ROD PUMPING SELECTION AND DESIGN

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INTRODUCTION

The selection and design of the artificial lift equipment will continue to be a significant event in the life of most oil wells. The resulting profitability will either be increased or diminished by the artificial lift equipment choices, and the credibility of the designer will be enhanced or lowered. The designer should be aware that the system efficiency is becoming increasingly more important, since energy costs continue to rise.

The best artificial lift selection and design will make the greatest amount of money (highest present value cash flow) over the life of the project. [2] In the planning stage the following three factors must be evaluated:

(1). the oil and gas production and revenue over well life,
(2). the operating cost over well life, and
(3). the capital cost of the equipment.

Of these --the most significant is the revenue. To maximize revenue, the oil reserves should be produced in a timely fashion. Thus a design is required that will produce the well near its capacity or at a high rate throughout its life.

The operating cost over life normally far exceeds the capital cost; thus, particular attention should be given to reducing recurring monthly costs (i.e. energy, maintenance, and repairs). Often the more efficient and trouble free equipment is initially more expensive. Good records must be kept and evaluated to determine the operating cost advantage of the installed equipment. Digging out the monthly costs may be a tedious process even in these days of data bases and computer manipulations.

The capital cost is easy to obtain. The vendors and service companies will help in this endeavor. Bids are often obtained and, unfortunately, sometimes not carefully reviewed. The bid specifications should be carefully written and the received bid then thoroughly analyzed. Do not waste money by purchasing excessively large equipment or buying features that are not needed. Be sure all equipment will meet the objectives of being efficient and trouble free. Do not overlook the service required and its availability and quality in the planned location.

This paper will discuss the more important considerations in rod pumping selection and design.

GENERAL CONSIDERATIONS

To make the proper artificial lift selection and the best design, various information is needed. Often the environmental and geographical considerations [10] are critical and may be the dominant factor in the decision process. One of the first steps is a good prediction of pressure-volume-temperature producing conditions over life. As pointed out by Neely and others [1], [2], [5], [8], the designer should match lift capabilities with well productivity. Paraffin, sand, gor, water cut, corrosion, casing size, crooked holes, lift depth, and temperature are important factors to consider. Also the designer should evaluate the simplicity, flexibility, and adaptability of the lift equipment. Both short and long term aspects must be reviewed.

ROD PUMPING

Rod pumping is generally recommended for artificial lift selection --- if it will produce the well near its maximum rate over life without excessive loads and ---if there is not some obviously bad operating problem. Examples of such problems are (a) excessive sand production, (b) extremely high gor's, (c) very crooked holes, (d) heavy paraffin, or (e) severe scale and corrosion. Environmental consideration may exclude rod pumping. Keep in mind that over 85 percent of the wells in the USA on artificial lift have rod pumping equipment---There must be some good reasons for its use.

Rod pumps have, in general, the highest system efficiency of all lift methods. They also are: able to produce at very low pump intake pressures, simple to diagnose, reliable, flexible, and able to tolerate high temperatures. Their initial cost may be more than other lift methods but rod pumping equipment has a long reliable operating life and a high salvage value. Typically, --- rod pumping systems earn you the most money.

Wells in West Texas and New Mexico are excellent candidates for rod pumping. The production rates are typically low and the lift depths are not excessive. Most wells are relatively straight (not crooked with severe dog-legs) and sand production is minimal as compared to wells in the Gulf of Mexico. Most operating problems such as corrosion, paraffin, scale, high glr's can be tolerated -- but cause increased lifting costs. The design problem for such wells is "what size and type rod pumping equipment to select."

A. Size

Size the pump displacement related equipment to produce the predicted daily production volume in about 20 hours. The slightly over-sized equipment for displacement allows for some errors in production forecasts, gas interference problems, and pump wear before pulling is required. The rate is based on well tests for the well or similar off-set wells. New fields may require using analog well data. The Vogel IPR curve gives reasonable predictions for maximum rates when good test data are available.

The big unknown comes from wells that will be water or wag CO_2 flooded. Review pumping experience for analog fields that are already under flood. Wells in depletion reservoirs will decline; therefore do not over-size equipment for such wells. Water drive field wells will often continue to have near the same total fluid rate but produce with increasing water cuts and lower glr's.

