Surface Pitting Fatigue Life of Noninvolute Low-Contact-Ratio Gears

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Abstract: Spur gear endurance tests were conducted to investigate the surface pitting fatigue life of noninvolute gears with low numbers of teeth and low contact ratios for use in advanced applications. The results were compared with those for a standard involute design with a low number of teeth. The gear pitch diameter was 8.89 cm (3.50 in.) with 12 teeth on both gear designs. Test conditions were an oil inlet temperature of 320 K (116° F), an oil outlet temperature of 350 K (170° F), a maximum Hertz stress of 1.49 GPa (216 ksi), and a speed of 10,000 rpm. The following results were obtained: The noninvolute gear had a surface pitting fatigue life approximately 1.6 times that of the standard involute gear of a similar design. The surface pitting fatigue life of the 3.43-pitch AISI 8620 noninvolute gear was approximately equal to the surface pitting fatigue life of an 8-pitch, 28-tooth AISI 9310 gear at the same load, but at a considerably higher maximum Hertz stress.

Introduction

Many gears used in aircraft and other transmissions have size limitations based on the minimum number of teeth that can be cut on a pinion without undercutting the teeth.[1] If the number of teeth is made fewer than this minimum, a weaker tooth will be produced because of the undercutting. One method currently used to allow smaller minimum numbers of teeth on a pinion is to change the involute radius at both the dedendum and the addendum. Smaller numbers of teeth can be manufactured on a given pinion with a standard addendum by increasing the involute radius in the dedendum region, where it normally becomes very short, and decreasing the radius in the addendum region, where it normally increases rapidly. This can be seen on an involute chart as a positive modification in the dedendum region and a negative modification, similar to a large tip relief, in the addendum region. In addition to allowing smaller numbers of teeth without undercutting, this method, sometimes called new tooth form,[2] also reduces the maximum Hertz stress in the dedendum region, where the very short involute radius has been increased. This increased involute radius may also improve the gear tooth's surface fatigue life and possibly improve its scoring resistance. The new tooth form can be used for most spur or helical gears with either normal or high contact ratios to reduce the effect of undercutting on gears with fewer than the minimum number of teeth.

The objectives of the research reported herein were (1) to investigate the noninvolute modifications for use as a design method for gears with small numbers of teeth, (2) to determine the surface endurance characteristics of a spur gear with the new tooth form, and (3) to compare the results with those for a standard involute gear of similar design parameters. In order to accomplish these objectives, tests were conducted with one

