Evaluation of High-Contact-Ratio Spur Gears With Profile Modification*

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A majority of current aircraft and helicopter transmissions have a spur-gear-contact ratio (average number of teeth in contact) of less than 2. The contact ratios are usually from 1.3 to 1.8, so the number of teeth in engagement is either one or two. Many gear designs use a pressure angle of 25° for improved tooth strength, giving a contact ratio of approximately 1.3. This low contact ratio causes increased dynamic loading of the gear teeth, increased noise, and sometimes, lower pitting fatigue life.

High-contact-ratio gears (greater than 2) have load sharing between two or three teeth during engagement and, therefore, usually have less load per tooth. These gears should operate with lower dynamic loads and thus less noise. High-contact-ratio gears have been in existence for many years but have not been widely used. High contact ratios can be obtained in several ways: (1) by smaller teeth (larger pitch), (2) by smaller pressure angle, and (3) by increased addendum. As a result, high-contact-ratio gears tend to have lower bending strength and increased tooth sliding. Because of the increased sliding, the high-contact-ratio gears may run hotter and have a greater tendency for surface-distress-related failures such as micropitting and scoring.

Profile modification (changing the involute profile at the addendum or dedendum or both) is normally done on all gears to reduce tip loading and scoring (ref. 1). If it is done improperly, however, it could increase the dynamic load (ref. 2). Several profile modifications have been proposed that would reduce scoring and improve the performance of high-contact-ratio (HCR) gears. One such proposal is the so-called new-tooth-form (NTF) gear, which has a large profile modification at both the addendum and dedendum. The profile radius of curvature is also reduced at the addendum and increased at the dedendum in an attempt to lessen sliding and thereby reduce scoring of HCR gears. However, a gear geometry analysis (ref. 3) indicates that sliding is independent of the profile radius of curvature.

Under NASA contract NAS3-18532 the Boeing Vertol Co. designed and manufactured two sets of NTF gears as well as two sets of standard gears for the purpose of evaluating the NTF gears and comparing them with standard gears.

The work reported herein was conducted to evaluate the NTF gears in relation to the standard gears by conducting three series of tests on both gear types. These tests were scoring tests, surface fatigue tests, and bending fatigue tests. The tests were performed by the Southwest Research Institute under NASA contract NAS3-20026.

Apparatus, Specimens, and Procedure

Gear Test Apparatus

The gear fatigue tests and the scoring tests were performed in two Wright Air Development Division (WADD) gear test rigs. The WADD gear test rig (fig. 1) was developed for high-temperature lubricant testing (ref. 4). It is very similar to the Ryder test rig (ref. 5), except that a roller bearing is used in place of sleeve bearings, and screw thread nonrubbing seals are used in place of shaft-rubbing lip seals. Load is supplied to the test gears through hydraulic pressure applied to the end of the driver gear shaft, forcing the slave gear to move a slight distance axially on the helix. This axial motion produces a torque on the test gears. The test rig is calibrated by strain gages on the test-gear shaft to determine test-gear torque as a function of hydraulic pressure.

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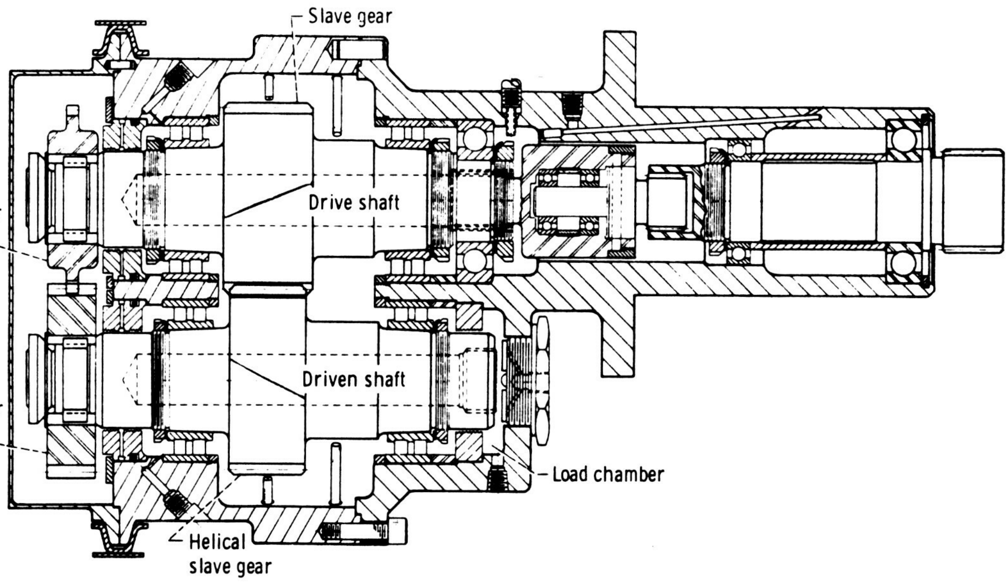


Figure 1. - Wright Air Development Division (WADD) gear test rig.

A hydraulic system provides regulated hydraulic pressure to the load piston end of the slave gear shaft. The system controls the gear torque to a sensitivity of approximately ± 1.13 N-m (10 in-lb) of the applied torque once the load has been set by the operator.

The two test spur gears are the same in all respects except that one gear is wider than the other. The wider gear allows full-face contact with the narrow gear during the slight relative axial movement between them as the tooth load is increased. The narrow gear has full and unvaried face contact and is the gear used for rating.

Each test rig is provided with an independent oil system. The oil used is a synthetic polyol ester C_5, C_6, C_7 substituted pentaerythritol oil. The physical and chemical properties of this oil are given in table I. A 3-μm-absolute oil filter is used to filter the oil during testing. Instrumentation provides for automatic detection of a lubrication or gear failure and for test rig shutdown.

Tooth Bending Fatigue Test Apparatus

The tooth bending fatigue tests were conducted using a special single-tooth test fixture, shown schematically in figure 2, that was designed to be used with a high-cycle-fatigue testing machine. The apparatus consists of the gear-supporting reaction frame, the loading clevis-load cell, the actuator, and the control system. A strain-gage bridge installed on the properly sized elastic elements of the clevis provides an output proportional to the applied load. This allows for real-time load control of the test. Instrumentation includes automatic shutdown of a test in the event of specimen fracture or after a preset number of load cycles are applied, whichever occurs first. The operating frequency of the load system is 1500 cycles per minute.

