POWER EFFICIENCY OF SUCKER-ROD PUMPING SYSTEMS

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INTRODUCTION

On average, two-thirds of the world's oil wells are produced by sucker rod pumping installations. Therefore, it is of utmost importance to ensure that these systems work at their peak efficiencies. Thus, calculating the energy efficiency of sucker-rod pumping is a very important task of the production engineer. To accomplish this task, one has to define the in-, and output powers of the system and the different kinds of losses occurring in the various parts of the downhole and surface equipment.

A review of the literature on the subject revealed that the useful power of the sucker rod pump is calculated by a widely accepted formula that gives inconsistent results. The formula, of which several variants are known, predicts different powers under the same conditions on the same well if the wellhead pressure is varied. Since this behavior does not allow the comparison of different scenarios.

POWER FLOW IN THE ROD PUMPING SYSTEM

The economy of a sucker-rod pumping system or any other type of artificial lift installation can best be evaluated by considering the lifting costs in monetary units per volume of liquid lifted. Since most of today's rod pumping installations are driven by electric motors, part of the operating costs is represented by the electric power bill. Because of the world-wide trend of increasing electric power prices, this single item has become the most decisive constituent of operating expenditure in sucker-rod pumped fields. Consequently, the never-ending pursuit of operating cost reduction can be translated to the reduction of energy losses both downhole and on the surface.

The power consumed by the pumping unit's motor comprises, in addition to the energy required to lift well fluids to the surface, all the energy losses occurring in the well and in the surface machinery. Therefore, any efforts to reduce these losses should start with a perfect understanding of their nature and magnitude. **Fig. 1** and the following discussion present the possible sources of energy losses along the wellstream's flow path, grouped into downhole and surface loss categories.

Input and Output Powers

The rod pumping system's useful output work is done by the downhole pump when it lifts a given amount of liquid from the pump setting depth to the surface. This work is usually described by the so-called hydraulic power, P_{hydr}, and can be calculated as the increase in potential energy of the liquid pumped. As will be detailed later, several formulae of different merit are available to calculate it, and this paper proposes a comprehensive equation that eliminates the many discrepancies previous models exhibited.

At the other end of the rod pumping system, the electric prime mover takes the required power from the surface power supply, that power being accurately measured. Since actual power requirements at the motor vary within the pumping cycle, an average input power value, P_e, valid for one pumping cycle is found from power meter readings. This power covers all requirements of the pumping system including the useful power used for fluid lifting and all energy losses occurring in the downhole and surface systems, and it represents the total energy input to the system.

Downhole Losses

The sources of downhole energy losses are the pump, the rod string, and the liquid column in the tubing string where irreversible mechanical as well as hydraulic losses take place.

Pump Losses

- Mechanical friction between the sucker-rod pump's barrel and plunger is usually unknown and can only be estimated.
- Hydraulic losses in improperly sized valves, especially when pumping highly viscous crudes, can increase downhole losses.

Losses in the Rod String

- Mechanical friction takes place wherever the rod string, while reciprocating in the tubing, rubs against the tubing inside wall, significantly increasing the energy losses in highly deviated or crooked wells or in wells experiencing rod buckling. The magnitude and severity of these frictional forces cannot accurately be determined, only estimates based on the well's inclination data can be made.

- Mechanical friction in the stuffing box is usually minimal, but extreme conditions (a dried-out or too tight stuffing box) may increase its magnitude.

Losses in the Liquid Column

- Fluid friction in the tubing-rod annulus adds to the irreversible losses because the pump action must overcome the resulting pressure differential between the pump setting depth and the wellhead. Since transient flow in an eccentric annulus is involved and the size of the annulus changes with depth in wells with tapered rod strings, accurate calculation of the frictional pressure drop, as well as the energy losses is practically impossible.
- Wellhead pressure imposes an additional power loss on the downhole pump that, by nature, cannot be considered a part of the useful work performed on the well fluid.
- Damping forces oppose the movement of the rod string because well fluids impart a viscous force on the rods' outside surface.

Surface Losses

On the surface, energy losses occur at several places from the polished rod to the prime mover's electrical connections. These can be classified according to their occurrence as mechanical losses in the drive train (pumping unit, gearbox and V-belt drive), and losses in the prime mover.

