Minimizing Residual Vibrations for Non-Zero Initial States: Application to an Emergency Stop of a Crane

Joaquim M. Veciana^{1, #}, Salvador Cardona¹, Pau Català¹

¹ Department of Mechanical Engineering, Universitat Politècnica de Catalunya, ETSEIB, Diagonal 647, Barcelona, Spain, 08028

Corresponding Author / E-mail: joaquim.maria.veciana@upc.edu, TEL: +34-93-4016718, FAX: +34-93-4015813

KEYWORDS : Residual vibration; Residual response; Vibration control; Command shaping; Input shaping; Forcing function

This paper presents an approach to design command inputs for the reduction of residual vibrations in mechanical systems for non-zero initial states. The technique starts with the development of a standard input shaper in a second order oscillator, and adds the initial states of the system to the formulation. However, it is proven that the obtained input shaper does not guarantee a null residual vibration when it is convolved with any unshaped command. To obtain an analytically proven zero-vibration response, it is required to include the parameters that characterize the response of the system to a selected normalized unshaped command in the formulation. By introducing an appropriate processing time delay and by measuring the initial states using a feedback sensor, the proposed method has been applied in a real time scenario where an emergency stop of a crane is required, assuming that initially the payload is swaying. The analyzed systems are those that can be modeled as discrete linear ones and can be described with constant parameter motion equations. Experimental results show the efficacy of the developed method with its application to a scaled crane test platform.

1. Introduction

Motion control has been one of the main areas of focus for the development of residual vibration reduction. Its application has been widely addressed to cranes ¹⁻⁴. Extensive work has been carried out within the methods based on command generation techniques applied to discrete linear systems ^{5,6}. In general, the state of the flexible modes at the initial condition is considered to be null, such as in the rest-to-rest maneuvers. This is the case of a standard crane. Most of its maneuvers, that include an input shaper in its control, will start in a null initial oscillatory condition and will achieve a null residual vibration in the stationary state. However, if there is a requirement of an emergency stop while the payload is swaying during the transient, the motion profile which is generated in a standard condition does not guarantee the stop of this load with a null residual oscillation. The main objective of the current development is to provide a method to design input functions which includes the non-zero initial states of the system in its formulation and to focus on a real time application such as an emergency stop of a crane. The new method can be included within the time-domain approaches. Other literature methods among the before mentioned time-domain approaches are the signal generation by means of trigonometric series and the so-called input shapers which are the most commonly used; other methods base their formulation in the frequency-domain, such as command conventional filtering and zero-placement techniques.

Within the time-domain techniques, early approaches ⁷⁻⁹ focused the definition of the transient motion profiles through trigonometric series expansion avoiding the terms related with the system natural frequency. Another technique defines the motion profiles through convolution of pulses multiplied by a negative exponential time function when damping is considered ^{10,11}.

The zero-vibration (ZV) input shaper was firstly proposed in 1950 ¹² and developed later ¹³⁻¹⁵: an input signal is shaped by convolving it with two impulses, properly located in time and with the appropriate amplitude. The vibration that is generated with the first impulse is suppressed with the vibration which is generated with the second one. This property remains invariable when any command signal is convolved with this impulse sequence. To provide robustness against variation of the system parameters, the zero-vibration and derivative (ZVD) and extra insensitive (EI) shapers were designed by adding one more impulse to the ZV shaper. Other approaches of this technique included: i) phasorial diagrams to reduce numerical calculations ^{16,17}; ii) negative impulses to reduce the transient time ¹⁸; iii) an optimal trade-off between performance, measured in terms of residual vibration and speed, and robustness ¹⁹; iv) the cancellation of a so-called pseudo-mode with lower frequency than any of the system component modes ²⁰; v) the reduction of the system response during the transient ^{21,22}.

Frequency domain approaches, such as conventional command filtering was also studied to eliminate the frequency content at the system resonance from the input ²³. However this technique is not considered an analytically proven ZV method: Although it is demonstrated that reducing the frequency content at the system resonance is equivalent to reducing the residual vibration ²⁴, the lower effectiveness of the filtering methods can be justified because this statement is only valid for undamped systems. Therefore, these methods do not give a total residual vibration reduction in a damped case.

