EHL Film Thickness, Additives and Gear Surface Fatigue

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Introduction

Aircraft transmissions for helicopters, turboprops and geared turbofan aircraft require high reliability and provide several thousand hours of operation between overhauls. In addition, they should be lightweight and have very high efficiency to minimize operating costs for the aircraft.

Most of the aircraft operating today use turbine engine lubricants to lubricate the transmissions. These provide good lubrication, thermal stability and low operation temperatures for the turbine engines, but they are less than optimum for reliability and long life for transmissions.

Tests with rolling element bearings have shown that bearing life is affected by the lubricant elastohydrodynamic (EHL) film thickness (Refs. 1 and 2). When the EHL film thickness divided by the composite surface roughness h/σ is less than one, the life of rolling element components is considerably reduced.

In gearing the effect of operating with an h/σ of less than one is more pronounced than it is with bearings. The higher sliding conditions encountered with gearing cause increased surface heating and higher friction coefficients, resulting in reduced EHL film thickness and surface fatigue life and increased wear or scoring risk.

Gear tests conducted with several lubricant additives have shown that the gear surface fatigue life can be improved somewhat with the right choice of additives (Refs. 3 and 4). Lubricants with the same viscosity, but with different additives, produced gear surface fatigue lives with a difference of five to one. The above-mentioned tests indicated the necessity of having the proper additive in the lubricant but did not determine what effect different lubricant viscosities of the same base stock would have on gear fatigue life.

The effect of the EHL film thickness on scoring and wear under various slide-to-roll ratios was determined in Reference 5 using rolling sliding cylinders. When the specific film thickness, Λ or h/σ, was less than or equal to 0.3, the rolling sliding cylinders experienced wear and scoring and elevated friction coefficient and temperature. These tests also showed an increase in scoring load capacity with EP additives in the lubricant.

Lubricant suppliers have recognized the need to supply better lubricants for modern gearboxes operating at increased power density (Ref. 6). Tests have shown that lubricants with the proper base stock, viscosity and additives can improve the load capacity and efficiency of transmissions.

The research work reported herein was undertaken to investigate the effects of lubricants with the same base stock, but with different viscosities on the surface fatigue life of AISI 9310 spur gears. The objectives were: (1) to investigate the effect of seven different lubricants on the surface fatigue life of hardened steel spur gears, (2) to compare

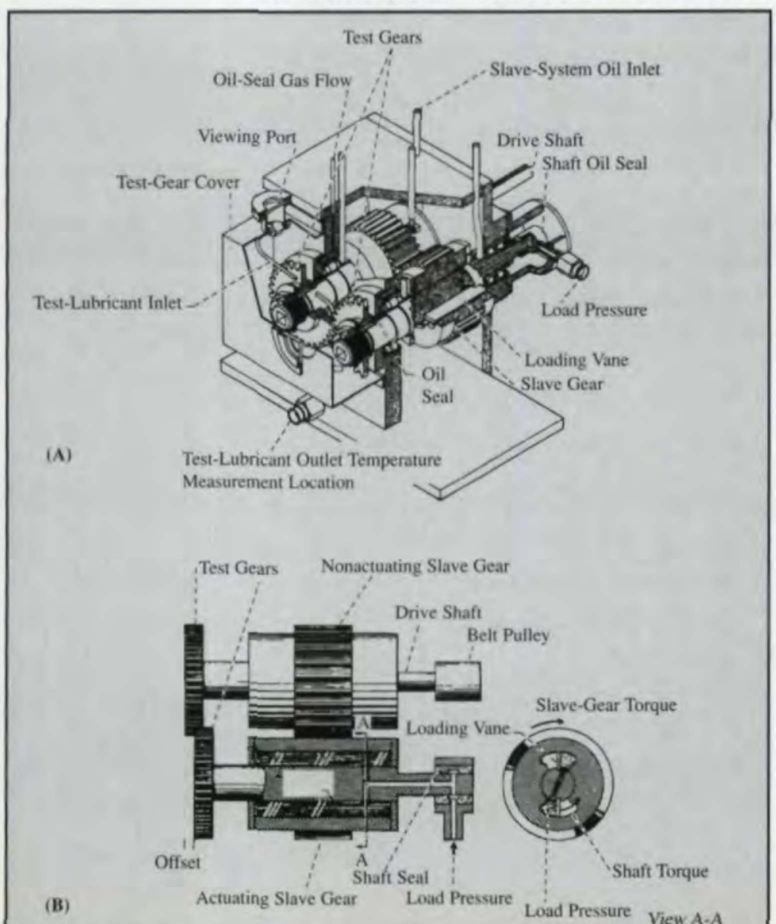

Fig. 1 — NASA Lewis Research Center's gear fatigue test apparatus.

