Journal of Engineering Science and Technology
Vol. 18, No. 1 (2023) 604 - 623

REFERENCE RESPONSE BASED TIME-VARYING VIBRATION SUPPRESSION CONTROL FOR FLEXIBLE DYNAMIC SYSTEMS

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Abstract

Flexible dynamic systems such as cranes, flexible robot manipulators, space structures are widely used in many applications. However, because of their natural flexibility, vibrations appear during their works and may become remarkable that cannot be ignored. To improve the system performance of these flexible systems, vibration suppression control is considered. Input preshaping is a popular vibration control technique that is very useful for invariant time vibratory systems but may not be adequate for time-varying vibratory systems that are popular in flexible dynamic systems. In this paper, a reference response based vibration control technique is proposed for time-varying vibratory system. The input that can suppress the time-varying vibration is calculated from the selected reference response and known time-varying vibration parameters. The selected reference response is the response of a reference time-invariant vibratory system with a shaped input that is created using the conventional input shaping technique. The simulations and experiments with the 2D overhead crane system have shown that the proposed approach can suppress the time-varying vibration outperform the conventional input shaping techniques such as ZV and ZVD.

Keywords: Feedforward control, Flexible dynamic system, Input shaping, Time-varying vibratory system, Vary rope length, Vibration suppression control.

1. Introduction

Flexible dynamic systems such as crane systems, flexible robot manipulators, space structures, flexible aircraft wings, etc., play an important role in many application fields. Nowadays, they are a powerful option because of their lightness, energy efficiency, and suitable applications for human-machine interaction. However, when a flexible dynamic system moves fast, it results in remarkable vibrations during and at the end of motion. These vibrations reduce the flexible machine's life and decrease the system's accuracy and sensitivity.

Due to sensor noise and unmodelled dynamic problems, feedback control often has strong limitations. Open-loop control is more effective and wildly used for vibration suppression control of flexible machines. In general, open-loop vibration suppression controls for flexible dynamics systems can be divided into three categories, including command smoothing [1-6], time parameter tuning [7-9], and preprocessing filters [10-13]. Among them, input preshaping (IS) [10] is one of the most attractive vibration suppression methods because it can offer efficient vibration suppression with a short time delay if accurate estimations of the natural frequency and damping are available for the controlled vibratory system. Various improvements and applications of input preshaping have been made; see [14] for more information.

Although most of above-mentioned methods, especially input preshaping are known to be very efficient and robust to modeling errors, their applications are limited to linear time-invariant vibratory systems or slowly varying systems. However, time-varying dynamic systems are frequently founded, and the previous vibration suppression control methods might be impossible to suppress the vibration. For this reason, time-varying input shaping is considered [15-17]. In [15], the magnitude and the time of preshaping impulses are adjusted according to the time-varying vibration frequency. In addition, the extension of the impulse shaping concept to linear time-varying dynamic systems has been proposed in [16]. Parameterization of pole-zero cancellation for linear time-varying systems was developed for vibration suppression of time-varying system [17]. Various improvements and applications have been made based on these approaches for flexible systems [18-21]. However, these methods have at least one of disadvantages, such as much computation effort is required; applications are limited with repetitive manoeuvres; and steady-state position may not be precise.

In this paper, a reference response based input shaping for vibration suppression control of linear time-varying vibratory systems is proposed. At first, the desired reference response without vibration is generated from the system input and reference time-invariant vibratory system using input preshaping method [10]. Then, the desired input is calculated with the assumption that the response of the linear time-varying vibratory system is the same as the reference response. To guarantee the precision of the steady-state position and the input constraints, the selection of reference time-invariant vibratory system is carefully evaluated. The robustness of the proposed approach is also considered.

The rest of the paper is organized as follows. The introduction to the conventional input preshaping method is shown in section 2. In section 3, the proposed reference response based input preshaping approach is described. The selection of reference model parameters is also analysed in this section. The simulation of the proposed control technique for vibration suppression control of a

2D overhead crane is presented in section 4. The robustness of the proposed approach is also considered in this section. Experiments with a 2D overhead crane model are done in section 5 to verify the effectiveness of the proposed approach. Section 6 presents conclusions and further studies.

2. Input Preshaping Method

Input preshaping (IS) [10] is a feed-forward control technique that defines a sequence of impulses with appropriate magnitude and time. The shaped command obtained from the convolution of the reference command and the impulse sequence is then used to drive the system and eliminate the vibration. To easily understand this technique, let's consider a vibratory system that can be modelled as a second-order system with the transfer function:

$$\frac{Y(s)}{U(s)} = G(s) = \frac{K\omega_n^2}{s^2+2\xi\omega_n s+\omega_n^2} \quad (1)$$

where $U(s)$ is input, $Y(s)$ is vibratory output, K is vibration gain, ξ is the damping ratio and ω_n is the natural vibration frequency.

At time t_1 if an impulse with amplitude A_1 is input to the system, a vibration appears. To suppress the vibration, at time t_2 the second impulse with amplitude A_2 is input to the system. The second impulse also generates the vibration in the system. However, if A_2 and t_2 are chosen appropriately, the vibration caused by the second impulse can eliminate the vibration caused by the first impulse, as shown in Fig. 1.

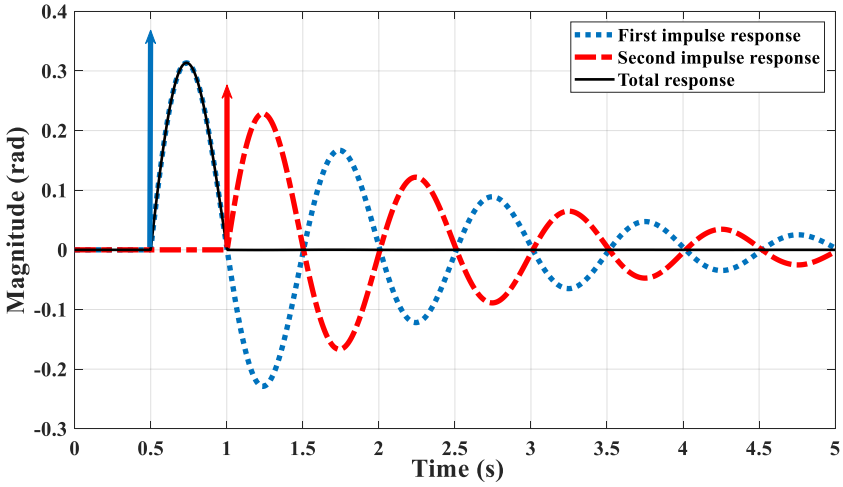

Fig. 1. Conventional input preshaping concept.

