

April 26, 1966

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3,247,803

HYDRAULIC PUMP

Original Filed March 20, 1963

4 Sheets-Sheet 1

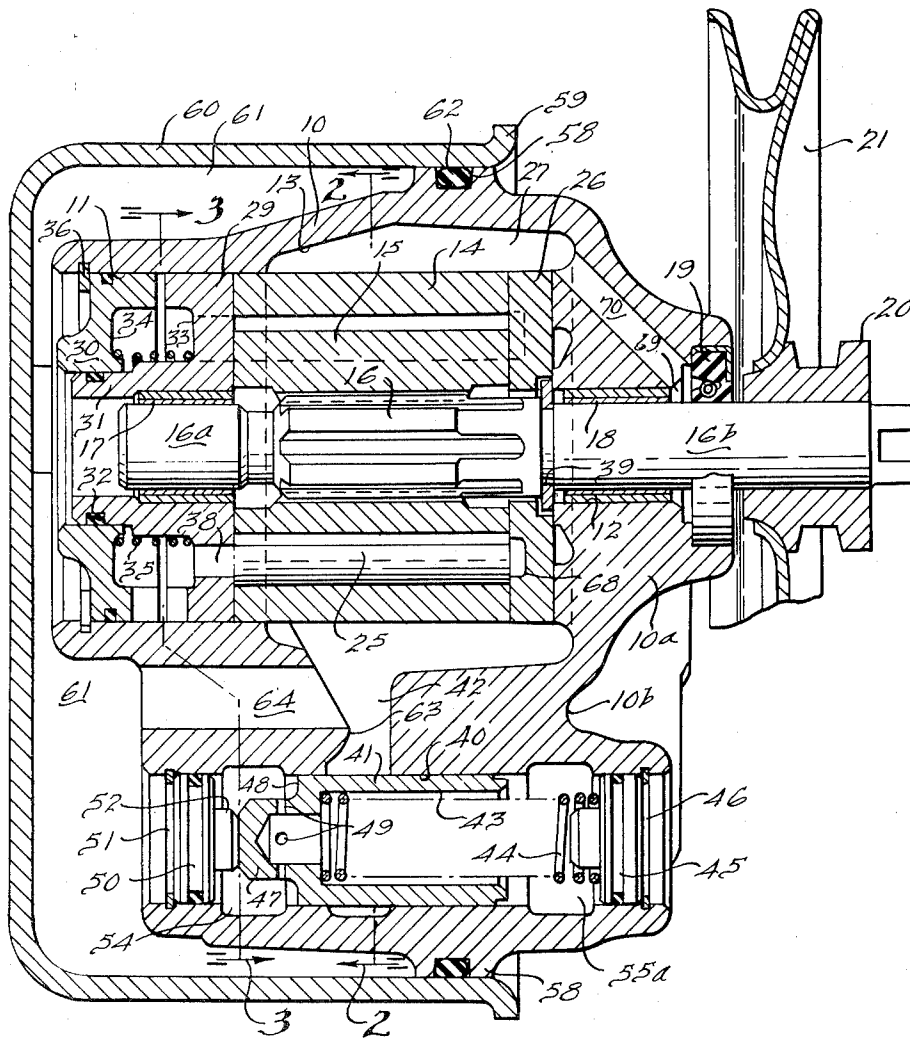


FIG. 1.

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4 Sheets-Sheet 2

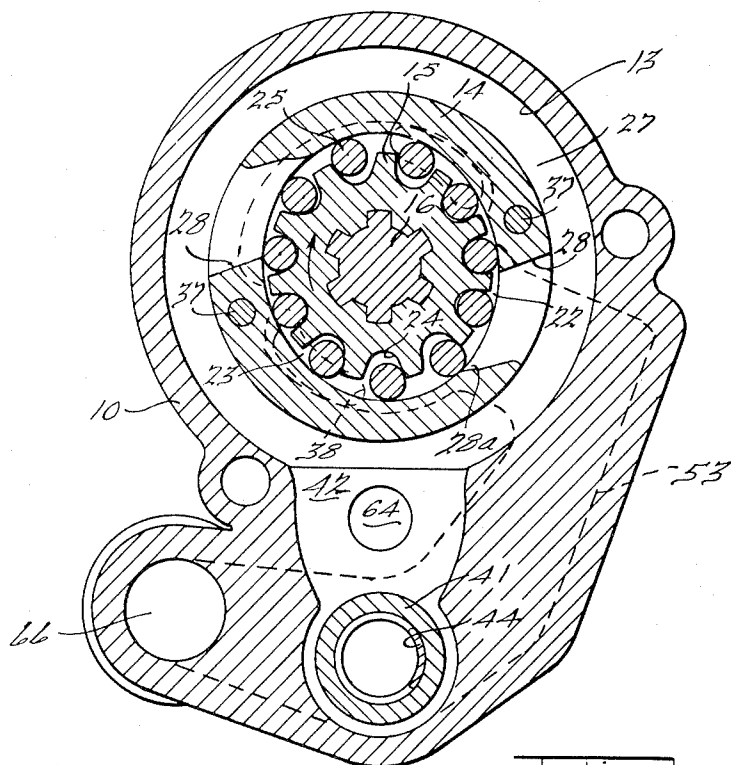


FIG. 2.

