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The Impact of Surface Condition and Lubricant on Gear Tooth Friction

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Abstract

Frictional losses in gear boxes are of significant interest to gear box designers as these losses transform into heat. The direct result is a reduction in the fuel efficiency of the vehicle involved. Further, in many instances, this heat has to be absorbed and dissipated so that lubricant properties and gear box performance are not significantly compromised. This effort is to measure and document the comparative friction losses in a gear mesh due to gear tooth surface condition and lubricant. Three distinct surface conditions are considered. They are ground, isotropic superfinished (REM ISF®) and tungsten incorporated diamond-like carbon coating (W-DLC) which is a wear resistant coating. Two lubricants, MIL-PRF-23699 (ISO VG 22) and Mobil SHC 626 (ISO VG 68) are considered.

The experimental effort is conducted on a high speed, power re-circulating (PC), gear test rig, which had been specially instrumented with a precision torque transducer to measure input torque to the four-square loop. The torque required to drive the loop is measured under various speeds and tooth loads within the torque loop, with test gears with different surface conditions and with different lubricants. Two operating torque levels within the four-square loop at speeds ranging from 4,000 rpm (pitch-line velocity of 19 m/sec) to 10,000 rpm (pitch-line velocity of 47 m/sec) are evaluated.

Input torque measurements, as measured by the precision torque transducer, on ground test gears operating in MIL-23699 lubricant are used as a base line. The increase or decrease in the input torque to the four-square loop is a measure of the change in friction losses at the test gear mesh due to changing surface condition, tooth load and or lubricant. Based on the collected data, a qualitative analysis of the effect of gear tooth surface condition on frictional losses is presented. Further, the surface characteristics of the tooth flanks of the ground, superfinished and coated gears are also described. Plans for future work, to obtain a quantitative measure of the effective coefficient of friction at the tooth surface, are also proposed.

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Introduction

The impact of gear tooth surface quality and treatments on frictional losses in a gear mesh is of significant interest to the aerospace gear community as these losses are converted to heat that has to be dealt with. Further, the impact of lubricant on frictional tooth mesh losses is also of interest. The most exhaustive experimental study quantifying gear tooth friction is by Yoshizaki [1], in which spur gears with various geometries were operated in a power re-circulating test rig and frictional losses were measured. Various lubricants and additives were also evaluated and tooth surface finishes (R_{max}) ranging from 0.5 to 4 μm were considered. In Britton [2], another experimental study, that specifically evaluated the effect of superfinishing on gear tooth friction on a power re-circulating gear test rig is described. A 30% reduction in frictional losses is measured and documented. In another experimental study on gear tooth friction, Petry-Johnson [3] measured frictional losses in a power re-circulating test rig operating ground and chemically polished gears with two different tooth sizes in three different lubricants. This data was further utilized to define guide lines for the design of gear meshes and transmissions. Martins [4] experimentally measured the friction coefficient in FZG (ground) gears utilizing two lubricants. Several attempts to model and predict the friction losses [5], [6], [7] are also evident in literature, where the experimental effort is utilized to correlate to analytical results.

Based on the available literature, a comprehensive experimental study to compare gear mesh friction losses with different tooth surface conditions, different lubricants and under various operating conditions was considered a worthwhile effort. In this study the special variables being evaluated include superfinishing and a W-DLC coating compared to a ground base line. Two lubricants are also evaluated.

Experimental set-up

A high-speed, power re-circulating (four-square) gear test rig was utilized for the purpose of this experimental study. This rig consists of a test gear box connected to a reversing gear box, as shown in Figure 1. An electrohydraulic torque applicator establishes and measures the torque within the four-square loop and consequently the load on the gear teeth. The motor driving the four-square kinematic loop is only supplying the power to overcome the frictional losses in the test gear box mesh and the reversing gear box mesh. This input torque, outside the four-square loop, was measured with a precision, bearing-less, digital torque-meter, under different experimental conditions to establish a comparative measure of the frictional losses in the test gear mesh under those experimental conditions.

