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ABSTRACT

Life tests were conducted at three different loads with three groups of 8.9 cm (3.5 in.) pitch diameter spur gears made of vacuum arc remelted VAR AISI 9310 steel. Life was found to vary inversely with load to the 4.3 and 5.1 power at the L_{10} and L_{50} life levels, respectively. The Weibull slope varied linearly with maximum Hertz contact stress, having an average value of 2.5. The test data when compared to AGMA standards showed a steeper slope for the load-life diagram.

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SUMMARY

Experiments were conducted to determine the influence of load on life for vacuum arc remelted (VAR) AISI 9310 steel spur gears. The results were used to modify the NASA life prediction method for surface endurance of gears. The test gears had a 20° involute profile and a 8.89-centimeter (3.5-in.) pitch diameter. Three groups of 19 gears were tested at transmitted tangential loads of 463×10^3, 578×10^3, and 694×10^3 N/M (2645, 3305, and 3966 lb/in.) which produced maximum Hertz stresses at 1531×10^6, 1710×10^6, and 1875×10^6 N/M^2 (222 000, 248 000, and 272 000 psi). The gears were run at 10 000 rpm and a temperature of 350 K (170° F). The lubricant was a superrefined naphthenic mineral oil with an additive package.

Life was found to be inversely proportional to load to the 4.30 and 5.1 power at the L_{10} and L_{50} life levels, respectively. The dispersion of the fatigue data as measured by the Weibull slope increased with load, varying linearly with contact stress. The average value for the Weibull slope was 2.5. The load-life relation developed has a steeper slope than the load-life relation given by the American Gear Manufacturers Association standards.

STAR Category 37

SYMBOLS

c	orthogonal shear stress exponent
e	Weibull's exponent
f	face width of tooth in contact, m (in.)
h	depth to critical stress exponent
K_2	constant of proportionality
L	pitting fatigue life, millions of revolutions
l	involute profile arc length, m (in.)
l_c	length of contact line, m (in.)
N	number of teeth
p	load-life exponent
S	probability of survival
S_{max}	maximum Hertz stress
W_t	transmitted tangential load, N (lb)
W_{tM}	dynamic capacity of gear-pinion mesh, N (lb)
$\sum \rho$	curvature sum, m^{-1} (in. $^{-1}$)
φ_t	transverse pressure angle, rad
ψ_b	base helix angle, rad

Subscripts:

1	driving member
2	driven member

INTRODUCTION

High quality aircraft gears may fail in several different ways. The most common modes of failure are scoring, tooth breakage, and surface fatigue. If the gears are properly lubricated and if the bending stresses are within reasonable limits, then scoring and tooth breakage will not be a problem. However, surface fatigue failure cannot be eliminated. It is a natural event that is brought about by cumulative damage to the gear material caused by repeated applications of contact stress.

Recently an equation for predicting surface fatigue life of gears was developed [1 to 3]. The theory was based on a modification of the Lundberg-Palmgren theory [4 to 6] which has been widely used for predicting rolling element bearing life. The new life equation includes the effect on life due to stress, stressed volume, and depth beneath the surface at which the maximum critical shearing stresses occur. In [3] the life equation was used to determine the 10-percent life for a group of AISI 9310 test gears. The load life exponent used in [3] was 1.5. In the discussion to [3], it was suggested that the exponent 1.5 was too small.

In hardened steel gears it may be reasonably expected that the inverse load-life exponent would be close to that for roller bearings. The industry standard for roller bearings is 10/3 and for ball bearings it is 3 [7]. In the work of [8] the life of automotive transmission gears was found to be proportional to the inverse 3.38 power of load. Others have shown similar results [9]. In AGMA standards [10] a load-life exponent of approximately 8.5 to 9.5 is used. The correctness of any life prediction method for gears, as well as bearings, is dependent upon the accuracy of the relationship between load and life. It is apparent that there is a large variance in the load-life exponent used by different investigators. It, therefore, becomes an objective of the research reported herein to (a) experimentally determine the load-life relationship for AISI 9310 steel gears, and (b) to amend the gear life prediction formula of [3].

In order to accomplish the aforesaid, three groups of AISI 9310 steel gears were tested at 463×10^3, 578×10^3, and 694×10^3 N (2645, 3305, 3966 lb/in.) which produced maximum Hertz stresses of 153×10^6 N/M^2, 1710×10^6 N/cm^2, and 187 500 N/cm^2 (222 000 psi, 248 000 psi, and 272 000 psi), respectively. The gears were manufactured from a single lot of vacuum-arc-remelted AISI 9310 steel (AMS 6265). Pitch diameter was 8.89 cm (3.5 in.), test temperature was 350 K (170° F), and speed was 10 000 rpm. All tests were run using a superrefined napthenic mineral oil from one lubricant batch having a proprietary additive package plus a 5-percent extreme pressure additive.

BACKGROUND

Under constant service conditions, the probability of survival for a gear mesh is given by the Weibull relation [1]

$$S = 0.9 \exp\left\{\left(\frac{L}{L_{10}}\right)^{e}\right\} \tag{1}$$

The 10 percent life a given gear mesh in millions of rotations of the input gear is given by the following equation [1]

$$L_{10} = \left(\frac{W_{tM}}{W_t}\right)^{p} \tag{2}$$

where W_t is the transmitted tangential load and W_{tM} is the gear mesh dynamic capacity.