Test Gears

The test gears were of either standard or new-tooth-form design. Three different gear sets were manufactured for each design: the bending test gears, the wide scoring and surface fatigue test gears, and the narrow scoring and surface fatigue test gears. The dimensions for the standard and NTF WADD test gears are given in table II. The narrow WADD gear of each design is shown in figure 3 for comparison. The contact ratio for the standard gear, not considering tip relief, was 1.30. All gears had a nominal surface finish of 0.406 μm (16 μin.). The standard gear had a pressure angle of 22.50° and a tip relief of 0.001 cm (0.0004 in.). The NTF gear had a contact ratio of 2.31, an increased addendum of 0.16 cm (0.063 in.), a pressure angle of 19°, a tip relief of 0.006 cm (0.0025 in.), and a flank addition of 0.006 cm (0.0025 in.). This is shown in the profile trace of figure 4(a). The involute radii at the tip and flank of the NTF gears were also modified to decrease the tip radius and to increase the flank radius.

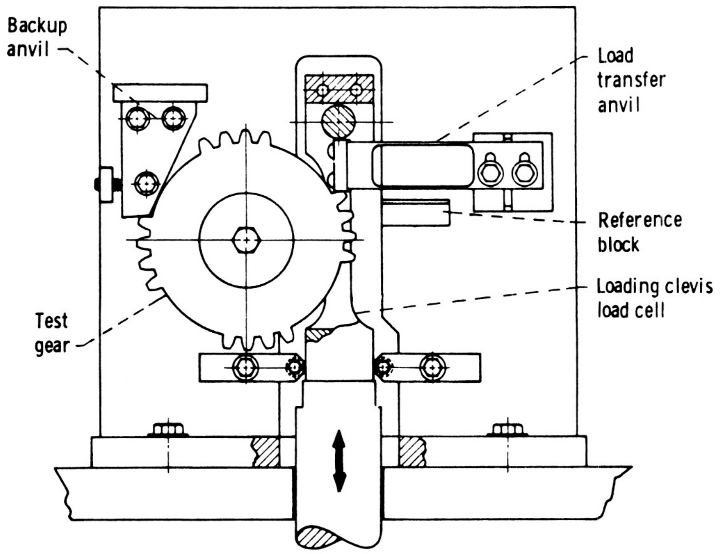
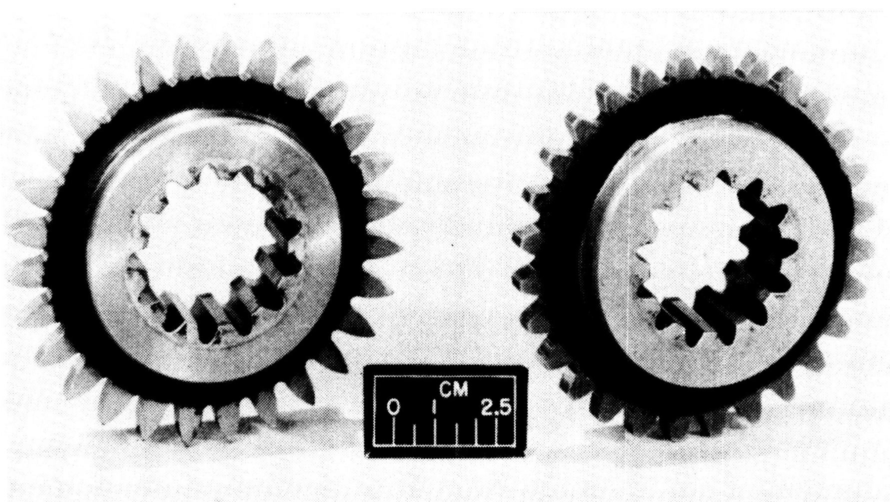


Figure 2. - Single-tooth bending fatigue test fixture.

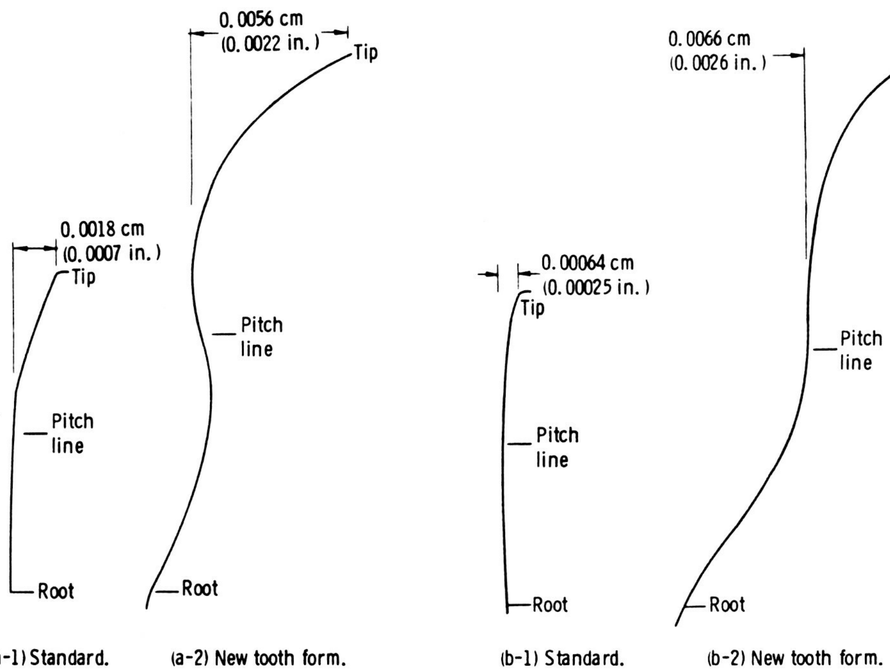


(a) New-tooth-form gear. (b) Standard gear.

Figure 3.- WADD test gears.

