

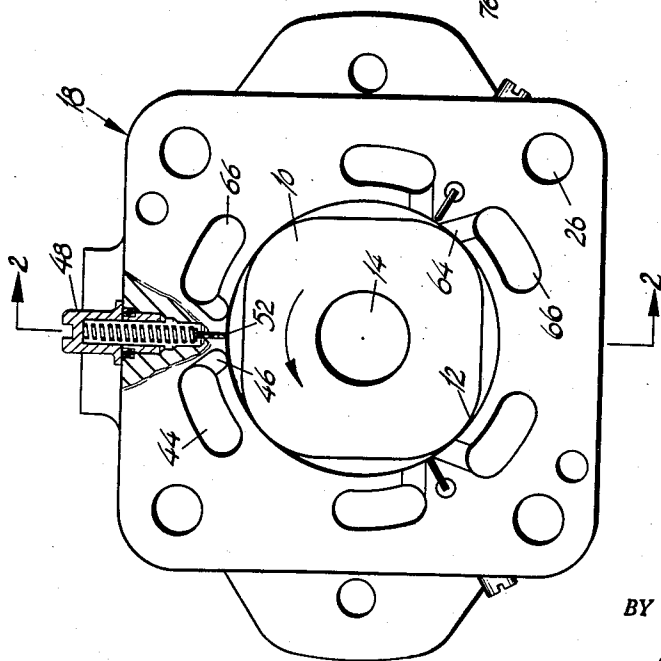
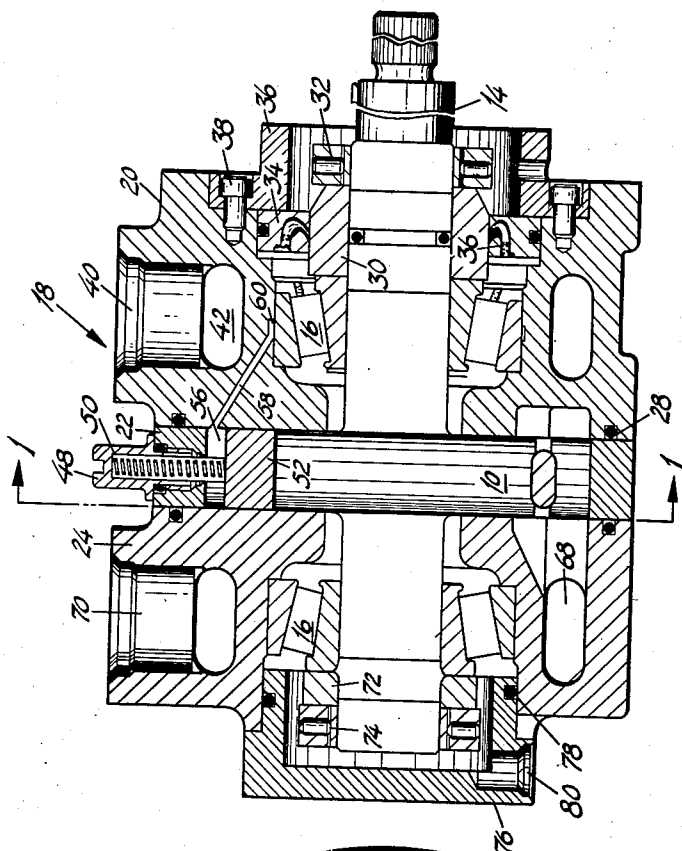
Aug. 5, 1958

