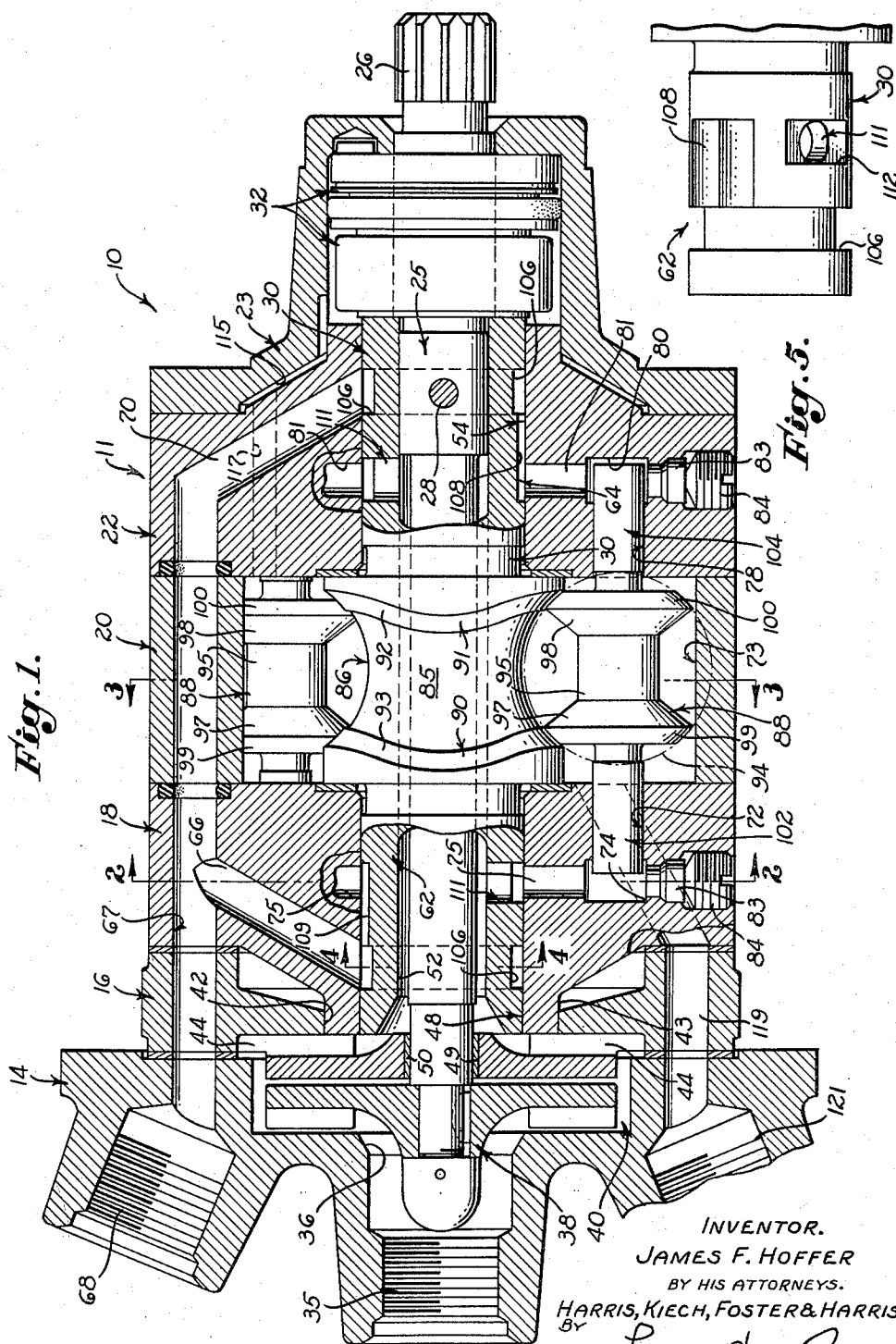
Feb. 16, 1960

J. F. HOFFER
ROTARY HYDRAULIC PUMP

2,925,047

Filed March 23, 1953

10 Sheets-Sheet 1



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10 Sheets-Sheet 3

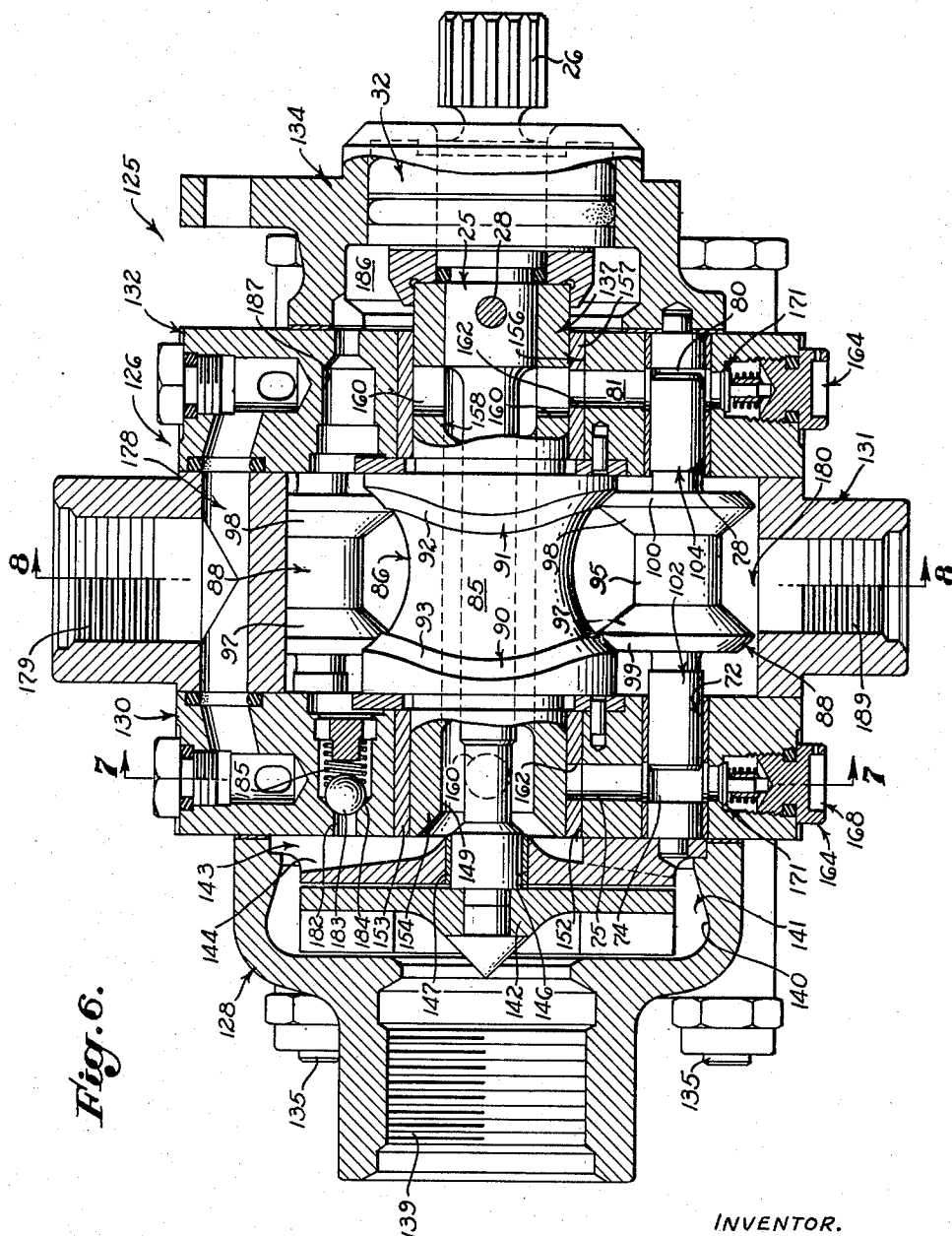


Fig. 6.

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ROTARY HYDRAULIC PUMP

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Fig. 7.

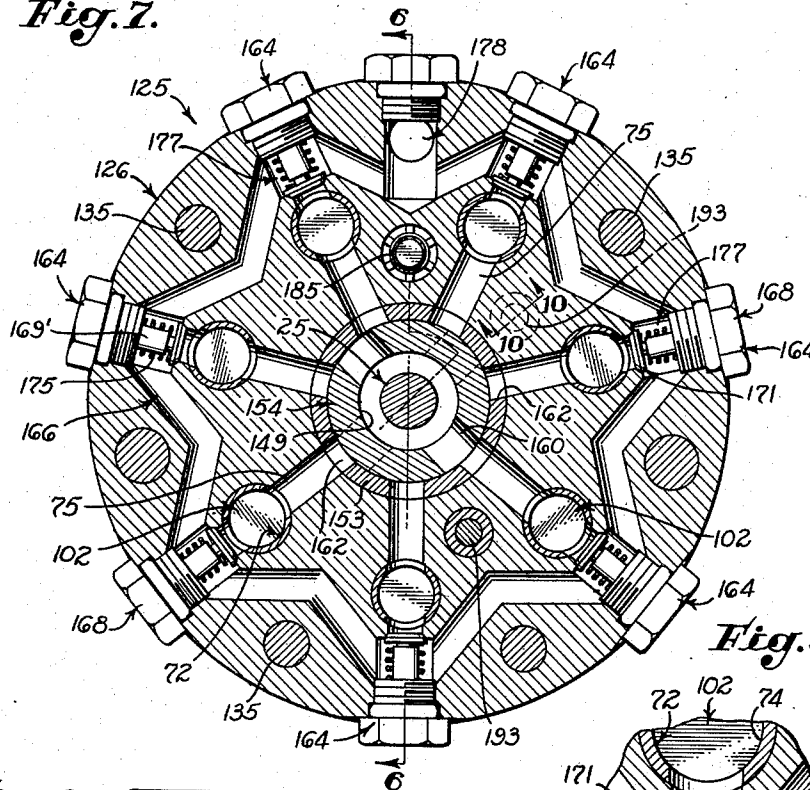


Fig. 8.

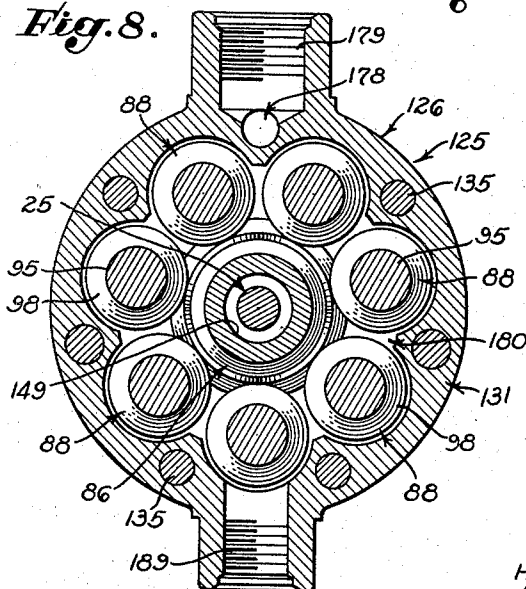


Fig. 9.

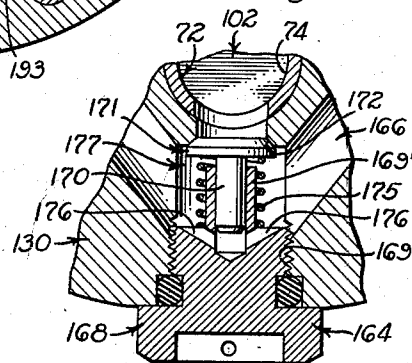
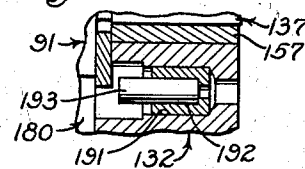


Fig. 10.



