PRACTICAL APPLICATIONS OF THE SUCKER ROD DIAGNOSTIC TECHNIQUE

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INTRODUCTION

As often discovered, determining downhole pump conditions by visual interpretation of a surface dynagraph card can be very difficult even for highly trained personnel. In addition, visual surface interpretations are more qualitative than quantitative. With the computerized method, surface measurements (load and displacement versus time) are used to calculate a downhole dynagraph card that is quantitative and much more easily interpreted. Basically, the computer program takes surface rod loads and displacements and removes rod weight, dynamic effects (harmonics) and rod stretch. The result is a pump card. Intermediate downhole cards are also calculated at critical stress points in the rod string such as at the junction points in a tapered rod design. Thus, rod taper designs can be easily evaluated.

Besides calculating downhole conditions, measured data are also used to analyze surface equipment loading such as gearbox torque, prime mover loading and structural loading.

All calculations can be made in a matter of minutes on the well site. Thus, conclusions can be drawn and changes can be initiated immediately for increasing production and/or reducing operating costs.

IMPLEMENTATION OF TECHNIQUE

Recent advances in computer technology and miniature circuitry have led to small and rugged machines. Thus, the portable digital computer unit was made practical. The advantages of performing an analysis at the well are threefold. First, the results are available immediately. Second, oil company engineers, foremen and lease operators can participate in the analysis by contributing their knowledge of well history and characteristics. And third, the analysis can be made more thorough. Some of the on-site steps that can be taken are pressuring tubing to help verify possible tubing leaks; and in gas engine driven installations, changes can be made in pumping speeds to reduce pounding or to increase production.

The portable analytical equipment consists of the following major components. (See Fig. 1.)

1. Load and position transducers are mounted on the polished rod to sense well load and polished rod position versus time.
2. A strip chart recorder is used to excite the transducers, amplify the return signal and record data on a two-channel strip chart.
3. A digital computer is programmed with a mathematical model which operates on data received from the strip chart recorder and teleprinter.
4. A teleprinter is used to input data into the computer and to output results.
5. An X-Y plotter is a computer output device which is used to plot dynagraph cards for interpretation.

Other components include a punched tape photoreader (for inputting programs quickly), vehicle, electric power plant, air conditioning unit and miscellaneous tools for mounting the transducers on the well.

INTERPRETATION OF DOWNHOLE PUMP CARDS

Figure 2 shows various types of pump cards. Since combinations of pump conditions can exist simultaneously, combined conditions can be thought of as being superimposed one upon the other. By scaling the pump card, quanti-

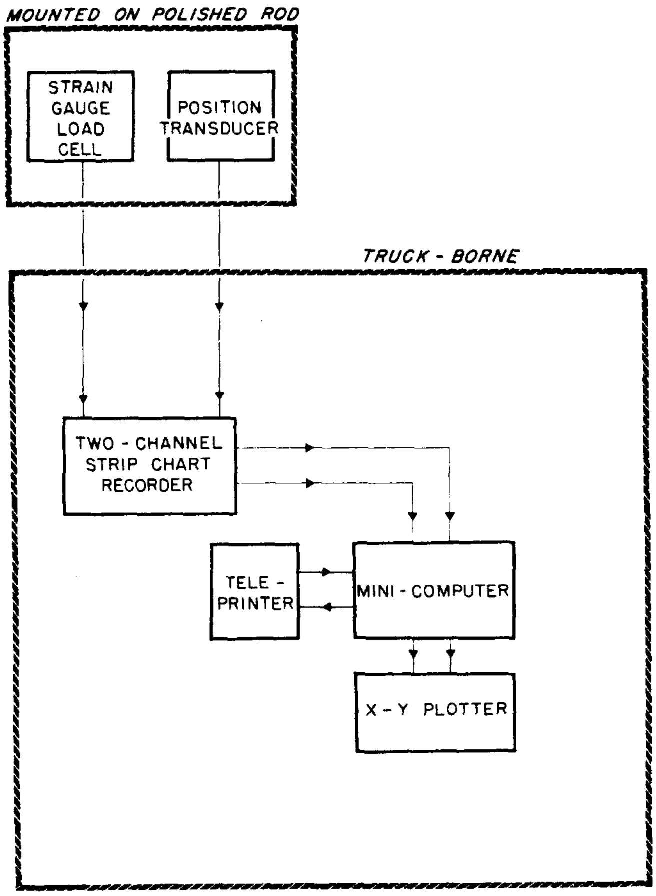


FIG. 1 ON-SITE ANALYTICAL EQUIPMENT

tative values are determined for gross pump stroke, net stroke, fluid load and friction. These measurements are used to compute pump liquid throughput, pump efficiency, pump intake pressure and frictional loads. Pump card interpretation is discussed further in the Case History examples that follow.