J. R. FARRON ET AL
CAM PUMP

2,845,872

Filed Sept. 16, 1953

8 Sheets-Sheet 1



INVENTOR.
JOHN R. FARRON
BY CHIEN-BOR SUNG

Edmund R. Roston
ATTORNEY

Aug. 5, 1958

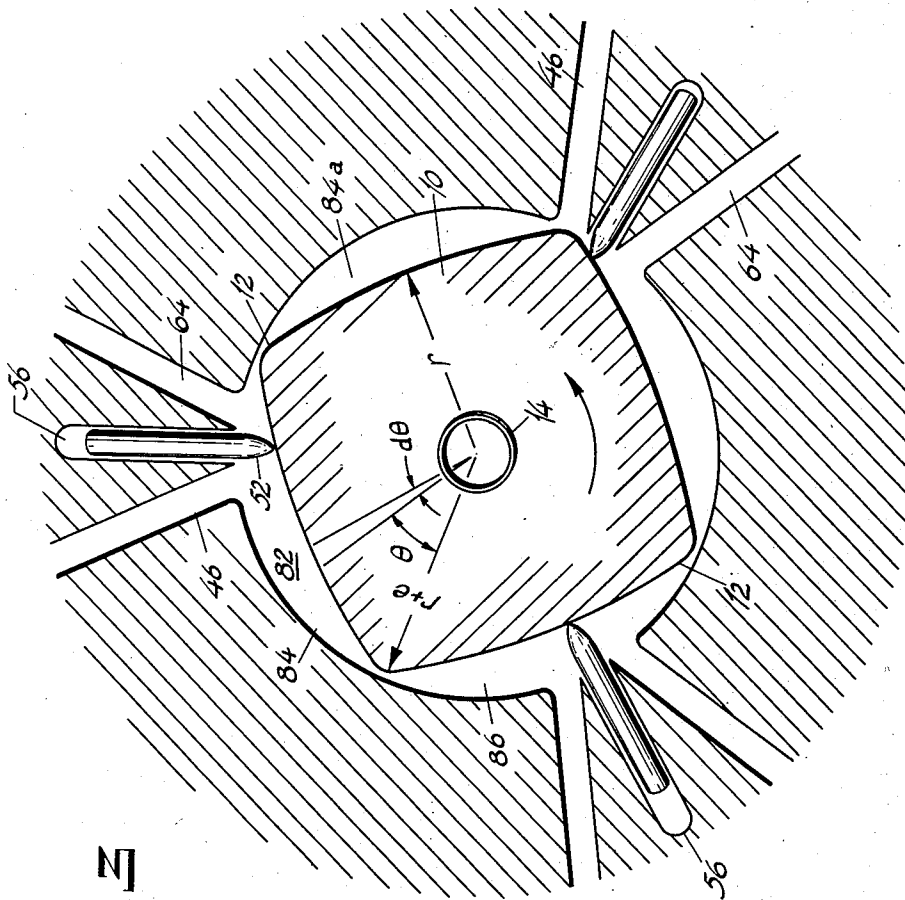
J. R. FARRON ET AL

2,845,872

CAM PUMP

Filed Sept. 16, 1953

8 Sheets-Sheet 2



2-1-1

INVENTOR.
JOHN R. FARRON
BY CHIEN-BOR SUNG

Edsworth R. Roston
ATTORNEY

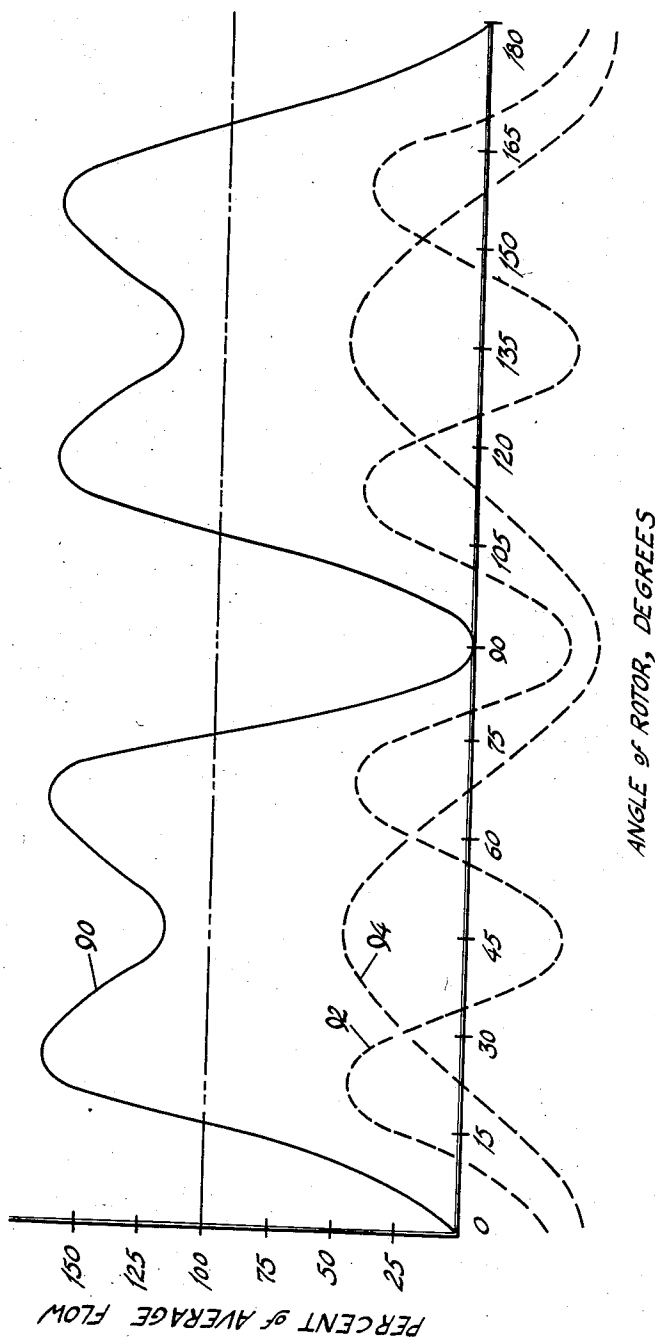
Aug. 5, 1958

J. R. FARRON ET AL
CAM PUMP

2,845,872

Filed Sept. 16, 1953

8 Sheets-Sheet 3



4-4

INVENTOR.
JOHN R. FARRON
BY CHIEN-BOR SUNG

Edw. R. Roston
ATTORNEY

Aug. 5, 1958

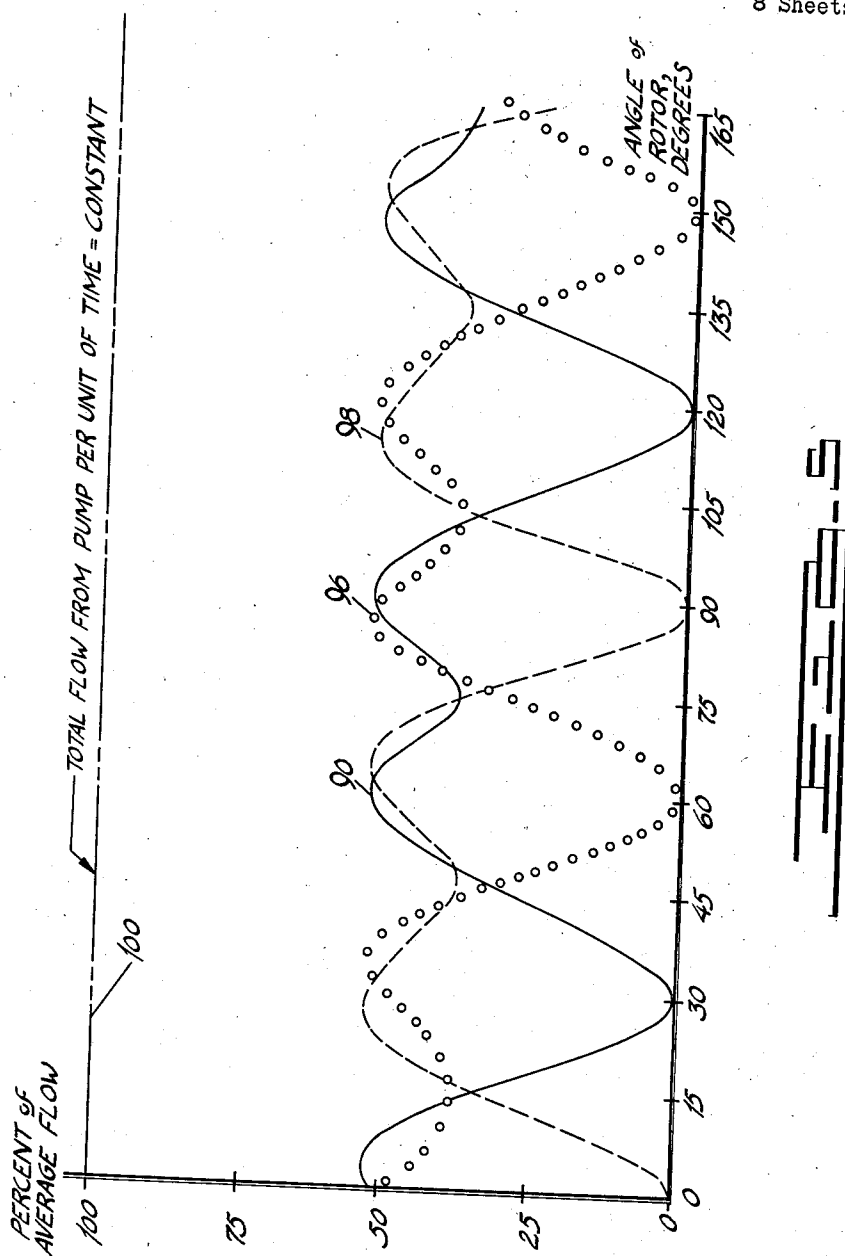
J. R. FARRON ET AL

2,845,872

CAM PUMP

Filed Sept. 16, 1953

8 Sheets-Sheet 4



INVENTOR.

