Effect of Fluid Buoyancy on Rod String Loads and Stresses

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Introduction:

The purpose of this article is to review the effect of fluid buoyancy has on rod stresses and rod loads. The hydrostatic pressure caused by the tubing fluids has an impact on rod stresses at the rod element level. Use of the Downhole Load Cell data can help resolve the question: What is the magnitude of the loads displayed as the downhole pump card? The effective force on the rods is the ideal method to display of the downhole pump cards. Some discussion will be based on dynamometer data collected on an Amerada Hess well in conjunction with SANDIA using the Glen Albert Downhole Dynamometer. The purpose of this paper is to discuss the true/effective load versus stress argument for display of downhole dynamometer data and analysis of stresses on a rod element.

Overview:

The Electronic Downhole Load Cells (DHLC) is used to measure rod loading and the DHLC was developed by Glen Albert Engineering (1). The DHLC is unique because it can be mounted at a particular location in the rod string (usually between two rod tapers) and the dynamometer data can then be collected while the well is operating. Prior to 1996, approximately 25 DHLC test had been conducted by the petroleum industry. These test were proprietary and the results were not generally available. In 1996 SANDIA coordinated collecting additional data using the DHLC for different types of well conditions. The petroleum industry provided wells and paid for the installation cost to run the DHLC. SANDIA and Glen Albert Engineering collected, de-coded and presented the data. NABLA provided the surface Dynamometer measurements at the same time as the DHLC collected data.

Amerada Hess participated in the project and provided SSAU 4115 well to SANDIA for purposed to run the DHLC. The test was identified as "Downhole Dynamometer Testing Gassy Well w/separator on Amerada Hess Corporation SSAU 4115 well" Selected extracts of the data are for tools ran on August 1996 08/04/96 10:00:00 AM.

Purpose of Test:

The reasons why Amerada Hess chose to participate in the test and run an DHLC in a rod pumped well were: 1) Verify calculations made by predictive and diagnostic wave equation programs. 2) Determine accuracy of predicted loads by modeling programs. 3) Compare compressive forces Calculated Vs Actual in bottom of rod string. 4) Understand the bottom of rod string because of high number tubing and rod failures in lower section are attributed to rod buckling, due to compressive (negative) force.

Data Collection Procedure:

The following list briefly displays the steps followed to collect the DHLC data on a well:
1. Install DHLC in Rod String, usually at rod taper junctions.
2. Produce well and allow time for well to stabilize
3. Run Surface Dynamometer simultaneously with DHLC
4. Collect both Dynamic and Static Dynamometer data at same time.
5. Retrieve DHLC

6. Using calibration coefficient Process data collected into Load, Pressure, Temperature, Acceleration, Position, etc.

7. Store data in "Downhole Dynamometer Database" for access through the DownDyn program.

Dynamometer Card:

A Dynamometer is a device that measures the applied load at increments of position over one complete stroke, this measurement is usually made by mounting the dynamometer in a spacer at the carrier bar, so that the entire weight of the rod string can be measured as the fluid is lifted to the surface. The DHLC should be thought of as a dynamometer that can be placed at any position in the rod string. A surface dynamometer card is the plot of the measured rod loads at the various positions throughout a complete stroke; the load is usually displayed in pounds of force and the position is usually displayed in inches. The pump dynamometer card is a plot of the loads at various positions of pump stroke and represents the load the pump applies to the bottom of the rod string. Identifying how the pump is performing and analysis of downhole problems is one of the primary uses of the pump dynamometer card. Dynamometer cards are displayed by commercial predictive and diagnostic software for the purposes of design and diagnosing Sucker Rod Systems.

One of the objectives for Amerada Hess, AHC, to participate in the DHLC project was to resolve the display of the downhole pump card. Analysis of the actual DHLC measured pump card would be used to verify that the downhole pump card calculated from the AHC internally developed program for diagnostic wave equation.

