A PRACTICAL MEANS OF CHECKING
COUNTERBALANCE CONDITIONS

By

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1954

Submitted to the faculty of the Graduate School of
the Oklahoma Agricultural and Mechanical College
in partial fulfillment of the requirements
for the degree of
MASTER OF SCIENCE
May, 1956

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No greater appreciation should be expressed than that due the Stanolind Oil and Gas Company for the vital role it played in unselfishly underwriting the financial phase of this investigation.

This together with its sincere, expert counsel and cooperation in making available the necessary facilities for testing, have been of inestimable value to the writer in the progress and completion of this project.

PREFACE

During the last few years, proper counterbalance as an equipment conservation measure, has been brought to the attention of the oil industry. However, measuring the existing counterbalance accurately, quickly, and inexpensively is the major problem.

In order to attack the problem logically, the investigator must be aware of all the previous significant work accomplished in this field, the practical limitations upon the solution, the variables suitable for measuring counterbalance, and the devices suitable to measure these variables.

The purpose of this investigation is to determine a variable that is indicative of counterbalance and to develop an instrument meeting the requirements to measure this variable.

I am deeply indebted to Professor W. H. Easton, my thesis advisor, for his conscientious and painstaking counsel, his sincere criticism and encouragement. My thanks and appreciation are extended to Professors R. E. Chapel and E. C. Fitch for their counsel and advice.

Acknowledgment is also given to National Supply Company for their cooperation in loaning one of their pumping units for my experimental tests.

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CHAPTER I

INTRODUCTION

The use of the walking beam sucker rod combination for the lifting of a fluid is an old invention. The idea is so ancient that the first date of application is unknown. It is known, however that in 476 A. D. the Egyptians used a device which bears a striking resemblance to the modern pumping unit. This simple mechanism employed a tripod which supported a wooden beam. Attached to one end of the beam was the rod string and the stone water jar, which was actuated by a servant, while on the other end hung a goat skin bag filled with rocks which served as a counterbalance.

It is interesting to note that no appreciable improvement has been made on the original idea, except the introduction of mechanical power and new materials of fabrication. The oil industry uses essentially the same Egyptian idea for the production of oil. The major quantity of all oil is produced by this method.

The first experience of our people with crude oil production was very distasteful. It was discouraging when water well drillers, such as Mr. Titus, discovered after their diligent labor in making a hole in mother earth, that instead of producing clear sparkling water from a well, they would sometimes produce a greenish brown, odorous and greasy fluid for which they had no use except to grease their wagon wheels. This unwanted and useless commodity was to become the sire of one of our greatest industries.

When oil production was first introduced, little or no attention was paid to the exorbitant cost of producing it. This was due to the limited use of this new commodity and to the lack of competition among the few manufacturing firms engaged in this new industry. Because of this, these companies had little incentive to develop new ideas or methods, or to improve on the design of existing equipment.

With the advent of the gasoline engine and the automobile, all of this was changed practically overnight with the consequence that in the 1920's the demand for oil and its by-products far exceeded the production from the then-established-shallow fields. To satisfy the unprecedented demand deeper zones were tapped and developed.

The greater depths at which oil was being produced caused production costs to soar and the attention of the entire oil industry became focused on production costs and the economics of oil production.

Since then, manufacturers have made tremendous strides in decreasing production costs and as a result of their efforts, counterbalance has again assumed its vital role, as a factor having great potential towards the further reduction in production costs.

CHAPTER II

STATEMENT OF PROBLEM

Since proper counterbalance contributes more than any other factor to the satisfactory operation of beam pumping units, a method must be found to measure qualitatively the existing counterbalance.

While present methods measure existing counterbalance, they are unacceptable because either a technically trained man is required to take and interpret the results of the method or the method used is not accurate.

Therefore, the problem is to develop a device, adaptable to most pumping unit installations, that measures qualitatively the existing counterbalance.

CHAPTER III

ANALYSIS OF PROBLEM

To analyze the problem completely, the term "counterbalancing" must first be clearly understood. Zaba explains counterbalancing as follows: "Counterbalancing provides for even distribution of loads and for reduction of peak horsepower and peak torque."[1]

This statement can be elaborated on as follows: Counterbalancing produces that mode of operation of a pumping unit which has the lowest possible stresses in the members of the pumping unit, particularly the sucker rods, the walking beam, the pitman, the gear box and the V-belts.

E. N. Kemler, a well-known authority in the oil industry, says this about counterbalance: "No other single item in the operation of pumping equipment can make such a large difference in operation and maintenance costs."[2]

The previous statement concerning proper counterbalance is a result of these known facts.

The cause for ninety per cent of gear failures, pitman failures, belt failures, bearing failures and shaft failures is overload due to improper counterbalancing. The other ten per cent are caused by improper lubrication and normal wear, faulty material or improper selection of size to begin with.

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[1] J. Zaba, Oil Well Pumping Methods (Tulsa, 1943), p. 30.
[2] E. N. Kemler, "Proper Counterbalance--An Equipment Conservation Measure," The Oil Field Weekly, (June, 1943), p. 108.

. . . . The only reason for counterbalancing is to save money in first cost of equipment and save operating cost.[2]

It is a demonstrated fact that as the depth of producing wells increases there is a comparable rise in production costs. Proper counterbalancing is one method of reducing these production costs.

Four methods of determining counterbalance are illustrated by the representative curves on the following page.

Figure I shows the torque curves for a pumping unit. The dotted line curve represents an out-of-balance condition, while the solid line curve represents correct counterbalance. The torque of this out-of-balance pumping unit varies from a maximum at point A on the upstroke to a minimum at point B on the downstroke. The use of the correct amount of counterbalance alters the curve substantially. When the two peak torques at A and B are equal or approximately equal, a condition of proper counterbalance exists.

Figure II is a wattmeter chart of an electrically powered pumping unit. As in the previous illustration, when point A equals point B the unit is in proper counterbalance.

Figure III is a load graph of a well with effective counterbalance shown. When the average of the sum of the maximum load on the upstroke and the minimum load on the downstroke is equal to the value of the effective counterbalance, the well is in proper counterbalance (assuming that the maximum loads occur approximately 180° apart).

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[2] "Counterbalancing," Sucker Rod Handbook, (Bethlehem Steel Company, 1950), p. 165.

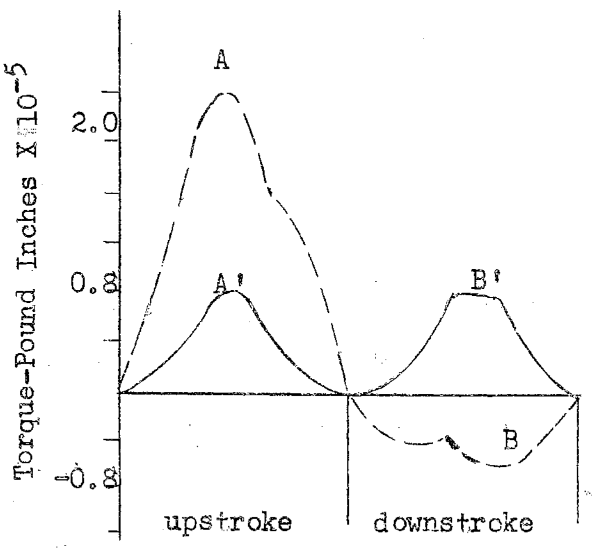

Figure I

Typical Torque Curves

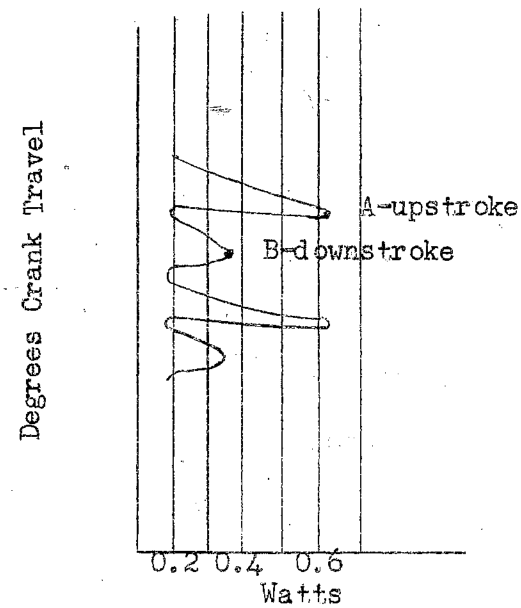

Figure II

A General Wattmeter Chart

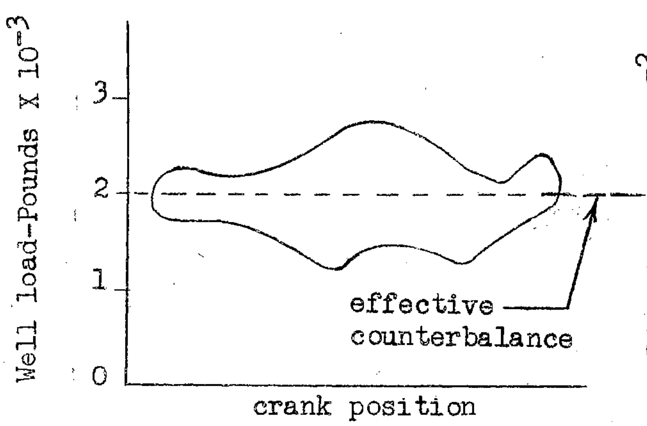

Figure III

Typical Dynamometer Card

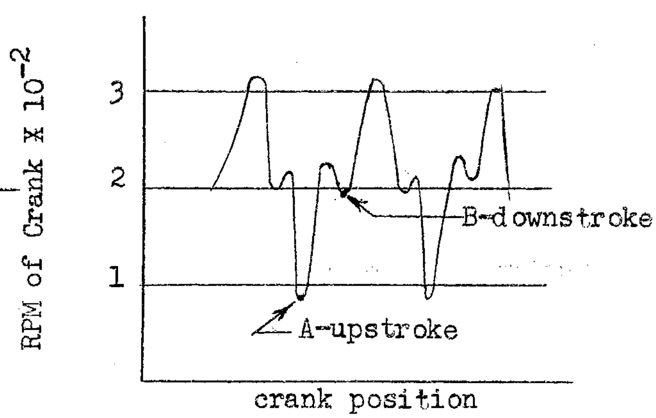

Figure IV

Typical Tachometer Recording

Figure IV is a curve illustrating the variation of the engine crank angular velocity. The angular velocity of the crank varies from a minimum at the approximate center of the upstroke (point A), to a similar point B at the approximate center of the downstroke, and points A and B are not of equal value indicating improper counterbalance. When proper counterbalance exists, point A will equal B.

A device which provides a suitable solution to the determination of proper counterbalance should meet the following requirements: it must not carry the full well load, it must not require a source of electrical power, it must not be sensitive to ordinary field conditions, it should be inexpensive, it must be simple to operate and maintain, and it must be adaptable to most pumping units.

Counterbalance is applied by three different methods: beam counterbalance, rotary counterbalance and beam-rotary counterbalance.

Beam counterbalance is the method by which weights are attached to the beam and the approximate amount of counterbalance necessary is based on the criterion of equal maximum engine loads for the upstroke and downstroke. The theoretical amount of counterbalance necessary for the pumping unit can be determined by investigating the effects of the static conditions.

In the following derivation many factors have been omitted in order to reduce the complexity of the theoretical concept for counterbalance. These factors are divided into two groups. The first group contains the intrinsic characteristics of the pumping cycle, the speeds and loads involved, the characteristics of the pumping mechanism and the prime mover used. The other group pertains to the well characteristics.

The pumping unit (Figure V) is beam counterbalanced. The centroid of the two beam weights acts through the upper pitman bearing.

F = upstroke force, lb
F' = downstroke force, lb
F_1 = well load, lb.
$W_1 + W_2 = F_c$ = counterbalance weight, lb
F_e = engine load, lb.
L_1 = lever arm to horsehead, in.
L_2 = lever arm to upper pitman bearing, in.

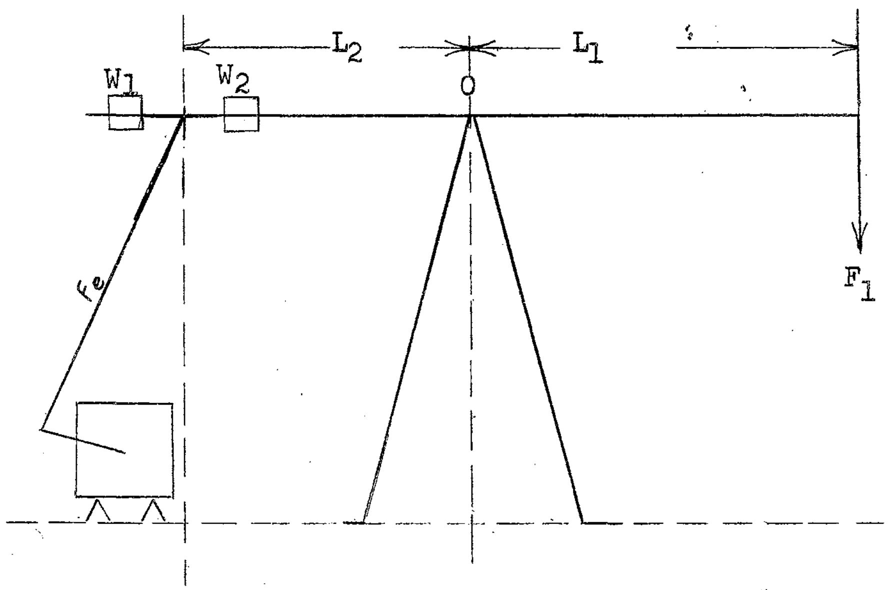

Figure V Pumping Unit With Beam Counterbalance

In order to simplify the development of the equations which follow the effect of angularity of the pitman will be neglected.

