Oct. 22, 1968 G. EYLER ETAL 3,406,581

PUMPING APPARATUS

Original Filed Oct. 19, 1964 5 Sheets-Sheet 1

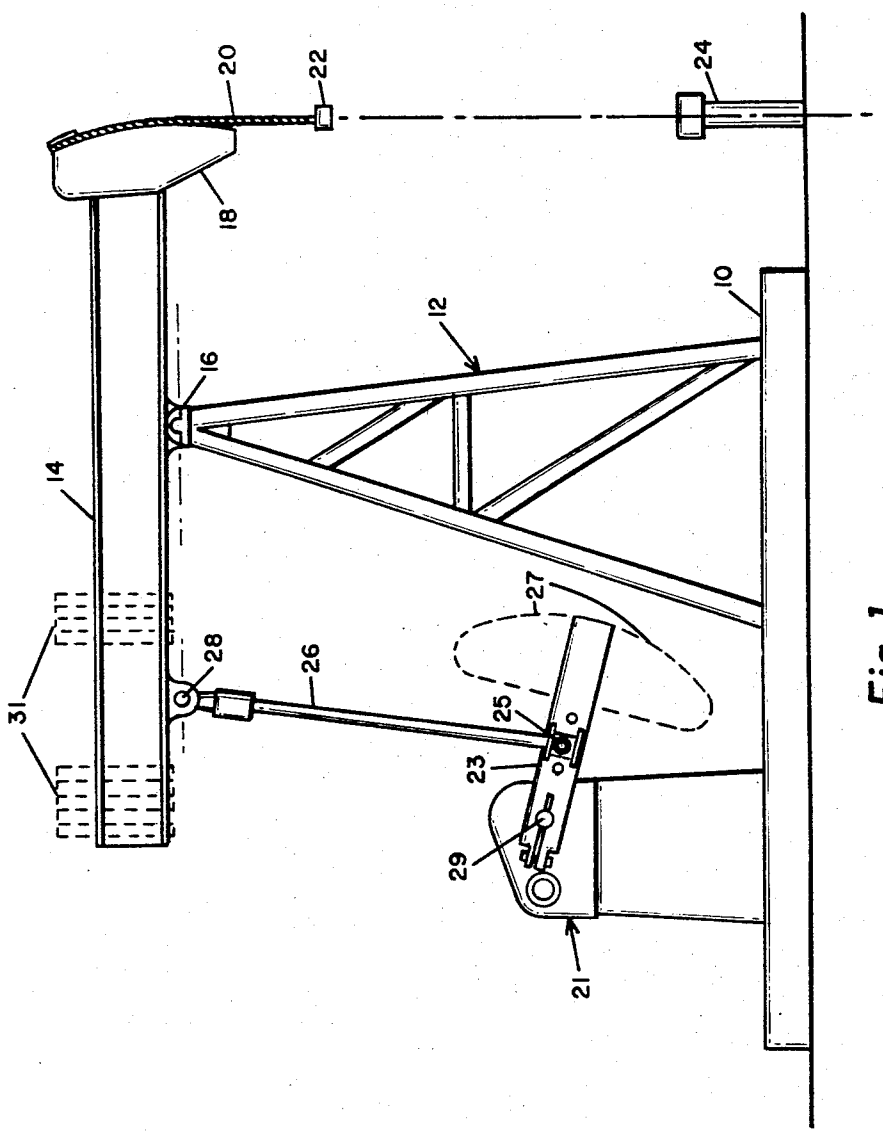


Fig. 1

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PUMPING APPARATUS

Original Filed Oct. 19, 1964 5 Sheets-Sheet 2

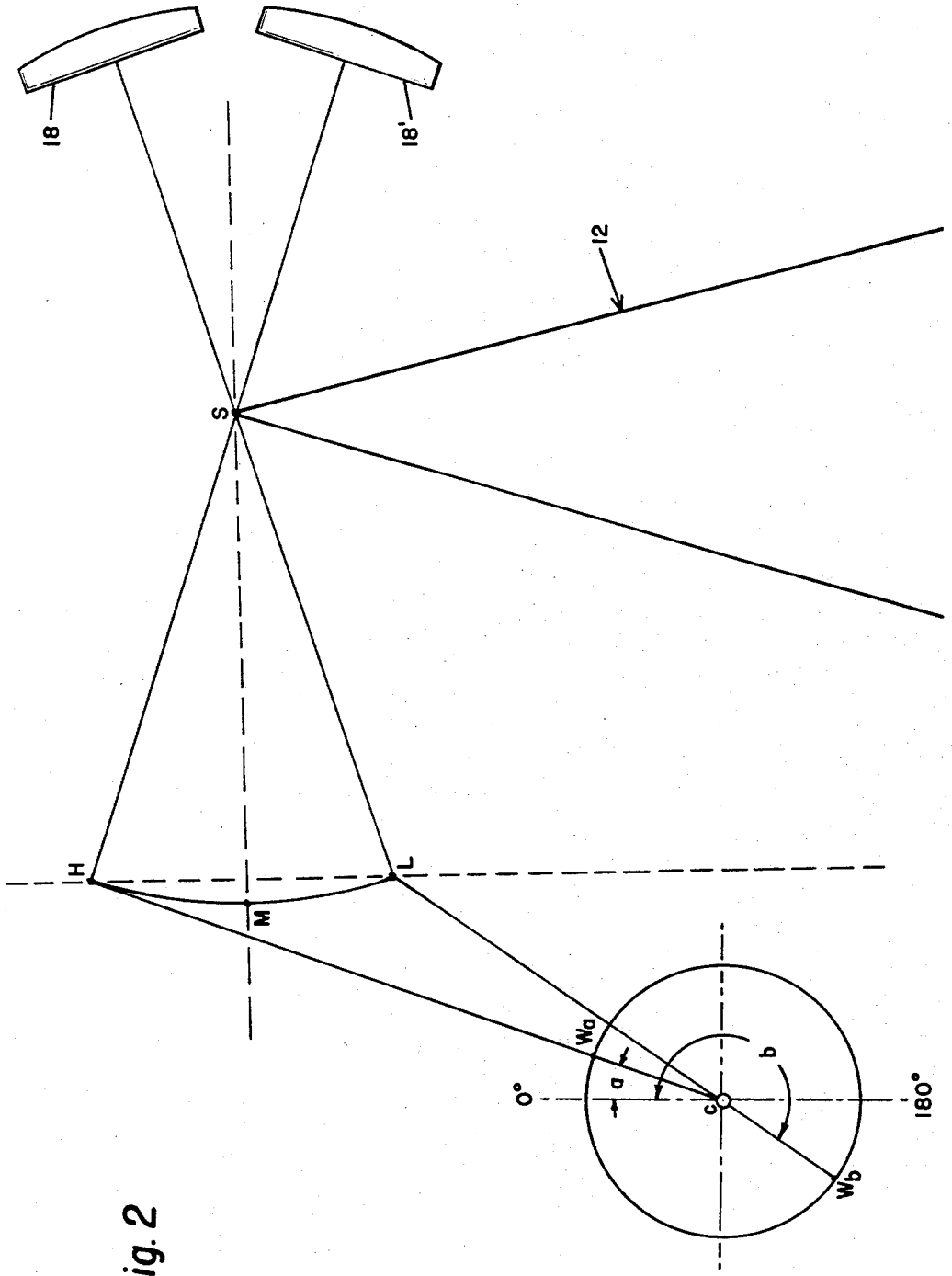


Fig. 2

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Original Filed Oct. 19, 1964 5 Sheets-Sheet 3

