PERFORMANCE CHARACTERISTICS OF THE MARK II IMPROVED GEOMETRY PUMPING UNIT

An Analysis Of The Mark II Pumping Unit Cycle To Illustrate The Effects Of Improved Geometry Machines On Torque And Power Requirements

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INTRODUCTION

The conventional pumping unit long ago set the standard by which all other beam lift systems are measured. It is one of the most recognizable symbols of the oilfield industry and yet, like most machines, it has limitations with respect to certain applications. During the last half of the 20th century, many other beam pumping systems were introduced targeting load applications in which the conventional geometry encountered these limits. One such unit is the Mark II. Over the decades since its introduction, the Mark II pumping unit has taken a place alongside the conventional gaining wide acceptance and expanding the range of applications where rod pumping may be employed.

The performance of beam pumping units can be measured relative to a variety of criteria. Two of the most important however, are power consumption and gear reducer net torque. It is not surprising to find that these quantities are related or that it is to the operator's advantage from a cost and reliability viewpoint to minimize them. Excessive torque peaks have a detrimental impact on, among other things, the life of gear teeth in pumping unit reducers. Likewise, these peaks are known to result in loss of efficiency - higher thermodynamic losses - during the conversion of energy to polished rod power.

Questions often arise related to which pumping unit suits which well conditions. Linkage modifications such as those resident in "improved geometry" machines like the Mark II pumping unit are often only vaguely understood. What effects, if any, do variations in pumping unit geometry create in torque and power consumption? Also, what range of well load conditions might benefit by these effects?

The focus of this paper will be to analyze the performance effects that geometry differences contained in the Mark II create in relation to a standard conventional pumping unit when considering net torque and power requirements. This will be accomplished by use of basic techniques contained in API Specification 11E and other generally accepted engineering principles. Comparisons between like-sized Mark II and conventional pumping units will be made in the context of a load scenario known to present problems for the latter. A modern rod pumping predictive program was utilized to simulate dynamometer information for both units under identical well conditions. It was also used to provide predictions related to reducer net torque and prime mover power requirements.

PUMPING UNIT LOADS

Figure 1 provides some insight into a common overload scenario for a conventional pumping unit. Simulated dynamometer cards are shown for a 912-305-192 operating at 6, 8, 10, and 12 SPM. The cards are based on the following well conditions: 1.5 inch pump set at 7500 ft, 2-7/8 tubing (anchored), API 76 taper using high strength rods. A positive torque permissible load diagram computed from counterbalance effect of 17,400 pounds is shown along with the dynamometer cards. It indicates the allowable polished rod load for this unit at any given rod position.

At 6 SPM, the pumping unit is operating at nearly full capacity according to the permissible load diagram. Wishing to increase production, the operator raises pumping speed with the unfortunate result of reducer overload. The problem here is not so much the capacity of the gear reducer as it is the timing of the peak and minimum polished rod loads. As is the case with most conventional units, peak polished rod loads that occur early in the up-stroke

stroke or minimum loads that occur early in the down-stroke can present problems as pumping speeds increase. Dynamometer cards that exhibit these early peaks and minimums are often referred to as over-travel cards. The figure illustrates that as pumping speeds increase, over-travel cards develop a shape that slopes downward to the right.

Generally speaking, over-travel cards result in greater down-hole plunger displacement, and therefore greater fluid production per stroke, than do cards in which peak and minimum loads occur later in their respective stroke intervals (under-travel cards). They are therefore considered desirable in a great number of cases. The locations of dips or humps in the permissible load diagram are dictated by the linkage of the particular pumping unit. It can be observed that the permissible load envelope for this conventional unit is more suited to peaks that occur later in the up-stroke or minimums occurring later in the down-stroke.

The operator might still attempt to achieve his new production goal by altering other parameters related to the well. He might, for example move these peaks and minimums into a more acceptable area of the permissible load diagram by increasing the differential load on the pump, perhaps enlarging the pump bore size while holding pumping speed and rod design constant. However, this approach often creates heavier structural loading in the pumping unit and the resulting under-travel dynamometer card will give-up substantially more of its polished rod power to unproductive rod stretch.

An alternative approach would be to change the pumping unit so as to modify the shape of the permissible load diagram and thereby accommodate the over-travel card.

ORIGIN OF THE MARK II UNIT GEOMETRY

Events leading to the creation of the Mark II pumping unit began in 1956. The Oilfield Equipment Corporation of Denver, Colorado initiated a design program intended to identify and develop a beam pumping unit geometry offering the most effective combination of production capability, peak loading, cost and reliability.
Gear reducer size requirements for pumping units were then as they are now dictated by peak net torque demands resulting from the interaction of well load and counterbalance. Dynamometer analyses had shown that peak torque demands on conventional pumping units occurred very often over only a small interval of time. The remainder of time during the pumping stroke, the net torque requirements were relatively small in relation to the gearbox rating. The designers realized that if these peaks could be reduced through modifications to the pumping unit geometry, an equivalent amount of polished rod work and fluid production could be accomplished with a smaller gear reducer and a correspondingly smaller prime mover.

J. P. Byrd, in his 1970 paper titled "The Functional Effectiveness of a Special Class III Lever System Applied to Sucker Rod Pumping" stated:
"To minimize peaks and smooth out the torque pattern for both the reducer and the 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."

Approaches to reducing net torque at the crankshaft revolved around two basic ideas. First, reduce the peak polished rod loads that are input to the pumping unit. The prescribed motion profile of the pumping unit mechanism was known to serve as a forcing function for pump and rod string dynamics. Modifications that could help reduce changes in fluid or rod string momentum for the targeted range of well conditions would be beneficial. The second idea was to alter the mechanical advantage of the pumping unit mechanism so that peak loads, once introduced to the pumping unit, would be converted to torque at a lower rate – particularly early in the upstroke where it was most needed.

CLASS I VS. CLASS III LEVER SYSTEMS

Beam pumping units can generally be divided into either class I or class III lever systems. A class I lever can be characterized as one in which the fulcrum (pivot) is located between the well load and the applied force. A conventional pumping unit is an example of a class I lever. A class III lever on the other hand will have its fulcrum located at one end of the lever, the well load at the other, and the applied force somewhere in between. Air balance and Mark II pumping units are examples of this type of lever system. It is sometimes referred to as a "front mounted" geometry.

