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Scientific and Technical
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Summary

Spur-gear surface fatigue tests were conducted with five lubricants using a single lot of consumable-electrode vacuum melted (CVM) AISI 9310 spur gears. The gears were case carburized and hardened to Rockwell C 60. The gear pitch diameter was 8.89 cm (3.5 in). The lot of gears was divided into five groups, each of which was tested with a different lubricant. The test lubricants can be classified as either a synthetic hydrocarbon, mineral oil, or ester based. All five lubricants have similar viscosity and pressure-viscosity coefficients. Test conditions included a bulk gear temperature of 350 K (170 °F), a maximum Hertz stress of 1.71 GPa (248 000 psi) at the pitch line, and a speed of 10 000 rpm. A pentaerythritol-base stock without sufficient antiwear additives produced a 10-percent surface fatigue life that was approximately 22 percent that of a pentaerythritol base stock of the same viscosity with chlorine and phosphorus type additive. The presence of a sulfur type antiwear additive in the lubricant did not appear to affect the surface fatigue life of spur gears at the conditions tested. No statistical difference in the 10-percent surface fatigue life was produced, with four of the five lubricants having similar viscosity and pressure-viscosity coefficients and various antiwear additives.

Introduction

Gear failure by surface pitting (rolling-element) fatigue is affected by the physical and chemical properties of the lubricant. Knowledge of how these chemical and physical properties affect rolling-element fatigue is a useful guide both in selecting existing lubricants for mechanical power transmission applications and in developing new lubricant formulations. For helicopter and turboprop transmission applications, it is important to know the effect of these lubricants on bearing and gear life and reliability.

The NASA Lewis fatigue spin rig and five-ball fatigue tester were used to determine the rolling-element fatigue lives at room temperature and at 422 K (300 °F) of groups of AISI M-2 and AISI M-1 steel balls run with nine lubricants having varied chemical and physical characteristics (refs. 1 to 3). These lubricants were classified as three basic types: esters, mineral oils, and silicones. At room temperature longer fatigue lives were obtained with the silicone and one of the esters, the dioctyl sebacates, than with the other lubricants. At 422 K (300 °F), however, the silicone (ref. 1) and the mineral oils gave the longer lives. Similar results were obtained with gears (ref. 4). The relative order of fatigue results obtained in the five-ball fatigue tester agreed with that obtained with 7208-size ball bearings (ref. 1) tested with the same lubricants. In the five-ball fatigue tester at 422 K (300 °F), the silicone induced such a high wear rate in the ball specimens because of ball spinning that long-term fatigue testing was precluded. The difference in life between these lubricants can be attributed to the elastohydrodynamic (EHD) film forming properties of the lubricants (refs. 2 and 3). For gears, which generally run at EHD film thicknesses lower than bearings, limited life data exists (refs. 4 to 6).

Lubricant additives are necessary to the operation of gear systems. These additives can prevent or minimize wear and surface damage to bearings and gears whose load-carrying surfaces operate under very thin film or boundary lubrication conditions (refs. 7 and 8). These antiwear or extreme pressure (EP) additives either adsorb onto the surfaces or react with the surfaces to form protective coatings or surface films. Lubricants essentially having the same base stock and viscosity characteristics and meeting the same specification can have significantly different additive packages. Some additives may alter the lubricant rheology of the base oil (viscosity or pressure-viscosity effects or chemical effects on the gear surface). As a result, additives may influence gear pitting (rolling-element fatigue) life (refs. 3 and 9 to 11).

The objective of the work reported herein was to determine the effect of five lubricants on the surface pitting (rolling-element fatigue) life of consumable-electrode vacuum melted (CVM) AISI 9310 spur gears. To accomplish the objective, one lot of spur gears was manufactured from a single heat of CVM AISI 9310 material. The gears were all case carburized, hardened, and ground to the same specifications. The gear pitch diameter was 8.89 cm (3.5 in). The lot of gears was divided into five groups, each of which was tested with a different test lubricant. These test lubricants can be classified as synthetic hydrocarbon, mineral oil, or ester-based (ref. 12). Test conditions included a bulk gear temperature of 350 K (170 °F), a maximum Hertz stress of 1.71 GPa (248 000 psi) at the pitch-line, and a speed of 10 000 rpm.

Apparatus, Specimens, and Procedure

Gear Test Apparatus

The gear fatigue tests were performed in the NASA Lewis Research Center's gear test apparatus. The test rig is shown in figure 1 and described in reference 13. 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 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.

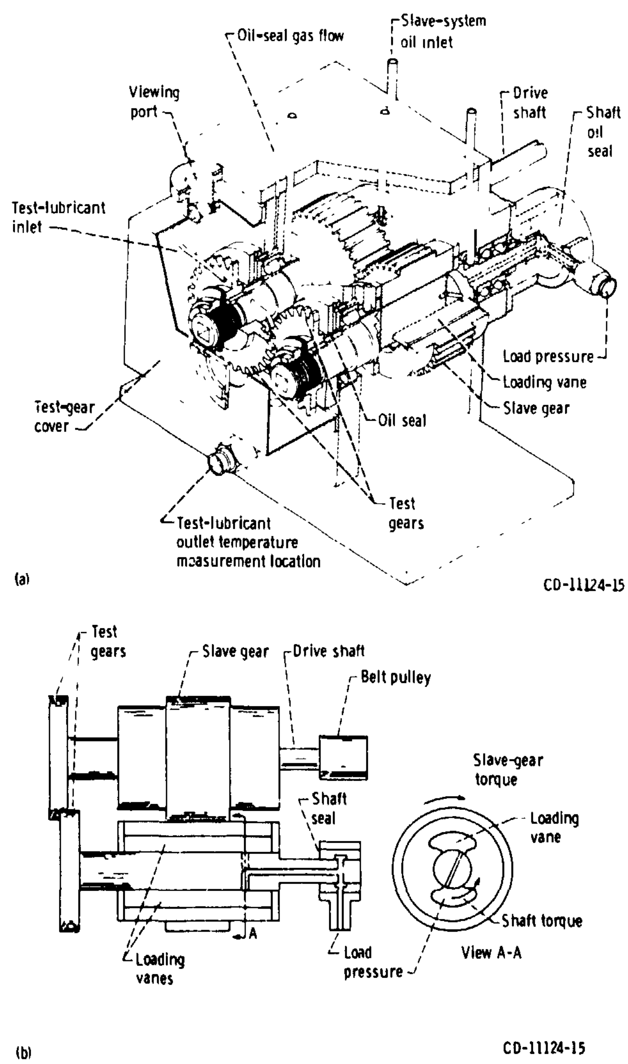


Figure 1.—NASA Lewis Research Center's gear fatigue test apparatus.

