Pumping Unit Geometry

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The principle aim of this paper is to present basic definitions and calculations for determining and using API Pumping Unit Torque Factors and Permissible Load Diagrams. And to determine these one must know how unit geometry variation affects their magnitude and location. The resulting knowledge can be used for the operator's benefit.

Although the use of torque factors was started some 15 years ago and the API has standardized them, their use has been almost completely limited to torque calculations from dynamometer cards. The general feeling by operators is that safety factors included in commonly used unit sizing equations eliminate requirements for more careful calculations or torque factor considerations. There are so many unknown variables that cannot be accurately predicted that operators feel that further accuracy is not warranted.

Another reason why more use is not made of torque factors is that many engineers who should be using them are not sure how they should be used and exactly how torque factors and permissible load diagrams are calculated. Perhaps a step by step presentation of the method of calculating torque factors will show their usefulness.

CALCULATING TORQUE FACTORS

First, one must assume that the walking beam is a simple "teeter board." If a weight of 1,000 pounds is hung on one end, balance can be obtained by hanging a 1,000 pound weight an equal distance from the fulcrum point.

With an actual walking beam the forces would be similar when the beam is horizontal; however, since the load is applied through a wire line along an arc of constant radius, any position of the beam other than horizontal would require additional force or weight.

The force required to balance the 1,000 lb weight would have a different value for each beam position. At 30° 1,155 lb would be required to balance 1,000 lb, or a load factor of 1.155.

In actual pumping unit geometry this balancing force must be supplied through the pitman arm which is connected at one end to the crankpin and at the other end to the tailbearing.

The instantaneous crankpin and tailbearing locations determine the angle of the pitman and therefore the angle

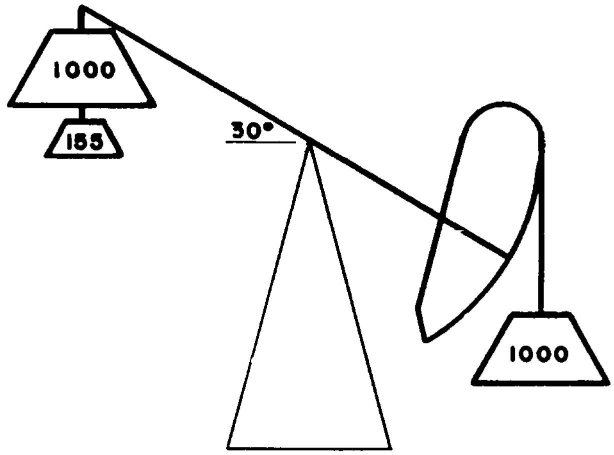

FIG. 2

along which the force must be applied. This pitman angularity also causes an increase in the force required to balance the load. With the beam horizontal it has been shown at 1,000 lb. could balance the load if applied vertically (Fig. 1). The force applied along the pitman, however, must be 1,155 lb if the pitman angle is 30° when the beam is level. One should note also that a horizontal force of 578 lb per 1,000 lb load is present: this force must be resisted at the saddle bearing and the slow speed shaft bearing of the gear reducer. And this is an internal force which must be resisted by the structural members of the unit. These forces act in opposite directions on each of these bearings and their net effect is to push them apart when the cranks are forward and to pull them together when the cranks are to the rear. In conventional design units these forces will cause flexure in the main base beam members unless they are solidly bolted to a base of sufficient stability to resist this flexure, or unless the structure is designed to resist this flexure.