The input preshaping method shows that to suppress the vibration, the time and magnitude of two impulses are calculated as

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

where $K = e^{-\frac{\xi\pi}{\sqrt{1-\xi^2}}}$ and $T_d = \frac{\pi}{\omega_n\sqrt{1-\xi^2}}$.

The above configuration is called ZV (Zero Vibration) shaper. ZV shaper is sensitive to model error. In order to increase the robustness of IS method, additional impulses are added to the system. The greater number of added impulses, the more robust the system is. For example, with three pulses, ZVD (Zero Vibration Derivative) configuration is obtained:

$$\begin{bmatrix} A_j \\ t_j \end{bmatrix} = \begin{bmatrix} \frac{1}{(1+K)^2} & \frac{K}{(1+K)^2} & \frac{K^2}{(1+K)^2} \\ 0 & 0.5T_d & T_d \end{bmatrix} \tag{3}$$

In the case of four pulses, ZVDD (Zero Vibration Derivative Derivative) configuration is obtained:

$$\begin{bmatrix} A_j \\ t_j \end{bmatrix} = \begin{bmatrix} \frac{1}{(1+K)^3} & \frac{3K}{(1+K)^3} & \frac{3K^2}{(1+K)^3} & \frac{K^3}{(1+K)^3} \\ 0 & 0.5T_d & T_d & 1.5T_d \end{bmatrix} \tag{4}$$

The conventional input preshaping method is very effective for time-invariant vibratory systems. The ZVD and ZVDD configurations are robust with a small damping ratio and vibration frequency variation. However, they are not adequate for time-varying vibratory system. There have been some modifications of input preshaping such as K and are calculated using the same formulas with IS, but the frequency is time-varying and is updated online [15].

This approach is simple, but the steady-state position may not be the same as the reference. Another approach recalculated the impulse sequence for a time-varying vibratory system with the same idea as IS [16]. The steady-state position is guaranteed to be the same as the reference, but it requires much computation that may not be applicable in practice. In the next section, a vibration suppression control for time-varying vibratory system with simple calculation is developed, and the steady-state position is guaranteed to be the same as the reference.

3. Reference Response Based Input Preshaping Approach

3.1. Time-varying vibration suppression input generation

Let's consider the time-varying vibration system with the following transfer function:

$$\frac{Y_t(s)}{U_t(s)} = G_t(s) = \frac{K_t\,\omega_t^2}{s^2+2\xi_t\omega_t s+\omega_t^2} \tag{5}$$

where K_t is vibration gain, ξ_t is damping coefficient, and ω_t is natural vibration frequency. They are time-varying, but it is assumed that they are known beforehand. The purpose of the proposed approach is to choose a reference time-invariant vibration system that we can apply the conventional input preshaping technique to create a reference response with vibration suppression from the reference input. Next, the modified input to suppress the vibration of the time-varying system is calculated from the reference response, shaped reference input, and time-varying vibratory model's parameters. Figure 2 describes the diagram of the proposed approach.

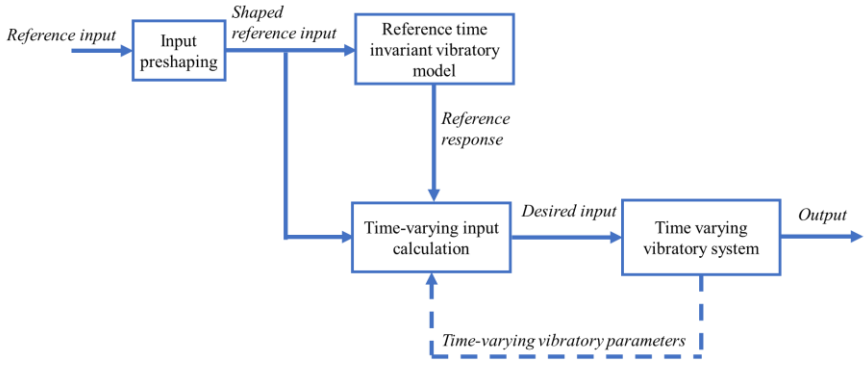

Fig. 2. Time-varying vibration suppression control input generation diagram.

In this paper, for simplicity, the reference time-invariant vibratory system is chosen as a second-order system with the transfer function as

$$\frac{Y_0(s)}{U_0(s)} = G_0(s) = \frac{K_0\,\omega_0^2}{s^2+2\xi_0\omega_0 s+\omega_0^2} \tag{6}$$

Rewrite the reference system under state-space model with the form

$$\dot{X}_0(t) = A_0X_0(t) + B_0u_0(t) \tag{7}$$

where $X_0(t) = [x_{01}; x_{02}]^T = [y_0; \dot{y}_0]^T$; $A_0 = \begin{bmatrix} 0 & 1 \\ -\omega_0^2 & -2\xi_0\omega_0 \end{bmatrix}$; $B_0 = \begin{bmatrix} 0 \\ K_0\omega_0^2 \end{bmatrix}$.

Similarly, the state-space model of the time-varying vibratory system is

$$\dot{X}_t(t) = A_t(t)X_t(t) + B_t(t)u_t(t) \tag{8}$$

where $X_t(t) = [x_{t1}; x_{t2}]^T = [y_t; \dot{y}_t]^T$; $A_t = \begin{bmatrix} 0 & 1 \\ -\omega_t^2 & -2\xi_t\omega_t \end{bmatrix}$; $B_t = \begin{bmatrix} 0 \\ K_t\omega_t^2 \end{bmatrix}$.

If $u_0(t)$ is shaped reference input that can suppress the vibration of the reference model using the conventional IS technique for the time-invariant vibratory system, then $y_0(t)$ will be the reference response of the time-varying system. By setting the response of the time-varying system is the same as the reference response we have

$$X_t(t) = X_0(t) \tag{9}$$

$$\dot{X}_t(t) = \dot{X}_0(t) \tag{10}$$

Substitute Eqs. (7) and (8) into Eq. (10), then Eq. (10) becomes

$$A_t(t)X_t(t) + B_t(t)u_t(t) = A_0X_0(t) + B_0u_0(t) \tag{11}$$

It is noted that $X_t(t) = X_0(t)$, then Eq. (11) can be rewritten more clearly as

$$\begin{bmatrix} 0 & 1 \\ -\omega_t^2 & -2\xi_t\omega_t \end{bmatrix}\begin{bmatrix} x_{01} \\ x_{02} \end{bmatrix} + \begin{bmatrix} 0 \\ K_t\omega_t^2 \end{bmatrix}u_t(t) = \begin{bmatrix} 0 & 1 \\ -\omega_0^2 & -2\xi_0\omega_0 \end{bmatrix}\begin{bmatrix} x_{01} \\ x_{02} \end{bmatrix} + \begin{bmatrix} 0 \\ K_0\omega_0^2 \end{bmatrix}u_0(t) \tag{12}$$

or

$$-\omega_t^2 x_{01} - 2\xi_t\omega_t x_{02} + K_t\omega_t^2 u_t(t) = -\omega_0^2 x_{01} - 2\xi_0\omega_0 x_{02} + K_0\omega_0^2 u_0(t) \quad (13)$$

