NASA Technical Memorandum 87044

(NASA-TM-87044) LUBRICANT AND ADDITIVE EFFECTS CN SPUR GEAR FATIGUE LIFE (NASA) 28 p HC A03/MF A01 CSCL 13I

N85-28373

Unclas
G3/37 21538

Lubricant and Additive Effects on Spur Gear Fatigue Life

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Prepared for the
1985 Lubrication Conference
cosponsored by the American Society of Mechanical Engineers
and the American Society of Lubrication Engineers
Atlanta, Georgia, October 8-10, 1985

NASA

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October 1985

The attached figure captions should be inserted in the report.

(a) Cutaway view.
(b) Schematic diagram.
Figure 1. - NASA Lewis Research Center's gear fatigue test apparatus.

(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.
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Figure 2. - 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 ksi).

(a) Typical fatigue spall.
(b) Cross section of typical fatigue spall.
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(a) Tetraester (reference) oil with 2.5-percent oxidation inhibitor plus 2.1-percent corrosion inhibitor (Lubricant L-1).
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(c) Tetraester oil with 0.1 wt % alkyl acid phosphate (phosphorous) EP additive (Lubricant L-3).
(d) Tetraester oil with 0.1 wt % alkyl amine thiocynate (sulfur) EP additive and without phosphate quinizarin corrosion inhibiter/metal passivator (Lubricant L-4).
(e) Summary.
Figure 4. - Surface pitting fatigue life of CVM AISI 9310 spur gears. Pitch diameter, 8.39 cm (3.5 in); speed, 10 000 rpm; lubricant, synthetic tetraester oil; gear temperature, 344 K (160 °F); maximum Hertz stress, 1.7×10^9 GPa (248 ksi).

LUBRICANT AND ADDITIVE EFFECTS ON SPUR GEAR FATIGUE LIFE

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ABSTRACT

Spur gear endurance tests were conducted with six lubricants using a single lot of consumable-electrode vacuum melted (CVM) AISI 9310 spur gears. The sixth lubricant was divided into four batches each of which had a different additive content. Lubricants tested with a phosphorus-type load carrying additive showed a statistically significant improvement in life over lubricants without this type additive. The presence of sulphur type antiwear additives in the lubricant did not appear to affect the surface fatigue life of the gears. No statistical difference in life was produced with those lubricants of different base stocks but with similar viscosity, pressure-viscosity coefficients and antiwear additives. Gears tested with a 0.1 wt % sulfur and 0.1 wt % phosphorus EP additives in the lubricant had reactive films that were 200 to 400 (0.8 to 1.6 μin) thick.

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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 and their additives on bearing and gear life and reliability.

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Lubricant additives are a necessity 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 (1,2).[1] These antiwear or extreme pressure (EP) additives either absorb onto the surfaces or react with the surfaces to form protective coatings or surface films. Although these boundary films are 1 μm (40 μin) or less thick (3), they can provide separation of the metal surfaces when the elastohydrodynamic (EHD) film becomes thin enough for the asperities to interact. The boundary film probably provides lubrication by microasperity-elastohydrodynamic lubrication as the asperities deform under load. The boundary film prevents contact of the asperities and at the same time provides low shear strength properties that prevent shearing of the metal while reducing the friction coefficient below that of the base metal.

The type of EP additive required will depend on the severity of the conditions of the meshing gear teeth, such as sliding velocity, surface temperature, and contact load. The boundary films can provide lubrication at different temperature conditions, depending on the materials used. Some boundary films melt at a lower temperature than others and thus fail to protect the surfaces (4). The failure temperature is the temperature at which the lubricant film fails. In EP lubrication, the failure temperature is that at which the boundary film melts. The melting point or thermal stability of surface films appears to be one unifying physical property that governs the failure temperature for a wide range of materials (4).

Virtually all effective EP additives are organic compounds that contain one or more elements such as sulfur, chlorine, phosphorous, carboxyl, or carboxyl salt that can react chemically with metal surfaces under the

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[1]Denotes references at end of text.

conditions of boundary lubrication (5). When these compounds react with metal surfaces they form inorganic films that can provide effective lubrication at high temperature and high sliding speed. These films are more stable than any physically or chemically absorbed film (6).

The NASA 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 (7-9). These lubricants were classified as three basic types: esters, mineral oils, and silicones. In the spin rig tests longer fatigue lives were obtained at room temperature with the silicone and one of the esters, a dioctyl sebacate, than with other lubricants. At 422 K (300 °F) in the spin rig where pure rolling occurs, the silicone and the mineral oils gave the longer lives (7). Similar results were obtained with gears (10). The relative order of fatigue results obtained in the five-ball fatigue tester compared favorably with that obtained with 7208-size ball bearings (7) 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 to preclude long term fatigue testing. The difference in life between these lubricants can be attributed to the elastohydrodynamic (EHD) film forming properties of the lubricants (8,9).

Rolling-element fatigue tests were conducted in the five-ball fatigue tester using a base oil with and without surface active additives. The 12.7-mm (0.500-in)-diameter test balls were either AISI 52100, AISI M-50, or AISI 1018 steel (9,11). The test lubricant was an acid-treated white oil containing either 2.5 percent sulfurized terpene, 1 percent didodecyl phosphite, or 5 percent chlorinated wax. Nine combinations of materials and lubricants (six containing additives) were tested at conditions including a maximum Hertz stress of 5.52 GPa (800 000 psi), a shaft speed of 10 700 rpm, and a race temperature of 339 K (150 °F). With the exception of the chlorinated wax additive, these additives showed essentially no statistical differences between the lives using the base oil with the additive and those without the additive. The presence of the chlorinated wax produced surface distress and a significant reduction in life.

