HIGH CYCLE LIFE FIBERGLASS - SINKERBAR DESIGNS

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SCOPE OF THE PAPER

The purpose of this paper is to document 8 years of performance of an improved artificial lift system installed in a secondary recovery project in Alberta Canada. There are estimated dates, production rates, and other information that must be confirmed by field research.

This improved artificial lift system matches a Fiberglass Sucker Rod and Steel Sinkerbar rodstring design to a specific pumping unit, installed with a Pump-off Controller and communicating to a Scada System. Each well's performance is closely monitored by field personnel and changes to the artificial system are constantly being made as well conditions change. A Failure Analysis Team reviews each failure and procedural or designs changes are implemented.

The most recent use of Fiberglass Sucker Rods and Steel Sinkerbars rodstring designs in artificial lift operations has been driven primarily by requirements for increased lift capacity.

In the past years, when an operator was experiencing a high frequency of downhole failures, little consideration was given to utilizing a Fiberglass Sucker Rod and Steel Sinkerbar rodstring design to both increase lift capacity and reduce downhole rod and tubing failures.

In addition, little or no consideration was given to power consumption or power costs when selecting a rod string design or an overall rod pumping system. The success of this improved artificial system has led to the following field performance improvements;

1. 285 % Increase in lift capacity from 260 BFPD (41 cmpd) to 1,000 BPFD (159 cmpd)
2. 95 % Reduction in downhole failures from 3.8 FPWPY to 0.2 FPWPY in 3 years
3. Runtimes without a failure in excess of 1,300 days (3.6 years), now determined by pump repairs
4. 10-20 % Reduction in power consumption
5. Typical power costs reductions from .005 cents / barrel / 1000 feet of lift / day

Utilization of this improved artificial lift system will increase lift capacity, reduce downhole failures, increase runtimes and reduce power consumption and related expenses.

THE EVOLUTION AND HISTORY OF RODSTRING DESIGN

The American Petroleum Association (API) in 1969 developed a sucker rod pumping system design book. The first edition of this book was published in May of 1970 as an API bulletin titled “Sucker Rod Pumping System Design Book” (API 11 Bul 11L3).

This “Sucker Rod Pumping Design Book” became the design standard for the Oil and Gas Industry. The recommended design practice for calculating sucker rod pumping systems was to refer to the Sucker Rod Pumping System Design Tables in the “Sucker Rod Pumping System Design Book”. These tables were constructed for a specific API taper, pump depth and production rate. These tables provided the required pump diameter, stroke length and strokes per minute. These tables also provided estimates of rodstring loading, rodstring stress, peak gearbox torque, peak polished rod horsepower, counterbalance effect, unanchored production and weight of rods in fluid.

These Sucker Rod Pumping System Design Tables showed a fixed production capacity based on 24 hour per day operations. To change this capacity required a change in strokes per minute, downhole pump size or length of surface stroke. The result of utilizing these designs from these tables were wells operating with either a high fluid level or incomplete fillage resulting in a fluid pound due to the inability, at that time to adjust runtimes from 24 hours per day.

The introduction of time clocks provided field operators the opportunity to begin to match system lift capacity to fluid inflow rates by adjusting the operational run time. This method was an improvement, but required trained operators to manually adjust run times.

Time clocks later evolved into automated pump-off controllers that matched production lift capacity to reservoir inflow rates by automatically varying the run time. Pump off controllers also protected surface and downhole equipment from overloads not able to be detected by time clocks.

The results of matching rod pumping lift capacity to reservoir inflow characteristics while minimizing equipment loading was and still is the prime reason behind reduced failure frequency and improved system efficiency for an artificial lift system.

Fiberglass Sucker rods were introduced in the Petroleum Industry in the mid-1970's to reduce corrosion related steel rodstring failures in beam pumping systems. Greater lift capacity for a given pumping unit was also realized due to the reduced weight and increased downhole stroke due to the elasticity of the Fiberglass Sucker Rods.

The benefits of these early Fiberglass Sucker Rod designs were limited to deeper applications because of the large amounts of steel sucker rods required to maintain tension in the Fiberglass taper. In shallow to intermediate depth applications, this necessary weight provided by the large footage of steel rods reduced the amount of Fiberglass that could be utilized in the rodstring designs. Consequently the associated benefits from the elasticity of the Fiberglass taper were limited due to the shortened Fiberglass taper.

The steel shortage of early 1980's reduced the availability of steel rods to the Oil and Gas Industry. This shortage of steel rods impacted the use of these steel rods for effective weight in designs using Fiberglass Rods and Steel Rods.

Sinkerbars were introduced during this time to take the place of these high demand steel rods on the bottom of Fiberglass Rod and Steel Rod rodstring designs. The use of Sinkerbars provided the weight required to protect the Fiberglass and Steel tapers from buckling due to downstroke compression. The Sinkerbars also provided a larger diameter that was less able to buckle under the same downstroke compressive force.

From 1983 through 1996, the Petroleum Industry experienced an increased awareness and understanding of sucker rod compression. Developers of downhole predictive programs offered improved output including bottom minimum stress readings at the bottom of all tapers and the ability to adjust pump friction. This attention by the Industry focused on the difference between true or effective compressive loads, the measurement of these loads, the compressive loads required to buckle sucker rods and Sinkerbars, tubing wall lose resulting from rod on tubing wear, Sinkerbar on tubing wear and determining total downstroke friction. The result was more industry wide use of Sinkerbars in all steel rodstring designs and fiberglass and steel rodstring designs. The result was a reduction in downhole compression related failures.

Today, the Petroleum Industry is searching for improved rodstring designs that provide matched lift capacity to various inflows with the lowest failure frequency at the lowest lifting cost.

DISCUSSION OF POWER COSTS

The evolution of power costs for rod pumping wells has advanced from rental charges for generator sets to paying a fixed monthly fee for installed horsepower to the current method of metered service which includes a charge for peak demand and a separate charge for kilowatt-hour usage.

Power costs are usually the second highest cost behind downhole failures. The next evolution of Rod Pumping Design must vary unit and rodstring selection to lower the poser cost per barrel of fluid.

DISCUSSION OF INSTALLED HORSEPOWER

When power costs are based on installed power, the size of the prime mover will control the power bill. Motors are usually triple rated such as a 30-40-50 hp. Depending on the way the motor is wired to the supply voltage, the motor will develop a different full load horsepower.

No credit is given for regenerative power or unused capacity. When the well is down the power bill continues. Time clocks or pump-off controllers will not reduce the power bill. The pumping system must constantly change with the change of the well inflow. Cyclic loading and regenerative power may cause the prime mover horsepower to be twice the value of the polished rod horsepower. Pumping systems that reduce or eliminate regenerative power will have reduced power bills.

DISCUSSION OF METERED SERVICE

With metered service, there are usually two parts to the power bill.

The demand charge is based on the peak demand for any 15-minute period during the billing cycle. There may be a minimum demand charge for supplying power to a location. The following field practices will affect the demand charge;

1. Unit balance and rotation
2. Abnormal conditions such as wax
3. Stuffing box friction, tubing pressure due to back pressure valves and flow line pressure

The usage portion of the bill is based on kilowatt-hours or KWH. KWH is the actual power used by the pumping system.

