IMPROVED SUCKER ROD PUMPING DESIGN CALCULATIONS

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INTRODUCTION

This paper presents an improved sucker rod pumping design method similar to the API RP 11L but provides more accurate answers. The design is based on developing look-up tables or graphs for the desired unit geometry and pumping conditions by using a wave equation program. The wave equation and simulation techniques are used to mimic pumping conditions and calculate the loads and displacements. Such an approach makes it possible to compare, on a general basis, the different pumping geometries.

In 1954, a group of users and manufacturers undertook a study of sucker rod pumping and retained the services of Midwest Research Institute. Based on results from an electrical analog computer model, a set of graphs were developed for design calculations for sucker rod pumping systems -- conventional units. This method is the now widely-used procedure outlined in API RP 11L.

The API RP 11L assumes the following conditions:

- Conventional Unit Geometry (Modified Sine Wave Motion)
- Steel Sucker Rods
- Full Pump Fillage
- Small Prime Mover Slip
- Anchored Tubing
- Negligible Fluid Acceleration Loads
- No Abnormal Friction
- No Unusual Mechanical Problems

Furthermore, calculations for tapered rod strings need a correction and the pump loads often need adjustments. These assumptions, corrections and adjustments led some to conclude that the API method is limited and applies to only a small percentage of pumping wells. Although there is some merit in this reasoning, I believe that the industry still needs a general sucker rod design method. Some changes need to be made which will make the design method suitable for the majority of our wells. Such a program can be used to make a first cut at screening the numerous displacement cases that are available. Once the design is narrowed to a few specific cases, it may be advisable to determine the best case by use of a good wave equation design program.

S. G. Gibbs gave the first paper on the use of a one-dimensional wave equation with viscous dampening to simulate behavior of the sucker rod string. It provided a mathematical model that solves the boundary problem. In recent years, a number of other wave equation programs were developed for sucker rod pumping system design. Use of these programs is possible through time-share computing or by purchasing or licensing one or more of the programs from one of the various developers.

Gibbs and Neely of Shell Oil Company used the wave equation program in 1962 to develop design graphs similar to the API RP 11L. Like the API RP 11L, a number of assumptions had to be made; however, there were some differences. The Shell design graphs were for no prime mover slip and for 25, 50, 75, and 100 percent pump fillage. Also, there were different design graphs for one, two, and three taper rod strings. Graphs were not developed for peak torque similar to those on API RP 11L; instead, peak torque was calculated using the following formula:

Peak Torque = (Stroke/2) X (PPRL - MPRL)/2

This formula assumed that the peak (PPRL) and minimum polished rod loads (MPRL) occurred halfway through the upstroke and downstroke and that the torque factor at that point was half the stroke length. Also, the unit was assumed to be ideally counterbalanced. This formula gave close answers for many cases but seems to break down for shallow wells and slow pumping speeds.

Shell Oil Company used the Gibbs-Neely design method for many years; but like API RP 11L, the program provided answers which often were in significant error -- especially the peak torque calculations. In 1986, a new computer program named OMNIROD was written by Tripp et al for the personal computer which calculates the performance of sucker rod pumping systems. It uses the same rod string model developed by Gibbs but allows the analysis of both steel and fiberglass rods.

APPROACH

The wave equation program OMNIROD was used to develop the graphs and look-up tables needed for the design calculations for various sucker rod systems. Three types of geometries were investigated: Conventional, Lufkin Mark II, and Baker Torqmaster. A representative size for each was picked. Other geometries or units can be easily investigated using the same approach as outlined below.

A Lufkin 228D-200-74 size was picked for the conventional geometry and the calculations made on the basis of the unit running clockwise. Other makes and sizes may differ slightly but not enough to alter the design significantly.

The Lufkin Mark II 640-256-144 and also the Baker Torqmaster 640-256-144 size units were investigated. The calculations were made on the basis of the units running in the recommended direction. The same non-dimensional values should be generated for a given type geometry (which has the same displacement and loads vs time) for various size units if the same design techniques are used. However, a cursory check showed that other size units will give slightly different non-dimensional values.

Possibly, there is a scaling variation in designs between units of a particular manufacturer, or another variable, not now thought significant, causes load variations.

Nearly full pump fillage was assumed for all cases. The highest polished rod horsepower and peak polished rod loads are normally obtained for complete pump fillage. Most wells need to be designed for this maximum condition because such conditions normally occur between startup and shutdown of a unit. Wells with time clocks and pump-off controllers see full pump fillage frequently. Typically with wells that pump off, the surface loads stay about the same, except in the first half of the downstroke. A typical bottom hole pump card is shown in Figure 1-A. Note that the transfer of the load from the standing valve to the traveling valve occurs in 0.1 feet with a slight amount of gas expansion. The transfer from the traveling valve to the standing valve on the downstroke takes .22 feet with a slight amount of gas compression.

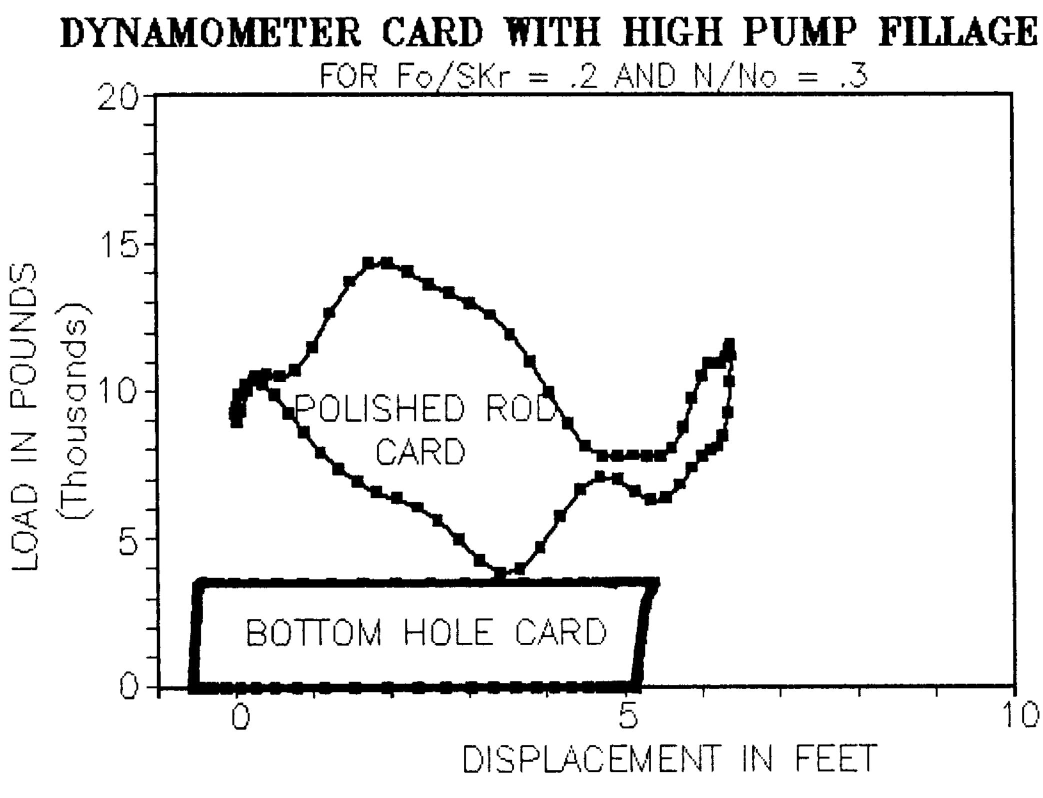

FIGURE 1-A

Typically, most wells run with only a small amount of motor slip. Slip tends to slightly reduce the peak loads in many cases; thus, a design with only a small amount of slip is on the conservative side. Assuming no prime mover slip results in the calculations giving loads that are too high. For all conditions, a Nema D electric motor with a seven percent slip was assumed. Use of such a motor will reduce the average speed about four percent. For example, if the sheaves size and gear ratio result in 10 spm, the well with a Nema D motor will average about 9.6 spm.

To develop the graphs and look-up tables, non-dimensional pumping speeds (N/No) of from 0 to 0.5 at 0.05 intervals were considered adequate. Closer intervals would improve the accuracy of the curves but would increase development time and result in increasing significantly the size of the look-up tables. Exact speeds are seldom known and the calculations between unit sizes vary only slightly. Thus, the effort required to increase the number of speeds was not deemed worthwhile.

The range of non-dimensional fluid loads (Fo/SKr) selected were values of 0.01, 0.1, 0.2, 0.3, 0.4, 0.5, and 0.6. The value of 0.01 is a very small load and was selected since the program will not run for a zero load. Values over 0.6 are rare and are not recommended.

A non-dimensional rod weight of 0.4 was selected as a base for all three type units. A single taper rod string was selected and for the base case a 0.75 inch rod was chosen as typical. A well depth was picked so that Wrf/SKr = 0.4 for the pumping conditions. The work in API RP 11L used a non-dimensional rod weight (Wrf/SKr) of 0.3. Since these values often run as high as 1.0, a higher value seemed appropriate.

The program OMNIROD will investigate only one unit at a time but will allow multiple speeds. A depth was selected so that Wrf/SKr was 0.4 and then a pump size and fluid level were picked to give a Fo/SKr equal to 0.01, .1, .2, .3, .4, .5, or .6. Also, pumping speeds were selected to give non-dimensional pumping speeds of .01, .05, .1, .15, .2, .25, .3, .35, .4, .45, and .5. Each run calculated 11 different speeds for a given Wrf/SKr and Fo/SKr. Since there were seven Fo/SKr cases to calculate, some 77 cases had to be run for each unit investigated.

