HISTORY, BACKGROUND, AND RATIONALE OF THE MARK II, BEAM TYPE, OIL FIELD PUMPING UNIT

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ABSTRACT

The Conventional beam pumping unit, a Class I lever system, was used almost exclusively in artificial lift applications from the 1700's until the late 1920's. At that time, a "reversed" Conventional geometry design (Class III lever system), called an Air Balance unit because of its pneumatic counterbalance system, made its appearance.

Later, in the mid 1950's, a second, Class III lever system, or "reversed" geometry unit, was introduced and named the Mark II. Like the Air Balance unit, the Mark II had some performance features different from those of the traditional Conventional unit, but used similar rotating counterweights instead of the pneumatic arrangement of the Air Balance unit.

The following paper will discuss some of the unique performance concepts of the Mark II design, and the background and rationale behind their development.

I. INTRODUCTION

Any discussion of beam pumping history must inevitably begin with one of the world's oldest and most remarkable machines - the Conventional, Class I, walking beam unit. (Figure 1)

It s likely beginnings are clouded in antiquity, but some scholars think the Chinese operated a beam pump - probably for irrigation purposes - several thousand years ago. Later the Romans, and possibly other civilizations used the concept of a simple beam pump for lifting fluid.

Perhaps modern beam pumping originated in the 1700's in Scotland where Scottish mining engineers used it to drain water from flooded coal mines.

Through the ages, a number of clever variations of the beam pump have been made - but most have not withstood the test of time, and the traditional Conventional unit has continued to dominate the world of artificial fluid lift since its very beginning.

A second, popular, beam unit was developed in the 1920's called the air balanced unit because the counterbalance system consisted of a piston, cylinder, and air pump. This unit was a close-coupled, "reversed" geometry design, employing a Class III, or push-up lever system. (Figure 2)

It is believed that the designers of the Air Balance unit had two chief goals in mind, (1) to reduce foundation size requirement, and, (2) to develop a beam pump that could be more easily counterbalanced. Little, if any, thought was originally given to the idea of changing the performance characteristics of the Conventional unit.

Later it was realized, perhaps by outsiders, that use of the "reverse", or Class III lever design of the Air Balance unit, might, in some applications, also reduce peak polish rod load and increase plunger stroke and production , though it is not believed that this was the thrust or intent of the inventor of the unit.

Use of the Air Balance unit has been widespread in cases where space limitations prohibited large foundations, and where a great number of pumping units were to be operated in a small and restricted area.

II. MARK II: BACKGROUND AND HISTORY

Some time in the late 1940's, an engineer was hired by one of the pumping unit manufacturing firms to work on the field development of a new type of Air Balance pumping unit.

Later, at the conclusion of an oil company meeting in Tulsa, Oklahoma, to discuss the pros and cons of the Air Balance unit, their highly regarded Production Superintendent, Wesley Moore, surprisingly stated, "One day, some manufacturer will develop an Air Balance pumping unit without an air system, and this unit, I believe, will be superior to all other beam units now available".

Passing on this comment to his chief, the engineer was told that the idea of a straight mechanical, Class III unit was impractical, that it had been tried before and was found unsuccessful.

In the mid 1950's, this same field engineer was discussing beam pumping with William Boyd, president of Kimbark, a newly formed oil company, when the oil man asked, "When was the last time the basic design and performance of the beam unit has been changed or significantly improved?". The answer was, with the exception of the Air Balance design, the beam unit has not been substantially modified in a very long time. Boyd replied, "Isn't it time to try and develop an improved beam pumping machine?". This conversation led to the formation of a small, pumping unit manufacturing firm, The Oilfield Equipment Corporation (O.E.C.), headquartered originally in Denver, Colorado.

One of the new company's first tasks was an in-depth study of existing beam pumping systems, with a goal of evaluating as many different pumping unit patents and beam pumping linkages as possible. After several months of study, the statement made many years before by Wesley Moore was recalled - "If an Air Balance unit without an air system is built - it will be a superior beam pumping machine".

With this rather vague admonition in mind, in 1956 work began on the design of a new (reverse) Class III geometry pumping unit, "without an air system", named the Mark II, and design work proceeded for many months.

Concurrently, a second equally important benchmark was constantly being reviewed - what were the most important advantages of the Conventional and Air Balance units, and how could these features be utilized - and what were these unit's disadvantages and how could they be minimized or eliminated?

Many unique and desirable performance features were conceived and attempted in this new design, which was made up of the same, simple, rugged, tried and proven components, used for many years on the conventional unit.

Nearing completion of the new Mark II design, it suddenly occurred to the designers that the new unit's crank shaft would have to be somewhat larger than that of the (torsionally) equivalent Conventional unit, since counterbalance and pitman forces were additive, rather than subtractive, as on a Conventional, Class I system.

Since an oversized crank shaft was not available on a standard (gear-driven) oil field pumping unit transmission, the Mark II design was stopped, and some thought was given to abandoning the project.

Some time later - and starting over - a second, Class III, or reverse geometry unit, completely different from the Mark II, and much more complicated, was slowly and painfully conceived, designed, manufactured, and named the Mark I. (Figure 3)

The Mark I had several unusual features which included; (1) the desired Class III reverse geometry which would accept a standard transmission (i.e., speed reducer); (2) with a semi-automatic, electro-mechanical system providing a wide range of adjustable counterbalance effect, while the unit was in operation; (3) maintenance-free rubber bushings at several locations, significantly reducing shock load on the transmission; (4) one man stroke-change capability without need of gin pole or crane, i.e. wrist pins ran in slots, and did not require pulling; (5) and an automatic belt tensioning system, etc.

The Mark I was constructed, purchased, and operated almost trouble-free for several years by two different firms, one of which was a major oil company. Unfortunately this unit had two fatal flaws - (1) it was too complicated; (2) it was too expensive to build. Because of these drawbacks, once again, a difficult decision had to be made - (1) should a new, Mark design be considered; or (2) should the project be abandoned?

The involved, Mark I arrangement had convinced the designers that if unique and desirable performance characteristics were to be included in a new Mark design, their attainment could not be complex, exotic, or expensive, as on the Mark I unit. Any new features, however desirable, must be conceived and incorporated into the simplest, most rugged and long-lived beam unit arrangement possible.

After much soul searching, it was decided to scrap the Mark I arrangement, continue the original Mark II concept, and to drive it with a newly designed double reduction chain transmission, with an oversized crank shaft.

This was accomplished and the first Mark II was put in operation in the Denver-Julesberg Basin in the early part of 1957. (Figure 4)

The initial run of some eight to ten units - the first production run - was subcontracted to a machine shop in Denver, Colorado, and upon completion, the units were sold to the parent company, and tested over various wells for the next several years.

The first manufacturing facility of the new Mark II company (O.E.C.) was located in Fort Morgan, Colorado where the next production run of units was made.

Some time later, realizing that the separation of the Denver office from the Fort Morgan plant was awkward, a larger and much improved facility was located at the Rocky Mountain arsenal in Denver, Colorado, where manufacture of the Mark II units was continued.

During the next several years, some fifteen different oil companies were bold enough to purchase and evaluate the Mark II themselves. Generally, the results of these company tests were favorable, but significant additional Mark II sales to these oil companies was considerably less than brisk.

During the first five or six years of operation, some 75 Mark II units were built and sold and most of them were tested and evaluated on problem wells. Of these first units, a number have been in continuous service for nearly 35 years.

About this time, several major pumping unit manufacturers became interested in the Mark II and its patents, and in 1961, after considerable study and evaluation of Mark II performance and claims, the Lufkin Foundry and Machine Co. of Lufkin, Texas, acquired the rights to manufacture and market the Mark II and have continued since that time.

III. MARK II DESIGN AND PERFORMANCE RATIONALE

The comprehensive goal of the Mark II designers in the mid 1950's was to develop a simple and rugged beam pump, having a number of unique and desirable performance features made possible by simply altering existing beam unit geometry.

The main thrust of this paper is to outline these unique performance features, one by one, to explain the designers' rationale in attempting their development, and where possible, to validate their desirability and worth by later industry confirmation.

If others had tried a similar design, or approach, or had attempted to develop similar beam pump features, their efforts were unknown to the Mark II designers.

Rather than highlighting the actual field experience of the Mark II, the following discussion addresses what the inventors hoped to accomplish - and why.

It is not the intention of this paper to compare theoretical and/or field performance of different types of beam pumping units except in cases where Mark II design rationale cannot otherwise be made understandable and meaningful.

Following is a list of the Mark II designers' goals and objectives: (Figure 5)

1) To lower rod stress and structural load;
2) To make counterbalancing easier by using the unit's energy of rotation;
3) To reduce and smooth out the torque load;
4) To increase productivity;
5) To decrease power consumption;
6) To improve and reduce foundation costs and mounting - and to combine all these features with unit reliability, efficiency, and economy.

IV. LOWERING ROD AND STRUCTURAL LOADS

In the mid 50's, although strides had been made in sucker rod manufacturing technology, rod breakage was still a formidable problem. As wells became deeper and fluid loads heavier, sucker rod stress mounted proportionately.

