Oct. 8, 1968 B. REMINIAC ETAL 3,404,632

ROTARY VOLUMETRIC PUMP

Filed Feb. 2, 1966 4 Sheets-Sheet 1

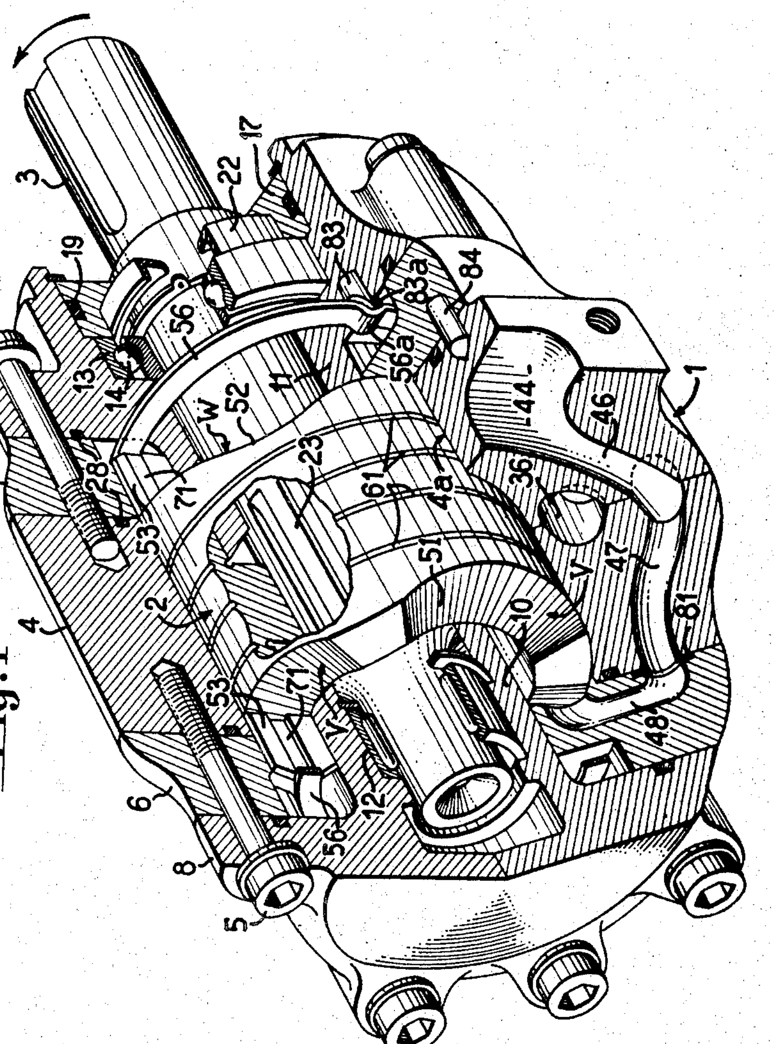

Fig. 1

ROTARY VOLUMETRIC PUMP

Filed Feb. 2, 1966 4 Sheets-Sheet 2

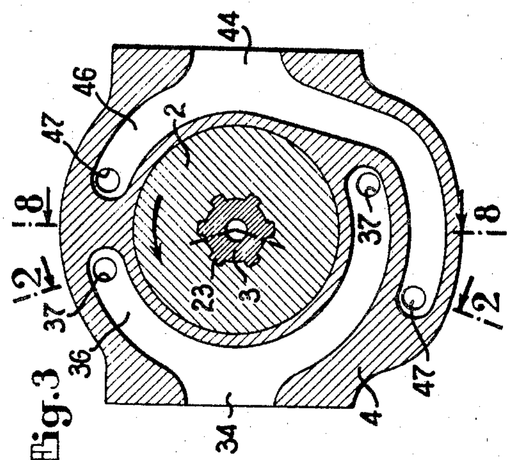

Fig. 3

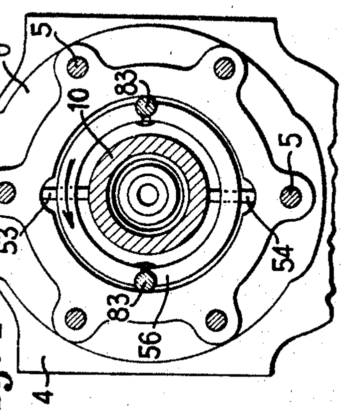

Fig. 4

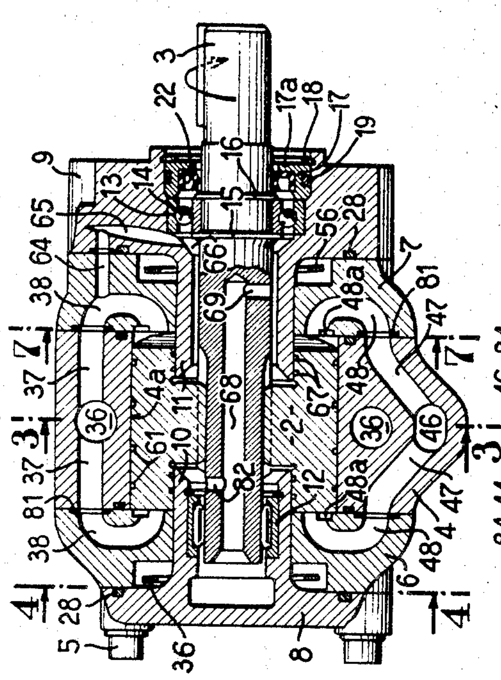

Fig. 2

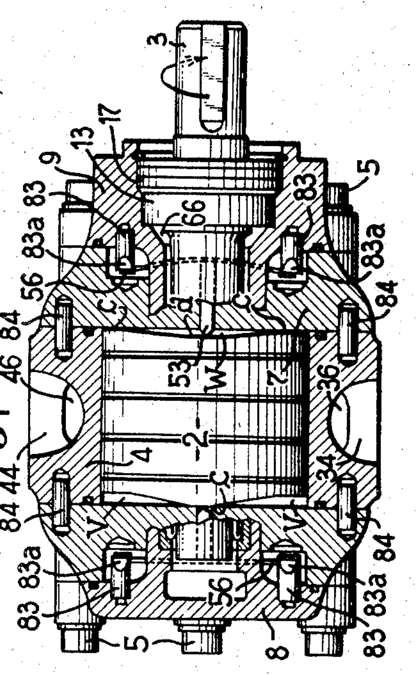

Fig. 5

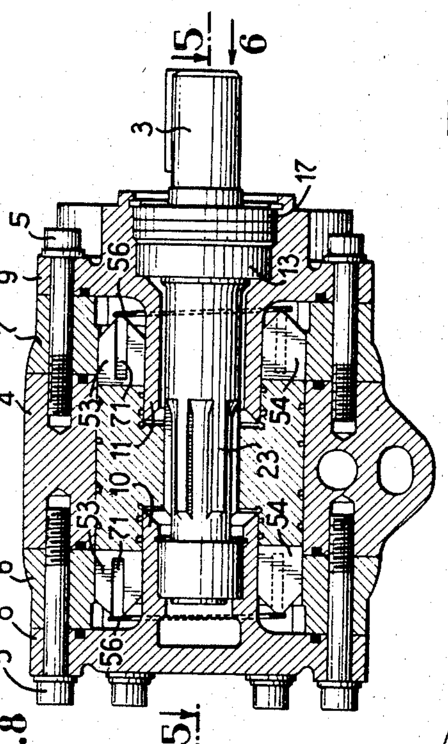

Fig.8

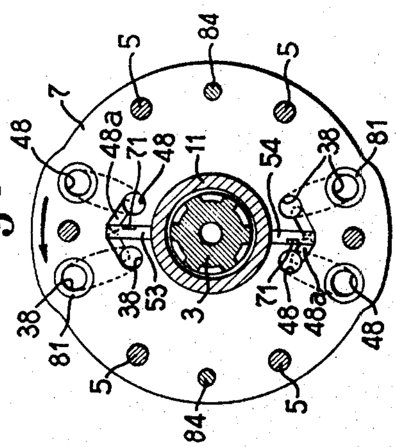

Fig.7

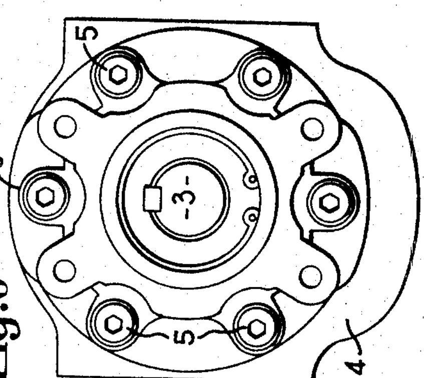

Fig.6

ROTARY VOLUMETRIC PUMP

Filed Feb. 2, 1966 4 Sheets-Sheet 4

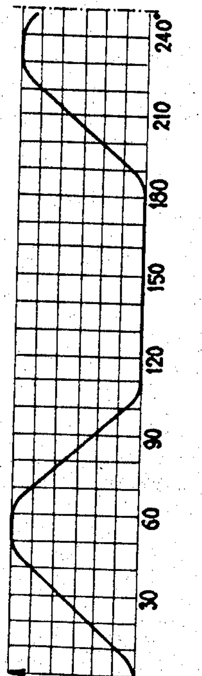

Fig. 11

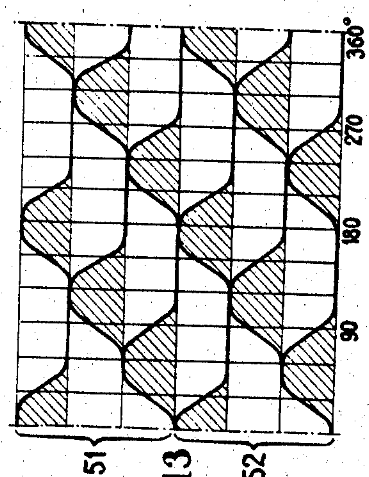

Fig. 13

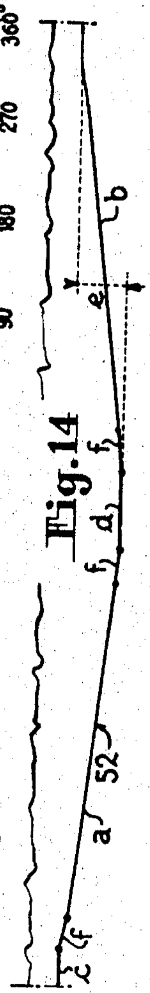

Fig. 14

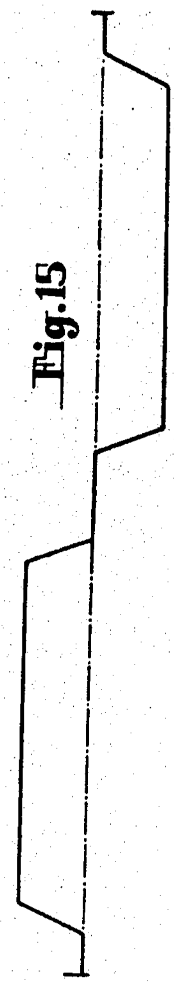

Fig. 15

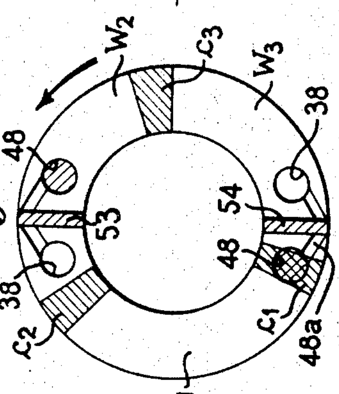

Fig. 12

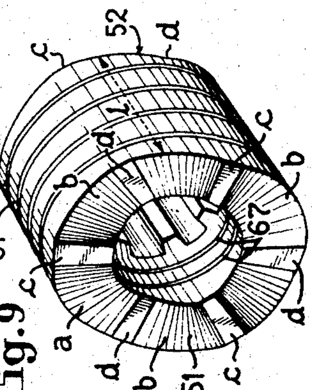

Fig. 9

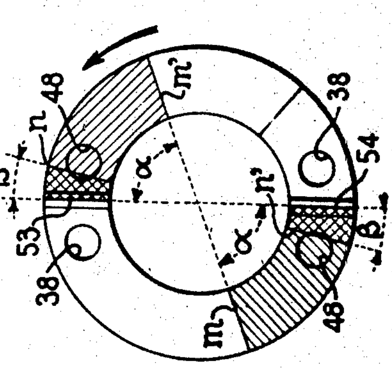

Fig. 10

United States Patent Office

3,404,632

Patented Oct. 8, 1968

1

3,404,632

ROTARY VOLUMETRIC PUMP

Bernard Reminiac, Dammarie-les-Lys, and Marcel Tournel, Chartrettes, France, assignors to Renou Dardel, Dammarie-les-Lys, Seine-et-Marne, France, a French body corporate

Filed Feb. 2, 1966, Ser. No. 524,418
Claims priority, application France, Feb. 4, 1965, 4,361