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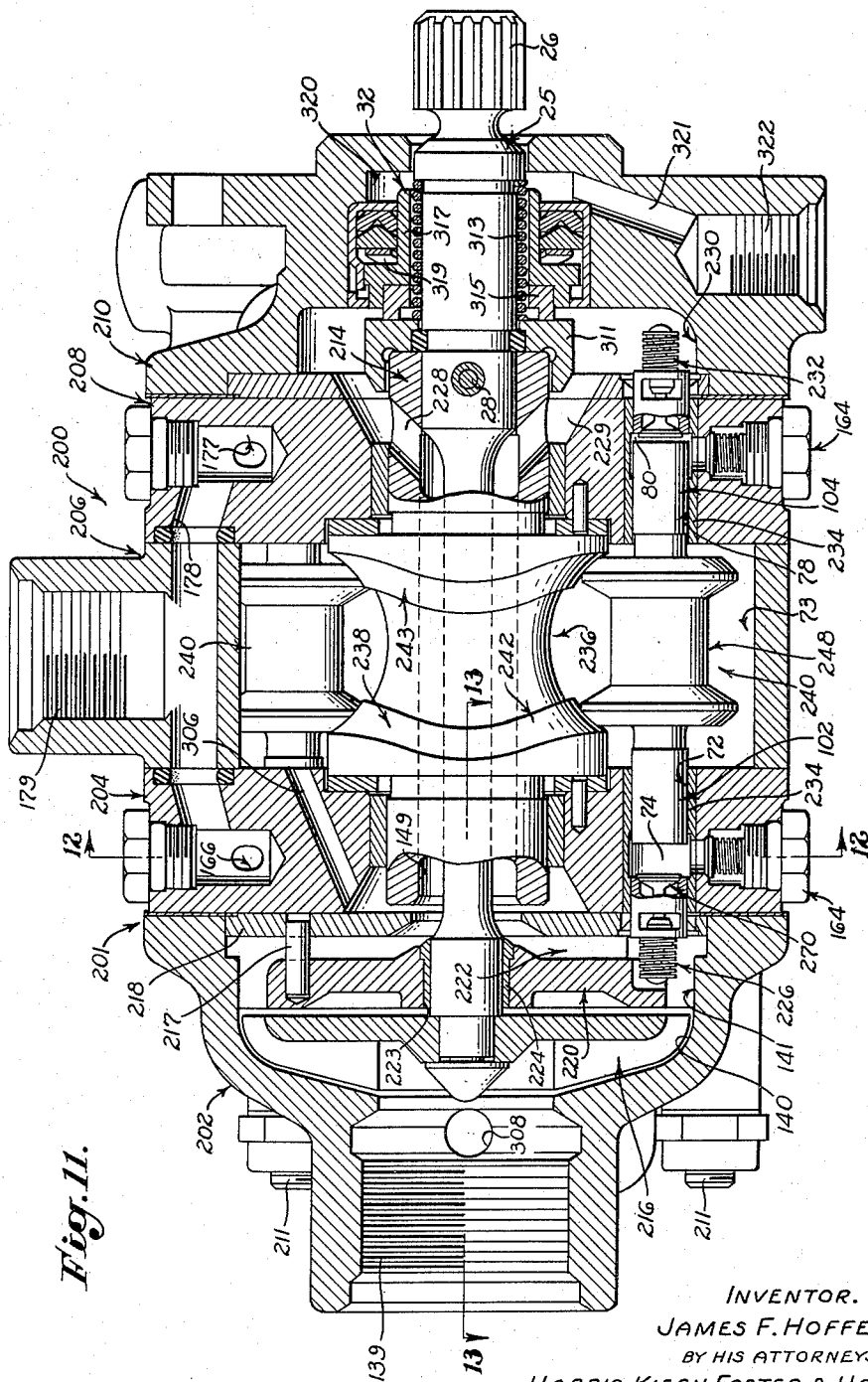


Fig. 11.

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Fig. 12.

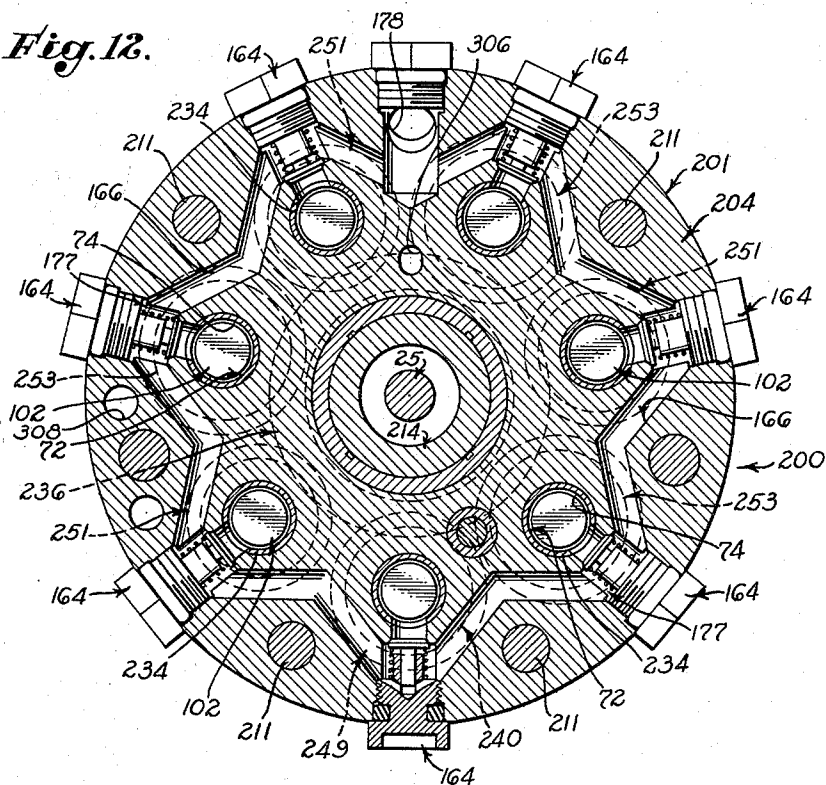
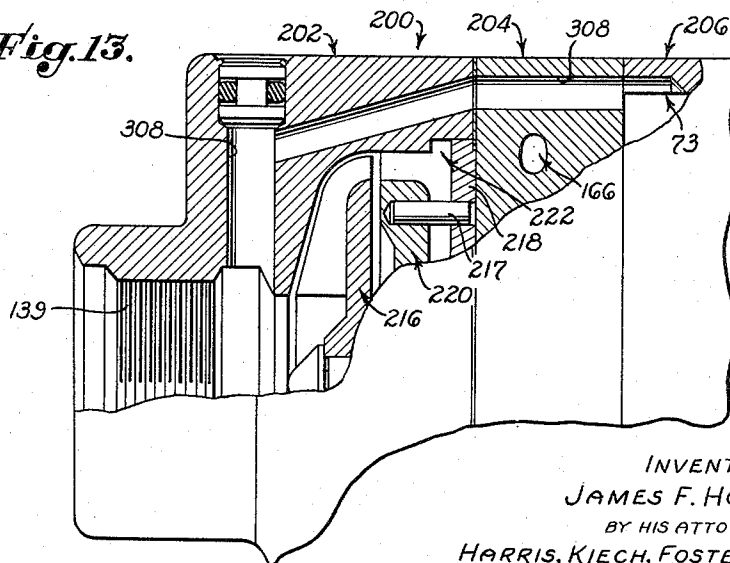


Fig. 13.



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ROTARY HYDRAULIC PUMP

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Fig. 14.

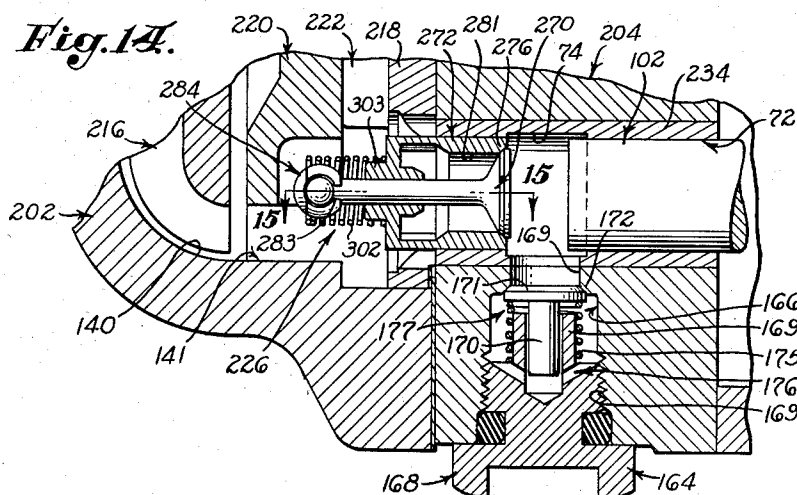


Fig. 15.

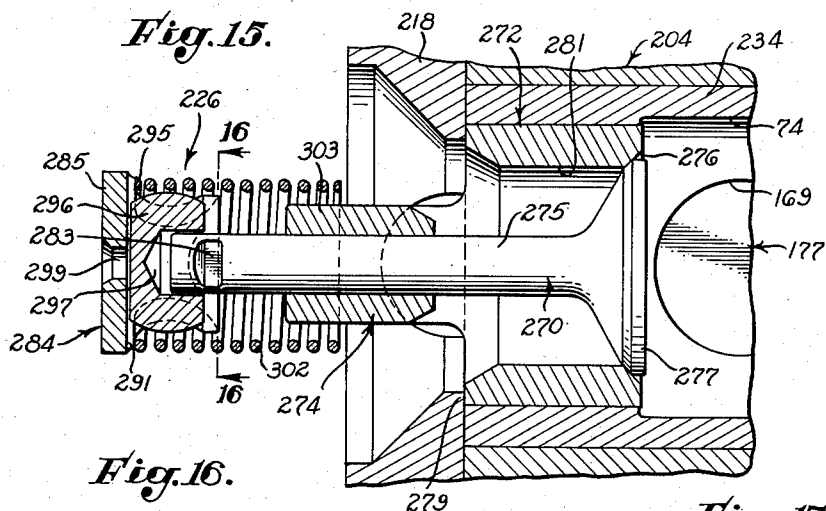


Fig. 16.

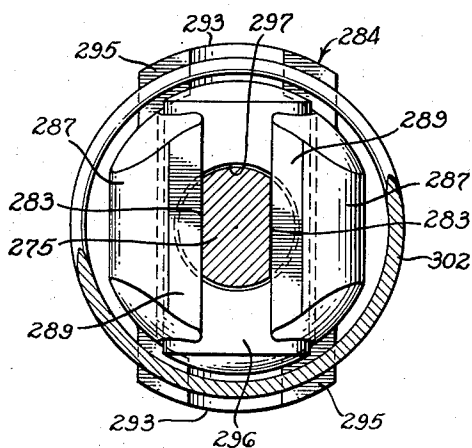
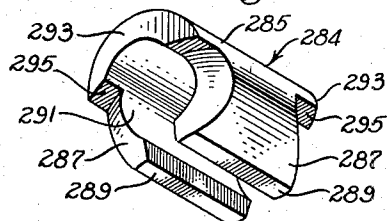


Fig. 17.



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Fig. 18.

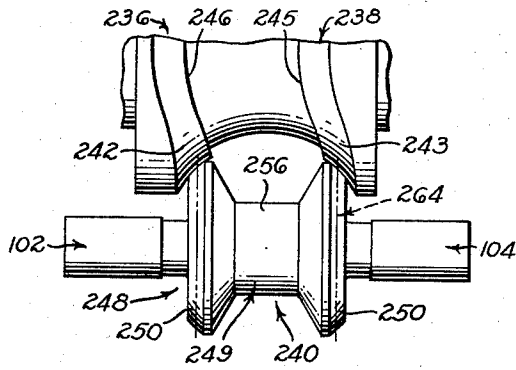


Fig. 19.

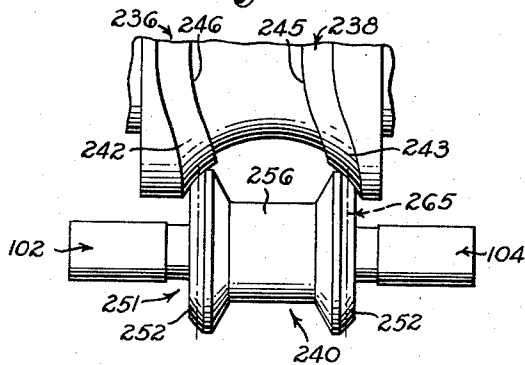


Fig. 20.

