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REDUCING OPERATOR-INDUCED MACHINE VIBRATION USING A COMPLEX POLE/ZERO PREFILTER

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A systematic prefilter design process for reducing operator-induced rigid body vibrations of rubber tire mounted machines is presented. The contribution of this work is the development of a systematic prefilter design process and interpretation of the results. The class of heavy equipment considered in this work are those machines having rigid body main frame vibrations dominated by linear dynamics. The reduction in machine vibrations is accomplished through the design of prefilters that reduce the machine resonant frequencies from the operator commands. The machine information required for the design process includes the bandwidth of the electro-hydraulic (E/H) valves and rigid body resonant frequencies of the machine mainframe. The prespecified performance in the design process is the desired attenuation of machine resonant frequencies which is related to the acceptable level of machine vibration. The design methodology has been applied to a telescopic boom lift to illustrate the procedure and the effectiveness of the design.

1. INTRODUCTION

Many machines have rigid body main frame vibration modes which can be excited by operator inputs. Input shaping is a method of modifying operator commands to reduce machine vibration levels. There have been many investigations involving the use of input-shaping techniques to reduce vibration levels.

Looking back to the 1950s, Smith [1] proposed one of the early forms of input shaping called posicast control. This method can be applied to change the setpoint of a system. For this method to completely remove an oscillation, the system must be linear and the oscillation frequency should be known exactly. Therefore, this method of input shaping is sensitive to changes in the natural frequency of the system.

The sensitivity of the posicast control led many researchers to investigate more robust methods. Singer and Seering [2, 3] developed a time-domain input-shaping technique where a sequence of impulses is designed such that when convolved with a command input, the resulting output of the system has little to no vibration. Variations of the natural frequency of the system are addressed using three to four impulse sequences to convolve with the command input. This reduces the sensitivity to changes in natural frequency. The trade-off for the additional robustness and vibration reduction is increased response time.

Singh and Heppler [4] extended this technique to address systems with multiple vibration modes. Knowing the modal frequencies and damping ratios, these investigators present a robust input shaper using three impulses. Other methods for addressing systems with multiple vibration modes are presented by Tuttle and Seering [5]; and Singer *et al.* [6]. Evaluations of several time-domain input-shaping methods are presented by Pao [7] and Crain *et al.* [8]. Other methods of generating input shapers include using frequency sampling [9], vector diagrams [10], probability distribution of system parameters [11], and dynamic programming [12]. This is by no means a complete list.

In the early 1990s Singh and Vadali [13] studied the input-shaper design technique developed by Singer and Seering [3]. They determined that the input shaper was designed with zeros that cancelled system poles. Singh and Vadali [14] continued their work addressing input shaping as a pole–zero cancellation problem. The specification of system poles for the development of the prefilter was also explored by Magee and Book [15] where a finite impulse response filter was used. Robustness was added to the pole–zero cancellation approach to input shaping by locating multiple prefilter zeros at the system pole locations.

Lin [16] introduced a prefilter which resembles a notch filter. The design procedure is relatively easy when compared to the impulse sequence design methods of references [2] and [3]. The pole–zero placement design method only requires the frequency and damping ratio of the vibration mode to be attenuated. Lin incorporates the ability to use complex poles and zeros in the prefilter structure. To provide robustness into the design, multiple pole–zero sets are used for each vibration mode. This method may be used to suppress multiple vibration modes by adding more pole–zero sets to the prefilter.

As an extension of a pole–zero cancellation method of prefilter design, Bodson [17] developed an adaptive algorithm to add robustness to these prefilters. Another adaptive algorithm for robust prefiltering was implemented by Jansen [18]. The prefilter design used the pole–zero placement method with the system natural frequencies known *a priori*. The prefilter developed in this paper is called the robust notch filter and can be designed without knowing the exact pole locations of the system. Like Jansen [18], Kotnik *et al.* [19] also incorporated the use of acceleration data and fast Fourier transforms as an alternate method for identifying system poles and zeros. The prefilter, designed by Jansen, was created to remove the dominant natural frequencies of the system from the input command. Similar to the pole–zero cancellation methods previously described, additional system resonant frequencies may be suppressed by cascading pole–zero prefilters together.

Kwon *et al.* [20] applied the robust notch filter to structurally flexible, long-reach manipulators. One of the advantages of this filter was the ease of design. Similar to other prefilters, it presented the reduction of vibration at the cost of responsiveness.

Another pole–zero prefilter design approach was applied to boom cranes by Lewis *et al.* [21]. The prefilters were again designed to filter the natural frequencies of the system from the input commands. Parker *et al.* [22] then experimentally verified this prefilter design at the Navy Crane Testbed at Sandia National Laboratories. The prefilters were successful in reducing system vibration attributed to operator commands. Again, the prefilters reduce the vibration levels or oscillations in the system at the cost of responsiveness.

In summary, there are several input-shaping techniques available to effectively reduce vibration levels due to operator command inputs. The approach taken in this paper utilizes complex/real poles and zeros in the frequency domain. A systematic complex pole/zero prefilter design methodology is presented for the reduction of machine vibration. This frequency domain approach allows prefilters to be designed based on the frequencies which contribute to the overall vibration levels of the machine, i.e., machine natural frequencies. By focusing attention to the natural frequencies of the machine, the prefilter reduces machine vibration. Presented in this paper are the original complex pole/zero design equations, the simplified design equations, and a step-by-step process for the prefilter

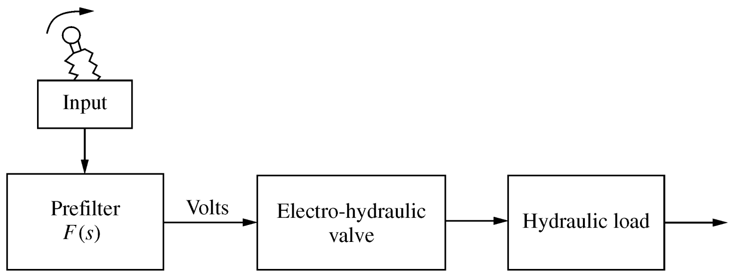

Figure 1. Prefilter configuration.

design. Finally, the design methodology will be applied to a telescopic boom lift to illustrate the procedure and the effectiveness of the design.

2. PROBLEM STATEMENT AND METHOD OF SOLUTION

The class of heavy equipment with E/H implements considered in this work are those machines which are susceptible to rigid body main frame vibrations due to the compressibility of their rubber tires. Such machines typically experience bounce and pitch rigid body vibration modes that are usually coupled. The initiation of these rigid body main frame vibrations are due to the operator commands which actuate the machine implements. The underlying assumption in this work is that the dynamics between the operator commands and E/H implements to the main frame vibrations are dominated by linear dynamics. The objective of this work is to reduce the levels of machine vibrations without reducing the responsiveness of the implements. Therefore, a qualitative balance between operator comfort and machine performance emerges. Consequently, rate and output saturation constraints on the electro-hydraulic valve(s) must be considered.