Then, the input $u_t(t)$ can be calculated as

$$u_t(t) = \frac{1}{K_t\omega_t^2}[(\omega_t^2 - \omega_0^2)x_{01} + 2(\xi_t\omega_t - \xi_0\omega_0)x_{02} + K_0\omega_0^2 u_0(t)] \quad (14)$$

Replace $x_{01} = y_0$ and $x_{02} = \dot{y}_0$ into Eq. (14), we obtain the desired input that can sup-press the vibration of the time-varying vibratory system is determined as

$$u_t(t) = \frac{1}{K_t\omega_t^2}[(\omega_t^2 - \omega_0^2)y_0 + 2(\xi_t\omega_t - \xi_0\omega_0)\dot{y}_0 + K_0\omega_0^2 u_0(t)] \quad (15)$$

3.2. Selection of reference time-invariant model

As shown in the previous section, the generated input strongly depends on selecting the reference response or reference model. The response of the time-varying system is as same as the reference response, i.e., it depends on the choice of reference gain, reference natural frequency and reference damping factor. However, the input constraint must be guaranteed. In addition, the generated input has also ensured the similarity between the steady-state position of the desired motion and the reference motion. The damping factor of a time-varying system is usually time-invariant. So, we can choose the reference model's damping factor as the same as the time-varying model. In this section we consider the choosing of model's gain and natural frequency.

3.2.1. Selection of vibration gain model

Similar to the conventional input shaping technique, the proposed approach not only has to eliminate the vibration but also controls the steady-state position to the desired position. For example, in a crane system, it is required to bring the trolley to the desired position while suppressing the payload oscillation. In this case, and in many mechanical systems, the input is the force or acceleration of the motion. Therefore, to obtain the steady-state position requirement, we have

$$\int_0^\infty \int_0^\infty u_0(t)dtdt = \int_0^\infty \int_0^\infty u_t(t)dtdt = L \quad (16)$$

Taking Laplace transform of Eq. (16) we have

$$\lim_{s\to 0}(s^2 U_0(s)) = \lim_{s\to 0}(s^2 U_t(s)) = L \quad (17)$$

Then

$$\lim_{s\to 0}\frac{s^2 U_0(s)}{s^2 U_t(s)} = \lim_{s\to 0}\frac{U_0(s)}{U_t(s)} = 1 \quad (18)$$

Because $y_t(t) = y_0(t)$ then $Y_t(s) = Y_0(s)$. Therefore

$$\frac{U_0(s)}{U_t(s)} = \frac{Y_0(s)/G_0(s)}{Y_t(s)/G_t(s)} = \frac{G_t(s)}{G_0(s)} \quad (19)$$

Then

$$\lim_{s\to 0}\frac{U_0(s)}{U_t(s)} = \lim_{s\to 0}\frac{G_t(s)}{G_0(s)} = \frac{\frac{K_t\,\omega_t^2}{s^2+2\xi_t\omega_t s+\omega_t^2}}{\frac{K_0\,\omega_0^2}{s^2+2\xi_0\omega_0 s+\omega_0^2}} = \frac{K_t}{K_0} = 1 \quad (20)$$

Thus, the condition for the steady-state position is the same as the reference position is

$$K_0 = K_t \tag{21}$$

It means that the gain coefficient of the reference model must be the same as the one of the time-varying system.

3.2.2. Selection of damping factor and natural vibration frequency

The reference model's natural vibration frequency affects the system response time and the input physical limitation. The higher the frequency, the faster the system response is. However, the high frequency may cause input limit violation. Therefore, it should be chosen as high as possible while guaranteeing the input limit.

According to the proposed control law Eq. (15), the desired input depends on the reference input. To evaluate the effect of the reference natural vibration frequency on the input limit, we choose the unit step signal as the standard reference input for evaluation. Then the desired input must be limited by 1.

With the reference input being the unit step, the shaped input is a sequence of steps with time t_i and magnitude A_i. Let's consider the reference output with the input is a single step at time t_i and magnitude A_i, $u_{0i}(t) = A_i.1(t - t_i)$. The output response of the reference model can be expressed as

$$Y_{0i}(s) = U_{0i}(s)G_0(s) = \frac{A_i e^{-t_i s}}{s}\frac{K_0\,\omega_0^2}{s^2+2\xi_0\omega_0 s+\omega_0^2} \tag{22}$$

In the time domain, the output can be calculated as

$$y_{0i}(t) = A_i K_0\left(1 - \frac{e^{-\xi_0\omega_0(t-t_i)}}{\sqrt{1-\xi_0^2}}\sin\left(\omega_0\sqrt{1-\xi_0^2}\,(t-t_i)+\phi\right)\right) \tag{23}$$

where $\phi = \cos^{-1}\xi_0$.

The output derivative is calculated as

$$\dot{y}_{0i}(t) = \frac{A_i K_0 e^{-\xi_0\omega_0(t-t_i)}}{\sqrt{1-\xi_0^2}}\sin\left(\omega_0\sqrt{1-\xi_0^2}\,(t-t_i)\right) \tag{24}$$

Finally, the desired input can be calculated as

$$u_t(t) = \frac{1}{K_t\omega_t^2}\left[(\omega_t^2-\omega_0^2)\sum_{i=1}^{N} y_{0i}(t) + 2(\xi_t\omega_t-\xi_0\omega_0)\sum_{i=1}^{N}\dot{y}_{0i}(t) + K_0\omega_0^2\sum_{i=1}^{N}u_{0i}(t)\right] \tag{25}$$

Substitute Eqs. (23) and (24) into Eq. (25), then Eq. (25) becomes

$$u_t(t) = \frac{1}{K_t\omega_t^2}\left[(\omega_t^2-\omega_0^2)\sum_{i=1}^{N} A_i K_0\left(1 - \frac{e^{-\xi_0\omega_0(t-t_i)}}{\sqrt{1-\xi_0^2}}\sin\left(\omega_0\sqrt{1-\xi_0^2}\,(t-t_i)+\phi\right)\right) + 2(\xi_t\omega_t-\xi_0\omega_0)\sum_{i=1}^{N}\frac{A_i K_0 e^{-\xi_0\omega_0(t-t_i)}}{\sqrt{1-\xi_0^2}}\sin\left(\omega_0\sqrt{1-\xi_0^2}\,(t-t_i)\right) + K_0\omega_0^2\sum_{i=1}^{N}A_0.1(t-t_i)\right] \tag{26}$$

where N is the number of input shaper impulses, N=2 with ZV, N=3 with ZVD, and N=4 with ZVDD.