Rod String Design:

In general a rod string is designed for a long operational life, where the planned failure mode would be due to fatigue, wear out the rods, rather than due to a tensile failure caused by applying a maximum load near the peak tensile strength of the rod. The rod at the top of each rod string taper supports the forces applied to the bottom of the rod string by the pump, plus the weight of the all the rods connecting the pump to the top rod in each taper. Design of the to resist fatigue failure is of primary interest because the rod string operates under repetitive cyclic loading conditions and is subjected to a minimum and maximum load during each stroke. In general, maximum allowable working stresses should not be higher than about 30,000 to 40,000 psi, although some sucker rods, such as the "high tensile strength" rods are rated at 40,000 to 50,000 psi maximum. Operation of rods in corrosive environments requires that the maximum allowable stress be decreased.

Maximum/minimum rod stress is determined by dividing the peak/minimum load on the top rod of each taper by the cross-sectional area of the top rod. Stress range is determined by subtracting maximum minus minimum stress. The normal criteria for rod design guidelines are the maximum stress must never exceed the *maximum allowable stress* for the particular grade of rod being used, nor should the *allowable stress range* be exceeded. This design method is based on the modified Goodman diagram and can be reviewed in the API publication RP 11BR. New manufactured rods have improved strength and resistance too fatigue failures (7) due to manufacture improved quality. The measured peak stress and stress range must be less than the Goodman guide allowable stress and stress limit the loading on the top most rod in each rod taper. This is compounded further in very corrosive environments and/or old rods where the service factor assumed is too high and not reflective of the operating environment. As the minimum stress approaches "zero" the maximum allowable stress approaches 25% of the tensile stress is then derated for service.

The loads on the top rod in a taper are the highest and the load decreases as the suspended rod weight decreases as you move closer to the pump. The largest diameter rods are located at the top of the string and the diameter rod string decreases when the length of each taper will result in the stresses on the top rod in each taper being equal. There are two general methods of designing the lengths of the tapered sucker rod string: 1) Assign to each of the

graduated sections of the string its maximum stress. In other words, a point in the string is determined at which the stress in the rod equals the arbitrarily selected maximum safe working stress; from this point up a larger size rod is used. 2) Design the string so the unit stresses are equal in the top rod of each of the different-sized sections of the string. The second method is usually safer, since it provides a greater safety margin as far as corrosion pitting is concerned. However, some companies prefer the first method, in which the maximum allowable stress is placed in the top rod of the smallest (lowest) size. With this method, rod breaks would occur in the smallest rod and would prevent any damage due to buckling of rods that occurs from breaks farther up the hole.

Buckling:

Rod strings have been shown to behave as a slender Euler column and are subject to buckling if the rod string experiences any (very small) compressive loading (4). During the pumping cycle: 1) On the upstroke with the traveling valve shut, the rod string will be under tension as the buoyed rods and fluids loads are lifted and 2) On the down stroke once the traveling valve is opened and the standing valve is shut, the rods are suspended in fluid and still under tension. But any upward force applied to the rod string from the plunger/pump assembly can result in compressive (negative) loads and cause rod buckling at the bottom of the rod string just above the pump. The bottom rods only carry the weight of the plunger, zero tension, and have the highest potential to experience compression on the down stroke. To keep the rod string in tension throughout the pump cycle, a section of large diameter rods, called *sinkerbar*, is placed just above the pump. The large-diameter sucker rods (such as 1.5 inch) replaces a length of sucker rods immediately above the pump and this section of the rod string is both heavy enough to keep the sucker rods in tension.

Display of the downhole dynamometer pump card may be used to identify buckling tendencies of the rods at the pump; a negative load would indicate a compressive load on the rods from the pump. During the presentation concerning the display of the downhole pump card (5) the following observation was made: "Example output from predictive and diagnostic programs available on the market show different negative loads on the downstroke. We have measured pump cards with the DHLC that shows very little negative loads. Why can't industry get together and consistently show the same type of pump card." Upon a review of the output from various diagnostic programs, the opinion concerning the presentation of the loads applied to the rods by the pump usually falls into two categories: 1) Minimum load of the pump card is shifted down by a negative load equaling the buoyancy force applied to the rods and 2) Minimum load at zero or near zero as the plunger moves down through the pump. Accounting for buoyancy effects the placement of the loads with respect of the zero load line for the downhole dynamometer pump card. The Figure 1 displays the lateral acceleration measured by the DHLC, review of this data indicates large lateral accelerations have occurred at some points along the trace. Correlation of large lateral accelerations with negative effective loads could be used to identify where buckling of the rod string would occur due to compressive loads on the rod string.