The gear mesh dynamic capacity W_{tM} is defined as the load which may be carried for one million revolutions and 90 percent probability of survival. The dynamic capacity of the gear is given by the following equation [2] which applies to helical gears and may be used for spur gears by setting $\psi_b = 0$ and $l_c = f$.

$$W_{tM} = K_2 l_c \cos \varphi_t (\cos \psi_b)^{\frac{h-c-3}{h-c-1}} \left(\sum \rho\right)^{\frac{h+c-1}{h-c-1}} \left[f l N_1 \left\{1 + \left(\frac{N_1}{N_2}\right)^e\right\}\right]^{\frac{2}{h-c-1}} \tag{3}$$

The Weibull slope e and the load-life exponent p may be directly determined by conducting life tests under several load conditions for a given group of gears. But to determine the exponents c and h the gear size must be varied. This is more difficult to do since gear testers normally will accept only a single size of gear. In addition, it is too expensive to conduct large numbers of fatigue tests with very large gears.

In [3] the values of K_2, c and h used were identical to those for rolling-element bearings in [4]. The value of e was taken as 3 based on NASA gear tests conducted at a maximum Hertz stress of 1896×10^6 N/m^2 (275 000 psi). Using these values in the relation

$$p = \frac{c - h + 1}{2e} \tag{4}$$

from Lundberg-Palmgren theory [4], gave an inverse load-life exponent of 1.5. This value of p is about 1/2 of that for rolling-element bearings [4 - 7] and up to 6 times smaller than reported elsewhere for gears [8 - 10].

APPARATUS, SPECIMENS, AND PROCEDURE

Gear Test Apparatus

The gear fatigue tests were performed in the NASA Lewis Research Center's gear-fatigue test apparatus (fig. 1(a)). This gear testing machine uses the four-square principle of applying the test-gear load so that the input drive need only overcome the frictional losses in the system.

A schematic of the gear testing machine is shown in figure 1(b). Oil pressure and leakage flow are supplied to the load vanes through a shaft seal. As the oil pressure is increased on the load vanes inside the slave gear, torque is applied to the shaft. This torque is transmitted through the test gears back to the slave gear, where an equal but opposite torque is maintained by the oil pressure. This torque on the test gears, which depends on the hydraulic pressure applied to the load vanes, loads the gear teeth to the desired stress level. The two identical test gears can be started under no load; and the load can be applied gradually, without changing the running track on the gear teeth. By testing both faces of the gears, a total of four fatigue tests can be run for each set of gears.

Separate lubrication systems are provided for the test gears and the main gearbox. The two lubrication systems are separated at the gearbox shafts by pressurized labyrinth seals, with nitrogen as the seal gas. The test-gear lubricant is filtered through a 5-micron nominal fiberglass filter. The test lubricant can be heated electrically with an immersion heater. The skin temperature of the heater is closely controlled to prevent a "hot spot" condition that would overheat the test lubricant.

A vibration transducer mounted on the gearbox is used to automatically shut off the test rig when gear-surface fatigue occurs. The gearbox is also automatically shut off if there is a loss of oil flow to either the main gearbox or the test gears, if the test gear oil overheats, or if there is a loss of seal gas pressurization.

The test rig is belt driven and can be operated at several fixed speeds by changing pulleys. The operating speed for the tests reported herein was 10 000 rpm which gave a pitch-line velocity of 46.55 meters per second (9163 ft/min).

Test Gears and Lubricant

The AISI 9310 gears were manufactured from a single lot of vacuum arc remelted (VAR) AISI 9310. The chemical composition of the AISI 9310 gears is given in table I. The heat treatment for the AISI 9310 gears is given in table II. A photomicrograph of etched and polished surface of the AISI 9310 gear is shown in figure 2. The case hardness of the gears was Rockwell C 62 to 64. The core hardness was Rockwell C 35 to 40.

Dimensions for the test gears are given in table III. All gears have a nominal surface finish on the tooth face of 0.406 micrometer (16 μin.) rms and a standard 20° involute profile with tip relief. Tip relief was 0.0013 centimeter (0.005 in.) starting at the last 30 percent of the active profile. The gears were also crowned to prevent excessive edge loading.

All tests were conducted with a single batch of superrefined naphthenic mineral oil lubricant having proprietary additives (antiwear, antioxidant, and antifoam). The physical properties of this lubricant are summarized in table IV. Five percent of an extreme pressure additive, designated Anglamol 81 (partial chemical analysis given in table V), was added to the lubricant. A nitrogen cover gas was used throughout the test as a baseline condition which allowed testing at the same conditions at much higher temperatures without oil degradation. By excluding oxygen the cover gas also reduced the effect of the oil additives on the gear surface boundary lubrication by reducing the chemical reactivity of the additive-metal system (ref. 11).

Test Procedure and Conditions

The test gears were cleaned to remove the preservative and then assembled on the test rig. All test specimens were run-in for one hour at a load of 7584×10^6 N/M^2 (110 000 psi) maximum Hertz stress. The load was then increased to the test condition which for the three loads used resulted in maximum Hertz stresses of 1531×10^6, 1710×10^6, and 1875×10^6 N/M^2 (222 000, 248 000, and 272 000 psi). Table VI summarizes the load-stress conditions.

The tests were continued 24 hours a day until they were shut down automatically by the vibration-detection transducer located on the gearbox, adjacent to the test gears.

