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Evaluation of Gears Manufactured by Roll Forming

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EVALUATION OF GEARS MANUFACTURED BY ROLL FORMING

By

Harold K. Frint

October 1972

EUSTIS DIRECTORATE
U. S. ARMY AIR MOBILITY RESEARCH AND DEVELOPMENT LABORATORY
FORT EUSTIS, VIRGINIA

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SIKORSKY AIRCRAFT
DIVISION OF UNITED AIRCRAFT CORPORATION
STRATFORD, CONNECTICUT

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13. ABSTRACT

The results of a 19-month gear fatigue research program on roll-formed gears are presented. The purpose of this research effort was to evaluate the fatigue strengths of spur gears produced by two representative roll-forming processes in comparison with those manufactured by conventional forging methods.

The two forming processes tested included roll forming, wherein continuous spline-like teeth are rolled on a long solid bar which is subsequently sliced up into individual gear blanks, and a roll-generating process, in which individual blanks are prehobbed and then rolled to finished blank size. Both forming methods produced gear blanks which were within 0.007 inch of finished gear size.

On the basis of both single-tooth and dynamic fatigue tests conducted on a limited number of test specimens, the roll-formed gears exhibited fatigue strengths which are essentially equal to the fatigue strength of gears produced by conventional means.

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Single-Tooth Tests						
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Gear Rolling						
Gear Testing						

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DEPARTMENT OF THE ARMY
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EUSTIS DIRECTORATE
FORT EUSTIS, VIRGINIA 23604

This report was prepared by Sikorsky Aircraft, Division of United Aircraft Corporation, under the terms of Contract DAAJ02-70-C-0034. It describes the results of a comparative evaluation of spur gears manufactured from conventional forgings and gear blanks produced by two representative roll-forming techniques.

The object of the contractual effort was to evaluate the fatigue strengths of spur gears formed by the roll-forming process, the roll-generating process, and the conventional pancake-type forging method. Single-tooth fatigue tests and dynamic tests on a four-square, closed-loop regenerative test rig were conducted to obtain data.

On the basis of both types of tests conducted on a limited number of test specimens, gears formed by the two roll-forming processes exhibited fatigue strengths that are essentially equal to the fatigue strength of gears produced by the conventional pancake-type forging method.

The technical managers for this contract were Messrs. Leonard M. Bartone and James Gomez, Technology Applications Division.

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USAAMRDL Technical Report 72-32
October 1972

EVALUATION OF GEARS MANUFACTURED BY ROLL FORMING

Final Report

Sikorsky Engineering Report 50749

By

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Prepared by

Sikorsky Aircraft
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Stratford, Connecticut

for

EUSTIS DIRECTORATE
U.S. ARMY AIR MOBILITY
RESEARCH AND DEVELOPMENT LABORATORY
FORT EUSTIS, VIRGINIA

SUMMARY

The results of a 19-month gear fatigue research program on roll-formed gears are presented herein. The purpose of this research effort was to evaluate the fatigue strengths of spur gears produced by two representative roll-forming processes in comparison with those manufactured by conventional forging methods.

The two forming processes tested included roll-forming, wherein continuous spline-like teeth are rolled on a long solid bar which is subsequently sliced up into individual gear blanks and a roll-generating process, in which individual blanks are prehobbed and then rolled to finished blank size. Both forming methods produced gear blanks which were within 0.007 inch of finished gear size.

On the basis of both single-tooth and dynamic fatigue tests conducted on a limited number of test specimens, the roll-formed gears exhibited fatigue strengths which are essentially equal to the fatigue strength of gears produced by conventional pancake-type forging.

FOREWORD

This report covers a comparative evaluation of spur gears manufactured from conventional forgings and gear blanks produced by two representative roll-forming techniques. The roll-forming processes tested included a roll-forming technique and a roll-generating method. The evaluation encompassed both single-tooth testing and dynamic testing at approximately 10,000 rpm. The program was conducted during the 19-month period from April 17, 1970 to November 17, 1971 for the Eustis Directorate, U.S. Army Air Mobility Research and Development Laboratory (USAAMRDL), Fort Eustis, Virginia, under Contract DAAJ02-70-C-0034, Task 1G162203D14414.

USAAMRDL technical direction was provided by Mr. Leonard M. Bartone and Mr. James Gomez of the Propulsion Division.

The program was conducted at Sikorsky Aircraft, Stratford, Connecticut, under the technical supervision of Mr. Lester R. Burroughs, Supervisor, Transmission Design and Development Section. Principal investigators for the program were Mr. S. Schuman of the Transmission Design and Development Section and Mr. J. Lucas and Mr. J. Bucci of the Materials Section.

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LIST OF SYMBOLS

E	endurance limit at an infinite number of cycles, pounds or psi
h	height of load above critical tooth section, inches
HPSTC	highest point of single-tooth contact
I	gear geometry factor - durability
J	gear geometry factor - strength
KHN	Knoop hardness number
n	number of test points
N	number of cycles
Rc	Rockwell hardness number
R_x	radius to load point at center of tooth, inches
S	equivalent stress, pounds per square inch
s	unbiased standard deviation, pounds
S/N	serial number
$s/\overline{X}$	coefficient of variation
t	tooth thickness at critical section, inches
W_n	normal tooth load, pounds
W_t	tangential tooth load, pounds
X	gear tooth form factor
$\overline{X}$	mean fatigue strength at 10^8 cycles, pounds
β	material constant in S-N relation
γ	material constant in S-N relation
ϕ	gear pressure angle, degrees
ϕ_N	angle which load line makes with a line perpendicular to the tooth center line, degrees

INTRODUCTION

The most readily observable advances in gear technology in recent years have been in the area of gear forging and manufacturing techniques. One such advance, reported by Sikorsky Aircraft in Reference 1 and by Western Gear Corporation in Reference 2, is the radial-extrusion forging of gears using high-energy forging processes which produce gear blanks with integrally forged teeth. In addition to lower material and manufacturing costs, higher fatigue strengths are realized from better control of grain size and orientation (grain flow). For example, in the program of Reference 1, increases in single-tooth bending endurance limits in the order of 24 to 44 percent were found when comparing integrally forged teeth with those produced by conventional means.

Another variation in gear manufacturing techniques which also has the potential of increasing gear tooth fatigue strengths is that of roll-forming, wherein integral teeth are produced on a long bar or on individual gear blanks by cold rolling. Principal differences in the various methods used to roll-form gear blanks lie in the amount of material displaced during the rolling process. In one such process, the teeth are rolled on a bar from the solid; whereas in another, gear blanks are pregashed or prehobbed before rolling to finished blank size.

This report presents the results of a two-phase test program conducted by Sikorsky Aircraft to evaluate the comparative fatigue strengths of gears produced by two representative roll-forming processes and gears fabricated by the conventional means. The test program covers both single-tooth static and fatigue testing, and also includes dynamic testing at approximately 10,000 rpm.

GEAR BLANK MANUFACTURE

DISCUSSION

The goal of this program was to evaluate, by test, the comparative fatigue strengths of spur gears produced by two representative roll-forming processes and by a conventional pancake-type forging. The test program encompasses both a single-tooth test phase and a dynamic test phase, and therefore, a sufficient quantity of raw material was purchased initially to supply gear blanks for both test phases.

RAW MATERIAL

The material selected was AMS 6265, an AISI 9310 vacuum-melt carburizing steel used extensively throughout the aircraft industry for power and accessory gearing. A sufficient amount of 5-inch-diameter bar stock for both the single-tooth and dynamic test programs was received from the supplier with proper documentation to certify that all bars were from the same heat. The material was inspected for conformity to AMS 6265 specifications, and a sufficient quantity was shipped to each gear blank and forging manufacturer for processing.

GEAR BLANK DESIGN

To reduce the number of machining operations from initial rolling of the gear blanks to final grinding of the finished test gears, and to assure that a maximum of the beneficial grain flow remained after machining, the gear blanks were designed so that a minimum of stock removal was required to produce the finished gear. As a result, the rolled gear blanks were produced with a tooth profile that was within 0.007 inch minimum of the finished tooth size specified in Figures 1 and 2.

PANCAKE FORGINGS

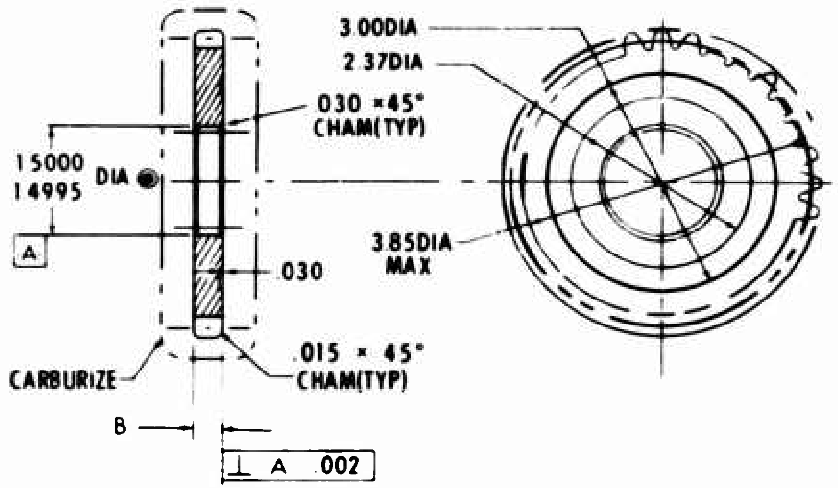
To produce each pancake forging, the 5-inch-diameter bar furnished to the forging manufacturer was first machined to a slug 2 inches in diameter by 4.500 inches in length. The forgings were then formed by a steam hammer in several successive blows at an initial temperature of 2200°F. After forming, the forgings were process annealed at 1200°F for one hour and then air-cooled, producing a Brinell hardness of 197. A typical forging blank produced by this method is shown in Figure 3.

ROLL-FORMED GEAR BLANKS

In this method of roll-forming, continuous spline-like gear teeth were formed on a long solid bar, the diameter of which conformed approximately to the pitch diameter of the gear blank. The rolling effort was provided by two pairs of opposing rollers, mounted in a planetary arrangement, which penetrated the bar for a short part of their cycle as the roller carrier rotated, forming an increment of a longitudinal groove into the bar. The rollers then left the bar for the rest of the cycle, during which time the bar was indexed. A new roller contact was then made, forming an increment of an adjacent groove. The bar was continuously longitudinally advanced. Since the teeth are not generated in this method, the roller shape conformed exactly to the gear space shape in the finished gear blank. The bar stock was rolled by guiding the outside diameter of the bar in bushings and passing the entire length of the bar through the machine. The synchronization and indexing were such that all of the teeth were formed in one lengthwise pass of the bar. Individual gear blanks were produced by sawing off gears from the roll-formed bar. The finished gear blank produced by this method is shown in Figure 4.

ROLL-GENERATED GEAR BLANKS

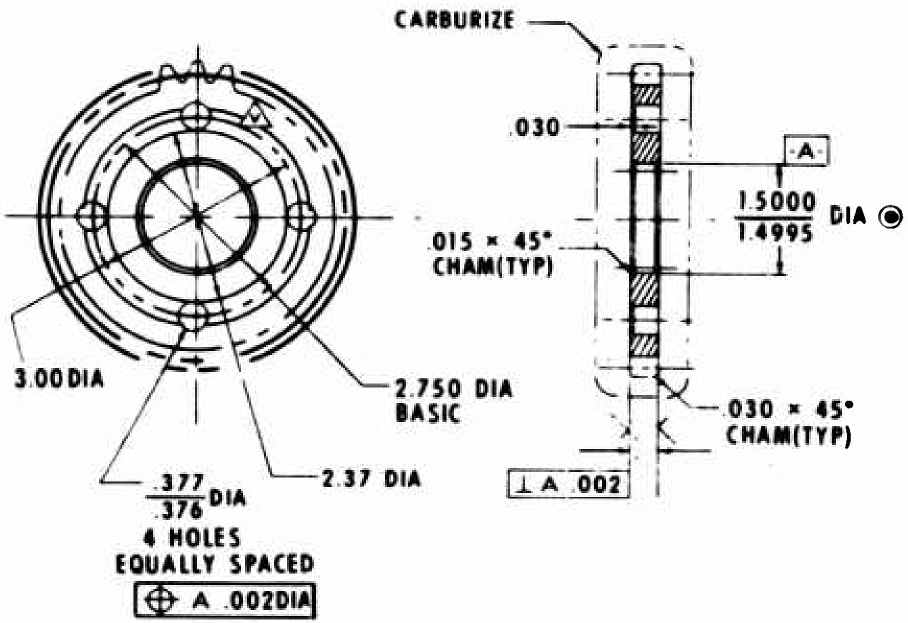
This roll-forming process we have termed the roll-generating method to distinguish it from the roll-forming method described above. Individual gear blanks were machined from the raw material and prehobbed to within approximately 0.005 inch of the finished blank dimensions. The prehobbed gear blank was then positioned between opposing circular dies having teeth with the same pitch as the finished blank. As the dies were set in under radial pressure and rotated in synchronization with the work piece, teeth were roll-formed on the gear blank to the required blank dimensions. This action is a generating action; therefore, the dies could have been any convenient diameter as long as they had the same pitch as the finished blank. The finished gear blank produced by the process is shown in Figure 5.



SPUR GEAR DATA	
NO. OF TEETH	32
DIAMETRAL PITCH	8
PRESSURE ANGLE	22½°
PITCH DIAMETER	4.000 IN.
BASE CIRCLE DIAMETER	3.6955 IN.
OUTSIDE DIAMETER	4.2500/4.2540 IN.
ROOT DIAMETER	3.6934/3.6834 IN.
CHORDAL TOOTH THICKNESS	.1943/.1933 IN.
CHORDAL ADDENDUM	.12735 IN.
DIMENSION OVER .2160 DIA BALLS	4.2949/4.2971
MAX. INVOLUTE PROFILE ERROR AT T.I.F.	+.0003 −.0003
AT TIP	−.0015/−.0019 IN.
MAX. ACCUMULATED SPACING ERROR BETWEEN ANY TWO TEETH	.0018 IN.
MAX. LEAD ERROR	.0002 IN.
MAX. T.I.F. DIA.	3.8088 IN.
FLAT ROOT CORNER RADII (APPROX)	
MATING PART	
BACKLASH FULL	.004/.006 IN.
FULL FILLET RADIUS (APPROX)	.0625 IN.

1. DIAMETERS MARKED ◉ TO BE CONCENTRIC TO EACH OTHER WITHIN 0.002 T.I.R.
2. CARBURIZE GEAR ALL OVER PER SS8015 TO PRODUCE 0.035 TO 0.045 DEPTH OF CASE IN FINISHED PART.
3. BEFORE FINISH MACHINING, HEAT TREAT TO R_c 58-64 OR EQUIVALENT CASE HARDNESS; CORE HARDNESS TO BE R_c34-40.
4. BREAK ALL SHARP EDGES 0.005-0.015 UNLESS OTHERWISE SPECIFIED.
5. SURFACES MARKED √ TO HAVE $\sqrt{}^{12}$ EXCEPT AS NOTED SS5100 SHALL APPLY FOR HOLES UNLESS OTHERWISE INDICATED. AREAS OF TRANSITION SHALL CONFORM TO THAT SPECIFIED FOR THE ROUGHEST ADJACENT FINISHED AREA UNLESS OTHERWISE INDICATED.
6. MARKING TO BE DONE PRIOR TO MACHINING OF GEAR TEETH PROFILE.

Figure 1. Single-Tooth Test Gear.



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6. MARKING TO BE DONE PRIOR TO MACHINING OF GEAR TEETH PROFILE.

Figure 2. Dynamic Test Gear.

Figure 3. Conventional Pancake Forging.

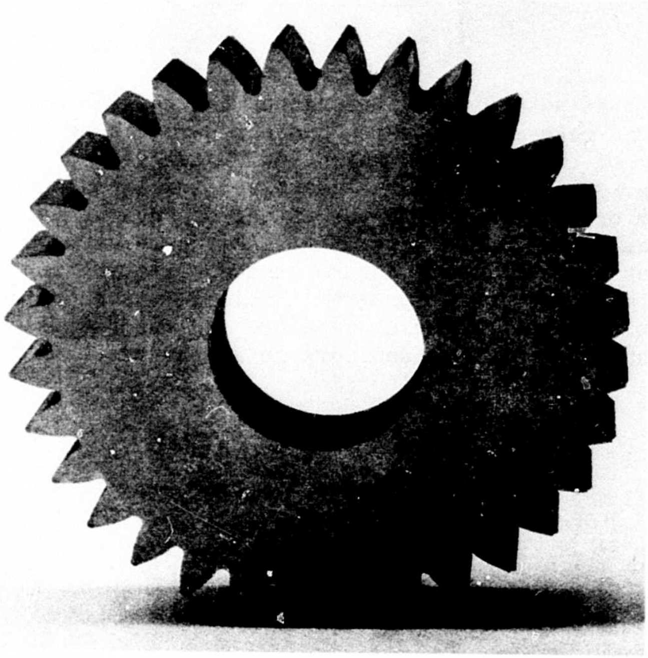


Figure 4. Roll-Formed Gear Blank.

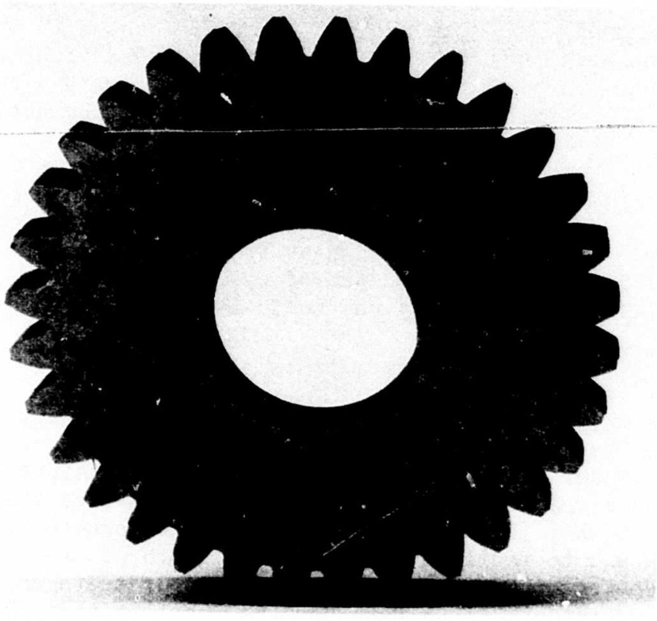


Figure 5. Roll-Generated Gear Blank.

GEAR MANUFACTURE

DISCUSSION

The procedure that was followed throughout the test program, and particularly during the gear manufacturing processes, was such as to minimize or eliminate many of the variables which could have lead to experimental error during testing. Consequently, the gears were machined and ground in a completely random order to minimize the effect of such variables as machine drift, tool wear, and machine operator. All gears were manufactured at the same manufacturing facility to assure common heat-treating and machining practices. A gear processing program, Table I, was supplied to the gear shop to control the manufacturing sequence and randomization of the gear blanks. Finished gear tolerances were in accordance with the AGMA Class 12 requirements.

TEMPERING

As each lot of gear blanks was received at the gear shop, it was identified and serialized. To facilitate machining, all gear blanks were then normalized and drawn to Rc 25-30 prior to machining.

MACHINING

The first machining sequence consisted of rough machining one side, the inside and outside diameters, and the undercut. The serial number was immediately stamped in the machined recess for identification during the subsequent processing. The second side was then rough machined. At this point in the sequence, the pancake forgings were hobbed to the tooth dimensions of the rolled gear blanks. The gear blanks destined to become the dynamic test specimen were then drilled and reamed (with allowance made for final grinding) to form the four attachment holes of the configuration shown in Figure 2.

CARBURIZING

All gear blanks were carburized and heat treated as a single lot to produce a 0.035 to 0.040 inch effective depth of case and an equivalent core hardness of Rc 34-40. Case hardness was Rc 58-64.

GRINDING

After carburizing, the gear blanks were intermixed in a systematically arranged sequence for grinding. The grinding operation was accomplished on a Detroit Gear Grinder using an 8-inch-diameter wheel. After grinding, 16 of the 32 teeth were removed from the single-tooth test specimen to produce the configuration shown in Figure 1. This was done to provide clearance around the test tooth and to facilitate loading the tooth at the highest point of single-tooth contact (HPSTC) in the fixture. Four teeth on each gear were designated as test teeth, four were used as reaction teeth, and the rest were retained for metallurgical evaluation.

