High Slip Motors Reduce Loading on Beam Pumping Installations

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INTRODUCTION

In the Shell-operated Denver Unit (Wasson Field), tests have been conducted on extra-high slip motors to determine if the predicted design-results from a computer program for predicting performance of rod pumping installations could actually be obtained so that a capital savings on pumping unit purchases could be realized. The computer program comes from the work by S. G. Gibbs, "Predicting the Behavior of Sucker Rod Pumping Systems", Journal of Petroleum Technology, July 1963. When designing a pumping unit installation, it takes into account the system rotary inertia caused by motor speed change. The results indicated that peak torque reduction in the range of 25 to 30 per cent could be realized with the use of high slip motors over conventional slip motors. The initial tests with the motor confirmed the computer results and also indicated the results to be conservative. It was then decided that more complete testing of this motor should be conducted to determine motor slip versus loading characteristic of beam installations. This paper shows some of these results for a 144-in. unit pumping 11+ SPM from 5130 ft with a 2-1/4 in. diameter pump. At present, about 90 beam installations with high slip motors are being operated in the Denver Unit.

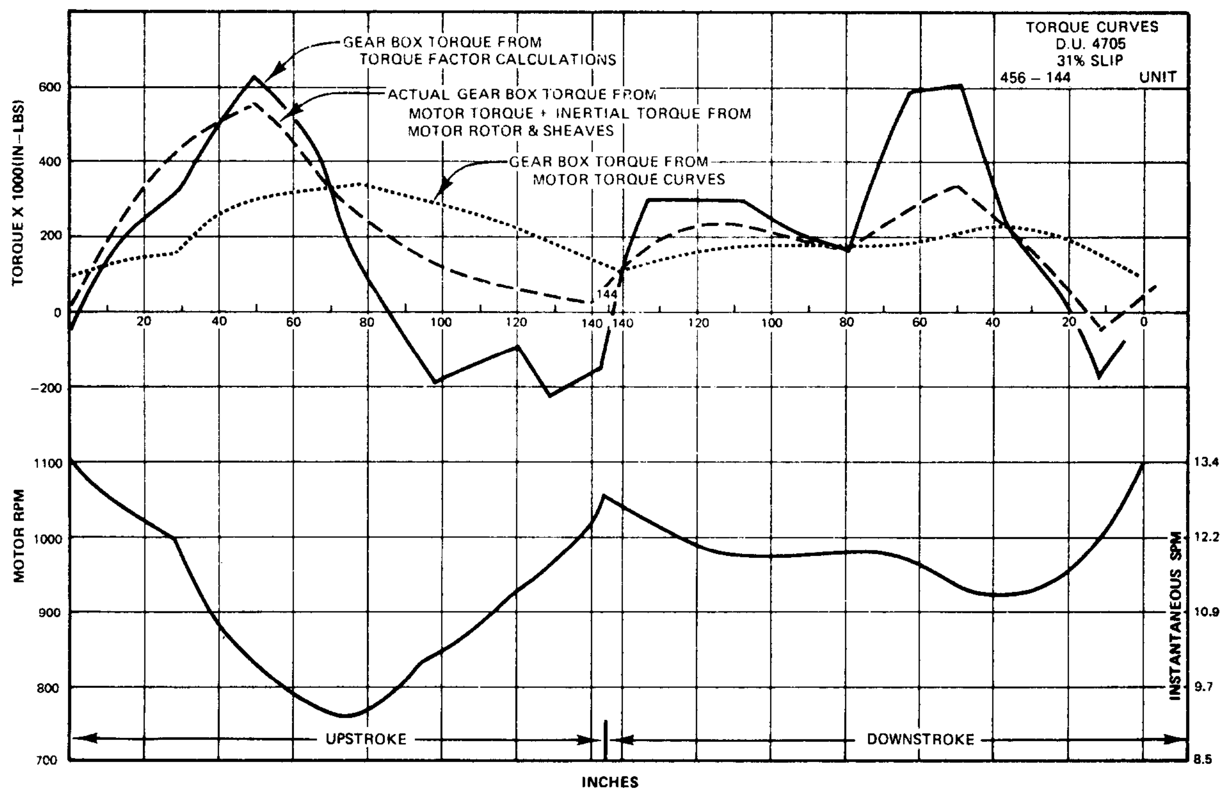

FIGURE 1

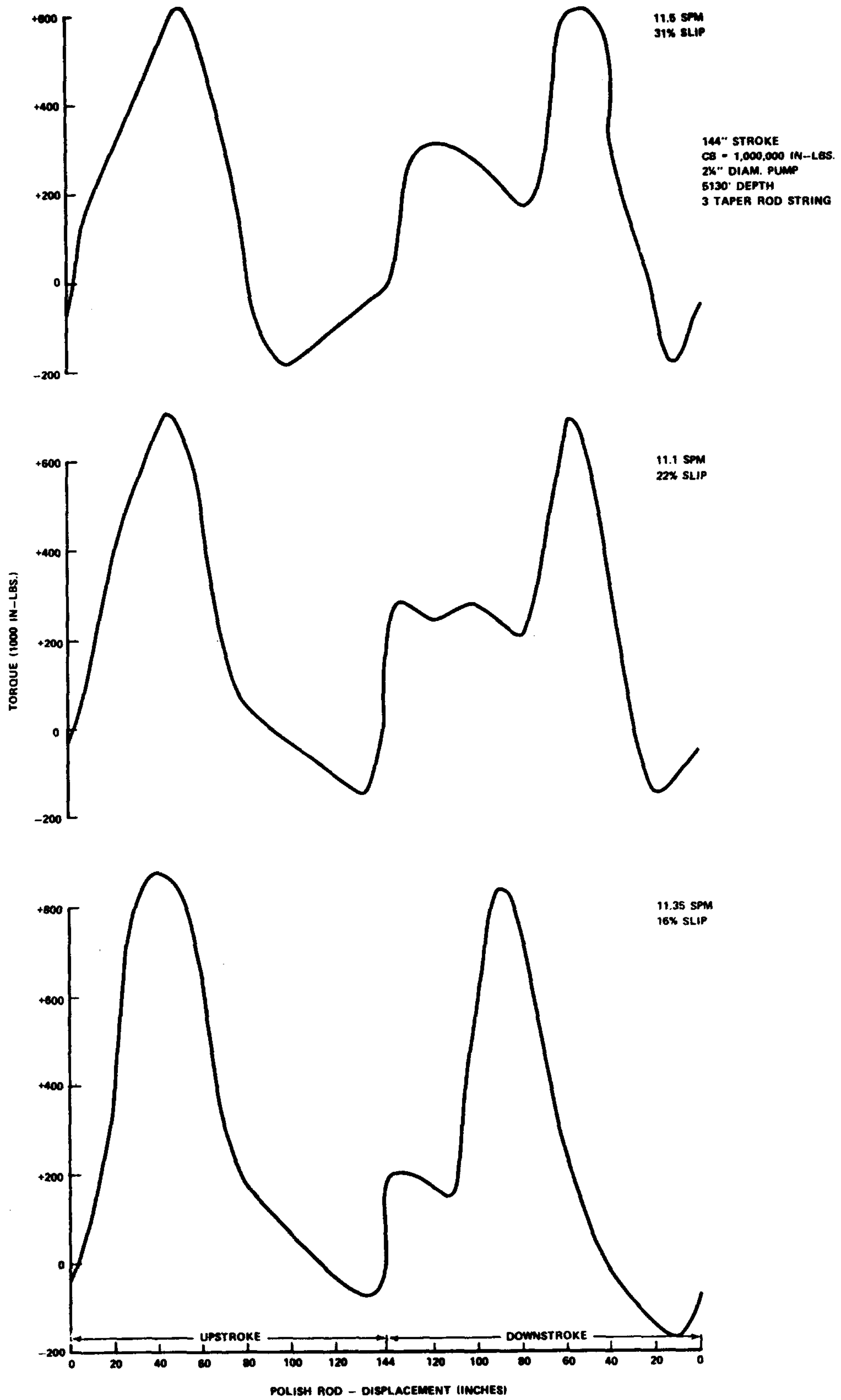
+600
+400
+200
0
-200
11.5 SPM
31% SLIP
144" STROKE
CB = 1,000,000 IN--LBS.
2¼" DIAM. PUMP
5130' DEPTH
3 TAPER ROD STRING
TORQUE (1000 IN-LBS.)
+600
+400
+200
0
-200
11.1 SPM
22% SLIP
+800
+600
+400
+200
0
-200
11.35 SPM
16% SLIP
UPSTROKE
DOWNSTROKE
0
20
40
60
80
100
120
144
120
100
80
60
40
20
0
POLISH ROD - DISPLACEMENT (INCHES)

FIGURE 2

TESTING PROCEDURE

A recording tachometer in conjunction with the Delta II Dynamometer was used to get motor rpm versus time, polish rod displacement versus time, and polish rod load versus time. Also, voltage was recorded versus time to assure that the motor characteristics were being used at correct voltage. From the collection of this type of data, conclusions were drawn pertaining to the loading of a beam installation in conjunction with high slip motors. Slip as used in this paper will be defined as the per cent speed change of the motor during a pumping cycle.

