**Assessing the Friction and Wear Performance of Self-lubricating Coatings in Dry Aerospace Actuation Gearboxes**

Thomas Finch\*1 2, Tomas Vitu 3, Tomas Polcar 1

Report number: CLS24391808

\* Corresponding Author: tmhf1a14@soton.ac.uk

1 nCATS, Faculty of Engineering and Physical Sciences, Building 5, University of Southampton, Highfield Campus, Southampton, SO17 1BJ, UK

2 Actuation Systems, Collins Aerospace, Wolverhampton, UK, WV10 5EW, UK

3 Faculty of Electrical Engineering, Czech Technical University in Prague, Technicka 2, 166 27 Prague 6, Czech Republic

**Keywords:**

Spur gear

Dry wear

Dry friction

Self-lubricating coating

**Abstract:**

Aerospace actuation gearboxes operate in low temperature environments where increased lubricant viscosity leads to significant no-load power losses. Replacing fluid lubricants with coatings applied to the gear teeth is one potential approach to improving gearbox efficiency. Here we develop an approach to determining average wear rates of coated gears using a power-recirculating test stand, profile measurements, and a model of the tooth contact. Worn gears are inspected using scanning electron imagery, and energy dispersive x-ray and Raman spectroscopy to understand the wear mechanisms and failure modes. Average coefficients of friction are determined at 20°C and -40°C using a power-absorbing test stand and isolation of tooth friction losses by calculation. These methods are then demonstrated on a promising C/Cr composite coating.

1. **INTRODUCTION**

The majority of gearboxes are designed to prevent wearing of the gear teeth through the use of fluid lubricants. Fluid lubricants prevent direct contact between the asperities of meshing gear teeth by maintaining a separating film between the two surfaces and help to reduce surface stresses and friction. They also contain chemical additives which, when asperity contact does occur, form easily sheared films to separate the metals and prevent wear.

The aerospace industry utilises fluid-lubricated gearboxes to actuate various control surfaces and utility systems on both civil and military aircraft. However, in the low temperature flight environment the high viscosity of the fluid causes significant power loss, and pressure and temperature variations make the sealing of dynamic interfaces challenging. This leads to both an increase in system weight and a reduction in reliability. A dry lubricated approach to geared actuation has the potential to reduce no-load losses and system weight, and to eliminate problems with leakage.

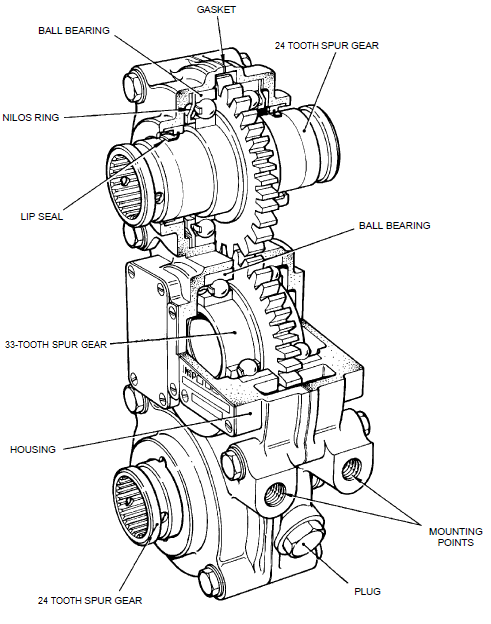
Modern physical vapor deposition (PVD) processes allow metal substrates to be coated with tailored nanocomposite films that combine the beneficial properties of multiple elements. This has produced a generation of wear-resistant coatings with the hardness and low friction characteristics of thin hard films, and the self-lubricating properties of softer lamellar materials.

PVD films, particularly those incorporating carbon, have been previously tested on gears to understand their impact on resistance to scuffing [1-7] and pitting [2-4, 8-10] in conventionally lubricated gearboxes. These have often focussed on the potential for coatings to be used to replace environmentally damaging lubricant additives in conjunction with biodegradable oils. Although the performance of dry PVD coatings on gears has been simulated on disc machines in mixed rolling and sliding [11-12], very little data is available for the performance and durability of these coatings when actually applied to gear teeth. The most notable effort came from Petrik et. al [13], who studied the dry performance of gear teeth coated with a:C-H:Me and a:C-H films on various substrates, including those pre-treated by nitriding. These tests were performed using a typical “load stage” approach, with the gears subjected to a fixed number of revolutions at increasing torque levels until seizure of the surfaces was evident. Little focus has been placed on understanding the evolution and distribution of wear on the tooth surfaces, the wear processes, or the extent of the performance benefit gained from the reduction of no-load losses.

