SOME OBSERVATIONS OF AN IMPORTANT THOUGH LARGELY OVER-LOOKED FACTOR IN THE UNDERSTANDING AND OPTIMIZING OF CERTAIN BEAM AND SUCKER ROD PUMPING APPLICATIONS

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Abstract
In many applications, an important aspect of beam and sucker rod pumping operation and analysis has been generally unrecognized. This paper reviews and emphasizes, not only this deficiency, but also the intrinsic value of several classes of dynamometer card shapes and slopes, to certain pumping unit geometries and applications.
A basic understanding of the nature, character and shape of the polished rod dynagraph, and its relationship to the particular pumping unit geometry selected - coupled with the modern analytical techniques of S. G. Gibbs and others - should materially assist the designer in determining a more effective beam and rod pumping system.

In the long and eventful history of producing oil from subterranean depths with a beam and sucker rod pumping device, two inventions or innovations have been of particular importance. (1) The invention of the polished rod dynamometer and (2) the development of the technology to simulate, measure, and interpret both the surface and subsurface information contained in the dynamometer card or dynagraph.
Recording dynamometers were first used for testing the performance of rod operated pumping wells as early as 1923, and have continued with increasing popularity thereafter.
Some of the early inventors, innovators, and students of this new science were: Hallan Marsh and E. V. (Jim) Watts, whose early work on and with the hydraulic polished rod dynamometer was of much importance to the industry; father of the bottom hole dynamometer was the distinguished scientist, Walton Gilbert; Dr. Emery Kemler, Douglas O. Johnson, and J. G. Umpleby pioneered the electronic polished rod dynamometer; Leo Fagg and D. O. Johnson developed a popular mechanical dynamometer; John Slonneger, R. W. Reinets, and Kenneth Mills did outstanding work on load prediction formulas and dynamometer card interpretation.
Another important development in the history of artificial lift came in the 1950's, when a consortium of major oil companies and oilfield equipment manufacturers banded together to form the Sucker Rod Pumping Research Institute, or simply SRI. One of the chief objectives of this group was to study the complex behavior of the beam and sucker rod system.
Using, at first, a mechanical simulator and later an analog computer, a large amount of valuable information was developed on sucker rod pumping technology. Professor Warren Snyder and A. J. Bossert were largely responsible for the analog computer work of the sucker rod pumping research institute (SRI) whose research formed the basis for the API simulation and tabulation of synthetic dynamometer card information.
Beginning in the early 60's, further major advances were realized with the work of S. G. Gibbs using a digital computer, rather than the analog device, to effect a solution of wave equation techniques applied to sucker rod pumping.
Following are several basic studies, some old, some new, which attempt to show in simple, diagrammatic fashion certain involved characteristics and trends in the relationship of the load displacement forces of the rod string, to the geometry of two particular pumping unit types, 1) the conventional unit, (Class I Lever), and 2) the Mark II, (Class III Lever). To simplify and make as

meaningful as possible, some familiar analogies and approximations are used throughout these studies. Also, **rod stretch** is considered to be roughly proportional to the distance between the bottom of the surface stroke, and the position of the peak polished rod load, while the **plunger travel** is approximately equal to the distance between the peak polished rod load and the top of the surface stroke. Steel rods are assumed throughout this discussion.

It is hoped that this series of studies will offer another dimension to the modern numerical solution of beam pumping load prediction and analysis, much as the dynamometer card illuminates the unique shape and anomalies of the polished rod load pattern.

I. A Systematic Arrangement of the Full Pumping Spectrum of Conventional Unit Dynamometer Cards

The SRI/API was the first to display dynamometer cards in a systematic fashion. Also, it was demonstrated for the first time the utility of a non-dimensional presentation of rod pumping data. In the process of creating design graphs, the analog computer generated synthetic dynamometer cards. These were cataloged versus F_0/SK_r and N/N'_0.

Figure 1 shows this familiar array of synthetic **conventional unit** dynamometer cards, along with the results of a statistical analysis which will be discussed later.

Broadly speaking, this full spectrum of dynamometer cards can be roughly divided into three classes, (1) **undertravel**, (2) **overtravel**, and (3) **neutral** or essentially symmetrical cards.

(a) **Undertravel** cards are characterized by having their major axis slope **upward** to the right, (b) **overtravel** cards slope **downward** to the right, and (c) **neutral** cards are those that are roughly symmetrical about a central vertical axis. This discussion will be chiefly concerned with only **undertravel** and **overtravel** card applications.

II. A Schematic Diagram of Typical Overtravel and Undertravel Dynamometer Cards

Figures 2-a and 2-b are simplified schematic diagrams of typical **overtravel** and **undertravel** dynamometer cards, both having the same peak and minimum polished rod loads, and both performing the same amount of polished rod work.

The principal difference between these two cards is that the **overtravel** profile, sloping downward to the right, shows limited rod stretch and substantial plunger travel, while the **undertravel** card exhibits greater rod stretch which often reduces plunger stroke.

In general, the factors that change an **overtravel** card application to an **undertravel** one are simply rod stretch and to a lesser extent the resulting harmonic forces. Pumping unit geometry may have little effect in this matter.

A second comparison of conventional unit **overtravel** and **undertravel** dynamometer cards is seen in Figure 2-c[3], showing a more realistic diagram of the reduced peak polished rod load and load range of the **overtravel** card as well as its increased plunger travel.

Figure 2-d[3] shows the same basic card as in Figure 2-c, but increased rod stretch has converted it to an **undertravel** card with higher rod loads, load range and a lesser plunger stroke.

III. A Series of Six Conventional Unit, Non-Dimensional Dynamometer Cards - Half Overtravel, Half Undertravel - All Having the Same Non-dimensional Pumping Speed

Figure 3 shows a series of six dynamometer cards taken directly from Figure 1, starting with the station $N/N'_0 = 0.20$, and $F_0/SK_r = 0.10$ and running horizontally to the right. The three cards on the left are typical **overtravel** card shapes, while the three cards on the right have **undertravel** profiles.

This comparison of **non-dimensional** cards clearly shows that a wide difference exists between the rod loading and plunger travel of the **overtravel** cards (Figures 3-a, b, c), and the **undertravel** ones (Figures 3-d, e, f).

