Putting Pumping Well Problems In Focus

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In order to get the maximum benefit from dynamometer well studies, it is essential that all of the factors involved in the pumping problems on each individual well be viewed as a complete picture. Each factor involved should be brought into focus and fitted into the picture as a whole. Fig. 1 illustrates a mechanical polished rod dynamometer.

As an amateur photographer, it has been my pleasure to try many types of cameras and to develop films and make enlargements. One of the first things any amateur learns is that the picture you are trying to portray must be in focus. If a negative is out of focus, the enlarging process cannot bring it back in. Sometimes a photographer will purposely throw the background out of focus by using a large lens opening to call specific attention to the foreground subject.

(See Fig. 2 for dynamometer card)

In a recent study made to try to diagnose sucker rod failures, the following pertinent facts were brought into focus:

1. The sucker rods were blamed for the trouble. The operator felt it might be bad steel, bad workmanship, bad heat treatment. The laboratory analysis showed rods up to standard.
2. The well was reported running with a pumping cycle of 22-24" spm, and load calculations were made with this as a basis.
3. When a dynamometer study was made, the well was found to be running 25.2-44" spm. When the pumper was questioned, he said he ran it as fast as the engine would run.
4. The stroke length was 44" but the crank holes would allow up to 74". When the question was raised, "why not lengthen the stroke and slow the well down?" The answer was, "the engine will not pull the well with a longer stroke."
5. Specific data on well production showed that water encroachment had increased recently due to the water disposal from offset wells in the same producing formation. This was the real reason for speeding up the pumping cycle.

Thus you see, the characteristics and speed of the engine determined the load and reversals on the sucker rods. The life of sucker rods in this well could be greatly increased with a longer slower stroke for the same production.

In order to get a true clear picture of any particular pumping well, we must focus our attention on all of the following factors:

1. Load
   A. Static
   B. Dynamic
2. Speed
3. Torque
4. Horsepower
5. Efficiency

Let us discuss each one separately; or staying on the photographic theme, let us focus on each one separately without regard to the other factors.

Load (Static)

When we speak of load, we are concerned with the load at the polished rod. The static forces contributing to that load will first be considered.

Dead weight of sucker rods in air is easily calculated from the weight per foot of the various sizes. Tables and charts are also available giving the average weight per foot of combination or tapered rod strings. Ref. 1.

(Reference numerals refer to the bibliograph at the end of this article.)

The weight of rods in fluid is simply the weight in air minus the bouy-

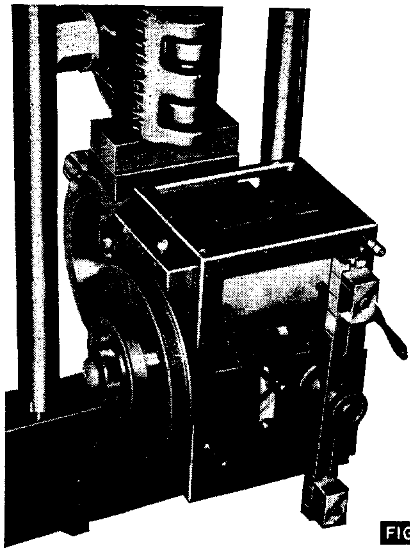


FIG. 1 — DYNAMOMETER

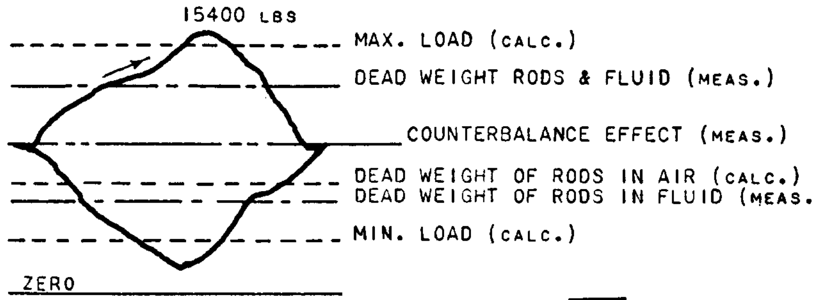


FIG. 2

WELL DATA

SPEED AND STROKE	25-44"
PRODUCTION	8 OIL 566 WATER
RODS	2970 FT. 7/8"
PLUNGER	2-1/2"

