# Recent Advances In Beam-Type Unit Design

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ABSTRACT

By arranging the geometry of the beam type oilfield pumping unit as a non-symmetrical front-mounted lever system, it is possible, in most cases, to fit more closely the counterbalance torque pattern to the well load torque pattern, with a significant reductions in peaks. This system results in a relatively uniform loading of both prime mover and gear reducer; the loading in turn allows the operator to handle many applications with smaller equipment and reduced first cost.

Indirectly, this increased torsional uniformity can be used to help reduce three of the primary causes of sucker rod fatigue failure and to minimize rod maintenance costs as well as attendant production losses.

This unique geometry further makes practical the semi-automatic counterbalancing of the pumping unit and employs its own energy of rotation to re-position a portion of the crank counterweight system. This semi-automatic system is constructed of simple mechanical components, which require little or no maintenance; and counterbalancing is accomplished while the unit is in operation by simply throwing a lever in or out of engagement.

INTRODUCTION

Although the engineer and designer have made substantial progress in improving the individual components of the beam type oilfield pumping unit, its geometry and hence its functional ability have changed little within the past century.

Until recently, the solution to several of the conventional beam unit's major functional problems had eluded the designer; and the industry had come to feel that the walking beam and sucker rod pumping system had reached the stage of development upon which few, if any, significant advances could be made.

Three of these major unsolved problems were (1) how to present the reciprocating, differential load of the polished rod string as a smooth and relatively uniform torsional load to the gear reducer and prime mover, (2) how to lift rods and fluid in such a way as to require the least amount of force, i.e., a reduced peak polished rod load for any given application, and (3) how to balance, effortlessly and safely, the pumping unit while in operation, by mechanical means, and by employment of the unit's energy of rotation to re-position a portion of the counterweight system.

Careful study of the torsional and structural problems mentioned above shows that it has been possible, by using dynamic and kinematic analysis, to design a new pumping machine (Fig. 1), that is capable of producing a smaller peak polished rod load and a relatively constant torque for any given application, and to balance the machine to a sizeable variation in well load. This variation is achieved by the use of a simple, mechanical, crank balanced system; the unit need not be stopped, and it requires only the actuating of a lever to increase or decrease counterbalance effect.

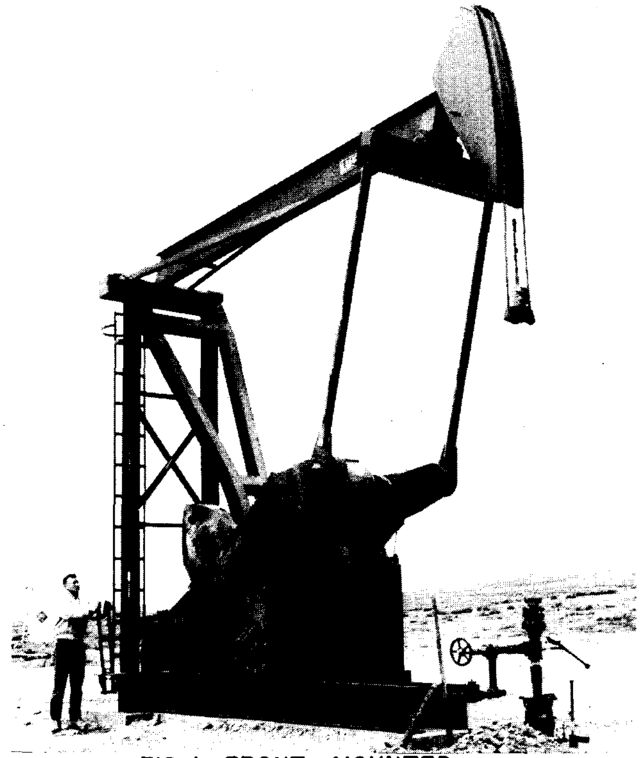


FIG. 1 FRONT MOUNTED PUMPING UNIT

These improved characteristics are brought about by a geometric re-arrangement of the components of the conventional unit and change it from a symmetrical Class I lever system (fulcrum at mid-beam) to a non-symmetrical Class III lever system (fulcrum at rear beam), and the well load and counterbalance torque are dephased by casting an angular off-set in the crank and moving the gearbox away from the well a proportionate amount, while driving the machine in a particular direction of rotation.