Perhaps the first objective of the new Mark II design was, all else equal; how can rod and structural loads be reduced?

Since maximum rod and structural loading normally occurred when the maximum mass was moved upward with maximum acceleration, it seemed logical to try and reduce dynamic rod and structural load by simply reducing the magnitude of maximum, off-bottom polished rod acceleration. One answer would be to reverse the Conventional Class I, pull-down geometry into a Class III, push-up type.

Because of their crank and pitman placement, the Mark II and Air Balance on the well side of the fulcrum, and the Conventional unit on the off side, these two lever systems have polished rod acceleration characteristics that are diametrically opposite.

Its cranks turning with constant angular velocity, the Conventional unit makes its bottom polished rod reversal with relatively high acceleration, and its top reversal with relatively low acceleration.

In the Mark II and Air Balance units acceleration characteristics are reversed, the front-mounted or push-up system comes off-bottom with low acceleration but makes its top reversal slightly faster than the Conventional unit.

Normally, all else equal, reversing the Mark II geometry lowered off-bottom maximum acceleration by some 25 - 30% and sometimes by as much as 40% (Fig. 5).

In many cases, this lower off-bottom acceleration did reduce rod and structural load but not always. If the rod and fluid acted much like a concentrated mass, reduced off-bottom acceleration would always lower rod and structural load - but the complexity of an elastic rod string and its harmonic behavior sometimes refuted this intuitive fact.

The Sucker Rod Research Institute (SRI) was just beginning about this time and Mark II designers knew little of SRI's valuable, new research on rod string behavior.

The designers assumed that any rod and fluid combination would develop less rod stress and structural load if the string were lifted off-bottom slowly, rather than rapidly. This beneficial effect of lower rod and structural loading happens frequently as a result of reduced Mark II off-bottom acceleration, but not in every case.

Some years later, T.E.W. Nind, Professor of Engineering at the University of Saskatchewan, made the following (slightly paraphrased) statement:

> "In the Conventional 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 lie some of the major drawbacks of the Conventional unit, namely, that at the bottom of the stroke just as the traveling valve is closing and the fluid load is being transferred to the rods, the acceleration force on the rods is at its maximum. These two factors combined to create a maximum stress on the rod system (as well as on the unit structure) that is one of the limiting factors in installation design."

In the final analysis, knowing that force is equal to the product of mass and acceleration, and believing that force reduction in rods and structure is desirable, the rationale of the designers was to lift the maximum mass of rods plus fluid with a lower acceleration, and return the lighter mass with a higher acceleration. It wasn't helpful in every case as the designers originally had hoped - but it was significantly beneficial in a large majority of beam pumping applications.

V. EASE OF COUNTERBALANCE

Counterbalancing the average conventional pumping unit is a laborious and sometimes dangerous job. Frequently beam pumps are left improperly balanced for long periods of time. In a study conducted some years ago, it was found that nearly 25% of a random sampling of field applications averaged about 25% out-of-balance.

One of the earliest Mark II concepts was to design a beam unit where the energy of rotation, or an electric motor, could be used to reposition a portion of the counterweights, <u>while the unit was in operation</u>.

This was first accomplished on the Mark I where saddle weights were screw-driven by an electric motor up or down the under beam, as desired. (Figure 3)

Later on, the first Mark II unit, which incidentally was driven by a multi-cylinder engine, had large trim weights inside the long counterweight stem, screw-driven by a star wheel on the end of the screw. As the star wheel passed a given point, a dual spring loaded finger was moved into engagement rotating the screw a given amount, either clockwise or counterclockwise, each stroke, thereby moving the trim weight out or in for more or less counterbalance effect, while the unit was in operation. (Figure 7)

The evolution of this concept finally led to the employment of a dual, stationary, sheave and belt system around the crankshaft, driving a small, dual sheave, bevel gearbox which furnished the screwdriven motive power to reposition the trim weights. A lever system loosened or tightened the appropriate trim weight belts which drove the weights in or out depending upon the desired counterbalance effect. (Figure 8)

A number of Mark II units, equipped with the semi-automatic counterbalance which Lufkin Industries had significantly upgraded, were purchased by the industry in the early 60's.

After a total of five or six generations of improvement, the high cost of manufacturing this semi-automatic counterbalance, and several other factors, caused the company to abandon the manufacture of the device.

If it can be made cost effective, using the unit's energy to safely and effortlessly change the counterbalance effect as needed, seems as valid today as it ever was.

VI. REDUCING AND SMOOTHING OUT THE TORQUE LOAD; THE UNIFORM TORQUE SYSTEM (UNITORQUE)

Perhaps the most important goal of the Mark II designers - and the most obscure - was torsional reduction and smoothing. From the beginning of beam pumping design, it was generally assumed that, in any balanced application, the torque pattern must go from zero (or below) up to a maximum, back down to zero (or below) back to maximum and down to zero (or below), every single crank rotation. Because almost all beam pumps were of the symmetrical, Conventional, Class I arrangement, this is the way the average unit behaved - regardless of the dynamometer card shape.

In studying the graphic relationship between the well load torque, counterbalance and net torque, the counterbalance torque curve was inadvertently plotted upside down. The surprising result was a picture, or "blue print" of how a relatively uniform torque system could be approached and perhaps developed. (Figure 9)

This chance diagram showed that the upstroke must be made in more than 180° - strongly complimenting the rod and structural load design objectives. It further indicated that a proper phasing should exist between well load and counterbalance torque. It emphasized that a significant disparity between up and downstroke maximum torque factors would be necessary. The diagram further suggested a non-symmetrical Class III component arrangement with a low pitman-to-crank ratio, turning in a preferred direction of rotation.

It's difficult to believe that a misdrawn diagram of the torsional components of a beam pumping system could so clearly define the design of a relatively uniform torque system - but that's exactly what happened.

By offsetting the transmission rearward the correct distance, casting a particular offset in the crank; turning the cranks in a preferred direction of rotation; having a low pitman-to-crank ratio on a Class III or push-up geometry - a relatively uniform torque system theoretically could be created. In a wide majority of applications, peak torque could be significantly lowered, though in a few applications, torsional reduction was modest or even negligible.

Torsional reduction and smoothing, the designers found, was a combination, or "package" of the several modifications noted above - and without all of them, the Unitorque system was considerably less effective.

Before Lufkin Foundry and Machine acquired the Mark II patents - and probably doubting the Mark II torsional reduction claims - their Engineering Department made an in-depth study of the Unitorque concept - unknown to the Mark II designers. This study is graphically summarized in Figures 10a and 10b.

In it a rectangular dynamometer was selected with an arbitrary peak polished rod load of 17,400 lbs. and minimum load of 10,000 lbs. This basic well load was then accelerated, throughout 360°, first by a Conventional unit turning 15 - 74 in. strokes per minute, and later by a Mark II turning the same speed, both with constant angular velocity - momentarily disregarding rod stretch and harmonics - but accounting for inertial forces and rod and fluid weight.

The resulting Conventional unit, inertial dynamometer card (i.e., well load pattern) was then applied to its own permissible load diagram, and the balanced net torque was plotted as shown in the lower diagram, Figure 10a. This exercise was then repeated for the Mark II, (Class III) geometry, Figure 10b.

Although harmonics and rod stretch were not included in this independent study, made by a respected and leading competitor, the results were favorable enough to the Mark II to at least give reasonable confirmation to the Unitorque concept.

The torsional implications of this independent study were that overtravel dynamometer cards (major axis sloping downward to the right) favored a non-symmetrical Class III geometry -

while undertravel cards (major axis sloping upward to the right) favored the Class I or Conventional unit geometry.

It is of importance to note that the birth of the Sucker Rod Research Institute with its in-depth studies into the dynamics of rod motion and loading - and the development of the Mark II, occurred about the same time in the mid 1950's. Unfortunately, the invaluable findings of SRI were thus unknown to the Mark II designers.

However, several elementary beam pumping "rules of thumb" were known or theorized by the Mark II inventors:

(1) That in slow to medium speed pumping the less the rod stretch (momentarily disregarding inertia and harmonics) the greater the plunger travel. (Fig. 11)

(2) When both inertia and modest rod stretch were considered, there was a strong tendency toward the generation of an overtravel card. (Figure 11)

Modern, sophisticated beam pumping analysis shows that in a number of cases these "rules of thumbs" were not always correct - but in a majority of practical field applications, of the 1950's, they were assumed to be generally true.

Years later, several studies seemed to spotlight and confirm the desirability of earlier Mark II Unitorque theory relative to an overtravel dynamometer card.

Figure 12 illustrates one way where, with a constant (non-dimensional) pumping speed, overtravel cards become undertravel type loads as static fluid loads become heavier or perhaps even excessive.

Theoretically, the three overtravel cards on the left produce approximately the same amount of fluid per stroke as the three undertravel cards on the right, but with a lower peak polished rod load, (H), a more narrow load range, (R), and with greater plunger travel (PT) - all of which are desirable characteristics. Differences in work area in these cards roughly reflects differential frictional components.