The lubricant flow rate was held constant at 800 cubic centimeters per minute, and lubrication was supplied to the inlet mesh of the gear set by jet lubrication. The lubricant was circulated through a 5-micron fiber glass filter to remove wear particles. A total of 3800 cubic centimeters (1 gal) of lubricant was used and was discarded, along with the filter element, after each test. Inlet and outlet oil temperatures were continuously recorded on a strip-chart recorder. The lubricant inlet temperature was constant at 319±6 K ($115^{o}\pm10^{o}$ F), and the lubricant outlet temperature was constant at 350±3 K ($170^{o}\pm5^{o}$ F). The outlet temperature was measured at the outlet of the test-gear cover.

The test gears were run in an offset condition with a 0.030-centimeter (0.120-in.) tooth-surface overlap to give a load surface on the gear face of 0.28 centimeter (0.110 in.) of the 0.635-centimeter (0.250-in.) wide gear, thereby allowing for edge radius of the gear teeth. This offset loading causes a slight twisting in the gear tooth. However, the mating tooth twists in the opposite direction approximately the same amount which, along with the crown radius, prevents edge loading.

The test gear shaft deflection resulting from the overhung load gives a tooth mismatch of 1.5×10^{-4} centimeter (6×10^{-5} in.) across the 0.28-centimeter (0.11-in.) contact face width. This amounts to approximately 10 percent of the Hertz deflection of 1.3×10^{-3} centimeter (5×10^{-4} in.). This could cause some edge loading effects. However, the crown radius in the tooth face prevents edge loading.

At the pitch-line load, the tooth bending stress was 2.48×10^{8} newtons per square meter (30 000 psi) at the intermediate load condition if plain bending is assumed. However, because there is an offset load there is an additional stress superimposed on the tooth bending stress. Combining the bending and torsional moments gives a maximum stress of 2.67×10^{8} newtons per square meter (38 700 psi). For this research, the dynamic load factor was assumed equal to unity.

The pitch-line elastohydrodynamic (EHD) film thickness was calculated by the method of Grubin [12]. It was assumed, for this film thickness calculation, that the gear temperature at the pitch line was equal to the outlet oil temperature and that the inlet oil temperature to the contact zone was equal to the gear temperature, even though the oil inlet temperature was considerably lower. It is probable that the gear surface temperature could be even higher than the oil outlet temperature, especially at the end points of sliding contact. The EHD film thickness for these conditions was computed to be 0.65 micrometer (26 μin.), which gave a ratio of film thickness to composite surface roughness (h/σ) of 1.13. According to [13] the life adjustment factor for surface fatigue would be approximately one-half.

RESULTS AND DISCUSSION

Three groups of vacuum arc remelted (VAR) AISI 9310 case carburized and hardened steel gears were fatigue tested under loads of 463×10^3, 578×10^3, and 694×10^3 N/m (2645, 3305, and 3966 lb/in.) which produced maximum Hertz stresses of 1531×10^6, 1710×10^6, and 1875×10^6 N/cm^2 (222 000, 248 000, and 272 000 lb/in.^{2}). The lubricant was a super-refined naphthenic mineral oil with a 5 percent extreme-pressure additive package. A representative fatigue spall is shown in figure 3.

The results of the fatigue tests are presented as Weibull plots in figure 4 using the method of [14]. A summary of the test results is presented in table VI. In each test group there were 19 failures out of 19 tests. All failures were due to subsurface originated fatigue and occurred in the zone of single tooth loading at or just below the pitch diameter.

The 90-percent confidence interval limits were determined for each group of test data. The confidence limits are shown in figure 4 and summarized for the L_{10} and L_{50} life loads in table VI. The interpretation of these limits is that the true life at each condition will fall between these limits 90 percent of the time. Where these confidence limits overlap, the life differences are not considered statistically significant. For an example, in table VI the lower confidence limit at the 1531×10^6 N/M^2 (222 000 psi) stress overlaps the upper confidence limit of the 1710×10^6 (248 000 psi) stress at the L_{10} life level. However, the limits do not overlap at the L_{50} life loads nor at the nonadjacent stress levels. This observation coupled with the consistent trend of decreasing life with increasing stress indicates good statistical significance in the data.

The slopes e of the Weibull plots of figure 4 which are summarized in table VI, are a measure of the dispersion of the fatigue data. The Weibull slopes increased with increased stress. That is, there was less dispersion of the data as the contact stress was increased. The average value of the slope for the plots of figure 4 is 2.5.

The L_{10} and L_{50} lives for each set of data of figure 4 are plotted against load in figure 5. For convenience, the 90-percent confidence bands are shown on each data point. From the slope of the least squares fit, the life was inversely proportional to load to the 4.3 and 5.1 power at the L_{10} and L_{50} life levels, respectively. Using standard methods of variance analysis [14], the 90-percent confidence interval limits for the load-life exponents were calculated as (3.5 to 5.1) at the L_{10} life level and (4.8 to 5.5) at the L_{50} life level. The values of K_2, p, and e in equations (1) to (3)) are determined from the experimental results reported herein. The value of K_2 was determined as 8.73×10^{7} when SI units (newtons and meters) are used in equation (3) and 21 800 when using English units (pounds, inches). If the effect of the life correction factor due to thin lubricant film is to be treated separately then the corresponding value of K_2 is 1.03×10^{8} (25 700). The stressed volume for the zone of single tooth contact was used in determining K_2. As a result (and using the values of c and h from rolling element bearing experience), equations (1) to (3) may be written as follows

$$S = 0.9 \exp \left\{ \left(\frac{L}{L_{10}} \right)^{2.5} \right\} \tag{4}$$

$$L_{10} = \left(\frac{W_{tM}}{W_t} \right)^{4.3} \tag{5}$$

$$W_{tM} = K_2 l_c \cos \varphi_t (\cos \psi_b)^{11/9} \left(\sum \rho\right)^{-35/27} \left[f l N_1 \left\{ 1 + \left(\frac{N_1}{N_2} \right)^{2.5} \right\} \right]^{-2/9} \tag{6}$$

The life distributions calculated by equations (4) to (6) are plotted for comparison with the experimental data in figure 6.