This new, front-mounted, mechanical pumping unit combines the simplicity, ruggedness, high-efficiency and relatively trouble free operation of the conventional unit, with the beneficial rod motion and ease of counterbalance of the air balanced system, and, in addition, provides a relatively uniform torque demand on both prime mover and gear reducer, which, in turn, permits the use of smaller and less expensive equipment to handle many applications.

To understand the uniform torque system of the front-mounted pumping unit, it is helpful to briefly review the torsional characteristics of the well known conventional beam pumping unit.

## THE BEAM PUMPING UNIT TORQUE SYSTEM

At the crankshaft of any beam type pumping unit, there are two torsional forces, or moments: one trying to rotate this shaft in one direction, the other attempting to turn it in the opposite direction. One of these torques is developed by the counterweight system, the other by the well load; the difference between these two moments is the net crankshaft torque, and it is all that the reducer and prime mover actually see. The more closely that the well load torque curve can be fitted to the counterbalance torque curve, the more uniform the net crankshaft torque becomes, with an attendant reduction in peaks, plus a proportionate reduction in the size requirement of both prime mover and gear reducer. By an individual examination of each of these two moments it is easier to see what factors must be adjusted to obtain this "perfect torque fit"

## CONVENTIONAL TORQUE REQUIREMENTS

Looking at the symmetrical Class I lever system (conventional unit) graphic torque analysis in Figure 2, the first of these two moments, or the counterbalance torque, describes a simple sine curve throughout the crank cycle, i.e., when the counterweight is vertically upward (bottom stroke) there is no counterbalance torque at the crankshaft; when horizontally forward (mid-upstroke) maximum counterbalance torque is developed; when vertically downward (top stroke) again there is no counterbalance torque; and when the weights are horizontally backward (mid-downstroke) once more maximum torque is developed by the counterweights alone.

The second of these two moments, the well load torque, is also zero at the crankshaft near the bottom of the polished rod stroke; but because of lifting its maximum load, rods and fluid accelerated upwardly it reaches its maximum value about mid-upstroke ($90^{\circ}$), then it drops back to zero torque at the top of the stroke ($180^{\circ}$), and returns to its maximum downward value at approximately mid-downstroke ($270^{\circ}$), and hence back once more to zero.

Now the difference between these two torque curves--the well load torque and counterbalance torque--is the net torque as seen by both gear reducer and prime mover; and the area between them is proportional to the work performed during one revolution of the crank, while the capacity of the gear reducer, required to handle this amount of work is proportional to the maximum height of this work pattern - i.e., ordinates "B" and "E" (Fig. 2).

For the conventional unit under ideal conditions the difference between the two torque curves -- i.e., their closeness of fit -- is shown as net crankshaft torque at the bottom of Figure 2, and varies from zero to maximum twice during each rotation of the crank. Superimposed on this fluctuating, conventional net crankshaft torque (or work) load (bottom Fig. 2), is the ideal torque rectangle described by the dotted line. This rectangle has the same area as do the two torque lobes and indicates equivalent work; but it has a maximum ordinate or peak torque (horsepower) requirement which is considerably smaller. The inability of this type of lever system to produce a better fit between well load and counterbalance torque makes it necessary to size the gear reducer and prime mover to the maximum differential ordinates "B" and "E", (Fig, 2) representing peak up and down stroke torque.