Using Eq. (26), one can easily calculate $u_t(t)$ corresponding to each value of ω_0 with known time-varying model parameters $\omega_t(t)$, K_t and ξ_t. Then we can choose the appropriate (maximum) value ω_0 so that $u_t(t)$ satisfies the limit requirement by numerical method. Figure 3 shows the maximum value of the proposed control input with various values of ω_0 in case the reference input magnitude is 10 V, and the model natural frequency $\omega_t(t)$ is a linear function of time and varies from 3 rad/s to 9 rad/s. The maximum input value increases when the reference natural vibration frequency increases. With the condition the maximum input value is limited by reference input, i.e., 10 V, the reference natural vibration frequency should be chosen as 6 rad/s. The choice guarantees the input limit while bringing the fast response.

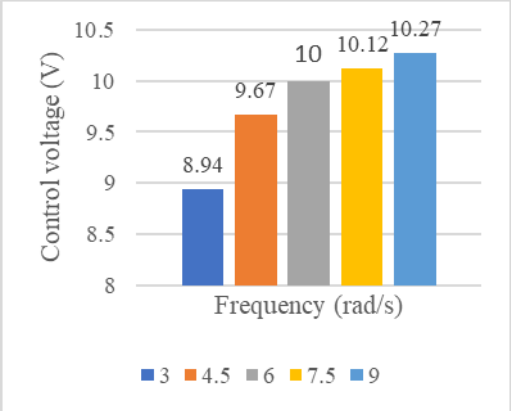

Fig. 3. Maximum control voltage corresponding to various values of reference nature vibration frequency.

4. Vibration Suppression Control of Vary Rope Length Overhead Crane

To validate the effectiveness of the proposed input preshaping approach, in this section, we consider applying the proposed control algorithm to vibration suppression control of an overhead crane with vary rope length. The following assumptions are made for the overhead crane

- The payload and trolley are connected by a massless and rigid link.
- The trolley moves along X-axis only.
- The rope length can be measured.
- The trolley and payload are considered as point masses and are known.
- The effects of disturbances are neglected.

4.1. Dynamical model of 2D overhead crane

Figure 4 demonstrates the model of the overhead crane. In this figure, F_x is the applied force to the trolley, f_{cx} is the friction force of trolley moving along x axis, and $f_{c\theta}$ is the friction force of the payload rotating around the trolley. Other parameters are described in Table 1.

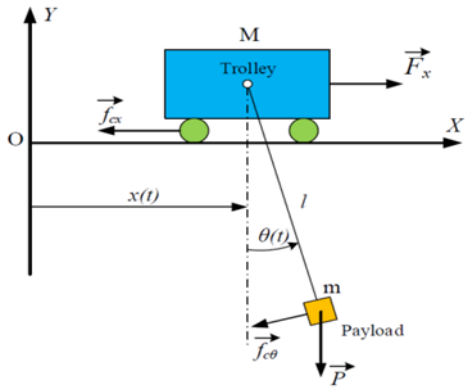

Fig. 4. The 2D overhead crane model.

Table 1. System parameter descriptions.

Symbol	Description
x(t)	Trolley position over time along X-axis (m)
θ(t)	Sway angle over time (rad)
l	Length of rope (m)
M	Mass of trolley (kg)
m	Mass of payload (kg)
g	Acceleration of gravity (=9.81 m/s^2)
B_{eq}	Equivalent viscous damping of trolley transition (Ns/m)
B_p	Equivalent viscous damping of payload rotation (Ns/rad)

Following the Euler - Lagrange equation, the dynamical model of the overhead crane is derived as follows [21]:

$$F_x - B_{eq}\dot{x} = (M+m)\ddot{x} + ml\ddot{\theta}cos\theta - ml\theta\dot{\theta}^2 sin\theta \quad (27)$$

$$-B_q\dot{\theta} = l\ddot{\theta} + \ddot{x}cos\theta + gsin\theta \quad (28)$$

Assuming that *θ(t)* is small, then $cos\theta \approx 1$; $sin\theta \approx \theta$. The dynamical model of the overhead crane is approximated by the following equations:

$$(M+m)\ddot{x} + B_{eq}\dot{x} + ml\ddot{\theta} - ml\dot{\theta}^2\theta = F_x \quad (29)$$

$$-x = l\ddot{\theta} + B_q\dot{\theta} + g\theta \quad (30)$$

Equation (30) shows that the overhead crane is a vibratory system with a natural vibration frequency is $\sqrt{g/l}$. The natural vibration frequency is a function of the rope length *l* that may be changed during crane operation. In the next subsection, the proposed input preshaping technique to suppress the payload time-varying frequency vibration is implemented via Matlab simulations and experiments.

4.2. Simulations

4.2.1. Simulation model

In practice, the moving of the trolley is done using a motor with a motor driver that can control the speed precisely, even with disturbance. Therefore, it can be assumed

that the velocity of the trolley can be easily controlled by the input voltage, and the transfer function of the trolley position with a control voltage input is:

$$\frac{X(s)}{U(s)} = \frac{K_m}{s(T_m s+1)} \tag{31}$$

To control the trolley to the desired position, a PD-type controller is chosen as:

$$F_{PD}(s) = \frac{U(s)}{E(s)} = K_P + K_D \frac{N_F}{1+\frac{N_F}{s}} \tag{32}$$

The transfer function of the vibratory system with the input is trolley acceleration, and the output is the payload sway angle can be obtained from Eq. (30) as follows:

$$\frac{Y_t(s)}{U_t(s)} = \frac{K_t\,\omega_t^2}{s^2+2\xi_t\omega_t s+\omega_t^2} \tag{33}$$

where $K_t = \frac{1}{g}$; $\xi_t = \frac{B_p}{2\sqrt{gl}}$; $\omega_t = \sqrt{g/l}$.

The control principal block diagram of the system is described in Fig. 5. The simulation parameters are as in Table 2. Based on the time-varying vibratory model, the reference model must have the parameters $K_0 = K_t = 1/g$ and $\xi_0 = \xi_t = 0$ to suppress the payload's swing. To guarantee the control voltage is limited to the motor driver's maximum control voltage input, the reference natural frequency is chosen as ω_0= 3 rad/s.

