THE DESIGN OF MULTIMODE VIBRATIONS SUPPRESSION COMPENSATORS FOR SPECIFIED OVERSHOT AND PEAK TIME

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Abstract: This paper presents a design method for an open-loop command preshaping vibrations suppression compensator that ensures a specified dynamic behavior for the original mechatronic system. A reference model is required in the form of a second order custom transfer function. This reference model is build to embed the specified overshot and peak time. Then, a multimodal representation of the original system is created using a parallel composition of independent second order transfer functions for each mode. The compensator calculation is made easy from the two previous models. A numerical example for a 7-DOF merchatronic system is also provided.

1. Introduction

The modern mechatronic systems require fast and precise motion. Being a multidisciplinary approach, the mechatronic systems design offers highly efficient solutions to single domain problems [1], such as mechanical vibrations suppression.

The mechanical vibrations are omnipresent and usually undesired. Apart from the cases such as vibrations based manufacturing or processing, the effect they have on particular systems is unwanted and alters the dynamic performances of the affected systems [2]. For example, in the case of manufacturing processing, the undesired mechanical vibrations severely alter the characteristics of the resulting surfaces and edges [3].

Many solutions for the counteracting of these undesired effects have been developed over time [4]. Most well known solutions are of purely mechanical nature such as the vibrations absorbers, vibrations dampers and oscillating masses [5]. They are effective if carefully designed and implemented, but have a low flexibility. Other solutions that cross the boundary of mechanical systems have better performances and flexibility. Some of the best examples are the electromechanical active force devices. As well as the purely mechanical solutions, electromechanical solutions are also invasive as they alter the structure of the target system. Also, both types of solutions are subject to fatigue and non negligible maintenance costs.

A much more flexible and non-invasive approach has been developed with the introduction of numerical automatic control and command preshaping techniques. Command preshaping appears in various forms such as Posicast [6], input shaping [7] or general purpose filters preshaping [8]. A compilation of time delay input shaping solutions has been made in [9]. However, given the fact that these solutions are sensitive to errors in the system model, more robust solutions have been proposed in [10].

Many positioning applications require precise timing and dynamical performances. It is important that the overshot caused by the residual vibrations is kept to an allowed value and also the moment it appears is precisely controlled. To this end, a solution of vibrations suppression compensation has been proposed. It requires a priori or online knowledge of the target system's modal behavior and a reference model embedding the desired performances. The reference model performance indicators of interest are the overshot and the peak time. These solutions prevent the effects of the undesired vibrations only by altering the system's original command and usually imply microprocessor based implementations.

2. The vibrations suppression compensator design

The proposed vibrations suppression compensator (or VSC) design method has been developed over time and lead to several flavors such as Zero Overshot (ZO), Specified Overshot (SO), Specified Settling Time (SST) and Specified Overshot and Settling Time (SOST) as described in [11].

This paper extends the main approach with a new combined version of Specified Overshot and Peak Time (SOPT).

In order to provide a system output with the desired dynamic behavior, the original step command signal must be reshaped into a more complex one that embeds the compensation information. The overall structure of the compensated system is given in Fig. 1.

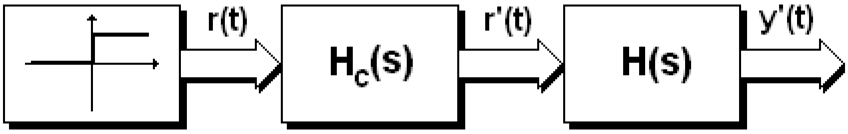

Figure 1. Command preshaping using the VSC approach

The original target system's continuous transfer function is denoted *H(s)*. The compensator's transfer function is $H_c(s)$. This shows that the vibrations suppression does not alter the original system's structure, but only the system's input *r(t)*. The goal of the design strategy is the calculus of the $H_c(s)$ function.

The compensator $H_c(s)$ is obtained by the division of the reference model $H_m(s)$ embedding the desired dynamic performances with the multimodal target system *H(s)* as shown in Eq. (1).

$$H_c(s) = \frac{H_m(s)}{H(s)} \tag{1}$$

There are two main topics to be covered in the following paragraphs: the multimodal transfer function identification and the design of the reference model.

