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Technical Report

No. 13443

COMBAT TRACKED VEHICLE FINAL DRIVE ANALYSIS

(PHASE I SBIR PROGRAM FINAL REPORT)

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PURPOSE

The purpose of this work effort was to furnish to TACOM an "Expert Design System" for the analysis of combat vehicle final drives. The system was to be tailored to account for the high stresses usually used in a military environment and to provide means of including a military duty cycle.

An existing final drive was analyzed for the purpose of comparing actual test data to the "Expert Design System" results. The design system was then "fine tuned" to account for the test data.

The existing final drive was then optimized for maximum life and reliability using the final form of the "Expert Design System."

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WORK ACCOMPLISHED

The following work was accomplished:

1. A design analysis of the M2/M3 final drive was done
2. Potential external influences on final drives were studied
3. Design optimization of the M2/M3 final drive life was done
4. An "Expert Design System" for final drive analysis was developed

POTENTIAL APPLICATION

The recommendations for optimization for the M2/M3 final drive should be considered since a considerable increase in life and reliability is indicated and the cost of the required changes is expected to be minimal.

The "Expert Design System" can be used to analyze any military final drive of the parallel axis gear-type where housing deflections are not significant or can be identified and torsional vibration is not significant or can be included in the duty cycle.

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1.0. INTRODUCTION

This technical report, prepared by Universal Technical Systems, Inc., for the U.S. Army Tank-Automotive Command (TACOM) under Contract DAAE07-88-C-R073, presents:

- A design study of potential problems and failure analysis of the M2/M3 final drive design.
- A design optimization of the M2/M3 final drive.
- An "Expert Design System" for military final drive analysis.

1.1 Design Analysis of the M2/M3 Final Drive

Geared transmission computer software originally formulated for commercial power transmission systems was used to analyze the M2/M3 final drive for three drive duty cycles furnished by TACOM and for two different gear sets. In addition, the specifications on the gear drawings made it necessary to consider both unground and ground gear teeth. Use of the software revealed potential problems in the M2/M3 final drive and provided specific recommendations for eliminating them.

Software correlation studies were based on data furnished by TACOM for two M2/M3 final drives subjected to a high torque test duty cycle. The test rig prevented any frame and housing deflections which might occur in vehicles operating over rough terrain. Also, the torsional vibration characteristics of a vehicle in the test rig are different from an operational vehicle. Evaluation of a possible need to modify the software is contingent on further testing to determine the effects of these external influences.

1.2. Design Optimization of the M2/M3 Final Drive

A design optimization of the M2/M3 final drive was undertaken using the "Expert Design System" software used in the design analysis of the M2/M3 Final Drive. The design parameters developed in the analysis of the existing final drives were used. The design analysis of the ground "MLRS" gears running with the 66,000 lb vehicle weight duty cycle was used as a "bench mark" for improvement.

1.3. "Expert Design System" for Military Final Drive Analysis

Development of an "Expert Design System" for military final drive analysis is a requirement of the contract. The methods developed during the analysis of two candidate final drives have been incorporated in the computer software system.

2.0 OBJECTIVES

The design analysis of the existing M2/M3 final drive had the primary goal of obtaining a correlation between the "Expert Design System" and actual field experience with:

- "Standard" gears with 50,000-lb vehicle duty cycle
  - Unground gear teeth
  - Ground gear teeth

- "MLRS" gears with 50,000-lb and 66,000-lb vehicle duty cycles
  - Ground gear teeth

- "MLRS" gears with "in-house" TACOM test duty cycle

The primary goal of the design optimization of the M2/M3 final drive was to increase the life of the drive by changes in the geometry of the gear teeth with little or no increase in the production cost of the drives. It was also desired to change dimensions and tolerances to allow operation at low temperatures.

The primary goal in developing the "Expert Design System" was to furnish to TACOM a set of computer software programs and instructions to enable TACOM engineers to analyze a military final drive design for suitability to perform a defined duty cycle.

3.0 CONCLUSIONS

The design analysis of the M2/M3 final drive led to the following conclusions:

- A change in the software code based on the "in-house" TACOM test duty cycle is not advisable.

- The increase in pitting and bending life of the ground gears over the unground gears is approximately 2.5 times.

- The strengths of the pinion and gear sets are not well "balanced" for equal bending fatigue life.

- The carburized case depth specified on the gear drawings is not enough to ensure that the case/core interface is safely below the depth to maximum sub-surface shear for some of the loads in the duty cycle.

- The operating backlash for both trains is insufficient to ensure safe operation at sub-zero temperatures.

- The probability of hot scoring of the high-speed train is unacceptable (58% to 87%) for the unground gears with a sump temperature of 180 °F.

- The probability of cold scoring of both trains is unacceptable (50% and 38%) for the unground gears with a sump temperature of 180 °F.

- The roller bearing on the intermediate shaft at the sprocket end has a much lower life than the other bearings. The life is, however, in the general range expected of the gears.

- The correlation between the results of the M2/M3 design studies and the TACOM test data was adequate with changes only in reliability factors to reflect military practice, method of estimating face mismatch, and surface finish. A change in the software code is therefore not advisable based on the TACOM test duty cycle and two test units; however, consideration of external influences led to the conclusions:

- Since the amount of gear misalignment caused by frame and housing deflection is not known and the torsional vibration characteristics of the test rig are different from an operational vehicle, the prediction accuracy of the software is not confirmed for field operation if these conditions are significant.
- If these external influences are not significant, the software is capable of good predictions of the suitability of final drives for a defined duty cycle.

The design optimization of the M2/M3 final drive indicated that a change in gear geometry can achieve an increase in final drive life with minimal or no increase in manufacturing costs:

- High Speed Train
  - Durability (Pitting): Net Increase in Life = 24%
  - Strength (Tooth Breakage): Net Increase in Life = 121%

- Low-Speed Train
  - Durability (Pitting): Net Increase in Life = 271%
  - Strength (Tooth Breakage): Net Increase in Life = 479%

  (NOTE: The onset of gear tooth pitting will not disable a vehicle, but tooth breakage will. An increase in life rating is not an increase in load rating. It requires relatively little change in load to change the life by large factors because of the flat character of the stress/cycle curves.)

- The change in life would be considerably more pronounced if the unground version of the "MLRS" gears and/or the optional flat root hobs were the "bench mark."

- The changes required do not require unusual materials or methods of manufacture. It would be necessary to specify the cutting edge geometry for the gear hobs and, depending upon the grinding method used, the geometry of the grinding wheel or grinding cams.

- H.S. Train, Hot Scoring: The hot scoring probability at the low end of the oil viscosity range decreases from 13% to 1% and at the high end from 3% to less than 1%.

- H.S. Train, Cold Scoring: The cold scoring probability remains at 6%.

- L.S. Train, Hot Scoring: The hot scoring probability remains at less than 1%.

- L.S. Train, Cold Scoring: The cold scoring probability increases from 5% to 9%. Since the scoring probabilities are less than 10%

they are not considered critical in the evaluation of the optimization. (If the unground gears were the "bench mark" scoring would be critical.)

- Low Temperature Operation: By changing the tooth thickness tolerance from +/-0.0015" to +/-0.001" and the center distance tolerance from +/-0.005" to +0.003" -0.000" it is possible to ensure operation with backlash for 95% of the drives after soaking in temperatures below -42 °F.

Based on the analysis of the M2/M3 final drive using a test duty cycle used at TACOM, a set of software and methods was developed which is suitable for TACOM engineers to use for analysis of final drives where housing deflection and torsional resonance are not significant.

4.0 RECOMMENDATIONS

The design analysis of the M2/M3 final drive led to the following recommendations:

- A change in the reliability factor from 1.0 (less than one failure in 100 units-commercial practice) to 0.9 (less than one failure in 20 units) to reflect military practice is advisable. (One "failure" in 20 units means that, out of 20 units, 19 units will run longer than predicted and 1 unit will not run as long as predicted.)

- When estimating the contact mismatch across the gear face it is recommended that the mean values of lead error, shafts out of plane and shafts out of parallel be used.

- It is recommended that the actual "run-in" values for gear surface finish be used for hot scoring and the listed values in Mobil Oil Corporation's EHL Guidebook, Third Edition, be used for cold scoring.

- Since the use of ground gears is an option on the gear drawings and the test gears examined were ground it is recommended that only ground gears be specified. A single tool geometry should also be specified instead of allowing two options.

- The pinions and gears should be adjusted for equal life in bending fatigue. (Equal life is maximum life for the drive.)

- The depth of the carburized case should be increased.

- The operating backlash should be increased to allow safe operation at cold temperatures. (The operating backlash problem is due to very wide tolerances on the tooth thickness of the gears and the housing center distances.)

- It is recommended that test data based on field operation of final drives be obtained prior to any changes in the software code or method of analysis outlined in the study.

- The test data should include dimensional inspection of the gears and housings of the test final drives.

The design optimization of the M2/M3 final drive led to the following recommendations:

- The optimization for drive life should be considered since the cost is expected to be minimal. No change in manufacturing methods is necessary compared to the ground production test gears examined by TACOM personnel. The major changes involve new perishable gear tools (hobs and grinding wheels).

- The changes in tolerance to ensure low temperature operation may increase manufacturing cost slightly but are mandatory if cold temperatures are encountered in service.

5.0 DESIGN ANALYSIS OF THE M2/M3 FINAL DRIVE

5.1 "Standard" Gears with 50,000-lb Duty Cycle

An analysis of both "standard" trains in the drive was made using the duty cycle for 50,000-lb vehicle weight furnished by TACOM.

The drawings of the "standard" gears indicate a rack form for generating the gears and, in all but one case, an alternate rack form. In addition, the specifications allow grinding the teeth as an option. The difference between gears that are put into service without post processing (grinding, honing, etc.) and gears which are ground can be very significant even though both gears meet the inspection tolerances. The tolerances were checked to find the approximate AGMA Q class for all 4 gears.

For the H.S. Train 19-tooth pinion:

12276787 Unground

```
==================== VARIABLE SHEET ==================================
St Input---- Name--- Output--- Unit----- Comment----------------------
   19        N                           Number of teeth
   3.5       Pnd                1/in     Normal pitch
   0         psi                deg      Helix angle
   1.75      F                  in       Face width

   .0035     VrT                in       Radial Runout Tolerance (TIR)
             QRUN    9                   Runout Quality Q#

   .0008     VpA                in       Allowable Pitch Variation +/-
             QPIT    9                   Pitch Quality Q#
```

```
   .0012       VoT                   in         Profile Tolerance
               QPRO     9                       Profile Quality Q#

   .0006       VyT                   in         Tooth Alignment Tolerance
               QLD      9                       Alignment Quality Q#
```

For the H.S. Train-32 tooth gear:

```
12291984 Unground
=================== VARIABLE SHEET ==================================
St Input---- Name--- Output--- Unit----- Comment----------------------
   32          N                                Number of teeth
   3.5         Pnd                   1/in       Normal pitch
   0           psi                   deg        Helix angle
   1.582       F                     in         Face width

   .004        VrT                   in         Radial Runout Tolerance (TIR)
               QRUN     9                       Runout Quality Q#

   .0008       VpA                   in         Allowable Pitch Variation +/-
               QPIT     10                      Pitch Quality Q#

   .0013       VoT                   in         Profile Tolerance
               QPRO     9                       Profile Quality Q#

   .0006       VyT                   in         Tooth Alignment Tolerance
               QLD      9                       Alignment Quality Q#
```

For the L.S. Train 18-tooth pinion:

```
12292025 Unground
=================== VARIABLE SHEET ==================================
St Input---- Name--- Output--- Unit----- Comment----------------------
   18          N                                Number of teeth
   3.5         Pnd                   1/in       Normal pitch
   0           psi                   deg        Helix angle
   3.5         F                     in         Face width

   .004        VrT                   in         Radial Runout Tolerance (TIR)
               QRUN     8                       Runout Quality Q#

   .0009       VpA                   in         Allowable Pitch Variation +/-
               QPIT     9                       Pitch Quality Q#

   .0014       VoT                   in         Profile Tolerance
               QPRO     8                       Profile Quality Q#

   .001        VyT                   in         Tooth Alignment Tolerance
               QLD      9                       Alignment Quality Q#
```

For the L.S. Train 53-tooth gear:

12292079 Unground

==================== VARIABLE SHEET ====================================

St	Input----	Name---	Output---	Unit-----	Comment----------------------
	53	N			Number of teeth
	3.5	Pnd		1/in	Normal pitch
	0	psi		deg	Helix angle
	2.88	F		in	Face width
	.005	VrT		in	Radial Runout Tolerance (TIR)
		QRUN	9		Runout Quality Q#
	.0011	VpA		in	Allowable Pitch Variation +/-
		QPIT	9		Pitch Quality Q#
	.0016	VoT		in	Profile Tolerance
		QPRO	9		Profile Quality Q#
	.0014	VyT		in	Tooth Alignment Tolerance
		QLD	7		Alignment Quality Q#

All 4 gears fall generally into AGMA class Q9. Since the computer software used for load analysis considers the tooth alignment tolerance (lead tolerance) separately, class Q9 was used for the unground gears.

The same gears after being ground will usually be at least AGMA class Q11. Class Q11 gears would, of course, meet the limits on the drawings but would be, in general, much closer than the limits. If some gears are ground and some unground it will cause difficulties in field evaluation of the drives. The drives with ground gears will exhibit better life and reliability than the drives with unground gears although all drives meet inspection limits.

Since it is not known what type of gears and which tooth forms are being used it was decided to analyze both ground gears with full fillet roots and unground gears with flat roots. The deviations for Q11 gears were estimated to be used in the comparison. Figures 5-5 through 5-8 show the results of determing the tolerances from a known gear classification.

For the H.S. Train 19-tooth pinion:

12276787 Estimated Ground

==================== VARIABLE SHEET ====================================

St	Input----	Name---	Output---	Unit-----	Comment----------------------
	11	Q			AGMA Quality Number
	19	N			Number of teeth
	3.5	Pnd		1/in	Normal pitch
	0	psi		deg	Helix angle
	1.75	F		in	Face width
		VrT	.0017	in	Radial Runout Tolerance (TIR)
		QRUN	11		Runout Quality Q#

```
VpA      .00047     in         Allowable Pitch Variation +/-
QPIT     11                    Pitch Quality Q#

VoT      .00059     in         Profile Tolerance
QPRO     11                    Profile Quality Q#

VyT      .00039     in         Tooth Alignment Tolerance
QLD      11                    Alignment Quality Q#
```

For the H.S. Train 32-tooth gear:

```
12291984 Estimated Ground
=================== VARIABLE SHEET ================================
St Input---- Name--- Output--- Unit----- Comment---------------------
             Q                           AGMA Quality Number
             m                           Message-Quality Number
   32        N                           Number of teeth
   3.5       Pnd                1/in     Normal pitch
   0         psi                deg      Helix angle
   1.75      F                  in       Face width

   .004      VrT                in       Radial Runout Tolerance (TIR)
             QRUN    9                   Runout Quality Q#

   .00051    VpA                in       Allowable Pitch Variation +/-
             QPIT    11                  Pitch Quality Q#

   .00064    VoT                in       Profile Tolerance
             QPRO    11                  Profile Quality Q#

   .00039    VyT                in       Tooth Alignment Tolerance
             QLD     11                  Alignment Quality Q#
```

For the L.S. Train 18-tooth pinion:

```
12292025 Estimated Ground
=================== VARIABLE SHEET ================================
St Input---- Name--- Output--- Unit----- Comment---------------------
            Q                           AGMA Quality Number
             m                           Message-Quality Number
   18        N                           Number of teeth
   3.5       Pnd                1/in     Normal pitch
   0         psi                deg      Helix angle
   3.5       F                  in       Face width

   .0037     VrT                in       Radial Runout Tolerance (TIR)
             QRUN    9                   Runout Quality Q#

   .00046    VpA                in       Allowable Pitch Variation +/-
             QPIT    11                  Pitch Quality Q#
```

.00059	VoT		in	Profile Tolerance
	QPRO	11		Profile Quality Q#
.00064	VyT		in	Tooth Alignment Tolerance
	QLD	11		Alignment Quality Q#

For the L.S. Train 53-tooth gear:

12292079 Estimated Ground

==================== VARIABLE SHEET ====================================

St Input----	Name---	Output---	Unit-----	Comment----------------------
11	Q			AGMA Quality Number
	m	'OK		Message-Quality Number
53	N			Number of teeth
3.5	Pnd		1/in	Normal pitch
0	psi		deg	Helix angle
2.88	F		in	Face width
	VrT	.0022	in	Radial Runout Tolerance (TIR)
	QRUN	11		Runout Quality Q#
	VpA	.00056	in	Allowable Pitch Variation +/-
	QPIT	11		Pitch Quality Q#
	VoT	.00069	in	Profile Tolerance
	QPRO	11		Profile Quality Q#
	VyT	.00056	in	Tooth Alignment Tolerance
	QLD	11		Alignment Quality Q#

The analysis was done on "nominal" gears to obtain a comparison between the ground and unground gears. Nominal means that the split limit was used on tooth thickness, ODs, etc. and the design center distances on the gear drawings were used.

A "Reliability Factor" of 0.9 was used. A factor of 0.9 results in less than 1 "failure" out of 20 drives. "Failure" means that 1 drive out of 20 will run less than the calculated life and 19 drives out of 20 will run longer than the calculated life. Commercial drives are usually designed for a 1 in 100 failure rate. A failure rate of 1 in 20 was used for military duty. This failure rate correlated well with an analysis of a TACOM test duty cycle performed under this contract. (See paragraph 5.3, "'MLRS' Gears with 'In-House' TACOM Test Duty Cycle.")

The surface finish limit set on the gear drawings is 63 microinches. While this finish is more or less standard for milled or hobbed finishes it is doubtful that any production gears have been produced with a finish this rough.

If the gears are shaved before heat treat (it is difficult to get class Q9 by hobbing only) the finish would be about 35 or 40 microinches after heat treat. After some running in shaved gears may be about 30 microinches; therefore, 30 microinches was used in the hot scoring calculations for the unground gears.

For the ground gears the pinion tooth surface should be no more than about 20 microinches (and may be as low as 10 microinches) after break-in. The gear should be no more than about 25 microinches (and may be as low as 15 microinches) after break-in. For hot scoring calculations, 20 microinches was used for the pinion and 25 for the gear.

It is recommended that the listed values in Mobil Oil Corporation's EHL Guidebook, Third Edition, be used for cold scoring calculations because the methods and equations were calibrated for these values. For the unground gears 28 microinches was used and for the ground gears 14 microinches was used.

5.1.1 High-Speed Train. The duty cycle table used for the "Miner's Rule" life predictions for the high-speed train was taken from "Original 500 Spec," Schedule A, furnished by TACOM. (Pinion torque is in lb-in.)

==== MINER'S RULE ====

Cond #	Pin RPM	Horsepower	Pin Tork	Mins	Pin-Cycles	Gear-Cycles
1	250	54.05	13620	20	5000	2968.75
2	250	30.48	7680	30	7500	4453.13
3	300	93.14	19560	10	3000	1781.25
4	1100	105	6060	10	11000	6531.25
5	900	75.43	5280	30	27000	16031.3
6	550	85.9	9840	20	11000	6531.25
7	1800	116	4080	20	36000	21375
8	450	144	20280	10	4500	2671.88
9	900	114	8040	30	27000	16031.3
10	2100	140	4200	30	63000	37406.3
11	1600	131	5160	20	32000	19000
12	1100	152	8760	30	33000	19593.8
13	2500	166	4200	30	75000	44531.3
14	1200	166	8760	30	36000	21375
15	3100	481	9780	20	62000	36812.5
16	300	74.29	15600	10	3000	1781.25
17	300	80	16800	10	3000	1781.25
18	900	86.57	6060	10	9000	5343.75
TOTALS				370	448000	266000

5.1.1.1 Bending strength and surface durability. UTS Gear Analysis program #500 was used to obtain the I and J factors for the high-speed gears. A semi-topping hob was used to simulate a nominal corner break at the tooth tips. The stress correction factor, Kf, is not the standard AGMA factor. The optional modified Kf in the program uses the radius of curvature where the J factor (and stress) is calculated, while the standard AGMA Kf uses the radius of curvature of the fillet at the root of the tooth. The output sheets and plots, labeled "H.S. Train-Unground Nominal" and "H.S. Train-Ground Nominal" are attached as Appendices A and B, respectively.

An estimate of the mismatch across the face of the gears is required along with a face mismatch factor, Cmf. An estimate of the mismatch was made from the lead errors allowed on the gears and the shaft misalignment allowed by the housing specifications. The mean misalignment was calculated as this correlated well with an analysis of a TACOM test duty cycle performed under this contract. (See paragraph 5.3, "'MLRS' Gears with 'In-House' TACOM Test Duty Cycle.") UTS Program #60-5406 (TK) was then used to calculate an "equivalent Cmf" for each load condition in the duty cycle to include the effect of the crown on the 32-tooth gear.

UTS Program #540 was run to obtain life predictions for the unground and ground high-speed train subjected to the duty cycle.

Program #540 Summary Sheet - Unground H.S. Gears

	Number of Duty Cycles
PINION PITTING:	
Life= 1003 hours	161+
to 838 hours	135+
PINION BENDING STRENGTH:	
Life Is More Than 100,000 hours	16000+
GEAR PITTING:	
Life= 1698 hours	272+
to 20422 hours	3293+
GEAR BENDING STRENGTH:	
Life Is More Than 100,000 hours	16000+

Program #540 Summary Sheet - Ground H.S. Gears

	Number of Duty Cycles	% of Unground Gear Life
PINION PITTING:		
Life= 2360 hours	380+	235%
to 85430 hours	13779+	704%
PINION BENDING STRENGTH:		
Life Is More Than 100,000 hours	16000+	----
GEAR PITTING:		
Life= 3975 hours	641+	235%
To more than 100000 hours	16000+	----
GEAR BENDING STRENGTH:		
Life Is More Than 100,000 hours	16000+	

NOTE: One duty cycle is 6.2 hours.

A range is given for the life of the gears if less than 100,000 hours. This is necessary as both values of Sac and Sat from Tables 5 & 6 of AGMA 218 have been used by the program. This range can be extensive due to the rapid change of cycles with the load. (See Fig. 20 & 21 of AGMA 218.) The higher values may be used if special care is used in gearbox design, manufacture, and heat treatment. The minimum case depth to the 50 Rc/C point specified on the gear drawings (0.055") is not enough to stay safely below the depth to maximum sub-surface shear for some of the duty cycle conditions. Since this is the case use of the higher life values for these gears is questionable.

Suggested Minimum Effective (50 Rc/C) Case Depth

Condition #	Unground	Ground
3	0.0606"	0.0582"
8	0.062"	0.0592"
16	0.0556"	
17	0.0772"	

5.1.1.2. Hot scoring. UTS Program 60-560 (TK) was used to obtain a probability of hot scoring for the unground and ground gears. This program is based on AGMA Std 217.

For hot scoring the maximum speed condition (Cond #15) is more critical than the maximum torque condition (Cond #8).

The sump temperature (oil inlet to mesh) was estimated to be 180 °F.

The oil is SAE 40 with no extreme pressure additives. (Mobil Oil Corporation viscosity specifications for their 15W-40 motor oil indicates that the viscosity at 140 °F is in the center of the range allowed by SAE for 40 weight motor oils. Since the supplier of the oil is not specified, the hot scoring probability was computed at both ends of the allowable SAE range.)

For the unground gears the hot scoring probability is 58% at the high end of the viscosity range and 87% at the low end.

For the ground gears the hot scoring probability is 5% at the high end and 19% at the low end of the viscosity range.

It should be noted that AGMA 217 does not give scoring probabilities for motor oils. The data used in UTS Program 60-560 for motor oils is from data gathered over a period of years by UTS staff and colleagues in the gear design field.

5.1.1.3. Cold scoring. UTS Program 60-5408 (TK) was used to obtain a probability of cold scoring for the unground and ground gears. This program is based on Mobil Oil Corporation's EHL Guidebook, Third Edition.

For cold scoring the maximum torque condition (Cond #8) is more critical than the maximum speed condition (Cond #15).

The sump temperature (oil inlet to mesh) was estimated to be 180 °F.

The oil is Mobil Delvac 1240. (Mobil Oil Corporation states that the lubricant parameter for Delvac 1240 engine oil would be suitable for SAE 15W-40.)

For the unground gears the cold scoring probability is 50%.

For the ground gears the cold scoring probability is below 5%

5.1.2. Low-Speed Train. The duty cycle table used for the "Miner's Rule" life predictions for the low-speed train was taken from "Original 500 Spec," Schedule A, furnished by TACOM. (Pinion torque is in lb-in.)

==== MINER'S RULE ====

Cond #	Pin RPM	Horsepower	Pin Tork	Mins	Pin-Cycles	Gear-Cycles
1	148	53.9	22944	20	2960	1005.28
2	148	30.39	12936	30	4440	1507.92
3	178	93.07	32940	10	1780	604.528
4	653	105	10212	10	6530	2217.74
5	534	75.37	8892	30	16020	5440.75
6	327	86.02	16572	20	6540	2221.13
7	1069	116	6876	20	21380	7261.13
8	267	144	34152	10	2670	906.792
9	534	114	13536	30	16020	5440.75
10	1247	139	7068	30	37410	12705.3
11	950	131	8688	20	19000	6452.83
12	653	152	14748	30	19590	6653.21
13	1484	166	7068	30	44520	15120
14	713	166	14748	30	21390	7264.53
15	1841	481	16476	20	36820	12504.9
16	178	74.22	26268	10	1780	604.528
17	178	79.95	28296	10	1780	604.528
18	534	86.56	10212	10	5340	1813.58
TOTALS				370	265970	90329.4

5.1.2.1. Bending strength and surface durability. UTS Gear Analysis program #500 was used to obtain the I and J factors for the low-speed gears. A semi-topping hob was used to simulate a nominal corner break at the tooth tips. The stress correction factor, Kf, is not the standard AGMA factor. The optional modified Kf in the program uses the radius of curvature where the J factor (and stress) is calculated, while the standard AGMA Kf uses the radius of curvature of the fillet at the root of the tooth. The output sheets and plots, labeled "L.S. Train-Unground Nominal" and "L.S. Train-Ground Nominal" are attached as Appendices C and D, respectively.

An estimate of the mismatch across the face of the gears is required along with a face mismatch factor, Cmf. An estimate of the mismatch was made from the lead errors allowed on the gears and the shaft misalignment allowed by the housing specifications. The mean misalignment was calculated as this correlated well with an analysis of a TACOM test duty cycle performed under this contract. (See paragraph 5.3, "'MLRS' Gears with 'In-House' TACOM Test Duty Cycle.") UTS Program #60-5406 (TK) was then used to calculate an "equivalent Cmf" for each load condition in the duty cycle to include the effect of the crown on the 53-tooth gear.

UTS Program #540 was run to obtain life predictions for the unground and ground low-speed train subjected to the duty cycle.

Program #540 Summary Sheet - Unground L.S. Gears

	Number of Duty Cycles
PINION PITTING:	
Life= 880 hours	142+
to 8604 hours	1387+
PINION BENDING STRENGTH:	
Life= 6489 hours	1046+
To More Than 100,000 hours	16000+
GEAR PITTING:	
Life= 2593 hours	418+
to 25335 hours	4086+
GEAR BENDING STRENGTH:	
Life= 8791 hours	1426+
Life Is More Than 100,000 hours	16000+

Program #540 Summary Sheet - Ground L.S. Gears

	Number of Duty Cycles	% of Unground Gear Life
PINION PITTING:		
Life= 2181 hours	351+	248%
to 38374 hours	6189+	446%
PINION BENDING STRENGTH:		
Life Is More Than 100,000 hours	16000+	1541%
GEAR PITTING:		
Life= 6423 hours	1035+	248%
To More Than 100,000 hours	16000+	395%
GEAR BENDING STRENGTH:		
Life Is More Than 100,000 hours	16000+	----

NOTE: One duty cycle is 6.2 hours.

A range is given for the life of the gears if less than 100,000 hours. This is necessary as both values of Sac and Sat from Tables 5 & 6 of AGMA 218 have been used by the program. This range can be extensive due to the rapid change of cycles with the load. (See Fig. 20 & 21 of AGMA 218) The higher values may be used if special care is used in gearbox design, manufacture, and heat treatment. The minimum case depth to the 50 Rc/C point specified on the gear drawings (0.055") is not enough to stay safely below the depth to maximum sub-surface shear for some of the duty cycle conditions. Since this is the case use of the higher life values for these gears is questionable.

Suggested Minimum Effective (50 Rc/C) Case Depth

Condition #	Unground	Ground
1	0.06"	0.0575"
3	0.069"	0.0658"
8	0.0704"	0.067"
16	0.0633"	0.0605"
17	0.0651"	0.0622"

5.1.2.2. Hot scoring. UTS Program 60-560 (TK) was used to obtain a probability of hot scoring for the unground and ground gears. This program is based on AGMA Std 217.

For hot scoring the maximum speed condition (Cond #15) is more critical than the maximum torque condition (Cond #8).

The sump temperature (oil inlet to mesh) was estimated to be 180 °F.

The oil is SAE 40 with no extreme pressure additives. (Mobil Oil Corporation viscosity specifications for their 15W-40 motor oil indicates that the viscosity at 140 °F is in the center of the range allowed by SAE for 40 weight motor oils. Since the supplier of the oil is not specified, the hot scoring probability was computed at both ends of the allowable SAE range.)

For the unground gears the hot scoring probability is 2% at the high end of the viscosity range and 8% at the low end.

For the ground gears the hot scoring probability is less than 1% over the viscosity range.

It should be noted that AGMA 217 does not give scoring probabilities for motor oils. The data used in UTS Program 60-560 for motor oils is from data gathered over a period of years by UTS staff and colleagues in the gear design field.

5.1.2.3 Cold Scoring. UTS Program 60-5408 (TK) was used to obtain a probability of cold scoring for the unground and ground gears. This program is based on Mobil Oil Corporation's EHL Guidebook, Third Edition.

For cold scoring the maximum torque condition (Cond #8) is more critical than the maximum speed condition (Cond #15).

The sump temperature (oil inlet to mesh) was estimated to be 180 °F.

The oil is Mobil Delvac 1240. (Mobil Oil Corporation states that the lubricant parameter for Delvac 1240 engine oil would be suitable for SAE 15W-40.)

For the unground gears the cold scoring probability is 38%.

For the ground gears the cold scoring probability is below 5%

5.1.3 Backlash. The gear drawings specify the tooth thickness at the reference pitch diameter for the gears. The size over pins is also given as an optional method of checking the tooth thickness. The tooth thickness given is not defined as actual thickness as measured by pins or effective tooth thickness. The backlash between gears is determined by the maximum material condition of the teeth. The effective tooth thickness of a tooth is larger than the measured tooth thickness except when measured with a parallel axis master gear which contacts from the specified start of active profile to the effective tooth tip. When measuring over two pins the effective tooth thickness is not measured, and allowance must be made for errors in those elements of the gear which are not measured. The measurement over two pins does not account for lead error, pitch error, profile error and runout. Errors in these elements all reduce the backlash between the teeth. (The increase in effective tooth thickness due to lead error is reduced considerably due to the crown on the teeth.) Calculations were made using the tolerances on the gear drawings and the size over pins to determine the effective tooth thickness. It was assumed that the gears were made to the size over pins given. Root mean square was used which covers more than 95% of cases.

Effective tooth thickness

19-tooth H.S. pinion:	0.5055"/0.5025"
32-tooth H.S. gear:	0.5265"/0.5225"
18-tooth L.S. pinion:	0.5043"/0.5012"
53-tooth L.S. gear:	0.5277"/0.5237"

The drawings of the housing indicate that the input and output shaft bores are to be within 0.005" of true location with respect to the intermediate shaft bores. The center distance limits are then as follows:

H.S. Train Center Distance = 7.435"/7.425"
L.S. Train Center Distance = 10.292"/10.282"

Calculations were then made to find the temperature at which the assembled backlash becomes zero when the gears and the housing are at the SAME temperature. The assumed inspection temperature is 68 °F.

H.S. Train:

At minimum machined center distance and maximum effective tooth thickness the backlash would become zero at +50 °F

At maximum machined center distance and minimum effective tooth thickness the backlash would become zero at -276 °F

L.S. Train:

At minimum machined center distance and maximum effective tooth thickness the backlash would become zero at +78 °F

At maximum machined center distance and minimum effective tooth thickness the backlash would become zero at -161 °F

5.1.4. Bearing Life. The bearings supporting the gears are cylindrical roller bearings. A calculation of the L-10 life was made from the duty cycle for "standard" gears and 50,000-lb vehicle weight. The sprocket load affects the 53-tooth gear shaft roller bearings as the inboard end of the sprocket output shaft is supported in a spline in the gear shaft. The direction of chain pull is 29.6 degrees from the gear housing center line in forward speed and about 50.6 degrees from the gear housing center line in reverse. Condition #17 is in reverse. See Figure 5-1 for the location of the bearings.

Calculations were made for the bearing loads in terms of the input torque (lb-in).

P = tangential gear load, lb
S = separating gear load, lb
Q = input torque, lb-in
R' = operating pitch radius, in
PA' = operating pressure angle

H.S. Train

$P = Q/R'_{HS\ Pin} = 0.361\ Q$
$S = P \tan(PA') = 0.186\ Q$

L.S. Train

$P = Q*(32/19)/R'_{LS\ Pin} = 0.646\ Q$
$S = P \tan(PA') = 0.324\ Q$

Bearing I & II

Forward
$P_I = +0.187\ Q$ | $P_{II} = +0.174\ Q$
$S_I = +0.0964\ Q$ | $S_{II} = +0.0896\ Q$

Reverse
$P_I = -0.187\ Q$ | $P_{II} = -0.174\ Q$
$S_I = +0.0964\ Q$ | $S_{II} = +0.0896\ Q$

Bearing III & IV

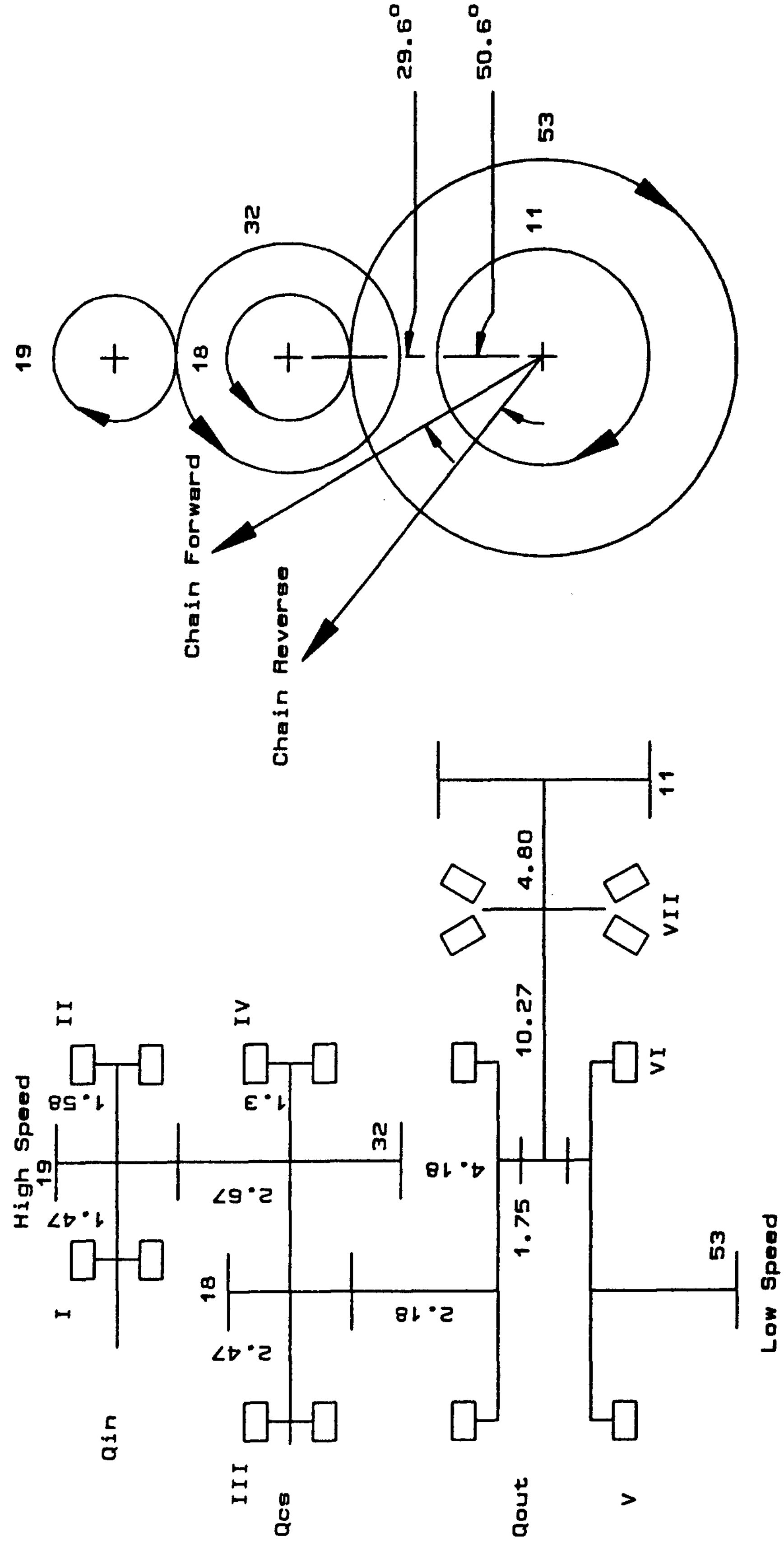
Forward, Due to H.S. Train
$P_{III} = -0.0729\ Q$ | $P_{IV} = -0.288\ Q$
$S_{III} = -0.0375\ Q$ | $S_{IV} = -0.148\ Q$

Forward, Due to L.S. Train
$P_{III} = -0.398\ Q$ | $P_{IV} = -0.248\ Q$
$S_{III} = +0.200\ Q$ | $S_{IV} = +0.124\ Q$

Reverse, Due to H.S. Train
$P_{III} = +0.0729\ Q$ | $P_{IV} = +0.288\ Q$
$S_{III} = -0.0375\ Q$ | $S_{IV} = -0.148\ Q$

Reverse, Due to L.S. Train
$P_{III} = +0.398\ Q$ | $P_{IV} = +0.248\ Q$
$S_{III} = +0.200\ Q$ | $S_{IV} = +0.124\ Q$

Figure 5-1. Bearing Arrangement, "Standard" Gears



Bearing V & VI

Forward, Due to L.S. Train

$P_V = +0.425\ Q$	$P_{VI} = +0.221\ Q$
$S_V = -0.213\ Q$	$S_{VI} = -0.111\ Q$

Forward, Due to Sprocket

Load at spline $= Q\ (32/19)(53/18)\ 1/R'_{SPKT}\ 4.80"/10.27"$
$= 0.108\ Q$ at -60.4 deg

$P_V = +0.0204\ Q$	$P_{VI} = +0.0329\ Q$
$S_V = -0.0359\ Q$	$S_{VI} = -0.0580\ Q$

Reverse, Due to L.S. Train

$P_V = -0.425\ Q$	$P_{VI} = -0.221\ Q$
$S_V = -0.213\ Q$	$S_{VI} = -0.111\ Q$

Reverse, Due to Sprocket

Load at spline $= 0.108\ Q$ at -39.4 deg

$P_V = +0.0319\ Q$	$P_{VI} = +0.0516\ Q$
$S_V = -0.0262\ Q$	$S_{VI} = -0.0424\ Q$

Total Loads

Forward

$R_I = 0.210\ Q$
$R_{II} = 0.196\ Q$
$R_{III} = 0.498\ Q$
$R_{IV} = 0.537\ Q$
$R_V = 0.479\ Q$
$R_{VI} = 0.259\ Q$

Reverse

$R_I = 0.210\ Q$
$R_{II} = 0.196\ Q$
$R_{III} = 0.498\ Q$
$R_{IV} = 0.537\ Q$
$R_V = 0.460\ Q$
$R_{VI} = 0.229\ Q$

UTS Program 20-370 (TK) was modified to provide L-10 life in addition to the exponential mean load, and the L-10 life was then calculated for each bearing.

The equation used for L-10 life:

$$L\text{-}10 = (16667/RPM)*(C/R)^{10/3}$$

where: RPM = mean exponential bearing speed, rev/min
C = bearing basic dynamic capacity, lb (10^6 cycles)
R = mean exponential radial load, lb

Tables 5-1 through 5-6 show the calculated life for each bearing illustrated in Figure 5-1.

Summary of L-10 Bearing Life

```
Bearing I   - 24967 hours
Bearing II  - 31424 hours
Bearing III - 22705 hours
Bearing IV  -  8230 hours
Bearing V   - 35798 hours
Bearing VI  - 200000+ hours
```

5.1.5. Computer Data. All computer data generated is furnished on two floppy discs labeled "H.S. Train, Standard Gears (Military), Ground and Unground" and "L.S. Train, Standard Gears (Military), Ground and Unground" and is part of the report. Appendix O contains an index of the files on these disks.

5.2. "MLRS" Gears with 50,000-lb and 66,000-lb Vehicle Duty Cycles

An analysis of both "MLRS" trains in the drive was made using duty cycles furnished by TACOM for 50,000-lb and 66,000-lb vehicle weight. (The "MLRS" low-speed gears are the same parts as the "standard" low-speed gears.)

The drawings of the "MLRS" gears indicate a rack form for generating the gears and an alternate rack form. In addition, the specifications allow grinding the teeth as an option. The analysis was run with a full tip radius hob and ground gears. (See paragraph 5.1, "Standard" Gears with 50,000-lb Duty Cycle, for a comparison of ground and unground gears.) The AGMA Q class for ground gears is at least Q11. Q11 tolerances were used in the analysis except for the runout for the intermediate shaft gears which is increased to allow for shaft assembly tolerance.

For the H.S. Train 18-tooth pinion:

```
12300307 Estimated Ground
==================== VARIABLE SHEET ==================================
St Input---- Name--- Output--- Unit----- Comment----------------------
   11        Q                           AGMA Quality Number
   18        N                           Number of teeth
   3.5       Pnd               1/in      Normal pitch
   0         psi               deg       Helix angle
   1.75      F                 in        Face width

             VrT     .0017     in        Radial Runout Tolerance (TIR)
             QRUN    11                  Runout Quality Q#

             VpA     .00046    in        Allowable Pitch Variation +/-
             QPIT    11                  Pitch Quality Q#

             VoT     .00059    in        Profile Tolerance
             QPRO    11                  Profile Quality Q#

             VyT     .00039    in        Tooth Alignment Tolerance
             QLD     11                  Alignment Quality Q#
```

Table 5-1. Exponential Mean Load - BRG I - STD

Cond #	Load lb	RPM	Time, hrs	Exponent	Dyn Rating,C
1	2860.000	250.0	.333	3.3333	17100
2	1613.000	250.0	.500		
3	4108.000	300.0	.167		
4	1273.000	1100.0	.167		
5	1109.000	900.0	.500		
6	2066.000	550.0	.333		
7	857.000	1800.0	.333		
8	4259.000	450.0	.167		
9	1688.000	900.0	.500		
10	882.000	2100.0	.500		
11	1084.000	1600.0	.333		
12	1840.000	1100.0	.500		
13	882.000	2500.0	.500		
14	1840.000	1200.0	.500		
15	2054.000	3100.0	.333		
16	3276.000	300.0	.167		
17	3528.000	300.0	.167		
18	1273.000	900.0	.167		
	Mean_Load 1800.568	Average 1210.8	Total_Hrs 6.167		
			L_10_Hrs 24967.258		

Table 5-2. Exponential Mean Load - BRG II - STD

Cond #	Load lb	RPM	Time, hrs	Exponent	Dyn Rating,C
1	2670.000	250.0	.333	3.3333	17100
2	1505.000	250.0	.500		
3	3834.000	300.0	.167		
4	1188.000	1100.0	.167		
5	1035.000	900.0	.500		
6	1929.000	550.0	.333		
7	800.000	1800.0	.333		
8	3975.000	450.0	.167		
9	1576.000	900.0	.500		
10	823.000	2100.0	.500		
11	1011.000	1600.0	.333		
12	1717.000	1100.0	.500		
13	823.000	2500.0	.500		
14	1717.000	1200.0	.500		
15	1917.000	3100.0	.333		
16	3058.000	300.0	.167		
17	3293.000	300.0	.167		
18	1188.000	900.0	.167		
	Mean_Load 1680.508	Average 1210.8	Total_Hrs 6.167		
			L_10_Hrs 31424.328		

Table 5-3. Exponential Mean Load - BRG III - STD

Cond #	Load lb	RPM	Time, hrs	Exponent	Dyn Rating,C
1	6783.000	148.0	.333	3.3333	33700
2	3825.000	148.0	.500		
3	9741.000	178.0	.167		
4	3018.000	653.0	.167		
5	2630.000	534.0	.500		
6	4900.000	327.0	.333		
7	2032.000	1069.0	.333		
8	10100.000	267.0	.167		
9	4004.000	534.0	.500		
10	2092.000	1247.0	.500		
11	2570.000	950.0	.333		
12	4362.000	653.0	.500		
13	2092.000	1484.0	.500		
14	4362.000	712.0	.500		
15	4871.000	1841.0	.333		
16	7769.000	178.0	.167		
17	8366.000	178.0	.167		
18	3018.000	534.0	.167		
	Mean_Load	Average	Total_Hrs		
	4269.406	718.8	6.167		
			L_10_Hrs		
			22704.942		

Table 5-4. Exponential Mean Load - BRG IV - STD

Cond #	Load lb	RPM	Time, hrs	Exponent	Dyn Rating,C
1	7314.000	148.0	.333	3.3333	26800
2	4125.000	148.0	.500		
3	10504.000	178.0	.167		
4	3254.000	653.0	.167		
5	2835.000	534.0	.500		
6	5284.000	327.0	.333		
7	2192.000	1069.0	.333		
8	10890.000	267.0	.167		
9	4317.000	534.0	.500		
10	2255.000	1247.0	.500		
11	2771.000	950.0	.333		
12	4704.000	653.0	.500		
13	2255.000	1484.0	.500		
14	4704.000	712.0	.500		
15	5252.000	1841.0	.333		
16	8377.000	178.0	.167		
17	9022.000	178.0	.167		
18	3254.000	534.0	.167		
	Mean_Load	Average	Total_Hrs		
	4603.596	718.8	6.167		
			L_10_Hrs		
			8229.537		

Table 5-5. Exponential Mean Load - BRG V - STD

Cond #	Load lb	RPM	Time, hrs	Exponent	Dyn Rating,C
1	6524.000	50.0	.333	3.3333	26800
2	3678.000	50.0	.500		
3	9369.000	60.0	.167		
4	2903.000	222.0	.167		
5	2529.000	181.0	.500		
6	4713.000	111.0	.333		
7	1954.000	363.0	.333		
8	9714.000	91.0	.167		
9	3851.000	181.0	.500		
10	2012.000	424.0	.500		
11	2472.000	323.0	.333		
12	4196.000	222.0	.500		
13	2012.000	504.0	.500		
14	4196.000	242.0	.500		
15	4685.000	625.0	.333		
16	7473.000	60.0	.167		
17	7728.000	60.0	.167		
18	2903.000	181.0	.167		
	Mean_Load	Average	Total_Hrs		
	4095.244	244.1	6.167		
			L_10_Hrs		
			35798.174		

Table 5-6. Exponential Mean Load - BRG VI - STD

Cond #	Load lb	RPM	Time, hrs	Exponent	Dyn Rating
1	3528.000	50.0	.333	3.3333	26800
2	1989.000	50.0	.500		
3	5066.000	60.0	.167		
4	1570.000	222.0	.167		
5	1367.000	181.0	.500		
6	2548.000	111.0	.333		
7	1056.000	363.0	.333		
8	5252.000	91.0	.167		
9	2082.000	181.0	.500		
10	1088.000	424.0	.500		
11	1337.000	323.0	.333		
12	2269.000	222.0	.500		
13	1088.000	504.0	.500		
14	2269.000	242.0	.500		
15	2533.000	625.0	.333		
16	4040.000	60.0	.167		
17	3847.000	60.0	.167		
18	1570.000	181.0	.167		
	Mean_Load	Average	Total_Hrs		
	2205.398	244.1	6.167		
			L_10_Hrs		
			281726.888		

For the H.S. Train 34-tooth gear:

```
12300301 Estimated Ground
=================== VARIABLE SHEET =================================
St Input---- Name--- Output--- Unit----- Comment---------------------
             Q                           AGMA Quality Number
   34        N                           Number of teeth
   3.5       Pnd               1/in      Normal pitch
   0         psi               deg       Helix angle
   1.582     F                 in        Face width

   .004      VrT               in        Radial Runout Tolerance (TIR)
             QRUN    9                   Runout Quality Q#

   .00051    VpA               in        Allowable Pitch Variation +/-
             QPIT    11                  Pitch Quality Q#

   .00064    VoT               in        Profile Tolerance
             QPRO    11                  Profile Quality Q#

   .00039    VyT               in        Tooth Alignment Tolerance
             QLD     11                  Alignment Quality Q#
```

For the L.S. Train 18-tooth pinion:

```
12292025 Estimated Ground
=================== VARIABLE SHEET =================================
St Input---- Name--- Output--- Unit----- Comment---------------------
             Q                           AGMA Quality Number
             m                           Message-Quality Number
   18        N                           Number of teeth
   3.5       Pnd               1/in      Normal pitch
   0         psi               deg       Helix angle
   3.5       F                 in        Face width

   .0037     VrT               in        Radial Runout Tolerance (TIR)
             QRUN    9                   Runout Quality Q#

   .00046    VpA               in        Allowable Pitch Variation +/-
             QPIT    11                  Pitch Quality Q#

   .00059    VoT               in        Profile Tolerance
             QPRO    11                  Profile Quality Q#

   .00064    VyT               in        Tooth Alignment Tolerance
             QLD     11                  Alignment Quality Q#
```

For the L.S. Train 53 tooth gear:

12292079 Estimated Ground

==================== VARIABLE SHEET ====================================

St	Input----	Name---	Output---	Unit-----	Comment----------------------
	11	Q			AGMA Quality Number
		m	'OK		Message-Quality Number
	53	N			Number of teeth
	3.5	Pnd		1/in	Normal pitch
	0	psi		deg	Helix angle
	2.88	F		in	Face width
		VrT	.0022	in	Radial Runout Tolerance (TIR)
		QRUN	11		Runout Quality Q#
		VpA	.00056	in	Allowable Pitch Variation +/-
		QPIT	11		Pitch Quality Q#
		VoT	.00069	in	Profile Tolerance
		QPRO	11		Profile Quality Q#
		VyT	.00056	in	Tooth Alignment Tolerance
		QLD	11		Alignment Quality Q#

The analysis was done on "nominal" gears. Nominal means that the split limit was used on tooth thickness, ODs, etc., and the design center distances on the gear drawings were used.