Buoyancy Force:

The buoyancy force acting on a rod was defined long ago and is basically Archimedes' Principle which was defined as a buoyancy force acts on a object equal to the weight of the volume of fluid displaced by the object. The impact of the buoyancy force on a rod this force is:

- Rods suspended in a fluid appear to weigh less, Wrf.
- The apparent weight equals the weight in air, Wra, minus the buoyant force, Fbuoy.
- The apparent change in (or loss in) weight of a rod immersed in a fluid is due to the buoyant force.
- The force does not depend on the depth at which the object is submerged, but the specific gravity, SG, of the fluid displaced.
- The net resultant force on the rod is upward, opposing the weight of the rods.

How buoyancy is accounted for effects the placement of the pump cards loads with respect of the zero load line for any downhole dynamometer pump card. When the pump card is plotted in terms of true load then the downhole pump card tends to be shifted below the zero load by a force usually equal to the buoyancy force. When the pump card is plotted in terms of effective load then the minimum downhole pump card load tends to be near zero.

True Vs Effective Load:

For comparison purposes Figure 3 plots both true load and effective load using the weight of the rods buoyed in fluid as an example, i.e. a standing valve check. Table 1. List the rod string design installed in SSAU 4115 during the DHLC test and the parameters used for the calculations of the true and effective stress.

Example Calculations:
Effective Stress @ Surface = 12,124 – 1225 = 10,899 lbs.
Effective Stress @ (0.875) rod = 10,899 – (4390 – 444) = 6953 Lbs.
True stress @ Bottom (1.0) rod = 10,899 – 4390 + 0.1841*506psi = 6602 Lbs.

As shown in figure 2 the **Effective Load** is determined by considering the rod weight is equal to the rod weight in air reduced by the buoyancy force on each section of rods. **True Load** represents the stress or load acting at a molecular level and is the sum of the weight of rods below a point minus by the exposed area of the rod taper junction times the hydrostatic pressure at the taper change depth. Notice at zero depth in the well the effective and true loads are equal. The slope of the true load is the weight of the rods in air, while the slope of the effective load is the buoyed rod weight per foot. At each rod taper the true load is discontinuous and is shifted upward by the area of the uncovered rod times the hydrostatic pressure. At the bottom of the rod string the effective load is equal to zero, while the true load is equal to the rod area times the hydrostatic pressure.

Pump Card Example:
In this study a series of test were made by installing the DHLC at each of the rod tapers, just above the pump and just below the pump intake. On August 04, 1996 data was collected on Amerada Hess Well SSAU 4115 using the DHLC. The collected data was stored on CD-ROM in the "Downhole Dynamometer Database" and a program, DownDyn (2), provided by SANDIA was used in extracting the data. The particular set of data referenced in this example comes from the test on 08/04/96 at 10:00:00 AM from Tool #2 approximately 2 feet above the pump at a depth of 4993 feet. The collected data is displayed in both True and Effective load for the dynamometer cards as shown in figure 3.

Figure 3 displays the same dynamometer trace for the effective load and the true load for a 0.75-inch diameter rod at the pump depth. The effective load is equal to the force applied to the rods by the pump. Both the fluid load, Fo, of 1833 pounds applied to the rods by the pump and the area of the pump dynamometer card, 4.9 Hp. of work the pump provides lift the fluid, are same for true or effective load. The true load is dependent on the diameter of the rod carrying the load, as shown in Table 1 the larger the diameter the more negative the load.

The following equation (6) is used determine the true vs. effective load relationship, assuming the rod size is continuous all the way to the surface. The calculations show how the true load for the pump card would be shifted based on a rod size of 0.75 or 1.5 inch diameter from the surface to the pump.