The additives used with the base oil did not change the life ranking of bearing steels in these tests where rolling-element fatigue was of subsurface origin. That is, regardless of the additive content of the lubricant, the lives with the three materials ranked in descending order as follows: AISI 52100, AISI M-50, AISI 1018 (9,11).

The research reported herein, which is based on the work reported in (12,13), was undertaken to investigate the effects of lubricant base stock and additive content on spur gear tooth surface life. The primary objectives were (a) to determine the effects of six lubricants on the spur gear life, (b) compare the surface pitting fatigue lives of the test spur gears using a tetraester oil with and without EP additives, and (c) determine the changes in chemical composition over the surface of the gear teeth and relate these changes to the effectiveness of the EP additives in forming the boundary film.

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 Fig. 1 and described in (14). 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. 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.

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 GEARS

Dimensions for the test gears are given in Table I. All gears have a nominal surface finish on the tooth face of 0.406 μm (16 μin) rms, with 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.

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 II. All sets of gears were case carburized and heat treated in accordance with the heat treatment schedule of Table III.

TEST LUBRICANTS

Six 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, and ester-based lubricants. Tests were conducted on these lubricants (15) 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 (15) 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 (16). 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 silicon seal compounds are used. Both oils are pentaerythritol esters with nearly the same viscosity and pressure viscosity characteristics. However, the measured wear rate (15) 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 military specifications for aircraft transmission application. However, it has been recommended for use in helicopter transmissions.

Lubricant L is a pentaerythritol tetraester of mixed alkyl acids. The lubricant was formulated in four batches (L-1 to L-4) with each batch having the constituents shown in Tables IV and V. Batch L-1 meets the MIL-L-23699 specification.

The pitch-line elastohydrodynamic (EHD) film thickness was calculated by the method of (17) and (18) and using the data of Table IV for each of the lubricants. It was assumed, for this film thickness calculation, that the gear surface temperature at the pitch line was equal to the oil temperature at the outlet of the gearbox and that the oil temperature entering the contact zone was equal to the gear temperature, even though the temperature of the oil jet to the gears 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 (12).

TEST PROCEDURE

After the test gears were cleaned to remove the preservative, they were assembled on the test rig. The test gears were run in an offset condition with a 0.305 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^3/min (0.21 gal/min) at 320$\pm$6 K (116$\pm$10 °F). The lubricant outlet temperature was nearly constant at 350$\pm$3 K (170$\pm$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.

RESULTS AND DISCUSSION

Gear life. The surface pitting fatigue life of the AISI 9310 gears run with lubricant F, the synthetic paraffinic oil is shown in Fig. 2. These data which are shown on Weibull coordinates were analyzed by the method of (19). 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 Fig. 3(a). A cross section of the spall is shown in Fig. 3(b). The failure index in Fig. 2 indicates the number of failures out of the number of tests run. For lubricant F, the failure index was 18 out of 19.

Lubricant F is the standard test oil used with all of the NASA gear material endurance tests (16). 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 life results are summerized in Fig. 2 and Table VI.

The surface pitting fatigue life of the CVM AISI 9310 steel spur gears run with lubricant E, a dibasic acid ester, are summarized in Fig. 2 and Table VI. 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 with lubricant F. Based upon a comparison of the physical properties of the two lubricants, this result is not unexpected if chemical differences are discounted. The surface pitting fatigue lives obtained with lubricant A, the mineral oil based lubricant, are shown in Fig. 2 and summarized in Table VI. 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 lubricant F, the synthetic paraffinic oil and lubricant E, the dibasic acid ester lubricant. The difference in these lives based upon the confidence numbers given in Table VI 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 the A lubricant will produce a higher life than the F lubricant. 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 meaningful in order to draw conclusions regarding life differences.

The surface pitting fatigue lives obtained with lubricant K, a pentaerythritol ester are shown in Fig. 2 and summarized in Table VI. 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 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 upon the physical properties alone and not on chemical differences, no statistical differences in life would be expected because resultant elastohydrodynamic film thickness would be nearly the same for all the lubricants.

The life results for lubricant C which is also a pentaerythritol ester are shown in Fig. 2 and summarized in Table VI. The 10- and 50-percent lives obtained with this fluid were 4.8 million and 25.9 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.

Hence, the life obtained with this lubricant is statistically significantly lower than the previous four lubricants. Based upon the physical characteristics of this fluid and the life results previously discussed, these results were not expected. It was speculated that undefined chemical effects due to the lubricants additive package may have contributed to the lower lives obtained.

Additive effects. From the results of lubricants C and K, it appears that the extreme pressure (EP) additive in lubricant K is responsible for the five to one improvement in the 10-percent surface fatigue life of the gears. Therefore, a series of tests were conducted to determine the effect of different EP additives on the gear surface fatigue life. In order to accomplish this evaluation, four groups of test gears were surface fatigue tested with a base lubricant containing different additives for each test group of gears. The base lubricant designated L-1 was a pentaerythitol tetraester which is a synthetic lubricant. This lubricant which has similar physical properties to lubricants C and K contained oxidation and corrosion inhibitors but no EP type additives.