the gear fatigue life with six of the seven lubricants to a reference lubricant and (3) to determine the effects of lubricant viscosity and specific EHL film thickness on the surface fatigue life of carburized and hardened spur gears.

To accomplish these objectives, one lot of spur gears was manufactured from a single heat of consumable-electrode, vacuum-melted (CVM) AISI 9310 material. The gears were case-carburized, hardened and ground to the same specifications. The pitch diameter was 8.89 cm (3.5"). The lot was divided into seven groups, each of which was tested with a different lubricant. All the test lubricants were synthetic polyolester with different viscosity properties and additives. Test conditions included a bulk gear temperature of 350 K (170°F), a pitch line maximum Hertz stress of 1.71 GPa (248 ksi) and a speed of 10,000 rpm.

Apparatus and Procedure

Gear Test Apparatus. The gear surface fatigue tests were performed in the NASA Lewis Research Center's gear test apparatus (Fig. 1A). This test rig uses the four-square principle of applying the test gear load so that the input drive only needs to overcome 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 load vanes through a shaft seal. As the oil pressure is increased on the load vanes inside the slave gear, the loop torque is applied. 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 contact or Hertz 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 pressurized labyrinth seals. Nitrogen is the seal gas. The test gear lubricant is filtered through a 5-µm-nominal fiberglass filter. The test lubricant can be heated electrically with an immersion heater. The temperature of the heater is controlled to prevent overheating the test lubricant. A water cooler and temperature controller are also provided in the test oil and slave oil system to control the inlet oil temperature.

A vibration transducer mounted on the gearbox is used to automatically shut off the test rig when a gear surface fatigue or tooth fracture 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.

Table 1 — Nominal Chemical Composition of AISI 9310 Gear Materials	
Element	AISI 9310
Carbon	0.1
Nickel	3.22
Chromium	1.21
Molybdenum	0.12
Copper	0.13
Manganese	0.63
Silicon	0.27
Sulfur	0.005
Phosphorous	0.005
Iron	Balance

Table 2 — Heat Treatment for AISI 9310 Gears				
		Temperature		
Step	Process	K	°F	Time, hr
1	Preheat in air	–	–	–
2	Carburize	1,172	1,650	8
3	Air cool to room temperature	–	–	–
4	Copper plate all over	–	–	–
5	Reheat	922	1,200	2.5
6	Air cool to room temperature	–	–	–
7	Austenitize	1,117	1,550	2.5
8	Oil quench	–	–	–
9	Subzero cool	180	-120	3.5
10	Double temper	450	350	2 each
11	Finish grind	–	–	–
12	Stress relieve	450	350	2

The operating speed for the test was 10,000 rpm. The four test rigs are operated 24 hours a day, seven days a week to provide the large number of test cycles required for surface fatigue testing.

Test Gears. Dimensions for the test gears are given in Table 3. The gear pitch diameter was 8.89 cm (3.5"). All gears have a nominal surface finish on the tooth face of 0.406 µm (16 µin), rms, and a standard 20° involute profile with tip relief. Tip relief was 0.0013 cm (0.0005"), starting at the highest point of single tooth contact.

The test gears were manufactured from (CVM) AISI 9310 steel from the same heat of material. The nominal chemical composition of the material is given in Table 1. All sets of gears were case-carburized and heat treated in accordance with the heat treatment schedule of Table 2. Fig. 2 is a photomicrograph of an etched and polished gear tooth showing the case and core microstructure of the AISI 9310 material. This material has a case hardness of Rockwell C60 and a case depth of 0.97 mm (0.038"). The nominal core hardness was Rockwell C38.