Separate lubrication systems are provided for the test gears and the main gearbox. The two lubricant 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 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 a gear-surface fatigue spall 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 Materials

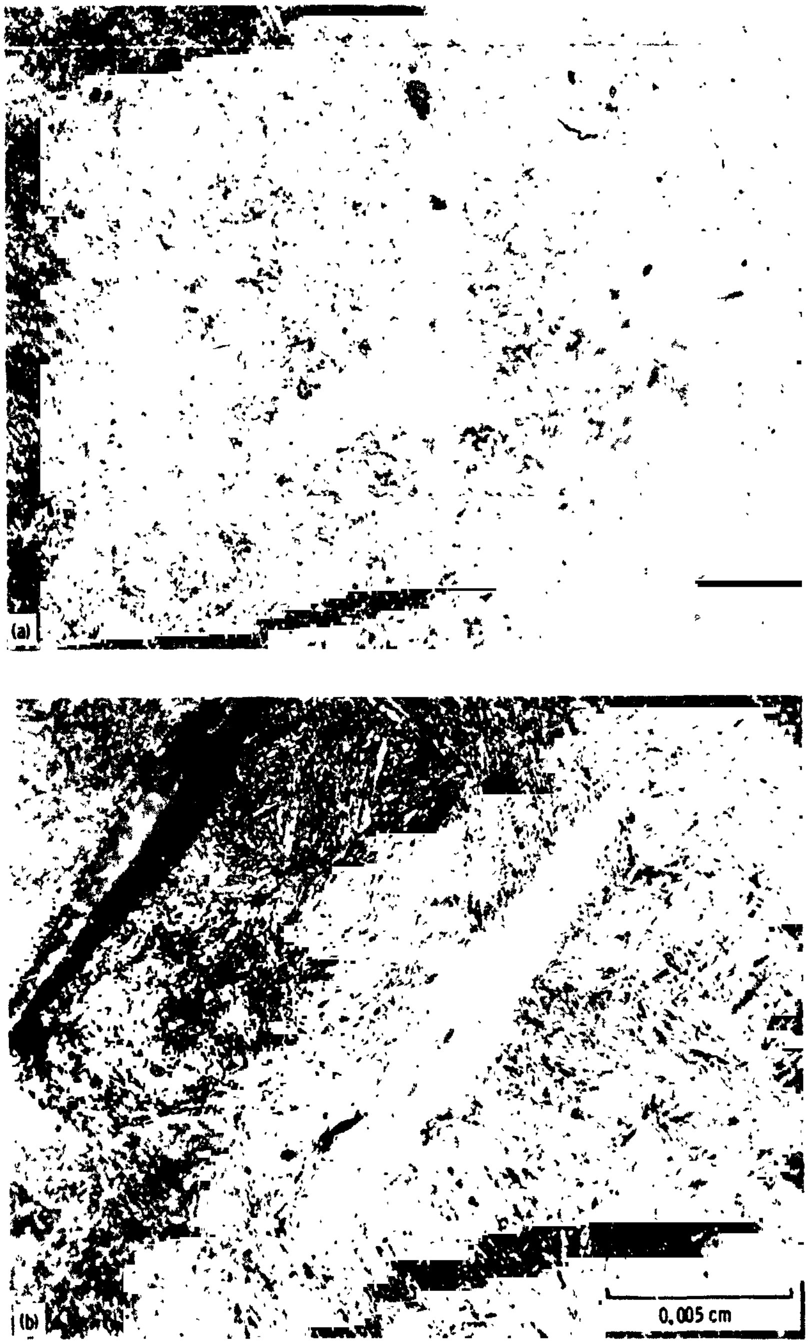
The test gears were manufactured from consumable-electrode vacuum-melted (CVM) AISI 9310 steel from the same heat of material. The nominal chemical composition of the material is given in table I. All sets of gears were case carburized and heat treated in accordance with the heat treatment schedule of table II. Figure 2 is a photomicrograph of an etched and polished gear tooth surface showing the case and core microstructure of the

TABLE I.—NOMINAL CHEMICAL COMPOSITION OF AISI 9310 GEAR MATERIAL

C	Mn	Si	Ni	Cr	Mo	Cu	P	S
Composition, wt %								
0.10	0.63	0.27	3.22	1.21	0.12	0.13	0.005	0.005

TABLE II.—HEAT TREATMENT FOR AISI 9310

Step	Process	Temperature, K	Temperature, °F	Time, hr
1	Preheat in air	----	----	-----
2	Carburize	1172	1650	8
3	Air cool to room temperature	----	----	-----
4	Copper plate all over	----	----	-----
5	Reheat	922	1200	2.5
6	Air cool to room temperature	----	----	-----
7	Austenitize	1117	1550	2.5
8	Oil quench	----	----	-----
9	Subzero cool	180	− 120	3.5
10	Double temper	450	350	2 each
11	Finish grind	----	----	-----
12	Stress relive	450	350	2



(a) Case.

(b) Core.

Figure 2.—Photomicrographs of case and core of CVM AISI 9310 spur gears.

AISI 9310 material. This material had a case hardness of Rockwell C 58 and a case depth of 0.97 mm (0.038 in). The nominal core hardness was Rockwell C 40.

Test Gears

Dimensions for the test gears are given in table III. 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 in), starting at the highest point of single tooth contact.

