TITLE PAGE

DEVELOPMENT OF A SWAY CONTROL SCHEME BASED HYBRID INPUT SHAPING AND PID CONTROLLERS FOR GANTRY CRANE SYSTEM

BY
IBRAHIM UMARU
(SPS/15/MEE/0029)
M.ENG. ELECTRICAL ENGINEERING
(CONTROL AND INSTRUMENTATION OPTION)

A DESSERTATION SUBMITTED TO THE DEPARTMENT OF ELECTRICAL ENGINEERING, BAYERO UNIVERSITY, KANO, IN PARTIAL FULFILMENT OF THE REQUIREMENT FOR THE AWARD OF DEGREE OF MASTER OF ENGINEERING (M.ENG.) IN ELECTRICAL ENGINEERING

FEBRUARY,2020

DECLARATION

I hereby declare that this work is the product of my research efforts undertaken under the supervision of Dr. Hassan A. Bashir and has not been presented anywhere for the award of a degree certificate. All sources have been duly acknowledged.

……………………………………… Date…………………………

Ibrahim Umaru
SPS/15/MEE/00029

CERTIFICATION

This is to certify that the research work for this dissertation and the subsequent write-up by (Ibrahim Umaru, SPS/15/MEE/00029) were carried out under my supervision.

……………………………………. Date…………………………………

Dr. Hassan A Bashir
Supervisor

……………………………………. Date………………………………

Professor NuraddeenMagaji
Head of Department

APPROVAL PAGE

This dissertation has been examined and approved for the award of Masters in Electrical engineering (Control and instrumentation).

……………………………………. Date…………………………………
Supervisor

……………………………………. Date………………………………
Head of Department

……………………………………. Date………………………………
Internal Examiner

……………………………………. Date…………………………………
External Examiner

……………………………………. Date…………………………………
Dean, School of Postgraduate studies

ACNOWLEDGEMENT

First and famous, I give special thanks to Allah for His guidance and protection throughout the period of this work.

My gratitude goes to my supervisor; Dr. Hassan A Bashir for his guidance, support and encouragement. My gratitude also goes to the entire staff of Department of Electrical Engineering, Bayero University, Kano for their support and cooperation.

Also, my sincere gratitude goes to my parents for their unquantifiable efforts and to my wife for her understanding and cooperation throughout the period. Jazakallahukhaira.

Finally, I wish to express my appreciation to my family, friends and well-wishers too numerous to mention.

DEDICATION

Dedicated to my parents

TABLE OF CONTENTS

CHAPTER THREE

MATERIALS AND METHODS

CHAPTER FOUR

RESULTS AND DISCUSSION

CHAPTER FIVE

SUMMARY, CONCLUSION AND RECOMMENDATIONS

LIST OF FIGURES

LIST OF TABLES

LIST OF ABBREVIATIONS, DEFINITION AND SYMBOLS

Abbreviation/Symbols	Definition
A_i	Impulse Amplitude
a_i	Filter Gains
AGVs	Automated Guide Vehicles
C.S	Control System
$c(t)$	Shaped Command signal
D	Derivative
3D	Three- Dimension
2DOF	Two Degree of Freedom
3DOF	Three Degree of Freedom
EI	Extra-Insensitive
f	Intensity of iteration
F(s)	Transfer Function of the input shaping
F(t)	Unshaped reference command
g	Gravitational Acceleration
GC	Gantry Crane
GDP	Gross Domestic Product
G(s)	Transfer Function of the target plant
I	Integral
IS	Input Shaping
K_d	Derivative gain
K.E	Kinetic Energy
K_i	Integral Gain

K_p	Proportional Gain
m	Mass of the Payload
M	Mass of the Trolley
MPC	Model Predictive Control
M(s)	Transfer Function of the reference Model
n	Number of Impulse
NCTF	Normal Characteristic Trajectory Following
P	Proportional
PC	Personal Computer
P.E	Potential Energy of the System
PID	Proportional- Integral Derivation
RBFNs	Radial Basis Function Network
S	Point of Suspension
t_0	Time the Impulse is applied
V	Residual Single-mode Vibration Amplitude
ω	Frequency
ω_d	Damped Frequency
ω_n	Natural Frequency
x	Horizontal Position of Trolley
$y_r(t)$	Unit step Response function of reference model
ZV	Zero Vibration
ZVD	Zero-Vibration Derivative
Δ_i	Time Delay
ζ	Damping Ratio
ϑ	Sway Angle

ABSTRACT

Gantry crane is a machine 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, but operating GCs as fast as possible results in payload sway coupled with inaccurate positioning of the load which degrades the speed, accuracy and safety. This research applies hybrid control system for sway suppression and precise trolley position control in gantry crane systems. The hybrid control system proposed in this work combines Zero-vibration(ZV) and Zero-Vibration Derivative(ZVD) input shaper 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. Simulation results show that hybrid ZV-ZVD was able to suppress the pay load sway to 99.69% and the combination i.e Hybrid ZV-ZVD and PID controllers offer precise trolley position control.

CHAPTER ONE

INTRODUCTION

1.1 Background Information

Gantry Crane (GC) finds application 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 the assembling of heavy equipment, and in power industry for the movement of heavy transformers.

Main 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 cause safety threat in its areas of application. For instance, in ports; since containers were introduced to the world-trade industry, an increasing number of goods are being put into these containers and loaded onto vessels to be carried to their respective destinations across the world. When a vessel arrives at a port, containers destined for the port must be unloaded and new containers which are bound for other ports must be loaded before the vessel can resume its trip. Demands on container ports to perform the loading and unloading process with maximum efficiency will become greater as transport companies continue to increase both the size of their fleets and also the capacity of the vessels. The problem with this increase is a pointer for port authorities that they will run out of space and the option to expand their operational areas on surrounding land is often hampered from local residents(Hong and Ngo, 2009). Therefore, a possible solution will be the reduction of the amount of time that the vessel needs to remain in dock. The way to do this is to ensure that the loading/offloading process is done as rapidly as possible, and this can be achieved by ensuring that the equipment in the port such as gantry cranes (GCs), automated guide vehicles (AGVs), and so on, can operate at their maximum efficiencies.

The loading/offloading process from/to the vessel to/from yard is performed by GCs, trucks or AGVs. One to six cranes are assigned to pick up containers depending on the vessel size (Hong and Ngo, 2009).

Each crane will be responsible for several rows of container storage slots along the ship. For each row, the GC must first offload all the containers destined for this port, and then load all the containers scheduled for leaving this port. Containers will be carried between the GCs and the yard by AGVs or trucks. An increasing number of trucks or AGVs will satisfy offloading/loading container demand, but because of the space limitation, more GCs cannot be assigned to serve the vessel. Thus, the offloading/loading efficiency depends on how fast the container is offloaded/loaded by GCs. However, the fast movement of trolley of GC sways the container during its movement (load swinging or vibration during motion).

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/anti-sway controllers.

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

This research focuses on GC control using hybrid controllers consisting of feed forward and feedback controllers. Hybrid Zero Vibration (ZV) input shaping technique is utilized to suppress vibration.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).

The feedback controllers to be utilized for precise input tracking are: proportional-Integral-Derivative (PID) controller.

PID is the most common and most popular feedback controller used in industrial process today. A PID gets the input parameter from the sensor which is referred to as actual process variable, it also accepts the desired actuator output, which is referred as set variable and calculate and combines the Proportional, Integral and Derivative responses to compute the output for 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 provide control action designed for specific process requirements.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, reputation and easy implementation. The merits of PID controllers are as follow:

1. It is obtainable in variety of structures such as; PD, PI, series PID, parallel PID and Internal Model Control-PID.
2. It supports online/offline tuning based on the process performance requirement.
3. It can be applied for advanced arrangements such as 2DOF and 3DOF (Rajinikanth and Latha, 2012).
4. It does not depend on system model but can be integrated into it.

1.2 Statement of the Problem

Three factors of paramount importance in crane operations include speed, accuracy, and safety (Thalapil, 2012).The faster the load is moved, the shorter the time it takes to reach the final desired destination. But moving the load very fast will result in an unwanted sway. This sway is detrimental to safe and efficient operation. Swinging degrades the speed, accuracy, and safety of transport operations. It lowers the speed of transport operations because the payload swing must die out before the payload can be safely lowered into position. The swing makes it difficult to perform alignment, position refinements, or other accuracy driven tasks. Swing also causes safety problems because of potential collisions with neighboring objects or people. It is against this backdrop that this research focuses on sway suppression and precise positioning of trolley using simple, but very efficient controllers.

1.3 Research Gap

From the literatures reviewed, it was observed that hybrids so far used are characterized by slow rise-time and very slow settling times due to delays resulting from more impulses of the input shaper derivatives. Hence this research intends to bridge this identified research gap by applying this hybrid Input Shaper and PID with the aid of toggle switch capable of selecting ZV with faster rise time to ZVD with faster settling time, to achieve fastest speed as compare to the individual Shaper's rise time and settling time

1.4 Aim and Objectives of the Study

The aim of this study is to develop a hybrid Input shaping for swaysuppression and PIDto achieve precise input tracking in GCs. The following are the objectives:

i. To design hybrid ZV and ZVD input shapers controller for sway control of Gantry crane system.
ii. To design a PID controller for trolley positioning of a Gantry crane.
iii. To develop a hybrid control scheme using the hybrid ZV- ZVD shaper and PID controller for Gantry crane system.
iv. To evaluate the performance of the developed hybrid controller for the Gantry crane system using Mean Square Error, Rise time and Settling time as the performance metrics.

1.5 Significance of the Study

GCs have a wide range of applications in disaster sites, nuclear plants, railway yards, warehouses, construction sites, and shipyards. Thus, improving crane efficiency and safety can have a great impact on productivity (operational efficiency) of a wide variety of industries which in turn will boost national gross domestic product (GDP).

1.6 Justification of the Study

The urge to increase the travelling and hoisting speed of GCs generally induces undesirable sway of the payload, and serious damage could occur during load transportation. Precise trolley position control also plays important role in the operation of GCs system. Therefore, a satisfactory control scheme is desirable to suppress load sway and to ensure precise trolley position control.Zero Vibration (ZV)is a method of shaping input signal to 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. Using this method, filters can be designed not only for simple open loop system, but also closed loop control systems with complex frequency characteristics. Also, the whole system can easily 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 reasons for selecting PID controller include its simplicity of design, good performance which includes low percent overshoot and short settling time(Pradeepkannan and Sathiyamoorthy, 2014) and ease of re-tuning on-line(Bansal *et al.*, 2012).

1.7 Report Organization

This report is organized in five chapters. Besides the introduction in this chapter, Chapter two discusses the description of GC, Derivation of dynamic equation equations and state space representation of the model, Control schemes for GC systems, reviewed relevant literatures, concept of input shaping and PID tuning.. Chapter three gives the design of ZV and ZVD Input shaper, Design of hybrid input shapers and tuning of PID. Chapter four presents the results and discusses the findings in the results. Chapter five presents the summary, conclusion and recommendations on further research.

CHAPTER TWO

LITERATURE REVIEW

2.1 Introduction

This chapter presents review of control schemes for Gantry crane (GC). It provides theReview of Control Schemes for Gantry Cranes system, detail concept of input shaping, background of Proportional integral derivative (PID), System Model Description and Derivation of Dynamic Equations of a Gantry and Review of relevant Literature.

2.2 Review of Control Schemes for Gantry Cranes system

Gantry cranes (GCs) have been in existence for more than four decades ago and since then major developments have been taken place. In 2008, a GC called Taisun capable of lifting 20,000 metric tons was installed in Yantai, China at the Yantai Raffles Shipyard. In 2012, a 22,000-ton capacity crane, the 'Honghai Crane' planned for construction in Qidong City, China (Soleiman et al.,2009) was realized.

GC is one of the most important equipment used for transportation of heavy material in factories,warehouses, shipping yards, building construction and nuclear facilities. Usually, GC is characterized by very strong structure in order to lift heavy payloads. In factories, Crane speed up the production processes by moving heavy materials to and from the factory as well as moving the product along production or assembly lines. In construction works, cranes facilitate the transport of building materials to high and critical spots. Similarly, on ships and in harbors, Cranes save time and consequently money in making the process of loading and offloading ships fast and efficient (Karajgikar et al.,2011).

Initially, cranes were manually operated. However, manual operation became difficult when cranes became larger, faster and higher. This necessitates the needs for efficientcontrollers as a way of automating their operation to ensure fast turn over time and tomeet safety requirements. (King, 2006 and Al-Mousa 2000).

2.2.1 Description of Gantry Crane

There are three main components in a gantry crane which are trolley, bridge and gantry. Figure 2.1 shows a typical gantry crane (King, 2006). Trolley with a movable or fixed hoisting mechanism is the load lifting component. It moves on and parallel to a bridge which is rigidly fixed to a supporting structure called gantry. The gantry extends downward from the bridge to the ground where it can be mobilized on wheels or set of tracks. The motion of the gantry on the ground, the trolley on the bridge and the hoisting of the payload provide the 3 degrees of freedom of the payload.

Figure 2.1: Description of a Gantry Crane. (King, 2006)

2.2.2 System Model Description and Derivation of Dynamic Equations

Figure 2.2 shows the model of two-dimensional gantry crane system with its payload considered in this study, where x is the horizontal position ofthe trolley, *l* is the length of the rope, θ is the sway angleof the rope, M and m are the masses of the trolley and payloadrespectively (King,2006).

