

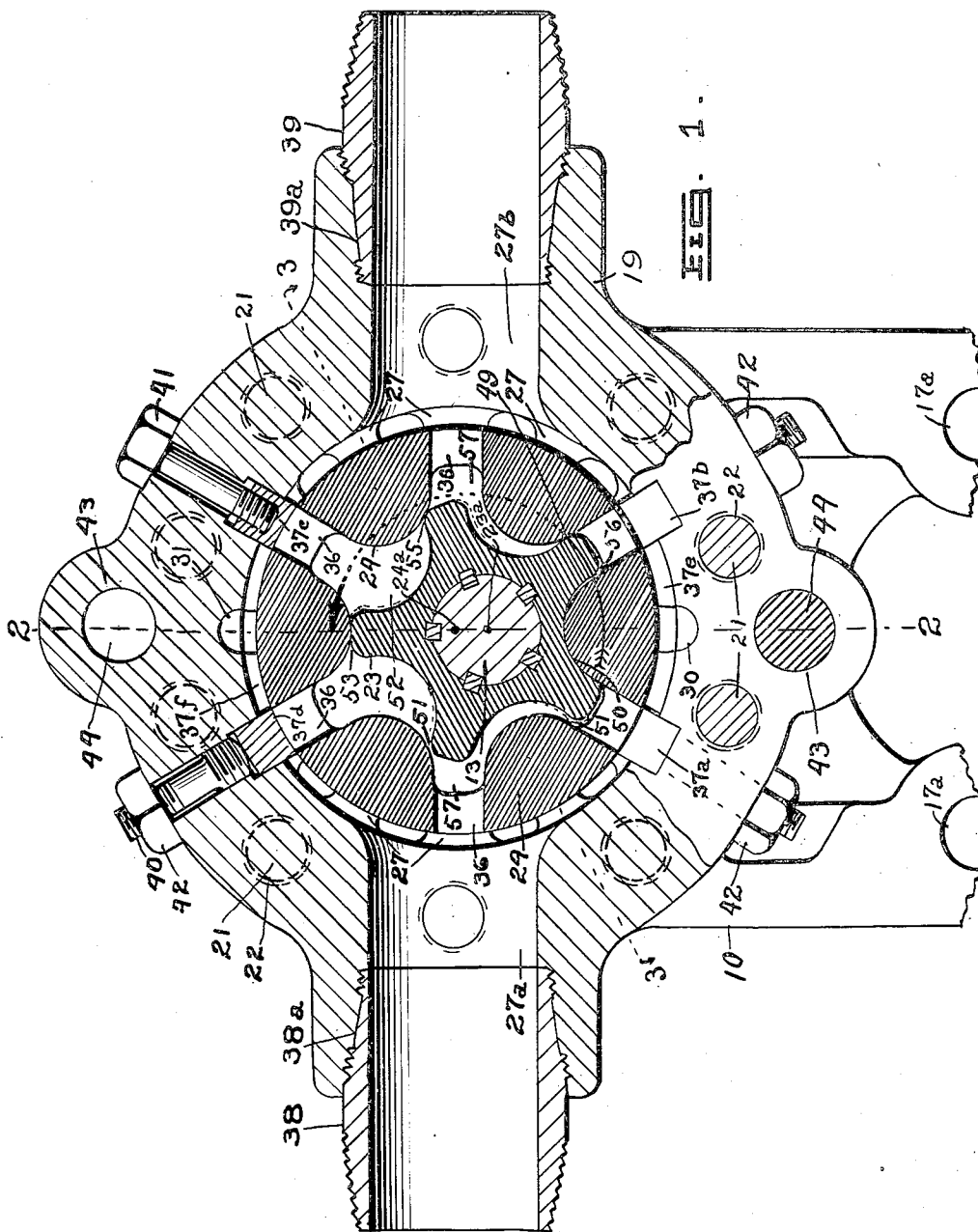
Aug. 14, 1934.

M. F. HILL

1,970,146

REVERSIBLE LIQUID PUMP

Original Filed March 1, 1926 3 Sheets-Sheet 1



INVENTOR

Myron F. Hill

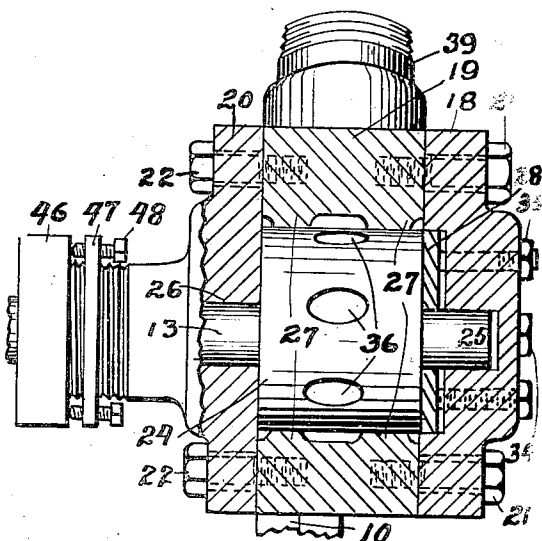
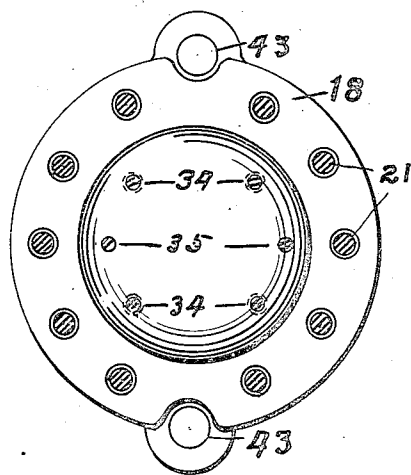
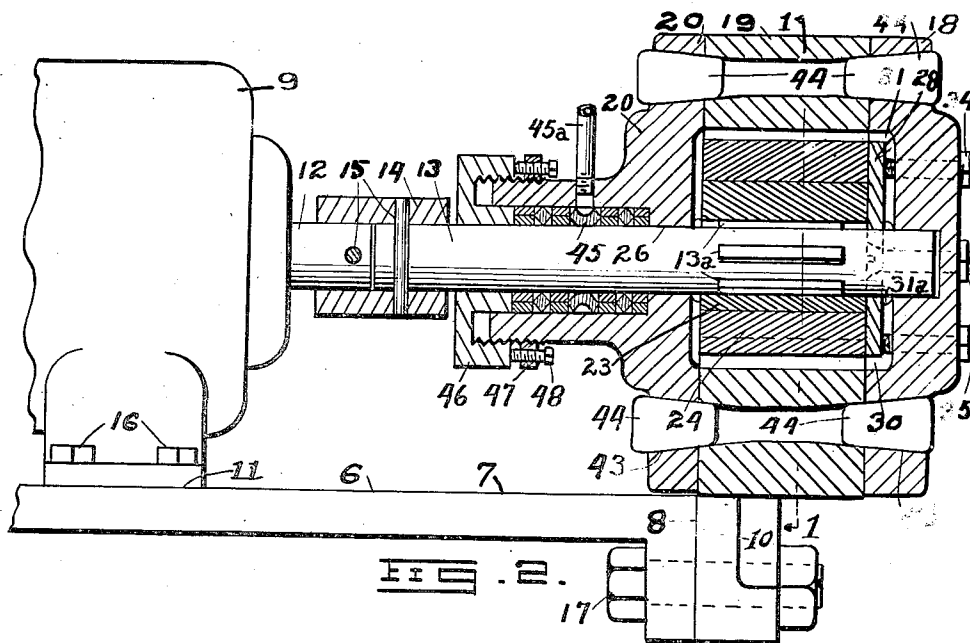
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3 Sheets-Sheet 3