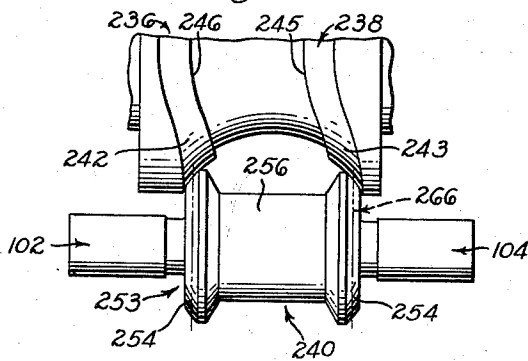
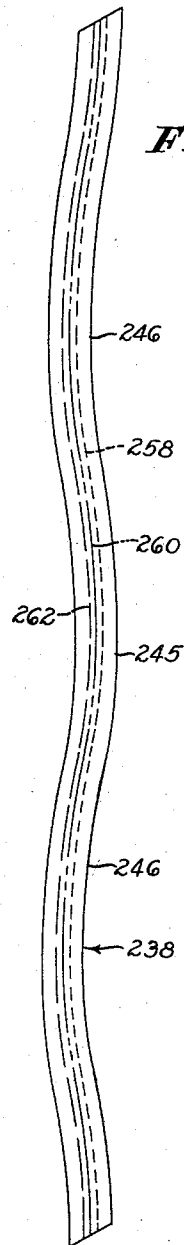


Fig. 21.



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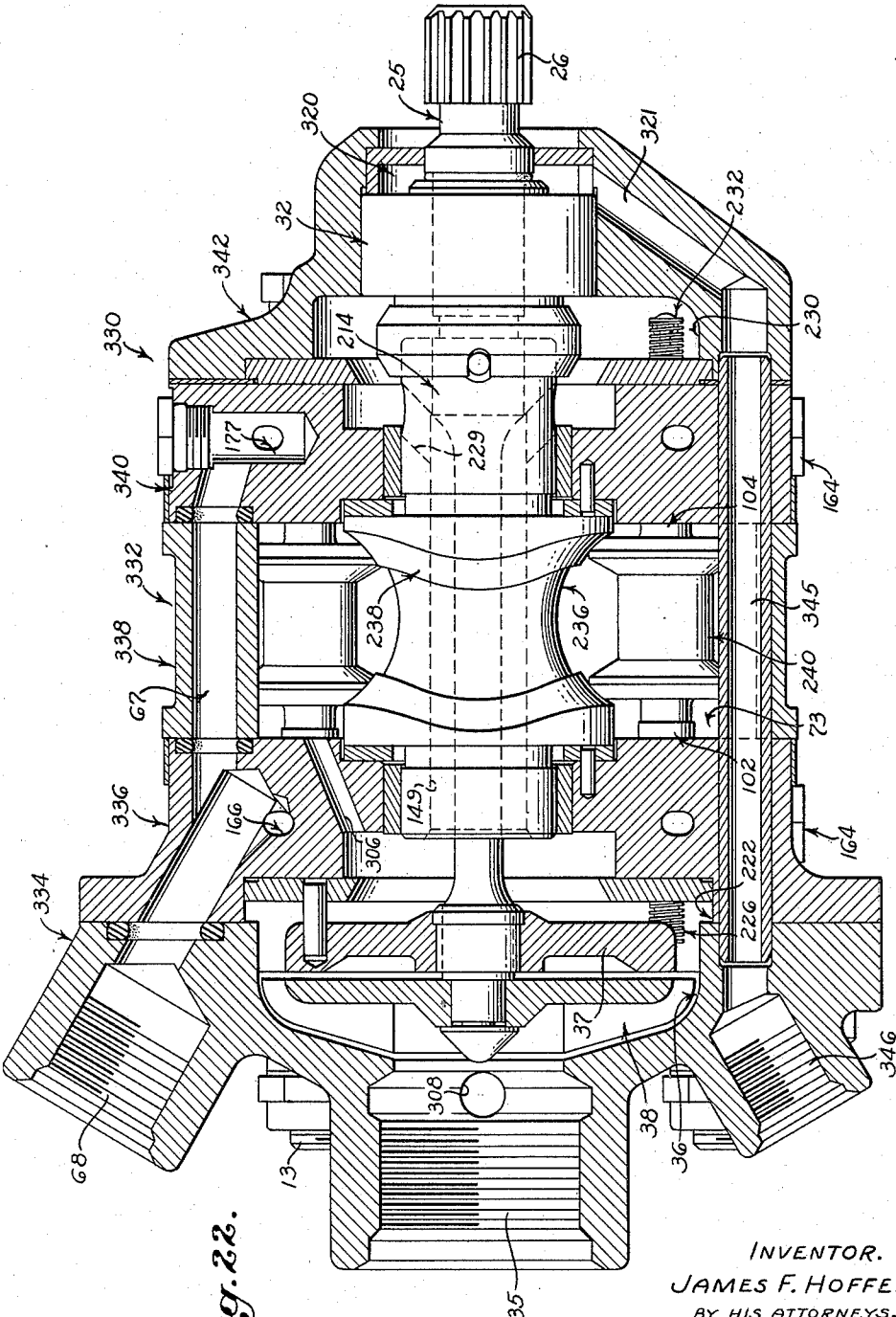


Fig. 22.

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10 Sheets-Sheet 10

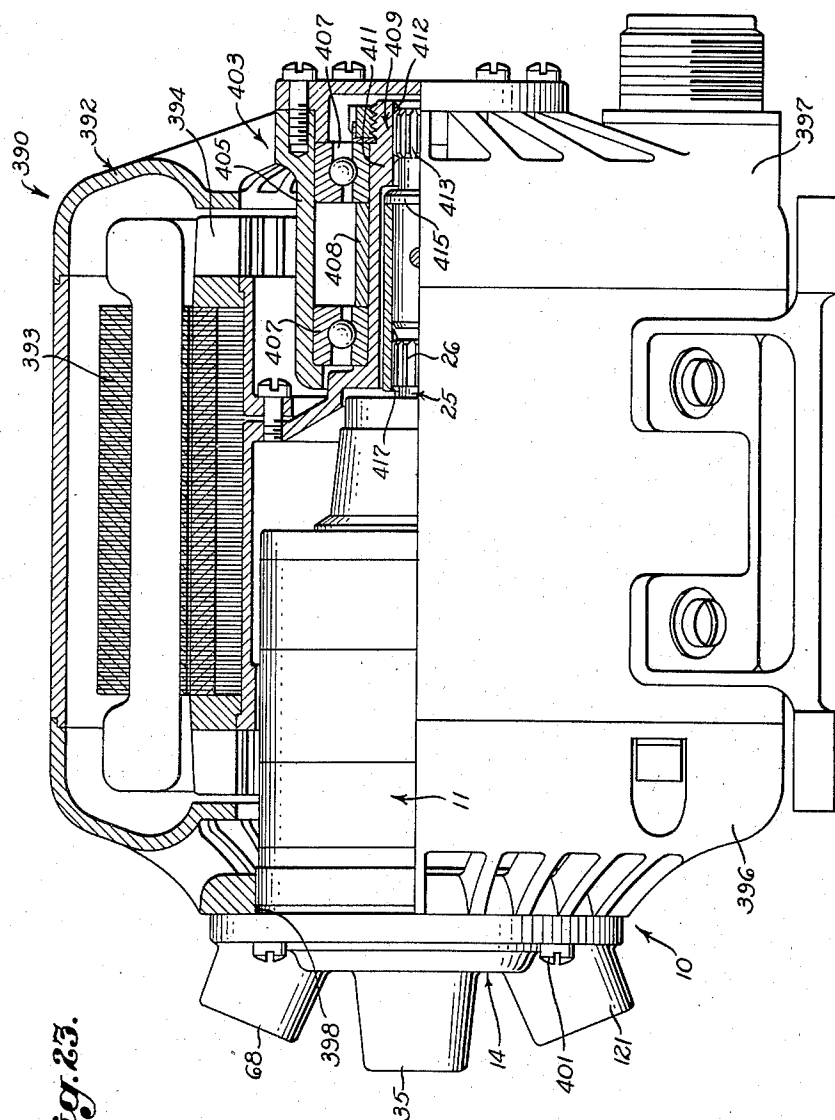


Fig. 23.

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ROTARY HYDRAULIC PUMP

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Application March 23, 1953, Serial No. 343,966

5 Claims. (Cl. 103—173)

This invention relates to hydraulic pumps and, more particularly, to a hydraulic pump of the rotary type.

Recent advances in hydraulic system designs have necessitated the provision of hydraulic fluid under high pressures for the actuation of various types of hydraulically actuable instruments and devices, necessitating the provision of a pump characterized by high pressure delivery, high speed operation, relatively large volume output and, yet, of exceedingly small size and light weight.

It is, therefore, a primary object of my invention to provide a hydraulic pump which is capable of maintaining a relatively large volume of hydraulic fluid under constant high pressure, the flow of fluid from said pump being substantially constant and being relatively free from heavy surge, pulsation, and high pressure peaks.

In order to achieve the aforementioned desirable results of large output and high pressure in a relatively small envelope, it is necessary that the component parts of the pump be energized at relatively high speed. When conventional pumps are utilized, the centrifugal force induced in the component portions thereof by such high speed results in rapid wear of the pump, necessitating the frequent dismantling thereof and the replacement of worn parts.

Another object of my invention is the provision of a high speed, large volume, high pressure pump in which the various components are so balanced that excessive bearing loads encountered in conventional pumps are eliminated, thus obviating the necessity for replacement of worn parts and for frequent dismantling of the pump.

An additional object of my invention is the provision of a high speed, high pressure pump which includes a plurality of reciprocating pistons, said pistons being actuable by a rotatable drive member associated therewith. An associated object of my invention is the provision of a pump of the aforementioned character in which the rotatable drive member is provided with a cam portion adapted to cause the reciprocation of said pistons.

A further object of my invention is the provision of a pump of the aforementioned type in which there is provided a plurality of banks of cylinders which are spaced from each other, said pump including a plurality of pumping members having provided at their opposite ends pistons adapted to be reciprocated in opposed cylinders of said banks.

Another object of my invention is the provision of a pump having a plurality of spaced banks of opposed cylinders adapted to receive oppositely directed pistons provided upon a plurality of pumping members which are actuable by a rotary cam, the rotation of said cam causing the movement of said pumping members and the reciprocation of said pistons in said cylinders.

An additional object of my invention is the provision of a pump of the aforementioned type in which said pumping members, in addition to being reciprocable to cause the reciprocation of said pistons, are rotatable about their axes by the rotation of said cam, the concomitant rotation of the cam and the pumping members

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causing the rotation of said pistons, which serve as journals for the members, to prevent the occurrence of hot spots or hot bands frequently encountered in closely fitted bearings and reducing frictional wear between the pistons and the cylinders in which they both rotate and reciprocate.

A further object of my invention is the provision of a pump of the aforementioned type in which the drive cam is so generated that four strokes of each pumping member are achieved during one revolution of the drive cam. Therefore, excessive pressure surge and pulsations are eliminated at the output of the pump since, in a pump containing seven pumping members, twenty-eight strokes per revolution occur and the pump can operate at speeds exceeding 8000 r.p.m.

An additional object of my invention is the provision of a pump of the aforementioned type in which the positive reciprocation of the pumping members in both directions of movement by the drive cam eliminates the necessity for return devices, such as springs, thus insuring the positive reciprocation of the pumping members and eliminating the possibility that the pumping members may become jammed because of the failure of the return devices.

Another object of my invention is the provision of a pump of the aforementioned type in which the reaction loads of the pumping members on the cam drive are substantially reduced by the symmetrical configuration of the cam which results in substantial dynamic balance of the cam loads axially and radially.

An additional object of my invention is the provision of a pump of the aforementioned type in which the drive member is provided with a plurality of valve means adapted to control the flow of inlet fluid to the opposed cylinders of opposite banks, the rotation of said drive member causing the concomitant rotation of said valve means.

Another object of my invention is the provision of a pump of the aforementioned type in which the fluid inlet to the cylinders is controlled by the aforesaid valve means and the discharge from each cylinder is controlled by a poppet valve, resulting in minimum inlet restriction and minimum leakage perimeter of the rotary valve.

A further object of my invention is the provision of a pump of the aforementioned type which, because of its cylinder construction and pumping member actuation, causes the cylinder pressure to be gradually elevated to equal the system pressure, thus reducing hydraulic shock and noise.

An additional object of my invention is the provision of an inlet booster adjacent the inlet passage of the pump which pressurizes the inlet fluid prior to its passage into the cylinders of the pump.

Another object of my invention is the provision of a pump of the aforementioned type in which inlet line and delivery line surge are reduced to a minimum because of the constant output characteristic of the pump and because of the double acting pumping members provided therein.

