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WAYS TO DECREASE PRODUCTION COSTS FOR SUCKER-ROD PUMPING

Mainly due to its long history, sucker-rod pumping is a very popular means of artificial lift all over the world, roughly two-thirds of the producing oil wells are on this type of lift. To maximize profits from these wells in the ever-changing economic situation with rising costs of electric power, installation designs must ensure optimum conditions. In the paper, basic considerations on ensuring profitable rod pumping operations are given. The key topics of installation design (pumping mode selection, optimum counterbalance, rod string design) are addressed and their role in the improvement of sucker-rod pumping operations and the reduction of lifting costs is discussed.

After a review of the surface and downhole energy losses in sucker-rod pumped wells, some key considerations on the ways to improve system efficiency are given. The most important task is the proper selection of the pumping mode, i.e. the combination of plunger size, pumping speed, stroke length, and rod taper design for lifting the prescribed amount of liquid to the surface. The best pumping mode maximizes the lifting efficiency and, at the same time, reduces prime mover power requirements and electrical costs. The operational efficiency of the surface equipment is improved by using an optimum counterbalancing of the pumping unit. To achieve an ideal sucker-rod pumping system the mechanical design of the tapered rod string must be properly made. The paper gives aspects and details of installation design improvements along with practical examples.

IMPROVING ENERGY EFFICIENCY

In order to increase the profitability of sucker-rod pumping installations, the reduction of operating costs is of prime importance. Since the majority of installations is driven by an electric motor and the cost of electric energy has steadily increased in recent years, energy losses both downhole and on the surface must be minimized. After a discussion of the possible sources of energy losses in the rod pumping system, an overall efficiency formula is derived. An evaluation of this formula allows important conclusions to be drawn on the most efficient pumping system.

Downhole Energy Losses

The rod pumping system does its useful work by lifting the given amount of liquid from the well bottom to the surface. The hydraulic power is easily calculated based on the depth of effective lift and the volume of the liquid produced:

$$P_{hydr} = 7.36\text{E}-6\ Q\ SpGr\ L_{dyn} \qquad (1)$$

where:

P_{hydr} = hydraulic power required for lifting the liquid, HP,

Q = liquid production rate, bpd,

$SpGr$ = specific gravity of the produced liquid, -, and

L_{dyn} = dynamic liquid level in the well, ft.

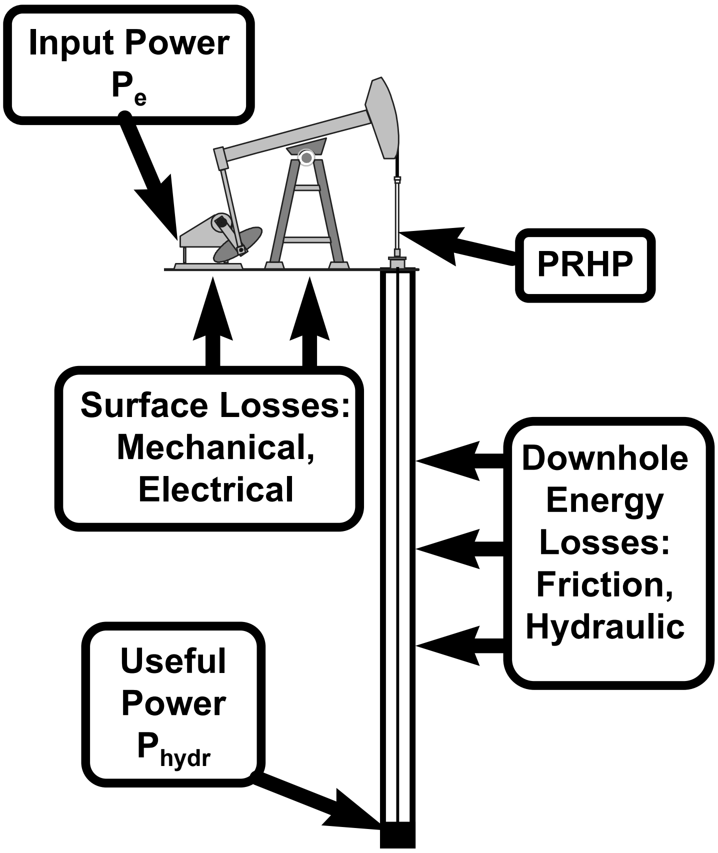

Fig. 1. Energy flow in the sucker-rod pumping system

The sources of downhole energy losses in the sucker-rod pumping system are the pump, the rod string, and the fluid column. In the pump, frictional and hydraulic losses as well as liquid leakage take place. The rod string, while reciprocating in the tubing, rubs against the tubing wall causing mechanical friction, especially in deviated wells. Produced liquids impart damping forces on the rod string and cause other hydraulic losses as well. All these losses, in addition to the hydraulic power required for fluid lifting, must be overcome by the mechanical work performed by the pumping unit at the polished rod, as seen in **Fig. 1**.