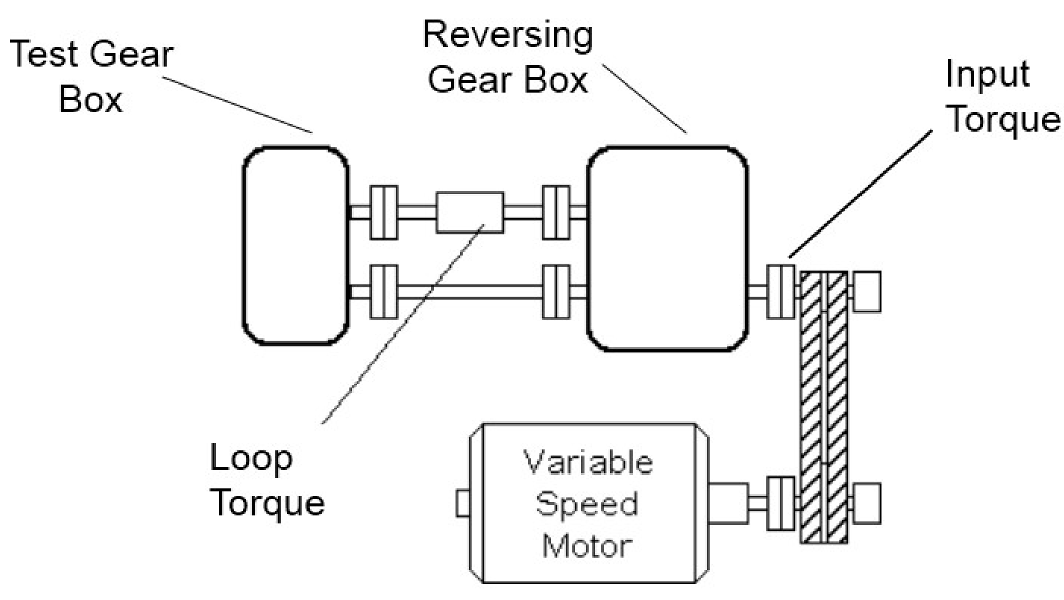

Figure 1. Four-square gear test rig schematic

As stated above the four-square gear test rig consists of a test gear box and a reversing gear box. The reversing gear box consists of very high accuracy helical gears with a face width of 100 mm. The gears in the test gear box are 28 teeth, 3.175 module, 20 degree pressure angle, 6.25 mm face width, spur gears fabricated from AMS 6308 steel, carburized and hardened to 60-64 on the Rockwell C scale. Due to the significant difference in face widths between the gears in the test gear box and the reversing gear box, gear failure in fatigue testing is restricted to the test gear box only. Figure 2 illustrates the test gears mounted in the test gear box with the direction of rotation illustrated by the arrow. Oil jet lubrication was employed in the tests and the "oil into the mesh" nozzle is at the bottom and the "oil out of the mesh" nozzle is at the top in Figure 2.

As the test gear box and the reversing gear box are dissimilar, the total frictional losses cannot be precisely assigned to either of the two gear boxes. However, a comparative estimate of changes in gear tooth frictional losses due to surface condition or lubricant change can be assessed. Further, an arbitrary assignment of frictional losses attributable to the two gear boxes allows an approximate assessment of the changes of frictional losses due to the variables of surface and lubricant.

Test effort

The initial effort focused on characterization of the surface of the test gears. Negatives of the tooth surface were first fabricated using surface replication epoxy (accuracy experimentally verified to be better than 0.1 micron). These replicas were analyzed utilizing optical interferometry to obtain surface characteristics of ground, superfinished and coated gears. The results of the surface characterizations are summarized in Table 1 and are considered to be consistent with what is normally obtained in industry.

A typical data output from one test run is illustrated in Figure 3. The blue line represents the measured loop torque within the four-square test rig and the orange line represents the input torque as measured by the torque transducer on the power input shaft. This particular figure shows the input and loop torques while operating with superfinished gears at 8,000 rpm in SHC 626 lubricant. Depending on the test conditions and the thermal inertia of this test rig, the set up requires up to an hour of operation before the input torque stabilizes for measurement purposes.

The repeatability of the measurements was evaluated. Figure 4 illustrates torque recordings of three different repetitions of superfinished gears operating at 8,000 rpm in SHC 626 lubricant. The range of the stabilized input torque in the three repetitions was 0.09 N-m. This computes to less than +/-1% of the measured torque of 6.95 N-m and was considered acceptable. The tests conducted are detailed in Table 2 with their respective pitch line velocities.