Comparison with AGMA Standards

The American Gear Manufacturers Association (AGMA) has published two standards for tooth surface fatigue [10, 16]. These standards are AGMA 210.02 and AGMA 411.02. AGMA 210.02 provides for an endurance limit for surface fatigue below which it is implied that no failure should occur. In practice, there is a finite surface fatigue life at all loads. AGMA 411.02 recognizes this finite life condition. Therefore, it does not contain an endurance limit in the load-life curve but does show a continuous decrease in life with increasing load. Both AGMA standards are illustrated in figure 7. The AGMA load-life curves shown are for a 99 percent probability of survival or the L_1 life [15]. The experimental L_1, L_{10}, and L_{50} lives are plotted for comparison.

It is evident that the load-life relation used by AGMA is different than the experimental results reported herein. The difference between the AGMA life prediction and the experimental lives could be the result of differences in stressed volume. The AGMA standard does not consider the effects of stressed volume which may be considerably different than that of the test gears used herein. The larger the volume of material stressed the greater the probability of failure or the lower the life of a particular gear set. Therefore, changing the size or contact radius of a gear set, even though the same contact stress is maintained would have an effect on gear life.

SUMMARY OF RESULTS

Experiments were conducted to determine the influence of load on life for vacuum arc remelted (VAR) AISI 9310 steel spur gears. The results were used to amend the NASA gear life prediction equation for surface endurance of gears. The test gears had a 20° involute profile and a 8.89-centimeter (3.5-in.) pitch diameter. Three groups of 19 gears were tested at transmitted tangential loads of 463×10^{3}, 578×10^{3}, and 694×10^{3} (2645, 3305, and 3966 lb/in.) which produced maximum Hertz stress of 1531×10^{6}, 1710×10^{6}, and 1875×10^{6} N/M^{2} (222 000, 248 000, and 272 000 psi). The gears were run at 10 000 rpm and a temperature of 350 K (170° F). The lubricant was a superrefined naphthenic mineral oil with an additive package. The following results were obtained.

1. Life was inversely proportional to load to the 4.3 and 5.1 power at the L_{10} and L_{50} life levels, respectively.

2. The dispersion of the fatigue data as measured by the Weibull slope was increased with load, varying linearly with contact stress. The average value for the Weibull slope was 2.5.

3. The load-life relation developed has a higher slope than the load-life relation given by AGMA, that is, the life increases less with decreasing load than the AGMA standard would predict.

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TABLE I. - CHEMICAL COMPOSITION OF VAR AISI 9310

GEAR MATERIALS BY PERCENT WEIGHT

Element	C	Mn	Si	Ni	Cr	Mo	Cu	P and S
Weight percent	0.10	0.63	0.27	3.22	1.21	0.12	0.13	0.005

TABLE II. - HEAT TREATMENT PROCESS FOR

VACUUM ARC REMELTED (VAR) AISI 9310

Step	Process	Temperature		Time, hr
		K	^{o}F	
1	Carburize	1172	1650	8
2	Air cool to room temperature	----	----	---
3	Copper plate all over	----	----	---
4	Reheat	922	1200	2.5
5	Air cool to room temperature	----	----	---
6	Austenitize	1117	1550	2.5
7	Oil quench	----	----	---
8	Subzero cool	189	-120	3.5
9	Double temper	450	350	2 each
10	Finish grind	----	----	---
11	Stress relieve	450	350	2

TABLE III. - SPUR GEAR DATA

[Gear tolerance per ASMA class 12.]

Number of teeth	28
Diametral pitch	8
Circular pitch, cm (in.)	0.9975 (0.3927)
Whole depth, cm (in.)	0.762 (0.300)
Addendum, cm (in.)	0.318 (0.125)
Chordal tooth thickness reference, cm (in.)	0.485 (0.191)
Pressure angle, deg	20
Pitch diameter, cm (in.)	8.890 (3.500)
Outside diameter, cm (in.)	9.525 (3.750)
Root fillet, cm (in.)	0.102 to 0.152 (0.04 to 0.06)
Measurement over pins, cm (in.)	9.603 to 9.630 (3.7807 to 3.7915)
Pin diameter, cm (in.)	0.549 (0.216)
Backlash reference, cm (in.)	0.0254 (0.010)
Tip relief, cm (in.)	0.001 to 0.0015 (0.0004 to 0.0006)

TABLE IV. - PROPERTIES OF LUBRICANT ADDITIVE ANGLAMOL 81

Phosphorous content, percent by weight	0.66
Sulfur content, percent by weight	13.41
Specific gravity	0.982
Kinematic viscosity at 372 K (210° F), cm^2/sec (cS)	29.5×10^{-2} (29.5)