If moments are taken about the samson post bearing (point O) and counterclockwise direction is assumed positive, the following is obtained:

1. $0 = (F_e)(L_2) + (F_c)(L_2) - (L_1)(F_1)$ upstroke

2. $0 = -(F_e')(L_2) + (F_c)(L_2) - (L_1)(F'_1)$ downstroke

The well load on the upstroke is made up of two components F_2 and F_3, where F_2 is the weight of the rods and F_3 is the weight of the fluid, therefore, $F_1 = F_2 + F_3$. The well load on the downstroke is equal to F_2 since the rods are moving downward. They are not supporting the fluid column. Equations 3 and 4 are solved for $(F_c)(L_2)$

3. $(F_c)(L_2) = (L_1)(F_1) - (F_e)(L_2)$

4. $(F_c)(L_2) = (L_1)(F_1') + (F_e)(L_2)$

Substituting $F_1 = F_2 + F_3$ and $F_1' = F_2$

$$(F_c)(L_2) = (L_1)(F_2) + (L_1)(F_3) - (F_e)(L_2)$$

$$(F_c)(L_2) = (L_1)(F_2) + (F_e)(L_2)$$

Adding, the result is:

$$2(F_c)(L_2) = 2(L_1)(F_2) + (L_1)(F_3).$$

Solving for F_c and assuming $L_1 = L_2$ the result is:

5. $F_c = F_2 + \frac{1}{2}F_3$.

This states that the weight of the counterbalance is equal to the weight of the rods (F_2) plus one half the weight of the fluid (F_3).

The results of the derivation apply equally as well to the rotary type of counterbalance as to the combination of beam and rotary counterbalance.

The value of F_c previously obtained is the effective counterbalance. The relationship between actual and effective counterbalance is shown in Figure VI.

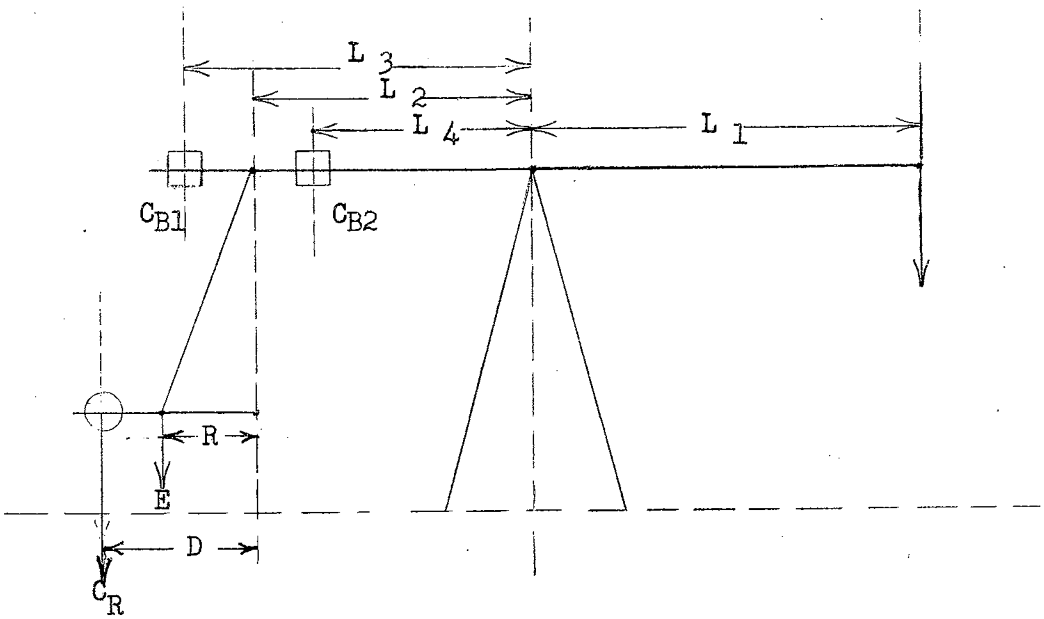

Figure VI Pumping Unit With Rotary-Beam Counterbalance

$$6.\quad C_{E_B} = \left(C_B \pm \frac{C_B}{g}(a)\right)\left(\frac{L_3 + L_4}{2L_1}\right) \quad \text{for}$$

beam counterbalance and,

$$7.\quad C_{E_R} = \left(\frac{L_2}{L_1}\right)\left(\frac{C_R \times D}{R}\right) \quad \text{for rotary counter-}$$

balance where:

C_{E_B} = the effective beam counterbalance as experienced by the well side of pumping unit, lb

C_B = $C_{B1} + C_{B2}$ actual static weight of the counterbalance, lb.

a = vertical acceleration of the beam weight at the upper pitman bearing, fpsps.

g = the acceleration of gravity, fpsps.

L_3 = lever arm for beam weight C_{B1}, in.

L_4 = lever arm for beam weight C_{B2}, in.

L_1 = lever arm from sucker rods to samson-post, in.

L_2 = lever arm from upper pitman bearing to samson-post, in.

C_R = actual weight of rotary counterbalance, lb.

C_{E_R} = effective rotary counterbalance weight as experienced by the well side of pumping unit, lb.

D = radius of rotation of center of gravity of the rotary counterbalance weight, in.

R = radius of crank pin, in.

Therefore, the speed of operation of a pumping unit has an influence on the effectiveness of the beam weights. The effectiveness of rotary weights is independent of speed.

The procedure for finding a method and a device which will indicate the degree of counterbalance of a pumping unit is divided into four parts: a study of the methods and devices now in use, an investigation of the variables which are indicative of counterbalance, a kinematic analysis of a typical pumping unit, and the conplete analysis of the various factors which are the most promising for the solution of the problem.

CHAPTER IV

RECOGNIZED METHODS OF CHECKING COUNTERBALANCE

It has previously been shown that the proper operation of a pumping installation depends to a large extent upon its counterbalance. Therefore, it is evident that methods müst be found to measure counterbalance. Some of the most common methods in use are: recording tachometer, slipping of the clutch, prime mover sound, belt tension, recording dynamometer, recording wattmeter, and vacuum gage readings. None of these methods have been wholly satisfactory because of the limitations associated with the solution of the problem.

The recording tachometer method is based on the theory that the engine slows down the same amount at the approximate center of each stroke because the maximum load or torque on the upstroke should be equal to the maximum load on the downstroke as experienced by the engine when the pumping unit is in proper counterbalance. This theory is based on the assumption that the engine is loaded heavily, that the governor is fixed, and that the so-called "flywheel effect" is negligible.

> A flywheel, interposed between prime mover and the crank, smoothes out the loads during different points of the stroke. The effectiveness of the flywheel is proportional to the square of its speed. Conditions may arise when the flywheel effect is sufficiently high to give constant speed although the well is not counterbalanced properly.[1]

Thus, "the recording tachometer is an excellent guide in some

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1 J. Zaba, Oil Well Pumping Methods (Tulsa, 1943), p. 41

cases; in others it is practically useless" because of the fly-wheel effects.[2]

The clutch slipping method was introduced by D. O. Barrett. This method is outlined as follows. The throttle is first fixed on the engine and the clutch is slowly released to the point where it will pull the peak load. When the pumping unit is in proper counterbalance the clutch will slip equally on the upstroke and downstroke, since the peak loads will be equal. When it is not in counterbalance the clutch will slow down and slip on one stroke, while on the other stroke the clutch will speed up and hold. By this method the speed variations can easily be observed on the crank.

Slipping of the clutch is a fairly good method but is not always practical, especially where large rotary weights are being used. Also this means that the clutch must be in good condition, which may or may not be the case.[3]

Thus, the cost of providing and maintaining a clutch which would handle the above operation is a determining factor.

The use of the prime mover sound is not satisfactory not only because of the human element involved, but also because the change of sound of the pumping unit is a function of the change of speed, while the change in speed is affected by the "flywheel effect". Therefore, the sound is affected by the "flywheel effect" and is as vulnerable to erroneous conclusions as the recording

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[2] Sucker Rod Handbook, (Bethlehem Steel Company, 1950), p. 166.

[3] D. O. Johnson, "Counterbalancing of Beam Pumping Units," Drilling and Production Practice, (New York, 1951) p. 234.

tachometer method. It can be stated, therefore, that this method is useless, except as a preliminary check.

While the prime mover sound method proves itself unsatisfactory for both electric motors and internal combustion engines, the recording wattmeter offers an excellent solution of the problem for electric motors, because of the instantaneous power measurement which can be made.

By the use of a recording wattmeter these measurements may be recorded on a continuous chart. Reasonably equal current peaks for the upstroke, when the well load is lifted, and for the downstroke, when the counterbalance weights are lifted, would indicate conditions of a satisfactory counterbalance.[4]

The wattmeter method shows immediately the existing load condition on the pumping unit, but the same type of information is not readily obtained from internal combustion engines. One attempt to solve the problem is the use of vacuum gage readings.

Experience shows that in relying on vacuum gage readings on the engine to determine best counterbalance, the result leads to an over-counterbalanced condition. In order to obtain results from this method, an experienced operator would be required, and each type of engine and well condition would need special interpretation.[5]

This method is unsatisfactory for general field use because of the required experience demanded of the person taking and interpreting the readings.

The most recognized method for determining counterbalance is the recording dynamometer method. This method employes a device

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4 Zaba, p. 41.
5 Johnson, p. 235.

which is placed between the connection of the polished rod and the horsehead and produces a dynamometer card. The effective counterbalance of the pumping unit is indicated on this diagram. Several quantities can be calculated from it such as: torque, instantaneous power, and the amount of counterbalance necessary. This method gives the most accurate measure of the amount of counterbalance required, but the time required to make and interpret the cards makes this method unsatisfactory.

The belt tension method, used in the past as a means to determine counterbalance conditions, was based on the idea that the engine power requirement reached a peak twice during a pumping cycle. Since belt tension is proportional to the horsepower and torque, the device indicated immediately, the existing counterbalance conditions. The two peak torques were approximately equal for proper counterbalance. This method was excellent for flat belt drives.

CHAPTER V

THE EVALUATION OF PROPOSED VARIABLES

In order to properly evaluate the proposed variables, the following considerations must be taken into account. The variable must be indicative of counterbalance and also sensitive to changing conditions. The devices to measure the variable or variables must fulfill the conditions specified by the Stanolind Oil and Gas Company, mentioned in the Statement of Problem.

The proposed variables to be evaluated as proper counterbalance indices are: velocity, acceleration, torque, horsepower, air-flow, back pressure.

Back pressure and air-flow are both indicative of the effective counterbalance, but there are two main reasons which eliminate them from further consideration. These reasons are: the two variables are peculiar to each engine and not to engines in general; and the devices to measure these variables, besides requiring a skilled operator, require an engineer for the interpretation of the results, immediately defeating the purpose as stated by Stanolind Oil and Gas Company.

The variables, horsepower and torque, can be determined from another variable which is belt tension. This variable, belt tension, is but one of the possible means of determining horsepower and torque. Another means is the wattmeter device which is easily adapted to the electrically powered pumping units, mentioned in Chapter IV. The use of belt tension is not a new means of indicating the effective counterbalance; in fact, it was one of the earliest ever employed. Belt tension is related to horsepower and torque, respectively, as shown by the following equations.

8. $hp_{tr} = CS(F_1 - F_2)$

9. $T = R(F_1 - F_2)$

Where: hp_{tr} = horsepower transmitted
C = constant
S = beltspeed, fpm
F_1 = tightside tension, lb
F_2 = slackside tension, lb
R = radius of belt pulley, in.
T = torque, lb-in.

The problem that remains is to design a device which will measure this quantity.

An idler pulley together with a hydraulic cylinder and pressure gage might be used to give a qualitative measure of

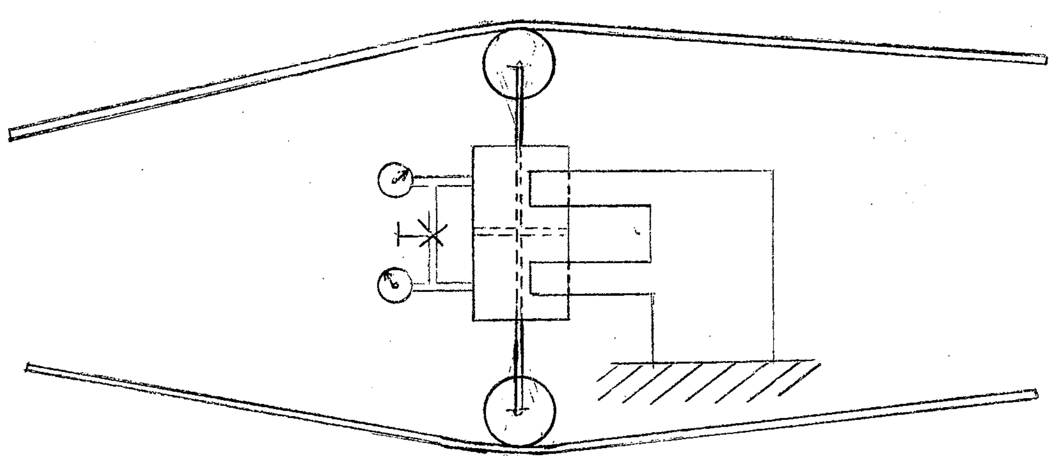

Fig. VII Belt tension device to measure peak torques belt tension. Figure VII shows a schematic view of such a device. The operation of this device is as follows. The pressures in the two gages are first equalized. Then pressure gage readings, p_1 and p_2, are taken on the upstroke and also on the downstroke. Thus the pressure gage readings may be used to obtain relative values of belt tensions. The peak value of the difference of belt tensions on the tight side and the slack side may then be obtained for the upstroke and the downstroke.

It appears as though this device should be a means of indicating the existing counterbalance condition based on the criterion of equal peak torques on the upstrokes and downstrokes.

