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Abstract

The objective of this study was to illustrate the importance of combining Health Usage Monitoring Systems (HUMS) data with usage monitoring system data when detecting rotorcraft transmission health. Three gear sets were tested in the NASA Glenn Spiral Bevel Gear Fatigue Rig. Damage was initiated and progressed on the gear and pinion teeth. Damage progression was measured by debris generation and documented with inspection photos at varying torque values. A contact fatigue analysis was applied to the gear design indicating the effect temperature, load and reliability had on gear life. Results of this study illustrated the benefits of combining HUMS data and actual usage data to indicate progression of damage for spiral bevel gears.

Background

Power train health is a critical part of a rotorcraft health management system since no other air vehicle relies on the propulsion system for propulsion, lift and maneuvering through a transmission with critical single load paths. Many rotorcraft are equipped with on-board Health Usage Monitoring Systems (HUMS) to detect anomalies in transmission dynamic mechanical components by monitoring vibration signatures produced by the damaged component. The results from vibration signatures, referred to as condition indicators (CI), are used to indicate component health or condition. Some failure modes of transmission components generate and introduce debris into the oil. Measurement of debris in the oil can be used to help assess transmission health. Oil debris measurements can be off-line, performing oil analysis at periodic maintenance intervals or on-line, using oil debris sensors installed on board the aircraft to detect metallic debris in real-time (Ref. 1).

Usage monitoring is another important HUMS function. Helicopter transmission components are designed per assumed flight regimes and usage spectrums. Usage monitoring consists of measuring torque, operating hours, flight regimes and operating conditions to track component "actual" usage. If "actual" usage is measured, and component usage is less severe than the life it was designed for, the time in operation of a component may be increased as shown in the green in Figure 1. If component usage is more severe than the life it was designed for, the time in operation of a component should be decreased for safety reasons as shown in the red in Figure 1.

In practice, fatigue critical components in helicopters are designed to a specific service life in hours and removed from service before they reach those operational hours. The service life hours are determined using the following information (Ref. 3):

1. Fatigue strength S-N (stress-cycles) curves that relate component design, stress, number of cycles, operating conditions and reliability.
2. Load usage spectra from flight regimes from assumed mission profiles.
3. Cumulative fatigue damage calculations that tie loads back to fatigue life. The Palmgren-Miner Rule is one fatigue damage theory that is applied to the component properties to determine life. Applying this theory, the total life of a part is estimated by adding the percentages of applied load cycles per allowable cycles for critical loads and calculating a damage fraction.

However, fatigue life calculations do not take into consideration manufacturing defects or damage due to the aircraft environmental conditions or operational conditions that can cause premature failure of components. These types of unanticipated failures can be detected by the condition indicators in the HUMS.

Typically, within a HUMS system, the CI and usage data are acquired, stored, tracked, trended and monitored separately (Ref. 1). Passing key information between both systems and integrating this information has the potential to improve the performance and reliability of both systems. An indication from the CI that damage has occurred can be fed into the usage monitoring system to increase the damage fraction, while the torques and rotor turn times monitored by the usage monitoring system can be fused with the CI to indicate progression of damage and remaining time in operation.

The objective of the research is a first step in illustrating how key operational parameters measured by the usage monitoring system can be used to improve determination of the health state of a dynamic mechanical component. The targeted components are spiral bevel gears. Oil debris generated and operational parameters were measured during tests performed on spiral bevel gear sets (pinion/gear) in the NASA Glenn Spiral Bevel Gear Fatigue Rig when pitting fatigue damage occurred on gear and pinion teeth. The effect of torque on damage progression, once gear tooth damage was initiated, were compared to the rate of oil debris generated. Cycles and loads were correlated to gear fatigue life through

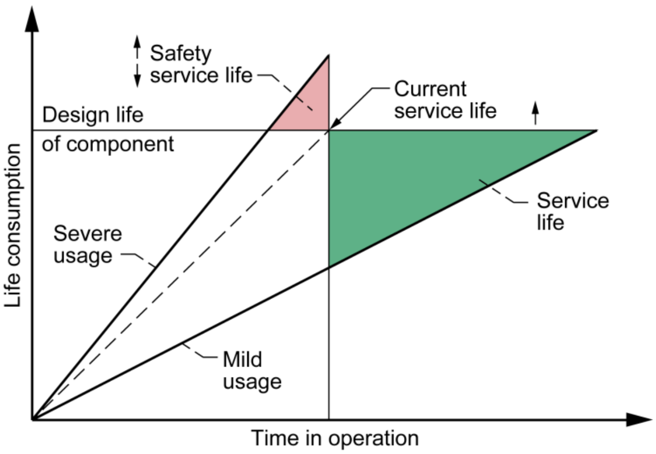

Figure 1.—Illustration of life consumption based on usage (Ref. 2).

the definition of an S-N curve based on the allowable contact stress, number of cycles and operating conditions. Results of this analysis will illustrate the benefits of combining usage operational parameters with condition indicators when detecting gear health.

Test Facility

Tests were performed in the NASA Glenn Spiral Bevel Gear Test Rig at NASA Glenn Research Center. A detailed description of this test facility can be found in References 4 and 5. The Spiral Bevel Gear Test Facility is illustrated in Figure 2. In addition to developing gear health monitoring tools, the test facility has been used to study the effects of gear material, gear tooth design, and lubrication on the fatigue strength of gears. Two sets of spiral bevel gears are installed in the test rig and tested simultaneously. Facing the gearboxes per Figure 3, the left gear set (pinion/gear) is referenced as left and the right gear set (pinion/gear) is referenced as right.