PUMP INTAKE PRESSURE AND WELL POTENTIAL

Pump intake pressure is calculated from data obtained from the downhole pump card (fluid load, gross and net strokes) and from produced fluid properties (specific gravity and PVT relationships). Thus, this method eliminates the need for fluid level measurements. Sonic fluid level measurements are often misleading in gassy (foamy) wells and impossible to obtain in wells equipped with packers or casing restrictions.

Fundamentally, the pump intake pressure is calculated as follows. The fluid load is measured from the pump card and divided by the pump area. This determines the pressure that the pump must supply to lift fluids from the well. By subtracting this pressure from the pressure in the tubing directly above the pump (obtained from produced fluid properties and from a family of modified gradient curves) the pressure that is filling the pump (pump intake pressure) is calculated.

The ultimate production capability can then be computed by using the well's PI (Productivity Index) or by using the dimensionless IPR (Inflow Performance Relationship) curve. (See Fig. 3.) The dimensionless IPR curve accounts for declining PI as a well's bottom-hole producing pressure is drawn down below the bubble point. In order to use the dimensionless IPR curve or to obtain a PI, an average static reservoir pressure or another pump intake pressure at a different stabilized producing rate is required.

Figure 4 is a good example of how one operator used the diagnostic technique to increase production. Gassy conditions were occurring which made fluid level information difficult to interpret. However, the downhole pump card indicated severe gas interference and a pump intake pressure of 1077 psi. The pump intake was set above perforations which resulted in very poor free gas separation. Much of the free gas entered the pump instead of being produced up the casing annulus. Subsequently, the pump intake was lowered and positioned below all perforations. Gas separation was improved which resulted in an increase in production from 100 BOPD and 2 BWPD to 140 BOPD and 6 BWPD and a corresponding decrease in pump intake pressure from 1077 psi to 623 psi. (See Fig. 5.) Using the dimensionless IPR curve, a total potential of 163 BOPD and 7 BWPD was predicted at a lower pump intake pressure of 100 psi. Larger equipment was installed to increase pump displacement and production was increased to 165 BOPD and 8 BWPD which compares very favorably to that predicted. Also, a static reservoir pressure of 1764 psi was calculated by extrapolating the dimensionless IPR curve backwards to zero producing rate. Having a static reservoir pressure is often useful in predicting potentials of offset wells.

Combinations of the following can exist simultaneously.

