

April 26, 1960

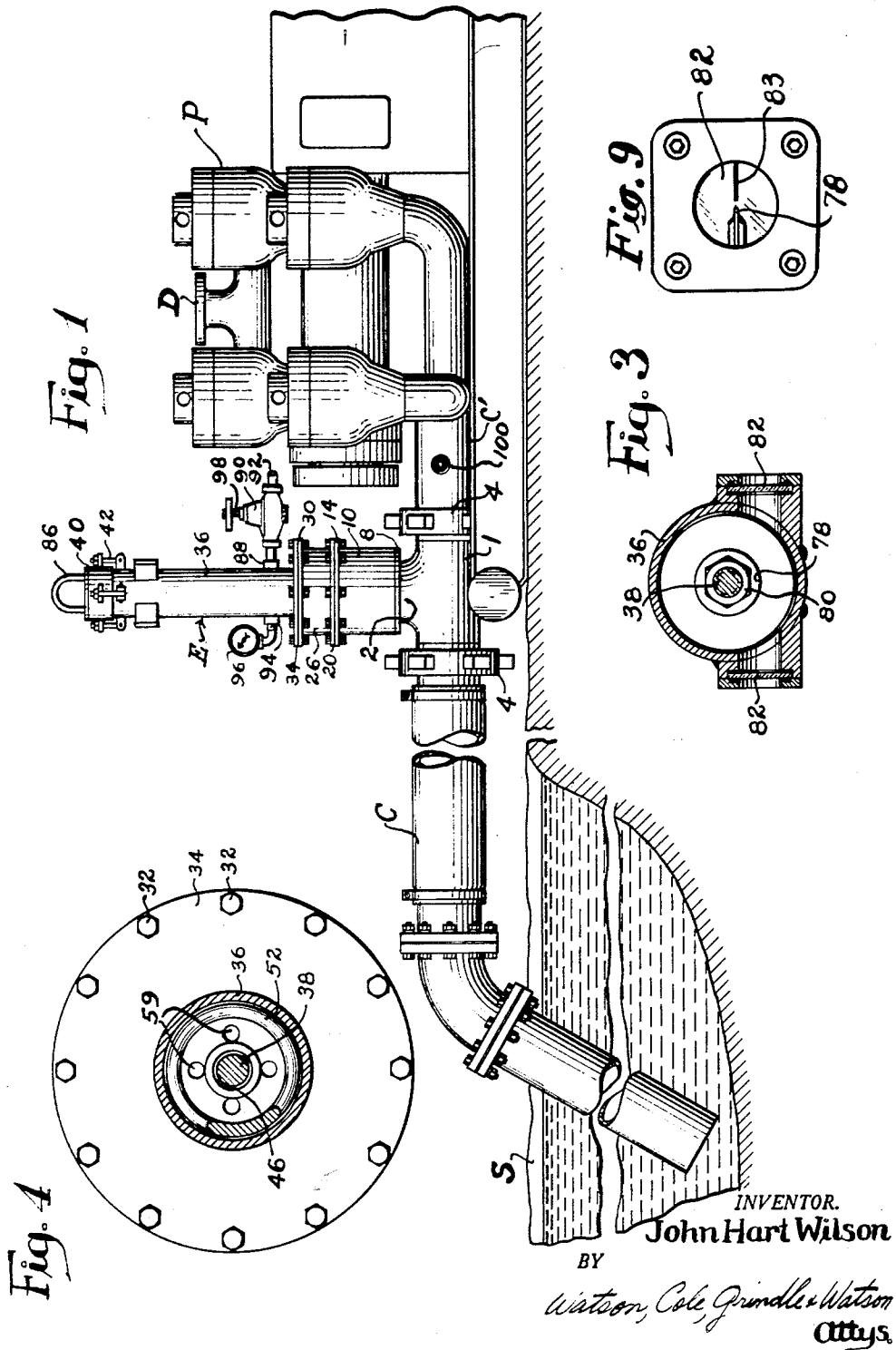
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2,934,025