7 Claims. (Cl. 103—123)

ABSTRACT OF THE DISCLOSURE

Rotary displacement pump which can also be employed as a hydraulic motor, said pump comprising a body, a single rotor having at both ends annular cams which define with the body a plurality of pump chambers through which sweep closing members which are axially slidably mounted in the body and urged against the cam.

The object of the present invention is to provide a constant-capacity rotary volumetric pump which can also be employed as a hydraulic motor, said pump having a single rotor, at least one of the axial ends of which constitutes an annular cam which defines one or a plurality of pump chambers through which sweeps at least one valve or closing member which is axially slidably mounted in the pump body and urged against the cam.

The cam or each cam has alternately crests and hollows which are interconnected by portions of helical appearance over which the slidable valve member or each slidable valve member sweeps so that the liquid which enters a pump chamber through a port in the body located on one side of the valve member is discharged under the effect of the sweeping through a port in the body located on the other side of the valve member. By suitably selecting the number and the angular setting of the crests of the cam and of the sliding valve members—for example by providing three crests angularly spaced 120° apart and two valve members spaced 180° apart—it is possible to obtain, even with a single cam, a substantially constant instantaneous flow which affords the pump according to the invention an important advantage over, for example, pumps having axial pistons or pistons arranged around an axis whose flow is, by construction, pulsatory.

The even nature of the flow is a result already obtained in a known rotary pump employing valve members slidably mounted in the pump body. In this known pump the valve members are radial and cooperate with radial cams formed at the periphery of at least two rotors. Compared to this known pump, the pump according to the invention consequently has the advantage of a simplified construction owing to its single rotor and the possibility of obtaining direct from the foundry the fluid conduits in the stator, which is also a single stator, with the result that drilling operations are restricted to the drilling of the fixing and drainage apertures.

The pump according to the invention is also advantageous in that it is easy to assemble. As the rotating part is in one piece no angular positioning of this part is necessary. As concerns the relative axial positioning of the rotor and the parts of the pump body disposed in axially adjacent relation at both ends of the stator, the required position and the operational clearance can be obtained by adequately dimensioning the stator, this stator being therefore the sole part of the pump whose thickness must be strictly calibrated whereas in pumps having multiple radial rotors several precise axial dimensions are required between the spacer rings and corresponding rotors.

2

It is advantageous that each end of the rotor have a cam of identical contour, the two cams, which cooperate with the valve members disposed in an identical manner, being angularly offset in such manner that the projection of their contour onto the outer cylindrical face of the rotor describes two parallel curves defining segments of a generatrix of equal length. This arrangement affords in particular the following advantages:

The axial forces produced in pump operation under the effect of the pressures exerted on the cams by the fluid are dynamically balanced.

It is possible to machine both cams on the rotor by means of radial tools having a constant spacing mounted on a common tool carrier which undergoes an axial motion of translation in coordination with the rotation of the rotor.

Further features and advantages of the invention will be apparent from the ensuing description with reference to the accompanying drawings to which the invention is in no way limited.

