Vol 5 (1) (2024)

Measurement, Control, and Automation

Website: https:// mca-journal.org

ISSN 1859-0551

Vibration Suppression Control for Overhead Crane using Inverse Notch Filter

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Abstract

Overhead cranes play an important role in many fields such as industry, transportation, etc. However, payload vibration is a common issue that can occur with overhead cranes. To overcome this problem, various control approaches have been applied including feedforward and feedback control approaches. One practical approach is feedforward control where the controller works as a filter to eliminate the vibration. The use of filter can exist in three structures including: the filter is outside the trolley position control loop, in the feedforward path of the trolley position control loop, and in the feedback path of the trolley position control loop. In this paper, the inverse Notch filter in the feedback path of the trolley position control loop is considered as a vibration suppression controller for the overhead crane. This proposed controller can suppress vibration caused by both reference trolley position input and disturbance in the feedback loop. The effectiveness of the proposed controller is verified via simulation and experiments.

***Keywords**: Inverse Notch Filter, Overhead Crane, Vibration Suppression Control, Feedforward Control,*

Symbols

Symbols	Description
M, m, l	Overhead crane parameters
$S(s), C(s), G(s), F(s)$	Transfer function
w, d, θ	System variables

Abbreviations

IS	Input Shaping
PID	Proportional–Integral–Derivative
ADRC	Active Disturbance Rejection Control

1. Introduction

Overhead cranes play an important role in industry and construction transportation, etc., especially in lifting, transporting, and unloading heavy and bulky materials [1]. However, one of the main natural problems of overhead cranes is the fluctuation of payload during operation. The payload fluctuations cause insecurity for surrounding people and equipment, reducing working accuracy, especially when working at high speed. To avoid these problems, many researchers have developed various methods to control the crane system to move the payload to the desired location while minimizing payload fluctuations. These control methods can mainly be separated into open-loop control and closed-loop control techniques.

Closed-loop control techniques, also known as feedback control, will measure and evaluate system states to minimize the oscillation angle and bring the payload to the desired position accurately. There are many control strategies that have been proposed for crane systems such as linear control [2-5], optimal control [6-8], adaptive control [9-12], sliding control [13-16], intelligent control [17-22], etc. The closed-loop control method has the advantage of being resistant to disturbances and uncertain parameters. However, the main disadvantage of this method is the need for sensors to measure the vibration angle. This increases the cost and complexity of the system. Therefore, the feedback control method has limited application in practice.

The dynamical model of an overhead crane is similar to the pendulum model, thus information about payload fluctuations can be known more or less in advance. Therefore, we can apply some feedforward control techniques to modify the input with the goal of eliminating payload oscillations. There are three main feedforward control techniques widely used for crane control including Input Shaping (IS) [23-26], filtering [27-29], and signal smoothing [30-32]. The feedforward control method is quite effective and is widely used to limit crane payload fluctuations because it is easy to implement and does not require a sensor to measure the angle of payload fluctuation.

The feedforward controller can exist in three structures including the filter is outside the trolley position control loop, in the feedforward path of the trolley position control loop [23], and in the feedback path of the trolley position control loop [33] as shown in Figure 2. Among these three structures, only the third structure where the filter is in the feedback path of the trolley position control loop can suppress the vibration cause by both reference position input of the trolley and disturbance inside the trolley position control loop [33].

In this paper, we apply the feedforward control structure where the filter is in the feedback path of the trolley position

Received: 27 December 2023; Accepted: 15 March 2024.

control loop to suppress the payload vibration of an overhead crane. Difference from [33] where the inverse input shaper that is non casual is used, in this paper inverse Notch filter is applied [34,35]. The inverse Notch filter can suppress the vibration caused by both reference position input of the trolley and disturbance inside the trolley position control loop. In addition, Notch filter is a casual component, thus the system become more stable, in comparison to a system with non-casual component.
The rest of the paper is organized as follows. In section 2, the mathematical model of an overhead crane is described. The analysis of three feedforward control structures for vibration suppression control is discussed in section 3. Section 4 presents the simulation/experimental results of the proposed controller. The conclusion and future study are presented in section 5.

