PERFORMANCE COMPARISON ON PID AND MIT RULE APPROACH FOR ELECTRO-PNEUMATIC POSITION SYSTEM

NURUL IZZATI BINTI MEDON

UNIVERSITI TEKNIKAL MALAYSIA MELAKA

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NURUL IZZATI BINTI MEDON

This report is submitted
in fulfillment of the requirement for the degree of
Bachelor of Mechanical Engineering (Plant & Maintenance)

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DECLARATION

I declare that this project report entitled "Performance comparison of PID and MIT rule approach for electro-pneumatic position system" is the result of my own work except as cited in the references.

Signature : ……………………………………..

Name : ……………………………………..

Date : ……………………………………..

APPROVAL

I hereby declare that I have read this project report and in my opinion this report is sufficient in terms of scope and quality for the award of the degree of Bachelor of Mechanical Engineering (Plant & Maintenance).

Signature	:	....
Supervisor's Name	:	....
Date	:	....

DEDICATION

This final year report is dedicated to families and loved ones for they are the ones who inspire me to finish this report. I would like to dedicate this report to my parents, Medon bin Ahmad and Zainab binti Kassim. This final year project would not have been possible without their love and continuous support**.**

ABSTRACT

Pneumatic system is one of fluid power system that commonly used in industry which generated by electrical source and normally called as electro-pneumatic system. This system convert the electrical energy to the mechanical energy in order to create motion. Normally, this system need a proper control system because of the high sensitivity of the system. There are various controllers that can be used in controlling the electro-pneumatic system and the performance is differ depends on the criteria of the performance required. In this project, the performance needed is to control the position of the electro-pneumatic system. The proper modelling of pneumatic system has been explained using derivation of mathematical modelling. All aspects in modelling the system such as the component of the system, the disturbances and sensitivity of the system as well as the nonlinearity of the system are considered. PID controller and MIT rule are the controllers that have been investigated in this project. The strategy in designing the controller is also has been discussed. PID controller used Ziegler-Nichols method as strategy to control the system while MIT rule used Model Reference Adaptive Control (MRAC) as the control strategy. From both controllers, the performance in position control of the system is analyzed. The performance characteristics used to evaluate the best controllers are the rise time, settling time, percentage overshoot, peak time and steady state based on the system response. The response is obtained from the simulation of the plant system in MATLAB Simulink software. The comparison of both controllers has been analyzed in this project.

ABSTRAK

Sistem pneumatik adalah salah satu sistem kuasa bendalir yang biasa digunakan dalam industri yang dijana oleh sumber elektrik dan biasanya dipanggil sebagai sistem elektro-pneumatik. Sistem ini menukar tenaga elektrik kepada tenaga mekanikal untuk mewujudkan pergerakan. Biasanya, sistem ini memerlukan sistem kawalan yang sewajarnya kerana mempunyai sensitiviti yang tinggi. Terdapat pelbagai pengawal yang boleh digunakan dalam mengawal sistem elektro-pneumatik dan prestasi yang didapati adalah berbeza bergantung kepada kriteria prestasi yang dikehendaki. Dalam projek ini, prestasi yang diperlukan adalah untuk mengawal kedudukan sistem elektro-pneumatik. Pemodelan sistem pneumatik telah dijelaskan dengan menggunakan terbitan pemodelan matematik. Semua aspek dalam pemodelan sistem seperti komponen sistem, gangguan dan sensitiviti sistem serta ketaklelurusan sistem telah dipertimbangkan. Pengawal PID dan peraturan MIT adalah pengawal-pengawal yang telah dikaji dalam projek ini. Strategi dalam merekabentuk pengawal juga telah dibincangkan. Pengawal PID menggunakan kaedah Ziegler-Nichols sebagai strategi untuk mengawal sistem manakala peraturan MIT menggunakan Rujukan Kawalan Penyesuaian Model (MRAC) sebagai strategi kawalan. Dari kedua-dua pengawal, prestasi dalam pengawalan kedudukan sistem dianalisa. Ciri-ciri prestasi yang digunakan untuk menilai pengawal yang terbaik adalah masa naik, masa penetapan, peratusan terlajak, masa puncak dan keadaan mantap berdasarkan tindak balas sistem. Tindak balas yang diperolehi daripada simulasi sistem dalam perisian MATLAB Simulink. Perbandingan kedua-dua pengawal telah dianalisa dalam projek ini.

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LIST OF ABBREVIATIONS

PMA	Pneumatic Muscle Actuator
LQR	Linear Quadratic Regulator
IPA	Intelligent Pneumatic Actuator
P	Proportional
PI	Proportional Integral
PID	Proportional Integral Derivative
DMRAC	Direct Model Reference Adaptive Controller
MIT	Massachusetts Institute of Technology
IAE	Integral Absolute Error
MRAC	Model Reference Adaptive Control
MATLAB	Mathematical Laboratory
DCV	Directional Control Valve
Z-N	Ziegler-Nichols

LIST OF SYMBOL

m_S	Mass Spool
c_S	Viscous Friction Coefficient
k_S	Spring constant
k_{fc}	Coil Coefficient
x_S	Spool displacement
i_c	Coil Current
C_f	Non-dimensional coefficient
R	Gas Constant
T	Temperature Constant
V_1	Volume At First Chamber
V_2	Volume At Second Chamber
V_{pc}	Total Volume Piston Chamber
P_1	Pressure At First Chamber
P_2	Pressure At Second Chamber
P	Pressure Drop
A_1	Area Piston At First Chamber
A_2	Area Piston At Second Chamber
M_T	Total Mass Piston And Load

N	Newton
A_P	Cross sectional area of piston
$\dot{x}_s$	First derivative of displacement spool
$\ddot{x}_s$	Second derivative of displacement spool
x_P	Piston displacement
$\dot{x}_P$	First derivative of piston displacement
$\ddot{x}_P$	Second derivative of piston displacement
$\dddot{x}_P$	Third derivative of piston displacement
K_P	Proportional Gain
K_i	Integral Gain
K_d	Derivative Gain
K_u	Ultimate Gain
T_u	Ultimate Period
T_r	Rise Time
T_s	Settling Time
T_p	Peak Time
m_p	Peak Amplitude
ω_n	Natural Frequency
ζ	Damping Ratio
y_m	Reference Model Output
y	Actual Plant Output
γ	Adaptation Gain

CHAPTER 1

INTRODUCTION

1.1 Background

From last few decades, a number of applications have been introduced to mankind in order to improve the productivity in industry comprehensively where pneumatic is one of them. Pneumatic system, which generates from the principle of fluid power is a system that uses compressed air in order to contribute work to the power transmission (Gill, Kumar and Kumar, 2015). Pneumatic comes from Greek word “Pneuma” which means “Breath”. This is due to the similar process of breathing and pneumatic wherein breathing, the air entered the body and released back to the surrounding while in pneumatic, the air from atmosphere is compressed in compressor and moves the specific parts or equipment where the equipment is considered as part of the machine and runs the whole system of the machine (Barala, Tiwari and Kumar, 2014). Several applications of the pneumatic system include packaging, open and closing doors, metal forming and clamping (Gill, Kumar and Kumar, 2015).

Electro-pneumatic system, on the other hand, is a system that worked by using air pressure and controlled by an electrical circuit in order to create forces and enhanced the motion of the system. It combines electrical and pneumatic system in one unit. The system varies with the general pneumatic system which the common pneumatic system only consists of a pneumatic system with several units. The early production of electro-pneumatic control systems was widely used in the industry of process control such as packaging, assembly and also in production. Type of components used in electro-pneumatic control system includes a timer, relays, counter and digital logic while pneumatic control system uses logic valve, stepper, sequencer etc. The motion of electro-pneumatic control is faster compared to normal pneumatic control because of the usage of electricity.

Electro-pneumatic controllers have some advantages over pneumatic controller systems which are:

- Fewer costs of electrical equipment than pneumatic equipment
- The system is controlled using electronic programmers and process computers.
- Control signal is reduced to control significant loads
- High reliability (Elsatar, 2010)

There are various applications of the electro-pneumatic system in the industry that can be divided into several types which are temperature control, level gauge, transportation, filling and packaging (Elsatar, 2010). In electro-pneumatic position system, the main focus is to control the movement of the cylinder. In order to control the movement, a controller is needed. PID controllers, defined as the Proportional Integral Derivative controller is used widely in the industry of process to control the desired position in the plant. It is known as the most desired and simple method and more popular rather than other controlling methods (Bansal, Patra and Bhuria, 2012). PID control is often used to build automation systems that are complicated such as energy production, manufacturing, and transportation where it is made from the combination of sequential functions, selectors, logics and simple function blocks (Astrom, 2002).