Figure 2 displays a hypothetical "2-headed" pumping unit. This is intended to illustrate that a class I and class III geometry differ fundamentally only in regard to which end of the walking beam the horse head is attached. If the well load is attached to the horse-head on the right the resulting system is a class I lever. If the well load is attached to the horse-head on the left a class III system is created. It can be readily observed that the motion of the walking beam will be the same for either case. Furthermore, since polished rod motion is proportional to movement of the walking beam, it can be deduced that the polished rod motion of the well to the left is simply opposite to that of the well to the right.

The polished rod motion profiles of different classes of lever systems owe much of what distinguishes them to one parameter – pitman length. Were the pitman links in either the class I or class III case extremely long in relation to the crank length, the resulting polished rod motion would approach that of simple harmonic motion (SHM). That is, the vertical component of motion created by an object traveling along a circular path (i.e. a sine-wave). However, since extremely long pitman links are not practical from a cost or functional stand point, pumping unit designers have been forced to contend with the effect of more moderately sized members.

As pitman lengths are reduced, the polished rod motion begins to become affected. Most noticeably, acceleration at the top and bottom ends of the pumping stroke begins to deviate from that of SHM.

Consider an example in which simple harmonic motion is used as a reference when comparing motion profiles created by class I and class III linkages. SHM will produce accelerations at the top and bottom of the stroke that are identical in magnitude although opposite in direction. Given equivalent stroke lengths and time intervals in which to complete the cycle, the class I linkage will generate acceleration at the bottom of the stroke that is larger than peak SHM acceleration and smaller than SHM acceleration at the top of the stroke. The class III lever on the other hand will be the reverse with smaller acceleration at the bottom of stroke and larger at the top.

Figure 3 displays acceleration vs. crank angle for a conventional (912-305-192), a Mark II (912-305-192), and a reference curve derived from simple harmonic motion (SHM) to yield the same stroke length over the prescribed time interval. The curves correspond to a pumping speed of 10 SPM. The horizontal axis of the graph indicates crank angle as measured from the bottom dead center polished rod position. Note that zero degrees crank angle as defined in API 11-E does not correspond to the bottom of the stroke for the conventional or Mark II. The curves in this and subsequent graphs are shifted 27 degrees to the right for the Mark II and 3 degrees to the left for the conventional in order to originate from the BDC position.

Polished rod force is at least fundamentally related to the product of mass and acceleration. The mass of the fluid and sucker rod system will be at its maximum during the upstroke so it follows that peak loads will typically occur during this interval. Polished rod load is also significantly modified by the dynamics of the elastic rod elements so that peak loads do not necessarily occur at the beginning of the upstroke. They do occur early in the up-stroke for over-travel dynamometer card shapes however, and the Mark II unit's lower off-bottom acceleration will generally have a favorable reducing effect on peak polished rod load in these cases.

VELOCITY AND TORQUE FACTORS

The task of reducing peak torques was aided by another design alteration. In addition to its front-mounted geometry, the Mark II designers shifted the reducer's crankshaft centerline aft of its walking beam connection. One reason for this modification was to improve the pumping unit's mechanical advantage during the up-stroke. By shifting the reducer so that the up-stroke was accomplished in approximately 195 degrees of crank rotation, the polished rod velocity was slowed and torque factor was lowered, particularly in the early stages of the upstroke. The curves shown in Figure 4 confirm that the polished rod velocity of the Mark II is below that of both the conventional and SHM through the first half of the upstroke. A consequence of this slower up-stroke however, is that the return stroke must be covered in only 165 degrees of crank rotation. The polished rod velocity and torque factor through that part of the cycle is therefore increased.

Torque factor is a way of numerically stating the effectiveness of the machine in converting polished rod load into crank shaft torque. It is a ratio between pumping unit well load and the resulting crank shaft torque for a particular beam pumping unit linkage at a specified crank angle. Given an assumption that the crank is rotating at constant speed through the entire pumping cycle, it can be proven that instantaneous polished rod velocity and torque factor are proportional. That is, where the polished rod is moving slowly such as at the top or bottom of the stroke, the torque factor will be low in magnitude meaning a load can be lifted with relatively little torque supplied by the reducer. However, if the polished rod is moving rapidly such as at mid-stroke, the torque factor will be higher and more reducer torque will be required to lift the same load. Figure 5 displays the torque factor vs. crank angle relationship for Mark II and conventional 912-305-192 geometries. Note the similarity in shape of these curves to the polished rod velocity curves in Figure 4.

The combined effect of the front mounted geometry and the rearward shift of the reducer on the Mark II was to reduce torque factor, polished rod velocity and acceleration during the initial portion of the stroke relative to its conventional counterpart.

NET TORQUE

Net torque on the gear reducer is ultimately given by the interaction of two primary torque components. The first, of course is the torque created by the well load and structural unbalance that is then transformed through the pumping unit mechanism to the crank shaft. The second is counterbalance torque. This is simply the torque on the crankshaft that is produced as a result of gravitational forces acting on the cranks and counterweights as they rotate through the cycle. Figure 6 displays curves intended to illustrate the interaction of well and counterbalance torques at the crank shaft. They correspond to a conventional 912-305-192 pumping unit operating clockwise at 10 SPM with 17400 lb. effective counterbalance. The load information comes from the simulated dynamometer data shown in Figure 1. The well torque is comprised of the difference of well load and structural unbalance which is then multiplied by the torque factor according to the following relation:

$$torque_{well} = \left(load_{well} - unbalance_{structural}\right) \times factor_{torque}$$

The general form of the counterbalance torque calculation is as follows:

$$torque_{counterbalance} = torque_{\max\_counterbalance} \times \sin\left(angle_{crank} - angle_{phase}\right)$$

Where: $angle_{phase} = 0$ for a standard conventional unit.

Net torque then, is simply the difference in well and counterbalance torque.

$$torque_{net} = torque_{well} - torque_{counterbalance}$$

Smaller effects related to structural inertia are also present but this is very nearly the torque that will be “felt” by the reducer at its output.