Losses in the Drive Train

- Mechanical friction in the pumping unit's structural bearings is usually very low, provided the unit is properly maintained.
- Mechanical friction in the gearbox occurs between well-lubricated gear surfaces, therefore power losses in the gearbox are usually low.
- Mechanical friction in V-belts and sheaves causes a minimal power loss if the right size sheaves with the proper number and tightness of V-belts are used.

Prime Mover Losses

- Mechanical losses due to friction occur in the structural bearings of the electric motor.
- Windage loss is consumed by the cooling air surrounding the motor's rotating parts.
- Electrical losses include iron (or core) and copper losses, of which the decisive is copper loss, resulting in the heating of the motor and is proportional to the square of the current drawn.

POWER EFFICIENCIES

If the in-, and output powers are known from actual measurements, an overall efficiency for the pumping system can easily be defined, see **Fig. 1.** Since the system's useful work is represented by the hydraulic power spent on fluid lifting, and the total energy input equals the measured electric power, the rod pumping system's energy efficiency is found from:

$$\eta_{system} = \frac{P_{hydr}}{P_e} \qquad \mathbf{1}$$

where:

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

In systems where energy losses of different nature in various system components are involved, the system's total efficiency can be broken down into individual efficiencies representing the different points in the energy flow. Total or overall system efficiency is then calculated as the product of the constituting efficiency items. In our case, one would have to assign separate efficiency figures to all or many of the individual kinds of energy losses detailed before, as was done by **Lea et al.** [1]. In this approach, it is necessary to designate efficiencies for the effects of: the rod-tubing friction, the fluid friction in tubing, etc. However, as it was discussed before, most of the individual energy losses in the pumping system are difficult or even impossible to predict, making this solution appear to be of a questionable value.

A more workable solution classifies energy losses according to their occurrence and utilizes two or three individual efficiencies for the description of the system's total energy efficiency. [2 – 4] As a natural choice, one item is assigned to describe the sum of all subsurface losses, with one or two additional items representing surface energy losses. This approach not only provides a more reliable solution for the determination of the rod pumping system's energy efficiency but allows one to identify the possible ways to increase the system's total effectiveness, as will be shown later.

LIFTING EFFICIENCY

The mechanical energy required to operate the polished rod at the surface is the sum of the useful work performed by the

pump and all the downhole energy losses detailed previously, i.e. those occurring in the sucker-rod pump, the rod string, and the fluid column. The amount of this work is directly proportional to the power required at the polished rod, the so-called **polished rod power** (PRHP), a basic pumping parameter. It represents the mechanical power exerted on the polished rod and can be found in several ways. The most reliable solution involves taking a dynamometer card and performing calculations based on the area of the card. If a dynamometer card is not available, as in the case of designing a new installation, the **API** RF' **11L** procedure [5] can be used for conventional pumping units. However, the solution of the damped wave equation provides good estimates for cases using any kind of pumping unit geometry.
Based on the above considerations, the energy efficiency of the downhole components of the pumping system is characterized by the relative amount of energy losses in the well. This parameter is called the **lifting efficiency,** η_{lift}**,** and is the quotient of the useful hydraulic power and the power required at the polished rod:

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

where:
η_{lift} = lifting efficiency, -
P_{hydr} = hydraulic power used for fluid lifting, HP
PRHP= polished rod power required at the surface, HP

The use of the lifting efficiency eliminates the need to assign individual efficiencies of mostly dubious value to each particular kind of downhole loss since it includes the effects of them all. In cases when polished rod power is obtained from a measured dynamometer card, the lifting efficiency represents the true energy effectiveness of fluid lifting in the well. If a new installation is designed, a reliable estimate of the predicted polished rod power, provided by either the **API RP 11L** procedure or by a wave analysis program can serve the same goal.
Actual values of lifting efficiency can vary in very broad ranges. At the lower end of possible values, consider the case of a worn-out pump producing a very low amount of liquid. The installation achieves a negligible hydraulic power, P_{hydr}, while consuming a definite power at the polished rod. This all adds up to a lifting efficiency value of almost nil. On the other hand, wells with big pump sizes and low pumping speeds can require little more than the hydraulic power at the polished rod under ideal conditions. **Lea et al. [** 1] gives estimates of lifting efficiencies between 95% and 70%, **Kilgore et al.** [3] presents measured values "for well designed systems" of 85% to 70%. **Gault** [6] and **Takacs** [7] point out the great impact of selecting the proper pumping mode (the combination of pump size, polished rod stroke length, pumping speed, and rod string design) on the value of lifting efficiency.

