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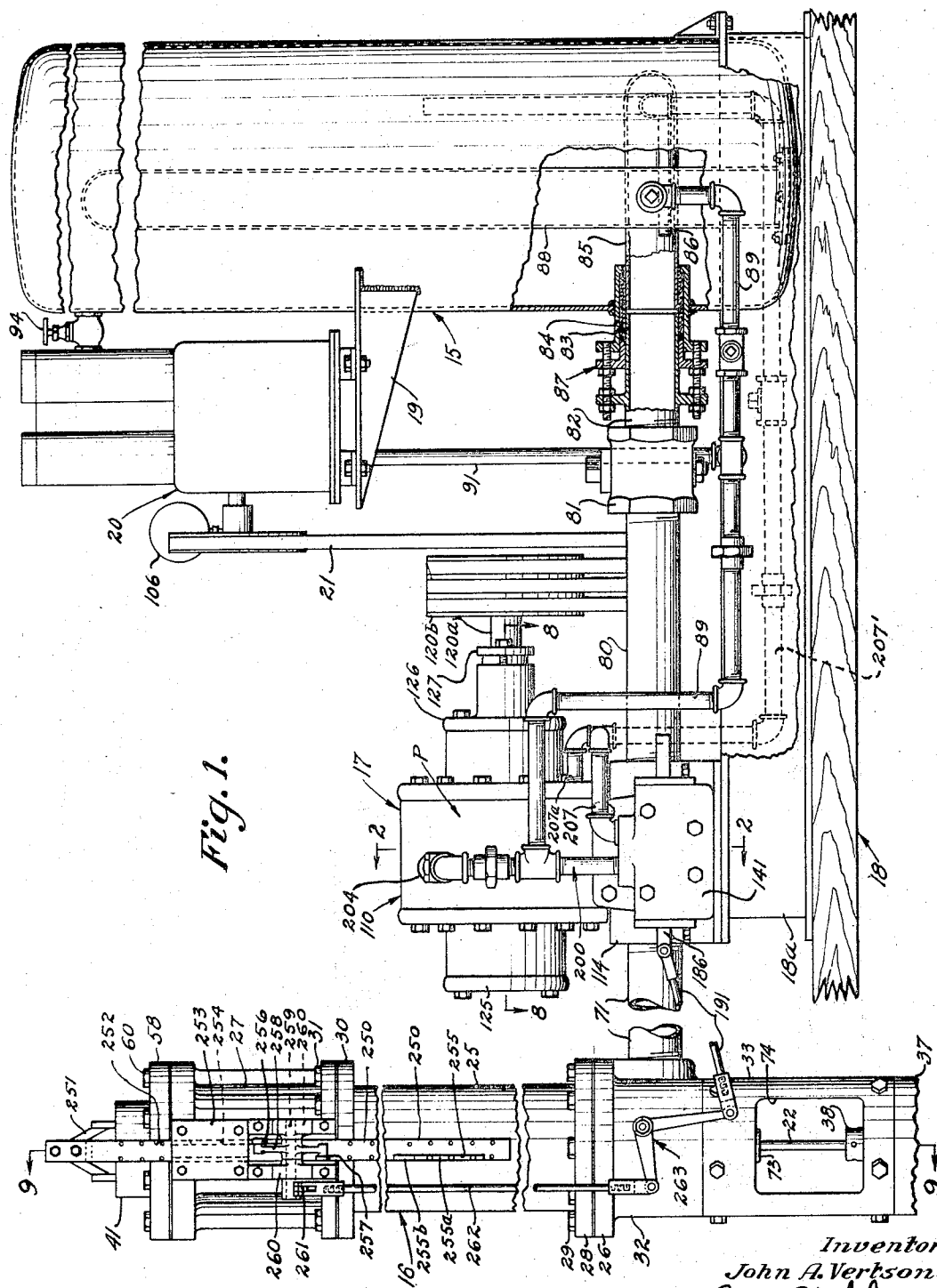
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2,292,331

HYDRAULIC JACK PUMPING PLANT

Filed June 16, 1939

5 Sheets-Sheet 1



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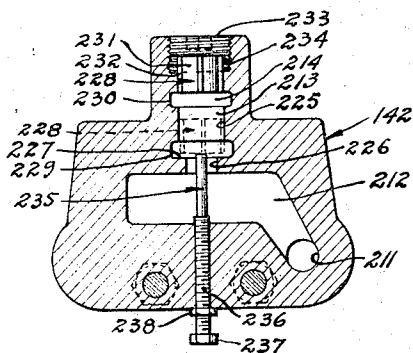
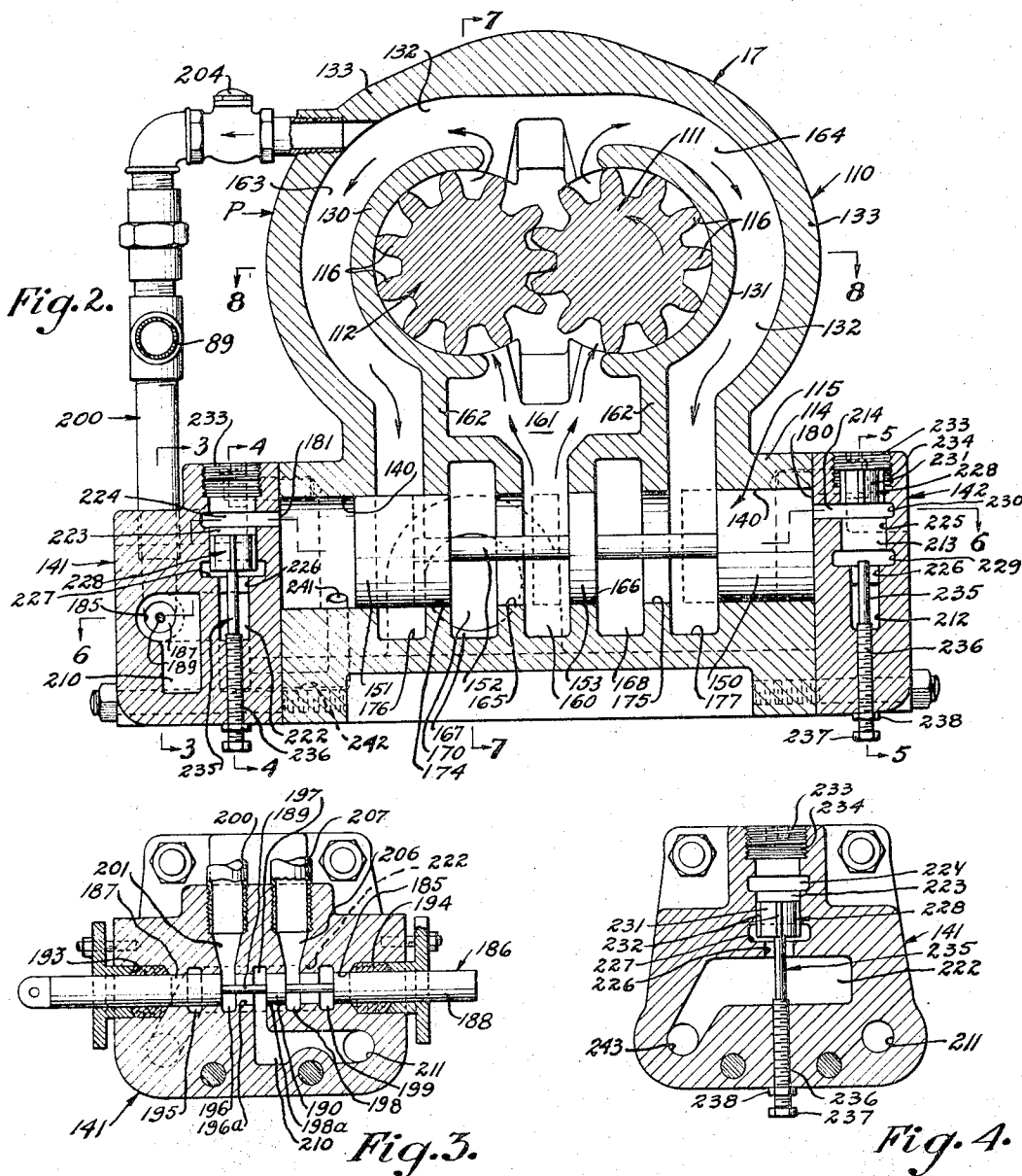
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HYDRAULIC JACK PUMPING PLANT