The method of solution advanced in this manuscript is a frequency-domain input-shaping approach which is accomplished through a prefilter. The prefilter is located between the operator commands and the E/H valve. The frequency response characteristics of the prefilter will attenuate those frequencies contained in the operator command that excite the machine bounce and pitch modes. A block diagram of the operator input, prefilter, valve, and load are shown in Figure 1.

3. MAIN RESULTS

Provided the dynamics between the operator command and the machine rigid body vibration is dominated by linear systems, input-shaping methods can reduce the vibration levels by modifying operator commands. This approach involves reducing the resonant frequencies of the machine vibration modes from the operator commands. This investigation utilizes the properties of complex poles and zeros to create a prefilter which can accomplish this task. The complex and real pole/zero prefilter is a type of notch filter which essentially attenuates the machine resonant frequencies from the operator commands. The desirable frequency response of the prefilter is based on measured vibrations from the machine.

The purpose of this investigation is to develop a systematic prefilter design process. The selected structure of the prefilter and the design of the prefilter parameters is presented in this section. The prefilter parameter design process will be segregated into three parts. The first part will describe the design for a single complex pole/zero pair that will reduce specific frequencies from an operator command. This signal complex pole/zero pair will match the natural frequency of the prefilter pole and zero. This prefilter will then be modified to include a real pole to avoid rate saturation of the E/H valve. Finally, the natural frequencies of the complex pole/zero pair for the third order prefilter will be mismatched to improve machine responsiveness to operator commands. Robustness issues will be discussed following this design section.

3.1. STRUCTURE OF THE PREFILTER

In general, the linear prefilter is comprised of complex poles and zeros. The parameters of the complex pole/zero pairs are designed to attenuate operator input frequencies which correspond to machine resonant frequencies. Cascading the complex pole/zero pairs creates the final prefilter. The resulting prefilter has the form

$$P_{TF}(s) = F_1(s) \cdot F_2(s) \cdots F_i(s) \cdot \left(\frac{1}{(s/\omega_{roll\text{-}off}) + 1} \right), \tag{1}$$

where

$$F_i(s) = \left(\frac{s^2/\omega_{zero_i}^2 + 2(\zeta_{zero_i}/\omega_{zero_i})s + 1}{s^2/\omega_{pole_i}^2 + 2(\zeta_{pole_i}/\omega_{pole_i})s + 1} \right), \tag{2}$$

ω_{zero_i} and ζ_{zero_i} are the natural frequency and damping ratio of the ith set of complex zeros; ω_{pole_i} and ζ_{pole_i} are the natural frequency and damping ratio of the ith set of complex poles, and $\omega_{roll\text{-}off}$ is the break frequency of the pole. The prefilter is designed to have unity gain at low frequencies to match the command input at steady state.

The parameters of the complex pole/zero prefilter shown in equation (2) are designed to attenuate a machine resonant frequency within an operator command. This leads to the specification of $\omega_{zero}$and ω_{pole}. The design of ζ_{zero} is based on the desired attenuation of machine resonant frequency within an operator command. The value of ζ_{zero} will be adjusted below $1/\sqrt{2}$ to achieve different levels of attenuation. The selection of ζ_{pole} essentially balances the trade-offs between desirable frequency response characteristics and transient time-domain characteristics. The damping ratio of the complex poles is set to $1/\sqrt{2}$ to allow the complex poles to roll off at a high frequency than any other damping ratio while not generating a resonant peak in the frequency domain. Finally, the break frequency of the pole is set to reject input noise and avoid rate saturation of the E/H valve. Note that $\omega_{zero}$and ω_{pole} may be independently selected. When ω_{zero} and ω_{pole} are not equal, this is called pole/zero frequency mismatching.

Essentially, each complex pole/zero pair in equation (2) is designed as a notch filter which attenuates machine resonant frequencies from the operator command. There are several advantages in using the complex pole/zero structure. First, the complex pole/zero prefilter is extremely responsive in the time domain. Second, each complex pole/zero pair contains only three adjustable parameters (ζ_{zero_i}, ω_{zero_i}, and ω_{pole_i}). Third, adjusting these parameters has an intuitive meaning. The smaller damping ratios of the complex zeros improve attenuation levels. The bandwidth of attenuation provides robustness to the prefilter to small variations of a machine resonant frequency. Another degree of freedom can be realized through pole/zero natural frequency mismatching and will be used when the E/H valve rate saturation is significant.

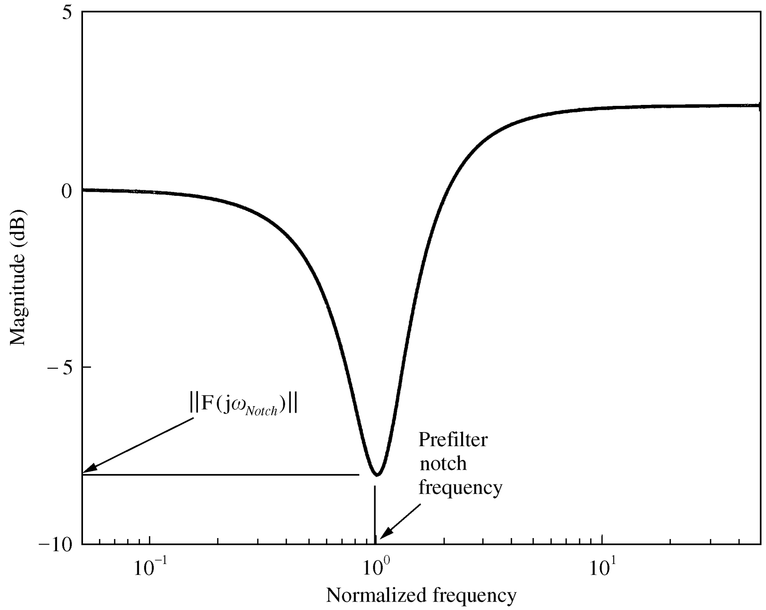

Figure 2. General frequency response function of $F_i(s)$.

The design of the complex pole/zero pairs with and without pole/zero natural frequency mismatching and a roll-off pole will be presented. This will provide the basic building blocks for robust prefilter design.

