(19) Europäisches Patentamt
European Patent Office
Office européen des brevets

(11) Publication number: 0 053 868
A2

(12) EUROPEAN PATENT APPLICATION

(21) Application number: 81301961.9

(22) Date of filing: 05.05.81

(51) Int. Cl.³: F 04 C 2/04
F 04 C 5/00

(30) Priority: 06.12.80 JP 172478/80

(43) Date of publication of application:
16.06.82 Bulletin 82/24

(84) Designated Contracting States:
AT BE CH DE FR GB IT LI LU NL SE

(71) Applicant: Ito, Kazuichi
1463-4, Ichibu-cho
Ikoma-shi Nara-ken(JP)

(72) Inventor: Ito, Kazuichi
1463-4, Ichibu-cho
Ikoma-shi Nara-ken(JP)

(74) Representative: Orr, William Maclean et al,
Haseltine Lake & Co Hazlitt House 28 Southampton
Buildings Chancery Lane
London WC2A 1AT(GB)

(54) Nutating piston pump.

(57) A rotary pump having an eccentric rotor (14) adapted to effect eccentric motion relative to the axis of the pump shaft (7) within a pump casing (1) to cause pumping action in a pump chamber. The pump comprises a pump casing (1), a pump shaft (7) mounted on the pump casing, a pair of eccentric rotors (14) mounted on the pump shaft (7) with angular spacing apart of 180°, a pair of diaphragms (24) on the eccentric rotors (14) to define a pair of pump chambers P, a partition member (36) adapted to divide the pump chamber into a suction chamber (Pi) and an exhaust chamber (Pe), and an intake port (42) and an exhaust port (43) in communication with the suction and exhaust chambers, respectively, the intake port being cut off from its communication with the exhaust port by the partition member during rotation of the pump shaft.

EP 0 053 868 A2

0053868

ROTARY PUMP

The present invention relates to a rotary pump having an eccentric rotor adapted to make an eccentric movement relative to the axis of the pump shaft within a pump casing, so as to cause a pumping action in a pump chamber formed between a diaphragm mounted on the outer periphery of the eccentric rotor and an inner peripheral surface of the pump casing.

Description of Prior Art

In the rotary pump of the type as described, the pumping action is performed by an eccentric movement of the eccentric rotor so that the fluid discharged from the pump chamber is disadvantageously pulsated, and a diaphragm is deformed by the eccentric movement of the eccentric rotor to form the pump chamber, which causes an amount of deformation of the diaphragm to be varied whereby the volume of the pump chamber cannot be maintained constant to bring forth a change in quantity of discharge of the fluid. In addition, there involves a problem in sealing between the diaphragm and the pump casing.

It is therefore a primary object of the present invention to provide a rotary pump of the aforementioned type wherein a pair of pump chmabers are formed within a pump casing, a pumping action is imparted to the pump chambers by means of an eccentric rotor different in phase by 180 degrees, and fluids which are discharged from each of the pump chambers and are different from each other in pulsation phase by 180 degrees, are merged together to thereby enable decrease in pulsations.

It is another object of the present invention to provide a rotary pump wherein an outer peripheral edge of a diaphragm mounted on the outer periphery of an eccentric rotor is firmly fixed on the pump casing to maintain at all times an amount of deformation of the diaphragm due to the eccentric movement of the eccentric rotor substantial constant thus providing a constant volume of the pump chamber and improving the sealing therebetween.

It is a further object of the present invention to provide a rotary pump wherein wearing of a wedge member interposed between a pump shaft and an eccentric rotor is decreased to be as small as possible, and a pressing force of the wedge member against the eccentric rotor resulting from rotation of the pump shaft is increased so that a power may be transmitted efficiently from the pump shaft to the eccentric rotor.

BRIEF DESCRIPTION OF THE DRAWINGS

The drawings show embodiments of a rotary pump in accordance with the present invention, in which:

Fig. 1 is a longitudinal sectional side view of a rotary pump;

Fig. 2 is a sectional view taken along the lline II-II of Fig. 1;

Fig. 3 is a sectional view taken along the line III-III of Fig. 1;

Fig. 4 is an illustration of operation of the rotary pump in accordance with the present invention; and

Fig. 5 shows characteristic curves describing quantities of fluids respectively discharged from two pump chambers due to the rotation of a pump shaft.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

One embodiment of the present invention will be described hereinafter with reference to the drawings. Referring now to Fig. 1, a pump casing 1 comprises a case body 3 composed of a pair of hollow cylindrical case halves 2, 2, and a pair of cover members 4, 5 adapted to close openings disposed at both ends of the case body 3, the case halves 2, 2 and the cover members 4, 5 being formed of synthetic resins such as polypropylene having chemical resisting properties and being integrally connected with each other by means of a plurality of connecting bolts 12. A pump shaft 7 is inserted into a central portion of a cylindrical hollow chamber 6 defined within the pump casing 1, the pump shaft 7 being rotatably supported at its middle portion and inner end portion on the cover members 4, 5 through bearings 8, 9 and being connected to a prime mover (not shown) at its outer end extending ourward from one cover member 4 through a seal 10 so that the shaft 7 may be rotatably driven by the prime mover.

