

March 23, 1954

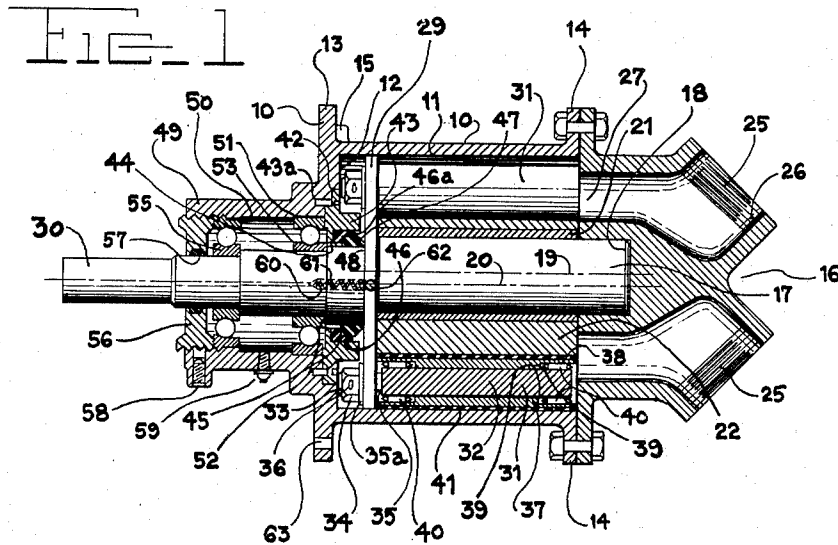
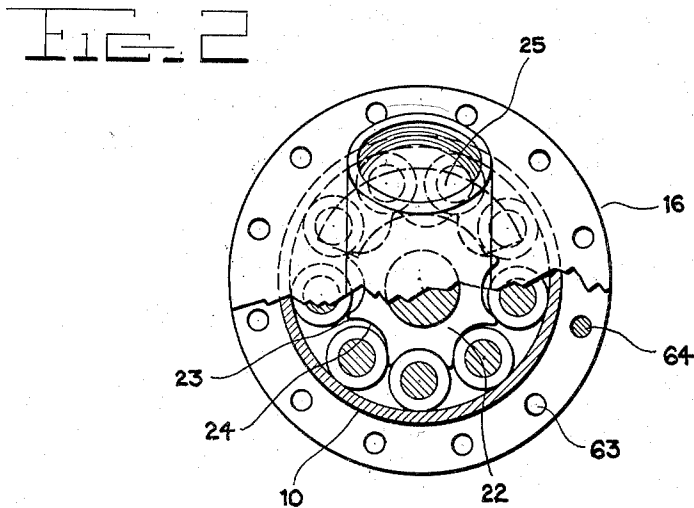
B. F. QUINTILIAN

2,672,825

HYDRAULIC PUMP AND MOTOR

Filed Feb. 17, 1945

3 Sheets-Sheet 1



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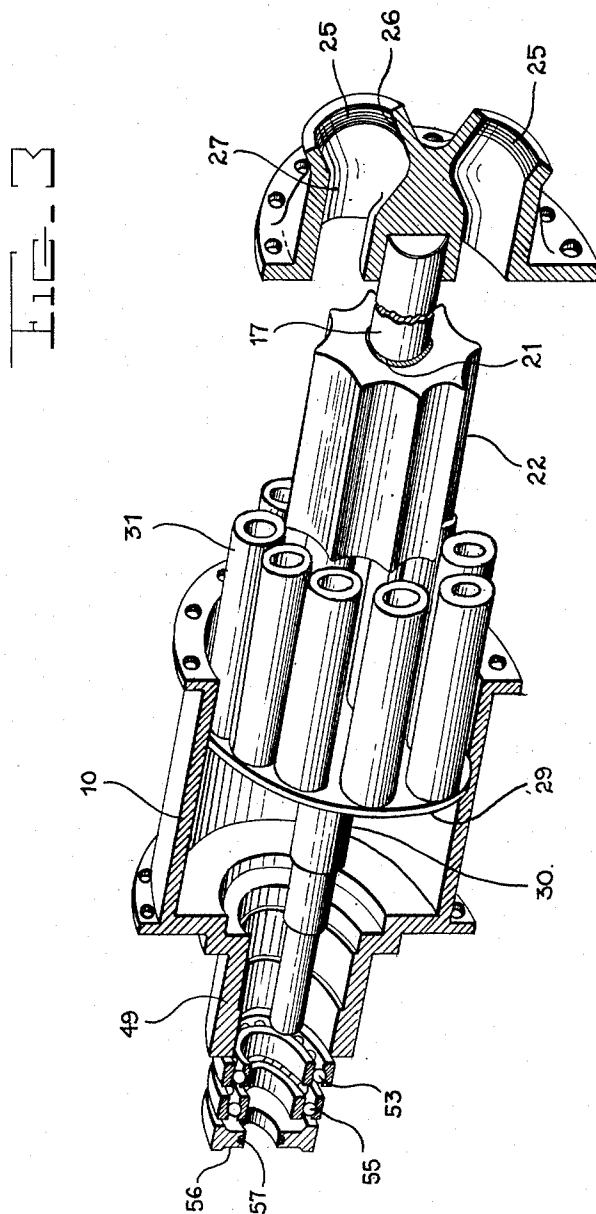
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Filed Feb. 17, 1945