Admittedly, the over-travel dynamometer card used in this example is not well suited for a conventional geometry, particularly at this pumping speed. However, it is useful in illustrating how certain torque overloads can occur. First, note that the peak net torques for the up and down strokes are nearly equal indicating that the unit is in balance. Next, the peak well load torque occurs at approximately 60 degrees from BDC. This is approximately 25 degrees in advance of the peak counterbalance torque. The mismatch in timing results in a high peak on the net torque curve followed very closely by a rather deep excursion into negative torque prior to reaching the top of the stroke. The peak net torque occurring on the down-stroke actually occurs as a result of excessive counterbalance. This is brought about by the need to offset the peak well torque during the up-stroke. It is clear that a reduction of the peak up-stroke well torque would reduce the necessary counterbalance torque and reduce the net torque peaks on both halves of the cycle. A somewhat similar improvement would result if the negative well load torque were actually larger in magnitude during the down-stroke.

PHASED COUNTERBALANCE

Crank mounted counterbalance torque obeys a sinusoidal relationship with crank angle. It will provide positive torque for exactly half of a revolution and negative torque for the other half. This is convenient for a conventional pumping unit since it accomplishes close to half of its stroke in 180 degrees of crank rotation. What about the Mark II? Shifting the reducer rearward had two impacts on counterbalance. First, the unequal spans of the up and down strokes might at first seem to create problems matching up with the exactly equal half-periods of counterbalance torque. Second, the top dead center and bottom dead center polished rod positions do not occur with the cranks near a vertical orientation. These issues were addressed by the use of phased counterbalance.

One consequence of the class 3 geometry is that the crank counterweights must be located opposite the crank pins with respect to the reducer crank shaft. This is due to the fact that the load transferred by the pitmans and the gravitational forces acting on the counterweights are both primarily downward in nature. It follows that in order for the counterweight force to be useful as well load counterbalance its location must be such that a torque opposite that of the well load will be produced. This gives rise to the counterweight mounting beam that is a part of the Mark II crank.

Phased counterbalance was accomplished on the Mark II by simply creating a bend in the crank arm. The phase angle was selected by review of the pumping unit torque factor curves. At BDC and TDC polished rod position the torque factor is equal to zero meaning torque is not being produced as a result of well load. Counterbalance torque is not required at zero torque factor locations but the unequal up and down stroke of the unit geometry made an exact match impossible. Rather, the phase angle was chosen so that the counterbalance curve nested equally inside the well torque curve on the up-stroke and outside it on the down-stroke. Therefore the well torque leads the counterbalance torque at stroke bottom by 7.5 degrees and lags it by the same amount at stroke top. This has the effect of introducing a slightly positive bias to the net torque curve near the top and bottom ends of the stroke.

Figures 7 & 8 illustrate the cumulative effect of the geometry differences resident in the Mark II. Figure 7 compares the simulated dynamometer card generated by a Mark II 912-305-192 pumping unit operating under identical conditions to that of the conventional unit in the previous section. Figure 8 displays the net torque curve derivation for the unit under this load scenario.

The peak and minimum polished rod loads encountered in this dynamometer card are both lower by an appreciable amount than those of the conventional unit. The peak polished rod load shown here is 24576 lb compared to 25894 lb for the conventional. Minimum loads are 2974 lb. and 4472 lb. for the Mark II and conventional respectively. The lower peak polished rod load for the Mark II can be largely attributed to the lower acceleration and velocity provided early in the up-stroke by its offset class III geometry. Conversely, the higher acceleration and velocity coming off-top contribute to the lower minimum load as well. It should be noted that the total load range encountered by the Mark II is slightly (180 lb.) more than that of the conventional unit. However, the 33% lower minimum load provides a significant increase in the allowable rod load range according to the Modified Goodman Diagram.

In addition to the reduction in polished rod load, the offset class III linkage further conditions this load as it is transformed into crankshaft torque. The lower torque factor through the first half of the up-stroke serves to proportionally reduce the well torque through that region. On the other hand, the higher torque factor through the beginning stages of the down-stroke magnifies the well torque in that area. Note that while the peak load on the Mark II dynamometer actually occurs prior to that shown for the conventional, the peak well torque does not occur until significantly later. The torque suppression characteristics early in the up-stroke tend to push well torque peaks over to the right providing better alignment with the peak counterbalance torque. Comparing this graph with the well torque curve previously shown for the conventional unit, the peak positive well torque is nearly equal for both units, although it occurs later by approximately 30 degrees of crank rotation for the Mark II. At the same time, the magnitude of the peak negative well torque is 34% larger. The result of all this is a well torque curve that more closely matches the magnitude, shape, and timing of the counterbalance curve. This allows the counterbalance to more readily cancel the majority of the well torque. Finally, the asymmetry of the up and down-stroke provides a positive net bias so that excursions into the negative net torque regime are relatively shallower. The final net torque curve for the Mark II is substantially more uniform having both lower peaks and higher minimums. Peak net torque is lower by approximately 36% in this case.

PERMISSIBLE LOAD DIAGRAM

The positive torque permissible load diagram for the Mark II 912-305-168 is shown in Figure 9. It is modified to include the structural load limit in addition to the reducer torque limit curves. A series of simulated dynamometer cards are also displayed for pumping speeds of 6, 8, 10, and 12 strokes per minute. The effective counterbalance was adjusted downward to 16460 lb for balancing purposes relative to the 10 SPM card. This is reflected in the previous graphs as well.

The effect of the offset class III geometry is readily apparent in this permissible load diagram. The dips and humps of the reducer limits occur much later in both the up and down strokes providing better accommodation for the over-travel shaped dynamometer cards. The diagram indicates that this pumping unit can operate safely up to and including 12 strokes per minute, with a slight counterbalance adjustment.

DIRECTION OF ROTATION

A consequence of the offset geometry is that the Mark II can operate effectively in only one direction of rotation – counterclockwise (viewed with the wellhead to the right). The same modifications that provide the improvements in mechanical advantage when rotated properly would in turn make it substantially weaker if operated in the opposite direction. Figure 10 provides illustration for this point.

The permissible load envelope, defined by L. Teel in his 1991 paper titled "*Permissible Load Envelopes for Beam Pumping Units,"* contains not only limit curves for positive reducer torque but negative as well. Both are shown here plotted against crank angle. The dynamometer curve generated by counterclockwise rotation easily avoids both sets of reducer limit curves. However, note that the dip in the negative torque limit for the down-stroke (195-360 deg.) extends much lower than does the corresponding dip during the upstroke. Were the direction of rotation reversed, such that the upstroke now occurred during this interval, the load curve would most likely exceed this limit.