When the beam is in any other position than horizontal, a combination of forces shown in Fig. 2 and Fig. 3 is

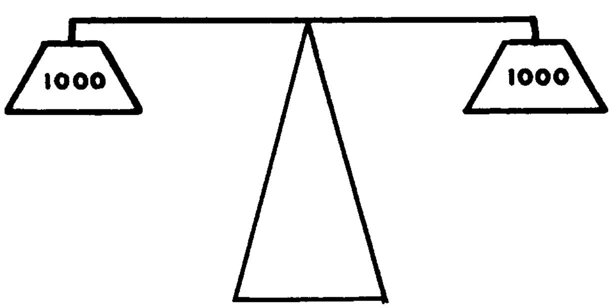

FIG. 1

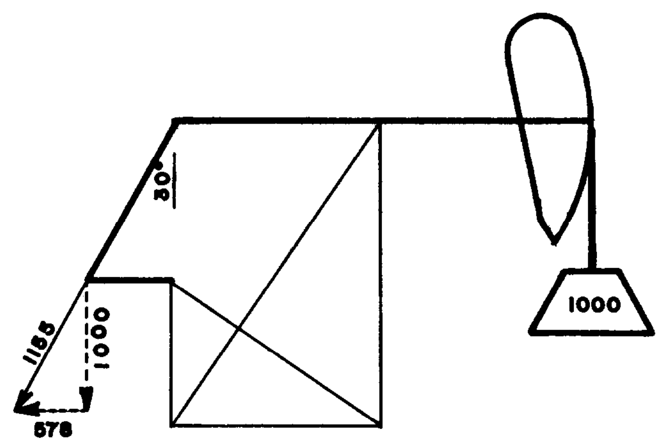

FIG. 3

necessary to balance the hanger bar load. For a given crank position this force would depend on the position of each of the bearing centers. However, it would be different for the same relative crank position with the pitman forward than with the pitman to the rear.

Using actual pumping unit geometry, the force required with the crank forward and 30 degrees above horizontal (Fig. 4) is 1,121 lb at the crank pin and the force required with the crank to the rear and 30° above horizontal (Fig. 5) is 1,050 lb at the crankpin. The difference in these forces is a result of the different angles along which the force is applied and the different location of bearing centers. A pitman load factor of 1.121 for Fig. 4 and 1.050 for Fig. 5 would represent load at the wrist pin.

This force which is transmitted by the pitman must come from the gear reducer which is the power source. If the crank pin radius is 37 in., this load must be lifted with a lever arm of 37 in. All of this force which is perpendicular to the crank arm must be offset by an equal and opposite force consisting of torque from the gear reducer. This force, which is parallel to the crank arm, is resisted by reducer mounting and is an internal force. The force which is perpendicular to the crank arm, Fig. 4, is 1,069 lb and torque which must be supplied is 39,550 in.-lb. Force perpendicular to crank arm, Fig. 5, is 1,028 lbs and torque which must be supplied is 38,040 in.-lb. Torque factor for Fig. 4 is 39.55 and for Fig. 5 is 38.04.

Torque factors simply are multipliers which indicate the inch-pounds of torque which must be furnished by the gear reducer to offset a load on the unit hanger bar. They may be applied directly to load as measured to determine gear reducer torque from load.

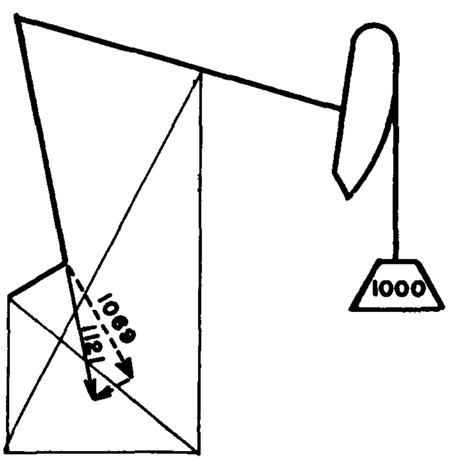

FIG. 4

All of the unit geometry is included in the torque factor for a given crank position and no separate calculations are necessary.