Figure 3 shows the cross-sectional view of the rig driveline. The facility operates in a closed-loop arrangement where the load is locked into the loop via a split shaft and a thrust piston that forces a floating helical gear axially into mesh. The drive motor supplies the facility with rotation and loop losses via v-belts to the axially stationary helical gear. The spiral bevel gears on the left side operates where the pinion drives the gear in the normal speed reducer mode and the right side of the facility acts as a speed increaser. However, the concave side of the pinion is always in contact with the convex side of the gear on either side of the test facility. Load and speed of the left side gear output shaft are monitored by torque and speed sensors.

Turbine engine oil that meets DOD-L-87354 specifications is used in the test rig. Both gear sets are lubricated with oil jets pumped from an oil reservoir. The lubrication from the gearbox then exits the gearbox and flows through an oil debris sensor. A strainer and a 3 µm filter are located downstream of the oil debris (OD) sensor to capture any debris before returning to the sensor and gearbox.

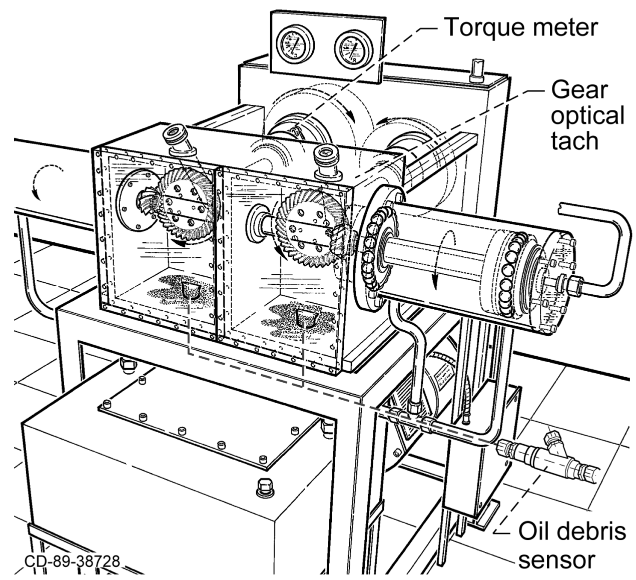

Figure 2.—Spiral Bevel Gear Fatigue Rig.

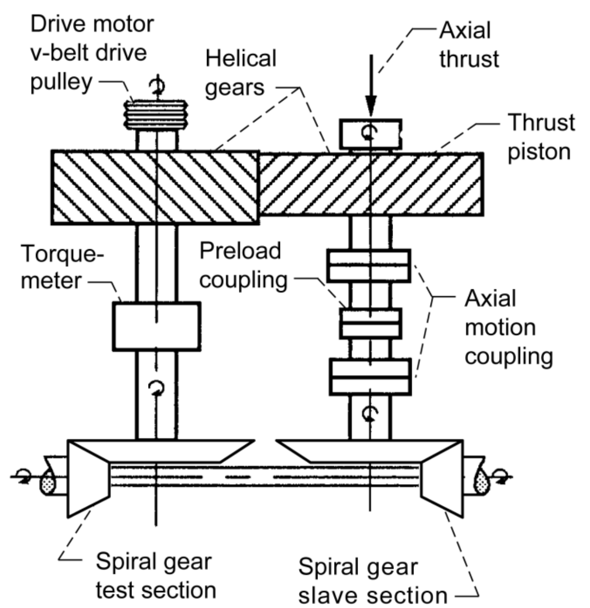

Figure 3.—Cross section of Spiral Bevel Gear Fatigue Rig.

A non-contact rotary transformer shaft mounted torque sensor was used to measure torque during testing. Speed was measured by an optical tachometer mounted on the right gear shaft that measures a pulse per each gear shaft revolution. Thermocouples were used to measure inlet and outlet oil temperatures. An inductance type oil debris sensor was used to measure the debris generated during fatigue damage to the gear teeth. The sensor measures the number of ferrous particles and their average size based on user defined particle size ranges or bins that is in turn used to calculate the cumulative mass by average particle size and the density of steel. Data was recorded once every minute with a facility data acquisition system.

The gears tested were prototypes of gears designed and specified to represent a rotorcraft drive system gear mesh and fit within the space available in the spiral bevel gear fatigue rig gearboxes. The gears were made from a steel alloy CEVM 9310, carburized, hardened and ground. However, the three gear sets tested, consisting of left pinion/gear and right pinion gear sets, differed from the final design in that they were not shot peened, the surface roughness on the pinion ranged from 11 to 15 μin. and the surface roughness on the gears ranged from 26 to 35 μin. In addition, copper plating used for masking parts of the gear during carburization was not removed from the edge of the gear teeth for this prototype design.

The prototype gear sets tested have 6.4 in. diametral pitch, 20° pressure angle, 25° spiral angle, 0.94 in. face width and 2.15:1 gear ratio. The gears have 41 teeth and the pinions have 19 teeth. The test gears were designed to operate at a gear speed of 3500 rpm and torque of 8000 in.-lb, pinion speed of 7553 rpm and torque of 3707 in.-lb and 240 to 265 °F inlet oil temperatures with an American Gear Manufacturers Association (AGMA) calculated contact stress of 237 ksi.

Experimental Data

For this study, three gear sets were tested at a gear speed of 3500 rpm and pinion speed of 7553 rpm. Gear torque varied from 4000 to 8000 in.-lb and pinion torque varied from 1854 to 3707 in.-lb. At the start of each test, a run-in was performed for a minimum of 1 hr at 4000 in.-lb gear torque/3500 rpm gear speed and 1854 in.-lb pinion torque/7553 rpm pinion speed. Contact cycles accumulated at a rate of 210,000 per hour for the gear and 453,180 per hour for the pinion.

At completion of the run-in, inspection photos were taken of the left and right gear and pinion teeth. Inspection photos were then taken throughout the test to document damage progression to the gear teeth.

Damage progression on the gear teeth depends on several factors that include: 1) the size, shape and location of the initiating flaw; 2) the type of failure mode; and 3) the gear set operational conditions. The effect of operational conditions and variance in damage progression rates indicates the importance of a diagnostic tool that can both detect the presence and location of damage and quantify the level, magnitude and rate of damage progression.