3.2. FREQUENCY RESPONSE CHARACTERISTICS OF $F_i(s)$

A general frequency response function for $F_i(s)$ is shown in Figure 2. The abscissa of the graph has been normalized with respect to the resonant frequency of the system. Changes in ζ_{zero} impact the attenuation level at the prefilter notch frequency. The prefilter notch frequency, ω_{Notch}, is defined as the frequency at which the prefilter complex pole/zero pair generates the greatest level of attenuation. The magnitude of the complex pole/zero pair at ω_{Notch}, $|F_i(j\omega_{Notch})|$, is defined as the maximum attenuation level achieved by the prefilter complex pole/zero pair.

To link the frequency response characteristics of $F_i(s)$ to its parameters, the relationship between the prefilter notch frequency and the parameters of the complex poles and zeros is developed. The solution is based on $|F_i(j\omega)|$ which gives

$$|F_i(j\omega)| = \sqrt{\frac{(\omega/\omega_{zero})^4 + (4\zeta_{zero}^2 - 2)(\omega/\omega_{zero})^2 + 1}{(\omega/\omega_{pole})^4 + 1}} \tag{3}$$

provided $\zeta_{pole} = 1/\sqrt{2}$. The value of ω_{Notch} is determined from

$$\omega_{Notch} = \left\{ \omega \left| \frac{\mathrm{d}}{\mathrm{d}\omega} |F_i(j\omega)| = 0 \right. \right\}. \tag{4}$$

Provided $\zeta_{pole} = 1/\sqrt{2}$ and $0 \leqslant \zeta_{zero} < 1/\sqrt{2}$, this solution is

$$\frac{\omega_{Notch}}{\omega_{pole}} = \sqrt{\frac{2(1-\alpha^4) - \sqrt{4(\alpha^4-1)^2 + \alpha^4(8\zeta_{zero}^2 - 4)^2}}{\alpha^2(8\zeta_{zero}^2 - 4)}}, \tag{5}$$

where $0 < \alpha \leqslant 1$ and

$$\alpha = \omega_{Zero}/\omega_{pole}. \tag{6}$$

Next, $|F_i(j\omega)|$ is developed for the case of $\zeta_{pole} = 1/\sqrt{2}$, $0 \leqslant \zeta_{zero} < 1/\sqrt{2}$, and $0 < \alpha \leqslant 1$. This closed-form solution is derived by substituting equation (5) into equation (3). This relationship is

$$|F_i(j\omega_{Notch})| = \sqrt{\frac{\begin{array}{c}[2(1-\alpha^4) - \sqrt{4(\alpha^4-1)^2 + \alpha^4(8\zeta_{zero}^2-4)^2}]^2 \\ + \alpha^4(4\zeta_{zero}^2 - 2)(8\zeta_{zero}^2 - 4)[2(1-\alpha^4) - \sqrt{4(\alpha^4-1)^2 + \alpha^4(8\zeta_{zero}^2-4)^2}] \\ + \alpha^8(8\zeta_{zero}^2-4)^2\end{array}}{\alpha^4\{[2(1-\alpha^4) - \sqrt{4(\alpha^4-1)^2 + \alpha^4(8\zeta_{zero}^2-4)^2}]^2 + \alpha^4(8\zeta_{zero}^2-4)^2\}}}. \tag{7}$$

Equations (5) and (7) provide a means to determine the complex pole/zero pair parameters (ω_{zero}, ω_{pole}, and ζ_{zero}) based on the performance specifications. However, the analytical relationships are complicated and will be simplified to streamline the design process.

3.3. DESIGN OF $F_i(s)$ WHEN $\omega_{zero} = \omega_{pole}$

The design of $F_i(s)$, equation (2), will be based on two primary pieces of machine information: rigid body main frame oscillation frequencies and performance (i.e., the desired attenuation of the resonant frequencies). These specifications will determine the three prefilter parameters (ζ_{zero_i}, ω_{zero_i}, and ω_{pole_i}).

The first step of the complex pole/zero pair design is to determine ω_{zero} and ω_{pole}. The following case is considered:

$$\alpha = 1 \Leftrightarrow \omega_{zero} = \omega_{pole}. \tag{8}$$

From equations (5) and (8),

$$\frac{\omega_{Notch}}{\omega_{pole}} = 1 \Leftrightarrow \omega_{Notch} = \omega_{zero} = \omega_{pole}. \tag{9}$$

Now ζ_{zero} must be calculated. Substituting equation (8) into equation (7) yields

$$\zeta_{zero} = |F_i(j\omega_{Notch})|/\sqrt{2}. \tag{10}$$

To summarize the design procedure for $F_i(s)$ in equation (2) when $\omega_{zero} = \omega_{pole}$, the rigid body main frame oscillation frequency (set equal to ω_{Notch}) and desired attenuation (set equal to $|F_i(j\omega_{Notch})|$) must be known. From equation (9), ω_{zero} and ω_{pole} are set equal to the prefilter notch frequency (i.e., rigid body main frame oscillation frequency). Next, equation

(10) provides ζ_{zero} given the desired attenuation of the rigid body main frame oscillation frequency. The next section presents a similar design procedure for $F_i(s)$ with a roll-off pole.

3.4. DESIGN OF $F_i(s)$ WITH A ROLL-OFF POLE WHEN $\omega_{zero} = \omega_{pole}$

A process has been developed in the previous section to design complex pole/zero pairs when $\omega_{zero} = \omega_{pole}$ in $F_i(s)$. These complex pole/zero pairs can be cascaded with a pole, where $\omega_{roll\text{-}off}$ is the bandwidth of the E/H valve. This roll-off pole will reject input noise and avoid rate saturation of the actuator. Consider a prefilter with one complex pole/zero pair and a roll-off pole. The transfer function of such a prefilter is

$$P(s) = F_1(s) \cdot \left(\frac{1}{s/(\omega_{roll\text{-}off}) + 1} \right), \tag{11}$$

where $P(s)$ is the prefilter transfer function, $F_1(s)$ is a complex pole/zero pair, and $\omega_{roll\text{-}off}$ is the break frequency of the pole.

If the break frequency of the roll-off pole is close to the notch frequency of a complex pole/zero pair, then the E/H implement performance will become unnecessarily sluggish to an operator. In such cases, pole/zero natural frequency mismatching can be used ($\omega_{zero} \neq \omega_{pole}$). The pole/zero natural frequency mismatching design is described in the next section. However, this section develops the criteria for determining if pole/zero natural frequency mismatching is necessary.