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5 Sheets-Sheet 2



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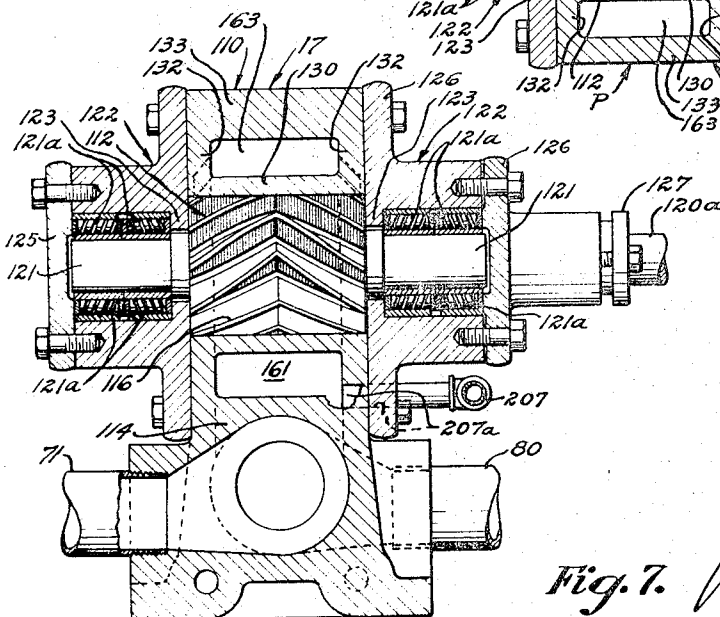
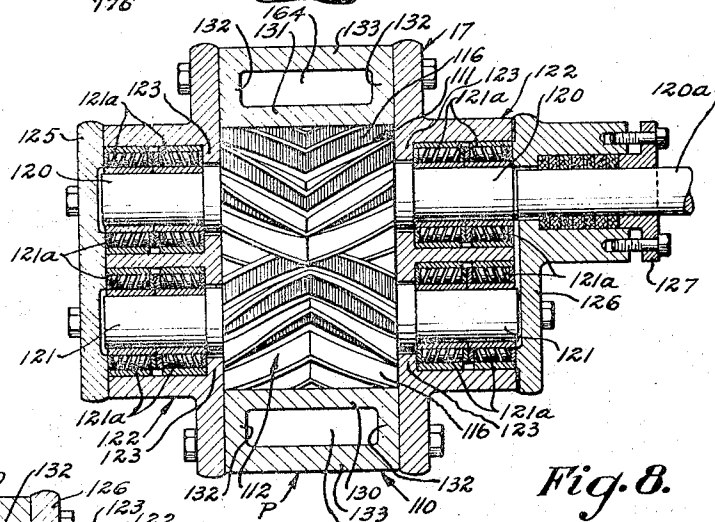
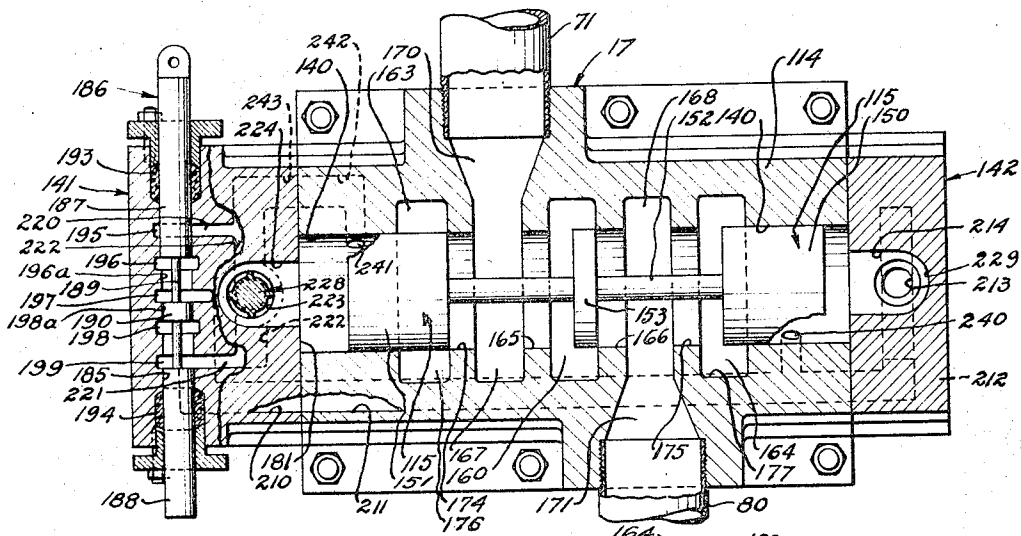
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HYDRAULIC JACK PUMPING PLANT

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5 Sheets-Sheet 3



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HYDRAULIC JACK PUMPING PLANT

Filed June 16, 1939

5 Sheets-Sheet 4

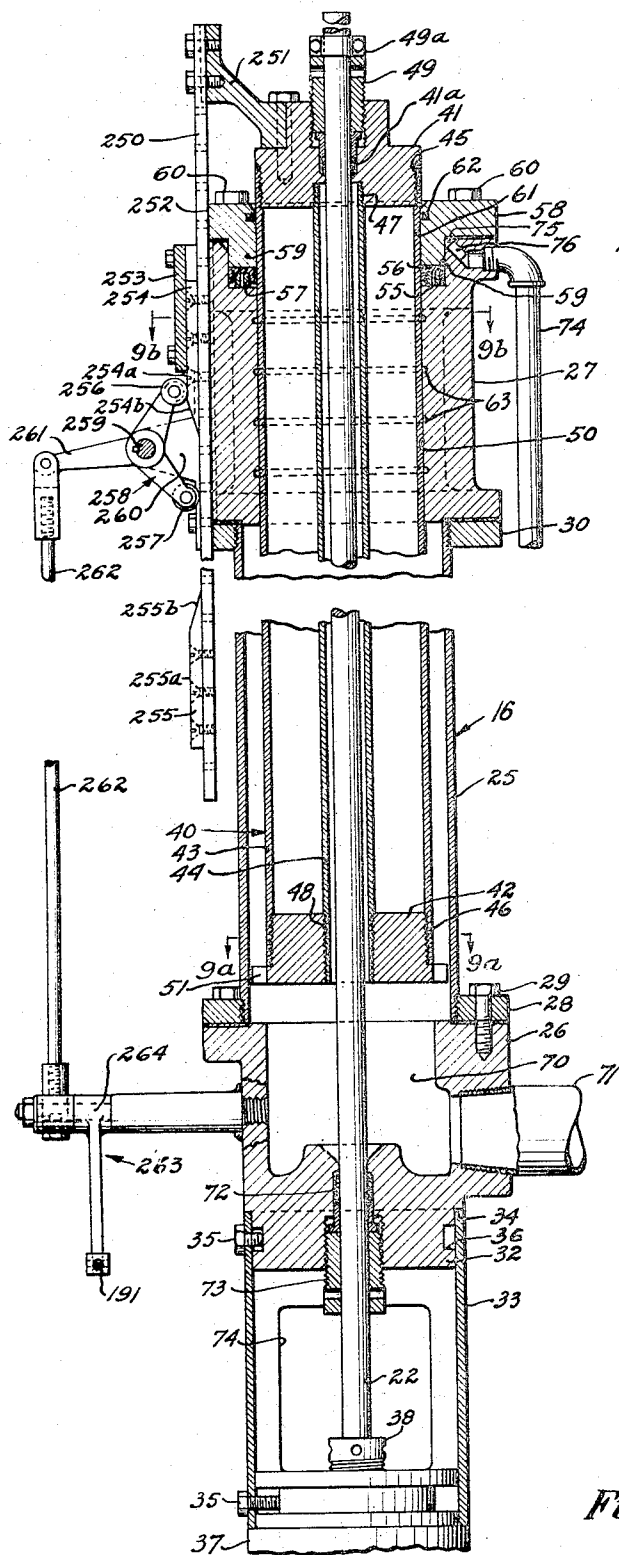


Fig. 9.b.

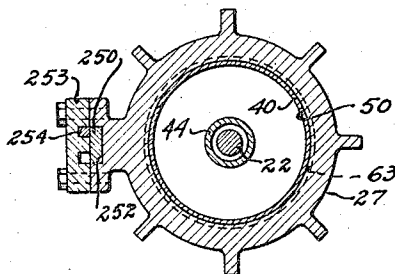


Fig. 9a.

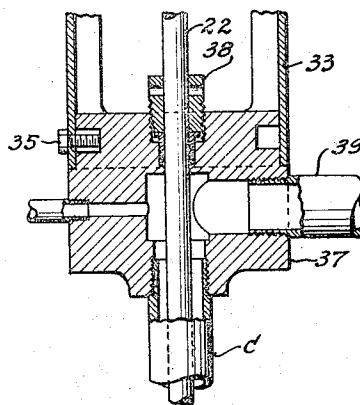
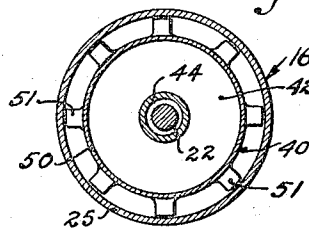


Fig. 9c.

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Fig. 9.

