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USAAVLABS TECHNICAL REPORT 67-83

AN EVALUATION OF CONFORMAL CONTACT GEARS

By
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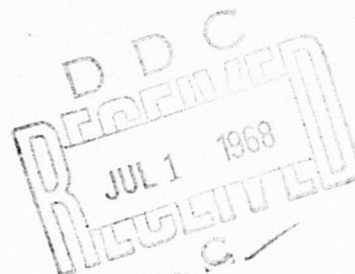
January 1968

U. S. ARMY AVIATION MATERIEL LABORATORIES
FORT EUSTIS, VIRGINIA
CONTRACT DA 44-177 AMC-389(T)
THE BOEING COMPANY
VERTOL DIVISION
PHILADELPHIA, PENNSYLVANIA

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AN EVALUATION OF CONFORMAL CONTACT GEARS

A. J. Lemanski, et al

**Boeing Company
Philadelphia, Pennsylvania**

January 1968



DEPARTMENT OF THE ARMY
U. S. ARMY AVIATION MATERIEL LABORATORIES
FORT EUSTIS, VIRGINIA 22604

This report represents a part of a research program established to investigate advanced gear design criteria. The main efforts of the program are directed toward dynamic load tests of specimen conformal contact gearing (Wildhaber-Novikov gear tooth form).

The Command concurs in the conclusions reported herein. The results obtained from this specific study indicate that extensive research investigations must be conducted before a complete evaluation of conformal contact gearing for aircraft application can be made.

Task LM125901A01410
Contract DA 44-177-AMC-389(T)
USAAVLABS Technical Report 67-83
January 1968

AN EVALUATION OF CONFORMAL CONTACT GEARS

Final Report

D8-0854A

By

**A. Lemanski
J. Alberti
J. Mack**

Prepared by

**The Boeing Company
Vertol Division**

for

**U S. ARMY AVIATION MATERIEL LABORATORIES
FORT EUSTIS, VIRGINIA**

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SUMMARY

This report concludes the program for the evaluation of one form of the conformal gear geometry conducted by The Boeing Company, Vertol Division, under Contract DA 44-177-AMC-389(T), initiated on 22 April 1966.

The program comprised the design, fabrication, inspection and load testing of one form of the conformal gear tooth geometry. The test gears were case carburized and ground in accordance with high aircraft quality standards and the applicable Vertol Division specifications. Load capacity of this particular tooth form was determined by a programmed increase in load until failure was attained. Results of this test program indicated that the gear member was the weakest in bending strength and that a better balance between contact stress and bending stress will be necessary in future designs. The tooth strain survey indicated that the trends of the strain readings at the gages, as the load zone passed across the face width, were similar to the results shown in USAAVLABS Technical Report 66-8.

FOREWORD

Acknowledgement is made to the following organizations for their assistance and contributions in the fabrication and inspection of the conformal gears during this program:

Fellows Gear Shaper Company, Springfield, Vermont

National Broach and Machine Company, Detroit, Michigan

Bendix Corporation, Industrial Metrology Division,
Ann Arbor, Michigan

Michigan Tool Division, Excello Corporation, Detroit,
Michigan

Acknowledgement is also made to Professors William Murphy and Paul Menz of the Villanova University Mechanical Engineering Department for their assistance and contributions during the test program.

Acknowledgement is also made to Mr. Robert Spencer of the Acoustics Unit of The Boeing Company, Vertol Division, for conducting the noise level test program.

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INTRODUCTION

The conformal (circular arc profile) gear has been investigated as a means for improving helicopter speed reduction drives. By overcoming certain inherent limitations of the involute tooth form, the developed conformal tooth form has shown promise of improving current power-to-weight ratios.

Both theoretical analysis and experimental evidence published to date indicate that convex profiles of mating involute teeth have less surface in contact than conformal (convex and concave circular arc) profiles. In addition, involute profiles are characterized by variable sliding conditions as contrasted by constant sliding of conformal teeth. As a result, for any given lubricant, the convex involute profiles develop a thinner oil film and generate more heat within the film than corresponding conformal profiles. Therefore, it follows that conformal teeth are inherently more efficient than involute teeth. For the same size and existing material, it should be possible to carry greater loads or to reduce the gear size, for the same load and material.

Under sponsorship of USAAVLABS, The Boeing Company, Vertol Division, has conducted theoretical and experimental programs on carburized and ground conformal gears. The initial program resulted in the first published work (USAAVLABS Technical Report 66-8)* covering aircraft quality gears. This first report presented the results of one tooth form of the conformal gear geometry. Problem areas were defined and specific recommendations were made for future design and development work. It was determined that in order to develop the full potential of the conformal gear and to improve further on the highly developed aircraft involute gear, a systematic approach of testing, analyzing, and modifying will be required. In addition, appropriate inspection fixtures and equipment must be designed and fabricated to permit the required dimensional control commensurate with aircraft involute gears.

In this direction, the current (second) program was conducted in an effort to explore the recommendations and problem areas presented at the conclusion of the previous program.

*Mack, J.C., Alberti, J.P., Lemanski, A.J., AN EXPERIMENTAL EVALUATION OF THE CONFORMAL GEAR, Vertol Division of Boeing, USAAVLABS Technical Report 66-8, U.S. Army Aviation Materiel Laboratories, Fort Eustis, Virginia, January 1966, DA 44-177-AMC-101(T), Modification 1. (AD 630925)

DESCRIPTION OF TEST ITEM

DESIGN

Six test gears utilized in this test program were fabricated by the National Broach and Machine Company, Detroit, Michigan. Two of these test gears were manufactured specifically for a destructive metallurgical (IP-1) evaluation.

The SK16207 Conformal Test Gears were manufactured to the following design specifications (Table I). For the complete dimensional data see Figure 1.

TABLE I. DESIGN SPECIFICATIONS		
	Pinion	Gear
Pressure angle, degrees	25°	25°
Diametral pitch, inches	4.500	4.500
Number of teeth	16	38
Pitch diameter, inches	3.5556	8.4444
Face width, inches	2.132	2.132
F/D ratio	0.60	0.60
Helix angle, degrees	25° 13'	25° 13'
Axial overlap	1.40	1.40

The design specifications selected for the test gears reflect the assumptions and recommendations suggested by USAAVLABS Technical Report 66-8. Both gear sets are identical except that the pinion in set number 2 (SK16207-7) was crowned, in the lead direction, at the entrance and exit ends. Crown tolerance at both ends was 0.0005 to 0.0008 inch for a nominal distance of 0.50 inch (Figure 1). This modification was accomplished to alleviate possible entrance distress or exit distress caused by deflection of the teeth as they pass through contact.



3. THE SPECIFIED LEAD TOLERANCE APPLIES TO THE TRUE LEAD PORTION ONLY
4. MAINTAIN ROOT LINE PARALLEL TO TOP LAND OF TOOTH WHEN MACHINING END CROWNS.

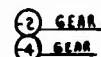
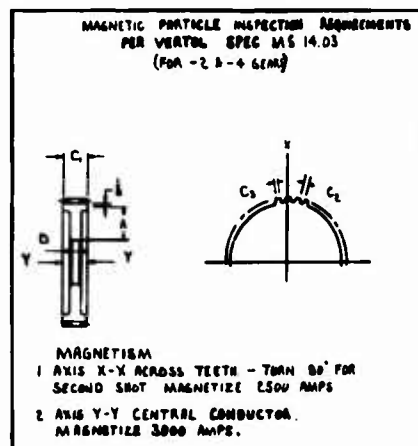


Figure 1. Conformal Gear Drawing - SK16207 (Sheet 1 of 2).

A.

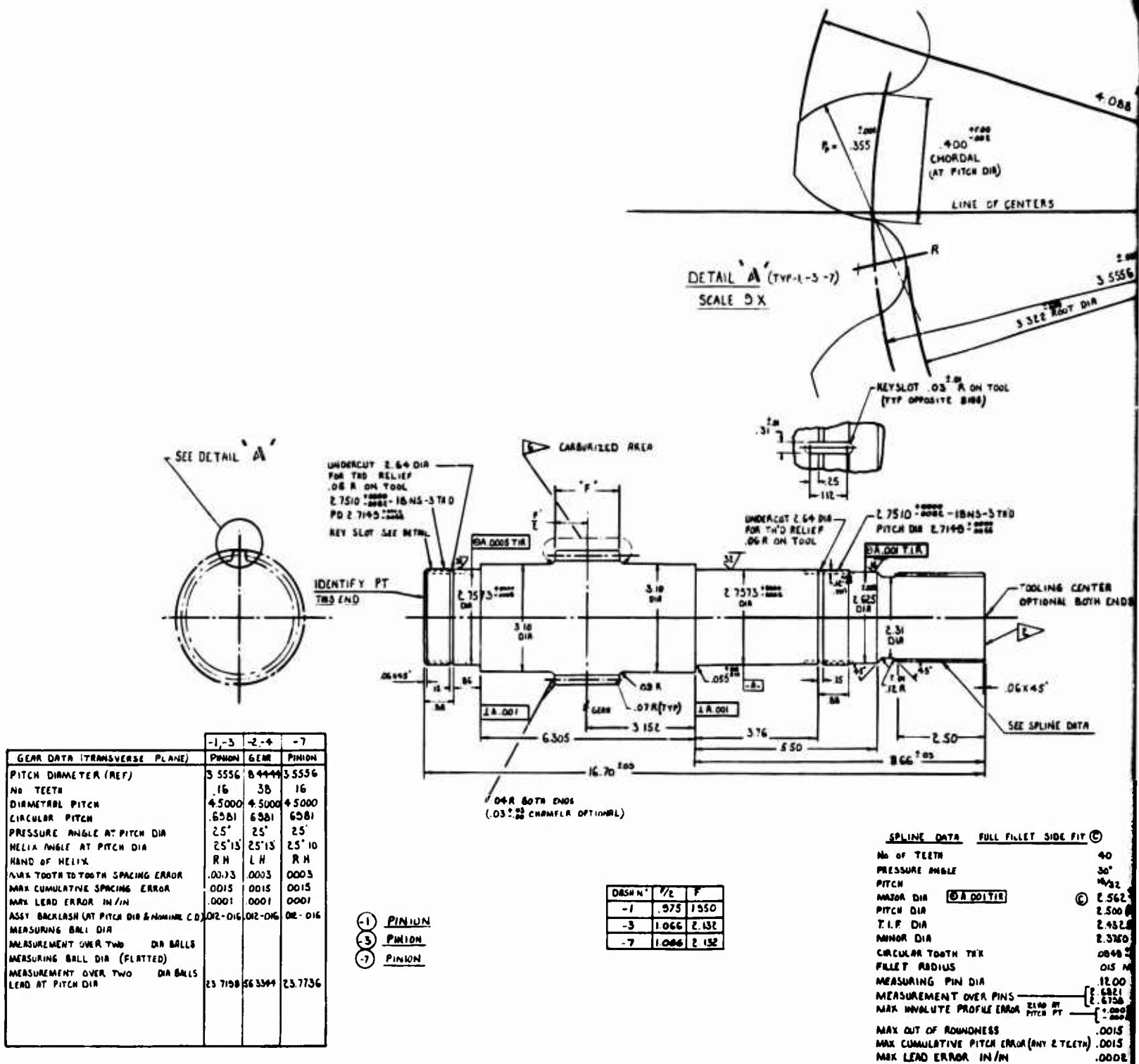


Figure 1. Conformal Gear Drawing - SK16207 (Sheet 2 of 2).

Comparison with SK13264 Test Gears

The current SK16207 conformal gears are the same as the SK13264 gears tested under the previous contract with respect to the following (See Figure 2):

1. Number of teeth
2. Helix angle
3. Pitch diameter
4. Diametral pitch
5. Axial overlap
6. Face width
7. Tooth thickness distribution

Profile radius of curvature of both the pinion and gear members was lengthened to increase the available tooth contact area, and thereby alleviated possible tip impingement experienced during the last program. This modification required an accompanying reduction in the pressure angle to maintain sufficient top land width. Profile mismatch was increased from 5 percent to 10 percent in order to permit greater center distance variation and more tolerance in manufacture of the test gears, and to prevent the last point of contact from reaching the tip of the concave teeth (gear member).

The gear member was undercut in the fillet area to insure that the profile tangency point was below the mating outside diameter of the pinion. This modification also enhances the grinding of the tooth flanks and negates the necessity for a blending operation at the root diameter, by leaving this undercut area unground. (See Figure 2.)

FABRICATION

The manufacturing procedure and process sequence for the SK16207 conformal contact test gears was as follows:

1. Forge individual biscuit of 9310 vacuum melt steel.
2. Rough machine and stress relieve gear blank.

3. Final machine gear blank.
4. Hob (cut) teeth.
5. Carburize gear teeth.
6. Heat-treat (harden and draw).
7. Grind faces and journals.
8. Grind teeth.
9. Make final inspection.
10. Shot peen teeth.

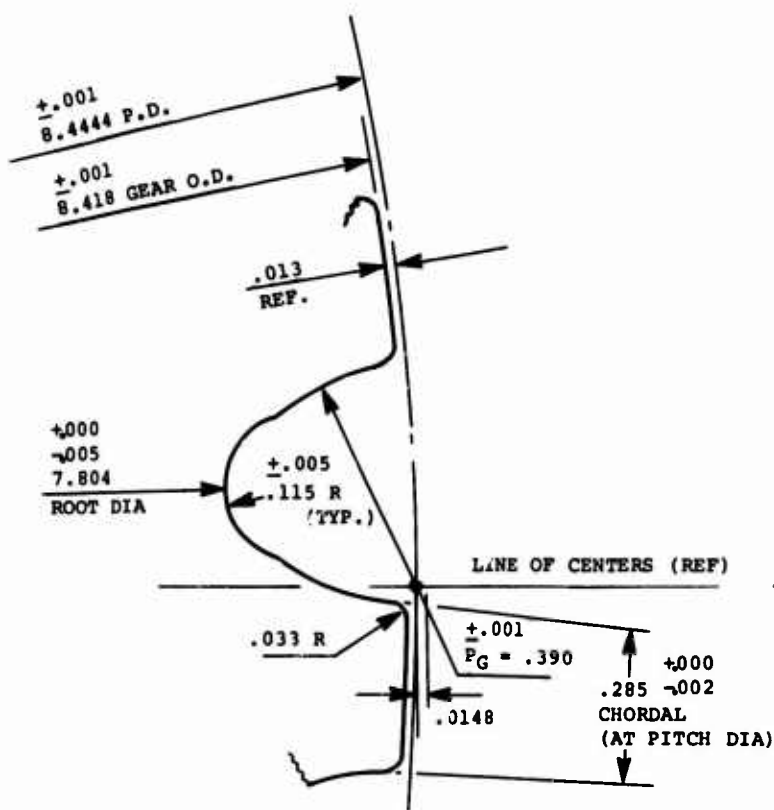
This sequence is normal for Vertol Division production gears. Standard gear production machine tools were used, although a special hob design was required to produce the conformal tooth.

The test gears were fabricated utilizing the nongenerating form ground method where the grinding wheel is formed to the required shape. Grinding stock allowance and equal stock removal on the test gears did not require any additional efforts as compared to production involute gears of comparable size. Accurate fabrication of the hob and controls of heat treat distortion contributed to the success of the final grinding operations. Nital etch inspection at Vertol Division did not indicate the presence of grinding burns on the subject test gears.

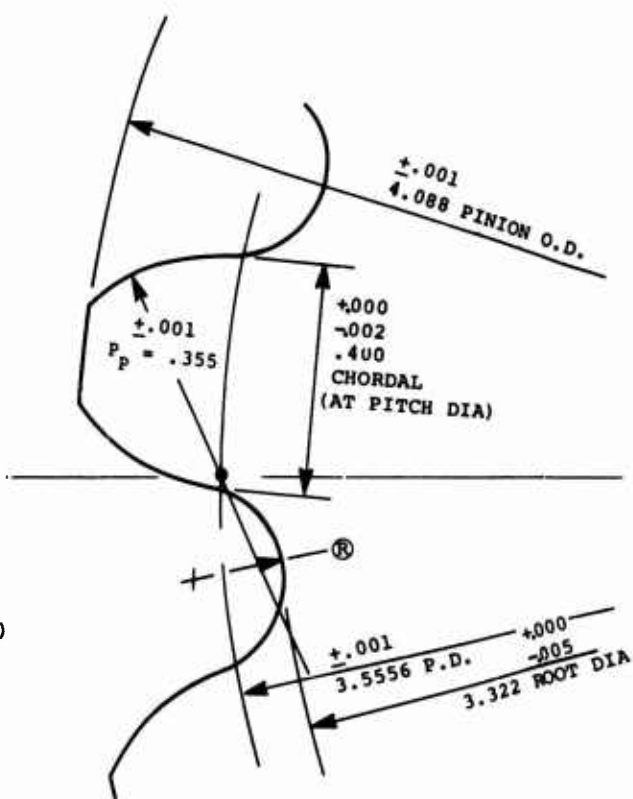
Both test gear sets had a 4-hour run-in (878,400 cycles) at approximately 50 percent (4,225 inch-pounds torque on the pinion) of the design load. Each gear set had three test runs at various load levels. A test run was concluded at 6 million cycles of load on the pinion teeth. Successful completion of a run without failure was considered as a runout point, and the applied stress was considered to be below the fatigue endurance limit for that particular test specimen.

GEAR TOOTH INSPECTION RESULTS

The results of the gear tooth element inspection are shown in Table II.