COUNTERBALANCE TORQUE

It is now established that with torque factors the torque can be calculated directly from a measured load. However, consideration must also be given to the effect of counterbalance which is applied to balance between upstroke and downstroke loads. The net reducer torque is the sum or difference of torque from load and torque

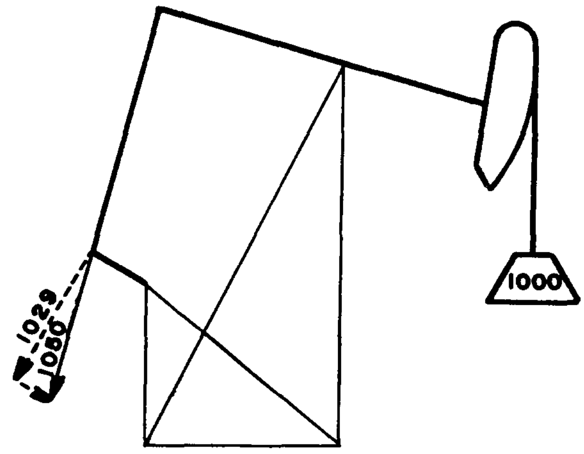

FIG. 5

from counterbalance. On the upstroke, net torque is the torque needed to lift the load which is reduced by the assistance of the counterbalance torque. On the downstroke, net torque is the torque necessary to lift the counterbalance which is reduced by the assistance from the well load.

First, it must be assumed that the units in Fig. 4 and Fig. 5 are both on the upstroke with the unit in Fig. 4 rotating clockwise and the unit in Fig. 5 rotating counterclockwise. It is also assumed that the units have a rotary counterbalance of 1,000 lb effective at the wrist pin when the crank is horizontal. At first it appears that, since the polished rod load is 1,000 lb and the counterbalance effect is 1,000 lb with the crank horizontal, the net torque on the gear reducer should be zero.

Fig. 6 shows that only the forces perpendicular to the crank result in torque. The torque resulting from counterbalance weight would be maximum with the crank horizontal and have a smaller value in any other crank position. But with the cranks vertical, no component of the weight would be perpendicular to the crank arm, and no torque would be received from counterbalance weight. The amount of torque in a certain crank position would vary as the horizontal distance to the center of gravity of the weight. This distance would be the crankpin radius times the cosine of the crank angle. With the crank 30° agove horizontal, a force of 866 lb would be perpendicular to the crank; and, with a crank radius of 37 in., the net torque from counterbalance would be 37 multiplied by 866 or 31,942 in.-lbs.

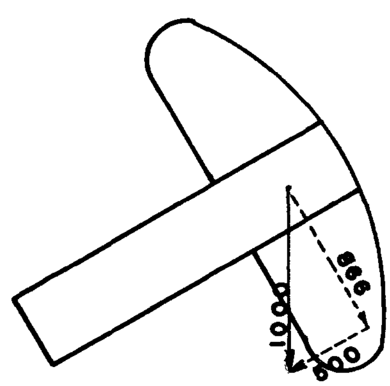

FIG. 6

Applying this counterbalance torque to the unit of Fig. 4, which had the pitman forward and which had a torque from load of 39,550 in.-lb, the net torque which must be supplied from the reducer is 39,550 - 31,942 = 7,608 in.-lb per 1,000 lb load. Applying this counterbalance torque to the unit of Fig. 5, which had the pitman to the rear and which had a torque from load of 38,040 in.-lb, the net torque which must be applied is 38,040 - 31,942 = 6,098 in.-lb per 1,000 lb. load. The net reducer torque with the unit operating clockwise is 7608/6098 or 125 per cent greater than it is with the unit operating counter-clockwise with identical crank angles, counterbalance and load. The torque factor found in Fig. 4 was 39.55 and in Fig. 5 was 38.04. With torque factor comparisons only we would expect this net torque to be only 39.55/38.04 = 1.04 per cent greater. It is apparent then that a comparison of torque factors only is not a valid consideration.