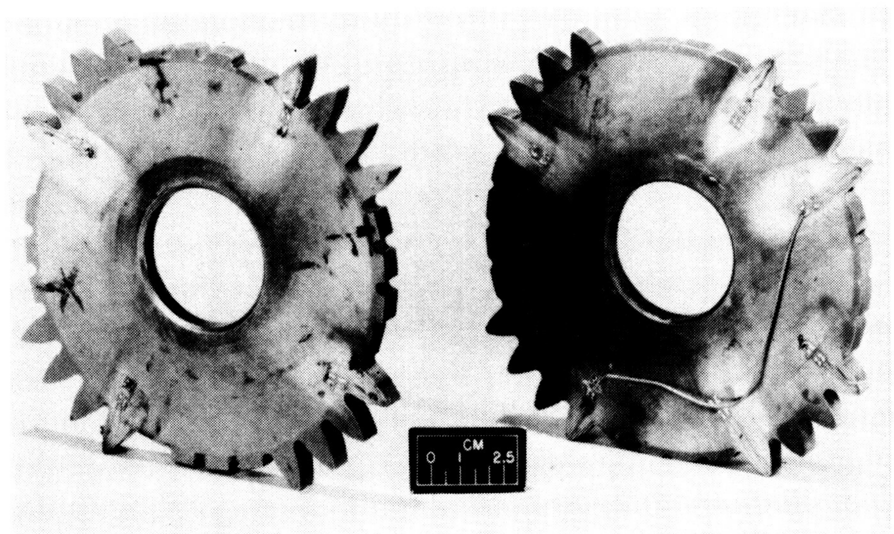
All gears were manufactured from AMS 6265C (AISI 9310) and were carburized and hardened to a case hardness on the tooth surface of 60 to 62 Rockwell C, with a core hardness of 35 to 40 Rockwell C. The chemical composition of the gear material is given in table III. The heat-treat schedule is given in table IV.

The bending-fatigue test gears were larger to fit the bending-fatigue test machine. The dimensions for the bending-fatigue test gears are given in table II. These gears, shown in figure 5, were similar to the smaller WADD test-rig gears shown in figure 3. The standard gear had a pressure angle of 25°, a tip relief of 0.001 cm (0.004 in.), and a contact ratio of 1.54. The NTF gear had a pressure angle of 20.5, an increased addendum of 0.222 cm (0.0875 in.), a tip relief of 0.006 cm (0.0025 in.), a flank addition of 0.006 cm (0.0025 in.), and a contact ratio of 2.25. An involute profile trace of the two gears is shown in figure 4(b).



(a) WADD test gears. (b) Bending fatigue test gears.

Figure 4. - Involute profiles for test gears.



(a) New-tooth-form gear. (b) Standard gear.

Figure 5. - Bending fatigue test gears.

Test Procedures

Test Lubrication

All gear scoring and surface-fatigue tests were conducted with a single batch of a synthetic polyol ester C_5, C_6, C_7 substituted pentaerythritol lubricating oil. The physical properties of this lubricant are summarized in table I. During the load scoring tests, the lubricant flow rate was held constant at 270 cm^3/min and was directed at the gear mesh exit. The lubricant inlet temperature was

TABLE I. - PROPERTIES OF SYNTHETIC POLYESTER

C_5, C_6, C_7 SUBSTITUTED PENTAERYTHRITOL

LUBRICATING OIL

Property	Typical data
Kinematic viscosity, cm^2/sec (cS), at-	
233 K (-40° F)	$10\ 764 \times 10^{-2}$ (10 764)
311 K (100° F)	28.51×10^{-2} (28.51)
372 K (210° F)	5.4×10^{-2} (5.4)
Flashpoint, K (°F)	533 (500)
Pourpoint, K (°F)	214 (-75)
Neutralization number, mg KOH/g	0.3

TABLE II. - TEST GEAR DATA

Dimension	Load scoring and surface fatigue (WADD) test gears		Tooth bending fatigue test gears	
	Standard gears	NTF gears	Standard gears	NTF gears
Tooth form	Involute	Noninvolute	Involute	Noninvolute
Pitch diameter, cm (in.)	8.890 (3.500)	8.890 (3.500)	15.240 (6.000)	15.240 (6.000)
Number of teeth	28	28	32	32
Diametral pitch, module in mm (DP in in^{-1})	3.2 (8)	3.2 (8)	4.76 (5.333)	4.76 (5.333)
Pressure angle at pitch point, deg	22.5	19.0	25	20.5
Face width, cm (in.):				
Wide gear	2.380 (0.937	2.380 (0.937)	---------------	--------------
Narrow gear	0.635 (0.250)	0.635 (0.250)	0.953 (0.375)	0.953 (0.375)
Outside diameter, cm (in.)	9.444 (3.718)	9.845 (3.876)	16.256 (6.400)	16.637 (6.550)
Base circle diameter, cm (in.)	8.214 (3.234)	8.301 (3.268)	13.813 (5.438)	14.275 (5.620)
Root diameter, cm (in.)	8.187 (3.223)	7.841 (3.087)	14.125 (5.561)	13.716 (5.400)
TIF diameter, cm (in.)	8.463 (3.332)	8.331 (3.280)	14.463 (5.694)	14.346 (5.648)
Full fillet radius, cm (in.)	0.165 (0.065)	0.165 (0.065)	0.188 (0.074)	0.188 (0.074)
Measurement over pins, cm (in.)	9.629 to 9.637 (3.791 to 3.794)	--------------	16.543 to 16.533 (6.513 to 6.509)	--------------
Pin diameter, cm (in.)		--------------	0.879 (0.346)	--------------
Backlash reference, cm (in.)	0.005 to 0.015 (0.002 to 0.006)	0.005 to 0.015 (0.002 to 0.006)	0.020 to 0.030 (0.008 to 0.012)	0.020 to 0.030 (0.008 to 0.012)
Contact ratio	1.30	2.25	1.54	2.25
Maximum specific sliding, v_s/v	0.283	0.540	---------------	--------------

TABLE III. - CHEMICAL COMPOSITION OF GEAR MATERIAL

CVM AISI 9310 - AMS 6265

	Element									
	C	Mn	P	S	Si	Ni	Cr	Mo	Cu	Fe
Content, wt %	0.10	0.62	0.008	0.003	0.28	3.36	1.33	0.15	0.19	Bal.

held constant at 347 K (165° F), but the lubricant outlet temperature varied with load as shown in figure 6. The test oil tank, which had a capacity of 2500 cm^3, was heated by an external resistance heater clamped on the copper-clad, stainless-steel oil tank. The oil was filtered through a 3-μm-absolute filter. During the surface fatigue tests the lubricant flow rate was 1300 cm^3/min, and two oil jets were used—one at the gear-mesh inlet and one at the gear mesh exit. The added oil flow for the surface fatigue tests was used to reduce the gear temperature and to prevent scoring failures.

