COMPUTER-AIDED STUDIES OF ROD PUMPING SYSTEM PERFORMANCE

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INTRODUCTION

Although it looks simple, rod pumping equipment requires complex simulation methods to mimic the variety of actal conditions encountered in the field. This paper describes such a method which relates the influence of unit geometry, prime mover slip, rod design and downhole pump condition. The method is mathematical, but non-technical means such as dynamometer cards and electrical terms are used to describe it.

The intent of the paper is to study a variety of practical rod pumping questions that frequently arise. It has been found that the computer aided simulation has suggested performance phenomena not widely known or understood.

Portions of the subject method have been under development for twenty years. Some of the theoretical details are shown in Reference 1. Recent additions to the technology have been in handling the effects of inertia on gearbox and prime mover loading (Ref. 4). Most recent has been simulation of electrical performance of motors.

STATE OF THE ART

To illustrate the type of predictions that are possible, Fig. 1-a shows predicted surface and pump dynamometer cards for a 7850 ft. well being lifted with a conventional unit at 8.99 SPM with an ultra high slip prime mover in the low torque mode. Shown on the surface card is the permissible load diagram as modified by the effects of rotary and articulating inertia. Figure 1-b shows (predicted) net gearbox torque versus stroke and Fig. 1-c shows (predicted) line current versus stroke. The torque plot shows at a glance the number of torque reversals and the severity of torque extremes. The current plot has utility in evaluating motor load and unit balance. Tables 1 and 2 show the specific values of predicted items in a report format produced by the computer.

SOME PRACTICAL APPLICATIONS

The following illustrations relate to practical problems that arise in the field. No attempt is made to give general solutions applicable to all wells. The results apply only to the specific conditions studied.

SELECTION OF BEAM UNIT TYPE

Several types of pumping units are commercially available and there is usually a "best" unit for any given well and operating condition. Thus it is desirable to impartially compare the various units so that a proper selection can be made. Pumping units are different because of their geometric characteristics and counterbalance arrangements. The method mimics any type of beam unit by specifying 6 appropriate dimensions. Together with prime mover characteristics (torque-speed curves), unit geometry specifies the driving motion at the top of the rod string.

To illustrate, Fig. 2 shows how three types of units would perform on the same deep well. Pump depth is 9403 feet. Full liquid fillage is prescribed and the tubing is anchored. The units all have 120 inch nominal strokes and are operating a 1-3/4 inch pump at 12.6 to 12.8 SPM. The units are Air Balanced, Conventional and Mark II. The differences in surface dynamometer cards are apparent as are other indices. Relative performance is summarized below:

	Air Balanced	Conventional	Mark II
In-balance torque	2	3	1
Rod loading (Goodman diagram)	1	3	2
Structure loading	2	3	1
Power required	2	3	1
Gross pump displacement	2	3	1
Uniformity of torque loading	2	3	1

The inset numbers denote the relative standing. For example, the Mark II rates first in all categories except rod loading and would be the likely choice for this well. It is important to note that other types of units could prove better in other regions of the pumping spectrum.

One cannot generalize as to overall superiority of a unit. Comparisons must be done on a case by case basis. In the case of Fig. 2, the Mark II is much better than the Conventional unit in torsional performance. The permissible load diagram for the Mark II favors the down-to-right type of surface dynamometer card that it produces under these conditions. In contrast, the Conventional unit has a permissible load diagram that does not favor its surface card. These are all predictable items which bear on unit selection.

COMPARISON OF PRIME MOVERS

The choice of prime movers is also an important design consideration. Figure 3 shows relative performance of a Nema D motor and an ultra high slip motor in the medium torque mode. A conventional beam unit is producing the 8000 ft. well with an 1-1/2 inch pump at about 11.5 SPM. All items in the comparision are the same except the prime mover and sheaving required to produce near equal pumping speed. Performance of the two motors is summarized as follows:

	Nema D	Ultra High Slip
In-balance torque	2	1
Rod loading (Goodman diagram)	2	1
Structure loading	2	1
Power required	2	1
Gross pump displacement	2	1
Uniformity of torque loading	2	1

In this application, the ultra high slip prime mover is best. However, general conclusions cannot be drawn that any given prime mover is always superior. Certain conditions can be stated that will favor any given prime mover.

As mentioned earlier, torque/speed characteristics of the prime mover and unit geometry prescribe the motion of the top of the rod string. In Fig. 3-b, the ultra high slip motor has altered the motion of the rod string and is producing a perceptably different surface card from that of the Nema D motor. The effects of inertia have also altered the permissible load diagram.

EFFECT OF TUBING ANCHOR IN POUNDING WELL

Tubing anchors are installed to eliminate production loss due to tubing stretch and/or to decrease wear between tubing and casing. A little known fact is that anchors can also increase load on the equipment. Figure 4 shows a 5940 ft. well being pumped with a Conventional unit and a 1-1/4 inch pump at 13.4 SPM. With the tubing unanchored, peak in-balance torque is about 26.8% less than if the tubing were anchored. Peak structure load is 814 lbs. less (6.2%) in the unanchored case. Rod loading according to the Goodman Diagram is 15.4% less with unanchored tubing.

The lower loading has a simple explanation. The unanchored tubing acts as a spring to cushion and smooth out the transfer of load to and from the rod string. The well is pumped-off (producing at capacity) so that loss of effective pump stroke due to unanchored tubing is not causing a loss in production.

Tubing anchors should be run in most deep wells. But if maximum pump displacement is not needed and if tubular wear is not an over-riding consideration, omission of the tubing anchor will usually decrease equipment loading.

EFFECTS OF PUMP FILLAGE ON UNIT BALANCE AND LOADING

Wells that pump-off and pound fluid pose a problem in balancing units. Figure 5 shows a sequence of simulations of a 7300 ft. well being produced with a Conventional 120 inch unit (in the second hole-107.6 inch stroke). Pumping speed is about 10.5 SPM and the downhole pump is 1-1/2 inches in diameter. Figure 5-a shows the well with complete liquid fillage (no fluid pound). The unit is ideally balanced with 710000 in-lbs. of counterbalance effect.

Figure 5-b shows the well after it has pumped-off and has developed a slight fluid pound (90% fillage). The unit is still well balanced.

A more severe fluid pound is shown in Fig. 5-c. This indicates the pound occurring near mid-stroke such that liquid fillage is only about 56%. The surface card shape has changed drastically from that of full fillage. The different pattern of loads has caused the unit to be badly out of balance (rod heavy). This is known because the predicted in-balance requirement (816600 in-lbs.) exceeds the counterbalance in current use (710000 in-lbs.).

Finally Fig. 5-d shows the well pumped down to only about 24% liquid fillage. The out-of-balance condition has grown worse.