An additional object of my invention is the provision of a pump of the aforementioned character which is provided with a centrally located cam chamber, said cam chamber being adapted to receive fluid under pressure which serves to equalize the temperature between various portions of the pump.

An additional object of my invention is the provision, in a pump of the aforementioned character, of a driving member which is engageable by a plurality of pumping members, the pumping members being provided with different configurations so that they engage the driving member in different contact zones to reduce the wear on the driving member.

A further object of my invention is the provision of a pump wherein the driving member has a spherically generated drive cam surface provided thereon, said cam surface being engageable by pumping members having spherically generated cam followers provided thereupon, and the contours of the spherically generated cam followers are selected from different segments of a sphere to distribute the load and wear across said drive cam surface.

Another object of my invention is the provision of a pump of the aforementioned character wherein the valve means for each of the various cylinders includes an inlet poppet valve located at the inlet end of the cylinder and operable by a pressure differential created across said inlet end. The provision of inlet poppet valves materially reduces the unswept volume of the cylinders which results with rotary valve constructions and improves the self-priming characteristics of the pump.

A further object of my invention is the provision of a simple but dependable poppet valve structure which can be readily assembled and which insures dependable operation at high speeds.

A further object of my invention is the provision of a pump which is operable at high speeds and which can thus be directly driven from the high speed generator and other accessory pads on an aircraft engine or from a high speed electric motor, without the necessity of interposing speed reduction means.

Another object of my invention is the provision of a power unit which includes a drive motor mounted in a housing and a pump adapted to be driven thereby and located within the armature of said motor.

A further object of my invention is the provision of a power unit of the aforementioned type which is provided with a first motor housing, said first motor housing including a motor armature and having one end adapted to receive a pump housing, said pump housing being partially encompassed by said armature.

An additional object of my invention is the provision of a power unit of the aforementioned type in which a wholly enclosed drive is provided at the end of the motor housing opposite the end in which the pump is mounted, thus providing a completely enclosed power unit and eliminating the necessary for external drive connections between the pump and motor.

A further object of my invention is the provision of a power unit of the aforementioned type in which the pump component thereof is readily removable from its driven relationship with the motor.

Other objects and advantages of my invention will be apparent from the following specification and the accompanying drawings which are for the purpose of illustration only and in which:

Fig. 1 is a vertical sectional view of a pump constructed in accordance with my invention and taken on the broken line 1—1 of Fig. 2;

Fig. 2 is a transverse, partly sectional view taken on the broken line 2—2 of Fig. 1;

Fig. 3 is a transverse sectional view taken on the broken line 3—3 of Fig. 1;

Fig. 4 is an enlarged, partly sectional view taken on the broken line 4—4 of Fig. 1;

Fig. 5 is an enlarged side elevational view of a valve constructed in accordance with my invention;

Fig. 6 is a vertical sectional view illustrating another embodiment of my invention and taken on the broken line 6—6 of Fig. 7;

Fig. 7 is a transverse sectional view taken on the broken line 7—7 of Fig. 6;

Fig. 8 is a transverse, reduced, partly sectional view taken on the broken line 8—8 of Fig. 6;

Fig. 9 is an enlarged, fragmentary, partly sectional view of one of the discharge check valves incorporated in the pump of my invention;

Fig. 10 is a transverse, fragmentary, enlarged, partly

sectional view taken on the broken line 10—10 of Fig. 7;

Fig. 11 is a vertical sectional view of still another embodiment of my invention;

Fig. 12 is a transverse sectional view taken on the broken line 12—12 of Fig. 11;

Fig. 13 is an enlarged, fragmentary, sectional view taken on the line 13—13 of Fig. 11;

Fig. 14 is an enlarged, fragmentary, sectional view of a valve means of my invention;

Fig. 15 is an enlarged, fragmentary, sectional view taken on the broken line 15—15 of Fig. 14;

Fig. 16 is an enlarged, transverse, sectional view taken on the broken line 16—16 of Fig. 15;

Fig. 17 is a perspective view of a spring seat constructed in accordance with my invention;

Fig. 18 shows a spherical configuration of the cam followers on certain of the pumping members;

Fig. 19 shows a different spherical configuration of the cam followers on other of the pumping members than Fig. 18;

Fig. 20 shows still a different spherical configuration of the cam followers on other of the pumping members than Figs. 18 and 19;

Fig. 21 is a development of the drive cam showing the relation of the various follower contact zones thereupon;

Fig. 22 is a vertical sectional view of still another embodiment of my invention; and

Fig. 23 is a partly sectional, partly elevational view of a power unit constructed in accordance with my invention.

Referring to the drawings, and more particularly to Figs. 1 through 5 thereof, I show a pump 10, said pump being enclosed in a substantially cylindrical housing indicated generally at 11. The housing 11 is constituted by a plurality of sections which are maintained in operative relationship with one another by means of elongated bolts 13 which pass through all of said sections to maintain them in the aforesaid operative relationship, as best shown in Figs. 2 and 3 of the drawings.

Among the sections constituting the pump housing 11 are a combined end plate and mounting flange 14 (which, in addition to other functions to be described in greater detail below, serves to mount the pump 10), a booster plate 16 juxtaposed to said end plate, a first stator plate 18 maintained in operative relationship with said booster plate 16, a cam chamber housing 20 interposed between said first stator plate 18 and a second stator plate 22, and an end plate 23 juxtaposed to said second stator plate 22. Mounted centrally of the pump housing 11 and having its longitudinal axis substantially coincident with the longitudinal axis of said housing is a drive shaft 25, said shaft having a splined outer end 26 engageable by a driving member, in a manner which will be described in greater detail below. Keyed to the drive shaft 25 by means of a shear pin 28 is a hollow rotor 30, said rotor being formed concentrically with the longitudinal axis of the drive shaft 25 and being rotatable thereby. An oil seal combination 32 prevents the leakage of oil past the end of the drive shaft 25.

An inlet port 35 provided by the end plate 14 communicates with a supercharger or booster chamber 36, also provided in the end plate 14. A centrifugal type booster impeller 38 located in the chamber 36 is keyed to the end of the drive shaft 25 and is rotatable thereby. The provision of the booster chamber 36 and the booster impeller 38 materially enhances the efficiency of the pump 10 since it insures constant delivery of fluid under constant pressure to the pump. Furthermore, at high altitudes, where pressure drops are encountered, it prevents the starving of the pump. In addition, the pressure boost self-priming the pump and renders it more effective on dry-line starting.

It should be noted here that the booster chamber 36 constitutes a portion of a fluid inlet passage 40 of the

pump. The supercharger or booster plate 16 is provided with a centrally located opening 42, said opening being adapted to receive a cylindrical boss or projection 43 formed on the left-hand side of the first stator 18, as best shown in Fig. 1 of the drawings, to facilitate the mounting and centering of the booster plate 16. Formed in the booster plate 16 are channels 44 which constitute continuations of the inlet passage 40 and communicate with the booster chamber 36 and a first inlet chamber 48 provided in the first stator plate 18. The booster plate 16 has a centrally located opening 49 in which is located a bushing 50 which serves as a bearing for the end of the drive shaft 25 adjacent the booster impeller 38.

When the hollow rotor 30 is mounted upon the drive shaft 25, an annular, elongated inlet channel 52 exists between the periphery of the drive shaft 25 and the hollow interior of the rotor 30. This annular, elongated inlet channel 52 constitutes a prolongation of the inlet passage 40 and is adapted to receive inlet fluid from the first inlet chamber 48 as it is fed thereinto from the channels 44 in the booster plate 16. The inlet fluid flows through the elongated inlet channel 52 between the drive shaft 25 and the interior of the rotor 30 into a second inlet chamber 54.

Formed integrally with the opposite ends of the rotor 30 are first and second valve means 62 and 64 which are located and rotatable in the first and second inlet chambers 48 and 54. Therefore, when the rotor 30 is rotated by the concomitant rotation of the drive shaft 25, the first and second valve means 62 and 64 are simultaneously rotated in the first and second inlet chambers 48 and 54, respectively.

Communicating with the first inlet chamber 48 is a first outlet passage 66 which, in turn, communicates with an outlet channel 67 which terminates in an outlet port 68 provided in the end plate 14. A second outlet passage 70 has one end in communication with the second inlet chamber 54 and its opposite end in communication with the outlet channel 67. A plurality of cylinders 72, as best shown in Figs. 1 and 2 of the drawings, is provided in the first stator plate 18, said cylinders being arranged about the perimeter of a circle concentric with the longitudinal axis of the drive shaft 25 and having their right-hand ends, as best shown in Fig. 1 of the drawings, communicating with a cam and temperature and pressure equalizing chamber 73 provided in the interior of the cam chamber housing 20. Each of the cylinders 72 communicates, at its left-hand end, with a chamber 74, each of the chambers 74 being maintained in communication with the first inlet chamber 48 by means of a passage 75. The cylinders 72 constitute a first bank of cylinders to which fluid is fed from the first inlet chamber 48 by the first valve means 62, in a manner which will be described in greater detail below.

The second stator plate 22 is provided with a plurality of cylinders 78 which are also arranged about the perimeter of a circle whose center is concentric with the longitudinal axis of the drive shaft 25, each of the cylinders 78 communicating at its left-hand end with the cam chamber 73 and at its right-hand end with a cylinder chamber 80 which is maintained in communication with the second inlet chamber 54 by means of a passage 81. The cylinders 78 constitute a second bank of cylinders, inlet fluid being fed to the cylinders from the second inlet chamber 54 through the passages 81 by the second valve means 64, in a manner which will be described in greater detail below.

Each of the cylinder chambers 74 associated with the cylinders 72 of the first bank and each of the cylinder chambers 80 associated with the cylinders 78 of the second bank is provided with a steel plug 83 and retained in position by means of a set screw 84. These plugs prevent the egress of fluid from the cylinder chambers 74 and 80 and may be removed to permit the servicing

of the cylinder chambers 74 and 80 and the passages 75 and 81 associated therewith.

Formed integrally with the rotor 30 and disposed in the cam and temperature and pressure equalizing chamber 73 is a rotatable driving member 85 having a cam 86, said cam being operatively engaged with a plurality of pumping members 88 which are also disposed in the chamber 73. The rotatable cam 86 includes first and second cam portions 90 and 91, defining a groove, each of which is provided with at least two rises 92 and two falls 93, the undulated rises 92 on the second cam portion 91, as best shown in Fig. 1 of the drawings, being disposed opposite the undulated falls 93 on the first cam portion 90. It should be noted that the cam portions 90 and 91 of the rotatable cam 86 are spherically generated on a circle indicated at 94, for a purpose which will be described in greater detail below. Each of the pumping members 88 includes a substantially cylindrical head portion 95 which has integrally formed therewith, at opposite ends thereof, enlarged cam follower lobes 97 and 98 which are provided with spherically generated beveled cam follower portions 99 and 100, respectively, engaging, respectively, the spherically generated cam portions 90 and 91 provided on the rotatable cam 86.

When the drive shaft 25 is rotated to cause the concomitant rotation of the rotor 30, the cam 86 is simultaneously rotated and the cam portions 90 and 91 move against the cam follower portions 99 and 100 of the cam follower lobes 97 and 98 thus causing the concomitant rotation and reciprocation of the pumping members 88. The significance of the spherically generated engaging cam and cam follower surfaces of the rotatable cam 86 and the cam lobes 97 and 98 at once becomes apparent since the rotation of the rotatable cam 86 causes not only the reciprocation of the pumping member 88 but also the concomitant rotation thereof and thereby preventing seizure and malfunction of piston bearings 102 and 104 which can occur on such closely fitted parts due to localized heat bands. The above mentioned reciprocation and concomitant rotation of the bearing surfaces prevents the generation of localized high temperature spots or bands by dissipating the latter over large areas of the bearing surface.

It should be noted here that the provision of the rotatable drive cam 86 with at least two rises 92 and two falls 93 on each of the first and second cam portions 90 and 91 causes each of the pumping members to be reciprocated four times during a single rotation of the rotatable cam 86, thus providing for double acting pumping members and also accomplishing an effective pumping action during rotation of the drive shaft 25. Therefore, when the pump is driven at 8,000 r.p.m., 32,000 strokes per minute of each of the seven pumping members occur, resulting in a fluid output characterized by the almost total absence of pulsation and pressure surges.