This study develops methods for quantifying the wear and friction performance of solid lubricant coatings when applied to gear teeth as a dry lubricant in aerospace actuation gearboxes. These methods are then applied to a promising carbon-chromium composite coating.

1. **MATERIAL AND METHODS**
   1. **Test Samples**

All tests were performed on coated gears assembled into a gearbox from a civil high-lift actuation application. The gearbox consists of three spur gears: a pair of identical 24-tooth input and output gears, and a central 33-tooth idler gear. Each gear is straddle mounted on a pair of deep groove ball bearings and housed within a set of cast and machined aluminium alloy housings. See figure 1.



*Figure 1: Sectional View of Test Gearbox*

The test gears were manufactured in BS S82 and gas carburised to a minimum surface hardness of 650Hv2.5 and a case depth of 0.5mm to 510Hv2.5. The tooth surface roughness was improved by wet superfinishing in a disc finishing machine to 0.1µm Ra, with the finished tolerances conforming to AGMA 2000-A88 class 12. The gear design parameters are summarised in table 1.

*Table 1: Test Gear Design Parameters*

|  |  |  |  |
| --- | --- | --- | --- |
| Parameter (units) | Designation | Input/Output Gear | Idler Gear |
| Number of Teeth | *z* | 24 | 33 |
| Module (mm) | *m* | 2.250 | |
| Pressure Angle (°) | α | 20 | |
| Modification Coefficient | *M* | 0 | |
| Face Width (mm) | *b* | 6.5 | |
| Centre Distance (mm) | *C* | 64.125 | |

The tested coating, Graphit-iC, was applied to the gears by Teer Coatings using closed field unbalanced magnetron sputtering (CFUBMS). It consists of a thin Cr interlayer to provide adhesion, followed by a transition layer in which the concentration of Cr is gradually reduced while the concentration of C is gradually increased. The final 2.8µm of the coating consists of Cr-doped (~5 at.%) amorphous carbon (a-C). Further details of the deposition process of the coating, and its performance in standard pin-on-disc tests have been documented by Yang et al. [14,15].

* 1. **Wear Testing** 
     1. **Wear Test Setup and Conditions**

Wear testing was performed on a test rig designed on the principal of power recirculation. Power-recirculating test rigs, sometimes referred to as “back-to-back” or “four-square” test rigs, are utilised widely in industry for testing the scuffing, pitting, and wear resistance of gears and lubricating oils.

Power-recirculating rigs function by mounting two gearboxes in-line, with the inputs and outputs connected by common shafts. A torque load is then trapped between the gearboxes using a slip coupling and torsion bar. One of the common shafts is driven by a motor, which results in one gearbox (the “test” gearbox) operating against an opposing torque, whilst the other (the “reversing” gearbox) operates with an aiding torque. The advantage of this approach is that the motor only drives the losses in the system. Consequently, a relatively small motor can be used to drive a highly loaded gearbox, and no separate load motor or control system is required.

For the test rig used in this study, both the test and reversing gearboxes used the same design shown in figure 1. The coated gears were installed to the test gearbox without any fluid lubrication. The supporting bearings were sealed to prevent egress of grease. For the reversing gearbox, uncoated gears were installed, and the gearbox was lubricated with ISO VG 150 EP gear oil to inhibit wear.

The applied torque and shaft speeds were monitored during testing using a combined torque transducer and encoder. Torque loads were applied to the torsion bar by first earthing the slave gearbox with a splined shaft and disconnecting the slip coupling, then suspending masses from an attachable moment arm. Once the applied torque load was verified by the torque transducer, the slip coupling was retightened to trap the torque in a loop, before removing the earthing shaft.

Diagram, engineering drawing

Description automatically generated

*Figure 2: Wear Test Rig*

The applied test duty was based on the normal operating conditions for the gearbox in its real-world application. Test cycles consisted of bursts of 100 revolutions at a speed of 400rpm, with a 165 second period between cycles to allow for dissipation of heat. All cycles were applied in the same direction of rotation. In a typical flight control actuation system, gearboxes of the type used in this experiment would operate against constantly varying torque loads. However, in order to better understand the impact of changes in torque load individual gearsets were tested against constant opposing torques of 10Nm, 20Nm, and 30Nm. An approximate equivalent life requirement for a gearbox of this type would be 120,000 test cycles at a mean torque of 20Nm.

* + 1. **Measurement and Inspection of Wear**

Prior to commencing testing 3 equally spaced teeth on each gear were marked with identifying numbers, then measured on a Klingelnberg P26 gear measuring centre. 7 equally spaced profile measurements were taken, with a single lead measurement to allow mapping of the tooth topography. The lead measurement was taken at the start of the profile trace, which was positioned below the start of active profile (SAP) to form a datum.