SUCTION FLOW EQUALIZER FOR MUD PUMPS

Filed Nov. 8, 1955

3 Sheets-Sheet 1



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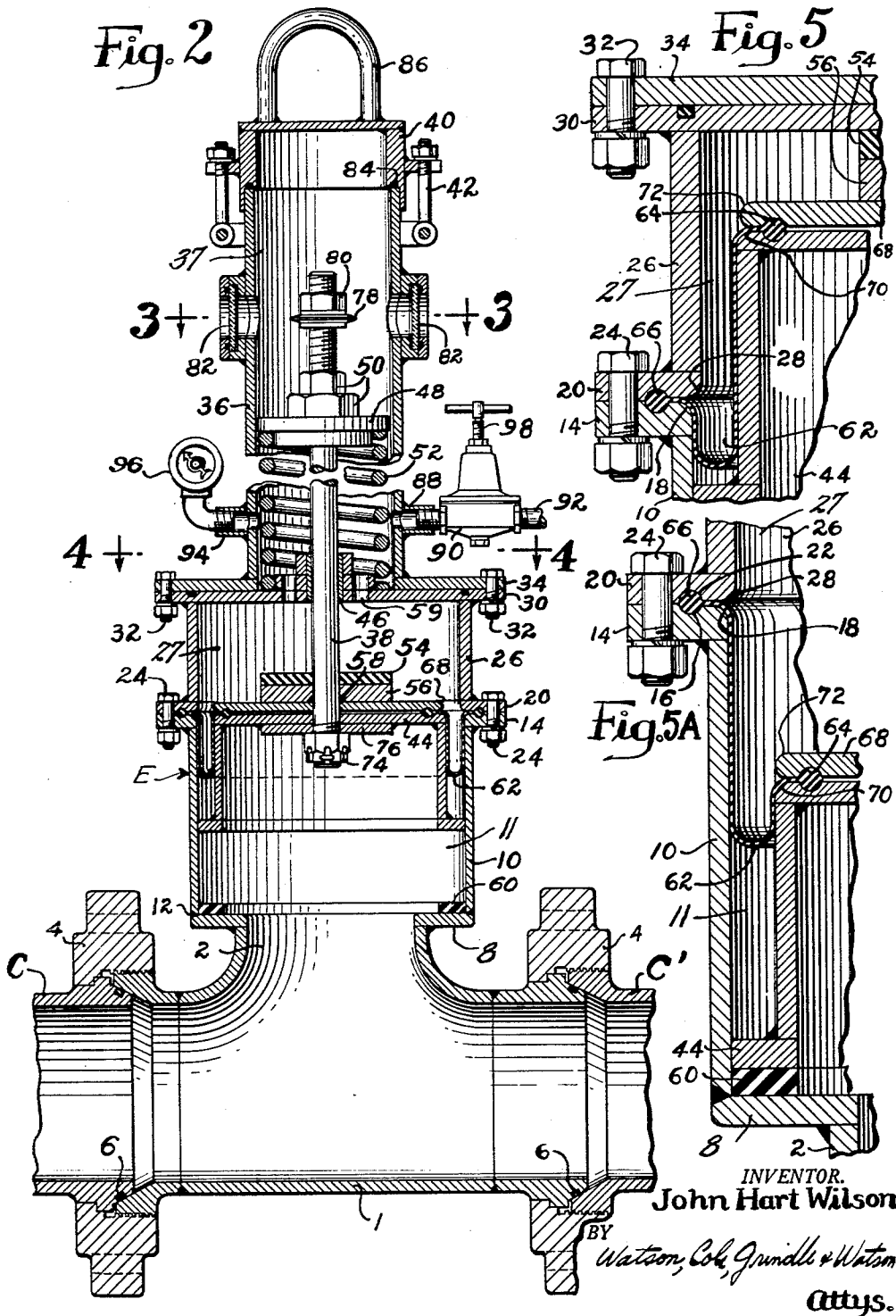
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3 Sheets-Sheet 2