This preliminary study focuses on the effect measured torque has on the progression of damage to the gear and pinion teeth. The failure modes observed for this study were defined by class (contact fatigue), general mode (macropitting) and degree (progressive) per AGMA standards for gear wear terminology outlined in Reference 6.

Using Reference 6 for tooth damage terminology, a numbering scheme per Reference 7 was developed to streamline the identification of gear damage. Table 1 illustrates this numbering scheme for the types of damage observed during these tests. Table 2, Table 3, and Table 4 summarize the observed failure modes observed on the gear teeth during each inspection interval. The tests ran until macropitting/spalling (4.3.4) was observed on two or more teeth. The reader should note that damage mode 6.0, fracture, was a failure mode observed due to the prototype gear design. In the prototype design, the edges of the teeth were not chamfered per design specifications and the copper masking was left on the edge of the teeth. This caused copper edges to fracture during testing. Since the oil debris sensor does not measure non-ferrrous particles, this type of debris was not measured by the sensor.

TABLE 1.—NUMBERING SCHEME FOR NOMENCLATURE OF GEAR FAILURE MODES (REF. 6)

Class	General mode	Specific mode or degree
2.0 Scuffing	2.1 Scuffing	
		2.1.1 Mild
		2.1.2 Moderate
		2.1.3 Severe
4.0 Contact Fatigue	4.1 Subcase Fatigue	
	4.2 Micropitting	
	4.3 Macropitting	
		4.3.1 Initial
		4.3.2 Progressive
		4.3.3 Flake
		4.3.4 Spall
6.0 Fracture	6.0 Fracture	

TABLE 2.—FAILURE MODES DURING TEST 1

Test 1	Rdg (min)	Hour(s)	Gear left	Pinion left	Gear right	Pinion right
Start	0	0.00	N/A	N/A	N/A	N/A
Run-in	60	1.00	N/A	N/A	N/A	N/A
Inspection	108	1.80	N/A	N/A	N/A	N/A
Inspection	379	6.32	6.0 at et	4.2 2t	6.0 at et, 2.1.2 at	4.2 at, 2.1.3 at
Inspection	559	9.32	6.0 at et	4.3.2 3t, 4.3.4 2t	6.0 at et, 2.1.2 at	4.2 at, 2.1.3 at
Inspection	662	11.03	6.0 at et	4.3.4 5t	6.0 at et, 2.1.2 at	4.2 at, 2.1.3 at

Key: xt= number of teeth; at=all teeth; et=edges of teeth

TABLE 3.—FAILURE MODES DURING TEST 2

Test 2	Rdg (min)	Hour(s)	Gear left	Pinion left	Gear right	Pinion right
Start	0	0.00	N/A	N/A	N/A	N/A
Run-in	69	1.15	N/A	N/A	N/A	N/A
Inspection	184	3.07	6.0 at et	N/A	N/A	N/A
Inspection	520	8.67	6.0 at et	4.3.2 at	6.0 at et, 2.1.2 at	6.0 at et, 2.1.2 at, 4.3.4 1t
Inspection	686	11.43	6.0 at et	4.3.2 at, 4.3.4 1t	6.0 at et, 2.1.2 at	6.0 at et, 2.1.2 at, 4.3.4 5t, 4.3.2 4t

Key: xt= number of teeth; at=all teeth; et=edges of teeth

TABLE 4.—FAILURE MODES DURING TEST 3

Test 3	Rdg (min)	Hour(s)	Gear left	Pinion left	Gear right	Pinion right
Start	0	0.00	N/A	N/A	N/A	N/A
Run-in	60	1.00	N/A	N/A	N/A	N/A
Inspection	489	8.15	6.0 at et	N/A	6.0 at et, 2.1.2 at	4.3.1 at, 2.1.2 at
Inspection	901	15.02	6.0 at et	6.0 at et, 4.3.1 at	6.0 at et, 2.1.2 at	4.3.1 at, 2.1.2 at
Inspection	1196	19.93	6.0 at et	6.0 at et, 4.3.1 at	6.0 at et, 2.1.2 at	4.3.1 at, 2.1.2 at
Inspection	1829	30.48	6.0 at et	6.0 at et, 4.3.1 at	6.0 at et, 2.1.2 at	4.3.1 at, 2.1.2 at
Inspection	4034	67.23	6.0 at et, 4.3.4 5t, 4.3.2 6t	6.0 at et, 4.3.1 at	6.0 at et, 2.1.2 at	4.3.1 at, 2.1.2 at

Key: xt= number of teeth; at=all teeth; et=edges of teeth

Figure 4 to Figure 9 are plots of the damage progression to the gear teeth and the effect of torque during the testing of the three gear sets. The first plot in Figure 4, Figure 6 and Figure 8, torque, in inch-pounds (in.-lb), plotted in blue per the left y-axis. Debris generated in milligrams, an indication of fatigue damage progressing, is plotted in orange per the right y-axis. The yellow diamonds on the x-axis indicate hours for gear teeth inspections. Representative photos of gear and pinion teeth damage modes and their progression at the inspections intervals are shown in the second figure. The six figures will be discussed in the following paragraphs.

Table 5 to Table 7 are summaries of the rate of debris generated during damage progression during each test per time in operation in hours. The debris generated per hour based on the inspection intervals is shown in the last column. Note that the amount of debris generated varied for each test due to the varying failure modes of the gear set. This data combined with the damage within each inspection interval is to be used to assess the overall trends of damage progression based on changes in torque applied to the gear set.

For test 1, Figure 5 illustrates the progression of damage to five left pinion teeth. The other photos are representative pictures of one left gear tooth, on right gear tooth and one pinion tooth. The damage to these gears did not change after the initial inspection. For the left pinion, no macropitting was observed on the pinion teeth 6 hr into the test. Nine hours into the test, 4.3.4 was observed on two pinion teeth. Eleven hours into the test, 4.3.4 was observed on five teeth.

