Sizing Pumping Unit Prime Movers

By CHARLES J FULLER
Continental-Emsco Company

INTRODUCTION

The problem of sizing pumping unit prime movers has been attacked in many ways with nearly all methods involved using different ways of arriving at approximately the same answer. If several factors can be kept in mind at one time, including the type of load with which we are dealing and the characteristics of the various types of prime movers available to us, a better mating of load to motive power can be achieved. There are a great many formulas which can be applied - some complicated, others more basic. With the method which is proposed in this paper, the same end results can be obtained with a quick, easy formula. Also, to better understand why this is possible, we will take a look at the characteristics of the various prime movers which we can use.

THE NATURE OF LOADS ENCOUNTERED

A series of changing loads, varying widely in their range, repeated from 10 to 20 times per minute over an extended period of time with a relatively constant repetition, is basically the type of load with which we must deal.

The well load, or load that is felt by the polished rod, can many times be best explained and measured through the use of a dynamometer card (Fig. 1), since all the prime mover "sees" or "feels" is the torque that is applied against it by the well load acting through the geometry of the pumping unit and the V-belt drive.

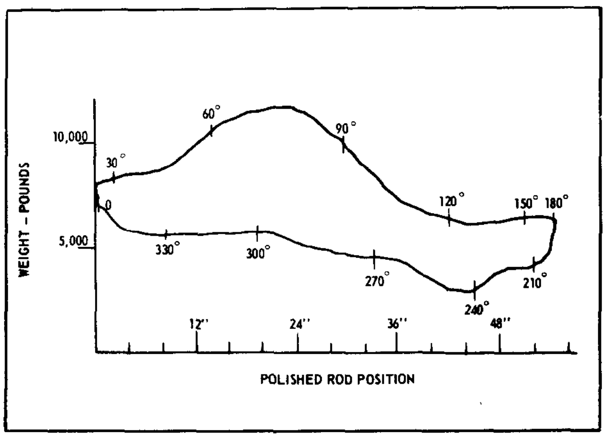


FIGURE 1

Let's convert the weight of this well load, as derived from the dynamometer card, to gross torque. The fact that the load is a cycling-type load begins to work in our favor to some degree, since we can counterbalance a good portion of it. This is shown (Fig. 2) as the sine wave curve that is the counterbalance torque, directly offsetting the gross torque caused by the well load. This is the load with which we must deal.

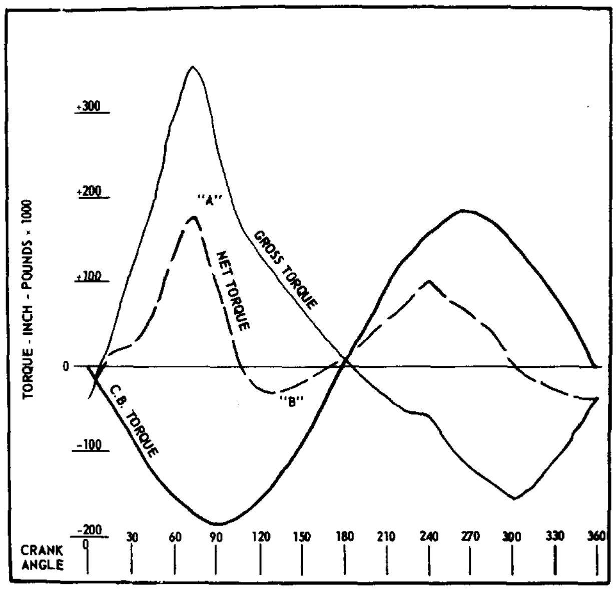


FIGURE 2

The net difference between the well load torque as determined by the dynamometer card, and counterbalance torque as determined by counterbalance weight, is the net torque. This is the load, expressed in torque, that the prime mover must overcome to keep the well pumping. Looking once again at Fig. 2, note the extreme ranges that are evident, the peaks of positive net torque which are the heaviest load demands, the periods of small, almost no load, and the negative torque peaks where the well load or counterbalance is actually driving the prime mover. In these cases, the prime mover is acting as a brake. This, then, is the load with which we must contend; changing from high peaks for a short period of time to no load, to braking, and a repetition of the same process repeatedly.

The factors that cause these widely varying loads are well known to all personnel connected with the oilfield and include depths and volumes and vary greatly as do the straightness of the hole, paraffin conditions, fluid viscosity, amount of gas; all making an infinite variety of combinations. To date, all past solutions for sizing pumping prime movers actually yield a good average method that has worked through experience, and has given good results in most cases, when a prime mover is neither greatly oversized nor undersized for the job.

BASIC HORSEPOWER REQUIREMENTS

As stated earlier, there are almost as many approaches to sizing pumping prime movers as there are companies involved. Most of these approaches involve the basic hydraulic formula with empirical factors included to allow for friction and other variables. Other approaches have included the sizing of pumping prime movers on the basis of torque delivered by the engine or motor being equal to or greater than the net pumping unit gear reducer torque required. Still other horsepower requirements for pumping service have been calculated as a multiple of the hydraulic horsepower or a multiple of the polish rod horsepower along with a number of other factors. Basically, they all include numerous empirical formulas, based on experience, and have been used with varying degrees of success.