Test Procedure

After the test gears were cleaned to remove the preservative, they were assembled on the test rig. The test gears

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 diamter, 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)	8.549 (0.216)
Backlash reference, cm (in).	0.0254 (0.010)
Tip relief, cm (in)	0.001 to 0.0015 (0.0004 to 0.0006)
Surface finish, μm (μin)	0.406 (16)

were run in an offset condition with a 0.30 cm (0.120-in) tooth-surface overlap to give a load surface on the gear face of 0.28 cm (0.110 in), 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 pair of gears. All tests were run-in at a load of 1225 N/cm (700 lb/in) for 1 hr. The load was then increased to 5784 N/cm (3305 lb/in), which gives a 1.71 GPa (248 000 psi) pitch-line maximum Hertz stress. At the pitch-line load the tooth bending stress was 0.21 GPa (30 000 psi) if plain bending is assumed. However, because of the offset load, an additional stress is imposed on the tooth bending stress. Combining the bending and torsional moments gives a maximum stress of 0.26 GPa (37 000 psi). This bending stress does not include the effects of tip relief which would also increase the bending stress. In general, 20 tests were run for each lubricant.

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^2/min at 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 they were automatically shut down by the vibration detection transducer, located on the gearbox adjacent to the test gears. The lubricant circulated through a 5-μm fiberglass filter to remove wear particles. After each test the lubricant and filter element were discarded. Inlet and outlet oil temperatures were continuously recorded on a strip-chart recorder. After each test the system was partially disassembled, flushed with trichloroethane and then with alcohol, dried, and reassembled before a new lubricant was used in the system.

Test Lubricants

Five lubricants were selected for endurance tests with the AISI 9310 gear test specimens. These lubricants either meet the MIL-L-23699 specification or are being used as gear or transmission lubricants. They can be classified as three basic types: synthetic hydrocarbon, mineral oil, or ester-based lubricants. Tests were conducted on these lubricants (ref. 12) to determine their physical and chemical properties. A summary of the properties of these lubricants is given in table IV. The lubricant designations are cross referenced between those of the NASA and the U.S. Army Fuels and Lubricants Research Laboratory (ref. 12) in the table. The additives contained in these oils are proprietary to their respective manufacturers except where indicated. However, it is expected that each of the oils would have antiwear and extreme-pressure (EP) additives as well as oxidation and rust inhibitors.

The lubricant F, synthetic paraffinic oil, is the standard lubricant used by the authors in their gear test facility (ref. 17). It is commercially available with an oxidation inhibitor. An EP additive package was added to the as-received oil. This additive package and the amount added is given in table IV.

Lubricant A is a common automotive automatic transmission fluid which is being used by some commercial helicopter users in the main rotor gearbox in place of MIL-L-7808 or MIL-L-23699 specification lubricants. The lubricant has been advocated for use by the military in place of those with the above military lubricant specifications. However, the oil does not meet all the MIL-L-23699 specifications for engine oil.

The lubricants C and K are from the same manufacturer and meet the MIL-L-23699 specification. However, lubricant K may have an adverse effect on seal clearances if silicone seal compounds are used. Both oils are pentaerythritol esters with nearly the same viscosity and pressure viscosity characteristics. However, the measured wear rate (ref. 12) with lubricant C is twice that of lubricant K. This would indicate a more effective additive package in lubricant K.

Lubricant E, which is a dibasic acid ester, is a commercially available gear lubricant. It does not meet existing

TABLE IV.—LUBRICANT CHEMICAL AND PHYSICAL SUMMARY (REF. 12)

	NASA code				
	A	C	E	F	K
	AFLRL code				
	11252	11250	11256	11258	11266
	Basestock				
	Mineral oil	Polyol ester pentaerythritol	Dibasic acid ester	Synthetic parafinic	Polyol ester pentaerythritol
Carboxylic acids, percent					
C-4	----------------	Trace	----------------	----------------	Trace
C-5	----------------	46	Di-63	----------------	22
C-6	----------------	10	Di-37	----------------	16
C-7	----------------	17	----------------	----------------	24
C-8	----------------	10	----------------	----------------	8
C-9	----------------	13	----------------	----------------	29
C-10	----------------	4	----------------	----------------	1
Additives	Proprietary	Proprietary	Proprietary	Lubrizol 5002, 5 vol % phosphorus, 0.6 wt % sulphur, 18.5 wt %	Proprietary
Specification	GM 6137-M	MIL-L-23699	----------------	----------------	MIL-L-23699
Type	Automatic transmission fluid	Turbine engine oil	Synthetic gear lubricant	NASA gear test lubricant	Type II turbine engine oil
Kinematic viscosity, mm^2/sec (cS), at—					
244 K (−20 °F)	3500 (3500)	3000 (3000)	5000 (5000)	2600 (2600)	3000 (3000)
311 K (100 °F)	40 (40)	28.5 (28.5)	36 (36)	3.03 (30.3)	28.5 (28.5)
372 K (310 °F)	7.2 (7.2)	5.3 (5.3)	6.0 (6.0)	5.5 (5.5)	5.3 (5.3)
477 K (400 °F)	1.5 (1.5)	1.3 (1.3)	1.4 (1.4)	1.3 (1.3)	1.3 (1.3)
Pressure viscosity coefficient, GPa^{-1} (psi^{-1}) at—					
311 (100 °F)	15.4 (10.6×10^{-5})	11.6 (8.0×10^{-5})	15.5 (10.7×10^{-5})	13.4 (9.2×10^{-5})	11.4 (7.9×10^{-5})
372 (210 °F)	11.2 (7.7×10^{-5})	10.0 (6.9×10^{-5})	11.5 (7.9×10^{-5})	11.1 (7.7×10^{-5})	9.5 (6.5×10^{-5})
422 (300 °F)	10.2 (7.0×10^{-5})	8.8 (6.1×10^{-5})	9.9 (6.8×10^{-5})	9.5 (6.5×10^{-5})	8.3 (5.7×10^{-5})
Flash point, K (°F)	433 (320)	527 (490)	513 (465)	508 (455)	533 (500)
Fire point, K (°F)	488 (347)			533 (500)	
Pour point, K (°F)	233 (−40)	211 (−80)	219 (−65)	219 (−65)	214 (−75)
Specific gravity	0.862	1.005	0.932	0.829	0.983
Specific heat at 311 K (100 °F), J/kg K (Btu/lb °F)	546 (0.42)	572 (0.44)	884 (0.68)	676 (0.52)	585 (0.45)
Vapor pressure at 311 K (100 °F), mm Hg or torr	Unknown	Unknown	Unknown	0.1	Unknown
Relative wear rate	1.3	2.2	1.3	1.7	1
Friction coefficient	0.053	0.024	0.035	0.034	0.022
Elastohydrodynamic film thickness, h, μm, (μin)	0.523 (20.6)	0.454 (17.9)	0.515 (20.3)	0.388 (15.2)	0.411 (16.2)
Λ ratio (h/σ)	0.87	0.76	0.86	0.65	0.69