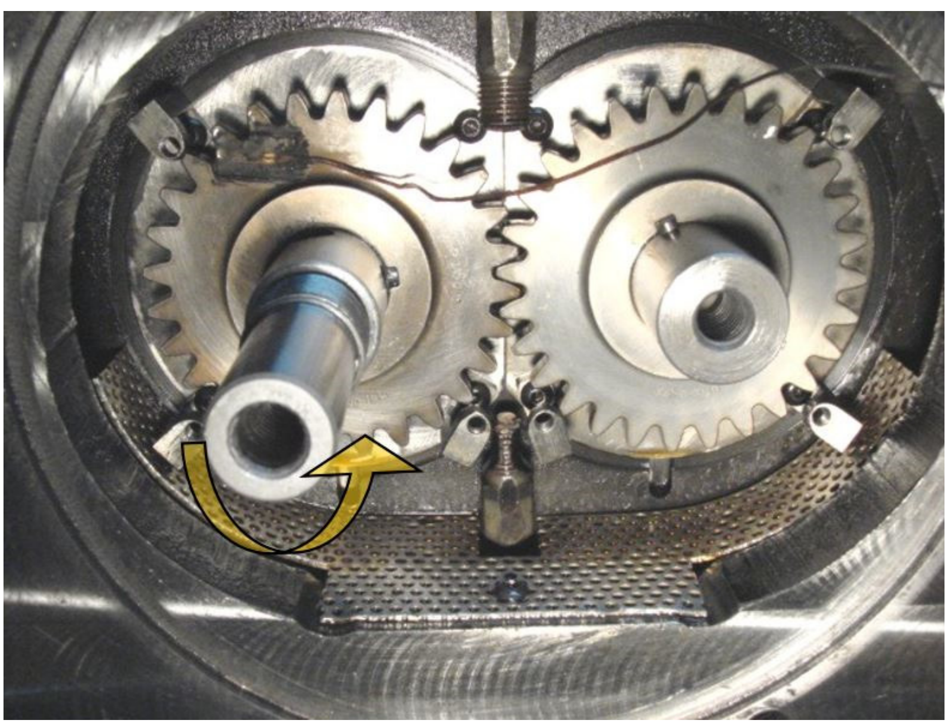

Figure 2. Test gears in gear box with oil inlet nozzles

Table 1. Surface roughness data of test gears

Test pair	S/N	Surface	Ra average	Ra standard development	Rz average	Rz standard development
1	006	As ground	0.241	0.028	1.26	0.11
1	008	As ground	0.257	0.028	1.33	0.22
2	064 (R)	REM ISF	0.084	0.010	0.56	0.16
2	057	REM ISF	0.081	0.015	0.53	0.11
3	054	W-DLC coating	0.084	0.015	0.62	0.12
3	064 (L)	REM ISF	0.084	0.010	0.56	0.16

NOTES:

1. Surface roughness measurements are reported in micrometers.
2. Measurements reported here are directional, taken orthogonal to grinding direction.
3. Averages are computed based upon 4 measurements at 6 similar locations for each sample.

