Oil-Off Characterization Method Using In-Situ Friction Measurement for Gears Operating Under Loss-of-Lubrication Conditions

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Introduction

The oil-off (also known as loss-of-lubrication or oil-out) performance evaluation of gears is of significant interest to the Department of Defense and various rotorcraft manufacturers, so that the aircraft can safely land in an accidental loss-of-lubricant situation. However, unlike typical gear failure modes such as pitting or bending fatigue where early detection is possible, gear failure in an oil-off situation is very rapid and likely catastrophic. Failures rapidly result in the loss of torque transmission and the inability to control the aircraft.

Interest in loss of lubrication gearbox performance testing is not new. In 1978, Hudgins and Schuetz described the need for improved survivability of helicopter drive systems based upon loss of lube events in the Vietnam conflict due to combat damage (Ref. 1). They noted that typical times to failure were five to nine minutes and that failure modes were inconsistent. Five to nine minutes is not enough time for an aircraft to escape hostile environments and land safely. A requirement of 30 minutes of operation after lubrication system failure was established and is still used today as outlined in the Title 14 Code of Federal Regulations: Airworthiness Standard (Ref. 2.)

Today's helicopters and tiltrotors are commonly used for missions where 30 minutes of loss of lube operation is insufficient, such as long range, over water flights to access drill rigs or aircraft carriers. This has led to review of this longstanding benchmark and the standard is currently being revised. Indications are that test survivability times for full gearboxes will be between 36 and 67 minutes and that more test repetitions will be required.

Related Research

Between 1978 and the early 2010s, there was very little related to oil-off testing reported in the open literature with the exception of some work done by NASA Glenn (Ref. 3) and Kaman Aerospace (Ref. 4), both of which focused on improvement of gearbox performance through auxiliary oil supplies. There were many lessons learned, the most obvious were that there are a large number of variables that can affect the result of the test and that the possibility exists for gearboxes to operate for a long time with minimal lubrication.

Many of the test reports available are for full-scale gearboxes (Refs. 1; 4–8). The complexity involved is staggering. The variables that can affect the outcome of each test are: gear arrangement (planetary vs. non-planetary); gear backlash; gear material; housing design (provides locations for oil to pool); type of oil leak (pressure loss vs. hole in sump); bearing and shaft clearances; bearing type; bearing material; surface roughness of sliding surfaces; ability to generate mist; heat transfer properties; and more. It is clear that component-level testing to optimize gear-related variables is a valuable step before undertaking a costly full-scale test.

Several computational tools have been built to model aspects of lubrication loss to a gear mesh (Refs. 9–15). The most sophisticated is likely the multi-physics-based approach taken by McIntyre, et al., incorporating aspects of contact mechanics; tribology; computational fluid dynamics modeling for the gearbox flow; conduction within the gears; housing and components; and free convection to the environment (Ref. 14). The model is based upon a component test rig at NASA Glenn and predicts temperature distribution across the teeth and time-to-failure (Refs. 13–14). Efforts are currently underway to correlate model output with temperature data collected at Penn State University. While useful insights can certainly be observed, the available models cannot currently account for variables such as oil mist variation, carbonaceous oxidation deposits that develop during testing, gear tooth profile loss and any other unknown influences.

Gear Failure Mechanisms

The lubricant inside of a gearbox serves two primary functions — to separate sliding metal surfaces and to distribute and remove the heat generated due to this sliding. Components begin to heat up immediately when the lubricant supply is removed. This leads to the thinning of the remaining oil film and the prompt occurrence of metal-to-metal contact, which is accompanied by a sharp rise in friction. Increased friction causes more heat generation, exacerbating the problem. Without oil, the system relies on conduction through the components and convection inside and outside the gearbox for heat dissipation. Most reported test results show a period of metastable thermal equilibrium after loss of lubrication where the gearbox operates at higher than design temperatures (Refs. 3, 5, 15–17). The gear teeth are often reported to be the initial component to fail (Refs. 1 and 7). Scuffing initiates immediately upon metal-to-metal contact (breakdown of lubricant film). The high heat eventually causes the steel to soften and leads to plastic deformation under load (Ref. 18). The extreme temperatures also result in thermal expansion that can cause seizure of the gears if backlash is insufficient. The gear backlash can typically be increased to account for the high temperatures expected during oil-off cases.

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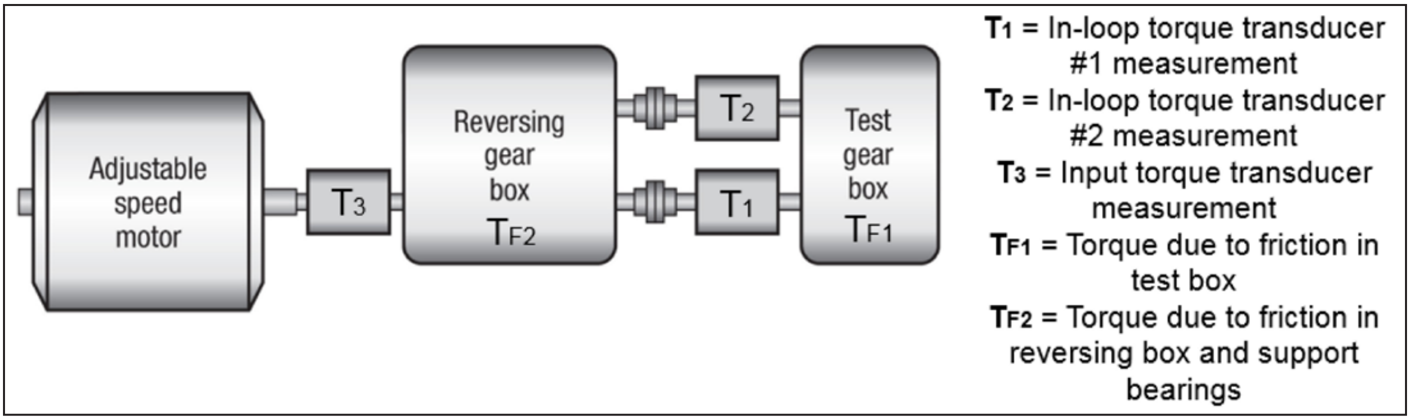

Figure 1 Test rig layout.