DISCUSSION

Effects of Motor Slip on Gear Box Loading

Two methods of calculating torque on the gear box were used to evaluate the effects of actual gear box loading with the use of extra-high slip motors. The first method is the conventional API method; i.e., the unit torque factors were multiplied times the polish rod loads and structural unbalance. Then the counterbalance torque was subtracted to give the gear box torque. This method excludes any inertial energy contributed to the system by the masses of the pumping unit changing speeds. With the second method, the motor torques from the motor torque curves were multiplied times the sheave ratio and the unit gear ratio to give an estimated gear box torque. This method excludes the inertial energy of the motor rotor, motor and unit sheaves, belts, and the low and high speed gears.

The actual gear box torque is the motor torque plus the inertial energy of the motor rotor, motor and unit sheave, and is somewhere between the torque factor and motor torque curves as shown on Fig. 1. It can be seen that the torque factor and motor torque curves are approximately equal when there is no motor speed change, since no inertial energy is being dissipated. The large difference in these two torque curves indicates the available inertial energy in the system which can be obtained from the use of extra-high slip motors. As the polish rod demands torque from the unit, the motor must slow down to develop the torque demand. At this point the torque demand on the motor is helped by the inertial energy dissipated by the counterbalance weights slowing down. On the other hand, the unit sheave must be as small as practical to reduce its available inertial energy from helping the motor supply the torque demand of the unit. The flywheel effect of a large unit sheave will not allow the motor to slip.

The difference between the torque factor curve and the actual torque curve shown in Fig. 1 is the torque developed by the counterbalance speed change. (The actual torque as shown in Fig. 1 is derived from the measured motor torque plus the calculated inertial torque developed from the motor rotor and sheaves.) Any error in the maximum counterbalance moment used to calculate the torque factor curve will be reflected in the differences between these two curves. The motor speed plot in Fig. 1 shows the unit to be under-counterbalanced since it slows down less during the downstroke, yet the torque factor

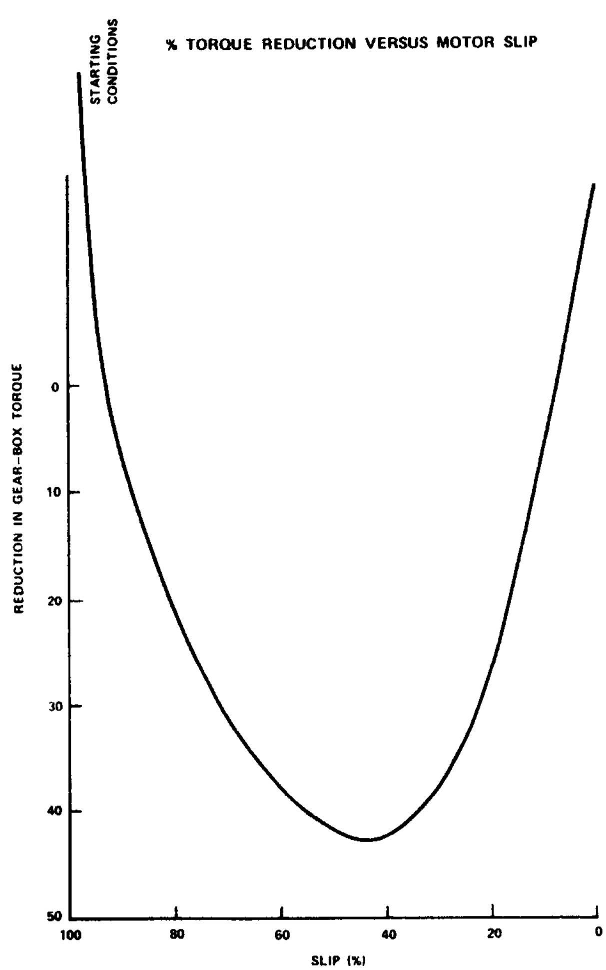

FIGURE 3

curve shows the unit to be in balance. This indicates the pumping unit manufacturer's counterbalance moment data used in the torque factor calculation curve to be in error (high) by a few per cent. On the torque factor curve the peak torque on the upstroke should be 100,000 in-lb higher and the peak torque on the downstroke 100,000 in-lb lower to account for the error in counterbalance moment. To eliminate the possibility of this error in net torque calculations, the counterbalance effect at the polish rod should be measured with a load cell at the polish rod. The difference between the motor torque and the actual torque curve is the torque developed by the speed changes in the motor rotor, motor and unit sheave and belts.