Testing was paused at set intervals to allow the measurements to be repeated. Following disassembly of the test gearbox the teeth were blasted with filtered air to remove any loose debris. No further cleaning was performed to ensure that any transfer films were not disturbed. Comparisons to the pre-test measurements were then used to quantify and map the wear distribution on each tooth. To monitor the accuracy of the measurements one uncoated sample of each gear type was retained as an artifact, and then rechecked at each inspection of the test gears. 95% of all data points remained within 0.27µm of the original pre-test measurements.

Each of the monitored teeth was also inspected and photographed under magnification to record visual evidence of coating damage or failure. Testing was discontinued once coating penetration could be visually verified by the presence of significant brown staining from oxidation of the substrate on multiple teeth.

* + 1. **Determination of Wear Rate**

The performance of a material couple in sliding wear is often quantified in the form of a wear rate, defined according to Archard’s Law. This states that the volume of material worn from rubbing surfaces is proportional to the applied normal force and the sliding distance [16]. When considering a coating, where the useful life is dependent on the thickness, this can be rearranged to consider the wear depth instead of volume:

(1)

*h* = wear depth

*k* = wear coefficient

*σ* = contact pressure

*s* = sliding distance

As the curvature, loads, and sliding velocities of the contacting gear teeth vary during the mesh cycle, the sliding distance and contact pressure must be calculated for each position on the tooth surfaces. The following method simplifies the contact between the gear teeth to a pair of loaded rollers of constantly changing radius and velocity and was originally proposed by Flodin and Andersson [17].

During the meshing cycle of the teeth the contact travels through a straight line of action that is cotangent to the base circles of the involutes. For each position on the line of action, where the pitch point is zero, the radius of curvature on the pinion (1) and wheel (2) can be found by:

(2)

(3)

*R* = radius of curvature

*dp* = pitch diameter

α = pressure angle

ξ = position on line of action

At different positions in the mesh cycle the load is carried by either one or two pairs of teeth. The regions of single and double tooth contact are established from the length of the line of action and the base pitch. When a single tooth pair is in contact the force per unit face width is found by:

(4)

*F* = force per unit face width

*T* = torque

*b* = face width

*db* = base diameter

At positions where two tooth pairs are in mesh the load share is determined by the stiffness of teeth, the position of the point of contact on the tooth surface, and the profile deviations. These deviations will be present due to inaccuracies in the manufacturing process but will also evolve during operation due to wear. As the points of contact between tooth pairs on the line of action are separated by a distance equal to the base pitch, the load share can be calculated by:

(5)

(6)

(7)

*CΣT* = tooth deflection constant

*pb* = base pitch

The tooth deflection constant is related to the modulus of elasticity of the material and is the combined stiffness of each tooth in bending. Treating each tooth pair as two springs in series:

(8)

Tooth deflection constants were selected from the work of Wang and Cheng [18], who studied variations in tooth stiffness by finite element analysis and produced a series of curves that related the deflection to the number of teeth and loading position. These curves were used to determine the deflection constants for the model, based on the stiffness when loaded at the pitch circle.

Having determined the radius of curvature and normal forces at each point, the contact conditions may now be calculated by treating the teeth as parallel cylinders in contact using Hertz’s formulae. The contact half width is calculated from:

(9)

(10)

(11)

*a* = contact half width

*R*\* = equivalent radius of curvature

*E*\* = equivalent modulus of elasticity

*ν* = Poisson’s ratio

The mean contact pressure is then:

(12)

The sliding distance is found by continuing to treat the contact as a pair of loaded cylinders in mixed rolling and sliding, and using the principle of single point observation as described by Andersson [19]:

(13)

(14)

(15)

(16)

*u* = peripheral velocity of tooth surface

Hence if the contact pressure and wear rate are considered as constant across the contact, the wear depth at a point *p* on the tooth surface after a single mesh cycle is:

(15)

(16)

This model can be extended to calculate the cumulative wear over many cycles by breaking up the total duty into discrete periods. This gives the following expression for the wear at a point *p* after *n* cycles:

(17)

A model was developed based on the approach described, with pre-test measurements of the test gears used to determine the starting conditions for *h*. On completion of testing the final set of measurements were used to determine the distribution of wear on the tooth surfaces. The average wear rate was then determined by adjusting its value to give a combined least-squares fit between the model outputs and the test measurements. The worn and pre-test profiles were generated based on the average of the 7 equally spaced profiles. It should be noted that as coating penetration occurred during testing, the reported wear rate is for the combined coating and substrate. An example comparison between the model and test data is shown figure 3. A schematic diagram of the model is shown in figure 4.