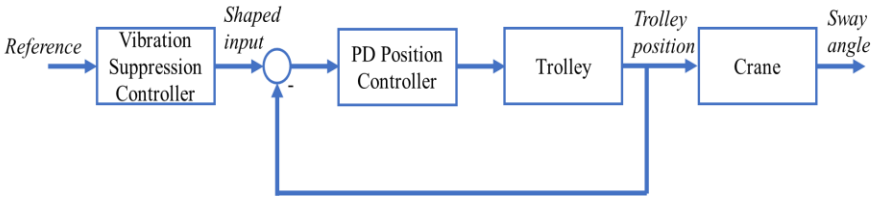

Fig. 5. Control principal block diagram of the crane system.

Table 2. Simulation parameters.

Symbol	Value
K_m	0.25 (m/sV)
T_m	0.1 (s)
K_p	1.2 (V/rad)
K_d	0.044 (V/rad)
N_F	221.6
g	9.81 (m/s^2)
B_p	0 (Ns/rad)
ω_t	3 + 0.3t (rad/s)

In the simulation, the reference vibration response is the response of the reference vibratory model, with the input being the shaped input created from the reference input using the ZV technique. We can also use other conventional input preshaping techniques to create shaped input for reference response.

4.2.2. Simulation results

The simulation results are shown in Fig. 6 for the trolley's position, Fig. 7 for sway angle, and Fig. 8 for controller output voltage. It is found that the trolly can get the desired position. There is a delay between the reference trajectory and the actual trajectory. It is because of the nature of the input preshaping method. The vibration is suppressed after the trolley's acceleration. The controller output is within the working limit (± 5V).

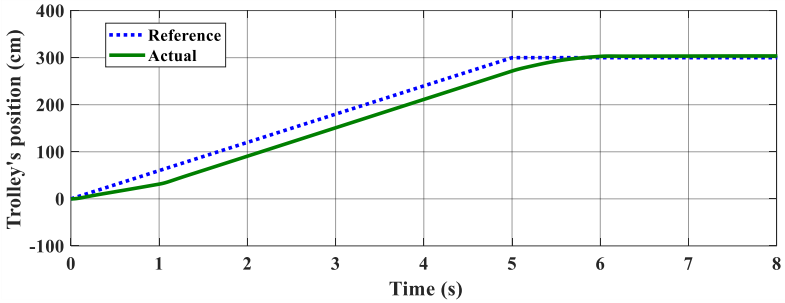

Fig. 6. Trolley's position response.

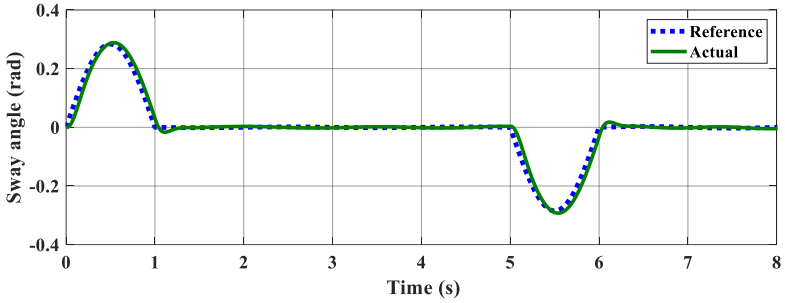

Fig. 7. Load's sway angle.

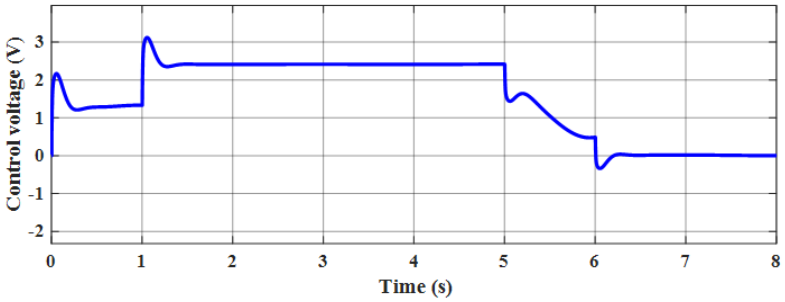

Fig. 8. Trolley's control voltage.

Figure 9 shows the comparison of the proposed IS and the conventional input shaping (ZV with constant natural vibration frequency). It is obviously found that the proposed approach is better than the conventional one that can suppress the time-varying frequency vibration. The swing of the proposed approach is mostly

eliminated when the swing of the conventional one still exists. Further, the peak of the proposed approach's response is smaller than the conventional one.

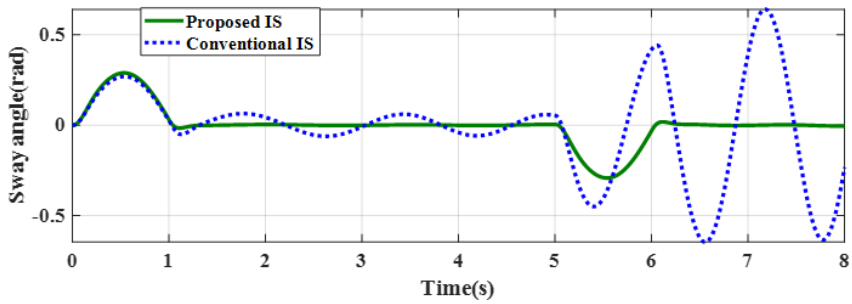

Fig. 9. Comparison between the proposed IS and conventional IS (ZV).

4.2.3. Robustness evaluation

In the previous section, the effectiveness of the proposed input preshaping approach was demonstrated by MATLAB Simulink. We used the precise time-varying frequency for simulation, but the frequency usually is not a clear expression in practice. In this paper, the robustness of the proposed input shaping approach is analysed via simulation.

To consider the robustness, we add a disturbance signal to the model's time-varying frequency. Two typical types of disturbance, including the constant and the time-varying sinusoidal signals, are considered. For each type of disturbance, we will simulate three cases of disturbance's amplitude including 5 percent, 10 percent, and 20 percent values of the initial frequency. The other simulation parameters and conditions are as same as in the previous section.

In the case of sinusoidal disturbance, the disturbance is shown in Fig. 10, and the sway angle responses are shown in Fig. 11. It is clear the proposed IS can still reduce the vibration significantly even with parameter uncertainty. In case of no disturbance, the residual vibration amplitude is about 0.016 rad. However, in the case of sinusoidal disturbance, the residual vibration amplitudes are 0.018 rad for 5% disturbance, 0.020 rad for 10%, and 0.022 rad for 20%, respectively. The greater the disturbance amplitude, the less robust the system is.

