May 28, 1946. F. A. QUIROZ 2,401,189

ROTARY PUMP CONSTRUCTION

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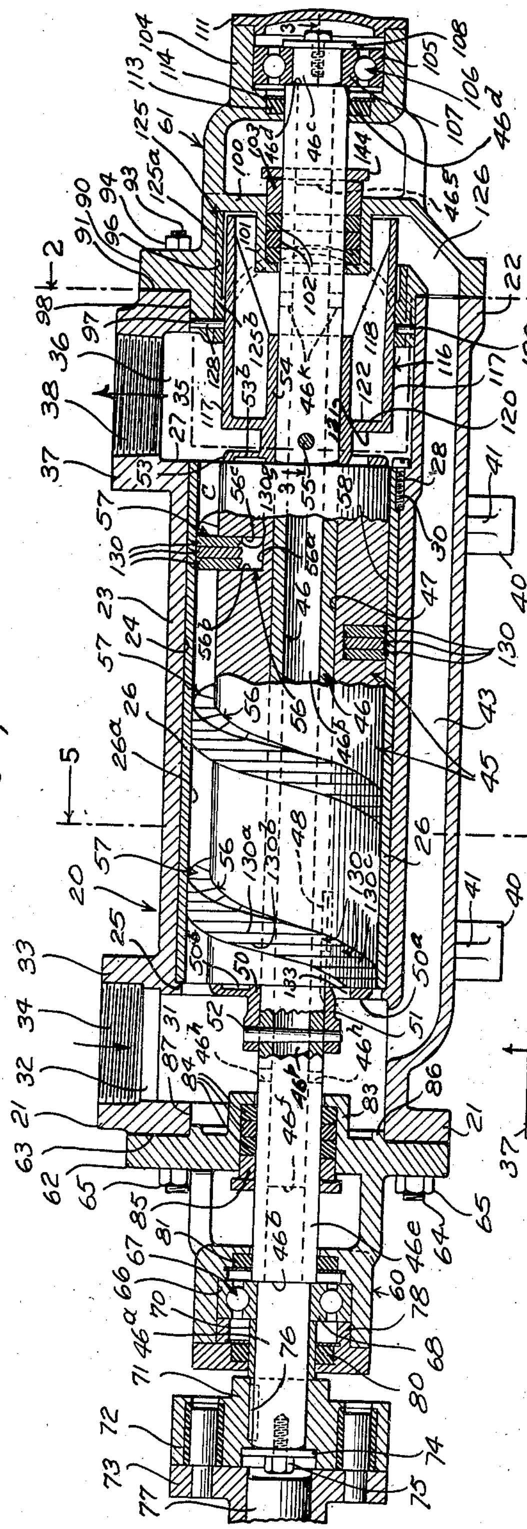


Fig. 1.

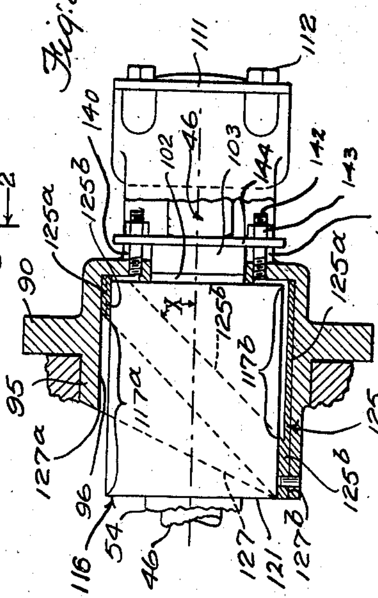


Fig. 3.

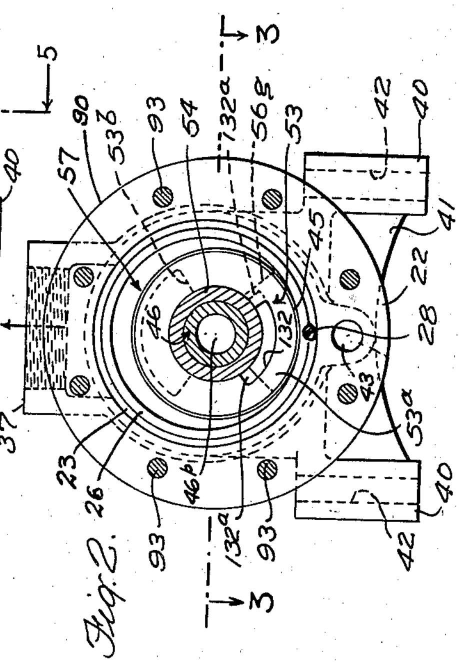


Fig. 2.

INVENTOR
FRANCISCO A. QUIROZ.
BY
William T. Kuiesner
ATTORNEY

Fig. 4.

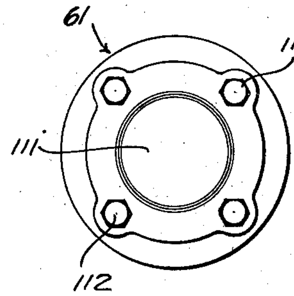


Fig. 5.

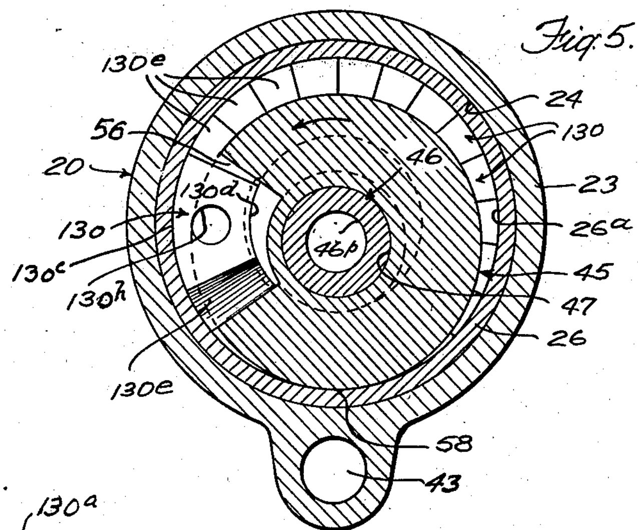


Fig. 8.

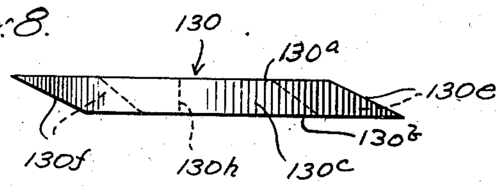
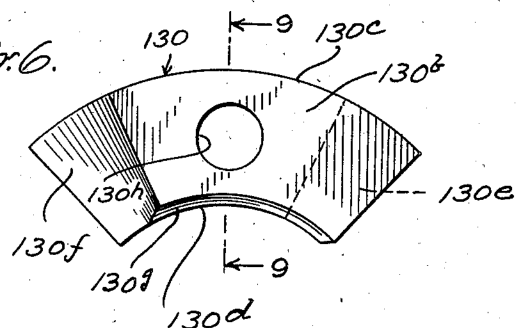


Fig. 6.



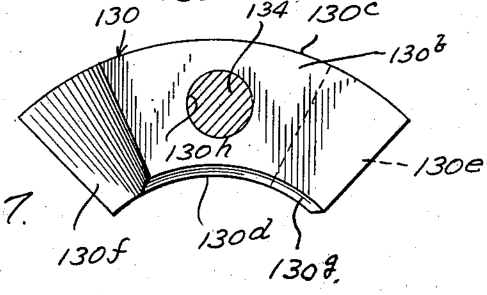


Fig. 7.

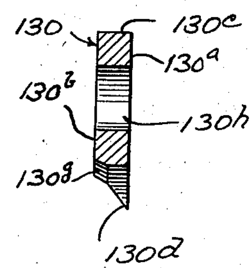


Fig. 9.

INVENTOR
FRANCISCO A. QUIROZ
BY
William T Kuesner
ATTORNEY

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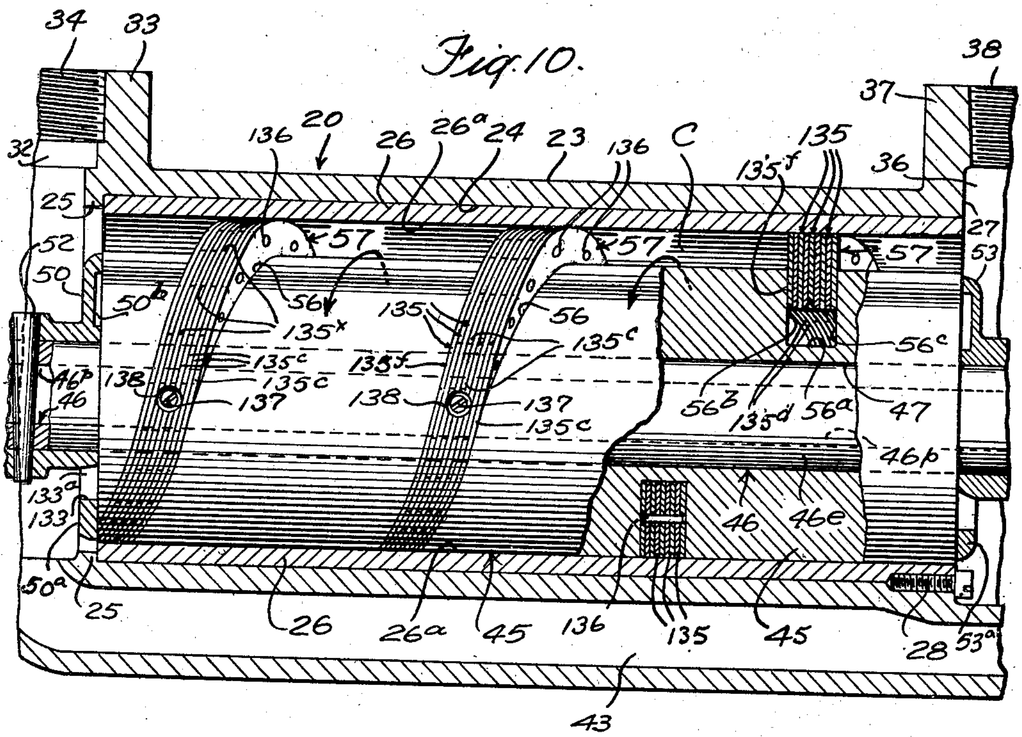
Fig. 10.



