TORQUE ANALYSIS OF PUMPING UNITS USING DYNAMOMETER CARDS

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ABSTRACT

A computer program has been developed which permits torque calculations in a much simpler and much accurate manner than previously presented in API publications. This is accomplished by using a highly accurate digitizing technique to input the necessary number of points from the dynamometer card. The proposed technique improves the accuracy of the data over manual input and allows a much greater number of points to be used for subsequent analysis. The program can handle all pumping unit geometries and calculates all the parameters for a complete analysis of the pumping unit: PRHP, PPRL, MPRL, PT, etc. The instantaneous torques are plotted vs. crank angle. The program also evaluates the counterbalancing of the unit by calculating the Cyclic Load Factor for actual conditions.

The author proposes a new technique to find the maximum counterbalance moment needed for ideal counterbalancing. The theoretically sound procedure seeks that counterbalance moment which results in the least value of Cyclic Load Factor for the given conditions. This approach ensures the minimum of prime mover power requirements, and can reduce the power consumption of existing installations, thus improving the economy of sucker rod pumping.

INTRODUCTION

Analyzing the performance of sucker rod pumping units involves the determination of gearbox torques on the slow-speed shaft during a complete pumping cycle. These torques are generally calculated by the use of the API Torque Factor method [1] using a dynamometer card taken on the given well. The API procedure gives accurate results except only for ultra high slip motors, that is why its application is widely accepted. In field practice, however, it has serious drawbacks which are due to the way data points are taken from the dynamometer card. Namely, the graphical solution proposed in API Std 11E involves hand constructions that are tedious, time consuming and may contain inherent data errors.

One aim of this paper is to present a calculation procedure for gearbox torque calculations which uses a faster and more accurate data acquisition method to obtain the necessary input data from dynamometer cards. The proposed technique involves the use of a plotter, used in digitizing mode, connected to a personal computer. The computer program developed allows direct input into the computer of a user-specified number of points from the card for subsequent calculations. This solution reduces the magnitude of input errors, as compared to manual calculations, and simultaneously improves the accuracy and reliability of gearbox torque calculations.

Knowledge of the variation of gearbox torques vs. crank angle enables the operator to analyze the effectiveness of counterbalancing the pumping unit and to decide on required modifications in order to arrive at optimum conditions. The paper proposes a new technique to find optimum counterbalancing which aims to minimize the value of cyclic load factor. This approach is based on more sound theory than previous methods as the objective of counterbalancing of pumping units lies in reducing the cyclic nature of gearbox and motor loading. An advantage of the proposed way to find optimum counterbalance conditions is that its use automatically results in the minimum of prime mover power required to drive the given unit. This, in consequence, reduces the power consumption and improves the economy of pumping.

The operation of the developed computer program is illustrated by presenting an example problem, the basic data of which are given in **Table 1.**

CALCULATION OF GEARBOX TORQUES

The prime mover drives the pumping unit through the slow-speed shaft of the gear reducer by overcoming the net torque needed to move the polished rod and all connected parts. As polished rod loads considerably vary during the pumping cycle, the amount of energy to be input into the system also varies with time. Therefore, in order to calculate the instantaneous and average energy requirements as well as other operating parameters of pumping, instantaneous torques on the crankshaft, resulting from polished rod loads, counterweights and other moving parts have to be calculated. Generally, four kinds of torques can be distinguished:

- Rod Torque is the result of polished rod loads and can be calculated with the help of the unit's kinematic parameters.
- Counterbalance Torque is a sinusoidal torque vs. crank angle, and is required to move the counterweights at a constant angular velocity.
- Articulating Torque is of inertial nature and represents the energy stored and released from accelerating-decelerating parts of the unit, such as the beam, horsehead, pitman, etc. As can be easily seen,

this torque exists even when crankshaft speed is constant.

- Rotary Inertial Torque occurs when crankshaft speed is not steady and it is the result of speed changes of rotating masses: counterweights, cranks, etc.

