Dynamics of Pumping for Maximum Products

By ROBERT H. GAULT
Bethlehem Steel Company, Supply Division

EVALUATING METHODS

Many of us today are being confronted with a new and different problem, that of obtaining maximum production from a given well. This problem can arise from a number of different sources. Perhaps we are starting a waterflood program and we need to utilize our present equipment as far as possible in the life of the flood or perhaps we are finding that our wells are making an increasing volume of water and that we must handle increasing amounts of fluid to obtain maximum production. In addition, there is the every day problem of those of us who are responsible for equipment selection. We need to know the maximum capabilities of our equipment design and the best methods to utilize these capabilities; we can then select adequate equipment for the full life of the well at minimum installation cost. This means that at some time in the well life, we must expect this equipment to deliver its maximum production capabilities.

Most of us today look at only three or, at the most, four possibilities in our search for maximum production. Many times we do not even attempt to evaluate these various methods to find out which one will give as the best utilization of the equipment at the least cost. The most common factors which are considered:

1. Increase the number of pumping cycles per minute.
2. Increase plunger diameter.
3. Increase stroke length.

It would be well, at this point, to evaluate these changes in view of installation capabilities so that from a design standpoint only we can realize the most production possible.

The most commonly used formulas for sizing equipment are:

$$\text{Peak Polished Rod Load} = Wf + Wr\left(1 + \frac{N^2L}{70{,}500}\right)$$

$$\text{Minimum Polished Rod Load} = Wr\left(1 - \frac{N^2L}{70{,}500}\right) - Wr\,\frac{62.5}{490} = Wr\left(.8725 - \frac{N^2L}{70{,}500}\right)$$

$$\text{Counterbalance} = \frac{PPRL + MPRL}{2} = .936Wr + \frac{Wf}{2}$$

$$\text{Peak Torque} = (PPRL + CB)\frac{L}{2} = \left(\frac{Wf}{2} + .064Wr + \frac{WrN^2L}{70{,}500}\right)\frac{L}{2}$$

As an illustrative example of how each of the three factors mentioned above (speed increase, plunger diameter increase, and stroke length increase) affect equipment capabilities, let us use the following:

We are presently pumping from 4000 feet using a 1-1/4 inch bore pump and 10 - 42 inch SPM with 3/4 inch rods and producing at the rate of 61 BPD, based on polished rod travel and 80 percent overall subsurface efficiency. Neglecting for the moment the changes in stretch, overtravel, and pump efficiency which will occur, let us see what will happen if we hold two of the above mentioned variables as they are at present and change the other to get 120 BPD based on polished rod travel and 80 percent subsurface efficiency.

1. Change SPM to 20 SPM. New conditions will be 20 - 42 inch SPM, 1-1/4 inch bore pump, 3/4 inch rods:

	PPRL	MPRL	CB	PEAK TORQUE	MAX. ROD STRESS	RANGE OF LOAD
Before Change	8318#	5330#	6824#	31370"#	18,800 PSI	36%
After Change	9488#	4160#	6824#	55,944"#	21,500 PSI	56%

2. Change plunger size to 1-3/4 inch. New conditions will be 10 - 42 inch SPM, 1-3/4 inch bore pump, 3/4 inch rods.

	PPRL	MPRL	CB	PEAK TORQUE	MAX. ROD STRESS	RANGE OF LOAD
Before Change	8318#	5330#	6824#	31,370"#	18,800 PSI	36%
After Change	10,358#	5330#	7844#	52,794"#	23,500 PSI	49%

3. Change stroke length to 84 inch. New conditons are 10 - 84 inch SPM, 1-1/4 inch bore pump, 3/4 inch rods.

	PPRL	MPRL	CB	PEAK TORQUE	MAX. ROD STRESS	RANGE OF LOAD
Before Change	8318#	5330#	6824#	31,370"#	18,800 PSI	36%
After Change	8708#	4940#	6824#	79,128"#	19,750 PSI	43%

From the above illustration it is evident that we need to evaluate each of these changes as they influence each of the component parts of the pumping equipment.

Sucker Rods

Changing stroke length is the most desirable from the standpoint of the rods and results in the least increase in both rod loadings and in stress range. Increased plunger size results in the largest load but the next to smallest range of stress and, in addition, fewer cycles of stress. The least desirable, from the standpoint of the sucker rods, is an increase in speed.

Pumping Unit Structure

Changing stroke length would again be most desirable. Where structural limitations are close, an increase in speed or in plunger size would most likely be settled by other factors. Probably the increased plunger size would cause less wear and tear on the unit structure, even though load is greater.

Counterbalance

A speed increase would require no additional counterbalance and is therefore best from counterbalancing standpoint. An increase in plunger diameter would require additional counterbalance equal to one-half the fluid weight increase. An increase in stroke length, while the indicated counterbalance remains the same, would require double the amount of iron to get the same counterbalance as before.

