

Accelerated Testing of UH-60 Viscous Bearings for Degraded Grease Fault

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Abstract

An accelerated aging investigation of critical aviation bearings lubricated with MIL-PRF-81322 grease was conducted to derive an understanding of the mechanisms of grease degradation and loss of lubrication over time. The current study focuses on UH-60 Black Hawk viscous damper bearings supporting the tail rotor driveshaft, which were subjected to more than 5800 hours of testing in a heated environment to accelerate the deterioration of the grease. The mechanism of grease degradation is a reduction in the oil/thickener ratio rather than the expected chemical degradation of grease constituents. Over the course of testing, vibration and temperature monitoring of bearings was conducted and trends for failing bearings are presented.

Keywords: bearing diagnostics, grease lubrication, accelerated test, helicopter drivetrain, seeded fault, mechanical diagnostics

Introduction

Rotary wing aircraft generally rely on complex mechanical power transmission systems to transfer mechanical energy from engine output shaft speeds to relatively slower rotors while changing axis orientation. Additionally, conventional tail rotor-based anti-torque systems, rotating aerodynamic surfaces requiring cyclic and collective pitch controls, and other mechanical systems integral to rotorcraft result in challenges related to system reliability and affordable maintenance. For many decades, failure phenomena and diagnostic techniques have been studied in order to improve reliability of these critical mechanical elements [1, 2].

The US Army in recent years has invested heavily in outfitting much its fleet of helicopters with health and usage monitoring systems (HUMS), with much of the associated instrumentation monitoring critical mechanical systems such as bearings and gearboxes. Various diagnostic algorithms drawn from the literature have had varying degrees of success in detecting component faults when implemented in actual helicopter applications, and with some of this operational experience, specific components have been identified for changes in maintenance practices due to the improved awareness of component state [3].

In consideration of changing maintenance practices, the mechanisms and behaviour of grease degradation in specific components must be better understood to extend maintenance intervals for grease lubricated bearings. Studies of grease lubrication reported in the literature have identified both chemical and physical processes as being responsible for degradation of grease

[4,5], with relative contributions depending heavily on operating and environmental conditions, as well as the composition of the grease. For *in situ* tests of grease aging in bearings, temperature and severity of operation (speed and load conditions) can be altered to accelerate the degradation processes for grease. However, care must be taken to ensure that the conditions chosen do not alter the mechanisms by which the grease degrades [6].

In the current work, a study was undertaken to subject UH-60 Black Hawk helicopter viscous damper bearings to accelerated life tests in order to determine failure phenomena associated with the degradation of the grease lubricant over time. The flight critical bearings support the tail rotor drivetrain, a singular path of power transfer from the main module transmission to the aircraft's tail rotor. To ensure realistic *in situ* grease aging processes, previously reported studies at the authors' laboratories [7] have developed methodologies appropriate for realistic accelerated aging for lightly loaded grease-lubricated bearings that operate at moderate speeds and temperatures.

In this study, test bearings were operated up to 5800 hours in a heated environment to accelerate the deterioration of the grease. The typical speed, load and temperature conditions for this bearing operating in the helicopter are moderate. As a result, degradation processes are reasonably simple and can be adequately simulated at operating conditions other than those in the application, and this permits some flexibility in setting the simulation conditions. Thus, an increase in operating temperature is perhaps the easiest means to effect acceleration of the grease aging *in situ*. Mechanical stresses play an important role in *in situ* aging [8], but it has been reported that the degradation processes are less sensitive to the specific value of mechanical stresses provided representative stresses are present. As a result, the bearings are run at speeds lower than those in the aircraft, and at lower loads.

This bearing aging test provides initial insight into the aging and lubrication failure of this specific helicopter drivetrain bearing, and validations of some of the methods employed. Following these initial tests with this particular test bearing, another series of tests will be conducted on identical bearings but with rotational speed and applied load conditions more closely representing those on the aircraft.

Methods

Test Rig

The test rig comprises an electric motor driving a rotating shaft assembly on which test bearings are mounted and operated within a heated test section. Eight test bearings are housed within the test section, with inner races fixed to the rotation of the test shaft and with stationary spring-loaded posts engaging slots machined in the outer race to constrain rotation. Four of these spring plungers engage from below and the other four engage from above. This alternating attachment provides a small radial load on each test bearing, while minimizing the total net radial load on the shaft and rig bearings and overall deflection. Two "rig bearings" on either end of the test section are mounted within the chamber structure and support and constrain the test shaft relative to the static rig structure. The arrangement constrains the rotation axis of the test bearing inner races more rigidly while allowing their outer races some freedom of movement in both axial and radial directions. The two rig bearings are identical to the test bearings, but operate at considerably lower temperatures. Other test bearings are accommodated by this rig with only minor configuration changes, but only the UH-60 viscous damper bearing is considered in this paper.

The rotating shaft assembly is attached to a motor through a flexible coupling. The eight bearings are encased in an insulated heated test chamber having sheet metal walls, with the heater located directly below the test bearings. A plastic guard is placed around the test chamber and motor to obstruct access to hot and rotating parts. Heater set points to 190 °C (375 °F) or higher can be achieved with the current configuration, although bearing temperatures may remain somewhat lower due to heat sinking to the outside environment through the solid bearing shaft. Proportional-integral-derivative (PID) control is used to regulate chamber temperature based on a thermocouple measurement, and while the chamber temperature can vary cyclically up to ± 10 °C (18 °F) the bearing race temperature does not reflect temperature oscillations greater than 0.5°C (1 °F). In later experiments, air flow has been introduced within the chamber encourage a more isothermal chamber environment, but that technique was not incorporated here and significant temperature gradients are observed between test bearings.