Meanwhile, MIT rule was started in 1960 by the researchers from Massachusetts Institute of Technology (MIT) where it can be used to design a controller using Model

Reference Adaptive Control (MRAC) from various systems including pneumatic systems. The controller design is sensitive to the changes in amplitude at reference input but it can provide a competent result. The use of MRAC is to design the controller so that the parameter that is being controlled can be adjusted in order to track the reference model from the actual plant that has the same reference input (Jain and Nigam, 2013).

1.2 Problem Statement

The modelling and designing controller plays a very important role in order to improve the dynamic as well as the static behaviour of the pneumatic system. For decades, numerous studies have been carried out regarding modelling and controller design for pneumatic systems. However, fewer studies have been carried out based on PID and MIT rule. This study looks into designing a controller based on PID and MIT rule approach for electro-pneumatic position system. The system, which has difficulties to control and always in inaccurate position is because of its non-linearity behaviour whereas the non-linearity occur due to the high frictions and the compressibility of air. Hence, restricting the use of the system in a high-performance control system (Roslan, 2015)

To design the required plant system with the implementation of the controller, mathematical modelling, control method and a software such as MATLAB is needed to analyse the performance of the system through simulation by considering two major criteria; a system with an application of controller and a system without an application of controller. The modelling, controlling and simulation techniques need to be developed so that the operating cost and energy consumption in a specific system could be reduced. By considering a good design of the controller, the performance of pneumatic system could be increased in terms of the positioning control.

1.3 Objective

The objectives of this project are as follows:

1. To design a control system for electro-pneumatic position system
2. To investigate the performance of the electro-pneumatic plant system without applying controller into the system.
3. To compare the performance of PID controller system against MIT rule approach in Model Reference Adaptive Control (MRAC) system.

1.4 Scope of Project

The scopes of this project are:

1. This project will only focus on the position or the movement of the cylinder in the system
2. The mathematical model of the system will be changed from nonlinear to linear using derivation method.
3. The simulation of PID controller and MIT rule are using MATLAB Simulink Software.
4. MRAC method will be used to distinguish the differences between actual plant and reference plant based on MIT rule approach that used to minimise the error function and tracking the perfect time between reference plant and actual plant output.

CHAPTER 2

LITERATURE REVIEW

2.1 Introduction

The uncertainties of pneumatic systems including their highly nonlinear characteristics make the systems difficult in achieving high performance. The nonlinearity occurs due to the flow of air and the friction force between the piston and the cylinder. Before exploring deeper to the main objectives of the paper, some background studies are presented in order to get a better understanding of the system and also the controllers. The review started with the attributes of the pneumatic system, followed by several applications of the pneumatic system, then focused on how the electro-pneumatic modelling system occur and finally the design of the controller that was implemented in the pneumatic system.

2.2 Attributes of Pneumatic System

There are many attributes of pneumatic actuator rather than hydraulic, that makes pneumatic actuator is attractive to be used in a difficult situation or in a certain environment (Ali, Mohd Noor, Bashi and M H Marhaban, 2009). Instead of using water as a source of energy in the hydraulic system, the source of energy used in the pneumatic actuator is based on air. Since air can be obtained freely, always available and does not require an external source to run the system, this lead pneumatic actuator to the front leaving the hydraulic actuator behind with conventional actuator that uses magnet, water and require external source in order to run the whole system (M.F. Rahmat *et al.*, 2011).

Electro-pneumatic systems are often chosen in automation industry due to various advantages which are low cost and clean operation as well as easy to handle based on electrical control and power translation in the system (Shih and Tseng, 1995). Pneumatic systems can be used in many applications since the system could provide a high dynamic response in order to position load and to extend the force needed for moving the load (Ali, Mohd Noor, Bashi and M H Marhaban, 2009) where the systems are also good in performance. The higher power-to-weight ratio by pneumatic actuator makes the air density is lesser than water density, thus, the weight-to-power ratio is also decreasing and the effectiveness of the system can be increased.

However, most of the pneumatic systems are in nonlinear form which most probably because of the compressibility of air inside the system, the characteristics of valve fluid flow itself and the Coulomb friction that higher than normal that makes the position control of pneumatic actuators are hard to accurate (M F Rahmat *et al.*, 2011).

2.3 Application of Pneumatic System

There are many applications of pneumatic system in the industry. Several researchers have proposed various applications regarding the pneumatic system due to its' availability in the market and the system does not require a high cost.

(Brubaker, 2015) presented a modern pneumatic braking control that combines the pneumatic valves and electronics valves in a single system. The system which consists of electronic valves improved the current pneumatic braking system with providing a better stability of the system and decreased the stopping distance and wheel slip when accelerating. The modern system was considered because of the existing models that did not explain clearly on how the effect of the dynamic response of the device related to the internal components. The modern system of foot brake valve was designed by considering several aspects, such as the difference in operating conditions and how the nominal systems react with nominal units.

A highly nonlinear Pneumatic Muscle Actuator (PMA) was proposed by (Zhao, Zhong and Fan, 2015) for position control by developing a phenomenological model to understand the physical behaviour of PMA based on Duhem Model. Duhem Model provides an analytical description of a smooth hysteresis behaviour where it describes two major components inside PMA which are a linear component and hysteresis component force. PMA experienced difficulty in controlling position because it consists of several physical properties that made the system become nonlinear such as force, pressure and displacement.

2.4 Electro-Pneumatic Modelling System

In designing an electro-pneumatic system for position control, modelling is the most important parts that need to be considered. Since most of pneumatic systems are in nonlinear form, the result obtained are sometimes unnatural and hard to achieve accuracy and produced an error. Hence, linearization is needed to be done in order to get the most accurate position control without error. Based on the previous studies made by researchers, there are several methods used in modelling which is system identification and derivation. System identification was used when such parameter could not be measured or identified and needed to estimated experimentally while derivation is from proposed equation that needs to analyse in order to get transfer function of the whole system.

(Richer and Hurmuzulu, 2001) developed a high-performance pneumatic force actuator system that consists of dual acting pneumatic actuators and four-way proportional spool valves to control the actuator. The aim was to gain an accurate model of the pneumatic system using proportional spool valve while considering several nonlinear characteristics such as flow effect through the valve, stroke inactive volume, the compressibility of air in cylinder chamber along with the leakage between chambers, time delay and attenuation in pneumatic lines. The author used system identification method in order to identify the valve discharge coefficient, piston friction force and valve spool viscous friction coefficient since these parameters could not identify easily and need to undergo the experimental process. The main equations of the system were limited to valve dynamic, chamber pressure and the piston load. There were some limitations in the system where the system was limited to a simpler application that position only at two ends of the stroke. When designing the system, friction force was neglected because it only consist of small magnitude in the system and the frequency is close to the valve bandwidth. The result obtained when modelling showed that the system was accurate and simple, so it can be used for on-line control application.

(Kaasa and Takahashi, 2003) designed an electro-pneumatic clutch actuator system that functions in position clutch disk during a gear shift. The proportional valve was operated to control the actuator position so that the necessary pressure against the piston can be produced to balance the load force at front chamber while the back chamber was connected to atmosphere by restriction. The friction force existed during the system operation. A mathematical model of the strongly nonlinear system was developed by deriving motion dynamic, pressure dynamic and valve dynamic equations until transfer function is obtained. To linearize the nonlinear system, the full nonlinear model of the system was simplified by considering several parameters. The pressure dynamic of the back chamber was neglected, coupled the pressure and temperature dynamic and neglected the dry friction force in friction model. The model presented showed that the system was strong in nonlinearities and controller need to be applied to the system.

2.5 Pneumatic Controller Design

Research of position control for pneumatic has outstanding growth in the 1990s where many researchers have investigated and developed the controller to solve the difficulties when dealing with pneumatic systems. The most used controller was PID (Proportional-Integral-Derivative) controller. The main focus of this review is based on the PID controller and MIT rule approach. There are also some reports based on the two main controllers that have been combined or compared with other controllers in order to achieve the best position control of the pneumatic system.

(Chaohui and Chenggang, 2010) compared PID and LQR (Linear Quadratic Regulator) controller in a pneumatic actuator system with proportional valve. The PID controller which consist of parameter K_P, $K_{I\ \text{and}}$ K_D is a linear regulator that was designed for measuring the system performance. The method of Bilinear transformation and zero order was used to obtain the discrete time model to the continuous time model. Square wave signal acted as input to the system in order to test and design the three types of PID controllers. The result obtained from square wave signal and measured output showed that the rise time was 0.5 seconds and the set time was 0.75 seconds. The overshoot percentage was 4.2%. There was steady state error of 2.2% in the system. The sign wave signal was proposed to get the estimated model of the system and compared with square wave signal before. The result showed that when comparing with input, the output response was 55.4 degree. Additional load with different weight was added to the system, with the PI controller was chosen for positioning control. Based on the result obtained, both controllers can be used to control the position of pneumatic actuator experimentally. However, in simulation, the LQR Controller has better characteristics with higher stability in the system, compared to PID controller.