Momentarily disregarding harmonics, Figure 13a shows a theoretical overtravel card (top left) with no rod stretch, resulting in maximum plunger travel, along with its accompanying in-balance torque load diagram. Increasing the rod stretch by increments, while holding the peak polished rod load, the minimum polished rod load, and work area, constant, successive, in-balance torque loads are plotted. When plunger travel is maximum, a high torque results - and when plunger travel is minimum, the lowest torque is produced.

Figure 13b illustrates the same theoretical dynamometer card and its successive stretch in increments, applied to a Class III geometry, the same as before. Here maximum plunger travel produces minimal torque, while successive increased stretch produces higher torque. Maximum torque obtains only when plunger travel is zero.

The index number shown to the right and left gives the ratio of the plunger travel to the net in-balance torque - the greater the ratio, the more effective the pumping mode. The numbers between the two studies indicate the ratio of the respective index numbers.

Admittedly, this is an early day, theoretical study of two different beam geometries pumping at slow to medium speeds - but to a significant extent, it emphasizes and confirms the desirability of an overtravel card applied to a Class III, non-symmetrical geometry like the Mark II. Spurious harmonics will alter these concepts to some degree as does excessive stretch - but in most cases, the basic idea still obtains.

Most of these concepts were known or theorized by the Mark II designers - but beam pumping technology had not yet progressed to the point where they could be confirmed under widely varying, dynamic field conditions.

The Mark II, Class III geometry was proficient at handling both overtravel and undertravel type loads - but its effectiveness was significantly increased when applied to overtravel applications.

Certain theoretical beam pumping concepts may be highly desirable - but of equal importance is, how often do these desirable or undesirable situations or circumstances actually occur in field practice?

Figure 14 shows the results of a theoretical study of some 400 dynamometer cards accepted at random from over 100 different operators or companies throughout the world. Assigning these cards to their proper non-dimensional pumping speed and fluid load parameters on the full spectrum of pumping, it can be seen that approximately 88% of the cards are essentially overtravel cards, while but approximately 12% are undertravel. In other words, only about one card in nine, statistically, is an undertravel, while approximately eight of each nine cards is overtravel.

Although only about one application in nine is of the undertravel variety, still perhaps one-half of this group could be converted to more desirable overtravel characteristics if the proper pumping mode and unit geometry had been selected.

There are two chief factors that account for the fact that approximately 12% of the dynamometer in the above spectrum are undertravel. They are:

(1.) Excessive rod stretch,
(2.) Spurious harmonics - with rod stretch being perhaps the more important of the two.

As mentioned above, although the Mark II can effectively accommodate both overtravel and undertravel types of rod loading - its preference is the overtravel profile, which in most cases it tends to generate.

This independent statistical study, made years after the original Mark II design, amplified the fact that a great proportion of existing field dynamometer cards favor Mark II geometry.

Sometime later a second independent statistical analysis of 500 random dynamometer cards was made, and though its objectives were slightly different, generally speaking, its findings closely confirmed those of the earlier study.

In the earliest days of the Mark II design, a theoretical dynamometer card was plotted using the Mark II torque factor schedule, to determine what card shape (i.e. well load pattern) would produce a constant torque load at the crankshaft. This was done and the result was a well defined overtravel card.

Thus, one of the designer's objectives was to, whenever possible, deal with an overtravel card shape, which, under normal stretch and low harmonic conditions would result frequently - regardless of the type of geometry used.

VII. INCREASING PRODUCTIVITY

Another goal of the designers was to hopefully increase Mark II productivity by three simple methods:

(A.) Increasing net plunger travel;
(B.) Extending fill-time;
(C.) Increasing the amount of safe work delivered to the bottomhole pump.

A. Maximizing plunger travel

By varying unit geometry to effect a faster downstroke and a slower bottom rod reversal rate, it is possible to significantly increase net plunger travel, and in many cases, without appreciably increasing polished rod loads.

Since the kinetic energy of a falling mass is a function of the square of its maximum velocity (K.E. = 1/2 MV^2) - and since the rod system cannot store up this kinetic energy over the bottom reversal, but must give it up as overtravel - the faster the downstroke, the greater the potential that exists for increasing overtravel.

Since the Mark II makes its upstroke in approximately 200° of crank rotation, it must make its downstroke in approximately 160°. Symmetrical Conventional, or Air Balance units make their return or downstroke in approximately 180° of crank rotation.

This means that the faster fall of the Mark II rods by only 12 1/2% tends to generate approximately 27% more kinetic energy, which is given up as overtravel at the bottom of the stroke.

In making its bottom (20%) rod reversal, the Mark II because of its reversed geometry dwells 35 - 40% longer over bottom stroke - allowing this extra amount of kinetic energy to reach out as additional overtravel, thereby generally increasing plunger stroke.

Years later, three major oil company studies reinforced the Mark designers earlier contentions: In his paper entitled, "Kinematics of Oil Well Pumping" by H. E. Gray, a distinguished Shell Oil Co. mathematician, shows where the front-mounted unit can generate as much as 10%

more net plunger travel excursion than a regular rear-mounted unit. Concluding his paper, Gray states:

> "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."

In another major oil company study, the following statement was made:

> "The limits of the pump stroke for the Mark II and Conventional units are determined in which an appreciable amount of difference of plunger stroke exists. The Mark II gives a greater total stroke because of greater overtravel at the bottom of the stroke." (Figure 15)

In comparing the Mark II to other beam geometries throughout the entire spectrum of pumping, a major oil company report concluded by stating:

> "Differences in pump stroke up to 10% exist between the Mark II and Conventional unit designs. The Mark II produces a longer stroke at speed and fluid load conditions most common in field use. Conventional design produces a longer (bottomhole) stroke at very high speeds which are seldom reached in practice."

Thus in determining maximum productivity of beam and sucker rod systems, not only must maximum pumping speed (for a given stroke length) be considered - but also the net plunger travel per stroke. A beam pumping unit with a 10% slower pumping speed, and a 10% increase in net plunger travel would produce the same amount of fluid per day as a unit pumping faster with a proportionately shorter net plunger stroke.

In most cases, the Mark II provides a greater net plunger travel, but because of the complex nature of the elastic rod string, it doesn't happen in every case.

B. Increased fill-time

Another factor in maximizing pumping unit productivity is fill-time. The only time new fluid can in-flow the pump is when the barrel is being emptied, and the only time this occurs is during the upstroke when the unit is elevating the fluid column.

In high volume pumping, when the barrel still has available capacity, the longer the time interval of the upstroke, the longer the fill-time to charge the barrel, and the greater the amount of fluid permitted to inflow during the upstroke. Thus, productivity is not only a function of the number of strokes per minute, or the net plunger stroke only, but fill-time as well.

Since the Mark II makes its upstroke in approximately 200° of crank rotation, this means that the upstroke fill-time interval for the non-symmetrical Mark II is approximately 8 - 10% greater than that of a symmetrical pumping unit whose upstroke is made in approximately 180° of crank rotation.

Here again, a faster downstroke on the Mark II affords a longer upstroke with accompanying increased fill-time - even if the pumping speeds are equal. If the Mark II, because of its faster return stroke, produces greater plunger travel, increased fill-time may even further enhance total productivity.

C. Delivering more safe work to the bottomhole pump.

It is fairly obvious that the more work the surface pumping unit can deliver to the bottomhole pump, without violating safe rod and structural capacities, the greater the potential productivity.

Observing the Goodman Diagram for a typical rod string - Figure 16, it can be seen that by lowering the peak polished rod load, a wider load range can be safely accommodated.

For a given stroke length and card shape, the wider the load range, the greater the amount of safe work delivered to the bottomhole pump - often making possible increased productivity. Figure 16 illustrates how, by reducing the peak polished rod load from P_c to P_m permits the accommodation of a wider load range from R_c to R_m, resulting in the ability of the Mark II unit to deliver more safe work to the bottomhole pump as shown in Figure 16b.

Although greater (potential) work area is shown in the right hand Mark II card, both applications are equally loaded as far as the rods are concerned - though the right hand card is even more beneficially loaded as regards structural capacity.

In many applications - though not in every one - the designers hoped that by lowering peak polished rod load, a wider load range could be accommodated on most applications, with increased safe bottomhole work, greater plunger travel, and increased fill-time, resulting in the probability of increased production.

VIII. REDUCING ENERGY CONSUMPTION

Mark II designers believed that, (1) if the torque could be substantially reduced and smoothed out, in most cases a smaller prime mover could be used that would operate closer to its rated (and more efficient) capacity, with attendant power savings; (2) less heat loss would result due to use of a straight mechanical system like the Mark II employed, rather than a less efficient system using hydraulics or pneumatics; and (3) lower electrical demand charges would result from a smoother torque load.

Input energy to the beam pumping system is employed in two different ways:

(1) To offset friction and other thermal losses
(2) And with the power that is left over, to beneficially lift fluid.