Methods involving discrete filters were also developed. The zero-placement technique ^{25,26} was extended to the discrete domain by constructing

the impulse sequences in the z-plane²⁷. The Optimal Arbitrary Time-delay Filter (OATF)²⁸, where its time-domain representation is a set of positive and negative impulses with variable time-delay. In these techniques, the desired signal is generated by convolving an unshaped command with a sequence of impulses.

The approach which is presented in this paper follows an analog development to that of the input shaper, by adding the initial state of the system. Nonetheless, in order to be successful with an analytically proven null residual vibration, the residual response of the system to a unitary normalized transient unshaped command is used instead of the impulse response. In addition, the usage of a feedback sensor to measure the initial state makes this approach slightly different from a pure feedforward method. Thus, it can be classified as an alternative approach which finds itself in the middle of the input shaping and the closed-loop input shaping controllers, which are based on a combined use of input shaping and feedback control³⁰⁻³². The method is developed by assuming a one degree-of-freedom damped linear system with a constant parameter motion equation. The application to a real time scenario implies some computation time during the emergency stop, which is included in the formulation as a time delay.

The rest of this paper is organized as follows. In Section 2, the problem formulation is introduced and it is proven that to design an input shaper by including the initial state of the system does not guarantee its effectiveness when it is convolved with any unshaped command. In section 3, the new input design method is proposed and it is implemented to a real time scenario. Furthermore, a qualitative comparison with closed-loop input shaping is included. In Section 4, some experimental results are shown by using a pendulum-carriage test bed which essentially simulates a crane. Finally, in Section 5, conclusions are drawn, pointing out the benefits of the method developed.

2. Problem formulation

The development which has been carried out in this section follows that of a standard input shaper and includes the input shaper formulation for non-zero initial states²⁹. A second order harmonic oscillator has been considered, in which $x(t)$ is the input and $y(t)$ the system response. The transfer function $G(s)$ and the response of the system $y_h(t)$ to an impulse of magnitude A are described, respectively, by:

$$G(s) = \frac{Y(s)}{X(s)} = \frac{\omega_0^2}{s^2 + 2\zeta\omega_0 s + \omega_0^2} \quad (1)$$

$$y_h(t) = \frac{A\omega_0}{\sqrt{1-\zeta^2}} e^{-\zeta\omega_0 t} \sin(\omega_d t) \quad (2)$$

where ω_0 is the system natural frequency, ζ is the damping ratio and $\omega_d = \omega_0 \sqrt{1-\zeta^2}$ is the oscillation frequency.

If the non-zero initial-state condition is introduced, the output yields:

$$Y(s) = G(s) \left[X(s) + \frac{\dot{y}(0) + y(0)(s + 2\zeta\omega_0)}{\omega_0^2} \right] = G(s)[X(s) + D_0(s)] \quad (3)$$

which is equivalent to adding a disturbance D_0 to the input of the oscillator (Figure 1).

Fig. 1 Block diagram of the oscillator $G(s)$ with a non-zero initial condition $D_0(s)$

2.1 Input shaper design and non-zero initial states

In a zero initial state condition, an input $x(t)$ which is described with a sequence of n impulses that cancels the residual vibration can be used as an input shaper (IS). The convolution of this sequence with any unshaped command $r(t)$ also cancels the residual vibration.

Fig. 2 Input shaping block diagram with a non-zero initial condition: $R(s)$ and $X(s)$ are the unshaped and shaped commands, respectively, and IS is the input shaper

This statement is not satisfied for the non-zero initial states case. The necessary and sufficient condition in order to obtain zero residual vibration is that the Laplace transform (LT) of the time-bounded control input to the system G has zero magnitude at the system poles^{20,25,26}. According to Figure 2, equation (4) is the input to the oscillating system. Consequently, both $D_0(s)$ and $R(s)$ are contributors to the zeros of this equation and, hence, the zero vibration condition is not guaranteed only with the first addend.

$$\mathbf{R}(s)\mathbf{IS}(s)+\mathbf{D}_0(s) \quad (4)$$

An example of this conclusion is described in Figures 3 and 4. Assume that the unshaped command $\mathbf{R}(s)$ is, in the time domain, a unitary impulse $r_1(t) = \delta(t)$, where $\delta(t)$ is the Dirac's delta function. Therefore, the resultant shaped command $x(t)$ is the IS impulse response. If the IS is a ZV shaper, $x(t)$ is written as:

$$x(t) = A_1\delta(t) + A_2\delta(t - t_2) \quad t \geq t_2 \quad (5)$$

where A_1 and A_2 are the magnitudes of the impulses that define this impulse response. Consider also that $y_0(t)$ is the free oscillation of the system for $t \geq 0$ to a given initial state. If the IS is designed to avoid residual vibrations for an unshaped command $r_1(t) = \delta(t)$ and for a given initial state²⁹, the total system response $y(t)$ shows null residual vibration for $t \geq t_2$ (Figure 3). Later, a rectangular pulse is used as an unshaped command $r_2(t)$, with the IS that is already defined. The results show non-zero residual vibration after the input (Figure 4).