A "Reliability Factor" of 0.9 was used. A factor of 0.9 results in less than 1 "failure" out of 20 drives. "Failure" means that 1 drive out of 20 will run less than the calculated life and 19 drives out of 20 will run longer than the calculated life. Commercial drives are usually designed for 1 in 100 failure rate. A failure rate of 1 in 20 was used for military duty. This failure rate correlated well with an analysis of a TACOM test duty cycle performed under this contract. (See paragraph 5.3, "'MLRS' Gears with 'In-House' TACOM Test Duty Cycle.")

The surface finish limit set on the gear drawings is 63 microinches. While this finish is more or less standard for milled or hobbed finishes it is doubtful that any production gears have been produced with a finish this rough.

For ground gears the pinion tooth surface should be no more than about 20 microinches (and may be as low as 10 microinches) after break-in. The gear should be no more than about 25 microinches (and may be as low as 15 microinches) after break-in. For hot scoring calculations, 20 microinches was used for the pinion and 25 for the gear.

It is recommended that the listed values in Mobil Oil Corporation's EHL Guidebook, Third Edition, be used for cold scoring since the methods and equations were calibrated for these values. For the unground gears 28 microinches was used and for the ground gears 14 microinches was used.

5.2.1. High-Speed Train. The duty cycle tables used for "Miner's Rule" life predictions for the high-speed train with 50,000-lb and 66,000-lb vehicle weight were taken from "Original 500 Spec," Schedule A, and "For 66K GVW," Schedule A, furnished by TACOM. (Pinion torque is in lb-in)

50,000-lb Duty Cycle

==== MINER'S RULE ====

Cond #	Pin RPM	Horsepower	Pin Tork	Mins	Pin-Cycles	Gear-Cycles
1	250	54.05	13620	20	5000	2647.06
2	250	30.48	7680	30	7500	3970.59
3	300	93.14	19560	10	3000	1588.24
4	1100	105	6060	10	11000	5823.53
5	900	75.43	5280	30	27000	14294.1
6	550	85.9	9840	20	11000	5823.53
7	1800	116	4080	20	36000	19058.8
8	450	144	20280	10	4500	2382.35
9	900	114	8040	30	27000	14294.1
10	2100	140	4200	30	63000	33352.9
11	1600	131	5160	20	32000	16941.2
12	1100	152	8760	30	33000	17470.6
13	2500	166	4200	30	75000	39705.9
14	1200	166	8760	30	36000	19058.8
15	3100	481	9780	20	62000	32823.5
16	300	74.29	15600	10	3000	1588.24
17	300	80	16800	10	3000	1588.24
18	900	86.57	6060	10	9000	4764.71
TOTALS				370	448000	237176

66,000-lb Duty Cycle

==== MINER'S RULE ====

Cond #	Pin RPM	Horsepower	Pin Tork	Mins	Pin-Cycles	Gear-Cycles
1	280	62.53	14070	20	5600	2964.71
2	280	45.33	10200	30	8400	4447.06
3	335	136	25620	10	3350	1773.53
4	1220	131	6780	10	12200	6458.82
5	1285	205	10080	30	38550	20408.8
6	615	105	10800	20	12300	6511.76
7	2015	145	4560	20	40300	21335.3
8	390	176	28500	10	3900	2064.71
9	1010	140	8760	30	30300	16041.2
10	2350	167	4500	30	70500	37323.5
11	1790	169	5970	20	35800	18952.9
12	1230	188	9660	30	36900	19535.3
13	2800	196	4410	30	84000	44470.6
14	1440	113	4980	30	43200	22870.6
15	3000	221	4650	20	60000	31764.7
16	335	100	18840	10	3350	1773.53
17	280	100	22500	10	2800	1482.35
18	1020	109	6780	10	10200	5400
TOTALS				370	501650	265579

5.2.1.1. Bending strength and surface durability. UTS Gear Analysis program #500 was used to obtain the I and J factors for the high-speed gears. A semi-topping hob was used to simulate a nominal corner break at the tooth tips. The stress correction factor, Kf, is not the standard AGMA factor. The optional modified Kf in the program uses the radius of curvature where the J factor (and stress) is calculated, while the standard AGMA Kf uses the radius of curvature of the fillet at the root of the tooth. The output sheets and plots, labeled "H.S. Train-MLRS," are attached as Appendix F.

An estimate of the mismatch across the face of the gears is required along with a face mismatch factor, Cmf. An estimate of the mismatch was made from the lead errors allowed on the gears and the shaft misalignment allowed by the housing specifications. The mean misalignment was calculated as this correlated well with an analysis of a TACOM test duty cycle performed under this contract. (See paragraph 5.3, "'MLRS' Gears with 'In-House' TACOM Test Duty Cycle.") UTS Program #60-5406 (TK) was then used to calculate an "equivalent Cmf" for each load condition in the duty cycle to include the effect of the crown on the 34-tooth gear.

UTS Program #540 was run to obtain life predictions for the high-speed train subjected to both duty cycles.

Program #540 Summary Sheet - 50000 lb Duty Cycle

	Number of Duty Cycles
PINION PITTING:	
Life= 1074 hours	173+
to 14513 hours	2341+
PINION BENDING STRENGTH:	
Life Is More Than 100,000 hours	16000+
GEAR PITTING:	
Life= 2030 hours	327+
to 27413 hours	4421+
GEAR BENDING STRENGTH:	
Life= 12095 hours	1950+
To More Than 100,000 hours	16000+

Program #540 Summary Sheet - 66000 lb Duty Cycle

	Number of Duty Cycles
PINION PITTING:	
Life= 135 hours	21+
to 1133 hours	182+
PINION BENDING STRENGTH:	
Life= 3233 hours	521+
To More Than 100,000 hours	16000+
GEAR PITTING:	
Life= 255 hours	41+
to 2140 hours	345+
GEAR BENDING STRENGTH:	
Life= 685 hours	110+
to 7959 hours	1286+

NOTE: One duty cycle is 6.2 hours.

A range is given for the life of the gears if less than 100,000 hours. This is necessary as both values of Sac and Sat from Tables 5 & 6 of AGMA 218 have been used by the program. This range can be extensive due to the rapid change of cycles with the load. (See Fig. 20 & 21 of AGMA 218.) The higher values may be used if special care is used in gearbox design, manufacture, and heat treatment. The minimum case depth to the 50 Rc/C point specified on the gear drawings (0.055") is not enough to stay safely below the depth to maximum sub-surface shear for some of the 6,6000-lb duty cycle conditions. Since this is the case, use of the higher life values for these gears is questionable.

Suggested Minimum Effective (50 Rc/C) Case Depth (66000 lb)

Condition #	
3	0.0601"
8	0.0628"

5.2.1.2. Hot scoring. UTS Program 60-560 (TK) was used to obtain a probability of hot scoring. This program is based on AGMA Std 217.

For hot scoring the maximum speed condition (Cond #15) is more critical than the maximum torque condition (Cond #8) for the 50,000-lb duty cycle. For the 66,000-lb duty cycle the critical condition is the maximum torque condition (Cond #8).

The sump temperature (oil inlet to mesh) was estimated to be 180 °F.

The oil is SAE 40 with no extreme pressure additives. (Mobil Oil Corporation's viscosity specifications for their 15W-40 motor oil indicates that the viscosity at 140 °F is in the center of the range allowed by SAE for 40 weight motor oils. Since the supplier of the oil is not specified, the hot scoring probability was computed at both ends of the allowable SAE range.)

For the 50,000-lb duty cycle the hot scoring probability is 77% at the high end and 96% at the low end of the viscosity range.

For the 66,000-lb duty cycle the hot scoring probability is 3% at the high end and 13% at the low end of the viscosity range. (The load for the maximum speed condition (Cond #15) is much lower for the 66,000-lb duty cycle.)

It should be noted that AGMA 217 does not give scoring probabilities for motor oils. The data used in UTS Program 60-560 for motor oils is from data gathered over a period of years by UTS staff and colleagues in the gear design field.

5.2.1.3. Cold scoring. UTS Program 60-5408 (TK) was used to obtain a probability of cold scoring. This program is based on Mobil Oil Corporation's EHL Guidebook, Third Edition.

For cold scoring the maximum torque condition (Cond #8) is more critical than the maximum speed condition (Cond #15).

The sump temperature (oil inlet to mesh) was estimated to be 180 °F.

The oil is Mobil Delvac 1240. (Mobil Oil Corporation states that the lubricant parameter for Delvac 1240 engine oil would be suitable for SAE 15W-40.)

The cold scoring probability is less than 5% for the 50,000-lb duty cycle and is 6% for the 66,000-lb duty cycle.

5.2.2. Low-Speed Train. The duty cycle tables used for "Miner's Rule" life predictions for the low-speed train with 50,000-lb and 66,000-lb vehicle weight were taken from "Original 500 Spec," Schedule A, and "For 66K GVW," Schedule A, furnished by TACOM. (Pinion torque is in lb-in)

50,000-lb Duty Cycle

==== MINER'S RULE ====

Cond #	Pin RPM	Horsepower	Pin Tork	Mins	Pin-Cycles	Gear-Cycles
1	148	53.89	22939	20	2960	1005.28
2	148	30.39	12935	30	4440	1507.92
3	178	93.08	32943	10	1780	604.528
4	653	105	10206	10	6530	2217.74
5	534	75.38	8893	30	16020	5440.75
6	327	86.02	16573	20	6540	2221.13
7	1069	116	6872	20	21380	7261.13
8	267	144	34156	10	2670	906.792
9	534	114	13541	30	16020	5440.75
10	1247	140	7074	30	37410	12705.3
11	950	131	8691	20	19000	6452.83
12	653	152	14754	30	19590	6653.21
13	1484	166	7074	30	44520	15120
14	713	166	14754	30	21390	7264.53
15	1841	481	16472	20	36820	12504.9
16	178	74.23	26274	10	1780	604.528
17	178	79.94	28295	10	1780	604.528
18	534	86.51	10206	10	5340	1813.58
TOTALS				370	265970	90329.4

66,000-lb Duty Cycle

==== MINER'S RULE ====

Cond #	Pin RPM	Horsepower	Pin Tork	Mins	Pin-Cycles	Gear-Cycles
1	148	62.43	26577	20	2960	1005.28
2	148	45.26	19267	30	4440	1507.92
3	177	135	48393	10	1770	601.132
4	646	131	12807	10	6460	2193.96
5	680	205	19040	30	20400	6928.3
6	326	105	20400	20	6520	2214.34
7	1067	145	8613	20	21340	7247.55
8	207	176	53833	10	2070	703.019
9	535	140	16547	30	16050	5450.94

66,000-lb Duty Cycle

Cond #	Pin RPM	Horsepower	Pin Tork	Mins	Pin-Cycles	Gear-Cycles
10	1244	167	8500	30	37320	12674.7
11	948	169	11277	20	18960	6439.25
12	651	188	18247	30	19530	6632.83
13	1482	195	8330	30	44460	15099.6
14	762	113	9407	30	22860	7763.77
15	1588	221	8783	20	31760	10786.4
16	177	99.98	35587	10	1770	601.132
17	148	99.84	42500	10	1480	502.642
18	540	109	12807	10	5400	1833.96
TOTALS				370	265550	90186.8

5.2.2.1. Bending strength and surface durability. UTS Gear Analysis program #500 was used to obtain the I and J factors for the low-speed gears. A semi-topping hob was used to simulate a nominal corner break at the tooth tips. The stress correction factor, Kf, is not the standard AGMA factor. The optional modified Kf in the program uses the radius of curvature where the J factor (and stress) is calculated, while the standard AGMA Kf uses the radius of curvature of the fillet at the root of the tooth. The output sheets and plots, labeled "L.S. Train-MLRS," are attached as Appendix G.

An estimate of the mismatch across the face of the gears is required along with a face mismatch factor, Cmf. An estimate of the mismatch was made from the lead errors allowed on the gears and the shaft misalignment allowed by the housing specifications. The mean misalignment was calculated as this correlated well with an analysis of a TACOM test duty cycle performed under this contract. (See paragraph 5.3, "'MLRS' Gears with 'In-House' TACOM Test Duty Cycle.") UTS Program #60-5406 (TK) was then used to calculate an "equivalent Cmf" for each load condition in the duty cycle to include the effect of the crown on the 53-tooth gear.

Program #540 Summary Sheet - 50000 lb Duty Cycle

	Number of Duty Cycles
PINION PITTING:	
Life= 2176 hours	350+
to 38170 hours	6156+
PINION BENDING STRENGTH:	
Life Is More Than 100,000 hours	16000+
GEAR PITTING:	
Life= 6408 hours	1033+
To More Than 100,000 hours	16000+
GEAR BENDING STRENGTH:	
Life Is More Than 100,000 hours	16000+

Program #540 Summary Sheet - 66000 lb Duty Cycle

	Number of Duty Cycles
PINION PITTING:	
Life= 154 hours	24+
to 1279 hours	206+
PINION BENDING STRENGTH:	
Life= 732 hours	118+
to 8703 hours	1403+
GEAR PITTING:	
Life= 453 hours	73+
to 3767 hours	607+
GEAR BENDING STRENGTH:	
Life= 6785 hours	1094+
To More Than 100,000 hours	16000+

NOTE: One duty cycle is 6.2 hours.

A range is given for the life of the gears if less than 100,000 hours. This is necessary as both values of Sac and Sat from Tables 5 & 6 of AGMA 218 have been used by the program. This range can be extensive due to the rapid change of cycles with the load. (See Fig. 20 & 21 of AGMA 218.) The higher values may be used if special care is used in gearbox design, manufacture, and heat treatment. The minimum case depth to the 50 Rc/C point specified on the gear drawings (0.055") is not enough to stay safely below the depth to maximum sub-surface shear for some of the duty cycle conditions. Since this is the case use of the higher life values for these gears is questionable.

Suggested Minimum Effective (50 Rc/C) Case Depth

Condition #	50000 lb	66000 lb
1	0.0575"	0.0607"
3	0.0659"	0.0761"*
8	0.067"	0.0793"*
16	0.0606"	0.0678"
17	0.0622"	0.0724"

* AGMA 218 recommends a maximum effective case depth of 0.0772" based on the tooth thickness at the tip of the pinion. To apply the suggested case depth, tip relief on the gears would be required to reduce the load on the tips of the teeth.

5.2.2.2. Hot scoring. UTS Program 60-560 (TK) was used to obtain a probability of hot scoring. This program is based on AGMA Std 217.

The maximum speed condition (Cond #15) is more critical than the maximum torque condition (Cond #8).

The sump temperature (oil inlet to mesh) was estimated to be 180 ^{o}F.

The oil is SAE 40 with no extreme pressure additives. (Mobil Oil Corporation's viscosity specifications for their 15W-40 motor oil indicates that the viscosity at

140 °F is in the center of the range allowed by SAE for 40 weight motor oils. Since the supplier of the oil is not specified, the hot scoring probability was computed at both ends of the allowable SAE range.)

For both duty cycles the hot scoring probability is less than 1% at both ends of the viscosity range.

5.2.2.3. Cold scoring. UTS Program 60-5408 (TK) was used to obtain a probability of cold scoring. This program is based on Mobil Oil Corporation's EHL Guidebook, Third Edition.

For cold scoring the maximum torque condition (Cond #8) is more critical than the maximum speed condition (Cond #15).

The sump temperature (oil inlet to mesh) was estimated to be 180 °F.

The oil is Mobil Delvac 1240. (Mobil Oil Corporation states that the lubricant parameter for Delvac 1240 engine oil would be suitable for SAE 15W-40.) The cold scoring probability is less than 5% for the 50,000-lb duty cycle and is 5% for the 66,000-lb duty cycle.

5.2.3. Backlash. The gear drawings specify the tooth thickness at the reference pitch diameter for the gears. The size over pins is also given as an optional method of checking the tooth thickness. The tooth thickness given is not defined as actual thickness as measured by pins or effective tooth thickness. The backlash between gears is determined by the maximum material condition of the teeth. The effective tooth thickness of a tooth is larger than the measured tooth thickness except when measured with a parallel axis master gear which contacts from the specified start of active profile to the effective tooth tip. When measuring over two pins the effective tooth thickness is not measured and allowance must be made for errors in the elements of the gear not measured. The measurement over two pins does not account for lead error, pitch error, profile error and runout. Errors in these elements all reduce the backlash between the teeth. (The increase in effective tooth thickness due to lead error is reduced considerably due to the crown on the teeth.) Calculations were made using the tolerances on the gear drawings and the size over pins to determine the effective tooth thickness. It was assumed that the gears were made to the size over pins given. Root mean square was used which covers more than 95% of cases.

Effective tooth thickness

18 tooth H.S. pinion: 0.5371"/0.5341"
34 tooth H.S. gear: 0.3560"/0.3520"
18 tooth L.S. pinion: 0.5043"/0.5012"
53 tooth L.S. gear: 0.5277"/0.5237"

The drawings of the housing indicate that the input and output shaft bores are to be within 0.005" of true location with respect to the intermediate shaft bores. The center distance limits are as follows:

H.S. Train Center Distance = 7.435"/7.425"
L.S. Train Center Distance = 10.292"/10.282"

Calculations were then made to find the temperature at which the assembled backlash becomes zero when the gears and the housing are at the SAME temperature. The assumed inspection temperature is 68 ^{o}F.

H.S. Train:

At minimum machined center distance and maximum effective tooth thickness the backlash would become zero at +44 ^{o}F

At maximum machined center distance and minimum effective tooth thickness the backlash would become zero at -294 ^{o}F

L.S. Train:

At minimum machined center distance and maximum effective tooth thickness the backlash would become zero at +78 ^{o}F

At maximum machined center distance and minimum effective tooth thickness the backlash would become zero at -161 ^{o}F

5.2.4 Bearing Life. The bearings supporting the gears are cylindrical roller bearings. A calculation of the L-10 life was made for both duty cycles. The sprocket load affects the 53-tooth gear shaft roller bearings as the inboard end of the sprocket output shaft is supported in a spline in the gear shaft. The direction of chain pull is 29.6 degrees from the gear housing center line in forward speed and about 50.6 degrees from the gear housing center line in reverse. Condition #17 is in reverse. See the sketch, Figure 5-2, for the location of the bearings.

Calculations were made for the bearing loads in terms of the input torque (lb-in.).

P = tangential gear load, lb
S = separating gear load, lb
Q = input torque, lb-in

R' = operating pitch radius, in
PA' = operating pressure angle

H.S. Train

$P = Q/R'_{HS\ Pin} = 0.389\ Q$
$S = P\tan(PA') = 0.181\ Q$

L.S. Train

$P = Q*(34/18)/R'_{LS\ Pin} = 0.724\ Q$
$S = P\tan(PA') = 0.363\ Q$

Bearing I & II

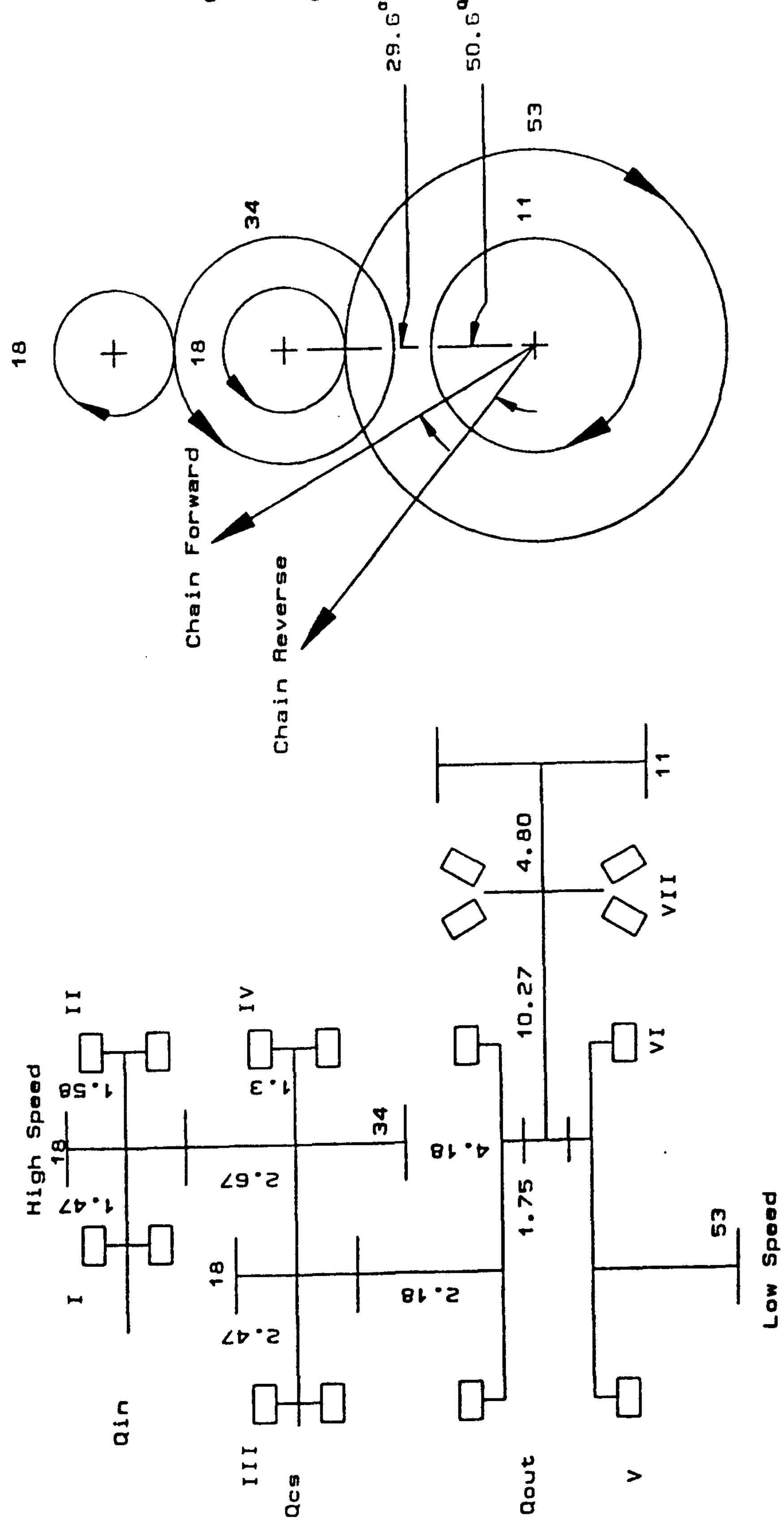
Forward

$P_I = +0.202\ Q$ $\quad$ $P_{II} = +0.187\ Q$
$S_I = +0.0938\ Q$ $\quad$ $S_{II} = +0.0872\ Q$

Reverse

$P_I = -0.202\ Q$ $\quad$ $P_{II} = -0.187\ Q$
$S_I = +0.0938\ Q$ $\quad$ $S_{II} = +0.0872\ Q$

Figure 5-2. Bearing Arrangement, "MLRS" Gears



Bearing III & IV

Forward, Due to H.S. Train

$P_{III} = -0.0785\ Q$ $\qquad$ $P_{IV} = -0.3105\ Q$
$S_{III} = -0.0365\ Q$ $\qquad$ $S_{IV} = -0.1445\ Q$

Forward, Due to L.S. Train

$P_{III} = -0.446\ Q$ $\qquad$ $P_{IV} = -0.278\ Q$
$S_{III} = +0.224\ Q$ $\qquad$ $S_{IV} = +0.139\ Q$

Reverse, Due to H.S. Train

$P_{III} = +0.0785\ Q$ $\qquad$ $P_{IV} = +0.3105\ Q$
$S_{III} = -0.0365\ Q$ $\qquad$ $S_{IV} = -0.1445\ Q$

Reverse, Due to L.S. Train

$P_{III} = +0.446\ Q$ $\qquad$ $P_{IV} = +0.278\ Q$
$S_{III} = +0.224\ Q$ $\qquad$ $S_{IV} = +0.139\ Q$

Bearing V & VI

Forward, Due to L.S. Train

$P_{V} = +0.476\ Q$ $\qquad$ $P_{VI} = +0.248\ Q$
$S_{V} = -0.239\ Q$ $\qquad$ $S_{VI} = -0.124\ Q$

Forward, Due to Sprocket

Load at spline = $Q\ (34/18)(53/18)\ 1/R'_{SPKT}\ 4.80"/10.27"$
$= 0.121\ Q$ at -60.4 degrees

$P_{V} = +0.0228\ Q$ $\qquad$ $P_{VI} = +0.0370\ Q$
$S_{V} = -0.0401\ Q$ $\qquad$ $S_{VI} = -0.0649\ Q$

Reverse, Due to L.S. Train

$P_{V} = -0.476\ Q$ $\qquad$ $P_{VI} = -0.248\ Q$
$S_{V} = -0.239\ Q$ $\qquad$ $S_{VI} = -0.124\ Q$

Reverse, Due to Sprocket

Load at spline = $0.121\ Q$ at -39.4 degrees

$P_{V} = +0.0357\ Q$ $\qquad$ $P_{VI} = +0.0578\ Q$
$S_{V} = -0.0293\ Q$ $\qquad$ $S_{VI} = -0.0475\ Q$

Total Loads

Forward

$R_I = 0.223\ Q$
$R_{II} = 0.206\ Q$
$R_{III} = 0.557\ Q$
$R_{IV} = 0.589\ Q$
$R_V = 0.572\ Q$
$R_{VI} = 0.342\ Q$

Reverse

$R_I = 0.223\ Q$
$R_{II} = 0.206\ Q$
$R_{III} = 0.557\ Q$
$R_{IV} = 0.589\ Q$
$R_V = 0.516\ Q$
$R_{VI} = 0.256\ Q$

UTS Program 20-370 (TK) was modified to provide L-10 life in addition to the exponential mean load and the L-10 life was calculated for each bearing.

The equation used for L-10 life:

$$L\text{-}10 = (16667/RPM) * (C/R)^{10/3}$$

where: RPM = mean exponential bearing speed, rev/min
C = bearing basic dynamic capacity, lb (10^6 cycles)
R = mean exponential radial load, lb

Tables 5-7 through 5-18 show the calculated life at 50,000 lb and at 66,000 lb for each bearing illustrated in Figure 5-2.

Summary of L-10 Bearing Life

	50000 lb	66000 lb
Bearing I	- 20445 hours	12358 hours
Bearing II	- 26619 hours	16098 hours
Bearing III	- 9279 hours	5609 hours
Bearing IV	- 3589 hours	2170 hours
Bearing V	- 4031 hours	2459 hours
Bearing VI	- 22864 hours	14108 hours

5.2.5. Computer Data. All computer data generated is furnished on two floppy disks labeled "H.S. Train, MLRS Gears (Military)" and "L.S. Train, MLRS Gears (Military)" and is part of the report. Appendix Q contains an index of files on these disks.

Table 5-7. Exponential Mean Load - BRG I - 50000 lb - MLRS

Cond #	Load lb	RPM	Time, hrs	Exponent	Dyn Rating,C
1	3037.000	250.0	.333	3.3333	17100
2	1713.000	250.0	.500		
3	4362.000	300.0	.167		
4	1352.000	1100.0	.167		
5	1177.000	900.0	.500		
6	2194.000	550.0	.333		
7	910.000	1800.0	.333		
8	4522.000	450.0	.167		
9	1793.000	900.0	.500		
10	937.000	2100.0	.500		
11	1151.000	1600.0	.333		
12	1953.000	1100.0	.500		
13	937.000	2500.0	.500		
14	1953.000	1200.0	.500		
15	2181.000	3100.0	.333		
16	3479.000	300.0	.167		
17	3746.000	300.0	.167		
18	1351.000	900.0	.167		
	Mean_Load	Average	Total_Hrs		
	1911.818	1210.8	6.167		
			L_10_Hrs		
			20444.712		

Table 5-8. Exponential Mean Load - BRG I - 66000 lb - MLRS

Cond #	Load lb	RPM	Time, hrs	Exponent	Dyn Rating,C
1	3138.000	280.0	.333	3.3333	17100
2	2275.000	280.0	.500		
3	5713.000	335.0	.167		
4	1512.000	1220.0	.167		
5	2248.000	1285.0	.500		
6	2408.000	615.0	.333		
7	1017.000	2015.0	.333		
8	6356.000	390.0	.167		
9	1953.000	1010.0	.500		
10	1004.000	2350.0	.500		
11	1331.000	1790.0	.333		
12	2154.000	1230.0	.500		
13	983.000	2800.0	.500		
14	1111.000	1440.0	.500		
15	1037.000	3000.0	.333		
16	4201.000	335.0	.167		
17	5018.000	280.0	.167		
18	1512.000	1020.0	.167		
	Mean_Load	Average	Total_Hrs		
	2149.320	1355.8	6.167		
			L_10_Hrs		
			12357.916		

Table 5-9. Exponential Mean Load - BRG II - 50000 lb - MLRS

Cond #	Load lb	RPM	Time, hrs	Exponent	Dyn Rating,C
1	2806.000	250.0	.333	3.3333	17100
2	1582.000	250.0	.500		
3	4029.000	300.0	.167		
4	1249.000	1100.0	.167		
5	1088.000	900.0	.500		
6	2027.000	550.0	.333		
7	840.000	1800.0	.333		
8	4178.000	450.0	.167		
9	1656.000	900.0	.500		
10	865.000	2100.0	.500		
11	1063.000	1600.0	.333		
12	1805.000	1100.0	.500		
13	865.000	2500.0	.500		
14	1805.000	1200.0	.500		
15	2015.000	3100.0	.333		
16	3214.000	300.0	.167		
17	3461.000	300.0	.167		
18	1248.000	900.0	.167		
	Mean_Load	Average	Total_Hrs		
	1766.295	1210.8	6.167		
			L_10_Hrs		
			26618.945		

Table 5-10. Exponential Mean Load - BRG II - 66000 lb - MLRS

Cond #	Load lb	RPM	Time, hrs	Exponent	Dyn Rating,C
1	2898.000	280.0	.333	3.3333	17100
2	2101.000	280.0	.500		
3	5278.000	335.0	.167		
4	1397.000	1220.0	.167		
5	2076.000	1285.0	.500		
6	2225.000	615.0	.333		
7	939.000	2015.0	.333		
8	5871.000	390.0	.167		
9	1805.000	1010.0	.500		
10	927.000	2350.0	.500		
11	1230.000	1790.0	.333		
12	1990.000	1230.0	.500		
13	908.000	2800.0	.500		
14	1026.000	1440.0	.500		
15	958.000	3000.0	.333		
16	3881.000	335.0	.167		
17	4635.000	280.0	.167		
18	1397.000	1020.0	.167		
	Mean_Load	Average	Total_Hrs		
	1985.432	1355.8	6.167		
			L_10_Hrs		
			16097.719		

Table 5-11. Exponential Mean Load - BRG III - 50000 lb - MLRS

Cond #	Load lb	RPM	Time, hrs	Exponent	Dyn Rating,C
1	7586.000	250.0	.333	3.3333	33700
2	4278.000	250.0	.500		
3	10895.000	300.0	.167		
4	3377.000	1100.0	.167		
5	2941.000	900.0	.500		
6	5481.000	550.0	.333		
7	2273.000	1800.0	.333		
8	11296.000	450.0	.167		
9	4478.000	900.0	.500		
10	2339.000	2100.0	.500		
11	2874.000	1600.0	.333		
12	4879.000	1100.0	.500		
13	2339.000	2500.0	.500		
14	4879.000	1200.0	.500		
15	5447.000	3100.0	.333		
16	8689.000	300.0	.167		
17	9358.000	300.0	.167		
18	3375.000	900.0	.167		
	Mean_Load 4775.322	Average 1210.8	Total_Hrs 6.167 L_10_Hrs 9279.194		

Table 5-12. Exponential Mean Load - BRG III - 66000 lb - MLRS

Cond #	Load lb	RPM	Time, hrs	Exponent	Dyn Rating,C
1	7837.000	280.0	.333	3.3333	33700
2	5681.000	280.0	.500		
3	14270.000	335.0	.167		
4	3776.000	1220.0	.167		
5	5615.000	1285.0	.500		
6	6016.000	615.0	.333		
7	2540.000	2015.0	.333		
8	15875.000	390.0	.167		
9	4879.000	1010.0	.500		
10	2507.000	2350.0	.500		
11	3325.000	1790.0	.333		
12	5381.000	1230.0	.500		
13	2456.000	2800.0	.500		
14	2774.000	1440.0	.500		
15	2590.000	3000.0	.333		
16	10494.000	335.0	.167		
17	12533.000	280.0	.167		
18	3776.000	1020.0	.167		
	Mean_Load 5368.501	Average 1355.8	Total_Hrs 6.167 L_10_Hrs 5609.033		

Table 5-13. Exponential Mean Load - BRG IV - 50000 lb - MLRS

Cond #	Load lb	RPM	Time, hrs	Exponent	Dyn Rating,C
1	8022.000	250.0	.333	3.3333	26800
2	4524.000	250.0	.500		
3	11521.000	300.0	.167		
4	3571.000	1100.0	.167		
5	3110.000	900.0	.500		
6	5796.000	550.0	.333		
7	2403.000	1800.0	.333		
8	11945.000	450.0	.167		
9	4736.000	900.0	.500		
10	2474.000	2100.0	.500		
11	3039.000	1600.0	.333		
12	5160.000	1100.0	.500		
13	2474.000	2500.0	.500		
14	5160.000	1200.0	.500		
15	5760.000	3100.0	.333		
16	9188.000	300.0	.167		
17	9895.000	300.0	.167		
18	3569.000	900.0	.167		
	Mean_Load	Average	Total_Hrs		
	5049.852	1210.8	6.167		
			L_10_Hrs		
			3588.757		

Table 5-14. Exponential Mean Load - BRG IV - 66000 lb - MLRS

Cond #	Load lb	RPM	Time, hrs	Exponent	Dyn Rating,C
1	8287.000	280.0	.333	3.3333	26800
2	6008.000	280.0	.500		
3	15090.000	335.0	.167		
4	3993.000	1220.0	.167		
5	5937.000	1285.0	.500		
6	6361.000	615.0	.333		
7	2686.000	2015.0	.333		
8	16786.000	390.0	.167		
9	5160.000	1010.0	.500		
10	2650.000	2350.0	.500		
11	3516.000	1790.0	.333		
12	5690.000	1230.0	.500		
13	2597.000	2800.0	.500		
14	2933.000	1440.0	.500		
15	2739.000	3000.0	.333		
16	11097.000	335.0	.167		
17	13252.000	280.0	.167		
18	3993.000	1020.0	.167		
	Mean_Load	Average	Total_Hrs		
	5676.729	1355.8	6.167		
			L_10_Hrs		
			2169.825		

Table 5-15. Exponential Mean Load - BRG V - 50000 lb - MLRS

Cond #	Load lb	RPM	Time, hrs	Exponent	Dyn Rating,C
1	7791.000	250.0	.333	3.3333	26800
2	4393.000	250.0	.500		
3	11188.000	300.0	.167		
4	3467.000	1100.0	.167		
5	3020.000	900.0	.500		
6	5628.000	550.0	.333		
7	2334.000	1800.0	.333		
8	11600.000	450.0	.167		
9	4599.000	900.0	.500		
10	2402.000	2100.0	.500		
11	2952.000	1600.0	.333		
12	5011.000	1100.0	.500		
13	2402.000	2500.0	.500		
14	5011.000	1200.0	.500		
15	5594.000	3100.0	.333		
16	8923.000	300.0	.167		
17	8669.000	300.0	.167		
18	3466.000	900.0	.167		
	Mean_Load 4876.920	Average 1210.8	Total_Hrs 6.167 L_10_Hrs 4030.762		

Table 5-16. Exponential Mean Load - BRG V - 66000 lb - MLRS

Cond #	Load lb	RPM	Time, hrs	Exponent	Dyn Rating,C
1	8048.000	280.0	.333	3.3333	26800
2	5834.000	280.0	.500		
3	14655.000	335.0	.167		
4	3878.000	1220.0	.167		
5	5766.000	1285.0	.500		
6	6178.000	615.0	.333		
7	2608.000	2015.0	.333		
8	16302.000	390.0	.167		
9	5011.000	1010.0	.500		
10	2574.000	2350.0	.500		
11	3415.000	1790.0	.333		
12	5526.000	1230.0	.500		
13	2523.000	2800.0	.500		
14	2849.000	1440.0	.500		
15	2660.000	3000.0	.333		
16	10776.000	335.0	.167		
17	11610.000	280.0	.167		
18	3878.000	1020.0	.167		
	Mean_Load 5467.355	Average 1355.8	Total_Hrs 6.167 L_10_Hrs 2459.388		

Table 5-17. Exponential Mean Load - BRG VI - 50000 lb - MLRS

Cond #	Load lb	RPM	Time, hrs	Exponent	Dyn Rating
1	4658.000	250.0	.333	3.3333	26800
2	2627.000	250.0	.500		
3	6690.000	300.0	.167		
4	2073.000	1100.0	.167		
5	1806.000	900.0	.500		
6	3365.000	550.0	.333		
7	1395.000	1800.0	.333		
8	6936.000	450.0	.167		
9	2750.000	900.0	.500		
10	1436.000	2100.0	.500		
11	1765.000	1600.0	.333		
12	2996.000	1100.0	.500		
13	1436.000	2500.0	.500		
14	2996.000	1200.0	.500		
15	3345.000	3100.0	.333		
16	5335.000	300.0	.167		
17	4301.000	300.0	.167		
18	2073.000	900.0	.167		
	Mean_Load	Average	Total_Hrs		
	2897.439	1210.8	6.167		
			L_10_Hrs		
			22863.915		

Table 5-18. Exponential Mean Load - BRG VI - 66000 lb - MLRS

Cond #	Load lb	RPM	Time, hrs	Exponent	Dyn Rating
1	4812.000	280.0	.333	3.3333	26800
2	3488.000	280.0	.500		
3	8762.000	335.0	.167		
4	2319.000	1220.0	.167		
5	3447.000	1285.0	.500		
6	3694.000	615.0	.333		
7	1560.000	2015.0	.333		
8	9747.000	390.0	.167		
9	2996.000	1010.0	.500		
10	1539.000	2350.0	.500		
11	2042.000	1790.0	.333		
12	3304.000	1230.0	.500		
13	1508.000	2800.0	.500		
14	1703.000	1440.0	.500		
15	1590.000	3000.0	.333		
16	6443.000	335.0	.167		
17	5760.000	280.0	.167		
18	2319.000	1020.0	.167		
	Mean_Load	Average	Total_Hrs		
	3237.315	1355.8	6.167		
			L_10_Hrs		
			14107.842		

5.3. "MLRS" Gears with "In-House" TACOM Test Duty Cycle

An analysis of both "MLRS" trains in the drive was made using the high torque test duty cycle furnished by TACOM for the purpose of checking the results produced by the software. This duty cycle was used during final drive high torque tests at the Warren, Michigan facility. The duty cycle tables used for "Miner's Rule" life predictions and scoring analysis were obtained from a letter of 10 January 1989 from Mr. Ted R. Zimmerman, Contracting Officer's Technical Representative, TACOM to Mr. Kenneth R. Gitchel, Vice President, Universal Technical Systems, Inc. The operating temperatures were obtained from Mr. Zimmerman by telephone on 13 January 1989.

5.3.1. Test Data. Two final drives were disassembled after test. TACOM reported to UTS that no evidence of gear breakage, pitting, spalling, or scoring was found upon visual examination. The parts were not dimensionally inspected. The gears were finished by grinding and had full radius fillets.

The correlation between the results of the M2/M3 design studies and the TACOM test data was adequate with changes only in reliability factors to reflect military practice, method of estimating face mismatch, and surface finish. However, the gears inspected by TACOM personnel were from a vehicle mounted in a test cell where the load was provided by dynamometers. The vehicle frame (and, therefore, the housing) was not subjected to any twisting or bending forces from operation on rough terrain. In addition, since the tracks were removed and the output of the final drives was transferred to dynamometers through drive shafts, the torsional mass-elastic system was different from an operational system.

No field data was available for analysis of drives from operational vehicles that had been subjected to rough terrain conditions.

5.3.2. Changes in Software. The analysis indicates that a change in the software code based on the TACOM test duty cycle and two test drives is not advisable.

Since the amount of gear misalignment caused by frame and housing deflection is not known and the torsional vibration characteristics of an operational vehicle differ from one in a test rig, the prediction accuracy of the software is not confirmed for field operation if these external influences are significant. If they are not significant, the software is capable of good predictions of the suitablility of final drives for a defined duty cycle.

Test data from operational vehicles should be obtained prior to any changes in the software code and/or method of analysis outlined in this report. The test data should include dimensional inspection of the gears and housings of the test final drives.

5.3.3. Changes in Method. A change in the reliability factor from 1.0 (less than one failure in 100 units - commercial practice) to 0.9 (less than one failure in 20 units) to reflect military practice is advisable. (One "failure" in 20 units means that, out of 20 units, 19 units will run longer than predicted and 1 unit will not run as long as predicted.)

It is recommended that the mean values of lead error, shafts out of plane and shafts out of parallel be used when estimating the contact mismatch across the gear face.

It is recommended that the actual "run-in" values for gear surface finish be used for hot scoring and the listed values in Mobil Oil Corporation's EHL Guidebook, Third Edition, be used for cold scoring.

5.3.4. Gear and Housing Data. The drawings of the "MLRS" gears indicate a full fillet rack form for generating the gears as one option and an alternate flat root rack form. In addition, the specifications allow grinding the teeth as an option. The analysis was run with a full tip radius hob and ground gears in keeping with the reported data for the test drives. The AGMA Q class for ground gears is at least Q11. Q11 tolerances were used in the analysis except for the runout for the intermediate shaft gears which are increased to allow for shaft assembly tolerance.

The analysis was done on "nominal" gears. Nominal means that the split limit was used on tooth thickness, ODs, etc., and the design center distances on the gear drawings were used.

5.3.5. High-Speed Train. Duty cycle table used for "Miner's Rule" life predictions for the high-speed train. (Pinion torque is in lb-in.)

TACOM Test Duty Cycle

==== MINER'S RULE ====

Cond #	Pin RPM	Horsepower	Pin Tork	Mins	Pin-Cycles	Gear-Cycles
1	306	41.92	8630	74.6	1.36966E+6	725112
2	551	149	17067	12.6	416556	220530
3	473	149	19915	7.6	215688	114188
4	362	130	22763	14.4	312768	165583
5	328	133	25611	13.9	273552	144822
6	295	133	28459	18.4	325680	172419
7	245	121	31307	8.7	127890	67706.5
8	234	122	33011	35.9	504036	266843
9	211	114	34155	9.4	119004	63002.1
10	184	108	37003	13.6	150144	79488
11	150	94.88	39851	1.5	13500	7147.06
TOTALS				210.6	3.82847E+6	2.02684E+6

5.3.5.1. Bending strength and surface durability. UTS Gear Analysis program #500 was used to obtain the I and J factors for the high-speed gears. A semi-topping hob was used to simulate a nominal corner break at the tooth tips. The root fillet stress correction factor, Kf, is not the standard AGMA (American Gear Manufacturers Association) factor. The optional modified Kf in the program uses the radius of curvature where the J factor (and stress) is calculated, while the standard AGMA Kf uses the radius of curvature of the fillet at the root of the tooth. The output sheets and plots, labeled "H.S. Train-MLRS," are attached as appendix F.

An estimate of the mismatch across the face of the gears is required along with a calculation of the face mismatch factor, Cmf. An estimate of the mismatch was made from the lead errors allowed on the gears and the shaft misalignment allowed by the housing specifications. The mean misalignment was calculated. UTS Program #60-5406

(TK) was then used to calculate an "equivalent Cmf" for each load condition in the duty cycle to include the effect of the crown on the 34-tooth gear.

UTS Program #540 was run to obtain life predictions for the high speed train subjected to the duty cycle with the following result:

	Number of Duty Cycles
PINION PITTING:	
Life is less than one duty cycle	1-
PINION BENDING STRENGTH:	
Life is less than one duty cycle	1-
to 3393 hours	16+
GEAR PITTING:	
Life is less than one duty cycle	1-
GEAR BENDING STRENGTH:	
Life is less than one duty cycle	1-
to 338 hours	1+

NOTE: One duty cycle is 210.6 hours.

A range is given for the life of the gears if less than 100,000 hours and more than one duty cycle. This is necessary as both values of Sac and Sat from Tables 5 & 6 of AGMA 218 have been used by the program. This range can be extensive due to the rapid change of cycles with stress. (See Fig. 20 & 21 of AGMA 218.) The higher values may be used if special care is used in gearbox design, manufacture, and heat treatment. The minimum case depth to the 50 Rc/C point specified on the gear drawings (0.055") is not enough to stay safely below the depth to maximum sub-surface shear for some of the duty cycle conditions. However, since no spalling was found upon examination of the gears it is indicated that the higher life numbers can be used.

The prediction of the software is that pitting (but not breakage) should occur before one duty cycle is reached by 1 out of 20 drives. It would require an increase in "allowable" compressive stress of only 7% to reach one duty cycle. Since only two drives were tested it is recommended that the prediction be considered adequate and that no change in the software code be made without further test data.

5.3.5.2. Hot scoring. UTS Program 60-560 (TK) was used to obtain a probability of hot scoring. This program is based on AGMA Std 217.

The duty cycle consists of an extensive warm up period at about 40 HP and the torque is then increased in small steps up to the maximum torque. The maximum torque is applied for a short period of time. This type of loading is ideal for "breaking in" the gears and will reduce the surface finish below the manufactured finish before the loads likely to produce hot scoring are applied. The pinion tooth surface should be no more than about 20 microinches (and may be as low as 10 microinches) after break-in. The gear should be no more than about 25 microinches (and may be as low as 15 microinches) after break-in. Surfaces of 20 microinches for the pinion and 25 microinches for the gear were assumed, resulting in an average composite surface finish of 22.6 microinches for the hot scoring calculations.

The oil is SAE 40 with no extreme pressure additives. (Mobil Oil Corporation viscosity specifications for their 15W-40 motor oil indicates that the viscosity at 140 degrees F is in the center of the range allowed by SAE for 40 weight motor oils. Since the supplier of the oil is not specified, the hot scoring probability was computed at both ends of the allowable SAE range.)

5.3.5.3. Cold scoring. UTS Program 60-5408 (TK) was used to obtain a probability of cold scoring. This program is based on Mobil Oil Corporation's EHL Guidebook, Third Edition.

The EHL Guidebook tabulates the composite surface finish (not the average composite) for various tooth finishing methods for which the probability of cold scoring was developed.

When a pinion surface finish of 20 microinches and a gear surface finish of 25 (as in the hot scoring calculations) are used with the software, the predicted probability of cold scoring exceeds 80%. When the values from the guidebook are used (composite finish of 20 microinches for ground, hardened gears) the maximum probability is 37%.

It is recommended that the guidebook surface finish values be used instead of the estimated actual finishes to achieve results that are consistent with the condition of the gears subjected to the test duty cycle. Even though only two drives were tested it is probable that, with a calculated probability of over 80%, scoring would have occurred during testing.

The oil used in the calculations is Mobil Delvac 1240. (Mobil Oil Corporation states that the lubricant parameter for Delvac 1240 engine oil would be suitable for SAE 15W-40.)

5.3.5.4. Scoring summary. Table 5-19 gives the load point from the TACOM duty cycle (Ld Pt), the pinion torque in lb-in (Tork), the pinion speed (RPM), the time at each load condition (Hours), the equivalent load distribution factor for crowned gears (Eq_Cm), the mesh oil inlet temperature in degrees F (oF) and the hot scoring (Hot Scr %) and cold scoring (Cold Scr %) probabilities in percent.

The scoring probabilities calculated by the software are not in conflict with the test data when the surface finish is estimated as noted and no change in the software code is recommended without further test data.

5.3.6. Low-Speed Train. The duty cycle information used for the "Miner's Rule" life predictions for the low-speed train is given on page 56. (Pinion torque is in lb-in.)

Table 5-19. Scoring Summary, MLRS H.S.Train (lb-in)

#	Ld Pt	Tork	RPM	Hours	Eq_Cm	oF	Hot Scr %	Cold Scr %
1	WU	8630	306	74.6	1.661		-----	-----
2	.3	17067	551	12.6	1.389	187	LT 1 to 5	7
3	.35	19915	473	7.6	1.339	208	4 to 17	28
4	.4	22763	362	14.4	1.298	198	1 to 6	15
5	.45	25611	328	13.9	1.266	198	2 to 8	14
6	.5	28459	295	18.4	1.241	220	6 to 22	37
7	.55	31307	245	8.7	1.221	187	LT 1 to 3	8
8	.58	33011	234	35.9	1.21	158	Under 1	Under 5
9	.6	34155	211	9.4	1.204	186	LT 1 to 3	7
10	.65	37003	184	13.6	1.19	165	Under 1	Under 5
11	.7	39851	150	1.5	1.178		-----	-----

Test Duty Cycle

==== MINER'S RULE ====

Cond #	Pin RPM	Horsepower	Pin Tork	Mins	Pin-Cycles	Gear-Cycles
1	162	41.92	16302	74.6	725112	246264
2	292	149	32237	12.6	220752	74972.4
3	250	149	37617	7.6	114000	38717
4	191	130	42996	14.4	165024	56045.9
5	174	133	48376	13.9	145116	49284.7
6	156	133	53755	18.4	172224	58491.2
7	130	122	59135	8.7	67860	23046.8
8	124	122	62355	35.9	267096	90711.8
9	112	114	64515	9.4	63168	21453.3
10	97.2	107	69894	13.6	79315.2	26937.2
11	79.5	94.99	75274	1.5	7155	2430
TOTALS				210.6	2.02682E+6	688355

5.3.6.1. Bending strength and surface durability. UTS Gear Analysis program #500 was used to obtain the I and J factors for the low-speed gears. A semi-topping hob was used to simulate a nominal corner break at the tooth tips. The stress correction factor, Kf, is not the standard AGMA factor. The optional modified Kf in the program uses the radius of curvature where the J factor (and stress) is calculated, while the standard AGMA Kf uses the radius of curvature of the fillet at the root of the tooth. The output sheets and plots, labeled "L.S. Train-MLRS," are attached as Appendix G.