This means that for a given energy input (into the prime mover), the smaller the heat or friction loss - the greater the amount of energy available to beneficially lift fluid.

Since Conventional unit geometry produced unwanted backdriven energy in most pumping applications, the Mark II designers recognized early that this negative, or backdriven energy, could not be given up unless additional input energy was delivered to the system at some other part of the pumping cycle.

This negative or backdriven energy not only suffered normal I^2R heat loss, but the additional input power, added to the regular positive energy, even further increased I^2R losses.

By means of the Unitorque system smoothing out and reducing torque peaks, the Mark II designers hoped to diminish, or in some cases, even eliminate negative or backdriven energy wherever possible in order to significantly lower prime mover heat loss.

Many years later, James Eckmier, a highly respected engineer, formerly of the Shell Oil Company of Canada, shows how the torsional fluctuations of a beam unit affects the efficiency of its electrical prime mover. Figure 17.

The cyclic load factor (CLF) of an electric motor is the ratio of the RMS (thermal) current to the average current. If an electric motor had no heat losses, the thermal current and the average current would be approximately equal, and the cyclic load factor value would be 1.0.

As the cyclic load factor increases, a greater proportion of the motor's input energy is devoted to thermal losses, with less energy being available for lifting fluid.

On the right hand side of Figure 17(c), is seen a widely fluctuating torque load driven by an electric motor prime mover. Because of the extreme torsional variation, the cyclic load factor is relatively high, i.e, 1.5. This means that about one-third of the input energy is consumed as heat loss and two-thirds devoted to fluid lift. The lower of the two dotted lines on the torque figure represent the average current demand of the system which is proportional to the unit's mechanical output, and the dotted RMS current line above it is proportional to the amount of energy delivered to the prime mover.

The distance between the average current line and the RMS current line is substantial. The wider the separation, and the greater the area of that particular rectangle between the two lines, the greater the heat loss for this excessive torque range.

The central diagram, Figure 17 (b), having a cyclic load factor of 1.25, shows the amount of mechanical work (i.e., average current) that the unit delivers to the rods is the same here as it was before with the higher cyclic load factor.

Since less of the electrical energy is dissipated as heat loss, the RMS current line has moved downward closer to the average line. This means that because of the lower torque fluctuation, the prime mover dissipates less thermal energy resulting in a higher percentage of the input energy devoted to lifting fluid. In this case, about one-fifth of the input energy is lost thermally while approximately four-fifths is devoted to beneficial fluid lift.

The left diagram, Figure 17(a), shows that, when the torque load is considerably smoother, the average current and the RMS current values are even closer together, with little difference in heat loss area between the two lines. This shows that even less of the input energy is lost thermally, leaving a higher percentage to perform the useful mechanical fluid lifting. It should be emphasized that each of the three examples noted above deliver the same amount of mechanical work to the rod string.

This smoother torque loading provides a more effective system, not because the pumping unit proper is more or less efficient, but because the prime mover driving it operates in a more efficient mode.

Generally, in the past, the Conventional unit's torsional load range on a majority of applications has been high. On the other hand, the Mark II torque fluctuation, because of the Uniform Torque System, has been relatively lower on the same applications - often with a significantly smaller cyclic load factor.

Substantial CLF reduction in a majority of pumping applications has shown the Mark II to generally minimize both power consumption and demand.

IX. IMPROVING FOUNDATION MOUNTING - TWO-POINT SUSPENSION

The final Mark II objective, a stable and less costly foundation was unplanned, and came about by accident.

The original Mark II was set up on a smooth, abandoned Conventional unit concrete foundation, over a 5,000 ft. well in north east Colorado. Knowing that the forces in the Mark II all acted downward, the unit was simply set on a series of 2" x 12" timbers spaced about 15 inches apart - <u>with no tie-downs</u>.

At a modest pumping speed, it was found that the unit rocked back and forth each stroke with the front and rear alternately moving up and down some 12 to 15 inches.

After the panic subsided, it was decided to remove the two middle 2 x 12's, and observe the results. This seemed to reduce the rocking motion enough that two more central timbers were removed, and then two more, and then still another two.

When all timbers except the two at the front and two at the rear were removed, the unit showed no movement at all - even at pumping speeds as high as 32 - 54 in. - spm.

The unit base sills were then stiffened to a safe allowable stress level and, from that time on, most Mark II's were simply mounted on transverse piers, fore and aft, for a stable and economical foundation. Figure 18.

Mark II units set on Conventional one-piece foundations occasionally had to be tied down in back to keep them from rocking. Fortunately, in most installations the stable two-point suspension mounting was used.

The designers hoped that use of the two transverse mounting piers would save significant foundation and installation expense, especially in areas where the cost of concrete was excessive, or where bases had to be eventually buried or the unit moved.

CONCLUSIONS

It is believed that few if any beam pumping unit performance advances were attempted, made, or even envisioned prior to the advent of the Mark II pumping unit in the mid 1950's. Hopefully the history, experience and rationale of this beam pumping unit has shown that improved performance is, in some cases, not only possible but practical.

AFTER WORD

Much of the Mark II technology was worked out "long hand" for it generally preceded both "computer assistance", and the invaluable beam and rod information developed by the Sucker Rod Research Institute, and later by the Nabla Corporation and others.

It would be a singular triumph for the designers if all the Mark II goals,attempted nearly 35 years ago, had been met and verified and understood by the industry. The fact is it did not happen in this tidy and convenient way - every time on every well. But a few or perhaps in some cases, many of these hoped for advances or goals did occur on a great number of Mark II applications - many without the operator's knowledge regardless of how beneficial they may have been to the pumping system.

Despite the relative "obscurity" of these alleged performance features, Lufkin Industries has placed some twenty-five thousand Mark II units in service in many of the oil producing regions of the world, among over a thousand different oil companies.

Mark II designers of the 1950's however would surely take pride and comfort in a recent statement made by one of the industry's most distinguished scientists, Dr. S. G. Gibbs, who said of the Mark II - "This outstanding development has withstood the test of time, and even in this computer age, it s original concept has not been improved upon".

ACKNOWLEDGMENT

This little history of a particular oil pumping device is dedicated with heartfelt gratitude to James Courtney Wright, a friend, partner, and the most brilliant engineer and Engineering Designer I have ever known .

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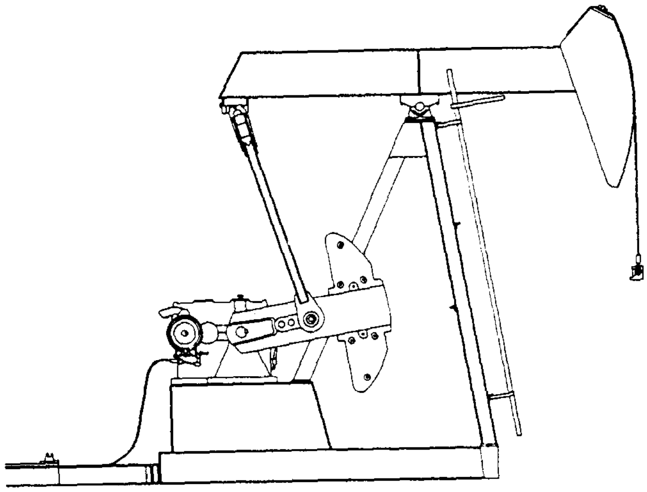
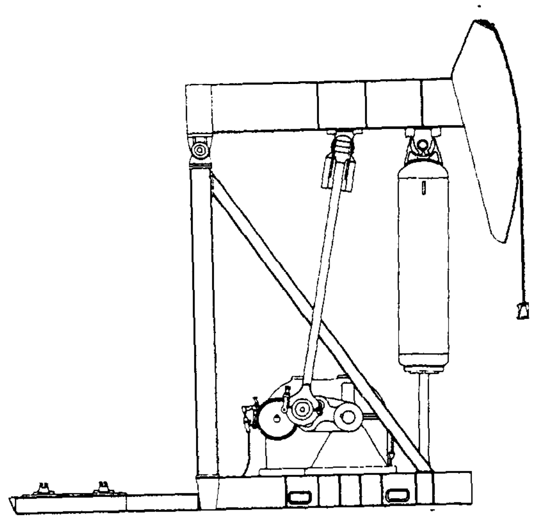