The brushless servo motor is placed on a translating mount on linear slides, allowing the motor to slide away from the shaft of one test package for removal of the test shaft and bearings, or for installation of an alternate test package. The rig is rated for continuous duty to maximize achievable run time. To date, over 9000 hours of experience have been accumulated on this test rig, and over 4000 hours of experience with a previous iteration of the rig and employing similar methodologies. This equipment provides a very economical means of accumulating large amounts of run time on many test bearings simultaneously.



Figure 1 – Grease Degradation Rig Setup with Insulation and Guard Shield Installed

Instrumentation

In order to monitor general health of the test rig and bearing specimens, a suite of instrumentation is installed in the test rig to be monitored and recorded at 1 Hz. Type J thermocouples are placed within the test chamber, on the motor housing, and are spring loaded in contact with the outer race of each rig bearing. The outer race of each test bearing is also instrumented with a type J “stick-on” thermocouple embedded within an adhesively mounted thin pad, as shown in Fig. 2. The test bearing thermocouples provide a practical, although indirect, indication of grease temperature.

In addition to the temperature instrumentation, an Endevco 7253C-10 triaxial accelerometer is mounted to the outboard end of the test chamber in the region of the rig bearing and the root mean square (rms) values of the accelerometer signals are logged to file. An analog voltage corresponding to the relative value of motor current is sampled and logged as well, although this signal has not been rigorously calibrated to motor torque and it serves more as a semi-quantitative indication of the torque to turn the shaft. The programmable automation controller compares all of the logged signals to user-defined limits, such that a limit exceedance on any of these signals effects a shutdown of the test rig.

In addition to the low speed data log, a high speed diagnostic system is installed to record tachometer and high frequency accelerometer data from the triaxial accelerometer. The tachometer signal is supplied by a fibre optic probe aimed at a metallic disk with a dark painted phase tick. The raw signal from each of the three accelerometer channels is recorded for monitoring of bearing fault frequencies and in computing other vibration-based condition indicators (CIs). The accelerometer signals have relatively low response to 20 kHz (at -4 dB), and the placement on the static support structure at one end of the rig diminish the ability for these signals to discern the vibration characteristics of individual bearings or of very high frequency vibration content. However, this data is recorded and available in the event of a failure to determine if useful diagnostic data is available.

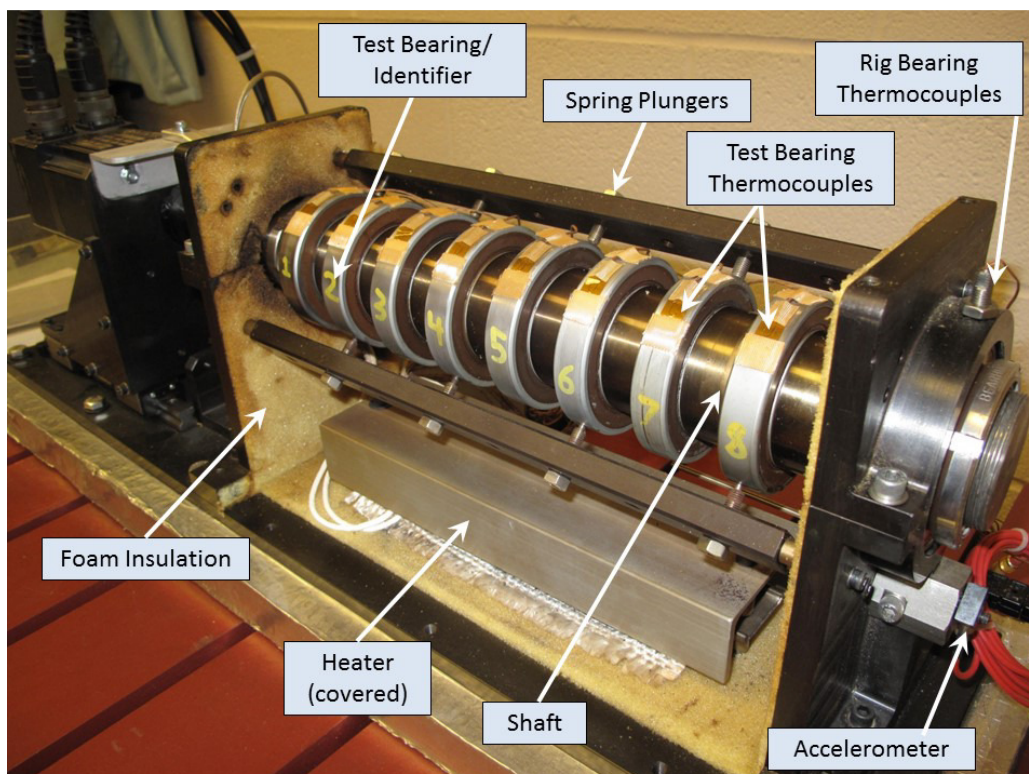


Figure 2 – Grease Degradation Rig with Test Package Chamber Exposed

Test Articles

The UH-60 tail rotor driveshaft “viscous bearing” is a sealed deep groove ball bearing, with a riveted two piece stamped cage. The bearing is housed within a viscous damper, loaded primarily in the radial direction on the order of 100 N (22 lb) and typically operates around 4114 rpm in the aircraft. Table 1 gives an overview of the technical specification of the test bearings. These operational conditions can be characterized as light load with a moderate $DN \approx 200,000$ (mm·rpm) using a bore diameter of 50 mm (1.968 in) and a nominal tail shaft

speed of 4115 rpm. DN is the product of the ball pitch diameter in millimetres and the speed in rpm, and gives a measure of the speed severity of a bearing, taking into account its size.