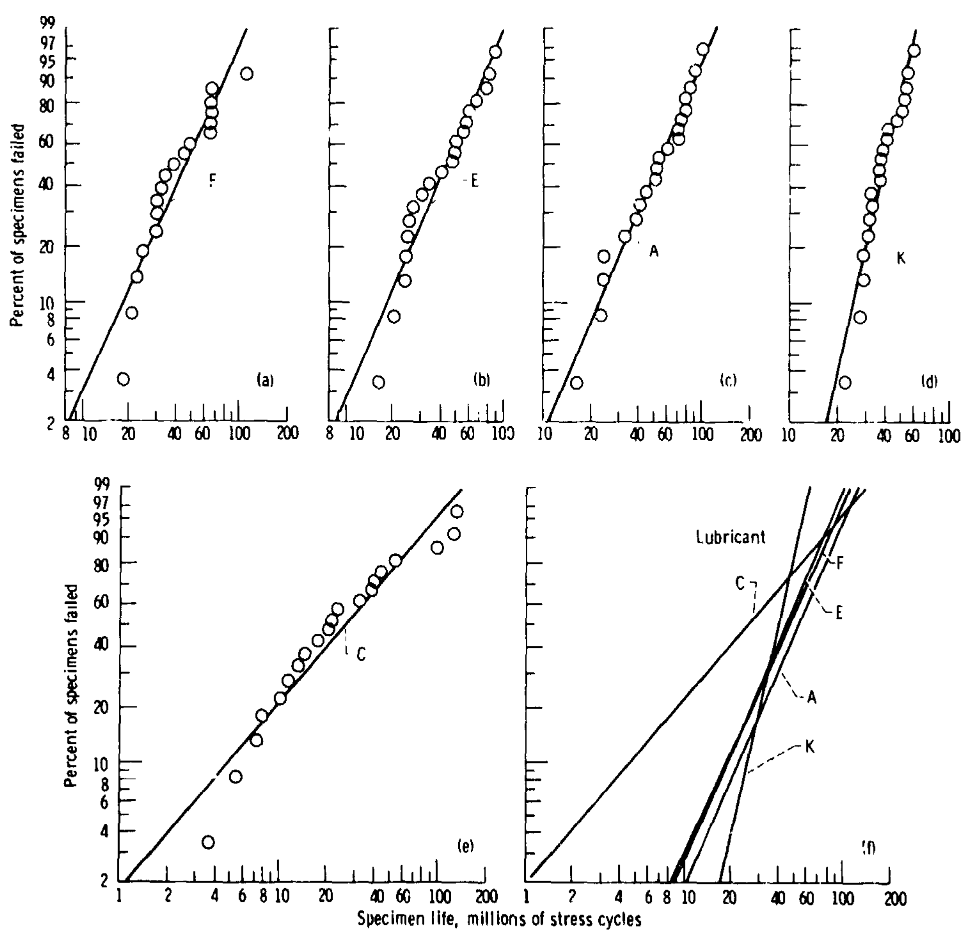
military specifications for aircraft transmission application. However, it has been recommended for use in helicopter transmissions.

The pitch-line elastohydrodynamic (EHD) film thickness was calculated by the method of references 14 and 15 and using the data of table IV for each of the lubricants. 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 possible that the gear surface temperature was even higher than the oil outlet temperature, especially at the end points of sliding contact. The computed EHD film thicknesses are given in table IV as are initial Λ ratios (film thickness divided by composite surface roughness (h/σ)) at the 1.71-GPa (248 000-psi) pitch-line maximum Hertz stress. Based on the Λ values, the gears lives obtained with each lubricant would not be expected to be significantly different except where additive effects become important (ref. 11).