In the drawings:

FIG. 1 is a perspective view of a pump according to the invention in which various cut-away portions show the construction of the stator and rotor;

FIG. 2 is an axial sectional view taken along line 2—2 of FIG. 3;

FIGS. 3 and 4 are sectional views taken along lines 3—3 and 4—4 respectively of FIG. 2;

FIG. 5 is an axial sectional view in plan of the pump, the upper portion of the stator having been removed;

FIG. 6 is an end elevational view of the driving shaft end of the pump showing the manner of securing the pump;

FIG. 7 is a sectional view taken along line 7—7 of FIG. 2;

FIG. 8 is a sectional view taken along line 8—8 of FIG. 3;

FIG. 9 is a perspective view of the pump rotor;

FIG. 10 is a diagrammatic view showing the dynamic balancing of the pump;

FIG. 11 is a diagram indicating the flow curve of one pump chamber;

FIG. 12 is a diagram showing a particular relative position of the cam and spacer plates;

FIG. 13 is a diagram showing in superimposed relation the flow curves of individual chambers;

FIG. 14 is a diagram showing the development of the profile of a part of a cam, and

FIG. 15 shows the curve of the velocities of a valve member cooperating with the cam shown in FIG. 14.

The pump illustrated in the drawing comprises, within a body **1** of generally cylindrical shape, a single rotor **2** driven by a shaft **3**.

The body—disposed at each end of a stator **4** in the bore **4***a* of which the rotor **2** is engaged—includes two plates **6**, **7** constituting spacer members which are positioned with respect to the stator by centering dowels **84** (FIG. 5), and end members **8**, **9** which extend inwardly and constitute sleeves **10**, **11**, the assembly being held together by screws **5**. The shaft **3** is journalled inside the sleeve **10** of the member **8** by a needle rolling bearing **12** and in a counterbore **13** of the member **9** by a ball rolling bearing **14**. Applied against the latter is a flange **15** of the shaft **3**, this bearing being retained by a retainer ring **16** and a flanged collar **17** which is held in position by a ring **18** (FIG. 2). The collar **17** also acts as sealing means owing to the provision of a flexible ring **19** inserted in a recess in the outer periphery of the collar and an annular membrane **22** which is located inside the collar backed by the flange **17***a* and slidably applied against the shaft.

The rotor 2 is prevented from rotating on the shaft 3 by a key 23 (FIG. 3) and is axially located by the lateral faces of the spacer plates 6, 7 which are sealed relative to the adjacent faces of the body 1 by rings 28. The precise axial position of these lateral faces, on which depends the operational clearance for the rotor 2, is obtained by calibrating the thickness of the stator 4.

The fluid to be pumped enters by way of an aperture 34 provided in one of the vertical faces of the stator 4, flows into a part-circular conduit 36 (FIGS. 2 and 3) and is distributed in two diametrically opposed axial manifolds 37 the ends of which communicate with conduits 38 which are formed in the plates 6 and 7 and supply the pump chambers. As concerns the delivery or discharge, it is effected by an assembly comprising conduits 48, manifolds 47, a part-circular conduit 46, and a discharge aperture 44 which correspond respectively to the elements 38, 37, 36, 34 of the inlet system. Rings 81 provide a seal between the manifolds 37 and the conduits 38 and between the manifolds 47 and the conduits 48. As can be seen in FIG. 5 the discharge aperture is provided at a point on the stator diametrically opposed to the point at which the fluid is admitted.