The next variable to be considered is the velocity of the polished rod. The velocity of the polished rod has one of the same drawbacks as angular velocity. The sensitivity of either the polished rod velocity or the angular velocity of the crank to existing counterbalance conditions is a function of the "flywheel effect". Therefore, if "flywheel effect" is very large, velocity may not be a satisfactory indicator of counterbalance conditions.

It is known that the polished rod velocity varies from a maximum at the approximate center of the stroke to zero at the ends and this is readily verified by assuming simple harmonic motion for the motion of the polished rod.

The problem is to find out how much the polished rod velocity varies and where the maximum variation is recognized. To analyze this more fully a velocity diagram was drawn. A pumping unit was selected which met the specifications of Mr. Hopper's Report. This in essence said, that the most all-round efficient pumping unit has the upper pitman bearing on a horizontal line with the samson post bearing and the crank shaft on a vertical line with the upper pitman bearing.[1] Using the DB - 114 Emsco Pump-Unit, velocities were calculated at 15° intervals. The assumed curves of Fig. VIII represent various counterbalance conditions. (Although the ordinate of the curves is shown as strokes per minute it also represents revolutions per minute.) Curve No. 1, constant angular velocity, is an indeterminate counterbalance condition because the "flywheel effect" is too great to allow any variation to occur. Curve No. 2 represents proper counterbalance condition because the minimum angular velocities are equal at the center of the strokes. Curve No. 3 is an approximate diagram for the condition of overcounterbalance. Curve No. 4 represents the condition of undercounterbalance with the minimum velocity on the upstroke less that that on the downstroke.

The instantaneous velocities of the polished rod obtained by analysis of the curves in Fig. VIII are tabulated in Table 1 and plotted in Figure IX. The results of the analysis indicate that the variable velocity of the polished rod has potential, as an indicator of counterbalance, in that the maximum variation occurs at the approximate center of the stroke. The results of this analysis must not be construed to be conclusive, because it was assumed that the maximum variation of the angular velocity curves

[1]Baye Hopper, "Efficiency and Load Text on Well Pumping Units", <u>Petroleum Engineer</u>, (Junuary, 1940, p. 121.

Fig. VIII — Angular Velocity Curves

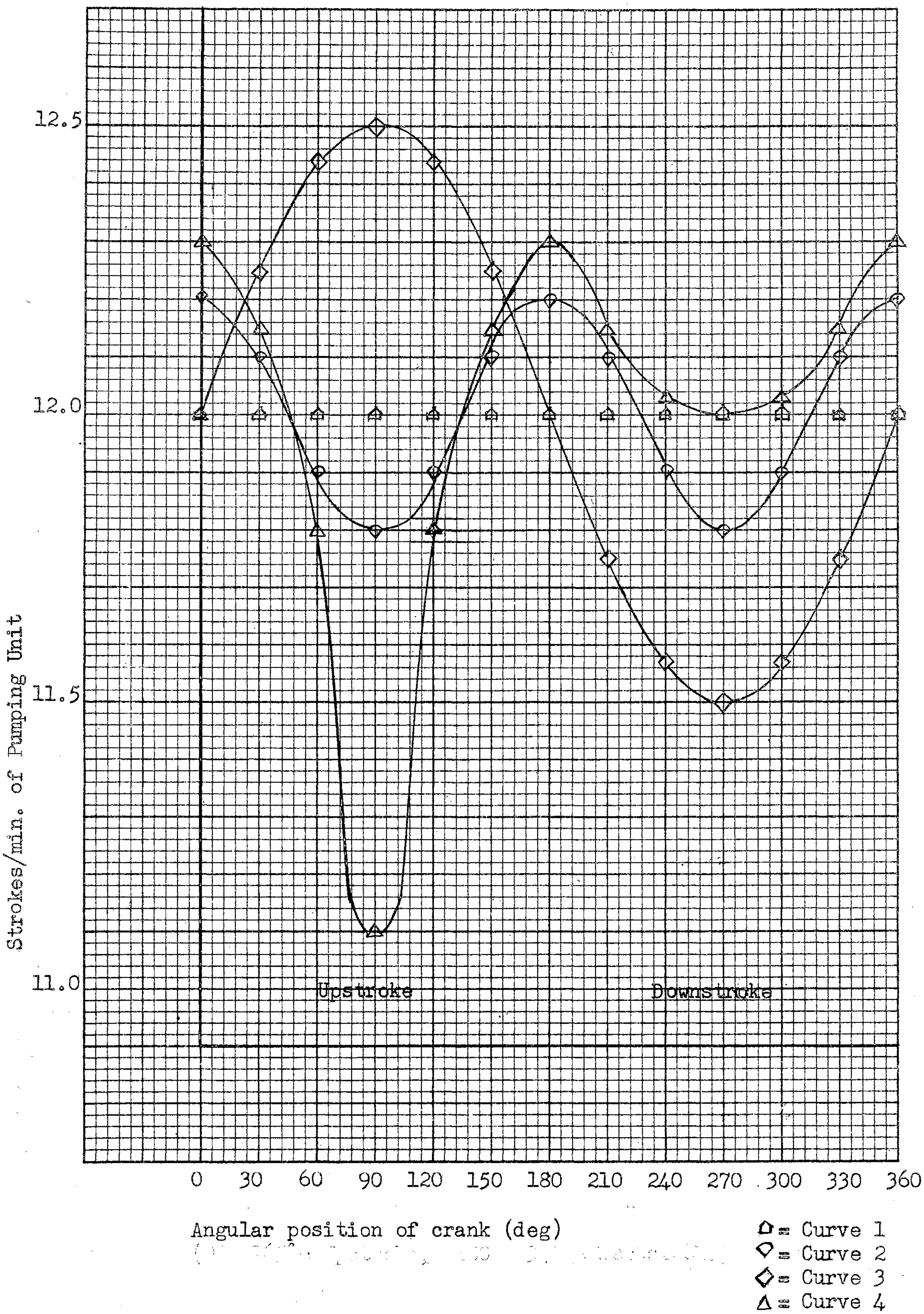

Fig. IX -- Tangential Velocity of Beam Versus Angular Position of Crank

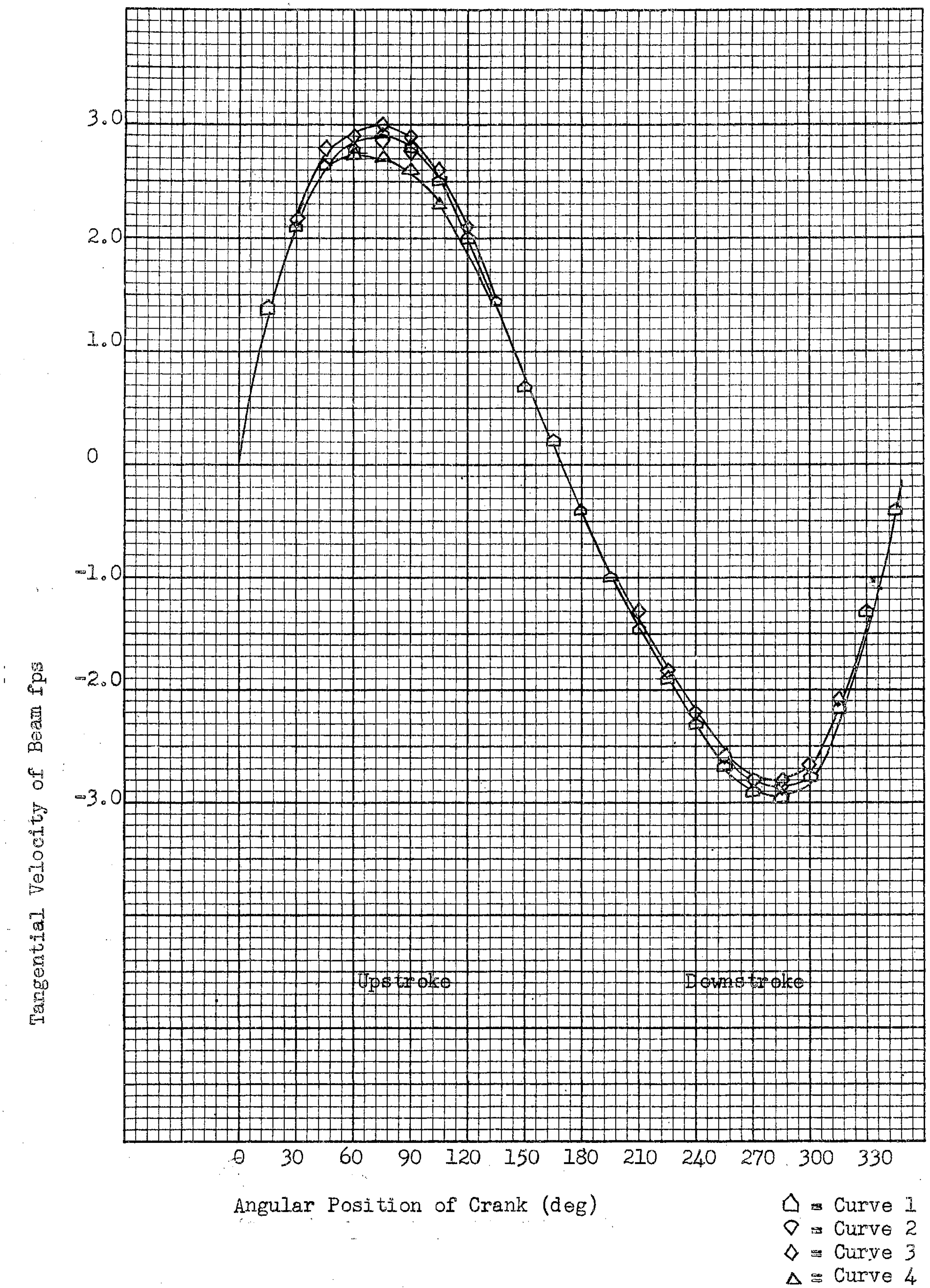

TABLE I POLISHED ROD VELOCITIES FOR FIGURE IX

No.	Pos. of Crank	Tangential Velocity of Beam (Curves)			
		No. 1	No. 2	No. 3	No. 4
1	0	0.60	0.61	0.60	0.62
2	15	1.40	1.42	1.42	1.44
3	30	2.10	2.11	2.15	2.12
4	45	2.65	2.65	2.82	2.65
5	60	2.80	2.78	2.92	2.75
6	75	2.90	2.86	3.02	2.72
7	90	2.80	2.75	2.92	2.61
8	105	2.50	2.47	2.61	2.33
9	120	2.00	1.99	2.08	1.98
10	135	1.45	1.45	1.49	1.45
11	150	0.70	0.705	0.72	0.71
12	165	+0.20	+0.20	+0.20	+0.21
13	180	-0.40	-0.41	-0.40	-0.41
14	195	-1.00	-1.02	-0.99	-1.031
15	210	-1.45	-1.46	-1.32	-1.47
16	225	-1.90	-1.90	-1.85	-1.92
17	240	-2.30	-2.28	-2.21	-2.31
18	255	-2.70	-2.65	-2.59	-2.70
19	270	-2.90	-2.85	-2.78	-2.90
20	285	-2.95	-2.90	-2.83	-2.95
21	300	-2.75	-2.73	-2.64	-2.76
22	315	-2.15	-2.15	-2.08	-2.16
23	330	-1.30	-1.32	-1.27	-1.33
24	345	-0.40	-0.41	-0.40	-0.41

occurred at the approximate center of the stroke. This is not the case for all pumping units especially those exhibiting unusual characteristics such as: gas lock, gas pound, plunger trouble, fluid pound, excessive friction, overtravel, undertravel, and synchronous speeds.

Several devices have been suggested and those assumed to be most suitable are: the voltmeter-generator principle of the electric tachometer, the liquid-filled-curved-tube steel-ball combination and the hydraulic cylinder-orfice-pressure gage arrangement. The theory of these devices and the manner in which they measure or indicate the amount of the variable will be taken up in the succeeding chapters.

The last variable to be considered is acceleration. The prospects of this variable as a means of indicating the counterbalance conditions are not particularly promising if the results of the velocity analysis graph, Figure IX are investigated further.

Since acceleration is the derivative of the velocity with respect to time (dv/dt) the maximum slope of the velocity curve indicates the maximum acceleration. It is noted that the maximum slope occurs where the beam is changing direction which is not the point of interest and further, the acceleration varies little if any from one curve to another. Therefore, acceleration was not considered.

As a result of this preliminary investigation the most promising variables appear to be torque measured by belt tension, and velocity. Since time was available to test only one of these devices fully, the variable, velocity, appeared to be the best from the standpoint of safety and ease in measurement.

CHAPTER VI

PRELIMINARY LABORATORY TESTS

This chapter deals with the equations which relate the results of the various devices with velocity and the equipment used to test these devices.

In the preceeding chapter, the difference between the maximum polished rod velocities was attributed primarily to the existing counterbalance conditions. This in general is not true because even when the angular velocity is constant, the intrinsic characteristics of the pumping unit cause the maximum polished rod velocities on the upstroke and downstroke to be different. This is due solely to the kinematics of the pumping unit. Figure X on the next page is the general type of pumping unit.

Table II shows the dependence of the velocity ratio (V_d max/V_u max) on the various parameters.

Thus with the aid of Table II, it is possible to predict whether the velocity ratio should be greater or less than one.