Chart, line chart

Description automatically generated

*Figure 3: Example of Predicted Wear Distribution fitted to Measurements from Tested Gears (Colour)*

*Diagram

Description automatically generated*

*Figure 4: Wear Model Schematic*

* + 1. **Post-Test Inspections**

On completion of the final set of measurements, the three monitored teeth were sectioned from each gear. After a final visual examination, the teeth were inspected using a Tescan Vega 3 scanning electron microscope (SEM), using both secondary and backscatter electron detectors. Material analysis was performed on areas of interest using an Oxford Instruments INCA energy dispersive x-ray spectroscopy (EDS) system. Further analysis of the worn and as-deposited coating was performed using a Horiba XploRA Raman spectrometer with laser excitation at 532nm.

* 1. **Efficiency Testing**
     1. **Efficiency Test Setup and Conditions**

Efficiency tests were performed to determine the frictional characteristics of the coated gears, and to provide a comparison of gearbox performance to those lubricated by common aerospace fluid lubricants. Testing was conducted on a power-absorbing test rig. The rig consisted of separate load and drive electric motors with torque transducers positioned to measure the input and output torques. The input torque transducer was a dual range type with separate outputs for 5Nm and 50Nm ranges to allow precise measurement of no-load and loaded input torques. This test setup was adopted in place of the power-recirculating wear test rig to allow the losses within the test gearbox to be isolated, and to allow the application of linearly increasing loads to generate efficiency profiles.

Testing was performed at 20°C and at -40°C. Temperature requirements for aerospace actuation equipment in unsealed zones generally specify -40°C as the lower temperature limit at which system performance requirements must be met, though lower temperatures are often specified as survival conditions. Low temperature tests were performed by surrounding the test gearbox with an insulated enclosure and circulating air with a centrifugal blower. Cooling of the air was performed using liquid N2 in a separate mixing chamber. T-type thermocouples were used to monitor the temperature of the gearbox, with a 2-hour cold soak period being implemented once the target temperature was reached to ensure uniform temperature distribution throughout the unit. This test setup is shown schematically in figure 5.

Diagram

Description automatically generated

*Figure 5: Test Setup Schematic for Efficiency Testing at Low Temperature (Colour)*

Tests were performed by running the gearbox at 400rpm against zero load until the input torque had stabilised, then applying a linearly increasing torque load. This test was repeated using uncoated gears, with the gearbox lubricated with common aerospace actuation fluid lubricants for comparison. The selected fluids were MIL-PRF-7808 turbine oil and AMS-3057 semi-fluid grease. The gearbox was filled sufficiently to immerse the lower 25% of the input and output seal lips.

During testing at -40°C the fluid lubricants exhibited a time dependent decay of the no-load losses prior to stabilising. As the starting conditions are often key to meeting performance requirements in actuation systems, which typically operate only for brief periods at a time, the gearbox was additionally operated against a range of discrete starting torques. During these tests the efficiency was measured immediately after the speed had stabilised in order to assess the impact of the initial, rather than steady state, no-load losses when the net efficiency is at its minimum. The initial no-load losses were then found by performing separate unloaded movements.

* + 1. **Calculation of Tooth Friction**

Power losses in gearboxes can be separated into the load-dependent and no-load losses. The load-dependent losses are caused by friction between the gear teeth and frictional moments of the bearing rolling elements. The no-load losses are caused by friction from gearbox and bearing seals, and churning losses of the lubricant. See equation (18), taken from [20].

(18)

*Pgbx* = gearbox power losses

*Pgl* = load-dependent power losses from gears

*Pg0* = no-load power losses from gears

*Pbl* = load-dependent power losses from bearings

*Pb0* = no-load power losses from bearings

*Ps* = no-load power losses from seals

The different components of the overall power loss can be determined either through experiment or by calculation in order to isolate the frictional losses from the gears. The combined no-load losses from the gears, bearings and seals during the experiment were found from the stabilised input torque readings when the output load was at zero.

(19)

*Pgbx0* = gearbox no-load power losses

The load dependent losses from each bearing were determined by calculation. This utilised the method defined by SKF [21], where the total power losses at the bearings are defined as the sum of sliding and rolling friction moment losses, drag losses, and seal losses.

(20)

*Mt* = bearing friction torque

*M’rr* = rolling friction moment

*Msl* = sliding friction moment

*Mdrag* = drag losses

*Mseal* = seal losses

Both the drag and seal loss terms are independent of the applied load, and hence these are included with the zero load drag measurements and only the rolling and sliding friction moment terms required calculation to determine the value of *Pbl*.