TABLE I. ROLL-FORMED GEAR PROCESSING PROGRAM

Step	Process	Purpose	Lot Size	Control
1	Serialize forging	Identification	Individually	Serialize in accordance with Table II
2	Normalize & draw to Rc 25-30	Machinability & structural refinement	All together	
3	Blank	Machine one side, ID & OD & undercut. Reserialize, machine second side	Individually	Random order
4	Hob or shape pancake forgings to within 0.007 to 0.010 inch of finished size	Form gear teeth	Individually	Random order
5	Drill & ream holes (allow for grinding) (S/N 31 thru 140 only)	Form mounting holes	Individually	Random order
6	Burr & buff	Remove sharp edges	Individually	Random order
7	Carburize, harden & draw	Heat-treat cycle	All together	
8	Finish grind (other than gear teeth)	Establish final dimensions	Individually	Random order
9	Jig-grind holes (S/N 31 thru 140 only)	Finish holes to size	Individually	Random order
10	Grind gear teeth	Establish final tooth dimensions	Individually	Random order

TABLE I - Continued

Step	Process	Purpose	Lot Size	Control
11	Nital etch	Inspection for grinding burns	Individually	Random order 100%
12	Remove teeth per drawing on S/N 01 thru 30	Single-tooth test configuration	Individually	Random order
13	Burr & buff	Remove sharp edges	Individually	Random order
14	Magnaflux	Inspection for grinding cracks	Individually	100% inspection
15	Parco-lubrize	Protective coating	All together	
16	Magnaflux & visually inspect	Final check	Individually	100% inspection

Profile charts showing deviations from a true involute form were taken for each test gear after grinding. Sample charts for the as-rolled gear blanks and for the finished hardened and ground test gear are shown in Figures 6, 7, and 8.

During grinding it was observed that the teeth on the roll-generated gear blanks failed to clean up when ground to the 3.6834-inch root dimension. Authorization was then given to grind to 3.6750 inches to clean up. This was done on all gear blanks to make strength comparisons valid.

The finished test gears are shown in Figures 9 and 10.

The test gears were serialized for identification as shown in Table II.

TABLE II. GEAR SERIALIZATION AND IDENTIFICATION

Forging	Forging Process	Part Number	Serial Number
61050-35059-001	Conventional Pancake	61050-35059-101	01-10
61050-35059-002	Roll-Formed	61050-35059-102	11-20
61050-35059-003	Roll-Generated	61050-35059-103	21-30
61050-35059-002	Roll-Formed	61050-35059-112	31-70
61050-35059-003	Roll-Generated	61050-35059-113	71-100

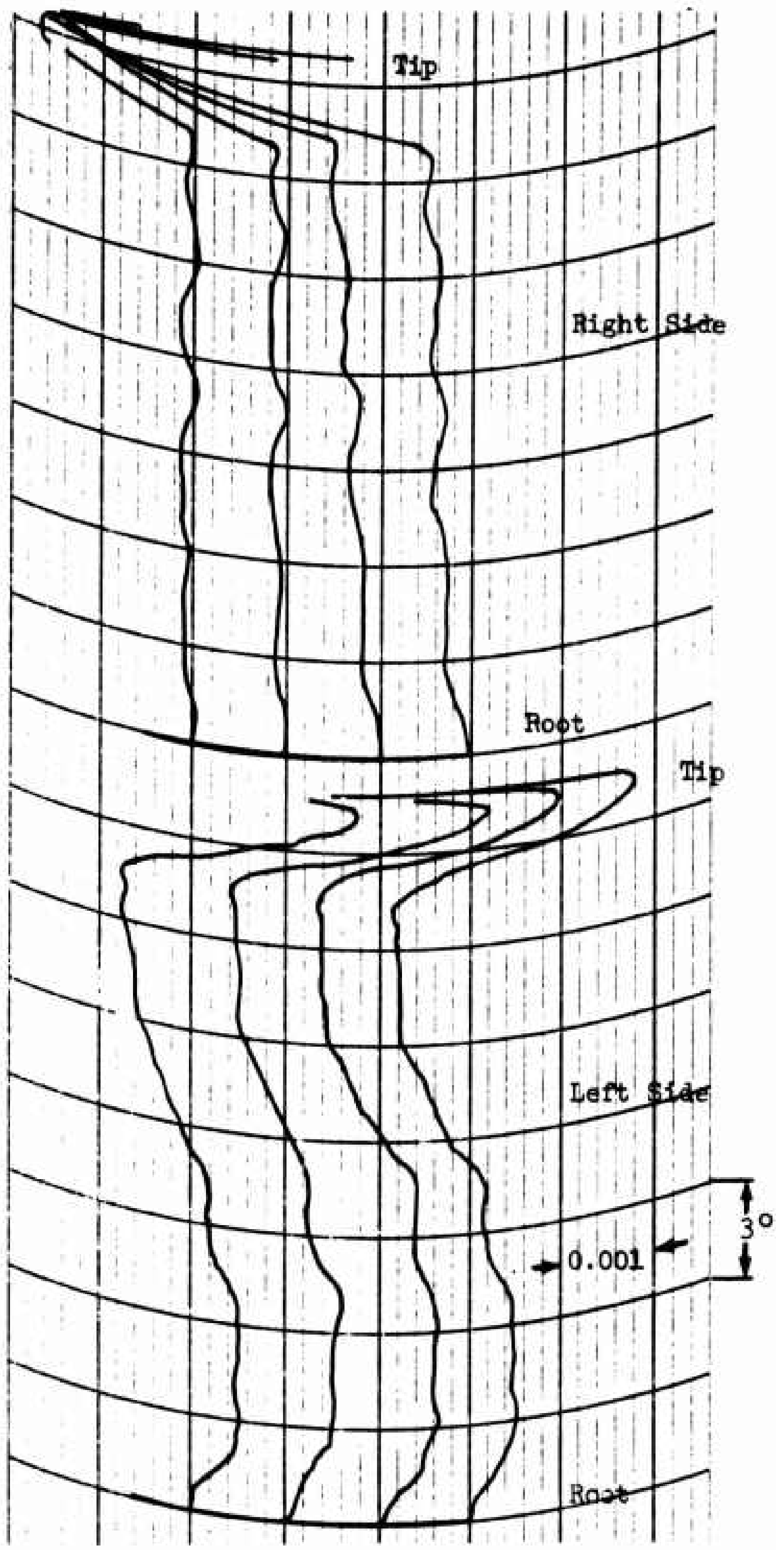


Figure 6. Involute Profile Chart for Roll-Formed Process - As Rolled.

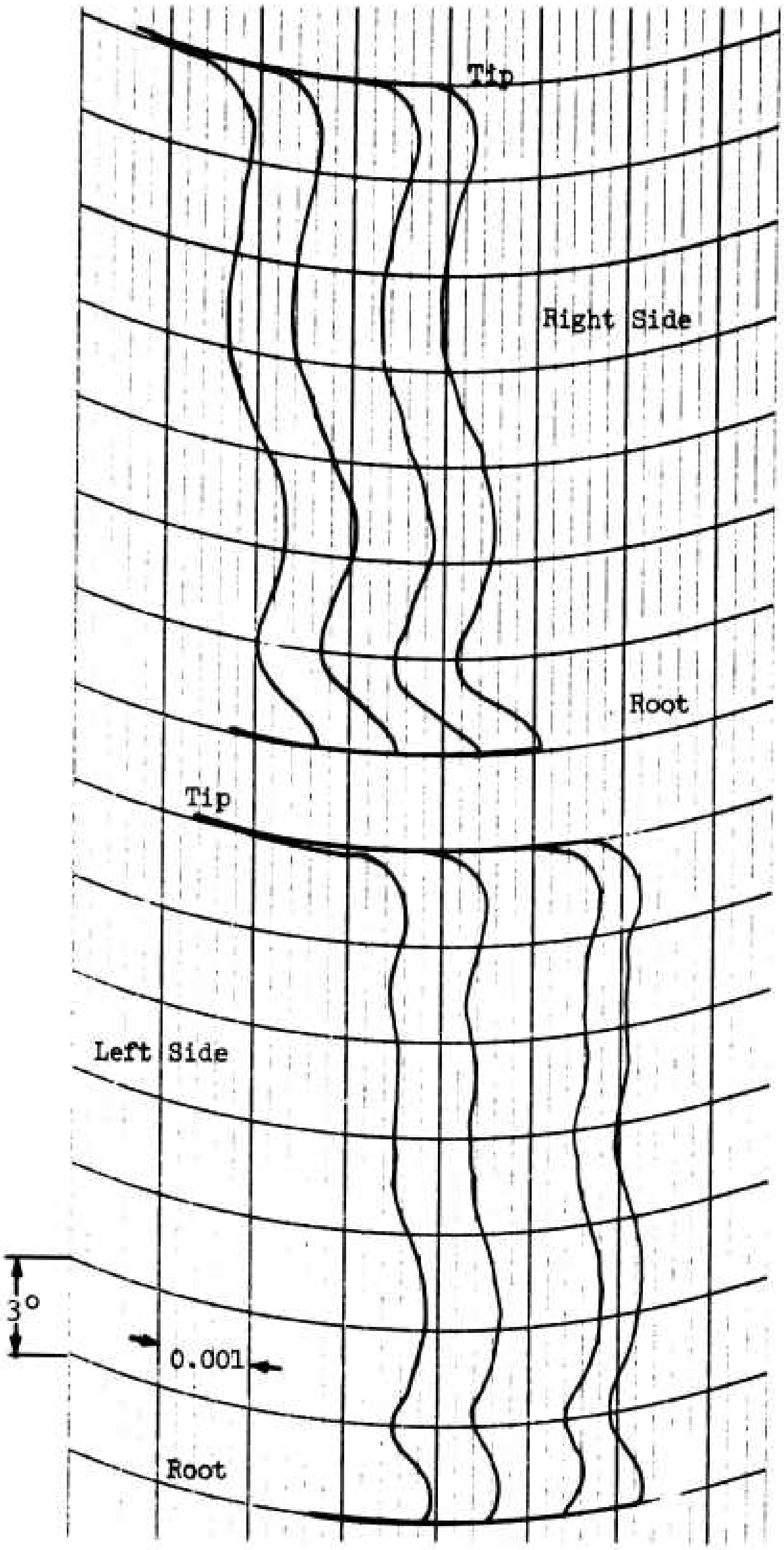


Figure 7. Involute Profile Chart for Roll-Generated Process - As Rolled.

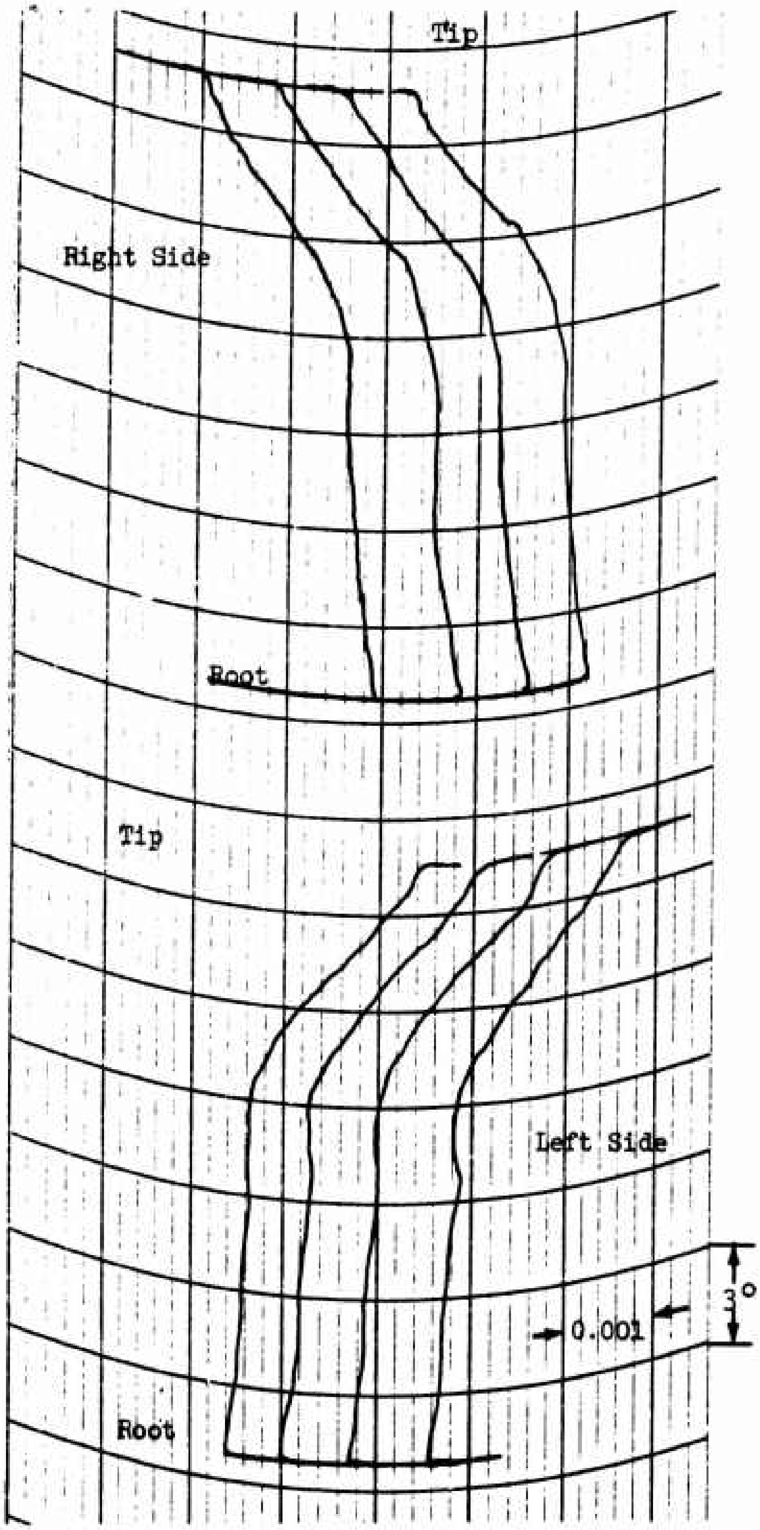


Figure 8. Involute Profile Chart for
Finished Test Gear.

Figure 9. Finished Single-Tooth Test Gear.

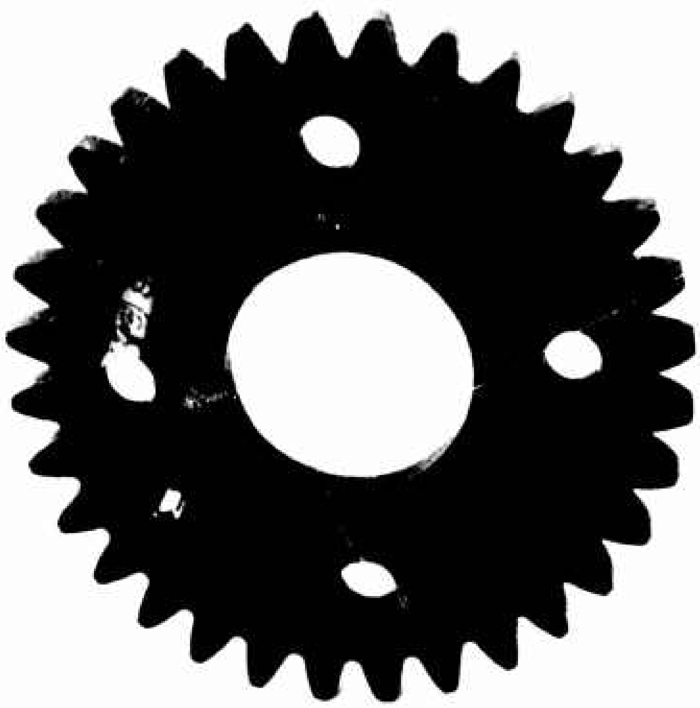


Figure 10. Finished Dynamic Test Gear.

TEST PROCEDURE

SINGLE-TOOTH TEST PHASE

Fatigue Test Setup

The single-tooth fatigue tests were conducted on special Sikorsky test fixtures. In this test series, three of the fixtures were mounted on Sonntag Model SF-1-U universal fatigue testing machines fitted with five-to-one load amplifiers. Figures 11 and 12 illustrate the Sikorsky test fixtures and Sonntag machines respectively. Figure 13 shows the complete test arrangement. The test load was applied by means of a loading pin which contacted the gear tooth normal to the involute profile at the "worst load position." A tungsten carbide tip was brazed to the loading pin to improve the durability of the surface contacting the gear tooth.

The normal tooth load was reacted by a reaction tooth which contacted a contoured support block. The contact was over the entire tooth profile to reduce the stress and prevent the reaction tooth from failing. The test fixture was designed so that the loading pin automatically contacted the test tooth at the "worst load" point when the gear was installed in the fixture, and the reaction tooth was positioned to make contact with the reaction block. Figures 14 and 15 show respectively the point of contact and direction on the gear tooth profile for the "worst load" condition and a drawing of the test fixture. A preload was maintained on the test tooth when the gear was tightened in the fixture to prevent separation between the reaction tooth and reaction block which could cause an error in the load application point.

Load cells were installed in series with the loading pin for static and dynamic load determination. Each load cell was calibrated statically in the Riehle PS-60 Tensile Machine at the beginning of the test and every two months thereafter while testing was in progress. An Ellis BA-12 bridge amplifier and cathode ray oscilloscope was used to read the strain gage bridge output. The calibrated load cells were used as the primary load measuring system for test setup and were used for checking the applied loads twice daily while a test was in progress.

Failure of the test tooth was considered to have occurred when a 1/16-inch crack was detected. A "microwire" technique was used for crack detection. Copper wire, approximately 0.005 inch in diameter, was cemented to the sides of the gear approximately 1/32 to 1/16 inch from the test tooth profile and was connected to the control system of the Sonntag machine. When a crack was initiated in the test tooth and propagated to the wire, the wire would break and the Sonntag machine would shut off.

Tests were conducted on twelve teeth of each of three manufacturing processes. Runs were made at four load levels on each of three gears of each process for a total of 36 test points. All test load levels varied sinusoidally from a 100-pound positive minimum load to whatever positive maximum load was required to obtain failures in the desired cycle ranges. This type of loading produced tooth bending in one direction only, which is typical of service pinions and gears. The positive minimum 100-pound load was maintained for all test load levels to prevent any separation and impact loading which would occur if the minimum load was allowed to reach zero. All four test teeth of each gear were tested at a different load level. The test sequence was randomized with respect to forming process, test tooth, test load, and testing machine.

The approximate single-tooth load levels and corresponding gear stresses calculated using the AGMA method (Reference 4) are shown in Table III.

TABLE III. SINGLE-TOOTH LOADS AND STRESSES

Normal Tooth Load (lb)	AGMA Bending Stress (psi)
2,700	126,700
3,100	145,500
3,900	183,000
5,500	258,100

Static Tests

Single-tooth static tests were conducted in the Riehle PS-60 Tensile Test Machine on the same gear configuration as previously discussed for fatigue tests. The same fixture used for the fatigue test was also used for the static tests. An adaptor was bolted to the top of the fixture to enable it to be held in the upper head of the tensile machine. The load was applied through the loading pin which contacted a compression plate in the lower head. Figure 16 shows the fixture mounted in the tensile machine. For the static tests, the loading point on the gear tooth was changed from the "worst load" condition to tooth tip loading by changing the reaction block. Figures 14 and 15 show respectively the point of contact and direction of the tip load on the tooth profile and a drawing of the test fixture with the dimensions used to obtain tip loading. This type of loading was considered necessary to prevent the loading point contact from rolling back into the root radius and off the edge of the loading pin because of the more ductile nature of the static fracturing. The ultimate load for each test tooth was read directly off the tensile machine dial.

DYNAMIC TEST PHASE

Test Facility

A Sikorsky-designed test facility, incorporating a four-square closed-loop regenerative test rig, Figure 17, was used to evaluate the test gears dynamically. In this test facility, two gearboxes, each containing two pairs of test gears, are connected by shafts mounted on flexible couplings to reduce any interactions between the two gearboxes. This arrangement permits four gears to be tested simultaneously.

The original configuration of this test facility incorporated an idler gear between the two test gears in each gearbox. The present improved configuration has two pairs of test gears at each end mounted in a staggered arrangement as shown in the schematic of Figure 17. This eliminates reversed bending loads on the idler gear and prevents progressive secondary fractures when a single test gear fails.

A 40-horsepower, 1750-rpm electric motor supplies the necessary power to overcome the friction of the system. A vee-belt drive with a 1.75 to 1.0 pulley ratio transmits the power to the test gearbox. A spur gear set with a 3.3 to 1.0 ratio delivers the power to the closed loop at 9200 rpm.

