Pitfalls of Pumping Unit Selection & Application

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ABSTRACT

The generally accepted methods of pumping unit selection usually are satisfactory in obtaining the correct sizes of pumping units. The methods used most often are (1) the Mills Formula for Peak Polished rod load,[1] (2) the average of the minimum load and peak load as the counterbalance required, and (3) the torque requirements obtained from the product of the unbalanced load times one-half of the stroke length.

These methods have certain limitations which can cause real trouble. However they should be used as a guide, or starting point, for correct unit selection. The correct size units will perform the job required at minimum cost and down time.

There are no average wells, and, if care is not taken, it is very easy to stumble into a pitfall in the selection. The average design conditions do not take into consideration such factors as down hole friction, fluid with a gravity greater than one, out-of-counterbalance conditions, and many others.

Each step in the procedure of proper selection will be discussed, and the pitfalls which may and can occur will be pointed out from examples. Proper unit selection to avoid trouble can be obtained quickly and easily, if the basic formulae are applied properly with due regard to their limitations and unusual well conditions.

PUMP SELECTION

To properly select a pumping unit, one must start with the given conditions of pumping depth, and desired fluid production in 24 hours. But the desired fluid production is often hard to determine. Should it be (1) the daily allowance of the well or twice the daily allowance or (2) the daily capacity of the well, or should it be (3) some maximum figure expected, from water flood operation, or water drive? Whatever the answer it is beyond the scope of this paper to determine the desired fluid production.

PRODUCTION

Pump size and type and pumping speed must be assumed. Pumping speeds on medium and deep wells are usually selected by the following formula:

$$\frac{237{,}000}{\text{Depth}} = \text{Frequency. Best Pumping Rate} = \frac{\text{Frequency}}{1.5,\ 2.5,\ 3.5}$$

For beam balanced units, speeds should be limited to 16 SPM. Others are limited as follows:

$$\text{SPM} = .7\sqrt{\frac{60{,}000}{\text{Stroke in inches}}}$$

Large casing pumps at shallow pump depths should be limited to 10 to 12 SPM.

Gross production is equal to desired fluid production divided by pump efficiency. The gross production is the product of net stroke length, pump constant, and strokes per minute. A pump size that might be used is selected. Then using pump constant of the selected pump and strokes per minute as determined above, net pump travel is calculated. Net pump travel is polished rod stroke, less rod stretch, plus overtravel. From this net travel a standard stroke length that might be used can be selected. Standard stroke length may be obtained from Table 2 of API Specification for Pumping Units (API STD 11E).[2]

Rod stretch may be determined from curves. And rod percentage for tapered rod strings can be taken from tables to give equal rod stress in the top rod of each string.

Overtravel is calculated from the following formula:

$$\text{Overtravel} = 1.55\ \frac{(LN^2)}{(70500)}$$

Where

L = Stroke in Inches
N = Number of Strokes Per Minute

The pump efficiency varies greatly. It may be as high as 85 per cent to 100 per cent on large casing pumps and 75 per cent to 80 per cent on tubing pumps. Several sets of calculations are sometimes necessary to obtain the required production. Or it may be necessary to use a longer stroke length, or different pump size. However, it is usually found that on deep wells increasing pump size only increases rod stretch but does not increase production. On deep wells it is usually best to stay with small pumps and use long stroke units.

Peak polished rod load is usually obtained from the Mills formula:[3]

$$Prl = Wr\,(C) + Wo$$

Prl = Peak Polished Rod Load
Wr = Weight of Rod
Wo = Weight of Fluid Column
C = Acceleration Factor

$$C = 1 + \frac{LN^2}{70500}$$

This is the best formula and is very widely used. The Mills formula was developed in the late 1930's and was based on information obtained from several thousand wells between 2,000 ft and 4,000 ft. This formula gives an average peak polish rod load to be expected, but no wells are average, and such factors as these may influence the peak load:

1. Fluid with a gravity greater than one
2. Down hole friction such as
   a. Rod Scrapers
   b. Paraffin Conditions
   c. Pump Friction
   d. Crooked Holes

e. Tight Stuffing Boxes
3. Fluid Pound
4. Poor Pump Action
5. And Others

Any one of these factors can and may increase the peak polish rod load as much as five to ten per cent or more. The average load obtained from the Mills formula is very good, but should be applied with the knowledge that it gives the Average Peak Polished Rod Load expected, and not the Maximum Peak Polished Rod Load that can occur on a given well. We have found several cases on medium depth (about 4,000 ft) water floods that peak polish rod load as measured was 10 to 15 per cent higher than calculated.