PEAK TORQUE

An increase in pump diameter would give the least increase in peak torque requirements as one-half the fluid can be counterbalanced, which means reducer must lift only the other half. The speed increase follows as a close second. In cases where rod load is small, this would actually result in lowest torque requirement, because increase in torque requirement due to speed increase is a function of rod weight. The highest torque requirement is for additional production through a longer stroke. By inspection of the torque formula it is evident why this is true, for one-half of the additional stroke length is multiplied by all other factors. Stroke length also is a factor in the acceleration constant. The total influence results in a much higher torque requirement.

To summarize the above, still with no consideration given to rod stretch, overtravel, or pump efficiency and rating: 1. best; 2. next best, and 3. poorest.

	SUCKER RODS	UNIT STRUCTURE	COUNTER-BALANCE	PEAK TORQUE
Increase Pump Size	2	2 - 3	2	1
Increase Pumping Speed	3	2 - 3	1	2
Increase Stroke Length	1	1	3	3

PROBLEMS OF CHOICE

We see then that there is really no clear cut case, so far, for any particular choice. So we must look still further for additional factors to aid us in our selection. We must analyze all of the many things which so far in this discussion have not been considered. If we are installing a new unit, we must weigh initial cost factors in all of these cases against the replacement and maintenance costs for successful operation. If we are utilizing present equipment, we must try to utilize the present equipment to its fullest and weigh costs of substitutions against repair and maintenance costs of present equipment.

As an example: If our present reducer would handle only a very slight increase in torque, we would look first at increasing pump size. With modern double displacement pumps this might be possible without making changes, but in many cases a larger diameter tubing might be required. Also, if rod loadings were critical, it might be necessary to change to a tapered string or replace the whole string. The cost of the replacement pump itself would be a factor in some areas. This cost must be weighed against the replacement of the unit and reducer with a larger one, possibly with a longer stroke length; it is seldom possible to obtain additional stroke length with present equipment. In practice, probably a balanced combination of the three are generally used, but a thorough knowledge of the changes that each makes is important in the overall design.

Ignore Important Aspect

Thus far in our discussion, we have ignored a very important aspect of maximum pumping. I am afraid that most equipment engineers and operators generally do the same. There has been a great deal of work done and many articles written on the subject of pumping dynamics, but the knowledge gained has not been put to a very good use. In the first place, the authors themselves make no attempt to correlate their findings with each other. Then, too, many of the formulas and charts are very hard to assimilate and too obtuse, making it difficult to gain a mental picture that can be easily applied. Each of us has used one formula or another to estimate rod and tubing stretch and rod and tubing overtravel, but, in general, we have had very little faith in the results and many times our lack of faith has been justified.

About the only really good use we have made of our knowledge is the acceptance of tubing anchors to eliminate tubing movement so we would gain the maximum benefit from the plunger movement we were able to get. We have made no real effort to operate our pumping string so that we might get the maximum stroke from our plunger, or to design or to utilize our present design so that the maximum capabilities may be used.

Think about it for a minute. Mr. John Slonneger in 1937 had already observed that cable tool drilling equipment made use of the second harmonic motion to get maximum impact from the bit. This was a case of "feel" and "art" and experience. But in this same period most of the basic formulas which we use in modern day equipment design were developed. Slonneger, Sargent, Rieniets, Gilbert, Coberly, Marsh, and others, developed rather completely the dynamics of rod pumping, but because each well was and is an individual thing and because some factors change so much, very little use has been made of their findings.

If you are called upon today to size pumping equipment, chances are you are using a rod and tubing stretch formula which is based wholly on transfer of a static load, with no consideration given to the point at which this transfer occurs or the rate of load transfer. The same thing holds true in the calculation of peak polish rod loadings, loads due to acceleration factors, and rod overtravel.

We have long used the acceleration of simple harmonic motion to calculate what the peak loads on the unit and rods would be. In fact, if you will recall, I used it just a short time ago when I evaluated the methods of obtaining maximum production. Why do we do this? It is very simple. We know from experience that the use of this acceleration factor provides ample margin of safety. Actually, this acceleration only occurs in simple harmonic motion at the beginning and ending of the stroke and occurs then only for a very short period of time. At the middle of the stroke (where we commonly apply (1) this acceleration factor), acceleration is zero. Mr. Slonneger has said:

"This acceleration force can only be properly applied in second harmonic order cards where the greatest load on the rods is at the beginning of the stroke. All other orders the acceleration due to crank motion does not enter into it at all, but that the magnitude of the vibration depends upon the velocity of the rods and the degree of synchronism." However, in unit design we continue to use it because it more than compensates for the variable forces which we are unable to isolate and calculate, such as friction, etc.