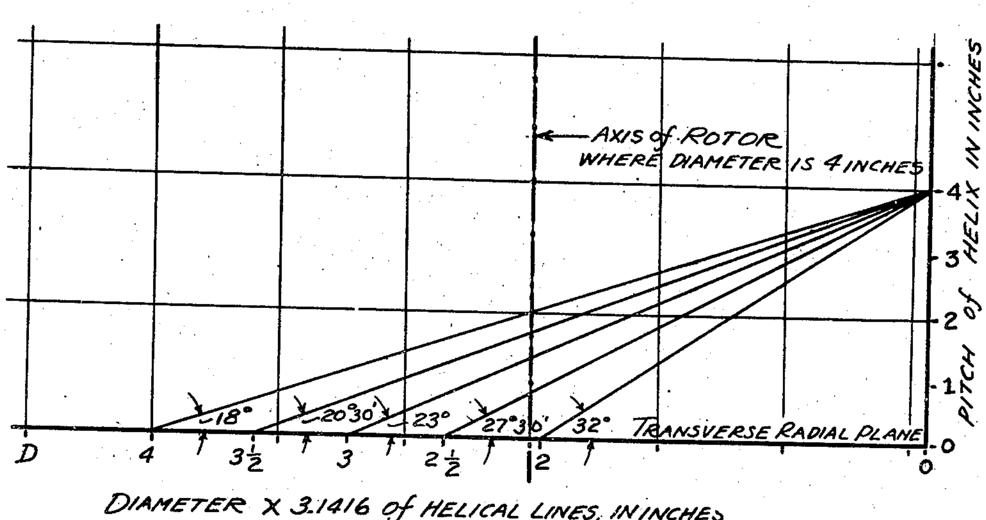


Fig. 11.

INVENTOR
FRANCISCO A. QUIROZ.
BY William T. Kiesner
ATTORNEY

Patented May 28, 1946

2,401,189

UNITED STATES PATENT OFFICE

2,401,189

ROTARY PUMP CONSTRUCTION

Francisco A. Quiroz, Newark, N. J.

Application May 12, 1944, Serial No. 535,283

22 Claims. (Cl. 103—117)

This invention relates to rotary pumps, more particularly to so-called screw or helical types of pumps.

An object of this invention is to provide an improved screw type of pump construction that will be inexpensive to manufacture and assemble and that will be of superior action and efficiency in practical use. Another object is to provide a screw type of pump so constructed that a wide variety of different materials or combinations of materials may be employed in making up its various parts and thereby readily accommodate the pump to any of the widely varying characteristics or requirements of different liquids to be handled. Another object is to provide a pump construction of the last-mentioned character in which various of its parts are so constructed as to be easily replaced either by parts of the same material or by parts made of some different material, according to practical requirements met with. Another object is to provide a pump construction in which its individual parts may be inexpensively fabricated and easily assembled or replaced.

Another object is to provide an improved screw or helical pump construction that will facilitate the use of, and achieve numerous advantages resulting from the employment of, relatively rigid though displaceable or movable helical or screw elements, but in a manner to maintain effective sealing throughout their range of relative displacement or movement in relation to their mounting or support and thereby achieve efficiency of performance in the pumping of liquids or the like. Another object is to carry out this last-mentioned object in a manner that will facilitate manufacture and assembly and achieve also dependability of action and operation throughout long continued use. Another object is to provide a pump construction of the above-mentioned character in which also multiple pumping action may be dependably achieved and the capacity and efficiency of the pump improved for each helix or screw element employed.

Another object is to provide a screw or helical pump construction that will have improved structural and operating characteristics. Another object is to provide a screw or helical pump construction in which inherent uni-directional end or axial thrusts will be overcome or opposed in a simple, practical and inexpensive manner. Another object is to provide a pump construction, in which the rotary impeller is eccentrically mounted, with simple, practical and inexpensive means for opposing or counteracting generally uni-directional lateral thrusts imposed upon the rotary impeller. Another object is to provide, in pumps of the above-mentioned character, a simple, inexpensive and dependable means for opposing or counteracting end or side thrusts as the case may be, and capable, where necessary, of variability of action where the thrusts to be opposed or counteracted vary within a cycle of operation of the pump.

Another object is in general to improve upon the construction, operation and efficiency of pumps of the rotary screw or helical type. Other objects will be in part obvious or in part pointed out hereinafter.

The invention accordingly consists in the features of construction, combinations of elements, and arrangements of parts as will be exemplified in the structure to be hereinafter described and the scope of the application of which will be indicated in the following claims.

In the accompanying drawings in which are shown by way of illustration several of the possible embodiments of my invention,

Figure 1 is a central vertical sectional view through the pump casing and its internal parts, showing in elevation and in fragmentary longitudinal section one form of helical impeller construction;

Figure 2 is a transverse vertical sectional view as seen along the line 2—2 of Figure 1;

Figure 3 is a fragmentary horizontal sectional view, as seen along the line 3—3 of Figure 1 and the line 3—3 of Figure 2, certain parts being shown in elevation and other parts being broken away or omitted;

Figure 4 is a fragmentary end elevation as seen from the right in Figures 1 and 3;

Figure 5 is a transverse vertical sectional view as seen along the line 5—5 of Figure 1;

Figures 6 and 7 are elevations of, respectively, two forms of elements that make up the helical impeller of Figures 1 and 5;

Figure 8 is an elevation as seen from the top in Figures 6 and 7;

Figure 9 is a transverse sectional view as seen along the line 9—9 of each of Figures 6 and 7;

Figure 10 is a view like that of Figure 1, with certain parts broken away, showing a pump construction embodying a modified form of built-up impeller element, the latter being shown in elevation and also in fragmentary section; and

Figure 11 is a development showing changes in angularity with change in radial distance from the axis, of the side walls of a helical groove.

Similar reference characters refer to similar parts throughout the several views of the drawings.

Referring first to Figure 1, I provide a casing 20 generally cylindrical, preferably in the form of a casting terminating at its left-hand end in a relatively heavy annular flange 21 and at its right-hand end in a relatively heavy annular flange 22, the two flanges being coaxial but displaced relative to the axis of the casing 20 so as to be eccentric relative thereto.

The casing 20 comprises what will hereinafter be termed the cylinder 23, the latter being bored out as at 24 preferably from the right-hand end so as to leave at the left-hand end thereof an annular shoulder 25 against which abuts the end of a preferably heavy-walled cylinder lining or liner 26 which at its right-hand end terminates substantially flush with a machined annular face 27 formed internally of the casing 20 at the end of the cylinder 23. The liner 26 is snugly fitted into the cylinder 23, being passed into it through the opening in the annular end flange 22 of casing 20 and it is anchored in position and against displacement or rotary movement by any suitable means, such as, for example, one or more screws 28 which preferably functions both as a clamping means and as a key. For this latter dual purpose, a hole 30 may be drilled parallel to the axis of the cylinder and liner, but so as to intersect the junction between the two, the hole is threaded, whence the screw 28 is threaded inward to hold the liner 26 clamped against the cylinder end shoulder 25 and also to hold it against rotary movement relative to the cylinder.

The casing 20, intermediate of the left-hand flange 21 and the left-hand end of the cylinder 23 is shaped to provide a chamber 31 of substantial size, hereinafter termed the inlet chamber, communicating with the left-hand end of the interior of the pump cylinder 23—26, and suitable means are provided for making a pipe connection to this inlet chamber 31. Thus in a wall of the latter is cast a transversely extending hole or passage 32 terminating in an annulus 33 which is internally threaded as at 34 to receive the threaded end of a connecting pipe.

Intermediate the right-hand end of the cylinder 23 and the end annular flange 22 of the casing 20, the latter is shaped to provide a substantial-sized space 35 communicating with the discharge end of the cylinder 23—26, and chamber 35 will hereafter be called the outlet or balancing chamber; like the inlet chamber 31 it is provided with a radially extending hole or passage 36 extending through its wall, surrounded by or terminating in a heavy annulus 37 internally threaded as at 38 so that pipe connection may be made to the discharge or high pressure side of the pump.

The casing 20 may be constructed, in any suitable way, as by casting integrally therewith, suitable standards or legs, indicated at 40, and suitably distributed, so that it may be secured to a suitable base or frame; thus the standards 40 may be in pairs, one pair near the inlet and the other pair near the outlet end of the casing 20, the standards of any one pair being preferably connected and reinforced by a suitable cross web 41 as is better shown in Figure 2. The standards 40 contain suitable holes 42 for the reception of bolts or the like.

