The Effectiveness of A Special Class III Lever System Applied to Sucker Rod Pumping

By J. P. BYRD

Lufkin Foundry & Machine Company

Since little change had been made in the geometry of the conventional beam-type oilfield pumping unit since its inception nearly a century before, a study was undertaken in 1956 by the Oilfield Equipment Corporation of Denver, Colorado, to review the current state of sucker rod pumping art and to explore the possibility of designing an improved pumping system.

To assure a fresh approach, the designers asked this question: "In order to lift a given amount of fluid, with a sucker rod system, from a particular depth, with the lowest (1) peak polished rod load, (2) peak torque, (3) energy requirement, (4) first cost, (5) highest efficiency, and (6) maximum reliability—what form would the surface geometry take?" Instead of beginning with the traditional Class I conventional unit geometry and assuming its functional characteristics and bottom-hole pump motion were optimum, a reverse approach was conceived.

Applying this "reverse" concept for the purpose of structuring a new and improved mechanical system that would produce these desired goals required the investigation of a large number of different pumping unit geometries—some newly conceived, others already patented.

After nearly a year of concentrated study, review, trial and error—the basic design goals, and at least their theoretical solutions, for a more effective beam pumping system could be stated as follows:

1. To reduce peak polished rod load, minimize rod failures, and lower structural load, a reduced, off-bottom polished rod acceleration was indicated. This could be achieved by a reversed conventional geometry (Class III lever), with a low pitman-to-crank ratio, and a specified direction of crank rotation. Also in many applications, the rods could be more beneficially loaded in order to transmit a greater amount of safe work to the bottom-hole pump.
2. To minimize peaks and smooth out the torque pattern for both the speed reducer and prime mover, possibly even reduce their size requirement, would necessitate several sophisticated, interrelated modifications; (a) a front-mounted unit, (b) an offset crank, (c) offset gearbox, (d) low pitman-to-crank ratio, and (e) specified direction of crank rotation.
3. To maximize net plunger travel without increasing rod loads, a faster downstroke and increased bottom-reversal dwell time would be needed. This, too, could be accomplished by offsetting the gearbox, using a single and preferred direction of crank rotation, along with a front-mounted unit and a low pitman-to-crank ratio. As an added extra, this arrangement could also increase bottom-hole pump fill time.
4. To maximize system efficiency and reduce energy costs, the pumping system should be constructed of simple mechanical components with negligible thermal losses. To assure highest prime mover efficiency, the unit must present a near-uniform torque load. Variations in torque loading would result in low prime mover efficiencies for either electric motors or internal combustion engines.

A study of these functional requirements, as well as their possible solutions, lasted many months and at least theoretically pointed to a special Class III lever system similar to the present day Mark II.

Unfortunately, further investigation showed that the crankshaft of the standard speed reducer would not be sufficiently stout to accommodate the heavy radial loads imposed by this new geometry and primarily because of this unforeseen drawback, the new design was abandoned.

Although it satisfied only a portion of the above goals, in late 1956 and early 1957 a work-ing field model was constructed called the Mark I (Fig. 1), which provided the reversed geometry characteristics so necessary to effect many of the desired improvements, but without requiring an oversized crankshaft. This was a relatively complex pumping system employing two walking beams, four pitmans, an elevated substructure, etc. The unit was put in service in the Denver Julesburg Basin of Northeast Colorado, and operated there for several years. While this more expensive and complicated machine achieved some of the required goals, it also, in turn, posed new problems—the principal one being the high cost of manufacturing a relatively complex machine.

Realizing the difficulty of the economic problems confronting the Mark I, after a renewed study it became apparent that with but one simple modification—i.e., "heavying-up" the unit's crankshaft—all of the desired features of the initial design could be accomplished, plus several additional ones.

Employing this new geometry, which satisfied the goals set for it, a prototype model called the Mark II (Fig. 2) was designed and put into service in the Plum Bush Creek Field of Northeast Colorado. This unit was studied over some 12 or 13 different wells throughout most of 1957.

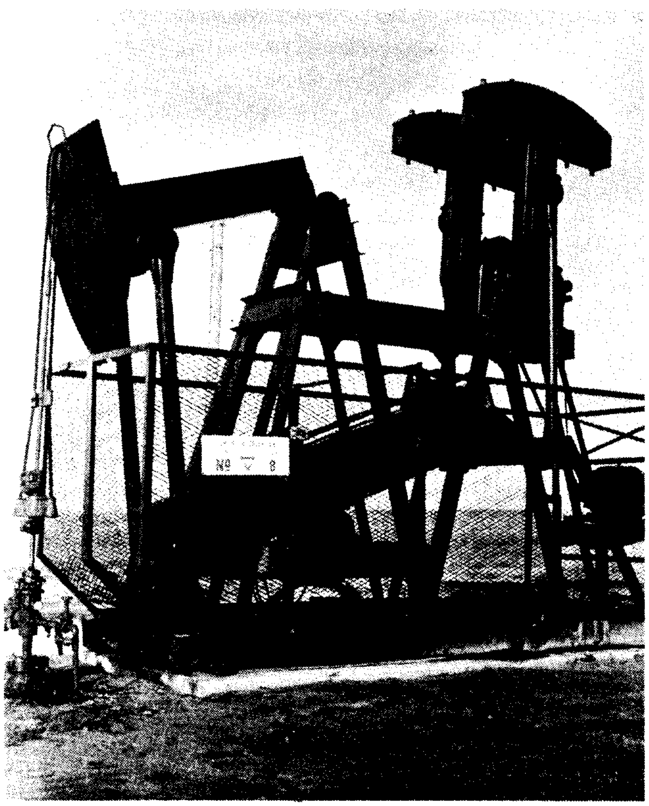


FIGURE 1

In order to further confirm the functional advances of the machine, a Mark II of variable geometry was constructed and field-tested over a number of wells under a wide variety of conditions in late 1957 and early 1958. Using this variable geometry unit, it was possible to vary much of the kinematic output of the unit to substantiate its functional improvements.

Shortly after this widespread field testing, both a graphical and mathematical method were developed to verify the torsional reduction (UNITORQUE) of the unit. The results of the field studies, and the mathematical and graphical confirmation, encouraged several companies to test the Mark II on a head-to-head basis, opposite similar-size conventional units. The results of these tests were favorable enough to warrant production of a number of additional Mark II units. During the next several years, a number of oil company studies were made of the Mark II pumping unit, and much of the information developed by these studies has been disclosed; some of it will be discussed later in the paper.

In 1961, Lufkin acquired the patent rights to the manufacture and sale of the Mark II, the machine was redesigned, and unit production was moved from Denver, Colorado, to Lufkin, Texas.

Even with widespread testing by a number of oil companies, often with encouraging field test results, many operators did not feel that the Mark II's geometric and kinematic arrangement could produce a significantly improved beam pumping system.