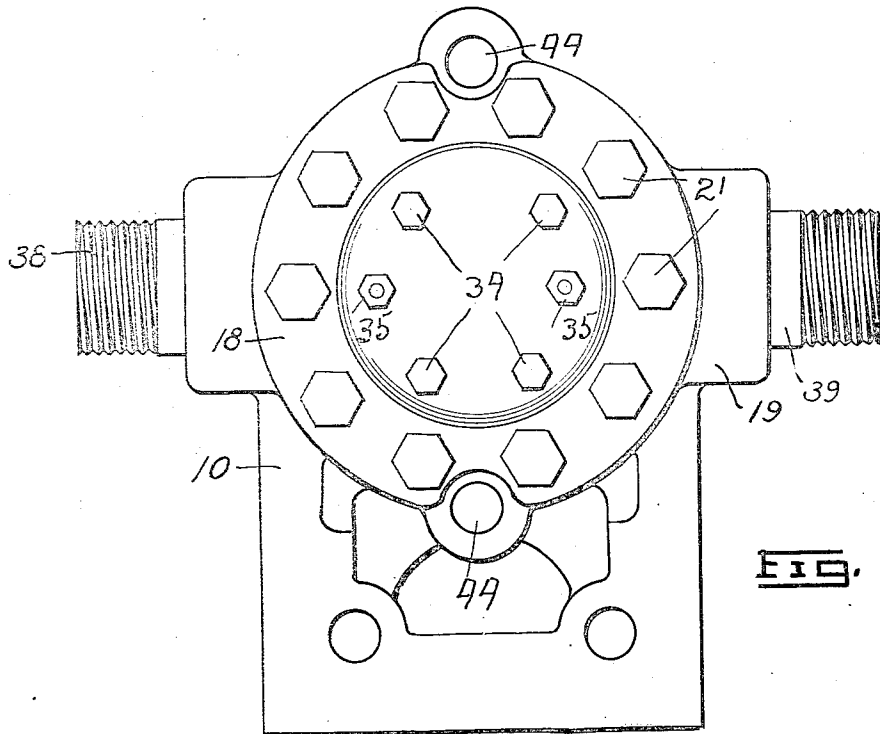


FIG. 5.

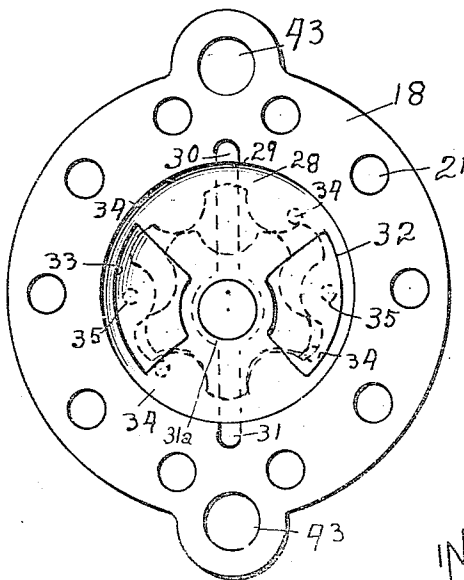


FIG. 6.

INVENTOR
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UNITED STATES PATENT OFFICE

1,970,146

REVERSIBLE LIQUID PUMP

Myron F. Hill, Philadelphia, Pa.

Application March 1, 1926, Serial No. 91,416

Renewed March 10, 1932

24 Claims. (Cl. 103—126)

The object of my invention is a reversible rotary pump or motor of great efficiency and noiseless in operation under pressure and at speed. My invention is aimed to eliminate the heavy loss of efficiency, due to wear, of the rotor movement described in my Patents 1,682,564 and 1,682,565 granted to me on Aug. 28, 1928. This result is attained by creating a new relation between the inlet and outlet ports and the rotor contours, such that there is always at least one travelling continuous tooth contact between the ports at full mesh and at open mesh. My treatise on Kinematics of Gerotors (Patent Office Library) describes rotor contours more in detail. The contours in this case are "prepicroids".

Another object is means for connecting the rotor rigidly to a driving shaft while avoiding the difficulties heretofore encountered in lining up the driving shaft with the pump shaft.

A third object is means to compensate for wear on the ends of the rotors or vary the clearance between the side walls confining the rotors.

A fourth object is rotor contours capable of easy assembly that will run themselves together forming perfect contours in service.

A fifth object is means to balance the rotors endwise to prevent resistance in starting under pressure.

Fig. 1 is a front elevation of the pump with the cover plate removed, and parts broken away to show features of construction.

Fig. 2 is a vertical section on line 2—2, Fig. 1.

Fig. 3 is an elevation with parts in section on line 3—3, Fig. 1.

Fig. 4 is a front elevation of the cover plate with bolts in section.

Fig. 5 is a front elevation of the pump.

Fig. 6 is an inside face view of the cover plate.

Upon a base 6 having two surfaces at right angles to each other, (Fig. 2) the top surface 7, and the end surface 8, are mounted respectively the motor 9 and the pump 10. The base 11 of the motor being parallel to the shaft, and the face of the pump member engaging the end 8, being at right angles to the pump shaft, the motor and pump, after being joined together with their shafts held rigidly in alignment, may be quickly located to allow both members to be bolted together to the base 6.

The shafts of the motor and pump, 12 and 13 respectively, are first driven into the sleeve 14 and pinned thereto by means of pins 15. After the units have thus been assembled and dowelled (dowelled by means of the pins if desired) the parts may be separated to allow the removal of

the motor bearing next the pump and then re-assembled. The bolts 17 may be used in the two holes 17a, Fig. 1 to locate the pump.