An estimate of the mismatch across the face of the gears is required along with a calculation of the face mismatch factor, Cmf. An estimate of the mismatch was made from the lead errors allowed on the gears and the shaft misalignment allowed by the housing specifications. The mean misalignment was calculated. UTS Program #60-5406 (TK) was then used to calculate an "equivalent Cmf" for each load condition in the duty cycle to include the effect of the crown on the 53-tooth gear.

UTS Program #540 was run to obtain life predictions for the low speed train subjected to the duty cycle with the following result:

	Number of Duty Cycles
PINION PITTING:	
Life is less than one duty cycle	1-
PINION BENDING STRENGTH:	
Life is less than one duty cycle	1-
to 405 hours	1+
GEAR PITTING:	
Life is less than one duty cycle	1-
to 239 hours	1+
GEAR BENDING STRENGTH:	
Life= 423 hours	1+
to 5793 hours	27+

NOTE: One duty cycle is 210.6 hours.

A range is given for the life of the gears if less than 100,000 hours and more than one duty cycle. This is necessary as both values of Sac and Sat from Tables 5 & 6 of AGMA 218 have been used by the program. This range can be extensive due to the rapid change of cycles with the stress. (See Fig. 20 & 21 of AGMA 218.) The higher values may be used if special care is used in gearbox design, manufacture, and heat treatment. The minimum case depth to the 50 Rc/C point specified on the gear drawings (0.055") is not enough to stay safely below the depth to maximum sub-surface shear for some of the duty cycle conditions. However, since no spalling was found upon examination of the gears it is indicated that the higher life numbers can be used.

The prediction of the software is that pitting (but not breakage) should occur before one duty cycle is reached by 1 unit out of 20. It would require an increase in "allowable" compressive stress of only 7% to reach one duty cycle. Since only two drives were tested it is recommended that the prediction be considered adequate and that no change in the software be made without further test data.

5.3.6.2. Hot scoring. UTS Program 60-560 (TK) was used to obtain a probability of hot scoring. This program is based on AGMA Std 217.

The duty cycle consists of an extensive warm up period at about 40 HP. The torque is then increased in small steps up to the maximum torque, which is applied for a short period of time. This type of loading is ideal for "breaking in" the gears and will reduce the surface finish below the manufactured finish before the loads likely to produce hot scoring are applied. The pinion tooth surface should be no more than about 20 microinches (and may be as low as 10 microinches) after break-in. The gear should be no more than about 25 microinches (and may be as low as 15 microinches) after break-in. Values of 20 microinches for the pinion and 25 microinches for the gear were assumed, resulting in an average composite surface finish of 22.6 microinches for the hot scoring calculations.

The oil is SAE 40 with no extreme pressure additives. (Mobil Oil Corporation's viscosity specifications for their 15W-40 motor oil indicates that the viscosity at 140 °F is in the center of the range allowed by SAE for 40 weight motor oils. Since the supplier of the oil is not specified, the hot scoring probability was computed at both ends of the allowable SAE range.)

5.3.6.3. Cold scoring. UTS Program 60-5408 (TK) was used to obtain a probability of cold scoring. This program is based on Mobil Oil Corporation's EHL Guidebook, Third Edition.

The EHL Guidebook tabulates the composite surface finish for various tooth finishing methods for which the probability of cold scoring was developed.

When a pinion surface finish of 20 microinches and a gear surface finish of 25 microinches are used with the software, the predicted probability of cold scoring is 73%. When the values from the guidebook are used (composite finish of 20 microinches for ground, hardened gears) the maximum probability is 18%.

It is recommended that the guidebook surface finish values be used instead of the estimated actual finishes to achieve results that are consistent with the condition of the gears subjected to the test duty cycle. Even though only two drives were tested it is probable that, with a probability of 73%, scoring would have occurred.

The oil used in the calculations is Mobil Delvac 1240. (Mobil Oil Corporation states that the lubricant parameter for Delvac 1240 engine oil would be suitable for SAE 15W-40.)

5.3.6.4. Scoring summary. Table 5-20 gives the load point from the TACOM duty cycle (Ld Pt), the pinion torque in lb-in (Tork), the pinion speed (RPM), the time at each load condition (Hours), the equivalent load distribution factor for crowned gears (Eq_Cm), the mesh oil inlet temperature in degrees F (oF) and the hot scoring (Hot Scr %) and cold scoring (Cold Scr %) probabilities in percent.

The scoring probabilities calculated by the software are not in conflict with the test data when the surface finish is estimated as noted and no change in the software code is recommended without further test data.

5.3.7. Computer Data. All computer data generated is furnished on a floppy disk labeled "MLRS Set, TACOM Test Data" and is part of the report. Appendix Q contains an index to the files on this disk.

Table 5-20. Scoring Summary, MLRS L.S.Train (lb-in)

#	Ld Pt	Tork	RPM	Hours	Eq_Cm	oF	Hot Scr %	Cld Scr %
1	WU	16302	162	74.6	1.906		-----	-----
2	.3	32237	292	12.6	1.585	187	Under 1	7
3	.35	37617	250	7.6	1.526	208	Under 1	18
4	.4	42996	191	14.4	1.478	198	Under 1	10
5	.45	48376	174	13.9	1.436	198	Under 1	10
6	.5	53755	156	18.4	1.399	220	Under 1	18
7	.55	59135	130	8.7	1.368	187	Under 1	6
8	.58	62355	124	35.9	1.35	158	Under 1	Under 5
9	.6	64515	112	9.4	1.339	186	Under 1	5
10	.65	69894	97.2	13.6	1.317	165	Under 1	Under 5
11	.7	75274	79.5	1.5	1.298		-----	-----

6.0. DESIGN OPTIMIZATION OF THE M2/M3 FINAL DRIVE

6.1. Optimization with 66,000-lb Vehicle Duty Cycle

6.1.1. Gears.

6.1.1.1 General. The optimized gears are ground to AGMA Class Q11 per ANSI/AGMA 2000-A88.1 The present gear specifications have tooth grinding as an option, and the test gears inspected by TACOM for this contract were ground. The optimized gears have controlled tip relief where the present gear specifications allow tip and root relief but do not require it. The same hob is used for all gears except the high speed pinion where a short lead hob is used to obtain finished involute down to the form diameter. If the gears are ground with the "Zero-Degree" method it would be necessary to use profile control cams for each of the four gears. If the "V-Wheel" or form grinding method is used a set of wheel dressing cams would be needed for each gear.

The depth to maximum sub-surface shear was calculated by the software for each load in the duty cycle and the minimum recommended case depth provided. The maximum recommended case depth is calculated based on the thickness of the tooth at the tip to avoid full case hardness completely through the tip. For the low speed gears, since some of the duty cycle loads are quite high, the required case depth based on load is higher than the recommended case depth based on the tooth tip thickness. The use of the required case depth for load is safe if the proper tip relief is used to reduce the load concentration at the tooth tips.

6.1.1.2 Tip relief and profile tolerance control. At the first point of contact (gear tip and pinion root) the deflection of the teeth already under load causes the incoming pinion tooth to seem to be ahead of where it should be. This causes the load to be picked up very abruptly at the tip of the gear tooth. (This condition is more serious on spur and low contact ratio helicals than on full helicals.) This condition applies a heavy shock load at the gear tooth tip where the "cantilever beam" is the longest and produces large root stresses. The same thing happens to the pinion tooth at the last point of contact but the effect of dropping the load abruptly is not as severe (mostly noise and vibration along with a rise in compressive stress from the tooth edge breaking the contact "foot print"). The action is approach action from the first point of contact to the pitch point. The gear tooth is approaching the pinion tooth and the tip acts much like a sharp edged scraper (or, in severe cases, a cutting tool). The action is recess action after contact has passed the pitch point. The gear tooth is retreating from the pinion tooth. Recess action is much more conducive to building and maintaining a lubricant film than is approach action, especially at start or end of action. Tooth spacing errors, either pitch or profile, add to this effect for those teeth that are "ahead" of where they should be. This means that even lightly loaded gears, with little deflection, suffer the same type of problem due to manufacturing error.

One method of reducing the deflection problem (or at least not making it worse) is to make the tolerances on profile such that the base pitch of the driver can never be less than theoretical and the base pitch of the driven never more than theoretical. This will reduce the impact at the first point of contact because it will ensure that the teeth on the driver are a little "behind" and on the driven a little "ahead". Of course, this will make things worse at the last point of contact but conditions there are much less severe. The practical application of this method means that the profile tolerance on the driver should always be plus at the tip, and on the driven always minus. As long as we have to live with these tolerances we may as well put them where they may help us and, at least, not hurt us. (Of course, this must not be overdone as we will lose conjugate action. For gears less than about Q9 accuracy a careful study should be made of the +/- tolerance zone.)

In addition to tolerance control the application of tip relief on the gears will reduce the engagement shock and the tendency to scrape the driver root. (This will also help the abrupt dropping of load at the last point of contact.) The load can then be picked up smoothly and the full load (plus shock) on the tooth tips eliminated. Since optimum tip relief can be built into the production tools, the advantages cost very little except getting the specifications right at the design stage.

6.1.1.3 Gear tolerance. Figures 5-1 through 5-4 give the tolerances for the HS18, HS34, LS18 and LS53 gears. The tolerances conform to AGMA Class Q11 tolerances; however, the shown profile tolerances must be modified later to include the tip relief. The modifications for the H.S. and L.S. gears are shown in Appendices H and I, respectively.

The crown on the present gears is retained. The gears would exhibit lower Cm values and higher life ratings with no crown at all, but only if there were no misalignment other than that allowed by lead errors and housing tolerance. The gears inspected by TACOM personnel were from a vehicle mounted in a test cell and the load was provided by dynamometers. The vehicle frame (and, therefore, the housing) was not subjected to any twisting or bending forces from operation on rough terrain. No field data was provided for analysis of drives from vehicles that had been subjected to rough terrain conditions. Since the amount of gear misalignment caused by frame and housing deflection is not known it is recommended that the crown be applied to reduce concentrated load on the tooth ends in case the housing deflections are significant.

6.1.1.4 Housing tolerance. The tolerance on the center distances must be changed to allow only a 0.003" variation instead of a 0.010" variation. The out of plane tolerance must be tightened slightly.

H.S. Train

Center distance = 7.430" +0.003" -0.000"
In Plane within 0.002"

L.S. Train

Center distance = 10.287" +0.003" -0.000"
In Plane within 0.004".

Figure 6-1. H.S. Train 18-tooth pinion:

```
=================== VARIABLE SHEET ========================================
St Input---- Name--- Output--- Unit----- Comment-----------------------------
                                          HS Train Ground - OPT
                                            60-100 (Rev 1.1) ANSI/AGMA 2000-A88
                                            Gear Classification and Inspection
                                            Handbook
   11         Q                           AGMA Quality Number
              m       'OK                 Message-Quality Number
   18         N                           Number of teeth
   3.5        Pnd               1/in      Normal pitch
   0          psi               deg       Helix angle
   1.75       F                 in        Face width
              Pd      3.5       1/in      Transverse pitch
              D       5.1429    in        Reference pitch diameter
              mn      7.2571429 mm        Normal module

              VrT     .0017     in        Radial Runout Tolerance (TIR)
              QRUN    11                  Runout Quality Q#
              m1      'OK

              VpA     .00046    in        Allowable Pitch Variation +/-
              QPIT    11                  Pitch Quality Q#
              m2      'OK

              VoT     .00059    in        Profile Tolerance
              QPRO    11                  Profile Quality Q#
              m3      'OK

              VyT     .00039    in        Tooth Alignment Tolerance
              QLD     11                  Alignment Quality Q#
              m4      'OK
```

Effective Case Depth to 50 Rc/C Point After Grinding = 0.060"/0.070"

Figure 6-2. H.S. Train 34-tooth gear:

==================== VARIABLE SHEET ====================================

St	Input----	Name---	Output---	Unit-----	Comment------------------------------
					HS Train Ground - OPT 60-100 (Rev 1.1) ANSI/AGMA 2000-A88 Gear Classificationa and Inspection Handbook
	11	Q			AGMA Quality Number
		m	'OK		Message-Quality Number
	34	N			Number of teeth
	3.5	Pnd		1/in	Normal pitch
	0	psi		deg	Helix angle
	1.582	F		in	Face width
		Pd	3.5	1/in	Transverse pitch
		D	9.7143	in	Reference pitch diameter
		mn	7.2571429	mm	Normal module
		VrT	.002	in	Radial Runout Tolerance (TIR)
		QRUN	11		Runout Quality Q#
		m1	'OK		
		VpA	.00052	in	Allowable Pitch Variation +/-
		QPIT	11		Pitch Quality Q#
		m2	'OK		
		VoT	.00065	in	Profile Tolerance
		QPRO	11		Profile Quality Q#
		m3	'OK		
		VyT	.00036	in	Tooth Alignment Tolerance
		QLD	11		Alignment Quality Q#
		m4	'OK		

Effective Case Depth to 50 Rc/C Point After Grinding = 0.060"/0.070"

Crown 0.0005/0.0010 inches
Max Crown Rise To Be Centered On Tooth +/-0.1 inches

Figure 6-3. L.S. Train 18-tooth pinion:

VARIABLE SHEET

St	Input----	Name---	Output---	Unit-----	Comment----------------------------------
					LS Train Ground - OPT
					60-100 (Rev 1.1) ANSI/AGMA 2000-A88
					Gear Classification and Inspection
					Handbook
	11	Q			AGMA Quality Number
		m	'OK		Message-Quality Number
	18	N			Number of teeth
	3.5	Pnd		1/in	Normal pitch
	0	psi		deg	Helix angle
	3.5	F		in	Face width
		Pd	3.5	1/in	Transverse pitch
		D	5.1429	in	Reference pitch diameter
		mn	7.2571429	mm	Normal module
		VrT	.0017	in	Radial Runout Tolerance (TIR)
		QRUN	11		Runout Quality Q#
		m1	'OK		
		VpA	.00046	in	Allowable Pitch Variation +/-
		QPIT	11		Pitch Quality Q#
		m2	'OK		
		VoT	.00059	in	Profile Tolerance
		QPRO	11		Profile Quality Q#
		m3	'OK		
		VyT	.00064	in	Tooth Alignment Tolerance
		QLD	11		Alignment Quality Q#
		m4	'OK		

Effective Case Depth to 50 Rc/C Point After Grinding = 0.070"/0.080"

Figure 6-4. L.S. Train 53-tooth gear:

```
=================== VARIABLE SHEET ======================================
St Input---- Name--- Output--- Unit----- Comment------------------------------
                                          LS Train Ground -OPT
                                            60-100 (Rev 1.1) ANSI/AGMA 2000-A88
                                            Gear Classification and Inspection
                                            Handbook
   11         Q                           AGMA Quality Number
              m       'OK                 Message-Quality Number
   53         N                           Number of teeth
   3.5        Pnd               1/in      Normal pitch
   0          psi               deg       Helix angle
   2.88       F                 in        Face width
              Pd      3.5       1/in      Transverse pitch
              D       15.1429   in        Reference pitch diameter
              mn      7.2571429 mm        Normal module

              VrT     .0022     in        Radial Runout Tolerance (TIR)
              QRUN    11                  Runout Quality Q#
              m1      'OK

              VpA     .00056    in        Allowable Pitch Variation +/-
              QPIT    11                  Pitch Quality Q#
              m2      'OK

              VoT     .00069    in        Profile Tolerance
              QPRO    11                  Profile Quality Q#
              m3      'OK

              VyT     .00056    in        Tooth Alignment Tolerance
              QLD     11                  Alignment Quality Q#
              m4      'OK
```

Effective Case Depth to 50 Rc/C Point After Grinding = 0.070"/0.080"

Crown 0.0010/0.0015 inches
Max Crown Rise To Be Centered On Tooth +/-0.1 inches

6.1.2 High-Speed Train. The duty cycle tables used for "Miner's Rule" life predictions for the high-speed train with 66,000 lb vehicle weight were taken from "Original 500 Spec," Schedule A, and "For 66K GVW," Schedule A, furnished by TACOM. (Pinion torque is in lb-in)

66000-lb Duty Cycle

==== MINER'S RULE ====

Cond #	Pin RPM	Horsepower	Pin Tork	Mins	Pin-Cycles	Gear-Cycles
1	280	62.53	14070	20	5600	2964.71
2	280	45.33	10200	30	8400	4447.06
3	335	136	25620	10	3350	1773.53
4	1220	131	6780	10	12200	6458.82
5	1285	205	10080	30	38550	20408.8
6	615	105	10800	20	12300	6511.76
7	2015	145	4560	20	40300	21335.3
8	390	176	28500	10	3900	2064.71
9	1010	140	8760	30	30300	16041.2
10	2350	167	4500	30	70500	37323.5
11	1790	169	5970	20	35800	18952.9
12	1230	188	9660	30	36900	19535.3
13	2800	196	4410	30	84000	44470.6
14	1440	113	4980	30	43200	22870.6
15	3000	221	4650	20	60000	31764.7
16	335	100	18840	10	3350	1773.53
17	280	100	22500	10	2800	1482.35
18	1020	109	6780	10	10200	5400
TOTALS				370	501650	265579

6.1.2.1 Bending strength and surface durability. UTS Gear Analysis program #500 was used to obtain the I and J factors for the high speed gears. A non-topping hob was used. Any corner break at the tooth tips should be held to no more than .007". The stress correction factor, Kf, is not the standard AGMA factor. The optional modified Kf in the program uses the radius of curvature where the J factor (and stress) is calculated, while the standard AGMA Kf uses the radius of curvature of the fillet at the root of the tooth. The output sheets and plots, labeled "H.S. Train-OPT," are attached as Appendix J.

An estimate of the mismatch across the face of the gears is required along with a face mismatch factor, Cmf. An estimate of the mismatch was made from the lead errors allowed on the gears and the shaft misalignment allowed by the housing specifications. The mean misalignment was calculated, as this correlated well with an analysis of a TACOM test duty cycle performed under this contract. UTS Program #60-5406 (TK) was then used to calculate an "equivalent Cmf" for each load condition in the duty cycle to include the effect of the crown on the 34-tooth gear.

UTS Program #540 was run to obtain life predictions for the "MLRS" and optimized high speed trains.

"MLRS" Program #540 Summary Sheet - 66000 lb Duty Cycle

	Number of Duty Cycles
PINION PITTING:	
Life= 135 hours	21+
to 1133 hours	182+
PINION BENDING STRENGTH:	
Life= 3233 hours	521+
To More Than 100,000 hours	16000+
GEAR PITTING:	
Life= 255 hours	41+
to 2140 hours	345+
GEAR BENDING STRENGTH:	
Life= 685 hours	110+
to 7959 hours	1286+

"Optimized" Program #540 Summary Sheet - 66000 lb Duty Cycle

	Number of Duty Cycles
PINION PITTING:	
Life= 168 hours	27+
to 1410 hours	227+
PINION BENDING STRENGTH:	
Life= 1551 hours	250+
To More Than 100,000 hours	16000+
GEAR PITTING:	
Life= 317 hours	51+
to 2663 hours	429+
GEAR BENDING STRENGTH:	
Life= 1513 hours	244+
To More Than 100,000 hours	16000+

NOTE: One duty cycle is 6.2 hours.

A range is given for the life of the gears if less than 100,000 hours. This is necessary because the program uses both the high and low values of Sac and Sat from Tables 5 & 6 of AGMA 218.2 This range can be extensive due to the rapid change of cycles with the load. (See Fig. 20 & 21 of AGMA 218.) The higher values may be used if special care is used in gearbox design, manufacture, and heat treatment.

The net increase in life for the H.S. Train is 24% for durability and 121% for strength.

6.1.2.2. Hot scoring. UTS Program 60-560 (TK) was used to obtain a probability of hot scoring. This program is based on AGMA Std 217.3

For hot scoring the critical condition is the maximum torque condition (Cond #8).

The sump temperature (oil inlet to mesh) was estimated to be 180 °F.

The oil is SAE 40 with no extreme pressure additives. (Mobil Oil Corporation viscosity specifications for their 15W-40 motor oil indicates that the viscosity at 140 °F is in the center of the range allowed by SAE for 40 weight motor oils. Since the supplier of the oil is not specified, the hot scoring probability was computed at both ends of the allowable SAE range.)

The hot scoring probability is below 1% at the high end and 1% at the low end of the viscosity range, compared to 3% at the high end and 13% at the low end for the "MLRS" gears.

It should be noted that AGMA 217 does not give scoring probabilities for motor oils. The data used in UTS Program 60-560 for motor oils is from information gathered over a period of years by UTS staff and colleagues in the gear design field.

6.1.2.3. Cold scoring. UTS Program 60-5408 (TK) was used to obtain a probability of cold scoring. This program is based on Mobil Oil Corporation's EHL Guidebook, Third Edition.4

For cold scoring the maximum torque condition (Cond #8) is more critical than the maximum speed condition (Cond #15).

The sump temperature (oil inlet to mesh) was estimated to be 180 °F.

The oil is Mobil Delvac 1240. (Mobil Oil Corporation states that the lubricant parameter for Delvac 1240 engine oil would be suitable for SAE 15W-40.)

The cold scoring probability is unchanged at 6% for both gear sets.

6.1.3. Low Speed Train. The duty cycle tables used for "Miner's Rule" life predictions for the low speed train with 66,000 lb vehicle weight were taken from "Original 500 Spec", Schedule A, and "For 66K GVW", Schedule A, furnished by TACOM. (Pinion torque is in lb-in.)

66000-lb Duty Cycle

==== MINER'S RULE ====

Cond #	Pin RPM	Horsepower	Pin Tork	Mins	Pin-Cycles	Gear-Cycles
1	148	62.43	26577	20	2960	1005.28
2	148	45.26	19267	30	4440	1507.92
3	177	135	48393	10	1770	601.132
4	646	131	12807	10	6460	2193.96
5	680	205	19040	30	20400	6928.3
6	326	105	20400	20	6520	2214.34
7	1067	145	8613	20	21340	7247.55
8	207	176	53833	10	2070	703.019
9	535	140	16547	30	16050	5450.94

66000-lb Duty Cycle

Cond #	Pin RPM	Horsepower	Pin Tork	Mins	Pin-Cycles	Gear-Cycles
10	1244	167	8500	30	37320	12674.7
11	948	169	11277	20	18960	6439.25
12	651	188	18247	30	19530	6632.83
13	1482	195	8330	30	44460	15099.6
14	762	113	9407	30	22860	7763.77
15	1588	221	8783	20	31760	10786.4
16	177	99.98	35587	10	1770	601.132
17	148	99.84	42500	10	1480	502.642
18	540	109	12807	10	5400	1833.96
TOTALS				370	265550	90186.8

6.1.3.1. Bending strength and surface durability. UTS Gear Analysis program #500 was used to obtain the I and J factors for the low speed gears. A non-topping hob was used. Any corner break at the tooth tips should be held to no more than .007". The stress correction factor, Kf, is not the standard AGMA factor. The optional modified Kf in the program uses the radius of curvature where the J factor (and stress) is calculated, while the standard AGMA Kf uses the radius of curvature of the fillet at the root of the tooth. The output sheets and plots, labeled "L.S. Train-OPT," are attached as Appendix K.

An estimate of the mismatch across the face of the gears is required along with a face mismatch factor, Cmf. An estimate of the mismatch was made from the lead errors allowed on the gears and the shaft misalignment allowed by the housing specifications. The mean misalignment was calculated as this correlated well with an analysis of a TACOM test duty cycle performed under this contract. UTS Program #60-5406 (TK) was then used to calculate an "equivalent Cmf" for each load condition in the duty cycle to include the effect of the crown on the 53-tooth gear.

UTS Program #540 was run to obtain life predictions for the "MLRS" and optimized low speed trains.

"MLRS" Program #540 Summary Sheet - 66000 lb Duty Cycle

	Number of Duty Cycles
PINION PITTING:	
Life= 154 hours	24+
to 4966 hours	206+
PINION BENDING STRENGTH:	
Life= 732 hours	118+
to 8703 hours	1403+
GEAR PITTING:	
Life= 453 hours	73+
to 3767 hours	607+
GEAR BENDING STRENGTH:	
Life= 6785 hours	1094+
To More Than 100,000 hours	16000+

"Optimized" Program #540 Summary Sheet - 66000 lb Duty Cycle

	Number of Duty Cycles
PINION PITTING:	
Life= 572 hours	92+
to 4966 hours	800+
PINION BENDING STRENGTH:	
Life= 4421 hours	713+
To More Than 100,000 hours	16000+
GEAR PITTING:	
Life= 1684 hours	271+
to 14624 hours	2358+
GEAR BENDING STRENGTH:	
Life= 4235 hours	683+
To More Than 100,000 hours	16000+

NOTE: One duty cycle is 6.2 hours.

A range is given for the life of the gears if less than 100,000 hours. This is necessary as both high and low values of Sac and Sat from Tables 5 & 6 of AGMA 218 have been used by the program. This range can be extensive due to the rapid change of cycles with the load. (See Fig. 20 & 21 of AGMA 218.) The higher values may be used if special care is used in gearbox design, manufacture, and heat treatment.

The net increase in life for the L.S. Train is 271% for durability and 479% for strength.

6.1.3.2. Hot scoring. UTS Program 60-560 (TK) was used to obtain a probability of hot scoring. This program is based on AGMA Std 217.

The maximum speed condition (Cond #15) is more critical than the maximum torque condition (Cond #8).

The sump temperature (oil inlet to mesh) was estimated to be 180 °F.

The oil is SAE 40 with no extreme pressure additives. (Mobil Oil Corporation viscosity specifications for their 15W-40 motor oil indicates that the viscosity at 140 °F is in the center of the range allowed by SAE for 40 weight motor oils. Since the supplier of the oil is not specified, the hot scoring probability was computed at both ends of the allowable SAE range.)

For both gear sets the hot scoring probability is less than 1% at both ends of the viscosity range.

It should be noted that AGMA 217 does not give scoring probabilities for motor oils. The data used in UTS Program 60-560 for motor oils is from information gathered over a period of years by UTS staff and colleagues in the gear design field.

6.1.3.3. Cold scoring. UTS Program 60-5408 (TK) was used to obtain a probability of cold scoring. This program is based on Mobil Oil Corporation's EHL Guidebook, Third Edition.

For cold scoring the maximum torque condition (Cond #8) is more critical than the maximum speed condition (Cond #15).

The sump temperature (oil inlet to mesh) was estimated to be 180 °F.

The oil is Mobil Delvac 1240. (Mobil Oil Corporation states that the lubricant parameter for Delvac 1240 engine oil would be suitable for SAE 15W-40.)

The cold scoring probability is 5% for the "MLRS" gears and increases to 9% for the optimized gears. Since the scoring probability is less than 10% this increase is not considered significant.

6.1.4 Backlash. The backlash between gears is determined by the maximum material condition of the teeth. The effective tooth thickness of a tooth is larger than the measured tooth thickness except when measured with a parallel axis master gear which contacts from the specified start of active profile to the effective tooth tip. When measuring over two pins the effective tooth thickness is not measured and allowance must be made for errors in those elements of the gear not measured. The measurement over two pins does not account for lead error, pitch error, profile error and runout. Errors in these elements all reduce the backlash between the teeth. Calculations were made to determine the effective tooth thickness for low temperature operation while keeping the "hot" backlash within reasonable values. Root mean square values were used which covers more than 95% of cases.

Effective tooth thickness of optimized gears

18 tooth H.S. pinion:	0.5152"/0.5132"
34 tooth H.S. gear:	0.3784"/0.3764"
18 tooth L.S. pinion:	0.5825"/0.5805"
53 tooth L.S. gear:	0.4455"/0.4435"

Center distance limits of optimized housing:

H.S. Train Center Distance = 7.433"/7.430"
L.S. Train Center Distance = 10.290"/10.287"

Calculations were then made to find the temperature at which the assembled backlash becomes zero when the gears and the housing are at the SAME temperature. The assumed inspection temperature is 68 °F.

H.S. Train:

At minimum machined center distance and maximum effective tooth thickness the backlash would become zero at -42 °F.

At maximum machined center distance and minimum actual tooth thickness the backlash would be 0.0219" at +180 °F. (The minimum actual tooth thickness was used instead of the minimum effective to find an absolute limit on "hot" backlash for 100% of all drives.)

L.S. Train:

At minimum machined center distance and maximum effective tooth thickness the backlash would become zero at -48 °F

At maximum machined center distance and minimum actual tooth thickness the backlash would be 0.0289" at +180 °F. (The minimum actual rather than minimum effective tooth thickness was used to find an absolute limit on "hot" backlash for 100% of all drives.)

6.1.5. Computer Data. All computer data generated is furnished on two floppy disks labeled "H.S. Train, OPT Gears (Military)" and "L.S. Train, OPT Gears (Military)" and is part of the report. Appendix R contains a list of the files on these disks.

7.0. "EXPERT DESIGN SYSTEM" FOR MILITARY FINAL DRIVE ANALYSIS

Presentation of the "Expert Design System" consists of step by step instructions for utilizing the software programs to analyze a candidate military final drive. Figure 7-1 is a flow chart summarizing the decision path for using the software programs. The given example is not an existing design and is used for instruction only.

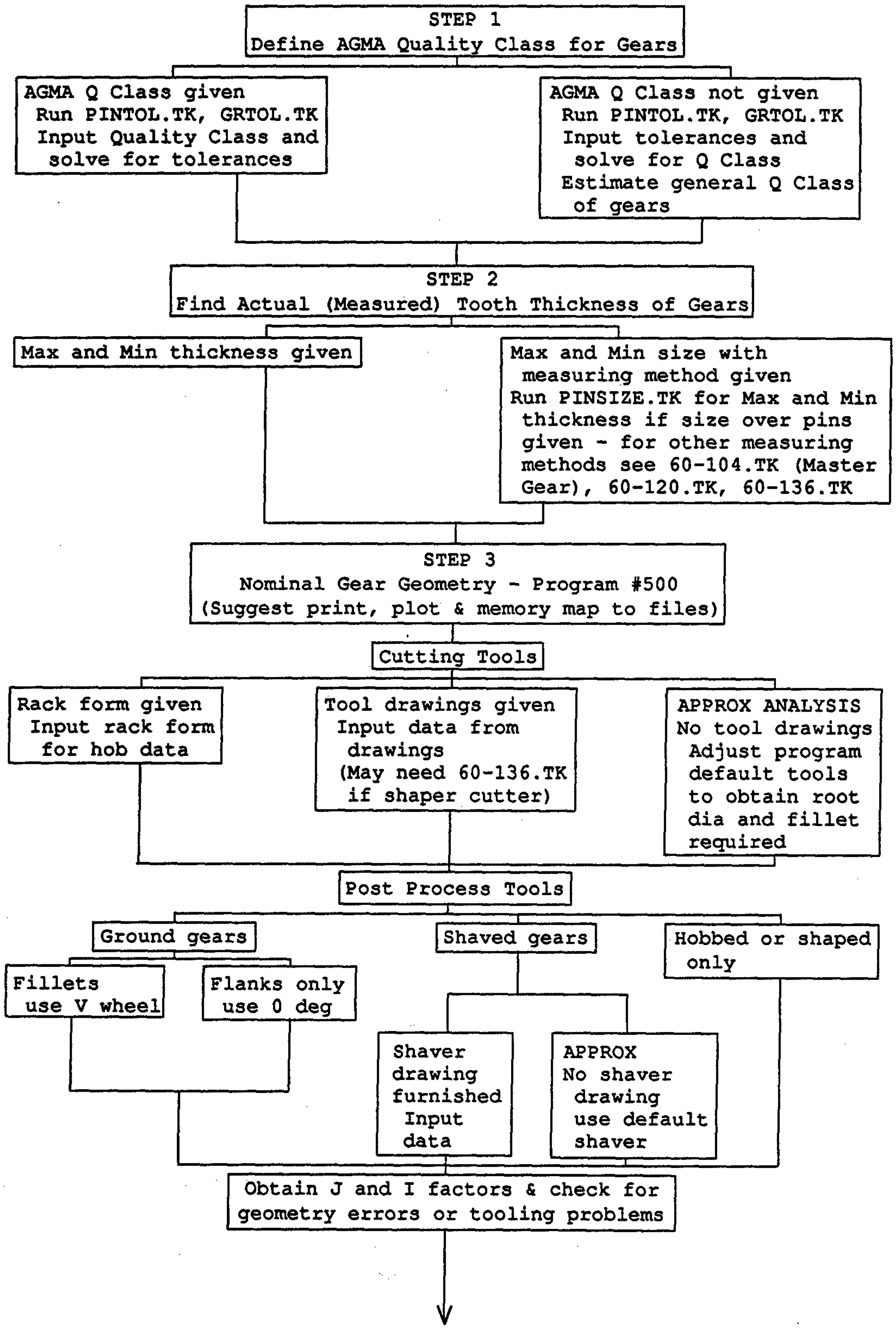
Before proceeding, run the programs "SETUP" and "TKSETUP" to configure the software for your computer, printer, plotter and any other devices you wish to use unless this has already been done.

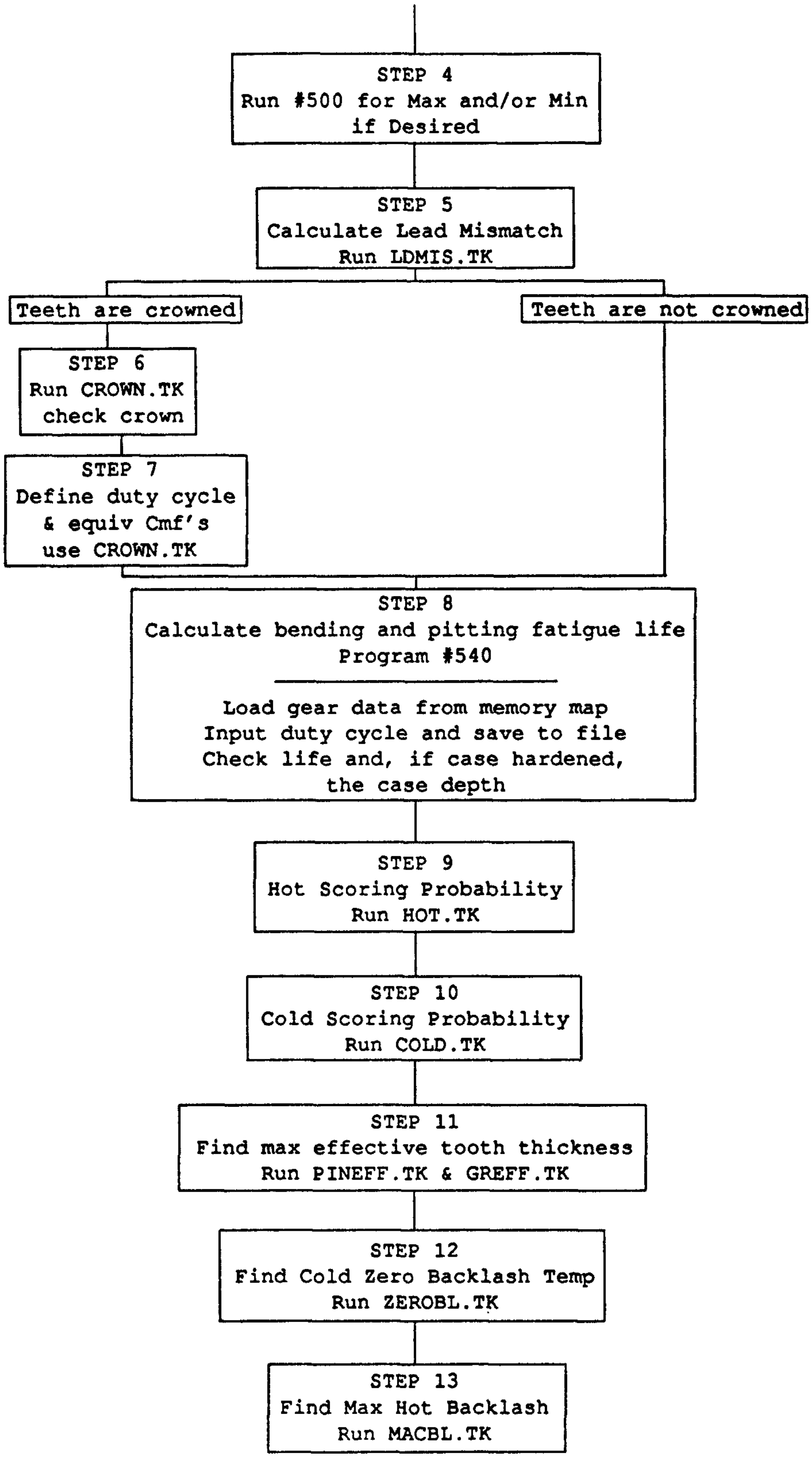
The UTS program TK Solver Plus will be abbreviated TK. It is assumed that TK is available on the same sub-directory as the TK models furnished for the analysis. It is also assumed that the user is familiar with the use of TK and of computer software generally.

The example set of gears are carburized and hardened steel spur gears running in an aluminum housing. There are 18 teeth in the pinion and 34 teeth in the gear. The normal diametral pitch is 3.5 and the nominal pressure angle is 25 degrees. Tip relief is not specified but one of the gears has crowned teeth. The input data for the various programs will furnish other details.

The drawings of the gears give the required root diameters but do not indicate a rack form (hob cutting edge dimensions) for generating the gears. (If a rack form is given the dimensions are input into program #500.) Without the rack form specified the best we can do is to adjust a computer generated form to match the full fillet or flat root condition specified and the root diameter given on the drawing. This would be done in program #500. Under these conditions the gear tooth is NOT fully defined and the possible variation in form

Figure 7-1. Flow Chart for Final Drive Gear Analysis




STEP 4
Run #500 for Max and/or Min
if Desired
STEP 5
Calculate Lead Mismatch
Run LDMIS.TK
Teeth are crowned
Teeth are not crowned
STEP 6
Run CROWN.TK
check crown
STEP 7
Define duty cycle
& equiv Cmf's
use CROWN.TK
STEP 8
Calculate bending and pitting fatigue life
Program #540
Load gear data from memory map
Input duty cycle and save to file
Check life and, if case hardened,
the case depth
STEP 9
Hot Scoring Probability
Run HOT.TK
STEP 10
Cold Scoring Probability
Run COLD.TK
STEP 11
Find max effective tooth thickness
Run PINEFF.TK & GREFF.TK
STEP 12
Find Cold Zero Backlash Temp
Run ZEROBL.TK
STEP 13
Find Max Hot Backlash
Run MACBL.TK

could cause a considerable difference in the bending fatigue life. All that the software can then provide is an approximation for bending fatigue life.

The AGMA Quality Class is not given but the individual tolerances are.

7.1. Step 1: Define the Quality Class of the Gears

Load TK and PINTOL. Clear the variables, input the known data, and solve by pressing F9.

For the 18-tooth pinion:

==================== VARIABLE SHEET ==

St	Input----	Name---	Output---	Unit-----	Comment------------------------------
					60-100 (Rev 1.1) ANSI/AGMA 2000-A88 Gear Classification and Inspection Handbook
		Q			AGMA Quality Number
		m			Message-Quality Number
	18	N			Number of teeth
	3.5	Pnd		1/in	Normal pitch
	0	psi		deg	Helix angle
	1.75	F		in	Face width
		Pd	3.5	1/in	Transverse pitch
		D	5.1429	in	Reference pitch diameter
		mn	7.2571429	mm	Normal module
	.0013	VrT		in	Radial Runout Tolerance (TIR)
		QRUN	12		Runout Quality Q#
		m1	'OK		
	.0005	VpA		in	Allowable Pitch Variation +/-
		QPIT	11		Pitch Quality Q#
		m2	'OK		
	.0005	VoT		in	Profile Tolerance
		QPRO	11		Profile Quality Q#
		m3	'OK		
	.0004	VyT		in	Tooth Alignment Tolerance
		QLD	11		Alignment Quality Q#
		m4	'OK		
		VqT		in	Tooth to Tooth Composite Tolerance
		QCOMP			Tooth Composite Quality Q#
		m5			
		VcqT		in	Total Composite Tolerance
		QTOT			Total Composite Quality Q#
		m6			

Ref:
Extracted from ANSI/AGMA 2000-A88 Gear Classification and Inspection Handbook, with permission of the publisher, American Gear Manufacturers Association, 1500 King Street, Suite 201, Alexandria, Virginia 22314

For the 34-tooth gear:

Reset TK (or load it) and load GRTOL. Clear the variables, input the known data and solve.

```
==================== VARIABLE SHEET ========================================
St Input---- Name--- Output--- Unit----- Comment------------------------------
                                         60-100 (Rev 1.1) ANSI/AGMA 2000-A88
                                           Gear Classification and Inspection
                                           Handbook
             Q                           AGMA Quality Number
             m                           Message-Quality Number
   34        N                           Number of teeth
   3.5       Pnd                1/in     Normal pitch
   0         psi                deg      Helix angle
   1.625     F                  in       Face width
             Pd      3.5        1/in     Transverse pitch
             D       9.7143     in       Reference pitch diameter
             mn      7.2571429  mm       Normal module

   .003      VrT                in       Radial Runout Tolerance (TIR)
             QRUN    10                  Runout Quality Q#
             m1      'OK

   .00051    VpA                in       Allowable Pitch Variation +/-
             QPIT    11                  Pitch Quality Q#
             m2      'OK

   .00064    VoT                in       Profile Tolerance
             QPRO    11                  Profile Quality Q#
             m3      'OK

   .00039    VyT                in       Tooth Alignment Tolerance
             QLD     11                  Alignment Quality Q#
             m4      'OK
```

Both gears fall generally into AGMA class Q11. Class Q11 will be used for the gears where required in the computer programs.

7.2. Step 2: Find Actual (Measured) Tooth Thickness of Gears

The gear drawings will specify the tooth thickness at the reference pitch diameter for the gears. If the tooth thickness is given we can use these values directly. Usually the size over pins is given as a method of specifying the tooth thickness. If only the size over pins is given we will need to calculate the tooth thickness at the reference pitch diameter. (If both are given you may still wish to check them.)

Suppose we have 6.0922"/6.0728" over 0.5600" pins for the pinion and 10.3944"/ 10.3914" over 0.4800" pins for the gear.

Load (or reset) TK and load PINSIZE. Clear the variables. Input the known data for the maximum size over pins for the pinion and solve.

```
==================== VARIABLE SHEET ==========================================
St Input---- Name--- Output--- Unit----- Comment-------------------------------
                                         60-1441 PIN MEASUREMENT-EXTERNAL GEARS
   18        n                           Number of teeth
   3.5       pn                1/in      Normal diametral pitch
   25        npa               deg       Normal pressure angle
   0         ha                deg       Nominal helix angle
             pd       5.1429   in        Ref pitch diameter
             db       4.661    in        Base diameter
             pt       3.5      1/in      Transverse diametral pitch
             tpa      25       deg       Transverse pressure angle
             ntt      .4881    in        Normal tooth thickness at ref pd
             ttt      .4881    in        Transverse tooth thickness at ref pd
             recpin   .53604   in        Pin dia for contact at tooth = space
             std_pin  .56000   in        Closest standard pin
   .56000    pin               in        Actual pin (or ball) diameter
             mea1     3.0461   in        Radius over one pin
             meab     6.0922   in        Measurement over two balls
   6.0922    mea2              in        Measurement over two pins
             mea3              in        Measurement over three pins
             minf              in        Minimum face width for three pins

             n_mod    7.2571429 mm       Normal module
             t_mod    7.2571429 mm       Transverse module
             ts_dia   5.2242718 in       Diameter where tooth = space
             r_pin_c  2.7661    in       Radius to center of actual pin
             pa_pin   .56884435 rad      Press angle at actual pin center
             inv_pa_  .07049497 rad      Involute actual pin center pa
             pa_cont  .47888442 rad      Press angle at contact of actual pin
             cont_d   5.2517876 in       Actual pin contact diameter
             inv_tpa  .02997535 rad      Involute of trans pa
             bha      0         deg      Base helix angle
```

For the minimum size over pins for the pinion:

```
==================== VARIABLE SHEET =========================================
St Input---- Name--- Output--- Unit----- Comment-------------------------------
                                         60-1441 PIN MEASUREMENT-EXTERNAL GEARS
   18        n                           Number of teeth
   3.5       pn                1/in      Normal diametral pitch
   25        npa               deg       Normal pressure angle
   0         ha                deg       Nominal helix angle
             pd      5.1429    in        Ref pitch diameter
             db      4.661     in        Base diameter
             pt      3.5       1/in      Transverse diametral pitch
             tpa     25        deg       Transverse pressure angle
             ntt     .4766     in        Normal tooth thickness at ref pd
             ttt     .4766     in        Transverse tooth thickness at ref pd
             recpin  .53074    in        Pin dia for contact at tooth = space
             std_pin .56000    in        Closest standard pin
   .56000    pin               in        Actual pin (or ball) diameter
             mea1    3.0364    in        Radius over one pin
             meab    6.0728    in        Measurement over two balls
   6.0728    mea2              in        Measurement over two pins
```

For the maximum size over pins for the gear:

```
==================== VARIABLE SHEET =========================================
St Input---- Name--- Output--- Unit----- Comment-------------------------------
                                         60-1441 PIN MEASUREMENT-EXTERNAL GEARS
   34        n                           Number of teeth
   3.5       pn                1/in      Normal diametral pitch
   25        npa               deg       Normal pressure angle
   0         ha                deg       Nominal helix angle
             pd      9.7143    in        Ref pitch diameter
             db      8.8041    in        Base diameter
             pt      3.5       1/in      Transverse diametral pitch
             tpa     25        deg       Transverse pressure angle
             ntt     .4655     in        Normal tooth thickness at ref pd
             ttt     .4655     in        Transverse tooth thickness at ref pd
             recpin  .51040    in        Pin dia for contact at tooth = space
             std_pin .48000    in        Closest standard pin
   .48000    pin               in        Actual pin (or ball) diameter
             mea1    5.1972    in        Radius over one pin
             meab    10.3944   in        Measurement over two balls
   10.3944   mea2              in        Measurement over two pins
```

For the minimum size over pins for the gear:

```
==================== VARIABLE SHEET ========================================
St Input---- Name--- Output--- Unit----- Comment-------------------------------
                                         60-1441 PIN MEASUREMENT-EXTERNAL GEARS
   34        n                           Number of teeth
   3.5       pn                 1/in     Normal diametral pitch
   25        npa                deg      Normal pressure angle
   0         ha                 deg      Nominal helix angle
             pd      9.7143     in       Ref pitch diameter
             db      8.8041     in       Base diameter
             pt      3.5        1/in     Transverse diametral pitch
             tpa     25         deg      Transverse pressure angle
             ntt     .464       in       Normal tooth thickness at ref pd
             ttt     .464       in       Transverse tooth thickness at ref pd
             recpin  .51005     in       Pin dia for contact at tooth = space
             std_pin .48000     in       Closest standard pin
   .48000    pin                in       Actual pin (or ball) diameter
             mea1    5.1957     in       Radius over one pin
             meab    10.3914    in       Measurement over two balls
   10.3914   mea2               in       Measurement over two pins
```

Specified tooth thickness

18 tooth pinion: 0.4881"/0.4766"
34 tooth gear: 0.4655"/0.4640"

7.3. Step 3: Run Program #500 for Analysis of Nominal Geometry of the Gear Set

Most of the analysis will be done using "nominal" center distance, max tooth thickness and outside diameters.

```
                              TACOM
                          Gear Analysis
                       ==================
     * Denotes Input Data
       * Normal Diam Pitch=    3.5000          Opr Diam Pitch=    3.4667
  * Normal Pressure Angle=    25.0000      Opr Pressure Angle=   26.1453
            * Helix Angle=     0.0000
         Trans Diam Pitch=     3.5000          Line of Action=    1.1311
     Trans Pressure Angle=    25.0000      % Approach Action=   47.81
                                             % Recess Action=   52.19
      Opr Center Distance=     7.5000             Profile C.R.=    1.3904
             * Face Width=     1.6250
          Basic Backlash=      0.0014
       Total Operating BL=     0.0121

                                DRIVER (Deg Roll)      DRIVEN (Deg Roll)
            * Number of Teeth=  18                     34
          * Outside Diameter=    5.8100  (42.64)       10.3300  (35.16)
  * Total Normal Finish Stock=   0.0150                  0.0150
```

HOB FORM DATA	NON-TOPPING		NON-TOPPING	
* Hob Pressure Angle=	25.0000		25.0000	
* Hob Tip to Ref Line=	0.3785		0.3800	
* Hob Tooth Thickness at Ref=	0.4338		0.4338	
* Both: Full Rad-Hob Tip Radius=	0.0772		0.0761	
* Hob Protuberance=	0.0080		0.0080	
Hob SAP from Ref Line=	0.2315		0.2564	
Hob Space Width at Hob SAP=	0.2479		0.2247	
Normal Tooth Thickness at OD=	0.1189		0.1510	
Normal Tooth Thickness,(Hobbed)=	0.5031		0.4805	
*Normal Tooth Thickness,(Ground)=	0.4881		0.4655	
Dia @ Mid-point of Line of Action=	5.2143	(28.73)	9.7859	(27.80)
Pitch Diameter, (Ref)=	5.1429	(26.72)	9.7143	(26.72)
Operating Pitch Diameter=	5.1923	(28.13)	9.8077	(28.13)
Base Diameter=	4.6610		8.8041	
Dia, (Start of Active Profile)=	4.8146	(14.83)	9.3477	(20.44)
Form Diameter=	4.8075	(14.48)	9.3406	(20.30)
Root Diameter=	4.4701		8.9901	
Root Clearance=	0.0999		0.0999	
Max Undercut=	0.0081		0.0083	
Diameter at Max Undercut=	4.7279	(9.74)	9.1800	(16.92)
* Finished Grind Diameter=	4.7279	(9.74)	9.1800	(16.92)
Roll, radians, (1 tooth load)=	0.608	(34.83)	0.542	(31.03)
Minimum Fillet Radius=	0.1009		0.0920	
Helical Factor, C(h)=	1.000		1.000	
Y Factor=	0.725		0.795	
Load Sharing Ratio, m(N)=	1.000		1.000	
MODIFIED Stress Corr Fact, K(f)=	1.633		1.666	
J-Factor=	0.444		0.477	
I Factor=		0.115		
Max Specific Sliding Ratio=	1.37	(14.83)	1.09	(20.44)

Steel Gears, Finish Ground
Case Carburized

Universal Technical Systems, Inc, Rockford, Ill 61101 (Program #500)
Date: 0/0/00
Job : Sample

The data marked with "*" are items that were input either by the operator or the program as a "default". (If a "default" is displayed, it will be entered if the enter key is pressed without typing an entry.)

