POSITION CONTROL OF ELECTRO-HYDRAULIC SYSTEM USING PID-MRAC APPROACH

MUHAMAD SYAZWAN BIN ABDUL KADIR

UNIVERSITI TEKNIKAL MALAYSIA MELAKA

POSITION CONTROL OF ELECTRO-HYDRAULIC SYSTEM USING PID-MRAC APPROACH

MUHAMAD SYAZWAN BIN ABDUL KADIR

A report submitted
in fulfilment of the requirements for the degree of
Bachelor of Mechanical Engineering

Faculty of Mechanical Engineering

UNIVERSITI TEKNIKAL MALAYSIA MELAKA

2019

DECLARATION

I declare that this report entitled "Position Control of Electro-hydraulic System Using PID-MRAC Approach" is the result of my own work except as cited in references.

.

Signature : ....

Name : ....

Date : ....

APPROVAL

I hereby declare that I have read this project report and in my opinion this report is sufficient in term of scope and quality for the award of the degree of Bachelor of Mechanical Engineering.

Signature	:	…………………………
Supervisor's Name	:	…………………………
Date	:	…………………………

DEDICATION

I dedicate this thesis to the Almightiest which is Allah, my creator, my strong pillar, which is one of my sources of inspiration and the one that give me strength when I thinking of giving up. Big thanks to my beloved family and my final year's project supervisor Sir Zairulazha Bin Zainal. I also dedicate this project to my fellow friends who has lend their hand to help me to complete the studies and give reinforcement when I'm in need all the way and whose encouragement has ensure me to give all I have no matters what it takes to finish the things that I have started.

ABSTRACT

Electro-hydraulic system is a type of feedback-controlled parameter regulates a hydraulic actuator, whether linear or rotary. The input, e.g. displacement or load, is commanded and the actual parameter is measured. An error signal is generating and apply to a servo-valve controller. The waveform produced can be any form of wave either static or dynamic, one shot or cyclic. It is very flexible and can be applied commanded complex pre-programmed load or displacement parameters. The range of loads can be applied is wide and the displacement precise. The purpose of this study is to design a PID controller with MRAC approach to improve the performance of the system. A model is obtained from a journal and all the parameters has been identified. The parameters are then will be derived to get the system transfer function.

ABSTRAK

Sistem elektro-hidraulik adalah sejenis parameter yang dikendalikan maklum balas mengawal penggerak hidraulik, sama ada secara linear atau berputar. Input, seperti pergerakan atau beban, dikawal dan parameter sebenar diukur. Isyarat ralat dijana dan dihantar kepada sistem servo-valve. Bentuk gelombang yang terhasil boleh menjadi apa jua bentuk gelombang sama ada statik atau dinamik, satu pukulan atau kitaran. Ia sangat fleksibel dan boleh digunakan untuk mengarahkan beban kompleks pra-program atau parameter pergerakan. Beban yang boleh digunakan adalah luas dan ketepatan pergerakan. Tujuan kajian ini dijalankan adalah untuk mereka bentuk pengawal PID bersama pendekatan MRAC untuk meningkatkan prestasi sesebuah sistem. Model diambil dari jurnal dan semua parameter akan dikenal pasti. Parameter-parameter itu kemudiannya akan diderivasi untuk mendapatkan fungsi pemindahan sistem itu.

ACKNOWLEDGEMENT

First off all, I would like to use this opportunity to give a big thanks to my supervisor Sir Zairulazha from the Faculty of Mechanical Engineering Universiti Teknikal Malaysia Melaka (UTeM) for his essential supervision, support and encouragement towards the completion of this project report. Special thanks to all my fellow friends, my beloved family and siblings for their words and advices for supporting me in completing this degree. Last but not least, thank you to everyone who had been involved directly or indirectly in completing this project.

TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

LIST OF ABBREVIATIONS, SYMBOLS, SPECIALIZED NOMENCLATURE

PID	-	Proportional Integral Derivative
MRAC	-	Model References Adaptive Control
LVDT	-	Linear Variable Differential Transformer
ω_γ	-	Natural frequency of servo valve
ζ_γ	-	Damping ratio of servo valve
Q_v	-	Related flow rate of servo valve
C_t	-	Total leakage coefficient
K	-	Torque motor grain of servo valve
P_s	-	Supply pressure
B	-	Bulk module of oil
ρ	-	Mass density of oil
M_e	-	Equivalent mass of both the load and the piston
d_{rod}	-	Diameter of load
d_{piston}	-	Diameter of piston
X_p	-	Max stroke of cylinder
L	-	Length of pipeline and hoses from pump of cylinder
F_{fc}	-	Cylinder coulomb friction force

LIST OF SYMBOL

c	=	Damping coefficient
ξ	=	Damping ratio
F_e	=	Excitation force
F_t	=	Force transmitted
i	=	$\sqrt{-1}$
m	=	Mass
k	=	Spring stiffness
ω	=	Operating frequency
ω_n	=	Natural frequency
T_f	=	Transmissibility
x	=	Displacement
$\dot{x}$	=	First derivatives of x
$\ddot{x}$	=	Second derivatives of x
Kp	=	Proportional Gain
Kd	=	Derivative Gain
Ki	=	Integral Gain

CHAPTER 1

INTRODUCTION

1.1 Background

The investigation for the looking of electro-hydraulic system with stable highlights, for example, elite level, quality guaranteed, environmentally-friendly and easy for maintenances has flooded explores to consider and grow facilitate on the capacity and abilities of electro-hydraulic system framework.

Electro-hydraulic system are generally utilized in industry and mechanical applications. The nonlinearities of an electro-hydraulic system including liquid nonlinearity, uneven mechanical attributes, and so on, cause the control execution changing with the moving course and the situation of the piston while utilizing a linear controller. The vulnerabilities of the pressure driven framework including load, erosion, and so on, will also influence the control execution. These two perspectives influence a conventional straight controller to be constrained to completely misuse the dynamic ability of the hydraulic system framework.

Electro-hydraulic system framework has become logically mainstream in different types of engineering equipment and by using it worthwhile and diverse applications, for example, flying machines, producing machines, fatigue testing, pressure driven excavator, sheet metal forming process and car applications built up so the actuator system will be more surely understood and important these days. However, the electro-hydraulic system is known as a convoluted framework which experiences vulnerabilities, nonlinearities and disturbances. These bothers may prompt degradation of control execution in trajectory tracking of the electro-hydraulic system.

Different kinds of criticism controller extending from direct to nonlinear compose are generally executed and distributed among the scholarly community and scientists for direction following control of electro-hydraulic system. The expanding quantities of works managing electro-hydraulic system over the previous decades included a linear control, intelligent control and nonlinear control methodologies, for example, Neural Network (NN), Self-tuning Fuzzy-PID, Model Reference Adaptive Control (MRAC), Generalize Predictive Control (GPC) and Sliding Mode Control (SMC).

Therefore, a feedback control methodology is dependably needed in outlining a great functioning trajectory tracking and positioning of electro-hydraulic system.

1.2 Problem Statement

The electro-hydraulic system is an electrically operated valve that control how hydraulic fluid transferred to an actuator. Servo valve operates by transforming the analog or digital signal into a movement in hydraulic cylinder. This system combined two control modes of electrical and mechanical. However, the dynamic properties of electro-hydraulic system is highly nonlinear and cause this system difficult to control. Non-linear system also may cause the system to be non-smooth and discontinuous due to directional change of valve opening, friction and valve overlap (Hong et al., 2004). Mathematical model of a system will use to model and simulate the system (Dechrit, 2009). Modelling and simulation is to optimize the dynamic performance by controlling the speed and displacement of the actuator.

This project will study on the characteristic of dynamic system execution of an electro-hydraulic system by using modelling and simulating in MATLAB and Simulink. The transfer function of the system will be developed for modelling and simulating in MATLAB and Simulink. Electro-hydraulic system will move with various speeds of piston motion. Displacement sensor was mounted to the hydraulic piston for measuring piston motion. The electro-hydraulic system will be run by a Simulink. This research also applied closed loop control during conducting experiments.

1.3 Objectives

The objectives for pursuing the current research topic are:

1. To develop mathematical models of a hydraulic servo system.
2. To model and simulates a hydraulic servo system for studying dynamic characteristics of the system.
3. To design an electro-hydraulic controller with PID-MRAC.
4. Testing the controller system by using PID-MRAC and drive the system to pursue perfectly the desired direction.

1.4 Scope

The scope of this project is:

1. Make a journal research on electro-hydraulic system on how to derive the equation and obtain complete parameters.
2. Deriving the equation and analyse all the parameters needed to the system.
3. Apply the final equation in Simulink apps and record the result.
4. Create the controller system by using PID-MRAC approach.

1.5 General Methodology

All the action that need to apply to achieve the objectives are listed below:

1. Literature Review

   Journals, articles, magazine and all the materials regarding to this project will be researched.

2. Simulation

   Simulation that will be used is Simulink apps to test the equation and PID-MRAC apps to test the controller system.

3. Analysis and Results

   Analysis will be calculated on how the product reacts to the control system whether it pursue the desire direction or not.

4. Report

   A report based on this project will be conducted at the end of this research.

CHAPTER 2

LITERATURE REVIEW

2.1 Introduction

The electro-hydraulic system is an electrically operated valve that controls the hydraulic fluid, which is ported to an actuator system. A valve is a device in the hydraulic system, which controls the flow of the hydraulic fluid. The electro-hydraulic system has the abilities to apply very large forces and torques, thus, it was being applied widely in heavy industry. Besides that, it also has high stiffness and fast response for heavy industry. Some applications of electro-hydraulic system in heavy industrial are in automotive, construction machinery, lifting and conveying devices (Markle et al., 1998). However, electro electro-hydraulic system is typically a non-linear system. It causes the system difficult to control due to problems with high non-linearity and motion friction (Shao et al, 2009).