3 Sheets-Sheet 2



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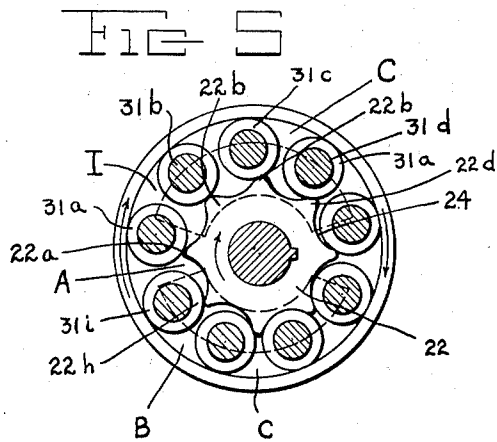
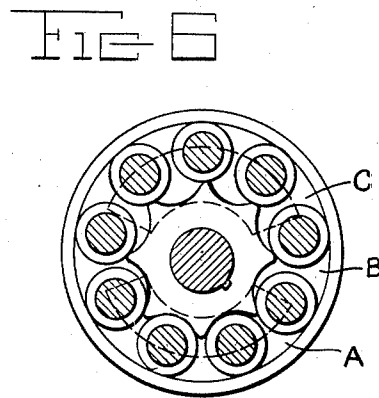
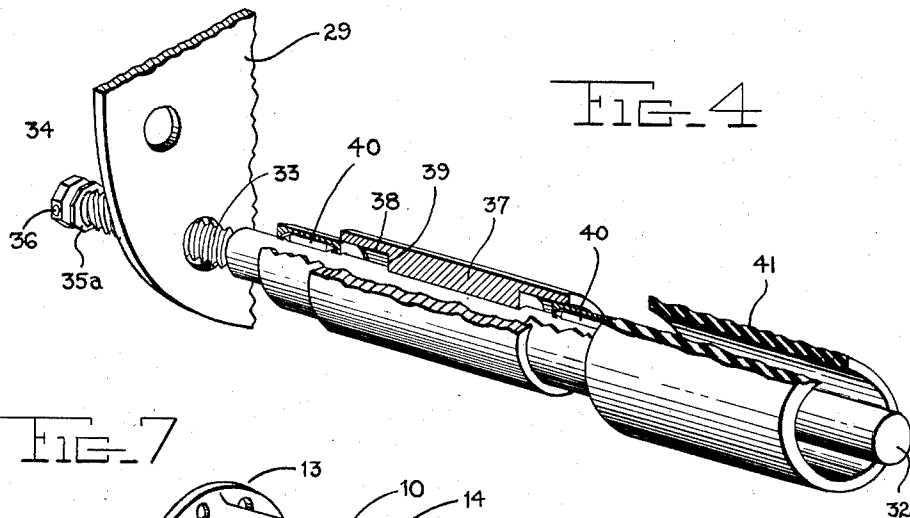
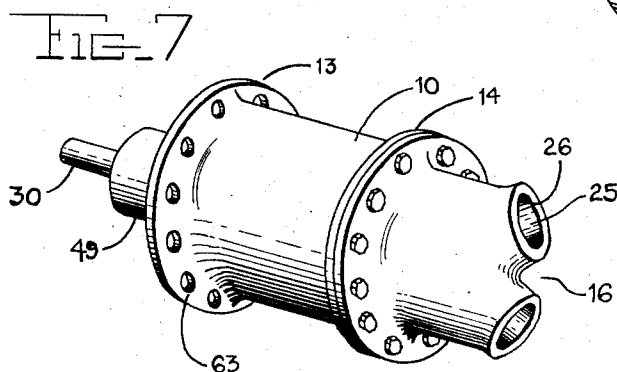
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Filed Feb. 17, 1945

3 Sheets-Sheet 3



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UNITED STATES PATENT OFFICE

2,672,825

HYDRAULIC PUMP AND MOTOR

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Application February 17, 1945, Serial No. 578,418

10 Claims. (Cl. 103—126)

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My invention relates in general to pumps, and more particularly to rotary pumps and to elements and assemblies for the same.

An object of my invention is the provision of a rotary pump for fluid which is simple, compact and rugged in construction, in which an effective fluid-tight seal is maintained between the working parts, and which pump with a minimum of maintenance and repair displays high efficiency and long life even under adverse high pressure and heavy duty operating conditions.

