

Nov. 30, 1948.

B. F. SCHMIDT
SUBMERSIBLE DOUBLE-ACTING FLUID
PISTON DEEP WELL PUMP

2,455,022

Filed Aug. 8, 1944

13 Sheets-Sheet 1

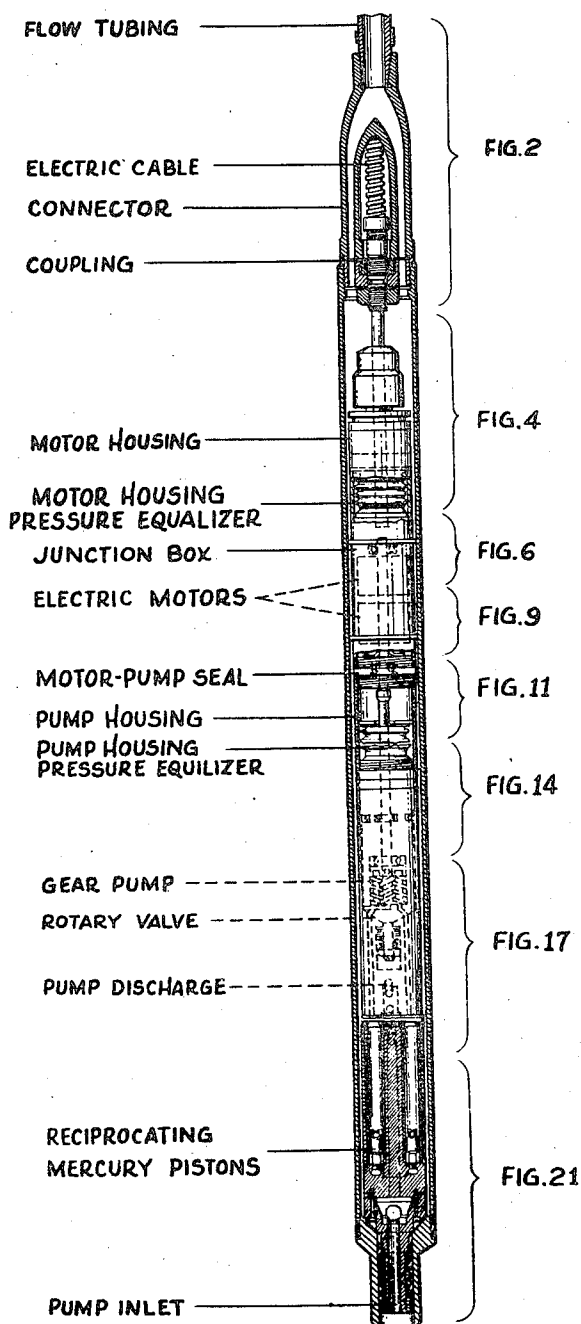


Fig. 1.

INVENTOR.
BENJAMIN F. SCHMIDT,
BY *Harold W. Mattingly*
Attorney

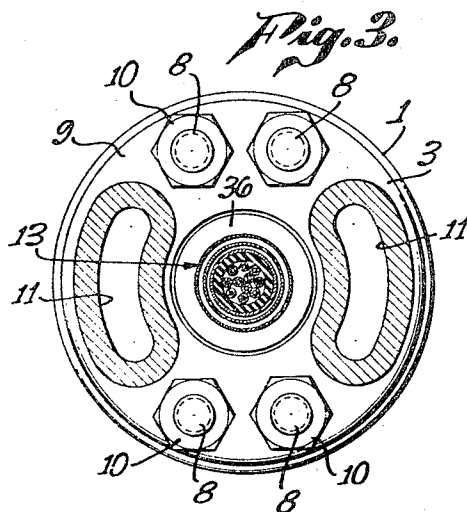
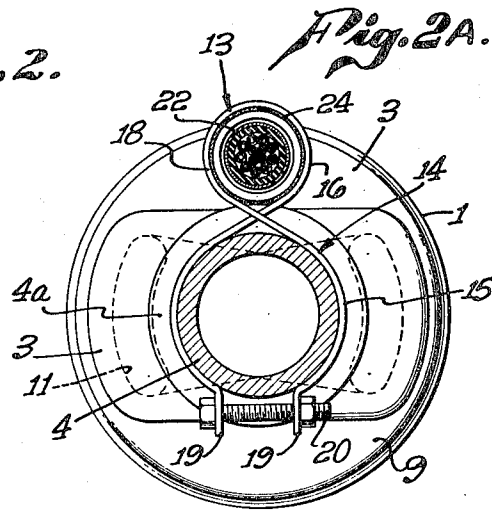
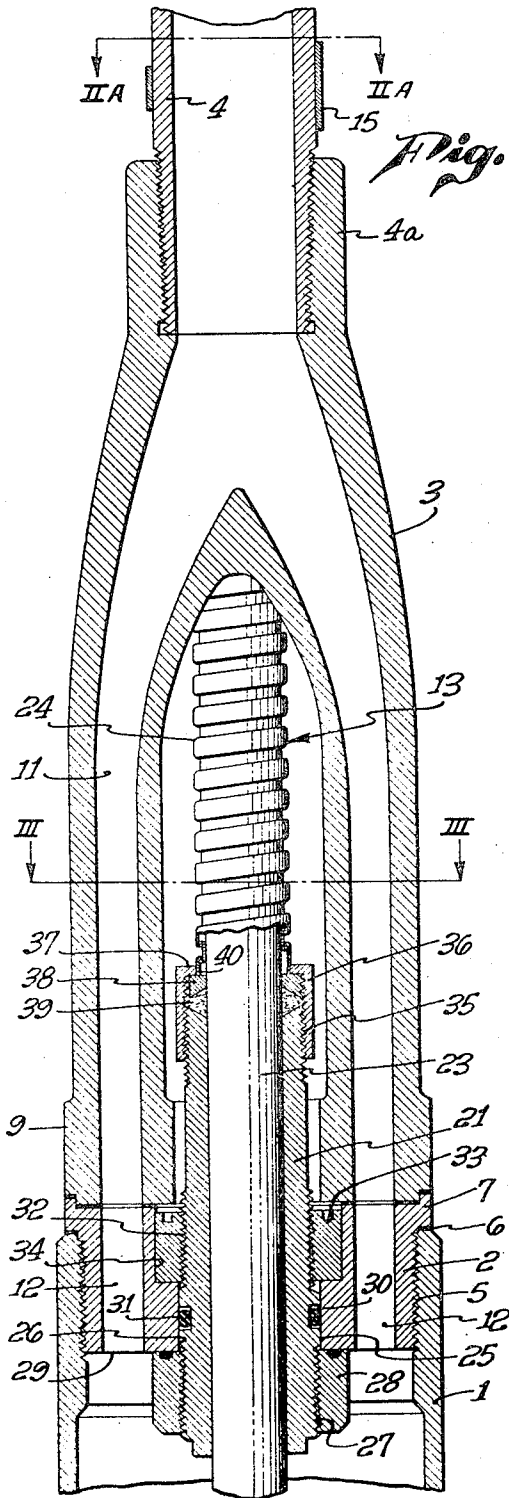
Nov. 30, 1948.

B. F. SCHMIDT
SUBMERSIBLE DOUBLE-ACTING FLUID
PISTON DEEP WELL PUMP

2,455,022

Filed Aug. 8, 1944

13 Sheets-Sheet 2



Inventor
BENJAMIN F. SCHMIDT,
By *Harold R. [Signature]*
Attorney

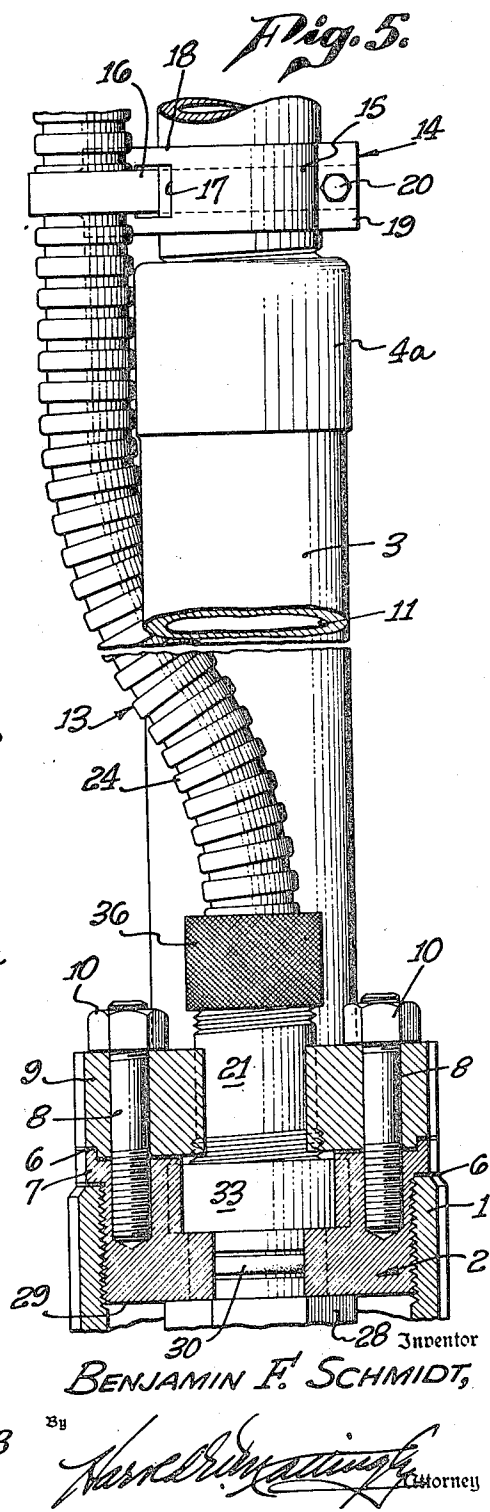
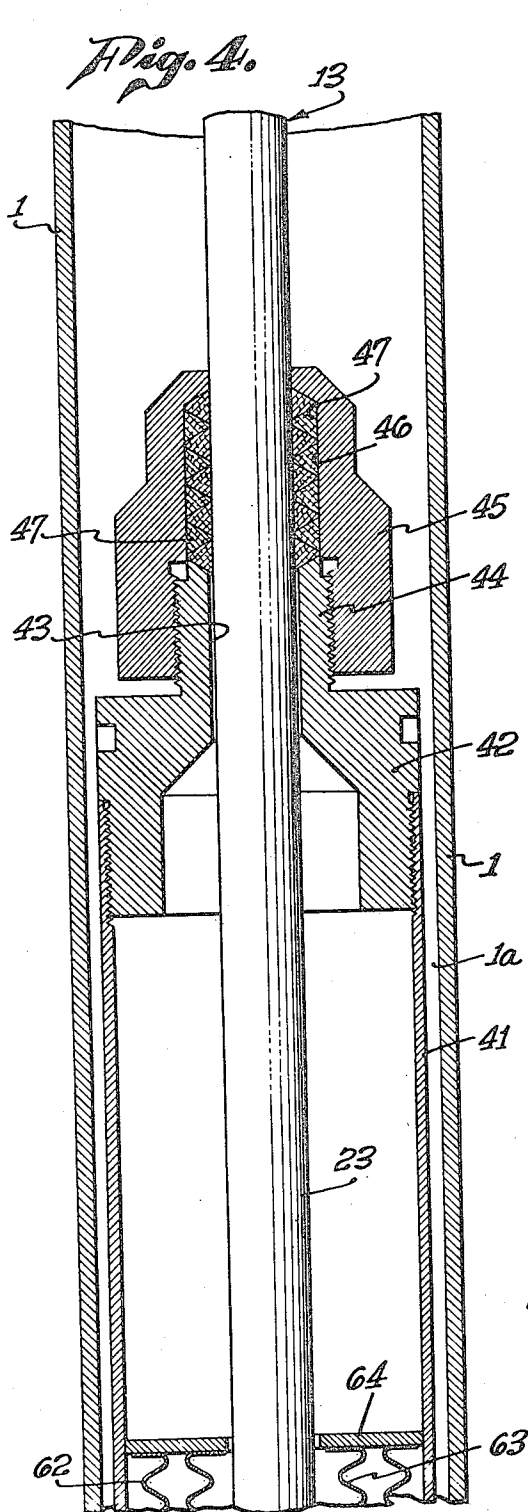
Nov. 30, 1948.