CRANK BALANCE UNIT

Figure 1



AIR BALANCE UNIT

Figure 2

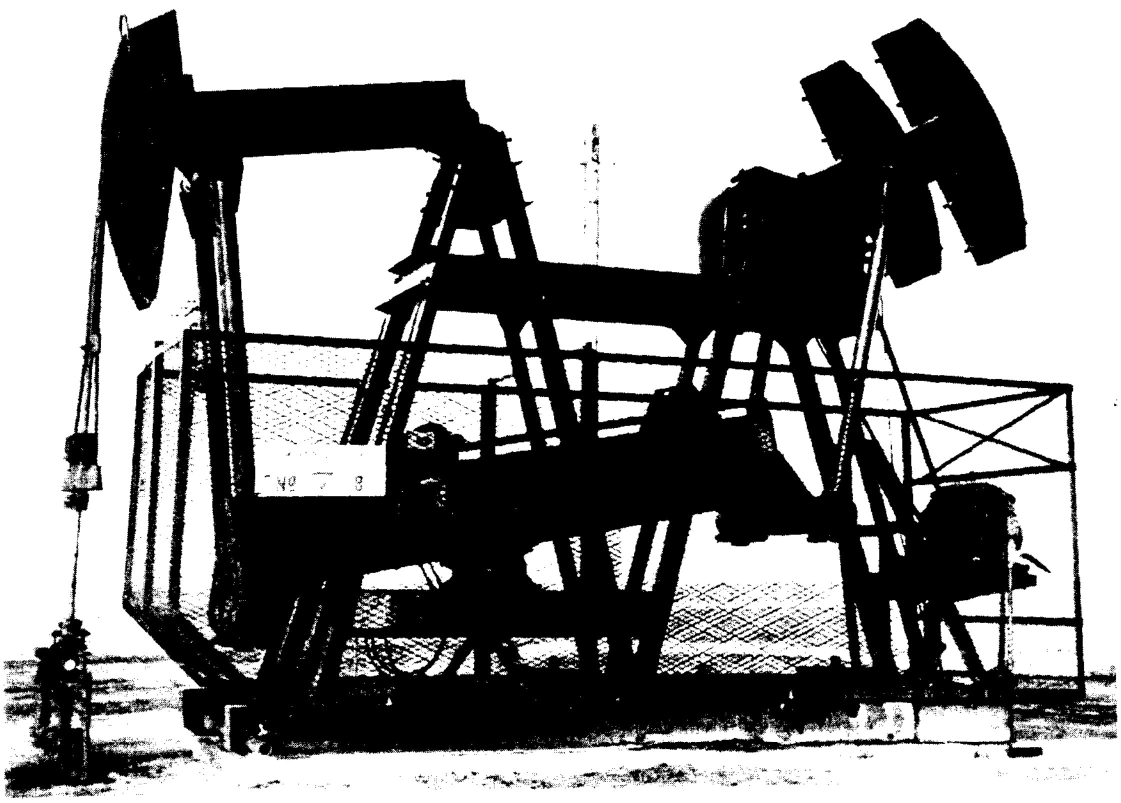


Figure 3

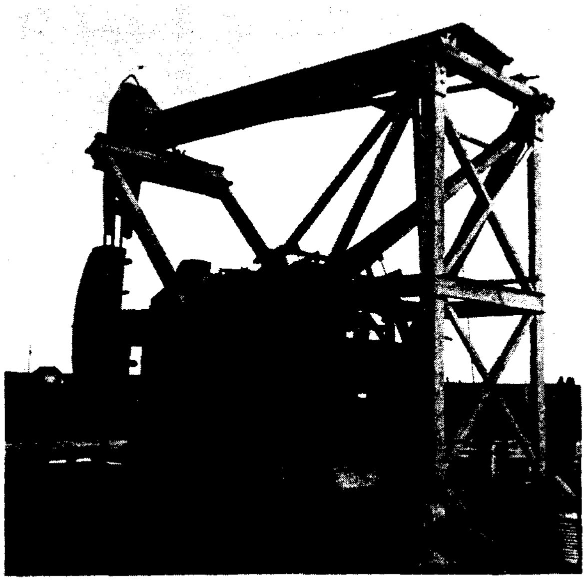


Figure 4

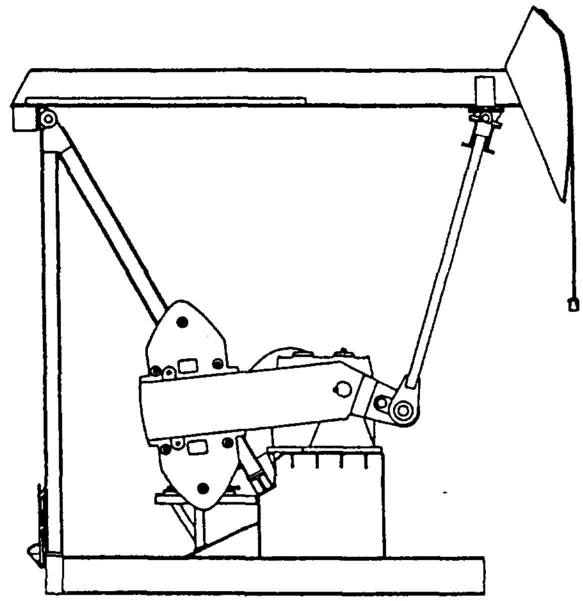


Figure 5

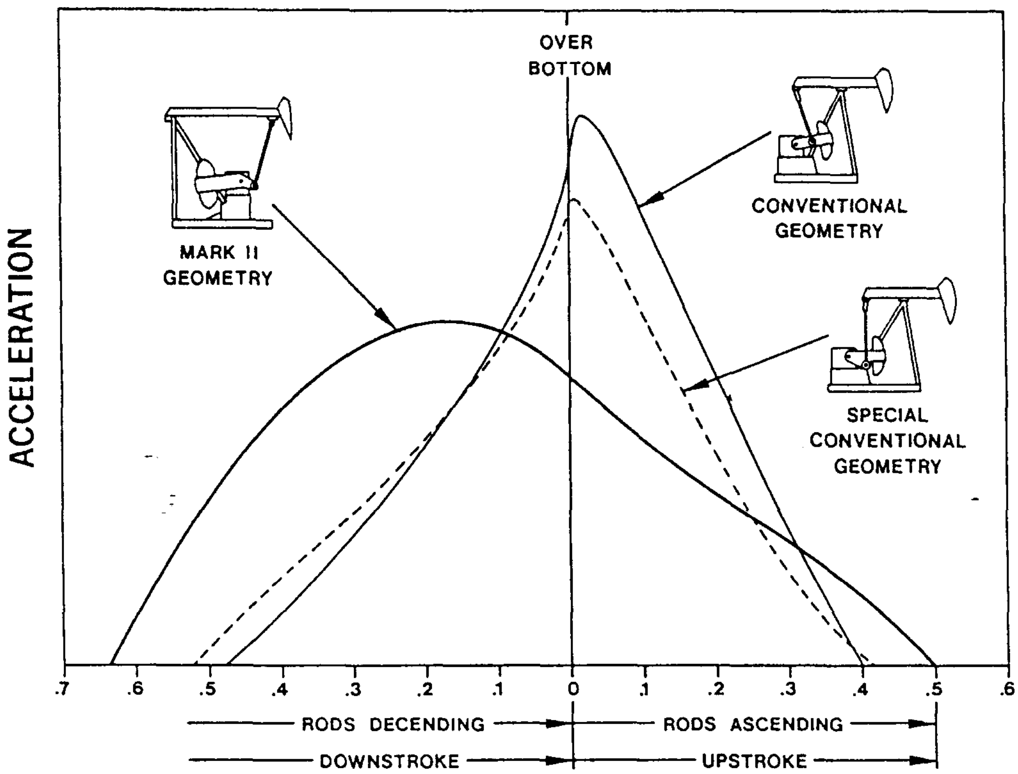


Figure 6

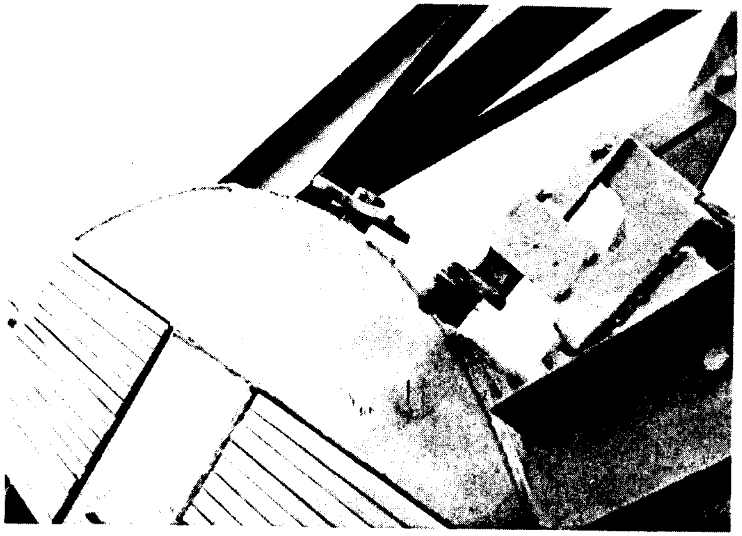


Figure 7

Figure 8

THE *UNITORQUE* GEOMETRY

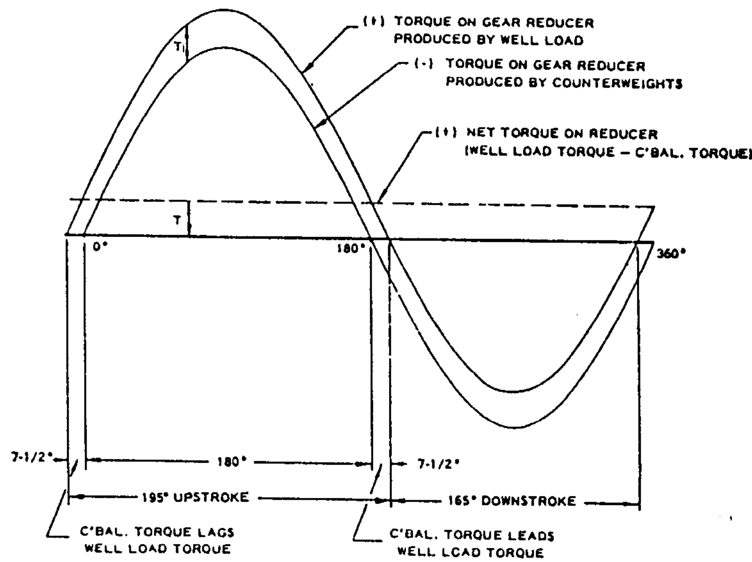
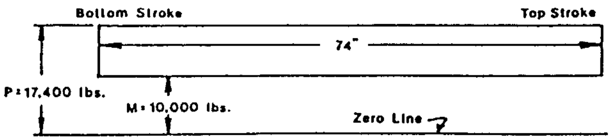
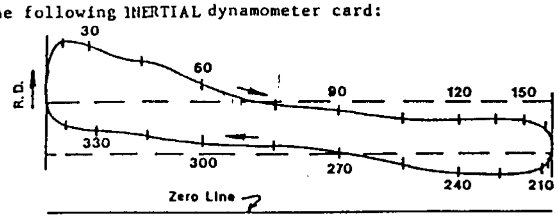


Figure 9

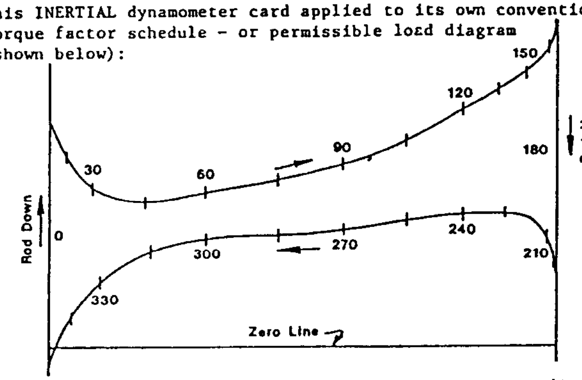
This conventional unit well load:



Driven 15 - 74" SPM by its own conventional unit geometry, develops the following INERTIAL dynamometer card:



This INERTIAL dynamometer card applied to its own conventional torque factor schedule - or permissible load diagram (shown below):



Results in the following conventional unit torque pattern (in-balance)