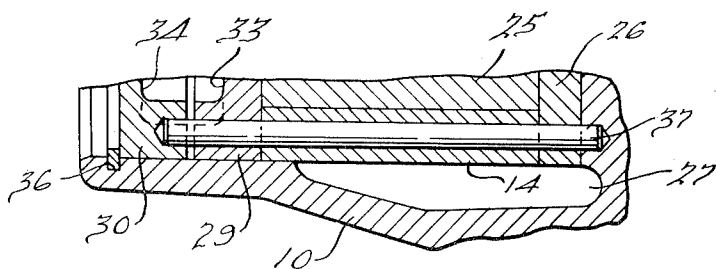


FIG. 4.

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4 Sheets-Sheet 3

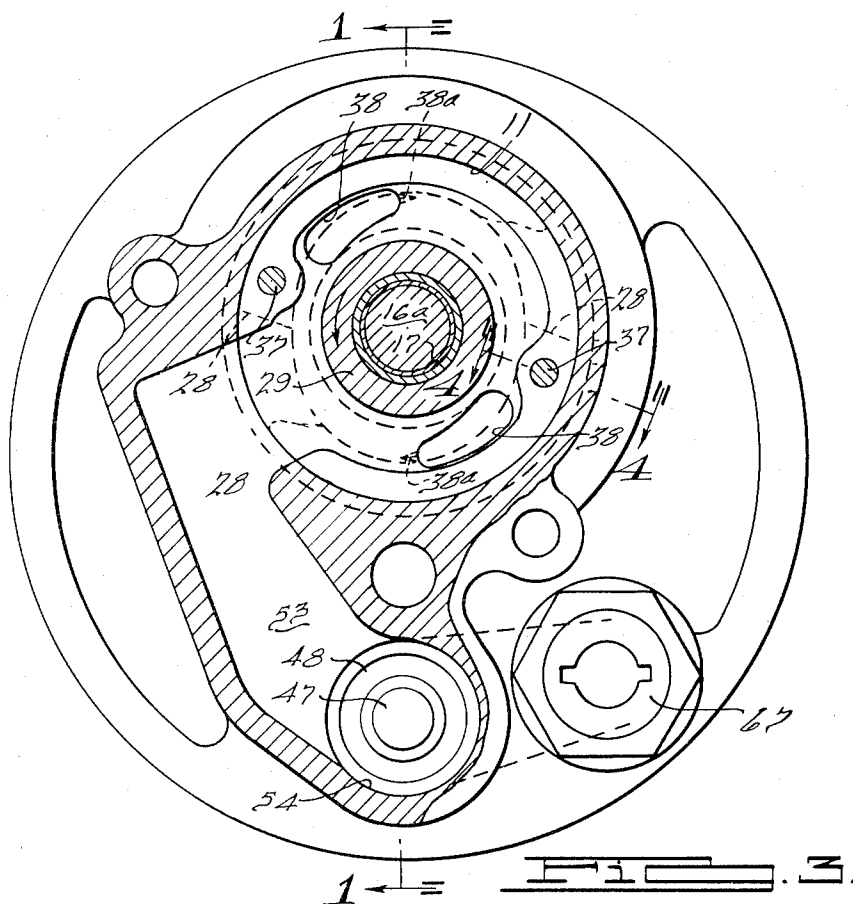
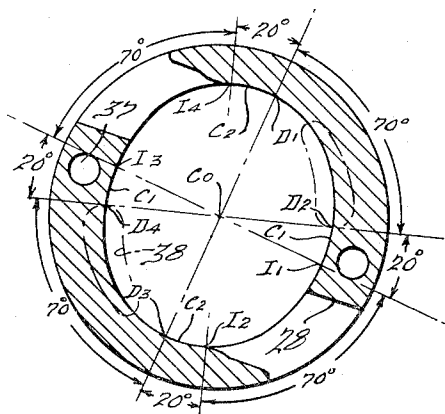


FIG. 7.