Within the pump casing 1, the pump shaft 7 is formed with a pair of recessed grooves 11, 11 in axially spaced relation and circumferentially displaced in phase by 180 degrees, the recessed grooves 11, 11 having, as may be clearly shown in Fig. 2, driving planes 13, 13 disposed parallel to the central axis O of the pump shaft 7, and a pair of eccentric rotors 14, 14 internally peripherally provided with needle bearings 15, 15 are arranged to surround the pump shaft 7 at the position of the recessed grooves 11, 11.

Wedge members 16, 16 are interposed between the driving planes 13, 13 of the recessed grooves 11, 11 and inner peripheral surfaces 19, 19 of the needle bearings 15, 15, the wedge member 16 comprising a wedge roller 18 having a driven plane 17 bearing on the driving plane 13 of the recessed groove 11 and a clutch element 21 being fitted in the outer periphery of the wedge roller 18 and having a circular surface 20 having the same curvature as of the inner peripheral surface 19 of the needle bearing 15 and in intimate contact with the latter over its entire periphery. The outer peripheral surface of the wedge member 16, that is, the circular surface

20 in the outer periphery of the clutch element 21 extends from the outer peripheral surface of the pump shaft 7 to cause the center A (Fig. 4) of the eccentric rotor 14 to be eccentric in a predetermined amount E relative to the central axis O of the pump shaft 7 through the needle bearing 15.

Accordingly, as the pump shaft 7 rotates, the pair of eccentric rotors 14, 14 effect their eccentric movement about the pump shaft 7 with a phase difference of 180 degrees through the wedge members 16, 16 and the needle bearings 15, 15 but the rotative force of the pump shaft 7 is not transmitted to the eccentric rotors 14, 14 by the action of the needle bearings 15, 15.

Also, since the center O' of the wedge roller 18 is arranged to be deviated from the center A of the needle bearing 15, the clutch element 21 is subjected to a reaction from the inner peripheral surface 19 of the needle bearing 15 as the pump shaft 7 rotates and tends to pivot about the center O' of the wedge roller 18. However, since the center O' of the wedge roller 18 is deviated from the center A of the needle bearing 15 as described, the needle bearing 15 is subjected to a great radial pressing force from the clutch element 21 by said pivotal movement of the clutch element 21, thus decreasing a slip therebetween to minimize the wear.

The eccentric rotor 14 is wholly formed into a channel section from a pair of annular members 22, 22 having an L-shaped section, in outer peripheral portions of which are formed inwardly extending annular projections 23, 23. An annular diaphragm 24 which is formed of a resilient material such as rubber and has an outer peripheral surface applied with teflon baking is radially slidably mounted on the outer periphery of the eccentric rotor 14, the disphragm 24 being slidably fitted in an annular hollow portion 25 of the eccentric rotor 14 of channel section and comprising a disc portion 27 provided with an annular engaging projection 26 engageable with an annular projection 23 formed in the outer periphery of the rotor 14 and an annular flange portion 28 integrally formed with the disc portion 27 at its external. The annular flange portion 28 is at its end edge formed with radially outwardly extending annular engaging projecting elements 29, 29, which are brought into engagement with annular engaging grooves 30, 30 formed in both ends of each case half member 2 and are firmly fixedly held in position by means of annular stepped portions 31, 31 formed internally of the cover members 4, 5 and an annular stepped portion 33 in the outer periphery of a clamping ring 32 arranged between both the diaphragms 24 and 24. In this case, the engaging projecting elements 29, 29 formed on the outer periphral edge of the annular flange portion 28 of each diaphragm 24 are provided with engaging surfaces 29', 29' for prevention of disengagement. Each engaging groove 30 of each case half member 2 and each annular stepped portion 33 in the outer periphery of the clamping ring 32 are also formed with an engaged surface 30' adapted to engage with said engaging surface 29' so that upon assembly of the pump casing 1, the engaging surface 29' is forced into engagement with the engaged surface 30' whereby the engaging projecting element 29 is immovably engaged by the engaging groove 30. Thus, upon the eccentric motion of the eccentric rotors 14, 14, the outer peripheral surfaces of the annular flange portions 28, 28 of the diaphragms 24, 24 are urged against the inner peripheral surfaces of the cylindrical case halves 2, 2 by means of the wedge members 16, 16 through the eccentric rotors 14, 14. By engagement of the engaging projections 26, 26 in the inner peripheries of the disc portions 27, 27 of the diaphragms 24, 24 with the annular projections 23, 23 in the outer peripheries of the eccentric rotors 14, 14, the central portions in the outer peripheries of the annular flange portions 28, 28 of the diaphragms 24, 24 are gradually strongly pulled inwardly as they move away in a peripheral direction from a point urged against the inner peripheral surfaces of the cylindrical case halves 2, 2 resulting from the eccentric motion of the eccentric rotors 14, 14 and then gradually move away greatly from the inner peripheral surfaces of the cylindrical case halves 2, 2 to form pump chambers P, P of crescent-shaped cross section between the inner peripheral surfaces of the cylindrical case halves 2, 2 and the outer peripheral surfaces of the diaphragms 24, 24, respectively. In this case, the diaphragm 24 is pulled towards the center upon eccentric rotation of the eccentric rotor 14, but since the engaging projecting elements 29, 29 on the both end edges of the annular flange portion 28 are fixedly held by the annular engaging grooves 30, 30 of the case half 2, the annular stepped portion 31 of the cover body 4 or 5 and the annular stepped portion 33 of the clamping ring 32, no leakage of fluid within the pump chamber P occurs therefrom or no entry lubricating oil in the bearings 8, 9 and needle bearings 15 into the pump chamber P occurs.