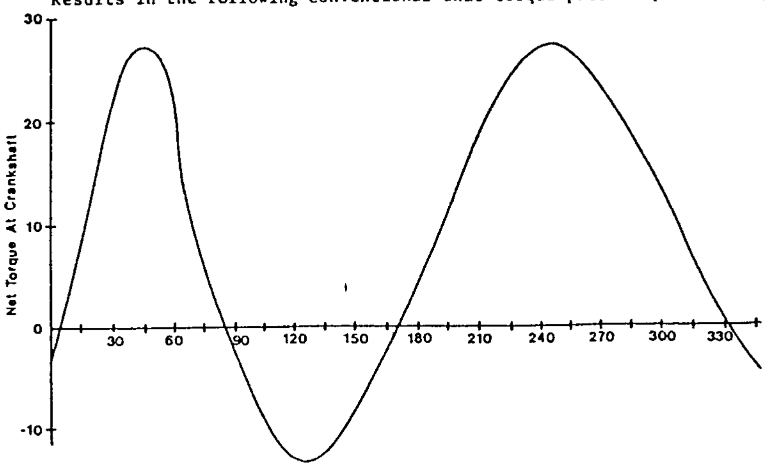
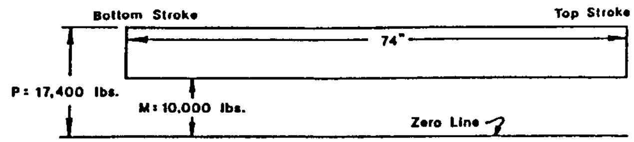
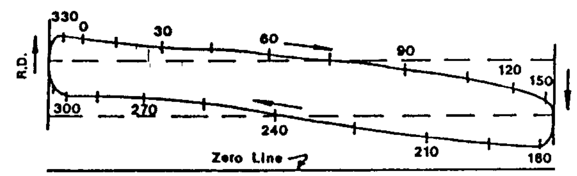


Figure 10a

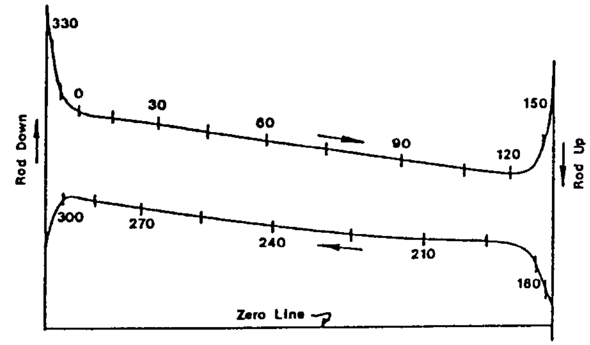
This Mark II unit well load:



Driven 15 - 74" SPM by its own Mark II unit goemetry, develops the following INERTIAL dynamometer card:



This INERTIAL dynamometer card applied to its own Mark II torque factor schedule - or permissible load diagram (shown below):



Results in the following Mark II unit torque pattern (in-balance):

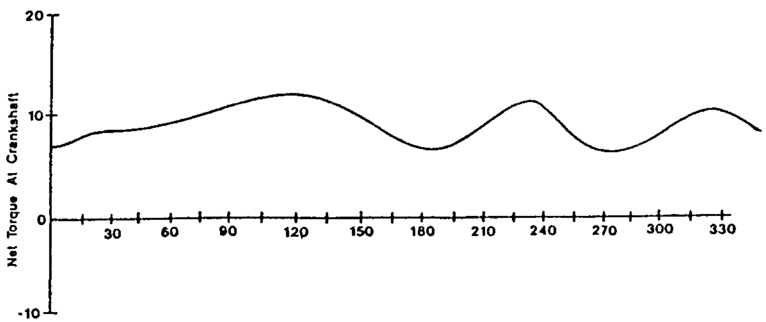
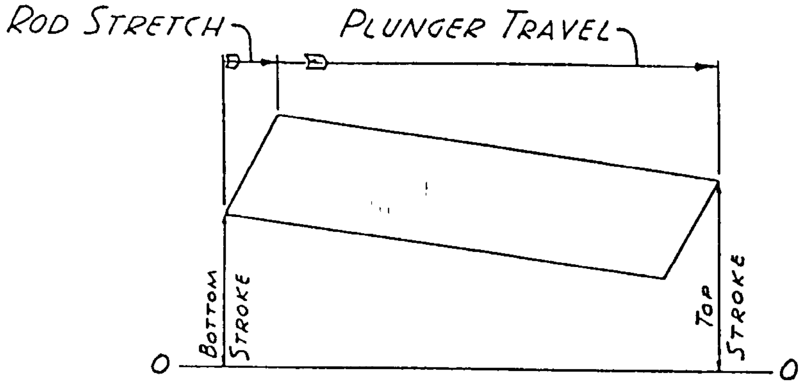
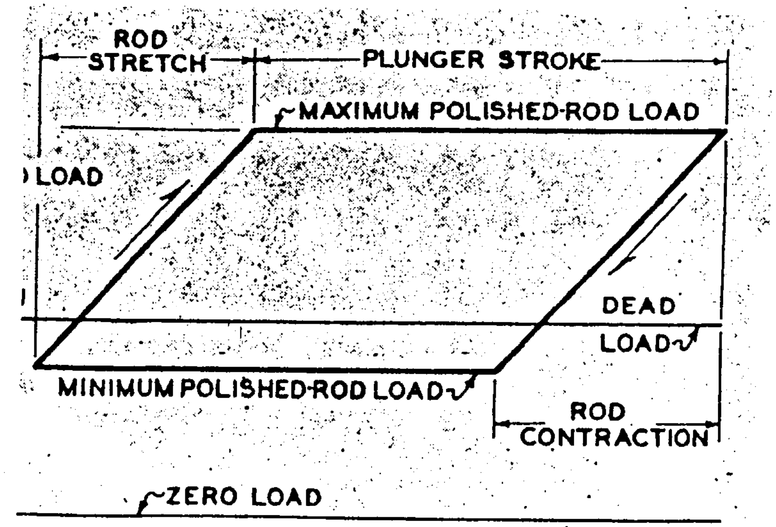


Figure 10b



DYNAGRAPH SHOWING INERTIA AND STRETCH

Figure 11a



DYNAGRAPH WITH STRETCH ONLY

Figure 11b

A SERIES OF TYPICAL SLOW SPEED <u>OVERTRAVEL</u> AND <u>UNDERTRAVEL</u> CARDS
(INCLUDING ROD STRETCH, INERTIAL, HARMONIC AND FRICTIONAL FORCES)