Fig. 3 Shaped command $x(t)$ and total system response $y(t)$ for an IS designed to avoid residual vibrations for a given initial state $y_0(t)$, when the unshaped command is $r_1(t) = \delta(t)$

Fig. 4 Shaped command $x(t)$ and total system response $y(t)$ when the unshaped command $r_2(t)$ is a pulse

3. Input design method

The system free oscillation $y_0(t)$ can be described as the free response of the system to one impulse with a magnitude of A_0 which is located at $t_0 \leq 0$. The equation for $t \geq 0$ is given by:

$$y_0(t) = \frac{A_0\omega_0}{\sqrt{1-\zeta^2}} e^{-\zeta\omega_0(t-t_0)} \sin(\omega_d(t-t_0)) \quad t \geq 0 \quad (6)$$

In a general case, if the initial state equation (6) is included, the response of the system after the impulse n is:

$$y(t) = \frac{\omega_0}{\sqrt{1-\zeta^2}} \left[A_0 e^{-\zeta\omega_0(t-t_0)} \sin(\omega_d(t-t_0)) + \sum_{i=1}^n [A_i e^{-\zeta\omega_0(t-t_i)} \sin(\omega_d(t-t_i))] \right] \quad t \geq t_n \quad (7)$$

On the other hand, the free response of the system G at the end of a known transient input $x(t)$ with a duration t_a (normalized to a unitary area) corresponds to a damped harmonic oscillation. It can be described by:

$$y_a(t) = \frac{A_a \omega_0}{\sqrt{1-\zeta^2}} e^{-\zeta \omega_0 (t-t_a)} \sin(\omega_d (t-t_a) + \varphi_a) \quad t \geq t_a \quad (8)$$

where A_a and φ_a are constant parameters such that $A_a \omega_0 / \sqrt{1-\zeta^2}$ is the amplitude and $(\varphi_a - \omega_d t_a)$ is the phase of the free oscillation. By evaluating the equation (8) and its time derivative at $t = t_a$, A_a and φ_a become:

$$A_a = \frac{y_a(t_a) \sqrt{1-\zeta^2}}{\omega_0 \sin(\varphi_a)} \quad (9)$$

$$\varphi_a = \text{atan}\left(\frac{y_a(t_a) \omega_d}{\dot{y}_a(t_a) + y_a(t_a) \zeta \omega_0}\right) \quad (10)$$

The response of the system to the input $x(t)$, evaluated at $t = t_a$, provides the values $y_a(t_a)$ and $\dot{y}_a(t_a)$. Notice that the values of A_a and φ_a depend on the system parameters ω_0 and ζ for a given input $x(t)$. Figure 5 depicts an example of $y_a(t)$ when the input is a rectangular pulse.

Fig. 5 Free response of the system $y_a(t)$ after applying a rectangular pulse as $x(t)$

If this input is used as an unshaped command $r(t)$ in the system of the Figure 2, the resultant response for $t \geq (t_n + t_a)$ is:

$$y(t) = \frac{\omega_0}{\sqrt{1-\zeta^2}} \left[A_0 e^{-\zeta \omega_0 (t-t_0)} \sin(\omega_d (t-t_0)) + \sum_{i=1}^n \left[A_a A_i e^{-\zeta \omega_0 (t-t_i-t_a)} \sin(\omega_d (t-t_i-t_a) + \varphi_a) \right] \right] \quad t \geq t_n + t_a \quad (11)$$

Following an analogous development to that of the input shaper¹³⁻¹⁵, the null residual vibration condition is given by equations (12) and (13).