POWER CONSUMPTION

It is not difficult to imagine that a pumping unit reducer operating under a lower net torque requirement would in turn need less power from the prime mover. There are virtually no differences between the Mark II and conventional in the string of mechanisms that transfer power from the prime mover all the way to the crank shaft. The discussion up to this point has circled around basic torque factor techniques for dynamometer analysis. It has neglected the dynamic effects of speed variation and of inertia whether from the pumping unit structure, cranks and counterweights, or the gear train and belt components. Fortunately, there are modern rod pumping predictive programs that do incorporate these effects. The remainder of the data presented in this paper was computed using the SROD program.

Table 1 provides data related to predicted reducer and prime mover performance for the pumping units and well conditions described in the previous sections. Note that the net reducer torque values are slightly lower than those shown in the previous figures owing to the effects of speed variation and inertia.

Regenerative power and cyclic load factor are both related to the net torque of the pumping unit. Regenerative power happens when the reducer net torque falls below zero. This negative torque will cause the direction of gear tooth loading to reverse and attempt to drive the prime mover above its synchronous speed thereby converting it into a generator. The associated load reversal in the gear teeth will typically be signaled by an audible "clunk" at the beginning and end of the negative torque excursion. Regenerative power that is restored to the utility line will have encountered thermodynamic losses related to two energy conversions rather than just one. The first loss occurs when electrical energy is converted to mechanical power and the second when it is converted back. It is therefore more efficient to minimize the amount of regenerative power being handled during the cycle.

Given a requirement for a certain amount of net work per stroke, a pumping unit that encounters large regenerative or negative work intervals will need to compensate with even higher magnitude positive work for the remainder of the cycle. This typically means higher torque swings and results in a larger cyclic load factor. The amount of electric current used by the motor is proportional to the torque that it must overcome. Larger torque peaks therefore mean larger current use. Electrical power consumption is affected by the square of current draw so it follows that high torque peaks consume power at a much higher rate.

The data from the predictive analysis indicates that the Mark II geometry helps to eliminate a large portion of the regenerative work encountered during the cycle. As a result, the positive work performed can be accomplished at a lesser, more uniform torque load. This is reflected in the lower cyclic load factor and ultimately contributes to the lower prime mover power requirement. The power required to operate the Mark II is 34% less than the conventional in this case.

SUMMARY

The peak loads in a large category of pumping unit dynamometer readings called over-travel cards occur relatively early in the up and down stroke intervals. The production efficiency associated with this type of card makes it desirable in a great number of cases. Class I lever systems like the conventional pumping unit tend to encounter the limitations of their gear reducer more often with this type of card than they do with under-travel cards.

The Mark II design was developed to more easily fit the loading scenario presented by the over-travel card. Its offset class III geometry provides lower off-bottom acceleration, torque suppression during the early stages of the up-stroke, and torque amplification early in the down-stroke. This along with its phased counterbalance and asymmetric stroke intervals yields a net torque curve that is more uniform with lower peaks and higher minimums than its conventional counterpart.

When used in its appropriate over-travel application, the power requirements necessary to operate a Mark II unit are significantly below those of the conventional unit owing to its lower, more uniform torque loading.

REFERENCES

L. Teel: Permissible Load Envelopes for Beam Pumping Units, Southwest Petroleum Short Course, 1991

J. P. Byrd: The Functional Effectiveness of a Special Class III Lever System Applied to Sucker Rod Pumping, Southwest Petroleum Short Course,1970

Table 1
Comparison of Power Requirements

PERFORMANCE DATA	MARK II	CONVENTIONAL
MAXIMUM REDUCER NET TORQUE (IN-LB)	640,000	1,015,000
PERCENT OF REDUCER RATING	70.2	111.3
PEAK REGENERATIVE POWER (HP)	-20.2	-59.3
CYCLIC LOAD FACTOR	1.2	1.9
POWER REQUIRED (HP)	58.2	88.4

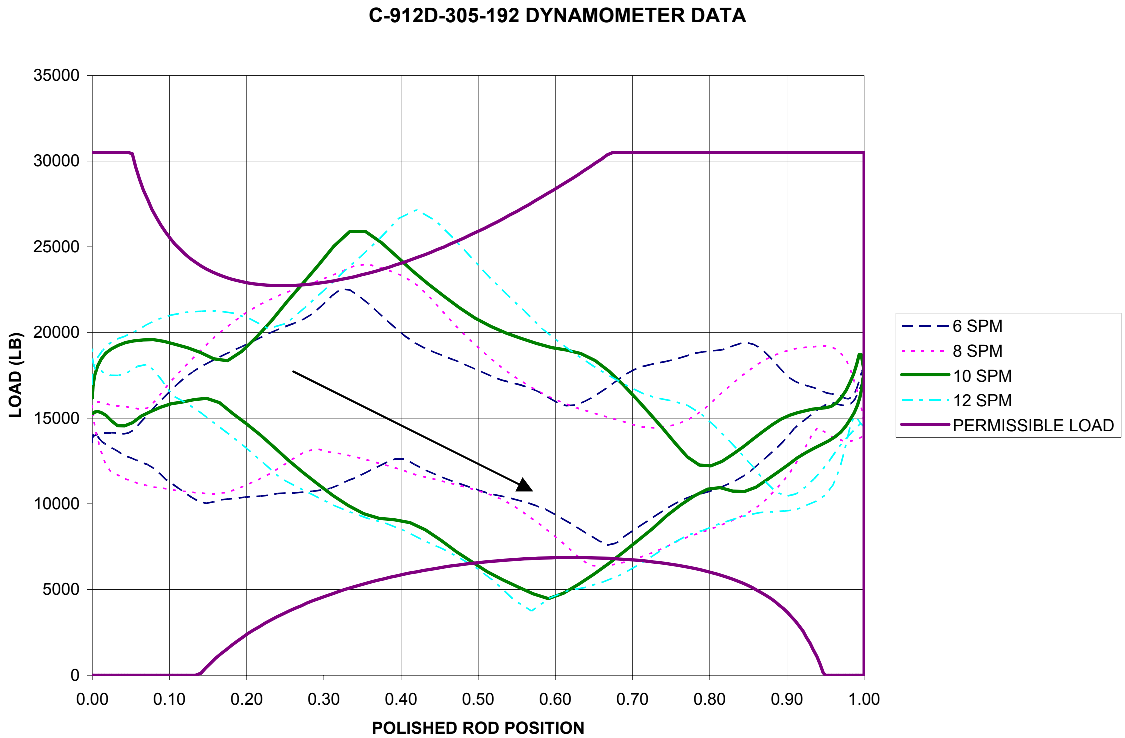

Figure 1 - Conventional (912-305-192) Dynamometer Cards and Permissible Load Diagram