Once a series of pump speeds has been run for a fixed Wrf/SKr and a given Fo/SKr, it is necessary to calculate the various design factors: S/Sp, F1/SKr, F2/SKr, F3/SKr, and 2T/SSKr. This was done by importing the output of OMNIROD into a Lotus 123 spread sheet. By use of Lotus macros, the importing of files and the calculation of the various design factors were easily accomplished. Also, graphs of the design factors vs non-dimensional speed can be quickly done. The results are summarized and shown in Figure 1-C to 5-C, Figure 1-M to 5-M, and Figure 1-T to 5-T.

PEAK TORQUE

Calculation of the peak torque factors (2T/SSKr) for the base case presented few problems. The program OMNIROD starts with a downhole card and generates the surface dynamometer card. The program then goes through an iteration procedure to determine the ideal counterbalance -- which by definition is where the up and down net peak torques are equal. To determine the peak torque factor, the calculated peak torque from OMNIROD is simply multiplied by two and divided by the stroke length squared (SS) and the rod elastic constant (Kr). The results are plotted for various non-dimensional fluid loads (Fo/SKr) against the non-dimensional speed (N/No). See Figures 5-C, 5-M, and 5-T.

Theoretically, no adjustments are needed for most load factors (Sp/S, F1/SKr, F2/SKr, and F3/SKr) even though the non-dimensional rod weight (Wrf/SKr) is different from 0.4. Basically, for a fixed non-dimensional fluid load (Fo/SKr) and non-dimensional speed (N/No) with a given shape downhole card, the surface dynamometer card will retain the same shape. The card will merely be positioned up or down due to the value of the rod weight in fluid (Wrf). (See API RP 11L - A13.) However, in the case of torque, the amount of rod load is important in determining the torque involved. If the non-dimensional rod weight (Wrf/SKr) is different from the base case, a torque adjustment (TA) must be made. In API RP 11L, the torque adjustment was determined as follows:

$$TA = 1 + AF * ((Wrf/SKr) - 0.3)/10$$

where AF was the adjustment factor percent found from Figure 4.6 for values of Wrf/SKr other than 0.3.

Use of this graph assumes that the adjustment factors are symmetrical and change directly in proportion to the difference from the base case. This is an over simplification for the cases investigated and can result in significant errors in calculated peak torques.

The torque adjustment is different for each geometry and requires checking of various non-dimensional rod weights (Wrf/SKr) for each non-dimensional fluid load (Fo/SKr). Conventional, Mark II, and Torqmaster unit calculations were based on non-dimensional rod weights of 0, .2, .4, .6, .8, and 1.0 and non-dimensional fluid loads (Fo/SKr) of .01, .1, .2, .3, .4, .5, and .6. These graphs are shown in Figure 6-C to 12-C, Figure 6-M to 12-M, and Figure 6-T to 12-T.

The torque adjustment (TA) is a straight forward determination; however, for each unit geometry a series of seven graphs is required. To find one TA, the program OMNIROD must be run for a particular N/No and Fo/SKr at a Wrf/SKr value different from 0.4. Once the peak torque is found, it is divided by the base torque that was previously determined. This must be done for all the 11 N/No values, the seven Fo/SKr values, and for the six Wrf/SKr values.

The torque adjustment (TA) normally requires interpolation between two graphs. Such an exercise is a little clumsy to do by hand but can be done rapidly on a computer.

RESULTS

The developed non-dimensional graphs provide the designer insight and a means of comparing the performance of the various geometry units. Most of the general conclusions that users of the API RP 11L reached are still valid. For example: The bottom hole stroke and loads increase with speed and increase more rapidly for higher N/No values. When fluid loads are low, the bottom hole stroke is normally greater. The minimum polished rod load factors are more erratic than other factors and do not change much due to the non-dimensional fluid load (Fo/SKr). Once the values of Fo/SKr reach 0.4, polished rod horsepower changes only slightly -- showing that larger pumps are more efficient. Also, at higher values of Fo/SKr, (.4), the torque adjustments are nearly equal for conventional units and the adjustment factor (AF) is minimal.

The non-dimensional graphs developed by using the wave equation OMNIROD for conventional units are, in general, very similar to the ones in API RP 11L. For the static condition (N/No = 0), the same values are calculated and the curves track each other up to N/No values of about 0.2. Thereafter, minor variations occur probably due to the influence of harmonics on the loads: variations are most noticeable at the second (N/No = .5), third (N/No = .33), and fourth (N/No = .25) order harmonics. The harmonic influence is more apparent if the non-dimensional factors are checked at more frequent speeds.

Conventional, Mark II, and Torqmaster comparisons can be easily made for each non-dimensional factor graph. As previously stated, the graphs of Sp/S, F1/SKr, F2/Skr, and F3/SKr have only minor variations. The Mark II has higher Sp/S factors at N/No values greater than 0.3 and also has higher F2/SKr and F3/SKr values at these relatively high non-dimensional speeds. See Figures 1-C to 4-C, 1-M to 4-M, and 1-T to 4-T. The Mark II has the "flattest" curves for the peak torque factors (2T/SSKr) - kicking up for N/No values greater than 0.35. At relatively low non-dimensional speeds (N/No values less than 0.2), the peak torque factors for all three units are reasonably close; thus, the special unit geometries (Mark II and Torqmaster) offer little advantage over Conventional geometry for such conditions. See Figures 5-C, 5-M, and 5-T.

The peak torque adjustment factor graphs for various Fo/SKr values are similar but have wider fluctuations for various unit types. The special geometry units require more adjustment at higher Fo/SKr values than the Conventional geometry unit. In fact, the Conventional unit requires only minor adjustments for Fo/SKr values greater than .2. Note that the lines for various non-dimensional rod weights (Wrf/SKr) are not symmetrical about the base case of Wrf/SKr = 0.4. For some cases, the "symmetry" may exist but, in general, symmetry about the base line does not occur.

EXAMPLE #1

A comparison of various design methods is shown in Table A. A relatively shallow well that was not pumped off was picked. In most such cases, the user attempts to get as much production as feasible without undue operating problems. Comparisons are shown in Column A for Conventional Units using API RP 11L, in Column B using procedures recommended by this paper, and in Column C by using the OMNIROD program. All three methods give slightly different answers. In this instant, the API RP 11L indicates a 160,000 in-lb gear box should be adequate but the other two methods show a 228,000 in-lb gear box would be needed.

The same type comparisons using a Mark II unit are shown in Columns D, E, and F. The API RP 11L calculation for a Conventional unit were modified as per the Lufkin formulas for a Mark II unit. See Column D.

Column E contains results from the new program ROD-MARK and Column F consists of results by using OMNIROD. Note that the API RP 11L shows a positive 1100 pounds for the minimum polished rod load but both ROD-MARK and OMNIROD show a negative load - indicating possible severe operating trouble. If these latter programs are correct, the Mark II unit would have to be run at a slower speed. Again, note that there is a fair amount of spread between most calculations.

The last comparison is for a Torqmaster unit. Program ROD-TM in Column G uses the graphs of this paper for a Torqmaster unit and OMNIROD shown in Column H is the wave equation program with a Torqmaster geometry. There are some minor differences but the same torque value is found indicating that a 160,000 inch-pound gear box could be used if the unit is operated near ideal counterbalance. In all cases, the API required counterbalance appears to be on the high side.

CONCLUSIONS

1. The wave equation can be used to develop non-dimensional graphs for various unit geometries, prime mover slip, and rod designs. The development of such graphs will permit better comparison of various special geometry units.

2. For the conditions selected, the developed graphs should be more accurate in most cases than the API RR11L.

3. The API torque adjustment procedure is over simplified and may lead to some serious sizing errors. Torque adjustment factor graphs can be developed that permit more accurate sizing.

4. The non-dimensional graphs developed are for specific conditions and deviation from these conditions may result in serious design errors. It is recommended that such graphs be developed for various geometry units. These graphs can then be used for general screening. For more precise answers, use of a good wave equation program adjusted for the design conditions should be considered.

NOMENCLATURE

CBE	Counterweight required, pounds
D	Plunger diameter, inches
E	Modulus of elasticity, psi
Fc	Frequency correction factor
Fo	Differential fluid load on full plunger area, pound
G	Specific gravity of produced fluid
H	Net lift, feet
Kr	Elastic constant of total rod string, pounds per inch
Kt	Elastic constant of tubing string, pounds per inch
L	Pump depth, feet
MPRL	Minimum polished rod load, pounds
N	Pumping speed, strokes per minute (spm)
No	Natural frequency of untapered rod string, spm
No'	Natural frequency of tapered rod string, spm
PD	Pump displacement, bpd
PPRL	Peak polished rod load, pounds

PRHP	Polished rod horsepower, hp
PT	Peak crank torque, inch-pounds
S	Surface polished rod stroke, inches
SKr	Pounds of load necessary to stretch the total rod string an amount equal to the polished rod stroke.
SPM	Strokes per minute
Sp	Bottom hole pump stroke, inches
W	Total weight of rods in air, pounds
Wr	Average unit weight of rods in air
Wrf	Total weight of rods in fluid, pounds

N/No	Nondimensional pump speed factor
Sp/S	Nondimensional stroke factor
F1/SKr	Non-dimensional PPRL factor
F2/SKr	Non-dimensional MPRL factor
F3/SKr	Non-dimensional PRHP factor
2T/SSKr	Non-dimensional PT factor
Wrf/SKr	Non-dimensional rod weight factor