A further object of my invention is the provision of comparatively simple and compact roller assemblies for use in a pump of the general class described, which are easy to manufacture, of low first cost, displaying high efficiency, long life, and great durability over long periods of use under heavy-duty conditions of operation, and in which the parts subject to wear can be readily replaced with minimum disturbance of the other elements.

A still further object of my invention is the provision of a rotor assembly for a pump of the general class described, embodying the aforementioned roller assemblies, and which contributes materially, in simple and entirely efficient and reliable manner, in the establishment of chambers of variable dimensions, and which enables a substantially perfect seal to be maintained between the several pumping chambers throughout the useful life of the pump.

Other objects of the invention will in part be obvious and in part pointed out hereinafter.

My invention accordingly resides in the several parts, elements, and features of construction, and in the several operational steps, as well as in the relation of each of the same to one or more of the others, all as is more fully set forth in the claims at the end of this specification.

In the accompanying drawings, Figure 1 constitutes a longitudinal vertical section through a pump embodying certain novel features of my invention;

Figure 2 is a fragmentary end elevation of the pump; parts being shown in section;

Figure 3 is an exploded perspective view, parts being omitted for clarity, illustrating the relation to each other of the several parts of my pump construction;

Figure 4 is an exploded perspective view, showing the details of my new outer pump rotor assembly;

Figures 5 and 6 are elevations, entirely schematic, showing the pump assembly in different phases of the same operating cycle; while

Figure 7 is a perspective view showing the pump assembly ready for operation.

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As conducive to a clearer understanding of certain features of my invention it may be noted at this point that rotary pumps of a class heretofore known include a plurality of pumping chambers between fluid intake and output lines. In this particular type of pump, teeth much on the order of gear teeth are provided on inner and outer rotor members, the number of teeth on the inner rotor being one less than the number of corresponding teeth on the outer rotor. The teeth and tooth valleys of the rotors form between themselves and with the pump casing a plurality of sealed pumping chambers which vary in volume during relative rotation of the rotors. With proper proportioning of the pump rotors and teeth thereof the pumping chambers are sealed against each other so that when a fluid such as air or liquid is introduced through a suitable port into the pumping chambers at a point where these latter have approximately their greatest dimensions, then in subsequent phases of any particular revolution of the driving rotor, the several pumping chambers continuously decrease in dimensions to a minimum, thereby compressing the fluid contained therein. Compressed fluid is discharged through a suitable port provided in the pump housing adjacent the region of minimum dimensions. Pumping action thus is based fundamentally on the relative rotation between the inner and outer rotors and is caused by differential movement of the teeth. A typical form of such construction is, for example, described in U. S. Patent to Hill, No. 2,011,338.

A multi-chambered pump, while retaining its smooth, continuous discharge, displays many practical advantages as contrasted with certain other heretofore known types of pumps. Possessing much greater efficiency for given volumetric capacity, and being capable of operation under pressure heads of important values, rotary pumps have received wide-spread acceptance in the art. The pumping units when properly constructed display hydrostatic balance, so that at least to a certain extent, a sort of floating action is occasioned between the moving pump parts.

Despite the manifest advantages of the multi-chambered rotor type construction, however, certain practical disadvantages are observed during operation. The rotors are found to wear too rapidly which is a serious problem considering that necessarily expensive broaching operations are required in the production of new parts. As a consequence of wear, the liquid seals between the several pumping chambers deteriorate so that following comparatively short periods of use, operating efficiency falls off appreciably from initial

value. After comparatively short periods of use, unless appreciable repair work is undertaken, the pumps are incapable of further operation especially against high-pressure heads. Clearly, further developmental work is required to reduce the wear encountered between the cooperating contacting rotor elements. It has been observed that the contact between the cooperating parts of the inner and outer rotors is frictional and sliding in nature. Particularly when the fluid undergoing compression is a gas, undue wear is found to occur between the contacting parts probably due to insufficient lubrication. Definite limitations thus are placed upon multi-chambered rotary pumps.

It has been suggested in my companion application, Serial No. 535,083, filed May 11, 1944 and entitled Hydraulic Pump and Motor that important increase in life could be achieved by providing one of the rotors, preferably the outer rotor, with a plurality of anti-friction rotating elements such as for example wheels or rollers in lieu of gear teeth. These anti-friction members serve to bear against the teeth of the inner roller, defining with them and with the pump housing, a plurality of separate pumping chambers. This novel construction is found to result in considerable improvement in durability; greater resistance to wear is achieved along with much higher efficiencies. The pumps retain their good seal between the rotors, and hence their usefulness, under high-pressure operation.