Table 1 – Test Article Specification, as described in its NSN technical listing

NATO Stock Number (NSN)	3110-01-329-8573
Bore Diameter	50 mm
Bore Shape	Straight
Load Direction	Radial
Material	All steel
Overall Outside Diameter	80 mm
Overall Width	16 mm
Style Designator	40A 4 Point Ball Contact, Solid Inner and Outer Rings, Non-Separable
Surface Finish	Ground

The test bearings are lubricated with grease conforming to the MIL-PRF-81322G standard. This standard specifies a high performance grease able to operate in a wide temperature range from -54 to 177 °C. The composition of the grease is not specified in the performance standard, but requirements for temperature range, dropping point, corrosivity, toxicity, and other physical properties restrict the compositions able to meet the standard. The specific grease chosen for this study is a commercially available formulation consisting of a polyalphaolefin synthetic base oil with an organo-clay (non-soap) thickener, which gives the grease a high dropping point over 300 °C. The grease is dark red in color and has a smooth buttery appearance when new. It is specified for use in the propulsion bearings under investigation in the current work.

In this study, a mixture of new and used test bearings was subjected to aging experiments. The new bearings had bright orange seals, but in some cases the packages were not sealed and so it could not be immediately confirmed that they had never been run. The used test specimens had darkened seals and in some cases liquid oil was present where it had separated from the grease. The used specimens were not received with a usage history, and seals were not removed to quantify grease volume and condition as-received. Consequently, the conclusions that can be drawn regarding the used bearings are somewhat limited, but observations from this initial series of experiments will be used for future tests with more controlled specimen condition and exposure. Table 2 provides an overview of the test specimens used in this study, where “TB” denotes a test bearing and “RB” denotes a rig bearing. Two bearings were removed and replaced during the course of testing (TB#6/a and RB#2/a) and a further two bearings, (TB#4 and TB#5) were removed but were replaced with spacers rather than additional test bearings. Finally, a fifth bearing (TB#7) was deemed failed as of the completion of testing. The details of the removal of bearings will be discussed in the results section.

Table 2 – UH-60 Oil Cooler Viscous/Hanger Bearings used in Rig Test

Bearing Identifier	Initial Condition	Exposure Time [h]	Avg. Bearing Temp. [°C] (°F)	
TB#1 B626-47134	Used – usage history unknown	5806+	114 (237)	
TB#2 B626-40571	New - sealed	5806	121 (249)	
TB#3 B626-56197	Used – usage history unknown	5806+	128 (262)	
TB#4 B626-80711	Appears new	5006 Failed	132 (270)	
TB#5 B626-56171	Used – usage history unknown	4167+ Failed	137 (278)	
TB#6/a B626-79706	Appears new	3278 Failed	136 (277)	137 (278)
TB#6/b B626-56175		2529	138 (280)	
TB#7 B626-55880	Used – usage history unknown	5806+ Failed	133 (272)	
TB#8 B626-40572	Appears new	5806	128 (262)	
RB#1	Arrived new, 2466 hrs as rig bearing in other test	5806+2466	79 (174)	
RB#2/a B626-25677	Arrived new, 2466 hrs as rig bearing in other test	3080+2466 “Failed”	90 (193)	91 (195)
RB#2/b B626-40629	New - sealed	2727	92 (197)	

For these experiments, the grease volume in the bearings was left as received to more realistically replicate the progression of grease degradation in this particular bearing. It should be noted, however, that some test procedures at the authors' laboratories call for the bearing to have a reduced charge of grease for accelerated aging, often 10-20% of the nominal charge. As a general rule, only a small percentage of the grease is actively involved in lubricating the tribological contacts at any given time, with the majority of the grease serving as a lubricant reservoir for when oil is consumed or lost over time. Thus, in other protocols the initial grease reduction is used as a further acceleration factor.

Test Procedure

Prior to the aging test, test bearings as received are photographed and their general condition is noted. Test specimens are then assigned to a position in the rig which can be assigned based on desired operating conditions, as heat transfer characteristics of the chamber cause specimens closer to the center of the chamber to be exposed to higher temperatures. Slots are then machined into the outer races of the bearings with ball mill, approximately 0.75 mm deep, where the spring plunger tips will engage the outer races to prevent rotation.

The test specimens are then assembled onto the test shaft with annular spacers placed between adjacent specimens. A bearing nut is then tightened onto the shaft, compressing the stack of inner races and spacers and locking them to the shaft. The shaft is then installed in the test rig, outer races of the test bearings are oriented and spring plungers screwed into engagement with the outer race slots such that some preload compresses the tip in the slot. The shaft is then spun by hand to verify smooth turning, then the shaft is coupled to the motor with a flex coupling. The motor is then turned on and spun at low-to-moderate speeds up to about 1000

rpm as a final check prior to closing the test chamber. Once closed, additional thermal insulation is placed over the chamber and appropriate guards are put in place.

To begin the aging tests, the motor speed is set to a moderate speed, often 1000 rpm, and the chamber temperature set point is set at an interim level of 50-75% of the final desired chamber temperature. A single thermocouple measures the chamber air temperature and thermocouples attached to the test bearing outer races provide further information on spatial distribution of temperature. The test is usually run at these conditions for approximately two to four hours to allow for temperature transients to stabilize, along with some settling of torque and vibration characteristics, as monitored by the instrumentation signals. Speed and temperature are then increased incrementally until they reach the final desired set points for a given test.

For the test specimens described by the current work, the motor speed was set at 2800 rpm for the first 3080 hours of the test and 3000 rpm for the remainder of test, due to some limitations of the motor controller in use at the time. This lower speed provides some mechanical working of the lubricant and kinematic motion of the bearing elements, even though it does not exactly match the conditions from the helicopter application. The lower speed also reduced the rig noise to more comfortable levels within the laboratory for these long duration tests.

The chamber temperature was set at 177 °C (350°F) for most of the duration of the test, with temperatures set slightly lower for some relatively short intervals such as when restarting the test after a shutdown. With this chamber set point, the maximum test bearing temperature as measured on the outer race, was approximately 140 °C (280 °F) as shown in Table 2. While the grease temperature within the bearing is expected to more closely follow the measured race temperature, chamber temperatures were restricted to 177 °C (350 °F) which is the limit of the performance range specified for MIL-PRF-81322 compliant grease.