Of the four above-mentioned types of crankshaft torque, for the majority of cases and especially for normal slip electric motors with a low variation of crankshaft speed, only the first two have to be considered. This is due to the facts that under such conditions rotary nertial effects are negligible and articulating torque can also be disregarded. [2]

The net torque loading of the gear reducer is the sum of torques acting on the crankshaft and can be calculated by the following formula as a function of crank angle:

$$T_{net}(\Theta) = TF(\Theta) \, [\, F_p(\Theta) - SU \,] + T_{CB}(\Theta)$$

The first term on the right-hand side is the Rod Torque which is calculated by multiplying the net polished rod load with the API Torque Factor. Torque factors represent imaginary lever arms and are determined from the pumping unit's geometrical data using available calculation models. [1,3,4]

Counterbalance Torque opposes Rod Torque in direction and varies with the sine function of crank angle as given below:

$T_{CB}(\Theta) = - M \text{ sine } \Theta$ for conventional units,
$T_{CB}(\Theta) = - M \text{ sine}(\Theta - \tau)$ for Torqmaster units,
$T_{CB}(\Theta) = - M \text{ sine}(\Theta + \tau)$ for Mark II units.

USE OF DYNAMOMETER CARDS

From the preceding it is obvious that rod torque can only be calculated if the F_p polished rod load is known in function of crank angle. As conventional dynamometer cards record polished rod loads vs. polished rod displacement, some procedure has to be used to get the required $F_p(\Theta)$ function. The API method [1] involves a manual construction and uses the $PR(\Theta)$ (position of rods) values that are supplied by pumping unit manufacturers. The application of this procedure has several drawbacks that impair its practical use, and which can be summed up as follows:

- Relying on hand construction and visual read-off, the accuracy of the results can be low.
- The number of points that can be read off the card is limited, as manufacturers usually supply $PR(\Theta)$ values for each 15 degrees of crank angle only.
- Actual peak torque values may be undetected, as these can lie between subsequent points.
- Its use is time-consuming.

To speed up calculations and to improve accuracy the author developed a computerized technique to find the $F_p(\Theta)$ function needed for torque analysis. This involves the digitizing of dynamometer cards by the use of a HP 7470A or 7475A Plotter. Since the proposed procedure utilizes user-specified number of points on the card, a sufficiently small $\Delta\Theta$ crank angle increment can be obtained. The basic steps of digitizing are detailed below.

- The number of points to be used, and thus the crank angle increment is set.
- The position of rods (PR) function is calculated for each Θ crank angle.
- The digitizing sight of the plotter is programmed to move over the card by taking given load steps on the actual $PR(\Theta)$ = const. line, starting from one of the extreme values of possible loads. The program automatically corrects for any rotation that might exist between card and plotter coordinate axes.
- After the user realized that the sight crossed the card, he signals to the computer to reverse the direction of movement and to decrease the steps taken by the sight. A second signal is sent when the sight is exactly over one point of the card. The plotter coordinate read off this way, after necessary transformations and scaling, will give the F_p polished rod load corresponding to the actual Θ crank angle.
- The procedure is repeated with the next value of crank angle, until the total number of points is reached.

The above procedure enables one to collect the necessary input data from the dynamometer card in a very efficient and highly accurate way. Not only the number of points can be selected but also the accuracy of load readings can be changed. These features make the proposed method superior to the API procedure both in the amount and accuracy of information that can be retrieved from dynamometer cards.

ANALYSIS OF GEARBOX LOADING

After the dynamometer card has been digitized, the program calculates gearbox torques as a function of crank angle. The shape and magnitude of these torque curves is a direct indication of gearbox loading, from which one can draw important conclusions on the operation of the pumping unit. The analysis of gearbox loading can be accomplished by the methods detailed below.

Permissible Load Diagrams

The permissible load concept was introduced by R.H.Gault [5] to evaluate the torque loading of pumping unit gear reducers. Permissible load is defined as the polished rod load value that gives a net torque equal to the gearbox torque rating, using the actual counterbalance. These loads can be plotted directly on the dynamometer card and the range between the values valid for up, and downstroke represents the range of polished rod load values that do not overload the gear reducer. **Figure 1** shows the permissible loads superimposed on the dynamometer card for the example given in **Table 1.**

