Feb. 13, 1940. J. S. FULLER 2,189,893

PUMPING APPARATUS

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Fig. 1.

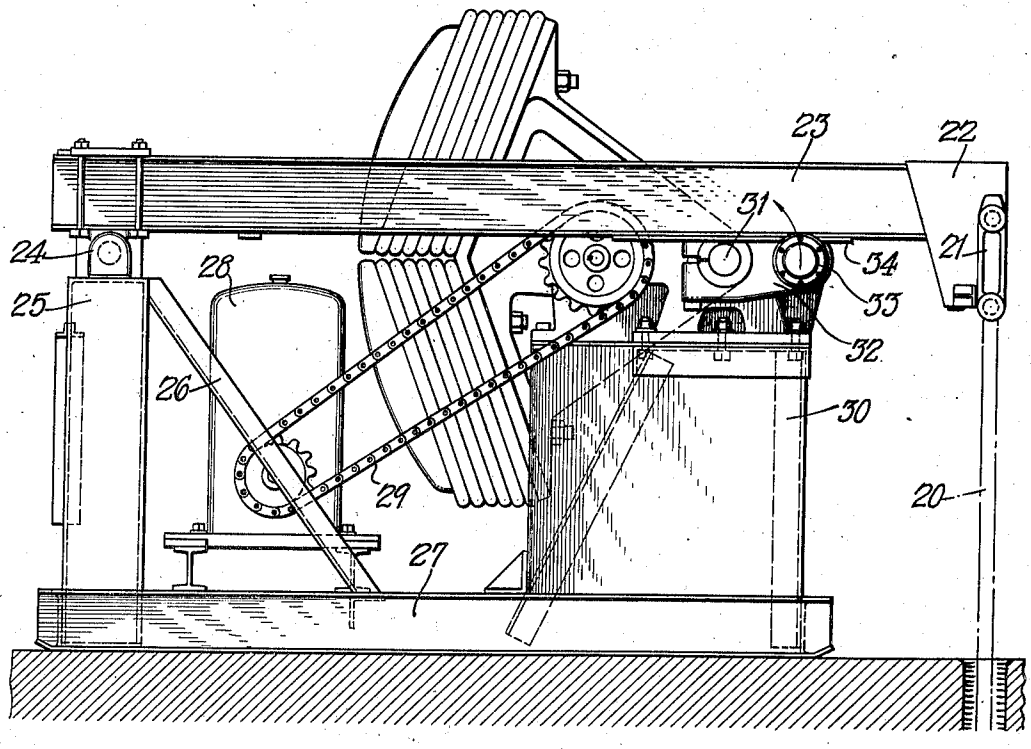
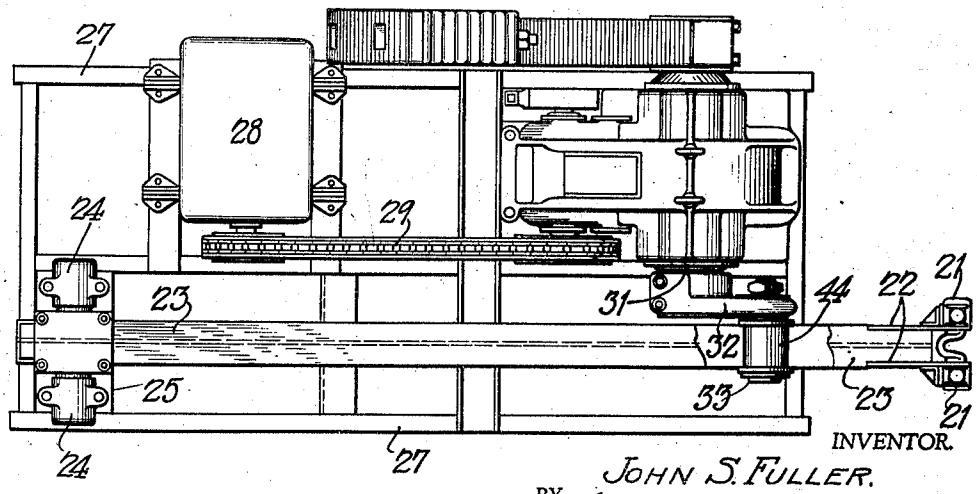


Fig. 2.



INVENTOR.
JOHN S. FULLER.
BY Thima & Rauber ATTORNEYS

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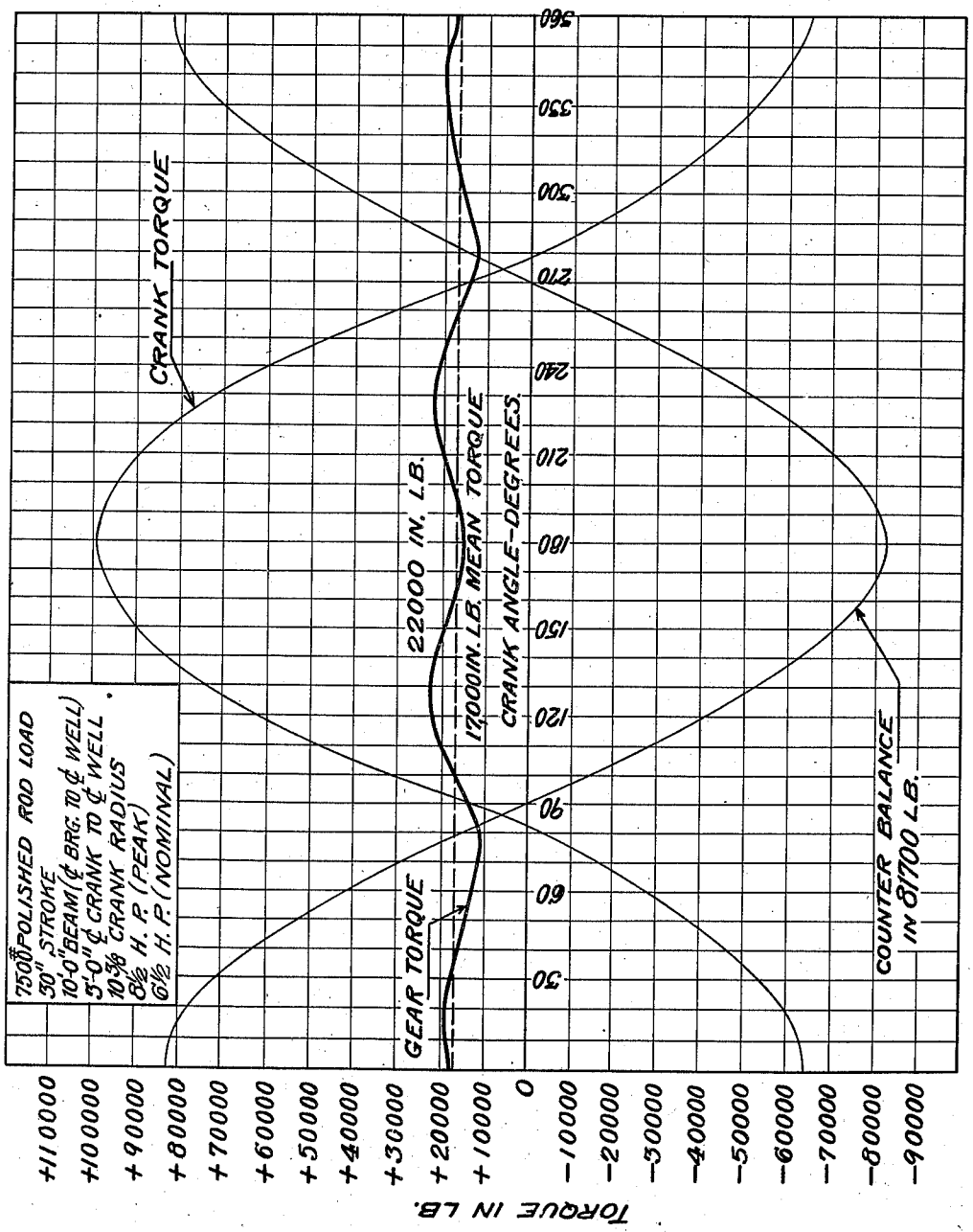


