June 23, 1942. J. E. WHITFIELD 2,287,716

FLUID DEVICE

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Fig. 1.

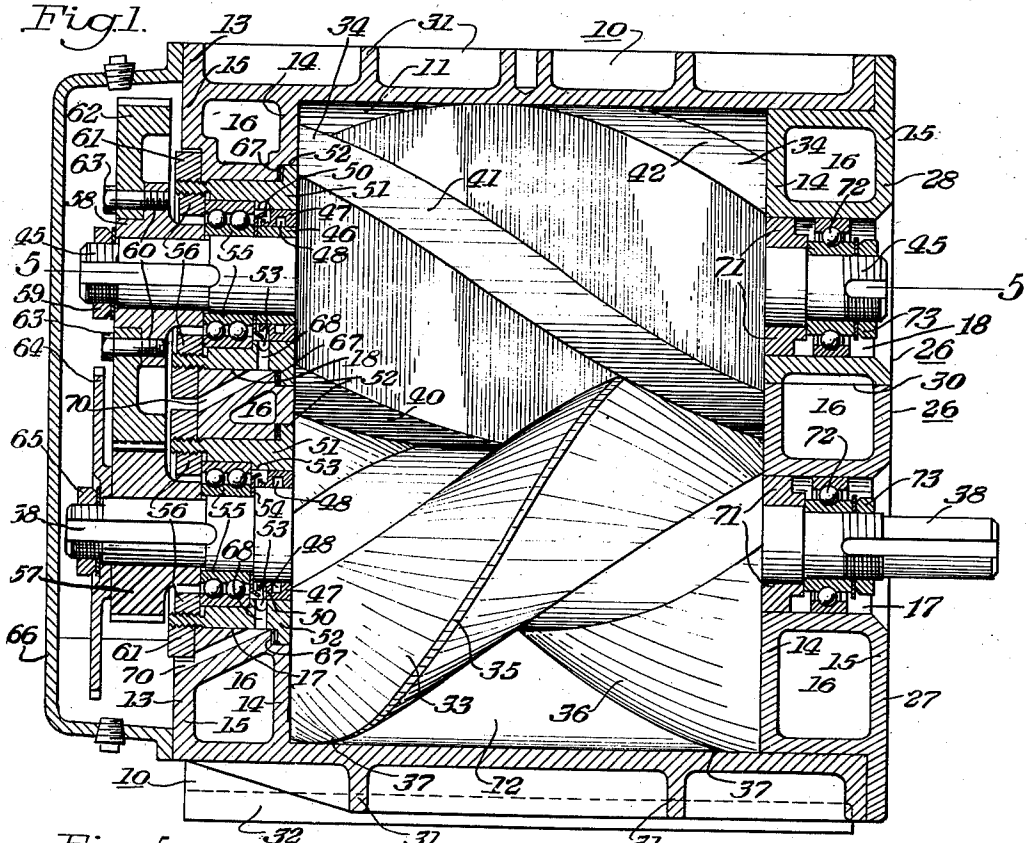
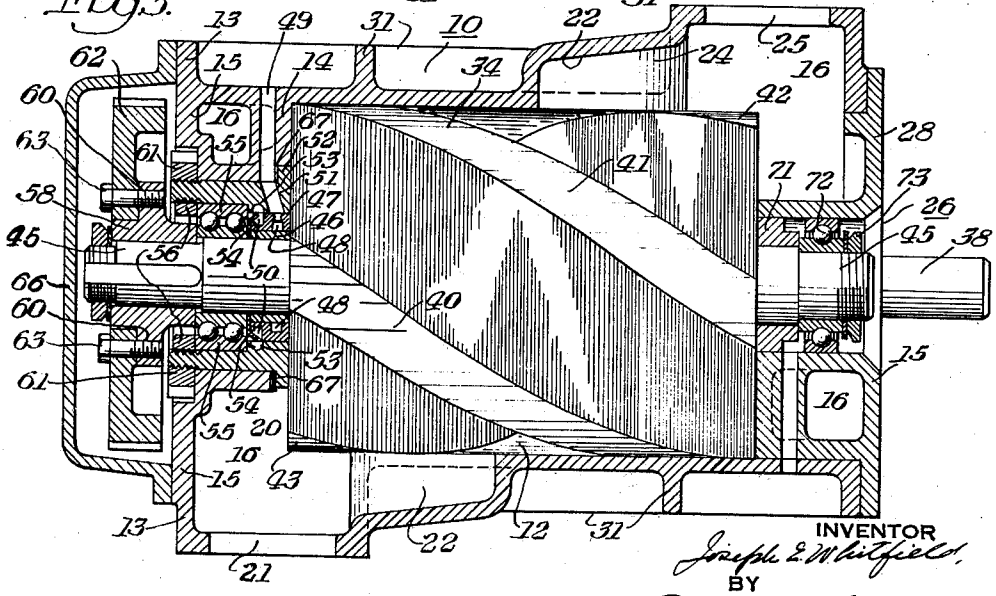


Fig. 5.



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Fig. 1.

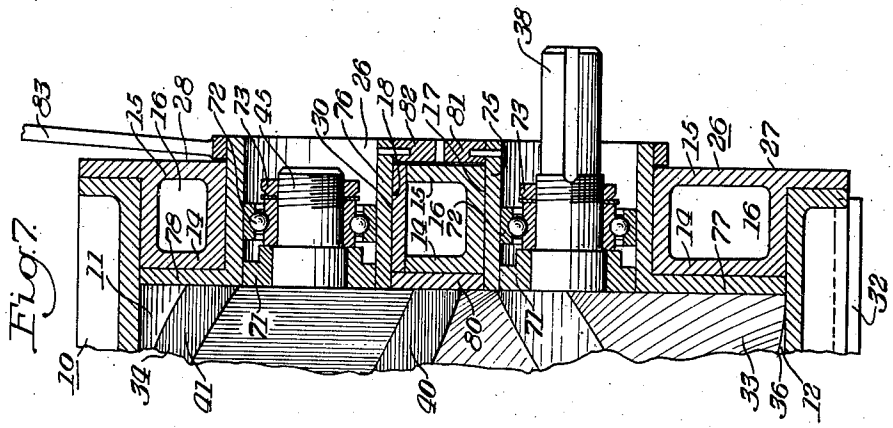
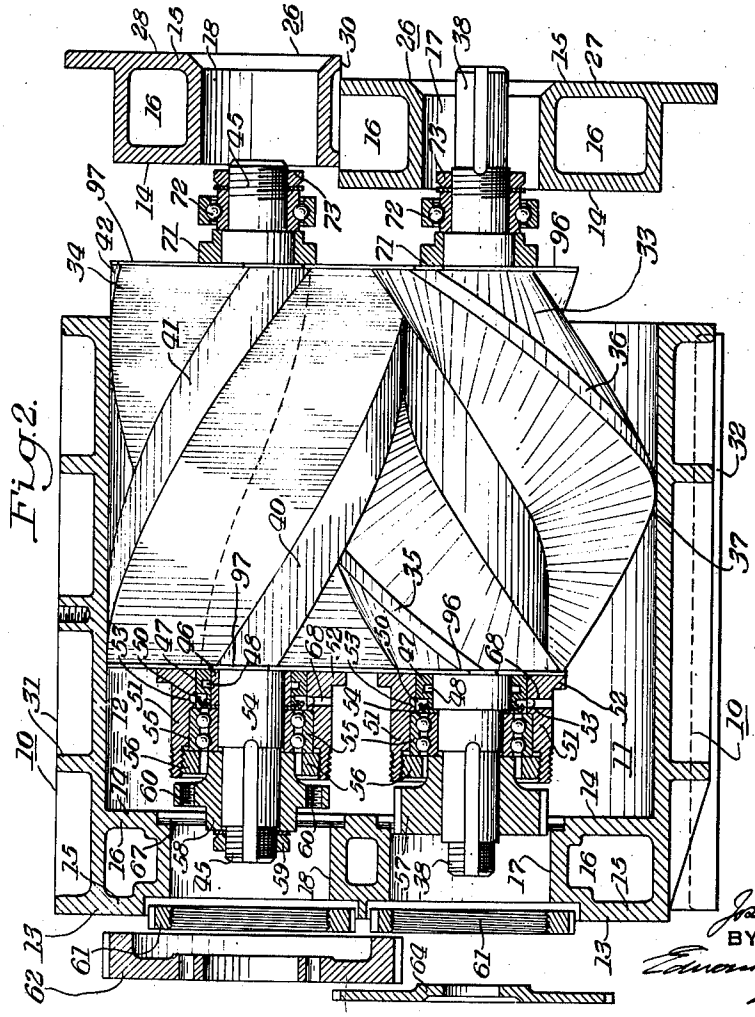


Fig. 2.



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Fig. 4.

