Engineering Failure Analysis 48 (2015) 185–200

Contents lists available at ScienceDirect

Engineering Failure Analysis

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

Vibration signal modeling of a planetary gear set for tooth crack detection

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ARTICLE INFO

Article history:
Received 3 May 2014
Received in revised form 20 October 2014
Accepted 17 November 2014
Available online 26 November 2014

Keywords:
Signal modeling
Dynamic simulation
Transmission path
Tooth crack
Sidebands

ABSTRACT

In a planetary gearbox, there are multiple vibration sources, and the transmission path of vibration signals changes due to the rotation of the carrier. This study aims to model the vibration signals of a planetary gearbox and investigate the vibration properties in the healthy condition and in the cracked tooth condition. A dynamic model is developed to simulate the vibration source signals. A modified Hamming function is proposed to represent the effect of the transmission path. By incorporating the effect of multiple vibration sources and the effect of transmission path, resultant vibration signals of a planetary gearbox are obtained. Through analyzing the resultant vibration signals, some vibration properties of a planetary gearbox are revealed and the fault symptoms of sun gear tooth crack are identified and located. Finally, the proposed approach is experimentally verified.

1. Introduction

The vibration signals of a planetary gearbox are more complicated comparing with that of a fixed-shaft gearbox. For a planetary gear set, several sun-planet gear pairs and several ring-planet gear pairs mesh simultaneously. The vibration signals generated by each sun-planet gear pair are similar but with different phases [1]. Similar comments apply to the ring-planet meshes. Due to the phase differences, some of the excitations are canceled or neutralized [2] while others are augmented. In general, vibration transducers, mounted on the housing of a gearbox or the housing of a bearing, are used to acquire vibration signals. Transmission paths of the vibration signals to a transducer change due to the revolution of the carrier. Multiple vibration sources and the effect of the transmission path will cause fault symptoms hard to be distinguished.

Even though many signal processing methods have been proposed to detect gear faults [1–4], the improvements of these methods are still desired. The transducer signal is comprised of many sub signals, such as the vibrations of the sun gear, planet gear, ring gear, bearings and shafts. Unfortunately, there are no mature signal processing methods which can effectively denoise and separate these signals [5,6]. If we can "open" the black box, "see" all the sub-signals, and understand the generation mechanisms of vibration signals, effective tools can be developed to detect gear faults.

Mathematical models have been used by several researchers to investigate the vibration properties of a planetary gearbox [7,8]. However, the mathematical models lack the connection with the physical parameters of a gearbox, like the mesh stiffness and damping. In addition, they can hardly model the process of the fault growth. Dynamic simulation is a better choice to investigate the vibration properties of a planetary gearbox. A dynamic model is more closely connected with the physical

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http://dx.doi.org/10.1016/j.engfailanal.2014.11.015
1350-6307/

parameters of a planetary gearbox than the mathematical model. It can model the process of fault growth and the corresponding effects. Compared with experimental or field systems, dynamic simulation models have the following advantages [9]: (1) environmental noises can be eliminated so that the changes in vibration signals caused by the faults can be identified easily; and (2) with a good dynamic simulation model, we can easily simulate different types and different levels of faults, and observe changes in the vibration signal due to the faults.

A large range of dynamic models have been developed to simulate the behavior of planetary gears. Kahraman [10] proposed a nonlinear dynamic model to investigate the load sharing characteristics of a planetary gear set. Inalpolat and Kahraman [11] used the same model as Ref. [10] to predict modulation sidebands of a planetary gear set having manufacturing errors. Lin and Parker [12] modified the model of Ref. [10] and investigated the free vibration properties of a planetary gear set. Cheng et al. [13] developed a pure torsional dynamic model to investigate the properties of a planetary gear set when a single pit was present on one tooth of the sun gear. Chaari et al. [14] used a similar model as Ref. [12] to investigate the effect of manufacturing errors on the dynamic behavior of planetary gears. The effects of tooth profile modification or manufacturing errors are not considered in this paper. Studies on the effect of manufacturing error are given in Refs. [11,14] while studies on the effect of tooth profile modification are given in Refs. [15–18].

