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Research paper

Damping models identification of a spur gear pair

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ABSTRACT

In this research paper, estimation damping in spur gear pair system using Integral method and the continuous wavelet transform based approach (CWT) is proposed. In the first method, the motion equations of the model describing the system are reformulated in terms of integrals that give a set of linear equations solved by linear least squares method to allow a new methodology that starts with a constant piecewise model to bootstrap the most suitable model of damping. The constant model is validated using the wavelet demodulation approach that's based on the envelope extraction procedure to obtain the damping ratio associated with structural modes. In order to show the ability to predict the response, damping estimation procedures are tested on a signal obtained from numerical simulation of the spur gear pair system.

1. Introduction

Dynamic modeling of the gear pair system has been an active research field in engineering. A numerous dynamic models have been proposed as it's presented by Ozguven and Houser [1], Blankenship and Kahraman [2], Theodossiades and Natsiavas [3] and more recently by Wang et al. [4]. These models typically employed two rigid disks that are connected with each other through a gear mesh interface model along the line of action characterized by time-varying stiffness and mesh damping to account for the energy loss at the gear mesh interface. The dynamic behavior of the gear system is largely affected by damping. Gregory et al. [5] highlighted the importance of damping on dynamic loading and the need for further research into the mechanism of damping in mechanical transmissions. Almost all of the previous gear dynamic models employed a user-defined constant viscous damping coefficient, stating that it must be determined empirically. Therefore, the estimation of damping from an observed response data of the gear system appears to be an important task of dynamic analysis. Because of the dependence of damping on complex mechanisms in the structure, the damping is the most difficult quantity to evaluate in comparison to the identification of another parameter. Several damping models exist in the literature. Liu et al. [6] recently proposed an analytical method to characterize damping and stiffness in lightly loaded, lubricated gear pairs at different operating speeds and lubricant temperatures. A common practice is to consider the damping as a Rayleigh damping [7] obtained by combining the mass matrix and stiffness matrix. The limitations of the current viscous damping model are discussed [8] and an advanced formulation for viscous damping, which depends on frequency using the Rayleigh damping, is developed.

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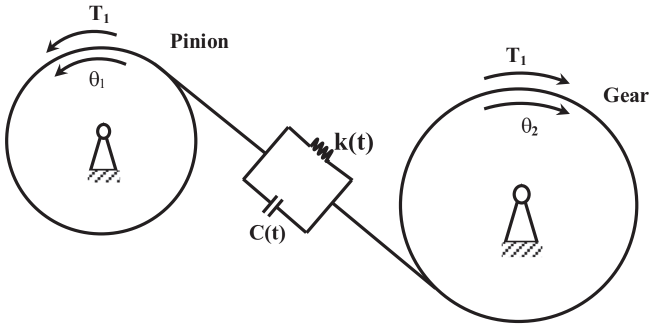

Fig. 1. The dynamic model of spur gear-pair system.

An original structural damping estimation procedure is created [9] using the well-known complex frequency response function matrix. The proposed method can be worked efficiently at simpler structures within a predicted frequency range. However, beyond the considered range, the identified FRFs do not correspond with experimental FRFs. Also, this approach based on experimental analysis [10] which requires a complex measurement for determining the frequency response functions.

The easy way to estimate the damping model is the analysis of the simulated free response of a vibrational system. In the case of SDOF systems, the signal envelope must be extracted to identify the damping from the response signal using Hilbert transforms [11]. But, in MDOF systems, the continuous wavelet transform method (CWT) that decomposes a signal in the time-frequency domain [12] can be used to overcome some limitations of the Hilbert transforms method. For example, in Ref. [13] wavelet ridge detection procedure is used to estimate the damping ratio associated with the structural modes in MDOF systems. In addition, CWT is considered as the best tool to identify a polynomial type nonlinear damping in rotors [14] and can be used to capture the nonlinearities at damping. These nonlinearities can be detected using a new methodology [15] which is developed to identify the damping model in an electro-mechanical cart system from the simply observed signal using an Integral method. But, this identification is not considering time-varying effects of stiffness and excitation force. The Integral method has been validated in biomedical applications and rocket roll dynamics [16,17] for first order systems with constant parameters. The estimation method described in this paper is based on such an approach with a new formulation which allows the use of the Integral method to estimate damping model in gear system for second order systems with time-varying stiffness and excitation force.

In spur gear pair system many models of damping existed [18–20]. The concept of the majority of these models is to begin with complex model structure first, then adjust them to the data that requires a strong initial assumption. However, the aim of this study is to start with a simplified constant model and add complexity as required to find more accurate damping model.

In this research, a one-stage gear system with mesh stiffness fluctuation is modeled. After that, we presented a theoretical approach of an integral method and we introduced the wavelet demodulation technique, including a numerical example to validate the proposed model. Then, the two procedures are tested with a simulated data from spur gear pair system. In Section 4, the proposed method is applied using different operating conditions and compared with the non-linear regression method [26] using standard functions in Matlab to finish by a conclusion in Section 5.

2. The dynamic modeling

The dynamic model of the gear system is illustrated in Fig. 1. The contact between the two gears is presented by a spring-damper. It consists of a spur gear pair with radius R_1 and R_2, masses m_1 and m_2, moments of inertia I_1 and I_2. Both gears are subjected to moments T_1. $\Theta_1(t)$ and $\Theta_2(t)$ are the two rotation degrees of freedom.

According to Newton's laws of motion, the differential equations of the gear system are given by

$$\begin{cases} I_1\dfrac{d^2\theta_1}{dt^2}+R_1c(t)\left(R_1\dfrac{d\theta_1}{dt}-R_2\dfrac{d\theta_2}{dt}\right)+R_1k(t)(R_1\theta_1-R_2\theta_2)=T_1 \\ I_2\dfrac{d^2\theta_2}{dt^2}-R_2c(t)\left(R_1\dfrac{d\theta_1}{dt}-R_2\dfrac{d\theta_2}{dt}\right)-R_2k(t)(R_1\theta_1-R_2\theta_2)=-T_1 \end{cases} \tag{1}$$

where $c(t)(R_1\frac{d\theta_1}{dt}-R_2\frac{d\theta_2}{dt})$ is the damping force, $k(t)(R_1\theta_1-R_2\theta_2)$ is the mesh elastic force and k(t) is the mesh stiffness fluctuation. In order to simplify the model, the variation of the meshing stiffness is expressed using the square waveform [21] in the following function

$$k(t)=\begin{Bmatrix} K_{max}.....for 0<t<(\varepsilon_\alpha-1)T_e \\ K_{min}.....for(\varepsilon_\alpha-1)T_e<t<T_e \end{Bmatrix} \tag{2}$$