The National Supply D-45N-10DP Pumping Unit was the type used for the preliminary tests of the devices. The D-45N-10DP Pumping Unit illustrated in Figure XI has its specifications Listed in Table III. This unit was powered by a five horsepower induction type motor. In order to vary the maximum velocity of the polished rod or horsehead, the stroke was changed from 20 inches to 15 inches resulting in a significant change of maximum velocity be-

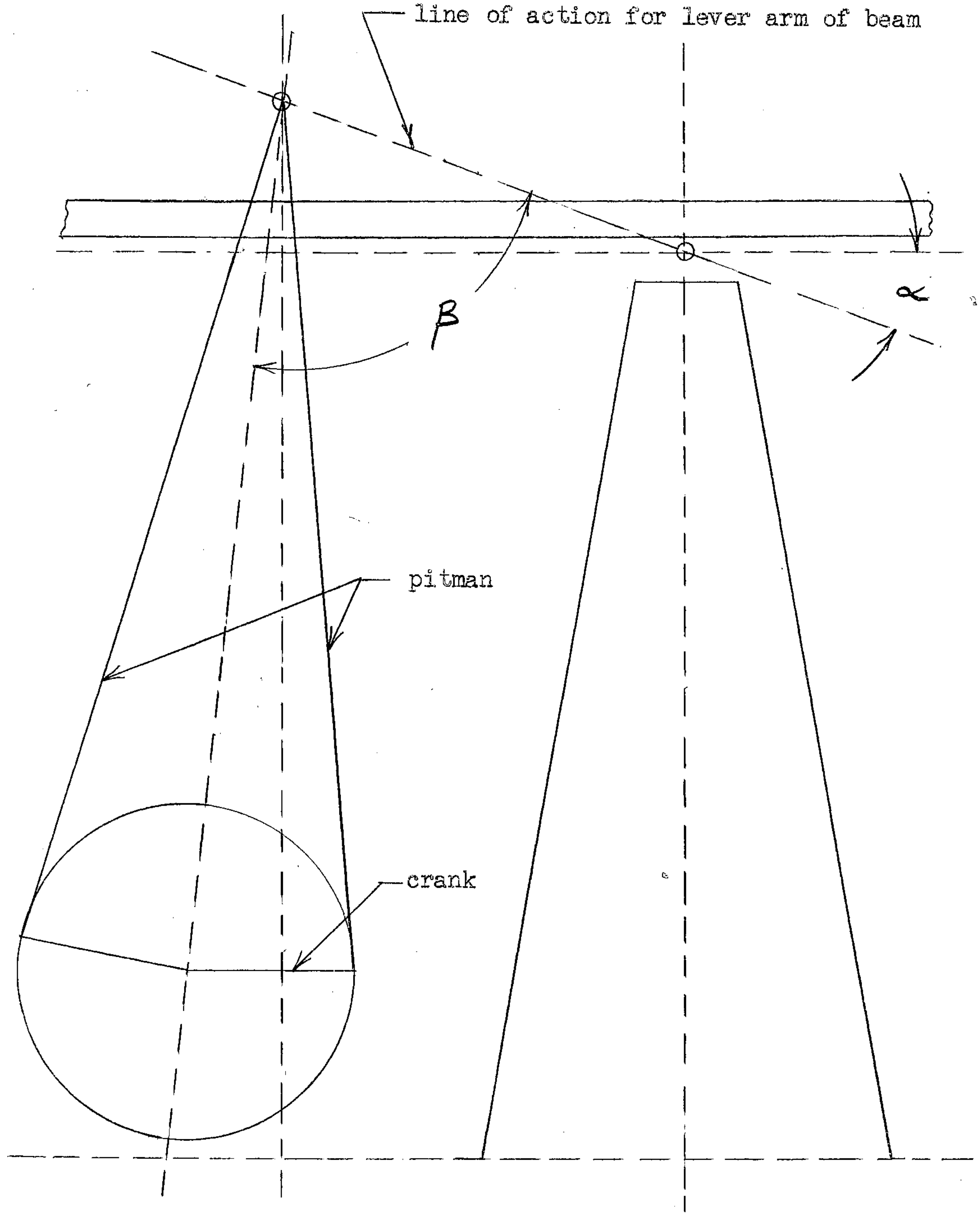

Figure X A General Pumping Unit

TABLE II EFFECTS OF PUMPING UNIT PARAMETER ON VELOCITY RATIO

No.	α (deg)	Θ deg	ϕ deg	Rotation	V_u/V_d
1	$=0$	$=90$	<90	CW	>1
2	$=0$	$=90$	$=90$	CW	$=1$
3	$=0$	$=90$	$=90$	CCW	$=1$
4	$=0$	$=90$	<90	CCW	<1
5	>0	$=90$	<90	CW	>1
6	>0	$=90$	$=90$	CW	$=1$
7	>0	<90	<90	CW	<1
8	>0	<90	$=90$	CW	<1
9	>0	$=90$	<90	CCW	>1
10	>0	$=90$	$=90$	CCW	$=1$
11	>0	<90	<90	CCW	<1
12	>0	<90	$=90$	CCW	<1
13	$=0$	>90	$=90$	CW	<1
14	$=0$	>90	>90	CW	<1
15	$=0$	>90	$=90$	CCW	>1
16	$=0$	>90	>90	CCW	>1

The angular velocity of the crank is assumed constant. All angles are measured when the beam is horizontal.

ϕ = Angle between crank and Pitman, deg

α = Angle between a horizontal line and the line of action for lever arm of the beam, deg

Θ = Angle between the line of action for the lever arm of the beam and a line through the crank shaft center and the upper Pitman bearing, deg

V_u/V_d Ratio of the maximum velocities on the upstroke and the downstroke

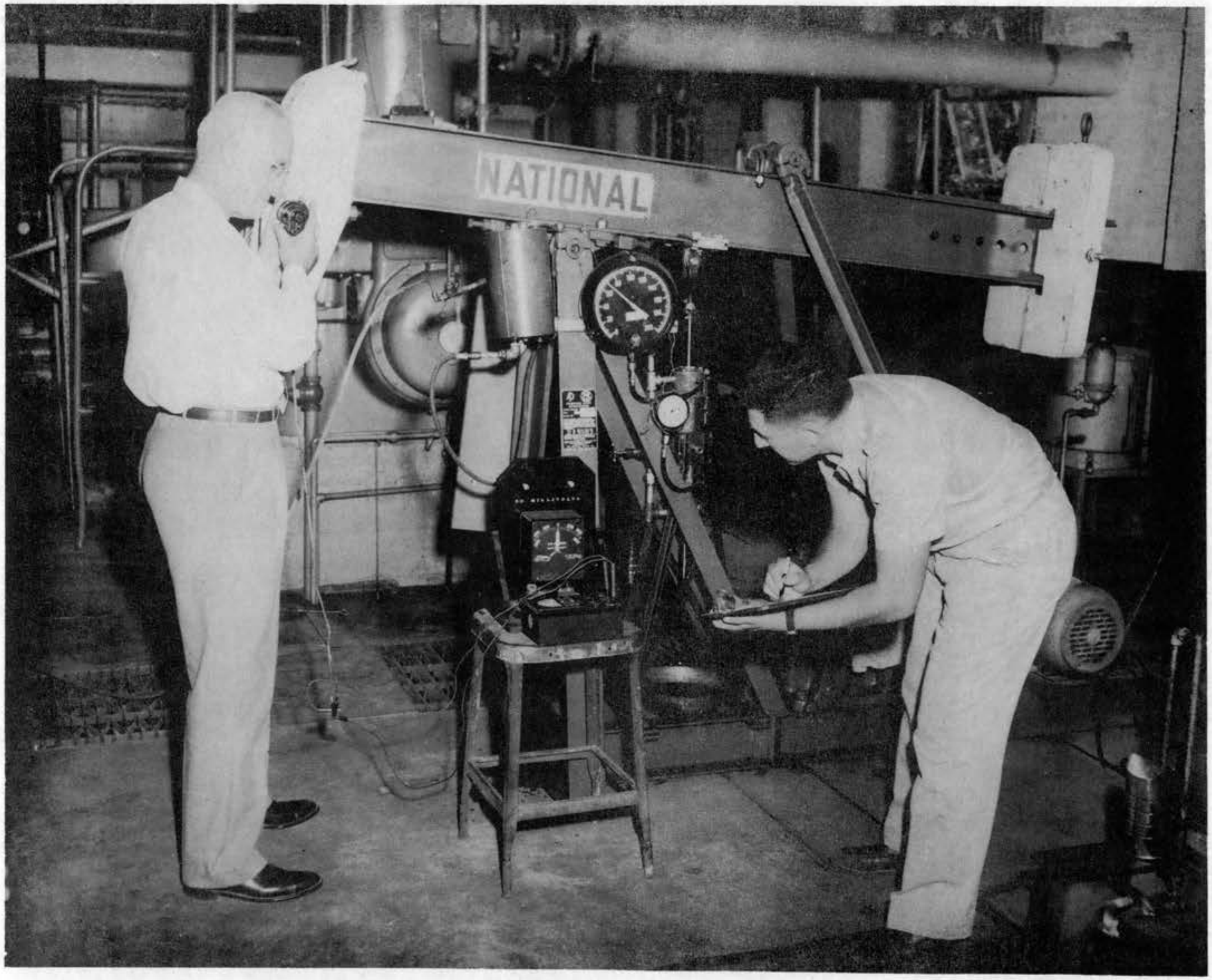

Figure XI. The D-45N-10DP Pumping Unit

TABLE III CONSOLIDATED SPECIFICATIONS OF

D-45N-10DP PUMPING UNIT [1]

API Walking Beam Rating lb.	4100
Walking beam section and weights	8 x $5\frac{1}{4}$"--17 lb
Polished rod strokes, inches	15, $17\frac{1}{4}$, 20
Bearings:	
Saddle, needle outer race assy. (2), shaft dia. x width, inches	2 x $1\frac{1}{4}$
Pitman, needle outer race assy. (2), shaft dia. x width, inches	$1\frac{1}{2}$ x $1\frac{1}{4}$
Wrist pin, self-aligning roller	
I.D. x O.D., inches	1.124 x 2.375
Width, inches	1.188
Wrist pins:	
Diameter at crank, inches	1 3/8
Overhang (centerline of bearing to face of crank), inches	1 5/16
Gear reducer (double reduction):	
API peak torque at 20 spm, in.- lb	10,000
Nominal horsepower rating at 20 spm	2.0
Overall gear ratio	29.5 to 1
Crankshaft diameter at bearing, in.	2 3/8
V-Belt sheave-pitch diameter by number and section of belt	$12\frac{1}{4}$"PD--3A
V-Belt sheave pitch diameter, maximum	14"PD
V-Belt sheave pitch diameter, minimum	5"PD
Oil capacity, gallons	2
Foundation bolts (not including extension base) number, size and length	8--3/4" x 18"
Weight regular equipment, less extension base and counterweights, lb	1338

[1] "National Beam Counterweighted Pumping Units For Light to Medium Loads," National Supply Company Bulletin No. 335 (Toledo, 1949).

tween the upstroke and downstroke. The maximum velocities for the 20 in. and 15 in. strokes were determined graphically. The ratios of the maximum velocity on the downstroke to the maximum velocity on the upstroke were 1.096 and 1.089 for the 20 in. and 15 in. stroke lengths respectively.

The first of the devices to be discussed is the hydraulic cylinder-orifice-pressure gage combination. The equation relating velocity and pressure drop is:[2]

$$Q = \frac{A_2\ C}{\sqrt{1 - \left(\frac{A_2}{A_1}\right)^2}} \sqrt{2g\ \frac{(P_1 - P_2)}{W}} \qquad 10.$$

$$Q = A_1 V = \frac{A_2\ C}{\sqrt{1 - \left(\frac{A_2}{A_1}\right)^2}} \left(\frac{2g}{W}\right)^{\frac{1}{2}} \sqrt{P_1 - P_2} \qquad 11.$$

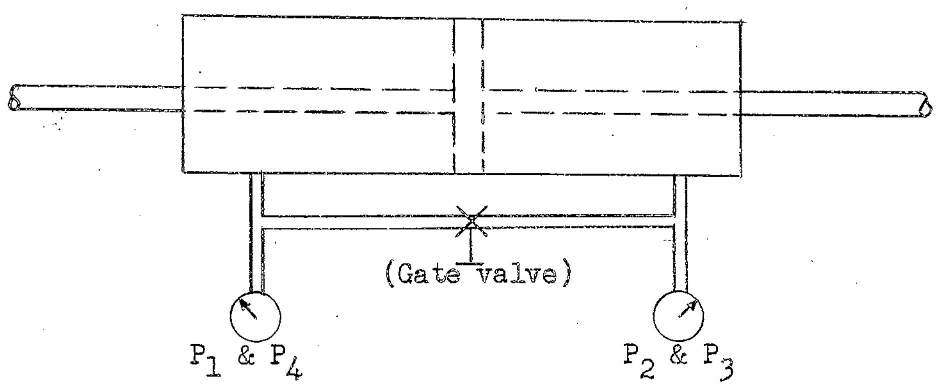

Figure XII Hydraulic Cylinder-Orifice Device

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[2] R. C. Binder, Fluid Mechanics (New York, 1950), p. 130

$$V = C\left(\frac{A_2}{A_1}\right)\left[1 - \left(\frac{A_2}{A_1}\right)^2\right]^{-\frac{1}{2}}\left(\frac{2g}{W}\right)^{\frac{1}{2}}\sqrt{P_1 - P_2} \qquad 12.$$

$$V = CK\sqrt{P_1 - P_2} \qquad 13.$$

$$\frac{V_u}{V_d} = \frac{C_u}{C_d}\left(\frac{P_1 - P_2}{P_3 - P_4}\right)^{\frac{1}{2}} \qquad 14.$$

Q = quantity rate of flow, cfs

C = discharge coefficient

A_1 = area of flow passage through gate valve, sq ft

A_2 = inside area of pipe connecting valve, sq ft

g = acceleration of gravity, fpsps

K = constant

P_1 = maximum pressure preceding orifice, upstroke, psf

P_2 = minimum pressure following orifice, upstroke, psf

P_3 = maximum pressure preceding orifice, downstroke, psf

P_4 = minimum pressure following orifice, downstroke, psf

C_u = discharge coefficient, upstroke

C_d = discharge coefficient, downstroke

W = specific weight of the fluid, lb per cu ft

The device in Figure XII was mounted on the walking beam and tested. The data and results from these tests are given in Table IV. It should be noted that the term "graph" used in Tables IV, V, and VI refers to the ratio of the maximum velocities obtained graphically for the pumping unit. The term "H.C." Table IV refers to the results of equation 14.