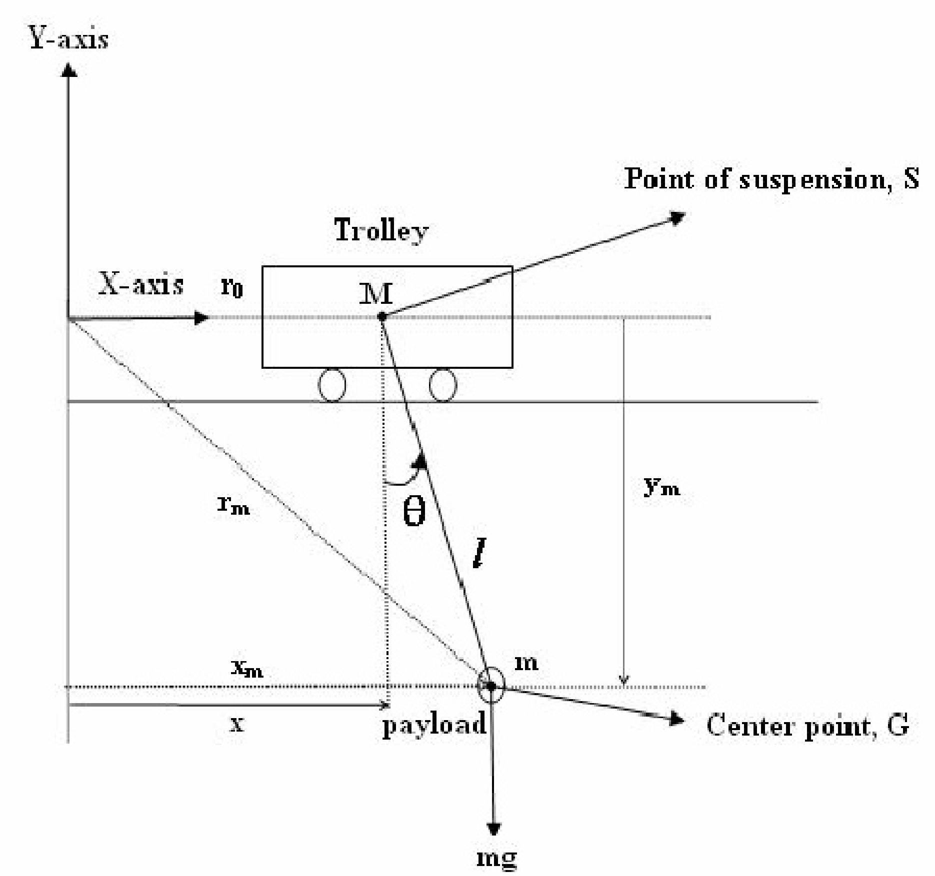


Figure 2.2: Model of a Gantry Crane (King, 2006)

Below are the assumptions to be made to simplify the process of system modeling.

(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 Position vector of center point G, r_o 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 2.2, the load and trolley position vectors are given by:

$$\bar{r}_m = \{x + l\sin\theta, -l\cos\theta\} \tag{2.1}$$

$$\bar{r}_0 = \{x, 0\} \tag{2.2}$$

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

$$K.E = K.E_{Trolley} + K.E_{Payload} \tag{2.3}$$

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

$$=\frac{1}{2}M\dot{x}^2 + \frac{1}{2}m(\dot{x}_m^2 + \dot{y}_m^2) \tag{2.4}$$

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

$$P = mgy_m$$

$$=-mgl\cos\theta \tag{2.5}$$

Where, g is the gravitational acceleration or g = 9.81ms^{-2}.

Using the Lagrangian function,

$$L = K.E - P.E \tag{2.6}$$

$$=\frac{1}{2}M\dot{x}^2 + \frac{1}{2}m(\dot{x}_m^2 + \dot{y}_m^2) + mgl\cos\theta \tag{2.7}$$

But $x_m = x + l\sin\theta$ and $y_m = -l\cos\theta$ Hence, the first derivative of x_m and y_m can be obtained as:

$$\dot{x}_m = \frac{dx_m}{dt}$$

$$= \frac{d(x + lsin\theta)}{dt}$$

$$= \dot{x} + \dot{l}sin\theta + \dot{\theta}(lcos\theta)$$

$$= \dot{x} + sin\theta\dot{l} + lcos\theta\dot{\theta}$$

and

$$\dot{y}_m = \frac{dy_m}{dt}$$

$$= \frac{d(-lcos\theta)}{dt}$$

$$= -[\dot{l}cos\theta + \dot{\theta}l(-sin\theta)]$$

$$= lsin\theta\dot{\theta} - \dot{l}cos\theta$$

Therefore,

$$\dot{x}_m^2 + \dot{y}_m^2 = (\dot{x} + sin\theta\dot{l} + lcos\theta\dot{\theta})^2 + (lsin\theta\dot{\theta} - \dot{l}cos\theta)^2$$

$$= \dot{x}^2 + \dot{l}^2 + l^2\dot{\theta}^2 + 2\dot{x}\dot{l}sin\theta + 2\dot{x}l\dot{\theta}cos\theta$$

Substituting for $\dot{x}_m^2 + \dot{y}_m^2$ in (3.7) we have:

$$L = \frac{1}{2}M\dot{x}^2 + \frac{1}{2}m\left(\dot{x}^2 + \dot{l}^2 + l^2\dot{\theta}^2 + 2\dot{x}\dot{l}sin\theta + 2\dot{x}l\dot{\theta}cos\theta\right) + mgl\cos\theta \qquad (2.8)$$

Let the generalized forces corresponding to the generalized displacements $\overline{q}=\{x,\theta\}$be $\overline{F}=\{F_x,0\}$. Constructing the Lagrangian$L = K.E - P.E$ and using Lagrangian's equation

$$\frac{d}{dt}\left(\frac{\partial L}{\partial \dot{q}_j}\right) - \frac{\partial L}{\partial q_j} = F_j \qquad j = 1,2 \qquad (2.9)$$

The equation of motion for the gantry crane system can be obtained.

Firstly, the equation of motion associated with the generalized coordinate $q = x$ can be derived as follows:

$$\frac{d}{dt}\left(\frac{\partial L}{\partial \dot{x}}\right) - \frac{\partial L}{\partial x} = F_x$$

Where, F_x is the input force (N).

$$\frac{\partial L}{\partial x} = 0$$

$$\frac{\partial L}{\partial \dot{x}} = M\dot{x} + \frac{1}{2}m\left(2\dot{x} + 2\dot{l}sin\theta + 2l\dot{\theta}cos\theta\right)$$

$$= M\dot{x} + m\dot{x} + m\dot{l}sin\theta + ml\dot{\theta}cos\theta$$

$$\frac{d}{dt}\left(\frac{\partial L}{\partial \dot{x}}\right) = M\ddot{x} + m\ddot{x} + m\left(\ddot{l}sin\theta + \dot{l}\dot{\theta}\ cos\theta\right) + m\left[\dot{l}\dot{\theta}cos\theta + l\ddot{\theta}cos\theta + l\dot{\theta}(-sin\theta)\dot{\theta}\right]$$

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

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

Thus, $\frac{d}{dt}\left(\frac{\partial L}{\partial \dot{x}}\right) - 0 = F_x$

$$F_x = (M + m)\ddot{x} + ml(\ddot{\theta}cos\theta - \dot{\theta}^2sin\theta + 2m\dot{l}\dot{\theta}cos\theta + m\ddot{l}\ sin\theta \qquad (2.10)$$

Secondly, the equation of motion associated with the generalized coordinate $q = \theta$ is as follows:

$$\frac{d}{dt}\left(\frac{\partial L}{\partial \dot{\theta}}\right) - \frac{\partial L}{\partial \theta} = 0$$

$$\frac{\partial L}{\partial \theta} = \frac{1}{2}m\left[2\dot{x}\dot{l}cos\theta + 2\dot{x}l\dot{\theta}(-sin\theta)\right] + mgl(-sin\theta)$$

$$= \frac{1}{2}m(2\dot{x}\dot{l}cos\theta - 2\dot{x}l\dot{\theta}(sin\theta)) - mglsin\theta$$

$$\frac{\partial L}{\partial \dot{\theta}} = \frac{1}{2}m(2l^2\dot{\theta} + 2\dot{x}lcos\theta)$$

$$\frac{d}{dt}\left(\frac{\partial L}{\partial \dot{\theta}}\right) = \frac{1}{2}m[2l^2\ddot{\theta} + 2\dot{l}\dot{\theta}(2l) + 2\ddot{x}lcos\theta + 2\dot{x}\dot{l}cos\theta + 2\dot{x}l\dot{\theta}(-sin\theta)]$$

$$= \frac{1}{2} m(2l^2\ddot{\theta} + 4l\dot{l}\dot{\theta} + 2\ddot{x}lcos\theta + 2\dot{x}\dot{l}cos\theta - 2\dot{x}l\dot{\theta}\ sin\theta)$$

Thus,

$$\frac{d}{dt}\left(\frac{\partial L}{\partial \dot{\theta}}\right) - \frac{\partial L}{\partial \theta} = 0$$

$$\tfrac{1}{2} m(2l^2\ddot{\theta} + 4l\dot{l}\dot{\theta} + 2\ddot{x}lcos\theta + 2\dot{x}\dot{l}cos\theta - 2\dot{x}l\dot{\theta}\ sin\theta) - \tfrac{1}{2} m(2\dot{x}\dot{l}cos\theta - 2\dot{x}l\dot{\theta}sin\theta) +$$

$$mglsin\theta = 0$$

$$2l^2\ddot{\theta} + 4l\dot{l}\dot{\theta} + 2\ddot{x}lcos\theta + 2\dot{x}\dot{l}cos\theta - 2\dot{x}l\dot{\theta}\ sin\theta - 2\dot{x}\dot{l}cos\theta + 2\dot{x}l\dot{\theta}sin\theta + \ 2glsin\theta = 0$$

$$2l\ddot{\theta} + 4\dot{l}\dot{\theta} + 2\ddot{x}cos\theta + \ 2gsin\theta = 0$$

$$l\ddot{\theta} + 2\dot{l}\dot{\theta} + \ddot{x}cos\theta + gsin\theta = 0 \qquad (2.11)$$

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

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

$$\theta: l\ddot{\theta} + 2\dot{l}\dot{\theta} + \ddot{x}cos\theta + gsin\theta = 0$$

2.2.3 Linearization of the system model

The above derived 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:

$$\theta \approx 0$$

$$\dot{\theta} \approx 0$$

$$sin\theta \approx \ \theta$$

And $\cos\theta \approx 1$

secondly, since the tension force that may cause the hoisting rope elongate was neglected, therefore 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:

$$x: \quad F_x = (M+m)\ddot{x} + ml\ddot{\theta} \quad (2.12)$$

$$\theta: l\ddot{\theta} + \ddot{x} + g\theta = 0 \quad (2.13)$$

2.2.4 State space representation of the system

After getting the linearized equation, equations (3.11) and (3.12) can be written in state space representation as:

$\dot{x} = Ax + Bu$

Where,

$$x = \begin{bmatrix} x \\ \dot{x} \\ \theta \\ \dot{\theta} \end{bmatrix}$$

$$\dot{x} = \begin{bmatrix} \dot{x} \\ \ddot{x} \\ \dot{\theta} \\ \ddot{\theta} \end{bmatrix}$$

From equation (2.12),

$$\ddot{x} = -l\ddot{\theta} - g\theta \quad (2.14)$$

Substitute equation (2.13) into (2.11) to get the following equation,

$F_x = (M+m)(-l\ddot{\theta} - g\theta) + ml\ddot{\theta}$

$$\ddot{\theta} = -\left[\left(\frac{M+m}{Ml}\right)g\theta + \frac{F_x}{Ml}\right] \tag{2.15}$$

Substitute equation (3.5) into equation (3.12),

$$F_x = (M+m)\ddot{x} - ml\left[\left(\frac{M+m}{Ml}\right)g\theta + \frac{F_x}{Ml}\right]$$

$$\ddot{x} = \frac{F_x}{M} + \frac{m}{M}g\theta \tag{2.16}$$

Equations (2.15) and (2.16) can be arranged into the matrix form as below:

$\dot{x} = Ax + Bu$

$$\begin{bmatrix} \dot{x} \\ \ddot{x} \\ \dot{\theta} \\ \ddot{\theta} \end{bmatrix} = \begin{bmatrix} 0 & 1 & 0 & 0 \\ 0 & 0 & \dfrac{mg}{M} & 0 \\ 0 & 0 & 0 & 1 \\ 0 & 0 & -\dfrac{(M+m)g}{Ml} & 0 \end{bmatrix} \begin{bmatrix} x \\ \dot{x} \\ \theta \\ \dot{\theta} \end{bmatrix} + \begin{bmatrix} 0 \\ \dfrac{1}{M} \\ 0 \\ -\dfrac{1}{Ml} \end{bmatrix} F_x \tag{2.17}$$

The output equation is,

$y = Cx + Du$

$$\begin{bmatrix} x \\ \theta \end{bmatrix} = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 \end{bmatrix} \begin{bmatrix} x \\ \dot{x} \\ \theta \\ \dot{\theta} \end{bmatrix} \tag{2.18}$$

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.

Having obtained the model of the system dynamics, simulation and analysis of the gantry crane system can be performed.