Torque is applied to the system by the relative angular displacement of vernier plates on one of the connecting shafts. System "wind-up" provides adequate sensitivity to obtain the desired torque levels. Strain gages in torque half-bridge configurations are used to measure system torque while the load is being applied.

Each gearbox has an independent lubrication system operating at flow rates of up to 5 gpm. Each oil reservoir has a capacity of 15 gallons of oil. The maximum heat rejection capacity for each cooling system is 50,000 Btu per hour. A 40-micron filter on each supply line maintains oil cleanliness and prevents oil jet blockage.

Temperatures are monitored using iron-constantan thermocouples on oil-in and oil-out lines and on bearing housings in twelve locations. A flow meter and pressure gage on each oil-in line provide lubrication rate information.

A failure detection system which would automatically shut down the test facility when a failure occurred is installed as part of the test facility circuitry. A low-oil-pressure switch protects the facility from failures due to malfunctioning oil pumps, ruptured oil lines, or low oil level in either sump. Excessive oil temperature also activates the shutdown system. Magnetic type chip detectors are incorporated to stop the test if metallic particles enter the lubrication system. A unique feature of this shutdown system is a missing-tooth detection device. This device compares an input signal from a magnetic tooth contactor (on a cycle per cycle basis) to an internal signal generated by an oscillator which is phase-locked to the contactor signal. If a tooth is missing, the comparison on that cycle will trigger a flip-flop which will trip the motor relay to shut off the machine. The time from detection to relay shutoff is approximately equal to the relay closing time. There is one circuit for each of the eight gear positions. When a bending failure occurs, which results in the loss of a gear tooth, lights on the instrument panel, Figure 18, will indicate not only which gearbox is affected but in which of the four possible gear positions the failed gear can be found. If the test machine shuts down because of low oil pressure, low oil level, chip detection, or a recorder malfunction (high temperature), this fact will also be indicated by an appropriate light on the instrument panel.

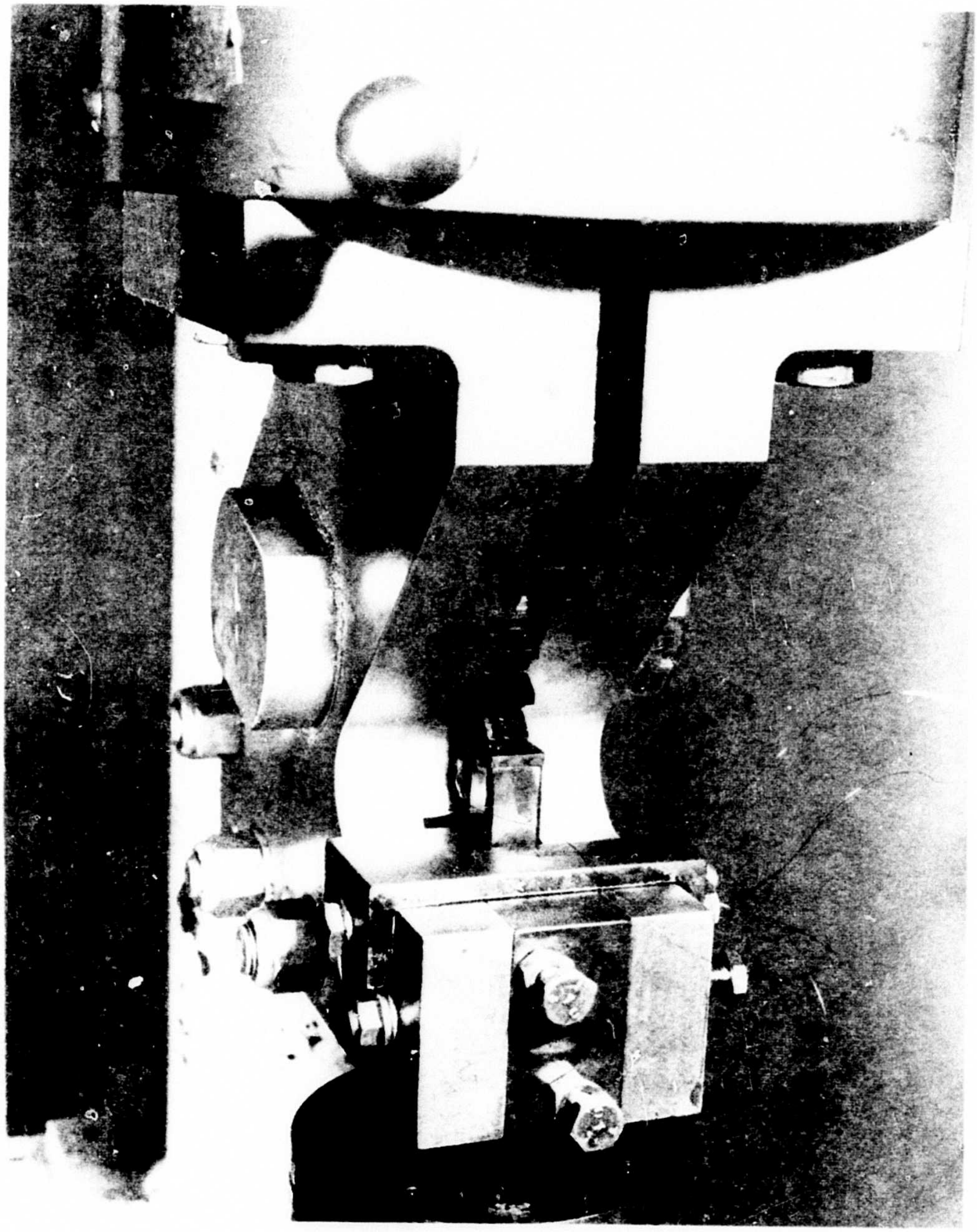


Figure 11. Single-Tooth Test Fixture.

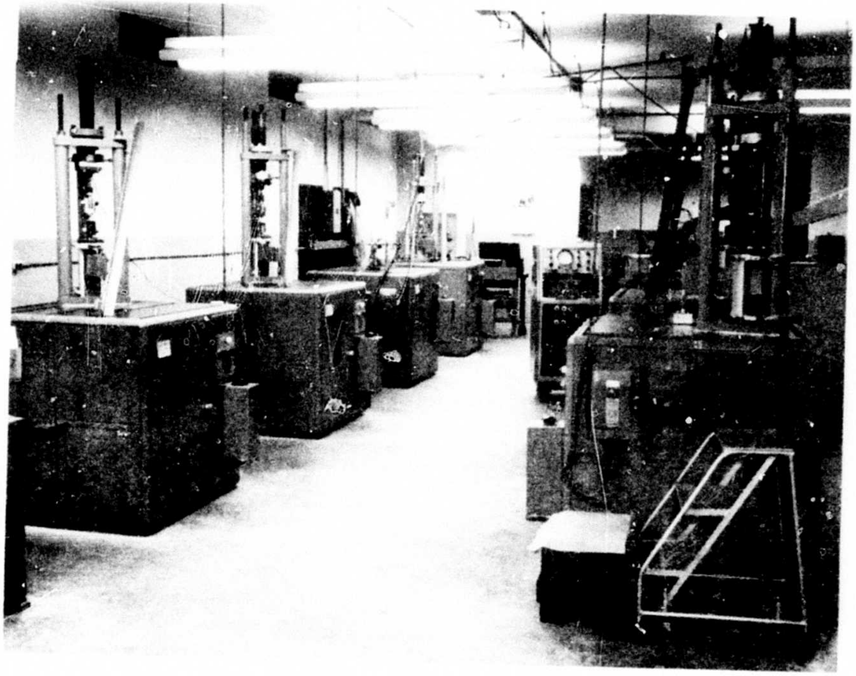


Figure 12. Sikorsky Fatigue Test Laboratory.

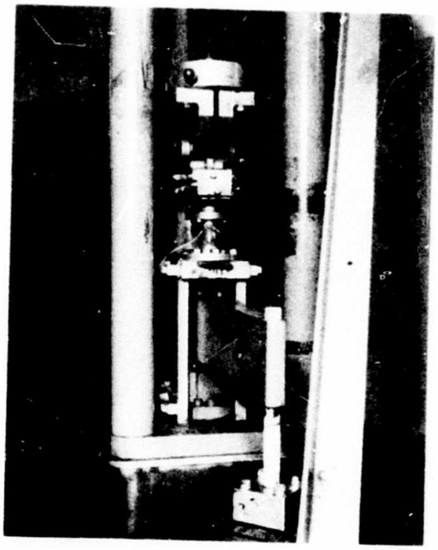
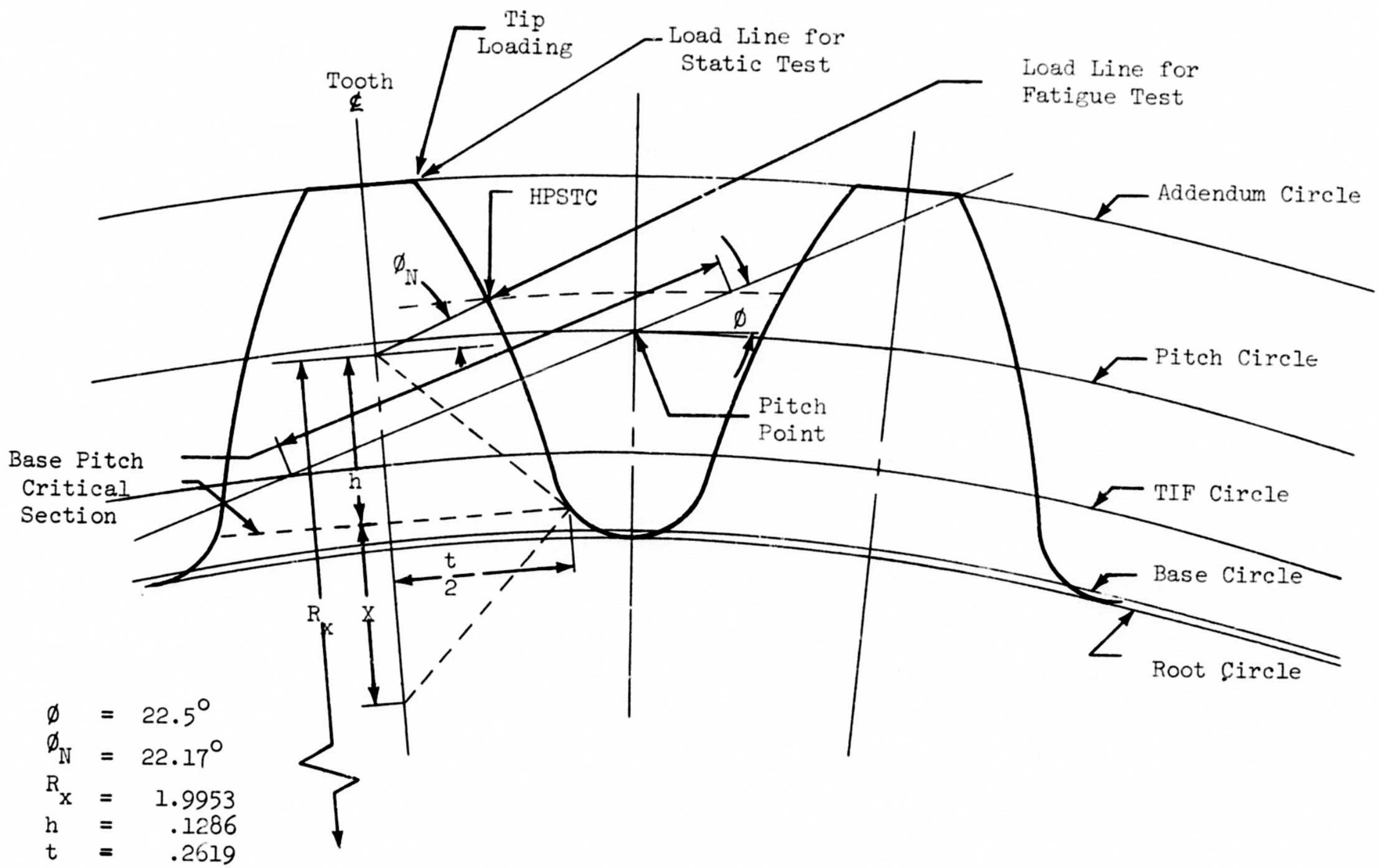


Figure 13. Fatigue Test Arrangement on the Sonntag Machine.



$\phi = 22.5^\circ$
$\phi_N = 22.17^\circ$
$R_x = 1.9953$
$h = .1286$
$t = .2619$
$X = .1333$
HPSTC is worse load position.

Figure 14. Single-Tooth Load Parameters.

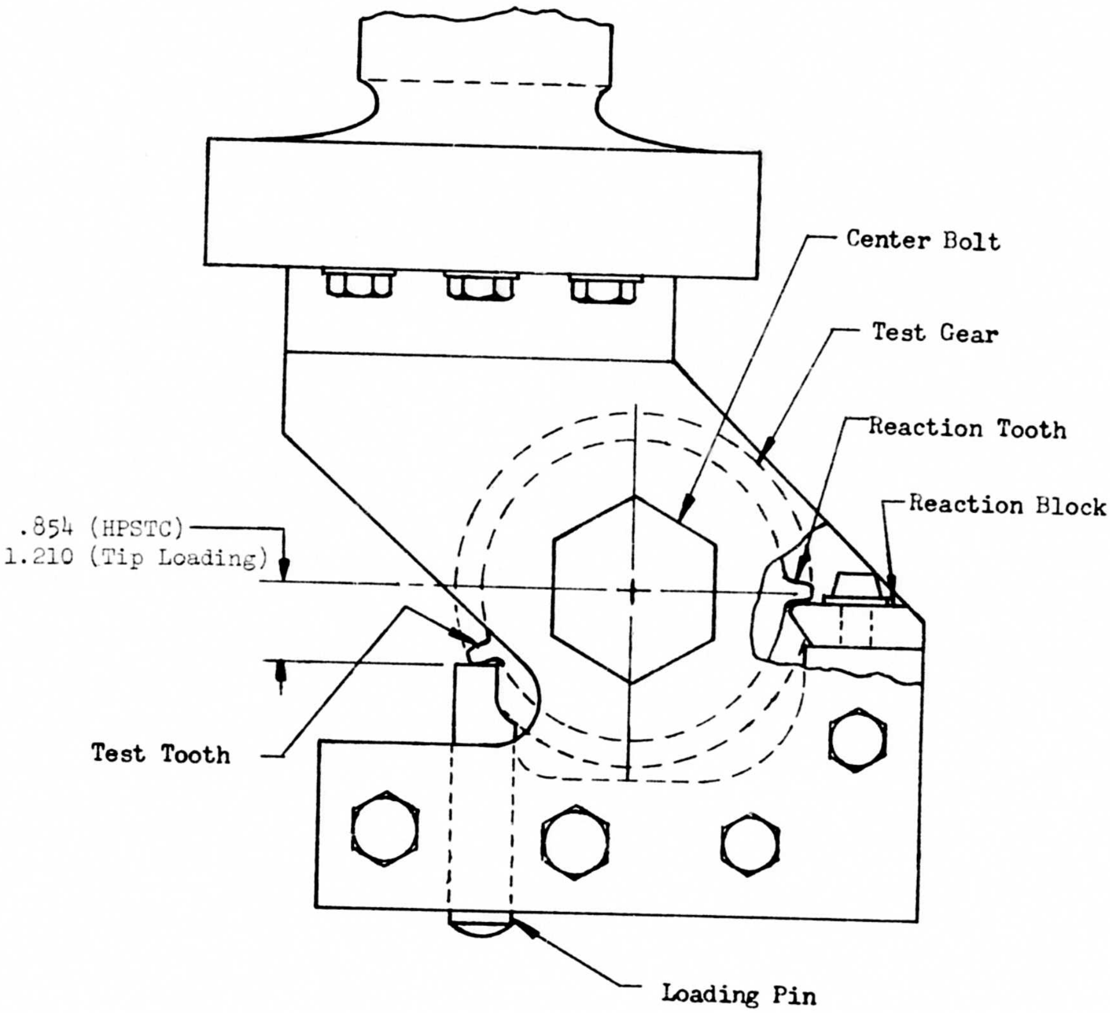


Figure 15. Schematic of Gear Test Fixture.

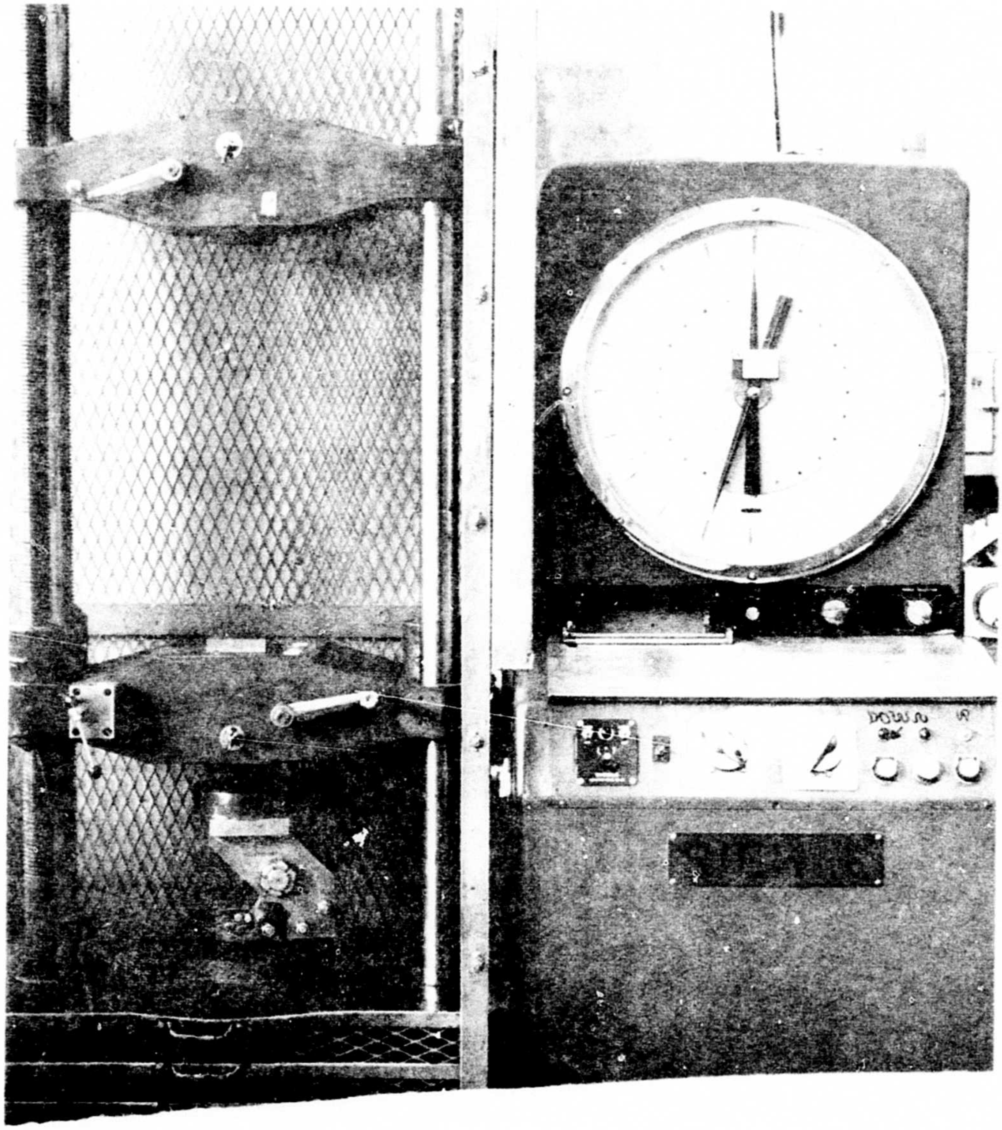
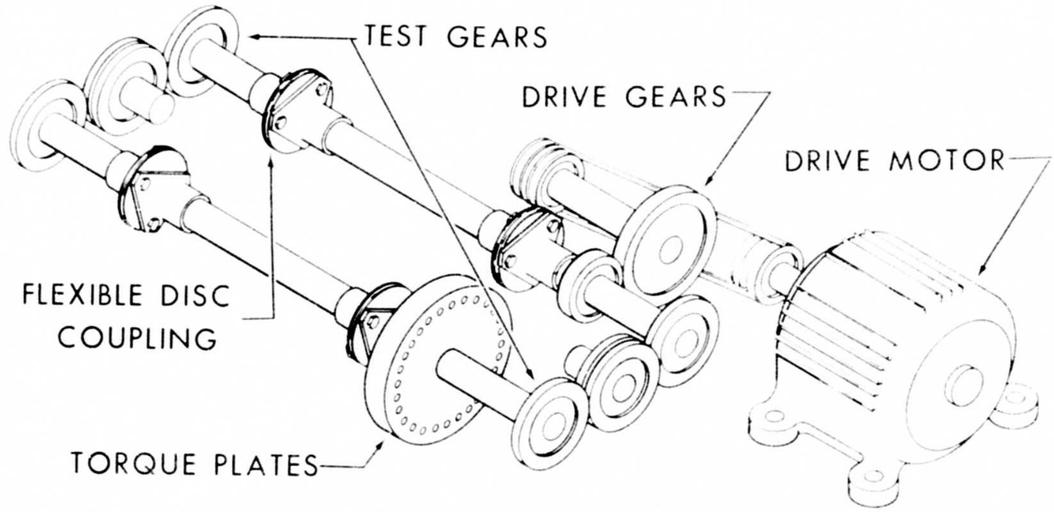


Figure 16. Static Test Arrangement in the Riehle Tensile Machine.