A long-awaited breakthrough occurred in 1963, when H. E. Gray of Shell Research and Development, presented a paper which utilized the newly developed and powerful Shell Diagnostic Technique to evaluate several different types of pumping unit geometries, including the Mark II, throughout the entire spectrum of pumping. In this paper entitled, "Kinematics of Oil Well Pumping Units", presented at the Spring Meeting of the A.P.I. in Amarillo, Texas, March 27-29, 1963 Gray stated:

> "It can be noted (from Table 4) that among several types of pumping units, peak polished rod load is found to vary

by 1,900 lbs. (10.1 per cent); the minimum polished rod load by 3,800 lbs. (54.5 per cent); plunger stroke by .65 ft. (10.3 per cent); peak well load torque by 213,000 in.lbs. (39.9 per cent); and minimum well load torque by 95,000 in.lbs. (26.2 per cent)."

Gray continues:

"THIS EXAMPLE, SELECTED AT RANDOM, ILLUSTRATES THE POINT THAT PUMPING UNIT GEOMETRY CAN HAVE SIGNIFICANT EFFECTS ON PUMPING SYSTEM PERFORMANCE."

He concludes his paper with the following statements:

"Pumping units can be classified into a few broad categories within which there is little kinematic variation from unit to unit."

And finally:

"PUMPING UNIT DESIGN EXERTS A SIGNIFICANT INFLUENCE ON THE POLISHED ROD LOADS, PLUNGER STROKE, AND TORQUES WHICH ARE OBTAINED DURING THE OPERATION OF THE SUCKER ROD PUMPING SYSTEM."

It should be emphasized that the illustration above was selected by the author as typical, and yet the magnitudes of the variation in plunger stroke and loading, both structural and torsional, were significant enough to confirm the fact that proper arrangement of the components could produce a superior beam pumping system. Here at last was unquestioned confirmation that the traditional beam pumping system could be significantly improved.

Following is a discussion of the four principal areas in which this new Mark II geometry sought to upgrade the sucker rod pumping system.

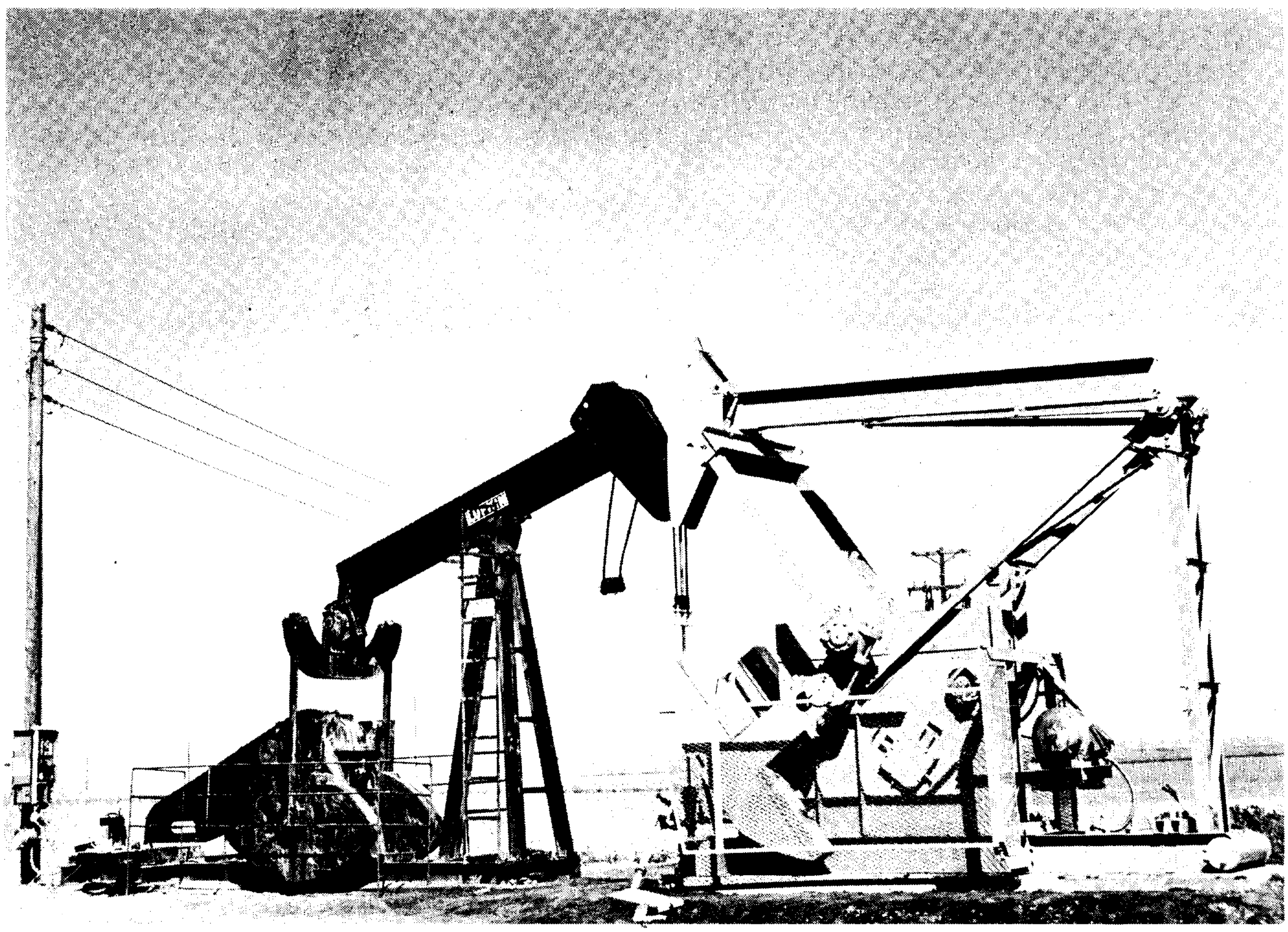


FIGURE 2

ROD AND STRUCTURAL LOADING

Admittedly, rod and structural loads, because of the elastic nature of the rod string, the continual and ever-changing harmonic stress waves, and the non-uniform polished rod motion, are complex in nature and cannot always be accurately described or predicted by simple mathematical formulas. Despite this complexity, by far the most important load considerations are rod and fluid masses and the maximum accelerations to which they are subjected. Review of these fundamental mechanical principles helps illuminate the chief factors which control rod and structural loading.