Non-linear system is a system, which is the output, is inversely proportional to its input for certain system such as dynamics of aircraft (nonlinear dependences on speed, angles, altitudes), helicopters, satellites, and that is most systems in aerospace engineering. Therefore, for some applications these systems are quite complicated system for perfect control. A non-linear phenomenon may cause non-smooth and discontinuous nonlinearities due to directional change of valve opening, friction and valve overlap (Yao et al., 1998). Thus, there are many previous researchers had studied the dynamic characteristic of an electro-hydraulic system to develop a controller for this system. The designed controller must work properly with dealing the non-linear phenomenon and dynamic of the hydraulic servo system parameters (Nitin, 2009).

2.2 Introduction to Electro-Hydraulic System

Basic servomechanism as shown in Figure 2.1 is an electro-hydraulic system which a criticism-controlled parameter directs a pressure hydraulic actuator, either straight of revolving. A parameter, e.g. removal or load, is applied and the real parameter estimated and a blunder flag created and connected to a servo-valve controller. The instructed waveform can be any wave shape, static or dynamic, one shot or cyclic. It is entirely adaptable means suitable to any system and can connected to complex pre-modified load or uprooting parameters. Various free channels can be connected and related. The scope of load can be connected is wide and the exactly located.

Typically, an electro-hydraulic system consists of various components:

1. A hydraulic power supply providing a motive source (e.g. electric motor), a hydraulic pump (often 3000 psi) an oil sump, oil cooler and accumulator, relief valves.
2. Manifolds, hard piping, flexible hoses and fitting,
3. Hydraulic manual or servo-valves,
4. Load or displacement measuring equipment (optional for high performance and automated control)
5. Feedback-control electronics, including PID controls (optional).

Advantages of electro-hydraulic are obviously known. Higher density power, easy to control the motion, higher reliability and strength, possibility to implement customized solutions, good resistance to vibrations and able to absorb impulsive loads and management of simple thermal exchange. Every one of these factors has made hydraulic systems capable to withstand for the power actuation in highly variation fields from aeronautics to earth handling, from bigger industrial applications such as steel mills to civil applications such as lifting bridges, dams and so on.

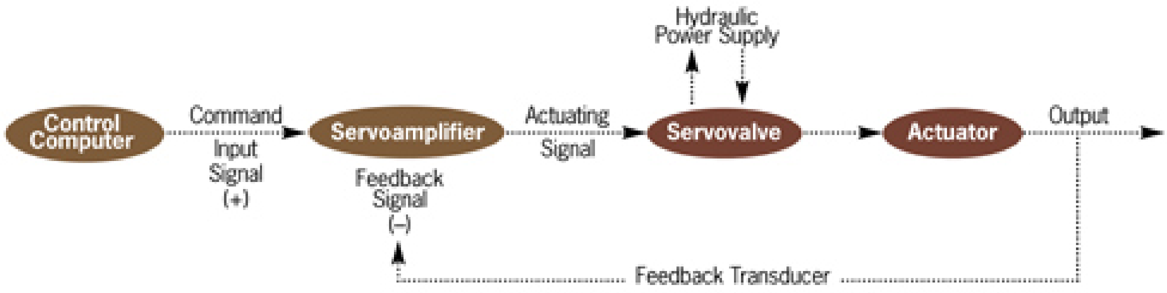


Figure 2.1 : Basic servomechanism (Karl Erik, 2008).

An electro-hydraulic system control contains of six major components showed in the picture above, which is control hardware such as a computer, microchip or system guidance which make an order input signal, a servo-intensifier which gives a low power electrical, activating signal which is the contrast between the order input signal, and the feedback signal created by the input transducer. The valve system reacts with this low electrical signal power and controls the flow of hydraulic fluid into the actuation, for example, a cylinder and chamber which positions of the devices are being controlled; and a power supply, for the most part an electric engine and siphon, which gives the flow of hydraulic fluid to deal with high pressure. The feedback transducer estimates the yield position of the actuator and changes over this estimation into a corresponding sign which is sent back to the servo-enhancer.

2.3 Design the PID controller

PID controller is one of the major control techniques. Its initial usage was in most gadgets, trailed by vacuum and strong state simple gadgets, previously touching base at the present computerized usage of chip. It contains a basic control structure which was comprehended by system administrators and which they found moderately simple to adjust. Since many controller systems utilizing PID control have demonstrated agreeable, despite everything it has an extensive variety of uses in mechanical control. As per study for a process control system led in 1989, more than 80 of the control circles are the PID type. The PID control has been used for exploration theme for some years. Since a lot of process plants are controlled by PID controllers has comparative elements, it has been discovered conceivable to set tasteful controller parameters from less plant data than an entire scientific model. These systems came about as a result of the desire to alter parameters of the controller in situ with the minimum of effort. Furthermore, in light of the fact that of the conceivable trouble and poor cost advantage of acquiring scientific models.

2.3.1 The PID Actions

Hydraulic servo system is a control system as shown in Figure 2.2, which made of by combining two modes of control of hydraulic and electric respectively. Movement of load and hydraulic transmission in the hydraulic servo system are being control by detecting the digital signal, transmitting the signal, and processing the signal by using electric and electronic components (Cheng et al., 2011). The servo systems can calculate their own output and force the output for the system to obey a command signal to the desired position and direction. The effect on incidence when the actual result under a given set of assumptions is varying from the expected result in control device and the load can be minimized as well as external disturbances in this system. For the block diagram of the hydraulic crane system. As shown in Figure 2.3.

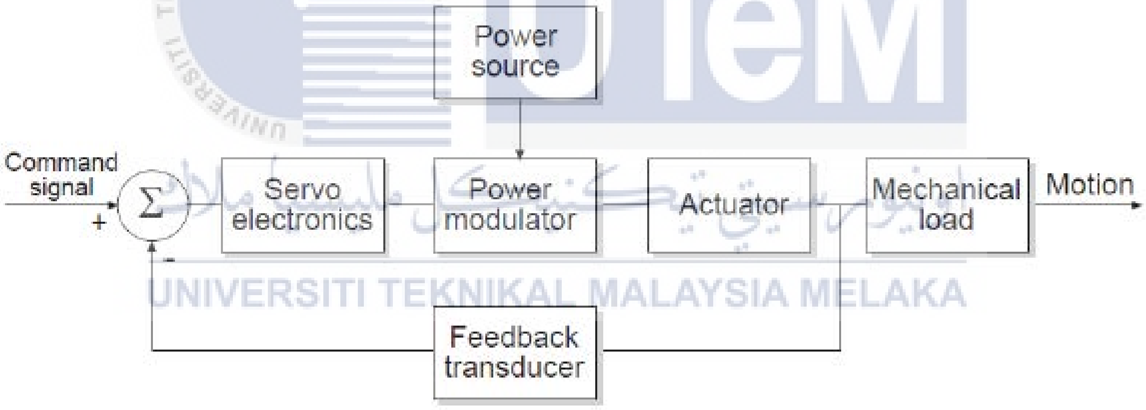


Figure 2.2: Basic servo mechanism (Karl Erik, 2008).

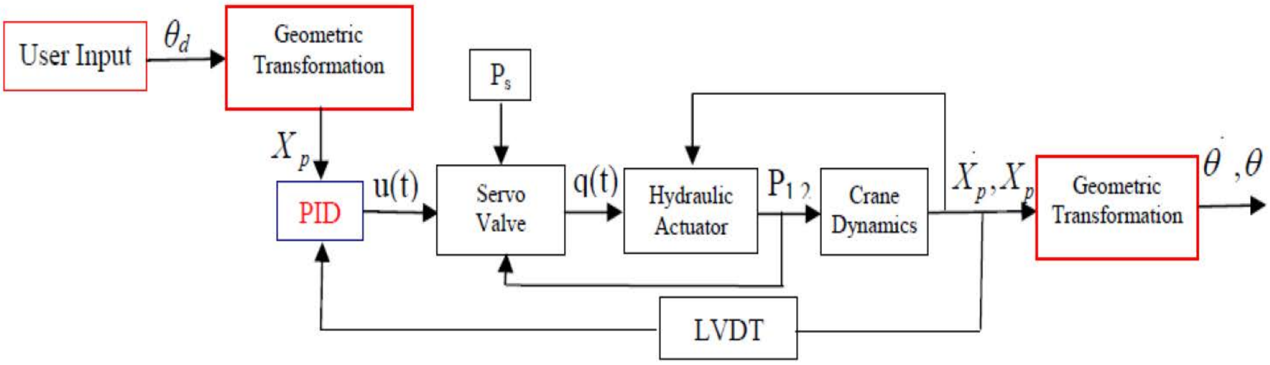


Figure 2.3: Block diagram of the hydraulic crane system.

Hydraulic servo system using an electrical signal to control the flow of hydraulic fluid with pressure for the movement of the piston. The hydraulic fluid flow is based on hydraulic power supply that will appropriately port a portion of the servo valves. The signal will trigger the fluid to navigate the actuators to move to the desired direction. A position transducer will generate electrical signals in voltage as output measures actuator. Since the hydraulic servo system is non-linear system, there might cause the system to become unstable and discontinuous due to the changes of direction of the valve opening and friction.

2.4 Modelling

The mathematical equation is a description of the system by using mathematical concepts and language. Mathematical equation of servo valves can be created by the relationship between the input voltage and the displacement for the proportional valves (Maneetham & Afzulpurkar, 2009). They had described a mathematical model of hydraulic mini press machine. The system contains electronic parts, position transducer, high speed, and hydraulic actuators. Based on basic theory of the hydraulic servo system, the transfer function can be found by simplifying the mathematical equation of each part (Shao et al., 2009). Transfer function is known as the function of the system and it represent the relation between input and output of a system.

