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Research Article

Robust adaptive vibration control of a flexible structure

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A B S T R A C T

Different types of L1 adaptive control systems show that using robust theories with adaptive control approaches has produced high performance controllers. In this study, a model reference adaptive control scheme considering robust theories is used to propose a practical control system for vibration suppression of a flexible launch vehicle (FLV). In this method, control input of the system is shaped from the dynamic model of the vehicle and components of the control input are adaptively constructed by estimating the undesirable vibration frequencies. Robust stability of the adaptive vibration control system is guaranteed by using the L1 small gain theorem. Simulation results of the robust adaptive vibration control strategy confirm that the effects of vibration on the vehicle performance considerably decrease without the loss of the phase margin of the system.

1. Introduction

Vibration and flexibility in large scale industrial products are inevitable problems and many research have been devoted to designing active vibration control systems [1,2]. Structural flexibility is more important for aerospace systems because of the size of structures and accuracy of missions. In this regard, launching vehicles from aerospace systems significantly encounter with vibrational challenges because of large size structures. The main types of vibration are bending vibration, fuel sloshing, and servo-oscillation [3]. Vibration control, including adaptive, robust, and optimal approaches, has been employed to remove the destructive effects of flexibility and undesired oscillations. Literature reviews on attitude control of flexible launch vehicles (FLVs) have been presented in several investigations [4–6].

Adaptive control is one of the most practical approaches for vibration control of aerospace structures [7]. Direct methods of adaptive control, indirect methods and adaptive filtering have been used for active vibration control and vibration suppression [8]. In new strategies, adaptive filtering as a practical and useful method has been applied to closed loop control systems of aerospace devices. From these approaches, Englehart and Krause have proposed an analog notch filter with a least square algorithm to reduce the bending vibrational effects on an aerospace launch vehicle [9]. Choi et al. [7,10] have also designed an adaptive control approach for the attitude control of a flexible aerospace vehicle. They used the root mean square method to estimate the bending frequency and one digital notch filter to reduce the flexible behaviors, considering the first and the second bending vibration modes. Oh et al. have proposed an attitude control of a flexible launch vehicle using an adaptive notch filter [4]. In another work, Khoshnood et al. have studied a model reference adaptive control for reducing undesired effects of bending vibration for an aerospace launch vehicle [5]. Elmelhi has designed a modified adaptive notch filter based on neural network for a flexible dynamic system [11]. In spite of simplicity and accuracy of these filtering strategies, linear and nonlinear changes in the phase margin of the closed loop control system as a result of the adaptive filter bandwidths produce considerable limitations. On the other hand, application of adaptive filters may lead to violation of the stability and performance of the preliminary control system. In this regard, Zhijian et al. have presented an interpolated Fourier transform based on an adaptive notch filter to solve the problem of phase lag in a conventional notch filter [12].

In the present study, a new framework of adaptive filtering strategy is proposed to improve the performance of this strategy for vibration control. The main idea of this method is the construction of vibration control input using undesirable vibration frequencies. All of the recent works related to the vibration control of FLVs have used the adaptive filtering strategy in the case of inserting the filters in the feedback of the main control system. In this paper, for vibration control the adaptive filtering approach is implemented based on modification of the control input of the system. Remarkably, this idea has been taken from the medical approaches in which anti-viruses like a noise rejection controller are made from the same viruses. Therefore, two basic improvements are applied to the adaptive filtering strategy. The

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first improvement is construction of the control input of the system using the results of filtering (estimated frequencies) instead of inserting the filter into the feedback of the closed loop control system. This change leads to set the undesired phase delay effects of the filter out of the closed loop control system. The second change is to analyze the robustness of the closed loop control system and the estimation algorithm. These two revisions produce significant results for vibration suppression, as shown in a numerical simulation.

The robust adaptive control design in which the adaptive control approaches are implemented based on robust theorems has been developed in the framework of L1 adaptive control systems [13]. Regarding the robust adaptive approaches, in this study, a model reference adaptive filtering scheme considering robust theories is investigated to extract a practical control system for vibration control of an FLV. The components of the control input are adaptively tuned with estimating undesirable vibration frequencies. The bending vibration parameters such as vibration frequencies are changed according to the mass consumption rate of the vehicle and accurate estimation of undesirable vibration frequencies is essential. As a result of this, sub-band adaptive filtering is considered instead of fullband analysis. On the other hand, sub-band analysis is employed to expand the estimation from one vibration frequency to two or more frequencies. Therefore, a sub-band adaptive filter is designed using the discrete Fourier transform (DFT) technique. This technique can be used for multi-frequency estimation with satisfactory persistence of excitation properties [14].

The vibration control system is employed for attitude control of the FLV. The FLV considered in this study is a multi-vibration mode flexible structure. The inertial navigation system (INS) of the vehicle measures the bending vibration as well as the rigid body motion. Simultaneous measurement of the rigid and the flexible body rotations may violate the closed loop control system, which in the worst cases may lead to resonance. The primary aim of the robust adaptive control system is to decrease this resonance.

In this study, a linear model of the FLV in the pitch channel is considered for vibration and attitude control. The control system is also applied to the yaw channel of the vehicle at the same structure as a result of dynamic symmetric properties of the vehicle. Moreover, according to flight conditions, three vibration modes are analyzed for modeling the bending vibration using the assumed modes method.