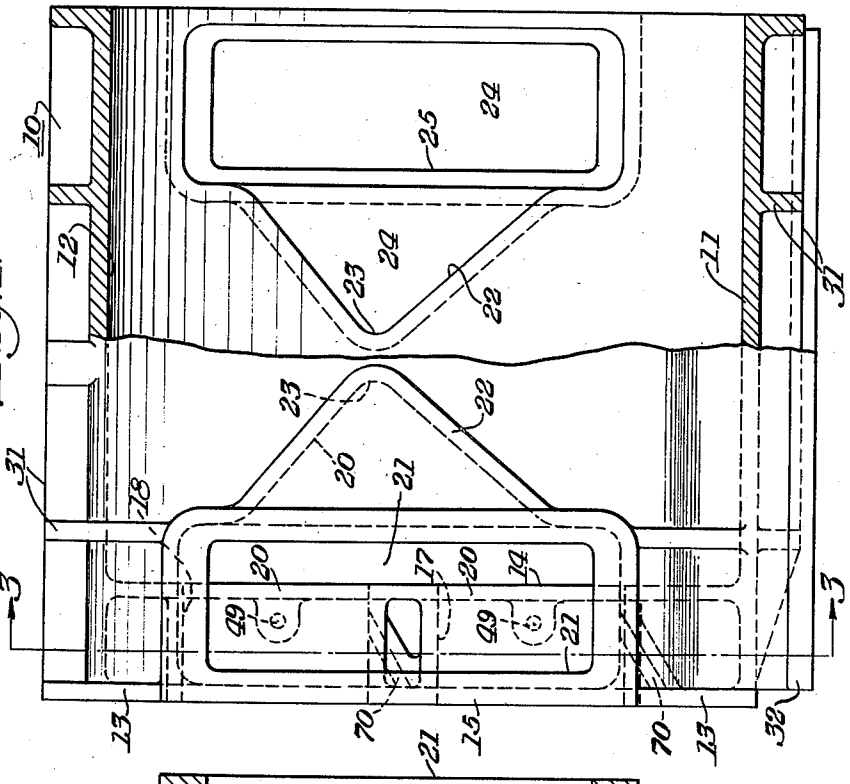
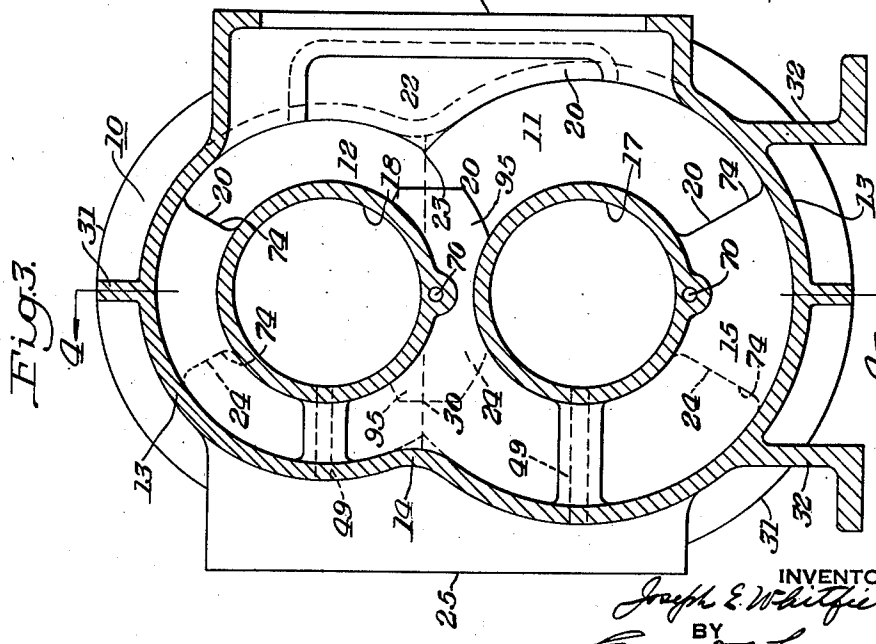


Fig. 3.



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Fig. 24.

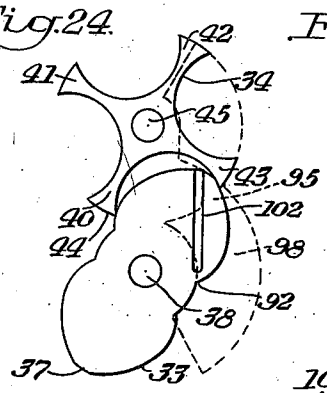


Fig. 25.

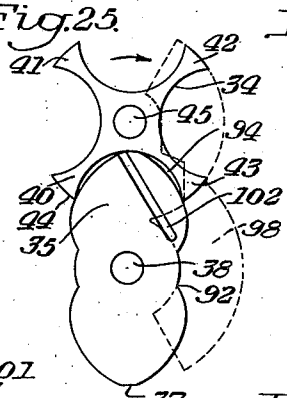


Fig. 26.

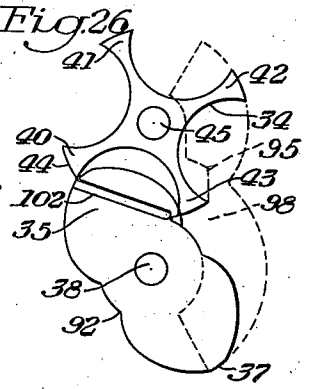


Fig. 27.

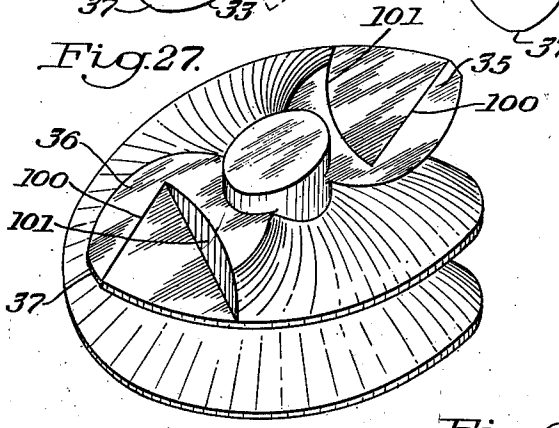


Fig. 31.

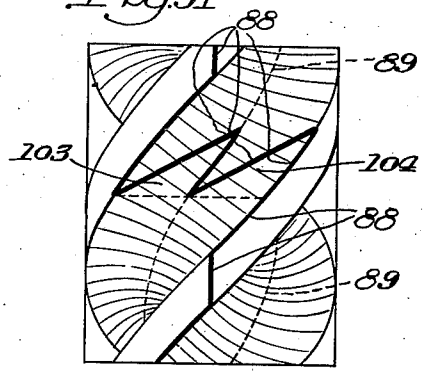
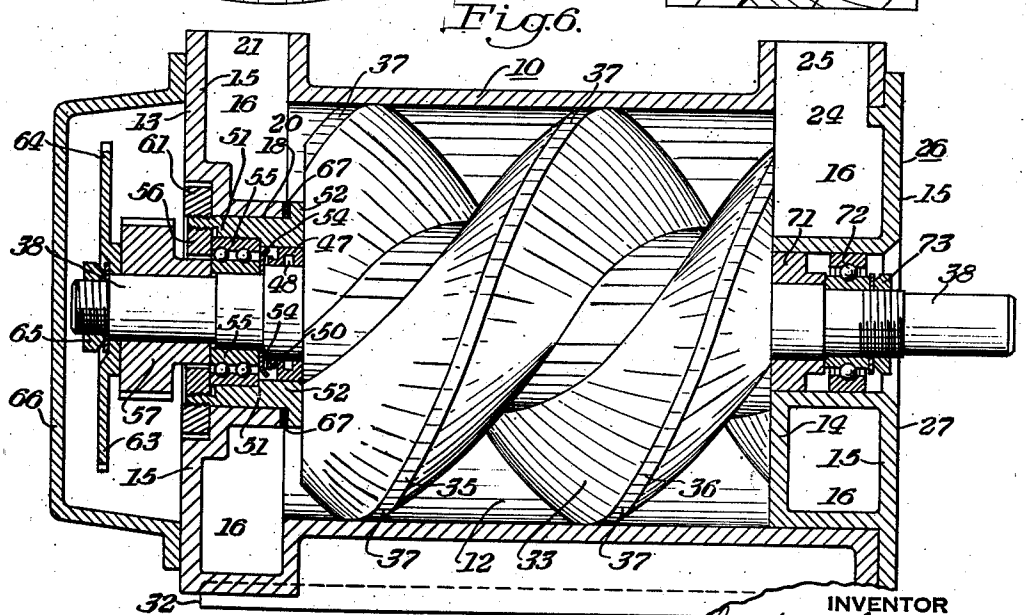


Fig. 6.



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Fig.8.

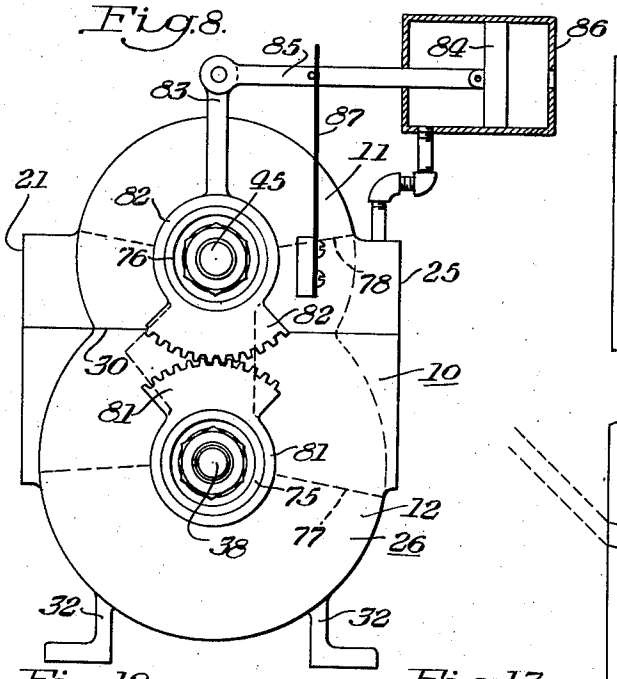


Fig.9.

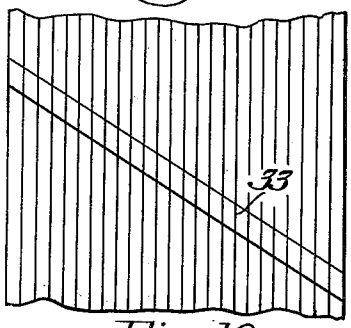


Fig.10.

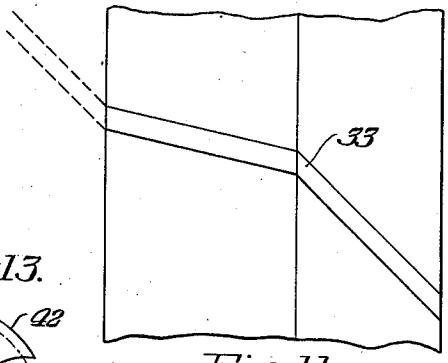


Fig.11.

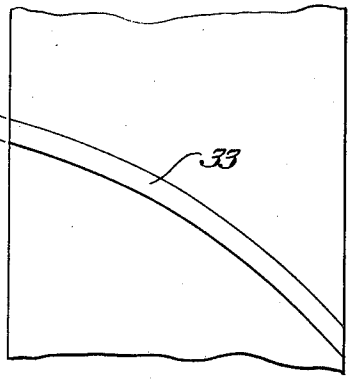


Fig.12.