The pumping is effected by two radial faces constituting annular cams 51, 52 formed on the axial ends of the rotor 2 and each cooperating with two diametrically opposed closing or valve members 53, 54 which slide axially in the spacer plate 6 or 7 in contact with the wall of the bore 4*a* of the stator 4. Each pair of valve members 53, 54 is biased against the corresponding cam by an annular spring 56 which is mounted to oscillate about an axis perpendicular to the line intersecting the valve members 53, 54 by means of two diametrically opposed studs 83 in the side member 8 or 9 and two V-shaped recesses 56*a* in the spring 56, the bottom of each recess bearing against the chamfered end 83*a* and the corresponding stud 83. The cam-contacting end of the valve members 53, 54 has preferably a semi-cylindrical shape. The profiles of the two cams 51, 52 are identical and consist of six arcs of a helix *a*, *b* of equal length, the pitch of which is the same but changes sign from one arc to the following, these arcs being interconnected by six flat portions *c* or *d* spaced 60° apart, which gives a developed curve (FIG. 14) resembling a sinusoid having fllattened crests *c* and hollows *d*. The flat portions *c* and *d* thus define two radial planes which are axially spaced apart a distance *e*. Formed between the plane intersecting the crests *c* which coincide (ignoring the operational clearance) with the inner end face of the spacer plate 6 or 7 and the contour of each cam are three pump chambers *v* or *w* through which the valve members 53, 54 sweep when the rotor rotates. In order to regulate the acceleration or deceleration of the valve members and to avoid an instability of the springs 56 which bias the valve members against the cam, each flat portion *c* or *d* is connected to the adjacent arc of a helix *a* or *b* by a suitable curved portion *f* (FIG. 14).

As can be seen more particularly in FIG. 9, the profiles of the two cams 51, 52 are angularly offset 60° from each other so that the two profiles form two parallel faces. Thus these faces can be machined simultaneously by a single tool carrier supporting two radial tools spaced apart a distance *l* corresponding to the generatrices defined on the cylindrical outer face of the rotor by the intersection therewith of the cam profiles.

In order to afford a radial balancing of the rotor 2 and relieve the pressure of the leakage liquid from the pump chambers *v* and *w* four grooves 61 are provided in the outer periphery of the rotor in contact with the bore 4*a* of the stator. Likewise, two groups of two grooves 67 are provided on the inner periphery of the rotor in contact with the sleeves 10 and 11.

The leakage flow on the outer periphery of the rotor is automatically conducted to the suction pump chambers. The leakage flow on the inner periphery of the rotor is drained towards the part-circular suction conduit 36. Connected to the latter through conduits 37, 38 (FIG. 2) is a drainage conduit 64 which is connected to a radial conduit 65 in the end member 9 for receiving at 66, in the vicinity of the ball bearing 14, the leakage liquid flowing from the pump chambers along the cylindrical parts of the rotor in contact with the sleeves 10 and 11, whence the liquid flows back along the shaft 3 and finally reaches the free annular space 66 around the part of the shaft 3 adjacent the ball bearing 14, either directly or in flowing through in succession, the needle bearing 12, the passageway 82, the axial passageway 68, the radial passageway 69, these three passageways being formed in the shaft 3.

It will be observed that the liquid from the pump chambers also flows along the sides of the valve members 53, 54 (biased by the springs 56) owing to the provision of an axial recess 71 in each valve member. This arrangement assists the springs 56 in pump operation.

It should be noted that these annular springs undergo no stress variations, since they are pre-stressed on assembly and rock on the studs 83 and therefore do not bend, owing to the alternating movement of the diametrically disposed valve members 53 and 54.

The pump just described operates in the following manner:

The fluid entering by way of the suction aperture 34 flows into the conduit 36, then into the two manifolds 37 which supply the two axially opposed groups of three pump chambers *v* and *w* by way of the conduits or ports 38. As can be seen in FIGS. 7 and 10 these conduits open onto the sweep path of the cam behind the valve members 53 and 54, with respect to the direction of rotation of the rotor 2 indicated by the arrow, whereas the discharge or delivery conduits or ports 48 communicate with the sweep path of the cams in front of the valve members.