One should never use a figure less than one for fluid gravity, and if greater than one the known figure should be used.

Peak polished rod load obtained will determine the size of unit structure to be used. Standard structure sizes can also be selected from Table 2 of API Specifications for Pumping Units (STD 11E). This structure size selected should be at least the peak polished rod load obtained. There are normal safety factors in pumping unit structures, so structure failures are not to be expected if loads are within name plate ratings.

Stress in rod string should be kept below 33,000 psi. However, salt water and hydrogen sulfide will usually alter this figure. If rod stress is higher than maximum figures allowed, it is necessary to select larger rods, or a different combination.

Minimum polished rod load is usually found from following formula:

Minimum Load = Wr (1.873-C)

Counterbalance required is usually taken as the average of peak and minimum loads. Care should be taken not just to obtain the exact counterbalance required. This counterbalance figure is based on one half the sum of two averages and can be different from actual requirements. Provisions should be made so that, if required, additional counterbalance can be obtained by adding auxiliary counterweights.

Peak torque is usually obtained as follows:

$$\text{Peak Torque} = (\text{Prl} - \text{C Bal}) \frac{(L)}{2}$$

This method is excellent except that it does not include any factors for friction of the bearings in the pumping unit and that it assumes a perfect geometric structure having a torque factor of one. We know that unit structures are not ideal and usually have about five per cent loss. Also, it has been found that there is usually about ten per cent friction loss in the center bearing, equalizer bearing, crank pin bearings, and crank shaft bearings. Combining these losses would result in a total of 15 per cent loss between gears and the polished rod.

In all cases one should take into account this 15 per cent loss by dividing a calculated peak torque requirement by .85 to give average peak torque expected.

It has been suggested by Hicks and Agnew that one should assume that pumping units in operation may be as much as five per cent out of counterbalance.[4] We would suggest that this figure may be as much as 5 to 20 per cent over or under balanced, and that ten per cent would be a more realistic figure to be used.

It is very important to have a unit properly counterbalanced, for proper counterbalance may mean the difference between gear failure and good operation. Several methods of proper counterbalancing are suggested in C. D. Richard's paper on counterbalancing beam type pumping units.[5]

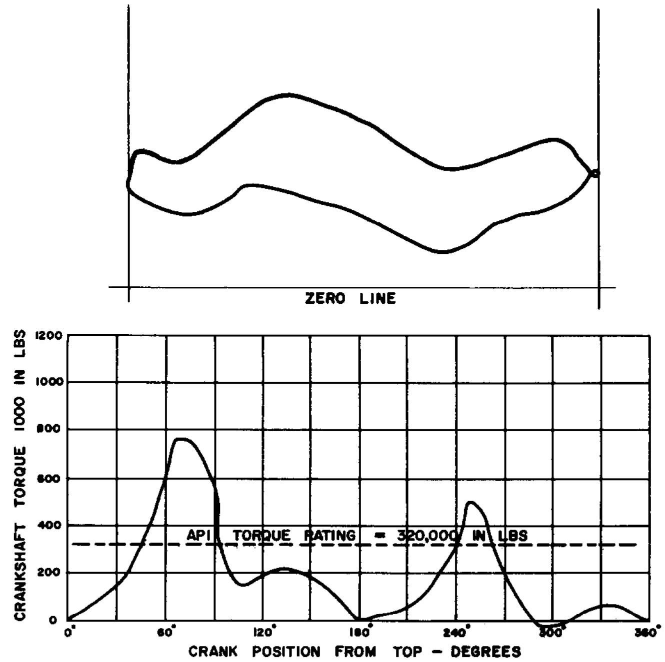

FIGURE I

After peak torque requirements have been determined, the next step is to pick out from Table 3 a standard API gear box. The standard assumption is that there is a safety factor in the rating of pumping unit gear boxes. However there is no safety factor in pumping unit gear rating. We quote as follows from API STD 11E, page 9:

> "22. Ratings are based on surface durability (which is independent of pitch). However, the gear manufacturer shall assume responsibility for selecting a pitch sufficiently coarse to provide adequate tooth strength.
> 26. Gear ratings shall be based on a nominal pumping speed of 20 strokes per minute."