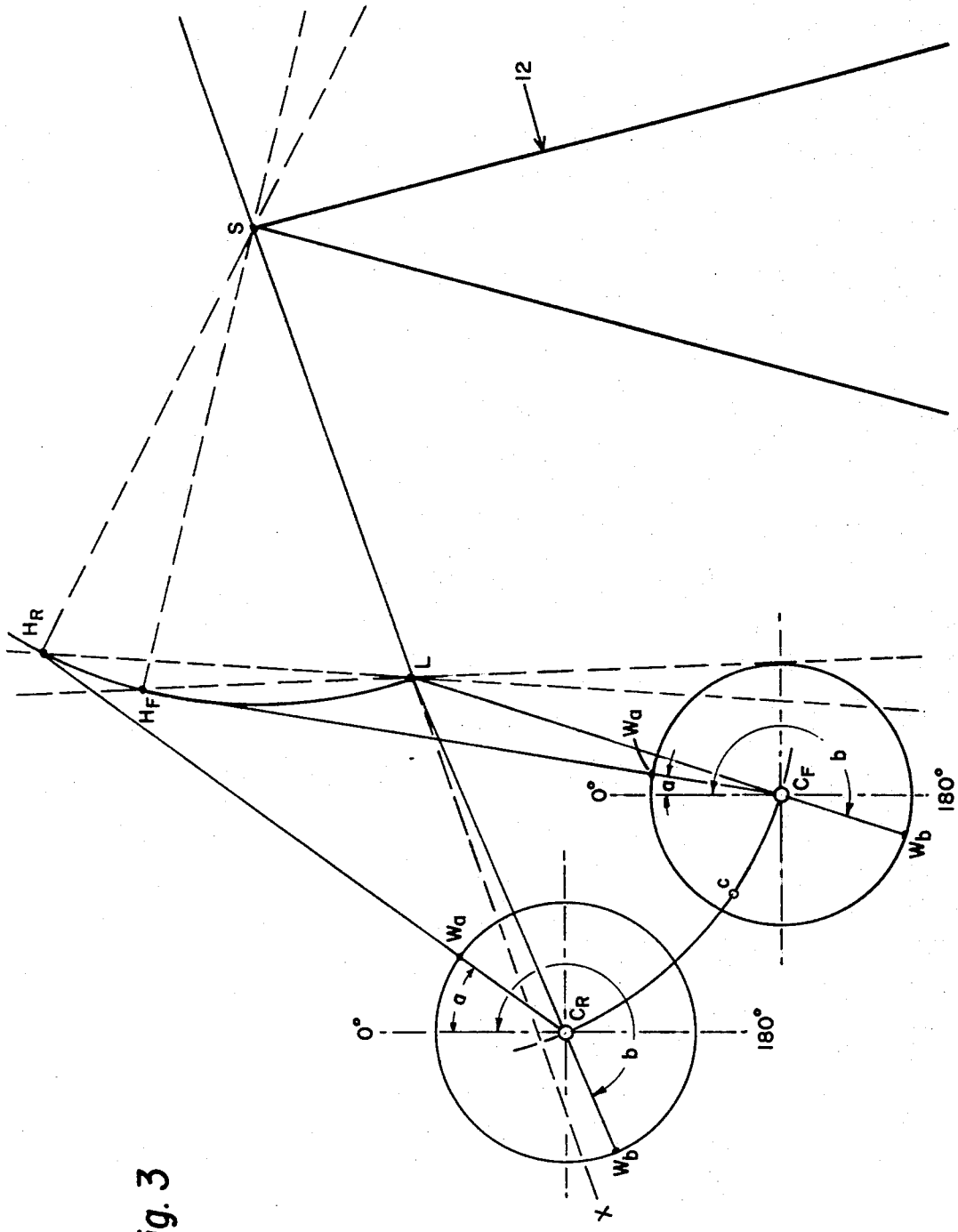


Fig. 3

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PUMPING APPARATUS

Original Filed Oct. 19, 1964 5 Sheets-Sheet 4

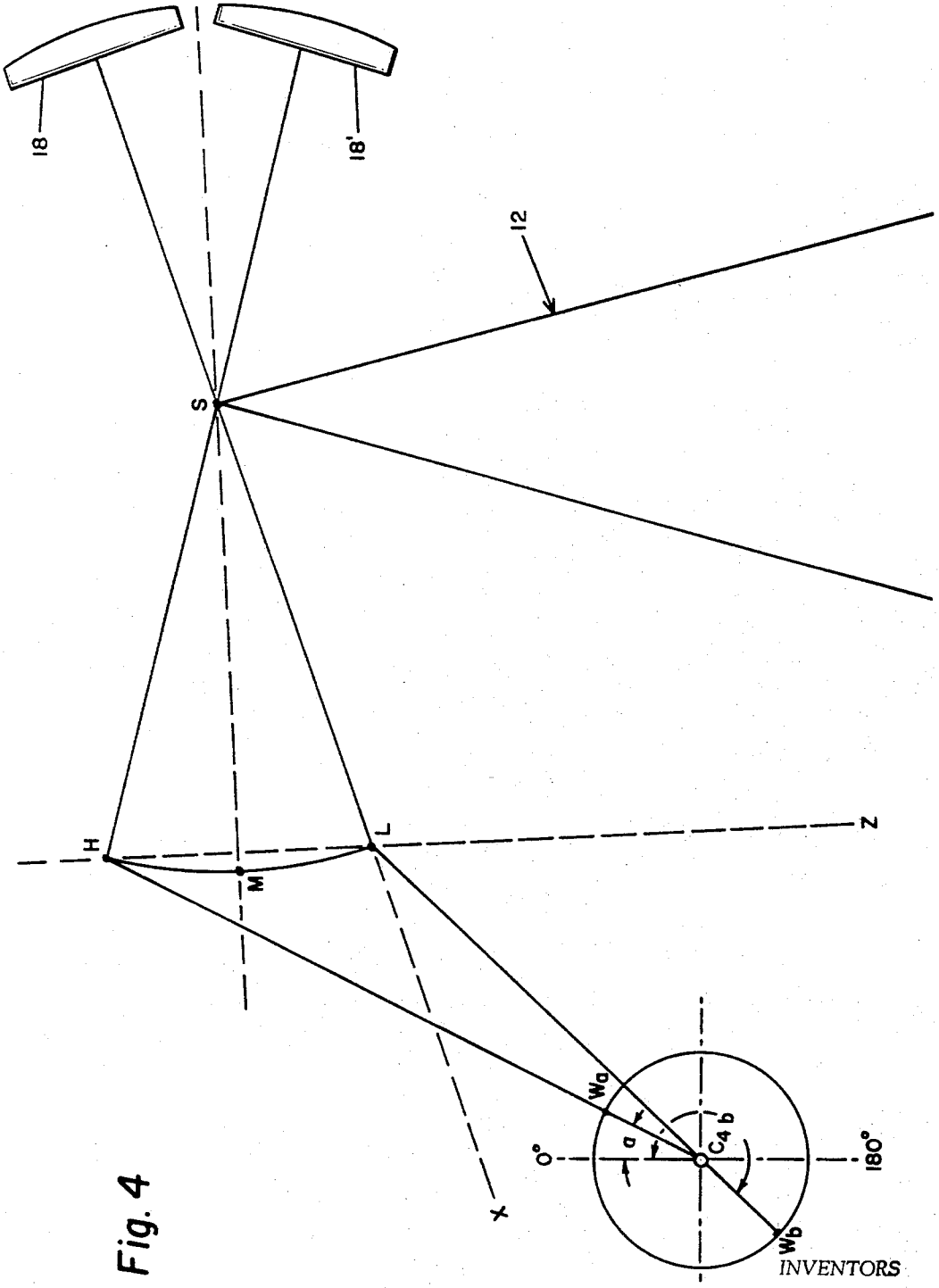


Fig. 4

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PUMPING APPARATUS

Original Filed Oct. 19, 1964 5 Sheets-Sheet 5

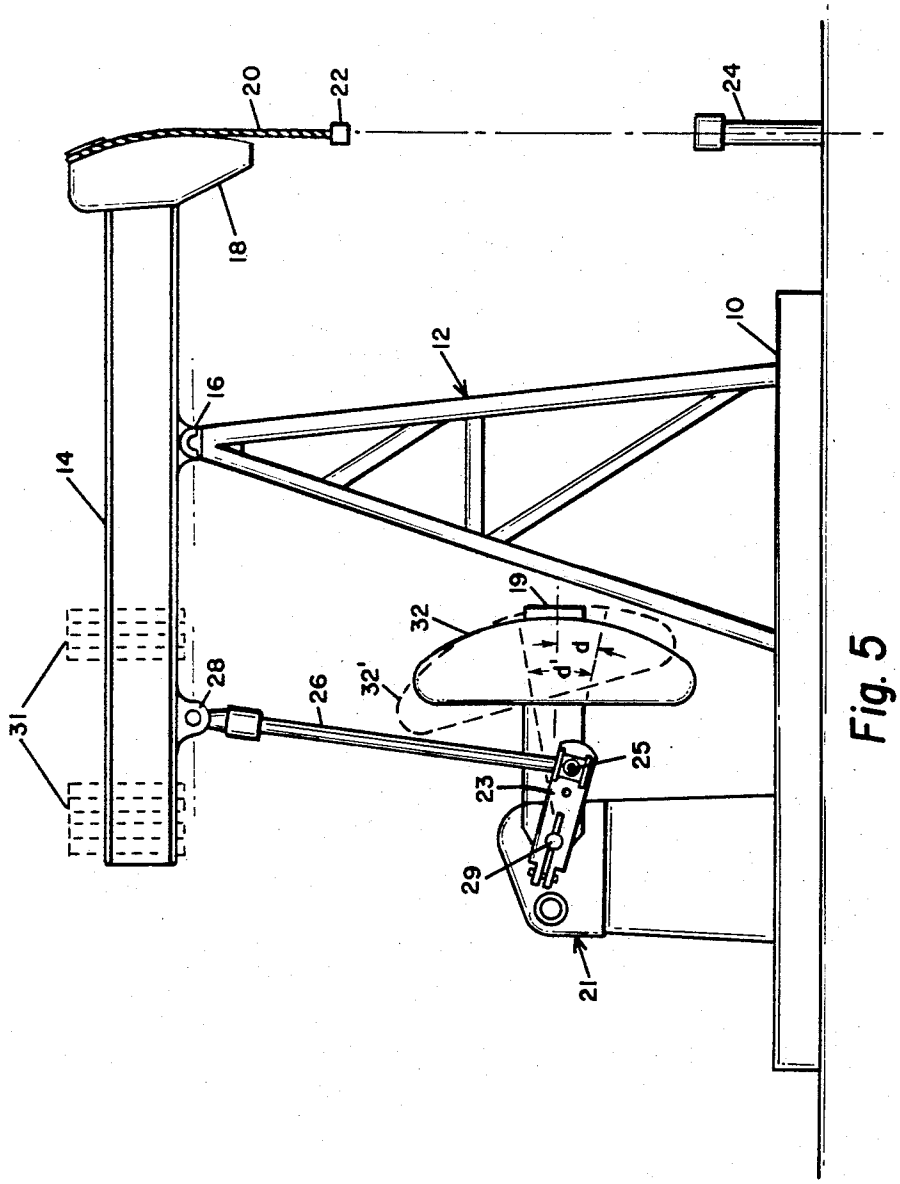


Fig. 5

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3,406,581

PUMPING APPARATUS

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Continuation of application Ser. No. 404,707, Oct. 19, 1964. This application Apr. 10, 1967, Ser. No. 629,833
6 Claims. (Cl. 74—41)

ABSTRACT OF THE DISCLOSURE

An improved well pumping unit is provided of the walking beam type in which the rotary drive means is located for maximum ease of servicing, optimum efficiency of power delivery and most consistent overall operating smoothness while the rotary counterbalancing weights are simultaneously mounted in specific geometrical relationship with respect to the exact timing of the well pumping stroke so as to exert maximum effect at the time of greatest need. The end result is a pumping unit constructed so that, without undue extension of the pitman arm length, the crank shaft is placed back completely out from under the normal area under the tail bearing connection on the walking beam while simultaneously minimizing peak torque factors and the mass of rotary counterweights needed.