The original system will be modeled as a parallel connection of independent low-pass second order transfer functions that represent each mode, regardless of the actual physical structure of the system. The general second order function used in this model is given in Eq. (2). The resulting multimodal system is shown in Eq. (3).

$$H_i(s) = \frac{K_i \cdot \omega_{ni}^2}{s^2 + 2 \cdot \xi_i \cdot \omega_{ni} \cdot s + \omega_{ni}^2} \tag{2}$$

$$H(s) = \sum_{i=1}^{n} H_i(s) \tag{3}$$

The reference model will be built in the simple yet sufficient form of a second order transfer function similar to the one shown in Eq. (2). However, the reference model parameters will be chosen in such way that they embed the desired dynamic behavior, in this case the specified overshot σ and the specified peak time t_p. The reference model parameters are shown in Eq (4).

$$K_m = \sum_{i=1}^{n} K_i$$
$$\omega_{nm} = \frac{\sqrt{\pi^2 + (\ln\sigma)^2}}{t_p} \tag{4}$$
$$\xi_m = \frac{|\ln\sigma|}{\sqrt{\pi^2 + (\ln\sigma)^2}}$$

The deduction of these parameters is not complicated, but still is not the goal of this paper. With these parameters the reference model transfer function becomes Eq. (5).

$$H_m(s) = \frac{\left[\pi^2 + (\ln\sigma)^2\right] \cdot \sum_{i=1}^{n} K_i}{t_p^2 \cdot s^2 + 2 \cdot |\ln\sigma| \cdot t_p \cdot s + \pi^2 + (\ln\sigma)^2} \tag{5}$$

The resulting compensator model is calculated using Eq. (2), Eq. (3) and Eq. (5) and given in Eq. (6).

$$H_c(s) = \frac{\left[\pi^2 + (\ln\sigma)^2\right] \cdot \sum_{i=1}^{n} K_i}{\left[t_p^2 \cdot s^2 + 2 \cdot |\ln\sigma| \cdot t_p \cdot s + \pi^2 + (\ln\sigma)^2\right] \cdot \sum_{i=1}^{n} \frac{K_i \cdot \omega_{ni}^2}{s^2 + 2 \cdot \xi_i \cdot \omega_{ni} \cdot s + \omega_{ni}^2}} \tag{6}$$

Although it may seem complicated, this transfer function can easily be implemented in modern numerical control systems after suitable conversion.

The main difficulties in applying this method are caused by the fact that the original system's model cannot be identified with infinite precision. This leads to an approximate model that can introduce errors in the overall control implementation. Improving the behavior to parametric estimation errors is an important issue, but is not covered in this paper.

3. An application of the specified overshot and peak time vibrations suppression compensator for a 7-DOF system

Let's consider the system described by the modal parameters in Eq. (7).

$$\begin{aligned}
&K_1 = 0.35; f_{n1} = 3.1\,\mathrm{Hz}; \xi_1 = 0.12;\\
&K_2 = 0.21; f_{n2} = 4.6\,\mathrm{Hz}; \xi_2 = 0.08;\\
&K_3 = 0.19; f_{n3} = 8.0\,\mathrm{Hz}; \xi_3 = 0.12;\\
&K_4 = 0.10; f_{n4} = 11.5\,\mathrm{Hz}; \xi_4 = 0.07;\\
&K_5 = 0.09; f_{n5} = 13.4\,\mathrm{Hz}; \xi_5 = 0.03;\\
&K_6 = 0.03; f_{n6} = 20.6\,\mathrm{Hz}; \xi_6 = 0.09;\\
&K_7 = 0.03; f_{n7} = 29.7\,\mathrm{Hz}; \xi_7 = 0.02;
\end{aligned} \tag{7}$$

The step response of this system is shown in Fig. 2. Its percentage overshot is of 34.15%, the settling time of 1.344 seconds and the peak time is 0.119 seconds.