Scuffing

The previously discussed work identifies scuffing as a key part of the oil-out failure process for gears (Refs. 1, 3, 5, 7, 9–10, 15, 17–19). Scuffing is a physical failure of the elastohydrodynamic lubrication (EHL) mechanism (Ref. 20). It can be described as the repeated welding and tearing apart of surface asperities due to insufficient lubrication. Scuffing resistance has traditionally been increased by improving surface finish, using EP additives in lubricants or other means of reducing the friction between the contacting surfaces (coatings for instance) (Ref. 20).

Scuffing has been studied using coupon level tests by many researchers. Tests such as pin-on-disc; four-ball; block-on-ring; twin-disc; ball-on-ring; and ball-on-disc are used to generate scuffing under controlled, specific test conditions (Ref. 19); the influence of roughness, oil additives, etc. can be quantified. There has been significant work done at both the U.S. Army Research Lab and Wedeven Associates, Inc. using ball-on-disc testing to evaluate loss-of-lube performance. While scuffing is the primary failure mechanism, some performance differences exist and are the subject of continued research. These differences are likely attributable to the variable stresses and variable sliding inherent to the involute gear tooth geometry and the fact that the meshing action of the gear teeth allows even the smallest amount of residual lubricant mist (or even vaporized carbon (Ref. 21)) to provide continued lubrication to the mesh.

Scuffing failures demonstrate a specific progression that is consistent — regardless of test type. The onset of scuffing, called "micro-scuffing" by Yagi, et al. (Ref. 22), is followed by a period of stability. What is occurring during this period varies depending on test type, roll/slide ratio and amount of residual lubricant present. It is most likely an oxidative wear phenomena (Ref. 23) created by the oxidation of the residual lubricant and the metal at high temperature, resulting in a high carbon deposit that reduces the friction between the contacting bodies. This is where the metastable thermal equilibrium occurs. The length of this period depends on a number of factors, but is essentially determined by how well the system is able to absorb and remove the generated heat from the contact. Eventually, the temperature rise is too much to overcome and another transition occurs where the friction and temperature rise sharply, leading to catastrophic failure (Refs. 19, 22–24).

Gear Testing

A gap exists between coupon testing and full-scale gearbox testing, allowing more realistic, cost-effective oil-out screening tests to be conducted using a component level test. These tests are currently being used to characterize the performance benefits of the most advanced gear steels, surface treatments, lubricants and even non-involute tooth profiles for oil-out operation (Refs. 17 and 25). The remainder of this paper outlines the test rigs and procedure developed for oil-out performance characterization.

Test Rig Hardware

A 3.5" center distance power recirculating four square test rig capable of speeds up to 10,000 RPM was used for the oil-off testing described in this paper. This test rig uses a dedicated reversing gearbox that hydraulically applies torque to the four-square loop, which enables torque changes during test operation. A schematic layout of the system is shown (Fig. 1). Several modifications to the test rig were necessary to allow for oil-off operation and are described in detail below.

Oiling System Modification

During typical oil-on testing, the test gearbox uses jet lubrication to supply oil to the test gears. Additionally, the test gear shafts are supported on each end by bearings which require a pressurized oil feed. The test box bearings use the same oil that is supplied to the gear mesh for lubrication, and the bearings drain to the interior of the test box. It was necessary to modify the oiling system of the rig to allow shutdown of oil flow to the gear mesh while maintaining lubrication to the test box bearings. The test box bearing oil feed lines was also equipped with needle valves so bearing oil flow could be controlled in order to ensure oil mist in the test box during oil-off was identical and repeatable from test to test. Bearing temperature limits were continuously monitored during testing to verify proper operation.

Oil Control Shroud

The test rig was also modified to shroud the perimeter of the test gears in order to control residual oil after lubrication flow is stopped. The shroud was based on the design shown (Ref. 9) and has 0.06"radial clearance from the shroud to the tooth top land, as well as 0.06" clearance to each end face. Slots positioned radially outward from the gear rotation allow residual lubrication to exit during the oil-off event. The assembled shroud (without front cover installed) is shown (Fig. 2).

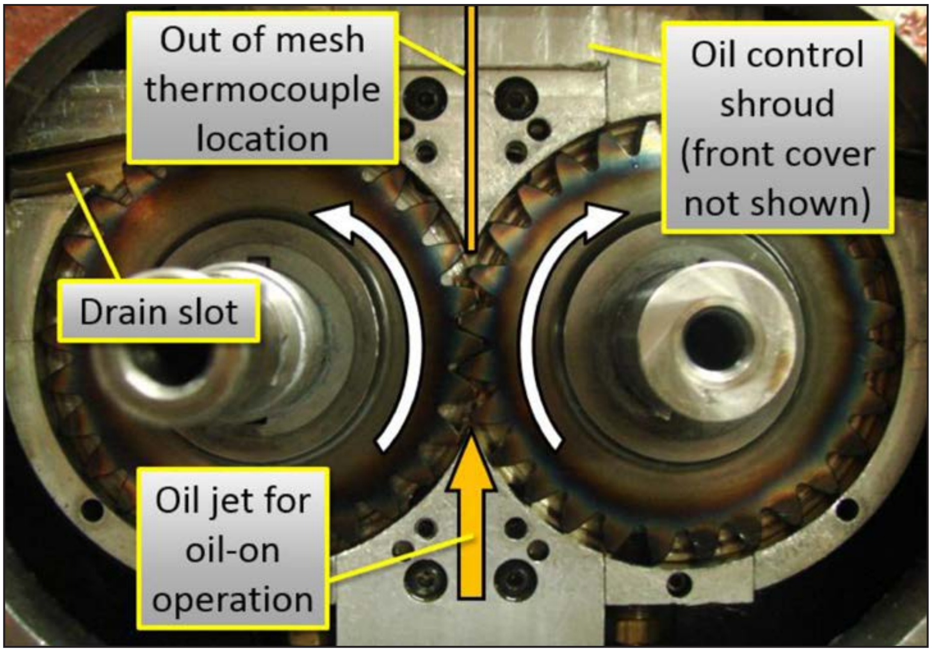

Figure 2 Oil control shroud.