A practical conclusion is that proper counterbalance for a well with complete fillage will not be proper if it later pumps off. In this sequence, the gearbox was never overloaded very much even when the unit was out of balance. Other cases could be constructed showing serious gearbox overloads resulting from fluid pound after the unit was first properly balanced with a completely filled pump.

In Fig. 5, it was found that the gearbox was never badly overloaded after first being balanced with full fillage. The converse is not ture. It can be shown that if this well is perfectly balanced to the fluid pound case of Fig. 5-c and if fillage later increased to 100%, the gearbox will be overloaded by 36.6%. This situation is common in waterfloods. The unit could have been balanced to the fluid pound condition of Fig. 5-c, before flood response occurred. Later, flood response took place and pump fillage became complete. At this time the unit would become weight heavy and seriously overloaded torsionally. The practical conclusion is that units operating in waterfloods should be watched closely and rebalanced as flood response progresses.

Many studies have been made concerning the accuracy of the various predictive methods. Often, the conclusion is drawn that a given method is unable to satisfactorily reproduce loads, powers and torques derived from actual surface dynamometer cards. The predictive error is usually caused by misapplication of the method. For example if a method which is only capable of simulating full fillage (see Refs. 2 and 3) is applied to a real card similar to that shown in the fluid pound case of Fig. 5-c, a large predictive error will result. For example, a limited method might only apply to full liquid fillage like that of Fig. 5-a. Analysis of the actual card would indicate an in-balance peak torque of 222300 in-lbs. whereas the method would predict an in-balance peak torque of about 341000 in-lbs. This is a large error of 53.3%. Other large errors can result if methods which can only simulate full fillage and anchored tubing are applied to actual cases involving gas interference, unanchored tubing, fluid acceleration effects, etc. Simular errors can result if a method based on conventional units and low slip prime movers is applied to Air Balance, Mark II, BG and ultra high slip motor installations.

SIMULATION OF FLUID FRICTION AND ACCELERATION EFFECTS

In fairly deep wells producing sizeable quantities of gas, the pump load is essentially constant on the upstroke. Little additional load is required at the beginning of the upstroke to accelerate the fluid column upward. These dynamic effects in the fluid column are largely dispersed and diffused by gas in the tubing. This is undoubtedly the reason for the rather constant fluid load on the upstroke normally observed.

In shallow wells that are producing large quantities of water with little entrained gas, the pump load on the upstroke is anything but constant. In addition to the dead fluid load, the pump must provide an additional force to accelerate the fluid in the tubing and flowline and to offset fluid friction. These can be sizeable effects if large volumes of fluid (mostly water) are being pumped. Viscous effects can also be significant if heavy crudes are being lifted.

The additional loads on the pump required to accelerate fluid and to overcome viscous friction can be very large - even larger than the dead fluid load itself. Thus in shallow wells which produce large quantities of water with little entrained gas, these extra loads must be considered in designing in installation. Figure 6 shows a 2049 ft. well which is producing about 2500 BPD (mostly water) with a 3-3/4 inch pump. The simulated pump card shows a peak pump load of about 11435 lbs. This is much greater than the dead fluid load of 7434 lbs. because of the additional forces required to accelerate the fluid upward and to overcome fluid friction. Had the additional pump load not been considered, the installation could have been under designed and the equipment, once installed, could have been overloaded.

PREDICTION OF GEARBOX LOAD PATTERN

Severe fluid pound is undesirable in beam pumping. A fluid pound can cause severe rod loads, unscrewed rods, damaged pumps and backlash in the gearbox. It is desirable to eliminate a severe pound when possible. This is usually done by decreasing pump capacity to be more nearly equal to well capacity.

Figure 7 shows a 7700 ft. well which is simulated as having a severe pound. The peak torque on the gearbox is negative even though the unit is almost prefectly balanced. The negative peak torque signifies that the pumping unit is driving the prime mover rather than vice versa.

Figure 7-b is the corresponding plot of net torque versus stroke. This shows clearly how the peak torque on the gearbox is negative. It further shows the drastic reversal from negative to positive torque when the pump pounds fluid just below midway on the downstroke. This conceivably could be damaging to teeth in the intermediate and high speed gear trains.

It is desirable to design and operate an installation so that torque loading is uniform as possible. This is a predictable item and makes possible the choice of equipment configurations that have desirable cyclic load characteristics.

REMOVING GEARBOX OVERLOADS IN POUNDING WELLS

Often a gearbox overload can be relieved in a pounding well without a loss in production. This is done by decreasing pump capacity.

There are several feasible ways to decrease pump displacement and possibly to eliminate a gearbox overload. These are:

1) Shorten the surface stroke
2) Decrease pumping speed
3) Install a smaller downhole pump
4) Combinations of the above.

A typical sequence of feasible attempts to relieve a gearbox overload in a pounding well is shown in Fig. 8. The well is 6600 feet deep and is being produced with a C456-304-144 unit at 10.9 SPM. A 1-3/4 inch downhole pump is initially in use. Figure 8-a shows the initial surface and pump dynamometer cards which reveal the severe fluid pound and a gearbox overload of 5%.

Figure 8-b shows what would happen if a smaller (1-1/4 inch) pump were installed. Note that the pump load decreases and the liquid fillage increases to maintain the same production rate (the fluid pound moves higher in the stroke). Unexpectedly, the smaller pump actually increased the gearbox overload to 17.1% even with the unit in perfect balance.

Decreasing pumping speed is another possible way for decreasing or eliminating a torque overload. Figure 8-c shows what would happen if the pumping speed were decreased to 7.7 SPM. Owing to sheave size limitations, this is the slowest practical speed attainable. Surprisingly gearbox overload is increased to 26.1%. Thus, decreasing pumping speed would not relieve the gearbox overload in this well.

Another possibility is to shorten the stroke. Figure 8-d shows what would happen if the stroke is shortened from 144 inches to 106.7 inches. The 1-3/4 inch pump and speed of 10.9 SPM are retained. Note that the fluid pound moves higher in the stroke. The net stroke is still sufficient to pump the well to capacity. The peak torque has been reduced to 78.7% of rating and the gearbox is no longer overloaded. Shortening the stroke is certainly the best strategy for this well.

These examples show how difficult it is to intuitively determine the best tactic to relieve a gearbox overload. As it turns out, shortening the stroke is the best (and only) option which will eliminate the gearbox overload in this case. The fact that torque loading can increase at a slower speed and with a smaller pump is not intuitively obvious. In this case, the torque increased because the dynamometer card shapes with the smaller pump and slower speed did not fit the permissible load diagrams as well. Other examples could have been chosen which would have favored a smaller pump or a decrease in pumping speed. A flexible computer program is a helpful tool in knowing how to combat gearbox overloads.