2.2.5 Concept of Input Shaping

Originally named ‘posicast control’, the initial development of Input shaping (I.S) is largely credited to O.J.M. Smith during the late 1950’s. Input shaping is a control strategy that uses a series of impulses to modify the reference command to suppress unwanted vibration. However, there seems to have been one notable precursor to Posicast Control’. In the early 1950’s, John Calvert developed a time-delay based vibrational filter named ‘Signal Component Control’. However, his solution did not contain the convenient closed loop form description offered by Smith (Huey, 2006, Park et al., 2001and Singhose, 2009). Smith’s method took a base line command and delayed part of command before given it to the system. The delay portion of the command canceled out the vibration induced by the portion of the baseline command that was not delayed.

2.2.6 Principle of Input Shaping

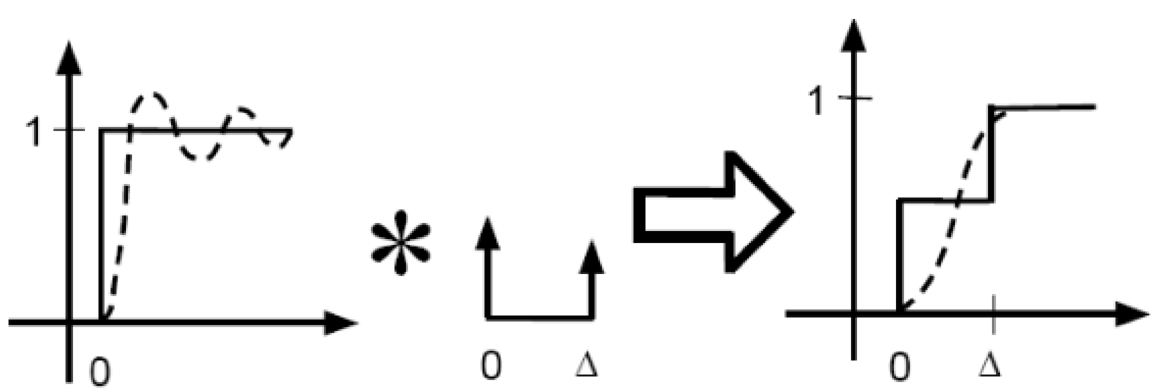
I.S can be said to be a command generation technique which attempt to impart zero energy into a system at the frequency at which it will vibrate. In order to ensure zero energy at the vibration frequencies, the commands given to the system must be modified, hence, the term input shaping (Arolovich and Agranovich, 2014). Input shaping is a feedforward control technique for improving the settling time and the positioning accuracy while minimizing residual vibration of computer controlling machines. The design objective of input shaping is to determine the amplitude and time locations of the impulsesin order to reduce the detrimental effects of system flexibility. These parameters are obtained from theNatural frequencies,$\omega_n$and Damping ratios, ζ of the system. It is a strategy for generation of time-optimal shaped commands using only a simple model; consist of the estimates of Natural frequencies and damping ratios. Hence, it is a simple method for reducing the residual vibration when positioning light damped systems. It offers several clear advantages over conventional approaches for trajectory generations:

1. Input Shaping does not require a dedicated transducer to measure the vibrations. It can be applied successfully to systems where vibrations cannot be observed at the feedback transducers.
2. Stability of the closed loop system is improved by input Shaping technique.

   Input Shaping simply modifies the command signal given to the system irrespective of length.
3. Input Shaping is designed to accommodate changes in the vibration frequency. The vibration frequency can change by 15% or more, but the vibrations will still be reduced by at least 95%. Furthermore, Input Shaping can compensate for changes in the vibration frequency without affecting the stability of the closed loop system.

4. Designing an input shaping does not require analytical model of the system, it can be generated from simple empirical measurement of the actual physical system.

Command shaping does not require feedback mechanism of closed-loop controllers. Vibration suppression is accomplished with a reference signal that anticipates the error before it occurs, rather than with a correcting signal that attempts to restore the system back to the desire state. i.e no need for any sensing of the payload sway. So control system technique (feed forward controller) is an easy solution in vibration reduction to feedback controller. In input shaping, an input shaper which is a sequence of impulses will be convoluted with an arbitrary reference signal to produce a shaped command that is used to drive a system. This process is described in Figure 2.1 with a shaper consist of two impulses. The input shaper is convolved with the unshaped input to produce the shaped input where the shaped input has the same oscillation reducing characteristic as the original set of impulses.



Initial Command Input Shaper Shaped Command

Figure 2.3: Generic Input Shaping Process (King, 2006)

The input shaping process can be represented in many ways in addition to the convolution process shown in Figure 2.3. It can be accomplished by time-delay blocks as in Simulink as shown in Figure 2.4. The unshaped input is fed into gain blocks that correspond to impulse amplitudes, A_i. The scaled function are then sent to time delay blocks, Δ_i, which corresponds to impulse time locations (Park et al, 2013, Singhose, 2009). Note that the first shaper impulse, A_1, is located at time zero, so it does not have an associated time delay block. At the output, the shaped command signal (filtered trajectory) is formed by summing the scaled and time-delay functions.

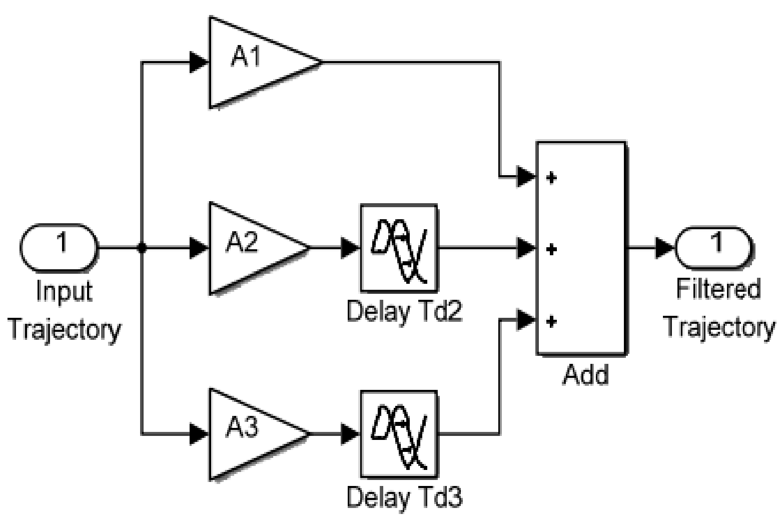


Figure 2.4: Simulink Block Diagram for Input Shaper

The amplitude and time locations of the impulses in an input shaper can be determined by solving a set of constraint equations (Singhose, 2009 and Sien, 2006), the constraint equations are usually categorized as follows:

(a) Zero Residual Vibration Constraints

(b) Impulse Amplitude Constraints

(c) Time Optimality Constraints

(d) Robustness Constraints

2.2.6.1 Zero Residual Vibration Constraints

This constraint is derived based on the concept that the impulse sequence must achieve zero vibration after the last impulse. Then, assuming the system can be modeled as a second -order harmonic oscillator or can be decomposed into a set of second-order system, then the vibration ratio can be determined from the expression for residual vibration amplitude from a single impulse as follows (King, 2006):

$$y(t)= \frac{A\omega_n}{\sqrt{1-\zeta^2}} \exp(-\zeta\omega_n(t-t_0))\sin(\omega_n\sqrt{1-\zeta^2}\ (t-t_0) \qquad (2.19)$$

where A is the amplitude of the impulse, t_0 is the time the amplitude is applied, ω_n is the natural, frequency, V is residual single-mode vibration, t_N is the time last response and ζ is the damping ratio.

The residual single-mode vibration amplitude of the response is obtained at the time of last impulse, I_N as(King,2006)

$$V=\sqrt{V_1^2+V_2^2} \tag{2.20}$$

where,

$$V_1=\sum_{i=1}^{N}\frac{A_i\omega_n}{\sqrt{1-\zeta^2}}\ \exp\left(-\zeta\omega_n(t_N-t_i)\right)\cos(\omega_d t_i) \tag{2.21}$$

$$V_2=\sum_{i=1}^{N}\frac{A_i\omega_n}{\sqrt{1-\zeta^2}}\ \exp\left(-\zeta\omega_n(t_N-t_i)\right)\sin(\omega_d t_i) \tag{2.22}$$

A_i and $t_i$are the amplitudes and time locations of the impulses and V_1 and V_2 are vertical and horizontal residual single-mode vibrations and

$$\omega_d=\omega_n\sqrt{1-\zeta^2} \tag{2.23}$$

To achieve zero vibration (ZV) after the last impulse, it is required that both V_1 *and* $V_2$are independently zero. This is known as zero residual vibration constraints (King, 2006).

2.2.6.2 Impulse Amplitude Constraints

As with any filtering method, a constraint must be applied to ensure that the shaped command produces the same rigid-body motion as the unshaped command. To satisfy this requirement, the impulse amplitudes must sum to one (King, 2006):

$$\sum_{i=1}^{n} A=1 \tag{2.24}$$

where n is the number of impulses.

2.2.6.3 Time Optimality Constraints

In order to avoid response delay, time optimality constraint is utilized. The first impulse is selected at $t_i = 0$ and the last impulse must be at minimum possible time, i.e,

$$\min(t_n)$$

2.2.6.4 Robustness Constraints

When the above constraints are solved for a two-impulse sequence, the ZV shaper is obtained as (King, 2006):

$$\begin{bmatrix} A_i \\ t_i \end{bmatrix} = \begin{bmatrix} \frac{1}{1+K} & \frac{K}{1+K} \\ 0 & \frac{\pi}{\omega_d} \end{bmatrix} \tag{2.25}$$

where,

The overshoot, $K = e^{\frac{-\pi\zeta}{\sqrt{(1-\zeta^2)}}}$ (2.26)

The residual single mode vibration amplitude of the response is obtained at the time of last impulse

Conventional C.S technique involves signal filtering such as Hamming and Butterwrth filters. However, the conventional technique has been marginally effective at reducing the residual oscillations in mechanical systems and it will increase the system settling time as well. Input shaping command is another form of command shaping that effectively improves the transient and steady state response of a mechanical system. Reference commands are modified in such a way that resonant mode of a system combine destructively, resulting in low residual oscillation.

There are several kinds of input shaping method, such as the Zero Vibration (ZV), ZV and Derivative (ZVD), and ZV and double Derivative (ZVDD) methods.

A Zero Vibration, ZV, input shaper is the simplest input shaper. The only constraints are minimal time and zero vibration at the modeling frequency. If these constraints are satisfied, the ZV shaper has the form of impulse amplitudes $A_i$and times t_i. The ZV shaper is useful in situations where the parameters of the system are known with a high level of accuracy. Also, if little faith is held in the input shaping approach, the application will never increase vibration beyond the level before shaping.

ZV shaper consists of two impulses. The shaper does not take robustness constraint into account. It only considers zero residual vibration, unity amplitude summation and time optimality constraints. By solving the constraints equations, the amplitudes and time locations of the ZV shaper are obtained as follows. Equation (2.7) shows the ZV shaper signal comprises of ZV shaper' s amplitudes$\left(\frac{1}{1+K}, \frac{K}{1+K}\right)$ and time location $(0, \frac{\pi}{\omega_d})$ based on the gantry crane' s natural frequency and damping ratio

A Zero Vibration and Derivative, ZVD, shaper is a command generation scheme designed to make the input-shaping process more robust to modeling error. If another constraint is added to the formulation of the shaper by setting the derivative of the vibration with respect to frequency equal to zero. The application of ZVD shapers is for systems where rise time is still important, but either the system will change with time or the model is not accurate. The matrix form of ZVD shaper is:

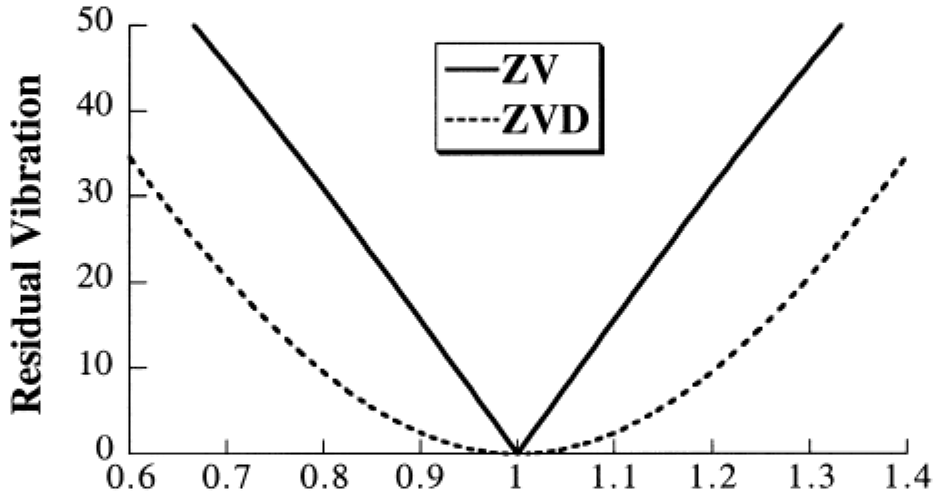
$$ZVD = \begin{bmatrix} \frac{1}{1+2K+K^2} & \frac{2K}{1+2K+K^2} & \frac{K^2}{1+2K+K^2} \\ 0 & \frac{\pi}{\omega_d} & \frac{2\pi}{\omega_d} \end{bmatrix} \tag{2.27}$$

If the model's inaccuracy cannot be controlled with a ZVD shaper then other shaping techniques are available, such as those described in the subsequent sections. It is possible to generate a more robust shaper by forming the second derivative of the residual vibration equation and setting it equal to zero. The shaper that results from satisfying this additional constraint is called a ZVDD shaper. This additional constraint increases the robustness, but also increases the shaper duration by one half period of the vibration. ZVDD shaper consists of four evenly spaced impulses lasting 1.5 periods of vibration. ZVDD is three ZV shapers convolved together. The advantage to doing this is that the input shaper parameters have less freedom, thereby simplifying the solution routine. However, by restricting the choice of input shaper parameters, the solution space is also restricted, meaning that there is the potential for optimal solutions to be missed. The matrix form of the three main Zero Vibration Shapers (Singhose 2009) is:

$$ZVDD=\begin{bmatrix} \frac{1}{1+3K+3K^2+K^3} & \frac{3K}{1+3K+3K^2+K^3} & \frac{3K^2}{1+3K+3K^2+K^3} & \frac{K^2}{1+3K+3K^2+K^3} \\ 0 & \frac{\pi}{\omega_d} & \frac{2\pi}{\omega_d} & \frac{3\pi}{\omega_d} \end{bmatrix} \tag{2.28}$$

In practice, ZVD input shaping is popular because it is robust to errors in the values of the natural frequency and damping.