DISCUSSION OF REGENERATIVE POWER

Regenerative power is created when the pumping unit and well load drives the motor above the no load speed. No credit is given for this power by the meter and must be offset by higher power consumption during the remainder of the pumping cycle. To reduce the power bill when metered service is the billing regime be very careful of the demand charge. There are two ways of reducing demand charges, design considerations and field practices.

Design factors affect the demand charges for the life of the well. The cyclic nature of rod pumping causes spikes in demand during the pumping stroke. The pumping unit, rod string, downhole pump size, and the combination of surface stroke length and strokes per minute cause this cyclic loading. The magnitude of this cyclic loading affects the efficiency of the pumping system. It is measured by the cyclic load factor which is the root mean squared of values divided by there average. The cyclic load factor or CLF should be as low as possible.

DISCUSSION OF DOWNHOLE PUMP CARDS

The downhole pump card is a graphic representation of the horsepower required at the pump to lift fluid to the surface. Factors that impact the horsepower required at the pump are as follows;

1. Specific gravity of produced fluids including all gas in the produced fluids
2. Path of the well bore, deviated or straight
3. Tubing pressure at the wellhead
4. Casing pressure
5. Flow line pressure
6. Pump depth
7. Fluid level over the pump
8. Pump size

Downhole pump horsepower or the area bounded by the pump card will peak just before pump-off or just before the start of incomplete pump fillage. The ideal situation would be if the beam lift system would operate continuously at this peak required horsepower and automatically adjust production capacity to match the inflow from the reservoir. This optimum situation would yield maximum production, excellent equipment life, and the potential for peak system efficiency if properly designed.

DISCUSSION OF LIFT CAPACITY

Downhole lift capacity is defined as; Downhole Lift Capacity = SPM x PC x DHPS x Percent Run Time

SPM – strokes per minute during the run time of the pumping unit
PC – pump constant for the downhole pump determined by the plunger diameter
DHPS – downhole pump stroke
Percent Run Time – run time for the pumping unit as a percentage of 24 hours per day

The downhole lift capacity of the beam lift system can be adjusted by changing any one of these variables.

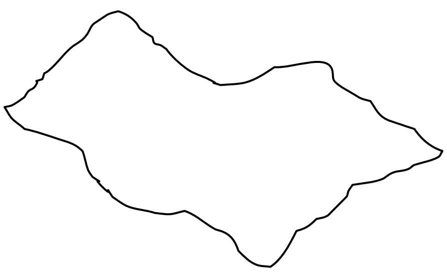
DISCUSSION OF SURFACE DYNAMOMETER CARD

The surface dynamometer card is created by load measurements obtained from a load cell and associated polished rod position measurements. The measured loads are a result downhole pump forces transmitted to the load cell at surface through the rod string and the weight of the rod string. The shape of the surface card can be affected by the following;

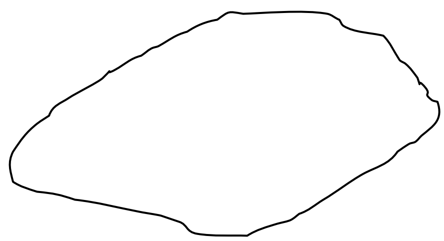
1. Physical characteristics of the rod string (steel or fiberglass)
2. Pumping Unit surface geometry
3. Strokes per Minute
4. Length of Stroke
5. Downhole Friction
4. Prime mover speed torque curves

For a given downhole card, there are an infinite number of available surface cards. The surface card could be manipulated by changing the components of the lift system to obtain some theoretical ideal shape. This ideal shape will result in the lowest rod loads and the best use of the surface equipment and the best system efficiency.

Below are examples of cards modified by changing only the rod material.



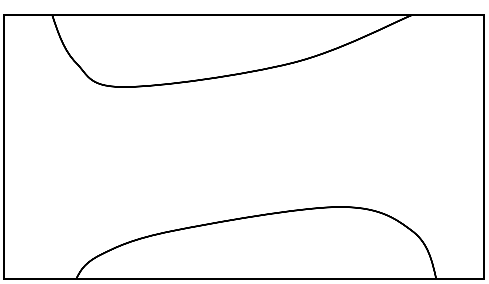
Steel Rods



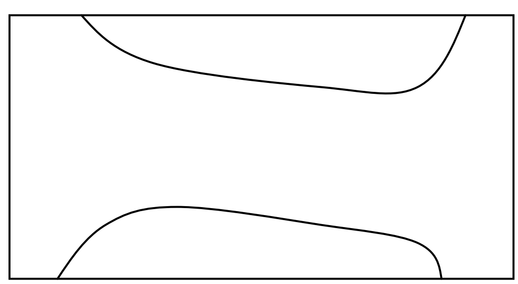
Glass Rod

DISCUSSION OF PERMISSIBLE LOAD DIAGRAM

Any surface pumping unit with a given counterbalance moment will develop a permissible load for a given position of the polished rod. These loads and positions when plotted are a graphical representation of the available horsepower for that particular beam lift system.



Conventional - Permissible Load Diagram



Mark II - Permissible Load Diagram

DISCUSSION OF COMBINING SURFACE DYNAMOMETER CARD INTO PERMISSIBLE LOAD DIAGRAM

If maximum production is required from a given pumping unit, the Surface Card must fill the maximum area provided by the Permissible Load Diagram. The surface Dynamometer Card must fit inside the Permissible Load Diagram to avoid the overload of surface equipment.

The goal of a best design practice must be to obtain maximum production without overloads, while achieving long equipment lift along with a low lifting cost per barrel of fluid.

When designing an artificial lift system certain variables become fixed for system. The most common fixed variable is the maximum tubing diameter and the pumping unit which is not easily exchanged.

The following steps should be followed in order.

1. Determine the work to done by the lift system
2. Determine the fixed components of the lift system.
3. Vary the remaining components and optimize the system

Lift systems can be optimized several ways.

1. Rod loading
2. Surface equipment loading
3. Power consumption
4. Others such as installed horsepower
5. Production increase.

HISTORY OF THIS PAPER

In 1995, Gulf Canada, which became ConocoPhillips began evaluating producing fields near Red Earth Creek in Alberta Canada for enhanced production and a reduction in lifting costs. These fields included; Trout, House Creek, Kidney Panny and the Sennex Fields.

These fields were producing under primary recovery and were averaging one (1) downhole failure every 96 days of operation. This failure rate was equal to a tubing leak, rod part or pump repair failure every 3.2 months. This was equivalent to a failure frequency of 3.75 failures per well per year (FPWPY).

This combination of this high failure rates and high repair costs for tubing leaks, rod parts and pump repairs in a remote field resulted in an examination of all artificial lift operations. Many of these wells were producing with high fluid levels and could not be pumped off with existing lift equipment. It was determined, enhanced recovery (waterflood operation) would not economically feasible without a reduction in the frequency of downhole failures. Downhole repair costs averaged in excess of $20,000 Canadian dollars per downhole failure.

The result was the need to challenge the standard assumptions concerning artificial list design.

All of the existing API 86 steel rodstring designs were reviewed to match lift capacity to current and future reservoir inflow characteristics. Results of this review are listed below;

1.	Pump Depths;	4,600' to 4,900'
2.	Initial Production Rates:	250 BFPD
3.	Final Production Rates:	650 BFPD
4.	Surface Equipment:	C320-256-120 Pumping Units
5.	Downhole Equipment:	API 86 Steel Rod Strings
6.	Rod Pumps:	1.5", 1.75", 2.0" and 2.25" Plunger Diameter

This rodstring design review recommended conversion to a Fiberglass Sucker Rod and Steel Sinkerbar rodstring design to better match reservoir inflow characteristics to the lift capacity of existing surface pumping units.