In making up the casing 20, as by casting it, I also prefer to embody into it, as by coring, a passage or channel 43, formed in a wall of the casing 20, conveniently along the bottom portion thereof (see Figure 2), the channel 43 terminating at its left-hand end in Figure 1 into the inlet chamber 31, preferably through a side wall thereof, and terminating at its right-hand end in the end flange 22 by which it is ultimately, as later described, to be placed in connection with certain elements in the outlet or balancing chamber 35.

Within the cylinder 23—26 is operative a rotor 45 that is of a radius less than the radius of the chamber formed by the cylinder 23—26 and which in length matches the length of the cylinder lining 26; the rotor 45 is secured to a shaft 46 which extends through and snugly fits within a suitable bore 47 within the rotor 45, and rotor and shaft are suitably keyed together as by the key 48 so that drive of the shaft will effect dependable drive of the rotor 45.

Rotor 45 is held against axial movement relative to the shaft 46 by a rotor head 50 that abuts against the left-hand end of rotor 45 and whose hub 51 is received on the shaft 46 to which it is pinned as by the taper pin 52, and by a rotor head part 53 that abuts against the right-hand end of the rotor 45, the part 53 being integrally formed with a hub 54 which carries certain other parts later described and which is secured to the shaft as by a taper pin 55.

In the solid-walled rotor 45 is formed as by milling, a helical groove or slot 56, later described in detail, and in the helical slot 56, which is preferably of substantial cross-sectional dimension, is mounted a helical impeller generally indicated by the reference character 57, and preferably built up of component elements in the manner and for the purposes and coactions later described; the elements comprising the impeller 57 extend throughout the entire length of the helical groove 56, from one rotor head 50 to the other rotor head 53 with which the impeller elements and also the helical groove 56 coact as later described.

The rotor assembly comprising the shaft 46 and rotor 45 is supported by bearing structures with the axis of the assembly sufficiently displaced from the axis of the interior cylindrical surface of the cylinder liner 26 so that the latter and the rotor 45 are tangent to each other along a line of tangency indicated in Figures 1 and 5 by the reference character 58. It is along this line of tangency that the two relatively rotatably and tangentially arranged parts form a seal that is continuous, along the straight line of tangency, from one end of the rotor or cylinder wall to the other. The elements that make up the helical impeller 57, however, have their outer faces of the same radius as the radius of the cylinder or lining wall which they continuously contact during rotation of the rotor to maintain throughout the extent of the helical impeller 57 a sealed contact with the cylinder wall, being constructed, as is later described, in relation to the side walls of the helical slot 56 to maintain a leak-proof seal between a wall or walls of the slot and themselves regardless of the extent of their displacement radially outwardly of the slot and regardless of certain peculiar variations in angularities in the side walls of the helical slot, all as is later described in detail.

The above-mentioned bearing structures for supporting the shaft 46 to maintain the above-described relationships of rotor and impeller to the cylinder wall during operation comprise two bearing and packing gland housings 60 and 61 assembled respectively to the annular end casing flanges 21 and 22.

Thus the housing 60 comprises a securing flange 62 one face of which is shaped to mate with the face of the casing flange 21, a suitable gasket 63 being interposed therebetween, flange 21 having threaded therein a suitable number of suitably distributed studs 64 of a length to extend through correspondingly positioned holes in the flange 62 and being threaded to receive nuts 65 whereby to achieve secure assembly of the housing 60 to the casing 20 and leak-proof compression of the gasket 63.

Housing 60, at its outer end, is bored out to receive the outer race 66 of an anti-friction bearing 67 whose inner race 68 is received over a reduced end portion 46^a of the shaft and abuts against a shoulder 46^b against which it is clamped by a sleeve 70 and the hub 71 of one member 72 of a flexible coupling 72—73, the clamping pressure being exerted by a washer 74 bearing against an internal shoulder of the hub 71 and pressed thereagainst by a cap screw 75 threaded coaxially in the left-hand end of the reduced portion 46^a of the shaft. The hub 71 is splined or keyed to the shaft portion 46^a, as indicated at 76, to transmit driving torque to the shaft, the driving power being applied by any suitable means to the coupling member 73 as by a shaft 77 which usually is the shaft of, for example, an electric motor, the flexible coupling avoiding need of precision of alignment of motor bearings and pump bearings.

The outer race 76 may be secured in its supporting counter-bore by any suitable means which may comprise also a ring member 78 fitted into the counter-bore and bearing against the outer race 66 and secured in position in any suitable manner as by screws (not shown) and the ring member may be suitably conformed as indicated to carry internally and about the sleeve 70 a packing or ring 80 of felt or the like to prevent ingress of foreign matter and to guard against leakage of lubricant from the bearing assembly; a similar felt ring 81, suitably housed in an appropriate counter-bore in the casing 60 can similarly close off the other end of the bearing assembly.

The inward extension of the mounting flange 62 is shaped and machined to provide a cylindrical housing 83 coaxial with the mounting flange 62 to form part of a stuffing box or packing gland construction which may comprise suitable gland packing 84 that extends about the shaft 46 and is compacted against an end flange of the housing 83 by a packing gland 85 adjustably pressed into the housing 83 by means later described.

Coaxial with the bearing and stuffing box structures is an annular rib 86 formed on the inside face of the flange 62 to be received within the bore 87 of the casing flange 21 to be thereby centered with respect to the bore 87; the axis of bore 87 is displaced in the same direction and to the same extent from the axis of the wall surface of the cylinder 23—26 as is required to achieve tangency between the rotor 45 and the cylinder wall surface, and hence assembly of the bearing and stuffing box housing 60 by its mounting flange 62 to the casing flange 21 achieves the desired eccentricity of support of the left-hand end of the shaft 46 and hence of the left-hand end of the rotor 45. The mounting flange 62, stuffing box and shaft 46 close off the bore 87 in the casing flange 21, thus closing off the inlet chamber 31 at the left-hand end thereof as seen in Figure 1.

Liquid supplied to the inlet chamber 31 through the pipe connection 33—34 upon rotation of the rotor assembly in the direction of the arrow in Figure 1, effects movement of the liquid toward the right in the space between the cylinder wall and the eccentric rotor 45 in a manner later described in detail, injecting it into the outlet chamber 35. In the process, an end or axial thrust is imposed upon the rotor and shaft assembly, tending to force the latter in axial direction toward the left as viewed in Figure 1, and there is also imposed a thrust upon the rotor 58 and its shaft 46 in a direction transverse to their coincident axes, tending to flex the shaft and thereby shorten the life of the bearings and cause excessive or detrimental friction and rubbing between the rotor and the casing, this radial thrust being exerted mainly at the discharge or right-hand end of the rotor 45 for a substantial distance inwardly of the rotor by a distance on the order of the pitch of the helical impeller 57 and in a direction displaced about 90°, in leading direction relative to the direction of rotation, from the line 58 of theoretical tangency between rotor and cylinder wall. These detrimental actions, dependent upon the above-mentioned leak-proof relationship between the elements of the helical impeller 57 and rotor and cylinder wall, all, as later described in detail, I dependably overcome by providing, in the other end casing structure 61, means for opposing or counterbalancing both the axial thrust and the radial thrust, and for that reason the other end casing 61 and the outlet or balancing chamber 35 and certain structural features related to the shaft and rotor at the discharge end are different from the structural features above described in connection with the left-hand bearing housing 60.

The housing 61 is constructed to coact to form part of the outlet chamber 35 and to control or determine the application of the relatively high discharge pressure of the pump to coact in opposing the above-mentioned thrusts; conveniently, it has a relatively large and heavy annular flange 90, comparable to the flange 62 of the left-hand housing 60, adapted to abut against the casing flange 22, with a gasket 91 therebetween. Studs 93, suitable in number and suitably distributed, are mounted in and project from the casing flange 22, and with the flange 90 provided with similarly positioned holes to take over the studs which are threaded at their outer ends, the casing 61 may be detachably secured to the casing flange 22 by the application of nuts 94 to the threaded studs 93, thus also pressing the gasket 91 to achieve fluid-tight connection.

The mounting flange 90 terminates inwardly in a cylindrical axially extending part 95 which is bored out as at 96 to provide an internal cylindrical wall that is coaxial with the axis of the bearing housing 60 at the other end of the casing 20, and for purposes of assembly and of achieving such coaxial relationship, the portion 95 is turned as at 97 to provide an external cylindrical surface that is eccentric or radially displaced from the axis of the bore 96 by the same amount that the axis of the rotor is displaced from the axis of the cylinder 23—26, the eccentric surface 96 being snugly received within the cylindrical bore 98 of the casing flange 22, the bore 98 being machined concentrically or coaxially with the bore in the casing cylinder 23 itself.

The cylindrical portion 95 in effect forms an endwise extension of the outlet or balancing chamber 35 being closed off by an annular wall

100 which is constructed to provide a cylindrical housing 101 to receive packing material 102 of a stuffing box, being placed under pressure to effect sealing relationship with the rotating shaft 46 by means of the sleeve or gland element 103 which is provided with suitable means for adjustably placing the packing material 102 under compression and hence under radial expansion.

From the end closing wall 100 of the housing 61, the latter is extended to the right in Figure 1 to terminate in a cylindrical portion 104 that is internally bored out to receive the outer race 105 of an anti-friction bearing 106 whose inner race 107 fits onto a reduced end portion 46^c of the shaft 46 and is clamped against the resultant shoulder by a washer or disk 108 held in clamping relationship by a cap screw 110 threaded into the end of the shaft 46.