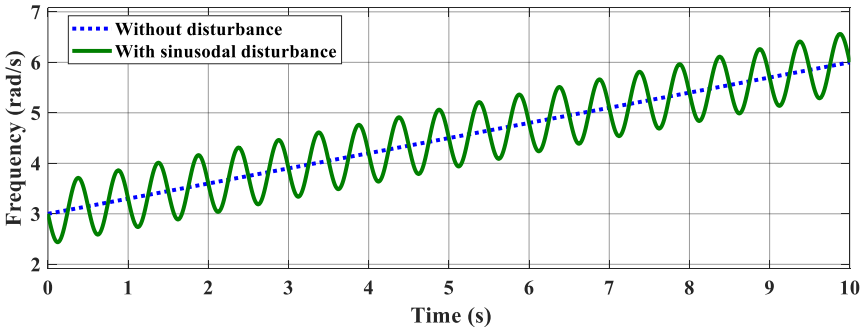

Fig. 10. The time-varying frequency with sinusoidal disturbance.

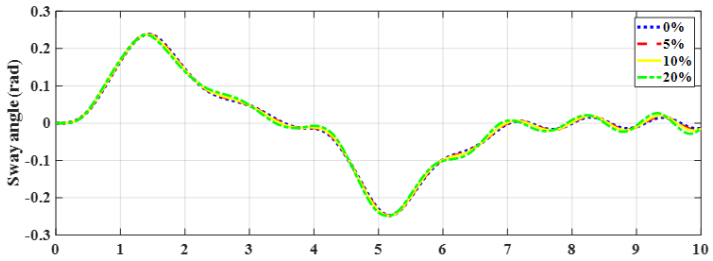

Fig. 11. Residual vibration in case of sinusoidal disturbances.

In the case of constant disturbance, the disturbance is shown in Fig. 12, and the sway angle responses are shown in Fig. 13. Similar to adding the sinusoidal disturbance, the proposed IS still can reduce the vibration sufficiently, and the greater the disturbance amplitude, the less robust the system is. However, the system is less robust in the case of sinusoidal disturbance. In case of constant disturbance, the residual vibration amplitudes are 0.022 rad for 5% disturbance, 0.042 rad for 10%, and 0.066 rad for 20%, respectively.

Robust evaluations with some other disturbance profiles have been done, and the results are similar to these two considered disturbance profiles, i.e., the proposed IS can still reduce the vibration. The residual vibration becomes more significant when the system uncertainty is more extensive. It is the nature of the feedforward controller. This existence is still acceptable. Depending on the precision requirement, the designers can accept the performance, or they have to improve the precision of time-varying frequency identification.

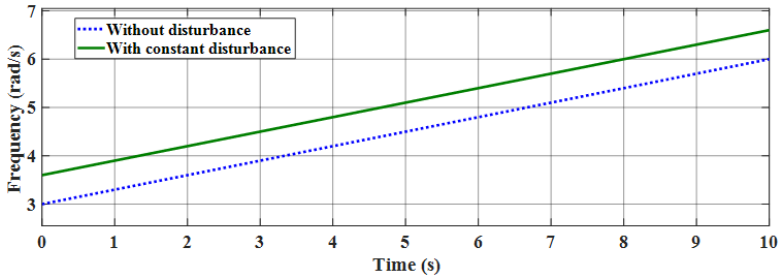

Fig. 12. The time-varying frequency with constant disturbance.

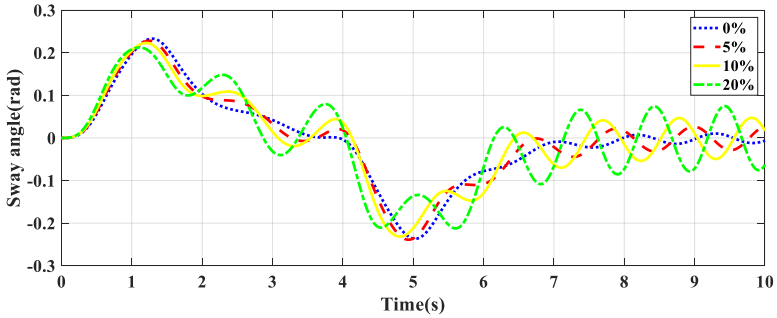

Fig. 13. Residual vibration in case of constant disturbances.

5. Experiments with a 2D Crane Model

To verify the effectiveness of the proposed input preshaping approach, experiments are conducted with a 2D experimental overhead crane, as shown in Fig. 14.

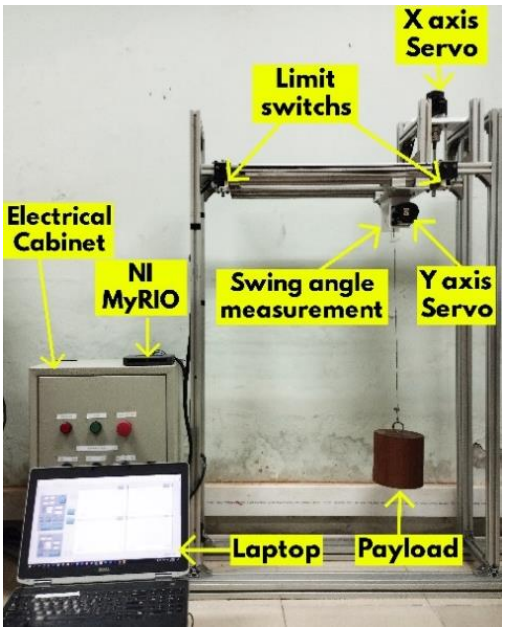

Fig. 14. 2D experimental overhead crane.

The experimental setup is as follows. The trolley moves according to the S-curve profile of about 0.4 m. While the trolley moving, the payload is lifted with cable length reduced linearly from 0.65 m to 0.20 m with a speed of 0.2 m/s. Three control techniques are done, including the proposed input preshaping, ZV, and ZVD input shaper. For the proposed input shaping controller, the reference model parameters are $K_0 = 1/g$, $\xi_0 = 0$, and ω_0= 3.14 rad/s. Similar to the simulation, in the experiment, the reference vibration response is the response of the reference vibratory model with the input being the shaped input created from the reference input using the ZV technique.

The conventional input shapers (ZV and ZVD) are designed for the vibration of payload in two cases: the cable length of 0.65 m (corresponding to the initial cable length) and the cable length of 0.2 m (corresponding to the final cable length). Figures 15 and 16 show the experimental results of the proposed input pre-shaping approach in comparison to the case without vibration suppression control. The vibration is suppressed significantly. In the case without vibration suppression control, the residual vibration magnitude is about 15 degrees, while using the proposed control algorithm, the magnitude is only 0.5 degrees. The control voltage of the proposed control technique is within the limit $\pm 5V$, as shown in Fig. 17. There is a position response delay of the proposed input shaping approach compared to the case without vibration control. This delay is the nature of the general input shaping method.