Following the rodstring design review project goals were discussed and determined as follows;

1. Increase Current Mean Time to Failure
2. Maintain Wells in a Pumped-Off Condition
3. Utilize Existing Surface Lift Equipment
4. Reduce Power Consumption and Power Cost

Following the establishment of project goals, methods to achieve these goals were discussed and determined as follows;

1. Modify shape of surface cards to fit permissible load diagram of existing surface equipment
2. Install diagnostic pump-off controllers with a scada system
3. Create and support a failure analysis and review team (FART)

STEPS USED TO MODIFY THE SHAPE OF CURRENT SURFACE CARDS

A Fiberglass-Sinkerbar rod string design replaced the “standard” API 86 steel taper rodstring design.
Increase strokes per minute
Reduced the length of the surface stroke

RESULTS OF MODIFYING THE SHAPE OF CURRENT SURFACE CARDS

Reduced Gearbox Torque
Extended Fiberglass and Steel rod life
Reduced Rod Loadings
Reduced power cost / barrel / foot of lift
Increase in polished rod horsepower
Reduction in rod load range
Increase in production without an increase in gearbox torque

STEP ONE

Utilizing Existing Surface Equipment to Lift Existing 260 BFPD and Reduce Downhole Failures

Fiberglass Sucker Rod and Steel Sinkerbar rodstring designs were installed in approximately 300 wells utilizing pump-off controllers communicating to a scada system with the initiation of a waterflood project. The following is performance of Step One Fiberglass-Designs to the existing API 86 Steel Designs:

	Original Designs	**Step One Designs**	
Input Data	API 86 Steel	Fiberglass-Sinkerbar	
Rodstring Design:	1,200’ – 1.000” Steel 1,350’ – 0.875” Steel 2,125’ – 0.750” Steel	3,750’ – 1.230” Fiberglass 925’ – 1.625” Sinkerbars	
Pumping Unit Manufacturer:	Ampscot	Ampscot	
Pumping Unit:	C-320-256-100	C-320-256-100	
Strokes Per Minutes:	11.0 spm	10.0 spm	9.1 % Decrease
Surface Stroke:	93.0”	93.0”	No Change
Average Polished Rod Velocity;	2,046 ipm	1,860 ipm	9.1 % Decrease
Pump Diameter:	1.75”	1.75”	
Motor Horsepower:	40 hp	40 hp	No Change
Output Data			
80% Efficient Production:	260 BFPD	260 BFPD	Maintained
Gearbox Torque:	313,000 in-lbs	295,000 in-lbs	5.8 % Decrease
Downhole Stroke:	83.0”	89.0”	7.2 % Increase
PRHP / PLHP:	38 %	40 %	5.3 % Increase
Rod Loading:	1.000” – 72 % 0.875” – 72 % 0.750” – 72 %	1.230” – 42 % 1.625” – 37 %	

The success of these Step One Designs was a reduction of downhole failures along with maintaining existing production rates of 260 bfpd.

<u>The following graphs are recreations from actual surface cards and torque plots and are 95 % accurate</u>
Refer to Graph 1 and Graph 2 for a comparison of Original API 86 Steel Design with Ampscot C-320-256-100 at 11 Spm to Step One Design with Ampscot C-320-256-100 at 10 Spm Surface Cards

Refer to Graphs 3 and Graph 4 for a comparison of Original API 86 Steel Design with Ampscot C-320-256-100 at 11 Spm to Step One Design with Ampscot C-320-256-100 at 10 Spm Torque Plots

Step One Designs were successful in reducing downhole failures while maintaining existing production rates of 260 bfpd. The next goal was to increase production rates from 260 bfpd to 540 bfpd while beginning to lower fluid levels utilizing larger surface equipment.

STEP TWO

Utilizing Larger Surface Equipment to Lift 535 - 540 BFPD

Larger surface equipment was installed on wells with the original all steel 86 designs to increase lift capacity. Existing pumping units were maintained on wells installed with the improved Step One Designs. The following is performance of Step Two Designs with existing C-320-246-86 pumping units to the Original Designs with larger C-640-305-120 pumping units.

	Original Designs	Step Two Designs	
Input Data	API 86 Steel	Fiberglass-Sinkerbar	
Rodstring Design:	1,200' – 1.000" Steel	3,500' – 1.230" Fiberglass	
	1,350' – 0.875" Steel	1,175' – 1.625" Sinkerbars	
	2,125' – 0.750" Steel		
Pumping Unit Manufacturer:	Ampscot	Ampscot	
Pumping Unit:	C-640-305-120	C-320-246-86	
Strokes Per Minute:	12.5 spm	13.7 spm	9.6 % Increase
Surface Stroke:	120.2"	76.8"	36.1 % Decrease
Average Polished Rod Velocity;	3,005 ipm	2,104 ipm	30.0 % Decrease
Pump Diameter:	2.00"	2.25"	
Motor Horsepower:	75 hp	60 hp	20.0 % Decrease

Output Data			
80% Efficient Production:	535 BFPD	540 BFPD	0.1 % Increase
Gearbox Torque:	563,000 in-lbs	291,000 in-lbs	48.3 % Decrease
Downhole Stroke:	114.0"	84.0"	26.3 % Decrease
PRHP / PLHP:	38 %	55 %	44.7 % Increase
Rod Loading:	1.000" – 88 %	1.230" – 63 %	
	0.875" – 88 %	1.625" – 37 %	
	0.750" – 88 %		

The success of the Step Two Designs was that these designs did not require the large C-640-305-120 pumping units to lift 540 bfpd. The Step Two Designs only required smaller C-320-246-86 pumping units to lift 540 bfpd. This success resulted in improved utilization of existing pumping units and no need to purchase larger and more expansive pumping units.

<u>The following graphs are recreations from actual surface cards and torque plots and are 95 % accurate</u>

Refer to Graph 5 and Graph 6 for a comparison of Original API 86 Design with Ampscot C-640-305-120 at 12.5 Spm to Step Two Design with Ampscot C-320-246-86 at 13.7 Spm Surface Cards

Refer to Graph 7 and Graph 8 for a comparison of Original API 86 Design with Ampscot C-640-305-120 at 12.5 Spm to Step Two Design with Ampscot C-320-246-86 at 13.7 Spm Torque Plots

Step Two Designs were successful in increasing existing production rates from 260 bfpd to 540 bfpd while maintain reduced downhole failures. All of this was accomplished utilizing significantly smaller surface lift equipment.

STEP THREE

Designing to Lift 1,000 BFPD While Maintaining Reduced Downhole Failures

With the confidence gained from the Step One and Step Two Designs, Step Three Designs were utilized to lift 1,000 bfpd, with small pumping units while maintain reduced downhole failures.