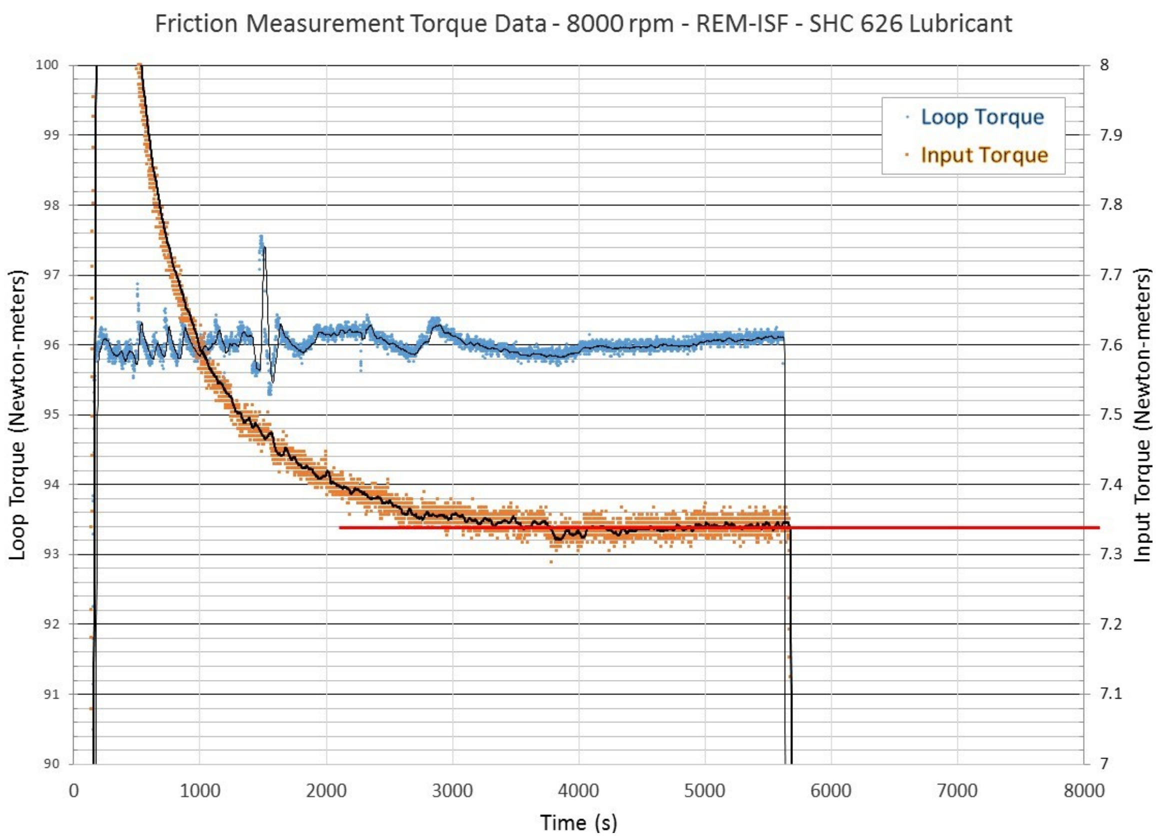

Figure 3. Record of measured torques

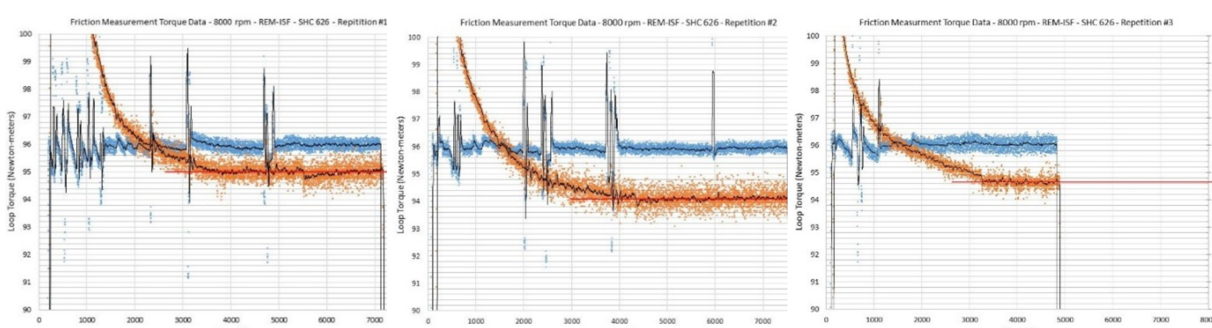

Figure 4. Three repetitive torque measurements

Table 2. Tests details[1)]

MIL-PRF-23699	4000 rpm (18.62 m/s)	8000 rpm (37.24 m/s)	10,000 rpm (37.24 m/s)
96 N-m	1, 2, 3	1, 2, 3	1, 2, 3
192 N-m	1, 2, 3	1, 2, 3	1, 2, 3
Mobil SHC 626	**4000 rpm (18.62 m/s)**	**8000 rpm (37.24 m/s)**	**10,000 rpm (37.24 m/s)**
96 N-m	1, 2, 3	1, 2, 3	2, 3[2)]
192 N-m	1, 2, 3	1, 2, 3	2, 3[2)]

NOTES:
1) 1 – Both gears are as ground;
2 – Both gears are REM/ISF;
3 – Specimen gear is W-DLC coated, mate gear is REM/ISF.
2) Tests 2 and 3 were not conducted as excessive vibration and scoring damage occurred during ground gear testing at prior 10K rpm test.

Results and discussions

In order to provide an adequate tribological basis for the collected data it was decided to compute and document the range of specific film thickness λ for the experimental effort [8]. The bulk temperature of the gear tooth in mesh was interpolated from an earlier experimental effort [9] in order to obtain the lubricant parameter that is required for the computation of the λ ratio. Based on an oil inlet temperature of 40.5ºC, the range of computed λ ratios for the MIL-PRF 23699 lubricant ranged from 0.31 to 2.5. For the SHC 626 lubricant the computed λ ratios ranged from 0.50 to 4.3. The lowest λ ratios are associated with ground gears at high torques and low speeds while the highest λ ratios are associated with superfinished gears at low torques and high speeds. The λ ratios for the coated gears could not be determined due to lack of experimental data on tooth bulk temperature and lack of coefficient of friction data to compute the same.