True Load = Effective Load – Hydrostatic Pressure * Area Rod
True Max Load (0.75) = 2215 – 1584 * 0.4418 = 1515 Lbs.

True Min Load (0.75) = 18 – 1608 * 0.4418 = -692 Lbs.
True Max Load (1.50) = 2215 – 1584 * 1.7671 = -584 Lbs.
True Min Load (1.50) = 18 – 1608 * 1.7671 = -2823 Lbs.

Negative Pump Card Load:
Presenting pump cards with large negative loads has generated much confusion in the oil industry. Sinkerbar are designed to over-come compressive loads applied to the rod string by the pump. The pump card's downward shift is caused the display of negative loads on the pump card only due to true stress.

Figure 4 shows that the dynamometer card @ 4993' from the DHLC versus the calculated card agrees closely. The fluid load on the plunger is approximately 2000 lbs. and the area of the card is 4.9 horsepower. The internally developed wave equation does a good of modeling rodstring dynamics. The DHLC displays a large negative compressive load exerted at bottom of rodstring this is based on a 0.75-inch diameter DHLC. This negative 700 Lb. force on the DHLC card exist at bottom of rods at a 4993' depth due to subtracting off the buoyancy force in calculating the loads as calculated in the following example.

Presentation of the DHLC downhole pump card has changed over time. The original data was presented in effective load. The second sets of data was presented in terms of true load, but every rod taper was treated a 0.785 inch diameter. The last presentation shows the downhole dynamometer with large negative loads on the pump card, true stress based on actual rod diameters. This change in presentation of the same data has resulted in additional confusion.   To correct this SANDIA has modified their DownDyn program to display effective stresses by default and only display of true stress if the user enters the rod diameter.

Please refer to figure 5 for the following Calculations:

<u>Standard API RP 11L Equation for Calculating Wrf by Displaced Volume</u>
Wrf = Wra – Fbuoy = Wra(1 – 0.128G)
Fbuoy = Wra * 0.128 * SG
Density Ratio (Water/Steel) = 62.4/487.5 = 0.128
Volume(Rod) = Wra/487.5
Fbouy = Volume(Rod) * 62.4 * SG

<u>Buoyancy Force calculated by Pressure Equilibrium, Figure 1</u>
P1 = Pressure
P2 = P1 + Length * Gradient
Fbuoy = (P2 -P1) * Area = (P1 + Length*Gradient - P1) * Area
Fbuoy = Length*Area*Gradient

<u>Is the Buoyancy Force equal by Displaced Volume and by Pressure Equilibrium.</u>
Fbouy = Volume * 62.4 * SG
If AREA is in units of Inches, then: Fbouy = (Volume/144) * 62.4 * SG
If Gradient is in units of Psi/Ft, then: Gradient = 62.4/144*Sg Fluid

Volume = Length * Area
Fbouy = Length * Area * Gradient

The Buoyancy Force calculated by Pressure Equilibrium, true stress, and Displaced Volume, effective stress, are exactly the same force.

Buoyancy Force calculated by Displaced Volume
Fbouy = Volume * 62.4 * Sg Fluid = 1226
 Buoyancy Force calculated by Pressure Equilibrium used in DHLC
Fbuoy = Area*Pressure

DHLC Measured Loads were adjusted by Fbouy
Fbouy   = .4418 Sq In * 1600 psi = 707 lbs.

Pressure = Depth * Gradient   = 4993 ft * 0.335 psi/ft = 1673 psi

The Buoyancy force subtracted from the downhole loads is equal to a 4993-ft rod submerged in a 0.335 gradient fluid.   In comparison of DHLC Vs diagnostic loads the dynamometer card @ 4993' from the DHLC versus the calculated card agrees closely.   This negative 700 Lb. force exist at bottom of rods at a 4993' depth due to subtracting off an arbitrary buoyancy force in calculating the loads.