Results and Discussion

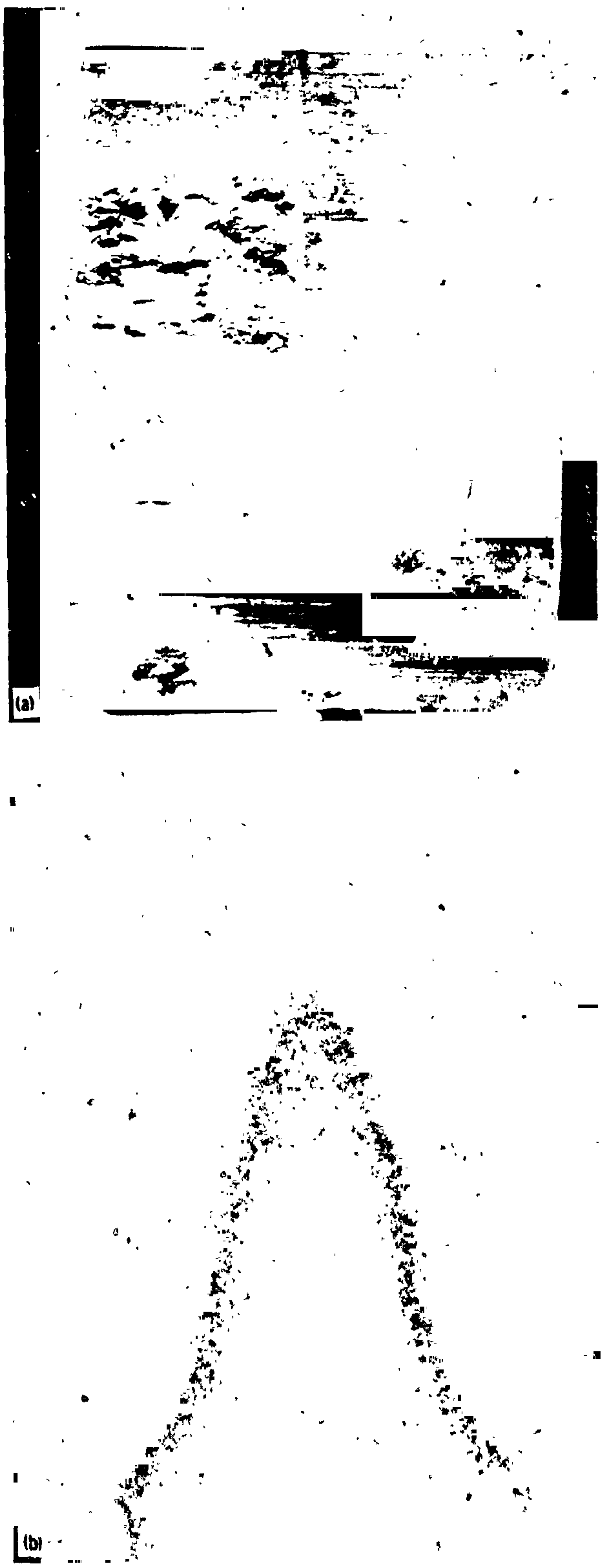
Effect of Lubricant

The gear fatigue life for each of the lubricants is shown in figure 3 and summarized in table V. The surface pitting fatigue life of the AISI 9310 gears run with lubricant F, the synthetic paraffinic oil is shown in figure 3(a). These data, which are shown on Weibull coordinates, were analyzed by the method of reference 16. The life shown is the life of gear pairs failed in millions of stress cycles. The gear teeth receive one stress cycle per revolution. A failure is defined as one or more spalls covering more than 50 percent of the width of the tooth Hertzian contact. A typical fatigue spall is shown in figure 4(a). A



(a) Lubricant F; failure index, 18 out of 19.
(b) Lubricant E; failure index, 20 out of 20.
(c) Lubricant A; failure index, 20 out of 20.
(d) Lubricant K; failure index, 18 out of 18.
(e) Lubricant C; failure index, 20 out of 20.
(f) Summary.

Figure 3.—Surface pitting fatigue life of lubricated, CVM AISI 9310 spur gears. Speed, 10 000 rpm; temperature, 350 K (170 °F); maximum Hertz stress, 1.7 mPa (248 000 psi).



(a) Typical fatigue spall.
(b) Cross section of typical fatigue spall.
Figure 4.—Fatigue spall for lubricant F.

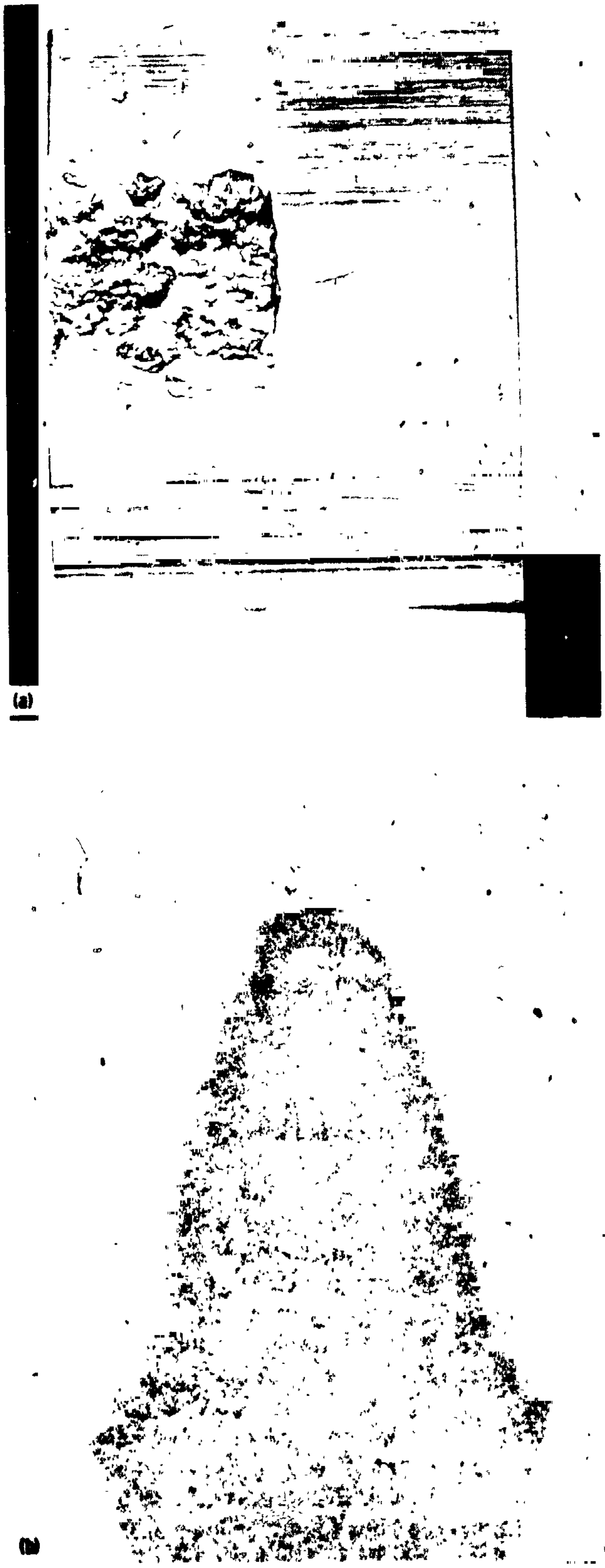
cross section of the spall is shown in figure 4(b). The failure index in figure 3 indicates the number of failures out of the number of tests run. For lubricant F, the failure index was 18 out of 19. The 10- and 50-percent system lives (these are the lives at a 90- and 50-percent probability of survival, respectively) were 18.8 million and 46.1 million revolutions or stress cycles, respectively.

