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Sensitivity analysis and parameter optimization for vibration reduction of undamped multi-ribbed belt drive systems

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Abstract

Tensioner is a critical mechanism to ensure a constant tension level within a serpentine belt drive that is widely used in modern passenger vehicles. For a belt drive with n pulleys, generic and explicit formulae about sensitivities of both frequency and steady harmonic responses are established in terms of system matrices with respect to any design parameter of the system. Deductions from the formulae results in frequency and steady response sensitivities relative to key tensioner parameters and the belt speed. Based on sensitivity analysis, optimizations are conducted on tensioner so as to suppress dynamic responses of the system by frequency detuning. A new approach for searching optimal parameters is put forward by incorporating sensitivity information into a classical coordinate alternating procedure. Examples are given to validate the analytical formulae of the frequency sensitivity and to demonstrate the effect of optimization.
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1. Introduction

A serpentine belt drive is widely used to power all front accessories of the engine in modern passenger vehicles. In order to keep a constant tension level in the belt, an automatic belt tensioner is usually adopted, which consists of a tensioner pulley, a tensioner arm and a torsional spring. Although a belt drive is much quieter compared with other drives such as gear drives and so on, evident vibration might occur at certain engine speed due to torque fluctuation from the crankshaft.

Typical vibrations can be roughly classified into three categories. The first category is rotational vibration of the pulleys and tensioner arm around corresponding shafts, where the belt spans act as axial springs [1,2]. The second one is transverse oscillation of belt spans with various boundary conditions supplied by the pulleys and the tensioner [3,4]. Lateral vibration of the belt spans is the third class [5]. It should be pointed out that vibrations of different categories might couple with each other under some circumstances, especially between rotational vibration of pulleys and transverse vibration of belt spans [6,7].

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Being able to capture the coupling between rotational and transversal vibrations and thus more accurately describe the actual dynamics of the system, hybrid models have become popular for dynamic analysis on front-end accessory belt drive. A typical hybrid model consists of both discrete variables for the dynamic behavior of pulleys and the tensioner arm, and continuous variables for the transverse dynamics of the belt spans.

For a model belt drive with two pulleys and a tensioner, which was decomposed to an independent belt span and a hybrid subsystem, a Holzer approach with two iteration loops was put forward by Beikmann to calculate the natural frequencies of the hybrid subsystem [6]. Zhang and Zu adopted this prototypical model and derived an explicit equation of frequency for free vibration analysis [7]. Starting from the explicit frequency equation presented in Ref. [7] for the model belt drive, a governing equation on frequency sensitivity with respect to the tensioner parameters was derived by the present authors [8]. The equation was then numerically solved for frequency sensitivity relative to some key design parameters. The approach was validated by results from directly applying a finite difference method to frequency-parameter results obtained by means of the Holzer approach.

It is obvious that the explicit characteristic equation varies with belt configuration and has to be derived case by case. Realizing the demerit, Parker [9] put forward a new method for eigensolution to the hybrid model by introducing Lagrange multipliers to a model of belt drives of n pulleys, where the belt spans adjacent to the tensioner were discretized. Sensitivity analysis of the frequency was also conducted with respect to various design parameters by means of a perturbation method.

The limitation of methods based on the explicit characteristic equation also motivates the present work. For a belt drive with n pulleys, generic and explicit formulae about sensitivities of both frequency and steady harmonic response are established in terms of system matrices with respect to any design parameter of the system. Deductions from the formulae results in frequency and steady response sensitivities relative to key tensioner parameters and the belt speed. Based on sensitivity analysis, optimizations are conducted on tensioner so as to suppress dynamic response of the system. A new approach for searching optimal parameters is put forward by combining a classical procedure and sensitivity analysis. Examples are given to, respectively, validate the analytical formulae of the frequency sensitivity and to demonstrate the effect of optimization.

2. Modeling and equations of motion

2.1. Model and assumptions

For a practical belt drive, the initial tension is usually far less than the tensile stiffness of the belt, i.e. $P_{0i} \ll \mathrm{EA}$. The longitudinal wave speed is much faster than the transverse wave speeds corresponding to lower order transverse vibrations. In addition, belt-pulley wedging and slip are not taken into consideration for free vibration analysis of a belt drive under normal operation and small belt motions. Furthermore, in quite a few researches, the flexural stiffness of the belt is neglected and the belt speed is regarded as constant.

The above assumptions are also adopted in this paper, which render the belt to behave pseudo-statically. As a sequel, the mass and stiffness of the belt are uniformly distributed and do not change during operation and vibration. Furthermore, the damping, which plays an important role for the behavior of the system at quasi-resonant vibration, is neglected for the simplicity of introducing the idea and method. It is also based on the same consideration to assume the belt speed as constant although the actual speed usually varies.

Fig. 1 schematically shows a serpentine belt drive that has n pulleys including the tensioner pulley. The pulleys and belt spans are numbered anti-clockwise, with the crankshaft pulley as the first pulley and the span from pulleys 1 to 2 as the first span, respectively. The tensioner arm is pivoted at one end by a torsional spring with k_r as the stiffness coefficient, and supports at another end pulley j ($j = 2$ in Fig. 1) that is also called as tensioner pulley. The constant translating-speed of the belt is notated as c.

The length of the tensioner arm and the moment of inertia of the arm with respect to the pivot are notated as L and J_t, respectively. For pulley $i (i = 1,2,\ldots,n)$, its radius and the moment of inertia relative to its center are denoted by r_i and J_i.

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