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A GANTRY CRANE CONTROL SCHEME USING HYBRID INPUT SHAPER AND PID CONTROLLER

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ABSTRACT

Gantry crane is a critical system used for point-to-point transportation of a payload in industries, ports, nuclear plants etc. Speed, accuracy and safety are of paramount importance in gantry cranes (GCs) operation. However, operating GCs as fast as possible generally results in payload sway coupled with inaccurate positioning of the load which degrades the speed, accuracy and safety. This work proposed a hybrid control system for sway suppression and precise trolley positioning in gantry crane systems. The hybrid control system combines Zero-vibration (ZV) and Zero-Vibration Derivative (ZVD) input shapers for sway suppression. Then a Proportional- Integral Derivative (PID) controller is designed for trolley position control. The effectiveness of the hybrid controller was studied and investigated by simulating the proposed hybrid ZV-ZVD shaper for sway suppression and the PID controller for precise position control using MATLAB/Simulink. Experimental results show that hybrid ZV-ZVD was able to suppress the pay load sway to 99.69% and the combination of hybrid ZV-ZVD and PID controllers offer precise trolley position control.

Keywords: Zero-Vibration (ZV), Zero Vibration Derivative (ZVD), Proportional Integral Derivative (PID), Gantry Crane (GC)

1. INTRODUCTION

Gantry Crane (GC) finds applications in many industries such as transportation, construction, manufacturing, power, etc. It is used in transport industry for the loading and unloading of freight, in construction industry for the movement of materials, in manufacturing industry for assembling of heavy equipment, and in power industry for the movement of heavy transformers.

In their early days, cranes were manually operated. But manual operation quickly became difficult when cranes became larger and faster (Al-Mousa, 2000). Due to this, efficient controllers are applied to the crane system to guarantee fast turnover and to meet safety requirement.

The typical problems associated with GC that have major adverse effects on its safety and performance include (Thalapil, 2012):

1. Lack of precise trolley positioning.
2. Load sway (vibration) during motion.

These problems can drastically reduce productivity and efficiency as well as heighten safety risks during operations. In addition, a residual sway occurs at the end of the trolley movement leading to inaccurate positioning of the load. These problems have been identified as bottlenecks in the operations of the transportation industry. Similar scenarios exist in other industries where GC finds application, hence the need for precise positioning and deployment of anti-sway controllers.

This work aims to control sway in GCs using hybrid feed forward controllers. A hybrid of Zero Vibration (ZV) and Zero Vibration Derivative (ZVD) input shaping techniques is utilized to suppress vibration.

Zero Vibration (ZV) is a method of shaping input signal (Singhose, 2009) for a flexible or vibratory system to suppress vibration. The application of this method requires the system natural frequency and damping ratio in order to estimate the amplitude gains and time locations respectively. A ZV system can be designed to have desired low-pass property to avoid excitation of high frequency system modes and reduce the possibility of saturation problems (Zhu et al., 2014).

The most important advantages of this method are:

1. Only the output signals of the system are required. As the model information is not needed, the problem of model uncertainty is avoided completely.
2. Adequate damping and bandwidth of the whole system can be chosen to yield desired system dynamics.
3. For multimode, high order filters can easily be designed (Han et al., 2015).

On the other hand, this work deploys the proportional-Integral-Derivative (PID) feedback controller for precise input tracking.

PID is the most common and most popular feedback controller used in industrial process today. At its input, PID controller gets signal from the output sensor which is referred to as actual process variable; it also accepts the desired actuator output, which is referred as set variable. The PID then evaluates the proportional, integral and derivative of the error signal which is then used to compute the control signal for deriving the actuator. The controller attempts to minimize the error by adjusting the process through use of a manipulated variable of future errors based on current rating of change (Silswal, 2012). PID controller is also known as three-term control: the proportional (P), integral (I) and derivative (D). By tuning the gains of these three parameters in the PID controller algorithm, the controller can adapt its control action to suit a specific process requirement. Despite the significant developments in advanced process control schemes such as predictive control, internal model control, sliding mode control, etc., PID controllers are still widely used in industrial control application because of their structural simplicity (Rajinikanth and Latha, 2012) good performance (Pradeepkannan and Sathiyamoorthy, 2014) and ease of implementation (Bansal et al., 2012)

In Section 2, this paper reviews the literature in this domain. In Section 3, the paper presents a new hybrid method for GC control. Section 4 presents experimental evaluations of the proposed method, its results and discussions. The paper concludes in Section 5.

2. LITERATURE REVIEW

This section presents review of the works done by other researchers in the control of Gantry crane (GC) system.

Ahmed (2009) developed a hybrid fuzzy logic control with input shaping for input tracking and sway suppression of a gantry crane system. The investigation deployed a Proportional-Derivative (PD)-type fuzzy logic control for cart position control of a gantry crane. It was then extended to incorporate input shaper control schemes for anti-sway control of the system. The positive and new modified Specified Negative Amplitude (SNA) input shapers were designed based on the properties of the system for control of system sway. The new SNA was proposed to improve the robustness capability while increasing the speed of the system response. Simulation results revealed that a significant reduction in the system sways was achieved with the hybrid controllers regardless of the polarities of the shapers.