The permissible load is a function of crank angle and has different shapes for the up, and downstroke. Its value approaches infinity at the start and end of upstroke as well as downstroke, as at these points the torque factors equal zero. This is easily observed from the equations used for different geometries:

$$F_{pe}(\Theta) = \frac{T_{net}(\Theta) + M \text{ sine } \Theta}{TF(\Theta)} + SU \qquad \text{Conventional,}$$

$$F_{pe}(\Theta) = \frac{T_{net}(\Theta) + M \text{ sine}(\Theta - \tau)}{TF(\Theta)} + SU \qquad \text{Torqmaster,}$$

$$F_{pe}(\Theta) = \frac{T_{net}(\Theta) + M \text{ sine}(\Theta + \tau)}{TF(\Theta)} + SU \qquad \text{Mark II.}$$

Gearbox Torques vs. Crank Angle

Figure 2 shows the variation of gearbox torques in the example case for actual counterbalancing. Such plots can also be useful to evaluate the operating conditions of pumping unit gear reducers.

Cyclic Loading

Electric motors, which are nowadays predominantly used to drive pumping units, are usually designed to work under steady loading conditions. The torque loading of gear reducers, however, is generally far from being constant. The basic reason for this is that during upstroke high positive torques are required to move the polished rod, while on the downstroke the polished rod drives the gear reducer, resulting in negative rod torque values. By applying counterbalance torque, which is the result of rotating counterweights, net torque fluctuations can be reduced (see **Figure 2**), but cannot be totally eliminated. This is why the selection of electric prime movers has to include considerations for cyclic loading.

It can be shown [6] that electric motors used for cyclic load service have to be oversized by a factor called Cyclic Load Factor (CLF). CLF allows for the overheating of the motor above designed temperature and is the ratio of thermal to average electric current values calculated for the pumping cycle. Thus, for constant loads CLF = 1, and its value increases as motor loading gets more uneven. It is also evident that a decrease in CLF results in less required motor power for the same conditions.

CLF can also be determined based on the net torque patterns, because current vs. torque characteristics of electric motors are usually linear. [7,8] The CLF value, calculated this way for the example case is given in **Table 1**. The equation for CLF using net torques is:

$$CLF = \frac{\int [\, T_{net}(\theta) \,]^2 \, d\theta}{\int T_{net}(\theta) \, d\theta}$$

OPTIMUM COUNTERBALANCING

Counterbalancing is used to even out the torque loading of pumping unit gear reducers, by the predominant use of crank counterweights. The counterweights produce a torque which is sinusoidal with crank angle, the maximum of which depends on their total weight and the distance between the crankshaft and their center of gravity. By changing either of these factors, the counterbalance and net torques change accordingly. This enables one to find the required amount of counterbalance for given conditions.

The concept of optimum counterbalancing has been a heavily discussed problem and different solutions have been given. These include minimizing the peak net torque, and setting the up, and downstroke peak torques or powers equal. [9,10,11] Of these methods the generally accepted one tries to keep the up, and downstroke peaks of net torque approximately equal. Although theoretically not very sound, this approach has a lot of practical advantages, hence its widespread use.

The author developed a novel technique to find optimum counterbalance conditions by setting the minimum of cyclic load factor as the goal of optimization. As discussed earlier, CLF is a very good indicator of the effectiveness of counterbalancing, and also directly impacts on the required prime mover power. The required motor power can be calculated by:

$$P_{mot} = \frac{PRHP \;\; CLF}{\eta_m}$$

Polished rod horsepower (PRHP) represents the power needed at the polished rod and is independent of the degree of counterbalancing. The mechanical efficiency of the pumping unit and gear reducer can also be considered constant, thus prime mover power is directly proportional to the value of CLF. This means that the lower the value of CLF, the lower the power required to drive the same unit. But CLF can be changed at will by adjusting the counterweights, therefore the logical solution for finding the optimum counterbalance conditions will be to minimize the cyclic load factor.

To illustrate the effect of counterbalancing on the value of CLF, **Figure 3** was prepared for the example case. It is clearly shown that, as maximum counterbalance torque increases from zero, CLF values rapidly decrease, but after reaching a minimum they start to increase with further increase in counterbalance torque. **Figure 4** gives, for the same conditions, the variation of the up, and downstroke peak net torque values vs. maximum counterbalance torque. Intersection of these curves represents the counterbalance required to set the two peaks equal. Comparison of **Figure 3** and **Figure 4** enables one to conclude:

- The proposed procedure results in a lower CLF value.
- The difference between the CLF values in the given case is not very great, showing the merits of the old procedure.