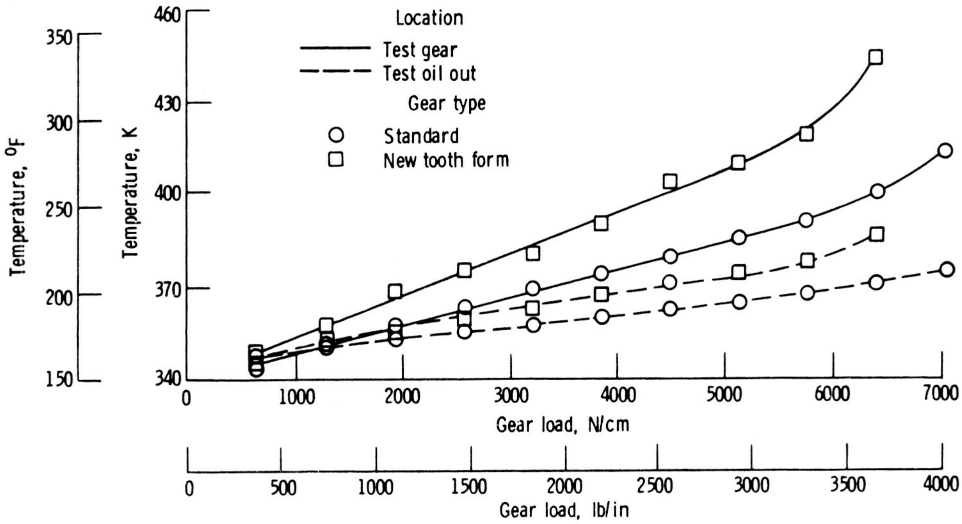


Figure 6. - Average maximum test-gear and oil-out temperatures for standard and NTF spur gears as function of gear load. Speed, 10 000 rpm; lubricant, synthetic polyol ester.

TABLE IV. - HEAT TREATMENT PROCESS FOR VACUUM-ARC-REMELTED (VAR) AISI 9310

Step	Process	Temperature		Time, hr
		K	°F	
1	Blast clean	----	----	7.5
2	Carburize 0.85 - 1.0 carbon potential	1200	1700	---
3	Cool under atmosphere to -	1090	1500	---
4	Air cool to room temperature	----	----	---
5	Stress relieve	922	1200	2
6	Copper plate all over	----	----	---
7	Austenitize	1117	1550	2.5
8	Oil quench	----	----	---
9	Subzero cool	189	-120	3.5
10	Double temperature	422	300	Two each
11	Finish grind	----	----	---
12	Stress relieve	422	300	2

Load Scoring Test Procedure

The test procedure used with the WADD gear test rigs for the load scoring tests conducted on both the standard and NTF test gears was ASTM test method D-1947 (ref. 4), with test conditions modified as follows: shaft speed, 10 000 rpm; test oil temperature, 347 K (165° F); and lubricant flow rate at the gear mesh exit, 270 cm^3/min. The test load increment was 63×10^3 N/m (360 lb/in) on the 0.635-cm (0.250-in.) wide test gear, and the load duration for each step load condition was 20 minutes instead of the 10 minutes used in the ASTM method. Test gears were considered scored when scoring had covered 22.5 percent of the test-gear face width as observed under an 18-power microscope.

Measurement of the test-gear temperature is not required in the ASTM procedure. However, the temperature of the narrow test gear was measured during each load step in all the load scoring tests. It was obtained from the infrared radiation that was emitted by the gear and detected by an infrared radiometer with a direct-reading digital readout. It was measured on the web of the narrow test gear just inside the root diameter, near the gear-mesh point. This point was used because its nearly constant emissivity factor minimized the error due to changing emissivity that would have resulted if the measurements were made at the gear teeth. A metal tube was used to prevent oil splashing and mist interference. A constant emissivity factor was further assured by applying black chromium plating on the gear web to closely approximate blackbody radiation.

Surface-Fatigue Test Procedure

Test conditions for the surface-fatigue tests were a shaft speed of 10 000 rpm, a test oil temperature of 347 K (165° F), and gear-mesh inlet and exit lubrication at a flow rate of 1300 cm^3/min. The endurance test load was 6440 N/cm (3680 lb/in) for both the standard gear and the NTF gear for the equal-load tests. This produced Hertz stresses of 173×10^7 Pa (250 000 psi) for the standard gear and 148×10^7 Pa (214 000 psi) for the NTF gear. The standard gear was also run at a load of 4720 N/cm (2700 lb/in) to produce a Hertz stress of 148×10^7 Pa (214 000 psi). These test loads were run to compare standard and NTF lives at equal loads and at equal Hertz stresses.

The test oil flow rate was increased from the 270 cm^3/min used in the load scoring tests to 1300 cm^3/min for the surface-fatigue tests. In addition, two test oil jets were used to provide lubricant to the test gears. One jet was directed toward the inlet side of the gear mesh, and the second jet was directed toward the exit side of the gear mesh. These jets increased the test oil flow rate and provided sufficient lubrication and cooling to the test gears to allow the higher gear load required for the surface-fatigue tests to be applied while decreasing the incidence of scoring or scuffing of the gear teeth. This condition gives an h/σ of 0.5 when calculated by the method of reference 6.