The case halves 2, 2 have partition grooves 35, 35 bored therein to open to the cylindrical hollow chamber 6, the partition grooves 35, 35 having partition members 36, 36 extended integrally with the annular flange portions 28, 28 of the diaphragms 24, 24, said partition members 36, 36 being radially slidably fitted in the partition grooves 35, 35 in fluid-tight manner. The crescent-shaped pump chambers P, P formed between the inner peripheral surafces of the case halves 2, 2 and the outer periperal surfaces of the diaphragms 24, 24 are divided by the partition members 36, 36 into an intake chamber Pi and an exhaust chamber Pe in fluid-tight manner, as shown by the two-dot contour lines in Fig.2.

Intake and exhaust passages 39 and 40 axially spaced from each other and extending in opposite directions are open to each of the partition grooves 35, the two exhaust passages 40, 40 being placed in communication with each other. The partition member 36 is inclined relative to a plane 41 which passes through the central axis of the cylindrical member formed by the outer peripheral surface of the diaphragm 24, that is, the central axis A of the eccentric rotor 14. The partition member 36 is formed at both sides thereof with intake and exhaust ports 42, 43 which are respectively open to the intake and exhaust chambers Pi, Pe and in communication with the intake and exhaust passages 39, 40, respectively. The intake and exhaust ports 42, 43 are arranged so that the intake port 42 is on the side of the exhaust chamber Pe and the exhaust port 43 is on the side of the intake chamber Pi with respect to the plane which passes through the central point between the ports and the central axis O of the cylindrical hollow chamber 6, that is, the plane 41 which passes through a tangent between the outer peripheral surface of the diaphragm 24 and the inner peripheral surface of the case half member 2 to define the pump chamber P within the cylindrical hollow chamber 6 when the diaphragm 24 is positioned shown by the solid line in Fig. 2. With the arrangement of the intake and exhaust ports 42, 43 as just mentioned, the communication between the intake and exhaust ports 42, 43 may be cut off at all times during rotation of the pump shaft 7 to reduce the pulsation of fluids supplied under pressure from the pump as small as possible, as will be described hereinafter.

While the intake and exhaust ports 42, 43 have been arranged as shown in Fig. 3, it will be understood that they can also be arranged in any other manner as long as the communication therebetween may be cut off at all times during rotation of the pump shaft 7.

In the drawings, reference numeral 44 designates a spacer interposed between the pair of needle bearings 15 and 15 and between the pair of eccentric rotors 14 and 14.

Next, the operation of the aforementioned embodiment will be described.