	Step Two Designs	Step Three Designs	
Input Data	Fiberglass-Sinkerbar	Fiberglass-Sinkerbar	
Rodstring Design:	3,500' – 1.230" Fiberglass	3,500' – 1.230" Fiberglass	
	1,175' – 1.625" Sinkerbars	1,175' – 1.625" Sinkerbars	
Pumping Unit Manufacturer:	Ampscot	Ampscot	
Pumping Unit:	C-320-246-86	C-320-246-86	
Strokes Per Minute:	13.7 spm	17.5 spm	27.7 % Increase
Surface Stroke:	76.8"	76.8"	No Change
Average Polished Rod Velocity;	2,104 ipm	2,688 ipm	27.7 % Increase
Pump Diameter:	2.25"	2.25"	
Motor Horsepower:	60 hp	60 hp	No Change
Output Data			
80% Efficient Production:	540 BFPD	1,000 BFPD	85.2 % Increase
Gearbox Torque:	291,000 in-lbs	314,000 in-lbs	7.9 % Increase
Downhole Stroke:	84.0"	101.0"	20.2 % Reduction
PRHP / PLHP:	55 %	67 %	21.8 % Increase
Rod Loading:	1.230" – 63 %	1,230" – 67 %	
	1.625" – 37 %	1.625" – 40 %	

The following graphs are recreations from actual surface cards and torque plots and are 95 % accurate

Refer to Graph 9 and Graph 10 to Compare Step Two Design with Ampscot C-320-246-86 at 13.7 Spm to Step Three Design with Ampscot C-320-246-86 at 17.5 Spm Surface Cards.

Refer to Graph 11 and Graph 12 to Compare Step Two Design with Ampscot C-320-246-86 at 13.7 Spm to Step Three Design with Ampscot C-320-246-86 at 17.5 Spm Torque Plots.

Step Three Designs were successful in increasing existing production rates from 260 bfpd to 1,000 bfpd while maintain reduced downhole failures. All of this was accomplished utilizing significantly smaller surface lift equipment.

COMPARISON OF ORIGINAL DESIGNS TO STEP THREE DESIGNS

Listed below is a comparison of the Original Designs to the final Step Three Designs

	Original Designs	Step Three Designs	
Input Data	API 86 Steel	Fiberglass-Sinkerbar	
Rodstring Design:	1,200' – 1.000" Steel	3,500' – 1.230" Fiberglass	
	1,350' – 0.875" Steel	1,175' – 1.625" Sinkerbars	
	2,125' – 0.750" Steel		
Pumping Unit Manufacturer:	Ampscot	Ampscot	
Pumping Unit:	C-320-256-100	C-320-246-86	
Strokes Per Minutes:	11.0 spm	17.5 spm	59.1 % Increase
Surface Stroke:	93.0"	76.8"	17.4 % Decrease
Average Polished Rod Velocity;	2,046 ipm	2,688 ipm	31.4 % Increase
Pump Diameter:	1.75"	2.25"	
Motor Horsepower:	40 hp	60 hp	50.0 % Increase
Output Data			
80% Efficient Production:	260 BFPD	1,000 BFPD	284.6 % Increase
Gearbox Torque:	313,000 in-lbs	314,000 in-lbs	0.3 % Increase
Downhole Stroke:	83.0"	101.0"	21.7 % Increase
PRHP / PLHP:	38 %	67 %	76.3 % Increase
Rod Loading:	1.000" – 72 %	1.230" – 67 %	
	0.875" – 72 %	1.625" – 40 %	
	0.750" – 72 %		

<u>The following graphs are recreations from actual surface cards and torque plots and are 95 % accurate</u>
Refer to Graph 13 and Graph 14 to compare Original API 86 Steel Design with Ampscot C-320-256-100 at 11 Spm to Step Three Design with Ampscot C-320-246-86 at 17.3 Spm Surface Cards.

Refer to Graph 15 and Graph 16 to compare Original API 86 Steel Design with Ampscot C-320-256-100 at 11 Spm to Step Three Design with Ampscot C-320-246-86 at 17.3 Spm Torque Plots.

CONCLUSIONS

Installation of Fiberglass Sucker Rod and Steel Sinkerbar rodstring designs in approximately 75 wells utilizing pump-off controllers communicating to a scada system with the initiation of a waterflood project resulted in the following;

Fiberglass-Sinkerbar Design Project Performance

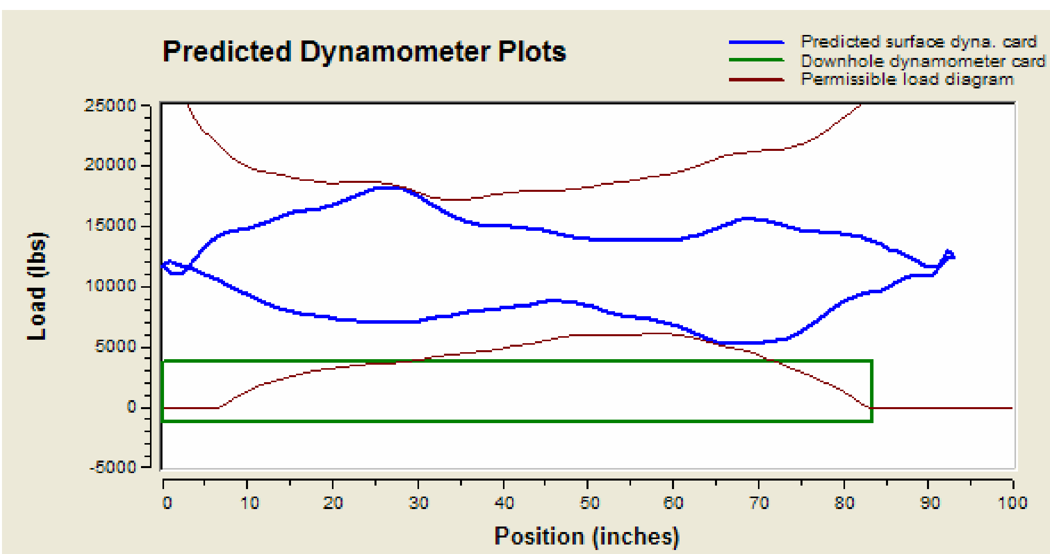
1. 285 % Increase in lift capacity with Existing Pumping Equipment
2. 95 % Reduction of downhole failures (FPWPY) over 3 years
3. 10-20 % Reduction of power consumption

Specific Performance of Fiberglass-Sinkerbar Designs

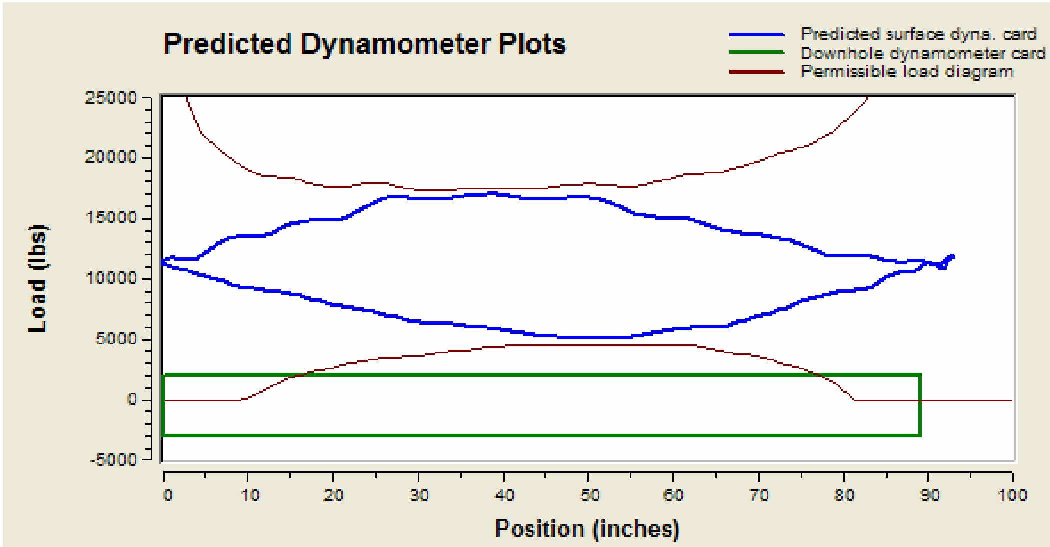
1. Increased Mean Time to Failure from 91 days in 1995 to greater than 1,300 days
2. Current Mean Time to Fiberglass Failure is due to pump repairs and is greater than 1,000 days
3. Average Fiberglass Cycle Life has increased; 4-8 million to an average of 30 million cycles
4. One Fiberglass-Sinkerbar design has operated in excess of 50 million cycles
5. A Total of 75 wells in 6 Fields are currently operating with a Fiberglass-Sinkerbar Design