Results

Test Data

During the course of the tests, several shutdowns of the test rig occurred due to either user intervention or limit exceedance on one of the instrumentation signals. When analyzing and presenting the test data, portions of the time trace were omitted where the motor speed fell below a threshold of 2000 rpm, as this was indicative of rig shutdown and no significant degradation of the lubricant is taking place. As shown in Figure 3, this resulted in slight compression of the time traces presented from about 6200 hours of elapsed time where data was being taken, to about 5806 hours of relevant aging time.

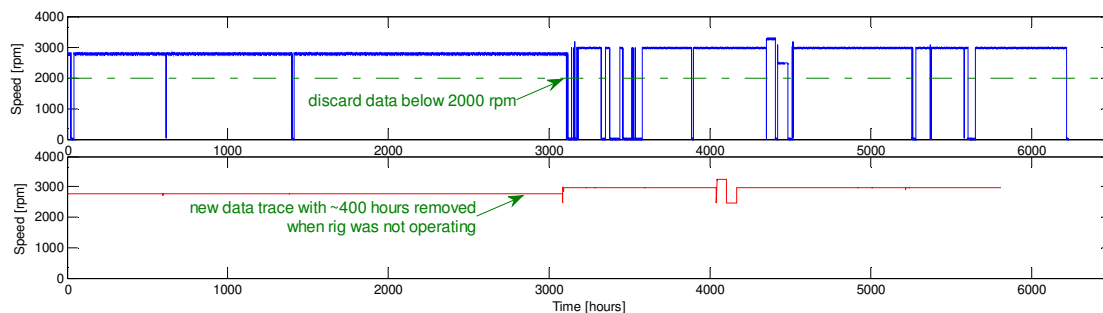


Figure 3 – Filtered and unfiltered speed signal for (a) all 6200 hours of recorded test data, and (b) only 5806 hours of test time when grease was aging

Based on the test duration and sampling rate of 1 Hz, the test produced over 2×10^7 lines of data which required some reduction to make analysis more manageable. Furthermore, due to signal ranging and resolution of the analog-to-digital conversion of the rms vibration signals, centered moving averages of the signals were performed to reduce the effects of quantization and as a de-noising step. From a small subset of the time domain data spanning approximately 195 hours, statistical moments of the signals up to the fourth (kurtosis) were compared before and after averaging as a check that important signal attributes including transient features were retained. The data was then parsed to retain only one out of every 301 samples.

Time traces of significant signals are shown in Figure 4 for the entire 5800 hours of test time. The speed and chamber temperature profiles show the operational conditions to which the test bearings were exposed. The temperature of test bearing #1 only is shown to more clearly illustrate the response of a typical bearing temperature to changes in the operational conditions. Two channels of rms vibration and the motor current signal are also plotted, and trends over time can be seen along with responses to defined bearing failures and changes in operating conditions. Figure 5 provides time traces of the test bearing temperatures for approximately 120 hours of the test, and illustrates typical temperature stability behavior and reaction to set point changes.

Over the course of the aging test, five bearing failures were identified and resulted in removal of a bearing. These are labeled in Fig. 4 as “BF1,” “BF2,” “BF3,” and “BF4,” with the fifth failure noted after the completion of testing. The specific bearings associated with these failures along with their circumstances are detailed in later sections.

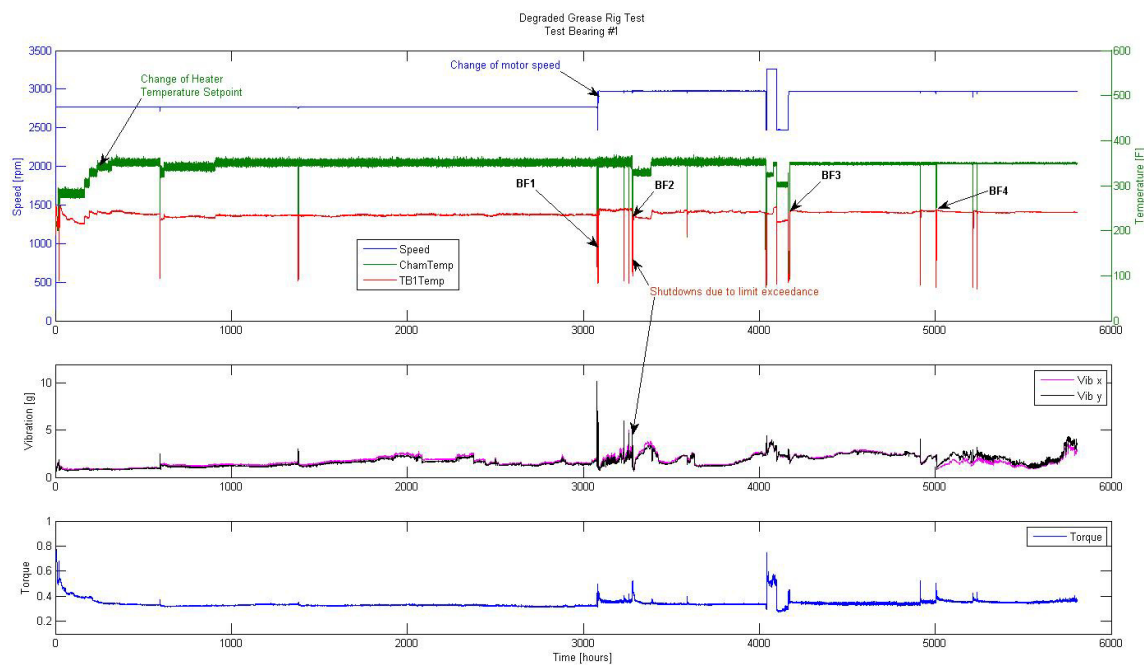


Figure 4 – Signal plots for Test Bearing #1
(BF1: RB#2/a, BF2: TB#6/a, BF3: TB#5, BF4: TB#4)