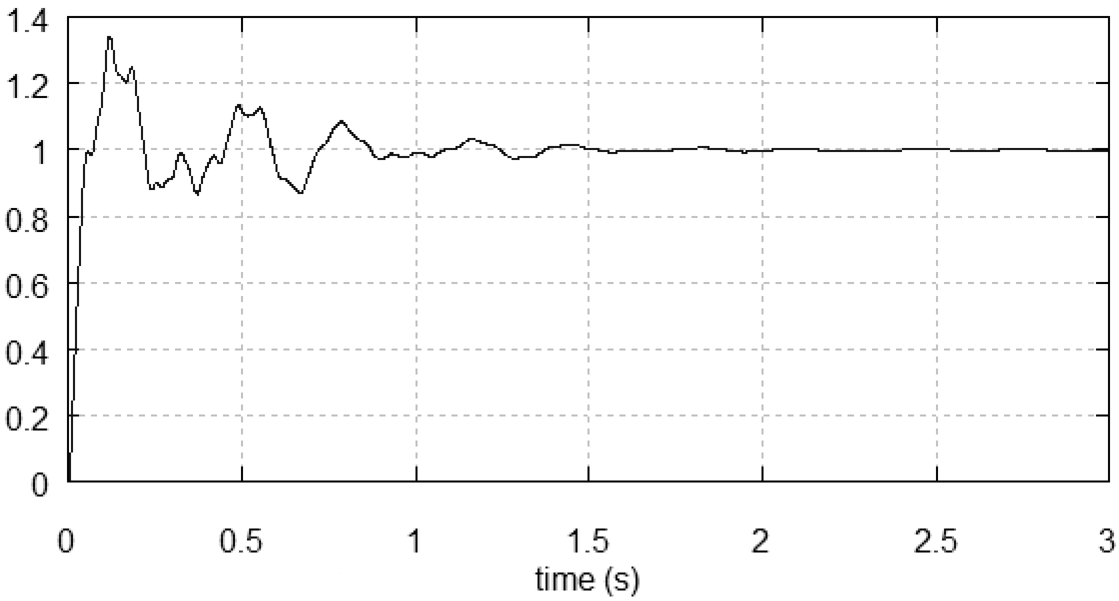

Figure 2. Non-compensated system's unit step response

The desired overshot is 10% and the peak time 80 ms. Therefore, the reference model parameters as calculated from Eq. (4) are shown in Eq. (8) and the reference model transfer function in Eq. (9).

$$\begin{aligned} K_m &= 1 \\ f_{nm} &= 7.749\,\mathrm{Hz} \\ \xi_m &= 0.591 \end{aligned} \tag{8}$$

$$H_m(s) = \frac{2370.5}{s^2 + 57.55 \cdot \omega_{ni} \cdot s + 2370.5} \tag{9}$$