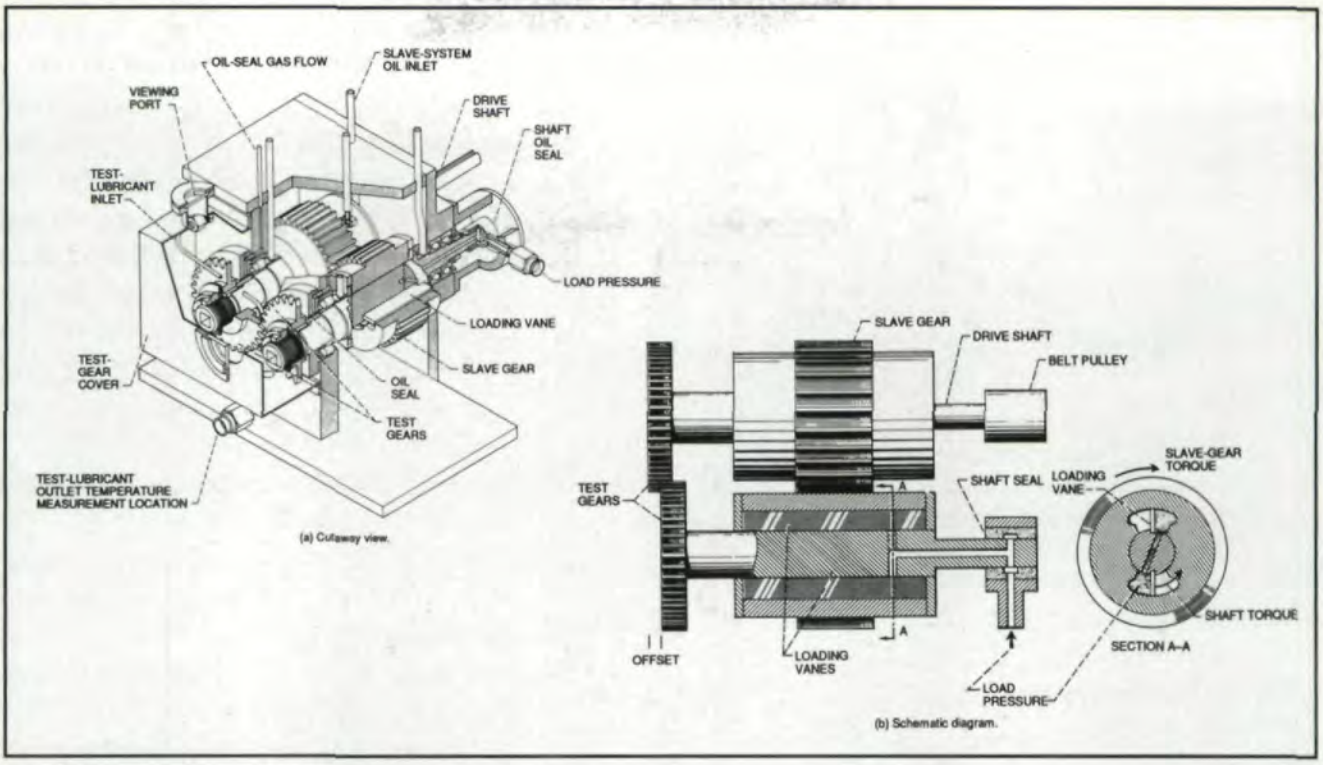

Fig. 1 NASA Lewis Research Center's gear fatigue test apparatus.

lot each of spur gears made from a single heat of AISI 8620 AMS 6274 material in the noninvolute design and in the standard involute design. The gear pitch diameter was 8.89 cm (3.50 in.). Test conditions included an oil inlet temperature of 320 K (116° F), which resulted in an oil outlet temperature of 350 K (170° F), a maximum Hertz stress of 1.49 GPa (216 ksi), and a shaft speed of 10,000 rpm.

The work reported herein was conducted as a joint effort of NASA Lewis Research Center, General Electric Co. Ordinance Systems Division, and ITW Spiroid Division.

Apparatus and Procedures

Gear Test Apparatus. The gear fatigue tests were performed in the NASA Lewis gear fatigue test apparatus (Fig. 1). This test rig uses the four-square principle (recirculating power) of applying the test gear load so that the input drive needs to overcome only the frictional losses in the system. A schematic of the test rig is shown in Fig. 1b. Oil pressure and leakage flow are supplied to the loading vanes through a shaft seal. As the oil pressure is increased on the loading 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 loading 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.

Separate lubrication systems are provided for the test gears and the main gearbox. The two lubrication systems are separated at the gearbox shafts by nitrogen-gas-pressurized labyrinth seals. The test gear lubricant is filtered through a 5μm nominal fiberglass filter. The test lubricant can be heated electrically with an immersion heater. The skin temperature of the heater is controlled to prevent overheating 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 belt-driven test rig can be operated at several fixed speeds by changing pulleys. The operating speed for the tests reported herein was 10,000 rpm.

Test Gears. The test gears are shown in Fig. 2. Their dimensions are given in Table 1. All the gears had a nominal surface finish on the tooth face of 0.82 μm (32 μin.) rms. The baseline gears had a standard involute profile; the noninvolute gear had a profile that deviated from a standard

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is a gear consultant for NASA and numerous industrial companies. During his career at NASA he has authored over 50 papers in the gear and bearing research fields and has done extensive research on gear materials and processes for improved gear life at increased operating temperatures. He is an active member of ASME.