The surface pitting fatigue lives obtained with lubricant L-1 are summarized in Figs. 2 and 4 and Table VII. The resultant 10- and 50-percent lives obtained with this fluid were 7.6 million and 31 million stress cycles, respectively. These lives were statistically similar to those obtained with lubricant C discussed herein above.

Lubricant L-1 was modified by the addition of 0.1 wt % sulfur type load carrying additive to formulate lubricant L-2. Additionally, lubricant L-2 was also formulated wherein the amount of oxidation inhibitor contained in the lubricant was reduced from 2.5 to 0.5 wt %. The batch so formulated with the 0.5 wt % oxidation inhibitor was designated as lubricant L-4. The results of lubricants L-2 and L-4 are shown in Fig. 4 and summarized in Table VII. The

10- and 50-percent lives for lubricant L-2 were 9.3 million and 65.6 million stress cycles, respectively. The 10- and 50-percent lives for lubricant L-4 were 12.5 million and 73.8 million stress cycles, respectively. There is an inference that the addition of the sulfur additive as well as the reduction in oxidation inhibitor improves life. However, there is no statistical differences in the lives obtained with lubricants L-1, L-2, and L-4. Hence, while it cannot be concluded with definite certainty that the presence of the sulfur EP additive improves life nor the presence of the oxidation inhibitor degrades life, it can be concluded with reasonable statistical certainty that the presence of a sulfur EP additive does not adversely affect gear surface pitting fatigue life.

Lubricant L-1 was also modified by the addition of 0.1 wt % phosphorus-type load carrying additive. This lubricant batch was designated lubricant L-3. The surface pitting fatigue lives obtained with lubricant L-3 were 19.8 and 67.1 million stress cycles for the 10- and 50-percent lives, respectively. These test results indicate a 10-percent life which is 2.6 times that of the gears tested with lubricant L-1 and statistically similar to those gears tested with lubricant K. These results would indicate that the phosphorus EP additive was beneficial in improving the surface fatigue life of the gears.

Scanning Auger microscope (SAM) images of gear teeth run with lubricants L-1 to L-4 were made (12). Results of the SAM studies revealed that the gears tested with the 0.1 wt % sulfur and 0.1 wt % phosphorus EP additives in the oil had reactive surface films that were 200 to 400 (0.8 to 1.6 μin) thick. The surface film across the tooth was nonuniform in thickness. This would suggest that protective films are formed by the reaction of the additives with the surface in response to some spacially varying environmental factor (i.e., mechanical load). No quantitative or qualitative conclusions could be reached on the basis of these tests and the SAM examination regarding the thickness of the film measured and the resultant surface fatigue life.

Chemical Effects

Except for lubricants L-1 to L-4, the other lubricants studied were commercial products having proprietary additives. As a result the additive contents could not be directly identified nor directly assessed. These lubricants were also tested in an OH-58 helicopter transmission. An x-ray fluorescence (XRF) filter method was used to identify and measure the metals contained in the lubricants (15) prior to and subsequent to testing in an OH-58 helicopter transmission at the authors' laboratory. With this method the wear metals and additive particulates are separated from the lubricant and subjected to energy-dispersive x-ray fluorescence analysis. Using this analysis the type and amount of lubricant additive can be identified. This method gives a sensitivity of 0.1 ppm. The results of these measurements are shown in Table VIII.

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 would be indicative of 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 XRF 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 zinc could be due to wear when present with copper or an additive when present alone. 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 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 (9) which showed a chlorinated wax additive to be detrimental to life. However, the tests in (11) were run at Hertz stresses more than three times (5.5 mPa (800 000 psi)) that used for the gears. Since the chemical effect of the additive are temperature dependent the higher stresses used in (9) may account for the difference in the results.

Lubricant C was the other pentaerythritol ester studied. This lubricant showed traces of both phosphorus and sulfur in much lower quantities than with the other lubricants in both the new and used samples. This lubricant produced the lowest gear life, approximately 22 percent of that obtained with the other lubricants. While the difference between the lives obtained with the other lubricants studied were considered statistically insignificant, 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 that lubricant K also being 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 phosphorous, chlorine and possible sulfur must be present in reasonable quantities (not less than 2.5 ppm is suggested but not substantiated from the results presented herein).

The fact that lubricant K gave the best life with on'y phosphorus and chlorine and no sulphur additive suggest 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 or nonselection have been based on friction and wear tests rather than on rolling-element fatigue tests. Referring to Table IV 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 vise versa. The 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. Based on the results of the gear tests conducted in this program, a phosphorous type EP additive should be added to gear lubricants to improve the surface fatigue life of steel gears.

SUMMARY

Research was conducted to investigate the effects of lubricant base stock and additive content on spur gear tooth surface (pitting) life. Spur gear endurance tests were conducted with six lubricants using a single lot of consumable-electrode vacuum method (CVM) AISI 9310 gear test specimens. The sixth lubricant was divided into four batches each of which had a different

additive content. 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. Lubricants tested with a phosphorus-type load carrying additive showed a statistically significant improvement in life over those lubricants without this type additive.

2. Lubricants with sulfur EP additives showed no statistical improvement in life compared with those lubricants without this type additive.

3. No statistical difference in life was produced with those lubricants of different base stock but with similar viscosity, pressure-viscosity coefficients, and antiwear additives. The EP and antiwear package in the lubricant appears to control the resultant surface fatigue life of the gears.