SIMULATION OF FIBER GLASS RODS AND SINKER BARS

Fiber glass sucker rods are gaining acceptance and pose a special simulative problem especially when combined with steel rods.

Figure 9-a shows a simulation of a well with a 2-1/4 inch pump set at 4512 ft. An Air Balanced unit (A640-305-168) is lifting the well at 10 SPM. The rod string consists of about 4100 feet of fiber glass rods with 400 feet of 1-1/2 inch steel sinker bars on bottom. Figure 9-b shows the same conditions except that a straight fiber glass string (no sinker bars) is being used. The role of the sinker bars is two-fold, i.e. to provide stiff rods and weight to off-set pump friction and to increase the pump stroke. Comparison indicates that the sinker bars add 40.8 inches to the pump stroke in this case which is equivalent to an additional 234 BPD in pump displacement. Owing to the relatively low modulus, straight fiber glass rods would tend to produce a shorter stroke than desired. However, when combined with sinker bars (which add mass near the bottom of the string and cause over-travel), a suitable pump stroke can be obtained.

Fiber glass rod strings produce surface dynamometer cards that are unlike those of conventional steel strings. They often have an undertravel appearance and frequently mate best with conventional unit geometry.

ELECTRICAL PREDICTIONS

Electrical items are predictable with good precision based on experience to date. These are logical outgrowths of the more established mechanical predictions. Once motor torque and speed variations are established (via the mechanical simulation), electrical quantities are predictable because these are directly related to torque and speed for any given motor size and design. Several practical examples follow which illustrate some of the field problems that can be studied.

A fairly common practice is to over-size prime movers. The reason is usually to make sure the prime mover is large enough after the well has responded to the waterflood. This may not always be desirable. Figure 10 shows a well with a 1-1/2 inch pump set at 8053 feet. A conventional unit and 60 HP motor are in use at 8.9 SPM. Power input to the motor is 34.6 KW.

Based on a power cost of $0.03 per KWH, monthly power bill would be $747.36 without consideration of line and transformer losses, demand charge and power factor penalty if any. The motor is fully loaded electrically.

If the same well had been powered by a 100 HP motor, the conditions of Fig. 11 are predicted. Note that the predicted dynamometer cards differ only slightly with those of the smaller motor. The larger motor is lightly loaded (81.8 thermal amps versus 118 amps full load rating). At $0.03 per KWH, the monthly power bill on the same basis would be $879.81. The saving for the smaller motor is $132.45 per month or $1589.40 per year. Power and capital cost savings for the smaller motor could make use of the 100 HP motor unattractive prior to flood response.

The potential power savings for pump-off controls or percentage timers can also be studied. Figure 12 shows predicted conditions for a similar well which has a severe pound. Production rate is about 200 BPD based on full-time pumping with the stabilized fluid pound as shown. The monthly power bill for full time pumping is $808.91. The predictions of Fig. 11 show average performance of the well being operated under a properly functioning pump-off controller or a precisely adjusted percentage timer. Comparison of net pump displacements suggests the pump-off controller (or percentage timer) would operate the well about 70 percent of the time. Under this intermittent producing cycle, monthly power cost would be $615.86. This indicates a saving of $193.05 per month ($2316.60 per year) if a pump-off control or percentage timer were used. At least two factors contribute to this saving, i.e.

1) the well is shut down 30% of the time,
2) overall system efficiency is lower in the severe fluid pound case.

Tending to offset power cost savings under intermittent pumping is the possibility of lost production. For example if the well has a high productivity index and a low static reservoir pressure, its montly production rate may be less under intermittent pumping conditions than under continuous pumping conditions. Flow into the wellbore in this type of well would be impeded during shut-down periods because of back pressure on the reservoir. Thus some wells might need to be produced continuously, regardless of power costs and the undesirable mechanical effects of fluid pound.

For wells that need to be pumped full-time, downhole pump displacement should be decreased to only slightly exceed the well's productive capability. Figure 13 shows the same well producing fulltime but at a slower pumping speed of 6.5 SPM. Monthly power bill under these conditions is predicted to be $694.49. Pump displacement and well production capability are closely matched so that no severe fluid pound is occurring. Rod and gearbox loading are less under continuous pumping at the reduced speed than under intermittent pumping at the higher speed. Further, if production rate is increased only slightly with continuous pumping at 6.5 SPM, the power cost saving ($78.63 per month) of intermittent pumping at 9.1 SPM will be completely offset. Thus it is seen that the pros and cons of continuous versus intermittent pumping are complex, but are questions that should be studied with both well productivity and equipment loading in mind.

CONCLUSIONS

1. A flexible simulation method is required to mimic the wide variety of real conditions encountered in beam pumping.
2. Various types of beam units, prime movers and rod strings perform differently. There is a best combination of unit, prime mover and rod string for given conditions. This best combination is predictable.
3. Downhole pump conditions exert as much influence on system behavior and surface loading as do the beam unit, prime mover and rod string. In order to properly reproduce actual measurements, the downhole pump simulation must be flexible enough to simulate full fillage, fluid pound, gas interference and unanchored tubing.
4. Electrical parameters and power costs are predictable quantities.

REFERENCES

1. Gibbs, S. G., "Predicting the Behavior of Sucker Rod Pumping Systems", J. Pet. Tech. (July 1963) 769-778; Trans., AIME, 228.

2. "API Recommended Practice for Design Calculations for Sucker Rod Pumping Systems (Conventional Units)", API RP 11L; Dallas, Texas (February 1977).

3. "API Catalog of Analog Computer Dynamometer Cards", API Bul. 11L2; Dallas, Texas (December 1969).

4) Gibbs, S. G., "Computing Gearbox Torque and Motor Loading for Beam Units with Consideration of Inertia Effects"; J. Pet Tech. (September 1975) 1153-1159.