Test Lubricant. Seven lubricants were selected for surface fatigue endurance tests with the CVM AISI 9310 steel gear test specimens. Lubricant A is an unformulated base stock lubricant with no

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additives and a viscosity between the MIL-L-7808J and MIL-L-23699 specifications, but it does not meet either specification. Lubricant A was used as the reference to compare the results with the other lubricants.

Lubricant B is a 5 cSt lubricant meeting the MIL-L-23699 specification. This lubricant had a small amount of a boundary lubrication additive to provide some boundary lubricating film. Lubricant C meets the MIL-L7808J specification and had the lowest viscosity of all the lubricants tested and also had a proprietary additive package. Lubricant D was developed for helicopter gearboxes under the specification DOD-L85734 and was also a 5 cSt lubricant with an anti-wear additive package. Lubricant E was a 7.5 cSt lubricant with an anti-wear additive package meeting a special development specification DERD-2487. Lubricants F and G were 9 cSt ester-based lubricants. Lubricant F was a base stock lubricant without additives, while G was a base stock industrial grade. Six of the seven could be classified as synthetic polyolester base stock lubricants, while E is a polyalkylene-glycol with a small amount of boundary lubrication additive.

The pitch line (EHL) film thickness was calculated by the method of Reference 8. The temperature used in the film thickness calculation was the gear surface temperature at the pitch line, which was assumed to be equal to the oil outlet temperature, even though the temperature of the oil jet lubricating the gear was much lower. Probably the gear surface temperature was higher than the oil outlet temperature based on temperature measurements made in Reference 5.

Table 4 shows the computed EHL film thicknesses and the initial Λ ratios (film thickness divided by composite surface roughness, h/σ) at the 1.71 GSPa (248 ksi) pitch line maximum Hertz stress.

Test Procedure. After the test gears were cleaned to remove the preservative, they were assembled on the test rig. The 0.635-cm (0.25") wide test gears were run in an offset condition with a 0.30-cm (0.12") tooth-surface overlap to give a load surface on the gear face of 0.28 cm (0.11"), thereby allowing for the edge radius of 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 pitch-line load of 1225 N/cm (700 lb/in.) for one hour, which gave a maximum Hertz stress of 0.756 GPA (111 ksi). The load was then increased to 5784 N/cm (3305 lb/in.), which gave a pitch-line maximum Hertz stress of 1.71 GPa (248 ksi) if plain bending is assumed. However, because there was an offset load, an additional stress was imposed on the tooth bending stress. Combining the bending and torsional moments gave a maximum stress of 0.26 GPa (37 ksi). This does not include the effects of tip relief, which would also increase the bending stress.

Operating the test gears at 10,000 rpm gave a pitch-line velocity of 46.55 m/s (9163 ft/min). Lubricant was supplied to the inlet mesh at 800 cm^3/min (0.21 gpm) at 321 K (120°F). The tests ran continuously until they were automatically shut down by the vibration detection transducer. The lubricant circulated through a 5 μm fiberglass filter to remove wear particles. After each test the lubricant and the filter element were discarded. Inlet and outlet oil temperatures were continuously recorded on a strip-chart recorder.

The pitch-line EHL film thickness was calculated by the method of Reference 8. It was assumed for this film thickness calculation that the gear temperature at the pitch line was equal to the outlet oil temperature, even though the inlet oil temperature was considerably lower. Possibly the gear surface temperature was even higher than the outlet oil temperature, especially at the end points of sliding contact. The EHL film thickness and the initial ratio of film thickness to composite surface roughness, Λ, was computed at the 1.7 GPa (248 ksi) pitch-line maximum Hertz stress. The values are shown in Table 4.

Table 3 — Spur Gear Data
[Gear tolerance per AGMA class 12]

Number of teeth	28
Diametral pitch	8
Circular pitch, cm (in.)	0.9975 (0.3297)
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)
Tooth width, cm (in.)	0.635 (0.25)
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)

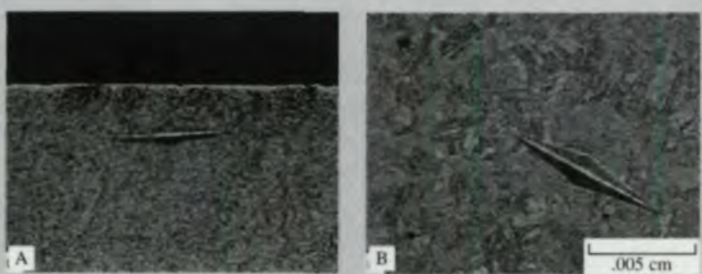

Fig. 2 — Photomicrographs of case and core of CVM AISI 9310 spur gears. (A) Case. (B) Core.