Calculation results of the proposed optimization technique are given in **Table 1**, the gearbox torques for optimum counterbalance conditions are plotted on **Figure 5**. These all show a considerable improvement compared to original conditions. Application of the optimum counterbalance torque has reduced the torque loading of the gearbox from the original 157% to 123%, and has also reduced the CLF. The decrease in CLF is approximately 12%, which shows that the power consumption of the given unit would decrease by the same amount. Proper counterbalancing of pumping units has therefore, besides several technical improvements, immediate financial advantages.

CONCLUSIONS

1. A highly accurate and efficient calculation model has been developed to input data from dynamometer cards for pumping unit torque calculations.
2. A new technique is proposed to find optimum counterbalance conditions that can reduce the power consumption of sucker rod pumping.

NOMENCLATURE

F_p	polished rod load, lbs
F_{pe}	permissible load, lbs
M	maximum counterbalance torque, in-lbs
PR	position of rods, -
PRHP	polished rod horsepower, HP
SU	structural unbalance of pumping unit, lbs
TF	torque factor, in
T_{CB}	counterbalance torque, in-lbs
T_{net}	net torque, in-lbs
θ	crank angle, degrees
τ	counterbalance offset angle, degrees
η_m	mechanical efficiency of pumping unit gear reducer, -

REFERENCES

1 "API Specification for Pumping Units." **API Std 11E** 12th Ed. (1982) with Supplement 1 (1983)

2 Gibbs, S.G.: "Computing Gearbox Torque and Motor Loading for Beam Pumping Units with Consideration of Inertia Effects." **J. Pet. Tech.** (Sept. 1975) pp 1153-59

3 Gray, H.E.: "Kinematics of Oil-Well Pumping Units." **API Drilling and Production Practice** (1963) pp 156-65

4 Svinos, J.G.: "Exact Kinematic Analysis of Pumping Units." paper **SPE 12201** presented at the 58th Annual Technical Conference and Exhibition, San Francisco, Oct. 5-8, 1983

5 Gault, R.H.: "Permissible Load Diagrams for Pumping Units." **7th Annual West Texas Oil Lifting Short Course** Lubbock, Tex. (1960) pp 67-71

6 Howell, J.K. and Hogwood, E.E.: **Electrified Oil Production** Petroleum Publishing Co. 1962

7 Eickmeier, J.R.: "Pumping Well Optimization Techniques." **J. Can. Pet. Tech.** (Apr.-June 1973) pp 53-61

8 Byrd, J.P. and Beasley, W.L.: "Predicting Prime Mover Requirements, Power Costs, and Electrical Demand for Beam Pumping Units." paper **No. 374035** presented at the 25th Annual Technical Meeting of the Petroleum Society of CIM, Calgary, May 7-10, 1974

9 Kemler, E.N.: "Counterbalancing of Oil-Well Pumping Machines." **API Drilling and Production Practice** (1943) pp 87-107

10 Johnson, D.O.: "Counterbalancing of Beam Pumping Units." **API Drilling and Production Practice** (1951) pp 243-41

11 Takacs, G. and Papp, I.: "Dynamometer Card Analysis Using Minicomputers." (in Hungarian) **Koolaj es Foldgaz** (March 1978) pp 65-73

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Table 1
Basic Input and Output Data
for Example Well

TORQUE ANALYSIS OF PUMPING UNITS

USING DYNAMOMETER CARDS

W E L L   D A T A of Well # Example

Pump Setting Depth :	4608.0 ft	Dynamic Liquid Level :	4608.0 ft
Liquid Prod. Rate :	121.0 bpd	Liquid Spec. Gravity :	0.9000