In another studied, (Cheng et al., 2011) they were stated the basic equation of a hydraulic servo system that later will simplify into transfer function. They studied asymmetrical cylinder position control of a hydraulic servo system that is used four - way slide valve. Figure 3.1 shows the servo valve controlled asymmetrical cylinder position control system diagram. The diagram notified how hydraulic servo valve operates. Hydraulic servo valve is a valve that is control of the output current signal from the servo amplifier in the system. The actuator or load position will measure by position transducer and gives signals in voltage as output. The mathematical equation of this system will develop using basic theory of hydraulic servo valve that involves flow rate in valve, pressure and actuator or piston motion.

2.5 Simulation

Simulation is an application to identify a system performance whether the existing system or still proposed under configurations of interest. Simulink is a software under MATLAB as shown in Figure 2.4, which is a graphical environment for dynamic system modeling, analysis interactively and most importantly is for simulation. The create model in Simulink as shown in Figure 2.5, block diagrams are used to represent the model for simulation. Simulation has to be applied to test the system before an existing system or a new system is built to minimize the chances of failures and to meet the specifications of the system (Maria, 1997). The purpose of the Simulink is the complex system can be developed to simulate and test the model. By running the simulation, the dynamic behaviour of the system can be identified and thus, optimize the performance of the system. Besides that, the aim for the simulation is to build a control system in controlling the performance of a hydraulic servo system.

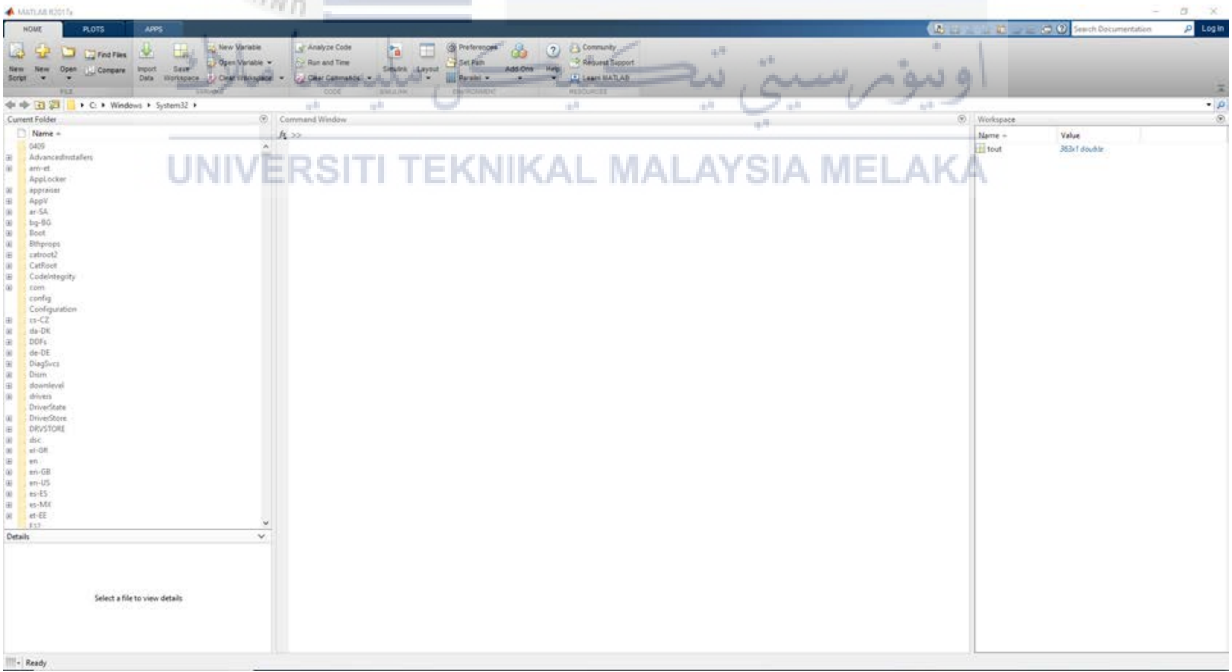


Figure 2.4: MatLab software that will be used in this project.

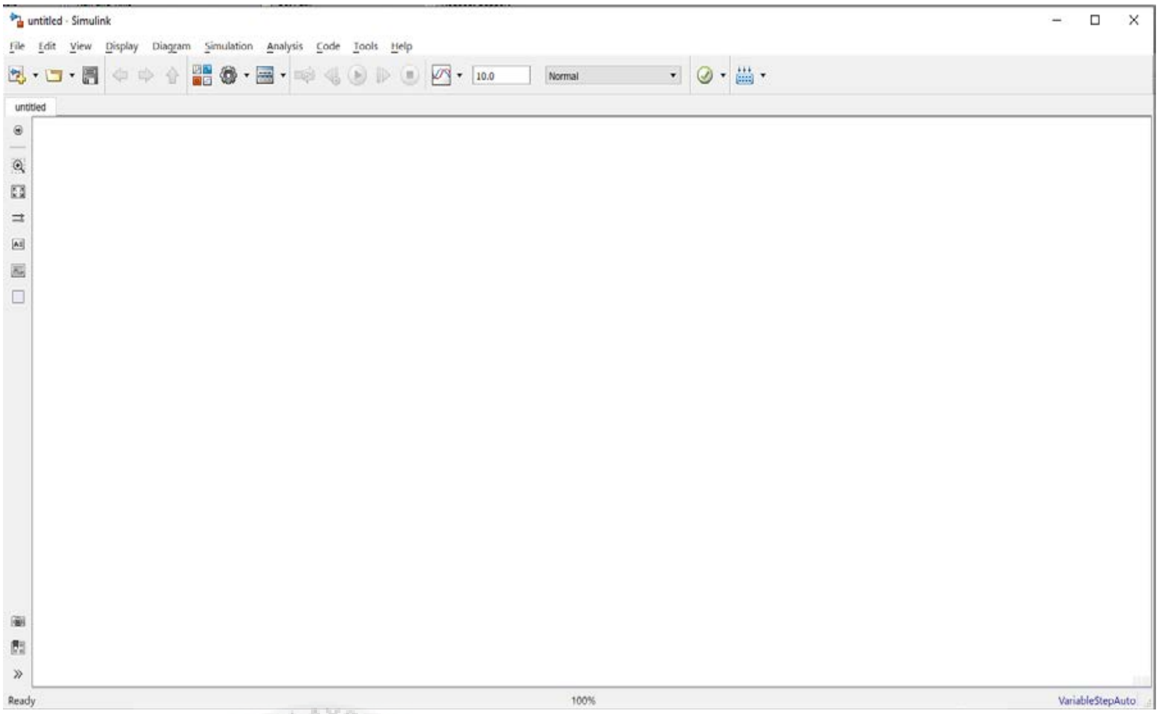


Figure 2.5: Simulink application which is used to draw a block diagram and simulation.

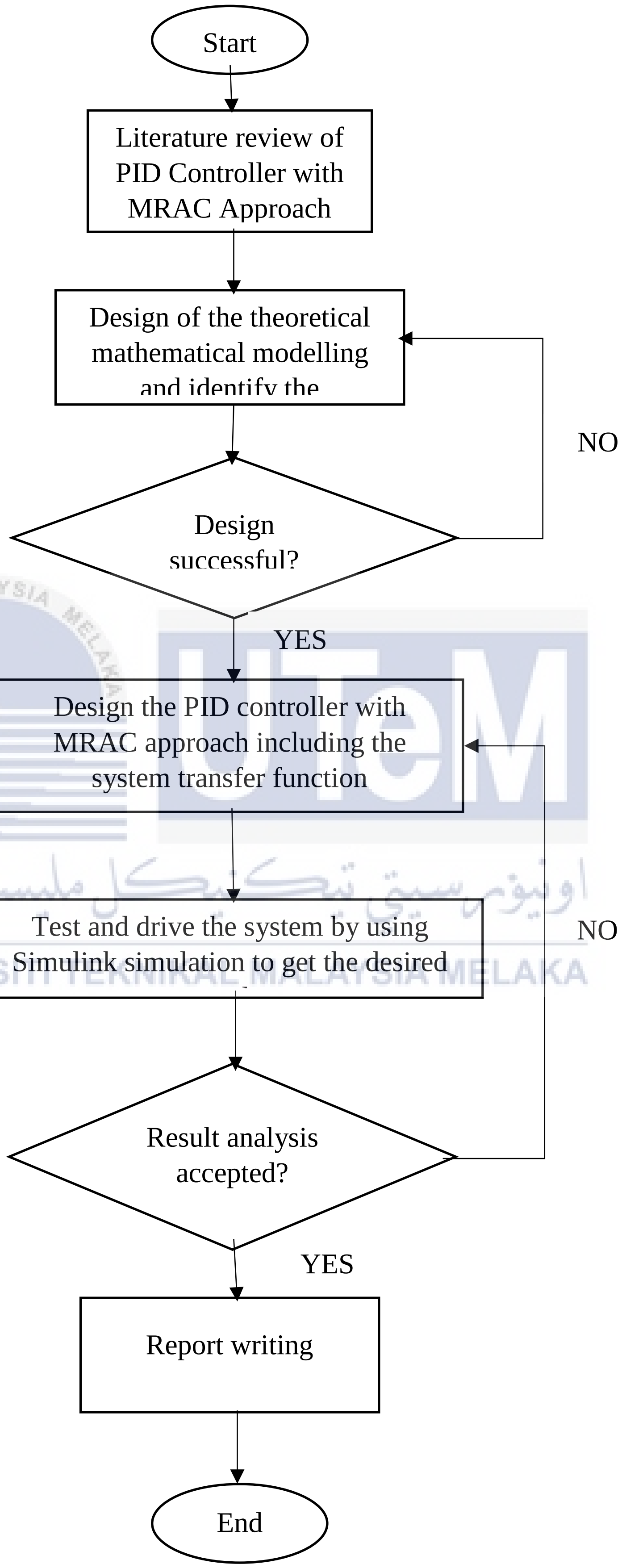
CHAPTER 3

METHODOLOGY

3.1 Introduction of methodology

In this chapter, the steps and procedure that are used to conduct this research project will be discussed. Besides, the literature studies and research will be explained. Next, the process of the transfer function will be derived as well as the hypothesis test that will be used in data analysis will be elaborated. The methodology will be used as a guide and reference for other researches who are doing studies on this similar topic. Below shows the flow chart of the methodology for this research.