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ACKNOWLEDGEMENT

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Table A

EXAMPLE # 1 OBJECT: TO COMPARE DESIGN METHODS FOR 2500 FT. WELL

ASSUMED DESIGN DATA:

PUMP DEPTH = 2500 FT	FLUID LEVEL = 2000 FT
PUMP SPEED = 15 SPM	PUMP DIAMETER = 2.25 IN.
STROKE LENGTH = 74 IN.	FLUID S.G. = 1.0
TUBING = 2.875 IN. OD	TUBING ANCHORED
API 66 TYPE C STEEL RODS	PUMP FILLAGE = .95%

CALCULATED NON-DIMENSIONAL FACTORS
Fo/SKr = 0.10 N/No = 0.153 Wrf/SKr = 0.105

RESULTS:

	A. API RP11L CONV.	B. ROD-CONV CONV.	C. OMNIROD CONV.	D. API RP11L MARK II	E. ROD-MARK MARK II	F. OMNIROD MARK II	G. ROD-TM TORQMASTER	H. OMNIROD TORQMASTER
BPD	629	594	590	629	585	574	574	572
PPRL	9533	10052	9796	8922	9764	9150	9835	9581
MPRL	1710	450	385	1100	-666	-104	8	217
PT/1000	158	190	187	133	129	125	156	156
PRHP	12.04	11.79	11.73	12.04	12.04	11.36	11.56	11.25
CBE	5692	5692	5090 **	5692	5692	4523 **	5692	4899 **
LOAD RATIO	.875 *	.986 *	.964 *	.842 *	1.021 *	.920 *	.989 *	.963 *

* LOAD RATIO IS THE PPRL STRESS DIVIDED BY THE PERMITTED MODIFIED GOODMAND STRESS FOR A SF = 1.0

** CALCULATED: CBE = (PPRL + MPRL)/2

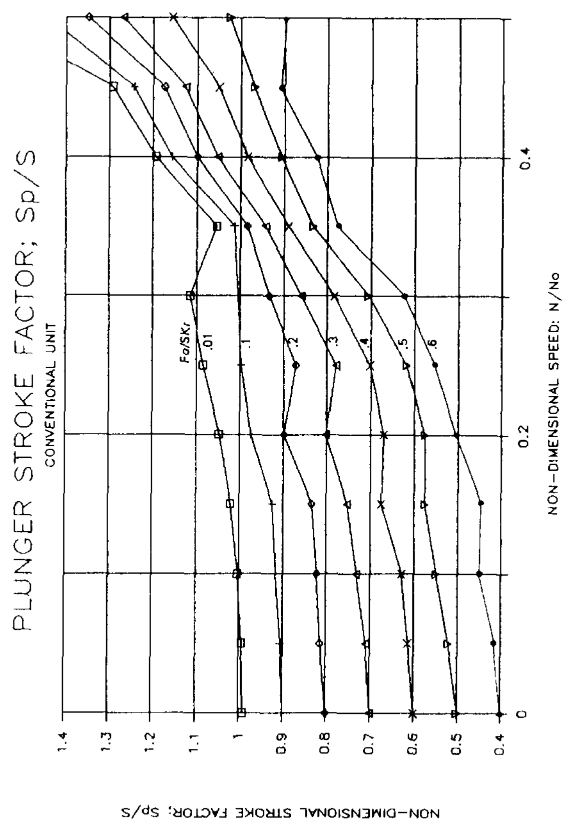


FIGURE 1-C

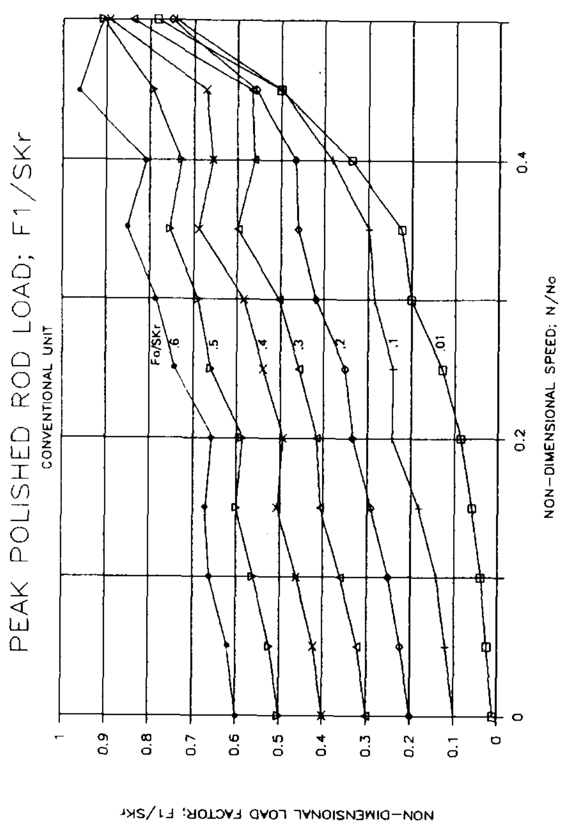


FIGURE 2-C

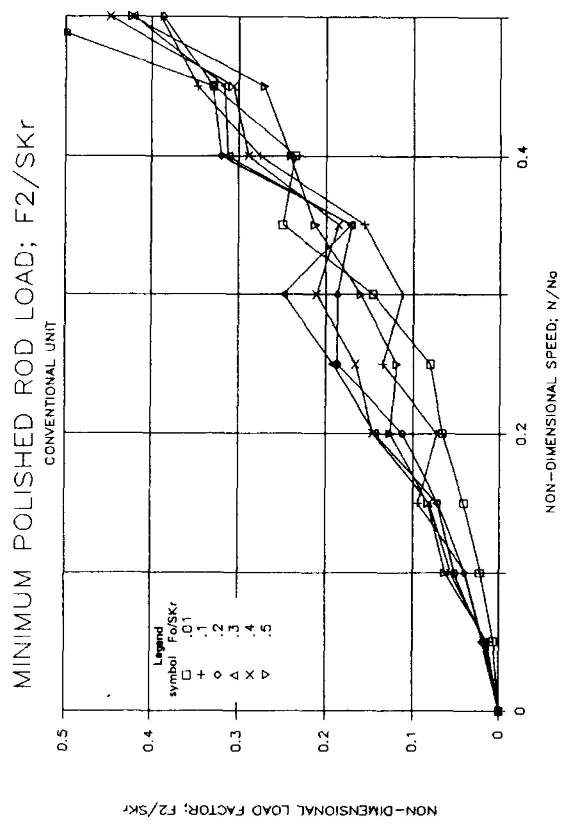


FIGURE 3-C

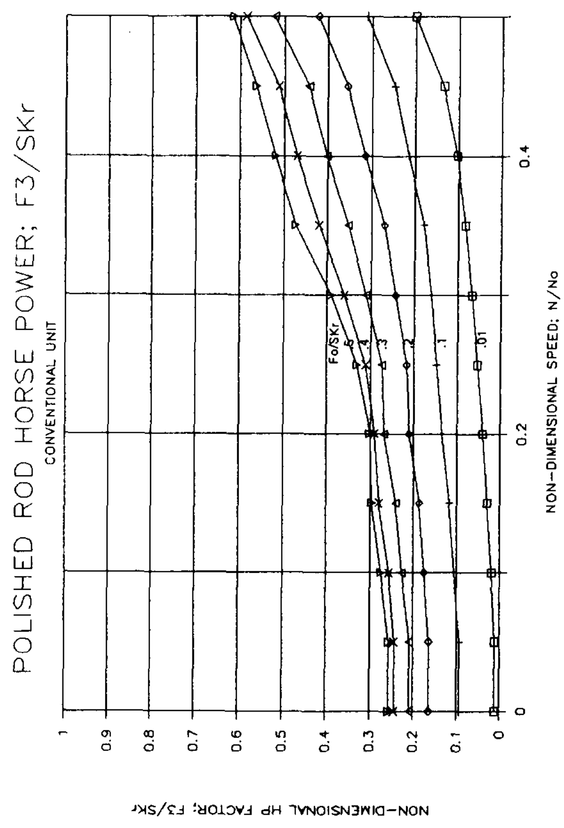
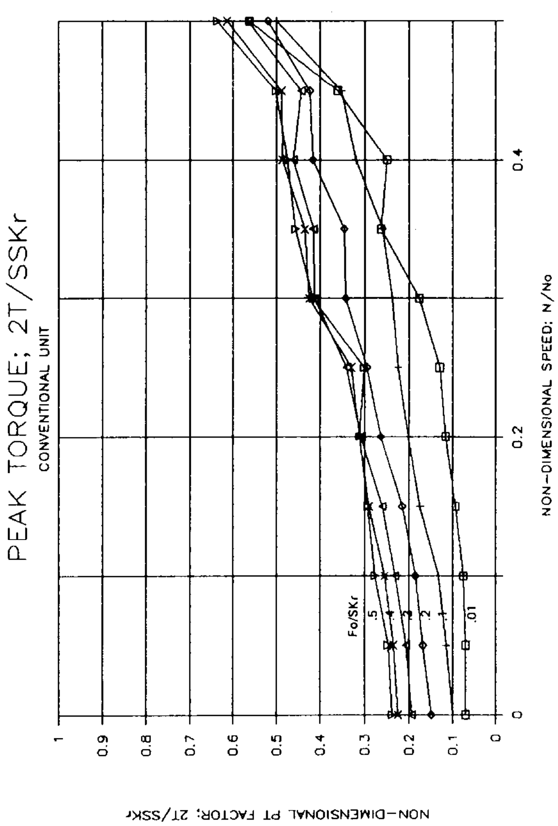