TABLE IV DATA AND RESULTS FROM HYDRAULIC CYLINDER DEVICE

Run	Stroke	Length	P_1	P_2	P_3	P_4	Velocity Ratio V_d/V_u Graph	H.C.	Error
1	Up	20	75	58			1.097	1.163	6.12
	Down	20			68	45			
2	Up	15	67	57			1.088	1.182	8.55
	Down	15			62	48			

$$\frac{V_{d_1}}{V_{u_1}} = \left(\frac{68 - 45}{75 - 58}\right)^{\frac{1}{2}} = \sqrt{1.353} = 1.163 \qquad 15A.$$

$$\frac{V_{d_2}}{V_{u_2}} = \left(\frac{62 - 48}{67 - 57}\right)^{\frac{1}{2}} = \sqrt{1.4} = 1.182 \qquad 15B.$$

It should be noted that the gate valve setting remained fixed for these two runs. The increase in error for a slight decrease in the velocity ratio may be attributed to two factors. The first is that it was impossible to remove all the air from the system although this was somewhat counteracted by pressurizing the system. The second is that the discharge coefficients (C_u) or (C_d) are not constant and their variation may be nonlinear.

Due to the unsatisfactory results of the laboratory tests on this device, and the difficulties that would be encountered in any attempt to conduct field tests on it, the tests on this device were discontinued.

The second device considered was the liquid-filled curved-tube steel-ball combination in Figure XIII. The combination operates on the principle that an increase in the velocity of the curved tube relative to the ball causes an increase in drag. This increase in drag causes the ball to be displaced from its lowest position an amount which is a function of the various forces acting in the system. The position of the ball is influenced by the following factors; time, displacement of the tube, the relative velocity of the tube with respect to the ball, the acceleration of the tube, gravity, inertia,

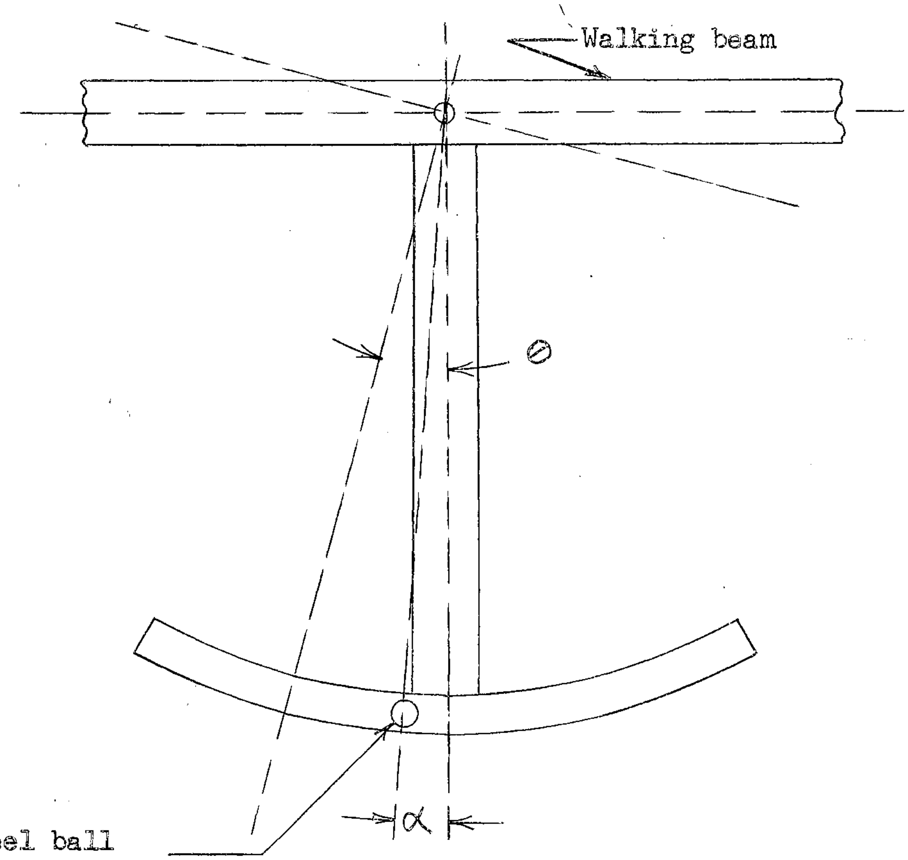

Figure XIII Liquid-Filled Curved-Tube Steel-Ball Device

θ = equals the angular position of beam, deg.

α = equals angular position of the ball, deg.

and the relation between drag and velocity. Some of the factors might have a diminishing effect as the system stabilized. A complete analysis of the motion of ball was considered to be beyond the scope of this investigation. On the basis of a preliminary analysis it seems probable that the displacement of the ball in the tube is not a linear function of the maximum tangential velocity of the tube.

A scale was attached to the glass tube and the steel ball positions were recorded by using the scale readings. The reference point was arbitrarily chosen as the position of the ball under static conditions with the beam at the mid-point of the stroke. The reference points for the 20 in. and 15 in. strokes were 0.45 ft and 0.522 ft respectively. Therefore with scale readings of 0.60 and 0.10 for the 20 in. stroke the relative positions are 0.15 and 0.35 respectively.

TABLE V DATA AND RESULTS FROM

LIQUID-FILLED-CURVED-TUBE STEEL-BALL DEVICE

			Position of ball		Velocity Ratio V_d / V_u		
Run	Stroke	Stroke length	Scale readings ft	Relative Scale readings ft	Graph	C.T.	Error
1	Down Up	15 15	0.69 0.25	0.168 0.272	1.088	0.62	43.0%
2	Down Up	20 20	0.60 0.10	0.15 0.35	1.097	0.43	60.7%

The results indicate that the error is not consistent and that the maximum ball displacement is not proportional to the maximum velocity. It was suggested that this equipment should be tested in the field since it was easily installed.

The last of the combinations to be discussed is the voltmeter-generator setup. This device is used to measure the maximum polished rod velocity by generating a voltage which is proportional to it. The voltage generated is a function of the flux and the angular velocity of the generator shaft.

$$\text{Velocity} = \frac{B}{\varphi}(\text{volts}) \qquad 16.$$

φ = flux density
B = constant

If the generator is composed of permanent magnets the flux is constant and the maximum voltage generated is proportional to the velocity.

$$\text{Velocity} = K(\text{Volts}) \qquad 17.$$

The validity of these conclusions was verified by the results of the preliminary laboratory tests as shown in Table VI.

TABLE VI DATA AND RESULTS FROM THE VOLTMETER-GENERATOR DEVICE

Run	Stroke	Length	Volts	Velocity Ratio V_d/V_u		
				Graph	V. G.	Error
1	Up Down	20 20	1.0 1.1	1.097	1.100	0.26%
2	Up Down	15 15	0.75 0.82	1.088	1.093	0.45%

This device appeared to be the best because it has the smallest amount of percentage error and was considered entirely worthy of further tests.

CHAPTER VII

FIELD TESTS

The purpose of this chapter is to examine the qualities of the various combinations thought to be most promising, namely, the liquid-filled-curved-tube steel ball combination and the generator voltmeter combination.

The tests are divided into two parts. The first test was to verify the assumption, that with a change in counterbalance condition there would be a corresponding change in the velocity ratio. The voltmeter-generator combination was used because it was found to be the most accurate.

The second test employed both combinations and was designed to compare the actual values of velocities under varied conditions with those obtained from the combinations (referred to as devices in the remaining portions of this manuscript).

The site for the first field test was at the Stanolind Oil and Gas Company's Field near Lucien, Oklahoma. The unit was a 320D OTC-29S-Bethlehem Series 50 Pumping Unit with specifications as shown in Table VII. These specifications were taken from the Bethlehem Pumping Unit Engineering Data sheet.

TABLE VII SPECIFICATIONS

OF

320D-29S PUMPING UNIT

Model number	320D-29S
Rating	29,500 pounds
Serial number	S-196
Peak torque	320,000 ft lb @20 SPM
Horsepower	64.7 @ 20 SPM
Stroke	86 inches
Gear reduction	29.6 to 1
Walking Beam center to front and center to rear	10 feet 9 inches
Engine	
Manufacturer	Minneapolis-Moline
Model	403

Effective counterbalance for position	Pounds
0	210
1	2,500
2	4,800
3	7,100
4	9,400
5	11,700
6	13,990
7	16,290
8	18,590
9	20,890
10	23,190

The well load was composed of:

Sucker Rod

30 - 1"Dia - 63#/sec = 1890
78 - 7/8" - 51#/sec = 3978
92 - 3/4" - 42#/sec = 3864

Sucker Rod Load - - - - - - - - 9732 (Static)

Depth of well - - - - - - - - - 5078 Feet

Weight of Fluid = W_f

$$W_f = \left[\frac{(D_1{}^2 - D_2{}^2)}{(144)}\right]\left(\frac{\pi}{4}\right)(D_3)(D_4)$$

Where

D_1 = Diameter of plunger, in.
D_2 = Average diameter of sucker rod, in.
D_3 = Depth of well, ft.
D_4 = Density of fluid, lb/cu.ft.

$$W_f = \left[\left(\frac{(2)^2-(0.842)^2}{144}\right)\right]\left(\frac{\pi}{4}\right)(5078)\ (62.4)$$

$$= 5700 \text{ Lb}$$

$$\text{Polished Rod Load} = w_f + w_R\left[1 + \frac{N^2L}{70,500}\right]$$

$$= 5700 + 9800\left[1 + \frac{(21)^2\ (86)}{70,500}\right]$$

$$= 20,700 \text{ lb}$$

Therefore, the pumping unit was operating at approximately two-thirds of the rated load. The data and results from the voltmeter-generator device tested at Lucien, Oklahoma are listed in Table VIII.

The results of this test were not conclusive but two important facts were noted. First, the generated voltage may vary some with different operators. This is due to the effect of the normal force

TABLE VIII DATA AND RESULTS OF LUCIEN FIELD TESTS

Run	Up-Stroke Volts	Down-Stroke Volts	C.B. Notes	V_u/V_d	Approx. C.B. lbs	Generator Operator
1	11.7	10.7	Field Correct	1.093	11,700	No. 1
2	12.5	11.25	Field Cor.	1.112	11,700	No. 2
3	12.5	11.5	Over C.B.	1.088	16,290	No. 1
4	12.5	11.25	Over C.B.	1.112	16,290	No. 2
5	12.5	11.25	Over C.B.	1.112	16,290	No. 2
6	11.7	11.0	Over C.B.	1.063	16,290	No. 1
7	11.6	11.0	Over C.B.	1.054	16,290	No. 1
8	12.1	11.0	Over C.B.	1.100	16,290	No. 1
9	12.6	11.4	Over C.B.	1.105	16,290	No. 2
10	12.4	11.6	Under C.B.	1.070	7,100	No. 2
11	11.5	10.7	Under C.B.	1.072	7,100	No. 1
12	12.6	11.7	Under C.B.	1.085	7,100	No. 2
13	11.9	11.0	Field Cor.	1.082	11,700	No. 2

All runs taken at 21 Strokes per minute
Runs 1-7 governor was fixed
Runs 8-13 governor was operative

between the polished rod and the rubber rim of the friction wheel on the effective diameter of the friction wheel. Second, the velocity ratio did change with a change of counterbalance. The reason that only a small change of velocity ratio was obtained with an appreciable change in the counterbalance conditions was attributed to the "flywheel effect" of the pumping unit. This "flywheel effect" becomes appreciable when a heavy duty pumping unit is lightly loaded.

Pumping Unit No. 31 of the Grief Creek Booch Sand Unit Field, located east of Holdenville, was chosen for the second test. This unit was tested with the cooperation of the Stanolind Oil and Gas Company on June 16, 1954. Table IX on the next page, contains the specifications of this pumping unit.

In evaluating these devices fully several questions presented themselves such as: Where does the maximum or minimum load occur? Does the maximum velocity on either stroke occur near the maximum or minimum loads positions? Does the average of the extreme loads ($\frac{1}{2}$ maximum load plus $\frac{1}{2}$ minimum load) approximately equal the correct counterbalance? What happens when the governor is free and when the governor is fixed? What is the effect of a change in the number of strokes per minute on the counterbalance requirement? What is the effect of time on the stabilization of the well condition?

During the test of the pumping unit fourteen dynamometer cards were taken. The field test data, and corresponding data from the dynamometer cards are tabulated in Table X.