$$A_0 e^{-\zeta \omega_0 (t-t_0)} \cos(\omega_d t_0) + \sum_{i=1}^n \left[A_a A_i e^{-\zeta \omega_0 (t-t_i-t_a)} \cos(\omega_d t_i + \omega_d t_a - \varphi_a) \right] = 0 \quad t \geq t_n + t_a \quad (12)$$

$$A_0 e^{-\zeta \omega_0 (t-t_0)} \sin(\omega_d t_0) + \sum_{i=1}^n \left[A_a A_i e^{-\zeta \omega_0 (t-t_i-t_a)} \sin(\omega_d t_i + \omega_d t_a - \varphi_a) \right] = 0 \quad t \geq t_n + t_a \quad (13)$$

By evaluating at $t = t_n + t_a$, the unknowns A_i and t_i can be obtained from:

$$A_0 e^{-\zeta \omega_0 (t_n+t_a-t_0)} \cos(\omega_d t_0) + \sum_{i=1}^n \left[A_a A_i e^{-\zeta \omega_0 (t_n-t_i)} \cos(\omega_d t_i + \omega_d t_a - \varphi_a) \right] = 0 \quad (14)$$

$$A_0 e^{-\zeta \omega_0 (t_n+t_a-t_0)} \sin(\omega_d t_0) + \sum_{i=1}^n \left[A_a A_i e^{-\zeta \omega_0 (t_n-t_i)} \sin(\omega_d t_i + \omega_d t_a - \varphi_a) \right] = 0 \quad (15)$$

Finally, constraints (16) and (17) should be included so as to ensure that the input produces the same rigid-body motion as the unshaped command, and the duration time is minimized.

$$\sum_{i=1}^n A_i = 1 \quad (16)$$

$$\min(t_n) \quad (17)$$

Depending on the initial state, the optimum time condition may imply positive, negative or alternate impulses. Large magnitude impulses can result from this last case. Therefore, to enforce the final motion profile under adequate limits, it is recommended to add a sign condition such as:

$$A_i \geq 0 \quad i = 1, \dots, n \quad (18)$$

Notice that the sequence of impulses that has been defined is only valid for a given unshaped command and, hence, does not have the functionality of a standard input shaper.

3.1 Relative performance versus closed-loop input shaping

Although the proposed method does not use the feedback sensor as a standard closed-loop control system, it is worthy of comparing its performance to the closed-loop input shaping, which is based on a combined use of input shaping and feedback control. Other authors suggest classifying these techniques within two groups³³: the outside-the-loop input shaping (OLIS) and the closed-loop signal shaping (CLSS).

1- Outside-the-loop input shaping (OLIS):

Fig. 6 Outside-the-loop input shaping block diagram

The OLIS architecture can be described by the block diagram shown in Figure 6, where IS , C and G are, respectively, the input shaper, the control and the plant. The IS shaper parameters are selected so that the oscillatory poles of the closed-loop transfer function H are cancelled by the zeros of the IS . The block C is a conventional PID controller. The transfer function H is:

$$H(s) = \frac{C \cdot G}{1 + C \cdot G} = \frac{C \cdot G_n}{G_d + C \cdot G_n} \quad (19)$$

where G_n and G_d are the numerator and denominator of G , correspondingly. As in the previous diagrams, the disturbance D_0 represents the initial states condition. As it is shown in Figure 6, this disturbance is not affected by the IS , and the oscillatory response to D_0 has the same behavior as the oscillatory response to the overall H . This can be seen through the transfer function between D_0 and Y , where its poles are contained in the poles of H :

$$\frac{Y(s)}{D_0(s)} = \frac{G}{1 + C \cdot G} = \frac{G_n}{G_d + C \cdot G_n} \quad (20)$$

Consequently, this architecture does not nullify the oscillations created by the initial conditions.

2- Closed-loop signal shaping (CLSS):

Fig. 7 Closed-loop signal shaping block diagram

The CLSS architecture can be described by the block diagram found in Figure 7. In this case, the IS shaper parameters are selected so that the oscillatory poles of the system G are cancelled by the zeros of the IS . As it is described by other authors³³, the initial conditions D_0 directly affect the plant and are not corrected by the input shaper. The transfer function between D_0 and Y is:

$$\frac{Y(s)}{D_0(s)} = \frac{G}{1 + IS \cdot C \cdot G} = \frac{G_n}{G_d + IS \cdot C \cdot G_n} \quad (21)$$

The expression shows that the oscillatory poles of $\mathbf{G}$ are contained in the poles of $\mathbf{Y}(s)/\mathbf{D}_0(s)$. Therefore, the oscillatory response to $\mathbf{D}_0$ has the same behavior as the oscillatory response of $\mathbf{G}$.