Ezuan (2013) developed a hybrid input shaping for anti-sway control of a three degree-of freedom (3-DOF) rotary crane system. To study the effectiveness of the controllers, initially a Linear Quadratic Regulator (LQR) control is developed for the tower rotation angle of the rotary crane. This controller is then extended to incorporate input shaping techniques for anti-swaying control of the system for different payload. Input shaping Positive Zero Vibration (PZV) and Positive Zero Vibration Derivative-Derivative (PZVDD) were designed based on the properties of the system. The performances of input shaping in hybrid control schemes were examined in terms of level of input tracking capability, swing angle reduction, and time response specifications. Acceptable anti-sway capability was achieved with both control strategies. A comparison of the results demonstrated that the PZVDD shapers provided higher level of sway reduction as compared to the cases using PZV shapers for different payload. In addition, by using the PZVDD shapers, the overshoot was reduced but with slower response as compared to the unshaped system.

Alhazza and Masoud (2013) proposed a waveform command shaper for overhead cranes. The response of the system to the proposed shaper was derived analytically and simulated numerically. The performance of the proposed shaper was further validated experimentally on a scaled model of an overhead crane. The design frequency of wave-form (WF) shaper was found to be independent of the maximum allowable crane acceleration. Numerical and experimental results demonstrated the ability of the WF shaper in eliminating inertia induced oscillation.

Mohammad (2015) proposed a simple but efficient technique to control 3D overhead crane. The method was based on the position error and projection of the swing angle to design crane controller. No plant information of crane is necessary in this approach. Therefore, the proposed method greatly reduces the computational efforts. Simulation results showed that the proposed method can greatly restrain the swing.

Conker et al. (2016) extensively reviewed command pre-shaping methods and investigates the compromise between rapidity of motion and shaper robustness. In total, the Author reviewed and compared the performances of fifteen (15) different input shaping methods. The reviewed methods cover almost all types of positive shapers and smoothly shaped reference commands reported in literature.

Auwalu et al. (2018), developed a hybrid controller which combines zero vibration derivative (ZVD) shaper with fuzzy logic control (FLC) technique for load hoisting control of a 2-D crane. In their work, ZVD was designed to minimize payload sway motion while, the FLC was designed for trolley positioning. Two different approaches were used for the design of the ZVD, first the ZVD was designed with parameters of the system at maximum hoisting length and secondly, with system's parameters at average travel hoisting length (ATL) to enhance sway reduction. Mean absolute error (MAE) of the payload sway was used as performance index of the controllers. The MAE values of the sway motion for crane with load hoisting without controller, with FLC, with FLC and ZVD and with FLC and ATL were found to be 11.3597, 3.5583, 2.0853 and 1.3508 degrees respectively. The simulation results revealed that the proposed hybrid control technique achieved precise payload position with acceptable payload sway reduction.

Recently, Tung et al. (2019) presented a control approach to a flexible gantry crane system. The equations of motion that characterized coupled transverse-transverse motions with varying rope length of the gantry is obtained From Hamilton's extended principle. The process results in a system of ordinary and partial differential equations including cable dynamics and boundary conditions at trolley and payload ends. Control forces that solve the control problem are designed based on Lyapunov's direct method. The effectiveness of the proposed control scheme was verified through extensive numerical simulations.

In another work, Tumary et al. (2013) compares the performance of various input shapers including positive sway zero vibration shaper and its higher derivatives (PZS), PZSD and PZSDD on pendulum angle sway suppression. The authors also deployed PID system to control the cart position of the gantry crane. Their investigation revealed that the higher derivative input shaper, PZSDD, achieves the most reduction of the pendulum sway, whereas the input shaper with lowest derivative (PZS) yields fastest tracking speed. It is important to note that, although their work (Tumary et al., 2013) was tagged hybrid, it did not actually hybridize multiple input shapers for the sway suppression. The method rather considers the use of PID controller in cart-position control and an input shaper (PZS, PZSD, or PZSDD) for sway suppression as a form of hybridization of Gantry control system.

From the literatures, it was observed that hybrid control in GC is a common practice. However, they are so far characterized by slow rise- and settling times due to delays resulting from more impulses of the input shaper derivatives. Hence, in the following section, this work proposed a new hybrid architecture that hybridizes two input shapers for sway suppression together with a PID for position control. A hybrid architecture that toggles between ZV (faster rise time) and ZVD (faster settling time) input shapers is designed to suppress GC sway. Also, a PID controller is applied for position control of the trolley.

3. Materials and Method

This section presents the method for the design of the input shapers (ZV and ZVD), the design and tuning of the PID controller and finally the development of the proposed hybrid scheme which comprised ZV-ZVD and PID controllers.

Figure 1 shows the model of two-dimensional gantry crane system with its payload considered in this study, where x is the horizontal position of the trolley, l is the length of the rope, θ is the sway angle of the rope, M and m are the masses of the trolley and payload respectively (King, 2006).