The energy required for operating the polished rod at the surface is thus the sum of the useful hydraulic work performed by the pump and the downhole energy losses. This power requirement is a basic pumping parameter called the polished rod power or **PRHP**. It represents the mechanical power input to the pumping system at the polished rod and can be experimentally found from the area of the dynamometer card taken on the well.

The energy efficiency of the downhole components of the sucker-rod pumping system can be characterized by the relative amount of energy losses in the well. The parameter widely used for this purpose is called **Lifting Efficiency** and is the quotient of the useful hydraulic power and the power required at the polished rod:

$$\eta_{lift} = \frac{P_{hydr}}{PRHP}, \tag{2}$$

where

η_{lift} = lifting efficiency,-,
P_{hydr} = hydraulic power required for lifting the liquid, HP, and
PRHP = polished rod power, HP.

Surface Losses

On the surface, from the polished rod to the prime mover, mechanical energy losses occur at several places in the rod pumping system. Starting from the polished rod: frictional losses arise in the stuffing box, in the pumping unit's structural bearings, in the speed reducer (gearbox), and in the V-belt drive. It is customary to include all these energy losses into a single mechanical efficiency η_{mech} to account for all these effects.

The prime mover (in most cases an electric motor) converts the electric energy input at its terminals into mechanical work at the motor's shaft and this involves certain inevitable losses. Thus the electrical power taken by the motor is always greater than the mechanical power developed at the motor's shaft. The power losses in an electric motor are classed as mechanical and electrical. Mechanical losses occur in the motor's bearings due to friction, other losses include windage loss consumed by air surrounding the rotating parts. Of the electrical losses, copper loss is the decisive one, resulting in the heating of the motor due to the electrical current drawn. Usually, an overall efficiency η_{mot} is used to represent all losses in the motor, which, for average electric motors, lies in the range of 85% to 93%.

Optimum Energy Efficiency

If all energy losses occurring from the well bottom to the prime mover are considered, an overall efficiency for the pumping system can be defined. Since the system's useful work is represented by the hydraulic power used for fluid lifting and the total energy input is proportional to the required electric power, the energy efficiency is found from:

$$\eta_{system} = \frac{P_{hydr}}{P_e}, \qquad (3)$$

where:

η_{system} = overall efficiency of the pumping system,-,
P_{hydr} = hydraulic power used for fluid lifting, HP, and
P_e = electrical power input at the motor's terminals, HP.

A more detailed formula can be reached if the individual efficiencies in the system components are substituted in the above equation:

$$\eta_{system} = \eta_{lift}\ \eta_{mech}\ \eta_{mot}, \qquad (4)$$

where:

η_{lift} = lifting efficiency,
η_{mech} = mechanical efficiency of the pumping unit and the gear reducer, and
η_{mot} = overall efficiency of the electric motor.

An investigation of this overall efficiency formula allows some basic conclusions to be drawn towards attaining a maximum energy efficiency in rod pumping. First the relative importance and the usual parameter ranges of the individual terms must be analyzed. Of the parameters figuring in the equation, the possible values of both the surface mechanical efficiency, η_{mech}, and the motor efficiency, η_{mot}, vary in quite narrow ranges. At the same time, their values are not easy to improve upon; that is why their effects on the system's total efficiency are not very significant. On the other hand, lifting efficiency can be considered as the governing factor since it varies in a broad range depending on the pumping mode selected. Thus considerable improvements on the pumping system's overall energy efficiency can only be realized by achieving a maximum of lifting efficiency.

In summary, the basic requirement for achieving an optimum overall pumping efficiency is increasing the lifting efficiency. Since lifting efficiency mainly depends on the pumping mode selected (i.e. the combination of plunger size, stroke length, pumping speed, and rod string design), the proper choice of the pumping mode for sucker-rod pumping cannot be overemphasized. When designing a new pumping system or improving the performance of an existing installation, this must be the primary goal of the rod pumping specialist's efforts.

