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EXTRACTION AND EVALUATION OF FAULT CONDITION INDICATORS FOR PLANETARY GEAR PLATE CRACK

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Abstract

In recent years, U.S. Blackhawk and Seahawk planetary gearbox carrier plates have experienced in-service cracking resulting in unscheduled removals and reduced operational availability. Vibration-based diagnostics is one method identified for mitigating failure risk of this critical drive train component and reducing costly scheduled inspections. This paper describes vibration data analysis and the derivation of fault condition indicators for state awareness monitoring of the helicopter gearbox systems. Such indicators are used to detect and identify incipient failures and predict the remaining useful life of failing components. We have developed and demonstrated a new class of features that indicate the progression of crack growth in a planetary gear plate. The features were evaluated based on the similarity (or linear correlation) between the evolution of the features and the true crack size.

INTRODUCTION

Epicyclic gears are important components for many rotorcraft transmission systems. An epicyclic gear system is defined by a sun/planet configuration, in which an inner "sun" gear is surrounded by two or more rotating "planets". Planetary systems are a subset of epicyclic gears defined by a stationary outer ring gear. Torque is transmitted through the sun gear to the planets, which ride on a planetary carrier. The planetary carrier, in turn, transmits torque to the main rotor shaft and blades.

A crack was recently found in the planetary carrier of the main transmission gear of the U.S. Army's UH-60A Blackhawk Helicopter [1] and the U.S. Navy's SEAHAWK helicopters. Figure 1 shows the carrier plate and the 10 in. and 3.25 in. crack of the main transmission gear. Since the planetary carrier is a flight critical component, a failure could cause an accident resulting in loss of life and/or aircraft. This resulted in flight restrictions on a significant number of the Army's UH-60A's. Manual inspection of all 1000 transmissions is not only costly in terms of labor, but also time prohibitive. There has been extensive work to develop a simple, cost-effective test capable of diagnosing this fault based on vibration data analysis techniques. A condition indicator (CI) or feature extracted from vibration data is typically used to describe gear health.

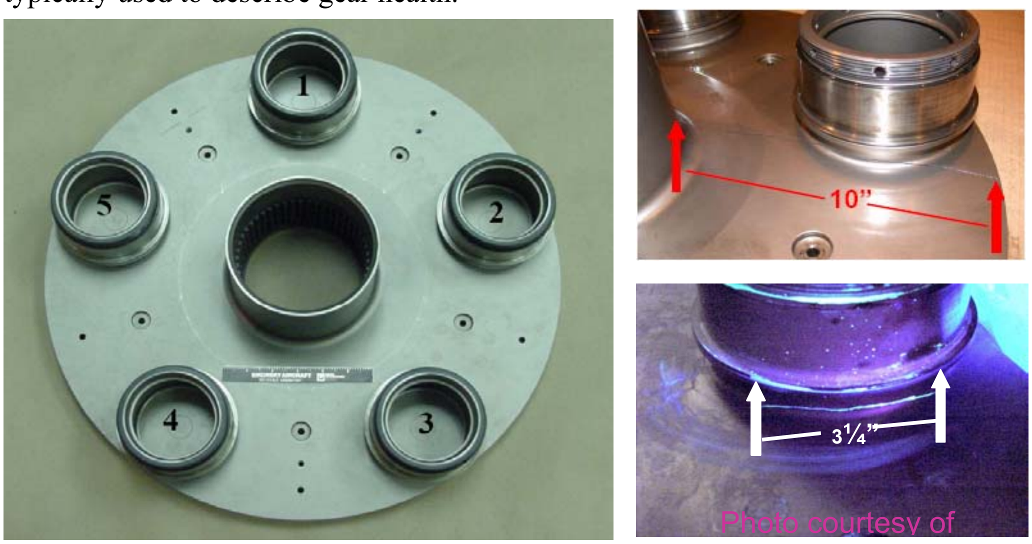

Figure 1 Planetary Carrier plate with a crack.