Excessively over-sized units increase capital costs and will actually result in higher operating costs due to lower system efficiency. As reported by Gipson and Swaim [8], unit gear boxes lose efficiency drastically below 50 percent load. For maximum system efficiency, operate a unit near its gear box rating.

B. Units

The conventional beam unit is the most used and is recommended as the standard choice for rod pumping. These units have proven to be reliable and can be rotated in either direction -- clockwise or counterclockwise -- which is possibly some benefit. However, every unit has a preferred direction of rotation and this direction should be selected to optimize the performance.

Other type units (such as the Mark II) must be justified by better or more efficient performance. Estimate the capital costs for the conventional and other possible type units. Check the predicted loads and polished rod horsepower using a good wave equation program. [7] Where there are significant energy and pulling cost savings, air balance or special-geometry units can be recommended.

For any unit, the production and resulting loads can be easily calculated using the method outlined in API RP 11l. More precise answers will result from use of a good wave equation design program. Typical maximum production rates vs. depths for a 320D-256-100 conventional unit are shown in Figure 1. At very shallow depths (< 1500 feet) the limiting factor is often the largest pump plunger size that can be practically used and the running speed. Peak polished rod loads (pprl), gear box rating and unit structure rating are not the controlling factors. Minimum polished rod loads (mprl) should not be less than zero. In fact, for design purposes, the mprl should be no less than 10 percent of the rod weight in air. Often for shallow wells where rod stress is not critical, use all 1-inch rods to help the rods fall and increase minimum loads.

Further investigation of the 320D-256-100 conventional unit shows that in the 2000 to 4000 foot depth range the gear box is the weak link. In the 5000 to 8000 foot range the pprl is often the controlling parameter, and use of higher strength rods (API "D") becomes advantageous. Once depths reach 9000 feet, the 25,600 pound structure rating limits production. Overloading rods, structure or gear box is not recommended since prolonged high load conditions will appreciably reduce equipment life.

Air balanced units normally cost more, are complicated, and more expensive to operate---especially with age. They have the advantage of being lighter, more compact, and easier to counter-balance. Keeping the units in good counter-balance should theoretically give longer gear box life but has only a minimal effect on power cost. Air balance units make excellent test units and can be used on offshore platforms where size and weight considerations are more important. These units come in large sizes that may be of some advantage in deep high volume applications. For shallow to medium wells with moderate lift volumes, there is little if any advantage to using air balance units.

Mark II units, which have a special geometry and an off-set crank, have been accepted as a proven design to reduce gear box loads. These units have been claimed to do more work using less energy than when using conventional units due to a higher system efficiency. Development of non-dimensional curves [3]--- similar to those in API RP 11l ---made it possible to compare the Mark II unit with conventional and special geometry units over a wide range. These comparisons indicated that peak gearbox torque was indeed reduced but peak polished rod load (pprl) was about the same as a conventional unit. The design of the unit causes an accelerated down stroke which reduces the mprl. This causes greater load ranges and limits the speed of the unit. Experience has shown that it is especially bad to pound fluid with a Mark II due to the faster down stroke. The Mark II's biggest advantage is for medium to deep wells with high gear box loads when running at moderate speeds. Often a size smaller gear box can be used with Mark II than needed for a conventional unit. System efficiency seems to vary and is normally close to that of a conventional unit. Further tests are required to define the possible benefits.

Special geometry units (e.g., Lufkin RM and American Producer II) result in lower gear box loads. The mprl's and the pprl's are comparable to conventional units; however, the mprl's are not as low as the Mark II. The special geometry units are designed to run in only one direction and have an improved linkage arrangement which gives lower torques and slightly lower prhp than conventional units. These units cost up to ten percent more for the equivalent structure conventional unit. Thus, if a size smaller gear box results for the special geometry, the unit will often cost less than the conventional unit. Use the wave equation program to determine the possible reductions in loads.