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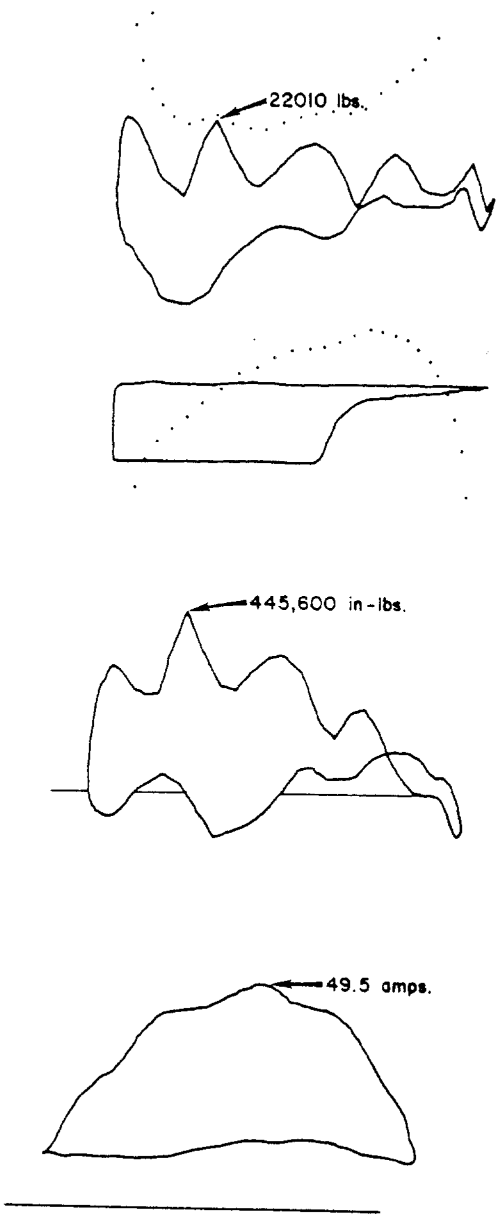