(Faudzi *et al.*, 2012) applied PI controller to a closed-loop control system of Intelligent Pneumatic Actuator (IPA) for position and force tracking control to validate the experimental and simulation test. The closed-loop was measured based on the different input signals which are step response, sine wave response and multi-step response. The closed-loop control system presented a better response when compared with open loop control system and provided a good steady state response. The similar result obtained from experimental and simulation, and this can be seen when the value of K_P and K_I was small, the overshoot response was low and the contraction movement was delayed in experimental. In the simulation, the value of K_P and K_I was set large and the result obtained was reversed from experimental. Hence, the result was both validated.

Model Reference Adaptive Control that uses MIT and Lyapunov rule were investigated by (Kochummen and Nasar, 2015) based on PID controller where the response then compared with conventional PID controller. The controller was designed to analyse the efficiency of thermodynamic in the tandem compound steam turbine. The MIT rule was functions to minimise error function to get a perfect tracing between the output of the actual plant and the reference model while Lyapunov rule consists of adjustment role that adapts proportional, integral and derivative parameter into the control law. The applied MRAC into the system showed that the settling time of MIT rule was 4 seconds, Lyapunov rule 4.5 seconds and conventional tuning was 40 seconds. The overshoot also lower than the conventional tuning. The precision improvement, fast response and reduced steady state error for the turbine speed was achieved by MRAC controller in this paper.

An adjustment of feed forward gain using MIT rule was applied by (Avinashe *et al.*, 2015) in Direct Model Reference Adaptive Controller (DMRAC) where the cylindrical task was considered as a nonlinear system. The cylindrical task was tuned for an adjustment gain of 0.6 at a setpoint of 30 cm. The performance is then compared with conventional PI controller based on the Integral Absolute Error (IAE), settling time and steady state error. The evaluation of the performance confirmed that MRAC gave better performance than conventional PI controller where the result is 1601.5 against 2645.9. The settling time was also higher for MRAC, hence it settles faster than PI controller.

CHAPTER 3

METHODOLOGY

3.1 Introduction

This chapter describes the methodology used in this project in order to achieve all the objectives that have been set up at early of the project. This chapter starts with the overall work that needs to be done for this project or also known as the flowchart of the project. Then, the derivation of several numbers of equations took place in this chapter. Several parameters that need to be considered when designing the electro-pneumatic system plant are also explained in this chapter. After the required transfer function has been obtained by the method of derivation, the system is then modelled in the MATLAB Simulink software to analyse the performance of the system which will be explained later in Chapter 4. This chapter also explains about the controllers that will be used in this paper which is PID controller and MIT controller as well as the method of tuning the controllers. The flowchart of the project is shown in Figure 3.1.

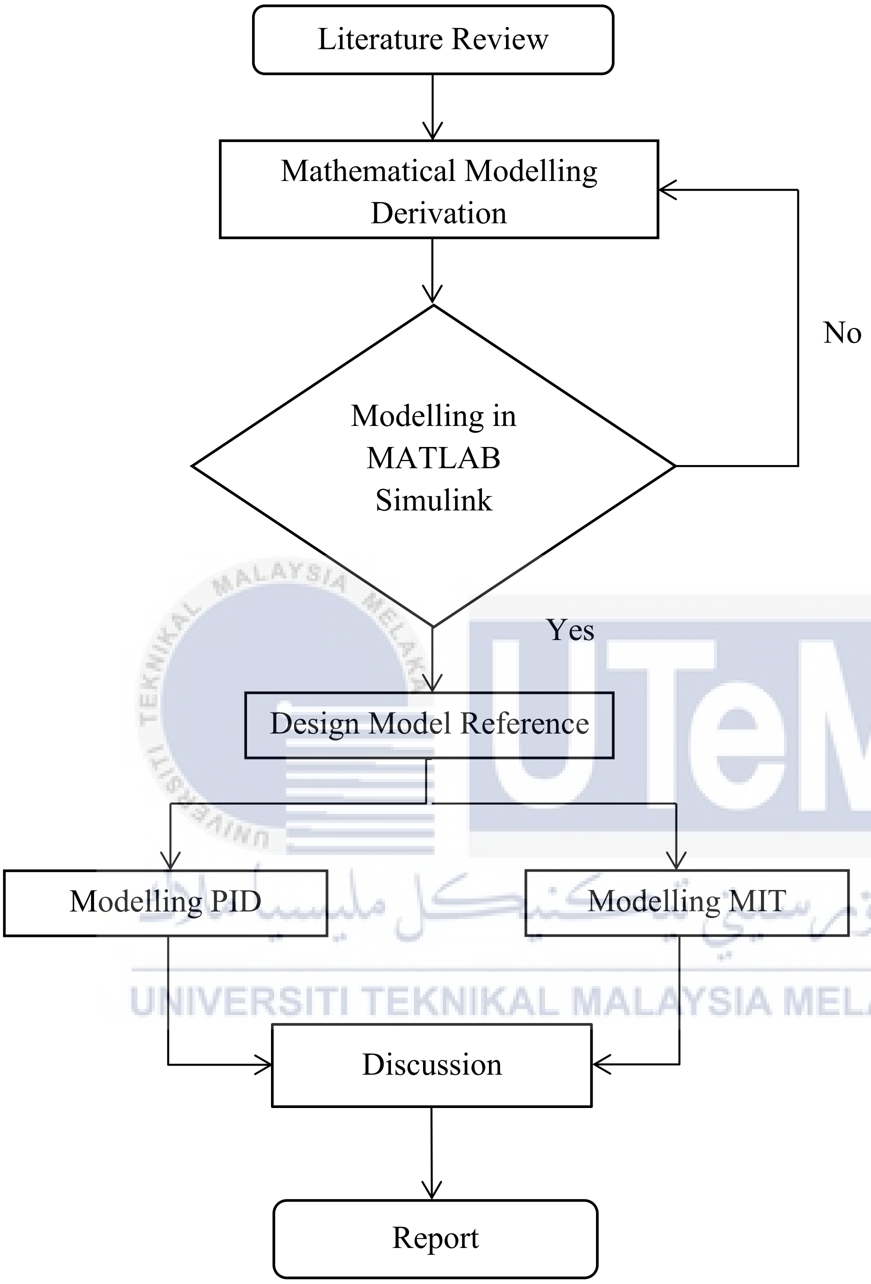


Figure 3.1: The flow chart of the methodology

3.2 System Modelling of Pneumatic Cylinder

A standard electro-pneumatic system consists of the pneumatic cylinder as an element of force, connecting tubes, valve as command device, current and sensors for pressure, force and position. The external load is the mass of external mechanical elements that connected to the piston along with the force produced by the interaction of the load with the environment. There are three main considerations in designing the pneumatic system which is load dynamic, volume, pressure and temperature of the air inside the cylinder and the mass flow rate through the valve. The chosen elements studied in this system are the double acting cylinder and directional 5/3 valve. The 5/3 valve means the valve has five ports; one port for supply pressure, two ports for ambient pressure and two ports for cylinder chamber, and three different modes of operation (Ilchmann, Sawodny and Trenn, 2005). The selected parameters in designing the system are the dynamic valve, flow at valve orifice and piston dynamic as well as pressure flow through the system.

The modelling of the pneumatic system is obtained from the theoretical mathematical analysis where the formation of the transfer function is based on the linearization of the nonlinear mathematical model obtained from the selected parameters. The mass flow rate is modelled through the changes of thermodynamic in a pneumatic cylinder with the application of Newton's second law of motion (Kothapalli and Hassan, 2008). Figure 3.2 shows the schematic diagram of a pneumatic system with the double acting cylinder, with the interests, are specified for every component that considers in the system.