FACTORS IN UNIT APPLICATION OR COMPARISON

Two factors must be considered in any unit application or comparison. First, the load is lifted with a linkage mechanism which necessarily cannot apply the best mechanical advantage in all positions. Second, rotary counterbalance effect does not provide either constant counterbalance effect or constant counterbalance torque. Torque factors and counterbalance effect both have been shown to be dependent on crank position. Thus true analysis must give consideration to their relationship with each other at each crank position.

Proposed is a method which includes both of these factors and which can be used either numerically or graphically. The numerical method is used to calculate "Permissible Load" and the graphical method uses these calculated "Permissible Loads" to construct "Permissible Load Diagrams." These "Permissible Load" values give consideration to all of the geometrical factors involved in pumping unit analysis or selection.

CALCULATING NET TORQUE

Net torque for a pumping unit is calculated by the following API formula:

$$\text{Net Torque} = (\overline{TF})(W) - (M \cos \theta)$$

where:

$\overline{TF}$ = Torque Factor
W = Polished Rod Load
M = Counterbalance Torque at 90° or 270°
θ = Angle of crank, starting with 0° at vertical position and reading clockwise.

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Net torque can be found by this formula when polished rod load, counterbalance torque and crank angle are known. The formula is used to find net torque from dynamometer cards.

PERMISSIBLE LOADS

The "Permissible Load" is the polished rod load necessary to give a resultant net torque equal to the API rating of the reducer with a certain amount of counterbalance. This load should be calculated for each 15 degree crank position. The formula used for this calculation is

$$W \text{ (Permissible Load)} = \frac{\text{Net Torque} + M \cos \theta}{\overline{TF}}$$

where:

Net Torque = API rating of the gear reducer
M = Counterbalance torque at 90° or 270° (Determined by counterbalance requirements for a particular pumping application)
θ = Crank Angle
W = Permissible Load (Polished rod load required to give net torque equal to rating of the gear reducer)

Permissible load calculations for an API unit with poor geometry, having 160,000 in.-lb API reducer rating, 74 in. maximum stroke, 480,000 in.-lb counterbalance torque requirements and clockwise rotation (calculated each 30° for brevity) are as follows (in actual practice permissible loads should be calculated for each 15° crank angle):

CRANK POS.	CRANK DEGREES	POSITION OF STROKE	TORQUE FACTOR	CB TORQUE	NET REDUCER TORQUE	NET REDUCER ± CB TORQUE	PERMISSIBLE LOAD
0	0	.005	7.45	0	160,000	160,000	21,500
1	30	.136	28.34	240,000	160,000	400,000	14,100 *
2	60	.368	38.04	416,000	160,000	516,000	15,100
3	90	.637	37.43	480,000	160,000	640,000	17,000
4	120	.863	26.19	416,000	160,000	576,000	22,000
5	150	.985	9.11	240,000	160,000	400,000	43,900
6	180	.995	- 5.42	0	160,000	160,000	29,600
7	210	.922	- 16.45	- 240,000	160,000	- 80,000	4,900
8	240	.769	- 27.16	- 416,000	160,000	- 256,000	9,400 *
9	270	.546	- 37.65	- 480,000	160,000	- 320,000	8,500
10	300	.272	- 39.55	- 416,000	160,000	- 256,000	6,500
11	330	.052	- 20.85	- 240,000	160,000	- 80,000	3,800
12	360	.005	7.45	0	160,000	160,000	21,500

The critical permissible load value on the upstroke would occur at 30° with a value of 14,100 lb. If a polished rod load greater than this weight should occur at this point, the resultant torque would exceed the rating of the gear reducer, and the critical permissible load value on the down stroke would occur at 240° with a value of 9,400 lb. On the other hand, if a polished rod load less than this should occur at this point, the resultant torque would exceed the rating of the gear reducer.

Experience with the dynamometer has shown that the calculated peak polished rod load can occur during any part of the stroke, and the minimum polished rod load can occur during any part of the stroke. The obvious concusion must be that, if a unit is applied with calculated peak and minimum polished rod loads which fall outside these critical values, a very strong possibility exists that the unit will be overloaded.