FIGURE 1

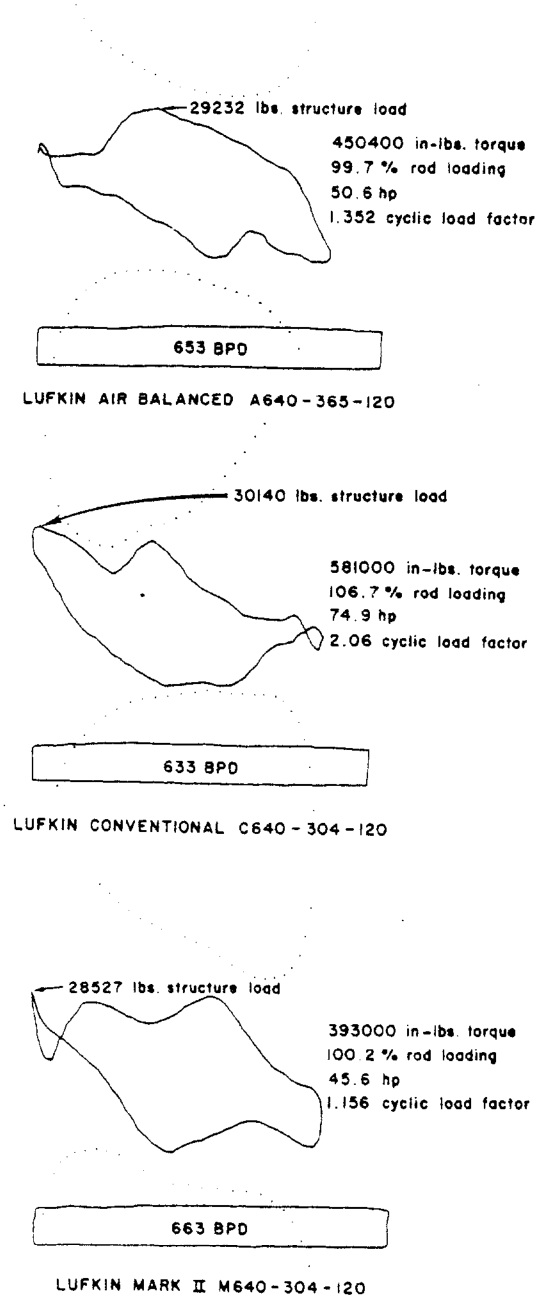


FIGURE 2

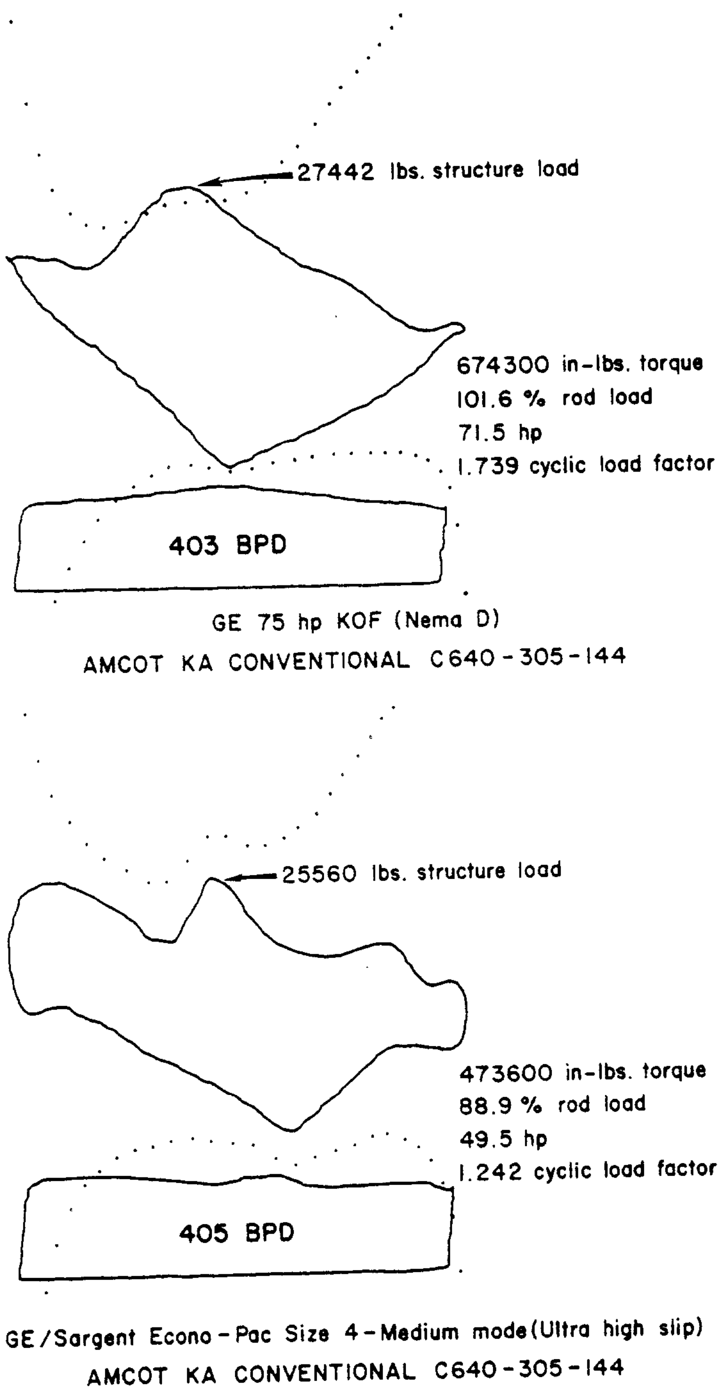


FIGURE 3

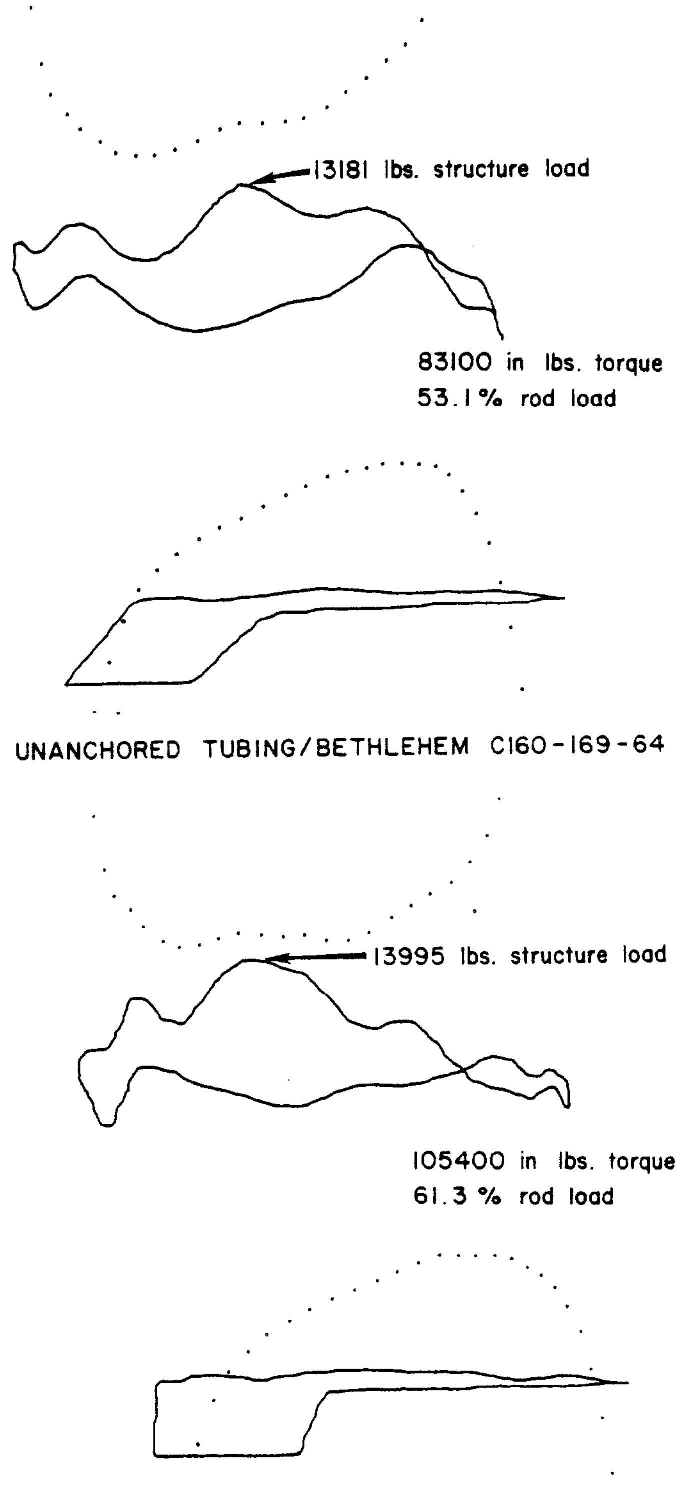


FIGURE 4

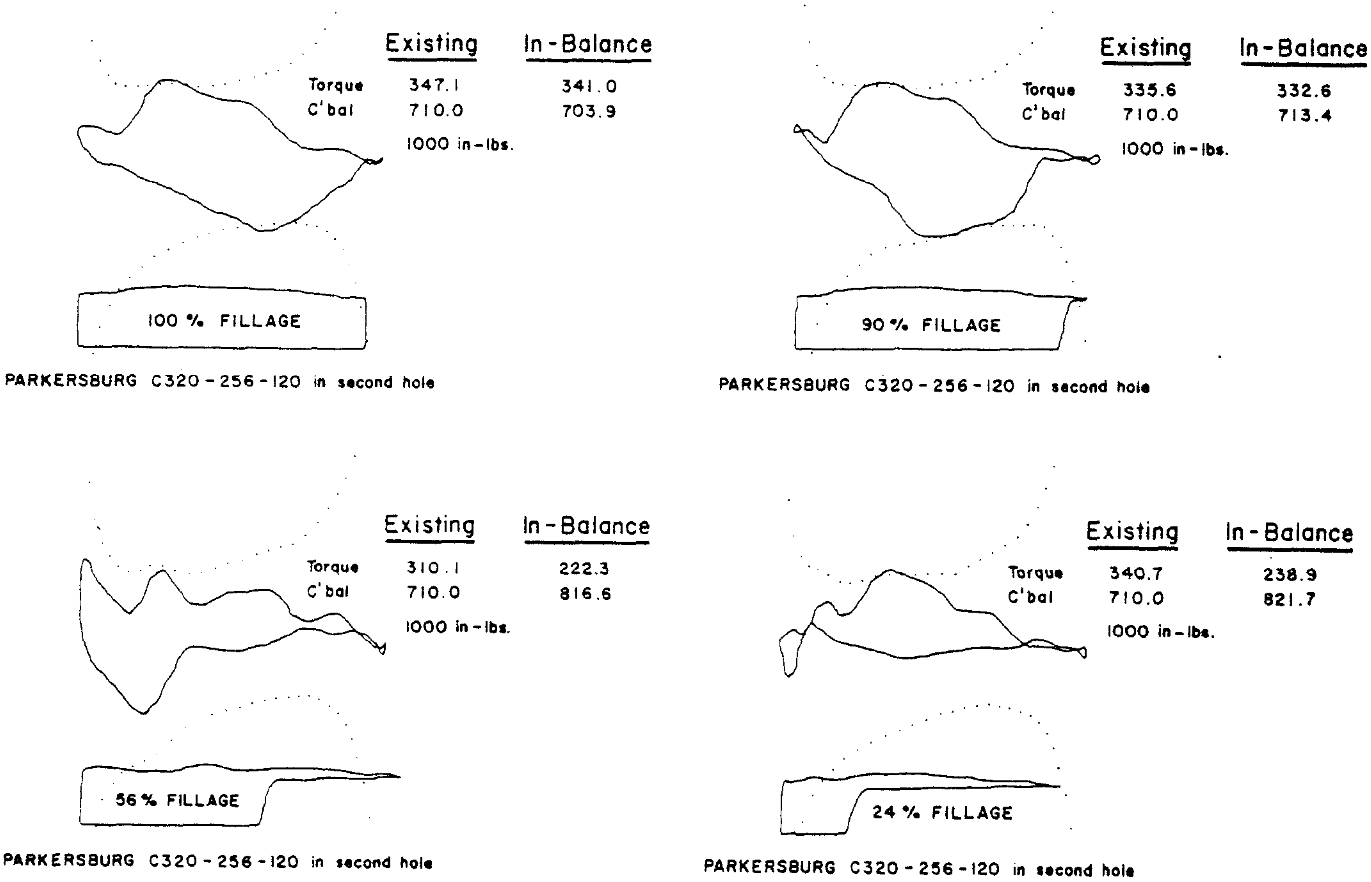


FIGURE 5

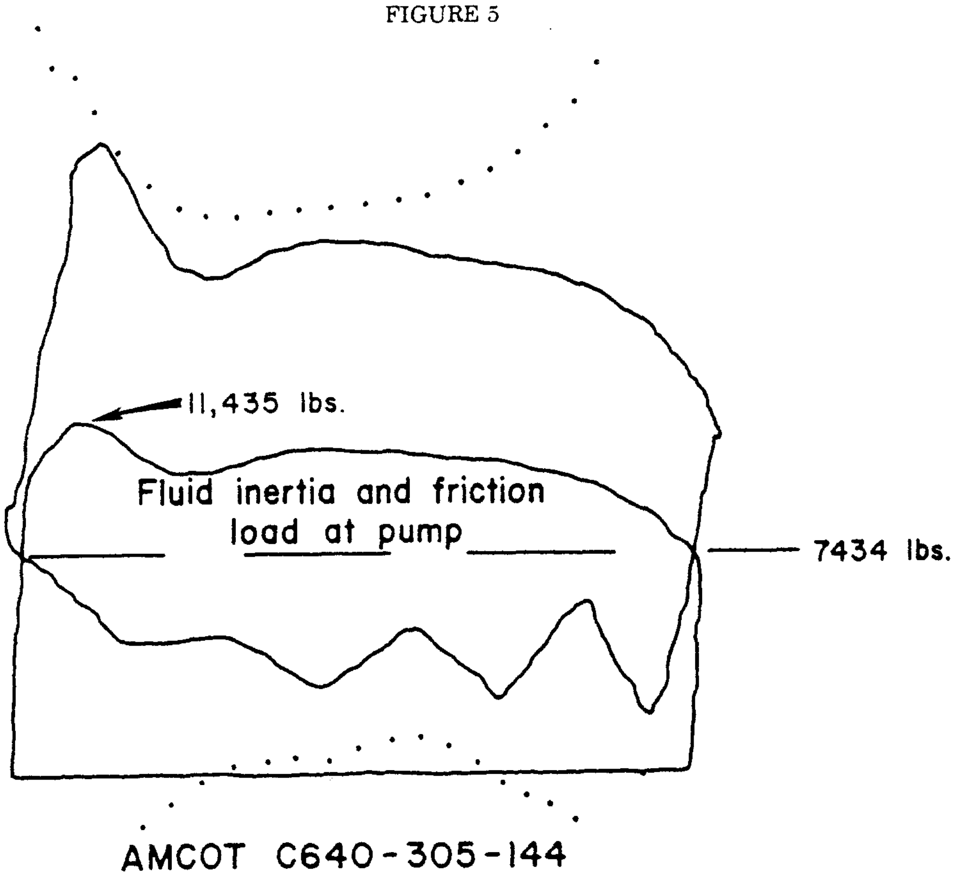
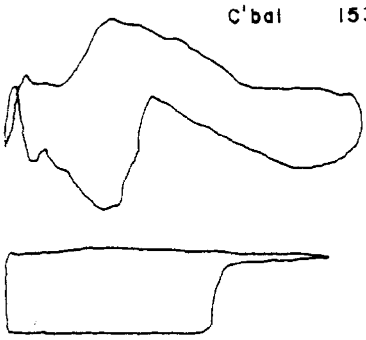


FIGURE 6