The pump is preferably provided with three main casing members, the cover plate 18, the body or middle casing member 19, and the back plate 20. These three members may have surfaces ground to an oil tight smoothness, and secured to each other by means of bolts 21. These bolts are preferably in loose holes in the cover and back plates and threaded into holes 22 in the body of the pump.

Dowels 44 may be driven into reamed holes 43 to fix the relative positions of these members as hereinafter described.

Within the middle casing member are mounted two rotors, 23 and 24. The pinion rotor 23 may be keyed to the driving shaft 13 of the pump by means of the keys 13a or in any other suitable way, the shaft being preferably journaled in the cover plate at 25 and in the back plate at 26. Mounted in the bore of the middle body of the casing upon bearing areas is the outer, or annular ring rotor 24. The actual bearing of this annular rotor is upon projections 27 near the two ends of the rotor (see Figs. 1 and 3). The pressure of the rotor against these projections is comparatively small, since the pressures inside the ring rotor and outside of it are balanced. The cut away portions between the bearing areas reduce film friction, which, in direct drive pumps, is a factor to be considered in the design. As shown in Figs. 1 and 2 each cut away portion opens into the neighboring port on the same side of the center line 2—2 to apply pressure to the outside of the rotor 24 to offset or balance the pressures upon the inside of the rotor, to remove journal pressure of the rotor against its bearing and reduce mechanical friction. When the rotors are travelling in the direction of the arrow in Figure 1, the discharge port of a pump is to the left, and pressure upon the outside of the outer rotor on the left side of the figure tends to push it to the right against the abutments 37^b and 37^c which would have the effect of maintaining tightness there. Pressure on the inside of the outer rotor tends to push it to the left against abutments 37^a and 37^d tending to maintain tightness there. Whether the rotors tend in one direction or the other depends on which pressure predominates, that on the outside of the rotor or that on the inside of it.

On one side of the center line thru the eccentric axes is the port 27a and on the other side the port 27b. Either port may be the suction or the

discharge according to the direction of rotation of the rotors.

The ends of the rotors slide between the back and front cover plates. The face of the front cover plate is shown in Fig. 6. Inserted in it is a circular disk 28 having the same diameter as the outer rotor and registering with it. It has a key 29 fitting the slot 30 to prevent the disk from turning. This disk 28 can be advanced toward or away from the ends of the rotors and thus vary the thickness of film to accommodate different pressures or different liquid viscosities, and to compensate for wear. The disk may be undercut at 32 and 33 to reduce film friction, care being taken not to connect the two under cuts together thru the rotor chambers at full mesh and at open mesh. The undercuts may be omitted. The disk is forced toward the rotors on the high pressure side of the center line 2—2 Fig. 1 thru the eccentric axes of the rotors. Screws 34 may be used to advance the disk toward the rotors. On the suction side of the center line the disk is drawn away from the rotors to prevent it from being sucked against the rotors and thus acting as a brake, and screws at 35, one for each side (the pump being reversible) in accordance with the direction in which it runs, may be employed for this purpose.

In Figure 1 the outer rotor is shown having ports of a radial character at 36, and as there are shown six teeth, or rotor chambers between them, there are six rotor chamber ports spaced 60° apart. At full mesh and at open mesh are shown shoes, abutments, or ribs, having substantially the same circular length as the ports and also spaced 60° apart. Two such ribs 37a and 37b are used at full mesh symmetrically located on either side of the center line, the rotor ports also being located half way between the teeth of the outer rotor. Two similarly located ribs 37c and 37d are used at open mesh. The space 37e between the ribs 37a and 37b; and space 37f between 37c and 37d may act as auxiliary relief ports. After a rotor port passes a rib 37a on the way to the center line 2—2 thru the eccentric axes, at full mesh, the liquid in its chamber is expelled into the auxiliary relief port 37e. After it passes the center line and until it passes the rib 37b the rotor chamber opens and tends to suck liquid from port 37e. While this is happening, the opposite action is taking place at port 37f. In order that these operations may balance each other, the auxiliary relief ports 37e and 37f may be connected together by the slots 30 and 31, (Figs. 1, 2, and 6), and the annular groove 31a around the shaft 13.

This arrangement provides noiseless operation at high speed. It has been found that without such compensating means vacuum holes are apt to be created in the liquid in the rotor chambers, particularly at open mesh, which are then closed so suddenly as to create a hammering action, and at high speed cause a buzzing sound. If the pipe connection 39 is the suction, and 38 the discharge, the ribs 37b and 37c on the right of Fig. 1 limit the range of suction of the rotor chambers as they turn, and the ribs 37d and 37a on the left limit the range of the rotor discharge. Between ribs 37c and 37d at open mesh a rotor chamber enlarges until it is on the center line and then closes again somewhat by the time it reaches the rib 37d. This closing of the rotor chamber at open mesh between the center line and rib 37d expels liquid in constantly increasing quantity into the relief port 37f until after it passes the rib 37d. Such liquid requires some space to re-

ceive it in order to prevent jamming of the rotors. Similarly as a rotor chamber at full mesh passes from rib 37a to the center line it closes at reducing speed also necessitating an outlet. And as a rotor chamber passes from the center line to the rib 37b it opens at increasing speed. Such closing and opening supplement the opening and closing action of the rotor chamber at open mesh above described.

Thus reverse actions take place between the ribs at full mesh and at open mesh. By connecting the auxiliary relief ports between the two pairs of ribs together the compensating action eliminates noise and radial pressures at full and open mesh upon the ring motor are balanced. The ribs are adjustable by push screws 40, one near each end of a rib, and the pull screws 41, one at the middle of each rib. The pull screws are naturally liquid tight. The push screws may be made liquid tight by the nuts 42. These screws adjust the ribs to a close running fit on the rotor.

The inner rotor is keyed to the shaft 13, and the shaft is journaled in the front cover plate at 25 and in the rear cover plate at 26. It is important that the rotors be mounted with correct relation to each other, that is, with the right eccentricity. The outer rotor is journaled in the middle casing member as has been noted, and the inner rotor and its shaft journaled in the two cover plates. These members are fastened together by means of bolts 21 in loose holes in the front and back plates. Before the bolts are tightened the middle member of the casing may be so shifted as to bring the teeth of the rotors together on the center line thru the two axes at open mesh at 53, Fig. 1. The bolts may then be tightened and the rotors run until they run freely after which the surfaces are inspected. If the crowns of the teeth of a liquid pump are shiny the parts may be re-assembled as before with their teeth in contact on the center line 2—2 at 53 and the dowel holes 43 may be reamed to fit the dowels 44 and the dowels driven into place holding the journals in permanent eccentric relation. As the rotors are running the screws 34 and 35 may be adjusted to locate the wearing plate 28 just clear of the rotors and yet close enough to prevent substantial leakage regardless of the viscosity of the liquid used. If the crowns are not cleanly polished the rotors may be pushed more tightly together at open mesh, and again run in to each other. This process is repeated until the teeth maintain contact with each other at least at open and closed meshes to prevent substantial leakage between the intake and outlet ports.