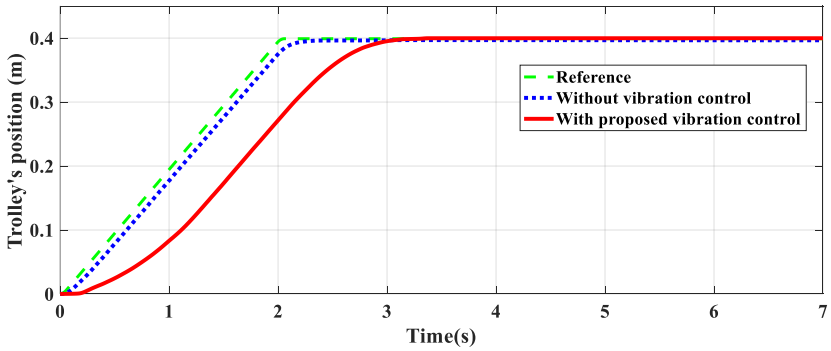

Fig. 15. Experimental trolley position response.

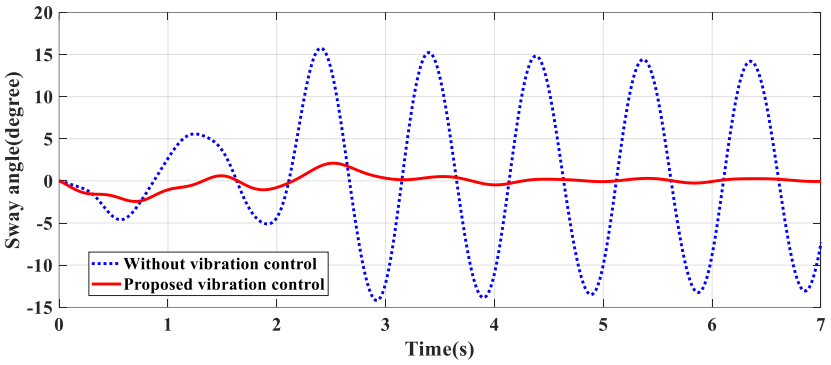

Fig. 16. Experimental load sway angle.

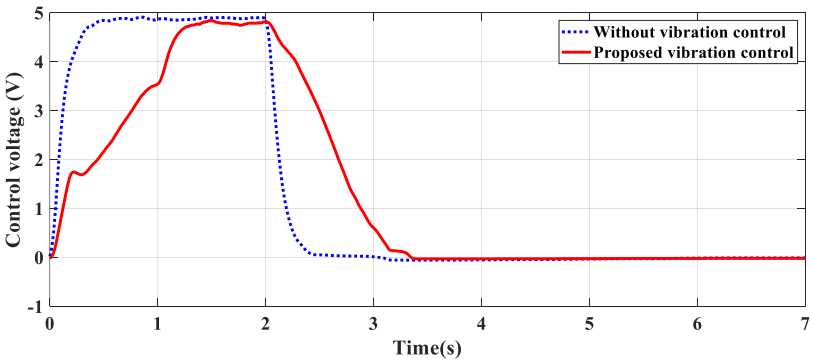

Fig. 17. Experimental trolley control voltage.

To verify the effectiveness of the proposed input shaping approach, a comparison to the ZV and ZVD input shaping are done. ZV1 and ZVD1 are designed with natural vibration frequency corresponding to the initial trolley position (rope length is 0.65 m). ZV2 and ZVD2 are designed with natural vibration frequency corresponding to the final trolley position (rope length is 0.20 m).

Figures 18, 19, and 20 show the trolley position, payload vibration, and control input voltage of ZV1, ZV2, and the proposed input shaping. It is obvious that the

proposed input preshaping outperforms ZV1 and ZV2. The residual vibration magnitude of ZV1 is about 6 degrees, ZV2 is a little smaller, about 5.6 degrees, while the residual vibration magnitude of the proposed approach is only 0.5 degrees. The ZV2 has faster response times since its corresponding frequency is higher. All the control voltages are within the limit.

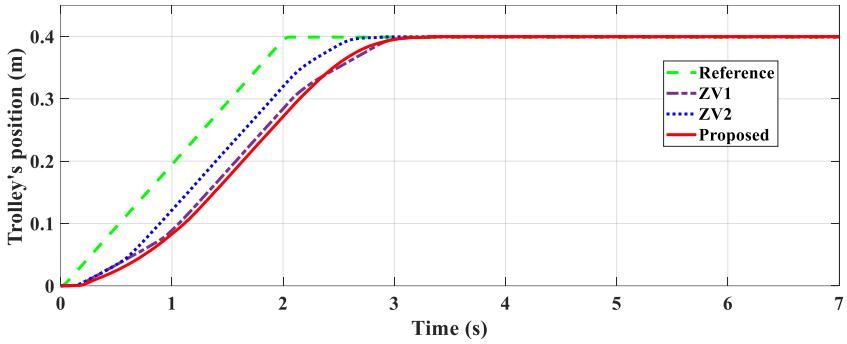

Fig. 18. Comparison of trolley's position response between the proposed IS and ZV input shaper.

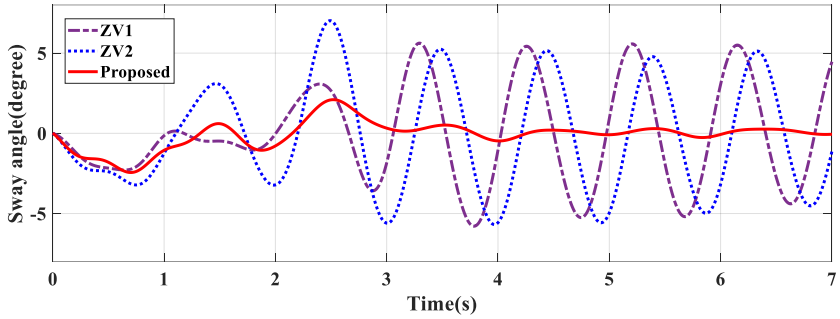

Fig. 19. Comparison of load sway angle between the proposed IS and ZV input shaper.