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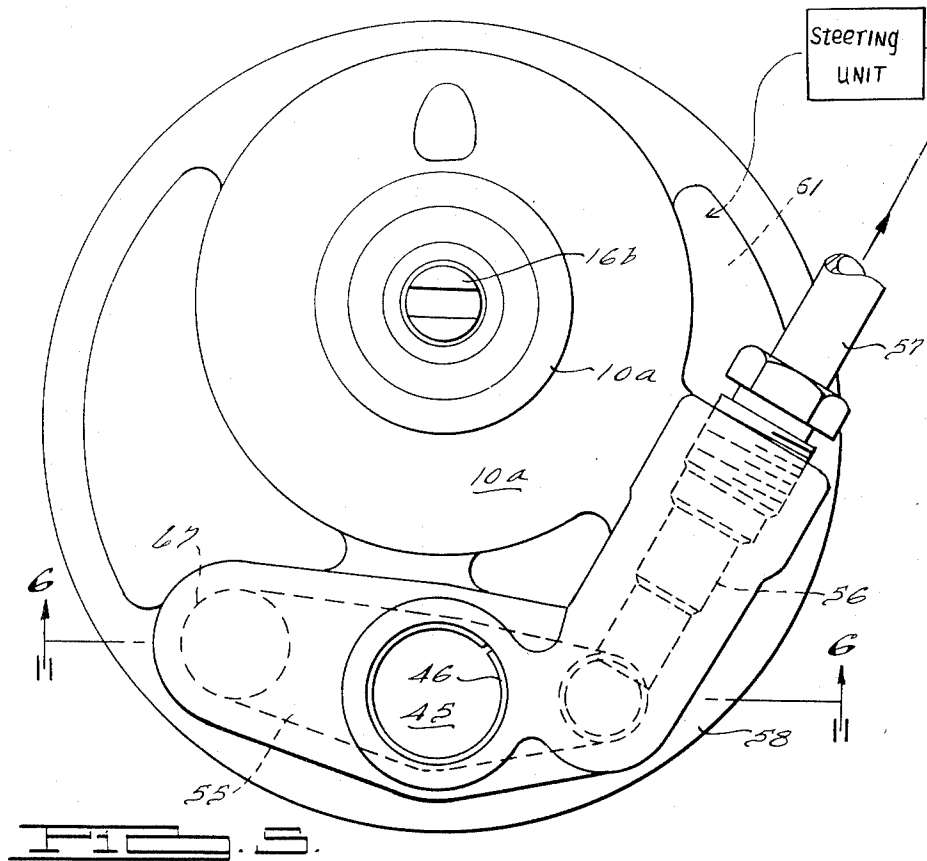
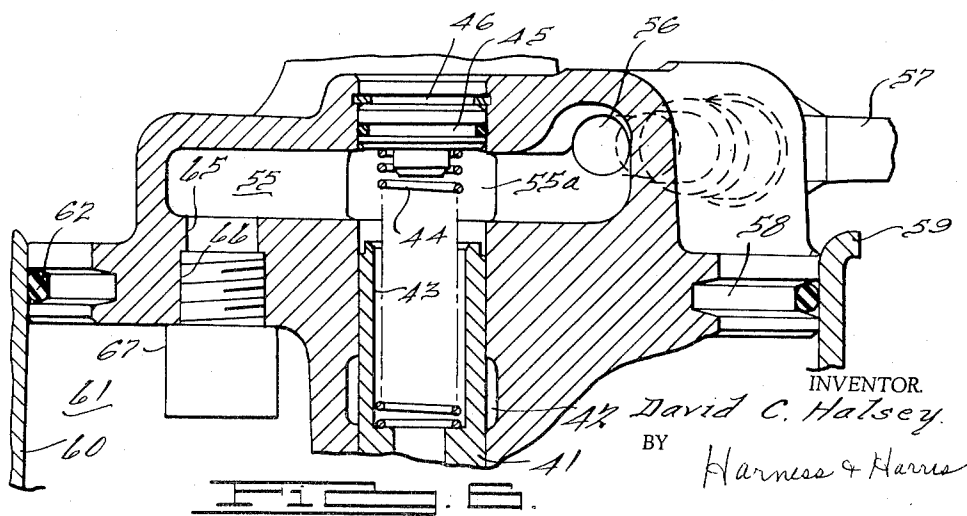
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4 Sheets-Sheet 4



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3,247,803

HYDRAULIC PUMP

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Original application Mar. 20, 1963, Ser. No. 266,569. Divided and this application June 23, 1965, Ser. No. 466,148

6 Claims. (Cl. 103—136)

This application is a division of my copending application, Serial Number 266,569, filed March 20, 1963.

This invention relates to improvements in hydraulic pumps, particularly high pressure pumps suitable for use with automobile power steering mechanisms.

An important object of the present invention is to provide an improved balanced pump of the above character comprising a housing containing a cylindrical pumping cam of out-of-round cross section. A cylindrical rotor mounted within the cam cooperates with the latter to provide a pair of diametrically spaced inlet chambers and another pair of diametrically spaced pumping chambers associated respectively with corresponding pairs of inlet and pumping arcs at the inner surface of the cam. The rotor carries a plurality of radially shiftable pumping elements such as slippers or rollers in fluid pumping and sealing engagement with the inner cylindrical surface of the cam, which latter also comprises a separate constant radius dwell or sealing arc spacing each pair of proximate ends of the pumping and inlet arcs and has a pair of inlet ports extending radially through its sidewall to connect the two inlet chambers with an annular fluid inlet header extending coaxially around the outer surface of the cylindrical cam.

By virtue of the annular inlet header enclosing the outer surface of the cam, free interchange of fluid between the inlet ports and also the supply of fluid to both inlet ports are readily accomplished regardless of constantly changing pump speeds. In consequence, means are avoided for prorating the inlet fluid to the two inlet chambers, as for example by varying the size of the radial inlet ports. The provision of the annular inlet header also renders the pump suitable for use with either an odd or even number of pumping elements without necessitating any change in the dimensions of the cam or ports or other features of the pump.

Another object is to provide such a pump wherein the inlet ports are located to assure a predetermined circulation of fluid in the inlet header, thereby to stabilize the temperature around the periphery of the cam to achieve efficient operation of the pump and reduction in noise by minimizing localized differential heating and consequent warping of the pump elements.

As the normal automobile operating speed range increases, the demand for a compact fast-acting power steering unit also increases. This demand is intensified as the road clearance and under-the-hood space for engine driven accessories decrease in accordance with the trend toward more compact and streamlined automobiles. This trend has in turn created a demand for a compact fluid pump capable of operating at high speed and at high fluid pressure for use with the power steering gear. Power steering pumps known heretofore, which have been adequate for use at comparatively low pressures, are noisy, and subject to excessive wear at the high pressures now required. Wear with consequent noise, is particularly objectionable between the pumping elements and cam. Where the curvature of the cam is too sharp at any region of the pumping are for example, the pumping element is urged against the pumping cam surface with such force that the latter is scored, with consequent noise and excessive wearing of the parts during operation under load at

high speed. Likewise during such operating conditions, where the cam curvature is too sharp at any region of the inlet arc, the pumping elements often fall away from the pumping cam surface and thereafter abruptly contact the latter with resulting noise and excessive wear.