Table 1 Oil-off test result cases

	Case	Description
Least Severe	I	No scuffing (runout)
⇕	II	Scuffing without progression to catastrophic failure (runout)
	III	Scuffing with progression to catastrophic failure
Most Severe	IV	Immediate catastrophic failure

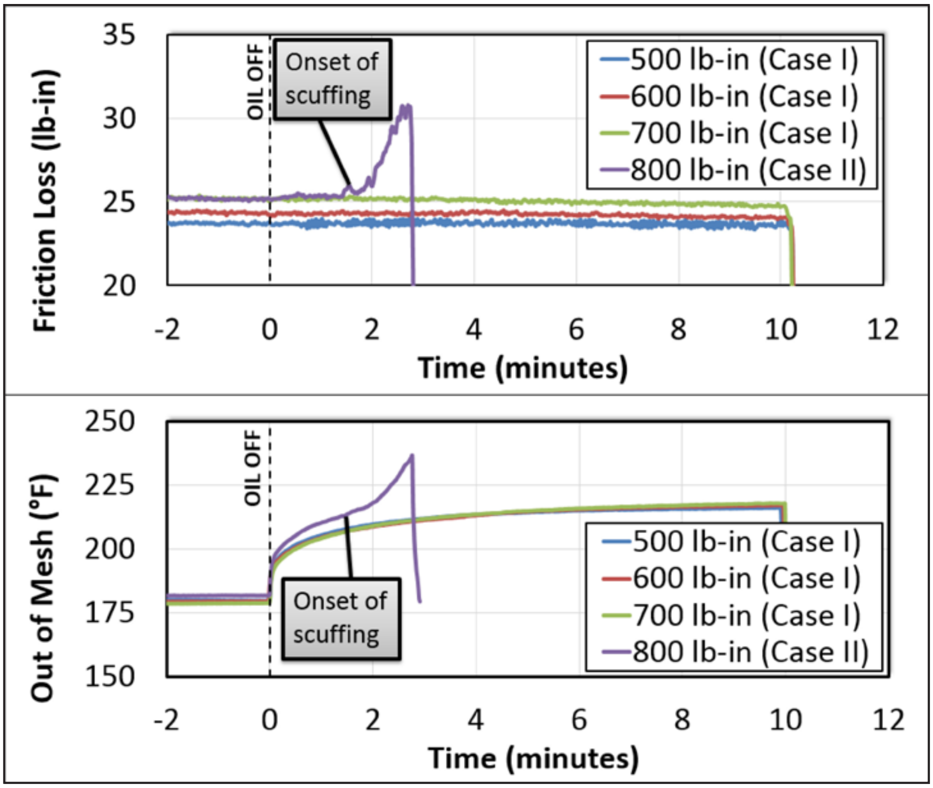

Figure 3 Detecting onset of scuffing.

Out-of-Mesh Temperature Measurement

A 0.125" diameter out-of-mesh thermocouple was incorporated into the oil-off shroud, as shown in the location shown (Fig. 2); it was positioned axially as close as possible to the rotating gear teeth. This thermocouple measures out-of-mesh oil temperature during oil-on break-in, but is also used to measure temperature of the entrained air near the gear tooth surfaces during oil-off. Although the absolute temperature reading of the entrained air does not directly predict tooth temperature, it was found that the temperature trend is a valuable metric for monitoring progression of gear failure during oil-off.

Friction Measurement

As shown (Fig. 1), the test rig was also instrumented with three torque transducers. Two transducers monitor torque inside the four-square loop, and an additional transducer monitors input torque from the drive motor. These measurements are then used to compute frictional losses in the test box and reversing box. Details of the loss calculations and friction measurement techniques were presented previously by the authors (Ref. 26). Test box friction loss measurements were found to be the most effective metric for monitoring oil-off progression to failure, and were also found to correlate well with out-of-mesh temperature measurement trends during catastrophic failure. To the knowledge of the authors, in-situ measurement of gear mesh friction during oil-off gear testing has not previously been reported in open literature.

Test Procedure

The oil off test procedure is defined as follows:

1. ***Break-in step #1*:** Run with oil on at break-in torque for 30 minutes.
2. ***Break-in step #2*:** Ramp to test torque, continue running with oil for 60 minutes.
3. ***Oil-off*:** Continue running at test torque and turn off oil; monitor for catastrophic failure.
4. ***Runout*:** If runout time limit is reached (typically 30 minutes), turn on oil and increase torque.
5. ***Stabilize*:** Allow to run at increased torque level with oil for 10 minutes.
6. ***Repeat*:** Repeat oil-off and continue to run up to runout time limit. If necessary, repeat steps 4 through 6 until catastrophic failure occurs.

Typical Test Results

Through extensive testing, it was found that oil-off results typically fall into one of the four cases summarized in Table 1, based on the severity of test conditions. Details and examples of each case are presented below.

Case I: no scuffing. In the least severe case, gear teeth can last the oil-off test period without scuffing. This can be caused by residual oil mist in the test box or test torques that is too low. This condition does not occur frequently, but is presented here to illustrate the friction loss and out-of-mesh temperature trends when the onset of scuffing is detected.

Figure 3 shows four tests which were completed at increasing torque levels with the same gear pair. These tests were part of an initial development effort to evaluate the effectiveness of friction loss and out-of-mesh temperature to detect scuffing. Tests were stopped for visual inspection after each torque step, and the 500, 600 and 700 lb-in steps did not show any scuffing. This indicates that the residual lubricant and/or tribological film were sufficient to prevent scuffing initiation under these conditions.