3.2 Flow chart



3.3 Derivation of the system

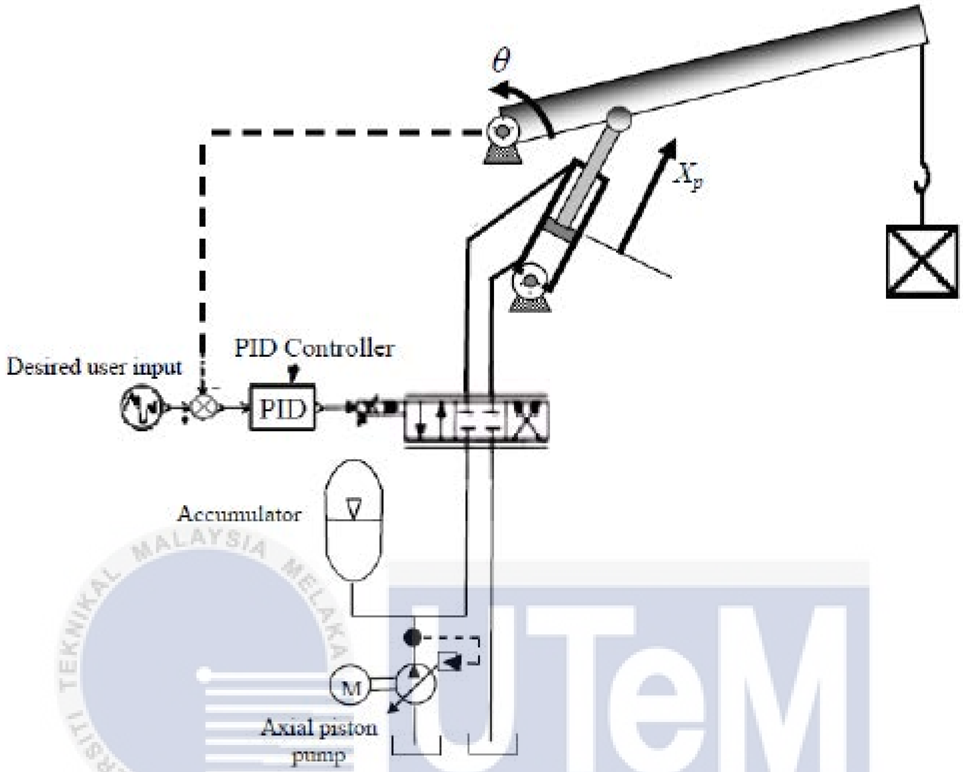


Figure 3.1: Hydraulic Crane System.

From the system Figure 3.1, we know that the position movement of the piston is controlled by the input which is voltage that is transmitted to the servo valve according to the desired position which is the controller signal current is generated. Then, there's a change of position of spool valve due to the input has been applied to the motor and the servo valve. The movement of the piston is controlled by the oil that was supplied to every chamber of the cylinder.

It is important to represent servo valve dynamics through a wider frequency range, a second-order transfer function must be used. The relation between the servo valve spool position x_v and the input current i_v is:

$$\frac{d^2x_v}{dt^2} + 2\xi_v\omega_v\frac{dx_v}{dt} + \omega_v^2 x_v = \omega_v^2 k_v i_v \tag{3.1}$$

$$i_v = \frac{1}{\omega_v^2 k_v}S^2 + \frac{2\xi}{\omega_v k_v}S + \frac{x_v}{k_v}2 \tag{3.2}$$

Where k_v is the servo valve gain, ω_v is the servo valve natural frequency, and ξ_v is servo valve damping ratio. Based on the electro-hydraulic servo system, three basic equations of the flow of the valve is:

$$Q_L = K_q X_v - K_c P_L \tag{3.3}$$

Where Q_L is load flow of the valve, K_q is the flow gain of the valve, x_v is the displacement of the valve, K_c is the flow pressure coefficient, and P_L is the load pressure.

$$Q_L = C_{tc}P_L + A_p\, sX_p + \frac{V_t}{4\beta e}sP_L \tag{3.4}$$

Where Q_L is load flow continuity of the cylinder, C_{tc} is total leak coefficient, P_L is load pressure, A_p is piston area, X_p is displacement of the piston, V_t is the total actuator volume, βe is the effective modulus.

$$F_g = P_L A_p = m_t S^2 X_p + B_p S X_p + F \tag{3.5}$$

Where *m* is the total mass of piston and payload. B_p is viscous damping coefficient. F is arbitrary external force applied on the piston. F_g is driving force occurred by cylinder.

According to the equation 3-5, we obtain the servo cylinder piston displacement calculation formula under the action of the servo valve spool load and displacement. Finally, we got the transfer function of the position control as below:

$$\frac{X_p}{X_v} = \frac{\frac{A}{K_c}}{\left(1+\frac{s}{\omega_h}\right)\left(\frac{s^2}{\omega_0^2}+\frac{2\xi}{\omega_0}s+1\right)} \tag{3.6}$$

$$X_p = \frac{\frac{K_q}{A}X_v - \frac{K_c}{A^2}\left(1+\frac{V_t}{4\beta_e K_c}\right)F_L}{\frac{m_t V_t}{4\beta_e A^2}s^3+\left(\frac{K_c m_t}{A^2}+\frac{V_t B_p}{4\beta_e A^2}\right)s^2+\left(1+\frac{B_p K_c}{A^2}+\frac{V_t}{4\beta_e A^2}\right)s+\frac{K_c}{A^2}} \tag{3.7}$$

$$\omega_h = \sqrt{\frac{4\beta_e A^2}{M_t V_t}} \tag{3.8}$$

$$\omega_h = \sqrt{\frac{4(7\text{x } 10^8)(0.0184)^2}{(1000)(1.5\text{x}10^{-3})}}$$

$$\omega_h = 794.97\ rad/s$$

Where ω_h is first order frequency, ω_0 is second-order natural frequency, and ξ is the damping ratio. Putting the data from table 3.1 into the system will produce transfer function as below:

$$G(s)\frac{225}{0.74s^3 + 1035s^2 + 2355.47s + 4696}$$

Equation of motion of the crane is given by:

$$P_{le}A_e = M_e\ddot{X}_p + B_e\dot{X}_p + F_d \tag{3.9}$$

Where, M_e represents the masss equivalent for both the inertial load and the piston, B_e is the viscous damping coefficient, and F_d is the disturbing forces such ass frictional force and so on.

Transfer function above is derived by referring to the journal which is then use for position control. However, for controlling the position of system, it requires a first order transfer function. Transfer function is an important for studying behaviors of hydraulic servo system. Furthermore, transfer function also required to develop control system. The analysis of dynamic system can be further analyzed by modeling using transfer function that obtained.

Table 3.1 : Variables symbol and value of equations.

Symbol	Name	Values
K_q	Flow gain	0.45 m^2 /s
ω_h	Natural frequency	794.97 rad/s
K_c	Pressure gain coefficient	1.6 10^6 Pa
P_L	Pressure load	14 10^6 Pa
C_{tc}	Total leakage coefficient	1.0 x 10^{-10}
A_p	Piston area	0.0184 m^2
d_p	Piston diameter	0.34 m
β_e	Effective modulus	7.0 x 10^8 Pa
V_t	Total actuator volume	1.5 x 10^{-3} m^3
m_t	Mass actuator	1000 kg
B_p	Viscous damping coefficient	100 N.s/m
ζ_h	Damping coefficient	0.5
X_p	Max cylinder stroke	1.2

3.4 Derivation of MRAC

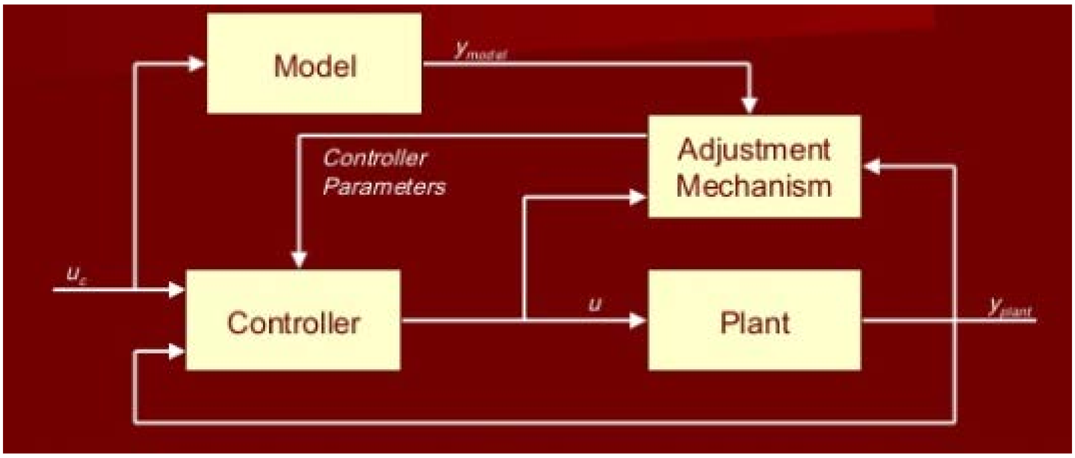


Figure 3.2: Basic Concept of MRAC (Nazir,2014).