Not knowing if the SRI/API non-dimensionalized dynamometer cards were properly scaled equally, application 3-b (**overtravel** example) and 3-e (**undertravel** example) were both **dimensionalized** by assuming that the conventional unit raised 500 barrels per day of fluid from 3500 feet with API 66 rods. This application conformed closely to the non-dimensionalized stations involved.

After dimensionalizing this conventional unit application, the peak polished rod load of the **overtravel** card showed a significant reduction of 18.3% and the load range of the **overtravel** card was reduced by 14.4%.

Though rod loading was dramatically improved in the **overtravel** example, concurrently the peak torque load was (as expected) increased from 235,000 in. lbs. to 262,000 in. lbs., while the NEMA D prime mover requirement increased from 28.9 hp (30 hp) to 37.7 hp (40 hp), one NEMA size larger. This presented a confused picture. If rod loading were the critical factor, the **undertravel** application would be superior - if torsional load and prime mover size were more important, then the **undertravel** application might be better - truly a mixed bag! (Figure 4, the tabular display of the above numbers, is not shown.)

This example emphasizes the need for the accurate modern wave equation solution in order to optimally balance the pros and cons of such a paradoxical application.

IV. Pumping Unit Geometry and Its Preference for Particular Rod Load Characteristics

Conventional units are known to prefer dynamometer cards that slope, up-to-right (see for example $F_0/SK_r = 0.5$ and $N/N_0' = 0.2$ in Figure 1). This conventional unit preference for **undertravel** cards is certainly correct, in most applications, as regards torsional reduction, smaller prime mover requirements and possibly power savings. Conversely, if rod loading is of major importance, perhaps the **overtravel** dynagraph shape may be preferable for the conventional unit, at least in most applications.

As regards torsional reduction, Mark II and Air Balanced units are known to prefer down-to-right cards (see for example $F_0/SK_r = 0.2$ and $N/N_0' = 0.3$). When optimizing gearbox load, it is the designers task to alter pumping parameters to attain the dynamometer card tilt preferred by the unit to be used. For example, a conventional gear box may be overloaded in a down-to-right condition $F_0/SK_r = 0.2$ and $N/N_0' = 0.3$. The designer knows to change pumping parameters to move downward and rightward in the array of cards shown in Figure 1. This can be accomplished in various ways, by shortening the unit stroke, installing a larger downhole pump or decreasing pumping speed. The card tilt then changes in the direction of the up-to-right in keeping with conventional unit preference and hopefully, gearbox loads will be reduced. Such understanding would have been more difficult to obtain without systematic presentation of predicted dynamometer cards.

V. The Influence of Dynamometer Card Slope on Peak Net Torque Values of Mark II and Conventional Units

A simulated **overtravel** card with modest stretch and no harmonics and with its full 360_ torque load curve developed using **Mark II** (Class III) torque factors, adjacent, is shown with dotted line in the lower left station of Figure 5-a.

A second torque curve of the same card developed with **conventional unit** torque factors (unbroken line) was superimposed on the Mark II torque curve, Figure 5-a.

It can be seen that with zero slope the Mark II in-balance torque is 11.2% lower than that of the conventional unit, all else equal, Figure 5-a.

Moving upward one station to Figure 5-b, the same **overtravel** dynamometer card is tilted 4_ clockwise and the torque history of both Mark II and conventional units are developed as before. Here the Mark II torque is reduced by some 28.9% under that of the conventional unit.

At the 8_ slope, the **overtravel** card of the Mark II shows a 41% torsional reduction over that of the conventional unit, all else equal. Figure 5-c.

At 12_ slope, Figure 5-d, the **overtravel** Mark II dynamometer card continues its 40.7% torsional reduction.

At 16_ **overtravel** slope, the Mark II torsional reduction is 39.4% under that of the conventional unit. And at 20_ **overtravel** slope, the Mark II torque reduction is 36.6%.

Thus for the Mark II geometry an **overtravel** dynamometer card slope of 8-16_ would provide maximum torque reduction compared to a similar conventional unit card, all else equal.

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Moving to the right hand column, Figure 5-g, with 0_ slope, the Mark II torsional reduction is still 11.2% lower than the conventional unit torque, all else equal.

Moving up to Figure 5-h, at 4_ **undertravel** tilt, peak torque of both conventional unit and Mark II unit are approximately the same.

At 8% slope **undertravel**, the conventional unit torque was 3.6% lower than the peak torque of the Mark II unit. Figure 5-i.

At 12_ slope (**undertravel**), conventional torque is 10.5% lower than the Mark II peak torque. Figure 5-j.

At 16_ **undertravel** slope, conventional torque is 11.5% lower than Mark II peak torque. Figure 5-k.

At 20_ **undertravel** slope, conventional unit torque is 16% lower than Mark II peak torque. Figure 5-l.

In this comparative study, the Mark II has lower and more desirable peak toque reduction values at 7 stations with up to 41% reduction, while the conventional unit has lower and more desirable torque values at 4 stations, up to 16.0% under that of the Mark II.

VI. Three Different Pumping Unit Geometries Pumping the Same Simulated Rectangular Rod Load with No Stretch or Harmonics Would Each Produce a Typical Overtravel Card

Figure 6 shows a synthetic rectangular dynamometer card (dashed line) with no stretch or harmonics and three overlapping dynamometer cards (solid Line) that resulted from three completely different pumping unit geometries, a, b, and c, pumping this same simulated rectangular load, i.e., 10-74 inch spm.

An important, but obscure, fact is that the three resulting inertial cards are all **overtravel** shapes regardless of the pumping unit type that produced them.

If it weren't for rod stretch and resulting harmonics, all beam and sucker rod units driven by a rotating crank system would produce only the **overtravel** dynamometer card, sloping downward to the right, regardless of the lever system or type of pumping unit employed.

VII. Dynamometer Card Slope Changes with Pumping Speed

Figure 7 shows that in most **overtravel** type loading, the dynamometer card rotates **clockwise**, as pumping speed increases. This is shown by selecting the vertical line of dynamometer cards with a light to medium (dimensionless) fluid load, shown at the extreme left of the full array of cards, beginning with the station $N/N'_0 = 0.10$; $F_0/SK_r = 0.1$(Figure 1).

On the other hand, in most applications, **undertravel** cards seem to rotate **counterclockwise** with an increase in pumping speed, beginning with station $N/N'_0 = 0.1$ and $F_0/SK_r = 0.5$, and running vertically upward (Figure 1).