For test 1, Table 5 displays the rate of debris generation measured was at 8.96 mg/hr when progressive macropitting (4.3.2) was first observed on several teeth as seen in Figure 4 and Figure 5. The rate decreased to 3.75 mg/hr when the torque dropped to 4000 in.-lb. The debris generated then increased from 3.75 to 27.69 mg/hr when torque increased from 4000 to 8000 in.-lb, and spalling macropitting (4.3.4) was observed on five gear teeth as seen in Figure 5.

For test 2, Figure 7 illustrates the progression of damage to six right and left pinion teeth. No macropitting was observed on the pinion teeth 3 hr into the test. Eight hours into the test, a pit was observed on a right pinion tooth. Several more pits were observed 11 hr into the test on the right pinion teeth and one pit on a left pinion tooth.

For test 2, Table 6 displays the rate of debris generation measured increased when torque increased from 6,000 to 8,000 in.-lb per Figure 6. For this test, the torque was not decreased when the macropitting was observed on the teeth per Figure 7. The 6.37 mg/hr rate of debris generation when spalling macropitting (4.3.4) was observed at 8,000 in.-lb was higher than the 3.75 mg/hr rate of debris generation when spalling macropitting (4.3.4) was observed at 4,000 in.-lb for test 1.

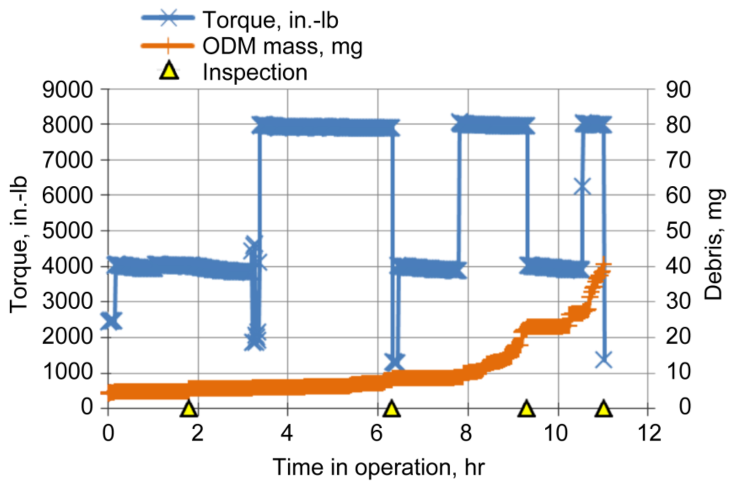

Figure 4.—Debris generated and torque during test 1.

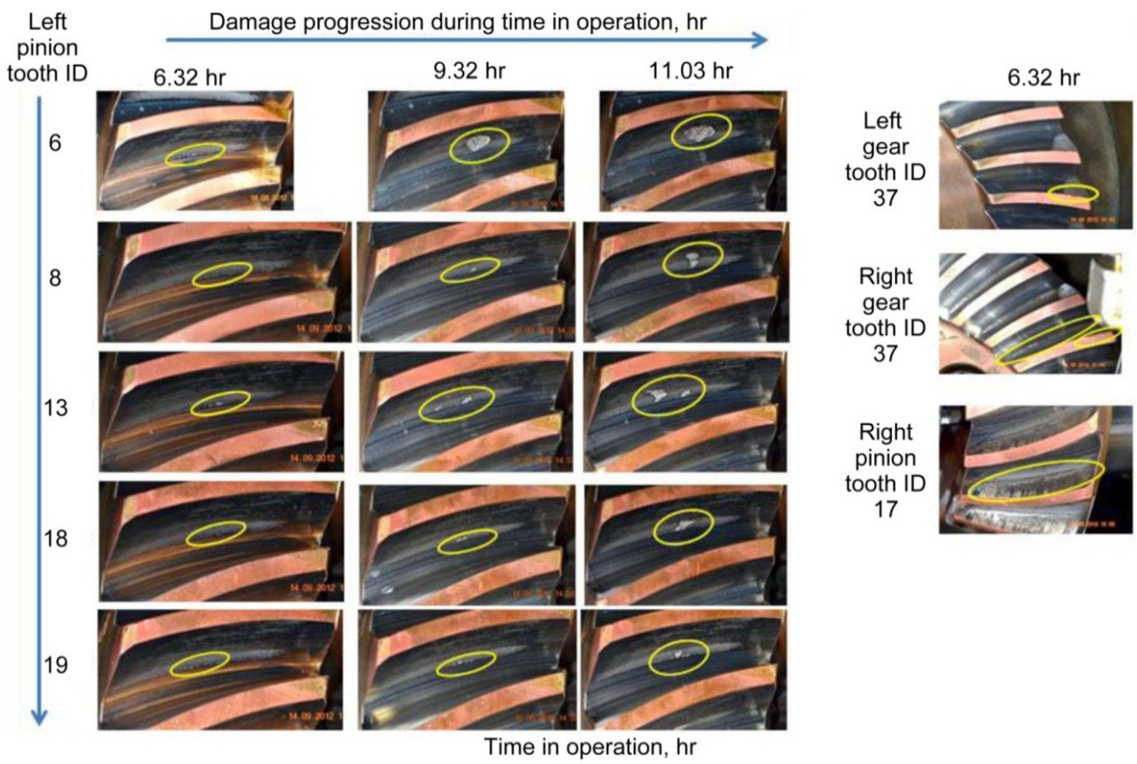

Figure 5.—Damage conditions on teeth during test 1.