From time to time, however, it is still necessary to replace either the rotating elements of the outer rotor or the teeth of the inner rotor, or both. Gradual decrease in efficiency is to be observed over a period of time before major replacement or repair actually is required. The seals between the separate fluid-pumping chambers gradually deteriorate. It is highly desirable, therefore to obtain further important improvements in the wearing qualities of the cooperating members and in the maintenance of a more perfect fluid seal between the several pumping chambers.

An important object of my present invention, therefore, is to achieve a substantial elimination of the disadvantages and deficiencies of the art as heretofore known; and to provide a smooth-action pump for continuous output, heavy-duty requirements, in which long life of the pump parts is realized in simple and expeditious manner; and in which wear is reduced to a minimum; and excellent sealing is maintained at substantially all times between the several separate pumping chambers.

Referring now more particularly to the embodiment of my invention disclosed in the drawings, and having particular reference to Figure 1, a pump housing 10 is provided, generally cylindrical in shape, and internally bored at 11, this bore terminating at its inner end in a machined wall 12. For example, the wall comprises a plane surface, substantially perpendicular to the axis of the bore and recessed in the manner hereinafter described. At the ends of the pump housing 10 peripheral flanges 13 and 14 are provided, preferably with reinforcing ribs 15 disposed on the back faces thereof. These flanges permit mounting the pump on a suitable base, and provide for attachment of a pump end cap indicated generally at 16, to the end of the pump housing 10 adjacent the end of bore 11.

Cooperating and nested outer and inner rotors are provided within the bore 11, the details of

which rotors will be more fully pointed out hereinafter. The inner rotor includes a shaft 17, extending substantially parallel to the axis of bore 11 and journaled in recess 18 in end cap 16 in eccentrically disposed relation with the bore axis. The center line 19 of recess 18 and shaft 17 is shown in Figure 1 as disposed eccentrically to the center line 20 of the bore 11. The recess or well 18 is sufficiently deep, and the shaft 17 is of sufficiently rugged construction that the shaft extends, with cantilever suspension and with requisite rigidity, into the bore 11, where it cooperates with the outer rotor. End cap 16 and housing 10 are made fast and fluid tight in a suitable manner as by bolting, or the like to serve as pump casing elements. In the present embodiment, bolts are employed.

The shaft 17 is held fast in well 18. A sleeve-type bearing 21, such as a journal, ball- or roller-bearing, is carried by the shaft 17 over that portion thereof extending into the bore 11. A toothed rotor or shell 22, of substantially the same length as the bearing 21, is rotatably carried by the latter (see also Figure 3). This rotor 22 is symmetrically disposed with respect to bearing 21, and carries a plurality of peripheral teeth suitably contoured, having for example epicycloidal or hypocycloidal contours, the number of these teeth being selected according to the dictates of convenience, usually ranging from about six to eight teeth, and the present construction being shown as having eight teeth. The teeth comprise a series of cusps 23 and troughs or valleys 24.

Ports 25 are provided in end cap 16, as illustrated in Figures 1 and 7, the upper port conveniently constituting the inlet port and the lower port constituting the outlet. In Figure 1 inlet port 25 is shown at the upper part of cap 16, having a wide bore 26 disposed centrally thereof and at an acute angle to the horizontal. As the bore 26 penetrates end cap 16, it tends to straighten out into a horizontal path as at 27. Fluid, such as gas, water or oil enters through upper port 25 to bore 11.

Also disposed within the bore 11, concentrically thereof and in cooperating but eccentric relation with the inner rotor, is an outer rotor assembly, perhaps best shown in Figures 3 and 4. This outer rotor comprises a disc or spider 29, of substantially the same diameter as that of bore 11 and rotatable therein. The outer rotor also includes a plurality of uniformly spaced substantially parallel roller shafts 32 supported as cantilevers on the spider at substantially uniform radial distances from the spider axis or axis of shaft 30. Rollers, indicated by the general reference numeral 31 are mounted on corresponding ones of the shafts 32 and are concentric with the same. These rollers are one greater in number than the number of peripheral teeth on the inner rotor 22, thus contributing to the differential action later to be described. In the particular embodiment undergoing description there are nine such rollers. Fast centrally to the outer face of spider 29 is a pump shaft 33, stepped for reasons which will be developed, and extending exteriorly through the closing end plate 10a of pump housing 10.