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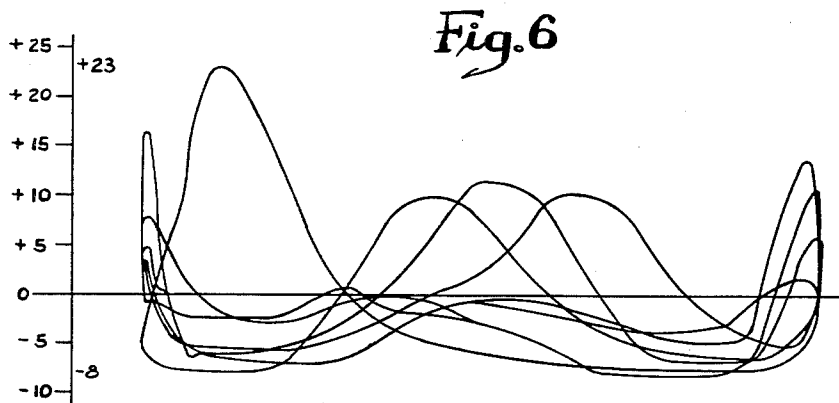
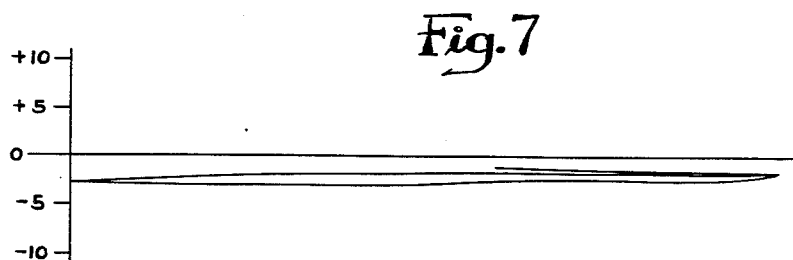
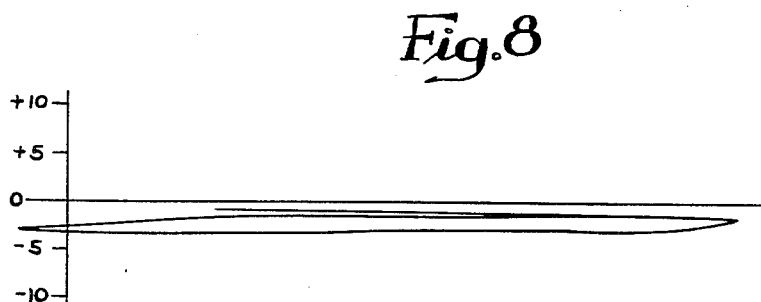
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SUCTION FLOW EQUALIZER FOR MUD PUMPS

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3 Sheets-Sheet 3



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SUCTION FLOW EQUALIZER FOR MUD PUMPS

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8 Claims. (Cl. 103—223)

This invention relates to the supplying of drilling fluid to oil wells; and is directed more particularly to a device for equalizing the suction flow of such fluids to the pump, which in turn, transfers the fluid to the well.

It is well known in the drilling art, that it is desirable to force a dense, viscous fluid, known as drilling "mud" into the bore of the well during the drilling thereof, the mud being pumped from a sump or reservoir and ultimately returned thereto. The function of the mud is to cool and lubricate the rotary drill bit, to remove borings as they are produced, and to coat or seal the face of the bore. Normally the mud consists of suspensions of natural clays weighted with such materials as barytes, hematite, and the like to increase the density, together with various natural or artificial gel-forming agents and other adjuncts.

Obviously, the high density and viscosity of such fluids causes them to be extremely difficult to pump, at least by means of conventional piston-type pumps. Thus, with the common duplex, double acting pump, and a fluid weighing ten or more pounds per gallon, the maximum pumping speed that can be attained, is about sixty strokes per minute. If this speed is exceeded, cavitation results, which is to say, that the face of the piston on the suction stroke pulls away from the mud in the cylinder, creating a void between the mud and the piston face, so that, on the return stroke, the piston rams into the mud, subjecting the piston and its driving mechanism to tremendous impact or shock, which, if repeated a sufficient number of times, will actually pound the pump to pieces. It is the opinion of the inventor that if it were possible to move the fluid from sump to pump at a uniform rate of flow, the speed of the pump, and accordingly, its capacity could be greatly increased, possibly as much as 50% to 60%.