VIII. Variation of Torque Load in Both a Conventional Unit and a Mark II Unit as Rod Stretch is Successively Increased While Plunger Travel Decreases, Automatically Changing the Dynamometer Card Shape from Overtravel to Undertravel

Momentarily disregarding harmonics, Figure 8 shows a theoretical (parallelogram type) **conventional** unit **overtravel** card (top left) with no rod stretch, resulting in the same plunger travel as the surface stroke, along with its accompanying in-balance torque load diagram. Increasing rod stretch by 15_ increments, while holding peak and minimum polished rod loads, and work area, constant, successive torque loads are plotted. In this **conventional unit** illustration, when plunger travel is maximum, a high torque results - and when plunger travel is minimum, the lowest torque is produced. By successively increasing stretch from top to bottom, the card profile changes from total **overtravel** to total **undertravel**.

Figure 8 (top right) illustrates the same theoretical **overtravel** dynamometer card and its successive stretch in 15_ increments applied to Mark II, Class III geometry, the same as before. Here maximum plunger travel results in minimal torque while successive, increased stretch produces higher torque, as the card profile changes from **overtravel** to **undertravel** and maximum torque occurs only when the 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 a theoretical study of two different beam geometries pumping at slow to minimum speeds - but to a reasonable 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 a majority of cases, the basic ideas presented are valid.

This comparison emphasizes that: in considering peak polished rod load, load range and plunger stroke in many or most applications, the **overtravel** card is more desirable than the **undertravel** card when the unit is properly applied.

IX. Statistical Analysis of 400 Dynamometer Cards (Random Samplings)

The SRI/API array of the total spectrum of conventional unit dynamometer cards is shown in Figure 1, and to the casual observer, it might be assumed from the above data, that in actual field practice, both **undertravel** and **overtravel** card shapes might be distributed approximately 50-50, and that their intrinsic value to effective rod pumping might be equally desirable. This would be an erroneous speculation as the statistical analysis will show. Figure 1 above shows the results of a study of some 400 dynamometer cards received 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 1 card in 9, statistically, is an **undertravel** card, while approximately 8 of each 9 cards are **overtravel**.

The chief factor accounting for the fact that approximately 12% of the dynamometer cards in the above spectrum are **undertravel** is excessive rod stretch and the resulting spurious harmonics, often considered undesirable.

This independent statistical study amplifies the fact that a great portion of existing field dynamometer cards are essentially **overtravel**.

Sometime after this initial study was made, 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.

These two statistical studies of some 900 or so dynamometer cards, randomly received, would strongly indicate that, in the past, the industry design of rod pumping systems, either inadvertently or intentionally, strongly favored the **overtravel** configuration as opposed to the **undertravel** class.

X. How Pumping Unit Geometry Influences Torque Values

Perhaps one of the most significant studies of conventional and Mark II unit comparative torque values was made sometime ago by Mr. Ben Elliott and Dr. Manuel Aguirre of Lufkin Industries (Figure 9).

The idea of the study was to take an arbitrary synthetic, rectangular dynamometer card, Figure 9, having a peak polished rod load of 17,400 lbs., and a minimum polished rod load of 10,000 lbs., and apply it, first, to a standard conventional unit, and secondly, to a Mark II, both pumping 15-74" strokes per minute with no stretch or harmonic content, using both unit's respective accelerative schedule to develop two inertial cards. This was done and as shown above, both conventional unit and Mark II inertial cards resulted in similar **overtravel** shapes, and when a regular API torque factor analysis was made of both inertial cards, the following in-balance net torque curves resulted.

It can be seen that the net torque values are significantly lower and smoother when the **overtravel** configuration of the Mark II dynamometer card is applied to its own Mark II (Class III Lever) geometry.

Superimposing the inertial card of the conventional unit with its **negative tilt** on its permissible load diagram with its **positive slope** emphasizes diagrammatically wide torsional fluctuations will result.

Conversely, the Mark II having its inertial card with a **negative slope**, the same as the tilt of its permissible load diagram, shows a much reduced and smoother torque load than the conventional unit, although both basic cards have the same peak and minimum rod loads and are performing the same rod work at the same rate.

If the **overtravel** inertial card of the **conventional unit** is applied to the Mark II geometry, it would result in a similar lower and smoother peak torque curve as well.

One of the Mark II's chief torsional features is that it often develops an **overtravel** card that its geometry favors, significantly lowering and smoothing out the torque.

The burden of this paper is to show, theoretically at least, that one type of dynagraph profile is often more desirable in most applications than other types, sometimes even independent of the geometry of the pumping unit.

XI. How Torque Reduction and Smoothing Often Lowers Power Consumption and Demand

If the torque load can be significantly reduced and smoothed out, the prime mover can operate more efficiently and a greater percentage of the input energy will go toward beneficially lifting fluid while resulting in lower thermal losses.

Figure 10 illustrates a study made by James Eickmeier, formerly an engineer with Shell of Canada, relating torque load range to thermal losses as well as with energy conservation.

Starting right-to-left, Figure 10-c' shows a widely fluctuating torque load whose beneficial torque work output is proportional to the area of the rectangle a, b, e, f, and whose total energy input is proportional to the area of the rectangle a, c, d, f.

The difference between the two energy rectangles noted above is a third rectangle, b, c, d, e, whose area is proportional to the thermal losses during the pumping cycle.

It can be seen from Figure 10-c' that the average rms current is proportional to the input energy. The horizontal line below that represents the average current which is proportional to the useful mechanical output work to the pumping unit. This wide torsional fluctuation has a high cyclic load factor equal to 1.50.

Moving to the left, Figure 10-b', the torque fluctuations are considerably reduced and the average rms current is also lowered, while the average current is constant and performing the same beneficial output work shown in Figure 10-c'. It can be seen that the rectangle b, c, d, e, is significantly smaller than the rectangle with widely fluctuating torque, showing that the thermal losses are considerably lower in this example than before, consequently requiring a smaller amount of input energy to develop the same work output. Here the cyclic load factor drops to 1.25.

Moving again to the left where the much more uniform torque loading is occurring, it can be seen that the thermal loss rectangle here is even smaller than that of the preceding Figure 10-b', resulting in a cyclic load factor of 1.05. This means that simply smoothing out the torque by mechanical means, with a much reduced torsional fluctuation, results in more of the input energy being applied to beneficial fluid lift, and less to thermal loss.