REFERENCES

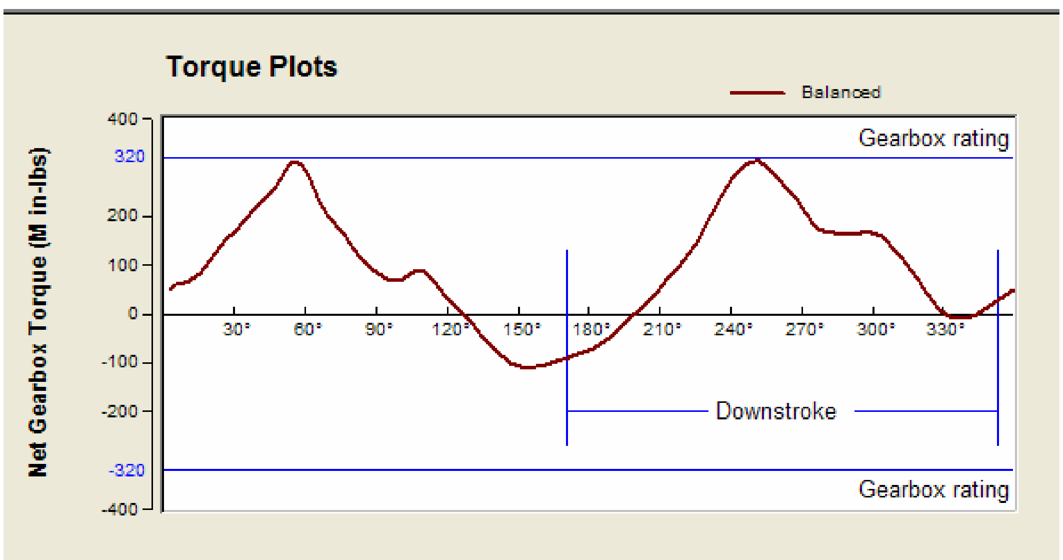
SPE #35214 – Euler Loads and Measured Sucker Rod / Sinkerbar Buckling
SPE #37502 – Measured Rodstring / Tubing Wear and Associated Side Loading
SPE #67274 – Total Downstroke Friction From Downhole Dynamometer Analysis
SPE #67270 – Best Practices in the Preston Spraberry Unit



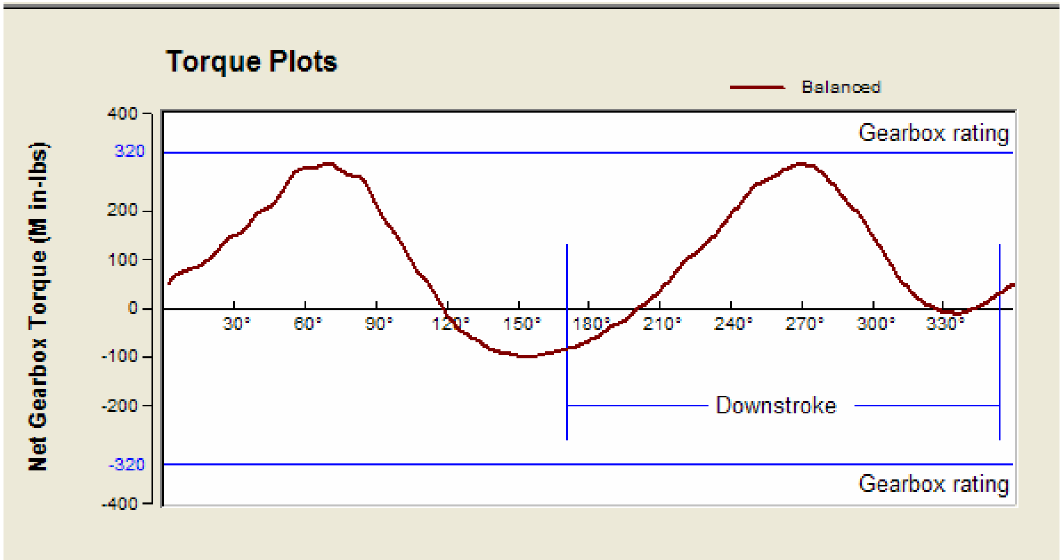
Graph 1 – Original API 86 Steel Design with Ampscot C-320-256-100 at 11 Spm – Surface Card



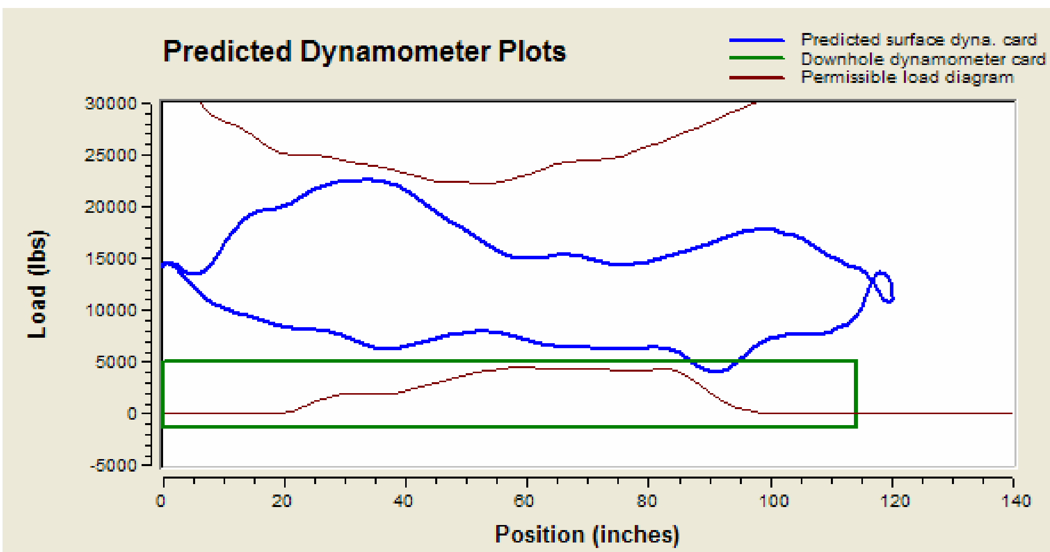
Graph 2 – Step One Design with Ampscot C-320-256-100 at 10 Spm – Surface Card