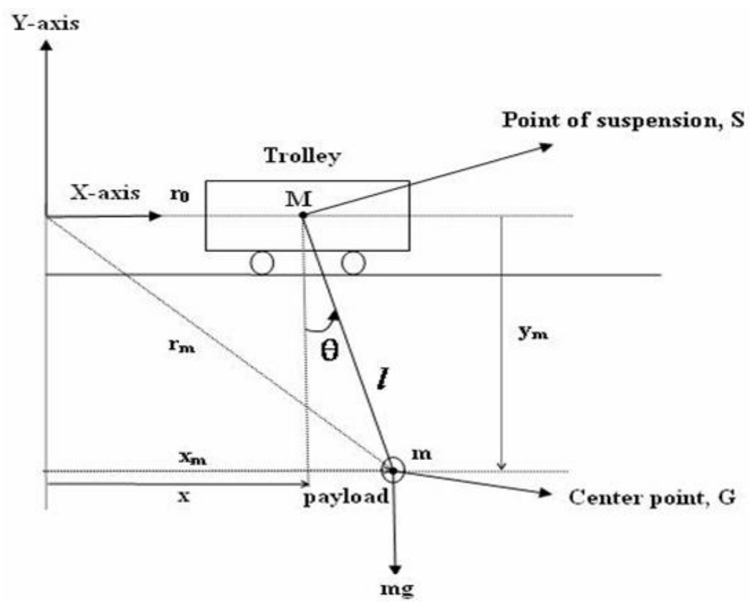

Figure 1: Model of a Gantry Crane (King, 2006)

3.1. System Model Description and Derivation of Dynamic Equations

The key assumptions made to simplify the process of modeling the gantry crane system include:

(a) Trolley frictional force was ignored.

(b) The trolley and the payload were considered as point masses.

(c) The tension force that may cause the hoisting rope elongate was considered negligible.

(d) The trolley and the payload were assumed to move in two dimensions only, i.e. x-y plane (King, 2006).

Where, M is the trolley mass in kg, m is the is the payload mass in kg, l is the length of hoisting rope in m, x is the trolley position in m, θ is the sway angle in rad, r_m is the position vector of center point G, r_0 is position vector of point of suspension S, x_m is horizontal position of payload, y_m is the vertical position of payload.

Based on Figure 1, the load and trolley position vectors are given by:

$$\bar{r}_m = \{x + l\sin\theta, -l\cos\theta\} \quad (1)$$

$$\bar{r}_0 = \{x, 0\} \quad (2)$$

The kinetic energy (K.E) of the system can thus be formulated as:

$$K.E = K.E_{Trolley} + K.E_{Payload} \quad (3)$$

$$= \frac{1}{2}M\,\dot{r}_0^2 + \frac{1}{2}M\,\dot{r}_m^2$$

$$= \frac{1}{2}M\,\dot{x}^2 + \frac{1}{2}m\,(\dot{x}_m^2 + \dot{y}_m^2) \quad (4)$$

The potential energy of the system, P can be represented as:

$$P.E = mgy_m = -mgl\cos\theta \quad (5)$$

where, g is the gravitational acceleration or g = 9.81ms-2.

Using the Lagrangian function,

$$L = K.E - P.E \quad (6)$$

The equations of motion of the gantry crane model associated with the generalized coordinates $\bar{q} = \{x, \theta\}$ can be summarized, respectively as:

$x: \rightarrow$

$$F_x = (M+m)\,\ddot{x} + ml(\ddot{\theta}\cos\theta - \dot{\theta}^2\sin\theta + 2m\,\dot{l}\,\dot{\theta}\cos\theta + m\,\ddot{l}\sin\theta)$$

$$\theta:\; l\,\dot{\theta} + 2\,\dot{l}\,\dot{\theta} + \ddot{x}\cos\theta + g\sin\theta = 0$$

3.1.1. Linearization of the system model

The above model is a nonlinear model. Therefore, for a better progress of modeling, the nonlinear dynamic model has to be linearized.

For safe operation, two assumptions were made. Firstly, it was assumed that the swing angle was kept small. So that:

$\theta \approx 0$

$\dot{\theta} \approx 0$

$\sin\theta \approx 0$

$\cos\theta \approx 0.$

Secondly, since the tension force that may cause the hoisting rope elongate was neglected, the length of the hoisting rope was assumed to be constant, which is:

$\dot{l} \approx \ddot{l} \approx 0.$

Using these two assumptions, the simplified equation of motion for the gantry crane system can be obtained as:

$$x: \rightarrow F_x = (M+m)\,\ddot{x} + ml\,\ddot{\theta} \quad (7)$$

$$\theta: \quad l\,\ddot{\theta} + \ddot{x} + g\theta = 0 \quad (8)$$

3.1.2. State space representation of the system

The state equation is

$\dot{x} = Ax + Bu$

$$| x | \ldots \frac{}{M} \ldots | + | \frac{}{M} | F_x \quad (9)$$

The output equation is,

$y = Cx + Du$

$$\ldots = \ldots \quad (10)$$

In this study, the length of hoisting rope, $(l) = 0.75\,m$, trolley mass $(M) = 3$ kg, payload mass $(m) = 0.75$ kg and $g = 9.81$ ms^{-2} were considered as in (Haliru, 2019). Having obtained the model of the system dynamics, simulation and analysis of the gantry crane system can be performed.