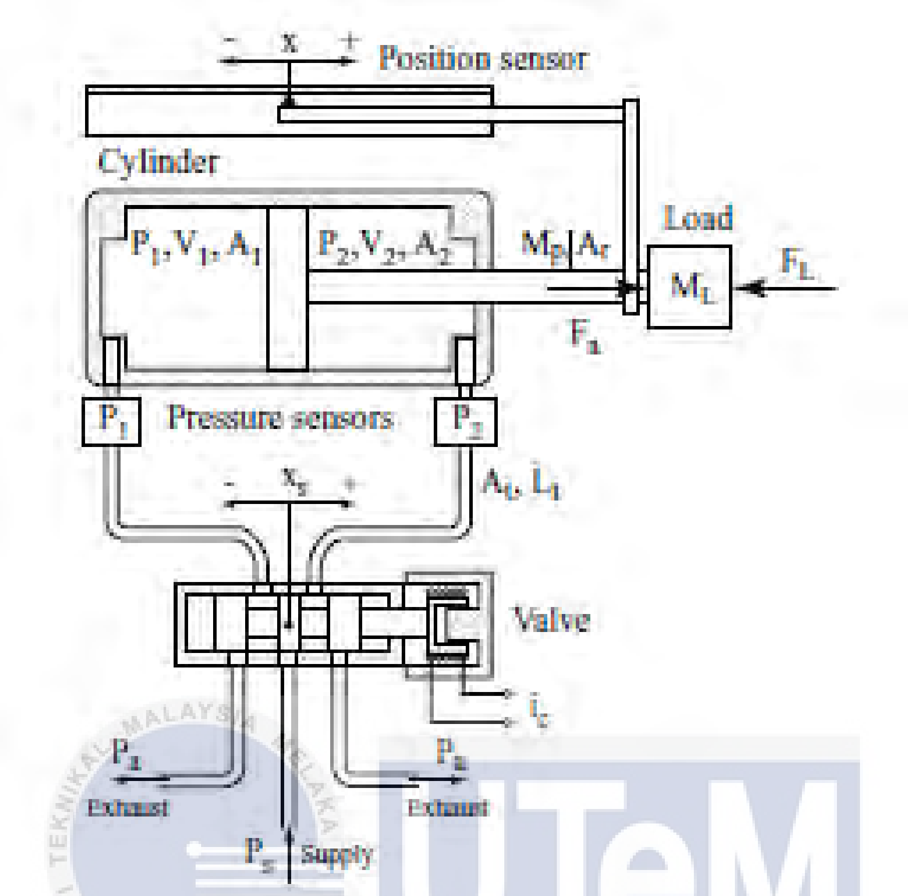


Figure 3.2: The schematic diagram of pneumatic system with double acting cylinder

(Richer and Hurmuzulu, 2001)

3.2.1 Dynamic Model of The Spool Valve

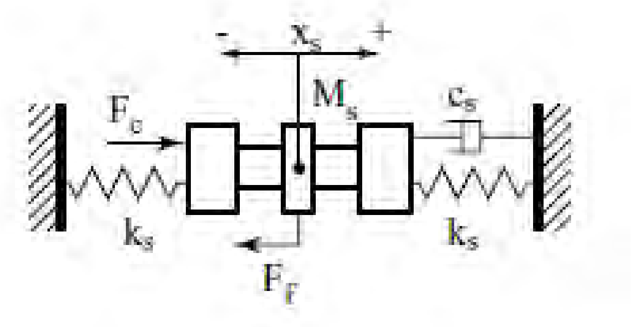


Figure 3.3: Dynamic equation of Spool valve

(Richer and Hurmuzulu, 2001)

Pneumatic valve is one of a critical element that consists of a pneumatic system where the airflows should be controlled fast and precisely. Figure 3.3 shows the dynamic equation of valve spool (M.F. Rahmat *et al.*, 2011). The modelling of this valve is based on two aspects which are the dynamic of the valve spool and the mass flow rate through the orifice of the valve (Ali, Mohd Noor, Bashi and Mohammad Hamiruce Marhaban, 2009). From the Figure 3.3 above, the equation of motion for the spool valve is expressed as (Richer and Hurmuzulu, 2001):

$$M_s\ddot{x}_s = -c_s\dot{x}_s - F_f + k_s(x_{so} - x_s) - k_s(x_{so} + x_s) + F_c \quad (3.1)$$

where M_s is the spool and coil assembly mass, x_s is the dispacement of the spool, c_s is coefficient of viscous friction, F_f is the friction force, k_s is the spring constant of spool, x_s is the compression of spring at equilibrium position and F_c is the force produced from the coil. The simpification of the spring force expressions produces :

$$M_s\ddot{x}_s + C_s\dot{x}_s + 2k_sx_s + F_f = F_c \quad (3.2)$$

The friction force F_f is neglected since in application of control, it is normal to apply dither signal to the coil with small magnitude and the frequency close to the valve bandwidth. The spool will vibrate slightly around equilibrium position and great reduction happen at Coulomb friction force. Neglecting F_f and using the force-current expression for the coil, equation (3.3) is generated :

$$M_s\ddot{x}_s + C_s\dot{x}_s + 2k_sx_s = K_{fc}i_c \quad (3.3)$$

where K_{fc} is coefficient of coil force and i_c is the coil current. To linearize the nonlinear equation of spool valve, laplace transform is applied to the equation (3.4). Laplace transform is one of the method that normally used to linearize nonlinear equations. The transfer function of the dynamic valve spool is presented after the linear equation is obtained from the laplace transform as in equation (3.5) :

$$M_s s^2 x_s + C_s s x_s + 2k_s x_s = K_{fc} I_s \tag{3.4}$$

$$J(s) = \frac{x_s}{I_s} = \frac{K_{fc}}{m_s s^2 + C_s s + 2k_s} \tag{3.5}$$

3.2.2 Spool Valve through Orifice Flow

Orifices are the fundamental of fluid power control where its flow characteristics play a significant role in designing pneumatic system. There are two flows exist which are laminar and turbulent which depends on inertia or viscous forces. To satisfy the continuity equation, the flow velocity needs to increase above in upstream state. The pressure drop across the orifice is due to the fluid particles acceleration from upstream velocity to the jet velocity at high Reynolds number while pressure drop at low Reynolds number occurs due to the internal shear force arise from fluid viscosity (Merrit, 1967).

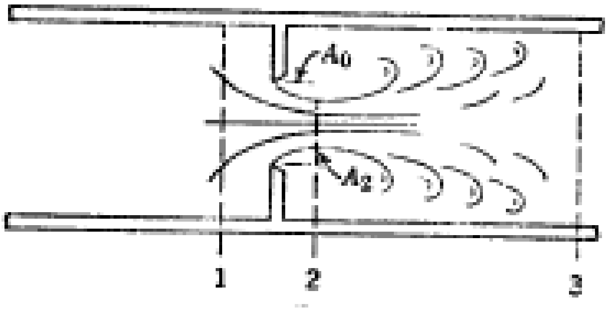


Figure 3.4: Turbulent flow through an orifice

(Merrit, 1967)

Considering Bernoulli equation for the flow:

$$P_1 + \frac{1}{2}\rho {V_1}^2 + \rho g h = P_2 + \frac{1}{2}\rho {V_2}^2 + \rho g h \tag{3.6}$$

where the height, h is negligible and V is substituted to u. By applying Bernoulli equation between the point 1 and 2, the pressure difference needed in order to accelerate fluid particle from upstream velocity u_1 to jet velocity u_2 can be found.

$${u_2}^2 - {u_1}^2 = \frac{2}{\rho}(P_1 - P_2) \tag{3.7}$$

Considering coefficient discharge at orifice area, $Q = Au$ and continuity equation for incompressible flow is $A_1 u_1 = A_2 u_2$, Hence, orifice equation that can be obtained are :

$$Q = C_d A_o \sqrt{\frac{2}{\rho}(P_1 - P_2)} \tag{3.8}$$

where C_d is discharge coefficient and A_o is orifice area.

The pressure drop is normally large across the valve orifice and the flow is usually turbulent and compressible. If the ratio of upstream to downstream pressure is larger than critical value P_{cr}, the flow is considered as choked flow and it depend linearly on upstream pressure while if the ratio is smaller, the mass flow will depends nonlinearly on both upstream and downstream pressure. The mass flow rate $(\dot{m}_v)$ through orifice area (A_o) is

$$\dot{m}_v = \begin{cases} C_f A_o C_1 \dfrac{P_u}{\sqrt{T}} & if\ \dfrac{P_d}{P_u} \leq P_{cr} \\ C_f A_o C_2 \dfrac{P_u}{\sqrt{T}} \left(\dfrac{P_d}{P_u}\right)^{1/k} \sqrt{1 - \dfrac{P_d}{P_u}^{(k-1)/k}} & if\ \dfrac{P_d}{P_u} > P_{cr} \end{cases} \tag{3.9}$$

where $\dot{m}_v$ is mass flow through valve orifice, C_f is nondimensional discharge coefficient, P_u is upstream pressure, P_d is downstream pressure, P_{cr} is critical pressure ratio, T is temperature and the constant of C_1 and C_2 is as follows :

$$C_1 = \sqrt{\frac{k}{R}\left(\frac{2}{k+1}\right)\frac{k+1}{k-1}} = 0.040418; \quad C_2 = \sqrt{\frac{2k}{R(k-1)}} = 0.156174;$$