Case II: scuffing without progression to catastrophic failure. In Figure 3, continuing testing to 800 lb-in showed a significant change in both friction and temperature trends, and visual inspection confirmed that scuffing occurred at this torque; the scuffed gear tooth surfaces are shown (Fig. 4). In a typical test sequence this test would have been allowed to continue to catastrophic failure or runout, but this test was stopped after friction and temperature trends indicated scuffing had occurred. This test result demonstrates the effectiveness of using friction and out-of-mesh temperature to detect the onset of scuffing.

As conditions increase further in severity, scuffing typically occurs rapidly after oil flow is stopped. This is shown in the friction loss trend in Figure 5, which peaks immediately after oil-off. Since heat generation increases with friction, this suggests that the maximum heat generation at the mesh interface occurs during the scuffing event immediately after oil-off. After the initial scuffing event, the friction loss and out-of-mesh temperatures stabilize as the test continues to run. In this case, the conditions are not severe

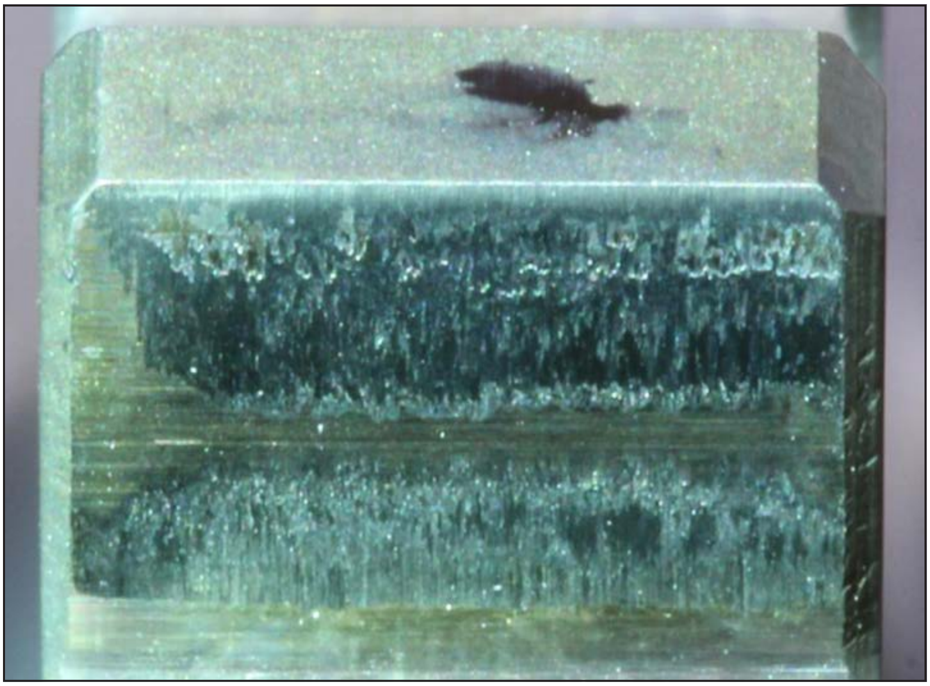

Figure 4 **Scuffed gear tooth.**

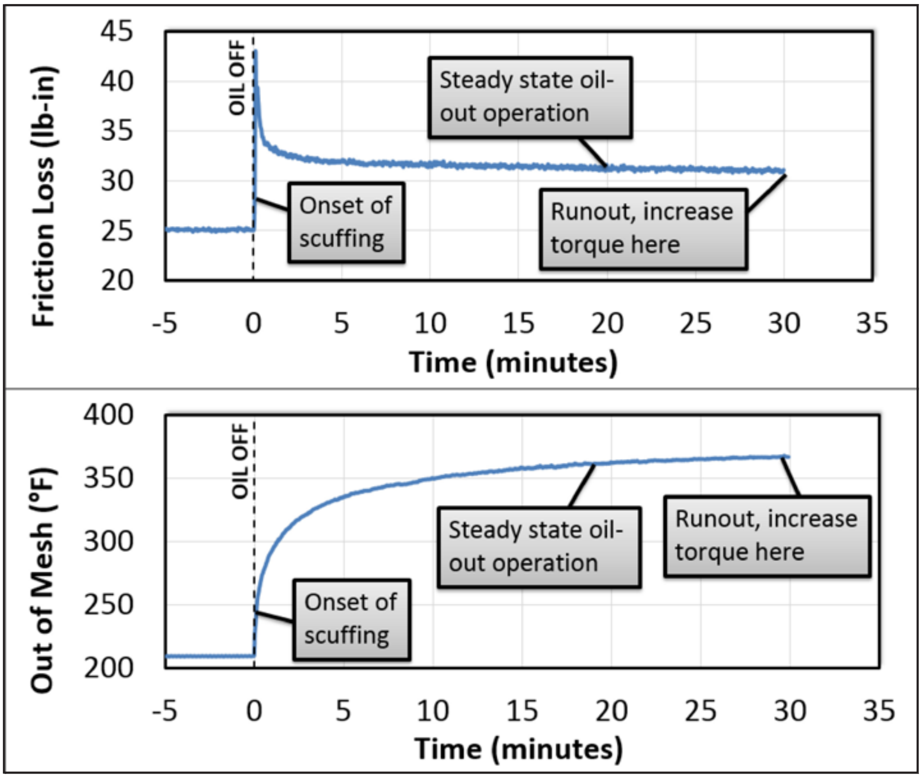

Figure 5 **Scuffing and runout.**

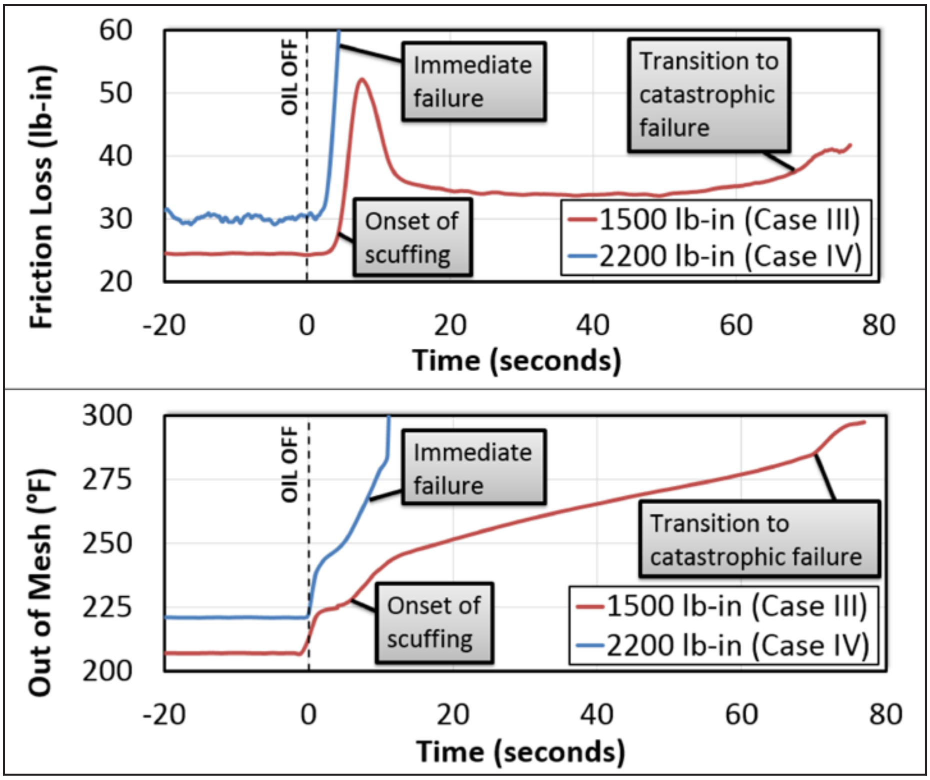

Figure 6 **Case III vs. Case IV severity comparison.**

Figure 7 **Catastrophic failure, run to loss of power transmission.**

enough to cause the test to progress to catastrophic failure.

A metastable thermal equilibrium is then reached where out-of-mesh temperature is asymptotic. The heat input to the mesh is being absorbed and removed from the system before runaway failure occurs. Although the tooth surfaces are scuffed, the gears still transmit torque and as such are not considered failures in the context of this test. Testing has shown that it is possible for gears to run in the scuffed oil-off state for several hours without progressing to catastrophic failure. If the run-out time limit is reached, the oil is turned back on and the torque is increased to prepare for the next oil-off step.

Case III: scuffing with progression to catastrophic failure. If test conditions are increased in severity further than in Case II, the test will eventually progress to catastrophic failure; a plot of the torque and temperature data for two different cases of catastrophic failure are shown (Fig. 6). With the failure progression of Case III, the initial friction peak from scuffing is followed by a period of stable operation. This period of stable operation that precedes the transition to catastrophic failure allows for the separation of performance of different test groups.