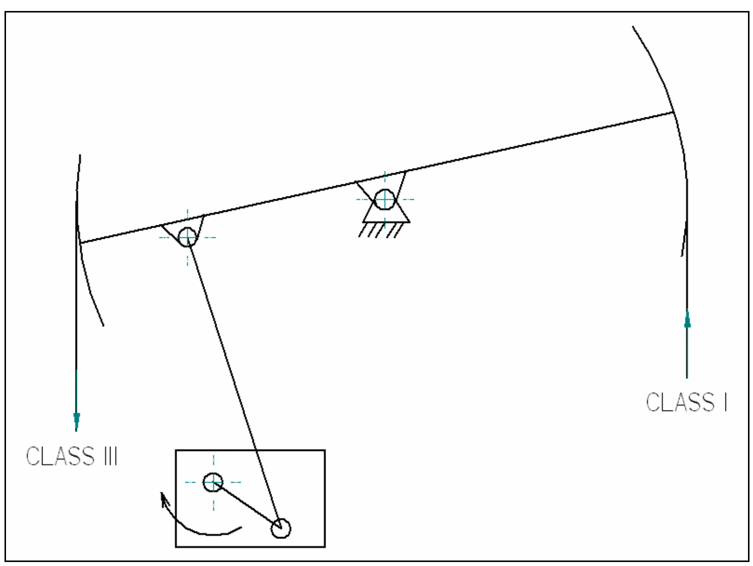

Figure 2 - Class I vs. Class III Lever

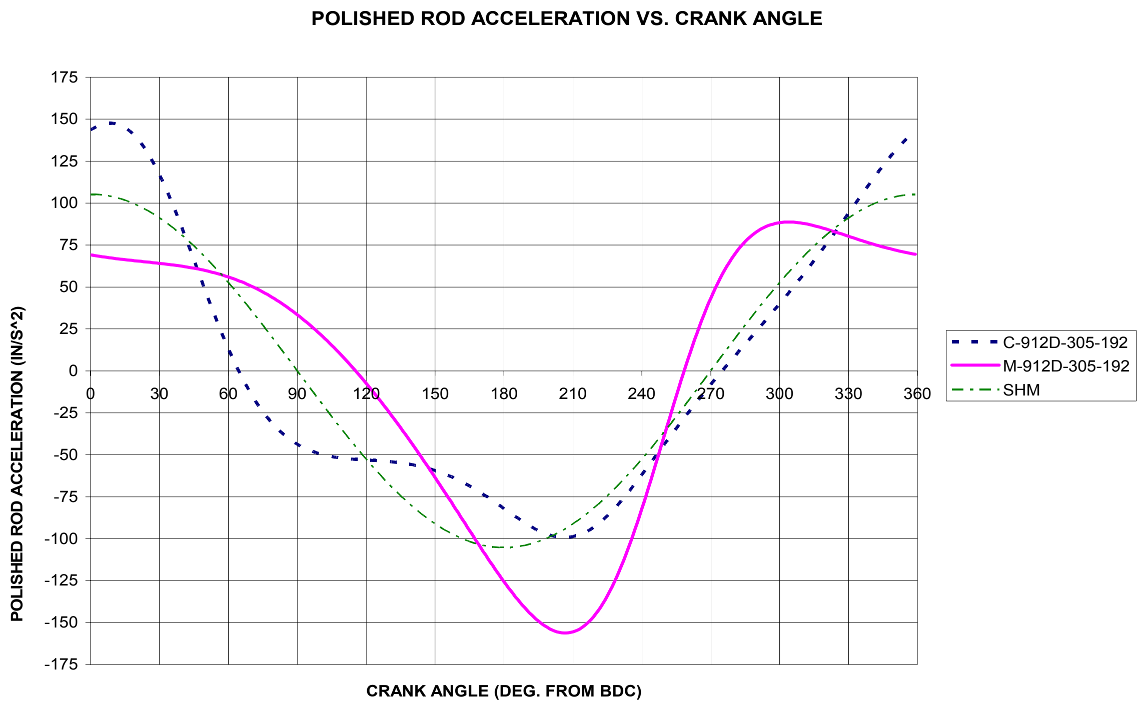

Figure 3 - Comparison of Polished Rod Acceleration for Conventional, Mark II, and Simple Harmonic Motion at 10 spm