When the pump shaft 7 is rotated counterclockwise (in the direction as indicated by the arrow in Fig. 2) as viewed in Fig. 2 by a prime mover not shown, the torque thereof is transmitted to the wedge members 16, 16 through the driving planes 13, 13 of the recessed grooves 11, 11 and further to the needle bearings 15, 15 through the wedging action of the wedge members 16, 16. At this time, the eccentric force transmitted to the needle bearings 15, 15 is transmitted to the eccentric rotors 14, 14 without modification but the torque is not transmitted to the eccentric rotors 14, 14 while being absorbed by the needle bearings 15, 15. As a result, the eccentric rotors 14, 14 revolve with the phase difference of 180 degrees from each other within the cylindrical hollow chamber 6 eccentrically with respect to the pump shaft 7 whereby one part of the central portion of the annular flange portions 28, 28 in the outer periphery of the pair of diaphragms 24, 24 mounted on the outer periphery of the eccentric rotors 14, 14 is pressed against the inner peripheral surface of the case halves 2, 2 by the wedge members 16, 16 while the other part of the central portion of the annular flange portions 28, 28 is pulled toward the center to form the crescent-shaped gaps, that is, the pump chambers P, P between the outer peripheral surface of the annular flange portions 28, 28 and the inner peripheral surface of the case halves 2, 2. Since the rotational phase of the eccentric rotors 14, 14 is deviated by 180 degrees, the phase of the pump chambers P, P is similarly deviated by 180 degrees. Each pump chamber P is formed integral with the outer periphery of the diaphragm 24 and divided into the intake chamber Pi and the exhaust chambers Pe in fluid-tight manner by the partition member 36 fitted in the partition groove 35 of each case half member 2. The volumes of these intake and exhaust chambers are changed by the eccentric rotational motion of the eccentric rotors 14,14 to cause a pumping action. In the state in which each eccentric rotor 14 is positioned at the top dead center of its stroke which is closest of the intake port 42 and exhaust port 43, i.e., the intermediate position between the intake and exhaust ports 42, 43, the intake chamber Pi leading to the intake port 42 is minimized in volume and is just to commence its stroke, whereas the exhaust chamber Pe leading to the exhaust port 43 is expanded in volume to the maximum and is just to commence its delivery. As the eccentric rotor 14 is rotated from this state in the manner described before, the volumes of the intake chamber Pi is gradually increased so as to suck the fluid thereinto whereas the exhaust chamber Pe is gradually decreasedin its volume to come into communication with the exhaust port 43, so that the fluid in the chamber Pe is compressed and delivered under pressure through the exhaust port 43. Then, as the eccentric rotor 14 reaches the bottom dead center which is farthest from the intake and exhaust ports 42, 43, the intake chamber Pi and the exhaust chamber Pe are just on the median position of the intake and exhaust strokes, respectively, where the volumes thereof come substatially equal to each other. As the eccentric rotor 14 further rotates, the intake chamber Pi further increases its volume, while the exhaust chamber Pe decreases its volume, so that these chambers come to resume the state as shown by the solid line in Fig. 2, thus completing the suction and delivery strokes. One cycle of suction and delivery strokes may be achieved by one rotation of the eccentric rotor 14, and a fluid of a volume equal to the volume of the pump chamber P is exhausted in one cycle.

As can be seen from Fig. 3, the communication between the intake and exhaust ports 42, 43 is always in cut-off state during the revolution of the eccentric rotor 14, and the suction and delivery strokes are never interrupted by the communication therebetween. Thus, the suction stroke and the delivery stroke are respectively continuously accomplished.

Next, referring to Fig. 4, the discharges flow rate ω_1 representative of the quantity of discharge per unit time of one pump chamber P is obtained relative to the rotational angle θ of the pump shaft 7. The center of the eccentric rotor 14, i.e. the center A of the outer peripheral circle C in the central portion of the annular flange portion 28 of the diaphragm 24 rotates at a uniform speed relative to the axis O of the pump shaft 7 and therefore, the discharge flow rate ω_1 is proportional to the distance y from the intersection D between the outer peripheral circle C and the line OO" (Y axis) connecting the center O" of the exhaust port 43 to the axis O of the pump shaft 7, to the exhaust port 43. Namely, the following relation is established between the discharge flow rate ω_1 and the distance y:

$\omega_1 = k\ y$ (k is the proportional constant)

As is apparent from Fig. 4, the distance y is represented by

$y = R - OD$

R: Radius of the cylindrical hollow chamber 6

OD: Distance from the center O (the axis of the pump shaft 7) of the chamber 6 to the point D.

And,

$OD = OB + BD$

OB: Component in the direction of Y-axis of the line AO connecting the center A of the eccentric rotor 14 to the center O of the cylindrical hollow chamber 6

BD: Component in the direction of Y-axis of the line AD connecting the center A of the eccentric rotor 14 to the point D

$OB = E \cos \theta$

E : Eccentricity (distance from the center A of the eccentric rotor 14 to the axis O of the pump shaft 7)

θ : Rotational angle of the eccentric rotor 14 or pump shaft 7

$$BD = \sqrt{r^2 - AB^2} = \sqrt{r^2 - E^2 \sin^2 \theta} = r\sqrt{1 - \frac{E^2}{r^2} \sin^2 \theta}$$

r : Radius of the outer peripheral circle C in the central portion of the diaphragm 24

AB : Vertical distance from the center A of the eccentric rotor 14 to the Y-axis.