The printout, tooth plots, and specific sliding plot labeled "Sample" are attached as Appendix L.

During operation, program #500 will ask if you wish to input "delta addendum". To analyze an existing design reply with "N" (no). This will turn off the dynamic default system and use entered data from the gear drawings.

The program was told to use non-topping full radius tip hobs. The program-generated (default) hobs did not provide the required root diameter. The program-generated hobs use the standard clearances to obtain the hob addendum.

The program uses a clearance of 0.025/Normal Pitch for gears that are hobbed only, 0.035/NP for shaved gears, and 0.040/NP for ground gears. The "Hob tip to reference line" was then adjusted to comply with the root diameter required.

Note: If a full radius hob is required, input a large number (such as 1") for the hob tip radius while in the hob design loop. The program will then adjust the hob tip radius to a full radius with the new "Hob tip to reference line".

The stress correction factor, Kf, is not the standard AGMA factor. The optional modified Kf in the program uses the radius of curvature where the J factor (and stress) is evaluated to calculate Kf, while the standard AGMA Kf uses the radius of curvature of the fillet at the root of the tooth instead. The AGMA approach is more conservative, but a more accurate evaluation is recommended for military final drives. The difference in the resulting J factor is not large, but the difference in calculated life can be substantial due to the flat character of the material stress/cycle ratio.

Note that the tooth plot of the pinion (Appendix L) shows that the start of active profile, Dia(sap), is below the point where the undercut meets the involute profile on the hobbed tooth. This indicates that the amount of finish stock is less than anticipated near the bottom of the active portion of the tooth if this hob geometry is used. In some cases heat treat distortion may be enough to prevent this area from cleaning up on some of the teeth. A short lead hob would place the undercut lower on the tooth and provide full finish stock in this area. Since the low stock situation could cause rejection of some gears at final inspection, a short lead hob could be considered; however, a short lead hob would decrease the radius of curvature in the root area and probably reduce the J factor to some extent.

7.4. Step 4: Run Program #500 for Analysis of Max/Min Geometries of Gear Set

If desired, the analysis can be run twice, once at "minimum" conditions and once at "maximum" conditions to "box in" the design.

"Minimum" conditions:

Minimum center distance
Maximum tooth thickness
Maximum outside diameters
Minimum finish stock

"Maximum" conditions:

Maximum center distance
Minimum tooth thickness
Minimum outside diameters
Maximum finish stock

7.5. Step 5: Calculate the Total Lead Mismatch Between Teeth

An estimate of the mismatch across the face of the gears is required along with a face mismatch factor, Cmf. An estimate of the mismatch is made from the lead errors allowed on the gears and the shaft misalignment allowed by the housing specifications.

From the gear drawings:

Tooth alignment (lead) error, 18 T = 0.0004"
Tooth alignment (lead) error, 34 T = 0.00039"

From the housing drawings:

Maximum out of parallel = 0.003" in 7.2"
Maximum out of plane = 0.004" in 7.2"

Load (or reset) TK and load LDMIS.

```
=================== VARIABLE SHEET ========================================
St Input---- Name--- Output--- Unit----- Comment--------------------------------
                                          LDMIS Total Lead Mismatch for
                                            Gear Pair in Housing

   26.14      opr_pa              deg      Operating Pressure Angle
                                           Pinion:
   1.75       face_p              in        Face Width
   .0004      lead_p              in        Tooth Alignment (Lead) Error
                                           Gear:
   1.625      face_g              in        Face Width
   .00039     lead_g              in        Tooth Alignment (Lead) Error

                                           Housing (Shaft Alignment):
   7.2        D                   in        Distance Between Bearings
   .003       paral               in        Shafts Maximum Out of Parallel
   .004       plane               in        Shafts Maximum Out of Plane

              ma       .001       in       Mean Average Mismatch
              rms      .0011      in       Root Mean Square Mismatch
              max      .002       in       Maximum Mismatch
```

The model calculates the mean average, root mean square and maximum possible mismatch. For military final drives the mean average misalignment is sufficient.

7.6. Step 6: Check the Effect of the Crown

Load (or reset) TK and load CROWN.

The crown specified for the 34-tooth gear is 0.0006" to 0.0010". With the "CROWN" model we can quickly find the compressive stress with straight and crowned teeth. Under conditions where the amount of gear face misalignment or mismatch is clearly defined, a straight tooth may result in lower compressive stress than a crowned tooth. However, it is probably risky to design with no crown because any unexpected deflection would increase the face mismatch and could then cause much higher stress than calculated.

Clear the variables, input the known data and solve (F9).

Use the speed and torque from the maximum torque condition of the duty cycle. For the first solution we will leave the amount of crown blank and obtain the compressive stress for a straight tooth. Set "Blank Previous Stress?" to 'y to clear any data in the plot lists left from a previous solution. Do not disturb the "L" in the status column of "PINION Speed","PINION Torque", "Cmf" or "Eq_cmf". We will need them later.

```
==================== VARIABLE SHEET =======================================
St Input---- Name--- Output--- Unit----- Comment------------------------------
                                         60-5406 CROWNED & STRAIGHT EXTERNAL
                                             GEAR COMPRESSIVE STRESS (Rev 1.2)
             m1      'STRAIGHT           Message Field
             m2      'TEETH
             m3      'PROF_NOT
             m4      'MODIFIED
                                         * Denotes Items Normally Input

                                         COMMON DATA:
             adj     'y                   * Adjust Tooth Load per AGMA 217?
                                                            Def='y=yes 'n=no
             pmod    'n                   * Profiles Modified? 'y=yes Def='n=no
             drv     'p                   * Driver:  Def='p=pinion 'g=gear
   11        Q                            * Quality Class (AGMA 390.03)
L  390       n                 rpm        * PINION Speed
L  22500     tork              lb-in      * PINION Torque
             power   139.23    hp           Power
             tan     8666.6667 lb           Tangential Load at Operating PD
             vt      530.1     ft/min       Pitch Line Velocity
   3.5       pn                1/in       * Normal pitch
             n_mod   7.2571429 mm           Normal Module
   25        npa               deg        * Normal Pressure Angle
   0         ha                deg        * Helix Angle
             pt      3.5       1/in         Transverse Pitch
             t_mod   7.2571429 mm           Transverse Module
             tpa     25        deg          Transverse Pressure Angle
             bha     0         deg          Base Helix Angle
   7.5       cd                in         * Operating Center Distance
             opr_tpa 26.1457   deg          Operating Trans Pressure Angle
             opr_ha  0         deg          Operating Helix Angle
             face    1.625     in           Effective Face Width
   .001      et                in         * Total Lead Mismatch Between Teeth
```

				AGMA 218 - Analytical Method
	Ca	1		* Application Factor (Default=1)
	Cv	.946		Dynamic Factor
L	Cmf	1.188		Face Load Distribution Factor
	Lcmf	1.625	in	Length of Contact, Straight Tooth
L	Eq_cmf	'NA		Equiv Face Load Distribution Factor for Crowned Teeth
	G	2E6	psi	Tooth Stiffness Constant
	dist	.8125	in	Distance from Center of Tooth to Initial Tooth Contact
	mg	1.8888889		Gear Ratio
	mp	1.3903		Profile Contact Ratio (Theoretical)
	mh	0		Helical Contact Ratio (Theoretical)
				PINION DATA:
18	Np			* Number of Teeth
5.81	odp		in	* Outside Diameter
	crn_p	0	in	* Transverse Crown Rise (Default=0)
1.75	face_p		in	* Face Width
	dist_p	.0625	in	Tooth End to Initial Contact Point
	ell_p		in	Tooth End to Crown Contact Ellipse With Crown Centered On Tooth
	Ep	3E7	psi	* Young's Modulus (Default=steel)
	poi_p	.3		* Poisson's Ratio (Default=.3)
	pdp	5.1923	in	Operating Pitch Diameter
	sap_p	4.8146	in	Start of Active Profile-No Undercut
	dbp	4.661	in	Base Diameter
				GEAR DATA:
34	Ng			* Number of Teeth
10.33	odg		in	* Outside Diameter
	crn_g	0	in	* Transverse Crown Rise (Default=0)
1.625	face_g		in	* Face Width
	dist_g	0	in	Tooth End to Initial Contact Point
	ell_g		in	Tooth End to Crown Contact Ellipse With Crown Centered On Tooth
	Eg	3E7	psi	* Young's Modulus (Default=steel)
	poi_g	.3		* Poisson's Ratio (Default=.3)
	pdg	9.8077	in	Operating Pitch Diameter
	sap_g	9.3477	in	Start of Active Profile-No Undercut
	dbg	8.8041	in	Base Diameter
	chk	36		* Number of Pinion Roll Angle Calculation Intervals (Default=36)
				COMPRESSIVE STRESS:
	sc_crit	250424	psi	Compressive Stress at LPSTC for
	loc	'LPSTC		Spur and LCR Helicals or at Mean
	E_loc	22.638	deg	Pinion Diameter for Full Helicals Includes Ca,Cv,Cmf (Straight Teeth)
	max_c	250424	psi	Maximum Compressive Stress
	E_max_c	22.638	deg	Pinion Roll Angle at Max Comp Stress

ROLL ANGLES:

			Pinion:
E_ld_p	14.831	deg	Start of Active Profile
E_lp_p	22.638	deg	Low Single Tooth Contact (LPSTC)
E_pdp	28.126	deg	Pitch Diameter, Operating
E_hp_p	34.831	deg	High Single Tooth Contact
E_od_p	42.638	deg	Outside Diameter, Effective
			Gear:
E_ld_g	20.443	deg	Start of Active Profile
E_lp_g	24.576	deg	Low Single Tooth Contact (LPSTC)
E_pdg	28.126	deg	Pitch Diameter, Operating
E_hp_g	31.031	deg	High Single Tooth Contact
E_od_g	35.164	deg	Outside Diameter, Effective

'y blk Blank Previous Stress?'y=yes Def='n=no

References:
Data Extracted from AGMA 218.01, AGMA Standard for Rating the Pitting Resistance and Bending Strength of Spur and Helical Involute Gear Teeth and AGMA 217.01, AGMA Information Sheet - Gear Scoring Design Guide for Aerospace Spur and Helical Power Gears with the permission of the publisher, the American Gear Manufacturers Association, 1500 King Street, Suite 201, Alexandria, Virginia 22314

Mobil EHL Guidebook, Third Edition Mobil Oil Corporation, Commercial Marketing, Technical Publications, 3225 Gallows Road, Fairfax, Virginia 22037

For a straight tooth the maximum compressive stress is 250424 psi at the lowest point of single tooth contact on the pinion. A plot of compressive stress from the start of active profile to the OD of the pinion is available. If you wish you may plot "stress" on the plot sheet but this information will be retained while we check the stress with the crown on the tooth.

Note: If the gears did not have a crown you could skip to step 8.

Input the crown on the 34 tooth gear. Set "Blank Previous Stress?" to 'n (or leave it blank) to retain the data in the plot lists for the straight tooth. Solve again (F9).

```
=================== VARIABLE SHEET ========================================
St Input---- Name--- Output--- Unit----- Comment-------------------------------
                                         60-5406 CROWNED & STRAIGHT EXTERNAL
                                             GEAR COMPRESSIVE STRESS (Rev 1.2)
             m1      'CROWNED            Message Field
             m2      'TOOTH
             m3      'CONTACT
             m4      'OFF_END
                                         * Denotes Items Normally Input

                                         COMMON DATA:
             adj     'y                   * Adjust Tooth Load per AGMA 217?
                                                                Def='y=yes 'n=no
             pmod    'n                   * Profiles Modified? 'y=yes Def='n=no
             drv     'p                   * Driver:  Def='p=pinion 'g=gear
   11        Q                            * Quality Class (AGMA 390.03)
L  390       n                 rpm        * PINION Speed
L  22500     tork              lb-in      * PINION Torque
             power   139.23    hp           Power
             tan     8666.6667 lb           Tangential Load at Operating PD
             vt      530.1     ft/min       Pitch Line Velocity
   3.5       pn                1/in       * Normal pitch
             n_mod   7.2571429 mm           Normal Module
   25        npa               deg        * Normal Pressure Angle
   0         ha                deg        * Helix Angle
             pt      3.5       1/in         Transverse Pitch
             t_mod   7.2571429 mm           Transverse Module
             tpa     25        deg          Transverse Pressure Angle
             bha     0         deg          Base Helix Angle
   7.5       cd                in         * Operating Center Distance
             opr_tpa 26.1457   deg          Operating Trans Pressure Angle
             opr_ha  0         deg          Operating Helix Angle
             face    1.625     in           Effective Face Width
   .001      et                in         * Total Lead Mismatch Between Teeth
                                            AGMA 218 - Analytical Method
             Ca      1                      * Application Factor (Default=1)
             Cv      .946                     Dynamic Factor
L            Cmf     'NA                      Face Load Distribution Factor
             Lcmf    'NA       in             Length of Contact, Straight Tooth
L            Eq_cmf  1.336                    Equiv Face Load Distribution
                                                Factor for Crowned Teeth
             G       2E6       psi            Tooth Stiffness Constant
             dist    .2539     in           Distance from Center of Tooth to
                                              Initial Tooth Contact
             mg      1.8888889              Gear Ratio
             mp      1.3903                 Profile Contact Ratio (Theoretical)
             mh      0                      Helical Contact Ratio (Theoretical)
```

				PINION DATA:
18	Np			* Number of Teeth
5.81	odp		in	* Outside Diameter
	crn_p	0	in	* Transverse Crown Rise (Default=0)
1.75	face_p		in	* Face Width
	dist_p	.6211	in	Tooth End to Initial Contact Point
	ell_p	-.4059	in	Tooth End to Crown Contact Ellipse With Crown Centered On Tooth
	Ep	3E7	psi	* Young's Modulus (Default=steel)
	poi_p	.3		* Poisson's Ratio (Default=.3)
	pdp	5.1923	in	Operating Pitch Diameter
	sap_p	4.8146	in	Start of Active Profile-No Undercut
	dbp	4.661	in	Base Diameter
				GEAR DATA:
34	Ng			* Number of Teeth
10.33	odg		in	* Outside Diameter
.0008	crn_g		in	* Transverse Crown Rise (Default=0)
1.625	face_g		in	* Face Width
	dist_g	.5586	in	Tooth End to Initial Contact Point
	ell_g	-.4684	in	Tooth End to Crown Contact Ellipse With Crown Centered On Tooth
	Eg	3E7	psi	* Young's Modulus (Default=steel)
	poi_g	.3		* Poisson's Ratio (Default=.3)
	pdg	9.8077	in	Operating Pitch Diameter
	sap_g	9.3477	in	Start of Active Profile-No Undercut
	dbg	8.8041	in	Base Diameter
	chk	36		* Number of Pinion Roll Angle Calculation Intervals (Default=36)
				COMPRESSIVE STRESS:
	sc_crit	265661	psi	Compressive Stress at LPSTC for
	loc	'LPSTC		Spur and LCR Helicals or at Mean
	E_loc	22.638	deg	Pinion Diameter for Full Helicals
				Includes Ca,Cv,Cmf (Straight Teeth)
	max_c	265661	psi	Maximum Compressive Stress
	E_max_c	22.638	deg	Pinion Roll Angle at Max Comp Stress
				ROLL ANGLES:
				Pinion:
	E_ld_p	14.831	deg	Start of Active Profile
	E_lp_p	22.638	deg	Low Single Tooth Contact (LPSTC)
	E_pdp	28.126	deg	Pitch Diameter, Operating
	E_hp_p	34.831	deg	High Single Tooth Contact
	E_od_p	42.638	deg	Outside Diameter, Effective
				Gear:
	E_ld_g	20.443	deg	Start of Active Profile
	E_lp_g	24.576	deg	Low Single Tooth Contact (LPSTC)
	E_pdg	28.126	deg	Pitch Diameter, Operating
	E_hp_g	31.031	deg	High Single Tooth Contact
	E_od_g	35.164	deg	Outside Diameter, Effective
	blk	'n		Blank Previous Stress?'y=yes Def='n=no

For the crowned tooth the maximum compressive stress is 265661 psi at the lowest point of single tooth contact on the pinion compared to 250424 psi for the straight tooth. Plots are already set up on the plot sheet for investigation of various conditions for straight and crowned gears. The plot "stress2" is attached as Appendix M. The addition of the crown results in higher compressive stress all along the tooth for this gear set. Since, for final drives, the additional mismatch due to deflection of the housing is usually somewhat unsure, the application of reasonable crown is good insurance against unexpected high tooth end load. (If the deflection is higher than anticipated the straight tooth could result in lower life than the crowned tooth.)

Of course the equivalent Cmf for the crowned teeth, 1.336, is higher than the Cmf for the straight teeth, 1.188, under these conditions.

7.7. Step 7: Define the duty cycle and calculate equivalent Cmf

Program #540 for stress and life will calculate Cmf for straight teeth as a default for all duty cycle conditions. If you had straight teeth you could skip this section, or you might wish to obtain the Cmf factors and define the duty cycle for separate reference from the output of program #540.

Since we are using a crowned tooth it will be necessary to calculate the equivalent Cmf and type over the default values in program #540. (AGMA Std 218 does not contain a method for analyzing crowned teeth.) The model "CROWN" contains a table to allow us to do this without entering the torque and speed for each duty cycle condition one at a time. (This is the reason some of the variables are associated with lists by typing 'L' in their Status fields on the Variable Sheet.)

Go to the Table Sheet in the model and dive twice into the interactive table "duty". (=t >>) Type in the duty cycle conditions.

==================== TABLE: duty ==

Title: Duty Cycle Table (Torque in lb-in)

Element	#--	RPM--	Torque-	Time,Min	Cmf-	Eq_Cmf	--------	--------	--------
1	1	280	12000	30					
2	2	390	22500	25					
3	3	1230	9300	50					
4	4	1440	4500	75					
5	5	3000	4600	90					

Now list solve (F10) and the equivalent Cmf will be calculated and placed in the table. (If we had straight teeth Cmf would be calculated and the column for Eq_Cmf would contain 'NA.)

```
==================== TABLE: duty ========================================
Title:       Duty Cycle Table (Torque in lb-in)
Element #--  RPM-- Torque- Time,Min  Cmf- Eq_Cmf -------- -------- --------
1        1     280   12000    30     'NA    1.568
2        2     390   22500    25     'NA    1.336
3        3    1230    9300    50     'NA    1.656
4        4    1440    4500    75     'NA    2.029
5        5    3000    4600    90     'NA    1.995
```

Surface back up to the Table Subsheet (<) and print the table for reference when we run program #540 later. Type "P" in the 'Printer or Screen' field, then press F8.

Duty Cycle Table (Torque in lb-in)

#	RPM	Torque	Time,Min	Cmf	Eq_Cmf
1	280	12000	30	NA	1.568
2	390	22500	25	NA	1.336
3	1230	9300	50	NA	1.656
4	1440	4500	75	NA	2.029
5	3000	4600	90	NA	1.995

7.8. Step 8: Calculate the bending and pitting fatigue lives

Load and run program #540.

The program #500 gear data required may be loaded into #540 from the memory map file 500.

Use only the "analytical" method for analyzing final drives. The "empirical" method is not exhaustive enough to obtain accurate life data.

Use an application factor, Ca, of one if you are using a duty cycle. Any overloads will be part of the duty cycle.

Use a reliability factor, CR, of 0.9. A factor of 0.9 results in less than 1 "failure" out of 20 drives. "Failure" means that 1 drive out of 20 will run less than the calculated life and 19 drives out of 20 will run longer than the calculated life. Commercial drives are usually designed for a 1 in 100 failure rate. A failure rate of 1 in 20 is compatible with military practice.

The duty cycle conditions for "Miner's Rule" may be entered from the reference table printed from the "CROWN" program or directly from the source data.

When the program displays the default value for the "Face Load Distribution Factor, Cmf" for each duty cycle condition, the default will be the Cmf for straight teeth. Enter the value of equivalent Cmf, Eq_Cmf, from the reference table printed from the model "CROWN". If you had straight teeth, of course, you would use the default value.

The complete printout from program #540 for this example is attached as Appendix N.

Program #540 Life Summary

	Number of Duty Cycles
PINION PITTING:	
Life= 331 hours	73+
to 2715 hours	603+
PINION BENDING STRENGTH:	
Life= 61901 hours	13755+
To More Than 100,000 hours	22000+
GEAR PITTING:	
Life= 625 hours	138+
to 5129 hours	1139+
GEAR BENDING STRENGTH:	
Life Is More Than 100,000 hours	22000+

NOTE: One duty cycle is 4.5 hours.

A range is given for the life of the gears if less than 100,000 hours. This is necessary as both values of Sac and Sat from Tables 5 & 6 of AGMA 218 have been used by the program. This range can be extensive due to the rapid change of cycles with the load. (See Fig. 20 & 21 of AGMA 218.) The higher values may be used if special care is used in gearbox design, manufacture, and heat treatment.

The minimum case depth to the 50 Rc/C point is usually specified based on pitch. AGMA 218 recommends a case depth for this pitch of 0.0408" to 0.0647". (See AGMA 218, Fig. 11 and the #540 printout, Appendix N.) This case is not enough to stay safely below the depth to maximum sub-surface shear for some of the duty cycle conditions. The #540 printout shows that for duty cycle condition #2, a case depth of 0.0602" is required for the load. The case depth on the gear drawings must be about 0.060" minimum or spalling may result. If a smaller case depth is used the higher life values for these gears is questionable.

7.9. Step 9: Hot Scoring Calculations

Hot scoring occurs when the temperature at the mesh point rises high enough to flash the lubricant off of the teeth. Welding of the gear teeth then takes place due to dry metal to metal contact. Subsequent sliding and rolling then tears the welds apart and produces radial score lines on the teeth. Hot scoring usually occurs when gears are first put into service at full speed and load. A "break-in" period at reduced speed and load can increase the resistance to scoring considerably.

For ground gears the pinion tooth surface should be no more than about 20 micro-inches (and may be as low as 10 micro-inches) after break-in. The gear should be no more than about 25 micro-inches (and may be as low as 15 micro-inches) after break-in. We will use 20 micro-inches surface finish for the pinion and 25 for the gear for hot scoring calculations.

For hot scoring the maximum speed condition (Cond #5) is more critical than the maximum torque condition (Cond #2). (If there is any doubt which condition is critical it is a simple matter to check them all.)

The sump temperature (oil inlet to mesh) is 180 °F.

The oil specified is SAE 15W-40 motor oil. The oil checked in the computer model is SAE 40 with no extreme pressure additives. (Mobil Oil Corporation viscosity specifications for their SAE 15W-40 motor oil indicates that the viscosity at 140 °F is in the center of the range allowed by SAE for 40 weight motor oils. Since the supplier of the oil is not specified, the hot scoring probability was computed at both ends of the allowable SAE range.)

UTS Program 60-560 (TK) will be used to obtain a probability of hot scoring for the gears. This program is based on AGMA Std 217.

Load (or reset) TK and load HOT. Clear the variables, input the known data and solve.

```
=================== VARIABLE SHEET ==========================================
St Input---- Name--- Output--- Unit----- Comment-------------------------------
                                         60-560 EXTERNAL GEAR SCORING (Rev 1.1)
             m1      '_                  Reference: AGMA Std #217.01
             m2      '_                   Information Sheet - Gear Scoring
             m3      '_                    Design Guide for Aerospace
             m4      '_                    Spur and Helical Power Gears

                                         * Denotes Items Normally Input

                                         COMMON DATA:
             pmod    'n                   *Profiles Modified? 'y=yes Def='n=no
             drv     'p                   *Driver:  Def='p=pinion 'g=gear
   3000      n                 rpm        *PINION Speed
   4600      tork              lb-in      *PINION Torque
             power   218.96    hp          Power
   3.5       pn                1/in       *Normal pitch
             n_mod   7.2571429 mm          Normal Module
   25        npa               deg        *Normal Pressure Angle
   0         ha                deg        *Helix Angle
   7.5       cd                in         *Operating Center Distance
             opr_tpa 26.1457   deg         Operating Trans Pressure Angle
   1.625     face              in         *Face Width
   1.995     Cm                           *Load Distribution Factor
```

	fin	22.64	uin	Composite Surface Finish, rms (After break-in)
	mg	1.8888889		Gear Ratio
	pt	3.5	1/in	Transverse Pitch
	t_mod	7.2571429	mm	Transverse Module
	tpa	25	deg	Transverse Pressure Angle
				PINION DATA:
18	Np			*Number of Teeth
5.81	odp		in	*Outside Diameter
	pdp	5.1923	in	Operating Pitch Diameter
20	pfin		uin	*Surface Finish, rms (After Break-in)
	sap_p	4.8146	in	Start of Active Profile
	dbp	4.661	in	Base Diameter
				GEAR DATA:
34	Ng			*Number of Teeth
10.33	odg		in	*Outside Diameter
	pdg	9.8077	in	Operating Pitch Diameter
25	gfin		uin	*Surface Finish, rms (After Break-in)
	sap_g	9.3477	in	Start of Active Profile
	dbg	8.8041	in	Base Diameter
				TEMPERATURES:
180	intemp		F	*Initial Lubricant (Inlet) Temp
	flash	313	F	Flash Temperature Index
				LUBE: (AGMA 217.01)
				Scoring Probability:
	probL	17.4	%	MIL-L-7808
	probO	74.6	%	MIL-O-6081 Grade 1005
				LUBE: Non-Reactive (NOT AGMA Std 217)
				See tables "NON_AGMA" & "vis"
'y	update			Update Tables to Flash Temp Index and Viscosity? 'y=yes Default='n=n
				PINION TEMP RISE: (Press F7 for Plot)
	chk	36		No. of Check Intervals (Default=36)
	R_ld_p	133	dF	Start of Active Profile
	R_lp_p	110	dF	Low Single Tooth Contact
	R_pdp	0	dF	Pitch Diameter, Operating
	R_hp_p	121	dF	High Single Tooth Contact
	R_od_p	113	dF	Outside Diameter, Effective
	max_t	133	dF	Maximum Temperature Rise
	E_max_t	14.831	deg	Roll Angle at Max Temp Rise
				ROLL ANGLES:
				Pinion:
	E_ld_p	14.831	deg	Start of Active Profile
	E_lp_p	22.638	deg	Low Single Tooth Contact

E_pdp	28.126	deg	Pitch Diameter, Operating
E_hp_p	34.831	deg	High Single Tooth Contact
E_od_p	42.638	deg	Outside Diameter, Effective
			Gear:
E_ld_g	20.443	deg	Start of Active Profile
E_lp_g	24.576	deg	Low Single Tooth Contact
E_pdg	28.126	deg	Pitch Diameter, Operating
E_hp_g	31.031	deg	High Single Tooth Contact
E_od_g	35.164	deg	Outside Diameter, Effective

The flash temperature is 313 °F.

It should be noted that AGMA 217 does not give scoring probabilities for motor oils. The data used in UTS Program 60-560 for motor oils is from data gathered over a period of years by UTS staff and colleagues in the gear design field.

The Variable Sheet does not give the scoring probability for oils that are not covered by AGMA Std 217. The scoring probability for AGMA gear oils and SAE crank oils are in the table "NON_AGMA". Go to the Table Sheet, place the cursor on "NON_AGMA", and press F8.

Scoring Probability - Non-Reactive AGMA & SAE Oils (Not AGMA 217)

Flash	AGMA	Score Prob	SAE	Score Prob	SAE	Score Prob
313_F	Gear_Oil		Crank_Oil		Gear_Oil	
	#1	9 %	#5W	54 %	#75	54 %
		4 %		33 %		5 %
	#2	2 %	#10W	33 %	#80	5 %
		Under_1%		15 %		Under_1%
	#3	Under_1%	#20W	15 %	#90	Under_1%
		Under_1%		Under_1%		Under_1%
	#4	Under_1%	#20	15 %	#140	Under_1%
		Under_1%		Under_1%		Under_1%
	#5	Under_1%	#30	Under_1%		
		Under_1%		Under_1%		
	#6	Under_1%	#40	Under_1%	MIL_L	
		Under_1%		Under_1%	_23699	8 %
	#7	Under_1%	#50	Under_1%		
		Under_1%		Under_1%		
	#8	Under_1%				
		Under_1%				

The hot scoring probability is under 1% at both ends of the viscosity range.

7.10. Step 10: Cold scoring Calculations

Cold scoring occurs when the asperities of the surface finish penetrate the elastohydrodynamic oil film on the teeth. This causes cutting and micro-welding of the tooth surface. Subsequent sliding and rolling then tears the micro-welds apart and produces radial score lines on the teeth. Cold scoring usually occurs when gears are first put into service at low speed and high load. A "break-in" period at reduced speed and load can increase the resistance to scoring considerably.

It is recommended that the values for surface finish listed in Mobil Oil Corporation's EHL Guidebook, Third Edition, be used for cold scoring since the methods and equations were calibrated for these values.

Composite Roughness, micro-inches

Tooth Finish	Initial	Run-In
Hobbed	70	40
Shaved	50	40
Ground Soft	35	
Ground Hard	20	
Polished (Honed)	7	

Our gears are carburized and ground so we will use 20 micro-inches composite roughness. (Composite roughness is for the pair. If both gears were the same the individual finishes would be about 14 micro-inches.)

UTS Program 60-5408 (TK) will be used to obtain a probability of cold scoring for the gears. This program is based on Mobil Oil Corporation's EHL Guidebook, Third Edition.

For cold scoring the maximum torque condition (Cond #2) is more critical than the maximum speed condition (Cond #5). (If there is any doubt which condition is critical check them all.)

The sump temperature (oil inlet to mesh) is °F.

The oil to be used is Mobil Delvac 1240. (Mobil Oil Corporation states that the lubricant parameter for Delvac 1240 engine oil would be suitable for SAE 15W-40.)

Load (or reset) TK and load COLD. Clear the variables.

We need the Mobil lubricant parameter for Delvac 1240 at 180 °F to calculate the oil film thickness. Input 180 °F for the initial lubricant temperature and solve.

```
180        intemp           F          * Initial Lubricant (Inlet) Temp
```

The table "LubeP" will then give us the lubricant parameter of various Mobil oils at 180 °F.

Mobil Lubricant Parameters

Mobil Lubricant	Parameter	
AUTOMOTIVE_GEAR_OILS		
Lube_SHC	29	EP
Lube_HD_80W	21	EP
Lube_HD_80W_90	36	EP
Lube_FE_80W_140	67	EP
Lube_HD_85W_140	80	EP
Lube_HD_90	46	EP
Lube_HD_140	82	EP
*		
*		
*		
AUTOMATIC_TRANS_FLUID		
ATF_220	9.4	
AUTOMOTIVE_ENGINE_OILS		
Mobil_1	14	
Delvac_1	13	
Delvac_1110	10	
Delvac_1120	14	
Delvac_1130	25	
Delvac_1140	33	
Delvac_1150	48	
Delvac_1210	11	
Delvac_1220	19	
Delvac_1230	30	
Delvac_1240	37	
Delvac_1250	55	
*		
*		
*		
INITIAL_TEMP___degF	180	
INITIAL_TEMP___degC	82.2	

Now that we have the lubricant parameter for Delvac 1240 (37) we can enter our data and solve the model.

```
==================== VARIABLE SHEET ==========================================
St Input---- Name--- Output--- Unit----- Comment-------------------------------
                                         60-5408  CROWNED & STRAIGHT EXTERNAL
                                              GEAR EHL FILM THICKNESS (Rev 1.2)
             m1      'CROWNED            Message Field
             m2      'TOOTH
             m3      'CONTACT
             m4      'OFF_END
             m5      '_
                                         * Denotes Items Normally Input

                                         COMMON DATA:
   11        Q                            * Quality Class (AGMA 390.03)
```

390	n		rpm	* PINION Speed
22500	tork		lb-in	* PINION Torque
	power	139.23	hp	Power
	tan	8666.667	lb	Tangential Load at Operating PD
	vt	530.1	ft/min	Pitch Line Velocity
3.5	pn		1/in	* Normal pitch
	n_mod	7.2571429	mm	Normal Module
25	npa		deg	* Normal Pressure Angle
0	ha		deg	* Helix Angle
	pt	3.5	1/in	Transverse Pitch
	t_mod	7.2571429	mm	Transverse Module
	tpa	25	deg	Transverse Pressure Angle
	bha	0	deg	Base Helix Angle
7.5	cd		in	* Operating Center Distance
	opr_tpa	26.1457	deg	Operating Trans Pressure Angle
	opr_ha	0	deg	Operating Helix Angle
	face	1.625	in	Effective Face Width
.001	et		in	* Total Lead Mismatch Between Teeth
				AGMA 218 - Analytical Method
	Cmf	'NA		Face Load Distribution Factor
	G	2E6	psi	Tooth Stiffness Constant
	dist	.2539	in	Distance from Center of Tooth to Initial Tooth Contact
	mg	1.8888889		Gear Ratio
	mp	1.3903		Profile Contact Ratio (Theoretical)
	mh	0		Helical Contact Ratio (Theoretical)
				FILM THICKNESS & SCORING PROBABILITY:
				Crowned Teeth:
	Ac	.043	in	Actual Width of Contact Ellipse
	Bcf	1.999	in	Length of Full Contact Ellipse
	Bc	1.5581	in	Length of Actual Contact Ellipse
				Straight Teeth:
	Lcmf	'NA	in	Contact Length
	As	'NA	in	Maximum Contact Width
37	lubeP			* Lubricant Parameter
	update	'y		* Set Lube Parameter Table to Initial Temp? 'n=no Def='y=yes
180	intemp		F	* Initial Lubricant (Inlet) Temp
	h	7.91	uin	EHL Film Thickness at Operating PD
20	fin		uin	Composite Surface Finish
	CT	.99		Correction Factor (Inlet Shear)
	lamda	.4		Specific Film Thickness
	prob	7	%	Probability of Cold Scoring (Non-Reactive Lubricants)
				PINION DATA:
18	Np			* Number of Teeth
5.81	odp		in	* Outside Diameter
	crn_p	0	in	* Transverse Crown Rise (Default=0)
1.75	face_p		in	* Face Width
	dist_p	.6211	in	Tooth End to Initial Contact Point
	ell_p	-.3784	in	Tooth End to Crown Contact Ellipse With Crown Centered On Tooth
	pfin		uin	* Surface Finish,rms (After break-in)
	Ep	3E7	psi	* Young's Modulus (Default=steel)

	poi_p	.3		* Poisson's Ratio (Default=.3)
	pdp	5.1923	in	Operating Pitch Diameter
	sap_p	4.8146	in	Start of Active Profile-No Undercut
	dbp	4.661	in	Base Diameter
				GEAR DATA:
34	Ng			* Number of Teeth
10.33	odg		in	* Outside Diameter
.0008	crn_g		in	* Transverse Crown Rise (Default=0)
1.625	face_g		in	* Face Width
	dist_g	.5586	in	Tooth End to Initial Contact Point
	ell_g	-.4409	in	Tooth End to Crown Contact Ellipse With Crown Centered On Tooth
	gfin		uin	* Surface Finish,rms (After break-in)
	Eg	3E7	psi	* Young's Modulus (Default=steel)
	poi_g	.3		* Poisson's Ratio (Default=.3)
	pdg	9.8077	in	Operating Pitch Diameter
	sap_g	9.3477	in	Start of Active Profile-No Undercut
	dbg	8.8041	in	Base Diameter
				ROLL ANGLES:
				Pinion:
	E_ld_p	14.831	deg	Start of Active Profile
	E_lp_p	22.638	deg	Low Single Tooth Contact
	E_pdp	28.126	deg	Pitch Diameter, Operating
	E_cm_p	28.734	deg	Center of Contact Interval
	E_hp_p	34.831	deg	High Single Tooth Contact
	E_od_p	42.638	deg	Outside Diameter, Effective
				Gear:
	E_ld_g	20.443	deg	Start of Active Profile
	E_lp_g	24.576	deg	Low Single Tooth Contact
	E_pdg	28.126	deg	Pitch Diameter, Operating
	E_cm_g	27.803	deg	Center of Contact Interval
	E_hp_g	31.031	deg	High Single Tooth Contact
	E_od_g	35.164	deg	Outside Diameter, Effective

References:

Data Extracted from AGMA 218.01, AGMA Standard for Rating the Pitting Resistance and Bending Strength of Spur and Helical Involute Gear Teeth and AGMA 217.01, AGMA Information Sheet - Gear Scoring Design Guide for Aerospace Spur and Helical Power Gears with the permission of the publisher, the American Gear Manufacturers Association, 1500 King Street, Suite 201, Alexandria, Virginia 22314

Mobil EHL Guidebook, Third Edition Mobil Oil Corporation, Commercial Marketing, Technical Publications, 3225 Gallows Road, Fairfax, Virginia 22037

The cold scoring probability is 7%.

7.11. Step 11: Find Maximum Effective Tooth Thickness

The tooth thickness given on drawings is usually not defined as actual thickness or effective thickness but if a measuring method is specified it is usually actual (measured) thickness. The backlash between gears is determined by the maximum material condition of the teeth. The effective tooth thickness of a tooth is larger than the measured tooth thickness except when measured with a parallel axis master gear which contacts from the specified start of active profile to the effective tooth tip. When the tooth thickness is measured by any other means, such as over two pins, the effective tooth thickness is not measured, and allowance must be made for errors in those elements of the gear which are not measured. For example, the measurement over two pins does not account for lead error, pitch error, profile error and runout. Errors in these elements reduce the backlash between the teeth. We will make calculations using the size over pins on the gear drawings to determine the maximum effective tooth thickness.

Load (or reset) TK and load PINEFF. Clear the variables.

The measuring method specified is over two pins. Enter the maximum specified actual tooth thickness for the pinion, 0.4881", under "MEASUREMENT OVER 2 OR 3 PINS" along with the other data required and solve.

```
=================== VARIABLE SHEET =========================================
St Input---- Name--- Output--- Unit----- Comment-------------------------------
                                         Pinion Max Eff Tooth Thickness
                                         60-EFF MEASURING EFFECTIVE AND ACTUAL
                                                TOOTH THICKNESS
             m1      'PARALLEL           Message
             m2      'AXIS
             m3      'GEAR
             m4      '_

             axis    'p                  Crossed or parallel axis? 'c or 'p=Def
   18        n                           Number of teeth
   3.5       pn                1/in      Normal pitch
   25        npa               deg       Normal pressure angle
   0         ha                deg       Helix angle
             hand                        Hand:'L,'R,'Spur (Crossed-axis master)
             bha     0.0000    deg       Base helix angle
   1.75      face              in        Eff face width
             pt      3.5       1/in      Transverse pitch
             tpa     25        deg       Transverse pressure angle
             pd      5.1429    in        Reference pitch diameter
             db      4.66101   in        Base diameter
             n_mod   7.2571429 mm        Normal module
             t_mod   7.2571429 mm        Transverse module
             ntt     .4895     in        Eff normal tooth thickness (Ref PD)
             ttt     .4895     in        Eff trans tooth thickness (Ref PD)
```

	Q			AGMA Quality Number
	m			Message-Quality Number
				Data may override AGMA Quality Number
.0013	Runout		in	Runout variation (TIR)
.0005	Pitch		in	Pitch (spacing) variation
.0005	Profile		in	Profile variation
.0004	Lead		in	Lead variation
				ACTUAL TOOTH MEASUREMENT:
				PARALLEL AXIS MASTER GEAR
	n_pm			MASTER, number of teeth
	pd_pm		in	Reference Pitch Diameter
	ntt_pm		in	Normal Tooth Thickness (Ref PD)
	ttt_pm		in	Trans Tooth Thickness (Ref PD)
	cd_pm		in	Master gear test center distance
	nmeffp	.4895	in	Measured eff normal tooth thickness
	tmeffp	.4895	in	Measured eff trans tooth thickness
	dtp	0.0000	in	(Eff-Meas Eff) trans tooth thickness
				SPAN MEASUREMENT
	nacts	.4882	in	Actual normal tooth thickness
	tacts	.4882	in	Actual trans tooth thickness
	dts	.0013	in	(Eff-Actual) trans tooth thickness
				MEASUREMENT OVER 2 OR 3 PINS
.4881	nact2		in	Actual normal tooth thickness
	tact2	.4881	in	Actual trans tooth thickness
	dt2	.0014	in	(Eff-Actual) trans tooth thickness
				TOOTH CALIPER MEASUREMENT
	nactc	.4881	in	Actual normal tooth thickness
	tactc	.4881	in	Actual trans tooth thickness
	dtc	.0014	in	(Eff-Actual) trans tooth thickness
				CROSSED AXIS MASTER GEAR MEASUREMENT
	n_xm			MASTER, number of teeth
	pt_xm			Transverse pitch
	tpa_xm		deg	Transverse pressure angle
	ha_xm		deg	Helix angle
	hand_xm			Hand:'L,'R,'Spur
	pd_xm		in	Reference pitch diameter
	bha_xm		deg	Base helix angle
	db_xm		in	Base diameter
	ntt_xm		in	Normal tooth thickness (Ref PD)
	ttt_xm		in	Trans tooth thickness (Ref PD)
	cd_xm		in	Master gear test center distance
	xaxis		deg	Master gear test cross-axis angle
	nactx	.4892	in	Actual normal tooth thickness
	tactx	.4892	in	Actual trans tooth thickness
	dtx	.0003	in	(Eff-Actual) trans tooth thickness

Ref:
Data Extracted from AGMA Handbook For Unassembled Gears - Volume 1 - Gear Classification, Materials, and Inspection (AGMA 390.03), with the permission of the publisher, the American Gear Manufacturers Association,1500 King Street, Suite 201, Alexandria, Virginia 22314

The maximum effective tooth thickness for the pinion is 0.0014" larger than the actual measured tooth thickness or 0.4895". Root mean square addition is used in the model which should cover more than 95% of cases.

Do the same thing for the gear. Load (or reset) TK and load GREFF. Enter the data and solve.

```
==================== VARIABLE SHEET ========================================
St Input---- Name--- Output--- Unit----- Comment---------------------------
                                          Gear Max Eff Tooth Thickness
                                          60-EFF MEASURING EFFECTIVE AND ACTUAL
                                                  TOOTH THICKNESS
             m1      'PARALLEL            Message
             m2      'AXIS
             m3      'GEAR
             m4      '_

             axis    'p                   Crossed or parallel axis? 'c or 'p=Def
   34        n                            Number of teeth
   3.5       pn                1/in       Normal pitch
   25        npa               deg        Normal pressure angle
   0         ha                deg        Helix angle
             hand                         Hand:'L,'R,'Spur (Crossed-axis master)
             bha     0.0000    deg        Base helix angle
   1.625     face              in         Eff face width
             pt      3.5       1/in       Transverse pitch
             tpa     25        deg        Transverse pressure angle
             pd      9.7143    in         Reference pitch diameter
             db      8.80413   in         Base diameter
             n_mod   7.2571429 mm         Normal module
             t_mod   7.2571429 mm         Transverse module
             ntt     .4684     in         Eff normal tooth thickness (Ref PD)
             ttt     .4684     in         Eff trans tooth thickness (Ref PD)

             Q                             AGMA Quality Number
             m                             Message-Quality Number
                                           Data may override AGMA Quality Number
   .003      Runout            in           Runout variation (TIR)
   .00051    Pitch             in           Pitch (spacing) variation
   .00064    Profile           in           Profile variation
   .0004     Lead              in           Lead variation
```

ACTUAL TOOTH MEASUREMENT:

				PARALLEL AXIS MASTER GEAR
	n_pm			MASTER, number of teeth
	pd_pm		in	Reference Pitch Diameter
	ntt_pm		in	Normal Tooth Thickness (Ref PD)
	ttt_pm		in	Trans Tooth Thickness (Ref PD)
	cd_pm		in	Master gear test center distance
	nmeffp	.4684	in	Measured eff normal tooth thickness
	tmeffp	.4684	in	Measured eff trans tooth thickness
	dtp	0.0000	in	(Eff-Meas Eff) trans tooth thickness
				SPAN MEASUREMENT
	nacts	.4655	in	Actual normal tooth thickness
	tacts	.4655	in	Actual trans tooth thickness
	dts	.0028	in	(Eff-Actual) trans tooth thickness
				MEASUREMENT OVER 2 OR 3 PINS
.4655	nact2		in	Actual normal tooth thickness
	tact2	.4655	in	Actual trans tooth thickness
	dt2	.0029	in	(Eff-Actual) trans tooth thickness

The maximum effective tooth thickness for the gear is 0.0029" larger than the actual measured tooth thickness or 0.4684".

Maximum effective tooth thickness

18 tooth pinion: 0.4895"
34 tooth gear: 0.4684"

7.12. Step 12: Find Cold Zero Backlash Temperature

Military specifications usually require that equipment operate after being subjected to a minimum temperature. We will check to find the temperature at which the backlash between the the gear teeth becomes zero when the gears and the housing are at the SAME temperature.

The drawings of the housing indicate that the shaft bores are to be within 0.003" of true location with respect to each other. The center distance limits are then as follows:

Center Distance = 7.503"/7.497"

Now we need to find the temperature at which the assembled backlash becomes zero when the gears and the housing are at the SAME temperature. The assumed inspection temperature is 68 °F.