DHLC Calibration Procedure:
Load - The load constants are measured by two separate calibration procedures. The first is to determine the tools' response to a pure axial load in the absence of external pressure variation. This is done usually with two tools at once, arranged in series with the calibration gage, a Transducer Techniques load cell. The tools under calibration and a calibration set of electronics to which the calibration load cell is attached are all programmed with the same sampling schedule and running synchronously during the calibration. Once the schedule is in the actively sampling period the load stimulus is applied, via a hydraulic cylinder, usually through a range of zero to over 12,000 lbs. This data is combined in a spreadsheet and a linear regression is performed to produce a slope and intercept for each tool that relates directly to applied loads.

Through repeated calibration it has been observed that the intercept can vary with application of the end cap on the tool but remains constant on any given application. For this reason the intercepts used to reduce the data from a given test can be adjusted to match what a tool is reading after it has been buttoned up or after it is installed into the rod string as this too can sometimes introduce a small offset.

Pressure - The second step is to determine the tools' response to external pressures. This is necessary because, the strain gages are mounted on the inside wall of the tool body and external pressures cause strain in the tool walls. This pressure-induced strain needs to be rejected from the field data to yield a pure measure of axial load. This calibration is done in a pressure vessel, usually three tools at a time, with the pressure stimulus supplied by a hydraulic pump, and measured by an external pressure sensor attached to the calibration electronics. The pressure sensors in the tools are also conveniently calibrated at this time.

Thus the load equations become
TrueDynamometerLoad =
= {RawSignal - Signal@ZeroLoad&ZeroPressure - (UnitSignal/psi)*Pressure} *(lbf/UnitSignal) -
Pressure*AreaDynamometer
EffectiveDynamometerLoad =
= {RawSignal - Signal@ZeroLoad&ZeroPressure - (UnitSignal/psi)*Pressure} *(lbf/UnitSignal)

The true load Vs. effective load relationship is:
TrueDynamometerLoad = EffectiveDynamometerLoad - Pressure*AreaDynamometer

In actual application of the DHLC to the well the term AreaDynamometer was set to a 0.75-inch diameter tool all for any depth the tool was placed in the well. Once the pressure and load measurements were made with the DHLC the data was adjusted so it would display properly. In general, the first of these formulas is what was used to calculate the loads of an individual strain gauge in the Downhole Dynamometer Database. That is, the reported loads are not loads in the dynamometer, they are true loads in "the" sucker rod connected to the dynamometer. The area used was the "nominal" area of the sucker rods for each well. The one exception was dynamometers below the pump. For these dynamometers, an area of zero was used, i.e. below the pump the load reported is the effective load, this is probably to avoid the display of a large negative load outside the pump.

Recommendation/Conclusions:
1. Display of the downhole dynamometer pump card should be such that the pump card may be easily used to identify buckling tendencies of the rods: a negative load would indicate a compressive load on the rods from the pump and would result in buckling of the rod string.
2. Presenting pump cards with large negative loads has generated much confusion in the oil industry and has resulted in designing sinker bars to overcome a buoyancy force that doe not cause rod buckling.
3. Calibration of the DHLC resulted in an accurate load measuring Dynamometer.
4. Correlation of large lateral accelerations with negative effective loads could be used to indicate buckling of the rod string has occurred.
5. SANDIA has modified the DownDyn program to display effective stresses at any rod taper depth by default. Only display of true stress is done if the rod diameter is entered this change will result in a consistent presentation of the data.
6. Buoyancy force does not cause the rods to buckle.