The end cap or cover 111 closes off the outer end of the cylindrical bearing support 104, bearing detachably secured in position as by studs and nuts 112 (Figures 3 and 4), thus sealing off the outer end of the bearing mounting against loss of lubricant or access thereto of foreign material. A suitable felt or like ring 113 held in a suitable counterbore by a retainer ring 114 seals off the other end of the bearing mounting against loss of lubricant or entry of foreign matter.

Due to the coaxial relationship between the cylinder bore 96, the stuffing box construction, and the bearing mounting, of the housing 61, in relation to which coaxiality the part 97 is turned to the desired eccentricity as above mentioned, the assembly of the housing 61, as above described, to the end casing flange 22 brings all of the coaxial parts into coaxial relationship with the bearing and stuffing box axis in the other end housing 60 and the rotor 45 thus becomes assembled to the cylinder with their axes parallel but displaced and with the surfaces just about tangent along the line 58 as above described.

Within the endwise extended outlet chamber 35 and mounted to rotate with the rotor, I provide a balancing rotor generally indicated by the reference character 116 and conveniently I form it integrally with the rotor and head 53 and its hub 54. Rotor 116 is of an external diameter on the order of the diameter of cylinder wall 26^a; thus it may be equal to the external diameter of the rotor 45 where the difference in diameters of cylinder wall 26^a and rotor 45 is not too great, and its external cylindrical face 117 is coaxial with the axis of the shaft 46 and of the rotor 45; this cylindrical face 117 coacts as later described to effect the above-mentioned radial thrusts and it is preferably constructed to be hollow, strong and rigid, having for example a plurality of ribs 118 extending radially from the internal face of the rotor 116 to the hub 54 to give the right-hand or outer portions of the rotor 116 cantilever-like internal support. This structural relationship aids in making it possible to have the stuffing box housing 101 extend axially inwardly within the right-hand open end of the balancing rotor 116, thus permitting shortening up of the axial distance between the end bearings supporting the shaft 46.

The hollow rotor 116 is otherwise joined to the hub 54 by an annular wall 120 which is spaced a substantial distance axially from the end face of the rotor 45, and it presents toward the rotor end face an annular end face 121, and it also presents internally of the cylinder 116 an annular face 122, both of which take part in offsetting axial thrust caused by the operation of the helical pump elements.

The end face 121 is subjected to the pump discharge pressure, being the pressure existent in the outlet chamber 35, and the pressure of liquid in the latter is thus exerted against the face 121, tending to thrust the rotating structure toward the right in Figure 1, thus to oppose the reaction of the helical impeller 57 and rotor 45 in exerting a thrust toward the left, provided that the outlet pressure is not made effective upon the opposed face 122 of the annular wall 120.

In part to realize this last-mentioned condition, I provide a seal between the external cylindrical face 117 of the balancing rotor 116 and the interior of the cylinder portion 95 of the end casing 61, and this seal I achieve by what I shall term a "sealing ring," generally indicated in Figure 1 by the reference character 125 secured to the face 96 of the cylinder 95 and relative to which the balancing rotor 116 rotates. Since the sealing ring 125 coacts with the rotor 116 for purposes later described, its construction and mounting are described later in greater detail.

The sealing ring 125 thus seals off the interior of the hollow balancing rotor 116 and hence also the internal annular face 122, from the high or outlet pressure in the chamber 35, and by way of a channel or passage 126 formed in the end casing 61 and mating with the channel 43 extending lengthwise of the pump casing 20 itself, the interior of the hollow balancing rotor 116 and the annular face 122 are connected to the inlet chamber 31 of the pump so that the inlet pressure is made effective upon such surfaces of the balancing rotor 116 as are thus placed in communication with the inlet chamber.

Axial thrust toward the left, caused during the operation of the rotor and helical impeller 57, is a function of the difference between discharge pressure and inlet pressure, and the actual thrust in pounds is the product of that difference in discharge and inlet pressures (in pounds per square inch) multiplied by the area (in square inches) throughout which that difference is effective, and that area, in general, will be the difference between the cross-sectional area of the bore of the cylinder 23—26 less the cross-sectional area of the hub 54 to the left of the annular face 121 of the balancing rotor 116. This axial thrust is opposed by the difference between the force exerted by the discharge pressure in the balancing chamber 35 against the external annular end face 121 of the balancing rotor 116 and the force exerted by the intake pressure, communicated to the balancing rotor 116 by the passages 43 and 126, against the effective surface areas of the balancing rotor 116 exposed thereto, and they comprise principally the internal annular face 122. As a result, the balancing rotor 116 exerts an axial thrust toward the right which, according to dimensional factors that are selected in relation to the difference between the discharge pressure and the inlet pressure for which the pump is designed may in whole or in part balance or oppose the axial thrust toward the left caused by the operation of the pump rotor 45 and impeller element 57.

The anti-friction bearings 67 and 106 are preferably of the combined radial and axial thrust type so that they can take up any resultant axial thrust that may exist or may be caused during the operation of the apparatus.

Considering now how the balancing rotor 116 compensates for radial thrust, it is first to be noted that the action of the rotor 45 and impeller 57 is such that this radial thrust upon the rotor is exerted mainly in the axial region of the rotor near its right-hand or discharge end as viewed in Figure 1, the principal radial effect adjacent the discharge end being throughout an axial distance at least equal to the pitch of the helical impeller 57 and its major effect, instead of being directly downwardly as seen in Figure 1, is in a horizontal direction, namely, at right angles to and toward the plane of Figure 1, or in a direction toward the right in Figure 5 from the 9 o'clock point of the cross-section. Accordingly, I provide, by appropriate construction, preferably by suitably shaping the sealing ring 125, for exposing a greater axial length of surface of the balancing rotor 116 to the outlet pressure always at the 3 o'clock point which is diametrically opposed to the 9 o'clock point in the cross-section of the pump cylinder 23—26 and from which 9 o'clock point the major effect of the radial thrust is exerted radially inwardly upon the rotor.

The sealing ring 125 comprises, in general, a truncated cylinder liner 125^a whose external diameter matches the internal diameter of the bore 96 of the cylindrical portion 95 of the end casing 61, the portion 95 being, as is better shown in Figure 3, and indicated by the broken line 127, also shaped in the form of a truncated cylinder, having its minimum axial dimension at the above-mentioned 3 o'clock position and its maximum axial dimension at the above-mentioned 9 o'clock point in the vertical cross-section, these two dimensions being better shown in Figure 3 by the lines 127^a and 127^b.

The external diameter of balancing rotor 116 is preferably less than the diameter of the cylinder wall 26^a and equal to or greater than the diameter of the pump rotor 45 and the liner 125^a is of lesser thickness than the difference between the diameter of the bore 96 in which it is tightly fitted and the external diameter of the rotor 116, excepting throughout the sealing portion 125^b thereof where its thickness matches the just-stated difference in diameters so as to provide a snug and sealing surface engagement with the rotating balancing rotor 116, somewhat like a packing gland. As shown in Figure 1, the axial extent of this sealing surface engagement is preferably substantial and peripherally the extent is 360° and continuous but at an angle to the axis equal to the pitch angle of the helical impeller 57 of the pump. This relationship appears better from Figure 3 where the angle is indicated at X. The sealing ring structure 125 is fixed within the bore 96 in any suitable way so that it will not be displaced axially or in a rotary direction, as by suitable locking pins 128 anchored in the cylinder portion 95 and extending into suitable holes in the member 125.

The above-described shape and disposition of the sealing portion 125^b, in relation to the pump rotor and pump cylinder, thus insures that a truncated portion 117^a of the external surface 117 of the balancing rotor 116, with the maximum axial dimension always at the 3 o'clock position, is always exposed to the pressure in the outlet chamber 35, and insures that the remaining truncated portion 117^b, having its major axial dimension always at the 9 o'clock position, is sealed from the effects of the discharge pressure being exposed to the pump inlet pressure by the channels 43 and 126. A thrust radially inwardly from the 3 o'clock position is thus always exerted upon the balancing rotor 116 and that thrust is in a direction opposed to the radially inward thrust from the 9 o'clock position exerted upon the pump rotor.

The parts are proportioned and dimensioned as above illustratively described, and the effective area of the balancing rotor that is exposed to the high pressure in the outlet chamber 35 is made large enough so that the product of the pressure multiplied by this area is substantially equal to the integrated product of the pressures (which increase progressively from the inlet to the discharge end of the rotor) effective upon the exposed areas of the pump rotor during the operation of the pump, thus achieving substantial balancing out of radial thrusts, preventing tendency to deflect the shaft 46 and preventing misalignment at the theoretical line of tangency 58; such misalignment could cause wear of the relatively moving parts with consequent impairment of efficiency and pumping action. The general proportioning of the parts of the balancing rotor 116, as shown in the drawings, is appropriate in relation to a pump structure of the number of turns of helical impeller 57 as shown in Figure 1; where the pitch of the turns is changed so that the radial thrust to be overcome is of a different order of magnitude, corresponding changes in proportioning of the balancing mechanism will, it is understood, be made.