The formula we would like to put forth has been used to good advantage for a good many years. It is extremely simple to use, and has been shown to be correct in a majority of cases. The relatively small number of incidences of undersizing prime movers that result in failures leads us to believe that the method is conservative enough, while actual measurement by dynamometer analysis shows that in few cases are we applying too large a prime mover for any particular job. If there had been no instance where a prime mover was applied using this method, and found too small, we believe the method would be too conservative.

To arrive at the simplicity found in this method, several assumptions, based on prior experience, have been made. These assumptions are:

(a) Sub-surface friction, lifting water from total pump depth, sucker rod stretch, and other widely varying factors will usually counteract each other, and will not be considered since this method relies solely on polished rod stroke.
(b) Efficiency of the surface equipment, from the polish rod to the prime mover output shaft, is approximately 85%.
(c) Different cyclic-load factors apply, dependent upon the characteristics of the type prime movers to be used.

To begin with, the basic hydraulic formula is used with the volume calculated, using the polished rod stroke length as the plunger stroke. The information that is required for applying this requirement is as follows:

(a) Plunger size
(b) Polished rod stroke length
(c) Strokes per minute
(d) Depth to the pump

As you can see, all these factors are easily obtainable before the installation is made, thus effecting the ease and simplicity of this method.

To derive this formula, we begin with:

$$HP_h = \frac{Q \times D \times W}{33{,}000 \times 60 \times 24} \qquad \text{(Equation 1)}$$

Where:

HP_h = Hydraulic Horsepower
Q = Volume in BPD based on the plunger size, strokes per min. and polish rod stroke at 100% pump efficiency
D = Depth to the pump in ft.
W = Weight of (1) bbl. of fluid

To convert this formula from hydraulic horsepower to its useable form in prime mover horsepower required, it is necessary to substitute 350# per bbl. for W, apply the 85% efficiency of the surface equipment, and an appropriate cyclic load factor (CLF):

$$HP = \frac{Q \times D \times 350}{33{,}000 \times 60 \times 24 \times 0.85 \times CLF}$$

or:

$$HP = \frac{Q \times D}{CLF \times 115{,}400} \qquad \text{(Equation 2)}$$

Where:

HP is the prime mover shaft horsepower for continuous service.

The cyclic-load factors that we will use will be as follows:

(a) For heavy flywheel single-cylinder engines:

CLF = .72 — .75

(b) For high-slip electric motors:

CLF = .72 — .75

(c) For multiple-cylinder, high-speed engines:

CLF = .54 — .57

(d) For normal-slip, electric motors:

CLF = .54 — .57

Substitution of these various cyclic load factors in Equation 2, yield our final useable formulas:

(a) For heavy flywheel, single-cylinder engines, or high-slip electric motors:

$$HP = \frac{Q \times D}{85{,}000} \qquad \text{(Equation 3)}$$

(b) For multiple-cylinder, high speed engines, or normal-slip electric motors:

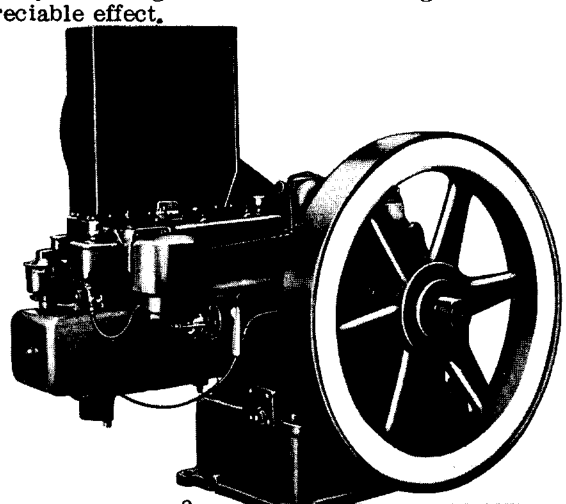
$$HP = \frac{Q \times D}{65{,}000} \qquad \text{(Equation 4)}$$

Equations 3 and 4 are extremely simple, require very little calculation and are difficult to misapply. As stated earlier in the article, this method is intended to be a good, fast, close approximation. Some other methods take a more technical approach, and therefore are somewhat more difficult to apply. All such methods, however, use some empirical method to allow for factors that vary from well to well. This is essentially what has been done in this case. In practice, this method has been successfully used for several years and its accuracy has been borne out through actual measurements by dynamometer analysis as well as by the very few instances where a prime mover has been overloaded. In these few instances extenuating circumstances usually have been present, such as abnormal friction caused by a crooked hole, paraffin deposition, or excessive pump friction.

PUMPING ENGINE CHARACTERISTICS

Since we have shown that the simplified formulas are based on the cyclic-load factor as determined by the prime mover characteristics, we should take a closer look at these characteristics. Exactly what is it that allows us to use a formula arriving at a lower required horsepower for single-cylinder engine applications as opposed to multi-cylinder engines?