	Existing	In-Balance
Torque	-483.8	-461.4
C'bal	1531.5	1553.8



LUFKIN M640-305-168 in second hole

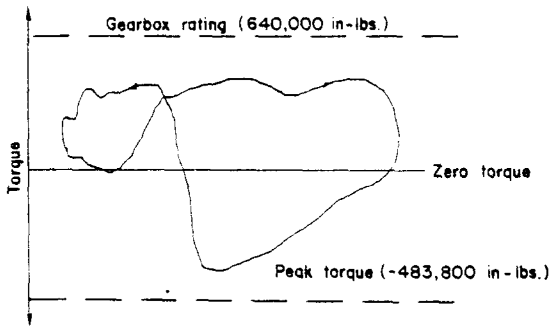
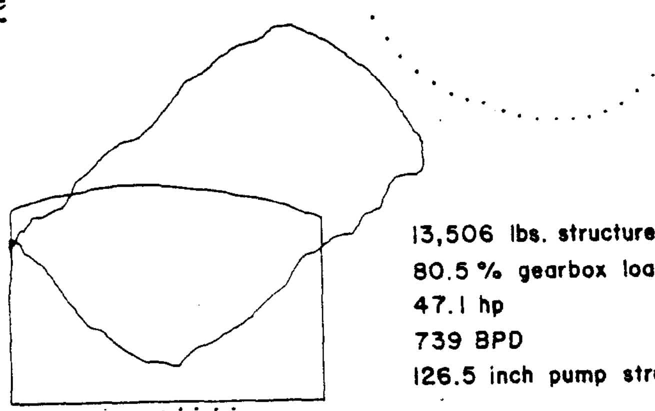
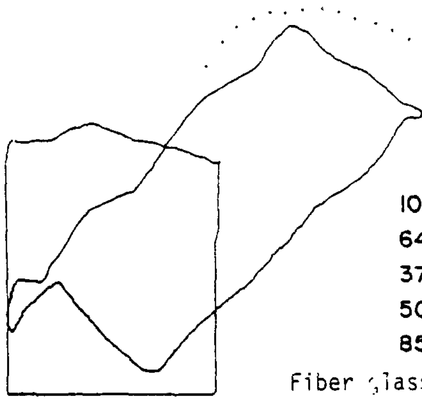


FIGURE 7



13,506 lbs. structure load
80.5 % gearbox load
47.1 hp
739 BPD
126.5 inch pump stroke