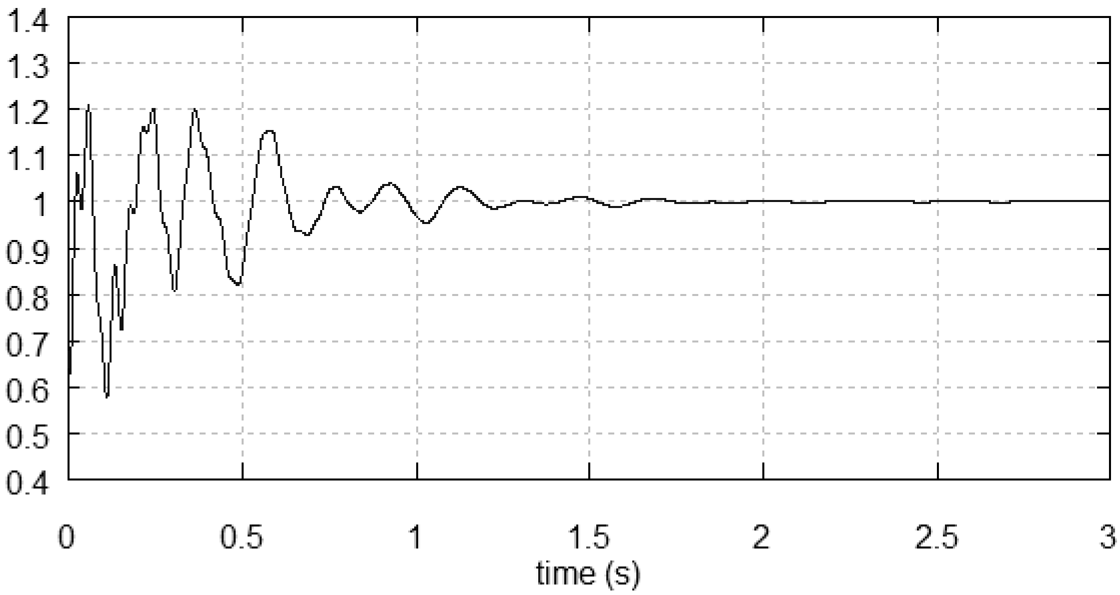

Figure 3. Vibrations suppression compensator's unit step response

The vibrations suppression compensator is obtained applying the design methodology described in the previous section. Its transfer function is not shown here due to the apparent complexity.

The compensator's and the compensated system's unit step responses are shown in Fig. 3 and Fig. 4, respectively.

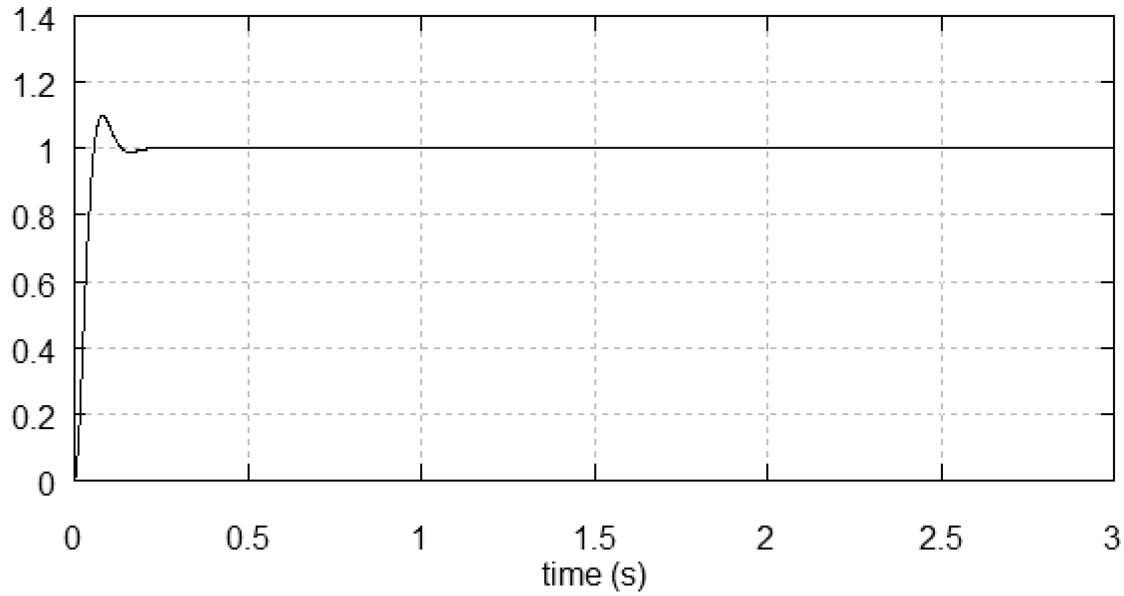

Figure 4. Compensated system's unit step response

As shown in Fig. 4, the response improvement is obvious. The system response is much faster and more accurate. The numerical values supporting this are the new overshot of 10%, as required, giving an improvement of 70.72%, and the peak time of 0.08 seconds, also as required, giving an improvement of 32.77%. Although not required, the settling time has also been dramatically improved having an absolute value of 0.122 seconds with an improvement of 90.92%.

All the numerical calculations and simulations have been made using the Open Source software GNU Octave 3 [12].

4. Conclusions

This paper presents a method of designing open-loop command preshaping compensators for vibrations suppression with specified overshot and peak time. The method requires a generic multimodal representation of the target system and a reference model with the specified performances. The compensator is obtained as a ratio between the reference model and the multimodal representation. This approach can be successfully applied in numerical control after suitable discrete conversion.

However, this method is sensitive to parameters estimation errors and future developments should focus on increasing the identification precision and system's robustness.

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