In Case III, friction then begins to increase again after the period of stable operation, which indicates the test is progressing to catastrophic failure. The out-of-mesh temperature rate of change also increases when catastrophic failure begins. Complete loss of power transmission typically occurs shortly after friction increases, and an example of a gear pair run to this extent is shown (Fig. 7). Testing to complete loss of power transmission can be damaging to the test rig, so tests are typically stopped once friction and out-of- mesh temperature trends suggest catastrophic failure is imminent. An example of a test stopped before loss of power transmission is shown (Fig. 8); note that there is significant plastic deformation present on the tooth.

Case IV: immediate catastrophic failure. If test conditions are increased in severity further than in Case III, catastrophic failure can occur immediately after oil-off, which is shown as Case IV in Figure 6. In this progression, the period of stable oil-off operation after initial scuffing from Case III is not present. Immediate catastrophic failure is not desirable since differentiating performance between test groups is difficult in this case.

Ideal Test Conditions

The ideal test conditions should produce a majority of Case III catastrophic failures at the established test torque. Per the test procedure, if a runout occurs (Case II) the torque is then increased, and the oil-off event is repeated. This will eventually lead to a catastrophic failure at a torque higher than the established test torque. It is desirable to select a test torque that is severe enough to minimize the number of runouts, since this requires multiple oil-off events to initiate a catastrophic failure. At the same time, the test torque must not be too high as to cause immediate catastrophic failures (Case IV) which do not produce useful performance data.

Load Step Searching Tests

The test torque which will produce a majority of Case III catastrophic failures will vary between test programs and is influenced by factors such as gear design, oil selection and surface finish. In order to establish an approximate value for the test torque using the minimum number of test gears, a load step approach is used. This is similar to the test procedure outlined previously, with the exception that a shorter time interval of 10 minutes oil-on and 10 minutes oil-off is used. A torque near the lower limit of the rig's capability is used as a starting torque, which is increased by 100 lb-in after 10 minutes of oil-off without catastrophic failure; a portion of a typical load step searching test is shown (Fig. 9).

Since a load step searching test typically subjects the test gears to multiple oil-off events before catastrophic failure occurs, the torque determined by this method may not be the final test torque that should be used. It will, however, establish a starting point for further testing to validate that new gears taken immediately to the test torque and subjected to oil-off will fail in the desired progression.