FIG. 3A — TESTING FOR STANDING VALVE LEAK

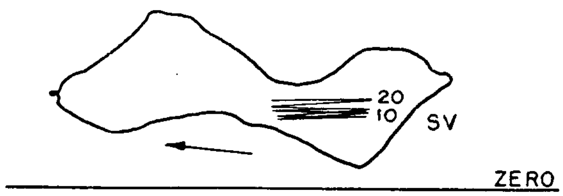


FIG. 3B — TESTING FOR WORN PLUNGER OR TRAVELING VALVE LEAK

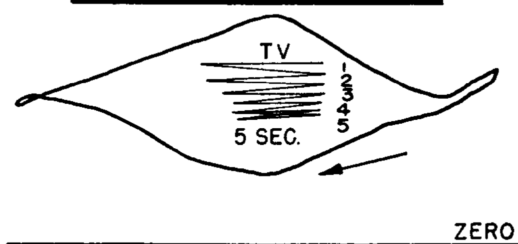


FIG. 4 — DEPTH CHANGE; CONSTANT SPEED

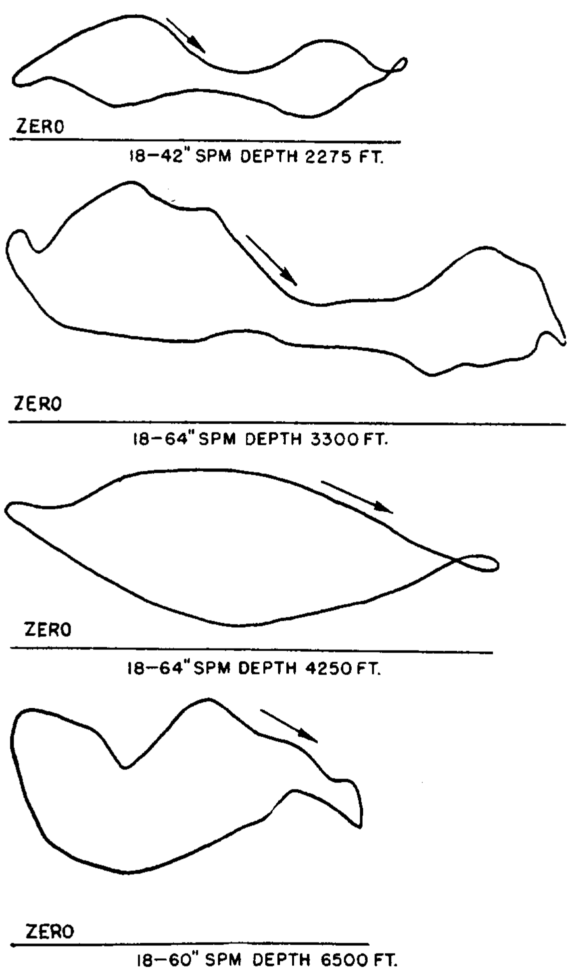
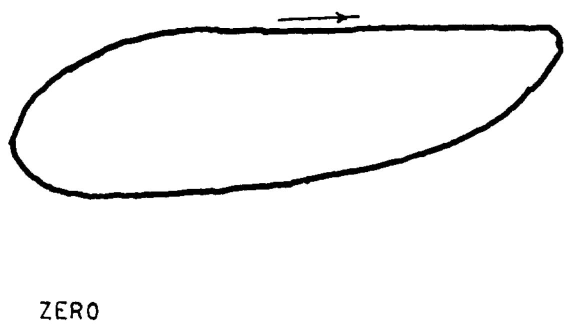


FIG. 5 PUMPING HEAVY VISCOUS OIL



WELL DATA

PUMPING CYCLE	14.8-62" SPM
PRODUCTION	114 OIL 10 WATER
RODS	2020-1" 5150'-7/8"