FIGURE 4-C



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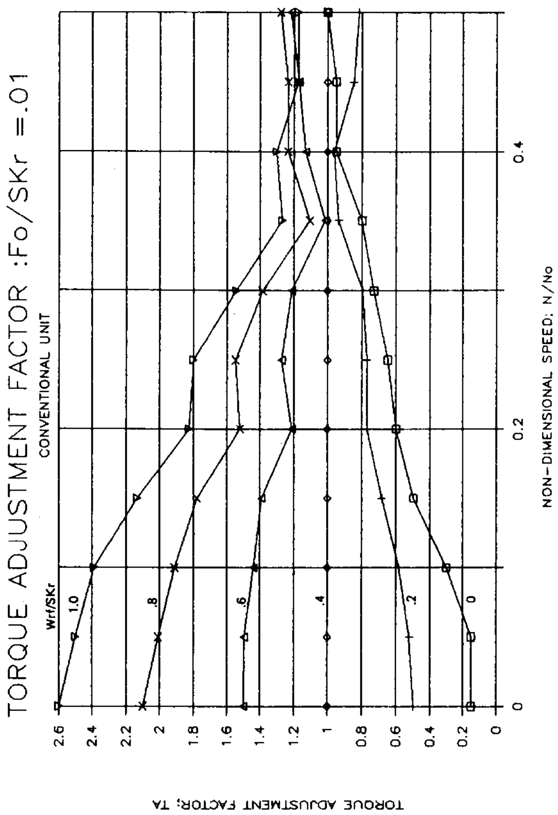

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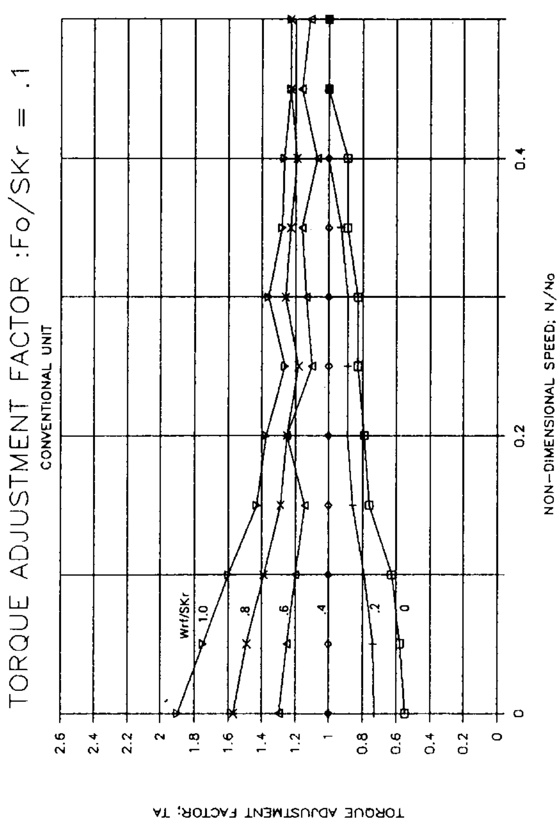


FIGURE 7-C

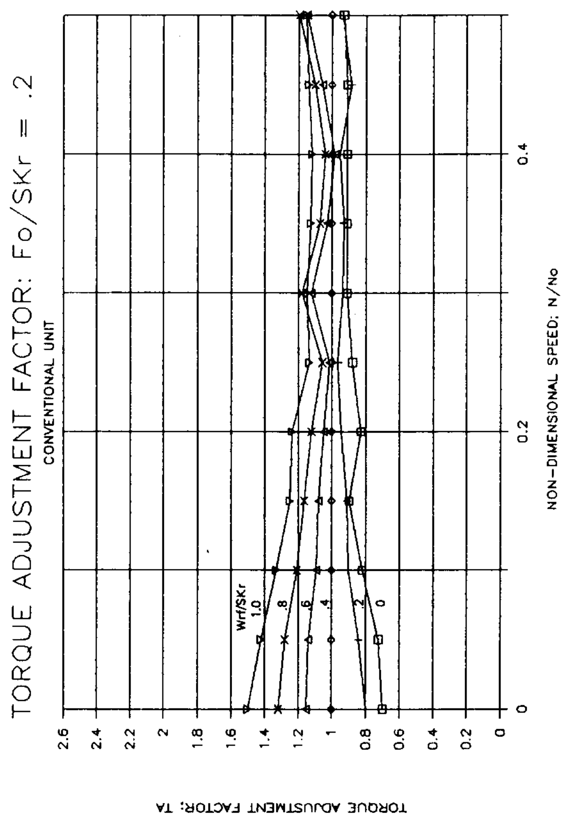
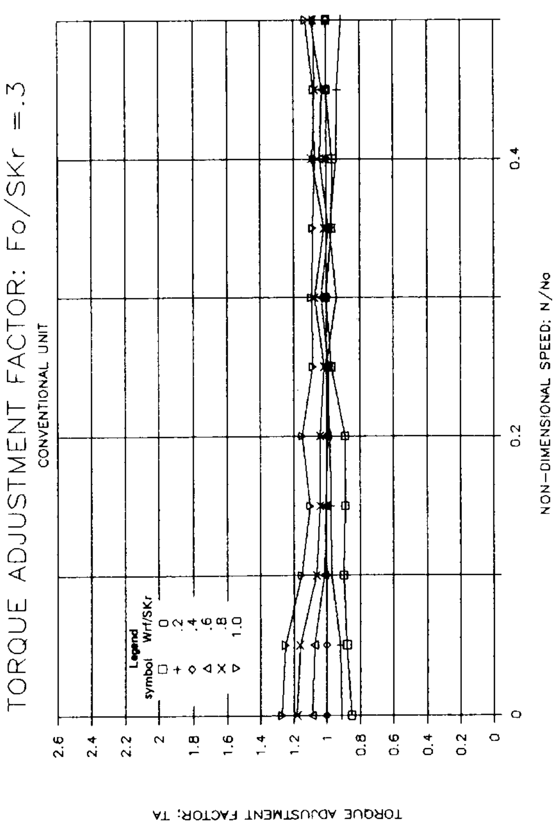


FIGURE 8-C



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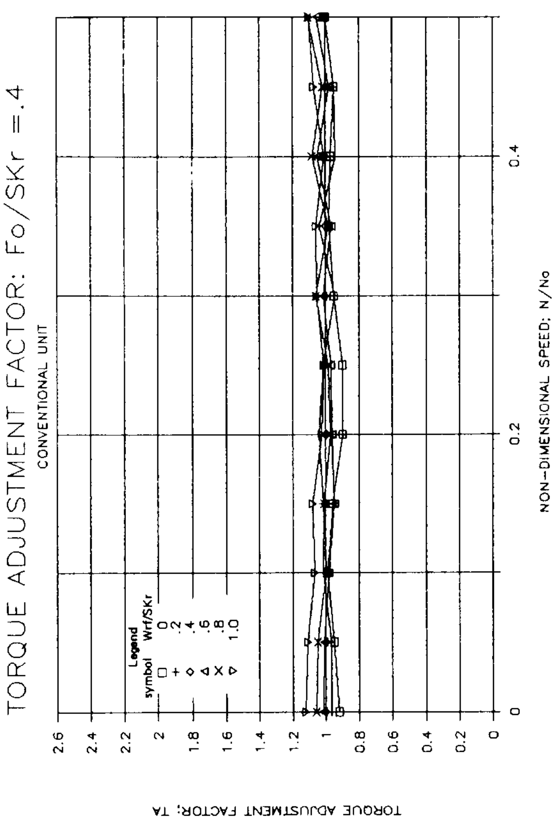

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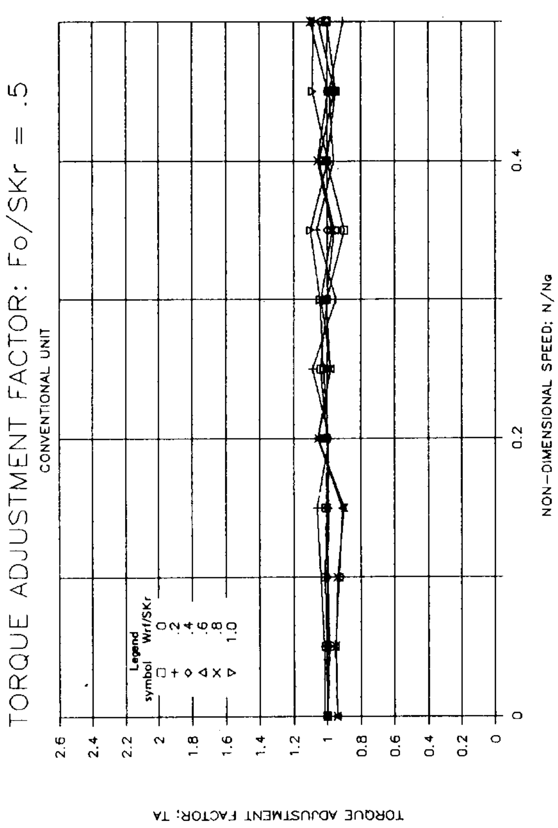
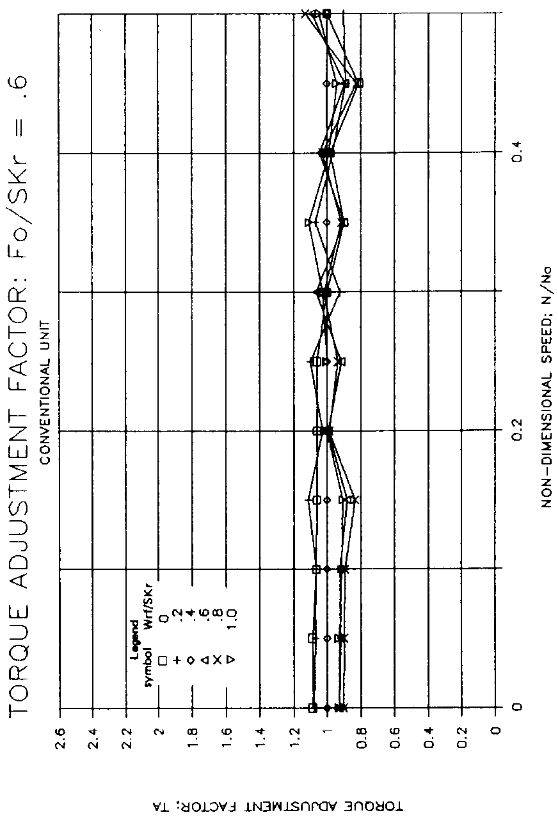