To achieve appropriate seal between the helical impeller element 57, the pump cylinder 23—26, and the helical groove 56, I construct the helical element 57 to have, and to maintain during its continuously changing radial position relative to the helical groove of slot 56, a reliable sealing relation with at least one wall of this groove. In the form shown in Figure 1, the helical impeller 57 is made up of a plurality of individual elements or members 130 which have leading and trailing faces 130^a and 130^b, respectively, that extend in planes at right angles to the axis of the helix 57 and hence also the axis of the rotor 45; the outer edge face 130^c of each has a radius equal to the radius of curvature of the wall 26^a of the cylinder 23—26 and an inner edge face 130^d (see Figures 6 and 7) that has a radius of curvature equal to a dimension that is the radius of the outer edge face 130^c less the depth of the helical groove 56.

Thus each outer edge face 130^c can make sealing contact, throughout its entire area, with the cylinder wall 26^a and all of the elements 130, when assembled in the helical groove 56, make such sealing engagement with the cylinder wall 26^a, while the inner edge faces 130^d can make tangential sealing contact with the groove bottom 56^a when necessary, namely, when the rotor and helical element 56 pass through the 6 o'clock position at the theoretical line of tangency 58, the radial dimension of the elements 130 being not greater than the radial depth of the helical groove 56.

The members 130 are all of identical shape and construction and there are enough of them so that, when assembled with their leading and trailing faces 130^a and 130^b in contact with each other, they make up a helix of the desired number of turns and of an outside diameter equal to the diameter of the cylinder wall 26^a; the just-mentioned contacting faces 130^a—130^b will be seen to be of substantial areas so that each member 130 finds and receives substantial support from an adjacent one as against thrusts exerted in an axial direction as when forcing liquid, in the pump chamber C, from the left toward the right in Figure 1, and they receive as a whole or as an entire unit, rigid support from the side walls 56b and 56c of the helical groove 56; in this latter connection, it is to be noted that it is preferred to so dimension the parts that, at the 12 o'clock position, the members 130 or the helical impeller element 57 as a whole project from the groove 56 by only about one-half of their radial dimension.

The length in a circumferential direction of the members 130, that is, of the outer edge face 130c, is equal to an arc of the same number of degrees as the outer arc of a cross-section of the helical groove 56 taken at right angles to the axis of the rotor 45, and with that dimension as a base, the circumferential dimension of the inner edge face 130d follows according to the shapes given the end faces 130e and 130f which are to contact directly with the side walls 56c and 56b of the helical groove 56; these end faces are helical, or segments of helixes.

Because the groove 56 has substantial depth, the walls 56b and 56c not only are helical but also of varying curvature with respect to a plane extending at right angles to the axis of the rotor, the curvature varying as the radius or diameter changes from the minimum radius at the bottom wall 56a of the groove to the maximum radius at the surface of the rotor 45. Though the pitch is the same throughout, the pitch angle, that is, the angle of a tangent to the curved groove side wall with a plane transverse to the axis of the helix, becomes less as the diameter increases, and in Figure 11 I have shown the change in this angularity for a rotor having a 4'' diameter, and a pitch of 4'' for the groove (that is, the helical groove completes one turn of 360° in 4'' of axial length of the rotor). Thus, the above-mentioned angle with the radial plane, at the maximum radius of 2'' is 18° and the angle at the minimum radius, that is, where the side wall of the groove intersects the bottom wall, is 32°. Now, the end edge faces 130e and 130f are given helical curvatures which vary progressively, with a maximum outward displacement of the members 130 from the groove 56 of ½'' (at the 12 o'clock position), of 27½° where these end edge faces intersect with the inner edge face 130d to 20½° where they intersect with the outer edge face 130c; this range of values will be seen to be the middle half range, so to speak, of the range of change in angularity of the side walls of the groove 56 itself.

The end edge faces 130e and 130f of the members 130 and hence the side faces of the impeller element 57, treated or considered as a whole, are thus helixes of the same pitch as the rotor groove but of such different range of change of pitch angularity that, I have discovered, gives the helical impeller 57 not only a desirable flexibility of self-adjustment or self-accommodation to the groove as to achieve dependability mechanically of operation, but also insure such contact with particularly the trailing side wall 56b of the groove, throughout the continuously varying displacement of the elements 130 radially, as achieves a dependable seal between the end edge faces 130f and the trailing side wall 56b, even though the lines or areas of generally tangential contact therebetween continuously shift radially according to the change in radial distance of the members 130 from the bottom of the groove.

I emphasize the contact between the trailing end faces 130f and the trailing side wall 56b because the resistance or back pressure of the liquid being moved toward the right along the chamber C, coupled with a reaction tending to force the members 130 individually in a direction opposite to the direction of rotation, presses the end faces 130f against the trailing side wall 56b of the groove and, with continuous though variable contact always maintained therebetween, reliable sealing against leakage from the high pressure sides of the helix 57 to the low pressure side thereof is achieved. This sealing action is supplemented by the pressure with which the members 130 are maintained flatwise against each other at their adjacent contacting plane side faces 130a and 130b and the substantial expanse of these engaging side faces coacts with the end edge faces in supporting the elements 130 against tilting. For example, viewing the left-hand half turn of the impeller 57 in Figure 1, the circumferential extent of any one element 130 and hence of its trailing side face 130b is great enough to overlie, in projection, two, three, or more helical end edge faces 130f of as many succeeding or trailing elements 130.

The above-mentioned sealing contact effected at the trailing end faces 130f guards against leakage not only as above described but also against leakage of liquid being pumped along the continuously varying (in volume) chamber B formed by the helical groove 56 and the helical impeller 57 whose pumping capacity supplements that of the sealed chambers C progressively advanced from the left toward the right externally of the rotor 45 per se. Liquid enters the left-hand terminus of the slot 56 through suitable passages in the rotor head 50 as that end terminus is progressively opened up in turning from the 6 o'clock position to the 12 o'clock position in Figure 5, whence progressive closure thereof ensues from 12 o'clock to 6 o'clock via 9 o'clock, the entered liquid being progressively advanced by the progressive (in helical direction) bottoming of the elements 130 against the bottom wall 56a of the helical groove 56, to be discharged at the right-hand end face of the rotor 45, the rotor head 53 being provided with suitable passageway for that purpose.

In this connection, the end head 53, shown in end elevation in Figure 2, is provided with a discharge opening 132 that overlies the right-hand terminus of the helical slot 56 in the end face of the rotor 45; the extent of the terminus of the slot in that end face, in circumferential direction, is substantial because the slot is helical and of substantial width in axial direction, and is indicated at 56g in Figure 2; the opening 132 in the end head 53 is of substantially matching peripheral or arcuate extent and the leading and trailing end walls 132a of the opening 132 are bevelled off at about 45° to effect smoothness of egress of liquid under pressure.

The discharge opening 132, however, overlies only about the inner half of the radial extent of the slot terminus 56g, so as to leave the arcuate portion 53a of the head 53 to close off the outer half portion of the slot terminus 56g and thus form a physical abutment for the endmost member 130 of the impeller 57, engaging the part 53a by way of a front or leading face 130a thereof. To avoid unbalance, because of the discharge opening 132, a diametrically opposed portion of the head 53 may be hollowed out as at 53b (Figures 2 and 1).

The head 50 at the inlet end of the rotor is provided with an arcuate inlet opening 133 similar to the just-mentioned discharge opening 132 in the head 53, and its leading and trailing end faces 133a are also preferably bevelled off, for example, at about 45°, in directions to function, with respect to the direction of rotation, as scoops to aid in injecting liquid from the inlet chamber 31 into the inlet terminus of the helical groove 56. Head 50 may be hollowed out as at 50^b, diametrically opposed to the inlet opening 133^a, for balance. In Figure 1 is shown in cross-section the portion 50^a, corresponding to the portion 53^a of Figure 2, that overlies about the outer half of the inlet terminus of the groove 56 and against which the endmost member 130 abuts and is supported, flatwise, by its trailing side face 130^b.

Maintenance of the elements 130 in position to make continuous sealing contact with the cylinder wall 26^a is aided not only by their structural interrelationships to the helical groove 56 but also by the centrifugal forces acting upon them as they are rotated. I prefer, however, also to supplement these actions and thus, referring to Figures 1, 6, 7 and 9, I preferably bevel off the inner edge faces 130^d, as at 130^g, throughout their arcuate extent but only along the trailing portions thereof; this insures postiveness of entry of liquid underneath the elements 130 when they are in engagement with the bottom wall 56^a of the groove 56 to insure against tendency to create a vacuum between them and the bottom wall 56^a, which would tend to resist outward radial movement of the elements 130, and with the forceful advancement of liquid along the helical groove 56 as above described, such entry of liquid under pressure operates in effect as a progressively acting hydraulic lift for the elements 130 to force them outwardly, aided by centrifugal force, and thus insure that they always are in sealing engagement with the cylinder wall 26^a. This bevelling off of the under or inner edge faces also aids, at the entry terminus of the groove 56 adjacent the inlet chamber 31, in achieving initial lift or outward radial movement of those members 130 that become directly exposed to the inlet terminus of the groove 56; in this latter connection, the scooping or impeller action effected by the bevelling of the radial end faces 133^a of the entry opening 133 in the rotor head 50 coact to inject liquid and give it force or pressure to provide initial lifting action if the latter becomes necessary.