To calculate the zero backlash temperature we will use "ZEROBL". Load (or reset) TK and load ZEROBL. Clear the variables. Iteration is required for the solution, so we will enter our data and make a guess of 0 °F for the zero backlash temperature. (The diameter to center of mesh is the diameter through the center of the working depth. It is equal to the outside diameter minus the working depth.)

```
==================== VARIABLE SHEET ==========================================
St Input---- Name--- Output--- Unit----- Comment------------------------------
                                         Zero BL Temp
                                          Min Case CD & Max Eff Tooth Thickness
                                         60-1101 OPERATING BACKLASH: EXTERNAL
             mess1                       CAUTION MESSAGE
             mess2
             mess3
             mess4

   18        np                          PINION, number of teeth
   34        ng                          GEAR, number of teeth
             mg                          Gear ratio (Gear/Pinion)

                                         NORMAL PLANE:
   3.5       pn                1/in       Diametral Pitch
   25        npa               deg        Pressure Angle
             n_mod             mm         Module
             pnb               in         Base Pitch

                                         TRANSVERSE PLANE:
             pt                1/in       Diametral Pitch
             tpa               deg        Pressure Angle
             t_mod             mm         Module
             ptb               in         Base Pitch

                                         COMMON:
   0         ha                deg        Helix Angle
             bha               deg        Base Helix Angle
             ap                in         Axial Pitch
             cd                in         Operating center distance
             std_cd            in         "Standard" center distance

                                         TOOTH THICKNESS:  (at Ref PD)
                                          Pinion:
   .4895     nttp              in          Normal Tooth Thickness
             tttp              in          Transverse Tooth Thickness
                                          Gear:
   .4684     nttg              in          Normal Tooth Thickness
             tttg              in          Transverse Tooth Thickness

                                         DIAMETERS:
                                          Pinion:
             dref_p            in          Reference Pitch Diameter
             pt_p              in          Pointed Tooth Diameter
             dbp               in          Base Diameter
```

			Gear:
	dref_g	in	Reference Pitch Diameter
	pt_g	in	Pointed Tooth Diameter
	dbg	in	Base Diameter
			OPERATING DATA:
	delta	in	Change in Opr CD from "Std" CD
0	nbl	in	Normal Backlash at Operating PD
	tbl	in	Transverse Backlash at Operating PD
	opr_tpa	deg	Transverse Pressure Angle
	opr_ha	deg	Helix Angle
	cirpitc	in	Circular Pitch
			Pinion:
	d_p	in	Pitch Diameter
	ttt_p	in	Transverse Tooth Thickness
			Gear:
	d_g	in	Pitch Diameter
	ttt_g	in	Transverse Tooth Thickness
			TOOTH THICKNESS / DIAMETER CHECK
			Pinion Diameter: (Input)
	cdp	in	Normal Tooth Thickness
	cnttp	in	Transverse Tooth Thickness
	ctttp	in	Gear Diameter: (Input)
	cdg	in	Normal Tooth Thickness
	cnttg	in	Transverse Tooth Thickness
	ctttg	in	
	gcd		Iteration trigger variable for solving CD from tooth thickness and backlash
			60-XXX EFFECTIVE CENTER DISTANCE DUE TO OPERATING TEMPERATURES
			ASSEMBLY CONDITIONS:
	Ta	F	Temperature (Default=68F=20C)
			Housing:
'Alum	Hmatl		Matl:('CIron,'Steel,'SS,'Alum,'Mag)
	Kh	/degF	Coefficient of expansion
7.5030	Cmax_a	in	Maximum center distance
7.4970	Cmin_a	in	Minimum center distance
			Gear #1
'CarbStl	G1matl		Matl:('CarbStl,'NicStl,'CIron, 'SS,'Brass)
	K1	/degF	Coefficient of expansion
5.2400	d1	in	Diameter to center of mesh
			Gear #2
'CarbStl	G2matl		Matl:('CarbStl,'NicStl,'CIron, 'SS,'Brass)
	K2	/degF	Coefficient of expansion
	d2	in	Diameter to center of mesh
	ms		

```
                                                OPERATING CONDITIONS:
G 0          Th                F                 Housing temperature
             Tg                F                 Gear temperature
             dC                in                Change in center distance
                                                 Effective center distance:
             Cmax_op           in                 Maximum
             Cmin_op           in                 Minimum
```

Press F9 to solve and the iterative solver will find the zero backlash temperature.

```
==================== VARIABLE SHEET =========================================
St Input---- Name--- Output--- Unit----- Comment-------------------------------
                                        Zero BL Temp
                                         Min Case CD & Max Eff Tooth Thickness
                                        60-1101 OPERATING BACKLASH: EXTERNAL
             mess1   'None              CAUTION MESSAGE
             mess2   '_
             mess3   '_
             mess4   '_

   18        np                         PINION, number of teeth
   34        ng                         GEAR, number of teeth
             mg      1.8888889          Gear ratio (Gear/Pinion)

                                        NORMAL PLANE:
   3.5       pn                1/in      Diametral Pitch
   25        npa               deg       Pressure Angle
             n_mod   7.2571429 mm        Module
             pnb     .8135     in        Base Pitch

                                        TRANSVERSE PLANE:
             pt      3.5       1/in      Diametral Pitch
             tpa     25        deg       Pressure Angle
             t_mod   7.2571429 mm        Module
             ptb     .8135     in        Base Pitch

                                        COMMON:
   0         ha                deg       Helix Angle
             bha     0         deg       Base Helix Angle
             ap                in        Axial Pitch
             cd      7.492     in        Operating center distance
             std_cd  7.4285714 in        "Standard" center distance

                                        TOOTH THICKNESS:  (at Ref PD)
                                         Pinion:
   .4895     nttp              in         Normal Tooth Thickness
             tttp    .4895     in         Transverse Tooth Thickness
                                         Gear:
   .4684     nttg              in         Normal Tooth Thickness
             tttg    .4684     in         Transverse Tooth Thickness
```

				DIAMETERS:
				Pinion:
	dref_p	5.1429	in	Reference Pitch Diameter
	pt_p	5.9682	in	Pointed Tooth Diameter
	dbp	4.661	in	Base Diameter
				Gear:
	dref_g	9.7143	in	Reference Pitch Diameter
	pt_g	10.5738	in	Pointed Tooth Diameter
	dbg	8.8041	in	Base Diameter
				OPERATING DATA:
	delta	.0634	in	Change in Opr CD from "Std" CD
0	nbl		in	Normal Backlash at Operating PD
	tbl	0	in	Transverse Backlash at Operating PD
	opr_tpa	26.021	deg	Transverse Pressure Angle
	opr_ha	0	deg	Helix Angle
	cirpitc	.9053	in	Circular Pitch
				Pinion:
	d_p	5.1868	in	Pitch Diameter
	ttt_p	.4726	in	Transverse Tooth Thickness
				Gear:
	d_g	9.7972	in	Pitch Diameter
	ttt_g	.4326	in	Transverse Tooth Thickness
				TOOTH THICKNESS / DIAMETER CHECK
	cdp		in	Pinion Diameter: (Input)
	cnttp		in	Normal Tooth Thickness
	ctttp		in	Transverse Tooth Thickness
	cdg		in	Gear Diameter: (Input)
	cnttg		in	Normal Tooth Thickness
	ctttg		in	Transverse Tooth Thickness
	gcd	1.0085399		Iteration trigger variable for solving CD from tooth thickness and backlash
				60-XXX EFFECTIVE CENTER DISTANCE DUE TO OPERATING TEMPERATURES
				ASSEMBLY CONDITIONS:
	Ta	+68	F	Temperature (Default=68F=20C)
				Housing:
'Alum	Hmatl			Matl:('CIron,'Steel,'SS,'Alum,'Mag)
	Kh	1.34E-5	/degF	Coefficient of expansion
7.5030	Cmax_a		in	Maximum center distance
7.4970	Cmin_a		in	Minimum center distance
				Gear #1
'CarbStl	G1matl			Matl:('CarbStl,'NicStl,'CIron, 'SS,'Brass)
	K1	6.40E-6	/degF	Coefficient of expansion
5.2400	d1		in	Diameter to center of mesh

```
                                               Gear #2
'CarbStl   G2matl                               Matl:('CarbStl,'NicStl,'CIron,
                                                                   'SS,'Brass)
           K2       6.40E-6   /degF             Coefficient of expansion
           d2       9.7600    in                Diameter to center of mesh
           ms       '_
                                              OPERATING CONDITIONS:
           Th       -27       F                Housing temperature
           Tg       -27       F                Gear temperature
           dC       -.0050    in               Change in center distance
                                               Effective center distance:
           Cmax_op  7.4980    in                Maximum
           Cmin_op  7.4920    in                Minimum
```

At minimum machined center distance and maximum effective tooth thickness the backlash would become zero at -27 °F

If this temperature does not meet the specifications the tooth thickness of the gears and/or the housing center distance must be changed.

7.13. Step 13: Find Maximum Hot Backlash

We should also check the maximum backlash at hot conditions to be sure that the maximum backlash is reasonable. For this check we will use the minimum actual rather than the minimum effective tooth thickness to find an absolute limit on "hot" backlash for 100% of all drives. We will assume that the gears and the housing are the same temperature at 180 °F, although the calculation can easily be done for different temperatures if desired. (The gears are usually hotter than the housing which would reduce the backlash somewhat.)

Load (or reset) TK and load MAXBL. Clear the variables, enter our data and solve.

```
==================== VARIABLE SHEET =========================================
St Input---- Name--- Output--- Unit----- Comment--------------------------------
                                         Max Possible BL at 180 oF
                                          Max Case CD & Min Act Tooth Thickness
                                         60-1101 OPERATING BACKLASH: EXTERNAL
             mess1   'None               CAUTION MESSAGE
             mess2   '_
             mess3   '_
             mess4   '_

   18        np                          PINION, number of teeth
   34        ng                          GEAR, number of teeth
             mg      1.8888889           Gear ratio (Gear/Pinion)
```

				NORMAL PLANE:
3.5	pn		1/in	Diametral Pitch
25	npa		deg	Pressure Angle
	n_mod	7.2571429	mm	Module
	pnb	.8135	in	Base Pitch
				TRANSVERSE PLANE:
	pt	3.5	1/in	Diametral Pitch
	tpa	25	deg	Pressure Angle
	t_mod	7.2571429	mm	Module
	ptb	.8135	in	Base Pitch
				COMMON:
0	ha		deg	Helix Angle
	bha	0	deg	Base Helix Angle
	ap		in	Axial Pitch
	cd	7.5062	in	Operating center distance
	std_cd	7.4285714	in	"Standard" center distance
				TOOTH THICKNESS: (at Ref PD)
				Pinion:
.4866	nttp		in	Normal Tooth Thickness
	tttp	.4866	in	Transverse Tooth Thickness
				Gear:
.464	nttg		in	Normal Tooth Thickness
	tttg	.464	in	Transverse Tooth Thickness
				DIAMETERS:
				Pinion:
	dref_p	5.1429	in	Reference Pitch Diameter
	pt_p	5.964	in	Pointed Tooth Diameter
	dbp	4.661	in	Base Diameter
				Gear:
	dref_g	9.7143	in	Reference Pitch Diameter
	pt_g	10.5666	in	Pointed Tooth Diameter
	dbg	8.8041	in	Base Diameter
				OPERATING DATA:
	delta	.0776	in	Change in Opr CD from "Std" CD
	nbl	.0213	in	Normal Backlash at Operating PD
	tbl	.0213	in	Transverse Backlash at Operating PD
	opr_tpa	26.2416	deg	Transverse Pressure Angle
	opr_ha	0	deg	Helix Angle
	cirpitc	.907	in	Circular Pitch
				Pinion:
	d_p	5.1966	in	Pitch Diameter
	ttt_p	.4658	in	Transverse Tooth Thickness
				Gear:
	d_g	9.8158	in	Pitch Diameter
	ttt_g	.4199	in	Transverse Tooth Thickness

				TOOTH THICKNESS / DIAMETER CHECK
	cdp		in	Pinion Diameter: (Input)
	cnttp		in	Normal Tooth Thickness
	ctttp		in	Transverse Tooth Thickness
	cdg		in	Gear Diameter: (Input)
	cnttg		in	Normal Tooth Thickness
	ctttg		in	Transverse Tooth Thickness
	gcd	1.0104472		Iteration trigger variable for solving CD from tooth thickness and backlash
				60-XXX EFFECTIVE CENTER DISTANCE DUE TO OPERATING TEMPERATURES
				ASSEMBLY CONDITIONS:
	Ta	+68	F	Temperature (Default=68F=20C)
				Housing:
'Alum	Hmatl			Matl:('CIron,'Steel,'SS,'Alum,'Mag)
	Kh	1.34E-5	/degF	Coefficient of expansion
7.5003	Cmax_a		in	Maximum center distance
7.4970	Cmin_a		in	Minimum center distance
				Gear #1
'CarbStl	G1matl			Matl:('CarbStl,'NicStl,'CIron, 'SS,'Brass)
	K1	6.40E-6	/degF	Coefficient of expansion
5.2400	d1		in	Diameter to center of mesh
				Gear #2
'CarbStl	G2matl			Matl:('CarbStl,'NicStl,'CIron, 'SS,'Brass)
	K2	6.40E-6	/degF	Coefficient of expansion
	d2	9.7573	in	Diameter to center of mesh
	ms	'_		
				OPERATING CONDITIONS:
+180	Th		F	Housing temperature
+180	Tg		F	Gear temperature
	dC	.0059	in	Change in center distance
				Effective center distance:
	Cmax_op	7.5062	in	Maximum
	Cmin_op	7.5029	in	Minimum

At maximum machined center distance and minimum actual tooth thickness the backlash would be 0.0213" at 180 °F.

7.14. Computer Models

All computer data generated and TK Solver Plus models used are furnished on a floppy disc labeled "Expert System Design - Military Final Drives". Appendix R contains an index of files on this disk.

LIST OF REFERENCES

[1]"ANSI/AGMA 2000-A88 Gear Classification and Inspection Handbook," American Gear Manufacturers Association, 1500 King Street, Suite 201, Alexandria, VA 22314

[2]AGMA 218.01, "AGMA Standard for Rating the Pitting Resistance and Bending Strength of Spur and Helical Involute Gear Teeth," American Gear Manufacturers Association, 1500 King Street, Suite 201, Alexandria, VA 22314

[3]AGMA 217.01, "AGMA Information Sheet—Gear Scoring Design Guide for Aerospace Spur and Helical Power Gears," American Gear Manufacturers Association, 1500 King Street, Suite 201, Alexandria, VA 22314

[4]"Mobil EHL Guidebook, Third Edition," Mobil Oil Corporation, Commercial Marketing, Technical Publications, 3225 Gallows Road, Fairfax, VA 22037

Appendix A

H.S. Train–Unground Nominal

TACOM
Gear Analysis

* Denotes Input Data

* Normal Diam Pitch=	3.5000	Opr Diam Pitch=	3.4327
* Normal Pressure Angle=	25.0000	Opr Pressure Angle=	27.2680
* Helix Angle=	0.0000		
Trans Diam Pitch=	3.5000	Line of Action=	1.0482
Trans Pressure Angle=	25.0000	% Approach Action=	50.97
		% Recess Action=	49.03
Opr Center Distance=	7.4286	Profile C.R.=	1.2885
* Face Width=	1.5820		
Basic Backlash=	0.0059		
Total Operating BL=	0.0138		

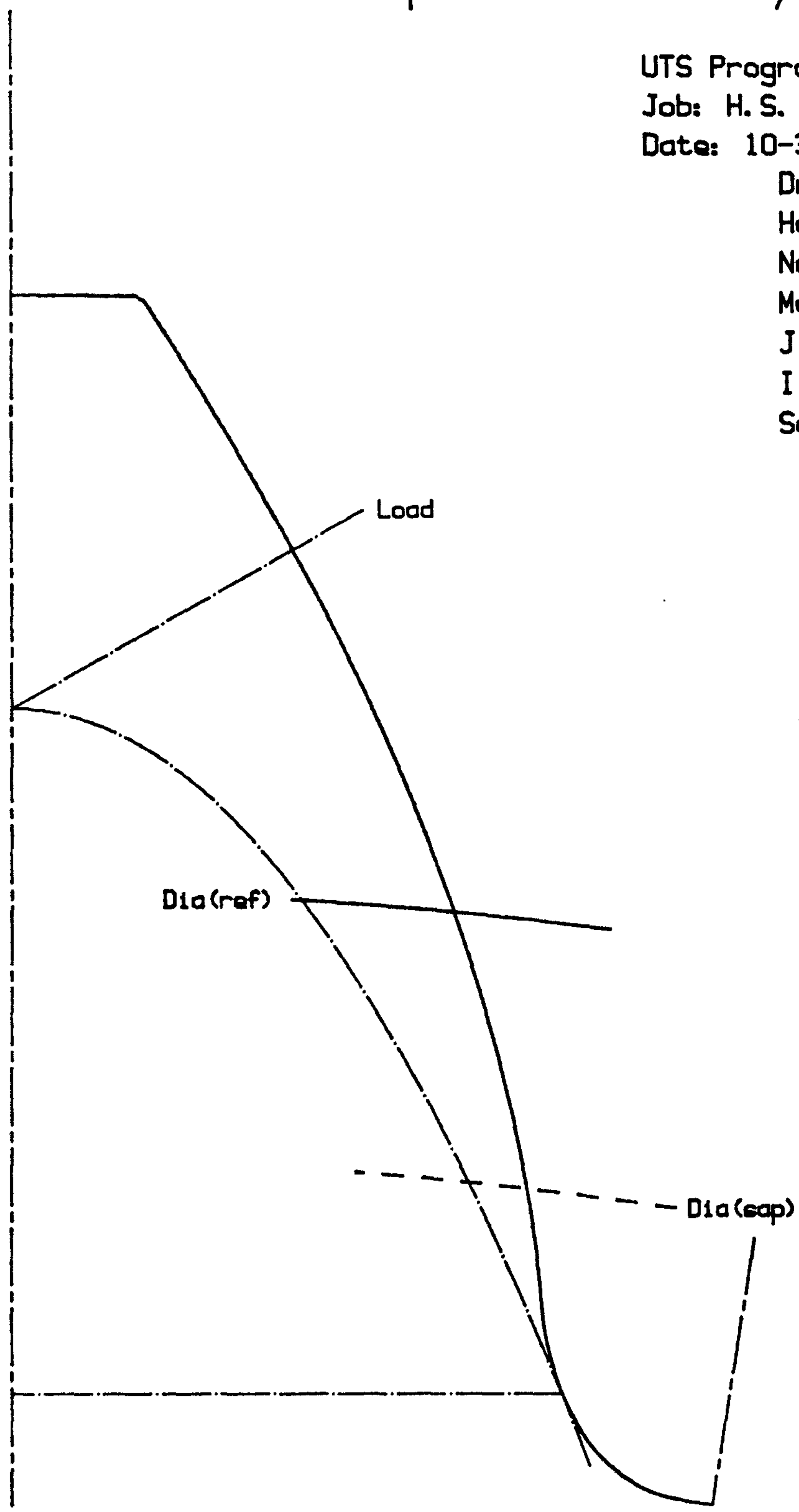
	DRIVER	(Deg Roll)	DRIVEN	(Deg Roll)
* Number of Teeth=	19		32	
* Outside Diameter=	6.0800	(41.60)	9.8625	(36.98)
Dia at Start of Tip Modification=	6.0750	(41.50)	9.8576	(36.92)
Circular Tip Relief at OD=	0.0028		0.0028	
* Total Normal Finish Stock=	0.0000		0.0000	
HOB FORM DATA	SEMI-TOPPING		SEMI-TOPPING	
* Hob Pressure Angle=	25.0000		25.0000	
* Hob Tip to Ref Line=	0.3615		0.3615	
* Hob Tooth Thickness at Ref=	0.4488		0.4488	
* Ref Line to Hob Mod Ramp=	0.2484		0.2631	
* Pressure Angle of Mod Ramp=	58.0000		58.0000	
* Hob Tip Radius=	0.0550		0.0550	
* Hob Protuberance=	0.0000		0.0000	
Hob SAP from Ref Line=	0.2636		0.2758	
Hob Space Width at Hob SAP=	0.1684		0.1628	
Normal Tooth Thickness at OD=	0.1411		0.1408	
Normal Tooth Thickness at Eff OD=	0.1502		0.1494	
*Normal Tooth Thickness, (Hobbed)=	0.5006		0.5225	
Dia @ Mid-point of Line of Action=	5.5257	(29.29)	9.3315	(29.67)
Pitch Diameter, (Ref)=	5.4286	(26.72)	9.1429	(26.72)
Operating Pitch Diameter=	5.5350	(29.53)	9.3222	(29.53)
Base Diameter=	4.9200		8.2862	
Dia, (Start of Active Profile)=	5.1341	(17.09)	8.8983	(22.43)
Form Diameter=	5.1341	(17.09)	8.8983	(22.42)
Root Diameter=	4.8167		8.5779	
Root Clearance=	0.0890		0.0996	
Max Undercut=	0.0000		0.0000	
Dia at Involute-Fillet Tangent=	5.0199	(11.60)	8.7081	(18.51)
Roll, radians, (1 tooth load)=	0.629	(36.04)	0.588	(33.68)
Minimum Fillet Radius=	0.0762		0.0658	
Helical Factor, C(h)=	1.000		1.000	
Y Factor=	0.734		0.817	
Load Sharing Ratio, m(N)=	1.000		1.000	
MODIFIED Stress Corr Fact, K(f)=	1.614		1.660	
J-Factor=	0.455		0.492	
I Factor=		0.111		
Max Specific Sliding Ratio=	1.16	(17.09)	0.85	(22.43)

Steel Gears

Universal Technical Systems, Inc, Rockford, Ill 61101 (Program #500)
Date: 10-31-88
Job : H.S. Train-Unground Nominal

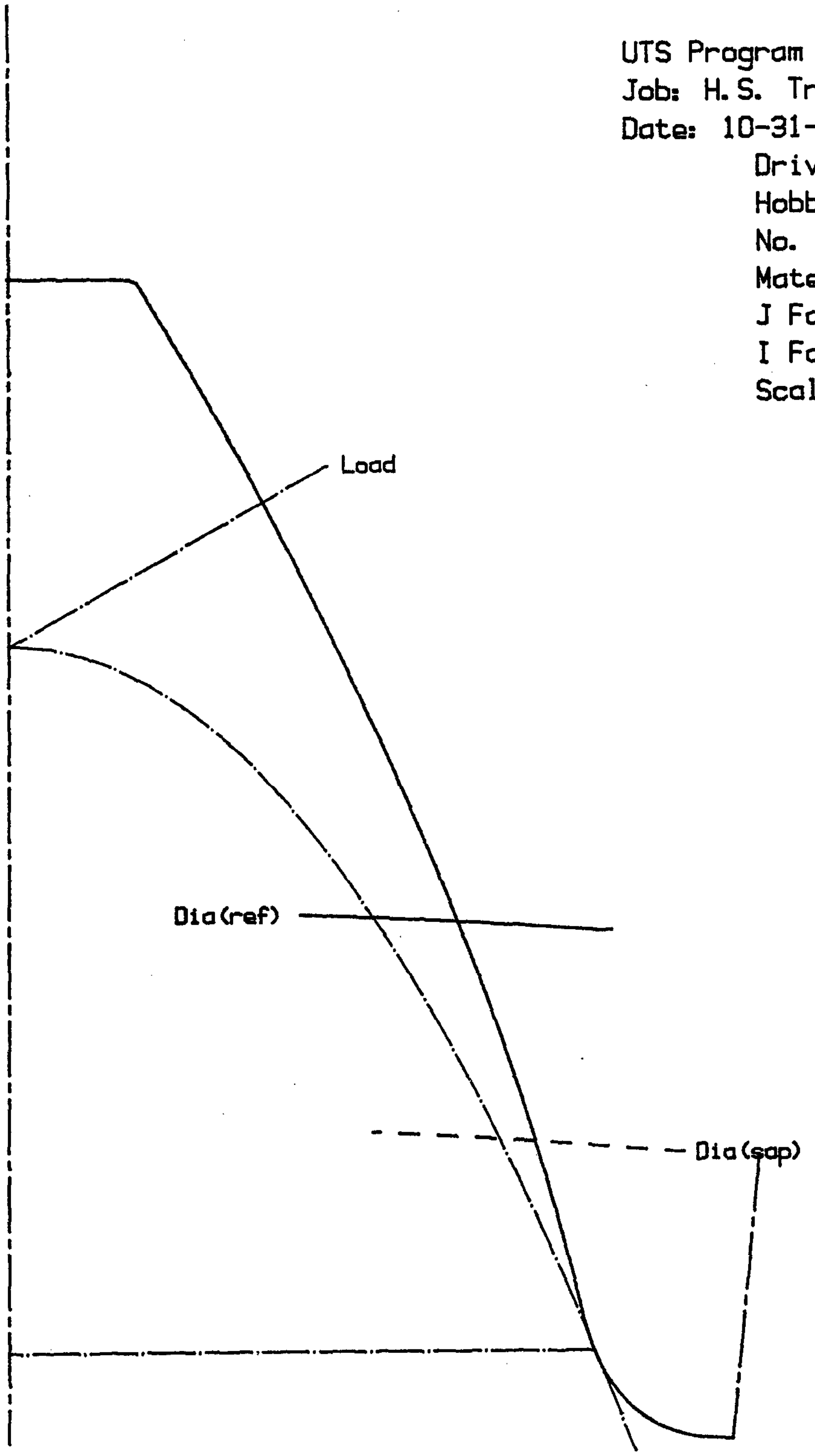
TACOM
Spur Gear Analysis

UTS Program #500
Job: H.S. Train-Unground Nominal
Date: 10-31-88
Driver
Hobbed Only
No. Teeth= 19
Mate= 32
J Factor= 0.455
I Factor= 0.111
Scale= 11

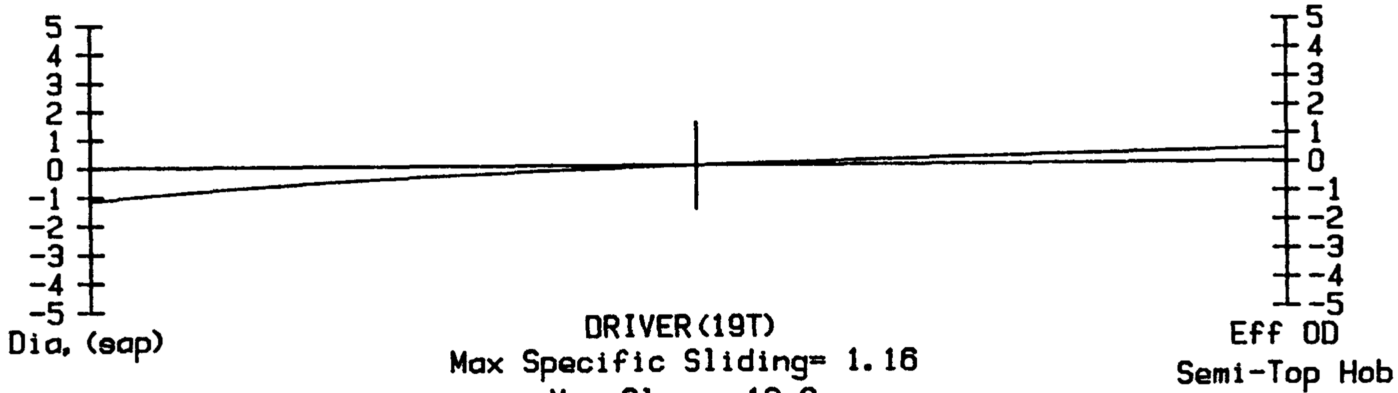


TACOM
Spur Gear Analysis

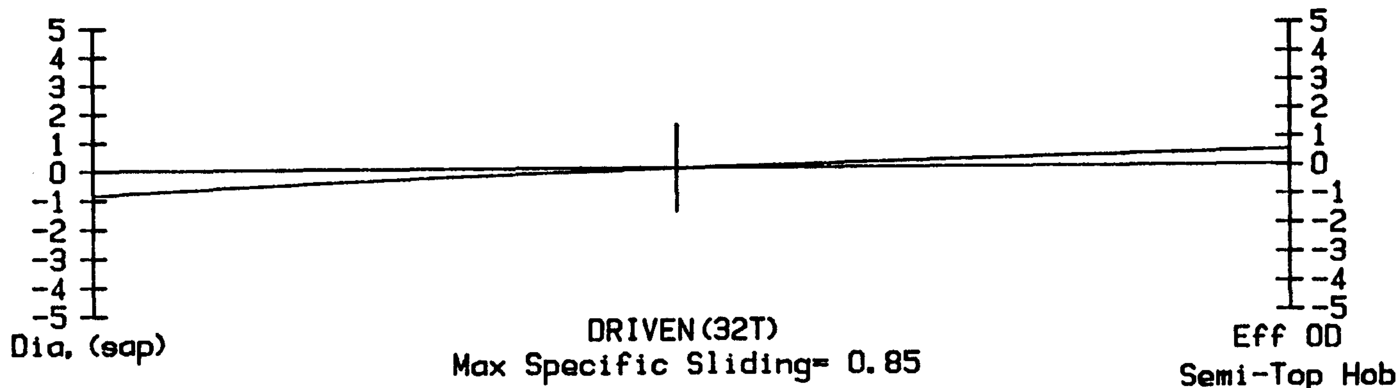
UTS Program #500
Job: H.S. Train-Unground Nominal
Date: 10-31-88
Driven
Hobbed Only
No. Teeth= 32
Mate= 19
J Factor= 0.492
I Factor= 0.111
Scale= 11



SPECIFIC SLIDING RATIOS (Slide/Roll)



DRIVER(19T)
Max Specific Sliding= 1.16
Max Slope= 12.9



DRIVEN(32T)
Max Specific Sliding= 0.85
Max Slope= 5.9

UTS #500 Job: H.S. Train-Unground Nominal Date: 10-31-88

TACOM

Appendix B

H.S. Train–Ground Nominal

TACOM
Gear Analysis

* Denotes Input Data

* Normal Diam Pitch=	3.5000	Opr Diam Pitch=	3.4327
* Normal Pressure Angle=	25.0000	Opr Pressure Angle=	27.2680
* Helix Angle=	0.0000		
Trans Diam Pitch=	3.5000	Line of Action=	1.0472
Trans Pressure Angle=	25.0000	% Approach Action=	50.94
		% Recess Action=	49.06
Opr Center Distance=	7.4286	Profile C.R.=	1.2873
* Face Width=	1.5820		
Basic Backlash=	0.0059		
Total Operating BL=	0.0138		

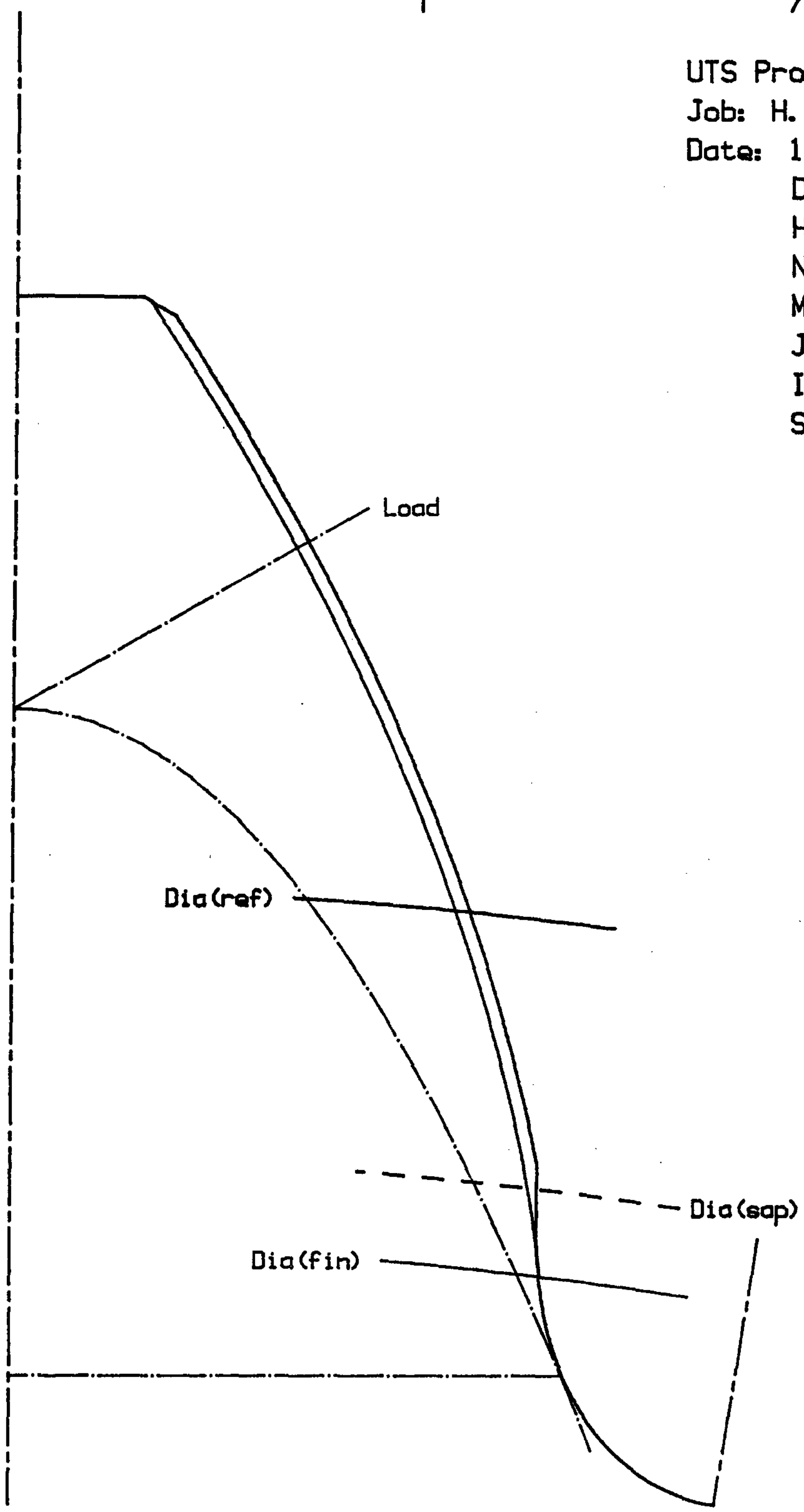
	DRIVER	(Deg Roll)	DRIVEN	(Deg Roll)
* Number of Teeth=	19		32	
* Outside Diameter=	6.0800	(41.60)	9.8625	(36.98)
Dia at Start of Tip Modification=	6.0748	(41.50)	9.8568	(36.91)
Circular Tip Relief at OD=	0.0030		0.0032	
* Total Normal Finish Stock=	0.0150		0.0150	
HOB FORM DATA	SEMI-TOPPING		SEMI-TOPPING	
* Hob Pressure Angle=	25.0000		25.0000	
* Hob Tip to Ref Line=	0.3615		0.3615	
* Hob Tooth Thickness at Ref=	0.4338		0.4338	
* Ref Line to Hob Mod Ramp=	0.2417		0.2561	
* Pressure Angle of Mod Ramp=	58.0000		58.0000	
* Both: Full Rad-Hob Tip Radius=	0.0896		0.0896	
* Hob Protuberance=	0.0080		0.0080	
Hob SAP from Ref Line=	0.2636		0.2758	
Hob Space Width at Hob SAP=	0.1682		0.1619	
Normal Tooth Thickness at OD=	0.1408		0.1399	
Normal Tooth Thickness at Eff OD=	0.1504		0.1499	
Normal Tooth Thickness, (Hobbed)=	0.5156		0.5375	
*Normal Tooth Thickness, (Ground)=	0.5006		0.5225	
Dia @ Mid-point of Line of Action=	5.5260	(29.30)	9.3313	(29.67)
Pitch Diameter, (Ref)=	5.4286	(26.72)	9.1429	(26.72)
Operating Pitch Diameter=	5.5350	(29.53)	9.3222	(29.53)
Base Diameter=	4.9200		8.2862	
Dia, (Start of Active Profile)=	5.1346	(17.11)	8.8985	(22.43)
Form Diameter=	5.1346	(17.11)	8.8985	(22.43)
Root Diameter=	4.8167		8.5779	
Root Clearance=	0.0890		0.0996	
Max Undercut=	0.0082		0.0084	
Diameter at Max Undercut=	5.0395	(12.71)	8.7376	(19.17)
* Finished Grind Diameter=	5.0395	(12.71)	8.7376	(19.17)
Roll, radians, (1 tooth load)=	0.629	(36.05)	0.588	(33.68)
Minimum Fillet Radius=	0.1056		0.0974	
Helical Factor, C(h)=	1.000		1.000	
Y Factor=	0.753		0.828	
Load Sharing Ratio, m(N)=	1.000		1.000	
MODIFIED Stress Corr Fact, K(f)=	1.619		1.660	
J-Factor=	0.465		0.499	
I Factor=		0.111		

Steel Gears, Finish Ground
Case Carburized

Universal Technical Systems, Inc, Rockford, Ill 61101 (Program #500)
Date: 10-31-88
Job : H.S. Train-Ground Nominal

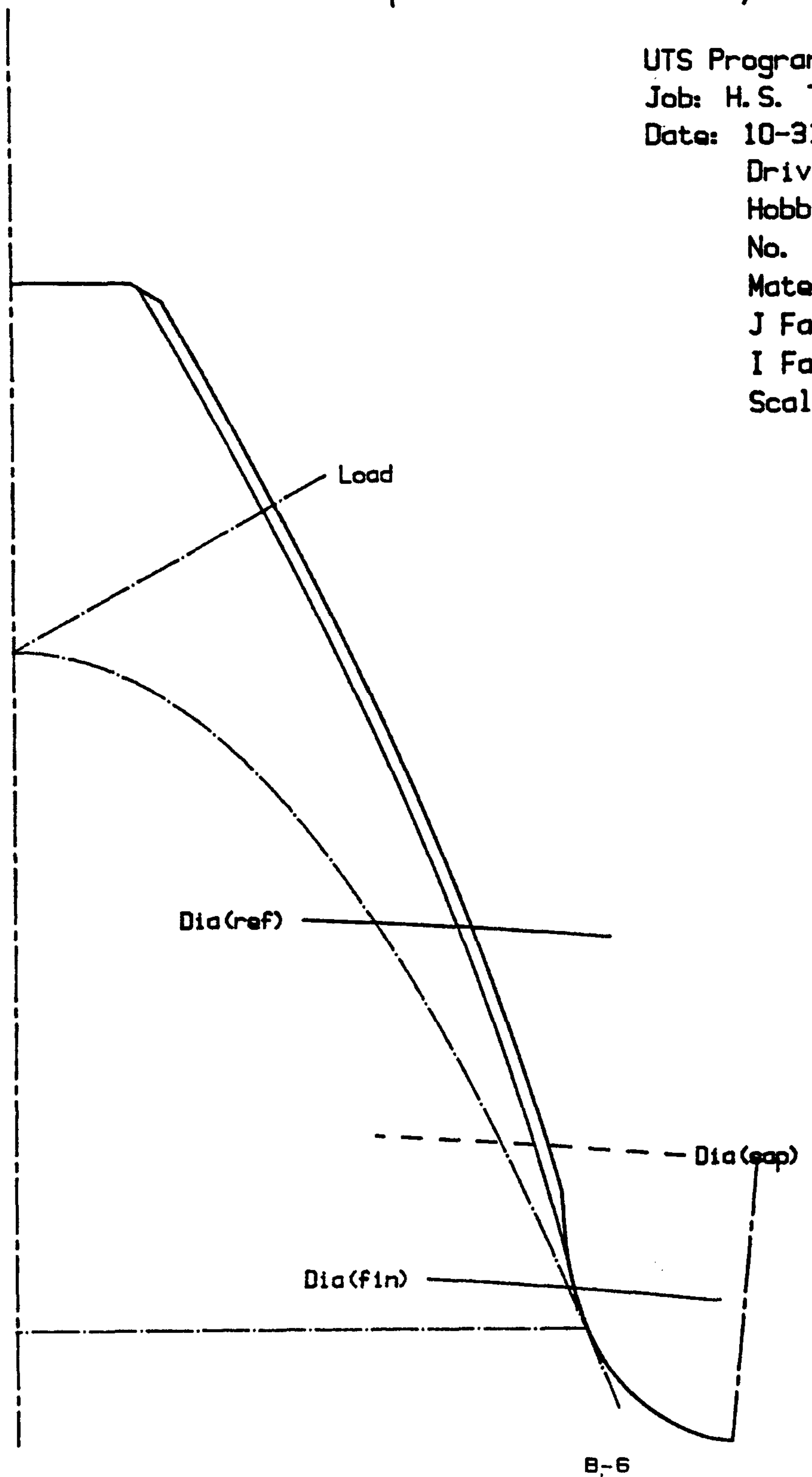
TACOM
Spur Gear Analysis

UTS Program #500
Job: H.S. Train-Ground Nominal
Date: 10-31-88
Driver
Hobbed-Ground
No. Teeth= 19
Mate= 32
J Factor= 0.465
I Factor= 0.111
Scale= 11



TACOM
Spur Gear Analysis

UTS Program #500
Job: H.S. Train-Ground Nominal
Date: 10-31-88
Driven
Hobbed-Ground
No. Teeth= 32
Mate= 19
J Factor= 0.499
I Factor= 0.111
Scale= 11



Appendix C

L.S. Train–Unground Nominal

TACOM
Gear Analysis

* Denotes Input Data

* Normal Diam Pitch=	3.5000	Opr Diam Pitch=	3.4514
* Normal Pressure Angle=	25.0000	Opr Pressure Angle=	26.6548
* Helix Angle=	0.0000		
Trans Diam Pitch=	3.5000	Line of Action=	1.0712
Trans Pressure Angle=	25.0000	% Approach Action=	48.88
		% Recess Action=	51.12
Opr Center Distance=	10.2857	Profile C.R.=	1.3168
* Face Width=	2.8800		
Basic Backlash=	0.0042		
Total Operating BL=	0.0136		

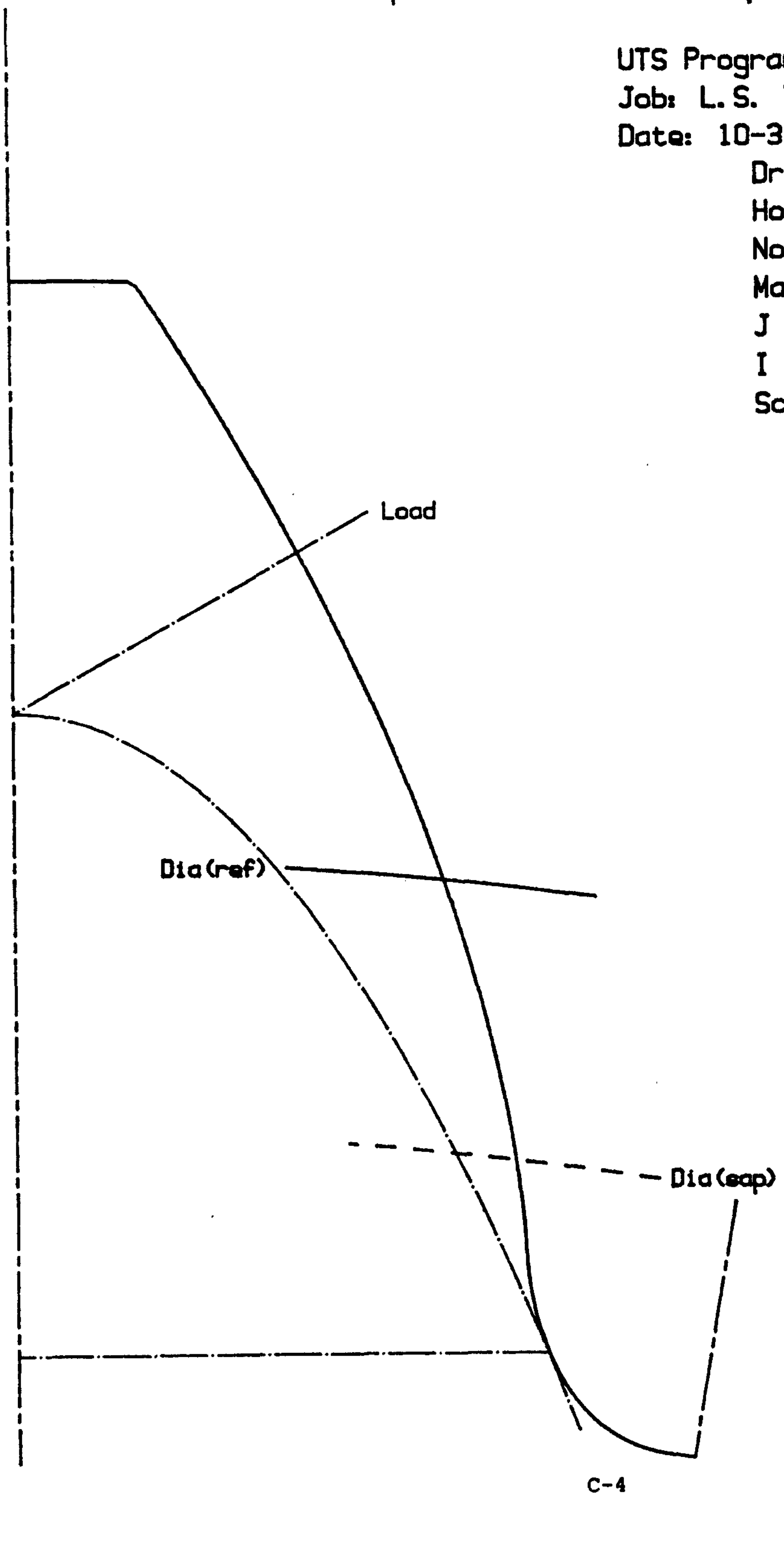
	DRIVER	(Deg Roll)	DRIVEN	(Deg Roll)
* Number of Teeth=	18		53	
* Outside Diameter=	5.7950	(42.33)	15.8585	(33.17)
Dia at Start of Tip Modification=	5.7900	(42.22)	15.8535	(33.13)
Circular Tip Relief at OD=	0.0029		0.0028	
* Total Normal Finish Stock=	0.0000		0.0000	
HOB FORM DATA	SEMI-TOPPING		SEMI-TOPPING	
* Hob Pressure Angle=	25.0000		25.0000	
* Hob Tip to Ref Line=	0.3615		0.3615	
* Hob Tooth Thickness at Ref=	0.4488		0.4488	
* Ref Line to Hob Mod Ramp=	0.2479		0.2682	
* Pressure Angle of Mod Ramp=	58.0000		58.0000	
* Hob Tip Radius=	0.0550		0.0550	
* Hob Protuberance=	0.0000		0.0000	
Hob SAP from Ref Line=	0.2637		0.2774	
Hob Space Width at Hob SAP=	0.1672		0.1692	
Normal Tooth Thickness at OD=	0.1381		0.1563	
Normal Tooth Thickness at Eff OD=	0.1474		0.1648	
*Normal Tooth Thickness, (Hobbed)=	0.5006		0.5209	
Dia @ Mid-point of Line of Action=	5.2261	(29.06)	15.3453	(28.66)
Pitch Diameter, (Ref)=	5.1429	(26.72)	15.1429	(26.72)
Operating Pitch Diameter=	5.2153	(28.76)	15.3561	(28.76)
Base Diameter=	4.6610		13.7241	
Dia, (Start of Active Profile)=	4.8369	(15.89)	14.8969	(24.19)
Form Diameter=	4.8369	(15.89)	14.8969	(24.19)
Root Diameter=	4.5309		14.5745	
Root Clearance=	0.0910		0.1010	
Max Undercut=	0.0000		0.0000	
Dia at Involute-Fillet Tangent=	4.7426	(10.77)	14.6780	(21.73)
Roll, radians, (1 tooth load)=	0.626	(35.89)	0.541	(30.98)
Minimum Fillet Radius=	0.0773		0.0617	
Helical Factor, C(h)=	1.000		1.000	
Y Factor=	0.750		0.871	
Load Sharing Ratio, m(N)=	1.000		1.000	
MODIFIED Stress Corr Fact, K(f)=	1.653		1.734	
J-Factor=	0.454		0.502	
I Factor=		0.125		
Max Specific Sliding Ratio=	1.09	(15.89)	0.75	(24.19)

Steel Gears

Universal Technical Systems, Inc, Rockford, Ill 61101 (Program #500)
Date: 10-31-88
Job : L.S. Train-Unground Nominal

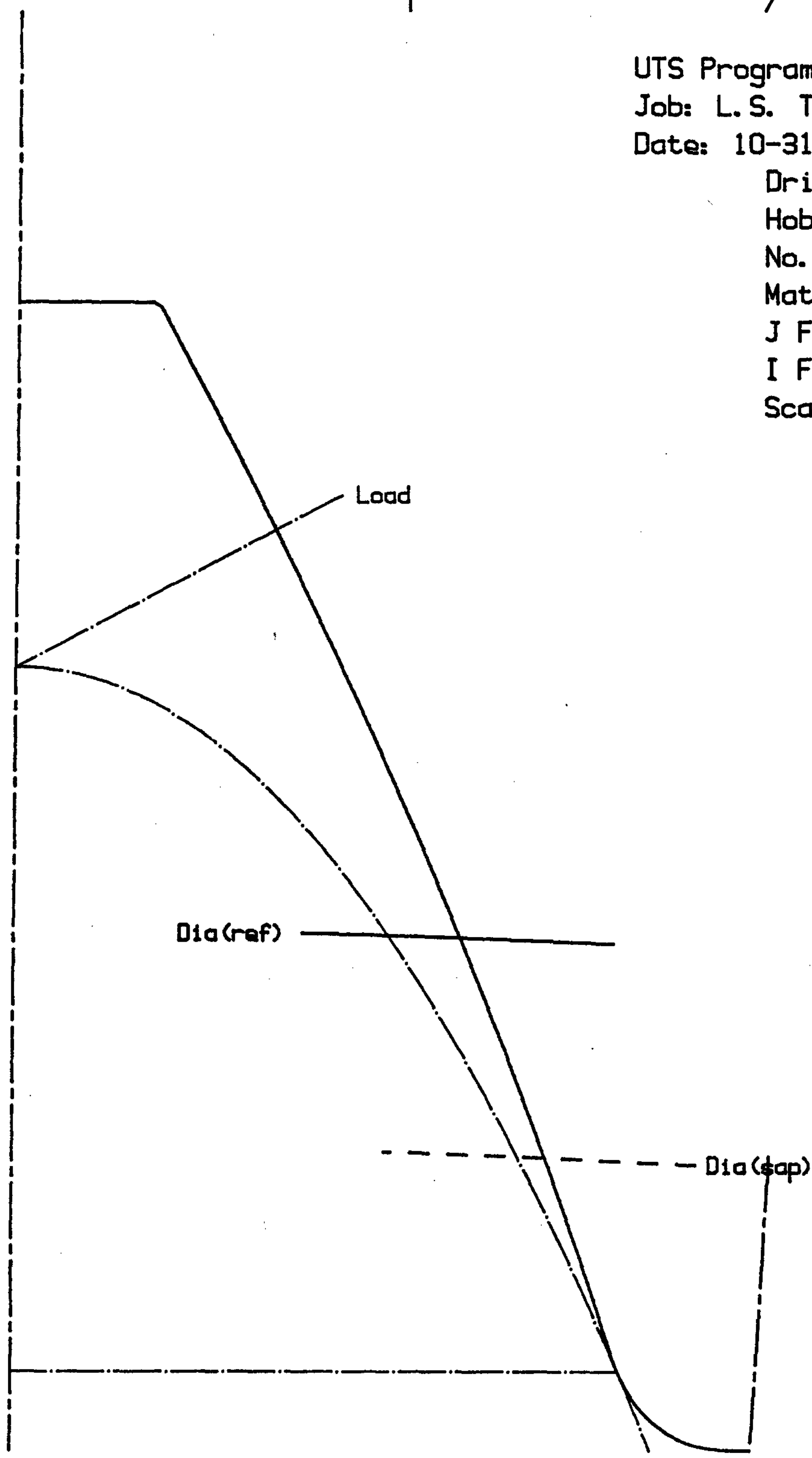
TACOM
Spur Gear Analysis

UTS Program #500
Job: L.S. Train-Unground Nominal
Date: 10-31-88
Driver
Hobbed Only
No. Teeth= 18
Mate= 53
J Factor= 0.454
I Factor= 0.125
Scale= 11

