

Sept. 17, 1963

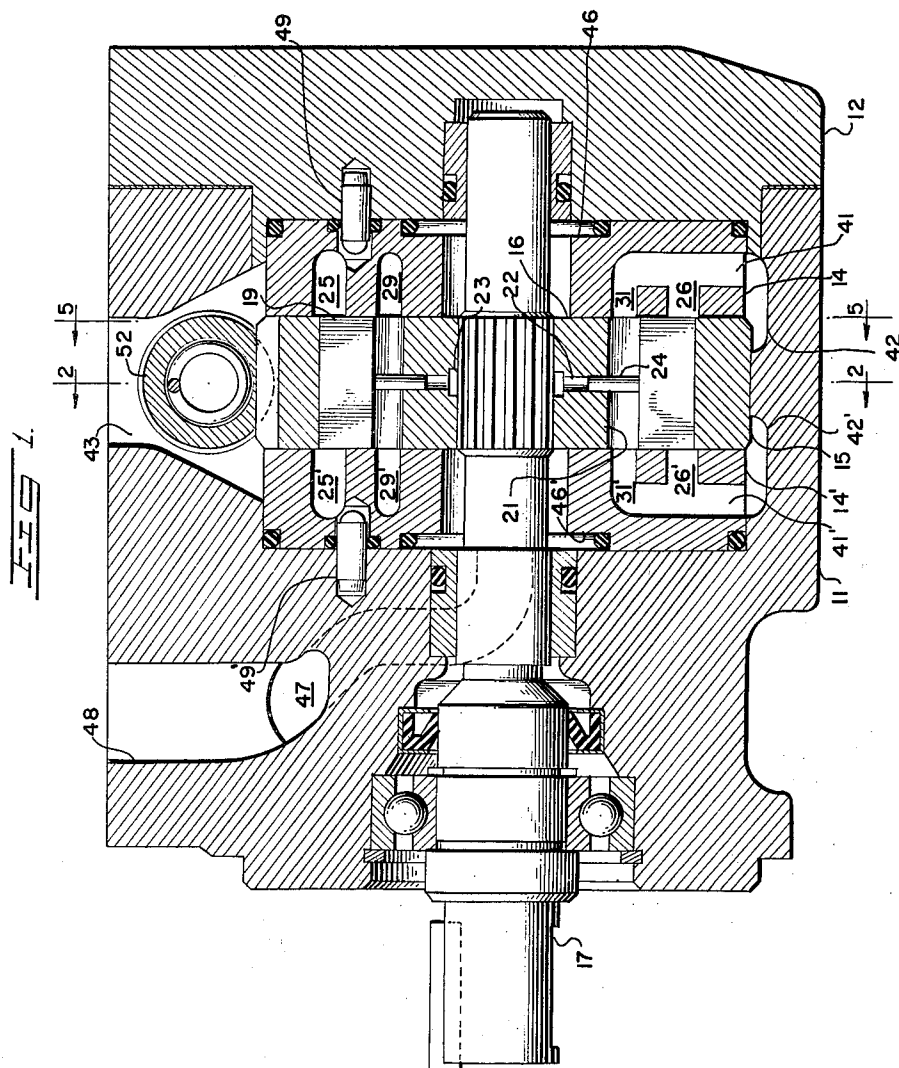
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3,103,893