Selection of the Proper Pumping Mode

The pumping mode of a sucker-rod pumping system is defined as the combination of pump size, polished-rod stroke length, pumping speed, and rod string design. The number of standard API pump sizes and stroke lengths is high and pumping speed can be varied within a broad range. Since many rod string taper combinations are available also, a fairly great number of pumping modes may be possible in any case. For producing a given amount of liquid, however, most of the theoretical pumping modes turn out to be impractical or uneconomical, but the elimination of these still leaves a multitude of options to consider. [1 - 2] As described above, the optimum design is chosen from among the remaining pumping modes based on the value of their lifting efficiencies and the one with the maximum η_{lift} is selected.

Table 1: Pumping modes with the best and worst lifting efficiencies for lifting 500 bpd from 6,000 ft.

Pumping Mode	Best	Worst
API Rod No.	86	85
Pump Size	2 ½"	1 ¼"
Stroke Length	120"	144"
Pumping Speed	8.3 SPM	18.4 SPM
PRHP	23.5 HP	58.3 HP
Lifting Efficiency	94.1%	37.9%
Pumping Unit Size	C-912D-305-168	

Maximizing the lifting efficiency coincides with the case of setting the polished rod power, **PRHP**, to be a minimum. Namely, lifting a given liquid volume from a given depth (i.e. for a given hydraulic power) lifting efficiency and **PRHP** are inversely proportional. Application of this optimization concept, therefore, gives the most energy-efficient and thus most economical pumping mode for the production of the required liquid rate from the given pump setting depth. A pumping system design utilizing this principle results in minimum operational costs and in a maximum of system efficiency.

To illustrate the dominant effect of pumping mode selection on the efficiency of the pumping system, an example problem [3] is presented. It involves a pump set at 6,000 ft with the liquid level at the pump and a desired liquid rate of 500 bpd. The tubing string is anchored and Grade D sucker rods are used with a Service Factor of 1.

From the multitude of possible pumping modes the ones with the best and worst lifting efficiencies are given in **Table 1**. If the best mode is selected, the energy input at the polished rod is only slightly greater than the pump's hydraulic power, ensuring a

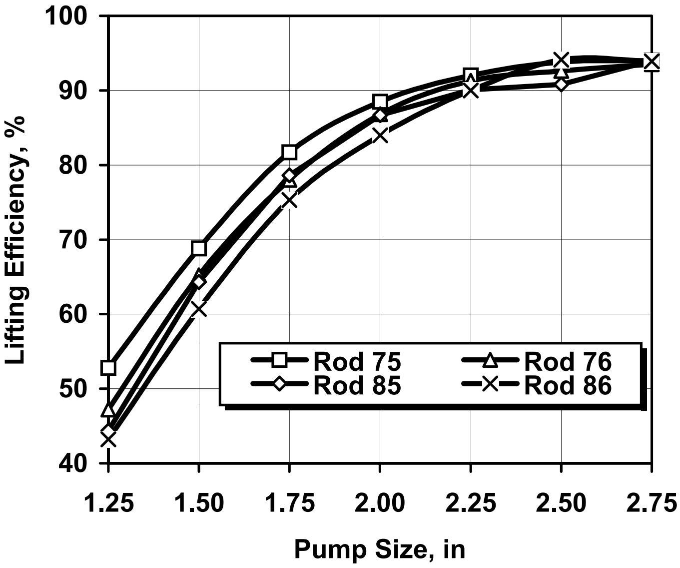

Fig. 2. Maximum lifting efficiencies for different rod taper combinations vs. pump size when producing 500 bpd from 6,000 ft

lifting efficiency of over 94%. The worst mode, on the other hand, requires almost three times as much energy as the best one for lifting the same amount of liquid from the same depth. It is also interesting to note that these two extreme pumping modes both require the same size of pumping unit. This example, therefore, demonstrates that big energy and operational cost savings can be realized by choosing the right pumping mode.

Additional calculation results are presented in **Fig. 2** where maximum values of lifting efficiencies for different rod combinations are plotted vs. pump size. An observation of the curves clearly shows that by increasing the pump size the attained maximum lifting efficiency values increase for all tapers. Therefore, use of bigger plungers with correspondingly slower pumping speeds is always advantageous and results in lower energy requirements.

