Model-Based Fault Diagnosis of a Planetary Gear Using Transmission Error

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ABSTRACT

A Planetary gear can transmit high torque ratio stably and, therefore, the gear is widely used in industrial applications, i.e., wind turbines, automobiles, helicopters. Unexpected failure of the planetary gear results in substantial economic loss and human casualties. Extensive efforts have been made to develop the fault diagnostic techniques of gears; however, the techniques are mostly concerned about spur gears. This is mainly because understanding of complex dynamic behaviors of a planetary gear is lacking, such as multiple gear contacts, non-stationary axis of rotation, etc. This study thus proposes model-based fault diagnostics for a planetary gear that is based upon its dynamic analysis. Instead of vibration signals, this study uses transmission error (TE) signals for fault diagnostics of the planetary gear because TE signals (a) are directly related to the dynamic behaviors of gear mesh stiffness and (b) increase as damages on a gear mesh reduce the gear mesh stiffness. A lumped parameter model was used for modeling dynamic behaviors of the planetary gear. For more precise modeling, mesh phase difference–between sun, ring, and planet gear– and contact ratio were taken into account in the lumped parameter model. After acquiring transmission error signals from the model, order analysis and data processing were executed to generate health related data for the planetary gear. Consequently, it is concluded that the use of transmission error signals helps gain understanding of complex dynamic behaviors of the planetary gear and diagnose its potential faults.

1. INTRODUCTION

A planetary gear is a kind of gear system composed of a ring gear, sun gear, planet gear and carrier as shown in Figure 1. While the ring gear is covering the whole gearbox, multiple planet gears connected by a carrier are rotating around the sun gear. As planet gears are distributing the loads a gear system delivers, the planetary gear can transmit high torque ratio in a stable way. So it is commonly used in many huge engineering applications like wind turbines, automobiles, helicopters. As unexpected failure of the planetary gear can result in substantial economic loss and human casualties, fault diagnostics for various gear system including the planetary gear has been developed.

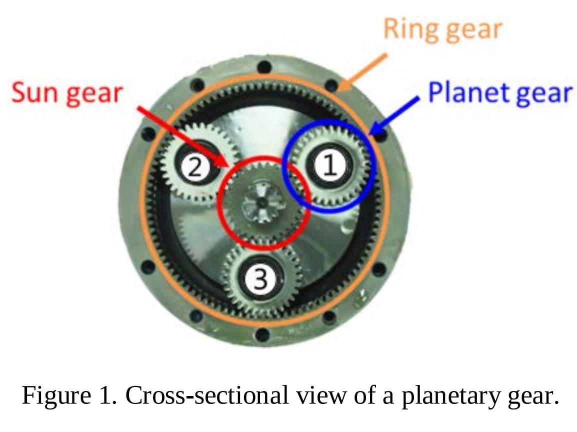


Figure 1. Cross-sectional view of a planetary gear.

Zheng, Li, and Chen (2002) developed fault diagnostics of a spur gear based on continuous wavelet transform. Samanta (2004) presented a comparative study for the performance of fault diagnostics for a spur gear between artificial neural networks (ANNs) and support vector machines (SVMs) which classify the normal and fault condition. Saravanan, Cholairajan, and Ramachandran (2009) used fuzzy classifier with vibration signal to detect the fault of a spur bevel gear box. Fault diagnostics for a planetary gear is relatively less developed (Lei, Kong, Lin, and Zuo, 2012). Barszcz and Randall (2009) applied spectral kurtosis technique to detect a tooth crack of the planetary gear. Lei et al. (2012) proposed two new diagnostics parameters for the planetary gear, root mean square of the filtered signal (FRMS) and normalized summation of positive amplitudes of the difference spectrum between the unknown signal and the healthy signal (NSDS). Feng and Liang (2014) exploited the adaptive optimal kernel (AOK) method to deal with the non-stationary signal of the planetary gear. Above literatures used vibration signals to detect the faults of the gear system. In recent years, Acoustic emission signals has been used to detect the faults of a gear due to the sensitivity to early faults than vibration signal. Qu, He, Yoon, Van Hecke, Bechhoefer, and Zhu (2014) performed comparative study between vibration signal and acoustic emission signal. They found that acoustic emission signal is more sensitive to small tooth damage in the low speed range.