TABLE IX SPECIFICATIONS

OF

80D-14SA Pumping Unit

Manufacturer	Bethlehem
Model	80D-14SA-50 Series
Peak Torque	80,000 in. lb @ 20 SPM
Horsepower	16.2 (old API) @ 20 SPM
Gear Ratio	24.492 to 1
Walking Beam Capacity	14,830 lb. API
Operational Stroke	34 inches
Counterbalance	Rotary
Filler Weights	None
Engine	
Manufacturer	Allis Chalmer
Model	B125
Bore	3.375 inches
Stroke	3.50 inches
Number of Cylinders	4
Serial Number	PU 92435B

Effective counterbalance for position	Pounds
0	1,420
1	2,680
2	3,930
3	5,190
4	6,450
5	7,700
6	8,960
7	10,210
8	11,470
9	12,730

TABLE X DATA FROM THE GRIEF CREEK BOOCH SAND UNIT FIELD TEST

Run	Card	V.M.#1 Up	V.M.#1 Down	SPM	V.M.#2 Upstr.	V.M.#2 Down	CB	Gover-nor	Time	Curved Tube up	Curved Tube Dn. Midpt-.51
1	3	4.75	4.25	20	---	---	4 @ 3+1 1/8"	Free	-----	0.31 0.32	0.68
2	4	4.25	3.75	17.5	---	---	4 @ 3+1 1/8"	Fixed	-----	0.25	0.71
2'	4	4.25	3.60	17.5	---	---	4 @ 3+1 1/8" 2@3+1 1/8"	Fixed	-----	0.27	0.72
3	5	5.75	5.0	23	---	---	2 @ 4 2@3+1 1/8"	Free	-----	---	---
4	6	4.9	4.5	20	---	---	2 @ 4	Free	-----	0.32	0.69
5	7	5.0	4.5	20	---	---	4 @ 4	Free	-----	0.34	0.69
5'	-	5.0	4.5	20	---	---	4 @ 4	Free	----- SU 1:28	0.35	0.69
6	8	5.0	4.25	20	---	---	4 @ 5	Free	SD 1:15	0.35	0.70
6'	-	5.1	4.35	20	---	---	4 @ 5	Free	MT 1:34	0.37	0.71
6"	-	5.1	4.35	20	---	---	4 @ 5	Free	----- SD 1:55	---	---
7	9	4.6	5.2	20	5.10	5.40	4 @ 1	Free	SU 2:10 SD 2:18	0.25	0.63
8	10	4.8	5.1	20	5.20	5.20	4 @ 2	Free	SU 2:25 SD 2:43	0.30	0.65
9	11	4.9	4.5	20	5.30	4.60	4 @ 3	Free	SU 2:50 SD 3:05	0.32	0.68
10	12	4.9	4.5	--	5.30	4.70	2 @ 3½ 2 @ 4	Free	SU 3:10 SD 3:13	0.34	0.69
10'	13	---	---	--	----	----	2 @ 3½ 2 @ 4	Free	SU 3:40	---- 0.32	---
11	14	4.8	4.4	20	5.30	4.60	4 @ 3½	Free	SU 4:25	0.33	0.69

TABLE XI DATA FROM DYNAMOMETER CARDS

Run No.	Card No.	Max. Load (lb)	Loc. Max. Load Deg.	Min. Load (lb)	Loc. Min. Load Deg.	Max. Load at 90°	Min. Load at 90°	A_1	A_2	Cal. CB	Actual CB	V_u / V_d
1	3	8969	82	2501	274	8886	2792	1.435	0.520	6280	5523	1.125
2	4	8969	93	2957	283	8928	3179	1.410	0.525	5700	5666	1.130
3	5	9715	77	2474	283	9468	2971	1.410	0.570	5830	5995	1.156
4	6	8886	87	2681	274	8789	2861	1.410	0.475	5500	5995	1.103
5	7	8955	90	2723	265	8955	2736	1.455	0.500	5900	6450	1.112
6	8	9052	87	2598	267	9011	2460	1.460	0.495	5920	7700	1.176
7	9	9121	93	2184	265	9121	2322	1.365	0.605	5670	1210	-----
8	10	9176	87	2460	270	9149	2515	1.410	0.556	5780	2180	0.939
9	11	9052	82	2653	278	8720	2709	1.450	0.475	5775	3930	1.080
10	12	8803	80	2598	275	8776	2792	1.405	0.485	5540	6185	1.079
10'	13	8900	80	2653	268	8734	2708	1.425	0.502	5930	----	1.090
11	14	8900	80	2557	277	8790	2902	1.425	0.550	5840	5820	1.091

The results of the dynamometer cards must be calculated before any questions can be answered. The numerical values were obtained from the cards with the aid of a Telecomputing Telereader which used $2\frac{1}{2}$ power lenses and a digital unit which measured 843 electronic counts per magnified inch. A timing wave generated from a tuning fork with a natural frequency of 10 cycles per second, was recorded on each dynamometer card. The displacement of each half cycle along the X axis provided a means of determining the maximum velocities of the upstrokes and downstrokes.

The values of calculated counterbalance were obtained from Formula 18.[1]

$$\text{Correct Counterbalance} = \frac{C(A_1 + A_2/2)}{L} \qquad 18.$$

C = Calibration constant of dynamometer card - lb/in. of height of card.
A_1 = Area below load curve, sq in.
A_2 = Area enclosed by load curve, sq in.
L = Length of card, in.

This formula results in the correct amount of counterbalance based on the criterion that the work done on the upstroke should equal the work done on the downstroke. The counterbalance in this equation is calculated by computing the mean ordinate of the dynamometer card and converting it to pounds. This equation neglects the shape of the dynamometer card and in some cases may give unsatisfactory results. In order to use the criterion of equal peak torques, another means was employed to determine which card repre-

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[1] J. Zaba, Oil Well Pumping Methods (Tulsa, 1943) P. 42

sented the best counterbalance conditions, eliminating the necessity of a complete analysis for each card. This method is solely used as an auxiliary method, but it determines rapidly the cards which approach correct counterbalance.

Given the effective counterbalance and the dynamometer card, a line which represents the effective counterbalance can be either drawn or superimposed on the card. The portion of the upstroke curve below the line corresponds to negative torque and the portion of the curve above the line on the downstroke also represents negative torque. Negative torque is detrimental to the pumping unit. The maximum torques may be also roughly compared. This is done by comparing the maximum vertical distance above the effective counterbalance line on the upstroke with the vertical distance at the crank angle 90° and selecting the one corresponding to the greater torque. This distance approximately represents the difference between the load at which peak torque occurs and the counterbalance. This procedure is repeated for the downstroke except the distances below the counterbalance line are used, and the maximum vertical distance is compared with the vertical distance at the crank angle 270°. The two selected distances should be approximately equal for equal peak torques. These maximum distances above and below the counterbalance do not always occur at either 90° or 270° ; therefore, the angular position of the crank must be taken into account when viewing the results as listed in Table XII.

Based on the result of Table XII the following cards (1, 2, 3, 4, 7, 8, 9, 10, 11, 12) were eliminated from further consider-

TABLE XII LOCATIONS OF NEGATIVE TORQUE ON DYNAMOMETER CARDS

Card No.	Negative Torque From (deg)	Negative Torque To (deg)	$\frac{1}{2}(F_u - F_d)$
1	310	360	$> C_e$
1	180	230	
2	315	360	$> C_e$
2	220	235	
3	160	180	$> C_e$
3	320	360	
4	320	360	$> C_e$
5	20	35	$\approx C_e$
5	155	180	
6	325	360	$\approx C_e$
6	170	180	
7	30	45	$< C_e$
7	145	180	
8	0	65	$\ll C_e$
8	115	180	
9	All negative torque on downstroke		$\gg C_e$
10	All negative torque on downstroke		$\gg C_e$
11	300	360	$\gg C_e$
11	180	260	
12	160	180	$< C_e$
13	155	180	$\approx C_e$
14	330	360	$\approx C_e$
14	165	180	

ation as the card which represents the best counterbalance conditions. This elimination is a result of either the presence of a significant amount of negative torque or because the average of the maximum and minimum loads does not equal the effective counter balance.

To determine the quality of counterbalance, the torque may be used as a criterion. Two requirements of a dynamometer card which indicate best counterbalance conditions are: the peak torques should be equal, and there should be no negative torque. It is realized that these conditions cannot always be met for every card because of the uniqueness and pecularities of each card, but in the limit as counterbalance conditions approach the best, the above requirements will be approached.

The above requirements may be easily checked by the following method. First, the effective counterbalance for a particular card is drawn to the scale of the card and used as an overlay for the card to determine the existence and location of any portion of of the curve which would indicate negative torque. Second, the load and location at which the peak torque occurs must be determined. Assuming that Figure XIV represents a typical dynamometer card, the peak torques can be determined by using equations 19 and 20. There are only two places the peak torque can occur on the upstroke for Figure XIV and at those two points the torques are compared to find the maximum torque on the upstroke. The same procedure is used for the downstroke.

$$T_1 = \frac{S}{2}(X_1 - CB_A)\sin 90^\circ \tag{19A.}$$

$$T_3 = \frac{S}{2}(X_3 - CB_A)\sin \alpha \tag{19B.}$$

$$T_u = \frac{S}{2}(X_u - CB_A)\sin \Theta \tag{19C.}$$

$$T_2 = \frac{S}{2}(X_2 - CB_A)\sin 270^\circ \tag{20A.}$$

$$T_4 = \frac{S}{2}(X_4 - CB_A)\sin \beta \tag{20B.}$$

$$T_d = \frac{S}{2}(X_d - CB_A)\sin \phi \tag{20C.}$$

where X_1 = Load at 90°, lb

X_2 = Load at 270°, lb

X_3 = Maximum load, lb

X_4 = Minimum load, lb

S = stroke length, in.

X_u = Load at which peak torque occurs on the upstroke, lb

X_d = Load at which peak torque occurs on the downstroke, lb

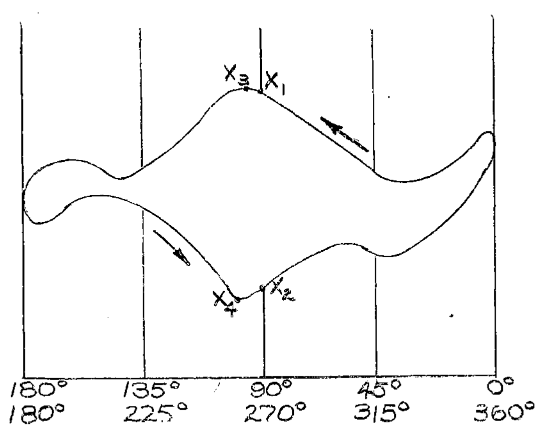

Figure XIV Typical Dynamometer Card

α = Angular position of crank at maximum load, deg

β = Angular position of crank at minimum load, deg

Θ = Angular position of crank at which peak torque occurs on the upstroke, deg

ϕ = Angular position of crank at which peak torque occurs on the downstroke, deg

CB_A = Actual effective counterbalance, lb

CB_c = Calculated counterbalance, lb

Equations 19 and 20 are only approximate, since they assume simple harmonic motion at the beam thus neglecting the angularity of the pitman. It should be noted that the peak torque on the upstroke may not occur at a point of maximum load or at 90°, if there are any peak loads present between $90^{\circ} = \Theta$. Therefore if such peak loads are present they should be considered in the peak torque determination. The same is true for the downstroke.

Therefore, without negative torque the calculated counterbalance can be found by equating the two peak torque equations 19C and 20C, using the criterion at equal peak torques for proper counterbalance.

$$T_u = T_d = \frac{S}{2}(X_u - CB_A)\sin\Theta = \frac{S}{2}(X_d - CB_A)\sin\phi$$

solving for counterbalance.

$$CB_c = \frac{X_u \sin \quad - X_d \sin\phi}{\sin\Theta - \sin\phi} \qquad 21.$$

In order to verify these equations, card fourteen was chosen and the value of the counterbalance was calculated from equation 21.

X_1 = 8790 lb at 90°

$X_2 = 2902$ lb at $270°$

$X_3 = 8900$ lb at $80°$

$X_4 = 2557$ lb at $277°$

$S = 34$ in.

$CB_A = 5820$ lb

$T_1 = (17) \quad [8790 - 5820] \quad (\sin 90°) = 50{,}490$ lb-in.

$T_2 = (17) \quad [2902 - 5820] \quad (\sin 270°) = 49{,}506$ lb-in.

$T_3 = (17) \quad [8900 - 5820] \quad (\sin 80°) = 51{,}867$ lb-in.

$T_4 = (17) \quad [2557 - 5820] \quad (\sin 277°) = 55{,}340$ lb-in.

Therefore, $T_u = T_3$ and $T_d = T_4$. Substituting for X_u, X_d and the angles in equation 21, then the counterbalance equals

$$CB_c = \frac{8900 \sin 80° - 2557 \sin 277°}{\sin 80° - \sin 277°}$$

$$CB_c = 5777 \text{ lb}$$

The actual counterbalance is equal to 5820 pounds as compared to the 5840 pounds obtained from the area method. The difference between results of the peak torque method and the area method may be due to the fact that area method does not take into account the shape of the load curve and the peak torque method assumed simple harmonic motion. The result of the peak torque method (5777 lb.) may also be compared to the result of equation 22. This equation assumes that $\sin\theta = -\sin\phi$ which is not the general case.

$$CB_c = \frac{X_u - X_d}{2} \qquad 22.$$

$$= \frac{8900 + 2557}{2}$$

$$= 5729 \text{ lb}$$

Equation 22 produced a value for calculated counterbalance which was 58 pounds less than the general results of the peak torque method (5777 lb). The error in equation 22 is further increased when the angular positions of the crank for peak torques are such that the difference between $\sin\theta$ and $-\sin\phi$ becomes greater.

For the purpose of comparing results obtained from the various cards, a particular card must be chosen which represents the best counterbalance conditions from the standpoint of equal peak torques, a minimum of negative torque and mean ordinate. The results of this comparison are tabulated in Table XIII. Therefore, the counterbalance conditions represented by card fourteen are nearest to those of a correctly counterbalanced pumping unit. This card will be used as the card representing the best counterbalance conditions.

Before proceeding further, the questions mentioned before in this chapter will be answered in the light of the data and results obtained from the cards.

The maximum and minimum loads occurred in an interval which was plus or minus 13 degrees from the midpoint of the card.

The maximum velocities did not coincide with the extremes of

TABLE XIII RESULTS OF BEST COUNTERBALANCED CARDS

Card No.	A	B	C	B-A	B-C	Peak Torque			
						Upstroke		Downstroke	
						Load	Loc.	Load	Loc.
5	6,094	6,058	5,830	- 36	-228	9,715	77°	2,474	283
6	5,788	6,058	5,500	-270	-558	8,886	87°	2,681	274
13	5,734	———	5,930	———	———	8,900	80°	2,653	268
14	5,777	5,820	5,840	- 43	- 20	8,900	80	2,557	277

A - - - - - Calculated Counterbalance (Peak Torque Method)

B - - - - - Actual Counterbalance

C - - - - - Calculated Counterbalance (Area Method) the loads. They appeared to be present on the low side of the midpoint of the downstroke and on the high side of the upstroke.