Since the polished rod load (or force) equals the product of mass times acceleration, the peak polished rod load normally occurs when the maximum mass (of rods and fluid) is elevated with maximum acceleration. In beam pumping, mass occurs only during the upstroke (i.e., lifting rods plus fluid), and since maximum acceleration during the upstroke takes place as the rods start off bottom, this then, is the critical area wherein the product of these two maximum quantities generally results in a maximum force—i.e., peak polished rod load. Even though rod string masses are elastic and there is a time lag between the movement of the polished rod and the accompanying response of the rod and fluid mass, the peak force required to lift rods and fluid is still dependent upon off-bottom (maximum) acceleration. In general, the lower this acceleration, the smaller the force required to accelerate the mass, and the lower the rod stresses and the unit's structural load*.

*If a mass is reciprocated up and down along a vertical line, the rate of the bottom reversal determines its maximum load, while the rate of the top reversal determines its minimum load. This is because, as a mass is accelerated away from the direction of the pull of gravity, its inertial component is additive to its static weight—while accelerating a mass in the direction of the pull of gravity, the inertial component of the mass is subtractive from its static weight.

The complex elastic nature of the rod string occasionally makes exceptions to this simple analogy but, by and large, a majority of all pumping applications conform to this intuitive analysis.

This fact has long been recognized by students of pumping unit technology; for instance, in his text, "Principles of Oil Well Production," Professor T. E. W. Nind of the University of Saskatchewan, states:

> "In the conventional type unit, the acceleration at the bottom of the stroke is somewhat greater than true simple harmonic acceleration; whereas, it is less at the top of the stroke. Herein lies one of the major drawbacks of the conventional unit, namely, that at the bottom of the stroke, just at the time the traveling valve is closing and the fluid load is being transferred to the rods, the acceleration force on the rod is at its maximum. These two factors combined to create a maximum stress on the rod system that is one of the limiting factors in installation design, as will be seen below."

Recognition of the high off-bottom acceleration rate of the conventional lever system prompted the designers of the Mark II to reverse this Class I geometry, replacing it with a Class III lever system which significantly reduces off-bottom acceleration! (Fig. 3).

Because of its Class III geometry*, and a low pitman-to-crank ratio, the Mark II lifts rod and fluid off-bottom with only approximately 60 per cent of the acceleration of the Class I, or conventional system.

*Although the Mark II and the air-balanced unit are both Class III (pushup) systems, the Mark II has a substantially lower pitman-to-crank ratio, i.e., about 4:1 vs. 7:1. The lower the pitman-to-crank ratio on a Class III lever system, the lower the off-bottom acceleration. Because of this fact, the Mark II makes its bottom polished rod reversal nearly as much slower than an air-balanced unit, as the air-balanced unit makes its bottom reversal slower than a conventional unit.

It was hoped, and experience has shown that the Mark II's lower off-bottom acceleration generally reduces its peak polished rod load, rod stress, and structural loading in most applications—often as much as 7 to 10 per cent.

Of the numerous Mark II field studies made by oil companies and operators, comparing it to other beam type units, in all cases except one, the Mark II has generated lower peak polished rod loads and lower structural loads.

Following are some typical examples from the field studies made. One of these studies compares peak polished rod load between different beam pumping geometries at around 1650 ft; another at around 3100 ft; while a third compares the peak polished rod loads between the Mark II and conventional unit on a 5000-ft well. Each of the three studies is made by a different oil company on a head-to-head, or turn-about basis, the units pumping under as near identical conditions, such as speed, stroke length, pump submergence, etc., as the wells would permit.

Field Study No. 1 (Continental)

A head-to-head study made over the same well between a conventional unit and an equivalent size Mark II, was made by the Continental Oil Company in the Denver Julesburg Basin several years ago. The pumping speed for both units was 16.5 SPM; stroke length approximately 54 in.; 3/4-in. rods; 2-in. tubing; 1-1/2 in. pump; and the same 25 HP motor; over a 5000 ft well. Results of this carefully controlled field study showed that,

> "The peak polished rod load with the Mark II unit was 10,400 lbs.; the peak polished rod load with the conventional unit was 11,500 lbs.—this represents a reduction in peak polished rod loading of 9.6 per cent in favor of the Mark II pumper".

Field Study No. 2 (Major Oil Co.)

This was another turn-about comparison made some time ago by a major oil company in Central Wyoming. In this study the operator made eight different runs under the same operating conditions, to be certain the system was completely stabilized. The units were operating

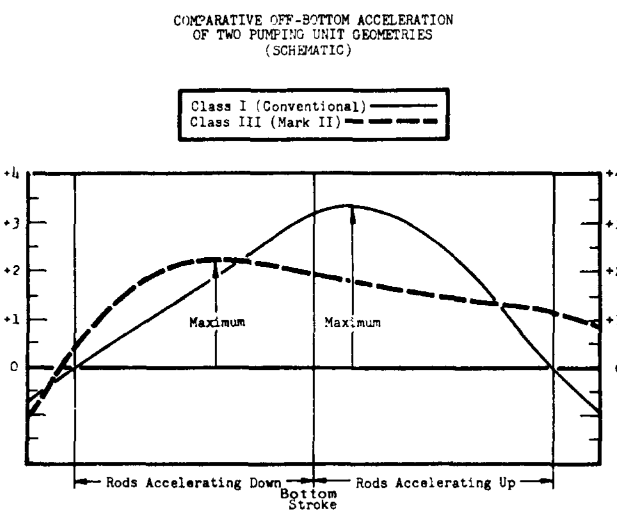


FIGURE 3

eighteen 86-in. SPM over a 1650-ft well with 2-3/4 in. pump, and 3/4-in. rods. Four of these trial runs were made on the Mark II, and four on an equivalent-size conventional unit. Conventional unit peak polished rod load measured 9713 lb, and under identical operating conditions the Mark II peak polished rod load ran 9241 lb. The difference in peak polished rod load showed the Mark II to have a 5.1 per cent lower load than that of the conventional unit.

Field Study No. 3 (Kewanee)

This comparison involved another head-to-head test made over the same well by the Kewanee Oil Company near Haysville, Kansas. The well depth was 3100 ft with 7/8-in. rods, 2-in. plunger, and a pumping speed of nineteen 74-in. SPM.

On an identical turn-about comparison, the Mark II peak polished rod load was 10,410 lb, the peak polished rod load of the comparable conventional unit was 11,280 lb. This reduction of 870 lb in favor of the Mark II represented a 6.6 per cent reduction in peak polished rod load.

Of equal importance with specific field tests is the comparison between these two geometries throughout the entire range of pumping, using the latest computerized diagnostic techniques. One operator stated:

> "Analysis shows that the Mark II design produces lower peak loads at low to intermediate speeds, with low fluid loads, and also at high speeds with heavy fluid loads. A higher peak load occurs only at high speeds with low fluid loads".

A study made several years ago by one of the research institutes showed that by dividing the entire pumping spectrum into 25 coordinate points, the conventional and Mark II peak polished rod loads were about even at 2 of the 25 points; at approximately 80 per cent of the remainder, the Mark II had a lower peak polished rod load, up to about 10 per cent, under identical pumping conditions.