3.2. Design of Simulink block of Nonlinear Crane System to Determine ω_n and ζ

The natural frequency ω_n and the Damping ratio, ζ of the system are determined by using Curve fitting toolbox in the MATLAB software, with the input and output data obtained from the nonlinear crane system. Figure 2 shows the Simulink Block diagram of the dynamic equation.

After the simulation, the desired values of the system's natural frequency and damping ratio are derived from the curve fitting results shown in Figure 3 as follows:

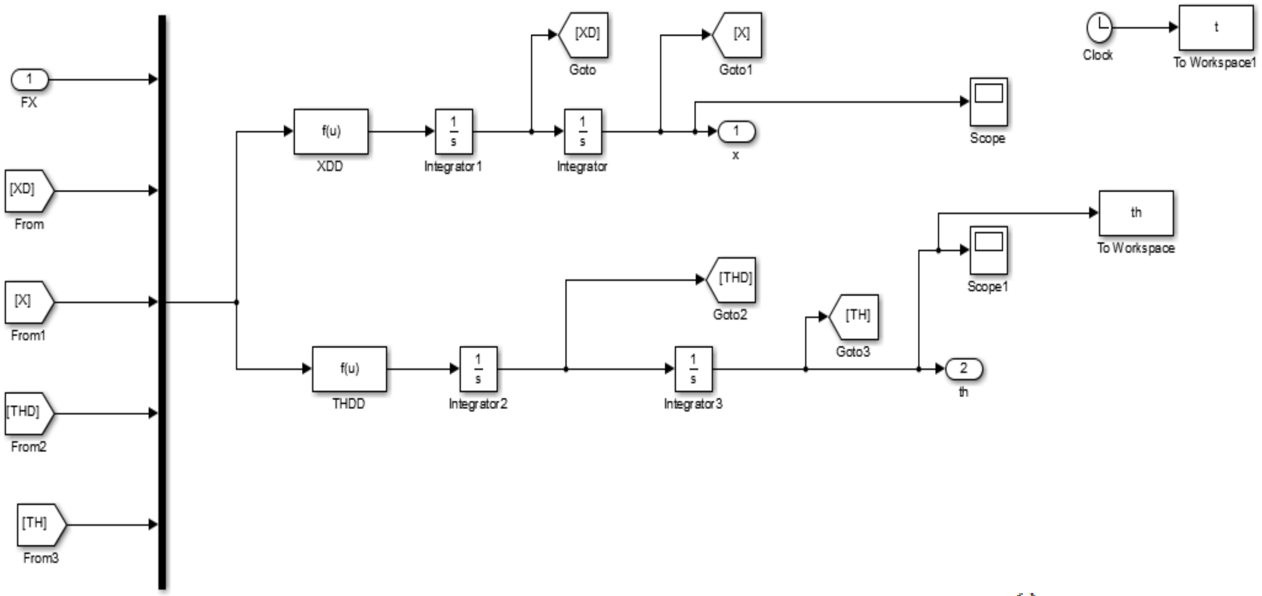

Figure 2: The Simulink Block diagram of nonlinear Crane system for determine determining ω_n and ζ

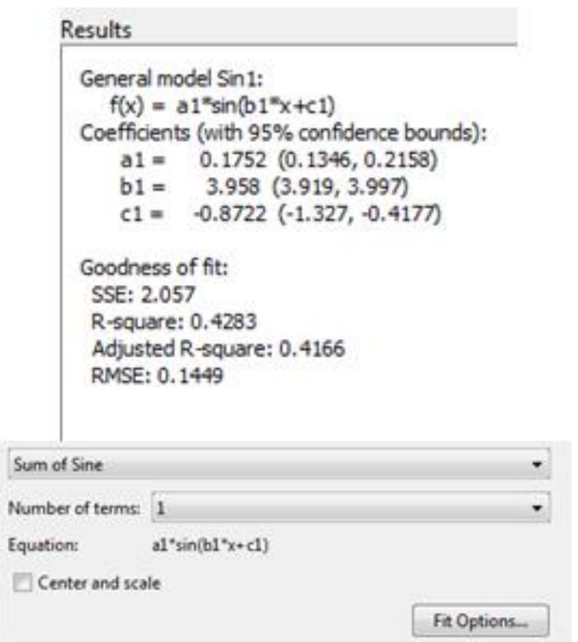

Figure 3: The Curve Fitting Toolbox Output for Determining ω_n and ζ

From Figure 3,
$b1 = \omega_n$

From the general model, $f(x)$ is then rewritten as:

$$f(x) = a_1 \exp(-ax)\sin(b_1 x + c_1)$$

And, ζ =0.03726 (0.005199, 0.06932)

$\omega_n = 3.985\,\mathrm{rads}^{-1}$ and the ζ = 0.03726

To calculate the damped natural frequency, ω_n equation 1 is applied

$$\omega_d = \omega_n \sqrt{1-\zeta^2} \tag{11}$$

$$\omega_d = 3.985\sqrt{1-0.03726^2} == 3.9822 \text{ rad s}^{-1}$$

To estimate the stiffness constant, K, equation (12) is applied

$$K = e^{-\frac{\pi\zeta}{\sqrt{1-\zeta^2}}} \tag{12}$$

$$= e^{-\frac{\ldots}{\ldots}} = 0.8895.$$

3.3. Design of ZV and ZVD Input Shapers for Sway suppression

The amplitudes, A_i and time locations, t_i of the ZV shaper are obtained with the aid of K and ω_d respectively.

$$A_1 = \frac{1}{1+K} = \frac{1}{1+0.8895} = \frac{1}{1.8895} = 0.5292$$

$$A_2 = \frac{K}{1+K} = \frac{0.8895}{1+0.8895} = \frac{0.8895}{1.8895} = 0.4708$$

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$t_{1=0}$, $\quad t_2 = \dfrac{\pi}{\omega_d} = \dfrac{\pi}{3.9822} = 0.7889$

Figure 4 shows the Simulink block diagram of the ZV shaper with the values obtained.