A comparative example is described in Figure 8, where $\mathbf{G}$ is a harmonic oscillator (1) with a natural frequency of 1 rad/s and a damping ratio of 0.05. The unshaped command $r(t)$ is a unitary step and the initial state is set to $y(0) = 0$ and $\dot{y}(0) = 0.5$. Gains of the PID controller are fixed to the unit. The IS is designed to cancel the oscillatory poles of $\mathbf{H}$ for the OLIS scheme, and the oscillatory poles of $\mathbf{G}$ for the CLSS.

Fig. 8 Comparative performance of OLIS, CLSS and the proposed method in the following cases: a) zero initial states and b) non-zero initial states

Results of this simulation show that the OLIS architecture provides better performance to reduce the initial state $\mathbf{D}_0$ than the CLSS. Nonetheless, in both cases an oscillatory response is obtained, as suggested by the transfer functions (20) and (21). Although both closed-loop architectures have an embedded IS , neither of them is able to nullify completely the disturbances $\mathbf{D}_0$ because they are not corrected by the IS . Figure 8 also shows that the proposed technique acts as a zero vibration method, since the new algorithm takes into account what will come into the system as a disturbance, and uses this information to calculate the IS .

3.2 Real time implementation on a crane

The application of the proposed method has been particularized on an emergency stop of a crane assuming that the carriage is moving and the load is swaying.

Fig. 9 Pendulum schematics and coordinates

As shown in Figure 9, the crane is formulated as a pendulum system with a mass m , inertia I_G and length l . The rod is articulated on a carriage which includes a rotary damper with damping constant c . The carriage displacement is defined by the coordinate x and the pendulum angle is defined by the coordinate θ . The motion equation of this pendulum for a given $x(t)$ is described by:

$$(ml^2 + I_G)\ddot{\theta} + c\dot{\theta} + mgl \sin \theta = -m\ddot{x}(t) \cos \theta \quad (22)$$

Consider that the oscillations during and after the transient are small and around the position $\theta=0^\circ$. Therefore, the motion equation can be linearized as:

$$(ml^2 + I_G)\ddot{\theta} + c\dot{\theta} + mgl\theta = -m\ddot{x}(t)l \quad (23)$$

this equation can be rewritten as:

$$\ddot{\theta} + 2\zeta\omega_0\dot{\theta} + \omega_0^2\theta = -\frac{\omega_0^2}{g}\ddot{x}(t) \quad (24)$$

where the system natural frequency and the damping ratio are, respectively:

$$\omega_0 = \sqrt{\frac{mgl}{ml^2 + I_G}} \quad (25)$$

$$\zeta = \frac{c}{2\sqrt{mgl(ml^2 + I_G)}} \quad (26)$$

The real time application of the proposed solution requires considering a time delay t_d that takes into account the computing time between when the stopping command is activated and the start of the motion profile. Figure 10 shows this situation. Assuming that the stopping command is activated at $t=0$, then the first impulse should be located at $t = t_d$.

Fig. 10 Computing time delay t_d between the stopping command and the first calculated impulse

1- Calculations and measurements to be made offline:

The different parameters involved in the equations should be ready when the real time algorithm is activated. Some experimental calibrations should be made offline: For a given unshaped command, a memory map of the parameters A_a and φ_a as a function of ω_0 should be included to cover the cable lengths range. Additionally, the rotary damping constant c can be derived easily from any test of the crane free response. Before each motion starts, the length and the static mass of the load must be measured and, hence, by neglecting the effect of the inertia I_G the natural frequency and the damping ratio can be approximated with equations (25) and (26). The omission of I_G in the calculations is quite acceptable in most of the cases. For example, if a standard European bulk pallet (base $0.8 \times 1.2 \text{ m}^2$) with 1500 kg of concrete bags (pallet height of 1 m approximately) is supposed, the error in ω_0 and ζ when the inertia is omitted (uniformly distributed mass) is less than 2% for cable lengths longer than 2.5 m.