Figure 2 shows the results of slip versus gear box torque as calculated from API methods. The motor sheave size was changed to keep the average SPM of each test approximately equal. Figure 3 shows a curve of gear box torque reduction versus motor slip for a 144-in. unit pumping 11+ SPM. From this curve the minimum obtainable gear box torque is between the 40 and 50 per cent motor slip range. Any slip greater than this is detrimental since the energy needed to speed up the unit masses is greater than the energy gained by the slowing down of the masses. The optimum gear box torque will be dictated by economic criteria.

Effect of Motor Slip on Polish Rod Load

To actually discuss the effects of motor slip on polish rod loading, a short discussion on the change of polish rod motion versus slip should

POLISH ROD MOTION CURVES

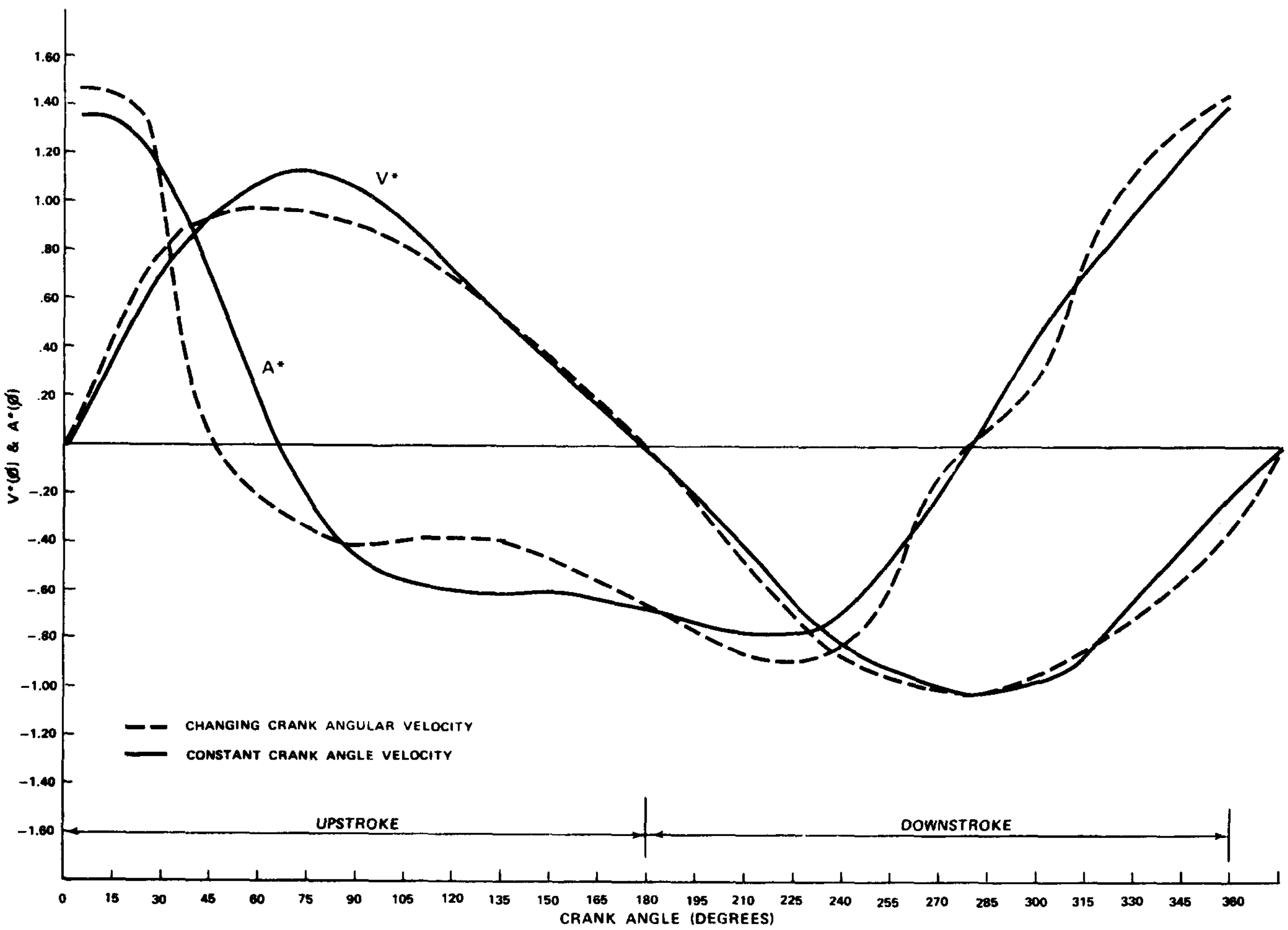

FIGURE 4

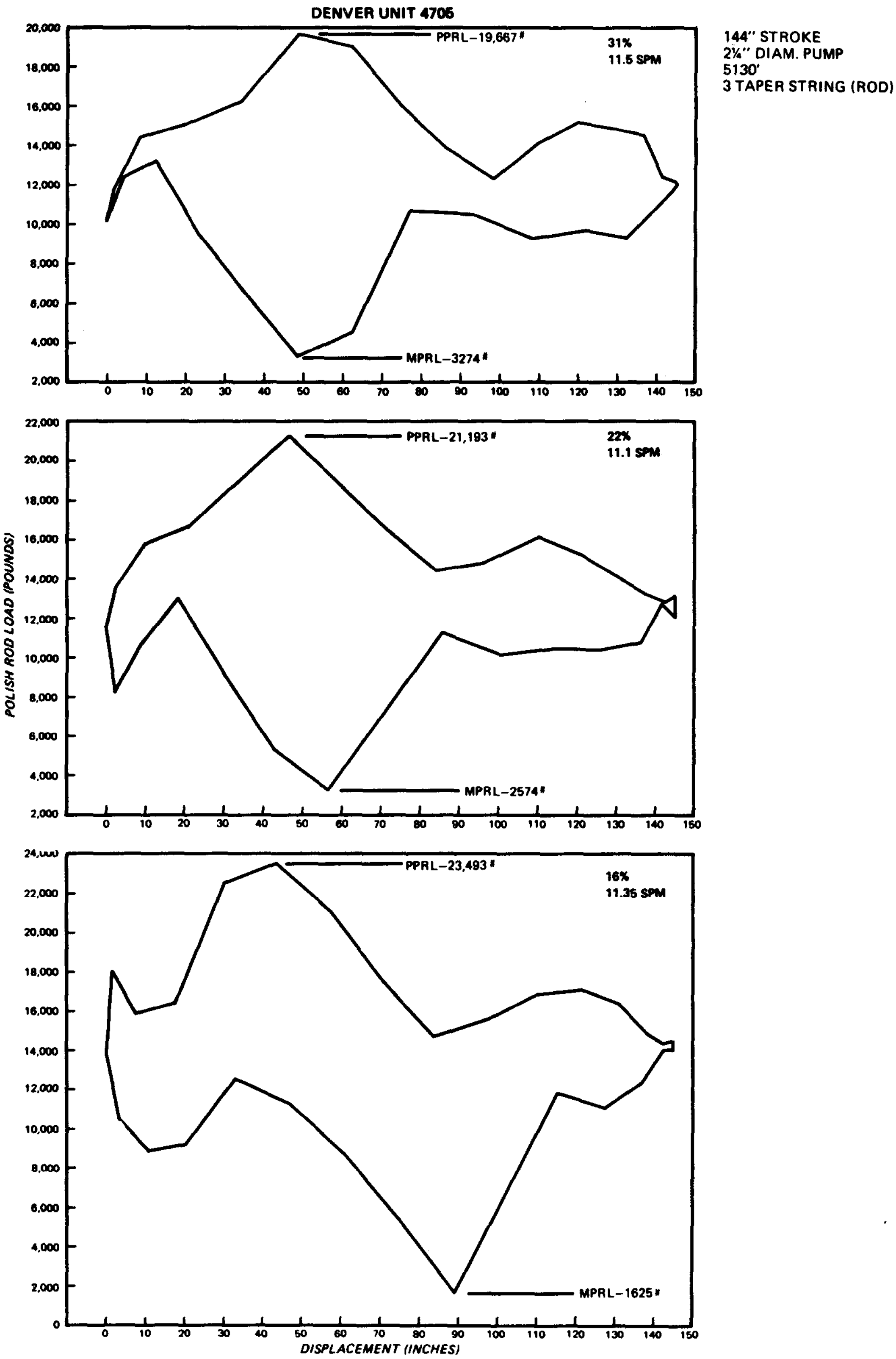
DENVER UNIT 4705
PPRL–19,667#
31%
11.5 SPM
MPRL–3274#
PPRL–21,193#
22%
11.1 SPM
MPRL–2574#
PPRL–23,493#
16%
11.35 SPM
MPRL–1625#
POLISH ROD LOAD (POUNDS)
DISPLACEMENT (INCHES)
144" STROKE
2¼" DIAM. PUMP
5130'
3 TAPER STRING (ROD)