References:
1. Albert, G.: "Downhole Dynagraph Measurement" A Thesis for MS at University of Oklahoma, 1986
2. Waggoner, John and Mansure, A.J.: "Development of the Downhole Dynamometer Database" SPE 37500 Production Operations Symposium, March 9-11, 1997 Oklahoma City, Oklahoma.
3. Hoffman, Edward: "Finite Analysis of sucker Rod Couplings with Guidelines for Improving Fatigue Life" (http://... apt.rodania 50 ... pdf).
4. Long, Scott W. and Bennett, Donald W.: "Euler Loads and Measured Sucker Rod / Sinkerbar Buckling" SPE 35124, presented at 42nd Annual Southwest Petroleum Short Course, April 19-20 at Texas Tech University, in Lubbock, Texas.
5. Sosa, Roberto L.: "Review of Downhole Dynamometer Testing", presented at 43rd Annual Southwest Petroleum Short Course, March 17-18 at Texas Tech University, in Lubbock, Texas.
6. Lea, J.F., Pattillo, P.D. and Studenmund, W.R.: "Interpretation of Calculated Forces on Sucker Rods" SPE 25416 Production Operations Symposium, March 21-23, 1993 Oklahoma City, Oklahoma.
7. Hein, N.W. Jr. and Hermanson D.E.: "A New Look at Sucker Rod Fatigue Life" SPE 26558, 68th Annual Technical Conference, Houston, TX October 3-6, 1993.

Table 1

Length (Ft)	Rod Dia. (In)	Rod Area (Sq In)	Hydro. Press. (Psi)	Wra (Lbs)	Fbouy (Lbs)	Eff. Rod Area (Sq In)
1512	1.000	0.7854	506	4390	-444	
1602	0.875	0.6013	1043	3562	-360	0.1841
1627	0.750	0.4418	1588	2658	-268	0.1595
252	1.500	1.7670	1672	1514	-153	-1.3252
4993				12124	-1225	1.7670

Table 2 - True vs. Effective Load Comparison

Description	Rod Diameter	Max Load	Min Load
Effective Load	N/A	2215	18
True Load for	0.750	1.515	-692
True Load for	0.875	1262	-949
True Load for	1.000	971	-1245
True Load for	1.500	-584	-2823
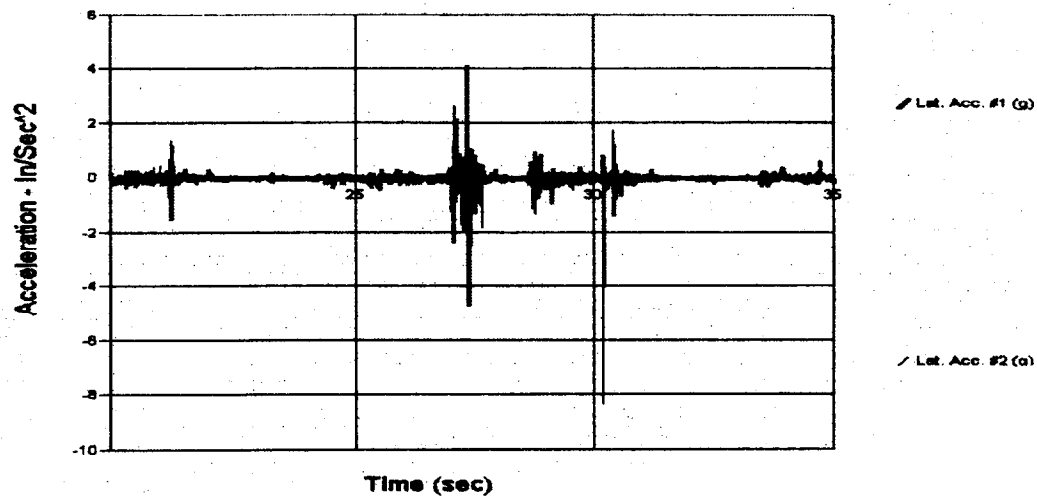
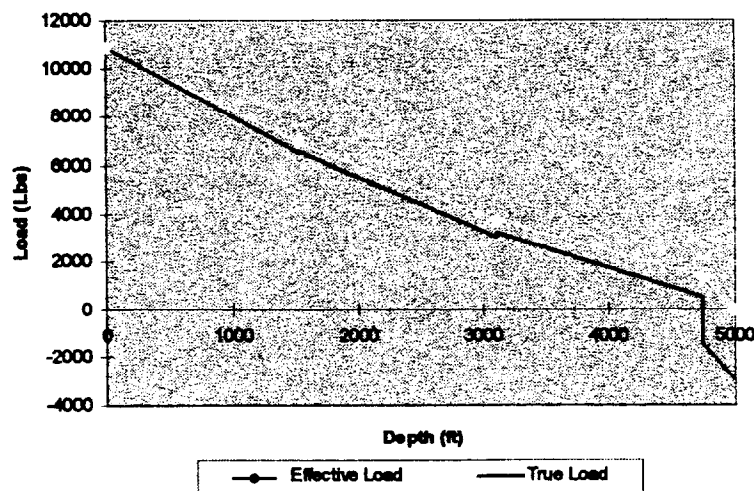Figure 1 - Lateral Acc. @ Pump vs. Time (2x5i03.csv)