With all other terms having been determined, the load dependent losses from the gears *Pgl* can be calculated from the overall power losses measured during the test. The average coefficient of friction during the meshing action may then be calculated according to Ohlendorf [22] from eq. (21).

(21)

*Pi* = input power

*Hv* = loss factor

*µa* = average coefficient of friction

The loss factor, according to Buckingham [23], is given by eq. (22)

(22)

*u* = gear ratio

*z1* = number of teeth on pinion

*βb* = base helix angle

*εα* = transverse contact ratio

*k0* = gear geometry factor

The gear geometry factor k0 is calculated from eq. (23)

(23)

*ra2* = gear tip radius

*rp2* = gear pitch radius

*αt*= transverse pressure angle

As the test gearbox includes two gear meshes, the overall no load losses of the gearbox were split evenly between the two meshes, and then the load-dependent losses were calculated in series. A single average coefficient of friction for both meshes was then calculated to match the input and output torques from the test data.

1. **RESULTS AND DISCUSSION**
   1. **Wear Test Results and Discussion**

On the vast majority of teeth and at all load cases the first 100 to 500 test cycles resulted in localised wear around the SAP on the teeth. However, during subsequent cycling this localised root wear deepened very slowly and remained bright and reflective. The overall contact area was clearly visible as a darker and more reflective region when compared with the as-deposited coating. A line of unworn, unreflective coating was visible at the pitch circle.

With further operation most samples began to display the anticipated wear pattern. Wear at the pitch circle was negligible, whilst in the dedendum the wear was deeper with increasing distance from the pitch circle, eventually blending with the root wear, which had broadened both across the face width and up the profile. On the addendum the wear tended to be highest approximately midway between pitch circle and the tips.

In most cases when testing at 10Nm and 20Nm the first evidence of coating failure was observed on the mid-addendum. This was initially evident as bright, reflective lines that were oriented across the face of the tooth. With further cycling these regions increased in size and developed a brown colouration, indicating complete removal of the coating and oxidation of the underlying substrate. At 30Nm coating failure occurred on the tooth addendum of one gear, and on the contacting point on the dedendum of the mating tooth. However, there were also instances on some teeth of coating failure and oxidation at the SAP and tips. Figure 6 shows the progression of wear on a typical tooth from testing at 20Nm.

Diagram

Description automatically generated with medium confidence

*Figure 6: Example Wear Maps, Visual Inspection, and Wear Rate Calculation for Monitored Tooth during Testing at 20Nm (Colour)*

As anticipated, increasing the applied torque load resulted in more rapid failure of the coating. The initial root wear, in terms of area and depth, was more severe at higher loads, but the calculated average wear rate for the post-test measured profiles showed reasonable consistency across the load range. The highest wear rates (4.25E-11 mm3/N.mm) occurred during cycling at 10Nm, whilst the lowest wear rates (2.34E-11 mm3/N.mm) occurred during cycling at 20Nm However there was significant overlap in the range of wear rates measured at different loads. Figure 7 shows the range of wear rates for each load at the input and output meshes of the gearbox on the monitored gear teeth. Figure 8 displays the maximum wear depth against test cycles, using the average of the maximum wear depth measured on each monitored tooth in the gearbox. When assessing the correlation in wear distribution between the measured components and the model output it was noted that the magnitude of addendum wear was often greater than predicted in comparison to the dedendum. Refer to figure 3.

Chart, waterfall chart

Description automatically generated

*Figure 7: Range of Calculated Post-Test Wear Rates by Torque Load and Mesh Position*

Chart, scatter chart

Description automatically generated

*Figure 8: Maximum Wear Depth vs. Test Cycles (Colour)*

During the post-test SEM inspection, backscatter imaging was used to further visualise the distribution of wear to the tooth surfaces. Due to the greater atomic number of the underlying substrate and the subsurface emission of backscattered electrons, the images display more heavily worn areas of the tooth with a brighter shade. The worn areas around the SAP were sharply contrasted with the adjacent coated areas, indicating that the process of initial damage to the coating is one of delamination rather than sliding wear. Areas of coating failure on the addendum were almost indistinguishable from the surrounding coating, which also appeared to be heavily worn. This is more consistent with a process of gradual material removal by sliding wear. The images also highlighted bands of wear variation extending across the faces of the teeth, with a large band of low wear evident around the pitch circle. Figure 9 provides a comparison of optical and backscatter imagery for a tooth at the end of testing.