The two effects of increasing the number of strokes per minute are that the maximum velocity increases with an increase in strokes per minute, and the counterbalance requirements increase because of the increased loads encountered. Examples of this are cards 5 and 6. The difference between cards 5 and 6 is that during the time the cards were recorded the pumping unit was operating at 23 and 20 strokes per minute respectively. The maximum load on card 5 was 9715 pounds as compared to a maximum of 8886 pounds on card 6.

The well stabilization was no problem in this test because within fourteen minutes after counterbalance adjustment, the voltmeter-generator device indicated identical voltages for consecutive readings.

The predominate effect of fixing the governor for a particular setting of the throttle was that it decreased the strokes per minute which in turn decreased the counterbalance requirements.

The results of the voltmeter-generator device and the curved-tube steel-ball device are listed in Table XIV and are compared to the maximum velocities of the polished rod determined from the tuning fork displacement curve.

The displacement of the steel ball in the tube was measured with respect to an arbitrarily chosen reference point on the tube. On the laboratory tests the reference point was the ball position under static conditions when the beam was at the midpoint of its travel. On the field tests the reference point was the position of the ball under static conditions when the beam was horizontal. The scale reading at this point was 0.51 ft. The ratio of the maximum displacement of the ball on the upstroke to the maximum displacement of the ball on the downstroke was determined for each run.

The ratio of the maximum steel ball positions relative to the point, 0.51 ft, are tabulated in Table XIV. These ratios are plotted against existing counterbalance in Figure XV, and are also plotted with respect to the out-of-balance conditions in Figure XVI. The variation of the ratio with a significant change of counterbalance, in the region of proper counterbalance seems to be too small for this device to be a dependable indicator of counterbalance. The curves in Figures XV and XVI exhibited the predicted non-linear characteristic of the device. This is evident from the error obtained when comparing the ratios of the maximum steel ball positions with the ratios of the maximum polished rod velocities. The error varied from a minimum of 2.04% to a maximum of 59.70%. The curves in Figures XV and XVI are similar and are approximately exponential in form.

The results for the ratio of the maximum voltages from the voltmeter-generator device are tabulated in Table XIV. These results are plotted with respect to the existing counterbalance in Figure XVII and with respect to the out-of-balance conditions in

Figure XVIII. The curves in Figures XVII and XVIII are similar in shape, and have a particular characteristic. This characteristic is the definite decrease in the velocity ratio in the region of proper counterbalance. This is a 3% decrease in the velocity ratio as compared to the maximum velocity ratio for the under counterbalanced condition. It seems possible that this low point on the curve in the region of proper counterbalance might be used as an indicator of proper counterbalance. It is recognized, however that a small error in the velocity ratio corresponds to a considerable change in the corresponding counterbalance as shown on the curve. The velocity ratio actually changed by 0.2317 with a change in counterbalance of 5,500 pounds.

Therefore the use of this device as a means indicating counterbalance conditions and counterbalancing is rather doubtful for two reasons. The first is that the variation in the ratio of maximum velocities with counterbalance is too small. The second is that the results of this test are not conclusive because only one well was tested, and it is not known if any other well will exhibit the same characteristics for proper counterbalance.

The voltmeter-generator device seems to be best suited for indicating counterbalance condition because the voltage is a linear function of velocity. The application of the curved tube device is limited because of its inherent characteristic to amplify most ratios. One feature of the curved tube device is that each ratio is distinct and a particular ratio can indicate only one counterbalance condition.

TABLE XIV RESULTS OF THE VOLTMETER AND CURVED TUBE DEVICES

Run No.	Card No.	Dynamometer Card			Voltmeter Generator Device				Curved Tube Device	
		V_u/V_d	(N)Lb	Spm	V_u/V_d #1-A	#1-B	#2	Error, %	Ratio	Error, %
1	3	1.125	-757	20	1.118	----	----	0.62	1.147	2.04
2	4	1.130	-177	17.5	1.133	1.181	----	0.27	1.218	7.80
3	5	1.156	-165	23	1.150	----	----	0.38	----	----
4	6	1.103	-495	20	1.089	----	----	1.27	1.055	5.30
5	7	1.112	-550	20	1.111	1.11	----	0.09	0.915	17.58
6	8	1.176	-1780	20	1.176	1.172	1.221	0.33	0.770	34.60
7	9	----	-4460	20	0.885	----	0.944	----	2.165	----
8	10	0.939	-3600	20	0.941	----	1.000	0.21	1.500	59.70
9	11	1.080	-1845	20	1.089	----	1.152	0.83	1.118	3.52
10	12	1.079	-635	--	1.089	----	1.127	0.93	0.945	12.40
10	13	1.090	----	--	----	----	----	----	----	----
11	14	1.091	-20	20	1.091	----	1.152	0.00	1.028	6.87

N = Existing Counterbalance - Required Counterbalance

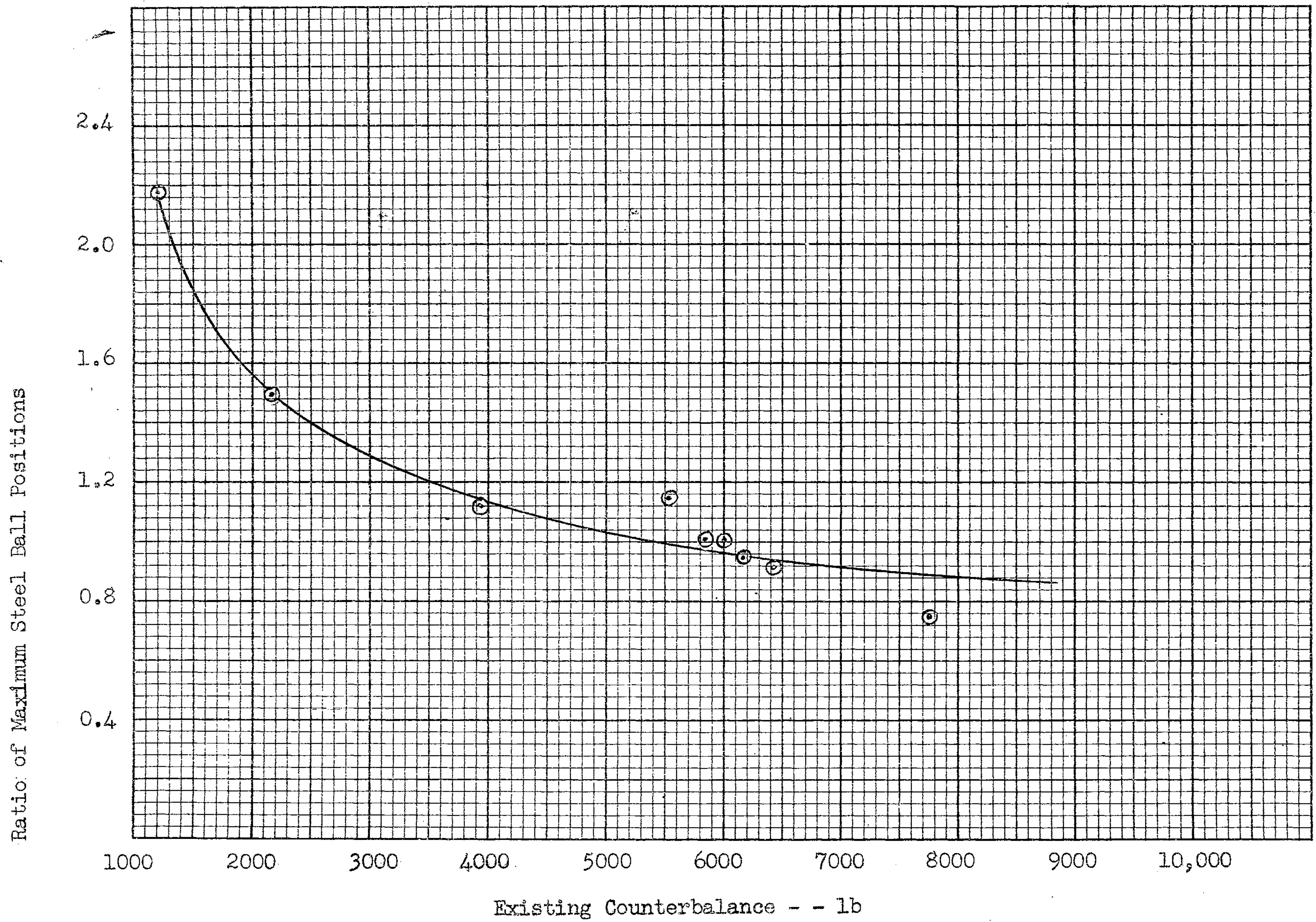

Fig XV -- Variation of Steel Ball Position Ratios With Counterbalance Conditions

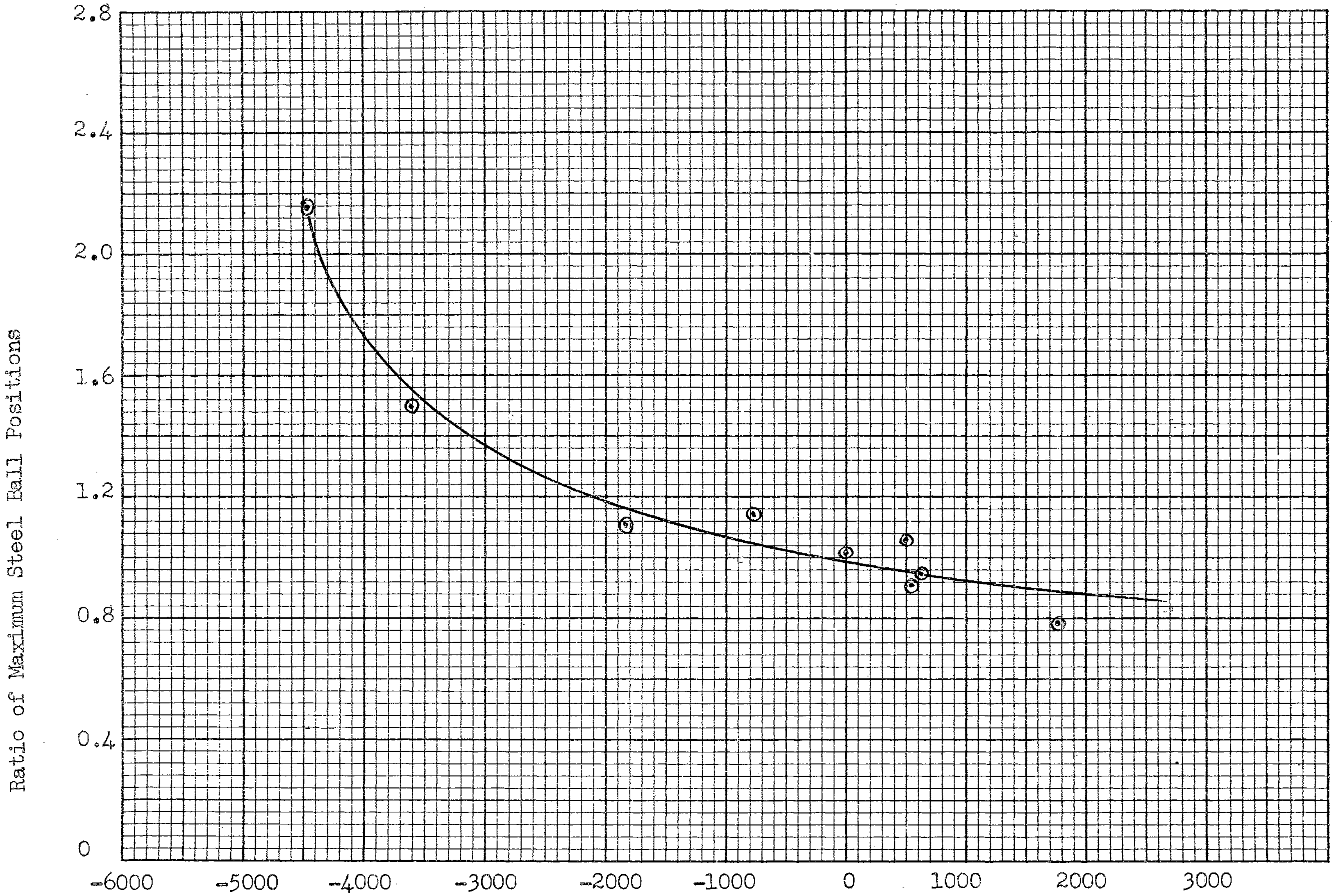

Fig. XVI -- Variation of Steel Ball Position Ratios With Difference Between Existing and Required Counterbalance