Accordingly,

$$y = R - \left\{E \cos \theta + r \sqrt{1 - \frac{E^2}{r^2} \sin^2 \theta}\right\}$$

However, since the eccentricity E is extremely smaller than the outside diameter r of the diaphragm 24, thus

$$\sqrt{1 - \frac{E^2}{r^2} \sin^2 \theta} = 1$$

and accordingly,

$$y \fallingdotseq R - E \cos \theta - r = R - r - E \cos \theta$$

$$= E - E \cos \theta = E (1 - \cos \theta)$$

$$\omega_1 = ky \fallingdotseq kE (1 - \cos \theta)$$

Also, the discharge flow eate ω_2 of the pump chamber P different in phase by 180 degrees from the rate ω_1 is given by

$$\omega_2 = ky \fallingdotseq kE\ (1- \cos(\pi + \theta)\ )$$
$$= kE\ (1 + \cos\theta\ )$$

Accordingly, the discharge flow rate ω_1 of one pump chamber P is shown by the curve I of Fig. 5, whereas the discharge flow rate ω_2 of the other pump chamber P different in phase by 180 degrees from the former is shown by the curve II of Fig. 5.

The sum ω of both the discharge flow rates ω_1 and ω_2 is given by

$$\omega = \omega_1 + \omega_2 \fallingdotseq 2\ kE$$

and is constant irrespective of the revolutional position of the eccentric rotors 14, 14.

Because of this, the fluid fed under pressure from the pump chamber P invólves no pulsation to provide a constant flow rate.

Incidentally, the explanation has been made in the foregoing on the assumption of $\sqrt{1 - \frac{E^2}{r^2} \sin^2\theta} \fallingdotseq 1$

and thus, the pulsation of the fluid fed under pressure decreases as the eccentricity E decreases relative to the radius r of the outer peripheral circle in the central portion of the diaphragm 24 so that $\frac{E}{r}$ comes close to zero.

In accordance with the present invention, as described above, there is provided an arrangement wherein a pump shaft rotatably journalled on a pump casing has a pair of eccentric rotors fitted thereon through bearings in an axially spaced relation and with a phase difference of 180 degrees in a peripheral direction, diaphragms are mounted on the outer periphery of the eccentric rotors to define a pair of pump chambers between the inner peripheral surface of the pump casing and the outer periphery of the diaphragm, the pump chamber being divided by a partition member into an intake chamber and an exhaust chamber in fluid-tight manner, and a communication between an intake port and an exhaust port which are respectively in communication with the intake chamber and the exhaust chamber is cut off by the partition member during the rotation of the pump shaft. With this arrangement, the communication between the intake port and exhaust port may be cut off at all times during the suction and delivery strokes of the pump chamber to avoid interruption of the suction and delivery strokes caused by the communication therebetween thus providing the continuous suction and delivery strokes, the rotational phase of the pair of the eccentric rotors may be diviated by 180 degrees to thereby differentiate the phase between both the intake and exhaust chambers through 180 degrees and the fluids fed under pressure from the exhaust chambers may be merged together to eliminate the pulsation of individual fluids diviated in phase by 180 degrees and to provide a substantially constant overall discharge quantity from the exhaust port.

Further, since the pump chamber formed between the inner peripheral surface of the pump casing and the outer peripheral surface of the diaphragm is closed by the diaphragm, fluid within the pump chamber may not be leaked into and out of the pump casing and lubricating oil for the bearings or the like within the pump casing may not be entered into the pump chamber, as a consequence of which components disposed within the pump casing are not subjected to the action of corrosion and the like thus increasing the durability, and the fluids within the pump chamber are neither affected nor contaminated by being mixed with lubricating oil for the bearings or the like and may be maintained clean.

In addition, the aforementioned diaphragm is formed at its outer peripheral edge with annular engaging projecting elements having an engaging surface for prevention of disengagement, the pump casing is provided with annular engaging grooves having an engaged surface for engagement with the engaging surface, and the engaging projecting elements are, upon assembly of the pump casing, pressed against and engaged with the annular engaging grooves so that the outer peripheral edge of the diaphragm is fixedly held by the pump casing. With this arrangemnt, even if the tensile force is exerted repeatedly on the diaphragm owing to the eccentric motion of the eccentric rotors, the outer peripheral edge thereof is not displaced, and accordingly, the amount of deformation can be made substantially constant to reduce the change in volume of the pump chamber thereby always maintaining the flow rate of the fluids discharged substantially constant and simultaneously enhancing the sealability of the pump chamber. Moreover, since the outer peripheral edge of the diaphragm may be evenly secured to the pump casing over the entire periphery thereof without using specific fastening means such as bolts, a heavy load is imposed on the diaphragm by the eccentric motion of the eccentric rotors but an excessive load is not imposed locally on the outer periphery thereof or a stress is not concentrated thereon, thus increasing the durability.