Fig. 5.

INVENTOR.
JOHN S. FULLER.
BY Hoima & Rauber ATTORNEYS

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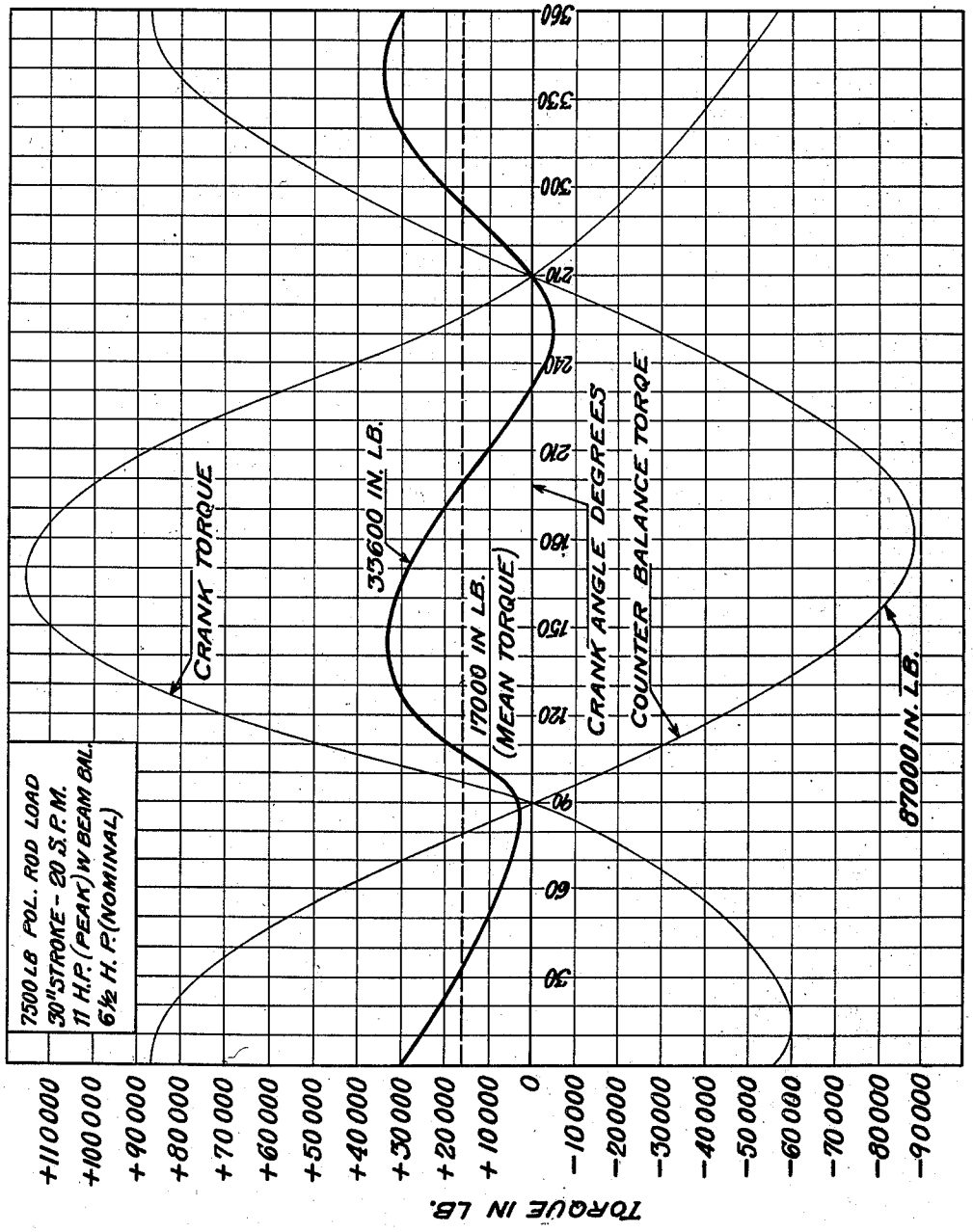


Fig. 4.

INVENTOR.
JOHN S. FULLER.
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Feb. 13, 1940. J. S. FULLER 2,189,893

PUMPING APPARATUS

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Fig. 5.