However, previous signals used for fault diagnostics of gears have defects because they didn't utilize the physical meaning of gear dynamics. Gear system is a very well organized system, especially a planetary gear has its own peculiarity due to the gear dynamics arising from pitch, contact ratio, phase difference. Therefore, we introduced new fault diagnostics signal, Transmission Error (TE) in a lumped parameter model. TE is defined as "the angular difference between the position that the output shaft of a gear drive would have if the gearbox were perfect (without errors or deflections) and the actual position of the output shaft" according to Remond and Mahfoudh (2005). This signal is deeply related with gear mesh stiffness. So, it has physical meaning in gear dynamics and could have potentials which could classify the fault condition in gear system. In this paper, we compared the TE signal from simulation model in both normal and faulty planetary gear and demonstrated the validity of TE for fault diagnostics of a planetary gear.

This paper is organized as follows. The development of the planetary gear lumped model is described in Section 2. In section 3, Description about how TE could have physical meaning and relation with fault is followed. Section 4 presents the way we processed the signal to effectively observe the fault symptom and results are shown. In section 5, health indices used for fault diagnostics of a planetary gear are introduced and they are calculated from TE signal for normal and faulty gear obtained from simulation model. Finally, section 6 states the conclusion and future work of this research.

2. PLANETARY GEAR MODELING

A Planetary gear used in this paper is constructed using DAFUL 4.2. Basic lumped parameter modeling strategies for planetary gears in DAFUL 4.2 are based on a thesis from Kim (2001).

2.1. Basic Specification of a Planetary Gear

Basic gear specification used in this paper is as shown in Table 1. These parameters are used as input parameters for lumped parameter model. For example, numbers of teeth for each gear are used for calculating the gear ratio (4.06:1), and pressure angle information is used for indicating the direction of interacting force, and so on. The system input is a low speed shaft connected with a carrier and the system output is a high speed shaft connected with a sun gear.

Table 1. Planetary gear specification.

Gear data	Sun	Ring	Planet
Number of teeth	31	95	31
Pressure angle (deg)	20	20	20
Module (mm)	1.5	1.5	1.5
Pitch circle diameter (mm)	46.5	46.5	142.5
Dedendum circle diameter (mm)	43.643	146.25	43.409
Tip diameter (mm)	50.693	139.5	50.459
Whole depth (mm)	3.525	3.375	3.525
Face width (mm)	16	16	16

2.2. Gear Mesh Stiffness

Another important parameter used in DAFUL is gear mesh stiffness. Gear mesh stiffness is defined as the ratio between the input torsional load and the total angular rotation of the gear (Sirichai, Howard, Morgan, and The, 1997). As mesh stiffness is closely related to the TE, which we would use as a fault signal, it is carefully parameterized in DAFUL. In DAFUL, gear mesh stiffness can be parameterized based on (a) one mesh, (b) all mesh, or (c) constant value.

2.2.1. Magnitude of Gear Mesh Stiffness

The magnitude of gear mesh stiffness has repeating patterns due to the repeating contact condition (single, double contact) in path of contact of gear mesh. This gear mesh stiffness can be obtained analytically (Cornell, 1985). However, in this paper, it is calculated by finite element analysis code, ABAQUS, and the result is as Figure 2. The Figure 2 is for ring-planet gear mesh stiffness calculated from ABAQUS code. Then, this values were parameterized as two values, 448900 and 536700N/mm for simplicity. Sun-planet gear mesh stiffness is achieved in the similar way and they were also parameterized as two values, 210600 and 274000N/mm.

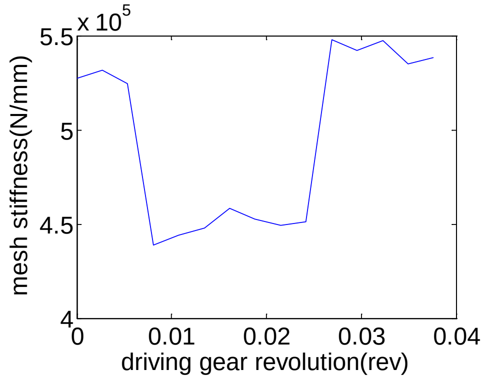


Figure 2. Ring-planet gear mesh stiffness result from finite element analysis.

We can also notice that ring-planet gear mesh stiffness is bigger than sun-planet gear mesh stiffness about two times. This is because ring-planet gear is an internal gear which shows high contact ratio.