## FRONT-MOUNTED TORQUE SYSTEM

By fitting the counterbalance the front-mounted lever

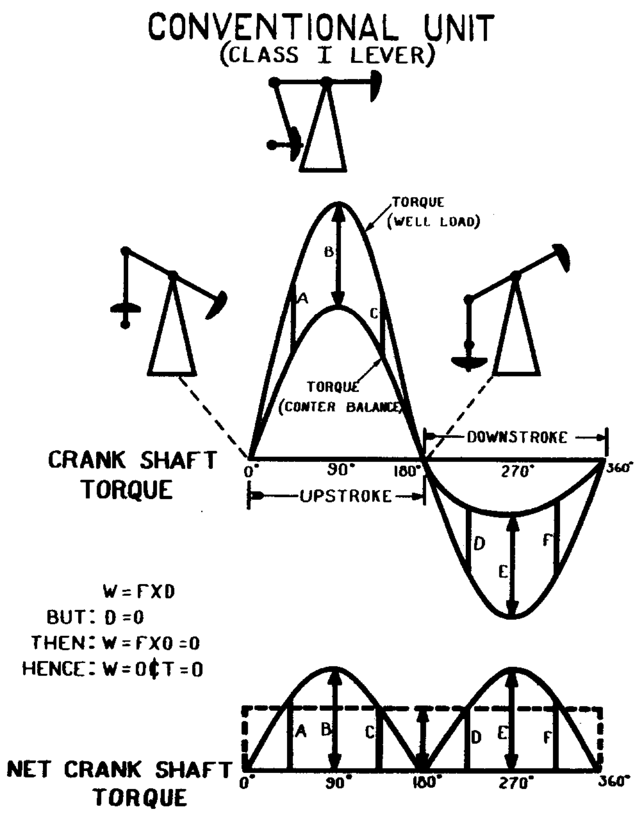



**FIG. 2 CONVENTIONAL UNIT TORQUE SYSTEM**

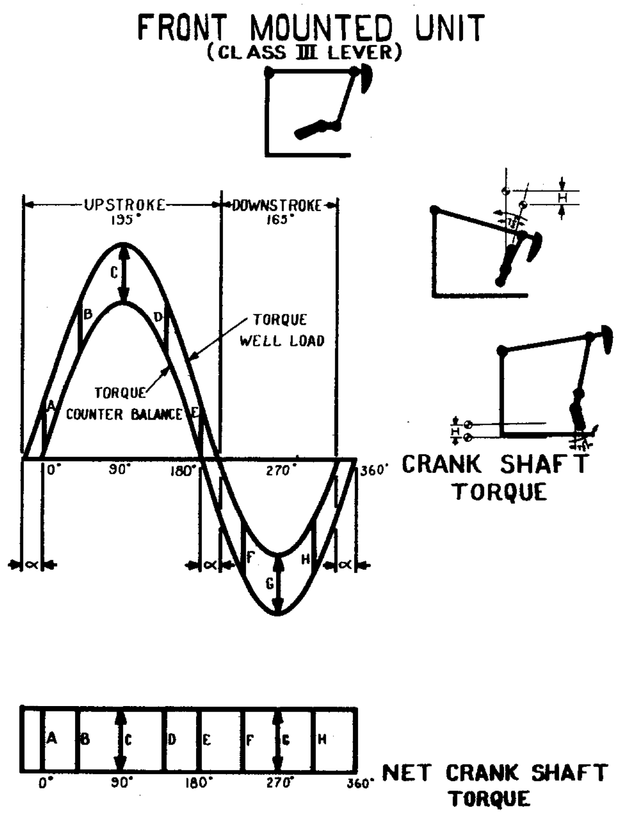



**FIG. 3 FRONT MOUNTED UNIT TORQUE SYSTEM**

system makes possible the performance of work at a much more constant rate torque and well load torque curves much more closely together as shown in Figure 3, and to produce a resultant net torque which approaches the ideal rectangle shown at the bottom of Figures 2 and 3.

Suppose the same counterbalance torque curve, i.e., sine wave, is taken as a fixed reference (Figs. 2 and 3), what will be necessary to produce a perfect fit between it and the well load torque curve so that their difference will always be a constant net crankshaft torque? The three adjustments (Fig. 3) that must be made for the well load torque to produce this constant difference, are shown below:

1. The well load and crankshaft torque must be dephased by some angle Alpha ( $\alpha$ ).
2. The upstroke cycle must be increased, and the downstroke cycle decreased.
3. The well load torque at mid-upstroke must be decreased and the mid-downstroke torque must be increased.

Fortunately, it is possible to make several simple geometric modifications to the front-mounted pumping unit structure so it will tend to generate a well load torque curve that, over a wide variety of field applications, more closely fits the counterbalance torque curve and thus approaches, in practice as well as in theory this desired uniform torque system.

These geometric structural modifications are:

1. By casting a certain angular offset in the cranks and driving in one particular direction of rotation, it is possible not only to dephase well load and counterbalance torque by the required amount, but also to work the machine over both top and bottom of the stroke, while proportionately reducing side load work.
2. By offsetting the gear reducer away from the well head and rotating in a particular direction of rotation, a long up cycle and a short down cycle are produced.
3. By varying the effective lever arm - lengthening it on the upstroke and shortening it on the downstroke - upstroke torque is reduced and downstroke torque is increased, and all requirements for the uniform torque system are fulfilled.