Results and Discussion

Gear Life. The surface pitting fatigue lives of the AISI 9310 gears run with all seven lubricants are shown in Fig. 3 and Table 5. These data are shown on Weibull coordinates and were analyzed by the method of Reference 9. The life shown is the life of gear pairs in millions of stress cycles or millions of revolutions. A failure is defined as one or more spalls covering more than 50% of the width of the Hertzian contact. Typical fatigue spalls for each lubricant are shown in Fig. 4.

Lubricant A (Fig. 3A) is the reference oil for these tests. The 10% and 50% system (two gears) lives (the life at 90% and 50% probability of survival) were 5.1 million and 20.4 million revolutions or stress cycles, respectively. The failure index was 30 out of 30, which is the number of failures out of the number of tests conducted. A typical fatigue spall with Lubricant A is shown in Fig. 4A.

The surface pitting fatigue lives of the AISI 9310 gears run with Lubricant B are shown in Fig. 3A. A typical fatigue spall for B is shown in Fig. 4B. A 5 cSt lubricant, B meets the MIL-L-23699 specification. The 10% and 50% system lives of the gears run with it were 12.1 million and 75 million revolutions or stress cycles, respectively. The failure index was 20 out of 20. These data indicate that the fatigue life of 9310 gears run with B is approximately 2.4 times that for A. The confidence number for the life difference between B and A was 84%, a statistically significant difference. The confidence number indicates the percentage of time the order of the test results will be the same. For a confidence number of 84%, 84 out of 100 times the test is repeated, the gear life with B will be higher than with A. Experience has shown that a confidence number of 80% or greater would indicate a meaningful life difference.

The life difference between A and B of over two to one would not be expected based on the small difference in viscosity and specific film thickness. However, given that A does not have an additive package including an EP additive, the life difference is more in line with expected results based on the test conducted in References 3 and 4.

The surface pitting fatigue lives obtained with Lubricant C are shown in Fig. 3B. A 3 cSt lubricant, C meets the MIL-L-7808J specification. A typical fatigue spall is shown in Fig. 4C. The 10% and 50% system lives of the 9310 gears run with C were 5.67 million and 20.7 million revolutions