All the surface-fatigue tests were run-in by using the step-load procedure described previously, except that the test oil flow rate was increased. Each pair of surface-fatigue test gears were step loaded and visually inspected after each 20-minute step load until the desired fatigue gear load was reached. Following the 20-minute step-load condition, the short-term timer was switched out of the machine control circuit to allow uninterrupted operation of the rig. The rig was then restarted and the test gears were loaded to the desired load for the fatigue portion of the test. The test was continued at constant conditions until at least one gear tooth on the narrow test gear was pitted 75 percent across the working face, or for 250 hours, whichever occurred first. The test-gear condition was monitored during the fatigue portion of the test by a vibration noise transducer connected to a control circuit that would automatically turn the test rig off in the event of a significant increase in the vibration noise level. In addition, the narrow test gear was visually inspected three to four times during each 24-hour test period by using a strobe light through a baffled inspection port in the test-gear end cover.

Single-Tooth Bending-Fatigue Test Procedure

In the single-tooth bending-fatigue test, the test tooth was loaded such that the load was applied at the highest point of single-tooth contact (HPSTC) and tangent to the base circle on the standard gear and at the highest load point of the center tooth with three teeth in contact on the NTF gear. The center tooth receives more load because of the greater flexibility at the ends of the two outer teeth. The HPSTC load point was 7.77-cm (3.06-in.) radius for the standard test gears and 7.67-cm (3.02-in.) radius for the NTF test gears.

Before starting the single-tooth bending-fatigue test, a small-gage crackwire, for failure indication, was bonded to the side of each gear tooth to be tested. In addition, foil strain gages were bonded to at least two test gear teeth of both the standard and NTF gears to allow for the determination of the initial test load for each gear type. The center of the strain gage was located near the root fillet of the tooth on a radius of approximately 7.17 cm (2.82 in.) for the standard gear and 6.96 cm (2.74 in.) for the NTF gear.

A test was started by mounting the test gear on the hub of the supporting fixture and rotating it until it came in firm contact with a properly sized beam mounted on the reference block of the reaction frame (fig. 2). The gear was then clamped by friction pads to the mounting hub and was blocked at the backup tooth by the backup anvil. After the correct load point was established, the gage block was replaced by the load transfer anvil, and the gear was ready for testing. A lubricant coating was applied to the gear tooth and anvil.

The initial test load for each tooth design was established by strain gaging the first test tooth at its critical section and loading the tooth statically to obtain a 103×10^7-Pa (150 000-psi) bending stress. The load thus obtained was used as the upper limit for the initial fatigue load span in each gear-type test sequence. The lower load limit was maintained at 445 N (100 lbf) throughout the test in order to assure constant contact between the test tooth and the load transfer anvil and thereby eliminate any impact loading of the test tooth. The test rig was then turned on and adjusted to provide 1500 load cycles per minute. The test was continued uninterrupted until the tooth failed or 3×10^6 load cycles were applied, whichever occurred first. If the test-gear tooth sustained the initial test loading for 3×10^6 load cycles without failure, the peak load on the next new test-gear tooth was increased from that of the previous test increment by 15 percent of the initial load for the standard gears and by 30 percent for the NTF gears. If a failure occurred before 10^5 load cycles on the test tooth, the peak load on the next new test-gear tooth was decreased by 50 percent from that used in the previous test increment.

Results and Discussion

Load Scoring Tests

Load scoring tests were conducted using three pairs of standard test gears and five pairs of NTF test gears. The load-carrying capacity of each pair of gears used was determined twice. The data from the load scoring tests were plotted on Weibull coordinates as the 22.5-percent-scuffed failure load versus the percentage of specimens tested (fig. 7). The scoring failure loads for the NTF gears were considerably less than those for the standard gears. There was also much more scatter in the failure loads for the NTF gears, as indicated by the greater slope of the line. The load at which 50 percent of the NTF gears failed was only 78 percent of that for the standard gears and the load at which 10 percent of the NTF gears failed was only 64 percent of that for the standard gears. Typical scoring failures for the standard and NTF gears are shown in figure 8.

Test-gear temperatures were measured during each load step for all the load scoring tests conducted. The test-gear temperature normally increased during the first 5 to 7 min of a load step and then reached a steady-state condition for the particular gear design and test load condition. The average maximum gear bulk and oil-outlet temperatures are shown as a function of load for both the

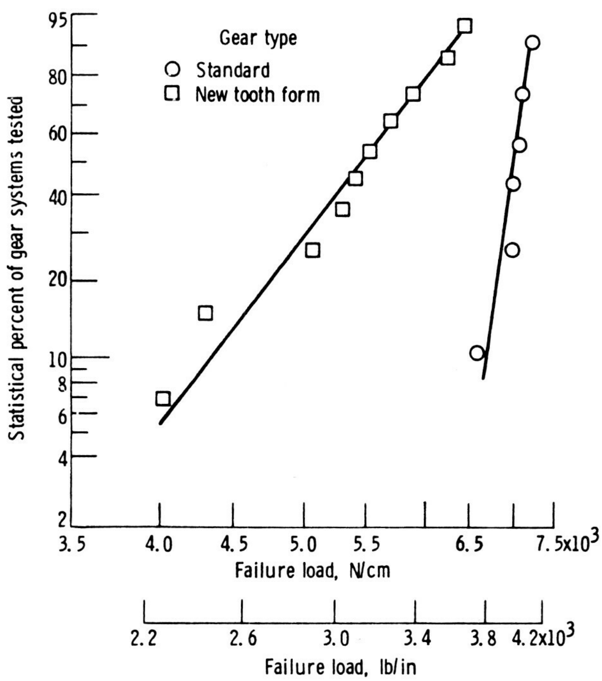


Figure 7. - Scoring load of standard and NTF spur gears as function of percentage of specimens tested. Speed, 10 000 rpm; lubricant, synthetic polyol ester; failure, surface 22.5 percent scuffed.