Typical of the conclusions is the following major oil company evaluation:

> "The Mark II pumping unit has a lower peak polished rod load over a wide

> range of likely field conditions. The load is higher on the Mark II only at very high speeds, with low fluid loads—conditions seldom encountered in practice."

Obviously, regarding rod and structural loading, the areas of the pumping spectrum wherein the operator benefits most, would be on deep wells where rod loads are heavy or high-volume wells where the inertial component of the mass is high, and the Mark II's slower off-bottom acceleration would significantly reduce rod and structural loads.

Another factor of importance is a statement from a recent major oil company field comparison of conventional, air balance, and Mark II units operating over some 8500-ft wells in Montana. It states:

> "As pump displacement is increased, maximum rod stress and load range increase less with the Mark II than with conventional or air balance units."

Though the complexity of elastic rod systems makes generalizations risky, in a majority of all pumping unit applications, the low off-bottom acceleration of the Mark II results in substantial reduction in structural loading and peak polished rod loads which, in turn, permits the surface unit to transmit more safe work to the bottom-hole pump. Normally, reduced rod loads result in lower rod maintenance costs, longer rod life, and reduced production losses.

IMPROVED TORQUE CHARACTERISTICS

One of the more important goals of any beam pumping system is the conversion of the widely varying polished rod force loads* into a smooth and uniform torque load at the speed reducer and prime mover. To the casual observer, this task might not seem mechanically feasible.

*Throughout this discussion it will be convenient to think of the well load in terms of its load displacement curve—i.e., dynamometer card or dynagraph. Also, constant angular velocity is assumed at the crankshaft in all cases.

However, by arranging the simple tried and proven conventional pumping unit's components as a Class III lever system with the following listed kinematic and geometric modifications, it is possible, in a majority of all pumping applications, to convert the irregular differential load at the polished rod into a relatively smooth torque load at the crankshaft.

The kinematic modifications for creating a uniform torque system are as follows:

1. Change from Class I (conventional unit) to a Class III lever system (front-mounted unit), with a low pitman-to-crank ratio.
2. The speed reducer offset away from the wellhead, resulting in:
   a. A long upstroke crank cycle of more than 180°
   b. A short downstroke crank cycle of less than 180°
   c. A low maximum upstroke torque factor
   d. A high maximum downstroke torque factor
3. A single and proper direction of rotation
4. An angular offset in the crank, which provides a properly phased counterbalance.

It is important to note that these modifications are completely interrelated, and to produce a relatively uniform torque system requires the careful harmonizing of all of them. Simply applying a portion of the above modifications will not produce the most effective uniform torque system.

In most pumping applications, this type of uniform torque system provides the following:

1. A smaller maximum upstroke torque factor reduces peak mid-upstroke well load torque, which in turn reduces net torque in this area.
2. The crank and speed reducer offset, and a preferred direction of rotation permits the unit to work over the top and bottom of the stroke at about the same rate as on the side loads, by storing up potential energy in the counterweight system.
3. The increased maximum downstroke torque factor reduces net torque in this area by increasing peak downstroke well load torque.

By equalizing the net torque load in these four areas, rather than the traditional two areas of the conventional and air balance units, the Mark II in many applications provides a relatively smooth uniform torque system. This UNITORQUE system tends to provide longer transmission, bearing, and gear life, since the work load is spread relatively uniformly around the entire crank circle, rather than peaking at two points.

In many cases, the amount of torsional reduction realized with a uniform torque system permits the use of a size smaller speed reducer. Often, when the speed reducer size requirement is lowered, the prime mover size can also be reduced, even though the same work load is performed at the polished rod.

Depending upon the general shape and slope of the dynamometer card, this UNITORQUE system can reduce the torsional load as much as 50 per cent—and even more.

Following are some typical examples which help illuminate this particular characteristic of the uniform torque system.

Field Study No. 1 (Continental)

This is the same head-to-head study mentioned earlier, made between Mark II and the conventional unit, pumping alternately over the same 5000-ft well at sixteen and one-half 54-in. SPM. The report concludes:

> "The peak torque generated at the crankshaft at the Mark II unit was 65,000 in.lbs. The peak torque of the conventional unit was 110,000 in.lbs. This represents a reduction in peak torque of 40.9 per cent in favor of the Mark II unit."

Field Study No. 2 (Major Oil Co.)

This same turn-about comparison was discussed earlier, and was made by a major oil company in Central Wyoming:

> "The peak torque of the Mark II unit was 114,520 in.lbs., whereas the peak torque for the conventional unit was 161,900 in.lbs. . . . . Although each unit handled essentially the same daily production, the Mark II unit was favored by 29.3 per cent reduction in peak torque compared to the conventional unit . . . . A one size smaller gearbox could have been used with the Mark II geometry to do the necessary work."

Field Study No. 3 (Kewanee)

The following head-to-head test was made by the Kewanee Oil Company near Haysville, Kansas, and was discussed in an earlier section. On an identical turn-about comparison, the AIME paper discussing this field test states:

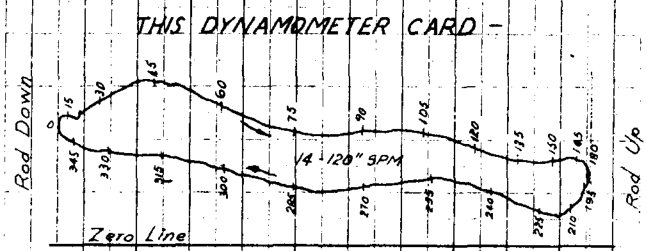
> "At 14 SPM, the peak torque of the conventional unit was 97,900 in.lbs., the peak torque of the front mounted unit was 55,500 in.lbs. (both in-balance). This represents a reduction in peak torque of 42 per cent in favor of the front mounted unit."
>
> "At 19 SPM, the peak torque of the conventional unit was 133,350 in.lbs., the peak torque of the front mounted unit was 74,410 in.lbs. This represents a reduction of 44 per cent in favor of the front mounted unit (both machines were in correct counterbalance)."
>
> "From this study, it should be evident that the front mounted unit imposes less peak torque on the gearbox than the conventional unit. In these particular applications, the reduction was 42 and 44 per cent. It is not contemplated that a reduction of this magnitude will be realized under all pumping conditions, but on many applications a size smaller gear reducer on a front mounted unit will handle the same well load as the conventional unit. This should be taken into consideration, and where possible our company should take advanatge of the economy in this uniform torque principle."