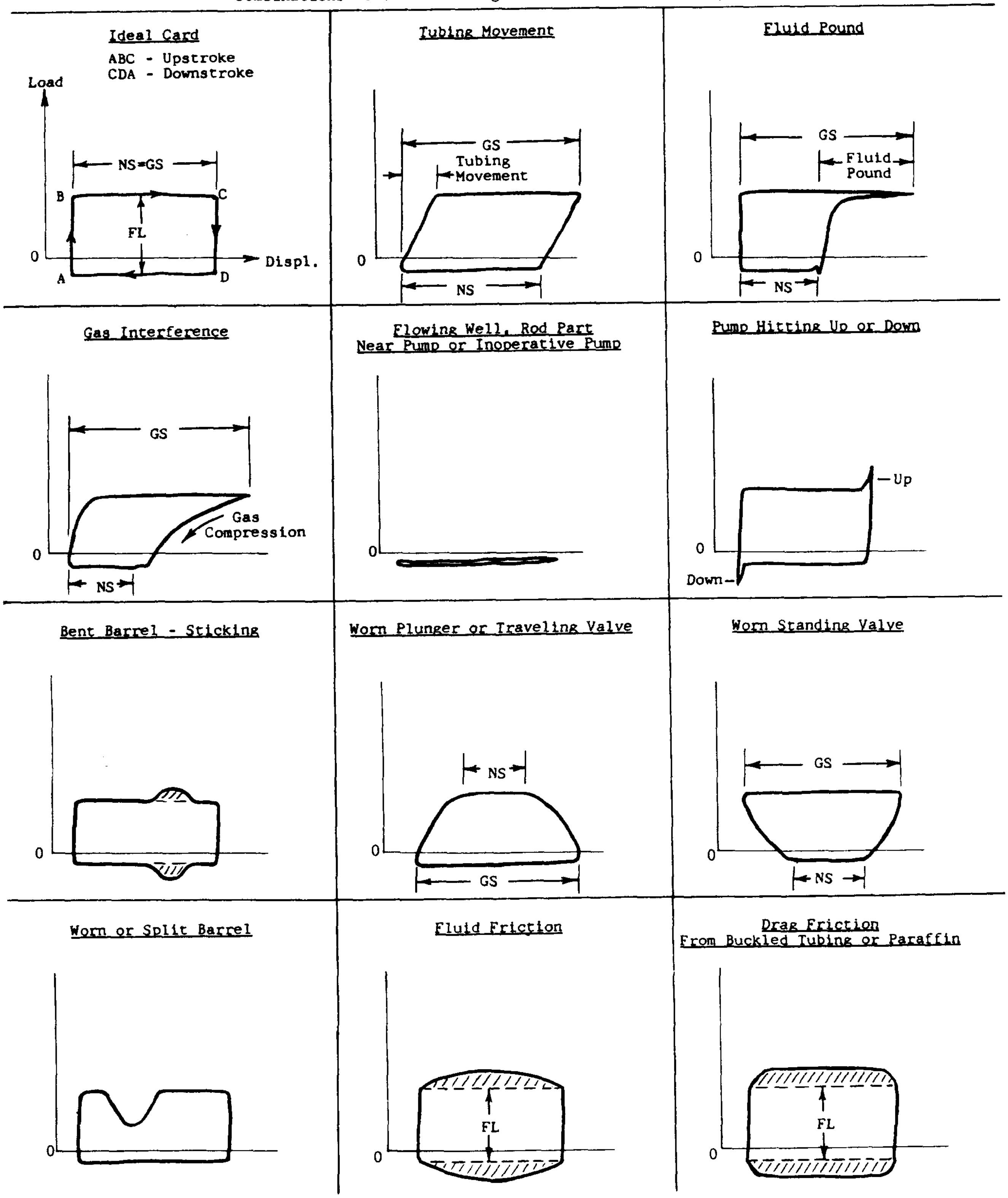


FIG. 2—INTERPRETATION OF PUMP DYNAGRAPH CARDS

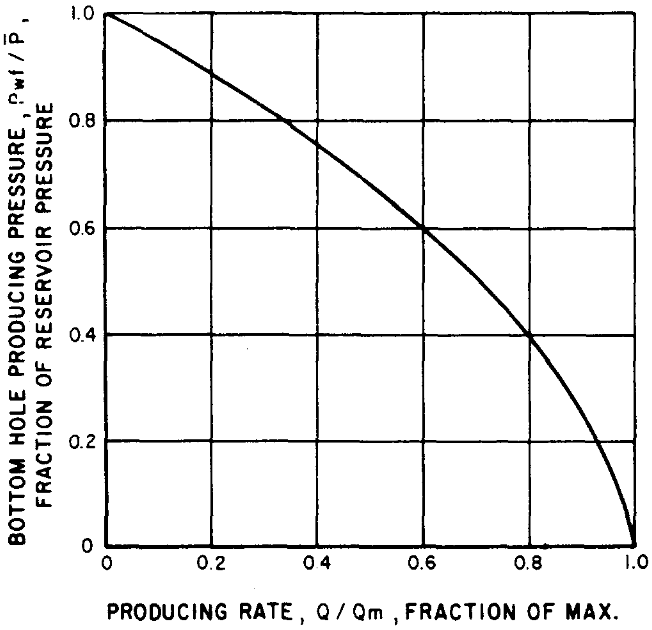


FIG. 3—DIMENSIONLESS IPR CURVE

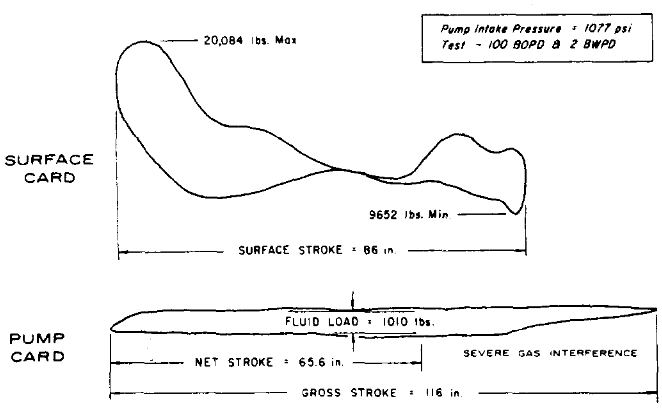


FIG. 4—BEFORE LOWERING PUMP

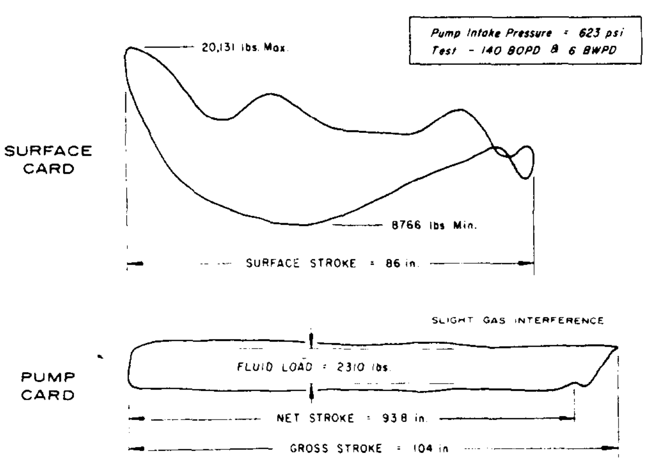
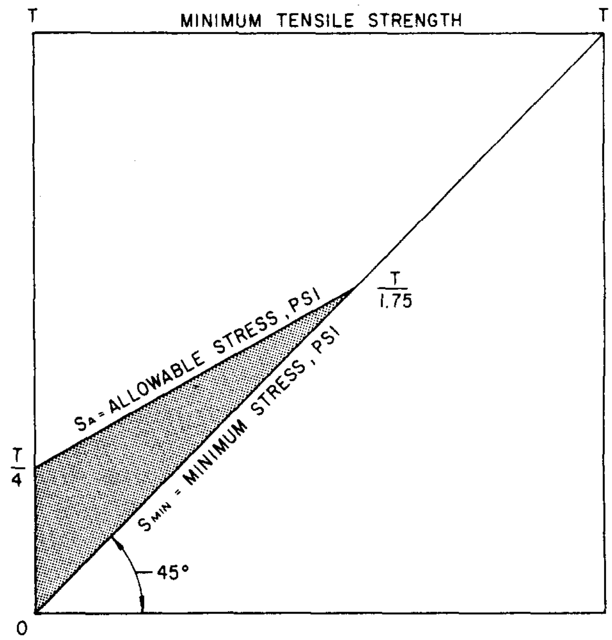


FIG. 5—AFTER LOWERING PUMP

EVALUATION OF SUCKER ROD STRESSES AND ROD TAPER DESIGN

In addition to surface rod stresses, downhole intermediate rod stresses are also computed to evaluate the rod taper design. The API Modified Goodman Diagram is used as an evaluation yardstick to compare stress levels for various grades of rods and for various service factors. (See Fig. 6.) A good taper design would be one in which the calculated percentages of the maximum allowable stress ranges are about the same for each rod size and where the maximum allowable stress ranges are not exceeded. Figure 7 shows examples of a good and a poor rod string design. In order to improve on the poor design, more ⅞-in. and ¾-in. rods are required.



$$S_A = \left(\frac{T}{4} + M\,S_{MIN}\right) SF$$

$$S_A = (0.25\,T + 0.5625\,S_{MIN})\,SF$$

$$\Delta S_A = S_A - S_{MIN}$$

Where :

S_A = MAXIMUM AVAILABLE STRESS, PSI
ΔS_A = MAXIMUM ALLOWABLE RANGE OF STRESS, PSI
M = SLOPE OF S_A CURVE = 0.5625
S_{MIN} = MINIMUM STRESS, PSI (CALCULATED OR MEASURED)
SF = SERVICE FACTOR
T = MINIMUM TENSILE STRENGTH, PSI