VARIABLE DISPLACEMENT ENGINE

Filed June 30, 1960

3 Sheets-Sheet 1



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FIG 2

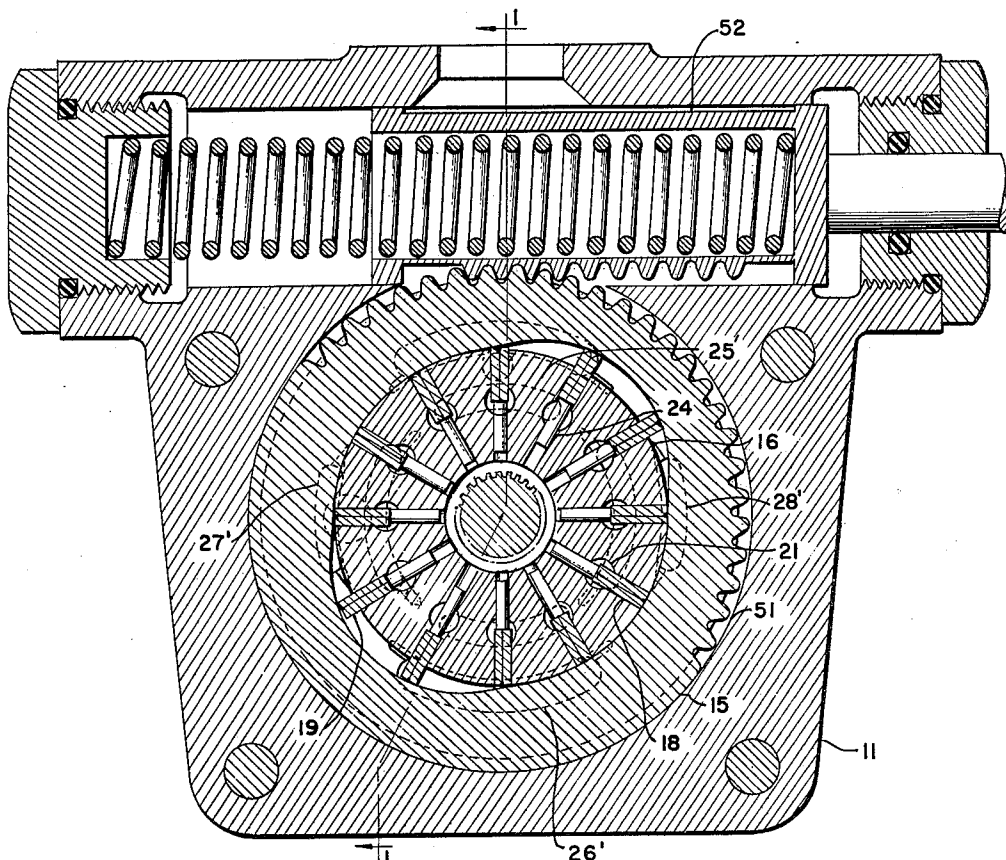


FIG 3.

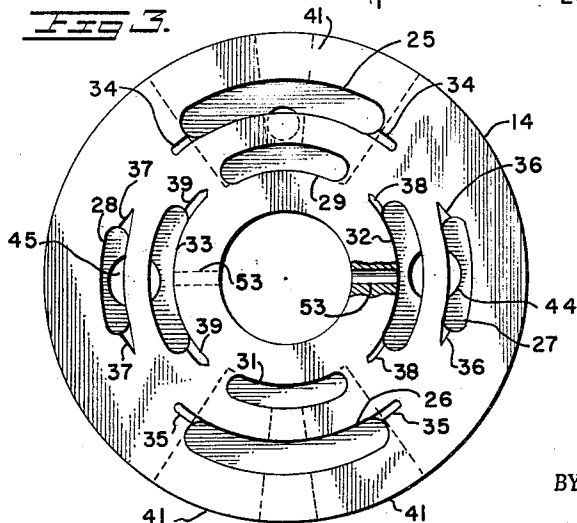
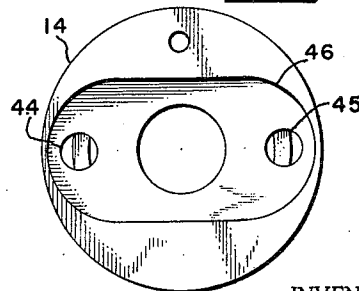