The construction thus far described makes possible a variety of combinations of materials capable of employment. Thus the liner 58 and the members 130 may be made of any suitable material or materials capable of appropriate coaction in relation to the liquid being pumped. For example, if oils or other self-lubricating liquids are being pumped, these parts may be made of metal or of any combination of metal and non-metallic parts that are appropriate. If water or other non-lubricating liquids are to be handled, the rotor may be made of any suitable non-corrosive metal, such as stainless steel, the members 130 may be made of a non-metallic material, such as hard rubber, any suitable plastic, compositions like so-called "Micarta" which operate with water as a lubricant, and the liner 58 may be made of suitable plastic water-lubricating compositions. If the members 130 are made of metal whose density might give them a weight that might result in excessive centrifugal forces, the members 130 may have one or more holes drilled in them, as indicated in Figure 6 at 130^h, the hole or holes being dimensioned or positioned so that they do not overlap when the members 130 are assembled to form the impeller 57, the hole or holes being of sufficient size to reduce the weight of the member.

If the members 130 are made of a non-metallic material and prove to be insufficiently heavy to give the desired effect under the action of centrifugal forces, such hole or holes 130^h may be filled with a metal like lead, as indicated at 134 in Figure 7, thus adding to the weight of each member 130 as may be desired.

In Figure 10, I have shown another form of construction for the helical impeller 57 which is also self-accommodating to the side wall or walls of the helical slot 56 to achieve suitable assurance against leakage between contacting faces of the varyingly-angled side walls of the groove and the faces of the impeller 57. The helical impeller of Figure 10 is also made up of a plurality of elements; these are indicated individually by the reference character 135, and each member 135 is preferably a complete helix in and of itself, of as many turns as there are turns in the helical groove 56, having an outside diameter matching the diameter of the cylinder wall 26^a and having an inside diameter of such dimension as will give each helical member 135 a radial dimension equal to the depth of the helical groove 56. The members 135 are of relatively small thickness and illustratively may be made of any suitable metallic or non-metallic material; their relatively small thickness gives them individually considerable flexibility, preferably spring-like or resilient, and a suitable number of them are assembled to give the helical impeller a thickness to be snugly received within the helical groove. At suitable intervals throughout the resultant composite helix, as for example every 3" or so, the members 135 may be secured together, as by cross-pins or rivets 136, the securing action being appropriate to hold the plurality of helical plate-like members 135 against relative displacement in a radial direction, thus to maintain their outer edge faces 135^c in alignment to form substantially a continuous face or surface for sealing contact with the cylinder wall 26^a. The composite helix may have, or may be given by construction, a tendency to expand and such action, being substantially spring-like or resilient, may if desired contribute toward maintaining the just-mentioned sealing contact. The space underneath the composite helix and between the aligned inner edge faces 133^d and the bottom wall 56^a of the helical groove, which varies progressively during the operation of the pump and is progressed from the inlet end to the outlet end of the groove so that it is effective also to pump liquid, can thereby also be utilized to insure internal outward radial pressure against the composite helix to achieve the above-mentioned sealing contact with the cylinder wall 26^a.

At the inlet and outlet ends of the helical groove 56, partially closed off by the portions 50^a and 53^a of the rotor end heads 50 and 53, respectively, the members 135 terminate in end edge faces which are oblique to the helix and fall in planes respectively coincident with the end faces of the rotor 45, thereby abutting at the respective ends of the composite helix against the arcuate head portions 50^a and 53^a to aid in holding the built-up helix against tendency to creep or thread itself along the helical groove. Preferably, at suitable intervals, for example, every 360°, I provide a radially extending stud 137, preferably in the form of a headless screw threaded into the rotor and thus projecting outwardly from the bottom wall 56^a of the helical groove and, in assembly, threaded through a hole 138 drilled through the composite helix, thus to form a row of parallel radially extending studs, one for each turn of the helix, to resist such reactions during pumping as tend to cause creepage of the helix along the helical groove.

The holes 138 for studs 137 are preferably given a shape approximating an hour-glass, or may contain a yieldable rubber bushing, and studs 137 terminate at their outer ends at or within the cylindrical plane of the surface of the rotor 45 so that they do not project out of the helical groove and still provide ample radial length to maintain sliding engagement with the composite helical impeller and permit change in relative angularities as the impeller partakes of relative radial inward and outward movements with respect to the bottom wall 56^a of the helical groove, as in passing from the 6 o'clock position to the 12 o'clock position and from the 12 o'clock position to the 6 o'clock position. The studs 137 thus prevent such reactions upon the helical impeller as tend to expand it from actually expanding the impeller to extents as would create excessive friction or as would cause jamming between the impeller and the cylinder wall 26^a.

The holes 138 in which the studs 137 engage, being preferably shaped as above described or containing a yieldable bushing, provide some extent of play toward the right or left as viewed in Figure 10 of the composite impeller relative to the groove 56; this relative play allows also for changes in angularity of the impeller, or portions of it, somewhat in a direction transversely of the helical groove. This play is desirable so as not to detrimentally interfere with the resilient flexing of the plate-like elements 135 individually and as a unit for purposes of achieving self-accommodation, due to the resultant flexibility of the helical impeller, of the trailing side face 135^f to the trailing side wall 56^b of the helical groove, throughout the changing angularities of this groove side wall 56^b at different radii thereof, as above explained, so as to maintain continuity of sealing engagement for whatever radial position that the composite helical impeller has relative to the groove 56. The reaction of the liquid pressure being built up as the pump forces liquid from the inlet chamber toward the outlet and balancing chamber is in a direction to press the helical impeller always against the groove side wall 56^b and the laminated structure of the impeller gives it such resiliency and flexibility as permits each turn or portion of a turn of the helix to yield or flex, about radial axes, relative to an adjacent turn or adjacent portion as will effect maintenance of sealing contact at the trailing groove side wall 56^b for any radial position of the impeller relative thereto.

This resilient flexibility, it will now be seen, may be easily predetermined for various practical conditions and for various sizes and designs or capacities of pump construction. Thus, according to the material employed in making up the members 135, they may be made of different thicknesses according to the yieldability desired or needed to maintain the above-described self-accommodating sealing action. The members 135 may be made up of metal or of non-metallic material, such as relatively hard or rigid rubber, resinous compositions, so-called plastics, or the like, and any combination therewith of different materials for the cylinder liner 26 may be made up according to the requirements of the particular liquid to be handled.

In achieving adequate maintenance of sealing between the built-up impeller of Figure 10 and the wall or walls of the helical groove 56, the axial and radial thrusts exerted by or upon the rotor become substantially determinate and the balancing rotor 116 and its related parts can be constructed and proportioned to achieve adequacy of compensation or counterbalance and thereby also achieve the advantages structurally and functionally earlier above described. With either of the illustrative forms of composite or built-up helical impellers, it is preferred that the helical groove 56, if only a single groove and impeller are employed, be not less than two turns and if two grooves and impellers are employed each groove and impeller should be not less than one and one-half turns, to avoid the existence of a continuous or free passage from the inlet chamber to the outlet chamber. In general, each additional turn adds another stage to the pump. The features of my invention are, it will now be seen, readily adaptable to these and other variations.

Whatever the number of turns or number of helixes employed, and this is true of the impeller structure of Figure 1 and also of the impeller structure of Figure 10, the members 130, being in a sense laminations in planes transverse of the axis of the rotor, and the members 135, being laminations in helical planes of the helical impeller, coact with each other and with the cylinder wall and walls of the helical groove to give multiple sealing actions. For example, these members individually engage the cylinder wall 26^a and in each case have enough freedom of relative movement to be individually seated snugly against the cylinder wall; each such seating achieves sealing action and these seatings and sealing actions are multiplied, due to the laminated arrangement or due to the overlapping of the members, in a direction lengthwise of the axis, that being the direction in which successive closed chambers are formed between the rotor, turns of the helical impeller, and cylinder wall and progress toward the right in Figures 1 and 10. Leakage out of such chambers would tend to take place in reverse direction, namely, toward the left; multiple sealing takes place between these impeller elements and the cylinder wall as just described and in a sense the sealing may be said to take place in a succession of individual stages. In both forms, leakage, in the just-stated direction, between the impeller and the rotor is guarded against by the flexibility of the helical impeller as a whole and the self-accommodation of its trailing helical face (made up in Figure 1 of the individual end faces 130^f and made up by the trailing end face 135^f in the form of Figure 10) to the varying angularities of the wall 56^b of the groove, varying with the radial distance from the axis.

If desired from the viewpoint of manufacture, each helical element or lamination 135 in the form of Figure 10 may be made up of sections of any desired arcuate extent, the sections of any one complete helical lamination 135 being in abutting end to end relation throughout the complete length of the helix, with butt joints of successive complete helixes 135 suitably staggered relative to one another; when so built up, the rivets or pins 136, which may be positioned in the lower half of the impeller assembly so as not to be exposed out of the helical groove 56 when in the 12 o'clock position may again be used to hold the laminations against material relative radial or circumferential displacement, and these pins may be located so as to assemble all of the sectionalized laminations into a single unitary helix or into sections of the helix of each composite section being of appropriate arcuate extent and adjacent ends of such helix sections being in effect interfitted as in a mortise and tenon joint, due to the above-described staggering of the sections of individual helixes.

In Figure 10, certain of the intermediate laminations or elements 135 are indicated as sectionalized by the lines 135x which indicate end to end abutting relationships of successive sections. In whatever way the members 135 are sectionalized it is preferred that the two outermost elements 135, being respectively the leading and trailing elements of the helical impeller be continuous throughout the length of the helix. Sectionalizing as above described also has the added advantage, particularly with certain relatively stiff materials, of giving wider range of flexibility, about radially extending axes, one portion of the helical impeller relative to an adjacent portion, thus to better achieve continuity of sealing engagement throughout the varying angularities of the trailing side wall 56^b of the helical groove.