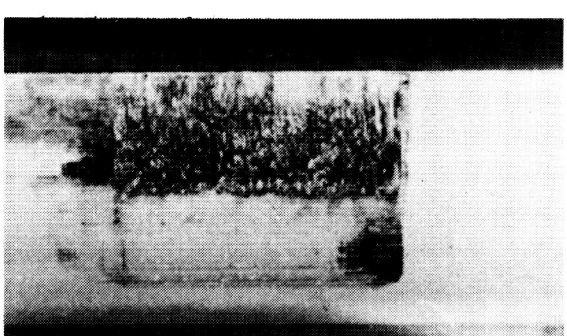
standard and NTF gears in figure 6. The NTF gear temperatures were significantly higher than the standard gear temperatures at each step load condition. The oil-outlet temperature was also higher with the NTF gears as a result of the higher gear temperature. The higher temperature operation for the NTF gears was not unexpected since they have a longer tooth that produces a higher sliding velocity (as shown in table II) and increased heating. The gear bulk temperatures for the 50-percent-failure load were nearly identical for both the NTF and standard gears. The NTF gears had a 50-percent-failure load of 5450 N/cm (3120 lb/in), which produced a gear bulk temperature of 409 K (277° F). The standard gears had a 50-percent-failure load of 6960 N/cm (3980 lb/in), which produced a gear bulk temperature of 407 K (274° F). The 10-percent-failure load produced gear bulk temperatures of 394 K (250° F) for the NTF gear and 401 K (262° F) for the standard gear. These temperatures are fairly close, considering the difference in loads. From these tests it was concluded that scoring failure is a function of gear-tooth bulk temperature for a given lubricant, where the temperature is a function of gear design, operating load, and speed.

Surface-Fatigue Tests

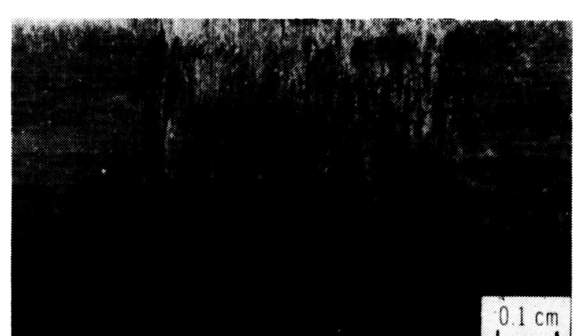
Surface-fatigue tests were conducted with two groups of standard gears and one group of NTF gears. Ten surface-fatigue tests were conducted with each group of gears. The three groups were tested to determine the relative surface-fatigue lives of the NTF and standard gears at the same Hertz stress and at the same load. The equal-stress condition was a maximum Hertz stress of 148×10^7 Pa (214 000 psi). The equal-load condition was 6440 N/cm (3680 lb/in), which gave maximum Hertz stresses of 173×10^7 Pa (250 000 psi) for the standard gears and 148×10^7 Pa (214 000 psi) for the NTF gears. The surface-fatigue tests were run with the oil flow increased to 1300 cm^3/min in order to reduce the gear temperature and the probability of scoring. With the higher oil flow, the temperature was 371 K (208° F) for both gear types at the higher load and 367 K (202° F) for the standard gear at the lower load. The methods of reference 7 were used to evaluate the fatigue data. Typical fatigue spalls for the standard and NTF gears are shown in figure 9.

The data for an equal Hertz stress of 148×10^7 Pa (214 000 psi) with both the standard and NTF gears are shown on the Weibull plot of figure 10(a). These data are summarized in table V. The 10-percent life was 14×10^6 cycles for the standard gears and 21×10^6 cycles for the NTF gears. The lower dynamic loading produced by the NTF gears could be the reason for the slight increase in surface fatigue life for these gears. However, the difference in life is statistically insignificant. Hence, the lives of both the standard and NTF gears are statistically equal for the same Hertz stress.

The results of the tests with equal loads, 6440 N/cm (3680 lb/in), for both the NTF and standard gears are shown on the Weibull plot of figure 10(b). These data are summarized in table V. The 10-percent life was 21×10^6 cycles for the NTF gears and 4×10^6 cycles for the standard gears. The NTF gears have approximately five times the 10-percent life of the standard gears at equal loads. This difference is statistically significant. The maximum Hertz stresses for the standard and NTF gears were 173×10^7 and 148×10^7 Pa (250 000 and 214 000 psi), respectively. Reference 8 states that the pitting fatigue life of gears is inversely proportional to the contact stress to the ninth power ($L \propto S^{-9}$).



(a) Standard gear.



(b) New-tooth-form gear.

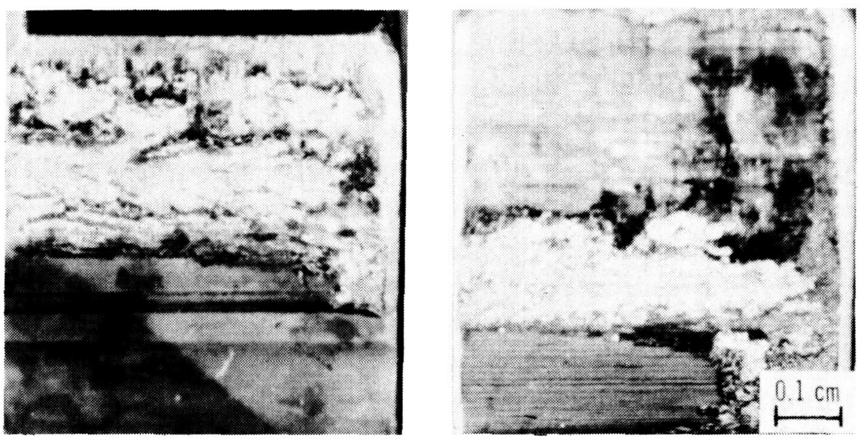
Figure 8. - Typical scoring failures.

Based on this relationship the relative difference in life between the standard and NTF gears could be contemplated. The tests reported herein show the relative lives of the standard and NTF gears to be a function of the contact stress for a given load.

Single-Tooth Bending-Fatigue Tests

Single-tooth, one-way bending-fatigue tests were conducted on both standard and NTF spur gears that were designed for the single-tooth bending tests. The standard gear was a 15.24-cm (6-in.) pitch-diameter, 4.76 module (5.333 DP) with a 25° pressure angle. The NTF gear was a 15.24-cm (6-in.) pitch diameter, 4.76 module (5.333 DP) with a 20.5° pressure angle and an increased addendum of 0.19 cm (0.075 in.). This increased addendum required a smaller root diameter. The NTF bending-fatigue gear, therefore, had a much thinner and longer tooth than the standard bending fatigue gear.