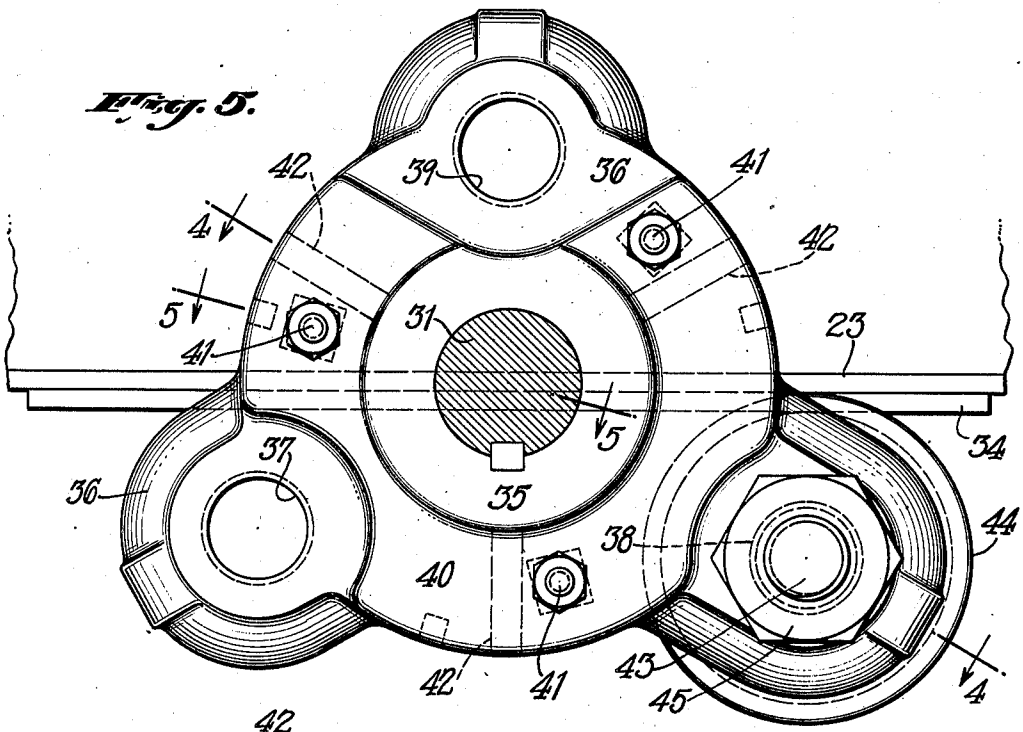


Fig. 6.

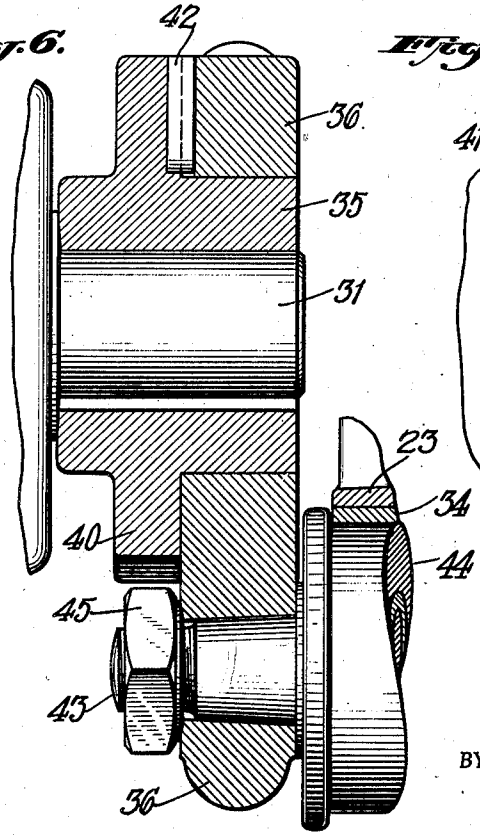
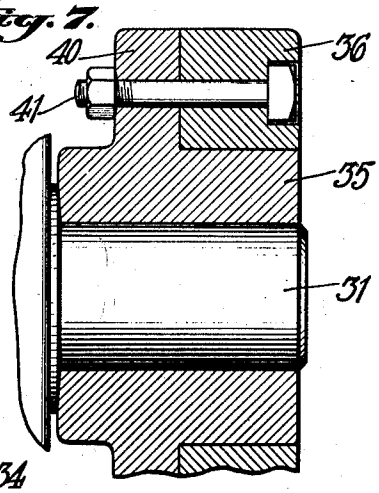


Fig. 7.



INVENTOR.
JOHN S. FULLER.

BY Howson & Sawyer ATTORNEYS

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Fig. 8.

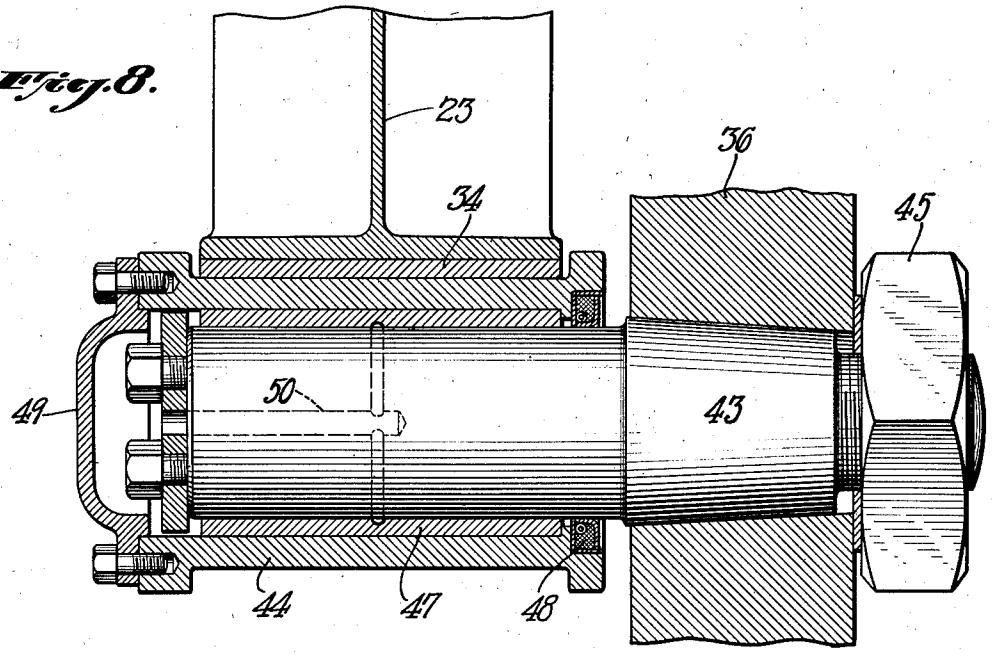


Fig. 9.