The gland element 103 of the stuffing box within the right-hand end casing 61 (Figure 1) is provided with suitable means for adjusting the pressure it exerts upon the packing material and for removing it to replace the packing material, and for this purpose the side walls of the housing 61 are cut away as at 140 and 141 to give access to suitably mounted studs 142 and nuts 143 by which a cross-bracket 144, which bridges across the right-hand end of the stuffing box element 103; a similar arrangement, not shown, is provided for adjusting and removably holding the packing gland at the left-hand end of the shaft 46, the housing 60 at that end being suitably cut away as at 145 and 146 to give access thereto.

As appears better from Figure 1, the sealing portion 125^b of the sealing ring will be seen to coact also in lessening the burden imposed upon the above-mentioned stuffing box in the end casing 61, for it seals the liquid under discharge pressure in the discharge chamber off from the inner end of the stuffing box which thus becomes exposed to the low pressure of liquid communicated to that portion of the balancing chamber by the passage 43—126 that leads from the inlet chamber 31.

The communication of low or inlet pressure liquid from the inlet chamber 31 to the low pressure surfaces or areas of the balancing rotor 116 in the balancing chamber 35 may be effected by other or additional means than the cored-out passages 43 and 126, and this I do by a preferred construction which I am enabled to cause to coact to achieve, when desired, certain practical advantages.

For example, for certain sizes or capacities of pump construction, it may be desirable or necessary that the shaft 46 be solid throughout, and in such case the inlet chamber 31 and the sealed-off portion of the balancing chamber 35 are connected together in, for example, the manner above described, such as the cored-out passages 43 and 126; but for certain other sizes or capacities of pump construction, particularly constructions where it may be undesirable to disrupt the external cylindrical surface of the casing 20 as by the longitudinally extending bulge formed by the passage or channel 43, I prefer to provide the communication between inlet chamber 31 and the sealed-off portion of the balancing chamber 35 by means of a passageway extending through and coaxially of the shaft 46 itself.

In the latter case, and also where it may be desired to supplement the communication provided by the cored passages 43—126, I construct the shaft 46 as is better shown in Figure 1. Thus, I may provide the above-mentioned shoulder 46^b adjacent the left-hand end of the shaft and the shoulder 46^d adjacent the right-hand end of the shaft by the respective annular faces of a heavy-walled tubular shaft portion 46^e having therefor an internal coaxial channel 46^p extending therethrough from end to end, in which case the lesser-diametered shaft portion 46^a at the left-hand end has a plug-like lesser-diametered extension 46^f fitted into and closing off the left-hand end of the channel 46^p and the lesser-diametered shaft part 46^c at the right-hand end has a similar lesser-diametered plug extension 46^g extending into and closing off the right-hand end of the channel 46^p. These interfitting parts may be mechanically secured against coming apart in any suitable way, as by force-fitting, welding, brazing, pinning, or the like.

Where the tubular shaft portion 46^e is exposed to the inlet chamber 31, I provide several equiangularly distributed holes 46^h and where it is exposed to the low pressure portion of the balancing chamber 35, I also provide several equiangularly displaced holes 46^k, communicating with the passageway 46^p in the shaft portion 46^e, thus to complete the communication via the shaft channel 46^p between the low pressure inlet chamber and the low pressure portion of the balancing chamber.

With such an arrangement and for certain capacities or pressure ranges of pump construction, I am also enabled to gain the advantage of the characteristics of the tubular cross-section of the shaft portion 46^e in offering greater resistance to flexing per cross-sectional area of metal employed than does a solid shaft and these features may be made to coact in lessening the radial-thrust-opposing burden carried by the balancing rotor.

It will thus be seen that there has been provided in this invention pump constructions in which the several objects hereinbefore pointed out together with many thoroughly practical advantages are successfully achieved. The construction, as a pump, is dependable, efficient, compact and durable, and manufacture and assembly of its various parts may be carried on with efficiency and at reasonable cost. For example, the component parts of the helical impeller constructions may be individually made up in quantity, being substantially identical or standardized for a given capacity or for respective practical conditions of operation, and may be readily and inexpensively assembled to the rotor in the form of a composite helical impeller. It will also be seen that a wide variety of practical pumping requirements, such as diversity of liquids to be handled, may be with facility and efficiency met in a thoroughly practical way.

As many possible embodiments may be made of the above invention and as many changes might be made in the embodiment above set forth, it is to be understood that all matter hereinbefore set forth or shown in the accompanying drawings is to be interpreted as illustrative and not in a limiting sense.

I claim:

1. A pump construction comprising a casing having an inlet chamber and an outlet chamber joined by a cylinder, with a lesser-diametered rotor within the cylinder rotatably mounted to

be substantially tangential to the cylinder wall along a line of the latter, said rotor having means comprising helical impeller means in sealed connection with the rotor and making sealed contact with said cylinder wall and means for accommodating said impeller means as it passes through the line of tangency, a balancing rotor coaxial with and connected to said first rotor and extending into said discharge chamber and having a cylindrical surface, said discharge chamber having sealing means in engagement with said balancing rotor to seal off from the pressure of liquid in said discharge chamber all of said cylindrical surface excepting a portion thereof displaced substantially 180° from the direction of radial thrust exerted upon said first rotor.

2. A pump construction comprising a casing having an inlet chamber and an outlet chamber joined by a cylinder, with a lesser-diametered rotor within the cylinder rotatably mounted to be substantially tangential to the cylinder wall along a line of the latter, said rotor having means comprising helical impeller means in sealed connection with the rotor and making sealed contact with said cylinder wall and means for accommodating said impeller means as it passes through the line of tangency, whereby said rotor and impeller means are subjected to an axial thrust that is a function of the difference between the pressure in said outlet chamber and the pressure in said inlet chamber, a balancing rotor coaxial with and connected to said first rotor and having opposed surface areas with means sealing the one off from the other, and means subjecting said two surface areas respectively to the pressure of liquid in said discharge chamber and to pressure of liquid in said inlet chamber whereby said balancing rotor exerts an axial thrust that is a function of the difference of said two pressures and in a direction opposed to said first-mentioned axial thrust.

3. A pump construction comprising a casing having a cylinder therein terminating at its respective ends in an inlet chamber and an outlet chamber, a shaft having thereon a rotor of lesser diameter than said cylinder and extending at its ends through and in sealed connection with the outer end walls of said casing with bearing means rotatably supporting it to rotate upon an axis such that said rotor is substantially tangential to the cylinder wall along a line of the latter, said rotor having means comprising helical impeller means in sealed connection with the rotor and making sealed contact with said cylinder wall and means for accommodating said impeller means as it passes through the line of tangency, whereby, during pumping action of said impeller means, said shaft is subjected to a radial thrust, said shaft having operative thereon at a point intermediate of its supports in said bearing means a radial thrust-transmitting means operating in response to liquid under substantially the same pressure as the pressure of liquid in said outlet chamber and exerting a radial thrust upon the shaft in a direction opposed to said first-mentioned radial thrust.

4. A pump construction comprising a casing having a cylinder therein terminating at its respective ends in an inlet chamber and an outlet chamber, a shaft having thereon a rotor of lesser diameter than said cylinder and extending at its ends through and in sealed connection with the outer end walls of said casing with bearing means rotatably supporting it to rotate upon an axis such that said rotor is substantially tangential to the cylinder wall along a line of the latter, said rotor having means comprising helical impeller means in sealed connection with the rotor and making sealed contact with said cylinder wall and means for accommodating said impeller means as it passes through the line of tangency, said shaft carrying a balancing rotor having a surface of revolution with means sealing off from the pressure of liquid in said discharge chamber all of said surface of revolution excepting a portion thereof displaced substantially 180° from the direction of said radial thrust exerted upon said first rotor.

5. A pump construction as claimed in claim 1, in which said sealing means extends peripherally about said cylindrical surface with a bounding edge thereof falling in a plane that intersects the axis of said balancing rotor at an angle whereby the said portion of said cylindrical surface that is subjected to the pressure of liquid in said discharge chamber is in the shape of a truncated cylinder whose effective area is displaced substantially 180° from the direction of radial thrust exerted upon said first rotor.

6. A pump construction comprising a casing having an inlet chamber and an outlet chamber joined by a cylinder, with a lesser-diametered rotor within the cylinder rotatably mounted to be substantially tangential to the cylinder wall along a line of the latter, said rotor having means comprising helical impeller means in sealed connection with the rotor and making sealed contact with said cylinder wall and means for accommodating said impeller means as it passes through the line of tangency, a balancing rotor coaxial with and connected to said first rotor and extending into said discharge chamber and having oppositely directed annular faces and also a surface of revolution, sealing means to seal off from the pressure of liquid in said discharge chamber all of said surface of revolution excepting a portion thereof displaced substantially 180° from the direction of radial thrust exerted upon said first rotor and excepting one of said annular faces, and means for subjecting the other of said annular faces to pressure of liquid in said inlet chamber.