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5 Sheets-Sheet 5

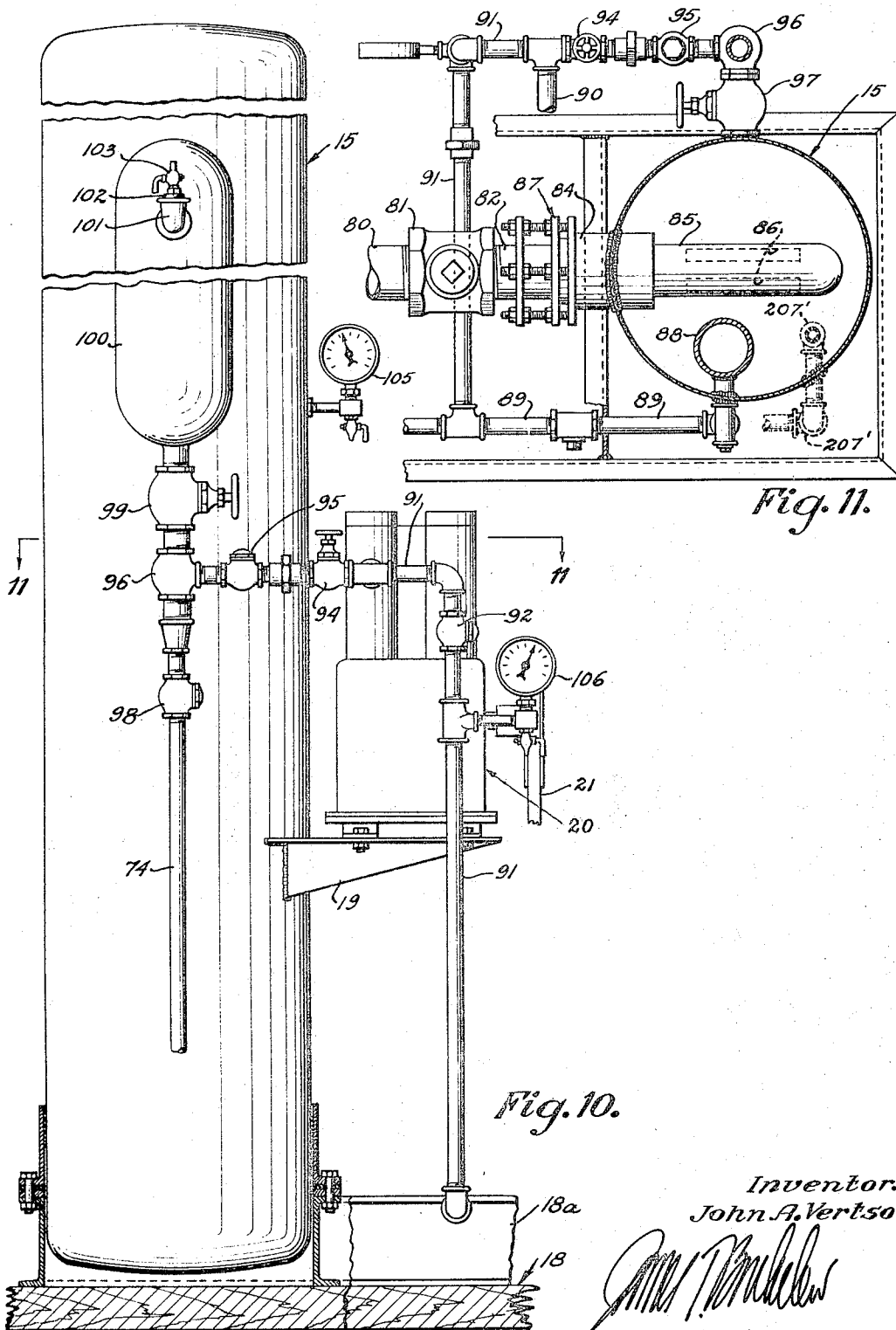


Fig. 11.

Fig. 10.

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UNITED STATES PATENT OFFICE

2,292,331

HYDRAULIC JACK PUMPING PLANT

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Application June 16, 1939, Serial No. 279,396

7 Claims. (Cl. 103—3)

This invention relates generally to deep well pumping plants, and more particularly to deep well pumping plants of the hydraulic jack type.

Hydraulic jack pumping plants of the class here referred to comprise an accumulator containing a fluid under pressure, a working cylinder and piston for reciprocating the pump rod string, and a pump and valve system for alternately pumping working liquid from the accumulator to the cylinder to elevate the piston and rod string, and then from the cylinder to the accumulator to allow the piston and rod string to descend. By maintaining a pressure in the accumulator equal to the pressure necessary in the working cylinder to counterbalance the weight of the pump rod string plus one half the weight of the column of liquid carried by the rod string on the up-stroke, the driving pump operates under a substantially constant head during both up-stroke and down-stroke as is well understood in the art.

It may be stated as a general object of the invention to provide a hydraulic jack of the class mentioned characterized by improvement in operating effectiveness, simplicity and relatively low cost of construction and installation, minimization of maintenance, and the accomplishment of a smooth deceleration and acceleration of the pump rod string at the end of each pumping stroke.

A further and more specific object of the invention is the provision of an hydraulic jack having a reversible flow circulating pump and valve system of increased simplicity as well as improved effectiveness.

Prior pumping plants of the hydraulic jack type have usually employed a flow reversing valve operated mechanically by a linkage driven or tripped by the working piston. However, a simple mechanical linkage will move the valve only to a dead center position at which flow between the tank and working cylinder ceases, since at this time the working piston ceases its motion and the linkage does not throw the valve on past dead center to flow reversing position. For this reason, a purely mechanical linkage must include some additional means for throwing the valve on past dead center. Mechanisms for accomplishing this result have been provided in the past, but have invariably been either quite complicated or else unsatisfactory in performance.

An object of the present invention is the provision of a simple valve system which will avoid the dead center difficulties mentioned immediately above.

The invention provides as a solution to the

valve problem set forth, an hydraulic jack having a master valve for reversing the direction of flow between tank and cylinder, and a pilot valve controlling a pressure fluid for operating the master valve, this pilot valve being mechanically operated by a linkage which is actuated, in turn, by the working piston. There is sufficient delay in the operation of the master valve by the pressure fluid that the working piston is not brought to a stop until after the pilot valve is thrown completely to flow-reversing position, and there is therefore positive assurance that the valves will not stop on "dead center."

A further object of the invention is the provision of an hydraulic jack which is so designed that the rates of deceleration and acceleration of the working piston at the end of each stroke may be readily adjusted within comparatively wide limits. This provision enables "timing" of the acceleration and deceleration of the working piston to the natural and characteristic behavior of the pump string, so that, for example, at the termination of the down stroke, the pump string is arrested in motion and then picked-up again in consonance with its natural resilient rebound. As is well known, if the rate of deceleration and acceleration at the bottom of the stroke are not well matched to the natural resilient stretch and contraction characteristics of the pump string, "whipping," jerkiness and loss of power, as well as undue strain on the string, are experienced. The provisions contemplated also enable adjustment of the rates of deceleration and acceleration at the upper end of the stroke, it being an important advantage to have available a means for adjusting these rates to most advantageous values for any given pumping speed. Thus, as indicated above, it is one primary purpose of the present invention to provide means whereby the deceleration and acceleration of the working piston may be adjusted to values best suited to the characteristics of the pump rod string to which the pumping plant has been applied.

In accordance with the present invention, the rates of deceleration and acceleration of the working piston are adjusted by regulating the speed of travel of the master valve, which is in turn adjusted by regulation of the rate of flow of the high pressure fluid that operates the master valve. This regulation is accomplished, in a present preferred embodiment of the invention, by adjustment of the size of a constriction in the fluid passageway system.

A still further object of the invention is the provision of a valve and pump system for an

hydraulic jack characterized by elimination of the usual water hammer during reversal of the valves.

In accordance with the invention, the master valve is so designed that while moving to reverse the direction of flow between the accumulator and working cylinder, the accumulator and working cylinder are placed in direct communication with one another through the chambers of the valve, so that the working piston is cushioned at such time by the air pressure in the accumulator, rather than coming to a jarring stop as the valves close off flow to the cylinder.

A feature of the invention is a novel type of action of the pump and valve system during reversal of the direction of flow between tank and cylinder. As the master valve moves to reverse the direction of flow, it gradually decreases the flow between tank and cylinder, and at the same time, a local ring circulation from the pump through the valve chamber and back to the pump is established. This local ring circulation through the valve increases until the entire output of the pump is diverted in this manner and no liquid is pumped toward or from either the cylinder or tank. Further travel of the master valve then causes liquid to be diverted from this local ring circulation through the valve and to be delivered to the tank, and at the same time causes the pump to begin to take liquid from the cylinder. The master valve thus decreases the local ring circulation to zero, while increasing the circulation between the cylinder and pump to a maximum in a direction contrary to its initial direction of flow. There is transition from maximum flow in one direction to maximum flow in the other without closing off communication between tank and cylinder, the pump circulating liquid locally through the valve during the transition, and all of the liquid discharged by the pump taking part in this local ring circulation through the valve at the time flow between tank and cylinder momentarily ceases preparatory to reversing its direction.

Further objects of the invention relate to features of the working cylinder, and to the flow connections and valves employed in connection with the pipe lines interconnecting the several units of the plant.

With this preliminary discussion in mind, the invention itself, as well as various additional objects and features not specifically mentioned, will be made clear from the following detailed description of a present preferred embodiment of the invention, reference for this purpose being directed to the accompanying drawings, in which:

Fig. 1 is a side elevation, with parts broken away, of an illustrative hydraulic pumping plant in accordance with the present invention;

Fig. 2 is a vertical section through the gear pump and main valve chest taken as indicated by line 2—2 of Fig. 1;

Fig. 3 is a vertical section through the pilot valve chest taken as indicated by line 3—3 of Fig. 2;

Fig. 4 is a vertical section taken as indicated by line 4—4 of Fig. 2;

Fig. 5 is a vertical section taken as indicated by line 5—5 of Fig. 2;

Fig. 6 is a section taken on broken line 6—6 of Fig. 2;

Fig. 7 is a section taken as indicated by line 7—7 of Fig. 2;

Fig. 8 is a section taken as indicated by line 8—8 of Figs. 1 and 2;

Fig. 9 is a vertical section through the working cylinder and piston taken as indicated by line 9—9 of Fig. 1;

Fig. 9a is a detail section taken on line 9a—9a of Fig. 9;

Fig. 9b is a detail section taken on line 9b—9b of Fig. 9;

Fig. 9c is a downward continuation of Fig. 9;

Fig. 10 is a view of the main pressure tank as seen from the rear in Fig. 1; and

Fig. 11 is a view taken on line 11—11 of Fig. 10.

With reference now to the drawings, and particularly to Fig. 1, numeral 15 designates generally a pressure tank or accumulator, numeral 16 the working cylinder, and numeral 17 a pump and valve unit for pumping working fluid alternately from tank 15 to cylinder 16 and then from cylinder 16 to tank 15. These units are indicated as mounted on or above a suitable platform or flooring, indicated at 18, unit 17 and the tank being indicated as furnished with a common base 18a resting on platform 18. Accumulator 15 is shown provided with a bracket 19 on which is mounted an air compressor of any suitable type, generally designated at 20, and indicated in Fig. 1 as driven by means of a belt 21 from any suitable source of power, as an electric motor, not shown.

I prefer to employ a single vertically disposed working cylinder positioned over and in alignment with the well casing, and the invention is so illustrated, though without implied limitation thereto. Thus, as here shown, working cylinder 16 is positioned over and in alignment with the upper end of well casing C, only a fragmentary upper end portion of the latter appearing in the drawings. The polish rod is designated by numeral 22, and extends upwardly through cylinder 16 as clearly illustrated in Fig. 9.