This paper is organized as follows: Section 2 provides linear equations of motion of the FLV. Section 3 describes the design procedure of the robust adaptive control system. Robust estimation of the control components and construction of the control input are given in this section. Section 4 discusses the robust stability of the new vibration control system and Section 5 provides the simulation results of the proposed control approach. The conclusions of the work are presented in Section 6.

2. Equations of motion

Equations of motion of an FLV (rigid and flexible body) are given in this section. In one of the studies, nonlinear equations of motion of a launch vehicle and the method of linearization have been addressed by Roshanian et al. [15]. Linear equations of motion of the rigid body dynamic are

$$\dot{\alpha}=\left(\frac{1}{2m_s}\rho U_0 S(C_{z\alpha}-C_{x0})-\frac{T}{m_sU_0}\right)\alpha+\left(\frac{1}{4m_s}\rho SDC_{zq}+1\right)q-\frac{C_T}{m_sU_0}(\delta_2+\delta_4)$$

$$\dot{\beta}=\left(\frac{1}{2m_s}\rho U_0 S(C_{y\beta}-C_{x0})-\frac{T}{m_sU_0}\right)\beta+\left(\frac{1}{4m_s}\rho SDC_{yr}-1\right)r+\frac{C_T}{m_sU_0}(\delta_1+\delta_3)$$

$$\dot{p}=\left(\frac{1}{4I_x}\rho U_0SD^2C_{lp}\right)p+\frac{C_T\,dy}{I_x}(\delta_1-\delta_2-\delta_3+\delta_4)$$

$$\dot{q}=\left(\frac{1}{2I_y}\rho U_0^2Sx_{ac}C_{z\alpha}\right)\alpha+\left(\frac{1}{4I_y}\rho U_0SD^2C_{mq}\right)q-\frac{C_T\,dx}{I_y}(\delta_2+\delta_4)$$

$$\dot{r}=-\left(\frac{1}{2I_y}\rho U_0^2Sx_{ac}C_{y\beta}\right)\beta+\left(\frac{1}{4I_y}\rho U_0SD^2C_{nr}\right)r-\frac{C_T\,dx}{I_y}(\delta_1+\delta_3) \quad (1)$$

where α and β represent the angle of attack and the side slip angles, respectively; p, q and r are the angular velocities of the vehicle; U_0 is the magnitude of the velocity in the x direction; m_s and $I_{x,y}$ are the mass and the mass moment of inertia in the x and y directions, respectively; and ρ is the air density. In addition, S, D and T are the reference surface, the reference length, and the thrust, respectively; C_{ij} is the aerodynamic coefficient; δ_i is the ith actuator deflection in the pitch and yaw directions; d_x, d_y and C_T are the thrust coefficients; and x_{ac} is the position of the aerodynamic center of the vehicle.

Using an arbitrary state space representation, the transfer function between the pitch (yaw) angular velocity and the pitch (yaw) actuator deflections are extracted as

$$\frac{q}{\delta_2}=\frac{a_1s+a_2}{s^2+a_3s+a_4} \quad (2)$$

where a_i is derived from Eq. (1). Because the vehicle is dynamically symmetric, only the pitch channel of the vehicle is considered. In addition, the actuator deflections of the pitch channel are assumed to operate uniformly ($\delta_2=\delta_4$).

The attitude determination system of the vehicle in the pitch channel as shown in Fig. 1 measures the rigid body motion and the elastic deflection. In this figure, θ is the pitch angle associated with the rigid body motion and θ_b is the pitch angle associated with the flexible body motion. The rigid and the elastic motions can be integrated as

$$q_T=q_r-q_b \Rightarrow \dot{q}_T=\dot{q}_r-\dot{q}_b \quad (3)$$

where q_b is the angular velocity arisen from the bending vibration; q_r is the pure rigid body angular velocity of the vehicle in the pitch channel (q from Eq. (2)); and q_T is the total angular velocity. The mathematical model of the vehicle structure is assumed to be a free–free Euler–Bernoulli beam as follows:

$$\rho A\frac{\partial^2 y}{\partial t^2}+\frac{\partial^2}{\partial x^2}\left[EI\frac{\partial^2 y}{\partial x^2}\right]=f(x,t) \quad (4)$$

where ρA is the mass of the vehicle per unit length; y is the deflection of the vehicle structure from the bending moment; E is the Young modulus; I is the area moment of inertia about the neutral axis; and $f(x,t)$ is the external force per unit length. Using the assumed mode method, Eq. (4) can be separated into a time dependent part as

$$\ddot{g}_{bj}+2\varsigma_j\omega_j\dot{g}_{bj}+\omega_j^2g_{bj}=\frac{Q_j}{M_j}=K_jf_j\delta_2 \quad (5)$$

where g_{bj} is the generalized coordinate for the jth mode of the bending vibration; M_j and Q_j are the generalized mass and the force of the jth bending vibration mode, ω_j is the natural frequency of the jth bending vibration mode; ς_j is the damping ratio of the jth bending vibration mode; K_j is a proportional constant; and f_j is

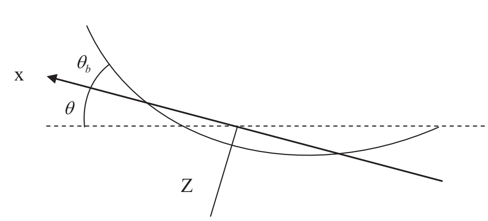

Fig. 1. Rigid and flexible coordinates defined on the vehicle body.

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