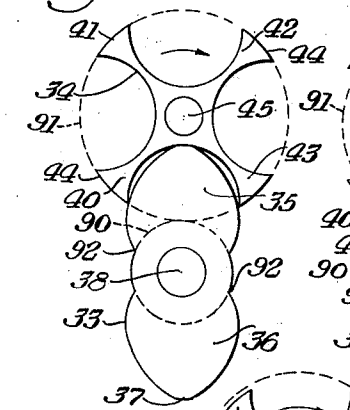


Fig.13.

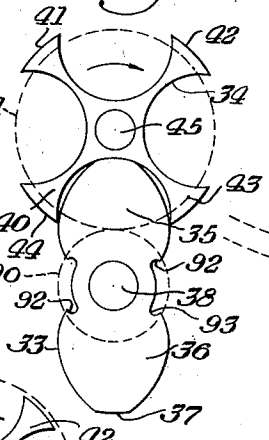
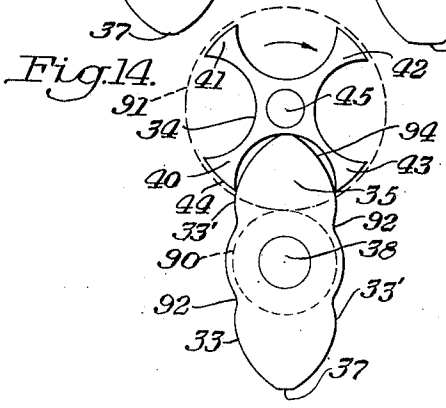


Fig.14.



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Fig.15.

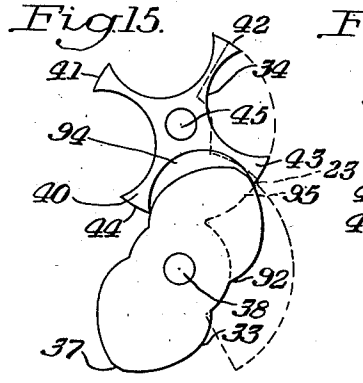


Fig.16.

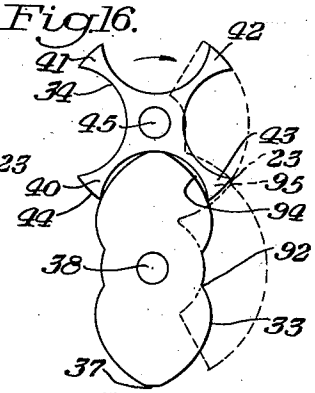


Fig.17.

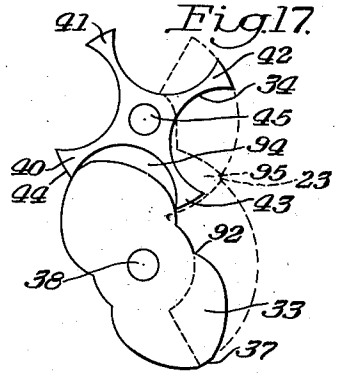


Fig.18.

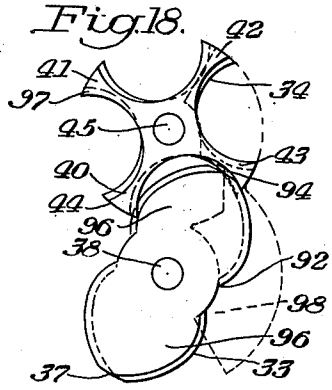


Fig.19.

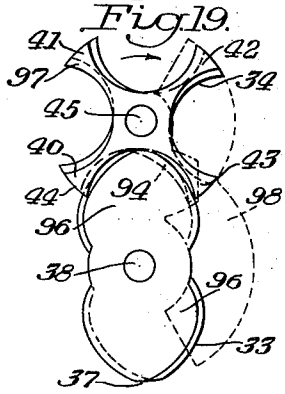


Fig.20.

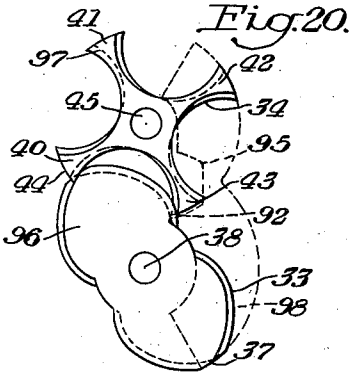


Fig.21.

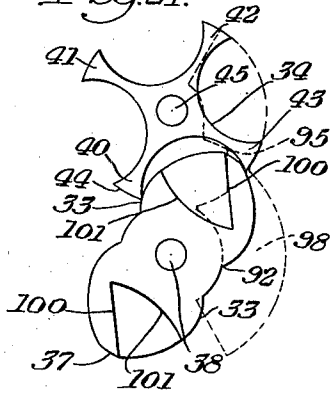


Fig.22.

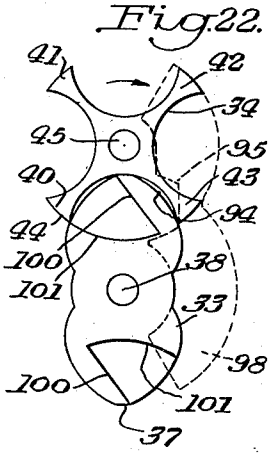
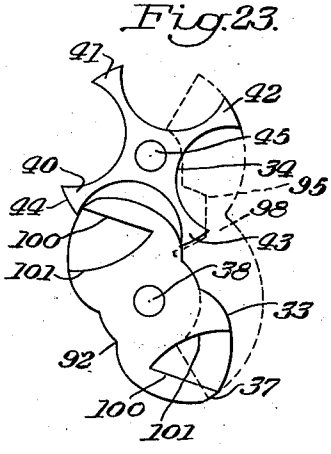


Fig.23.



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Fig. 28.

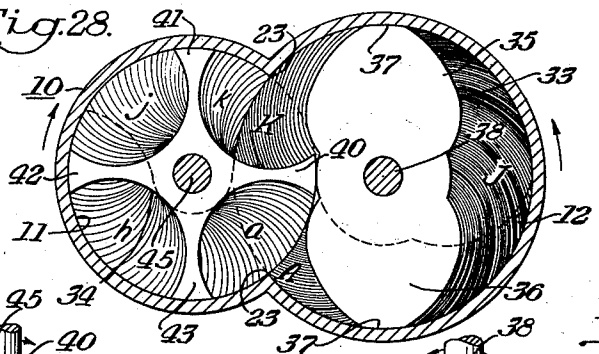


Fig. 29.

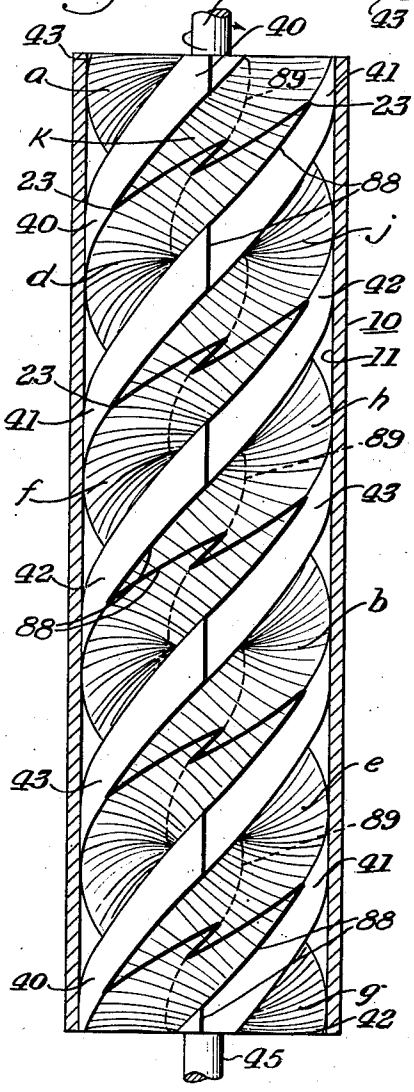
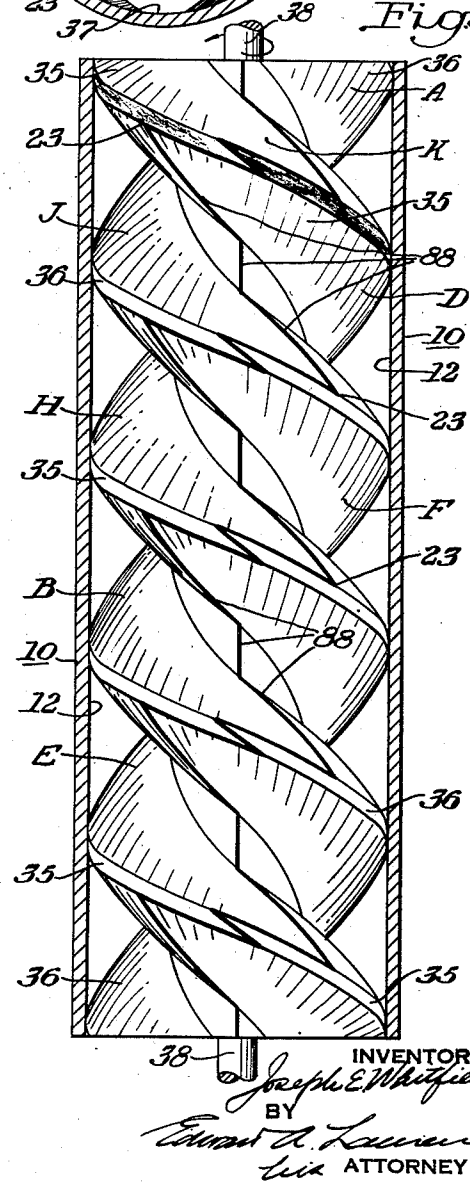


Fig. 30.



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Patented June 23, 1942

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UNITED STATES PATENT OFFICE

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FLUID DEVICE

Joseph E. Whitfield, Grove City, Pa.

Application April 22, 1941, Serial No. 389,729

22 Claims. (Cl. 230—143)

This invention relates generally to fluid pumps, motors, blowers, compressors and similar devices in which the rotary engaging members are provided with helical intermeshing threads, and more particularly to the novel structure of the rotary members, the case in which they operate and other associated parts making up any one of these devices.