While the working cylinder and piston may be of any desired or appropriate construction, that here shown presents certain novel features of advantage and will therefore be described in some detail. The illustrated cylinder 16 embodies a barrel 25, mounted at its lower end on a lower cylinder head fitting 26, and having mounted on its upper end a cylinder head 27. The preferred mounting for barrel 16 on fitting 26 consists of a ring 28 screw threaded on the lower end of barrel 16 and secured to the upper end of fitting 26 as by means of screws 29. In a similar manner, head 27 is preferably mounted on the upper end of barrel 25 by means of a ring 30 screw-threaded on the upper end of the latter and secured to head 27 by screws 31. Fitting 26 has at the bottom a reduced rounded portion 32 received within the upper end of a depending tube or sleeve 33, fitting 26 and the parts above being supported by the engagement of the upper end of sleeve 33 with shoulder 34. Screws 35, threaded in the upper end of sleeve 33 and engaging in a peripheral groove 36 formed around the lower end 32 of fitting 26, secure members 26 and 33 in assembly. Sleeve 33 is mounted in any suitable manner on the usual tubing head 37 on the upper end of the well casing C (see Fig. 9c), the tubing head being provided with a packing gland 38 for the polish rod, and having connected thereto a pipe line 39 for receiving the well fluid.

The present preferred piston 40 for cylinder

5 is made up of an upper head 41, a lower head 2, and concentric outer and inner sleeves 43 and 44 joining with and extending between heads 1 and 42. Sleeve 43 is internally threaded at its upper and lower ends for connection at 45 and 46 with external threads on heads 41 and 42, respectively. Inside sleeve 44, which is preferably of an inside diameter slightly greater than that of polish rod 22, so as to leave an annular clearance therebetween, is externally threaded at its upper and lower ends, being screwed at its upper end within a screwthreaded socket 47 formed in the underside of head 41, and being screwed at its lower end inside internally threaded central bore 48 of lower head 42. Cylinder head 27 is formed with a finished cylinder bore 50, within which outside sleeve 43 of piston 40 has a working fit, bore 50 being of a diameter somewhat less than the inside diameter of cylinder 25, so as to leave an annular clearance space between cylinder 25 and the outside sleeve 43 of the piston. The lower end of the piston is guided inside cylinder 25 by means of a notched or interrupted flange 51 projecting outwardly from lower piston head 42 (see Fig. 9a), this interrupted flange having a comparatively free sliding fit inside cylinder 25.

The upper end of cylinder head 27 has a countersink 55, in which is received a U-leather packing ring 56, the latter being spread by a split-resilient ring 57 of any suitable type. A cover 58 mounted on the upper end of cylinder head 27 has an annular downwardly projecting portion 59 received within countersink 55 and following up and supporting U-leather 56. Cover 58, which is secured to the upper end of cylinder head 27 as by screws 60, has a bore 61 slidably embracing piston 40, and is shown as provided with a wiper ring at 62. Cylinder head bore 50 may be provided with suitable oil grooves, as indicated at 63. The described construction provides for an exceptionally long stroke of the piston without necessitating an unduly long finished cylinder bore, the latter being only a fraction of the full length of cylinder 16.

Fitting 26 has a chamber 70 opening through its upper side to the interior of cylinder 16, and connected to this fitting and communicating with said chamber 70 and therefore with the interior of cylinder 16 is a pipe 71 which leads from the driving pump. The lower portion of fitting 26 is bored to pass polish rod 22, and suitable packing means is provided between the polish rod and the lower portion of fitting 26, as indicated. For example, the fitting may be provided with a packing gland 72, set up by a bushing 73 screwthreaded into the lower end portion of the fitting. Access to bushing 73 and this packing is had by way of aperture 74 in sleeve 33.

Upper cylinder head 41 is bored to pass the upper end portion of polish rod 22, and to provide for a packing gland 41a which packs between the polish rod and cylinder head. This packing gland is tightened by means of a bushing 49 screwed into the upper end of head 41. A rod clamp 49a on the upper end of the polish rod engages the upper end of bushing 49, and thus suspends the polish rod and pump rod string from working piston 40.

A further feature of the invention illustrated in part by Fig. 9 is a means for collecting oil which passes U-leather 56 and delivering it back to tank 15. Any such oil passes upwardly between the vertical surfaces between the annular portion 59 of cover 58 and of countersink 55 to an annular collection space 75. This oil then

drains by way of downwardly and outwardly inclined passageway 76 to a pipe line 74, which leads to a drip tank as later to be described.

A pipe 80 connected to pump and valve unit 17 leads via valve 81 to a fitting 82 coupled to accumulator 15. The coupling employed preferably employs packing in such a way as to avoid direct metal to metal contact between coupling member 82 and the tank, so as to avoid transmission of vibrations from the pump to the tank, which sometimes cause the tank to provide noise. In the arrangement illustrated, fitting 82 is received within a packing ring or sleeve 83, which is mounted inside a tubing 84 extending through and welded in the side wall of the tank. Pipe fitting 85 is received and supported within packing sleeve 83, and projects inwardly across the interior of the tank. This fitting 85 is provided with a suitable liquid opening, as for instance the downwardly directed slots 86. Packing sleeve 83 may be tightened by any suitable means, as for example by the follower and tightening device illustrated at 87.

The pumping plant of the present invention includes, in addition to main pressure tank 15, a high pressure fluid source which preferably comprises a smaller high pressure tank 88, and this small tank is conveniently and preferably mounted inside main tank 15. This high pressure tank 88 has connected thereto a pipe line 89, which will be understood to extend through the wall of tank 15 and to communicate with the interior of tank 88.

The discharge line from compressor 20 is designated by numeral 90 (see Fig. 11, in which the compressor itself, however, has been omitted). Compressor line 90 is connected to a pipe line 91 that connects to the aforementioned high pressure line 89 communicating with the interior of high pressure tank 88, and this line 91 contains a check valve 92 adapted to hold the pressure in high pressure tank 88. Compressor line 90 also connects via valve 94 and check valve 95 with a fitting 96 that communicates by way of valve 97 with the interior of main tank 15. Check valve 95 permits flow of air from compressor 20 to tank 15, but holds the pressure in the tank when the compressor is stopped.

The previously mentioned oil line 74 that collects oil leaking past the packing in cylinder head 27 is coupled to the bottom connection of fitting 96, this line 74 having, just below fitting 96, a check valve 98 preventing back flow in line 74. Assuming the pumping plant to be in normal running condition, valves 94 and 97 will be closed, and the oil collected from cylinder head 27 will flow by way of line 74 into fitting 96, thence upwardly through its upper connection and via valve 99 to drip tank 100, which latter is thus positioned above the valve 97 which communicates with the interior of tank 15. Drip tank 100 has near its upper end an oil inlet fitting 101, provided with a closure plug 102. This closure plug 102 is provided with a pet-cock 103, which normally stands open and permits tank 100 gradually to fill from line 74 leading from the working cylinder, as before described. The accumulation of oil in tank 100 is from time to time discharged into main tank 15. This is accomplished by first closing pet-cock 103 and then opening valve 97. This causes the pressures in main tank 15 and tank 100 to become equalized, whereupon the oil in tank 100 will drain into tank 15. Valve 97 is then closed and pet-cock

103 opened, putting the system back in its normal operating condition.

Inlet fitting 101 is also utilized in initially introducing the working liquid (oil) to tank 15, valve 97 of course being opened at such time to permit the oil to drain into tank 15. Enough oil is introduced to supply the operating needs of the system; typically, tank 15 is about one-fourth full of oil when the pumping plant is in normal operation. To build up the air pressure in tank 15, valve 99 is closed and valves 94 and 97 opened. Compressor 20 is then operated until the necessary pressure has been built up in tank 15, as read from pressure gauge 105. This pressure, as explained elsewhere, is substantially that which is necessary in the working cylinder to counterbalance the weight of the piston and pump rod string, plus one-half the weight of the column of well fluid, and may be designated as

$$\left(R + \frac{C}{2}\right)$$

where R is the pressure necessary to counterbalance the pump rod string, and

$$\frac{C}{2}$$

is the pressure necessary to counterbalance one-half the weight of the column of well fluid. In practice, it is usually desirable to initially overcharge the tank somewhat, that is, to a pressure somewhat above

$$\left(R + \frac{C}{2}\right)$$

so as to allow for the certain inevitable gradual loss of pressure without immediately falling below an operating minimum. The compressor at the same time pumps air through connection 91 to high pressure line 89 leading to high pressure tank 88. When the proper pressure is reached in main tank 15, valve 94 is closed, check valves 95 and 98 holding the pressure in main 15, while the compressor continues to increase the pressure in high pressure tank 88. When a pressure of say 50 lbs. greater than the pressure in tank 15 is reached in tank 88, as read by gauge 106 connected into line 91, the compressor is stopped, the high pressure in tank 88 being held by check valve 92 (Fig. 10).

Preferably, and as a feature of the invention, the driving pump and valves are embodied in a single unit 17. This unit embodies a main casting 110 serving as a housing for the rotors or gears 111 and 112 of the pump proper, designated at P, and providing a chest 114 for master valve 115. While the pump proper may be of any desired or appropriate type, I prefer to employ a gear pump of the type embodying a pair of rotors formed with intermeshing double helical gear teeth 116, a pump of this type being particularly efficient and furnishing a constant, positive pressure. Rotors 111 and 112 have axles 120 and 121, respectively, journaled in suitable bearings 121a supported by end castings 122, the latter fitting closely around said axles adjacent the rotor ends and presenting walls 123 defining the sides of the rotor chamber. A pump drive shaft 120a carrying a driving pulley 120b extends from one of the axles 120 (Figs. 1 and 8). A cover plate 125 closes one of the bearing supporting castings 122, while the other bearing supporting casting 122 carries a closure member 126 furnished with a packing gland 127 for drive shaft 120a.