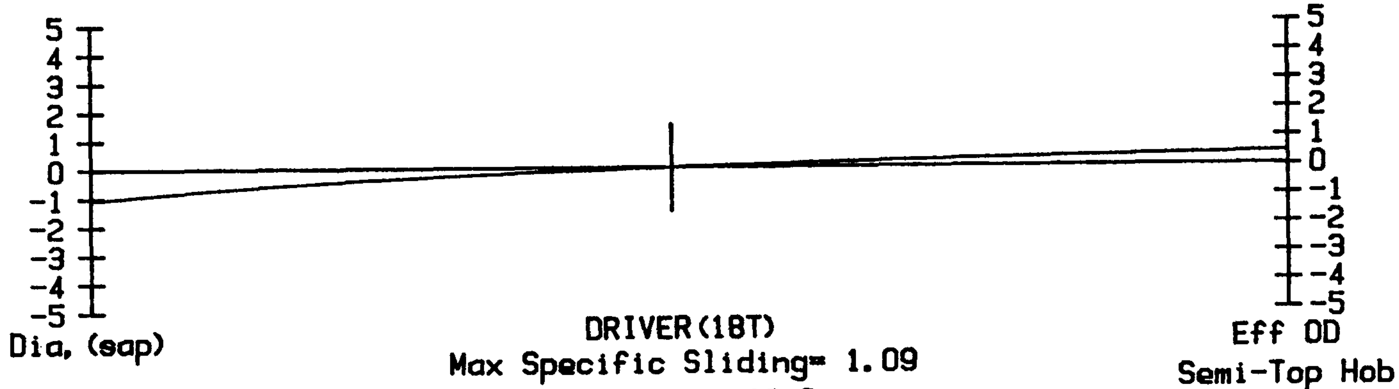


TACOM
Spur Gear Analysis

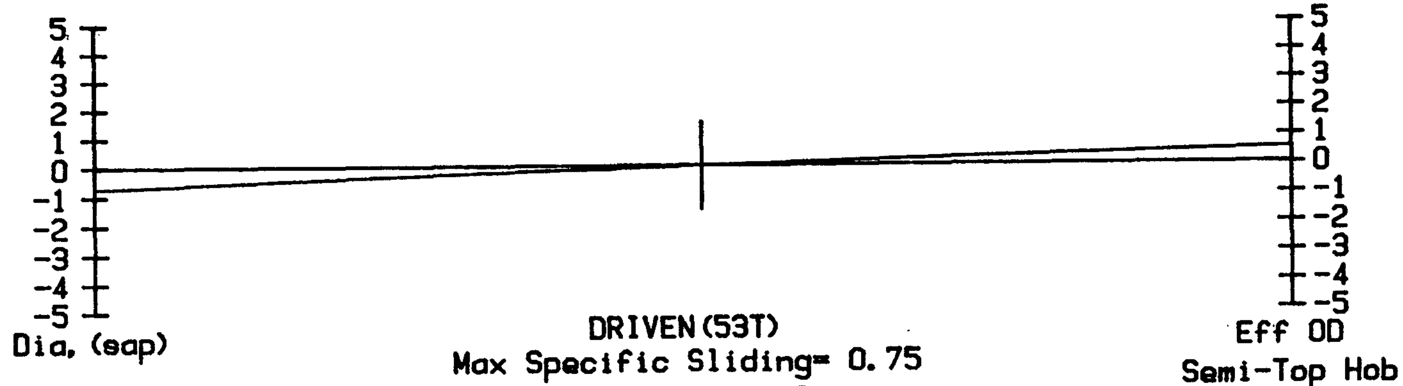
UTS Program #500
Job: L.S. Train-Unground Nominal
Date: 10-31-88
Driven
Hobbed Only
No. Teeth= 53
Mate= 18
J Factor= 0.502
I Factor= 0.125
Scale= 11



SPECIFIC SLIDING RATIOS (Slide/Roll)



DRIVER(18T)
Max Specific Sliding= 1.09
Max Slope= 13.8



DRIVEN(53T)
Max Specific Sliding= 0.75
Max Slope= 4.2

UTS #500 Job: L.S. Train-Unground Nominal Date: 10-31-88

TACOM

Appendix D

L.S. Train–Ground Nominal

TACOM
Gear Analysis

* Denotes Input Data			
* Normal Diam Pitch=	3.5000	Opr Diam Pitch=	3.4514
* Normal Pressure Angle=	25.0000	Opr Pressure Angle=	26.6548
* Helix Angle=	0.0000		
Trans Diam Pitch=	3.5000	Line of Action=	1.0701
Trans Pressure Angle=	25.0000	% Approach Action=	48.84
		% Recess Action=	51.16
Opr Center Distance=	10.2857	Profile C.R.=	1.3154
* Face Width=	2.8800		
Basic Backlash=	0.0042		
Total Operating BL=	0.0136		

	DRIVER	(Deg Roll)	DRIVEN	(Deg Roll)
* Number of Teeth=	18		53	
* Outside Diameter=	5.7950	(42.33)	15.8585	(33.17)
Dia at Start of Tip Modification=	5.7898	(42.22)	15.8525	(33.12)
Circular Tip Relief at OD=	0.0030		0.0034	
* Total Normal Finish Stock=	0.0150		0.0150	
HOB FORM DATA	SEMI-TOPPING		SEMI-TOPPING	
* Hob Pressure Angle=	25.0000		25.0000	
* Hob Tip to Ref Line=	0.3615		0.3615	
* Hob Tooth Thickness at Ref=	0.4338		0.4338	
* Ref Line to Hob Mod Ramp=	0.2412		0.2611	
* Pressure Angle of Mod Ramp=	58.0000		58.0000	
* Driven: Full Rad-Hob Tip Radius=	0.0550		0.0896	
* Hob Protuberance=	0.0080		0.0080	
Hob SAP from Ref Line=	0.2637		0.2774	
Hob Space Width at Hob SAP=	0.1670		0.1681	
Normal Tooth Thickness at OD=	0.1379		0.1551	
Normal Tooth Thickness at Eff OD=	0.1476		0.1654	
Normal Tooth Thickness, (Hobbed)=	0.5156		0.5359	
*Normal Tooth Thickness, (Ground)=	0.5006		0.5209	
Dia @ Mid-point of Line of Action=	5.2265	(29.07)	15.3449	(28.66)
Pitch Diameter, (Ref)=	5.1429	(26.72)	15.1429	(26.72)
Operating Pitch Diameter=	5.2153	(28.76)	15.3561	(28.76)
Base Diameter=	4.6610		13.7241	
Dia, (Start of Active Profile)=	4.8374	(15.91)	14.8970	(24.19)
Form Diameter=	4.8374	(15.91)	14.8970	(24.19)
Root Diameter=	4.5309		14.5745	
Root Clearance=	0.0910		0.1010	
Max Undercut=	0.0081		0.0086	
Diameter at Max Undercut=	4.7426	(10.77)	14.7118	(22.12)
* Finished Grind Diameter=	4.7426	(10.77)	14.7118	(22.13)
Roll, radians, (1 tooth load)=	0.627	(35.91)	0.541	(30.98)
Minimum Fillet Radius=	0.0773		0.0945	
Helical Factor, C(h)=	1.000		1.000	
Y Factor=	0.744		0.879	
Load Sharing Ratio, m(N)=	1.000		1.000	
MODIFIED Stress Corr Fact, K(f)=	1.649		1.720	
J-Factor=	0.451		0.511	
I Factor=		0.125		

Steel Gears, Finish Ground
Case Carburized

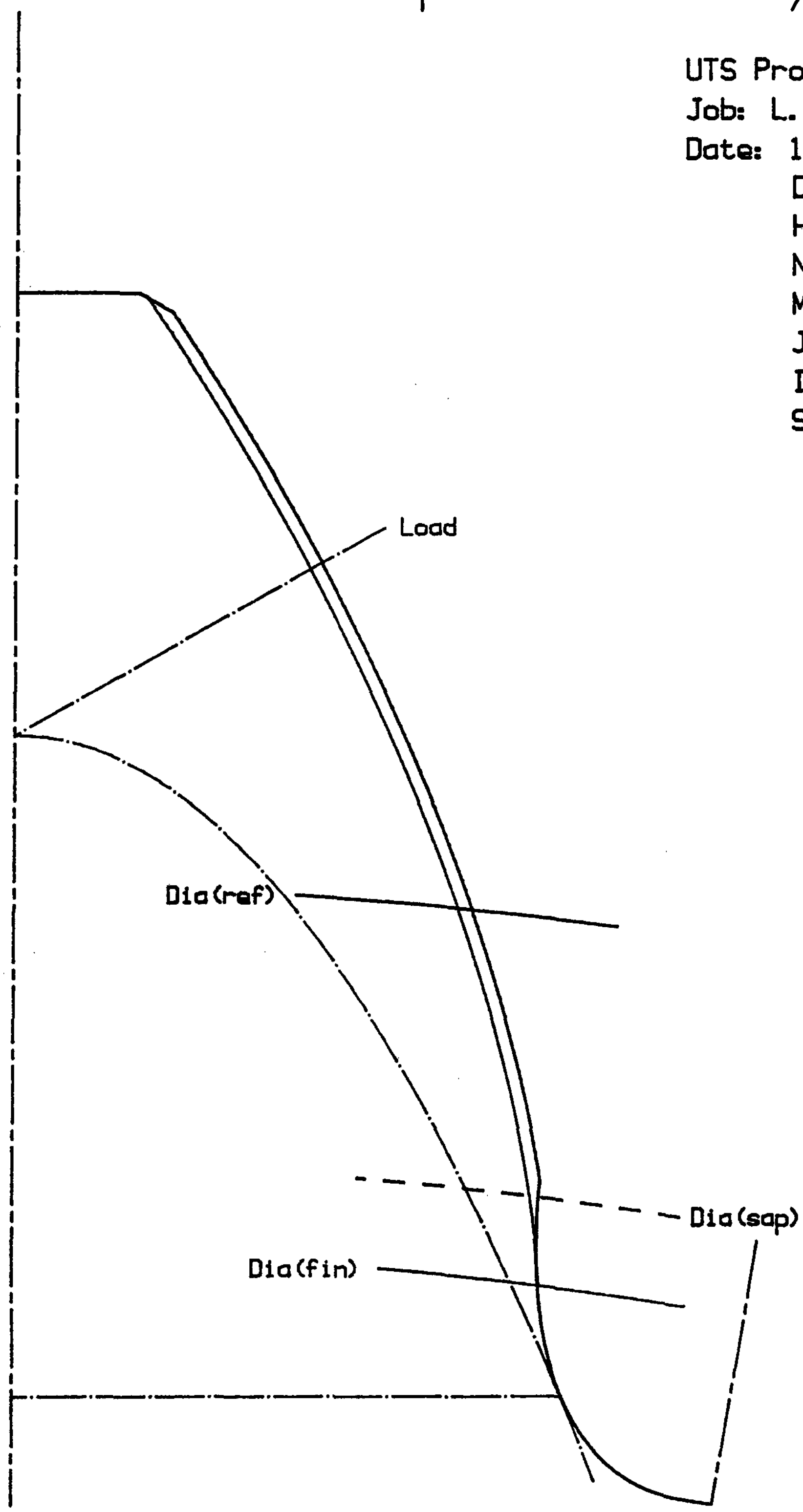
Job: L.S. Train-Ground Nominal (Program #500) Page 1 of 2

Universal Technical Systems, Inc, Rockford, Ill 61101 (Program #500)
Date: 10-31-88
Job : L.S. Train-Ground Nominal

TACOM

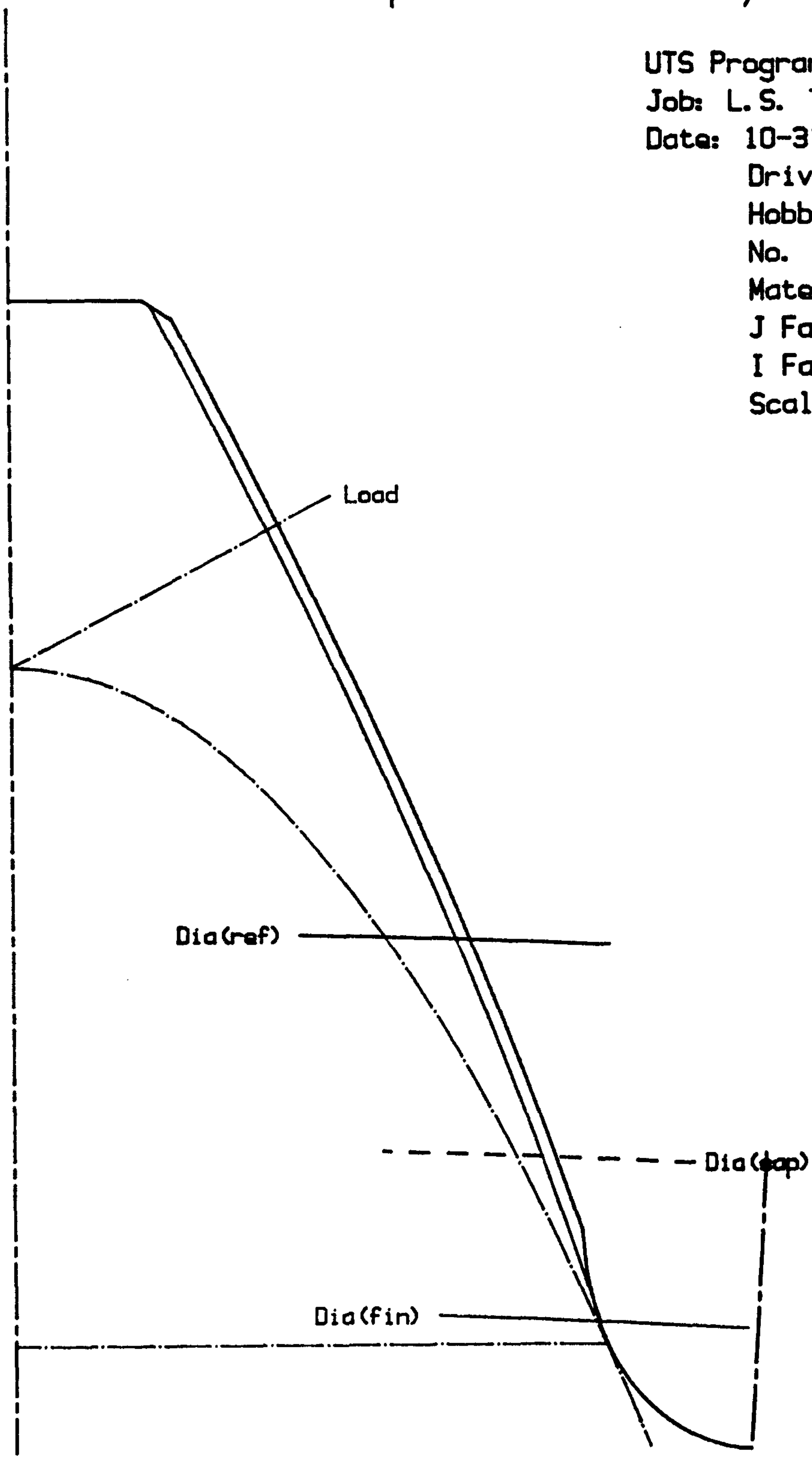
Spur Gear Analysis

UTS Program #500
Job: L.S. Train-Ground Nominal
Date: 10-31-88
Driver
Hobbed-Ground
No. Teeth= 18
Mate= 53
J Factor= 0.451
I Factor= 0.125
Scale= 11



TACOM
Spur Gear Analysis

UTS Program #500
Job: L.S. Train-Ground Nominal
Date: 10-31-88
Driven
Hobbed-Ground
No. Teeth= 53
Mate= 18
J Factor= 0.511
I Factor= 0.125
Scale= 11



Appendix E

UTS Data Memory Map

Format Diagram of 500 (Ver 4.0) ASCII Data File

	DRIVER	DRIVEN	GENERAL
Row 1			
Row 2			
Row 3			
⋮	⋮	⋮	⋮
Row 25			
Row 26			
Row 27			
⋮	⋮	⋮	
Row 82			
Row 83			

Effective March 1,1988

MEMORY MAP OF UTS PROGRAM 500 (Ver. 4.0) DATA FILES

Data files written to disk by program 500 are organized into three columns of data. The first column of data contains information about the driver gear, the second column about the driven gear, and the third column about the gear set in general.

There are 83 data elements in the driver column, 83 elements in the driven column, and 26 data elements in the general column:

DRIVER (1)	DRIVEN (1)	GENERAL (1)
•	•	•
•	•	•
•	•	GENERAL (26)
•	•	
•	•	
DRIVER (83)	DRIVEN (83)	

MAP OF DATA ELEMENTS AND DESCRIPTIONS OF THEIR CONTENTS:

DRIVER (1) • • Driver– Number of Teeth
DRIVER (2) • • Driver– Outside Diameter
DRIVER (3) • • Driver– Cut Transverse Backlash
DRIVER (4) • • Driver– Delta Addendum (Generating Rack Shift +/-)
DRIVER (5) • • Driver– Normal Finish Stock on Tooth Thickness
DRIVER (6) • • Driver– NPA of Hob or Number of Teeth in Shaper Cutter
DRIVER (7) • • Driver– Hob Tip to Ref Line or OD of Shaper Cutter
DRIVER (8) • • Driver– Hob Tooth Thick@ Ref Line or NTT of Shaper @ Ref PD
DRIVER (9) • • Driver– Hob Ref Line to Modification Ramp on Hob
DRIVER (10) • • Driver– Hob Ref Line to Hob Root for Topping Hob
DRIVER (11) • • Driver– Secondary Pressure Angle for Hob Mod Ramp
DRIVER (12) • • Driver– Radius in Hob Root fo Topping Hob
DRIVER (13) • • RESERVED FOR FUTURE USE
DRIVER (14) • • RESERVED FOR FUTURE USE
DRIVER (15) • • Driver– Tip Radius on Hob or Shaper Tooth
DRIVER (16) • • Driver– Protuberance on Hob or Shaper Tooth
DRIVER (17) • • Driver– Normal Tooth Thickness @ Actual OD
DRIVER (18) • • Driver– Normal Tooth Thickness @ Effective OD
DRIVER (19) • • Driver– Normal Tooth Thickness @ Ref PD Hobbed or Shaped
DRIVER (20) • • Driver– Normal Tooth Thickness Finished
DRIVER (21) • • Driver– Diameter @ Line of Action Mid-point
DRIVER (22) • • Driver– Reference Pitch Diameter
DRIVER (23) • • Driver– Operating Pitch Diameter
DRIVER (24) • • Driver– Base Diameter
DRIVER (25) • • Driver– Diameter @ Start of Active Profile
DRIVER (26) • • Driver– Form Diameter

DRIVER (27) . . Driver– Root Diameter
DRIVER (28) . . Driver– Helical Lead for Helical Gear
DRIVER (29) . . Driver– Number of Shaving Cutter Teeth / Grind Wheel PA, radius
DRIVER (30) . . Driver– Helix Angle of Shaving Cutter / Grind Wheel Tip to Ref
DRIVER (31) . . Driver– Normal Tooth Thickness of Shaving Cutter / Grind Wheel Thickness at Ref
DRIVER (32) . . Driver– Shaving Cutter OD / Grind Wheel Tip Radius
DRIVER (33) . . Driver– Center Distance Between Shaving Cutter and Gear / Grinding Helix Angle, deg
DRIVER (34) . . Driver– Clearance Between Shaving Cutter OD and Gear Root / Root Grind Stock: +Stock, -Clearance
DRIVER (35) . . Driver– Maximum Undercut on Gear (Extended Profile to Fillet)
DRIVER (36) . . Driver– Diameter @ Max U.C. or Involute/Fillet Tangent Diameter
DRIVER (37) . . Driver– Diameter to Which Gear is Finished by Grinding or Shaving
DRIVER (38) . . Driver– Maximum Specific Sliding Ratio
DRIVER (39) . . Driver– Roll Angle to Load Line in Radians
DRIVER (40) . . Driver– Minimum Fillet Radius
DRIVER (41) . . Driver– AGMA: Ch (Full Helicals) or C(Psi) (LCR Helicals)
DRIVER (42) . . Driver– AGMA: Y-factor
DRIVER (43) . . Driver– AGMA: mN (Load Sharing Factor)
DRIVER (44) . . Driver– AGMA: Kf (Root Fillet Stress Correction Factor)
DRIVER (45) . . Driver– AGMA: J-factor (Strength)
DRIVER (46) . . Driver– AGMA: I-factor (Durability)
DRIVER (47) . . Driver– Roll Angle @ Contact Limit Diameter in Radians
DRIVER (48) . . Driver– Finish tool clearance (Involute Profile to Fillet)
DRIVER (49) . . Driver– Distance from SAP to Fin Involute if SAP is Below Fin Involute
DRIVER (50) . . Driver– Roll Angle @ Outside Diameter, degrees
DRIVER (51) . . Driver– Roll Angle @ Start of Modification on Finished Tooth, degrees
DRIVER (52) . . Driver– Roll Angle @ Reference Pitch Diameter, degrees
DRIVER (53) . . Driver– Roll Angle @ Operating Pitch Diameter, degrees
DRIVER (54) . . Driver– Roll Angle @ Contact Limit Diameter, degrees
DRIVER (55) . . Driver– Roll Angle @ Form Diameter, degrees
DRIVER (56) . . Driver– Roll Angle @ Finish Diameter, degrees
DRIVER (57) . . Driver– Roll Angle @ Max UC or Inv-Fil Tan Diameter, degrees
DRIVER (58) . . Driver– Normal Tooth Thickness @ Shave Cutter Tooth Tip
DRIVER (59) . . Driver– Normal Space Width @ Shave Cutter Start of Active Profile
DRIVER (60) . . Driver– Hobbed Transverse Circular Tip relief @ Gear OD
DRIVER (61) . . Driver– Hobbed Normal Tip Relief Normal to Involute
DRIVER (62) . . Driver– Finished Transverse Circular Tip relief @Gear OD
DRIVER (63) . . Driver– Finished Tip Relief Normal to Involute
DRIVER (64) . . Driver– Effective Finished Gear OD (Start of Chamfer)
DRIVER (65) . . Driver– Effective Hobbed Gear OD (Start of Chamfer)
DRIVER (66) . . Driver– Final Effective Gear OD (Hobbed/Shaped or Finished)
DRIVER (67) . . Driver– Final Actual Gear OD (Machine, Topped or Pointed Tip)
DRIVER (68) . . Driver– Developed Involute Arc Length of Modification with Tip Rel Hob
DRIVER (69) . . Driver– Distance from Form Dia to Inv Profile if Form is Below Profile
DRIVER (70) . . Driver– Light Load Profile C.R. (No Contacts on Modified Profile)

DRIVER (71) . . Driver– Hob Ref Line to Deepest Point of Contact on Hob
DRIVER (72) . . Driver– Hob Space Width @ Deepest Point of Contact
DRIVER (73) . . Driver– Roll Angle @ Line of Action Mid-Point, degrees
DRIVER (74) . . Driver– Clearance Between Driver Root and Driven OD
DRIVER (75) . . Driver– Almen-Straub Strength Factor
DRIVER (76) . . Almen-Straub Pitting Factor
DRIVER (77) . . Driver– AGMA Stress Correction Factor, K(f)
DRIVER (78) . . Driver– MODIFIED Stress Correction Factor
DRIVER (79) . . Driver– Rad of Curvature at Max Stress Point, Hob Shape
DRIVER (80) . . Driver– Rad of Curvature at Max Stress Point, Fil Grind
DRIVER (81) . . Driver– Grinding Wheel Normal Linear Pitch
DRIVER (82) . . Driver– Process: Hobbed = 1, Shaped = 2
DRIVER (83) . . Driver– Post: None = 0, Shave = 1, Grind = 2, Fil Grind = 3

DRIVEN (1) . . Driven– Number of Teeth
DRIVEN (2) . . Driven– Outside Diameter
DRIVEN (3) . . Driven– Cut Transverse Backlash
DRIVEN (4) . . Driven– Delta Addendum (Generating Rack Shift +/-)
DRIVEN (5) . . Driven– Normal Finish Stock on Tooth Thickness
DRIVEN (6) . . Driven– NPA of Hob or Number of Teeth in Shaper Cutter
DRIVEN (7) . . Driven– Hob Tip to Ref Line or OD of Shaper Cutter
DRIVEN (8) . . Driven– Hob Tooth Thick@ Ref Line or NTT of Shaper @ Ref PD
DRIVEN (9) . . Driven– Hob Ref Line to Modification Ramp on Hob
DRIVEN (10) . . Driven– Hob Ref Line to Hob Root for Topping Hob
DRIVEN (11) . . Driven– Secondary Pressure Angle for Hob Mod Ramp
DRIVEN (12) . . Driven– Radius in Hob Root fo Topping Hob
DRIVEN (13) . . RESERVED FOR FUTURE USE
DRIVEN (14) . . RESERVED FOR FUTURE USE
DRIVEN (15) . . Driven– Tip Radius on Hob or Shaper Tooth
DRIVEN (16) . . Driven– Protuberance on Hob or Shaper Tooth
DRIVEN (17) . . Driven– Normal Tooth Thickness @ Actual OD
DRIVEN (18) . . Driven– Normal Tooth Thickness @ Effective OD
DRIVEN (19) . . Driven– Normal Tooth Thickness @ Ref PD Hobbed or Shaped
DRIVEN (20) . . Driven– Normal Tooth Thickness Finished
DRIVEN (21) . . Driven– Diameter @ Line of Action Mid-point
DRIVEN (22) . . Driven– Reference Pitch Diameter
DRIVEN (23) . . Driven– Operating Pitch Diameter
DRIVEN (24) . . Driven– Base Diameter
DRIVEN (25) . . Driven– Diameter @ Start of Active Profile
DRIVEN (26) . . Driven– Form Diameter
DRIVEN (27) . . Driven– Root Diameter
DRIVEN (28) . . Driven– Helical Lead for Helical Gear
DRIVEN (29) . . Driven– Number of Shaving Cutter Teeth / Grind Wheel PA, radius
DRIVEN (30) . . Driven– Helix Angle of Shaving Cutter / Grind Wheel Tip to Ref

DRIVEN (31) . . Driven– Normal Tooth Thickness of Shaving Cutter / Grind Wheel Thickness at Ref
DRIVEN (32) . . Driven– Shaving Cutter OD / Grind Wheel Tip Radius
DRIVEN (33) . . Driven– Center Distance Between Shaving Cutter and Gear / Grinding Helix Angle, deg
DRIVEN (34) . . Driven– Clearance Between Shaving Cutter OD and Gear Root / Root Grind Stock: +Stock, -Clearance
DRIVEN (35) . . Driven– Maximum Undercut on Gear (Extended Profile to Fillet)
DRIVEN (36) . . Driven– Diameter @ Max U.C. or Involute/Fillet Tangent Diameter
DRIVEN (37) . . Driven– Diameter to Which Gear is Finished by Grinding or Shaving
DRIVEN (38) . . Driven– Maximum Specific Sliding Ratio
DRIVEN (39) . . Driven– Roll Angle to Load Line in Radians
DRIVEN (40) . . Driven– Minimum Fillet Radius
DRIVEN (41) . . Driven– AGMA: Ch (Full Helicals) or C(Psi) (LCR Helicals)
DRIVEN (42) . . Driven– AGMA: Y-factor
DRIVEN (43) . . Driven– AGMA: mN (Load Sharing Factor)
DRIVEN (44) . . Driven– AGMA: Kf (Root Fillet Stress Correction Factor)
DRIVEN (45) . . Driven– AGMA: J-factor (Strength)
DRIVEN (46) . . Driven– AGMA: I-factor (Durability)
DRIVEN (47) . . Driven– Roll Angle @ Contact Limit Diameter in Radians
DRIVEN (48) . . Driven– Finish tool clearance (Involute Profile to Fillet)
DRIVEN (49) . . Driven– Distance from SAP to Fin Involute if SAP is Below Fin Involute
DRIVEN (50) . . Driven– Roll Angle @ Outside Diameter, degrees
DRIVEN (51) . . Driven– Roll Angle @ Start of Modification on Finished Tooth, degrees
DRIVEN (52) . . Driven– Roll Angle @ Reference Pitch Diameter, degrees
DRIVEN (53) . . Driven– Roll Angle @ Operating Pitch Diameter, degrees
DRIVEN (54) . . Driven– Roll Angle @ Contact Limit Diameter, degrees
DRIVEN (55) . . Driven– Roll Angle @ Form Diameter, degrees
DRIVEN (56) . . Driven– Roll Angle @ Finish Diameter, degrees
DRIVEN (57) . . Driven– Roll Angle @ Max UC or Inv-Fil Tan Diameter, degrees
DRIVEN (58) . . Driven– Normal Tooth Thickness @ Shave Cutter Tooth Tip
DRIVEN (59) . . Driven– Normal Space Width @ Shave Cutter Start of Active Profile
DRIVEN (60) . . Driven– Hobbed Transverse Circular Tip relief @ Gear OD
DRIVEN (61) . . Driven– Hobbed Normal Tip Relief Normal to Involute
DRIVEN (62) . . Driven– Finished Transverse Circular Tip relief @Gear OD
DRIVEN (63) . . Driven– Finished Tip Relief Normal to Involute
DRIVEN (64) . . Driven– Effective Finished Gear OD (Start of Chamfer)
DRIVEN (65) . . Driven– Effective Hobbed Gear OD (Start of Chamfer)
DRIVEN (66) . . Driven– Final Effective Gear OD (Hobbed/Shaped or Finished)
DRIVEN (67) . . Driven– Final Actual Gear OD (Machine, Topped or Pointed Tip)
DRIVEN (68) . . Driven– Developed Involute Arc Length of Modification with Tip Rel Hob
DRIVEN (69) . . Driven– Distance from Form Dia to Inv Profile if Form is Below Profile
DRIVEN (70) . . Driven– Light Load Profile C.R. (No Contacts on Modified Profile)
DRIVEN (71) . . Driven– Hob Ref Line to Deepest Point of Contact on Hob
DRIVEN (72) . . Driven– Hob Space Width @ Deepest Point of Contact
DRIVEN (73) . . Driven– Roll Angle @ Line of Action Mid-Point, degrees
DRIVEN (74) . . Driven– Clearance Between Driver Root and Driven OD

DRIVEN (75) . . Driven– Almen-Straub Strength Factor
DRIVEN (76) . . Almen-Straub Pitting Factor
DRIVEN (77) . . Driven– AGMA Stress Correction Factor, K(f)
DRIVEN (78) . . Driven– MODIFIED Stress Correction Factor
DRIVEN (79) . . Driven– Rad of Curvature at Max Stress Point, Hob Shape
DRIVEN (80) . . Driven– Rad of Curvature at Max Stress Point, Fil Grind
DRIVEN (81) . . Driven– Grinding Wheel Normal Linear Pitch
DRIVEN (82) . . Driven– Process: Hobbed = 1, Shaped = 2
DRIVEN (83) . . Driven– Post: None = 0, Shave = 1, Grind = 2, Fil Grind = 3

GENERAL (1) . . Normal Diametral Pitch
GENERAL (2) . . Nominal Normal Pressure Angle
GENERAL (3) . . Helix Angle @ Reference Pitch Diameters, rad
GENERAL (4) . . Operating Center Distance
GENERAL (5) . . Transverse Pressure Angle @ Reference Pitch Diameters, rad
GENERAL (6) . . Operating Transverse Pressure Angle @ Operating Pitch Diameters, rad
GENERAL (7) . . Operating Helix Angle @ Operating Pitch Diameters, rad
GENERAL (8) . . Transverse Diametral Pitch @ Reference Pitch Diameters
GENERAL (9) . . Face Width
GENERAL (10) . . Profile Contact Ratio
GENERAL (11) . . Axial (Face, Helical) Contact Ratio
GENERAL (12) . . Helix Angle on the base Cylinders
GENERAL (13) . . AGMA: Contact Height Factor (Cx)
GENERAL (14) . . AGMA: Contact Height Factor for Equiv Normal LCR Helicals (Cxh)
GENERAL (15) . . AGMA: Helical Factor (C(Psi))
GENERAL (16) . . Roll Angle on Driver @ Calc Point for I-Factor (LCR Helical & Spurs)
GENERAL (17) . . Roll Angle on Driver @ Calc Point for I-Factor (Full Helicals)
GENERAL (18) . . Total Length of Zone of Action in the Transverse Plane (Z)
GENERAL (19) . . Diameter on Driver @ the Radial Mid-point of the Mesh
GENERAL (20) . . Operating Transverse Diametral Pitch.
GENERAL (21) . . Total Operating Transverse Backlash.
GENERAL (22) . . Basic Transverse Backlash.
GENERAL (23) . . % Approach.
GENERAL (24) . . % Recess.
GENERAL (25) . . Minimum Contact Line Length
GENERAL (26) . . Maximum Contact Line Length

Appendix F

H.S. Train–MLRS

TACOM
Gear Analysis

* Denotes Input Data

* Normal Diam Pitch=	3.5000	Opr Diam Pitch=	3.5000
* Normal Pressure Angle=	25.0000	Opr Pressure Angle=	25.0000
* Helix Angle=	0.0000		
Trans Diam Pitch=	3.5000	Line of Action=	1.0476
Trans Pressure Angle=	25.0000	% Approach Action=	35.01
		% Recess Action=	64.99
Opr Center Distance=	7.4286	Profile C.R.=	1.2878
* Face Width=	1.5820		
Basic Backlash=	0.0000		
Total Operating BL=	0.0135		

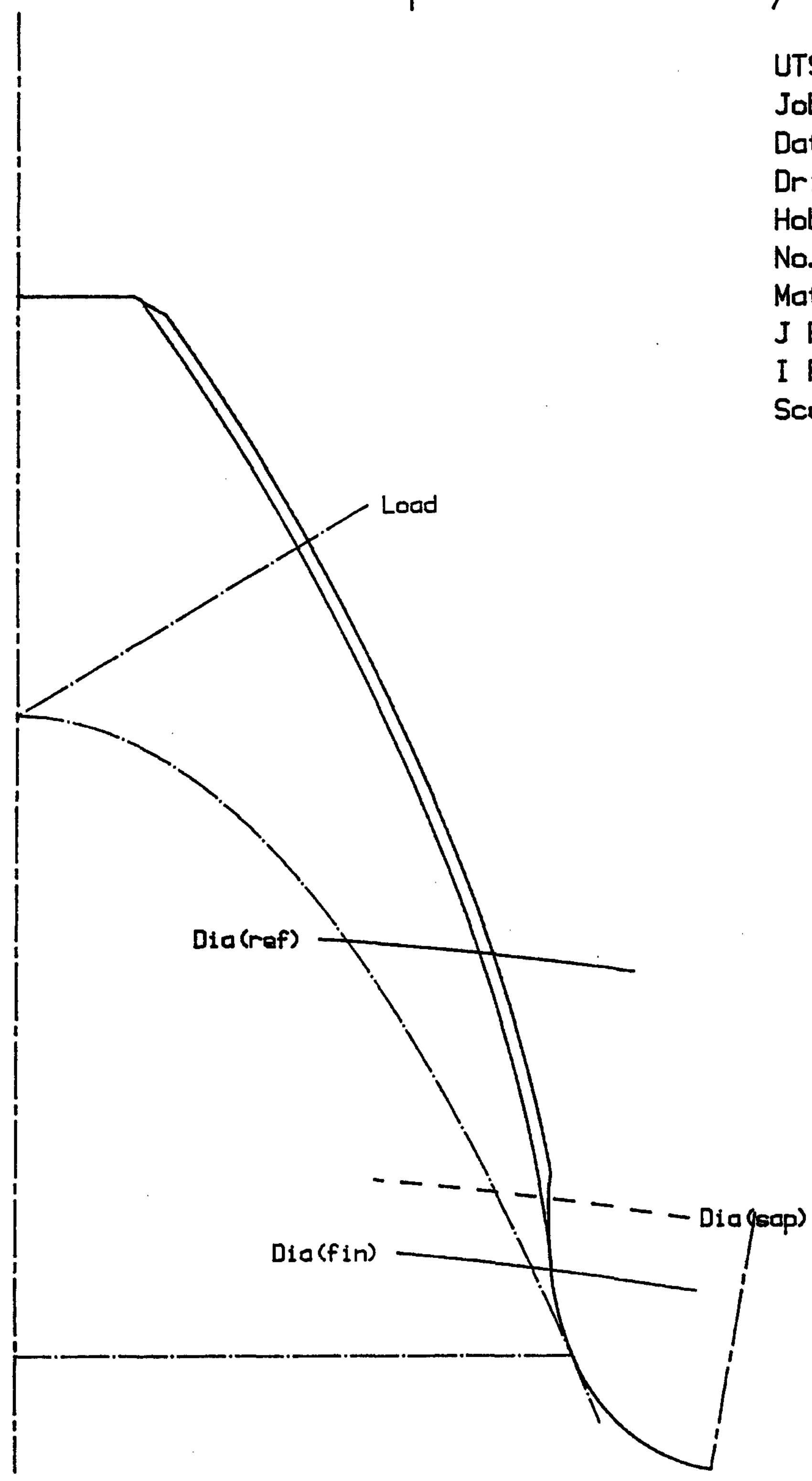
	DRIVER	(Deg Roll)	DRIVEN	(Deg Roll)
* Number of Teeth=	18		34	
* Outside Diameter=	5.8550	(43.56)	10.0525	(31.57)
Dia at Start of Tip Modification=	5.8499	(43.46)	10.0464	(31.49)
Circular Tip Relief at OD=	0.0029		0.0035	
* Total Normal Finish Stock=	0.0150		0.0150	
HOB FORM DATA	SEMI-TOPPING		SEMI-TOPPING	
* Hob Pressure Angle=	25.0000		25.0000	
* Hob Tip to Ref Line=	0.3615		0.3615	
* Hob Tooth Thickness at Ref=	0.4338		0.4338	
* Ref Line to Hob Mod Ramp=	0.2322		0.2617	
* Pressure Angle of Mod Ramp=	58.0000		58.0000	
* Both: Full Rad-Hob Tip Radius=	0.0896		0.0896	
* Hob Protuberance=	0.0080		0.0080	
Hob SAP from Ref Line=	0.2567		0.2737	
Hob Space Width at Hob SAP=	0.1687		0.1812	
Normal Tooth Thickness at OD=	0.1323		0.1802	
Normal Tooth Thickness at Eff OD=	0.1417		0.1904	
Normal Tooth Thickness, (Hobbed)=	0.5489		0.3652	
*Normal Tooth Thickness, (Ground)=	0.5339		0.3502	
Dia @ Mid-point of Line of Action=	5.2832	(30.58)	9.5858	(24.67)
Pitch Diameter, (Ref)=	5.1429	(26.72)	9.7143	(26.72)
Operating Pitch Diameter=	5.1429	(26.72)	9.7143	(26.72)
Base Diameter=	4.6610		8.8041	
Dia, (Start of Active Profile)=	4.8783	(17.70)	9.2218	(17.86)
Form Diameter=	4.8783	(17.70)	9.2218	(17.86)
Root Diameter=	4.6024		8.7798	
Root Clearance=	0.1012		0.1112	
Max Undercut=	0.0082		0.0082	
Diameter at Max Undercut=	4.7982	(14.01)	9.0603	(13.92)
* Finished Grind Diameter=	4.7982	(14.00)	9.0603	(13.92)
Roll, radians, (1 tooth load)=	0.658	(37.70)	0.496	(28.44)
Minimum Fillet Radius=	0.1015		0.1168	
Helical Factor, C(h)=	1.000		1.000	
Y Factor=	0.836		0.597	
Load Sharing Ratio, m(N)=	1.000		1.000	
MODIFIED Stress Corr Fact, K(f)=	1.722		1.522	
J-Factor=	0.485		0.392	
I Factor=		0.117		
Max Specific Sliding Ratio=	0.78	(17.70)	1.43	(17.86)

Steel Gears, Finish Ground
Case Carburized

Universal Technical Systems, Inc, Rockford, Ill 61101 (Program #500)
Date: 1-3-89
Job : H.S. Train-MLRS

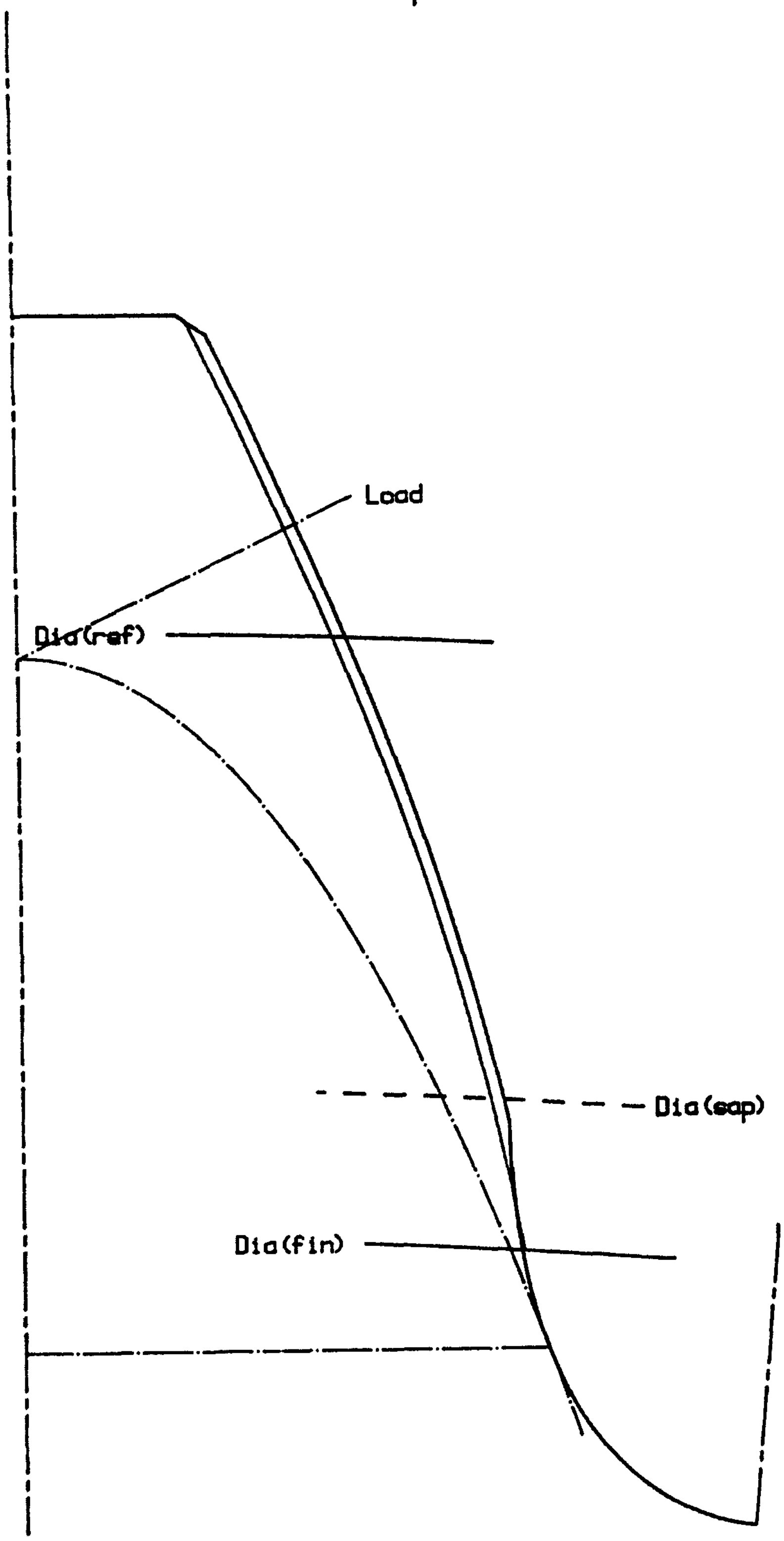
TACOM
Spur Gear Analysis

UTS Program #500
Job: H.S. Train-MLRS
Date: 1-3-89
Driver
Hobbed-Ground
No. Teeth= 18
Mate= 34
J Factor= 0.485
I Factor= 0.117
Scale= 11

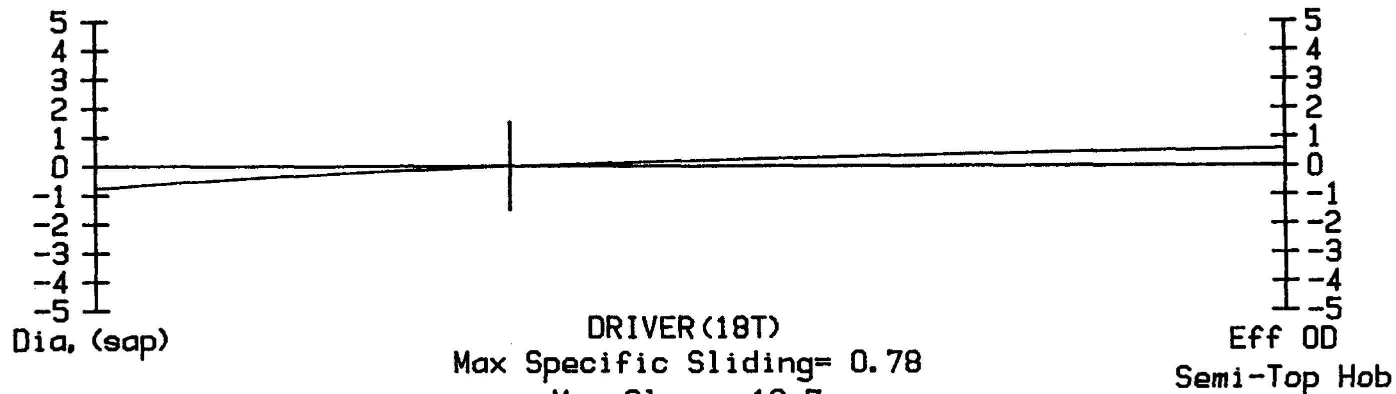


TACOM
Spur Gear Analysis

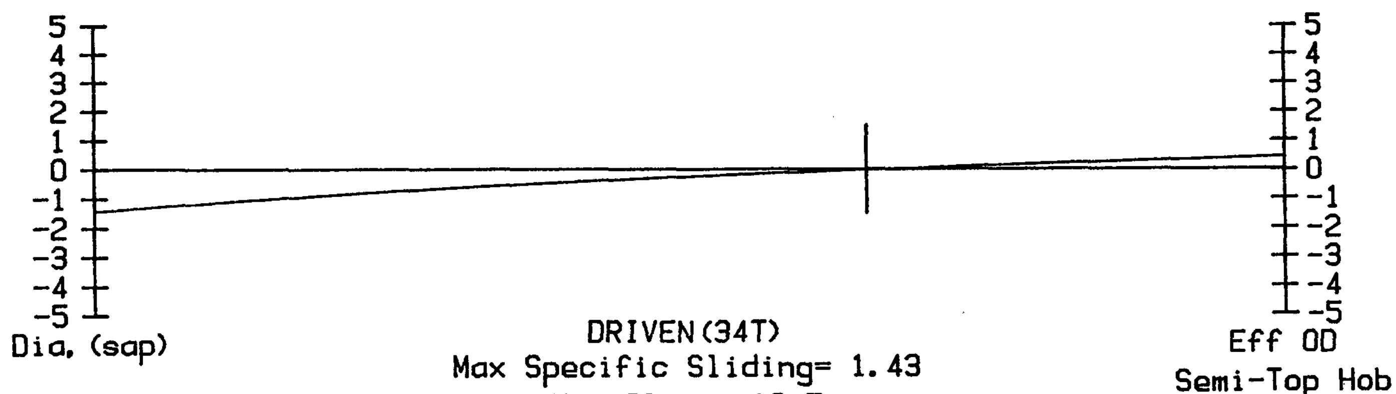
UTS Program #500
Job: H.S. Train-MLRS
Date: 1-3-89
Driven
Hobbed-Ground
No. Teeth= 34
Mate= 18
J Factor= 0.392
I Factor= 0.117
Scale= 11



SPECIFIC SLIDING RATIOS (Slide/Roll)



DRIVER(18T)
Max Specific Sliding= 0.78
Max Slope= 10.7



DRIVEN(34T)
Max Specific Sliding= 1.43
Max Slope= 10.5

UTS #500 Job: H.S. Train-MLRS Date: 1-3-89

TACOM

Appendix G

L.S. Train–MLRS

TACOM
Gear Analysis

* Denotes Input Data

* Normal Diam Pitch=	3.5000	Opr Diam Pitch=	3.4514
* Normal Pressure Angle=	25.0000	Opr Pressure Angle=	26.6548
* Helix Angle=	0.0000		
Trans Diam Pitch=	3.5000	Line of Action=	1.0701
Trans Pressure Angle=	25.0000	% Approach Action=	48.84
		% Recess Action=	51.16
Opr Center Distance=	10.2857	Profile C.R.=	1.3154
* Face Width=	2.8800		
Basic Backlash=	0.0042		
Total Operating BL=	0.0136		