TABLE 5.—TEST 1 DEBRIS GENERATED DURING DAMAGE PROGRESSION

Test 1	Rdg (min)	Hour(s)	Torque	ODM	Δ ODM	Δ hr	mg/hr
Start	0	0.00	0	4.15			
Run-in	60	1.00	4,000	4.69			
Inspection	108	1.80	4,000	4.69	0.00	0.80	0.00
	202	3.37	4,000	5.70	1.01	1.57	0.64
Inspection	379	6.32	8,000	7.72	2.02	2.95	0.68
	468	7.80	4,000	8.59	0.87	1.48	0.59
Inspection	559	9.32	8,000	22.18	13.59	1.52	8.96
	632	10.53	4,000	26.75	4.57	1.22	3.75
Inspection	662	11.03	8,000	40.59	13.84	0.50	27.69

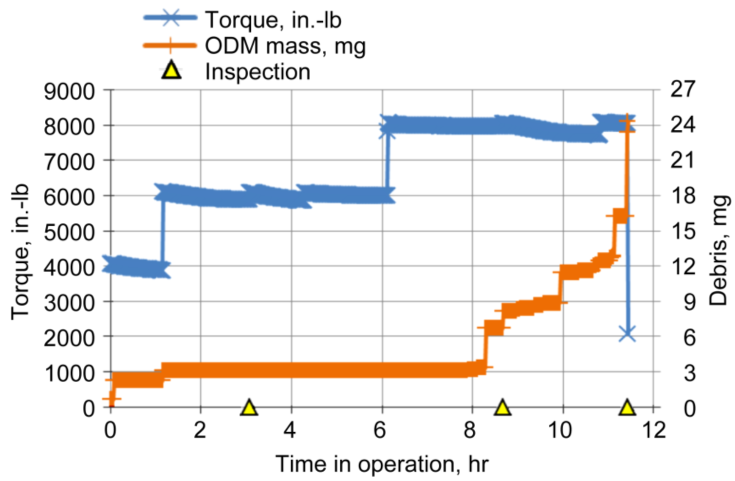

Figure 6.—Debris generated and torque during test 2.

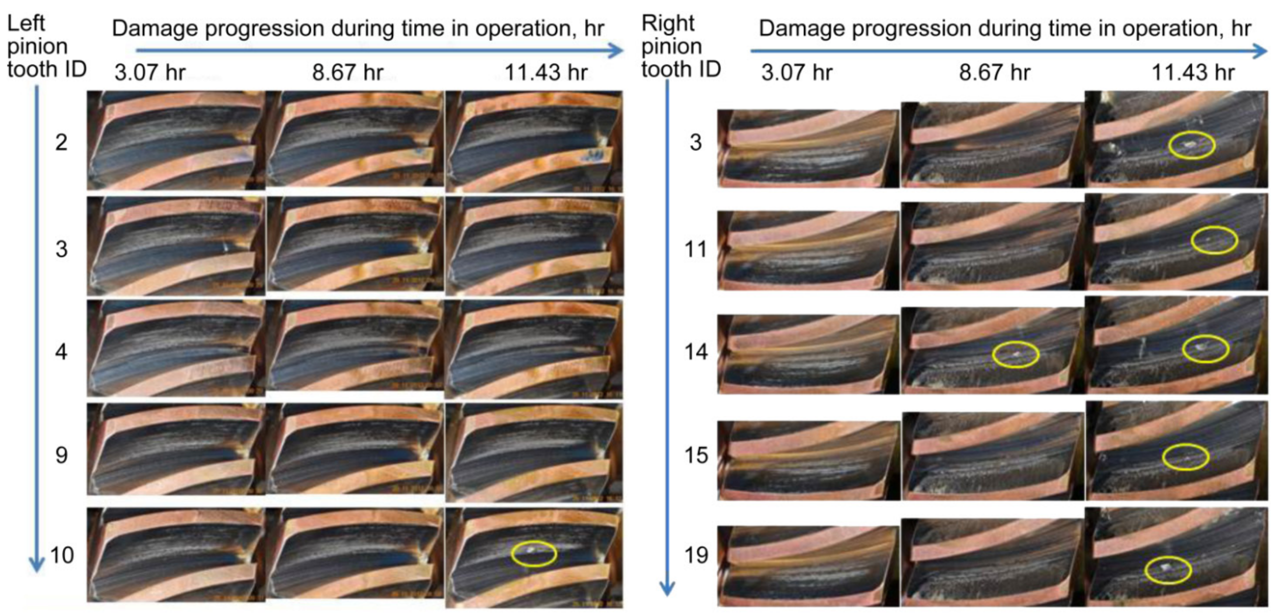

Figure 7.—Damage conditions on teeth during test 2.

TABLE 6.—TEST 2 DEBRIS GENERATED DURING DAMAGE PROGRESSION.

Test 2	Rdg (min)	Hour(s)	Torque	ODM	Δ ODM	Δ hr	mg/hr
Start	0	0.00	4,000	0.63			
Run-in	69	1.15	4,000	2.20			
Inspection	184	3.07	6,000	3.08	0.87	1.92	0.45
	367	6.12	6,000	3.08	0.00	3.05	0.00
Inspection	520	8.67	8,000	6.68	3.60	2.55	1.41
Inspection	686	11.43	8,000	24.30	17.62	2.77	6.37

For test 3, Figure 9 illustrate no macropitting was observed on the pinion teeth 30 hr into the test. Sixty-seven hours into the test, several pits were observed on the left gear teeth and micropitting on both the left and right pinion teeth. The torque remained at 6000 in.-lb during the test.

For test 3, Table 7 displays the rate of debris generation measured was at 0.51 mg/hr rate of debris generation when spalling macropitting (4.3.4) was observed at 6000 in.-lb per Figure 8 and Figure 9. This rate was lower than the 3.75 mg/hr rate of debris generation when spalling macropitting (4.3.4) was observed at 4000 in.-lb for test 1, and lower than the 6.37 mg/hr rate of debris generation when spalling macropitting (4.3.4) was observed at 8000 in.-lb for test 2.