Casting 110 includes two substantially semi-

circular walls 130 and 131 fitting closely around the outer sides of rotors 111 and 112, these walls joining at their edges with horizontally spaced vertical walls 132. A peripheral wall 133 joining with walls 132 forms the outer sides and top of the pump casing, and, together with walls 132, merges with the top of valve chest 114.

The aforementioned master valve 115 may be embodied in various mechanical forms, though it is preferably a balanced, linearly travelling valve. This valve 115 is mounted for reciprocation in a valve bore 140 in valve chest 114, said bore extending transversely of the axes of pump rotors 111 and 112, in the disposition clearly shown in Fig. 2. Valve bore 140 opens through opposite ends of valve chest 114, and communicates with certain later described ports or passages formed in end castings 141 and 142 secured to opposite ends of valve chest 114.

Master valve 115, in the present preferred and specific form here illustrated, comprises a pair of valve elements 150 and 151 in the form of piston sections or heads, connected by a reduced valve rod 152, and a central valve element 153 in the form of a valve disk mounted or formed on rod 152 at a position half way between valve elements or heads 150 and 151. These valve elements 150, 151 and 153 all slidably fit valve bore 140.

An annular port 160 surrounds valve bore 140 at a point midway of the length of the latter, this port being of a dimension, measured axially of the valve bore, equal to twice the thickness of valve disk 153. Port 160 communicates with a pump intake chamber 161 leading directly to the lower side of pump rotors 111 and 112, via the port defined by the lower ends of semi-circular walls 130 and 131, this intake chamber 161 being enclosed by the aforementioned walls 132 of the main casting and by walls 162 joining with and extending downwardly from walls 130 and 131, as clearly shown in Fig. 2. Thus the intake side of the pump is at all times in open communication with port 160, and, the pump rotors being understood to be driven in the directions indicated by the arrows in Fig. 2, it will be understood that suction will be developed in chamber 161 drawing liquid from port 160 upwardly through chamber 161, thence around the inside of walls 130 and 131 within the spaces between gear teeth 116, to be discharged to the two pump discharge passageways 163 and 164 via the port defined by the upper ends of walls 130 and 131, the oil travelling in the paths indicated by the arrows in Fig. 2. It should be pointed out that while one direction of operation for the pump is here assumed, this is not at all limitative, since the system will operate with the pump rotors running in either direction.

On opposite sides of port 160 are valve seats 165 and 166 for valve disk 153, these seats each being of a width substantially equal to the thickness of valve disk 153. Surrounding valve bore 140 outside valve seats 165 and 166 are annular ports or valve chambers 167 and 168, of dimensions, measured axially of valve bore 140, equal to twice the thickness of valve disk 153. Port 167 communicates by way of passage 170 (see Fig. 6) with the aforementioned pipe 71 leading to working cylinder 16, while port 168 communicates by way of passage 171 with pipe 80 connecting with the interior of accumulator 15. Immediately adjacent and to the outside of ports 167 and 168 are valve seats 174 and 175, of the same dimensions as valve seats 165 and 166.

Surrounding valve bore 140 immediately adjacent and to the outside of valve seats 174 and 175 are annular ports 176 and 177, of dimensions, measured axially of the valve bore, again equal to twice the thickness of valve disk 153.

Port 176 is in open communication with pump discharge passageway 163, while port 177 is in open communication with pump discharge passageway 164, as shown in Fig. 2. The master valve is so proportioned that when valve element 153 is, for instance, seated on its seat 166, as in full lines in Fig. 2, piston section or element 150 will be outside of port 177, establishing free communication between pump discharge passageway 164 and port 168 leading to tank 15, whereas piston section or element 151 will at this time be seated on valve seat 174 and will therefore form a barrier between pump discharge passageway 163 and port 167 leading to the working cylinder. In the particular form of valve here illustrated, and with the proportions of valve seat and liquid port dimensions as given above, these relations are obtained by making the spacing between the valve disk 153 and each one of the piston heads 150 and 151 substantially equal to five times the thickness of the central disk 153. With the valve in the full line position of Fig. 2, the outer end of piston section 150 is in engagement with the abutment presented by the opposed face 180 of end casting 142, as shown in Fig. 2. Valve disk 153 being seated on valve seat 166, suction chamber 161 of the pump is then in open communication with port 167 leading from working cylinder 16, so that liquid will be pumped from the working cylinder via pipe 71, ports 167 and 160, suction chamber 161, discharge passageway 164, ports 177 and 168, and pipe 80 to tank 15, permitting the piston and pump string to descend.

It will be understood that when the master valve is in its extreme position toward the left, valve element 153 will be seated on seat 165, valve element 151 will be to the left of port 176 and in engagement with the abutment presented by the opposed face 181 of end casting 141, with valve element 150 seated on valve seat 175. Under such conditions, liquid will be pumped from tank 15 via pipe 80, ports 163 and 160 (then in open communication), pump suction chamber 161, pump discharge passageway 163, ports 176 and 167 (then in open communication) and pipe 71 to working cylinder 16 to elevate piston 40 and the pump rod string.

A feature of the invention in one of its primary aspects involves operation of master valve 115 under the differential pressure of fluids at two different pressure levels. In accordance with the present preferred form of the invention, the high pressure fluid source constitutes the aforementioned high pressure tank, which contains oil under a pressure substantially above that in main tank 15, while the lower pressure is supplied from any suitable low pressure point in the oil circulation system. While this feature is not indispensable to certain other aspects of the invention, and is therefore not specified in all of my claims, it leads to certain very important advantages and results, especially in conjunction with the other aspects of the invention, as will presently appear, and incorporation of this feature is therefore deemed preferable and important. For control of this high pressure valve-operating fluid, a four-way pilot valve is preferably employed, and while this pilot valve may be embodied in various me-

chanical forms, and may or may not be incorporated within the body of the pump unit 17, I prefer to employ a balanced pilot valve working in a pilot valve bore 185 in end casting 141.

This preferred pilot valve, which is designated generally by numeral 186, comprises two rod or piston sections 187 and 188, a connecting valve rod 189 of reduced cross section, and a central valve disk or element 190 mounted on rod 189 midway between piston sections 187 and 188. Pilot valve 186 is adapted to be mechanically moved by means of a link 191 pivotally connected to the outer end of piston section 187 and operated in accordance with the reciprocation of working piston 40, in a manner later to be explained. Section 187 is packed by means of a suitable gland at 193, and section 188 is packed by similar gland at 194. A series of five equally spaced annular ports 195 through 199 surround valve bore 185, the spacing between said ports being substantially equal to the length dimension of valve element 190, and the wall portions of bore 185 between central port 197 and the adjacent ports 196 and 198 providing annular seats 196a and 198a, respectively, for valve element 190.

It is preferable and convenient, and a feature of the present invention, that the high pressure fluid utilized to operate master valve 115 be supplied or kept up from the discharge side of pump P. For this purpose, a high pressure line 200 is connected into pump discharge passageway 163 (Fig. 2), and this high pressure line 200 is connected to the previously mentioned line 89 leading to small high pressure tank 88, and also to a passageway 201 leading to the aforementioned port 196 (Fig. 3). A check valve 204 in line 200 between the pump and the connection with line 89 holds the pressure in tank 88 when the pump is stopped. The discharge side of pump P as well as tank 88 are thus in communication with pilot valve passageway 201 and port 196. It has previously been mentioned that a certain air pressure, as for example 50 lbs. pressure in excess of the pressure in tank 15, is preferably initially built up in tank 88 by means of compressor 20. The pump P, which during a part of the operating cycle delivers liquid at a pressure equal to $(R+C)$, discharges liquid via line 200 and 89 to tank 88, the pressure in which is thus kept up to approximately $(R+C)$.

Annular port 196 (Fig. 3) communicates via passage 206 with a discharge pipe line 207 that preferably discharges to a low pressure point in the working liquid system. By thus discharging the valve operating liquid back to the working liquid system, the necessity for frequent replenishing of the working liquid is avoided. As here shown, this line 207 is connected to a port 207a opening inside pump intake chamber 161. An alternative arrangement, indicated in Figs. 1 and 11 in dotted lines, consists of a line 207' discharging back to tank 15.

Central valve bore port 197 communicates via passageway 210 in end casting 141 with one end of a passageway 211 extending longitudinally through valve chest 114, the other end of the latter communicating with a passageway 212 in end casting 142. Passageway 212 communicates in turn with a vertical, check-valved passageway 213, from which leads a horizontal port 214 opening through surface 180 of the end casting and communicating with the liquid chamber at the right-hand end of master valve

bore 140, as viewed in Fig. 2. This liquid chamber will be observed to be defined by wall surface 180, the end portion of valve bore 140, and to be closed by the pressure area on the end of master valve piston section 150.

The two end or outside valve bore ports 195 and 199 (Fig. 3) communicate via passageways 220 and 221, respectively, with a passageway 222 (see Figs. 6 and 4), and the latter communicates with a vertical, check-valved passageway 223, from which leads a horizontal port 224 opening through surface 181 of end casting 141 and communicating with the liquid chamber at the left-hand end of master valve bore 140, this liquid chamber being defined by wall surface 181, the end portion of valve bore 140, and being closed by the pressure area on the end of master valve piston section 151.

The check-valved passageways 213 and 223 and the adjustable check valve elements contained therein are similar and a detailed description of one will suffice for both. Referring particularly to Figs. 2, 5 and 6, it will be seen that passageway 213 consists of a vertical bore 225 extending downwardly from the upper end of end casting 142 to a point somewhat short of passageway 212, and a reduced bore 226 continuing downwardly from bore 225 through to passageway 212, the construction providing an upwardly facing annular stop shoulder 227 for the check valve element, designated by numeral 228. As shown, bore 225 is enlarged somewhat at its lower end, just above shoulder 227, for example as indicated at 229, and is formed with an enlargement at 230 at its junction with horizontal port 214. The check valve 228, in the typical form here shown, comprises a cylindrical body 231 formed with a plurality of longitudinally extending ribs 232 slidably fitting bore 225. The upper end of bore 225 is closed by means of a plug 233 screwed into an internally threaded socket 234 formed at its upper end.