Independently these modifications will not produce uniform torque, but by working together, the net crankshaft torque can be reduced as much as 40 per cent. This reduction depends upon the type of application encountered.

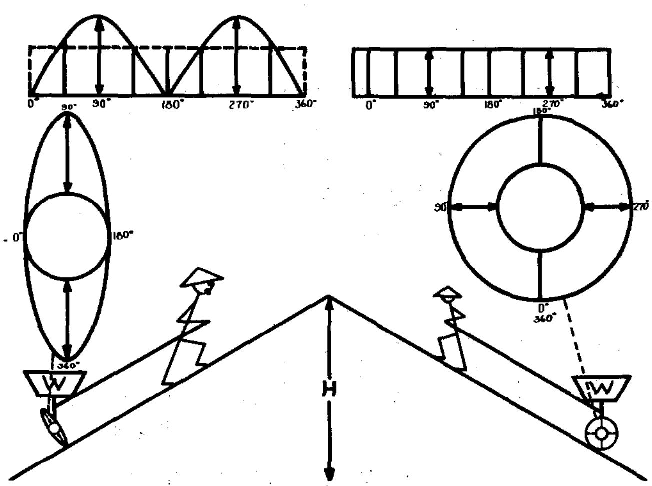



FIG. 4 WORK, TORQUE & HORSEPOWER COMPARISON

Laying out the net torque load pattern of the conventional unit (Fig. 4, left hand side) around its crank circle and using it for a wheel and doing the same for the front-mounted unit (Fig. 4, right hand side), one can be seen that the horsepower requirements for the left weight (W) would be greater than for the right weight (W), although the work done in lifting both loads to the top of the incline is identical. Thus, by minimizing peak torque at both gearbox and prime mover, not only has equipment first cost been reduced but also continued operating costs as well.

## SEMI-AUTOMATIC COUNTERBALANCING

Of the front-mounted unit the second major feature that makes possible many of its desirable characteristics is the semi-automatic counterbalancing device shown in Figure 5. Within the long, uninterrupted, counterbalance arms (a), which are located on the opposite side of the shaft from the crank and pin holes, is placed a massive iron slug (b) with a hole (c) cast longitudinally through it. A nut (d, not shown) is attached to the slug, with a long drive screw (e) running through both the hole in the slug and the nut.

As the drive screw is rotated in one direction, the slug is screwed in toward the center of crank rotation and reduces the counterbalance. When screwed oppositely the slug is drawn away from the center of rotation, and counterbalance is increased.

Mounted at the head of the drive screw is a small right angle gearbox (f) which has two coaxially mounted v-belt sheaves (g, h) for driving it. When the inboard sheave (g) is actuated, the drive screw is turned in one direction, and the slug moves in toward the center of crank rotation and actuating the outboard sheave (h) which drives the slug oppositely.

A simple v-belt clutch (i) is mounted around the crankshaft and in line with the inboard and outboard sheaves and v-belts. When more counterbalance is needed, a lever at the rear of the pumping unit (Fig. 1) is held momentarily outward - engaging the outboard belt whose relative motion turns the outboard sheave, thus driving the right angle gearbox and screw in such a manner as to draw the slug outward. Actuating the rear lever oppositely engages the inboard belt, whose relative motion turns the inboard sheave; thus the right angle box and screw are drivin in a manner that draws the slug inwardly.

Releasing the lever causes the slug to remain at a constant radial distance from the center of rotation. Slugs are installed in both cranks, and right and left levers are used to actuate them.

Running these two trim weight slugs from completely in to out affects a counterbalance adjustment approximately equal to 25 per cent of the torsional capacity of the gear reducer.