Field Study No. 4 (Canada)

Recently, Mobil Oil Company of Canada weighed its South Success Well No. 14-26-16-16 with the attached dynagraph resulting (Fig. 4). The unit was a C-228D-143-120 (conventional) unit, pumping from 3103 ft with a 2-1/4 in. plunger, a standard 7/8 in.-3/4 in. tapered rod string, pumping fourteen 120-in. SPM. The net in-balance torque developed by the regular API method measured 305,113 in.lb. This amounts to a 34 per cent overload on a standard 228,000 in.lb. conventional unit.

TORQUE COMPARISON - CONVENTIONAL VS. MARK II

THIS DYNAMOMETER CARD -



TAKEN ON A CONVENTIONAL UNIT *(C-228 D-143-120) OPERATING 14-120" SPM WAS RUN THROUGH TWO SETS OF TORQUE FACTORS -

CONVENTIONAL UNIT (C-228D-143-120)

CRANK POSITION	PRL	NET TORQUE
0	9195	- 54,626
15	9271	+ 23,246
30	10,411	+ 155,454
45	11,855	+ 305,113
60	10,031	+ 193,161
75	8,283	+ 20,303
90	8,359	- 37,815
105	8,207	- 88,561
120	6,991	- 154,200
135	6,231	- 159,004
150	6,155	- 113,421
165	6,231	- 51,765
180	5,471	+ 9,406
195	4,939	+ 129,362
210	2,811	+ 232,718
225	2,659	+ 305,113
240	3,951	+ 285,731
255	4,863	+ 243,359
270	4,407	+ 265,986
285	4,331	+ 249,174
300	5,547	+ 133,795
315	6,059	+ 42,671
330	6,839	+ 1,036
345	7,623	- 22,249
360	9,195	- 5,691

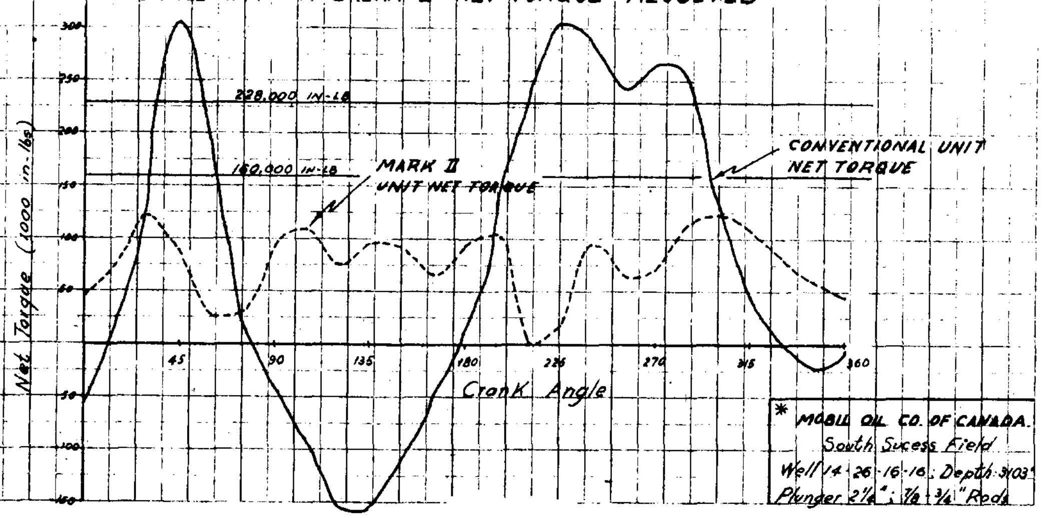
cw Rotation From 12 O'Clock
Max. Torque Up = 305,113 (45°)
Max. Torque Dwn = 305,113 (225°)

MARK II UNIT (M-160D-143-120)

CRANK POSITION	PRL	NET TORQUE
0	9727	+ 44,654
15	11,019	+ 73,868
30	11,855	+ 121,950
45	10,487	+ 88,330
60	8,663	+ 26,043
75	7,979	+ 28,621
90	8,283	+ 94,953
105	7,751	+ 109,654
120	6,307	+ 73,851
135	6,231	+ 96,348
150	6,155	+ 93,196
165	4,939	+ 66,462
180	2,887	+ 93,306
195	2,811	+ 104,077
210	4,559	- 140
225	4,483	+ 17,412
240	4,027	+ 95,673
255	5,319	+ 66,479
270	6,459	+ 68,161
285	6,687	+ 112,647
300	7,143	+ 122,627
315	7,827	+ 107,603
330	8,663	+ 83,304
345	9,347	+ 61,546
360	9,727	+ 44,654

C'Bal Torque Req'd = + 609,597
Max. Torque Up = + 121,950
Max. Torque Dwn = + 122,627

THE FOLLOWING IN-BALANCE NET TORQUE RESULTED



* MOBIL OIL CO. OF CANADA.
South Sucess Field
Well 14-26-16-16; Depth 3103'
Plunger 2¼"; 7/8 - 3/4" Rods

FIGURE 4

When this same dynagraph load was run through the torque factors of an M-160D-143-120 (Mark II) unit (i.e., one size smaller speed reducer), the in-balance net torque measured 122,289 in.lb., or only 76 per cent of its full load capacity (160,000 in.lb.). The measured torque here was also developed by the same API torque factor method.

In this particular case it can be seen that by selecting the proper geometry for a particular load, peak torque can be reduced from 305,113 in.lb. to 122,289 in.lb.—assuming the same stroke length, strokes per minute, polished rod and torsional work. In other words, by selecting the proper geometry to handle this particular type of load, the torsional picture changed from a 34 per cent overload on a 228 conventional unit, to a 24 per cent underload on a 160 Mark II unit, one API gearbox size smaller. This particular study has been confirmed by a torsional work analysis. This means that the net torsional work area beneath both curves is exactly the same.

It should be emphasized that this is not an average torsional comparison, and reductions of this magnitude cannot be expected in a majority of pumping unit applications.

Perhaps the Mark II torsional characteristics can best be summed up by a paragraph from a comprehensive major study made by an oil company using the latest diagnostic technique, which covered the entire pumping range:

> "The Mark II has a lower peak torque under all conditions investigated. This reduction is sufficient in many cases to allow use of the next smaller API size gearbox. In addition, the Mark II yields a lower torque range even though the polished rod horsepower is unchanged, the prime mover requirements may be lowered because of higher load ratings on the equipment."

Smoothing out the torque load without reducing system efficiency is one of the more desirable aspects of the Mark II system, for in many cases it enables the operator to drop one API transmission size which often permits use of a size smaller prime mover. Because of this torque-smoothing effect, both prime mover and speed reducer can operate closer to their rated capacity, and generally with significantly higher efficiencies.

BOTTOM-HOLE PUMP DISPLACEMENT AND PRODUCTIVITY

Sometime after the advent of the first air-balanced pumping units in the early 1930's, several operators noted that increased productivity seemed to result from this new front-mounted geometry, even though the pumping speed and stroke length were the same as that of similar conventional pumping machinery.