JOHN R. FARRON

BY CHIEN-BOR SUNG

Ellsworth R. Roston

ATTORNEY

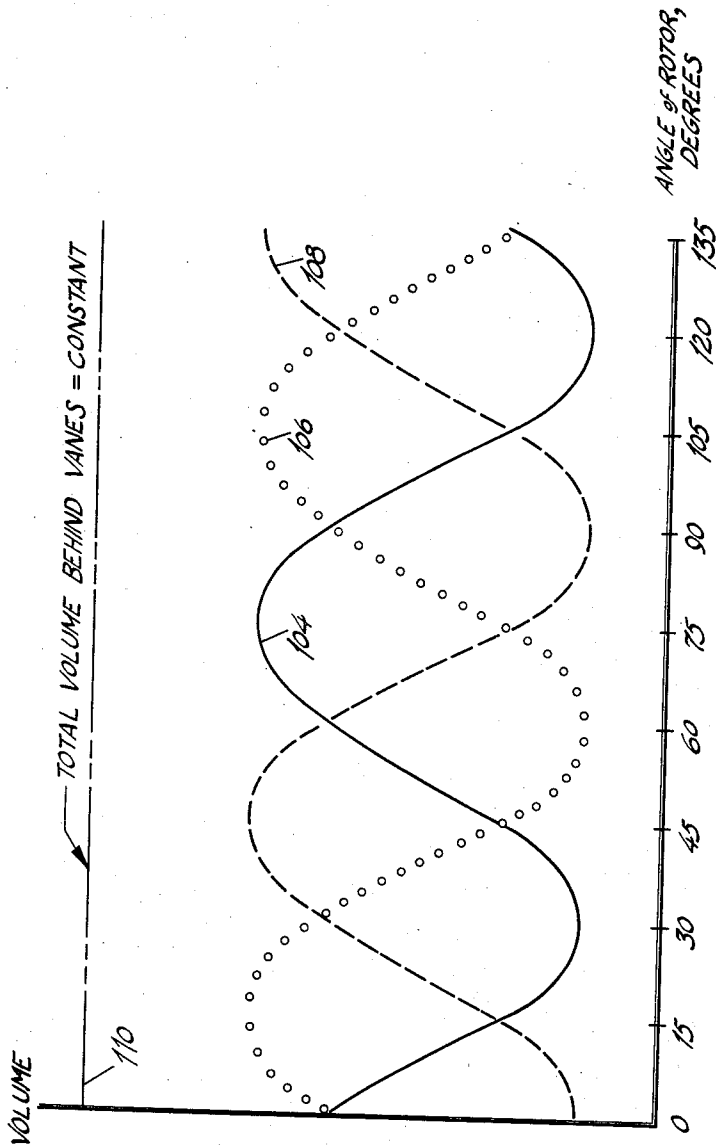
Aug. 5, 1958

J. R. FARRON ET AL
CAM PUMP

2,845,872

Filed Sept. 16, 1953

8 Sheets-Sheet 5



INVENTOR.
JOHN R. FARRON
BY CHIEN-BOR SUNG

Ellsworth R. Roston
ATTORNEY

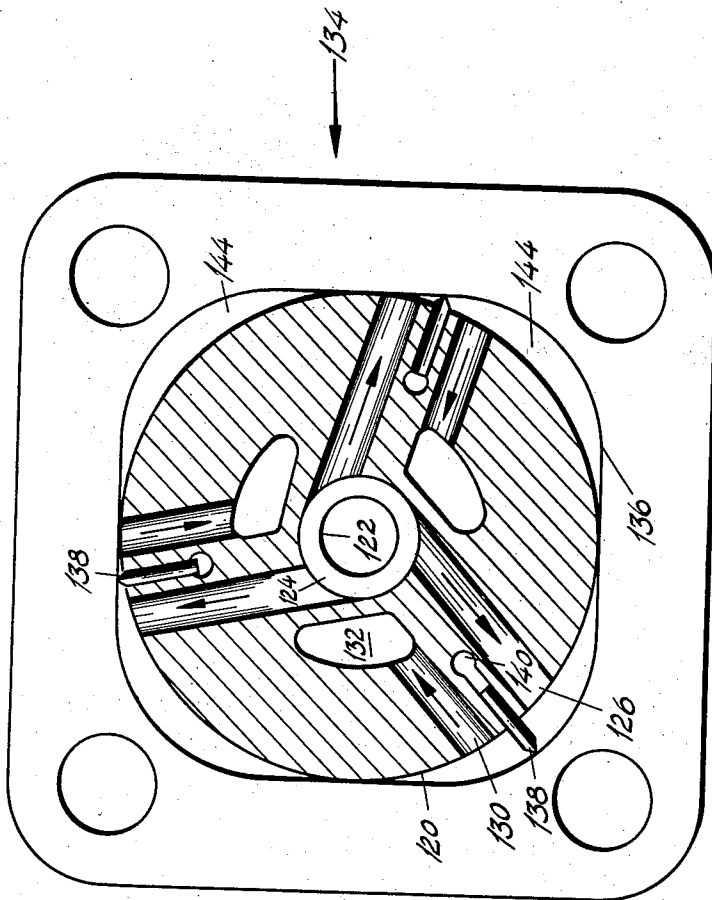
Aug. 5, 1958

J. R. FARRON ET AL
CAM PUMP

2,845,872

Filed Sept. 16, 1953

8 Sheets-Sheet 6



7-133-1

INVENTOR.
JOHN R. FARRON
BY CHIEN-BOR SUNG

Ellsworth R. Roston
ATTORNEY

Aug. 5, 1958

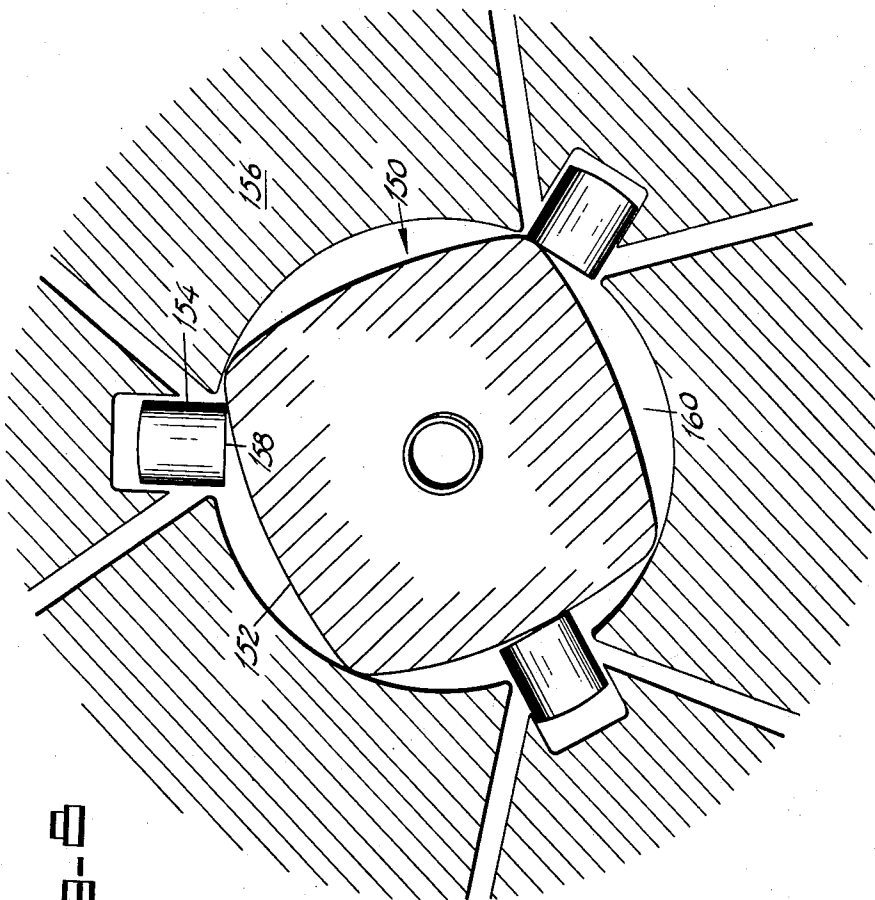
J. R. FARRON ET AL

2,845,872

CAM PUMP

Filed Sept. 16, 1953

8 Sheets-Sheet 7



INVENTOR.