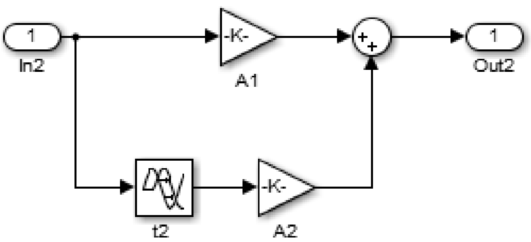

Figure 4: Simulink block of ZV Shaper

The A_i and t_i of the ZVD shaper:

$$A_1 = \frac{1}{1 + 2K + K^2}$$

$$= \frac{1}{1 + 2(0.8895) + 0.8895^2} = 0.2801$$

$$A_2 = \frac{2K}{1 + 2K + K^2} = \frac{1.779}{3.5702} = 0.4983$$

$$A_3 = \frac{K^2}{1 + 2K + K^2} = \frac{0.7912}{3.5702} = 0.2216$$

$t_1 = 0, \rightarrow t_2 = \dfrac{\pi}{\omega_d} = \dfrac{\pi}{3.9822} = 0.7889$

$t_3 = \dfrac{2\pi}{\omega_d} = 1.5778.$

Figure 5 shows the Simulink block diagram of the ZV shaper with the values obtained.

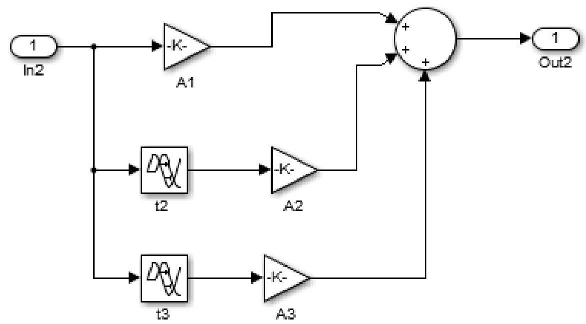

Figure 5: Simulink block of ZVD

3.4. Design and Tuning of PID Controller for Position Control

The goal in this section is to find the values of the gains K_p, K_i and K_d that give satisfactory result. Figure 6 represents the Simulink block diagram of the GC hybrid Shaper and PID that was used to design and tune PID controller.

The tuning method employed was the auto tuning tool of the optimization tool in Matlab to obtain the best optimal values for K_p, K_i and K_d.

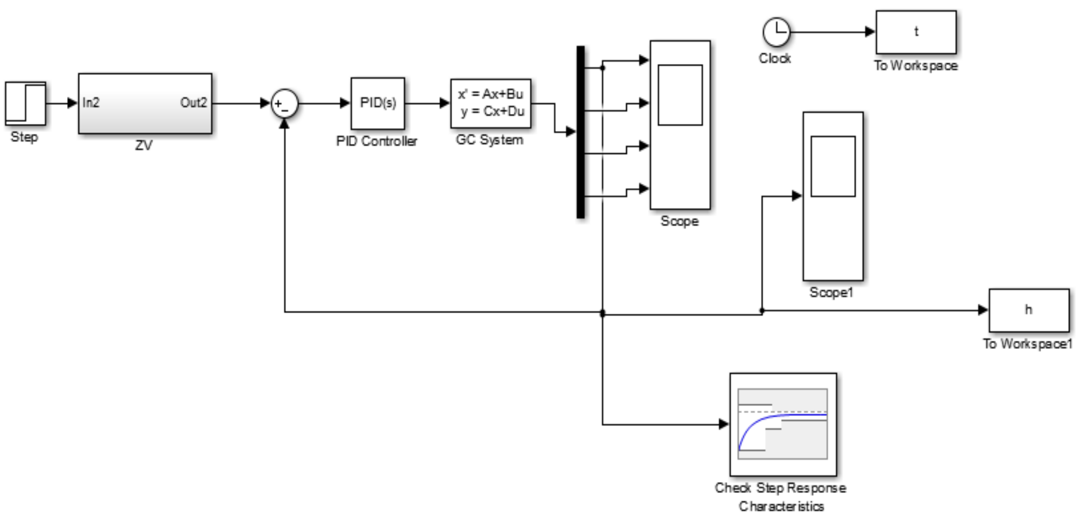

Figure 6: Simulink Model for Optimization PID Parameters for Gantry Crane Trolley Position Control

The optimization tool converged and returned the following gains as optimal values of the PID parameters (Table 1).