FIG 4



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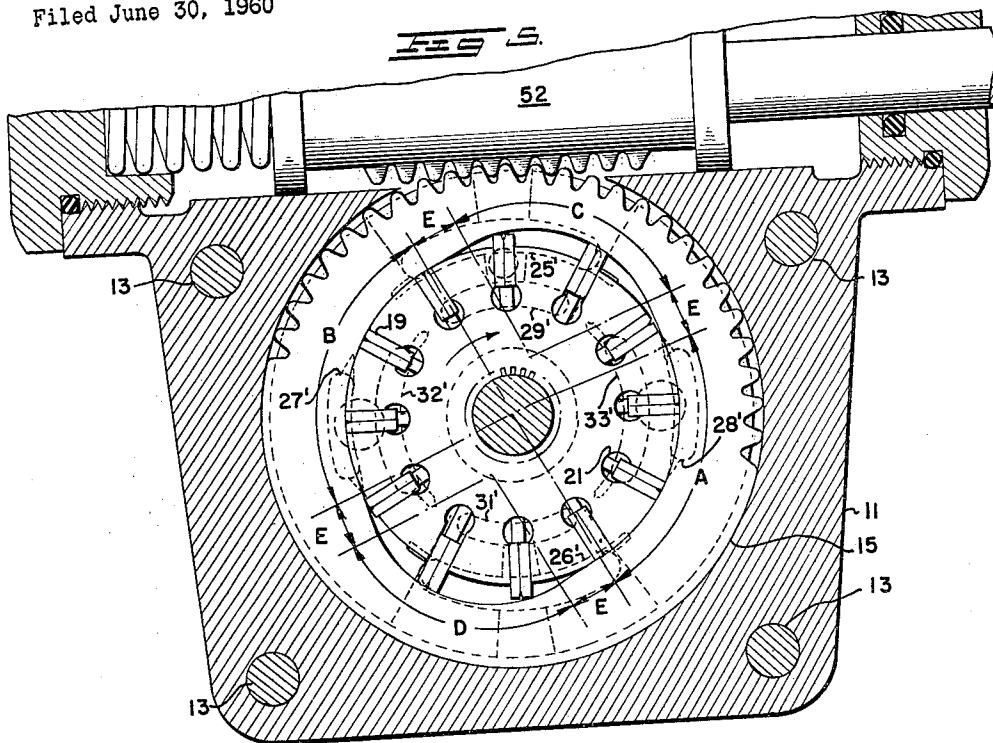
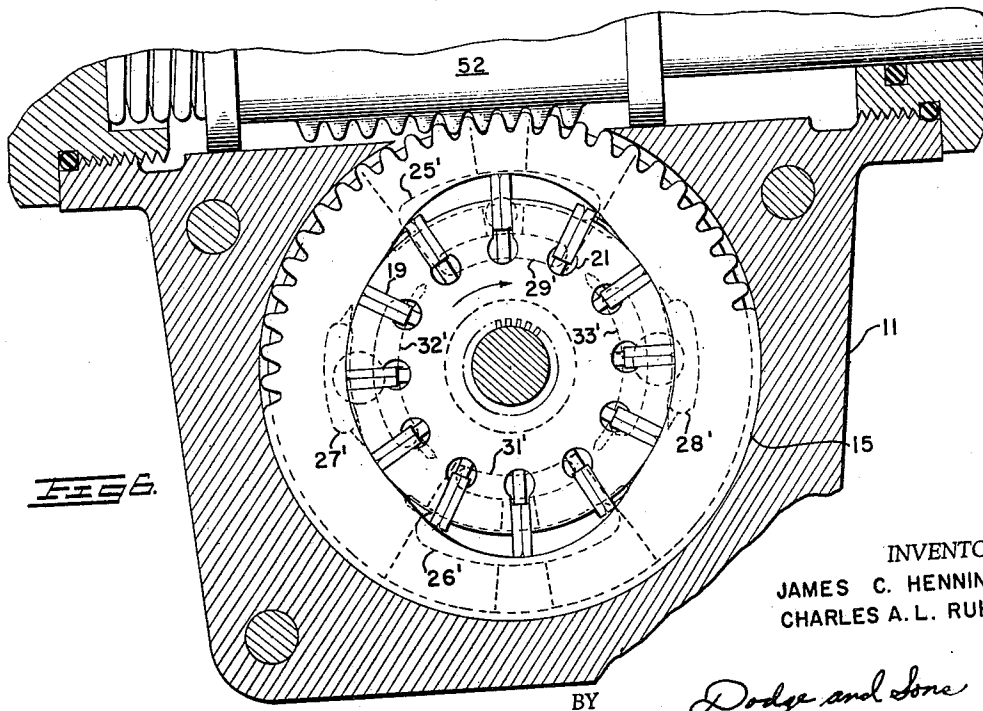
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3,103,893

VARIABLE DISPLACEMENT ENGINE

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2 Claims. (Cl. 103-120)

This invention relates to variable displacement rotary fluid pressure engines of the vane type. The term engine is used herein in its generic sense and it will be understood that it includes pumps as well as motors.

It is known in the art to vary the displacement of a double lobe balanced vane engine by rotating the cam ring about the axis of the rotor relatively to the stationary inlet and discharge ports. In this known method, as the cam ring is rotated from its maximum displacement-establishing position to its minimum displacement-establishing position, the volume of the intervane working space leaving each inlet port and entering the next succeeding discharge port decreases progressively while the volume of the intervane space leaving each discharge port and entering the next succeeding inlet port increases progressively. In this scheme, some fluid always is carried from the inlet ports to the discharge ports and the effective displacement of the engine depends on the ratio of the volumes of the intervane spaces entering and leaving the region of each discharge port. When these volumes are equal, effective displacement is zero.