The result of this torsional reduction and smoothing makes possible a lesser amount of input energy required to perform a given amount of system work. Also in many cases, the prime mover size is reduced because of peak torque reduction and smoothing. Thus it would seem that the goal of torque reduction and smoothing would be beneficial to the pumping system wherever it is feasible.

Conclusions

1. Use of the full pumping spectrum of dynamometer cards (Figure 1), along with the appropriate permissible load diagrams, can, by inspection, often suggest more beneficial ways of lowering and smoothing the torque load by properly altering pumping speed, stroke length, plunger and rod size. Also, it can help in the selection of optimum pumping unit geometry for a desired application. When rod loading is critical, the full spectrum of pumping, (Figure 1) can also assist in the selection of the best pumping unit geometry and pumping mode.
2. In order to insure highest performance and maximum economy with a beam and rod system, it is of importance to use a modern predictive analysis of several pumping unit geometries for a particular application to see which unit type is best.
3. As regards torque reduction, there is an optimum, preferred, and significantly different, dynamometer card slope for both conventional and Mark II pumping unit geometries.
4. Except for rod stretch and resulting harmonics, all types of beam pumping units driven by a rotating crank system will produce an **overtravel** card.
5. Increased pumping speeds in all applications with low to medium fluid loads tend to rotate the dynamometer card clockwise, which tends to produce an **overtravel**-type card.
6. Minimizing rod stretch often tends to help optimize the beam and rod pumping system performance.
7. Proper selection of pumping unit geometry may help increase the effectiveness of the pumping system.
8. Torsional reduction and smoothing can, in most cases, reduce power costs and electrical demand charges.
9. In a majority of all beam and rod pumping applications, the industry has selected **overtravel** applications as opposed to **undertravel** applications.

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Acknowledgments

Deepest appreciation to those dedicated pioneers of past and present cited at the beginning of this paper, who have given so much to advance our knowledge and understanding of the technology of artificial lift.

Also, sincerest gratitude to: E. G. Pittman, Cecil Hunt, Robert Estes, Shirley Williamson, and for the memory of James Courtney Wright, my partner, teacher and friend.

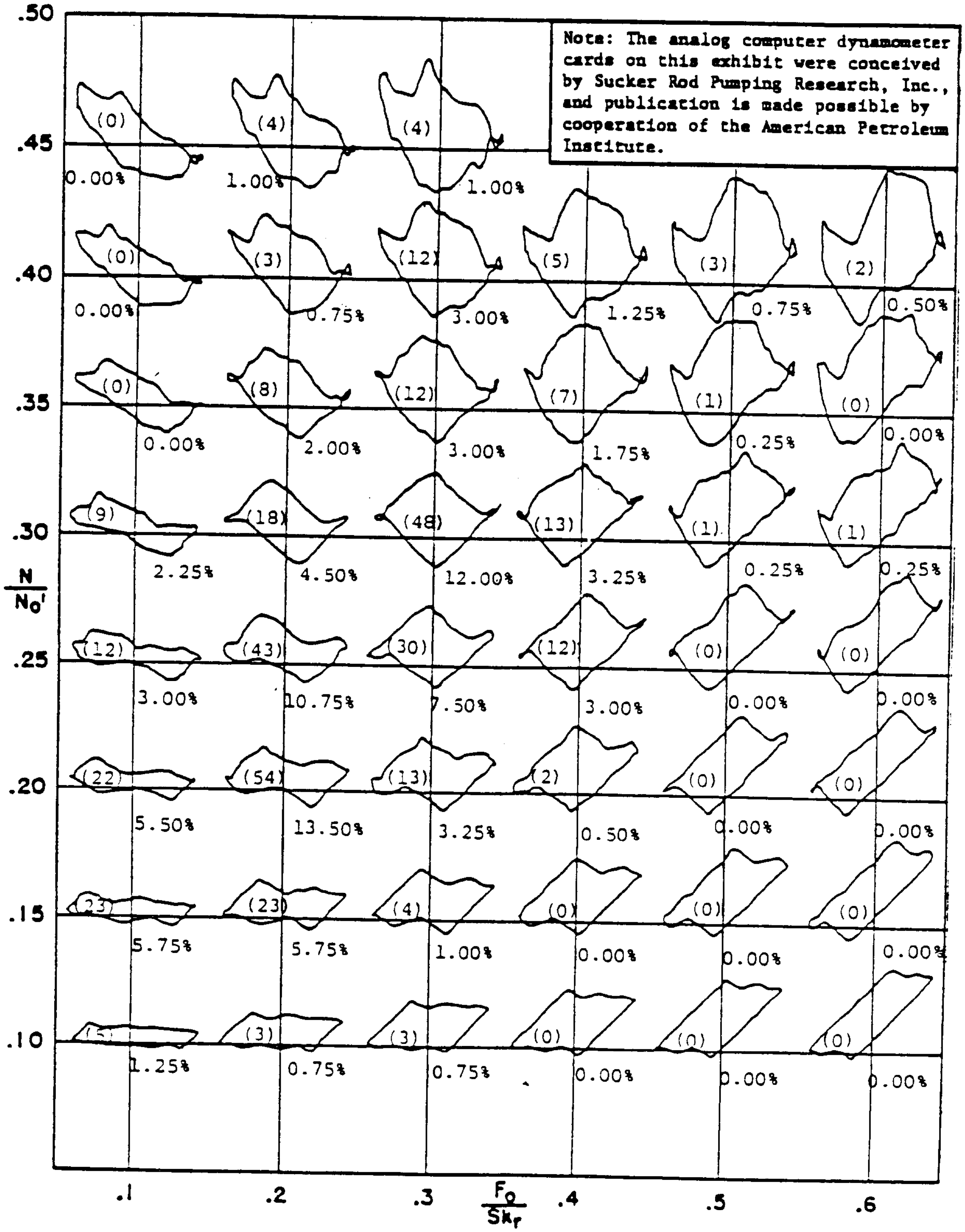


Figure 1 - Statistical Analysis of 400 Dynamometer Cards
(Random Sampling)