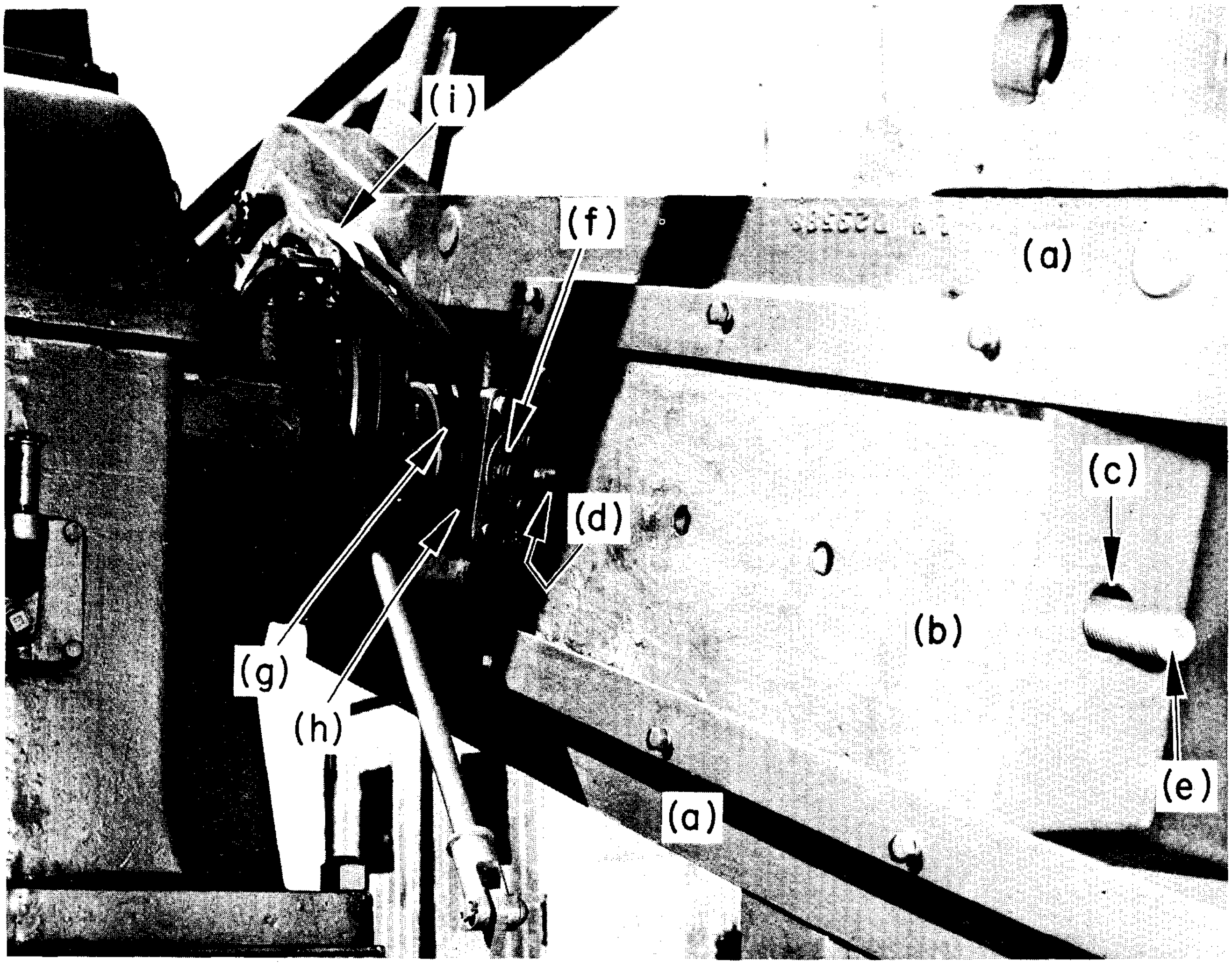
The semi-automatic system is so designed that, depending upon whether or not variation in well loading warrants its installation, it may be added or removed from the unit at any time.

The standard master weights on the outside of the counter-weight arm may be adjusted in the normal manner if radical changes in counterbalance are required.

This simple, safe, mechanical, and trouble-free method of keeping the unit in correct counterbalance without stopping it helps to insure maximum life for all unit components and prime mover as well as it helps to reduce power costs.

## ROD MOTION

One of the most significant mechanical properties of the




## FIG. 5 SEMI—AUTOMATIC COUNTER BALANCE SYSTEM

front-mounted unit is its ability to reduce peak polished-rod loading in most applications; in turn, this reduction of loading helps to decrease rod breaks caused by fatigue, and thus helps to minimize rod maintenance expense and consequent loss of production.

In practically all pumping unit applications the sucker rod string is designed to operate within its safe allowable load limit. If successful, a rod break seldom, if ever, would occur because of fatigue. But rod strings do part, sometimes all too frequently; and one of the most common causes of failure is overloading the string.

All else equal, three principal factors control peak polished rod loading: (1) the weight of the rods themselves, (2) the weight of the fluid column, (3) the acceleration that rods and fluids must undergo as they are lifted off bottom. Now it is this off-bottom acceleration that punishes the string, because it is here that the upward acceleration is the greatest; and it is this maximum upward acceleration that helps load the rods and unit so heavily, although it may take some time for this peak load impulse to be felt at the surface.