Furthermore, the rotary pump according to the present invention comprises a pump shaft, a wedge member interposed between the pump shaft and a bearing fitted on the outer periphery thereof, a wedge roller having a driven plane bearing on a driving plane of a recessed groove formed in the pump shaft and being arranged eccentrically with respect to the center of an eccentric rotor, and a clutch element having a circular surface rotatably fitted in the outer periphery of the wedge roller and in surface contact with the inner peripheral surafce of the bearing. Accordingly, as the pump shaft is rotated to cause the eccentric rotation of the eccentric rotor through the bearings and the wedge member, a frictional sliding movement occurs mainly between the inner peripheral surface of the bearings and the circular surface of the clutch element, but the clutch element is pivoted about the wedge roller along the outer peripheral surface thereof due to reaction from the inner peripheral surface of the bearings to apply a great radially outward pressing force to the bearings thereby enhancing the transmission efficiency of power from the pump shaft to the eccentric rotor and restraining the relative rotation between the inner peripheral surface of the bearings and the circular surface of the clutch element. And, the circular surface in the outer periphery of the clutch element and the inner peripheral surface of the bearing are in surface contact with each other and the contact area is large so that wearing therebetween may be considerably reduced, and as a result, the durability of the wedge member may be improved to effectively prevent the lowering of the pumping efficiency resulting from decrease in eccentricity of the eccentric rotors.

CLAIMS

1. A rotary pump comprising a pump casing (1) having a cylindrical hollow chamber (6) formed therein, a pump shaft (7) rotatably supported by said pump casing, and an arrangement of an eccentric rotor and a diaphragm mounted on the pump shaft so as to exert a pumping action in a pump chamber formed between the diaphragm and the periphery of the pump casing;

characterised in that a pair of eccentric rotors (14) are mounted on said pump shaft (7) in axially spaced relation and peripherally spaced apart by 180° from each other;

a pair of diaphragms (24) are mounted one on the outer periphery of each of said eccentric rotors (14) to define therewith a pair of pump chambers (P) in cooperation with the inner peripheral surface of the pump casing (1);

a respective partition (36) is arranged to divide each of said pump chambers (P) in a fluid-tight manner into a suction chamber (Pi) and an exhaust chamber (Pe);

and a respective intake port (42) communicates with each suction chamber (Pi) and a respective exhaust port (43) communicates with each exhaust chamber (Pe), said intake port (42) and said exhaust port (43) being separated from each other by means of said partition (36) at all times during the rotation of said pump shaft (7).

2. A rotary pump according to claim 1, characterised in that said partition (36) is arranged to be inclined relative to a plane which passes through a central axis of a cylindrical member formed by the outer peripheral surface of said diaphragm (24), and said intake and exhaust ports (42,43) are arranged on the opposite sides of said partition (36).

3. A rotary pump according to claim 1 or 2, characterised by an annular engaging projecting element (29) formed in the outer peripheral edge of said diaphragm (24) and having an engaging surface (29') for prevention of disengagement, and an annular engaging groove (30) disposed in said pump casing (1) and having an engaged surface (30') for engagement with said engaging surface, said engaging projecting element (29) being , upon assembly of said pump casing, pressed against and engaged with said annular engaging groove (30) to be fixedly held by the pump casing (1).

4. A rotary pump according to any one of the preceding claims, characterised by a recessed groove (11) formed in said pump shaft (7) and having a driving plane (13) parallel to the central axis of said pump shaft, and a wedge member (16) arranged between said recessed groove (11) and a bearing (15) in the inner periphery of the respective eccentric rotor (14) to locate said eccentric rotor in an eccentric position with respect to the central axis of said pump shaft.

5. A rotary pump according to claim 4, characterised in that said wedge member (16) comprises a wedge roller (18) having a driven plane (17) which bears on said driving plane (13) and being arranged eccentrically with respect to the center of said eccentric rotor (14), and a clutch element (21) rotatably fitted over the outer periphery of said wedge roller (18) and having a circular surface in contact with the inner peripheral surface of said bearing (15).

6. A rotary pump cpmprising: a pump casing having a cylindrical hollow chamber formed therein; a pump shaft rotatably supported by said pump casing through bearings and

connected to a driving source; a pair of eccentric rotors fitted over said pump shaft through bearings in an axially spaced relation and with a phase difference of 180° degrees in a peripheral direction; a pair of diaphragms mounted on the outer peripheries of said eccentric rotors to define a pair of pump chambers in cooperation with the inner peripheral surface of said pump casing; a partition member adapted to divide each of said pump chambers in a fluid-tight manner into a suction chamber and an exhaust chamber; an intake port communicating with said suction chamber; and an exhaust port communicating with said exhaust chamber; said intake port and said exhaust port being separated from each other by means of said partition member at all times during the rotation of said pump shaft.