These screw type fluid pumps, motors and the like have two or more helically threaded members rotatably supported with their axes parallel and with their complementary threads intermeshing to provide a continuous seal line for the full extent of their engagement. The casing encloses both of the members and the perimetral edge of each thread forms a seal therewith. Thus any flow of fluid from one end of the members to the other must pass through the grooves of the threaded members.

The threads must be complementary to permit rotation of the members. Thus one member has left hand threads while the other member has right hand threads.

One of the threaded members is referred to as the rotor and the other as the gate. When rotated the rotor threads, in effect, act a continuous series of pistons which slide axially in the same direction through the grooves of the gate and produce a successive series of pockets which convey the fluid from one end of the members to the other. The openings in the casing at the ends of the rotary members may be either the inlet or the discharge, depending upon the direction of rotation of the members.

These types of screw pumps, motors and the like are old in the art but they are impractical commercially because they inherently possess a leakage path back through the screw members from the discharge to the inlet. The volume of this leakage path is substantially the same as the pump displacement which renders them impractical.

To make these pumps practical it is necessary to seal this leakage path and increase their efficiency so that they may compete commercially with other types of pumps. This object may be accomplished as disclosed herein, by partially closing and sealing one or both ends of the rotary members and by carefully selecting the character of the coacting helical threads on the rotary members.

The most efficient combination of coacting helical threads on the rotary members is in the ratio of two to one. This thread ratio provides the largest capacity for a given size of the members and the ends of the rotary members are easily sealed. The speed of the rotary member having the least number of threads must be greater, and the pitch of its threads must be less than the rotary member having the most threads in order that they may properly coact to produce a continuous sealing engagement between each other.

If the ratio of the threads is one to one the rotary members would have the same speed and the pitch of the threads would be the same. It is practically impossible to prevent an end to end backward or leakage flow in a structure of this character. To function the whole of the ends of the rotor members must be covered, leaving a very small size opening or port, the location of which is critical. This makes the device inoperative as a liquid pump and impractical as a gas pump or blower, since the capacity and the efficiency are very low.

With the threads on the members being in the ratio of two to one a larger port opening may be used, which is an important advantage over a device where the threads have a one to one ratio.

Again if the ratio of the threads on the rotary members were two to five the capacity of the device is decreased for a given diameter of the rotary members which offsets what other advantages it might have.

After having selected the ratio of one to two for the threads of the rotary members it will be found advantageous to employ a plurality of parallel threads on both members because a greater number of independently sealed pockets may be produced in a shorter length of the rotary members.

The selection of two parallel threads for the rotor member provides a greater pumping capacity for a given diameter than three or more threads. The choice of two parallel threads on the rotor thus fixes the number of threads on the gate to four. In order that these threads may roll together the pitch of the rotor threads must be less than the pitch of the gate threads and the first rotor thread slides through the odd gate troughs while the second rotor thread slides through the even gate troughs. This thread selection also fixes the speed ratio of the rotor and the gate at two to one respectively.

The accurately formed complementary threads of the members will operate as smoothly as a set of gears and one member can drive the other and transmit power therebetween. However it is preferable to provide a slight clearance between

the members and between the perimetral edges of their threads and the casing to avoid wear. Timing gears are therefore employed to maintain the proper angular relation of the intermeshed threads.

The principal object of this invention is to obtain a novel and commercially practical fluid pump, motor and the like employing the above recited advantages.

Another object is the provision of a fluid transfer device through which the fluid may travel in either direction.

Another object is the provision of an improved form of casing structure for screw type blowers, compressors, pumps, motors and the like.

Another object is the provision of a fluid transfer device having a pair of rotary members with complementary coacting threads, one member being smaller in diameter than the other and having a greater number of threads which are disposed at a steeper pitch and wherein the coacting threaded surfaces of the members do not roll but slide relative to one another.

Another object is the provision of improved rotor and gate members having complementary helical threads in the ratio of one to two and wherein the rotor threads extend through a helix of approximately 180° and the gate threads one-half that amount.

Another object is the provision of rotor and gate members having intermeshing complementary helical threads wherein the helix angle of the threads change intermediate the ends of the members.

Another object is the provision of means for sealing the inherent leakage path through the rotary members.

Another object is the provision of means for varying the opening at the discharge end of the rotor and gate members to produce a variable rotary screw type compressor.

Another object is the provision of means for closing and maintaining a seal on the discharge ends of the rotor and gate members to prevent a succession of pockets from being prematurely opened to discharge when the seal line between the members has been interrupted as the pocket being discharged runs out.

Another object is the provision of a relief for fluids entrapped in pockets as they are running out at the discharge of a fluid pump.

Another object is the provision of complementary helical threads that coact to produce equal exposed areas on each side of the gate troughs which balance each other and eliminate power consumption in the gate member.

Other objects and advantages appear in the following description.

In the accompanying drawings a practical embodiment, illustrating the principles of the invention is shown wherein:

Fig. 1 is a vertical sectional view taken through the axes of the rotary members of the device.

Fig. 2 is a view similar to Fig. 1 with the parts being partially disassembled to illustrate their correlation.

Fig. 3 is a vertical sectional view of the casing, with the rotary parts removed, taken on the line 3—3 of Fig. 4.

Fig. 4 is a side elevation of the casing shown in Fig. 3, part of which is shown in section along the line 4—4 of Fig. 3.

Fig. 5 is a horizontal sectional view taken through the axis of the rotary gate member and the casing along the line 5—5 of Fig. 1.

Fig. 6 is a horizontal sectional view taken through the axis of the rotor member having a helical screw thread longer than that shown in Fig. 1, one end of which member is completely exposed within the case while the other end is adjacent a port, the rotor being shown in full.

Fig. 7 is a partial vertical sectional view taken through the axes of the rotary members and illustrating variable port members at one end of the device.

Fig. 8 is an end elevation of the device illustrating a fluid pressure operated servo motor for actuating the variable port members shown in Fig. 7.

Fig. 9 is a diagrammatic illustration of a helical thread of a rotary member wherein the pitch of the thread is constant.

Fig. 10 is a diagrammatic illustration of a helical thread of a rotary member wherein the pitch of the thread changes progressively from one end to the other.

Fig. 11 is a diagrammatic illustration of a helical thread of a rotary member wherein the pitch of the thread changes continuously from one end to the other.

Fig. 12 is an end view of the rotary members constructed coincidental with the pitch circles of their intermeshing gears.

Fig. 13 is an end view of the rotary members constructed with the gate member larger in diameter than the pitch circle of its gear.

Fig. 14 is an end view of the rotary members constructed with the gate member smaller in diameter than the pitch circle of its gear.

Fig. 15 is an end view of the rotary members showing a rotor thread proceeding into the gate.

Fig. 16 is an end view of the rotary members showing a rotor thread in full engagement with the gate.

Fig. 17 is an end view of the rotary members showing a rotor thread about to leave the gate.

Fig. 18 is an end view of the rotary members showing a rotor thread proceeding into the gate and sealing plates covering the trailing edges of the threads.

Fig. 19 is a view similar to Fig. 18 showing the rotor thread in full engagement with the gate.

Fig. 20 is a view similar to Fig. 19 showing the rotor thread about to leave the gate.

Fig. 21 is an end view of the rotary members showing a rotor thread proceeding into the gate and illustrating the cutting away of the leading face of the rotor threads to extend the discharge.

Fig. 22 is a view similar to Fig. 21 showing the pocket being shut off from the discharge and opening up to a successive pocket.

Fig. 23 is a view similar to Fig. 21 showing the pocket being relieved by the cut away portion to a successive pocket.

Fig. 24 is a view similar to Fig. 15 showing a slot cut from the leading edge to adjacent the trailing edge of the rotor threads for relieving the pockets to discharge.

Fig. 25 is a view similar to Fig. 24 showing the rotor thread in full engagement with the gate.

Fig. 26 is a view similar to Fig. 25 showing the slot relieving the pocket to discharge at its fully run out position.

Fig. 27 is a side elevation of the rotor member shown in Figs. 21 to 23 showing the leading portions of the threads cut away.

Fig. 28 is an end view of gate and rotor members at the discharge end showing one gate thread in full engagement with a trough of the rotor.

Fig. 29 is a side elevation of an elongated gate with the sealing line inscribed thereon and along which line engagement is made by the rotor when the members are intermeshed, as illustarted in Fig. 28.

Fig. 30 is a view similar to Fig. 29 showing the rotor with the same sealing line inscribed thereon.

Fig. 31 is a side elevation of a rotor thread designed to produce equal exposed areas on each side of the gate trough.

Referring to Figs. 1 to 5 of the drawings, the casing 10 of the pump contains two cylindrical chambers 11 and 12 disposed side by side in parallelism and merging into one another, forming a large chamber the cross-section of which is somewhat in the form of a figure 8. One end of the casing is provided with an integral chambered head 13 having an inner wall 14 which forms one end wall of the chambers 11 and 12 and an outer wall 15 with a space or chamber 16 therebetween. The head 13 is provided with the tubular openings 17 and 18, concentrically disposed with respect to the axes of the cylindrical chambers 11 and 12, respectively, for receiving the bearing and sealing structure which supports the rotary members.

A portion of the inner wall 14 is cut away to form the port 20 which may be either the intake or exhaust port, depending upon the direction of rotation of the rotary members of the pump. This port connects the cylindrical chambers 11 and 12 with the chamber 16. An opening 21 is provided in the wall of the casing adjacent the head 13 for connecting the chamber 16 to a conduit external of the casing. The port 20 is extended along the side walls of the chambers 11 and 12 as indicated by the V-shaped wall 22 which is wide at the opening 21 and narrows down to the sharp edge 23 near the center of the casing which is formed by the juncture of the walls of the cylindrical chambers 11 and 12.

At the other end of the casing the chambers 11 and 12 are open to permit the assembly of the rotary members therein. After the rotary members are inserted in the casing it is closed by the removable head 26. A port 24 is provided in the head 26 and is connected to without the casing by the opening 25. A second V-shaped wall 22 is provided in the side walls of the casing diagonally opposite to the port 20 and the opening 21. The apices of the V-shaped walls 22 extend to points just short of one another as shown in Fig. 4.

The removable head 26 may be made in one piece but it is preferably constructed in two pieces 27 and 28 separable along the plane 30 which intersects the edges 23 of the casing. The removable head 26 is also provided with the spaced inner and outer walls 14 and 15 forming the included chamber 16 that is connected with the interior of the casing through the port 24, similar in perimetral contour to the port 20. The head 26 is also provided with the tubular openings 17 and 18 which are aligned with their corresponding openings in the fixed head.