B. F. SCHMIDT
SUBMERSIBLE DOUBLE-ACTING FLUID
PISTON DEEP WELL PUMP

2,455,022

Filed Aug. 8, 1944

13 Sheets-Sheet 3



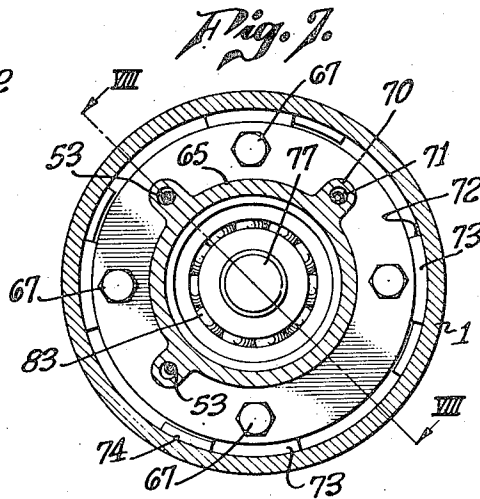
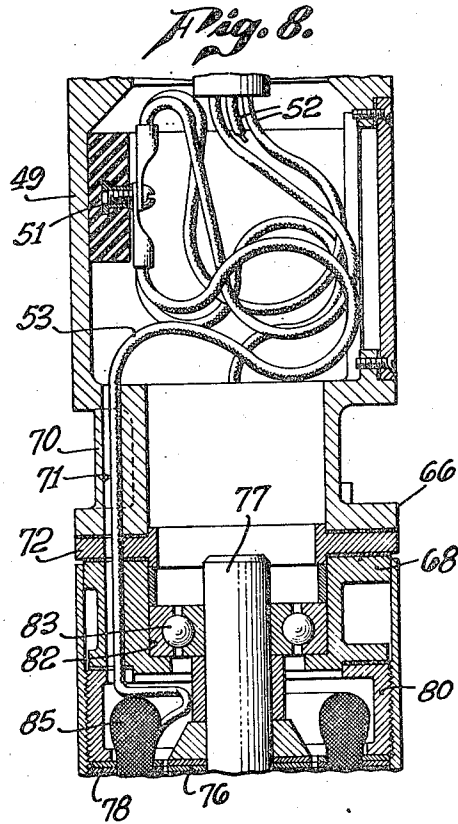
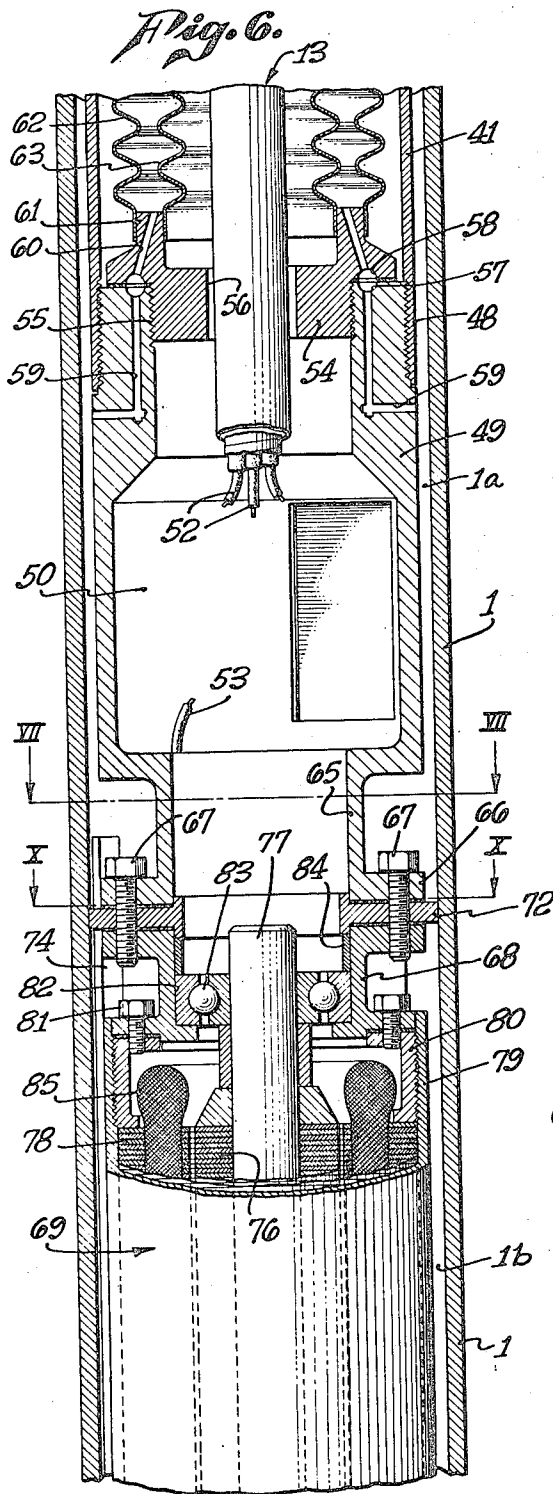
Nov. 30, 1948.

B. F. SCHMIDT
SUBMERSIBLE DOUBLE-ACTING FLUID
PISTON DEEP WELL PUMP

2,455,022

Filed Aug. 8, 1944

13 Sheets-Sheet 4



Inventor
BENJAMIN F. SCHMIDT,

By *Harold R. Manning*
Attorney

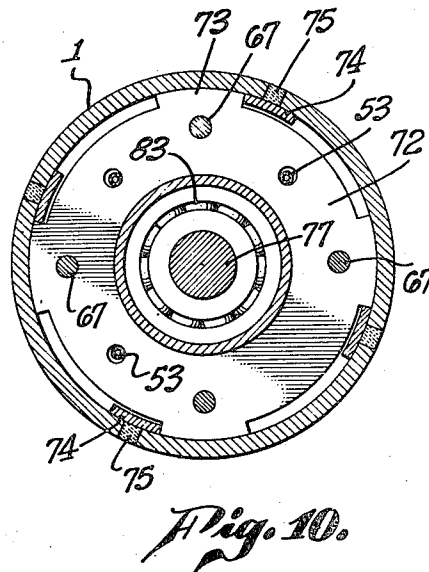
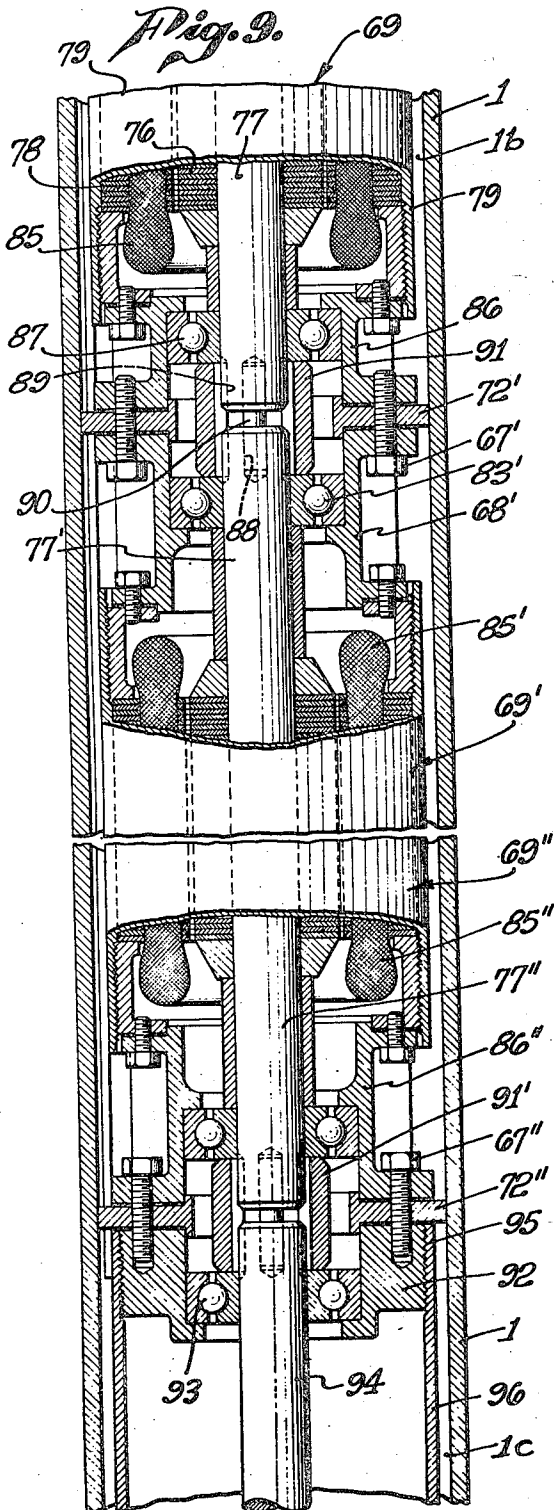
Nov. 30, 1948.

B. F. SCHMIDT
SUBMERSIBLE DOUBLE-ACTING FLUID
PISTON DEEP WELL PUMP

2,455,022

Filed Aug. 8, 1944

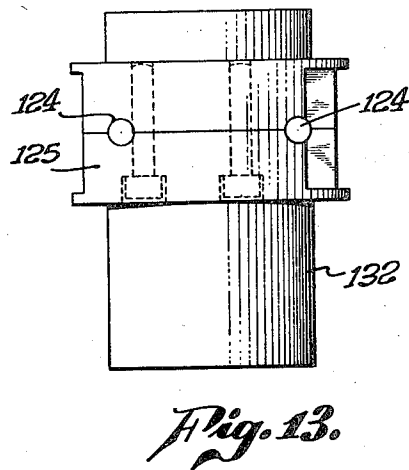
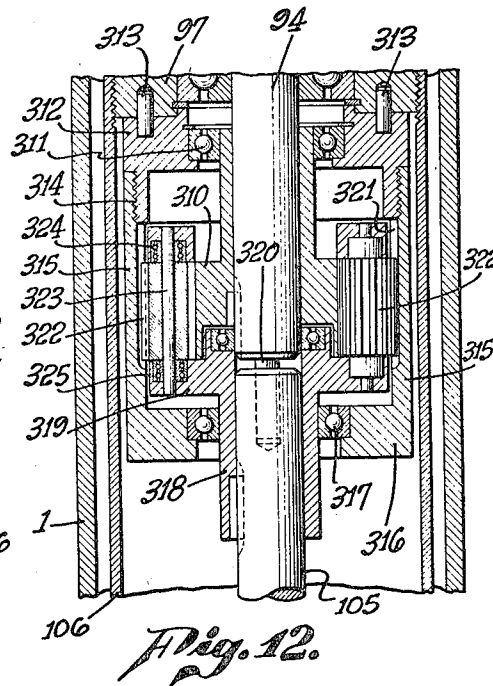
13 Sheets-Sheet 5



Inventor
BENJAMIN F. SCHMIDT,
By *Harold W. Mattingly*
Attorney

B. F. SCHMIDT
SUBMERSIBLE DOUBLE-ACTING FLUID
PISTON DEEP WELL PUMP

13 Sheets-Sheet 6

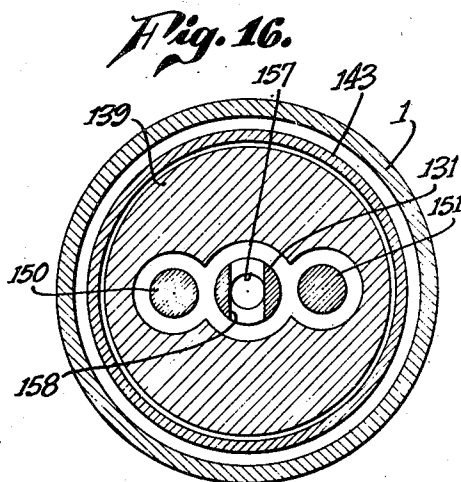


Inventor
BENJAMIN F. SCHMIDT,
By *Kenneth W. Hastings* Attorney

Filed Aug. 8, 1944

2,455,022

13 Sheets-Sheet 7



Inventor
BENJAMIN F. SCHIMDT,
By *Harold R. Dunning*
Attorney

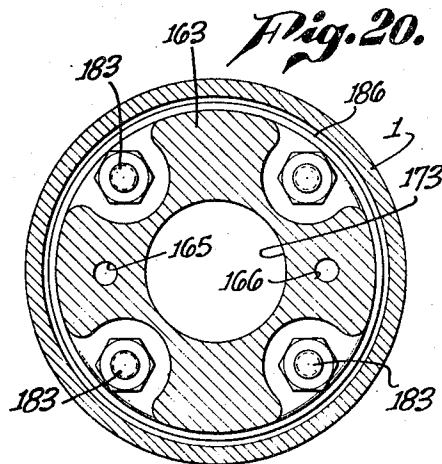
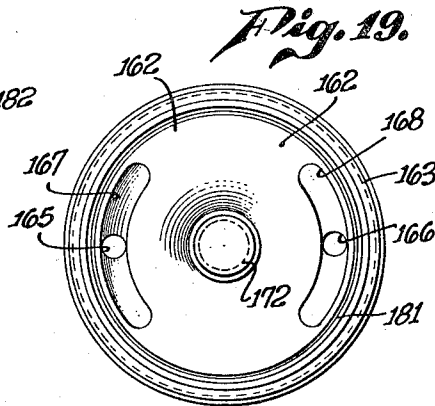
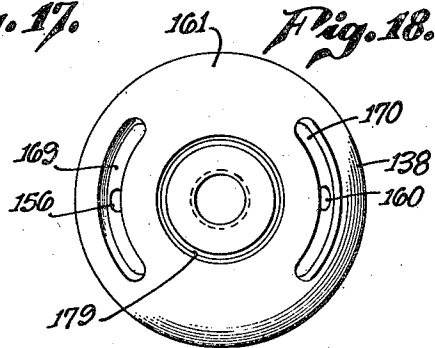
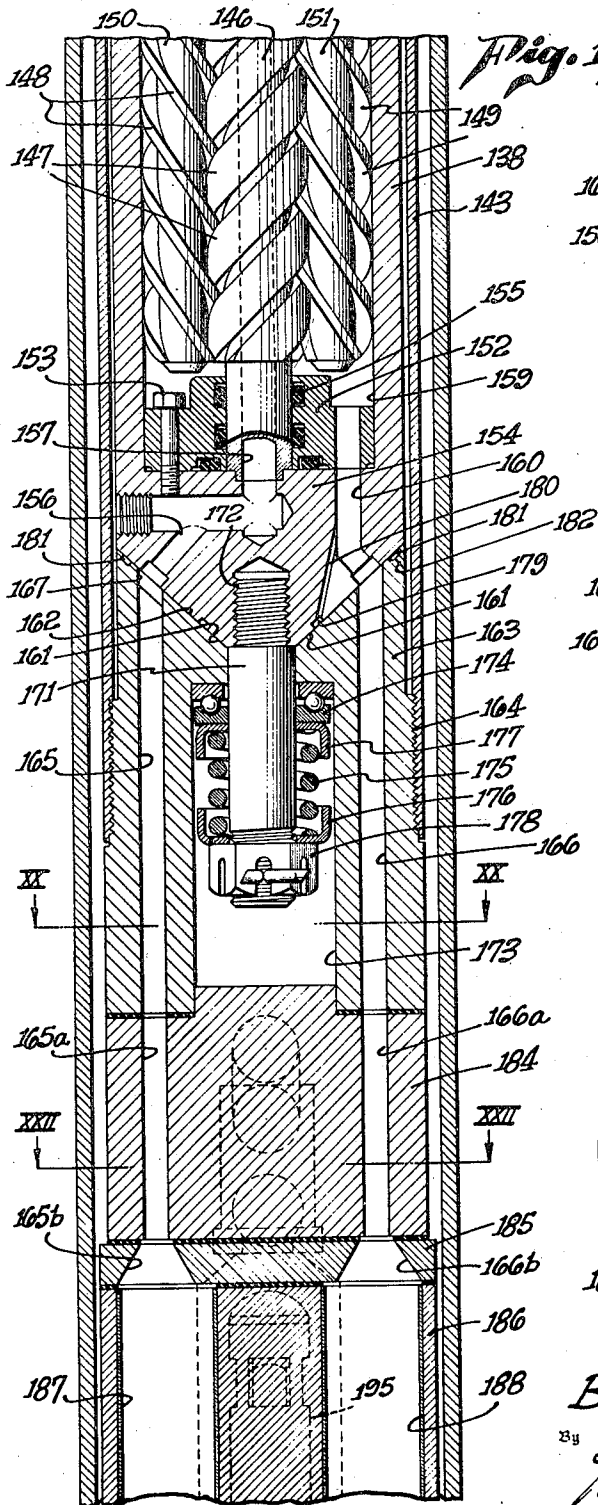
Nov. 30, 1948.