C. Motors

NEMA D motors (normally 440 volts and 1200 rpm) are designed to slip 5 to 8 percent under full load conditions and are recommended for most rod pumped wells--where electric power is available. For such a motor the starting torque must not exceed the locked-rotor torque - which is 2.8 times the full load torque. "High slip" motors are not as efficient. If run in the low torque mode, ultra high slip motors will reduce peak rod and gearbox loads. However, in the low-torque mode the system efficiency may be 10 percent lower than the high torque mode. The designer must decide if the slightly reduced loads will offset the extra power cost.

Motors can be sized using the rule-of-thumb of twice the polished rod horsepower--and selecting the closest available motor hp standard size on the high side. Work by Tripp, Padden and Kilgore and others found that motors should be sized closely to the actual required hp. This practice results in the best efficiency. Typically the motor shaft hp should be between 50 and 70 percent of the rated hp. If a 40 percent or lower load results, the motor efficiency will fall off significantly. In such cases, the motor should be changed for a smaller size. The rms amperage should not exceed the rated motor amperage. Some additional hp value is needed for poor counter-balacing practices; however the NEMA D design permits the motor to operate overloaded up to 280 percent for brief time periods. The best design approach is to use actual field measurements. Remember--installing too big a motor increases the initial cost and the power bill.

D. Sheaves and Belts

System efficiency can be improved by selecting the proper sheaves and belts. Big unit sheaves permit running the unit at the needed slow speed. Design for as much speed variation as practical. In the 456 and larger gearbox sizes, select "C" size grooves rather than "D" since slower speeds are possible using the "C" V-belt without causing belt damage. Use the required number of belts as outlined by Gipson and Swaim [8].

It has been reported that standard V-belts have a maximum efficiency of 94 percent. The efficiency will be less if the V-belts are not properly tightened and when smaller driver sheaves are used. (Thus the need for large unit sheaves). Use of the high efficiency V-belts (Cog belt) are recommended and they can be used with standard sheaves.

E. Rods

Rod selection and design is extremely important since this choice often has a great effect on operating performance and efficiency. Normally steel rods in the API grade of C, D, or K are selected. Special high strength steel rods are available and the fiber glass rods are becoming more widely used. Corods are popular in Canada.

The designer should normally use the Modified Goodman Diagram for picking maximum stress levels for steel API rods to provide for long life. Service factors (SF) are a method of derating rods due to the environment. The designer should not use SF's over 1.0. The need to use SF under .8 indicates severe corrosion or poor operating conditions. Use of lower SF's result in heavier rod strings and may reduce system efficiency. In such cases the rod design and handling practices plus the inhibition program should be reviewed.

Typically rod stress is not an important factor in shallow wells and grade C rods can be used. Normally stresses of less than 25,000 psi are suitable for API grade C rods. Once stresses reach 30,000 psi most operators use API D rods if there is less than .05 psi partial pressure H2S in the produced fluids. API D rods are susceptible to stress corrosion cracking and must be inhibited when used in sour service. For stresses beyond the range of API D rods, specialty rods are available and are rated up to 50,000 psi stress levels. Good handling practices and corrosion inhibition are highly recommended for all type rods and especially for the higher strength rods. Some operators prefer API K rods in lieu of API C rods in corrosive fields. A good choice must be made; otherwise, frequent rod parts will result. In general, rods parts should be less than two per year per well.

The API rod size selection is often restricted by tubing size. Good operating practices require that the rods and their associated couplings permit fishing in the available tubing size. Tapered strings result in lower stresses and are recommended for medium to deep wells. Both two and three tapers should be evaluated but four tapers are of questionable benefit. The use of API 64 or 75 rod strings is risky since a high rod part probably will result in cork-screwing the smaller rods. For 2 3/8 inch OD tubing the API 76 rod string and for 2 7/8 inch OD tubing the API 86 rod strings are often the best tapered choice. (Note: slim hole 7/8-inch rods couplings may require some derating for high load applications.)

Fiber glass rods are lighter and have a higher modulus of elasticity which results in more stretch. Increased production is possible with fiber glass rods in cases where the unit structure load is close to rated values when using steel rods. Overloading the structure significantly shortens bearing life--and this can lead to unit self destruction. Use of the wave equation is recommended for design of fiber glass strings. Generally a relatively high speed is required to increase production. Caution should be used when considering fiber glass rods in crooked holes, high glr wells or under pumped-off conditions because fiber glass rods have very little lateral or compressive strength.