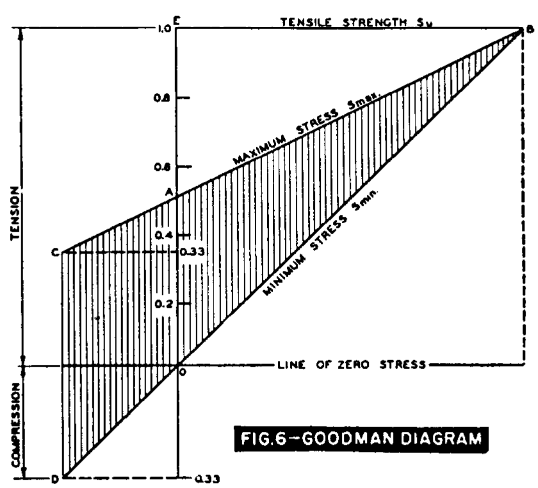


FIG. 6 — GOODMAN DIAGRAM

ancy of the fluid. Bouyancy is equal to the weight of the volume of fluid displaced by the sucker rods. The volume of sucker rods is equal to the wt. in air divided by wt. of a cubic foot of steel (490 lbs.) The volume of the rods multiplied by the weight of a cubic foot of water (62.5 lbs.) would give the bouyancy if the rods were suspended in fresh water. To correct this factor for the individual well, multiply by the specific gravity of the well fluid.

For wells producing solid fluid, the actual specific gravity may be easy to obtain; but in gassy wells, it may have no definite value.

The dead weight of rods in fluid may be checked with the dynamometer by stopping the motion on the down stroke so the full fluid colum is supported by the standing valve and tubing 3. See Fig. 3A.

If this reading shows the rod load too light, there may be some excessive down-the-hole friction. This reading is important in dynamometer well study work because it is a quick check to see whether the load readings make sense. It also checks the operator as to whether he is reading the right scale on the dynamometer. Sometimes this reading may reveal discrepancies in rod records.

Dead weight of fluid is the next consideration and in addition to knowing the specific gravity of the fluid the pumping fluid level should be known in order to make proper calculations.

The dead weight of the fluid is checked with dynamometer by stopping the pumping motion on the upstroke so that the fluid column in the tubing is carried by the traveling valve, pump plunger, and sucker rods.

In making this test, the reading must be taken quickly because the leakage past the plunger will soon reduce it. If the traveling valve is leaking it will reduce more rapidly as the fluid column load shifts to the standing valve and tubing. This change over is not instantaneous because of the relative stretch of the rods and tubing as the load is transferred. It should be noted that in checking the dead weight of fluid, the fluid level is the working fluid level and not the static level.

In checking fluid load, the dynamometer is used to detect leakage of the pump valves 2. Some experience is needed in each type and fit of pump at various depths to know what might be the normal rate of leakage past the plunger. See Fig. 3B.

Of course the pressure differential across the pump plunger affects the rate of leakake and this differential increases with depth and lower fluid level.

In a very high fluid level well, the upstroke dead weight and the downstroke dead weight will be very nearly the same because there is very little fluid load.

If abnormal dead weight readings are recorded, look for friction factors like crooked holes, paraffin, slack in tubing, tight pump plungers, etc.

One other static load determination made with the dynamometer is the peak counterbalance effect at the polished rod. The method of doing this and its significance is discussed under the general heading "torque."

Figure 2 shows the position of the dead weight of rods and fluid as referred to the load recorded by the dynamometer.

After the static forces have been considered, the problem is to get a picture of what should happen to the polished rod load under pumping conditions.

Load (Dynamic)

To determine the force on any mass such as the sucker rod string, we go back to fundamental physics and the relationship that force equals mass X acceleration. The conventional beam pumping unit is so constructed that when running at constant speed the motion at the polished rod is approximately simple harmonic. This is the same type of motion generated by a freely swinging pendlum.

It should be pointed out that calculation of peak dynamic rod load is complicated by the fact that the whole mass of the rod string is not accelerated at once because it acts like a long spring.

Mathematical analysis of simple harmonic motion shows that the peak acceleration which comes at the end of the stroke is given by the following expression N2L over 70500 where N is number of strokes per minute and L is the length of the stroke in inches.

To facilitate calculations, the term "impulse factor" was born. It is the factor by which the weight of the rods is multiplied to give force (Ref. 1).