It is accordingly another object to provide an improved vehicle power steering pump which avoids the above objections and has an improved cam contour along the pumping and inlet arcs, and in particular to provide such a cam having a contour which is characterized by a moderate curvature and gradual changes in curvature throughout its length and an exceptionally smooth transition between the pumping and inlet cam arcs and the intermediate sealing cam arcs of the constant radius, whereby a comparatively noiseless and long wearing power steering pump is provided for use under the most severe operating conditions of a modern automobile.

Other objects are to provide an improved compact arrangement in an automobile power steering pump and flow control valve for regulating the rate of flow of pressurized fluid from the pump to the power steering motor, which is particularly efficient in operation, simple in construction, and economical to manufacture and assemble.

Other objects of this invention will appear in the following description and appended claims, reference being had to the accompanying drawings forming a part of this specification wherein like reference characters designate corresponding parts in the several views.

FIGURE 1 is an axial section of a pump embodying the present invention taken in the direction of the arrows substantially along the line 1—1 of FIGURE 3.

FIGURE 2 is a reduced transverse sectional view with reservoir cover removed, taken in the direction of the arrows substantially along the line 2—2 of FIGURE 1.

FIGURE 3 is a sectional view taken in the direction of the arrows substantially along the line 3—3 of FIGURE 1.

FIGURE 4 is a fragmentary sectional view taken in the direction of the arrows substantially along the line 4—4 of FIGURE 3.

FIGURE 5 is a right end view of the pump illustrated in FIGURE 1 with the pulley removed.

FIGURE 6 is a fragmentary sectional view taken in the direction of the arrows substantially along the line 6—6 of FIGURE 5.

FIGURE 7 is a reduced transverse sectional view through the pumping cam element, illustrating the inlet and pumping arcs spaced by the constant radii dwell arcs.

It is to be understood that the invention is not limited in its application to the details of construction and arrangement of parts illustrated in the accompanying drawings, since the invention is capable of other embodiments and of being practiced or carried out in various ways. Also it is to be understood that the phraseology or terminology employed herein is for the purpose of description and not of limitation.

Referring to the drawings, an embodiment of the present invention is illustrated by way of example comprising a cast steel housing 10 having a large bore 11 opening at one end and a smaller coaxial bore 12 opening at the opposite end through a thickened portion 10a of the housing. The bores 11 and 12 open inwardly into an enlarged intermediate chamber 13 containing a hollow cylindrical cam 14 having an outer cylindrical surface closely confined at one end within bore 11. A roller 15 coaxial with the outer surface of cam 14 is contained within the hollow cam and is splined on a central rotor shaft 16 coaxial with the bores 11 and 12 so as to rotate with shaft 16 and slide axially thereon. Extending coaxially in opposite directions from the rotor 15 are bearing portions 16a and 16b of the shaft 16 journaled within bearings 17 and 18 respectively, the latter bearing being

supported within the housing portion 10a which is adapted to withstand the major side loading of the bearing 18 resulting from operation of the pump under load. The shaft extension 16b projects outwardly through a suitable seal 19 contained within housing portion 10a and is keyed to the hub 20 of a power driven pulley 21.

The interior cylindrical surface of cam 14 is out of round and cooperates with rotor 15 to provide a pair of diametrically opposed inlet chambers 22 and a pair of diametrically opposed pumping or discharge chambers 23 mutually spacing each other, FIGURE 2. The periphery of rotor 15 is provided with a plurality of cylindrical axially extending and radially outward opening notches 24 for a corresponding number of cylindrical rollers 25 of circular cross section. The present pump is designed to employ approximately eleven or twelve rollers 25 which make sliding and sealing engagement with the interior surface of cam 14 and with the trailing surface of the associated notch 24 in consequence of the fluid pressure and centrifugal and inertia forces acting thereon during operation of the pump.

The opposite axial ends of cam 14 and rotor 15 as well as the axially inner surface of housing portion 10a lie in transverse planes perpendicular to the rotor axis, the right ends of cam 14 and rotor 15 being spaced from housing portion 10a by an annular wear plate 26 having the same outside diameter as cam 14. The outer surfaces of cam 14 and wear plate 26 are spaced from the inner circumference wall of housing 10 to provide an annular inlet header 27 extending coaxially around the axis of rotor shaft 16 and communicating with the inlet chamber 22 via a corresponding pair of inlet ports 28 extending generally radially through cam 14.

The bore 11 at the left end of housing 10 is provided with a closure including an annular pressure plate 29 having a plane axial inner end surface abutting the left ends of cam 14 and rotor 15 and having an outer cylindrical surface closely fitting within bore 11 in sliding relation. The closure for bore 11 also includes an annular cap or closure member 30 seated on an annular axial extension 31 of plate 29. Suitable annular seals 32 and 32a between the juxtaposed surfaces of extension 31 and cap 30 and also between cap 30 and the inner wall bore 11 prevent axial loss of pressurized fluid between these members. The confronting surfaces of plate 29 and cap 30 are formed with mating grooves 33 and 34 which cooperate to provide an annular discharge header extending coaxially around the axis of rotor shaft 16 and communicating with the discharge chambers 23 via a corresponding pair of discharge ports 38 extending axially through plate 29. A coil spring 35 within the header 33, 34 urges plate 29 to the right against the end surfaces of cam 14 and rotor 15. Leftward movement of cap 30 is prevented by an annular snap ring 36 partially embedded within the interior surface of bore 11.