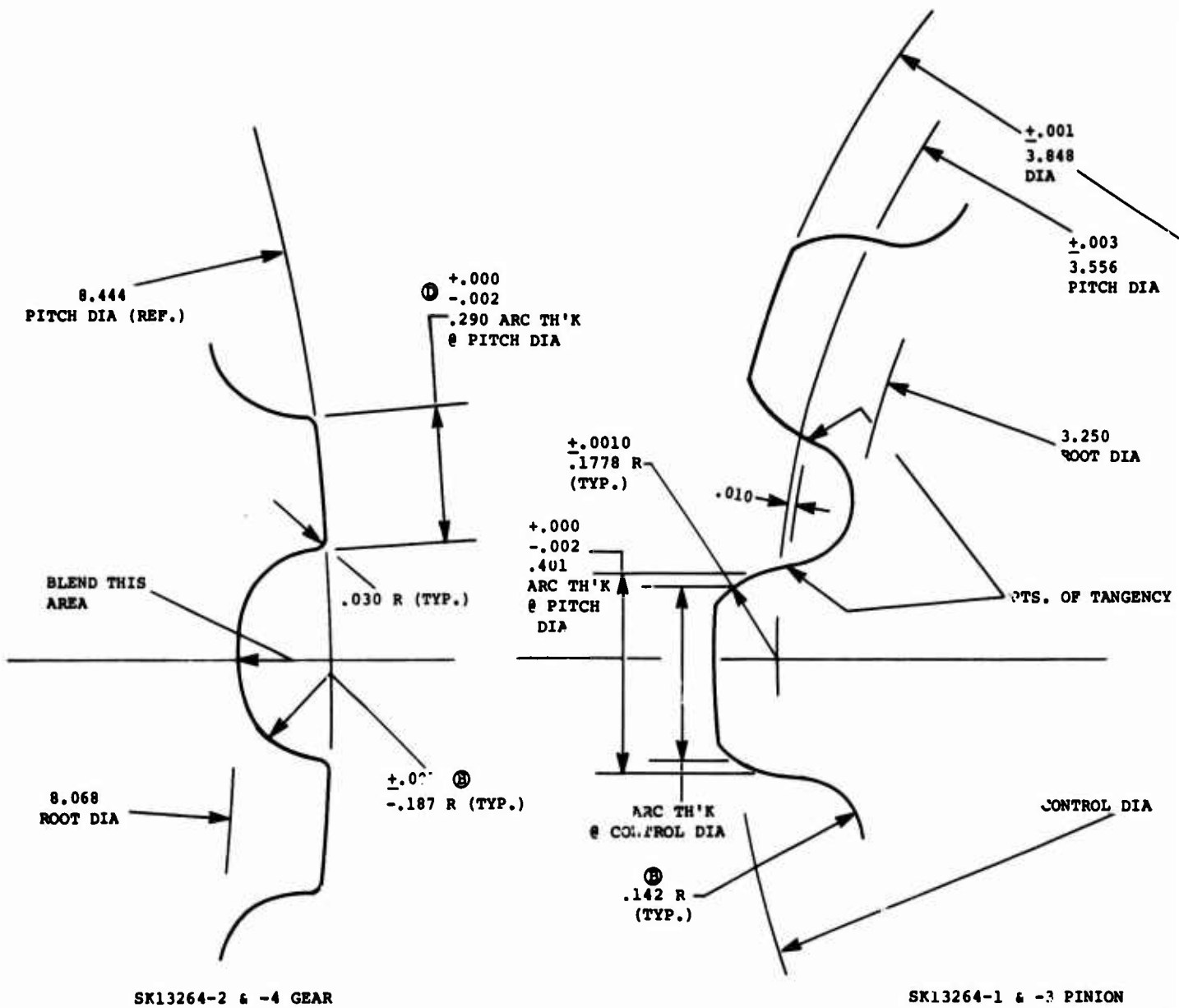


SK16207-4 GEAR



SK16207-3 & -7 PINION

Figure 2. Comparison of Geometry of SK16207 Test Gears With SK13264 Test Gears.



B.

TABLE II. GEAR TOOTH INSPECTION RESULTS					
Tooth Element	Pinion		Gear		Gear Actual (inch)
	SK16207-3 Tolerance (inch)	SK16207-3 Actual (inch)	SK16207-4 Tolerance (inch)	SK16207-4 Actual (inch)	
Set No. 1					
Profile	.0020	Within .0003	.0020	*Not checked	
Lead of Helix	.0002	Within .0002	.0002	Within .0001	
Tooth to Tooth Spacing	.0003	Within .0001	.0003	**Not checked	
Accumulated Spacing	.0015	Within .0002	.0015	**Not checked	
Pitch Circle Runout	.0010	Within .0002	.0020	Within .0005	
Set No. 2					
Profile	.0020	Within .0003	.0020	*Not checked	
Lead of Helix	.0002	Within .0002	.0002	Within .0001	
Tooth to Tooth Spacing	.0003	Within .0001	.0003	**Not checked	
Accumulated Spacing	.0015	Within .0002	.0015	**Not checked	
Pitch Circle Runout	.0010	Within .0003	.0020	Within .0005	
Crown of Helix	.0005-.0009	.0006-.0008	None	None	
* Requires special fixture for use on involute measuring instrument.					
** Requires special ball contact fingers for use on index measuring instrument.					

METHODS OF INSPECTION

As a result of this program, inspection techniques have been determined which are capable of providing suitable dimensional control of aircraft quality conformal (circular arc profile) gears. All the circular arc tooth elements can be inspected to the same accuracy as involute teeth.

The inspection equipment used for dimensional control of circular arc teeth is the same as that for involute teeth with one exception. (Refer to Table III.) The circular arc profile inspection requires a special offset fixture when using a conventional involute measuring instrument. When a practical need arises in the future, current equipment manufacturers will probably market circular arc profile-measuring instruments which do not require a fixture.

TABLE III. INSPECTION EQUIPMENT FOR
DIMENSIONAL CONTROL

Circular Arc Tooth Element	Inspection Technique
Profile	Involute Measuring Instrument with Special Offset Fixture
Lead of Helix	Lead Measuring Instrument
Tooth to Tooth Spacing	Index Measuring Instrument
Accumulated Spacing	Calculated from Tooth Spacing Checks
Pitch Circle Runout	Cone Checking Instrument
Tooth Thickness	Vernier Gear-Tooth Caliper and Over Two Ball Measurement

To demonstrate the feasibility of circular arc profile inspection, an offset gear fixture was developed for the convex pinion member for use on a standard involute measuring instrument. A special pointer (stylus) is required which has a 0.0005-inch-radius diamond for contacting the profile. (Refer to Figure 3.) The primary purpose of the fixture is to position the conformal pinion in both a radial and a tangential

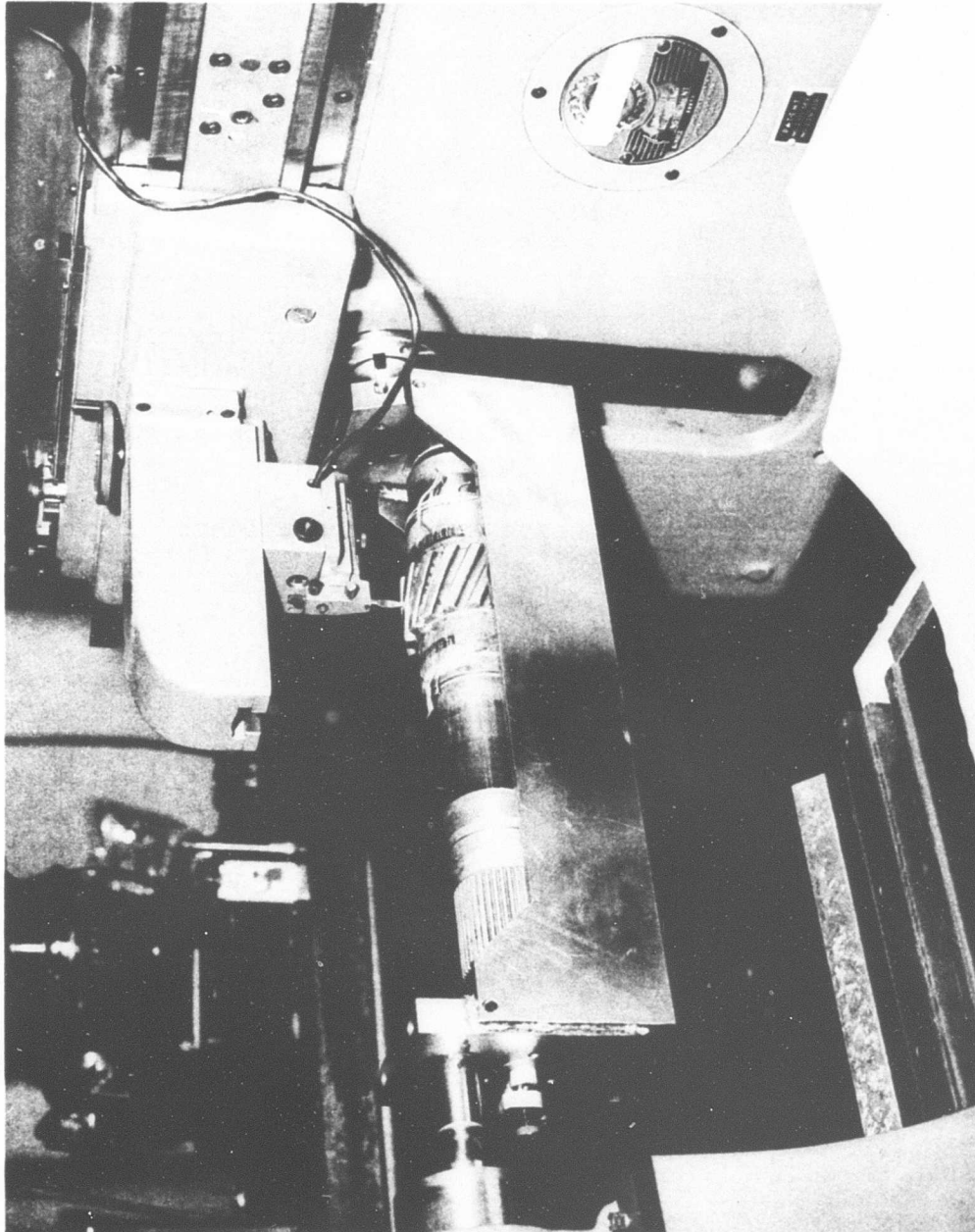


Figure 3. Offset Pinion Fixture for Involute Measuring Instrument.

direction so as to permit rotation about the center of the profile radius while the fixture is mounted between the center of the involute measuring instrument. The generating action of the instrument is locked out by setting the base circle radius to zero.

The pointer is located by means of a gage block on the cross slide to a tangential offset position with respect to the measuring instrument centers equal to the profile radius. With this arrangement, the fixture and conformal pinion profile rotate tangent to the pointer. A circular arc profile with a perfect radius will not actuate the pointer, and thereby the recorder would produce a straight line. (Refer to Figure 4.)

In the past, the inspection of precision circular arc tooth profiles (both convex and concave configurations) was limited to the following relatively crude techniques (with regard to accuracy) as compared to the more sophisticated analytical methods employed for measuring involute profiles:

1. Optical projection of convex and concave tooth profiles
2. Tracing of convex tooth profiles
3. Tooth-bearing patterns on rolling fixture

OPTICAL PROJECTION TECHNIQUE

This method has inherent limitations on accuracy, usually less than the accuracy of the precision-ground circular arc profile to be measured. Fundamentally, an optically projected image does not have a sharply defined edge as the human eye observes it. The black image texture actually becomes shades of gray at the edges, and the eye cannot distinguish small variations in the image contour.

Magnification is also a limitation for coarse pitch teeth. For example, a 4.5-pitch tooth size can be magnified only 20 times on a 30-inch optical comparator screen. A 0.0001-inch profile error would be a 0.002-inch error on the screen image. The human eye is not capable of discerning a 0.002-inch variation in the projected image. In addition, the tooth profile outline on the screen has a line thickness of approximately 0.002 inch with some variation in thickness and location.

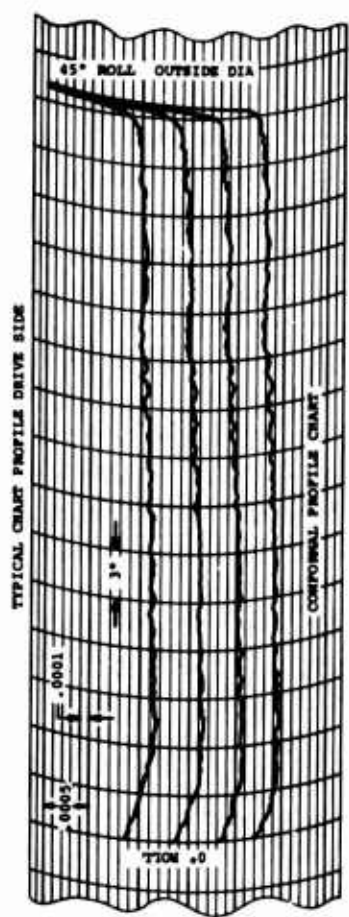
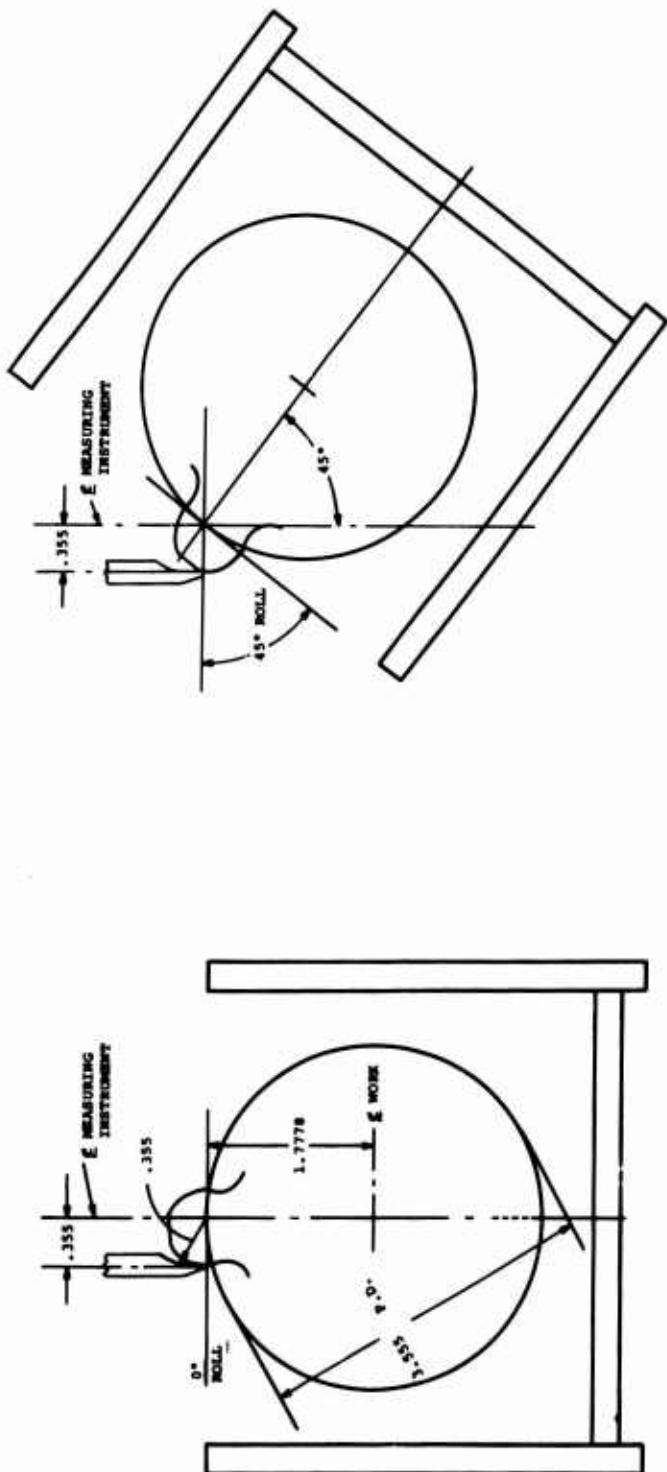


Figure 4. Pinion Offset Fixture - Principle of Operation.

The tooth helix angle of conformal gear limits direct inspection of the tooth profile to the ends of the gear face, and edge breaks or rounding distorts the resulting projected image. Both of these limitations can be overcome by the use of indirect methods which are subject to limited accuracy, as explained above.

The use of a coordinate tracing fixture on the table of an optical comparator will permit inspection of a convex or concave circular arc profile at any point along the helix. Refer to Figure 5 for a schematic of the system.

As an aid to fabrication, a 0.002-inch shim is attached to the end of the gear blank during the cutting and grinding operations. The shim is removed from the gear and becomes a template of the profile which can be projected on an optical comparator. The template image is compared to the chart of the profile on the screen.

TRACING TECHNIQUE

This method adapts an involute measuring instrument to tracing a circular arc profile by setting the instrument to a reference involute base circle which closely approximates the circular arc. The deviations of the perfect circular arc profile from the reference involute are computed at several locations, and thereby a master inspection profile curve is established. The recorded tracing of the inspected circular arc profile is then compared to the master profile curve. Deviations from the master curve are considered errors.

The tracing technique is subject to an indeterminate degree of error. Some error results from the tracing pointer. The shape of the pointer and its position with respect to the circular arc profile are critical and can be set only by eye. In most cases, the deviations of the perfect circular arc profile from the reference involute are several thousandths of an inch. The accuracy of the recording equipment is doubtful over so broad a range. In other cases, the circular arc profile trace extends beyond the chart width.

TOOTH BEARING PATTERNS

This method provides only functional information or the composite of all errors as they affect the amount of contact area under some light load. Individual tooth element errors cannot be ascertained from this method of inspection.

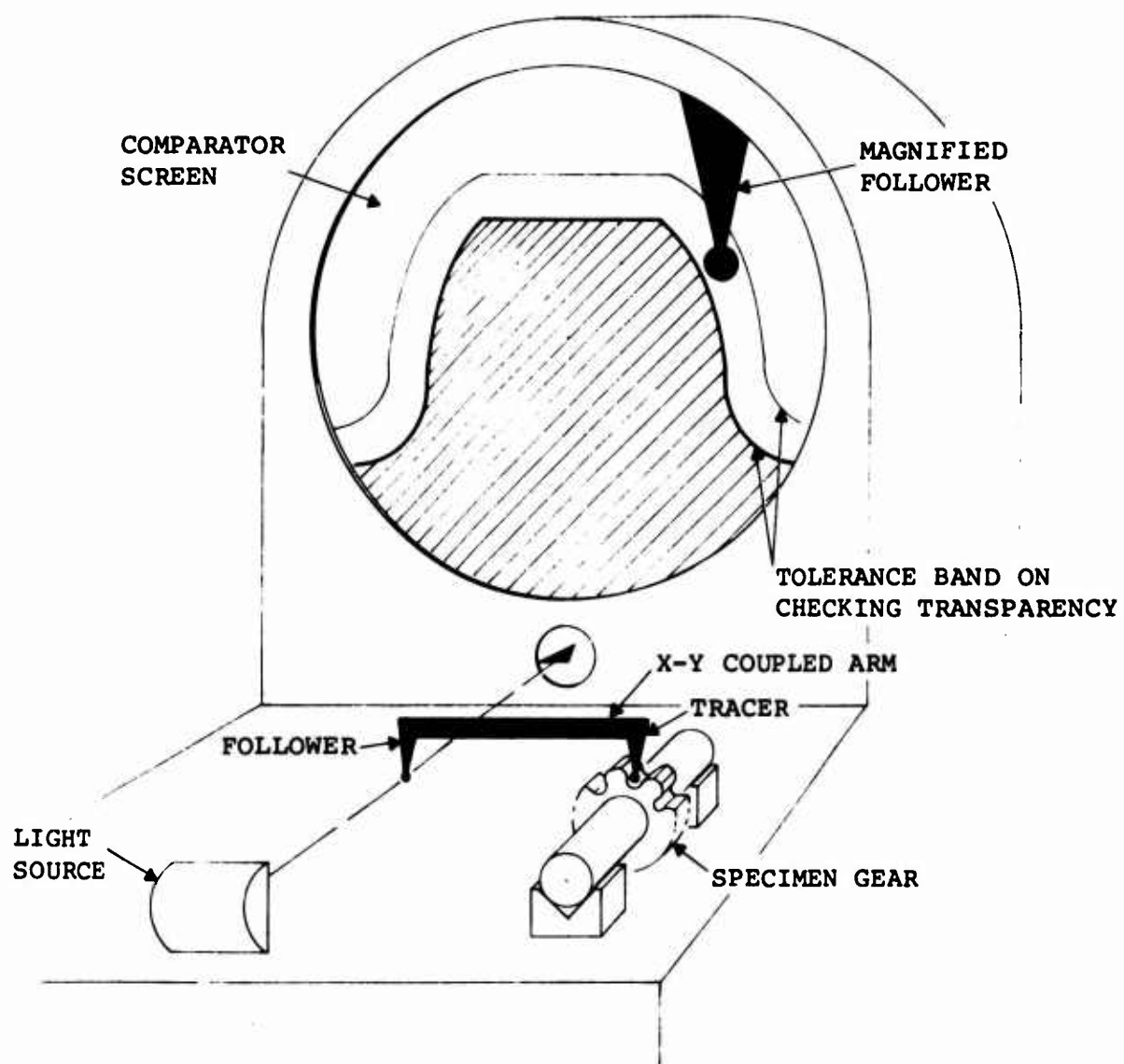


Figure 5. Coordinate Tracer Schematic.