B. F. SCHMIDT
SUBMERSIBLE DOUBLE-ACTING FLUID
PISTON DEEP WELL PUMP

2,455,022

Filed Aug. 8, 1944

13 Sheets-Sheet 8



Inventor
BENJAMIN F. SCHMIDT,
By *Harold R. Manning*
Attorney

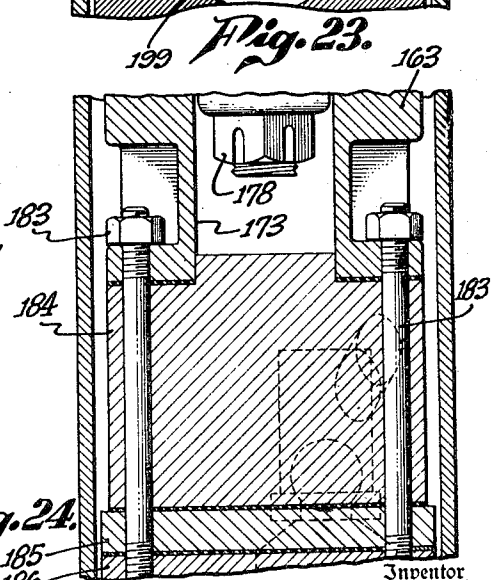
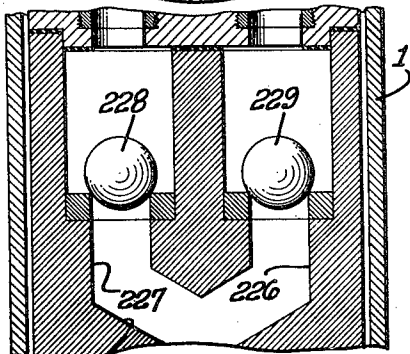
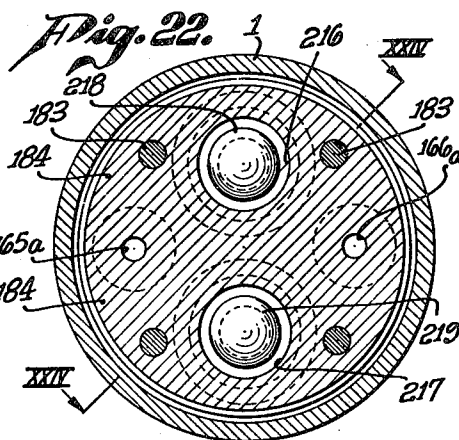
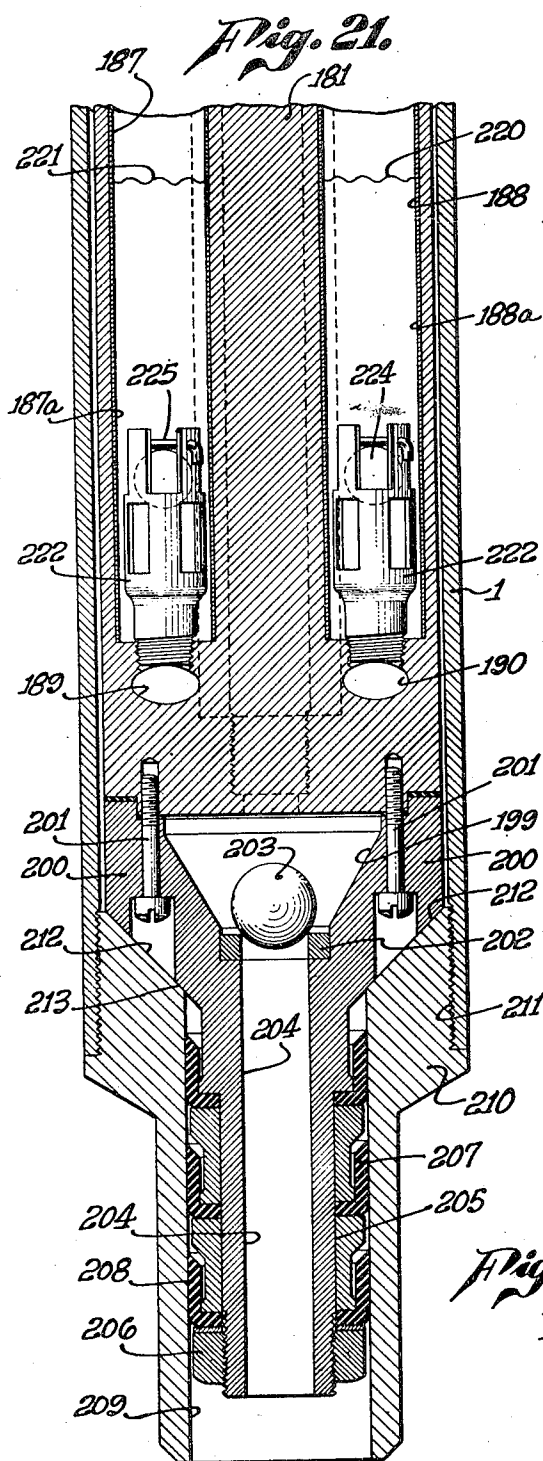
Nov. 30, 1948.

B. F. SCHMIDT
SUBMERSIBLE DOUBLE-ACTING FLUID
PISTON DEEP WELL PUMP

2,455,022

Filed Aug. 8, 1944

13 Sheets-Sheet 9



BENJAMIN F. SCHMIDT,

By

Harold R. Mattingly
Attorney

Nov. 30, 1948.

B. F. SCHMIDT
SUBMERSIBLE DOUBLE-ACTING FLUID
PISTON DEEP WELL PUMP

2,455,022

Filed Aug. 8, 1944

13 Sheets-Sheet 10

Fig. 25.

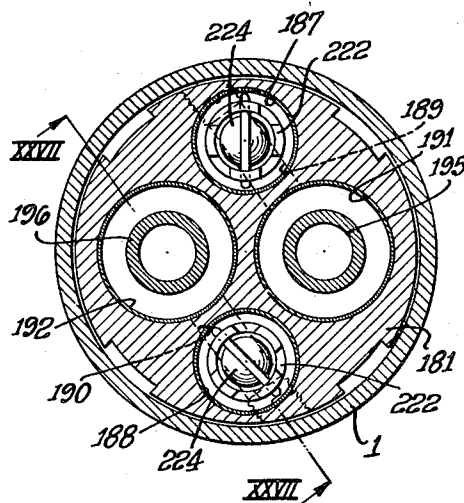
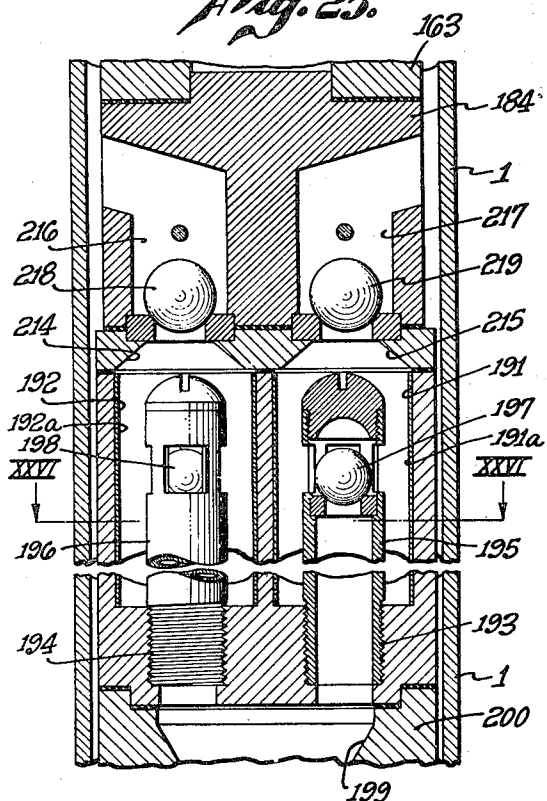
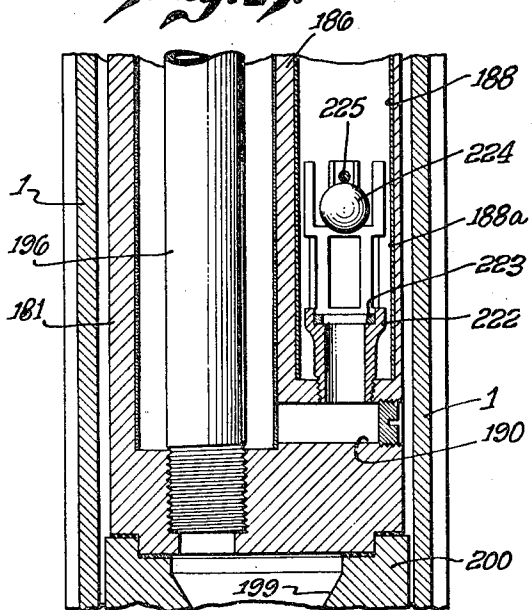


Fig. 26.

Fig. 27.



Inventor
BENJAMIN F. SCHMIDT,
By *Norval D. Mattings*
Attorney

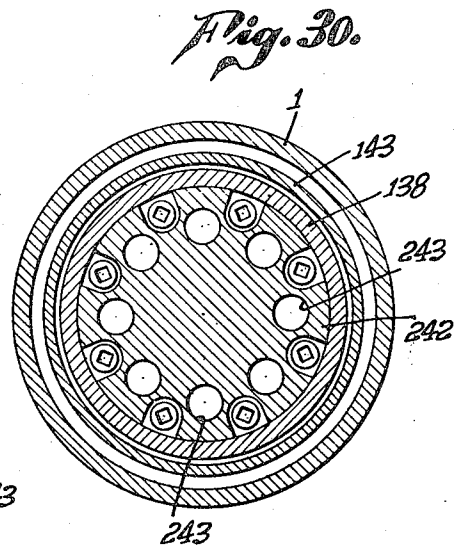
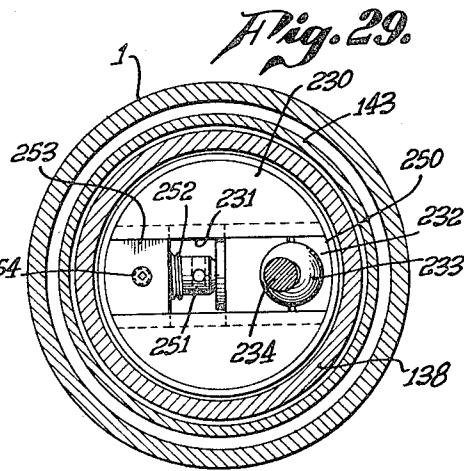
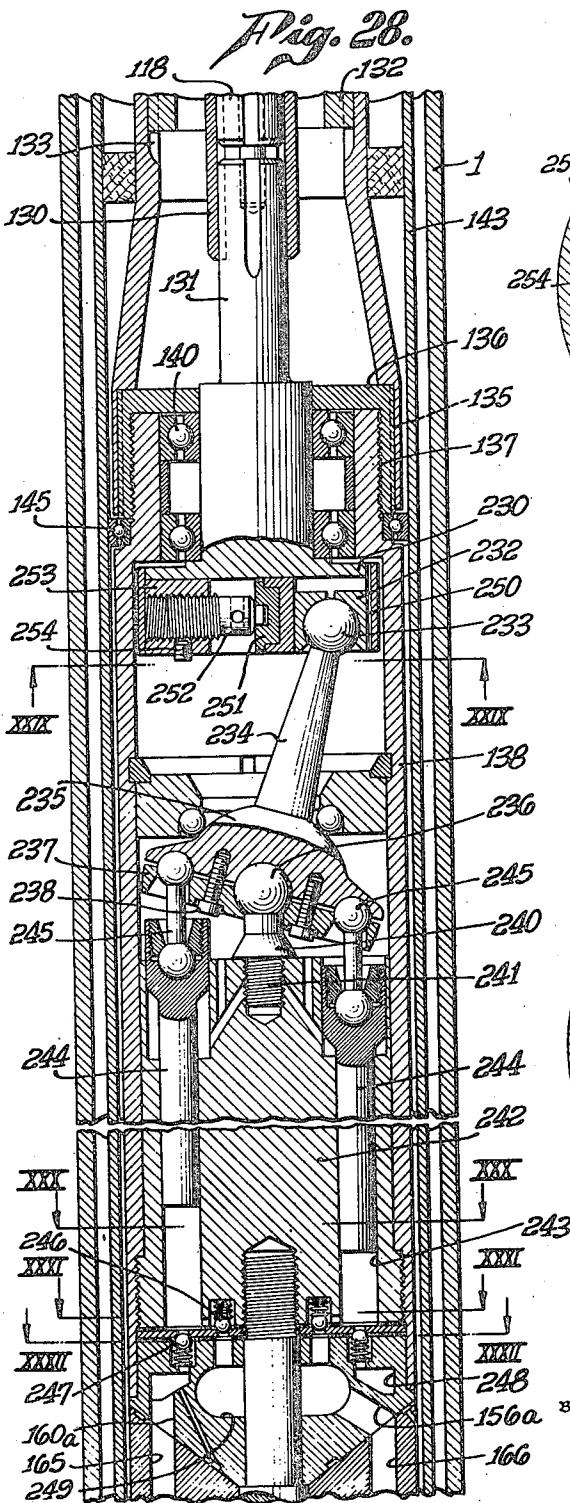
Nov. 30, 1948.