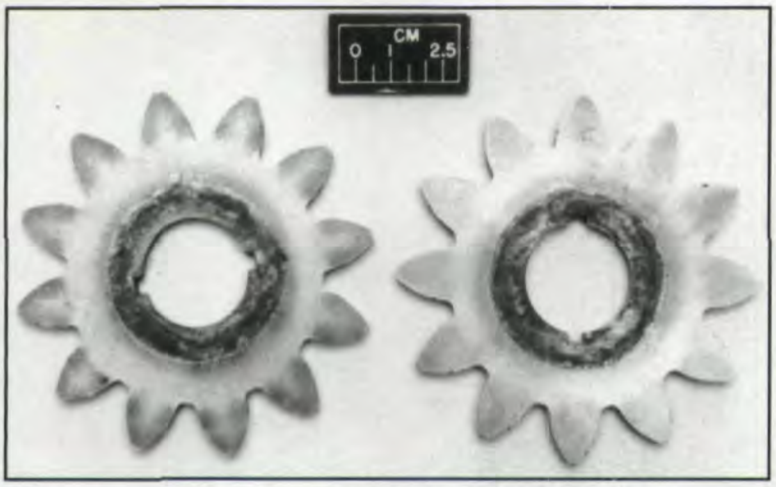


Fig. 2 Test gear configurations. (a) Standard involute. (b) Noninvolute.

Table 1 Description of Test Gears*	
Pitch diameter, cm (in.)	8.89 (3.50)
Number of teeth	12
Module (diametral pitch)	7.4083 (3.4286)
Pressure angle (deg)	20°
Face width, cm (in.)	0.635 (0.250)
Outside diameter cm (in.)	10.2558 (4.0377)
Root diameter, cm (in.)	7.248 (2.854)
Tooth thickness (arc), cm (in)	1.3528 (0.5326)
Fillet radius, cm (in.)	0.198 (0.078)
Surface finish (min.), m (in.)	0.8 (32)

*Gears were identical except for the tooth form, which was involute for the standard gear and noninvolute for the other gear.

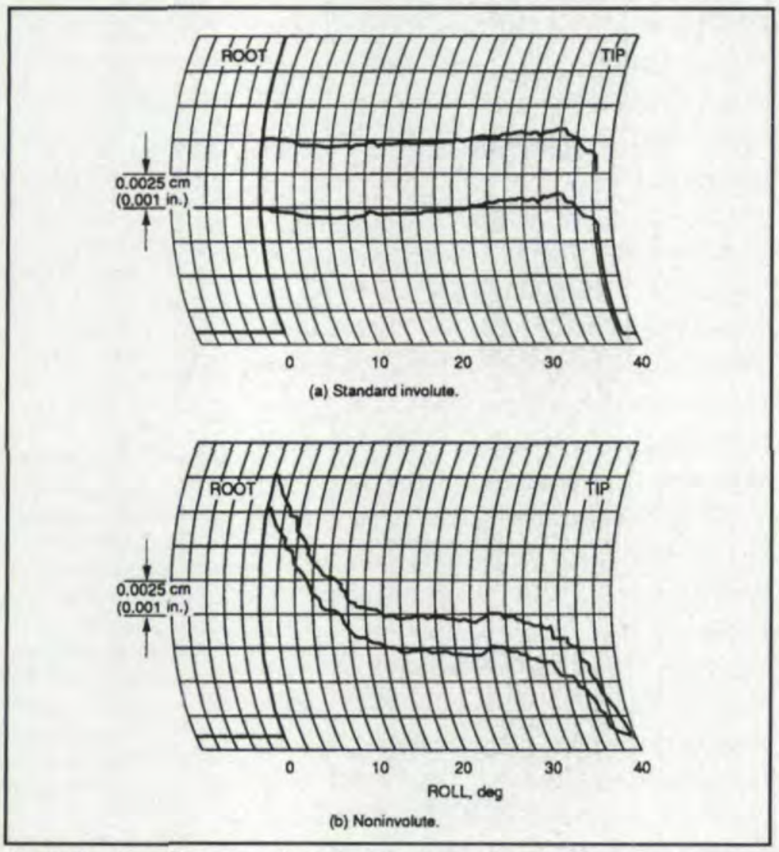


Fig. 3 Tooth profile traces of standard involute and noninvolute gears.