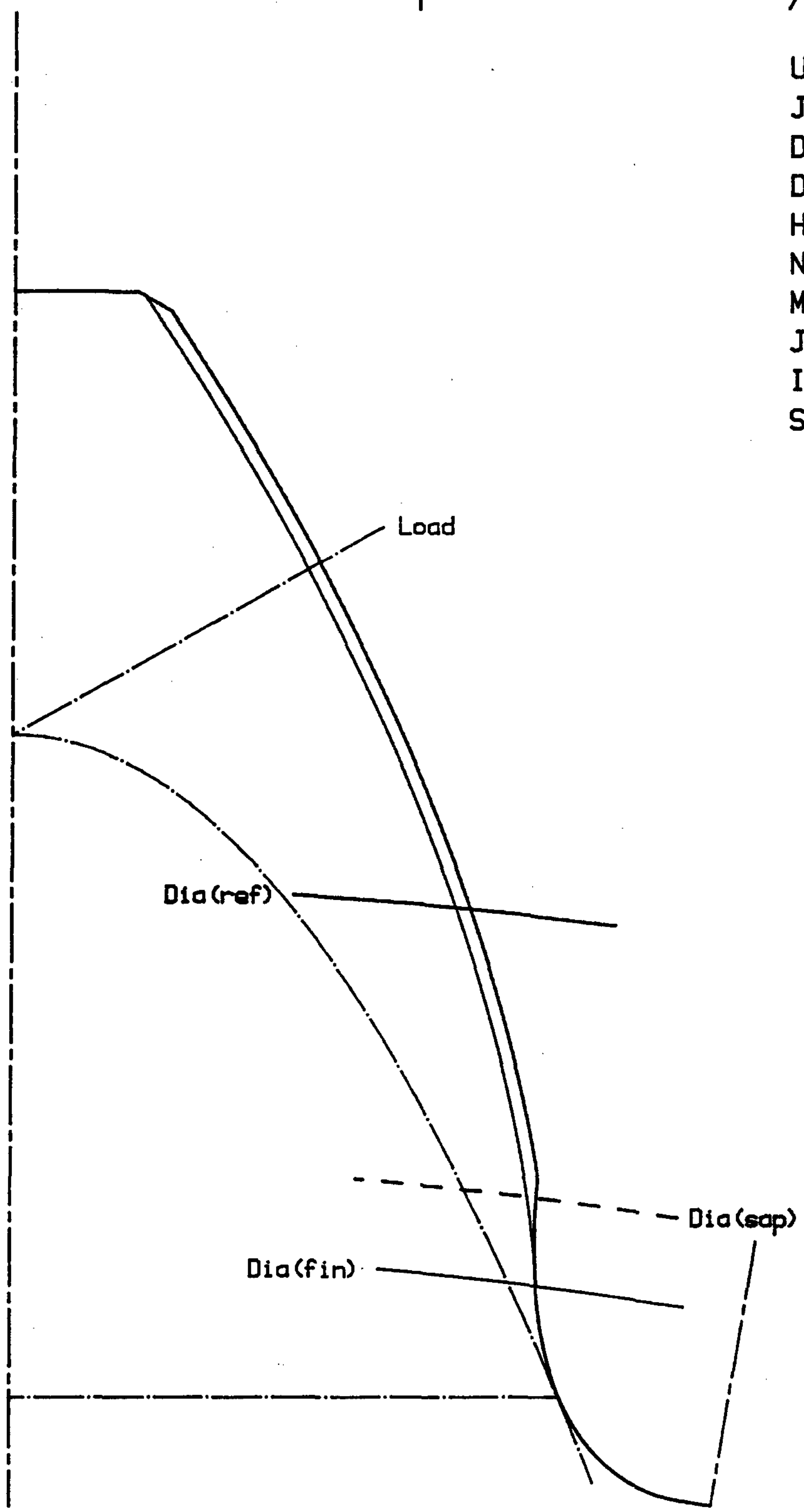
	DRIVER	(Deg Roll)	DRIVEN	(Deg Roll)
* Number of Teeth=	18		53	
* Outside Diameter=	5.7950	(42.33)	15.8585	(33.17)
Dia at Start of Tip Modification=	5.7898	(42.22)	15.8525	(33.12)
Circular Tip Relief at OD=	0.0030		0.0034	
* Total Normal Finish Stock=	0.0150		0.0150	
HOB FORM DATA	SEMI-TOPPING		SEMI-TOPPING	
* Hob Pressure Angle=	25.0000		25.0000	
* Hob Tip to Ref Line=	0.3615		0.3615	
* Hob Tooth Thickness at Ref=	0.4338		0.4338	
* Ref Line to Hob Mod Ramp=	0.2412		0.2611	
* Pressure Angle of Mod Ramp=	58.0000		58.0000	
* Driven: Full Rad-Hob Tip Radius=	0.0550		0.0896	
* Hob Protuberance=	0.0080		0.0080	
Hob SAP from Ref Line=	0.2637		0.2774	
Hob Space Width at Hob SAP=	0.1670		0.1681	
Normal Tooth Thickness at OD=	0.1379		0.1551	
Normal Tooth Thickness at Eff OD=	0.1476		0.1654	
Normal Tooth Thickness, (Hobbed)=	0.5156		0.5359	
*Normal Tooth Thickness, (Ground)=	0.5006		0.5209	
Dia @ Mid-point of Line of Action=	5.2265	(29.07)	15.3449	(28.66)
Pitch Diameter, (Ref)=	5.1429	(26.72)	15.1429	(26.72)
Operating Pitch Diameter=	5.2153	(28.76)	15.3561	(28.76)
Base Diameter=	4.6610		13.7241	
Dia, (Start of Active Profile)=	4.8374	(15.91)	14.8970	(24.19)
Form Diameter=	4.8374	(15.91)	14.8970	(24.19)
Root Diameter=	4.5309		14.5745	
Root Clearance=	0.0910		0.1010	
Max Undercut=	0.0081		0.0086	
Diameter at Max Undercut=	4.7426	(10.77)	14.7118	(22.12)
* Finished Grind Diameter=	4.7426	(10.77)	14.7118	(22.13)
Roll, radians, (1 tooth load)=	0.627	(35.91)	0.541	(30.98)
Minimum Fillet Radius=	0.0773		0.0945	
Helical Factor, C(h)=	1.000		1.000	
Y Factor=	0.744		0.879	
Load Sharing Ratio, m(N)=	1.000		1.000	
MODIFIED Stress Corr Fact, K(f)=	1.649		1.720	
J-Factor=	0.451		0.511	
I Factor=		0.125		
Max Specific Sliding Ratio=	1.08	(15.91)	0.75	(24.19)

Steel Gears, Finish Ground
Case Carburized

Universal Technical Systems, Inc, Rockford, Ill 61101 (Program #500)
Date: 1-3-89
Job : L.S. Train-MLRS

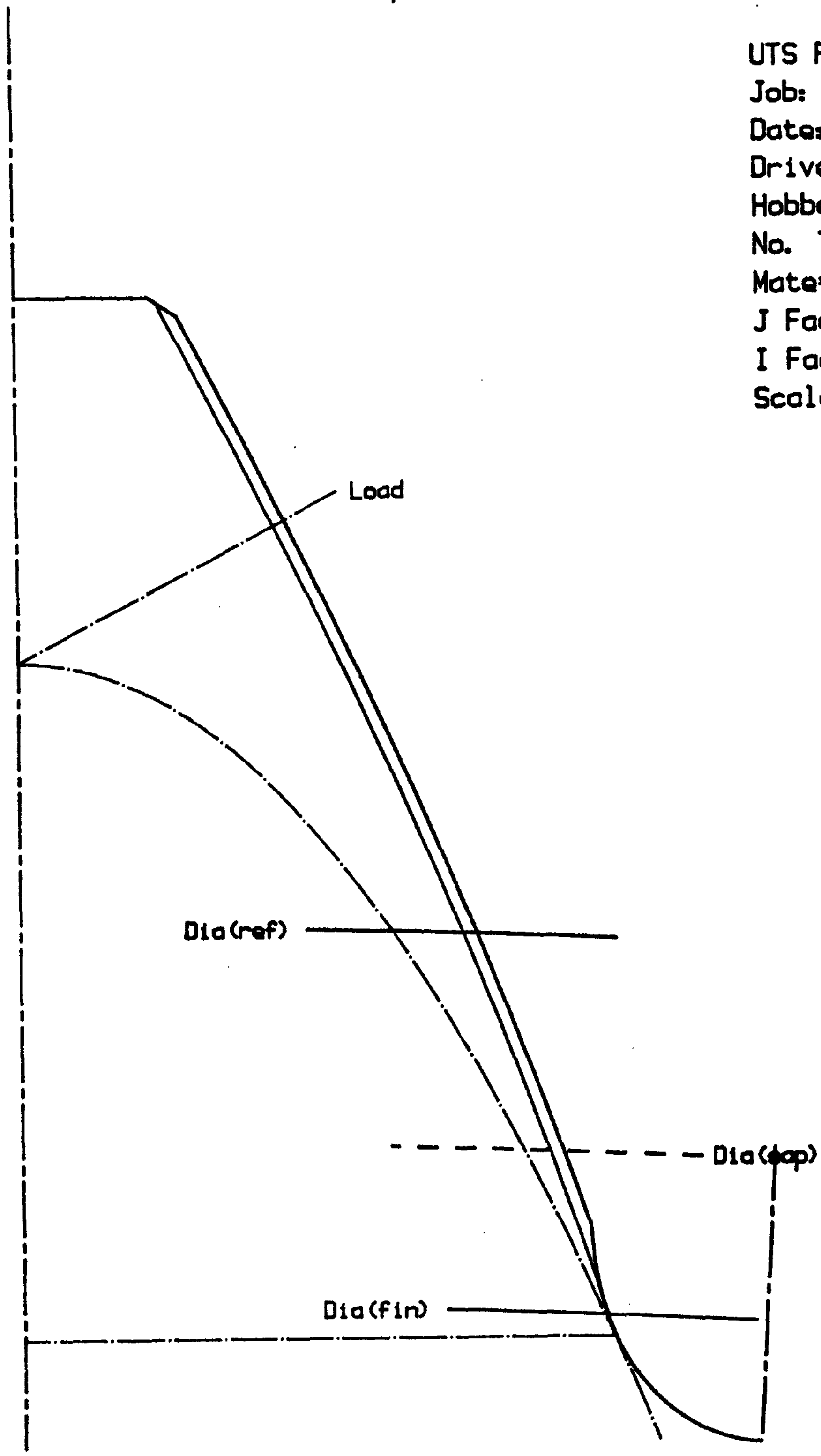
TACOM
Spur Gear Analysis

UTS Program #500
Job: L.S. Train-MLRS
Date: 1-3-89
Driver
Hobbed-Ground
No. Teeth= 18
Mate= 53
J Factor= 0.451
I Factor= 0.125
Scale= 11

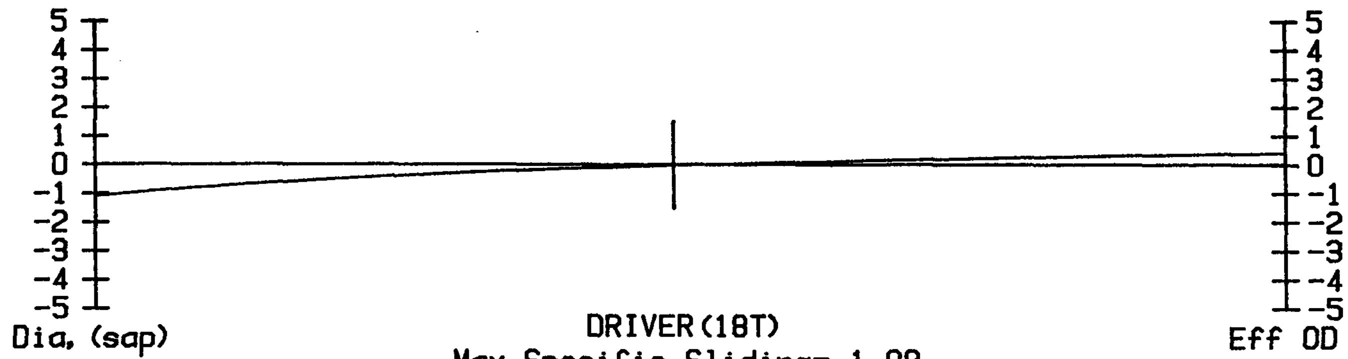


TACOM
Spur Gear Analysis

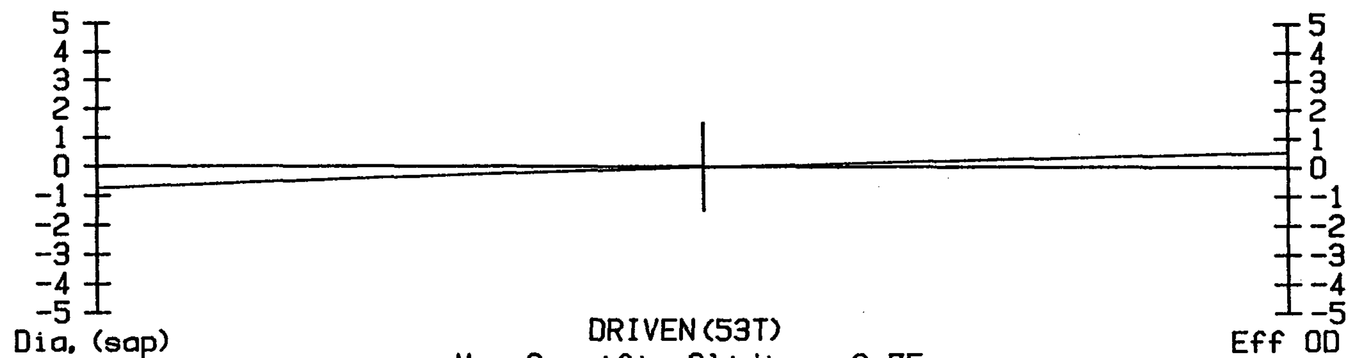
UTS Program #500
Job: L.S. Train-MLRS
Date: 1-3-89
Driven
Hobbed-Ground
No. Teeth= 53
Mate= 18
J Factor= 0.511
I Factor= 0.125
Scale= 11



SPECIFIC SLIDING RATIOS (Slide/Roll)



DRIVER (18T)
Max Specific Sliding= 1.08
Max Slope= 13.7



DRIVEN (53T)
Max Specific Sliding= 0.75
Max Slope= 4.1

UTS #500 Job: L.S. Train-MLRS Date: 1-3-89

TACOM

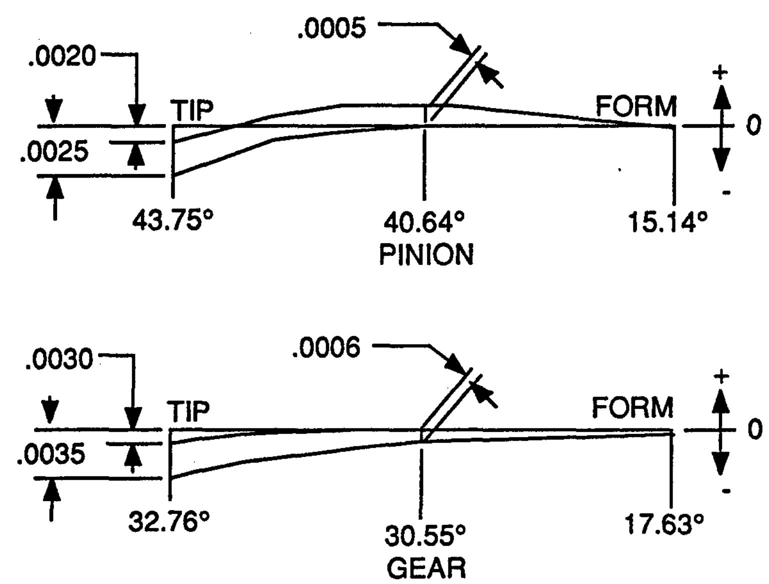
Appendix H

Profile Diagram--H.S.

Recommended Tip Relief:

H.S. 18 Pinion: Profile tolerance = .0006*(40.64–15.14)/(43.75–15.14) = .0005

H.S. 34 Gear: Profile tolerance = .0007*(30.55–17.63)/(32.76–17.63) = .0006



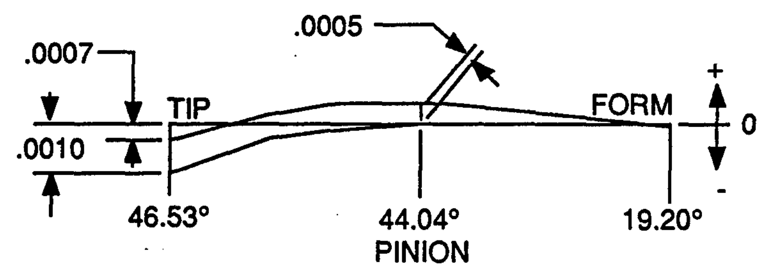
Appendix I

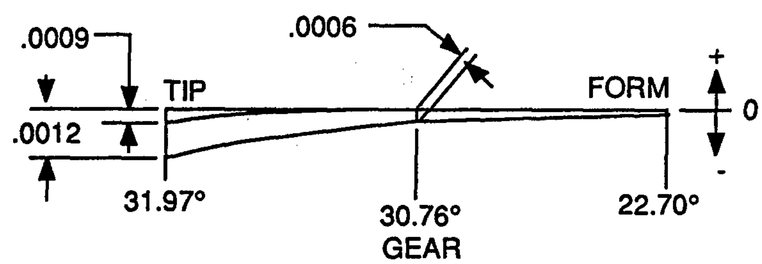
Profile Diagram--L.S.

Recommended Tip Relief:

L.S. 18 Pinion: Profile tolerance = .0006*(44.04–19.20)/(46.53–19.20) = .0005

L.S. 53 Gear: Profile tolerance = .0007*(30.76–22.70)/(31.97–22.70) = .0006





Appendix J

H.S. Train–OPT

TACOM
Gear Analysis

* Denotes Input Data

* Normal Diam Pitch=	3.5000	Opr Diam Pitch=	3.4986
* Normal Pressure Angle=	25.0000	Opr Pressure Angle=	25.0479
* Helix Angle=	0.0000		
Trans Diam Pitch=	3.5000	Line of Action=	1.1499
Trans Pressure Angle=	25.0000	% Approach Action=	39.97
		% Recess Action=	60.03
Opr Center Distance=	7.4315	Profile C.R.=	1.4135
* Face Width=	1.5820		
Basic Backlash=	0.0000		
Total Operating BL=	0.0130		

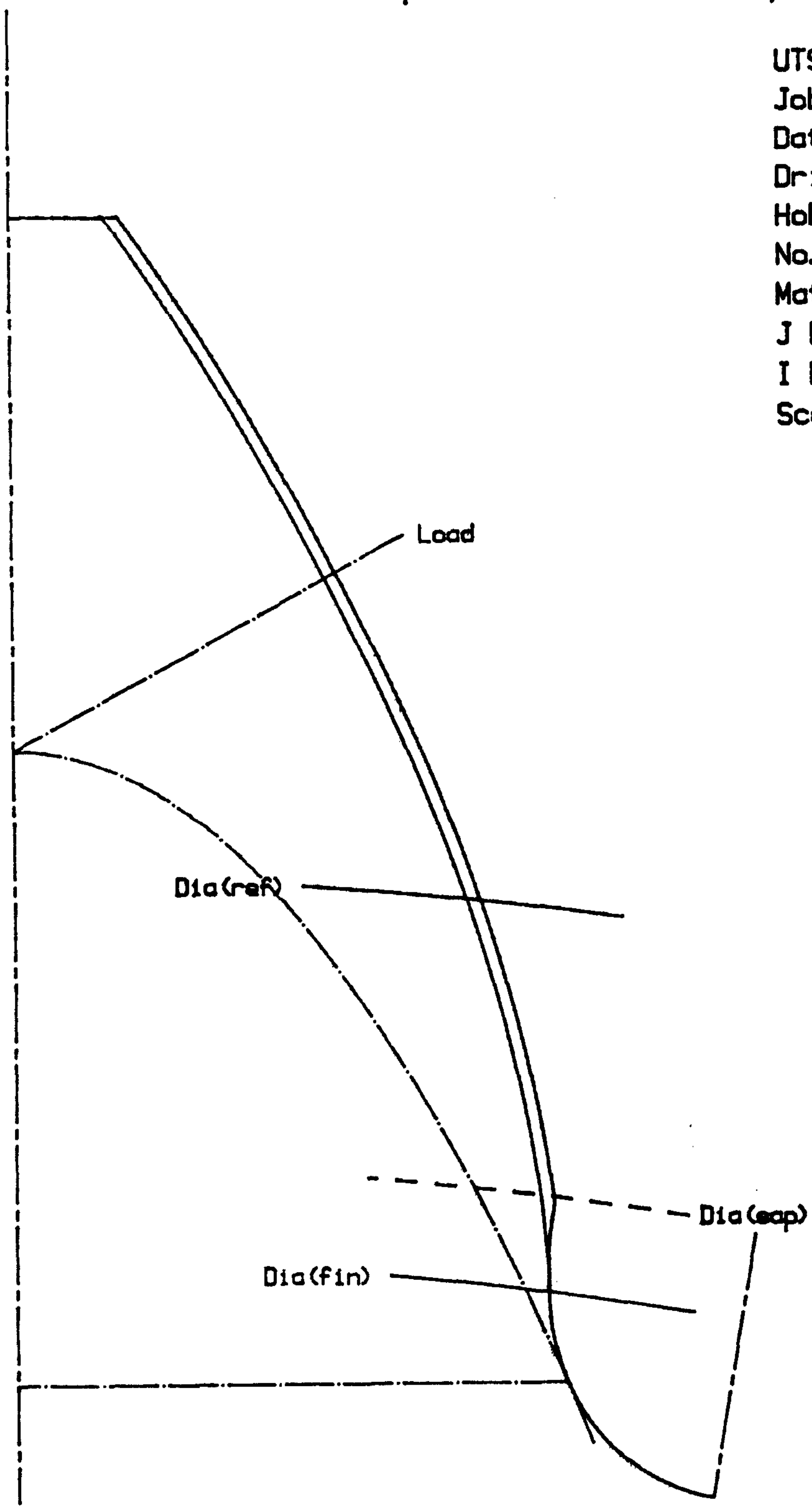
	DRIVER	(Deg Roll)	DRIVEN	(Deg Roll)
* Number of Teeth=	18		34	
* Outside Diameter=	5.8643	(43.75)	10.1415	(32.76)
* Cut Transverse Backlash=	0.0065		0.0065	
* Delta Addendum=	0.0750		-0.0721	
* Total Normal Finish Stock=	0.0150		0.0150	
HOB FORM DATA	NON-TOPPING		NON-TOPPING	
*Driver Off Lead: Hob Press Ang=	17.5000		25.0000	
* Hob Tip to Ref Line=	0.3773		0.3615	
* Hob Tooth Thickness at Ref=	0.3772		0.4188	
* Both: Full Rad-Hob Tip Radius=	0.1063		0.0778	
* Hob Protuberance=	0.0080		0.0080	
Hob SAP from Ref Line=	0.1182		0.2913	
Hob Space Width at Hob SAP=	0.4012		0.2071	
Normal Tooth Thickness at OD=	0.1065		0.1646	
Normal Tooth Thickness, (Hobbed)=	0.5272		0.3901	
Normal Tooth Thickness, (Ground)=	0.5122		0.3751	
Dia @ Mid-point of Line of Action=	5.2467	(29.61)	9.6227	(25.27)
Pitch Diameter, (Ref)=	5.1429	(26.72)	9.7143	(26.72)
Operating Pitch Diameter=	5.1449	(26.78)	9.7181	(26.78)
Base Diameter=	4.6610		8.8041	
Dia, (Start of Active Profile)=	4.8281	(15.48)	9.2188	(17.79)
Form Diameter=	4.8210	(15.14)	9.2117	(17.63)
Root Diameter=	4.5244		8.8010	
Root Clearance=	0.0985		0.0989	
Max Undercut=	0.0081		0.0082	
Diameter at Max Undercut=	4.7222	(9.31)	9.0645	(14.04)
* Finished Grind Diameter=	4.7222	(9.31)	9.0645	(14.04)
Roll, radians, (1 tooth load)=	0.619	(35.48)	0.495	(28.38)
Minimum Fillet Radius=	0.1085		0.1052	
Helical Factor, C(h)=	1.000		1.000	
Y Factor=	0.792		0.665	
Load Sharing Ratio, m(N)=	1.000		1.000	
MODIFIED Stress Corr Fact, K(f)=	1.747		1.582	
J-Factor=	0.454		0.420	
I Factor=		0.118		
Max Specific Sliding Ratio=	1.12	(15.48)	1.46	(17.79)

Steel Gears, Finish Ground
Case Carburized

Universal Technical Systems, Inc, Rockford, Ill 61101 (Program #500)
Date: 2-10-89
Job : HS Train - OPT

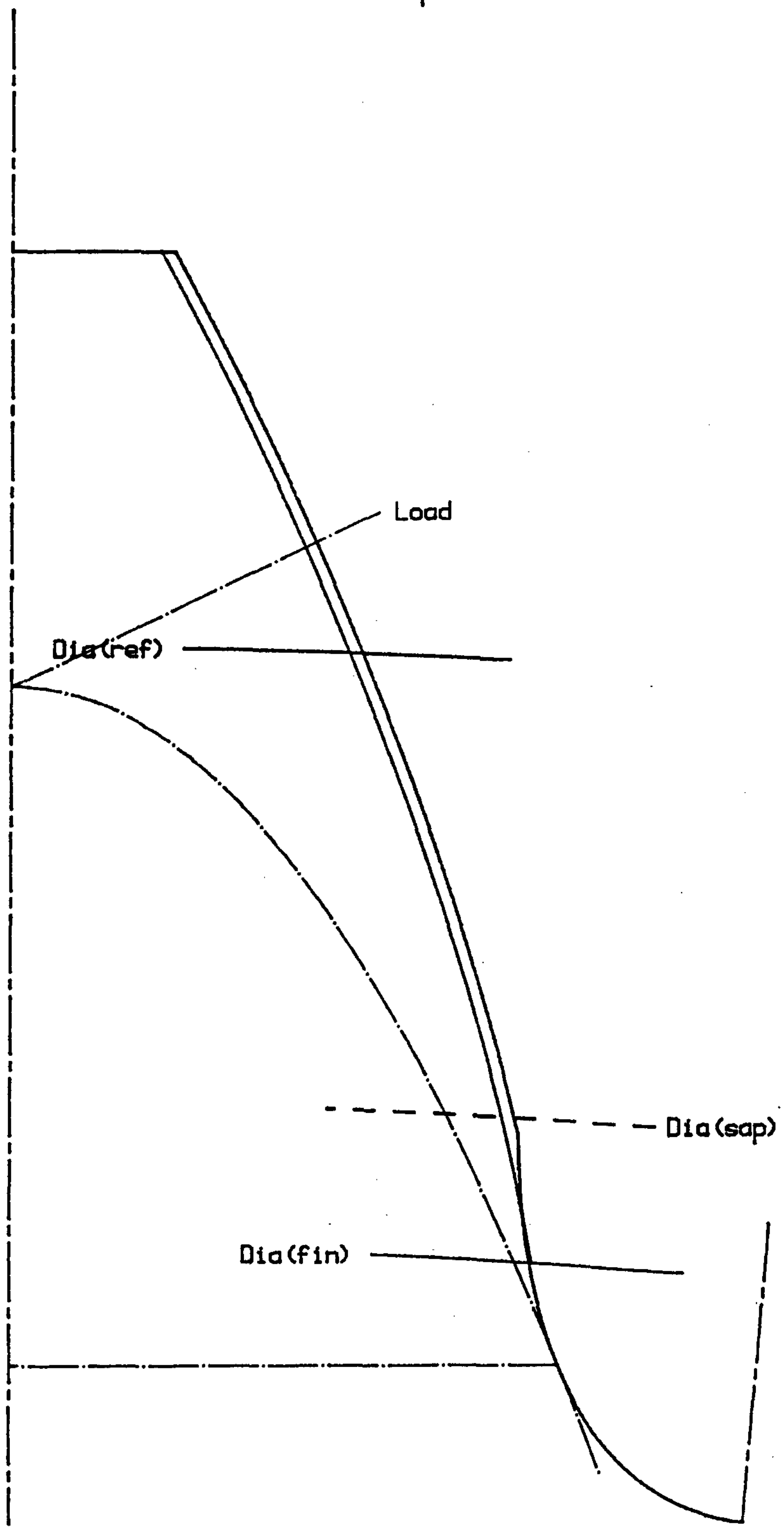
TACOM
Spur Gear Analysis

UTS Program #500
Job: HS Train - OPT
Date: 2-10-89
Driver
Hobbed-Ground
No. Teeth= 18
Mate= 34
J Factor= 0.454
I Factor= 0.118
Scale= 11

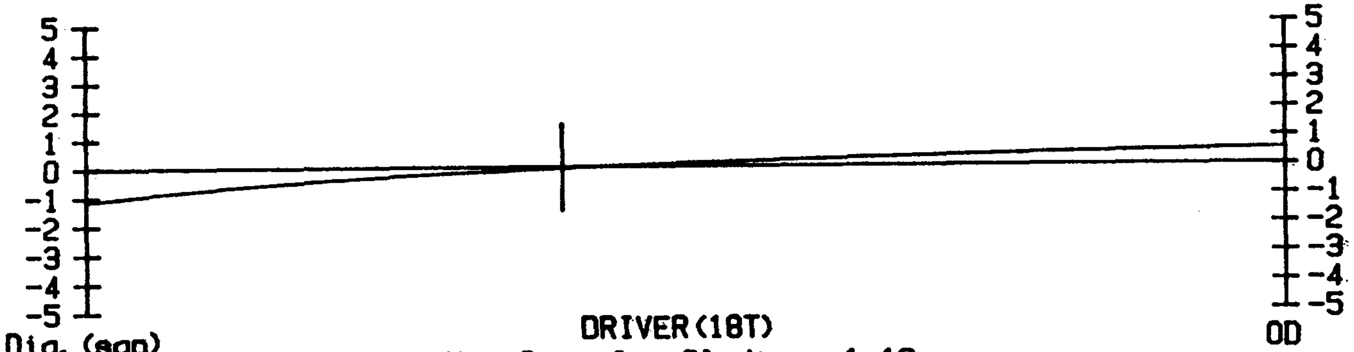


TACOM
Spur Gear Analysis

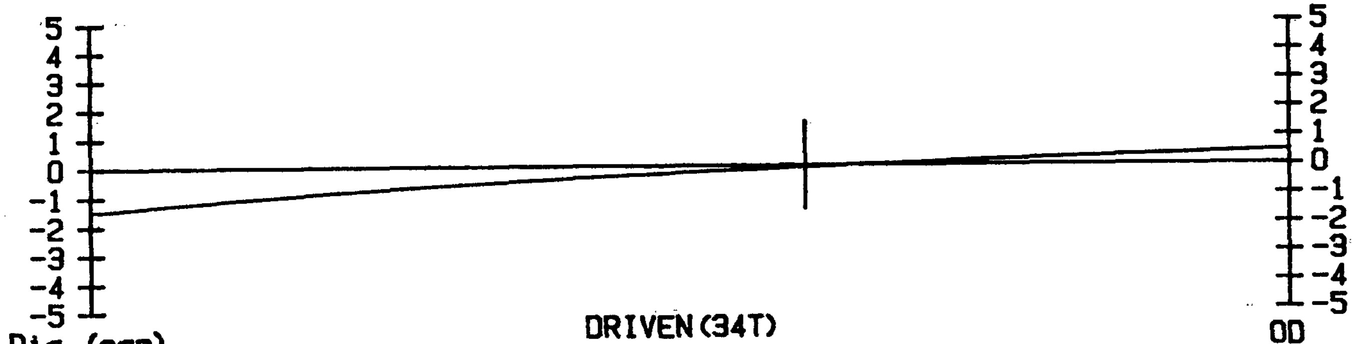
UTS Program #500
Job: HS Train - OPT
Date: 2-10-89
Driven
Hobbed-Ground
No. Teeth= 34
Mate= 18
J Factor= 0.42
I Factor= 0.118
Scale= 11



SPECIFIC SLIDING RATIOS (Slide/Roll)



DRIVER(18T)
Max Specific Sliding= 1.12
Max Slope= 15.8



DRIVEN(34T)
Max Specific Sliding= 1.46
Max Slope= 10.6

UTS #500 Job: HS Train - OPT

Date: 2-10-89

TACOM

Appendix K

L.S. Train–OPT

TACOM
Gear Analysis

* Denotes Input Data			
* Normal Diam Pitch=	3.5000	Opr Diam Pitch=	3.4505
* Normal Pressure Angle=	25.0000	Opr Pressure Angle=	26.6859
* Helix Angle=	0.0000		
Trans Diam Pitch=	3.5000	Line of Action=	1.1004
Trans Pressure Angle=	25.0000	% Approach Action=	34.47
		% Recess Action=	65.53
Opr Center Distance=	10.2885	Profile C.R.=	1.3526
* Face Width=	2.8800		
Basic Backlash=	0.0043		
Total Operating BL=	0.0171		

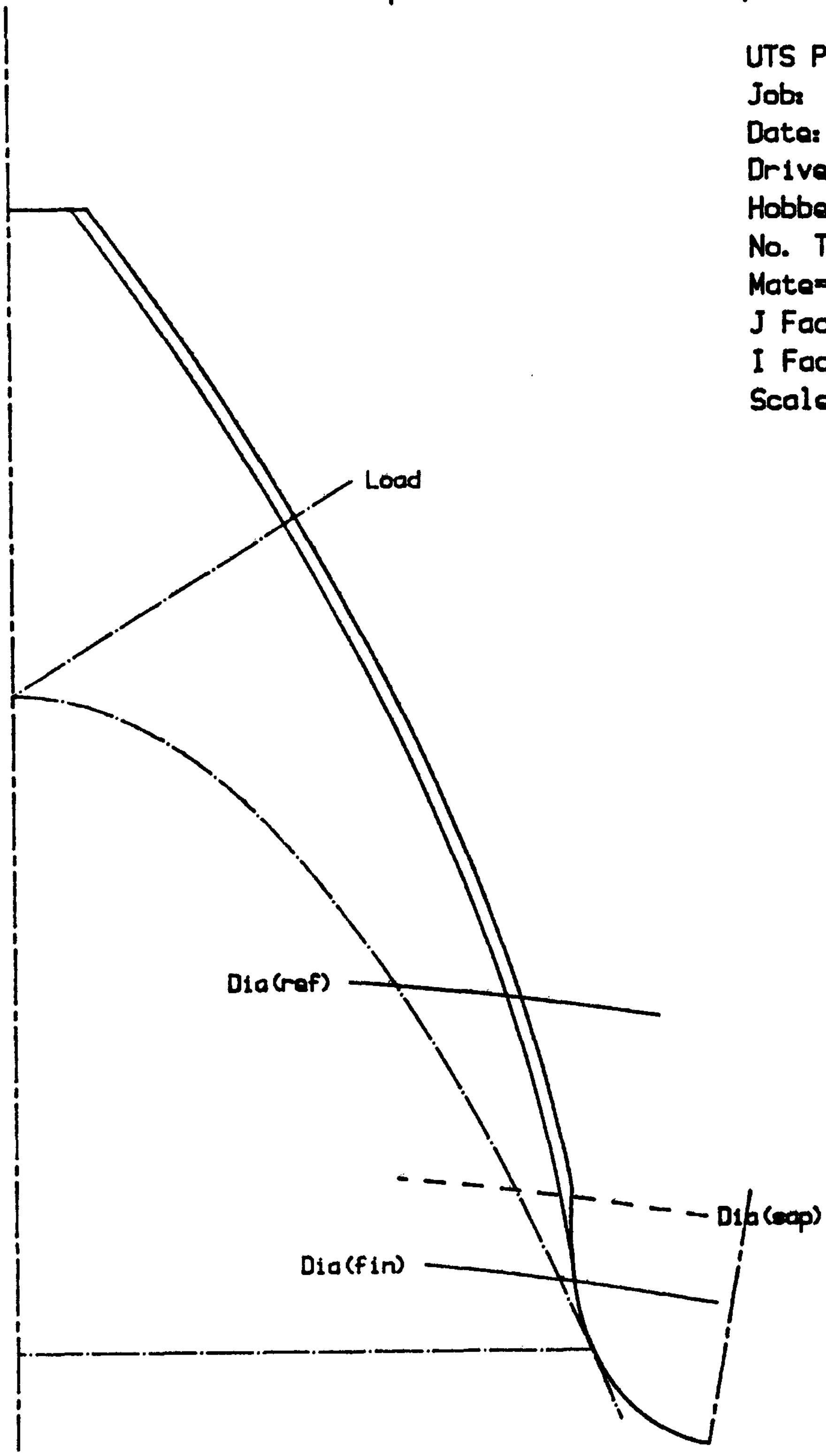
	DRIVER	(Deg Roll)	DRIVEN	(Deg Roll)
* Number of Teeth=	18		53	
* Outside Diameter=	6.0043	(46.53)	15.7155	(31.97)
* Cut Transverse Backlash=	0.0050		0.0078	
* Delta Addendum=	0.1450		0.0006	
* Total Normal Finish Stock=	0.0150		0.0150	
HOB FORM DATA	NON-TOPPING		NON-TOPPING	
* Hob Pressure Angle=	25.0000		25.0000	
* Hob Tip to Ref Line=	0.3615		0.3615	
* Hob Tooth Thickness at Ref=	0.4188		0.4188	
* Both: Full Rad-Hob Tip Radius=	0.0778		0.0778	
* Hob Protuberance=	0.0080		0.0080	
Hob SAP from Ref Line=	0.2169		0.2894	
Hob Space Width at Hob SAP=	0.2765		0.2089	
Driver:((0.3/NDP) Normal TT at OD=	0.0755		0.1592	
Normal Tooth Thickness,(Hobbed)=	0.5941		0.4567	
Normal Tooth Thickness,(Ground)=	0.5791		0.4417	
Dia @ Mid-point of Line of Action=	5.3789	(33.00)	15.2098	(27.37)
Pitch Diameter, (Ref)=	5.1429	(26.72)	15.1429	(26.72)
Operating Pitch Diameter=	5.2167	(28.80)	15.3603	(28.80)
Base Diameter=	4.6610		13.7241	
Dia, (Start of Active Profile)=	4.9229	(19.47)	14.7688	(22.78)
Form Diameter=	4.9158	(19.20)	14.7617	(22.70)
Root Diameter=	4.6671		14.3724	
Root Clearance=	0.0972		0.1001	
Max Undercut=	0.0083		0.0085	
Diameter at Max Undercut=	4.8284	(15.49)	14.5357	(19.99)
* Finished Grind Diameter=	4.8284	(15.49)	14.5357	(19.99)
Roll, radians, (1 tooth load)=	0.689	(39.47)	0.516	(29.57)
Minimum Fillet Radius=	0.0872		0.0898	
Helical Factor, C(h)=	1.000		1.000	
Y Factor=	0.920		0.775	
Load Sharing Ratio, m(N)=	1.000		1.000	
MODIFIED Stress Corr Fact, K(f)=	1.734		1.623	
J-Factor=	0.530		0.478	
I Factor=		0.142		
Max Specific Sliding Ratio=	0.64	(19.47)	1.04	(22.78)

Steel Gears, Finish Ground
Case Carburized

Universal Technical Systems, Inc, Rockford, Ill 61101 (Program #500)
Date: 2-11-89
Job : LS Train - OPT

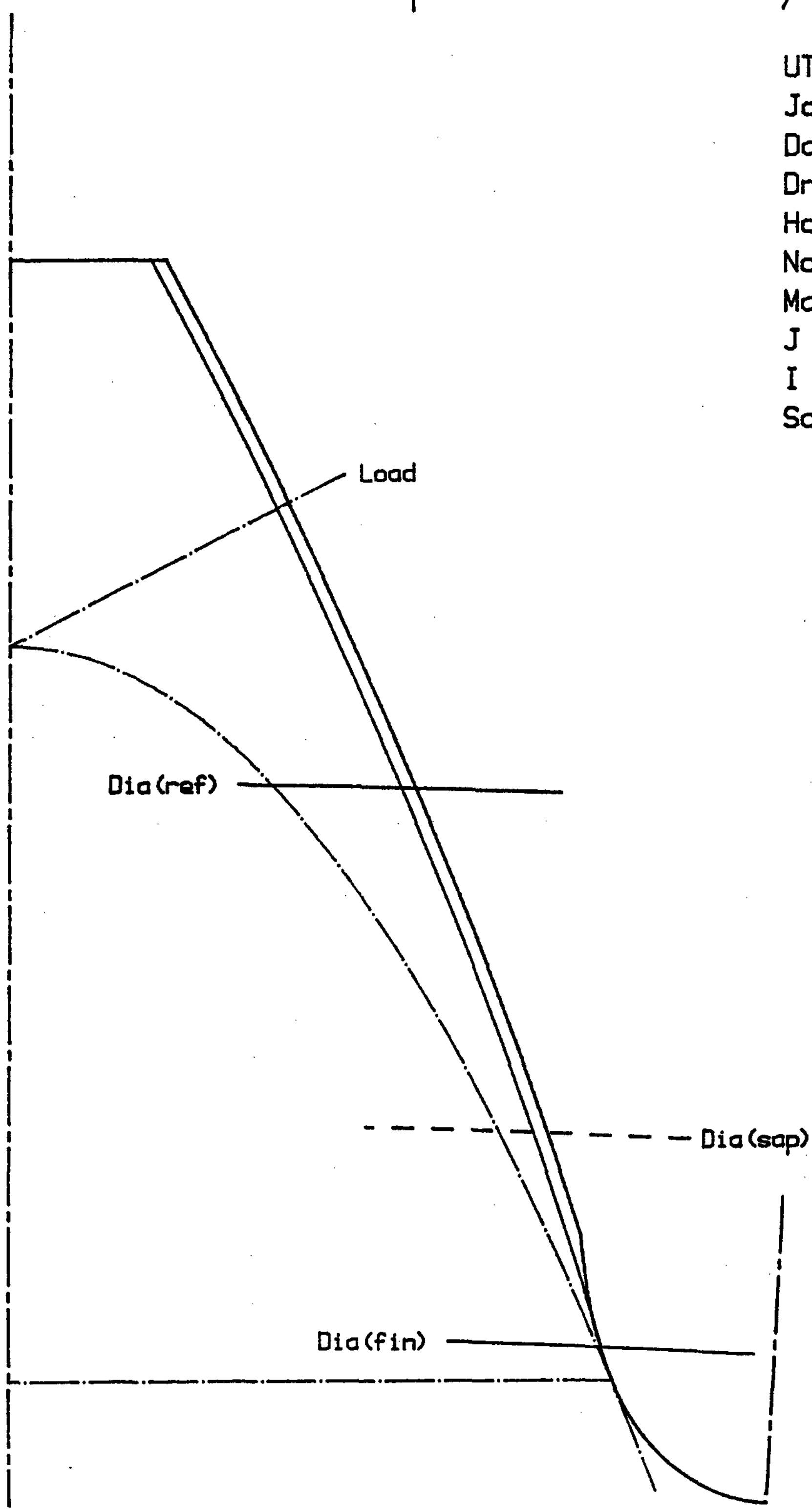
TACOM
Spur Gear Analysis

UTS Program #500
Job: LS Train - OPT
Date: 2-11-89
Driver
Hobbed-Ground
No. Teeth= 18
Mate= 53
J Factor= 0.53
I Factor= 0.142
Scale= 11

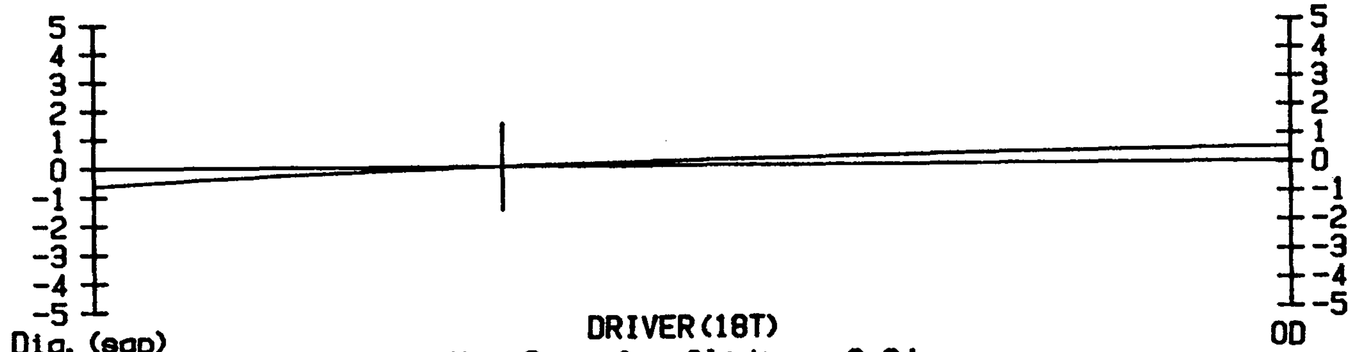


TACOM
Spur Gear Analysis

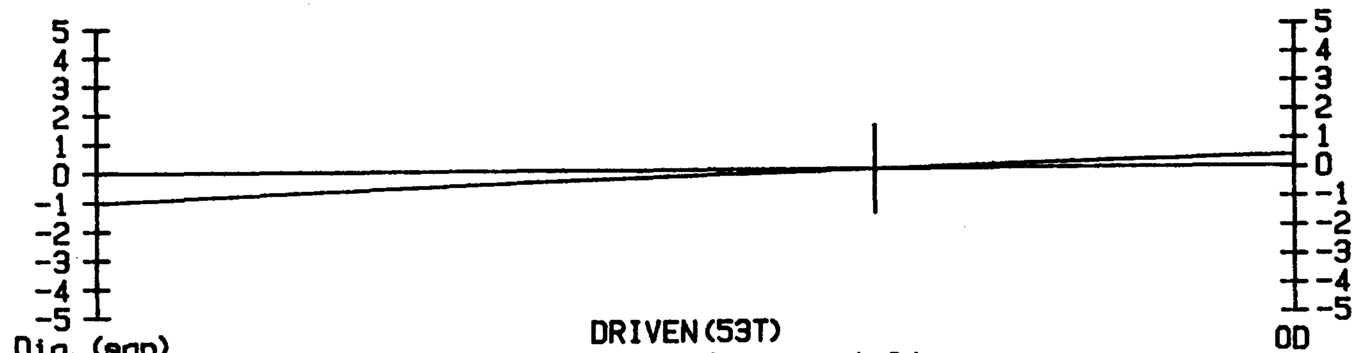
UTS Program #500
Job: LS Train - OPT
Date: 2-11-89
Driven
Hobbed-Ground
No. Teeth= 53
Mate= 18
J Factor= 0.478
I Factor= 0.142
Scale= 11



SPECIFIC SLIDING RATIOS (Slide/Roll)



DRIVER(18T)
Max Specific Sliding= 0.64
Max Slope= 7.7



DRIVEN(53T)
Max Specific Sliding= 1.04
Max Slope= 4.9

UTS #500 Job: LS Train - ØPT

Date: 2-11-89

TACOM

Appendix L

Sample Printout, Tooth Plots, and Specific Sliding Plots

TACOM
Gear Analysis

* Denotes Input Data

* Normal Diam Pitch=	3.5000	Opr Diam Pitch=	3.4667
* Normal Pressure Angle=	25.0000	Opr Pressure Angle=	26.1453
* Helix Angle=	0.0000		
Trans Diam Pitch=	3.5000	Line of Action=	1.1311
Trans Pressure Angle=	25.0000	% Approach Action=	47.81
		% Recess Action=	52.19
Opr Center Distance=	7.5000	Profile C.R.=	1.3904
* Face Width=	1.6250		
Basic Backlash=	0.0014		
Total Operating BL=	0.0121		

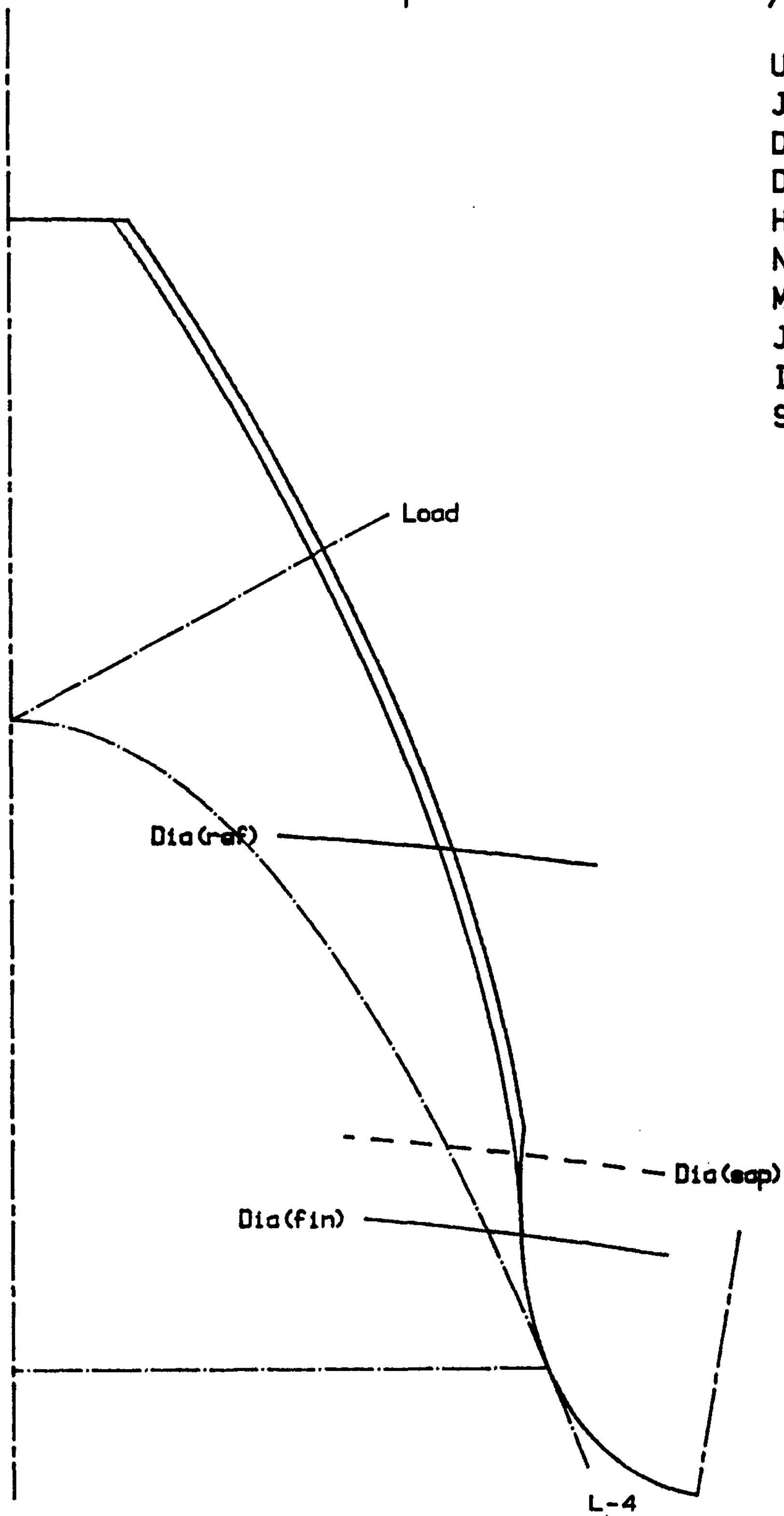
	DRIVER	(Deg Roll)	DRIVEN	(Deg Roll)
* Number of Teeth=	18		34	
* Outside Diameter=	5.8100	(42.64)	10.3300	(35.16)
* Total Normal Finish Stock=	0.0150		0.0150	
HOB FORM DATA	NON-TOPPING		NON-TOPPING	
* Hob Pressure Angle=	25.0000		25.0000	
* Hob Tip to Ref Line=	0.3785		0.3800	
* Hob Tooth Thickness at Ref=	0.4338		0.4338	
* Both: Full Rad-Hob Tip Radius=	0.0772		0.0761	
* Hob Protuberance=	0.0080		0.0080	
Hob SAP from Ref Line=	0.2315		0.2564	
Hob Space Width at Hob SAP=	0.2479		0.2247	
Normal Tooth Thickness at OD=	0.1189		0.1510	
Normal Tooth Thickness, (Hobbed)=	0.5031		0.4805	
*Normal Tooth Thickness, (Ground)=	0.4881		0.4655	
Dia @ Mid-point of Line of Action=	5.2143	(28.73)	9.7859	(27.80)
Pitch Diameter, (Ref)=	5.1429	(26.72)	9.7143	(26.72)
Operating Pitch Diameter=	5.1923	(28.13)	9.8077	(28.13)
Base Diameter=	4.6610		8.8041	
Dia, (Start of Active Profile)=	4.8146	(14.83)	9.3477	(20.44)
Form Diameter=	4.8075	(14.48)	9.3406	(20.30)
Root Diameter=	4.4701		8.9901	
Root Clearance=	0.0999		0.0999	
Max Undercut=	0.0081		0.0083	
Diameter at Max Undercut=	4.7279	(9.74)	9.1800	(16.92)
* Finished Grind Diameter=	4.7279	(9.74)	9.1800	(16.92)
Roll, radians, (1 tooth load)=	0.608	(34.83)	0.542	(31.03)
Minimum Fillet Radius=	0.1009		0.0920	
Helical Factor, C(h)=	1.000		1.000	
Y Factor=	0.725		0.795	
Load Sharing Ratio, m(N)=	1.000		1.000	
MODIFIED Stress Corr Fact, K(f)=	1.633		1.666	
J-Factor=	0.444		0.477	
I Factor=		0.115		
Max Specific Sliding Ratio=	1.37	(14.83)	1.09	(20.44)