It has been assumed in the diagram shown in FIG. 11 that the point of origin of the angular motion of the rotor 2 corresponds to a position of one of the crests c_1 of the cam on the right side of the rotor in respect of which there is coincidence with the lower discharge conduit or port 48, the pump chamber w_1 located at the rear of this crest c_1 being filled with fluid owing to the supply of fluid thereto which occurred previously in the course of the passage of the chamber w_1 in front of the upper inlet conduit or port 38. As soon as the illustrated position has been passed through the fluid of the chamber w_1 starts to be fed through the discharge conduit 48, thence into the manifold 47 and the conduit 46 until it reaches the discharge aperture 44 (FIG. 3). To be more precise, if it is assumed that the flat portions *c* and *d* of the cam correspond to an angle subtended at the centre of 10° and that the conduits are offset 20° relative to the valve members, it will be clear, as shown in the graph of FIG. 11, that the flow in the discharge circuit of the chamber w_1 increases between the zero angular position shown and the position of 50° to which corresponds the passage across the port 48 of the hollow *d* of the portion of the cam defining the chamber w_1. The flow remains constant over an angular distance of 10° and then decreases for the angular positions of *c* between 60° and 110°. For this latter position, the discharge stops owing to the passage of the remaining volume of fluid between the discharge port 48 and the valve member 54 by way of the recess 48*a*. The flow remains zero between the positions 110° and 180° of the crest c_1, a new supply of fluid to the chamber w_1 having started in front of the valve member 54 for the angular position 20° and terminating in the position 140°. In the position 180° of the crest c_1 a discharge cycle identical to the preceding cycle once again starts for the chamber w_1 by utilization of the upper discharge conduit 48 located in front of the valve member 53. Thus there are two discharges for each rotation of the rotor in respect of each pump chamber *v* or *w* and this means that the capacity of the pump is twelve times the volume of one elementary chamber *v* or *w*.

FIGURE 13 shows the juxtaposition of six flow curves of respective chambers which are identical to the curve shown in FIGURE 11 but offset from each other in accordance with an arrangement providing a constant instantaneous discharge or delivery of the pump, namely a non-pulsatory discharge. At each instant, the flow in respect of a cam of the rotor is equal to the value of the maximum flow of a pump chamber, that is, either a single chamber discharges in the zone of the apex of its curve or the flows of two chambers complement each other so as to give this constant value.

The diagram shown in FIGURE 10 is intended to show that the axial thrusts due to pumping pressures exerted on the rotor are always balanced. In the illustrated angular position of the rotor (which has been chosen arbitrarily), the referances *m, n and m', n'* represent the crests of the left and right cams respectively which define the rear limits of the chambers in which the discharge is in course. It is clear from this diagram that the discharge pressure is exerted on the rotor in one direction, in respect of the left cam, on an angular segment of angle α between the crest *m* and the valve member **54** and on an angular segment of angle β between the crest *n* and the valve member **53**, and, in the opposite axial direction in respect of the right cam, on an angular segment α between the crest *m'* and the valve member **53** and on an angular segment β between the crest *n'* and the valve member **54**. Thus the pressures are balanced in the illustrated position—as it is in any other angular position of the rotor.

FIGURE 15 is a diagram of the linear velocity of a sliding valve member. It is clear that the velocity is constant in the passage of the valve member in the region of the helical arcs *a* or *b* but that the velocity changes direction in the region of the crests and hollows of the adjacent cam.

Although specific embodiments of the invention have been described, many modifications and changes may be made therein without departing from the scope of the invention as defined in the appended claims.

Thus the pump according to the invention can be constructed with a single cam on the rotor, the axial thrust then exerted on the rotor being compensated by a reaction device or thrust bearing employing mechanical or hydraulic means.

Further, a pump according to the invention having two cams and ensuring the balancing of the axial forces could be provided with a hydraulic compensating device on one side or on both sides of the rotor for automatically taking up clearances or play thereby resulting in performances which are unaffected by wear of the component parts and consequently strictly constant over a period of time.

It is also clear that the pump according to the invention could comprise in the region of each cam, or of the single cam, of the rotor a different number of pump chambers cooperating with a different number of sliding valve members.