2. Overhead Crane Model

Figure 1 describes a simple crane model with the trolley moving in the X direction and the payload being released in the Y direction. In this figure, F_x is the force acting on the trolley, f_{cx} is the friction force of the moving trolley. $f_{c\theta}$ is the friction force of the moving payload, *P* is the force of gravity, *x(t)* is the position of the trolley, *l* is the length of the rope, *m* is the mass of the payload, *M* is the mass of the trolley, and θ(t) is the angle between the rope and the vertical axis (Y axis) - which is the oscillation angle. According to [16], the linearized model of the crane can be described as follows:

$$F_x = (M+m)\ddot{x} + b_x\dot{x} + ml\ddot{\theta} \quad (1)$$
$$-\ddot{x} = l\ddot{\theta} + b_\theta\dot{\theta} + g\theta \quad (2)$$

where b_x and b_θ are the friction coefficients of the trolley and the payload, respectively. Equation (1) describes the relationship between the force acting on the trolley and the trolley position, where the oscillation angle acts as a disturbance signal that affects the control of the trolley position. Equation (2) describes the influence of the motion of the trolley on the payload fluctuation. In fact, the trolley is controlled by a motor with a motor controller that allows us to control the trolley speed accurately and eliminate external disturbances.

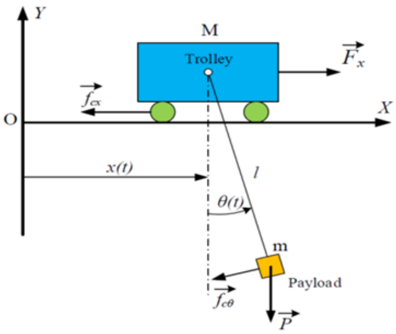


Figure 1: Overhead crane model

Therefore, to design the trolley position controller, instead of using model (1), we will use the motor model with the motor controller as follows [36]:

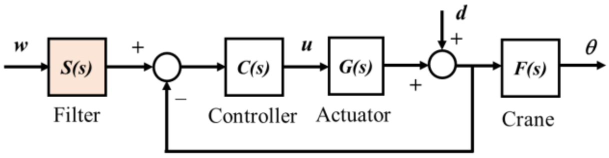
$$G(s) = \frac{X(s)}{U(s)} = \frac{K_c}{(T_c s+1)s} \quad (3)$$

where *U(s)* is the Laplace transform of the control voltage signal *u(t)*, *X(s)* is the Laplace transform of the trolley position *x(t)*, K_c is the amplification coefficient, and T_c is the system time constant.
In practice, the amplification coefficient K_c, and the system time constant T_c are determined by model identification process.

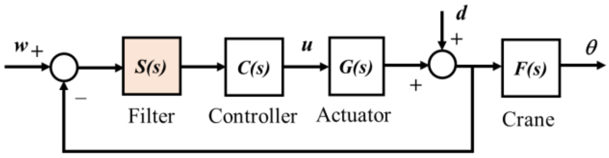
3. Control of overhead crane

3.1. Feedforward control structures

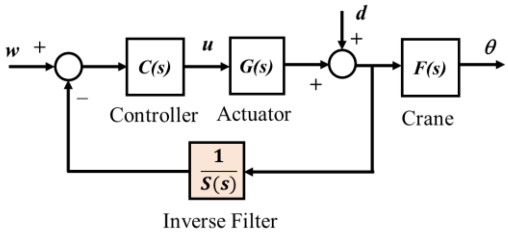
Crane control is to move the payload to the desired position. However, we cannot move the payload directly to the desired position. Instead, we will move the trolley to the desired position and if there is no vibration, the payload will reach the desired position. According to equation (2), the motion of the trolley causes payload fluctuations. Therefore, two tasks must be done, including controlling the trolley to the desired position and at the same time suppressing the oscillations of the payload as quickly as possible. To control the trolley position, a PD position controller will be used for the object in equation (3). Additionally, to suppress payload oscillations, a filter is used. There are three control structures as shown in Figure 2. In these structures, *w* is reference input or reference trolley position, *d* is disturbance to the system, and θ is payload angle. We will analyze the effect of filter *S(s)* in suppression of vibration caused by reference input and disturbance.



(a) Control structure with the filter is outside the position control loop



(b) Control structure with the filter is in the feedforward path of the trolley's position control loop



(c) Control structure with the inverse filter is in the feedback path of the trolley's position control loop

Figure 2: Three vibration suppression control structures for overhead crane

In the first structure, as shown in Figure 2a, the filter *S(s)* is outside the position control loop and plays a role suppress the payload's vibration. The feedback controller *C(s)* guarantees the precise positioning for the trolley. The advantage of this structure is that the control design process is simple, and we can design the filter and the feedback controller independently.