From the Figure 3.2 show that, the basic concept of MRAC. The controller is design to drive plant response to mimic ideal response (error = $y_{plant} - y_{model}$ => 0). Parameters that has been choose is reference model, controller structure, and tuning gains for adjustment mechanism.

$$e = y_{plant} - y_{model} \tag{3.10}$$

$$J(\theta) = \frac{1}{2} e^2(\theta) \tag{3.11}$$

$$\frac{d\theta}{dt} = -\gamma \frac{\delta J}{\delta \theta} = -\gamma e \frac{\delta e}{\delta \theta} \tag{3.12}$$

Where e is the tracking error formula, $J(\theta)$ is the form cost function formula, and $\frac{d\theta}{dt}$ is the update rule formula. Change in θ is proportional to negative gradient of J.

$$J\ddot{\theta} + c\dot{\theta} + mgd\ sin\theta = dT \tag{3.13}$$

$$\frac{\theta(s)}{T(s)} = \frac{d}{Js^2 + cs + mgd} \tag{3.14}$$

$$\frac{\theta(s)}{T(s)} = \frac{0.69}{s^2 + 0.389s + 676.9}$$

By assuming the controller takes the form:

$$e = y_{plant} - y_{model} = G_p u - G_m u_c \tag{3.15}$$

$$y_{plant} = G_p u = \left(\frac{0.69}{s^2 + 0.389s + 676.9}\right)(\theta_1 u_c - \theta_2 y_{plant})$$

$$y_{plant} = \frac{0.69\theta_1}{s^2 + 0.389s + 676.9 + 0.69\theta_2} u_c$$

$$e = \frac{0.69\theta_1}{s^2 + 0.389s + 676.9 + 0.69\theta_2} u_c - G_m u_c$$

$$\frac{\delta e}{\delta \theta_1} = \frac{0.69\theta_1}{s^2 + 0.389s + 676.9 + 0.69\theta_2} u_c$$

$$\frac{\delta e}{\delta \theta_2} = -\frac{0.69\theta_1}{s^2 + 0.389s + 676.9 + 0.69\theta_2} y_{plant}$$

From MIT Rule, update rules will be:

$$\frac{d\theta_1}{dt} = -\gamma \frac{\delta e}{\delta \theta_1} e = -\gamma \left(\frac{a_{1m}s + a_{2m}}{s^2 + a_{1m}s + a_{2m}} u_c\right) e \tag{3.15}$$

$$\frac{d\theta_2}{dt} = -\gamma \frac{\delta e}{\delta \theta_2} e = \gamma \left(\frac{a_{1m}s + a_{2m}}{s^2 + a_{1m}s + a_{2m}} y_{plant}\right) e \tag{3.16}$$

CHAPTER 4

RESULTS AND DISCUSSION

4.1 Simulation using and without using PID controller

The system is testing in Simulink simulation with sampling period to be 10 seconds. The input was a step signal which occurs at 0 to 10. The parameters has been identified by using the self-tuning method and has been tested with a lot of suitable amount to achieve stability. The best value tested is as follow with the proportional gain is set to Kp=900 and tune with various value of Ki=150, and Kd=100. The system without PID Controller as shown in Figure 4.1, System response without PID Controller as shown in Figure 4.2, System with PID Controller as shown in Figure 4.3, System respond with PID Controller using step input as shown in Figure 4.4.

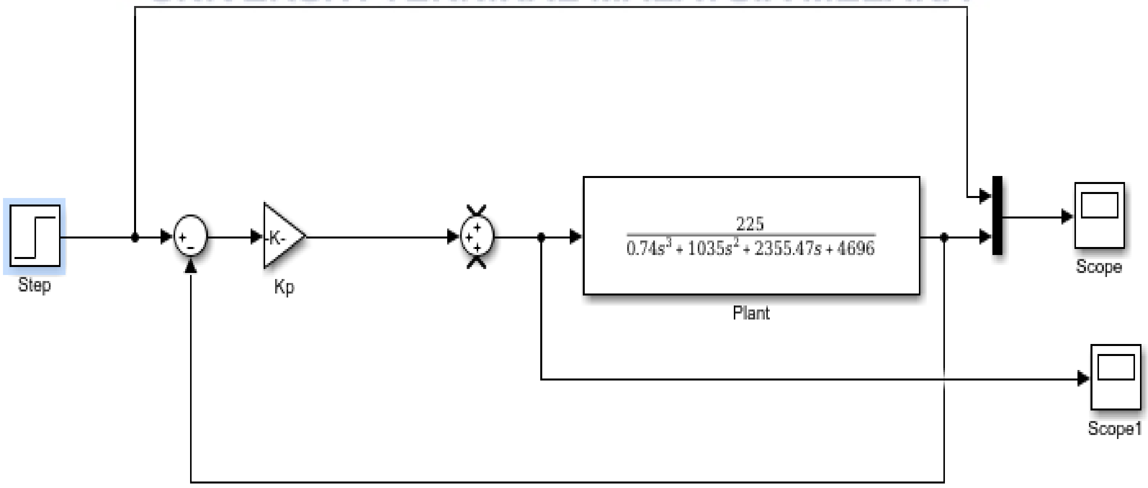


Figure 4.1: System without PID Controller.

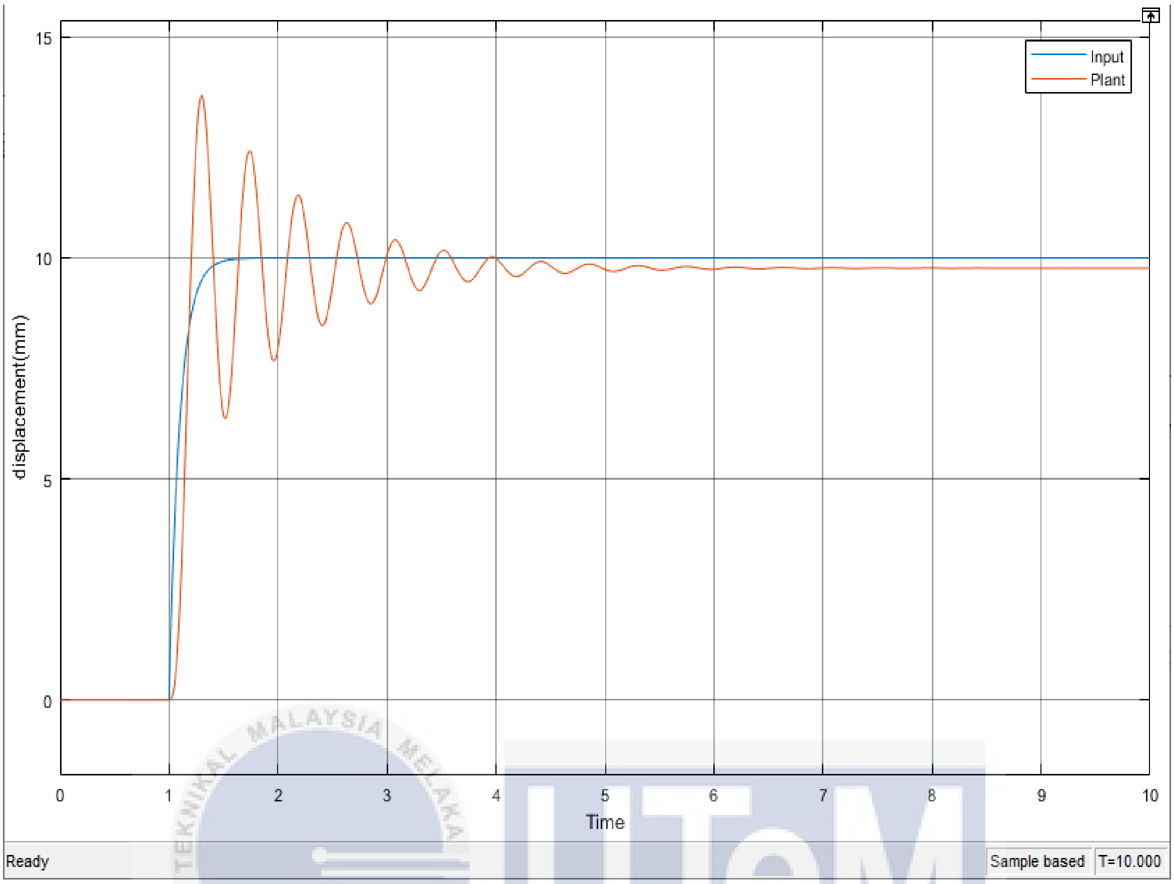


Figure 4.2: System response without PID Controller.