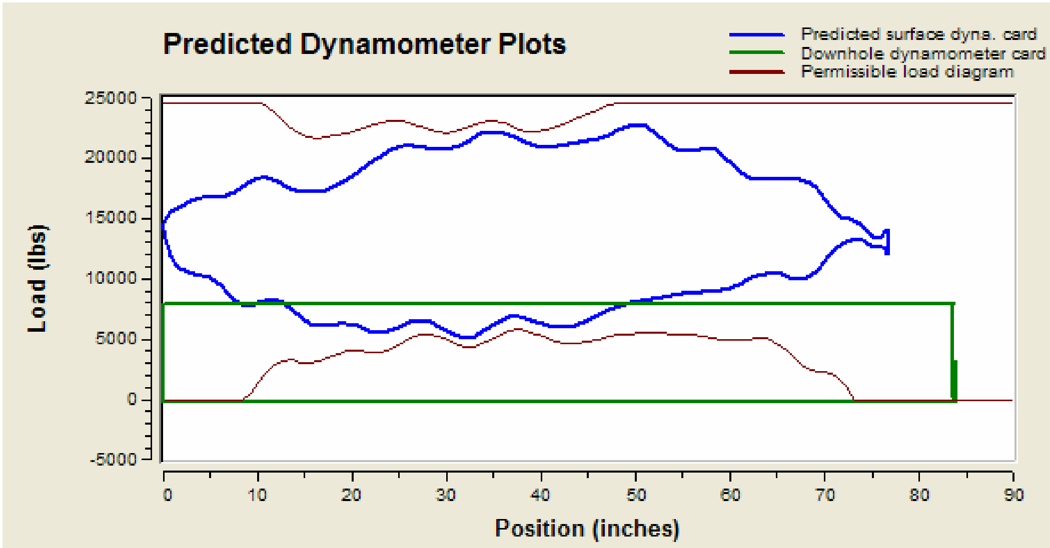
Graph 3 – Original API 86 Steel Design with Ampscot C-320-256-100 at 11 Spm – Torque Plot



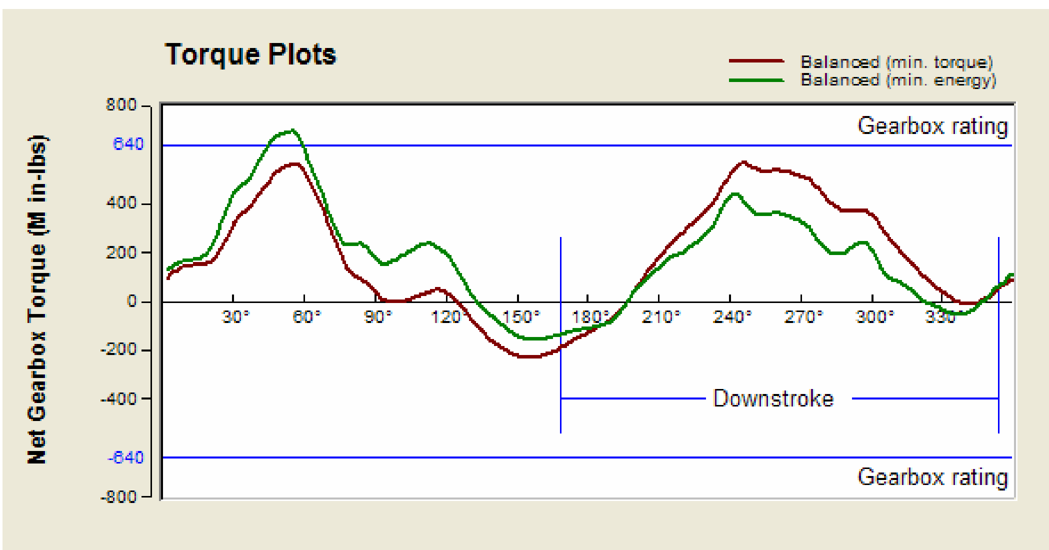
Graph 4 – Step One Design with Ampscot C-320-256-100 at 10 Spm – Torque Plot



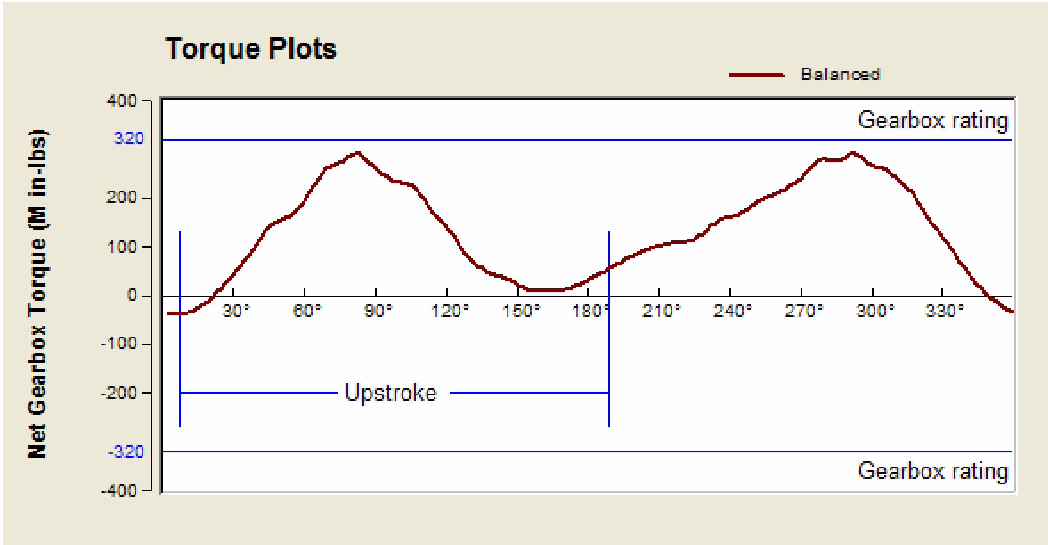
Graph 5 – Original API 86 Steel Design with Ampscot C-640-305-120 at 12.5 Spm - Surface Card



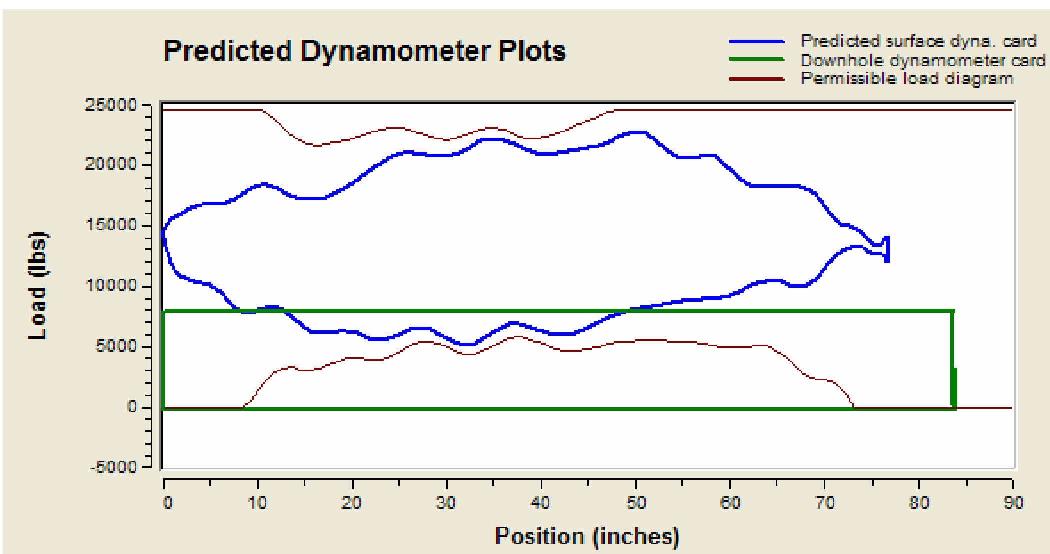
Graph 6 – Step Two Design with Ampscot C-320-246-86 at 13.7 Spm – Surface Card