This application is a continuation of application Ser. No. 404,707, which was filed on Oct. 19, 1964, now abandoned.

This invention relates to improved pumping apparatus of the walking beam type such as is used primarily for lifting fluids, particularly oil, from substantial depths to the earth's surface. More particularly, the invention is concerned with an improved arrangement of the drive linkage elements in a pumping unit of the most familiar walking beam type in which the samson post supports the walking beam near its medial portion by means of a saddle bearing pivot connection and the drive equipment by means of which power is applied to the walking beam is located on the opposite side of the samson post from the well working end of the walking beam.

U.S. Patent 3,029,650 indicates that, in order to minimize undesirable "back-drive" (or imposition of negative torque on the drive elements of a pumping unit) during various periods of each complete stroke cycle and to insure good over-all power efficiency, it is necessary to locate the drive equipment on the well side of the samson post. Such an arrangement of components, however, is not only strange and unfamiliar to most oil field operators, but it also renders more hazardous the job of servicing the well head area and associated equipment elements due to the closer proximity of the drive equipment.

Accordingly, it is now a primary object of the present invention to minimize the possibility of encountering high peaks or sudden changes to excessive levels in the net torque and to tend to equalize the net torque level applied to the crank shaft during various portions of the pumping cycle of a unit in which the walking beam is mounted in more conventional fashion, i.e., on a pivot point near the medial portion thereof. Moreover, in achieving this more nearly uniform net torque level throughout the pumping cycle of a more conventional unit it is a further object of the present invention to reduce the degree of dependence on crank weight counterbalancing and, therefore, the amount of such counterweights required for a given job, by providing an arrangement of component parts which inherently also more nearly equalizes the *actual* well load torques applied to the crank during various parts of the pumping cycle.

Essentially, this means a unit design which inherently compensates to an unexpected degree for the extra well load always carried on the upstroke of a functioning pumping unit due to the weight of fluid lifted and delivered to ground level on each stroke.

Some of the immediate practical advantages of achieving the above objects, in addition to the saving on counterbalance requirements, would be the realization of a reduction in the size of prime mover, speed reducer and other elements of drive equipment for a given unit of specified capacity and an over-all saving in power demand on a given job. Other objects and advantages will be obvious or will become apparent from the detailed description of the invention which follows.