Tripp [15] suggests that a 50 percent steel on bottom and 50 percent fiber glass on top should give good results. The use of all fiber glass strings is not recommended since fiber glass rods will not tolerate compression loads. Fiber glass rods are more expensive than steel rods and have a short service life based on load range. (Steel rods will theoretically run forever in a non-corrosive environment if stresses are kept below the endurance limit.) The question the designer of fiber glass strings must answer is whether the life will be long enough to pay out the extra initial costs.

Steel sucker rods on bottom are mandatory when using fiber glass strings. Whether the use of large rods or sinker bars just above the down-hole pump helps in steel rod strings is debatable. (Neely vs. Gipson). A few large sucker rods will not hurt. An examination of wear in the lower rods and a visit to the pump repair shops to examine pull rods and guides should help the designer in making the decision.

There are several other considerations to be reviewed in the selection of the rod string. Couplings often result in 50 percent of the failure; thus, their selection can be critical. Sprayed metal couplings (with a polished surface) are recommended where box wear problems occur. Paraffin scrapers and rod guides should be avoided unless absolutely required. Rod guides may be essential in crooked holes.

F. Pumps

Rod insert type pumps --that are as large as practical in size-- are recommended for most wells. Gault suggested using large pumps with big units to achieve higher system efficiencies. Size the pump to get the production with the given unit stroke and planned spm in about 20 hours. Table "A" lists displacements needed to produce 300 bpd from 5000 feet on a well equipped with a conventional 320D-256-100 unit and 2 7/8-inch OD tubing.

Size for a displacement of 360 bpd. Note that use of small pumps results in a higher prhp--generally running at a high spm. Three different stroke lengths were available and pump plunger sizes from 1.25 to 2.0 were evaluated. The obvious displacement choice for this unit would be the 2.0 pump, 85-inch stroke, running 10.8 spm with a prhp of 14. In this case the rods, gear box and structure are not overloaded. (See Table A).

Insert type pumps can be more easily pulled and repaired than tubing pumps; however, the tubing pump may be more efficient. Thus the designer must evaluate the benefits for the local conditions. Simple bottom hold-down pumps are recommended for moderate to deep wells [13]. Shallow wells with trash problems may find the top hold-down better. Bottom hold-down pumps can use top seals if trash is a problem. Single balls and seats are generally recommended. Close spacing is mandatory for high glr wells. The use of 60 inch plungers is adequate if the fit can be kept tight--say -.003 inches. See Figure 2. Avoid buying and running worn out pumps. Also be careful in selecting special pump features such as bottom hole discharge valves, ring valves, double displacement pumps, and three-tube pumps unless good field data support their use for unusual downhole conditions.

The best pump materials are often found through trial and error by the pump repair service company. In general, buy "premium" parts so that pump life will exceed two years. The designer is missing a great opportunity to reduce pump repairs if detailed records are not kept on each well. Incentive bids should be considered to get the pump repair service companies on your side. The API rod pump teardown report should be filled out and computer data base started. Good statistical data are needed to make good decisions--not just the recent failures that you remember.

Total lifting costs (including run life) should be evaluated not just the pump repair costs. Quarterly or yearly meeting should be arranged between the repair service company and the operator to discuss the data and ways to improve operations. If time permits, frequent trips to the repair shop are needed to define a policy on when to replace parts. Improving pump repair costs performance is an effort that probably takes at least two years to be meaningful and show results.

G. Gas Anchors

Gas interference can reduce system efficiencies drastically [4], [12]. Free gas should be separated downhole and vented up the annulus and liquid with minimal gas actually pumped. Natural gas anchors -- where the pump intake is below the producing interval -- should be utilized whenever possible. A key to using natural gas anchors is to install large casing and provide the well with a 30 foot sump below the perforations. Table B is a suggested guideline for selecting casing size where high gor's are anticipated and the natural gas anchor is planned.

