Importance of Direction of Rotation of A Pumping Unit

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For many years people in the oil industry have discussed the direction of rotation of pumping equipment. The first point to realize is that the pumping equipment can be designed to operate to an advantage in either direction, and some are designed to give equal effect in either direction. It then becomes apparent that in most cases the observer must know the purpose for which the pumping equipment was designed and built. This purpose in itself will answer, automatically, the best direction of rotation.

In some cases in the earlier days of manufacturing individual well units, the gear reducers have been built to rotate in only one direction. The direction of rotation was founded on two factors of manufacturing economics.

In the first instance it took much more time to machine both sides of the gears. Therefore, manufacturers prepared only the contacting surfaces of the various gears. Consequently, if the gear train rotated differently than indicated, the very rough surfaces would cause excessive pressures and wear. This action took place and eventually necessitated a standard marking on the gear case to indicate direction of rotation.

The second factor to cause a marking on the case for direction of rotation was the manner in which the intermediate shaft was installed in the case. It can be seen that the direction of rotation of the slow speed shaft can cause forces upward on the intermediate shaft and forces downward on the opposite direction. Economy was also affected here and the amount of strength necessary could amount to dollars, and affect the manufacturer's volume of sales.

These ideas and uses did not affect the quality of the work being done as long as the equipment was used as directed. These markings on the case were the first cause of concern in the direction of rotation. It must be realized that gear cases were first installed on old rig fronts to reduce maintenance on band wheels, belt halls, gas engines, etc. The second step was to build metal rig fronts so the gear reducers could be added as soon as the well was drilled with cable tools.

The rotary rig equipment had no part that could be left on the well site to aid in pumping, so it was an easy step to buy the fabricated steel walking beam, samson post, and gear case as a pumping unit. These were bolted directly on a concrete foundation.

But the installation had only a single pitman and resulted in many types of failure. Further, the large size of the structure did not lend to economy and was soon in competition with small two-pitman rotary balanced units on skids.

Competition is said to be the "spice of life," and it has really seasoned and flavored the competition in that branch of the industry which makes pumping units. In many cases some copied another. Occasionally, one fellow could pump a duplicate set of well conditions with less horsepower that did the other; and this competition started another tide of changes and study of kinematics. It is understandable that this type of explanation is difficult to make simple and, as a result, too many people who were involved just accepted beliefs without proof.

Another thing regarding these explanations or comparison of the various units is the "basis of comparison" For example the original identity of a gear reducer was in horsepower which is usually a force creating effort. In those days, the power of the engine was known so the gears were just rated to take the load. Some began to realize that this rating was a poor way to compare units so they started to look at "in. lb of torque" to twist the slow speed shaft and a conversion factor resulted. In 1940, the formula was:

$$hp = \frac{T \times RPM}{63025} \quad \text{or } T = \frac{63025 \times hp}{RPM}$$

One hp at one RPM would generate 63,025 in. lbs of torque, or at 10 RPM, 6,302 in. lbs of torque

The oilfield hands wanted a short cut, so the assumption was that, since the average unit was operated at 14 SPM, then the hp = 4500 in. lbs of torque, and that figure was used for fast conversion.

The torque figure seemed to be a real comparison for evaluating competitors units. In this manner it was a positive figure to compare and everyone switched to torque for identification.

There was apparently still some difference in opinions about how to get the correct size of gear reducer on a well. One would recommend a full API size less for a unit sale. However, it was realized by the purchaser that this was still not a fair comparison. Some discovered that the "old formula" of calculating the unit size used 40 per cent of loads as a figure for counter balance. This was a figure derived from "guesstamation" Others claimed that the actual counter balance had to be measured after the equipment was installed and that the cranks were so short that they could not carry a larger weight. Some manufacturers saw the advantage of longer cranks and to allow greater effective counter balance built sub-bases under the reducer.

The idea now was to figure the weight of the upstroke and the weight of the downstroke. By adding the two loads and dividing by two, the best counterbalance would be found. But the actual measuring of the unit with the crank at the 90° point more or less established the actual counterbalance and, in many cases, was found to be as much as 80 to 90 per cent of maximum load instead of the 40 per cent as in original formulas.

The next phase of competition brought in the torque factors. API has set up a standard on how to report the "torque factors" and counterbalance weights used for certain counterbalance effect. A torque factor is really just a factor or number to change vertical reciprocal loading at the front end of the beam or rods to a quantity of rotating force or torque at the gear reducer, for each angle of the crank, so to speak, a conversion factor. But of course these conversion factors or torque factors are good for only one type and size of unit. To use in torque calculation, each size and type of unit must have a different set of torque factors.