SURFACE MECHANICAL EFFICIENCY

Mechanical energy losses occurring in the drive train cover frictional losses arising in the pumping unit, in the gearbox, and in the V-belt drive. Due to their effects, the mechanical power required at the prime mover's shaft, P_{mot}, is always greater than the polished rod power, PRHP. It is customary to describe these losses by a single **mechanical efficiency** as given below:

$$\eta_{mech} = \frac{PRHP}{P_{mot}} \qquad 3$$

where:
η_{mech} = mechanical efficiency of the surface drive train, -
P_{mot} = mechanical power required at the motor shaft, HP

Average values of surface mechanical efficiency are high, usually over 90% in favorable conditions. [1, 2]. There is a consensus in the technical literature that efficiencies increase as gearbox loading increases. **Gipson and Swaim [** 8] present the correlation shown in **Fig. 2** for the estimation of the pumping unit's overall mechanical efficiency. The curves presented for new and worn units are both highly affected by the average torque load on the gearbox and efficiencies improve as gearbox loading approaches the rated capacity of the unit.

MOTOR EFFICIENCY

If the power demand on the motor shaft were steady over the pumping cycle, a motor with a power rating P_{mot}, calculated from **Eq. 3,** would be sufficient. The energy requirement of pumping, however, is always cyclic in nature because even in an ideally counterbalanced case the fluctuations in net gearbox torque cannot completely be eliminated. Thus, the mechanical load on the motor shaft is also fluctuating and the mechanical power P_{mot} only represents the average power demand over the complete cycle. Consequently, electric motors for rod pumping service must be oversized by using a derating factor that equals the so-called **cyclic load factor** (CLF). Although no industry standard exists, derating factors between 1.2 and 2.5 are used. [9 – 11]

To represent all losses in the motor, an overall efficiency factor can be used, that allows the calculation of the electric power drawn from the power supply based on the mechanical power at the motor's shaft:

$$\eta_{mot} = \frac{P_{mot}}{P_e} \qquad 4$$

where:
η_{mot} = overall efficiency of the electric motor, -
P_e = required electrical power input, HP

Although electric motors used in pumping service may have full load efficiencies close to 90% under steady loads, because of the motor's cyclic loading during pumping, actual values belong to load ranges between 30% and 80%. **Lea et al. [** 1] presents motor efficiencies of 78% - 91% for NEMA D motors of 5 HP – 60 HP sizes.

OPTIMUM SYSTEM EFFICIENCY

The rod pumping system's energy efficiency, defined in **Eq. 1,** can now be written in terms of the individual efficiencies discussed earlier as follows:

$$\eta_{system} = \eta_{lift}\ \eta_{mech}\ \eta_{mot} \qquad 5$$

where:
η_{system} = overall efficiency of the pumping system, -
η_{lift} = lifting efficiency, -
η_{mech} = mechanical efficiency of the surface drive train, -
η_{mot} = overall efficiency of the electric motor, -

An investigation of this formula allows one to draw important conclusions on the possible ways of attaining maximum energy efficiencies in rod pumping. To do so, the relative importance and the usual parameter ranges of the individual terms must be analyzed. Of the three parameters figuring in the equation, the possible values of the surface mechanical efficiency, η_{mech}, and the motor efficiency, η_{mot}, vary in quite narrow ranges. At the same time, their values can be maximized if the right size of gearbox and electric motor are selected. As shown before, a properly maintained pumping unit with a gearbox operated near its torque capacity ensures mechanical efficiencies greater than η_{mech} = 90%. **A** properly selected electric motor can also provide relatively high η_{mot} values. Thus the combined efficiency of the drive train and the motor can lie in the range of 70% - 82%, as given by **Lea et al.** [1]

In contrast to the usual ranges of the above efficiencies, lifting efficiency, η_{lift}, can vary in very broad ranges depending on the pumping mode (plunger size, stroke length, and pumping speed) selected. For example, **Takacs [** 7] reports lifting efficiencies between 94% and 38% when producing 500 bpd from 6,000 ft with different pumping modes. As supported by **Gault [** 6], considerable improvements on lifting efficiencies can be realized by selecting the optimum pumping mode.