Many devices to achieve this result have been made such as flooded suctions; or pressure charging has been used, and they do permit an increase in speed, but flooded suctions are not always possible, and pressure charging has many disadvantages.

It is therefore an object of this invention to provide a device adapted to be interposed in the suction conduit of a mud pump or the like, between the sump and pump, which will tend to equalize the flow of the fluid in the conduit.

More particularly, it is an object of the invention to provide a suction flow equalizing device for piston pumps comprising a housing, movable means associated with said housing to form a variable volume chamber in communication with the suction line of a pump. Spring means for acting upon said movable means to urge the latter in one direction, the variable fluid pressure acting on said movable means to urge said movable means in the other direction. Variable pressure means connected to said chamber for maintaining fluid pressure thereon to adapt a pump suction system to various conditions of operation.

A further object of the invention is the provision, in

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a device of the character described, of indicating means to facilitate the initial setting of the device, i.e. its adjustment to the particular pump with which it is to be used, and for observing the operation of the device from time to time.

Other objects and advantages will be revealed by the following detailed description, when read in conjunction with the accompanying drawings, in which like reference characters designate like parts in the several views thereof, and in which:

Fig. 1 is a fragmentary side elevational view of a mud pump, with parts broken away and shortened, and showing a suction line leading to the mud pump, with a suction flow equalizer interposed within the mud pump suction line, and showing the relationship of the suction flow equalizer to the mud pump;

Fig. 2 is an enlarged vertical sectional view through the suction flow equalizer and through a portion of the suction line leading to the pump;

Fig. 3 is a sectional view taken on the line 3—3 of Fig. 2, looking in the direction indicated by the arrows;

Fig. 4 is a sectional view taken on the line 4—4 of Fig. 2, looking in the direction indicated by the arrows;

Fig. 5 is an enlarged fragmentary sectional elevational view of the equalizer cylinder showing the diaphragm in its uppermost position; Fig. 5A shows diaphragm of Fig. 5 in its lowermost position;

Fig. 6 is a reproduction of an actual synchronized pressure graph, as traced on a card by a steam indicator graph instrument, before the installation of the suction flow equalizer in the suction flow line, with the pump working fifty-four strokes per minute;

Fig. 7 is a view similar to Fig. 6, but of a card graph made on the same pump operating under the same conditions, but after the installation of the flow equalizer on the suction line of the pump;

Fig. 8 is a view similar to Fig. 7, but the graph is taken from a pump using a slightly smaller diameter cylinder, and with the speed of the pump being increased to seventy-five strokes per minute; and

Fig. 9 is an elevational view showing one of the windows and the window frame, together with a pointer, of the spring housing.

In accordance with the present invention it is proposed to place in direct communication with the suction conduit, on the suction side of the pump and as close to the pump as possible, a device which functions as an accumulator, to receive a portion or charge of the mud during one part of the pumping cycle, and to release that portion during another part of the pumping cycle.

This device comprises a cylinder, which cylinder opens into the suction conduit of the pump, a piston in the cylinder, a diaphragm seal between the piston and the cylinder, and a compression spring acting on the piston in a direction to draw mud fluid into the receiving chamber 11; and air pressure in the upper chamber 27 pushing downward on the diaphragm 62 and piston 44. The upper chamber 27 is sealed at the top and is connected to an air supply 92 through a regulator 90 to maintain a constant reduced air pressure, in this chamber.

Turning now to a detailed description of the invention, in the drawings, the letter S designates a sump or reservoir containing a suitable supply of drilling fluid or mud. A suction conduit C extends from the sump S to the pump P, the discharge port D of which is connected to a suitable pipe (not shown), which extends to the bore hole of the well (not shown). The pump is preferably of the reciprocating, duplex, double acting type, and is well known in the art of drilling. The two crank units do not operate 180 degrees out-of-phase, but approximately ninety degrees out-of-phase, so that one unit is just beginning a stroke when the other unit is at about

the mid-point of its stroke. Consequently, at least one of the units is actively pumping at all times, and the rate of flow tends to be much more uniform than would be the case with a single unit.