For many years, the expected peak load was calculated by multiplying the sum of the dead weight of rods and dead weight of fluid by the impulse factor and the resulting load was generally greater than the peak load measured by the dynamometer. This led one investigator (Ref. 3) to suggest that since there was a time delay in first accelerating the rods and then the fluid that the acceleration factor be applied only to the rods. In other words, since the acceleration of rods and fluid do not come simultaneously, the forces do not add up to give the expected peak load.

In calculating peak load, the bouyant force on the rods is generally omitted since friction forces act in the opposite direction.

This simple method of calculation has proved very valuable in the general application of pumping equipment. Other formulas have been presented from time to time, and no doubt others can be presented to give closer results to measured loads for specific depths and speeds.

The short comings of any formula may be easily seen when it is evident that all of the following factors affect the peak load.

1. The motion of the pumping unit is not truly simple harmonic because of the geometry of the linkage even if constant crank speed is assumed.

2. Crank speed is not constant due to the speed variation of the prime mover. In shallow high volume wells this may be as high as 30 percent.

3. The operation of the governor on the engine might be such as to cause acceleration at different points in the stroke.

4. The well behavior can at various times considerably alter the fluid load.

5. The acceleration of the polished rod is greatly affected by the natural fundamental vibration of the rod string and the harmonics of this vibration, Ref. 1-4. This is true to such an extent that under normal condition a definite pattern of dynamometer card may be expected at specific depths and speeds. See Fig. 4. This changes the position of the peak load in the pumping cycle. No formula yet proposed takes into account the rod vibration or tries to define the position of the peak load.

6. Abnormal friction can greatly affect the load and the shape of the load characteristic.

Fig. 5 is an example of pumping a heavy viscous fluid. The load characteristics would normally be a second harmonic at the particular speed and depth involved. All vibrations are clamped out in this card.

Peak load alone is not sufficient in an evaluation of sucker rod and pumping unit performance. Equally important is the minimum load.

If we assume simple harmonic motion, the peak acceleration on the downstroke will be the same as on the upstroke but in the opposite direction. It is then logical to subtract the same percentage from the weight of the rods as was added on the upstroke to give the dynamic weight. Bouyant force must be considered because it acts in the same direction as friction.

The simple formula for minimum load is then:

PL (min) equals:

Wr (1-accelleration factor)

—Bouyancy

Wr. (1-c)—Wr 62.5 over 490

In evaluating the performance of sucker rods the range of load is just as important to know as the peak. If we refer to Figure 6 which is known as a Goodmon diagram, we can readily see that range of load greatly affects the endurance limits of steels.

Speed

In the discussion of pumping installations, speed generally means strokes per minute. However, there are other speed considerations that need to be added to the picture.

The strokes per minute generally determine the number of major reversals on the rod string which, with load and range of load, is the other important factor in producing fatigue failures.

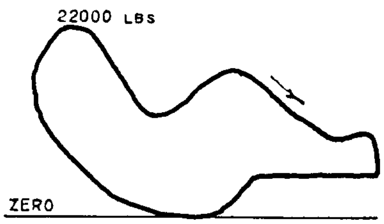
The limiting number of strokes per minute is generally that point at which the hanger runs ahead of the sucker rods on the downstroke. This of course depends on the friction forces which retard rod fall as well as the natural acceleration of gravity on the rods.

Operating with the minimum load at zero is generally bad because of the slack and jerks resulting and their effect on the surface unit.

At high operating speeds, stuffing box friction can be a big factor in retarding the natural fall of the sucker rods.