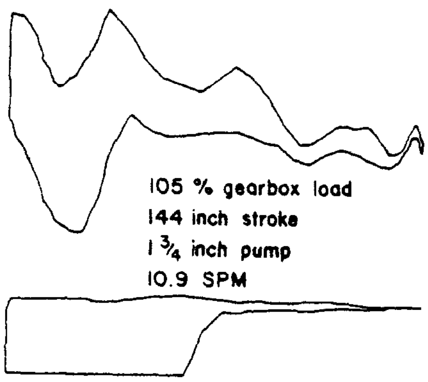
Fiber glass rods with 400 ft. of 1½ inch sinker bars



10,609 lbs. structure load
64.2 % gearbox load
37.5 hp
505 BPD
85.7 inch pump stroke

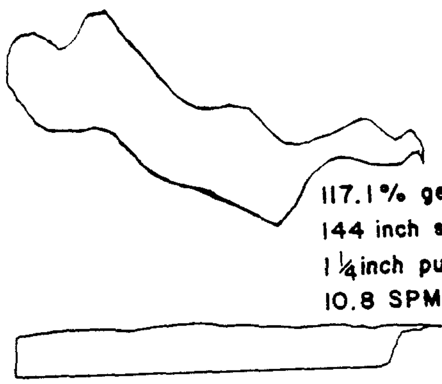
Fiber glass rods w/o sinker bars

FIGURE 9



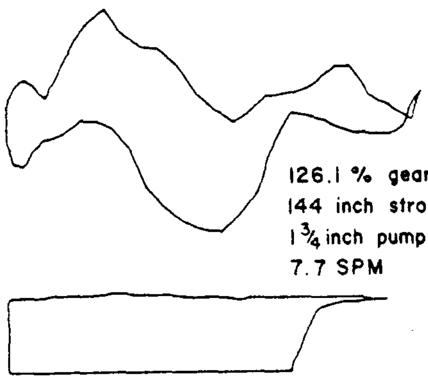
105 % gearbox load
144 inch stroke
1 ¾ inch pump
10.9 SPM

Base case - Overloaded gearbox in pounding well



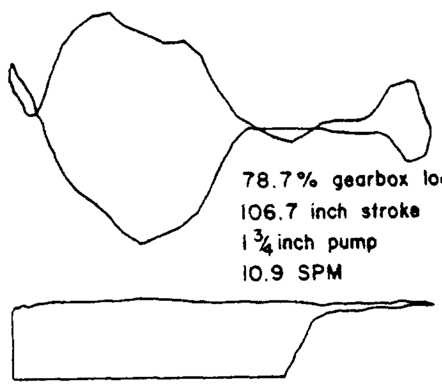
117.1 % gearbox load
144 inch stroke
1 ¼ inch pump
10.8 SPM