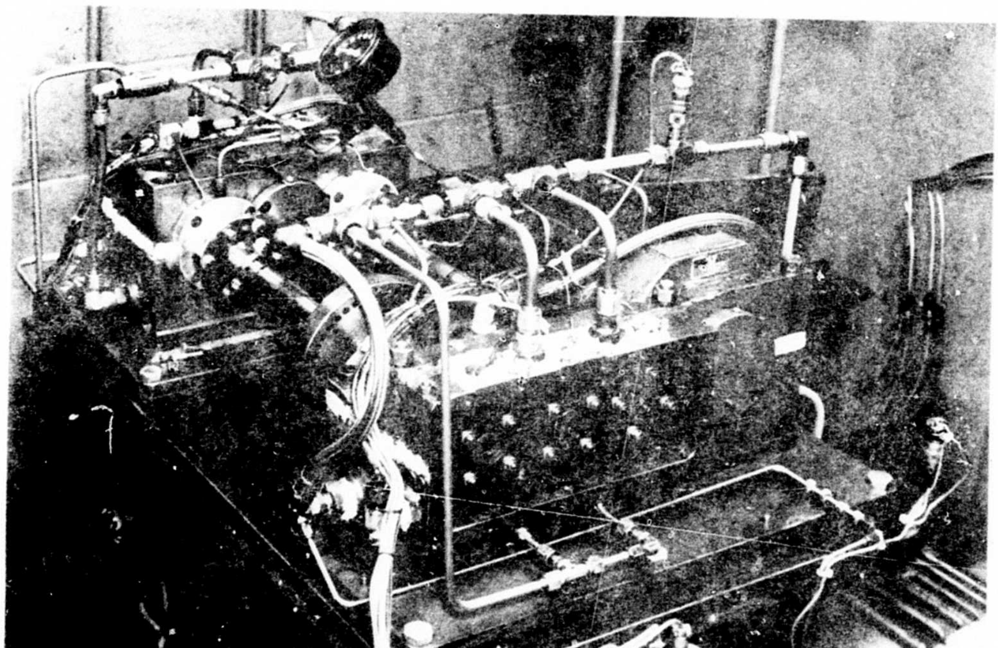


Figure 17. Sikorsky Dynamic Test Facility.

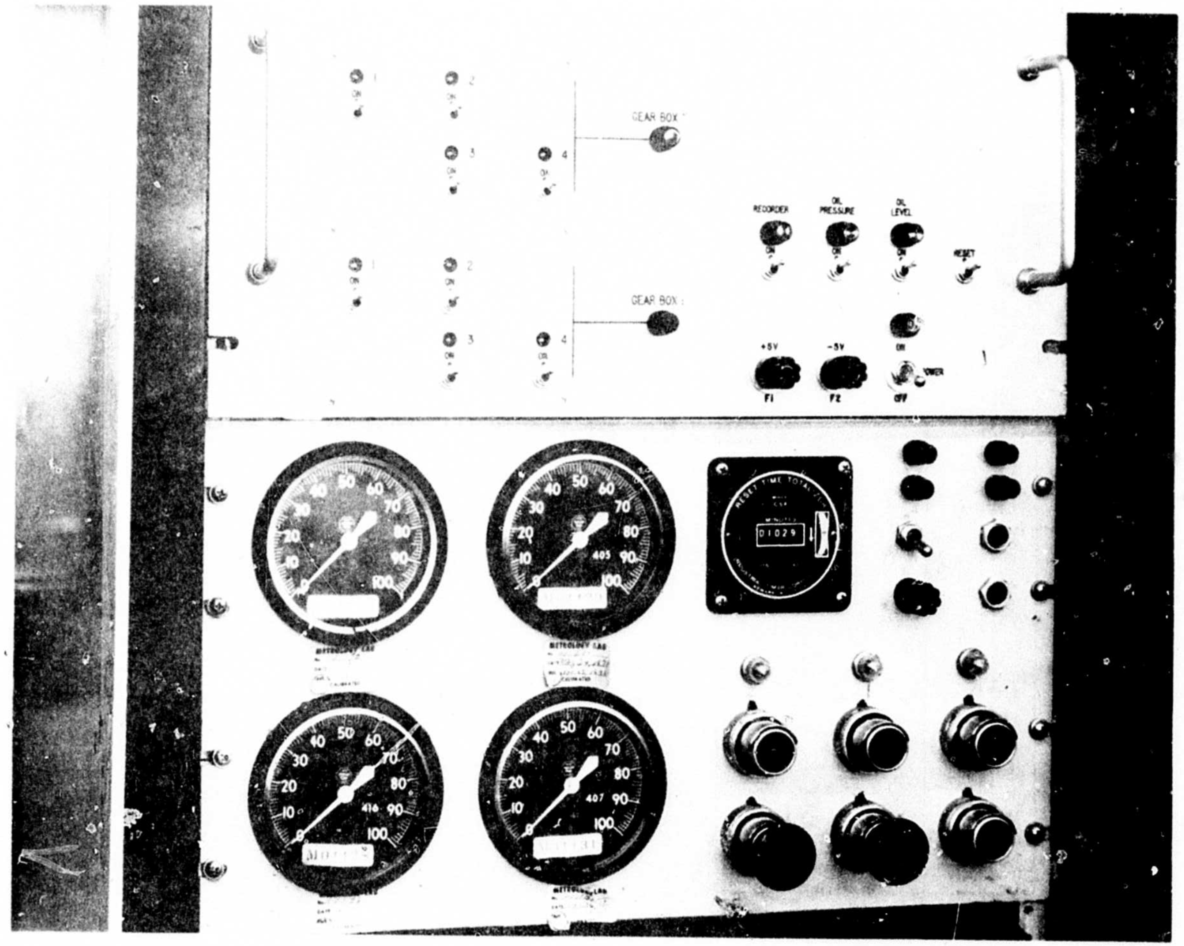


Figure 18. Instrument Panel - Four-Inch Gear Tester.

Fatigue Tests

The dynamic fatigue tests were conducted in a randomized order on a minimum of sixteen gears of each roll-forming process. Three gears of each process were tested at each of four load levels, for a total of 32 test points.

The gear tooth loads used and the corresponding gear tooth stresses are shown in Table IV.

TABLE IV. DYNAMIC TOOTH LOADS AND STRESSES

Tooth Load (lb)	AGMA Bending Stress (psi)	Compressive Stress (psi)
1,500	76,200	244,600
1,750	88,900	264,200
2,000	101,600	282,500
2,250	114,300	299,600

Since the design of the test facility permits four gear pairs to be tested at one time, the position of each gear in the tester was randomized to preclude the possibility of error due to gear location. When a gear failure occurred in a test run, all test gears were removed from the tester and tagged with a notation of the test run. The gears which did not fail were then stored for a later run at the same load levels. The procedure was followed for each test run until failure or runout of all test gears occurred.

GEAR STRESS CALCULATIONS

The gear stresses presented in this report were calculated using the methods and equations outlined in References (4) and (5). The geometry factor for the single-tooth test gear was based on an assumed gear ratio of one-to-one and thus is identical with the geometry factors for the dynamic test gear.

The equation for calculating the bending stress at the critical section of the gear tooth when loaded at the HPSTC is:

$$S_b = \frac{W_t K_o}{K_v} \times \frac{P_d}{F} \times \frac{K_s K_m}{J} \qquad (1)$$

where	S_b	=	bending stress at the critical section
	W_t	=	tangential load
	K_o	=	overload factor
	K_v	=	dynamic factor
	P_d	=	diametral pitch
	F	=	face width
	K_s	=	size factor
	K_m	=	load distribution factor
	J	=	geometry factor

For the test gears utilized in this program, assume

$$K_o,\ K_v,\ K_s,\ K_m, = 1.0$$

$$F = .375$$

$$P_d = 8$$

$$J = .420$$

Therefore, the bending stress is given by

$$S_b = \frac{W_t \times 1}{1} \times \frac{8}{0.375} \times \frac{1 \times 1}{0.420} \tag{2}$$

$$S_b = 50.79\ W_t$$

For the single-tooth specimen, the normal tooth load is

$$W_n = \frac{W_t}{\cos \phi} \tag{3}$$

and

$$S_b = 50.79 \cos \phi\ W_n$$

$$S_b = 50.79 \times .92388\ W_n \tag{4}$$

$$S_b = 46.93\ W_n$$

The surface compressive stress at the pitch point is given by

$$S_c = C_p \sqrt{\frac{W_t\ C_o}{C_v} \times \frac{C_s}{D_p\ F} \times \frac{C_m\ C_f}{I}} \tag{5}$$

where	S_c	=	compressive stress
	C_p	=	elastic coefficient
	W_t	=	tangential tooth load

C_o = overload factor

C_v = dynamic factor

D_p = operating pitch diameter

F = face width

C_s = size factor

C_m = load distribution factor

I = geometry factor

C_f = surface factor

For the test gears used in this program, assume

C_o, C_v, C_s, C_m, C_f, = 1.0

F = .375

D_p = 4.000

I = .0884

C_p = 2300

Then

$$S_c = 2300 \sqrt{\frac{W_t \times 1}{1} \times \frac{1}{4.0 \times .375} \times \frac{1 \times 1}{.0884}}$$

$$S_c = 6316 \sqrt{W_t} \qquad (6)$$

SINGLE-TOOTH TEST RESULTS

TEST DATA

The single-tooth test program was designed to determine the relative bending fatigue strengths of gears produced by the two representative roll-forming techniques as well as the bending strength of gears produced by the conventional pancake-type forging, which was established as the baseline process.

For simplification, gears made from the conventional pancake-type forging are identified as PAN and gears produced by roll-forming and roll-generating are identified as RF and RG respectively.

Fatigue Tests

A summary of the 12 fatigue test data points has been made for each process and is presented in Table V. This table contains all of the pertinent test information, including maximum test load, process, serial number, cycles to failure, and test fixture number, and comments on mode of failure and surface condition.

An analysis of the test data was made to determine if the test results were influenced to any degree by the utilization of a particular test fixture. On the basis of this analysis, it was concluded that the influence of a particular test fixture was insignificant.

To verify that the load position on the test tooth was at the correct position, random test gears were selected for dimensional inspection. The inspection results confirmed that the center of the wear pattern was located at the HPSTC.

Static Tests

A summary of single-tooth static test data is presented in Table VI.

DATA ANALYSIS

Fatigue Tests

For data analysis purposes, in order to make maximum use of the test data, the runout points generated at the 2700-pound and 3100-pound load levels were treated as normal bending failures. The rationale for this assumption is that any termination of a test prior to a bending failure is considered a runout. Consequently the conservative approach was to consider a runout point to be a failure point.

The data from the single-tooth test program was analyzed statistically with the aid of a Sikorsky Aircraft computer program. This statistical approach is based on the theory that for a particular process, there exists a stress or load level below which a failure will never occur (endurance limit), no matter how many stress cycles are imposed. In line with this theory, the mean stress/life (S-N) curve can be written in the general form:

$$S = E + \frac{\beta}{N^{\gamma}} \qquad (7)$$

where S = stress or load level

E = endurance limit

β = material constant

γ = material constant

N = cycles to failure x 10^{-6}

A curve equation of this form was used in Reference 6 to plot the results of single-tooth tests on advanced gear materials. A "best fit" S-N curve conforming to Equation (7) was derived from the test data for each process by the computer utilizing the method of least squares. The constants determined by this method satisfy the condition that the sum of the squares of the deviations of stress from the mean curve is a minimum. Using the given set of test data as input, the computer calculates the curve parameters E, β, and γ from a purely objective and unbiased viewpoint, thus eliminating the need for preplotting and curve adjusting.

After evaluation of the constants, Equation (7) was used to evaluate the mean load at various values of N for each process. Figures 19, 20, and 21 are the resulting plots of the data points and respective mean S-N curves. For comparative purposes, a composite of the three mean curves is presented in Figure 22.

The distribution of the data about the mean was analyzed statistically by assuming that each individual data point lay on an S-N curve of its own which had the same curve shape as the mean curve for the total group. The mean curve was thus shifted up or down to pass through the specified test point. The value of stress, or load, at 10^8 cycles based on an individual data point was then calculated by

$$\bar{S}_i = \frac{S_i}{\left[1 + \frac{\beta/E}{N_i^{\gamma}}\right]} \left[1 + \frac{\beta/E}{100^{\gamma}}\right] \qquad (8)$$

where $\bar{S}_i$ = equivalent load at 10^8 cycles for each data point

S_i = load at failure for each individual data point

N_i = stress at failure for each individual data point

After the data points were "stacked up" at 10^8 cycles, the standard deviation and coefficient of variation were calculated by

$$s = \sqrt{\frac{\sum(\bar{S}_i)^2 - n(\bar{X})^2}{n - 1}} \tag{9}$$

where s = unbiased standard deviation

$\bar{X}$ = mean fatigue strength at 10^8 cycles for the total group

n = number of test points

and $\frac{s}{\bar{X}}$ = coefficient of variation

The mean fatigue strengths and standard deviations at 10^8 cycles, the coefficients of variation, and the constants of the S-N curve equation for the mean curves are summarized in Table VII.

The probable error between the mean of the test sample at 10^8 cycles and the true mean was calculated using the statistical tables of Reference 3. Based on the probable error of the mean, the spread in the mean fatigue strength at 10^8 cycles for a confidence level of 99 percent was calculated for each process. This data is presented in Table VIII.

Static Tests

The mean ultimate test loads, standard deviations, and coefficients of variation were determined for the gears of each process. Comparative evaluations were performed using a single tail "t" test which is a statistical method to determine the significant difference between the means of two sets of test data. The results of these analyses, for a 90 percent confidence level, are presented in Table IX.

TABLE V. TEST RESULTS - SINGLE-TOOTH FATIGUE TESTS

Test Run	Maximum Test Load (lb)	Process	S/N	Cycles $\times 10^{-6}$	Test Fixture	Comments
1	2700	RF	13 E	0.221	2	Fractured-pitted
2	2700	PAN	03 A	17.654	2	Runout-pitted
3	2700	RG	25 D	17.520	1	Runout-pitted
4	5500	RG	26 C	0.00290	2	Fractured-clean
5	3100	RF	12 C	0.153	2	Fractured-pitted
6	3900	PAN	01 B	0.032	2	Fractured-pitted
7	3100	RG	26 A	0.140	2	Fractured-clean

TABLE V - Continued						
Test Run	Maximum Test Load (lb)	Process	S/N	Cycles x 10^{-6}	Test Fixture	Comments
8	3100	RG	25 C	0.116	2	Fractured-pitted
9	3100	RF	13 D	0.211	1	Fractured-pitted
10	5500	PAN	01 D	0.0033	2	Fractured-pitted
11	5500	RF	12 B	0.0083	1	Fractured-pitted
12	2700	RG	26 D	21.830	1	Fractured-clean
13	2700	PAN	01 C	0.304	2	Fractured-pitted
14	3100	PAN	03 C	0.120	2	Fractured-clean
15	3100	PAN	01 A	0.165	2	Fractured-clean
16	3900	RG	25 B	0.0520	2	Fractured-pitted
17	3900	RF	13 A	0.0390	2	Fractured-pitted
18	5500	RF	13 C	0.00610	2	Fractured-pitted
19	5500	PAN	03 B	0.00380	2	Fractured-pitted
20	5500	RG	25 A	0.00230	2	Fractured-clean
21	2700	RF	12 A	0.310	2	Fractured-clean
22	2700	RG	22 D	77.495	2	Fractured-clean
23	3100	RF	15 B	17.790	1	Runout-clean
24	2700	PAN	08 A	24.946	1	Fractured-clean
25	2700	RF	15 A	25.310	2	Runout-clean
26	3100	RG	22 B	0.272	1	Fractured-clean
27	3900	RG	26 B	0.0280	1	Fractured-clean
28	3900	PAN	03 B	0.0370	1	Fractured-pitted
29	5500	RG	22 A	0.00410	1	Fractured-clean
30	3900	RG	22 C	0.0440	1	Fractured-clean
31	3900	PAN	08 C	0.120	1	Fractured-clean
32	5500	PAN	08 B	0.00930	1	Fractured-clean
33	3900	RF	15 D	0.0780	1	Fractured-clean
34	3100	PAN	08 D	19.870	1	Runout-clean
35	5500	RF	15 C	0.0100	1	Fractured-clean
36	3900	RF	12 L	0.0760	1	Fractured-clean

TABLE VI. TEST RESULTS - SINGLE-TOOTH STATIC TESTS

Process	S/N	Tooth	Ultimate Load (lb)
Conventional	02	A	8825
		B	9700
		C	9850
		D	9550
Roll-Generated	23	A	7800
		B	8230
		C	7650
		D	7550
	24	A	7550
		B	8150
		C	8250
		D	7900
Roll-Formed	14	A	8700
		B	9000
		C	8900
		D	9000
	16	A	8000
		B	9125
		C	9500
		D	8450

TABLE VII. ANALYSIS OF SINGLE-TOOTH FATIGUE TEST RESULTS

Manufacturing Process	No. of Test Points (n)	E (lb)	β	γ	$\overline{X}$ * (lb)	s** (lb)	s/$\overline{X}$ (%)	Mean Fatigue Strength (%)
Roll-Formed	12	2634	260	0.493	2661	245	9.4	100.1
Roll-Generated	12	2463	465	0.322	2568	129	5.0	96.6
Conventional	12	2585	397	0.366	2659	255	9.6	100.0

*Mean fatigue strength at 10^8 cycles

**Unbiased estimate of standard deviation at 10^8 cycles

TABLE VIII. MEAN FATIGUE STRENGTHS AT 99 PERCENT CONFIDENCE LEVEL, SINGLE-TOOTH TESTS

Manufacturing Process	No. of Test Points (n)	s (lb)	Probable Error at 99% Confidence (lb)	Upper* Limit (lb)	Mean* (lb)	Lower* Limit (lb)
Roll-Formed	12	245	±220	2881	2661	2441
Roll-Generated	12	129	±116	2684	2568	2452
Conventional	12	255	±229	2888	2659	2430

*Applicable at 10^8 cycles of vibratory stress

TABLE IX. RESULTS OF STATISTICAL TESTS ON STATIC TEST DATA

Processes Compared	$\overline{X}$ (lb)	s (lb)	$s/\overline{X}$ %	n	Results of Single-Tailed "t" Test
Conventional versus	9480	454	5	4	The ultimate static strength of conventionally forged gear teeth is at least 265 pounds higher than that of roll-formed gear teeth.
Roll-Formed	8830	454	5	8	
Conventional versus	9480	454	5	4	The ultimate static strength of conventionally forged gear teeth is at least 1300 pounds higher than that of roll-generated gear teeth.
Roll-Generated	7890	295	4	8	
Roll-Formed versus	8830	454	5	8	The ultimate static strength of roll-formed gear teeth is at least 691 pounds higher than that of roll-generated gear teeth.
Roll-Generated	7890	295	4	8	