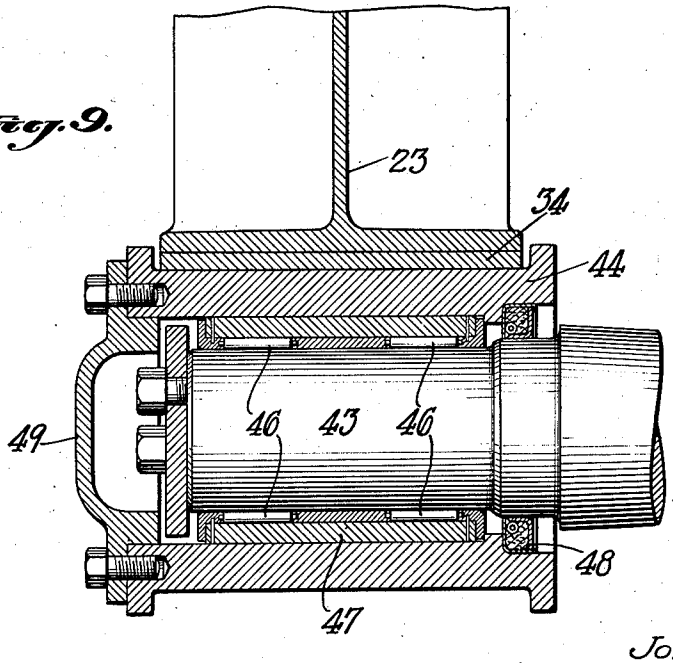
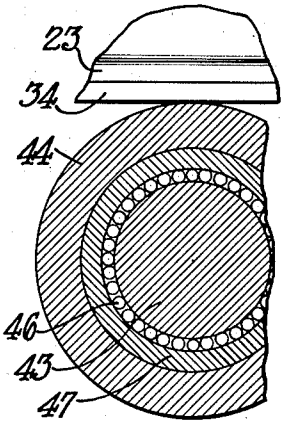


Fig. 10.



INVENTOR.

JOHN S. FULLER.

BY Shinn & Pauber ATTORNEYS

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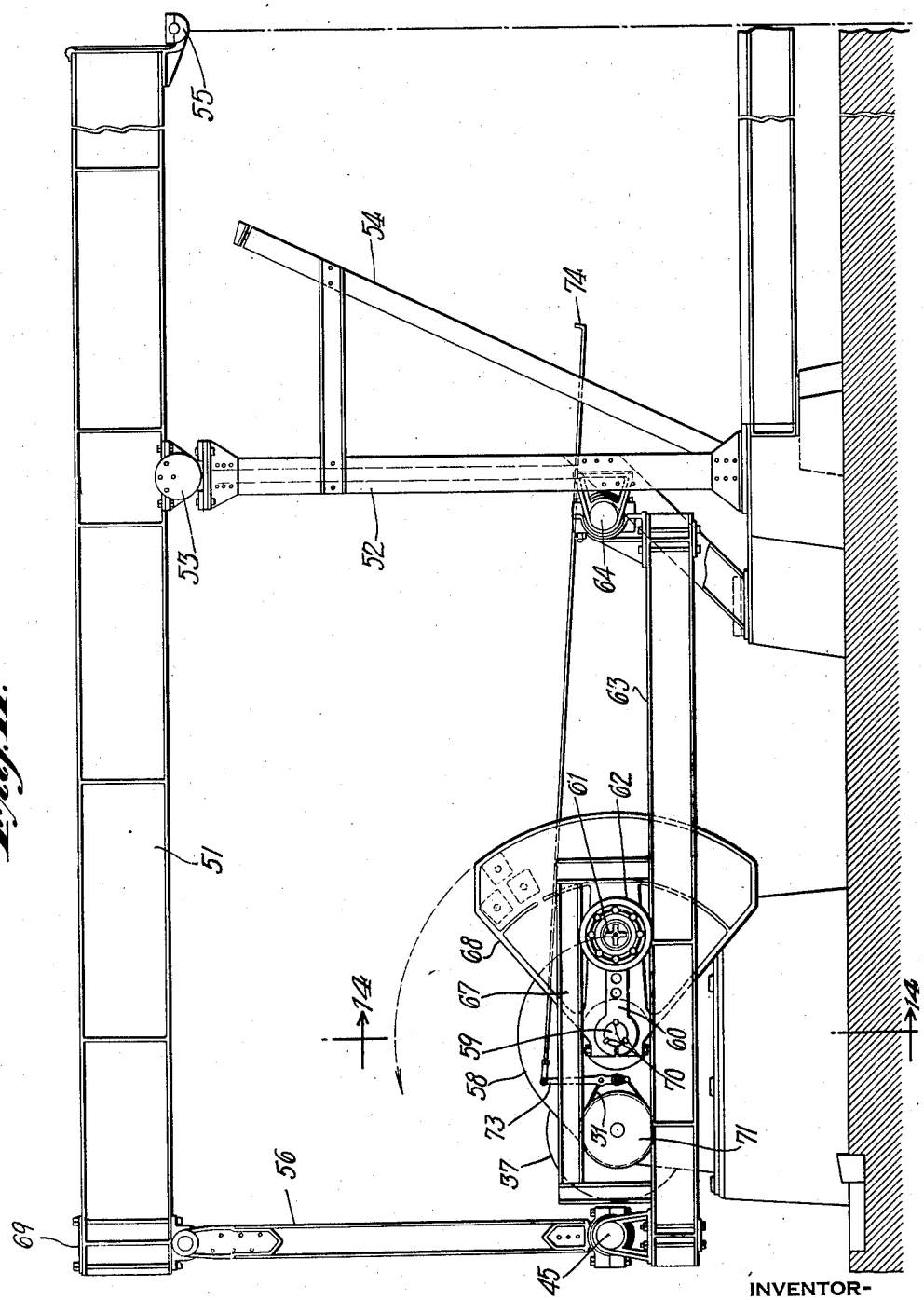


Fig. 11.