Check valve 228 when in its lower position rests on the upper end of an adjustment member, here shown in the form of a vertical rod 235 adjustably mounted in casting 142, so as to provide a restricted clearance space between its lower end and shoulder 227 (see the dotted line position in Fig. 5). As here typically shown this rod 235 has a screwthreaded section 236 threaded within casting 142 and extending through the lower side thereof, having on its lower end a head 237 by means of which it may be turned. A lock nut 238 may be employed to tighten member 235 in adjusted position.

As previously mentioned, passageway 223 in end casting 141 may be identical with that just described, being provided with a check valve 228 and check valve adjustment member 235 exactly as described in connection with liquid passage 213 in end casting 142.

When liquid flows upwardly via passage 212 and engages the lower side of check valve 228, it lifts the latter as to the position illustrated in Figs. 2 and 5, permitting free flow of fluid through passageway 213 to horizontal port 214 leading to the right hand end of master valve bore 140. Similarly, upward flow of liquid against the check valve 228 in end casting 141 will elevate that check valve in a like manner to permit free flow of fluid via passageway 223 and port 224 into the left hand end of master valve bore 140. Check valves 228 normally assume their lowermost positions, in engagement with the upper ends of supporting rods 235. In such

positions downward flow of fluid therepast is checked or restricted, downward flow taking place through the spaces between check valve ribs 232 to the reduced portion 226 of the check valved passageway. The restriction effected by the check valve depends finally upon the elevation of the check valve above annular shoulder 227, so that downward flow through these passageways is easily adjustable simply by adjustment of rods 235. The purpose of this adjustment will be more fully explained at a later point in the specification.

Opening within master valve bore 140 are two ports 240 and 241 (Fig. 6), which are adapted to be controlled by piston sections 150 and 151, respectively, of master valve 115. Port 240 leads to the aforementioned passageway 211, while port 241 leads to a fluid passageway 242 that communicates with a fluid passageway 243 leading through end casting 141 to the aforementioned passage 222. Port 241 is so located that it is completely covered by piston section 151 of master valve 115 when the latter has moved toward the left (as viewed in Figs. 2 and 6) from its extreme right-hand position a distance sufficient that valve elements 153 and 151 are just ready to crack or separate from valve seats 166 and 174, respectively, this position being shown in full lines in Fig. 6. In a similar manner, port 240 is so located that it is completely closed by piston section 150 of valve 115 when said valve has moved from its extreme left-hand position a distance toward the right sufficient to bring valve elements 150 and 153 to the point of cracking from seats 175 and 165, respectively.

Pilot valve 186 is movable between the position shown in Fig. 3, in which its central valve element 190 is seated on valve seat 198a, and a position shifted toward the left, as viewed in said figure, in which valve element 190 is seated on valve seat 196a. As previously indicated, this pilot valve is operated by the working piston, means being provided for shifting the pilot valve just as the piston reaches or approaches the desired limiting positions of its stroke. Figs. 1 and 9 show a typical and present preferred linkage for accomplishing this result, though it will be understood that any other found simple or appropriate may be substituted if desired. As shown in Fig. 9, a vertical camming bar 250 is mounted on the upper end of piston 40 by means of a bracket arm 251, bar 250 extending downwardly alongside the cylinder. Bar 250 is received in a way 252 extending downwardly along the side of cylinder head 27 and through cover member 58 and ring 30. This bar 250 is confined within way 252 by a keeper plate 253 secured to the side of cylinder head 27 and overlying way 252. Upper and lower cam elements or plates 254 and 255 are secured to the outer face of bar 250, being mounted at opposite edges of said bar so as to be horizontally offset from one another (see Fig. 1). The inner side of keeper plate 253 is notched, as indicated in Fig. 9b, to accommodate cam elements 254 and 255. These cam elements 254 and 255 are provided with straight vertical edges 254a and 255a, respectively, and with tapered ends 254b and 255b, respectively, the tapered portions being on the lower end of upper cam element 254 and on the upper end of lower cam element 255.

These cam elements 254 and 255 engage follower rollers 256 and 257, respectively, rotatably mounted on the ends of the two arms of a bell crank 258, the latter being rigidly mounted on a

stub shaft 259 journaled in brackets 260 secured to the side of cylinder head 27. Tightly mounted on stub shaft 259 and extended horizontally therefrom is a swinging arm 261, which is connected by a vertical link 262 with one arm of a bell crank 263 pivotally mounted at 264 on fitting 26. The other arm of bell crank 263 is connected by the aforementioned link 191 to the pilot valve, as previously mentioned. The two rollers 256 and 257 borne by the two arms of bell crank 258 are on opposite sides of said arms and are in alignment with cam elements 254 and 255, in such manner that roller 256 will be exclusively engaged and operated by cam element 254 and roller 257 will be exclusively engaged and operated by cam element 255.

Fig. 9 shows piston 40 near the bottom limit of its stroke, and it will be understood that follower roller 256 has ridden up the incline of the tapered portion 254b of upper cam element 254, causing bell crank 258 to swing in a counter-clockwise direction, thereby moving link 262 in a downward direction and swinging bell crank 263 in a counter-clockwise direction, which results in moving link 191 in a direction to shift pilot valve 186 to the position illustrated in Fig. 3. Bell crank 258 having swung to the position illustrated in Fig. 9, cam bar 250 may move downwardly somewhat further before reversal occurs, roller 256 at this time riding on vertical edge 254a of cam element 254 and roller 257 riding on the vertical surface of bar 250, as will be clear from inspection of Fig. 9.

As the piston reverses direction and travels through its up stroke bell crank 258 remains in the position illustrated in Fig. 9, roller 257 continuing to rise on the surface of bar 250, and roller 256 maintaining its separation from bar 250 notwithstanding the fact that it is no longer supported in such position by cam element 254. In other words, during the up stroke the pilot valve operating linkage remains in the position of Figs. 1 and 9, friction in the various connections holding the linkage against displacement.

As the upper end of the stroke is approached, the inclined surface 255b of lower cam element 255 engages lower follower roller 257 and causes it to swing out, bell crank 258 moving in a clockwise direction until roller 256 is in engagement with the surface of bar 250, this causing the pilot valve linkage to withdraw pilot valve 186 to a position with central valve element 190 on seat 196a, between ports 196 and 197. The pilot valve linkage then remains in this position until, near the bottom of the down stroke, it is operated by cam element 254 to swing again to the position of Fig. 9.

The complete cycle of operations of the system may now be described. Assume that piston 40 is approaching the lower limit of its down stroke (Figs. 1 and 9), and that the pilot valve 186 has been moved to the position illustrated in Fig. 3. Master valve 115 will have been in its extreme right-hand position, as viewed in Fig. 2, and liquid will have been flowing from cylinder 16 by way of pipe 71 and the valve and pump system to tank 15 as follows: pipe 71, valve port 167, valve port 160, pump suction chamber 161, pump discharge passageway 164, valve port 177, valve port 168, and pipe 80 to tank 15. The pressure in the cylinder at this time is substantially that necessary to counterbalance the pump rod string, or just less than counterbalancing pressure, and the pump is raising the pressure of the liquid from that pressure to tank pressure.

The described shift of pilot valve 186 to the position of Fig. 3 as the lower limit of the piston stroke is approached allows high pressure liquid to flow from high pressure line 200, see Fig. 3, via passageway 201, valve port 196, port 197, passageways 210, 211, 212 (Fig. 5) passageway 213 (check valve 228 becoming elevated to the position of Fig. 5 by the pressure of the liquid), and port 214, to be discharged to valve bore 140 to act against the outer end of piston section 150 of the master valve.

This high pressure liquid, at a pressure of approximately $(R+C)$, acting against the outer end of the piston section 150 of the master valve, is opposed by liquid at a lower pressure, for example, the pressure at the intake side of the pump, acting against the outer end of the piston section 151 at the other end of the master valve. The master valve accordingly travels toward the left, the liquid ahead of piston section 151 being expelled as will presently be traced.

The sequence of operations and conditions passed through during such movement of the master valve from right to left is as follows: The liquid in piston bore 140 ahead of piston section 151 is expelled by way of port 224 and check-valved passageway 223 to passageway 222, and also, during the first portion of the stroke of the master valve, by way of port 241 and passageways 242 and 243 to passageway 222. From passageway 222 this liquid flows by way of passageway 221 (Fig. 6) into valve port 199, and from there flows to valve port 198, from which it is discharged by way of passageway 206 to pipe line 207 leading to the intake side of the pump. Check valve 228 in passage 223 being at this time, in its lowermost position, resting on rod 235, out-flow of liquid by way of port 224 and passageway 223 is considerably restricted. Out-flow by way of port 241, however, is comparatively free, port 241 being of such diameter as will permit the liquid to be rapidly expelled until it is covered by the piston section 151 of the master valve. This occurs, as previously described, when the master valve has moved toward the left a distance sufficient that cracking of valve elements 153 and 151 from seats 166 and 174 is imminent (Fig. 6). The travel of the master valve from the position of Fig. 2 to the position of Fig. 6 has not restricted the free flow of liquid from port 167 through port 160 to suction chamber 161, nor the flow of liquid from exhaust passageway 164 via port 177 into port 168 leading to the tank. The only effect of the travel of the master valve from the position of Fig. 2 to the position of Fig. 6 is, therefore, to close port 241 and restrict all further discharge of liquid from ahead of piston section 151 to port 224 and check valve passageways 223, thereby reducing the speed of travel of the master valve to a determined rate depending upon the adjustment of check valve 228.