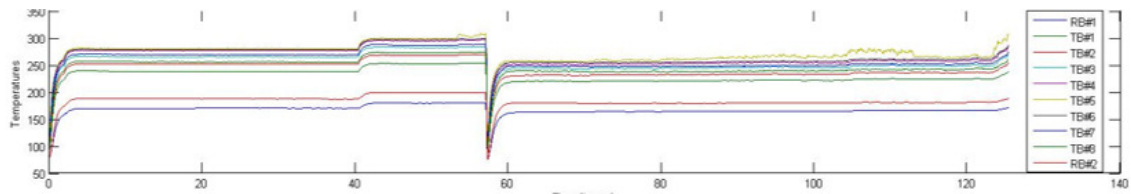


Figure 5 – Measured time traces of bearing outer race temperature for 120+ hours

Figure 6 shows the average temperature profile of the test and rig bearings over the duration of the test. The bearings marked in red are the ones that failed during testing. The outer race temperature averages fall within a total range of approximately 40 °F.

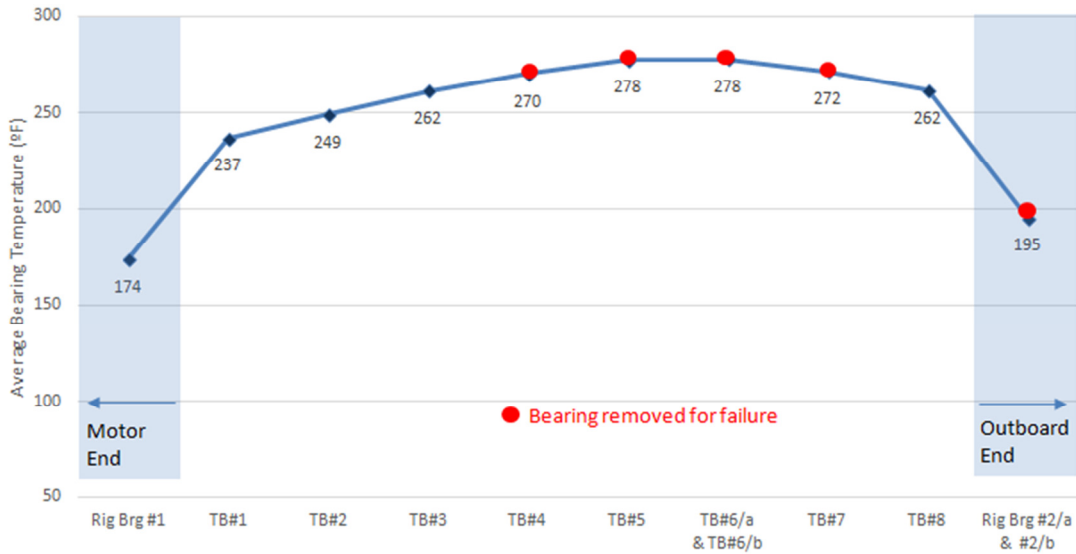


Figure 6 – Mean outer race temperatures of test and rig bearings

Condition of Failed Bearings

Bearing Failure #1 – S/N B626-25677 at Rig Brg #2/a

The failure of the rig bearing RB#2/a (bearing failure #1, or “BF#1”) appeared to be the most conservative removal, as this was done mostly as a precaution since it was not specifically identified as a test bearing and a rig bearing failure was to be avoided. At approximately 3077 hours of elapsed test time, a limit exceedance on rms vibration of 3.0 g was recorded and the test stopped automatically (denoted “SD5” in Fig. 7). After restart, vibration and torque started considerably higher than their previous steady values, although decreasing. This behaviour is quite typical for a cold restart of the test rig. Following the restart, the rig bearing temperature triggered a shutdown at 200 °F after only a few hours of additional runtime, even though the vibration by this time had dropped to a lower value than before the first shutdown. It was thought that the temperature exceedance so soon after the rms vibration exceedance could be indicative of an impending failure of this rig bearing, and so it was removed and replaced with a new bearing.

After removing the seal from this rig bearing (S/N B626-25677), the grease appeared dry and depleted though still somewhat pliable. Grease remaining on the seal had a slight appearance

of cracking like dried mud, and detailed chemical studies are reported separately. Turning by hand, excessive clearance or substantially rough turning was not noted when compared to another bearing, but the rotation was somewhat noisy. By comparison, a control bearing turned more slowly due to the increased grease drag. Little grease was present in the bearing but it remained in small clumps on the inner diameter of the between the balls. Fig. 7 shows the behaviour of torque, vibration, and the rig bearing temperature prior to removal. Subsequent attempts to restart the rig caused the new bearing to trip the temperature limit, perhaps as the new bearing had initially higher torque during break-in. Exceedance limits were set higher to allow the test to run longer.