$$P_{cr} = \left(\frac{2}{k+1}\right)^{\frac{k}{k-1}} = 0.528 \tag{3.10}$$

for constant k = 1.4 and R = 0.287. During charging, upstream is classified as the pressure in supply tank and downstream is the pressure in the cylinder chamber. For discharging, the upstream is the pressure chamber while downstream is the ambient pressure (Richer and Hurmuzlu, 2001). Assuming the flow for the mass flow rate of upstream pressure, $\dot{m}_1$ is choked flow and downstream pressure, $\dot{m}_2$ is unchoked flow, the mass flow rate equations used to obtain the mass flow rate of the whole orifice area are as in equation (3.11) and (3.12) :

$$\dot{m}_1(choked\ flow) = C_f A_{choked} C_2 \frac{P_1}{\sqrt{T}} \left(\frac{P_d}{P_u}\right)^{1/k} \sqrt{1 - \left(\frac{P_d}{P_u}\right)^{k-1/k}}$$

$$= 7.5851 \times 10^{-3}\, A_{choked} \tag{3.11}$$

$$\dot{m}_2(unchoked\ flow) = C_f A_{orifice} C_1 \frac{P}{\sqrt{T}}$$

$$= 1.0316\ \times 10^{-8}\, P \tag{3.12}$$

where $A_{orifice}$ = 6.2413 mm^2 (Zhu, 2006). The valve effective area for choked flow at Eq. (3.11) is represented as :

$$A_{choked} = \frac{\pi}{4} {x_s}^2 \tag{3.13}$$

Then, substitute Eq. (3.13) into Eq. (3.11) to get the mass flow rate of the valve orifice area at upstream pressure:

$$\dot{m}_1 = 0.0059573\, {x_s}^2 \tag{3.14}$$

The nonlinear equation of (3.14) is linearized by applying Taylor series expansion at operating point of valve spool displacement, $x_0 = 0$ into the equation with the second and higher order term is neglected along with valve leakage. The equation is as follows :

$$\dot{m}_1 = 0.0119\, x_s \tag{3.15}$$

Hence, the mass flow rate of the whole orifice area is expressed as:

$$\dot{m}_v = \dot{m}_1 - \dot{m}_2 \;\; = 0.0119\, x_s - \; 1.0316\; 10^{-8}\, P \tag{3.16}$$

3.2.3 Piston Dynamic in Cylinder

The derivation of load dynamics is from dynamics of piston load where it is based on the motion second law of Newton, $F = ma$. The equation of motion for piston-rod-load is represented as :

$$M_T \ddot{x}_p = (P)A_p - F_f + F_L \tag{3.17}$$

where M_T is the total mass of M_L(external load mass) and M_P (piston and rod assembly mass), x_p is the position of piston, F_f is the Coulomb friction force, F_L is the external force acting on the external load, $P = P_1$ - P_2 are the pressure drop in the chambers and A_p is the effective areas of piston, (Kothapalli and Hassan, 2008). However, the friction force and the external load force can be neglected. The new equation of motion can be expressed as :

$$M_T \ddot{x}_p = (P)A_p \tag{3.18}$$

Eq. (3.18) is arranged until P become an unknown, then it is substituted into Eq. (3.16) to obtain the mass flow rate at orifice area without the value of P as shown in equation (3.20):

$$P = \frac{M_T \ddot{x}_p}{A_p} \tag{3.19}$$

$$\dot{m}_v = 0.0119\, x_s - \, 1.0316\, 10^{-8} \left(\frac{M_T \ddot{x}_p}{A_p} \right) \tag{3.20}$$

3.2.4 Pressure in Chamber

The general model for gas volume is combined with three equations; energy equation, continuity and ideal gas law. The assumptions made are the gas is a perfect gas, the pressure and temperature are homogenous within the chamber and the negligence of kinetic and potential energy. The energy equation can be expressed as:

$$\dot{Q} - \dot{W} = \dot{U} \tag{3.21}$$

$$q_{in} - q_{out} + kC_v(\dot{m}_{in}T_{in} - \dot{m}_{out}T_{out}) - \dot{W} = \dot{U}) \tag{3.22}$$

where $\dot{Q}$ is the term of heat transfer, $(q_{in} - q_{out})$, $\dot{W}$ is workdone by the piston, $\dot{U}$ is the change of internal energy, C_v is specific heat at contant volume, T_{in} is the temperature of gas flow incoming and T_{out} is the temperature of the gas the flow is leaving. The equation is further simplifies. The process of the whole system is considered adiabatic, where there is no heat loss to surrounding, $q_{in} - q_{out} = 0$. The energy is also considered as isothermal, where T = constant. Hence, the change of internal energy, $\dot{U}$ as in equation (3.23) and the equation of $\dot{W}$ can be expressed as in equation (3.24):

$$\dot{U} = kC_v\dot{m}T = k\frac{d}{dt}C_v\dot{m}T \tag{3.23}$$

$$\dot{W} = \mathrm{P}\dot{V} + V\dot{P} = P\dot{V}_{pc} + V_{pc}\,\dot{P} \tag{3.24}$$

where $C_v = (\frac{R}{k-1})$. After simplification of the three general equations, the final equation obtained for the pressure in chamber is as in equation (3.25) and it is then arranged in form of mass flow rate, $\dot{m}_v$

$$\dot{P} = \frac{RT\dot{m}_v - 2A_pP\dot{x}_p}{V_{pc}} \tag{3.25}$$

$$\dot{m}_v = \frac{\dot{P}V_{pc} + 2A_pP\dot{x}_p}{RT} \tag{3.26}$$

where $V_{pc} = V_{c1} + V_{c2}$ which makes $\dot{V}_{pc} = 2A_p\dot{x}_p$. Then, in order to eliminate $\dot{P}$, differentiate both left and right hand side of Eq. (3.18) so that both equations will have $\dot{P}$

$$\frac{d}{dt}PA_p = \frac{d}{dt}m_T\ddot{x}_p \tag{3.27}$$

$$\dot{P}A_p = M_T\ddot{x}_p \tag{3.28}$$

and substitute Eq. (3.28) into Eq. (3.26) to get the final equation of pressure chamber in terms of mass flow rate:

$$\dot{m}_v = \frac{\left(\frac{M\ddot{x}_p}{A_p}\right)V_{pc} + 2A_pP\dot{x}_p}{RT} \tag{3.29}$$

In order to eliminate the equation of mass flow rate, Eq. (3.20) and Eq. (3.29) are combined and finally Laplace transform is performed into equation (3.30) to linearize the equation:

$$[0.0119A_pRT]x_s = V_{pc}M_T\,\dddot{x}_p + 1.0316x10^{-8}RTM_T\ddot{x}_p + 2{A_p}^2\Delta P\,\dot{x}_p \tag{3.30}$$

$$[0.0119A_pRT]x_s = [V_{pc}M_T\,s^3 + 1.0316x10^{-8}RTM_T\,s^2 + 2{A_p}^2\Delta P\,s]x_p \tag{3.31}$$

The final transfer function for the pressure chamber in the cylinder is illustrated as follows:

$$K(s) = \frac{x_p}{x_s} = \frac{0.0119A_pRT}{V_{pc}M_T\,s^3 + 1.0316x10^{-8}RTM_T\,s^2 + 2{A_p}^2\Delta P\,s} \tag{3.32}$$

The functions were calculated and valued based on the parameters in Table 3.1. Some parameters are constant while others are identified based on work of several authors.

Table 3.1: Values of parameters used in transfer function

(Šitum, Novaković and Petrić, no date; Zhu, 2006; Kothapalli and Hassan, 2008)

System Parameter	**Value**
Coil coefficient	$K_{fc} = 2.78\ N/A$
Non-dimensional coefficient	$C_f = 0.7$
Gas constant	R=287
Air temperature	T=293 K
Piston cross-section area	A_p=1.767.10^{-3} m^2
Total volume of piston chamber	$V_{pc} = 8.835x10^{-4}\ m^3$
Total mass of piston and load	M_T=2.33 kg
Mass spool	m_s = 1.5 kg
Viscous friction coefficient	c_s = 12 Ns/m
Spring constant	k_s = 108
Pressure drop	P = 1 atm

3.3 Controller Design

Controller design is about how to create a system that behaves in useful ways dynamically where it involves physical and non-physical systems such as fly jets, hydraulic or pneumatic actuators and also macroeconomics. The fundamental concept of the controller is the output was created based on a set of input variables that acts through a given plant which is the system.

3.3.1 PID Controller

Proportional Integral and Derivative (PID) control is a widely used controller in the industry where it is very useful in stabilising an unstable control system. The controller is based on feedback where the output of error is gained based on the error characteristics and perform a good result to the system. The three elements of PID; Proportional, Integral and Derivative offer the simplest and efficient solution to solve lots of control problems in the real world. PID controller is often used as a controller for hydraulic, pneumatic, mechanical controller and also can be used for manual tuning of the certain controller. To make it simple, P controller depending on the current error while I depending on the summation of past errors and lastly, D controller depend on future errors by considering the current rate of changes of errors (Bhagwan, Soni and Kumar, 2016).