FIG. 1

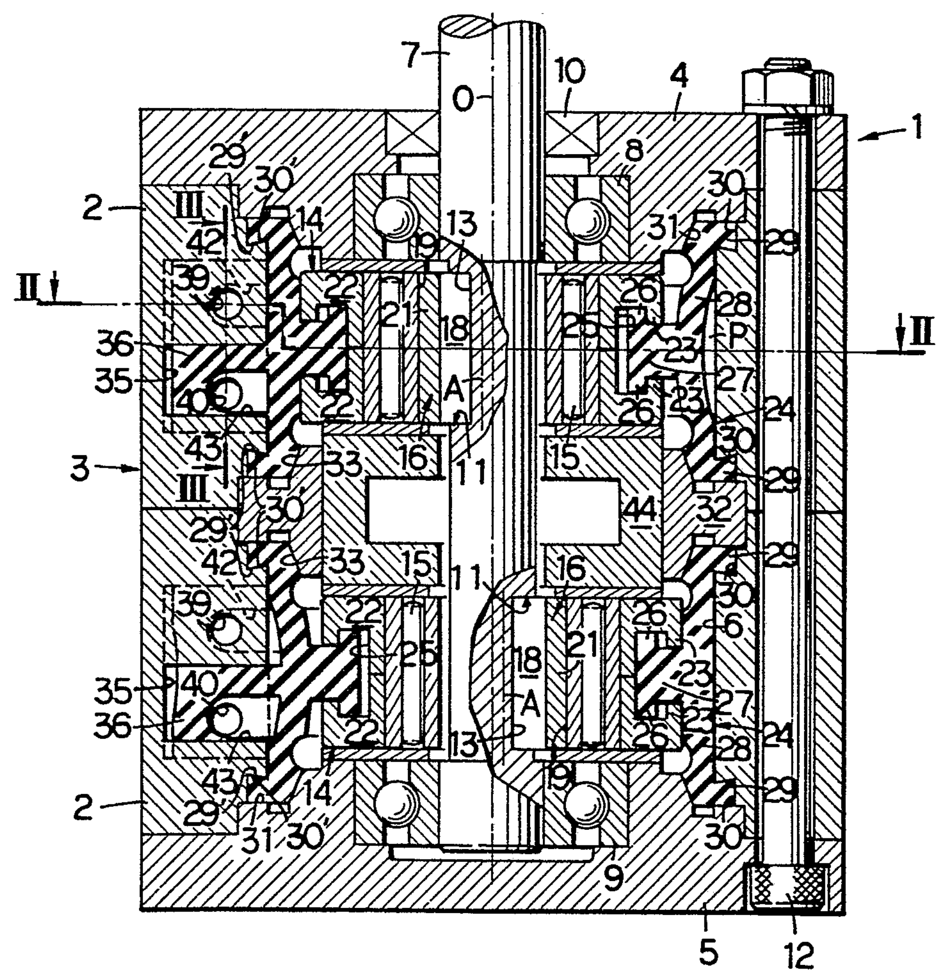
7
0
10
4
1
8
2
29
30'
III
42
14
13
19
30
31
29
39
22
26
28
II
II
36
21
18
25
P
23
35
A
27
40
22
26
23
24
43
16
11
15
30
3
33
29
III
30'
44
32
29
42
33
15
11
16
29
30
39
22
26
6
25
21
18
23
35
40
A
27
36
22
13
9
26
23
24
28
43
14
29
2
29
31
30'
30
9
5
12