*Figure 9: Post-Test Microscopic and SEM Backscatter Imagery of Graphit-iC Coated Tooth (Colour)*

Areas of interest were studied further using secondary electron imagery. Figure 10 shows the regions of coating damage at the SAP and mid-addendum of the tooth shown in figure 9. At the SAP the lower edge of the worn area was marked with an abrupt boundary between coating and exposed substrate, indicating that the coating in this region had suffered from delamination. The upper edge of the area was more irregular in shape, and the edges of the coating were not visible. The edges from the initial delamination appeared to have been blended by the subsequent sliding wear. The region immediately above the exposed substrate was pockmarked with round delaminated areas, mostly 10µm to 20µm in diameter. This may have been due to debris impacts from the initial failure. The exposed substrate was marked with scratches oriented in the direction of sliding.

On the region of mid-addendum coating failure both the upper and lower edges were jagged and irregular in appearance, with the peaks and troughs oriented in the direction of sliding. The brown-stained areas of the exposed substrate were covered in a thick oxide layer, which was cracked and spalling away in places. The edges of the coating were not visible. Some small patches of intact coating remained within the zone of failure and were also oriented in the direction of sliding.

A picture containing background pattern

Description automatically generated

*Figure 10 – SEM Secondary Electron Imagery of Wear to Addendum and SAP (Colour)*

EDS analysis for the as deposited coating and underlying substrate detected 54% by weight C, 22% Cr, 20% Fe, with 1-2% O, Ar, and Ni. Partially worn areas showed a reduction in the detected C content, and a corresponding increase in the detected Cr and Fe due to the wear of the coating bringing more of the Cr interlayer and steel substrate within the penetration depth of the electron beam. In heavily worn areas on the addendum where the coating was no longer visible the EDS continued to detect 6-11% C. The presence of O varied significantly between 32% in regions with thick oxide layers, to as little as 2% in others. In and around the heavily oxidised regions very little C or Cr content was present: 6-9% and 1-6% respectively. In the delaminated regions on the SAP, although the C content was also greatly reduced, all areas examined continued to exhibit >17% Cr. This may provide an explanation for the absence of brown staining in these regions: sufficient Cr was present to protect the surface from gross oxidation. It may also indicate that the apparent delamination had not occurred between the interlayer and substrate, but between the interlayer and transition layer instead.

Post-test Raman spectra from for the as deposited and partially worn coating are displayed in figure 11. The spectra have been deconvoluted into multiple peak fits using gaussian curves.

**Chart, line chart, histogram

Description automatically generated**

*Figure 11 – Raman Spectra (532nm Excitation)*

*As-deposited and Worn Graphit-iC Coating (Colour)*

Raman spectra of disordered C are dominated by two bands [24]. The G (graphite) band is attributed to in-plane stretching of pairs of C atoms and can be found between 1500cm-1 to 1600cm-1. In the spectra for the coated gear teeth, the partially worn coating exhibited an increase in the G peak wavenumber from 1567cm-1 to 1583cm-1 when compared to the as-deposited coating. As the sp2 content in a-C increases the size of sp2 clusters also increases. The longer chains result in higher vibrational frequencies. The worn coating’s increase in G peak wavenumber indicates that the wear process has caused a conversion of sp3 bonding to sp2, and a transition towards a nanocrystalline graphite structure.

The second band prominent band, the D (disorder) band, is found around 1335 cm-1 and is caused by breathing of sp2 rings. However multiple peaks of significant intensity were required to obtain satisfactory fitment to the spectra in this region. Consequently, the D band position and intensity could not be accurately determined.

* 1. **Efficiency Test Results and Discussion**

During steady state efficiency testing at room temperature, the no-load losses for both the coated and fluid-lubricated gearboxes were broadly similar, but the load-dependent losses on the coated gears were higher. Consequently, the coated gears exhibited a slightly lower net efficiency across the load range than the fluid-lubricated gears.

At -40°C the no-load losses became far more significant and were the dominant source of power loss. Whilst the load dependent losses of the coated gears were again higher than those of the fluid lubricated gears, due to the lower no-load losses on the coated gears the net efficiency was higher. The efficiency benefit became smaller with increasing load, and by 40Nm the net efficiencies for all three lubrication regimes were roughly equal. Due to the effects of kinematic starvation and inlet shear heating, the increased base oil viscosity within the bearings resulted in a reduction of the M’rr and Msl terms, to the point that the load-dependent losses in the bearings were calculated to be negligible during the tests at -40°C. During the initial efficiency tests, the benefit of the Graphit-iC coated gears was even more pronounced. The no-load losses formed a more significant proportion of the overall losses for the fluid-lubricated gears; hence a net efficiency benefit was maintained over the entire range of output loads tested. The net efficiencies are plotted against output load in figure 12. Figure 13 provides a breakdown of the different sources of power loss at discrete torque loads for each lubricant.