This peculiar pumping unit behavior remained unstudied and a partial mystery for a number of years, until the early 1950's. At that time William G. Corey presented a paper at the Petroleum Engineers Conference, American Society of Mechanical Engineers, Tulsa, Oklahoma, giving a simple graphical and mathematical proof which showed how the Class III (push-up) or front-mounted pumping unit, often generated more overtravel at its bottom-hole pump and, hence, obtained a greater net plunger travel and productivity. In a section of his paper entitled, "Effect on Pump Travel from the Second Harmonic", Corey explained how the increased net plunger travel is generated ONLY in a Class III lever system. Corey concludes this discussion:

> "The distorted sine wave produced by the crank cycle when the necessarily short pitmans are used, introduces a second harmonic to the rod string impulses; in the case of the front mounted unit, it serves to aid the overtravel without adding to the rod stress that might be produced."

With the introduction of the front-mounted Mark II pumping unit in the mid-1950's, this mechanical characteristic of greater net plunger travel (for the same surface pumping speed and stroke length) was again predicted for the Class III lever system. In a paper entitled, "Pumping Unit Geometry and Its Effect on Torsional and Structural Loading", presented by James C. Wright before the Canadian Institute of Mining and Metallurgical Engineers and the Society of Petroleum Engineers of AIME, Calgary, Canada, May 4, 1960, the following statement was made:

> "The energy relationship for a moving mass is as follows:
>
> $$E = 1/2\ MV^2$$
>
> Thus, if the maximum downward velocity of a Class I (conventional) system

is proportional to 4.7 units per second and the maximum downward velocity of the comparable Class III (Mark II) system is 5.6 units per second (both units turning with the same speed and stroke length), their relative kinetic (rod string) energies are as 22 to 31. This additional energy developed by the unique geometry of the Class III system is given up at the bottom reversal of the rod string. It is believed that this additional kinetic energy is transformed into greater (plunger) overtravel."

Not only is increased kinetic energy of the falling rods a factor in creating additional net plunger travel but also of much importance is the increased dwell time at the bottom of the stroke.* Increased bottom reversal time is a unique characteristic of the Class III lever system—the lower the pitman-to-crank ratio, the longer the dwell time. All else equal, the longer the bottom reversal time interval, the greater the outreach and overtravel of the bottom-hole pump.

*In general, the rod string with the most kinetic energy—having to give up this energy on reversing—would tend to have greater bottom overtravel if the rods were allowed to delay longer at the bottom of the stroke. Going through the bottom 20 per cent of its rod reversal, Mark II cranks turn through an arc of 133° (267° to 40°), while the conventional system, going through the bottom 20 per cent of its rod reversal travels through 95° (130° to 225°) of crank rotation. This means that not only is the increased kinetic energy of the Mark II some 50 per cent greater, but the delay time across the bottom of the stroke is increased by nearly 30 per cent—permitting the rods to reach out to a greater length, thus tending to maximize overtravel, with resulting increased production for a given pump cycle.

As previously mentioned, the Continental Oil Company in northeast Colorado (Field Study No. 1), ran a head-to-head field comparison between a Mark II unit and a conventional unit, pumping alternately over the same well with the same pumping speed and approximate stroke length. The results of this field test were published in an issue of the Oil & Gas Journal in 1960, and showed that under similar pumping conditions, Mark II produced more fluid.

Following is a paragraph from the summary of Continental's results:

> "It was of interest to note that the Mark II unit pumped more efficiently than the conventional unit . . . The Mark II produced an average of 4.5 barrels of additional fluid per day, which was an increase of 3.2 per cent*, over the conventional unit. The explanation for the increase apparently lies in the difference in pumping motion between the two units. The Mark II, which completes its downstroke in 165°, has a noticeably faster downstroke than upstroke. The combination of the greater velocity on the downstroke and the slower velocity change across the bottom reversal apparently causes more plunger overtravel. This, of course, would create more net effective plunger stroke."

*Using the advanced diagnostic technique, the predicted increase in net plunger travel for the Class III lever system at this point ($N/N_O = .317$; $W_f/SK = .333$) is approximately 3 per cent. The measured value was 3.2 per cent.

Because of the unstable nature of the average well and the large number of variables involved, a survey of the entire pumping spectrum using the advanced diagnostic technique to determine net plunger travel may be an even better method of comparison than field tests. Following are some examples from different studies that considered the net plunger travel throughout the pumping spectrum.

As noted, Gray showed that the front-mounted unit generates approximately 10 per cent greater net plunger excursion than the regular conventional unit in many modes of pumping.

Some years later, in a major oil company study using the most advanced computerized technique, the following was made:

> "The limits of the pump stroke for the Mark II and conventional units are determined in which an appreciable amount of difference in (plunger) stroke exists. The Mark II gives greater total stroke because of greater overtravel at the bottom of the stroke."

Looking once more at the entire pumping spectrum, a major oil company's conclusions were:

> "Differences in pump stroke up to 10% exist between the Mark II and conven-

tional designs. The Mark II produces a longer stroke, its speed and fluid load conditions most common in field use. Conventional design produces a longer stroke only at very high speeds which are seldom reached in practice."

These and other similar studies have shown the Mark II (Class III lever system) geometry often tends to maximize net plunger travel over a wide range of likely pumping conditions.

A second important item relating to increased productivity, whenever additional fluid is available, and the pump is not completely filled each stroke, is pump fill-time. Since the

TABLE 1

COMPARISON OF NET PLUNGER DISPLACEMENT CONVENTIONAL VS. MARK II

(Expressed as percentage of net plunger travel.)

A (+) percentage indicates greater Mark II plunger stroke.

A (−) percentage indicates greater conventional plunger stroke.

$\frac{N}{N_o}$ \ $\frac{W_f}{SK}$	.10	.15	.20	.25	.30	.35	.40	.45	.50
.10	0	0	0	0	0	0	0	0	0
.15	−.5	+.5	+1.0	+1.0	+1.0	+1.0	+1.0	+.5	+.5
.20	−1.0	+.5	+2.0	+2.0	+2.0	+1.5	+1.5	+1.0	+1.0
.25	−1.5	+.5	+3.0	+2.5	+2.0	+2.5	+2.5	+3.0	+3.0
.30	−2.0	+1.0	+4.0	+3.0	+2.0	+3.0	+4.0	+5.0	+6.0
.35	+4.5	+5.5	+7.0	+6.0	+5.0	+4.0	+3.5	+3.0	+2.0
.40	+11.0	+10.5	+10.0	+9.5	+9.0	+6.0	+4.5	+2.0	0
.45	+13.0	+11.0	+9.0	+7.0	+5.0	+4.0	+3.0	+2.0	+1.0
.50	+17.0	+12.5	+8.0	+4.5	+1.0	+2.0	+2.5	+3.0	+4.0