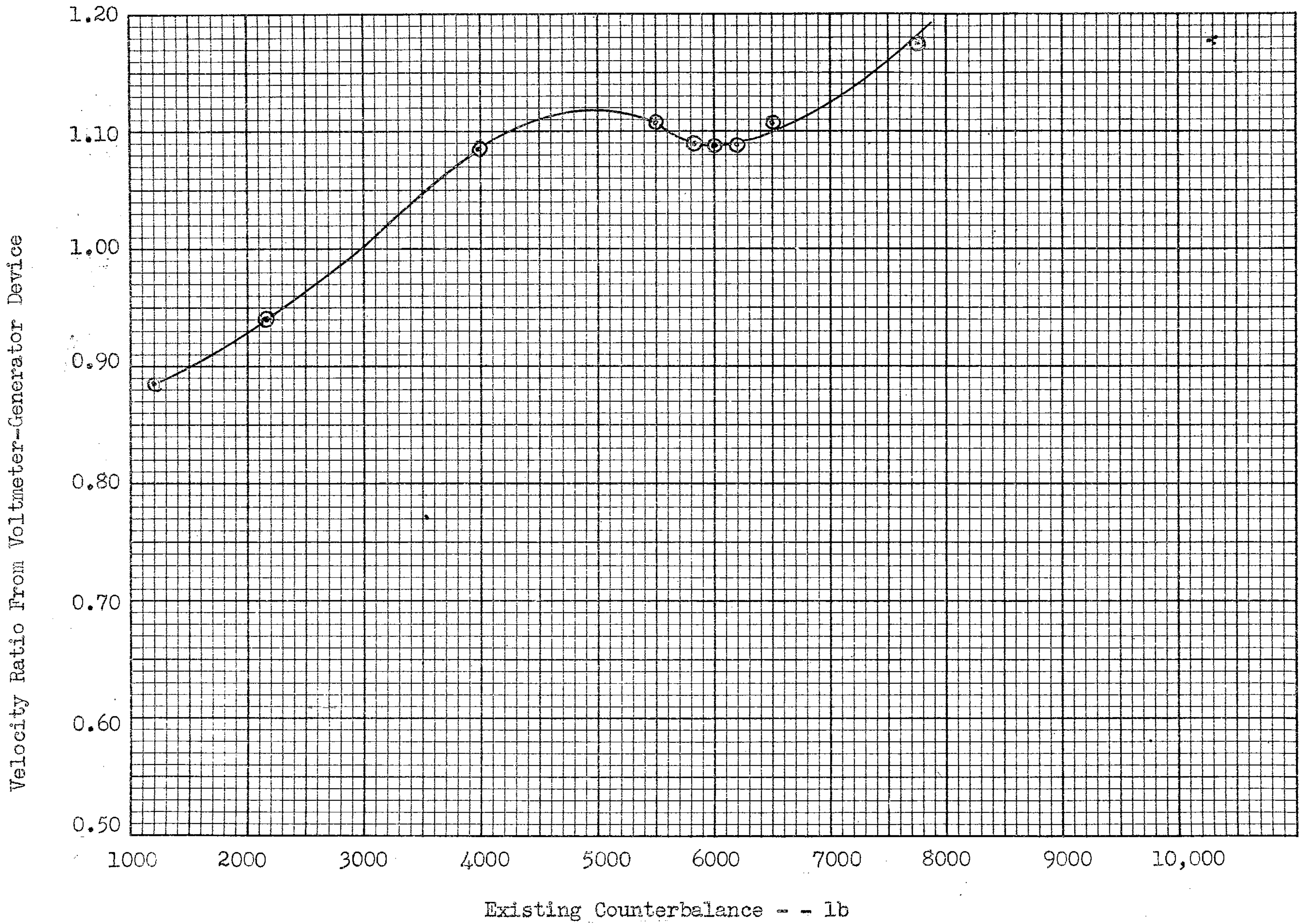

Fig. XVII -- Variation of Voltage Ratio With Counterbalance Conditions

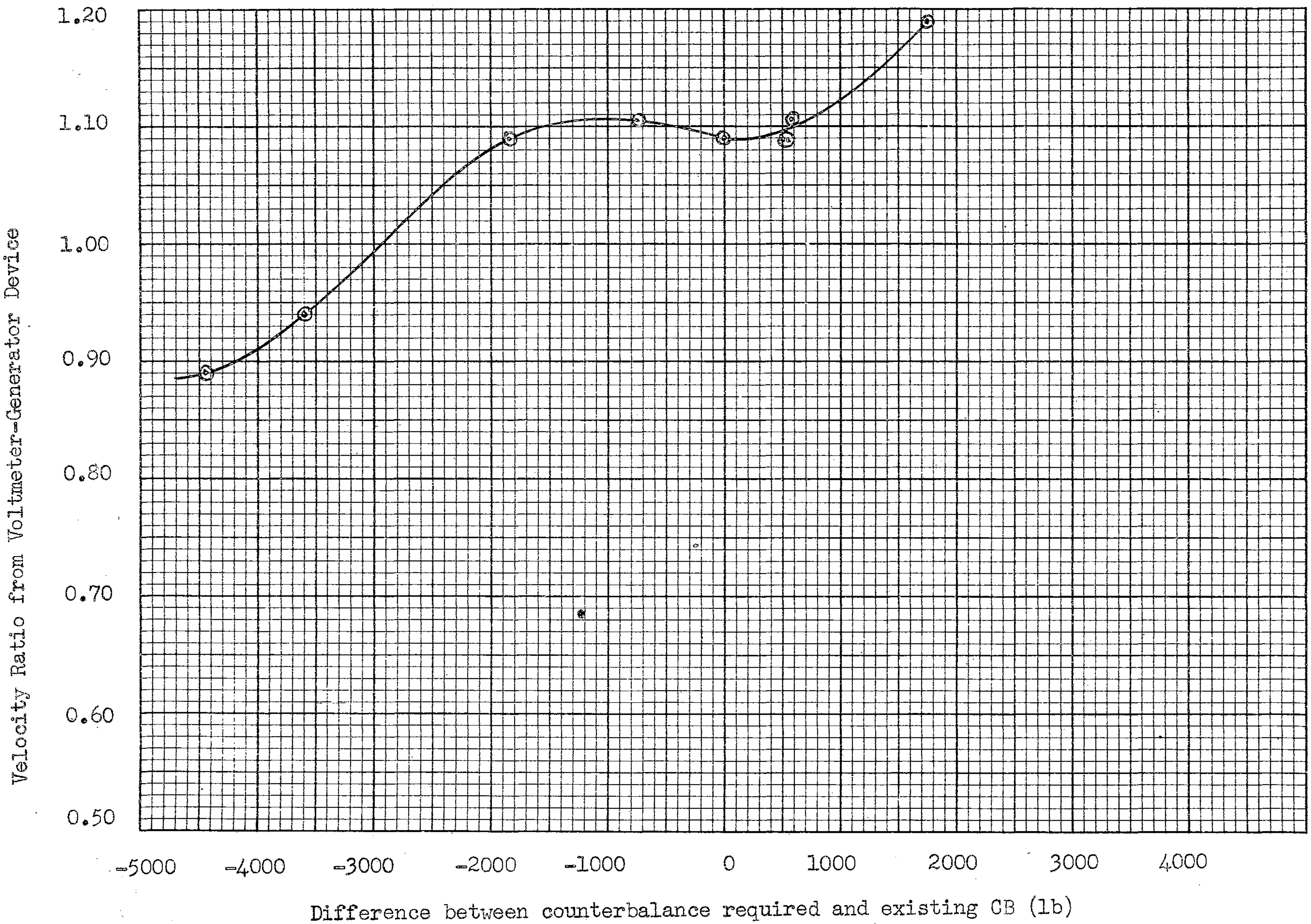

Fig. XVIII -- Variation of Voltage Ratio With Difference Between Existing and Required Counterbalance Conditions

CHAPTER VIII

SUMMARY AND CONCLUSIONS

The problem as stated was to develop a device, adaptable to most pumping unit installations, that would determine qualitatively, the existing counterbalance conditions.

The following variables, thought indicative of counterbalance conditions were proposed: velocity, acceleration, horsepower, torque, air-flow and back pressure. As a result of the preliminary analysis, torque and velocity were found to be best suited for the necessary index of counterbalance conditions. Because of time limitations, only one variable could be tested. The velocity of the polished rod was chosen since it could be measured easily and the devices employed to measure it were safer and more adaptable. It was also determined that the ratio of the maximum polished rod velocity on the upstroke to the maximum polished rod velocity on the downstroke should indicate these counterbalance conditions.

In order to measure qualitatively the selected variable, two devices were selected. These devices were the hydraulic cylinder-orifice-pressure gage device and the voltmeter-generator device. Another device, the liquid-filled curved-tube steel-ball device was also used to indicate counterbalance conditions.

The first device tested was the liquid-filled curved-tube steel-ball device. The displacement of the steel ball, relative to an arbitrarily chosen reference point on the tube, was measured. The ratio of the maximum displacement on the upstroke to the maximum displacement on the downstroke was considered as a possible indicator of counterbalance conditions. The position of the ball in the tube is a function of the acceleration of the beam, the angular position of the beam, gravity, and the relationship between the drag of the fluid and the size of the ball.

The results of the preliminary laboratory tests indicated that the ratio of the maximum positions of the steel ball was not a linear function of the ratio of the maximum polished rod velocities. The results were compared with the ratio of the graphically determined maximum velocities of the polished rod. The ratios of the steel ball positions varied in error from 43.0% for the 15 in. stroke to 60.7% error for the 20 in. stroke indicating the steel ball position was probably a non linear function of the relative velocity of the tube with respect to the ball. The device was field tested in order to obtain more conclusive data.

The field test conducted at Holdenville showed that the device was of little value, at least in its present form, for the purpose of counterbalancing. This was concluded from the ratios when plotted against either existing counterbalance or the difference between existing and required counterbalance. The curves actually showed a significant variation of 1.395 with a corresponding change in counterbalance of 6490 pounds. The variation in the region of proper counterbalance is of most importance. This variation amounted to only 0.173 with a corresponding change of counterbalance of

2,255 pounds.

The device might prove to be of greater value if the readings could be taken more accurately; if the influence of the forces in the system were known; and if the device could be made more sensitive to counterbalance conditions. The actual movement of the ball can be increased by varying the parameters of the device, such as the arm length, the fluid, the diameter of the ball, and the density of the ball.

The second of the devices was the hydraulic cylinder-orifice pressure gage device. This device operated on the principle that the velocity of the fluid passing through an orifice could be measured by a pressure difference. The results of this device were compared with the ratio of the graphically determined maximum velocities. These ratios were 1.088 and 1.1097 for the 15 in. and 20 in. strokes respectively. The ratios of the square root of the maximum pressure differences were 1.182 and 1.163 for the 15 in. and 20 in. strokes respectively. The error in these ratios when compared to velocity ratios was 6.12% for the 15 in. stroke and for the 20 in. stroke it was 8.55%. This amounted to an increase of 2.43% in the error with a corresponding decrease at 0.64% in the velocity ratio. A reason was sought to explain this peculiar error increase. The error was partially attributed to the air in the system. Although it was impossible to remove all of the air from the system its effect was reduced somewhat by pressurizing the system. The error might also be partially due to variable discharge coefficients. This device was eliminated from further consideration because it lacked the safety and the adaptability of the other devices, and becuase of its peculiar characteristics.

The last device considered was the voltmeter-generator device. This device operated on the principle that a voltage would be generated which was proportional to the velocity of the driving unit if the flux was constant. The results of the preliminary laboratory tests verified this relationship between the voltage generated and the velocity of the driving unit. The ratios of the maximum voltages compared with the velocity ratios were within 0.5%. The second phase of the test was concerned with determining how much the ratios of maximum velocities should vary with a change in the counterbalance conditions. This test was the first field test. It was conducted at a field near Lucien, Oklahoma. The test resulted in two important findings. First, the voltages generated were found to be a function of the normal force applied to the friction wheel, since the rim of the wheel is made of a compressible material. Therefore the effective diameter of the wheel is reduced with an increase in the normal force, causing an increase in the voltage generated. Second, the voltage ratio decreased by 0.03 with a corresponding decrease in counterbalance of 9,190 pounds. This variation was not as large as was anticipated. One reason might be that the unit was operating at approximately two-thirds of its rated maximum load, and the "flywheel effect" might have been significant. This small variation might have been accounted for in light of the Holdenville tests.

The last phase of the tests on this device was conducted at a field near Holdenville, Oklahoma. The well loads were recorded on dynamometer cards. The polished rod velocities in the form of a timing wave generated by a tuning fork were also recorded on the cards. The test results indicated that the variation in the ratio of maximum polished rod velocities with a change in counterbalance of 5,500 amounted to 0.237. The variation of the velocity ratio in the region of proper counterbalance amounted to 0.046 with a 1,392 pound counterbalance change. The ratios of the maximum voltmeter readings were within 1.3% of the ratio of the maximum velocities of the polished rod obtained from the timing waves on the dynamometer card.

This device does not seem to be capable of indicating counterbalance conditions accurately, even though a definite drop in the velocity ratios is present in the region of best counterbalance. This drop only amounts to about a 3% decrease in the velocities for the under-counterbalanced condition, Figure XVIII. Since the variation in the velocity ratio is small compared to a counterbalance change it indicates that the selected variable is not entirely dependent upon counterbalance conditions.

Several factors influence the velocity ratio. An important factor in addition to the magnitude of the loads at which peak torques occur, is the location of these loads. The location is a function of the pumping speed, the sensitivity and response characteristics of the governor-engine combination to changing loads, and the amount of counterbalance. The velocity ratio is also a function of the percent utilization of rated load capacity, which in turn determines the significance of the "flywheel effect". The importance of the location of the peak torque is exemplified by considering a pumping unit operating at a constant pumping speed having the same velocity ratio at two different counterbalance conditions. This phenomenon may be explained by considering such factors as the location of peak torque, the sensitivity and response characteristics of the engine-governor combination, and how these factors might influence the velocity ratio.

Therefore it appears as though the variable, maximum polished rod velocity is not entirely a function of counterbalance conditions, but also a function of other factors which influence the velocity, and they are neither negligible nor necessarily dependent upon counterbalance conditions.

Although the voltmeter-generator device measured the selected variable (velocity) with satisfactory accuracy, the velocity seems to be a poor index of counterbalance because of the small variation of the velocity ratio obtained with a change in counterbalance.

The curved tube device should be investigated further.

One possible solution to the problem is to use the variable, torque, as an index. A device which may measure the variation in torque is the belt tension device. With this device it should be possible to indicate the counterbalance conditions, and determine when the pumping unit is properly counterbalanced.

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The content and form have been checked and approved by the author and thesis adviser. The Graduate School Office assumes no responsibility for errors either in form or content. The copies are sent to the bindery just as they are approved by the author and faculty adviser.

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