FIG. 6—API MODIFIED GOODMAN DIAGRAM

GOOD ROD TAPER DESIGN FOR UNITY SERVICE FACTOR

TOTAL STRING LENGTH(FT): 8675
API ROD GRADE AND MIN TENSILE: GRADE D, WCN-75; 125000 PSI

DIMENSIONS AND ACTUAL STRESSES ***

DIAMETER (IN)	INTERVAL LENGTH (FT)	MAX STRESS (PSI)	MIN STRESS (PSI)	PERCENT OF API/GOODMAN RANGE FOR SERVICE FACTOR OF 1.0	0.8	0.6
1.000	2150	35000	15880	78.7	117.6	232.4
.875	2500	31942	12477	75.5	107.3	185.7
.750	4025	28933	7720	76.1	102.2	155.6

POOR ROD TAPER DESIGN FOR UNITY SERVICE FACTOR

TOTAL STRING LENGTH(FT): 7100
API ROD GRADE AND MIN TENSILE: GRADE D, JONES 4; 120000 PSI

DIMENSIONS AND ACTUAL STRESSES ***

DIAMETER (IN)	INTERVAL LENGTH (FT)	MAX STRESS (PSI)	MIN STRESS (PSI)	PERCENT OF API/GOODMAN RANGE FOR SERVICE FACTOR OF 1.0	0.8	0.6
.875	1825	26000	14000	50.3	73.6	137.5
.750	1825	25700	11000	58.4	81.9	137.2
.625	3450	26700	6000	75.6	100.0	147.6

FIG. 7 GOOD AND POOR ROD TAPER DESIGN FOR UNITY SERVICE FACTOR

FINDING UPHOLE LEAKS

Uphole leaks (tubing or casing check valve leaks) can be indicated by comparing pump liquid throughput with the production test. The net liquid stroke is determined from the pump card. Thus, by knowing the stroke, pump bore and pumping speed, production through the pump can be calculated. If calculated pump throughput significantly exceeds the test then an uphole leak is suspected. Figure 8 pointed out an uphole leak. Pump throughput was calculated at 239 BFPD as compared to a test of 125 BFPD. Since the analysis is done on-site, the tubing can be pressured on wells with poor comparisons between pump throughput and test to quickly help prove or disprove an uphole leak.

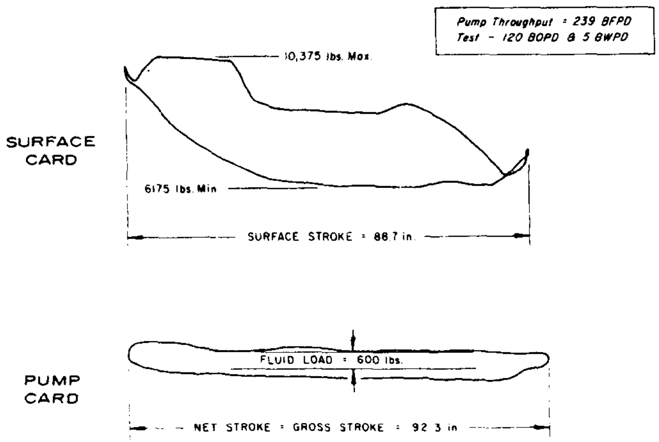


FIG. 8—UPHOLE LEAK INDICATED

DOWNHOLE FRICTION

Downhole friction can be separated into two basic types—fluid friction and drag friction.

Excessive fluid friction results when large volumes of produced fluids are forced up relatively small tubing strings. An example of excessive fluid friction is shown by Fig. 9.

Excessive drag friction results from tubing deflection (crooked hole or set-down weight on packers or anchors) or from paraffin deposition. An example of excessive drag friction is shown by Fig. 10.

Fluid friction is velocity-dependent and normally reaches maximum near midstroke. Drag friction is relatively independent of velocity; thus, the interpreter can normally distinguish between the two.