However, the filter must be chosen carefully so that there is no steady state error between the signals before and after the filter. The main disadvantage of this structure is that the vibration caused by disturbance cannot be suppressed.

Let us consider the transfer function from input *w* and disturbance *d* to the output θ as follows:

$$\frac{\Theta(s)}{W(s)} = \frac{S(s)C(s)G(s)F(s)}{1+C(s)G(s)} \tag{4}$$

$$\frac{\Theta(s)}{d(s)} = \frac{F(s)}{1+C(s)G(s)} \tag{5}$$

From equation (4), it is clear that the zeros of the filter *S(s)* can be assigned to cancel the oscillatory modes of crane *F(s)* given by (2), i.e. the filter can suppress the vibration of the crane. However, *S(s)* does not appear in the transfer function from disturbance d to payload angle θ as shown in equation (5). Therefore, *S(s)* cannot cancel the oscillation caused by disturbance.

In the second structure, as shown in Figure 2b, the filter $S(s)$ is in the feedforward path of the position control loop. The feedback controller $C(s)$ guarantees the precise positioning for the trolley. The feedback control design process is quite complicated since the filter is in the loop. However, the steady-state error can be avoided. As same as the first structure, the main disadvantage of this structure is that the vibration caused by disturbance cannot be suppressed. Let us consider the transfer function from input *w* and disturbance d to the output θ as follows:

$$\frac{\Theta(s)}{W(s)} = \frac{S(s)C(s)G(s)F(s)}{1+S(s)C(s)G(s)} \tag{6}$$

$$\frac{\Theta(s)}{d(s)} = \frac{F(s)}{1+S(s)C(s)G(s)} \tag{7}$$

As shown in equation (6), it is clear that the zeros of the filter *S(s)* can cancel the oscillatory modes of crane *F(s)* given by (2), i.e. the filter can suppress the vibration of the crane. However, in equation (7), $S(s)$ cannot cancel the oscillation modes of crane $F(s)$ caused by disturbance.

In the third structure, as shown in Figure 2c, the inverse filter $1/S(s)$ is in the feedback path of the position control loop. The feedback controller $C(s)$ guarantees the precise positioning for the trolley. The feedback control design process is quite complicated since the inverse filter is in the loop. However, the steady state error can be avoided. The main advantage of this structure is that the vibration caused by both reference input and disturbance can be suppressed. Let us consider the transfer function from input *w* and disturbance d to the output θ as follows:

$$\frac{\Theta(s)}{W(s)} = \frac{S(s)C(s)G(s)F(s)}{S(s)+C(s)G(s)} \tag{8}$$

$$\frac{\Theta(s)}{d(s)} = \frac{S(s)F(s)}{S(s)+C(s)G(s)} \tag{9}$$

From equation (8) and (9), it is clear that the zeros of the filter $S(s)$ can be assigned to cancel the oscillatory modes of crane $F(s)$ given by (2), i.e. the filter can suppress the vibration of the crane.

From this analysis, we choose the third structure for control of the overhead crane. However, inverse of the popular filters *S(s)* such as input shapers are non-casual. The non-casual filter may not be realizable and therefore the approximation must be used. To avoid this problem, the casual inverse filter should be chosen. In this paper, we propose to use Notch filter as the filter *S(s)* to suppress the vibration. The Notch filter and its inverse are casual and can be realizable in practice.

After choosing the inverse filter $1/S(s)$, we can design the controller *C(s)* as PID controller that stabilizes the system and guarantees good position tracking. We have the models of components for the overhead control system as in Figure 2c as follows:

- The PID feedback controller: $C(s) = K_P + \frac{K_I}{s} + K_D s$
- The actuator (motor and driver): $G(s) = \frac{K_c}{(T_c s+1)s}$
- The Notch filter: $S(s) = \frac{s^2+2\omega_0 Ds+\omega_0^2}{s^2+2\omega_0 DK_n s+\omega_0^2}$, where ω_0 is filter frequency, D is damping ratio, and K_n is coefficient to control the depth and width of the filter.
- Crane vibration model: $F(s) = \frac{s^2}{ls^2+b_\theta s+g}$