4. Scanning Auger microscope (SAM) images of the gear tooth surfaces made after testing showed that gears tested with 0.1 wt % sulfur and 0.1 wt % phosphorus EP additives in the lubricant had reactive surface films that were 200 to 400 (0.8 to 1.6 μin) thick.

ACKNOWLEDGMENTS

The authors wish to acknowledge the advise, assistance and cooperation of Richard S. Fein who has since retired from the Texaco Co., and Frank M. Newman of the Fuels and Lubricants Research Laboratory, Southwest Research Institute.

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TABLE I. - SPUR GEAR DATA

[Gear tolerance per ASMA class 13.]

Number of teeth	28
Diametral pitch	8
Circular pitch, cm (in)	0.9975 (0.3927)
Whole depth, cm (in)	0.762 (0.300)
Addendum, cm (in)	0.318 (0.125)
Chordal tooth thickness reference, cm (in)	0.485 (0.191)
Pressure angle, deg	20
Pitch diameter, cm (in)	8.890 (3,500)
Outside diameter, cm (in)	9,525 (3.750)
Root fillet, cm (in)	0.102 to 0.152 (0.04 to 0.06)
Measurement over pins, cm (in)	9.603 to 9.630 (3.7807 to 3.7915)
Pin diameter, cm (in)	0.549 (0.216)
Backlash reference, cm (in)	0.0254 (0.010)
Tip relief, cm (in)	0.001 to 0.0015 (0.0005 to 0.0007)

TABLE II. - CERTIFIED CHEMICAL COMPOSITION OF GEAR MATERIAL

CVM AISI 9310 AMS 6265D

	Element									
	C	Mn	P	S	Si	Ni	Cr	Mo	Cu	Fe
Contents, wt %	0.11	0.64	0.0007	0.004	0.34	3.18	1.33	0.17	0.26	Balance

TABLE III. - HEAT TREATMENT FOR AISI 9310

Step	Process	Temperature,		Time, hr
		K	°F	
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

TABLE IV. - LUBRICANT CHEMICAL AND PHYSICAL PROPERTIES SUMMARY

	Lubricant					
	A Mineral oil	C Polyol ester pentaerythritol	E Dibasic acid ester	F Synthetic paraffinic	K Polyol ester pentaerythritol	L[a] Pentaerythritol tetraester
Carboxylic acids, percent						
C-4	---------------	Trace	---------------	---------------	Trace	---------------
C-5	---------------	46	Di-63	---------------	22	100
C-6	---------------	10	Di-37	---------------	16	100
C-7	---------------	17	---------------	---------------	24	100
C-8	---------------	10	---------------	---------------	8	---------------
C-9	---------------	13	---------------	---------------	29	---------------
C-10	---------------	4	---------------	---------------	1	---------------
Additives	Proprietary	Proprietary	Proprietary	Lubrizol 5002 (5 vol %) Content of additive Phosphorus 0.6 wt % sulphur 18.5 wt %	Proprietary	See Table 5
Specification	GM 6137-M	MIL-L-23699	---------------	---------------	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	Type II turbine engine oil
Kinematic viscosity, cm^2/sec (cS) at - 244 K (-20 °F)	35. (3500)	30. (3000)	50. (5000)	26. (2600)	30. (3000)	---------------
311 K (100 °F)	0.40 (40)	0.285 (28.5)	.36 (36)	.303 (30.3)	.285 (28.5)	.276 (27.6)
372 K (210 °F)	.07 (7.2)	.053 (5.3)	.06 (6.0)	.055 (5.5)	.053 (5.3)	.052 (5.2)
477 K (400 °F)	.015 (1.5)	.013 (1.3)	.014 (1.4)	.013 (1.3)	.013 (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) Fire point, K (°F)	433 (320) 488 (347)	527 (490)	513 (465)	508 (455) 533 (500)	533 (500)	527 (490)
Pour point, K (°F)	233 (-40)	211 (-80)	219 (-65)	219 (-65)	214 (-75)	213 (-76)
Specific gravity	0.862	1.005	0.932	0.829	0.983	0.982
Specific heat at 311 K (100° F), J/kg K (Btu/lb °F)	1758 (0.42)	1842 (0.44)	2847 (0.68)	2177 (0.52)	1884 (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 (15.3)	0.411 (16.2)	0.731 (28.8)
Λ ratio (h/σ)	0.87	0.76	0.86	0.65	0.69	1.22

[a]See Table 5.

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TABLE V. - LUBRICANT COMPOSITION AND PROPERTIES

(a) Compositions

	Lubricant			
	L-1[a]	L-2	L-3	L-4
	Composition, wt %			
Tetraester base oil	95.4	95.3	95.3	99.4
Amine-type oxidation inhibitor	2.5	2.5	2.5	0.5
Phosphate quinizarin corrosion inhibitor plus metal passivator	2.1	2.1	2.1	----
Alkyl amine thiocynate EP additive	----	.1	----	.1
Alkyl acid phosphate EP additive	----	----	.1	----
Water	.019	.020	.010	.016

(b) Properties

	L-1[a]	L-2	L-3	L-4
Kinematic viscosity (CS) at-				
311 K (100 °F)	27.58	27.90	27.90	26.40
373 K (212 °F)	5.22	5.17	5.18	5.08
Flash point, K (°F)	527 (490)	533 (500)	527 (490)	533 (500)
Pour point, K (°F)	213 (-76)			
Specific gravity at-				
311 K (100 °F)	0.982	0.982	0.982	0.979
373 K (212 °F)	0.943	0.943	0.943	0.939
Total acid number (TAN), mg KOH/g oil	0.02	0.31	0.29	0.15

[a]Meets MIL-L-213699 specifications.