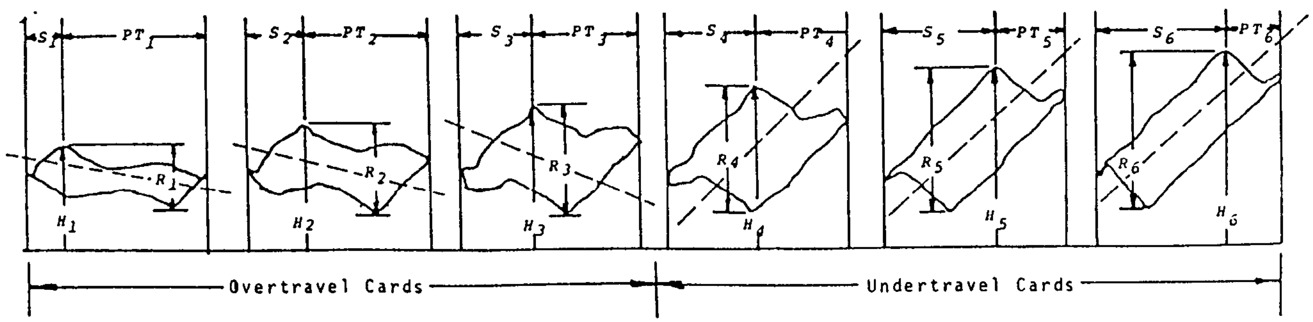


Figure 12

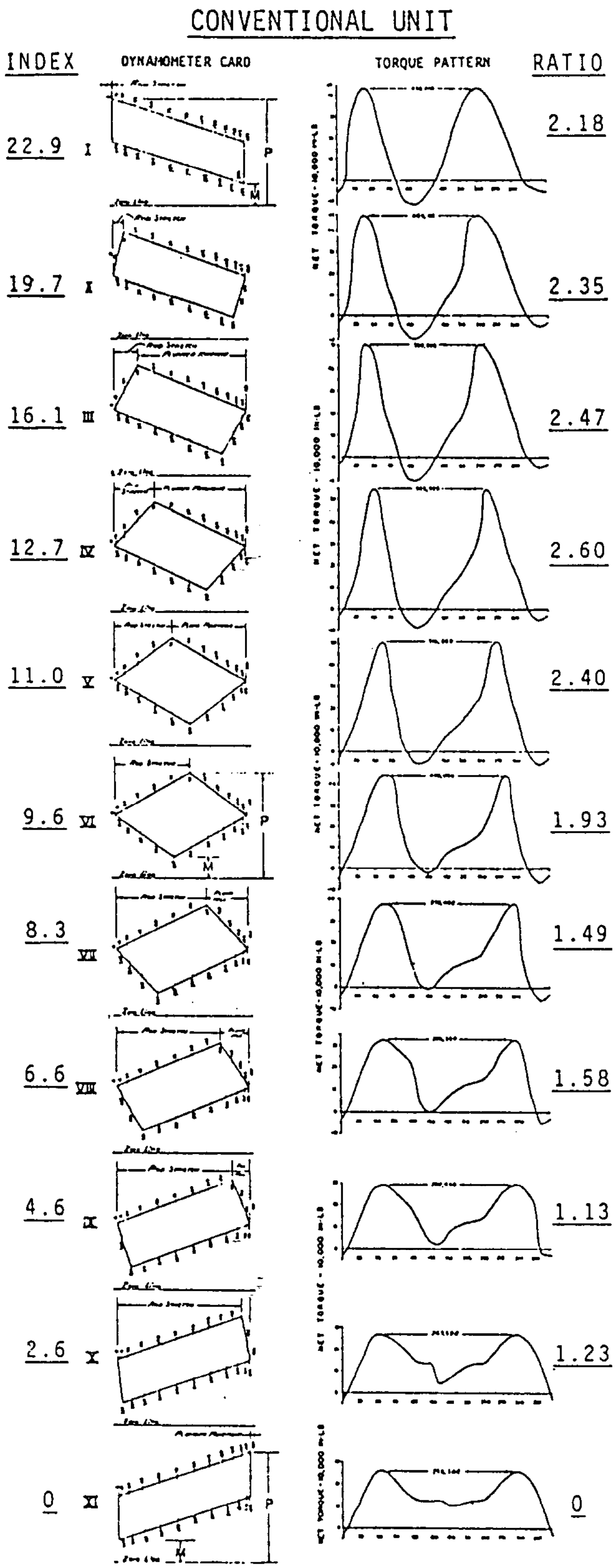


Figure 13a

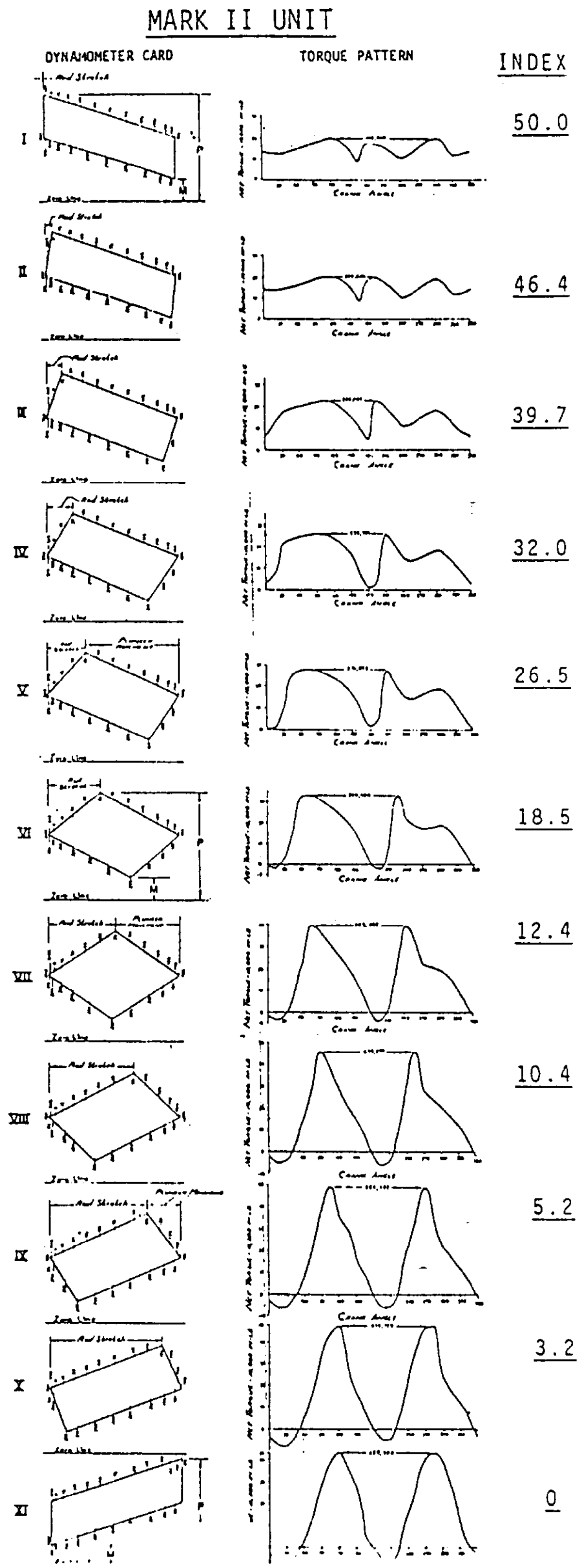


Figure 13b

STATISTICAL ANALYSIS OF 400 DYNAMOMETER CARDS
(RANDOM SAMPLING)