In accordance with the present invention it has now been found that the above objects and advantages can, in fact, be realized in a mid-mounted type of walking beam pumping unit provided the rotary crank which, via a pitman arm or arms, supplies the oscillating motion to the walking beam is located at a substantially more rearward position (from the samson post and the well) than has been previously used or recommended. More specifically, the crank shaft should be located sufficiently toward the rear so that substantially the entire circle described by the crank wrist pin lies to the opposite side from the well of a straight line passing through the tail bearing pivot point at its lowest position of travel and running at right angles to the straight line drawn between the saddle bearing pivot point of the beam and said tail bearing pivot point at the midstroke position of the unit.

A further most surprising discovery of this invention is the fact that, when the aforesaid condition is met, then, for those pitman arm to crank arm ratios of greatest practical interest (namely from about 2.5 to 1 to 6 to 1), the maximum effective lever arm (i.e., the maximum torque factor) of the unit during the upstroke plus the maximum effective lever arm or torque factor during the downstroke add up to a sum *less than* the actual total stroke of the unit. This is an indication of the amazingly good overall efficiency of which the pumping units of the present are capable since it has long been known that, in conventional pumping units having an essentially symmetrical location of the crank with respect to the tail bearing (i.e., such that said line through the tail bearing pivot point and at right angles to the straight line connecting saddle bearing and tail bearing joints at the midstroke passes through or near the crank shaft), then the sum of the maximum effective lever arms of the unit on the upstroke and downstroke must necessarily be greater than the actual stroke realized by the unit except in the impractical or idealized case where the ratio of the pitman length to crank arm approaches infinity. See U.S. Patent 3,109,313 which discusses these previous limitations on efficiency and describes the possibility of approaching such idealized efficiencies in a practical unit by substituting for the conventional tail bearing connections between rigid pitman arm(s) and the beam, a flexible spring connection to an arcuate mulehead on the driven end of the walking beam, thus permitting the angle between the pitman arm and the acting line of the beam to approximate 90° at all times during the complete stroke cycle. The preferred embodiments of the present invention are represented by those crank shaft locations which, as stated above, are sufficiently toward the rear so that substantially the entire circle of rotation of the crank arm lies to the rear of said straight line passing through said tail bearing as described above but not so far to the rear as to necessitate pitman arms of impractical length or tail bearings of impractical size. As will be seen from more quantitative considerations presented hereinafter, this generally means that the horizontal distance of the crank

behind said line through said tail bearing should not be over about two-thirds of the pitman arm length.

In order to explain our invention in greater detail including quantitative aspects and preferred specific embodiments thereof, reference is now made to the accompanying drawings, in which:

FIGURE 1 is a side elevation of a pumping unit constructed in accordance with the present invention;

FIGURES 2, 3 and 4 are largely diagrammatic representations of unit layouts similar to that of FIGURE 1 and illustrating representative variations in the range of preferred positions for the drive crank;

FIGURE 5 is a side elevation view of a pumping unit similar to that of FIGURE 1 but showing a slightly different method of applying crank counterbalance to units of the present invention.

Referring now to the drawings and particularly to FIGURE 1, it will be seen that the basic components of the pumping units of this invention are decidedly conventional, including a base frame **10**, a samson post or upright frame **12** and a walking beam **14** mounted for oscillation in a vertical plane by means of saddle bearing **16** at the top of the samson post. At the forward or right-hand end of the walking beam there is a mulehead **18** from the top of which there is suspended a wire line **20** to a hanger **22** by means of which the rod string (not shown) is held in order to pump the well **24**. Oscillation of the walking beam **14** is effected by means of pitman arm **26** pivotally connected by means of tail bearing **28** to the rear or left hand end of the beam and by means of wrist pin **25** to the crank **23** of speed reducer **21**. Reducer **21**, in turn, is driven by means of a prime mover (not shown) of any suitable type. Counterbalance can be provided in any of the usual manners, such as by mounting crank counterweights **27** off of crank shaft **29** or by placing weights **31** directly on the walking beam **14** or by a combination of crank weights **27** with beam weights **31**.

Referring now to FIGURES 2, 3 and 4, simplified diagrammatic layouts of units similar to that of FIGURE 1 are presented. In order to demonstrate the quantitative aspects of the present invention, these layouts of FIGURES 2, 3 and 4 incorporate a considerable variation in the location of the crank shaft within the scope of the present invention. Thus, FIGURE 2 represents a most preferred arrangement embodying the present invention in that the pitman arm to crank arm ratio of 4.0 is not only within the optimum central area of the practical range but the crank shaft location is also in the middle ground of the invention, well removed from any of the boundary limits. FIGURE 3 actually depicts two different layouts representing approximately the extreme or limiting positions within the scope of this invention for locating the crank shaft of a unit having exactly the same pitman and crank arm lengths as the unit represented by FIGURE 2. FIGURE 4, on the other hand, depicts a typical but nonlimiting location for a unit with a substantially different pitman arm to crank arm ratio. In said FIGURES 2 through 4, point S represents the saddle bearing pivot point on top of the samson post **12**. Points H, M and L represent the pivot point of the tail bearing at its highest, middle and lowest positions respectively during a stroke cycle. Thus, straight lines LS, HS and MS represent the effective lines of action of the walking beam at the top, bottom and midstroke positions, respectively, of the mulehead, while **18** and **18'** represent the mulehead in its extreme positions. Likewise, points C, C_R and C_F represent centers of the crank shaft, and points W_a and W_b represent the positions of the wrist pin connecting crank and pitman arms when the tail bearing pivot point is located at extreme points H and L, respectively. Thus, the circle on which points W_a and W_b lie represents the circle of rotation described by said wrist pin. For the purposes of the present invention, it is convenient to define the location of the crank shaft in terms of its distance behind the dotted lines drawn through point L in FIGURES 2 through 4, which represent in each case a straight line drawn at right angles to a straight line (such as line MS), connecting the pivot points of the saddle bearing and the tail bearing at the midstroke position of the unit. The key quantitative relationships of the unit arrangements represented by the layouts of FIGURES 2, 3 and 4 are set forth in the following table.