The rotors are provided with contours, particularly on the tooth, sides and crowns, that cause the teeth of either rotor to wipe over the contour of the teeth of the other rotor as they rotate, to maintain fluid tight engagements with each other during the performance of pressure functions in the chambers formed by the rotor teeth. One rotor preferably has one less tooth than the other and is eccentric to it.

In order to form such rotors I prefer to cause a circular cutter representing the circular tooth of the outer rotor to operate. If in a milling machine the cutter is a circular mill. If in a shaping machine, the cutter is a shaping tool having a circular edge for cutting, at cutting speed. A blank for a pinion rotor is caused to have a rotary movement around one axis (its own) as that axis has a relatively rotary movement around the eccentric axis on which the other rotor is to be mounted. By varying the relative directions of

these rotary movements, hypo or epi-curves may be formed. Hypo-curves would be formed on the inside of a ring blank for the outer rotor. In this pump I prefer to form epi-curves on the outside of a blank for a pinion rotor. In the generating machine set forth in the joint application of Hugo Bilgram and myself (Serial Number 573,559 filed July 8, 1928, Patent No. 1,798,059 issued March 24, 1931) the worm shafts were geared together by an intermediate pinion meshing with two gears and driving them in the same rotational direction, one gear upon each worm shaft. By meshing the gears upon the worm shafts directly to each other and meshing the driving pinion to either one or the other, so that the worm shafts rotate in opposite directions, epi-curves are generated. With a five to one ratio the contour of the inner rotor is produced.

An outer rotor to mate with such a five tooth inner rotor preferably has six teeth, circular in form. When made perfectly the inner rotor tooth spaces preferably fit these outer circular teeth across the portions nearer the rotor axes for a substantial distance when centered on the center line thru the two rotor axes. In my patent 1,682,563 I described how tooth spaces of either rotor might be generated by the teeth of the other rotor, or by tools having their forms.

Geometrically speaking, the axis of the milling cutter may be carried by one circle rolling in or on another circle of a different diameter (with the axes at the eccentric distance of the rotor axes), the blank to be formed being mounted on the axis of the other circle. If two circles vary by an integer or whole unit in their relative diameters a number of complete teeth will be formed on the blank. In the rotors shown in Figure 1, for example, the axis of a milling cutter having the radius of the convex crowns of the teeth of the outer rotor is carried by a circle (the pitch circle of the outer rotor) having the diameter of six units, while the circle rolls relatively around the pitch circle of the inner rotor (23) five times, that circle being five units in diameter. Thus the teeth of the inner rotor are based upon the pitch circle of five units in diameter. As a result of this relation the eccentricity of the pitch or ratio circles of the two rotors, and of the rotors themselves, is equal to substantially one half the radial depth of the tooth space of the inner rotor from a circle touching the crowns of the teeth to a circle touching the bottoms of the tooth spaces.

In forming this rotor the milling cutter is started at a safe distance outside of its theoretical position with relation to the outer pitch circle when cutting the curve, and it is then advanced toward the center of the outer pitch circle that carries it so that it cuts deeper and deeper into the blank until, preferably, the minimum radius of the curve at the two ends of the crown of the tooth that is being formed is perhaps a $\frac{3}{4}$ and of an inch, in rotors of half the size shown in Fig. 1, or even less. If the cutter is advanced to cut too deep the corners of the crowns of the teeth may become too sharp. If carried in too far the milling cutter fails to cut a continuous true curve desirable for the driving relation across full mesh from 50 to 49 or for across open mesh. In 7 by 8 rotors (the inner rotor having 7 and the outer 8 teeth) having an eccentricity of $\frac{3}{8}$ units (any unit), the milling cutter may have a diameter of $1\frac{3}{4}$ units, the outer diameter of the inner rotor curve may be 5.85 units, and the outer diameter of the curve of the outer rotor may be 6.6 units.

Such rotors engage near the pitch circles across full mesh.

In order to form the outer rotor, a tool may be formed having a complement to the correct form of the tooth division of the outer rotor. Such a tool has a hollow circular curve or curves to cut the circular teeth of the outer rotor.

The space curves of the outer rotor have little importance. Since the drive on the outer rotor is limited to the circular convex tooth curves, and the space curves of the outer rotor do not participate in driving and do not engage the inner rotor for tightness, their form, particularly when the spaces have a narrow form—or are angularly short—so long as they are loose with the crowns of the teeth of the inner rotor during rotation, is of negligible consequence. In this case, if the curve is generated by the same circular form of mill or shaping tool carried by a seven unit pitch circle rolling on the six unit pitch circle of the outer rotor, and then deepened, or whether they are generated by the teeth of the inner rotor and then deepened, makes no difference in small liquid pumps in results worth paying attention to. For these reasons the tool for cutting the outer rotor, in addition to having circular hollows for cutting the circular teeth may have an edge for cutting the tooth spaces of the outer rotor so that they clear the teeth of the inner rotor during rotation. This tool may then be hardened, and if of steel that does not change size during the hardening process, it will cut the correct teeth of the outer rotor. This tool may be mounted in a vertical slotting machine, the blank being indexed between cuts the length of a tooth division.