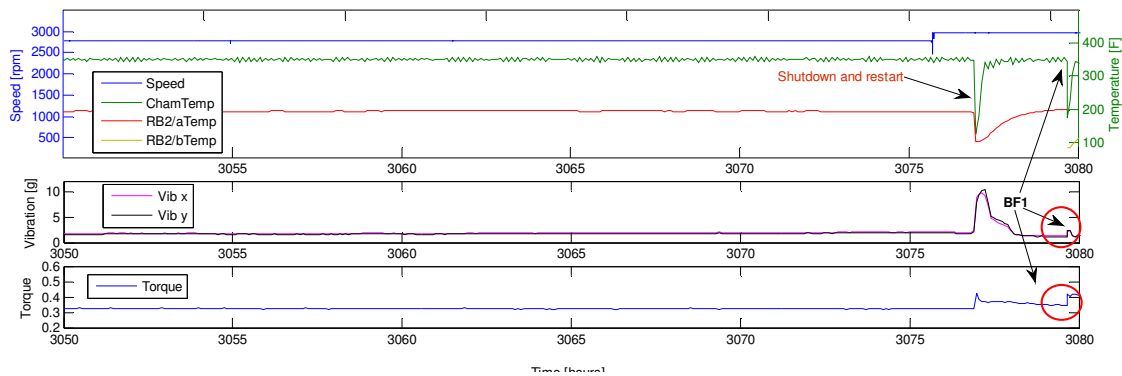


Figure 7 – Detail of bearing failure BF1 on Rig Bearing RB#2/a, with a conservative shutdown around 3077 hours and then another shutdown at 3079 hours with a precautionary removal of the rig bearing

Bearing Failure #2 – S/N B626-79706 at Test Brg #6/a

The second failed bearing was removed from the rig following a series of shutdowns that occurred not long after the replacement of rig bearing #2. At approximately 3230 hours of test elapsed time, the test stopped on an exceedance of rms vibration at 4.0 g. After restarting the test with slightly higher limits, the test again stopped about 17 hours later due to an exceedance of the TB#6 outer race temperature of 300 °F. After another restart with higher temperature limits, another shutdown was triggered approximately 21 hours later due to the temperature of TB#6 exceeding 310 °F. Earlier quasi-steady readings from this race thermocouple were between 270-280 °F, so this final limit exceedance corresponded to a 30 – 40 °F increase. The bearing was declared to have failed and was removed and replaced. At this point it was noted that the bearing was very difficult to turn by hand, though it had not completely seized. Figure 8 show time traces of approximately 80 hours of test time around the point where the bearing was deemed to have failed, “BF2” at approximately 3278 hours.

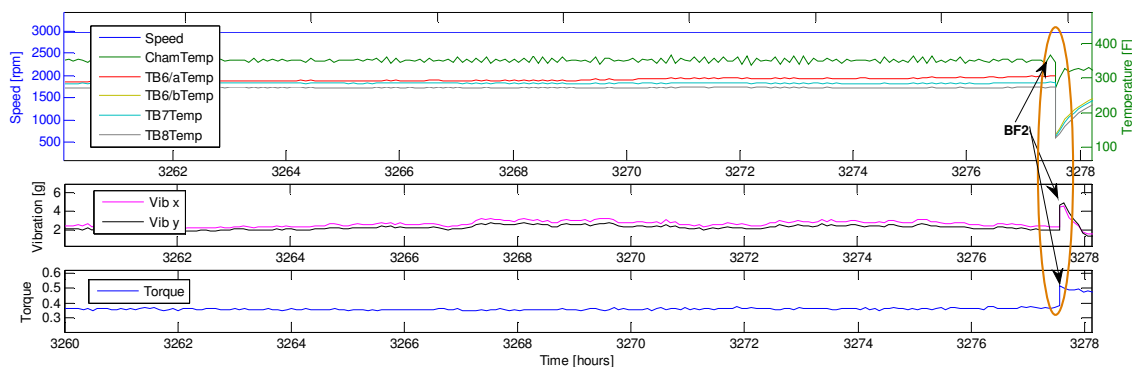


Figure 8 – Detail of bearing failure BF2 on Test Bearing TB#6/a, showing slight trending upward of TB#6/a temperature prior to shutdown and removal

Bearing Failure #3 – S/N B626-56171 at Test Brg #5

After approximately 4000 hours, the test was interrupted and the rig was moved to a new location. There was a resulting period of approximately one year during which the rig remained in conditioned storage with the test specimens left installed. The test was then resumed, and within the first 150 hours of operation after storage, the third bearing failure occurred on test bearing #5. As shown in Fig. 9, the race temperature for this bearing showed some signs of excursion for a period of at least 35 hours prior to failure. Since restarting the test following storage, speed set points were varied from 2500 rpm up to 3250 rpm to determine noise and vibration characteristics in the new location. This departure from the usual steady test conditions result in some uncertainty in the contribution to aging rate and appropriate threshold criteria for rig shutdown. Although race temperature and torque variations preceded the bearing failure by at many hours, the test shutdown occurred immediately following an increase in speed set point from 2500 rpm to 3000 rpm, and chamber temperature set point from 300 °F to 325 °F. The bearing (SN #B626-56171) was removed for inspection, but due to a lack of additional test specimens it was replaced with spacers.

At the initial inspection the bearing seals appeared darkened. After its removal from the test rig the grease had a dried appearance with almost a granular texture, and was adhered to the cage. Some areas along the seal lip at the inner race had darkened, brittle and continuous sections of grease. The rivets were pressed out of the cage and the cage removed for inspection of the race surfaces, which is shown in Fig. 10.