Interposed in the suction conduit C, as near the pump P as possible, is the accumulator or suction flow equalizer designated generally by the letter E. To form the equalizer, a pipe T 1 is situated within the suction conduit C, with the lateral opening 2 thereof being connected to receiving chamber 11 which is formed by cylinder 10, with the cylinder preferably extending vertically, the arms of the T being connected to the suction side of the pump P and to the conduit C leading to the sump S, respectively, by means of the respective threaded union couplings 4, with sealing gaskets 6 (Fig. 2) interposed between the respective adjacent faces of the T and conduits. The remote end of the outlet 2 is provided with an outwardly extending flange 8 on which cylinder 10 is secured, as by welding as indicated at 12. The upper end of cylinder 10 has a flange 14 thereon, which flange has an annular groove 16 formed therein, as is best brought out in Figs. 5 and 5A. The flange 14 has the inner corner 18 thereof rounded, as will be more fully explained hereinafter.

A flange 20 has an annular groove 22 formed therein, which annular groove is complementary to the annular groove 16 of the flange 14. The flange 20 is superposed on the flange 14 so that holes formed in the respective flanges will receive bolts 24. A short length of cylindrical tubing 26 is secured to flange 20, so that the tubing 26 will be in axially aligned relation with the cylinder 10. The flange 20 has a rounded corner 23 thereon, as will be more fully explained hereinafter. The upper end of the cylindrical tubing 26 is closed by a plate 30 of larger diameter than that of tubing 26, providing an outturned flange, which flange has circumferentially spaced holes therein to receive bolts 32 which extend therethrough and through the out-turned flange portion of plate 34, which is secured to the lower end of cylindrical spring housing 36.

The respective plates 30 and 34 have holes therein, which holes are axially aligned with the cylinders 10 and 26, so as to permit the piston rod 38 to pass therethrough. The cylindrical spring housing 36 is closed at its remote end by a cap 40 which is held in place by detachable connecting bolts 42.

The term "housing" as employed in the appended claims is intended to include the structure jointly exemplified by the elements 10, 26, 36 and 40.

Fitted within the bore of cylinder 10, is an inverted cup-shaped piston head 44, which is secured to piston rod 38, which piston rod 38 extends upward through axial bushing 46 within plates 30 and 34. A circular plate 48 is mounted near the upper end of piston rod 38 and is held in adjusted position with respect thereto by means of nuts 50, which nuts are screw-threaded onto rod 38. The circular plate 48 is slightly smaller in diameter than the diameter of the cylindrical opening in the cylindrical spring housing 36, so the plate will perform the dual function of guiding the upper end of the piston rod, and will serve as a retainer for one end of a compression spring 52 which functions as a resilient means urging the diaphragm 62 and piston 44 in a direction opposite to the direction in which these parts are urged by the gas or air pressure hereinafter described in more detail. The other end of the compression spring 52 rests upon the upper face of plate 30.

The compression spring 52 is a spring of a strength to give the correct calibrated suction equalization, as will be more fully brought out hereinafter. The spring is positioned so as to urge the plate 48 and piston rod 38 upward. A cushion disc 54 is provided on top of plate 56, which plate is secured to the piston rod 38, as by welding, as indicated at 58. The cushion disc 54 is preferably made of an elastomer material so as to serve as

a cushion for the piston head in its extreme upward position. Holes 59 which are formed through plates 30 and 34, form connecting passages between chambers 27 and 37.

An annular ring 60 is positioned within cylinder 10 on flange 8 so as to serve as a cushion member for piston 44, when said piston is in the lowermost position. In this manner the piston 44 is cushioned at each end of the cylinder, so as to absorb shock, thereby preventing the piston from coming to an abrupt stop.