TABLE V. - PROPERTIES OF SUPERREFINED NAPHTHENIC

MINERAL OIL USED AS TEST LUBRICANT

Kinematic viscosity, cm^2/sec (cS), at-	
266 K (20^{o} F)	2812×10^{-2} (2812)
311 K (100^{o} F)	73×10^{-2} (73)
372 K (210^{o} F)	7.7×10^{-2} (7.7)
478 K (400^{o} F)	1.6×10^{-2} (1.6)
Flashpoint, K (^{o}F)	489 (420)
Autoignition temperature, K (^{o}F)	664 (735)
Pour point, K (^{o}F)	236 (-35)
Density at 289 K (60^{o} F), g/cm^3	0.8899
Vapor pressure at 311 K (100^{o} F), mm Hg (or torr)	0.01
Thermal conductivity at 311 K (100^{o} F), J/(m)(sec)(K) (Btu/(hr)(ft)(^{o}F))	0.04 (0.0725)
Specific heat at 311 K (100^{o} F), J/(kg)(K) (Btu/(lb)(^{o}F))	581 (0.450)

TABLE VI. - SUMMARY OF FATIGUE LIFE RESULTS WITH 8.89-CM (3.5-IN.)

PITCH DIAMETER SPUR GEARS AT THREE LOADS

[Material, VAR AISI 9310 steel; speed, 10 000 rpm; lubricant, superrefined naphthenic mineral oil with additive package.]

Transmitted tangential load, wt (lb/in.)	Maximum Hertz stress, N/M^2 ($lb/in.^2$)	Gear set life, millions of revolutions (hrs)						Weibull slope	Failure index
		L_{10}			L_{50}				
		Lower 90-percent confidence limit	Experimental	Upper 90-percent confidence limit	Lower 90-percent confidence limit	Experimental	Upper 90-percent confidence limit		
463×10^3 (2645)	1531×10^6 (222×10^3)	12.4 (20.7)	23.6 (39.3)	44.8 (74.6)	47.9 (79.9)	63.8 (106.4)	85 (141.6)	1.9	19 out of 19
578×10^3 (3305)	1710×10^6 (248×10^3)	7.1 (11.8)	11.4 (19)	18.2 (30.4)	19.2 (32)	23.8 (39.6)	29.3 (48.8)	2.6	19 out of 19
694×10^3 (3966)	1875×10^6 (272×10^3)	2.8 (4.6)	4.3 (7.1)	6.4 (10.7)	6.7 (11.2)	8.1 (13.5)	9.8 (16.3)	2.9	19 out of 19

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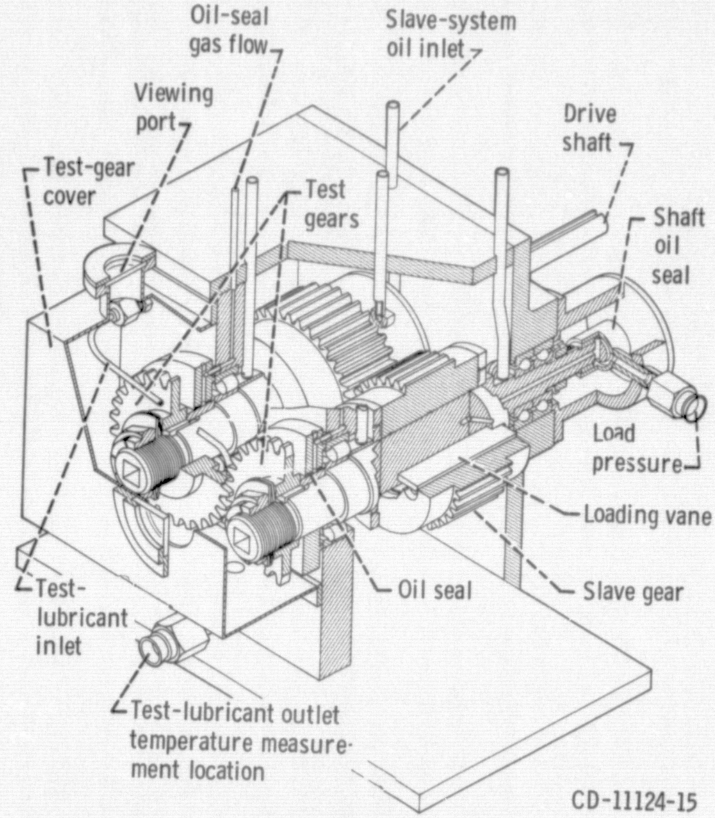
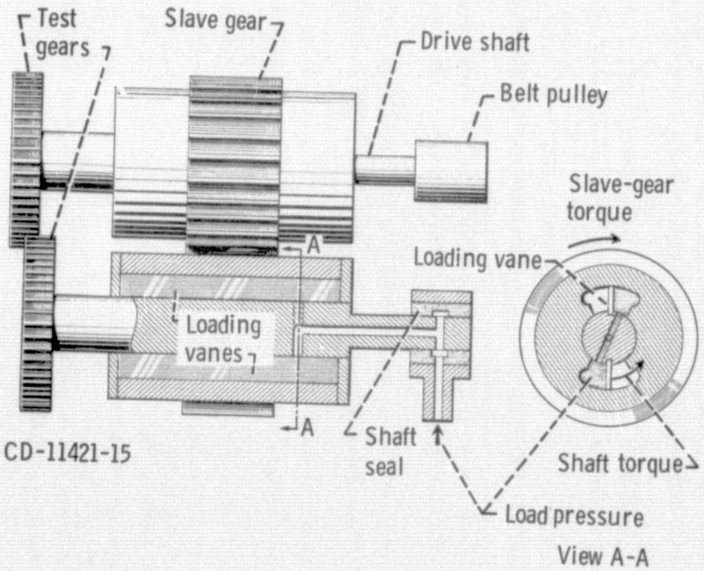
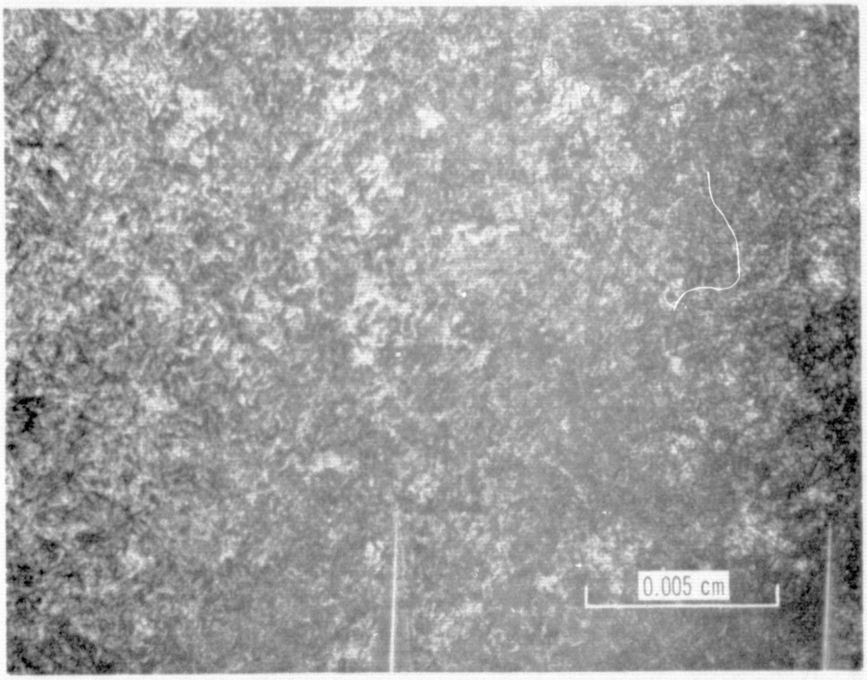


