EXPERIMENTAL ANALYSIS OF HARD COATING ON SPUR GEAR

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ABSTRACT
Hard coatings have potential for increasing gear surface fatigue lives. Experiments were conducted using gears both with and without a metal-containing, Carbon-based coating. The gears were case-carburized AISI 9310 steel spur gears. Some gears were provided with the coating by magnetron sputtering. Lives were evaluated by accelerated life tests. For uncoated gears, all of 15 tests resulted in fatigue failure before completing 275 million revolutions. For coated gears, 11 of the 14 tests were suspended with no fatigue failure after 275 million revolutions. The improved life owing to the coating, approximately a six fold increase, was a statistically significant result.
KEYWORDS: Hard coating, Gears, Heat treatment, tribological behavior

1. INTRODUCTION

The power density of a gearbox is an important consideration for many applications and is especially important for gearboxes used on aircraft. One factor that limits gearbox power density is the need to transmit power for the required number of cycles while avoiding gear surface fatigue failure (micro pitting, pitting, or spalling). Effective and economical methods for improving surface fatigue lives of gears are therefore highly desirable. Some product durability improvements have been achieved by the application of thin, hard coatings to gears.

Diamond like carbon and related materials have the potential to improve machine performance for a wide variety of applications that require wear protection and/or low friction properties. Because of the widely recognized potential, the deposition methods and resulting properties of the films have been studied extensively. Today's deposition technology allows for the production of a great diversity of coatings, but the ability to reliably tailor the tribological behavior of a coating for a particular application has been elusive.

Aerospace gearing requirements are demanding, calling for high power density, long life, and excellent reliability. The low friction properties and high hardness of diamond like and related coatings offer the possibility to improve the performance of aero space gearing.

The purpose of the present investigation was to compare the surface fatigue lives of coated and uncoated gears using accelerated life tests. The testing is considered as accelerated in that the contact stresses used for testing exceeds the stressed used for design of the target application (helicopter gearing). The metal containing, carbon-based coating selected for this study was designed specifically for aerospace gearing application.

2. DESCRIPTION OF THE TEST GEARS, LUBRICANT, AND COATING

The test gears used for this work were manufactured in one lot from a single heat of consumable electrode vacuum-melted (CVM) AISI 9310 steel. The nominal chemical composition of the AISI 9310 material is given in Table 2.1. The gears were case carburized and heat treated according to Table 2. The nominal properties of the carburized gears were a case hardness of Rockwell C60 a case depth of 0.97mm (0.038 in), and a core hardness of Rockwell C38.

The dimension for the test gears are given in Table 3. The gear pitch diameter was 89 mm (3.5 in), and the tooth form was a 20° involute profile modified to provide a tip relief of 0.013 mm (0.0005 in) starting at the highest point of single tooth contact. The gear tooth surface finish after final grinding was specified as a maximum of 0.406μm (16μin).

The lubricant used for testing was from a single batch of synthetic paraffin oil. Physical properties of this lubricant are summarized in Table 1.

Table 1 Nominal Chemical Composition of AISI 9310 gear material

Element	Weight %
Carbon	0.10
Nickel	3.22
Chromium	1.21
Molybdenum	0.12
Copper	0.13
Manganese	0.63
Silicon	0.27
Sulphur	0.005
Phosphorous	0.005
Iron	balance

W(tungsten) coatings were deposited onto carburized spur gears after final grinding using an unbalanced magnetron sputter deposition process. The nature of the deposition process results in substrate temperatures that are no uniform. The temperature during the coating process depends on both the part geometry and the thermal properties of the substrate. The coating was applied in a manner to avoid tempering of the substrate. Measurements of case and core hardness on cross sections have confirmed that tempering of the substrate was avoided. Prior to coating deposition, the spur gears were "vapor honed" using pressurized water based media containing 10μm diameter A12 O3 particles.The vapor honing process is applied t the functional surfaces of the gears to promote coating substrate adherence. The vapor honing of the ground surfaces results in a slight refinement of the surface roughness, producing a roughness of about 0.30 – 0.36 μm (11-14 μin) . The coating/gear system consisted of an elemental Cr (chromium) adhesion layer adjacent to the steel substrate, followed by an intermediate, transition region, featuring alternating lamellae composed of Cr and WC, and an outermost W-containing hydrocarbon (W-C:H) layer.