Facts Necessary

We must, however, in the utilization of maximum installed equipment capacities forget this "booger factor" and get completely honest with ourselves. In our evaluation of the problem we must forget all claims about increased rod and unit loads due to increased acceleration factors and trust only what our facts show. Our starting point therefore ought to be a dynamometer, sonolog, tachometer, dynagraph and other available instruments. This method tolerates no guessing or supposition; it records only the facts. Let us then utilize as many of these facts as we are able to isolate. What are we now able to use from the mass of information available on a simple dynamometer card? There are a great many facts which we can now use from a dynamometer card! To itemize a few:

1. The true weight of the rod string in the well fluid.
2. The true weight of the fluid mass which must be carried on the pump plunger by the rods (any help from any source is already subtracted).
3. What the true pumping loads are and the exact point at which they occur.
4. Whether any disturbing forces are present and

their magnitude.

What are the practical uses of this knowledge?

1. We can know exactly the loads on our sucker rods and pumping unit.
2. We are able to determine what these loads mean in terms of pumping unit torques.
3. We are able to evaluate the work required to pump this well.
4. We can more closely estimate our plunger movement.

But another bit of knowlege of which we have made very little use is that we can predict in almost every case what the general configuration of this card shape will be. Charts and tables have been made which tell us what the card in general will look like. This is true! If you will go back through your dynamometer cards you can prove what I am saying. We have pictures in all of our technical journals that show the general appearance of cards that are called "second harmonic", "third harmonic", "fourth harmonic", etc., and usually in this chart or description, mention is made of destructive or synchronous harmonics. Possibly because of this, and because there have actually been found an extremely few cases of synchronism, very few people have seen fit to utilize this knowledge. One common usage made by men who have a lot of dynamometer experience is a guidepost. They look for the cause of this deviation when it is found. Gradually they are realizing that in most normal cases it can also be used to predict what will happen when a change is made in the pumping conditions.

Why Cards Have Particular Shape

But perhaps I should first attempt to illustrate why these cards do have this particular shape. We know that in any medium the transmission of force is not instantaneous. Even electrical impulses travel only at the speed of light, so that there is always a finite time between the initiation of a force and its arrival at its destination. We have established that the rate of force transmission in steel is around 15,800 feet per second and 17,000 feet per second, depending on the shape of the steel bar and the damping forces involved. Therefore, any force applied to a sucker rod string at the polished rod takes a finite time to arrive at the plunger. This force does not travel as uniform motion throughout the string, but in a localized area exactly as a shock wave travels in a water system to give water hammer.

In other words, if we hit a long steel bar with a hammer, the force would travel as a localized compression until it reached the end of the bar. The end of the bar would be elastically distorted and when it returned to its original position would start a return shock wave back to its source. If the hammer were resting on the end of the bar, it would actually be forced away from the end of this return force wave. If we know then the rate of travel of this shock wave in steel, we can tell how long ago a certain event occurred, much as an astronomer can tell with the speed of light how long ago a certain event occurred on a star.

Applying and simplifying this illustration then to a sucker rod string, we can see that the motion of the polished rod imparts a force at the top of the string which must be transmitted to the plunger through the rod string. If the rod string were 15,800 feet/4 or 3,900 feet deep, it would take one second to make the round trip. If our unit were pumping 60 SPM, it would also make one cycle each second, so that the return wave would coincide and we would have first order harmonic motion. Again, if we were pumping at 30 SPM, it would take two seconds to complete a cycle and in this time interval the original force application would go to bottom, return, go to bottom and return again, so that two complete cycles would be completed. If these were in phase, we would have second harmonic motion. And, of course, at 15 SPM we would have third harmonic motion and at 7-1/2 SPM, we would have fourth harmonic motion. Each of these conditions could conceivably be in phase and amplification could take place, but in practice this happens so seldom that to worry about it is inconceivable. Let us forget all about amplified harmonics and use this phenomonen to our useful purposes.

As both Slonneger and Rieniets have pointed out, the general configuration of a card changes from only two causes over which we may exercise some control.

1. The length of the sucker rod string, and
2. The number of cycles per minute (number of times the force is applied).

There is also a secondary consideration which affects the speed of transmission that we can change in tapered strings; that is the percentage of the taper. (A 50-50 taper gives highest transmission speed). We can, with this knowledge, put harmonics to work for us. The other major factor in the shape of a particular card is the fluid load that we are lifting. It will in general make the card thin or full.

Utilize Knowledge

First, let us consider how we can utilize this knowledge to realize maximum benefit from it. What limitations does it have? There are quite a few limitations, but not so many that this knowledge is of no use to us. There are certain wells where outside factors, such as damping, gas lock, fluid pound, friction, viscous crudes with slow plunger fall, etc., prevent or hamper the transmission of forces and they do not follow the prescribed patterns. In these wells our knowledge must be of limited application. In the vast majority of wells where maximum production is required, however, it can by flexible application do amazing things.