Fig. 2 proves also, that, in line with practical experience, use of the heavier rod strings (like API 85 or 86 instead of API 75 or 76) considerably increases the power requirement for smaller pump sizes. As pump size increases, however, the difference in power requirement tends to be less pronounced. This phenomenon is explained by the relative importance of rod string weight in the total pumping load since bigger pumps involve greater fluid loads and rod string weight thus becomes a smaller fraction of the total rod load.

OPTIMUM COUNTERBALANCING OF PUMPING UNITS

Proper counterbalancing of a pumping unit evens out the torsional loads on the speed reducer during the pumping cycle. Without counterbalancing the torque loading on the gearbox would be high during the upstroke due to the high loads on the polished rod. During the downstroke, on the other hand, the rod string falling in the produced fluid would drive the pumping unit resulting in a negative torque loading on the gear reducer. Since no motor can operate under such heavily fluctuating loads, some means of counterbalancing the pumping unit had to be devised. These can take the form of beam or rotary counterweights or an air cylinder.

When counterbalancing a pumping unit, ideal counterbalance conditions are desired that can have many beneficial effects on the operation of the sucker-rod pumping system:

- Gearbox size can be reduced when compared to an unbalanced condition,
- The size of the required prime mover is smaller, and
- The smoother operation of a properly balanced speed reducer lowers maintenance costs and increases equipment life.

The measure of the evenness of the torsional load on the gear reducer is the mechanical **Cyclic Load Factor** (CLF). It can be calculated from the variation during the pumping cycle of the net torque on the reducer as the ratio of the root mean square and the average net torques:

$$CLF = \frac{\sqrt{\dfrac{\int_{\theta=0}^{2\pi} [T_{net}(\theta)]^2 \mathrm{d}\theta}{2\pi}}}{\dfrac{\int_{\theta=0}^{2\pi} T_{net}(\theta) \mathrm{d}\theta}{2\pi}}, \tag{5}$$

where

CLF	=	mechanical CLF,
$\mathbf{T_{net}}$	=	net torque on the gearbox at crank angle θ, in-lbs, and
θ	=	crank angle, degrees.

Due to its practical and economical importance, optimum counterbalancing of pumping units is a heavily discussed topic. From the many practical solutions developed over the years several different approaches [4] must be mentioned. These methods try to find the maximum counterbalance moment satisfying one of the following criteria:

- The peak motor currents are equal during the up-, and downstroke,
- The peak net torques on the up-, and downstroke are equal,
- The required mechanical powers for the up-, and downstroke are equal, or
- A minimum of the cyclic load factor is achieved.

Minimizing CLF

In this age of the computer the old field procedures for achieving optimum counterbalance conditions must give way to theoretically more sound procedures. Thus, from the above models, the sucker-rod pumping specialist's first choice should be the one that minimizes the value of the **CLF**.

The merits of minimizing the **CLF** for achieving optimum counterbalance conditions are shown in **Fig. 3**, which displays calculation results for an example well. All parameters are plotted vs. maximum counterbalance moment on the gearbox since this is the variable that can be changed at will by either moving the existing counterweights on the crank or by replacing them. As seen, the Cyclic Load Factor (**CLF**) curve clearly exhibits a minimum at about 300,000 in-lbs for the given case.

This figure can also serve to compare the recommended and the field procedures. For this reason, the variations of the peaks of upstroke and downstroke net torques are also plotted. The intersection of these curves represents the counterbalance moment required to set the two peaks equal, in order to meet one of the optimum counterbalance criteria mentioned above. The **CLF** value valid for this case is greater than the minimum **CLF** achieved by the recommended optimization model. Therefore, a proper approach to the optimization of counterbalance conditions is to find the counterbalance moment ensuring a minimum of the Cyclic Load Factor.

PITFALLS IN ROD STRING DESIGN

A properly designed rod string should provide failure-free pumping operations for an extended period of time. Rod string design aims at the determination of:

- The rod sizes to be used in the string,
- The lengths of the individual taper sections, and
- The rod material to be used.

In order to find an ideal solution to the above problems, detailed calculations should be performed with a proper consideration of actual well conditions. The two basic problems in rod string design concern: (1) how rod loads are calculated, and (2) what principle to use for the determination of taper lengths. At the time of design, of course, the anticipated rod loads are not known, and they also depend on the taper lengths that are about to be determined. Therefore, one has to rely on approximate calculations to find probable rod loads that will occur during pumping.