The location of the exterior connecting openings 21 and 25 are preferably positioned as shown in Figs. 3 and 4. However they may be positioned at any other location on the casing so long as they are connected to their respective chambers 16 in the heads of the casing.

The casing and the removable head are provided with suitable abutting flanges for securing the latter in place. The exterior of the casing is reinforced by the ribs 31 to provide strength to the walls of the casting which are of uniform thickness. The feet 32 may be formed integral with the casing and positioned at any convenient place which is determined by the location of the exterior connections of the ports 21 and 25, so that the pump as a unit may be properly adapted to other apparatus associated therewith. However it is advisable to arrange the feet so that the shafts of the rotary members lie horizontally, for obvious reasons.

The rotary members which are mounted in the casing comprise the mating helical screw thread members 33 and 34 which are arranged to operate in the cylindrical chambers 11 and 12, respectively. For convenience the member 33 is referred to as the rotor and the member 34 as the gate. However as they may either act upon or are rotated by the fluid medium, they do not engage for the purpose of transmitting power.

The rotor 33, shown in Figs. 1 to 5, is provided with two matched and oppositely disposed screw threads 35 and 36, the contour of which is later discussed in detail. Each thread is helically disposed and extends through approximately 180° from one end of the rotor to the other. Thus the profile of the thread 35 is congruent with the profile of the thread 36 at opposite ends of the rotor, and vice versa. The perimetral surfaces or crests 37 of these threads are cylindrical and snugly fit the cylindrical wall 11 of the casing. The rotor is preferably made hollow to reduce its weight and mounted on the shaft 38.

The gate 34 has four helical threads 40, 41, 42 and 43, which are formed complementary to the threads of the rotor. Each of these threads are helically disposed and extend through 90° from one end of the gate to the other. Thus the profile of each thread at one end of the gate is congruent with the profile of the next adjacent thread at the opposite end of the gate. The cylindrical crest surface 44 of the threads on the gate snugly fit the cylindrical wall 12 of the casing. The gate is mounted on the shaft 45.

As stated above the threads of the rotor and gate are complemental to one another and the threads of each member form grooves or troughs therebetween. The threads of the rotor in this instance are disposed in a left hand helical path, whereas the threads of the gate are disposed in a right hand helical path and the overall diameter of the rotor is larger than the gate.

The shafts 38 and 45 project beyond the ends of the rotor and gate, and since the drive is connected with the shaft 38 it extends beyond the head 26, as shown in Fig. 1, for receiving a pulley or gear to provide for the transmission of power to or from the device as the case may be.

The openings 17 and 18 in the heads are sufficiently large to receive bearing assemblies which are mounted on the shafts before the rotary members are assembled in the casing. The bearing assemblies on the shafts 38 and 45, which are mounted in the head 13, are the same with the exception that the shaft 38 carries a small gear and the shaft 45 carries a hub for a large gear which cannot be passed through the opening 18.

Each of these bearing assemblies in the head 13 comprises the sleeve 46 which has a press fit on the shaft. A seal ring 47 is placed over this sleeve. This ring has a running fit with the sleeve 46 and is provided with an internal annular groove 48 which is vented through a radial hole to the passageway 49 and thence to atmosphere, as shown in Figs. 3 and 5. An annular

groove 50 is formed on the outer end of each sealing ring.

A sleeve 51 is stepped over the sealing ring 47 and is provided with a radially extending flange 52 which abuts against the rotary member. An internal annular groove 53 is provided in the bore of the sleeve opposite to the groove 50 on the sealing ring.

An oil slinging disk 54 is placed on the shaft and held between the inner race of the bearing 55 and the end of the sleeve 47 and thus rotates with the shaft. The outer race of the bearing 55 engages a shoulder formed in the bore of the sleeve 51 and is held in place by the spanner nut 56.

The small gear 57 is keyed to the shaft 38 and abuts the inner race of the bearing 55, thus holding it in place. The mounting of the gear 57 completes the bearing assembly for the gear end of the rotor shaft 38. It will be noted in Fig. 2 that the gear 57 has ample clearance for passing through the tubular opening 17.

The gate shaft 45 has keyed thereto the gear hub 58 which is locked by the nut and washer indicated at 59. The gear hub 58 has ample clearance for passing through the tubular opening 18. The radially extending flange on the hub 58 is provided with a series of threaded openings 60. When the rotor and gate members are placed in mesh and the bearing assemblies are mounted in the head 13 through the tubular openings 17 and 18, the spanner nuts 61 are screwed on the outer ends of the threaded sleeves 51 to lock them in place and the gear 62 is slid in position in mesh with the gear 57 and is secured to the hub 58 by means of the bolts 63 as shown in Fig. 1.

A large oil slinger disk 64 is then mounted on the end of the shaft 38 by means of the washer and nut indicated at 65 in Fig. 1.

The left or gear end of the pump is thus completely assembled and a cover 66 is placed thereover to protect the gear and to provide an oil sump.

In order to provide the proper clearance between the ends of the rotor and the gate members and the wall 14 of the head 13, annular shims 67 are placed in the recesses of the tubular openings 17 and 18 which receive the flanges 52 of the sleeves 51.

The internal annular grooves 53 in the bore of the sleeves 51 are connected by the radial passages 68 through the sleeves 51 to the passages 70 which lead back to the oil sump formed by the cover 66. Thus the oil that is picked up by the large slinger 64 splashes over the case and gears and finds its way to the bearings where it is again picked up by the small oil slingers 54 and is discharged into the grooves 53 and drains back through the passages 68 and 70 to the sump.

Thus one end of the sealing ring 47 discharges any fluid that has leaked past the sleeve 46 directly to atmosphere and the oil slinger disk 54 at the other end directs the lubricating oil back to the sump within the case. With this novel arrangement the fluid and the oil are never permitted to mingle with one another.

The bearing assembly on the other end of the shafts 38 and 45 comprises the sleeves 71 which abut against the rotor and gate members at one end and are flush with a shoulder on the shafts at the other end. The bearings 72 are sealed by the sleeves 71 and the inner race abuts the shoulder on the shaft and holds the sleeve in place. These bearings are locked in position by a suitable washer and nut as indicated at 73.

The sleeves 71 have a running fit with the tubular openings 17 and 18 in the head 26. Any leakage of the fluid past the sleeves 71 is permitted to escape directly to atmosphere.

Referring now to Fig. 6, in this instance the threads on the rotor 33 extend through 360° or one complete revolution, and the threads of the complementary gate member extend through 180°. Thus the rotary members in this view are equal in effect to twice the length of the rotary members shown in Figs. 1 to 5. The actual length of the rotary members shown in Fig. 6 need not be twice the length of the rotary members shown in Fig. 1, for the same effect may be obtained by reducing the pitch of the screw thread without altering its shape.

In the structure shown in Fig. 6 the inner wall 14 of one of the heads is eliminated, and the corresponding ends of the cylindrical chambers 11 and 12 thus open directly into the chamber 16. The other head retains its inner wall with its port. As illustrated in the drawings, the wall 14 of the head 13 has been eliminated, thus converging the whole left end of the casing into the port 20. When the rotary screw members are rotated as indicated on the drawings, the fluid is drawn in through the opening 21 into the chamber 16 and is picked up by the rotary members and conveyed to the other end of the casing where it is discharged through the port 24 and the opening 25. When the threads of the rotor extend through 360° and the threads of the gate extend through 180° it is necessary to seal off selected troughs of the gate at one end of the pump. Thus the inner wall 14 of the head 26 with the port 24 are left intact at the right end of the casing as shown in this view.

Regardless of how many turns the threads of the rotor and gate are extended, it is necessary to partially seal off the rotors at one of the inner walls 14 of the heads 13 or 26. However the pump may be materially shortened by the use of the oppositely disposed ports in the walls 14 as shown in Figs. 1 to 5, and operated at a higher efficiency.

In the structure illustrated in Figs. 1 to 5 the radial edges 74 of the ports 20 and 24, as indicated in Fig. 3, may be positioned at an angle approximately 30° from the plane which intersects the axes of the two rotary members, and in the structure illustrated in Fig. 6 the port 24 may retain the same degree of opening. To convert these structures into compressors the radial edges 74 are moved toward one another, thereby delaying the opening of the successive pockets. The distance that the radial surface 74 is extended determines the degree of delay in the opening of the pockets to the discharge side of the pump and thus determines the degree of compression of the fluid within the pocket before it is discharged. If the port setting is the same as that shown in Figs. 1 to 5 and the system connected to the discharge of the pump is subjected to pressure, then, as each successive pocket opens to discharge, the fluid under pressure rushes back into the pocket to equalize the pressure of discharge. This momentary back flow is a loss, and to be avoided the same radial edges 74 are advanced to a position where the fluid in the contracting pocket independently reaches the same pressure as that in the discharge as the pocket is opened to the discharge. If the pressure in the system varies, then provision must be made to control the advance or retraction of the

radial edges 74 of the ports to regulate the instant at which the pocket opens to the discharge.

In Figs. 7 and 8 the tubular openings 17 and 18 in the head 26 are increased in diameter to receive the sleeves 75 and 76, the bores of which are of proper diameter for receiving the rotatable collars 71. Radially disposed arcuate sectors 77 and 78 are formed integral with the inner end of the sleeves 75 and 76, respectively. The inner faces of these sectors abut against the ends of the rotary members and the outer faces against the wall 14 of the head 26 to prevent the passage of liquid. The space intermediate of the sleeve 75 and 76 in the plane of the sectors 77 and 78 is occupied by the filler plate 80.

The port 24 is formed in the head 26 in the usual manner. However the edges of the arcuate sectors 77 and 78 provide the radial edges 74 of the port.

The intermeshing segmental gears 81 and 82 are secured to the outer ends of the sleeves 75 and 76 to provide the proper movement of the radial edges 74 at opposite ends of the port 24. As shown in Fig. 8, one gear sector may be provided with a radially extending arm 83 which is operably connected to a piston 84 by a link 85. The piston is arranged to reciprocate in the chamber 86 which is connected to the discharge 25 of the compressor. Thus the fluid under pressure in the discharge is effective on the piston for opening or closing the ports in the compressor against the force of a spring 87. By properly selecting the spring 87 the pressure of the fluid in the discharge of the compressor may be effective to set the edges 74 of the port to delay the opening of the pockets until the pressure of the fluid therein reaches the pressure of the fluid in the discharge system.