The construction of the spider-carried rollers assemblies constitutes an important feature of my invention. A typical one of these assemblies is illustrated in section at the bottom of Figure 1, also see Figure 4. The roller assembly comprises the central roller-supporting shaft 32 having a length slightly shorter than bore 11,

and extending through spider 29. The roller shaft terminates adjacent the spider in a flanged or offset head 35 and threaded end 33, and is made fast to the spider as a cantilever by suitable means such as washer 35a and hexagonal nut 34. Conveniently, a keyway 36 is provided through the nut 34 and threaded end 33 for the reception of a cotterpin or the like, not shown. It is essential that the rollers 31 rotate about their shaft 32 as they revolve about the spider axis 20. To ensure this, a roller sleeve 37 is fit to fairly close tolerances about shaft 32 with reliance being placed on any suitable type or bearing for establishing the rotational relation of the sleeve and shaft. In the preferred embodiment, I prefer to recess the sleeves 37 annularly adjacent each internal end thereof, as at 38, to provide annular shoulders 39, and to mount in the annular spaces provided by recesses 38 and shaft 32, a plurality of needle bearings 40. These needle bearings enable anti-friction rotation of the sleeves 37. For many advantageous purposes which will be more fully pointed out, I jacket the sleeve 37 with a snugly fitting cover 41 of material which is elastic or flexible or both.

It is to be noted that the pump shaft 30, carrying the outer rotor assembly, is subjected to heavy loads. Moreover, proper consideration must be given to the construction of the shaft bearings to provide requisite support to permit free rotation of the shaft without binding and without wear while at the same time providing for proper seal between the shaft and the pump bore 11.

Accordingly, the inner face of the wall 10a of the pump housing 10 is centrally recessed at 42 for the reception of a circular disc-like sealing plate 43 which, when mounted to end wall 10a by suitable means such as threaded bolts 43a has its working surface substantially flush with the machined end surface 12 of bore 11. Certain of these details are best illustrated in Figure 3. Plate 42 (Figure 1) is centrally apertured at 44 for the passage therethrough of pump shaft 30, freely but with small clearance. The plate 42 is annularly and outwardly shouldered at 45, to serve as a retainer for nested and cooperating bushings 46 and 47 disposed in the space provided between retainer 45 and the shouldered base 48 of pump shaft 30 where the latter joins spider 29. These bushings 46 and 47 are formed of suitable and well-known materials, such as hard rubber, rubber composition, fiber, or the like. Bushing 47 takes the form of a ring encircling base 48, and preferably is of softer material than bushing 46. This latter ring-shaped bushing has a peripheral flange 46a engaging about the left end of bushing 47 and terminating short of the base 48.

A boss 49 is struck centrally and outwardly from end 31 of pump housing 10. This boss is internally recessed at 50. Bore 50 is annularly shouldered at 51 at a point short of its right end in Figure 1. The stepped shaft 30 is similarly shouldered at 52. A suitable anti-friction bearing, shown as roller bearing 53, is provided on the right end of shaft 30 and is snugly received and retained between the two annular shoulders 51 and 52.

A further annular shoulder is provided at 54 (Figure 1) intermediate the length of the shaft 30, and at a point short of the left end of bore 50. A suitable anti-friction bearing fitting snugly against shoulder 54, typically a ball bearing 55, is provided between boss 49 and shaft 30. A threaded bearing retainer 56, annular in shape

and having a sealing insert 57 of soft rubber, or the like, bearing against shaft 30, threads into the open outer end of boss 50 into locking position against bearing 55, firmly retaining the same in position. Suitable means are provided, such as a setscrew 58, for locking the bearing retainer 56 against loosening or backing off after it has once been set in position. The bearing and liquid seal assembly just recited, effectively serves to protect the pump against leakage through the shaft bearings. The bearings themselves operate in a bath of grease or oil injected through a suitable fitting 59 provided in the boss 49. Because the bushings 46, 47 and 57 come into contact with this lubricating oil or grease, it is desirable that they be formed of suitable material possessing the qualities hereinbefore mentioned, but which is preferably grease-resistant.

In a recommended sequence for assembling the pump, the rollers 31 are first mounted on the spider 29. The bushings 47 and 46 thereafter are slipped over shaft 30 and seated home on the base 48 thereof. In the meantime, bearing 53 is inserted through bore 11, from the right end thereof, until it seats against shoulder 51 in bore 50. Plate 42 is then screwed fast in position in the casing wall 10a.

The pump shaft with spider and roller assemblies attached is introduced from the left through the housing 10 and through plate 42. Bearing 55 afterwards is inserted from the left end of bore 50 until it seats against shoulder 54 on shaft 30. The threaded retainer ring 56 then is threaded down on boss 49, so as to hold down the bearing 55 in operable position. Set screw 58 is tightened to lock the retainer ring 46 in its proper working position, and grease is inserted through fitting 59. The inner rotor, bearing 21 and shaft 17 are nested in the outer rotor, following which the end cap 16 is fastened in place, and the pump then is in condition for operation. The appearance of the assembled pump is illustrated in Figure 7.

While it is possible for either the outer rotor or the inner rotor to comprise the pump-driving rotor and thus comprising the driven rotor, I prefer, in the present embodiment, to have the outer rotor serve as the driving member, receiving its driving impulse through shaft 30, which may be rotated in any desired manner, such as by gearing, belting, or by being otherwise connected with a source of rotational energy.