involute profile as shown in Fig. 3. The pressure angle for all the gears was 20° and the contact ratio was 1.15.

Test Materials. The test gears were manufactured from one lot of AISI E8620 AQHR AMS 6274 material. The chemical composition of the gear material is shown in Table 2. The heat treatment for the test gears is described in Table 3. The case hardness was R_c 60 with a case depth of 0.147 cm (0.058 in.); the core hardness was R_c 40. Photomicrographs of the case and core regions of the gear material are shown in Figs. 4a and b.

Lubricant. All the gears were lubricated with a single batch of synthetic paraffinic oil. The physical properties of this lubricant are summarized in Table 4. Five volume percent of an extreme-pressure additive, designated Lubrizol 5002 (partial chemical analysis given in Table 4), was added to the lubricant.

Test Procedure. After the test gears were cleaned to remove their protective coating, they were assembled on the test rig. The test gears ran in an offset condition with a 0.30 cm (0.120 in.) tooth-surface overlap to give a 0.28 cm (0.110 in.) load surface on the gear face after allowing for the edge radius on the gear teeth. If both faces of the gears were tested, four fatigue tests could be run for each set of gears. All tests were run-in at a load per unit length of 1230 N/cm (700 lb/in.) for one hour. The load was then increased to 5800 N/cm (3300 lb/in.), which resulted in a 1.49 GPa (216 ksi) pitch-line maximum Hertz stress. The tooth bending stress at the worst load point was calculated to be 0.10 GPa (15 ksi).

Operating the test gears at 10,000 rpm gave a pitch-line velocity of 46.55 m/sec (9163 ft/min). Lubricant was supplied to the inlet mesh at 800 cm^3/min (49 $in.^3$/min) and 320 ± 6 K (116 ± 10° F). The lubricant outlet temperature was nearly constant at 350 ± 3 K (170 ± 5° F). The tests ran continuously (24 hr/day) until the rig was automatically shut down by the vibration detection transducer (located on the gearbox adjacent to the test gears) or until 500 hours of operation without failure were completed. The lubricant circulated through a 5μm fiberglass filter to remove wear particles. For each test, 3.8 liters (1 gal.) of lubricant was used. At the end of each test, the lubricant and the filter element were discarded. Oil inlet and outlet temperatures were continuously recorded on a strip-chart recorder.

The pitch-line elastohydrodynamic (EHD) film

Table 2 Chemical Composition of AISI 8620 Gear Material	
Element	Content, wt %
Carbon (core)	0.22
Manganese	.82
Phosphorus	.013
Sulfur	.01
Silicon	.27
Copper	.16
Chromium	.49
Molybdenum	.16
Nickel	.54
Iron	Balance

Table 3 Heat Treatment Procedure (Material, AISI 8620)