Steel Gears, Finish Ground
Case Carburized

Universal Technical Systems, Inc, Rockford, Ill 61101 (Program #500)
Date: 0/0/00
Job : Sample

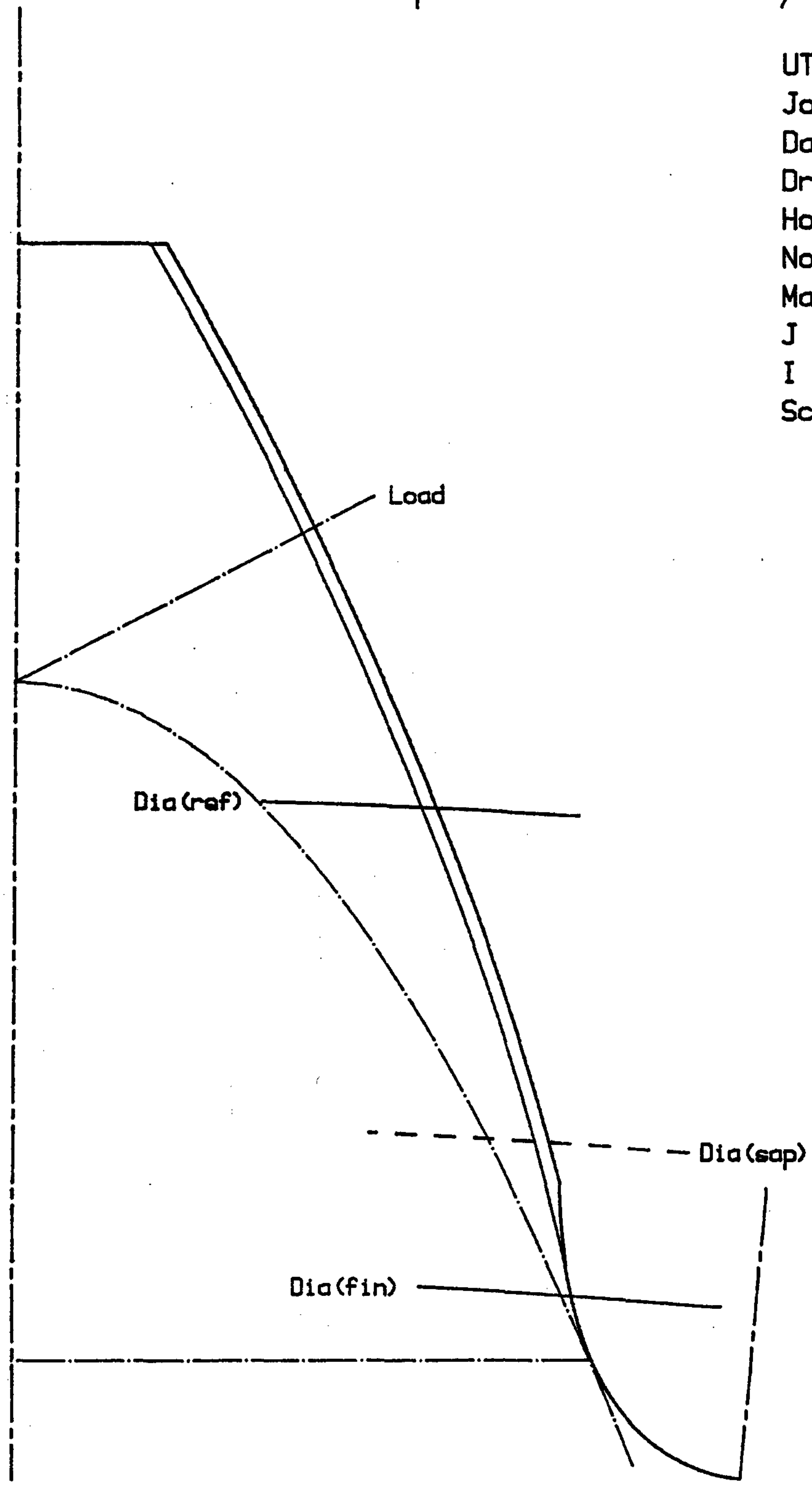
TACOM
Spur Gear Analysis

UTS Program #500
Job: Sample
Date: 0/0/00
Driver
Hobbed-Ground
No. Teeth= 18
Mate= 34
J Factor= 0.444
I Factor= 0.115
Scale= 11

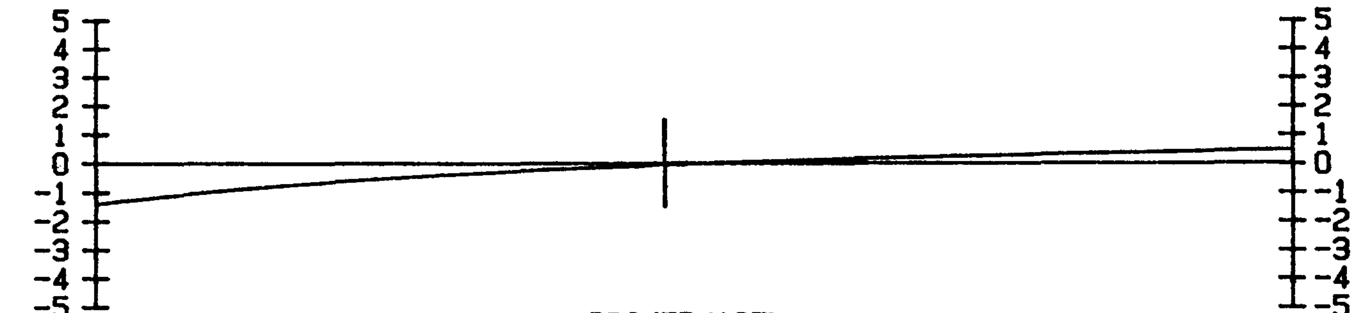


TACOM
Spur Gear Analysis

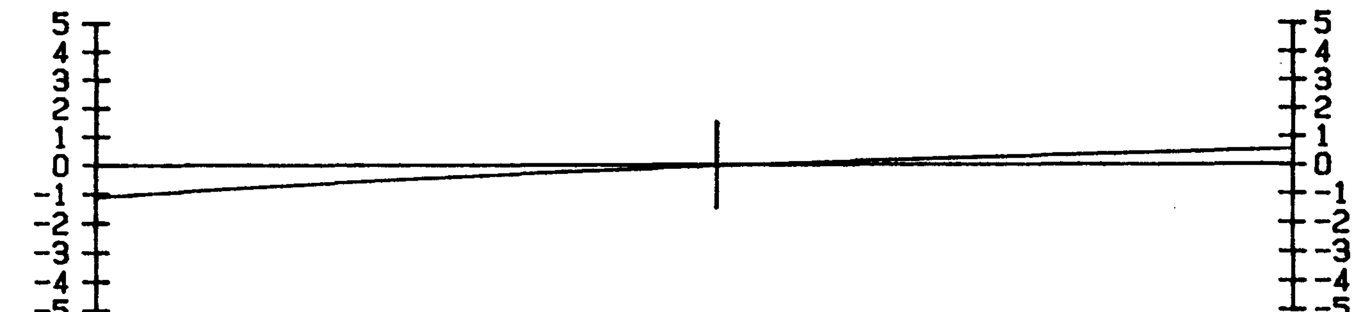
UTS Program #500
Job: Sample
Date: 0/0/00
Driven
Hobbed-Ground
No. Teeth= 34
Mate= 18
J Factor= 0.477
I Factor= 0.115
Scale= 11



SPECIFIC SLIDING RATIOS (Slide/Roll)



DRIVER(18T)
Max Specific Sliding= 1.37
Max Slope= 18.7



DRIVEN(34T)
Max Specific Sliding= 1.09
Max Slope= 7.5

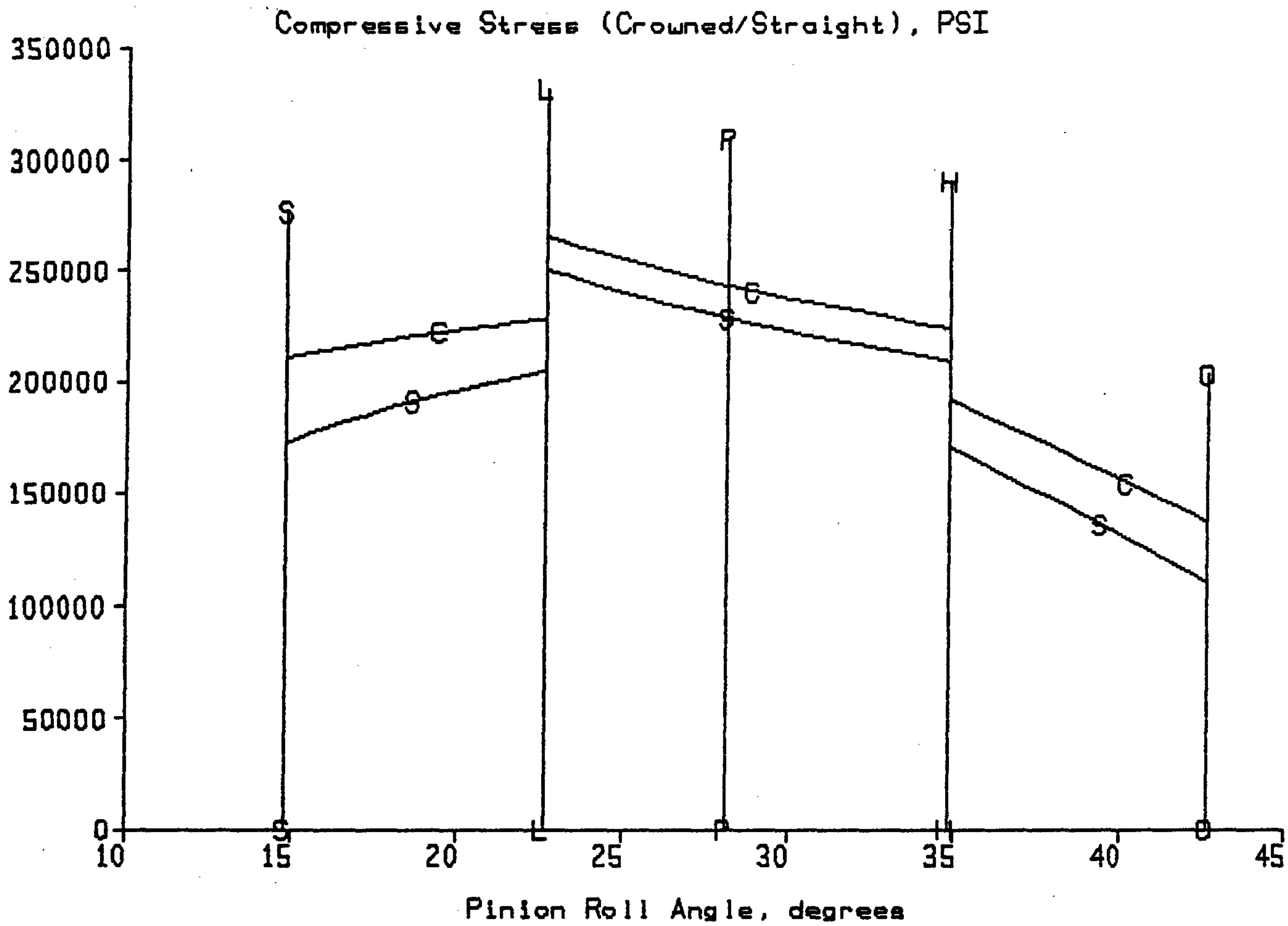
UTS #500 Job: Sample

Date: 0/0/00

TACOM

Appendix M

Compressive Stress (Crowned/Straight)



Compressive Stress (Crowned/Straight), PSI
350000
300000
250000
200000
150000
100000
50000
0
10
15
20
25
30
35
40
45
Pinion Roll Angle, degrees
S
L
P
H
O
C
S

Appendix N

#540 Printout, Pitting Resistance and Bending Strength

TACOM

Gear Rating Program Using AGMA 218.01

Pitting Resistance and Bending Strength of

Spur and Helical Involute Gear Teeth (#540)

==

Job ID: Sample Date: 0/0/00

Pinion Teeth= 18
Gear Teeth= 34
AGMA Q Class= 11
Normal Diametral Pitch, Nominal= 3.5
Diametral Pitch, Plane of Rotation= 3.5
Face Width= 1.625 inches
Operating Pitch Diameter of pinion= 5.1923 inches
Helix Angle at Standard PD= 0 degrees
Operating Trans Press Angle= 26.1453 degrees
AGMA I-factor= 0.115
AGMA Pinion J-factor= 0.444
AGMA Gear J-factor= 0.477

Pinion Material is Steel
Carburized & Case Hardened
Hardness= 60 Rc/C
Recommended Case Depth= 0.0408 to 0.0647
(From AGMA 218.01, Fig. 11)
Pinion Normal Tooth Thickness at OD= 0.1189

Gear Material is Steel
Carburized & Case Hardened
Hardness= 60 Rc/C
Recommended Case Depth= 0.0408 to 0.0647
(From AGMA 218.01, Fig. 11)
Gear Normal Tooth Thickness at OD= 0.151

ANALYTICAL METHOD FOR FACE DISTRIBUTION FACTOR
Lead Mismatch, et= 0.001 inches
Tooth Stiffness Constant, G= 2.00E+6 psi
Temperature Factor, C(T)=K(T)= 1

SPUR GEARS
Size Factor, Cs=Ks= 1
Elastic Coefficient, Cp= 2291
Surface Condition Factor, Cf= 1
GEAR Hardness Ratio Factor, C(H)= 1
Transverse Load Distribution Factor, Cmt= 1

```
PINION RPM= 280  (Miner's Rule Cond #1)
 Pitch Line Velocity, vt= 380.6 ft/min
PINION:
 Pitting:
  Dynamic Factor, Cv= 0.953
    Face Load Distribution Factor, Cmf= 1.568
    Cmf (= 2 indicates load across total face
    Load Distribution Factor, Cm= 1.568
    Horsepower= 53.33
    Pinion Torque= 12000 lb-in
    Tangential Load= 4622 lb
     Separating Force= 2268 lb
    Compressive Stress= 202960 psi
    Life= 5.156E+8 to 8.637E+10 cycles
    Life= 30691 hours
      To More Than 100,000 hours
    NOTE: High Cycle Curve Used
            (See AGMA 218.01, Fig. 20)
    Number of same flank pinion contacts
       per revolution= 1
    Application Factor, Ca= 1
    Reliability Factor, CR= 0.9
     ((1 Failure in 20)
    Life Factor, CL= 0.913 to 0.812
    MIN REQUIRED EFFECTIVE CASE DEPTH
       FOR THIS LOAD= 0.0474
    MAX SUGGESTED EFFECTIVE CASE DEPTH= 0.0666
  Strength:
   Dynamic Factor, Kv= 0.953
    Face Load Distribution Factor, Cmf= 1.568
    Cmf (= 2 indicates load across total face
    Load Distribution Factor, Km= 1.568
    Horsepower= 53.33
    Pinion Torque= 12000 lb-in
    Tangential Load= 4622 lb
     Separating Force= 2268 lb
    Pinion Bending Stress= 36882 psi
    Life= 5.588E+19 to 4.278E+25 cycles
    Life Is More Than 100,000 hours
    NOTE: High Cycle Curve Used
            (See AGMA 218.01, Fig. 21)
    Number of same flank pinion contacts
       per revolution= 1
    Application Factor, Ka= 1
    Reliability Factor, KR= 0.9
     ((1 Failure in 20)
    Life Factor, KL= 0.604 to 0.474
```

GEAR: (Gear RPM= 148)
Pitting:
Dynamic Factor, Cv= 0.953
Face Load Distribution Factor, Cmf= 1.568
Cmf <= 2 indicates load across total face
Load Distribution Factor, Cm= 1.568
Horsepower= 53.33
Pinion Torque= 12000 lb-in
Tangential Load= 4622 lb
Separating Force= 2268 lb
Compressive Stress= 202960 psi
Life= 5.156E+8 to 8.637E+10 cycles
Life= 57972 hours
To More Than 100,000 hours
NOTE: High Cycle Curve Used
(See AGMA 218.01, Fig. 20)
Number of same flank gear contacts
per revolution= 1
Application Factor, Ca= 1
Reliability Factor, CR= 0.9
((1 Failure in 20)
Life Factor, CL= 0.913 to 0.812
MIN REQUIRED EFFECTIVE CASE DEPTH
FOR THIS LOAD= 0.0474
MAX SUGGESTED EFFECTIVE CASE DEPTH= 0.0846
Strength:
Dynamic Factor, Kv= 0.953
Face Load Distribution Factor, Cmf= 1.568
Cmf <= 2 indicates load across total face
Load Distribution Factor, Cm= 1.568
Horsepower= 53.33
Pinion Torque= 12000 lb-in
Tangential Load= 4622 lb
Separating Force= 2268 lb
Gear Bending Stress= 34309 psi
Life= 3.246E+21 to 2.485E+27 cycles
Life Is More Than 100,000 hours
NOTE: High Cycle Curve Used
(See AGMA 218.01, Fig. 21)
Number of same flank gear contacts
per revolution= 1
Application Factor, Ka= 1
Reliability Factor, KR= 0.9
((1 Failure in 20)
Life Factor, KL= 0.561 to 0.441

PINION RPM= 390 (Miner's Rule Cond #2)
Pitch Line Velocity, vt= 530.1 ft/min
PINION:
Pitting:
Dynamic Factor, Cv= 0.946
Face Load Distribution Factor, Cmf= 1.336
Cmf <= 2 indicates load across total face
Load Distribution Factor, Cm= 1.336
Horsepower= 139
Pinion Torque= 22500 lb-in
Tangential Load= 8666 lb
Separating Force= 4254 lb
Compressive Stress= 257530 psi
Life= 718300 to 5.885E+6 cycles
Life= 30 hours, 42 min
to 251 hours
Number of same flank pinion contacts
per revolution= 1
Application Factor, Ca= 1
Reliability Factor, CR= 0.9
(<1 Failure in 20)
Life Factor, CL= 1.159 to 1.03
CAUTION: Compressive Stress is Too High for
Recommended Case Depth: See AGMA 218.01
MIN REQUIRED EFFECTIVE CASE DEPTH
FOR THIS LOAD= 0.0602
MAX SUGGESTED EFFECTIVE CASE DEPTH= 0.0666
Strength:
Dynamic Factor, Kv= 0.946
Face Load Distribution Factor, Cmf= 1.336
Cmf <= 2 indicates load across total face
Load Distribution Factor, Km= 1.336
Horsepower= 139
Pinion Torque= 22500 lb-in
Tangential Load= 8666 lb
Separating Force= 4254 lb
Pinion Bending Stress= 59380 psi
Life= 1.341E+8 to 1.027E+14 cycles
Life= 5731 hours
To More Than 100,000 hours
NOTE: High Cycle Curve Used
(See AGMA 218.01, Fig. 21)
Number of same flank pinion contacts
per revolution= 1
Application Factor, Ka= 1
Reliability Factor, KR= 0.9
(<1 Failure in 20)
Life Factor, KL= 0.972 to 0.763

GEAR: (Gear RPM= 206)
Pitting:
Dynamic Factor, Cv= 0.946
Face Load Distribution Factor, Cmf= 1.336
Cmf <= 2 indicates load across total face
Load Distribution Factor, Cm= 1.336
Horsepower= 139
Pinion Torque= 22500 lb-in
Tangential Load= 8666 lb
Separating Force= 4254 lb
Compressive Stress= 257530 psi
Life= 718300 to 5.885E+6 cycles
Life= 57 hours, 59 min
to 475 hours
Number of same flank gear contacts
per revolution= 1
Application Factor, Ca= 1
Reliability Factor, CR= 0.9
((1 Failure in 20)
Life Factor, CL= 1.159 to 1.03
CAUTION: Compressive Stress is Too High for
Recommended Case Depth: See AGMA 218.01
MIN REQUIRED EFFECTIVE CASE DEPTH
FOR THIS LOAD= 0.0602
MAX SUGGESTED EFFECTIVE CASE DEPTH= 0.0846
Strength:
Dynamic Factor, Kv= 0.946
Face Load Distribution Factor, Cmf= 1.336
Cmf <= 2 indicates load across total face
Load Distribution Factor, Cm= 1.336
Horsepower= 139
Pinion Torque= 22500 lb-in
Tangential Load= 8666 lb
Separating Force= 4254 lb
Gear Bending Stress= 55238 psi
Life= 7.791E+9 to 5.965E+15 cycles
Life Is More Than 100,000 hours
NOTE: High Cycle Curve Used
(See AGMA 218.01, Fig. 21)
Number of same flank gear contacts
per revolution= 1
Application Factor, Ka= 1
Reliability Factor, KR= 0.9
((1 Failure in 20)
Life Factor, KL= 0.904 to 0.71

```
PINION RPM= 1230  (Miner's Rule Cond #3)
 Pitch Line Velocity, vt= 1672 ft/min
PINION:
 Pitting:
  Dynamic Factor, Cv= 0.912
    Face Load Distribution Factor, Cmf= 1.656
    Cmf <= 2 indicates load across total face
    Load Distribution Factor, Cm= 1.656
    Horsepower= 181
    Pinion Torque= 9300 lb-in
    Tangential Load= 3582 lb
     Separating Force= 1758 lb
    Compressive Stress= 187690 psi
    Life= 1.546E+10 to 2.59E+12 cycles
    Life Is More Than 100,000 hours
    NOTE: High Cycle Curve Used
            (See AGMA 218.01, Fig. 20)
    Number of same flank pinion contacts
       per revolution= 1
    Application Factor, Ca= 1
    Reliability Factor, CR= 0.9
     ((1 Failure in 20)
    Life Factor, CL= 0.845 to 0.751
    MIN REQUIRED EFFECTIVE CASE DEPTH
       FOR THIS LOAD= 0.0439
    MAX SUGGESTED EFFECTIVE CASE DEPTH= 0.0666
  Strength:
   Dynamic Factor, Kv= 0.912
    Face Load Distribution Factor, Cmf= 1.656
    Cmf <= 2 indicates load across total face
    Load Distribution Factor, Km= 1.656
    Horsepower= 181
    Pinion Torque= 9300 lb-in
    Tangential Load= 3582 lb
     Separating Force= 1758 lb
    Pinion Bending Stress= 31541 psi
    Life= 3.665E+23 to 2.806E+29 cycles
    Life Is More Than 100,000 hours
    NOTE: High Cycle Curve Used
            (See AGMA 218.01, Fig. 21)
    Number of same flank pinion contacts
       per revolution= 1
    Application Factor, Ka= 1
    Reliability Factor, KR= 0.9
     ((1 Failure in 20)
    Life Factor, KL= 0.516 to 0.406
```

```
GEAR:  (Gear RPM= 651)
 Pitting:
  Dynamic Factor, Cv= 0.912
    Face Load Distribution Factor, Cmf= 1.656
    Cmf <= 2 indicates load across total face
    Load Distribution Factor, Cm= 1.656
    Horsepower= 181
    Pinion Torque= 9300 lb-in
    Tangential Load= 3582 lb
     Separating Force= 1758 lb
    Compressive Stress= 187690 psi
    Life= 1.546E+10 to 2.59E+12 cycles
    Life Is More Than 100,000 hours
    NOTE: High Cycle Curve Used
            (See AGMA 218.01, Fig. 20)
    Number of same flank gear contacts
       per revolution= 1
    Application Factor, Ca= 1
    Reliability Factor, CR= 0.9
     ((1 Failure in 20)
    Life Factor, CL= 0.845 to 0.751
    MIN REQUIRED EFFECTIVE CASE DEPTH
       FOR THIS LOAD= 0.0439
    MAX SUGGESTED EFFECTIVE CASE DEPTH= 0.0846
 Strength:
  Dynamic Factor, Kv= 0.912
   Face Load Distribution Factor, Cmf= 1.656
   Cmf <= 2 indicates load across total face
   Load Distribution Factor, Cm= 1.656
   Horsepower= 181
   Pinion Torque= 9300 lb-in
   Tangential Load= 3582 lb
    Separating Force= 1758 lb
   Gear Bending Stress= 29340 psi
   Life= 2.129E+25 to 1.63E+31 cycles
   Life Is More Than 100,000 hours
   NOTE: High Cycle Curve Used
           (See AGMA 218.01, Fig. 21)
   Number of same flank gear contacts
      per revolution= 1
   Application Factor, Ka= 1
   Reliability Factor, KR= 0.9
    ((1 Failure in 20)
   Life Factor, KL= 0.48 to 0.377
```

```
PINION RPM= 1440  (Miner's Rule Cond #4)
 Pitch Line Velocity, vt= 1957.5 ft/min
PINION:
 Pitting:
  Dynamic Factor, Cv= 0.907
    Face Load Distribution Factor, Cmf= 2.029
    Cmf > 2 INDICATES PARTIAL FACE LOADING
    Load Distribution Factor, Cm= 2.029
    Horsepower= 102
    Pinion Torque= 4500 lb-in
    Tangential Load= 1733 lb
     Separating Force= 850 lb
    Compressive Stress= 144967 psi
    Life= 1.165E+15 to 1.951E+17 cycles
    Life Is More Than 100,000 hours
    NOTE: High Cycle Curve Used
            (See AGMA 218.01, Fig. 20)
    Number of same flank pinion contacts
       per revolution= 1
    Application Factor, Ca= 1
    Reliability Factor, CR= 0.9
     ((1 Failure in 20)
    Life Factor, CL= 0.652 to 0.58
    MIN REQUIRED EFFECTIVE CASE DEPTH
       FOR THIS LOAD= 0.0408
    MAX SUGGESTED EFFECTIVE CASE DEPTH= 0.0666
  Strength:
   Dynamic Factor, Kv= 0.907
    Face Load Distribution Factor, Cmf= 2.029
    Cmf > 2 INDICATES PARTIAL FACE LOADING
    Load Distribution Factor, Km= 2.029
    Horsepower= 102
    Pinion Torque= 4500 lb-in
    Tangential Load= 1733 lb
     Separating Force= 850 lb
    Pinion Bending Stress= 18816 psi
    Life= 1.471E+36 to 6.338E+36 cycles
    Life Is More Than 100,000 hours
    NOTE: High Cycle Curve Used
            (See AGMA 218.01, Fig. 21)
    Number of same flank pinion contacts
       per revolution= 1
    Application Factor, Ka= 1
    Reliability Factor, KR= 0.9
     ((1 Failure in 20)
    Life Factor, KL= 0.308 to 0.3
```

```
GEAR:  (Gear RPM= 762)
 Pitting:
  Dynamic Factor, Cv= 0.907
    Face Load Distribution Factor, Cmf= 2.029
    Cmf > 2 INDICATES PARTIAL FACE LOADING
    Load Distribution Factor, Cm= 2.029
    Horsepower= 102
    Pinion Torque= 4500 lb-in
    Tangential Load= 1733 lb
     Separating Force= 850 lb
    Compressive Stress= 144967 psi
    Life= 1.165E+15 to 1.951E+17 cycles
    Life Is More Than 100,000 hours
    NOTE: High Cycle Curve Used
            (See AGMA 218.01, Fig. 20)
    Number of same flank gear contacts
       per revolution= 1
    Application Factor, Ca= 1
    Reliability Factor, CR= 0.9
     ((1 Failure in 20)
    Life Factor, CL= 0.652 to 0.58
    MIN REQUIRED EFFECTIVE CASE DEPTH
       FOR THIS LOAD= 0.0408
    MAX SUGGESTED EFFECTIVE CASE DEPTH= 0.0846
 Strength:
  Dynamic Factor, Kv= 0.907
    Face Load Distribution Factor, Cmf= 2.029
    Cmf > 2 INDICATES PARTIAL FACE LOADING
    Load Distribution Factor, Cm= 2.029
    Horsepower= 102
    Pinion Torque= 4500 lb-in
    Tangential Load= 1733 lb
     Separating Force= 850 lb
    Gear Bending Stress= 17503 psi
    Life= 6.338E+36 cycles
    Life Is More Than 100,000 hours
    NOTE: High Cycle Curve Used
            (See AGMA 218.01, Fig. 21)
    Number of same flank gear contacts
       per revolution= 1
    Application Factor, Ka= 1
    Reliability Factor, KR= 0.9
     ((1 Failure in 20)
    Life Factor, KL= 0.3 to 0.3
```

```
PINION RPM= 3000  (Miner's Rule Cond #5)
 Pitch Line Velocity, vt= 4078 ft/min
PINION:
 Pitting:
  Dynamic Factor, Cv= 0.877
    Face Load Distribution Factor, Cmf= 1.995
    Cmf <= 2 indicates load across total face
    Load Distribution Factor, Cm= 1.995
    Horsepower= 219
    Pinion Torque= 4600 lb-in
    Tangential Load= 1771 lb
     Separating Force= 869 lb
    Compressive Stress= 147800 psi
    Life= 5.021E+14 to 8.411E+16 cycles
    Life Is More Than 100,000 hours
    NOTE: High Cycle Curve Used
            (See AGMA 218.01, Fig. 20)
    Number of same flank pinion contacts
       per revolution= 1
    Application Factor, Ca= 1
    Reliability Factor, CR= 0.9
     ((1 Failure in 20)
    Life Factor, CL= 0.665 to 0.591
    MIN REQUIRED EFFECTIVE CASE DEPTH
       FOR THIS LOAD= 0.0408
    MAX SUGGESTED EFFECTIVE CASE DEPTH= 0.0666
  Strength:
   Dynamic Factor, Kv= 0.877
    Face Load Distribution Factor, Cmf= 1.995
    Cmf <= 2 indicates load across total face
    Load Distribution Factor, Km= 1.995
    Horsepower= 219
    Pinion Torque= 4600 lb-in
    Tangential Load= 1771 lb
     Separating Force= 869 lb
    Pinion Bending Stress= 19558 psi
    Life= 1.672E+35 to 6.338E+36 cycles
    Life Is More Than 100,000 hours
    NOTE: High Cycle Curve Used
            (See AGMA 218.01, Fig. 21)
    Number of same flank pinion contacts
       per revolution= 1
    Application Factor, Ka= 1
    Reliability Factor, KR= 0.9
     ((1 Failure in 20)
    Life Factor, KL= 0.32 to 0.3
```

```
GEAR:  (Gear RPM= 1588)
 Pitting:
  Dynamic Factor, Cv= 0.877
    Face Load Distribution Factor, Cmf= 1.995
    Cmf <= 2 indicates load across total face
    Load Distribution Factor, Cm= 1.995
    Horsepower= 219
    Pinion Torque= 4600 lb-in
    Tangential Load= 1771 lb
     Separating Force= 869 lb
    Compressive Stress= 147800 psi
    Life= 5.021E+14 to 8.411E+16 cycles
    Life Is More Than 100,000 hours
    NOTE: High Cycle Curve Used
            (See AGMA 218.01, Fig. 20)
    Number of same flank gear contacts
       per revolution= 1
    Application Factor, Ca= 1
    Reliability Factor, CR= 0.9
     (<1 Failure in 20)
    Life Factor, CL= 0.665 to 0.591
    MIN REQUIRED EFFECTIVE CASE DEPTH
       FOR THIS LOAD= 0.0408
    MAX SUGGESTED EFFECTIVE CASE DEPTH= 0.0846
 Strength:
  Dynamic Factor, Kv= 0.877
   Face Load Distribution Factor, Cmf= 1.995
   Cmf <= 2 indicates load across total face
   Load Distribution Factor, Cm= 1.995
   Horsepower= 219
   Pinion Torque= 4600 lb-in
   Tangential Load= 1771 lb
    Separating Force= 869 lb
   Gear Bending Stress= 18194 psi
   Life= 6.338E+36 cycles
   Life Is More Than 100,000 hours
   NOTE: High Cycle Curve Used
           (See AGMA 218.01, Fig. 21)
   Number of same flank gear contacts
      per revolution= 1
   Application Factor, Ka= 1
   Reliability Factor, KR= 0.9
    (<1 Failure in 20)
   Life Factor, KL= 0.3 to 0.3
```

==== MINER'S RULE ====

Cond #	Pin RPM	Horsepower	Pin Tork	Mins	Pin-Cycles	Gear-Cycles
1	280	53.33	12000	30	8400	4447.06
2	390	139	22500	25	9750	5161.76
3	1230	181	9300	50	61500	32558.8
4	1440	102	4500	75	108000	57176.5
5	3000	219	4600	90	270000	142941
TOTALS				270	457650	242285

NOTE: The effects of C(H), C(T), CR, K(T) & KR are included in this data.

PINION PITTING:
Life= 331 hours
to 2715 hours
PINION BENDING STRENGTH:
Life= 61901 hours
To More Than 100,000 hours
GEAR PITTING:
Life= 625 hours
to 5129 hours
GEAR BENDING STRENGTH:
Life Is More Than 100,000 hours

NOTE: A range has been given for cycles and life of the gears. This is necessary as both values of Sac and Sat from Tables 5 & 6 have been used by the program. This range can be extensive due to the rapid change of cycles with the load. (See Fig. 20 & 21) The higher values may be used if special care is used in gearbox design, manufacture, and heat treatment.

Appendix O

Index to Computer Disks for "Standard" Gears

Index of files for "standard" gearset analysis

Disk A - "H.S. Train, Standard Gears (Military), Ground and Unground"

File Name	Description	UTS Program #
UHSSTD2.TK	HS Train Duty Cycle and Eq Cmf-Unground	TK
GHSSTD2.TK	HS Train Duty Cycle and Eq Cmf-Ground	TK
6787A.TK	19 tooth unground tolerance	60-100 TK
6787B.TK	19 tooth ground tolerance	60-100 TK
1984A.TK	32 tooth unground tolerance	60-100 TK
1984B.TK	32 tooth ground tolerance	60-100 TK
HSCONT.TK	Nominal Contact Conditions	60-104 TK
HSUN.PRT	#500 Output Sheet-Unground	500
HSUN.PLT	#500 Plot File-Unground	500
HSUN.500	#500 Gear Data File-Unground	500
HSGR.PRT	#500 Output Sheet-Ground	500
HSGR.PLT	#500 Plot File-Ground	500
HSGR.500	#500 Gear Data File-Ground	500
HSDTY50.MNR	Duty cycle - 50000 lb - H.S.Gears	540
2HSUNCR.TK	Crowned Gear Contact-Unground-Max Torque	60-5406 TK
2LHSUN.PRT	#540 Output Sheet-Unground	540
2HSGRCR.TK	Crowned Gear Contact-Ground-Max Torque	60-5406 TK
2LHSGR.PRT	#540 Output Sheet-Ground	540
2HSUNHSC.TK	Hot Score Prob - Max Speed - Unground	60-560 TK
2HSGRHSC.TK	Hot Score Prob - Max Speed - Ground	60-560 TK
2HSUNCSC.TK	Cold Score Prob - Max Torque - Unground	60-5408 TK
2HSGRCSC.TK	Cold Score Prob - Max Torque - Ground	60-5408 TK
HS19EFX.TK	19 T - Max Effective Tooth Thickness	60-EFF TK
HS19EFN.TK	19 T - Min Effective Tooth Thickness	60-EFF TK
HS32EFX.TK	32 T - Max Effective Tooth Thickness	60-EFF TK
HS32EFN.TK	32 T - Min Effective Tooth Thickness	60-EFF TK
HSCLDX.TK	Zero BL Temp-Min CD, Max Eff TT	60-1101/DTEMP TK
HSCLDN.TK	Zero BL Temp-Max CD, Min Eff TT	60-1101/DTEMP TK
LBRGI.TK	L-10 Life - Bearing I	20-370 MOD TK
LBRGII.TK	L-10 Life - Bearing II	20-370 MOD TK
LBRGIII.TK	L-10 Life - Bearing III	20-370 MOD TK
LBRGIV.TK	L-10 Life - Bearing IV	20-370 MOD TK
LBRGV.TK	L-10 Life - Bearing V	20-370 MOD TK
LBRGVI.TK	L-10 Life - Bearing VI	20-370 MOD TK

Disk B - "L.S. Train, Standard Gears (Military), Ground and Unground"

File Name	Description	UTS Program #
ULSSTD2.TK	LS Train Duty Cycle and Eq Cmf-Unground	TK
GLSSTD2.TK	LS Train Duty Cycle and Eq Cmf-Ground	TK
2025A.TK	18 tooth unground tolerance	60-100 TK
2025B.TK	18 tooth ground tolerance	60-100 TK
2079A.TK	53 tooth unground tolerance	60-100 TK
2079B.TK	53 tooth ground tolerance	60-100 TK
LSCONT.TK	Nominal Contact Conditions	60-104 TK
LSUN.PRT	#500 Output Sheet-Unground	500
LSUN.PLT	#500 Plot File-Unground	500
LSUN.500	#500 Gear Data File-Unground	500
LSGR.PRT	#500 Output Sheet-Ground	500
LSGR.PLT	#500 Plot File-Ground	500
LSGR.500	#500 Gear Data File-Ground	500
LSDTY50.MNR	Duty cycle - 50000 lb - L.S.Gears	540
2LSUNCR.TK	Crowned Gear Contact-Unground-Max Torque	60-5406 TK
2LLSUN.PRT	#540 Output Sheet-Unground	540
2LSGRCR.TK	Crowned Gear Contact-Ground-Max Torque	60-5406 TK
2LLSGR.PRT	#540 Output Sheet-Ground	540
2LSUNHSC.TK	Hot Score Prob - Max Speed - Unground	60-560 TK
2LSGRHSC.TK	Hot Score Prob - Max Speed - Ground	60-560 TK
2LSUNCSC.TK	Cold Score Prob - Max Torque - Unground	60-5406 TK
2LSGRCSC.TK	Cold Score Prob - Max Torque - Ground	60-5406 TK
LS18EFX.TK	18 T - Max Effective Tooth Thickness	60-EFF TK
LS18EFN.TK	18 T - Min Effective Tooth Thickness	60-EFF TK
LS53EFX.TK	53 T - Max Effective Tooth Thickness	60-EFF TK
LS53EFN.TK	53 T - Min Effective Tooth Thickness	60-EFF TK
LSCLDX.TK	Zero BL Temp-Min CD, Max Eff TT	60-1101/DTEMP TK
LSCLDN.TK	Zero BL Temp-Max CD, Min Eff TT	60-1101/DTEMP TK

Files with suffix .TK can be loaded directly into TK Solver Plus. Files with suffix .PRT or .PLT can be output to printer or plotter with UTS program "PLOT". Files with suffix .500 are gear data ASCII files and can be loaded into UTS Program #540 as input data or accessed by other programs for use of complete gear data. (The structure of the gear data ASCII files is detailed in the UTS Data Memory Map attached as Appendix E.) Files with suffix .MNR are duty cycle files and can be loaded into UTS Program #540.

Appendix P

Index to Computer Disks for "MLRS" Gears

Index of files for "MLRS" gearset analysis

Disk A - "H.S. Train, MLRS Gears (Military)"

File Name	Description	UTS Program #
HSMLRS50.TK	HS Train Duty Cycle and Eq Cmf-50000 lb	TK
HSMLRS66.TK	HS Train Duty Cycle and Eq Cmf-66000 lb	TK
0307B.TK	18 tooth tolerance	60-100 TK
0301B.TK	34 tooth tolerance	60-100 TK
HSCONT_M.TK	Nominal Contact Conditions	60-104 TK
HSMLRS.PRT	#500 Output Sheet	500
HSMLRS.PLT	#500 Plot File	500
HSMLRS.500	#500 Gear Data File	500
HSD50_M.MNR	Duty cycle - 50000 lb - H.S.Gears	540
HSD66_M.MNR	Duty cycle - 66000 lb - H.S.Gears	540
2HS50CR.TK	Crowned Gear Contact-50000 lb	60-5406 TK
2HS66CR.TK	Crowned Gear Contact-66000 lb	60-5406 TK
2LHS50.PRT	#540 Output Sheet-50000 lb	540
2LHS66.PRT	#540 Output Sheet-66000 lb	540
2HSHCR50.TK	Hot Score Prob - 50000 lb	60-560 TK
2HSHCR66.TK	Hot Score Prob - 66000 lb	60-560 TK
2HSCCR50.TK	Cold Score Prob - 50000 lb	60-5408 TK
2HSCCR66.TK	Cold Score Prob - 66000 lb	60-5408 TK
HS18EFXM.TK	18 T - Max Effective Tooth Thickness	60-EFF TK
HS18EFNM.TK	18 T - Min Effective Tooth Thickness	60-EFF TK
HS34EFXM.TK	34 T - Max Effective Tooth Thickness	60-EFF TK
HS34EFNM.TK	34 T - Min Effective Tooth Thickness	60-EFF TK
HSCLDX_M.TK	Zero BL Temp-Min CD, Max Eff TT	60-1101/DTEMP TK
HSCLDN_M.TK	Zero BL Temp-Max CD, Min Eff TT	60-1101/DTEMP TK
L50I.TK	L-10 Life - Bearing I - 50000 lb	20-370 MOD TK
L66I.TK	L-10 Life - Bearing I - 66000 lb	20-370 MOD TK
L50II.TK	L-10 Life - Bearing II - 50000 lb	20-370 MOD TK
L66II.TK	L-10 Life - Bearing II - 66000 lb	20-370 MOD TK
L50III.TK	L-10 Life - Bearing III - 50000 lb	20-370 MOD TK
L66III.TK	L-10 Life - Bearing III - 66000 lb	20-370 MOD TK
L50IV.TK	L-10 Life - Bearing IV - 50000 lb	20-370 MOD TK
L66IV.TK	L-10 Life - Bearing IV - 66000 lb	20-370 MOD TK
L50V.TK	L-10 Life - Bearing V - 50000 lb	20-370 MOD TK
L66V.TK	L-10 Life - Bearing V - 66000 lb	20-370 MOD TK
L50VI.TK	L-10 Life - Bearing VI -50000 lb	20-370 MOD TK
L66VI.TK	L-10 Life - Bearing VI -66000 lb	20-370 MOD TK

Disk B - "L.S. Train, MLRS Gears (Military)"

File Name	Description	UTS Program #
LSMLRS50.TK	LS Train Duty Cycle and Eq Cmf-50000 lb	TK
LSMLRS66.TK	LS Train Duty Cycle and Eq Cmf-66000 lb	TK
2025B_M.TK	18 tooth tolerance	60-100 TK
2079B_M.TK	53 tooth tolerance	60-100 TK
LSCONT_M.TK	Nominal Contact Conditions	60-104 TK
LSMLRS.PRT	#500 Output Sheet	500
LSMLRS.PLT	#500 Plot File	500
LSMLRS.500	#500 Gear Data File	500
LSD50_M.MNR	Duty cycle - 50000 lb - L.S.Gears	540
LSD66_M.MNR	Duty cycle - 66000 lb - L.S.Gears	540
2LS50CR.TK	Crowned Gear Contact-50000 lb	60-5406 TK
2LS66CR.TK	Crowned Gear Contact-66000 lb	60-5406 TK
2LLS50.PRT	#540 Output Sheet-50000 lb	540
2LLS66.PRT	#540 Output Sheet-66000 lb	540
2LSHCR50.TK	Hot Score Prob - 50000 lb	60-560 TK
2LSHCR66.TK	Hot Score Prob - 66000 lb	60-560 TK
2LSCCR50.TK	Cold Score Prob - 50000 lb	60-5408 TK
2LSCCR66.TK	Cold Score Prob - 66000 lb	60-5408 TK
LS18EFXM.TK	18 T - Max Effective Tooth Thickness	60-EFF TK
LS18EFNM.TK	18 T - Min Effective Tooth Thickness	60-EFF TK
LS53EFXM.TK	53 T - Max Effective Tooth Thickness	60-EFF TK
LS53EFNM.TK	53 T - Min Effective Tooth Thickness	60-EFF TK
LSCLDX_M.TK	Zero BL Temp-Min CD, Max Eff TT	60-1101/DTEMP TK
LSCLDN_M.TK	Zero BL Temp-Max CD, Min Eff TT	60-1101/DTEMP TK

Files with suffix .TK can be loaded into TK Solver Plus. Files with suffix .PRT or .PLT can be output to printer or plotter with UTS program "PLOT". Files with suffix .500 are gear data ASCII files and can be loaded into UTS Program #540 as input data or accessed by other programs for use of complete gear data. (The structure of the gear data ASCII files is detailed in the UTS Data Memory Map attached as Appendix E.) Files with suffix .MNR are duty cycle files and can be loaded into UTS Program #540

Appendix Q

Index to Computer Disks for "MLRS" Gears

Index of files on disk "MLRS Set, TACOM Test Data"

H.S.GEARS

File Name	Description	UTS Program #
HSMLRS.PRT	#500 Output Sheet	500
HSMLRS.PLT	#500 Plot File	500
HSMLRS.500	#500 Gear Data File	500
HSDTEST.MNR	Test Duty cycle - H.S.Gears	540
HSTEST.TK	Scoring Summary - H.S.Gears	TK
HSTESTCR.TK	Crowned Gear Contact	60-5406 TK
LHSTEST.PRT	#540 Output Sheet	540
HSHST1.TK	Hot Score Prob	60-560 TK
HSCST1.TK	Cold Score Prob	60-5408 TK

L.S.GEARS

File Name	Description	UTS Program #
LSMLRS.PRT	#500 Output Sheet	500
LSMLRS.PLT	#500 Plot File	500
LSMLRS.500	#500 Gear Data File	500
LSDTEST.MNR	Test Duty cycle - L.S.Gears	540
LSTEST.TK	Scoring Summary - L.S.Gears	TK
LSTESTCR.TK	Crowned Gear Contact	60-5406 TK
LLSTEST.PRT	#540 Output Sheet	540
LSHST1.TK	Hot Score Prob	60-560 TK
LSCST1.TK	Cold Score Prob	60-5408 TK

Files with suffix .TK can be loaded into TK Solver Plus. Files with suffix .PRT or .PLT can be output to printer or plotter with UTS program "PLOT". Files with suffix .500 are gear data ASCII files and can be loaded into UTS Program #540 as input data or accessed by other programs for use of complete gear data. (The structure of the gear data ASCII files is detailed in the UTS Data Memory Map attached as Appendix E.) Files with suffix .MNR are duty cycle files and can be loaded into UTS Program #540

Appendix R

Index to Computer Disks for Optimized Gears

Disk Title: "H.S. Train, OPT Gears (Military)"

File Name	Description	UTS Program #
HSOPT.TK	HS Train Duty Cycle and Eq Cmf	TK
HS18OPT.TK	18 tooth tolerance	60-100 TK
HS34OPT.TK	34 tooth tolerance	60-100 TK
HSOPT.PRT	#500 Output Sheet	500
HSOPT.PLT	#500 Plot File	500
HSOPT.500	#500 Gear Data File	500
HSD66_M.MNR	Duty cycle - 66000 lb - H.S.Gears	540
OHS66CR.TK	Crowned Gear Contact	60-5406 TK
OLHS66.PRT	#540 Output Sheet	540
OHSHCR66.TK	Hot Score Prob	60-560 TK
OHSCCR66.TK	Cold Score Prob	60-5408 TK
HS18EFXO.TK	18 T - Max Effective Tooth Thickness	60-EFF TK
HS18EFNO.TK	18 T - Min Effective Tooth Thickness	60-EFF TK
HS34EFXO.TK	34 T - Max Effective Tooth Thickness	60-EFF TK
HS34EFNO.TK	34 T - Min Effective Tooth Thickness	60-EFF TK
HSCLDX_O.TK	Zero BL Temp-Min CD, Max Eff TT	60-1101/DTEMP TK
HSMAXBLO.TK	Max BL - Max CD, Min Actual TT	60-1101 TK
HSTIPREL.TK	Tip Relief and Location	60-1111 TK

Disk Title: "L.S. Train, OPT Gears (Military)"

File Name	Description	UTS Program #
LSOPT.TK	LS Train Duty Cycle and Eq Cmf	TK
LS18OPT.TK	18 tooth tolerance	60-100 TK
LS53OPT.TK	53 tooth tolerance	60-100 TK
LSOPT.PRT	#500 Output Sheet	500
LSOPT.PLT	#500 Plot File	500
LSOPT.500	#500 Gear Data File	500
LSD66_M.MNR	Duty cycle - 66000 lb - L.S.Gears	540
OLS66CR.TK	Crowned Gear Contact	60-5406 TK
OLLS66.PRT	#540 Output Sheet	540
OLSHCR66.TK	Hot Score Prob	60-560 TK
OLSCCR66.TK	Cold Score Prob	60-5408 TK
LS18EFXO.TK	18 T - Max Effective Tooth Thickness	60-EFF TK
LS18EFNO.TK	18 T - Min Effective Tooth Thickness	60-EFF TK
LS53EFXO.TK	53 T - Max Effective Tooth Thickness	60-EFF TK
LS53EFNO.TK	53 T - Min Effective Tooth Thickness	60-EFF TK
LSCLDX_O.TK	Zero BL Temp-Min CD, Max Eff TT	60-1101/DTEMP TK
LSMAXBLO.TK	Max BL - Max CD, Min Actual TT	60-1101 TK
LSTIPREL.TK	Tip Relief and Location	60-1111 TK

Files with suffix .TK can be loaded into TK Solver Plus. Files with suffix .PRT or .PLT can be output to printer or plotter with UTS program "PLOT." Files with suffix .500 are gear data ASCII files and can be loaded into UTS Program #540 as input data or accessed by other programs for use of complete gear data (see UTS Data Memory Map). Files with suffix .MNR are duty cycle files and can be loaded into UTS Program #540.

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