The pitch of the threads of the rotary members shown in Fig. 6 is different than the pitch of the threads of the rotary members shown in Fig. 1. However the pitch of each set is uniform throughout its length. Fig. 9 is a diagrammatic view illustrating threads having a uniform pitch from one end of the rotary members to the other. Slight pulsations are produced by the discharge of the successively formed pockets when the device is in operation. Thus each pocket being an independent conveying chamber for the fluid passing through the device produces a slight unbalance of the torque load. This unbalancing of the torque load may be compensated for by changing the pitch of the threads of the rotary members intermediate of their length. This change in pitch may be made progressive or in steps, as illustrated diagrammatically in Fig. 10, or by continuously changing the pitch as illustrated in Fig. 11.

It will be noted that in Figs. 10 and 11 the pitch of the threads decreases as the threads approach the discharge end of the device. Thus the pockets with the fluid therein travel at a lower rate of speed as they approach the discharge end. This structure is particularly advantageous for use in compressors wherein the delay of the opening of the pockets is made to compress the gas to the proper pressure before permitting it to discharge. The work expended in compressing the gas within the pocket is thus spread over a longer period of time, thereby reducing the peak of the torque load that is normally formed with threads of uniform pitch. In this manner the torque curve is flattened and the torque characteristics of the device are more uniform. If the compressor is to operate in both directions under the same conditions it is necessary to provide the same thread structure on each end of the rotary members as indicated by the extension to the left in Figs. 10 and 11. Thus the threads on each rotary member may be symmetrical from end to end or at both ends in that the end sections of the continuously advancing thread have a similar pitch which is different than the pitch of the intermediate portion of the thread and corresponding points along the generated curves of the threads in such case would be equidistant from a transverse plane passing through the center of each member.

Figs. 12, 13 and 14 illustrate three different forms of the thread profiles of the rotary members. These views are diagrammatic and the specific contours which they illustrate are too small to accurately portray the exact forms of the thread surfaces as these surfaces are generated curves and not true circles as they import. Since the rotor is provided with two threads and the gate has four, the rotor travels at twice the speed as the gate. Thus the ratio of the timing gears 57 and 61 is one to two. If the profiles of the threads of the rotary members are as shown in Fig. 12, the rotors will have the same diameter as the pitch circles of the timing gears wherein 90 is the pitch circle of the gear 57 and 91 is the pitch circle of the gear 62. With this construction the perimetral portions 44 of the gate threads have a rolling contact with the hub of the rotor and in all three forms the sharp edges of the gate threads have a relative sliding action with the sides of the rotor threads. The normal rolling surfaces of the rotary members are equivalent in diameter to the pitch circles of their respective timing gears the dimensions of which are determined by the number and the axial pitch of threads of the respective members. It should be noted that this design provides a sharp intersection where the sides of the rotor teeth join the rotor hub, as indicated at 92. This sharp angle must be accurately formed to produce a small clearance between the rotary members. This is very difficult to cut and is therefore considered to be impractical. However in this design a perfect continuous seal is formed between the members.

In the form shown in Fig. 13 the diameter of the gate is larger than its pitch circle, requiring the rotor surfaces to be undercut at 93. The edge surfaces 44 of the gate pass over the gap and reach the hub of the rotor and then follow the undercut arc. This form is also regarded as impractical because the undercut is difficult to produce and provides a leakage path. Again these two types of threads tend to roll and pack any foreign material rather than remove it.

It has been discovered that by making the diameter of the gate 34 smaller than its pitch circle, as shown in Fig. 14, and the hub surface or root surface of the rotor larger in diameter than its pitch circle, any foreign material is wiped out rather than being pressed between the members. The edges defining gate thread surfaces 44 may be kept sharp and continuously seal along the rotor thread surfaces intersecting at 92 and the perimetral surfaces of the gate threads follow a reverse curve and still maintain the proper sealing relation between the rotary members. This reverse curve, shown

in Fig. 14, is purposely exaggerated to emphasize the radius at the intersection of the sides of the rotor threads with the hub surface indicated at 92. The actual size of this radius is of course determined by the diameters of the gate and rotor members relative to the diameters of their true pitch circles. In this preferred form of thread section the perimetral surfaces 44 of the gate must have a combined rolling and sliding movement relative to the hub or root surfaces of the rotor because of the difference between the diameter of these surfaces and their respective pitch circles. As stated above the sharp edges of the gate threads describe the curved sides of the rotor threads and conversely the edges of the crest of the rotor threads describe the curved sides of the troughs of the gate, producing the continuous sealing line which is later described with reference to Figs. 29 and 30. It is apparent that the thread structures must be generated in this manner to produce the continuous sealing line described hereinafter.

It will be noted that the diameter of the rotor member is the same in Figs. 12, 13 and 14 and the hub of the rotor in Fig. 14 is larger than in the other two views, but the space or void created between the threads is larger because of the shape of the sides 33' of the threads. Thus with this preferred form of thread the capacity of the pump is increased.

Again the void 94 formed between the gate and the rotor when they are in mesh, as shown, is smaller. These advantages are attributed to the fact that the diameter of the gate is smaller than the pitch circle of the gear. This is a very important discovery in this invention. It aids in providing a gate trough which is substantially circular in cross section and the bottom of the troughs do not have a hump, yet the extent of the perimetral surfaces 37 of the rotor threads is sufficient to provide an adequate seal. It is impossible to obtain these important advantages if the diameters of the rotary members are the same as the gear pitch circle.

As stated above, screw type pumps of this character inherently possess a leakage path from the discharge to the inlet end of the pump. When two or more threads are employed on each member, the volume of this leakage path is substantially as great as the pump displacement, thus rendering them impractical unless this leakage path is checked.

This leakage path may be readily traced on Figures 29 and 30 which illustrate an elongated gate and rotor members of the same character as that shown in Fig. 1. These screw members were made sufficiently long to clearly show that added length will not seal off the leakage path.

When the gate and rotor members are assembled in intermeshing relation, as shown in Fig. 28, their adjacent surfaces engage each other along a continuous line 88 which extends from one end of the members to the other. This continuous line of contact forms the sealing line between the two members and prevents any flow of fluid transversely between the members.

If the members were dipped in paint while intermeshed as shown in Fig. 28 and then separated, the heavy continuous sealing line 88 would appear along the full length of each member as shown in Figs. 29 and 30. Since the gate has a right hand thread and the rotor has a left hand thread the sealing line on these figures is correspondingly right and left handed, and if the drawing were folded to bring the figures face to face these sealing lines would match.

Assuming that the ends of the rotor and gate are fully exposed and the top of these figures represent the discharge end of the pump, fluid may flow from *a* through the gate trough between the threads 40 and 43 down in back of the screw to *b* where this trough again appears three-fourths of the way down Fig. 29. Further flow down this groove is checked by the seal line 88. However the fluid may transfer from the gate groove to B on the rotor in Fig. 30 which is on the corresponding side of the seal line. B is in the rotor groove above the thread 36.

The fluid may then flow up in back of the rotor screw to D where it appears on the opposite side of the seal line. The fluid then transfers from the rotor back to *d* between the threads 40 and 41 on the gate and on the corresponding side of the seal line. Thus the fluid upon making a complete circuit, down one gate trough across and up one of the rotor troughs and back to the gate, arrives in the next adjacent gate trough toward the inlet from which it started. The oftener the fluid circulates in this manner the nearer it approaches the inlet.

The fluid may then continue to flow down between the threads 40 and 41 from *d* to *e*, and if this trough is closed by the seal line, as illustrated in Fig. 29, it will make another circuit by transferring to E on the rotor and flow up to F and across to *f* on the gate and thence flow down between the gate threads 41 and 42 to *g* where it may pass directly to the inlet of the pump.

This leakage path has no restrictions whatsoever and it is quite evident that the volume per unit of time, flowing through this leakage path under given pressure and speed conditions, approaches the displacement output of the pump.

During the particular instant that the above leakage path has been traced the gate trough between the threads 42 and 43 is closed by the seal line below *h* in Fig. 29 and the corresponding trough H on the rotor is connected to the discharge at A in Fig. 30. This path is thus blocked.

Likewise the gate trough between the threads 41 and 42 is closed by the seal line below *j* and the corresponding trough J on the rotor is open to discharge.

The corresponding gate and rotor trough portions *k* and K represent a pocket that is running out at discharge.

Since the upper end of the gate and rotor is the discharge end of the pump the gate must rotate in a clockwise direction and the rotor in a counterclockwise direction because of the direction of their threads. If the gate and rotor in Figs. 29 and 30 were rotated the sealing line on each figure would appear to continuously travel up the screw members and vanish at the top. The same effect may be obtained by sliding a transverse straight edge down over the figures.

By closely observing the seal line 88 in Figs. 29 and 30 it will be noted that the vertical straight section at the top is made between the cylindrical edge of the gate thread 40 and the adjacent cylindrical trough of the rotor. The contacting surfaces being cylindrical this section of the seal line must be straight. The bottom of this straight section intersects the bottom edge of the gate thread 40 and from thereon the edge of this thread seals across the upper face of the rotor thread 35 to where the edges of both threads intersect, which occurs at the juncture 23

in the casing. The upper edge of the rotor thread **35** then seals across the left side of the gate trough to the gate trough center line indicated by the dotted line **89**. The seal then retraces the center line **89** until it intersects the lower edge of the rotor thread **35** which seals across the right side of the gate trough to the point where the edges of the gate thread **41** and rotor thread **35** intersect at **23** on the opposite side of the casing. The zigzag line just described represents the sealing line made by the rotor thread extending into and out of the gate trough.

From the intersection **23** the seal line continues down along the top of the gate thread **41** across the under side of the rotor thread **35** to the position where the cylindrical edge surface of the gate thread **41** intersects the cylindrical surface in the bottom of the trough in the rotor. This represents a complete sealing path from one rotor trough to another which is duplicated down the faces of the rotary members from one end to the other.