2- Real time algorithm

To implement the real time algorithm, it is necessary to have feedback of the angular state of the crane. For this reason, some sensors that provide instant measurements of the angular position $\theta(t)$ and velocity $\dot{\theta}(t)$ are required. Figure 11 shows the flow diagram of this algorithm. Notice that equations (27) and (28) are used instead of (16) and (18) because it is a decelerating motion.

$$\sum_{i=1}^n A_i = -1 \quad (27)$$

$$A_i \leq 0 \quad i = 1, \dots, n \quad (28)$$

Fig. 11 Flow diagram of the real time algorithm

4. Experimental results

A carriage-pendulum test bed was built to check the goodness of the proposed method (Figure 12). The carriage displacement follows a command input driven with a control, a DC motor and a belt-pulley transmission. The feedback for this control is provided by an incremental rotary encoder, which measures the carriage displacement through the angular position of the idler pulley. The angular velocity $\dot{\theta}(t)$ is measured by means of a gyroscope integrated circuit that is assembled on the rod near to the pendulum rotary axis. The angular displacement $\theta(t)$ is obtained by integrating the angular velocity from a rest condition. Table 1 shows a summary of the values and devices which were used for this test bed.

Fig. 12 Carriage-pendulum test platform: 1) pendulum, 2) gyroscope, 3) carriage

Table 1 Test bed parameters and specifications

Parameter / device	Value / specification
m	0.797 kg
I_G	$15.8 \cdot 10^{-3} \text{ kg} \cdot \text{m}^2$
l	270 mm
c	$14.7 \cdot 10^{-3} \text{ Nm/(rad/s)}$
Motor	Permanent magnet DC brushed
Encoder	Rotary, 7200 cycles per revolution
Gyroscope	ADXRS300, $\pm 300^\circ/\text{s}$, 5mV/($^\circ/\text{s}$)
Power supply	DC 24V, 10A
Power electronics	Galil Motion Control MSA12-80
Control electronics	NI PCI-6036E
Control software	Matlab Simulink®
Computer	Pentium 4, 2.8 GHz, 512 MB RAM

The test-bed setup gives a measured oscillation frequency of $\omega_d=5.342$ rad/s and a damping ratio of $\zeta=0.019$. Therefore, the system natural frequency obtained by calculation is $\omega_0=5.343$ rad/s. The bandwidths of the motor-carriage system and the electronics have cut-off frequencies that are far from the system resonance and, hence, their filtering effect is not significant.

Two sample tests were performed where the carriage acceleration $\ddot{x}(t)$ was taken as the input. In both tests, there is an initial arbitrary command that accelerates the carriage up to a desired constant velocity. This acceleration excites the oscillation behavior of the pendulum. Once the velocity of the carriage is stable and the pendulum is swaying, the user activates an emergency stop.

The number of impulses is fixed to $n=2$ and the selected unshaped acceleration is a rectangular pulse of 0.4 s with 2.5 m/s^2 amplitude (area normalized to 1 m/s). The constants A_a and φ_a are obtained by testing: a pulse is generated as $\ddot{x}(t)$ while $\dot{\theta}(t)$ is measured and integrated to obtain $\theta(t)$, that is the desired output. Later, for $t \geq t_a$, an equation analog to (8) is adjusted to the measured data through the least squares method. The resultant parameters for this unshaped command are $A_a=-0.702$ m/s and $\varphi_a=-0.77$ rad. It is convenient to obtain them by testing to consider the sources of uncertainty like the actuator and the control. The total duration of the unshaped command is the nominal duration 0.4 s added to the delay caused by the control. The measured delay of the control for such motion profiles is around 0.3 s. Thus, $t_a=0.7$ s.

The measured computing time t_d between the stopping order and the generation of the impulse sequence is less than 1 ms. To provide some margin, t_d is fixed to 2 ms.

Fig. 13 Experimental results: acceleration input $\ddot{x}(t)$ and angular velocity $\dot{\theta}(t)$ when the stop order is activated at 1.0 s.

Fig. 14 Experimental results: acceleration input $\ddot{x}(t)$ and angular velocity $\dot{\theta}(t)$ when the stop order is activated at 1.6 s.

In the former test, the stop order is activated at 1.000 s, the first impulse is fixed at $t_1 = 1.000 + t_d = 1.002$ s, the second one is obtained by computing at $t_2=1.267$ s and the crane stops at $t_2 + t_a = 1.967$ s. In the second test, the stop is activated at 1.600 s, the first impulse is fixed at $t_1=1.602$ s, the second one is obtained at $t_2=2.404$ s and the crane stops at $t_2 + t_a = 3.104$ s.