TABLE I

Case	P/R ratio	Relative distance of crankshaft behind line LZ
Figure 2	4.0	.43P
Figure 3 (forward)	4.0	.25P
Figure 3 (rear)	4.0	.66P
Figure 4	5.3	.49P

Standard methods of calculating the effective or equivalent over-all lever arm or "torque factor" for a pumping unit at any given position of its stroke from the basic geometrical relationships of the unit layout are by now well known, having been worked out under the auspices of the A.P.I. and adopted throughout the industry. (See, for example, the brochure by D. O. Johnson entitled "Torque Factors for Pumping Units," published as part of the Proceedings of the West Texas Oil Lifting Short Course of April 1959.) Thus, it is known that these "torque factors" can be calculated from the following equation:

$$\text{T. Factor} = \frac{AR}{C} \times \frac{\sin \alpha}{\sin \beta}$$

where

A is the length of the well working arm of the walking beam, i.e., the horizontal distance between the saddle bearing pivot point and the center line of the well,

R is the crank arm length, i.e., the radius of the circle of rotation of the wrist pin,

C is the distance between saddle bearing and tail bearing pivot points,

α is the angle between the pitman arm and the crank arm, and

β is the angle between the pitman arm and the straight line connecting tail bearing and saddle bearing pivot points.

Since A, R and C all remain constant throughout the full stroke cycle of a unit, the variations which occur in "torque factor" values during the stroke will obviously be determined by the interrelationships between angles α and β.

Using the above relationship, the "torque factors" were calculated at frequent intervals throughout the pumping stroke cycles of each of the unit layouts represented by the cases set forth in Table I above, the length of the well working arm, A, employed in each set of calculations having been first selected so that each layout would deliver the same actual well working stroke, namely 74 inches. In this way the maximum "torque factor" during the upstroke and the maximum "torque factor" during the downstroke were determined for each of the four cases specified in Table I with the following results:

TABLE II

[all values in inches]

Case	Max. T. factor (upstroke)*	Max. T. factor (downstroke)*	Sum of Max. T. factors
Figure 2	32.43	39.93	72.36
Figure 3 (forward)	35.07	38.58	73.65
Figure 3 (rear)	26.32	42.10	68.42
Figure 4	32.38	39.73	72.11

*For clockwise rotation of the cranks.

It will be seen that in each of the four cases the sum of the two maxima in "torque factors" is at least slightly less than the total 74 inch working stroke of the unit. Although the over-all reduction in these maximum "torque factors" is most impressive in the rear layout of FIGURE

3, i.e., with the crankshaft located at point C_R, as a practical matter this location is less desirable than the others illustrated because, as a result of the magnitude of the forces involved, excessively large and heavy bearing assemblies would be required. Furthermore, said crank location, C_R, as depicted in the rear layout of FIGURE 3, is in close proximity to the limit line (dotted line LX) at which the pitman could break downward at the tail bearing connection when it reaches point L (i.e., at the top of the unit stroke).

The forward layout of FIGURE 3 with the crank shaft at point C_F represents a preferred limiting position in the other direction for a unit layout in which the crank and pitman arm lengths are exactly the same as in the setup depicted in FIGURE 2 and in the rear layout of FIGURE 3. Thus, as suggested by the relatively small reduction in sum of maximum "torque factors" shown in Table II above for the case of FIGURE 3 (forward), further movement of the crank shaft toward the well along the arc C_R—C_F will cause a general increase in "torque factors" so that the sum of the two maxima therein will exceed the stroke. Therefore, location of the crank shaft in accordance with the present invention should be sufficiently to the rear of the tail bearing so that substantially the entire circle of rotation of the wrist pin lines to the rear of the straight line through the tail bearing at its lowest point L and running at right angles to the line (MS) between the saddle bearing and the tail bearing at its mid-stroke position.

For best results the crank will be located in the middle ground between said extreme positions such as at point C of FIGURE 3 (which represents the location laid out in detail in FIGURE 2), or such as that shown in FIGURE 4 for a unit with a different pitman arm to crank arm ratio. In short, it can be seen from the above considerations balancing over-all efficiency with utility, soundness and economy of construction, and ease of maintenance, the crankshaft should be located so that substantially the entire circle of rotation of the wrist pin lies with an elevational area such as that in the lower left hand corner of FIGURE 4 (below line XL and to the left of line LZ).

In addition to the improved over-all efficiency indicated by the over-all reduction in "torque factors" attained, several other definite advantages accrue from the present invention. Thus, as is indicated by the figures on maximum "torque factors" given in Table II above, the "torque factors" on the upstroke for clockwise rotation of the crank in units of this invention are substantially lower over practically the full cycle than the respective figures at corresponding positions of the downstroke. This is most beneficial since it tends to provide inherent compensation for the increased well load at the polished rod due to the differential weight of fluid lifted and delivered each stroke. Consequently, the *actual* well load torques applied to the crank during the downstroke tend to approach those on the upstroke much more closely than in previous designs of conventional units, thus greatly decreasing one's dependence upon the use of crank weight counterbalancing and, accordingly, the amount of counterweights which must be provided and used in practice.