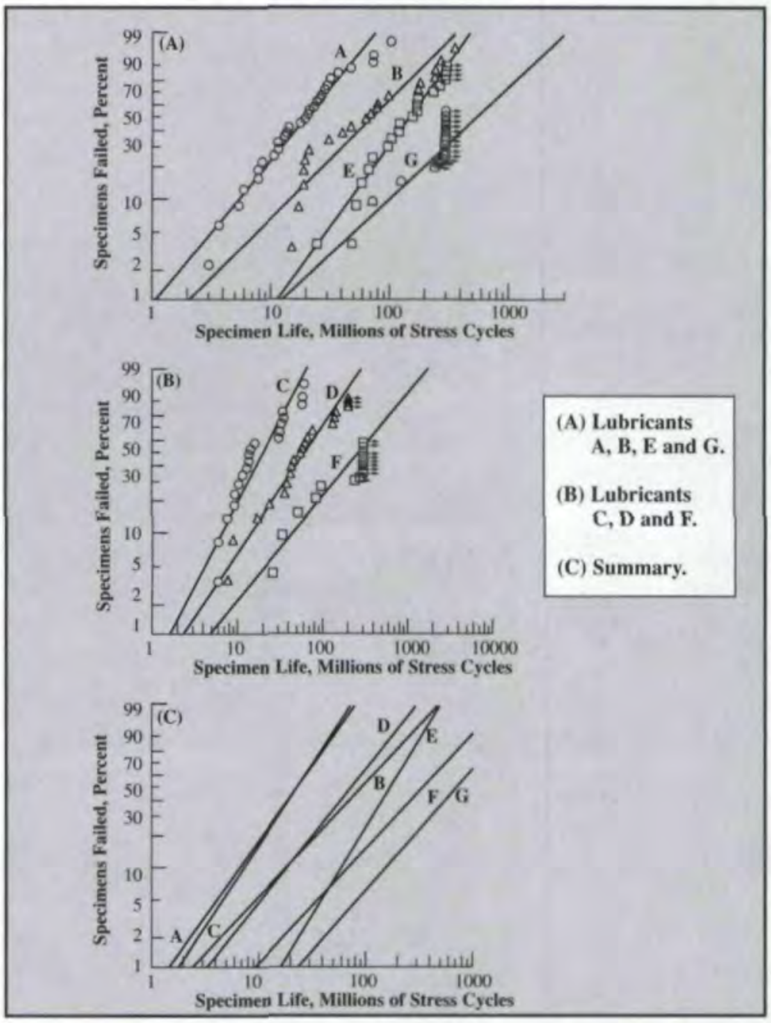

Fig. 3 — Surface pitting fatigue lives of AISI 9310 gears run with seven different lubricants. Pitch diameter 8.39 cm (3.5"); speed 10,000 rpm; maximum Hertz stress 1.71 GPa (248 ksi); gear temperature 350 K (170°F).

Table 4 — Lubricant Properties

NASA Identification	Lubricant						
	A	B	C	D	E	F	G
Kinematic Viscosity 331 K (100°F) 372 K (210°F)	21.0 4.31	29.7 5.39	12.2 3.2	27.6 5.18	34.7 7.37	60.54 8.84	52.4 8.98
Flash Point, K (°F) Pour Point, K (°F)	516 (470) 200 (-100)	539 (510) 217 (-70)	489 (420) —	544 (520) 211 (-80)	519 (475) 214 (-75)	519 (475) 228 (-49)	561 (550) 213 (-76)
Specific Gravity at 289 K (60°F)	1.00	1.00	—	0.995	0.947	0.96	0.986
Total Acid Number (tan) Mg Koh/g oil	0.07	0.03	0.15	0.40	0.06	0.00	1.01
EHL Film Thickness h mm (min) Λ ratio (h/σ)	0.43 (17) 0.75	0.52 (20) 0.90	0.34 (13) 0.58	0.50 (20) 0.87	0.66 (26) 1.15	0.76 (30) 1.33	0.76 (30) 1.33
Specification	none b.stock	MIL-L- 23699	MIL-L 7808J	DOD-L- 85734	DERD- 2487	none	none

or stress cycles, respectively. The failure index was 20 out of 20. These data indicate that the fatigue life of 9310 gears run with C was nearly equivalent to that with A. The confidence number for the life difference between C and A was 55%. The gear life with C would not be expected to equal the gear life with A based on the lubricant viscosity alone. However, C is a formulation that contained some EP additives, while A is a base stock lubricant without EP additives. Since the tests with both A and C were run with specific film thickness in the mixed or boundary regime, the EP additives in C would improve the gear life over that for A. This points out the need for EP additives in lubricants used for gears operating with specific film thicknesses less than one, as demonstrated in other tests.