Table 2 Heat treatment for AISI 9310 gears

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	Austentize	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 relieve	450	350	2

Table 3 spur gear data

Number of teeth	28
Module, mm	3.175
Diametric pitch	8
Circular pitch, mm	9.975
Whole depth, mm	7.62
Addendum, mm	3.18
Chordal tooth thickness reference, mm	4.85
Tooth width, mm	6.35
Pressure angle, deg	20
Pitch diameter, mm	88.90
Outside diameter, mm	95.25
Root fillet, mm	1.02 to 1.52
Measurement over pins, mm	96.03 to 96.30
Pin diameter, mm	5.49
Backlash reference, mm	9.254
Tip relief, mm	0.010 to 0.015

Table 4 lubricant properties

Additive	Lubrizol 5002
Kinematic viscosity, cSt 311 K (100 °F) 372 K (210 °F)	 31.6 5.7
Specific gravity	0.83
Flash point, K	544
Pour point, K	211

3. TEST APPARATUS AND PROCEDURE

The gear fatigue tests were performed in the NASA Glenn Research Center's gear test apparatus. The test rig is shown In Fig.2 (a). The input drive only needs to overcome the frictional losses in the system. The test rig is belt driven and operated at a fixed speed of 10000 rpm for the duration of a particular test.

A schematic of the apparatus is shown in Fig 2(b). Oil pressure and leakage replacement flow is supplied to the load vanes through a shaft seal. As the oil pressure is increased on the load vanes located inside one of the slave gears, torque is applied to its shaft. This torque is transmitted through the test gears and back to the slave gears. In this way power is circulated, and the desired load and corresponding stress level on the test gear teeth may be obtained by adjusting the hydraulic pressure. The two identical test gears may be started under no load, and the load can then be applied gradually. This arrangement also has the advantage that changes in load do not affect the width or position of the running track on the gear teeth. To enable testing at the desired contact stress, the gears are tested with the faces offset as shown in fig 2. By utilizing the offset arrangement for both faces of the gear teeth, a total of four surface fatigue tests can be run for each pair of gears.

Separate lubrication systems are provided for the test and slave gears. The two lubrication systems are separated at the gearbox shafts by pressurized labyrinth seals, with nitrogen as the seal gas. The test gear lubricant is filtered through a 5 - µm (200 - µin) nominal fiberglass filter.

A vibration transducer mounted on the gearbox is used to automatically stop the test rig when the broadband rms vibration magnitude increases beyond a threshold, indicating the occurrence of gear surface fatigue damage. The gearbox is also automatically stopped if there is a loss of oil flow to either the slave gearbox or the test gears, if the test gear oil overheats, or if there is a loss of seal gas pressurization.

The test gears were run with the tooth faces offset by a nominal 3.3mm (0.130 in) to give a contact width on the gear face of 3.0 mm (0.120 in). The actual tooth face offset for each test uses the measured face width of the test specimen and is verified on installation using a depth gage. The nominal 0.13-mm- (0.005 in) radius edge break is accounted for to calculate load intensity. All tests were run in using a load 9normal to the pitch circle) of 123 N/mm 9700 lb/in) for 1h. The load was then increased to the desired test load. For the uncoated gears, all tests were conducted using a test load of 580 N/mm 93300 lb/in) which resulted in a 1.7 – GPa (250 – ksi) pitch line maximum Hertz stress. For the coated gears, six tests were conducted at the same test load that was used for the uncoated gears and eight tests were conducted using a test load of 7220 N/mm (4100 lb/in) which resulted in a 1.9 –GPa (280 – ksi pitch line maximum hertz stress. The Hertz stress is an idealized stress index assuming static equilibrium, perfectly smooth surfaces, and an even pressure distribution across a 2.79 mm 90.11. in) line contact (the line length is a less than the face width allowing for the face offset and the radius edge break). Typical dynamic tooth forces using the same rigs and gears of the same specification have been measured 9Kratz [10]). And the results are provided in fig 3. The tooth forces reported in fig 3. Are the dynamic forces normal to the tooth surface for nominal pitch line test load intensity of 580 N/mm (3300 lb/in). The contact force used for stress calculations for such load intensity was 1720 N (387 lb). This value for the contact force is the value required for static equilibrium, and it is somewhat less than the measured dynamic forces.