In summary, the basic requirement for achieving high overall system efficiency is finding the maximum possible value of the lifting efficiency. Since this is accomplished by the proper selection of the pumping mode, the choice of the right combination of pump size, polished rod stroke length, and pumping speed is of prime importance. 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.

HYDRAULIC POWER CALCULATIONS

Introduction

In general, the power required to move fluids through a pump is found from the volumetric rate of the fluid pumped and the pressure increase developed by the pump. Using oil field units, the following equation can be developed:

$$\boldsymbol{P} = 1.7 \times 10^{-5}\ \boldsymbol{Q}\ \Delta \boldsymbol{p} \qquad 6$$

where:

P = power requirement, HP

Q = volumetric pumping rate, bpd

Ap = pressure increase through pump, psi

When applying this formula to sucker rod pumps, it is customary to use the liquid rate, Q, actually produced and measured at the surface. In this fashion, the pump's volumetric losses are automatically accounted for, since the measured rate includes the effects of the following volumetric losses along the flow path:

- Improper pump fillage due to gas interference or insufficient inflow from the well.
- Leakage losses in the barrel-plunger clearance as well as in the pump's valves due to mechanical wear.
- Leakage in tubing and flowline decreases the liquid amount produced by the pump.

All of the above effects tend to decrease the pump's useful output, therefore the energy losses associated with them must not be considered as part of the pump's useful work, and must be viewed as wasted power components.

It should be noted here that all subsequent calculations in this paper assume that incompressible liquids are produced by the sucker-rod pump. The presence of considerable amounts of free gas in the tubing and the annulus can greatly modify the pressure conditions and power calculations must be modified accordingly.

Previous Models

As pointed out by **Lea and Minissale** [12], the technical literature quite consistently assumes that the sucker-rod pump's useful pressure increase, Dp, equals the hydrostatic pressure calculated from the "net lift", the depth of the dynamic liquid level measured from the surface in the well's annulus, and the specific gravity, SpGr, of the produced liquid, valid in the tubing string. Thus, **Eq. 6** takes the form:

$$P_{hydr} = 7.36 \times 10^{-6}\ Q\ SpGr\ L_{dyn} \qquad 7$$

where:

P_{hydr} = hydraulic power used for lifting, HP

Q = liquid production rate, bpd

SpGr = specific gravity of the produced liquid, -

L_{dyn} = dynamic liquid level in the well's annulus, ft

An analysis of this formula provides an explanation of the problem discussed by **Lea et al.** [1] who revealed that hydraulic powers calculated from the above formula give greater values for increased wellhead pressures while pumping the same liquid rate. For this purpose, let us express the pump's intake pressure from the pressures valid in the well's annulus (see **Fig. 3):**

$$PIP = p_{wh} + \Delta p_g + 0.433\ SpGr_a \left(L_{pump} - L_{dyn}\right) \qquad 8$$

Solving for L_{dyn} and substituting it into **Eq. 7** we get:

$$P_{hydr} = 1.7 \times 10^{-5}\ Q \left[0.433\ SpGr_t\ L_{pump} - PIP \frac{SpGr_t}{SpGr_a} + \left(p_{wh} + \Delta p_g\right)\frac{SpGr_t}{SpGr_a}\right] \qquad 9$$

where:

L_{pump} = pump setting depth, ft

PIP = the pump's suction pressure, called pump intake pressure, psi

p_{wh} = wellhead pressure, psi

Δp_g = static gas pressure increase in the annulus, psi

The power calculated from the above formula, as can be seen, includes the power required to overcome the surface wellhead pressure and, therefore, increases with an increase in wellhead pressure. Calculations performed on the same well producing the same liquid rate at different wellhead pressures will obviously give different values of hydraulic power. This results in different system efficiencies under the same conditions as well as prevents the comparison of two different pumping well's overall efficiency, making this formula of dubious value. In order to properly compare different pumping conditions, a standardized formula for hydraulic power and overall system efficiency calculations is desirable.