JOHN R. FARRON
BY CHIEN-BOR SUNG

Ellsworth R. Roston
ATTORNEY

Aug. 5, 1958

J. R. FARRON ET AL
CAM PUMP

2,845,872

Filed Sept. 16, 1953

8 Sheets-Sheet 8

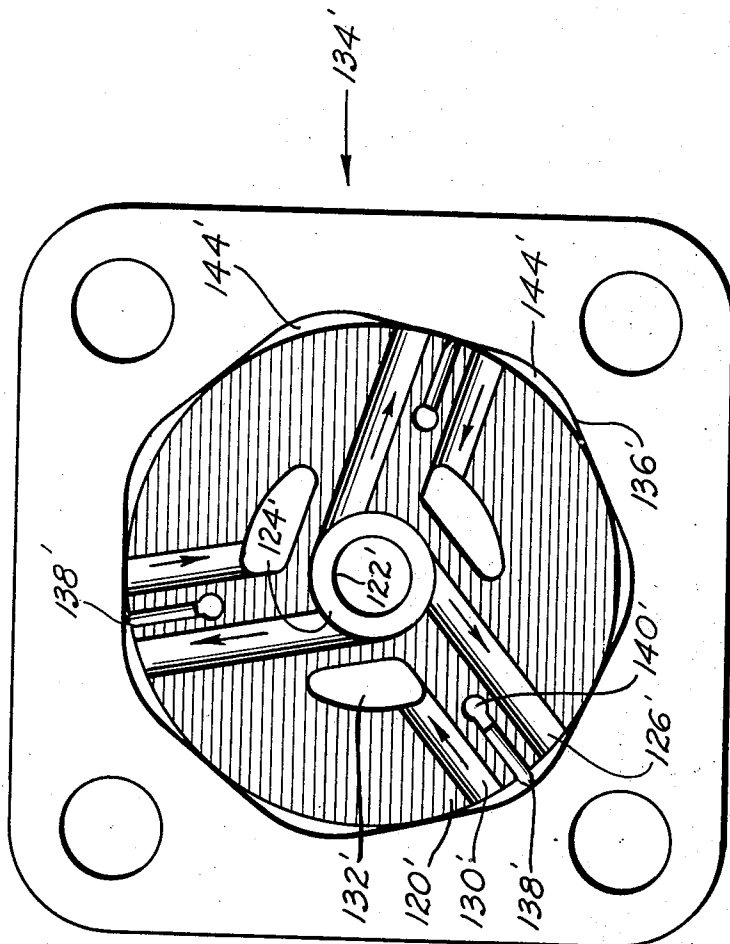


FIG. 9-

INVENTORS
JOHN R. FARRON.
CHIEN-BOR SUNG.
BY

William P. Hickey
ATTORNEY.

1

2,845,872

CAM PUMP

John R. Farron, Oak Park, and Chien-Bor Sung, Detroit, Mich., assignors to Bendix Aviation Corporation, Detroit, Mich., a corporation of Delaware

Application September 16, 1953, Serial No. 380,406

8 Claims. (Cl. 103—123)

This invention relates to positive displacement fluid handling devices and more particularly to cam pumps for producing a substantially uniform output when driven at a uniform speed.

In recent years, the use of hydraulic equipment has increased considerably. For example, in the automotive field automatic transmissions and power steering apparatus have become relatively popular. Certain problems exist in such apparatus and cause inconvenience to the users of the apparatus. For example, one of the important problems existing in the design of power steering and other apparatus results from the tendency of the apparatus to produce pulses during use. These pulses are transmitted from the source through the mechanical linkages and hydraulic media to the passenger compartment. In this way, the pulses produce a discomfort to the driver.

It has been determined that the pulses transmitted to the driver of an automobile having power steering equipment largely results from the operation of pumps which supply fluid for operating the equipment. The pulses are produced because the pumps do not have the desirable characteristics of generating a pulse-free and uniform output. Attempts have been made to eliminate these pulses but such attempts have not as yet been entirely successful.

This invention provides a positive displacement fluid handling device having a rotor member and a stator member one of which is provided with a surface of revolution and the other of which is provided with a multi-lobed camming surface—the extremity of the lobes of which extend into substantial sealing engagement with the surface of revolution. Each of the lobes has a peripheral contour corresponding substantially to that of the other lobe and establishes a continuity at its extremities with the contiguous extremities of the adjacent lobes. The periphery of the cam is preferably defined throughout its contour by a function which includes trigonometric functions such as sines and cosines.

A plurality of vanes extend from the member having the surface of revolution and extending into engagement with the camming surface of the other member. The vanes co-operate with the peripheral contour of the multi-lobed member to describe a purely trigonometric function such as a sine or a cosine. By employing vanes, the fluid handling device is effectively divided into a plurality of dynamic chambers. The amount of fluid flowing in or out, as the case may be, of each chamber at any instant differs from the amount of fluid flowing in or out of the other chambers, but the resultant amount of fluid flowing in or out of all of the chambers remains a constant at all times. In this way, a relatively simple and inexpensive cam-type fluid handling device is provided for producing a substantially constant displacement.

An object of this invention is to provide a positive displacement fluid handling device having a stator and a rotor one of which is provided with a surface of revolution and the other of which is provided with a multi-lobed camming surface the extremities of which

2

are in substantial sealing engagement with the surface of revolution so as to produce a substantially pulse-free fluid flow through the device.

Another object is to provide a positive displacement fluid handling device of the above character having a multi-lobed camming member, each lobe having a continuous contour and establishing a continuity at its extremities with the extreme positions of the contiguous lobes.

A further object is to provide a positive displacement fluid handling device of the above character having a multi-lobed camming surface in which the peripheral contour of the cam is defined by a function which includes trigonometric functions such as sines and cosines.

Still another object is to provide a pump of the above character in which a plurality of vanes are associated with a multi-lobed camming member to produce a plurality of dynamic chambers and in which the output from all of the chambers remains a constant at all times.

A still further object is to provide a pump of the above character in which the movement of the vanes during the operation of the pump does not interfere in any way with the production of a substantially constant flow of fluid from the pump.

Other objects and advantages will be apparent from a detailed description of the invention and from the appended drawings and claims.

In the drawings:

Figure 1 is a sectional view of a hydraulic pump and is taken substantially on the line 1—1 of Figure 2, parts of the view being broken away to show other parts more clearly;

Figure 2 is a sectional view substantially on the line 2—2 of Figure 1;

Figure 3 is an enlarged schematic view in section of the pump shown in Figures 1 and 2 is included to illustrate the operation of the pump more clearly;

Figures 4, 5 and 6 are curves which illustrate the operation of the pump shown in the previous figures;

Figure 7 is a somewhat schematic view in section of another embodiment of a pump for producing a substantially uniform output;

Figure 8 is a somewhat schematic view in section of a third embodiment of the invention; and

Figure 9 is a somewhat schematic view in section of another embodiment similar to the one shown in Figure 7, but having a larger lobe to vane ratio.

In one embodiment of the invention, a rotor 10 having a plurality of lobes 12 extends from a shaft 14 at an intermediate position in the shaft. Each lobe 12 has a contour corresponding substantially to that of the other lobes and establishes a continuity at its extremities with the contiguous extremities of the adjacent surface. Each lobe 12 has a continuous contour such that no sharp breaks occur in its periphery. On the basis of analysis, we have found that there should be at least four lobes on the rotor.