It will also be seen from the diagrams of FIGURES 2 through 4 that the angle *b* representing the position of the crank in a clockwise direction beyond its 12 o'clock or 0° position at the end of the upstroke is always more than 180° greater than angle *a*, representing the position of crank at the beginning of the upstroke. The accurate figures for the four cases illustrated in FIGURES 2 through 4 are given in the following table.

TABLE III

Case	Angle *b* (deg.)	Angle *a* (deg.)	Difference (*b*−*a*) (deg.)
Figure 2	212	17	195
Figure 3 (forward)	195	7	188
Figure 3 (rear)	247	35	212
Figure 4	219	26	193

Thus, it will be seen that, for clockwise rotation of the crank as viewed in the drawings, the period of the crank rotation cycle devoted to the upstroke is always somewhat greater than 180°. If the speed of rotation of the crank is at all constant, this means that the present unit will favor movement of the rod string on the downstroke at an average velocity higher than that on the upstroke, thus providing a situation much more in harmony with the way other factors and natural phenomena, such as gravity, tend to influence the motion of the rod string. Thus, the decreased instantaneous speeds on the upstroke help to avoid adding extra dynamic load due to frictional effects, etc., at the very time when the inertial loading itself is already high. Furthermore, moderate increases in down hole instantaneous speeds are generally in line with the natural acceleration of the rod string due to gravitational pull. However, the downstroke acceleration must not be increased to the point of causing separation of the polished rod clamp and wire line hanger to occur since this might damage the polished rod or other parts of the pumping unit. For this reason, it is not recommended that the upstroke period of the pumping cycle occupy substantially more than about 210° out of the 360° total.

Considering all the above factors, it can be stated as a general rule, therefore, that the location of the crank shaft should be to the rear of the specified straight line through the tail bearing (such as LZ) by a distance at least as great as about one quarter the length of the pitman arm but not greater than about two thirds of said pitman arm length so that the angle *a* at the beginning of the upstroke will be at least about 7° and not substantially greater than about 35° and the difference between angle *b* and angle *a* will not be substantially less than about 190° nor substantially more than about 210°.

As already mentioned in connection with the description of the unit shown in FIGURE 1, the most common method of providing counterbalance in pumping units of the walking beam type is the addition of weights either on the crank arm or on the walking beam itself. It is, of course, obvious that the moment exerted by a given mass of such counterweights will increase as the center of mass of said weights is moved out from the crank shaft or the saddle bearing respectively, but there are naturally practical limits on such distances. Thus, any decrease in need for and dependence on counterweights is always a definite advantage. However, it is of even greater advantage in the present invention because, the counter-moment produced by a *crank-attached* counterweight being inherently a perfectly regular sinusoidal function, special problems are presented in the use of crank-attached counterweights to counterbalance a unit like the present ones in which the deviation of the actual dynamic well load on the crank from a regular sine wave is further exaggerated due to the appreciable differential between upstroke and downstroke periods. Accordingly, in many units designed in accordance with the present invention, the reliance on crank-attached counterweights can be sufficiently minimized that counterweights can be mounted on the crank shaft in the same manner most commonly used for conventional units, that is in line radially with the crank arm as shown in FIGURE 1.

Naturally, if the effect of the crank-attached counterweights assumes sufficient importance, they can also be mounted in such a way as to act somewhat out of phase with the drive crank itself. For example, this can be accomplished by using adjustable, or spreadable multipiece (e.g., sheave or butterfly) type counterweights, which permit angular adjustment in the center of mass of such weights with respect to the crank mounting even when mounted directly on the crank arm as shown in FIGURE 1. Alternatively, as shown in FIGURE 5, a separate counterweight arm **19** can be provided which is angularly offset with respect to crank arm **23** by an angle *d* for example. Also, here again, the crank weights **32** can be adjustable, e.g., from position **32** to position **32'**, so

as to change the radial location of their center of gravity, thus changing the effective phase angle d to a different value, such as d'. However, it must be realized that the shape of the dynamic well load curves (or Dynamometer cards) varies so widely for different wells that any fixed phase location of a crank-attached counterweight represents at best some compromise. This explains the preference with the present invention for the use of an angularly adjustable mounting for crank-attached counterweights of major size or significance as illustrated in the dotted lines of FIGURE 5. As a compromise arrangement for those intermediate cases where the crank-attached counterweights represent a factor too important to permit the use of an ordinary fixed mounting radially in line with the crank arm itself but not sufficiently important to justify an angularly adjustable mounting, it is recommended that the counterweight mounting arm be offset so that the crank weights exert their maximum effect (i.e., so that their center of gravity reaches the 90° and 270° positions of rotation about the crankshaft) when the pumping unit is at approximately its midstroke position.

In order to demonstrate a preferred embodiment of the present invention and the valuable advantages obtainable through actual operation thereof, the following detailed specific example is presented.

EXAMPLE 1

A complete pumping unit was assembled with a drive system conforming closely in layout with that shown in FIGURE 2. Thus, the height of the saddle bearing pivot point S on top of the samson post **12** was about 17 feet above the bottom of the base, while the pitman arms and crank arms were 118.4" and 29.6" long respectively (P/R=4.0). With a distance of 96.2" between saddle bearing and tail bearing pivot points, the mulehead was located on the walking beam so as to give a pumping stroke length of 74 inches, which required a well working arm of about 105.7 inches (i.e., the horizontal distance from saddle bearing pivot point to vertical centerline of the well). The maximum "torque factors" and the angular positions of the wrist pins at the beginning and end of the upstroke are accordingly as set forth in Tables II and III hereinabove for the FIGURE 2 case.