B. F. SCHMIDT
SUBMERSIBLE DOUBLE-ACTING FLUID
PISTON DEEP WELL PUMP

2,455,022

Filed Aug. 8, 1944

13 Sheets-Sheet 11



Inventor
BENJAMIN F. SCHMIDT,
Benjamin F. Schmidt
By *Harold W. Mattingly*
Attorney

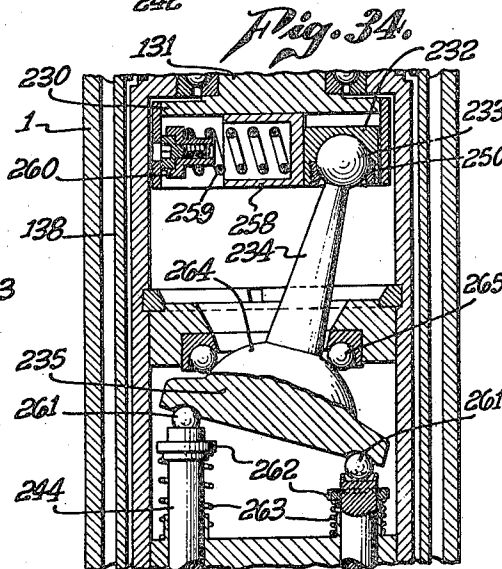
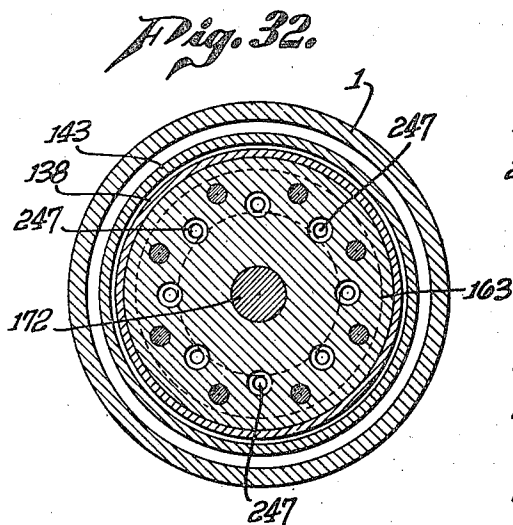
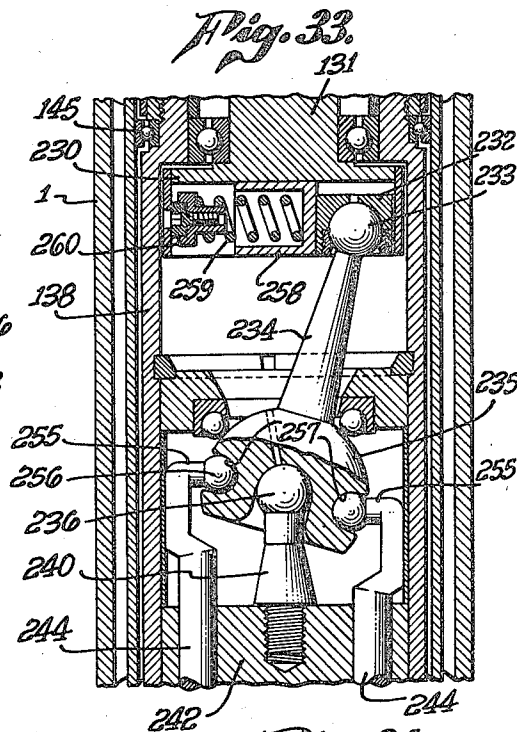
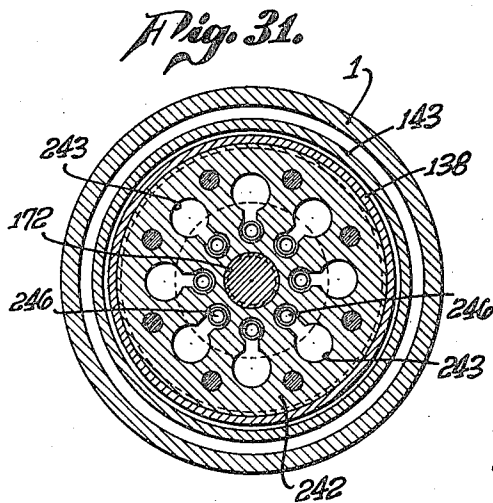
Nov. 30, 1948.

B. F. SCHMIDT
SUBMERSIBLE DOUBLE-ACTING FLUID
PISTON DEEP WELL PUMP

2,455,022

Filed Aug. 8, 1944

13 Sheets-Sheet 12



Inventor
BENJAMIN F. SCHMIDT,

By *Aswell & Smith*
Attorney

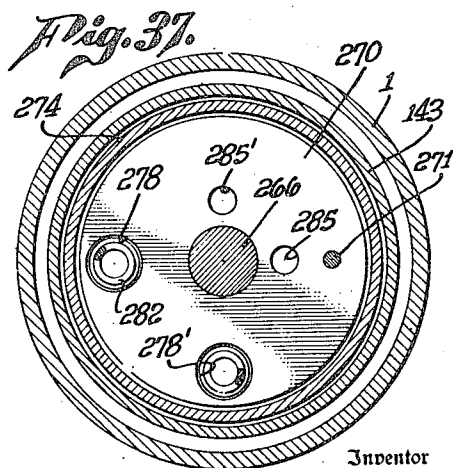
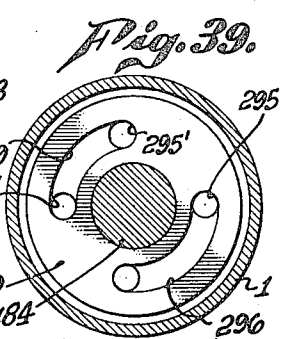
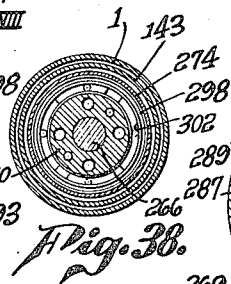
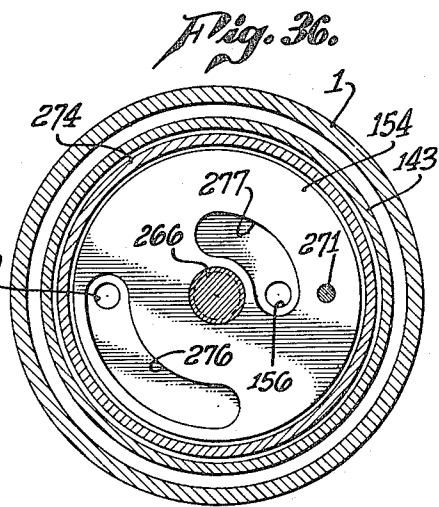
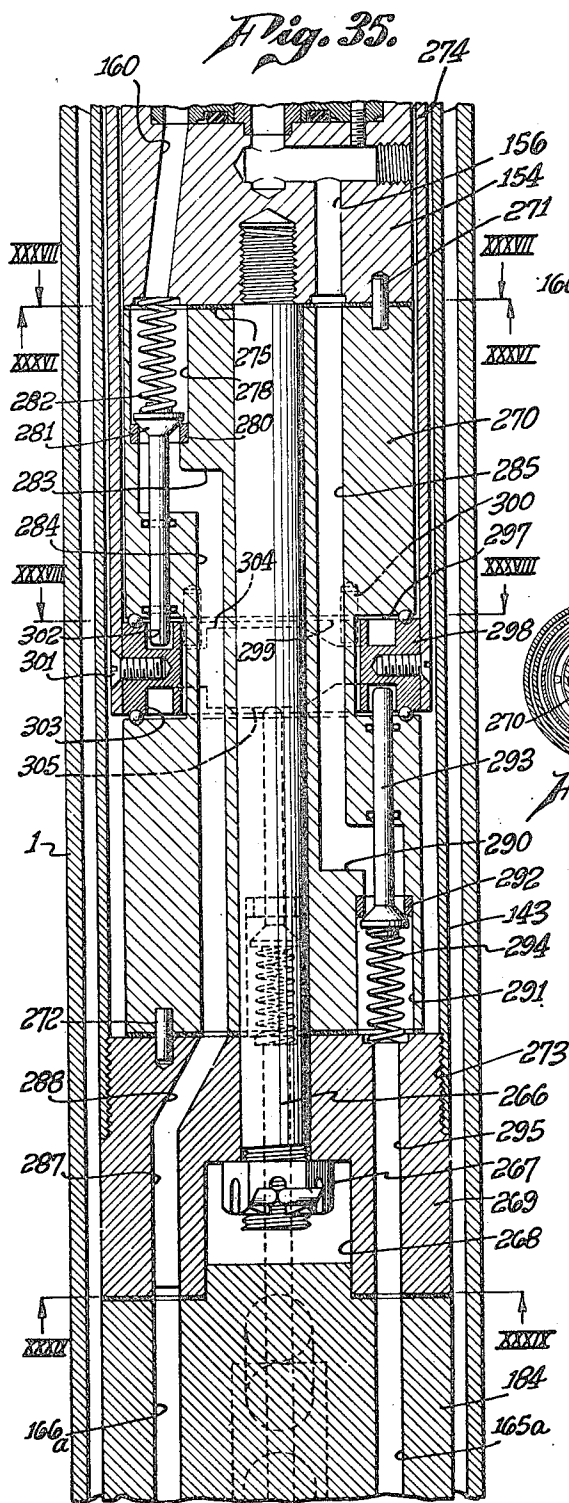
Nov. 30, 1948.

B. F. SCHMIDT
SUBMERSIBLE DOUBLE-ACTING FLUID
PISTON DEEP WELL PUMP

2,455,022

Filed Aug. 8, 1944

13 Sheets-Sheet 13



Inventor
BENJAMIN F. SCHMIDT,
Harold D. Spaulding
Attorney

UNITED STATES PATENT OFFICE

2,455,022

SUBMERSIBLE DOUBLE-ACTING FLUID
PISTON DEEP WELL PUMP

Benjamin F. Schmidt, Los Angeles, Calif.

Application August 8, 1944, Serial No. 548,518

15 Claims. (Cl. 103-46)

1

My invention relates to submersible pumps and has particular reference to an electric motor driven fluid piston reciprocating insert pump which is particularly adapted to the pumping of deep wells such as oil wells and the like.

In the oil producing art it is the common practice to pump oil from great depths by means of reciprocating pumps which are mounted deep within the well bore upon the lower end of a string of flow tubing through which the pumped oil is conveyed to the surface and to suitable storage receptacles. The conventional type of oil well pump comprises a cylinder or barrel within which is moved a reciprocating piston or plunger. The plunger is connected to a string of sucker rods which extend from the pump through the flow tubing to the ground surface where the end of the sucker rod string is connected to a walking beam or pumping jack, by means of which the string of sucker rods is reciprocated to effect a reciprocation of the pump plunger.

This type of pumping apparatus is notoriously inefficient and only a small fraction of the total energy put into the system is useful in raising oil to the ground surface. The principal losses are in the transmission of power to the pump through the sucker rod string because of the hysteresis losses attendant upon the alternate elongation and shortening of the sucker rod string and friction losses resulting from the rubbing of the sucker rod string on the flow tubing. Large losses are also encountered in the pump itself due to the high friction between the plunger and the barrel caused by the high pressure against which the pump operates.

The present invention is directed to a pump which will eliminate the energy losses mentioned above. This is accomplished by employing an electric motor drive at the pump location and transmitting the power to the pump by means of an electric cable which is extended down the well alongside of the flow tubing. Friction losses are also reduced to a minimum by employing a reciprocating pump of the fluid piston type which is driven by the electric motors.

This type of pump assembly can be successfully made only if certain design requirements are met as regards the size of the pump which must be small enough to be inserted within the well bore, the pressures under which the pump must operate which may amount to as much as 4,000 lbs. per square inch, and the temperatures encountered at the depth of the pump location which may approach the boiling point of water.

2

It is accordingly an object of my invention to provide a deep well submersible pump of the character above described which embodies certain novel features of construction permitting the above mentioned design requirements to be met.

It is an additional object of my invention to provide a pump assembly of the character above described which includes a novel mechanism for preventing the transfer of fluids between parts of the assembly which must be kept isolated from each other.

It is also an object of my invention to provide a pump assembly of the character set forth in the preceding paragraphs which is assembled as a completely self-contained unit including means for attachment to the lower end of a string of flow tubing, allowing the pump to be readily inserted within a deep well bore.

It is a still further object of my invention to provide a pump assembly of the character set forth in the preceding paragraphs in which the component parts of the assembly are removably contained within a single housing serving as a carrier for the component parts of the assembly and also as a means for supporting the component parts in their required relative operating positions.

It is additionally an object of my invention to provide in a pump assembly of the character set forth hereinbefore a novel sealing arrangement which establishes a seal between the pump assembly and the carrier when the pump assembly is inserted within the carrier.

It is also an object of my invention to provide a pump assembly of the character set forth in the preceding paragraphs which includes a means for equalizing the pressure exerted by insulating and operating fluids within the pump assembly with the static pressure exerted by the column of fluid extending from the pump to the ground surface.

Other objects and advantages of my invention will become apparent from a study of the following specification, read in connection with the accompanying drawings, wherein:

Fig. 1 is a schematic longitudinal section of a pump constructed according to the preferred embodiment of my invention and indicating figure numbers to which reference may be made for a detailed showing of the various parts of the apparatus.

Fig. 2 is a sectional view through the upper portion of the pump shown in Fig. 1 and showing the manner in which the pump assembly is

attached to the lower end of a string of flow tubing and showing the details of a sealing arrangement used to establish a fluid seal between the interior of the pump assembly and an electric cable used to carry power to the pump motors;

Fig. 2A is a cross sectional view taken substantially along the line IIA—IIA of Fig. 2 and illustrating the manner in which the electric cable is secured to the outside of the flow tubing;

Fig. 3 is a cross sectional view taken substantially along the line III—III of Fig. 2 and showing the arrangement of passages for conducting the pumped fluid to the flow tubing;

Fig. 4 is a longitudinal sectional view of that portion of the assembly which lies immediately below the portion illustrated in Fig. 2.