(a) Cutaway view.

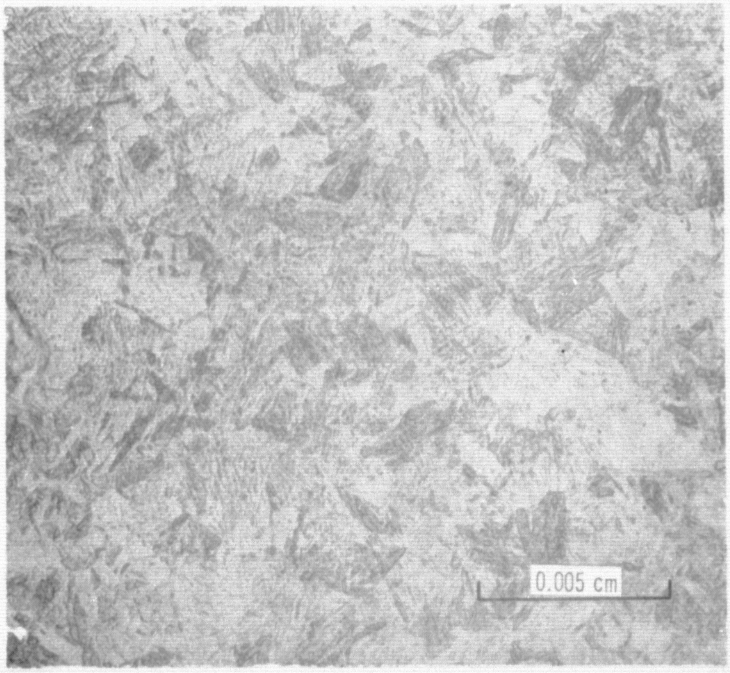


(b) Schematic diagram.

Figure 1. - NASA Lewis Research Center's gear fatigue test apparatus.



(a) Carburized and hardened case of the VAR AISI 9310 gear showing high carbon fine grain martensitic structure.



(b) Core structure of VAR AISI 9310 gear showing low carbon refined austenitic grain size.

Figure 2. - Photomicrographs of case and core regions of test gears.

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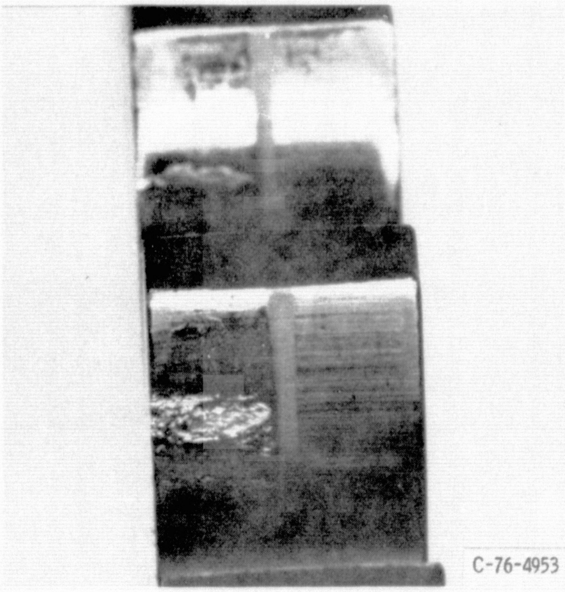


Figure 3. - Representative fatigue spall of test gear material VAR AISI 9310 steel. Speed, 10 000 rpm; lubricant, superrefined naphthenic mineral oil with additive package.