The general equation of PID controller can be described as follows (Triantafyllou and Hover, 2013):

$$u(t) \ = k_p e(t) + k_i \int_0^t e(\tau) d\tau + k_d e'(t) \tag{3.33}$$

where k_p is proportional gain for proportional controller, k_i is integral gain integral controller, k_d is derivative gain for derivative controller, e is control error, t is time domain and τ is time characteristics. The general equation is then converted into laplace transform to get :

$$C(s) = \frac{U(s)}{E(s)} = k_p + \frac{k_i}{s} + k_d s = k_p\left[1 + \frac{1}{\tau_i s} + \tau_d s\right] \tag{3.34}$$

where τ_i is the time characteristics of integral part and τ_d is the time characteristics of derivative part. The PID controller is developed based on the closed loop in control system in Figure 3.5 and the categories of PID is describes as in Figure 3.6 below :

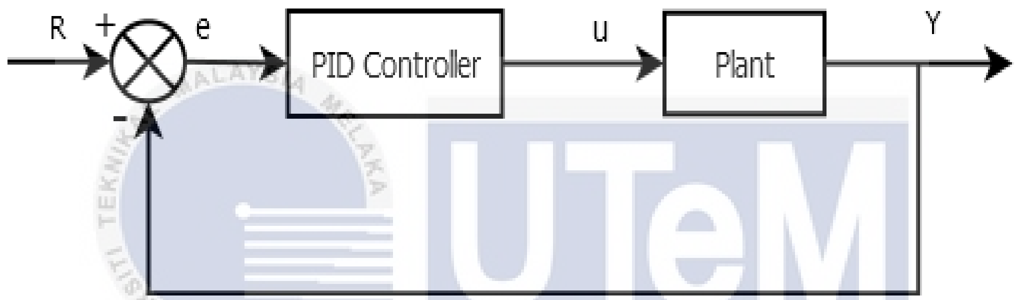


Figure 3.5: Design of PID controller in control system

(Bhagwan, Soni and Kumar, 2016)

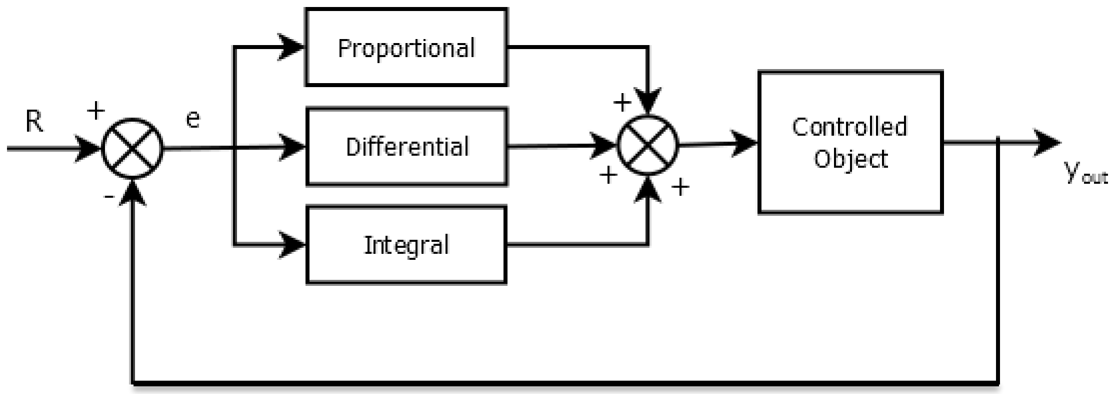


Figure 3.6: Categories of PID controller on the closed loop in control system

(Bhagwan, Soni and Kumar, 2016)

The three controller gains have their own roles when applied to the control system. The short description of the difference between P, PI and PID controllers are described as follows (Andrighetto, Valdiero and Vincensi, 2004):

3.3.1.1 Proportional Controller (P)

The Proportional controller (P) produces a control signal that linearly proportional to the error in the output that is measured, where the K_p gain needs to be small for the system to reach stability. Due to the low robustness of the controller, the system could experience instability if the gain is large and there are some disturbance or variation of parameters used in the system. The transfer function for the Proportional controller is expressed as in equation (3.35) as follows:

$$D(s) = K_p \tag{3.35}$$

3.3.1.2 Proportional Integral Controller (PI)

The Proportional Integral controller added signal fraction that proportional to the error integral where the integral parts reduce steady state error but, the system will produce more oscillation and increase the possibility to reach instability. The transfer function of the Proportional Integral controller is described as follows:

$$D(s) = K_p + \frac{K_i}{s} \tag{3.36}$$

3.3.1.3 Proportional Integral Derivative Controller (PID)

The Proportional Integral Derivative controller has a derivative part that increases damping in the system. The stable limit cycles are achievable due to the presence of integral. The small gains are applied to the system so that the system would be stable. However, the response is slow. The transfer function of the Proportional Integral Derivative controller is shown as:

$$D(s) = K_p + K_d s + \frac{K_i}{s} \tag{3.37}$$

3.3.2 Control Strategy of PID Controller

In order to control the system, every gain in PID controller needs to be tuned to get desired response. There are several tuning methods used for PID controller which are Manual Tuning, Ziegler-Nichols, Software tools and Cohen-coon. However, this paper will only focus on Ziegler-Nichols tuning method. The Ziegler-Nichols method was introduced in 1942 by John G. Ziegler and Nathaniel B. Nichols where the method is classified as step response and frequency response. The explanation on the response methods is as follows:

3.3.2.1 Step Response Method

The method of designing this response is based on open loop step response where it is categorised by two parameters that intersect between the tangent and coordinate axes that produced L as dead time and A as seen in the step response model in Figure 3.7 (Astrom and Hagglund, 1995):

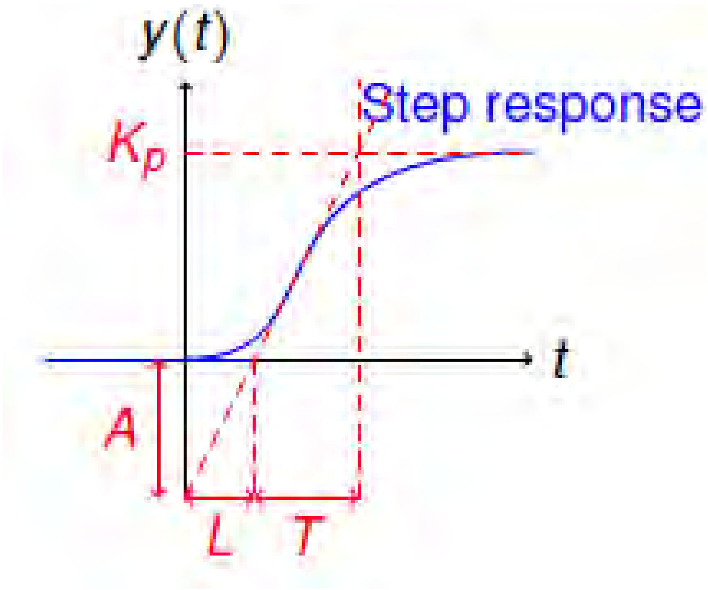


Figure 3.7: Step response model

(Astrom and Hagglund, 1995)

The parameters for PID controller based on step response tuning method is shown in Table 3.2 with the parameter of A and L obtained from Figure 3.7 above:

Table 3.2: PID Controller parameters obtained from step response method

(Astrom and Hagglund, 1995)

Controller	**K**	T_p	T_i	T_d
P	$1/a$	4 L	0	0
PI	$0.9/a$	5.7 L	3 L	0
PID	$1.2/a$	3.4 L	2 L	$L/2$

3.3.2.2 Frequency Response Method

The method of designing this response is based on the closed loop frequency response by Nyquist curve that intersects at negative real axis then produces two parameters which are $K_u$as ultimate gain and T_u as ultimate period. The parameters can be determined by setting the parameters $T_i = \infty$ and $T_d = 0$ after the controller is connected to the process to make the control action become proportional. The parameters of PID controller obtained from frequency response method can be expressed as in Table 3.3.