The competition now is trying to obtain the smallest conversion factor to show the advantage of each piece of equipment. There are all sorts of designs being

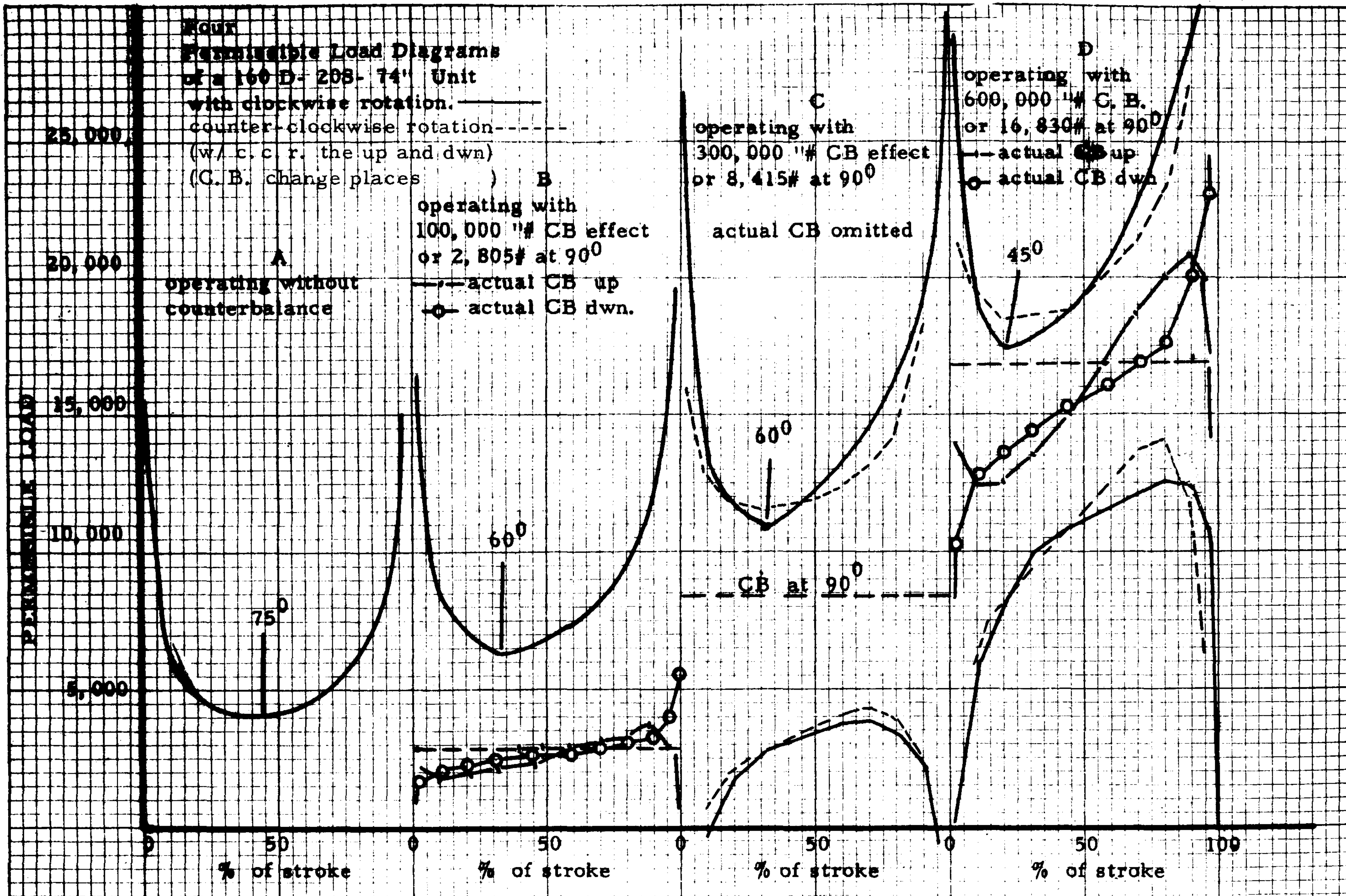
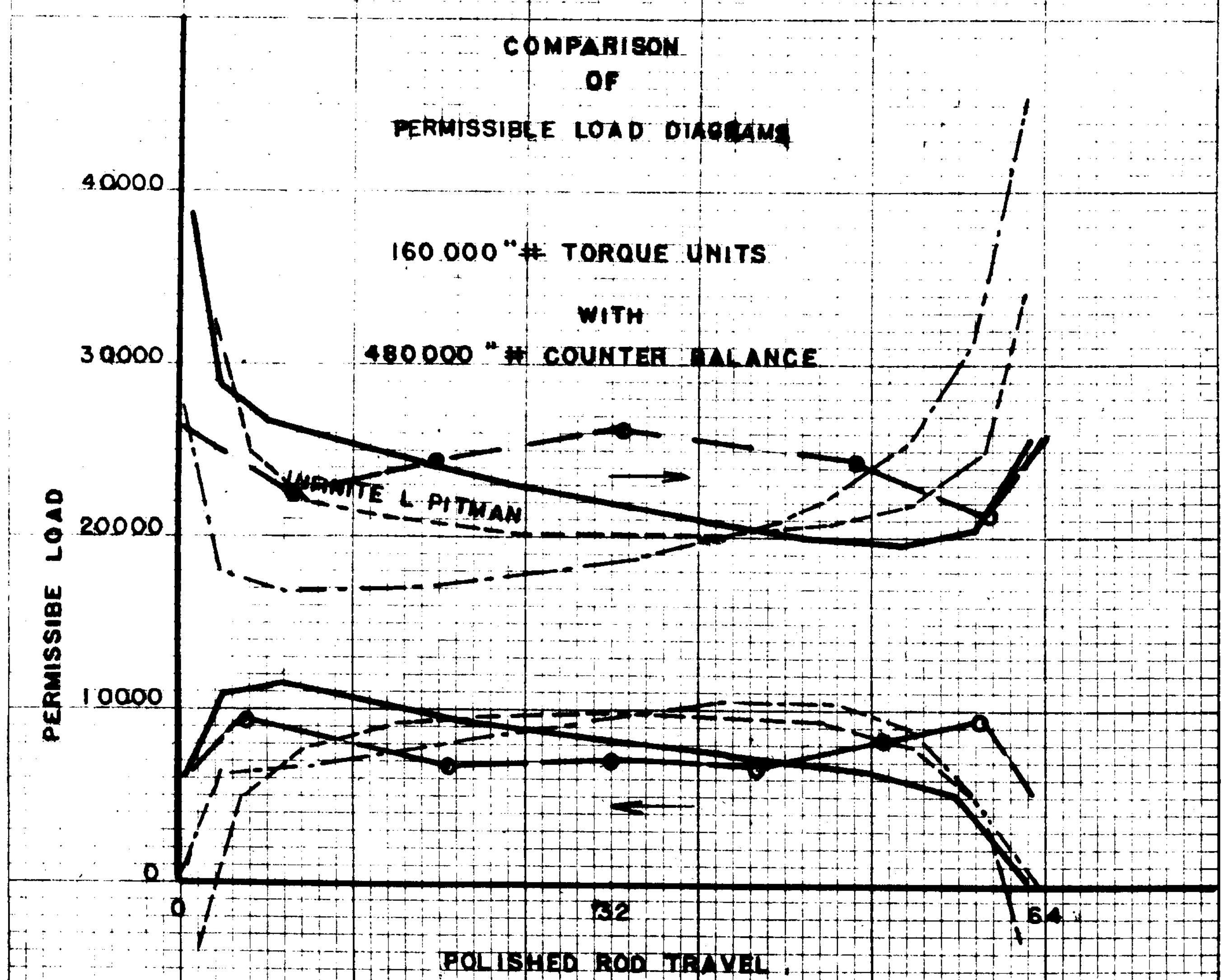