Another aspect of the simultaneous rotation and reciprocation of the pumping members 88 by the rotatable drive cam 86 through the inter-engagement of the spherically generated cam and cam follower surfaces thereof is the fact that high speed rotation of the drive shaft 25 is permitted and facilitated which would not be the case if the inter-engagement between the rotatable cam 86 and the pumping members 88 were a conventional rubbing cam inter-engagement since the high speeds of the drive shaft 25 would cause not only tremendous frictional losses between the drive cam 86 and the pumping members 88, but would also cause such great wear that the cam and its associated pumping members would have a relatively short useful life. The engagement between the drive cam 86 and the cam follower portions 99 and 100 of the pumping members 88 is the substantial equivalent of a ball bearing action and is characterized by minimal wear and minimal friction. The manner in which the drive cam surfaces engage the spherically generated surfaces of the pumping members is a most important feature of my in-

vention because of the highly desirable results achieved by its utilization. Furthermore, the fact that the driving member 85 is surrounded and partially supported by the pumping members results in substantially dynamic stability and prevents the imposition of unbalanced radial loads on the shaft 25 and the rotor 30 and thus avoids the frictional losses and the wear otherwise attendant thereupon.

Each of the pumping members 88 is provided at its opposite ends with opposed pistons 102 and 104, the opposed pistons 102 and 104 being reciprocable and rotatable, respectively, in the opposed cylinders 72 and 78. Therefore, in addition to providing their ordinary functions of permitting the pumping of inlet fluid from the cylinders 72 and 78, the pistons 102 and 104 serve as bearing spindles which permit the rotation of the pumping members 88 by the rotatable cam 86. It can readily be perceived that, when the rotation of the cam 86 causes the concomitant rotation and reciprocation of the pumping members 88, the pistons 102 and 104 are simultaneously rotated and reciprocated in the opposed cylinders 72 and 78 of the first and second banks of cylinders, the exhaust stroke of one piston being accompanied by a concomitant intake stroke of its opposed piston.

The construction of the first and second valve means 62 and 64 will now be considered in greater detail, it being understood that since the valve means 62 and 64 are substantially identical in configuration but oppositely oriented with respect to one another, the first valve means 62 only will be described in detail, identical reference numerals being utilized to indicate identical portions of the second valve means 64. As best shown in Figs. 1, 2, 4, and 5, the first valve means 62 is of substantially cylindrical configuration and is provided at one end with an annular exhaust channel 106 which communicates continuously with the first outlet passage 66 and is interposed between said outlet passage 66 and a pair of horizontal exhaust bays 108 and 109 which are disposed on opposite sides of the valve 62. The bays 108 and 109 of the valves 62 and 64 are adapted to communicate, as best shown in Figs. 1, 4, and 5 of the drawings, with the passages 75 and 81 to the cylinder chambers 74 and 80, respectively, thus exhausting pressure fluid from the cylinder chambers 74 and 80 upon the pressure strokes of the pistons 102 and 104, respectively, associated therewith.

Also provided in the first valve means 62 is a transverse inlet passage 111 which, as best shown in Figs. 2 and 4 of the drawings, is bisected by the inlet channel 52 between the rotor 30 and the drive shaft 25 and has its ends communicating with inlet bays 112 and 113 on opposite sides of the valve 62 to permit inlet fluid to flow from the inlet channel, through the inlet bays 112 and 113 and into the passages 75 and 81 associated, respectively, with the cylinder chambers 74 and 80 to permit the ingress of fluid to the cylinders 72 and 78. It should be noted here that the second valve means 64, while identical in configuration with the first valve means 62, is not only oppositely oriented with respect to the first valve means 62 so that the inlet passages 111 of the two valves face each other, but is also rotated 90° about the longitudinal axis of the drive shaft 25 so that, as best viewed in Fig. 1 of the drawings, those cylinders which are directly opposed to each other, as in the case of the cylinders 72 and 78 of Fig. 1, will be opened, respectively, to communication with the inlet channel 52 and the exhaust bay 108 in order that fluid may be drawn into the cylinder chamber 74 by the intake stroke of the piston 102 and expelled from the cylinder chamber 80 by the pressure stroke of the piston 104.

Therefore, as the rotatable drive cam 86 is rotated by the rotation of the drive shaft 25, the first and second valves 62 and 64 are simultaneously rotated, the rotation of the rotatable cam 86 and the valves 62 and 64 being so coordinated that, when the pumping members 88 are

shifted to the right, as best shown in Fig. 1 of the drawings, the inlet passage 111 in the valve 62 registers with the cylinders 72 and the cylinder chambers 74 of said cylinders in which the pistons 102 are initiating or passing through their intake strokes while, on the other hand, the exhaust bays 108 and 109 of the valves 62 and 64 are registered with those cylinders in which the pistons are completing their pressure strokes so that fluid under pressure may be fed by means of the exhaust bays and the annular exhaust channel 106 into the outlet channel 67.

The manner in which the valve 62 controls the flow of inlet fluid and pressure fluid into and from the various cylinders is graphically illustrated in Fig. 2 of the drawings, those arrows which point toward the cylinders 72 and away from the longitudinal axis of the drive shaft 25 indicating the flow of inlet fluid from the inlet channel 52 to the inlet passage 111 in the valve 62 and into the cylinders 72 in which the pistons 102 are at varying positions in their intake strokes. It will be noted that the arrows which are directed away from the cylinders 72 and toward the center of the drive shaft 25 indicate the flow of fluid under pressure from the cylinders 72 and into the oppositely disposed exhaust bays 108 and 109 whence the pressure fluid flows into the annular exhaust channel 106 of the valve 62 and thence through the first outlet passage 66 and the outlet channel 67 into the outlet port 68. It is thus apparent that, by the provision of a valve of the configuration disclosed herein, and by the integration of the action of the valve 62 with the valve 64 and the provision of double acting pumping members 88, it is possible to provide a pump characterized by the maintenance of a continuous and uniform high pressure head in which discharge surge is practically eliminated.

A by-pass chamber 115 is constituted by a space between the right-end of the second stator plate 22 and the end plate 23, as best shown in Fig. 1 of the drawings. The by-pass chamber 115 is maintained in communication with the cam and temperature and pressure equalizing chamber 73 by means of a by-pass passage 117 so that pressure fluid leaking past the second valve 64 and the drive shaft 25 may be returned to the chamber 73. The chamber 73 is connected by means of a drain passage 119 to a drain port 121 in the end plate 14, said drain port being, in turn, connected to a reservoir of the fluid being pumped. Since it is intended that the pump 10 will generally be utilized in a closed system, the cam chamber 73 is continually maintained full of fluid at low pressure because of the by-passing of fluid therein from the by-pass chamber 115 through the by-pass passage 117 and also because of the leakage past the pistons from the cylinders of the pump. The pressure fluid in the cam chamber 73 serves the valuable function of equalizing the temperature between the various operating components of the pump and also serves as a lubricant and hydraulic cushion for the various operating components of the pump. The end plate 14 contains both the inlet and outlet ports 35 and 68, respectively, permitting the incorporation of the pump 10 in pump and motor combinations in a manner which will be described in greater detail below.

Although, in describing the invention as a pump, stress has been laid upon the pumping function thereof, it is obvious that the principles of my invention could be utilized in a motor of somewhat similar structure.

Another embodiment of my invention, as shown in Figs. 6-9 of the drawings, is a pump 125 including a housing 126. In describing the pump 125, those portions of the pump 125 which are identical with portions of the previously discussed embodiment will be designated by identical reference numerals for convenience in description and to prevent duplication. The housing 126 is substantially cylindrical in configuration and is constituted by a first end fitting 128, a first stator plate 130,

a cam chamber plate 131, a second stator plate 132 and a second end fitting 134. The various components of the housing are maintained in operative relationship with one another by means of bolts 135 extending there-through.

Mounted in the housing 126 and having its longitudinal axis disposed substantially centrally thereof is a drive shaft 25 which is provided with a splined outer end 26 adapted to establish a driving connection with a source of power to cause the operation and rotation of the drive shaft 25. Mounted on the drive shaft 25 and secured thereto by means of a shear pin 28 is a rotor 137, said rotor being rotatable by and concomitantly with the drive shaft 25. Disposed in the second end fitting 134 and encompassing the periphery of the drive shaft 25 is an oil seal combination 32 adapted to prevent the leakage of fluid from the pump around the periphery of the drive shaft 25.

The first end fitting 128 provides an inlet port 139 which communicates with a booster chamber 140, it being understood that the inlet and booster chambers 139 and 140, respectively, constitute portions of a fluid inlet passage 141, the other components of which will be described subsequently. Disposed in the booster chamber 140 is a booster impeller 142 which is secured to the left-hand end of the drive shaft 25, as shown in Fig. 6 of the drawings, and which is adapted to boost the pressure of the inlet fluid as it passes through the booster chamber 140. Disposed in the first end fitting 128 and supported thereby against the left-hand face of the first stator plate 130 is a booster plate 143 which defines one wall of the booster chamber 140 and is provided with a plurality of vertical inlet openings 144. The booster plate 143 has a centrally located opening 146 provided therein in which is mounted a bushing 147 for supporting the end of the drive shaft 25 adjacent the booster impeller 142.

The rotor 137 is hollow for the major portion of its length and has its interior wall spaced from the periphery of the drive shaft 25 to define an elongated inlet channel 149 which constitutes a portion of the fluid inlet passage 141 and communicates at its left-hand end, as best shown in Fig. 6 of the drawings, with the inlet openings 144 in the booster plate 143. A first inlet chamber 152 is provided in the first stator plate 130 and is lined with a sleeve or bushing 153 which encompasses a first valve means 154 formed integrally with the left-hand end of the rotor 137. A second inlet chamber 156 is provided in the second stator plate 132, said second inlet chamber being lined by a sleeve or bushing 157 which receives a second valve means 158 formed integrally with the right-hand end of the rotor 137, as best shown in Fig. 6 of the drawings.

Since the first and second valve means 154 and 158 are of identical construction with the exception that their positions at the opposite ends of the rotor 137 are reversed and the second valve means 158 is rotated 90° about the axis of the drive shaft 25, the description of the first valve means 154 will apply with equal cogency to the description of the second valve means 158, like portions of the second valve means 158 being designated by identical reference numerals utilized in designated like portions of the first valve means 154. The first valve means 154 is provided with a transverse inlet port 160 which communicates with the elongated inlet channel 149 in the rotor 137 and is adapted to transfer inlet fluid from said channel through inlet orifices 162 in the sleeve 153 which communicate with the passages 75 between the first inlet chamber 152 and the cylinder chambers 74 of the cylinders 72, thus permitting inlet fluid to flow into the cylinder chambers 74 and to be acted upon by the pistons 102 therein. Similarly, fluid flows through the transverse inlet port 160 in the second valve means 158 through the passages 81 communicating with the cylinder chambers 80.

Since the constructions of the rotatable drive cam 86

and the pumping members 88 are identical with the constructions of these components in the previously discussed embodiment of my invention, they will not be referred to in detail herein, it being understood that the rotation of the rotatable cam 86 and the first and second valve means 154 and 158, respectively, is synchronized with the reciprocation of the pumping members 88 and the pistons 102 and 104 associated therewith so that the first and second valve means 154 and 158 are appropriately disposed when the pistons 102 and 104 are reciprocated by the action of the rotatable cam 86, so that inlet fluid may be fed into the cylinder chambers associated therewith at the appropriate moment.

One of the major distinctions between the pump 125 of my invention and the previously discussed pump 10 of my invention is the manner in which pressure fluid is fed from the cylinder chambers 74 and 80, respectively, by the action of the pistons 102 and 104, respectively. Each of the cylinder chambers 74 and 80 communicates with a check valve 164, at best shown in Figs. 6, 7, and 9 of the drawings, which permits pressure fluid to be fed into a substantially circumferential outlet passage 166 when the pressure of the fluid reaches a predetermined value.