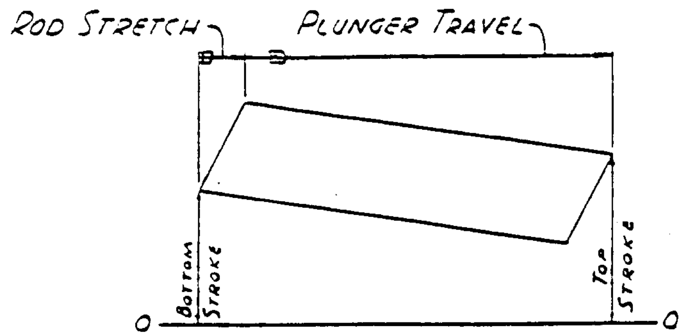


Figure 2a - Typical Overtravel Dynagraph

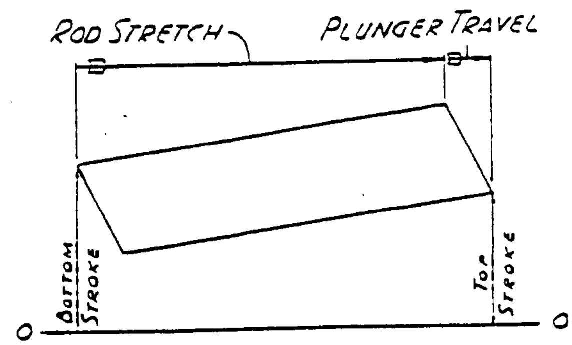


Figure 2b - Typical Undertravel Dynagraph

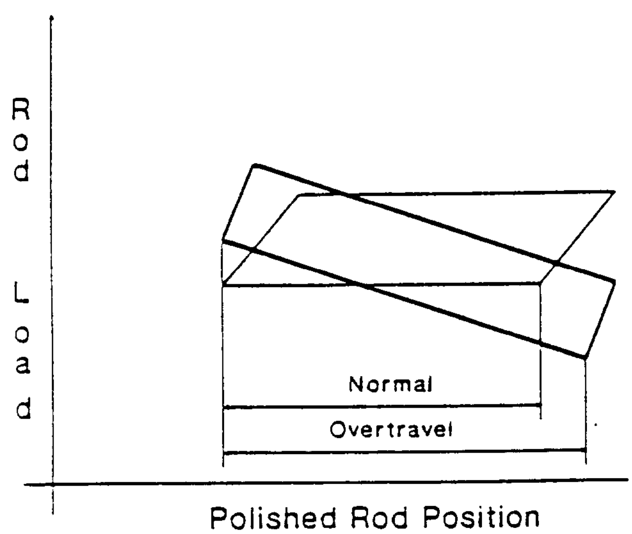


Figure 2c - The general shape of an overtravel card

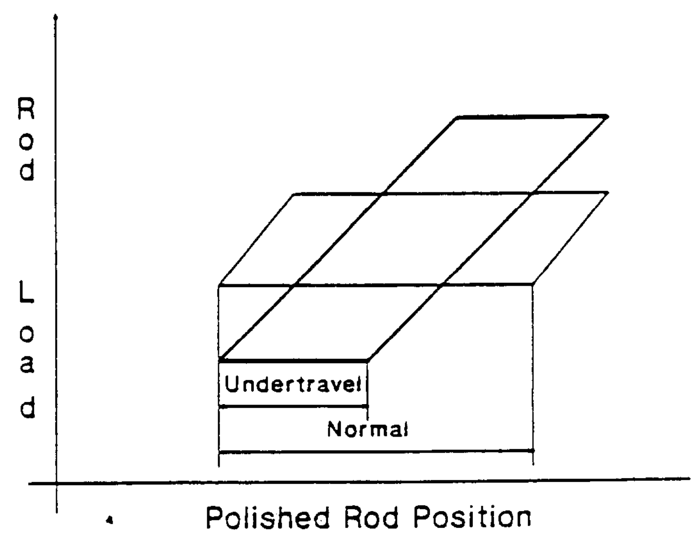


