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Research paper

Elastic dynamics modelling and analysis for a valve train including oil film stiffness and dry contact stiffness

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ABSTRACT

An elastic dynamics model for a valve train including combined stiffness of oil film and dry contact between a cam and a tappet, is proposed. Multi-directional elastic deformations of the valve train are solved by discretizing slender components into Rayleigh beam and bar elements. The oil film stiffness is determined by non-Newtonian elastohydrodynamic lubrication in the line contact, whereas the dry contact stiffness is achieved by using finite element contact analysis. A numerical solution for the dynamic model is developed and verified by conducting a dynamic stress experiment. The influences of rotation speed on the oil film stiffness and contact force between the cam and the tappet are discussed, and the combined effects of rotation speed and oil film stiffness on a dynamic transmission error of the valve train are analyzed. Results show that the oil film stiffness decreases when the cam rotation speed increases within a certain range, whereas the sharp increased contact force may enlarge the oil film stiffness when the speed exceeds a critical value. The oil film stiffness variation slight influences the contact force, but such variation significantly affects the dynamic transmission error.

1. Introduction

Valve trains are used to control the opening and closing of a valve, and the dynamic properties of these valve trains are crucial for output power and torque, fuel consumption and idling stability of engines. Elastic deformations of slender bars and beams in a valve train result in reduced accuracy of a valve lift curve. Moreover, oil film stiffness on the dynamic response of a cam mechanism is significantly affected when lubrication impact on valve train dynamics increases [1,2]. Therefore, a dynamic model with elastic deformations and elastohydrodynamic lubrication (EHL) should be developed to enhance the operating performance of valve trains.

Numerous significant works have focused on the dynamics of valve trains or cam mechanisms. Hrones [3] demonstrated that cam mechanisms with constant acceleration demonstrate a violent vibration at high speed, and the result was verified by Mitchell [4] using a dynamic experiment on the basis of an analytical solution. Furthermore, an increasing number of studies have concentrated on the dynamic performances in cam mechanisms. A two degrees-of-freedom (DOFs) dynamics model for a cam follower was built by Eiss [5], and extended by Chen and Polvanich including residual vibration [6] and nonlinear factors [7]. However, Pisano and Freudenstein [8] verified in the 80s that the dynamic model with lumped parameter for a valve train cannot accurately predict its dynamics behaviour by conducting an experiment and thus presented a

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dynamic model with distributed mass for a high-speed cam mechanism. This distributed mass model has been developed by considering various factors [9–12]. Multi-rigid-body dynamics theory is used to investigate the dynamic behaviours in cam mechanisms or valve trains given the conveniences and uniformity of this theory [13–15]. However, the rigid-body model cannot attain the mode of vibration of parts. Consequently, Guo et al. [16,17] established a rigid–flexible coupled dynamic model for a valve train on the basis of finite difference method. However, a dynamic transmission error (DTE) of the valve train remains underrated in this model.

With the development of EHL theory and its numerical solutions, the dynamic properties of the valve train including lubrication have become a major concern of many researchers. The lubrication of cam-tappet has been accorded with considerable attention since the 1950s [18–20]. Ai and Yu [21] and Dowson et al. [22] obtained transient EHL results for the cam and follower. Hooke [23–25] obtained a minimum film thickness by examining the change rate of entrainment velocity when the change in entrainment direction was observed in a running period of cam-tappet. Subsequently, additional factors, such as temperature [26–28], finite-length line contact [29], boundary lubrication [30] and mixed lubrication [28], have been considered. In their studies, the lubrication performance of the valve train was analyzed, but the interaction between the dynamic and lubrication characteristics was disregarded. Literature [17,31–34] analyzed the dynamic response of valve trains using the dynamics and EHL theory. However, these works neglected the effects of oil film stiffness on dynamic characteristics, and the EHL and dynamics equations were separately solved. The oil film stiffness between the cam and the tappet significantly influences dynamic transmission error, jump and bounce of the valve. Thus, the oil film stiffness is necessary to be included in the dynamics analysis. Qin et al. [1] and Tsuha et al. [2] analyzed the influences of oil film stiffness on the dynamic response of a cam-follower mechanism. These authors firstly solved the EHL equations by assuming a load increment to obtain the oil film stiffness and then introduced the stiffness to solve the dynamics equations. The prediction results in Refs. [1,2] indicated that a large load or low speed increases oil film stiffness.

