Mechanical Systems and Signal Processing 100 (2018) 706–728

Contents lists available at ScienceDirect

Mechanical Systems and Signal Processing

journal homepage: www.elsevier.com/locate/ymssp

Torsional vibration signal analysis as a diagnostic tool for planetary gear fault detection

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ARTICLE INFO

Article history:
Received 17 February 2017
Received in revised form 17 June 2017
Accepted 19 July 2017

Keywords:
Torsional vibration
Planetary gear fault diagnosis
Lumped-parameter model
Finite element model
Signal processing techniques

ABSTRACT

This paper aims to investigate the effectiveness of using the torsional vibration signal as a diagnostic tool for planetary gearbox faults detection. The traditional approach for condition monitoring of the planetary gear uses a stationary transducer mounted on the ring gear casing to measure all the vibration data when the planet gears pass by with the rotation of the carrier arm. However, the time variant vibration transfer paths between the stationary transducer and the rotating planet gear modulate the resultant vibration spectra and make it complex. Torsional vibration signals are theoretically free from this modulation effect and therefore, it is expected to be much easier and more effective to diagnose planetary gear faults using the fault diagnostic information extracted from the torsional vibration.

In this paper, a 20 degree of freedom planetary gear lumped-parameter model was developed to obtain the gear dynamic response. In the model, the gear mesh stiffness variations are the main internal vibration generation mechanism and the finite element models were developed for calculation of the sun-planet and ring-planet gear mesh stiffnesses. Gear faults on different components were created in the finite element models to calculate the resultant gear mesh stiffnesses, which were incorporated into the planetary gear model later on to obtain the faulted vibration signal.

Some advanced signal processing techniques were utilized to analyses the fault diagnostic results from the torsional vibration. It was found that the planetary gear torsional vibration not only successfully detected the gear fault, but also had the potential to indicate the location of the gear fault. As a result, the planetary gear torsional vibration can be considered an effective alternative approach for planetary gear condition monitoring.

1. Introduction

Planetary gears, also known as epicyclic gear sets, commonly include several planet gears meshing simultaneously to split the torque and power. The planet gear can not only rotate around its own axis, but also around the planetary gear common axis. However, this rotating mechanism poses a big challenge for planetary gear condition monitoring compared to that of the parallel shaft gear. Additionally, the vibration level of the planetary gear system is also typically lower than the parallel shaft gear because of the self-centring capability of central members (sun gear, ring gear and the carrier) [1].

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http://dx.doi.org/10.1016/j.ymssp.2017.07.038

The traditional approach for condition monitoring of the planetary gear involves using a stationary transducer mounted on the ring gear casing to measure the vibration data when the planet gear passes by [2]. However, in this way, the measured vibration data exhibits a modulation effect caused by the carrier arm rotation. In the 1990s, a technique was developed for the time domain averages of the tooth meshing vibration of the individual planet gears and of the sun gear in an epicyclic gearbox [3–5]. Various window functions [5] could be used to capture the vibration data from the individual planet gear tooth as it was observed that the vibration level reached its peak when the corresponding planet gear was closest to the transducer [3]. Based on this, further signal processing techniques which have been used in the fixed axis gear could be used here to detect the gear fault. Moreover, numerous publications have also been focused on explaining the vibration spectrum caused by this carrier arm passing modulation [6–9] and it indicated that the sidebands would appear in the position of $f_m \pm nf_c$ ($n = 1, 2, 3, \ldots$), with sideband spacing equal to the carrier arm rotation frequency f_c. McFadden and Smith pointed out that the asymmetry vibration spectra was caused by the fact that each planet gear had varying phase angles [6]. McNames applied Fourier series analysis to explain the source of the asymmetry observed in the spectrum and identified the location of the dominant frequency components near all harmonics of the meshing frequency [7]. Furthermore, Kahraman classified all the possible modulation sidebands into five different conditions based on the assembly condition and parameters of the planetary gear system [8]. Mark predicted additional sidebands in the frequency spectra produced by planet-carrier torque modulations, which might potentially mask the sidebands caused by damage in planetary gearboxes [2,10]. Recently, Feng and Zuo summarised the planetary gear fault characteristics [11]. For the planetary gear system with a localized sun gear fault, the sidebands would appear in the position of $f_m \pm kf_s \pm nf^c_{sf} = f_m \pm kf_c \pm (k/N \pm n)f^c_{sf}$, with sideband spacing equals to the faulty sun gear rotational frequency f^c_{sf}. For the planetary gear system with a faulty planet gear, the sidebands would appear in the position of $f_m \pm kf_c \pm nf^c_{pf}$, with sideband spacing equals to the faulty planet gear rotation frequency f_{pf}. For the planetary system with a faulty ring gear, the sidebands would appear in the position of $f_m \pm nf^c_{rf}$, with sideband spacing equals to the faulty ring gear rotation frequency f^c_{rf}.