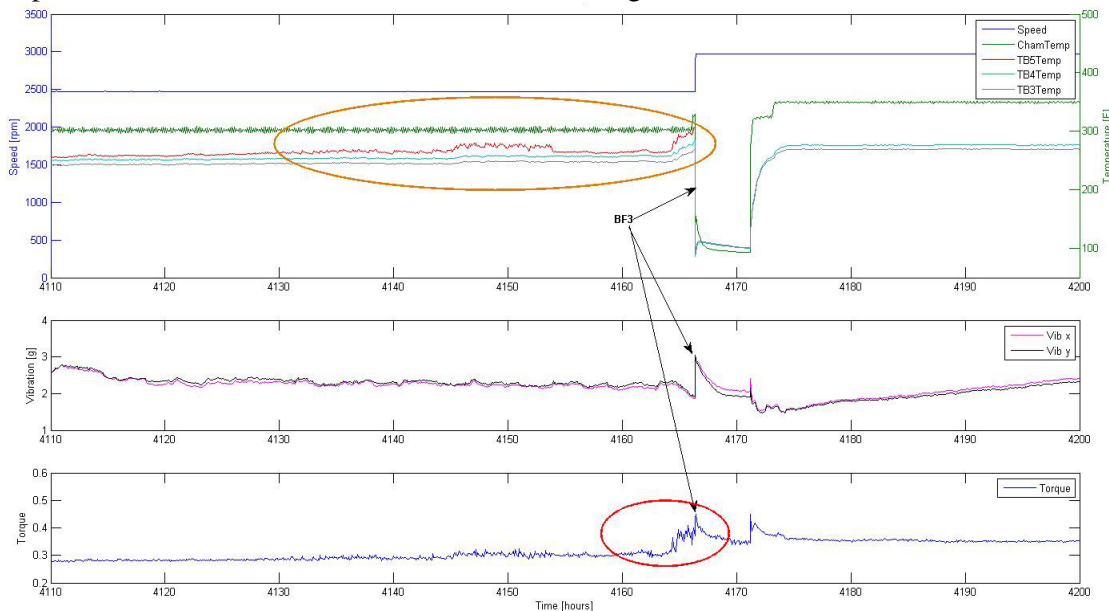


Figure 9 – Detail of bearing failure BF3 on Test Bearing TB#5, showing significant test bearing temperature excursions and torque increase prior to shutdown and removal

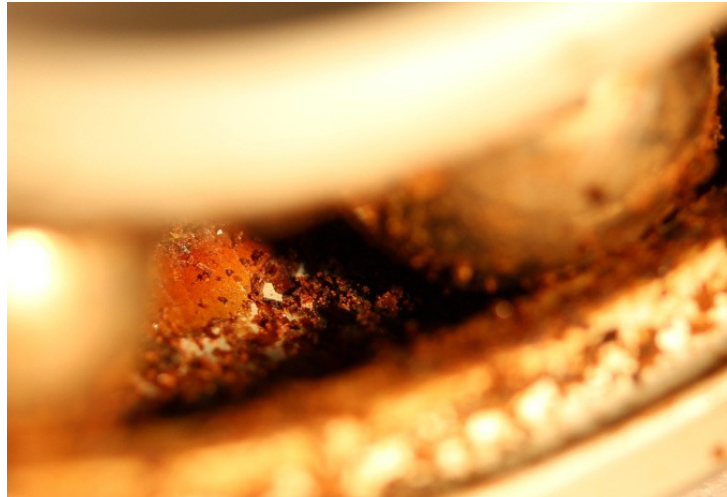


Figure 10 – Photograph of failed test bearing #5 after removal of cage, showing visual condition of grease on the outer race in different locations

Bearing Failure #4 – S/N B626-80711 at Test Brg #4

The fourth bearing deemed to have failed was removed following an exceedance of torque. An attempt to restart the test with higher torque shutdown limits resulted in the outer race breaking loose from the spring plunger and ripping the thermocouple open, so the bearing was deemed to have failed and removed from the rig to be replaced with spacers. The final 100 hours of the bearing operation is given in Fig. 11, showing upward trends in vibration somewhat erratic torque in the final hours prior to failure, and some fluctuations in temperature, though not as substantial as in the previous test bearing failures. The bearing had seized with a few degrees of rotational freedom, where dried grease pockets had settled on the outer race between rolling elements and were not easily overrun. Fig. 12 shows the visual condition of the bearing after it was removed for failure.

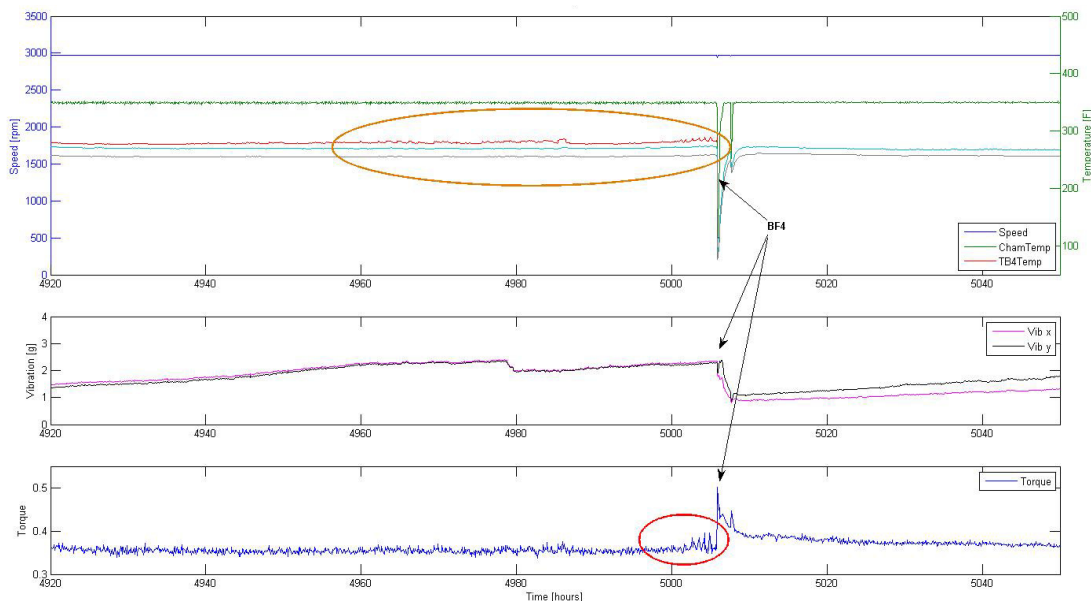


Figure 11 – Detail of bearing failure BF4 on Test Bearing TB#4, with temperature, vibration and torque signals showing deviations in the hours preceding failure



Figure 12 – Photographs of failed test bearing #4, (left) showing cage areas and (right) showing outer race surface between rolling elements

Bearing Failure #5 – S/N B626-55880 at Test Brg #7

Finally, near the end of the recorded data at 5800 hours, indications of high vibration, torque, and temperature of test bearing #7 (SN # B626-55880) indicate potential impending failure, with the values of these signals trending upward and exhibiting seemingly erratic behavior for at least 200 hours prior to the termination of the test. This behavior is shown in Fig. 13.