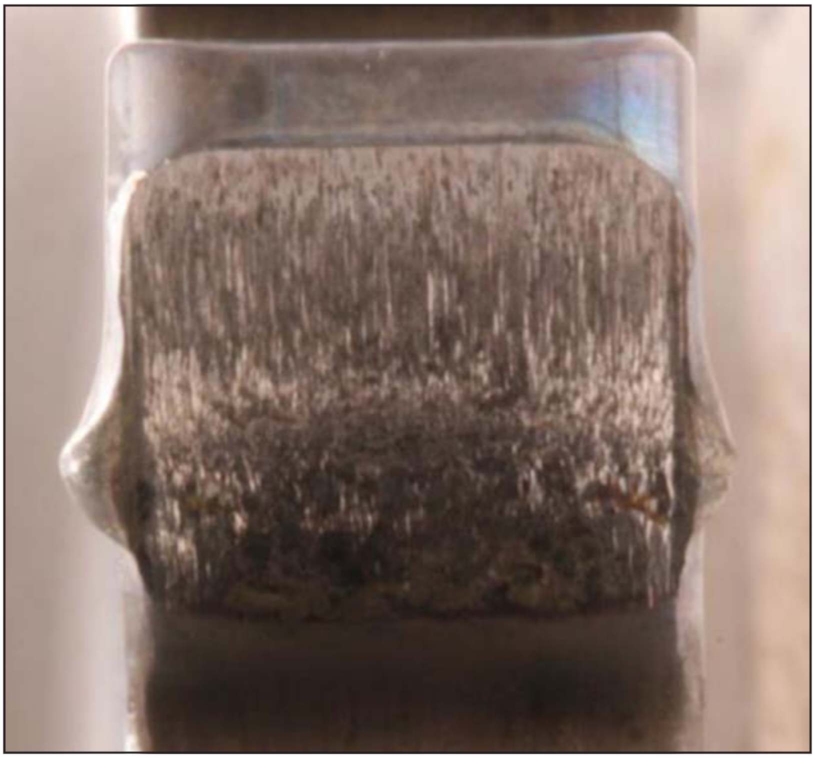

Figure 8 Catastrophic failure, stopped prior to loss of power transmission.

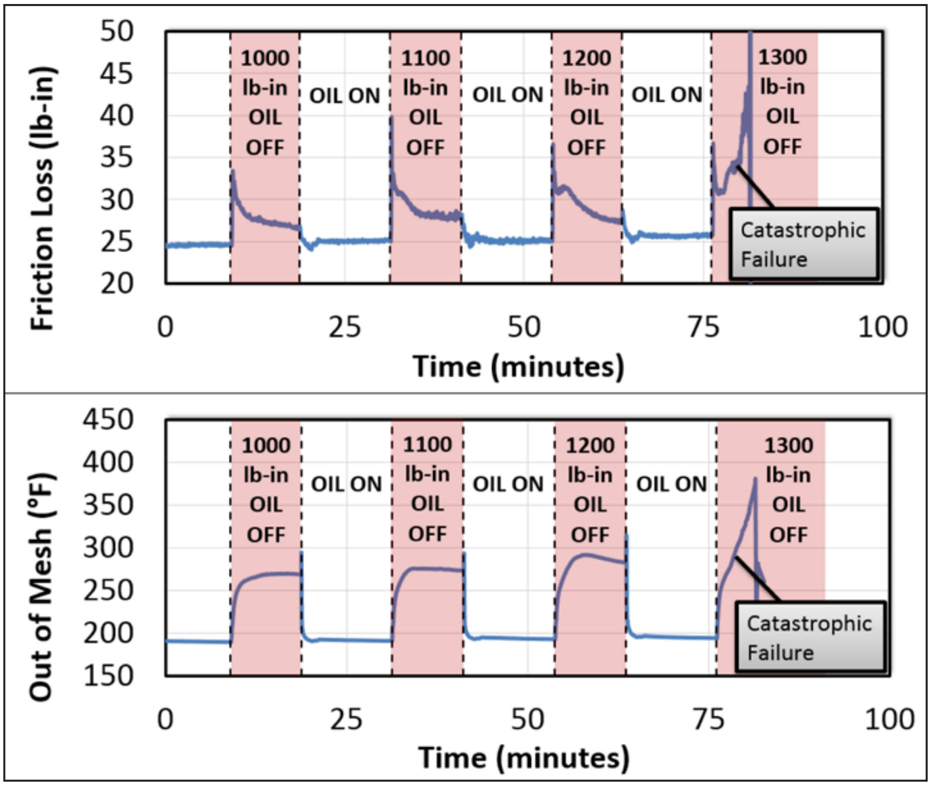

Figure 9 Load step searching test.

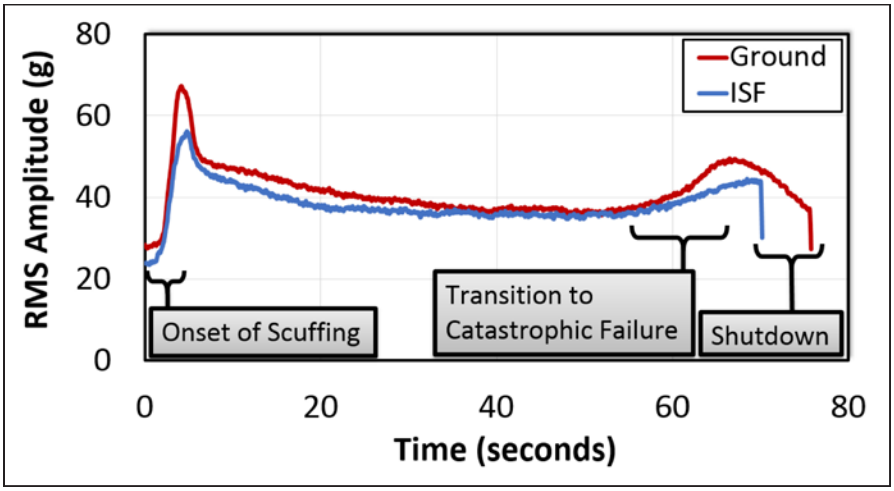

Figure 10 RMS accelerometer data.