Fig. 5 is a side elevational view with parts broken away and parts shown in section illustrating additional details of construction of the portion shown in Fig. 2;

Fig. 6 is a longitudinal sectional view illustrating that part of the pump assembly which lies immediately below the portion shown in Fig. 4;

Fig. 7 is a cross sectional view taken substantially along the line VII—VII of Fig. 6 and illustrating the manner in which a junction box section is secured to the upper end of the first electric motor assembly;

Fig. 8 is a sectional view taken substantially along the line VIII—VIII of Fig. 7 showing the manner in which the electric motor lead wires are connected to the electric cable conductors;

Fig. 9 is a longitudinal sectional view of that part of the assembly lying immediately below the portion shown in Fig. 6;

Fig. 10 is a cross sectional view taken substantially along the line X—X of Fig. 6 and illustrating additional details of the construction of the joint between the motor section and the junction box section;

Fig. 11 is a longitudinal sectional view of that portion of the assembly which lies immediately below the portion shown in Fig. 9;

Fig. 12 is a fragmentary sectional view showing one type of reduction gearing which may be connected to the output shaft of the lower motor;

Fig. 13 is an elevational view of the reduction gearing housing shown in section in Fig. 14;

Fig. 14 is a longitudinal sectional view of that portion of the assembly which is disposed immediately below the portion shown in Fig. 11 and showing a reduction gearing arrangement which may be used as an alternative to the arrangement shown in Fig. 12;

Fig. 15 is a cross sectional view taken substantially along the line XV—XV of Fig. 14 showing additional details of construction of the reduction gearing;

Fig. 16 is a cross sectional view taken substantially along the line XVI—XVI of Fig. 14 showing the relative positions occupied by the rotating members of a rotary pump;

Fig. 17 is a longitudinal sectional view of that portion of the assembly which is disposed immediately below the portion shown in Fig. 14;

Fig. 18 is a bottom view of the lower end of the pump which is shown in the upper portion of Fig. 17;

Fig. 19 is a top view of a seat member which receives the pump assembly shown at the top of Fig. 17;

Fig. 20 is a cross sectional view taken substantially along the line XX—XX of Fig. 17 showing the form of attachment means used to se-

cure to each other the component parts shown in the lower portion of Fig. 17;

Fig. 21 is a longitudinal sectional view of that portion of the assembly which is disposed immediately below the portion shown in Fig. 17, the portion shown in Fig. 21 constituting the lowermost portion of the pump assembly;

Fig. 22 is a cross sectional view taken substantially along the line XXII—XXII of Fig. 17 and showing the arrangement of the various fluid ports and valves operating as the working valves of the pump;

Fig. 23 is a fragmentary longitudinal section of the portion of the assembly shown in Fig. 21, Fig. 23 being a section along a plane at right angles to the plane of the section of Fig. 21;

Fig. 24 is a fragmentary longitudinal section taken substantially along the plane XXIV—XXIV of Fig. 22 showing the manner in which the valve assembly is secured between the rotary pump unit and the reciprocating pump unit shown in Fig. 21;

Fig. 25 is a fragmentary sectional view of that portion of the structure shown in the lower portion of Fig. 17, Fig. 25 being taken along a plane at right angles to the plane of Fig. 17;

Fig. 26 is a cross sectional view taken substantially along the line XXVI—XXVI of Fig. 25 and showing the relative arrangement of the working valves and the pump cylinders;

Fig. 27 is a longitudinal sectional view taken substantially along the line XXVII—XXVII of Fig. 26 and showing additional details of the valving and port arrangement;

Fig. 28 is a longitudinal sectional view showing a modified form of primary pump which may be substituted for the rotary pump shown in Figs. 14 and 17;

Fig. 29 is a cross sectional view taken substantially along the line XXIX—XXIX of Fig. 28 and illustrating details of the pump driving mechanism;

Fig. 30 is a cross sectional view taken substantially along the line XXX—XXX of Fig. 28 and showing the relative arrangement of the parts of the pump;

Fig. 31 is a cross sectional view taken substantially along the line XXXI—XXXI of Fig. 28 showing the port arrangement employed at the lower end of the pump shown in Fig. 28;

Fig. 32 is a cross sectional view taken substantially along the line XXXII—XXXII of Fig. 28 and showing the relative positions occupied by the check valves shown in Fig. 28;

Fig. 33 is a fragmentary longitudinal sectional view similar to Fig. 28 but showing another modification of the invention which is particularly adapted for lower pressure operation;

Fig. 34 is a sectional view similar to Fig. 33 but showing a different modified form of the invention employing a spring return and spring supporting mechanism for the operating member of the pump;

Fig. 35 is a longitudinal sectional view showing a modified distributing valve arrangement which may be used in place of the distributing valve arrangement;

Fig. 36 is a cross sectional view taken substantially along the line XXXVI—XXXVI of Fig. 35 and illustrating details of the fluid port arrangement;

Fig. 37 is a cross sectional view taken substantially along the line XXXVII—XXXVII of Fig. 35 showing the relative locations of the fluid passages and the operating valves;

5

Fig. 38 is a cross sectional view taken substantially along the line XXXVIII—XXXVIII of Fig. 35 and illustrating the relative positions of the cam operated valves; and

Fig. 39 is a cross-sectional view taken substantially along the line XXXIX—XXXIX of Fig. 35 and showing the fluid passage and port arrangement employed at the lower end of the distributing valve.

Referring to the drawings wherein I have illustrated the preferred embodiment of my invention, I have shown the entire pump assembly as being contained within a carrier 1 which is connected as by means of a coupling unit 2 (Fig. 2) to a wishbone member 3 which is provided at its upper end with an integrally formed female coupling element 4a adapted to be threadedly secured to the lower end of a string of flow tubing 4.

The coupling member 2 is an annular member which is attached to the upper end of the carrier 1 as by means of screw threads 5, a gasket 6 being interposed between a flanged out portion 7 of the member 2 and the upper end of the carrier 1 to establish a fluid seal. The coupling member 2 is secured to the wishbone member 3 as by means of studs 8 (Fig. 5) screw threaded into the member 2 and extending upwardly through a base portion 9 of the wishbone member 3 to receive nuts 10 on their upper ends.

The wishbone member 3 is Y-shaped as shown in Figs. 1 and 5 and is cored out to provide interior passages 11 positioned to register with and communicate with similarly located passages 12 formed in the coupling member 2 to provide a fluid communication between the interior of the carrier 1 and the interior of the flow tubing 4. The Y-shaped wishbone member 3 is employed to permit an electric cable 13 to be extended from the exterior of the flow tubing 4 to within the interior of the carrier 1 without interfering with the fluid passage provided between the interior of the carrier 1 and the interior of the flow tubing 4.

As is shown in Figs. 2, 2A and 5, the electric cable 13 is preferably of the steel-clad armored type supported for the full length of the tubing 4 by means of figure eight clamps 14 shown in Figs. 2A and 5. This clamp may comprise a metal band 15 which is passed around the flow tubing 4 in one direction and between the flow tubing and the cable 13. It then encircles the cable 13 in the opposite direction and completes the encirclement of the flow tubing 4 in the first-named direction. The cross-over between the cable 13 and the tubing 4 is effected by extending the return portion 16 through a suitable aperture 17 provided in the first portion 18 (see Fig. 5). The ends of the band 15 turn outwardly to form ears 19 which may be drawn toward each other as by means of a nut and screw 20 to securely clamp the cable 13 to the tubing string 4.

The coupling member 2 supports a packing assembly for effecting a fluid seal between the coupling member 2 and the cable 13. This packing assembly may comprise an inner metal sleeve 21 which is adhesively secured to the inner rubber insulation 22 of the cable 13 as, for example, by vulcanizing, the metal protecting armor 24 of the cable 13 being terminated at a point above the sleeve 21. The sleeve 21 includes an outer cylindrical portion 25 which is received within a bore 26 formed in the center of the coupling member 2. The lower end of the sleeve 21 is threaded as

6

shown at 27 to receive a stop nut 28 adapted to bear against the under surface 29 of the coupling member 2.

The cylindrical portion 25 is preferably grooved as at 30 to receive a rubber packing ring 31. That portion of the sleeve 21 positioned above the coupling member 2 is externally threaded as shown at 32 to receive a lock nut 33 which enters a recess 34 formed in the upper surface of the coupling member 2, the lock nut 33 serving to clamp the coupling member 2 between the lock nut 33 and the stop nut 28. The exterior upper portion of the sleeve 21 is externally threaded as indicated at 35 to receive a packing gland 36 which is flanged inwardly at its upper end as at 37 to bear against an upper metal ring 38. Suitable packing material 39 is interposed between the ring 38 and the upper end of the sleeve 21. A mechanical attachment to the cable armor 24 is preferably effected by means of an upwardly extending tubular portion 40 formed on the member 38 and turned inwardly to engage the spiral recesses characterizing the armor 24.

This structure, it will be seen, establishes a fluid seal around the cable 13 between the interior of the housing 1 and the exterior thereof. The wishbone arrangement permits the cable 13 to pass from the interior of the carrier 1 to the exterior of the flow tubing 4 while permitting the passages 11 and 12 to fluidly interconnect the flow tubing with the carrier 1. These passages serve to conduct the pumped fluid from the pumping mechanism to be described hereinafter to the flow tubing and thence to the ground surface.

Since the electrical apparatus to be described hereinafter to which the cable 13 is connected must be adequately insulated and protected from the corrosive effects of sulphur, water and acids contained within the pumped fluid, this electrical apparatus is isolated from the pumped fluid by means of an inner housing 41 (Fig. 4). The housing 41 is mounted concentrically within the carrier 1 and spaced therefrom to define an annular space 1a forming a continuation of the fluid passage to the flow tubing 4. As shown in Fig. 4, the inner housing 41 comprises a section of thin wall tubing which is threadedly connected at its upper end to a coupling member 42. The coupling member is bored as at 43 to receive the cable 13 and is provided at its upper end with an upwardly extending threaded boss 44. A fluid seal between the coupling member 42 and the cable 13 is effected by means of a packing gland comprising a gland member 45 threadedly connected to the threaded boss 44 and bored as indicated at 46 to receive multiple packing rings 47. The lower end of the inner housing 41 is connected as by means of threads 48 to a junction box member 49 (Fig. 6) within which is supported a suitable terminal box 50 including connection studs 51 (Fig. 8) or an equivalent arrangement for effecting an electrical connection between conductors 52 contained within the cable 13 and lead wires 53 of an electric motor to be described hereinafter.

The space within the inner housing 41 is normally filled with an insulating oil, such as a transformer oil. It is desired to reduce to a minimum the possibility of contamination of the insulating oil contained within the inner housing 41 resulting from fluid leakage through the threads 48 or around the packing 47. I accordingly employ a novel pressure equalizing mechanism for producing within the insulating oil contained in the inner housing 41 a pressure

equal to that exerted by the fluid contained within the carrier 1 and within the annular space 1a. This mechanism includes a supporting ring 54 which is connected as by means of threads 55 to the upper end of the junction box member 49 and which is bored as at 56 to permit the cable 13 to pass into the junction box member 49.

The upper surface of the junction box member 49 and the lower surface of the supporting ring 54 are annularly grooved as shown at 57 and 58. Fluid passages 59 communicate the annular space 1a with the annular groove 57. The annular groove 58 registers with the groove 57 and is communicated by means of fluid passages 60 with the upper surface of an annular extension 61 of the ring 54.

To the outer periphery of the upstanding portion 61 there is suitably secured as by brazing or soldering an outer pressure expansible element or bellows 62. An inner like element 63 is similarly secured to the inner surface of the flange 61. The space between the bellows 62 and 63 communicates with the aforementioned passages 60. The upper ends of the bellows 62 and 63 are mutually secured to an annular ring or washer 64 (Fig. 4) mounted within the inner housing 41 and loosely received therein so as to be axially movable in response to expansion and contraction of the bellows 62 and 63.

The passages 59, 57, 58 and 60 apply to the space between the bellows 62 and 63 a fluid pressure equal to that existing within the annular space 1a, such pressure comprising the pressure exerted by the column of pumped fluid which extends to the ground surface. This pressure tends to expand the bellows 62 and 63 and so tends to displace the insulating oil contained within the inner housing 41. This process proceeds only to the point where the pressure opposing expansion of the bellows 62 and 63 exerted by the insulating oil balances the pressure exerted between the bellows by the fluid in the annular space 1a.

This mechanism thus serves to maintain within the inner housing 41 a pressure which is precisely equal to the pressure exerted in the annular space 1a. There is thus no pressure differential available to cause fluid from the annular space 1a to enter the interior of the inner housing 41 and contamination of the insulating oil is thus prevented.

The lower end of the junction member 49 is made with a reduced diameter portion 65 (Fig. 6) terminating in a flange 66 to permit attachment bolts 67 to be employed to secure to the junction member 49 an upper end bell member 68 of an electric motor indicated generally by the reference character 69. As is shown in Figs. 7 and 8, the reduced diameter portion 65 of the junction member 49 is provided with a plurality of longitudinally extending ribs 70 which are bored as indicated at 71 to receive the aforementioned motor lead wires 53. Between the flange 66 and the end bell member 68 there is placed an alignment ring 72 which serves to axially align the end bell 68 with the junction member 49 and which serves also to prevent rotation of the motor 69 within the carrier 1. To this end the member 72 is formed of a diameter substantially equal to the diameter of the flange 66 but includes radially extending lugs 73 (Fig. 10) adapted to bear against the inner surface of the carrier 1 and engage longitudinal splines 74 secured to the interior of the carrier 1 as by means of plug welds 75.

The inter-engagement of the lugs 73 and splines 74 prevents rotation of the flange 72 and the motor 69 secured thereto while permitting the motor 69 to be moved axially with respect to the carrier 1. The motor 69 may be of any suitable type, though a preference is expressed for a three-phase squirrel cage motor comprising a rotor 76 mounted upon a shaft 77 and adapted to rotate within a stator 78 contained within a motor housing 79 and secured thereto as by means of a lock ring 80 threaded into the upper end of the housing 79.

The lock ring 80 is secured as by means of bolts 81 to the aforementioned end bell member 68 which is counterbored as at 82 to receive the outer race of a ball bearing 83 by means of which the shaft 77 is supported for rotation. A spacer ring 84 may be placed between the outer race of the bearing 83 and the aforementioned aligning ring 72 to hold the ball bearing 83 against axial movement. The diameter of the motor housing 79 is preferably made somewhat smaller than the interior diameter of the carrier 1 to define between the housing 79 and the carrier 1 an annular space 1b constituting a continuation of the annular space 1a hereinbefore mentioned.

The aforementioned bores 71 which receive the motor lead wires 53 are extended through the aligning ring 72 and the end bell member 68 to form a continuous channel through which the lead wires may be extended from the motor windings shown in outline at 85. The lower end of the motor housing 79 supports a lower end bell 86 (Fig. 9) which is identical to the end bell 68 previously described. The bell end 86 supports a lower ball bearing 87 which journals the lower end of the motor shaft 77.

It is many times advisable to use a plurality of motors 69 and I have, therefore, shown in Fig. 9 the manner in which a second motor 69' may be secured to and drivably connected to the upper motor 69. The motor end bell 86 of the motor 69 is connected to the upper end bell 68' of the motor 69', it being understood that the motor 69' is identical in all respects with the motor 69. An alignment ring 72' identical to the alignment ring 72 previously described is interposed between the end bells 86 and 68' and the attachment of these end bells to each other is effected by means of a plurality of bolts 67'.

The shaft 77 of the motor 69 and the shaft 77' of the motor 69' are coaxially bored at their adjacent ends as shown at 88 and 89 to receive an aligning dowel pin 90. For convenience in assembly, the dowel pin 90 is preferably more or less permanently secured to one of the shafts as by being pressed into the counterbore in that shaft and removably associated with the other shaft as by a sliding fit between the pin and the bore in such other shaft.

The lower end of the shaft 77 and the upper end of the shaft 77' are preferably machined to a hexagonal cross section or splined to permit a similarly contoured coupling member 91 to effect a driving interconnection between the two shafts. The length of the coupling 91 is preferably so selected as to bear against the inner races of the ball bearings 87 and 83' and hold those inner races in their proper location in the same manner as the spacing sleeve 84 previously described. The lower end of the lowermost motor identified generally by the reference character 69'' includes a lower end bell 86'' which is secured as by means of bolts 67'' to a lower supporting ring 92, a lower aligning ring 72'' being inter-

posed between the end bell 68' and the supporting ring 92.