A pair of diametrically spaced locating dowels 37, FIGURES 2 and 4 extend slidably through plate 29, cam 14 and wear plate 26 and are confined at their axially opposite ends in housing portion 10a and cap closure 30. Thus fluid pressure within discharge header 33, 34 urges members 29, 14 and 26 closely together and toward the inner wall of housing portion 10a to prevent fluid leakage between these parts with a force proportional to the pump discharge pressure, which force is also proportional to the tendency for fluid leakage. At the same time, thermally induced dimensional changes in these parts, particularly between cam 14 and rotor 15, is readily accommodated. Furthermore, the axial dimension of rotor 15 and cam 14 may feasibly be almost identical to achieve optimum pump efficiency by minimizing a bypass flow of fluid around the ends of the rotor 15 during operation of the pump. Also in accordance with the construction shown and as explained more fully below, axially directed forces on the

cam 14 and rotor shaft 16 are substantially in balance at all times. Small transient unbalanced forces are assumed by the annular thrust bearing 39 having its inner circumference closely confined within a mating groove in shaft 16 and having an outer portion confined between plate 26 and housing portion 10a.

Extending radially from bore 11 and header 27 is an extension 10b of the housing 10 which contains a bypass bore 40 parallel to the coaxial bores 11 and 12. An axially slidable single-land spool valve 41 within bore 40 normally closes a supply conduit 42 extending radially inwardly from a central portion of bore 40 into the inlet head 27. The valve 41 has a hollow interior 43 opening to the right in FIGURE 1 and containing a coil spring 44 under compression between a closure plug 45 and the left end of the interior 43. The closure 45 seals the right end of bore 40, which extends completely through housing extension 10b, and is retained against rightward movement by snap ring 46 embedded within the interior wall of bore 40. The left end of valve 41 is reduced at 47 to provide a radial shoulder 48 and contains four radial metering ports 49 opening into the interior 43 of valve 41. The left end of bore 40 is closed by a closure plug 50 retained against leftward movement by snap ring 51 partially embedded within the inner wall of bore 40. The plug also provides an integral axially inwardly directed stop 52 which abuts valve 41 to limit its leftward movement.

Pressurized fluid from discharge header 33, 34 is conducted by a generally radially extending duct 53 in housing 10, FIGURE 3, to an enlargement 54 of bore 40 at the left end of valve 41. Pressurized fluid from conduit 53 enters the left end of bore 40 at the enlargement 54, then flows through metering orifices 49 at a substantially constant rate of approximately two gallons per minute into the interior 43 then to an enlargement 55a at the right end of bore 40, from whence the metered fluid is conducted via transverse duct 55 to duct 56 within housing 10 and an exterior conduit 57 to the vehicle power steering unit, FIGS. 5 and 6.

Extending around the circumference of the housing 10 is an annular groove and support 58 on which is supported the flanged end 59 of a cup shaped container 60. The latter encloses the left end of housing 10 and cooperates therewith to form a reservoir 61 into which the fluid from the power steering unit may be returned, FIG. 5. Fluid leakage between the support 58 and the juxtaposed side walls of the container 60 is prevented by an O-ring 62.

In order to supercharge the flow of fluid into the inlet header 27 via supply conduit 42, the latter is provided with a venturi restriction 63 at the opening of a duct 64 in communication with the reservoir 61. The bypass bore enlargement 55a and transverse duct 55 are also connected by means of a pressure relief conduit 65 in housing portion 10b with an axially extending threaded conduit 66 opening into the reservoir 61 and normally closed by a pressure relief valve 67 which may be of conventional construction.

In operation of the structure described, upon the application of power to pulley 21, rotor 15 is rotated in the direction of the arrow, FIGURE 2, to draw fluid into the inlet chamber 22 from header 27 via the radial inlet ports 28 by operation of the rollers 25. The fluid drawn into the inlet chambers 22 is carried clockwise to the discharge chambers 23 and is then forced into the discharge header 33, 34 via the axial discharge ports 38. In order to balance the pump hydraulically, axially inwardly opening notches 68 shaped identically with the discharge ports 38 are formed in the surface of the wear plate 26. Small tapered precompression notches 28a and 38a are provided at the trailing edges of the inlet ports 28 and in the leading edges of the discharge ports 38 to accommodate inertial effects of the fluid and to minimize cavitation and noise during pump operation in accordance with customary practice. Also by the construction de-

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scribed, leakage from discharge header 33, 34 returns to inlet header 27. Endwise leakage at bearing 17 returns to the reservoir 61. Endwise leakage at bearing 18 is collected in annular chamber 69 upstream of seal 19 and is returned via duct 70 to inlet header 27.

As the discharge pressure increases at high pump load, as for example during a parking maneuver when the pump is normally operating at low speed, the resulting high pressure in discharge header 33, 34 urges pressure plate 29 more firmly against the juxtaposed ends of cam 14 and rotor 15 to compact these members more tightly between plates 26 and 29 than when the pump is under light load, as for example during straight ahead steering when the pump is usually operating at high speed. The high pressure fluid discharged from header 33, 34 is conducted via conduit 53 to the inlet or upstream end 54 of bypass bore 40, whereby fluid at the rate of approximately two gallons per minute is supplied to the power steering unit via metering ports 49 as described above. The pressure of the pump discharge in excess of the two gallons per minute flow will act on the left end of valve 41 to move the latter rightward against the tension of spring 44 until shoulder 48 establishes communication between supply conduit 42 and the upstream inlet enlargement 54. The pressure at which valve 41 opens to bypass the pump discharge into supply conduit 42 will of course be determined by the tension of spring 44. In the event that the pump discharge pressure exceeds a predetermined limit downstream of ports 29, pressure relief valve 67 will open to discharge directly from duct 55 into reservoir 61.