The surface pitting fatigue life of the CVM AISI 9310 steel spur gears run with lubricant E, a dibasic acid ester, is shown in figure 3(b). The 10- and 50-percent system lives with this fluid were 18.8 million and 43.7 million stress cycles, respectively. This lubricant exhibited fatigue lives almost identical to lubricant F. Based on a comparison of the physical properties of the two lubricants, this result is not unexpected if chemical differences are discounted. A typical fatigue spall for lubricant E is shown in figure 5(a), a cross section of the spall is shown in figure 5(b).

The surface pitting fatigue lives obtained with lubricant A, the mineral-oil based lubricant, are shown in figure 3(c). A typical fatigue spall for lubricant A is shown in figure 6(a); a cross section through the fatigue spall is shown in figure 6(b). This lubricant produced 10-and 50-percent lives of 22.8 million and 53.7 million stress cycles, respectively. The 10-percent life is more than 20 percent greater than the life obtained for the synthetic paraffinic oil (F) and the dibasic acid ester lubricant (E). The differences in these lives, based on the confidence numbers given in table V, are not statistically significant. (The confidence number indicates the percentage of time the order of the test results would be the same. For a confidence number of 62 percent, 62 out of 100 times the test is repeated lubricant A will produce a higher life than lubricant F. Generally, a 2-σ or a 95-percent confidence is considered statistically significant. However, experience has shown that a confidence number of 80 percent or greater is necessary to draw useful conclusions regarding life differences.)

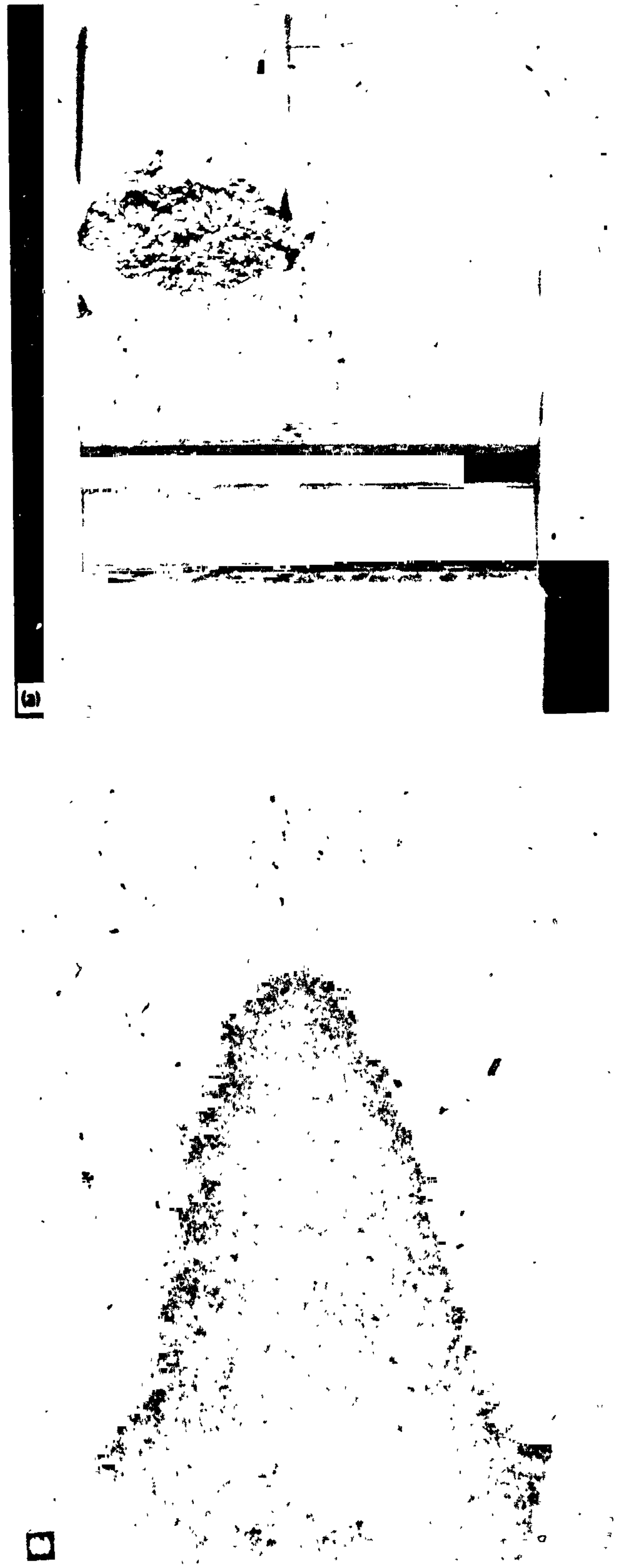
The surface pitting fatigue lives obtained with lubricant K, a pentaerythritol ester are shown in figure 3(d). The 10- and 50-percent lives were 24.7 million and 37.5 million stress cycles, respectively. While this lubricant had a higher 10-percent life than the reference oil (lubricant F), it had a lower 50-percent life. The confidence number for the 10-percent life was 72 percent, which indicates no statistical life differences between this fluid and the three previous lubricants discussed. Again, based on the physical properties alone and not on chemical differences, no statistical differences in life would be expected because the resultant elastohydrodynamic film thickness would be nearly the same for all the lubricants. A typical fatigue spall for lubricant K is shown in figure 7(a); a cross section through the spall is shown in figure 7(b).

The life results for lubricant C, which is also a pentaerythritol ester are shown in figure 3(e). The 10- and 50-percent lives with this fluid were 4.8 million and 25.9



(a) Typical fatigue spall.
(b) Cross section of typical fatigue spall.

Figure 5.—Fatigue spall for lubricant E.