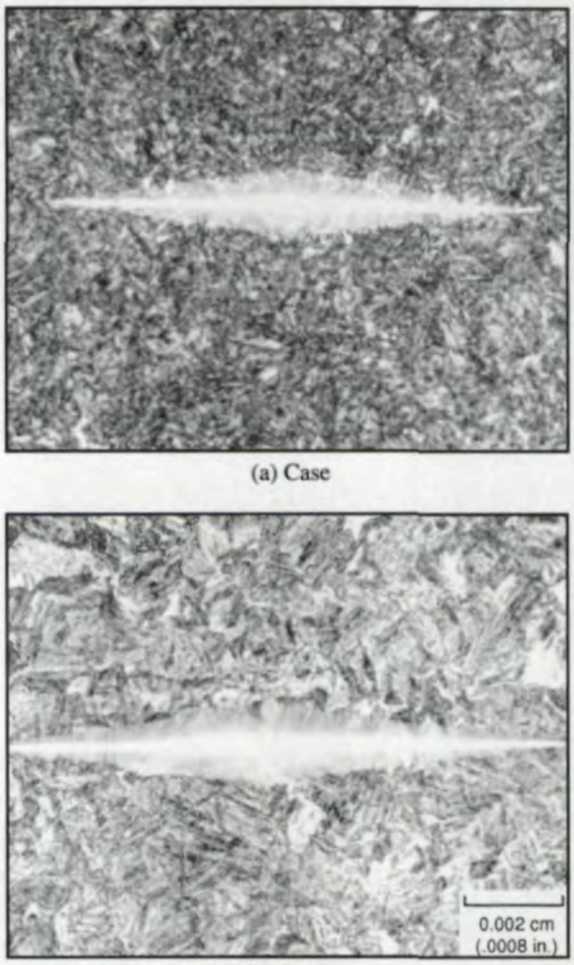
Step	Process
1	Carburize at 1200 K (1700° F) for 8 hrs.
2	Temper at 922 K (1200° F) for 1 hr.
3	Austenize or harden at 1118 K (1550° F) for 2.5 hrs.
4	Oil quench
5	Deep freeze at 190 K (-120° F) for 3.5 hrs.
6	Temper at 436 K (325° F) for 2 hrs.

thickness was calculated by the method of Dowson and Higginson.[3] It was assumed, for this calculation, that the gear temperature at the pitch line was equal to the oil outlet temperature and that the oil inlet temperature to the contact zone was equal to the gear temperature, even though the oil inlet temperature was considerably lower. It is possible that the gear surface temperature was 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 0.94 μm (37 μin.), which gave an initial ratio of film thickness to composite surface roughness h/σ of 0.82 at the 1.49 GPa (216 ksi) pitch-line maximum Hertz stress.

Each test conducted with a pair of gears was considered as a system and, hence, a single test. A maximum of four tests were conducted with each pair of gears. Test results were evaluated by using Weibull plots calculated by the method of Johnson.[4] (A Weibull plot is the number of stress cycles versus the statistical percentage of gear systems failed.)

Results and Discussion

One lot each of AISI 8620 standard involute gears and noninvolute gears with the modified involute profile was endurance tested. Test conditions included a tangential tooth load of 5800 N/cm (3300 lb/in.), which produced a maximum Hertz stress of 1.49 GPa (216 ksi), and a speed of 10,000 rpm. The gears failed by classical subsurface pitting fatigue. The surface pitting fatigue life results of these tests are shown in the Weibull plots of Fig. 5 and are summarized in Table 5. Surface pitting fatigue life results for the standard involute gears are shown in Fig. 5a. The 10% and



(a) Case

(b) Core.

Fig. 4 Photomicrographs of case and core of AISI 8620 test gear material.

Table 4 Lubricant Properties
(Lubricant, synthetic paraffinic oil plus additive.*)

Kinematic viscosity, cm^2/sec (cS) at:		
244 K (-20° F)	2500×10^{-2}	(2500)
311 K (100° F)	31.6×10^{-2}	(31.6)
372 K (210° F)	5.5×10^{-2}	(5.5)
477 K (400° F)	2.0×10^{-2}	(2.0)
Flash point, K (°F)	508	(455)
Fire point, K (°F)	533	(500)
Pour point, K (°F)	219	(-65)
Specific gravity	0.8285	
Vapor pressure at 311 K (100° F). torr	0.1	
Specific heat at 311 K (100° F), J/kg K (Btu/lb °F)	2190 (0.523)	

*Additive: 5 vol % Lubrizol 5002 (phosphorus, 0.03 vol %; sulfur, 0.93 vol%).