TABLE VI. - SUMMARY OF GEAR FATIGUE LIFE RESULTS

[NASA spur gear test apparatus; material, CVM AISI 9310; gear bulk temperature, 350 K (170 °F); maximum Hertz stress, 1.71 GPa (248 ksi); speed, 10 000 rpm.]

Lubricant	Lubricant basestock	Gear system life, millions of stress cycles		Weibull slope	Failure index[a]	Confidence number,[b] percent
		10 percent	50 percent			
A	Mineral oil	22.8	53.7	2.2	20 out of 20	62
C	Pentaerythritol ester	4.8	25.9	1.1	20 out of 20	96
E	Dibasic acid ester	18.8	43.7	2.2	20 out of 20	50
F	Synthetic Paraffinic	18.8	46.1	2.1	18 out of 19	--
K	Pentaerythritol ester	24.7	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 F.

TABLE VII. - SUMMARY OF GEAR FATIGUE LIFE RESULTS

[Material, CVM AISI 9310 steel; pitch diameter, 8.89 cm (3.5 in); speed, 10 000 rpm; maximum Hertz stress, 1.71 GPa (248 ksi); lubricant, synthetic tetraester oil with various additives; gear temperature, 344 K (160 °F).]

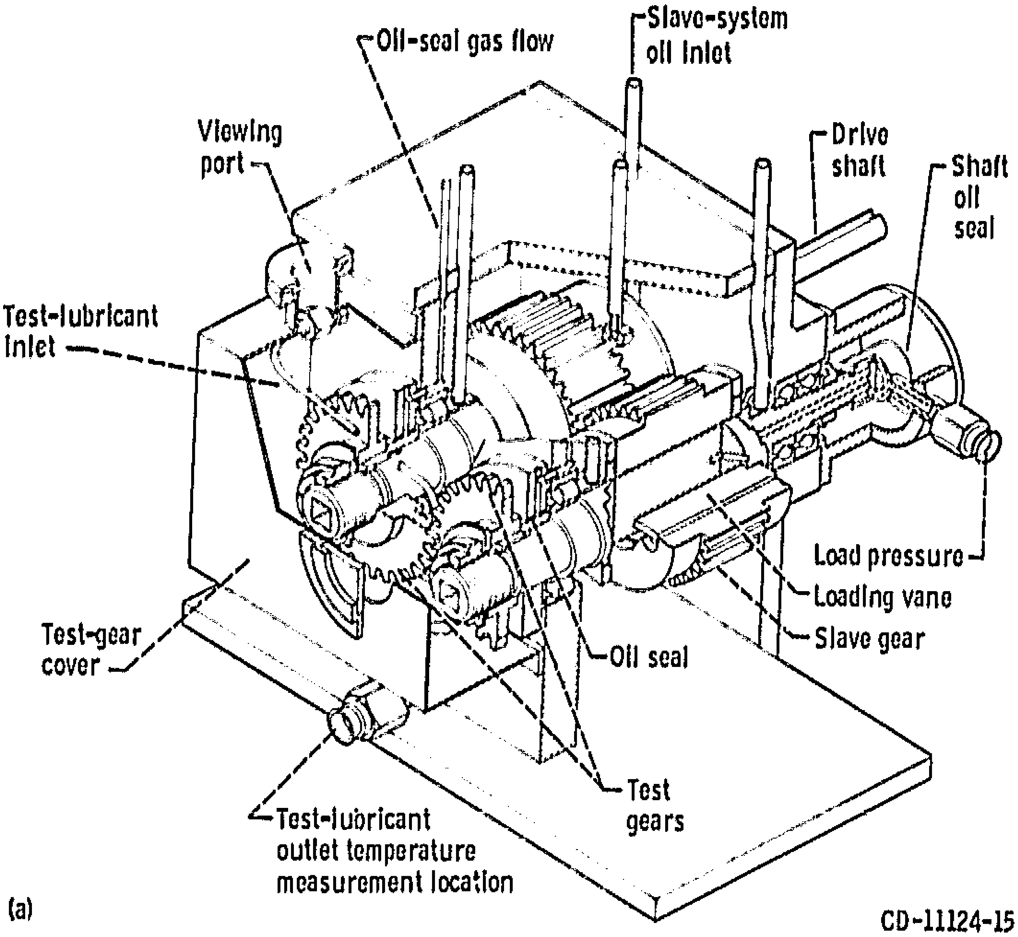
Lubricant	Gear system life millions of stress cycles		Weibull slope	Failure index[a]	Confidence number[b]
	10-percent	50-percent			
L-1-Tetraester reference oil + 2.5 percent oxidation inhibitor +2.1 percent corrosion inhibitor	7.61	31.0	1.34	30 out of 30	95
L-2-Same as L-1 + 0.1 percent sulfur-type load-carrying additive	9.26	65.6	.96	22 out of 23	81
L-3-Same as L-1 + 0.1 percent phosphorous-type load-carrying additive	19.8	67.1	1.55	20 out of 20	52
L-4-Tetraester reference oil + 0.5 percent oxidation inhibitor + 0.1 percent sulfur-type load-carrying additive	12.5	73.8	1.06	21 out of 21	69

[a]Number of fatigue failure out of number tested.
[b]Percent of time that 10 percent life obtained with each lubricant will have the same relation to the 10-percent life of lubricant F.