A diaphragm 62 is employed to enhance the effectiveness of the piston 44 and to form a positive seal between the piston 44 and the cylinder 10, with a minimum of friction loss. The diaphragm is substantially annular in form, and is preferably constructed of a combination of a fabric and an elastomer, with the inner and outer edges 64 and 66 of the diaphragm 62 taking the form of large O-shaped beads, as will best be seen in Figs. 5 and 5A. The inner O-shaped bead edge fits in complementary grooves formed in the element comprising the upper face of the cup-shaped piston 44 and in the lower face of a plate 68, as will best be seen in Figs. 5 and 5A, while the outer O-shaped bead fits in the previously described complementary grooves 16 and 22. The piston 44 is spaced inward from the cylinder 10 so as to form an annular space therebetween, which space may be from one-half to three-quarters of an inch in width, so that the diaphragm may work freely without sharp bending, which would result in damage to the diaphragm 62 which spans the annular space between the cylinder 10 and the piston 44. In this manner the piston 44 may work through a relatively long stroke with a minimum of fatigue to the diaphragm 62, which will prolong the life of the diaphragm materially. The particular type of diaphragm used is of a character which does not stretch materially, but performs a positive sealing action between the piston 44 and the cylinder 10, and obtains the necessary movement by a bellows or rolling action of the diaphragm 62 during the movement of the piston 44. The circumferential edges of the upper portion of piston 44 and plate 68, which are in contact with the diaphragm 62, have rounded corners 70 and 72, respectively, as do the inner edges of flanges 14 and 20, respectively, thereby relieving the diaphragm 62 of a substantial amount of fatigue at these points.

A nut 74 bears against a plate 76, which plate is welded to piston 44 so as to hold the piston securely against longitudinal movement with respect to piston rod 38.

The upper end of piston rod 38 is threaded to receive a correspondingly threaded pointer 78, which may be adjusted precisely therealong, which pointer is annular and is held in fixed adjusted position on piston rod 38 by means of a lock nut 80. The cylindrical spring housing 36 has a transparent sight window 82 on each side thereof in transverse aligned relation and preferably offset with respect to the center. Each sight window 82 has a transverse horizontal mark thereon to enable the correct aligning of the adjustable pointer 78 with the mark, when the piston rod 38 is correctly positioned, that is, with the piston 44 at the mid-point of the travel of cylinders 10—26, which is the preferred setting for the pointer 78 to align with transverse mark 83, to give the maximum working movement of piston 44 in each direction. It is preferably to have the sight windows 82 fitted between gaskets so that air pressure may be maintained within the cylindrical spring housing 36.

A gasket 84 is provided intermediate the upper end of the cylindrical spring housing 36 and the lower end of cap 40 so as to form a seal therebetween. The cap 40 has a bail 86 thereon for convenience in handling the device with a hoist line or the like.

An inlet opening 88 is provided in a side of housing 36 for introduction of air under pressure, through an adjustable pressure regulator 90, from a line 92 used in

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supplying air to the housing for impressing a positive pressure on diaphragm 62, as will be more fully explained hereinafter.

A threaded connection 94 connects housing 36 with a pressure gauge 96, which enables the determining of the pressure within the housing 36. An adjustment screw 98 is provided to adjust the pressure within housing 36 to the desired degree, irrespective of the pressure in the line 92.

Operation

In the pumping of drilling fluid with a two cylinder, double-acting pump, which pump is usually known as a duplex pump, four fluid pressure impulses and four fluid suction impulses are had on each revolution of the crankshaft of the pump, and since the intake of the suction line or conduit C is a substantial distance from the intake valve of the pump P, and with the viscosity of the drilling fluid being considerably in excess of that of water, a lag in the movement of the drilling fluid is present. However, once this heavy fluid starts moving it possesses considerable inertia.

In the operation of a conventional pump during part of the cycle, two pistons are sucking mud, and this amount of suction accelerates the flow of mud in the suction conduit C to a velocity sufficient to take care of the demand of the two pistons. As may be seen from Fig. 6, this requires up to eight pounds suction pressure or vacuum to accelerate this flow. The mud hardly reaches this velocity until one of the cranks reaches dead center and that piston stops sucking mud. When this occurs, the mud flowing in conduit C is of a greater amount than is required for the suction of one piston, therefore, something must decelerate the flow of mud in conduit C to the amount required by the suction of one piston. The only thing that can decelerate this flow is positive pressure, and this positive pressure actually builds up in conduit C, upon the closing of an inlet valve, as shown by the indicator card, Fig. 6, and in one instance, as shown by this card, the pressure built up to twenty three pounds above atmospheric pressure. This repeats four times each revolution of the pump. If the pump runs too fast the fluid will fail to follow the piston on the suction stroke, and on the return stroke, the piston will meet the fluid coming into the cylinder at considerable velocity. This will cause a pounding action, which, if continued, will actually destroy the pump, or some parts thereof.