50% lives were 14.6×10^6 and 45.8×10^6 stress cycles (24.3 and 76.3 hr), respectively. The failure index (i.e., the number of fatigue failures out of the number of sets tested) was 20 out of 20. A typical fatigue spall that occurs near the pitch line on a standard involute gear is shown in Fig. 6a. This spall is similar to those observed in rolling-element fatigue tests. Pitch-line pitting is the result of a high subsurface shearing stress, which develops subsurface cracks. The subsurface cracks propagate into a crack network that results in a fatigue spall slightly below the pitch line, where the sliding condition is more severe.

Surface pitting fatigue life results for the noninvolute gear systems are shown in Fig. 5b. The 10% and 50% surface pitting fatigue lives were 23.2×10^6 and 62.5×10^6 stress cycles (38.1 and 104.2 hr), respectively. The failure index was 18 out of 18. Fig. 6b shows a typical fatigue spall for a noninvolute gear. The fatigue spalls and tooth wear were very similar for both types of gears. The 10% life of the noninvolute gear was approximately 1.6 times that of the standard involute gears. The confidence number was 77%, which indicates that the difference in surface fatigue life is statistically significant. (The confidence number indicates the percentage of time that the relative lives of the two types of gears will occur in the same order.)

The gear life data are summarized in Fig. 5c. The surface pitting fatigue test data show the noninvolute gear to be superior in surface pitting fatigue life to the standard involute gear for the gear sets tested. It is not clear why there was an improvement in surface pitting fatigue life for the noninvolute gear, since the fatigue failures occurred near the pitch line, where the load, Hertz stress, and involute radius are the same for both types of gears. Since the gears had a very low contact ratio on only 1.15 because of the low number of teeth, it is possible that the dynamic load for the noninvolute gear was less than that for the standard involute gear. Data from Lin et al.[5] indicate that certain types (or length) of profile modification give reduced dynamic loads. Since the noninvolute gear is a special form of profile modification, it may have a reduced dynamic load.

The 10% pitting fatigue life of the noninvolute

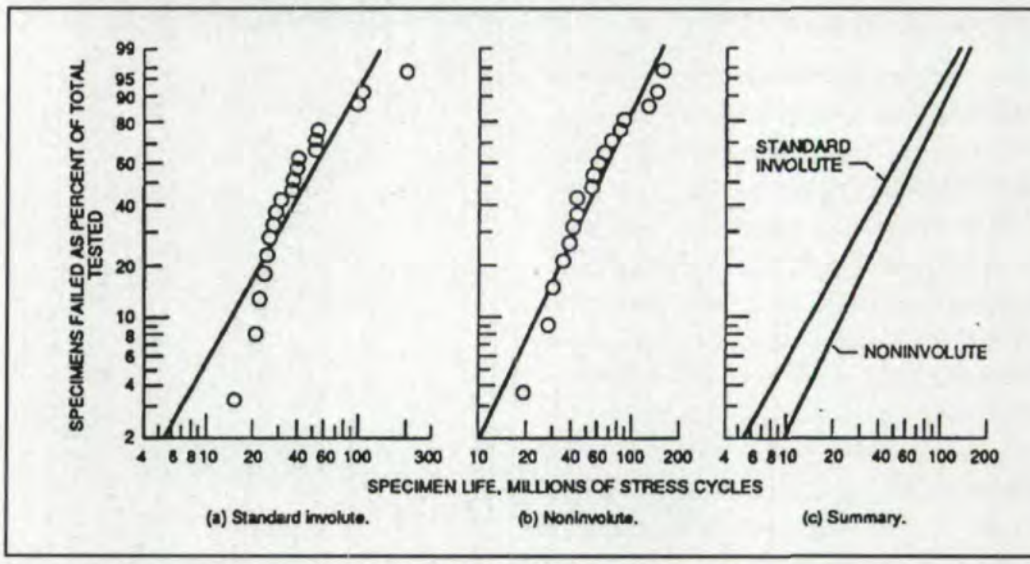


Fig. 5 Pitting fatigue lives of carburized and hardened AISI 8620 AMS 6274 standard involute and noninvolute gears. Speed, 10,000 rpm; lubricant, synthetic paraffinic oil with 5 vol % extreme-pressure additive; maximum Hertz stress, 1.49 GPa (216 ksi); temperature, 350 K (170° F).