Since it was found that the wrist pins reached a clockwise position of about 110° at the midpoint in the upstroke of this unit, a counterweight arm was mounted on the crank shaft having a centerline 20° behind the radial line between the wrist pins and the center of the crank shaft so that the center of gravity of the crank mounted counterweights would reach the 90° or 3 o'clock position of maximum torsional effect exactly at this midstroke position.

This unit was tested by using it to lift oil from an actual well having a fluid level about 2750 feet below ground. Its performance on this well was compared directly against a conventional pumping unit of the same structural capacity and stroke length of 74". Both units were tested at pumping speeds of 10, 15 and 20 strokes per minute using the same electric prime mover (a 25 H.P.–440 volt GE motor) for both in order to avoid motor speed torque variations. Since the conventional unit had a substantially symmetrical crank shaft location, i.e., substantially directly under the tail bearing, the counterweights for this unit were mounted radially in line with the center of the crank shaft and the wrist pins, rather than out of phase therewith.

During these comparison tests described above, a reduction in peak torque at the crank of between about 21 and 34% (depending mostly on pumping speed) was experienced on the unit of the present invention over that on the conventional unit, with an average reduction of about 28%. Likewise a significant reduction in crank counterweights (averaging about 13.5%) was successfully effected with the experimental unit. Likewise, the average power demand was reduced by about 17% with the new unit over that of the conventional apparatus. This indicates that significant reductions in size and capacity of both prime movers and speed reducers would be possible through use of the apparatus of this invention.

Investigations with other pumping units constructed in accordance with the present invention and having different pitman arm to crank arm ratios, such as 5 to 1 or 3 to 1 indicate that similar advantages can be obtained over the entire practical range of P/R ratios extending between about 2.5 and 6.0.

Having thus described our invention in detail together with preferred, illustrative embodiments thereof, what we claim and desire to secure by U.S. Letters Patent is:

1. In an oil well pumping device of the type in which reciprocation of the sucker rod string in the well casing is effected by attaching said rod string to one arm of an oscillatory walking beam pivotally mounted by a saddle bearing near the medial portion thereof to a samson post, said walking beam having on the other arm thereof a tail bearing pivotally connected to a pitman arm the opposite end of which is equipped with a wrist pin attached pivotally to a rotary crank the length of which is not less than ⅙ the length of said pitman arm, said crank being mounted on a shaft which bears counterbalancing weights thereon and is positively driven in a clockwise direction as viewed with the well working end of the unit to the right, the improvement which comprises locating said crank shaft out from under the area directly beneath the operating arc of said tail bearing and so that substantially the entire circle of rotation of said wrist pin lies to the rear of a straight line passing through the center of said tail bearing and drawn at right angles to the straight line connecting said tail bearing and said saddle bearing at the midstroke position of the unit.

2. The improvement in an oil well pumping device as specified in claim **1** in which the position of said counterbalancing weights on said shaft is such that the center of gravity of said weights lags behind the wrist pin position as the crank rotates in said clockwise direction.

3. The improvement in an oil well pumping device as specified in claim **2** in which the position of the center of gravity of said counterweights is set so that said center of gravity will be at the 3 o'clock or 90° position of its rotation clockwise about the crank shaft when the well working end of the walking beam has moved to the midpoint of its upstroke.

4. In oil well pumping devices of the type in which reciprocation of the sucker rod string is effected by attaching said string to one arm of an oscillatory walking beam mounted near the medial portion thereof by means of a pivotal connection atop a samson post, said beam being driven by a pitman pivotally depending from a tail bearing on the other arm of said beam which pitman is pivotally connected in turn to a rotary crank mounted on a driven shaft having counterbalancing weights mounted thereon, the improvement which comprises proportioning and arranging the location of the several elements so that the effective pitman to crank ratio is between 2.5/1 and 6/1 and the crank shaft is sufficiently spaced from the samson post so that it is out from under the area beneath the operating arc of said tail bearing and so that substantially the entire circle of rotation of the pivot point of the connection between crank and pitman lies to the opposite side from the well and samson post of a straight line passing through the tail bearing pivot point at its lowest point of travel and drawn at a right angle to the straight line between the saddle bearing pivot point and the tail bearing pivot point at its midstroke position thereby producing a combination of torque factors such that the sum of the peak upstroke and peak downstroke torque factors is less than the actual stroke length delivered by the pumping device so assembled.

5. The improvement in oil well pumping devices as specified in claim **4** in which the said counterbalancing weights are mounted from said crank shaft in a position

sufficiently offset in a counterclockwise direction from the radial position of the crank that the center of gravity of said weights will be at the 3 o'clock or 90° position of clockwise rotation about the crank shaft just as the walking beam reaches the midpoint of the upstroke on the rod string, thereby providing maximum torsional effect of said counterbalancing weights on said crank shaft at about the same time as the peak upstroke torque factor occurs.

6. The improvement in oil well pumping devices as specified in claim 4 in which the location of the drive shaft is also such as to place the center of said shaft not more than about two thirds of the effective pitman length behind the straight line drawn through the tail bearing pivot point at its lowest position and at right angles to the straight line connecting the saddle bearing pivot point of the beam on top of the samson post with the tail bearing pivot point at the midstroke position of the unit.

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MILTON KAUFMAN, *Primary Examiner.*