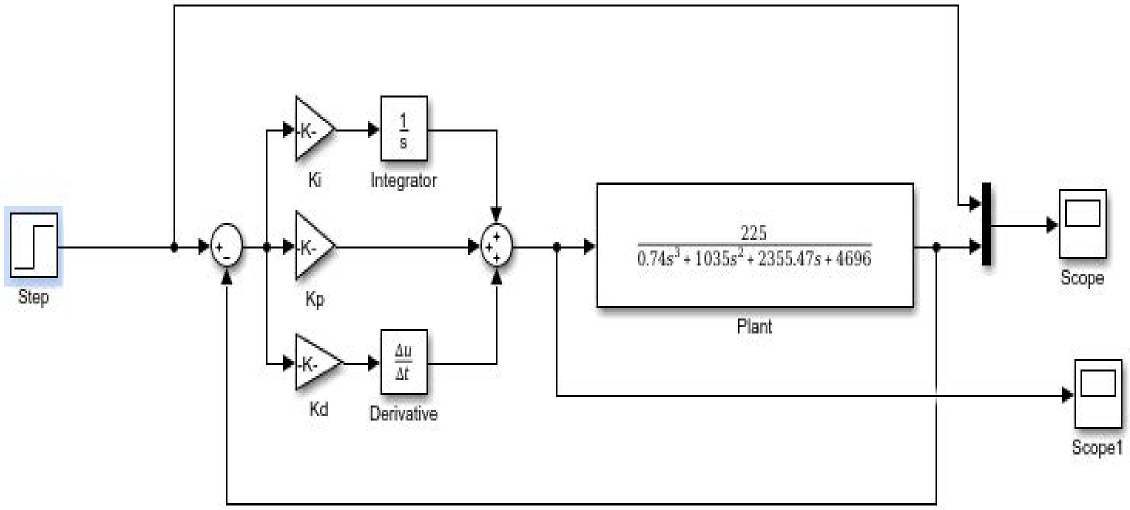


Figure 4.3: System with PID Controller.

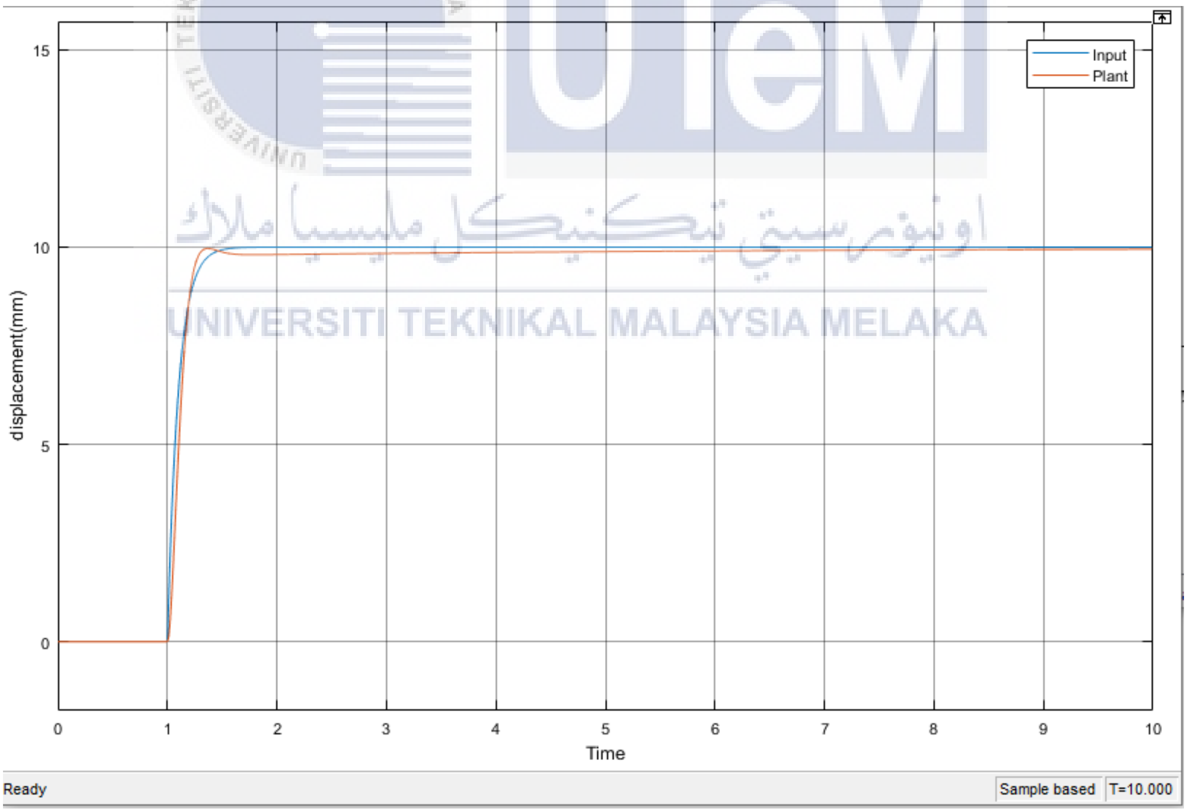


Figure 4.4: System respond with PID Controller using step input.

4.2 Simulation by using Model Reference Adaptive Control (MRAC)

Model reference adaptive controller (MRAC) technique initially purpose to overcome all incoming problem that was associated with the required fulfilment specification that was delineate in terms of reference model. The objective of using the MRAC technique is to obtain the transfer function from reference signal, r to output signal, y that is equal to the transfer function of the desired reference model. Figure 4.5 shows a basic model of MRAC including some main components. The main components are the things that we want to control which is plant, reference model, adaptive mechanism and the controller.

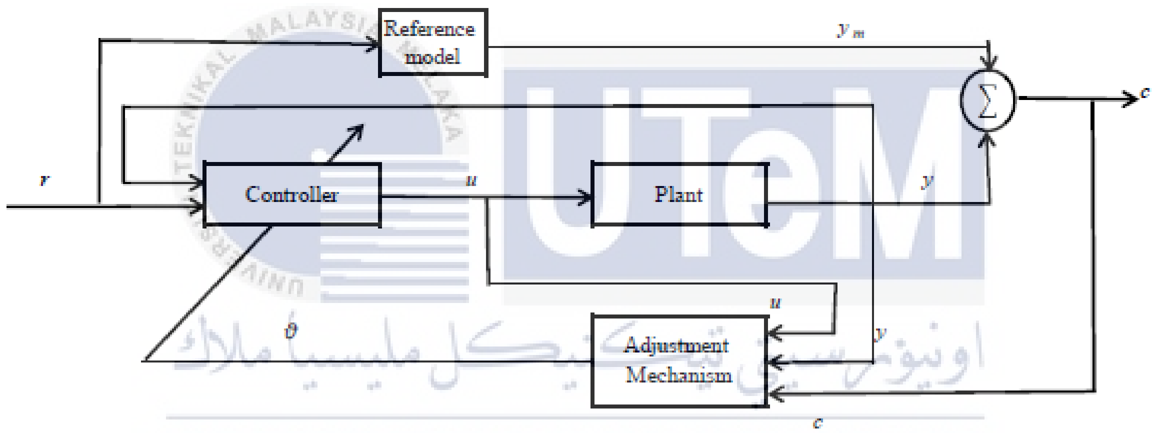


Figure 4.5: Basic Model of MRAC.

After conducting some literature review, watching YouTube, and reading Control Engineering book about MRAC, a system with PID controller and MRAC has been designed with the help of some journal about MRAC technique. The plan transfer function has been applied to the MRAC technique. The model reference has been chose from a journal (Ayman, 2012) which the writer used to test his hydraulic system.

During the earlier tuned with smaller parameters of Kp, Ki, and Kd, it’s hard to get the desired output to be as the reference model. After increasing the parameters to reach hundreds to thousands value, the output seems to reach the reference model. Finally, the parameters tested has achieve to the nearest output of the reference model. The value of the parameters is Kp=5000, Ki=1500, and Kd=1200. The simulation of the MRAC technique has been conducted with step input and sine wave. The sine wave properties have been set the amplitude, A=10 mm, and phase=0. The system with PID Controller and MRAC as shown in Figure 4.6. System response of PID Controller and MRAC using step input as shown in Figure 4.7

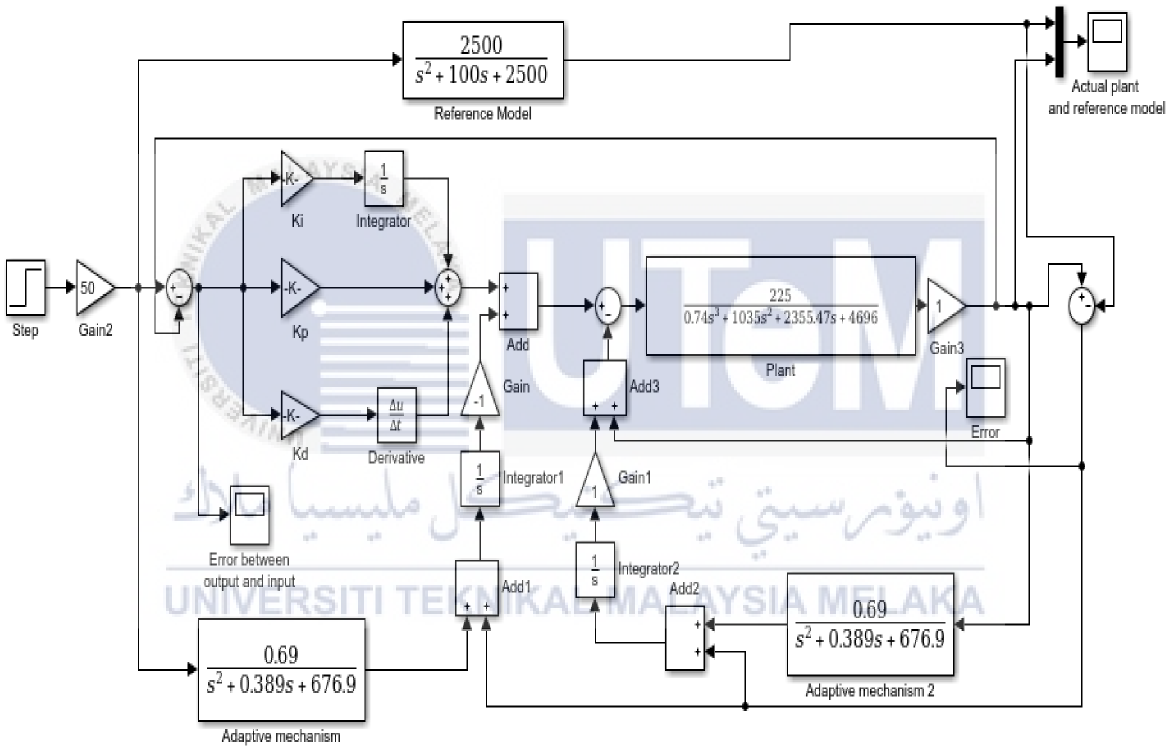


Figure 4.6: System with PID Controller and MRAC.