STRAIN SURVEY

In order to evaluate the load-carrying capability of the con-formal gear, it is necessary to understand the nature of the load-sharing and concentration, as the teeth rotate through mesh and under load. To accomplish this, three consecutive pinion teeth were gaged in such a way that records of load-sharing were obtained as the gear and pinion were rolled through mesh under load. The application of torque and direction of rotation were identical to the load test runs.

INSTRUMENTATION

Strain gages (type C6-111) were applied to the tension root fillet of the pinion. Figures 6 and 7 show the gage placement and position by numbered location. Resistance calibration methods were used to establish a known strain level in each gage. Gage locations with respect to distance from the pinion pitch diameter were held within 0.007 inch of each other. The gages did not extend into the contact surface of the teeth. Seven gages were applied to the middle tooth at equal spacing of 0.330 ± 0.015 , with gages numbers 3 and 9 as close as possible to the tooth ends. Gages numbers 1 and 2 were applied to the preceding tooth with the same spacing between gages. Gages 10 and 11 were similarly applied to the entering side of the succeeding tooth. Additional information was obtained from a rotary potentiometer connected to the pinion shaft and from a continuous record of the gear torque.

PROCEDURE

The test procedure was to rotate the pinion and gear slowly through mesh, while observing and recording strain gage outputs, torque, and angular position. The first roll-through was accomplished with the pinion as the driving member. The second roll-through was achieved with the gear driving and returning to the original starting point. The driving member is the one where the loaded side of the tooth is advancing with respect to direction of rotation. In normal speed decreasing drive, the pinion is driving. This was the condition prevailing during the actual test runs under load. All of the data were recorded on a multichannel oscillograph while the gears were rotated through mesh.

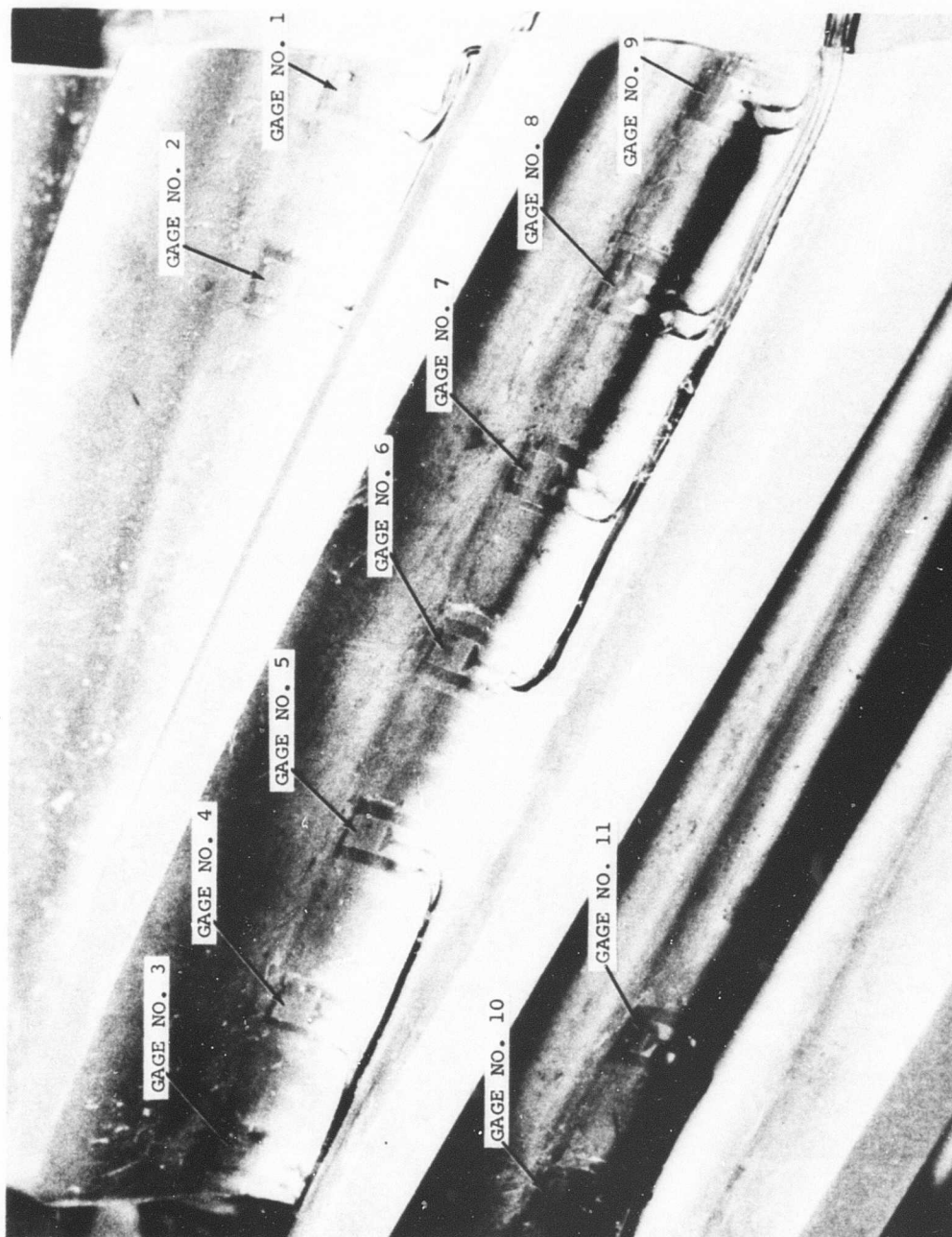


Figure 6. Strain Gage Locations - Pinion.

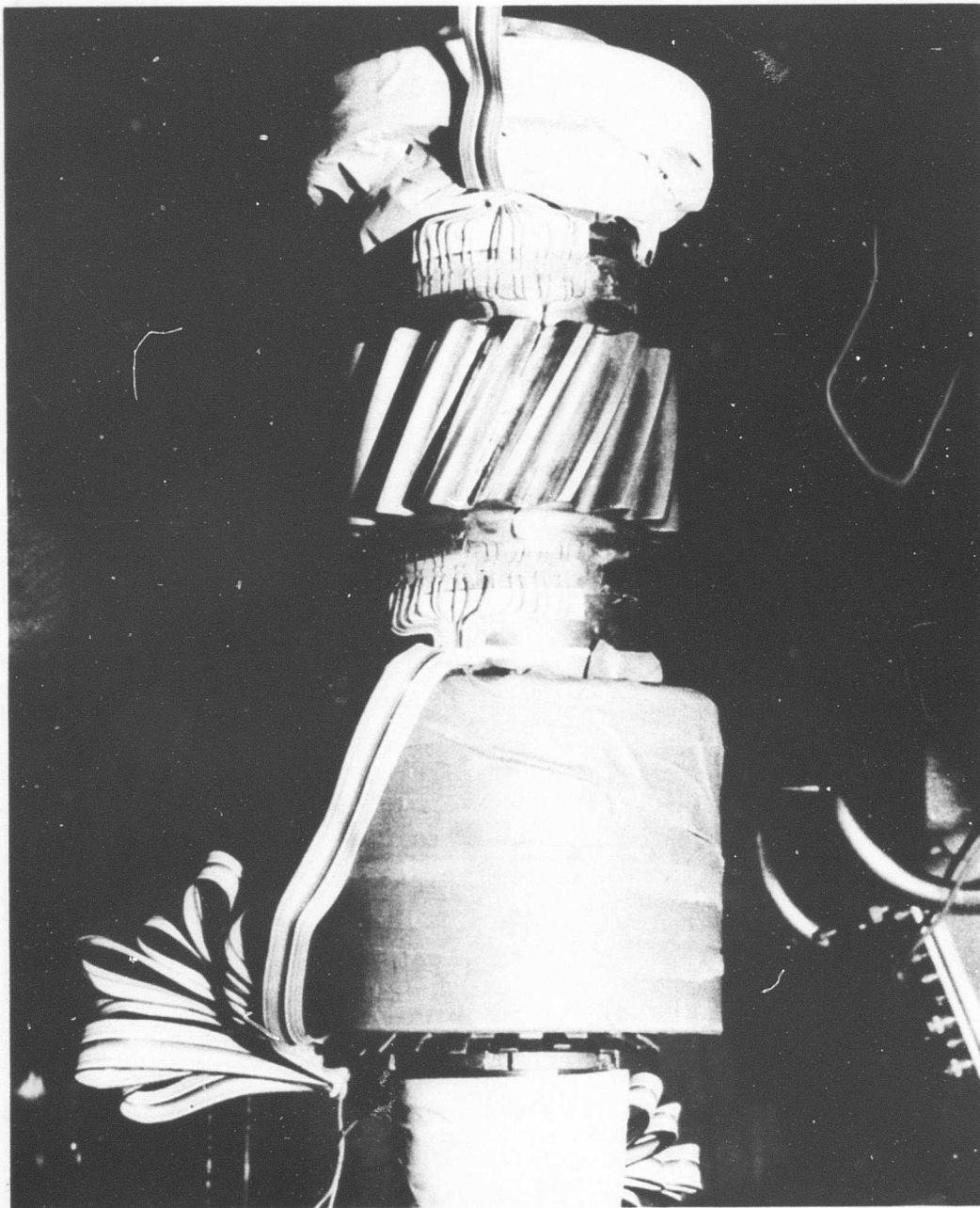


Figure 7. Strain Gage Installation - Pinion.

RESULTS

A typical record of test results is shown in Figure 8. Outputs of gages 1 through 11 are shown as they rise and fall during the meshing cycle. The middle tooth is in mesh from the first rise of gage 9 to the return to zero of gage 3. The return to zero of gage 10 indicates the instant of disengagement of the first tooth. The rise of gage 2 indicates the engagement of the third tooth. The interim period is when the middle tooth carries all the load. The period of single-tooth engagement was found to be of very short duration (Figure 9).

The left-hand peaks were obtained with the pinion driving; the right-hand peaks were obtained in reverse with the gear as the driving member. During the meshing cycle, strain amplitudes were found to be consistently higher with the pinion driving. This result is in agreement with the testing accomplished in USAAVLABS Technical Report 66-8.

Variations in the torque trace were observed and recorded during roll-through. The maximum variation shown is less than 1 percent of the total applied torque. Torque variations did not appear to repeat in a cyclic pattern and were not clearly identifiable with other rotating components. Maximum strains at each gage across the pinion face are recorded in Table IV. Two test conditions are shown.

Strain outputs by a particular gage may be affected by variables such as gage location, inclination, and bonding of the gage to the specimen. With this in mind, the trends and directions of the maximum strain at each gage across the pinion face width compare favorably with the results presented in USAAVLABS Technical Report 66-8 (see Figures 10 and 11). The two test conditions for the strain survey were conducted at 100 and 150 percent of the design load. Results indicate that the maximum strain level for the same gage location is approximately proportional to the increase in load level. Gage number 3 (on the exiting end of the middle tooth) recorded a strain 48 percent lower than gage number 9 (on the entrance end of the middle tooth). This condition may be partially responsible for the tapered patterns, larger width at the entrance end, and tapering to a smaller width at the exit end. It is reasonable to expect that an increased strain can result in more area in contact and a larger pattern.

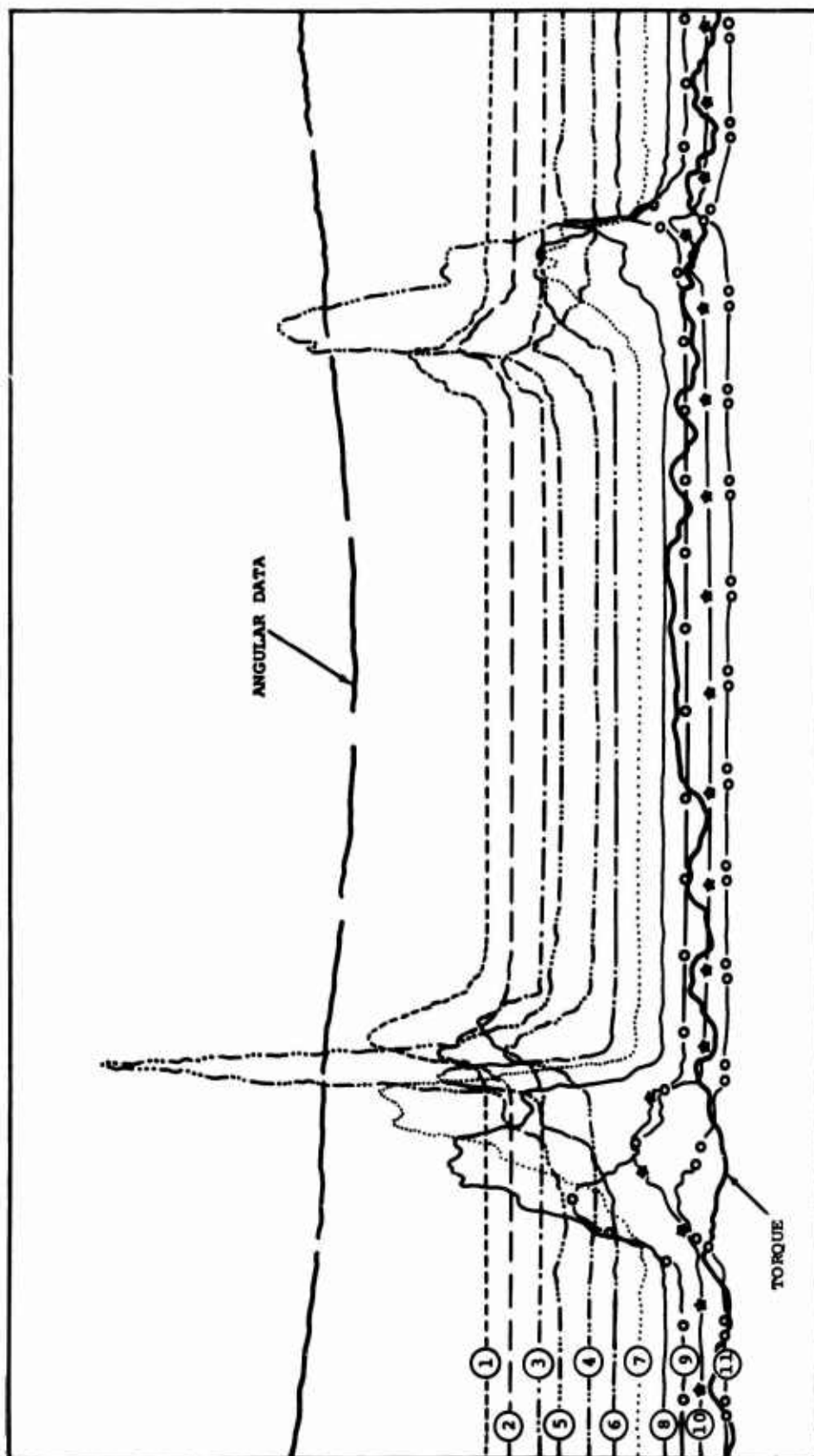


Figure 8. Oscilloscope Record of Strain Gage Data.

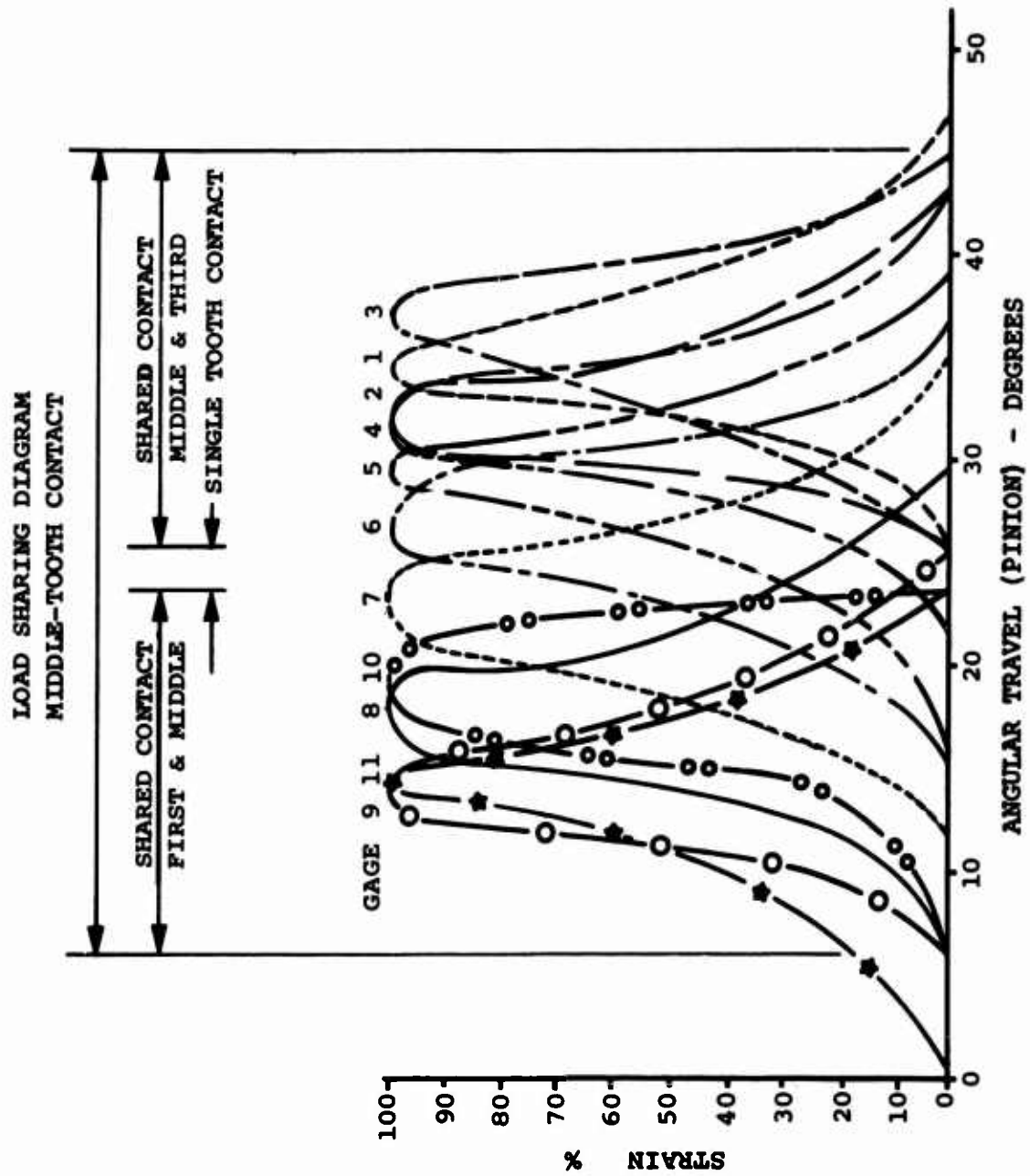


Figure 9. Load Overlapping Effect.