It is desirable that the inner rotor, constituting the driven member, receive its driving impetus solely from the inter-engagement of rollers 31 with the teeth and tooth valleys on rotor member 22. For this reason, it is important that frictional contact between spider 29 and shaft 17 be effectively prevented at all times. I accomplish this by the provision of an ingenious ball thrust bearing illustrated in Figures 1 and 3. A recess or well 60 of substantial depth is provided centrally through the right end of shaft 30 and spider 29. A small coiled spring 61 is carried in this well, and thrusts a ball or other anti-friction bearing member 62 against the adjacent end of the shaft 17 of the inner rotor. This effectively thrusts against the shaft 17, to the right in Figure 1, seated within well 18 in end cap 16, and insures that the shaft is held out of engagement with spider 29. In this manner, frictional losses of unpredictable character are effectively precluded between the spider and the shaft. Only

rolling contact, anti-frictional in nature, occurs between the two rotors.

Both flanges 13 and 14 at the ends of the pump housing 10, as well as the periphery of pump end cap 16 are provided with bolt holes 63 (Figure 1) adapted for the reception of bolts 64. These bolts serve to secure the end cap 16 on housing 10, and to make the pump fast to a suitable base, not shown.

As has been suggested, the inner rotor 22 comprises a plurality of radially disposed tooth cusps 23 between which are formed troughs or valleys 24 of cycloidal or other configuration (Figures 1, 3, 5 and 6). These teeth are uniform in contour and spacing and their number is one less than that of the rollers 31. The eccentric disposition of the inner rotor with respect to the outer rotor ensures that at any given time when one tooth 23 of the inner rotor is substantially fully enmeshed and engaged momentarily with two adjacent rollers 31 of the outer rotor, then the remaining inner rotor teeth are in various stages of contact with the remaining rollers 31 (Figures 5 and 6). When either of the two rotors receives a rotational impetus, these rotors will be moved, at high speed relative to the housing 10 and to each other. A slow change of phase occurs between the two rotors, inasmuch as the outer rotor will rotate at slightly lower angular velocity than does the inner rotor. Assuming nine outer rollers 31 for example, with eight teeth 23 provided on the inner rotor, then the outer rotor rotates at 8/9 the angular velocity of the inner. In other words, the outer rotor loses one revolution in the course of every nine revolutions of the inner rotor. Under such conditions, the outer rotor will pass through eight revolutions during the time required for the inner rotor to make nine revolutions. Calculation will show, by way of illustration, that if the outer rotor is driven at approximately 1680 R. P. M., the inner rotor will operate at about 1800 R. P. M.

The number and size of teeth 23 and rollers 31 in large measure determines the ultimate pump capacity. Moreover, proper choice of tooth contour appreciably affects the pump efficiency. As suggested in an earlier part of this discussion, these factors have all been the subject of careful study and investigation by earlier workers in the field. Their findings comprise the subject of earlier patents as well as articles in the general technical literature. Since such design does not per se comprise part of this invention, no effort will be devoted herein to a development of such constructional details. By the use of proper epicycloidal and hypocycloidal contours, or other suitable shapes, it is ensured that tight, yet rolling and anti-frictional contact is maintained between the cusps 23 and valleys 24 of the teeth of the inner rotor and the several rollers 31 of the outer rotor.

Figures 5 and 6, wherein are disclosed the momentary relation of the inner and outer rotors are different phases of the same cycle of my device. Fluid enters through upper port 25, serving as the pump intake port. This fluid may be either gas, vapor, or liquid or any mixture thereof. The fluid courses to the upper port of bore 11. With the two rotors revolving at different speeds, there is a continuous opening and closing of spaces or compartments defined between the two rotors. The constant contact of the teeth of the inner rotor with the rollers of the outer rotor create a series of small spaces filled with fluid. During any small increment of time, some of these spaces

are increasing in size. This creates a suction and draws fluid from the intake manifold into the spaces. Others of the spaces are decreasing in size causing pressure. These latter spaces discharge as they pass the exhaust manifold.

Compartment A is defined at the moment by roller 31a, tooth 22a, tooth 22h, roller 31i, and the pump casing. The contact of teeth 22a and 22h with rollers 31a and 31i, respectively, is such that the volumetric dimensions of the compartment A are increasing. An ensuing increase in volume of the compartment or space creates a suction drawing in fluid from bore 11.

The fluid continues to enter compartment A during rotation of the two rotors upward through the arc adjacent manifold 28, increase in dimensions of compartment A. This is during the suction or intake phase. During the interval, compartment A is substantially filled. Meanwhile as compartment A sweeps rapidly in a clockwise direction, indicated by the arrow, it is sequentially replaced at point of expanding dimensions by successive compartments B, C, etc., which in turn follow through the same cycle as compartment A. It is to be noted that as the rotor 22 continues in its clockwise rotation, compartment A, reaches a condition of diminishing volume at the right in Figure 6, to discharge fluid through the manifold and out through the cooperating lower or outlet port 25.