*Figure 12 – Efficiency Curves for Dry-Coated and Conventionally Lubricated Gears (Colour)*

A screenshot of a computer

Description automatically generated with low confidence

*Figure 13 – Power Loss Contributions (Colour)*

Both fluid lubricants exhibited lower coefficients of tooth friction at low temperature, with the lowest values being achieved for MIL-PRF-7808 at -40°C. This behaviour is consistent with greater film thicknesses due to increased lubricant viscosity, and a transition from a boundary or mixed lubrication regime towards a hydrodynamic regime. For the Graphit-iC lubricated gears this relationship was inverted, with the coefficient of friction increasing at -40°C by up to 16.4%. Similar behaviour has been observed in other studies [25] and has been attributed to the inhibition of the transfer film formation and graphitization processes, which benefit from high flash temperatures in the contact and are responsible for the low friction properties DLC coatings. Additionally adsorbed water from ice layers formed during the cold soak period may be drawn into the contact by capillary action resulting in additional viscous drag. The coefficients of friction at 20°C and -40°C for 20Nm torque load are shown in Figure 14. The output torque to load-dependent loss relationship was highly linear in all cases.

Chart, bar chart

Description automatically generated

*Figure 14 – Tooth Average Coefficients of Friction (20Nm) (Colour)*

* 1. **Future Work**

As the most common cause of failure was found to be sliding wear on the mid-addendum, and the extent of this wear is not predicted by the rolling cylinder model, efforts should be made to better understand the contact conditions at this point in the mesh cycle.

In terms of the coating itself, an obvious improvement would be to simply increase the thickness to extend its life. However, increasing the thickness of the coating would also lead to an increase in intrinsic stresses, which may exacerbate the issues with delamination. Therefore, an increase in coating thickness should also be accompanied by measures to improve coating adhesion, whether through changes to the coating interlayer, or improvements to the pre-deposition cleaning procedures. The early delamination of the coating at the SAP could potentially be addressed by the introduction of micro geometric tooth profile modifications, specifically tip relief, to ease the loads at the point of tooth engagement and disengagement.

As the efficiency testing showed an increase in tooth friction when tested at low temperature, the implications of low temperature operation with regards to wear rates also requires further study. Long-term wear testing at low temperature requires a significant increase in test setup complexity and monitoring requirements. However, in order to confirm suitability for aerospace applications any temperature sensitivity with respect to the useful life of the coating needs to be quantified.

Similarly, this study covers operation at a single gearbox speed. Efficiency and wear testing should be expanded to cover a range of speeds to determine any effects on friction or wear performance.

1. **CONCLUSIONS**

* Whilst the tested gears could potentially have continued to operate satisfactorily for a further period following partial coating failure, the useful life of the coating is far less than required for this gearbox in its real-world application. The coating does, however, show potential for military applications where flight cycle requirements are lower, and with improvements it is possible that the required life could be met. The primary cause of failure for the coating and operating conditions tested in this study are delamination at the SAP and sliding wear on the addendum for both driven and driving gears.
* The results show the value of conducting periodic dimensional inspections during wear testing to understand the evolution of wear distribution to the tooth surfaces. The rolling/sliding cylinder model is a useful tool for calculating the average wear rate of coated gears but requires refinement due to the tendency to underestimate the extent of wear on the addendum.
* Adopting a dry lubricated approach may offer a significant benefit to overall efficiency, particularly for gearboxes that are required to operate at low temperatures for brief periods, and where the system starting efficiencies are significant design drivers. The extent of this benefit may be quantified and understood through loaded efficiency testing and isolation of the sources of power loss by calculation.

1. **ACKNOWLEDGEMENTS**

The authors would like to thank Teer Coatings for performing the coating deposition and Collins Aerospace for funding the manufacture of the test rig and samples. This work was partially supported by the MEYS CR project "Novel Nanostructures for Engineering Applications" No. CZ.02.1.01/0.0/0.016 026/0008396.