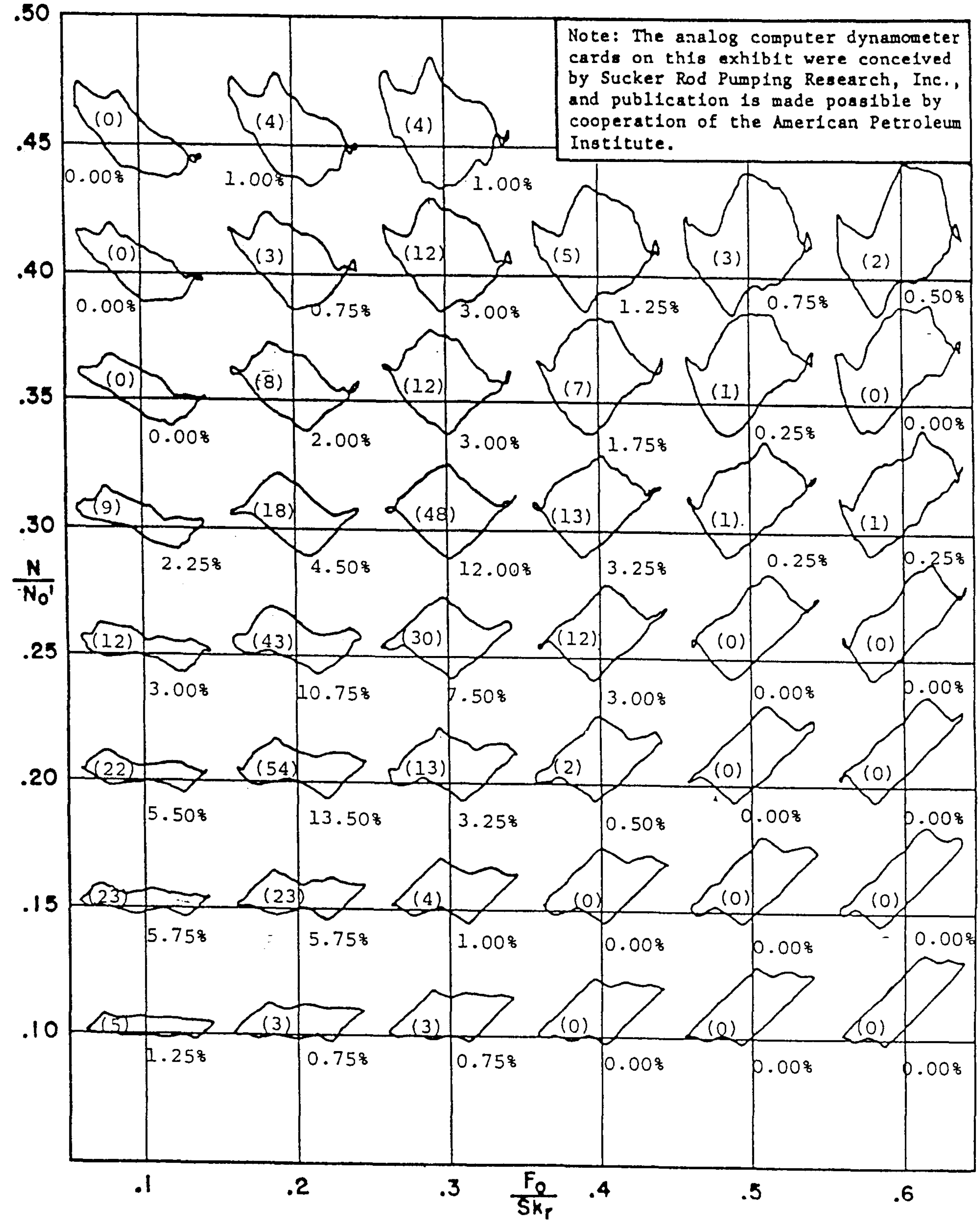


Figure 14

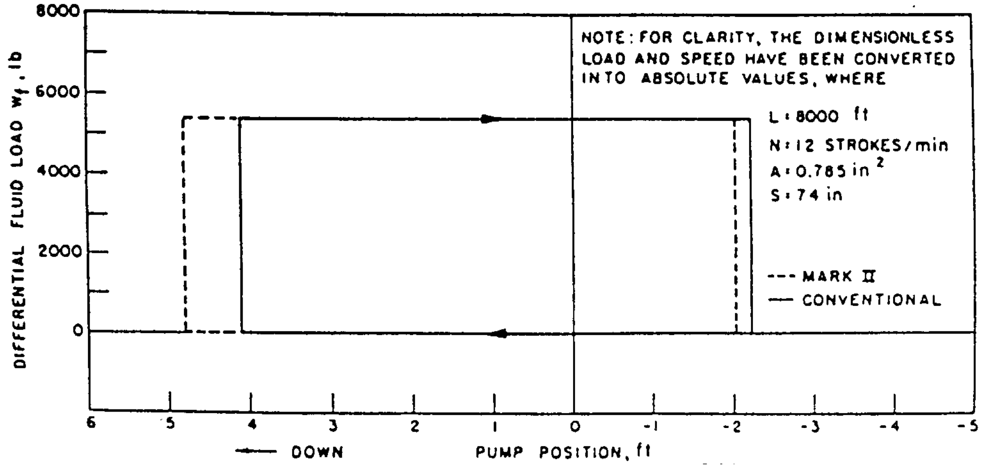
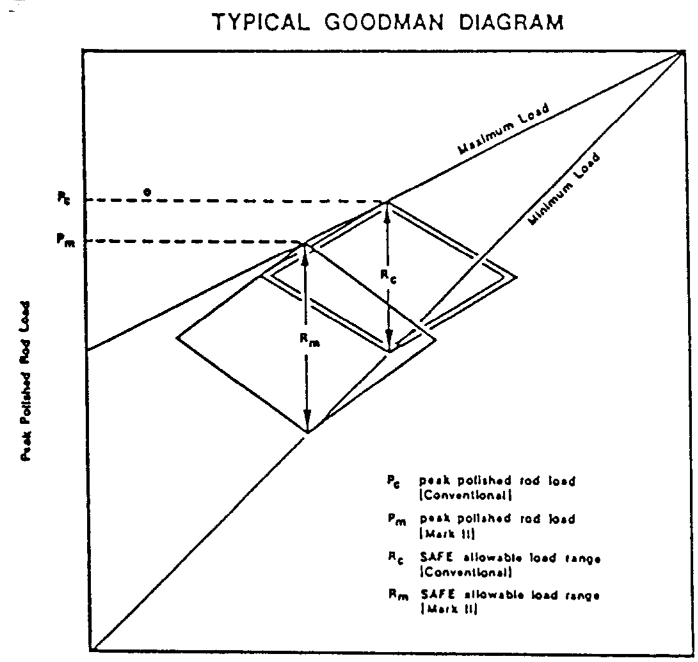


Figure 15 - Comparison of pump dynagraphs of $W_f/SK=0.3$ and $N/N_0=0.4$ showing greater overtravel resulting from Mark II design



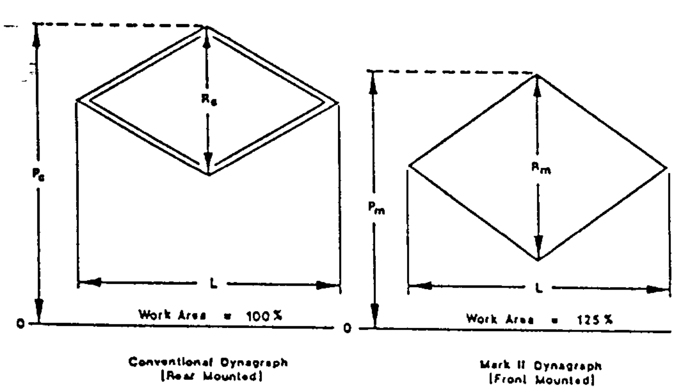
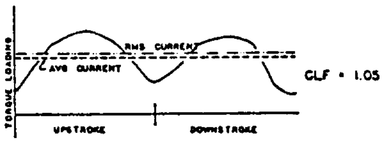


Figure 16a

Figure 16b

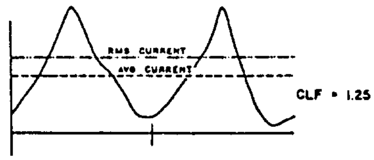
TYPICAL CYCLIC LOAD FACTORS* FOR BEAM PUMPING SYSTEMS
(after Eickmeier)



CLF = 1.05

Pumping Unit, Type "A"

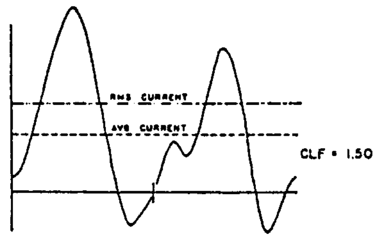
Figure 17a



CLF = 1.25

Pumping Unit, Type "B"

Figure 17b



CLF = 1.50

Pumping Unit, Type "C"

Figure 17c

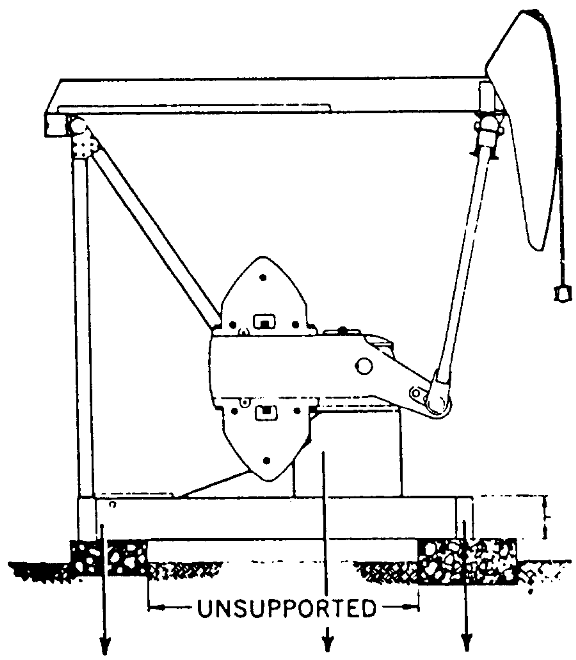


Figure 18