In practice, ZV shaper can be sensitive to modeling errors. To demostrate this effect, the amplitude of residual vibration can be plotted as a function of modeling errors. Figure 2.5 shows such a senstivity curve for the ZV shaper. The vertical axis is the percentage vibration, while the horizontal axis is the normalised frequency formed by dividing the actual frquency of the system, $\omega_n$by modeling frequency, ω_m. Notice that the residual vibration increases rapidly as the actual frequency deviates from the modeling frequency. The robustness can be measured quantitatively by measuring the width of the curve at some low level of vibration . This non-demensional robustness measure is called the input shaper frequency insensitivty, I_n. The 5% insensitivity of the ZV shaper is 0.06, as shown in Figure 2.5 (Singhose, 2009). This means that the shaper can keep the residual vibration below the 5% level for frequency changes of only ±3 percent.



Normalised frequency ($\omega_{actual/\omega_{model}}$)

Figure 2.5: Input Shaper Sensitivity Curves (Singhose, 2009)

At the 1988 IEEE Conference on Robotics and Automation, Singer and Seering presented a paper on acausal command-shaping methods for controlling robot vibration (Singhorse 2009). However, they gave only a brief overview of the paper and then used their remaining time to present a robust command-shaping method that they had developed. This method was a significant leap forward, as it greatly expanded the possible applications for command shaping. In a very short time, several other research groups adopted the idea and were making extensions and experimental verifications of Singer and Seering's input shaping method. Equation 2.28 is the matrix form of Extra-Insensitive shaper

$$EI = \begin{bmatrix} A_1 & 1-(A_1 + A_3) & A_3 \\ 0 & t_2 & T_D \end{bmatrix} \tag{2.29}$$

Where, EI is Extra-Sensitive

A_1 is the initial amplitude

A_3 is the last amplitude

T_D is the final time location

t_2 is the second time location

In order to increase the robustness of the input-shaping process, Singer and Seering used additional contraints to design their input shaper. Their constraints forced their derivative of the residual vibration, with respect to frequency, to equal zero (King 2006):

$$\frac{\partial}{\partial \omega} V(\omega, \zeta) = 0 \qquad (2.30)$$

When equqtions (2.10), (2.24), and min(t_n) are satisfied with $V(\omega, \zeta) = 0$, the result is zero vibration and Derivative (ZVD) shaper containing three impulses given by:

$$ZVD = \begin{bmatrix} \frac{1}{1+2K+K^2} & \frac{2K}{1+2K+K^2} & \frac{K^2}{1+2K+K^2} \\ 0 & \frac{\pi}{\omega_d} & \frac{2\pi}{\omega_d} \end{bmatrix} \qquad (2.31)$$

By comparing the 5% insensitivities shown in Figure 2.5, it is obvious that the ZVD shaper is significantly more robust than the ZV Shaper. Its 5% insenstivity is 0.286 – a 480% increase over the ZV shaper. Note that the cost of this robustness is a lenghtening of the input shaper. The ZV shaper is 0.5 vibration periods in duration, while the ZVD shaper is one full period. This means that when the command is shaped, its rise time will be increased by one vibration period. This increase in rise time is usually a small price to pay for the robust vibration reduction.

An extension was proposed to this idea where additional Higher- order Derivatives are formed and set equal to zero (Singhose, 2009). When this additional constraints are used, the resulting shapers get more and more robust by further flattening the sensitivity curve at the modeling frequency. The cost for each additional robustness contraints is an additional lenghtening of the input shaper by 0.5 vibrational periods, and the need for one additional impulse in the input shaper. In their book that first described ZV shaping (Posicat control), O.J.M Smith demostrated that the shaper was effectively placing zeros over the flexible poles of the system- thereby canceling their vibratory effects (Singhose, 2009). When derivative constraints are added to the problem formulation, the input shaper places additional zeros over the flexible poles of the plant. Soon after the development of the Zero-derivative shapers, many other resarchers sought to extend the robustness idea. The extensions can largely be categorised as:

1. Built-in robustness
2. Adaptive robustness

Built-in robustness seeks to make the input shaper inherently robust (Singhose, 2009), as for example the ZVD shaper. A fundamental tradeoff in this approach is that obtaining more robustness leads to an increase in rise time. A challenge is to obtain significant robustness with very little rise time penalty.

Adaptive input shaping seeks to use feedback measurements of the system states to continually change the input shaper to improve its effectiveness. For example, the ZV shaper given in equation (2.8) can be continually changed during operation by updating the frequency, ω, which is used to calculate the shaper impulses. A significant cost of adaptive input shaping is that sensors must be added to control the system. Also adaptive methods require a large amount of online calculations. Thus, high speed processors are required to ensure the feasibilty when input shapers filters are designed for quick dynamic plants. Furthermore, when filters are designed for multimode plants, the algorithms can be very complex with even larger amount of online calculations. Many plants are time invariant, or variation can be neglected in practice. For input shaping of these plants, an offline filter design method that only needs the signals of the system outputs is desirable. Han et al,(2015) proposed a novel input shaping filter offline design method. This method has the following advantages:

(a) Only the signals of the system output are required. As the model information is not needed, the problem of model uncetainty is completely avoided.

(b) Adequate damping and bandwidth of the whole system can be chosen to yield desired system dynamics.

(c) For multimode plants, high-order filters can be easily designed (Han et al, 2015). Although the designed filter has zeros that cancels some of the poles of the controlled system, this is a byproductof the proposed algorithm, rather than a designed target. Infact, the designed method does not require parameter identification. For multimode plants of which the orders are unknown a piori, a second-order filter can be tried. If the filtering result is not satisfactory, then the filter order can be progressively increased.

To explain the basic idea of input shaping based on system output, the plant is assumed to be a second-order system with a transfer function:

$$G(S) = \frac{K\omega_n^2}{S^2+2\zeta\omega_n S+\omega_n^2} \qquad (2.32)$$

Where the damping ratio ζ is small (much less than 1), leading to poles closed to the $j\omega$- axis and to highly oscillatory step responses. K and ω_n are the static gain and natural frequency, respectively. The input shaping filter to be designed for G(s) is denoted as F(s). For the design of an input shaping filter for G(s), the desire transfer function (i.e reference model) is defined as follows(Han et al, 2015):

$$\mathrm{M(s)} = \frac{K_m\omega_m^2}{S^2+\omega_m^2 S^2+2\zeta\omega_m S+\omega_m^2} \qquad (2.33)$$

When the input shaping filter is desgned as

$$F_0(s) = \frac{K_m \omega_m^2 S^2 + 2\zeta\omega_n S + \omega_n^2}{K\omega_n^2 S^2 + 2\zeta_m \omega_m S + \omega_m^2} \qquad (2.34)$$

Then the combined transfer function $F_0(s)G(s)$ is equal to M(s). thus, K_m, ζ_m *and* ω_m can be chosen to yield an adequate static gain, damping ratio and bandwidth respectively.

Once K_m, ζ_m *and* ω_m are chosen, the filter to be designed can be written as:

$$F(s) = \frac{a_2 S^2 + a_1 S + a_0}{S^2 + 2\zeta_m \omega_m S + \omega_m^2} \qquad (2.35)$$

Then, the goal is to obtain the values of $\{a_{2,} a_{1,} a_0\}$ to make F(s) = $F_0(s)$, or F(s)G(s) = M(s).

Because the zero of $F_0(s)$ can precisely cancel the poles of G(s), realising F(s) = $F_0(s)$ means that the poles of G(s) are precisely identified.

The unit step response functions of M(s) and F(s)G(s) are respectively denoted as $y_r(t)$ and y(t).

$y_r(t)$ is the reference step response output.

The transfer fuction of a linear-time invariant system is uniquely determined from the step response function of the system under the assumption that all initial conditions are zero (Han et al, 2015). Thus, to make the transfer function of F(s)G(s) precisely equal to that of M(s), the difference between y(t) and $y_r(t)$ needs to be minimised. The following cost function is defined to evaluate the difference between y(t) and $y_r(t)$:

$$E = \int_0^{\alpha} (\mathrm{y(t)} - \mathrm{yr(t)})^2 \, dt \qquad (2.36)$$

Thus, the design problem is formulated as obtaining the values of $\{a_{2,} a_{1,} a_0\}$ that minimize E. if the target system G(s) is a multimode system and the number of the under-damped modes of G(s) is *l*, then the transfer function of the reference model can be set as:

$$M(s) = K_m \prod_{j=1}^{l} \frac{\omega_{mj}^2}{S^2 + 2\zeta_{mj} S + \omega_{mj}^2} \qquad (2.37)$$

where, $K_{m,}$ $\omega_{mj,}\zeta_{mj}$ can be chosen to yield adequate properties.

Correspondingly, the input shaping filter is initially written as

$$F(s) = \frac{a_n S^n + a_{n-1} S^{n-1} + \cdots + a_0}{\prod_{j=1}^{l} (S^2 + 2\zeta_{mj}\, \omega_{mj} S + \omega_{mj}^2)} \tag{2.38}$$

where, n + 2l, and a_n, $a_{n-1} \ldots a_0$ are to be designed.

To decomposed the system output; the input shaping filter F(s) given in (2.20) is written as

$$F(s) = \frac{a_n S^n + a_{n-1} S^{n-1} + \cdots + a_0}{F_d(S)} \tag{2.39}$$

where, $F_d(s) = \prod_{j=1}^{l} (S^2 + 2\zeta_{mj}\ \omega_{mj} S + \omega_{mj}^2)$

If the times are define as H(s) = F(s)G(s), $f_i(s) = {S^1}/{F_d(s)}\, G(s), i = 0, 1, 2, 3 \ldots n$

Then,

$$F(s) = \sum_{i=0}^{n} a_i\, f_i(s) \tag{2.40}$$

$$H(s) = \sum_{i=0}^{n} a_i\, h_i(s) \tag{2.42}$$

As stated earlier, the unit step response functionof H(s) is denoted as y(t), Here the unit step response function of $h_i(s)$ is denoted as $y_i(t)$. before the design of F(s) is accomplished, y(t) is unknown, but $y_i(t)$ can be obtained once $F_d(s)$ is determined.

$y_i(t)$ is defined as acomponent of y(t), so

$$y(t) = \sum_{i=0}^{n} a_i\, y_i(t) \tag{2.43}$$

where, a_i is Filter Gains, n is the number of impulses

Figure 2.4 illustrate the relationship given in equation (2.24)

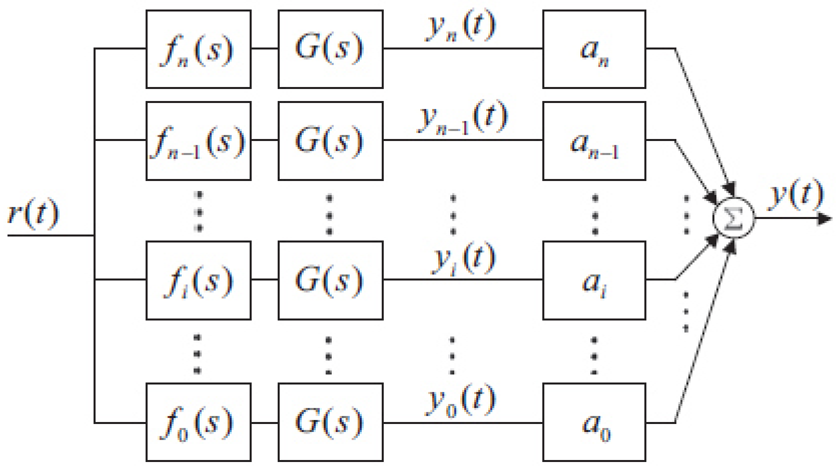


Figure 2.6:The Decomposition of a System with Input Shaping Filter (Han et al, 2015).