Further travel of the master valve toward the left then causes separation of valve element 153 from seat 166, separation of piston section 151 from seat 174, and an approach of piston section 150 toward seat 175, with the following results:

Liquid from pump discharge passageway 164 begins to travel past valve element 153 directly to pump suction chamber 161, and at the same time liquid begins to flow from pump discharge passageway 163 past piston section 151 to port 176 and thence via port 160 to pump suction chamber 161. Tank 15 is in communication with pump intake chamber 161 via a gradually widening passageway, and the pressure in pump suction

chamber 161 accordingly increases from its original value of R toward tank pressure

$$\left(R + \frac{C}{2}\right)$$

The pressure in pump discharge passages 163 and 164 correspondingly increases from the original value of

$$\left(R + \frac{C}{2}\right)$$

toward a value of $(R+C)$, the pump operating to raise the head of the liquid by an amount equal to

$$\frac{C}{2}$$

The flow of liquid to the tank becomes less and less, both because of the increasing restriction between piston section 150 and valve seat 175, and because of the by-pass of an increasing quantity of liquid from port 168 directly back to suction chamber 161. At the same time, the flow of liquid from the cylinder becomes less and less, by reason of increasing flow of the high pressure liquid from pump discharge passageway 163 to port 167, and increasing restriction of the flow from port 167 to port 160 and suction chamber 161 due to approach of valve element 153 toward seat 165.

By the time the master valve has reached its central position (Fig. 2 in dotted lines), flow from the cylinder toward the pump and from the pump toward the tank has stopped, and downward travel of the working piston and upper end of the pump rod string has ceased, though the lower end of the string may continue downwardly for a short time due to inertia forces stretching the pump string. The total of the liquid flowing from discharge passageways 163 and 164 into ports 167 and 168 at this time goes past valve seats 165 and 166 to port 160 and pump suction chamber 161, and no liquid is taken in from pipe 71 connecting with the cylinder nor being discharged through pipe 80 to the tank. At this time the pressures in tank 15 and in the cylinder are substantially equalized and the pump is merely circulating liquid locally.

As the master valve then continues to travel further toward the left, i. e., from the position indicated in dotted lines in Fig. 2 toward the position in which valve element 153 just closes the passageway between ports 167 and 160 and in which piston section 150 just closes the passageway between ports 177 and 168, the following occurs: Because of the approach of piston section 150 toward valve seat 175, flow from pump discharge passage 164 is gradually reduced to zero, while at the same time movement of valve element 153 away from seat 166 and toward seat 165 initiates a flow from port 168 toward suction chamber 161 and gradually decreases the flow from port 167 toward said suction chamber to zero. At the same time, further recession of piston section 151 from seat 174 is gradually increasing flow from pump discharge passage 163 to port 167. It follows that as the master valve travels toward the left from its central position (dotted lines in Fig. 2), liquid begins to flow from the tank via ports 168 and 160 to pump suction chamber 161, and liquid begins to be pumped toward the cylinder by way of discharge passage 163 and port 167. The pressure of the liquid in the cylinder is not yet up to $(R+C)$, such as is required to lift the entire pump string and column of well fluid, but the liquid being pumped

toward and into the cylinder at this time is increasing toward that value and is under sufficient pressure to begin slowly to lift the working piston and rod string, which have not yet assumed the fluid column load. As a matter of fact, it may occur that the working piston will already have begun a slow upward movement, due to increase in pressure of the liquid within the cylinder because of opening of communication between the cylinder and tank, and this may be especially noticeable when the tank is operating under an overcharge of pressure. In the course of the travel of the master valve toward the position in which valve elements 153 and 150 just reach seats 165 and 175, respectively, the lower end of the rod string terminates its downward travel and begins to elevate by reason of resilient contraction, causing the travelling valve to close. Further upward travel of the working piston under the gradually increasing pressure in the cylinder then stretches out the rod string, until the full load of the rod string plus the weight of the fluid column is on the piston. By this time valve element 153 is on its seat 165, flow from passage 164 is interrupted by piston section 150, pump intake chamber 161 is in unrestricted communication with tank 14, and pump discharge passageway 163 is in unrestricted communication with the working cylinder and therefore supplies liquid to the cylinder at full high pressure $(R+C)$, so that the piston then elevates the entire rod string and column of well fluid. The speed of this travel of course depends upon the speed at which driving pump P is operated.

As previously noted, the master valve moves comparatively quickly from its extreme right-hand position to the position of Fig. 6, where outlet port 241 is closed, at which point its speed is reduced as determined by the setting of check valve adjustment member 235. The time within which the deceleration and acceleration of the working piston and upper end of the pump string must occur depends upon this reduced speed of travel of the master valve, and hence upon the setting of the controlling check valve, which may thus be readily set to provide a smooth and easy deceleration, and an acceleration which is timed with the rebound of the pump string, the action being such that the stretch is substantially out of the rod string and the travelling valve is closed when the full output pressure of the circulation pump is applied to the piston and the piston and rod string are accordingly up to maximum speed. The timing thus made possible is conducive to smoothest and most efficient performance, and to the elimination of undesirable whipping of the rod string as the load of the well fluid is assumed.

Master valve 115 having been moved to its extreme left-hand position, liquid will be taken from tank 15 and delivered to the cylinder to elevate the working piston and pump rod string, as mentioned above. Near the upper end of the stroke, cam element 255 of car bar 250 strikes follower roller 257 and swings bell crank 258 in a clockwise direction, thereby operating the pilot valve linkage to move pilot valve 186 in a left-hand direction, as viewed in Fig. 3, at which time valve element 190 is seated on seat 196a, valve port 199 is closed by piston section 188, valve port 198 is in communication with valve port 197, and valve port 196 is in communication with valve port 195. At this time, high pressure liquid flows by way of high pressure line 200 and passageway 201 to valve port 196, from there to

port 195, and thence via passageways 220 (see Fig. 6), 222, 223, and 224 to be discharged against the end of piston section 151 of master valve 115. It will be understood that such passage of high pressure liquid upwardly through passageway 223 will elevate the corresponding check valve 228 to provide a free passageway for flow of high pressure liquid to and against piston section 151. At the same time, liquid ahead of piston section 150 is expelled by way of port 214, passageway 213 (then restricted by its check valve 228, which is at such time in its lowermost position) thence flowing by way of passageways 212, 211, 210, port 197, past seat 198a (from which valve element 190 is at this time removed), to port 198 leading via passage 206 to pipe line 207 discharging to the intake side of pump P.

Liquid also flows from ahead of piston section 150, during the initial portion of the stroke of the master valve toward the right, by way of port 240 discharging directly into passageway 211.

When the master valve has moved sufficiently toward the right that cracking of valve elements 153 and 150 from seats 165 and 175 is imminent, port 240 is just closed by piston section 150, so that the master valve is checked in speed and the remainder of its travel toward the right reduced to a rate permitted by the degree of restriction of passageway 213 by its check valve element 228.

As before, no change in operation, other than reduction in speed of travel of the master valve, is effected by the initial travel of the master valve prior to cracking of valve element 153 from seat 165. As valve element 153 separates from seat 165 and piston section 150 separates from seat 175, however, liquid from pump discharge passageway 163 begins to travel past valve element 153 directly back to pump suction chamber 161, and liquid begins to flow from pump discharge passageway 164 past piston section 150 to port 168 and thence via port 160 to pump suction chamber 161. The flow of liquid to the cylinder becomes less and less, both because of increasing restriction between piston section 151 and valve seat 174, and because of the by-pass of an increasing quantity of liquid from port 167 past valve seat 165 to pump intake chamber 161. At the same time, the flow of liquid from the tank toward the pump becomes less and less, by reason of increasing flow of liquid from pump discharge passageway 164 to port 168, and decreasing flow from port 168 via port 160 to suction chamber 161 due to the approach of valve element 153 toward seat 166. Travel of valve element 153 toward the right within port 160 gradually restricts the passageway from the tank past seat 166 and through port 160 to the pump intake chamber, while at the same time increasing the size of the passageway between port 167 and pump suction chamber 161, this having the effect of gradually reducing the intake pressure existing within chamber 161 from

$$\left(R + \frac{C}{2}\right)$$

to R during the course of travel of piston element 153 to the full line position of Fig. 6, the discharge pressure of the pump falling correspondingly to a value of

$$\left(R + \frac{C}{2}\right)$$

By the time the master valve has reached its

central position (dotted lines in Fig. 2) flow from the tank and toward the cylinder has stopped, and upward travel of the working piston and the upper end of the pump rod has ceased. The travelling valve of the well pump will still be closed, and because of the communication now established between the cylinder and tank, the pressure in the cylinder will fall below pressure $(R + C)$, whereupon the piston and upper end of the rod string will begin slowly to descend. The total of the liquid flowing from discharge passageways 163 and 164 into ports 167 and 168 at this time goes past valve seats 165 and 166 to port 160 and pump intake chamber 161, and no liquid is taken in from pipe 80 connecting with the tank nor discharge through pipe 71 to the cylinder.

As the master valve then continues to travel further toward the right, i. e., from the position indicated in dotted lines in Fig. 2 toward the position shown in full lines in Fig. 6, liquid is drawn from the cylinder at a gradually increasing rate and liquid is pumped toward the tank at a correspondingly increasing rate. The downward travel of working piston thus accelerates, the rate of acceleration being governed by the speed of travel of the master valve, which is regulated as previously explained. After the master valve position of Fig. 6 is reached, liquid is pumped from the cylinder toward the tank at normal operating speed (depending upon the speed of operation of the pump), the piston and pump string then descending at maximum speed. The lowering of the pump string causes the travelling valve to open, after which pump P takes liquid from the working cylinder, raises it to the pressure of tank 15, and delivers it to the tank.