To quantify the allowable closeness of the roll-off pole to the complex pole/zero pair, a relationship is developed between $\omega_{roll\text{-}off}$ and $\omega_{Notch} = \omega_{zero} = \omega_{pole}$. The roll-off pole, $\omega_{roll\text{-}off}$, will be referred to as close to $\omega_{zero} = \omega_{pole}$ when

$$\|F_i(j\omega_B)\| < -0{\cdot}5 \text{ dB}, \tag{12}$$

where ω_B is the bandwidth frequency of the E/H valve. If $\omega_B = \omega_{roll\text{-}off}$ is close to $\omega_{zero} = \omega_{pole}$, the roll-off pole can create a sluggish response. Given $\alpha = 1$, a sufficient condition for equation (12) is

$$\omega_{roll\text{-}off} \geqslant 5 \cdot \omega_{Notch}. \tag{13}$$

If equation (13) is satisfied, then the roll-off pole is not close to $\omega_{zero} = \omega_{pole}$ and no natural frequency mismatching is required.

For the design of $P(s)$ with $\omega_{zero} = \omega_{pole}$, it is assumed that equation (13) is satisfied. This validates the use of equation (8) for this design. With equation (8) in place, the design procedure described in the previous section is valid of specifying ω_{zero}, ω_{pole}, and ζ_{zero}. Only one parameter of the prefilter, $P(s)$, remains to be determined, $\omega_{roll\text{-}off}$. The value of $\omega_{roll\text{-}off}$ is set equal to the bandwidth of the E/H valve.

3.5. DESIGN OF $F_i(s)$ WITH A ROLL-OFF POLE WHEN $\omega_{zero} \neq \omega_{pole}$

The final prefilter design that must be considered is the simple complex pole/zero pair with a roll-off and ω_B too close to ω_{Notch}. This prefilter will use the same structure as equation (11). For this design, the system violates equation (13) and the prefilter will be designed such that $\omega_{zero} \neq \omega_{pole}$. The design procedure described in the previous section is not valid for specifying ω_{zero}, ω_{pole}, and ζ_{zero} and may cause the E/H implement performance to be unnecessarily sluggish to the operator. However, $\omega_{roll\text{-}off}$ is still set equal to the bandwidth of the E/H valve.

To specify ω_{zero}, ω_{pole}, and ζ_{zero}, equations (5) and (7) provide two necessary relationships. Now the third relationship must be developed to introduce the system bandwidth limitation and uniquely determine the value of α. The value of α is determined by a relationship between ω_B, ζ_{zero}, ω_{zero}, and ω_{pole}. The relationship is derived based on the transfer function given in equation (11). Specifically, $|P(j\omega_B)| = 1$, where $\omega_{roll\text{-}off}$ is set equal to ω_B. This relationship is given by

$$\alpha = \left[\frac{1}{2} + \frac{\omega_{zero}^2(2\zeta_{zero}^2 - 1)}{\omega_B^2} - \frac{\omega_{zero}^4}{2\omega_B^4}\right]^{1/4} \tag{14}$$

provided $\zeta_{pole} = 1/\sqrt{2}$, $0 \leqslant \zeta_{zero} < 1/\sqrt{2}$, and $0 < \alpha \leqslant 1$.

The relationships given in equations (5), (7), and (14) will be used to produce a desirable set of simplified design equations. The systematic complex pole/zero prefilter methodology will use these simplified equations to determine the prefilter parameters given information about the bandwidth of the system, resonant frequencies, and acceptable machine vibration levels.

3.5.1. *Simplification of equations* (5) *and* (7) *where* $\omega_{zero} \neq \omega_{pole}$

The design of $F_i(s)$ when $\omega_{zero} \neq \omega_{pole}$ will be considered in this section. To develop a set of prefilter design tools, approximations of equations (5) and (7) will be made using regression analysis. The approximation of equation (5) will be based on a range of values for α and ζ_{zero}. Substituting $\alpha = \{1{\cdot}0\ 0{\cdot}8\ 0{\cdot}6\ 0{\cdot}4\}$ and $\zeta_{zero} = \{0{\cdot}01\ 0{\cdot}1\ 0{\cdot}2\ 0{\cdot}3\}$ into equation (5) yields

$$\omega_{pole} \approx \frac{\omega_{Notch}}{B}, \tag{15}$$

where B is tabulated in Table 1. Performing a least-squares polynomial fit of B tabulated in Table 1 gives the line

$$B = 1{\cdot}023\alpha - 0{\cdot}02726. \tag{16}$$

The exact values of B are plotted against the polynomial fit of equation (16) in Figure 3.

The approximation of ζ_{zero}, using equation (7), must also be derived for a range of α and ζ_{zero}. Substituting $\alpha = \{1{\cdot}0\ 0{\cdot}8\ 0{\cdot}6\ 0{\cdot}4\}$ and $\zeta_{zero} = \{0{\cdot}01\ 0{\cdot}1\ 0{\cdot}2\ 0{\cdot}3\}$ into equation (7) yields

$$|F_i(j\omega_{Notch})| = \gamma, \tag{17}$$

TABLE 1

Tabulation of B in equation (15)

	$\zeta_{zero} = 0{\cdot}01$	$\zeta_{zero} = 0{\cdot}1$	$\zeta_{zero} = 0{\cdot}2$	$\zeta_{zero} = 0{\cdot}3$
$\alpha = 1{\cdot}0$	1·000000	1·000000	1·000000	1·000000
$\alpha = 0{\cdot}8$	0·799966	0·796594	0·785674	0·764753
$\alpha = 0{\cdot}6$	0·599954	0·595329	0·580723	0·554217
$\alpha = 0{\cdot}4$	0·399962	0·396176	0·384407	0·363718

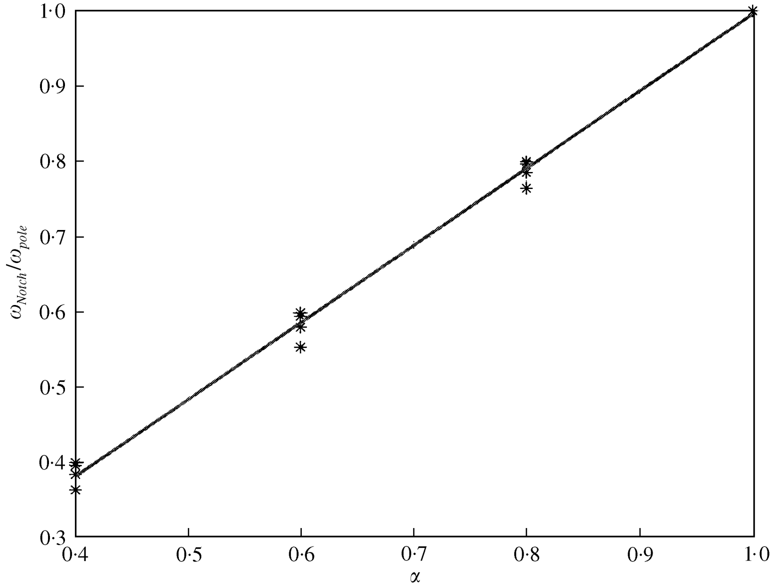

Figure 3. Least-squares polynomial fit of parameter B.