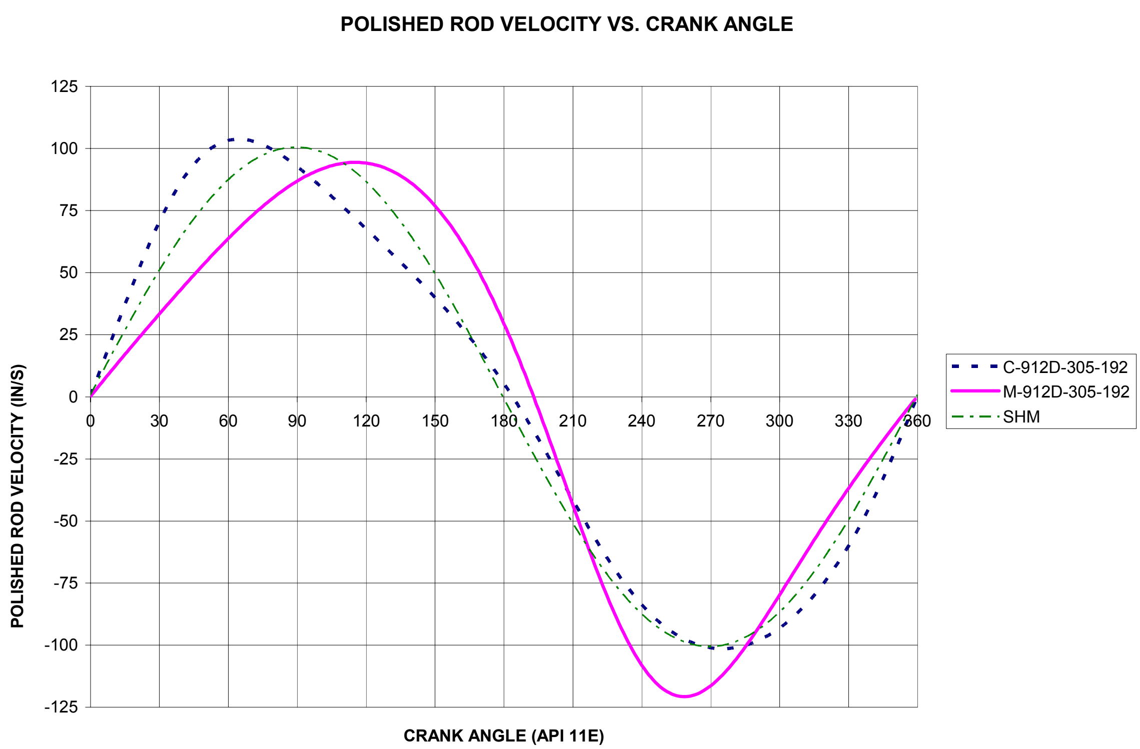

Figure 4 - Comparison of Polished Rod Velocity for Conventional, Mark II, and Simple Harmonic Motion at 10 spm

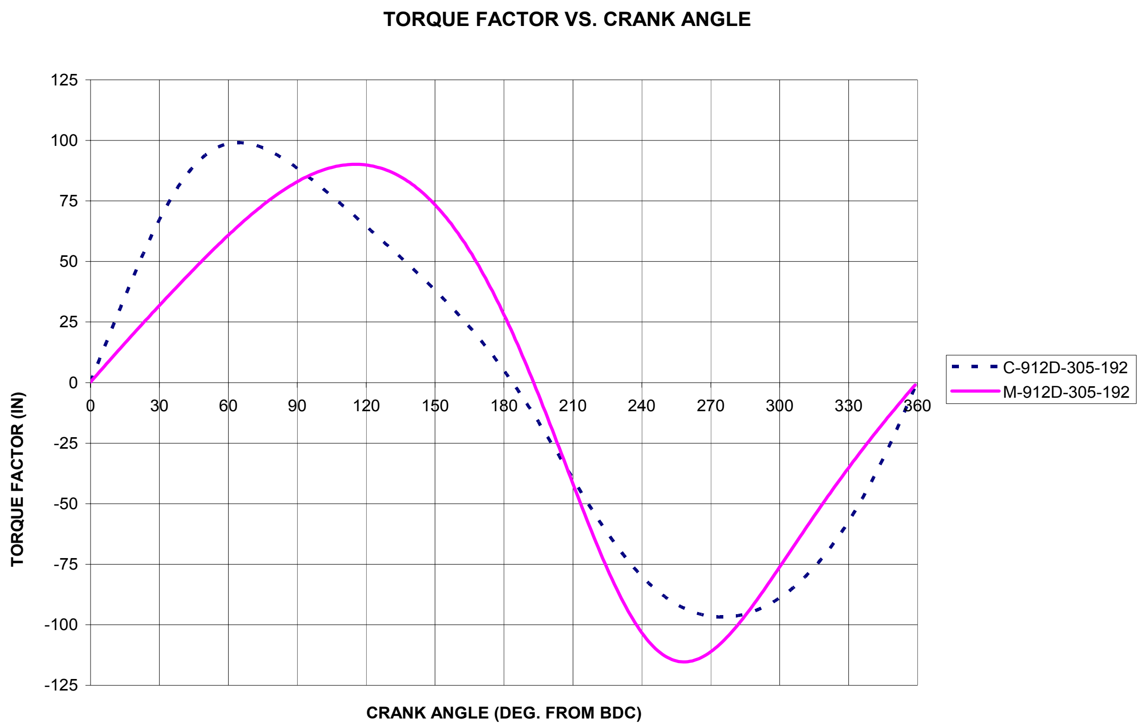

Figure 5 - Comparison of Torque Factors for Conventional and Mark II Pumping Units