When the rotors are assembled the teeth are brought together at open mesh with the pinion tooth on the center line at full mesh if possible, and the rotors run together until the crowns of the teeth are sufficiently polished to prevent leakage between ports 27a and 27b. The driving contacts at full mesh in my contours take care of themselves in the running in process. In making the curves by mechanical processes the mechanical error or tolerance, perhaps a fraction of a thousandth of an inch may, if desired, leave the crowns of the teeth correspondingly high, and the space curves undercut without disturbing curves necessary for driving and for tightness at uniform angular speed in order that they may fit together loosely and permit the teeth to wear each other to the proper generative contour. When rotors are assembled as described and run in as described they correct any slight variations in contours that may result from the method of manufacture above described. If in machining the tool for the concave spaces of the annular or outer rotor, a flat portion providing the cutting edge for the concave spaces is set at an angle of 88° or more from the axis of the milling cutter, and then set perpendicular to the surface of the rotor which it is cutting, the spaces will be slightly deeper and while the inner teeth and outer concave tooth spaces fit together loosely, the teeth will require tight pressure holding and driving contours while being operated under pressure by wearing each other to the correct contours. The outer rotor may have the teeth slightly narrower than the above theory requires to permit easy assembly of the rotors. This may be accomplished after the teeth have been cut in the gear slotting machine by means of a tool having the circular teeth and undercut tooth spaces, in reverse, by turning the rotor slightly to a degree

depending on the particular rotors being made and their sizes and shapes on its axis in the machine and repeating the slotting operation. This side stepping process results in a compound curve system on the rotor. Whether applied to one rotor or the other the result is the same. Such a compound curve is in reality providing all the forward faces and flanks of a rotor with curves of one contour and all the back faces and flanks of that rotor with curves of a separately generated and sidestepped, tho identical form contour. Instead of rotors having teeth that are angularly tight when new, they are angularly loose. Yet regardless of whether the rotors are travelling one way or the opposite, the faces of the teeth of the so "doctored" outer rotor after being run together maintain tight relations with the teeth and concave tooth spaces of the inner rotor. If the inner rotor is connected to the driving shaft this tight relation is as the rotor chambers open, and if the outer rotor is driven the tight relation is as the rotor chambers close. One side of the "doctored" teeth engage the other rotor in one direction, and other conditions being the same the other side of the teeth engage in the other direction. My Patents 1,682,564, and 1,682,565 show constructions in which the outer rotor is attached to the driving shaft. This side stepping of the faces and flanks of a rotor may be applied to the rotors in those patents.

In other words, the rotor that is "doctored",—if the outer one is so treated—has its circular teeth cut as in a vertical indexing slotting machine and then the rotor is sidestepped angularly (circumferentially) about its own axis (the amount is a matter of choice) depending on the diameters of the rotors and the sizes of the milling cutters used, and the process repeated so that a second circular contour is imposed, cutting away the contour, already formed, on one side of the teeth and not touching the circular contour upon the other side of the teeth. Where these two contours meet in the middle of a tooth one curve practically merges into the other and after running in leaves no visible line of demarkation. The side-stepping of course shifts one side of the tooth spaces.

The ends of the ports are accommodated to this slight alteration of the theoretical curve system. Production tools may be modified to manufacture such rotors at a single operation.

It has been explained how, when a shaft is connected to the inner rotor for driving, the teeth (and inner rotor tooth spaces at full mesh) make contact as the rotor chambers are opening, on the right hand side of the center line 2—2 in Fig. 1, when the rotors are travelling in the direction of the arrow. This range extends as explained from some curve contact region 50 on the left of the center line a distance of half a tooth crown of the outer rotor around on the right side for a distance of 180°, that is, to the center line at open mesh. The total theoretical driving length of the arrow is the total contact range, including the actual driving range where most of the actual driving results after running in, from 50 to 49 a little shorter than the length of a tooth division. A rotor chamber at full mesh extends from such contact 49 to the nearest part of the curve contact region 50. A full tooth division is measured by the distance between adjacent tooth contacts anywhere except at full mesh. When first assembled the pressure between the teeth may begin at around the point 55, and the wearing in

begins so that contact may creep around to the point 49 and finally to 50. Then the contacts from 49 to 53 wear approximately pressure free so that the driving friction is limited to an almost pure rolling action at full mesh from 50 to 49, and the contacts over the rest of the range create substantially no rubbing friction but merely slide lightly over each other. As a rule these contact surfaces remain polished in service. In order that each tooth curve along this range may maintain this continuous pressure holding engagement with each tooth of the other rotor, all rotor curves thus cooperating should be symmetrical and developed or generated as described,—in this case 5:6. Any variation of the speed of generation from the tooth ratio is a departure from the true curves of the rotors and creates leakage and loss of efficiency, the degree depending upon the aberration involved. It is therefore apparent that the term "pressure holding engagement" means either a contact or such a near approach to a contact that the principle of the invention is maintained. Sometimes viscous fluids might make possible more aberration from steady angular velocity in generation than if thin and nonviscous fluids are employed, but whatever the fluids the curves must approximate the true curves as above described in order to attain efficiency in service.

In the driving range the tooth surfaces lie so closely against each other a convex rolling on a concave that they tend to entrain the liquid used in the mechanism and to actually separate the metal surfaces of the teeth, so that the contact is scarcely a contact, just as a shaft may not touch the metal in its journal when well lubricated. Nevertheless such a tooth relation maintaining pressure against the mating teeth performs the function of a "pressure holding engagement". Such a contact is maintained tight by the relatively uniform angular speed between the rotors due to correct generated curves in the driving range.

From the previous description of rotor curves, ports and contact and driving ranges, it should be evident that correct interrelations between them are essential in order that rotor chambers may handle fluids without leakage and serious loss of efficiency. It may be evident that the reversible factor of the pump in which the intake becomes the discharge port and vice versa, further complicates the problem; since a port that might be efficient for a one direction pump might be inefficient when the pump runs only in the opposite direction. If the pump runs only in the direction of the arrow for example, rib 37d might be removed without injury. Since the contact range is along the opening chambers on the right hand of the center line 2—2, Fig. 1, the suction port 27b has its ends or abutments within the contact range or so located as to cut the tight rotor chambers off from the outlet port 27a at each end in such a way that one travelling tight tooth contact at least is always interposed between the ports at both ends. As the outlet port 27a lies along the loose contact—or absence of contact—range of side stepped or worn rotors it may be longer than the intake port except when the pump is reversible in which case the outlet port becomes the intake port. If the mechanism is used as a motor instead of as a pump it has pressure fluids admitted through one of the ports and the other is employed as the discharge. The inlet is of higher pressure than the discharge, in such a motor, of course, and if admitted thru port 27a the rotors will travel in the direction op-

posite to the arrow, Fig. 1, and the contact range will be along the arrow just the same. It is a pump in reverse. If pressure is admitted thru port 27b the rotors turn with the arrow, but the contact range is from 53 down on the left side of the center line 26 and around across the center line at the lower side of the rotors as far as the point 49. The mechanism, in other words is a reversible pump and motor. It is efficient in power consumption as a liquid motor but for expansible fluids the form of mounting shown in my Patent No. 1,682,565, dated Aug. 28, 1928, in which the outer rotor is fastened to the driving shaft, is preferable, as it utilizes the expansive factors of such fluids in applying torque to the shaft.

Whether employed as a pump or motor, it is evident that if the mechanism is to be reversible the inlet and outlet ports must be interchangeable and therefore the same,—in other words, symmetrical. The width of the ports of course might vary without serious effect.