TABLE 2

Tabulation of γ in equation (17)

	$\zeta_{zero} = 0{\cdot}01$	$\zeta_{zero} = 0{\cdot}1$	$\zeta_{zero} = 0{\cdot}2$	$\zeta_{zero} = 0{\cdot}3$
$\alpha = 1{\cdot}0$	0·014142	0·141421	0·282843	0·424264
$\alpha = 0{\cdot}8$	0·016845	0·168304	0·335641	0·500664
$\alpha = 0{\cdot}6$	0·018817	0·187613	0·371709	0·548056
$\alpha = 0{\cdot}4$	0·019748	0·196593	0·387724	0·567460

where γ is tabulated in Table 2. Note that $\gamma/\zeta_{zero} = A$, where A is approximately constant for a given α. Performing a least squares polynomial fit of A as a function of α gives

$$A = -\,1{\cdot}141\alpha^2 + 0{\cdot}7134\alpha + 1{\cdot}8408. \tag{18}$$

The exact values of A are plotted against the polynomial fit of equation (18) in Figure 4. Therefore, equation (17) can be approximated as

$$\zeta_{zero} \approx \frac{|F_i(j\omega_{Notch})|}{A}. \tag{19}$$

Now equations (14)–(16), (18), and (19) must be solved simultaneously to determine the prefilter parameters (ζ_{zero_i}, ω_{zero_i}, and ω_{pole_i}). Substituting equations (15) and (19) into equation (14) yields the relationship

$$\omega_{zero}^4\left(\frac{B^4}{\omega_{Notch}^4} + \frac{1}{2\omega_B^4}\right) + \omega_{zero}^2\left[\frac{A - 2|F(j\omega_{Notch})|^2}{\omega_B^2 A^2}\right] - \frac{1}{2} = 0, \tag{20}$$

where A and B are defined in equations (16) and (18).

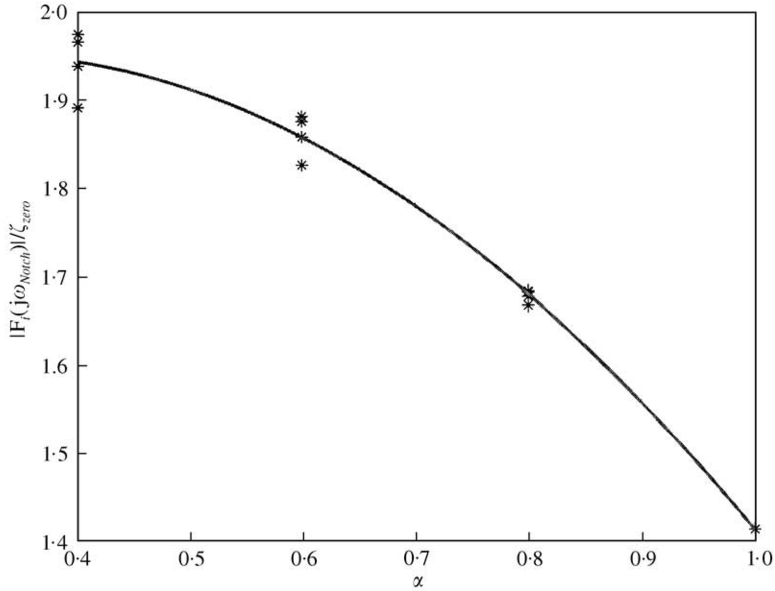

Figure 4. Least-squares polynomial fit of parameter A.

To design a single complex pole/zero pair with pole/zero natural frequency mismatching and a roll-off pole, the rigid body main frame oscillation frequency (set equal to ω_{Notch}), desired attenuation (set equal to $|F_i(j\omega_{Notch})|$), and E/H valve bandwidth (ω_B) must be known. First, set $\omega_{roll\text{-}off}$ equal to ω_B. Next, solve equations (14), (15), and (19) simultaneously to determine the prefilter parameters ($\zeta_{zero_i}, \omega_{zero_i}$, and ω_{pole_i}). Equation (20) provides a relationship between ω_{zero} and α. To determine ω_{zero} from this relationship, choose a value of α and plot the frequency response of the prefilter. The value of α may be varied, if necessary, to find the solution of equation (20) where the magnitude of the prefilter is equal to one at ω_B. Once ω_{zero} has been determined, ω_{pole} and ζ_{zero} may be determined from equations (15) and (19) respectively. This process may be automated.

3.6. ROBUST PREFILTER DESIGN

Now that the design of the prefilter is complete, issues concerning robustness and performance may be addressed. Multiple resonant frequencies or uncertain resonances may be addressed through cascading complex pole/zero pairs. Due to the approximations made during the design process, the prefilter performance must be verified and parameters adjusted if required. These issues are discussed in this section.

3.6.1. *Cascading complex pole/zero pairs*

To account for multiple machine resonant frequencies or to create a more robust prefilter, multiple complex pole/zero pairs can be cascaded together. An example of this is shown in equation (1). Each complex pole/zero pair can be designed to attenuate a machine resonant frequency or several complex pole/zero pairs may be designed to attenuate a broad band of machine resonant frequencies. This provides a method for designing a robust prefilter and addresses the uncertainty of machine resonant frequencies.

3.6.2. *Final parameters for the complex pole/zero prefilter design*

The final step in any design process is to verify that the design has met all the specifications. Calculating the frequency response of the prefilter may be used to validate the design. The validation stage is needed since many approximations were used to simplify the original closed-form relationships of equations (5), (7), and (14) into a set of design-friendly equations. Due to these approximations, the complex pole/zero prefilter parameters may require some adjustment given certain design specifications. The prefilter parameters may be adjusted manually using classical loop-shaping techniques.

4. APPLICATION

The reduction of telescopic boom lifts rigid-body vibration modes via complex pole/zero prefilter design is investigated in this section. A telescopic boom lift is a machine used to elevate an operator to a work site. For example, a telescopic boom lift may be used to elevate an operator near the top of a water tower or building.

4.1. MODEL DESCRIPTION

A simple model of a telescopic boom lift illustrates the effectiveness of the prefilter design methodology. This model is essentially a spring–mass–damper system with a hydraulically operated rigid boom (Figure 5). The telescopic boom can extend the operator platform approximately 24·4 m, while the hydraulic lift cylinder is used to raise and lower the boom.