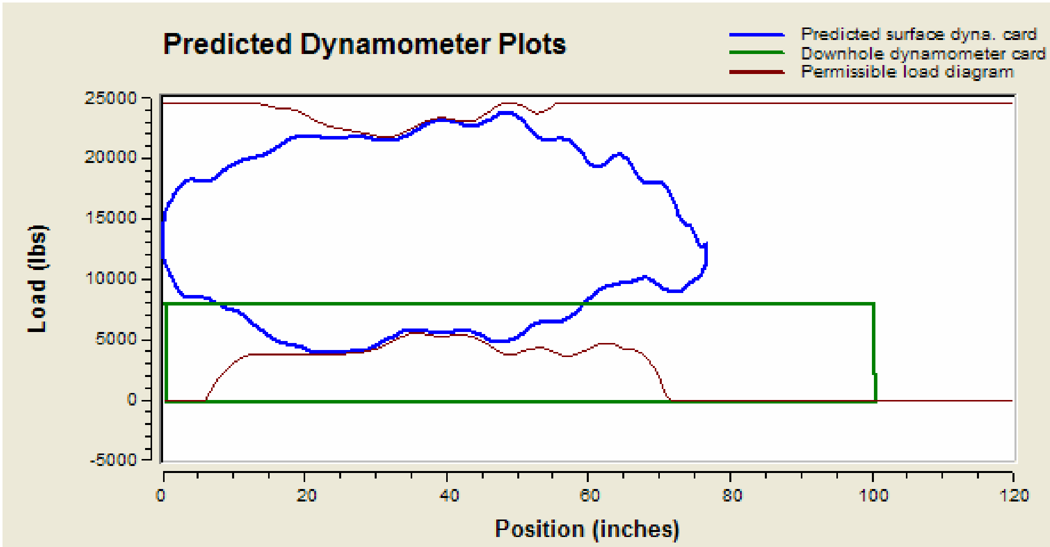
Graph 7 – Original API 86 Steel Design with Ampscot C-640-305-120 at 12.5 Spm – Torque Plot



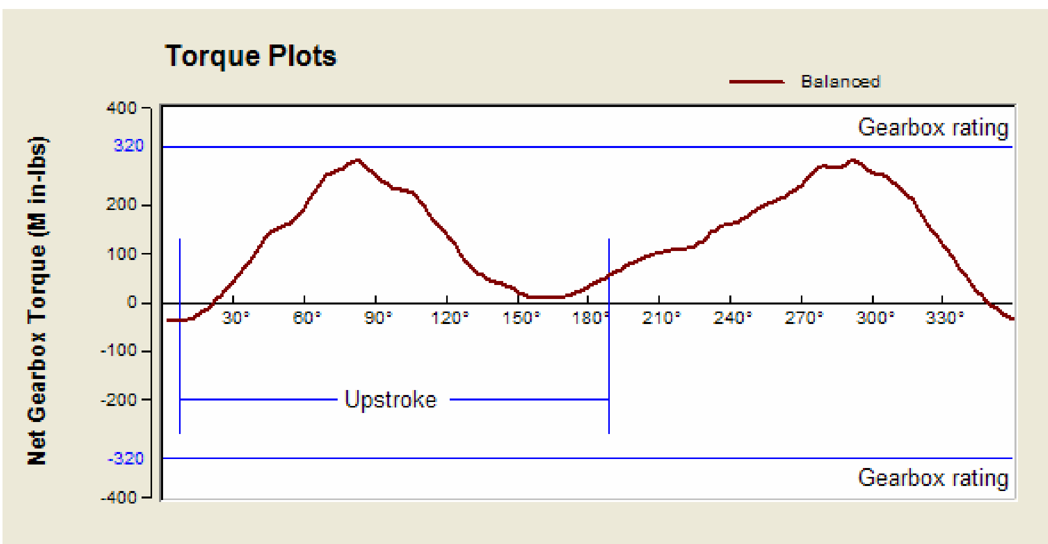
Graph 8 – Step Two Design with Ampscot C-320-246-86 at 13.7 Spm – Torque Plot



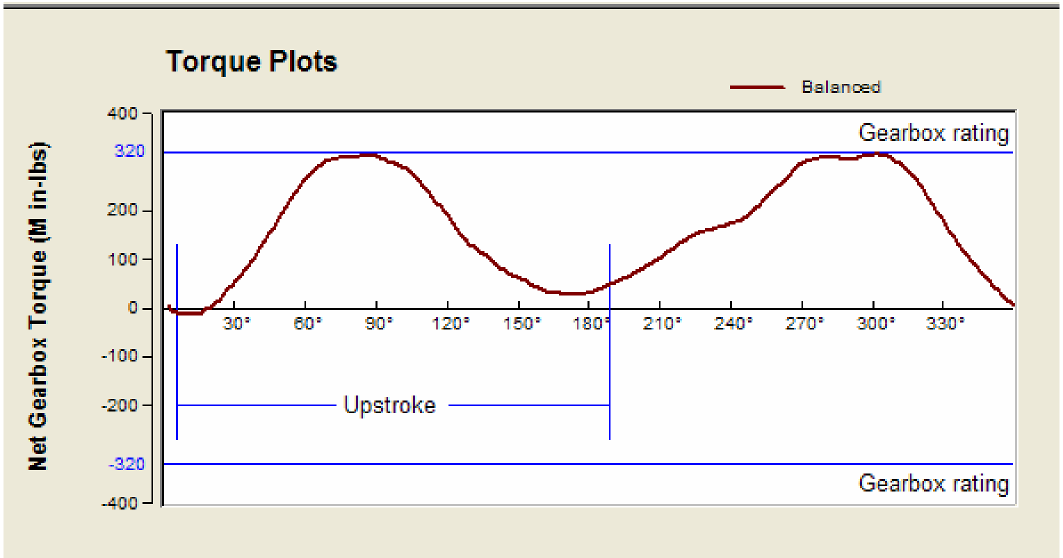
Graph 9 – Step Two Design with Ampscot C-320-246-86 at 13.7 Spm – Surface Card