As side stepped or worn rotors, when running as a pump in the direction of the arrow, may have angular looseness between the teeth from 53 to 50, Fig. 1, on the left of the centerline, it is evident that liquid can pass freely between them. Such tooth relations might become useless for tightness between the ports. Before this fact was understood pumps were built which when new, and for some time afterward had high efficiency, but then developed leakage between the ports which in extreme cases rendered the pumps useless. For commercial utility therefore, pumps and motors should have a tight tooth relation separating the ports at full mesh after substantial wear and open mesh. The end of a port should be so located that after passing it a tight tooth relation may travel the length of a rotor chamber before the beginning of the next port can begin to function. In Fig. 1 for example, it can be noted that the rotor chamber outlet or port 36 is blocked by the abutment 37c and the tooth contact of rotor chamber 52 ends at 53. When rotor chamber 52 travels beyond this position the tooth relation at 53 is opened, as it has passed beyond the contact range. If in this advanced position chamber 52 was still connected to the port 27b there would be leakage from port 27a between the teeth at 51 into rotor chamber 54 thru the open tooth relation just mentioned, and thru chamber 52 into the low pressure port 27b. Therefore the abutment 37c is located to cut rotor chamber 52 off from the suction port 27b just as the contact at 53 parts company. And at this moment the contact at 55 prevents leakage from chamber 52 into the intake port 27b, and continues to do so until it has reached the position at 53. One tooth contact is maintained for nearly the length of a rotor chamber until the next succeeding tooth contact can take its place. Contact 55 for example is maintained after chamber 52 leaves the suction port 27b until it reaches the position at 53. While running in this direction and under these conditions the rib 37d is not needed. It is when the pump is rotated in the reverse direction that the abutment 37d is needed, as the terminus of the intake port which then is 27a.

Similar relations between the teeth and ports are needed at full mesh. When the pump is running in the direction of the arrow and when the lowest tooth of the outer rotor is centered upon the center line 2—2, the range of tooth contact begins at 50 where the circular tooth engages a circular portion of the tooth space of the inner

rotor. This is so because this tooth preferably has a circular crown, characteristic of the prepi-croid system of rotor curves and a portion of the tooth space of the inner rotor nearest the rotor center is also circular. The rotor chamber 56 extends from contact 50 to contact 49 in this position. The port 36 of the rotor chamber 56 is covered by the abutment 37b. Pressure in port 27a in order to reach port 27b would have to enter rotor chamber 57, pass thru contact 50 thru rotor chamber 56, thru tooth contact 49, rotor chamber 57 and then port 27b. Tooth contacts 50 and 49 being closed prevent this leakage. Before the teeth engage at 50 the contact at 49 alone prevents such leakage. While the tooth contact is travelling from 50 to 49, it alone prevents leakage from port 27a to 27b. This is the driving range, and the drive of one tooth against the other keeps the tooth contact tight. It will be noted that at 50 the surfaces of the contacting teeth—or rather of a tooth of the outer rotor and a tooth space of the inner rotor—lie against each other for a considerable distance. In the actual curves the contact at 49 is very fair for driving. In designing rotors the numbers of teeth affect the driving contact at 49, more teeth improving it. Rotors are designed as to number of teeth to provide a driving relation adequate to the service conditions of the rotors to provide durability, it being understood that the more teeth in a given diameter the less the rotor chamber capacity. The relations shown for rotors having an outside diameter of two inches is satisfactory. The teeth of the inner rotor may be as narrow as strength permits. A change in widths of teeth and curve diameters requires the port abutments 37a, 37b, 37c &c. to be altered to suit the conditions illustrated in Fig. 1. These differences are matters of degree.

The position of the location 50 may be at any part of the contact of the outer tooth with the inner space curve in the positions upon the center line referred to, provided it lies to the left of the center line 22 in Fig. 1.

From the foregoing description it should be evident that from the moment that a rotor chamber such as 56 connects with the suction port as 27b, it must have a contact such as 50 to prevent leakage from 27a to 27b, and this contact travels the length of a tooth division—50 to 49—until a following rotor chamber with its protecting contact succeeds it. Correspondingly at open mesh the moment a rotor chamber as 52 leaves the intake port 27b it must provide a contact as from 55 to 53 to prevent leakage into rotor chamber 57 from rotor chamber at 54. And when the contact at 53 parts company—beyond the arrow point—the contact 55 succeeds it. Furthermore the moment rotor chamber 52 leaves port 27b the contact at 53—or just beyond 53—is no longer necessary to prevent leakage and can part company. Since the curves of the sides of the rotor teeth (and spaces) of one of the rotors have been shifted from the curves shown in broken lines in a direction opposite to that of the arrow the teeth fail to make contact from 55 to 50, as indicated particularly at 51. When the pump is reversed of course the contact range reverses as described, and the teeth part company on the other side of line 2—2.

It has been explained that the teeth press against each other along the driving range at 50 to 49. From 49 to 53 the teeth wear off quickly and thereafter lightly slide over each other keep-

ing each other polished usually, but without measurable friction, a relation which is not altered by wear of the rotor curves.

The contact range for phycocroid curves is quite different—see Kinematics, supra.

In cutting the inner rotor due to tolerances in manufacture, the spaces may be cut deeper sometimes perhaps a fraction of a thousandth of an inch, and the crowns likewise left correspondingly higher than the theoretical form to compensate for variations in manufacture. With such additions to and substrations from the true curves the rotors may be run together on their correct centers, or as near to them as possible, the excess metal along the contact range from 50 to 49 being worn or burnished off, and the rotors set to make contact at 53, as has been described, before putting them into use. Those portions of the curves employed for continuous contact in the driving range above described should be correct since they have so close to a rolling contact with one another that wear is infinitesimal—more theoretical than practical. To prevent chatter and leakage they should be indexed accurately to maintain steady angular velocity and pressure engagement between the rotors.

In Fig. 1 the outer rotor is shown with an even number of teeth. For a liquid pump this is of importance in eliminating noise as in the manner heretofore set forth. Otherwise the outer rotor may have an odd number of teeth. In air compression whether the teeth are odd or even in number is of little importance per se.

The pump may act as a motor if fluid pressure is connected to one of the ports.

The shaft may be supplied with any suitable stuffing box as at 45, adjustable by means of a nut 46 split at 47 and locked in place by set screws 48. Oil may be fed at 45a to the middle of the stuffing box 45 if desired—particularly if used for vacuum purposes.

It is evident that this pump or motor may be used for liquids, or for gases if a liquid seal is provided and fed into the intake port. It may also operate as an air motor, water motor, or even as a steam or gas engine if desired, tho' for compressible fluids the long high pressure port shown is not very efficient.