3.2. Control design

From equation (9), to suppress the the payload vibration, zeros of *S(s)* have to cancel the poles of *F(s)*. This leads to the parameters ω_0 and *D* of the Notch filter is calculated as

$$\omega_0 = \sqrt{\frac{g}{l}} \qquad D = \frac{b_\theta}{2\sqrt{gl}} \tag{10}$$

The parameters K_P, K_I, K_D of controller *C(s)* and coefficient K_n of the Notch filter is chosen such as the following polynomial *P(s)* is Hurwitz:

$$P(s) = s^2(s^2 + 2\omega_0 DK_n s + \omega_0^2)(T_c s + 1) + \mathrm{K_c}(K_D s^2 + K_P s + K_I)(s^2 + 2\omega_0 Ds + \omega_0^2) \tag{11}$$

4. Simulation and Experimental Results

To validate the proposed control structure, simulations and experiments have been done. The experiment apparatus is shown in Figure 3.

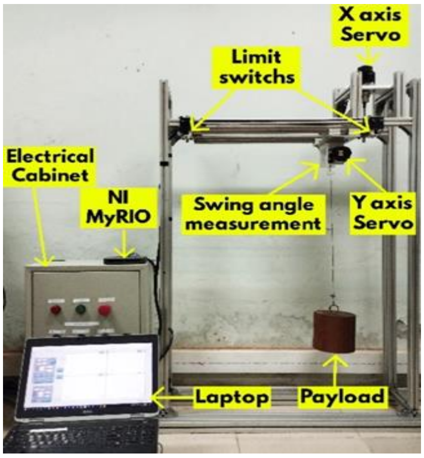


Figure 3: Experimental overhead crane system

The system parameters are identified and set as in Table 1. The parameters in Table 1 are used for simulations and experiments.

Table 1: System and controller parameters

Parameter	**Value**
Crane parameters	
Actuator gain (K_c)	30 [cm/s/V]
Actuator time constant (T_c)	3 [s]
Rope length (l)	0.4 [m]
Gravitational acceleration (g)	9.81 [m/s^2]
Notch filter	
Frequency (ω_0)	4.9 [rad/s]
Damping ratio (D)	0.017
K_n	35
PID controller	
K_P	0.713
K_I	0.001
K_D	0.835

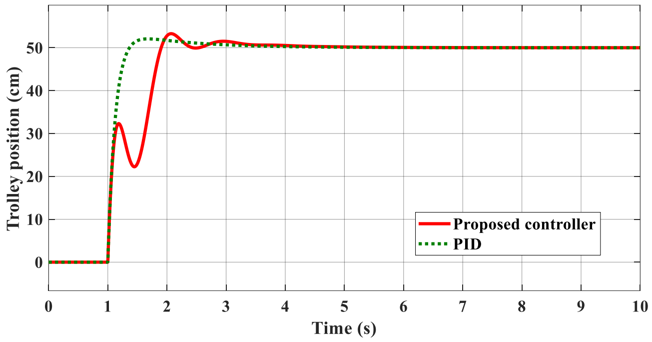
4.1. Simulation results

Three simulations have been conducted for the proposed controller. The simulation of overhead crane control using PID position control for trolley and input shaper for vibration suppression control have also been conducted to compare with the proposed controller.

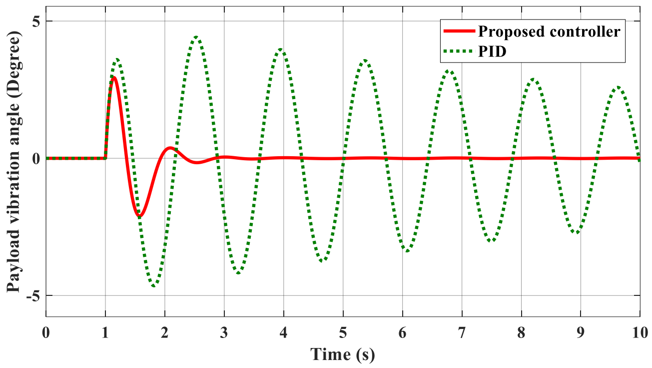
In the first simulation, the proposed control system is simulated with step input. The results are compared to the response of the overhead crane system using PID controller for trolley position without any filter for vibration suppression. As shown in Figure 4, the proposed control system can suppress the payload vibration significantly right after the trolley gets to the destination. The precise position tracking of the trolley is guaranteed with fast response time, in comparison to PID controller. The PID controller for trolley alone cannot suppress the payload vibration.