P U M P I N G   U N I T   D A T A

	Manufacturer :	LUFKIN			
API Designation :	C- 160D-173- 86	Rotation :	CLOCKWISE		
Structural Unbalance :	450 lbs	CB Offset Angle :	0.0 deg		
Meas. Stroke Length :	86.00 in	Pumping Speed :	14.86 SPM		
Linkage Dimensions :	L1 = 37.000 in	L2 = 151.340 in	L3 = 96.050 in		
	L4 = 114.000 in	L5 = 111.000 in	L6 = 96.000 in		

D Y N A M O M E T E R   C A R D   D A T A

Dynamometer Constant : 1000 lbs/cm   No. of Points on Half Stroke : 25
Maximum CB Torque : 250000 in-lbs

C A L C U L A T E D   P A R A M E T E R S

Min. Polished Rod Load :	4108 lbs	Hydraulic Power :	3.7 HP
Peak Polished Rod Load :	10996 lbs	Polished Rod Power :	12.7 HP
Percent of Rating :	63.6 %	Lifting Efficiency :	29.2 %

		Actual COUNTERBALANCING	Ideal COUNTERBALANCING
Max. CB Torque	in-lbs	250000	309789
Min. Net Torque	in-lbs	-35277	-23675
Peak Net Torque	in-lbs	252004	197330
Percent of Rating	%	157.5	123.3
Cyclic Load Factor	-	1.594	1.400