The tooth bending load was applied tangent to the base circle at 7.77-cm (3.06-in.) radius for the standard gear and at 7.67-cm (3.02-in.) radius for the NTF gear. The initial NTF-gear load that gave a strain-gage bending stress of 10.35×10^8 Pa (150 000 psi) was 13 878 N (3120 lbf). The blending stress for this load was computed, by the AGMA method of calculation (ref. 9), to be 10.3×10^8 Pa (149 400 psi) for a J-factor of 0.297. The load was then increased in increments of 30 percent of the initial load until failure occurred. The first failure load was 160 percent of the initial load. The load was then decreased to 50 percent of the previous increment when no failure occurred. The results are plotted in figure 11(a). From a line drawn through the lowest failure load to the cutoff point, the predicted failure load for the NTF gear at 3×10^6 stress cycles would be 21 084 N (4740 lbf), which would give an AGMA calculated bending stress of 15.7×10^8 Pa (227 000 psi).



(a) Standard gear. (b) New-tooth-form gear.

Figure 9. - Typical pitting fatigue failures.

TABLE V. - FATIGUE LIFE RESULTS FOR STANDARD AND NEW-TOOTH-FORM GEARS AT EQUAL LOAD AND EQUAL HERTZ STRESS

Gear type	Hertz stress		Tooth load		Average test-gear temperature		Failure index	Life, cycles		Confidence number (10-Percent life)
	N/m^2	psi	N	lbf	K	°F		10 Percent	50 Percent	
Standard	148×10^7	214 000	3002	675	368	202	8 of 10	14×10^6	44×10^6	--
New tooth form	148×10^7	214 000	4092	920	370	207	8 of 10	21×10^6	61×10^6	65
Standard	173×10^7	250 000	4092	920	371	208	10 of 10	4×10^6	12.6×10^6	99

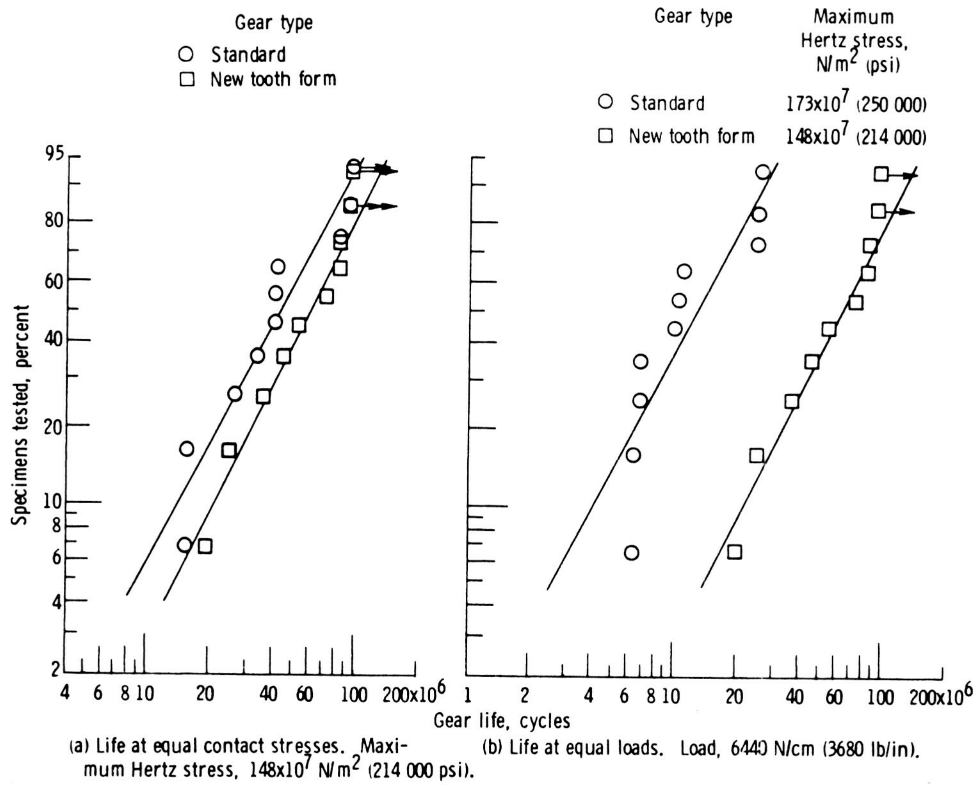


Figure 10. - Pitting fatigue lives of standard and NTF spur gears. Speed, 10 000 rpm; lubricant, synthetic polyol ester; failure, temperature, 370 K (207° F).