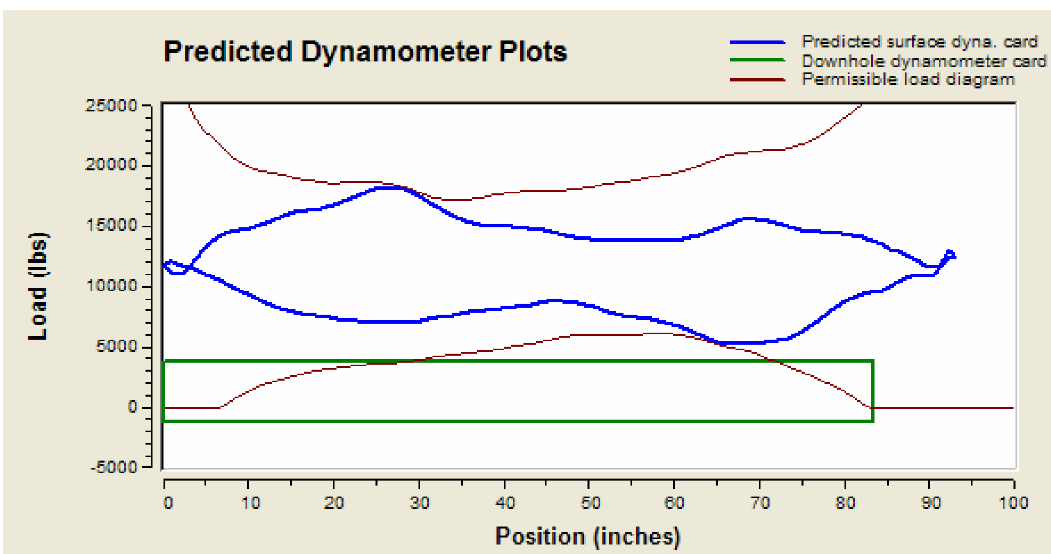
Graph 10 – Step Three Design with Ampscot C-320-246-86 at 17.5 Spm – Surface Card



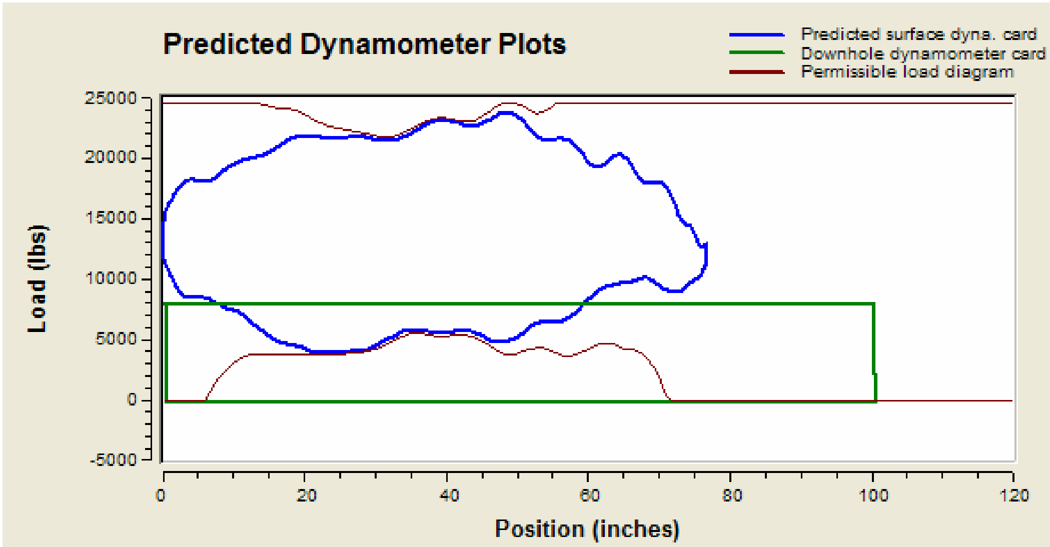
Graph 11 – Step Two Design with Ampscot C-320-246-86 at 13.7 Spm – Torque Plot



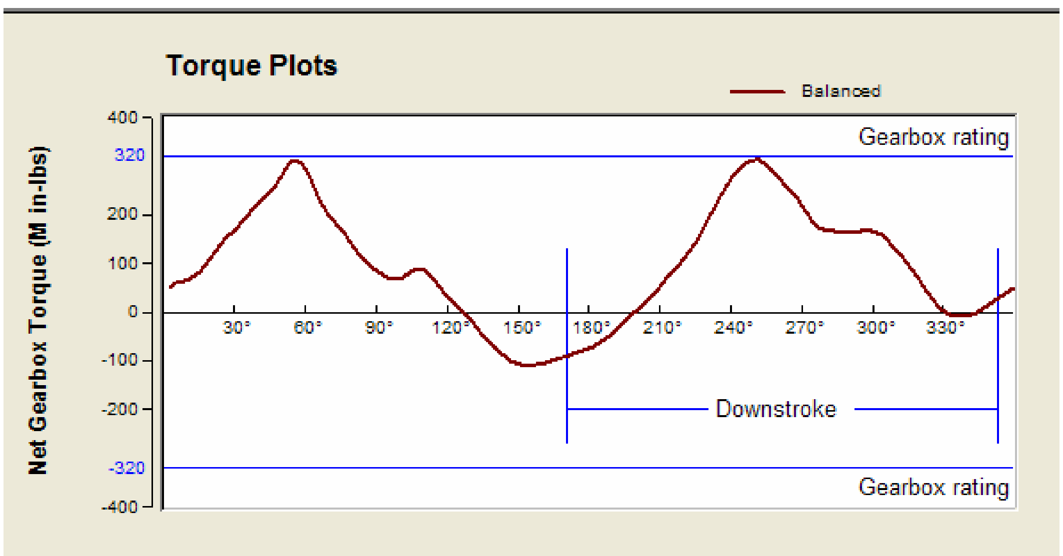
Graph 12 – Step Three Design with Ampscot C-320-246-86 at 17.5 Spm – Torque Plot



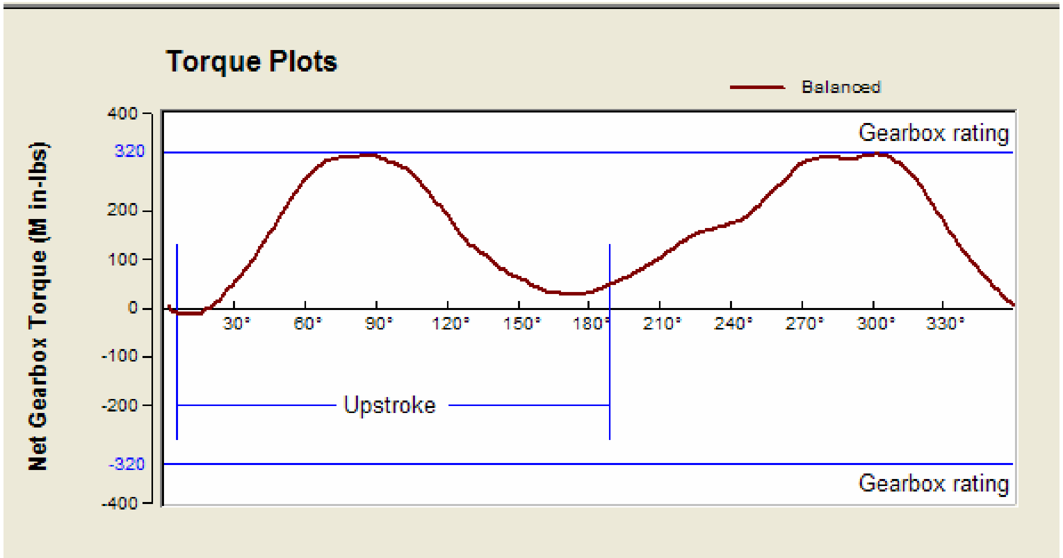
Graph 13 – Original API 86 Design with Ampscot C-320-256-100 at 11 Spm– Surface Card



Graph 14 – Step Three Design with Ampscot C-320-246-86 at 17.3 Spm – Surface Card



Graph 15 – Original API 86 Design with Ampscot C-320-256-100 at 11 Spm– Torque Card



Graph 16 – Step Three Design with Ampscot C-320-246-86 at 17.5 Spm – Torque Plot