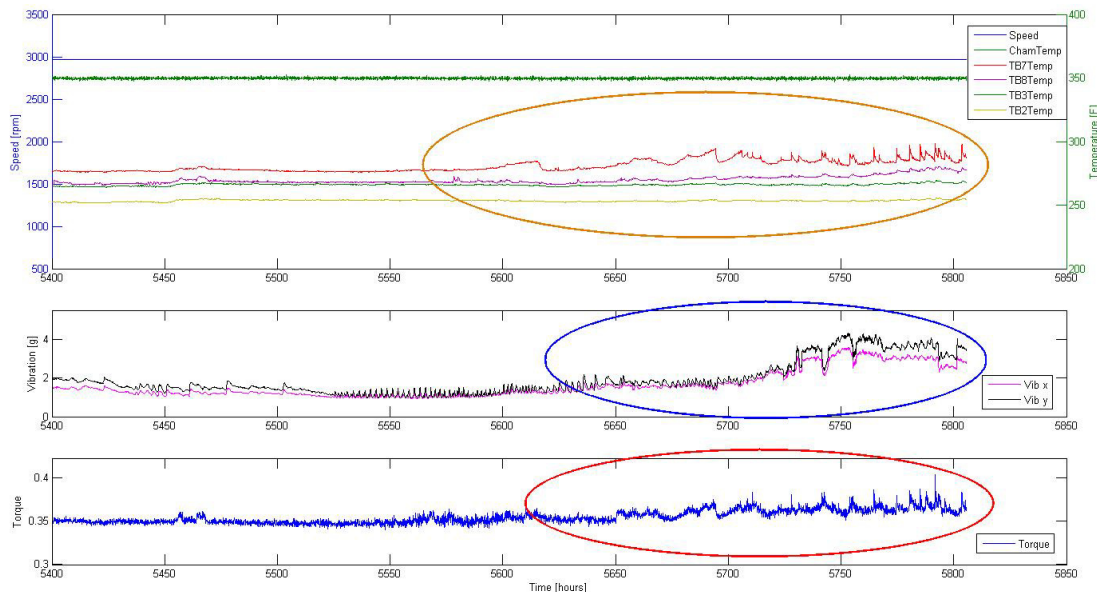


Figure 13 – Final hours of rig testing, showing erratic temperature, vibration and torque

Grease Condition

The specimens declared to have failed were inspected to determine the failure mechanism. The seals were found to remain intact and functional, although the seal material was found to have become brittle from the exposure to elevated temperature. With the seals removed, the grease in the failed bearings was found to have a dry, brown and cracked appearance and was retained on the seals or clinging to the cage. When pried with a pick, the grease flaked off in small dry clumps or flakes in some cases, while in others it remained slightly pliable. On some of the more advanced failures, small mounds of dried caked grease were noted on the outer race between rolling elements. When the remaining grease products were washed from

the bearings with solvent, corrosion damage was not observed despite what may appear similar to corrosion in the above photographs. When turning by hand, there was a very large periodic torque variation notable as each successive ball attempted to overrun those grease deposits. In some instances this torque variation was large enough to trip an overcurrent protection in the motor controller.

A somewhat qualitative and non-rigorous definition of bearing failure is employed in this work, where generally repeated shutdowns due to high torque, vibration or bearing temperature are taken as indicative of *impending* failure. Given the normally light loads on these bearings, the failure mode if allowed to progress is expected to be cage failure rather than rolling contact fatigue. Since the failures were not permitted to progress until gross failure of bearing elements (such as cage failure) or generation of large wear debris, it is not certain how much longer the bearings would have been able to operate acceptably prior to complete failure. The intention of the study was to characterize grease condition and behaviour near the impending failure rather than to generate exact lifetimes and catastrophic damage for forensic examination.

Using some of the techniques described in [7], Fourier Transform Infrared (FTIR) spectroscopy was performed on some samples and the ratio of oil to clay was estimated. Where typical practice assumes that a grease should be considered depleted when approximately 50% or less of the original oil remains, the specimens drawn from the failed bearings here were found to have only 20% to 30% of the original concentration of oil remaining. A more detailed study of the chemical interrogation of degraded grease samples will be reported separately.

Conclusions

Accelerated testing of UH-60 viscous bearings supporting the tail rotor drive shaft has been conducted. Over the course of approximately 5800 hours of aging tests at elevated temperature, four bearings were declared to have failed and were removed and inspected. Slight increases in bearing temperature, as well as increases in rms vibration and torque generally accompanied the onset of failure, however sensor placement, measurement accuracy and the simultaneous operation of ten identical bearings make it difficult to reach strong conclusions on the implications for diagnostic monitoring by those sensor types. Despite challenges in development of exact thresholds for bearing temperature limits especially outside of the very carefully controlled conditions on this particular test rig,

It is significant that the mechanism of grease degradation seen in this bearing in a simulation environment is more likely one of lubricant depletion and not mechanical or chemical breakdown of the base oil or thickener. As a lightly loaded bearing, it may be possible for various methods of diagnostic monitoring to provide robust detection without significant risk of catastrophic failure. Further testing of these bearings at conditions better representing those in a helicopter, along with better information on specimen and lubricant pedigree and follow on rig test of the degraded components, may allow the component to transition away from scheduled maintenance in favour of diagnostic monitoring.

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