TABLE IV. STRAIN GAGE SURVEY DATA
(SK16207-3, PINION; SK16207-4, GEAR)

Gage No.	Maximum Amplitude (inch)		Maximum Strain (Micro in./in.)	
	Pinion Driver	Gear Driver	Pinion Driver	Gear Driver
Run No. 1 - 21,800 Inch-Pounds Torque				
1	0.69	0.45	1750	1140
2	0.43	0.31	2033	1465
3	0.39	0.26	1229	819
4	0.54	0.38	1257	885
5	2.93	1.77	2040	1231
6	1.06	0.49	2650	1225
7	1.55	0.57	2323	855
8	1.23	0.63	1891	968
9	0.68	0.39	2206	1265
10	0.45	0.23	1263	646
11	0.31	0.17	1451	796
Run No. 3 - 32,550 Inch-Pounds Torque				
1	1.02	0.66	2585	1672
2	0.61	0.44	2881	2080
3	0.55	0.34	1731	1070
4	0.75	0.52	1741	1208
5	3.95	2.48	2747	1725
6	1.53	0.69	3824	1725
7	2.26	0.88	3387	1320
8	1.83	0.90	2813	1383
9	0.97	0.51	3143	1653
10	0.62	0.30	1740	843
11	0.43	0.23	2016	1078

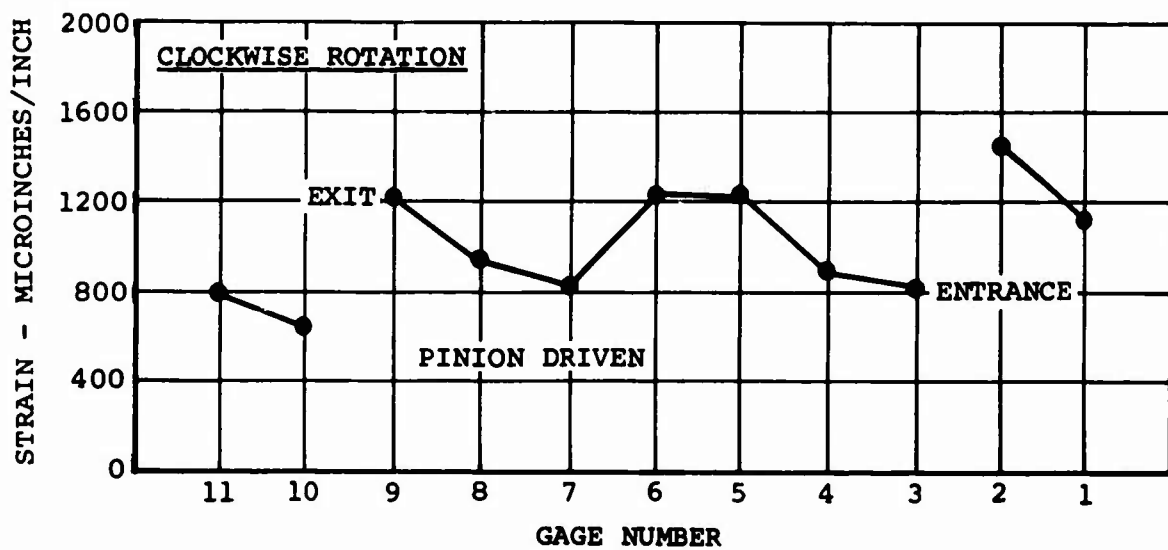
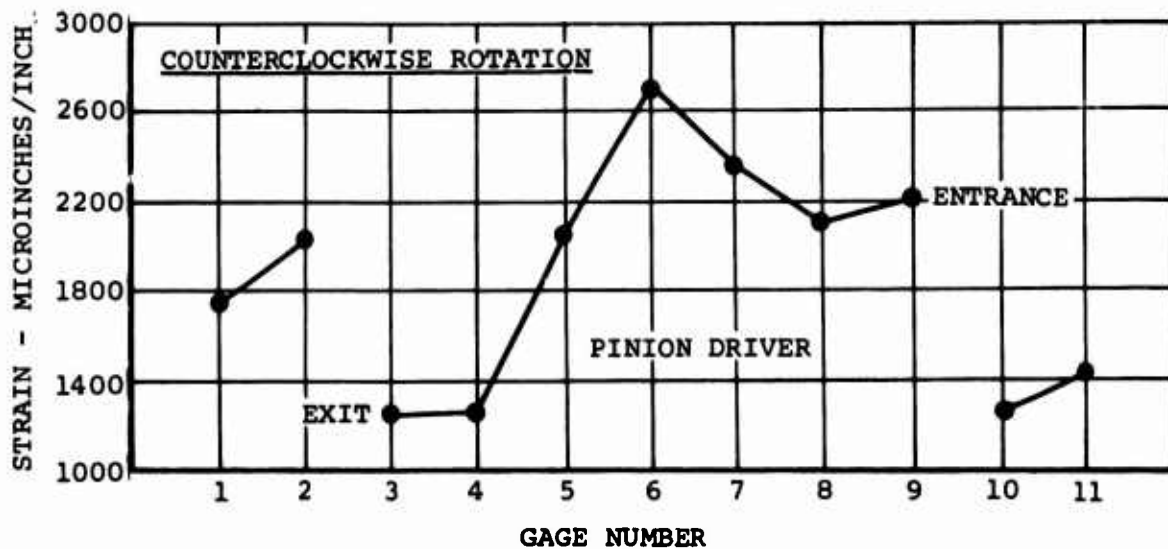


Figure 10. Maximum Strain at 21,800 Inch-Pounds Torque.

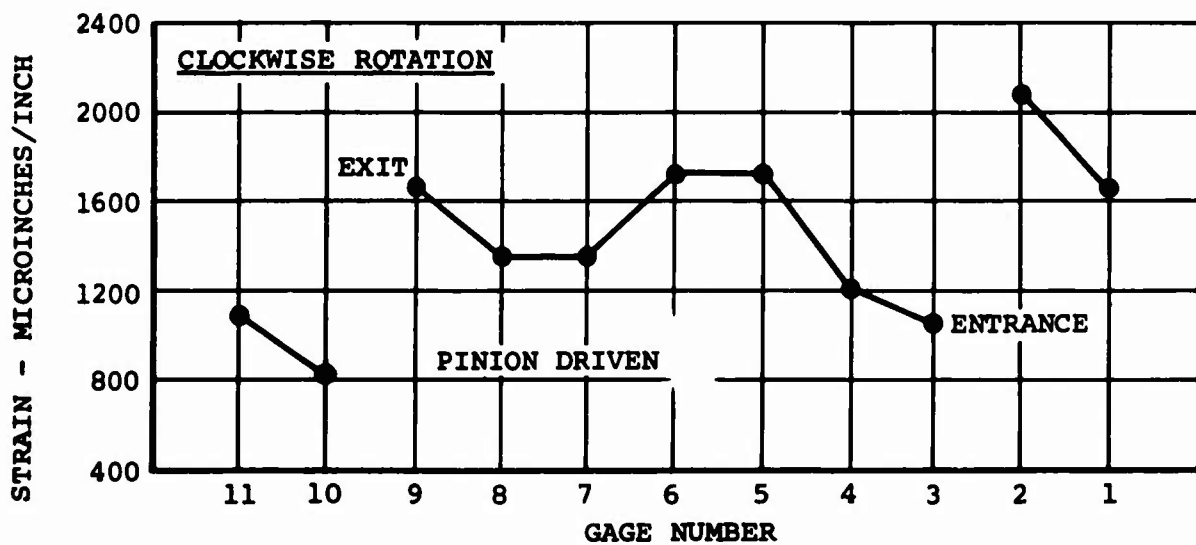
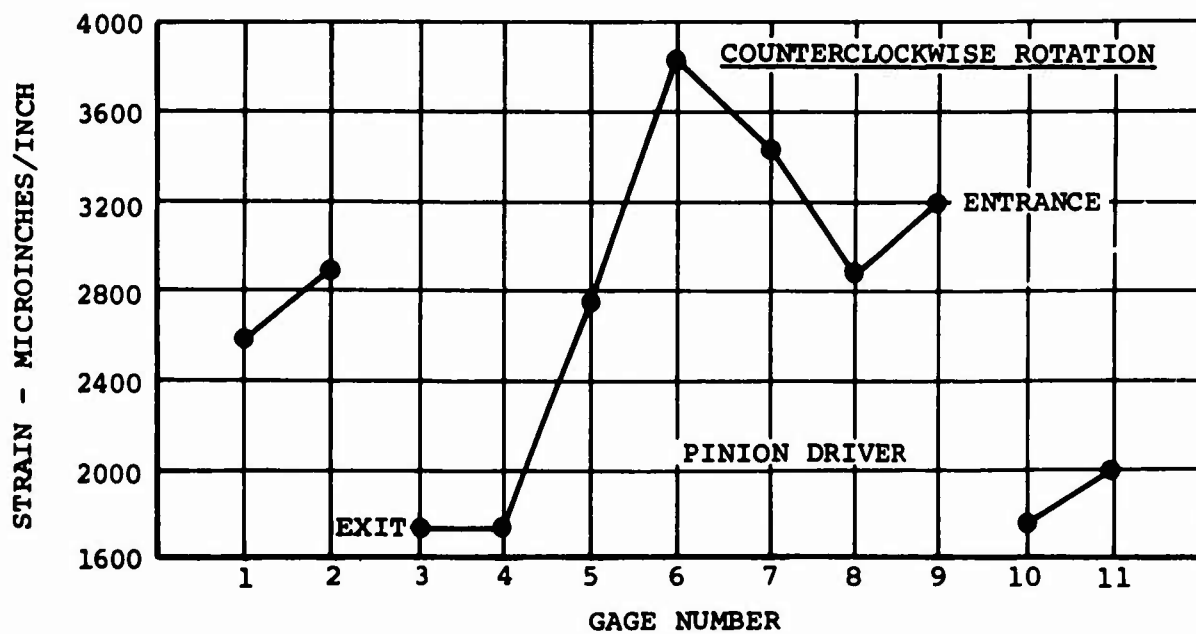


Figure 11. Maximum Strain at 32,550 Inch-Pounds Torque.

COMPARISON OF STRAIN SURVEY RESULTS WITH RESULTS IN USAAVLABS
TECHNICAL REPORT 66-8

Comparison of the results of this strain survey with the results of the strain survey accomplished during the previous contract indicates that the current SK16207 conformal gears recorded larger strains. Load levels for the two programs were not the same; the present loads were 1.42 times greater than those for the previous program. The resulting maximum strain readings were 1.84 times greater for the current program. This can be attributed to the fact that the present tooth form has a larger radius of curvature and more tooth height.

The ratio of the minimum strained and maximum strained axial sections is in the order of two to one. This result closely correlates with the results of the previous test program where the tooth geometry was different in some respects to the present test gears. Comparison of the results of both strain surveys indicates that the pattern across the tooth has a much higher strain at the entrance end. This is in agreement with the tooth pattern checks made during both programs. These checks indicated a larger load concentration at the entrance end as evidenced by the tapered pattern (heavier at the entrance end of the mesh).

LOAD TESTS

EQUIPMENT

Testing was conducted in a Vertol Division-owned regenerative (four-square) load test stand built specifically for gear research (see Figure 12). The two gear housings are designed to be rigid and stable under all loading conditions. Housings were through-bored for maximum accuracy. Torque, rpm, and lubrication can be varied over a wide range.

Accelerometers were attached to both the slave and test gearboxes and connected to a dual-channel oscilloscope. Changes in vibration signatures for both gear housings can be observed simultaneously and used as an indication of impending gear failure.

Each gearbox lubrication system includes an individual reservoir, pump, control, indicators, filler, and jets. All controls and indicators are mounted in the console. Lubrication oil volume is controlled by a bypass valve. Temperature is measured both in and out of the test gearbox. Pressure is measured at a point just upstream of the jets.

Atlantic Refining Company Premier 12 gear lubricant was used throughout the test program; it contains extreme pressure additives in the form of lead soaps. Vertol Division has used Premier 12 in previous gear research testing.*

Lubrication was provided by oil jet on the out-of-mesh side. The flow rate was approximately 2.0 gallons per minute. Lubricant pressure, temperature, and viscosity were controlled to the following values during the tests:

1. Operating Temperature: $130 \pm 15^{\circ}\text{F}$ Oil In
2. Operating Pressure: 50 ± 15 psi
3. Viscosity at Operating Temperature: 300 SSU

* Mack, J.C., D'Angelo, B., EVALUATION OF THE EFFECT OF GEAR TOOTH GRINDING METHODS ON FATIGUE LIFE OF SPUR GEARS, Boeing Document No. R-342A, the Vertol Division of Boeing, Morton, Pennsylvania, August 1964.

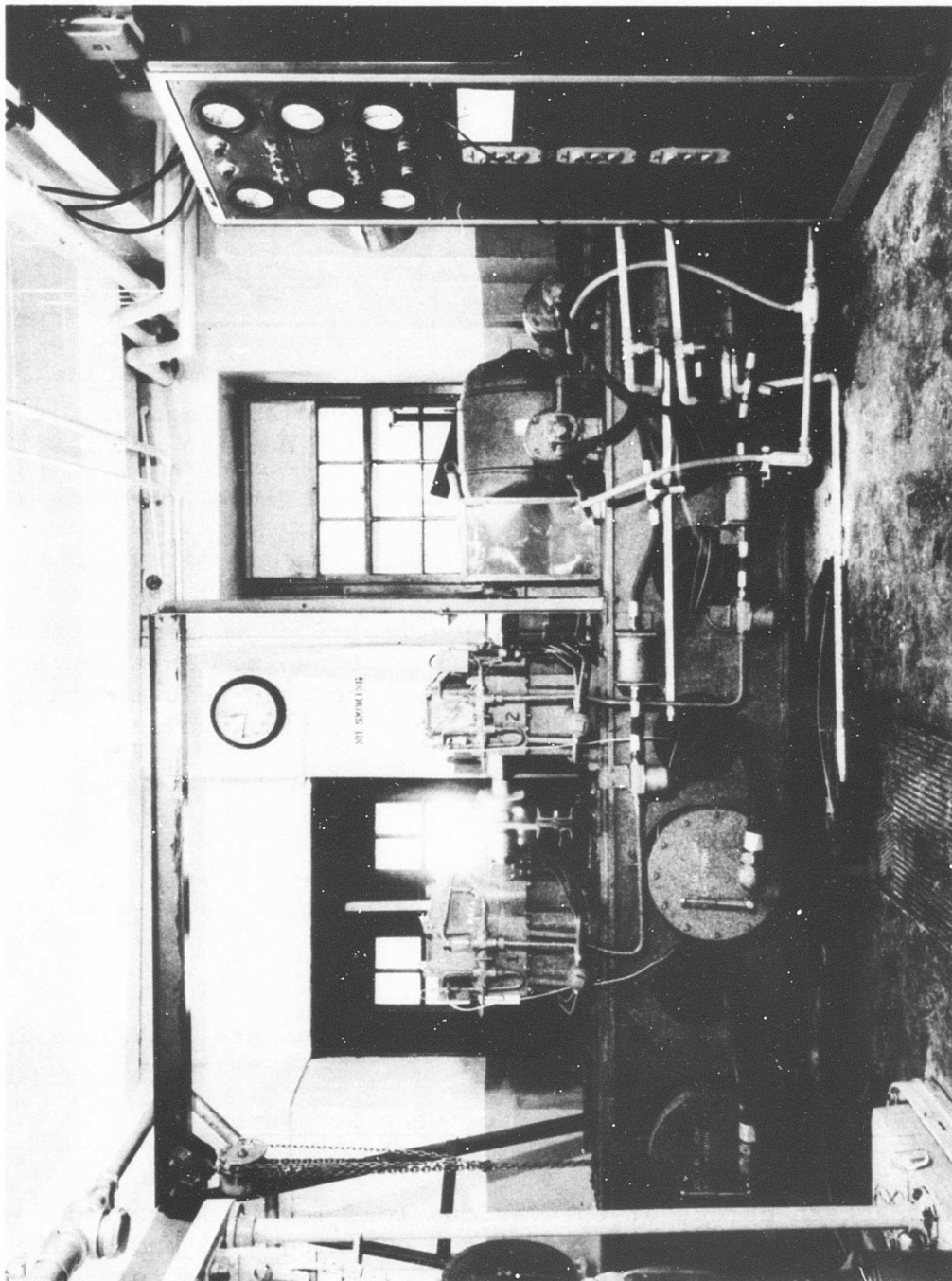


Figure 12. Gear Research Test Stand.

Variables

The primary test variables were shaft torque and test cycles (time). Gear load is a function of shaft torque.

Torque

Torque was applied with a lever system at the beginning of each test run. Torque levels were observed on a Strainert instrument (see Figure 13). The procedure for this test program was to check torque after 30 minutes of running, then to recheck every 4 hours during the test. Deviation from the initial target torque was held to ± 5 percent.

The torquemeter shaft was calibrated before and after the test series on a deadweight torsion test machine (see Figure 14). The same instrument was used during the calibration and testing. Recalibration curves agreed with the initial curve within 2 percent.

Test Cycles

Test cycles were determined by a log record of running time and an elapsed time meter in the test stand console. An electric motor driving the input shaft through a toothed belt maintained pinion speed at 3660 rpm (219,600 tooth-loading cycles per hour).

Procedure

At the initial installation of both gear sets, static tooth patterns were recorded at various load levels (see Figure 15).

RESULTS

The complete results of the load test program are contained in Table V.