With progressive movements along the periphery of the lobe, the distance from the center of the rotor 10 to the periphery of each lobe preferably varies as a function which includes trigonometric functions such as sines and cosines. For example, the distance from the center of the rotor 10 to the median position along the periphery of each lobe 12 may be indicated as r . The distance from the center of the rotor 10 to the two extremities of each lobe 12 may be defined as $r + e$, where e indicates an increment in the value of r . The distance from the center of the rotor 10 to any position along the periphery of each lobe 10 may then be designated as $r + e \cos n \theta$, where θ = the angle between an extremity of each lobe 12 and the position in question; and n = the number of lobes 12 on the rotor.

3

The rotor 10 is adapted to rotate on bearings 16 (Figure 2) within a stator, generally indicated at 18. The stator 18 is formed from three members 20, 22 and 24 which are aligned relative to one another by dowel pins (not shown) and secured by bolts 26. The members 20, 22 and 24 are sealed relative to one another by O rings 28. The rotor 10 is positioned within the member 22, which is provided with an annular interior having a radius of approximately $r+e$ to form seals with the extremities of each lobe 12.

A spacer 30 is mounted on the shaft 14 at one end of the shaft and is axially maintained in position by lock nuts 32. An annular seal 34 is in turn supported on the spacer 30. The seal 34 is axially fixed in position by a retainer 36 and studs 38, which extend through the retainer 36 into the member 20.

A threaded socket 40 extends radially inwardly from the peripheral surface of the member 20 to provide a connection for the introduction of fluid to the pump. The socket 40 communicates with an annular manifold 42 in the member 20. A plurality of passages 44 (Figure 1) extend at spaced intervals from the manifold 42 through the member 20 to the member 22. The passages 44 form a continuous circuit with ports 46 which extend radially inwardly through the member 22 to the inner periphery of the member.

A plurality of caps 48 are supported by the member 22 at spaced intervals around the periphery of the member. Each cap 48 retains a spring 50 under compression between it and an associated vane 52. The vanes are provided with pointed tips at their inner ends for reasons which will be disclosed in detail hereafter. The pointed tips on the vanes can be defined by a substantially zero radius of curvature. Each vane 52 is positioned adjacent the inner end of an associated port 46 on the up stream side of the passage—in other words, in a clockwise direction from an associated port 46 for a counter-clockwise movement of the rotor 10.

We have found that a definite relationship exists between the number of lobes 12 and the number of vanes 52. For example, we have found that any odd number of vanes 52 greater than 1 may be used. When such a number of vanes 52 is used, a number of lobes may be used such that the number of lobes is not evenly divisible by the number of vanes. For example, when 3 vanes 52 are utilized, the number of lobes 12 may be 4, 5, 7, 8, 10, 11, 13, etc. To prevent a sudden exposure of low pressure chambers to high pressure ports, it has been found advantageous for certain applications to use a number of lobes such that the difference between the vanes and lobes is greater than one (1). For example, a 3 vane 8 lobe arrangement is advantageous over a 3 vane 4 lobe arrangement in that intermediate chambers are provided in the former so that pre-compression may take place in the intermediate chambers, as a result of normal leakage between chambers, prior to the exposure of such chambers to the high pressure ports. An even number of vanes 52 may be utilized when the number of vanes and the number of lobes 12 are both integrally divisible by 2. For example, for a cam pump having six vanes 52, the number of lobes 12 may be 8, 10, 14, 16, 20, 22, 26, etc. The utilization of an even number of vanes and lobes is advantageous for certain applications in that a balanced bearing load is provided in the pump. For example, a 4 vane 6 lobe pump would be properly balanced and in addition such a pump would be advantageous because of the pre-compression occurring in the intermediate chambers as mentioned above. We have also found it desirable to have the number of vanes 52 less than the number of lobes 12.

The vanes 52 are angularly separated by substantially equal distances from one another. Thus, when three vanes are used as shown in Figure 3, the angular separation may be 120° and the angular separation between vanes may be 72° for a pump having five vanes. The

4

vanes are pointed at their inner ends to make line contacts with the periphery of the rotor in the axial direction of the rotor as shown in Figures 1 and 3. The vanes 52 are slideable in a radial direction within pockets 56 formed in the member 22. The pockets 56 communicate by slots 58 (Figure 2) with an annular recess 60 in the member 20 for reasons which will be disclosed in detail hereinafter.

During the time that the pump is in operation, the dynamic action of the fluid in the compartments 56 causes the vanes 52 to be pressed against the lobes 12 of the rotor 10. The vanes 52 are pressed against the rotor 10 because of the leakage of fluid through a hydraulic circuit which includes the space between the rotor 10 and the stator 18 and also includes the radial walls of the compartments 56. Because of the dynamic action resulting from the leakage of the fluid into the compartments 56, the springs 50 are actually required to press the vanes 52 against the peripheral contour of the rotor 10 only when the pump is being started.

Ports 64 (Figure 1) corresponding in construction to the ports 46 are adapted to receive the fluid in the spaces between the lobes 12 and the interior surface of the member 22. Each of the ports 64 is positioned adjacent an associated vane 52 in a clockwise direction from the vane. The ports 64 extend from the member 22 in a radial direction through the member 24 and communicate with passages 66 in the member 24. The passages 66 form a continuous circuit with an annular manifold 68 (Figure 2), which in turn communicates with a threaded socket 70. The socket 70 provides a connection for the flow of fluid from the pump.

A spacer 72 is supported on the shaft 10 at the left end of the shaft as seen in Figure 2 and is held in axially fixed position by lock nuts 74. A cap 76 fits within the member 24 and co-operates with an O ring 78 to form a seal. The cap 76 is secured to the member 24 as by suitable screws 80.

As may be seen in Figure 3, a chamber 82 is produced between each pair of adjacent vanes 52. Each chamber 82 is in turn divided into a pair of pockets 84 and 86 by the seal formed between the extremity of a lobe 12 and the annular interior of the member 22. When the volume of a pocket 84 is increasing at any instant, the volume of an associated pocket 86 is decreasing.

During the time that the volume of one of the pockets 84 is increasing, fluid flows into the pocket through a fluid circuit including the socket 40, the manifold 42, one of the passages 44 and one of the ports 46. This causes the volume of the fluid in the pocket to increase. The volume of the fluid in a pocket reaches a maximum value when the trailing extremity of a lobe 12 is positioned adjacent a port 46. For example, a pocket 84a formed between the lower right port 46 in Figure 3 and the upper lobe extremity has substantially a maximum value.

After the pocket 84a has reached a maximum volume, the rotor 10 moves through a relatively short angle before the forward edge of the pocket 84a reaches the vane 52 positioned at the top of Figure 3. With further angular movement of the rotor 10, the volume of the pocket 84a decreases and causes fluid to flow into the port 64 at the top of Figure 3. The fluid then flows through a circuit including the port 64, the associated passage 66, the annular manifold 68 and the socket 70.