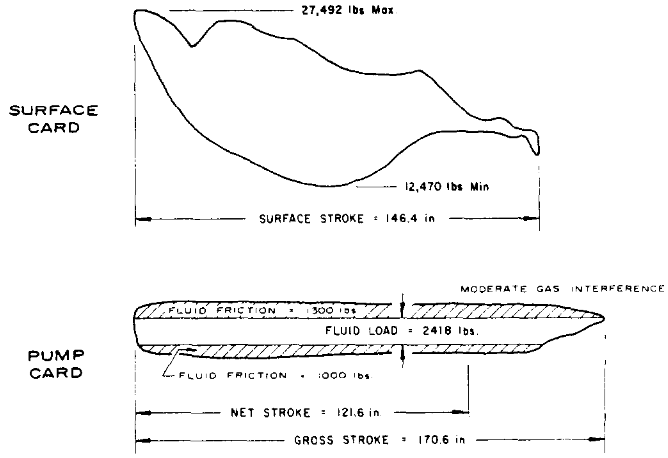


FIG. 9—EXCESSIVE FLUID FRICTION

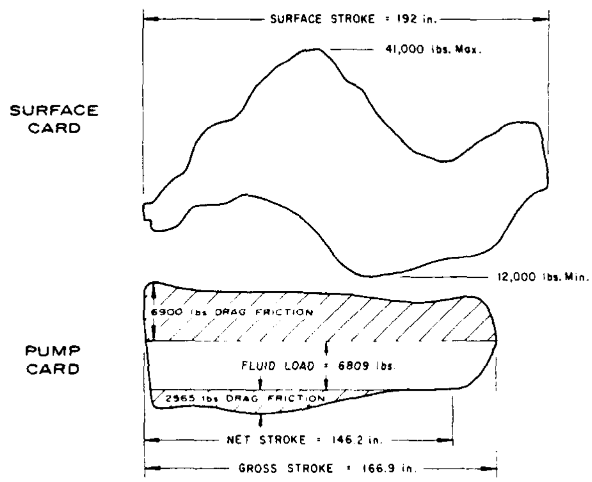


FIG. 10—EXCESSIVE DRAG FRICTION

OTHER DOWNHOLE INFORMATION

Downhole pump conditions such as wear, hitting up or down, tubing movement and excessive pump friction are easily recognized by the shape of the downhole pump card.

An example of each is shown respectively by Figs. 11, 12, 13 and 14.