Figures 13 and 14 present the results obtained. The white area corresponds to the stop sequence duration. Notice that in the grey area at the right hand side, the system response $\dot{\theta}(t)$ shows almost zero residual vibration.

5. Conclusions

An approach to design command inputs for the reduction of residual vibrations in mechanical systems for non-zero initial states has been developed. It has been proven, firstly, that an input shaper designed for a non-zero initial state condition does not guarantee the complete cancellation of the residual vibration when it is convolved with an arbitrary unshaped command. The proposed workaround follows an analog development, but additionally includes the parameters that characterize the response of the system to the selected normalized unshaped command in the formulation, as well as the measurement of the initial conditions by means of a feedback sensor. In this case, the obtained solution is a proven zero vibration method, which shows better performance in comparison to the closed-loop input shaping architectures. Nevertheless, it is only valid for the selected unshaped command and, hence, it does not have the filter functionality of an input shaper. By introducing an appropriate processing time delay, the proposed method has been applied in a real time scenario where an emergency stop of a crane is required assuming that, initially, the payload is swaying. Finally, a pendulum test bed, which is essentially a crane, was used to conduct some emergency stop experiments, confirming the effectiveness and usefulness of the method.

REFERENCES

1. Starr, G. P. "Swing-Free Transport of Suspended Objects with a Path-Controlled Robot Manipulator," *ASME Journal of Dynamic Systems, Measurement, and Control*, Vol. 107, No. 1, pp. 97-100, 1985.
2. Sorensen, K., Singhose, W., Dickerson, S., "A Controller Enabling Precise Positioning and Sway Reduction in Bridge and Gantry Cranes," *Control Engineering Practice*, Vol. 15, No. 7, pp. 825-837, 2007.
3. Kim, D., Singhose, W., "Performance Studies of Human Operators Driving Double-Pendulum Bridge Cranes," *Control Engineering Practice*, Vol. 18, No. 6, pp. 567-576, 2010.
4. Stergiopoulos, J., Konstantopoulos, G., Tzes, A., "Experimental Verification of an Adaptive Input Shaping Scheme for Hoisting Cranes," *Mediterranean Conference on Control and Automation*, pp. 730-735, 2009.
5. Singhose, W.E., "Command Shaping for Flexible Systems: A Review of the First 50 Years", *International Journal of Precision Engineering and Manufacturing*, Vol. 10, No. 4, pp. 153-168, 2009.
6. Singhose, W.E., Seering W., "Command Generation For Dynamic Systems", Singhose, W.E. (www.lulu.com), 2011.
7. Aspinwall, D.M., "Acceleration Profiles for Minimizing Residual Response", *ASME Journal of Dynamic Systems, Measurement, and Control*, Vol. 102, No. 3, pp. 3-6, 1980.
8. Meckl, P.H., Seering, W.P., "Minimizing Residual Vibration for Point-to-Point Motion", *ASME Journal of Vibration, Acoustics, Stress, and Reliability in Design*, Vol. 107, No. 3, pp. 378-382, 1985.
9. Meckl, P.H., Seering, W.P., "Experimental Evaluation of Shaped Inputs to Reduce Vibration for a Cartesian Robot," *ASME Journal of Dynamic Systems, Measurement, and Control*, Vol. 112, No. 2, pp. 159-165, 1990.
10. Veciana, J.M., Cardona, S., "Residual Vibration Reduction in Low Damping Systems. Generation of Regular Piecewise Algebraic Polynomial Inputs," *Journal of Vibroengineering*, Vol. 13, No. 4, pp. 739-754, 2011.
11. Veciana, J.M., Cardona, S., "Residual Vibration Reduction in Mechanical Systems: A Time-domain Approach," *International Journal of Precision Engineering and Manufacturing*, Vol. 13, No. 8, pp. 1327-1339, 2012.
12. Smith, O. J. M., "Posicast Control of Damped Oscillatory Systems," *Proc. The IRE*, Vol. 45, No. 9, pp.1249-1255, 1957.
13. Singer, N.C., Seering, W.P., "Preshaping Command Inputs to Reduce System Vibration", *ASME Journal of Dynamic Systems, Measurement, and Control*, Vol. 112, No. 1, pp. 76-82, 1990.
14. Hyde, J.M., Seering, W.P., "Using Input Command Pre-Shaping to Suppress Multiple Mode Vibration", *IEEE Proceedings of the International Conference on Robotics and Automation*, pp. 2604-2609, 1991.
15. Hyde, J.M., Seering, W.P., "Inhibiting Multiple Mode Vibration in Controlled Flexible Systems," *Proceedings of the American Control Conference*, pp. 2449-2454, 1991.
16. Singhose, W.E., Seering, W.P., Singer, N.C., "Residual Vibration Reduction Using Vector Diagrams to Generate Shaped Inputs", *ASME Journal of Mechanical Design*, Vol. 116, pp. 654-659, 1994.
17. Gürleyük, S.S., Cinal, S., "Robust Three-impulse Sequence Input Shaper Design", *Journal of Vibration and Control*, Vol. 13, No. 12, pp. 1807-1818, 2007.
18. Singhose, W.E., Seering, W.P., Singer, N.C., "Time-Optimal Negative Input Shapers", *ASME Journal of Dynamic Systems, Measurement, and Control*, Vol. 119, No. 2, pp. 198-205, 1997.
19. Chang, T., Godbole, K., Hou, E., "Optimal Input Shaper Design For High-Speed Robotic Workcells", *Journal of Vibration and Control*, Vol. 9, pp. 1359-1376, 2003.
20. Singh, T., Heppler, G.R., "Shaped Input Control of a System With Multiple Modes", *ASME Journal of Dynamic Systems, Measurement, and Control*, Vol. 115, No. 3, pp. 341-347, 1993.
21. Singhose, W., Banerjee, A., Seering, W., "Slewing Flexible Spacecraft with Deflection-Limiting Input Shaping", *AIAA Journal of Guidance, Control, and Dynamics*, Vol. 20, No. 2, pp. 291-298, 1997.
22. Kojima, H., Singhose, W., "Adaptive Deflection Limiting Control for Slewing Flexible Space Structures", *AIAA Journal of Guidance, Control, and Dynamics*, vol. 30, No. 1, pp. 61-67, 2007.
23. Singer, N.C., Singhose, W.E., Seering, W.P., "Comparison of Filtering Methods for Reducing Residual Vibration", *European Journal of Control*, Vol. 5, No. 2-4, pp. 208-218, 1999.