A sample unit calculation with consideration given critical permissible load values would be the following example problem:

Use the above 160,000 in.-lb peak torque unit with 74 in. maximum stroke with the following conditions:

16 strokes per minute, 74-in. stroke, 1 1/4 in. pump plunger, 7/8 in. and 3/4 in. tapered rod string, 5,500 ft pumping depth.

The following formulae are the most commonly used in calculations for pumping unit sizing:

$$\text{PPRL} = \text{Wr}\left(1 + \frac{N^2L}{70500}\right) + \text{Wf}$$

$$\text{MPRL} = \text{Wr}\left(1 - \frac{N^2L}{70500}\right) - \frac{62.5\ \text{Wr}}{490}$$

$$\text{CB} = \frac{\text{PPRL} + \text{MPRL}}{2}$$

$$\text{Peak Torque} = (\text{PPRL} - \text{CB})\frac{L}{2}$$

where:

PPRL = Peak Polished Rod Load, lb
Wr = Weight of rods, lb
N = Number of strokes per minute
L = Length of Stroke, in.
Wf = Weight of Fluid, lb
MPRL = Minimum Polished Rod Load, lb
CB = Counterbalance, lb

For the above conditions. the following values are calculated:

$$\text{PPRL} = (1.79)(5500)(1.281) + (.322)(5500) = 14{,}380 \text{ lb}$$

$$\text{MPRL} = (1.79)(5500)(1.281) - (1.79)(5500)\left(\frac{62.5}{490}\right) = 5{,}820 \text{ lb}$$

$$\text{CB} = \underline{14{,}380 + 5820}/2 = 10{,}100 \text{ lb}$$

$$\text{Peak Torque} = (14{,}380 - 10{,}100)37 = 158{,}430 \text{ in.-lbs}$$

Using the torque factor table above, the torque factor at 90° is 37.43, and the calculated counterbalance required is 10,000 lb effective. Counterbalance torque is calculated by multiplying torque factor at 90° by effective counterbalance measured. The resulting counterbalance torque is (10,000)(37.43) = 378,043 in.-lb.

The critical values occur at 30° and 270°, as already determined, so it is assumed that calculated values occur at these points — PPRL at 30° and MPRL at 240° — and actual torque is calculated if this did happen.

$$\text{Net Torque} = (\overline{\text{TF}})(\text{W}) - (\text{M} \cos\theta)$$

For PPRL at 30°:

$$\text{Net Torque} = (28.34)(1{,}830) - (378{,}043)(.5)\ 231{,}262 \text{ in.-lb.}$$

For MPRL at 240°:

$$\text{Net Torque} = (-27.16)(5820) - (-378{,}000)(.866) = 169{,}280$$

It is evident that more counterbalance could be used to equalize torque on upstroke and downstroke, but even with equalized torque of approximately 200,000 in.-lb, the 160,000 in.-lb reducer would be severely overloaded if calculated loads were to occur at these critical values.

To simplify unit selection, there can be made calculations of critical permissible load values for regular counterbalance increments. Peak and minimum polished rod loads are calculated in the normal manner, and, from these calculated permissible load values, reducer size and counterbalance are selected by inspection. For example, a 228,000 in.-lb reducer and a 160,000 in.-lb reducer with the above structure would be

C. B.	C. B.	160D REDUCER		228D REDUCER	
EFFECT	TORQUE	MAX. P.L.	MIN. P.L.	MAX. P.L.	MIN. P.L.
5,000#	187,150#	8,948#	99#	11,347#	0#
6,000#	224,580#	9,608#	1,270#	12,007#	0#
7,000#	262,010#	10,268#	2,463#	12,669#	0#
8,000#	299,440#	10,929#	3,657#	13,330#	1,153#
9,000#	336,870#	11,589#	4,850#	13,991#	2,346#
10,000#	374,300#	12,249#	6,043#	14,652#	3,540#
11,000#	411,730#	12,909#	7,236#	15,313#	4,733#
12,000#	449,160#	13,569#	8,429#	15,974#	5,927#
13,000#	486,590#	14,229#	9,622#	16,635#	7,121#
14,000#	524,020#	14,889#	10,815#	17,296#	8,315#