Figure 2d - The general shape of an undertravel card

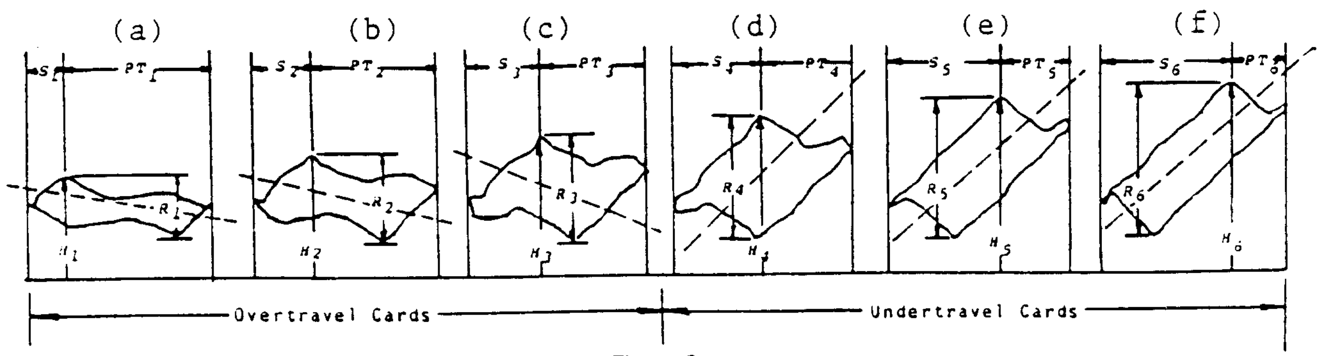


Figure 3

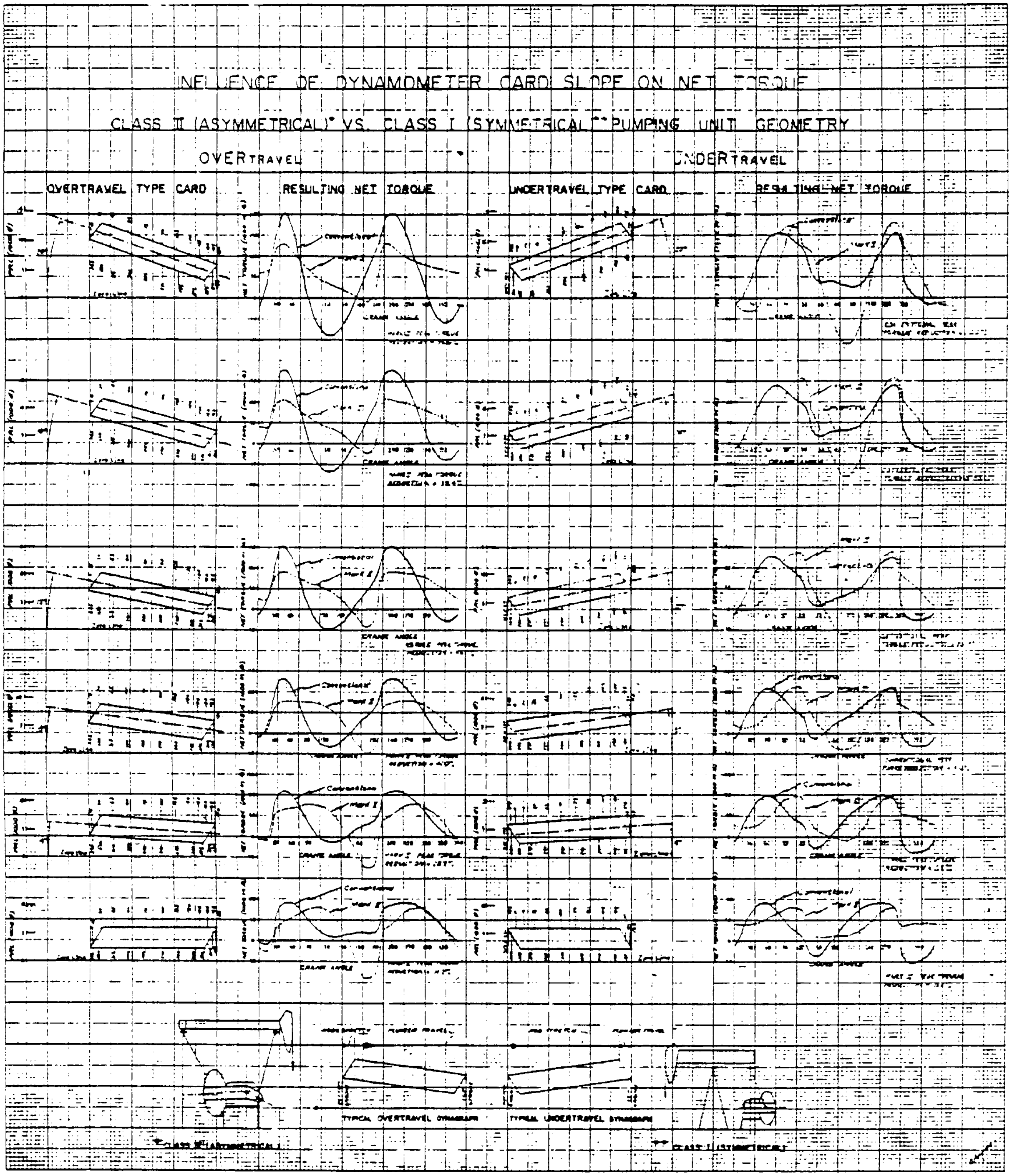


Figure 5 - Influence of Dynamometer Card Slope on Net Torque
Class III (Asymmetrical)* vs. Class I (Symmetrical)** Pumping Unit Geometry