INVENTOR-
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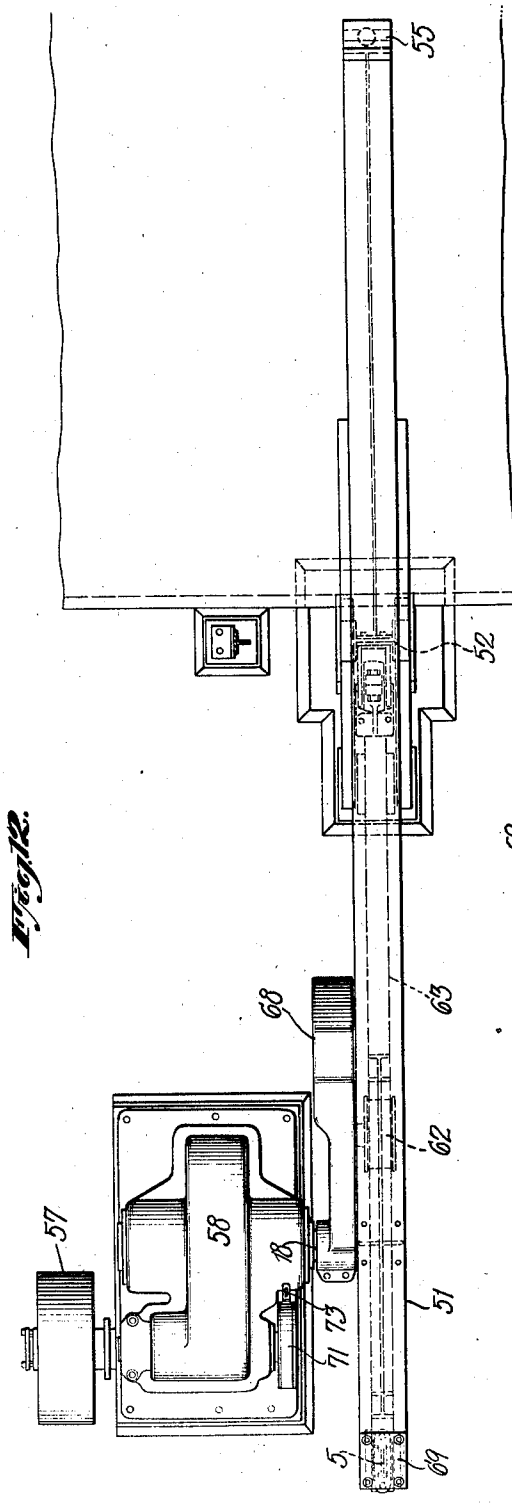


Fig. 12.

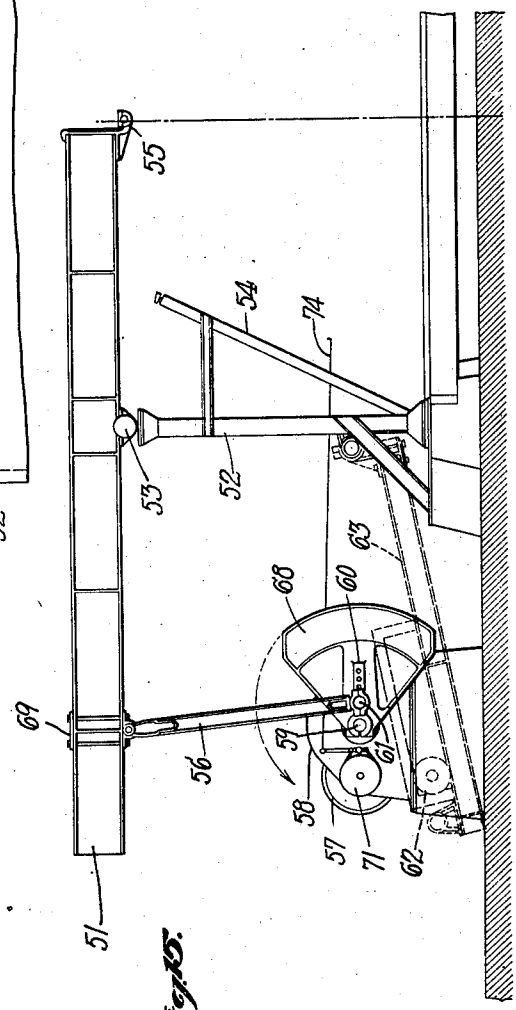


Fig. 13.

INVENTOR
JOHN S. FULLER
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Feb. 13, 1940. J. S. FULLER 2,189,893

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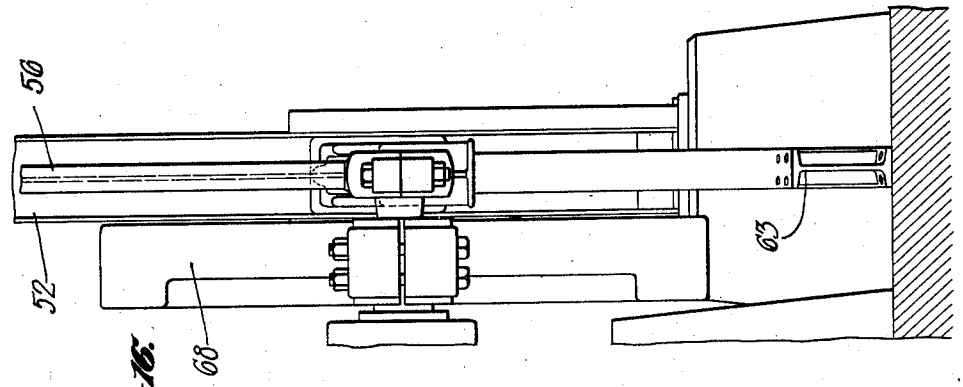


Fig. 16.

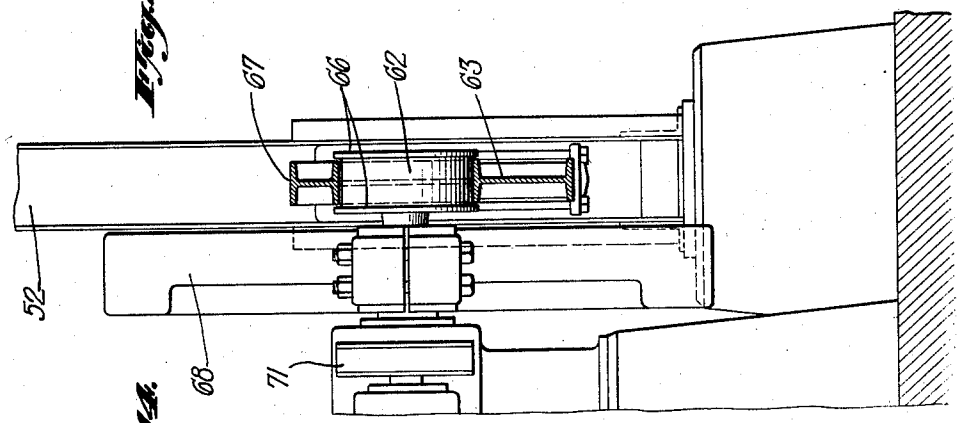


Fig. 14.