Design Principles

Early rod string design methods utilized the simplifying assumption that the string was exposed to a simple tension loading. An examination of the rod loads during a complete pumping cycle, however, shows that the rod string is under a cyclic loading. The nature of the loading is pulsating tension because the whole string is under tension

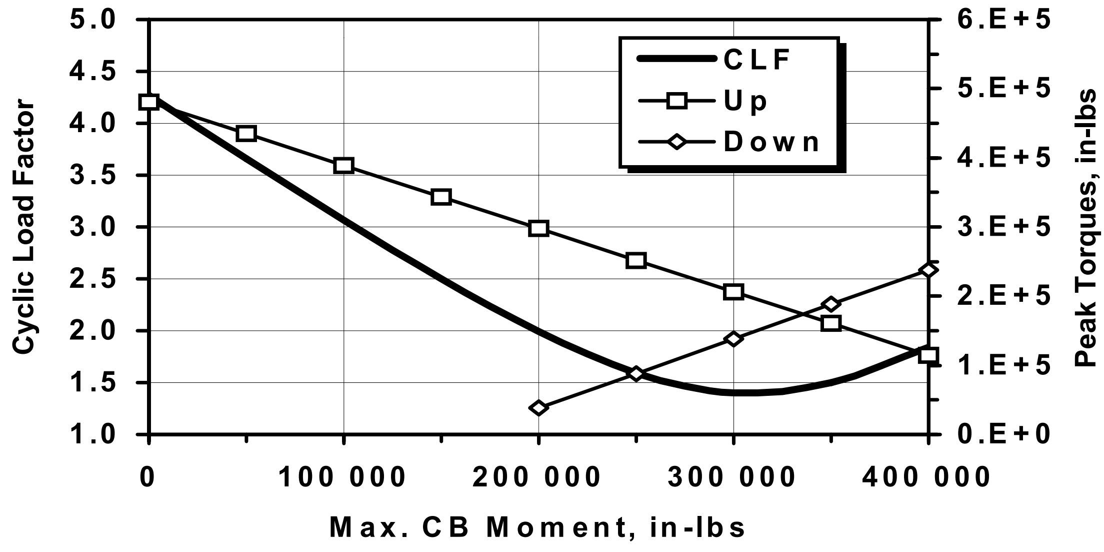

Fig. 3. Variation of CLF and peak upstroke and downstroke net torques with maximum counterbalance moment for an example case

at all times, but rod stress levels change for the up-, and the downstroke. This is why, in contrast to early design principles, sucker-rod strings should be designed for fatigue endurance.

After the pioneering work of **West** [5], the string design procedure of **Neely** [6] gained wide acceptance. It was adopted in 1976 by the **American Petroleum Institute** and rod percentages calculated by this procedure were included in the editions of **API RP 11L**. [7] Since then, the tables published by the **API** have been used to install thousands of rod strings, saving the time required for detailed string designs.

The use of the **API** rod taper percentages, as revealed by several investigators, has some inherent errors. These are related to the assumptions that were used but never disclosed for the calculation of taper lengths. The **API** taper percentages, as given in the **RP 11L** tables, do not vary with well depth, pumping speed, or stroke length. The only input variable being the plunger size, taper lengths determined from the tables for actual conditions can considerably differ from those calculated with the original **Neely** procedure. The differences between API tapers and actual calculation results lead to the conclusion that accurate rod string designs should be based on actual pumping conditions.

Recommended Design Procedure

The above discussion of sucker-rod string design methods reveals that each of them contains either questionable or very simplifying assumptions. The early designs do not consider fatigue loading and are thus inadequate for designing rod strings for cyclic loading. **West** [5] uses the almost obsolete **Mills** acceleration factor for calculating dynamic forces. **Neely**'s design principle [6] and his proposal for the distribution of dynamic loads are also subject to question. The **API** tables [7], as shown above, were developed for unknown conditions and cannot be used for design purposes without significant errors.

An ultimate rod string design, of course, would eliminate the above disadvantages and would calculate rod loads in each taper section using the solution of the damped wave equation. This approach, however, would involve very complex iterative procedures necessitating computation times prohibitive for design purposes. To solve this problem, the method of **Gault and Takacs** [8] gives a more theoretically sound design procedure than previous designs while requiring only moderate computational time. The goal of this string design method is to have the same degree of safety in every taper section. Actual Service Factor (**SF**) values are the same for all taper sections and the rod string is subjected to a uniform level of fatigue loading all along its length.