2.2.7 Single Mode Shaping

When a system has just one identical vibration as response to step command, it is reduced by Single Mode Shaping. An early form of input shaper was the use of Posicats control. This control splits the reference signal into two parts. The size of the steps and the delay before introducing the second step are derived from the system dynamics. The modern technology that we call impulse shaping is a few generations removed from police control idea. The original work is done in this area by Singer. The process relates back to the use of a series of impulses, which will cause zero vibration in the system; this series of impulses or shaper when convolved with the original command to the system yields a response that also causes zero vibration (Arolovich, 2014).

2.2.8 Multimode Shaping

In systems with more than one mode of vibration, two schemes can be used to generate commands for the system. The first is by simply convolving together multiple Input Shapers designed for each specific mode. The result of this process is a longer input shaper, which can deal with each mode specifically. Another way to design the shaper is to solve the constraint equations for the two modes simultaneously. This method results in vibration reduction near the modeling frequencies, but does not yield as much suppression of the high modes. However, a simultaneous shaper is never longer than a convolved shaper, and it is often significantly shorter. This advantage in speed can be important for slow oscillations (Igor and Arolovich, 2014).

2.2.8 Concept of Proportional-Integral-Derivative (PID) Controller

PID Controller dated back to 1890s governor design (Bansal et al 2012). Despite having been around for a long time, majority of industrial applications still use PID controllers. According to a survey in 1989, 90% of process industries use them (Pradeepkannan and Sathiyamoorthy 2014, Osama et al., 2013 and Bansal et al., 2012). This widespread use of PID in industry can be attributed to their simplicity and ease of re-tuning on-line.

PID control gives the simplest and yet efficient solution to various real-world control problems. Both the transient and steady state responses are taken care of with its three-term (i.e P, I andD) functionality. Since its invention, the popularity of PID control has grown tremendously, particularly at the lowest level, as no other controllers can match the simplicity, clear functionality, applicability and ease of use offered by PID controllers.

An error value which is the difference between the measured process variable and desire response is calculated. The controller attempts to minimize the error by manipulating the process

input. The PID controller calculation involves the determination of three constant parameters called the Proportional (P), Integral (I) and Derivative (D) values. These values can be interpreted in term of time. P depends on the present error, I on the accumulation of past error and D is prediction of future error, based on current rate of change. By combining the three different control actions, a PID controller output u(t) is obtained (Nimbalkar et al, 2015) as:

$$u(t) = K_p e(t) + K_i \int e(t)dt + K_d \frac{de(t)}{dt}$$

(2.44)

The effects of the three gains i.e K_p, K_i, and K_d are as on Table 2.1 (Bhagwan *et al.*, 2016)

Table 2.1: Effects of PID gains

Parameter	Rise time	Overshoot	Settling time	Steady state error

K_p	Decrease	Increase	Small Change	Decrease
K_i	Decrease	Increase	Increase	Eliminate
K_d	Small Change	Decrease	Increase	Small Change

And the transfer function of the controller is:

$$\frac{U(s)}{E(s)} = K_P\left(1 + \frac{1}{T_i(s)} + T_d(s)\right) = K_p\left(\frac{T_i T_d s^2 + T_i S + 1}{T_i s}\right) \quad (2.45)$$

the research in tuning PID parameters started in early 1940s. Several tuning methods have been proposed for the tuning of process control loops, with the most popular method being that of Ziegler and Nichols (1942) (Pradeepkannan and Sathiyamoorthy, 2014). They proposed two methods, one is open loop step response method and the second is closed loop frequency response method. The Ziegler and Nichols first PID tuning method is the technique made based on certain controllers assumptions. Hence, there is always a requirement of further tuning; because the controller settings derived are rather aggressive and thus results in excessive overshoot and Oscillatory response. Also the controller parameters are rather difficult to estimate in noisy environment. The second method is based on knowledge of the response to specific frequencies. The idea is that the controller setting can be based on the most critical frequency points for stability. This method is based on determining the point of marginal stability experimentally. This frequency can be found by increasing the proportional gain of the controller, until the process becomes marginally stable. The gain is called ultimate gain k_{pc} and the time period T_c. Other methods include: the method of Cohen Coon, ・str・m and H・gglund, De Paor and O' Malley, Zhuang and Atherton, Venkatashankar and Chidambaram, Poulin and Pomerleau and Haung and Chen. In spite of this large range of tuning technique, to date there still seems to be no general consensus as which tuning method works best for most applications (Osman et al., 2013). Some methods rely heavily on experience, while others rely more on mathematical considerations.

Table 2.2 shows advantages and Disadvantages of some common methods of tuning PID controller (Bhagwan*et al.,2016*)

Table 2.2: PID Tuning Methods

Tuning method	**Advantage**	**Disadvantage**
Manual Tuning	1. No math required 2. Online method	1.Requires experienced personnel
Ziegler-Nichols	1. Proven method 2. Online method	1.Process upset 2. Some trial and error 3.Very aggressive tuning
Software Tools	1. Consistent tuning 2. Either online or offline 3. May include actuator and sensors analysis	1.Involves some cost and training
Cohen-coon	1. Good process models	1.Requires some math 2.Offline method only 3.Only good for first order process

2.3 Review of Related Works

Many research works have been carried out on Gantry crane control. Open loop and closed loop controllers are the two major categories of the developed controllers. This section reviews some of the key research works in this direction.

Singhorse et al. (2000) investigated the dynamic behavior of a planar GC with hoisting of load. The command generation method of input shaping was proposed for reduction of the residual vibration. Several versions of input shaping were evaluated and compared with time-optimal rigid-body commands over a wide range of parameters. Input shaping provided significant reduction in both the residual and transient oscillations, even when the hoisting distance was a large percentage of the cable length. Experimental results from a 15-ton gantry crane at the Savannah River Technology Centre were used to support the numerical result.

Wahyudi and Jalani (2005) proposed fuzzy logic controllers for GC system. Fuzzy logic control was designed based on information of skillful operators. The performance of the proposed intelligent gantry crane system was evaluated experimentally using a large scale GC. On a range of performance indices, the fuzzy controlled GC system performed better compare to the classical crane system.

King (2006) worked on Command shaping control of a crane system.The work was able to show that the vibration of the gantry crane system can be reduced effectively by input shaper technique. The original system with unshaped bang-bang input force will oscillate at its own natural frequency when the input force is taken off. But this oscillation was effectively reduce or even removed from the system when the bang-bang input force is shaped by the input shaper technique before applied or input to the gantry crane system. From the result: the payloads of the system with ZV shaper still oscillate when the input force is taken off. System with ZVD and ZVDD shaper show that the payload oscillation is zero radian after the input force is taken off which payload will be static at the final destination. This means the input shaper with more number of impulses has better performance in reducing the oscillation of the payload.

Steven et al. (2007) investigated the control of GC by using model predictive control (MPC). One of the main motivations to apply MPC to control GC was based on its ability to handle control constraints for multivariable systems. A pre- compensator was constructed to compensate the input nonlinearity (non-symmetric dead zone with saturation) by using its inverse function. The algorithm was implemented for the control of GC system in system control lab of university of technology, Sydney (UTS) and achieved desired experimental result.

Wahyudi et al. (2007) introduced and evaluated experimentally a practical and intelligent method for GC. The proposed method consisted of Nominal Characteristic Trajectory Following (NCTF) and fuzzy logic controller for position and anti-swing control respectively. The method is practically applicable without the need to have the exact model of the plant during controller design process. The design of NCTF controller was based on a simple open loop experiment while the fuzzy logic controller was based on human experience. The performance of the proposed system was evaluated experimentally on a large scale GC system. It was also compared with model-based PID and non-model based fuzzy logic control methods. The results showed that the proposed method was not only effective for controlling the crane but also robust to parameter variation.

Soleiman et al. (2009) constructed a hardware platform for controlling a model of GC. The status of GC was to be monitored on a personal computer (PC), while control was implemented on microcontroller level. A theoretical PID-controller was developed and documented, but it was not implemented on microcontroller level. The sway of the crane was reduced by a digital PID controller, in order for the crane to move from one side to the other with minimal sway. A theoretical PID-controller was developed and documented, but due to time issues it is not implemented on microcontroller level.

Singhose (2009) reviewed first 50years of Command shaping of flexible systems. The paperprovides a review of command-shaping researches that have provenuseful since it was first proposed in the late 1950's. The importantmilestones of the research advancements, as well as applicationexamples, are used to illustrate the developments in this importantresearch field. He stated major advantages of command shaping which include: no requirement forsensor measurements; although sensors can be used in adaptivecommand-shaping methods to improve performance, it suppresses vibration ina preemptive way that is faster than anything possible withfeedback control. Feedback control must wait for an error to ariseand be sensed before it starts to suppress it, usesa dynamic model to anticipate the occurrence of vibration, so it caneffectively start to act as soon as the system starts to move, also, robustness even when largemodeling errors exist, require simple dynamic model – just estimates of the natural frequencies anddamping ratios, Command shaping has high versatility that manytypes of auxiliary constraints, such as actuator limits, fuel usage,and transient deflection limits, can be integrated into the design ofthe commands.

Ahmed (2009) Developed Hybrid Fuzzy Logic Control with Input Shaping for Input Tracking and Sway Suppression of a Gantry Crane System. The investigation revealed the effectiveness of the controllers. A Proportional-Derivative (PD)-type fuzzy logic control was developed 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 of the response of the gantry crane with the controllers were presented in time and frequency domains. The performances of the hybrid control schemes were examined in terms of input tracking capability, level of sway reduction and robustness to parameters uncertainty.A significant reduction in the system sways was achieved with the hybrid controllers regardless of the polarities of the shapers.

Bruins (2010) investigated and compared the performances of four control algorithms. These algorithms were compared on the basis of simplicity, stability and robustness. Parallel P-controller, cascade P-controller, fuzzy controller and an internal model controller were used. In order to validate the designed controllers, a model was derived. The controllers and the model were implemented in Matlab/Simulink. Finally, the controllers were validated and tuned in Labview on a laboratory scale of GC model. Although, it was concluded that all the presented controllers are suitable for control of GC system, the fuzzy controller showed the best performance.

Abe (2011) proposed a new anti-sway open loop control method to allow an overhead crane to reach its target position without oscillation of attached load. Radial Basis Function Networks (RBFNs) were adopted to generate a smooth trajectory of the trolley position. In order to obtain a trajectory of the trolley that suppressed residual sway motion, RBFNs were trained using particle swarm optimization (PSO). Simulation and experiment were performed to demonstrate the validity and effectiveness of this method. The simulation and experimental results confirm that the residual sway was perfectly suppressed using the proposed approach.

Karajgikar et al. (2011) compared Proportional-Derivative (PD) feedback control and an input shaping control for double-pendulum crane. The study comprised ten novice crane operators using representative two- mode input shaping and feedback control methods. It was found that, both feedback control and input shaping reduced average task completion time from the manually controlled-case; however, input shaping provided the lowest average completion time. Input shaping also allowed the operators to move the trolley over a shorter total path length to complete the tasks, suggesting that input shaping also resulted in zero obstacle collisions during the tests thereby improving safety.

Ryan and Shan (2011) conducted Experimental Study on Combining a Simple Input Shaper and Adaptive Positive Position Feedback Control. The paper presents a vibration control strategy for a flexible manipulator with a collocated piezoelectric sensor/actuator pair. The experimental system is first outlined and then a control law developed. The proposed vibration controller combined the input shaping technique with multi-mode adaptive positive position feedback. An adaptive parameter estimator based on the recursive least square method was developed to update the system's natural frequencies which are used by the adaptive positive position feedback. A proportional-derivative controller was combined with the proposed vibration controller to suppress vibration while slewing the manipulator. Experimental results shows that combination of input shaping with the multi-mode adaptive positive position feedback control clearly offers advantages in the vibration suppression over either of these control methods alone, particularly for systems where there are frequency uncertainties.

Similarly, Syvertsen (2011) modeled and simulated an offshore crane model with input shapers. Three input shapers and two different workspace controllers were developed in order to reduce residual vibration. The three input shapers successfully reduced the payload swing. Without the shapers the vibration was approximately 1m in amplitude. When the operator input was shaped by the zero-vibration derivative (ZVD) shaper, it was reduced to about 5cm. the down side of it was that the shaped command was delayed compared to the unshaped one.

Thalapil (2012) developed a technique for improving the control of two mode double pendulum crane called input shaping. In this work, a robust input shaper called a specified Insensitivity Input Shaper was optimized using the knowledge of amplitude contributions of each mode to the overall response. The results showed that input shaping was an effective feedforward control technique that can be implemented on GC to reduce the payload sway.

Alhazza and Masoud (2013) proposed a Novel waveform (NW) 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.

Kolar and Schlacher (2013) presented flatness control of a laboratory model GC. The design of the tracking control was accomplished in two steps. First, the system was exactly linearized by quasi-static state feedback.

Subsequently, for the linear system a feedback with integral parts was designed such that the motion of the load was stabilized about the reference trajectories. Moreover, the control law was extended by terms which approximately compensated for the friction occurring at the GC. The presented control algorithm yielded excellent result.

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 PZV andPZVDD were designed based on the properties of the system. Implementation results of the response of the rotary crane with the controllers were presented in time and frequency domains. 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.

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 the 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.