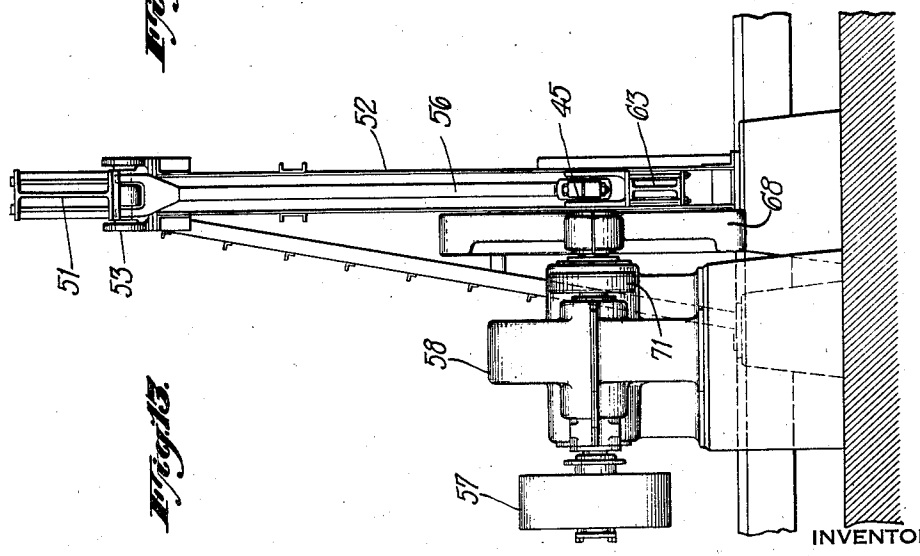


Fig. 15.

INVENTOR
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Patented Feb. 13, 1940 2,189,893

UNITED STATES PATENT OFFICE

2,189,893

PUMPING APPARATUS

John S. Fuller, Tulsa, Okla., assignor to Oil Well Supply Company, Dallas, Tex., a corporation of New Jersey

Application July 15, 1937, Serial No. 153,823

3 Claims. (Cl. 74—48)

My invention relates to apparatus for pumping deep wells, as for example wells for the production of petroleum. In the pumping of such wells, the piston of a pump located at the bottom of the well is actuated through a line of "sucker" rods connected end to end and reaching to a pumping mechanism at the top of the well. The top end of the line of sucker rods is connected through any suitable means, usually through a "polish rod" extending through a packing at the top of the well tubing to a walking beam or pumping jack, which gives the line of sucker rods and the pump piston an up and down oscillation.

It is apparent that the entire weight of the sucker rods and of the pump piston, diminished by the buoyant effect of the liquid they displace, is at all times suspended from the end of the walking beam, pumping jack, or other apparatus, which would cause an added burden to the walking beam in its upward stroke and an accelerating force in its downward stroke.

The weight of the sucker rods and pump piston have heretofore been counterbalanced by a weight mounted on a crank arm of the pumping apparatus. If the moment arm or torque of this counterbalance be made just equal to the displacement weight of pump piston and sucker rods suspended from the walking beam, the torque on the pumping apparatus during a downstroke being balanced in both directions, will be zero. On the upstroke, the torque will be that imposed by the weight of liquid being lifted and the power required to give it the necessary upward acceleration. The net weight of sucker rods and pump piston is, of course, balanced in this upper stroke. In this case, the torque will vary from zero on the down stroke to a maximum on the upstroke, this maximum being, therefore, about twice the average or mean torque. If the net weight of the sucker rods and pump piston be overbalanced, the maximum torque required on the upstroke will be correspondingly reduced, but there will be an accelerating torque on the down stroke. Heretofore, therefore, it has been impossible to avoid a wide variation in the torque required to pump the well.

My invention greatly minimizes and largely eliminates these variations in pumping torque required to pump a well and provides means by which wells may be pumped with a torque that varies only slightly from the mean throughout a complete pumping cycle.

In my invention, I provide a counterbalance weight that acts on the pumping apparatus with a varying mechanical advantage in such a way that the greatest mechanical advantage is exerted by the counterbalance weight during the upstroke and correspondingly diminished mechanical advantage exerted on the downstroke. Thus, in the upstroke, the counterbalance weight may act on the walking beam or equivalent element of the pumping apparatus with a greater moment arm than on the downstroke. For example, a counterbalanced crank arm having weights at a fixed radius from the axial center of the crank arm may act through a crank arm on a lever or walking beam at a greater distance from the axial center of the latter during the upstroke than during the downstroke and in this way give an effective longer moment arm to the counterbalance weights during the upstroke and a diminished moment arm during the downstroke.

I also provide an apparatus wherein the force from the engines can likewise have an effective leverage change as well.

The various features of the invention are illustrated by way of example in the accompanying drawings, in which—

Fig. 1 is a side elevation and Fig. 2 a plan of a pumping apparatus embodying a preferred form of the invention; Fig. 3 is a chart showing the torques on the walking beam or crank, the counterbalancing torque of the counterbalance weight and the resultant gear torque in the apparatus embodying my invention; Fig. 4 is a similar diagram showing a typical torque in apparatus having a counterbalance weight of constant mechanical advantage or moment arm; Figs. 5, 6 and 7 are, respectively, front elevations and sections on the lines 4—4 and 5—5 of a modified form of crank arm applicable to the apparatus shown in Figs. 1 and 2; Figs. 8, 9 and 10 are, respectively, longitudinal sections and a cross-section of the crank arm pin; Figs. 11, 12, 13 and 14 are respectively side elevation, plan, end elevation and section on line 14—14 of Fig. 11 of a pumping apparatus embodying another form of the invention; Fig. 15 is a view similar to that of Fig. 11 on a smaller scale, showing certain parts of the apparatus disconnected; Fig. 16 is a vertical end view of a part of the apparatus taken from the left of Fig. 15.

Referring more particularly to the pumping apparatus shown in Figs. 1 to 10 inclusive, the end, or end connection, of a line of sucker rods 20 indicated by the dot and dash line of Fig. 1 is connected by means of a link 21 to a beam hanger 22 mounted on one end of a walking beam 23. The other end of the walking beam 23

is journalled in a pair of bearings 24 supported on an upright pedestal or post 25 commonly known as a Samson post. This post may be braced by a bracing strut 26 and is mounted on the base frame 27 of the pumping apparatus. The walking beam 23 is rocked or oscillated vertically on its supporting pivot in the bearings 24 by power supplied from a prime mover, such as the internal combustion or other engine 28. Power is transmitted from the engine 28 through any suitable transmission, such as a chain and sprocket drive 29 to a speed reduction mechanism 30 and thence to a crank shaft 31 to rotate the latter counter-clockwise. As shown in Fig. 2, the crank shaft 31 terminates just short of the beam 23 and has rigidly mounted on it a crank arm 32. The end of the crank arm 32 carries a roller 33, which bears upwardly against a bearing plate 34 on the underside of the beam 23, so that, as the crank 32 rotates, the roller 33 acting against the plate 34 alternately lifts and lowers the beam 23.