In the second case, a step signal with a magnitude of 5 is added to the system at time t = 6s for 1 second as a disturbance. As shown in Figure 5, for both controllers, the trolley moves out of the stable position when the disturbance appears. Then the trolley returns to the reference position when the disturbance disappears. However, the proposed controller can suppress the vibration caused by both reference input and disturbance. There is a little fluctuation when the disturbance appears, but it is then suppressed by the filter.

In the third simulation, the proposed controller is compared to the overhead control system using PID controller as trolley position controller and input shaper outside the trolley position control loop as vibration filter. In this case, at t = 6, a step signal with a magnitude of 5 is applied to the system and remains constant thereafter, which is considered as a system disturbance. As shown in Figure 6, at the trolley moving period without disturbance, the response of the input shaper is better than the proposed controller with smaller vibration magnitude. When there is a disturbance, the proposed controller can still suppress the vibration. However, with the input shaper, the vibration appears when there is a disturbance, and it cannot be suppressed. Note that the position controller takes a long time to return the trolley to set-point position.

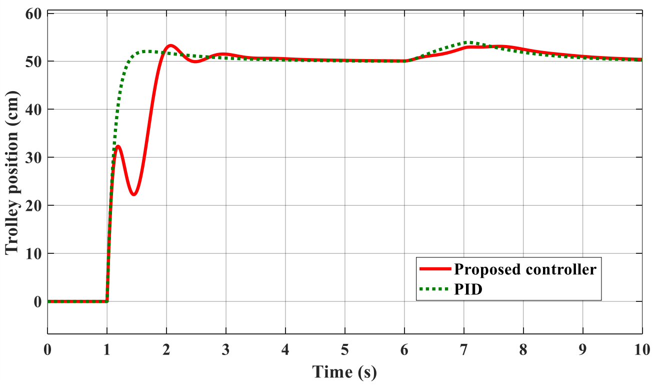


(a) Trolley position

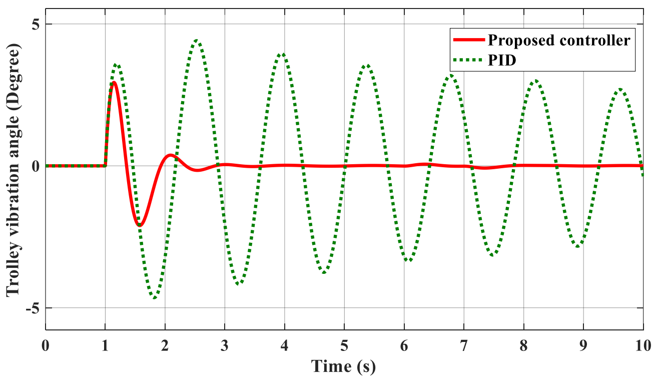


(b) Payload vibration angle

Figure 4: Comparison of the control system using only PID controller with the proposed control for overhead crane in the case of without disturbance

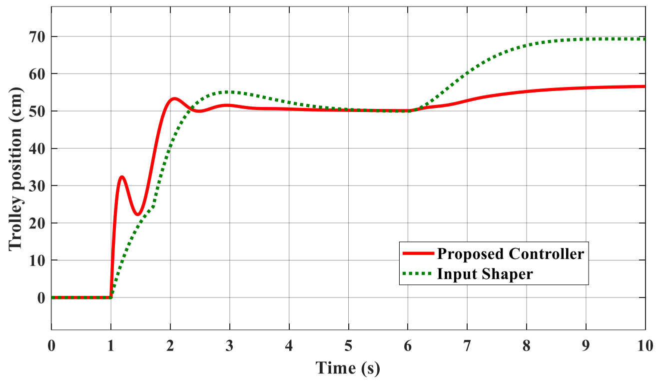


(a) Trolly position

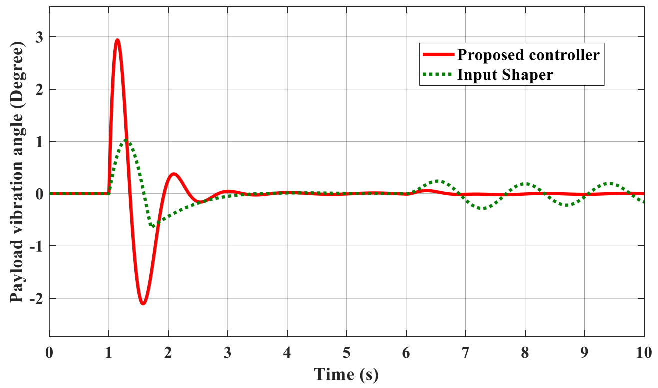


(b) Payload vibration angle

Figure 5: Comparison of the control system using only PID controller with the proposed control for overhead crane in present of disturbance