Because of the repetitive pattern of the peripheral contour on the rotor 10, the amount of fluid flowing through each of the ports 64 also varies in a repetitive pattern. When the rotor 10 has a peripheral contour which varies in a pattern including a trigonometric function and corresponding to

$$r+e \cos \theta$$

the amount of fluid flowing per unit of time through each port 64 varies in a pattern corresponding to the curve 90 shown in Figure 4. This curve is formed from

a sinusoidal curve 92 having a frequency of n cycles per revolution of the rotor and a sinusoidal curve 94 having a frequency of $2n$ cycles per revolution of the rotor.

For a rotor having three vanes as shown in the drawings, the fluid flowing through each of the ports 64 is indicated by the curves 90, 96 and 98 in Figure 5. As may be seen in Figure 5, the curves 90, 96 and 98 have a phase relationship of 30° with respect to each other. The resultant amount of fluid flowing through the socket 70 at any instant is indicated in Figure 5 by a curve 100, which remains a constant at all times.

Since the vanes follow the contour of the lobes 12, they move radially in a trigonometric pattern corresponding to contour of the lobes. This causes the fluid retained in each of the vane compartments 56 to vary in a trigonometric pattern. Thus, for a pump having three vanes as shown in the drawings, the amount of fluid in each of the vane compartments 56 varies in accordance with one of the patterns 104, 106 and 108 shown in Figure 6. Because of the relative phases of the patterns 104, 106 and 108, the resultant amount of fluid retained in all of the vane compartments 56 at any instant remains a constant. This is indicated at 110 in Figure 6. Since the resultant amount of fluid in all of the vane compartments 56 remains a constant at all times, the flow of fluid into and out of the compartments does not affect the pulse-free operation of the pump.

We have found that it is necessary for the vanes 52 to move in a substantially pure trigonometric function such as a sine or a cosine. The reason for this is the requirement that at least the first two derivatives of the function be continuous so that the vanes will not be subjected to shock loading during their movement. If the first two derivatives of the function, representing velocity and acceleration, are not continuous then the third derivative or the "jerk" of the function becomes infinitely large and the vanes will be subjected to shock loading during their movement. Such loading will cause the vanes to be jerked away from the surface along which the vanes are moving. In this way a pump is prevented from delivering a constant output because a continuous contact between the vanes and the surface is not maintained. Since all of the derivatives of a function such as a sine or cosine are continuous, shock loading of the vanes is substantially eliminated. A movement of the vanes in accordance with a function such as a sine or cosine is accomplished by utilizing a rotor having the peripheral contour disclosed above and by utilizing vanes 52 which contact the rotor at a substantially constant position on the vanes because of the zero radius of curvature at the tips of the vanes. The vanes can also have tips with an infinite radius of curvature or with any radius of curvature between zero and infinity, provided that the rotor has a complementary contour to produce a sinusoidal movement of the vanes.

A cam utilizing vanes having an infinite radius of curvature is shown in Figure 8 and will be disclosed in detail hereinafter.

The operation of the cam pump disclosed above to produce a substantially uniform output may be clearly seen by the following mathematical analysis. The volume enclosed by the stator 18 within an angular sector defined by an angle $d\theta$ may be given as:

$$dv_s = \frac{C_1}{2} (r_b + e)^2 d\theta$$

where

$r_b + e$ = the distance from the center of the pump to the wall of the stator 18; and

C_1 = the axial length of the rotor 10 and is a constant.

The volume occupied by the rotor in the angular arc $d\theta$ may be given by the expression

$$dV_r = \frac{C_1}{2} (r_b + e \cos n\theta)^2 d\theta$$

where

dV_r = volume occupied by the rotor;

θ = the angle from the extremity of a rotor lobe 12 to the position at which a vane 52 contacts the lobe,

n = the number of lobes 12 on the rotor; and

$r_b + e \cos n\theta$ = the distance from the center of the rotor 10 to the periphery of the rotor at the position defined by the angle θ .

Since the volume of a pocket 84 or 86 may be obtained by determining the difference between the volume of the stator and the volume occupied by the rotor, the volume of the pocket may be given as

$$dV_c = dV_s - dV_r = \frac{C_1}{2} [(r_b + e)^2 - (r_b + e \cos n\theta)^2] d\theta$$

where

V_c = the volume occupied by a pocket 84 or 86.

But the volume occupied by a pocket 84 or 86 continues from the point of contact between the vane 52 and the rotor lobe 12 to the extremity of the rotor lobe. Since the rotor lobe 12 has a radial length of

$$\frac{2\pi}{n}$$

$$V_c = \int_0^{\frac{2\pi}{n}} dV_c = \int_0^{\frac{2\pi}{n}} \frac{C_1}{2} [(r_b + e)^2 - (r_b + e \cos n\theta)^2] d\theta$$

Since $\cos^2 n\theta$ can be resolved trigonometrically into

$$\frac{1 + \cos 2n\theta}{2}$$

the volume V_c upon integration can be determined as

$$V_c = C_2 - C_1 (r_b e + \frac{1}{4} e^2) \theta + C_1 \frac{r_b e}{n} \sin n\theta + C_1 \frac{e^2}{8} \sin 2n\theta$$

where C_2 = a constant.

The volume displacement per unit of time of this pocket is

$$\frac{-dV_c}{dt} = \frac{-dV_c}{d\theta} \cdot \frac{d\theta}{dt} = C_1 \left[(r_b e + \frac{1}{4} e^2) - r_b e \cos n\theta - \frac{e^2}{4} \cos 2n\theta \right] \frac{d\theta}{dt}$$

As is well known, the rate at which fluid discharges from each pocket is directly dependent upon the rate at which the volume of the pocket decreases. Furthermore, since the number of pockets 84 and 86 undergoing compression at any instant corresponds to the number of outlet ports 64, there are $(n-1)$ compression chambers. This causes the cumulative rate of discharge from all of the pockets 84 and 86 to be

$$-\sum_{k=1}^{n-1} \frac{dV_k}{dt} = C_1 (n-1) (r_b e + \frac{1}{4} e^2) \frac{d\theta}{dt}$$

$$-\sum_{k=1}^{n-1} C_1 r_b e \cos n \left(\theta + \frac{2(k-1)\pi}{n-1} \right) \frac{d\theta}{dt}$$

$$-\sum_{k=1}^{n-1} C_1 \frac{e^2}{4} \cos 2n \left(\theta + \frac{2(k-1)\pi}{n-1} \right) \frac{d\theta}{dt}$$

where k may vary from 1 to $(n-1)$.

In the above equation, the expression

$$\left(\theta + \frac{2(k-1)\pi}{n-1} \right)$$

is obtained by starting from a position θ along a first lobe 12 as defined above and by adding

$$\frac{2\pi}{n-1}$$

75 radians to obtain a corresponding position along suc-

cessive rotor lobes. The position θ on the first lobe is chosen so that it corresponds with a position at which a vane 52 contacts a rotor lobe. In this way, the position θ defines the initial position of one of the pockets 84 and 86 which passes fluid through the adjacent outlet port 64.

In order to have a pulse-free pump, the following conditions must exist:

$$\sum_{k=1}^{n-1} C_1 r_1 e \cos n \left(\theta + \frac{2(k-1)\pi}{n-1} \right) = 0 \text{ or a constant}$$

$$\sum_{k=1}^{n-1} C_1 \frac{e^2}{4} \cos 2n \left(\theta + \frac{2(k-1)\pi}{n-1} \right) = 0 \text{ or a constant}$$

or the sum of the two above expressions must be equal to a constant. When the number of lobes 12 is such that $n=4, 6, 8, 10$ and successive even numbers and the number of vanes is $n-1$, the above conditions are satisfied. A pulse-free pump is also produced when an odd number of surfaces greater than 4 are provided and the number of vanes is less than the number of rotor lobes by certain integers greater than 1. For example, three vanes may be used with a rotor having either five or seven lobes, as disclosed above.