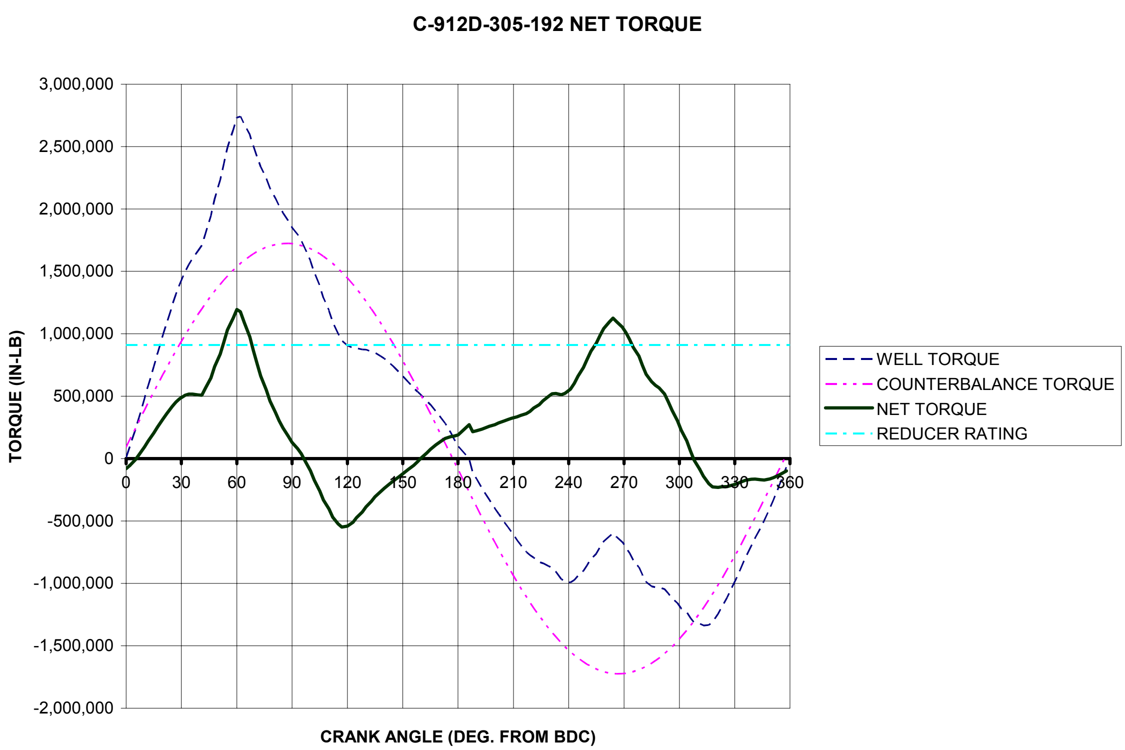

Figure 6 - Net Torque Derivation for Conventional Unit at 10 spm

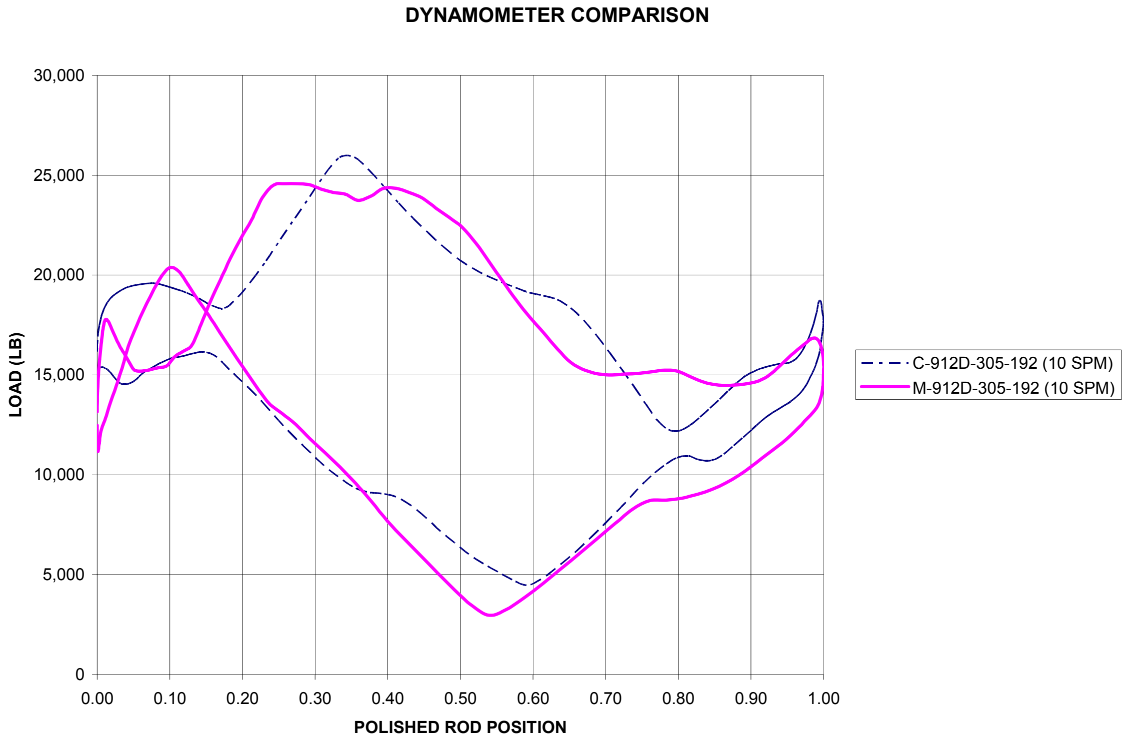

Figure 7 - Comparison of Dynamometer Cards (10 spm) for Conventional and Mark II Units