Tests were conducted with the various gear pairs in the test gear box and under load, speed and lubricants as defined in Table 2. One typical set of results is shown in Figure 5. As can be seen from Figure 5, the ground gears had the highest measured input torque at all speeds at 192 N-m with MIL-PRF-23699 lubricant.

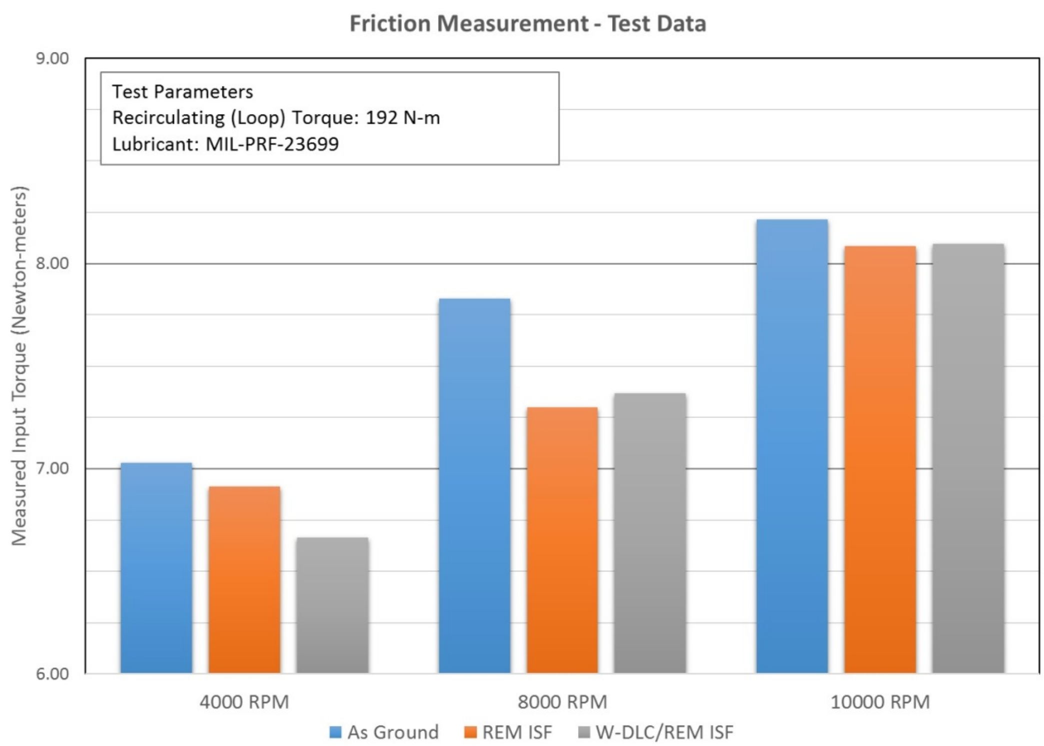

Figure 5. Typical input torque measurements

The results of all the tests conducted are summarized in Table 3. To examine these results analytically, the measured input torque for the ground gear pair was subtracted from the measured input torques for the superfinished and coated gear pairs, under the same load, speed and with the same lubricant. These changes in input torques can be entirely attributed to the change in the surface condition of the gear pair under test and the change in frictional losses at the tooth flank, as all test conditions are otherwise identical. The frictional loss changes range from -0.72 N-m (ID No. 6) to +0.08 N-m (ID No. 2), as shown in Table 3.

The superfinished and coated gears generally required lower input torques (compared to ground) except for one instance where the coated gear had a higher input torque (ID No. 2) than the ground gear set. As this experiment was the first test conducted with the coated gear, some “breaking in” of the coating may have influenced the measurement. If time and budget allowed, the test would be repeated with new gears for better characterization of the break in process or to confirm an anomaly in the data. The ground and superfinished gears were also new at the start of testing. No data was observed that would indicate a similar break in characteristic.

A comparison between superfinished and coated gears was inconsistent. In some instances the superfinished gears had a lower or the same input torque as the coated gear. In some instances the coated gear performed better with a lower loss measurement. The more viscous SHC 626 oil appears to play a greater role in reducing frictional losses at lower speeds and higher loads at the same speeds. The MIL-PRF-23699 appears to more effective at reducing losses at higher speeds and lower loads. As all other conditions are maintained the same, this difference in input torques at each speed, each loop torque and utilizing the same lubricant is entirely due to the changes in frictional losses in the meshing gear teeth mesh.