There is very definitely a maximum

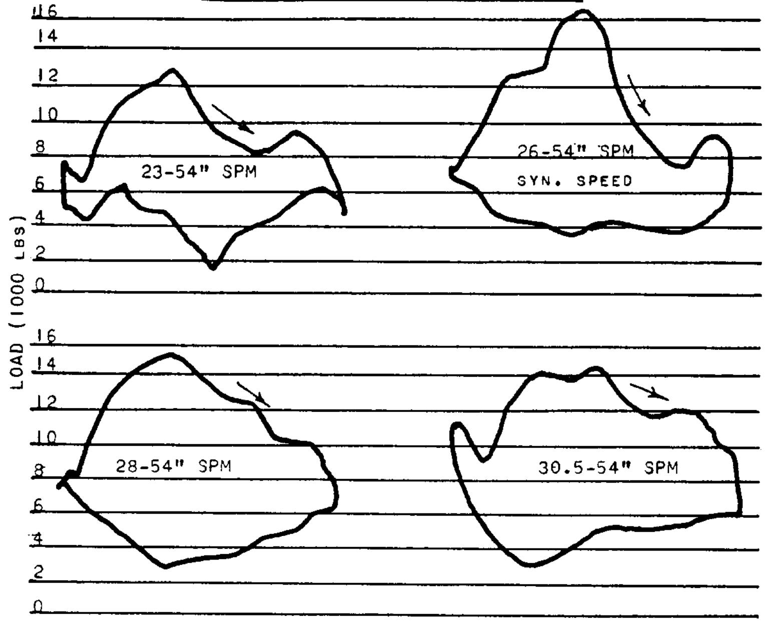
FIG. 7—FLUID POUND DUE TO HIGH SPEED



WELL DATA

PUMPING CYCLE 32-54" SPM
PRODUCTION 43 OIL 923 WATER
RODS 700'-1" 950'-7/8" 2300'-3/4"
PLUNGER 2-1/4"

FIG. 8—EFFECT OF SYNCHRONOUS SPEED



WELL DATA

PUMPING CYCLE AS SHOWN
RODS 1150'-7/8" 1150'-3/4"
PLUNGER 2-3/4"

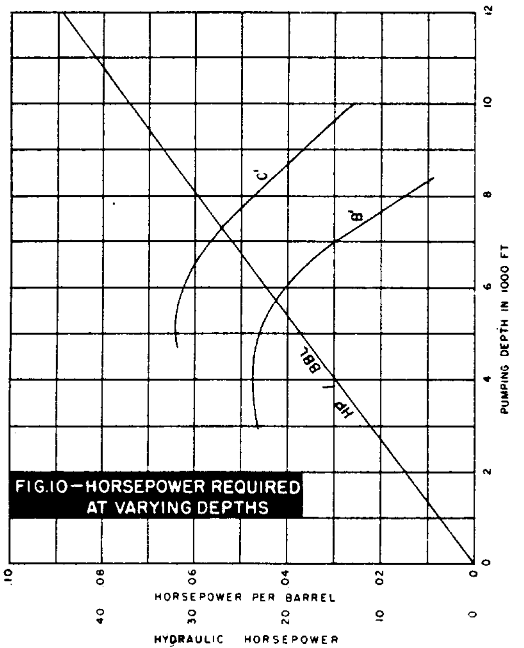
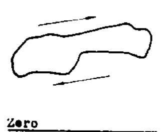
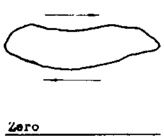


FIG. 10—HORSEPOWER REQUIRED AT VARYING DEPTHS

FIG. 11—ILLUSTRATING ELIMINATION OF FLUID POUND BY DECREASING PUMP DISPLACEMENT

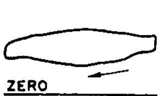


Maximum load 13000 lbs.
Minimum load 6500 lbs.
Range 6500 lbs.
Cycle 19 - 34" SPM
Pump 1-3/4" Bore
Prod. 83 bbls/day
Apparent Volumetric Efficiency 36%



Maximum load 12800 lbs.
Minimum load 6500 lbs.
Range 6300 lbs.
Cycle 19 - 34" SPM
Pump 1¼" Bore
Prod. 86 bbls/day
Apparent Volumetric Efficiency 71%