During normal operation of the pump, the bypass flow from enlargement 54 into supply conduit 42 will result in a reduced pressure at the venturi 63 whereby the flow of fluid from reservoir 61 through duct 64 and into inlet header 27 will be accelerated to supercharge the flow to the inlet ports 28. In this regard it is to be noted that the opening of duct 64 into supply conduit 42 is adjacent the juncture of the latter with inlet header 27 to assure optimum supercharging. Also the two diametrically opposed inlet ports 28 are arranged asymmetrically with respect to the supply conduit 42, the port 28 which is located on the clockwise side of duct 42, FIGURE 2, being more remote than the diametrically opposed inlet port 28, so that these two ports may both be of the same size and yet receive the same amounts of inlet fluid. The length of the path of the inlet flow in header 27 to the inlet port 28 in the same clockwise direction as the rotation of rotor 15 is determined with respect to the length of the path of the corresponding inlet flow in the opposite direction to the other inlet port 28 so that the amount of flow to each inlet port 28 from supply conduit 42 is approximately balanced. The ports 28 are located however so that a portion of the inlet flow into the port 28 at the counterclockwise side of supply conduit 42 will enter in a clockwise direction from the far side of the inlet header 27. The amount of such flow will vary in accordance with the speed of rotation of the pump, but will always be adequate to prevent stagnation of fluid within the far side of the inlet header 27.

In consequence of the arrangement described, the fluid flow within header 27 will serve as a coolant to minimize localized overheating of the pump and assure a more uniform thermal expansion and contraction of its parts. In addition, a portion of the header 27 connecting the inlet ports 28 at the side of the header opposite the supply conduit 42 assures fluid supply to both ports 28 and enables the use of the same cam element 14 and inlet ports 28 with a rotor having either eleven or twelve pumping elements 25.

Referring to FIGURE 7, the contour of the interior cam surface of element 14 is illustrated where points I_1 and I_2 designate the beginning and end of a fluid inlet arc I_1I_2 along which the rollers 25 rise or move radially outwardly a total radial distance H from the circumference of a circle

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of small radius C_1 to the circumference of a concentric circle of larger radius C_2 . The arc I_3I_4 represents a second inlet arc, whereas the arcs D_1D_2 and D_3D_4 represent first and second pumping arcs along which the rollers 25 fall the radial distance H , the arcs D_2I_1 and D_4I_3 represent dwell arcs of constant radius C_1 , and the arcs I_4D_1 and I_2D_3 represent dwell arcs of constant radius C_2 . As the rollers 25 rise along each inlet arc I_1I_2 and I_3I_4 upon clockwise rotation of the rotor 15, fluid is drawn through the associated inlet port 28 into the inlet chambers 22 comprising the spaces between the rotor 15 and each inlet arc. The fluid is then carried clockwise along the length of the dwell arcs I_2D_3 and I_4D_1 and, thereafter as the rollers fall along each pumping arc D_1D_2 and D_3D_4 , is forced outwardly through the discharge ports 38 from the pumping chambers 23 comprising the spaces between the rotor 15 and each pumping arc.

In order to minimize noise and wearing in the high-speed high-pressure pump, the inlet and pumping arcs must blend smoothly at their opposite ends with the curves of constant radius C_1 and C_2 . Also the curvature of the cam 14 must satisfy the conditions that the rollers 25 will follow the cam surface closely without changing direction too abruptly at any point therealong, so as to avoid excessive wearing of the parts and noise during high-speed high-pressure operation of the pump. Thus the rollers 25 must not fall away from the inlet arc portions of increasing radius and thereafter strike the cam contour with excessive force, nor should the rollers be urged with excessive force against the pumping arc portions of decreasing radius.

Any curve of finite length, such as profile of one of the inlet or pumping cam arcs, can be approximated by the polynomial:

$$(1) \quad y = a_0 + a_1x + a_2x^2 + a_3x^3 + \dots a_nx^n$$

where y is the ordinate and x is the abscissa in the cartesian coordinate system.

The corresponding curve in polar coordinates suitable for representing the cam contour about the center C_0 of the circles C_1 and C_2 is:

$$(2) \quad R = b_0 + b_1A + b_2A^2 + b_3A^3 + \dots b_nA^n$$

where for the sake of convenience the unit of radial measurement is H , the unit of angular measurement is the angle from the center C_0 subtended by the inlet or pumping cam arc involved, R is the radial measurement from the circumference of inner circle C_1 to the circumference of outer circle C_2 , and A is the angular measurement about the center C_0 in the direction of decreasing R . Thus for the pumping arcs, A is measured from the point D_1 to the point D_2 , and from the point D_3 to the point D_4 . For the inlet cam arcs, A is measured from the point I_4 to the point I_3 , and from the point I_2 to the point I_1 . It is also apparent that the radial velocity of a roller with respect to its angular displacement as it moves along the cam arc is represented by the differential of R with respect to A , that is $R' = dR/dA$. Similarly radial acceleration $R'' = d^2R/dA^2$; and the rate of change of acceleration with respect to changes in A , referred to herein as "jerk," is $R''' = d^3R/dA^3$.

A concept of the present invention has been to provide inlet and pumping cam contours such that their terminals blend so smoothly with the circles C_1 and C_2 that as a roller 25 leaves the constant radius C_2 or rides onto the constant radius C_1 , the radial velocity R' in each case will be zero to avoid sudden stress and resultant wear and noise. Similarly inertial and vibratory forces resulting from acceleration R'' and the rate of change of acceleration or jerk R''' are avoided at the terminal points of the inlet or pumping cam arcs if both R'' and R''' are also zero. We accordingly have as terminal conditions:

When $A=0$; $R=1$, and $R'=R''=R'''=0$.