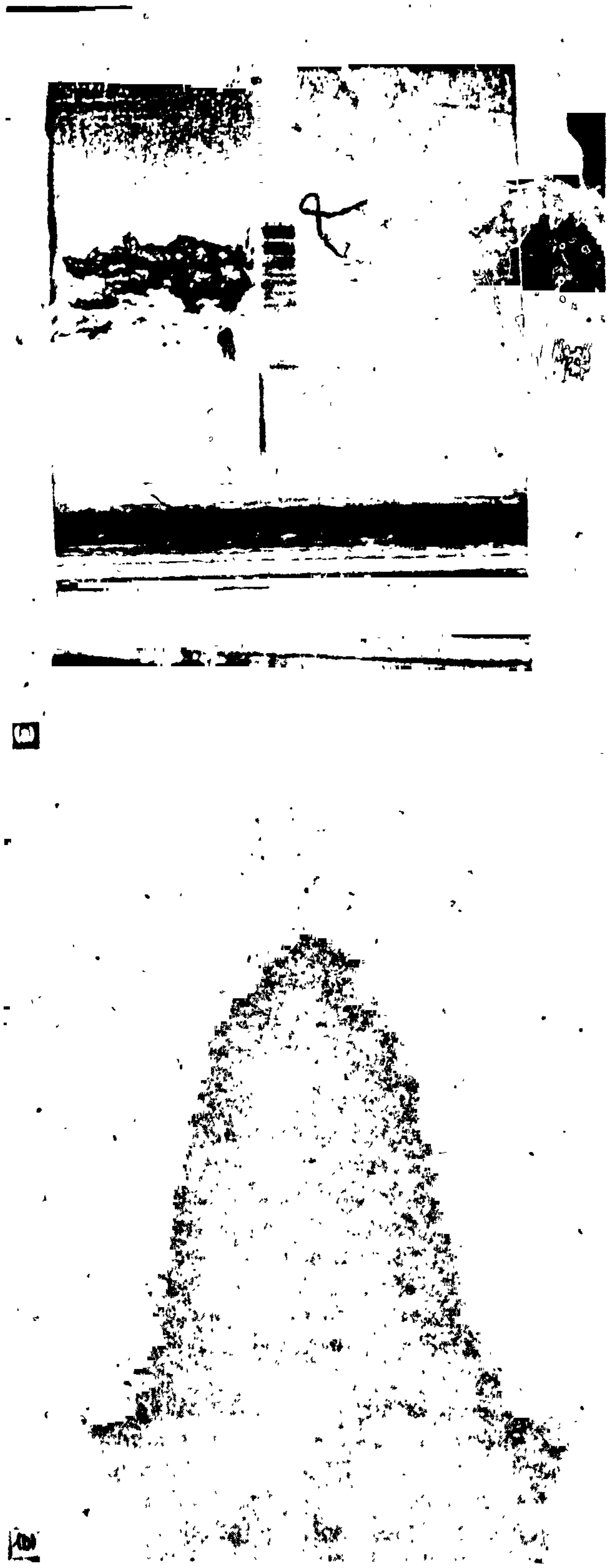


(a) Typical fatigue spall.
(b) Cross section of typical fatigue spall.

Figure 6.—Fatigue spall for lubricant A.

(a) Typical fatigue spall.
(b) Cross section of typical fatigue spall.

Figure 7.—Fatigue spall for lubricant K.



(a) Typical fatigue spall.
(b) Cross section of typical fatigue spall.

Figure 8.—Fatigue spall for lubricant C.

TABLE V.—SURFACE PITTING (ROLLING-ELEMENT) FATIGUE LIVES

[NASA spur gear test apparatus; material, CVM AISI 9310; gear bulk temperature, 350 K (170 °F); maximum hertz stress, 1 GPa (248 000 psi); speed, 10 000 rpm.]

Lubricant code		Lubricant basestock	Gear system life, millions of stress cycles		Weibull slope	Failure index[a]	Confidence number,[b] percent
NASA	AFLRL		10 percent	50 percent			
A	11252	Mineral oil	22.8	53.7	2.2	20 out of 20	62
C	11250	Pentaerythritol ester	4.8	25.9	1.1	20 out of 20	96
E	11236	Dibasic acid ester	18.8	43.7	2.2	20 out of 20	50
F	11258	Synthetic Paraffinic	18.8	46.1	2.1	18 out of 19	--
K	11266	Pentaerythritol ester	24.[illegible]	37.5	4.5	18 out of 18	72

[a]Number of failures out of number of tests

[b]Percent of time that 10 percent life obtained with each lubricant will have the same relation to the 10 percent life of lubricant NASA F

million stress cycles, respectively. This 10-percent life is approximately 20 percent of the life obtained with lubricant K, which is the same base stock with different additives. The confidence number was 96 percent, which is a 2-σ confidence. Statistically, then, the life obtained with this lubricant is significantly lower than the previous four lubricants. The life results are summarized in figure 3(f). Based on the physical characteristics of this fluid and the life results previously discussed, these results were unexpected. It is speculated that undefined chemical effects due to the lubricants additive package may have contributed to the lower lives obtained. A typical fatigue spall for lubricant C is shown in figure 5, a cross section through the fatigue spall is shown in figure 8(b).

Chemical Effects

An x-ray fluorescence (XRF) (filter method) was used to identify and measure the metals contained in the lubricants (ref. 12) before and after testing in an OH-58 helicopter transmission. With this method the wear metals and additive particulates are separated from the lubricant and subjected to energy-dispersive x-ray fluorescence analysis. This method gives a sensitivity of 0.1 ppm. The results of these measurements are shown in table VI.

For the reference lubricant (F), the synthetic paraffinic oil, the chlorine, phosphorus, and sulfur present are from the additive package. Approximately 51 ppm of sulfur was measured in the used oil, which indicates a predominance of this element in the additive package.

Lubricant A, the mineral oil, showed barium, chlorine, phosphorus, and sulfur as probably being part of the additive package. Both chlorine and sulfur showed their presence to be approximately 1.12 ppm in the used oil. The amount of sulfur is significantly less than that measured for lubricant F. The life difference between these two lubricants does not suggest any effect of the presence of sulfur on the surface pitting fatigue life of the gears.