As concerns the contour of the cams **51** and **52**, from a technological point of view, it might be difficult to generate the helical profile of the operative faces defined by the arcs *a* or *b* by means of a cylindrical rotary grinding wheel whose spindle intersects the axis of the rotor and is perpendicular to the latter, this radial grinding wheel embodying so to speak the generatrices of the helicoid. A different manner of generating the profile of the cams could however be employed, and has in fact been employed in the construction of a prototype. It comprises imparting to the rotor blank a helical motion and machining the end faces of the blank by means of a rotary circular grinding wheel in the form of a disc whose plane approximately coincides with a plane intersecting the axis of the rotor and which effects alternating radial travels in this plane. In this method, the radial generatrices of the cam faces obtained between the flat portions *c* and *d* are not rectilinear and have a slightly helical appearance. The contact of the part-cylindrical end of the valve members **53**, **54** with the corresponding cam then occurs on a curve of the face of the cam which is inscribed on the cylinder of said end so that the contact is fluidtight. In this respect it is advantageous that the grinding edge of the grinding wheel has a profile of a semi-torus whose radius is equal to that of the part-cylindrical profile of the contacting end of the valve members.

Having now described our invention what we claim as new and desire to secure by Letters Patent is:

1. Rotary displacement pump, also of utility as a fluid motor, said pump comprising a fixed part and a rotating part, said fixed part comprising a pump body having an axis, said rotating part consisting of a single rotor mounted in said body to rotate about said axis and means for driving said rotor in rotation, said rotor having at each end thereof an annular face cam, said body having inner annular end faces respectively cooperative with said cams, each cam having a progressively varying profile defining recess portions and crests, said crests being in sliding contact with said annular end faces whereby said recess portions of each cam define with said body a plurality of pump chambers which rotate with said rotor, closing members mounted in said body to slide in a direction substantially parallel to said axis and cooperative with said cams, elastically yieldable means biasing said closing members against said cams, whereby upon rotation of said rotor, said closing members sweep through said rotating pump chambers, and inlet ports and discharge ports provided in said body in positions to communicate with said chambers as said rotor rotates.

2. Rotary displacement pump as claimed in claim **1**, wherein said crests of the cam are plane.

3. Rotary displacement pump as claimed in claim **1**, wherein the means biasing each closing member is a spring.

4. Rotary displacement pump as claimed in claim **1**, comprising two diametrically opposed closing members coacting with each cam, said biasing means comprising a single annular spring mounted in the body to rock about an axis perpendicular to the diametrical plane intersecting the closing members and resiliently bearing against the diametrically opposed closing members, whereby the spring is not subjected to variations in bending stress.

5. Rotary displacement pump as claimed in claim **1**, wherein said inlet ports and discharge ports are respectively located adjacent and on opposite sides of said closing members.

6. Rotary displacement pump as claimed in claim **5**, wherein each closing member comprises a recess adjacent the discharge port for the passage of fluid under pressure from the pump chambers to the end of the closing member remote from the end thereof in contact with the cam, whereby the pressure of said fluid biases the closing member into contact with the corresponding cam.

7. Rotary displacement pump, also of utility as a fluid motor, said pump comprising a fixed part and a rotating part, said fixed part comprising a pump body having an axis, said rotating part consisting of a single rotor mounted in said body to rotate about said axis and means for driving said rotor in rotation, said rotor having at each end thereof an annular face cam, said body having inner annular end faces respectively cooperative with said cams, each cam having a progressively varying profile defining three recess portions and three plane crests spaced 120° apart, said crests being in sliding contact with said annular end faces whereby said recess portions of each cam define with said body three pump chambers

which rotate with said rotor, four closing members mounted in said body to slide in a direction substantially parallel to said axis, two of said closing members being diametrically opposed and cooperative with one of said cams and two of said closing members being diametrically opposed and cooperative with the other of said cams, elastically yieldable means biasing said closing members against said cams, whereby upon rotation of said rotor, said closing members sweep through said rotating pump chambers, and an inlet port and a delivery port provided in said body adjacent each of said closing members and on opposite sides of said closing members and communicating with said chambers as said rotor rotates.

References Cited

UNITED STATES PATENTS

8,893	4/1852	Spalding et al.	103—123
878,600	2/1908	Berrenberg	103—123
1,686,767	10/1928	Saxon	123—14
1,936,467	11/1933	Deubel	103—123
2,672,099	3/1954	Deubel	103—123
2,812,719	11/1957	Nash	103—123
2,925,779	2/1960	Pelladeau	103—123

FRED C. MATTERN, Jr., *Primary Examiner.*

WILBUR J. GOODLIN, *Assistant Examiner.*