Example Problems

To show the consequences improper rod string design procedures can have, we present **Fig. 4** taken from a recent MS Thesis [9]. This figure involves three wells from the Hamada field in Libya and contains relevant parameters for current and optimum sucker-rod string designs. The designs are compared based on the minimum and maximum stresses occurring at the top of each taper. After plotting the calculated stresses on the modified Goodman diagram for the two cases, i.e.: the current state and an optimum design, one can draw important conclusions for improving sucker rod pumping operations in the given field.

The current taper percentages (shown in normal weight lines) clearly indicate very poor original designs. Generally, the strings are only lightly loaded and consequently over-dimensioned. Total rod string weights are thus more than necessary making some additional power requirements for lifting the heavier-than-needed string. The properly designed strings (shown in heavy lines) were calculated with the **Gault and Takacs**

[8] procedure. These designs, in contrary to the field practice, constitute properly and uniformly loaded rod strings.

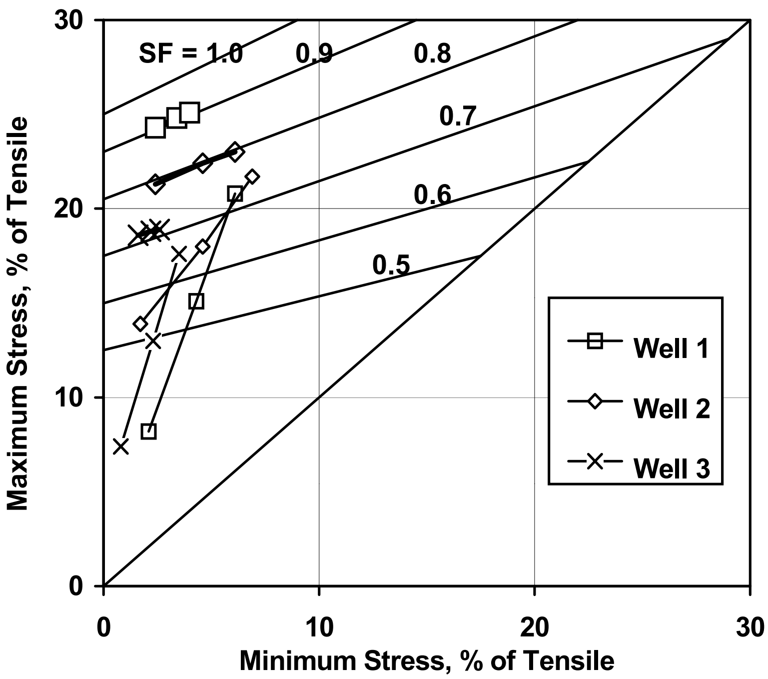

Fig. 4. Comparison of rod string designs in the Hamada field

The greatest improvement the proper design procedure ensures over field practice is the change in the level of fatigue loading in the different rod tapers. Field designs clearly exhibit a very biased rod loading, the top rod tapers always being much highly loaded than bottom ones. Under these conditions the upper, and especially the top tapers are bound to fail first. This general trend is very much in agreement with the rod failure pattern of the wells in the given field.

The properly designed strings, as seen in **Fig. 4**, have a very uniform fatigue loading in each taper. This is proved by the fact that points representing the individual tapers fall on lines parallel to **SF = const.** lines. Therefore, the level of fatigue loading in each taper is uniform and none of the tapers is predetermined to fail first.

CONCLUSIONS

Sucker-rod pumping is a well-known and widespread means to artificially lift oil wells all over the world. To ensure optimum conditions and minimize operational costs, rod pumping specialists should constantly look for ways to improve over-all operations. The present paper gives details on some of the possible actions that ensure optimum conditions. Some more important recommendations are given below.

- The pumping system’s energy efficiency depends primarily on the amount of downhole power losses. Maximum system efficiency is ensured by achieving a maximum of lifting efficiency.
- The proper selection of pumping mode can ensure maximum lifting efficiency and thus a most energy-efficient sucker-rod pumping system.
- Optimum counterbalancing of pumping units has many beneficial effects. Minimizing the **CLF** is the preferred method for finding optimum counterbalance conditions.
- Available rod string design procedures can have many pitfalls. Proper designs should provide a uniform fatigue loading of all rod taper sections.

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