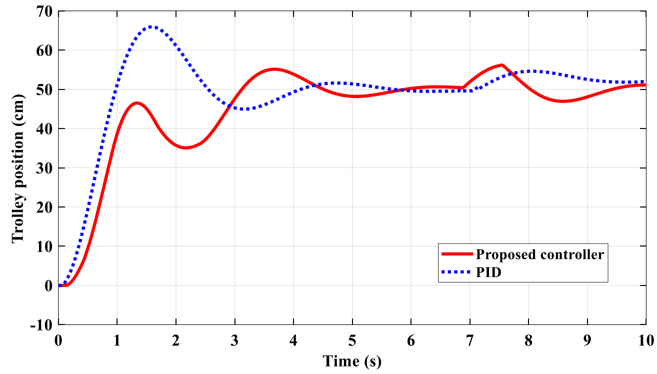


(a) Trolley position

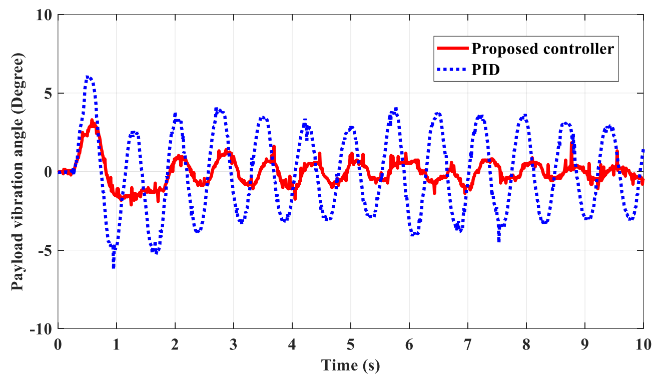


(b) Payload vibration angle

Figure 6: Comparison of the input shaper and the proposed control for overhead crane in the present of disturbance



(a) Trolley position



(b) Payload vibration angle

Figure 7: Experimental results of PID and the proposed control for overhead crane in the present of disturbance

4.2. Experimental results

To confirm that the proposed control can be applied to the practical system, an experiment of overhead crane control has been conducted. Similar to simulation case, the step input is applied to the system. The experiment of overhead crane with PID controller has also been conducted to compare with the proposed controller.

The experiment results are shown in Figure 7. With the same simulation parameters, the performance of the practical system is reduced, the transition time is longer with higher overshoot. However, the system is still stable and with small steady state errors (1% for PID Controller and 2% for proposed controller). The payload vibration angle in the case of PID controller is quite high. The maximum vibration angle is 6 degrees at transient period and 4 degrees at steady state period. While in the case of the proposed controller, the payload vibration angle is significantly reduced. The maximum vibration angle is 3 degrees at transient period and 1 degree at steady state period.

When a disturbance appears (a voltage with a value of 0.75V is added to the system at time t = 7 for a period of about 1 second), in the case of PID controller, since the payload vibration is still large, the effect of disturbance to the payload vibration is difficult to recognize. In the case of the proposed controller, the effect of the disturbance to the payload vibration is very small and we cannot find any increase of payload vibration in this situation. It is noted that because of measurement noise, there is some sudden change in the payload vibration response. It is not the real response that we can observe in practical experiments.

5. Conclusion

In this paper, a feedback vibration suppression control structure for overhead cranes is developed. In this structure, an inverse Notch filter is put in the feedback path of trolley position control loop. This allows to suppress the payload vibration caused by both reference position input and disturbance. In addition, the use of inverse Notch filter that is a casual component instead of input shaper type filter can guarantee the system more stable.

The use of a Notch filter instead of input shaper can avoid delay component in the system. Since the order of the system is quite high when inverse Notch filter is used, the selection of controller parameters to obtain the desired performance is difficult. In the next step, an optimization design process is considered. In addition, the PID controller can be replaced by more powerful controllers such as ADRC to improve the system performance.

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