But to utilize them, consider the load capacities of present day pumping units. If we start with a set of torque factors and work backwards, we can establish what we shall call a "permissible load diagram." The construction of this diagram is very simple and outlines an area within which all well loads must be confined to stay within load and torque capacities of the unit.

If it is constructed to scale and copied on transparent plastic, it can be laid on the measured counterbalance line and it will tell you immediately if the loads measured by the dynamometer are causing a torque overload. But this is not its whole value! It also defines an area within which the position of loads can be shifted and changed, by changes in speed, depth or rod taper, so that we can utilize the maximum torque capacity of our unit.

Let us construct this "Permissible Load Diagram." This can be a "tailor made" diagram which outlines loads which can be carried by a particular unit with a particular stroke length without overloading the unit. If it is to be used in the form of an overlay, the length of the diagram should be the same as the card length taken on this unit and the scale used should be the same as the ring and setting of the dynamometer. It can also be used as a guide only to show what general card shape permits the maximum work to be done by the unit without overload.

The formula to determine torque from load with Torque Factors is:

$$\text{Net Reducer Torque} = \overline{TF}\,(W-B) - M\sin\theta$$

Where Θ = Position of crank, degrees
M = Maximum moment of counterbalance "#
B = Structural Unbalance, lb.
W = Measured polished rod load (lb.) at position of rods corresponding to θ
$\overline{TF}$ = Torque Factor corresponding to θ

To construct a "Permissible Load Diagram" for a particular unit, the API Pumping Unit Stroke and Torque Factor sheet which applies to the particular unit should be used. To find permissible loads which would not exceed the unit rating, we would insert the API reducer rating as the "Net Reducer Torque." And using as counterbalance moment the present counterbalance setting of the unit, we would solve the equation for load.

$$W = \frac{\text{API Reducer Rating} + (\text{Present Counterbalance Moment})\ (\text{Sin } \theta)}{TF} + B$$

Values for the "API Reducer Rating" would be positive on the unit upstroke and negative on the unit downstroke.

When loads are calculated for each position of the polished rod and crank, they should be plotted using the length of the card as the base line. The indicated polished rod position would determine the placement of the load on the constructed card. The load scale of the dynamometer would be used to calculate the placement of the counterbalance on this "Permissible Load Diagram" from the following:

$$(\text{Measured CB, lbs.}) = \frac{M}{TF \text{ at } 90^o} + B$$

This calculated line should be drawn across the constructed card in the same manner as a counterbalance line is recorded on a dynamometer card. (It should be noted here that theoretically a new "Permissible Load Diagram" should be drawn for each counterbalance position, but the error is very small unless there is a wide variation in the counterbalance setting).

If an overlay of this diagram is made, it can be laid directly over the dynamometer card in the field, matching the two counterbalance lines. Then any loads which lie outside this diagram will show that the unit is overtorqued at that point. This device would prove very useful in an area where all wells are equipped with the same unit and pumping from about the same depth.

We know that in a particular well at a particular speed, the general configuaration of the card changes very little with change in stroke length or pump size. Of course, the areas enclosed will vary greatly but usually it will remain in the same order it originally was and load peaks and valleys will generally maintain their relative position. Using this knowledge and varying the speed or depth until we get the best desirable card shape to fit the unit diagram, we can then consider stroke length and plunger changes. As we become more experienced in the use of this device, we can probably predict necessary changes very closely.

Maximum Bottom-hole Plunger Stroke

There is one other very important use we can make of this knowledge. We can use it to gain maximum bottom-hole plunger stroke. We know that to gain maximum overtravel of the plunger, it is desirable to have a load as much heavier than the static rod load line as is possible at the beginning of the upstroke. This means the rods have stretched through dynamic load and the plunger is at a lower point, than it would be due to static rod weight only.

In addition, it is desirable that the rod load be as low as possible (below the static rod load line, if possible) at the end of the upstroke. This indicates that the rods have contracted below their normal length and the plunger is above the position it would normally be if the rods were extended by their static weight. What happens in the other portions of the stroke are of little importance from the standpoint of overtravel. Only the ends have any bearing on the total plunger travel. If we can adjust speed, depth, or speed of transmission slightly so that we can gain additional production by the use of harmonics, we have in effect increased the pumping efficiency. The Slonneger graphic method of determining plunger travel bears out the above and in many cases a slight change in pumping conditions could result in a very large increase in production through utilization of natural rod harmonics.

CONCLUSION

It is realized that the practical application of the above theories and suggestions might in many cases prove more difficult than the gain would warrant. However, we have had harmonics around a long time and we will continue to have them for a lot longer time. Let's put them to work. Almost every other industry is using them now. Why should we wait any longer?