The supporting ring 92 supports a ball bearing 93 which journals the upper end of a jack shaft 94. The jack shaft 94 is drivably connected to the lower end of the lowermost motor shaft 77' by means of a coupling assembly 91' identical with the coupling 91 previously described. The supporting ring 92 is also threadedly attached as by means of threads 95 to a supporting sleeve 96 which is mounted concentrically within the carrier 1 and spaced therefrom to define an annular space 1c comprising a continuation of the annular spaces 1a and 1b.

The lower end of the supporting sleeve 96 is secured to a coupling and sealing member 97 (Fig. 11) serving as a closure for the lower end of the space within the sleeve 96. This space, it will be appreciated, is filled with the insulating oil which fills likewise all of the spaces within the motors 69—69'' and the junction member 49. Since the shaft 94 extends through the member 97, a fluid seal must be effected between the shaft and the member 97. Accordingly, the member 97 is bored as indicated at 98 to receive a plurality of spring-pressed packing elements 99 of conventional construction. An auxiliary seal or wiper comprising an annular ring of leather or synthetic rubber 100 may be secured as indicated in Fig. 11 to the upper surface of the member 97.

The lower end of the shaft 94 is also journaled in the member 97 as by means of a ball bearing 101 mounted within a suitable counterbore 102 and held in position as by means of a snap ring 103. The shaft 94 is drivably connected as by means of a splined connection 104 to a shaft extension 105 mounted within an inner housing 106 secured as by means of threads 107 to the lower end of the coupling member 97.

As shown in Fig. 14, the inner housing 106 is mounted concentrically within the carrier 1 and spaced therefrom to define an annular passage 1d forming a continuation of the annular passages 1a—1c. The housing 106 is connected as by means of threads 108 to a supporting member 109. The supporting member 109 is similar to the upper end of the member 49 shown in Fig. 6 and mounts a ring member 110 to which is secured concentrically disposed bellows 111 and 112. The bellows 111 and 112 are secured at their upper ends to a washer 113 (Fig. 11) mounted within the inner housing 106. A fluid passage arrangement identical to that used in connection with the bellows 62 and 63 of Fig. 6 is employed to apply to the space between the bellows 111 and 112 the same fluid pressure as exists within the passage 1d.

The space within the inner housing 106 is intended to be filled with a fluid lubricant and the bellows 111 and 112 operate in a manner identical to that described in connection with the bellows 62 and 63 to maintain within the lubricant filling the housing 106 the same pressure as exists within the annular space 1d.

The coupling member 109 includes a downwardly depending flange portion 114 which is internally threaded to receive an upwardly extending flange portion 115 of the reduction gear housing 116. The reduction gearing which is contained within the housing 116 is indicated generally by the reference character 117 and comprises the subject matter of my copending application Serial No. 548,517, filed August 8, 1944, and entitled Reduction gearing. Briefly stated, it includes a central shaft 118 connected as by a

splined coupling 119 to the lower end of the shaft extension 105 and having formed thereon a worm 120. The shaft 118 is journaled as by means of bearings 121 and 122. The worm 120 meshes with a pair of worm gears 123 (Fig. 15) mounted upon cross shafts 124 suitably supported within a carrier 125, the carrier 125 in turn being journaled as by means of bearings 126 and 127 within the gear housing 116.

Special spiral gears 128 are formed integrally with the worm gears 123 and mesh with gear teeth 129 formed on the interior of the gear housing 116. The action of the mechanism is such that when the worm 120 is rotated, the carrier 125 is caused to rotate at a slow speed within the gear housing 116. The reduction gearing is shown in elevation in Fig. 13.

The lower end of the shaft 118 is coupled as by means of a splined coupling 130 to the upper end of a pump shaft 131, and a downwardly depending cylindrical portion 132 of the slowly rotating carrier 125 is secured as by means of a splined connection 133 to the upper portion of a coupling member 134 which is secured at its lower end as by means of splines 135 to an upper closure member 136 threadedly connected as at 137 to the upper end of a pump housing 138.

The pump shaft is journaled for rotation relative to the pump housing 138 as by means of a supporting member 139 which mounts ball or other suitable anti-friction bearings 140 and 141. A fluid seal around the shaft 131 is obtained by means of packing elements 142.

Attention is directed to the mounting of the ball bearing 140 within the space provided in the supporting member 139 within which the packing elements 142 are placed. This arrangement serves to hold the shaft 131 in a fixed position within the packing 142 and maintains permanent fixed alignment between the shaft and the packing. This avoids the difficulties often encountered with packed joints in which the packing crowds to one side and forces the shaft out of line, increasing the friction and causing rapid wear of the packing and rapid wear of the shaft.

Since the pump housing 138 is slowly rotated by means of the reduction gearing 117 acting through the coupling member 134, it is desirable to reduce to a minimum the friction between the pump housing 138 and an outer stationary housing 143 which is tubular in form and secured as by means of threads 144 to the stationary gear housing 116. This may be accomplished by encircling the pump housing 138 with an anti-friction bearing 145 as is shown in Fig. 14.

The pump shaft 131 and the pump housing 138 may comprise parts of a rotary pump of any suitable design. I have shown, by way of illustration, a well known type of rotary pump which includes a power rotor 146 (Fig. 17) which is formed integrally or suitably secured to the pump shaft 131. This rotor has formed thereon a plurality of spiral vanes 147 which mesh with oppositely inclined vanes 148 and 149 formed on a pair of idler rotors 150 and 151 disposed on opposite sides of the power rotor 146 and suitably journaled for rotation. The lower end of the power rotor 146 is supported for rotation in a backing member 152 suitably secured as by means of bolts 153 to a lower head member 154 which may, as indicated, be formed integrally with the pump housing 138. A fluid seal is established between the rotor 146 and the stationary sealing member 152 by any suitable means such as the packing rings shown at 155.

The head member 154 is the lower end of the pump proper and includes a suction passage 156 communicating with an inlet bore 157 formed in the power rotor 146. The bore 157 extends the full length of the power rotor 146 and communicates with a transverse passage indicated by dotted lines 158 in Fig. 14 so that fluid drawn in through the passage 158 will be discharged into the interior of the pump 138 at the upper end of the rotors 146, 150 and 151. As the rotors are rotated, this fluid is carried downwardly through the chambers in which the rotors rotate, is collected in an annular space 159 at the lower end of the rotors and discharged through a discharge passage 160 formed in the lower head member 154.

According to my invention, the lower head member 154 serves in a dual capacity acting also as the rotary member of a distributing valve. To this end the head member is formed with a cone-shaped external contour as shown at 161 to be received within a similarly cone-shaped depression 162 formed in a stationary valve member 163. The valve member 163 is secured as by means of threads 164 to the lower end of the stationary housing 143. By virtue of this threaded interconnection, the stationary valve member 163 serves as a bottom closure for the housing 143.

That portion of the apparatus just described which lies below the coupling and sealing member 97 and above the stationary valve member 163 is filled with a lubricating oil chosen for its lubricating properties to reduce to a minimum the friction losses and wear in the various bearings, gears and the like included in the stated portion of the device. The rotary pump which is driven by the drive shaft 131 is intended to pump what may be termed an "operating fluid." This fluid may be oil of the same character as that used for lubrication or it may comprise any one of a number of suitable liquids. This oil is contained in the space and passages to be described hereinafter and situated below the stationary valve member 163.

The stationary valve member 163 includes a pair of diametrically opposed vertically extending passages 165 and 166. As is shown in Fig. 19, these passages terminate in arcuate grooves 167 and 168 each extending approximately one-quarter of a circle about the inclined surface 162. They are positioned to register with similarly disposed arcuate channels 169 and 170 (Fig. 18) formed on the under surface of the head member 154. The inlet passage 156 communicates with one of the channels 169 and the discharge passage 160 communicates with the other channel 170. The surfaces 161 and 162 are in abutting relation to each other and serve as a fluid seal to prevent leakage of fluid from the aforementioned passages while permitting the head member 154 to be slowly rotated with respect to the stationary valve member 163.

The head member 154 is preferably guided during its rotation and held in pressure contact with the female cone surface 162 by means of a stub shaft 171 which is secured as by means of threads 172 to the lower end of the head member 154. The shaft 171 extends into a counterbore 173 formed in the under side of the stationary valve member 163 and receives a ball thrust bearing 174. A compression spring 175 confined in suitable retainers 176 and 177 is interposed between the thrust bearing 174 and a nut 178 screw threaded onto the lower end of the stub shaft 171. Rotation of the nut 178 may be used to effect an

adjustment of the downward force exerted on the shaft 171 by the spring 175.

In many cases the force urging the head member 154 onto the conical seating surface 162 will be unduly large despite any adjustment that may be made on the spring 175. It is, therefore, desired to balance a part of the weight resting on the surface 162 by hydraulic means and to this end I have formed in the under surface of the head member 154 a circular groove 179 which is connected by means of a port 180 to the pump discharge port 160. A similar circular groove 181 formed in the seating surface 162 may be communicated by means of a small passage 182 with the annular space lying between the pump housing 138 and the stationary housing 173. The fluid pressures which are conveyed to the grooves 179 and 181 exert an upward force on the head member 154 balancing at least a part of the weight of the structure resting on the head member 154. It is preferred to adjust the area of the grooves 167, 168, 169, 170, 179 and 181 to provide a separating force due to fluid pressure which exceeds the weight resting on the head member 154. The spring 175 and associated mechanism may then be used to obtain the desired contact pressure between the head member 154 and the stationary valve member 163.

The stationary valve member 163 is secured as by means of bolts 183 (Figs. 20 and 24) to a valve block 184 (Figs. 17 and 24), a distributing member 185 and a pump housing 186 (Fig. 21). Through the enumerated members, there are formed continuations 166a, 166b and 165a, 165b of the aforementioned fluid passages 165 and 166. The passages 165b and 166b communicate with similar vertically extending bores 187 and 188 formed in the pump housing 186. The bores 187 and 188 are extended nearly to the bottom of the housing 186 where they are communicated by means of diagonally extended passages 189 and 190 (see Figs. 26 and 27) with corresponding vertical bores 191 and 192 spaced circumferentially between the bores 187 and 188.

The passages 191 and 192 have reduced diameter portions at their lower ends which are internally threaded as indicated at 193 and 194 to receive the threaded lower ends of a pair of tubular stand pipes 195 and 196. The stand pipes are each fitted at their upper end with ball standing valves 197 and 198 arranged to discharge fluid from within the tubes 195 and 196 into the chambers 191 and 192 near the upper ends thereof. The tubes 195 and 196 communicate with a common inlet passage 199 (Figs. 21 and 25) formed in a foot valve member 200 which may be secured to the pump housing 186 in any suitable manner as by means of screws or bolts 201. The foot valve member 200 mounts a seating ring 202 upon which is rested a ball foot valve 203. The member 200 is bored as indicated at 204 to communicate with the ring valve seat 202 and is provided with a reduced diameter external portion 205 which is threaded at its lower end to receive a lock nut 206 used to hold on the tubular portion 205 a series of pump leathers or packing cups 207 and 208. These cups are normally received within a bore 209 formed in the lower end of a lower closure member 210 which is secured as by means of threads 211 to the central lower end of the housing 1. The upper surface of the member 210 is, by preference, tapered as shown at 212 to receive a similarly tapered lower surface 213 formed on the foot valve member 200, the mating surfaces 212 and 213 serving to hold the assembly

in a concentrically aligned position with respect to the housing 1.

The aforementioned chambers 191 and 192 communicate at their upper ends as by means of passages 214 and 215 formed in the distributing member 185 with discharge ports 216 and 217 formed in the valve block 184 (Fig. 25). The passages 216 and 217 each communicate as shown in Fig. 25 with the annular space disposed immediately within the outer housing 1. This space, it will be recalled, is continuous the full length of the instrument and communicates with the discharge passages 11 formed in the upper wishbone member 3, and it is by this means communicated with the tubing string 4. The passages 216 and 217 are each fitted with ball check valves 218 and 219 serving to supplement the standing valves 197 and 198 previously described.

The bores 187 and 188 and the bores 191 and 192 each contain a heavy fluid such as mercury. The level of the fluid in one of its possible positions is indicated by the wavy lines 220 and 221 in Fig. 21.

The operation of the device may best be understood by assuming the parts to be in the positions shown in Figs. 17, 21 and 25. With the rotary pump in operation, fluid is drawn in through the passage 156 and discharged through the passage 160. This fluid under pressure flows into the passage 166, through the mating of the arcuate channels 168 and 170 and flows downwardly through the passages 166a and 166b into the pump chamber 188. Fluid pressure is thus exerted on the surface 220 of the mercury column contained in the passage 188 and so forces the mercury through the diagonal passage 190 into the chamber 192, causing the mercury to rise in the chamber 192 and displace from that chamber any fluid which may be contained therein and resting upon the surface of the mercury. This fluid cannot flow downwardly through the stand pipe 196 because the standing valve 198 is closed. It therefore flows upwardly past the check valve 218, through the passage 216 and into the annular space immediately within the housing 1, whence it is conducted upwardly to the wishbone 3 and into the tubing string 4.

At the same time the fluid which flows through the suction port 156 into the rotary pump 147 is withdrawn from the passage 165 and the passages 165a and 165b to produce a materially reduced pressure within the pump cylinder 187. The reduction in pressure on the surface 221 of the mercury column contained therein serves to withdraw mercury through the diagonal passage 189 from the bore 191. Since the top of the bore 191 is closed by the check valve 219, the void tending to be produced by the withdrawal of the mercury is filled by contact oil or pumped fluid which flows upwardly through the passage 204 past the foot valve 203 and into the space 199 from whence it is conducted upwardly through the stand pipe 195 and past the standing valve 197.

During the time this operation is taking place, the head member 154 which is secured to the pump housing 138 is slowly rotating with respect to the stationary valve member 163 by virtue of the reduction gearing 117. The speed of this rotation is adjusted to interchange the passages 156 and 160 with respect to the passages 165 and 166 before all of the mercury has been displaced from the bores in which it is contained.

It will be noted by referring to Figs. 18 and 19 that during approximately one-half of a revolution, the fluid connections effected by the valve

154, 163 are as shown in Fig. 17. After a one-half revolution of the valve member 154, the passage 156 is communicated with the passage 166 and the passage 160 is communicated with the passage 165. The flow of oil or other working fluid caused by the operation of the rotary pump is thus reversed in the passages 165 and 166 with the result that the rise of mercury in the chamber 188 causes contact oil to be drawn in through the stand pipe 196, while displacement of mercury from the chamber 187 causes the contact oil previously drawn into the chamber 191 to be discharged past the check valve 219 and into the annular space disposed immediately within the housing 1.