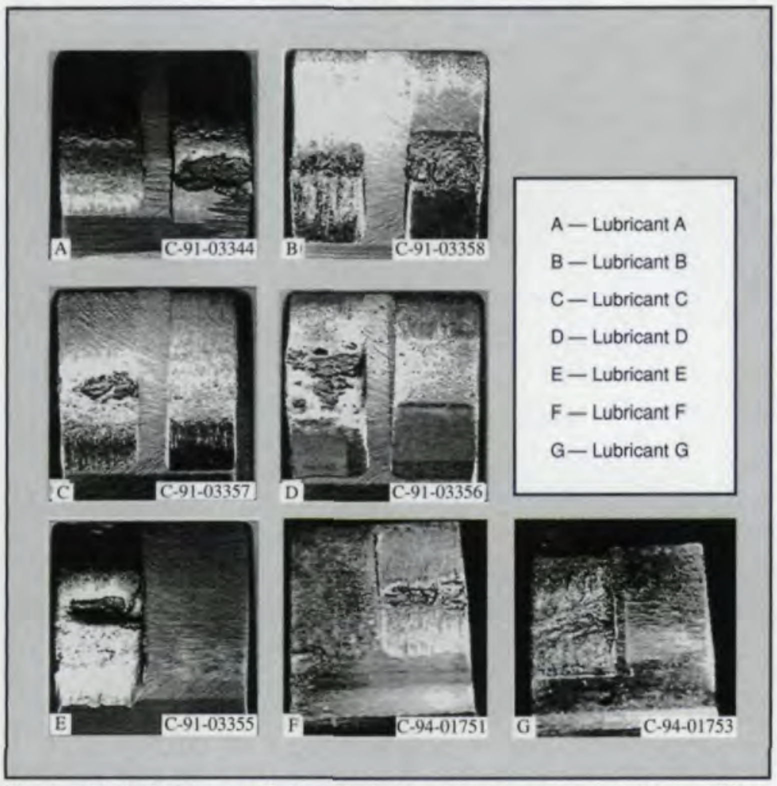

Fig. 4 — Typical fatigue spalls for AISI 9310 steel spur gears run with seven different lubricants. Pitch diameter 8.39 cm (3.5"); speed 10,000 rpm; maximum Hertz stress 1.71 GPa (248 ksi); gear temperature 350 K (170°F).

Table 5 — Surface Pitting Fatigue Lives of AISI 9310 with Different Lubricants

Lubricant Code	Lubricant Basestock	Gear System Life, millions of stress cycles		Weibull Slope	Failure Index[a]	Confidence Index[b] Percent
NASA		10 percent	50 percent			
A	polyolester	5.1	20.4	1.36	30 of 30	—
B	polyolester	12.1	76	1.02	20 of 20	84
C	polyolester	5.7	20.7	1.46	20 of 20	55
D	polyolester	11.8	50.8	1.29	17 of 20	83
E	polyalkylene-glycol	46.5	152	1.59	15 of 19	99
F	polyolester	45.2	276	1.04	7 of 17	99
G	polyolester	103	568	1.1	5 of 18	99

a Number of failures out of number of tests.
b Percent of time that 10% life obtained with each lubricant will have the same relation to the 10% life of Lubricant NASA A.

The gear surface pitting fatigue lives obtained with Lubricant D are shown in Fig 3B. The 10% and 50% system lives of the 9310 gears tests with D were 11.75 million and 50.8 million revolutions or stress cycles, respectively. The failure index was 17 out of 20, and there were three suspended tests that completed 300 million stress cycles without failure. A typical fatigue spall for D is shown in Fig. 4D. The life for D was 2.3 times that for A and was nearly identical to the life for B. Lubricant D has more boundary additive and nearly the same viscosity as B, which could provide better surface fatigue life. However, both B and D had nearly identical fatigue lives. Lubricant A, on the other hand, has only slightly less viscosity than D, but does not have an additive package or EP additive, which is the most probable reason for the shorter life of A. The confidence number for the life difference between D and A was a statistically significant 83%.

The gear surface pitting fatigue lives obtained with Lubricant E are plotted on Weibull coordinates shown in Fig. 3A. A typical fatigue spall for E is shown in Fig. 4E. The 10% and 50% system lives of the 9310 spur gears tested with E were 46.5 million and 152 million stress cycles or revolutions, respectively. The failure index for E was 15 out of 19, with four tests that were suspended after 500 hours or 300 million stress cycles without failure. The confidence number for the life difference between E and A was 99%.

The gear pitting life obtained with Lubricant F is shown in Fig. 3B. The 10% and 50% system lives of the AISI 9310 gears for this lubricant were 45 million and 276 million stress cycles respectively. The failure index was 7 out of 17. Ten tests completed 300 million cycles without failure. A typical fatigue spall for F is shown in Fig. 4F. The 10% surface fatigue life for F was nine times that for A, a 5 cSt lubricant, and about equal to that for E, a 7.5 cSt lubricant. The main reason that F did not produce a better gear surface fatigue life than E appears to be that F is a base stock lubricant without an anti-wear additive package, while E contains a good additive package. The confidence number for F compared to A was 99%. The confidence number for F compared to E was only 50%, which means the lives were approximately equal.