When $A=1$; $R=R'=R''=R'''=0$.

These eight conditions enable determination of the eight constants in the equation:

$$(3) R = b_0 + b_1A + b_2A^2 + b_3A^3 + b_4A^4 + b_5A^5 + b_6A^6 + b_7A^7$$

Additional terms can be added by extending the terminal conditions to include $R''''=0$, etc., but the additional terms have negligible effect on the operating efficiency of a power steering pump of the character disclosed at the terminals of the inlet and pumping cam arcs and necessarily increase the maximum radial acceleration R'' of rollers riding on the cam surface at some point between the terminals. Thus the additional terms are objectionable because, in order to minimize wearing between the rollers and cam surface, the minimum force therebetween resulting from radial acceleration is desired. This is especially true with a balanced pump of the type shown having two inlet ports and two discharge ports, where the rise or fall H must be accomplished in a comparatively small arcuate segment. On the other hand, less than the eight terms shown results in objectionable noise and wear at the terminals during high speed operation, because if acceleration R'' and jerk R''' are not zero at the terminals, the resulting sudden change in radial velocity and acceleration will cause infinite jerk at the terminals. By reducing jerk R''' to zero, the transitions between the inlet and pumping cam contours and the portions of constant radius C_1 and C_2 are accomplished with optimum smoothness without unduly increasing the maximum acceleration R'' at any intermediate portion of the cam contours.

It is apparent from the terminal condition $A=0$ when $R=1$, FIGURE 7, that $b_0=1$ in Equation 3. It is also obvious that $b_n=0$ for values of $n=1, 2$ or 3 , because, for the aforesaid values of n , $d^nR/dA^n=0$ when $A=0$. Thus $b_1=b_2=b_3=0$ and Equation 3 reduces to:

$$(4) R = 1 + b_4A^4 + b_5A^5 + b_6A^6 + b_7A^7$$

From the remaining four terminal condition

$$R=R'=R''=R'''=0$$

when $A=1$, the four coefficients b_4, b_5, b_6 , and b_7 can be readily determined by simultaneous equations as follows: $b_4=-35, b_5=84, b_6=-70$, and $b_7=20$.

Equation 4 becomes:

$$(5) R = 1 - 35A^4 + 84A^5 - 70A^6 + 20A^7$$

Equation 5 determines either a pump or an inlet cam arc wherein, not only is an exceptionally smooth transition with the seal arcs of constant radius obtained, but also a moderate curvature characterized by gradual changes throughout the length of the cam arc is obtained enables production of a highly efficient, quietly operating, and long wearing power steering pump.

By virtue of the units of A and R specified above, each value of R along the curve of one of the pumping or inlet cam arcs will determine one and only one corresponding value for A . For a pump of the type shown, it has been found adequate to calculate the value of R at each twenty minutes of arc. Inasmuch as the total angular measurement from D_1 to D_2 in FIGURE 7 is 70° , each $20'$ of arc (one-third of a degree) represents $(20^\circ/3)$ $(1/70^\circ) = 2/21$ of the unit angular measurement employed in Equation 5. The greater the angular measurement of the pumping and inlet arcs, the less will be the acceleration forces on the rollers. The smallest feasible inlet arc has been found to be approximately $58'$ for a pump of the type shown where the distance H is approximately 10% of the radius of circle C_2 . This limit is determined by the magnitude of the rise H , the operating pressure and speed of the pump, and the related factor—the centrifugal force urging the rollers 25 into sealing engagement with the inner surface of cam 14. If the inlet arc is so small that the roller cannot follow the cam surface, the sealing engagement is lost and noise and pump inefficiency will result. A somewhat smaller pumping arc can be employed, the limit again being determined by the

fall H and by the operating speed and pressure of the pump.

The maximum value of R in Equation 5 is the unit distance H between the inner and outer circles C_1 and C_2 . The value of R corresponding to each assigned value of A can readily be converted to any other unit of measurement by multiplying the value obtained from Equation 5 by the total value of the distance H in the other unit, as for example the value of H measured in inches. In the present instance, the contour of the pumping cam arc D_3D_4 is identical with the contour of arc D_1D_2 , whereas the contours of the inlet cam arcs I_1I_2 and I_3I_4 are mirror images of arc D_1D_2 .

Having thus described my invention, I claim:

1. In a fluid pump, a housing, a rotor rotatable within said housing, a cam element extending around said rotor and cooperating therewith to provide two pumping chambers and two inlet chambers mutually spacing each other circumferentially, said cam element comprising a pair of pumping cam arcs associated with said pumping chambers respectively, a pair of inlet cam arcs associated with said inlet chambers respectively, and a separate constant radius sealing cam arc connecting the ends of each inlet cam arc with the proximate ends of the next adjacent pumping cam arcs, said pair of inlet cam arcs and said pair of pumping cam arcs mutually spacing each other circumferentially, inlet and outlet ports for supplying fluid to said inlet chambers and for discharging fluid from said pumping chambers respectively, a plurality of pumping elements carried by said rotor and cooperable with said cam element to pump said fluid from said inlet chambers to said pumping chambers, each inlet and pumping cam arc being defined by a seventh power polynomial

$$R = b_0 + b_1A + b_2A^2 + b_3A^3 + \dots + b_7A^7$$

R and A being radial and angular measurements respectively and $b_0, b_1, b_2, b_3 \dots b_7$ being constants such that the first, second, and third derivatives of R with respect to A are each equal to zero at the terminals of each inlet and pumping cam arc, the radius at the beginning and ending of each inlet and pumping cam arc being equal to the radius of the adjacent sealing cam arc.