The gears were tested at 10000 rpm resulting in a pitch line velocity of 46.5 m/s(9154 ft/min). Inlet and outlet oil temperatures were continuously monitored. Cooled lubricant was supplied to the inlet of the gear mesh at 0.8l/min (0.2 gal/min)and 320 ±7 K (116 ± 13 °F) The lubricant outlet temperature was recorded and observed to have been maintained at 348 ± 4.5 K 9116 ± 8 °F). The lubricant was circulated through a 5 - µm- (200-µ in) nominal fiberglass filter to remove wear particles. For each test, 3.8 1 (1 gal) of lubricant was used.

The tests ran continuously (24 h/day) until a vibration detection transducer automatically stopped the rig. The transducer was located on the gearbox housing. The gears were also inspected visually at intervals of approximately 50 million cycles. For purposes of this work, surface fatigue failure was defined as one or more spells or pits covering at least 50 percent of the width of the Hertzian line contact on any one tooth. If the gear pairs operated for more than 460 h (corresponding to 275 million stress cycles) without failure, the test was suspended.

4. RESULT AND DISCUSSIONS

Results of the gear surface fatigue testing are summarized in Table 5. A total of 29 tests were completed, 15 tests using the uncoated gears and 14 tests using the coated gears. For coated gear tests, both gears (driving and driven members) were coated for all tests. All of the baseline tests were conducted at a Hertzian stress index o f1.7 GPa (250 ksi), with all tests resulting in failures. The range of duration for the tests using uncoated baseline gears was 25-272 million revolutions. The coated gears were tested at two loads. Six of the coated gears were tested at a Hertzian stress index of 1.7 GPa (250 ksi), and eight of the coated gears were tested at a stress index of 1.9 GPa (280 ksi). The range of duration for the tests using coated gears was 63-311 million revolutions, and 11 of the 14 tests were completed with no failure after at least 275 million revolutions.

The distributions of the fatigue lives were modeled as two parameter Weibull distributions. The fatigue lives are system lives, the system consisting of two identical gears. Tests that were suspended after pre-specified times showed no signs of impending failure, and so such tests were treated as right-censored life tests for the purpose of statistical analysis. It was decided not to estimate the slope of the coated gear population with only three failure data points. Instead, it was assumed that the coated and uncoated gears had life distributions with equal Weibull slope parameters. Alikelihood ration statistical test was used to verify that the assumption of equal slopes was indeed a reasonable assumption. Software employing the maximum likelihood method is used to estimate the Weibull parameter values from the test data. Figure 4 is a Weibull plot displaying the test data and the lines representing the maximum likelihood fit Weibull distributions. Data points are plotted at the positions of exact median ranks with adjustments to the order numbers to account for suspended tests . The results of the statistical analysis are summarized in Table 6. The ten percent lives of the uncoated and coated gear populations were estimated to be 28 X 10 6 and 180 X 10 6 cycles, respectively.