Table 3.3: Parameters of PID controller obtained from frequency response method (Astrom and Hagglund, 1995)

Controller	**K**	T_p	T_i	T_d
P	0.5 K_u	T_u	0	0
PI	0.4 K_u	1.4 T_u	0.8 T_u	0
PID	0.6 K_u	0.85 T_u	0.5 T_u	0.125 T_u

3.3.3 Control Strategy of MIT Rule

Model Reference Adaptive Control (MRAC) is a strategy used in designing an adaptive controller that relates to the controller parameters adjustment in order for the actual plant output to follow or behave like the reference model output with both consists of same input reference. The model output is compared to the actual plant output and the difference between both plants is used to adjust feedback controller parameters. There are three components used in designing MRAC method besides the system plant itself. The components are (Pankaj, 2011):

i. Adjustment Mechanism: A component used to alter the controller parameters for the tracking of the actual plant to the reference model by using mathematical approaches such as Lyapunov theory, MIT rule and augmented error theory in order to develop the adaptation mechanism.

ii. Controller: A number of adjustable parameters that has been parameterized. The adaptive controller design usually requires a linear parameterization to obtain adaptation mechanism that will stabilise the system and track the reference model successfully. The values for the control parameters are dependent on adaptation gain that will change the control algorithm of adaptation mechanism.

iii. Reference Model: Used to provide an idyllic response to reference input from the adaptive control system. The ideal behaviour that has been specified by reference model is to be achieved by the adaptive control system.

The block diagram of MRAC system is shown as in Figure 3.8. In the figure, y(t) is the actual plant output and $y_m(t)$ is reference model output. The difference between both output is presented by e(t).

$$e(t) = y(t) - y_m(t) \tag{3.38}$$

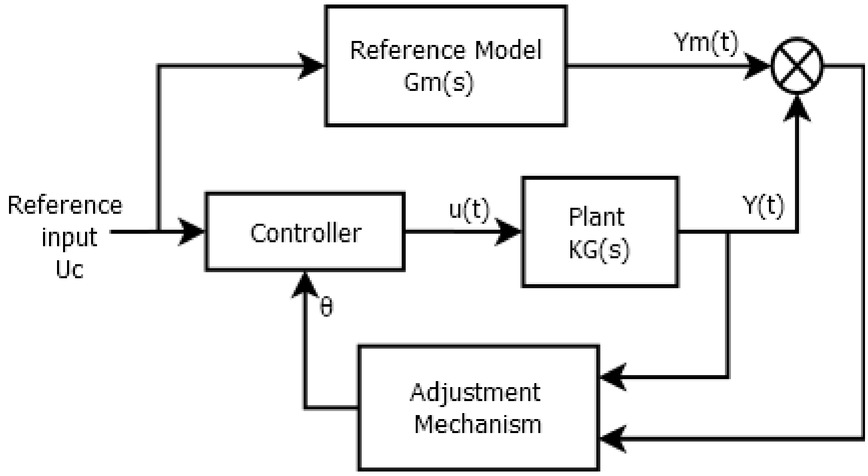


Figure 3.8: Block diagram of MRAC system

(Jain and Nigam, 2013)

In this MRAC system, MIT rule is related to the system by becoming one part of the components which is an adjustment mechanism that locates inside the block diagram. The adaptation gain, γ of the controller is used in adjusting the plant system output, $y(t)$ in order to follow the reference model output, $y_m(t)$.

3.3.4 MIT Rule

The rule of MIT was developed by Massachusetts Institute of Technology (MIT) in 1960. The controller design based on MIT rule can be produced with any system by using MRAC scheme including electro- pneumatic position system. The equations for this rule are described as follows (Jain and Nigam, 2013):

The cost function or loss function of this rule is expressed as:

$$J(\theta) = e^2/2 \tag{3.39}$$

where θ is adjustable parameter and e is the output error and indicate the differences between plant output model and reference model. To minimize the cost function to zero, parameter θ is adjusted where the change in parameter is put in negative gradient in direction of J :

$$\frac{d\theta}{dt} = -\gamma \frac{\partial J}{\partial \theta} \tag{3.40}$$

where γ is indicated as adaptation gain for the controller and also in a positive quantity. For equation (3.39), the cost function $J(\theta)$ will be reduced to zero because of the changes of parameter θ with respect to times. The combination of Eq. (3.39) with (3.40) produces

$$\frac{d\theta}{dt} = -\gamma e \frac{\partial e}{\partial \theta} \tag{3.41}$$

where the sensitivity derivative $\frac{\partial e}{\partial \theta}$ indicates the change of error which relates to the parameter θ.

CHAPTER 4

RESULTS AND DISCUSSIONS

4.1 Introduction

Based on the transfer functions which obtained from the derivation of a mathematical model in Chapter 3, the final transfer function is calculated and displayed in this chapter. This chapter explains in detail about the simulation and analysed the result obtained from the simulation based on open loop and closed loop in order to identify the basic performance of the system without a controller. This chapter also distinguished the differences obtained when designing the system using opened loop and closed loop. A baseline of second order system is created based on the values obtained from the closed loop response of the plant system. Based on the baseline that has been created, another second order system using root locus method is created where the method is carried out due to the need of specific criteria that PID controller and MIT rule need to follow when they are designed. The model reference also will be referred when comparing both PID controller and MIT rule later at the end of this paper as well as the behaviour of the controllers in terms of position control, thus achieving the third objective of this paper. Finally, discussions are made based on the results that have been acquired in this project.

4.2 Transfer Function for Electro-Pneumatic System

The transfer functions were split into two different processes which are valve and cylinder actuator. The transfer functions were expressed as:

i. The transfer function for valve:

$$J(s) = \frac{x_s}{I_s} = \frac{2.78}{1.5s^2 + 12s + 108} \tag{4.1}$$

ii. The transfer function for cylinder actuator

$$K(s) = \frac{x_p}{x_s} = \frac{1.7682}{2.059x10^{-3}\, s^3 + 2.021x10^{-3}\, s^2 + 0.624\, s} \tag{4.2}$$

Both transfer functions for valve and cylinder actuator are then combined to form the final transfer function that will be used in designing electro-pneumatic system plant in Simulink software. The final transfer function obtained is as follows:

$$L(s) = \frac{x_p}{I_s} = \frac{4.915596}{3.0885x10^{-3}\, s^5 + 2.7739x10^{-2}\, s^4 + 1.182624\, s^3 + 7.706268\, s^2 + 67.392\, s} \tag{4.3}$$

After gaining the transfer function of the plant, it then becomes the system that will be designed in form of open loop and closed loop system. The transfer function will then be used for designing PID controller and also to become the plant in MIT rule diagram that needs to follow the reference plant that will be created based on a closed loop system.

4.3 Simulation of Electro-Pneumatic System based on Open Loop and Closed Loop System

The transfer function of the electro-pneumatic system was simulated in Simulink in order to identify the system performance without the implementation of the controller into the system. The system was identified in two different control systems which are open loop and closed loop. An open loop system also called as the non-feedback system is a continuous system where the output does not affect the action of the input signal. On the other hand, a closed-loop control system or feedback system is a control system that uses the open loop as a forward path but contain one feedback loop between the output and its input. The output of the system is compared with the required condition and the error is converted to become control action which designed to reduce error and bring the system output to its desired response.

In this simulation, the input method that was used to identify the system is step input. The simulation started with analysing the open loop of the system. The system was first identified in the open loop to determine whether the system is stable or not. In general, if the system was not stable in the open loop, the simulation for the closed loop need to be done for determining whether the system might be stable when it is in closed loop behaviour. After the data for the open loop has been taken, the simulation for a closed loop was carried out. Then, both graphs response obtained from the simulation was analysed and compared. Figure 4.1 and 4.2 shows the block diagram to simulate the open loop of the electro-pneumatic system and graph of the system using step response.