Static Pattern Check and Run-in of Gear Set Number 1 (SK16207-3, Pinion and SK16207-4, Gear)

Static pattern checks at the various load levels indicated a tapered tooth pattern, larger at the entering side and smaller at the exiting side (see Figure 16). The pinion pattern extended further down the tooth flank than did the gear pattern. The gear tooth pattern was at or beyond the break edge radius

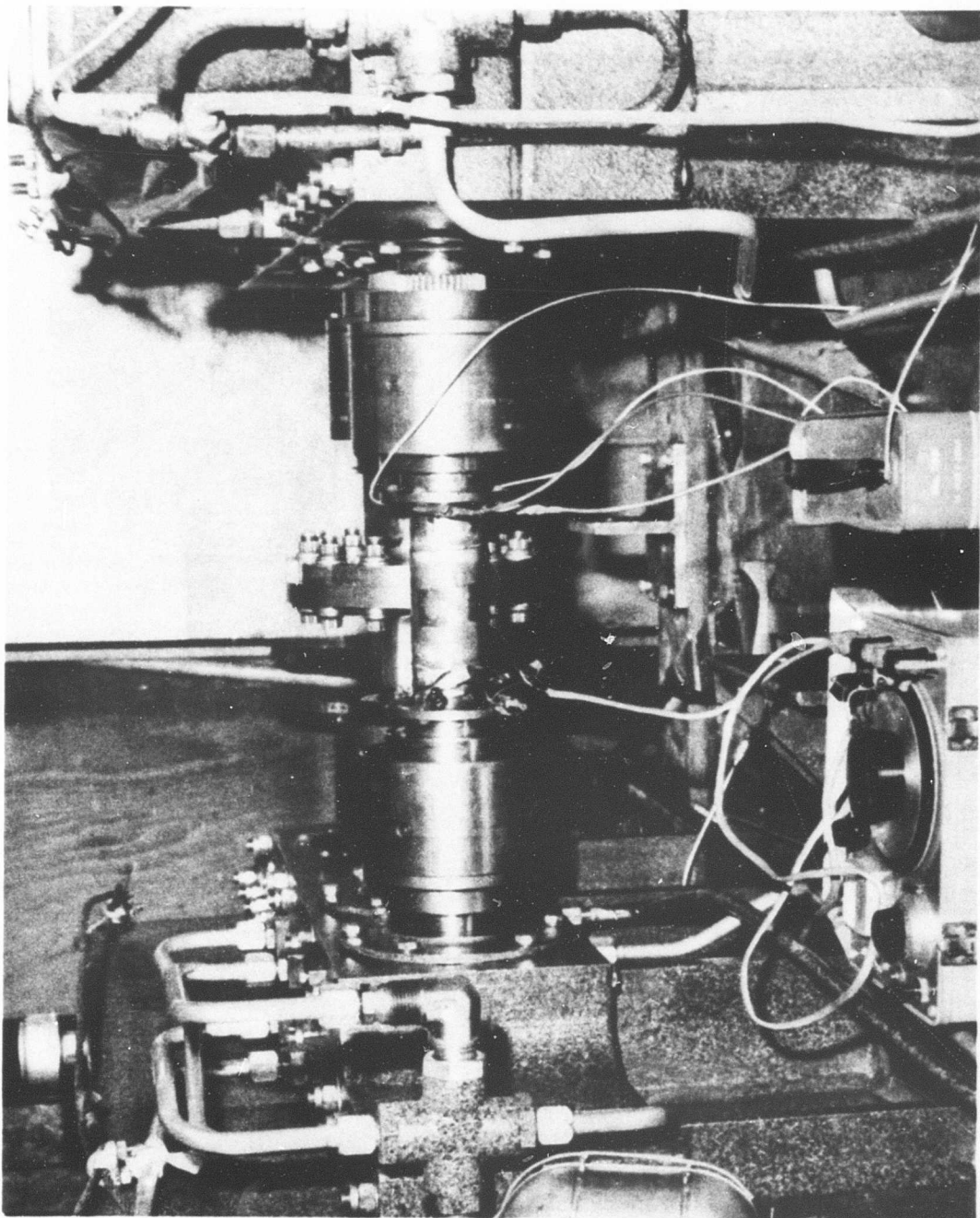


Figure 13. Test Stand Torquemeter Installation.

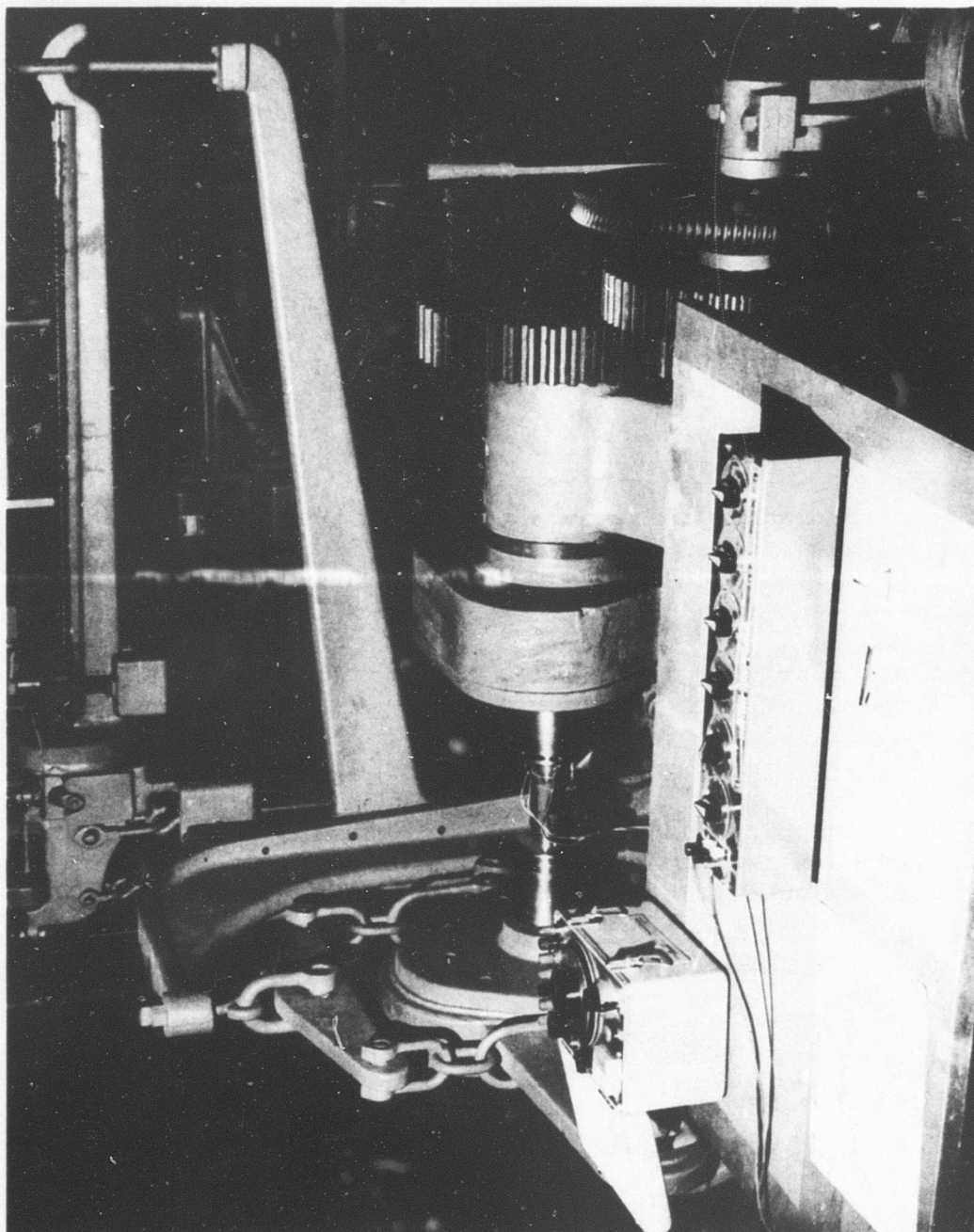


Figure 14. Deadweight Torsion Test Machine.

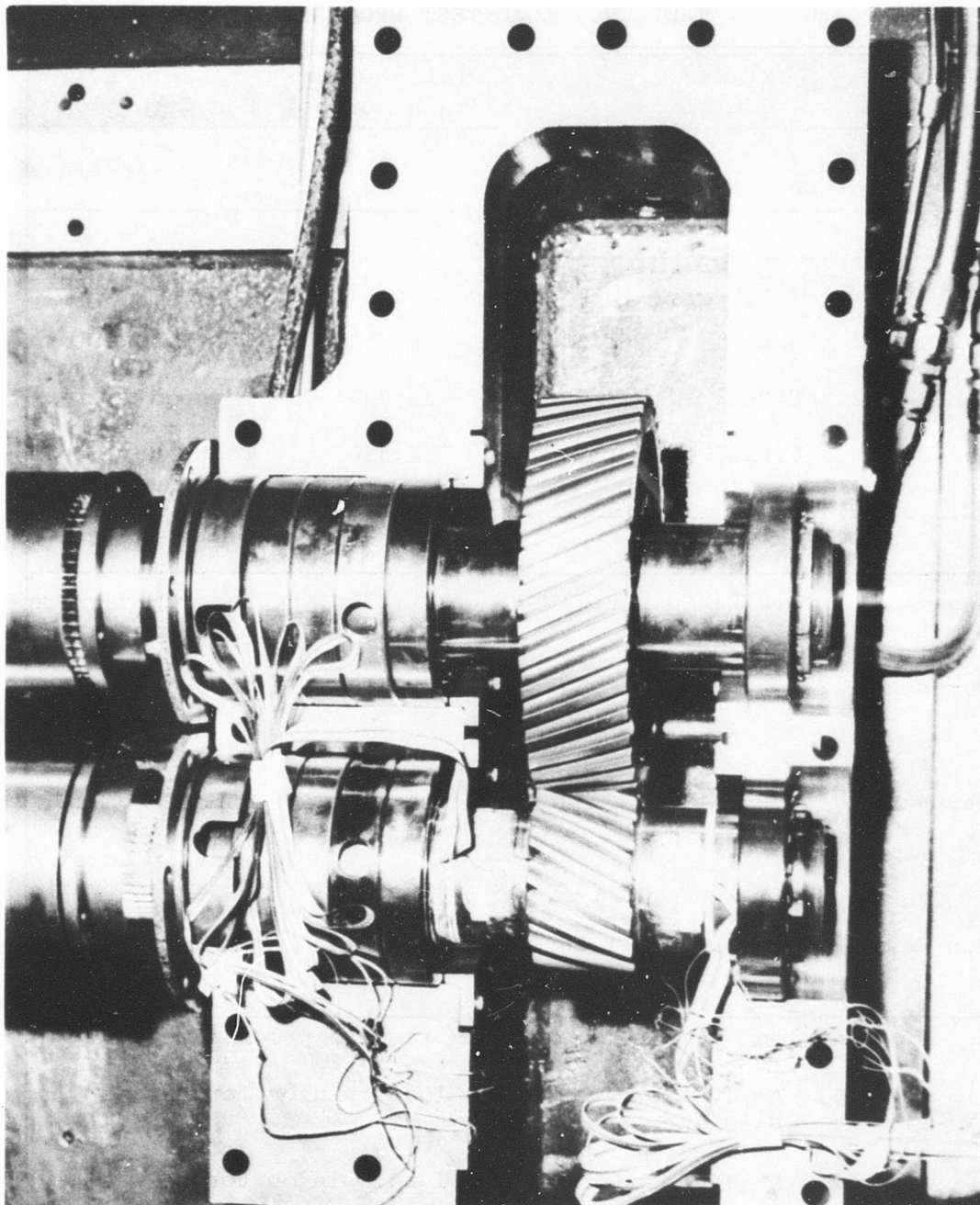


Figure 15. Installation of Conformal Gears - Static Pattern Checks.

TABLE V. LOAD TEST PROGRAM				
Test Run	Stress Cycles (millions)		Pinion Torque (inch-pounds)	End Condition
	Pinion SK16207-3	Gear SK16207-4		
Set No. 1				
Run-in	0.8784	0.37	4,380	Tapered pattern
Run No. 1	6.0	2.53	8,760	Tapered pattern
Run No. 2	6.0	2.53	13,650	Tapered pattern
Run No. 3	3.623	1.53	14,850	Tooth failure, gear member
Set No. 2				
Run-in	0.8784	0.37	4,475	Tapered pattern mild scoring
Run No. 1	6.0	2.53	8,410	Tapered pattern slight increase in scoring
Run No. 2	6.0	2.53	12,380	Tapered pattern small improve- ment in scoring (healing over)
Run No. 3	2.987	1.26	15,610	Tooth failure, gear member
Target load levels established for the program were: 1. At 100-percent design load, pinion torque was 8450 inch-pounds. 2. At 150-percent design load, pinion torque was 12,675 inch-pounds. 3. At 175-percent design load, pinion torque was 14,788 inch-pounds.				

at the outside diameter, at the higher load levels. Visual inspection after the initial run-in (at 50 percent of the design load) did not reveal any adverse conditions in regard to surface distress (see Figure 17a).

Load Test Runs of Gear Set Number 1

Test Run Number 1

Run number 1 was completed at 100 percent of the design load for 6 million pinion cycles. Visual inspection at the completion of this run did not reveal surface distress. There was some evidence of an improvement in the tooth pattern. Taper at the entrance and exiting ends was less pronounced (see Figure 17b).

A close examination of the tooth patterns revealed a local heavy bearing area on the lower end (entrance side) of the pinion and along the top of the gear member for the full length of the pattern.

Test Run Number 2

Run number 2 was completed at 150 percent of the design load for 6 million pinion cycles. Visual inspection revealed that at the lower portion entrance side of the pinion, there was an indication of a secondary local heavier bearing area within the original pattern, approximately 0.4 inch in length and 0.03 inch wide (see Figure 17c). On the gear member, the secondary local bearing area was apparent just below the break edge, at the outside diameter for the length of the original pattern. Patterns on the gear member ended approximately 0.05 inch short of the end of the gear at the exit side. Visual inspection of both members did not reveal any indications of surface distress.

Test Run Number 3

Run number 3 was conducted at 175 percent of the design load level. This test run was terminated at 3,623,400 pinion cycles, due to bending failure of one tooth on the gear member. Subsequent Magnaglo inspection of this gear member did not reveal any other teeth with failure indications. Vertical scratches on every other tooth profile of the pinion members (in the first inch of face width) were attributed to the tooth failure on the gear member. Magnaglo

STATIC GEAR PATTERNS - GEAR SET #1

PINION-P/N

SK16207-3, S/N 1

GEAR-P/N

SK16207-4, S/N 1

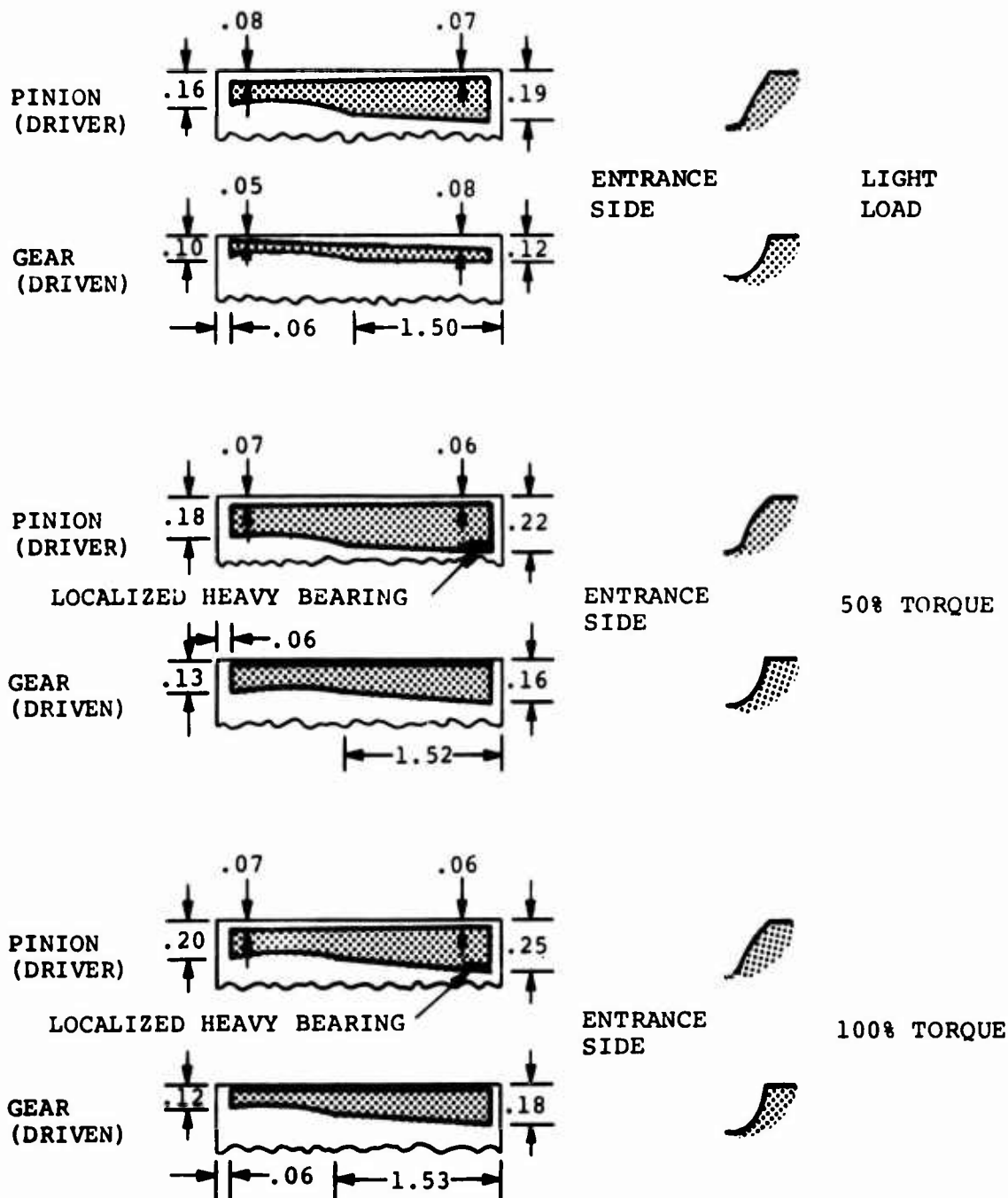


Figure 16. Static Gear Patterns - Gear Set No. 1.

GEAR PATTERNS AFTER RUNNING - GEAR SET #1

PINION-P/N SK16207-3, S/N 1

GEAR - P/N SK16207-4, X S/N 1

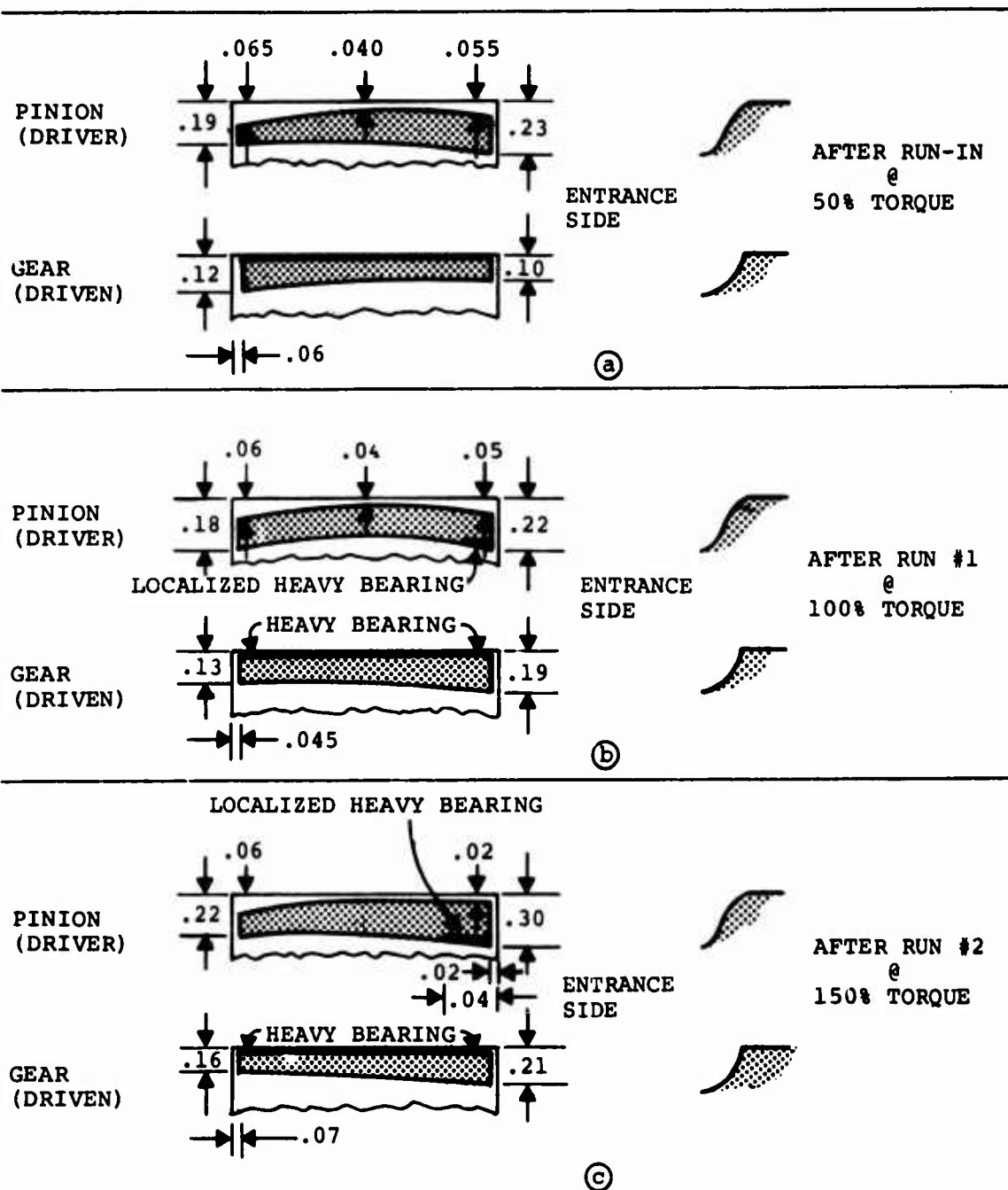


Figure 17. Gear Patterns After Running - Gear Set No. 1.

inspection of the pinion did not reveal any cracks. Approximately 20 minutes before failure, there was an increase in signature amplitudes and a change in the original pattern.