Summary of the type of control and the nature of controllers used by different researchers reviewed are given in table and table

Table 2.3 only Sway Suppression

AUTHOR/PLANT	SWAY	NATURE OF CONTROLLER
1. Singhose(2000) G.C	ZV, NEG.ZV, ZVD & SD	Feedforward (F.F)
2. King (2006) G.C	ZV, ZVD & ZVDD	F.F
3. Ahmed (2009) G.C	PD-type Fuzzy logic & ZSDD PD-type Fuzzy logic & modified SNA/ZSDD	F.F & F.B
4. Soleiman (2009) G.C	Microcontroller MSP430F. sway reduced by digital PID	F.B
5. Singhose (2009) C.S	ZV, ZVD, EI and SI	F.F
6. Abe (2011) O.C	Radial basis function network (RBFNs)	F.B
7. Karajgikar (2011) D.P	Compared I.S (two-mode ZV shaper) and PD	F.F & F.B

8. Syvertsen (2011) Offshore crane	ZV & PPF	F.F
9. Thalapil (2012) G.C	Specified Insensitve shaper (I.S)	F.F
10. Ryan (2011) Crane system	ZV & PPF	F.F
11. Alhazza (2013) O.C	Wave form command shaper (W.F)	F.F

Table 2.4 Sway and Position Control

AUTHOR/PLANT	SWAY	POSITION	NATURE OF CONTROLLER
Wahyudi (2005) GC	Fuzzy logic	Fuzzy logic	F.B & F.B
Steven (2007) GC	MPC algorithm	Different MPC algorithm	F.B & F.B
Wahyudi (2007) GC	Fuzzy logic	NCTF	F.B & F.B
Kolar (2013) GC	Quasi static state feedback	Feedback with integral part	F.B & F.B
Ezuan (2013) 3DOF R..Crane	PZV & PZVDD	LQR for rotation angle	F.F & F.B
Mohamed (2015) 3D O.C	Fuzzy control	PID	F.B & F.B

From the tabular presentation of the reviewed of literatures, it is clear that certain group of authors concentrate only on the sway suppression in crane system leaving out the trolley position control in such cases manual control may be adopted at the hoisting point. Many of these categories of authors adopt feedforward controllers and few among them used combination of feedforward and feedback controllers. On the other hand, some authors controlled both payload sway and trolley position simultaneously and this ensure both safety to human and accurate positioning. In this category more of feedback controllers are adopted for both tasks than the combination of feedforward and feedback. In all the reviews especially those involving input shapers, time delay is always noticed when it involves derivatives, a cost pay for accuracy and Safety.

2.7 Conclusion

The Chapter reviewed the control schemes for GC system, explained the model description and derivation of dynamic equation of the Gantry crane, the concept of Input shaping, the Proportional Integral Derivative (PID) controller and Review of Related works. The next chapter will address ways to accomplish the research gap by the designing the controllers and implement them on the GC system.

CHAPTER THREE

MATERIALS AND METHODS

3.1 Introduction

This chapter presents the procedure for the design of the input shapers (ZV and ZVD), the design and tuning of the PID controller, the development of the hybrid scheme comprises of ZV-ZVD and PID and finally the simulation of the designed circuits.

3.2 Materials

The material used for the actualization of this research includes:

- Computer system
- Matlab/ Simulink software.

3.2.1 Personal Computer

- A HP computer that has the following specification was used: Windows 8.1 Pro, Processor: Intel(R) Pentium(R) @ 1.8GHz, RAM: 4.00GB, System type: 64-bit Operating System and Hard Disk: 500GB.

3.2.2 MATLAB/ Simulink

MATLAB is a programming language developed by MathWorks, which stands for MATrixLABoratory. It is mathematical software package that is used extensively in both academia and industry. It is an interactive program for numerical computation and data visualization that along with its programming capabilities provides a very useful tool for almost all areas of science and engineering. Unlike other mathematical packages, such as MAPLE or MATHEMATICA, MATLAB cannot perform symbolic manipulations without the use of additional Toolboxes. It remains however, one of the leading software packages for numerical computation. As you might guess from its name, MATLAB deals mainly with matrices. A scalar is a 1-by-1 matrix and a row vector of length say 5, is a 1-by-5 matrix. One of the many advantages of MATLAB is the natural notation used. It looks a lot like the notation that you encounter in a linear algebra course. This makes the use of the program especially easy and it is what makes MATLAB a natural choice for numerical computations. Some of the futures and uses of MATLAB are presented as follows:

3.2.2.1 Features of MATLAB

Following are the basic features of MATLAB:

a) It is a high-level language for numerical computation, visualization and application development.
b) It also provides an interactive environment for iterative exploration, design and problem solving.
c) It provides vast library of mathematical functions for linear algebra, statistics, Fourier analysis, filtering, optimization, numerical integration and solving ordinary differential equations.
d) It provides built-in graphics for visualizing data and tools for creating custom plots.
e) MATLAB's programming interface gives development tools for improving code quality, maintainability, and maximizing performance.
f) It provides tools for building applications with custom graphical interfaces.
g) It provides functions for integrating MATLAB based algorithms with external applications and languages such as C, Java, .NET and Microsoft Excel.

3.2.2.2 Uses of MATLAB

MATLAB is widely used as a computational tool in science and engineering encompassing the fields of physics, chemistry, math and all engineering streams. It is used in a range of applications including:

1) Signal processing and Communications
2) Image and video Processing
3) Control systems
4) Test and measurement
5) Computational finance
6) Computational biology

3.2.2.3 Simulink

Simulink is a graphical extension to MATLAB for modeling and simulation of systems. In Simulink, systems are drawn on screen as block diagrams. Many elements of block diagrams are available, such as transfer functions, summing junctions, etc., as well as virtual input and output devices such as function generators and oscilloscopes. These virtual devices will allow you to perform simulations of the models you built. Simulink is integrated with MATLAB and data can be easily transferred between the programs. In this dissertation, we used Simulink to modeled systems, build controllers and simulate the model.

3.3 Methodology

The design of the ZV and ZVD input shapers after the determination of the natural frequency (ω_n) and damping ratio (ζ) of the Gantry crane system is established for sway suppression, the shaped command is linked with the PID;auto-tuned using Simulink optimization tool and finally, the simulation of the developed hybrid configuration.

3.3.1 Design of Simulink block of Nonlinear Crane System to Determine ω_n&ζ

The natural frequency, ω_n and the Damping ratio, ζ of the system are determined by using Curve fitting toolbox in theMATLAB software,with the input and output data obtained from the nonlinear crane system. Figure3.1 shows the Simulink Block diagram of the dynamic equation.

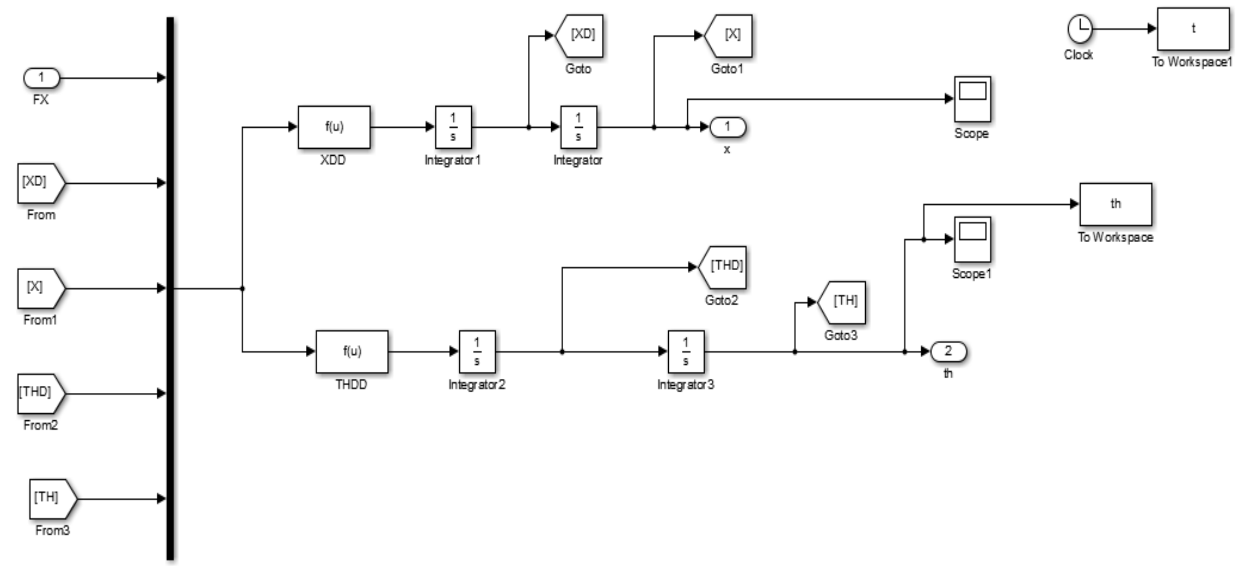


Figure 3.1: The Simulink Block diagram of nonlinear Crane system for determine ω_n and ζ

Figure 3.1 is the Simulink block diagram representation for obtaining the $\omega_n$and ζ in the curve fitting tool of matlab from the non-linear dynamic equation

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

$$\theta: l\ddot{\theta} + 2\dot{l}\dot{\theta} + \ddot{x}cos\theta + gsin\theta = 0$$

For ease of configuration in Simulink, the equation is simplified as:

XDD= (u(1)-m2*l*u(4)*cos(u(2))+m2*l*u(3)*sin(u(2))*u(3))/(m1+m2)

THDD= (-g*sin(u(2))-u(1)*cos(u(2)))/l

Where, $F_x = u(1), \ddot{x} = u(2), x = u(3), \dot{\theta} = u(4) and\ \theta = u(5)$ M=m_1 m=m_2

And the numerical values:

m_1= 3Kg, m_2= 0.75Kg, l= 0.75m and g 9.81ms^2

After the simulation and necessary adjustments, the desire values of the system's natural frequency and damping ratios are obtained as:

$\omega_n = 3.985\text{rad}s^{-1}$ and the $\zeta = 0.03726$

To calculate the damped natural frequency, $\omega_{d,}$ equation 3.5 is applied

$$\omega_{d} = \omega_n\sqrt{1-\zeta^2} \quad 2.23$$

$$\omega_{d} = 3.985\sqrt{1-0.03726^2}$$

$$= 3.985\sqrt{0.9986}$$

$$= 3.985X0.9993 \qquad = 3.9822\text{rad}s^{-1}$$

To also, estimate the stiffness constant, K, equation 2.26 is applied

$$K = e^{-\frac{\pi\zeta}{\sqrt{1-\zeta^2}}} \quad 2.26$$

$$K = e^{-\frac{\pi X0.03726}{0.9993}} \quad = 0.8895$$

3.3.2 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.

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

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

$$t_1 = 0, \qquad t_2 = \frac{\pi}{\omega_d} = \frac{\pi}{3.9822} = 0.7889$$

Figure 3.2 is the Simulink block diagram of the ZV shaper with the values obtained.

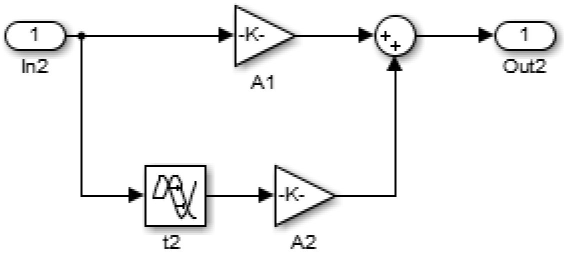


Figure 3.2: Simulink block of ZV Shaper

The A_i and t_i of the ZVD shaper:

$$A_1 = \frac{1}{1+2K+K^2} = \frac{1}{1+2(0.8895)+0.8895^2} = \frac{1}{3.5702} = 0.2801$$

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

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

$$t_1 = 0, \quad t_2 = \frac{\pi}{\omega_d} = \frac{\pi}{3.9822} = 0.7889 \text{ and } t_3 = \frac{2\pi}{\omega_d} = 1.5778$$

Figure 3.3 is the Simulink block diagram of the ZV shaper with the values obtained.

