Mechanical Systems and Signal Processing 39 (2013) 461–470

Contents lists available at SciVerse ScienceDirect

Mechanical Systems and Signal Processing

journal homepage: www.elsevier.com/locate/ymssp

Vibration reduction for flexible systems by command smoothing

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ARTICLE INFO

Article history:
Received 5 September 2012
Received in revised form
20 December 2012
Accepted 25 February 2013
Available online 19 March 2013

Keywords:
Command smoothing
Flexible system
Vibration reduction
Input shaping
Modeling error

ABSTRACT

A method is presented for limiting vibration in flexible systems by smoothing the original command. The original command is smoothed to drive the flexible systems without inducing vibration. The proposed smoother is designed as a function of system natural frequency and damping ratio. The comparison between the new method and the zero vibration and triple derivative (ZVDDD) shaper for modeling error is explored and quantified. The theoretic analysis shows that the new method is more insensitive at higher frequencies, while the ZVDDD shaper has more insensitivity at lower frequencies. Moreover, the designed smoother benefits vibration reduction for multi-mode systems. Experimental results for single-mode and multi-mode systems validate the simulated dynamic behavior and the effectiveness of the new method.

1. Introduction

Flexible dynamic systems suffer from unwanted transient deflection and residual vibration. Those detrimental effects cause significant problems for positioning accuracy, effectiveness, fatigue and safety for many types of systems ranging from nano-positioning devices to large industrial cranes [1]. Therefore, there exists a need for a control system that can effectively reduce vibration.

Numerous researchers have worked to provide solutions to the challenging problems posed by the flexible dynamic systems. The work can roughly be broken into three categories: feedback control, input shaping and command smoothing. The feedback control strategies use measurements and estimation of system state to suppress vibration. Qiu designed an adaptive nonlinear controller for vibration reduction. The gains of the controller were varied with the vibration amplitude. The performance of the controller was verified on a flexible Cartesian manipulator [2]. Yu et al. proposed a robust controller for vibration suppression of rotor systems supported by magnetic bearings. Simulations demonstrated the effectiveness of the controller and its robustness to uncertainties, such as sudden disk mass loss and speed variation [3].

Input shaping can effectively reduce the oscillatory dynamics of many types of flexible dynamic systems including bridge cranes [4], tower cranes [5], boom cranes [6], container cranes [7], coordinate measurement machines [8], spacecrafts [9], robotic arms [10], robotic workcells [11], demining robots [12], micro-milling machines [13], nano-positioning stages [14], and linear step motors [15]. The input shaping process is demonstrated as follows. The original command produces an oscillatory response. To eliminate the oscillatory response, the original command is convolved with a series of impulses, called the input shaper, to create the shaped command. The shaped command can move the system without inducing vibrations. For a three-impulse zero vibration and derivative (ZVD) shaper, the amplitudes, A_i, and times,

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http://dx.doi.org/10.1016/j.ymssp.2013.02.021

t_i, of the impulses are given by [10]

$$\begin{bmatrix} A_i \\ t_i \end{bmatrix} = \begin{bmatrix} \frac{1}{1+2K+K^2} & \frac{2K}{1+2K+K^2} & \frac{K^2}{1+2K+K^2} \\ 0 & 0.5T_d & T_d \end{bmatrix} \quad (1)$$

where T_d is the damped period of vibration and K is given by

$$K = e^{(-\zeta\pi/\sqrt{1-\zeta^2})} \quad (2)$$

where ζ is the damping ratio. The convolution is performed by simply multiplying the original command by the amplitude of the first impulse, and adding it to the original command multiplied by the amplitude of the second impulse and shifted in time by one-half of the damped vibration period, and then adding it to the original command multiplied by the amplitude of the third impulse and shifted in time by a damped vibration period. Note that the rise time of the shaped command is increased by the duration of the input shaper.

There have been hundreds of papers on smooth command profiles to drive flexible systems and reduce vibration, such as S-curves [16], trigonometric transition functions [17], Gaussians [18], spline function [19] and cam polynomials [20]. Smooth profiles have been used to reduce the tendency to excite flexible system vibration. The trajectory ramps up to peak acceleration and ramps down to constant velocity, thereby producing a smooth velocity profile. The smooth transitions between boundary conditions avoid vibration. Note that vibration suppression comes at the cost of increased duration of this ramp-up time.

While significant work has been directed at smooth profiles to reduce vibration, however, these methods usually fail to fully exploit the known properties of the system. Instead, they simply provide a low-pass filtering effect [21]. The contribution of this paper is a novel method for command smoothing in order to limit the residual vibrations. The proposed smoother is a function of the system parameters, such as natural frequency and damping ratio. An increase in robustness to modeling errors in natural frequency and damping must be traded off against an increase in rise time. However, the robustness for a given rise time will differ between design methods. Therefore, an effective method for vibration reduction should provide the optimal robustness for a given rise time. Note that the proposed smoother can only reduce the vibration induced by intentional motions commanded by the human operator. However, it cannot reject external disturbances.

The rest of this paper is organized as follows. The Section 2 presents the design of the smoother for vibration reduction. In Section 3, the comparison between the smoother and the zero vibration and triple derivative (ZVDDD) shaper for modeling error in natural frequency and damping ratio is investigated. The higher-order derivative smoother, which increases the robustness, is presented in Section 4. Section 5 demonstrates the effectiveness of the new method on single pendulum and double pendulum crane.

2. Design of command smoothing for vibration reduction

The command smoothing is a control method that dramatically suppresses motion-induced vibration by intelligently smoothing the original command. Using estimates of system natural frequency and damping ratio, the smoother is designed.

If the system can be modeled as a second-order harmonic oscillator, then the system response from the smoothed impulse command, u, is

$$f(t) = \int_{\tau=0}^{+\infty} u(\tau)\frac{\omega}{\sqrt{1-\zeta^2}} e^{-\zeta\omega(t-\tau)} \sin(\omega(t-\tau)\sqrt{1-\zeta^2})\mathrm{d}\tau \quad (3)$$

where ω is the natural frequency of the system. The vibration amplitude of system response (3) is

$$A(t) = \frac{\omega}{\sqrt{1-\zeta^2}} e^{-\zeta\omega t}\sqrt{[S(\omega,\zeta)]^2 + [C(\omega,\zeta)]^2} \quad (4)$$

where,

$$S(\omega,\zeta) = \int_{\tau=0}^{+\infty} u(\tau)e^{\zeta\omega\tau}\sin(\omega\tau\sqrt{1-\zeta^2})\mathrm{d}\tau \quad (5)$$

$$C(\omega,\zeta) = \int_{\tau=0}^{+\infty} u(\tau)e^{\zeta\omega\tau}\cos(\omega\tau\sqrt{1-\zeta^2})\mathrm{d}\tau \quad (6)$$

If Eqs. (5) and (6) are limited to zero, the smoothed impulse command, u, would cause no residual vibration. In order to add the robustness of the smoother under variations of the system natural frequency and damping ratio, a new constraint must be added. The derivative of Eqs. (5) and (6) with respect to ω and ζ can also be set equal to zero [10]. Changes in