It should be noted that, with the 160,000 in.-lb reducer and with the satisfaction of the PPRL calculated of 14,380 lb, a minimum load no smaller than 9,622 lbs can be tolerated without overload. Thus, it is quickly evident that this reducer is too small for the previous application. However, with the 228,000 in.-lb reducer, 10,000 lbs of counterbalance effect will allow a PPRL of 14,380. The minimum permissible load is well below the calculated value of 5,820 lbs at 3,540 lbs, so minimum load could fall as low as 3,540 lbs without overloading reducer.

The use of critical permissible load values will allow proper sizing of pumping units with accuracy. This accuracy is of special importance in waterflood pumping where calculated and measured values are very close together. Many cases have been found where produced water brine is created with fluid level at or near pump seat and with calculated loads not conservative enough. It should also be noted that, as counterbalance becomes greater, the difference between maximum and minimum critical load values becomes less. With 160D this difference is 8,338 lb with 6,000 lbs counterbalance effect, but is only 4,074 lbs with 14,000 lbs counterbalance effect. Critical permissible loads assume extreme importance when counterbalance requirements are high.

CONSTRUCTING PERMISSIBLE LOAD DIAGRAMS

The next step in the use of permissible loads is to construct permissible load diagrams. Torque factor tables show the position of the polished rod for each crank angle. If the calculated permissible loads are plotted with a vertical scale reading in pounds load, this value can be plotted. The location of the point on the horizontal axis is determined by the percentage of total stroke at which this permissible load would occur (Fig. 7).

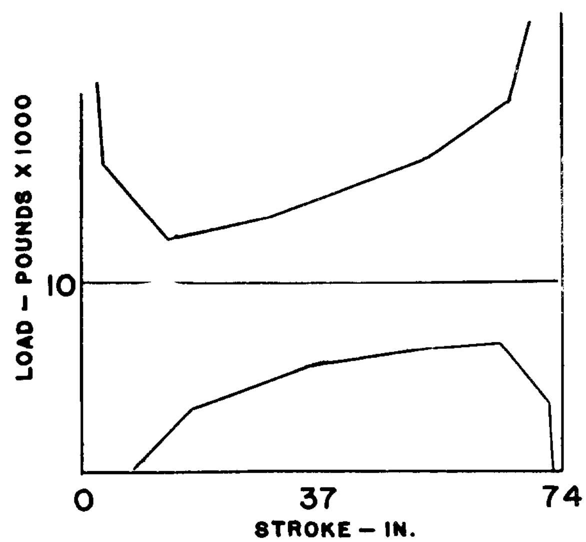

FIG. 7

For the 30° crank angle permissible load is 14,100 lb and polished rod has moved .136 per cent of total, or 10.06 in. Counterbalance is 10,100 lbs effective. Each point calculated can be plotted, and, when all points are connected, permissible load diagram for a certain counterbalance effect is determined. Any load on the upstroke greater than the permissible load at that crank angle will result in torque overload and any downstroke load less than the permissible load will result in torque overload.

The permissible load diagram has exactly the same value on each axis as has a dynamometer card. Thus, if the diagram is constructed to the same scales as the dynamometer card being taken, it can be used in the field for direct comparison to determine torque overload without necessity for any other calculation.

Each manufacturer of units has his own particular geometry, and comparison will show that the permissible load diagram will be different for each unit. These diagrams have been used to compare capacity and efficiency of these various geometrical constructions. In fact, the differences found in capacities of units with identical API ratings, but with different geometry will prove surprising.