Based on this preliminary analysis, for test 1 and 2, the rate of debris generated and damage progression to the gear and pinion teeth increased with increased torque values. Since damage progression depends on the size, location and shape of the initiating flaw, further testing is required to determine if once initiated, the damage progression can be slowed by decreasing the torque value.

TABLE 7.—TEST 3 DEBRIS GENERATED DURING DAMAGE PROGRESSION.

Test 3	Rdg (min)	Hour(s)	Torque	ODM	Δ ODM	Δ hr	mg/hr
Start	0	0.00	4,000	4.96			
Run-in	60	1.00	4,000	4.96			
Inspection	489	8.15	6,000	12.86	7.89	7.15	1.10
Inspection	901	15.02	6,000	19.82	6.96	6.87	1.01
Inspection	1196	19.93	6,000	23.40	3.59	4.92	0.73
Inspection	1829	30.48	6,000	41.07	17.67	10.55	1.67
Inspection	4034	67.23	6,000	59.74	18.66	36.75	0.51

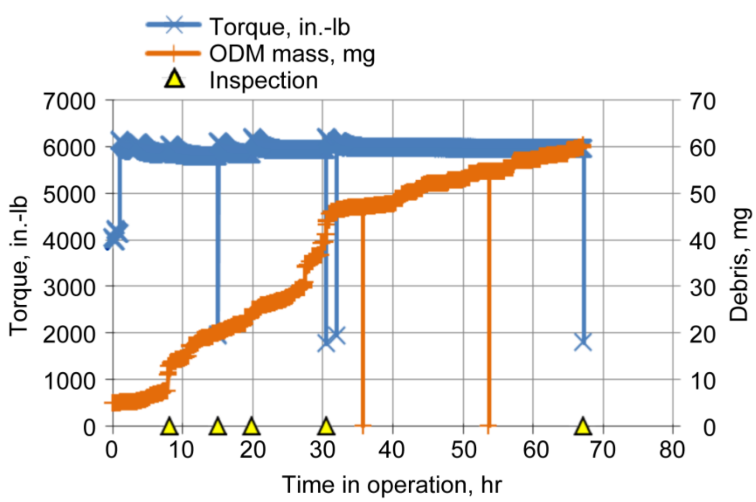

Figure 8.—Debris generated and torque during test 3.

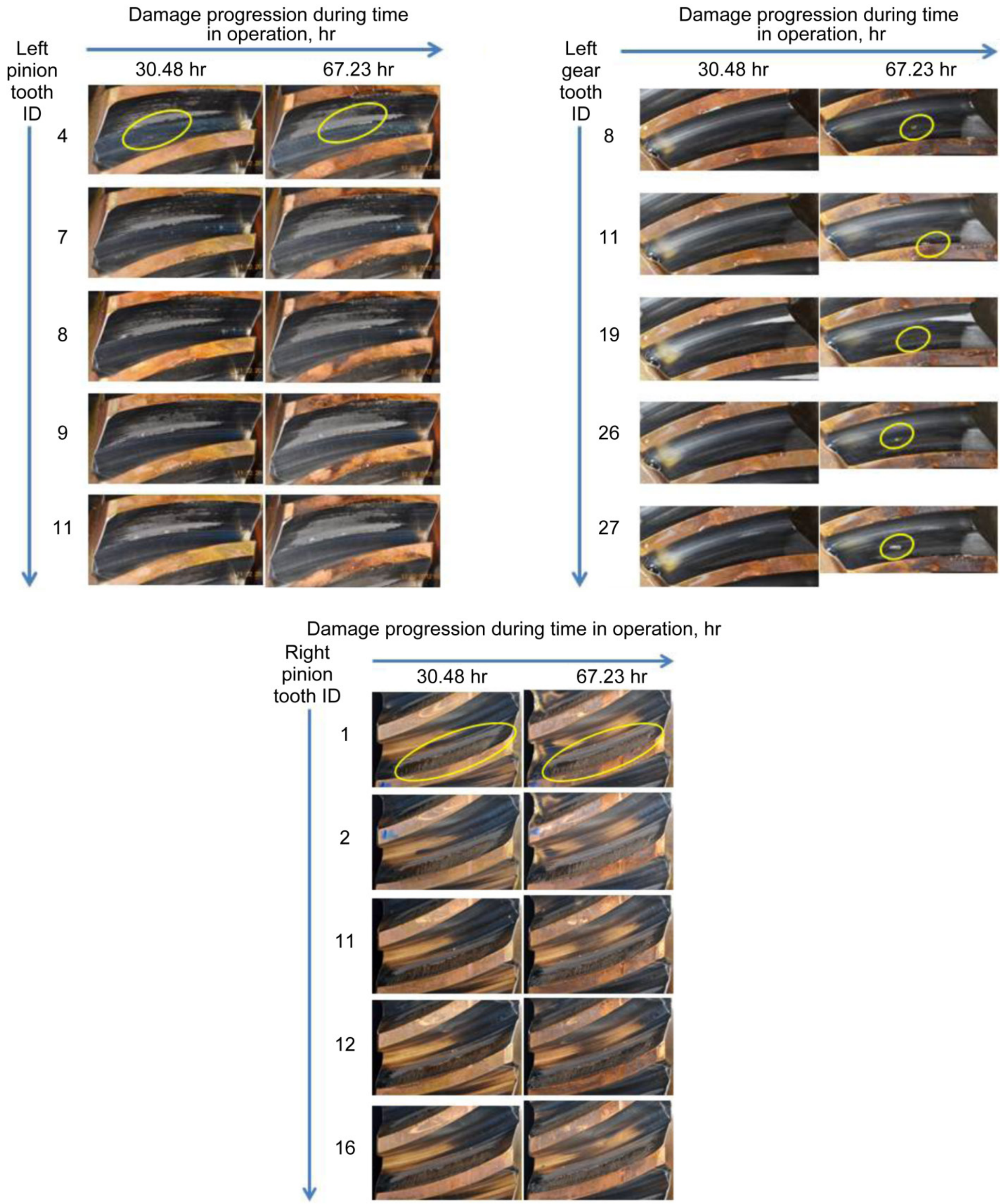

Figure 9.—Damage conditions on teeth during test 3.