The main thing is the heavy flywheel (Fig. 3) that stores a large amount of energy in comparison to the engine horsepower rating. This is expressed as the WR^2 effect, or inertia of the rotating parts of an engine. Obviously, the WR^2 of a single-cylinder engine with its heavy flywheel is a factor to be reckoned with, while a multi-cylinder engine WR^2 is small enough to be without appreciable effect.



Since the WR^2 is not a very descriptive term, let's develop a somewhat more useable term and express this flywheel effect as a "horsepower effect". This is possible only because we are dealing with a cyclic-type load.

To do this, we need to select a typical oilfield pumping engine (a heavy-flywheel type) and look at some of its characteristics. E.g., this engine has, as follows:

(1) Rated horsepower - 14
(2) Operating speed - 700 RPM
(3) WR^2 - 600 lb.-ft.^{2}
(4) Flywheel weight - 400 lb. (approximately)

The stored energy of the flywheel at the rated RPM is 66,000 ft.-lb. After attaining operating speed, no energy is required to keep it rotating, other than to overcome friction losses. Only when the speed changes, due to some outside force, is energy taken from or put into the flywheel. Assume, for instance, that during a peak demand, during the pumping cycle (see point A, Fig. 1), lasting approximately one-third second, the speed of the engine is reduced 3%. From the basic kinetic energy formulas, we know that kinetic energy is proportional to the RPM of the rotating mass, or:

$$KE = CV^2$$

And:

$$\Delta KE = 2CVdv$$

Where:

(KE) is kinetic energy.
(C) is a constant representing the mass of the rotating parts, moment of inertia, etc.
(V) is the rotating speed of the flywheel.
(dv) is the change in rotating speed.
(ΔKE) is the change in kinetic energy.

From the formula $\Delta KE \doteq 2CVdv$ we see that a change of 3% in speed will cause a 6% change in kinetic energy.

From the typical engine having 66,000 ft.-lbs. stored energy, the change of 3% in speed releases 3,960 ft.-lbs. energy. This energy is used in overcoming the peak load that caused the slow-down. Since this change in speed took place in one-third second, it can be translated into horsepower as follows:

$$HP = \frac{3,960 \div 1/3}{550} = 21.5$$

Adding this 21.5 HP released from the stored energy in the flywheel to the 14 HP available from the engine itself, gives us approximately 35.5 HP to overcome this peak load. The period of small or no lead that follows this peak lasts much longer (see point B, Fig. 2), and allows the engine to restore this energy to the flywheel for the next peak load encountered; all with a very small speed fluctuation. This feature is not available in a multi-cylinder engine.

ADVANTAGEOUS CHARACTERISTICS OF HIGH-SLIP ELECTRIC MOTORS

Basically, electric motors are rated on the load they will carry without excessive overheating. Since heat is directly proportional to current, the lower the current peaks when confronted with a cycling load, the smaller the electric motor that can be used. In a high-slip electric motor when a peak load is applied, instead of attempting to maintain synchronous speed and thus causing a high current peak, the motor slows down or "slips". Just the opposite is true in a normal-slip electric motor, which has "stiff" speed characteristics and will try to power through these peaks to maintain a near synchronous speed; therefore, resulting in high current peaks. The current peaks encountered with either high-slip or normal-slip motors will be considerably higher than the full-load ampere rating of a properly sized motor used on a cycling load. However, since we are dealing with a cyclic-type load, the average or RMS current, as measured by a thermal ammeter, gives a much more accurate indication of the work being done. All other factors being equal, since the high-slip motor characteristically causes lower current peaks than a similar normal-slip motor, we are able to apply smaller high-slip motors than normal-slip motors in this service.

SUMMARY

This paper has been written to explain a method that is both quick and sufficiently accurate for sizing prime movers for pumping service. In addition, the characteristics of the available prime movers have been presented to show the advantages to be gained by judicious selection.

BIBLIOGRAPHY

1. J. Taylor Hood. "Selection and Application of Prime Movers for Oil Well Pumping", Proceedings of the Third Annual West Texas Oil Lifting Short Course, 1956.
2. J. S. Martin. "Peak Torque and Pumping Unit Counterbalance", The Petroleum Engineer, October, 1958.
3. John H. Day, Jr. "Production Testing of Oil Well

Pumping Motors", Proceedings of the Sixth Annual West Texas Oil Lifting Short Course, 1959.
4. Max H. Halderson. "Determining Actual Power Output and Derating Factors of Electric Motors that Drive Oil Well Pumping Units", AIEE Petroleum Industry Conference, Dallas, Texas, September, 1958.
5. Frank W. Beach. "Sizing Electric Motors for Oil Well Pumping", Third AIEE Electrical Conference, Kansas City, Missouri, September, 1956.

ACKNOWLEDGEMENT

The author wishes to express his gratitude to Mr. J. C. Slonnegar for his advice and consultation in the preparation of this paper.