Table 1: Optimum Values of PID gains Obtained using Matlab

S/N	PID Gains	Optimal Value
1	K_p	0.00101000
2	K_i	0.00001000
3	K_d	5.00000002

3.5. Configuration of the Proposed Hybrid Controller Combining Two Input Shapers

Since on one hand, the ZV shaper is characterized by fast rise time due to fewer impulses associated with it but has a disadvantage of long settling time. On the other hand, the ZVD shaper with higher number of impulses has a longer rise time but with fast settling time. In order to achieve fast rise and settling time coupled with improved steady state performance, a switching circuit was designed to initially track the input shaper with shorter rise time (ZV) until the 90% of the desired response is reached. Thereafter, the system automatically toggles to an input shaper (ZVD) having a shorter settling time and minimal steady state error. Figure 7 shows the configuration of the proposed architecture hybridizing the ZV-ZVD input shapers and a PID controller for the control of Gantry crane system. In the next section, the proposed system is simulated and evaluated against traditional its traditional counterparts.

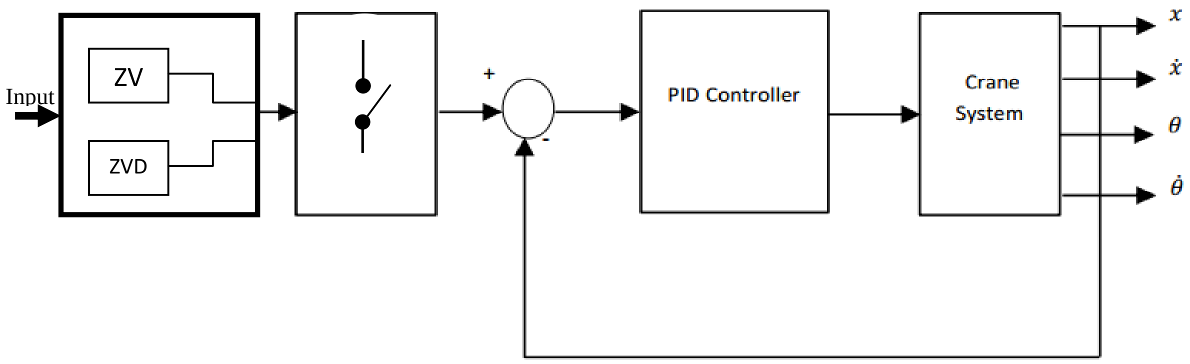

Figure 7: Block Diagram of the Hybrid controller combining two input shapers

4. RESULTS AND DISCUSSION

4.1. Experimental Setup

This section presents an experimental simulation and evaluation of the proposed hybrid controller. The Matlab/Simulink software is used to simulate the GC model operation and the designed controllers.

Figure 8 shows the Simulink block diagram of the experimental setup. It comprised the GC model controlled with a PID alone, PIDZV, PIDZVD and the proposed hybrid PID/ZV-ZVD. The output of first three controllers formed the basis for the design of the switching circuit for the hybrid PID/ZV-ZVD. The hybrid system was designed to initially take the advantage of shorter risetime of ZV to track position until 90% rise time and then switch to ZVD with shorter settling time at minimum of 0.75s. The performance evaluation of the proposed controller is compared with that of the aforementioned individual PID, PID-ZV and PIDZVD, controllers.

4.2. Simulation Results

This section presents and discusses the results of simulations of the system; with PID-ZV, PID-ZVD and hybrid PID/ZV-ZVD controllers carried out on MATLAB (R2015a) software platform.

Firstly, the speed and input tracking capability of the proposed method is investigated. Figure 9 shows the trolley position control responses using PID controller alone, PIDZV, PIDZVD and the hybrid PID/ZV-ZVD.

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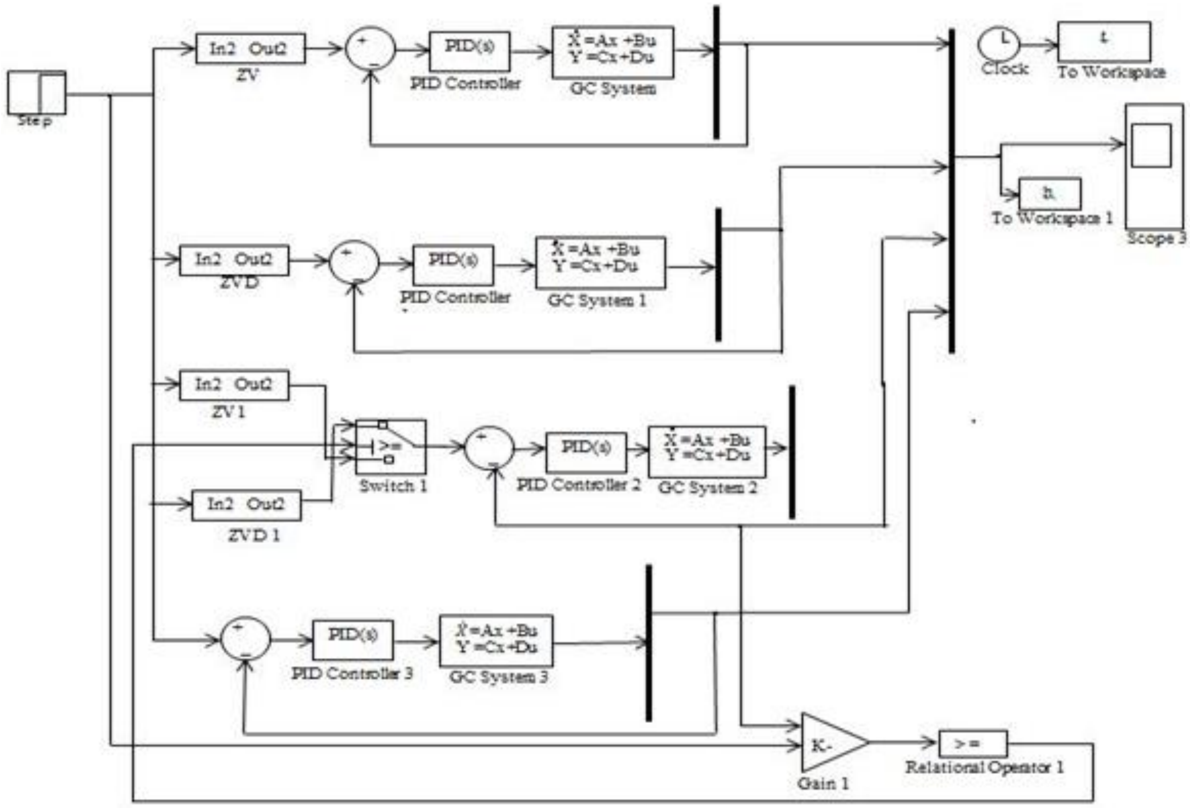