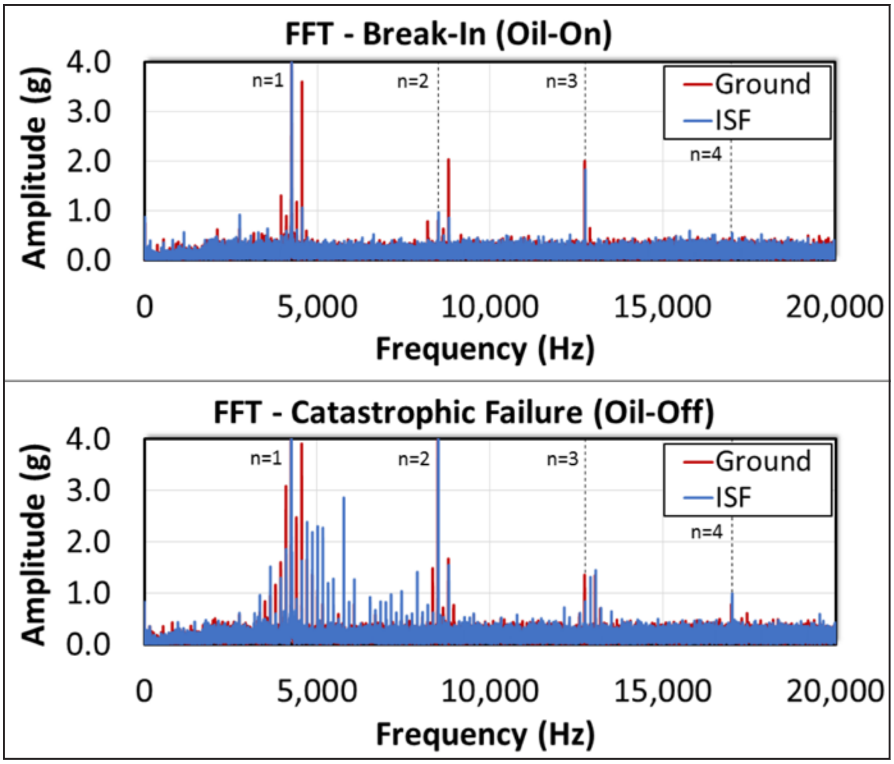

Figure 11 FFTs of accelerometer data (tooth mesh fundamental frequency and harmonics denoted by n = 1...4).

Figure 12 Thermocouple instrumented gear.

Additional Data Collected

Noise and vibration measurements. Noise and vibration data were also collected during testing. A sample rate of 51.2 kHz was used for the data presented (Figs. 10–11), which in this case captures up to the fourth harmonic of the tooth meshing fundamental frequency.

Figure 10 shows the RMS amplitude of the accelerometer data for two tests conducted with as-ground and isotropic superfinished (ISF) test surfaces. The accelerometer data is shown to have trends similar to the friction measurement data. An initial peak during the scuffing event is followed by a period of stable operation, leading to a second increase when the test transitions to catastrophic failure.

Data from the same tests are shown in the frequency domain in Figure 11 — both before and after oil-off.

As expected, peaks in the FFT (fast Fourier transform) data at the tooth meshing fundamental frequency and harmonics are visible during oil-on break-in. The progression to catastrophic failure shows the appearance of additional sidebands as the tooth profile degrades from severe scuffing.

Side-of-tooth thermocouple data. In an effort to correlate out-of-mesh entrained air temperatures to gear tooth surface working temperatures, gears were instrumented with a thermocouple on the side of one tooth using thermally conductive epoxy (Fig. 12). Figure 13 shows the friction loss result of an oil-off test, together with out-of-mesh and side-of-tooth thermocouple data. The side-of-tooth temperature shows an initial maximum followed by a gradual reduction to near steady state (thermal equilibrium). This validates the theory that the maximum heat generation at the mesh interface occurs during the initial scuffing event, indicated by the friction peak immediately after oil-off. This particular test did not catastrophically fail and ran for over 90 minutes without oil. A steady state temperature of approximately 440°F was measured on the side of the tooth when the test was terminated as a runout.

Figure 14 shows an example of side-of-tooth thermocouple data for a test which catastrophically failed. At the onset of catastrophic failure the rate of change of the side-of-tooth temperature increases, similar to the trends observed for out-of-mesh temperature. Side-of-tooth temperatures of over 700°F were measured before the test was terminated. This data is useful for validating computational models of the loss-of-lubrication event, examples of which can be found in (Figs. 13 and 14).

Application Example: Comparison of Experimental Group vs. Baseline.

An example of this test method's ability to differentiate performance between test groups is illustrated in Table 2 and Figure 15. The results of oil-off testing with a baseline test group and a second experimental test group with an advanced material, coating, and oil are shown. Multiple test repetitions were used, since scatter in oil-off performance test data is well documented (Refs. 1, 5, 10, 17 and 27). As shown in Table 2, although scatter is present in the experimental group data, a significant increase in performance is shown over the baseline group in all cases.

Figure 15 highlights the differences in friction and temperature trends between the two test groups. The experimental group shows a delayed scuffing onset and slower progression to catastrophic failure, along with reduced out-of-mesh temperatures in both oil-on and oil-off conditions.

Summary

The testing method and data presented show that the friction loss in the gear mesh and out-of-mesh temperature are effective means of evaluating the operation and performance of aerospace gears in loss-of- lubrication conditions. Specifically, gear mesh friction was shown to be a sensitive indicator of scuffing and the progression to catastrophic failure during oil-off events. The effort described in this paper provides an experimental methodology for evaluating oil-off performance of gears — a phenomenon that has been difficult to characterize in the past owing to its catastrophic and sudden nature. Typical failures were found to fall into one of four categories based on severity of the test conditions, and guidelines were proposed to establish the desired failure progression. Examples of data from baseline and experimental test groups demonstrate the ability of the test method to highlight performance improvements of advanced materials, coatings and lubricants in oil-off conditions.