It should be noted that the compartment A is entirely sealed from the other compartments B, C, etc., except through the intake and exhaust manifolds. The seal is accomplished by snug fit of the cooperating elements; by the snug rolling contact of the rollers 31 against the cylindrical wall of bore 11; by the snug fit of these rollers against the pump casing end plates; and by the snug contact of the rotor 22 with the rollers 31. Each roller 31 is at all times in engagement with some one or the other of the teeth 23 of the inner rotor 22.

Returning now to further and somewhat more detailed consideration of the volumetric capacity of the several compartments, it will be seen that when the compartment A travels from the position shown in Figure 5 to the momentary position now occupied by compartment I, a considerable increase in volume takes place. Compartment I, which corresponds to a successive phase or position of compartment A, is seen momentarily to be comprised between roller 31a, tooth 22a, tooth 22b, roller 31b and bore 11. Subsequently, during the rotational movement, compartment A reaches station G. This represents the position of approximate maximum volume of the compartment. Compression is just beginning. It will be observed that compartment G for the moment is comprised of roller 31c, tooth 22c, tooth 22d, roller 31d and the pump casing. The fluid begins to discharge through outlet port 25. Pressure discharge continues during the travel of compartment A down through its arcuate passage where compression occurs. What has been described with respect to compartment A is true of all of the other compartments such as B and C. As the rotors continue past the position G of approximate maximum volumetric capacity, the compartments defined at this position decrease in volume with further clockwise rotation until a position of minimum volume is reached at approximately the lower end of the exhaust manifold.

Pumping chambers of continuously varying dimensions are formed due to the eccentric re-

lation of the inner rotor with respect to the outer and to the provision of the former with one less tooth than the number of rollers 31. Despite the high velocity of the rotors relative to the pump housing, it is nevertheless insured that the phase change between the rotors is relatively slow. Elimination of intake and outlet manifolds in bore 11 ensures that smooth pump action is had both on the intake and discharge sides. Constant change and replacement occurs between the teeth and rollers comprising each pumping compartment, and due to the relative phase displacement between the rotors, the elements defining any given compartment will be different from one revolution to another.

The purpose of jackets 41 about roller sleeves 37 (Figures 1 and 4) of material which is either flexible or elastic, or both, will now be explained. These jackets, may be formed of any suitable material possessing the aforementioned qualities, but preferably are formed of rubber or sharkskin, which are soft and yieldable. The rollers, thus readily adapt themselves to the contours of the cooperating teeth 23 and bore 11 and serve admirably to provide a perfect seal therebetween, substantially fluid-tight in nature. The characteristic yieldability of the rolls tends admirably to substantial reductions in cost and time of production. Nicety of fitting no longer is of utmost importance. The jacket adapts itself to and adjusts for any moderate regularities between the surface of the outer and inner rotors. A sort of squeegee action insures that fluid cannot escape between the rotor elements from one pumping chamber to another. Moreover, this perfect seal is maintained throughout the life of the jacket. Wear on both the inner rotor teeth and outer roller sleeves is reduced to negligible value. Eventually should the jackets be worn through they may be readily, simply and inexpensively replaced by backing off end cap 16 and either replacing the jackets with the outer rotor spider in position, or after the latter has been removed from housing 10. New jacketing material is applied for example either by frictional fit, adhesive, or the like with minimum difficulty.

The new outer rotor roller elements which I provide rotate freely and easily against the cooperating cusps and valleys of the teeth of the inner rotor. The bearings 40 are either lubricated by the fluid entering the intake port, or they may be made self-lubricating if desired. The rollers do not of themselves involve any substantial element of first cost, and require no expensive manufacturing process. They are simple and inexpensive to produce, and tend inherently towards long wear. The slow phase change between the rollers contributes to this tendency, to long life and low rate of wear. Rolling contact with a minimum of sliding friction contributes towards minimum dimensional change during use.

I have described my new device as a pump and predict its principal sphere of utility in that field. Motor action, however, can be accomplished simply by reversing the course of the fluid through the ports of the housing and by permitting the fluid under pressure to enter the chambers of minimum volume and thereafter to expand while each chamber is moving into its position of maximum volume. In such instance, it is possible to achieve power take-off through either the inner rotor or the outer rotor.

As many possible embodiments may be made of my invention and as many changes may be

made in the embodiments hereinbefore set forth, it will be understood that the matter described and illustrated herein is to be interpreted as illustrative and not as a limitation.