Proposed Model

Eq. 9 can easily be modified to exclude the effects of wellhead and gas column pressures by eliminating the appropriate term when the following equation is found:

$$P_{hydr} = 1.7 \times 10^{-5}\ Q \left[0.433\ SpGr_t\ L_{pump} - PIP \frac{SpGr_t}{SpGr_a} \right] \qquad 10$$

This can be reduced to:

$$P_{hydr} = 7.36 \times 10^{-6}\ Q\ SpGr_t \left[L_{pump} - \frac{PIP}{0.433\ SpGr_a} \right] \qquad 11$$

It can be shown that the above formula represents the **absolute minimum power** required to lift the given amount of liquid from the well and can thus be used as a standard reference in comparing sucker-rod pumping system efficiencies. The pressure distribution in a sucker-rod pumped well is shown in **Fig. 3.** The pump's discharge pressure, p_d, is greater than the liquid's hydrostatic pressure because it must overcome the wellhead pressure plus all possible hydraulic losses arising in the tubing string. The pressure and energy losses occurring in the annulus between the tubing and the rod string, as discussed before, cannot readily be calculated and an estimation of their magnitude is possible only through the evaluation of the dynamometer card. However, since the energy used against the wellhead pressure and the hydraulic losses is considered wasted, they must not be included in the calculation of useful power. Since the pump's suction pressure is positive and equals the pump intake pressure, PIP, the useful pressure increase developed by the pump equals Dp_u.

The theoretical minimum power should exclude all hydraulic losses in the tubing – rod string annulus as well as the effect of the surface backpressure and will be found by assuming a zero wellhead pressure. In this case the net lift is modified and is found by projecting the line representing the pressure distribution in the annulus. Net lift will be equal to L_{pump} – PIP / (0.433 $SpGr_a$) and the basic equation for power calculations **(Eq. 7)** can be used to arrive at the above expression which is very similar to that of **Lea et al.** [1]
It can be easily observed that, in contrary to previous models, **Eq. 11** excludes the power wasted for overcoming the wellhead pressure and all hydraulic losses occurring in the well. Therefore, it represents the possible minimum power required to lift well fluids to the surface. Since its value is constant as long as the pump intake pressure is constant, it provides a standard way to compare the energy efficiency of the same system under different conditions or the efficiencies of different pumping systems. Because of its beneficial features, the general application of this equation for calculating the power efficiency of sucker-rod pumping systems is recommended.

EXAMPLE PROBLEMS

Hydraulic Power Calculations

The pump is set at 6,000 ft in a 6,500 ft deep well, the measured liquid rate is 500 bpd with a water-oil ratio of WOR = 3. At a wellhead pressure of 200 psi the dynamic liquid level was found at 4,500 ft, oil and water specific gravities are 0.85 and 1.03, respectively. Find the system's hydraulic power at the original wellhead pressure and at 400 psi, if the system's lifting capacity is not altered. Static gas column pressure calculations resulted in gradients of 5 psi/1,000 ft and 6 psi/ 1,000 ft for the two cases.
First, find the liquid specific gravities in the annulus and tubing:
$SpGr_a$ = 0.85, since the annulus contains oil above the pump, due to gravitational separation.
$SpGr_t = SpGr_o / (1 + WOR) + SpGr_w\ WOR / (1 + WOR) = 0.85 / (1 + 3) + 1.03\ 3 / (1 + 3) = 0.985$
Now the pump's intake pressure, PIP, at 200 psi wellhead pressure can be found from **Eq. 8:**
PIP = 200 + 5 4,500 / 1,000 + 0.433 0.85 (6,000 – 4,500) = 774.6 psi
Because the pump operates with the same settings, the flowing bottomhole pressure, consequently the PIP does not change for the other wellhead pressure of 400 psi. PIP being fixed, the new dynamic liquid level is found from **Eq. 8,** after expressing Ap_g with the gas gradient:

$$L_{dyn2} = \frac{p_{wh} + 0.433\ SpGr_a\ L_{pump} - PIP}{0.433\ SpGr_a - grad_g} = \frac{400 + 0.433\ 0.85\ 6{,}000 - 774.5}{0.433\ 0.85 - 0.006} = 5{,}065\ \mathbf{ft}$$