Compared with the measured vibrations produced by fixed axis gear systems, those produced by epicyclic gear systems are fundamentally more complicated. For fixed axis gear systems, the vibration signal consists mainly of the gear meshing frequency, which is $N_t \times f_{sh}$ with N_t being the gear tooth number and f_{sh} being the shaft frequency, and its harmonics. For an accelerometer mounted on the gearbox's housing, the vibration damage to individual gear teeth modulates the vibration signal, and, in the frequency domain, the damage appears in the form of symmetric sidebands around the gear meshing frequency. And the gear meshing frequency and its harmonics are still the dominant components. Many of the features for fixed axis gears detect damage based upon the signal amplitude at the gear meshing frequency/harmonics and/or the modulating sidebands [3][5].

For epicyclic gear systems, an accelerometer mounted on the housing will measure a periodic amplitude-modulated vibration signal as each planet rotates past the sensor. The vibration signal measured by the accelerometer is stronger when a planet is closer to the sensor, and is modulated by the transmission path to be lower when it is father away. Thus, even healthy gears will produce a vibration spectrum resulting in naturally occurring sidebands. Furthermore, there are multiple planets located at different positions relative to the sensor, and because the vibrations produced by the planets are generally not in phase, the dominant frequency component often does not occur at the gear meshing frequency and its harmonics. In fact, the gear meshing frequency component is often completely suppressed. This phenomenon was first recognized and explained by McFadden and Smith (1985) and later McNames (2002). In [1], the features developed for fixed axis gears were modified for the case of an epicyclic gearbox.

In this paper, we present the accelerometer data analysis techniques and the results for the seeded fault test conducted under the DARPA Structural Integrity Prognosis System (SIPS) Program. The project was mainly aimed to develop, demonstrate, and validate a prognosis system for a cracked planetary carrier plate. A seeded fault test was designed and conducted to provide a comprehensive database of accelerometer and crack measurements. The testing started with an initial 1.344 inch crack that propagated from a 0.25in starter EDM notch in a carrier plate post. The carrier plate was then stressed with a loading spectrum emulating a severe high engine torque mission and considered Ground Air Ground (GAG), 1P geometric vibratory, 980Hz gear vibratory, and transverse shaft bending. Vibration data was collected from accelerometers mounted on the external transmission casing, far removed from the seeded fault. These sensors represent existing sensors from Health and Usage Monitoring Systems (HUMS) that are beginning to be installed in the Army's and Navy's rotorcraft fleet. Snapshots of data were collected as the crack was growing.

VIBRATION DATA AND FAULT FEATURE (OR CONDITION INDICATOR) EXTRACTION

The vibration data from the testing consist of snapshots that are 5 seconds long and acquired at a rate of 100 kHz. A raw tachometer signal synchronized to the revolution of the planetary gear carrier plate and the main rotor is also included. The tachometer signal indicates the start of each data segment corresponding to each individual revolution of the carrier plate. The rotational frequency of the main rotator is slightly more than 4 Hz (it also varies a little over time), and therefore the snapshots of the test data contain 20 complete revolutions. Primarily, the analysis results on the VMEP2 sensor signal are presented here.

Data Pre-processing

The raw vibration data are pre-processed using a technique called Time Synchronous Averaging (TSA). The vibration data are segmented into numerous revolutions according to the tachometer signal before they are ensemble averaged resulting in an averaged data segment with a length corresponding to a single revolution. The TSA technique is intended to enhance the vibration frequencies that are multiples of the shaft frequency, which in many cases are mainly vibration related to the meshing of the gear teeth, while averaging out other components such as random vibrations and external disturbances.

Since the rotational speed of a transmission typically varies slightly during normal operation, the numbers of the data samples per revolution are different for a given sampling frequency. Interpolation is required to make the sample numbers per revolution the same before ensemble averaging can be carried out. Since interpolation is computationally demanding and time-consuming, it is undesirable especially for on-line real-time vibration monitoring. A simple technique is used that achieves the same result with much lower computational complexity since interpolation is not required [6]. The result obtained using this preprocessing technique is equivalent to the DFT of the time synchronous averaged data in the time domain using interpolation, or the TSA in the Frequency Domain.