While this prior displacement-varying scheme appears to offer several theoretical advantages, as a practical matter it produces intolerable shocks and noise, at least when the engine is used as a pump. The exact cause of this condition is not known but it is thought that it stems from the fact that the discharge port, which is under high pressure, communicates with expanding intervane spaces.

The object of the present invention is to provide an improvement in the prior displacement-varying method discussed above which is not subject to the shock and noise problems heretofore encountered. According to this invention, the cam surface of the cam ring is so designed that the discharge ports never communicate with an expanding intervane space and displacement is varied solely by regulating the volume of the intervane spaces leaving each inlet port and entering the next succeeding discharge port. Tests have shown that this change in basic concept does eliminate the shock and noise problems.

The preferred embodiment is employed in a pump and will be described in detail with reference to the accompanying drawings, in which:

FIG. 1 is a sectional view on line 1-1 of FIG. 2.

FIG. 2 is a sectional view taken on line 2-2 of FIG. 1, showing the cam ring in maximum displacement-establishing position.

FIG. 3 is a view of the inner face of port plate 14.

FIG. 4 is a view, on reduced scale, of the outer face of port plate 14.

FIG. 5 is a sectional view taken on line 5-5 of FIG. 1, showing the cam ring in a position intermediate the minimum and maximum displacement-establishing positions.

FIG. 6 is a sectional view similar to FIG. 5 but showing the cam ring in its minimum displacement-establishing position.

As shown in the drawings, the pump comprises a housing having two separable sections 11 and 12 which are joined by bolts 13 (see FIG. 2) and are cored and bored to receive two identical port plates 14 and 14' and a cam ring 15. Mounted between port plates 14 and 14' and within cam ring 15 is a rotor 16 that is splined to the drive shaft 17 and contains a plurality of radial slots 18 for receiving the reciprocable vanes 19. The vanes 19 are formed of two leaves and are of the type shown in

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Rosen Patent 2,393,223, granted January 15, 1946. A plurality of longitudinal bores 21 extend through rotor 16 and are arranged so that one intersects the inner end of each vane slot 18. Extending inward from each bore 21 is a radial bore 22 which intersects the annular chamber 23 and receives a sliding pin 24. These pins 24 are not attached to the vanes 19 but are urged into contact with them by the fluid pressure in annular chamber 23.

The two port plates 14 and 14' are identical so only the port plate 14 is described in detail. The corresponding parts of port plate 14' carry the same reference numerals with primes added for clarification. The inner face of port plate 14 (as shown in FIG. 3) contains a pair of diametrically opposed arcuate inlet ports 25 and 26 and a pair of diametrically opposed discharge ports 27 and 28 which are centered on an axis which makes an angle of 90° with the axis which bisects inlet ports 25 and 26. These four ports 25-28 are positioned radially to register with the intervane working spaces. Located radially inward from the ports 25-28 are four similar biasing ports 29, 31, 32 and 33 which are arranged to register with the through bores 21 formed in rotor 16. The ports 25, 26, 27, 28, 32 and 33 are provided with pairs of shallow tail ports 34, 35, 36, 37, 38, and 39, respectively. The tail ports, in effect, constitute extensions of the main ports with which they are connected and therefore, it will be understood that further references to the main ports 25-28 mean the main ports as extended by their tail ports. The angular distances between adjacent ends of the inlet ports 25 and 26 and the discharge ports 27 and 28 are equal to the intervane interval, i.e., the angular distance between the trailing edge of one vane and the leading edge of the next succeeding vane.

Four milled radial slots 41 open through the outer periphery of port plate 14 and connect ports 25, 26, 29 and 31 with annular inlet manifold 42 and inlet port 43 formed in housing section 11. Two longitudinal bores 44 and 45 connect the ports 27, 32 and 28, 33, respectively, with a shallow O-ring-encircled chamber 46 located in the outer face of port plate 14. The pressure in this chamber develops a biasing force which balances the forces acting on the inner face of the port plate and thus permits free rotational movement of cam ring 15 without excessive leakage. The shallow chamber 46', in the rear face of port plate 14', communicates with a pair of discharge manifolds 47 (only one shown) which lead to the pump discharge port 48. The port plates 14 and 14' are restrained against rotation by pins 49 and 49', respectively.