**REFERENCES**

1. B. Kržan, M. Kalin, J. Vižintin, “The Lubrication of DLC Coated Gears with Environmentally Adapted Ester-Based Oil”, *Gear Technology*, July/August, pp 36-40, 2006.
2. F. Joachim, N. Kurz, B. Glatthaar, “Influence of Coatings and Surface Improvements on the Lifetime of Gears”, *Gear Technology*, pp 50-56, July/August 2004.
3. M. Weck, O. Hurasky-Schönwerth, C. Bugiel, “Service Behaviour of PVD-Coated Gearing Lubricated with Biodegradable Synthetic Ester Oils”, *Proceedings of the International Conference on Gears*, 2002
4. R. Michalczewski, W. Piekoszewski, M. Szczerek, W. Tuszynski, “The Lubricant-Coating Interaction in Rolling and Sliding Contacts”, *Tribology International*, Vol. 42, pp 554-560, 2009
5. R.I. Amaro, R.C. Martins, J.O. Seabra, S. Yang, D.G. Teer, N.M, Renevier, “Carbon/Chromium Low Friction Surface Coating for Gears Application”, *Industrial Lubrication and Tribology,* Vol 57, Iss 6, pp 233-242, 2005
6. R. Martins, R. Amaro, J. Seabra, “Influence of Low Friction Coatings on the Scuffing Load Capacity and Efficiency of Gears”, *Tribology International*, Vol. 41, pp 234-243, 2007
7. M. Kalin, J. Vižintin, “The Tribological Performance of DLC-Coated Gears Lubricated with Biodegradable Oil in Various Pinion/Gear Material Combinations”, *Wear*, Vol. 259, pp 1270-1280, 2005
8. M. Fujii, M. Seki, A. Yoshida, “Surface Durability of WC/C-Coated Case-Hardened Steel Gear”, *Journal of Mechanical Science and Technology*, Vol. 24, pp 103-106, 2010
9. V. Moorthy, B. Shaw, “Contact Fatigue Performance of Helical Gears with Surface Coatings”, *Wear*, Vol. 276-277, pp 130-140, 2012
10. V. Moorthy, B. Shaw, “Effect of As-Ground Surface and the BALINIT C® and Nb-S Coatings on Contact Fatigue Damage in Gears”, *Tribology International,* Vol. 51, pp 61-70, 2012
11. M. Yilmaz, D. Kratzer, T. Lohner, K. Michaelis, K. Stahl, “A study on highly-loaded contacts under dry lubrication for gear applications”, *Tribology International*, Vol 128, pp 410-420, 2018.
12. A. Rempp, A. Killinger, R. Gadow, “New Approach to Ceramic/Metal-Polymer Multilayered Coatings for High Performance Dry Sliding Applications”, *Journal of Thermal Spray Technology*, Vol 21, pp 659-667, 2012.
13. M. Petrik, R.W. Titel, H. Thomsen, P. Kaestner, J.P. Kropp, “Beschichtungen für Zahnräder”, *Vakuum in Forschung und Praxis*, Vol 20, pp 6-12, 2008.
14. S. Yang, D. Camino, N. Renevier, A. Jones, and D. Teer, “Deposition and Tribological Behavior of Sputtered Carbon Hard Coatings”, *Surface and Coatings Technology*, Vol. 124, 2000.
15. S. Yang, X. Li, N. Renevier, and D. Teer. “Tribological Properties and Wear Mechanism of Sputtered C/Cr coating”, *Surface and Coatings Technology*, Vol. 142, 2001.
16. J.F. Archard, “Contact and Rubbing of Flat Surfaces, *Journal of Applied Physics, Vol. 24,* pp 981-988, 1953.
17. S. Andersson, A. Flodin, “Simulation of Mild Wear in Spur Gears”, *Wear*, Vol. 207, pp 16-23, 1997
18. K. Wang, H. Cheng, “A Numerical Solution to the Dynamic Load, Film Thickness and Surface Temperature in Spur Gears, Part I – Analysis”, *ASME Journal of Mechanical Design*, Vol. 103, pp 177-187, 1980
19. S. Andersson, “Wear Simulation with a Focus on Mild Wear in Rolling and Sliding Contacts”, *Friction, Wear, and Wear Protection*, pp 3-20, 2008
20. C. Fernandes, P. Marques, R. Martins, J. Seabra, “Gearbox Power Loss. Part II: Friction Losses in Gears”, *Tribology International*, Vol. 88, pp. 309-316, 2015.
21. SKF Group, “The SKF model for calculating the frictional moment”, *SKF Rolling Bearings Catalogue*, 2018.
22. H. Ohlendorf, “Verlustleistung und Erwärmung von Stirnrädern”, Dissertation, TU München, 1958.
23. E. Buckingham, “Analytical mechanics of gears”, Dover Publications, 1963
24. A.C. Ferrari, J. Robertson, “Interpretation of Raman spectra of disordered and amorphous carbon”, *Physical Review B*, Vol 21, 1999.
25. T. Ye, J. Ma, Z. Jia, T. Li, W. Liu, W. Yu, ,”Microstructure, Mechanical Properties and Low-Temperature Tribological Behavior of Cr/Cr-W/W-DLC/DLC Multilayer Coatings on 5A06 Al alloy”, [*Journal of Materials Research and Technology*](https://www.sciencedirect.com/journal/journal-of-materials-research-and-technology), Vol. 18, pp. 810-819, 2022.