FIGURE 5

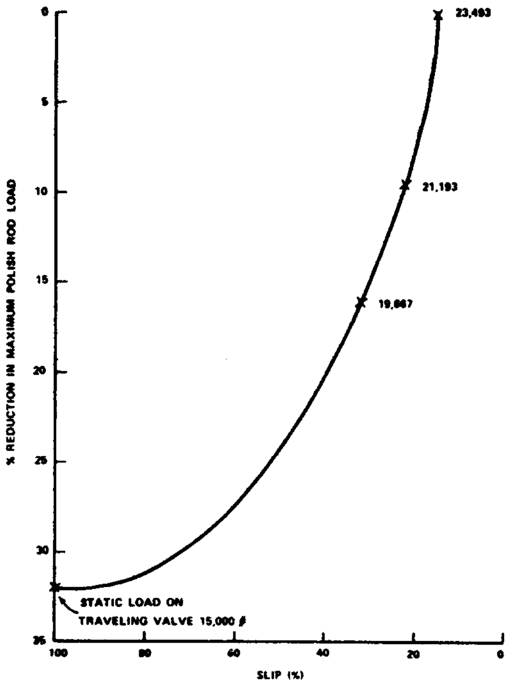

FIGURE 6

be made. The base curves used in this discussion will be the non-dimensional superficial polish rod velocity and acceleration curves for conventional units defined in the paper "Kinematics of Oil Well Pumping Units" by H. E. Gray, presented at the API Division of Production, March 1963.

Figure 4 shows the effect motor slip and changing crank angle velocity have on polish rod motion. From Fig. 4 it can be seen that the major difference in the velocity curves on the upstroke is the decrease in the peak and the flatting out over a portion of the upstroke. This change will, in general, give a slight increase in time for the upstroke. The major difference in the velocity curves on the downstroke is the higher velocity going into the bottom of the downstroke. This should cause more overtravel of the pump plunger, if the pump's motion is responding as the polish rod. At present, no general conclusions pertaining to overtravel have been drawn. Field tests have shown both slight gains and losses in pump overtravel.

The differences in the acceleration curves show less upstroke acceleration through the majority of the first 90° of crank rotation on the high slip motor curve. This reduction in acceleration will, in general, reduce the maximum polish rod load. The possibility of overtravel at the end of the downstroke can also be seen in the increase in acceleration at the bottom of the downstroke.

Figure 5 shows the surface dynamometer cards for the tests. From these cards it can be seen that the peak polish rod load is in phase with the position of the reduction of upstroke acceleration realized with use of the high slip motor. With the peak polish rod load in phase with the polish rod's reduction in upstroke acceleration, a plot can be made of slip versus peak polish rod load reduction. This plot is shown on Fig. 6. The minimum possible polish rod load on the upstroke would be the 100 per cent slip or static condition, and this load would be the static traveling valve load at the polish rod.

ELECTRICAL BENEFITS

The extra-high slip motor is more fully loaded through the pumping cycle than a conventional motor. This is accomplished by using a smaller horsepower-rated motor to do the same work. The smaller motor in turn improves the power factor due to being loaded heavier and reduces transformer size and distribution line requirements due to reduced currents. The high slip motor is also a triple-rated motor; this flexibility will enable the motor to be loaded throughout its life to gain added electrical benefits. The triple-rated motor has three torque ratings; (1) low torque, maximum 45 per cent slip, (2) medium torque, maximum 35 per cent slip, and (3) high troque, maximum 25 per cent slip. The motor should always be operated in the lowest possible torque mode to keep it fully loaded and thus maximize motor speed change.

CONCLUSIONS

It is concluded from the computer results and actual field tests that the gear box rating on beam installations from 640,000 in-lb-size down can be reduced one API size with the proper use of the extra-high slip motor. This reduction in unit size means a substantial savings in capital cost/BFPD. The extra-high slip motor also gives added benefits by reducing peak polish rod loads and polish rod load range. These load reductions will reduce that part of operating costs due to rod failures.