For the purpose of illustrating the method of operation of the pump, it may be assumed that the flow equalizer E is installed in the suction line of a pump, and that the spring 52 is a 100 lb. spring, that is, a spring such that an additional 100 lbs. of pressure is required to compress the spring each additional inch within the elastic limits thereof, and that the spring 52 is adjusted so that, with the piston at the top of the stroke, the spring will have a pressure of 600 lbs. and at the bottom of the 6" stroke, a pressure of 1200 lbs. If an air pressure of about 8 lbs. above atmospheric is exerted upon the diaphragm 62 on the upper side, and 3 lbs. suction or negative pressure acts on the lower side of the diaphragm, a total of 11 lbs. p.s.i. acts upon the diaphragm 62, to which the piston rod 38 is connected through piston 44. If the effective piston area is approximately 90 square inches, the total pressure acting on the piston is approximately 1000 lbs., under which condition the piston would stop approximately 2" from the bottom of travel at the point where the spring would have a compressive tension of 1000 lbs.

With 8 lbs. pressure above the diaphragm and 1 lb. vacuum pressure below, or a total of 9 lbs. pressure on 90 square inches which would be about 800 lbs. total pressure, the piston 44 would come to rest at a point two inches below the top of the possible stroke or four inches from the bottom of the possible stroke. Operating under

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these conditions, the piston 44 would float within cylinder 10 between two inches from the bottom and two inches from the top position, in other words, would have a movement of 1" above and 1" below the midpoint of the cylinder 10. Thus the piston displacement is approximately 180 cubic inches, and calculation indicates that a displacement of this amount is required to equalize the flow of the fluid in the suction conduit C to compensate for the variable demand of the four suction chambers of a duplex pump of conventional 7½" x 14", size.

The conditions as set out above, present an ideal operating condition, and in view of this, air pressure is introduced through pipe 92, and regulator 90 is set to maintain 8 lbs. pressure within the spring housing 36, which is sealed and is in communication, by means of vent holes 59, with a chamber in cylinder 26 above diaphragm 62. By maintaining the pressure and with the spring balanced in this relation, the pump may be run seventy five or more strokes per minute without a positive pressure being created within the suction line, and the pump will run smoothly and the cylinders will load perfectly. But no pump can be run faster than that speed where the fluid fails to follow the piston on the suction stroke, for if this speed is exceeded, the pump will pound and ultimately destroy itself.

As pointed out, the chart shown in Fig. 6, was graphed by a pressure indicator attached to connection 100 on the suction side of the pump, which shows the pressures and suctions present in the suction lines C and C', through approximately four revolutions of the crank shaft of the pump. It is to be noted that a positive pressure, as high as twenty three pounds per square inch, existed in the suction line for a period during one revolution of the crankshaft, and that suction pressures of 8 p.s.i. below atmospheric pressure occurred on many strokes.

Since the spring 52 must be selected to meet the requirements of the pump with which it is to be used, no set figure can be given for the tension of the spring. However, for purposes of illustration, the following discussion will be based upon a 7.5" diameter by a 14" stroke mud pump, running fifty-four strokes per minute, with or without the flow equalizer, and also upon the same pump with a 7" liner, running 75 strokes per minute.

Fig. 7 shows an actual indicator card taken at slightly more than one revolution of the pump, with the pump running fifty four strokes per minute, with 7½" liners, sucking from a pit four feet below the level of the pump and pumping with nine pound mud through an 8" suction more than thirty feet long, and with the pump running smoothly and without pounding. It is obvious, from this card, that the pump could be run much faster, therefore the pulley size was changed and the pump speeded up to 75 strokes per minute, and an indicator card, as shown in Fig. 8, was taken. The operating conditions were otherwise the same as those under which the card in Fig. 7 was made, except the liner size was reduced to seven inches to avoid pumping too much mud. As can be seen from the card, Fig. 8, the suction pressure was at all times, below atmospheric, and varied from about one pound below to about three pounds below atmospheric pressure. The pump ran quietly and the cylinders loaded perfectly.