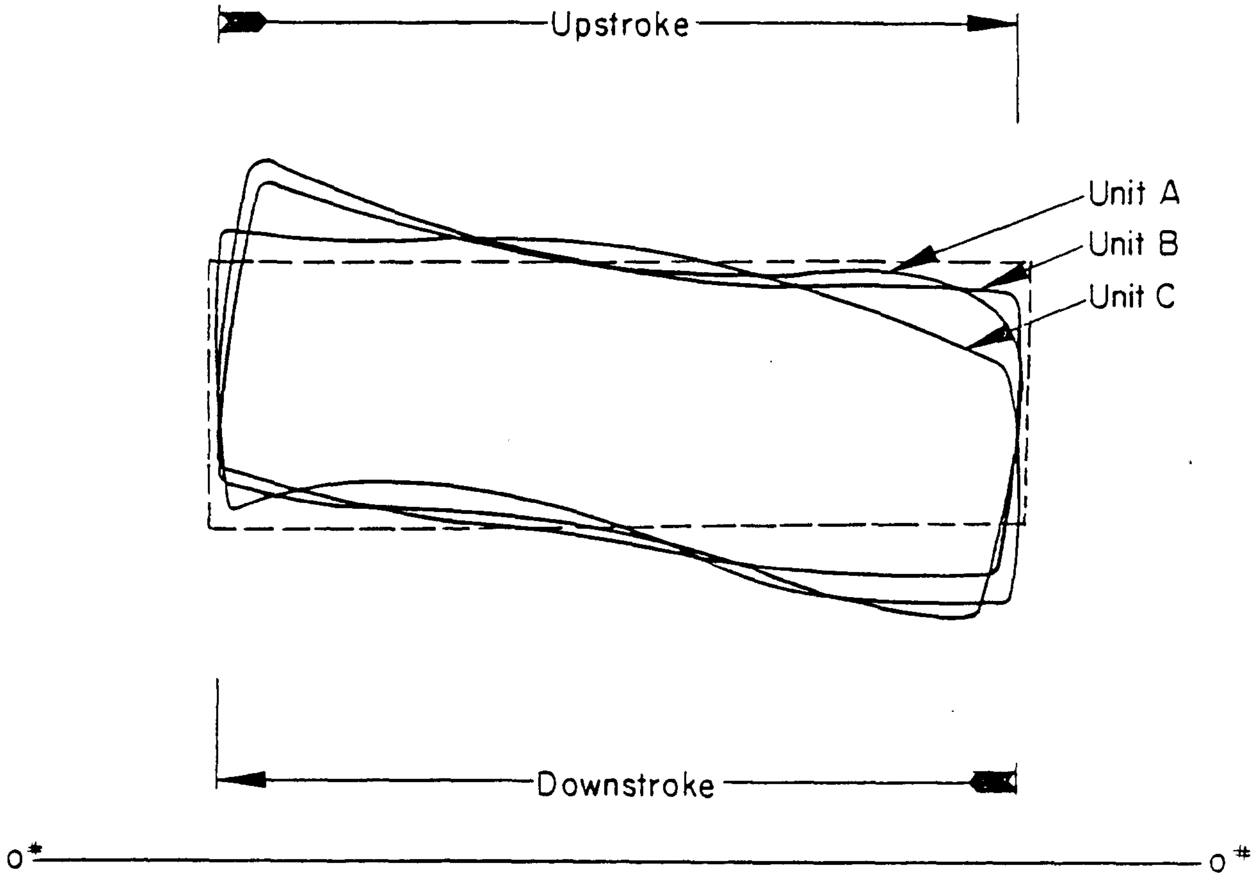


Figure 6

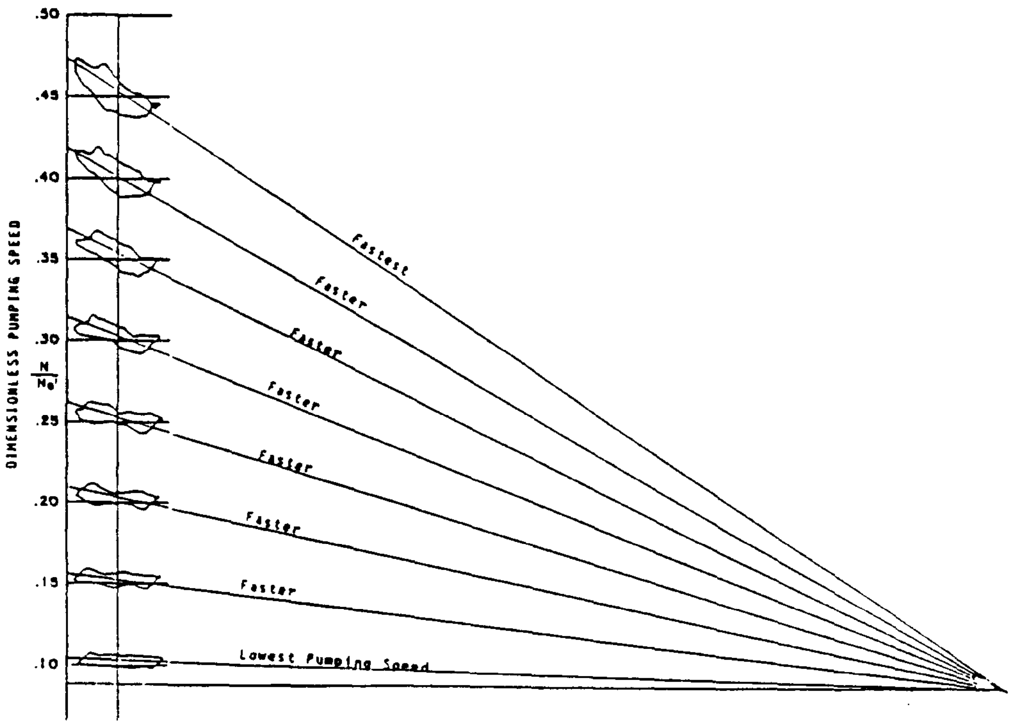


Figure 7 - The Effect of Pumping Speed on Dynamometer Card Slope
Showing Clockwise Rotation—Inertia, Friction, and Harmonic Forces are Included