However, the investigation of the vibration properties of a planetary gearbox with cracked teeth is limited. Barszcz and Randall [19] applied spectra kurtosis to detect a ring gear tooth crack in the planetary gear of a wind turbine. Lewicki et al. [20] utilized vibration separation techniques to detect the tooth damage in the sun gear, planet gear and ring gear, respectively. Chen and Shao [21] investigated the vibration properties of a planetary gear set when there was a tooth root crack in the ring gear. Chaari et al. [22] applied the dynamic model developed in Ref. [14] to investigate the vibration properties of the sun gear and the carrier of a planetary gear set with tooth crack or a single pit on the sun gear. In their studies, the gear mesh stiffness was approximated as a square waveform which would generate unwanted frequency components in the dynamic response [23]. Chen and Shao [24] studied the dynamic features of a planetary gear set when a tooth crack was under different sizes and inclination angles. The displacement signals of the sun gear and the planet gear were investigated when a crack was present on the sun gear or the planet gear. But, Refs. [21,22,24] did not considered the effect of transmission path in their studies. In this study, we focus on cracks in the sun gear only considering the effect of transmission path.

All the above mentioned dynamic models can be divided into two categories: fixed coordinate model and rotating coordinate model. In order to consider inertia effect caused by the rotation of the carrier, a rotating coordinate model is more convenient to use. The inertia force contains gyroscopic force and centrifugal force. Some previous studies [14,24] considered the gyroscopic force in their dynamic models; however, they did not consider the centrifugal force properly in the equilibrium equations of planet gears. In this study, both the gyroscopic force and the centrifugal force will be considered.

A few studies considered the effect of the transmission path in the vibration signal modeling. Inalpolat and Kahraman [7] expressed the resultant acceleration signal of a planetary gear set as follows:

$$a(t) = \sum_{n=1}^{N} C w_n(t) F_{rpn}(t) \tag{1.1}$$

where C is a constant; N represents the number of planet gears; $w_n(t)$ denotes the effect of transmission path for the n-th planet gear which is a weighting Hanning function [7] with a time duration of T_c/N; and $F_{rpn}(t)$ is the dynamic force of the n-th ring-planet mesh.

Later, Inalpolat and Kahraman [11] improved the modeling considering the dynamic forces of both sun-planet meshes and ring-planet meshes. They used the same Hanning function as Ref. [7] to cover the effect of the transmission path.

$$a(t) = \sum_{n=1}^{N} (C_s w_n(t) F_{spn}(t) + C_r w_n(t) F_{rpn}(t)) \tag{1.2}$$

where C_s and C_r are constants facilitated to establish the relation between the gear mesh forces; and $F_{spn}(t)$ and $F_{rpn}(t)$ represent the dynamic forces of the n-th sun-planet mesh and the n-th ring-planet mesh, respectively.

However, the correctness of Eq. (1.2) is worth discussing. The line of force $F_{spn}(t)$ is the internal common tangent of the base circles of the sun gear and the n-th planet gear. While the line of force $F_{rpn}(t)$ is the internal common tangent of the base circles of the ring gear and the n-th planet gear. Since dynamic forces $F_{spn}(t)$ and $F_{rpn}(t)$ are not in the same direction, it is not proper to add weighted $F_{spn}(t)$ and weighted $F_{rpn}(t)$ together as two scalars.

Feng and Zuo [8] investigated possible transmission paths of vibration signals in a planetary gearbox. In Fig. 1, three transmission paths are illustrated from its origin to the transducer. According to their studies, the transducer perceived signal arriving along paths 2 and 3 will have negligible amplitude. Therefore, only the first transmission path was considered in their studies, and the transmission path was modeled by a Hanning function with a time duration of T_c.

All the previous studies modeled the effect of transmission path using a Hanning function [7,8,11]. They all assumed that as planet n approached the transducer location, its influence would increase, reaching its maximum when planet n was the closest to the transducer location, then, its influence would decrease to zero as the planet went away from the transducer. However, even if the planet n is in the farthest location from the transducer, its influence may not be zero. In this study, we propose an approach to overcome this shortcoming.

In this paper, a planetary gear dynamic model is developed to simulate the vibration signals of each gear, including the sun gear, the planet gears, and the ring gear. In this dynamic model, both the gyroscopic force and the centrifugal force are