2.2.2. Phase of Gear Mesh Stiffness

Parker and Lin (2003) calculated phase difference of gear mesh stiffness not only among planets with a ring gear and sun gear but also between ring-planet gear mesh stiffness and sun-planet gear mesh stiffness in a planetary gear.

For the case of gear mesh stiffness among planets with a ring gear and sun gear, the phase difference can be calculated by the following equation when the planet rotation is counter-clockwise.

$$\gamma_{sn} = -\frac{Z_s \psi_n}{2\pi} \quad \gamma_{rn} = \frac{Z_r \psi_n}{2\pi} \tag{1}$$

where γ_{sn} is relative phase difference between nth sun-planet gear mesh stiffness and the reference sun-planet gear mesh stiffness, γ_{rn} is relative phase difference between nth ring-planet gear mesh stiffness and the reference ring-planet gear mesh stiffness, $Z_{r,s}$ is ring and sun gear tooth numbers and ψ_n is circumferential angle measured at reference planet gear. In this equation reference planet gear can be selected arbitrarily as 1st planet gear in Figure 1. For our case, Z_r, Z_s are 95, 31 and $\psi_{1,2,3}$ are 0, 2π/3, 4π/3 respectively. So $\gamma_{r1}, \gamma_{r2}, \gamma_{r3}$ are 0, 2/3, 1/3 and $\gamma_{s1}, \gamma_{s2}, \gamma_{s3}$ are 0, -1/3, -2/3 respectively, which means same phase difference to the same planet with a ring gear and sun gear as phase difference of 2/3, 1/3 is identical to phase difference of -1/3, -2/3.

For the case of gear mesh stiffness between ring-planet gear mesh stiffness and sun-planet gear mesh stiffness in a planetary gear, the phase difference ($=\gamma_{rs}$) can be calculated analytically based on pitch contact point which is the midpoint of the lower stiffness region. It is indicated as a red