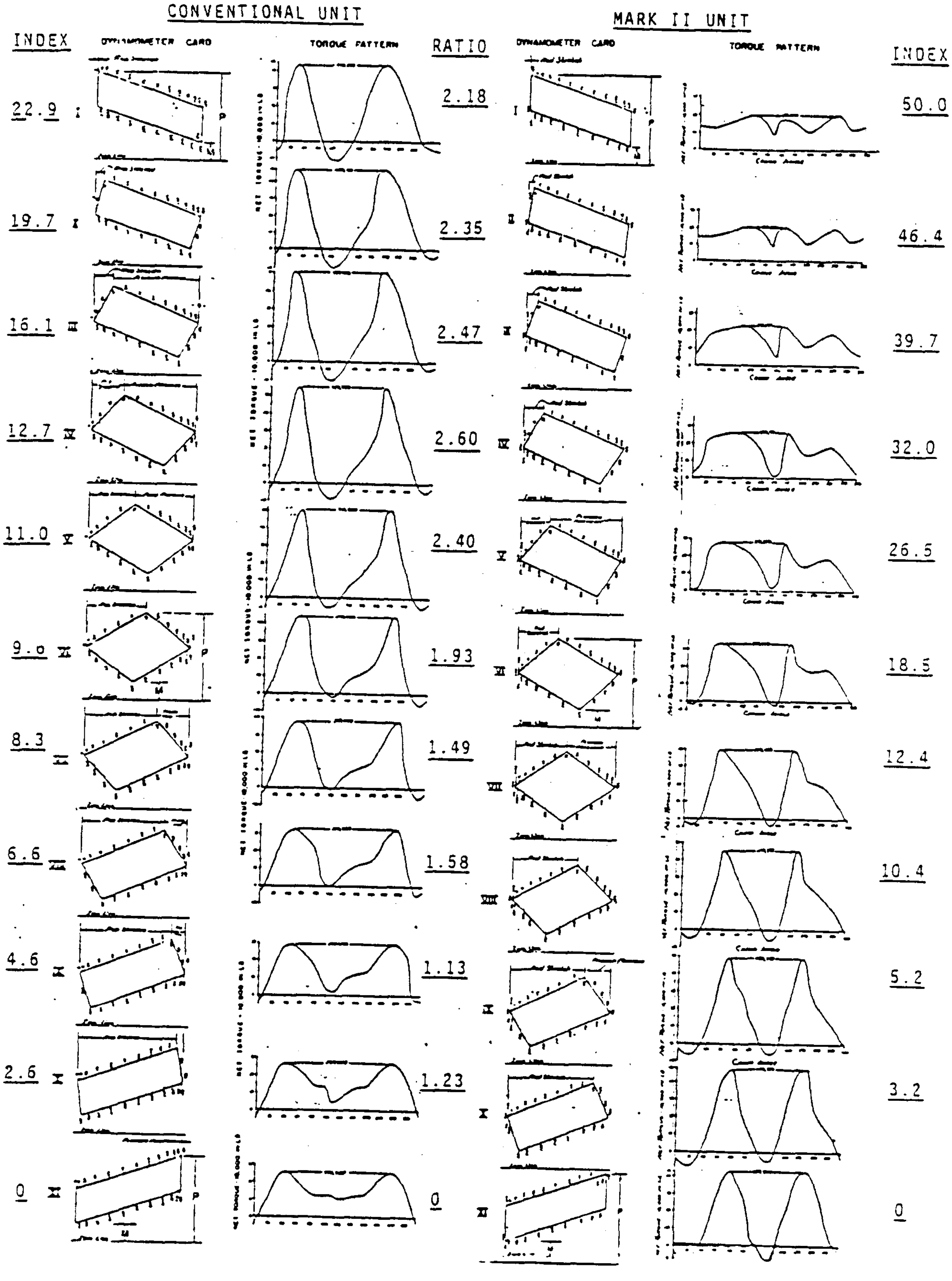

CONVENTIONAL UNIT
MARK II UNIT
INDEX
DYNAMOMETER CARD
TORQUE PATTERN
RATIO
DYNAMOMETER CARD
TORQUE PATTERN
INDEX
22.9
19.7
16.1
12.7
11.0
9.0
8.3
6.6
4.6
2.6
0
2.18
2.35
2.47
2.60
2.40
1.93
1.49
1.58
1.13
1.23
0
50.0
46.4
39.7
32.0
26.5
18.5
12.4
10.4
5.2
3.2
0

Figure 8

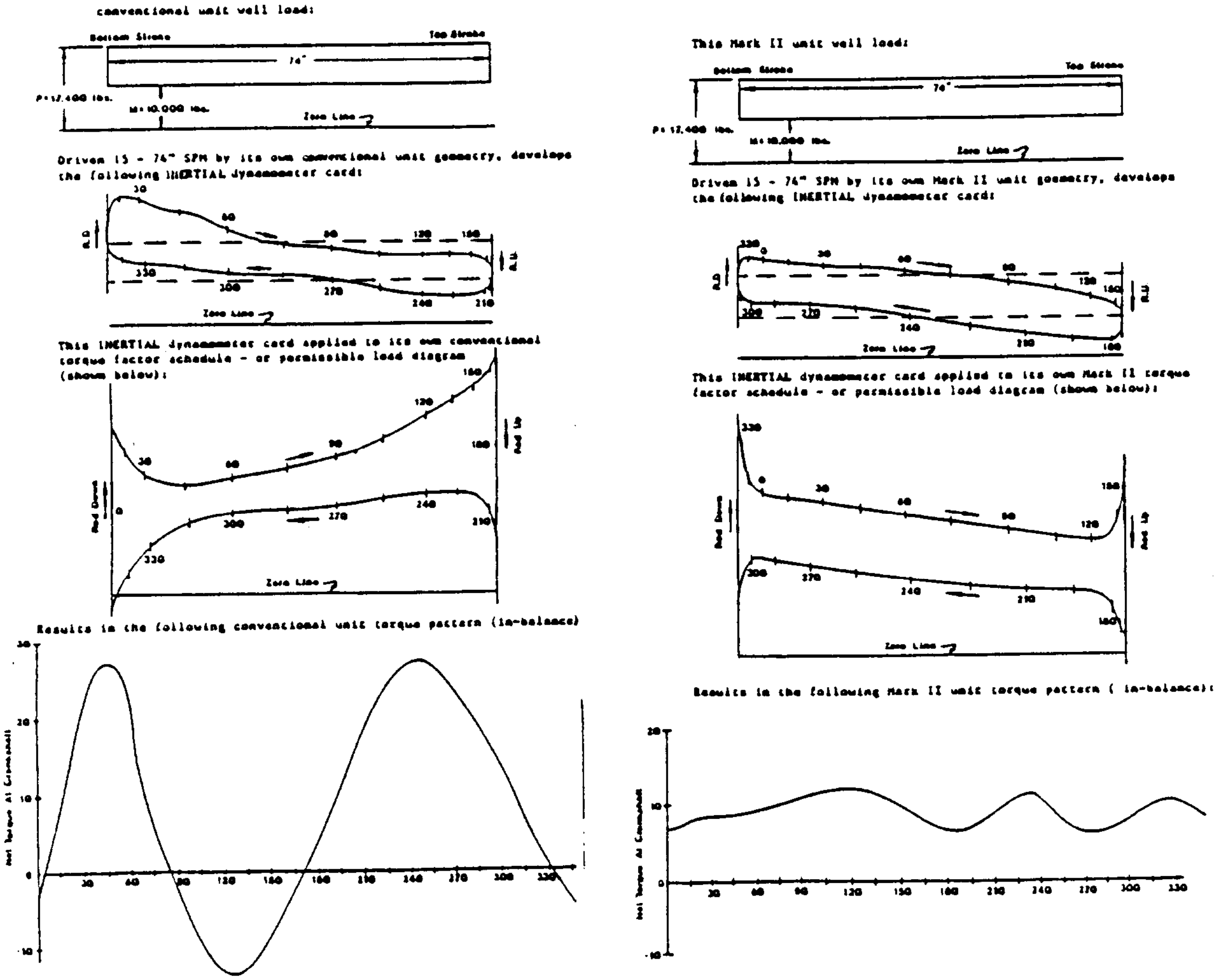
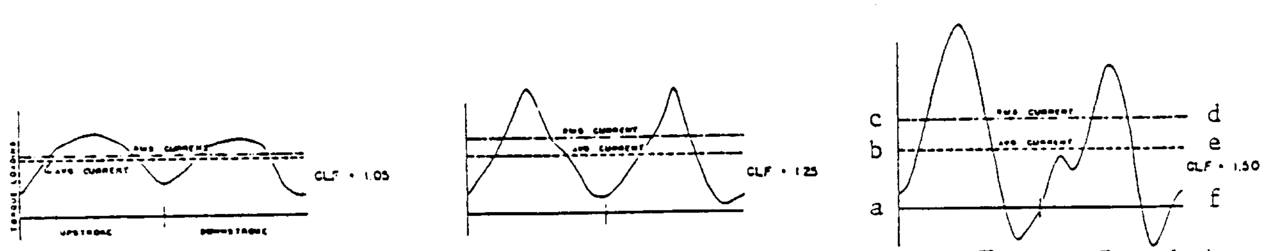


Figure 9



Relatively Uniform Torque Loading
(a')

Nominal Torque Fluctuations
(b')

Widely Fluctuating Torque Load
(c')

Figure 10 - Relationship of the RMS Current to the (Constant) Average Current in a Typical Beam Pumping Application, as Torque Load Range Increases