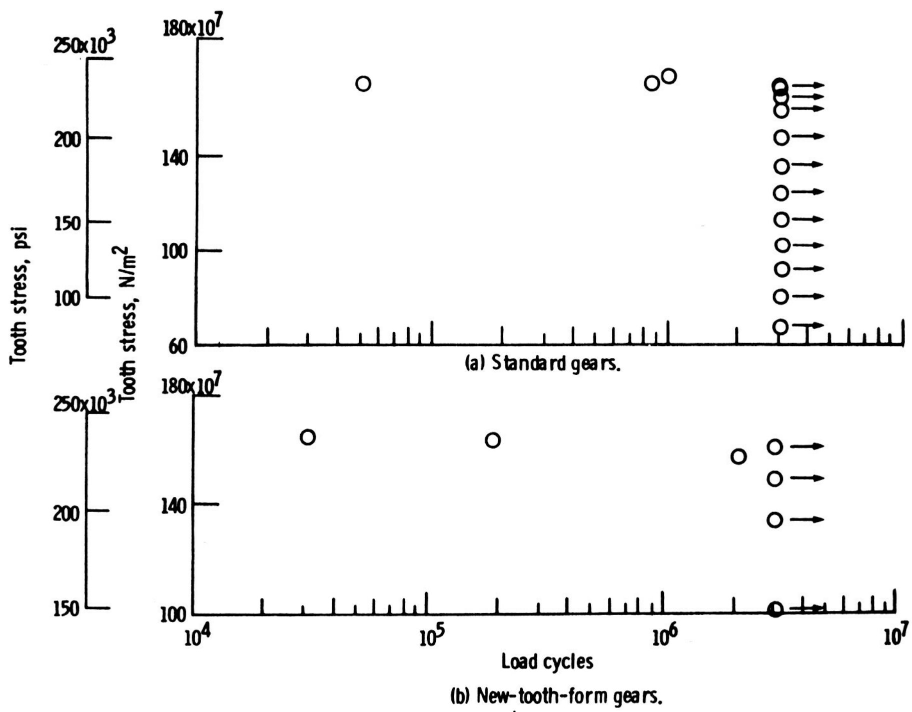
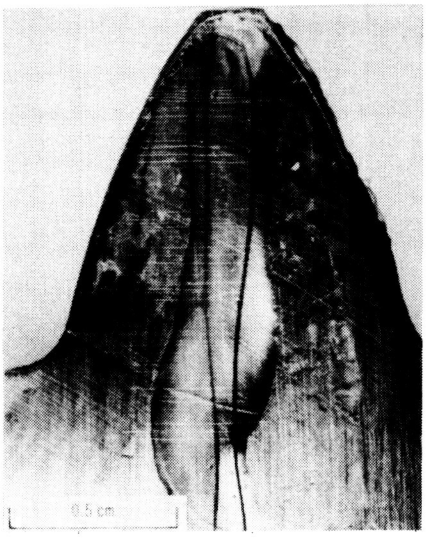
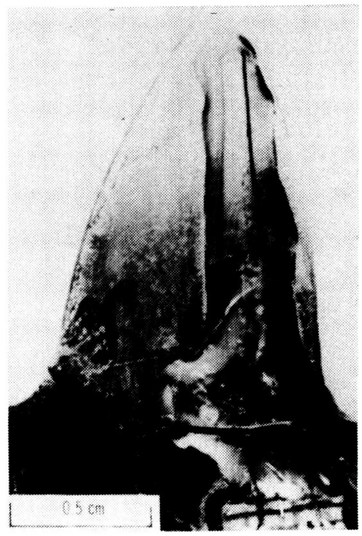


Figure 11. - Maximum test cycle load for standard and NTF spur gears as function of load cycles to failure. Cycle rate, 1500 per minute.



(a) Standard gear.



(b) New-tooth-form gear.

Figure 12. - Typical bending fatigue failures.

The initial standard-gear load that gave a strain-gage bending stress of 10.2×10^8 Pa (147 650 psi) was 22 262 N (5005 lbf). This load gave an AGMA-computed bending stress of 10.2×10^8 Pa (147 650 psi) for a J-factor of 0.483. The load for the standard gear was increased in increments of 2566 N (577 lbf) until failure occurred. The load was then decreased to 50 percent of the previous increment. Three failures occurred at less than 3×10^6 stress cycles, including one backup tooth failure. The results are plotted in figure 11(b). From a line drawn through the lowest failure points to the cutoff point, the predicted failure load at 3×10^6 stress cycles would be 40 032 N (9000 lbf). This load would give an AGMA-computed bending stress of 18.3×10^8 Pa (265 000 psi), which is approximately 17 percent higher than the failure stress for the NTF gear.

This difference in calculated bending stress is most likely the result of the peak bending stress occurring at a slightly different location from that determined by the AGMA method. The strain gages would also have to be located exactly to measure the peak stress. The crack in the failed gear started close to the junction of the root radius and involute curve, but the AGMA layout indicated that the highest stress point was slightly into the root radius. In addition, the small number of failures at the load condition makes the difference in failure load statistically insignificant.

Typical bending failures for both gears are shown in figure 12. The failure load for the standard gears was nearly double (190 percent) that the NTF gears. This would indicate that, in many cases, the NTF gear would fail in bending before the standard gear when the load sharing does not reduce the center-tooth load to approximately one-half that on the standard gear. However, this condition could be changed by good design practice using a good tooth-load-sharing-and-dynamic-gear-strength analysis. If this were done, the NTF gear would have less dynamic load, good load sharing, and better load-carrying capacity than a standard gear designed for the same conditions. However, the test results do show that there are many cases where the NTF, or high-contact-ratio, design will have less load-carrying capacity than the standard gear design.

Summary of Results

Scoring tests, surface fatigue tests, and single-tooth bending-fatigue tests were conducted with four sets of spur gears of standard design and three sets of spur gears of the new-tooth-form (NTF) design. Scoring tests were conducted in a Wright Air Development Division (WADD) gear test rig at a speed of 10 000 rpm using a synthetic polyol ester C_5, C_6, C_7 substituted pentaerythritol oil. Surface fatigue tests were conducted in the same rig at a speed of 10 000 rpm and Hertz stresses of 173×10^7 and 143×10^7 Pa (250 000 and 214 000 psi). Single-tooth bending-fatigue tests were conducted on both the standard and NTF gears at an initial load that produced a 10.35×10^8-Pa (150 000-psi) bending stress. The gears were load cycled to failure or for 3×10^6 cycles, whichever occurred first. The load was increased after each test until failure occurred at 3×10^6 cycles or less. The following results were obtained:

1. Both the standard and NTF gears scored at a gear bulk temperature of approximately 409 K (277° F). At this temperature the load on the NTF gears was 22 percent less than the load on the standard gears. The scoring failure was a function of gear bulk temperature,where for a given lubricant the temperature is a function of gear design, operating load, and speed.

2. The pitting fatigue lives of the standard and NTF gears are statistically equal for the same maximum Hertz stress.

3. The pitting fatigue life of the NTF gears was approximately five times that of the standard gears at equal loads. The difference in life was a function of stress to the ninth power.

4. The standard gear tooth failed at a 17 percent higher bending stress than the NTF gear when stress was calculated by the AGMA method. This difference is not statistically significant for this test.

5. The minimum load to produce a bending fatigue failure at 3×10^6 stress cycles for the standard gears was 1.9 times that for the NTF gears.

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