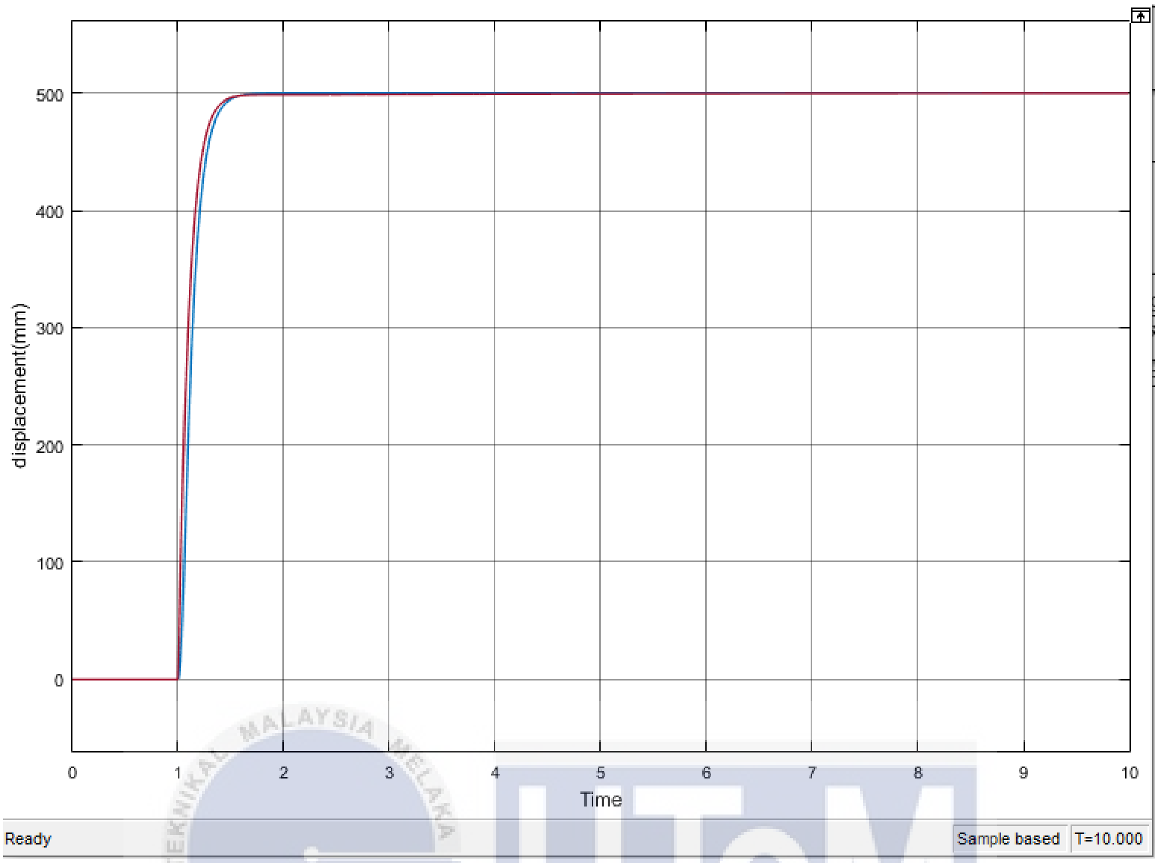


Figure 4.7: System response of PID Controller and MRAC using step input.

4.3 Performance evaluation

The performance of the system from the system without PID controller until using PID-MRAC controller can be seen from the overshoot, rise time, and vibrating from the graph.

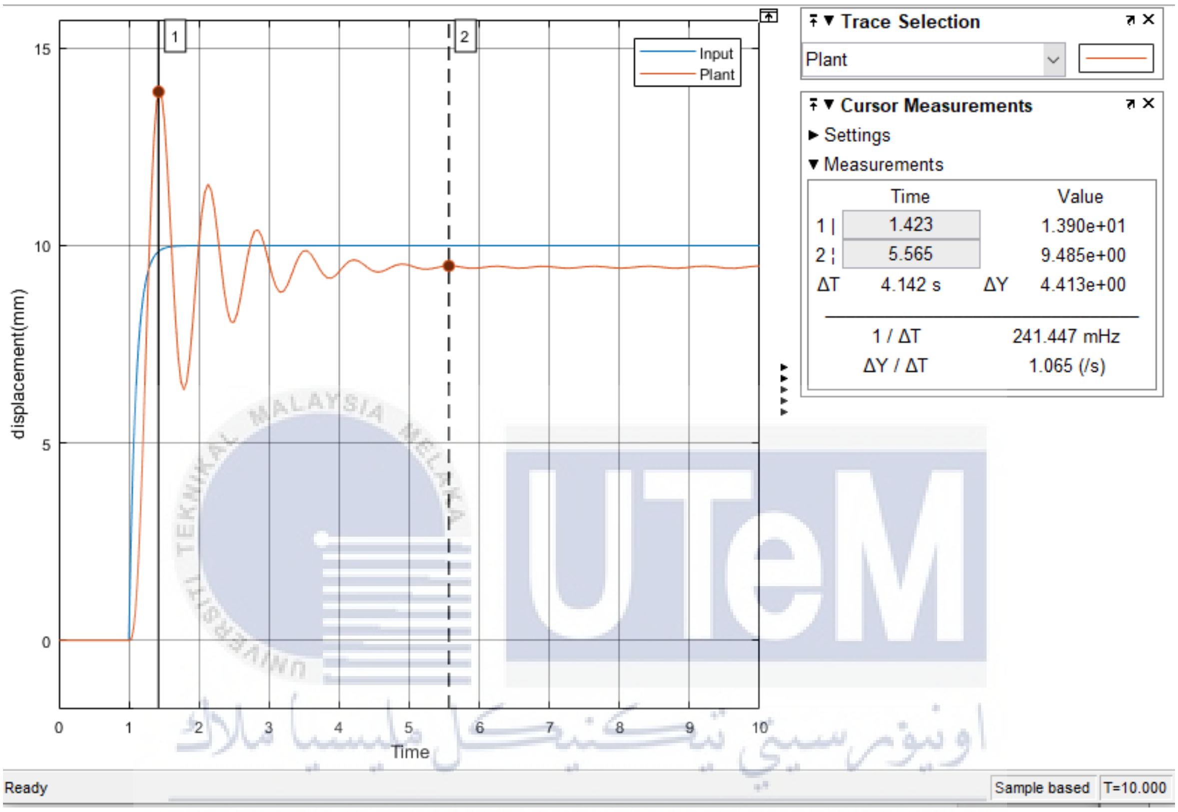


Figure 4.8: System performance without PID.

Based on Figure 4.8, the performance of the plant is worst as the graph produce larger overshoot. It is also vibrating which will make the system become unstable. The rise time also lagging about +/- 0.2s. The system stops vibrating at +/- 5.565s.

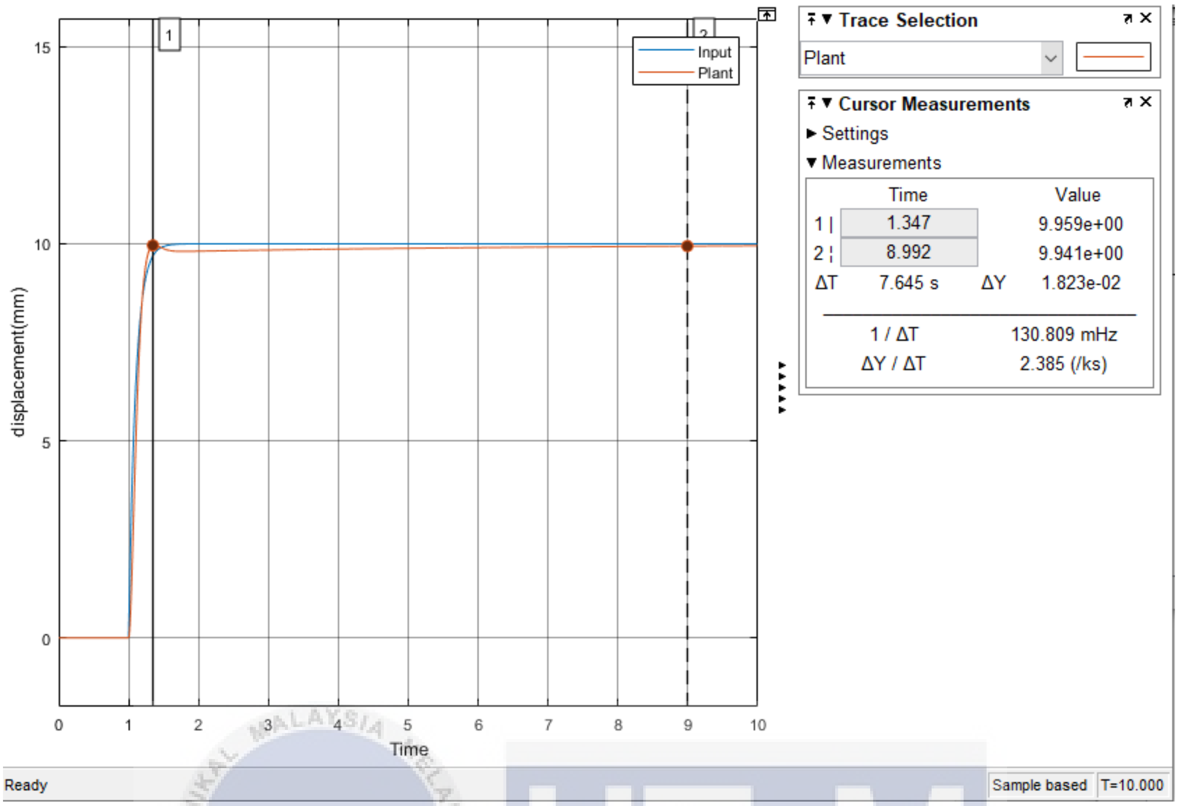


Figure 4.9: System performance with PID.