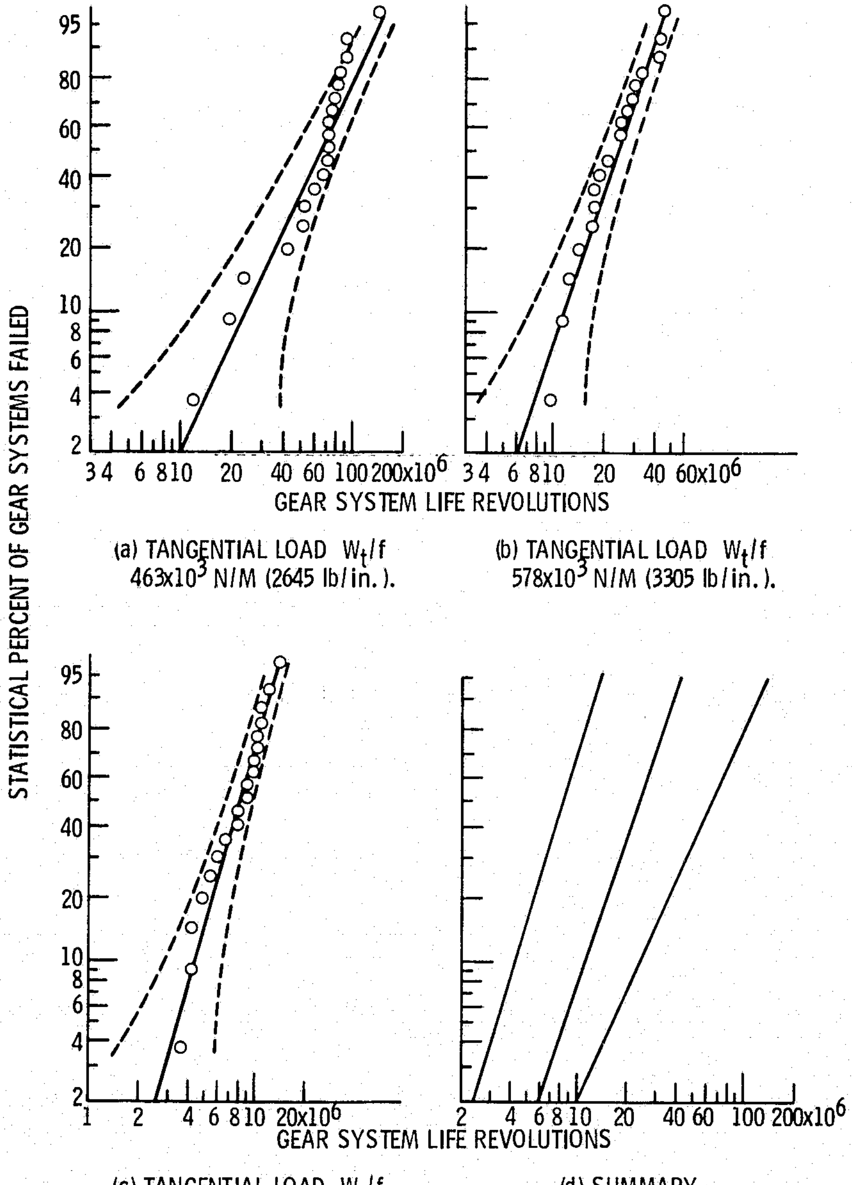


Figure 4. - Effect of load on the life of 8.39 cm (3.5 in.) pitch diameter spur gears, material, VAR AISI 9310 steel, speed 10 000 rpm lubricant superrefined naphthenic mineral oil with additive package.