FIG. 12—FRICTION IN STUFFING BOX



MAX. LOAD 11,000 LBS.
MIN. LOAD 4,730 "
RANGE 6,270 "
SPEED 14.5 SPM
STROKE 44 IN.
POL. ROD HP 7.04

MAX. LOAD 10,500 LBS.
MIN. LOAD 5,570 "
RANGE 4,930 "
SPEED 14.5 SPM
STROKE 44 IN.
POL. ROD HP 5.5

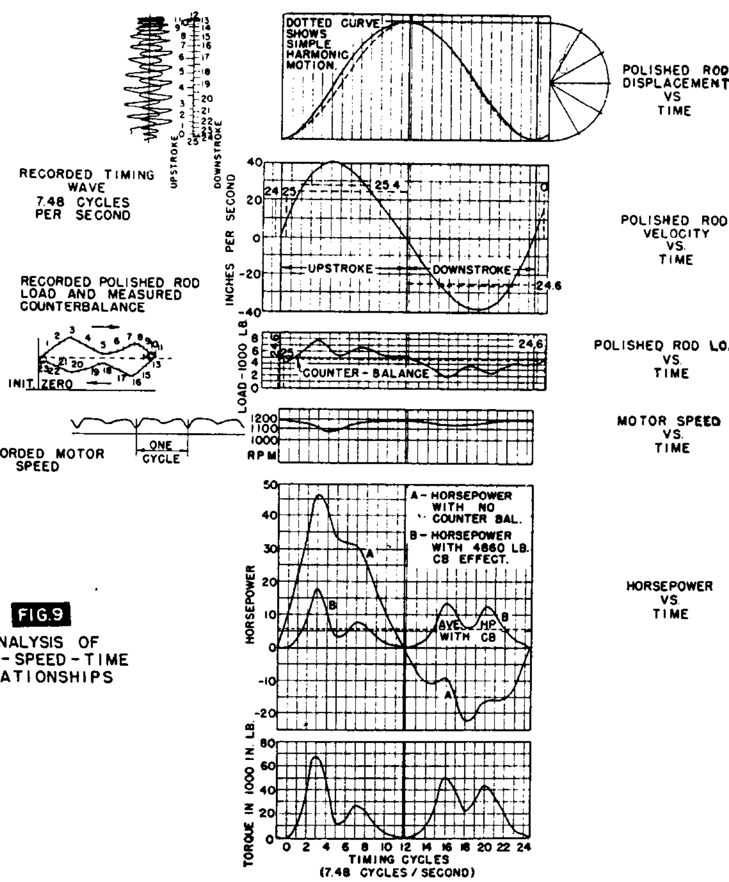


FIG. 9

ANALYSIS OF LOAD-SPEED-TIME RELATIONSHIPS average polished rod speed above which it is impractical to operate and that is 4,000 inches per minute.

Speed also must be considered in the operation of the bottom hole pump. The pump barrel can fill only so fast depending on the character of the fluid, the restrictions through the pump valves, and the bottom hole pressure.

Fig. 7 is an example of a well running as fast as the rods will fall. The pump barrel is not filling because of lack of fluid but because there simply is not sufficient time for it to fill. The pump has good submergence at all times.

When the prime mover is a gas engine, it should be operated at a speed which is in line with its design.

The counterbalance effect produced by beam counterbalence weight is greately affected by speed whereas that of rotary weight is affected very little.

Sometmies it is possible that abnormal loads may occur at harmonic speeds. Fig. 8 shows a well where the load is appreciably higher at the 4th harmonic speed of 26 SPM than at higher speed.

Torque

Mathematically expressed, torque is the product of the tangential force on a crank multiplied by the distance from this force to the center of rotation. It might be crudely stated as the amount of twist present at any moment on the crank shaft of a pumping unit.

Torque is also expressed by the following relationship:

Torque equals horsepower x 33000 x 12 divided by 2 TT rpm

If the gear box of a pumping unit could transmit the necessary horsepower to pump a well under a steady stated condition, the torque requirements would be very low. As an example if a steady 25 HP were transmitted at 20 spm the torque would be 78700 in.lbs. Pumping unit gear boxes were at one time rated on a horsepower basis and as a comparison a unit rated at 23.1 horosepower must also have a 114000 in. lbs. torque capacity. In order to measure the peak torque of a gear box conveniently, we must use the dynamometer card and measured counterbalance.

First, let us consider the counterbalance effect at the polished rod and how it affects the torque. The peak counterbalance effect is measured when the crank and pitman are at right angles to each other. This peak effect is the one given in tables furnished by pumping unit manufacturers.