The two liquid levels known, the hydraulic powers according to the conventional formula can be found from **Eq. 7:**

$P_{hydr1} = 7.36E\text{-}6\ 500\ 0.985\ 4{,}500 = 16.3$ HP, and
$P_{hydr2} = 7.36E\text{-}6\ 500\ 0.985\ 5{,}065 = 18.4$ HP.
The proposed expression yields a single value for both cases, as calculated from **Eq. 11:**
$P_{hydr3} = 7.36E\text{-}6\ 500\ 0.985\ [\,6{,}000 - 774.6 / (0.433\ 0.85)] = 14.1$ HP.
By comparison, the formula recommended by **Lea et al. [** 1] resulted in 15.2 HP.
As seen, the old model estimated an increase in hydraulic power for the greater wellhead pressure and therefore, in contrary to the proposed model, cannot be used for comparisons. **Fig. 3** illustrates the pressure distributions in the well at a wellhead pressure of 200 psi. The flowing tubing pressure, as shown, is only an estimate of the probable pressure losses occurring in the tubing-rod annulus.

System Efficiency Calculations

A 1 ¼” pump is set at 4,329 ft and the dynamic liquid level is at 1,449 ft. With a polished rod stroke length of 99.8” and a pumping speed of 9.4 SPM the pump produces 170 bpd of liquid. Oil and water specific gravities are 0.82 and 1.02, respectively, and the well produces with WOR = 0.8. Gas production is negligible and the surface pressure is 30 psi. Under these conditions the average motor power was measured as 6 kW, and the polished rod power was found from the dynamometer diagram as 4.9 HP.
Liquid specific gravities in the annulus and the tubing:
$SpGr_a = 0.82$, and
$SpGr = SpGr_o / (1 + WOR) + SpGr_w\ WOR / (1 + WOR) = 0.82 / (1 + 0.8) + 1.02\ 0.8 / (1 + 0.8)$
$= 0.9’1$
Now the pump intake pressure is found from **Eq. 8,** by assuming a static gas gradient of 0.002 psi/ft:
$PIP = 30 + 0.002\ 1449 + 0.433\ 0.82\ (4{,}329 - 1{,}449) = 1{,}055$ psi
Hydraulic power is calculated from **Eq. 11:**
$P_{hydr} = 7.36E\text{-}6\ 170\ 0.91\ [\,4{,}329 - 1{,}160 / (0.433\ 0.82)] = 1.55$ HP.
The system’s overall power efficiency can now be found from **Eq. 1:**
$\eta_{system} = 1.55 / (6 / 0.746) = 0.193 = 19.3\%$
Also, lifting efficiency can be calculated from **Eq. 2:**
$\eta_{lift} = 1.55 / 4.9 = 0.316 = 31.6\%$
After optimizing the well’s operation, the optimum pumping mode was found [4] with the following parameters: 2 ¼” pump, a stroke length of 82.2”, and a pumping speed of 4 SPM. Polished rod power was calculated as 2 HP, and the lifting efficiency is found as:
$\eta_{lift} = 1.55 / 2 = 0.775 = 77.5\%$
If mechanical and motor efficiencies are identical to the original case, the system efficiency for the optimized case can be estimated. From previous data and **Eq. 5:**
$\eta_{mech}\ \eta_{mot} = 0.193 / 0.316 = 0.611 = 61.1\%$
The system efficiency for the optimum pumping mode is:
$\eta_{system} = 0.775\ 0.611 = 0.473 = 47.3\%.$
As seen, the use of the optimum pumping mode has considerably increased the pumping system’s efficiency.

CONCLUSIONS

The paper investigates the power conditions of sucker-rod pumping installations and draws the following conclusions.

1. The overall energy efficiency of a sucker-rod system is best described by a three-term formula that includes the efficiencies of:
   - the downhole system,
   - the surface mechanical parts, and
   - the electrical prime mover.
2. The most important constituent of the system’s total energy efficiency is the lifting efficiency describing all energy losses in the well.
3. Maximum system efficiency is achieved by the proper selection of the pumping mode, i.e. the proper combination of pump size, polished rod stroke length, and pumping speed.
4. Since the formula most often used for the calculation of the pump’s useful power gives inconsistent results, a new formula is proposed that represents the minimum power requirement for lifting the given amount of liquid to the surface.