AGMA Discussion

The previous section of this paper provides a preliminary assessment of the effect actual usage, using torque and rotational hours, has on damage progression to gear teeth. Although contact fatigue analysis applied to a new gear design defines a specific service life, aircraft environmental conditions can cause premature failure of components. Sometimes these conditions, such as operational climate and use, are not accounted for in the original design. A component failure per Reference 1 is defined as when a component cannot perform its intended function or the level of damage has reached a detectable level. Some component damage progresses slowly and the component can continue to function with a detectable level of damage.

From an operations standpoint, if the type of component failure mode is dependent on load, and the HUMS CI is capable of indicating this type of failure mode before the fault reaches a critical size, it may be possible to modify an operational condition, such as decrease the load, to extend the useful life of the component. However, contact fatigue analysis is used to define the probability of survival, to avoid initiation of surface durability damage. Once the flaw initiates and starts to progress, based on varying operational conditions, a new approach may be required.

A preliminary contact fatigue analysis was applied to this gear design and its operational conditions to show the effect on the life of these gears based on their operational conditions. The spiral bevel gear pinion was evaluated for various loading scenarios at different operational temperatures, to determine the effect on the expected time for damage initiation at a predetermined level of reliability.

The pinion design information and operational conditions within the Spiral Bevel Gear Fatigue Rig were utilized in a computer program (Ref. 8) that provides contact and bending stress indices. The design information included gear material, heat treatment, lubricant and desired reliability. The operational conditions included rotational speed, input torque and operating temperature. For a given design, rotational speed, and torque transmitted, the analysis provides the contact and bending stress indices. Contact and bending stress indices are criteria used to define gear tooth load capacity between mating teeth and its resistance to pitting damage on the gear tooth contact surface and the gear tooth base for bending (Ref. 9).

All of the input information described above has an effect on the allowable stress indices. Based on the input data, the program then calculates the margin of safety for the input design and operational conditions for the contact and bending fatigue behavior of the gear mesh. Note that 100 percent gear torque is 8000 in.-lb. The effect of torque on contact stress and bending stress indices in (ksi) for this design is given in Table 8.

TABLE 8.—EFFECT OF TORQUE ON CONTACT AND BENDING STRESS INDICES

% torque	Contact stress index, ksi	Bending stress index, ksi
10	75.1	4.2
20	106.1	8.4
30	130	12.7
40	150.1	16.9
50	167.8	21.1
60	183.8	25.3
70	198.6	29.6
80	212.3	33.8
90	225.1	38
100	237.3	42.2
110	248.9	46.5
120	260	50.7
130	270.6	54.9
140	280.8	59.1
150	290.7	63.4

Gear and pinion temperatures and applied loads have significant effects on gear and pinion life. The effect of load on the progression of damage to the gear teeth was observed in Figure 5, Figure 7 and Figure 9. The applied torque affects contact and bending stresses on all spiral bevel geared systems. In many cases the gears must operate at elevated temperatures. Elevated temperatures can also have detrimental effects on the aspects of a design that have to do with bending fatigue (material capability to carry load at elevated temperatures), contact fatigue (pitting—surface degradation) and scoring (surface interaction through the lubricating film thickness) that decrease the life of the gears.

In addition to loads and temperatures, the risk associated with a given design must be assessed. This is known as the reliability of the design. This is typically described as a percentage of reliability for a given load(s) for a number of hours (or cycles) of operation. The non-uniform loading a gear mesh might see can be evaluated utilizing a "Miner's Rule" summation (Ref. 10). The level of reliability the gears must maintain also affects gear and pinion life. Reliability refers to the probability the gear will fail. A 99 percent reliability is defined as 99 out of 100 gears will survive to x number of cycles at a given load. A 50 percent reliability is defined as one out of two gears will survive x number of cycles at a given load. These effects are illustrated in Table 9. Per the last column of the table, if you can accept a lower reliability, you can increase the number of life cycles since lower reliability allows higher allowable stress indices.

TABLE 9.—CALCULATED LIFE ASSESSMENT IN CYCLES FOR PINION AT DIFFERENT OPERATING CONDITIONS

% torque	500 °F life cycles (millions) 99% reliability	400 °F life cycles (millions) 99% reliability	300 °F life cycles (millions) 99% reliability	500 °F life cycles (millions) 50% reliability
10	>10,000	>10,000	>10,000	>10,000
20	>10,000	>10,000	>10,000	>10,000
30	>10,000	>10,000	>10,000	>10,000
40	>10,000	>10,000	>10,000	>10,000
50	>10,000	>10,000	>10,000	>10,000
60	700	7,000	>10,000	>10,000
70	340	1,600	>10,000	6,000
75	160	1,100	10,000	3,300
80	100	600	5,000	1,800
90	40	240	2,000	700
100	19	100	700	300
110	8	44	340	140
120	3.6	23	160	64

Per the highlighted line in Table 9 at 100% torque (8000 in.-lb gear torque), the life cycles at 500 °F pinion temperature is significantly lower than at 400 and 300 °F. An S-N curve of cycles versus allowable stress index was generated for the contact fatigue reliability based on blank or gear tooth temperatures for this gear design and is shown in Figure 10. The number of cycles is shown on the x-axis and the contact stress index is shown on the y-axis. The red dashed horizontal line is the contact stress index at 8000 in.-lb gear torque, 3707 in.-lb pinion torque. The blank temperature is the pinion tooth temperature. During the three gear tests the inlet oil temperatures ranged from 240 to 265 °F. The gear and pinion temperatures can be significantly higher than the inlet oil temperatures. Pinion tooth root temperatures were measured in this test rig on another gear design with 100 °F oil inlet temperatures at comparable load that ranged from 200 to 225 °F, over 100 °F above the oil inlet temperature (Ref. 11).