In gassy wells that must be pumped from above the perforations, the packer anchor is recommended. Its performance is similar to the natural gas anchor. Careful selection of the packer type gas anchor should be made to avoid operating problems. In unusually severe gas interference situations, the specialty anchors should be evaluated. In low volume (<100 bfpd) wells, a modified improved poor-boy anchor may be satisfactory [4].

H. Tubing and Tubing Anchors

Tubing size is often limited by the casing size. Thus some key decisions should be made on the size casing and the additional cost of larger casing justified before the drilling of the well. If 4 1/2-inch casing is selected, the common practice is to use only 2 3/8-inch OD tubing. If the selection causes serious pumping problems, then 2 7/8-inch tubing with small OD connections should be considered. In many fields, the sticking and fishing of tubing is rare. A few joints of integral joint tubing on bottom may solve all realistic operating problems since near bottom is where most sticking and fishing problems occur. The same philosophy can be applied to using 3 1/2-inch tubing inside 5 1/2-inch casing. Experience indicates that larger tubing reduces drag and in turn prhp.

API EUE tubing is recommended for most wells. Tapered tubing strings can be used to accommodate tapered rod stings. Internal plastic coated tubing is seldom justified as the coating often is knocked or worn off.

Tension tubing anchors are recommended for all but shallow wells with small plungers. Tubing stretch can easily reduce system efficiency 10 percent. The cost of the anchors and the infrequent repair problems can be quickly paid out by the lower power bills.

I. Pumpoff Control

Sizing the displacement to fit exactly the wells actual production is virtually impossible. Thus pumpoff controllers can be easily justified. Neely and Tolbert [11] listed the benefits of pumpoff controlers to Shell Western E&P in the Permian Basin. Their use resulted in an average 20 percent reduction in energy, a 25 percent reduction in pulling jobs and a 1 to 4 percent increase in production. The analog and local logic microprocessor types are recommended. In most large fields, central communication to host computer (via radio) is beneficial and should be evaluated.

J. Pump Intake Pressure

The selected artificial lift system should be able to produce the well near its capacity and have low operating costs. Trying to get the very last barrel of oil may prove to be expensive. Rod pumping has the capability to pump at very low intake pressures--possibly 10 psig. Keeping pump intake pressures (pip) to less than 100 psig with rod pumping is accomplished in many fields. A realistic goal is to plan for a minimum of 50 psig pip and keeping the pump fillage to at least 75 percent by using pumpoff controllers. These goals will still permit a high system efficiency.

Resorting to vacuum pumps on the annulus to reduce casing pressures and producing bottom hole pressures can only be justified in very high PI wells with very low reservoir pressures. Vacuum pumps create a number of corrosion and explosion problems that the designer needs to avoid.

SYSTEM EFFICIENCY

As previously stated, the rod pumping system efficiency is the highest of any conventional artificial lift method. The system efficiency in rod pumping can be significantly improved by the proper choices. Just what system efficiencies and power bills are obtained in a typical installation? This problem has been addressed by Gault [6] and by Lea and Wilson [9] in recent papers. (See Appendix "A" - Nomenclature).

Field data were obtained from the Wasson Field Denver Unit. These 5000 foot plus wells are equipped with both rod and ESP artificial lift. Mostly conventional beam units of 74 to 168 inch strokes are used and ESP with pump capacities of 400 to 1000 bpd pumps are employed. Most of the previous recommendations are followed for rod pumping. A number of the installations had separate electrical meters that permitted obtaining monthly power bills.

The hydraulic horsepower (hhp) can be easily calculated. (See Appendix A). Based on well tests and assuming a PIP of 100 psig, the theoretical monthly power costs were generated for system efficiencies of 100 to 10 percent. (See Figure 3.)