FIGURE 11-C



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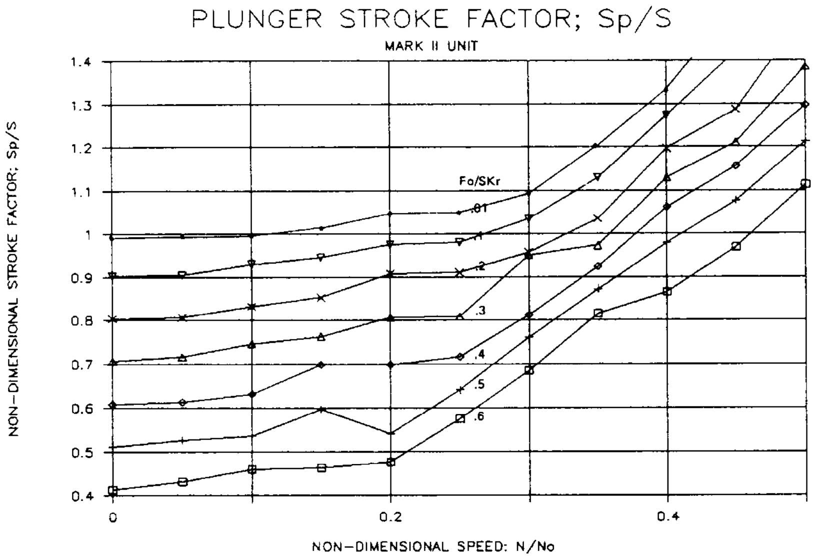

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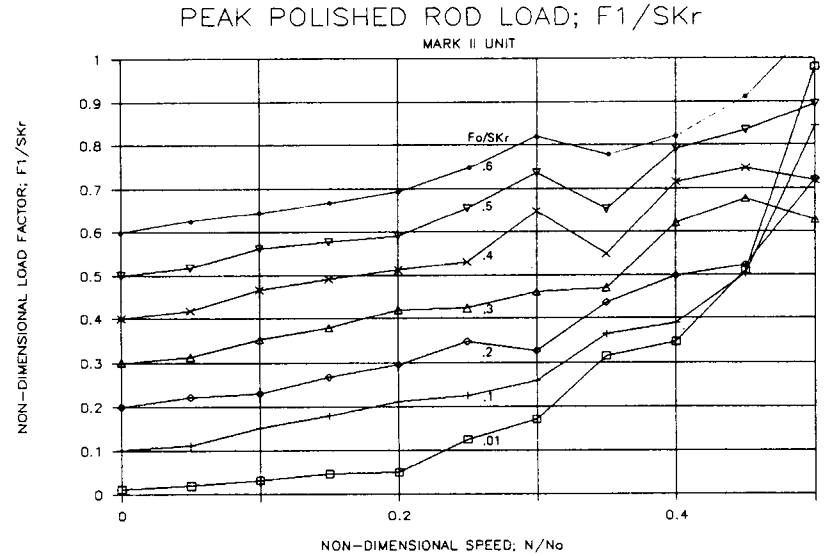

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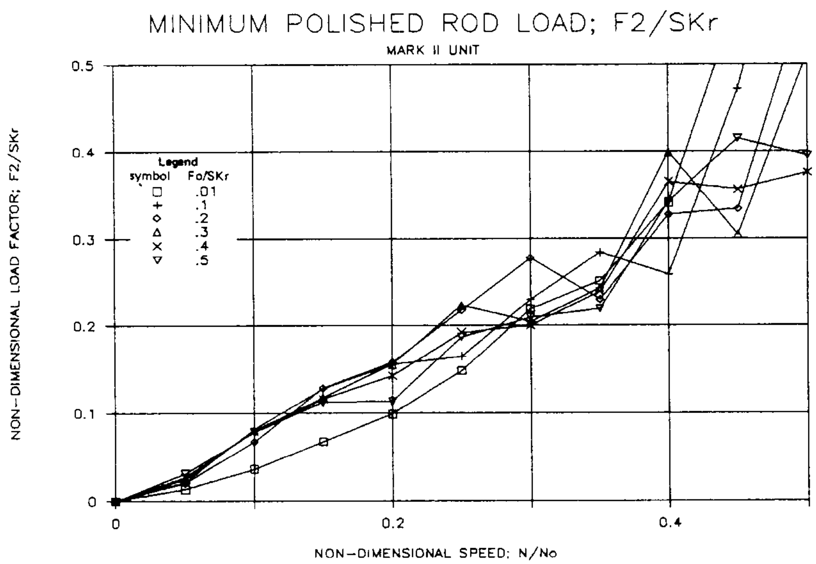

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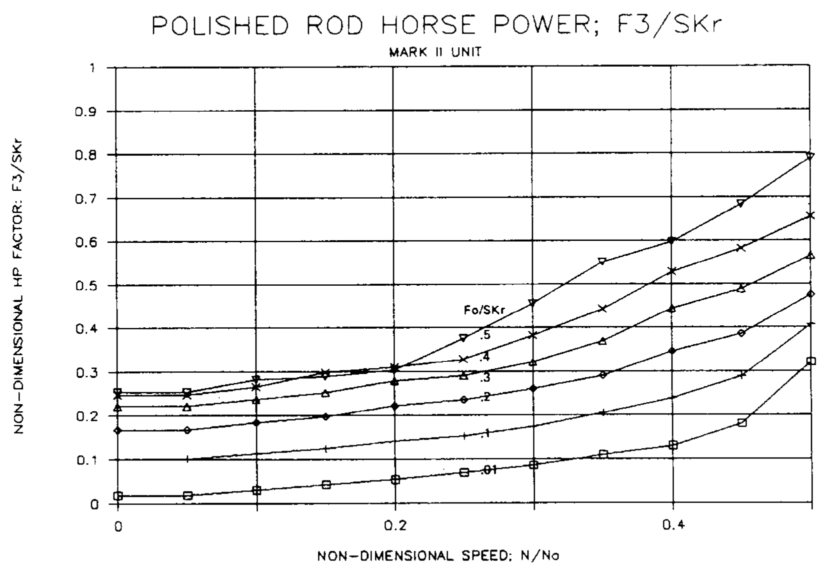

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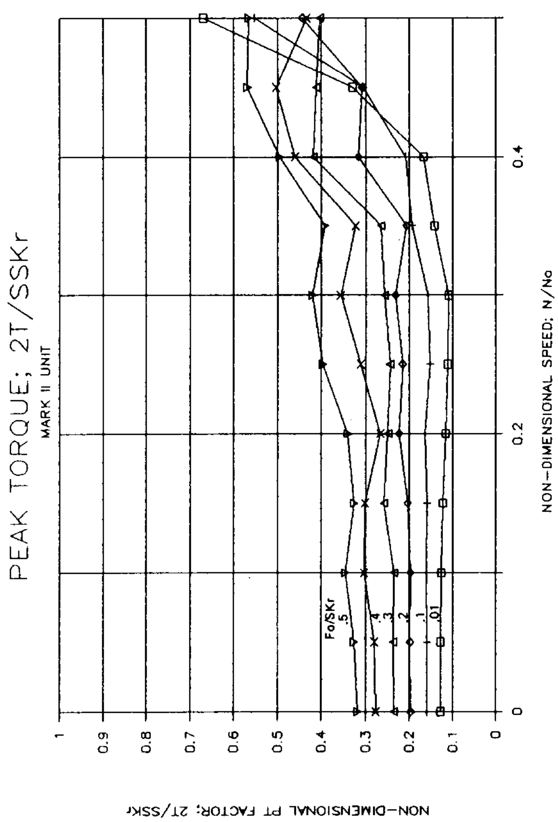

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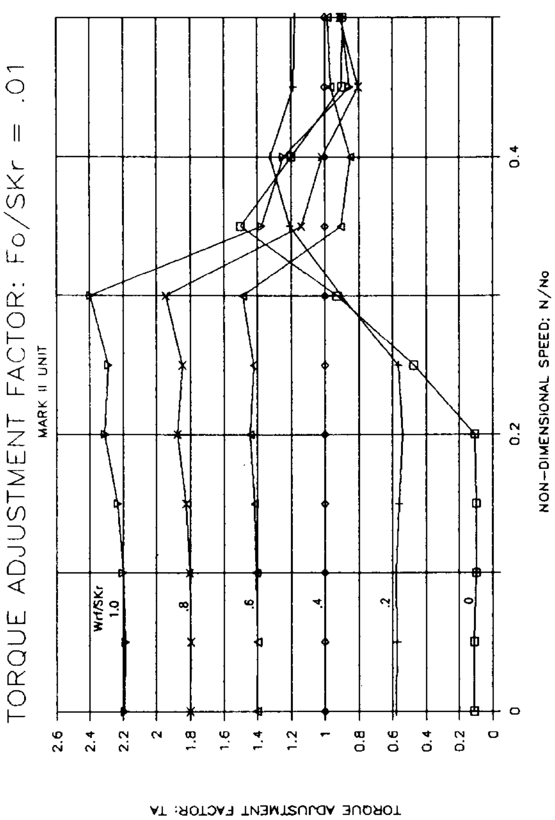