I claim:

1. In a pump assembly, the combination of a housing having a cylindrical bore therein and inlet and outlet passages therefrom, and nested driving and driven members eccentric to one another having continuously cooperating elements thereon, the number of elements of the inner member being one less than the number of elements of the outer member, thereby providing a plurality of separate pumping compartments of continuously varying dimensions, the elements of the outer member comprising rollers jacketed with flexible material.

2. A pump comprising a casing having a pair of substantially diametrically opposed inlet and outlet ports, a peripherally-toothed inner member disposed eccentrically in said casing, an assembly of flexible-material-jacketed rollers disposed within said casing and about said toothed member, said rollers engaging and cooperating with the teeth on said toothed member to provide a plurality of separate pumping compartments, each of continuously variable dimensions with those of the increasing dimensions cooperating with inlet port and those of decreasing dimensions with outlet port, said assembly constituting a driving member, and shaft means extending exteriorly of said casing for powering the driving member.

3. A pump comprising, in combination, a pump casing with inlet and outlet openings therein, a stub shaft mounted eccentrically therein, a toothed member mounted on said stub shaft, a cooperating member comprising a plurality of rollers disposed within said casing and about said toothed member, the teeth and rollers of said members inter-engaging and forming individual pumping compartments of varying sizes between rollers and teeth with expanding compartments in communication with said casing inlet opening and the contracting in communication with said casing outlet opening and one of said members being adapted to drive the other, means for powering the driving member, and a thrust bearing provided between said powering means and the adjacent end of the stub shaft for thrusting apart the driving and driven members, whereby the torque of the driving member is transmitted to the driven member only through the cooperating rollers and teeth.

4. A pump comprising, in combination, a cylindrical pump casing having substantially diametrically opposed inlet and outlet passages therein, an outer rotor in said casing including a plurality of substantially uniformly spaced rollers in rolling contact with the casing wall and spider means at corresponding ends of said rollers supporting the same as cantilevers, a stub shaft extending from said pump casing substantially parallel to the rollers and having an axis eccentrically disposed with respect to the outer rotor, and an inner rotor concentrically mounted on said stub shaft and having gear teeth in variable conditions of mesh and contact with said rollers so as to define within the pump casing a plurality of pump compartments of variable volume, the compartments of increasing volume being in communication with the casing inlet passage and those of decreasing volume with said outlet passage.

5. In a pump assembly, an outer rotor, a pe-

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ripherally toothed inner rotor disposed eccentrically within the outer rotor, said outer rotor having spider means disposed at one end of said inner rotor carrying a plurality of substantially uniformly spaced rollers disposed about and engaging the teeth of said inner rotor, and an anti-friction, spring-pressed thrust member disposed centrally in said spider means of said outer rotor and thrusting against one end of the supporting shaft of the inner rotor to separate the same from the spider means.

6. As an element of an assembly having inner and outer rotor assemblies, a roller frame for said outer rotor assembly including about its periphery a plurality of substantially uniformly spaced parallel rollers, and a spring-pressed ball thrust member disposed substantially centrally of said frame in a well provided therein for contacting said cooperating inner rotor assembly.

7. As an element of a pump assembly, a roller frame, a pump shaft fast substantially centrally of said frame for rotating the same, said frame including along its periphery a multiplicity of substantially uniformly spaced parallel, roller-supporting shafts fixed at one end and free at the other, and a flexibly-jacketed roller sleeve snugly disposed in rotatable manner about each such roller-supporting shaft.

8. In an assembly, an outer rotor, a peripherally toothed inner rotor disposed eccentrically within the outer rotor, said outer rotor comprising spider means with supporting shaft disposed at one end thereof and having a plurality of substantially uniformly spaced rollers disposed about and engaging the teeth of said inner rotor, and spaced bearing supports for said rotor shaft.

9. A pump comprising, in combination, a cylindrical pump casing having opposed fluid inlet and outlet ports communicating therewith, an outer rotor in said casing including a plurality of substantially uniformly spaced rollers in rolling contact with the casing wall and spider means at like ends of said rollers supporting the

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same and a central drive shaft for such spider, spaced bearing supports for said spider shaft, sealing means between spider shaft and housing, a stub shaft extending from said pump casing substantially parallel to the rollers and having an axis eccentrically disposed with respect to the outer rotor, and an inner rotor concentrically mounted on said stub shaft and having gear teeth in variable conditions of mesh and contact with said rollers so as to define within the pump casing a plurality of pump compartments of variable volume, the eccentricity of inner and outer rotors assuring communication of pump compartments of increasing volume with inlet port and decreasing volume with outlet port.

10. In a pump assembly, the combination of a housing having a cylindrical bore therein and having inlet and outlet passages therefrom, and nested outer and inner members received in said bore and having cooperating elements engaging with each other to form a plurality of separate pumping compartments, the elements of the outer member comprising anti-friction rollers jacketed with flexible material.

BARTHOLOMEW FRANK QUINTILIAN.

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FOREIGN PATENTS

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