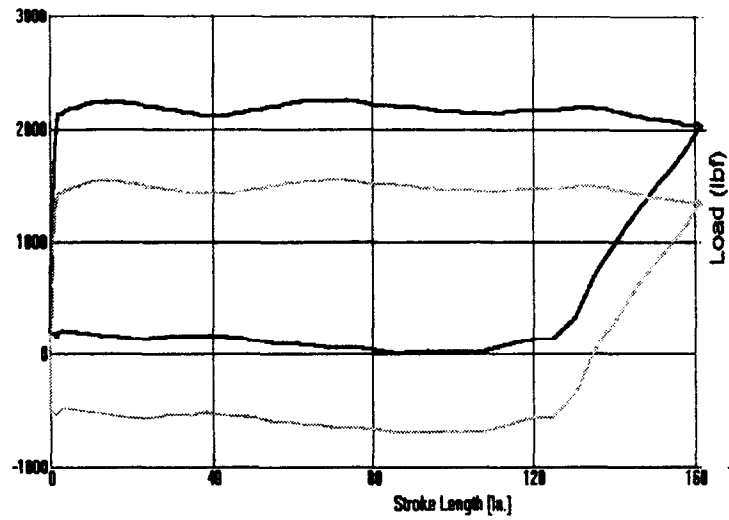Figure 2 - True vs. Effective Load

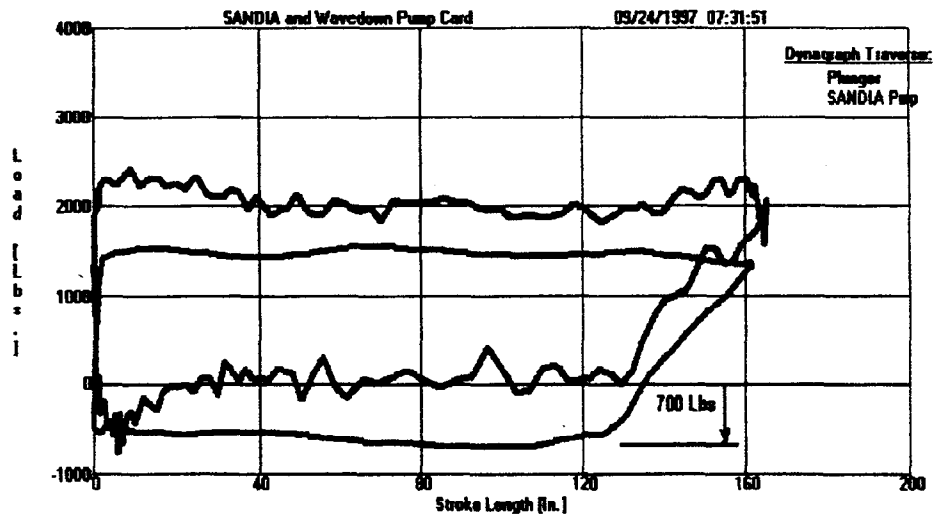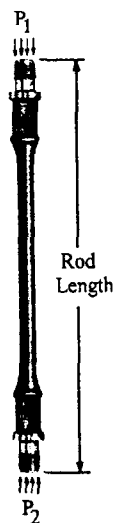Figure 3 - True vs. Effective Load from DHLC (205i03.csv) File (Red - 0.75 Dia. Rod)

Figure 4

Figure 5