Table 3. Test data summary showing changes in input torque

ID No.	Surface condition	Loop torque, N-m	Speed, rpm	Lubricant	Input torque change, N-m, relative to as ground at same condition	Reduction compared to as ground (50% model)	Reduction compared to as ground (67% model)
1	REM/ISF	96	4000	MIL-PRF-23699	-0.03	98.8%	98.2%
2	W-DLC	96	4000	MIL-PRF-23699	0.08	102.7%	104.1%
3	REM/ISF	96	8000	MIL-PRF-23699	-0.19	94.3%	91.4%
4	W-DLC	96	8000	MIL-PRF-23699	-0.32	90.7%	85.9%
5	REM/ISF	96	10,000	MIL-PRF-23699	-0.67	82.9%	74.1%
6	W-DLC	96	10,000	MIL-PRF-23699	-0.72	81.4%	71.9%
7	REM/ISF	192	4000	MIL-PRF-23699	-0.11	96.8%	95.1%
8	W-DLC	192	4000	MIL-PRF-23699	-0.36	89.7%	84.4%
9	REM/ISF	192	8000	MIL-PRF-23699	-0.53	86.4%	79.4%
10	W-DLC	192	8000	MIL-PRF-23699	-0.46	88.2%	82.1%
11	REM/ISF	192	10,000	MIL-PRF-23699	-0.13	96.8%	95.2%
12	W-DLC	192	10,000	MIL-PRF-23699	-0.15	97.1%	95.6%
13	REM/ISF	96	4000	Mobil SHC 626	-0.18	94.1%	91.1%
14	W-DLC	96	4000	Mobil SHC 626	-0.14	95.6%	93.3%
15	REM/ISF	96	8000	Mobil SHC 626	-0.27	92.5%	88.6%
16	W-DLC	96	8000	Mobil SHC 626	-0.31	91.6%	87.2%
17	REM/ISF	192	4000	Mobil SHC 626	-0.28	92.4%	88.5%
18	W-DLC	192	4000	Mobil SHC 626	-0.29	92.1%	88.0%
19	REM/ISF	192	8000	Mobil SHC 626	-0.38	90.6%	85.7%
20	W-DLC	192	8000	Mobil SHC 626	-0.37	91.0%	86.4%

As the input torque measurement includes the losses in the reversing gear box, which is very dissimilar to the test gear box, an assumption on the amount of losses in the reversing gear box was necessary in order to compute the input loss change as a percentage of the loss with a pair of ground gears. Splits of 50% split and 67% in the losses between the test gear box and reversing gear box were assumed, with the 67% split being more appropriate (large face width, helical gears in the reversing gear box). The percentage reductions in losses are listed in Table 3 based on the loss split assumption model. Based on a 67% split model, superfinishing alone provided a reduction in frictional losses of up to 26% (ID No. 5) while the addition of this coating increase this reduction to 28% (ID No. 6).

The test tooth surfaces were characterized after testing with the same tooth negative optical interferometry technique that was used for pre-test inspection. A reduction in surface finish (Ra) of approximately 0.005 to 0.010 microns was observed for both the superfinished and the W-DLC coated gears. A similar reduction was measured for the corresponding superfinished mate gears. The ground gears were damaged during the 10,000 RPM testing. As expected, the post-test surface characterizations of the ground gears showed an increase in roughness of 0.02 to 0.07 microns (Ra).

Conclusions

The impact of surface treatments such as superfinishing and coating on frictional losses is of significant interest since this loss is converted into heat that has to be accounted for. This experimental effort described above demonstrates that these surface treatments can reduce frictional losses by as much as 28% over ground gears, based on an assumption of the loss split between the two gear boxes on the four-square test rig. Considering that this reduction can be obtained at each gear pair and most gear boxes have many gear meshes, the total impact on the heat generated by the gear box can be significant. It is difficult to state, based on this study if W-DLC coating has any added benefit on frictional losses, though it may improve contact fatigue life and oil out performance of the gear pair.

In order to obtain a more precise estimate of the impact of the surface treatments on gear losses it is necessary to obtain an accurate measure of the losses in the reversing gear box. It would then be possible to isolate the actual losses in the test gear box. From a measurement of total losses a more precise estimate of changes in frictional losses due to surface treatments could be estimated. Identifying the losses solely in the test gear box and eliminating other losses would also lead to an estimate of the effective coefficient of friction at the tooth flank and the impact of the surface treatment on this parameter. This effort, to measure the actual losses in the reversing gear box of the four-square test rig and estimate the effective coefficient of friction at the test gear tooth flank, is currently underway.

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