The cam ring 15 is journaled in a bore 51 formed in housing section 11 and, on its outer periphery, is provided with gear teeth which mate with a rack carried by plunger 52. On its inner periphery, the cam ring 15 carries a cylindrical cam surface (see FIG. 5) which, in transverse cross-section, comprises two diametrically opposed true circular arcs A and B which are centered on the axis of the rotor 16 and which have a radius of curvature slightly larger than the radius of rotor 16, two diametrically opposed eccentric arcs C and D whose centers are displaced in opposite directions from the axis of rotor 16 along an axis normal to the axis which bisects arcs A and B, and four blend curves E which are straight lines tangent to the adjacent ends of the true and eccentric arcs. The radius of curvature of arcs C and D is less than the radius of curvature of arcs A and B but, because of the eccentric location of their centers, the arcs C and D are farther from the axis of rotor 16 than the arcs A and B.

The lengths of the true arcs A and B and their angular positions relative to the discharge ports 28, 28' and 27, 27', respectively, are critical. When the cam ring is in the maximum displacement-establishing position of FIG. 2, some fluid must be discharged from the pump so the

approach ends of arcs A and B (i.e., those portions of arcs A and B which are first passed by a point on rotor 16 moving in the desired direction of rotation) must overlie the discharge port 28, 28' and 27, 27', respectively. As far as the leaving ends of the arcs A and B are concerned, there are two requirements which must be satisfied. First, in order to prevent communication between the discharge ports 28, 28' and 27, 27' and expanding intervane spaces, it is essential that the leaving ends of arcs A and B be spaced from the leaving edges of these ports in the direction of rotation. Second, in order to prevent cavitation in the intervane spaces, the leaving ends of these arcs must not be farther from the approach edges of the inlet ports 26, 26' and 25, 25', in the direction opposite to the direction of rotation, than the intervane interval. Since these two conditions must be fulfilled in all positions of the cam ring 15 within its operating range, the leaving ends of arcs A and B actually overlie the inlet ports 26, 26' and 25, 25', respectively, in the maximum displacement-establishing position of FIG. 2.

In the minimum displacement-establishing position (see FIG. 6), the leaving ends of arcs A and B are positioned between discharge ports 28, 28' and inlet ports 26, 26' and between discharge ports 27, 27' and inlet ports 25, 25', respectively. It will be apparent that this position satisfies the two conditions given above. The positions of the approach ends of arcs A and B depend on the actual displacement which is desired when the cam ring is in the minimum displacement-establishing position. In the preferred embodiment, the desired minimum displacement is zero so the approach ends of these arcs are nearly aligned with the leaving ends of inlet ports 25, 25' and 26, 26', respectively. Actually, the approach ends of the arcs are spaced from the leaving edges of the inlet ports 25, 25' and 26, 26' a distance, in the direction of rotation, equal to the width of one vane 19 in order to develop a very small flow which compensates for internal leakage in the pump. It will be apparent that, in the minimum displacement-establishing position, the approach ends of the arcs A and B can assume any positions between those shown in FIGS. 2 and 6.

When the pump is put in operation, inlet port 43 is connected with a reservoir and discharge port 48 is connected with the hydraulic circuit which is to be supplied with high pressure oil. The rotor 16 is rotated in the direction of the arrow in FIGS. 2, 5 and 6 and this causes the oil entering inlet port 43 to flow into those intervane spaces overlying arcuate inlet ports 25, 25' and 26, 26' along paths comprising annular inlet manifolds 42 and 42', and radial slots 41 and 41'. Just after communication between the intervane spaces and arcuate inlet ports 25, 25' and 26, 26' is interrupted, these spaces are connected with the arcuate discharge ports 28, 28' and 27, 27', and the inward slant of the cam surface causes oil to be discharged through these ports and bores 44' and 45', chamber 46', discharge manifold 47 and discharge port 48 into the system being supplied.