$\overline{X}$ = Arithmetic Mean Load

s = Sample standard deviation

n = Number of test points

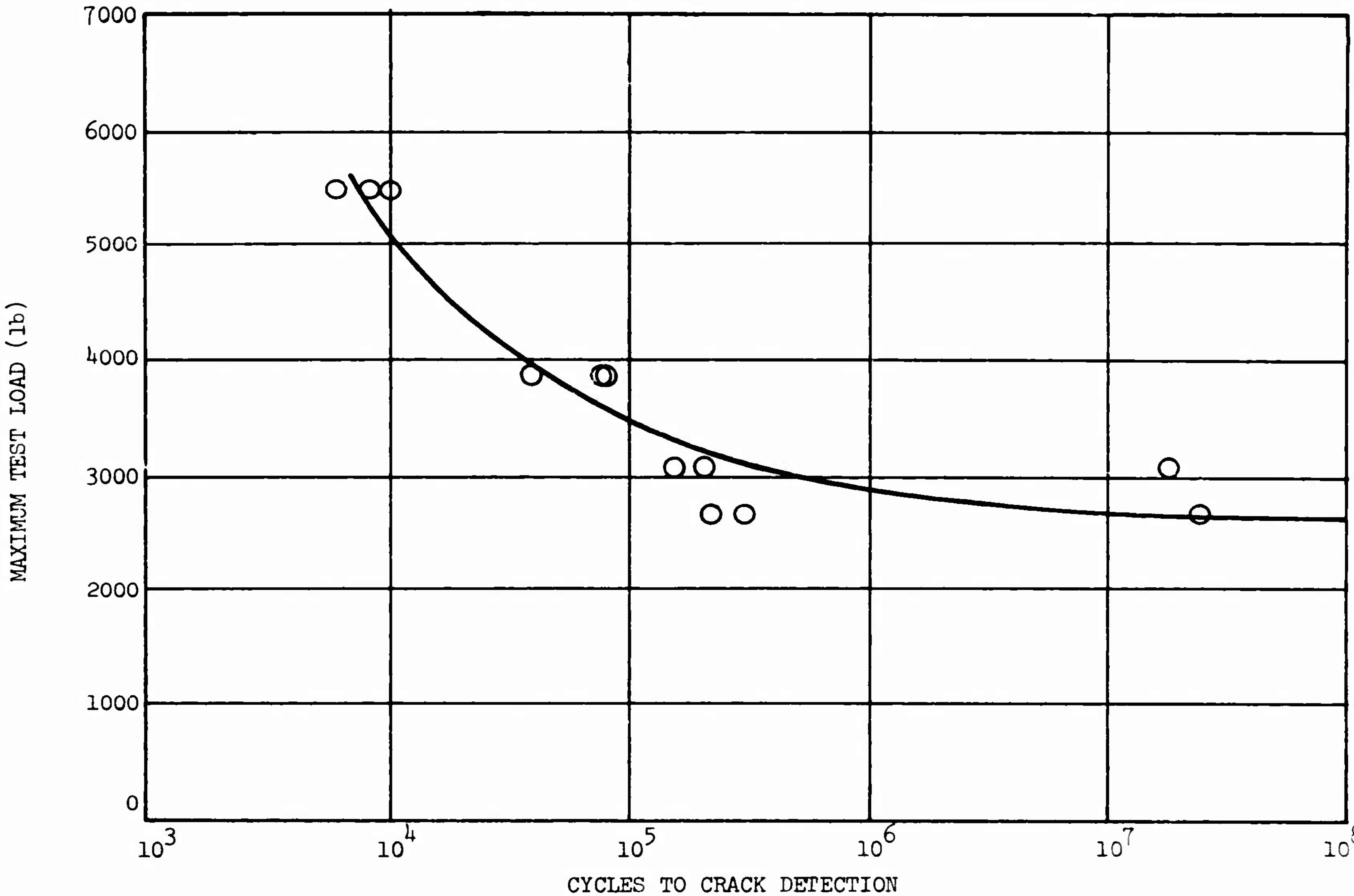


Figure 19. Single-Tooth Fatigue Test Results - Roll-Formed Gears.

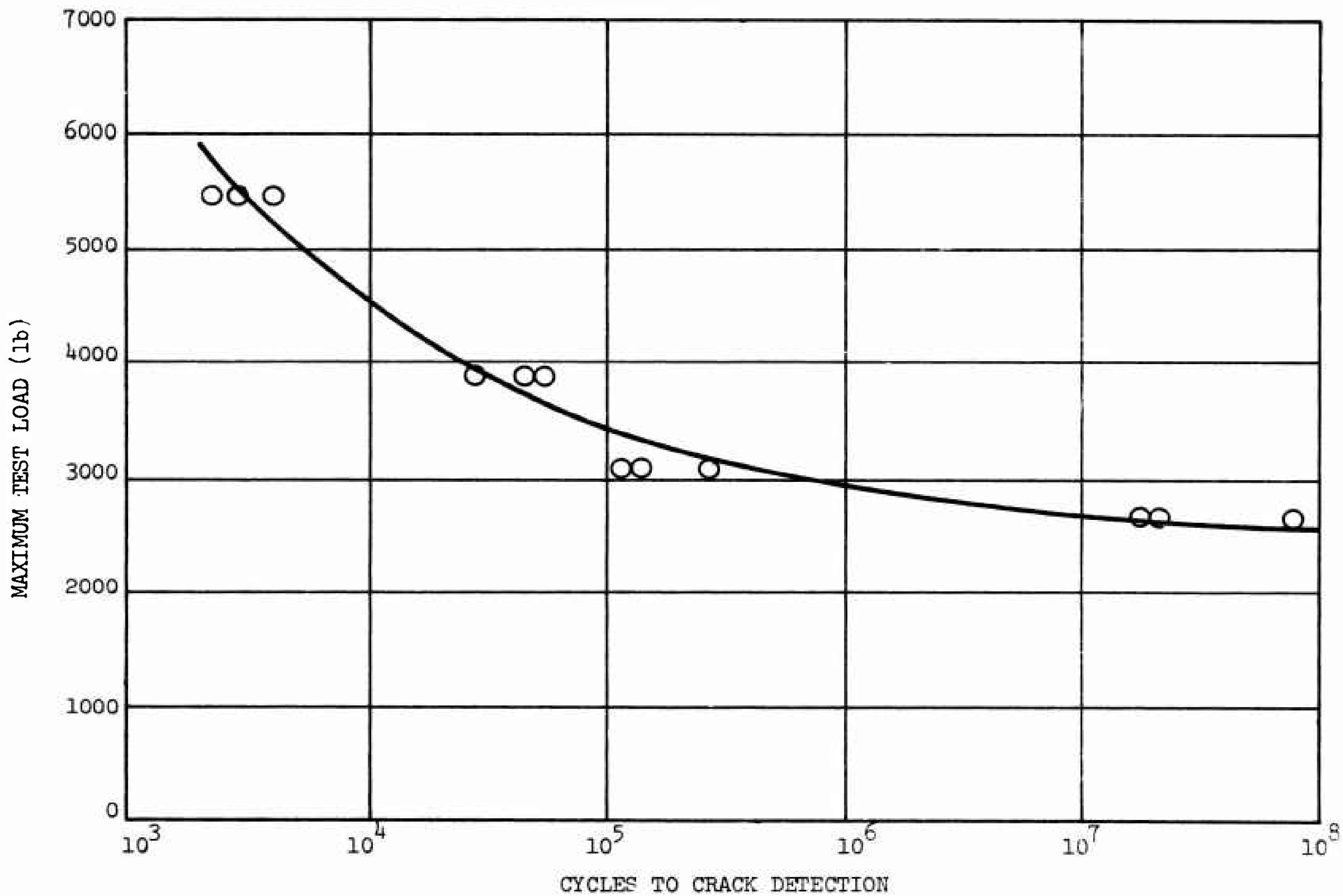


Figure 20. Single-Tooth Fatigue Test Results - Roll-Generated Gears.

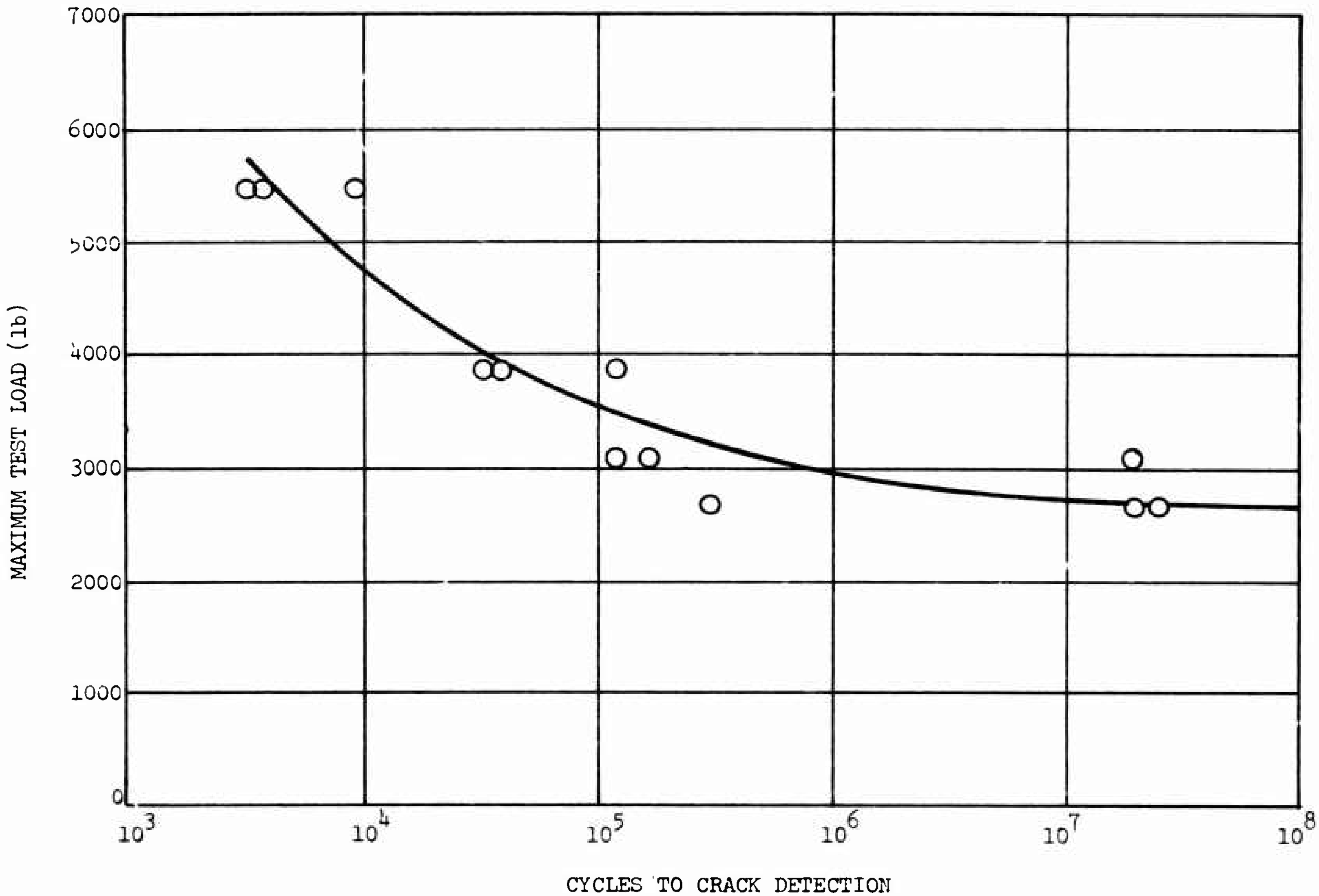


Figure 21. Single-Tooth Fatigue Test Results - Conventionally Processed Gears.

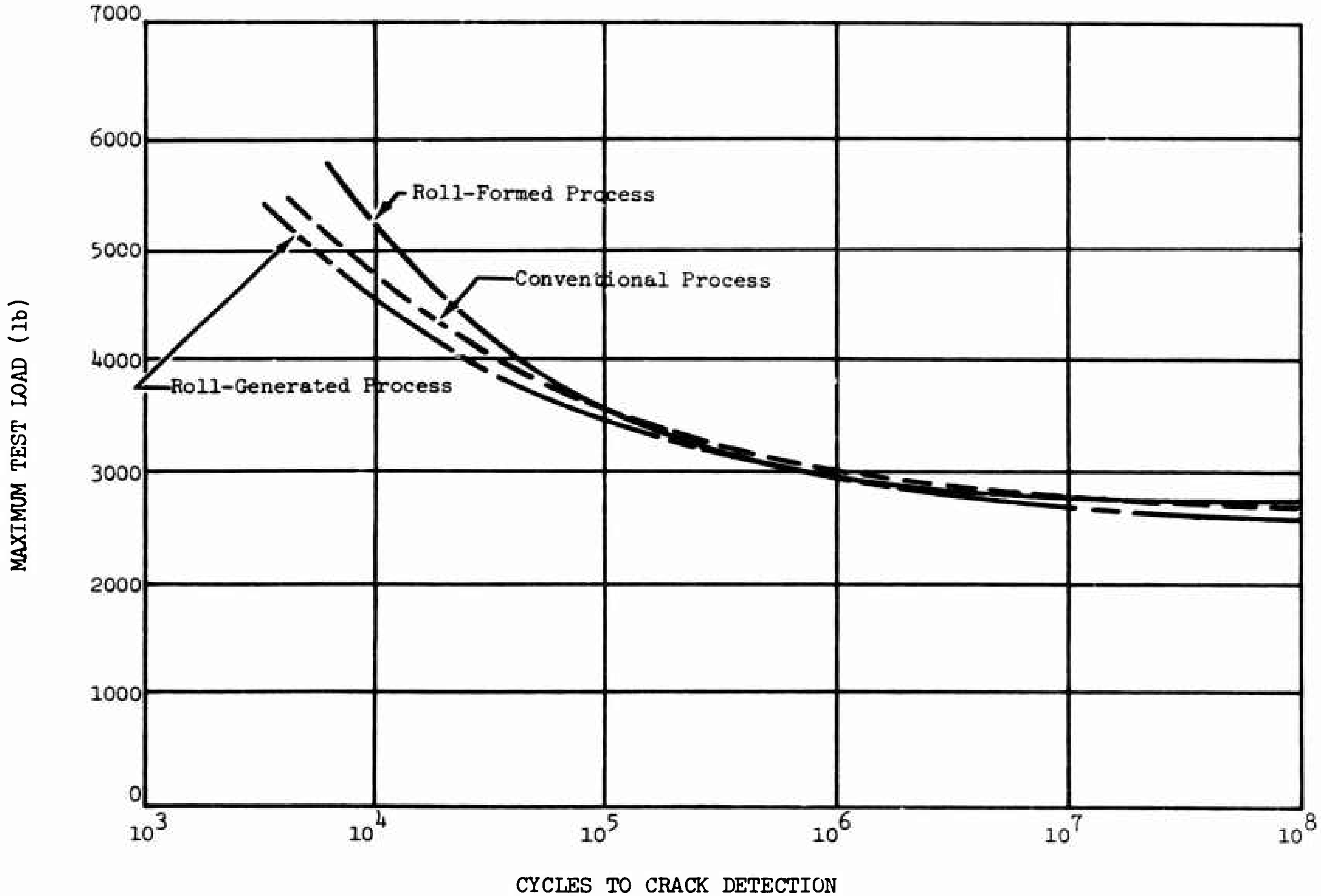


Figure 22. Comparative Results of Single-Tooth Fatigue Tests.

DYNAMIC TEST RESULTS

TEST DATA

The dynamic test program was established to determine the relative dynamic fatigue strengths of gears produced by the two roll-forming processes when run in a gearbox environment at 9200 rpm. Although the conventional forging process was not included in this test series, the test data can be correlated with the baseline process when the results of a current program[1] on forged gears become available.

Based on the initial results obtained in this program, which was conducted on the same test rig, the lower load level for this test phase was set at 1500 pounds. This choice resulted in failure times in excess of 10 million cycles. Such high-time points have a significant influence in determining the endurance limit and S-N curve shape. The upper load level was selected at 2250 pounds, again based on previous experience with this test gear, to give failure times in excess of 10,000 cycles. Two other load levels were spaced between these extremes. Runout was established at 100 million cycles. A summary of the fatigue test data is presented in Table X. The table includes transmitted tangential tooth load, process, serial number, and total accumulated cycles.

DATA ANALYSIS

The mean S-N curves for each process were drawn through the test data points using the same procedures used in the single-tooth test phase. The resulting curve plots are shown in Figures 23 and 24. For comparative purposes, the two curves are shown superimposed in Figure 25. The equivalent loads at 10^8 cycles for each data point were analyzed statistically to determine the mean fatigue strength, standard deviation, and coefficient of variation for each process. These values and the constants of the S-N relation are summarized in Table XI.

The probable error between the mean of the test sample at 10^8 cycles and the true mean was calculated using the statistical tables of Reference 3. The spread in the mean fatigue strength at 10^8 cycles for a confidence level of 99 percent was calculated for each process. This data is presented in Table XII.

[1]Dynamic Test of Gears Manufactured by Advanced Forging Techniques, USAAMRDL Contract DAAJ02-69-C-0060

TABLE X. TEST RESULTS, DYNAMIC FATIGUE TESTS				
Tangential Tooth Load (lb)	Process	S/N	Total Accumulated Cycles x 10^{-6}	Comments
1500	RF	35	26.514	Tooth fracture
1500	RG	72	70.371	Tooth fracture
1500	RG	94	70.371	Tooth fracture
1500	RG	88	76.434	Tooth fracture
1500	RF	61	34.535	Tooth fracture
1500	RG	81	76.654	Tooth fracture
1500	RF	36	100.280	Runout
1500	RF	32	81.034	Tooth fracture
1750	RG	87	22.015	Tooth fracture
1750	RG	95	2.870	Tooth fracture
1750	RG	92	23.616	Tooth fracture
1750	RG	86	16.247	Tooth fracture
1750	RG	79	13.082	Tooth fracture
1750	RF	50	18.225	Tooth fracture
1750	RF	66	11.077	Tooth fracture
1750	RF	46	10.525	Tooth fracture
1750	RF	37	6.550	Tooth fracture
1750	RF	31	6.339	Tooth fracture
2000	RG	78	0.506	Tooth fracture
2000	RF	34	0.672	Tooth fracture
2000	RF	39	0.754	Tooth fracture
2000	RG	77	0.690	Tooth fracture
2000	RG	85	1.072	Tooth fracture
2000	RG	91	1.195	Tooth fracture
2000	RF	64	0.497	Tooth fracture
2000	RF	52	1.444	Tooth fracture
2250	RG	71	0.396	Tooth fracture
2250	RG	93	0.408	Tooth fracture
2250	RF	49	0.460	Tooth fracture
2250	RG	73	0.202	Tooth fracture

TABLE X - Continued				
Tangential Tooth Load (lb)	Process	S/N	Total Accumulated Cycles x 10^{-6}	Comments
2250	RF	45	0.791	Tooth fracture
2250	RF	60	0.414	Tooth fracture
2250	RG	82	0.286	Tooth fracture
2250	RF	33	0.595	Tooth fracture

TABLE XI. ANALYSIS OF FATIGUE TEST RESULTS, DYNAMIC TESTS

Manufacturing Process	No. of Test Points (n)	E (lb)	β	γ	$\bar{X}$ * (lb)	s** (lb)	$s/\bar{X}$
Roll-Formed	17	1331	683	.351	1467	79.2	5.3
Roll-Generated	17	1422	536	.361	1524	76.7	5.0

*Mean fatigue strength at 10^8 cycles

**Unbiased estimate of standard deviation at 10^8 cycles

TABLE XII. MEAN FATIGUE STRENGTHS AT 99 PERCENT CONFIDENCE LEVEL, DYNAMIC TESTS

Manufacturing Process	No. of Test Points (n)	s (lb)	Probable Error at 99% Confidence (lb)	Upper* Limit (lb)	Mean* (lb)	Lower* Limit (lb)
Roll-Formed	17	79.2	± 56	1523	1467	1411
Roll-Generated	17	76.7	± 54	1578	1524	1470

*Applicable at 10^8 cycles of vibratory stress

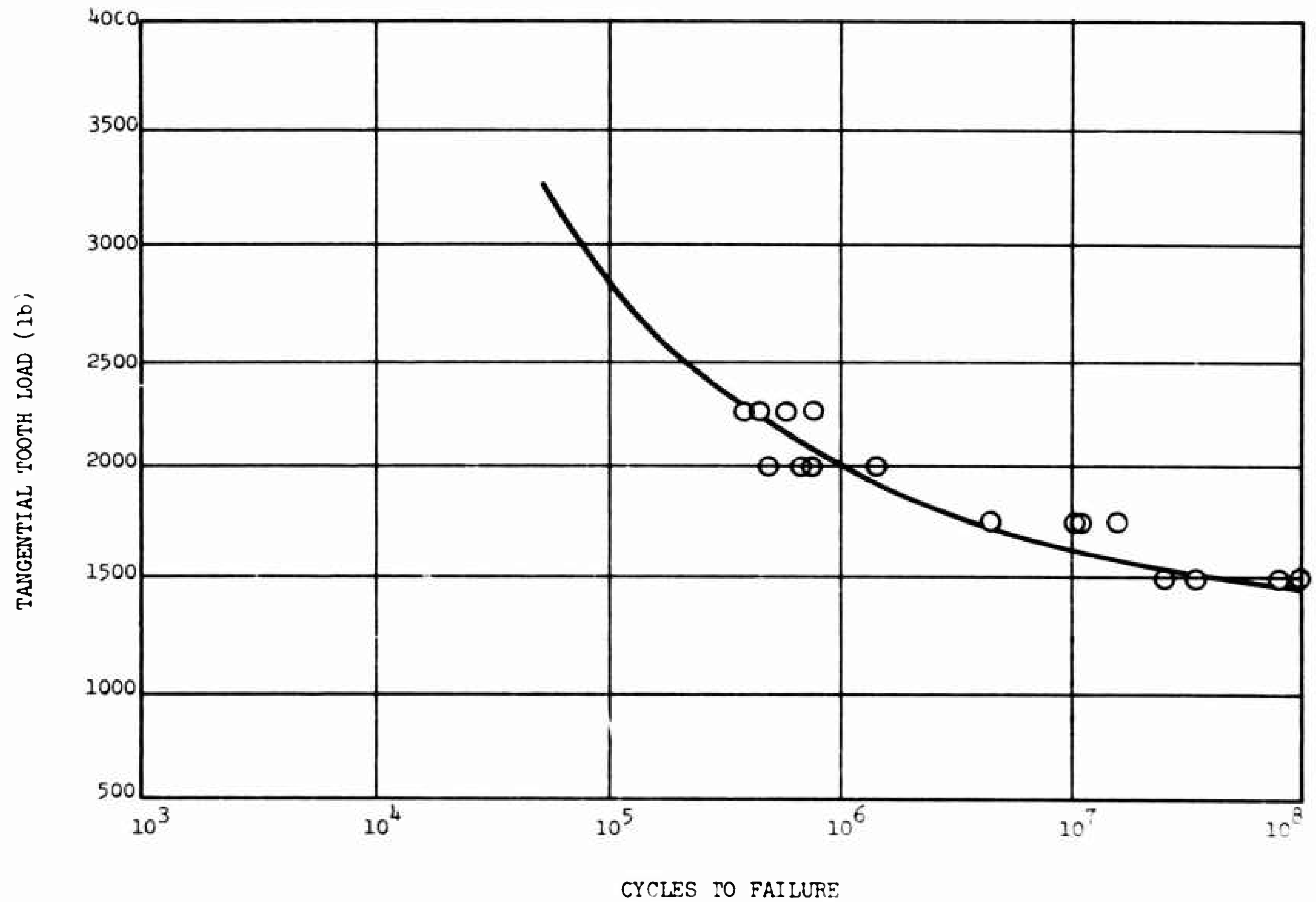


Figure 23. Dynamic Test Results for Roll-Formed Gears.