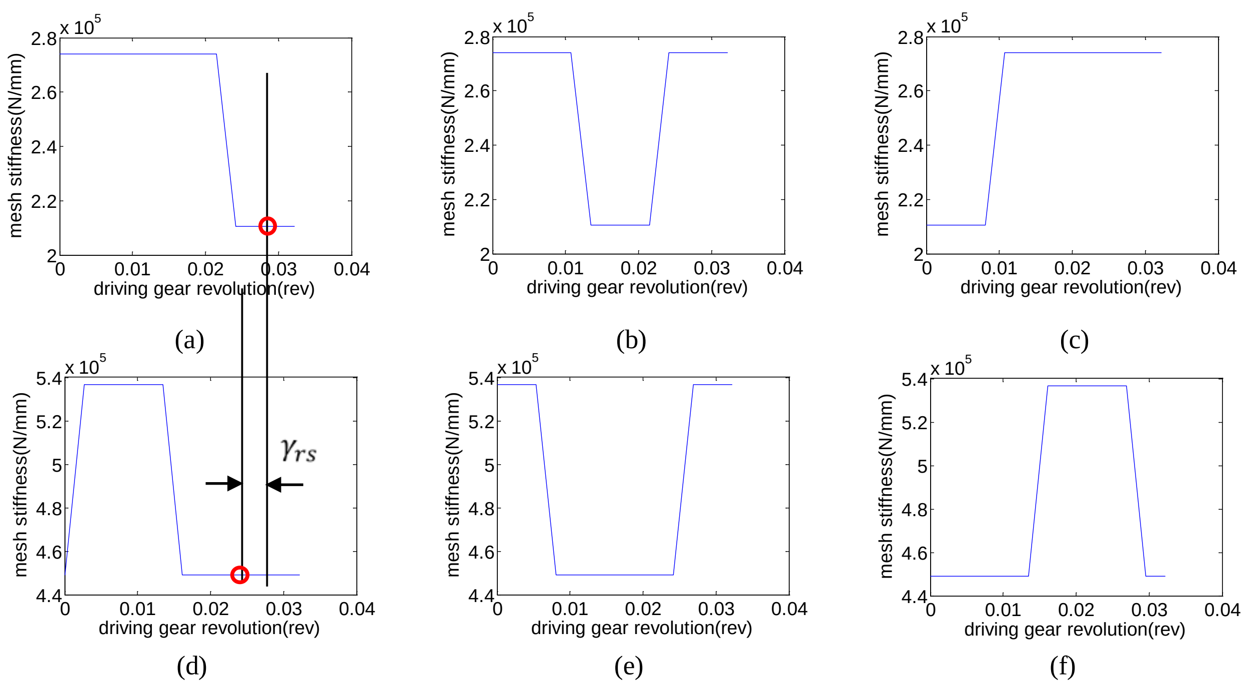


Figure 3. Gear mesh stiffness of (a) 1st sun-planet gear, (b) 2nd sun-planet gear, (c) 3rd sun-planet gear, (d) 1st ring-planet gear, (e) 2nd ring-planet gear, (f) 3rd ring-planet gear.

circle in Figure 3 and it is applied to in the phase of a planetary gear.

It is proved that no phase difference in sun-planet gear mesh stiffness could make equal load distribution at planets, and differing phase difference could have significant effect in reducing vibration and noise. (Parker & Lin, 2003) In our case, as phase difference is equally distributed at each planet, we could guess our planetary gear is designed to reduce the noise and vibration rather than to distribute the loads the system carries.

2.3. Mesh Stiffness of a Faulty Gear

In this paper, we define gear fault as a crack in a planet gear tooth. Chaari and Haddar (2009) studied the relationship between crack size and mesh stiffness reduction. In above literature, gear mesh stiffness for a spur gear gets smaller and smaller as a crack in a gear tooth gets larger. And this literature showed that 1/4 of tooth thickness- cracked gear induces 10% mesh stiffness reduction to the one of whole gear mesh stiffness. In this research, therefore, as each ring-planet gear and sun-planet gear interaction can be thought as a spur gear interaction, mesh stiffness reduction would happen to the both ring-planet and sun-planet gear mesh stiffness by 10% in the same way if we assume a crack in a gear tooth is 1/4 of tooth thickness. However, in the planetary gear, we should also consider the fault phase difference between ring-planet gear mesh stiffness reduction and sun-planet gear mesh stiffness reduction. The planet gear makes one rotation around a sun gear while it is meshing with a ring gear and sun gear repeatedly. So for the crack in a gear tooth, it contact with a sun gear, ring gear, sun gear at 0, 1/2, 1 rotation of planet gear like Figure 4. So, the mesh stiffness phase difference in fault condition is 1/2 rotation of a planet gear like Figure 5. So this stiffness values were parameterized for the faulty gear mesh stiffness of the planetary gear.

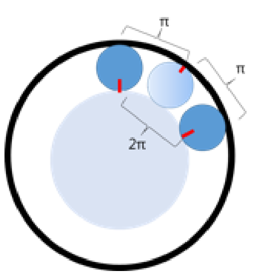


Figure 4. A cracked planet gear rotation behavior.

3. SIGNALS FOR FAULT DETECTION

This section will discuss TE, the signal used for fault detection in this research. First, we explain about why TE is related with health condition and how TE varies when the fault is seeded into gear sets. And then, TE behavior of a planetary gear in normal condition will be discussed.

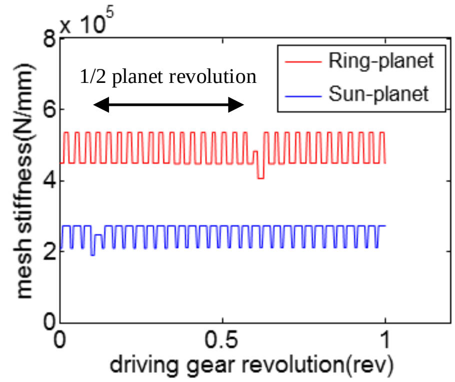


Figure 5. Ring-planet and sun-planet gear mesh stiffness for a cracked planetary gear.

3.1. Transmission Error

TE can be simply defined as “the output gear difference between the expectation and reality”. TE occurs due to many sources like tooth profile error, tip relief error, mesh stiffness, etc. In our case, we only consider the effect from mesh stiffness. Let’s say the gear is rotating clockwise and inverse torque is applied to output gear counterclockwise. Then the gear teeth will deflect counterclockwise due to inverse torque. This is the reason TE happens in a gear. That is, for the single contact condition, gear mesh stiffness is low, and TE would show higher value. Then, for the double contact condition, gear mesh stiffness is high, and TE would show lower value. In this way, TE fluctuates repeatedly along the stiffness fluctuation. Then, what would happen if a gear tooth is cracked? As we discussed in section 2, crack in a gear tooth makes gear mesh stiffness reduction. So, TE would increase as the stiffness is reduced. In this way, TE signal can be a physically meaningful signal differently from other signals in relation with mesh stiffness. Also, as stiffness reduces gradually along the crack size propagation, TE signal can be a more useful signal for fault prognostics.