The x-ray fluroescence analysis results for lubricant E, the dibasic acid ester, showed lesser amounts of elements associated with the additive package with the used oil than with the new oil. Specifically large reductions were indicated in chlorine, zinc, and sulfur. Sulfur appears to be the major constituent in the additive package. The presence of zinc could be due to wear when present with copper or, when present alone, to its incorporation as an additive. This additive package is more typical of reciprocating piston engine oil. Strong acid was indicated in the oil, which was probably due to free sulfonic acid from the additives. The life with this lubricant was nearly identical to that with lubricant F.

From the results for lubricant K, a pentaerythritol ester, a predominance of chlorine is indicated, which is significantly more than with the other lubricants. The other additive present appears to be phosphorus. No sulfur appears present. This lubricant produced the highest life of all the lubricants studied, although the higher life was not statistically significant. These results may contradict those rolling-element fatigue results from reference 9, which showed a chlorinated wax additive to be detrimental to life. However, the tests in reference 9 were run at Hertz stresses more than three times (5.5 mPa (800 000 psi) that used for the gears. Since the chemical effects of the additive are temperature dependent, the higher stresses used in reference 9 may account for the difference in the results.

Lubricant C, the other pentaerythritol ester studied, showed traces of both phosphorus and sulfur in much lower quantities than with the other lubricants in both the new and used samples. It produced the lowest gear life,

TABLE VI.—ANALYTICAL REPORT SYNTHETIC LUBRICANT X-RAY FLOURESCENCE ANALYSIS

Lubricant code		Element content, ppm[a]														Limit[d] of detection, ppm
NASA	AFLRL	Mg	Al	Cl	Fe	Ni	Cu	Pb	Zn[b]	P[c]	S[c]	Ca[c]	Ba[c]	Si	Mn	
A-New	-11252-	0.48	---	2.47	---	---	---	0.21	---	0.18	4.71	---	0.23	---	---	0.11
A-Used	-11253-	---	5.91	1.12	0.51	0.10	0.14	---	0.11	17	1.12	---	.12	---	---	.09
C-New	-11250-	.28	---	73	.13	---	---	---	---	26	----	---	----	--	---	.09
C-Used	-11251-	---	2.97	1.04	2.19	.21	.12	---	.15	.19	.20	---	----	---	---	.09
E-New	-11256-	.16	.19	7.57	.10	---	---	1.28	7.27	2.15	13.01	.29	10.16	-	---	.09
E-Used	-11257-	.12	1.69	1.61	.26	---	.11	---	3.71	.94	4.29	---	2.43	---	---	.09
F-New	-11258-	.31	---	.45	---	--	---	---	---	.19	7.08	---	----	---	---	.10
F-Used	-11259-	5.36	---	2.39	---	---	---	---	---	2.42	51.0	---	----	---	- -	.55
K-New	-11266-	.60	---	9.80	.28	---	---	---	--	2.51	----	---	----	---	---	.24
K-Used	-11267-	1.26	.39	7.30	.56	---	---	.65	---	1.86	----	---	----	---	---	.37

[a]See page Notes on XRF Particulate Wear Metal Analysis

[b]Presence could be due to wear when present with copper, or as an additive when present alone

[c]Probably present as additives

[d]Limit of detection for sample, when shown, element is less than this value

approximately 22 percent of that of the other lubricants. While the difference between the lives obtained with the other lubricants studied were not considered statistically significant, the life obtained with this lubricant is, in fact, statistically lower. The major differentiating factor appears to be the small amounts of sulfur and phosphorus. Considering the fact that lubricant K, also a pentaerythritol ester, has no sulfur present, it is strongly suggested that the phosphorus additive has no detrimental effect and could have a beneficial effect on the surface pitting fatigue life of gears. What appears to be important is that, in order to obtain reasonable gear life expectancy, a phosphorus additive or a combination of phosphorus, chlorine, and possibly sulfur must be present in reasonable quantities (not less than 2.5 ppm is suggested, but the suggestion is not substantiated by the results presented herein). The fact that lubricant K gave the best life with only phosphorus and chlorine additives and no sulphur additive suggests that a sulphur additive is not necessary for good gear life at the conditions tested.

For a long time it has been a practice in the gear industry to require EP and antiwear additive packages for gear oils. The additive packages are generally proprietary, and the scientific and engineering basis for their selection have been based on friction and wear tests rather than on rolling-element fatigue tests. Tables IV and V show that the lubricant exhibiting the lowest wear and friction was lubricant K which also produced the longest-gear life. The lubricant producing the highest wear and the lowest life was lubricant C. Friction and wear are not related in that low friction is not indicative of low wear and vice versa. Wear rate is a measure of the effectiveness of the EP and antiwear additive packages. This would explain the past success in gear lubricant selection and field experience.

Summary of Results

Spur-gear surface pitting fatigue tests were conducted with five lubricants using a single lot of consumable-electrode vacuum melted (CVM) AISI 9310 spur gears. The gears were case carburized and hardened to Rockwell C 60. The gear pitch diameter was 8.89 cm (3.5 in.). The lot of gears was divided into five groups, each of which was tested with a different test lubricant. The test lubricants can be classified as either synthetic hydrocarbon, mineral oil, or ester-based lubricants. All five lubricants have similar viscosity and pressure viscosity coefficients. Test conditions included a bulk gear temperature of 350 K (170 °F), a maximum Hertz stress of 1.71 GPa (248 000 psi) at the pitch line, and a speed of 10 000 rpm. The following results were obtained:

1. A pentaerythritol-base stock without sufficient antiwear additive produced a 10-percent surface fatigue life that was approximately 22 percent that of a pentaerythritol-base stock of the same viscosity with chlorine and phosphorus type additives.

2. The presence of a sulphur type antiwear additive in the lubricant does not appear to affect the surface fatigue life of spur gears under these test conditions.

3. No statistical difference in the 10-percent surface fatigue life was produced with four of the five test lubricants, having similar viscosity, pressure-viscosity coefficients, and various antiwear additives.

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