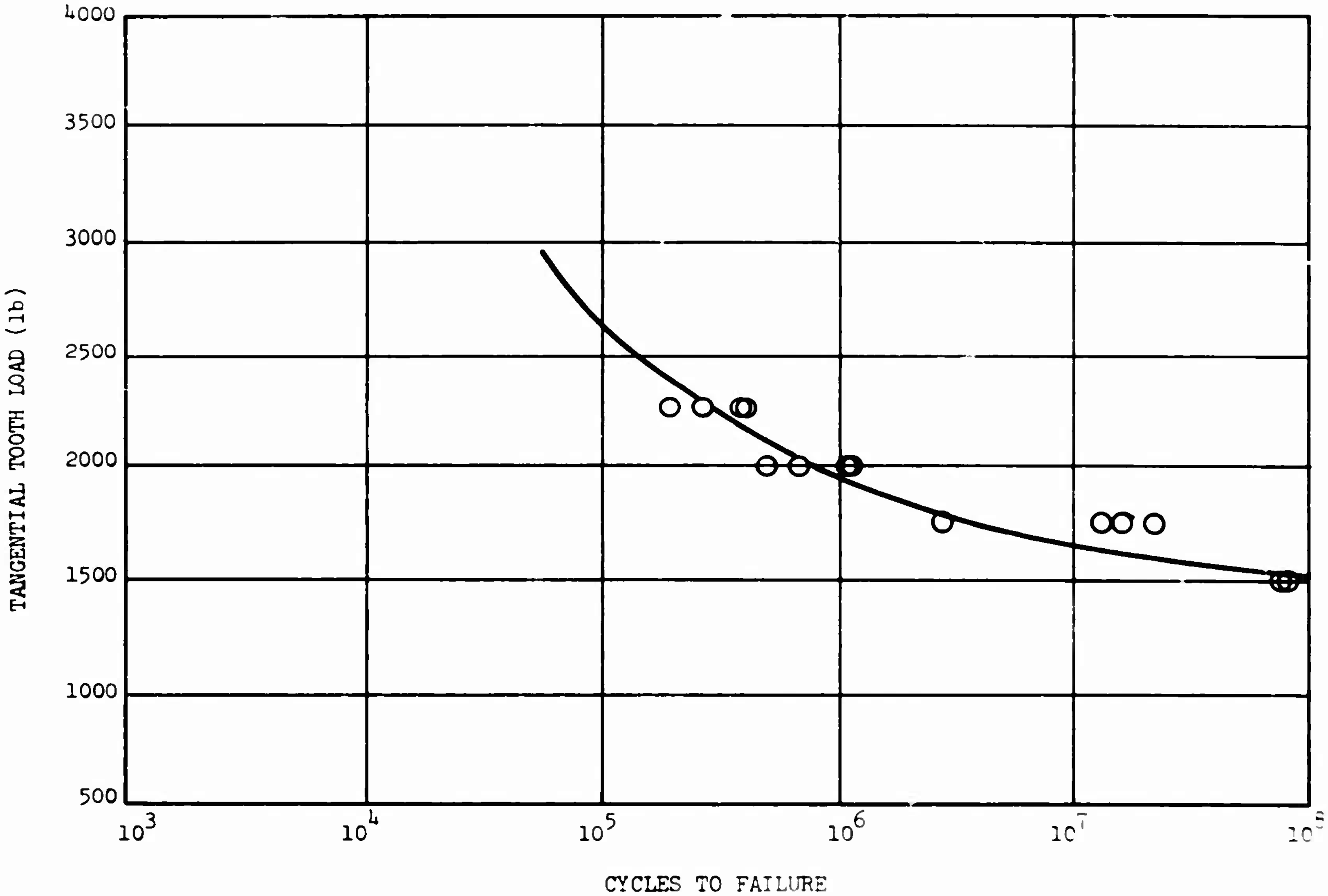


Figure 24. Dynamic Test Results for Roll-Generated Gears.

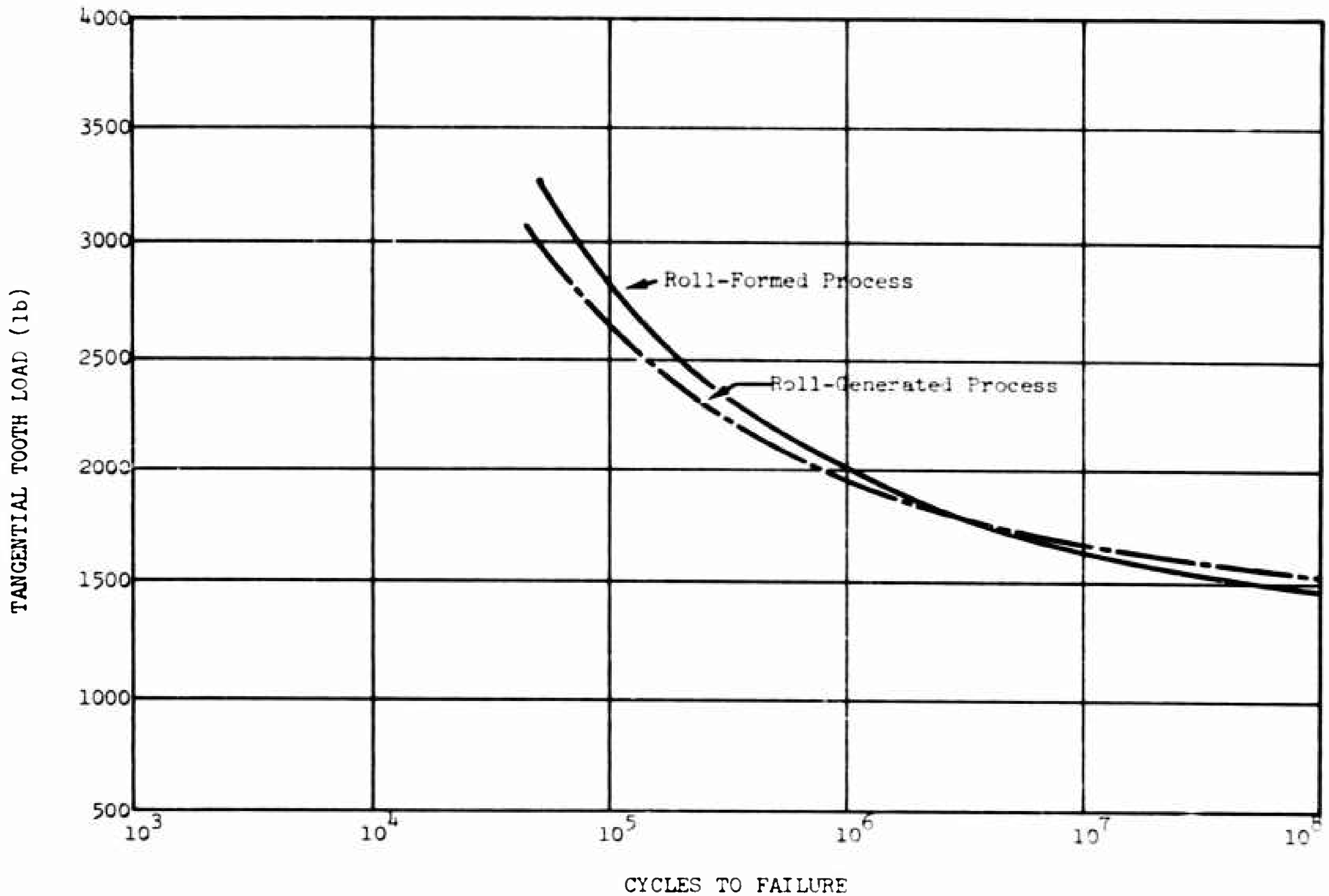


Figure 25. Comparative Results of Dynamic Tests.

METALLURGICAL EVALUATION

INVESTIGATION PROCEDURE

The following methods were used to determine the mode of failure, origin of failure, microstructure of case and core, chemical composition, grain size, grain flow, case depth, and hardness of the case and core:

1. The fractured gear teeth were examined with a low-power stereomicroscope to determine the mode and origin of failure.

2. One fractured tooth from each gear was further examined as follows:

   a. The Rockwell hardness was determined for the case and core.

   b. Fractured teeth from each process were mounted, etched with 2 percent nital solution, and examined on a metallograph to determine the microstructure of the case and core.

   c. Total case depth was determined by examination of the etched mounts under a Brinell microscope. The effective case depth was determined in terms of "Knoop" hardness on a Tukon microhardness tester for one tooth from each forging process. The case-core transition point was taken at KHN 542 (approximately equal to Rc 50). The results presented have been converted to Rc readings.

3. One gear from each forging process was analyzed on a spectrograph to determine the chemical composition. A volumetric carbon determination of the core was also conducted.

4. Grain flow was determined for a series of teeth from each of the rolling processes in the as-rolled condition. Grain flow was also determined on the finished gears for all the processes. Transverse sections from gear teeth were cut, mounted and polished. The grain flow was revealed by etching with a saturated solution of ammonium persulfate followed by a light polish with a 1-micron paste to remove smutting.

5. Prior austenitic grain size was determined on several gear teeth from each process by using a modified McQuade-Ehn test. The samples were packed in carbon and subjected to a 2-hour soak at 1600°F and then slow-cooled. The samples were then etched with a nital and picral solution to reveal the grain boundaries. The ASTM grain size was determined by using method E112-63, Reference 7, and comparison with ASTM Plate IV at 100X magnification.

FRACTURE ANALYSIS

Single-Tooth Gear Tooth Fractures

Examination of the gear-tooth-fracture interfaces revealed multiple origins extending across the tooth width, shown typically for each process in Figure 26. All the fractures were located in the area adjacent to the start of the root radius. The first gear tooth fatigue tested (S/N 13, roll-formed process) failed after 221,000 cycles at the 2700-pound load level. Examination revealed that the fatigue origins were coincident with pits in the surface as shown in Figure 27, View A. This condition was apparent on every tooth on the gear, shown typically in View B, Figure 27. This pitting occurred during the manganese phosphate coating process and is a rejectable condition. Analysis of the test data revealed that the four test points from gear S/N 13 failed lower than the other test points at that load level. Examination of the remaining gears revealed pitting of varying degrees. The surface condition at the crack site of each gear tooth is noted in Table V. A typical example of a static fractured tooth is shown in Figure 28. The static fractures are considerably more crystalline in appearance and coarse grained in texture.

Dynamic Gear Tooth Fractures

Two gears of each roll-formed process were analyzed. The examination revealed single-tooth fractures in all four gears extending across the tooth face approximately at the start of the tooth-to-root fillet radius. A typical single-tooth fracture is shown in Figure 29.

Figure 30 shows the fracture interface and origin site for a typical fracture. The gear teeth on all of the gears examined exhibited varying degrees of scuffing, surface pitting, and a contact pattern which favored one side of the tooth face.

Fluorescent magnetic particle inspection revealed an additional cracked tooth on one of the gears examined. The crack extended from a spalled area at the pitch line, down the side of the tooth, and across approximately one-half of the tooth width, as shown in Figure 31.

HARDNESS AND CASE DEPTH

Hardness of the case and core and case depths are listed in Table XIII. The results of these tests show that all the teeth tested conformed to the hardness and case depth requirements specified in Figures 1 and 2. Microhardness surveys of each process, shown in Figure 32, revealed a uniformly decreasing hardness gradient with no sharp drop-off, indicating a good transition from case to core. The effective case depth (Rc 50 min.) in each case was at the minimum drawing requirement, and for gear S/N 13 it was 0.001 inch below.

CHEMICAL ANALYSIS

Chemical analysis was performed on one of each type of gear. A tabulation of the elements is listed in Table XIV. The results of the analysis indicated that the material conformed to the requirements of AMS 6265.

MICROSTRUCTURE

Metallographic examination revealed a typical martensitic case microstructure with no evidence of retained austenite or carbide network. Typical microstructures of case and core for each process are shown in Figures 33 and 34. No significant difference exists between manufacturing process or individual gears in either case or core structure. These structures meet the present requirements of Sikorsky Standard, SS-8015, Carburizing Procedures and Requirements, dated October 5, 1967.

GRAIN FLOW ANALYSIS

Figures 35 and 36 show the grain flow of the gear blanks from each rolling process before machining or heat treatment. Figures 37 and 38 show these same processes in the finished gear. Comparison revealed that roll-formed gears exhibited grain flow which followed the contour of the gear tooth from the root to the crest and extended approximately 0.016 to 0.020 inch below the surface. Roll-generated gears exhibited no grain flow in the root radius and a very small amount, 0.002 inch, around the pitch line to the crest. Grain flow in the roll-generated gears is similar to that found in conventionally processed gears. Comparison of the root radii revealed some cutting of the grain flow lines of roll-formed gears (arrows in Figure 39, View A), while in contrast, all the grain flow lines in roll-generated gears are cut in finish grinding (Figure 39, View B).

GRAIN SIZE

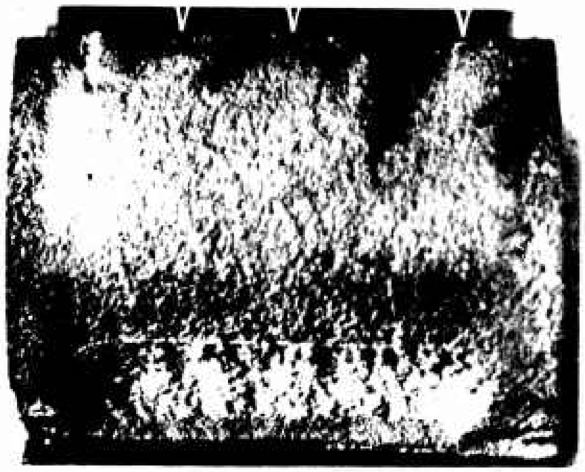
The prior austenitic grain size for all three manufacturing processes was found to be equivalent to size 8 when compared with ASTM plate No. IV using ASTM method E112-63. This grain size is within the requirements of AMS 6265, which calls for grain size predominantly 5 or finer with occasional grains as large as 3 permissible. Figure 40 shows the grains from a typical specimen at 1000X magnification.

TABLE XIII. SUMMARY OF GEAR HARDNESS AND CASE DEPTH RESULTS

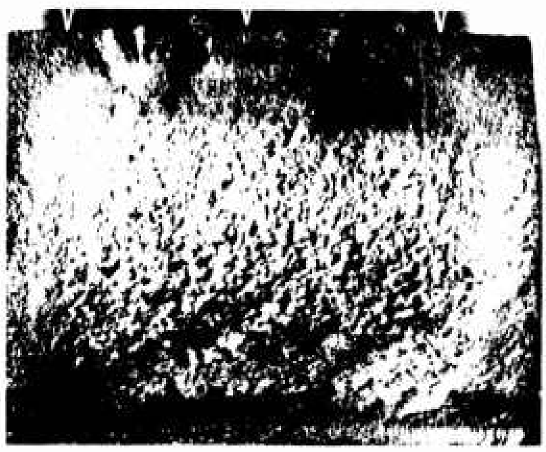
Process	Serial No.	Case Hardness Rc	Core Hardness Rc	Total Visual Case Depth (in.)	Effective Case Depth, Rc 50 (in.)
Conventional	1	59.5 - 60.5	36	.056	
	3	60.5	36	.056	.036
	8	58.5 - 59.5	38	.059	
	9	59	36.5	.058	
Roll-Formed	12	59-60	37 - 37.5	.058	
	13	58-60	36.5 - 37.5	.051	.034
	14	58	35-36	.058	
	15	59-59.5	37 - 37.5	.062	
	16	58-58.5	35.5 - 36.5	.059	
Roll-Generated	22	61 - 61.5	35-36	.057	
	23	60 - 60.5	36.5 - 37	.055	
	24	59 - 61.5	36 - 37	.064	
	25	60-61	37	.047	.036
	26	60.5 - 61	36 - 36.5	.058	
Required		58-64	34-40	No Requirement	.035 - .045

TABLE XIV. CHEMICAL ANALYSIS

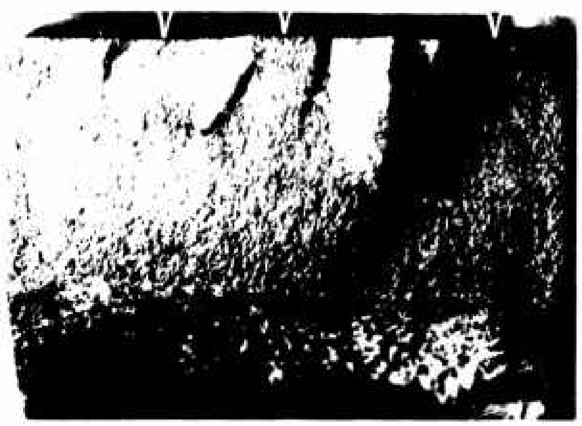
Process	C	Mn	Ni	Cr	Mo	Si	Fe
Roll-Formed	.12	0.52	3.18	1.34	0.09	0 29	Balance
Roll-Generated	.11	0.52	3.20	1.28	0.09	0.29	Balance
Conventional	.11	0.52	3.18	1.30	0.09	0.29	Balance
Requirements of AMS 6265	0.07-0.13	0.40-0.73	3.00-3.50	1.00-1.40	0.08-0.15	0.20-0.35	Balance



Roll-Formed

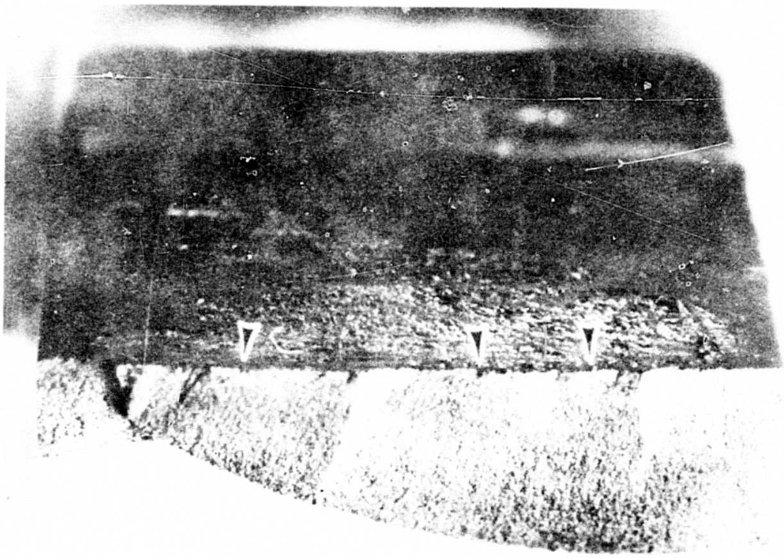


Roll-Generated



Conventional

Figure 26. Single-Tooth Fatigue Origins and Fracture Interfaces.