TABLE VIII. - ANALYTICAL REPORT SYNTHETIC LUBRICANT X-RAY FLOURESCENCE ANALYSIS

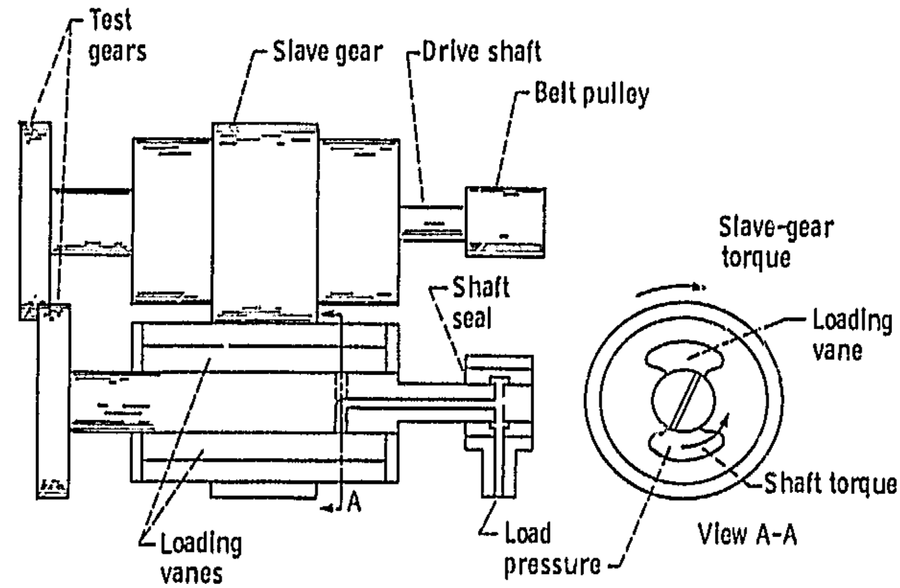
Lubricant code		Element content, ppm[a]														Limit[d] of detection, ppm
NASA	AFLRL	MG	Al	Cl	Fe	Ni	Cu	Pb	Zn[b]	P[c]	S[c]	Ca[c]	Ba[c]	Si	Mn	
A-New	-11252-	0.48	----	2.47	----	----	----	0.21	----	0.18	4.71	----	0.23	---	---	0.11
A-Used	-11253-	---	5.91	1.12	0.51	0.10	0.14	----	0.11	.17	1.12	----	.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	0.29	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	----	2.49	----	----	----	----	----	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.



(a)

CD-11124-15



(b)

CD-11124-15

Figure 1.