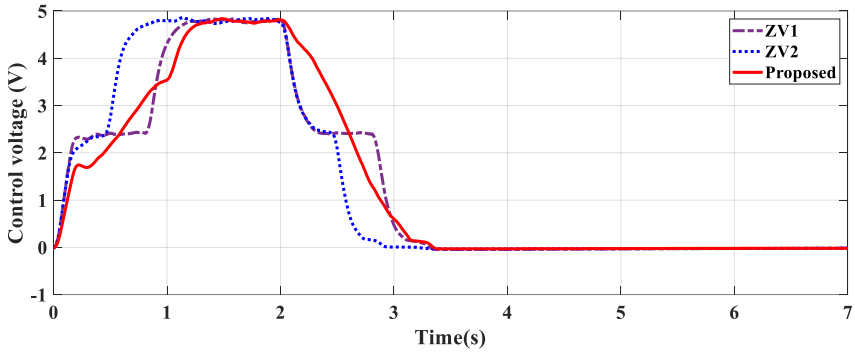

Fig. 20. Comparison of trolley control voltage between the proposed IS and ZV input shaper.

ZV technique is usually less robust than ZVD, so its performance is not so good when the vibration frequency is changed. Next, we do the experiments with the ZVD technique. Figures 21, 22, and 23 show the trolley's position, payload vibration, and control input voltage of ZVD1, ZVD2, and the proposed input preshaping.

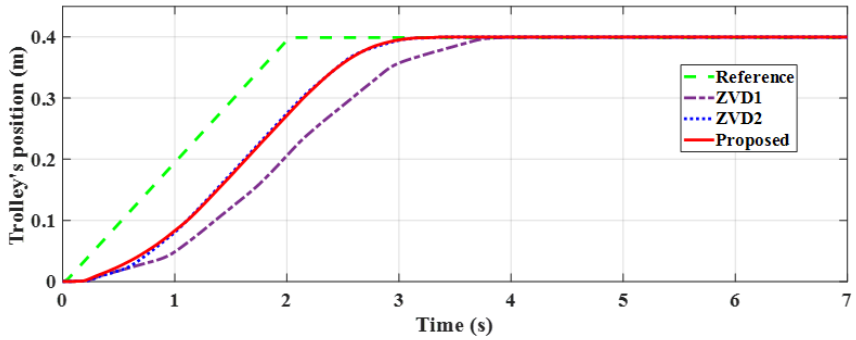

Fig. 21. Comparison of trolley's position response between the proposed IS and ZVD input shaper.

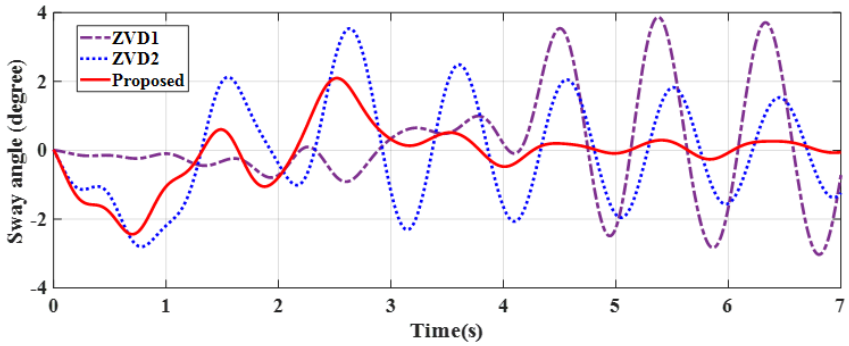

Fig. 22. Comparison of load sway angle between the proposed IS and ZVD input shaper.

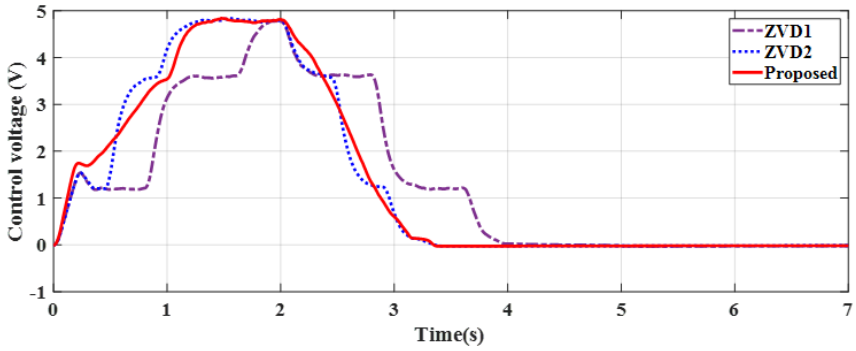

Fig. 23. Comparison of trolley control voltage between the proposed IS and ZVD input shaper .

It is clear that the proposed input preshaping also outperforms ZVD1 and ZVD2, although ZVD1 and ZVD2 can reduce vibration better than ZV1 and ZV2. The residual vibration magnitude of ZVD1 is about 3.8 degrees, ZV2 is a little smaller, about 2.1 degrees, while the residual vibration magnitude of the proposed approach is only 0.5 degrees. In addition, the proposed IS has the same response times as ZVD2 and is faster than ZVD1. All the control voltages are within the limit. Thus, ZVD is more robust than ZV, but it is still ineffective in the time-varying vibratory system. Furthermore, the proposed IS is effective for the time-varying vibratory system.

6. Conclusions

In this paper, a reference response base input preshaping technique to suppress the time-varying vibration has been developed. The reference response is the desired response of the system, and it is the response of a reference time-invariant vibratory system with the input is shaped by conventional input preshaping technique. The simulation and experiments show the effectiveness of the proposed IS technique. The residual vibration is reduced significantly even with vibration model parameter uncertainty. The computation is simple that can apply easily to practice. At present, only the ZV technique is used to create a reference response, but any effective IS technique for a time-invariant vibratory system can also be used instead.

In the future, the extension of the proposed IS approach to multi-mode time-varying vibratory systems will be done. The proposed IS will be applied to other flexible dynamics systems to prove its effectiveness. In addition, the comparison to other adaptive input preshaping techniques is also considered.

Acknowledgements

The paper is supported by Hanoi University of Science and Technology project T2021-PC-002.

Nomenclatures

B_{eq}	Equivalent viscous damping of trolley transition, Ns/m
B_p	Equivalent viscous damping of payload rotation, Ns/rad
F_x	Applied force to the trolley, N
f_{cx}	Friction force of trolley moving along x axis, N
$f_{c\theta}$	Friction force of the payload rotating around the trolley, N
g	Acceleration of gravity, g=9.81 m/s^2
K_d	Derivative gain
K_m	Motor velocity control gain
K_p	Proportional gain
l	Length of rope, m
M	Mass of trolley, kg
m	Mass of payload, kg
N_F	Filter coefficient
T_m	Motor velocity control time constant
$x(t)$	Trolley position over time along X-axis, m

Greek Symbols

θ	Sway angle, rad
ξ	Damping factor
ω	Vibration frequency, rad/s

Abbreviations

IS	Input preShaping
ZV	Zero Vibration
ZVD	Zero Vibration Derivative
ZVDD	Zero Vibration Derivative

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