The polished rod horsepower (prhp) was calculated using a program similar to API RP 111. The expected input electrical horsepower was calculated using the assumptions of Gault (6) and using a power cost of 4.5 cents per kw-hr. Actual power costs were also plotted vs. rate. (See Figure 4). The hhp represents 100 percent system efficiency. Wells with such a high efficiency have questionable data or might be flowing/flumping. The assumption used by Gault (1.67*prhp) results in about a 50 percent system efficiency. Note that the actual data at best has a 50 percent system efficiency. Most of the rod pumped wells power bills fall between 50 and 25 percent system efficiency. Thus in actual practice, poorer system efficiency performance often occurs. More realistic power bill estimates result if twice the prhp is used. This results in a system efficiency of 40 percent for rod pumping. The suggested value of 1.35*prhp by Lea and Wilson seems low for the reviewed cases.

For comparison purposes, the ESP's in the same field were also reviewed. PIP's of 250 psig were assumed. The Reda Lotus program by Pelton (version 1.4) was used to calculate the monthly power bill for several size pumps used in this field. These results were plotted on the graph for rates of 350 to 1000 bfpd. Operating at near peak efficiency was assumed for all pumps. Actual power bills were then plotted for a few esp equipped wells. Note that the actual bills were significantly higher than the calculated values. This could result from selecting pumps oversized to the well's maximum capacity. The system efficiencies for ESP's are poorer (< 40 percent) at low rates. Some of the above wells are directional, which may account for the disappointing system efficiencies. Others are probably over pumped; however, all rod pumped wells have pumpoff controllers. Some improvement seems possible.

CONCLUSIONS

1. Rod pumping systems deserve to be the most widely used means of artificial lift on relatively low volume wells.

2. Normally use conventional units with NEMA "D" motors. Other type units and prime movers must be justified. Use the wave equation to evaluate possible improvements for non-conventional units. Size the units to get the production in about 20 hours.

3. Rod design (and proper handling) becomes exceedingly important when rod stresses exceed 25,000 psi.

4. If feasible, use relatively large plungers, simple insert pumps with tight fits and long lasting components. Space out closely in wells with gas interference problems. Work with the pump service companies to reduce repair costs.

5. Use the natural gas anchor whenever feasible and the packer gas anchor in gassy wells when pumping from above the perforations. The poor-boy gas anchor may be used for low volume wells.

6. Use large tubing to accommodate the rods and insert pumps that are needed for efficient operation. Tubing anchors are recommended for all medium to deep wells.

7. Pumpoff controllers are normally worthwhile in wells that can be pumped off.

8. Check your system efficiency. Wells that are out of line may require some redesign.

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APPENDIX A
NOMENCLATURE

L = NET LIFT (SURFACE TO FLUID LEVEL), FEET
Q = RATE, BPD
SG = SPECIFIC GRAVITY
HHP = HYDRAULIC HORSEPOWER , HP
PRHP = POLISHED ROD HORSEPOWER, HP
MSHP = MOTOR SHAFT HORSEPOWER, HP
RMHP = RATED MOTOR HORSEPOWER, HP
IEHP = INPUT ELECTRICAL HORSEPOWER, HP

HYDRAULIC HORSEPOWER = HHP = L * Q *SG / 136000
RMHP = APPROXIMATELY 2 * HHP (FOR DESIGN PURPOSES)
MSHP = APPROXIMATELY 1.25 * PRHP (AS PER GAULT AND G & S)

IEHP = (TYPICALLY)	1.25 *PRHP /.75 = 1.67 * PRHP	(GAULT)
IEHP = (SUGGESTED)	1.35 *PRHP	(LEA AND WILSON)
IEHP = (EXPERIENCED)	2.00 *PRHP	(CLEGG)

SYSTEM EFFICIENCY FACTOR = SEF

SEF = USEFUL WORK / TOTAL INPUT WORK = HHP / IEHP

EXAMPLE: L = 5000 FT; Q = 300 BPD ; SG = 1.03
HHP = 5000*300*1.03/136000 = 11.36 HP
HHP = 12 HP (AS CALCULATED BY API RP 11L)
IEHP = 1.67 * 12 = 20.04 (GAULT)
SEF = 11.36 / 20.04 = .567
MONTHLY POWER BILL =IEHP*.746*24*30.4 * $/KWH
TYPICAL $/KWH = .06 AND IF ACTUAL USED HP = 20.04
MONTHLY POWER BILL =20.4*.746*24*30.4*.06 = $654.44