Figure 8: Simulink Model: Performance Comparison of the Proposed Hybrid ZV/ZVD with PID Controller with non-hybrid Input Shapers and PID alone

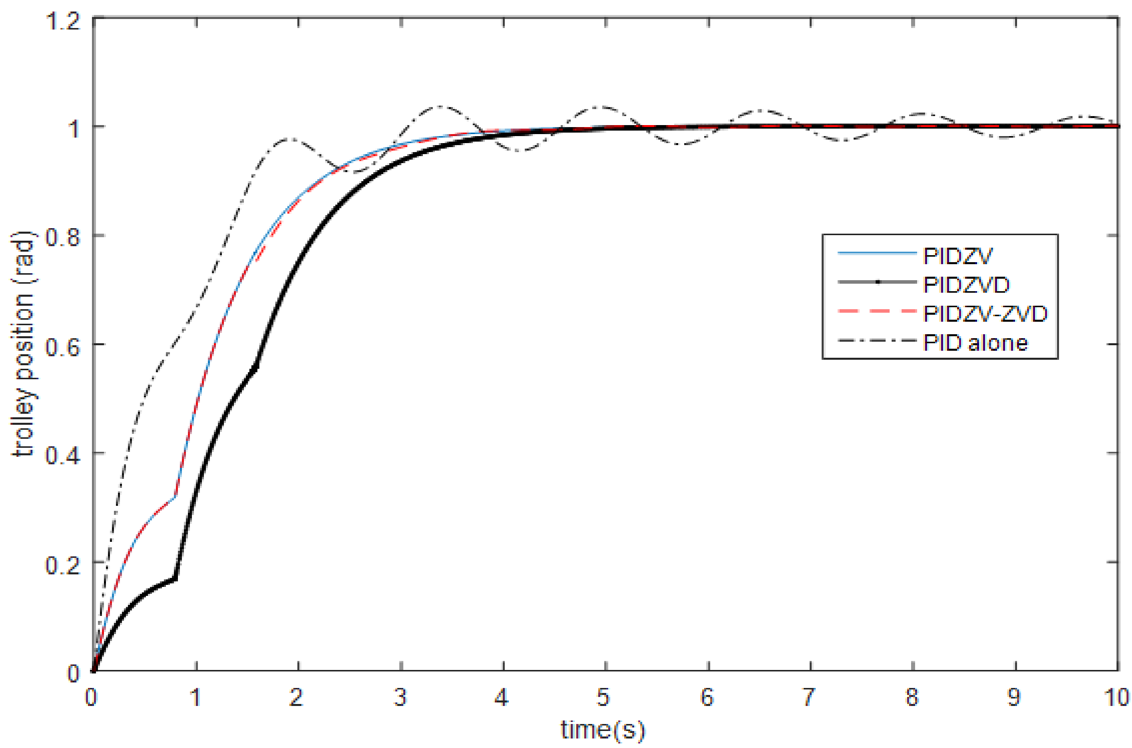

Figure 9: Trolley Position Control Responses with Hybrid PID Shapers

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Table 2: Performance Comparison of the Hybrid PID/ZV-ZVD Controller against its Nonhybrid Counterparts

Controller	Rise time (s)	Peak time (s)	Overshoot (%)	Settling time (s)
PID	1.4738	1.923	14.5	-
PIDZV	2.1027	-	0	4.513
PIDZVD	2.3923	-	0	3.852
PIDZV-ZVD	2.0652	-	0	3.369