The pipe nipples at 38 and 39 may be cast in the casing by putting them into the core box and ramming the cores into them, so that when the cores are placed in the mold the cast iron runs around the nipples and grasps them firmly in cooling. To facilitate pressure tight joints the threads are removed on the inner ends of the nipples and circular corrugations or sharp edges 38a and 39a are substituted. The edges fuse into the cast iron and make a liquid tight joint.

The foregoing description of my invention applies particularly to fluid mechanisms of small size capable of high volumetric efficiency and of high pressures. For lower efficiencies, lower pressures, and larger sizes, some degree of perfection of form may be sacrificed without departing from its spirit.

While I have shown and described separate strips or shoes 37a, 37b, 37c, and 37d, it is understood that strips 37a and 37b may be considered as constituting one abutment or wall portion lying along the contact range at full mesh, and 37c and 37d constituting another abutment or wall portion lying along the contact range at open mesh.

The curves in this rotor system have been described as of the class of epicycloids. The

curves, being curves of envelopment described by successive positions of rotor teeth, with the teeth of the outer rotor of a circular nature, are termed prepicroids.

While I have described my invention in the form I prefer to make it, it is evident that it may be varied in many of its features and some may be used without others. My invention comprises the novel features alone or in combination, shown or described herein.

What I claim is:—

1. In a rotor mechanism for operation on fluids or by fluid pressures, a casing having therein inlet and outlet ports for relatively high and low pressures, and toothed rotors, one within, eccentric to, and having a lesser number of teeth than the other, said toothed rotors having contours maintaining continuous contacts at relatively uniform angular motion while performing fluid pressure functions, and forming rotor chambers between the contours which open and close during rotation to receive and discharge fluid thru said ports; the ends of said ports at open mesh being so disposed and limited that in cooperation with the shape of said contours at least one continuous pressure holding relation between the teeth of said rotors is provided between the ends of said ports; and the ends of said ports at full mesh being so disposed and limited in cooperation with the shape of said contours that at least one pressure holding and driving contact at uniform angular motion is provided from the full mesh end of one port to the full mesh end of the other port.

2. The combination claimed in claim 1 having said pressure holding contacts continuous on one side of the center line thru the rotor axes between open and full mesh, and driving means for the rotors to maintain said contacts.

3. The combination set forth in claim 1 having said pressure holding contacts continuous on one side of the center line thru the rotor axes between open and full mesh, and driving means for the rotors to maintain said contacts, said driving means including a shaft, and a connection between said shaft and the inner rotor.

4. The combination set forth in claim 1 having said pressure holding contacts continuous on one side of the center line thru the rotor axes between open and full mesh, and driving means for the rotors to maintain said contacts, said pressure holding and driving contacts being continuous at full mesh on both sides of the said center line.

5. The combination set forth in claim 1 wherein the ends of said two ports at the full mesh region and the other ends of said ports at the open mesh region are disposed symmetrically on either side of the center line thru the rotor axes, thereby providing a reversible pump.

6. The combination set forth in claim 1 wherein a bearing is provided in said mechanism, said bearing having landings and fluid spaces, a rotor supported thereon, said spaces during rotation being hydraulically connected with successive rotor chambers as said successive chambers are connected to the high pressure port, said spaces and landings being so distributed that the pressure therein opposes the pressure in said successive chambers while they are connected to the high pressure port, and in cooperation with said rotor contours that the pressure in said high pressure port is prevented from being dissipated into said low pressure port.

7. The combination set forth in claim 1 where-

in a bearing is provided in said mechanism, said bearing having landings and fluid spaces, a rotor supported thereby, said spaces being connected hydraulically with successive rotor chambers as said successive chambers are connected to the low pressure port, said spaces and landings being so distributed that the pressure therein opposes during rotation, the pressure in successive chambers while they are connected to said low pressure port, and in cooperation with said contours that pressure from said high pressure port is prevented from being dissipated into said low pressure hydraulic connections.

8. In a rotor mechanism for operation on fluids or by fluid pressure, a casing having therein inlet and outlet ports for relatively high and low pressures, and toothed rotors, one within, eccentric to, and having a lesser number of teeth than the other, said toothed rotors having contours forming chambers between the teeth, said chambers closing as they approach full mesh and opening as they leave full mesh, a bearing provided in said mechanism, a rotor supported thereby, said bearing having landings and fluid spaces, said spaces having hydraulic connections with successive rotor chambers as said successive chambers are connected to a pressure port, said spaces and landings being so distributed that during rotation the pressure therein opposes the pressure in said successive chambers while they are connected to said port, and in cooperation with said contours that pressure from said port is maintained, said bearing having therein also a depressed area in said full mesh region disposed and limited in cooperation with said contours to prevent dissipation of pressure from one port to the other, and hydraulic connections between said depressed area and said successive rotor chambers as they cross the full mesh region while unconnected with either port.

9. In a rotor mechanism for operation on fluids or by fluid pressure, a casing having therein a bearing, and intake and outlet ports for different pressures, toothed rotors in said casing, one mounted on said bearing, one rotor within, eccentric to, and having a lesser number of teeth than the other, said toothed rotors having contours forming chambers between them which vary in their displacements across both full and open mesh region, interconnected depressed areas in said bearing in both full and open mesh regions limited to prevent dissipation of pressure between said ports, hydraulic connections between said depressed areas and successive rotor chambers as they pass from one port to the other while unconnected with either port, whereby chambers opening and closing between said ports permit ebb and flow of fluid between them thru said interconnected depressed areas.

10. In a rotor mechanism for operation on fluid or by fluid pressure, a casing having therein inlet and outlet ports for different pressures for receiving and delivering fluids, and toothed rotors meshed with each other, the teeth of said rotors forming displacement areas opening and closing during rotation to transfer fluid from one port to the other; a bearing in said casing, a rotor supported upon said bearing having successive tooth areas exposed to the pressure in one of said ports, a series of landings and spaces separating said landings in said bearing, said spaces being hydraulically connected to said port and so distributed that the fluid pressure from said port opposes the fluid pressure upon said teeth and thru said teeth upon said bear-

ing, whereby pressure in said port tends to float said rotor off from said bearing.

11. In a rotary mechanism for operation on fluids or by fluid pressures, a casing having therein inlet and outlet ports for different pressures, a bearing in said casing, a rotary member supported on said bearing, said rotary member having successive areas exposed to fluid pressure in one of said ports, a series of landings in said bearing, spaces separating said landings, said spaces being hydraulically connected to said port and its fluid pressure, and said spaces being so distributed that the pressure from said port opposes the pressure upon said successive areas in a radially opposite direction.