In conclusion, studies of the above literatures indicate that EHL obviously affects the dynamic behavior of the valve train, and the dynamic response (such as dynamic load and vibration velocity) in turn change the oil film stiffness in EHL. It is clear that EHL and dynamics equations are coupled with each other, and these coupled equations should not be independently solved like the above literatures. Therefore, the present work proposes a numerical iterative approach to solve the coupled equations, and aims to further investigate the interaction between EHL and dynamics. In this work, a valve train dynamics model, including oil film stiffness and elastic deformations in multi-directions, is firstly developed by coupling beam and bar elements with non-Newtonian line contact EHL. In the proposed model, the compression, shear and bending deformations in a rocker arm and the compression deformations in other components (e.g. tappet, pushrod and valve) are simultaneously considered. A combined contact stiffness model is established by combining the oil film and dry contact stiffness between the cam and the tappet. The oil film stiffness is obtained using the non-Newtonian EHL in a line contact, and the dry contact stiffness is calculated by using finite element (FE) contact analysis. An experiment is then conducted to verify the predicted pushrod stress. The effects of a cam rotation speed on oil film stiffness, contact force and minimum oil film thickness between the cam and the tappet are analyzed, and the combined impacts of oil film stiffness and cam rotation speed on the DTE of the valve train are discussed.

This work is organized as follows. In Section 2, an elastic dynamic model for the valve train including oil film stiffness and deformations in multi-directions is developed. In Section 3, the combined stiffness model between the cam and the tappet is established by combining the oil film stiffness and the dry contact stiffness. In Section 4, the numerical solutions to the dynamics model are defined, and a dynamic experiment is performed to verify the predicted pushrod stress. The theoretical natural frequency and mode shape of the valve train, the oil film stiffness, contact force and minimum oil film thickness between the cam and the tappet, and the DTE of the valve train under different rotation speeds are discussed in Section 5. The conclusions drawn from this work are presented in Section 6.

2. Dynamics model

Fig. 1 illustrates a valve train dynamics model with combined stiffness and multi-directional elastic deformations, where the combined stiffness K_c comprises the oil film stiffness K_{oil} and dry contact stiffness K_d between the cam and the tappet. C_{oil} and C_d represent the oil film and dry contact damping, respectively, between the cam and the tappet. K_{ij} represents the contact stiffness between components i and j, and C_{ij} represents the contact damping between the two components. K_{4x}, K_{4z}, C_{4x} and C_{4z} denote the bearing stiffness and damping in directions X or Z. I–VII denote the element numbers and ①–⑪ denote the node numbers.

The valve train is divided into 7 elements with 11 nodes based on the Rayleigh beam and bar elements. Thus, the proposed dynamic model in XOZ with 20 DOFs can be described as follows:

$$\mathrm{M}\ddot{\mathrm{U}}+\mathrm{C}\dot{\mathrm{U}}+\mathrm{K}\mathrm{U}=\mathrm{F} \tag{1}$$

where

$$\mathbf{U}=\begin{bmatrix} u_1, & u_2, & u_3, & \cdots & u_{20} \end{bmatrix}^{\mathrm{T}}$$

In Eq. (1), **M** refers to the mass matrix, and **K** denotes the stiffness matrix. Their values are derived by Lagrange equation, and the detailed derivation is shown in Ref. [35]. **C** represents the damping matrix, and Rayleigh damping is applied in this work, that is $\mathbf{C}=\alpha\mathbf{M}+\beta\mathbf{K}$ (i.e. damping ratio $\xi_i=(\alpha\omega_i^{-1}+\beta\omega_i)/2$). In general, the damping ratio of the valve train is less