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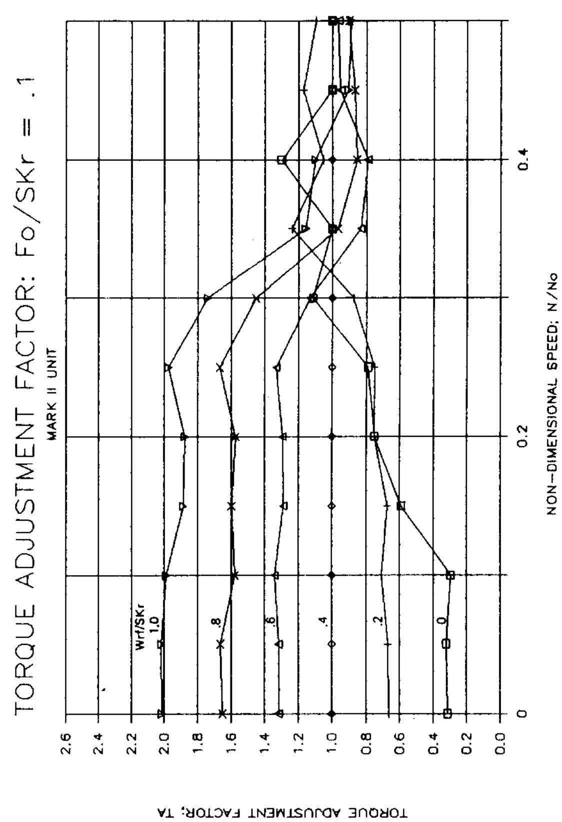

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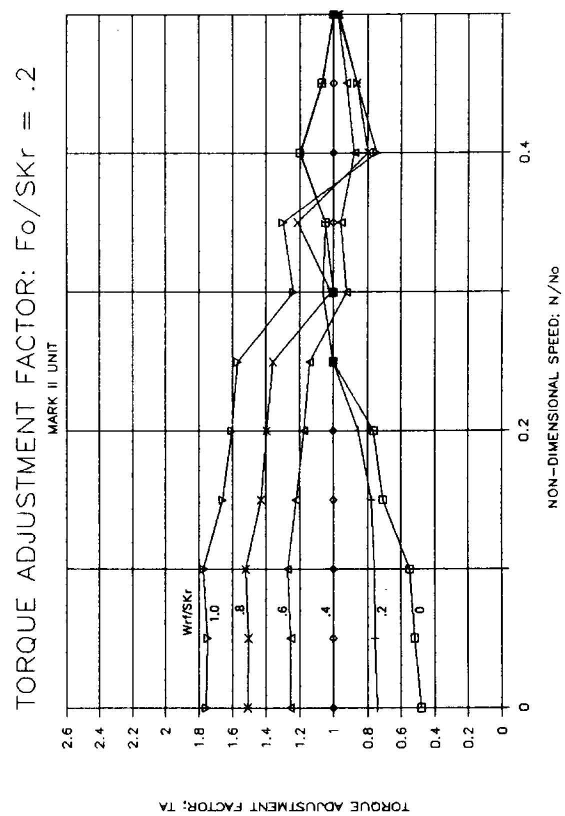

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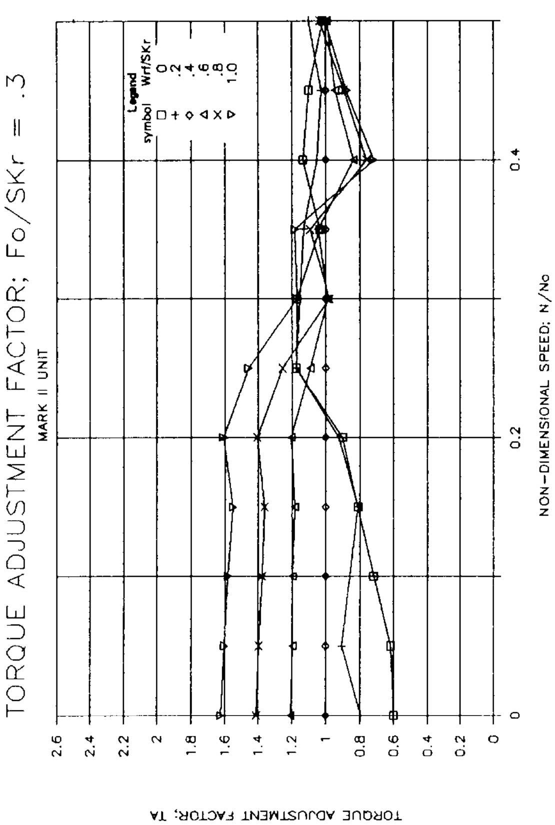

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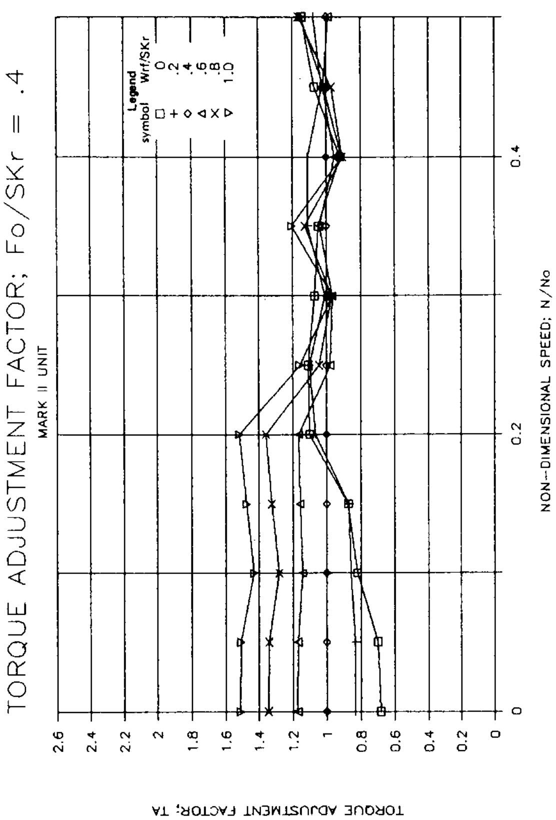