12. The combination set forth in claim 11 wherein said bearing surrounds said rotary member and said spaces are hydraulically connected to said successive areas within said rotary member, whereby the pressure upon said areas is opposed by pressure in said spaces and over said landings upon the outside of said rotary member.

13. In a rotary mechanism for operation on fluids or by fluid pressures, a casing having therein inlet and outlet ports for different pressures, toothed rotors in said casing having chambers between the teeth which open and close during the performance of pressure functions to receive and discharge fluid thru said ports; a bearing in said casing, landings and separating spaces in said bearing, hydraulic connections between the spaces and one of said ports and to successive chambers opening or closing along said port, said spaces and landings being so disposed that the pressure in said chambers is opposed by the pressure in said spaces and over said landings connected to said port; depressed areas in said bearing located between said ports in both full and open mesh regions, hydraulic interconnections between said depressed areas at full and open mesh, said hydraulic interconnections and depressed areas being so disposed and limited that, in cooperation with said chambers, the dissipation of pressure from one hydraulic pressure region to another is prevented.

14. In a rotary mechanism, a casing, a bearing in said casing, a rotary member supported on said bearing, sources of varying pressures, successive areas of said rotary member being subjected to said pressures, alternating landings and spaces in said bearing, hydraulic connections between said spaces and one of said sources, said sources, spaces and hydraulic connections being so disposed that the pressure on said successive areas is opposed by the pressures in said spaces from said source.

15. In a rotor mechanism for operation on fluids or by fluid pressures, a casing having therein inlet and outlet ports for said fluids, and toothed rotors, one within, eccentric to, and having a lesser number of teeth than the other, said toothed rotors having contours maintaining continuous contacts while performing pressure functions at relatively uniform angular speed, and forming rotor chambers between the rotors which open and close during rotation to receive and discharge fluid through said ports, said casing for said mechanism having different portions or members, one member centering one rotor and one member centering the other rotor, and attaching means providing looseness in the direction of the eccentricity to attach said members to one another with the rotors in contact at open mesh whereby their eccentricity is determined by the rotor diameters.

16. In a rotor mechanism for operation on fluids or by fluid pressures, a casing having therein inlet and outlet ports for said fluids, and toothed rotors, one within, eccentric to, and having a lesser number of teeth than the other, said toothed rotors having contours maintaining continuous contacts while performing pressure functions at relatively uniform angular speed, and forming rotor chambers between the rotors which open and close during rotation to receive and discharge fluid through said ports, said casing for said mechanism having different portions or members, one member centering one rotor and one member centering the other rotor, and attaching means providing looseness in the direction of the eccentricity to attach said members to one another with the rotors in contact at open mesh whereby their eccentricity is determined by the rotor diameters, and means to permanently secure said eccentric relation.

17. In a rotor mechanism for operation on fluids or by fluid pressures, a casing having therein inlet and outlet ports for said fluids, and toothed rotors, one within, eccentric to, and having a lesser number of teeth than the other, said toothed rotors having contours maintaining continuous contacts while performing pressure functions at relatively uniform angular speed, and forming rotor chambers between the rotors which open and close during rotation to receive and discharge fluid through said ports, said casing for said mechanism having different portions or members, one member centering one rotor and one member centering the other rotor, and attaching means providing a looseness in the direction of the eccentricity to attach said members to one another with the rotors in contact at open mesh whereby their eccentricity is determined by the rotor diameters, means to permanently secure said eccentric relation and a cover plate to act as a detachable cover for the rotor cavity.

18. In a rotor mechanism for operation on fluids or by fluid pressures, a casing having therein inlet and outlet ports for said fluids, and toothed rotors, one within, eccentric to, and having a lesser number of teeth than the other, said toothed rotors having contours maintaining continuous contacts while performing pressure functions at relatively uniform angular speed, and forming rotor chambers between the rotors which open and close during rotation to receive and discharge fluid through said ports, the contour flanks or sides of the teeth of one of said rotors circularly displaced to provide lost motion between the teeth on one side of a plane passing through the axes of the two rotors to provide spaces for grit or foreign matter to pass through said rotor teeth with freedom from binding while still maintaining the pressure holding relations specified on said circularly displaced contours.

19. The combination claimed in claim 1 having the outer rotor journaled in said mechanism, fluid spaces in said mechanism outside of said rotor at full mesh and at open mesh having an

interconnecting passageway also connecting with a journal for the inner rotor, and connected during rotation with a rotor chamber crossing full mesh.

20. The combination claimed in claim 1 having the outer rotor journaled in said mechanism, fluid spaces in said mechanism outside of said rotor at full mesh and at open mesh having an interconnecting passageway also connecting with a journal for the inner rotor, and with a rotor chamber crossing open mesh.

21. The combination claimed in claim 1, said mechanism including three casing members, two carrying journals for the inner rotor, and a third middle member journaled the outer rotor, and separating the other two members; and means to center them in assembled relation in a position determined by the eccentricity of the rotors.

22. The combination claimed in claim 1, said mechanism including three casing members, two carrying journals for the inner rotor, and a third middle member journaled the outer rotor, and separating the other two members; means to fix them in assembled relation in a position determined by the eccentricity of the rotors, and ports in said third member.

23. In a rotor mechanism for operation on fluids or by fluid pressure, a casing therein having inlet and outlet ports for said fluids, and toothed rotors, one within, eccentric to, and having a lesser number of teeth than the other, said toothed rotors having contours maintaining continuous contacts while performing fluid pressure functions at uniform angular speed, and forming rotor chambers between the rotors which open and close during rotation to receive and discharge fluid through said ports; a bearing in said casing, said outer rotor supported on said bearing, a series of landings separated by spaces in said bearing, hydraulic pressure connections between said spaces and one of said ports, said landings and spaces being so distributed that the pressure in said spaces opposes the radial fluid pressure within successive chambers in said rotors while said chambers are in communication with said port, whereby fluid pressure in said spaces and over said landings tends to lift said rotor off from said landings.

24. In a rotor mechanism for operation on fluids or by fluid pressure, a casing therein having inlet and outlet ports for said fluids, and toothed rotors, one within, eccentric to, and having a lesser number of teeth than the other, said toothed rotors having contours, maintaining continuous contacts while performing fluid pressure functions at uniform angular speed and forming rotor chambers between the rotors which open and close during rotation to receive and discharge fluid through said ports; said casing including a plurality of members, one rotor centered in a member, another rotor centered in another member, said rotors so assembled therein as to transfer the eccentricity of said rotors to said members, and attached means to positively locate said members for subsequent reassembly.

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