3.2. Transmission Error in a Planetary Gear

Transmission error in a planetary gear can be calculated as

$$TE = h.s.s\,rotation - gear\,ratio \times l.s.s\,\,rotation \qquad (2)$$

where *h.s.s* denotes high speed shaft connected with a sun gear and *l.s.s* denotes low speed shaft connected with a carrier. Differently from a spur gear, TE in a planetary gear shows complicated behavior due to the effect from multiple meshing condition from ring, planet, and sun gear as in Figure 1. Figure 6 shows a TE signal result from DAFUL when input velocity is 20rad/s, inverse torque is 2×106 Nmm with sampling frequency 1000hz. 3 peaks in one fluctuation are repeatedly appearing. We could guess this could happen due to the effect from three planets.

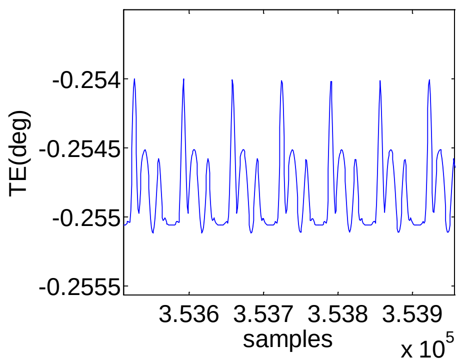


Figure 6. Simulated transmission error signal from a planetary gear.

4. SIGNAL PROCESSING

To effectively observe the fault characteristics of a planetary gear, TE signals were processed with three steps like Figure 7, (1) DC component subtraction, (2) Time synchronous averaging (TSA), (3) Order analysis. In this section, we discuss the principles of each procedure for signal processing and explain why each procedure was performed.

4.1. DC Component Subtraction

The first step for signal processing is to subtract DC component in raw TE signal. TE fluctuates while the DC component is shifted due to the deflection like Figure 6. To effectively analyze the TE in a frequency domain, mean value of the TE should be subtracted from original signal. Comparing with Figure 7 (a) and 7 (b), you can see TE value is shifted along y-axis.

4.2. Time Synchronous Averaging (TSA)

Time synchronous averaging (TSA) for gear signal analysis was originally proposed to suppress the noisy signal - (a) non-synchronous coherent signal, (b) non-coherent random signal (Hochmann & Sadok, 2004.). However, in this research, TSA was adopted to effectively observe the gear mesh frequency of interest in TE signal. Eq. (3) is the equation used for TSA in this paper.

$$\bar{x} = \frac{1}{N}\sum_{k=1}^{N} x_k \tag{3}$$

where $\bar{x}$ is time synchronous averaged data, N is number of planet rotation and x_k is TE data in time domain for kth planet rotation.

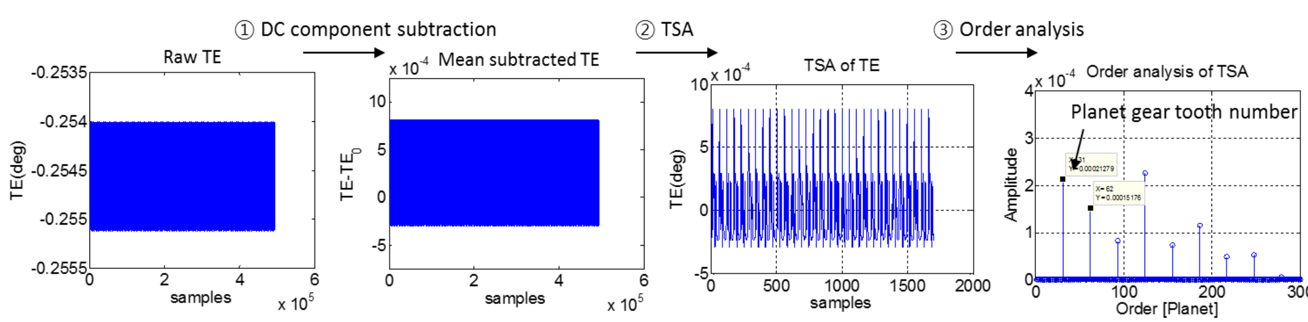
By calculating equation (3), we can observe the only planet-oriented behavior of TE signal. In Figure 7 (c), there are 31 fluctuations which contain 3 peaks in a fluctuation as in Figure 6. 31 is the number of a planet gear and we can observe how the TE is varying for the 1 rotation of a planet gear by performing TSA.