2. In a fluid pump, a housing, a rotor rotatable about an axis within said housing, a cam element extending around said rotor and cooperating therewith to provide two pumping chambers and two inlet chambers mutually spacing each other circumferentially, said cam element comprising a pair of pumping cam arcs of variable radius associated with said pumping chambers respectively, a pair of inlet cam arcs of variable radius associated with said inlet chambers respectively, and a separate constant radius sealing cam arc connecting the ends of each variable radius inlet cam arc with the proximate ends of the next adjacent variable radius pumping cam arcs, said pair of variable radius inlet cam arcs and said pair of variable radius pumping cam arcs mutually spacing each other circumferentially, the constant radius of the sealing cam arc at the beginning of each variable radius pumping cam arc being greater than the constant radius of the sealing cam arc at the end of that pumping cam arc, inlet and outlet ports for supplying fluid to said inlet chambers and for discharging fluid from said pumping chambers respectively, a plurality of pumping elements carried by said rotor and cooperable with said cam element to pump said fluid from said inlet chambers to said pumping chambers, each variable radius cam arc being defined by the polynomial $R = 1 - 35A^4 + 84A^5 - 70A^6 + 20A^7$, R being the radial measurement from the smaller radius sealing cam arc to the larger radius sealing cam arc at the ends of the associated variable radius cam arc, the unit of radial measurement being the radial difference between said sealing cam arcs at said ends of the associated variable ratio cam arc, A being the angular measurement about said axis in the direction from the large radius end to the small radius end of the associated variable radius cam arc, and the unit

of angular measurement being the angle from said axis subtended by the associated variable radius cam arc.

3. In a fluid pump, a housing, a rotor rotatable within said housing, a cam element extending around said rotor and cooperating therewith to provide an inlet chamber and a discharge chamber, said cam element having an inlet cam arc and a discharge cam arc associated with said inlet chamber and discharge chamber respectively, inlet and discharge ports for supplying fluid to said inlet chamber and for discharging fluid from said discharge chamber respectively, a plurality of pumping elements carried by said rotor in sliding engagement with said cam element to pump said fluid from said inlet chamber to said discharge chamber, the radius of said inlet cam arc progressively increasing in the direction of rotor rotation and being defined by the seventh power polynomial

$$R=b_0+b_1A+b_2A^2+b_3A^3+\dots b_7A^7$$

wherein R and A are radial and angular measurements respectively, and $b_0, b_1, b_2, \dots, b_7$ are constants such that the first, second, and third derivatives of R with respect to A are each equal to zero at both ends of said inlet cam arc.

4. In a fluid pump, a housing, a rotor rotatable within said housing, a cam element extending around said rotor and cooperating therewith to provide an inlet chamber and a discharge chamber, said cam element having an inlet cam arc and a discharge cam arc associated with said inlet chamber and discharge chamber respectively, inlet and discharge ports for supplying fluid to said inlet chamber and for discharging fluid from said discharge chamber respectively, a plurality of pumping elements carried by said rotor in sliding engagement with said cam element to pump said fluid from said inlet chamber to said discharge chamber, the radius of one of said cam arcs progressively changing in one direction only from one end thereof to its other end and being defined by the seventh power polynomial $R=b_0+b_1A+b_2A^2+b_3A^3+\dots b_7A^7$, wherein R and A are radial and angular measurements respectively and $b_0, b_1, b_2, \dots, b_7$ are constants such that the first, second, and third derivatives of R with respect to A are each equal to zero at both said ends of said one cam arc.

5. In a fluid pump, a housing, a rotor rotatable about an axis within said housing, a cam element extending around said rotor and cooperating therewith to provide an inlet chamber and a discharge chamber, said cam element having an inlet cam arc and a discharge cam arc associated with said inlet chamber and discharge chamber respectively, inlet and discharge ports for supplying fluid

to said inlet chamber and for discharging fluid from said discharge chamber respectively, a plurality of pumping elements carried by said rotor in sliding engagement with said cam element to pump said fluid from said inlet chamber to said discharge chamber, the radius of said inlet cam arc increasing from its beginning to its end and being defined by the polynomial

$$R=1-35A^4+84A^5-70A^6+20A^7$$

wherein R is the radial measurement starting from zero at the small radius end and measuring outwardly toward the large radius end of said inlet cam arc, the unit of radial measurement is the radial difference between said ends, A is the angular measurement about said axis in the direction from the large radius end to the small radius end of said inlet cam arc, and the unit of angular measurement is the angle from said axis subtended by said inlet arc.

6. In a fluid pump, a housing, a rotor rotatable about an axis within said housing, a cam element extending around said rotor and cooperating therewith to provide an inlet chamber and a discharge chamber, said cam element having an inlet cam arc and a discharge cam arc associated with said inlet chamber and discharge chamber respectively, inlet and discharge ports for supplying fluid to said inlet chamber and for discharging fluid from said discharge chamber respectively, a plurality of pumping elements carried by said rotor in sliding engagement with said cam element to pump said fluid from said inlet chamber to said discharge chamber, the radius of at least one of said cam arcs increasing progressively from one end to the other and being defined by the polynomial

$$R=1-35A^4+84A^5-70A^6+20A^7$$

wherein R is the radial measurement starting from zero at the small radius end and measuring outwardly toward the large radius end of said one cam arc, the unit of radial measurement is the radial difference between said ends, A is the angular measurement about said axis in the direction from the large radius end to the small radius end of said one cam arc, and the unit of angular measurement is the angle from said axis subtended by said one arc.

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