Table 5 Summary of test results

Gear type	Hertz stress index (GPa)	Number of tests	Number of failures	Number without failure(after $275*10^6$ cycles)
Uncoated	1.7	15	15	0
Coated	1.7	6	1	5
	1.9	8	2	6

Table 6 Summary of Weibull statistical analysis:

Gear type	Weibull slope	Scale (10^6 cycles)	10-percent lives(10^6 cycles)	50-percent lives(10^6 cycles)
Uncoated	1.7	105	28	83
Coated	1.7	673	180	530

From the data plot and the statistical analysis, it is clear that the lives of the coated gears were longer than the lives of the uncoated gears by a factor of approximately 6. To test that the measured life difference was a statistically significant one, the null hypothesis was set forth that the coated and uncoated gears represented a single fatigue life population. If the null hypothesis were true, the observed life difference would have come about from random sampling effects. The null hypothesis was tested using the likelihood ration method and it was found that the null hypothesis can be rejected with greater than 99.5% confidence, a statistically significant difference.

5. CONCLUSIONS

The purpose of the present investigation was to compare the surface fatigue lives of coated and uncoated gears using accelerated life tests. The test gears used were manufactured in one lot form a single heat of consumable-electrode vacuum-melted (CVM) AISI 9310 steel. The gears were case carburized and ground to aerospace quality. A subset of the ground gears was provided with a thin, hard, low-friction carbon-based coating (Me-DLC) by magnetron sputtering, a physical vapor deposition (PVD) process. Tests were conducted using a four square type gear fatigue rig. Tests were run until either of surface fatigue failure of any one gear tooth or until a predetermined number of cycles had occurred with no failure. The following specific results were obtained.

Fifteen tests were completed using the uncoated gears using a load intensity corresponding to a Hertzian stress index of 1.7 GPa (250 ksi).

Fourteen tests were completed using the coated gears. Two loads were used for the coated gears, with six tests conducted at a Hertzian stress index of 1.7 GPa (250 ksi) and eight tests conducted at a stress index of 1.9 GPa (280 ksi).For the uncoated gears, all tests resulted in failure with test durations ranging from 25-272 million revolutions.

For the coated gears, three of the tests resulted in failure while eleven tests were suspended without failure. The test durations ranged from 63-311 million revolutions.

The distributions of the fatigue lives were modeled as two parameter Weibull distributions. From the Weibull analysis, the ten percent lives of the uncoated and coated gear populations were estimated to be $28*10^6$ and $180*10^6$ cycles, respectively. The measured life difference is a statistically significant difference to a greater than 99.5 percent statistical confidence.

REFERENCES

1. Hans, M., Buchel, R., Grischke, M., Hobi, R., Zach,M., 2000, "High-volume PVD Coating of Precision Components of Large Volumes at Low Process Costs", Surface and coatings Technology, 123(2-3) pp. 288-293

2. Grill, A., 1997, "Tribology of Diamondlike Carbon and Related Materials: An Updated Review," Surface and Coatings Technology, 94-95, pp. 507-513.

3. Bewilogua, K., Cooper, C. V., Specht, C, Schroder,J., Wittorf, R., and Grischke, M., 2000, "Effect of target material on deposition and properties of metalcontaining DLC (Me-DLC) coatings," Surface and Coatings Technology, 127(2-3) pp. 224-232.

4. Alanou, M. P., Snidle, R. W., Evans, H. P., and Krantz, T. L., 2002, "On the Performance of Thin Hard Coatings for Gearing Applications," Tribology Transactions, 45(3) pp. 334-344.

5. Townsend, D. P., and Shimski, J., 1994, "Evaluation of the EHL Film Thickness and Extreme Pressure Additives on Gear Surface Fatigue," NASA TM- 106663.

6. Anderson, N., Lowenthal, S., and Black, J., 1984, "An Analytical Method to Predict Efficiency of Aircraft Gearboxes," NASA TM-83716, AIAA Paper 84-1500.

7. Mercer, C, Yao, N., Allameh, S., Evans, A. G., and Cooper, C. V., 2003, Surf. Coat. Technol., in press.

8. Townsend, D.P., 1992, "Improvements in Surface Fatigue Life of Hardened Gears by High-Intensity Shot Peening," NASA TM-105678.