It will be apparent that the moment arm of the suspended sucker rods and pump elements is equal to the distance from the axis of the bearing 24 to the sucker rod line 20 at a right angle to the latter. Also that the upward force exerted by the roller 33 acts with a moment arm equal to the distance between the centers of the roller 33 and bearings 24. The moment arm of the sucker rod line and associated element remains unchanged for the upstroke and downstroke. The moment arm of the force exerted by the roller 33 is less during the downstroke than during the upstroke because the crank arm is then on the opposite side of its center of rotation, bringing the axial center of the roller 23 nearer the center of the bearings 24. Accordingly, the change in the moment arm of this crank varies the torque exerted by the counterweight between the upstroke and downstroke to more nearly approximate the changed load on the sucker rod line.

This is shown graphically in Fig. 3 of the drawings, in which the ordinates express in inch pounds the resistance to rotation exerted by the counterbalance and the walking beam, respectively, for each angle starting with the position 180 degrees from that of Fig. 1. As the resistance to rotation is shown as positive and accelerating torques as negative, the resultant force to be driven by the gears or the gear torque required to drive the crank shaft will be the algebraic sum of the crank and counterbalance torques.

Thus, at the beginning of the stroke, the counterbalance will have a retarding torque of somewhat over 80,000 inch pounds (that is equal to 8,000 pounds at 10 inch moment arm) while the crank torque will urge the crank forwardly with a torque of nearly 65,000 inch pounds. The difference of about 18,000 inch pounds is the gear torque required to drive the crank shaft. As shown on the graph, the crank torque rises, and the counterbalance torque falls to zero at a 90 degree angle from the starting point at which the crank arm 32 will be in its lowermost position. As it swings upwardly, the crank torque rises rapidly to about 100,000 inch pounds at the 180 degree mark, while the counterbalance torque has dropped correspondingly to a negative value of somewhat over 80,000 inch pounds. It will be noticed that the crank torque has increased. This is due to the fact that it is lifting liquid as well as the sucker rod and associated parts. The crank torque would be larger than 100,000 inch pounds, except for the fact that the effective moment arm of the crank roller 33, has now been increased from that of its starting position by an amount equal to twice the radius of the crank arm. In passing to the 270 degree angle and thence to the original position, the crank torque decreases and the counterbalance torque increases until they reach the same magnitudes as at their starting position.

It is to be noted that the resulting gear torque obtained by the algebraic sum of the crank torque and counterbalance torque does not become negative and varies only slightly from the mean torque indicated in broken lines as 17,000 inch pounds. Thus, the actual gear torque in this particular example does not drop to 10,000 inch pounds, nor rise above 24,000 inch pounds and is generally well within these limits at all angles. Moreover, it is to be noticed that there are no sharp or steep changes in the resulting gear torque.

In Fig. 4, the torques are plotted in a similar manner for corresponding installation, in which the crank arm was connected to the walking beam, by a pitman as in installations prior to my invention. In this case, it was necessary to use a somewhat larger counterbalance torque, namely, 87,000 inch pounds maximum instead of 81,700 inch pounds maximum as in Fig. 3. This is because the crank torque is subjected to wider variations, varying from 60,000 inch pounds to about 115,000 inch pounds. As shown in Fig. 4, the gear torque obtained by the algebraic addition of the crank torque and counterbalance torque varies widely and sharply from the mean torque. At its maximum, it is very nearly twice the mean torque and at its minimum, it is actually negative, so that the resultant of the crank torque and counterbalance torque actually tend to drive the gears, thus wearing them on the opposite side of the teeth from the driving side. The wide and sudden variation in gear torque require an excess of power over the mean, in order to drive the apparatus during the upstroke and require heavier counterbalance weights and heavier structure throughout in order to stand the maximum stresses and the sudden changes or shocks imposed. In contrast, the changing mechanical advantage obtained in applicant's invention enables a much smaller power unit to be employed with smaller and lighter construction and with less wear due to reversals and sudden variations in the driving stresses or power. Moreover, the period of upstroke is increased and that of the downstroke decreased, the upstroke starting before 90° and lasting until after 270° from the starting point of the graph, whereas in the graph of Fig. 4 the upstroke starting exactly at 90° and terminating at 270° from the starting position.

Where the pumping power is to be used under such conditions that the stroke of the pump is to be varied or changed, the crank arm 32 must be lengthened or shortened accordingly. In the modification shown in Figs. 5, 6 and 7, a modified form of crank arm is provided, in which three crank arm lengths may be obtained selectively. In this modification, a hub 35 is mounted on a crank shaft 31 and carries a crank arm ring 36 having three crank pin openings 37, 38 and 39 of progressively larger radius from the axis of the shaft 31. The crank ring 36 is bolted to a flange 40 of the hub by means of bolts 41 and is keyed thereto by keys 42 inserted in common keyways

between the hub and crank ring. It will be apparent, therefore, that the crank ring is replaceable and when mounted in position is rigidly secured and rotates with the crank shaft 31. A crank pin 43 may be inserted selectively in any one of the three openings 37, 38 or 39 and carries on its free end a roller 44, as shown in Figs. 8, 9 and 10. The pin 43 may be removably secured in any of the openings 37, 38 and 39 by means of a nut 45 threaded onto the end of the pin 43. The roller 44 is mounted on the pin 43 by any suitable anti-friction construction, as for example by the roller bearing elements 46 acting between an outer raceway 47 in the roller 44 and an inner raceway formed by the pin 43. The roller elements are enclosed at one end by means of a gasket type of closure 48 and the other by a plate 49, so as to form a closed chamber, in which a lubricant may be carried to be supplied to the roller elements through the canal system 50.

It will be apparent that by changing the pin 43 from one to the other of the three openings 37, 38 and 39, three different crank arm radii may be obtained selectively. Also that the crank hub may be replaced by other crank hubs having different crank arm radii, or by a single crank arm. The action in any case is similar to that illustrated in connection with Figs. 1 and 2 and Chart 3.