It will be noted that ratings are based on 20 SPM operating speeds. Lower operating speeds reduce the torque ratings of gear reducers. Purchasers buy pumping unit reducers on the basis of gear rating, i.e., a certain number of dollars for a given torque rating. No more is expected to be furnished, and no more is guaranteed by the manufacturer.

Gear loads in the order of 100 to 150 per cent of ratings will cause pitting and accelerated tooth wear. This gradual reduction of tooth cross section will allow tooth deflection and will result in tooth breakage after a period of time. This period of time before failure may range from a few months to several years.

Large overloads in the order of 150 per cent (or greater) of gear ratings will usually result in tooth deflection and tooth breakage in a very short period of time. The period may range from a few weeks to a few months, but is usually less than a year.

Large numbers of failures have been stopped by rapid decline in the production.

We have found that approximately 25 per cent additional torque should be added to the calculated figure to take care of fluid pound on large casing pumps at shallow

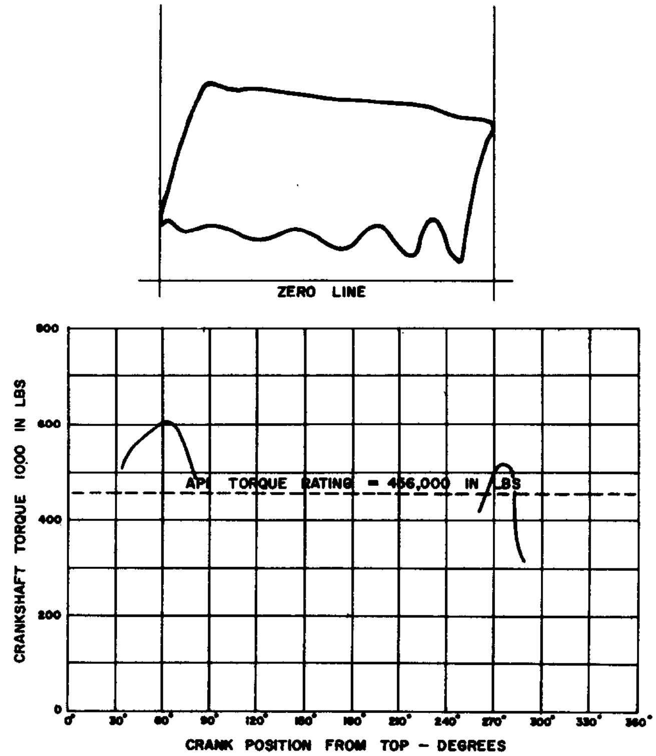

FIGURE 2

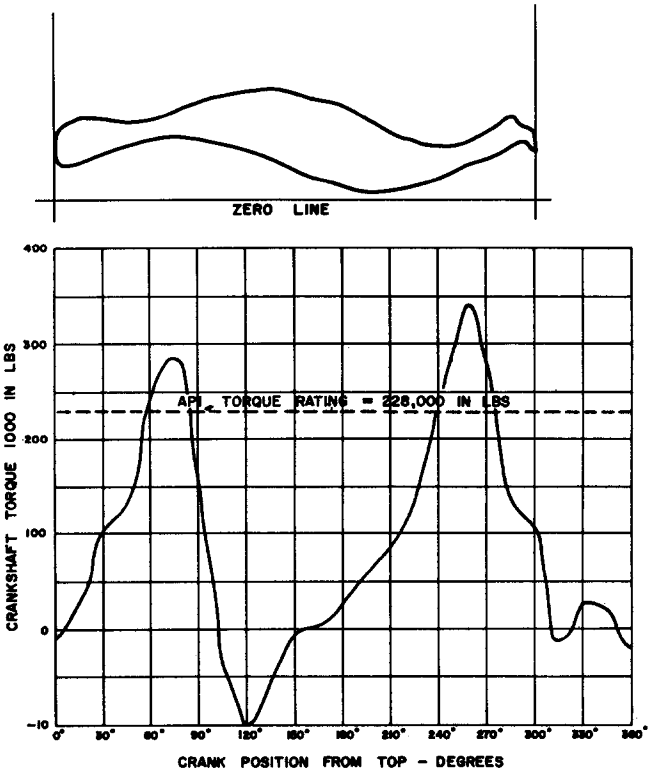

FIGURE 3

depths. The large casing pumps are often used on water supply wells.

EXAMPLES

Example 1

Producing Oil Well	Deep Water Flood
Pump Set 5,000 ft;	Two in. Pump
1 in. 7/8 in. 3/4 in. Rods:	12.12 SPM 122 7/8 in. Stroke Length
MAJOR OIL COMPANY	BOWIE, TEXAS
CARD TAKEN JULY 28, 1955	UNIT 320D
Approximately 500 bbls.	Fluid Produced in 24 hr

AS CALCULATED USING	GEAR LOADS AS A PER CENT OF GEAR BOX RATING
No corrections	121
Unit efficiency correction	138
Unit efficiency, and 10% out of counterbalance corrections	152
AS MEASURED	
Sine curve method of card interpretation	237
IF BALANCED AND REMEASURED	197

Gear Box failed from tooth breakage caused by severe torque overload, resulting in high tooth bending stresses, and subsequent fatigue failure in shear. Unit should have been 640D. Stroke length reduced and unit still in operation.