FIG. 2

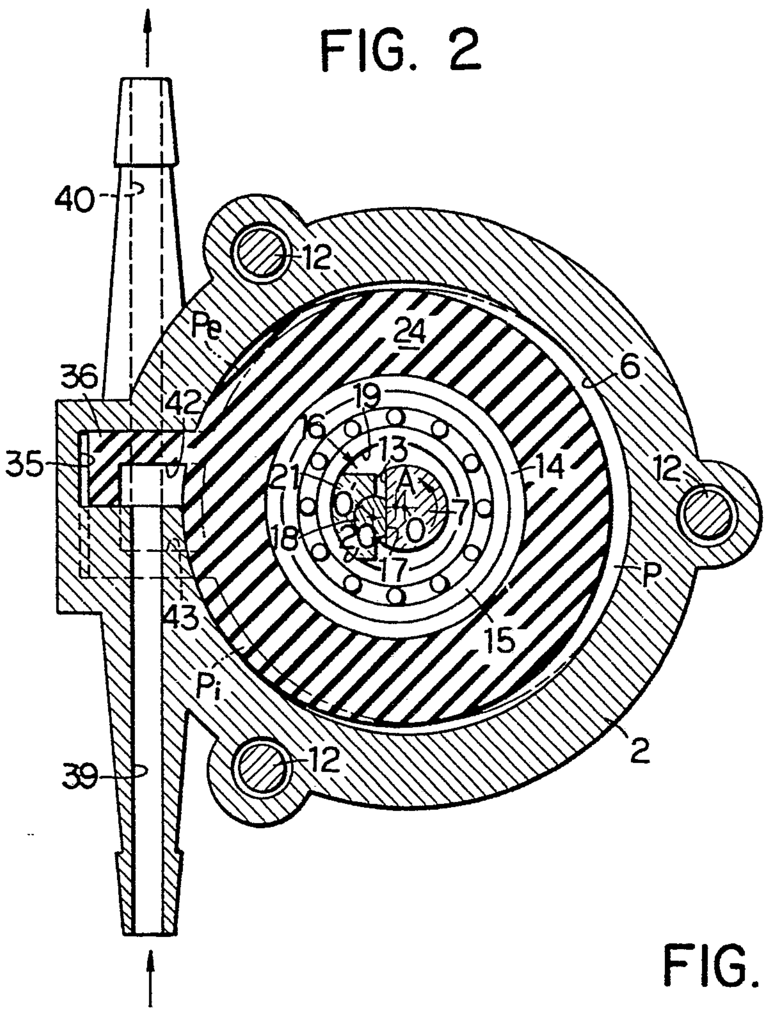
40
12
Pe
24
6
36
42
19
16
13
35
21
A
14
12
7
18
20
17
P
43
15
Pi
12
2
39

FIG. 3

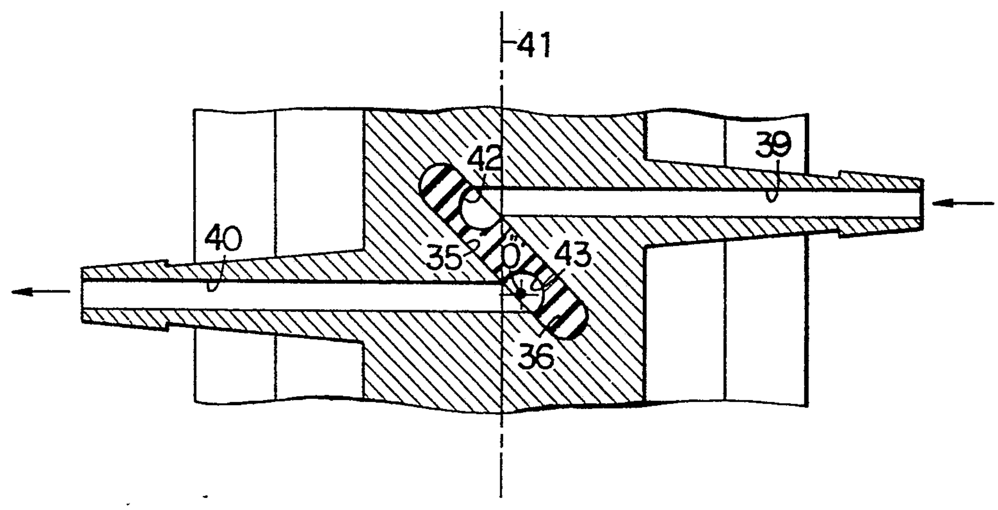
41
42
39
40
35
43
36

FIG. 4

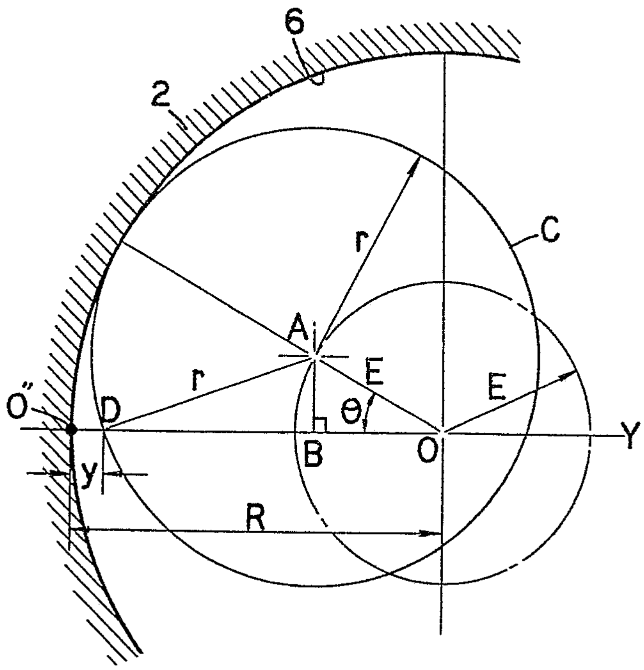

FIG. 5