Each of the check valves 164, as best shown in Fig. 9 of the drawings, includes a plug 168 which is threadedly engaged in a bore 169 formed in the periphery of the first stator plate 130. The plug 168 provides a valve guide 169' in which is disposed the stem 170 of a poppet type valve 171 which seats on a seat 172 to prevent the flow of pressure fluid from an associated cylinder chamber 74 until the pressure of the fluid reaches a value sufficient to overcome the compression of a spring 175 encompassing the valve guide 169' and biasing the valve 171 on the seat 172. The plug 168 is provided with a plurality of drainage ports 176 adjacent the base of the valve guide 169' to permit pressure fluid seeping between the periphery of the valve stem 170 and the interior of the valve guide 169' to be drained off and prevent it from locking the stem 170 and thus preventing the movement of the poppet valve 171. The plug 168 is also provided with a transverse port 177 registering with the bore of the outlet passage 166 to permit free flow of outlet fluid therepast.

Therefore, when, as best shown in Fig. 7 of the drawings, the body of the first valve means 154 blocks the inlet passages 75 to the cylinder chambers 74 and the pistons 102 are being driven into the cylinders 72, the pressure of the fluid is elevated to a point where the poppet valves 171 are driven off their seats 172 to permit the egress of pressure fluid into the circumferential outlet passage 166. The circumferential outlet passage 166 communicates with a main outlet passage 178 which, in turn, feeds the pressure fluid into an outlet port 179 on the cam plate 131. It should be noted here that the second stator plate 132 is provided with an identical circumferential outlet passage 166 which also communicates with the main outlet passage 178. It is, therefore, apparent that the pressure fluid is fed from the various cylinder chambers of the pump 125 as the rotation of the first and second valve means 154 and 158 causes the various cylinder chambers to be sealed and permits the pressure in the cylinder chambers to rise to a value which will overcome the check valves 164. Since the discharge of pressure fluid from the cylinder chambers is controlled by check valves, minimum inlet restriction is encountered and minimum leakage about the perimeter of the first and second valve means 154 and 158 occurs.

It may be noted here that the rotary cam 86 and the pumping members 88 are disposed in a cam and pressure chamber 180 which is adapted to receive inlet fluid through a passage 182 which communicates at one end, as best shown in Fig. 6 of the drawings, with one of the openings 144 in the booster plate 143 and is thus adapted to transmit boosted inlet fluid directly to the interior of the cam and pressure chamber 180. The flow of inlet

fluid through the passage 182 is controlled by a ball check 183 which seats on a seat 184 by means of a compression spring 185. When the pressure of the fluid in the cam and temperature and pressure equalizing chamber 180 is less than the pressure of the boosted inlet fluid passing through the opening 144 in the booster plate 143, the pressure differential between the chamber 180 and the opening 144 permits the ball check 183 to be driven off its seat 184 and the passage of boosted inlet fluid into the cam and temperature and pressure equalizing chamber 180 is thus permitted. However, when the pressure of the fluid in the chamber 180 rises above the pressure of the boosted inlet fluid, the pressure differential between the fluid in the chamber 180 and the boosted inlet fluid in the opening 144 in the booster plate 143 causes the ball check 183 to be seated on the seat 184. Of course, the compression spring 185 can be so calculated that a pressure differential of a desired maximum value must be achieved between the chamber 180 and the opening 144 in the booster plate 143 before the ball check 183 can be driven off its seat 184 or, conversely, can be so calculated that a minimum pressure differential is necessary to cause the ball check to be driven off its seat.

Another source of pressure fluid for the cam and temperature and pressure equalizing chamber 180 is provided by a by-pass chamber 186 in the second end fitting 134, said by-pass chamber being connected to and maintained in communication with the chamber 180 by a by-pass passage 187. When the pressure fluid in the chamber 180 increases beyond the capacity of the chamber 180, it is drained off through a drain port 189 on the underside of the cam and fluid pressure plate 131 and continuous circulation of the fluid through the chamber is thus accomplished.

As previously indicated in the discussion of the pump 10, the presence of pressure fluid in the cam and temperature and pressure equalizing chamber 180 equalizes the temperature between the various operating components located in the chamber, acts as a hydraulic cushion for the component operating parts and also serves to drain fluid by-passed about the various operating parts to prevent leakage at the exterior of the pump 125.

Since there is a possibility that particles of metal from poorly filtered fluid may enter the chamber 180, there is provided in the second stator plate 132, as best shown in Figs. 7 and 10 of the drawings, an opening 191 which provides a mounting for a sleeve 192 in which is located a cylindrical, magnetic element 193, the forward end of which projects into the chamber 180 and is adapted to gather upon its surface particles of iron and steel which are present in the pressure fluid in the chamber 180 so that the particles will not remain in circulation in the chamber with possible damage to the mechanism.

Another embodiment of my invention, as shown in Figs. 11-21 of the drawings, is constituted by a pump 200 which includes a housing 201. In describing the pump 200, those portions thereof which are identical with portions of the previously discussed embodiments of the invention will be designated by identical reference numerals for convenience in description and to avoid duplication. It should be noted that the pump 200 is, in many details of construction, quite similar to the pump 125 shown in Figs. 6-9 of the drawings and described in detail hereinbefore.

The housing 201 is substantially cylindrical in configuration and is constituted by a first end fitting 202, a first stator plate 204, a cam chamber plate 206, a second stator plate 208, and a second end fitting 210. The various components of the housing are maintained in operative relationship with one another by means of bolts 211 extending therethrough.

Mounted in the housing 201 and having its longitudinal axis disposed substantially centrally thereof, is a drive shaft 25 which is provided with a splined outer end 26 adapted to establish a driving connection with a source

of power to cause the operation and rotation of the drive shaft 25. Mounted on the drive shaft 25 and secured thereto by means of a shear pin 28 is a rotor 214, said rotor being rotatable by and concomitantly with the drive shaft 25. Disposed in the second end fitting 208 and encompassing the periphery of the drive shaft 25 is an oil seal combination 32 for preventing major leakage of fluid from the pump around the periphery of the drive shaft 25. However, the oil seal combination 32 which has been schematically shown in Fig. 6 of the drawings is adapted to by-pass pressure fluid when the pressures encountered in the pump housing 201 reach extremely high levels, as will be described in detail hereinafter.

The first end fitting 202 provides an inlet port 139 which communicates with a booster chamber 140, it being understood that the inlet 139 and the booster chamber 140 constitute portions of a fluid inlet passage 141, the other components of which will be described below. Disposed in the booster chamber 140 is a booster impeller 216 which is secured to the left-hand end of the drive shaft 25, as shown in Fig. 11 of the drawings, and which is adapted to boost the pressure of the inlet fluid as it passes through the booster chamber 140. Disposed in the first end fitting 202, and supported therein in spaced relationship with the left-hand wall of the first stator plate 204 by means of dowels 217 which are secured in a mounting plate 218 juxtaposed to the stator plate 204, is a booster plate 220 which acts as a stator with respect to the booster impeller 216 associated therewith. The space between the booster plate 220 and the mounting plate 218 therefor constitutes a first, vertical inlet chamber 222. The booster plate 220 has a centrally located opening 223 provided therein in which is mounted a bushing 224 for supporting the end of the drive shaft 25 adjacent the booster impeller 216.

The rotor 214 is hollow for the major portion of its length and has its interior wall spaced from the periphery of the drive shaft 25 to define an elongated inlet channel 149 which constitutes a portion of the fluid inlet passage 141 and communicates, at its left-hand end, as best shown in Fig. 11 of the drawings, with the vertical inlet chamber 222 between the mounting plate 218 and the booster plate 220.

Communicating with the first inlet chamber 222 is first inlet valve means 226, said first inlet valve means being exposed to the full flow of fluid from the booster impeller 216 prior to the entry thereof into the elongated inlet channel 149 in the rotor 214. A plurality of outlet ports 228 are provided in the right-hand end of the rotor 214, as best shown in Fig. 11 of the drawings, which communicate continuously with channels 229 formed in the second stator plate 208 which, in turn, are maintained in continuous communication with a second inlet chamber 230 provided in the second end fitting 210. Therefore, the inlet fluid pressurized by passage through the booster impeller 216 flows through the first inlet chamber 222 into the elongated inlet channel 149 defined by the interior of the rotor 214 and the periphery of the drive shaft 25. From the elongated inlet channel 149 the fluid flows through the radially oriented ports 228 in the right-hand end of the rotor 214 and into the second inlet chamber 230. Disposed in fluid communication with the second inlet chamber 230 are second inlet valve means 232, said second inlet valve means 232 being constantly subjected to fluid pressure in the second inlet chamber 230 because of the constant communication of the second inlet chamber 230 with the first inlet chamber 222.

A plurality of cylinders 72, as best shown in Figs. 11 and 12 of the drawings, is provided in the first stator plate 204, said cylinders being arranged about the perimeter of a circle concentric with the longitudinal axis of the drive shaft 25 and having their right-hand ends, as best shown in Fig. 11 of the drawings, communicating

with a cam and pressure-equalizing chamber 73 provided in the interior of the cam chamber plate 206. Each of the cylinders 72 is disposable in fluid communication with the first inlet chamber 222 by means of the first inlet valve means 226, the first inlet valve means 226 controlling the flow of inlet fluid into the cylinder chambers 74 of the cylinders 72 in a manner which will be described in greater detail below.

The cylinders 72 constitute a first bank of cylinders into which inlet fluid is fed and from which pressure fluid is discharged, each of the cylinders 72 being lined with a bronze or brass bushing 234 to provide a bearing surface in the interior of each of the cylinders 72 for a purpose which will be described in greater detail below.

The second stator plate 208 is provided with a plurality of cylinders 78 which are also arranged about the perimeter of a circle whose center is concentric with the longitudinal axis of the drive shaft 25, each of the cylinders 78 communicating at its left-hand end with the cam chamber 73 and at its right-hand end being disposed in fluid communication with the second inlet chamber 230 through the second inlet valve means 232. Each of the cylinders 78 incorporates a cylinder chamber 80 and is provided with a brass or bronze bearing bushing 234. The cylinders 78 constitute a second bank of cylinders, said second bank of cylinders being disposed on the opposite side of the chamber 73 from the first bank of cylinders previously described.

Formed integrally with the rotor 214 and disposed in the cam, temperature, and pressure-equalizing chamber 73 is a rotatable driving member 236 having a spherically generated cam surface 238, said cam surface 238 being operatively engaged with a plurality of pumping members, indicated generally at 240, which are also disposed in the chamber 73.

The rotatable cam surface 238, as best shown in Figs. 11 and 18-21 of the drawings, includes first and second cam portions 242 and 243, each of which is provided with at least two rises 245 and two falls 246, the undulated rises 245 on the second cam portion 243 being disposed opposite the undulated falls 246 on the first cam portion 242. It should be noted that the first and second cam portions 242 and 243 are spherically generated in the same manner as the cam portions on the drive members of the previously discussed embodiments of the invention to accomplish the same desirable purposes previously adverted to in the description of the drive members.

Each of the pumping members 240 includes a substantially cylindrical head portion 248 and is similar in that respect to the pumping members of the previously described embodiments of my invention. However, the pumping members 240 of the embodiment of my invention presently under consideration differ in that, while the head portions thereof are provided with cam follower lobes having spherically generated cam follower surfaces provided thereupon, the cam follower lobes on adjacent pumping members are located in different spatial relationships with respect to the spherically generated cam surface 238 of the driving member 236 and are also, as will be described in greater detail below, individually generated on spheres of different diameters to accomplish certain desirable results which will be indicated hereinbelow.

As best shown in Figs. 18-20 of the drawings, a first pumping member 249 is provided with relatively closely spaced cam follower lobes 250, a second pumping member 251 is provided with cam follower lobes 252 which are more widely spaced than the cam follower lobes 250 of the pumping member 249, while a third pumping member 253 is provided with the most widely spaced cam follower lobes 254, it being noted that the spacing of the lobes on the various pumping members is accomplished by the variation in the length of the intermediate cylindrical section 256 of each of the pumping members.

Due to the different spacings of the cam follower lobes 250, 252, and 254, respectively, the surface of the drive cam 238 is contacted in three major zones; an inner zone 258, an intermediate zone 260, and an outer zone 262.

Therefore, the most closely spaced cam follower lobes 250 will have their surfaces engage the spherically generated drive cam surface 238 of the rotatable driving member 236 on the inner zone 258, the cam lobes 252 and 254 of the pumping members 251 and 253, respectively, being disposed, respectively, in engagement with the intermediate and outer contact zones 260 and 262.

Of course, due to the limited transverse area of the drive cam surface 238, the zones of contact of the various cam follower lobes overlap, to a certain extent, but the major areas of contact are distinguishable, one from the other.