24. Meckl, P.H., Seering, W.P., "Reducing Residual Vibration in Systems with Time-Varying Resonances", IEEE International Conference on Robotics and Automation, pp. 1690-1695, 1987.
25. Bhat, S.P., Miu, D.K., "Precise Point-to-Point Positioning Control of Flexible Structures", ASME Journal of Dynamic Systems, Measurement, and Control, Vol. 112, No. 4, pp. 667-674, 1990.
26. Bhat, S.P., Tanaka, M., Miu, D.K., "Experiments on Point-to-Point Position Control of a flexible Beam Using Laplace Transform Technique-Part 1: Open-Loop", ASME Journal of Dynamic Systems, Measurement, and Control, Vol. 113, pp. 432-437, 1991.
27. Tuttle, T.D., Seering, W.P., "A Zero-placement Technique for Designing Shaped Inputs to Suppress Multiple-mode Vibration", Proceedings of the American Control Conference, pp. 2533-2537, 1994.
28. Magee, D.P., Book, W.J., "Optimal Filtering to Minimize the Elastic Behavior in Serial Link Manipulators", Proceedings of the American Control Conference, pp. 2637-2642, 1998.
29. Dhanda, A., Vaughan, J., Singhose, W.E., "Optimal Input Shaping Filters for Non-Zero Initial States," Proceedings of the American Control Conference, pp. 4141-4146, 2009.
30. Kapila, V., Tzes, A., Yan, Q., "Closed-loop Input Shaping for Flexible Structures Using Time-delay Control", Journal of Dynamic Systems, Measurement, and Control, Vol. 122, pp. 454-460, 2000.
31. Zuo, K., Wang, D., "Closed Loop Shaped-input Control of a Class of Manipulators with a Single Flexible Link. Proceedings of the IEEE International Conference on Robotics and Automation, pp. 782-787, 1992.
32. Zuo, K., Drapeau, V., Wang, D., "Closed Loop Shaped-input Strategies for Flexible Robots", International Journal of Robotics Research, Vol. 14, No. 5, pp. 510-529, 1995.
33. Huey, J.R., Sorensen, K.L., Singhose, W.E., "Useful Applications of Closed-loop Signal Shaping Controllers", Control Engineering Practice, Vol. 16, pp. 836-846, 2008.