When a horizontal straight edge is moved down over Figs. 29 and 30 the sealing line **88** vanishes in the same manner that it would if the gate and rotor members were rotated. As the straight edge proceeds down Figs. 29 and 30 it breaks the seal line **88** and the upper part of the space *d* in the gate and D in the rotor are open to discharge. The break thus formed in the sealing line is the void **94** which produces a leakage path from the discharge to the inlet parallel to the path starting at *a* as described above.

The leakage path starting at *a* ceases the instant that the pocket formed by *k* and K completely runs out and the gate groove between the threads **40** and **41** becomes the sole leakage path until the next void **94** is formed by breaking the sealing line. Thus for the short space of time that the sealing line is broken the leakage path from *a* to *b* and B to D and back to *d* is short circuited and remains in this condition until the pocket formed by *k* and K runs out, at which time the space *d* becomes in effect the space *a* described above.

In order to stop this leakage path through the gate and rotor members, where the threads extend through one or more complete turns, it is necessary to close one end of the members with a sealing wall or land covering the overlapping areas of the gate and rotor and extending around the back thereof, as shown by the dotted line in Fig. 28. But for this area the ends of the rotor and gate are exposed and this exposed portion represents the discharge port.

If the rotor threads are less than one complete turn, as illustrated in Fig. 1, it is then necessary to employ a sealing wall or land at both ends of the rotary members, one at the inlet and one at the outlet. These sealing walls are diagonally disposed to one another and extend over the areas illustrated by the port lines **20** and **24** shown in Figs. 3 and 15 to 17. When the end sealing walls are disposed in this manner, the rotor and the gate may be made very short in length, which is an important advantage in this art. The rotor threads must extend at least substantially one-half turn in order that a complete seal line is formed over one rotor thread which is necessary to seal the members. The curve of the seal line **88** thus extends through one complete cycle.

When used as a blower or compressor the port edges can be disposed substantially at 30° to the plane intersecting the axes of the members as shown in Figs. 3, and 15 to 17 because the fluid is compressible. However when used as a liquid pump the port edges should be located as indicated in Fig. 28 so that there is no tendency to compress the liquid. The size and shape of the rotors in such a case would determine the exact position and contour of the port edges.

These rotary members may be made with little or no tolerance and then assembled in an open machine where they can be operated to lap in their engaging surfaces. Again if the device is very large, a clearance of one thirty second of an inch may produce a negligible leakage in comparison to the volume of fluid transmitted therethrough. Thus tolerances and clearances are, as a practical question, determined largely as to size and use to which the device is to be applied.

The void **94** which is produced between the trailing side of the rotor and the leading side of the gate threads after a thread of the rotor has entered a trough in the gate directly connects the discharge with the inlet when the rotor and the gate members are provided with threads that extend only 180° and 90° respectively. This void may be closed by the arrangement of the sealing end walls as explained above or by the use of sealing plates fastened on the ends of the rotary members.

The sealing end walls may be arranged as shown in dotted lines in Figs. 16 and 17, wherein a single port at one end of the rotary members is divided into two openings separated by the land **95**. The land is an integral part of the inner wall **14** and its edges represent the ultimate port opening whether the ports are fixed or variable as described above. It will be noted that the land **95** covers up the void **94** and thus prevents a leak back.

The sealing plates are shown in Figs. 18, 19 and 20 at **96** and **97**. The plate **96** is shown to have exactly the same profile as the rotor, and the plate **97** is shown to have the exact profile as the gate. These sealing plates are mounted on the ends of the rotary members in an offset position sufficient to cover the initial part of the void **94** formed by the rotary members.

The direction of the offset of these sealing plates is inconsequential within the limits of the thickness of the gate threads. It is preferable to extend the trailing edge of the rotor plate **96** over the void and thus overlap the gate thread to produce an effective seal. These sealing plates are fastened to the ends of the rotary members in any suitable manner, such as by screws. To avoid undue complication of the drawings these sealing plates have been illustrated only on the right of Fig. 2. By employing these sealing plates the ports may be enlarged to the extent that they produce a seal and the remote edges **74** of the port **98**, similar to the ports **20** and **24** in Figs. 1–5, may follow radially disposed arcuate lines having the same curvature as the trailing edge of the thread of the members associated therewith. Thus the port **98** leaves a goodly part of the ends of the rotary members exposed.

Another important advantage lies in the fact that the whole of the rotary members may be made up of a series of such plates with the teeth portions beveled to the degree of pitch of the thread desired, as indicated in Fig. 9, and then mounted on a shaft with each succeeding plate angularly disposed with respect to the plate adjacent thereto to produce a smooth continuous

thread. In this instance the end plates become the sealing plates and they are disposed at a greater angle relative to the next adjacent plate to cover the voids. However the edges of the sealing plates are preferably not beveled but normal to their faces.

Theoretically each pocket *k*, K completely runs out when the thread of the rotor is about to leave the trough of the gate, at which time they are completely covered by the sealing wall. As a practical proposition the pockets may be kept vented to the discharge until the rotor thread approaches the full depth of the gate trough, as illustrated in Fig. 22. This is accomplished by cutting away the leading side of the rotor threads along an angularly disposed plane intersecting the radial line **100** and bounded by the arcuate line **101** as shown in Figs. 21 to 23 and 27. Thus the fluid is permitted to escape from the pocket to the discharge up until these lines reach the sealing end wall. In Fig. 22 the port **98** is employed and is similar to the port shown in Figs. 3 and 18, the sealing plates being eliminated for the sake of simplicity. Immediately after the radial line **100** of the cut away portion passes behind the edge of the port **98** and thus seals the included pocket from discharge, the outer end of the cut away portion defined by the arcuate line **101** opens the pocket to the leading side of the thread and permits the fluid to escape. Thus as the pocket becomes smaller it is being continuously relieved until it completely runs out as the rotor thread passes from the gate trough.

If it is desired to continuously relieve the fluid trapped in the pockets to the discharge of the pump up until the time that they completely run out, a groove **102** may be cut in the end face of the rotor member as shown in Figs. 24 to 26. Just as the groove **102** is about to be opened to the space in front of the rotor thread, the other end of the groove is sealed under the port **98**. This represents the ultimate position of the pocket as it runs out.

A third method of avoiding the squeezing of the fluid in the pocket as the latter runs out may be obtained by changing the pitch of the threads of the rotary members so that the pocket is completely run out when the rotary members reach the edge of the port, in which case the whole of the pocket would be vented past the edge **100** or the groove **102** when the rotor and gate are about to pass under the sealing end wall.

Fig. 31 shows a seal line so arranged and balanced that in each pocket the same amount of area is exposed to pressure on one side of the center line as on the other side of the center line, as indicated by the equivalent areas **103** and **104**. Therefore, the pressure does not tend to resist rotation of the gate, nor does it tend to overrun the gate ahead of the rotor. This condition eliminates all gear loads, except the friction incidental to the bearings and seals. All of the pumping or compression load is transmitted through the rotor and the gate merely idles.

I claim:

1. In a fluid device of the character described, the combination of a casing having parallel cylindrical chambers which intersect to form a common chamber, a rotor member and a gate member rotatably disposed in said chamber, said members having complementary intermeshing helical threads and grooves of such shape as to form a continuous seal line therebetween which extends from one end to the other and is maintained as the members rotate, the threads of the rotor member extending through no more than one complete turn, both chambers being open at one end providing a fluid inlet, an end wall closing a portion of both chambers at the other end providing a fluid outlet therein, the cylindrical walls of the chambers and the threads and grooves of the members cooperating to form fluid pockets which advance toward the outlet end when the members rotate, said end wall covering substantially one side of the corresponding ends of said members and extending between and to the other side of said members to the intersection of said chambers to completely close each pocket until said pockets are closed at the other end by the cooperation of the threads and grooves of the members forming the sealing line across the common chamber at the intersection of the two chambers.

2. In a fluid device of the character described, the combination of a casing having parallel cylindrical chambers which intersect to form a common chamber, a rotor member and a gate member rotatably disposed in said chambers, said members having complementary intermeshing helical threads and grooves of such shape as to form a continuous seal line therebetween which extends from one end to the other and is maintained as the members rotate; the threads of the rotor member extending through approximately one half turn and the threads of the gate extending through approximately one quarter turn, the cylindrical walls of the chambers and the threads and grooves of the members cooperating to form fluid pockets which form at one end and advance toward the other end as the members rotate, walls closely adjacent the ends of said members partially closing both ends of said chambers and defining an opening in each wall on opposite sides of the plane passing through the axes of both members, said openings extending from the intersection of said chambers by an amount determined by the twist of the gate threads to avoid a leakage from one end to the other.

3. A pair of rotary elements for use in a fluid device of the character described, comprising a rotor member and a gate member rotatably disposed in parallelism and having complementary intermeshing helical threads and troughs, the crests of the threads of the members being cylindrical and bounded by extending helical edges, the rotor having two threads which extend through not more than one-half turn and the gate having four threads which extend through not more than one quarter turn, the sides of the rotor threads being described by the continuous edges of the helical threads of the gate, and the troughs of the gate being described by the edges of the crest of the helical threads of the rotor.

4. In a fluid device of the character described, the combination of a casing having a pair of parallel cylindrical chambers which intersect to form a common chamber, heads closing the ends of the common chamber, a rotor member and a gate member arranged to rotate in the cylindrical chambers with their ends closely adjacent said heads and having complementary intermeshing helical threads and troughs, a port in each head diagonally disposed with respect to each other and positioned on opposite sides of the plane intersecting the axes of said members, each port being open for substantially one-half of the areas of the ends of said cylindrical chambers, rotor and gate sealing plates secured on each end of said members and having the same cross sec-

tional contour as the threads of the respective members but angularly displaced relative thereto for closing the void formed between the trailing edges of the rotor threads and the leading edges of the gate troughs as they intermesh before passing beyond said ports.