Vibration Data

Figure 2 shows one revolution of the VMEP2 raw data collected at GAG #260 of the test (left plot) and the TSA data that resulted from ensemble averaging of the 20 revolutions of the raw data (right plot), respectively. Five peaks, caused by the five planets passing the sensor on the gearbox housing, are visible in the TSA data. The frequency spectrum of the TSA data is shown in the left plot of Figure 2, in which the scale along the x axis is the integer multiple of the shaft frequency. The figure shows that significant frequency components are around the meshing frequency ($228 \times f_{sh}$ with 228 being the tooth number of the ring gear and f_{sh} the rotational frequency of the planet carrier) and the harmonics up to the 7th.

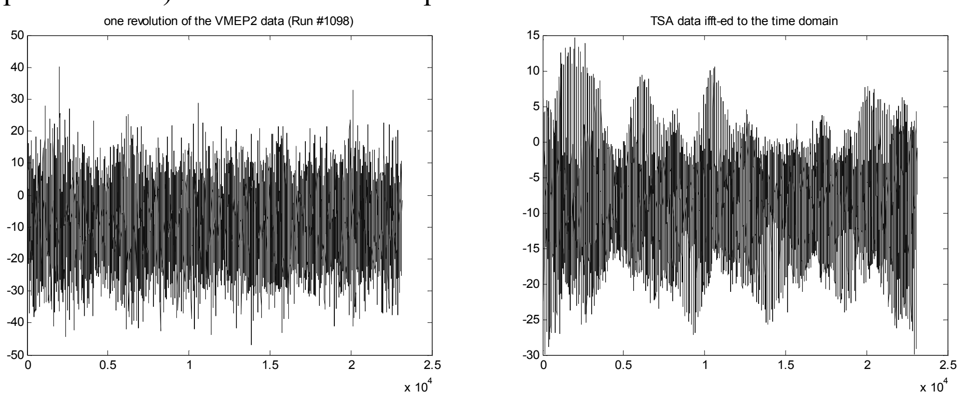

Figure 2 One revolution of the raw data (left), and the TSA of 20 revolutions (right).

The right plot of Figure 3 shows the spectrum content around the fundamental meshing frequency. We can observe that every 5 indices apart there is an apparent component, and the dominant frequency is not the meshing frequency (228), but occurs at 230. The principal component of the spectrum is slightly removed from the gear meshing frequency, that is, the sidebands are nontrivial and asymmetric about the meshing frequency.

Macfadden (1985) gave an explanation of this asymmetry suggesting that it is caused by the superposition of vibrations produced by the planets at different positions as they move relative to the transducer location. The motion of a single planetary gear past an accelerometer fixed on the housing produces symmetrical sidebands about the tooth meshing frequency due to the modulation effect of the vibration transmission path from the meshing teeth to the accelerometer fixed on the housing. Different planets in the same system have different phase angles relative to the first planet, and it is the relationship between the different phases of vibration produced by the planets which may cause asymmetry of the observed spectrum.

Keller (2003) gave a good summary of the explanation and concluded that only components with frequencies $(mN_t \pm n)$ that are multiples of the number of planets, N_p, appear for a healthy gear system and are measured by a sensor in the fixed frame, where m is the harmonic number, N_t is the tooth number (228 in this case), and n is an integer number. For the fundamental meshing frequency, $m=1$, so the apparent frequencies occur at 225, 230, 235, etc.

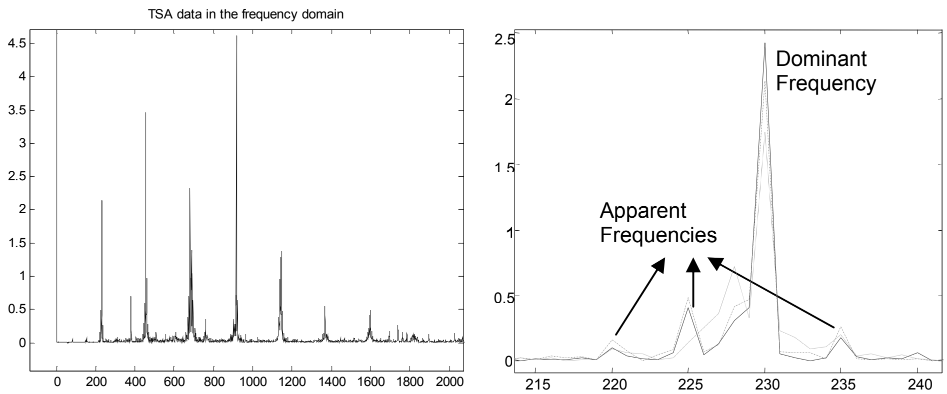

Figure 3 The spectrum of the TSA data (left), and around the fundamental meshing frequency (right).