By providing pumping members having cam follower lobes which engage the drive cams on the rotatable drive member 236 in different contact zones, it is possible to materially reduce the wear on the drive cam surface 238 in addition to materially equalizing any wear that may occur because all of the load on the drive cam surface 238 is not concentrated in a single zone but is equally distributed across the entire surface of the drive cam.

Furthermore, in addition to providing cam lobes which are spaced at different distances with respect to each other, I provide cam follower lobes whose spherical configuration is slightly different from the spherical configuration of the drive cam so that substantially linear contact between the cam follower lobe surfaces and the drive cam surface is achieved. This is illustrated in Figs. 18-20 by the lines 264, 265, and 266, respectively, to indicate that contact is being made between the respective cam follower lobes 250, 252, and 254 in line contact. It should be noted that the spherical deviation of the respective cam follower lobes from the spherical configuration of the drive cam surface 238 is relatively minute so that, while the major portion of the contact load is transmitted to the cam followers through a line contact, the remainder of the surfaces of the cam followers are in substantial contact with the drive cam surface 238. As previously indicated in discussing the different spacing of the cam follower lobes, the provision of cam follower lobes whose sphericity is slightly different from the sphericity of the drive cam prevents significant overlapping of the contact zones of the respective cam follower lobes and results in equalization and material reduction of wear on both the cam follower lobes and the drive cam itself.

In order to accomplish the most effective distribution of wear on the drive cam surface 238, the various pumping members are so arranged in relation to one another that, as best shown in Fig. 12 of the drawings, adjacent pumping members all have different contact zones on the drive cam surface 238. For instance, the pumping member 249 is juxtaposed to the pumping member 251 which is, in turn, juxtaposed to the pumping member 253 which is, in turn, juxtaposed to a pumping member 251. The different spacings of the cam follower lobes, the generation of slightly different sphericities on the cam follower lobes from the sphericity of the drive cam surface and the interspersing of pumping members whose lobes are differently spaced and/or generated on spheres of different diameters are most important features of my invention because they materially reduce the wear which one would expect to encounter at the tremendously high speeds of operation of the pump 200 and, also, serve to reduce the frictional heat which one would expect to encounter at such high speeds, materially enhancing and prolonging the life of the pump and improving the operational characteristics thereof.

When the drive shaft 25 is rotated to cause the concomitant rotation of the rotor 214, the driving member 236 is simultaneously rotated and the cam surface 238 thereof moves against the cam follower portions of the

pumping members 249, 251 and 253, respectively, thus imparting thereto both rotary and longitudinal movement for a purpose which will be described in greater detail below. As indicated in the consideration of the previously discussed embodiment of my invention, the spherical generation of both the drive cam surface 238 and the spherical generation of the cam follower lobes which are disposed in driven relationship therewith, provide substantially ball bearing characteristics materially reducing the frictional wear encountered and frictional heat which might normally be expected in constructions of the character under consideration.

Each of the pumping members 240 is provided at its opposite end with opposed pistons 102 and 104, said pistons being reciprocable and rotatable, respectively, in the opposed cylinders 72 and 78 and, more particularly, in the bushings 234 disposed in and lining the walls of said cylinders. Therefore, in addition to their ordinary function of pumping inlet fluid from the cylinders 72 and 78, the pistons 102 and 104 serve as bearing spindles which permit the rotation of the pumping members 240 by the rotatable drive member 236. It can readily be perceived that, when the rotation of the drive member 236 causes the concomitant rotation and reciprocation of the pumping members 240, the pistons 102 and 104 are simultaneously rotated and reciprocated in the bushings 234 lining the opposed cylinders 72 and 78 of the first and second banks of cylinders, the exhaust stroke of one piston being accompanied by a concomitant intake stroke of its opposed piston. The simultaneous rotation and reciprocation of the pistons 102 and 104 in the cylinders 72 and 78, respectively, prevent the formation of hot bands on the pistons since the rotation thereof dissipates heat generated by the movement of said pistons throughout the entire wall of each cylinder and obviates the possibility that localized over-heated areas may occur.

The construction of the first and second inlet valve means 226 and 232 will now be considered in greater detail, it being understood that since the first and second inlet valve means 226 and 232 are substantially identical in configuration, but oppositely oriented with respect to one another, the first inlet valve means 226 only will be described in detail, identical reference numerals being applied to indicate identical portions of the second inlet valve means 232. As best shown in Figs. 14 and 15 of the drawings, the first inlet valve means 226 is of the poppet valve type and includes a poppet valve 270 which is mounted and reciprocable in a combination guide and seat member 272 which is of substantially cylindrical configuration. The combination seat and guide member 272 is disposed in the left-hand end of the cylinder 72 internally of the bushing 234, as best shown in Figs. 14 and 15 of the drawings, and includes, at its extreme left-hand end, a centrally located valve guide 274 which encompasses the stem 275 of the poppet valve 270 and prevents lateral deflection of said stem within the member 272. Provided on the extreme right-hand end of the member 272 is a valve seat 276, said seat being engageable by the head 277 of the poppet valve 270 to prevent inlet of fluid into the valve chamber 74 and outlet of fluid from said valve chamber.

It will be noted that, as best shown in Fig. 15 of the drawings, an overlapping portion 279 of the mounting plate 218 engages the combined guide and seat member 272 to prevent its dislocation from the end of the bushing 234. It should also be noted that the member 272 is provided with an elongated inlet channel 281 which is maintained in constant communication with the first inlet chamber 222.

Provided adjacent the end of the stem 275 is a pair of oppositely disposed notches 283, best shown in Figs. 15 and 16 of the drawings. A spring seat member or retainer 284 is mounted on the end of the stem 275, said member including an arcuate top portion 285 which has formed integrally therewith depending arcuate side por-

tions 287, the lowermost ends of which constitute retaining fingers 289 receivable in the oppositely disposed notches 283 on the stem 275.

The top portion 285 and the depending side portions 287 of the spring seat member 284 define an elongated cavity 291 and the top portion 285 is provided with overhanging ends 293 which serve as spring seats 295. A substantially cylindrical keeper or retainer 296 is disposed in the elongated cavity 291 and is provided with a centrally located recess 297 which receives the outermost end of the valve stem 275 of the poppet valve 270. A fluid by-pass port 299 is provided in the top portion 285 of the spring seat member 284 to permit by-passing of pressure fluid entrapped in the elongated cavity 291.

Encircling the valve stem 275 and having its outermost or left-hand end, as best shown in Fig. 15 of the drawings, seated on the spring seat 295 provided by the spring seat member 284 is a compression spring 302, the opposite or right-hand end of the spring 302 being seated on a spring seat 303 provided on the combined guide and seat member 272, as best shown in Fig. 14 of the drawings. By the provision of the first and second inlet valve means 226 and 232 in the form of poppet valves, frictional wear and possible seizure which may be encountered with rotary valves of the character incorporated in the previously discussed embodiments of my invention are eliminated. Furthermore, the unswept volume of the cylinder chambers 74 and 80 is reduced to a minimum.

The compression spring 302 normally biases the head 277 of the poppet valve 270 on the seat 276 provided on the combined guide and seat member 272, thereby preventing the ingress or egress of fluid from the cylinder chamber 74 through the inlet passage 281 in the member 272. However, when the piston 102 is moved by the pumping member 240 with which it is integrally formed through its exhaust stroke into the position shown in Fig. 14 of the drawings, a negative pressure area is created in the cylinder chamber 74 across the head 277 of the poppet valve 270. This negative pressure, coupled with the positive pressure imposed upon the other side of the valve by the boosted pressure fluid in the first inlet chamber 222, overcomes the bias of the compression spring 302 permitting the head 277 of the poppet valve 270 to move out of engagement with the seat 276 and permitting the inflow of fluid into the cylinder chamber 74. When the piston 102 starts its return or pumping stroke in which it moves toward the left-hand end of the cylinder, as best shown in Fig. 14 of the drawings, the poppet valve 270 is seated on the seat 276 to prevent the egress of pressure fluid therepast into the inlet channel 281. It is readily apparent, therefore, that the operation of the first inlet valve means and, more particularly, the poppet valve 270 constituting an essential portion thereof, is directly responsive to the energization of the piston with which it is associated and is attributable both to the action of the piston and to the positive pressure exerted thereupon by the inlet fluid passing into the first inlet chamber 222 from the booster impeller 216. The provision of poppet-type valve means for the first and second inlet valve means eliminates the disadvantages of rotary-type valve means such as binding and high frictional losses and, furthermore, reduces the unswept volume of the cylinder chambers to a minimum, the unswept volume being defined as that volume of the cylinder chamber which is not traversed by the reciprocation of the piston. The unswept volume of the cylinder chambers can be materially reduced since there is no necessity for providing the elongated entry and exit ports which must necessarily be provided with rotary-type valve means as is clear from the previously discussed embodiment of the invention. Therefore, since the relatively large volume of inlet and outlet channels to and from the cylinder chamber which is found when rotary valves are utilized is eliminated by the use of direct

valving means such as the poppet valve means described above, more effective pumping action may be achieved and operation of the pump with lower leakage losses, due to positive inlet and outlet valving, may be obtained.

It should also be noted that the construction of the first and second inlet valve means by my invention is characterized by ease of assembly, minimum number of parts, and relatively long life. In the first place, the assembly of the valve structure is accomplished by the insertion of the poppet valve 270 in the member 272, the compression spring 302 then being placed on the seat 303 on the member 272. The keeper 296 is then placed over the outer end of the stem 275 of the valve 270, the end of the stem being received in the centrally located recess in said keeper, and the spring seat member 284 is then secured to the end of the stem 275 by slidable engagement of the retaining fingers 289 with the oppositely disposed notches 283 on the opposite sides of the stem 275. By means of this relatively simple and compact assembly, the spring seat member 284 is secured against inadvertent displacement from the end of the stem 275 but, nevertheless, can be easily installed or voluntarily removed when necessary.

Thus, as the driving member 236 is rotated in the cam chamber 73, the simultaneous rotation and reciprocation of the associated pumping members 240 is accomplished thereby, causing the concomitant reciprocation and rotation of the oppositely disposed pistons 102 and 104 formed integrally therewith. As indicated previously, the configuration of the drive cam surface 238 on the rotary driving member 236 causes the pumping members 240 to be reciprocated four times during each rotation of the driving member, resulting in four strokes per member or twenty-eight strokes for each revolution of the driving member 236 in a pump such as the pump 200 which is provided with seven pumping members.

It should be noted that the pump 200 is provided with outlet check valve means 164 installed in each of the cylinder chambers 74 and 80 which check valve means are substantially identical with the check valve means previously shown and discussed in the disclosure of the embodiment of the invention shown in Figs. 6, 7, and 9 of the drawings. The check valve means 164 permits pressure fluid to be fed from the individual cylinder chambers under the action of the associated pistons into substantially circumferential outlet passages 166 when the pressure of the fluid in the individual cylinder chambers reaches a predetermined value.

Of course, circumferential outlet passages 166 are located in both the first stator plate 204 and the second stator plate 208, the first outlet passage 166 in the first stator plate collecting the outlet fluid from the first bank of cylinders and the second outlet passage 166 in the second stator plate 208 collecting the outlet fluid from the second bank of cylinders.

Since the construction of each of the outlet check valves is identical, only one of the outlet check valves 164 in the first stator plate 204, as best shown in Fig. 14 of the drawings, will be described. Each of the check valves 164 includes a plug 168 which is threadedly engaged in a bore 169 formed in the first stator plate 204, the bore 169 continuing inwardly to traverse the fluid outlet passage 166 and terminate in communication with a cylinder chamber 74. The plug 168 provides a valve guide 169' in which is disposed the stem 170 of a poppet-type valve 171 which seats on a seat 172 to prevent flow of pressure fluid from its associated cylinder chamber 74 until the pressure of the fluid, under the action of the piston 102, reaches a value sufficient to overcome the compression of a spring 175 encompassing the valve guide 169' and biasing the valve 171 on the seat 172. The plug 168 is provided with a plurality of drainage ports 176 adjacent the base of the valve guide 169' to permit pressure fluid seeping between the periphery of the valve stem 170 and the interior of the valve guide 169' to be drained off

and prevented from locking the valve stem 170 and thus preventing the movement of the poppet valve 171.