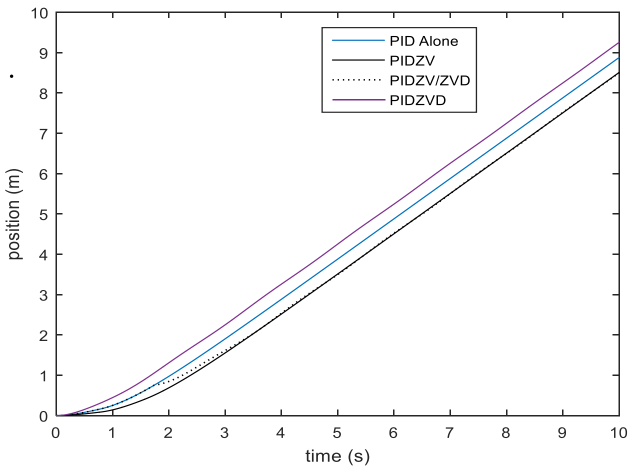

Figure 10: Gantry Crane Tracking Response to a Ramp Input Excitation

In this experiment, the trolley was excited to move to the position 1 m away from the origin. The uncontrolled GC behaves as an open loop unstable system while for the controlled system, the performance parameters are as shown in Table 2. From Table 2, it is evident that, the proposed hybrid ZV-ZVD/PID controller has an improved performance both in speed and accuracy of position tracking (rise/settling time of 2.0652s/3.369s) as compared to the non-hybrid ZV/PID and ZVD/PID controllers with 2.1027/4.513s and 2.3923/3.852s respectively.

Similar behaviour is observed when the gantry crane is excited with a ramp and bang-bang inputs as shown in Figures 10 and 11 respectively.

Secondly, the efficacy of sway suppression by the input shapers is also investigated. The ZV is adopted initially due to its fast response and then ZVD for its better suppression characteristics. Consequently, the proposed ZV-ZVD produced an improved response both in speed and pay load sway suppression. Figure 12 shows the sway angle response comparisons with and without input shapers. It can be noticed from Table 3 that sway suppression is achieved in ascending order of perfection ranging from 93.84% by ZV, 98.797% by ZVD and 99.69% by the proposed hybrid of ZV-ZVD.

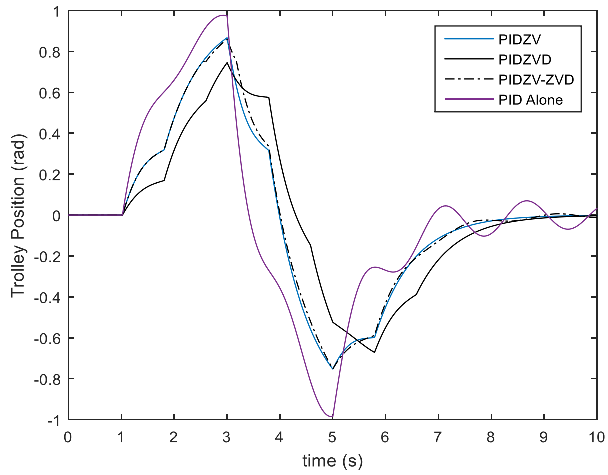

Figure 11: Gantry Crane Tracking Response to a Bang-Bang Input Excitation

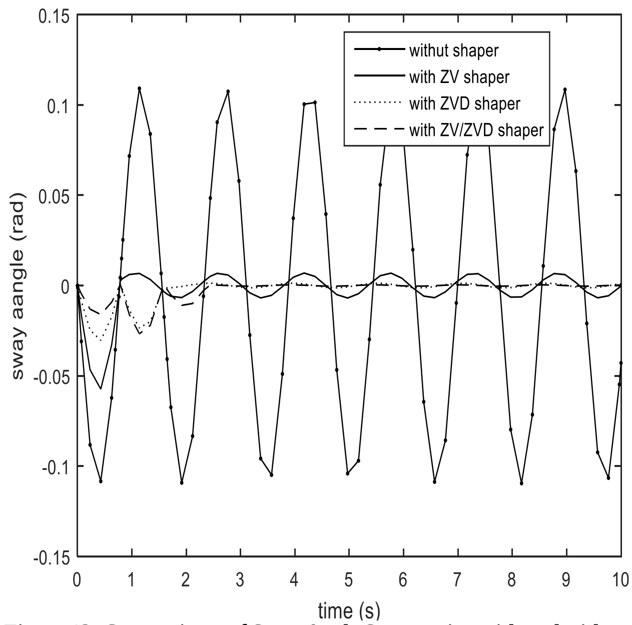

Figure 12: Comparisons of Sway Angle Suppression with and without Input Shapers

Table 3: Comparison of Vibrations Suppression with Input Shapers

Input Shaper	Peak-Peak sway (rad)	Reduction (rad.)	Percentage Reduction (%)
Uncontrolled	0.2174	-	-
ZV	0.0134	0.204	93.84
ZVD	0.0026	0.2148	98.80
ZV-ZVD	0.0007	0.2167	99.69

5. CONCLUSION

This work presented application of hybrid controllers for sway suppression and precise trolley position control in gantry crane system. Performance evaluation of all the controllers was carried out on MATLAB/Simulink software platform and the result showed that the proposed hybrid ZV-ZVD and input shaper was able to suppress the sway appreciably resulting in up to 99.6% reduction in sway angle. Also, a more precise trolley position control was achieved with the Hybrid controllers (ZV-ZVD and PID).

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