Load Test Runs of Gear Set Number 2
(SK16207-7, Pinion and SK16207-4, Gear)

Static Pattern Check and Run-in Gear Set Number 2

Static pattern checks at the various load levels indicated a tapered pattern at both the entering and exiting ends (see Figure 18). At the 100-percent load level, this condition was not as evident as the result at the lower load level. The pattern on the gear member moved out to the break edge at the outside diameter at the higher load levels. Visual inspection after the initial run-in (50 percent of the design load) revealed very light scoring on both pinion and gear members (Figure 19a).

Test Run Number 1

Run number 1 was completed at 100 percent of the design load for 6 million pinion cycles. Visual inspection upon completion of this run revealed a slight progression in the scoring condition at the completion of the 4-hour run-in. Some improvement in the tooth pattern was noted. The taper at the entrance and exiting ends was not as pronounced. There was a small, heavy bearing area at the exit end of the gear member, approximately 0.3 inch wide, running vertically along the tooth flank (Figure 19b).

Test Run Number 2

Run number 2 was completed at 150 percent of the design load for 6 million pinion cycles. Visual inspection upon completion of this run revealed a slight improvement in the scoring condition (healing over). On the gear member, there was an indication of a secondary local heavier bearing within the original pattern at the entrance end of mesh (Figure 19c).

Test Run Number 3

Run number 3 was conducted at 175 percent of the design load level and was terminated at 2,986,560 pinion cycles, due to a bending failure of one tooth on the gear member.

STATIC GEAR PATTERNS - GEAR SET #2

PINION - P/N SK16207-7, S/N 2
GEAR - P/N SK16207-4, S/N 2

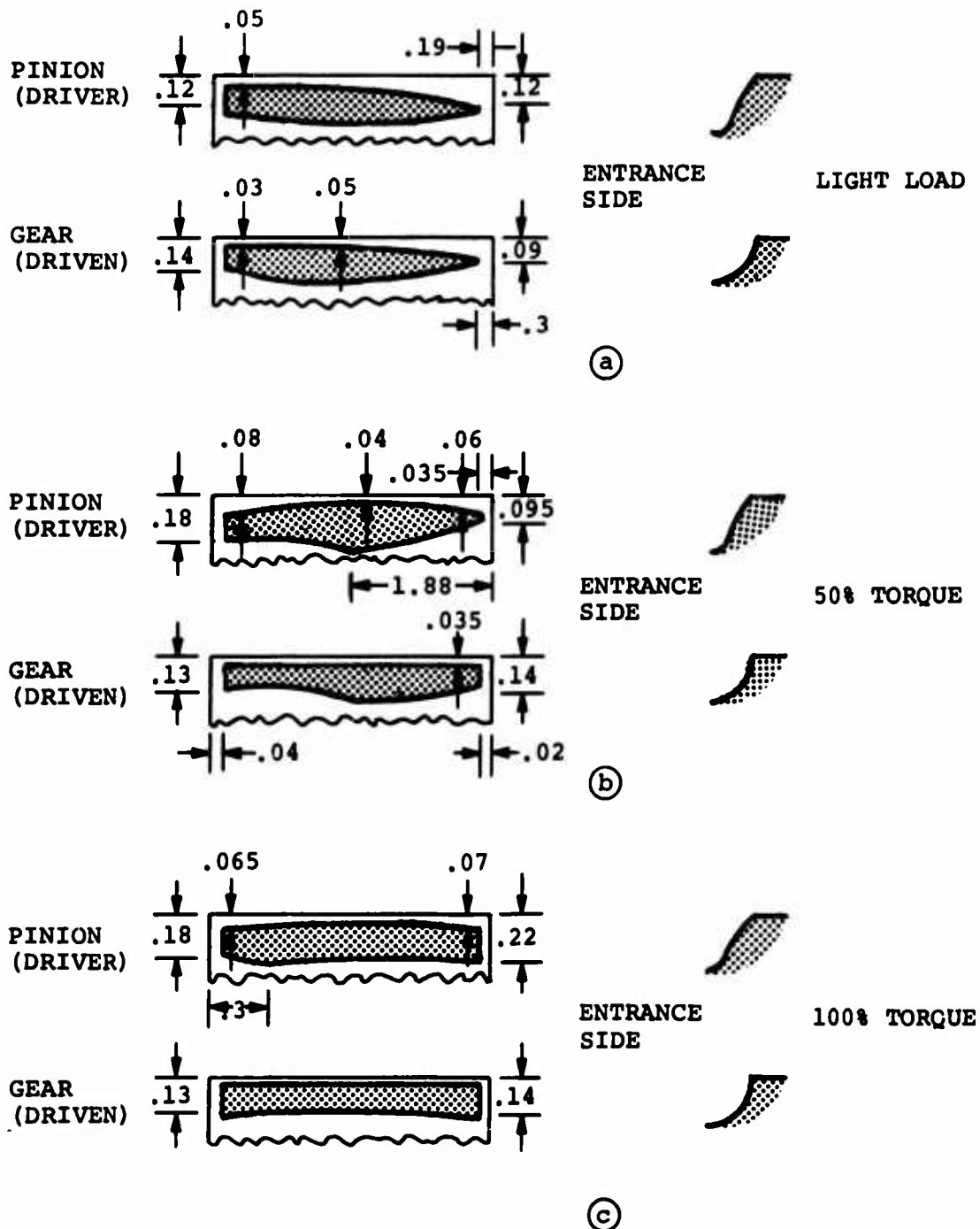


Figure 18. Static Gear Patterns - Gear Set No. 2.

GEAR PATTERNS AFTER RUNNING - GEAR SET #2

PINION - P/N SK16207-7, S/N 2
GEAR - P/N SK16207-4, S/N 2

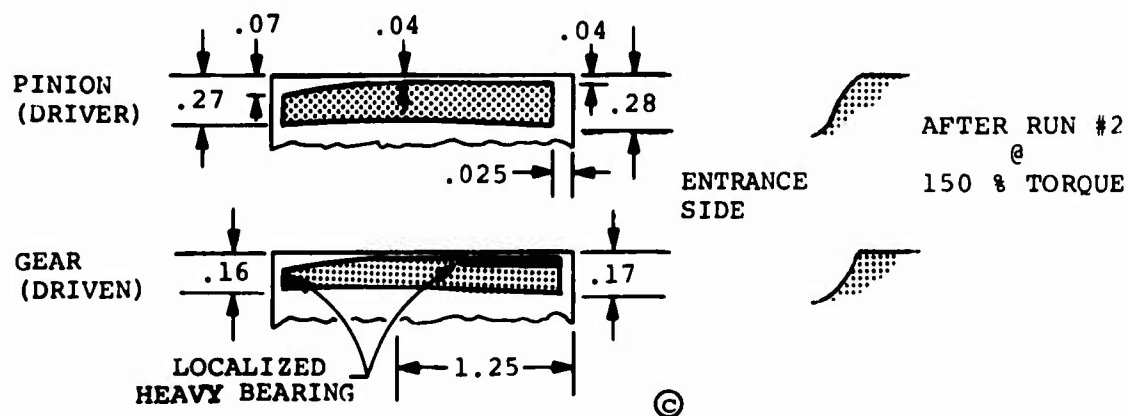
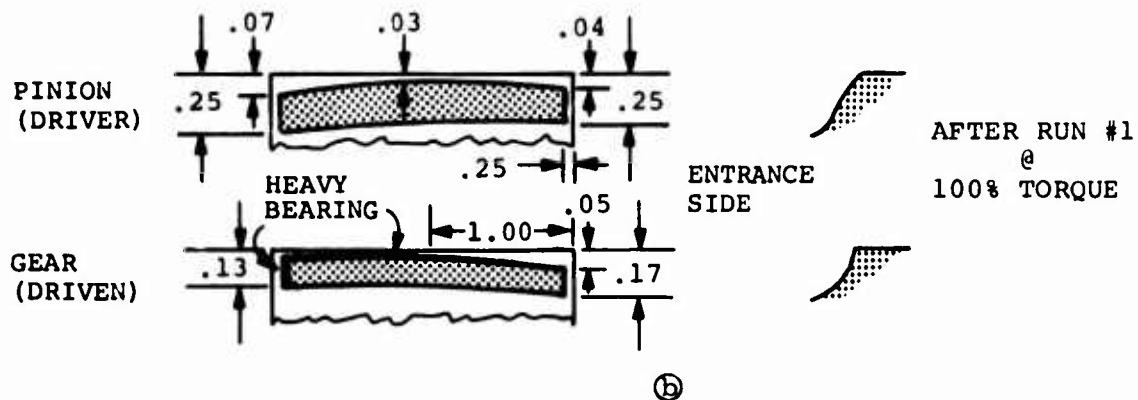
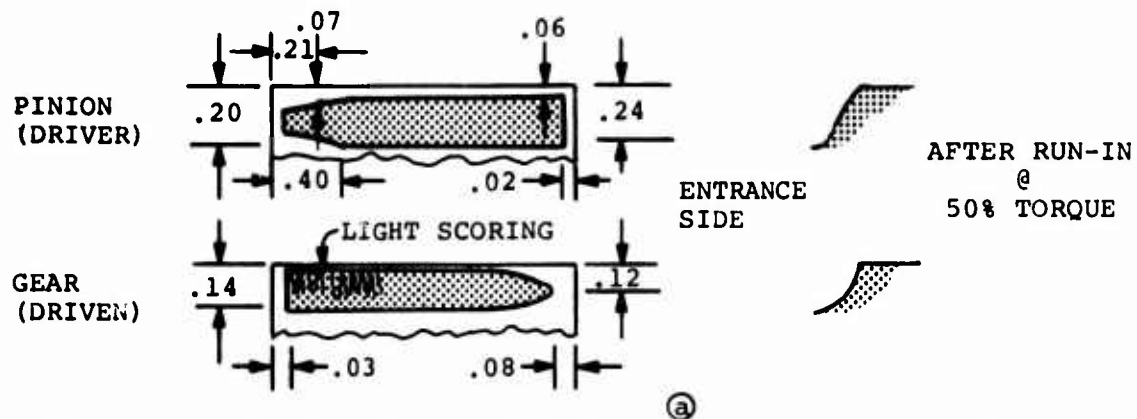


Figure 19. Gear Patterns After Running - Gear Set No. 2.

Subsequent Magnaglo inspection of the gear member did not reveal failure indications on any other teeth. The vertical scratches on every other tooth of the pinion member in the first inch of face width were attributed to the tooth failure of the gear member. Magnaglo inspection of the pinion did not reveal any failure indications. A change in trace trends on the oscilloscope were seen prior to failure. They were similar to the trends seen on the first test set.

METALLURGICAL FINDINGS

The metallurgical analysis of gear sets numbers 1 and 2 made upon completion of testing included the following (Figures 20 through 31):

1. Nital Etch Examination

Nital etch examination showed no evidence of grinding damage on either gear set.

2. Microhardness Traverse

A microhardness traverse was made on the failed tooth of the gear member of both sets. The results are plotted in Figure 32. The effective case depth of the gear from set number 2 was 0.014 inch over the drawing specifications.

3. Microexamination

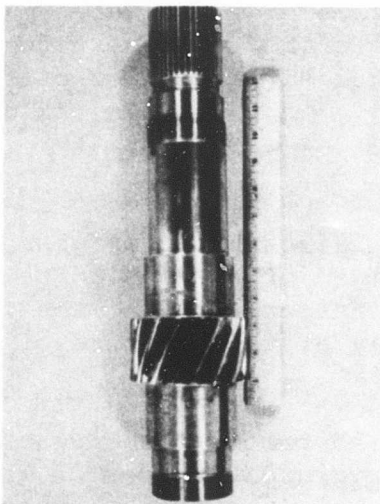
A microexamination for retained austenite, carbide network, and cleanliness showed that the material was satisfactory in all respects.

4. Fractographic Examination

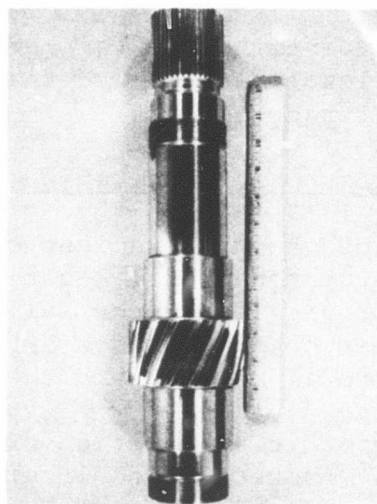
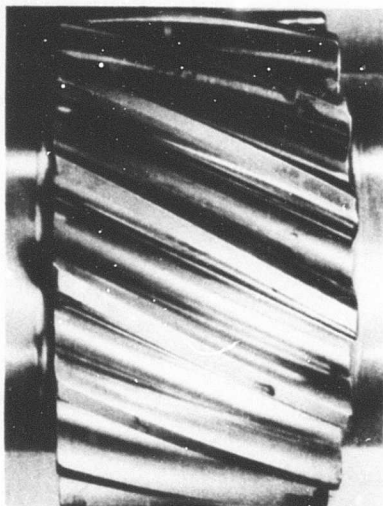
A fractographic examination of the failed tooth of gear member set number 1 located the origin of fatigue to be on the drive side of the fillet. The failure origin was located on the surface approximately at the juncture of the profile radius and the undercut.

A fractographic examination of the failed tooth on the gear member of set number 2 also located the origin of fatigue on the drive side of the fillet. The origin was subsurface, approximately 0.042 inch below the junction of the profile radius and the undercut, and 0.008 inch below the surface (Figure 33).

PINIONS



Set No. 1
SK16207-3
PINION



Set No. 2
SK16207-7
PINION

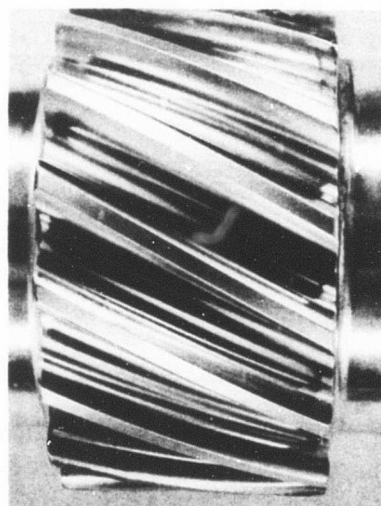


Figure 20. Test Pinions After Final Load Run.

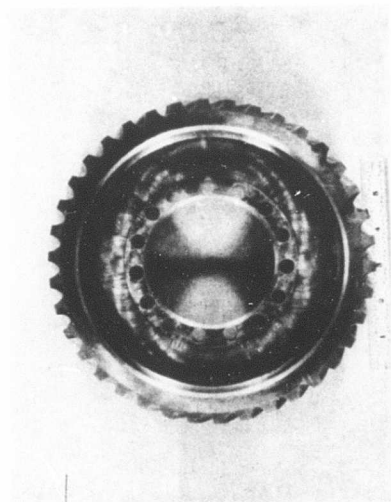


Figure 21. Conformal Gear (SK16207-4) - Figure 22. Fractured Tooth of Conformal Gear - Set No. 1.

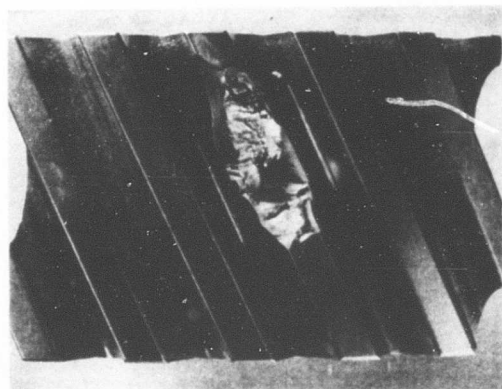
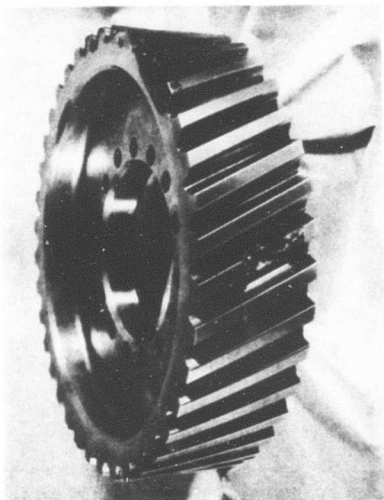


Figure 23. Fractured Face of Gear Set No. 1.

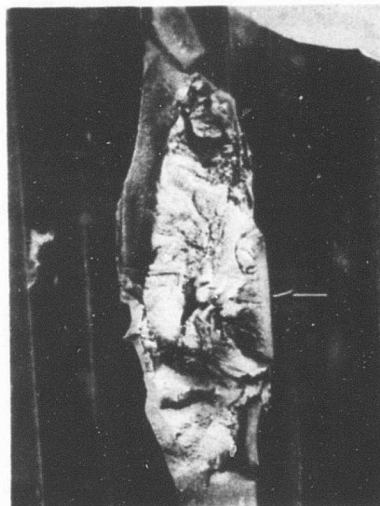


Figure 24. Closeup View of Fractured Tooth. (Arrow Indicates Origin.)