The above described apparatus of Figs. 1 to 10, inclusive, is particularly advantageous because it provides the longest possible lever arm for a beam of a given length, thereby reducing the angle and the side sway on the sucker rod line for a given lift. Special hangers to compensate for this side sway may generally be avoided by the form of the invention shown in Figs. 1 to 10. However, the invention may also be applied to pumping power in which the beam is suspended midway of its ends, and may thus be applied to existing pumping apparatus. A modification of this type is shown in Figs. 11 to 16 inclusive. In this modification the walking beam 51 is pivotally supported on the Samson post 52 by means of a beam saddle or bearing 53 which may be provided with the usual "headache" post 54 to support the pumping end of the beam, or to be disconnected for changes or repairs. The pumping end of the beam is provided with a hanger 55 above the center line of the well or in alignment with the sucker rod as indicated in the dot and dash lines. The opposite end of the beam is connected by means of a pitman 56 to the driving or pump operating means.

The pump driving mechanism is driven from a pulley 57 through a speed reduction mechanism 58 which drives a rotating shaft 59 on the end of which is a crank arm 60. On the end of the crank arm 60 is mounted a wrist pin 61 which, in turn, carries a rotatable wheel or roller 62. The wheel 62 rides freely on a lever beam 63 pivoted at one end on a bearing or fulcrum 64 on the Samson post 52, and connected at its other end to the pitman 56 through a connecting bearing 65. The roller 62 is preferably provided with flanges 66, as shown in Fig. 14, to ensure its accurate travel on the beam 63. The lever beam 63 is held upwardly against the roller 62 by the weight of the pumping apparatus acting through the walking beam 51 and the pitman 56. The roller 62 and the lever beam 63 may be additionally held in association by the guide 67 mounted on the lever beam 63 and extending over the pulley 62. A counter-weight 68 is also mounted on the shaft 59 either as an integral part of the crank 60, or independently thereof, but in rigid relation thereto so that the weight of the counter-weight acts through the crank 60 on the lever beam 63.

In the operation of the apparatus, the shaft 59 and the crank 60 and counter-weight 68 rotate counter-clockwise as indicated by the arrow in Fig. 11. During the downstroke of the crank, it tilts the lever beam 63 downwardly about its fulcrum 64 and carries with it the pitman 56 and the left arm of the walking beam 51, thereby tilting upwardly the opposite arm and carrying with it the sucker rod line of the pump. As the crank swings from its lowermost position counter-clockwise to its uppermost position, the beam 63 moves upwardly and permits the walking beam 51 to swing clockwise to lower the opposite end to which the pumping mechanism is attached through the hanger 55.

During the downstroke, the crank 60 acts on the lever beam 63 with a greater moment arm than during the upstroke inasmuch as the distance of the fulcrum 64 from the wheel 62, through which the power and weight of the counter-weight 68 are transmitted through the lever beam 63, is greater on the downstroke than on the upstroke. Accordingly, the effect of the counter-weight is increased during the upstroke of the sucker rod line, and decreased during the downstroke, thereby compensating for the unequal pull of the line during the respective strokes. The result is similar to that shown in Fig. 3.

In the construction shown in Figs. 11 to 16, it is at times desirable to use the walking beam 51 for drilling, or cleaning out operations. In this case the variable counter-weight 68 is not required. The lever beam 63 is disconnected and the walking beam 51 is driven directly from the crank 60. For this apparatus the connection of the pitman 56 to the walking beam 51 is shifted by moving a connecting hanger 69 from the position shown in Fig. 11 to that shown in Fig. 15. The roller or wheel 62 is disconnected from the crank arm, and connecting the bearing 66 of the pitman directly to the end of the crank arm 60.

In the operation of the apparatus, unequal stresses may at times be imposed on the shaft 59 which, being transmitted to the gearing, would cause unequal wear on the latter. To compensate for this wear, the crank arm 60 is keyed to the shaft 59 by a plurality of keys 70, three being shown by way of example, so that the position of the crank 60 relatively to the transmission gears may be adjusted to equalize the wear. In order to hold the walking beam stationary while adjusting the pump mechanism, a brake drum 71 is provided on the driving shaft of the speed mechanism 58 on which a brake band 72 may be caused to operate by a lever 73 and control rod 74.

By means of the above invention it is, therefore, possible to so counter-balance the driving apparatus as to maintain a torque resistance against the driving gear that is very nearly constant and approximately that of the mean stroke. As a result of this and of the lessening or elimination of shock due to sudden changes in, or reversals of, the torque resistance against the driving gears, a smaller power equipment is required and the entire construction may be made lighter and less cumbersome. The lengthening of the time of the upstroke and relative shortening of the downstroke period are also of advantage, affording more time to get the liquid into motion during the upstroke for a given speed of driving.

The modification shown in Figs. 1 to 10 inclusive, has the further advantages of a longer angle arm and less deviation of the free end of the walking beam from vertical alignment with the well and also, inasmuch as the greatest weight is supported nearer to the pull of the pump and sucker line, this beam or lever may be made of somewhat lighter construction.

The modification shown in Fig. 11 has the further advantage of eliminating any horizontal thrust on the Samson post as all of the forces acting on the walking beam are in a vertical plane.

What I claim is:

1. Well pumping apparatus comprising in combination a walking beam, means to pivotally sustain said beam for vertical oscillation, a pump mechanism dependingly secured from the beam at a fixed distance from the pivotal point of the beam, and means including a rotating crank arm to apply power to the beam intermediate the pivotal point and the point of suspension of said pump mechanism, said means including means to progressively shift the point of application of power away from and towards the said pivotal point to obtain a maximum moment arm at the time of maximum load during the upstroke of the pump mechanism and a minimum moment arm at the time of minimum load during the downstroke of the pump mechanism, and a counterbalance weight secured to rotate about the axis of rotation of the crank shaft supporting said crank arm in a position relative to the crank arm to effectively balance the load of said pump mechanism.

2. The combination of claim 1, wherein said means including a rotating crank arm to apply power to the beam intermediate the pivotal point and the point of suspension of said pump mechanism comprises a crank pin mounted on said crank arm and a roller mounted on said crank pin and means locating the crank arm, pin and roller in a position to operatively engage the roller with the said walking beam thereby to raise and lower the said beam during rotative movement of the said crank arm and pin.

3. The combination of claim 1, wherein said means to apply power to the said beam comprises a crank shaft rotatively driven, a crank arm mounted on said shaft, a roller pin extension on said crank arm and means to position the crank shaft, crank arm and roller pin extension in a position between the pivotal point and the point of suspension on the beam with the axis of the roller pin extension normal to the beam axis and with the roller surface in engagement with the under side of the said beam in all positions of rotation of the said crank arm, and a bearing plate member on the under side of the beam covering the beam surface contacting said roller.

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