4.3. Order Analysis

Then the order analysis was performed to analyze the effect from the planet gear mesh frequency. This can be performed by transforming time-domain TE data into frequency domain by Fast fourier transform (FFT) code in MATLAB. As TSA data were averaged with planet rotation, we can observe the planet gear tooth number component and its harmonic in order analysis result in Figure 7 (d).

4.4. Results from Normal and Cracked Gear

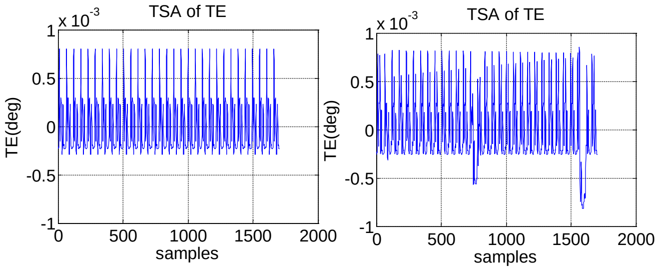
After following these procedures, simulated TE results from normal and cracked planetary gear were obtained like Figure 8, 9. Figure 8 shows the TSA of TE from normal and cracked planetary gear. In advance, we can see the two sparks in Figure 8 (b). In Figure 4, we showed that a crack in a planet gear contacts with a ring, sun gear repeatedly while planet gear makes one rotation. So, this behavior makes TE in a planetary gear spark from normal TE. Also there is magnitude difference in TE sparks. As there are difference in stiffness between ring-planet and sun-planet gear mesh stiffness, TE sparks, which arose from stiffness, also has difference in magnitude.



(a) (b) (c) (d)

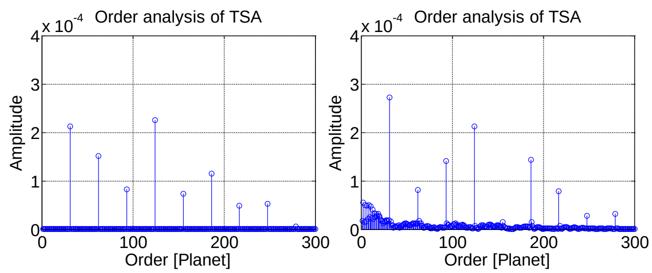
Figure 7. Procedures for TE signal processing.

Figure 9 shows the order analysis results. In Figure 9 (b), we can observe the sub-harmonic and sideband near the main harmonic.



(a) (b)

Figure 8. TSA results of TE in a (a) normal and (b) cracked planetary gear.



(a) (b)

Figure 9. Order analysis results of TE in a (a) normal and (b) cracked planetary gear.

5. HEALTH INDEX CALCULATION

TE results from section 4.4 need to be quantified to properly represent health state of the system. Lebold, McClintic, Campbell, Byington, and Maynard (2000) organized health index frequently used for gearbox diagnostics. In section 5, we adopted two health index and compared the results from normal and cracked gear.

5.1. Health Index

In this study, we adopted root mean square (RMS) and FRMS to quantitatively classify a cracked gear from a normal gear.

First, RMS can be formulated as

$$RMS = \sqrt{\frac{1}{N}\sum_{k=1}^{N} x_k^2} \tag{4}$$

where x_k is kth time data point and N is number of total data. By calculating RMS, overall noise level can be easily detected.

Secondly, FRMS can be formulated as

$$FRMS = \sqrt{\frac{1}{T}\sum_{t=1}^{T} (s(t))^2} \tag{5}$$

where *s(t)* is the ith data of data point of the filtered signal S and T is the number of total data. Filtered signal is obtained by filtering out the shaft frequency and its five-order harmonics and gear mesh frequency and its three-order harmonics in frequency domain. Then the signal is transformed into time domain again. This signal is effective in planetary gear analysis because shaft frequency and its harmonics, gear mesh frequency and its harmonics mainly dominates the vibration signal of planetary gear (Yaguo, et al., 2012).