Acknowledgements. *The authors would like to thank NASA Glenn Research Center, Army Research Laboratory, Gary Pozarnsky of Applied Colloids, Boeing and the Vertical Lift Consortium for their collaboration and/*

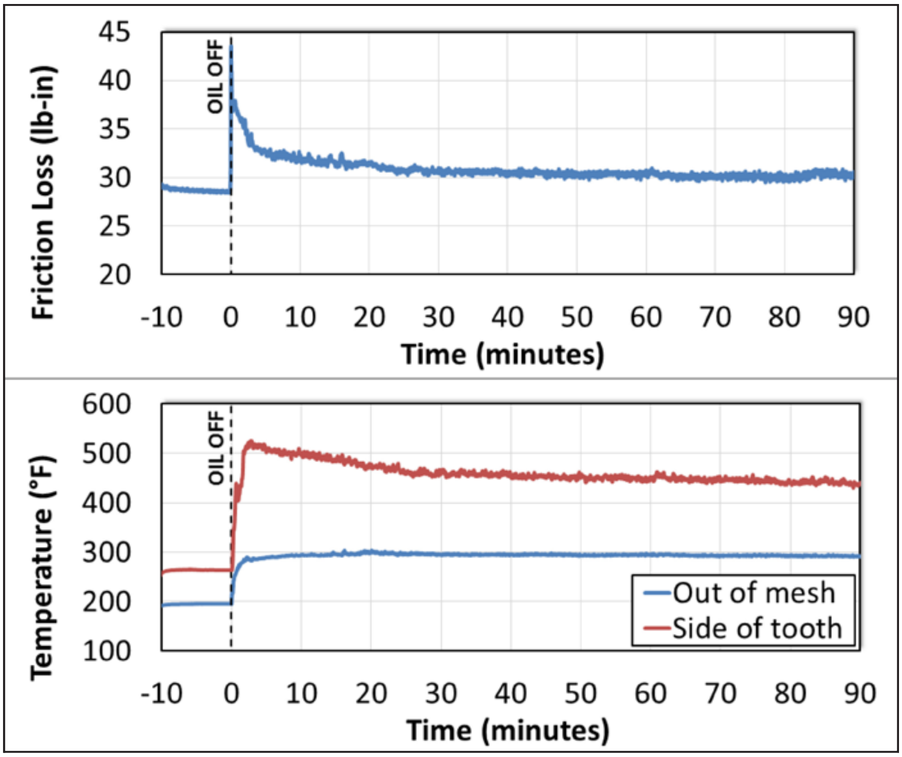

Figure 13 Oil-off results with side-of-tooth thermocouple data (runout).

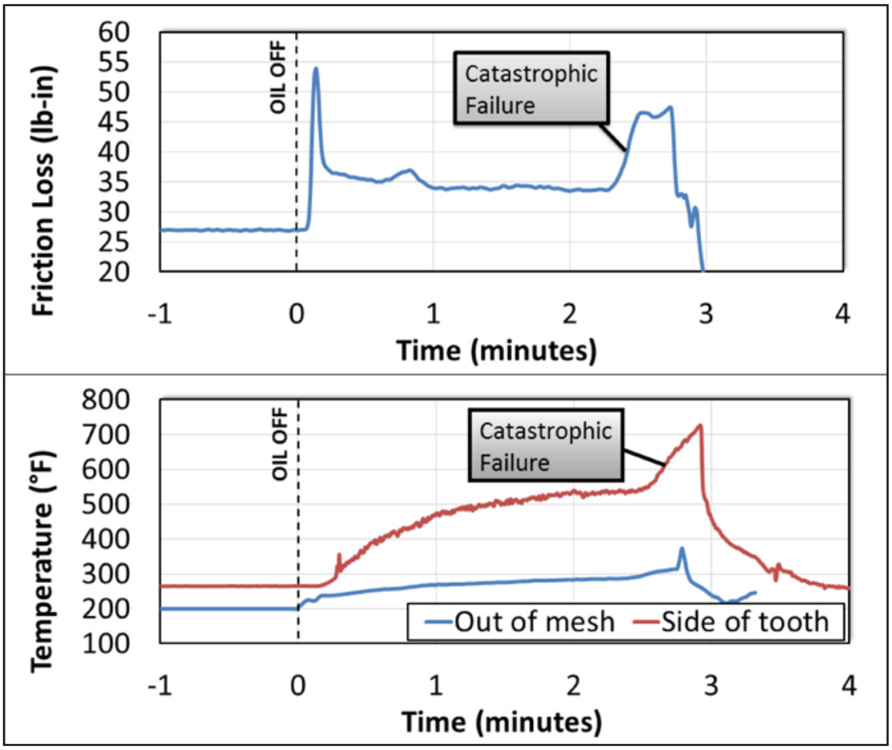

Figure 14 Oil-off results with side-of-tooth thermocouple data (catastrophic failure).

Table 2 Time to failure for baseline vs. experimental group

	Time to Catastrophic Failure (minutes)	
	Baseline Group	**Experimental Group**
Test #1	0.7	9.1
Test #2	0.7	12.5
Test #3	0.8	5.3

or financial contribution in support of this work.

For more information. Questions or comments regarding this paper? Contact Aaron Isaacson at *aci101@arl.psu.edu.*

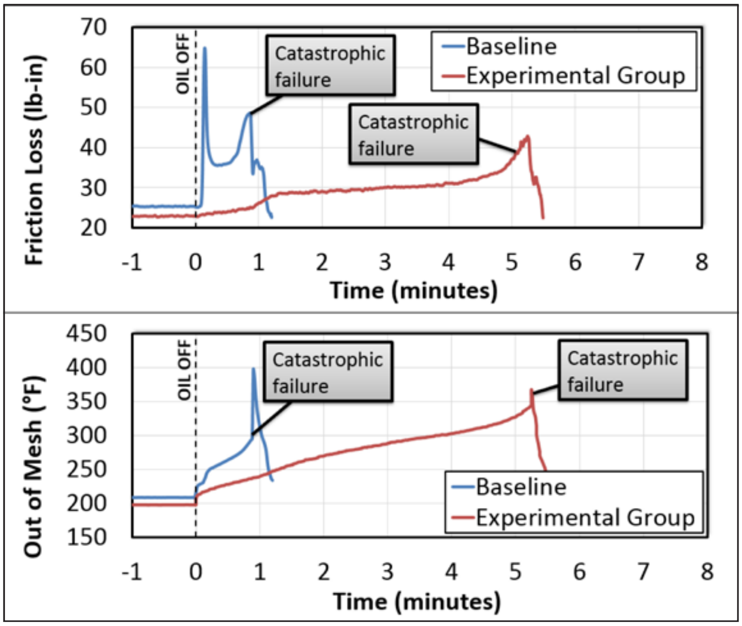

Figure 15 **Experimental vs. baseline performance comparison.**

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