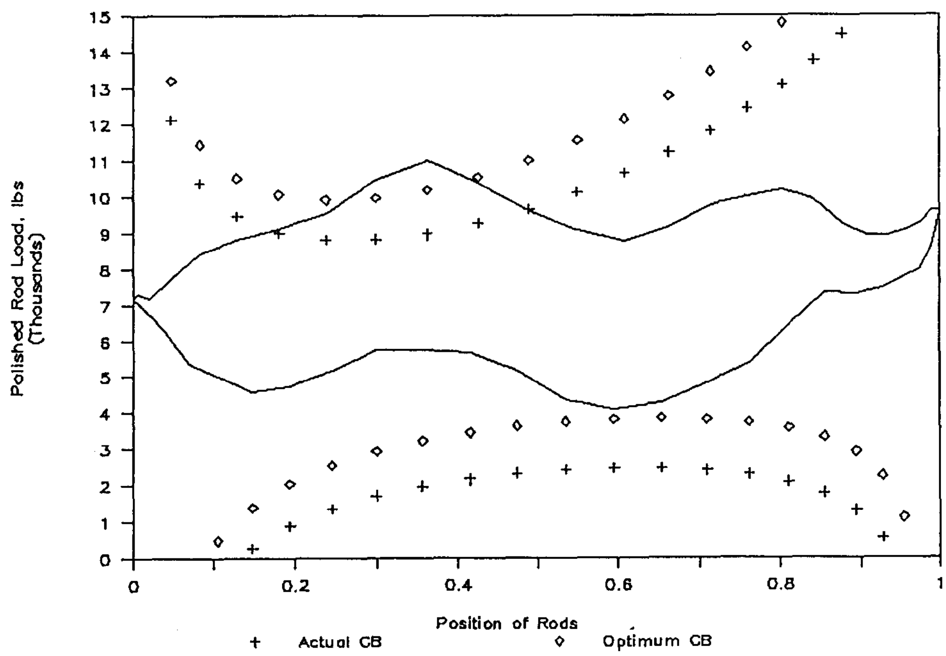

Figure 1 — Dynamometer card taken on example well including permissible loads

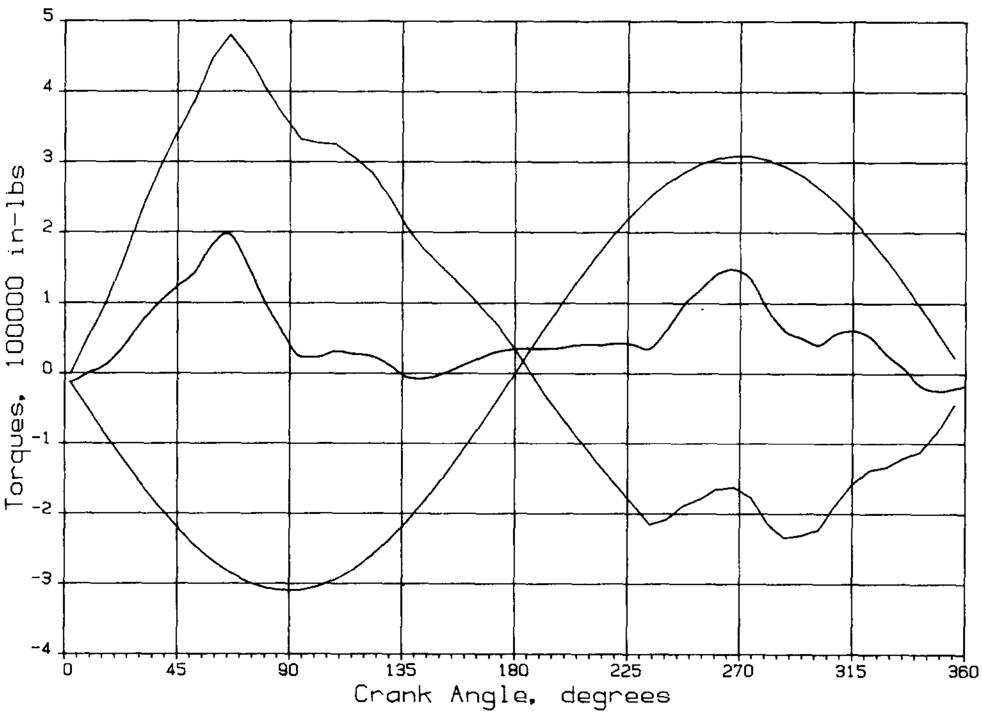

Figure 2 — Calculated gearbox torques vs. crank angle for example well with actual counterbalancing

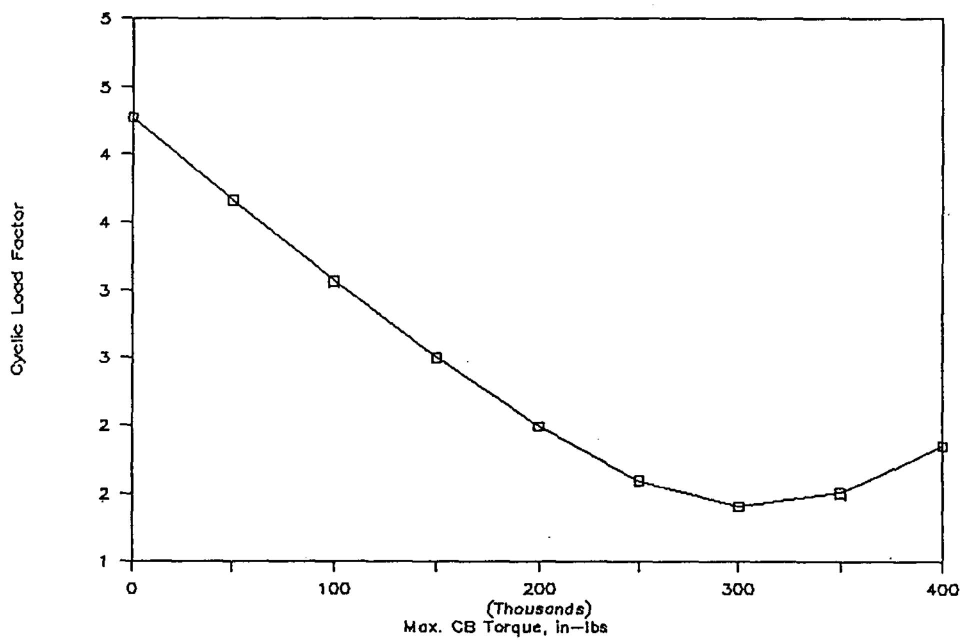

Figure 3 — Calculated cyclic load factors vs. maximum counterbalance torque for example well