Example 2

Water Supply Well	
Pump Set 1,457 ft;	4 3/4 in. Pump
1 in. Rods; 9.5 SPM; 120 in. Stroke	
INDEPENDENT OIL COMPANY	HOLLIDAY, TEXAS
CARD TAKEN APRIL 20, 1960	UNIT 456D
2880 bbls.	Salt Water Produced in 24 hrs

AS CALCULATED USING	GEAR LOADS AS A PER CENT OF GEAR BOX RATING
Fluid gravity correction	88
Fluid gravity and unit efficiency corrections	104
Fluid gravity, unit efficiency and 10 per cent out of Counterbalance correction	110
AS MEASURED	
Torque Factor method of car interpretation	132
IF BALANCED AND REMEASURED	122

Unit still in operation slightly overloaded. Unit should have been 640D.

Example 3

Producing Oil Well	Water Flood
Pump Set at 3,800 ft;	Two in. Stroke
3/4 in. Rods; 16 SPM;	86 in. Stroke
INDEPENDENT OIL COMPANY	KAMAY, TEXAS
CARD TAKEN JANUARY 4, 1961	UNIT 228D
409 bbls. Fluid 86 per cent Water Produced in 24 hrs.	

AS CALCULATED USING	GEAR LOAD AS PERCENTAGE OF GEAR BOX RATING
No corrections	86
Unit efficiency correction	103
Unit efficiency, and 10 per cent out of counterbalance correction	112
AS MEASURED	
Torque Factor Method of Card Interpretation	151
IF BALANCED AND REMEASURED	137

Unit still in operation, overloaded. Unit should have been 320D.

CONCLUSION

Existing methods of calculation give average results and do not take into account any unusual operating conditions. There are safety factors in pumping unit structure ratings, but none in gear ratings. Torque calculations should take into account the following:

1. Fluid gravity greater than one
2. Pumping unit efficiency
3. Out of counterbalance conditions
4. Fluid pound on casing pumps

Allowances should be made for inaccuracies in calculation methods. But one is trying to do with a yardstick a job that requires a micrometer. One must select gear reducers large enough for the job. If a gear box is too large for the job, it will just last longer. But if it is too small, it will fail and cause considerable trouble and expense.

REFERENCES

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