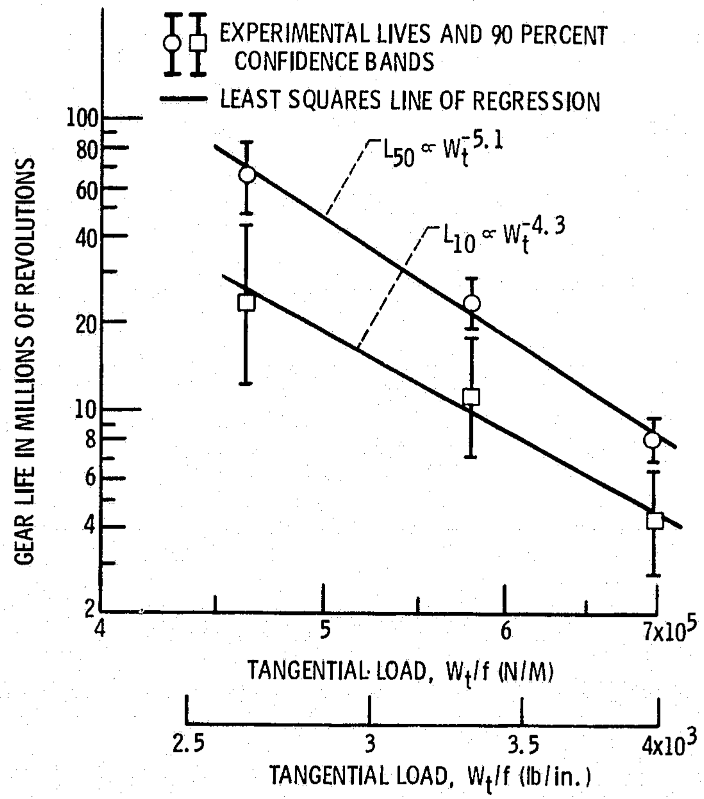
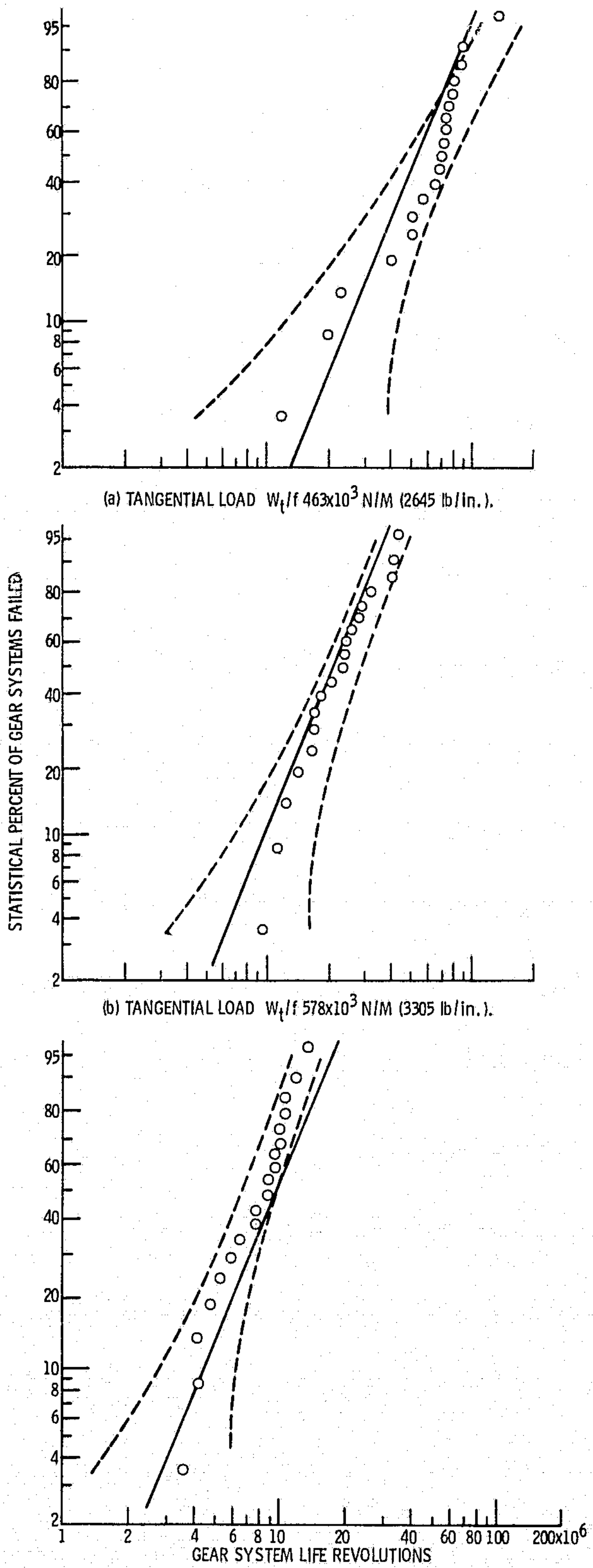


Figure 5. - Load life relationship for VAR AISI 9310 steel spur gears, speed 10 000 rpm, lubricant superrefined naphthenic mineral oil with additive package.



(a) TANGENTIAL LOAD W_t/f 463x10^3 N/M (2645 lb/in.).

(b) TANGENTIAL LOAD W_t/f 578x10^3 N/M (3305 lb/in.).

(c) TANGENTIAL LOAD W_t/f 694x10^3 N/M (3966 lb/in.).

Figure 6. - Comparison of revised life prediction theory to experimental results for VAR AISI 9310 steel spur gears, speed 10 000 rpm, lubricant superrefined naphthenic mineral oil with additive package.

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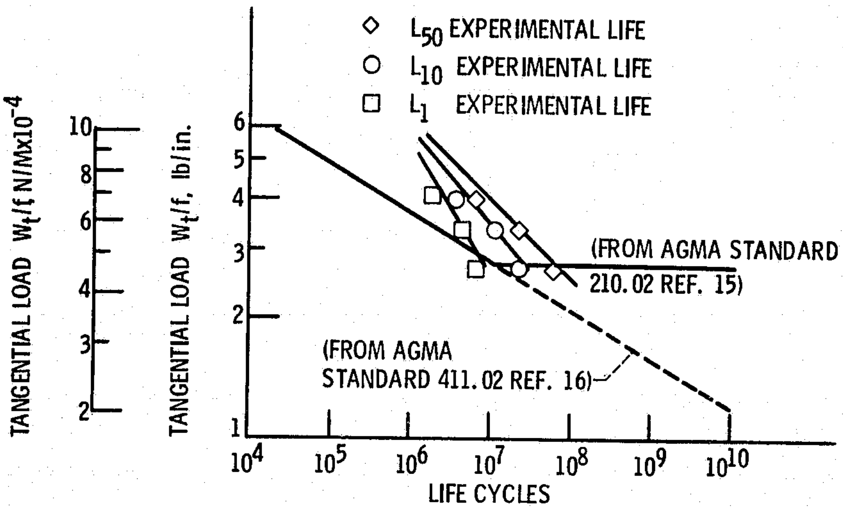

Figure 7. - Comparison of experimental life for VAR AISI 9310 spur gears with AGMA life prediction. Speed 10 000 rpm, lubricant superrefined naphthenic mineral oil with additive package.