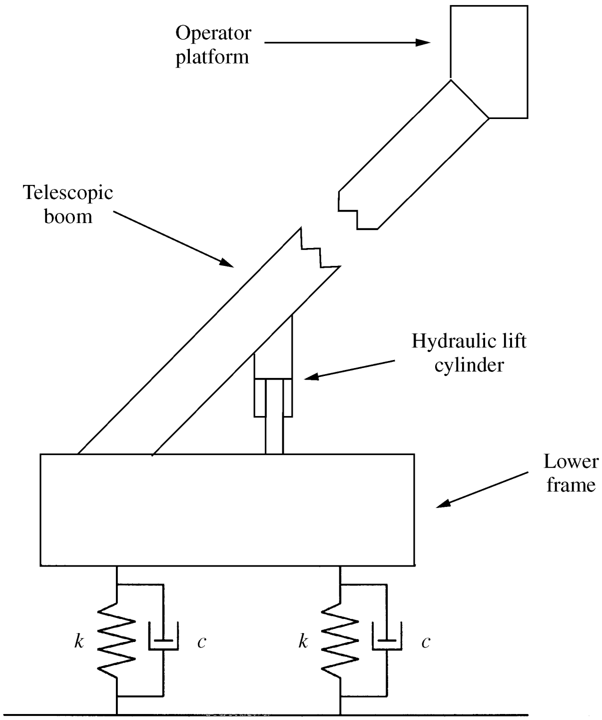

Figure 5. Telescopic boom lift model.

The suspension of the telescopic boom lift is provided by the dynamic tire properties. The damping and spring rates for the model used for this investigation were determined from the tire properties.

The mass of lower frame, telescopic boom, and operator platform are: 12 369·6, 1187·8, and 186·6 kg respectively. The total mass of the telescopic boom lift is 13 744 kg. The spring rate of the front and rear tires (k) is 3·117 MN/m. Furthermore, the damping rate of the front and rear tires (c) is 7·005 Ns/mm.

4.2. DETERMINATION OF TELESCOPIC BOOM LIFT PROPERTIES

The prefilter design only requires information about the bandwidth of the system, acceptable machine vibration levels, and machine resonant frequencies. The prefilter designed in this investigation removes the resonant frequencies from operator commands to the boom lift circuit. For the telescopic boom lift application, the bandwidth of the system is determined by the bandwidth of the hydraulic value used to actuate the boom. In this case, the bandwidth of the hydraulic lift cylinder valve given as 4·5 Hz.

Acceptable machine vibration levels must also be known. These vibration levels translate to the required amount of attenuation of the machine resonant frequencies by the prefilter. The level of attenuation is dependent on the machine and may require operator evaluative testing to determine. For this example application, acceptable machine vibration levels are achieved when the primary machine resonance is attenuated -20 dB. Note that a compromise exists between reduction in machine response and reduction in machine vibration levels.

Finally, the telescopic boom lift resonant frequencies must be determined. For this investigation, a model of a telescopic boom lift used to calculate the required operator acceleration information. To determine the most severe machine resonant frequencies, the acceleration of the operator platform is calculated for a series of step commands that excite the machine dynamics at various boom positions.

To determine the actual resonant frequencies of the telescopic boom lift, a discrete Fourier transform (DFT) of the modelled acceleration is implemented to determine the frequency content of the vehicle acceleration. The modelled operator acceleration is broken into segments, each containing one step of the boom. A DFT is performed on each segment to determine the contribution of each frequency to the overall operator acceleration levels. The segments are averaged together to obtain the frequency content of the vehicle acceleration. The peaks of the averaged DFTs occur at the resonant frequencies of the machine rigid body vibration modes. The greatest contribution to the vehicle acceleration levels is in the vertical direction at approximately 1·15 Hz. Since the fore/aft and vertical resonant frequencies agree, the prefilter will be designed to attenuate a resonant frequency of 1·15 Hz. Now that the machine resonant frequency is known, the prefilter design may take place.

4.3. COMPLEX POLE/ZERO PREFILTER DESIGN

A prefilter is now designed using complex poles and zeros to attenuate the machine resonant frequency from the operator boom lift command. Note that for this particular application equation (13) is not satisfied. Therefore, this design requires the natural frequencies of the complex pole/zero set to be mismatched. Following the design procedure for $F_i(s)$ with roll-off pole when $\omega_{zero} \neq \omega_{pole}$, equation (20) is solved using $\omega_{Notch} = 7{\cdot}2257$ rad/s, $\omega_B = 28{\cdot}2743$ rad/s, $|F(j\omega_{Notch})| = 0{\cdot}1$, and the value of A and B given

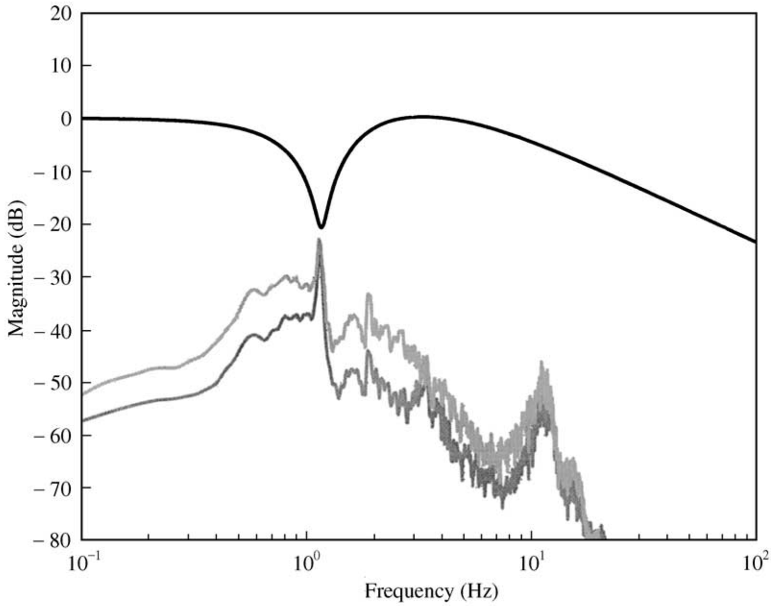

Figure 6. Frequency response function of final prefilter design (black) and average DFTs of operator fore/aft (dark gray) and vertical (light gray) acceleration.

by equations (18) and (16) respectively. The solution to equation (20) may be determined by iterating on the value of α until $\|P(j\omega_B)\| \approx 0$ dB. With the value of α known, equation (20) provides $\omega_{zero} = 8{\cdot}11$ rad/s. The values of ω_{pole} and ζ_{zero} are determined from equations (15) and (19) respectively. The frequency response of the prefilter is plotted to verify the design. At this point the actual prefilter notch frequency is slightly higher than desired. By slightly decreasing the notch frequency used for the design, the desired prefilter FRF is determined. The transfer function of the final prefilter designed for this application is given by equation (11), where $\omega_{zero} = 7{\cdot}33$ rad/s, $\zeta_{zero} = 0{\cdot}0573$, $\omega_{pole} = 8{\cdot}91$ rad/s, $\zeta_{pole} = 1/\sqrt{2}$, and $\omega_{roll\text{-}off} = 28{\cdot}27$ rad/s. Loop-shaping methods may have been implemented to determine the final prefilter design as well. Notice that $\alpha < 1$. This has the effect of providing less attenuation between the resonant frequency of the system and the break frequency of the pole. By providing less attenuation of the frequencies not contributing to the operator vibration levels, a faster overall response is achieved. The FRF of the prefilter as well as the averaged DFTs of the operator acceleration are shown in Figure 6.