The pump disclosed above has several important advantages. As may be seen by the above analysis, the pump operates under ideal conditions to produce a constant output. This constant output is not affected in any way by the flow of fluid in the vane compartments 56, since the resultant flow of fluid through all of the compartments remains a constant at all times. Furthermore, the pump requires only a minimum number of components, each of which can be inexpensively manufactured. Even the rotor 10 can be made simply and inexpensively because of its continuous contour. The pump is also advantageous in that there is no shock loading of the vanes. This results from the continuous contour of the rotor periphery.

In the embodiment of the invention shown in Figure 7, an annular rotor 120 is mounted on the shaft 122. An annular manifold 124 is provided in the shaft 122 in communication with a plurality of ports 126 which extend at spaced intervals through the rotor 120 to the annular periphery of the rotor. Outlet ports 130 are also provided in the rotor 120 in communication with passages 132 for providing for the flow of fluid out of the pump.

The shaft 122 and the rotor 120 are adapted to rotate within a stator, generally indicated at 134 and having a plurality of lobes 136. Vanes 138 are positioned within compartments 140 in the rotor to contact the surface of the stator 134 as the rotor turns. The vanes 138 produce chambers 144 between the rotor 120 and the stator 134 corresponding to the chambers 82 disclosed above.

The pump shown in Figure 7 operates in the same manner as the pump shown in Figures 1 to 3, inclusive to obtain a substantially constant output. In addition to the advantages disclosed for the pump shown in Figures 1 to 3, inclusive, the pump shown in Figure 7 has further important advantages. The centrifugal force which is produced upon a rotation of the rotor 120 causes the fluid in the manifold 124 to be forced through the ports 126 into the chambers 144 between the rotor and the stator. Because of the force exerted on the fluid, no air spaces can be produced in the chambers 144 to prevent the flow of a constant amount of fluid out of the pump.

The pump shown in Figure 7 has the further advantage of utilizing a relatively simple hydraulic circuit. For example, no inlet passages corresponding to the outlet passages 132 are required in the pump to produce a flow of fluid to the chambers 144. In addition, no

springs corresponding to the springs 50 shown in Figure 2 are required to press the vanes 138 against the stator 134 when the pump is just being started.

The cam pump shown in Figure 8 also has a rotor generally indicated at 150 with a plurality of lobes 152, each of which is continuous along its periphery and each of which has a contour corresponding substantially to that of the other lobes. However, each of the lobes is defined by a trigonometric function which may be indicated in polar co-ordinates as:

$$r = \sqrt{r_0^2 + e^2 - 2r_0 e \cos n\theta + (n^2 - 1)e^2 \sin^2 n\theta}$$

A plurality of vanes 154 are provided at spaced angular intervals in a stator 156 to contact the lobes 152 of the rotor. Each of the vanes 154 has a flat face 158 in contiguous relationship to the rotor. The flat faces 158 can be defined by an infinite radius of curvature. As in the previous embodiments, a chamber 160 is produced between the rotor 150 and the stator 156 and is defined at its extremities by a pair of adjacent vanes 154.

As the rotor turns, each vane 154 moves radially in and out of its recess in the stator 156 in a pattern defined by a substantially pure trigonometric function such as a sine or a cosine. However, the line of contact between the face 158 of each vane 154 and the rotor varies along the face in a pattern corresponding to the trigonometric function defining the contour of each lobe 152.

We have found that the cumulative flow of fluid from the different outlet ports in the cam pump remains substantially constant as the pump rotates. This can be shown mathematically for an ideal pump. We have further found that such a pump is advantageous because of the ease with which the peripheral surface of the rotor can be machined to the proper contour.

The pump shown in Figure 9 is similar in construction and operation to that shown in Figure 7. Those parts in Figure 9 which correspond to those shown in Figure 7 are designated by a like reference numeral and are characterized further in that a prime mark is affixed thereto. Suffice it to say that the surfaces of the stator are formed according to the equation $r = r_0 + e \cos \theta$, that its output is constant; and that it has the further advantage of having pumping chambers which are completely filled with liquid but are not immediately communicated with the discharge passages of the pump. Leakage from the chambers which are connected to the discharge of the pump enters these isolated chambers to raise their pressure to a pressure which approaches discharge pressure before the isolated chamber is moved to a point where its leading lobe opens communication with a discharge port of the pump.

It should be appreciated that the apparatus disclosed above can also be used as a motor as well as a pump. The motor would have advantages similar to those of the pump in that it would be able to deliver a substantially constant output.

Although this invention has been disclosed and illustrated with reference to particular applications, the principles involved are susceptible of numerous other applications which will be apparent to persons skilled in the art. The invention is, therefore, to be limited only as indicated by the scope of the appended claims.

What is claimed is:

1. In a hydraulic power converter: a rotor member, a stator member, one of said members having a surface of revolution and the other of said members having a cooperating camming surface having valleys separated by lobes which slidably sealingly abut said surface of revolution to form individual fluid confining chambers, a plurality of generally radially extending slots in the member having the surface of revolution, a vane in each slot extending into sliding sealing engagement with the other of said surfaces and being constructed and arranged to sweep fluid from said chambers, suitable passage means

in said members for removing fluid from and adding fluid to the chambers, said surface having its lobes and valleys formed substantially in accordance with the following equation:

$$r=r_b+e \cos n \theta$$

wherein:

r =the distance from the axis of rotor rotation to any point on the cam surface,

r_b =the radius of the base circle about which the cam surface is formed,

e =the maximum height of both lobes and valleys from the base circle,

n =the number of lobes spaced around the cam surface, and

θ =any angle measured from a reference plane passing through a radially outermost point on the camming surface.

2. In a hydraulic power converter: a rotor member, a stator member, one of said members having a surface of revolution and the other of said members having a cooperating camming surface having valleys separated by lobes which slidably sealingly abut said surface of revolution to form individual fluid confining chambers, a plurality of generally radially extending slots in the member having the surface of revolution, a vane in each slot extending into sliding sealing engagement with the other of said surfaces and being constructed and arranged to sweep fluid from said chambers, suitable passage means in said members for removing fluid from and adding fluid to the chambers, said surface having its lobes and valleys formed substantially in accordance with the following equation:

$$r=r_b+e \cos n \theta$$

wherein:

r =the distance from the axis of rotor rotation to any point on the cam surface,

r_b =the radius of the base circle about which the cam surface is formed,

e =the maximum height of both lobes and valleys from the base circle,

n =the number of lobes spaced around the cam surface, and

θ =any angle measured from a reference plane passing through a radially outermost point on the camming surface,

the total number of lobes in the surface exceeding the number of vanes in the power converter by at least two, and the ratio of lobes to vanes being greater than one but not being equal to an integer.