In order to check the counterbalance effect with the dynamometer, the procedure is as follows: Ref. 5. Stop unit with crank and pitman at right angles to each other on the upstroke and set the brake. Then provide a chain to carry the difference between the dead weight of the rods and the counterbalance effect. Release the brake and make a recording of load by engaging the stylus and pulling the string on the dynamometer.

It is well to check the counterbalance effect in the same manner both on the up and down stroke because pumping rig geometry may make these forces quite different. The rotation of the prime mover (particularly electric motors which are easily reversed) should be such as to have the higher counterbalance effect on the upstroke provided there is an appreciable difference in the up and down stroke.

The only other place torque need be considered in a pumping installation is in connection with the prime mover if it is a gas engine installation with little fly wheel effect. Engines are de-rated for pumping application to take care of these requirements, but then must be run at near rated speed in order to give them the chance of good performance.

Torque is further discussed under the consideration of instantaneous horsepower. See Fig. 9.

In order to lift a certain weight of fluid a given number of feet in a given time, a very definite horsepower is required because the factors mentioned are in foot pounds per minute, and we know 33000 of these make one horsepower.

The theoretical (hydraulic horsepower) required to lift fluid in a well is then expressed as follows:

HP equals bpd x H x Spgr divided by 135600

Or it may be expressed as follows: HP equals 0.000017 x bpd x psi where bpd equals barrels per day and psi the pressure represented by the net lift in feet and H equals net lift in feet.

In order to get the true hydraulic horsepower, the working fluid must be determined by measurement.

In general the prime mover horsepower is selected on the basis of 2 to 2 1/2 times the hydraulic which is assuming the overall efficiency to be between 40 and 50 percent. Because the horsepower load on the prime mover is cyclic, it must be rated with enough excess capacity to take care of peaks.

It is common practice for electric motor manufacturers to apply a cycle load factor based on polished rod speed in selecting motors for pumping.

The polished rod horsepower is a measurement of what it is actually requiring to pump the well. It may be less than the hydraulic horsepower if the well is semi-flowing. It may be overly large when a well is pumping too fast or has excessive friction.

When the horsepower of a jet airplane is computed it is the product of thrust (pushing force in pounds) times the speed. The horsepower being expended by a train locomotive is the draw bar pull times the speed.

If we know the instantaneous velocity of the polished rod on a pumping well with the corresponding instantaneous load, we may readily compute an instantaneous horsepower curve. Then if we know the instantaneous speed of the crank, we can get a full torque curve. (Ref. 6 and 7).

Figure 9 is an example of how this is applied. This particular example shows a dynamometer card taken at a speed that produces a dumb-bell shaped card because it is operating near the harmonic. The unbalanced load near the middle of the stroke shows very little unbalanced load hence it might be assumed that the peak torque is low. Due to the rig geometry, the product of velocity and load is greatest at a point approximately one quarter of the upstroke travel.

In considering the maximum productions which may be obtained at various depths and translating this into horsepower at the pump, it is soon evident that the peak horsepower it is possible to transmit by sucker rods is at a depth between 4000 and 5000 feet. The limit below 5000 feet is set by the size of plunger and plunger speed. Fig. 10—Curve B' is for pumping units with a maximum stroke of 74". Curve C' is for long stroke units up to 30 feet.

Efficiency

Every effort should be made to pump a well with apparent volumetric efficiency that is as high as possible. When efficiency is low, the well is often speeded up to make up for it and this means unnecessary loading and fatigue of rods and unit.

To correct low volumetric efficiency, we must look at all the factors which may cause it. Some of these factors are listed below:

1. Pump plunger too large resulting in plunger undertravel.
2. Fluid gassy resulting in partial gas lock of pump.
3. Solution gas coming out of solution after going through the pump.
4. Improper pump submergence.
5. Plunger speed too high. See fig. 7.
6. Viscous fluid not allowing pump to fill in time.
7. Pump off and fluid pound. See Fig. 11.

By comparing the polished rod horsepower to the theoretical hydraulic horsepower, a factor called lift efficiency may be calculated. This may be used to check down-the-hole losses.

A tight stuffing box can cause unnecessary use of power as shown in Figure 12.

The efficiency of the surface unit from prime mover input to polished rod may be checked on electrified installations when suitable instruments are available to measure the motor input.

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