REFERENCES

1. **Lea, J. F. – Rowlan, L. – McCoy, J.:** "Artificial Lift Power Efficiency." Proc. 46th Annual Southwestern Petroleum Short Course, Lubbock, Texas, 1999, 52-63.
2. **Butlin, D. M.:** "A Comparison of Beam and Submersible Pumps in Small Cased Wells." Paper SPE 21692 presented at the Production Operations Symposium, Oklahoma City, Oklahoma, April 7-9, 1991.
3. **Kilgore,** J. J. **– Tripp, H. A. – Hunt, C. L.:** "Walking Beam Pumping Unit System Efficiency Measurements." Paper SPE 22788 presented at the 66th Annual Technical Conference and Exhibition of SPE, Dallas, Texas, October 6-9, 1991.
4. **Takacs, G.:** "Modern Sucker-Rod Pumping." PennWell Books, Tulsa, Oklahoma, 1993.
5. "Recommended Practice for Design Calculations for Sucker-Rod Pumping Systems (Conventional Units)." API RP 11L 4th Ed. American Petroleum Institute, Washington, D.C., 1988.
6. **Gault, R. H.:** "Designing a Sucker-Rod Pumping System for Maximum Efficiency." SPE Production Engineering, November 1987,284-90.
7. **Takacs, G.:** "Program Optimizes Sucker-Rod Pumping Mode." Oil and Gas Journal, October 1, 1990, 84-90.
8. **Gipson, F. W. – Swaim, H.W.:** "The Beam Pumping Design Chain." Proc. 31st Annual Southwestern Petroleum Short Course, Lubbock, Texas, 1984, 296-376.
9. **Durham, M. O. – Lockerd, C. R.:** "Beam Pump Motors: The Effect of Cyclic Loading on Optimal Sizing." Paper SPE 18186 presented at the 63rd Annual Technical Conference and Exhibition of SPE, Houston, Texas, October 2-5, 1988.
10. **Durham, M. O. – Lockerd, C. R.:** "Pumping Unit Effect on Motor Efficiency." Proc. 36th Annual Southwestern Petroleum Short Course, Lubbock, Texas, 1989, 308-20.
11. **Clegg, J. D.:** "Rod Pumping Selection and Design." Proc. 38th Annual Southwestern Petroleum Short Course, Lubbock, Texas, 1991,274-88.
12. **Lea, J. F. – Minissale, J. D.:** "Beam Pumps Surpass ESP Efficiency." Oil and Gas Journal, May 18, 1992, 72-5.