Effect of installing smaller pump



126.1 % gearbox load
144 inch stroke
1 ¾ inch pump
7.7 SPM

Effect of decreasing pumping speed



78.7 % gearbox load
106.7 inch stroke
1 ¾ inch pump
10.9 SPM

Effect of shortening stroke

FIGURE 8

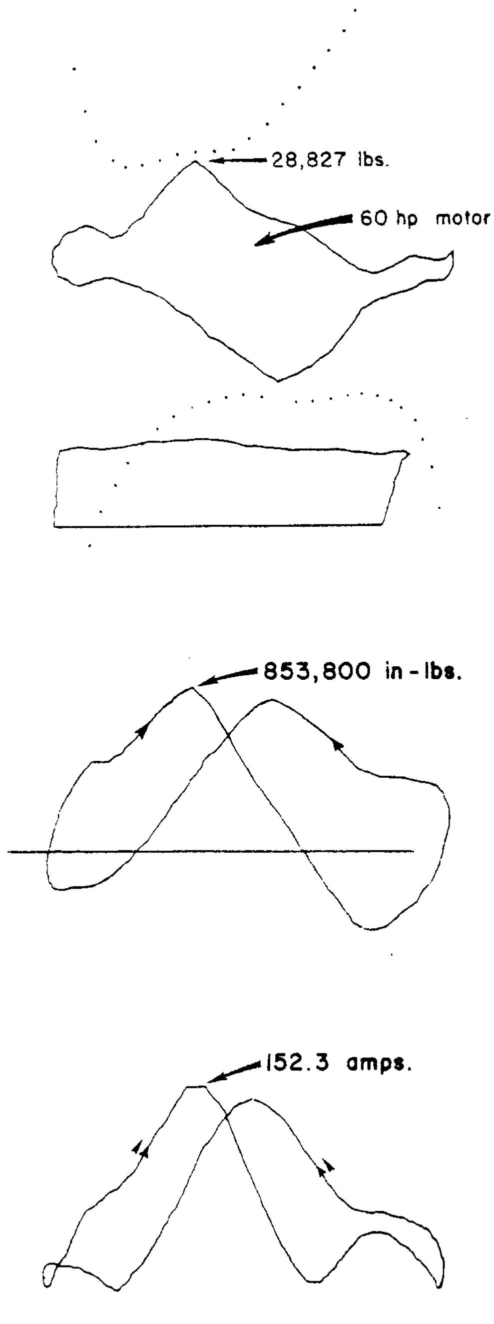


FIGURE 10

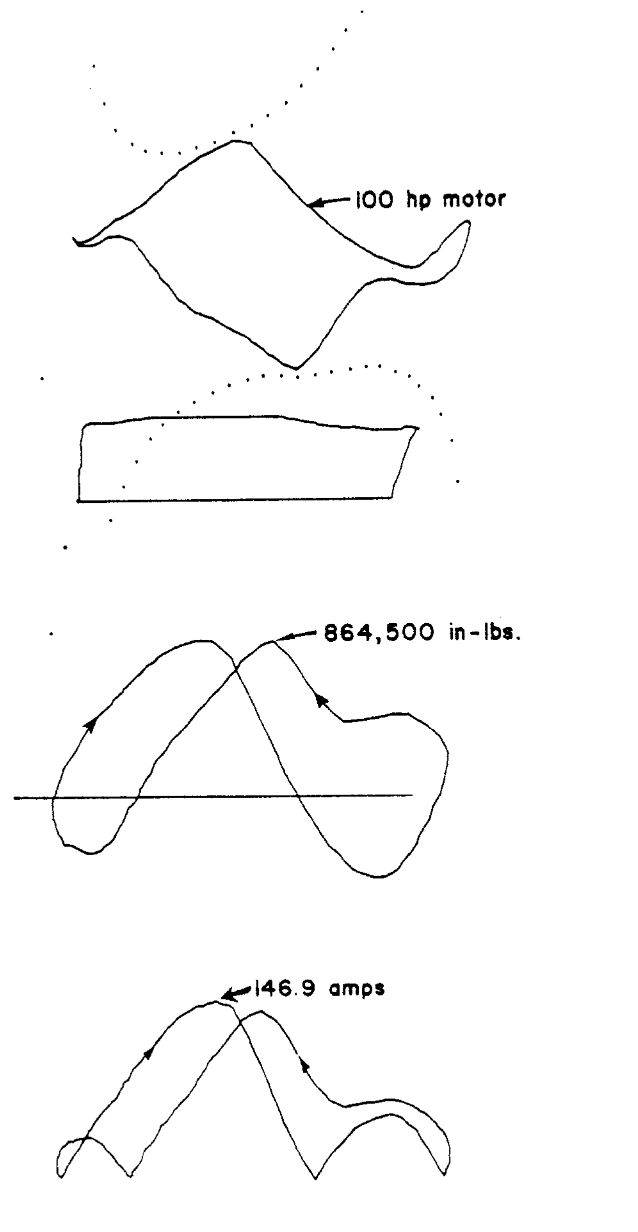
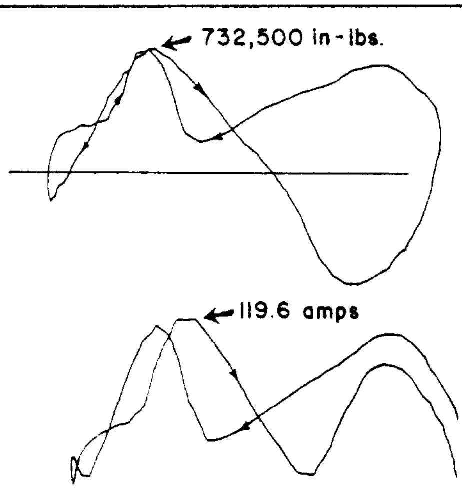


FIGURE 11



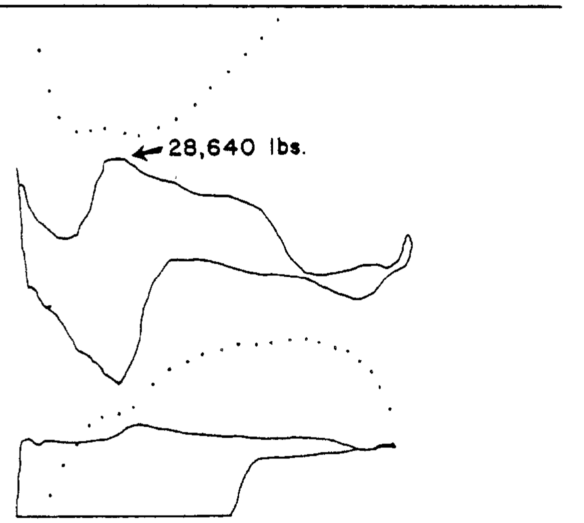


FIGURE 12

TABLE 1

PRIME MOVER ********************

MFGR AND TYPE: SARGENT ECONO-PAC SIZE 4 (LOW TORQUE MODE)
MAX SPEED (RPM): 1214.9 SPEED VARIATION (%): 32.6
MIN SPEED (RPM): 818.4 MOTOR TORQUE CLF: 1.4607
PEAK POWER (HP): 30.9 REGENERATIVE POWER (HP): -2.3
POWER REQUIRED (HP): 19.2

PUMPING UNIT ********************

MFGR AND TYPE: AMCOT C456-365-120 (TF-119 CRANKS)
ACTUAL MAX LOAD (LBS): 22010. ACTUAL MIN LOAD (LBS): 10099.
AVERAGE PUMPING SPEED (SPM): 8.9 MAX LOAD (% OF RATING): 60.3
POLISHED ROD POWER (HP): 11.8

SUMMARY OF GEARBOX LOADING ***

	EXISTING	IN BALANCE
PEAK TORQUE (M IN-LBS):	445.6	281.3
COUNTERBALANCE (M IN-LBS):	865.2	1070.6
PERCENT OF GEARBOX RATING:	97.7	61.7

SURFACE ROD LOADING *************

API TAPER DESCRIPTION: 76
MAX STRESS (PSI): 36602. MIN STRESS (PSI): 16794.

ROD LOADING AS % OF API-GOODMAN GUIDE

SERVICE FACTOR	CLASS C RODS	CLASS D RODS	CLASS K RODS
1.0	130.7	92.6	150.6
0.9	165.7	112.7	195.0
0.8	226.0	143.9	276.5

TABLE 2

DOWNHOLE PUMP PERFORMANCE **********

GROSS PUMP STROKE (IN): 119.3 NET PUMP STROKE (IN): 65.6

DISPLACEMENT BASED ON GROSS PUMP STROKE (BPD): 380.4
DISPLACEMENT BASED ON NET PUMP STROKE (BPD): 209.1

TUBING STRETCH (IN): .0 LOST DISPLACEMENT (BPD): .0

OTHER BASIC DATA *******************

GEARBOX RATING (M IN-LBS): 456.0
COMPUTED SURFACE STROKE (IN): 119.8
DIRECTION OF ROTATION: COUNTER CLOCKWISE - WELL TO RIGHT

ASSUMED DRIVE TRAIN EFFICIENCY (%): 95.0
OVERALL SPEED RATIO: 119.1

MOMENTS OF INERTIA (SLUG FT-FT)
ARTICULATING ELEMENTS: 12422.
ALL ROTARY ELEMENTS FROM MOTOR TO C'WEIGHTS: 68323.
CRANKS, C'WEIGHTS AND SLOW SPEED SHAFT/GEAR: 13230.

ROD STRING DAMPING FACTOR: .090 UP, .13 DOWN
BOUYANT ROD WEIGHT (LBS): 13500.
PUMP DEPTH (FT): 7850.

FLUID LOAD ON PUMP (LBS): 4950. PUMP BORE SIZE (IN): 1.7500

OPTIONAL POWER SUMMARY *************

OUTPUT OF MOTOR (HP): 12.6 INPUT TO MOTOR (HP): 20.5
PEAK UPSTROKE AMPS: 49.5 PEAK DOWNSTROKE AMPS: 15.5
THERMAL CURRENT (RMS AMPS): 25.1 POWER FACTOR: .766
ELECTRICAL CLF: 1.164

LOSSES IN SURFACE UNIT AND DRIVE TRAIN (HP): .9
POLISHED ROD POWER (HP): 11.8
LOSSES ALONG ROD STRING (HP): 3.2
USEFUL DOWNHOLE PUMP OUTPUT (HP): 8.5

SURFACE EQUIPMENT EFFICIENCY (%): 57.3
OVERALL SYSTEM EFFICIENCY (%): 41.5