Four
Permissible Load Diagrams
of a 160 D- 208- 74" Unit
with clockwise rotation.
counter-clockwise rotation------
(w/ c. c. r. the up and dwn)
(C. B. change places)
A
operating without
counterbalance
B
operating with
100, 000 "# CB effect
or 2, 805# at 90°
actual CB up
actual CB dwn.
C
operating with
300, 000 "# CB effect
or 8, 415# at 90°
actual CB omitted
D
operating with
600, 000 "# C. B.
or 16, 830# at 90°
actual CB up
actual CB dwn
PERMISSIBLE LOAD
25, 000
20, 000
15, 000
10, 000
5, 000
75°
60°
60°
45°
CB at 90°
0
50
100
% of stroke

Figure 1



COMPARISON
OF
PERMISSIBLE LOAD DIAGRAMS
160 000 "# TORQUE UNITS
WITH
480 000 "# COUNTER BALANCE
INFINITE L PITMAN
PERMISSIBE LOAD
40000
30000
20000
10000
0
0
32
64
POLISHED ROD TRAVEL

Figure 2

studied, and all are trying to obtain the best torque factors. The difficulty is that, until recently, there has not been a very good comparison technique.

Mr. Robert H. Gault presented in West Texas Oil Lifting Short Course for the years 1959, 1960 and 1961, information dealing with the "permissible load" that could be applied to the horsehead end of the unit without overloading the API rated gear reducer. From this permissible load he presented a "permissible load diagram" which could be drawn on clear plastic and overlaid on the dynamometer card taken to see if the loading of the unit was within the permissible load range.

This permissible load and diagram are the true ways to compare units, and it is easy to visualize the peculiar characteristics that some units may have.

The diagrams should be in about four phases: one diagram with no counterbalance load, one with maximum counterbalance load and at a couple of levels of counterbalance in between. The counterbalance level should be at the approximate 90° or 270° level. Figure 1 is a true example of how increased counterbalance affects the shape of the permissible load.

It can be seen that true counterbalance line is not level as often thought; and this fact may come as a surprise to many. The only way that a horizontal counterbalance line could be drawn would be in the condition where the unit had infinite length pitmans.

The permissible load diagram shown in Figure 1 indicates that the only way by which a dynamometer card could guarantee no overload would be to have undertravel shape. If the dynamometer card indicated overtravel the unit would really be overloaded in some instances, as one can visualize. In many instances the normal horizontal type dynamometer card would show overloading at the first part of the upstroke and at an early point on the down stroke.

The permissible loads on Figure 1 may be a surprise, but if one tried to use this unit, heavily loaded, such as "D" position, the maximum load point would occur at 45° on upstroke; then only about 600 lbs more load would be permitted than was on the counterbalance at the 90° position. This situation is somewhat different from what most of us have thought. Another aspect of the counterbalance line is that it is also tilted. Its angularity is also a function of rotary counterbalance affecting the load through the torque factors. And still another point to notice is that the counterbalance line is not straight, nor can it be.

Some pumping units have a less desirable shape than does this type unit, and a few have a more desirable shape, too. There is one unit that has a permissible load diagram on which the area between the upstroke and downstroke is rotated clockwise almost 40° about the center. This area would then be suitable for a dynamometer card of overtravel conditions.

Figure 2 is a family of permissible load diagrams showing as a comparison four different types of designs. The area between the upstroke and the downstroke is the permissible work area. The narrowest is the conventional type unit which indicates less than 160,000 in. lb of torque at mid-point. The next is an infinite length pitman type which has exactly 160,000 in. lb of torque. The next wider has about 220,000 in. lb of torque; and the widest of all has a torque capacity of 352,000 in. lb at the mid-way point. Each of these units is rated on 160,000 in. lb of torque and comparable structures.

These discussions should make it clear that before the correct direction of rotation is known it will be necessary to use Mr. Gault's permissible load diagrams application. However, Mr. Gault's ideas have uncovered some information that can be used in just a visual examination of a pumping unit as it sits in the field. For example, one must learn to look at the saddle bearing and tail bearing as one side of a 90° angle when the beam is horizontal. If the side of the angle pointing down from the tail bearing falls behind the slow speed shaft, the best direction of rotation is counter-clockwise. However, if the side of the angle falls in front of the slow speed shaft, then the direction would be better to rotate clockwise. Thus, if it results in the low angle side running through the slow speed shaft, it does not affect the direction of rotation. However, in either case, the permissible load diagram will be tilted.

The correct direction of rotation does not eliminate the poorly designed unit from having a lack of load capacity as proven by the permissible load diagram.

Conclusions

The most important point in pumping unit operation is to know not only in which direction the unit should operate, but also to know how much abuse a unit receives when operated incorrectly.

The counterbalance is important and a deficiency of only 20 per cent counterbalance can cause some units operating in the wrong direction to be overloaded. When properly counterbalanced and rotating correctly the unit would only consume a little more than one half the total torque capacity.

This paper then indicates that torque, structure, and a few other factors may not constitute a complete basis for a pumping unit comparison.