I have found that it is possible to maintain a clean surface on the cylinder bores 187—188, 191 and 192 by silver plating the surface of these bores as indicated at 187a, 188a, 191a and 192a. The mercury will readily form an amalgam with the silver and thus continually "wets" the surface to obtain a scraping or cleaning action, preventing the adherence of dirt particles to the surface of the cylinder walls and thus avoiding the development of encrustations on the walls of the cylinders.

I have shown in Figs. 21 and 27 a safety valve arrangement which is employed by preference to prevent inadvertent loss of the mercury contained within the pump chambers. This valve comprises a cage 222 suitably secured in the lower end of the bore 187, 188 and defining a seat 223 for a ball valve 224. The ball valve 224 is preferably formed of steel or other metal having a density less than that of the mercury or fluid used as a fluid piston. The ball valve, therefore, consequently tends to rise to the surface of the fluid piston. This is prevented by means of a cross pin 225 extended across the cage above the valve in a position to arrest upward movement of the valve. Should, however, the level of the mercury be depressed to the level of the ball 224, the ball will follow the mercury level down until the ball seats on the seat 223, closing off the passage 190 and preventing further discharge of mercury.

If desired, the single foot valve member 203 may be replaced by a double valve arrangement of the character shown in Fig. 23 wherein the passage 199 is divided into two passages 226 and 227 each closed by a separate ball valve member 228, 229.

It is desired at this point to direct attention to the fact that the connection of the above described assembly to the outer housing 1 is effected at the upper end by the female member or upper closure 2 and at the lower end by the packing elements 207 and 208 which are received within the lower restricted bore 209. With this arrangement, parting the screw threaded joint 5 (Fig. 1) allows the entire assembly to be withdrawn as a unit from the housing 1, the cup packing members 207 and 208 merely sliding upwardly through the bore 209. Installation is accomplished by a reverse operation, the cup form of the packing members 207 and 208 permitting that assembly to be readily slipped downwardly through the bore 209. Making up the threaded connection 5 completes the assembly.

I have illustrated in Figs. 28-34 alternative rotary pump constructions which may be employed if desired instead of the rotary pump shown in Figs. 14 and 17. Reference to Fig. 28 will show that in the modification illustrated therein, the pump shaft 131 is terminated in a rotating disk 230. This disk is provided with a transverse slot 231 within which is mounted a

drive member 232. The drive member is formed with a partially spherical socket to receive a ball 233 formed upon the upper end of an operating arm 234. The operating arm 234 is attached to a pump operator 235 which is mounted for universal movement upon a ball 236 received within a suitable socket formed in the operator 235 and retained therein as by means of a lock ring 237 secured to the operator 235 as by means of bolts 238. The ball 236 may be formed integrally with a pillar 240 mounted as by means of threads 241 upon a pump body 242. The pump body 242 is provided with a plurality of cylinder bores 243 (Fig. 30) within which are reciprocally mounted pump plungers 244. The pump plungers 244 are drivably connected as by means of ball and links 245 to the pump operator 235.

With the parts disposed as shown in Fig. 28, rotation of the drive disk 230 through rotation of the shaft 131 will cause the ball 233 to describe a circle about the axis of rotation of the shaft 13. This causes the operator 235 to wobble and in so doing progressively raises and lowers the pistons 244 in their bores 243. The lower ends of these bores are communicated with inlet valves 246 and discharge valves 247 which are manifolded as indicated at 248 and 249 to communicate, respectively, with an inlet passage 156a and a discharge passage 160a comprising counterparts of the previously described inlet and discharge passages 156 and 160, respectively.

The lower end of the pump just described is made cone-shaped in the same manner as the rotary pump shown in Fig. 17 and operates in the same manner as a rotary valve to distribute the flow between the two vertical passages 165 and 166 in the manner previously described.

Referring again to the rotating disk 230 and the slot 231 which is formed therein, it will be noted that the socket member 232 which is mounted within the slot 231 and which carries the ball 233 is slidable in the slot 231. It is arranged to be held in a selected position by means of shims 250 placed on one side of the socket member 232 and a pusher member 251 placed on the other side of the socket member 232. The pusher member 251 is engaged by an adjusting screw 252 which is threaded in a block 253 suitably secured in position as shown in the slot 231. Inadvertent rotation of the adjusting screw 252 may be prevented by means of a lock screw 254.

It will be noted that by withdrawing the screw 252, the block 251 and the socket member 232 may be slid left as viewed in Fig. 28, allowing thicker or additional shims 250 to be placed to the right of the socket members 232. This reduces the eccentricity and defines a smaller diameter circle of movement of the bore member 233 and so correspondingly reduces the length of the stroke which is imparted to the pump plungers 244. Thus the production capacity of the pump may be changed, as desired, by means of the relatively simple adjustment just described.

A modification of the multiple plunger pump just described is shown in Fig. 34 wherein the connecting links 245 are omitted and the upper ends of the plungers 244 are carried above the surface of the cylinder block 242 and turned inwardly as shown at 255 to support ball ends 256 received in suitable spherical depressions 257 formed in the sides of the pump operator 235. Also, the adjustment screw 252 and associated mechanism described in connection with Fig. 28 has been replaced by means of a pusher member 258 which is spring urged to the right as viewed

in Fig. 33 by means of a compression spring 259 interposed between the pusher member 258 and an adjustment device 260.

With this construction it will be seen that if the force required to reciprocate the pistons 244 should for any reason rise above the force which may be applied through the spring 259, the spring will be compressed and the pump operating arm 234 will move inwardly toward the center of rotation of the shaft 131 to reduce the effective length of stroke of the plungers 244. This provides an automatic return adjustment which may vary as changing operating conditions requires.

In the forms of the invention shown in both of Figs. 28 and 33, the pump plungers 244 are so connected to the pump operator 235 as to be positively drawn upwards. In the modification of the invention which is shown in Fig. 34, the pump operator 235 is arranged to exert a downward pressure only upon the plungers 244. This is accomplished by forming the ends of the plungers with a small cup-like depression within which a ball member 261 may be received. The ball member 261 is likewise received in a cup-like depression formed in the under surface of the operator 235. A suitable abutment flange 262 is secured to the upper ends of the plungers 244 and a compression spring 263 is confined between the abutment member 262 and the upper surface of the pump body 242.

It will be noted that this modification of the invention permits the elimination of the supporting column 240 as the upward force exerted by the plurality of compression springs 263 serves to hold the pump operator 235 in an upper position in which a spherical outer surface 264 thereof is engaged by an annular ball bearing 265 suitably mounted upon the pump housing 138.

I have shown in Figs. 35-39 a valve arrangement which may be used in place of the rotary valve 154, 163 previously described. In Fig. 35 a lower pump head is shown at 154. A longitudinally extending threaded stud 266 depends from the lower head 154 and carries a clamping nut 267 which is received within a counterbore 268 formed in the under side of a transition block 269 positioned immediately above the valve block 164 (shown also in Figs. 17 and 24). Between the transition piece 269 and the lower pump head 154 there is positioned a valve block 270 which is held against rotation relative to the pump head 154 and the transition piece 269 as by means of dowel pins 271 and 272. The transition piece 269 is secured as by means of threads 273 to the lower end of the outer stationary housing 143 (see Fig. 14). This serves to hold the transition piece 269 against rotation and because of the dowel pins 271 and 272, it serves also to hold the valve block 270 and the pump head 154 against rotation.

It will be recalled that in connection with Fig. 14 the coupling member 134, which was slowly rotated by the reduction gear 117, was described as being drivably connected to the rotary pump. In the modification of the invention shown in Fig. 35 this connection is not made but instead the coupling member 134 is connected to a drive sleeve 274 which is disposed in the annular space between the inner housing 140 and the valve block 270. The drive sleeve 274 is employed in a manner hereinafter described to operate valves mounted within the valve block 270.

The head member 154 includes inlet and exhaust ports 156 and 160, respectively. These ports communicate with a lower plane surface

275 which is by preference formed on the under side of the lower head member 154 instead of the conical under surface 161 shown in Fig. 17. The passages 156 and 160 are so positioned as to intersect the plane surface 275 at materially different radii as is clearly shown in Figs. 35 and 36. The under surface 275 is provided with a pair of arcuate grooves 276 and 277, each extending substantially a quarter of a revolution, as is shown in Fig. 36. The groove 276 serves to communicate the discharge passage 160 with a pair of valve chambers 278 and 278' spaced at equal radii and a quarter of a circle apart, as is shown in Fig. 37. These chambers are similar and are each formed like the chamber 278 shown in Fig. 35. These chambers each mount valve seats 280 upon which poppet valves 281 are rested. Compression springs 282 urge the poppet valves 281 to closed positions. Below the seats 280 the fluid passages are offset as shown at 283 and continue in a downward direction in the form of passages 284 situated at the same radius as a pair of passages 285 and 285' which are also formed in the valve block 270 in positions to communicate with the inner arcuate slot 277.

The passages 284 each continue downwardly through the valve block 270 and register with continuation passages 287 and 287' formed in the transition piece 269. The passages 287 and 287' are offset as shown at 288 to dispose the lower end of the passages 287 and 287' at the same radius as the passage 166a formed in the valve block 184 (see Fig. 17). The interconnected passages 285 and 285' previously mentioned are, as shown in Fig. 35, offset outwardly as indicated at 290 to communicate with a pair of valve chambers 291 situated at the same radius as the chambers 278 and 278' previously mentioned. Valve seats 292 receive body valves 293 which are disposed below the seats 292. These valves are urged upwardly to their closed positions by means of compression springs 294 interposed between the valves 293 and the upper surface of the transition piece 269. The transition piece 269 includes a pair of passages 295 and 295' positioned to register with each of the valve chambers 291.

The passages 295 and 295' are positioned at the same radius as the passages 287 and 287' previously mentioned. A pair of arcuate slots 289 and 296 are formed in the under surface of the transition piece 269, the former serving to interconnect the passages 289 and 295' and the latter serving to interconnect passages 295 and 287'. It will be seen that this construction provides a fluid connection between the passage 165a in the valve block 184 (Fig. 17) and the pair of passages 287 and 295' and likewise intercommunicates the passage 165a which is formed in the valve block 184 (Fig. 17) with the two passages 287' and 295.

The valve block 270 is provided with a circumferential groove or recess 297 which is located at a point between the passage offsets 283 and 290. Within this groove there is placed a rotary cam 298. The cam may be placed in the recess as described by forming it of two separate halves or, alternatively, as is shown in Fig. 35, the valve block 270 may be formed in two parts meeting at a joint line 299 and held against relative rotation by means of dowel pins 300. The rotary cam 298 is secured as by means of screws 301 to the lower end of the sleeve 274 which, it will be recalled, is rotated at a slow speed by virtue of its connection through the

coupling member 134 to the reduction gearing 117.

In operation the rotary cam 298 is slowly rotated as the rotary pump is operated. The rotary cam 298 is annularly grooved in its upper surface as shown at 302 and includes a similar annular groove 303 formed in its lower surface. These grooves are formed with a radius equal to that of the passages 278, 278' and 291 and are, therefore, so positioned as to receive the stems of the two poppet valves 281 and of the two poppet valves 293. The bottoms of the grooves 302 and 303 act as the camming surfaces for operating the enumerated valves and to this end the grooves 302 and 303 include relatively deep portions, as shown in section in Fig. 35, which are diametrically opposed with intermediate portions which are relatively shallow as shown by the dotted lines 304 and 305.

With the parts positioned as shown in Fig. 35, the poppet valves 281 and 293 are closed. Fluid discharged through the port 160 is therefore conveyed through the arcuate channel 276 of the valve chamber 278', the valve 281 of which is held open by the shallow cam portion 304. The fluid therefore passes the valve 281 and continues downwardly through the valve block through the passages 284 and 287' where, by virtue of the arcuate slot 296, the fluid is conveyed to the passage 165a in the block 184 to apply pressure to the mercury column contained within the cylinder bore 187. Similarly, fluid displaced by the rise of mercury in the cylinder bore 188 is passed downwardly through the passage 166a in the valve block 184 to the arcuate slot 289, by means of which it is conveyed to the passage 295'. The passage 295' communicates with one of the valve chambers 291, the valve 293 of which is at that time held open by the shallow cam portion 305. Fluid therefore passes the valve and continues upwardly through the valve block 270 by way of the passage 285' to reach the arcuate slot 277, by means of which it is conveyed to the inlet passage 156 of the rotary pump.

A quarter of a revolution of the cam 298 after the operating condition above described, the valves previously described as closed will be opened and the other pair of valves will be closed. A fluid discharged through the passage 160 will, therefore, be conducted through passages 278, 284, 287 and 166a to apply pressure to the mercury column contained in the cylinder bore 188. Similarly, fluid displaced by the rise of mercury in the cylinder bore 187 will be conveyed upwardly through the passages 165a, 295 and 285 to the pump inlet 156. Thus, as the cam 298 is rotated the pairs of valves 281, 293 are alternatively opened and closed to connect the rotary pump discharge first to one of the mercury pump cylinders and then to the other of those cylinders while connecting the rotary pump inlet to the opposite one of the pair of mercury cylinders. The mercury is alternately raised and lowered in each cylinder to effect the pumping action in the manner described.

It has recently been established that it is possible to obtain a greater horsepower output per pound of motor for per cubic inch of space required by operating alternating current induction motors at very high speed through the use of a high frequency alternating current supply. The use of 400 cycle alternating current is common in certain fields. My invention therefore comprehends the use of such high speed electric motors as the motors 69, 69', 69'' previously de-

scribed. However, when very high speed motors are used, it is desirable to interpose a speed reduction between the motor shaft and the shaft of the rotary pump as such pumps do not operate and are not efficient at very high speeds.

I have illustrated in Figs. 12 and 13 a simple form of gear reduction unit which may be interposed between the jack shaft 94 and the shaft extension 105. This gear reduction unit may include a pinion 310 secured to the shaft 94 and journaled for rotation by means of ball bearings 311 mounted in a housing carrier 312 non-rotatably secured to the lower end of the sealing member 97 as by means of dowels 313. To the housing carrier 312 there is secured as by means of threads 314 a gear housing 315 which is turned inwardly at its lower end as shown at 316 to support a ball bearing 317 serving to journal for rotation a downwardly depending sleeve portion 318 of a spider 319, the sleeve portion 318 being non-rotatably secured to the upper end of the shaft extension 105.