Methods to calculate pinion temperature based on operating conditions and pinion design were researched in the literature. Based on the operating conditions of the test rig, pinion temperatures were calculated using methods outlined in Reference 12. The method uses the following equations and parameters to calculate pinion temperatures.

$$\Delta T = \frac{G}{C_1}\sqrt{C_P}\,{K_T}^{0.75}\left(\frac{50}{50-s}\right)P_d^{0.6875}n_p^{0.3125} \qquad (1)$$

$$\frac{\sqrt{C_P}}{C_1} = 1.2096 \qquad (2)$$

$$K_T = \frac{T_P C_o C_m}{C_v F} \qquad (3)$$

Where ΔT is the temperature rise in °F; G is a geometry factor; C_1 is a thermal constant for gear material; C_p is an elastic coefficient; K_T is a load factor; s is the surface finish in micro-inches; P_d is the transverse diametral pitch in (1/in.); n_p is the pinion speed in rpm; factors C_o, overload, C_m, load distribution, and C_v, dynamic, are set equal to 1; and F is equal to the gear face width. Per Equations (1) through (3), if the pinion surface finish is 10 µin., the pinion temperature is calculated as 561 °F. This value increases to 631 °F if the surface finish of the pinion is 20 µin.

Table 10 is a summary of the hours the pinions will be free of contact fatigue damage at 99 percent reliability at the rig operating conditions of 8000 and 6000 in.-lb gear torque/3500 rpm gear speed and 3707 and 2780 in.-lb pinion torque/7553 rpm pinion speed at pinion temperatures of 300, 400, and 500 °F. The gear sets were tested at these conditions during test 1 and 2 as shown in Figure 4 and Figure 6. Per Table 10, 1 out of 100 gears tested should have failed at 42 hr at 8000 in.-lb torque. Based on the damage progression data collected during the three tests, macropitting was observed less than 10 hr into each test. For tests 1 and 2, 8000 in.-lb torque was applied for less than 5 of the 10 hr. For test 3, macropitting was observed within 10 hr at 6000 in.-lb load. This AGMA analysis over predicted the life of this gear design.

TABLE 10.—CALCULATED LIFE ASSESSMENT IN HOURS FOR PINION AT TEST CONDITIONS

% torque	500 °F life cycles, hr, 99% reliability	400 °F life cycles, hr, 99% reliability	300 °F life cycles, hr, 99% reliability	500 °F life cycles, hr, 50% reliability
75	353	2,427	22,067	7,282
100	42	221	1,545	662

At this time, surface finish effects are not part of the input data to the AGMA analysis. However, the effect of surface finish on contact fatigue has been documented in previous studies (Ref. 13). If surface finish effects were part of the input data for the AGMA contact stress calculations, the stress levels would have been calculated to be a higher value or have a lower allowable index stress decreasing the life of the gears. Using a pinion temperature higher than 500 °F would have also decreased the calculated life of these gears. Further studies, outside the scope of this paper, are required to capture these other design and operational conditions into the AGMA to more accurately predict gear life.

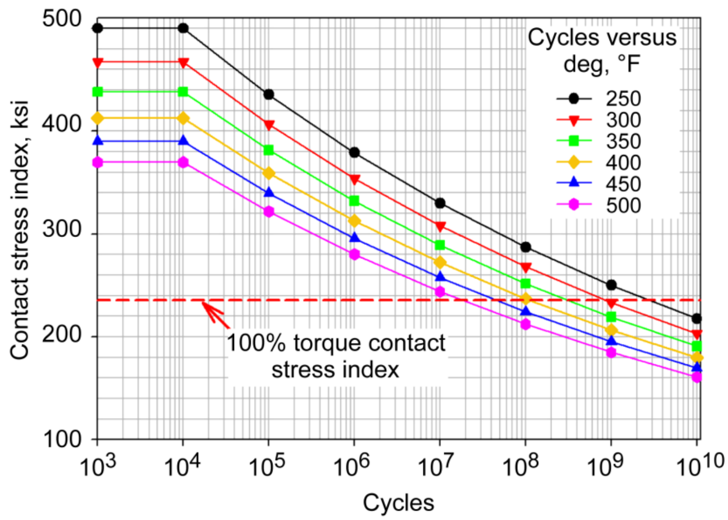

Figure 10.—Blank temperature effects on allowable contact stress, 99 percent reliability.

Summary

The benefit of combining usage operational parameters with HUMS condition indicators has the potential to improve the performance and reliability of both systems to indicate progression of damage and remaining time in operation. This analysis illustrates the importance of combining gear condition indicators and gear operational conditions, such as load and temperature, to indicate spiral bevel gear health. Decreasing both temperature and load can increase the fatigue life of the components. Per accepted AGMA practice, torque and temperature have significant influence on time for damage to appear. Once surface fatigue damage was initiated, the experimental data indicated decreasing load can slow the progression of damage to the gear teeth. Further studies are required to determine if usage monitoring of temperature and torque can be used to increase operational life for rotorcraft gears.

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14. ABSTRACT
The objective of this study was to illustrate the importance of combining Health Usage Monitoring Systems (HUMS) data with usage monitoring system data when detecting rotorcraft transmission health. Three gear sets were tested in the NASA Glenn Spiral Bevel Gear Fatigue Rig. Damage was initiated and progressed on the gear and pinion teeth. Damage progression was measured by debris generation and documented with inspection photos at varying torque values. A contact fatigue analysis was applied to the gear design indicating the effect temperature, load and reliability had on gear life. Results of this study illustrated the benefits of combining HUMS data and actual usage data to indicate progression of damage for spiral bevel gears.

15. SUBJECT TERMS
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