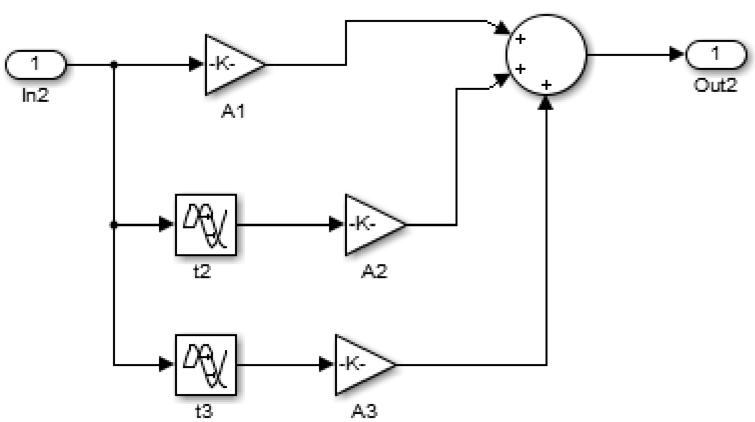


Figure 3.3: Simulink block of ZVD

3.3.3 Design and Tuning of PID Controller for Position Control

The goal here is to find the values of the gains K_p, $K_i$and K_d that give satisfactory result. Figure 3.4 is a representation of Simulink block diagram of hybrid Shaper/PID that was used to design and tune PID controller

The tuning method employed was Simulink Optimization tool in Matlab to obtain the best optimal values for K_p, K_i, and K_d

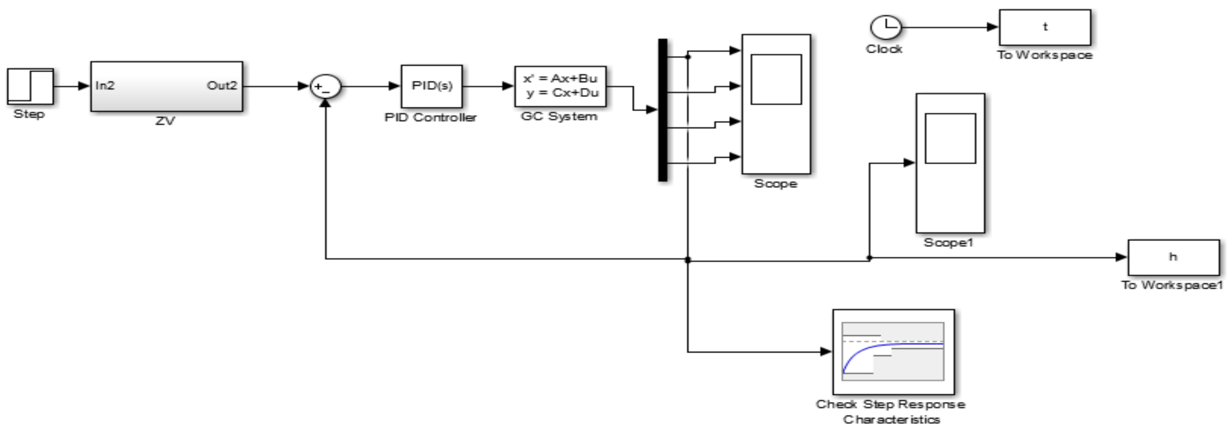


Figure3.4 Simulink Optimization Tool for PID Tuning

The convergence is at:

0.0010100 0.00001000   5.00000002

The result shows the optimum value of the PID parameters. Table 3.1 gives the tabular representation of he values.

Table 3.1: Values of PID gains

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

3.3.4 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 risetime 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 toggles to an input shaper (ZVD) having a shorter settling time and minimal steady state error. Figure 3.5 is the block diagram configuration of input shapers with switch, PID and the Gantry crane system.

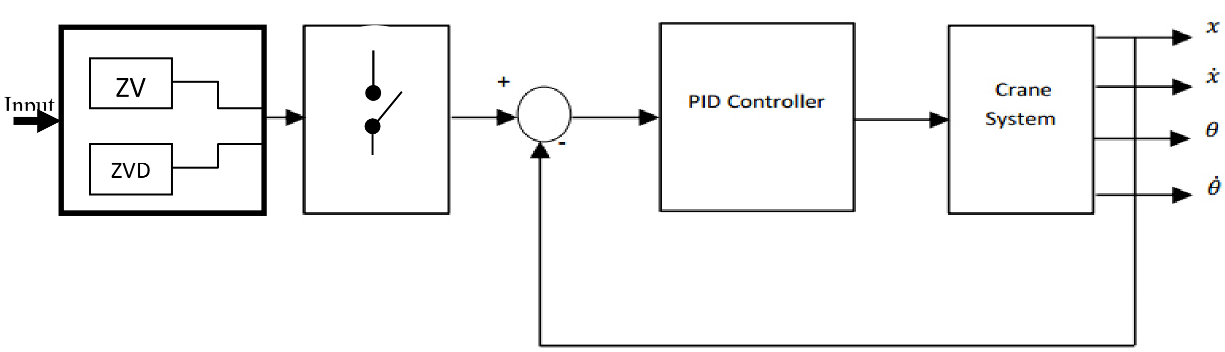


Figure 3.5: Block Diagram of the Hybrid controller combining two input shapers

3.3.5 Sway Suppression using Zero-Vibration Shapers

The figure 3.6 below is the Simulink block diagram for payload sway suppression made up of the combined input shaper (ZV and ZVD) connected by a switch designed to toggle between the ZV and the ZVD, when ZV is at the range of ≥0.75s, the appropriate portion for ZVD triggering, the switch selects ZVD for suppression. The block diagram incorporates the individual ZV, ZVD and uncontrolled system for response comparison.

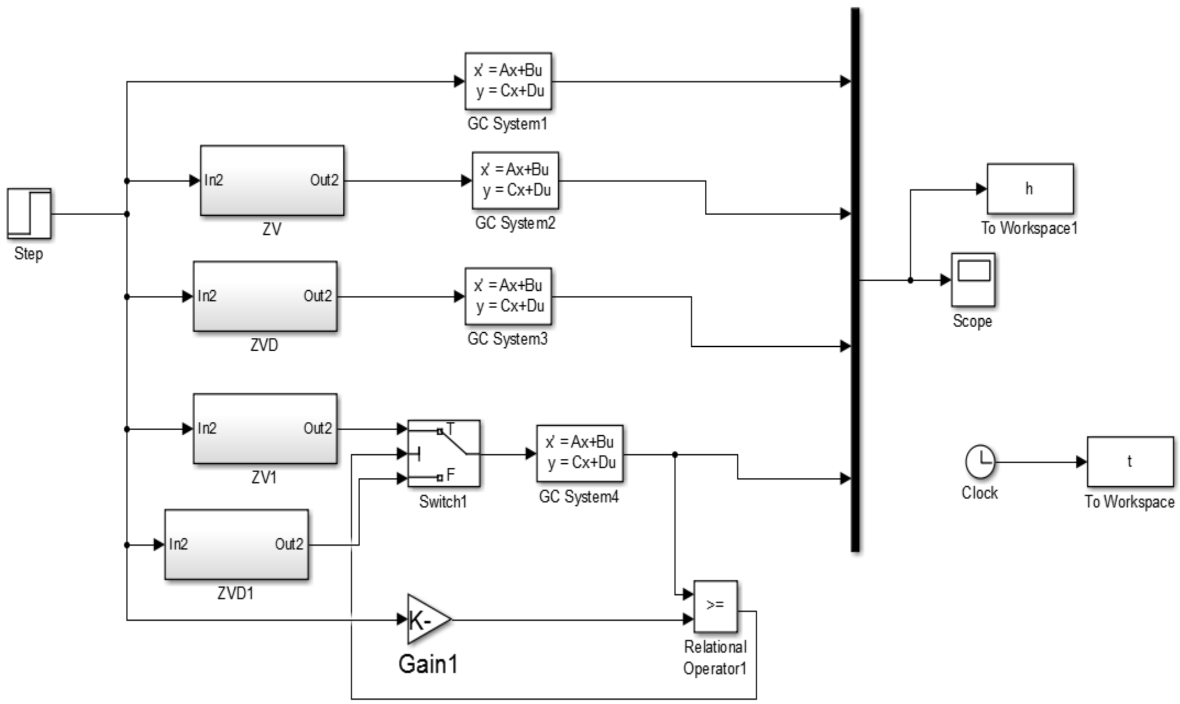


Figure3.6: Combine Simulink block diagram of ZV, ZVD, ZV-ZVD Controllers

3.3.6 Trolley Position Control using hybrid PID/Shapers Controller

Figure 3.7 below is the Simulink block diagram consisting of the PID alone, PIDZV, PIDZVD and 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 switch was designed to initially take the advantage of shorter risetime of ZV to track position until 90% and then change to ZVD with shorter settling time at minimum gain of ≥0.75s. The performance evaluation of the proposed controller is compared with that of the aforementioned controllers.

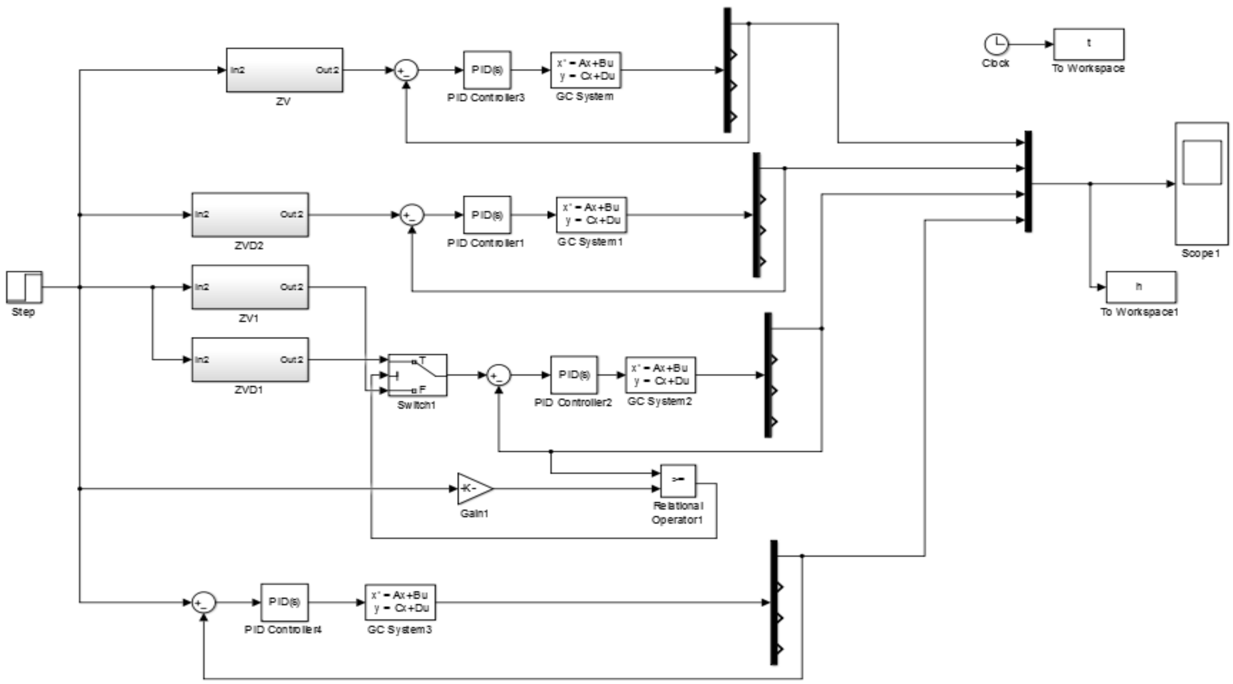


Figure 3.7 Simulink block diagram of Input Shapers with PIDs and PID alone

CHAPTER FOUR

RESULTS AND DISCUSSION

4.1 Introduction

This chapter presents and discusses the results of simulations of the system; with ZV filter alone, with ZVD alone, Hybrid ZV-ZVD and also the result of simulation of PID-ZV, PID-ZVD and hybrid PID/ZV-ZVD controllers carried out on MATLAB/Simulink (R2015a) software platform.

4.2 Result of Sway Suppression with the Input Shapers

The ZV is adopted initially due to its fast response and then ZVD for its better suppression characteristics and consequently, ZV-ZVD produced an improved response both in speed andpay load sway suppression. Figure 4.1 show the Sway angle Response Comparisons of Input Shapers and without input shaper

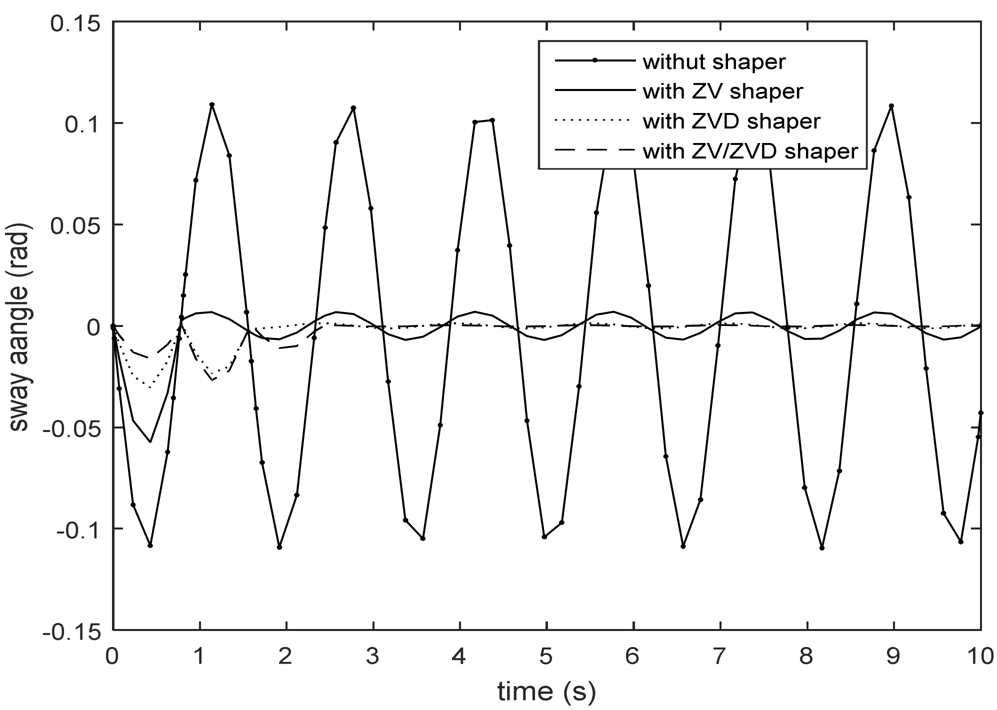


Figure 4.1 : Sway Angle Response Comparisons of Input Shapers and without Input Shaper

Figures 4.1 show the level of sway angle reduction achieved by the input shapers. From the graphs, it can be seen that ZV filter was able to suppress the sway significantly; ZVD shows better suppression while the proposedhybrid of the two gives the best sway suppression as shown in Table 4.1

Table 4.1: Zero Vibrations Shaper Performance Results

Input Shaper	**Peak-Peak sway angle (rad)**	**Reduction (rad.)**	**Percentage Reduction (%)**
Uncontrolled	0.2174	-	-
ZV	0.0134	0.204	93.84
ZVD	0.002615	0.2148	98.797
ZV-ZVD	0.0006669	0.2167	99.69

It can be observed from the combined plots of Figure 4.1 that the sway suppression is significantly achieved in ascending order of perfection ranging from 93.84% by ZV, 98.797% by ZVD and 99.69% achieved by hybrid of ZV-ZVD. Table 4.2 summerises the findings:

Table 4.2 Result Discussions of the Input Shapers

Input Shaper	System Response	Payload Oscillation	Max. Sway Amplitude	Sway suppression (%)
Uncontrolled	Fastest	Highest	Highest	
ZV	Slow	Low	Low	93.84
ZVD	Slower	Lower	Lower	98.80
ZV-ZVD	Slowest	Lowest	Lowest	99.69

4.3 Results of the Trolley Position Control using Hybrid PID/Shapers Controller

Figure 4.2 are the trolley position control response using PID controller alone PIDZV, PIDZVD and PID/ZV-ZVD.

