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Transient vibration analytical modeling and suppressing for vibration absorber system under impulse excitation

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ABSTRACT

The impulse excitation of mechanism causes transient vibration. In order to achieve adaptive transient vibration control, a method which can exactly model the response need to be proposed. This paper presents an analytical model to obtain the response of the primary system attached with dynamic vibration absorber (DVA) under impulse excitation. The impulse excitation which can be divided into single-impulse excitation and multi-impulse excitation is simplified as sinusoidal wave to establish the analytical model. To decouple the differential governing equations, a transform matrix is applied to convert the response from the physical coordinate to model coordinate. Therefore, the analytical response in the physical coordinate can be obtained by inverse transformation. The numerical Runge-Kutta method and experimental tests have demonstrated the effectiveness of the analytical model proposed. The wavelet of the response indicates that the transient vibration consists of components with multiple frequencies, and it shows that the modeling results coincide with the experiments. The optimizing simulations based on genetic algorithm and experimental tests demonstrate that the transient vibration of the primary system can be decreased by changing the stiffness of the DVA. The results presented in this paper are the foundations for us to develop the adaptive transient vibration absorber in the future.

1. Introduction

Vibration caused by impulse excitation widely exists in engineering applications. For example, the vibration caused by impulse excitation should be attenuated to micrometers or even nanometers level to improve the image quality and line-of-sight precision of satellites [1]. The water waves and missile firing bring impulse excitation to the ships and vessels whose structural strength is always enough under the impulse excitation, but the transient vibration caused by the impulse excitation is difficult to control [2, 3]. The vibration of building floor subjected to footsteps and the spacecraft subjected to impact or thruster force both need to be suppressed [4, 5]. In order to control the vibration caused by the impulse excitation, a method to exactly model the impulse response should be proposed and analyzed.

Dynamic vibration absorbers (DVA) are usually applied to suppress the steady state vibration caused by a harmonic excitation with a given frequency [6]. Due to the narrow effective bandwidth, the vibration may increase significantly and the dynamic vibration absorbers (DVAs) lose efficiency when small changes occur with respect to the exciting frequency of

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the primary system [7]. Therefore, lots of achievements focus on making the DVAs' natural frequency tunable [8–10].

Active control is an effective method to suppress the impulse vibration by adapting characteristics and parameters of the absorber according to the variation of exciting force, but it has disadvantages of all-time power consumption, complex control law, and low reliability. A virtual-vibration-absorber was introduced in [11], its stiffness, interia and damping coefficient were online tunable to mathematically equivalent to DVA, and the efficient algorithm allowed improving the frequency resolution by active vibration control method. Similarly, voice coil motors were employed to serve as actuators in an active dynamic vibration absorber which could adopt to different conditions [12], and the voice coil motor was analyzed in [13]. However, these achievements all aim to control the steady state vibration of harmonic excitation.

On the contrary, passive and semi-active vibration control are more simple and stable, however, few efforts are focused on transient vibration control [14]. Semi-active method is easily to achieve the adjustment of stiffness or interia value by getting the feedback signal from the response. A flexible cantilever beam attached with a mass was used to develop the DVA, and the stiffness of the beam varied with the change of the effective length of the beam [15]. Similar to the cantilever beam ATVA reported in [15], an absorber with circular cross-section beam was used to compare two tuning algorithms [16]. In order to avoid the uncertainty in mechanical direct driving process, an electromagnetic DVA was presented in [17], and it consisted of a clamped-clamped beam and a permanent magnet which was embedded in the center of the beam, and the electromagnetic force was produced by the C-shaped electromagnetic coil and the permanent magnet, moreover, the stiffness changed when the current in the coil was adjusted. Piezoelectric element combined with a resonant electrical circuit was applied to realize stiffness variation for the semi-active ATVAs in [18, 19], and the natural frequency of the absorbers was tuned to the required frequency electrically, furthermore, the electrical signal of piezoelectric element was able to be used as sensors in DVAs. Shape memory alloy was another kind of smart material which could be used as a variable stiffness element of DVAs, the Young's modulus of the element was changed by adjusting the temperature of the alloy by passing electrical current through it, thus it resulted in the changing of the natural frequency [20–22]. Moreover, magneto-rheological fluid was used to achieve stiffness changing, and the shear stiffness of the fluid was adjusted by changing the magnetic field applied to the fluid [23–25]. Although the passive and semi-active DVAs are simple and stable, few efforts aim to suppress the transient vibration caused by impulse excitation.

An adaptive DVA using magnetorheological elastomer was applied to reduce vehicular powertrain transient vibration, but it was designed for the vehicular powertrain vibration happened during the transient stage of acceleration [26]. A control algorithm was proposed for semi-passive absorber to control the transient response [27], and the analytical model is based on the perturbation method [28, 29], but this model has assumed the natural frequencies of the primary system and the absorber are equal all the time, so it can't be applied to the tunable transient absorber.

We aim to develop the adaptive transient vibration absorber, which can suppress the vibration of the primary system to the ideal minimum level when the impulse force and impact time are acquired by a sensor. Analytical modeling of the response is a key step to obtain the optimum parameters of the transient absorber. However, to the best of our knowledge, few efforts have been devoted to developing the analytical model of the DVA under impulse excitation and suppressing the transient response of the primary system by optimizing the parameters of the DVA, which can help us study the adaptive transient vibration absorber in the future. Thus in this paper an analytical model is proposed to obtain the response of the DVA attached to the primary system under impulse excitation, and the stiffness of the DVA is optimized based on the genetic algorithm to suppress the transient vibration of the primary system. The rest of this paper is organized as follows. In Section 2, after introducing dynamic model of the primary system with DVA, the instantaneous exciting force is simplified as sine wave to establish the analytical model, and the analytical model of the transient response is proposed. In Section 3, Runge-Kutta method shows that the numerical simulations coincide with the analytical model, and the wavelet of the response indicates that it consists of components with multiple frequencies. In Section 4, experimental tests are employed to verify the effectiveness of the analytical model proposed. In Section 5, the optimizing simulations and experimental tests show that the transient vibration of the primary system can be decreased by changing the stiffness of the DVA. Finally, some conclusions are summarized.

2. Analytical model of the transient response

2.1. The system and the impulse excitation

The DVA and the primary system can be modeled as in Fig. 1, the governing equations are

$$\begin{aligned} &m_1\ddot{x}_1 + c_1\dot{x}_1 + k_1x_1 + c_2(\dot{x}_1 - \dot{x}_2) + k_2(x_1 - x_2) = F \\ &m_2\ddot{x}_2 + c_2(\dot{x}_2 - \dot{x}_1) + k_2(x_2 - x_1) = 0 \end{aligned} \tag{1}$$

Where x_1 is the displacement of the primary system, x_2 is the displacement of the DVA. The primary system subjected to impulse excitation F is modeled as a single-degree-of-freedom (SDOF) structure with stiffness k_1, mass m_1, and the damping value c_1. The DVA is also modeled as a SDOF structure with stiffness k_2, mass m_2, and the damping value c_2.

In order to establish the common analytical model which can be applied to analyze the primary system attached with DVA under impulse excitation, the instantaneous force is simplified as sine wave shown in Fig. 2. Fig. 2(a) and Fig. 2(b) show