Table 5 Results of Spur Gear Fatigue Life Tests

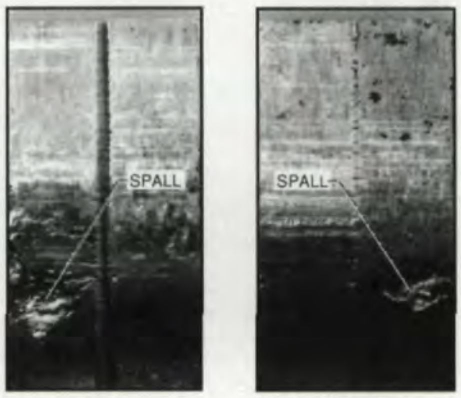
Pitch diameter, 8.2542 cm (3.2497 in.); maximum Hertz stress, 1.49 GPa (216 ksi); speed, 10,000 rpm; lubricant, synthetic paraffinic oil; gear temperature, 350 K (170 °F.)

Tooth form	10% life	50% life	Weibull slope	Failure index*	Confidence no. at 10% level#
	Gear system life, revelutions				
Involute	14.6×10^6	45.8×10^6	1.64	20 out of 20	–
Noninvolute	23.2	62.5	1.9	18 out of 18	77

* Number of surface fatigue failures out of number of gears tested.

Percentage of time that 10% life obtained with involute gears will have the same relation to 10% life obtained with noninvolute gears.

gear (23.3 million cycles) at the 1.49 GPa (216-ksi) maximum Hertz stress was approximately equivalent to that of a standard AISI 9310 8-pitch gear (19 million cycles) at the same load, but with a much higher maximum Hertz stress of 1.71 GPa (248 ksi).[6] The 8-pitch gears had a contact ratio of 1.638, in contrast to 1.15 for the 3.43-diametral-pitch gears used in these tests. Normally, the gear life is inversely proportional to the stress to the ninth power.[7] In these tests the low contact ratio may have resulted in higher dynamic loads on the teeth and therefore in a higher dynamic maximum Hertz stress and a reduced life. Results form the NASA gear dynamic analysis program show the 12-tooth gear to have a dynamic load factor of 1.5, in contrast to 1.04 for the 8-pitch, 28-tooth gear. In addition, the AISI 8620 material may have a lower fatigue life at the same stress than the AISI 9310 material.



(a) Standard gear. (b) Noninvolute gear.

Fig. 6 Typical fatigue spall.

Summary of Results

Spur gear endurance tests were conducted to investigate the effect of the noninvolute tooth form on the surface pitting fatigue life of gears with low numbers of teeth. The results were compared with those for a standard involute design with the same number of teeth. The gear pitch diameter was 8.89 cm (3.50 in.) with 12 teeth on both gear designs. Test conditions were on oil inlet temperature of 320 K (116° F), an oil outlet temperature of 350 K (170° F), a maximum Hertz stress of 1.49 GPa (216 ksi), and a speed of 10,000 rpm. The following results were obtained:

1. The noninvolute gear had a surface pitting fatigue life approximately 1.6 times that of a standard involute gear of similar design.

2. The surface pitting fatigue life of the 3.43-pitch, 12-tooth AISI 8620 noninvolute gear was approximately equal to the surface pitting fatigue life of an 8-pitch, 28-tooth AISI 9310 gear at the same load, but at a considerably higher maximum Hertz stress.

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