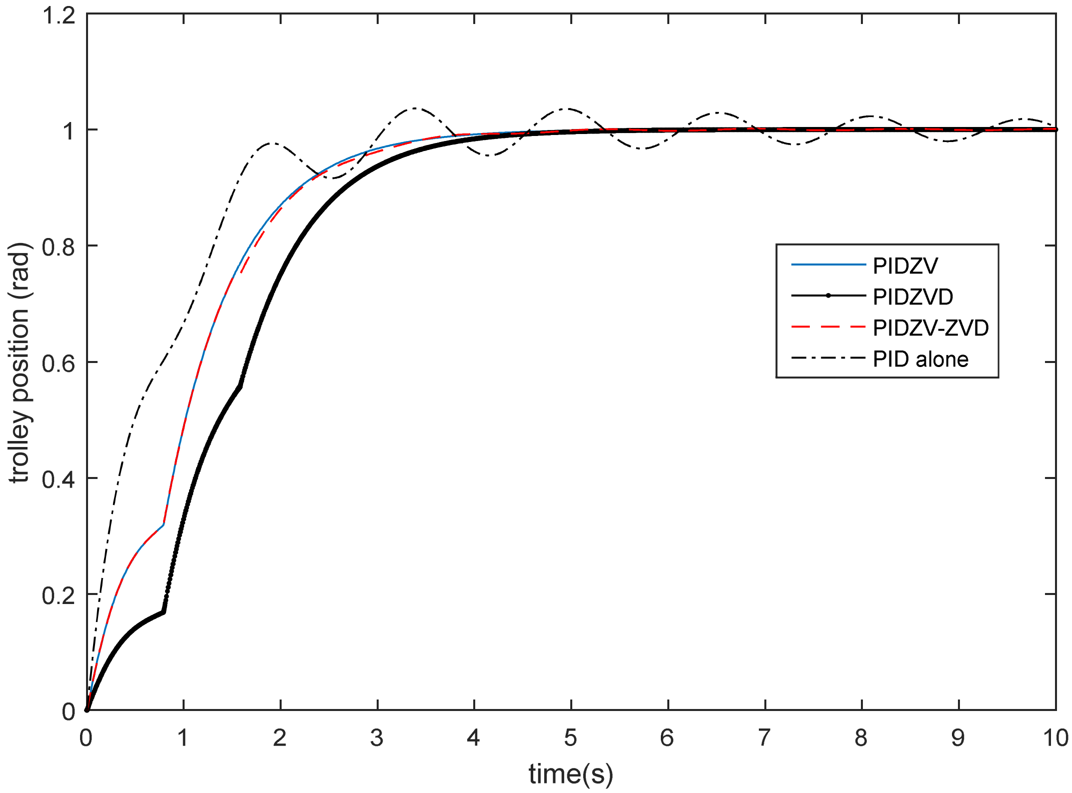


Figure 4.2 Trolley Position Control Responses with Hybrid PID Shapers.

Here the trolley was excited to move to the position 1m 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 4.3.

Table 4.3 Input shaper/PID performance Parameter

Controller	**Rise time (s)**	**Peak time (s)**	**Maximum 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

From the Table 4.3, it is evident that, the proposed hybrid PID/ZV-ZVD controller has an improved performance both in speed and accuracy of position tracking (rise time of 2.0652s and settling time3.369s) as compare to PIDZV and PIDZVD with 2.1027/4.513 and 2.3923/3.852 respectively.

CHAPTER FIVE

SUMMARY, CONCLUSION AND RECOMMENDATIONS

5.1 Summary

In this research, different strategies of gantry crane system control were reviewed and the design of hybrid controllers for sway suppression and precise trolley position control was successfully achieved. The hybrid controllers designed in this work include: a hybrid of ZV and ZVD input shaper combined with a proportional-integral-derivative (PID) Controller. In both cases, ZV with ZVD; which are feedforward controllers were responsible for sway suppression and the PID was responsible for trolley position control. The responses of gantry crane system without any controller/Shaper, with Hybrid Input Shaper, with PID alone and with Hybrid Shaper/PID were simulated using MATLAB/Simulink (R2015a) Software. Finally, the performance of the developed Hybrid Controller was presented.

5.2 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 (R2015a) software platform and the result showed that the Input Shaper was able to suppress the sway appreciably resulting in up to 99.6% reduction in sway angle. Also, the precise trolley position control was achieved with the Hybrid controllers (ZV-ZVD and PID).

5.3 Contribution to Knowledge

This research work yields the following contributions:

1. Hybrid Zero-Vibration Input Shaper was developed for a 2D GC which yields a 99.6% reduction in payload sway angle thereby, improves the safety and efficiency of crane operation.
2. Improvement in trolley position control as a result of incorporation of Zero-Vibration Input Shaper (feed-forward controller) and PID (feedback controller). The proposed hybrid PID/ZV-ZVD position controller successfully eliminate the overshoot and reduced the settling time to 3.369s

5.4 Recommendations

Areas for future work can include:

1. Using real time implementation results to validate simulation results.
2. Application of intelligent controllers such as fuzzy logic controllers, neural network etc. and investigating their performances.
3. Application of other vibration controllers such as higher order differentiators (HODs), investigating their performances and finally compare their performances with that of Hybrid ZV/ ZVD.

REFERENCES

Abe,A., (2011). Anti-Sway Control for Overhead Cranes Using Neural Networks. *International Journal for Innovative Computing, Information and Control, 7(7),4251-4262.*

Ahmad M.A.,Mohamed Z.,(2009). Hybrid Fuzzy Logic Control with Input Shaping for Input Tracking and Sway Suppression of a Gantry Crane System. *American J. of Engineering and Applied Sciences 2 (1): 241-251.*

Alhazza k. and Masoud Z., (2013). A Novel Waveform Command Shaper for Overhead cranes. *Journal of Engineering Research, 1(3),181-209.*

Al-Mousa, A. A. (2000). *Control of Rotary Cranes Using Fuzzy Logic and Time-Delayed Position Feedback Control*.Unpublished, master's Thesis, Virginia Polytechnic Institute and State University.

ArolovichI.andAgronovich G., (2014). Control Improvement of Under-Damped and Structure of Input Shaping. *Ariel University of Samari*.

Bansal H.O.,Sharma, R., &Shreeaman, P.R., (2012). PID Controller Tunning Technique: A review, *Journal of Control Engineering and Technology, 2(4), 168-176.*

Bhagwan, S., Kumar A.,andSoni, J.S., (2016). A review on PID Controller.*International Journal on Recent Technologies in Mechanical and Electrical Engineering, 3(2) 17-22.*

Binitha, S., &Sathya, S.S.,(2012). A Survey of Bio inspired Optimisation Algorithm. *International Journal of Soft Computing and Engineering, 2(2), 137-151.*

Bruins S. (2010). Comparison of Different Control Algorithms for a Gantry Crane System.*Intelligent Control and Automation*.**1**, 68-81.

Eiben, A.E., &Schippers, C.A., (1998). On Evolutionary Exploration and Exploitation

FundamentaImformaticae, 35(1-4),35-50.

Samin R.E., Mohamed Z., Jalani J., Ghazali R., (2013). A Hybrid Controller for Control of a 3-DOF Rotary Crane System.*First International Conference on ArtificialIntelligence, Modelling & Simulation,*https://www.researchgate.net/publication/288484302.

Han, J., Zhu, Z., He, Y., Qi, J., (2015). A novel input Shaping Method base on system output.*Journal of Sound and Vibration 335,338-349.*

Hong, K. & Ngo, Q.H. (2009). Port Automation: Modeling and Control of Container Cranes. *International Conference on Instrumentation ,Control and Automation ICA2009,19-26.*

Huey,J.R.(2006).*The Intelligent Combination of Input Shaping and PID Feedback Control*.Unpublished, Ph.D Thesis, School of Mechanical Engineering Georgia Institute of Technolgy.

Igo, A., Grigory A.,(2014). Control Improvement of Under –damped Systems and Structures by Input Shaping. *Ariel University of Samaria.*

Karajgikar, A., Vaughan, J., and Singhorse, W., (2011). Double Pendulum Crane Operator Performance comparing PD –feedback control and Input Shaping. *Advances in Computational Multibody Dynamics, 1-14*

King, S.S., (2006). *Command Shaping Control of a Crane System.*Unpublished, Masters Thesis, UniversitiTeknologi Malaysia.

Kolar, B. &Sclacher, K., (2013). Flatness based Control of a Gantry Crane. 9th IFAC Symposium on Nonlinear Control System.487-492.

Mandal,A. J., (2006). Introduction to Control Engineering. (1st edition), *New Age International (P) Ltd.*

Mohammad, S., (2015). Anti-Swing Fuzzy Controller Design for a 3-D Overhead Crane. *Journal of modern Processes in Manufacturing and Production, 4(2),57-66*

Nimbakar, R. B., Lavhate,S. B., &Toradmal, N. D., (2015). A review of Significance of PID Controller for Speed Control of DC motor.*International Journal on Recent and Innovation Trends in Computing and Communication,3(4)2080-2088.*

Ogata, K., (2002). Modern Control Engineering.(4th edition), Prentice Hall, New Jersey.

Osma, M., Abied, W., and Elfandi, S., (2013).Tunning PID Controller Based on the SWARM Intelligence. *International Conference on Innovations in Engineering and Technology.64-67.*

Park, C.H., Park, D., & Park, J. H. (2001).Vibrational Control for a Beam-type substrate Transport Robot.*International Journal of Advanced Robotic Systems, 10(303), 1-7.*

Pradeepkannan, D. &Sathiyamoorthy, S., (2014). Control of Nonlinear Spherical Tank Process Using GA tunned PID Controller.*International Journal of Innovative Research in Science, Engineering and Technology, 3(3), 580-586.*

Rajinikanth, V. &Latha, K., (2012), Setpoint Weighted PID Controller tunning for unstable system using heuristic algorithm. *Archives of Control Sciences.229(4), 481-505.*

Ryan R. O.&Jinjun S. (2011). Experimental Study on Combining a Simple InputShaper and Adaptive Positive Position Feedback Control.*American Institute of Aeronautics and Astronautics Guidance, Navigation, and Control Conference.*1-12.

Singhorse W., Lisa, P., Michael, K., Eric, K., (2000). Effect of Hoisting on Input shaping Control of Gantry Cranes.*Control Engineering Practice 8.1159-116*

Singhose, W., (2009). Command Shaping for flexible systems: a Review of First 50 years. *International Journal of precision engineering and Manufacturing, 10(4),pp.153-168.*

Soleiman, K., Andersen, C.L., Lauritzen, R., Dalsgaard, J.,Krautz, M., (2009). Gantry Crane Control. Unpublished, master's Thesis, Aalborg University Esbjerg.

Steven, W.S., Hung N., & Rob, J.,(2007). Model Predictive Control of Gantry Crane with Input Nonlinearity Compensation. Retrieved September 18, 2018, 11:22am from www.Gooogle.com.

Syvertsen, P.G., (2011). *Modelling and Control of Crane on offshore Vessel.*Unpublished, Master's Thesis, Nowegian University of Science and Technology.

Thalapil, J., (2012). Input Shaping For Sway Control in Gantry Cranes.*IOSR Journal of Mechanical and Civil Egineering (IOSRJMCE) 1(2).36-46.*

Wahyudi and Jalani, J., (2005). Design and Implementation of Fuzzy Logic Controller for Intelligent Gantry Crane System. Proceedings of 2nd International Conference on Mechatronics, ICOM'05.pp345-351Kuala Lumpur, Malaysia

Wahyudi, Jamaludin, J., Riza, M. &Momoh, J. E. S. (2007). Control Strategy for Automatic Gantry Crane Systems: A Practical and Intelligent Approach. *International Journal of Advanced Robotic Systems.***4**. (4)*,* 447-456*.*

Zhu, Z., Liu, K., He, Y. & Han, J. (2014).An Input Shaping Method Based on System Output. *Sensors & Transducers.***172**. (6)*,* 254-262*.*