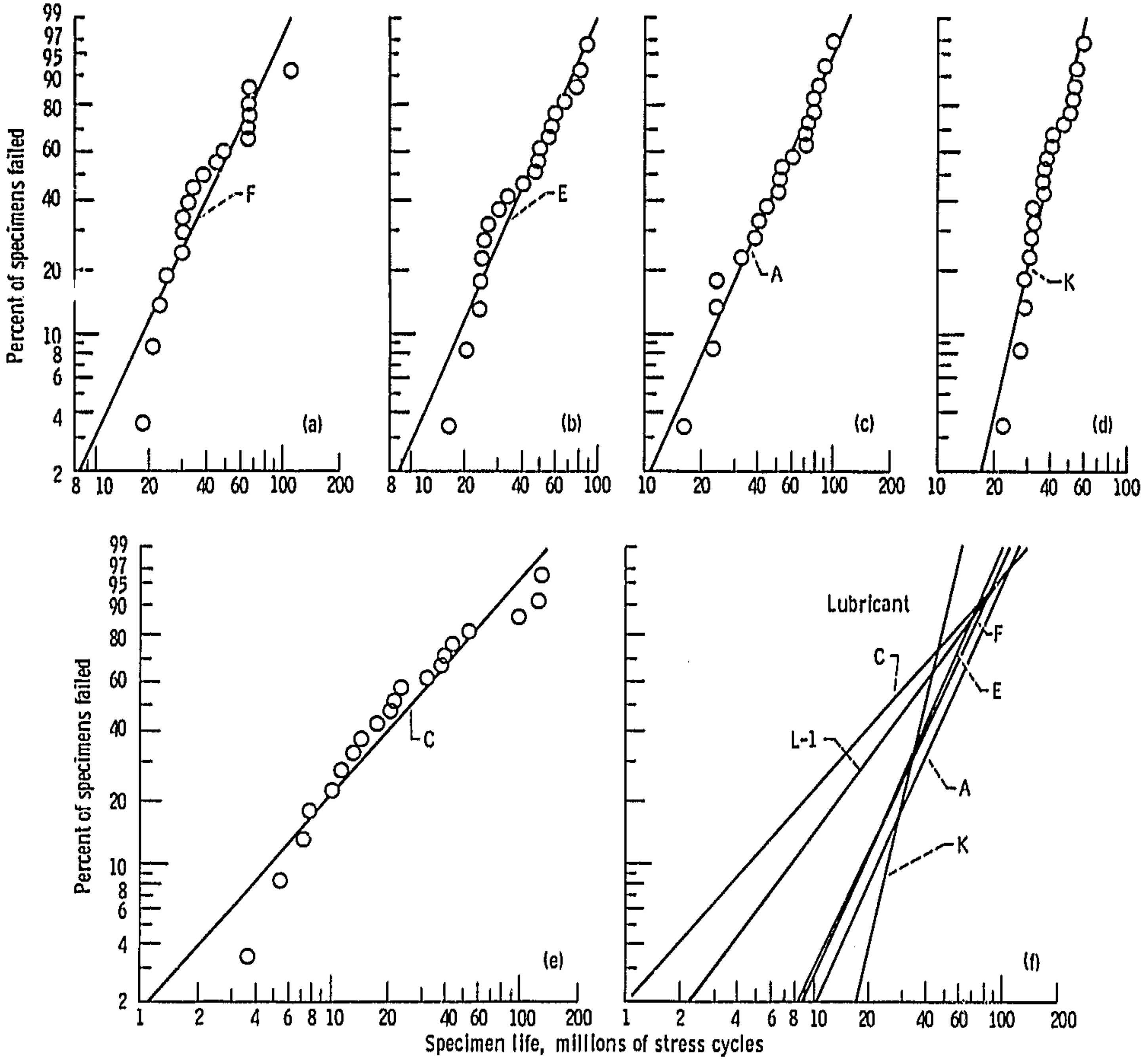

Percent of specimens failed
F
E
A
K
(a)
(b)
(c)
(d)
C
(e)
Lubricant
C
F
E
L-1
A
K
(f)
Specimen life, millions of stress cycles

Figure 2.

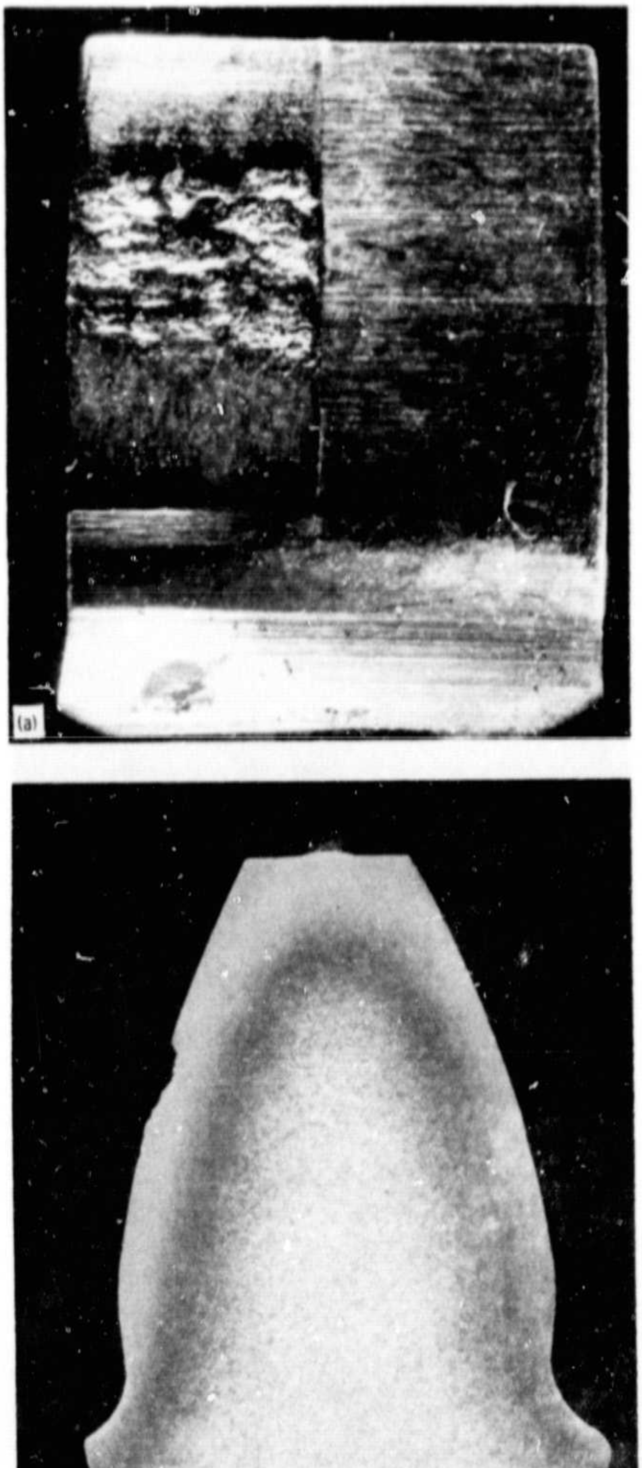
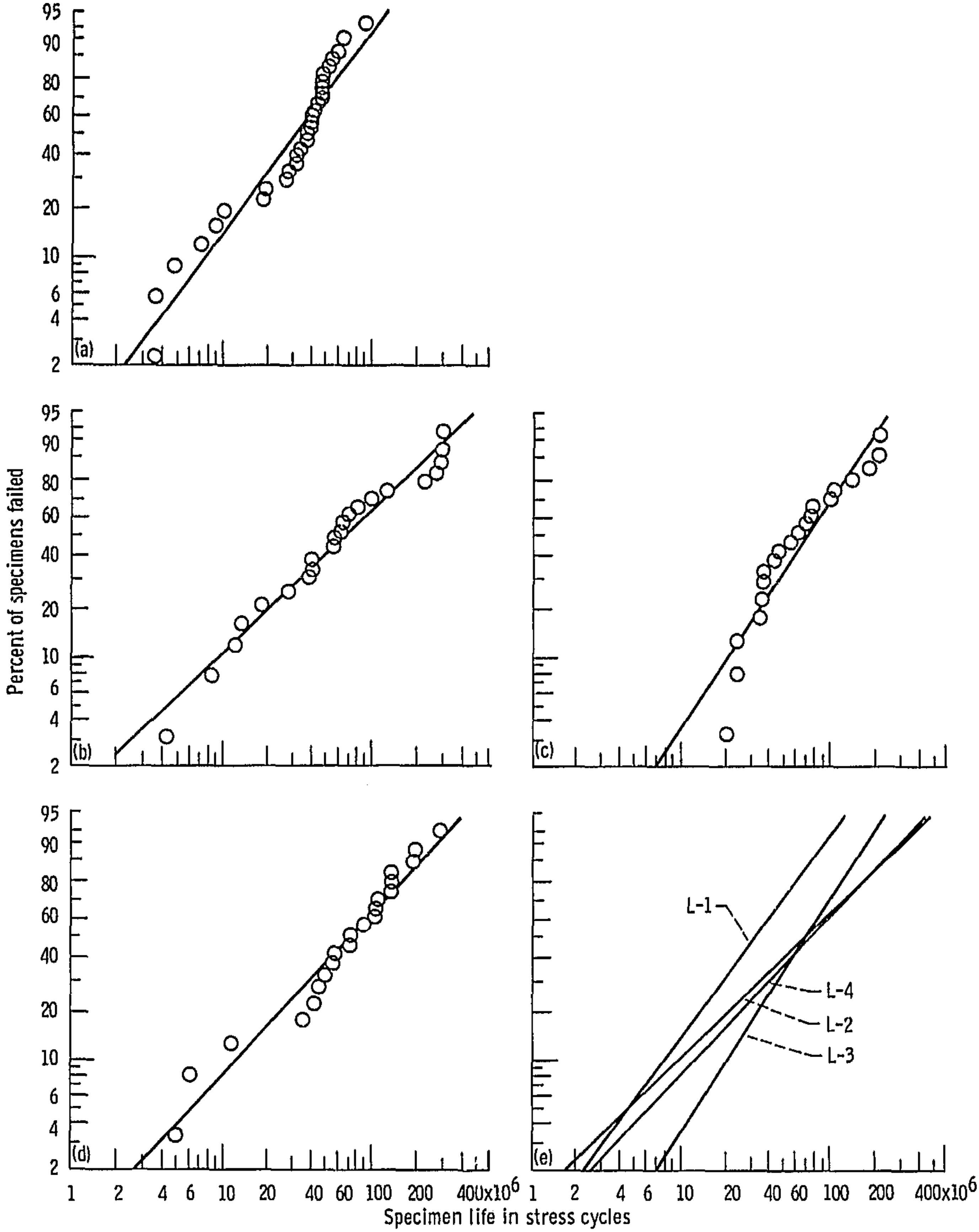