TABLE 2

COMPARISON OF TOTAL PRODUCTIVE CAPACITY; NET PLUNGER TRAVEL PLUS FILL TIME—CONVENTIONAL VS. MARK II

(Referenced to conventional unit productivity)

A (+) indicates additional Mark II productivity (%)

$\frac{N}{N_o}$ \ $\frac{W_f}{SK}$	.10	.15	.20	.25	.30	.35	.40	.45	.50
.10	+8.5	+8.5	+8.5	+8.5	+8.5	+8.5	+8.5	+8.5	+8.5
.15	+8.0	+9.0	+9.5	+9.5	+9.5	+9.5	+9.5	+9.0	+9.0
.20	+7.5	+9.0	+10.5	+10.5	+10.5	+10.0	+10.0	+9.5	+9.5
.25	+7.0	+9.0	+11.5	+11.0	+10.5	+11.0	+11.0	+11.5	+11.5
.30	+6.5	+9.5	+12.5	+11.5	+10.5	+11.5	+12.5	+13.5	+14.5
.35	+13.0	+14.0	+15.5	+14.5	+13.5	+12.5	+12.0	+11.5	+10.5
.40	+19.5	+19.0	+18.5	+17.5	+17.5	+14.5	+13.0	+10.5	+8.5
.45	+21.5	+19.5	+17.5	+15.5	+13.5	+12.5	+11.5	+10.5	+9.5
.50	+25.5	+21.0	+16.5	+13.0	+9.5	+10.5	+11.0	+11.5	+12.5

bottom-hole pump is filled only during the upstroke, the longer the upstroke time interval, the greater the fluid production, all else equal. Since the upstroke time interval of the non-symmetrical Mark II is approximately 8.5 per cent greater than that of a comparable symmetrical conventional or air-balanced unit, further productive capacity often results, in addition to that obtained from increased plunger travel.

Table 1 shows the increased productive capacity of the Mark II pumping unit (in percentage) resulting from predicted increased net plunger travel throughout the entire spectrum of pumping. Table 2 shows increased productive capacity resulting from BOTH additional net plunger travel and increased fill-time. These tables were derived from computerized curves showing relative pump strokes throughout the entire pumping spectrum.

SYSTEM EFFICIENCY AND POWER SAVINGS

One of the chief advantages of the rotary counterbalance beam pumper is its high efficiency. Unlike hydraulic, pneumatic, and electrical pumps, the mechanical components of the conventional and Mark II units have negligible thermal losses, with counterweight systems incapable of dissipating energy. A foot-pound of energy applied at the high-speed shaft of the conventional or Mark II unit, is delivered almost intact as useful reciprocating work at the horsehead.

Perhaps even more important than high mechanical efficiency, is the Mark II's uniform torque system as it applies to the prime mover.

An electric motor prime mover is reasonably efficient when operating at its rated capacity and driving a relatively uniform torque load; however, its efficiency falls off rapidly under variable loading, such as the driving of a conventional or air-balanced pumping unit.

The electric motor's capacity for performing work is dependent upon the amount of heat it can dissipate in a given time interval. This heat, chiefly resulting from driving a widely varying cyclic load—such as that of an oilfield pumping application—is a function of the square of the current, (I^2R). With a smoother and more uniform torsional load (resulting in a proportionately lower and more even current), a smaller amount of heat will be generated for any given polished rod work load, thereby allowing the motor to be applied more efficiently. Under this more uniform loading, more of the electrical energy is converted into useful rotating work for driving the pumping unit, rather than being dissipated as heat loss.

The power factor, and hence the efficiency of an electric motor decreases as the load drops off—as it must do twice each conventional or air-balanced crank cycle. For example, a 20 HP NEMA D oilfield motor has a full load power factor of 90 per cent; but at half load it drops to 77 per cent and at one-quarter load, it is further reduced to 55 per cent. Thus, a widely varying cyclic load sharply reduces the motor's effectiveness and increases the amount of heat dissipated, as well as raising its energy requirements to perform a given amount of polished rod work.

Following are several studies comparing the power cost saving characteristics of the uniform torque system to those of other beam type pumping units.

Field Study No. 5 (Forest Oil)

In a six months field study of some 15 conventional and Mark II pumping units in a field in Wyoming, the Forest Oil Company found that power requirements could be reduced by some 30 to 35 per cent when using a uniform torque system.

> "For an average pumping well in the field, from the same depth, under the same conditions, a Mark II unit will pump 58 BPD for $26.40 per month less than a conventional unit—a savings of 32.2 per cent in energy charges. Using similar calculations, the savings on energy charges at 35 BPD would amount to $25.55 or 33.3 per cent, and at 110 BPD, $38.50 or 39.5 per cent would be saved by using the front mounted unit."

These are cost figures from rigorous field studies, and this paper was published in the Journal of Petroleum Technology.

Field Study No. 3 (Kewanee)

In a head-to-head field test made sometime ago by the Kewanee Oil Company and disclosed in the SPE-AIME paper ("Field Testing a Front Mounted Mechanical Oilfield Pumping Unit"), Kewanee found a reduction in electrical consumption of about 10 per cent and a reduction in demand charges of approximately the same value. This dual electrical savings combined to reduce the monthly power bills about 17 to 18 per cent.

Field Study No. 6 (Major Oil Co.)

In a field study made several years ago by a major oil company, between a conventional unit and a Mark II in Wyoming—but not pumping over the same well—the section on prime mover and surface unit efficiency was concluded as follows:

> "In addition, based on the figures above, the power bill should be 39.6 per cent lower for driving the Mark II unit than for the conventional unit."

Field Study No. 2 (Major Oil Co.)

In November 1963, a major oil company ran a head-to-head field study over the same well in Central Wyoming, between a conventional unit and a Mark II. Pumping conditions were as near identical as possible. From one of the paragraphs is the following statement:

> "From this study, it is apparent that a long range savings in power cost and motor repair may be realized."

Field Study No. 7 (Major Oil Co.)

Following is a passage from a report made by a major oil company comparing conventional unit and a Mark II in South Central Texas:

> "Also, due to its more uniform torque demand, the electric power consumption has been reduced by 32 per cent over the conventional unit previously on the well."
>
> "However, the savings in electric power consumption alone should amount to more than $600 per year under present rates when using the Mark II unit."

Field Study No. 8 (Shell)

Several years ago, the Shell Oil Company evaluated the Mark II pumping unit against an equivalent size conventional unit. In the section on power savings, the report states:

> "The Mark II unit requires about a 15 per cent smaller motor than does a comparable size crank balanced unit, lifting an equal amount of fluid."

Further in the report, the following:

> "On the first five tests, 2 conventional and 3 Mark II, the motor on the Mark II required an average of 23 per cent less RMS current (17.2 amps vs. 22.2 amps) than did the same motor on the conventional unit."