View A

View B

Figure 27. General Pitting Condition Observed on Some Test Gears.

Figure 28. Typical Single-Tooth Static Fracture.

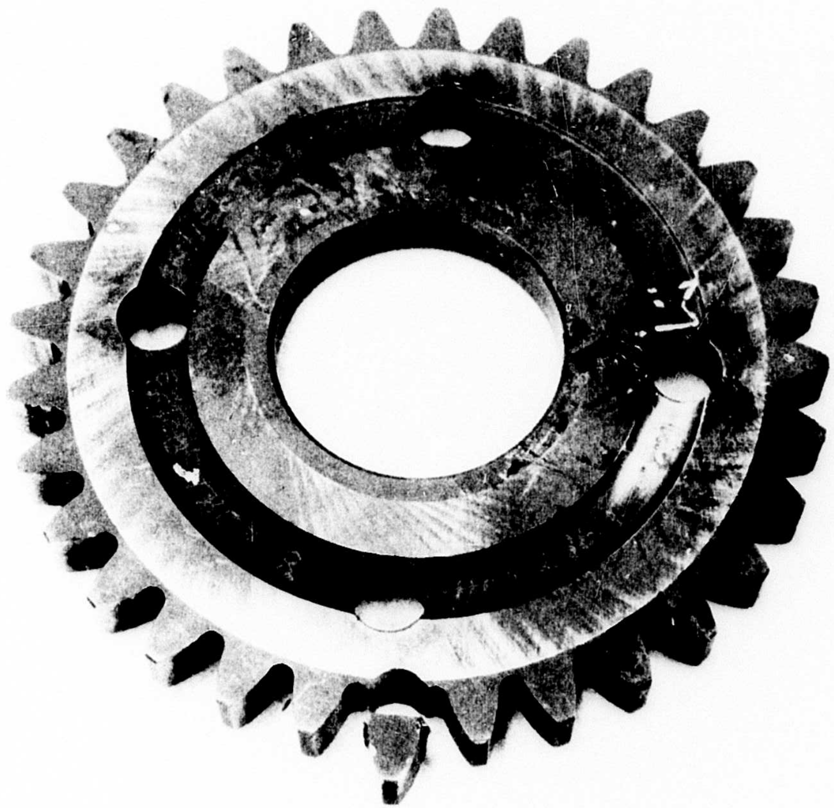


Figure 29. Dynamic Gear Tooth Fracture.

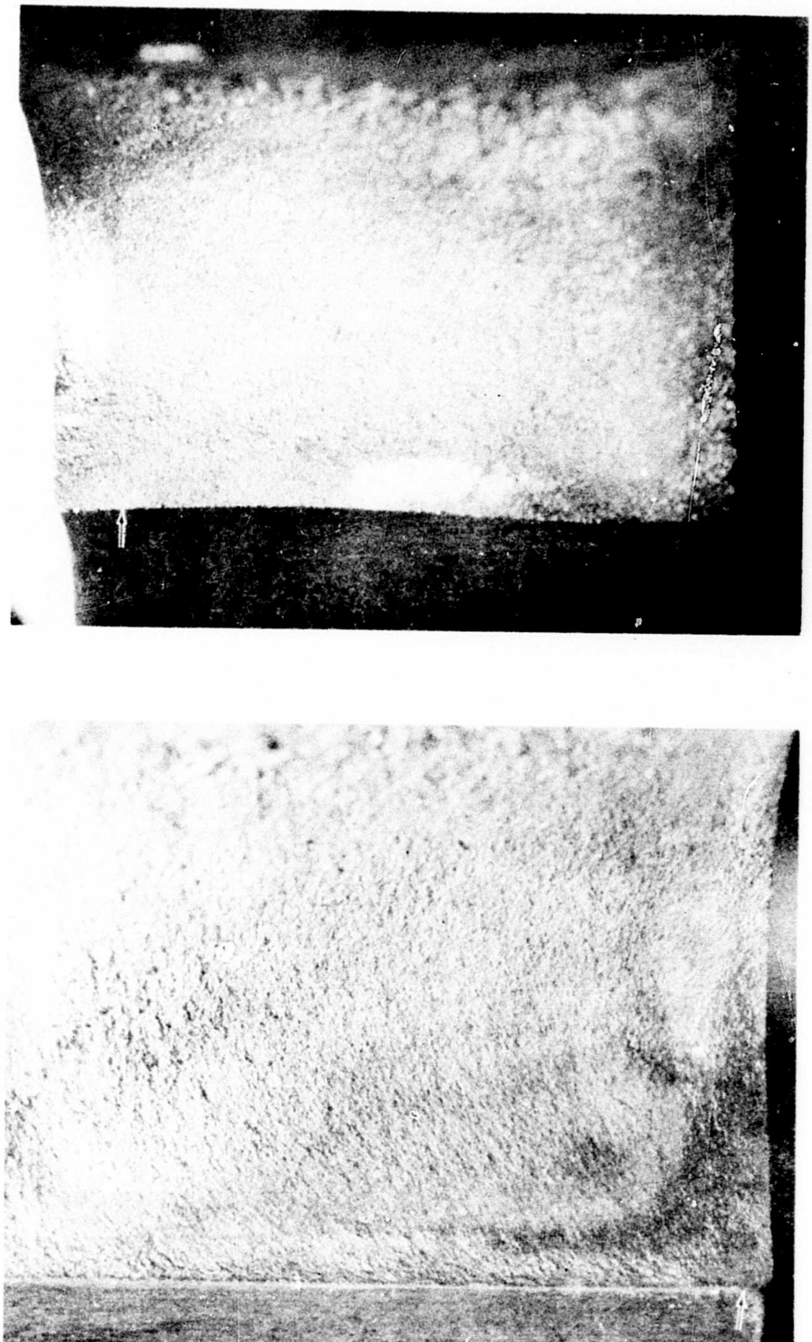


Figure 30. Dynamic Fatigue Origins and Fracture Interfaces.

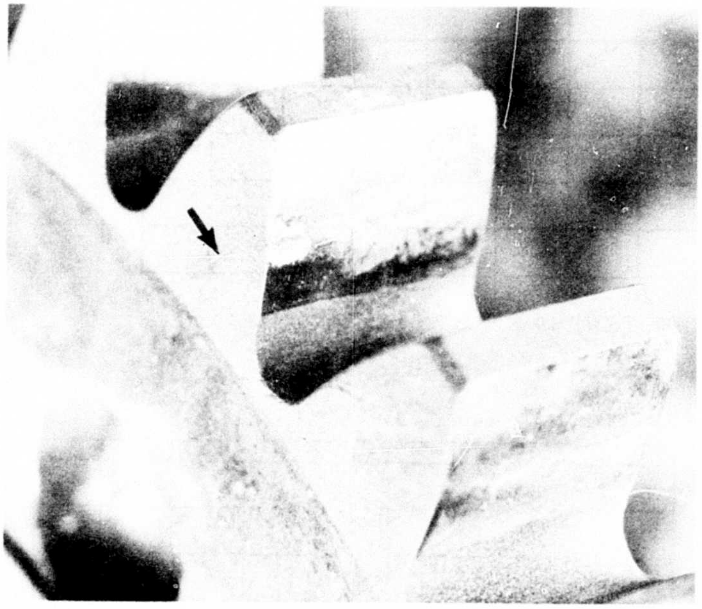
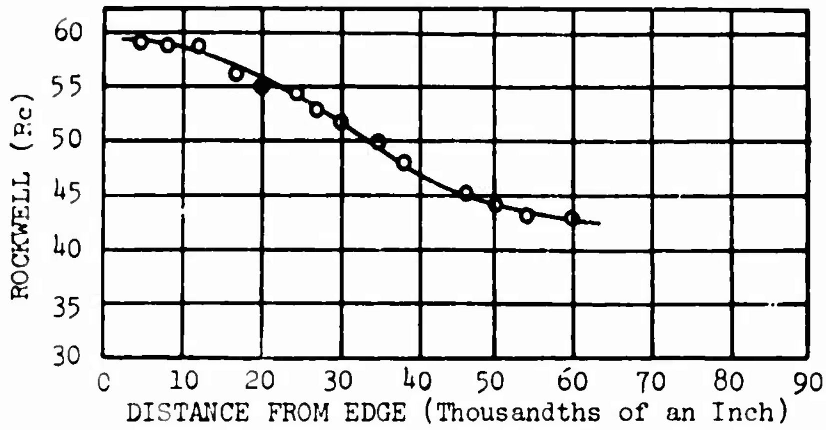
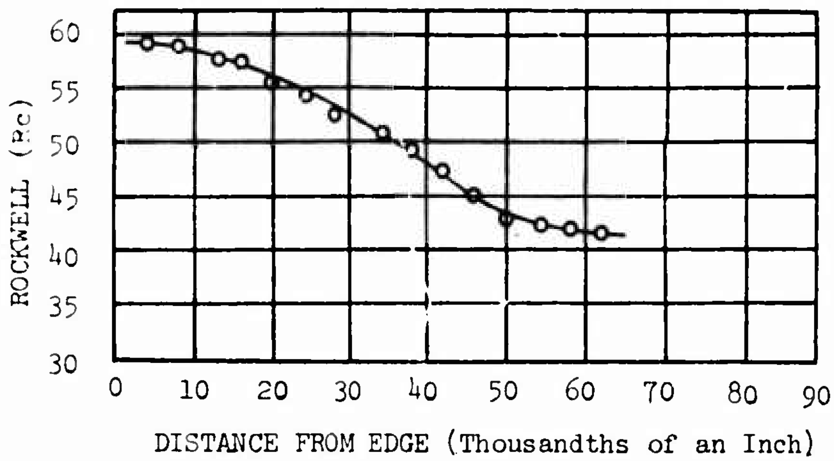


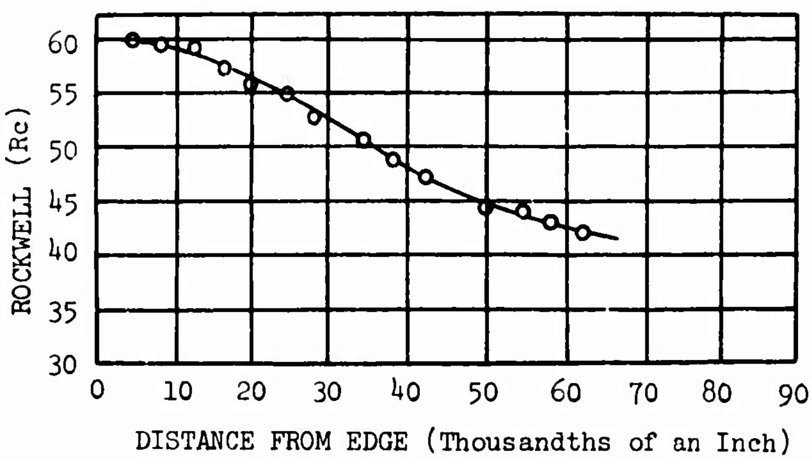
Figure 31. Fatigue Crack Originating From Surface Pit.



ROLL-FORMED PROCESS

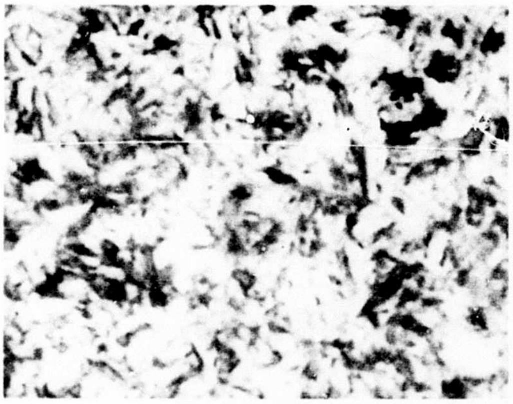


ROLL-GENERATED PROCESS

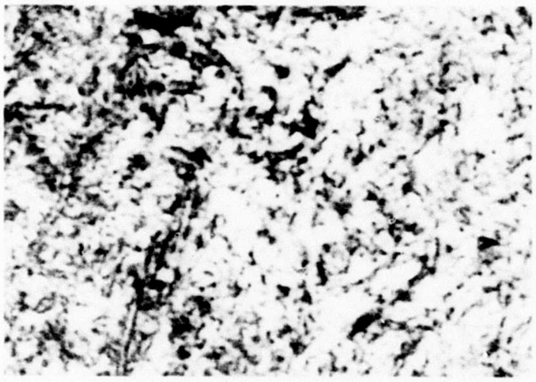


CONVENTIONAL PROCESS

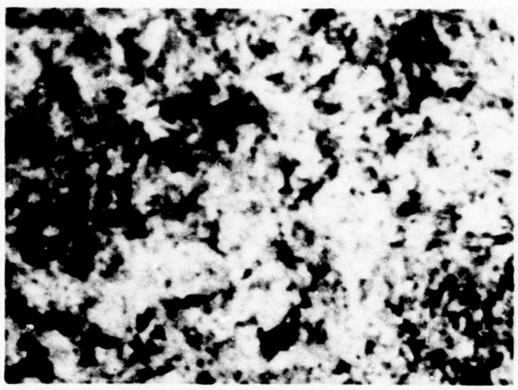
Figure 32. Typical Microhardness Surveys.



Roll-Formed



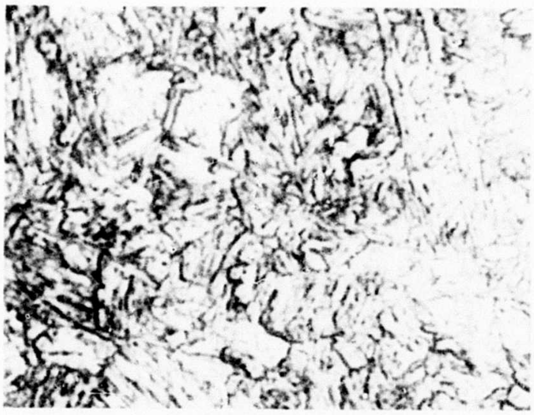
Roll-Generated



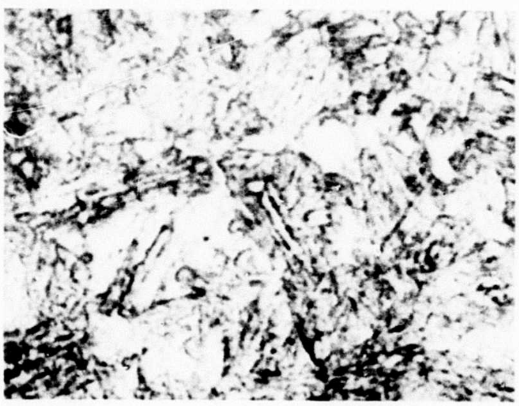
Conventional

Figure 33. Typical Case Microstructures.

Roll-Formed



Roll-Generated



Conventional

Figure 34. Typical Core Microstructures.

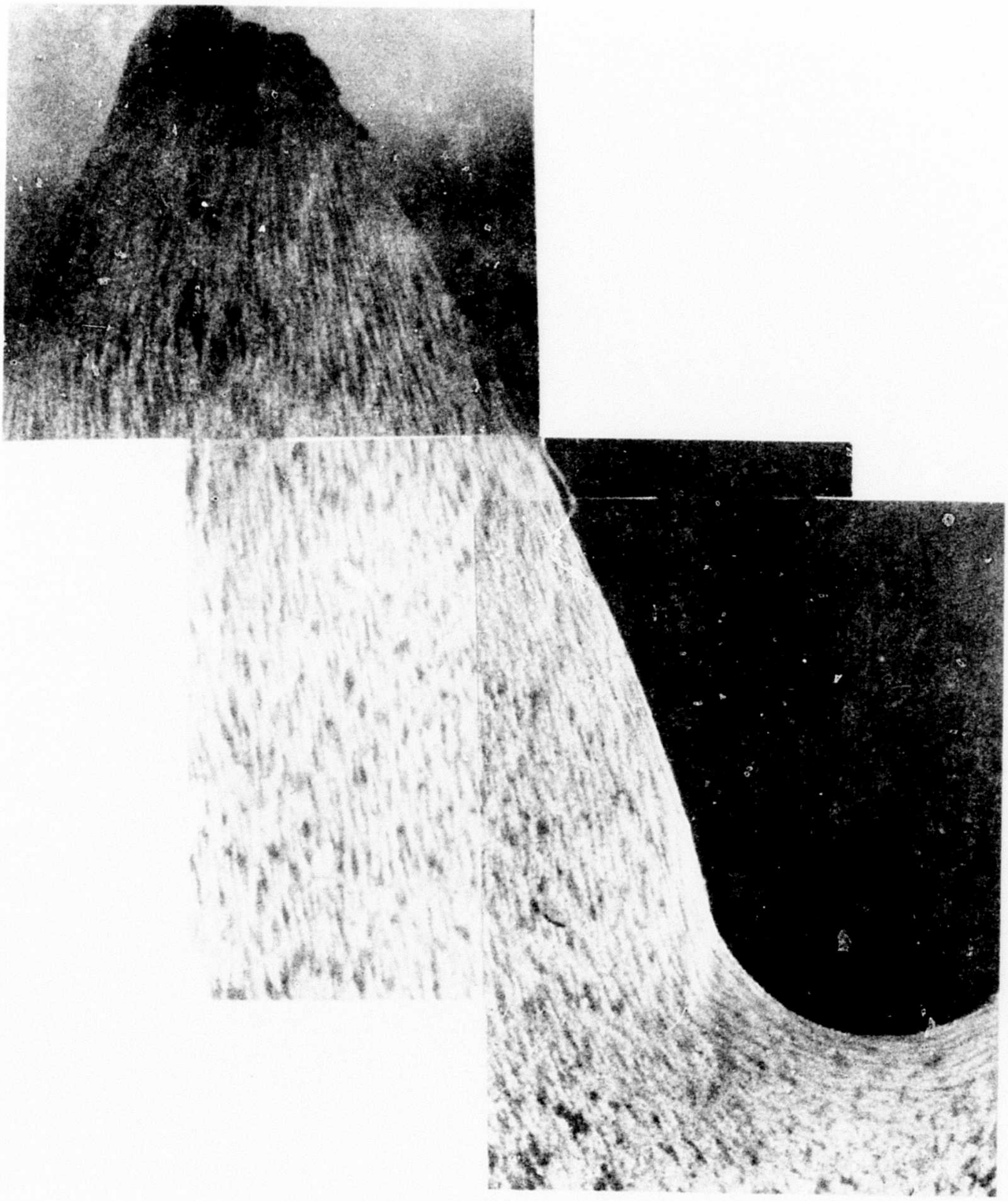


Figure 35. Grain Flow Pattern - Roll-Formed Gear Blank.