With the proper pressure on the diaphragm 62 and the proper spring pressure on piston rod 38, the surge of mud will move the piston upward to fill or to partially fill cylinder 10 below the diaphragm 62, whereupon, upon the suction being increased, the mud will flow uniformly through the valves into the pump cylinders, thereby maintaining the flow through the pump at a substantially uniform pressure, hence volume, as indicated in the graphs of Figs. 7 and 8.

The operating conditions of the equalizer may be visually observed by noting the movements of the an-

nular pointer 78, which should have a mean position in register with the transverse mark 83 on transparent window 82. Should the mean position of the pointer change, the pressure indicated by gauge 96 can be immediately noted and the proper pressure restored to housing 36 by manipulation of screw 98 of regulator 90, so as to maintain the mean position of the pointer 78 in proper relation with the transverse mark 83. By so doing, the piston 44 is brought back to the midpoint operating position within the cylinder 10.

Having thus described the invention, what is claimed is:

1. In combination with a piston pump of the reciprocating type and a suction line communicating with said pump, a suction flow equalizer comprising a cylindrical accumulator, a piston reciprocable axially in said accumulator, sealing means between said piston and said accumulator, said piston and said sealing means forming two variable sized chambers within said housing, means for placing one of said chambers in communication with said suction line, resilient means operatively connected to said piston exerting relatively opposing axial forces substantially greater than the effect of atmospheric pressure against said piston and normally urging said piston to an axial position of balance intermediate of its stroke in the accumulator, said suction line communicating with the accumulator on one side of said piston, whereby the creation of high pressures in the suction line will store pumped fluids in said accumulator and displace the piston against the force of one said means, and responsive to a subsequent reduction in pressure said last mentioned means will urge the piston in a direction to expel fluid from said accumulator.

2. The combination defined in claim 1, wherein said piston and its sealing means divide the accumulator into two chambers, one of which communicates with said suction line, said resilient means comprising a pressurized gas in the other said chamber acting on said piston.

3. The combination of claim 2 wherein said resilient means further includes a spring acting on said piston in opposition to said gas.

4. A suction flow equalizer for a piston pump having a suction line, comprising a fluid tight housing, a diaphragm within said housing connected at its periphery to said housing, a piston connected to said diaphragm and therewith forming two variable size chambers within the housing, means for placing one of said chambers in communication with the suction line of the pump, a piston rod within the other chamber affixed to and extending from said piston, means communicating with said other chamber for maintaining a gas above atmos-

pheric pressure within said other chamber to urge said diaphragm in one direction, and a spring of a strength approximately equalling the thrust of said gas pressure acting upon said piston rod to urge said diaphragm in the other direction and toward a position of balance intermediate of the stroke of said diaphragm.

5. The device defined in claim 4, including means to regulate said gas pressure in said other chamber.

6. In combination with a piston pump, and a suction line communicating with the intake side of said pump, a suction flow equalizer comprising a fluid tight housing, a diaphragm within said housing and attached thereto and forming therewith chambers of variable volume on opposite sides of said diaphragm, means for placing one of said chambers in communication with said suction line, means for maintaining a gas pressure above atmospheric against said diaphragm in the other of said chambers to urge the diaphragm in one direction, and resilient means of a strength approximately equalling the thrust of said gas pressure for urging said diaphragm in the other direction and toward a position of balance intermediate of the stroke of said diaphragm.

7. The device defined in claim 6, including means to regulate said gas pressure and means to adjust the pressure exerted by said resilient means operatively associated with said means for maintaining gas pressure.

8. The device defined in claim 6, including means connected to said diaphragm and to indicate visibly the position within said housing of said movable means, and transparent means within the wall of said housing to permit such visibility.

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