The vanes 19 are held in sealing engagement with the cam ring 15 by centrifugal force and also by fluid pressure forces. When the vanes overlie the arcuate inlet ports 25, 25' and 26, 26', the bores 21 register with ports 29, 29' and 31, 31' and thus the inner ends of the vanes are subjected to inlet pressure. The inner ends of pins 24, however, are subjected to discharge pressure which is transmitted to the bores 22 from arcuate discharge ports 27, 27' and 28, 28' along paths comprising bores 44, 44' and 45, 45', ports 32, 32' and 33, 33', radial passages 53, the clearances between the splines on rotor 16 and shaft 17, and annular chamber 23. The force developed on pins 24 is constant and small but it is sufficient to produce a seal when the vanes are in the regions of the arcuate inlet ports. When the vanes 19 overlie the arcuate discharge ports 27, 27' and 28, 28', the bores 21 register with ports 32, 32' and 33, 33' which contain fluid under discharge pressure. Therefore, an

additional fluid pressure bias is exerted on the vanes when they are within the region where large sealing forces are required.

When the cam ring 15 is in the position shown in FIG. 2, the volume of the intervane spaces leaving inlet ports 25, 25' and 26, 26' is a maximum and so the displacement of the pump is a maximum. As the cam ring 15 is rotated counterclockwise, as viewed in FIG. 2, the volumes of the intervane spaces leaving the arcuate inlet ports 25, 25' and 26, 26' progressively decrease thus causing the vanes 19 to carry progressively smaller quantities of oil to the arcuate discharge ports 28, 28' and 27, 27'. When the cam ring reaches the zero displacement-establishing position of FIG. 6, the volumes of the intervane spaces leaving the arcuate inlet ports are essentially zero (neglecting the small leakage compensating flow) and consequently the displacement of the pump is zero. Movement of the cam ring 15 in the opposite direction results in a progressive increase in displacement.

It will be observed that, because of the configuration of the cam ring, only that quantity of oil which is to be discharged through port 48 actually is transmitted from arcuate inlet ports 25, 25' and 26, 26' to the arcuate discharge ports 28, 28' and 27, 27', respectively. In no position of the cam ring (within its operating range) is oil allowed to flow from the arcuate discharge ports 27, 27' and 28, 28' into expanding intervane spaces and thence into the arcuate inlet port next in line in the direction of rotation.

As stated previously, the drawings and description relate only to the preferred embodiment of the invention. Since many changes can be made in the structure of this embodiment without departing from the inventive concept, the following claims should provide the sole measure of the scope of the invention.

What is claimed is:

1. In a variable displacement vane engine of the double lobe balanced type, including a housing, a rotor formed with a plurality of circumferentially-spaced vane slots for receiving a plurality of sliding vanes, stationary valve means located adjacent at least one end of the rotor and containing circumferentially-spaced inlet and discharge ports that lie in planes normal to the axis of the rotor and are positioned radially to register with the intervane spaces, there being two diametrically-opposed inlet ports and two diametrically-opposed discharge ports and adjacent ports being spaced from each other a distance not less than the intervane interval, and a cam ring encircling the rotor and having a cam surface which engages the outer ends of the vanes, the cam ring being mounted in the housing for substantial rotational adjustment about the axis of the rotor between minimum and maximum displacement establishing positions, the improvement which comprises a cylindrical cam surface on the cam ring which in transverse cross-section is defined by two diametrically-opposed true circular arcs which are centered on the axis of the rotor, two diametrically-opposed eccentric circular arcs which are centered at points displaced in opposite directions from the axis of the rotor, the distance from the axis of the rotor to any point on each of these arcs being greater than the radius of the true circular arcs, and four blend curves joining the adjacent ends of the four arcs, the lengths of the arcs being so selected and correlated with said rotational movement of the cam ring that in all positions of the cam ring between and including said minimum and maximum displacement establishing positions

(a) a portion of each true arc lies between one discharge port and the next succeeding inlet port in the direction of rotation,

(b) the point of maximum radius on each eccentric arc, measured with respect to the axis of the rotor, is spaced from the approach end of a discharge port a distance, measured in the direction opposite to

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the direction of rotation, not less than the intervane interval, and

(c) the approach end of each true arc lies between the leaving end of a discharge port and a point spaced from the approach end of that discharge port, in the direction opposite to the direction of rotation, a distance not greater than the intervane interval.

2. The improvement defined in claim 1 in which, in all positions of the cam ring between the minimum and maximum displacement establishing positions and including the maximum displacement establishing position, the leaving end of each true arc is located between a point overlying an inlet port and a point spaced from the approach end of that inlet port, in the direction opposite

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to the direction of rotation, a distance not greater than the intervane interval.

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