In the right plot of Figure 3 three spectra for three sets of vibration data are plotted, with the black solid line for data at GAG #9, the dash dotted line for GAG #260, and the dashed line for GAG #639. The three GAG cycles represent the starting, middle, and later stages of the test. As the test progresses, the crack grows from its initial size of 1.4 in. The magnitude decreasing of the dominant frequency as well as the other apparent frequencies and the magnitude increasing of the rest may be a good sign of the crack growth, and is quantified as features for state awareness monitoring of the helicopter gearbox plate in the following section.

Feature Extraction

All the following features are calculated using the TSA data in the Frequency Domain (TSAinFD) obtained from the data pre-processing.

Feature #1: Averaged Harmonic Ratio For an epicyclic gearbox, the "regular" meshing components (RMC) of a specific gear meshing component (the fundamental component or the harmonics) are defined as the dominant frequency and the other apparent frequencies around the component. The Harmonic Ratio of a particular gear meshing component is the ratio between the non-RMC and the RMC. The Averaged Harmonic Ratio is the average of the feature for a certain number of meshing components. The left plot in Figure 4 shows the Averaged Harmonic Ratio averaged up to the 3th harmonic, as a function of the GAG cycles for the whole duration of the test. The feature values for different torque conditions are different. We plot the feature values of vibrations acquired when the gear system is under the 20% (red curve) and 40% (green curve) load conditions for all GAG cycles. The 100% load condition was changed to 93% at GAG cycle #320 to slow down the crack growth, and they are represented by the blue curve in the figure.

Feature #2 Non-RMC Ratio The Non-RMC Ratio is defined as the standard deviation of the difference signal normalized by the standard deviation of the regular signal,

$$NRMC\_R^e = \frac{RMS(d^e)}{RMS(r^e)}, \tag{1}$$

where the difference signal is the inverse Fourier transform of the TSAinFD data excluding the RMC up to the a certain harmonic such as the 6th harmonic for our implementation

$$d^e = F^{-1}(TSAinFD - \sum_{i=1}^{6} RMC_i) \tag{2}$$

and the regular signal is defined as the inverse Fourier transform of the RMC, as given by

$$r^e = F^{-1}(\sum_{i=1}^{6} RMC_i) \tag{3}$$

The right plot in Figure 4 shows the feature values of the Non-RMC Ratio for the whole test duration.

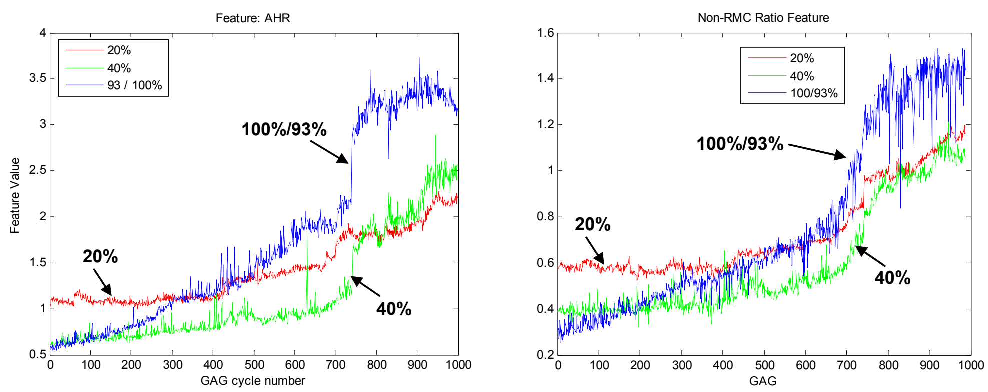

Figure 4 The Averaged Harmonic Ratio (left) and the Non_RMC Ratio (right).