TYPICAL SYSTEM EFFICIENCY

PEF = PUMP EFFICIENCY FACTOR
MEF = MOTOR EFFICIENCY FACTOR
EEF = ELECTRICAL EFFICIENCY FACTOR
UEF = UNIT EFFICIENCY FACTOR
FEF = FRICTION EFFICIENCY FACTOR
I2RF = POWER LOSS FACTOR FOR ESP'S

ROD/BEAM UNIT SYSTEM EFFICIENCY=PEF*MEF*EEF*UEF*FEF
TYPICAL=.90*.88*.985*.80*.98* =.612
61.2 % (FOR GOOD PUMPING CONDITIONS)

THEORETICAL ESP UNIT SYSTEM EFFICIENCY=PEF*MEF*EEF*FEF*I2RF
=.60*.85*.985 *.98 *.96 = .47 OR 47 %
WILL BE LOWER IF USE SMALL CABLES, VSD, AND ROTARY GAS ANCHOR.
TYPICALLY DO NOT EXPECT A SYSTEM EFFICIENCY FACTOR OVER .40 FOR
SMALL (<600 BFPD) ESP INSTALLATIONS.

Table A

320D-256-100 UNIT

BFPD RATE	MEAS DEPTH	PMP EFF	ROD GRD	ROD LTH	ANCH DEPTH	API GRAV	H2O SG	TBG GRAD	UNIT TYPE	GIVN PIP	CSG GRAD	FLAP
360	5000	95	C-86	25	5000	30.0	1.05	.417	C	100	.379	264

PUMP BORE	STK LTH	SPM	MIN LOAD	MAX LOAD	STRESS RATIO	TORQUE X1000	HP	NON-DIMENSIONAL N/No	Fo/SKr	
1.25	70	23.7	1727	20066	1.076	296	22	0.418	0.128	
1.25	85	21.3	1700	19029	1.022	357	23	0.375	0.105	
1.25	100	19.2	2458	18176	0.954	413	24	0.339	0.090	
1.50	70	19.9	5041	17943	0.875	251	18	0.350	0.180	
1.50	85	16.4	4587	18070	0.892	307	18	0.288	0.149	
1.50	100	14.0	3966	17242	0.866	363	17	0.245	0.126	
1.75	70	16.1	4625	18951	0.935	274	16	0.283	0.238	
1.75	85	13.1	4767	18639	0.916	322	15	0.230	0.196	
1.75	100	11.0	6545	18782	0.880	355	15	0.193	0.167	
2.00	70	14.1	5640	19771	0.949	251	15	0.248	0.300	
2.00	85	10.8	7048	19853	0.918	306	14	0.191	0.246	###
2.00	100	9.2	6982	19221	0.890	347	14	0.161	0.209	

Table B

Rate (bpd)	Casing Size (inches ID)	Tubing Size (inches OD)
<265	4 1/2	2 3/8
<410	5 1/2	2 7/8
<850	7	2 7/8
>850	9 5/8	3 1/2

* Based on limiting the downward velocity to less than 0.3 fps--a practice followed in the Denver Unit.