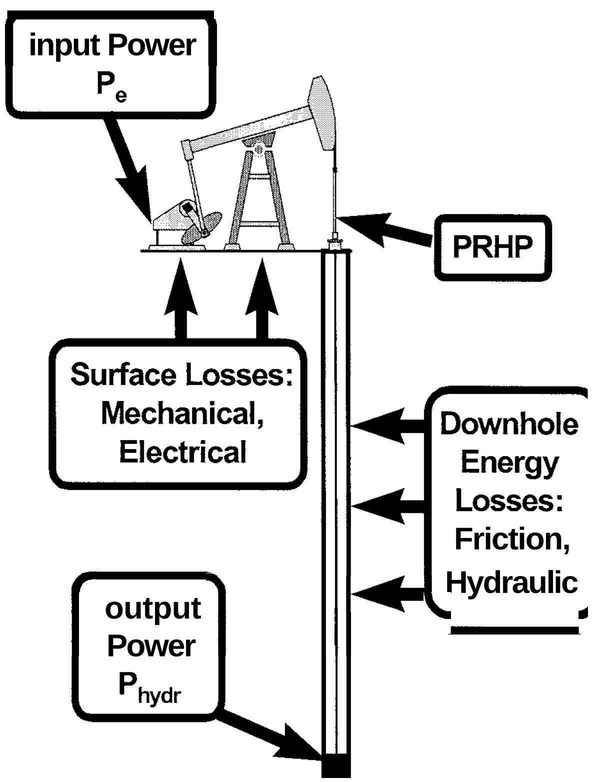

Figure 1 - Energy Flow in the Sucker-rod Pumping System

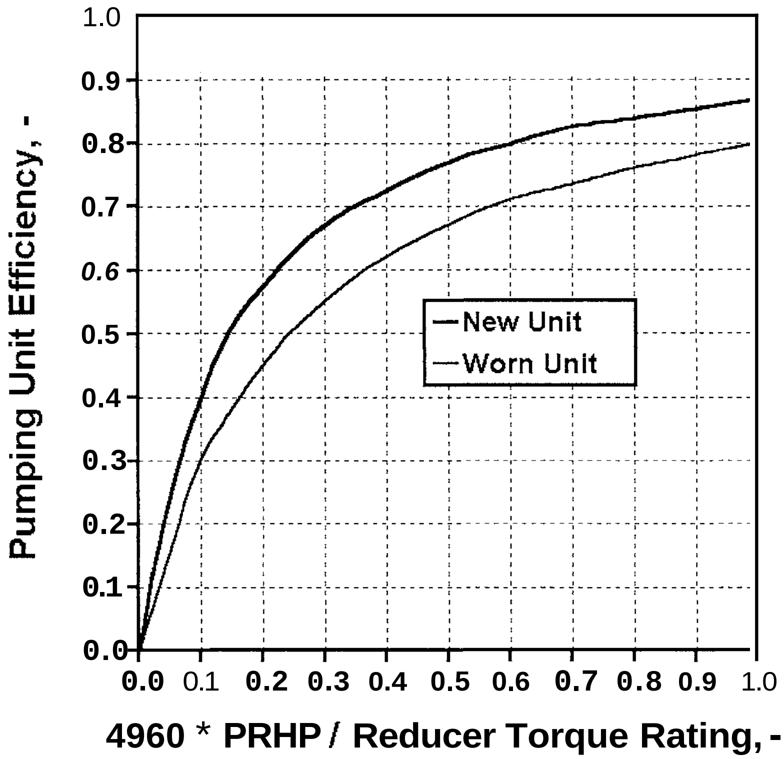

Figure 2- Overall Mechanical Efficiency of Pumping Units

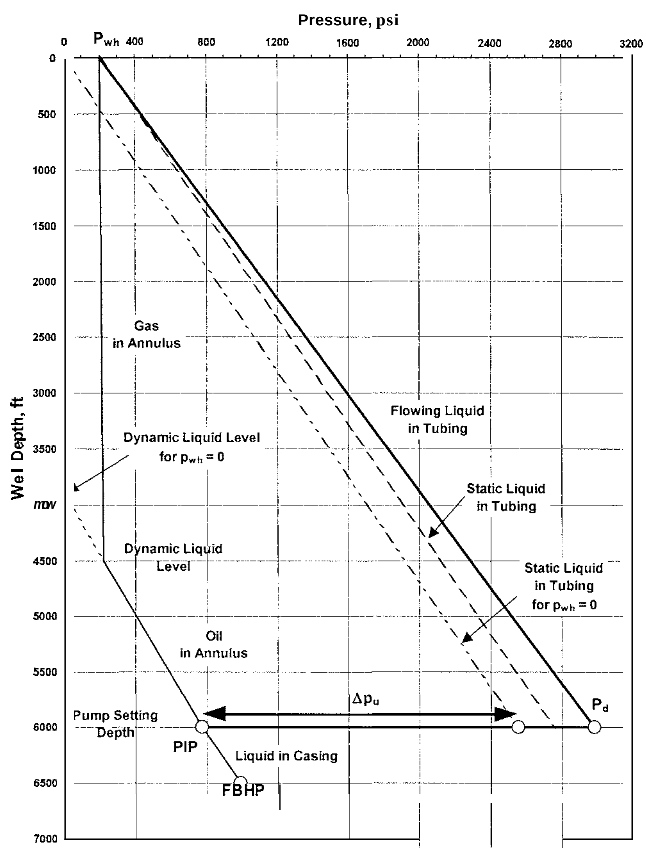

Figure 3 - Pressure Distribution in a Sucker-rod Pumped Well