FIGURE 10-M

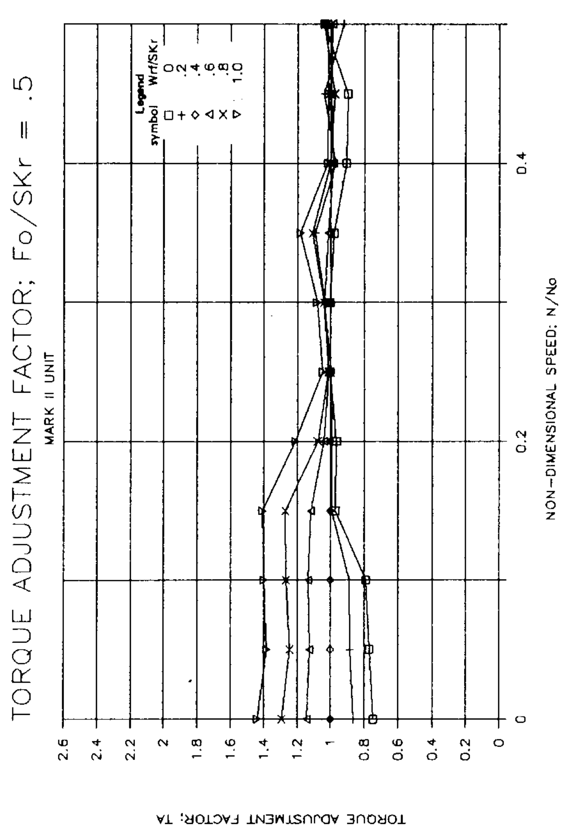

FIGURE 11-M

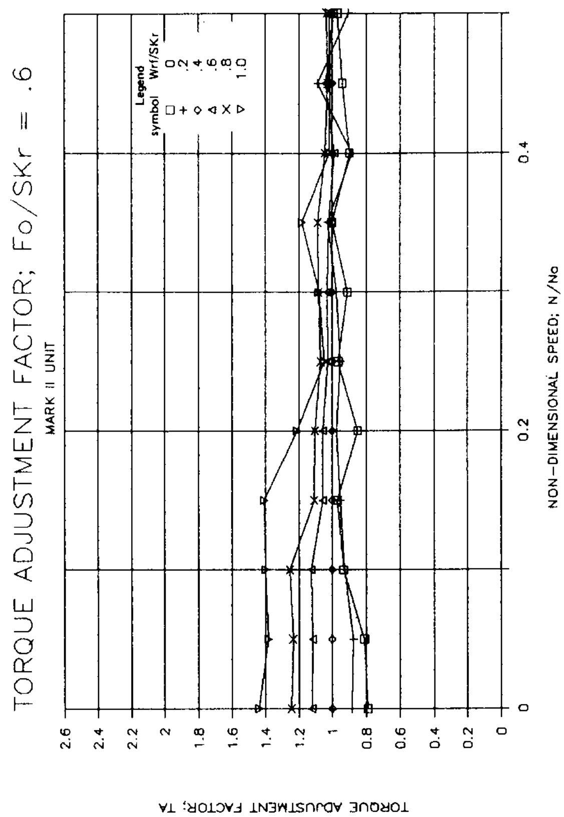

FIGURE 12-M

PLUNGER STROKE FACTOR; Sp/S

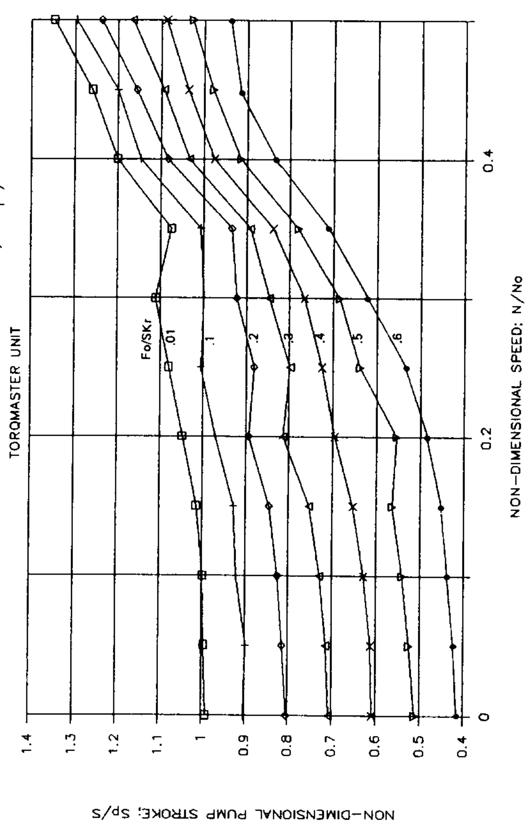

FIGURE 1-T

PEAK POLISHED ROD LOAD; F1/SKr

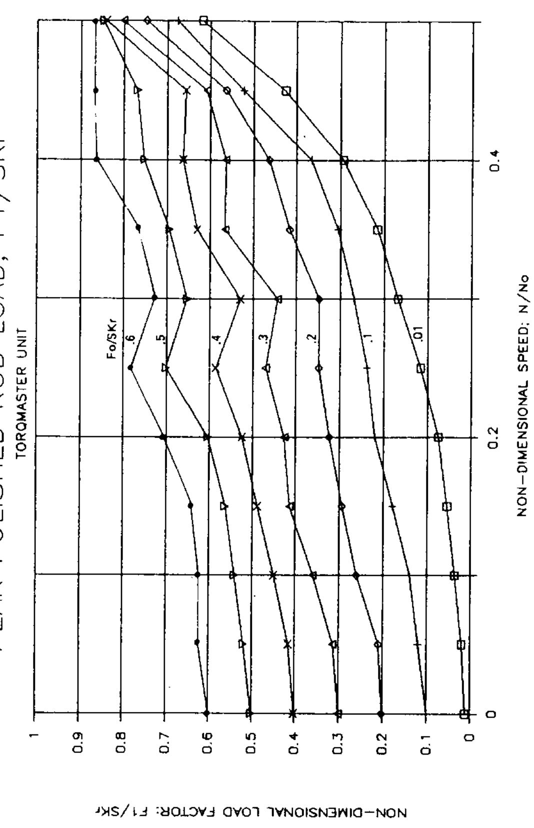

FIGURE 2-T

MINIMUM POLISHED ROD LOAD; F2/SKr

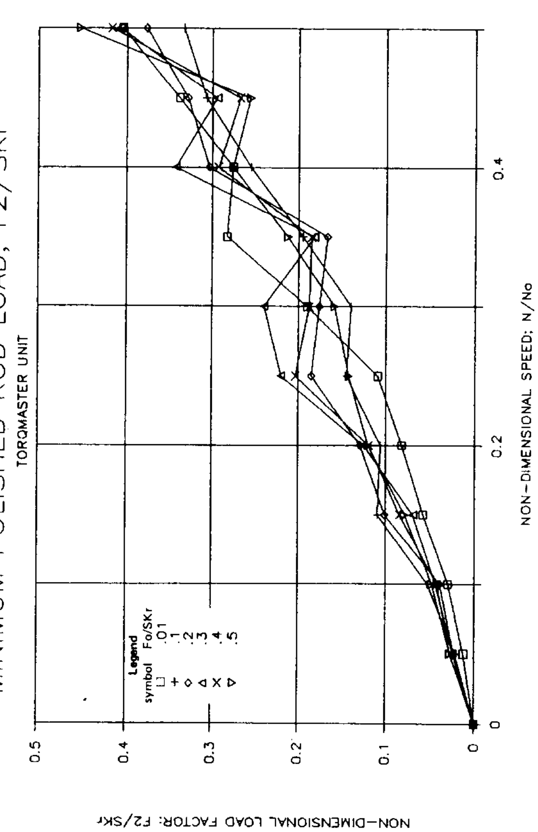

FIGURE 3-T

POLISHED ROD HORSE POWER; F3/SKr

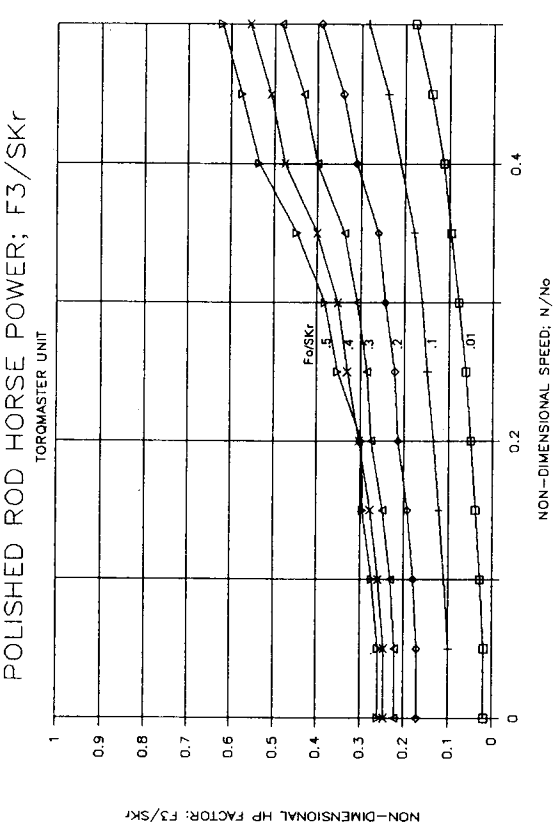

FIGURE 4-T

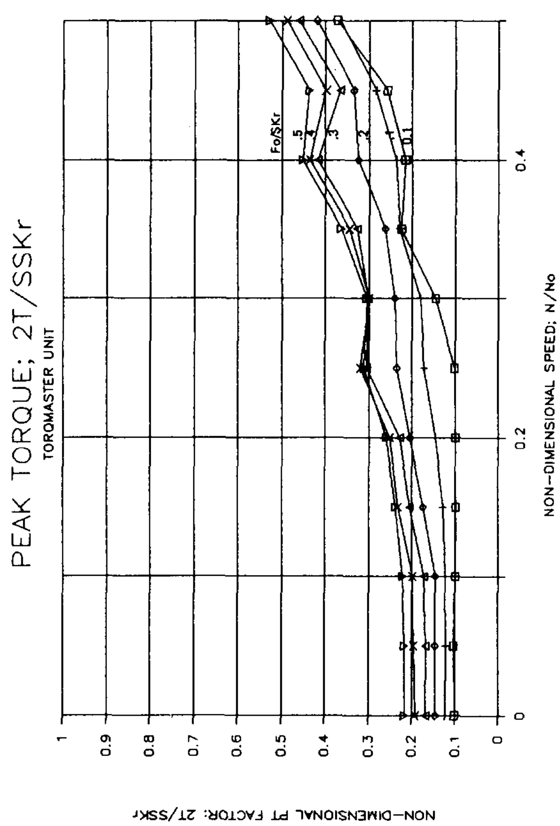

FIGURE 5-T

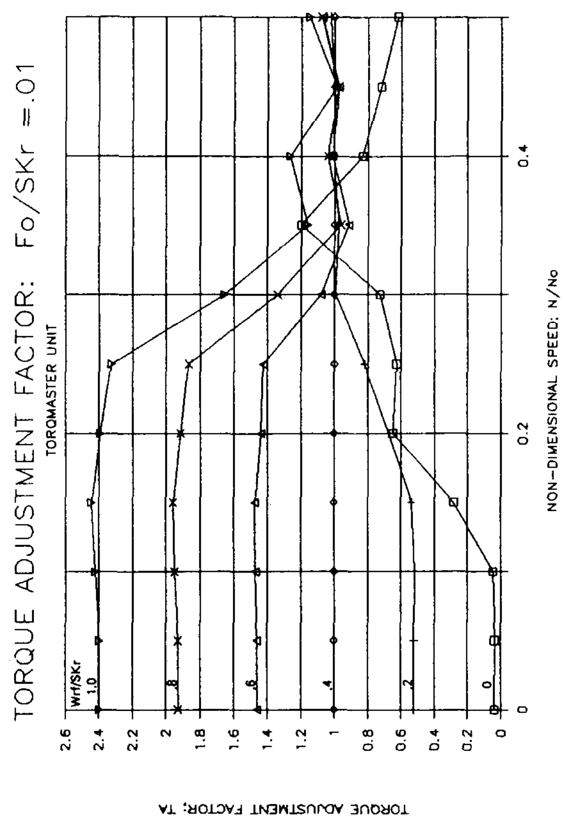