The report concludes:

> "In some instances, it will be possible to use smaller motors with their attendant savings in first cost, distribution system, and energy cost."

It must be recognized that in all the above power costs studies, the Mark II is assumed to do the same polished rod work as the conventional unit. Reduction in electric power savings results primarily from one factor—Mark II in most cases presents a more uniform torque load to the prime mover, permitting it to perform with a greater efficiency and fewer thermal losses.

Interestingly enough, the internal combustion engine has characteristics that are somewhat analogous to those of the electric motor. Somewhat like an electric motor, the internal combustion engine runs more efficiently when it is driven to its rated capacity handling a relatively uniform torque load.

It is believed that an internal combustion engine driving a relatively smooth torque load and operating close to its rated capacity, tends to run more efficiently and as a result will probably require less energy or fuel to perform a given job.

Undoubtedly, if and when the uniformity of the load is great enough that the engine size can be reduced, fuel savings of a substantial magnitude can be realized.

CONCLUSIONS

Despite the mechanical "attractiveness" or "fascination" of any artificial lift system—rod or rodless—in the final analysis, its true worth lies primarily in its ability to reduce lifting costs. How successful the Mark II (Fig. 5) has been in achieving this goal is often dependent upon (1) how well the operator understands the functional goals and characteristics of the unit, and (2) whether or not the machine does in fact produce the desirable functions and greater economy claimed for it.

Generally, the functional advantages of a pumping system are hard to explain in simple terms, and their true economic worth and merit are often difficult to measure and evaluate. Frequently, the operator has neither the time nor the inclination to study and compare subtle and complex mechanical effects—all of which often make the economics of an artificial lift system involved and obscure.

Perhaps the most accurate and mechanically sound method of comparing beam type pumping systems is to relate peak in-balance torque, peak and minimum polished rod load, and energy consumed to the simultaneously measured polished rod work per stroke—on the same stable well under identical conditions of stroke length and pumping speed.

One of the chief features of the Mark II is that its kinematic output is automatically programmed to produce certain desirable functions in a particular sequence.

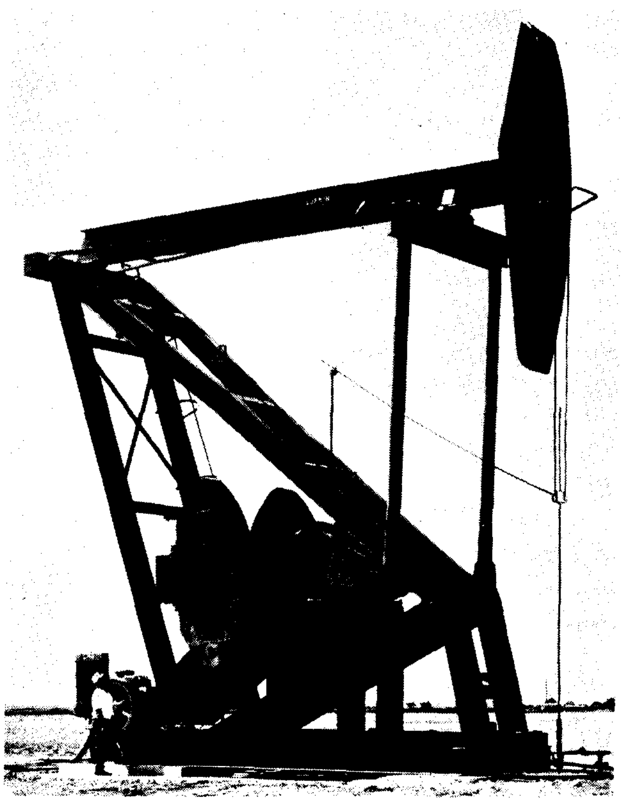


FIGURE 5

With the prime mover facing a relatively smooth torque load, it tends to drive the unit cranks with near constant angular velocity, insuring (1) that the maximum load is lifted off bottom with a reduced acceleration, often lowering rod and structural loads, (2) that the longer upstroke results, tending to increase productivity, (3) a faster top reversal, helping to create a more favorable rod load range, without appreciably limiting maximum pumping speed, (4) a faster downstroke, tending to generate greater net plunger travel and productivity without increasing rod load, and (5) generation of a relatively uniform torque system providing maximum efficiency and often requiring smaller torsional components.

Perhaps these functional characteristics are best summed up by two oil company statements. A six-week study, made in the early '60s, reported:

> "The motion, torque characteristics, semi-automatic counterbalancing, and other new features of the Mark II unit combined to make a marked advance in beam type pumping unit design . . . Our evaluation indicates that a major advance in pumping unit design has been made by the manufacturers of the Mark II pumping unit."

Recently, a second operator has stated:

> "The push-up geometry and phased counterbalance of the Mark II give it the best pumping characteristics of any unit now being manufactured. The unit geometry tends to decrease both the maximum polished rod load and the minimum polished rod load, thus, creating a better operating range with the sucker rods. This type of geometry tends to maximize the overtravel at the pump—thus, increasing the amount of production per stroke. The negative torque on the gear reducer is kept to a minimum—thus, reducing the operating costs. In many cases it is possible to use a smaller size Mark II where a larger size conventional unit would be needed. The choice of a Mark II will also allow the use of a smaller prime mover which will reduce operat-

ing costs even further. Occasionally a less expensive sucker rod string can be used due to the lessening of the well loads."

BIBLIOGRAPHY

1. Gray, H. E.: Kinematic of Oil Well Pumping Units, API—Division of Production, Amarillo, Texas, March 29, 1963.
2. Gibbs, Dr. S. G.: Predicting the Behavior of Sucker Rod Pumping Systems, SPE of AIME Paper No. 588, Denver, Colorado, May 27, 1963.
3. Nolen, K. B.: Deep High Volume Rod Pumping, SPE of AIME Paper No. 2633, Denver, Colorado, September 28, 1969.
4. What Operators Learn from Competitive Pumping Tests, Oil & Gas Journal, December 12, 1960.
5. Jackson, B. C., and Byrd, J. P.: Field Testing a Front Mounted Mechanical Oil Field Pumping Unit, SPE of AIME Paper 382, Billings, Montana, May 24, 1962.
6. Patton, L. D.: Comparative Energy Requirements of Oil Field Pumping Units—Conventional vs. Front Mounted, Journal of Petroleum Technology, SPE of AIME, January 1965.
7. Hammond, K.: Pumping Unit Comparison—Conventional Unit vs. Mark II Unit, Continental Oil Company Report January 22, 1960.
8. Nind, T. E. W.: "Principles of Oil Well Production", University of Saskatchewan, 1964.
9. Studies and Reports—Charles L. Wheatley, Texas Pacific Oil Company, Inc., Dallas, Texas.