From the Figure 4.9, the performance is getting better due to the overshoot is very small compare to the input. There is also no vibrating on the system which make the system more stable. The rise time is a bit lagging about +/- 0.05s from the input. However, the system does not achieve the desired position and take about 8.992s to achieve the desired position.

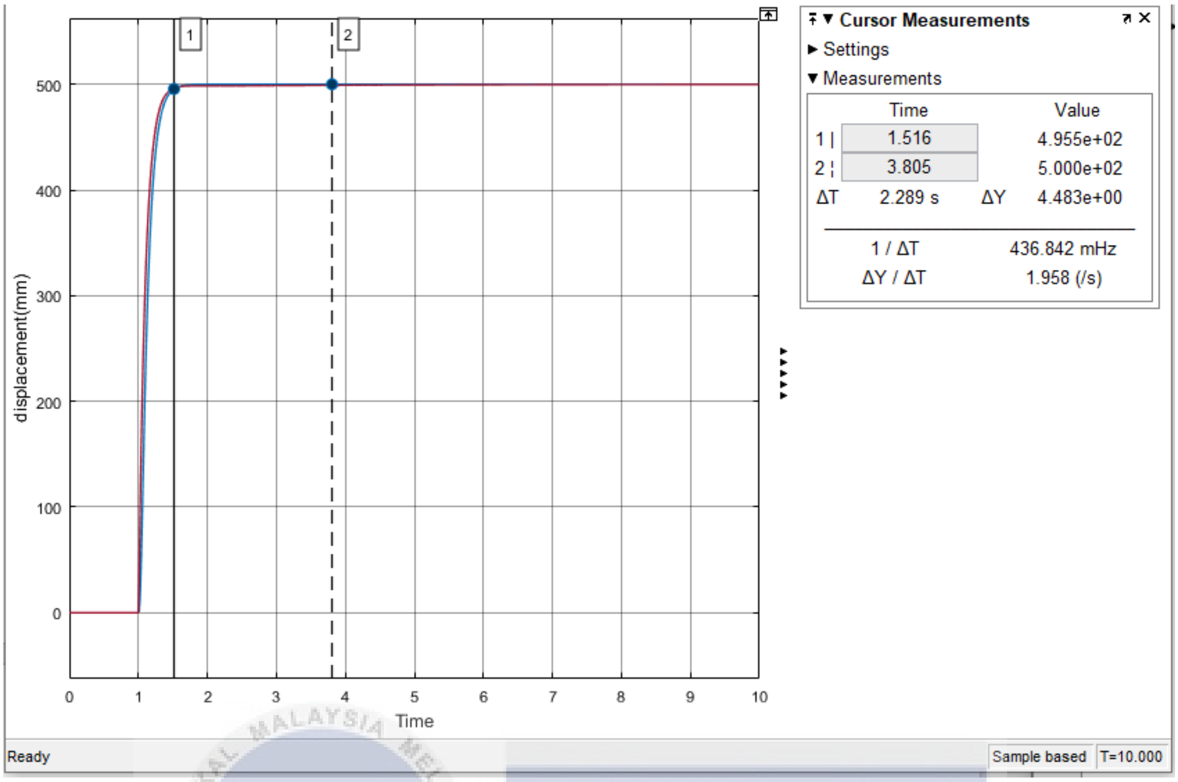


Figure 4.10: System performance with PID-MRAC.

Based on the Figure 4.10, it is clearly shown that the output response manages to follow the input signal given with minimum phase lagging and small tracking error. The rise time difference is about 0.04 s and the steady state error is ± 0.04s. There is also no overshoot and vibrating on the system.

CHAPTER 5

CONCLUSION AND RECOMMENDATION

5.1 Conclusion

After conducting this project, electro-hydraulic system is a quite difficult system to understand the nonlinear dynamic behaviors due to the its unique and diverse. It is too complex to derive all the equation regarding to the system and to identify the global nature from local nature through linear analysis. Thus, the system is frequently tuned. To achieve the effective, assess the execution of the calibration methodology, the system is tested through simulations.

Due to the highly nonlinear, the system maximum the performance of the of the linear controllers that was used for his purpose. After it has been tested, it shows that PID-MRAC controller might be a successfully completion in the hydraulic crane control system. However, the nonlinearities of hydraulic position control make it quite difficult to obtain the high-precision in tracking the output with only using linear PID controllers.

The performance of the tracking the controllers was conducted by differing the phase difference and transient response which has been calculated. Based on the result, the hydraulic crane system performance was improved due to the designed controller achieve to tracked the reference input signal in the simulation and produce a vigorous transient response and minimum phase lagging.

5.2 Recommendation for future work

In order to improve the detection of the performance of the position control of the electro-hydraulic system, some suggestions and improvements can be done in the future.

Firstly, modelling of the system must be complete with all parameters which in this research, some of the parameters is unknow so it has to neglected the missing parameters. Moreover, by having a complete mathematical modelling can help to increase the performance of the system.

Secondly, the structure of the system must be simple. There's a researcher that once have a complex structure of adaptive position control based on Radial Basis Function neural network and LQ controller was created and it has a problem of computation burden which make the real-time performance cannot be guaranteed.

Finally, to increase the performance of position control is by using several control approaches such as sliding mode control, improved PID, adaptive control, robust control and fuzzy control. Try to run the project by applying the model to all of this approaches and compare the result between them in terms transient response and phase lagging.

REFERENCES

Ayman A. Aly., (2012). Model Reference PID Control of an Electro-hydraulic Drive.

Ai X., Yongkun F., (2007). Application of a PID Controller using MRAC Techniques for Control of the DC Electromotor Drive.

Ayman A. Aly, and. Ohuchi, H ―Fuzzy Hybrid Control Using Indirect Inference Control For Positioning An Electro-Hydraulic Cylinder‖, Conference of Fluid Power System, Yamaguchi, JAPAN, 2001.

Ayman A. Aly, Aly S. Abo El-Lail, Kamel A. Shoush and Farhan A. Salem ―Intelligent PI Fuzzy Control of An Electro-Hydraulic Manipulator‖ I.J. Intelligent Systems and Applications, 7, 43-49, 2012.

Ayman A. Aly and A. Abo-Ismail, ―Intelligent Control of A Level Process By Employing Variable Strcure Theory‖, Proceeding of International Conference in The Mechanical Production Engineering, Warso, Poland, 1998.

A. Bonchis!, P. I. Corke, D. C. Rye and Q. P. Ha, ―Variable Structure Methods In Hydraulic Servo Systems Control‖, Automatica (37), pp 589-595, 2001.

Che-Pin C., and Mao-Hsiung C., (2018). Mathematical Simulations and Analyses of Proportional Electro-Hydraulic Brakes and Anti-Lock Braking Systems in Motorcycles.

Dechrit, M., Nitin A., (2009). Modeling, simulation and control of high speed nonlinear hydraulic servo system.

Dechrit Maneetham and Nitin Afzulpurkar (2009). Modelling Simulation and Control of High Speed Nonlinear Hydraulic Servo System.

Daohang Sha a, Vladimir B. Bajic b and Huayong Yang, ―New Model and Sliding Mode Control of hydraulic Elevator Velocity Tracking System‖, Simulation Practice and Theory Journal (9), pp 365–385, 2002.

G. Bartolini, A. Ferrara and E. Usai, ―Chattering Avoidance By Second-Order Sliding Mode Control‖, IEEE Trans. Autom. Control 43 (2), pp 241–246, 1998.

H.E. Merrit, ―Hydraulic control systems‖, John Wiley & Sons, 1976.

J. Watton. ―Fluid power systems: Modeling, simulation, analog and microcomputer control‖, Prentice-Hall International, London, UK, 1989

J.W. Dobchuk, ―Control of a Hydraulically Actuated Mechanism Using a Proportional Valve and a Linearizing Feedforward Controller‖, PhD, Department of Mechanical Engineering, University of Saskatchewan, Saskatoon, Canada, Aug. 2004.

Ning-Bo Cheng Li-Wen Guan (2014). Position Control of an Electro-hydraulic Servo System Based on Switching between Nonlinear and Linear Control.

Sepehri, N., Lawrence, P.D., Sassani, F., and Frenette, R., ―Resolved-Mode Teleportation Control of Heavy-Duty Hydraulic Machines‖, ASME Journal of Dynamics Systems, Measurement and Control, V116, N2, pp 232-240, 1994.

Wilson J. Rugh, ―Analytical Framework for Gain scheduling‖, American Control Conference, San Diego, California, USA, May 1990.

Ziegler, J. G., & Nichols, N. B., ―Optimum Settings for Automatic Controllers‖, Transactions of ASME, 64, pp 759-768, 1942.