Thus, for any given combination of rods and fluid, the greater the off-bottom acceleration, the greater the peak polished rod load. And since the mass of rods and fluid cannot be reduced, peak rod load reduction must come as the result of reducing maximum off-bottom acceleration.

The top rod reversal, however, is of much less significance, because it is here that the rod string tends to unload by an amount equal to the top acceleration component. This tendency is intuitive, for if a person holding two heavy suitcases, steps on an elevator, they will seem heavier as the elevator accelerates upward from the first floor, but will seem lighter as the elevator de-accelerates and comes to rest at the top.

In a pumping unit of any particular class lever system, i.e., Class I or Class II or Class III with the cranks turning at constant angular velocity, the bottom reversal rate, or off-bottom acceleration, is primarily a function of only one variable, the pitman to crank ratio.

On a Class I lever system such as the conventional unit, the greater the pitman to crank ratio, the higher the off-bottom acceleration; but on a Class III system, like the front-mounted mechanical unit or the air balanced unit, just the reverse is true: the greater the pitman to crank ratio, the slower the off-bottom reversal.

Because of the low pitman to crank ratio of the front-mounted unit, the acceleration at the bottom of the polished rod reversal is decreased as much as 40 per cent under that of a conventional unit whose cranks run in synchronism and have the same pitman to crank ratio. This more beneficial rod motion reduces peak loads as much as 10 to 15 per cent.

Although reduction of peak polished rod load alone is beneficial to the rod string, there are two other factors, of near equal importance, that can further reduce rod fatigue failures. These factors are (1) the reduction of the number of rod reversals, and (2) the reduction of the stress range of each reversal.

In many applications it is possible to combine the benefits of the low off-bottom acceleration of the front-mounted unit with its uniform torque characteristics to reduce not one but all three of the above mentioned factors controlling sucker rod life. The following typical example illustrates how this combination is accomplished.

Suppose that there arose a requirement to lift approximately 200 BPD (at 100 per cent volumetric efficiency) from 5000 ft with a 1-1/2 in. pump and 3/4 in. rod string. In use is a 64 in., 160,000 in. lb conventional unit, pumping 14 strokes per minute. The approximate loading would be:

(1) Peak Polished Rod Load . . . . . . 12,510 lb
(2) Minimum Polished Rod Load . . . . 5,690 lb
(3) Stress Range . . . . . . . . . . . . . . . . 6,820 lb
(4) Rod Stress . . . . . . . . . . . . . . . . . 28,400 psi
(5) Counterbalance . . . . . . . . . . . . . . . 9,100 lb
(6) Peak Torque (at 93% Mechanical Efficiency) . . . . . . . . . . . . . 123,000 in. lb

By taking advantage of the torque reduction of the uniform torque system, (without overloading the gear reducer) the front-mounted unit can frequently employ longer cranks, affording a longer stroke which permits a slower pumping rate, yet produces the same amount of fluid. Applying a longer stroke, front-mounted unit to this same problem and pumping 12 74 in. SPM with a 114,000 in. lb reducer, instead of the 14 64 in. SPM driving the conventional 160,000 in. lb unit. The new loading would be:

(1) Peak Polished Rod Load . . . . . . . 11,790 lb
(2) Minimum Polished Rod Load . . . . . 5,550 lb
(3) Stress Range . . . . . . . . . . . . . . . . 6,240 lb
(4) Rod Stress . . . . . . . . . . . . . . . . . 26,650 psi
(5) Counterbalance . . . . . . . . . . . . . . . 8,670 lb
(6) Peak Torque (at 93% Mechanical Efficiency) . . . . . . . . . . . . . . . . 87,000 in lb

In this typical example, the longer stroke, front-mounted unit has reduced the three major causes of fatigue failure: (1) the number of rod reversals; by lengthening the stroke, (2) peak polished rod load; by slower, long stroke pumping, and (3) the stress range; by lowering the peak load and spreading the polished rod work area over a longer stroke distance.

Reduced torque, maximum rod life, semi-automatic counterbalancing, decreased horsepower demand, lower operating and first costs are some of the advances made by altering the geometry of the conventional beam type pumping unit to that of a front-mounted system.