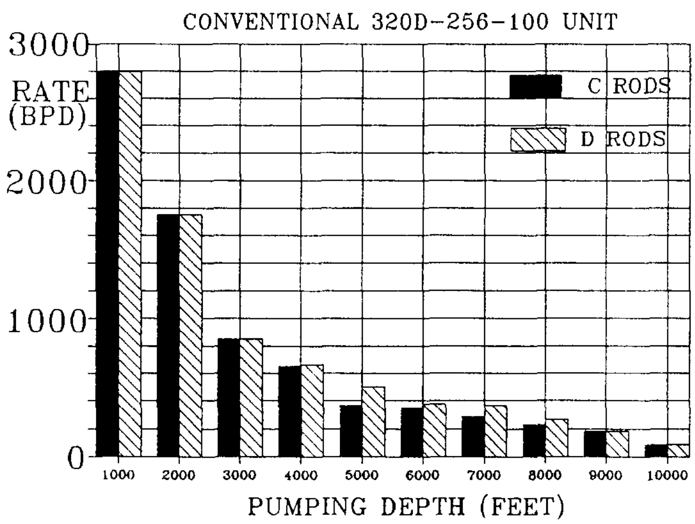

Figure 1 - Maximum pumping rate

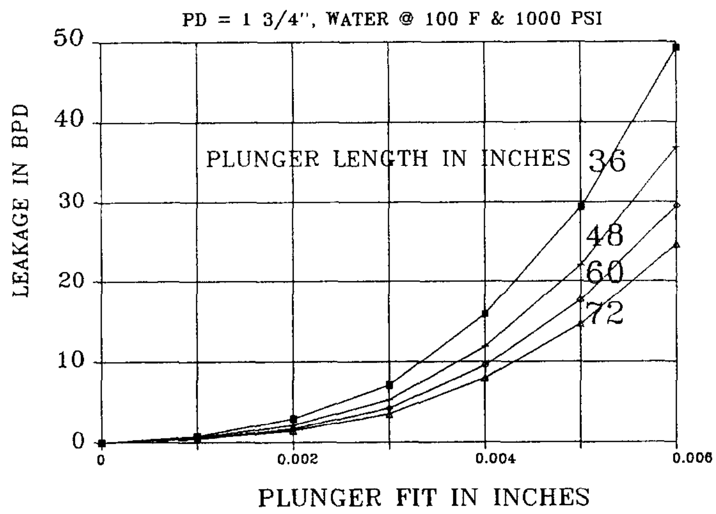

Figure 2 - Rod pump plunger slippage

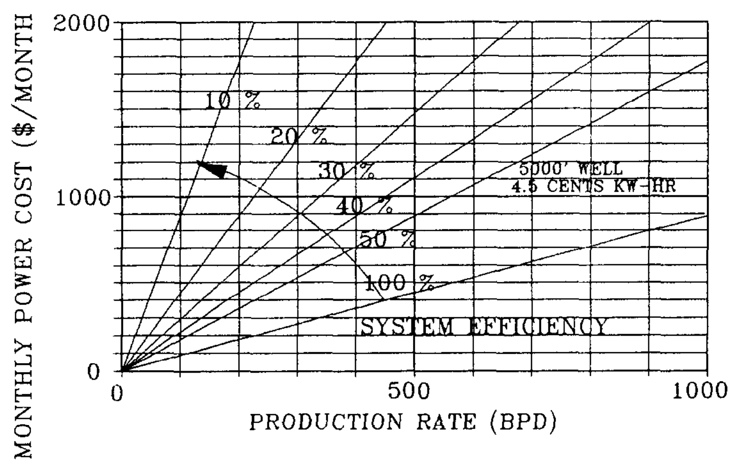

Figure 3 - Theoretical monthly power costs based on hydraulic horsepower

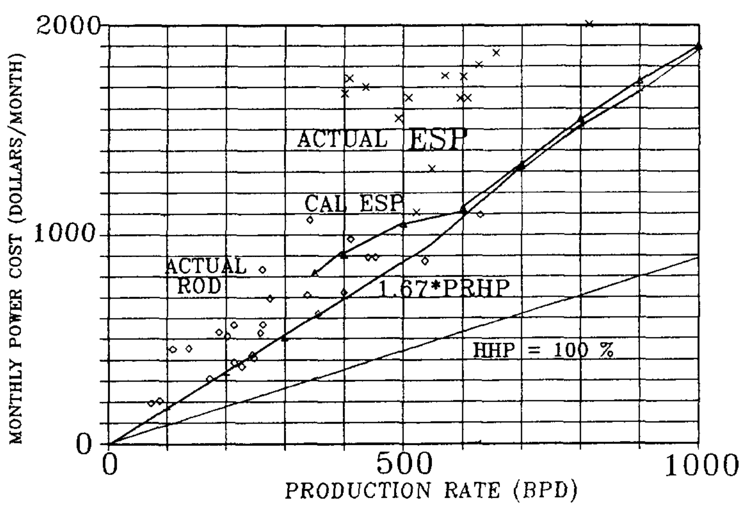

Figure 4 - Actual power costs vs. calculated costs: rod and ESP