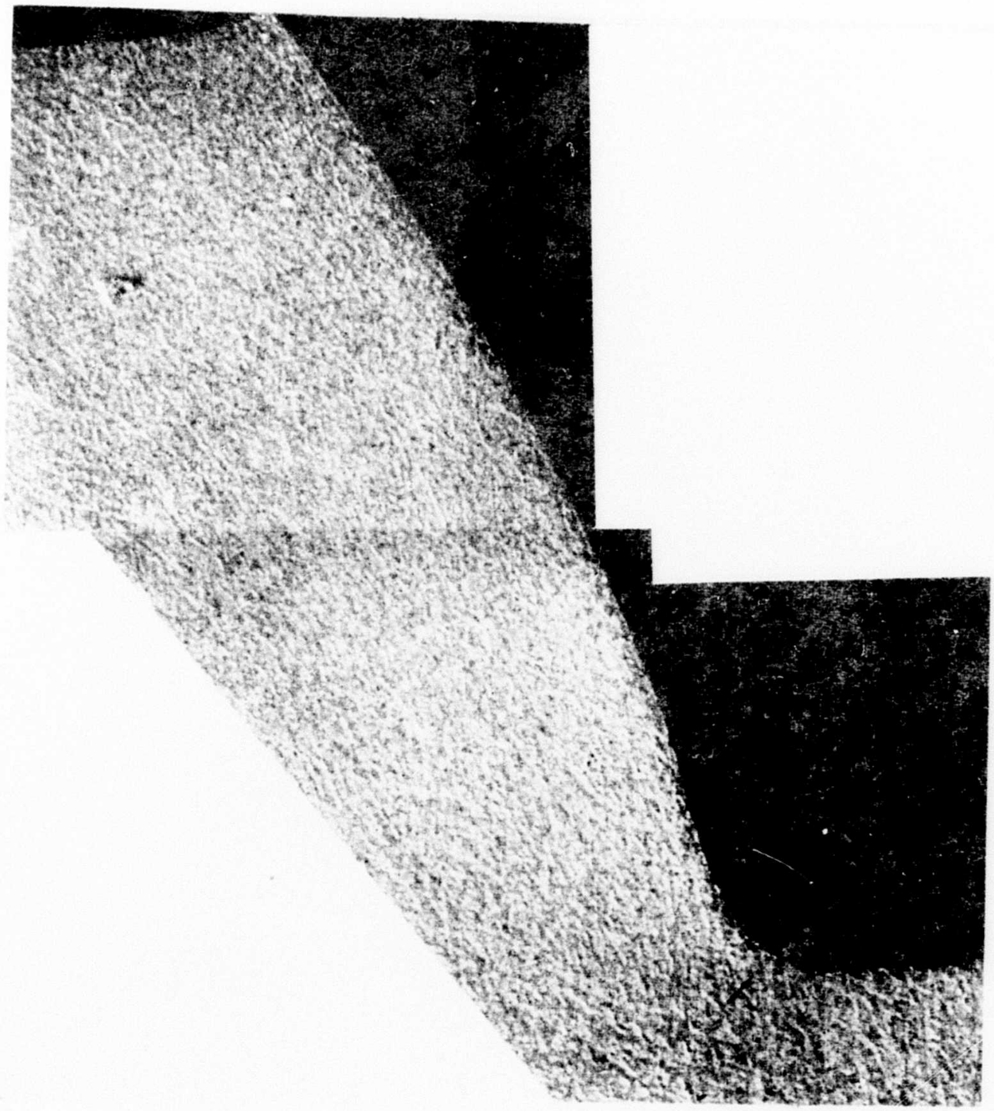


Figure 36. Grain Flow Pattern - Roll-Generated Gear Blank.

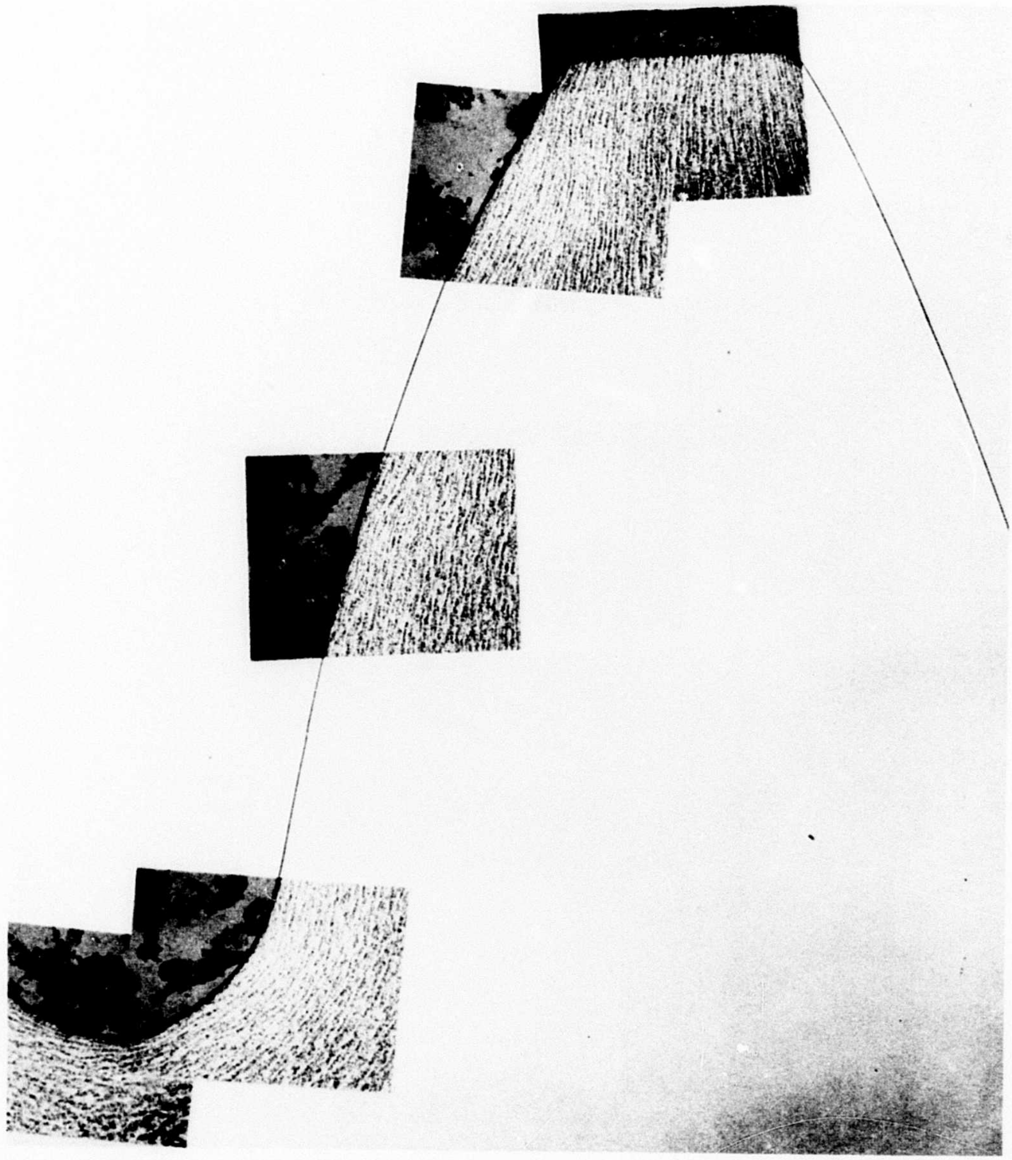


Figure 37. Grain Flow Pattern After Grinding - Roll-Formed Gear.

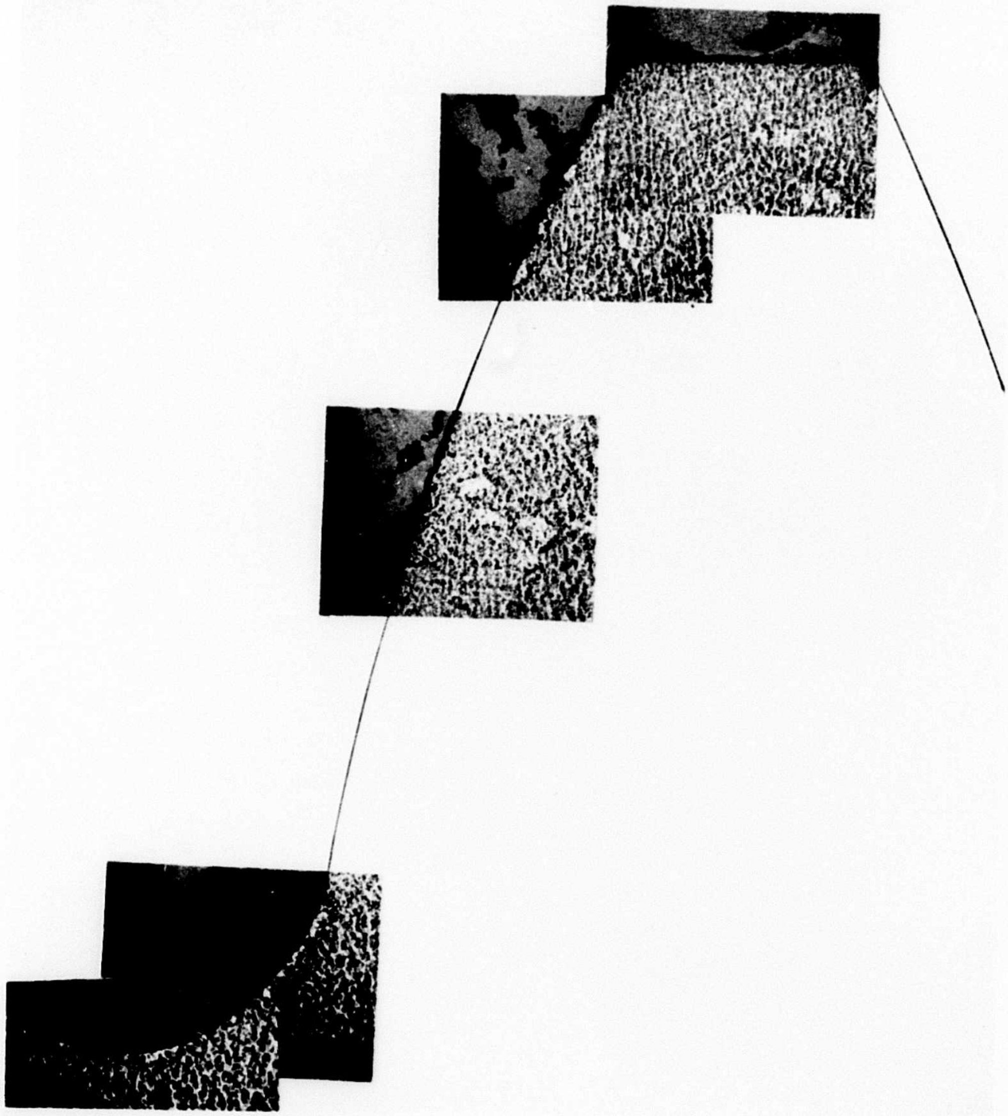
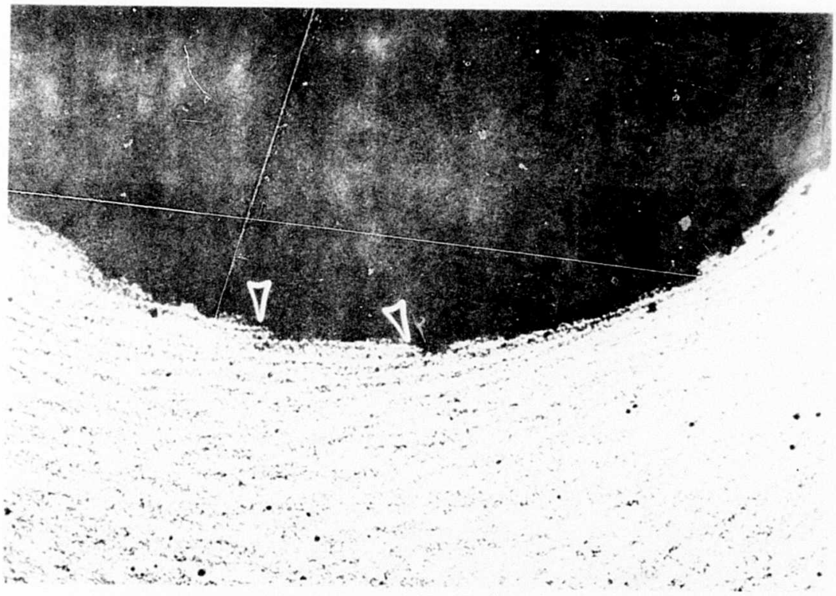
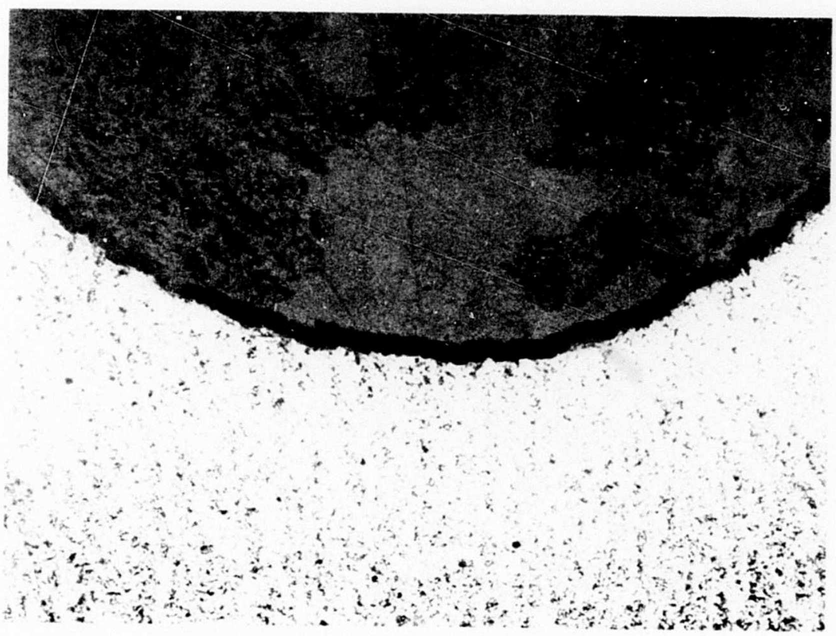


Figure 38. Grain Flow Pattern After Grinding - Roll-Generated Gear.



View A



View B

Figure 39. Grain Flow in Root Fillet.

Figure 40. Typical Prior Austenitic Grain Size.

SUMMARY OF RESULTS

SINGLE-TOOTH TESTS

The comparative results of the single-tooth fatigue tests presented in Table VII and plotted in Figure 22 show that the mean-fatigue-strength difference between the two roll-forming processes at 10^8 cycles is 93 pounds, or approximately 4 percent, favoring the roll-formed process. As shown in Figure 22, the mean fatigue strength at 10^8 cycles for the conventionally forged gears falls between the values obtained for the two roll-forming processes; thus, the fatigue strengths of all three processes tested lie within a relatively narrow band.

The results for the conventionally forged gears tested in this program are significantly higher (28 percent) than the results obtained in the single-tooth program of Reference 1. In both of these programs, the conventionally forged test gears were of the same material (AISI 9310 CVM carburizing steel) and forging design, were manufactured by the same gear manufacturer, and were tested on the same test fixtures. The only differences were that the raw material in each case was from a different heat and it was processed at a different time. This indicates that a variability can exist between heats of material, and this difference can be as much as 28 percent.

Examination of the test data and the results of the post-test inspection revealed that the pitting condition which was in evidence to some degree on 15 of the 36 teeth did not influence the test results significantly. In the majority of cases the pitted gears failed earlier than the clean teeth, although this was not true 100 percent of the time. The divergence between the failure times for pitted teeth and clean teeth at each load level is considered to be within the normal scatter to be expected in a fatigue test of this sort.

The results of the single-tooth static tests shown in Table IX reveal substantial differences in the static strengths of the three processes tested. The conventionally forged gears evidenced ultimate static strengths at least 265 pounds greater than those of roll-formed gears and at least 1300 pounds greater than those of roll-generated gears based on a single-tailed "t" test. Since only four test points on one test gear were required for the conventional process, this result cannot be considered statistically significant. The comparison also reveals the static strength of roll-formed teeth to be 690 pounds greater than that of roll-generated teeth.

DYNAMIC TESTS

The results of the dynamic tests on the two roll-forming processes, presented in Table XI and Figure 25, show that the gears produced by the roll-generated process have a mean dynamic fatigue strength at 10^8 cycles which is 57 pounds, or approximately 4 percent, greater than that obtained for the roll-formed process. On the basis of these results, the two processes exhibited essentially equal fatigue strengths.

The comparison of Figure 25 also shows that, in the high-load low-cycle region, the roll-formed gears are stronger than the roll-generated gears, which is in general agreement with the results of the single-tooth static tests.

The dynamic test results of this program can be correlated with the baseline conventionally forged gears upon the completion of a current USAAMRDL program (Contract DAAJ02-69-C-0060) on forged gears.

Examination of the gear tooth wear patterns and the fracture origins indicate that the gears were end-loaded to some degree. This end-loading was undoubtedly caused by shaft deflection due to the fact that the gears are overhung for ease in loading and unloading and to facilitate inspection. All gears examined, however, showed a consistent wear pattern; therefore, the end-loading condition did not affect the relative strengths of the gears tested, but it does require the use of an appropriate load distribution factor when gear tooth stresses are calculated.

The gear teeth on all of the gears examined exhibited varying degrees of scuffing and surface pitting, which is characteristic of a balanced design in which the tendency to pit or scuff is roughly equal to the tendency of fracture. A coarser-pitch gear would be expected to exhibit predominantly surface failures, whereas a finer-pitch gear would fail almost exclusively by root bending.

STRESS COMPARISON

The average fatigue strengths at 10^8 cycles for all of the processes tested are 2629 pounds for the single-tooth tests and 1495 pounds for the dynamic test. The corresponding average stresses are 123,400 psi for the single-tooth specimen and 98,700 psi for the dynamic gear when a load distribution factor of 1.3 is used to account for end-loading. The ratio of these two stresses is a measure of the dynamic effect at 9200 rpm. Thus,

$$K_v = \frac{98700}{123400}$$

$$K_v = .80$$

where K_v is the experimental dynamic factor. This value agrees with the results of the dynamic tests of Reference 8 within 10 percent.

DISCUSSION

The results obtained in the single-tooth test program show little difference in the fatigue strengths of the two roll-forming processes and the conventional forging process. This result, especially in the case of the roll-formed process, is surprising since the results of previous tests conducted on advanced gear forgings (Reference 1), which had almost identical grain flow structure as the roll-formed gears, showed a substantial improvement (20 to 40 percent) in fatigue strength over the conventional process. The small difference obtained between the two roll-forming processes themselves was also surprising since the fatigue strength of gears formed by any metal-displacing method was expected to increase with the amount of material displaced during forming. This small difference was apparent in both the single-tooth and dynamic test results. The gear processes tested in this program had varying degrees of cold working and metal flow as indicated by the orientation and depth of the grain flow pattern which remained after final grinding. The roll-formed gears had a considerable amount of material displaced, since they were rolled from the solid, and exhibited a good grain flow pattern which followed the contour of the tooth profile. The roll-generated gears had a relatively small amount of metal displaced, since the gears were prehobbed before rolling, and had little or no residual grain flow. The roll-generated teeth were similar to the conventionally forged gears in this respect.

Two possible explanations are offered for the apparent differences between the test results of this program and those of the advanced gear forging program. First, the differences could be purely statistical and the direct result of too small a test sampling to adequately represent the total population of roll-formed gears. If the sample size in each case had been larger, no doubt a truer representation of the relative strengths would have resulted.

Another possibility which could account for the results obtained is that there may exist an optimum degree of cold working and metal displacement which produces maximum benefits in terms of increased fatigue strengths. If this is true, working the metal beyond this point would produce results similar to those processes where the metal was underworked or not worked at all. This suggests that perhaps too much material was displaced during the roll-formed process, thus weakening the gear, and too little during the roll-generated process, producing no increase in strength over the conventional process. More testing at various degrees of cold working is needed to fully validate this theory.

Regarding gear testing in general, it is the author's opinion that a minimum of 20 test points, excluding runouts, is necessary to adequately define the S-N curve relation. It is recommended that to get the most useful data from this limited number of test points, more specimens should be run at loads and stresses close to the endurance limit, where the scatter is greatest, than at the high-load low-cycle region, where the test points are more closely spaced.

CONCLUSIONS

1. On the basis of both single-tooth and dynamic fatigue tests conducted on a limited number of test specimens, the roll-formed gears exhibited fatigue strengths at 10^8 cycles which are essentially equal to the fatigue strength of gears produced by conventional means.

2. The results obtained for the roll-formed process are different from what was expected based on the results obtained for the advanced gear forgings (which had similar grain flow structure) tested in a previous program (Reference 1). It is believed that this result is due, in part, to the small size of the test sample, which may not be representative of the total population.

3. A possibility for the different results, which should be explored further, is that there is a degree of cold working which could produce optimum results. This suggests that perhaps too much metal was displaced during the roll-forming process and too little during the roll-generating process, thus producing results which were similar to the conventional forging process.

4. The average fatigue strength of the test gears when run under dynamic conditions at 9200rpm is 80 percent of the average fatigue strength of the gears tested in the nonrotating or single-tooth test phase.

5. The results of the single-tooth static tests revealed the static strength of roll-formed teeth to be 690 pounds greater than that of roll-generated teeth.

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