Fig. 3.



95
90
80
60
40
20
10
6
4
2
(a)
(b)
(c)
(d)
(e)
L-1
L-4
L-2
L-3
Percent of specimens failed
1
2
4
6
10
20
40
60
100
200
400x10^6
Specimen life in stress cycles

Figure 4.

1. Report No.	2. Government Accession No.	3. Recipient's Catalog No.
NASA TM-87044		

4. Title and Subtitle

Lubricant and Additive Effects on Spur Gear Fatigue Life

5. Report Date

6. Performing Organization Code

505-33-72

7. Author(s)

Dennis P. Townsend, Erwin V. Zaretsky, and Herbert W. Scibbe

8. Performing Organization Report No.

E-2482

10. Work Unit No.

9. Performing Organization Name and Address

National Aeronautics and Space Administration
Lewis Research Center
Cleveland, Ohio 44135

11. Contract or Grant No.

13. Type of Report and Period Covered

Technical Memorandum

12. Sponsoring Agency Name and Address

National Aeronautics and Space Administration
Washington, D.C. 20546

14. Sponsoring Agency Code

15. Supplementary Notes

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16. Abstract

Spur gear endurance tests were conducted with six lubricants using a single lot of consumable-electrode vacuum melted (CVM) AISI 9310 spur gears. The sixth lubricant was divided into four batches each of which had a different additive content. Lubricants tested with a phosphorus-type load carrying additive showed a statistically significant improvement in life over lubricants without this type additive. The presence of sulphur type antiwear additives in the lubricant did not appear to affect the surface fatigue life of the gears. No statistical difference in life was produced with those lubricants of different base stocks but with similar viscosity, pressure-viscosity coefficients and antiwear additives. Gears tested with a 0.1 wt % sulfur and 0.1 wt % phosphorus EP additives in the lubricant had reactive films that were 200 to 400 (0.8 to 1.6 μin) thick.

17. Key Words (Suggested by Author(s))

Gears; Lubrication; Additives; Pitting fatigue; Rolling-element fatigue

18. Distribution Statement

Unclassified - unlimited
STAR Category 37

19. Security Classif. (of this report)	20. Security Classif. (of this page)	21. No. of pages	22. Price*
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