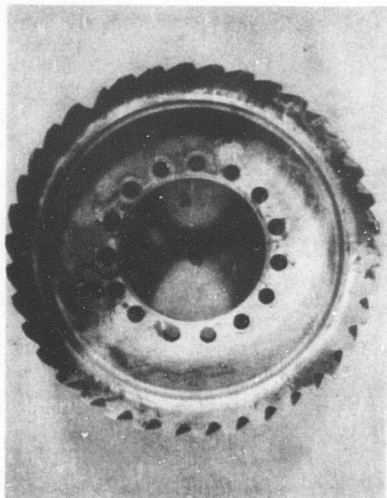


Figure 25. Conformal Gear (SK16207-4) - Set No. 2.

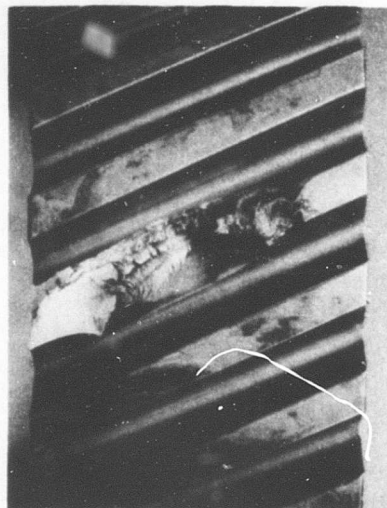


Figure 26. Fractured Tooth of Conformal Gear - Set No. 2.

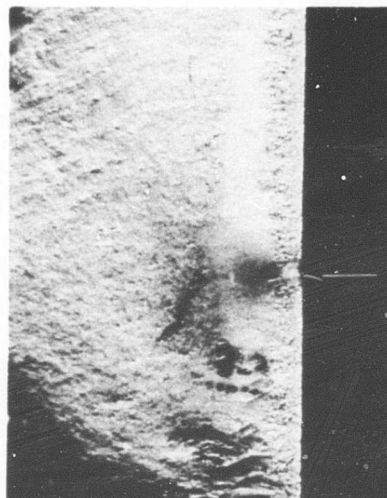


Figure 27. Fractured Face of Conformal Gear - Set No. 2. (Arrow Indicates Subsurface Origin.)

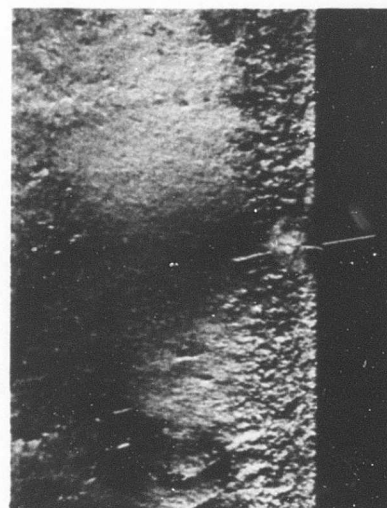
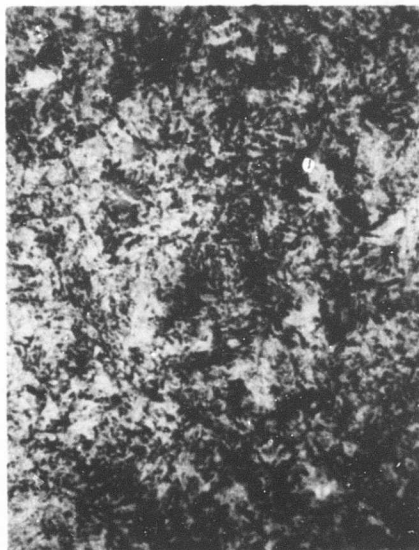
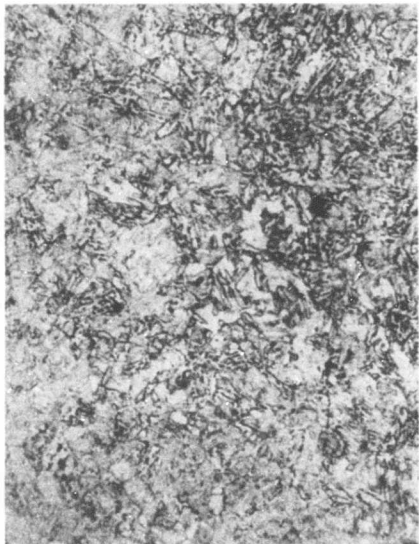


Figure 28. Origin Area With Arrow Denoting Fatigue Area.



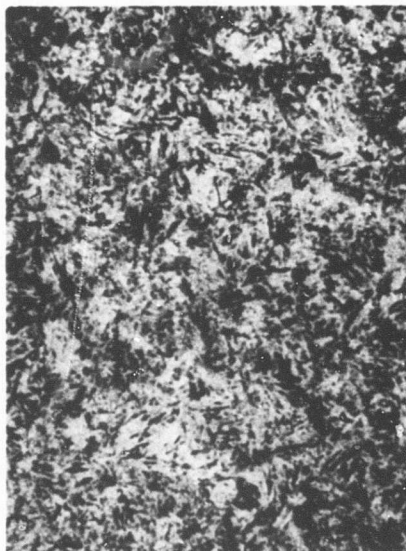
500 MAG

Figure 29. Case Microstructure of Set No. 1 (Origin Area).



500 MAG

Figure 30. Core Microstructure of Set No. 1 (Origin Area).



500 MAG

Figure 31. Case Microstructure of Set No. 2 (Origin Area).

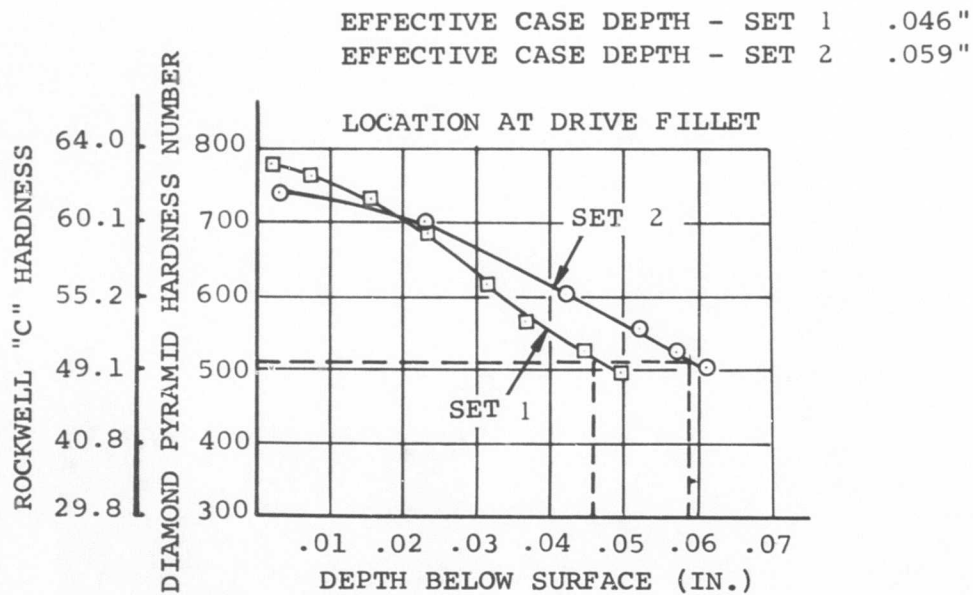


Figure 32. Graphical Analysis of Gear Sets Nos. 1 and 2 for Effective Case Depth Evaluation.

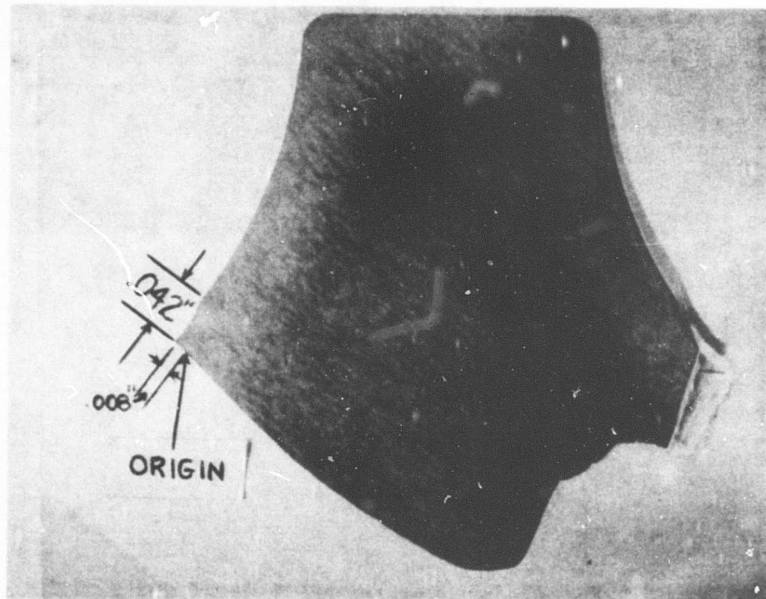


Figure 33. Origin Location of Failed Tooth Gear Member - Set No. 2.

DISCUSSION OF TEST RESULTS

As a result of the problems encountered in the previous test program (USAAVLABS Technical Report 66-8), the following design changes were incorporated in the current test gears:

1. The profile radius was increased by 100 percent on both the pinion and gear.
2. The profile mismatch was increased between the pinion and gear by 100 percent.
3. The pinion member of one gear set was crowned 0.0005 to 0.0008 inch.

By increasing the profile radius on both members, the surface contact area was also increased. This design change resulted in a substantial improvement in surface load capacity of the current test gears as evidenced by the lack of any pitting at the 175-percent load level.

The combined effect of the increased profile radii and increased profile mismatch resulted in a solution to the tip impingement problem on the pinion member of the previous test gear sets. In addition, the effect of heavy load concentration at the entrance end of engagement was diminished for the load levels tested.

Gear set number 1 successfully completed a load run of 150 percent of the design load for 6 million cycles. During the testing of this set, there were no indications of surface distress evident at any time. The detrimental condition of entrance distress experienced during previous testing (USAAVLABS Technical Report 66-8) was not evident in this test set.

Set number 2 successfully completed a load run of 150 percent of the design load for 6 million cycles. During the testing of this set, mild scoring was evident at the lower load levels. This condition healed over at the higher load levels and may be attributed to the lead relief (crowning) present on the pinion. At the lower load levels, tooth deflection was not sufficient to allow the entire face width to accept load; however, entrance distress was not evident on this set.

Failure of both sets numbers 1 and 2 (one tooth on the concave gear member) at 175 percent of the design load was attributed to a fatigue failure in bending on the drive side in the fillet area. In the previous program (USAAVLABS Report 66-8), both sets of test gears experienced surface distress (pitting) at a load level approximately equal to the 100-percent load level as established for the test gears in the current program. Visual inspection of the former test gears revealed a heavy load concentration at the entrance end of engagement where pitting was first encountered. A tooth bearing pattern which tapered toward the exit end of engagement was further evidence of this load concentration. In addition, tip impingement by the gear member occurred in the vicinity of the root on the pinion, resulting in a heavy localized bearing.

The gear member with concave profiles failed in bending. Since indications of surface distress were not present on the pinions or gears at the conclusion of testing, it is evident that the current design favored contact strength capacity. In any future designs, it will be necessary to redistribute tooth thickness to favor the gear member in order to obtain a better balance between bending and contact strength capacity.

Although significant design improvements were achieved in the current test gears, the tapered tooth bearing patterns which were encountered in the previous test gears were also evident to a lesser degree in the current gear sets.

On the basis of load concentration at the entrance end of engagement, strain gage survey results, and tapered tooth patterns as experienced on the previous test gears, one of the current test pinions was crowned (SK16207-7). The test results of gear set number 2, with the crowned pinion, did not indicate an improvement over that of set number 1. At the beginning of load running, set number 2 encountered light scoring on both members. However, operation at the higher load levels resulted in a healing over of the initial scoring condition. Scoring at the lower load levels may be attributed to the pinion crowning. The teeth may not have deflected sufficiently to fully load the crown ends, thereby resulting in a concentration of load in the remaining face width. The effect of crowning on entrance distress and tip impingement cannot be definitely ascertained because both gear sets (crowned and uncrowned) successfully operated without encountering these problems.

In order to provide a better understanding and possible solution to the tapered tooth patterns, the following investigations were made:

1. Inspection of the test box bores for alignment, concentricity, and center distance was made and found to be within the drawing tolerances. Center distance and runout checks were made with the test gears and couplings installed in the test stand. The results indicated that these parameters were within acceptable limits. All of the inspections were performed by the Vertol Division's Quality Control department.
2. Shaft deflection and gear face runout were inspected during the static pattern checks and were found to be negligible.
3. Lead-error checks of both gear sets indicated that deviations were within 0.0002 inch across the entire face width.
4. The conformity of the pinion profile radius was inspected utilizing the special inspection fixture; it was found to be within 0.0003 inch (on the drive side) across the entire face width.

These results indicate that all parameters are within acceptable limits; therefore, their effect on the tapered pattern condition may be considered negligible.

The strain gage survey indicated that gage number 9, located on the middle tooth at the entering side, had approximately 80 percent more strain than gage number 3 at the exit end of the same tooth. This result was repeated for both load levels tested. An explanation for differences in the amount of strain at the ends of the middle tooth may be in the amount of tooth material available for backing. At the extreme entrance end, the helix angle limits the backing material available at the load point to approximately one-half of the tooth thickness. When the load shifts to the exit (opposite) end, the helix angle has no effect, since the full tooth thickness is available for backing behind the load; therefore, it appears reasonable to assume that the patterns at the entrance end would be heavier than at the exit end.

Figures 34 and 35 reflect the bending and contact stress levels for a comparable involute gear, with the same pitch diameter,

number of teeth, diametral pitch and face width. However, in good design practice, 16-tooth involute pinions are not normally used. A finer pitch with more teeth would be standard practice. The bending stress resulting would thus be higher than is shown in Figure 35.

The teeth on all the test gears were shot peened (flanks and fillets) per the appropriate Vertol specification. The effect of shot peening primarily is to increase the residual compressive stress at the fillet. However, visual examination of the test gears indicates that the shot peening coverage in the fillet area of the gear member could be improved. This may be attributed to the inaccessibility of the root area to the shot and the effect of shot reverberating from the flanks, thereby blocking the path to the root. In future designs a reciprocating wand directly in the undercut area will vastly improve the shot distribution in the fillet area.

An indication of gearing efficiency may be obtained by a comparison of the temperature rise of the lube oil and oil flow of the slave box and the conformal test box. The heat generation in the conformal test box was 13 percent lower than the heat generation in the involute slave box. This result appears to indicate an improved efficiency for the conformal gear as compared to the slave (involute) gears. This is in agreement with the analytical predictions of improved efficiency for the conformal tooth form, due to a constant sliding velocity, as contrasted to a variable sliding velocity for the involute tooth form.

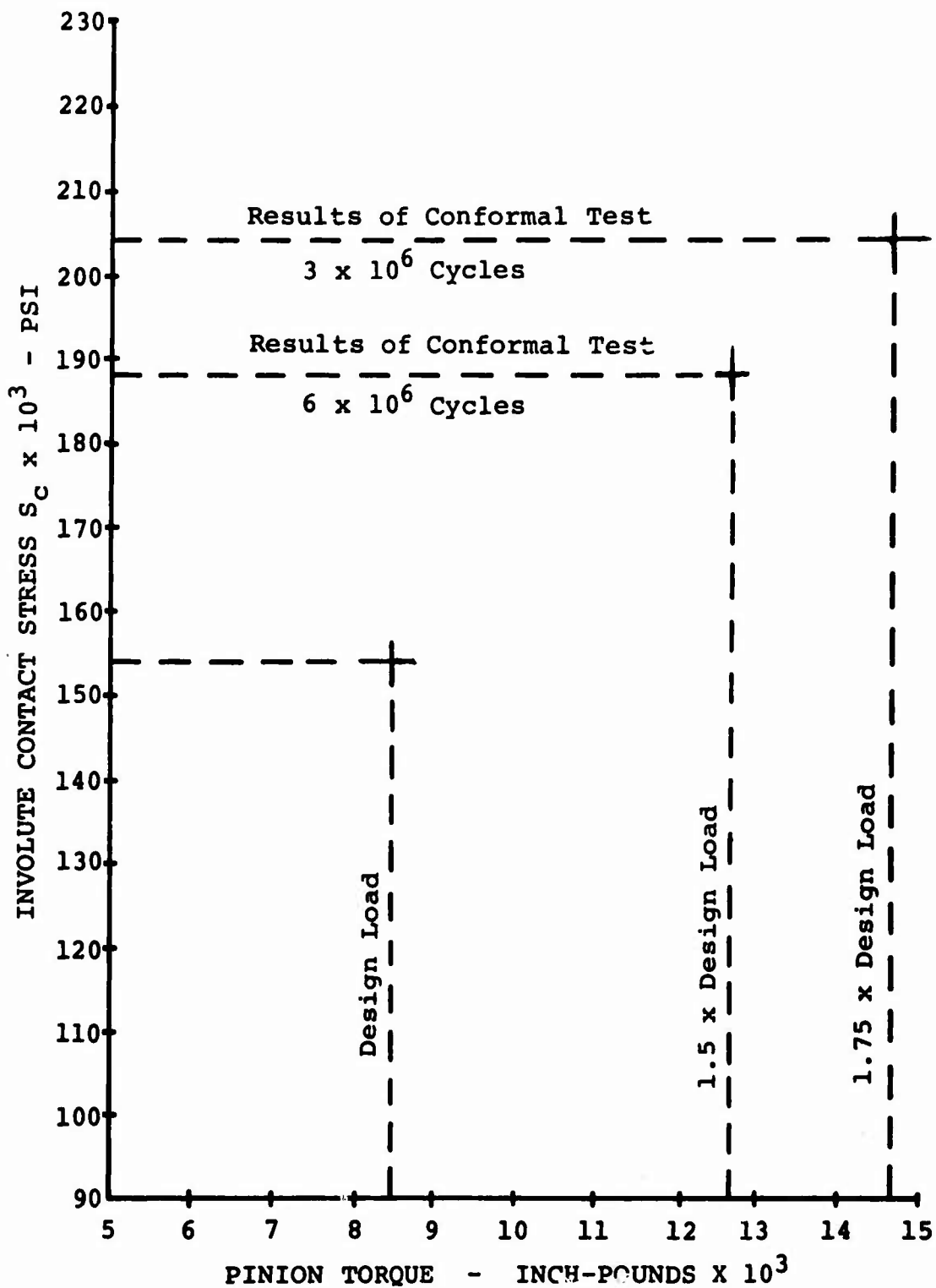


Figure 34. Involute Contact Stress Levels at Pitch Line Vs. Torque.