The plug 168 is also provided with a transverse port 177 registering with the bore of the outlet passage 166 to permit free flow of outlet fluid therepast. Therefore, when the piston 102 initiates its pumping stroke and thus seats the inlet poppet valve 270 to prevent the egress of fluid from the cylinder chamber 74 past the poppet valve 270, the pressure of the fluid in the cylinder chamber 74 is elevated to a point where the poppet valve 171 is driven off its seat 172 to permit the egress of pressure fluid into the circumferential outlet passage 166.

The circumferential outlet passage 166 communicates with a main outlet passage 178 which, in turn, feeds the pressure fluid into an outlet port 179 in the cam plate 206. In this manner, pressure fluid is fed from both outlet passages 166 in the stator plates 204 and 208, respectively, into the opposite ends of the main outlet passage 178. Pressure fluid is thus fed from the various cylinder chambers of the pump 200 and the first and second inlet valve means 226 and 232 are closed and the outlet check valve means 164 are opened by the increase in pressure of the fluid in the various cylinder chambers. Since the inlet and discharge of fluid into and from the cylinder chambers are controlled by poppet-type valves, minimum inlet and outlet restriction is encountered and minimum leakage loss due to inadequate valving occurs.

In order that the cam chamber 73 may be continuously subjected to a constant flow of pressure fluid to accomplish both heat transfer from the rotatable driving member 236 and its associated pumping members 240, and also to lubricate the same, a pressure inlet passage 306 is provided in the first stator plate 204, as best shown in Fig. 11 of the drawings, one end of said pressure passage being maintained in communication with the pressure inlet chamber 222, while the other end thereof communicates with the cam chamber 73.

In the previously discussed embodiments of my invention, the return fluid from the cam chamber 73 has been returned to the reservoir, but in the presently discussed embodiment of my invention, as best shown in Fig. 13 of the drawings, I have provided a fluid return passage 308 which communicates directly with the fluid inlet port 139 and thus returns fluid from the chamber 73 directly to the inlet port 139. By the above described construction, continuous circulation of lubricating and cooling fluid through the cam chamber 73 is accomplished and the additional port provided in the housings of the previously discussed embodiments of my invention for the return of the fluid from the cam chamber 73 has been eliminated, thus providing an external housing with only the major inlet and outlet ports 139 and 179 and also accomplishing the elimination of a return line to the reservoir.

The aforementioned seal 32 at the right-hand of the drive shaft 25 includes a rotatable sealing member 311 which directly engages the end of the rotor 214 on the shaft 25 and which is biased into operative relationship therewith by means of a compression spring 313. The rotatable sealing member 311 is engaged by a sealing ring 315 which is backed up by a collar 317 which is, in turn, biased into engagement with the ring 315 by means of a spring washer 319. This resilient sealing construction embodied in the seal 32 is interposed between the second inlet fluid chamber 230 and a drain chamber 320 so that any fluid by-passing the seal 32 may be received in the drain chamber 320 and conducted therefrom through a drain passage 321 to a drain port 322 whence the fluid may be discharged to atmosphere or to a drain tank or, if desired, returned to the reservoir.

An alternative embodiment of my invention is shown in Fig. 22 of the drawings and is exemplified in a pump 330, the housing structure of which is somewhat similar in character to that of the pump shown in Figs.

1-4 of the drawings, but the valving structure of which is substantially identical with that of the pump 200 discussed immediately hereinabove. Therefore, in describing the construction of the pump 330, identical reference numerals will be applied to those portions of the structure which are identical with portions of the previously described embodiments of the invention.

The pump 330 includes a housing 332 of substantially cylindrical configuration, said housing being constituted by a plurality of housing sections which are maintained in operative relationship with each other by means of bolts 13. Among the sections constituting the housing 332 are a first combined mounting and end plate 334, a first stator plate 336, an intermediate cam chamber housing plate 338, a second stator plate 340, and a second end plate 342. Mounted centrally of the housing 332 and having its longitudinal axis substantially coincident with the longitudinal axis of said housing, is a drive shaft 25, said shaft having a splined end 26 engageable by a driving member, in a manner which will be described in greater detail below.

A rotor 214 identical with the previously discussed rotor of the pump 200 is mounted on the shaft 25 for rotation therewith. An oil seal 32 is provided at the right-hand end of the housing, as best shown in Fig. 22 of the drawings, to prevent leakage of oil from the interior of the pump 330.

An inlet port 35 communicates with a booster chamber 36, both the inlet port and booster chamber being provided in the first end plate 334. Disposed in the booster chamber 36 is a centrifugal-type booster impeller 38 which is keyed to the left-hand end of the drive shaft 25 for rotation thereby. A booster plate 37 cooperates with the booster impeller 38 and permits boosted inlet fluid to be fed into a first inlet fluid chamber 222 whence it is permitted to flow into an elongated inlet channel 149 provided between the interior of the rotor 214 and the periphery of the drive shaft 25.

The inlet fluid flows from the channel 149 through ports 229 at the end of the rotor 214 into a second inlet chamber 230. The pump 330 is provided with first inlet valve means 226 and second inlet valve means 232 whose construction is identical with the first and second inlet valve means of the previously disclosed pump 200 and will not be described here in detail.

A rotary drive member 236 is disposed in a cam chamber 73 and is designed to drive a plurality of pumping members 240 to cause the reciprocation of pistons 102 and 104 formed integrally with said pumping members and disposed in cylinders (not shown) arranged in first and second banks in the first and second stator plates 336 and 340, respectively, in a structure identical with that previously disclosed in the consideration of the pump 200. As in the previously disclosed pump 200, the cam chamber 73 is maintained in continual fluid communication with the inlet port 35 by means of inlet passage means 306 and outlet passage means 308. Each of the cylinders has associated therewith an outlet check valve 164 identical in construction with the previously disclosed and discussed outlet check valves and designed to perform an identical function.

The first and second stator plates 336 and 340 are provided with circumferential outlet passages 166, said outlet passages receiving pressure fluid from the cylinders, not shown, when the outlet check valves 164 are open. The pressure fluid from the outlet passages 166 is discharged into an outlet channel 67 which terminates in an outlet port 68 provided in the first end plate 334. Thus far the construction of the pump 330 is distinguishable from that of the pump 10, shown in Fig. 1 of the drawings, by the provision of poppet-type inlet and outlet valves, by the provision of means for continuous circulation of fluid through the cam chamber and its direct return to the inlet port, and by the provision of rotary drive means 236 and 240 constructed in accord-

ance with the invention disclosed in the consideration of the pump 200, that is, incorporating the varied spacings of the cam follower lobes and the different sphericities of the cam follower lobes from the sphericities of the cam surfaces 238 provided on the rotary drive member 236.

Another distinguishing feature of the construction of the pump 330 over the pump 10 lies in the provision of a drain chamber 320 which receives pressure fluid from the second inlet chamber 230 which by-passes the seal 32 and discharges said by-pass fluid into a drain port 321 which is connected with an elongated drain channel 345 running the entire length of the housing 332 and terminating in a drain port 346. The drain chamber 320 serves the same function as the drain chamber of the pump 200 and will not be discussed in detail.

There is shown in Fig. 23 of the drawings a power unit 390 constructed in accordance with my invention. The power unit 390 includes a primary or motor housing 392 in which is supported a field coil 393 which encompasses an armature 394 adapted to be rotated when said field coil is energized. The motor housing 392 includes first and second end plates 396 and 397, respectively, the first end plate 396 being provided with a centrally located opening 398 through which the housing 11 of the above described pumps 10 or 330 is insertable. The housing 11 of the pump 10 is shown as being disposable in the interior of the armature 394 while it is supported from the motor housing 392 by the first end plate 14 which is secured to the first end plate 396 of the motor housing 392 by means of bolts 401 or similar fasteners. This method of mounting the pump housing 11 within the armature 394 of the motor with which it is associated is facilitated by the location of the inlet, discharge, and drain ports 35, 68, and 121, respectively, in the first end plate 14 of the pump 10. It is obvious that by so mounting the pump 10 with its housing 11 disposed in the interior of the armature 394, an extremely compact power unit 390 is achieved.

Located in the opposite ends of the motor housing 392 adjacent the second end plate 397 is a driving connection 403, said connection including a substantially cylindrical mounting member 405 which is secured to or formed integrally with the second end plate 397. Disposed in the mounting member 405 and supported thereby are ball bearings 407 which are spaced from each other by means of a sleeve 408. A connector 409 is secured to the interior of the armature 394 adjacent the exterior of the housing 11 of the pump 10 and has a cylindrical portion 411 mounted for rotation in the bearings 407. Thus, the rotation of the armature 394 within the field coil 393 causes the concomitant rotation of the cylindrical portion 411 of the connector 409. The right-hand end of the connector 409 is provided with a female spline 412 which is adapted to receive a male spline 413 supported on the outer end of a coupling 415 which is disposed in the interior of the connector 409. The opposite end of the interior of the coupling 415 is provided with a female spline 417 which receives the splined outer end 26 of the drive shaft 25 and thus establishes a driving relationship between the armature 394 and the pump 10.

The provision of the driving connection 403 on the interior of the motor housing 392 eliminates the necessity for cumbersome and complex external driving connections between the armature 394 and the splined outer end 26 of the pump 10. As a matter of fact, the armature 394 is supported for rotation on the driving connection 403 between the armature 394 and the pump 10 and the conventional armature shaft is eliminated, the drive shaft 25 of the pump 10 being aligned with the longitudinal central axis of the armature 394. In this manner, the pump housing 11 is disposed in the waste space normally occupied by the armature shaft and the supporting structure for the armature of the motor.

Furthermore, the nature of the driving connection 403 and the manner in which the spline 26 on the end of the drive shaft 25 of the pump 10 is slidably received in the spline 417 in the end of the coupling 415 of the driving connection 403 permit the pump 10 to be removed from the motor housing 392 by the removal of the bolts 401 and the disengagement of the splined end 26 of the drive shaft 25 of the pump 10 from the splined end 417 of the coupling 415 by a sliding movement of the pump housing 11 to the left, as best shown in Fig. 23 of the drawings. Therefore, when a pump failure is encountered, or a motor failure occurs, the two units can be easily separated from each other by the mere disengagement of the bolts 401.

The extreme simplicity of the driving connection 403 of the power unit 390 and the compactness of said power unit together with the ease of separating the motor component from the pump component and the over-all relatively small envelope size of the power unit are important features of my invention.

I claim as my invention:

1. In a pump, the combination of: a housing having an axis; pumping means in said housing and spaced from said axis, said pumping means including a pumping member which has a centerline parallel to and spaced from said axis and which is reciprocable along said centerline thereof; a cam follower rigidly mounted on said pumping member and having two annular cam follower surfaces thereon which encircle said centerline and which are spaced apart therealong, said cam follower surfaces being annular bands of a single sphere having its center on said centerline; and a cam member in said housing and rotatable about said axis and having thereon two annular, undulating, cross-sectionally concave, cam surfaces respectively engaging and respectively complementary to said cam follower surfaces.

2. In a pump, the combination of: a housing having an axis; a plurality of circumferentially spaced pumping means in said housing and spaced from said axis, each of said pumping means including a pumping member which has a centerline parallel to and spaced from said axis and which is reciprocable along said centerline thereof; a plurality of cam followers rigidly mounted on said pumping members, respectively, each of said cam followers having two annular cam follower surfaces thereon which encircle the centerline of the corresponding pumping member and which are spaced apart therealong, said cam follower surfaces of each of said cam followers being annular bands of a single sphere having its center on the centerline of the corresponding pumping member; and a cam member in said

housing and rotatable about said axis, said cam member having thereon two annular, undulating, cross-sectionally concave, cam surfaces respectively engaging and respectively complementary to said cam follower surfaces on each of said cam followers.

3. A pump as set forth in claim 2 wherein each of said pumping members is provided with two axially aligned pistons respectively disposed at opposite ends thereof and respectively reciprocable in axially aligned cylinders in said housing, said housing having inlet and outlet ports and said pump including inlet and outlet valve means for establishing fluid communication between each of said cylinders and said inlet and outlet ports alternately.

4. A pump according to claim 2 wherein said cam follower surfaces on at least some of said cam followers engage said cam surfaces in different contact zones spaced apart in the direction of said axis.

5. A pump as set forth in claim 2 wherein said cam follower surfaces of at least some of said cam followers differ in sphericity so that they engage said cam surfaces in different contact zones spaced apart in the direction of said axis.

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