4.4. RESULTS

Simulations were run with and without the boom lift circuit prefilter using lift cylinder step commands. The simulation results show a substantial reduction in vibration levels with the prefilter active. Figure 7 compares the fore/aft and vertical acceleration levels with and without the prefilter. Clearly, the fore/aft and vertical peak vibration levels are much lower as well as the residual vibrations or ringing of the system. This can be explained by the fact the resonant frequency in the force/aft and vertical case is the same. The prefilter was designed to attenuate this resonant frequency and therefore vibration reduction was accomplished in both the fore/aft and vertical directions.

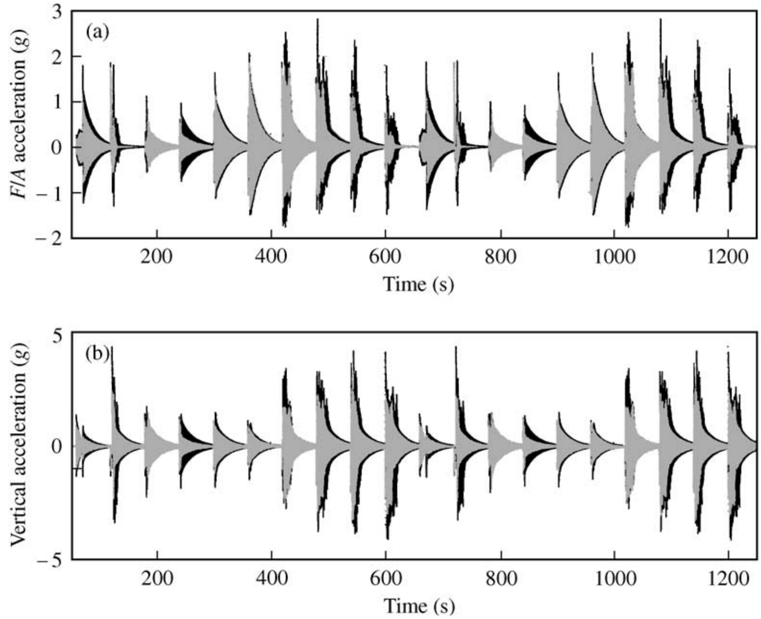

Figure 7. Acceleration, no prefilter (black), prefilter active (gray). (a) Fore/aft direction; (b) vertical direction.

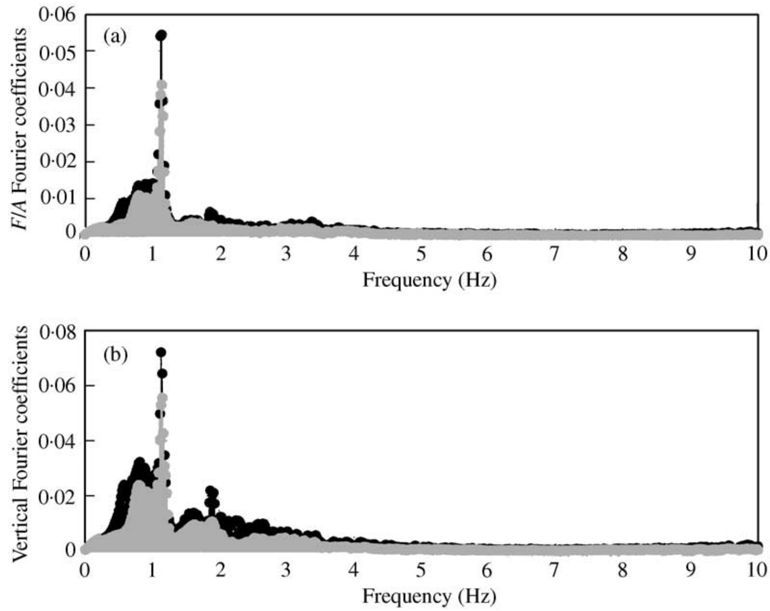

Figure 8. DFT of acceleration, no prefilter (black), prefilter active (gray). (a) Fore/aft direction; (b) vertical direction.

To further evaluate the prefilter, a DFT is performed on each segment of the modelled acceleration to determine the contributions of each frequency to the overall operator acceleration levels. Again, the segments are averaged together to obtain an overall picture of the frequency content contributing to the operator acceleration (Figure 8). Figure 8 clearly shows a substantial reduction in the contribution of the fore/aft and vertical resonant

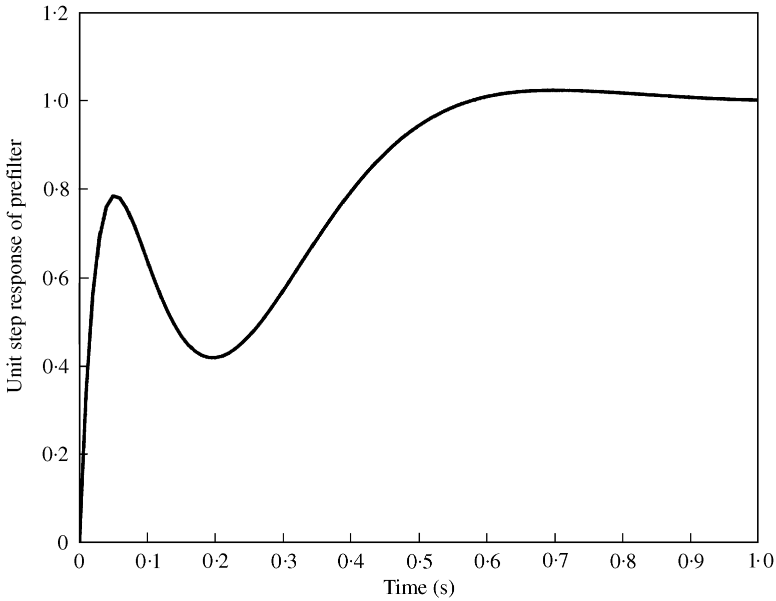

Figure 9. Unit step response of a complex pole/zero prefilter.

frequency to operator vibration levels. Once again this can be explained since the prefilter is designed to attenuate the same fore/aft and vertical resonant frequency.