The gear pitting life obtained with Lubricant G is shown in Fig. 3A. The 10% and 50% system lives of the AISI 9310 gears were 103 million and 568 million stress cycles respectively. The failure index was 5 out of 18, which means that there were 13 tests that completed 300 million cycles without failure. A typical fatigue spall for G is shown in Fig. 4G. The 10% surface fatigue life for

G was 20 times that for A, a 5 cSt lubricant, and about 2.3 times that for E, a 7.5 cSt lubricant and F, a 9 cSt lubricant. It appears that the main reason for the life improvement of G over F, which has the same viscosity, is the fact that F has no antiwear additive package, while G has some additive that provided boundary lubrication. The confidence number for G compared to A was 99%, which means that the life difference was statistically significant. The confidence number for G compared to F was 70%, which means the life difference was also statistically significant.

The life results are summarized in Fig. 3C and Table 5. The life of Lubricant G was more than twenty times that for A and more than two times that for E and F. A 9cSt lubricant, G had a calculated specific film thickness Λ of 1.33. It was, therefore, expected that it would produce longer fatigue life than the other, less viscous lubricants. However, it could not be analytically determined just how much improvement in surface fatigue could be obtained with this higher viscosity lubricant. The surface fatigue testing was, therefore, necessary to determine gear life with these lubricants.

Fig. 5 is a plot of specific film thickness ratio Λ versus the relative gear surface fatigue life and shows how the gear life is affected by the specific film thickness ratio. The test results are conclusive in showing that when gears are operated with lubricants that provide specific film thickness around one or greater, the surface fatigue will show large improvements over some of the turbine engine lubricants that provide lower EHL specific film thickness. In addition, as the EHL specific film thickness ratio increases above 1, the surface fatigue is further improved. The above results also point to the need to provide separate lubricants with higher viscosities than the engine lubricants for power transmissions such as turboprop or turbofan reduction gearboxes and helicopter gearboxes to provide increased life and reliability of these systems.

Conclusions

The following results were obtained:

1. Lubricants with a viscosity providing a specific film thickness greater than one and with an additive package produced surface fatigue lives 4 to 8.6 times those of lubricants with a viscosity providing a specific film thickness less than one.

2. As the lubricant viscosity is increased to give EHL specific film thickness ratios Λ well above 1, the gear surface fatigue life is further improved.

3. A low viscosity lubricant with an additive package produced surface fatigue lives equivalent to similar base stock lubricant with 30% high viscosity, but without an additive package.

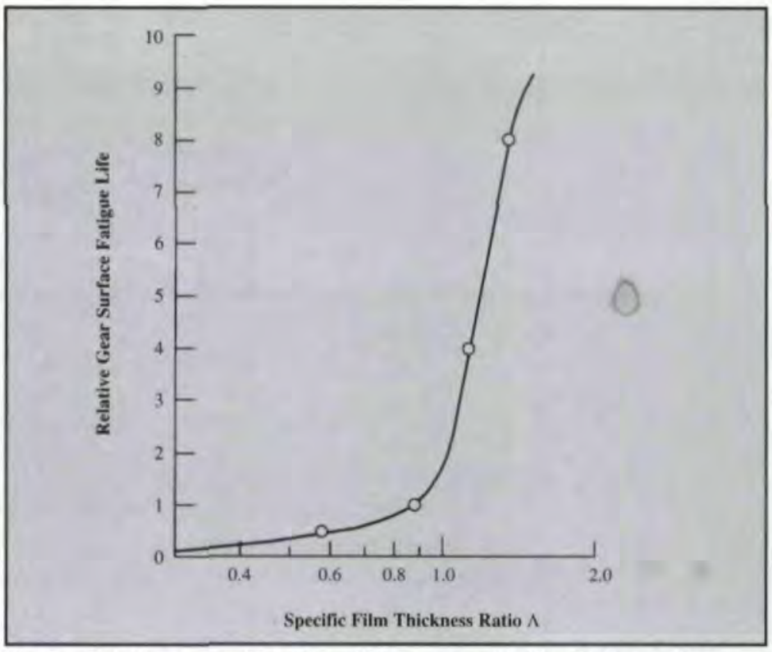

Fig. 5 — Relative gear surface fatigue life versus specific film thickness ratio Λ. Test conditions: 10,000 rpm, maximum Hertz stress 1.71 GPa (248 ksi), and a bulk gear temperature of 350 K (170°F).

4. Lubricants with the same viscosity and similar additive packages gave equivalent gear surface fatigue lives.

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