3. In a hydraulic power converter: a rotor member, a stator member, one of said members having a surface of revolution and the other of said members having a cooperating camming surface having valleys separated by lobes which slidably sealingly abut said surface of revolution to form individual fluid confining chambers, a plurality of generally radially extending slots in the member having the surface of revolution, a vane in each slot extending into sliding sealing engagement with the other of said surfaces and being constructed and arranged to sweep fluid from said chambers, suitable passage means in said members for removing fluid from and adding fluid to the chambers, said surface having its lobes and valleys formed substantially in accordance with the following equation:

$$r=r_b+e \cos n \theta$$

wherein:

r =the distance from the axis of rotor rotation to any point on the cam surface,

r_b =the radius of the base circle about which the cam surface is formed,

e =the maximum height of both lobes and valleys from the base circle,

m =the number of vanes in the power converter,

n =the number of lobes spaced around the cam surface, and

θ =any angle measured from a reference plane passing through a radially outermost point on the camming surface,

the total number of lobes in the surface exceeding the number of vanes in the power converter by at least two, and the ratio

$$\frac{2n}{m}$$

being greater than one but not being equal to an integer.

4. In a hydraulic power converter: a rotor member, a stator member, one of said members having a surface of revolution and the other of said members having a cooperating camming surface having valleys separated by lobes which slidably sealingly abut said surface of revolution to form individual fluid confining chambers, a plurality of generally radially extending slots in the member having the surface of revolution, a vane in each slot extending into sliding sealing engagement with the other of said surfaces and being constructed and arranged to sweep fluid from said chambers, suitable passage means in said members for removing fluid from and adding fluid to the chambers, said surface having its lobes and valleys formed substantially in accordance with the following equation:

$$r=r_b+e \cos n \theta$$

wherein:

r =the distance from the axis of rotor rotation to any point on the cam surface,

r_b =the radius of the base circle about which the cam surface is formed,

e =the maximum height of both lobes and valleys from the base circle,

n =the number of lobes spaced around the cam surface, and

and

m =the number of vanes in the power converter,

θ =any angle measured from a reference plane passing through a radially outermost point on the camming surface;

and wherein the ratio of n to m is such that

$$\frac{n}{x} \text{ equals } \frac{m}{x} + y$$

where x is the largest integer which can be divided into both n and m to give an integer in both case, y =any integer greater than one; and the ratio

$$\frac{n}{x} - \frac{m}{x}$$

does not equal an integer.

5. In a hydraulic power converter: a rotor member, a stator member, one of said members having a surface of revolution and the other of said members having a cooperating camming surface having valleys separated by lobes which slidably sealingly abut said surface of revolution to form individual fluid confining chambers, a plurality of generally radially extending slots in the member having the surface of revolution, a vane in each slot extending into sliding sealing engagement with the other of said surfaces and being constructed and arranged to sweep fluid from said chambers, suitable passage means in said members for removing fluid from and adding fluid to the chambers, said surface having its lobes

and valleys formed substantially in accordance with the following equation:

$$r=r_b+e \cos n \theta$$

wherein:

r =the distance from the axis of rotor rotation to any point on the cam surface,
 r_b =the radius of the base circle about which the cam surface is formed,
 e =the maximum height of both lobes and valleys from the base circle,
 n =the number of lobes spaced around the cam surface,
 and

m =the number of vanes in the power converter,
 θ =any angle measured from a reference plane passing through a radially outermost point on the camming surface;

and wherein the ratio of n to m is such that

$$\frac{n}{x} \text{ equals } \frac{m}{x} + y$$

where x is the largest integer which can be divided into both n and m to give an integer in both cases, y equals any integer greater than one, and

$$\frac{2n}{m}$$

does not equal an integer.

6. In a pump: a rotor member, a stator member positioned outwardly of said rotor member, said rotor having a surface of revolution and said stator having a cooperating camming surface having valleys separated by lobes which slidably sealingly abut said surface of revolution to form individual fluid pumping chambers, a plurality of generally radially extending slots in said rotor, a vane in each slot extending into sliding sealing engagement with said stator and being constructed and arranged to sweep fluid from said pumping chambers, suitable passage means in said members for removing fluid from and adding fluid to the pumping chambers, said surface having its lobes and valleys formed substantially in accordance with the following equation:

$$r=r_b+e \cos n \theta$$

wherein:

r =the distance from the axis of rotor rotation to any point on the cam surface,
 r_b =the radius of the base circle about which the cam surface is formed,
 e =the maximum height of both lobes and valleys from the base circle,
 n =the number of lobes spaced around the cam surface,
 and

m =the number of vanes in the pump,
 θ =any angle measured from a reference plane passing through a radially outermost point on the camming surface;

and wherein the ratio of n to m is such that

$$\frac{n}{x} \text{ equals } \frac{m}{x} + y$$

where x is the largest integer which can be divided into both n and m to give an integer in both cases, y =any integer greater than one; and the ratio

$$\frac{n}{x} + \frac{m}{x}$$

does not equal an integer.

7. In a hydraulic power converter: a rotor member, a stator member, one of said members having a surface of revolution and the other of said members having a cooperating camming surface having valleys separated by lobes which slidably sealingly abut said surface of revolution to form individual fluid confining chambers, a plurality of generally radially extending slots in the member having the surface of revolution, a vane in each slot extending into sliding sealing engagement with the other of said surfaces and being constructed and arranged to sweep fluid from said chambers, suitable passage means in said members for removing fluid from and adding fluid to the chambers, said surface having its lobes and valleys formed substantially in accordance with the following equation:

$$r^2=r_b^2+e^2-2r_b e \cos n \theta + (n^2-1)e^2 \sin^2 n \theta$$

wherein:

r =the distance from the axis of rotor rotation to any point on the cam surface,
 r_b =the radius of the base circle about which the cam surface is formed,
 e =the maximum height of both lobes and valleys from the base circle,
 n =the number of lobes spaced around the cam surface,
 and

θ =any angle measured from a reference plane passing through a radially outermost point on the camming surface.

8. In a pump: a rotor member, a stator member, said rotor having a surface of revolution and said stator having a cooperating camming surface having valleys separated by lobes which slidably sealingly abut said surface of revolution to form individual fluid pumping chambers, a plurality of generally radially extending slots in said rotor, a vane in each slot extending into sliding sealing engagement with said stator and being constructed and arranged to sweep fluid from said pumping chambers, suitable passage means in said members for removing fluid from and adding fluid to the pumping chambers, said surface having its lobes and valleys formed substantially in accordance with the following equation:

$$r^2=r_b^2+e^2-2r_b e \cos n \theta + (n^2-1)e^2 \sin^2 n \theta$$

wherein:

r =the distance from the axis of rotor rotation to any point on the cam surface,
 r_b =the radius of the base circle about which the cam surface is formed,
 e =the maximum height of both lobes and valleys from the base circle,
 n =the number of lobes spaced around the cam surface,
 and

θ =any angle measured from a reference plane passing through a radially outermost point on the camming surface.

References Cited in the file of this patent

UNITED STATES PATENTS

1,348,103	George	July 27, 1920
2,013,397	Balsiger	Sept. 3, 1935
2,191,172	Lisowski	Feb. 20, 1940
2,452,468	Johnson	Oct. 26, 1948
2,475,391	Johnson	July 5, 1949
2,609,754	Prendergast et al.	Sept. 9, 1952

FOREIGN PATENTS

70	29,195	Great Britain	of 1912
----	--------	---------------	---------