On the other hand, torsional vibration, measured from the gear shaft, also carries diagnostic information [12]. It would be an advantage to measure the torsional vibration directly to naturally separate the signal from the modulation effect. As a result, measuring the torsional vibration could be treated as an alternative way for condition monitoring of the planetary gear system. Limited publications on the planetary gear torsional vibration studies could be found. It was demonstrated by Wang in his thesis that the torsional vibration signal is far superior than the transverse signals for frequency analysis for all rotating components of planetary gearboxes [13]. Feng and Zuo gave explicit equations for the planetary gear torsional signal model indicating that the only modulation effect in the torsional vibration signal was the amplitude and phase modulation caused by gear faults [14]. It was found that for the planetary gear system with a localized sun gear fault, the sidebands of the torsional vibration signal would appear in the position of $f_m \pm nf^c_{sf}$, with sideband spacing equals to the faulty sun gear rotational frequency f^c_{sf}. For the planetary gear system with a faulty planet gear, the sidebands of the torsional vibration signal would appear in the position of $f_m \pm nf^c_{pf}$, with sideband spacing equals to the faulty planet gear rotation frequency f^c_{pf}. For the planetary system with a faulty ring gear, the sidebands would appear in the position of $f_m \pm nf^c_{rf}$, with sideband spacing equals to the faulty ring gear rotation frequency f^c_{rf}. However, the big challenge of using this method is that there are so many faulty characteristic frequencies in the frequency spectrum and it becomes hard to determine which frequency component is the best choice. As a result, continuous research should be conducted on the analysis of the planetary gear torsional vibration.

Dynamic modelling of planetary gear vibration can be used to further the understanding of the vibration generation mechanisms in gear transmissions as well as the dynamic behaviour of the transmission in the presence of several types of gear faults. Kahraman presented a single-stage planetary gear train including the rigid body motions of the gears and the carrier arm to study the load sharing characteristics, the manufacturing error and the wear effect on the planetary gear dynamic response [15]. Lin and Parker further developed this model to including the gyroscopic effect to investigate a series of factors influencing the planetary gear natural frequency [16]. An extended three-dimensional model was developed by Velex to calculate the dynamic tooth loads on a planetary gear [17]. A planetary gear lumped parameter model considering the eccentricity error and planet position error was developed by Gu and the instantaneous gear geometry was used in the model [18,19]. Chen incorporated the mesh stiffness of the internal gear pair with a crack into a 21DOF planetary gear model to investigate the dynamic response [20]. Based on a lump-parameter model, Liang investigated the vibration signal features of each component in the planetary gear system in the perfect and cracked situations [21].

The sun-planet and ring-planet tooth mesh stiffness variations and the resulting transmission errors were found to be the main internal vibration generation mechanisms for planetary gear systems. There were mainly analytical methods [20] and finite element (FE) methods to calculate the gear mesh stiffness [22–24]. The use of FE modelling was found to be most suitable for capturing the extended tooth contact phenomenon, especially when there was a crack at the gear tooth root, which could aggravate this effect [23]. Another issue was that the deviation between the gear mesh stiffness obtained from the analytical method and the FEA method was found to become larger with the increase in the crack size. The FE model was found to be more suitable for modelling the gear tooth mesh stiffness with larger gear tooth crack size [24].

Some advanced signal processing techniques could be used further to help in extracting the gear fault feature. The residual signal can be obtained by removing the regular gearmesh component and its harmonics while keeping the local variations. Initially, the residual signal was obtained from the frequency domain by removing the known gear mesh harmonics and then inverse Fourier transforming back to the time domain [25]. Narrowband envelope analysis was one of the prominent vibration signal processing techniques initially developed for the rolling bearing failure detection [26]. The idea behind