FIGURE 6-T

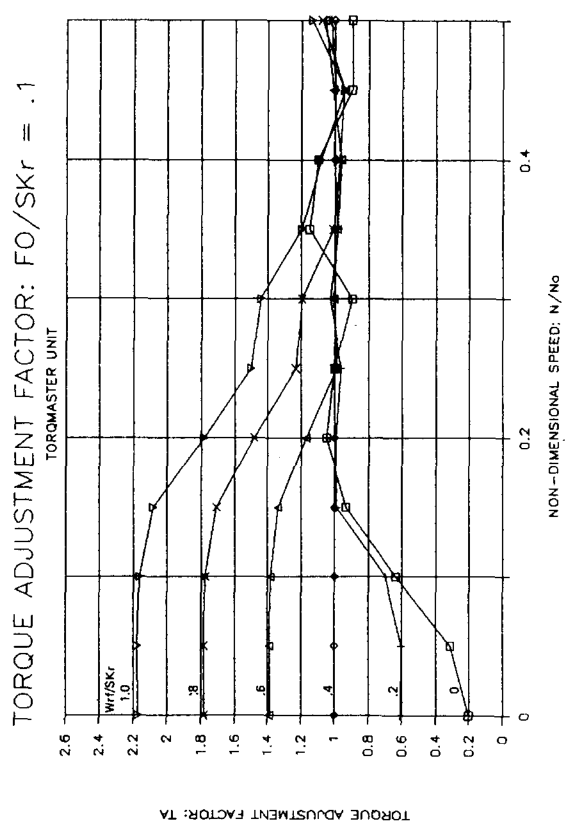

FIGURE 7-T

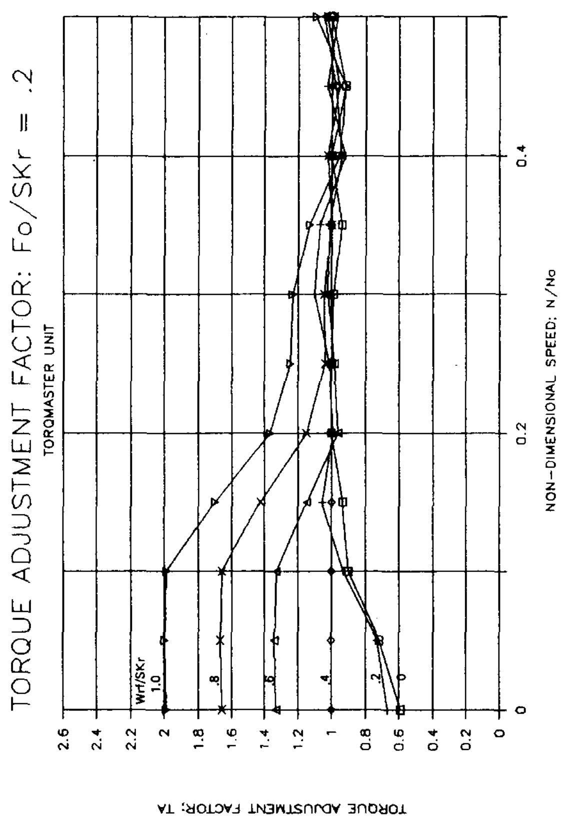

FIGURE 8-T

TORQUE ADJUSTMENT FACTOR: Fo/SKr = .3

TOROMASTER UNIT

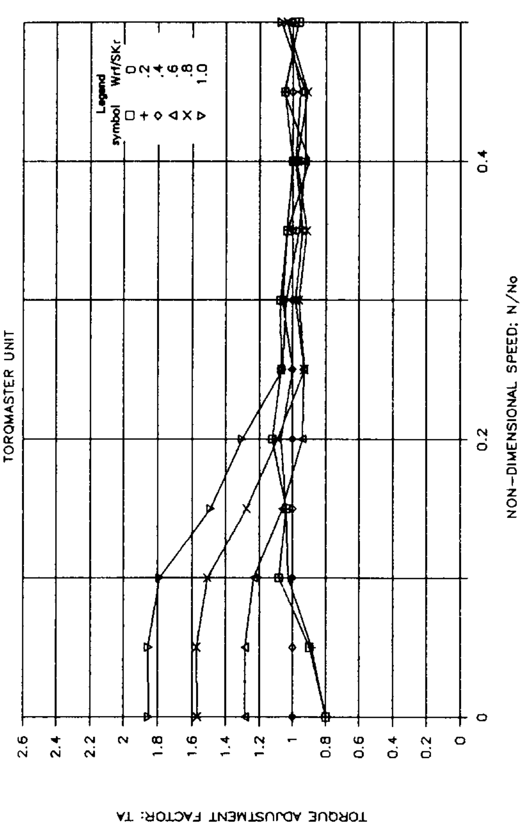

FIGURE 9-T

TORQUE ADJUSTMENT FACTOR: Fo/SKr = .4

TOROMASTER UNIT

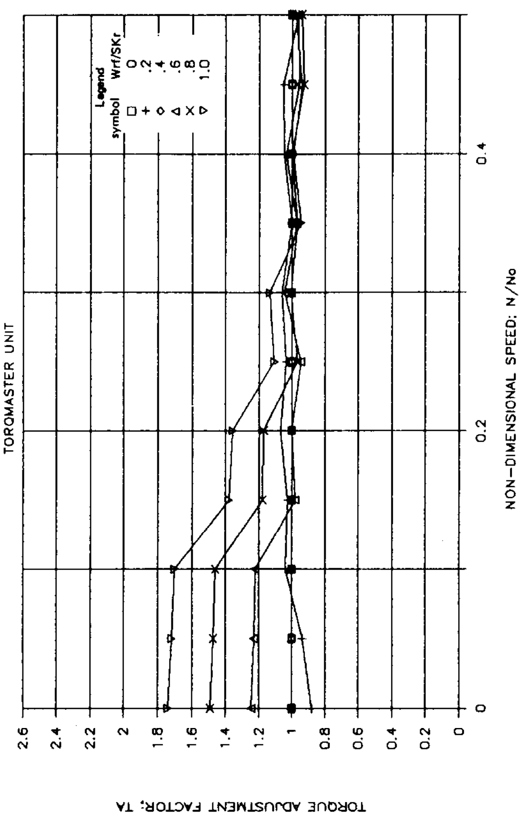

FIGURE 10-T

TORQUE ADJUSTMENT FACTOR: Fo/SKr = .5

TOROMASTER UNIT

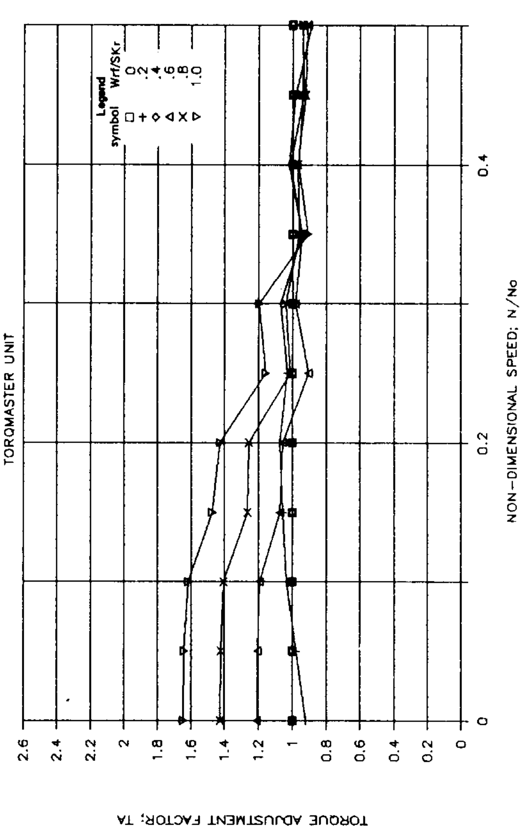

FIGURE 11-T

TORQUE ADJUSTMENT FACTOR: Fo/SKr = .6

TOROMASTER UNIT

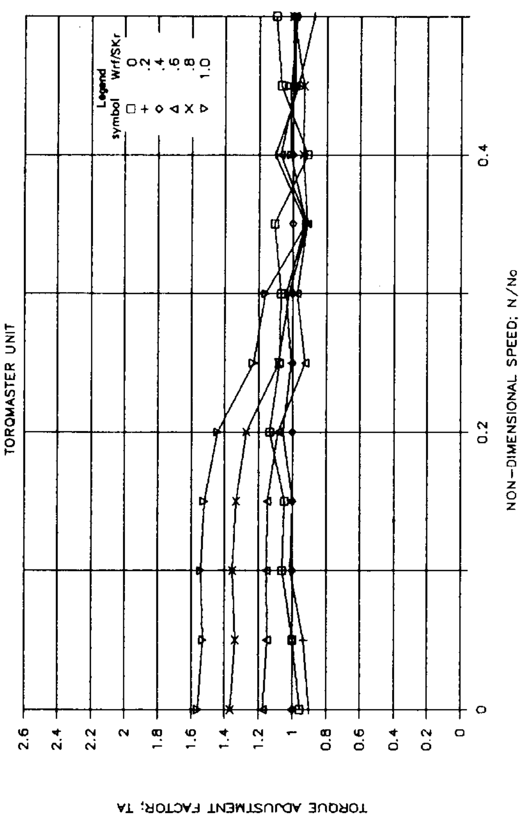

FIGURE 12-T