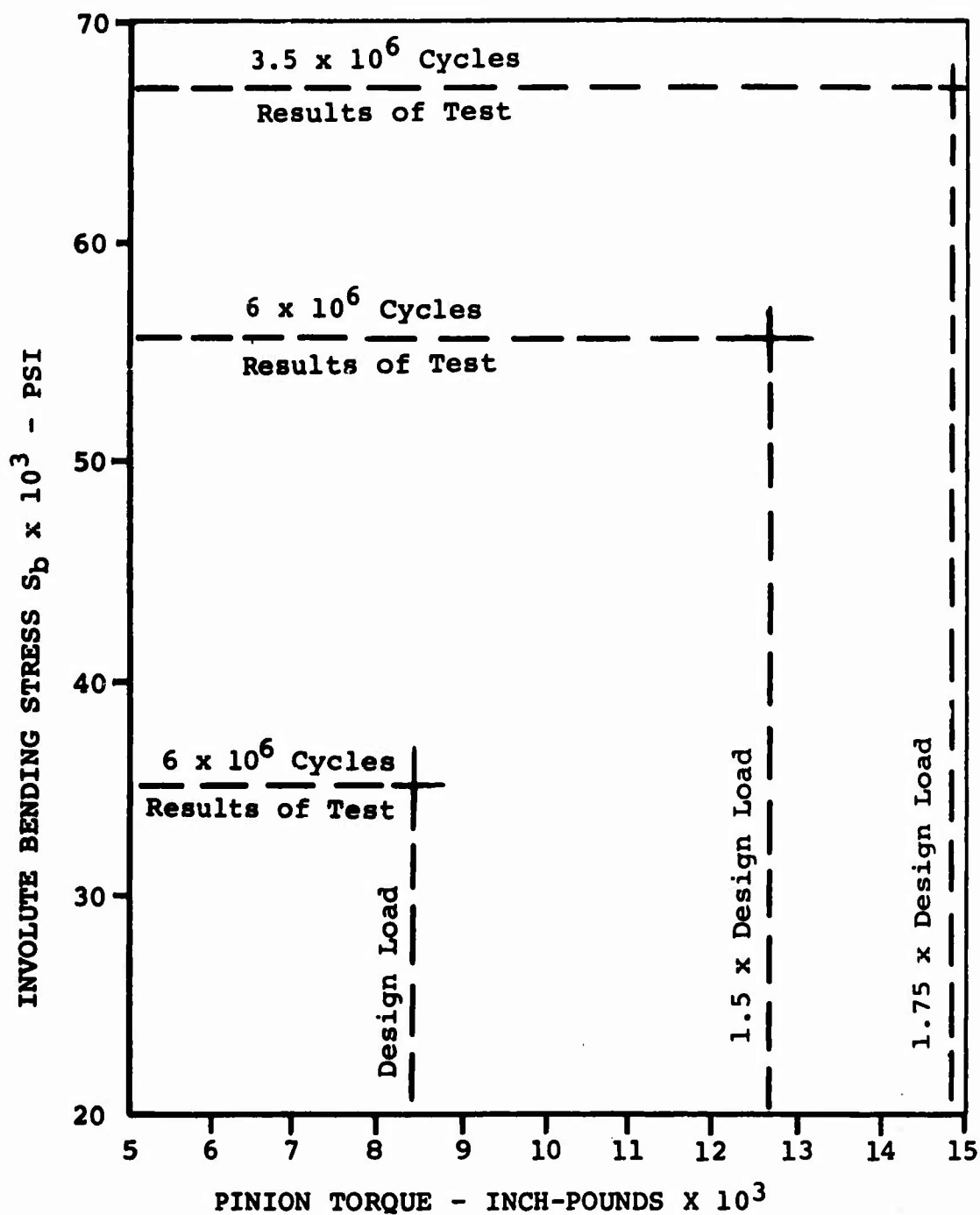


Figure 35. Involute Bending Stress Levels Vs. Pinion Torque.

CONCLUSIONS

1. The conformal test gears successfully demonstrated a load-carrying capacity equal to 150 percent of the established design load.
2. Inspection techniques have been determined which are capable of providing dimensional control of conformal gears commensurate with that of aircraft involute gears.
3. Load-carrying capacity can be improved by a better balance of contact and bending strength and modification of the following respective design factors: Reproportioning the tooth thickness of the pinion and gear members to improve the balance between contact and bending strength, relieving the entrance end of the pinion tooth face by machining at an angle at the ends of the teeth, thereby eliminating the condition where entrance load does not have the full tooth thickness as backing, and increasing the outside diameter to prevent the tooth bearing pattern from shifting over the top land break edge, at the higher load levels.
4. The gear member was the weaker member in bending.
5. The contact stress which was attained at the failure load level was approximately 36 percent higher than the design load (which represents normal operating design practice). It should be noted that the test gears were operated in a rigid test stand using an oil containing extreme pressure (E.P.) additives.
6. Failure of both test gear sets was attributed to a bending fatigue failure in the fillet area. The tooth surface condition of the test gears at the conclusion of the final runs indicates that the upper limit for contact strength capacity had not been attained.
7. Both test pinions (with and without crowning) in this test program operated without developing indications of entrance distress as experienced during the first program.

8. Realization of the full potential of the conformal gear for application to helicopter speed reduction gear drives will require extensive design and development programs to ascertain the effects of the following design factors:
 - a. The relative radius of curvature for maximum pitting resistance
 - b. The effect of axial overlap on load capacity and noise level
 - c. The maximum difference in profile curvature mismatch between pinion and gear to accommodate a suitable center distance tolerance and still provide maximum load capacity
 - d. The effect of pressure angle on both bending and contact strength

Determination of the effects of these design factors may require three-dimensional photoelastic analysis, single tooth bending fatigue tests, additional rotating surface fatigue tests, and additional strain surveys of rotating teeth.

APPENDIX
NOISE LEVEL TESTS - CONFORMAL GEARS

INTRODUCTION

As a further effort in the evaluation of the conformal tooth form, the Vertol Division of Boeing conducted a noise measurement program in August 1967 at the Villanova Department of Mechanical Engineering Gear Laboratory to evaluate the effects of gear tooth design on radiated sound pressure levels. The gear set evaluated was of conformal profile and 4.5 diametral pitch. These gears were run simultaneously with 9.0 diametral pitch finish ground involute spur slave gears (to aircraft standards) with the same pitch diameters in a pretorqued configuration. Although this experimental arrangement was preliminary in nature from a noise measurement standpoint, some useful information was obtained.

PROCEDURE

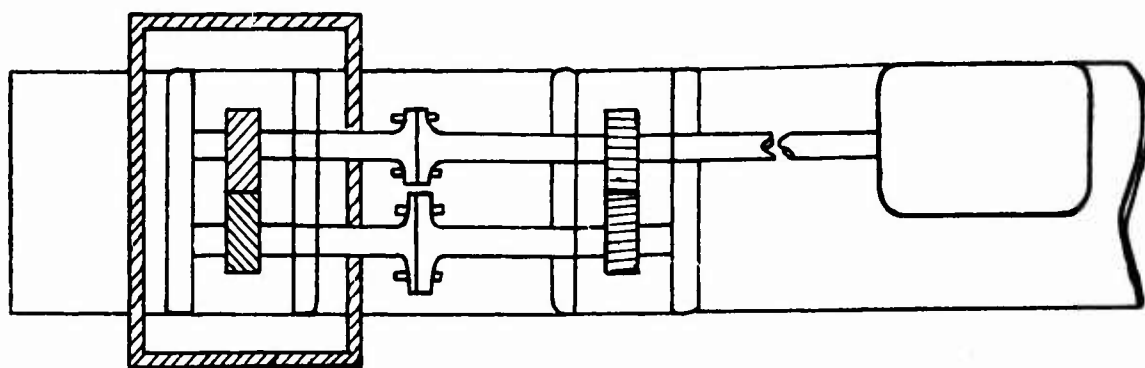
In order to isolate the noise of each gearbox, an acoustical enclosure was constructed of 1/2-inch plywood (see Figure 36). The box was lined with aircraft-type blanketing to increase the sound transmission loss and to provide a semifree field environment within the enclosure. All measurements were made by inserting the microphone through predrilled openings in the walls of the enclosure. When the survey on the slave box had been completed, the enclosure was transferred to the conformal gearbox and the survey repeated.

RESULTS AND CONCLUSIONS

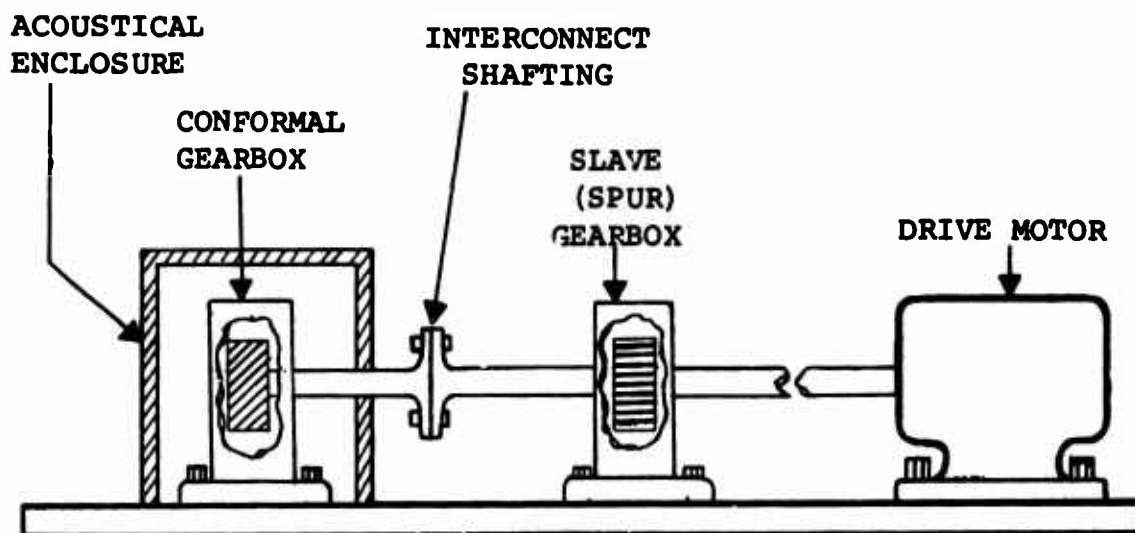
Data were obtained at 100 percent of the design torque for both gears. An analysis of the noise at two locations for each gear configuration (midposition on the top of the box and an inboard side location) was performed on a wave analyzer using a 3-Hz filter, at 0.5-inch-per-minute chart speed, and 3-inch-per-second writing speed. Although the gear mesh frequency of the spur gears was twice that for the conformal set, some general conclusions may be drawn. At the third harmonic conformal gear mesh frequency and below, the conformal gears display larger amplitude noise levels than the spur gear set (see Table VI). Above the third harmonic of the conformal gears, the level of the spur gear generally exceeds the conformal, and at integer harmonics it is as much as 20 db higher. The odd harmonics of the spur gear correspond to the midpoint between conformal

TABLE VI. NARROW BAND ANALYSIS																			
Location	Frequency	480	960	1440	1920	2400	2880	3360	3840	4320	4800	5290	5770	6250	6720	7200	7690		
	Spur Harmonic	1n	2n	3n	4n	5n	6n	7n	8n	9n	10n	11n	12n	13n	14n	15n	16n		
	Conformal Harmonic	1n	2n	3n	4n	5n	6n	7n	8n	9n	10n	11n	12n	13n	14n	15n	16n		
2	Spur Level	90	88	96	106	78	87	88	9n	90	79	64	82	68	74	64	70		
	Conformal Level	99	99	109	109	94	94	83	83	82	82	59	59	54	54	58	58		
	db re Spur	+11	+11	+3	+3	+7	+7	-15	-15	+3	+3	-23	-23	-20	-20	-12	-12		

TABLE VII. ONE-THIRD OCTAVE BAND ANALYSIS																			
Location	Frequency	25	315	40	50	63	80	100	125	160	200	250	315	400	500	630			
	Spur Harmonic	25	315	40	50	63	80	100	125	160	200	250	315	400	500	630			
	Conformal Harmonic	25	315	40	50	63	80	100	125	160	200	250	315	400	500	630			
2	Spur Level	79	77.5	73.5	78.5	82.5	88.5	88.5	78	82	87	95	80.5	83	93.5	78.5			
	Conformal Level	81	83.5	75	82.5	94.5	90.5	92	95	81.5	87.5	93	79.5	84	81	79			
	db re Spur	+2	+2	+1.5	+4	+12	+2	+3.5	+17	-0.3	+0.5	-2	-1	+1	-12.5	+0.5			
Location	Frequency	800	1000	1250	1600	2000	2500	3150	4000	5000	6300	8000	10000						
	Spur Harmonic	2	2	3	4	5,6	7	8,9	10,11	13,14	15,16	17,18	19,20						
	Conformal Harmonic	1	1	2	2	3	3	4	5,6	7,8	9,10	11,12	13,14						
2	Spur Level	84	93	92.5	98.5	109.5	92	95	101	82.5	89	81.5	75.5						
	Conformal Level	83.5	97	90.5	95.5	108.5	92.5	97	93	84	78.5	75.5	70.5						
	db re Spur	-0.5	+4	-2	-3	-1	+0.5	+2	-8	+1.5	-10.5	-6	-5						



TOP VIEW



SIDE VIEW

Figure 36. Schematic of Gear Test Stand Showing Installation of Acoustical Enclosure.

harmonics and display levels well above conformal gear noise at these frequencies. The wave analysis for Location 2 is shown in Figure 37. No attempt was made to determine if any of the reported levels were a direct result of the gear case response at or near a natural frequency. The assumption is made that this was not occurring.

A comparison of one-third octave band analyses for each gear configuration is shown in Table VII for Location 2, a typical location. This also illustrates the higher levels produced by the conformal gears in the low frequency spectrum and the generally lower levels produced by the conformal profile above 1000 Hz.

Due to the many variations in noise level which can occur owing to effects associated with the test setup and not with the gears themselves, the results of the program described here should be treated as a trend and not as absolute information. However, additional data would serve to confirm these results.

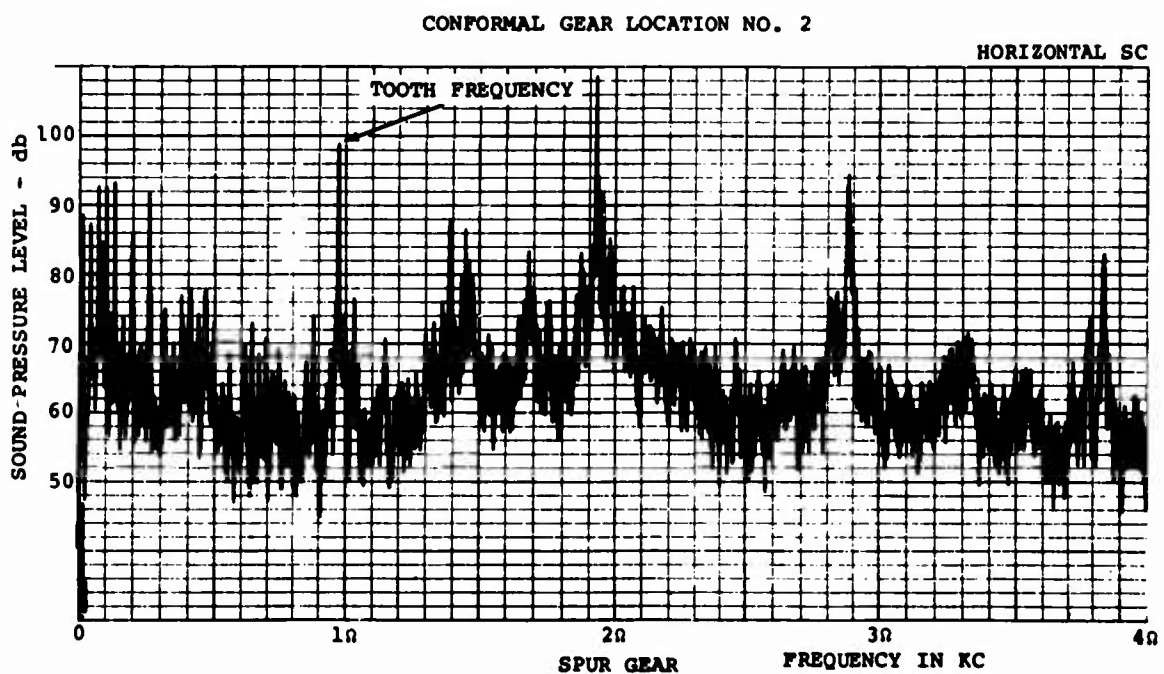
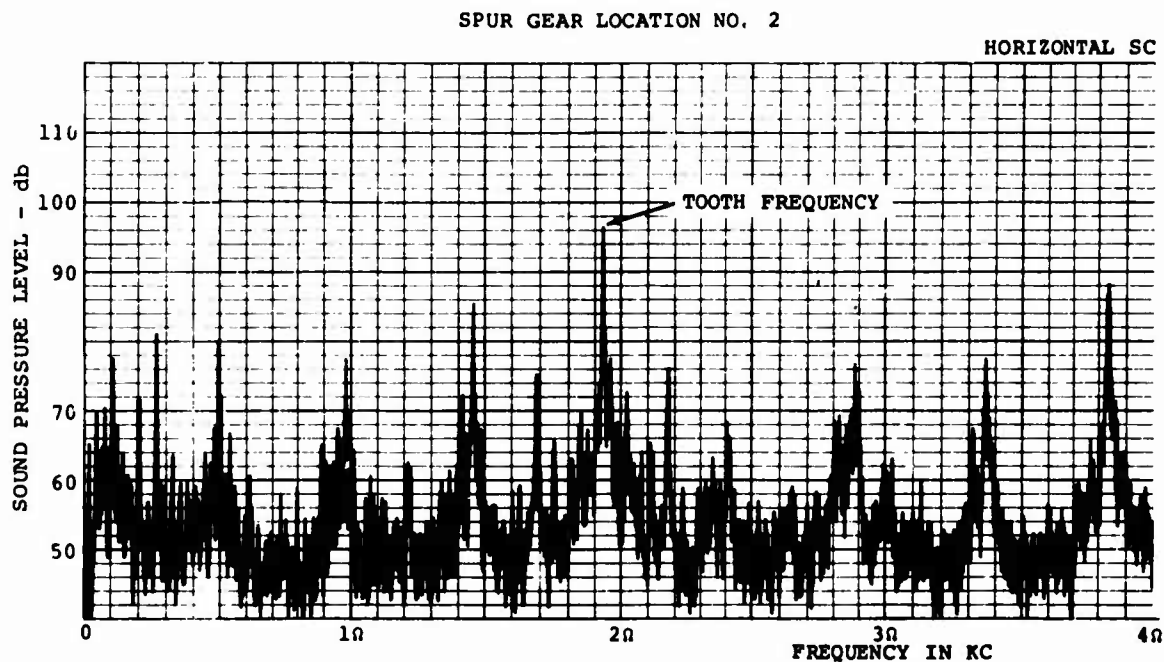


Figure 37. Wave Analysis - Conformal and Spur Gears.

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13. ABSTRACT This report concludes the program for the evaluation of one form of the conformal gear geometry conducted by the contractor. The program comprised the design, fabrication, inspection, and load testing of one form of the conformal gear tooth geometry. The test gears were case-carburized and ground in accordance with high air-craft quality standards and the applicable Vertol Division specifications. Load capacity of this particular tooth form was determined by a programmed increase in load until failure was attained. Results of this test program indicated that the gear member was the weakest in bending strength, and that a better balance between contact stress and bending stress will be necessary in future designs. The tooth strain survey indicated that the trends of the strain readings at the gages, as the load zone passed across the face width, were similar to the results shown in USAAVLABS Technical Report 66-8.		

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