5.2. Health Index from Various Condition

To verify the validity of the TE as a fault diagnostics signal, health indices proposed from section 5.1 are calculated using TE in various conditions.

First, RMS, FRMS were calculated from various input speed at 1~20 rad/s like Figure 10. Then, RMS, FRMS were calculated from various inverse torque at 1~10×10^5 Nmm magnitude like Figure 11. We can observe that at faster input speed and higher inverse torque magnitude we can more easily differentiate the cracked gear from a normal gear.

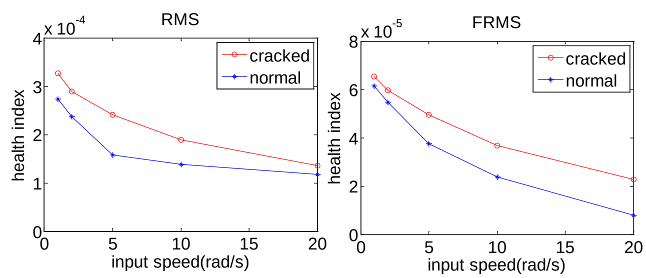


Figure 10. RMS, FRMS values from a normal, cracked planetary gear at various input speed.

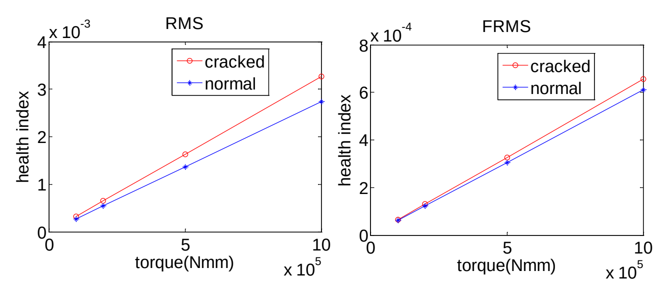


Figure 11. RMS, FRMS values from a normal, cracked planetary gear at various inverse torque.

Also, we observed the RMS and FRMS change along the relative stiffness like Figure 12. Relative stiffness means the ratio of the stiffness to the stiffness from normal planetary gear. As bigger crack size indicates larger gear mesh stiffness reduction, we tried to estimate health index from different crack size from relative stiffness. From Figure 12, we can notice that as crack size is getting bigger, health indices indicates larger values.

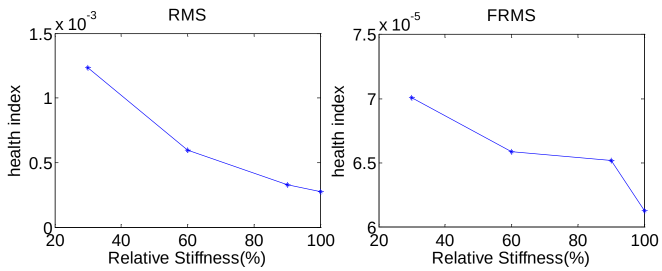


Figure 12. RMS, FRMS values from different crack size.

6. CONCLUSION

This paper proposed a new signal, TE for model-based fault diagnostics of the planetary gear. First, we developed a planetary gear with lumped parameter model. In this step, we closely studied phase difference in ring-planet gear mesh stiffness and sun-planet gear mesh stiffness considering pitch contact point. To simulate the fault condition in a gear as a crack in a gear tooth, we studied the relationship between crack size and gear mesh stiffness, which is directly related with TE signal. We also considered the fault phase occurring from planet gear rotation. Then we analyzed the TE signal in an organized signal processing procedures and calculated health indices. By calculating health indices from various condition, we could conclude that TE can be a good signal for diagnosing the fault in a planetary gear. Moreover, as TE is a physically meaningful signal related with stiffness, it can not only differentiate fault level but also be a signal for fault prognosis.

Future work will include development of lumped parameter model and validation using test-bed data. As we considered many things in modeling the planetary gear, it can be developed more precisely to simulate a real planetary gear. Then, finally, validation using a real planetary gear TE data should be performed. To accurately measure the TE signal, many methods have been developed using encoder. So, by obtaining and analyzing the TE data, proposed idea could be validated.

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