A novel and important feature of my pump and valve system is that during reversal of the direction of pumping stroke, the working cylinder is in constant communication with the tank, so that the usual jarring and water hammer occasioned by complete cut-off of the line to the working cylinder is entirely eliminated and instead the piston in the working cylinder is constantly cushioned by the air pressure in tank 15. Attention is particularly directed to the novel action of the pump and master valve during reversal of the direction of flow between tank and cylinder. As the master valve moves, it gradually decreases flow, e. g., from the pump through the valve chamber to the cylinder, and gradually correspondingly decreases flow from the tank through the valve chamber to the pump. At the same time, a local ring circulation from the pump through the valve chamber and back to the pump is initiated, which increases until the entire output of the pump is diverted in this manner. Further travel of the master valve then causes liquid to be diverted from this local ring circulation and delivered to the tank, and at the same time causes the pump to begin to take liquid from the cylinder. In this phase of the cycle the master valve gradually decreases the local ring circulation to zero, and gradually increases the circulation from the cylinder to the pump and from the pump to the tank to maximum. There is thus a gradual transition from maximum flow in one direction to maximum flow in the other, without ever closing off communication between tank and cylinder, the pump circulating liquid locally through the valve during the transition, and all of the liquid discharge by the pump taking part in this local ring circulation through the valve at the time flow between

tank and cylinder momentarily ceases preparatory to reversing its direction.

One of the primary features of the invention is the provision of means for regulating the rates of deceleration and acceleration of the working piston, independently of its speed of travel between the deceleration and acceleration zones. This enables adjustment of the acceleration and deceleration to rates best suited to any given pump rod string, as well as permitting operation at a comparatively high linear pumping speed.

And an outstanding feature of advantage in the pumping system of the invention is the use of a novel, simple and effective fluid pressure means for operation of the main or master valve, this fluid pressure valve operating means avoiding all dead center problems and complicated and unsatisfactory mechanical expedients for throwing the main valve past dead center position, as inevitably encountered in purely mechanical valve operating mechanisms.

I have now described a present preferred embodiment of my invention in considerable detail; it will be understood, however, that this is for illustrative purposes only, and so that the principles inherent in my invention will be fully understood, the invention being susceptible to various changes in design, structure and arrangement without departing from the spirit and scope of the invention or of the appended claims. I claim:

1. A reversible flow pump and valve unit, comprising a housing, a pair of meshing gear type pump rotors rotatably mounted in said housing on parallel axes, a pair of arcuate walls in said housing positioned closely adjacent opposite sides of said pair of pump rotors, corresponding end edges of said arcuate walls being spaced apart to permit liquid flow therebetween, a valve bore in said housing positioned to one side of said pair of pump rotors and extending in a direction substantially transversely of said rotor axes, a series of five axially spaced annular ports around and communicating with said valve bore, walls in said housing establishing direct communication between the central one of said annular ports and a space at the adjacent side of the two pump rotors between the two adjacent end edges of said arcuate walls, said housing including walls which, together with the outer surface of said arcuate walls, define two passageways extending between a space at the opposite side of the two pump rotors, between the two adjacent end edges of said two arcuate walls, around the outer sides of said two arcuate walls, and communicating with the two outside annular ports, passageways extending inside said housing and communicating with the two annular ports between said middle and outside ports, and a valve reciprocable in said valve bore between two positions and adapted to establish communication between the central annular port and one of the annular ports next outside and communication between the other annular port next outside and the adjacent outside annular port when in one of said positions, and communication between the central annular port and the other of the annular ports next outside and communication between the other annular port next outside and the adjacent outside annular port when in the other of said positions.

2. A reversible flow pump and valve unit, comprising a housing, a pair of meshing gear type pump rotors rotatably mounted in said housing

on parallel axes, a pair of arcuate walls in said housing positioned closely adjacent opposite sides of said pair of pump rotors, corresponding end edges of said arcuate walls being spaced apart to permit liquid flow therebetween, walls closely spaced to the ends of said pump rotors and completing the enclosure of said pump rotors except for the spaces between said end edges of said arcuate walls, a valve bore in said housing positioned to one side of said pair of pump rotors and extending in a direction substantially transversely of said rotor axes, a series of five axially spaced annular ports around and communicating with said valve bore, walls in said housing establishing direct communication between the central one of said annular ports and a space at the adjacent side of the two pump rotors between the two adjacent end edges of said arcuate walls, said housing including walls which, together with the outer surface of said arcuate walls, define two passageways extending between a space at the opposite side of the two pump rotors, between the two adjacent end edges of said two arcuate walls, around the outer sides of said two arcuate walls, and communicating with the two outside annular ports, passageways extending inside said housing and communicating with the two annular ports between said middle and outside ports, and a valve reciprocable in said valve bore between two positions and adapted to establish communication between the central annular port and one of the annular ports next outside and communication between the other annular port next outside and the adjacent outside annular port when in one of said positions, and communication between the central annular port and the other of the annular ports next outside and communication between the other annular port next outside and the adjacent outside annular port when in the other of said positions.

3. A reversible flow pump and valve unit, comprising a housing, a pump embodying a pair of interacting pump rotors rotatably mounted in said housing on parallel axes, at pair of arcuate walls in said housing positioned closely adjacent opposite sides of said pair of pump rotors, corresponding ends of said arcuate walls being spaced apart to permit liquid flow therebetween, a valve bore in said housing positioned to one side of said rotors, a series of five axially spaced ports opening within said valve bore, a passageway in said housing establishing communication between the central one of said ports and the liquid flow space between corresponding ends of said two arcuate walls, passageways in said housing extending from the liquid flow space between the other corresponding ends of said two arcuate walls around the outer sides of said two arcuate walls and communicating with the two outside ports, external liquid connections with the two ports between said central and outside ports, and a valve reciprocable in said valve bore between two positions and adapted to establish communication between the central port and one of the ports next outside and communication between the other port next outside and the adjacent outside port when in one of said positions, and communication between the central port and the other of the ports next outside and communication between the other port next outside and the adjacent outside port when in the other of said positions.

4. A reversible flow pump and valve unit, comprising a housing, a pump embodying a pair of interacting pump rotors rotatably mounted in

said housing on parallel axes, a pair of arcuate walls in said housing positioned closely adjacent opposite sides of said pair of pump rotors, corresponding ends of said arcuate walls being spaced apart to permit liquid flow therebetween, a valve bore in said housing positioned to one side of said pair of pump rotors and extending in a direction substantially transversely of the rotor axes, a series of five axially spaced ports opening within said valve bore, a passageway in said housing establishing communication between the central one of said ports and the liquid flow space between corresponding ends of said two arcuate walls, passageways in said housing extending from the liquid flow space between the other corresponding ends of said two arcuate walls around the outer sides of said two arcuate walls and communicating with the two outside ports, external liquid connections with the two ports between said central and outside ports, and a valve reciprocable in said valve bore between two positions and adapted to establish communication between the central port and one of the ports next outside and communication between the other port next outside and the adjacent outside port when in one of said positions, and communication between the central port and the other of the ports next outside and communication between the other port next outside and the adjacent outside port when in the other of said positions.

5. A reversible flow pump and valve unit, comprising a housing, pump rotor means in said housing, curved walls in said housing defining the periphery of a rotor chamber for said rotor means, spaced end walls in said housing closing the ends of said rotor chamber, liquid intake and discharge ports communicating with said rotor chamber, a valve bore in said housing positioned to one side of said pump rotor means, a series of five axially spaced ports opening inside said valve bore, a liquid passageway in said housing extending around the outside of said curved rotor chamber walls and establishing communication between one of said rotor chamber liquid ports and the two outside valve bore ports, a liquid passageway in said housing establishing communication between the other of said rotor chamber liquid ports and the central one of the valve bore ports, external liquid connections with two valve bore ports between said central and outside ports, and a valve reciprocable in said valve bore between two positions and adapted to establish communication between the central port and one of the ports next outside and communication between the other port next outside and the adjacent outside port when in one of said positions, and communication between the central port and the other of the ports next outside and communication between the other port next outside and the adjacent outside port when in the other of said positions.

6. A reversible flow pump and valve unit, comprising a housing, pump rotor means in said housing, curved walls in said housing defining the periphery of a rotor chamber for said rotor means, spaced end walls in said housing closing the ends of said rotor chamber, liquid intake and discharge ports communicating with said rotor chamber, a valve bore in said housing positioned to one side of said pump rotor means and extending substantially transversely thereof, a series of five axially spaced ports opening inside said valve bore, a liquid passageway in said housing extending around the outside of said curved rotor chamber walls and establishing communication between one of said rotor chamber liquid ports and the two outside valve bore ports, a liquid passageway in said housing establishing communication between the other of said rotor chamber liquid ports and the central one of the valve bore ports, external liquid connections with two valve bore ports between said central and outside ports, and a valve reciprocable in said valve bore between two positions and adapted to establish communication between the central port and one of the ports next outside and communication between the other port next outside and the adjacent outside port when in one of said positions, and communication between the central port and the other of the ports next outside and communication between the other port next outside and the adjacent outside port when in the other of said positions.

7. A reversible flow pump and valve unit, comprising a housing, a pump embodying a pair of interacting pump rotors rotatably mounted in said housing on parallel axes, a pair of arcuate walls in said housing positioned closely adjacent opposite sides of said pair of pump rotors, a valve bore in said housing positioned to one side of said rotors, a series of five axially spaced ports opening within said valve bore, a passageway in said housing establishing communication between the central one of said ports and one side of said pair of interacting pump rotors, passageways in said housing around the outer sides of said arcuate walls and communicating at their one ends with the other side of said pair of interacting pump rotors, and at their other ends with the two outside ports, external liquid connections with the two ports between said central and outside ports, and a valve reciprocable in said valve bore between two positions and adapted to establish communication between the central port and one of the ports next outside and communication between the other port next outside and the adjacent outside port when in one of said positions, and communication between the central port and the other of the ports next outside and communication between the other port next outside and the adjacent outside port when in the other of said positions.

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