FEATURE EVALUATION

Proposed feature evaluation metrics include the similarity (or linear correlation) between the feature and the true crack size. The left plot in Figure 5 shows the ground truth data (in the blue lines) of the crack size versus the GAG cycles when the information is available. The plate crack grows from 1.4 in. to approximately 7.5 in. The red curve is the Piecewise cubic Hermite interpolation (pchip) interpolated data of the ground truth data for every GAG cycle. A feature is desirable if it shows a similar growth pattern to that of the ground truth data. The middle plot shows the

Non_RMC Ratio feature for 100%/93% load condition. It shows a similar growth pattern to the crack size and therefore is desirable. The right plot is the feature value versus the crack size (or the mapping between the feature value and the crack size). It can be seen that the feature value increases as the crack grows, and ideally it is close to a linear relationship between the feature and the fault dimension.

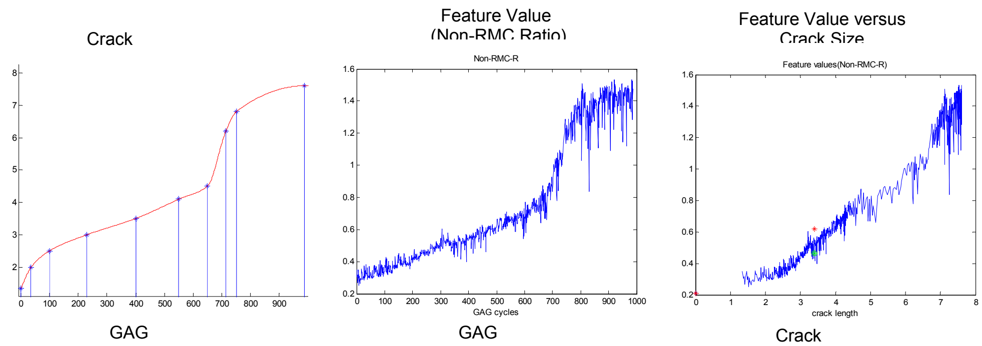

Figure 5 Feature Values versus crack size.

We use a correlation coefficient as a metric for feature evaluation. A correlation coefficient is the covariance between the two signals divided by their standard deviations. The most general form of the correlation coefficient is stated as

$$\rho = \frac{E\left(\left(x-\bar{x}\right)\left(y-\bar{y}\right)\right)}{\sigma_{xL}\sigma_{y}} \tag{4}$$

where $\bar{x}$ and $\bar{y}$ are the mean values of the two stochastic variables x and y defined by their probability density functions, and σ_x and σ_y are the standard deviations of x and y. For implementation purposes, the correlation coefficient is also written as

$$r = \sqrt{\frac{ss_{xy}^{2}}{ss_{xx}ss_{yy}}} \tag{5}$$

where $ss_{xx} = \sum(x_i - \bar{x})^2$, $ss_{yy} = \sum(y_i - \bar{y})^2$, and $ss_{xy} = \sum(x_i - \bar{x})(y_i - \bar{y})$.

The correlation coefficient is a number between 0 and 1 which measures the degree to which two variables are linearly related. If there is perfect linear relationship between the two variables, we have a correlation coefficient of 1. A correlation coefficient of zero means that there is no linear relationship between the two variables. It is the proportion of the observed data (such as a feature) which consistently reflects the change of a deterministic physical variable (such as a gear plate crack).

Table 1 shows the feature evaluation results for different loading conditions of the gear box in the test. We can see that the metric gives us fair evaluations of the features, and that the Non-RMC Ratio and the Averaged Harmonic Ratio reflect the growth of the crack very well, especially at the 100%/93% loading condition.

Table 1 the correlation coefficients of the two features for different loading conditions of the gear box.

Feature	20%	40%	100%/93%
AHR	0.9654	0.9332	0.9821
NonRMC Ratio	0.9579	0.9551	0.9838

SUMMARY

This paper presented vibration data analysis and the derivation of fault condition indicators for state awareness monitoring of a planetary gearbox system. The features or fault condition indicators introduced reflect the progression of crack growth in a planetary gear plate very well. The technology has potential applications in real-time condition monitoring of military and commercial aircrafts with planetary gearbox main transmission systems. It is useful for vibration-based diagnosis and prognosis to mitigate failure risk of this critical drive train component and reduce costly scheduled inspections.

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