The shaft extension 105 is held aligned with the shaft 104 as by means of a pin 320 formed upon or pressed into the shaft 94 and received within an axial counterbore formed in the upper end of the shaft 105. The gear housing 315 is cylindrical in form and has formed on the inner face thereof gear teeth 321. Planet gears 322 are mounted for rotation upon the spider 319 as by means of pivot shafts 323 and associated bearings 324 and 325. The planet gears 322 mesh with the pinion 310 and with the ring gear teeth 321, the planet gears 322 are caused to progress around the periphery of the ring 321 in an orbital motion about the pinion 310, thus driving the shaft extension 105 at a reduced speed.

The drive ratio of the gearing just described is so chosen as to reduce the high operating speed of the shaft 94 to a suitable speed for operating the rotary pump shown in Fig. 17.

From the foregoing it will be observed that I have provided a self-contained deep well pump which employs a rotary prime mover operating a reciprocating pump of the fluid passage type. Attention is particularly directed to the mechanical arrangement by means of which the entire pump assembly is made up as a unit and mounted within a single outer housing 1. Breaking the upper joint at the wishbone 3 allows the entire pump structure to be removed from the housing for repairs, replacements and adjustments.

While I have shown and described the preferred embodiment of my invention, I do not desire to be limited to any of the details of construction shown or described herein, except as defined in the appended claims.

I claim:

1. In a deep well pump, the combination of: a tubular carrier; a pump assembly in said carrier including a pumping unit and a drive unit, and means securing said units to each other to form a unitary assembly, said pumping unit having means defining an inlet passage and said drive unit being spaced from said carrier to define a discharge passage, said unitary assembly being freely slidable axially in said carrier; means at one end of said carrier surrounding said discharge passage and engaging said assembly for holding said assembly in said carrier; and means on the other end of said carrier defining a seal between said assembly and said carrier.

2. In a deep well pump, the combination of: a tubular carrier, a pump assembly in said car-

rier including a pumping unit and a drive unit, and means securing said units to each other to form a unitary assembly, said pumping unit having means defining an inlet passage and said drive unit being spaced from said carrier to define a discharge passage, said unitary assembly being freely slidable axially in said carrier; means at one end of said carrier surrounding said discharge passage and engaging said assembly for holding said assembly in said carrier; means at the other end of said carrier defining a seal; and means on said pumping unit surrounding said inlet passage defining a seating element for engaging said seal when said assembly is placed in said carrier to establish a fluid seal between said assembly and said carrier.

3. In an electrically driven deep well pump for attachment to the lower end of a string of flow tubing extending to the ground surface, the combination of: a tubular carrier; a pumping unit including an electrical motor and a pump driven thereby mounted in said carrier and extended into the interior of said carrier and spaced from said carrier to form a passage for pumped fluid; an electrical cable extending into said carrier for conducting electrical power to said motor; closure means for the upper end of said carrier, the said closure means comprising a sealing unit establishing a fluid seal between said carrier and said electrical cable; a pair of conduits on opposite sides of said sealing unit each communicating with the interior of said carrier for discharge of pumped fluid; and means joining said conduits at a point spaced from said sealing unit and including means for attachment to said flow tubing, whereby said cable may be displaced laterally at a point between said carrier and said flow tubing and extended to the ground surface alongside of said string of flow tubing.

4. In a deep well pump, the combination of: a housing adapted to be disposed deep within a well bore and subjected to the hydrostatic pressure exerted by a column of liquid extending to the ground surface from the location of said housing in said well bore; an electric drive mechanism mounted in said housing; insulating fluid in said housing and immersing said electric drive mechanism; a driven unit mounted in said housing and drivably connected to said electric mechanism, a lubricating fluid in said housing immersing said driven unit; sealing means in said housing interposed between said electric mechanism and said driven unit for separating said insulating fluid from said lubricating fluid; and pressure equalizing means mounted in said housing in contact with each of said fluids and communicating with the exterior of said housing to exert upon each of said fluids a pressure substantially equal to said static pressure to prevent leakage between said fluids and between said liquid and either of said fluids.

5. In a deep well pump, the combination of: a prime mover; a rotary pump connected to be driven by said prime mover and having inlet and discharge passages; a fluid-piston pump having a first and a second chamber; a poppet valve mechanism interposed between said rotary pump and said fluid-piston pump operable to connect said inlet and discharge alternately to said first and second chambers of said fluid-piston pump; a rotary member for operating said valve mechanism; a gear reduction unit connected to said prime mover and having a slow speed take-off; and means connecting said slow speed take-off to said rotary member.

6. In a deep well pump, the combination of: a prime mover; a rotary pump including a pumping element mounted within a stator, said stator being secured to a lower head member having inlet and discharge passages formed therein; a fluid-piston pump having at least two cylinders; a stationary valve member interposed between said fluid-piston pump and said head member and defining passages communicating with the cylinders of said fluid-piston pump; and gear reduction means connected between said prime mover and said rotary pump to rotate said pump stator and said head member to thereby connect said inlet and discharge passages to each of said cylinders in succession.

7. In a submersible oil well pump, the combination of: a structure arranged to be secured to the lower end of a string of flow tubing extended into a well and including a cylindrical housing having a reduced lower end portion; means defining a passageway through said end for intake of fluid to be pumped; a motor driven pump assembly in said housing; a removable head in the upper end of said housing to provide for installation and removal of said pump assembly, said removable head having ports therethrough for passage of pumped fluid and having a central counterbored passageway for an electric cable; a locking ring in said counterbore for effecting a fluid seal between said head and said electric cable; a removable connector having converging fluid passageways terminating in one opening centrally of said connector and providing means for attachment to said flow tubing for conveying pumped fluid from said pump assembly, said connector also having a passageway centrally thereof for receiving said electric cable; and means effecting a rigid fluid-tight connection between said head and said connector.

8. In a submersible oil well pump, the combination of: a structure arranged to be secured to the lower end of a string of flow tubing extended into a well and including a cylindrical housing having a reduced lower end portion; means defining a passageway through said end for intake of fluid to be pumped; a motor driven pump assembly in said housing; a removable head at the upper end of said housing to provide for installation and removal of said pump assembly, said removable head having ports therethrough for passage of pumped fluid and having a central counterbored passageway for an electric cable; a locking ring in said counterbore for effecting a fluid seal between said head and said electric cable; a removable connector having converging fluid passageways terminating in one opening centrally of said connector and providing means for attachment to said flow tubing for conveying pumped fluid from said pump assembly, said connector also having a passageway centrally thereof for receiving said electric cable; means effecting a rigid fluid-tight connection between said head and said connector; said removable motor driven pump assembly having sealed connections with said electric cable at its upper end, and including means defining an intake orifice at its lower end communicating with the fluid passageway in the lower end of said cylindrical housing; means effecting a fluid seal between the lower end of said pump assembly and the lower end of said housing, said pump assembly having a diameter less than the inside diameter of said housing to define an annular fluid flow space between the exterior of said assembly and the interior of said cylindrical housing, said pump assembly in-

cluding fluid discharge ports near its lower end communicating with said space; and means for centralizing said assembly in said housing and preventing said assembly from rotating in said housing upon operation of said pump assembly.

9. In a deep well pump for attachment to the lower end of a string of flow tubing, the combination of: a shell adapted to be lowered into a well and having an outlet for fluid to be pumped and having an electric cable inlet; means on said shell for attachment to said string of flow tubing and defining a fluid passage interconnecting said outlet and said tubing; an electric cable secured to said tubing; a motor driven oil filled pump assembly unit removably housed within said shell; means effecting a sealed connection between said electric cable and said pump assembly, said pump assembly including inlet means defining a valved inlet port for admitting pumped fluid; and means effecting a fluid-tight junction of said inlet means with said fluid inlet of said shell, said pump assembly also including outlet ports positioned to discharge pumped fluid inside of said shell, whereby said pumped fluid is conveyed through said outlet of said shell to said string of flow tubing.

10. In a deep well pump for attachment to the lower end of a string of flow tubing, the combination of: a shell adapted to be lowered into a well and having an outlet for fluid to be pumped and having an electric cable inlet; means on said shell for attachment to said string of flow tubing and defining a fluid passage interconnecting said outlet and said tubing; an electric cable secured to said tubing; a motor driven oil filled pump assembly unit removably housed within said shell; means effecting a sealed connection between said electric cable and said pump assembly, said pump assembly including inlet means defining a valved inlet port for admitting pumped fluid; means effecting a fluid-tight junction of said inlet means with said fluid inlet of said shell, said pump assembly also including outlet ports positioned to discharge pumped fluid inside of said shell, whereby said pumped fluid is conveyed through said outlet of said shell to said string of flow tubing; a flexible pressure responsive member contained within said motor driven pump assembly and defining an interior space of variable volume; and means defining a passage intercommunicating said space with the inside of said shell, whereby pumped fluid is conveyed to said space to vary the volume thereof in accordance with changes in the pressure of said pumped fluid as compared to the pressure in said motor driven pump assembly.

11. In an oil well pump, the combination of: a shell adapted to be lowered into a well bore; a motor driven oil filled fluid actuated double acting mercury piston pump assembly in said shell, said shell having formed therein fluid intake and fluid discharge ports and having an electric cable passageway formed therein; and means for effecting a fluid seal between said shell and an electric cable in said passageway.

12. In a submersible oil well pump assembly, a fluid filled housing containing the following instrumentalities: an electric motor; a high pressure pump operatively connected to said motor for pumping an operating fluid; means mounting said pump for rotation as a unit within said housing; a head member in said housing defining inlet and outlet ports, said pump including intake and discharge ports spaced from each other and positioned to alternately register with said inlet

and outlet ports upon rotation of said pump; means defining a pair of actuating chambers each containing a quantity of said operating fluid and a quantity of mercury, said inlet and outlet ports communicating respectively with said chambers; means defining a pair of crude oil pumping chambers each containing a quantity of mercury; means interconnecting each of said pumping chambers with said actuating chambers; means defining crude oil inlet and outlet passages for each of said pumping chambers; check valves in each of said passages; a sealed leak-proof pressure balancing connection between said high pressure pump and said actuating chambers; means for regulating the sealing pressure between said pump and said head; a plurality of flexible elements within said high pressure pump defining an interior space of variable volume; means for conveying crude oil from said discharge passages to said space to produce within said high pressure pump substantially the same pressure as that against which said pumping chambers are operating; a reduction gearing interposed between said motor and said high pressure pump for rotating said pump as a unit to thereby locate said pump intake and discharge ports with respect to said inlet and outlet ports in said head to thereby produce a reciprocating flow of operating fluid in said actuating chambers to effect a rise and fall of the mercury in said pumping chambers to produce a crude oil pumping action through the cooperation of said check valves; means on said housing defining inlet and outlet orifices for crude oil to be pumped; means on said housing defining an inlet for an electric cable supplying power to said motor; means in said housing preventing rotation of the assembly contained therein upon operation of said motor; and means associated with one or more sections of said housing for maintaining said housing in centrally spaced relation with respect to the instrumentalities contained therein.

13. In a submersible oil well pump structure, a fluid filled housing containing the following instrumentalities: an electric motor; a high pressure pump operatively connected to said motor for pumping an operating fluid, said pump including intake and discharge ports; a plurality of valves for opening and closing said ports; a rotatable ring cam for actuating said valves; a reduction gearing operatively connected to said motor; means connecting said reduction gearing to said ring cam; means defining a pair of actuating chambers each containing a quantity of said operating fluid and a quantity of mercury; means communicating said intake and discharge ports with said actuating chambers; means defining a pair of crude oil pumping chambers each containing a quantity of mercury; means interconnecting said pumping chambers with said actuating chambers; means defining inlet and outlet connections to said pumping chambers; check valves in each of said inlet and outlet connections; a plurality of flexible elements disposed within said housing and defining an interior space of variable volume; means communicating said spaces with said crude oil outlet connections to produce within said housing substantially the same pressure as that against which said crude oil is pumped; a shell for enclosing said housing and having inlet and outlet orifices for crude oil to be pumped and having an electric cable inlet; means in said shell for maintaining said housing

in centrally spaced relation with respect to said shell; and means within said shell for preventing rotation of said housing upon operation of said motor.

14. In a deep well pump, the combination of: a pump housing defining an interior chamber filled with fluid, said housing having formed therein a discharge port communicating with the exterior of said housing; pumping means in said housing for discharging pumped fluid under pressure to the exterior of said housing; a sealed pressure expansible element mounted within said chamber; and fluid passage means connecting said pressure expansible element to the exterior of said housing at a point in fluid communication with said port, whereby the pressure of said fluid in said chamber is balanced against the pressure exerted on the outside of said housing by pumped fluid discharged through said port.

15. In a deep well pump, the combination of: a string of flow tubing adapted to conduct fluid from the bottom of the well to the ground surface; a sealed pump housing disposed within the bottom end of the flow tubing and having an inlet port sealed to the flow tubing and having a discharge port; a pump disposed within the housing and adapted to pump fluid from the inlet port to the discharge port; a non-compressible fluid disposed within the housing; and a pressure expansible element mounted on the housing and exposed to the incompressible fluid on one side and the discharge fluid on the other side, whereby the static head of the pump will be restrained by the well tubing permitting the pump and housing to be lightly constructed, and whereby the interior of the housing will automatically be at the same pressure as the exterior.

BENJAMIN F. SCHMIDT.

REFERENCES CITED

The following references are of record in the file of this patent:

UNITED STATES PATENTS

Number	Name	Date
159,105	Light	Jan. 26, 1875
588,015	Riggin	Aug. 10, 1897
765,826	Felker	July 26, 1906
1,519,018	Boudreau	Dec. 9, 1924
1,593,820	Gates	July 27, 1926
1,643,616	Zinn et al.	Sept. 27, 1927
1,736,094	Schmidt	Nov. 19, 1929
1,830,869	Charles	Nov. 10, 1931
1,845,179	Penrod	Feb. 16, 1932
1,891,424	Kirkpatrick	Dec. 20, 1932
1,891,771	Mendenhall et al.	Dec. 20, 1932
1,907,191	Thomas	May 2, 1933
1,911,856	Stephenson	May 30, 1933
2,039,621	Behnke	May 5, 1936
2,098,958	Ekstromer	Nov. 16, 1937
2,180,400	Coberly	Nov. 21, 1939
2,207,226	Schmidt	July 9, 1940
2,218,937	Myers et al.	Oct. 22, 1940
2,230,787	Swain	Feb. 9, 1941
2,236,887	Arutunoff	Apr. 1, 1941
2,240,901	Ferris	May 6, 1941
2,253,022	Evans	Aug. 19, 1941
2,253,754	Bousman	Aug. 26, 1941
2,265,393	Place	Dec. 9, 1941
2,269,789	Schmidt	Jan. 13, 1942
2,270,666	Arutunoff	Jan. 20, 1942
2,279,498	Schmidt	Apr. 14, 1942
2,283,439	Herman	May 19, 1942