7. A pump construction comprising a casing having an inlet chamber and an outlet chamber joined by a cylinder, with a lesser-diametered rotor within the cylinder rotatably mounted to be substantially tangential to the cylinder wall along a line of the latter, said rotor having means comprising a helical groove in its surface having therein a laminated helical impeller means of outside diameter equal to the diameter of said cylinder with the individual laminations thereof made of substantially rigid material whereby said laminations as a whole are substantially self-accommodating to the surfaces with which they engage for substantially sealing contacts of said impeller means with said cylinder wall and with a wall or walls of said groove, whereby there are exerted upon said rotor an axial thrust and a radial thrust substantially unimpaired by material leakage of liquid, and means operating upon said rotor and responsive to the pressure differential between the liquid in said outlet chamber and the liquid in said inlet chamber for exerting upon the rotor axial and radial thrusts respectively opposed to the afore-mentioned axial and radial thrusts.

8. A pump construction comprising a casing having an inlet chamber and an outlet chamber joined by a cylinder, with a lesser-diametered

rotor within the cylinder rotatably mounted to be substantially tangential to the cylinder wall along a line of the latter, said rotor having means comprising a helical groove in its surface having therein a laminated helical impeller means of outside diameter equal to the diameter of said cylinder with the individual laminations thereof made of substantially rigid material whereby said laminations as a whole are substantially self-accommondating to the surfaces with which they engage for substantially sealing contacts of said impeller means with said cylinder wall and with a wall or walls of said groove, whereby there is exerted upon said rotor a radial thrust substantially unimpaired by material leakage of liquid, means responsive to pressure of liquid in said discharge chamber, and means coacting with said pressure-responsive means for transmitting force exerted by the latter to the rotor as a radial thrust exerted in a direction displaced substantially 180° from the direction of said first-mentioned radial thrust.

9. A pump construction as claimed in claim 8 in which said individual laminations have leading and trailing side faces that extend in respective planes that are parallel to each other and at right angles to the axis of said rotor whereby each lamination presents an arcuate outer edge face for contact with the cylinder wall with outer edge faces of successive laminations progressively spaced angularly from one another to thereby form multiple sealing contact with the cylinder wall, each lamination presenting leading and trailing end edge faces for making multiple sealing contact with the leading and trailing side wall of said groove.

10. A pump construction as claimed in claim 8 in which said individual laminations have leading and trailing side faces that extend in respective substantially parallel helical planes of the same pitch as said helical groove and thereby present outer edge faces for multiple sealing contact with the cylinder wall and said impeller means as a whole is given a flexibility for self accommodation of its trailing side face for sealing engagement with the trailing side wall of said groove.

11. A pump construction comprising a casing having an inlet chamber and an outlet chamber joined by a cylinder, with a lesser-diametered rotor within the cylinder rotatably mounted to be substantially tangential to the cylinder wall along a line of the latter, said rotor having a helical groove of substantial depth whereby the side walls of the groove vary in pitch angle according to the radial distance from the axis of the rotor, and a laminated helical impeller means in said helical groove, said impeller means being of outside diameter equal to the diameter of said cylinder wall and of radial dimension materially greater than the displacement between the axis of the rotor and the axis of said cylinder, whereby the radial displacement between the axes of said impeller means and of said rotor varies during rotation and engagement and said impeller means partakes of radial movement relative to the groove side walls of varying pitch angle, the individual laminations of said impeller means presenting a plurality of surfaces for multi-stage sealing contact and giving the helical impeller means substantial self-accommodation to the surfaces with which said impeller means engages and sealing engagement is maintained between the impeller means and the trailing side walls of the helical groove throughout said radial movement therebetween and the varying pitch angle of said trailing side wall.

12. A pump construction comprising a casing having an inlet chamber and an outlet chamber joined by a cylinder, with a lesser-diametered rotor within the cylinder rotatably mounted to be substantially tangential to the cylinder wall along a line of the latter, said rotor having a helical groove of substantial depth whereby the side walls of the groove vary in pitch angle according to the radial distance from the axis of the rotor, and a laminated helical impeller means of outside diameter equal to the diameter of said cylinder wall and of radial dimension materially greater than the displacement between the axis of the rotor and the axis of said cylinder, the individual laminations of said impeller means having leading and trailing side faces lying in respective planes that are parallel to each other and extend at right angles to the axis of the rotor and thereby presenting multiple outer edge faces for multi-stage sealing engagement with said cylinder wall and each lamination having leading and trailing end edge faces for respective multiple engagement with the leading and trailing side walls of said helical groove, the trailing end edge faces of said laminations having their faces conformed to a helical plane of the same pitch as said groove and of varying pitch angles throughout a range less than but within the limits of the range of change of pitch angles of the trailing side wall of said helical groove.

13. A pump construction comprising a casing having an inlet chamber and an outlet chamber joined by a cylinder, with a lesser-diametered rotor within the cylinder rotatably mounted to be substantially tangential to the cylinder wall along a line of the latter, said rotor having a helical groove of substantial depth whereby the side walls of the groove vary in pitch angle according to the radial distance from the axis of the rotor, and a laminated helical impeller means of outside diameter equal to the diameter of said cylinder wall and of radial dimension materially greater than the displacement between the axis of the rotor and the axis of said cylinder, the individual laminations of said impeller means having leading and trailing end edge faces for respective contact with the leading and trailing side walls of said helical groove, with the faces of at least the trailing end edge faces thereof conformed to a helical plane of the same pitch as said helical groove with variable pitch angles throughout a range less than but within the limits of the range of change of pitch angle of the trailing side wall of said groove.

14. A pump construction comprising a casing having an inlet chamber and an outlet chamber joined by a cylinder, with a lesser-diametered rotor within the cylinder rotatably mounted to be substantially tangential to the cylinder wall along a line of the latter, said rotor having a helical groove of substantial depth whereby the side walls of the groove vary in pitch angle according to the radial distance from the axis of the rotor, helical impeller means of outside diameter equal to the diameter of said cylinder wall and of radial dimension greater than the difference in diameters of said cylinder wall and said rotor whereby said impeller means projects from the groove in all regions excepting in the region where the rotor and cylinder wall are substantially tangential, said helical impeller means being made up of a plurality of individual parts made of a material inherently substantially rigid to thereby give

the impeller means substantial self-accommodation for sealing contact with at least the variable pitch-angled trailing side wall of said groove throughout the range of radial displacement of the impeller means relative to said groove.

15. A pump construction comprising a casing having an inlet chamber and an outlet chamber joined by a cylinder, with a lesser-diametered rotor within the cylinder rotatably mounted to be substantially tangential to the cylinder wall along a line of the latter, said rotor having a helical groove of substantial depth whereby the side walls of the groove vary in pitch angle according to the radial distance from the axis of the rotor, and a laminated helical impeller means of outside diameter equal to the diameter of said cylinder wall and of radial dimension materially greater than the displacement between the axis of the rotor and the axis of said cylinder, the laminations comprising a plurality of helical elements made of a material inherently substantially rigid whereby said helical impeller means is inherently yieldable for substantial self-accommodation of its trailing side face to sealing engagement, throughout the varying radial displacements of the impeller means relative to the groove, with the variable pitch-angled trailing side face of said helical groove.

16. A pump construction as claimed in claim 15 in which at least one of said helical laminations is sectional.

17. A pump construction as claimed in claim 15 in which a plurality of said helical laminations are in sections with successive sections in substantially end to end abutting relationship, the junctions between sections of successive helical laminations being angularly displaced.

18. A pump construction as claimed in claim 15 provided with means holding said laminations against relative rotary displacement therebetween and means for holding said helical impeller means at a plurality of points throughout its extent against relative rotary displacement between it and said rotor.

19. A pump construction comprising a casing having an inlet chamber and an outlet chamber joined by a cylinder, with a lesser-diametered rotor within the cylinder rotatably mounted to be substantially tangential to the cylinder wall along a line of the latter, said rotor having a helical groove of substantital depth whereby the side walls of the groove vary in pitch angle according to the radial distance from the axis of the rotor, and a laminated helical impeller means of outside diameter equal to the diameter of said cylinder wall and of radial dimension materially greater than the displacement between the axis of the rotor and the axis of said cylinder, said rotor having end faces exposed respectively to said inlet and outlet chambers and having the ends of said helical groove respectively terminating in said end faces, and means closing off the ends of said helical groove and thereby form abutments to hold the laminated helical impeller against endwise emergence from the ends of the groove, said means providing passageway means for the entry therethrough and at the end of the groove adjacent said inlet chamber of liquid into the variable helical space between the bottom of said helical groove and said helical impeller means and for the exit of liquid from said variable space at the other end of the groove for discharge into said discharge chamber, whereby the liquid is forced along the bottom portions of the groove as the displacement of the laminations relative to the groove progresses from the inlet end to the outlet end.

20. A pump construction as claimed in claim 19 in which the inner edge faces of said laminations that are exposed toward the bottom wall of said groove are conformed to form with the bottom wall of the groove a mouth-like entry for ingress of liquid thereinunder to effect projection thereof outwardly of the groove.

21. A pump construction as claimed in claim 6 in which the last-mentioned means comprises channel means formed in a wall of said casing and communicating at one end with said inlet chamber and at the other end with that portion of said discharge chamber that is sealed off from the discharge pressure liquid in said discharge chamber.

22. A pump construction as claimed in claim 6 in which said last-mentioned means comprises a tube-like shaft rotatably supporting said rotor and having a passageway therethrough communicating at one end with said inlet chamber and at the other end with that portion of the discharge chamber that is sealed off from the discharge pressure liquid in said discharge chamber.

FRANCISCO A. QUIROZ