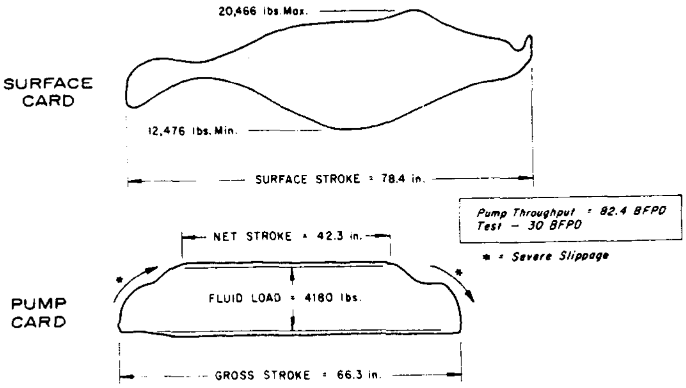


FIG. 11—SEVERELY WORN PUMP

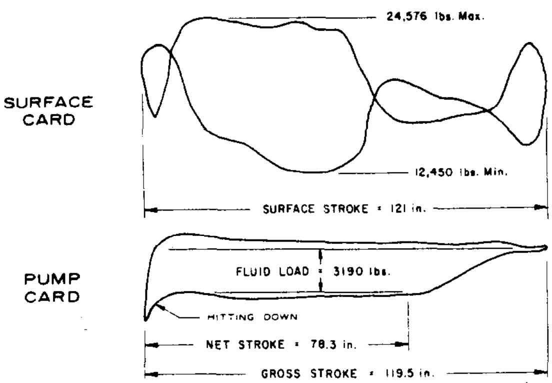


FIG. 12—POOR PUMP SPACING—PUMP HITTING DOWN

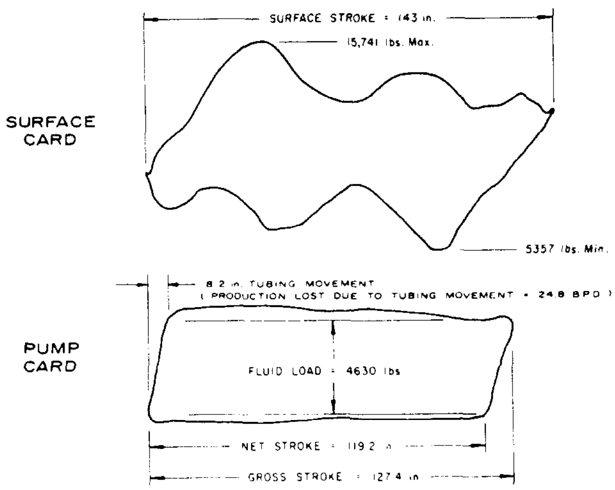


FIG. 13—TUBING MOVEMENT

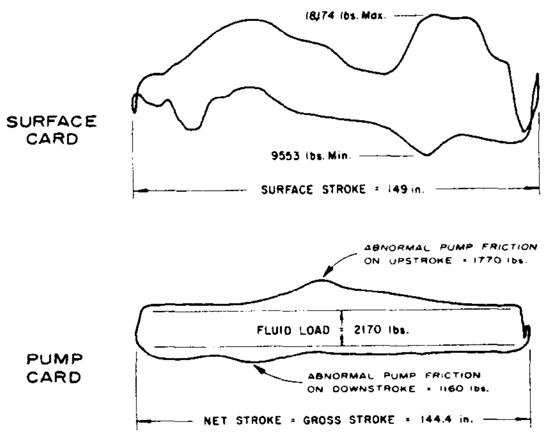


FIG. 14—EXCESSIVE PUMP FRICTION

CALCULATION OF NET GEARBOX TORQUE

Gearbox torque is computed by the API torque factor method when a normal slip prime mover is being used. Measured polished rod load, polished rod displacement, maximum counterbalance and the unit manufacturer's torque factors are input into the computer. Net gearbox torque is then calculated for each 15 degrees of crank rotation in accordance with the following formula:

$$NT = TF\,(PRL\text{-}S) - M \sin(\Theta + \gamma)$$

where:

- NT = Net torque (in.-lb)
- TF = Torque factor (in.)
- PRL = Polished rod load (lb)
- S = Structural unbalance (lb)
- M = Maximum counterbalance moment (in.-lb)
- Θ = Crank angle (degrees)
- γ = Offset angle of weights (0 degrees for conventional units)

Figure 15 shows a tabulation of torque history versus crank angle followed by a Summary of Gearbox Performance. In the Summary the computer prints out the peak net torque, maximum counterbalance moment and percent of gearbox rating. As shown the gearbox is 50.3% overloaded with a maximum counterbalance moment of 1,224,400 in.-lb. Also shown in the Summary is the peak net torque that would exist if the unit were perfectly balanced under existing conditions. For ideal unit balance the maximum counterbalance moment is computed to be 1,627,400 in.-lb, and with this amount of counterbalance the gearbox would operate at 95.7% of rated capacity (no overload). Once the counterbalance requirement is known, the manufacturer's counterbalance tables can be used to properly position the weights on the cranks to attain ideal balance. If present weights are inadequate, larger counterweights can be selected.

NORMAL SLIP PRIME MOVER

TORQUE HISTORY ******************

CRANK ANGLE (DEG)	TORQUE(M IN-LBS) ROD	COUNTERBALANCE	NET
0	-131.8	.0	-131.8
15	511.6	316.9	194.7
30	863.2	612.2	250.9
45	1520.2	865.8	654.4
60	2022.6	1060.4	962.2
75	1848.4	1182.7	665.7
90	1345.1	1224.4	120.7
105	960.8	1182.6	-221.8
120	951.9	1060.2	-108.4
135	884.1	865.6	18.5
150	639.2	611.9	27.3
165	344.1	316.6	27.6
180	88.7	-.4	89.1
195	-138.9	-317.3	178.4
210	-423.5	-612.6	189.1
225	-700.0	-866.1	166.1
240	-989.4	-1060.6	71.2
255	-1179.8	-1182.8	3.1
270	-1301.6	-1224.4	-77.2
285	-1400.3	-1182.5	-217.7
300	-797.4	-1060.0	262.6
315	-619.6	-865.3	245.7
330	-675.7	-611.6	-64.1
345	-483.4	-316.2	-167.2

SUMMARY OF GEARBOX PERFORMANCE ***

	EXISTING	IN BALANCE
PEAK TORQUE(M IN-LBS):	962.2	612.3
COUNTERBALANCE(M IN-LBS):	1224.4	1627.4
PERCENT OF GEARBOX RATING:	150.3	95.7

FIG. 15

Computation of gearbox torque on units driven by an ultra high-slip prime mover (speed variation from 20-40%) requires additional measurements and calculations. Besides continuous tachometer measurements to determine rotating speed variations, moments of inertia are required for rotating and articulating unit components. As shown by Fig. 16 two types of inertia are calculated, i.e. rotating inertia (cranks, weights, gears, sheaves, and motor rotor) and articulating inertia (beam, horsehead, equalizer and pitman arms). The sign convention is such that when inertia is resulting from a decrease in angular velocity, gearbox torque is reduced and when inertia is resulting from an increase in angular velocity, gearbox torque is increased. Net torque is calculated for each 10 degrees of crank rotation according to the following formula:

$$NT = TF\ (PRL\text{-}S) - M \sin(\quad +\gamma) + I_R + I_A$$

where:

I_R = Rotating inertia torque (in.-lb)
I_A = Articulating inertia torque (in.-lb)

ULTRA HIGH SLIP PRIME MOVER

TORQUE HISTORY ******************

TORQUE(M IN-LBS) ROD	CNTR-BAL	ROTARY INERTIA	ARTICULATING INERTIA	NET	SPM
-45.9	0	14.8	-2.4	-33.6	12.9
184.6	-117.2	2.8	9.8	80.1	12.93
410.3	-230.8	-35.2	19.9	164.1	12.78
607.2	-337.4	-63.1	21.9	228.5	12.34
737.6	-433.8	-84.1	14.5	234.2	11.65
1002.6	-517	-100.5	1.2	386.3	10.66
922	-584.5	-88.8	-7.6	241	9.59
876.9	-634.2	-32.4	-5.5	204.7	8.85
756.9	-664.7	22.7	-1.2	113.7	8.83
737.2	-675	34.7	-2.6	94.3	9.17
755	-664.7	29.4	-5.1	114.5	9.52
655.1	-634.2	29.7	-6	44.4	9.83
580.1	-584.5	36.5	-6	26.1	10.16
450.9	-517	49.9	-5.6	-21.9	10.59
337.6	-433.8	64	-5.4	-37.7	11.09
278.9	-337.5	67.3	-5.6	3.1	11.68
219.1	-230.8	49.9	-5.8	32.4	12.23
141.6	-117.2	22.8	-4.7	42.6	12.53
61.5	0	-.2	-2.2	59	12.63
-34.3	117.3	-3.2	1.5	81.2	12.63
-126.1	230.9	-15.7	6.1	95.2	12.51
-206.1	337.5	-30.6	10.6	111.5	12.31
-291.6	433.9	-42.1	13.4	113.6	11.97
-357.6	517.1	-35.2	14.4	138.7	11.61
-437.2	584.6	-15	14.6	146.9	11.36
-496.7	634.3	-6.9	12.6	143.3	11.28
-386.4	664.8	-22.5	6.3	262.1	11.15
-417.5	675.1	-44.5	-2.1	210.8	10.79
-523.5	664.8	-42.3	-6.6	92.5	10.3
-569.1	634.3	-4.6	-4.9	55.6	10.02
-641	584.6	42	-2.7	-17	10.22
-541.5	517.1	65.3	-5.1	35.8	10.77
-434.7	433.9	58	-11.2	46.1	11.4
-380.7	337.6	43.2	-15.5	-15.5	11.87
-292.9	230.9	39.1	-15.6	-38.5	12.24
-200.9	117.3	37.6	-11.4	-57.5	12.6

SUMMARY OF GEARBOX PERFORMANCE ***

	EXISTING	IN BALANCE
PEAK TORQUE(M IN-LBS):	386.3	331.9
COUNTERBALANCE(M IN-LBS):	675	745.7
PERCENT OF GEARBOX RATING:	84.7	72.7

NOTE: TORQUES ARE SHOWN EACH 10 DEGREES OF CRANK ROTATION STARTING FROM ZERO WHICH IS VERTICAL POSITION NEAREST BOTTOM OF STROKE. IN BALANCE COMPUTATIONS ASSUME INERTIA EFFECTS WILL BE RELATIVELY UNCHANGED.

FIG. 16

PRIME MOVER AND STRUCTURAL LOADING

Prime mover loading can be determined by comparing the calculated average polished rod horsepower from the surface dynagraph card with the prime mover horsepower rating. Normally, the prime mover horsepower rating should be at least twice the average polished rod horsepower. This is because the peak power demand on the prime mover which occurs twice each pumping cycle (upstroke and downstroke) is normally several times the average power.

Structural loading is determined by comparing the maximum polished rod load with the unit manufacturer's rating.

CONCLUSIONS

The on-site sucker rod diagnostic technique is a unified analysis of mechanical equipment performance and the capability of the well. Interpretation of a surface dynagraph card by the computer makes downhole analysis a science (quantitative) rather than an art (qualitative). Thus, the task of obtaining the maximum profit from a well by assuring optimum equipment performance is made easier.

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