5. In a fluid device of the character described, the combination of a casing having a pair of parallel cylindrical chambers which intersect to form a common chamber, heads closing the ends of the common chamber, a rotor member and a gate member arranged to rotate in the cylindrical chambers with their ends closely adjacent said heads and having complementary intermeshing helical threads and troughs which cooperate with the casing walls to form fluid-conveying pockets therebetween, a port in each of said heads, said ports being diagonally disposed with respect to each other and positioned on opposite sides of the plane intersecting the axes of said members, each port being open for substantially one-half of the area of the ends of said cylindrical chambers, rotor and gate sealing plates secured on each end of said members and having the same cross sectional contour as the threads of the respective members but angularly displaced relative thereto for closing the void formed between the trailing edges of the rotor threads and the leading edges of the gate troughs as they intermesh before passing beyond said ports, and slots extending diagonally across the ends of thread portions of the rotor plates from the crest of the leading side to a point adjacent the root of the thread but spaced from its trailing edge for venting the pockets to the ports up until the time each of said pockets runs completely out.

6. In a fluid device of the character described, the combination of a casing having parallel cylindrical chambers which intersect to form a common chamber, heads closing the ends of the common chamber, fluid connections in said heads opening into the cylindrical chambers, rotor and gate rotating members arranged to operate in the cylindrical chambers and having complementary intermeshing helical threads and troughs of such shape as to form a continuous sealing line therebetween and extending from one end to the other of said members and which coact with each other and the walls of said chambers to form pockets traveling from one end of the casing to the other, the extent and pitch of the intermeshing threads and troughs together with said heads being arranged to seal each pocket from communication with one of said connections before it opens to the other thereof, the axial pitch of the helical threads on said members becoming less as the threads approach the discharge end to compensate for the unbalancing torque characteristics created in the independent traveling pockets.

7. A pair of rotary elements for use in a fluid device of the character described, comprising a rotor member and a gate member rotatably disposed in parallelism and having complementary intermeshing helical threads and troughs, the crests of the threads of the members being cylindrical and bounded by extending helical edges, the sides of the rotor threads being described by the continuous edges of the helical threads of the gate, and the troughs of the gate being described by the edges of the crest of the helical threads of the rotor, the overall diameter of the gate member being smaller than the diameter of the pitch circle thereof and the radius of the troughs of the rotor member being larger than the radius of the pitch circle thereof.

8. A pair of rotary elements for use in a fluid device of the character described comprising a rotor member and a gate member rotatably disposed in parallelism and having complementary intermeshing helical threads and troughs, the overall diameter of the gate member being smaller than the diameter of the pitch circle thereof, and the radius of the troughs of the rotor member being larger than the radius of the pitch circle thereof.

9. A pair of rotary elements for use in a fluid device of the character described comprising a rotor member and a gate member rotatably disposed in parallelism and having complementary intermeshing helical threads and troughs of different pitch, the overall diameter of the gate member being smaller than the diameter of the pitch circle thereof, and the radius of the troughs of the rotor member being larger than the radius of the pitch circle thereof.

10. A pair of rotary elements for use in a fluid device of the character described comprising a rotor member and a gate member disposed in parallelism and having complementary intermeshing helical threads and troughs, the ratio of the threads on said members being one to two respectively, the overall diameter of the gate member being smaller than the diameter of the pitch circle thereof, and the radius of the troughs of the rotor member being larger than the radius of the pitch circle thereof.

11. A pair of rotary elements for use in a fluid device of the character described comprising a rotor member and a gate member rotatably disposed in parallelism and having complementary intermeshing helical threads and troughs, the ratio of the threads on said members being one to two respectively, the overall diameter of the gate member being smaller than the diameter of the pitch circle thereof, the radius of the troughs of the rotor member being larger than the radius of the pitch circle thereof, and the overall diameter of the rotor member being greater than the overall diameter of the gate.

12. A pair of rotary elements for use in a fluid device of the character described, said members being disposed in parallelism and having complementary intermeshing helical threads and grooves, the threads and grooves of each member twisting in the same general direction about the axis of said member, the axial pitch of said complementary threads of each member adjacent the ends thereof being less than the pitch of the intermediate portion of said threads.

13. A pair of rotary elements for use in a fluid device of the character described, said members being disposed in parallelism and having complementary intermeshing helical threads and grooves, the threads and grooves of each member twisting in the same general direction about the axis of said member, the axial pitch of said complementary threads of each member changing from a small pitch to a large pitch and back to a small pitch from one end of each rotary member to the other end thereof.

14. A pair of rotary elements, for use in a fluid device of the character described, comprising a rotor member and a gate member rotatably disposed in parallelism and having complementary intermeshing helical threads and troughs, a plate secured to an end of each member having the same contour as the threads of said member but angularly displaced thereto for closing the void formed between the trailing edge of the rotor and

the leading edge of the gate threads as they intermesh to substantially their full depth.

15. A pair of rotary elements, for use in a fluid device of the character described, comprising a rotor member and a gate member rotatably disposed in parallelism and having complementary intermeshing helical threads and troughs, a plate secured to an end of each member, said rotor plate having the same contour as the trailing edge of the rotor threads, said gate plate having the same contour as the leading edges of the gate threads, said plates being angularly displaced relative to their respective rotor or gate threads to overlie and close the voids formed between the rotor thread and the gate trough as the threads intermesh to substantially their full depth.

16. A pair of rotary elements, for use in a fluid device of the character described, comprising a rotor and a gate member rotatably disposed in parallelism and having complementary intermeshing helical threads and troughs, the crest of the threads of the members being cylindrical and bounded by extending helical edges, a portion of the end of the rotor threads, bounded by the leading side from the edge of the crest, and from a point where the edge of the crest of the gate thread intersects the leading side of the rotor thread when the rotor thread is in the fully meshed position, to a point adjacent the root of the trailing side of the rotor thread, being cut away for venting the space between the leading side of the rotor threads and the gate to one side of the rotary members as the threads approach their full meshing position and thereafter venting said space to the other side of said rotary members.

17. A pair of rotary elements, for use in a fluid device of the character described, comprising a rotor member and a gate member rotatably disposed in parallelism and having complementary intermeshing helical threads and troughs, the crest of the threads of the members being cylindrical and bounded by extending helical edges, a slot extending diagonally across the end face of each rotor thread from the edge of the crest on the leading side to a point adjacent the root of the thread but spaced from its trailing side for venting the space between the leading side of the rotor threads and the gate until the time this space runs completely out.

18. In a fluid device of the character described, the combination of a casing having a pair of parallel cylindrical chambers which intersect to form a common chamber, a rotor member and a gate member arranged to rotate in the cylindrical chambers and having complementary intermeshing helical threads and grooves which cooperate with the casing walls to form fluid-conveying pockets therebetween, heads closing the ends of the common chamber, fluid connections in said heads opening into the cylindrical chambers, arcuate plates rotatable relative to the axes of said members and immediately interposed between the ends of said members and the fluid connection in one of said heads for varying the time at which the pockets become connected to the opening of the fluid connection in said head, and means responsive to the discharge pressure for actuating said arcuate plates.

19. In a fluid device of the character described, the combination of a casing having a pair of parallel cylindrical chambers which intersect to form a common chamber, heads closing the ends of the common chamber, fluid connections in said heads opening into the cylindrical chambers, a rotor member and a gate member rotatably disposed in the cylindrical chambers and having complementary intermeshing helical threads and grooves which cooperate with the casing walls to form fluid-conveying pockets therebetween, shafts supporting said members, bores in said heads axially aligned with said cylindrical chambers, sleeves rotatably mounted in the bores of one head, bearing assemblies mounted on each shaft adjacent the ends of said members and arranged to be inserted into the bores in one head and the sleeves in the other head for rotatably supporting said members, an arcuate plate connected to the inner end of each sleeve and lying between the fluid connection in said one head and the end of its respective member for varying the time at which the pockets become connected to the opening of the fluid connection, intermeshing gear segments fixed to the other ends of said sleeves for moving the arcuate plates in unison, and means responsive to the discharge pressure for actuating said gear segments.

20. In a fluid device of the character described, the combination of a casing having a pair of parallel cylindrical chambers which intersect to form a common chamber, a rotor member and a gate member rotatably disposed in said chambers and having complementary intermeshing helical threads and troughs of such shape as to form a continuous sealing line therebetween and extending from one end to the other of said members and which coact with each other and the walls of said chambers to form and advance pockets along the grooves of the threads as the members rotate, and walls at both ends of the common chamber closely adjacent the ends of said members covering portions of said ends when interengaged and defining inlet and outlet fluid connections to the chambers, the extent and pitch of the intermeshing threads and grooves together with said last named walls being arranged to seal each pocket from communication with the inlet before it opens to the outlet and to maintain said seal until and while such outlet communication takes place.

21. A pair of rotary elements for use in a fluid device of the character described, comprising a rotor member and a gate member rotatably disposed in parallelism and having complementary intermeshing helical threads and troughs, timing gears operably connecting said members, the crests of the threads of the members being cylindrical and bounded by extending helical edges, the sides of the rotor threads being described by the respective edges of the helical threads of the gate and the sides of the troughs of the gate being described by the respective edges of the crest of the helical threads of the rotor, said threads being so shaped and dimensioned to produce a continuous sealing line between the members that is symmetrical on both sides of the plane which includes the axes of the members, said sealing line following the lower edge of the gate crest along the coacting side of the rotor thread from said plane to the point of intersection of the crest edges of the rotor and gate members at one side of the members at which point the sealing line transfers to the upper edge of the crest of the succeeding rotor thread by extending back up through the gate trough to the center line of the bottom thereof and thence down along the center line of the gate trough across the rotor crest to a point in the same

transverse plane as said point of transfer, and back up along the lower edge of the crest of the rotor thread to the point at which the crest edges of the members intersect at the other side of the members and in the same transverse plane as the upper end of the sealing line along the bottom of the gate trough, said sealing line extending thence downwardly along the upper edge of the next adjacent gate crest to said first mentioned plane and thence across the face of the gate crest to the lower edge thereof, thus completing a sealing cycle, whereby the extent of the sealing line along the bottom center line of the gate trough is such as to provide equal areas on each side of said center line exposed to the same fluid pressure thereby producing a balanced load condition on the gate member and reducing the timing gear load.

22. A pair of rotary elements for use in a fluid device of the character described, comprising a rotor member and a gate member rotatably disposed in parallelism and having complementary intermeshing helical threads and troughs, the crests of the threads of the members being cylindrical and bounded by extending helical edges, the rotor having two threads which extend through less than one complete turn and the gate having four threads which extend through less than one half turn, the sides of the rotor threads being described by the continuous edges of the helical threads of the gate, and the troughs of the gate being described by the edges of the crest of the helical threads of the rotor.

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