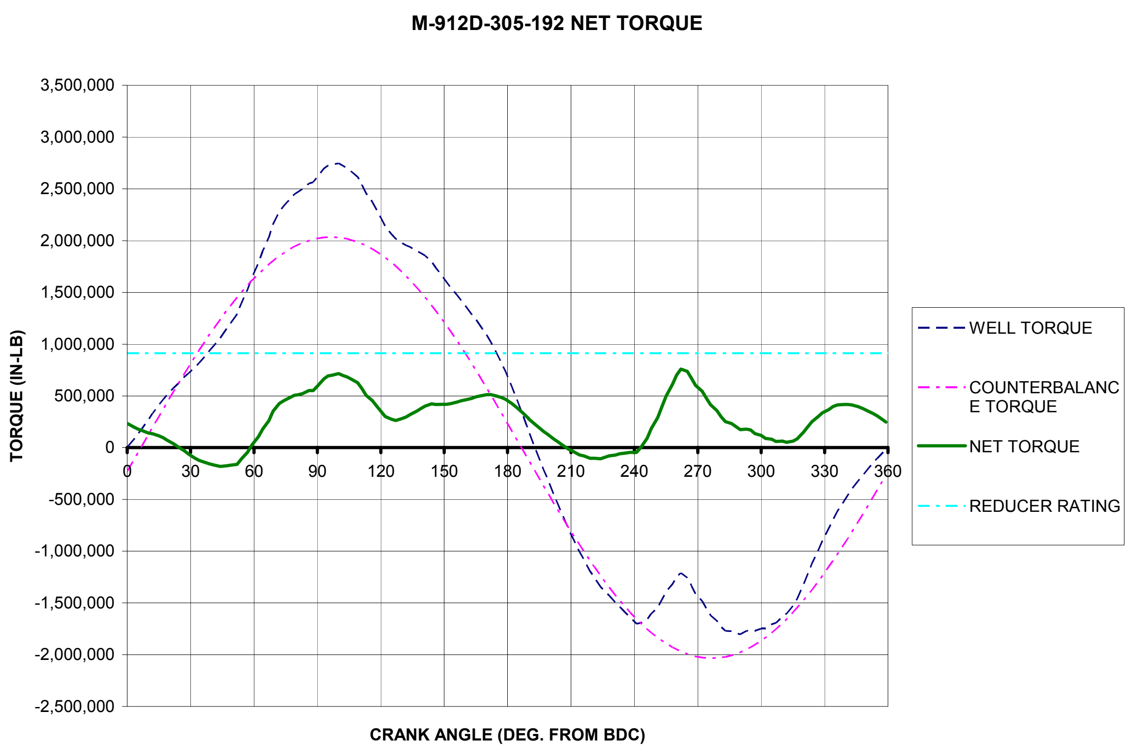

Figure 8 - Net Torque Derivation for Mark II at 10 spm

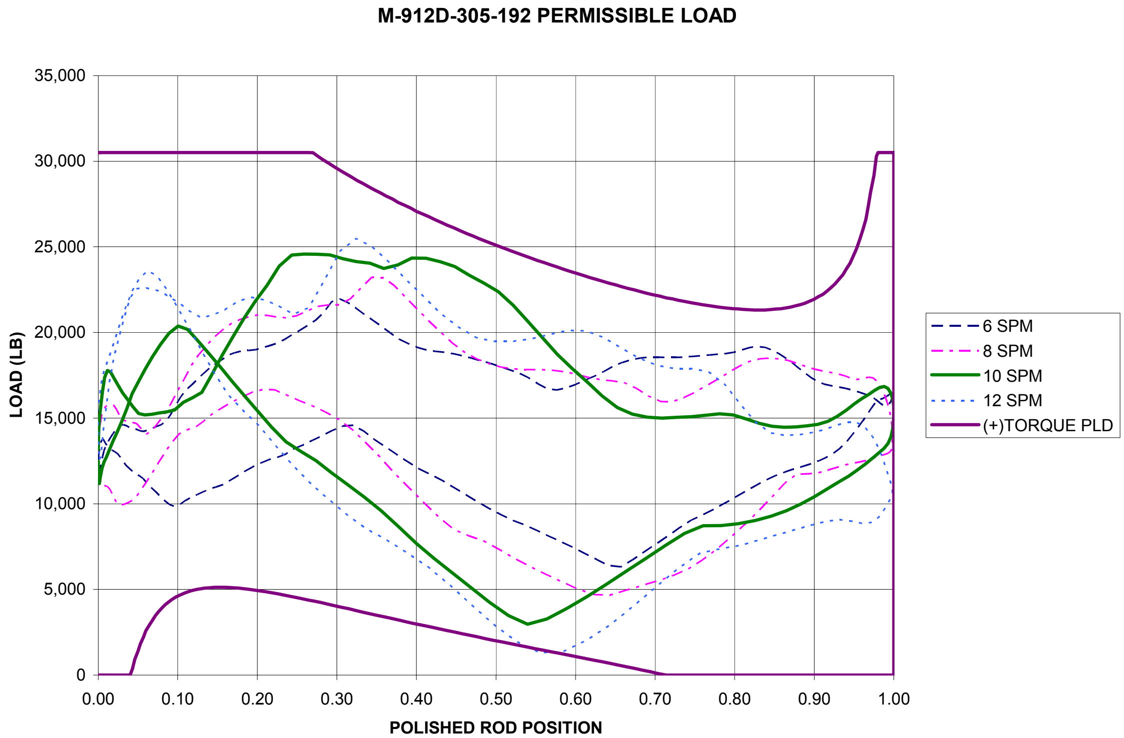

Figure 9 - Mark II Dynamometer Cards at Different Speeds With Permissible Load Diagram

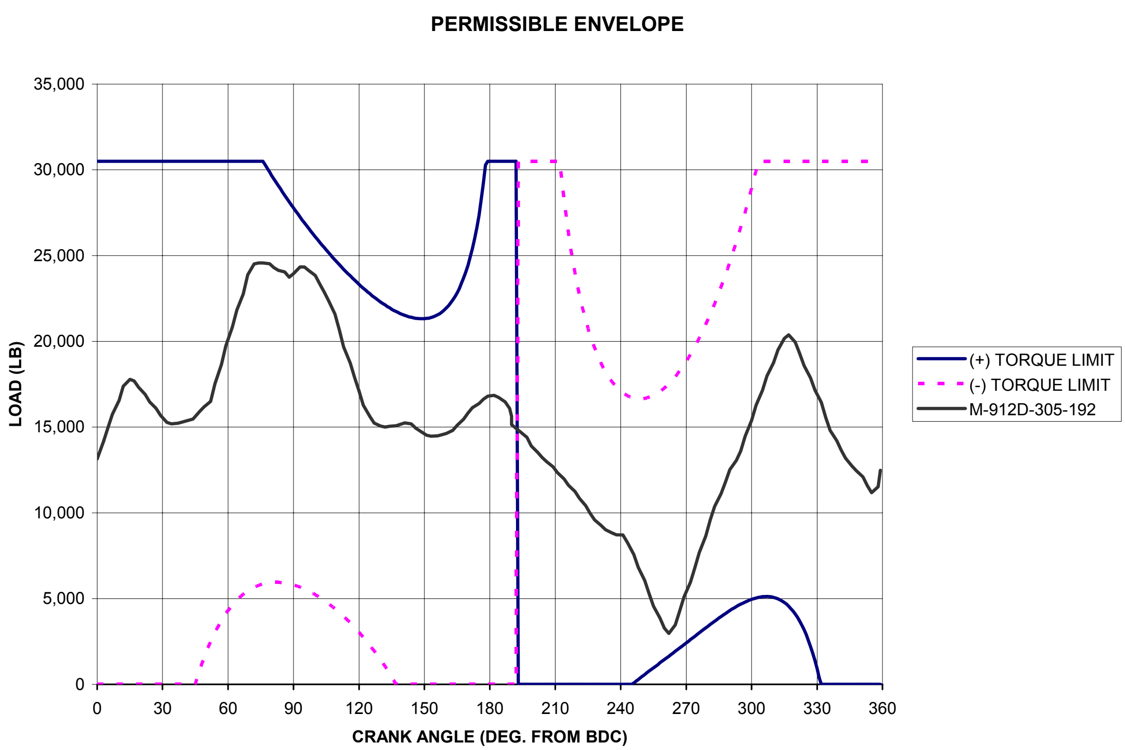

Figure 10 - Permissible Load Envelope vs. Crank Angle