The trade-off for the additional robustness and vibration reduction is increased response time. Figure 9 provides a unit step response of the complex pole/zero prefilter. Note the behavior of the unit step response at time equal to zero. The slope of the prefilter response is non-zero at this point. This is a property of complex poles and zeros which allows the prefilter to maintain a degree of responsiveness while reducing machine vibration.

If further reduction of vibration is required, the attenuation level at the prefilter notch frequency may be changed to provide more attenuation of the machine resonant frequency. Furthermore, if there existed another resonant frequency requiring attenuation, additional complex pole/zero sets can be designed to attenuate this additional resonant frequency and cascaded with the current prefilter.

5. CONCLUSIONS

The design of prefilters using complex poles and zeros is a successful method for the reduction of vibration levels. However, there is a compromise made between vibration reduction and machine responsiveness during the prefilter design process. The complex pole/zero prefilter design approach outlined in this investigation offers several advantages over other vibration reduction methods. First, this method only modifies operator commands and does not require any other machine modifications. Second, only knowledge of the bandwidth of the system, machine resonant frequencies, and desired attenuation of machine resonant frequencies are required for the prefilter design. Third, the prefilter design accommodates the reduction of any number of machine resonant frequencies and addresses issues of robustness with varying machine resonant frequencies.

Finally, this methodology has simplified the prefilter design process by making it systematic. Prefilter design is broken into three categories, each with a set of design-friendly equations linking performance specifications and machine constraints to prefilter design parameters. This presents an effective tool for handling the compromises involved between reduction in response and reduction in vibration levels.

REFERENCES

1. O. J. M. SMITH 1958 *Feedback Control Systems*. New York: McGraw-Hill.
2. N. C. SINGER and W. P. SEERING 1989 *Proceedings of the 1989 IEEE International Conference of Robotics and Automation*, Vol. 2, 883–893 Scottsdale, AZ, USA. Design and comparison of command shaping methods for controlling residual vibration.
3. N. C. SINGER and W. P. SEERING 1990 *ASME Journal of Dynamic Systems, Measurement, and Control* **112**, 76–82. Preshaping command inputs to reduce system vibration.
4. T. SINGH and G. R. HEPPLER 1993 *ASME Journal of Dynamic Systems, Measurement, and Control* **115**, 341–347. Shaped input control of a system with multiple modes.
5. T. D. TUTTLE and W. P. SEERING 1994 *Proceedings of the American Control Conference* **3**, 2533–2537. A zero-placement technique for designing shaped inputs to suppress multiple-mode vibration.
6. N. C. SINGER, L. Y. PAO, W. E. SINGHOSE and W. P. SEERING 1996 *Proceedings of the 1996 IEEE International Conference on Control Applications*, 373–378 Dearborn, MI, USA. An efficient algorithm for the generation of multiple-mode input shaping sequences.
7. L. Y. PAO 1997 *Journal of Guidance, Control, and Dynamics* **20**, 909–915. Analysis of the frequency, damping, and total insensitivities of input shaping designs.
8. E. A. CRAIN, W. E. SINGHOSE and W. P. SEERING 1996 *Japan/USA Symposium on Flexible Automation* **1**, 315–318. Evaluation of input shaping on configuration dependent systems.
9. N. C. SINGER and W. P. SEERING 1992 *Proceedings of the* 1992 *IEEE International Conference on Robotics and Automation*, Vol. **1**, 800–805. Nice, France. An extension of command shaping methods for controlling residual vibration using frequency sampling.
10. W. E. SINGHOSE, W. P. SEERING and N. C. SINGER 1990 *Proceedings of the 1990 IEEE International Conference on Robotics and Automation*, Vol. 2, 922–927 Cincinnati, OH, USA. Shaping inputs to reduce vibration: a vector diagram approach.
11. L. Y. PAO, T. N. CHANG and E. HOU 1997 *Proceedings of the American Control Conference* **6**, 3542–3546. Input shaper designs for minimizing the expected level of residual vibration in flexible structures.
12. G. G. PARKER, B. PETTERSON, C. DOHRMANN and R. D. ROBINETT 1995 *Proceedings of the American Control Conference* **1**, 934–938. Command shaping for residual vibration free crane maneuvers.
13. T. SINGH and S. R. VADALI 1993 *ASME Journal of Dynamic Systems, Measurement, and Control* **115**, 303–306. Robust time-delay control.
14. R. SINGH and S. R. VADALI 1993 *Journal of Guidance, Control, and Dynamics* **16**, 1061–1068. Input-shaped control of three-dimensional maneuvers of flexible spacecraft.
15. D. P. MAGEE and W. J. BOOK 1995 *Proceedings of the American Control Conference* **1**, 924–928. Filtering micro-manipulator wrist commands to prevent flexible base motion.
16. T.-C. LIN 1993 *Proceedings of the American Control Conference* **3**, 2502–2506. Design an input shaper to reduce operation-induced vibration.
17. M. BODSON 1997 *Proceedings of the American Control Conference* **3**, 1340–1344. An adaptive algorithm for the tuning of two input shaping methods.
18. J. F. JANSEN 1992 *ORNL/TM-12198, Oak Ridge National Laboratory*. Control and analysis of a single-link flexible beam with experimental verification.
19. P. T. KOTNIK, S. YURKOVICH and ÜMIT ÖZGÜNER 1988 *Journal of Robotic Systems* **5**(3), 181–196. Acceleration feedback for control of a flexible manipulator arm.
20. DONG-SOO KWON, DONG-HWAN HWANG, SCOTT M. BABCOCK and BARRY L. BURKS 1994 *Proceedings of the 1994 IEEE International Conference on Robotics and Automation*, **4**, 3259–3264. Input shaping filter methods for the control of structurally flexible, long-reach manipulators.

21. D. Lewis, G. G. Parker, B. Driessen and R. D. Robinett 1988 *Proceedings of the American Control Conference* **5**, 2643–2647. Command shaping control of an operator-in-the-loop boom crane.
22. G. G. Parker, K. Groom, J. E. Hurtado, J. Feddema, R. D. Robinett and Frank Leban 1999 *Proceedings of the American Control Conference* **1**, 86–90. Experimental verification of a command shaping boom crane control system.