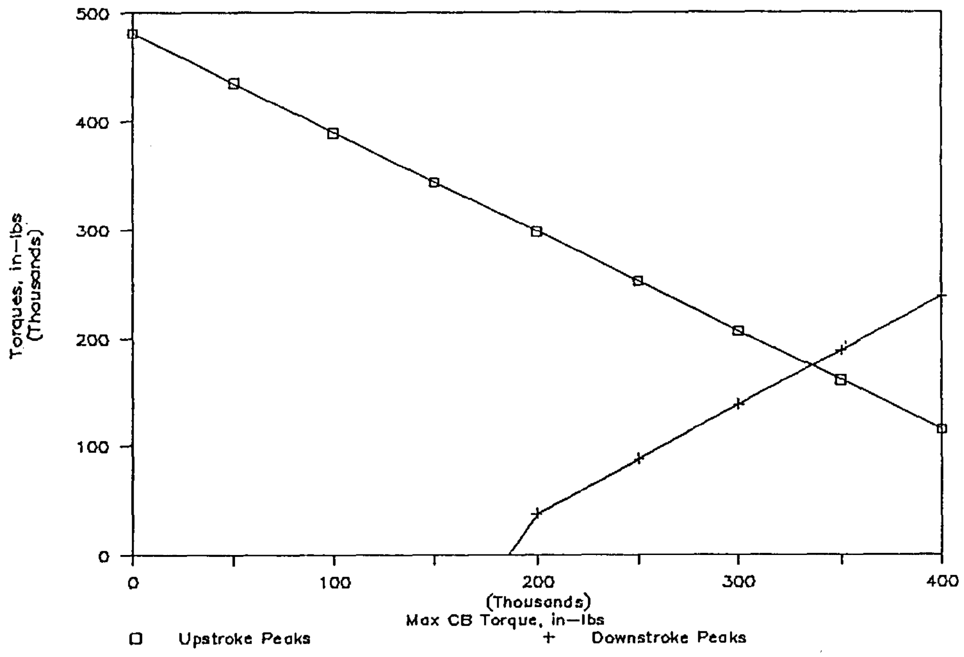

Figure 4 — Variation of peak net torque values during up and downstroke with varying maximum counterbalance torque

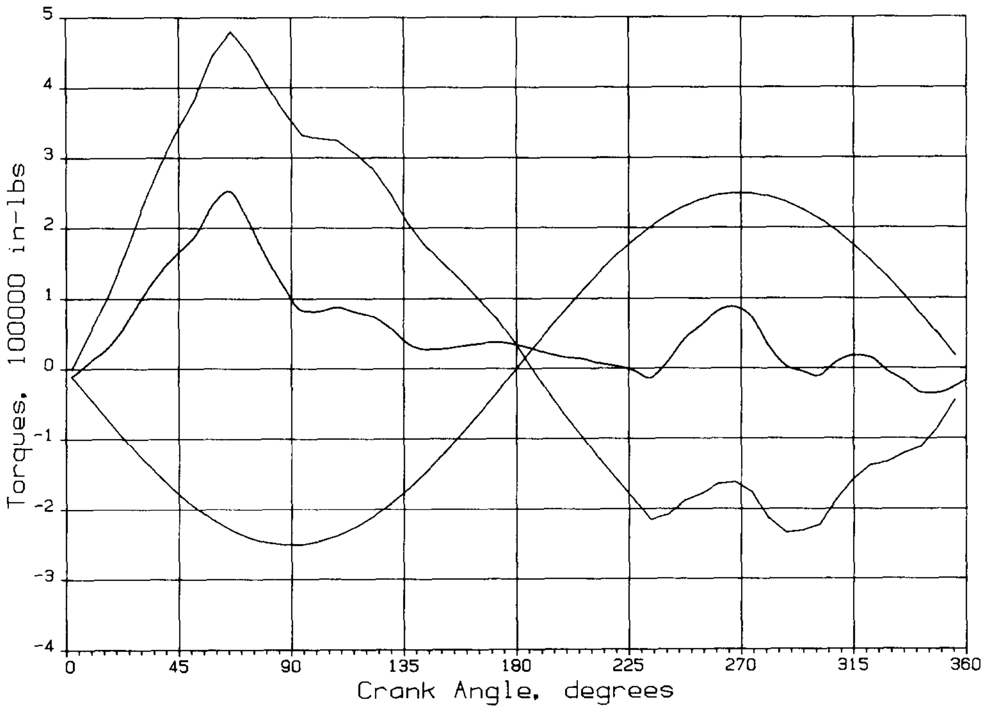

Figure 5 — Calculated gearbox torques vs. crank angle for optimum counterbalancing