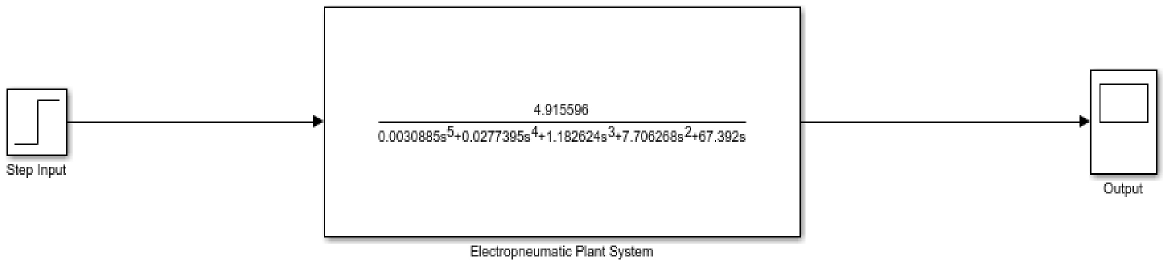


Figure 4.1: The block diagram for open loop system of electro-pneumatic plant

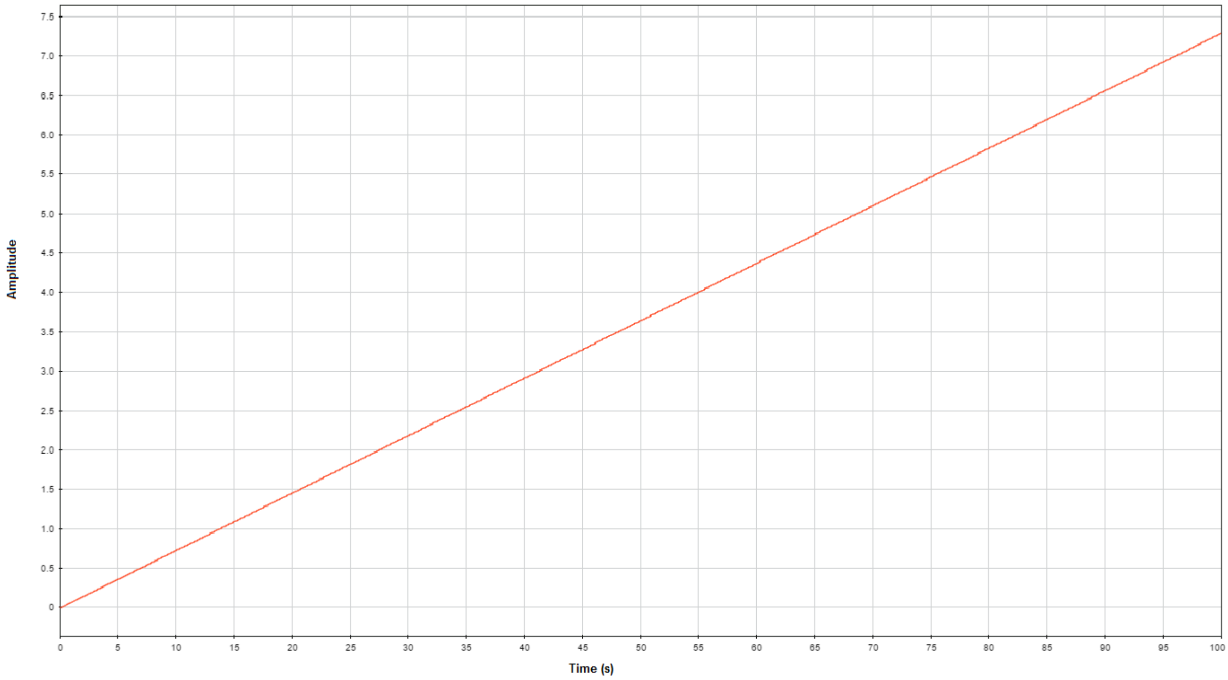


Figure 4.2: Graph of open loop control system

Based on the graph in Figure 4.2, the system started at 0 seconds and it is supposed to stop at amplitude one since the final value was set to one amplitude. However, the system did not stop since it keeps rising and it cannot be estimated when the system will stop due to the system reach infinity. When the system reaches 10 seconds, the amplitude was below than one while after reaching 20 seconds, the amplitude was already more than one. In 100 seconds simulation time, the amplitude of the system was seven. The behaviour of the electro-pneumatic system in open loop shows that the system was not stable. The system also moves slow right from the beginning where in the range of 0 to 100 seconds, the amplitude was still below ten.

Due to the non-stability and unknown output of the open loop system, the system was then simulated in the closed loop system. In closed loop control system, there are two main types of feedback control which are negative and positive feedback. The positive feedback was functioned to increase overall gain of the feedback system. However, too much gain in the system could increase the oscillation and the magnitude of the input signal, thus making the system become unstable. In contrast with the positive feedback, the negative feedback function to reduce the gain of the system thus provides stable responses and improves the stability of the system. In simulating the closed loop system of electro-pneumatic system, the system was designed using negative feedback system. Figure 4.3 and 4.4 shows the block diagram used to simulate the closed loop of electro-pneumatic system and graph of the system using step input response.

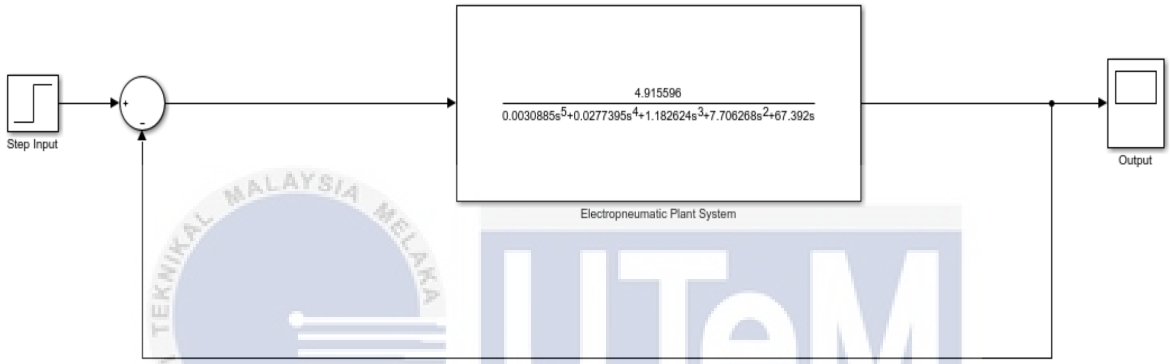


Figure 4.3: Block diagram for closed loop system of electro-pneumatic plant

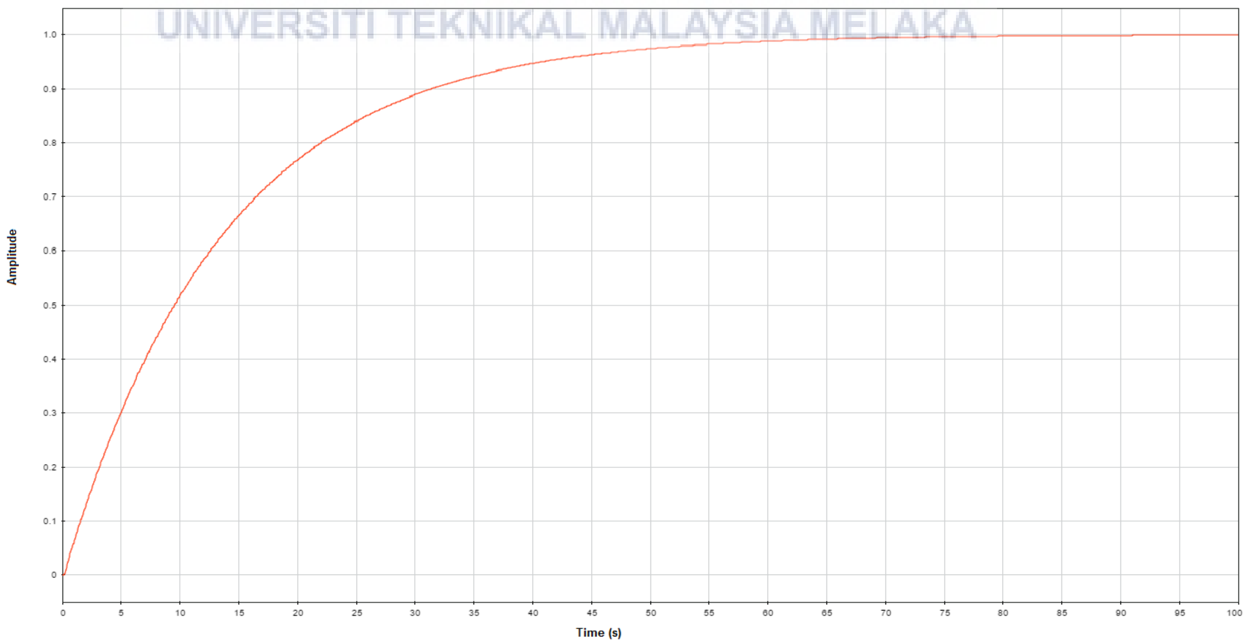


Figure 4.4: Graph of closed loop control system

Based on the graph in Figure 4.4, the closed loop system was set to start at zero amplitude and stop at amplitude one in step input. The results obtained was different from the open loop system where the closed loop system settles in the required amplitude which is one. However, the time taken for the system to settle at amplitude one was 53.3 seconds. It shows that the system was moving slowly before it reaches maximum amplitude which is for almost a minute. The system does not have a delay and no overshoot occur to the system. Hence, the system is stable. In other words, the system follows the path that it need to follow directly, only that it is in a slow motion. However, due to the movement of the cylinder was slow, there is need to apply controller to the system. Figure 4.5 and 4.6 shows the rise time, steady state and settling time of the electro-pneumatic system in closed loop control system. Meanwhile, Table 4.1 shows the difference between open loop system and closed loop system of the electro-pneumatic plant.

Table 4.1: Value differences between open loop and closed loop system

Open Loop	**Characteristics**	**Closed Loop**
Infinity	Peak Time (s)	>90
Infinity	Peak Amplitude	0.999
No	Steady State	1
No	Overshoot (%)	0
Infinity	Rise Time (s)	29.923
No	Settling Time (s)	53.3

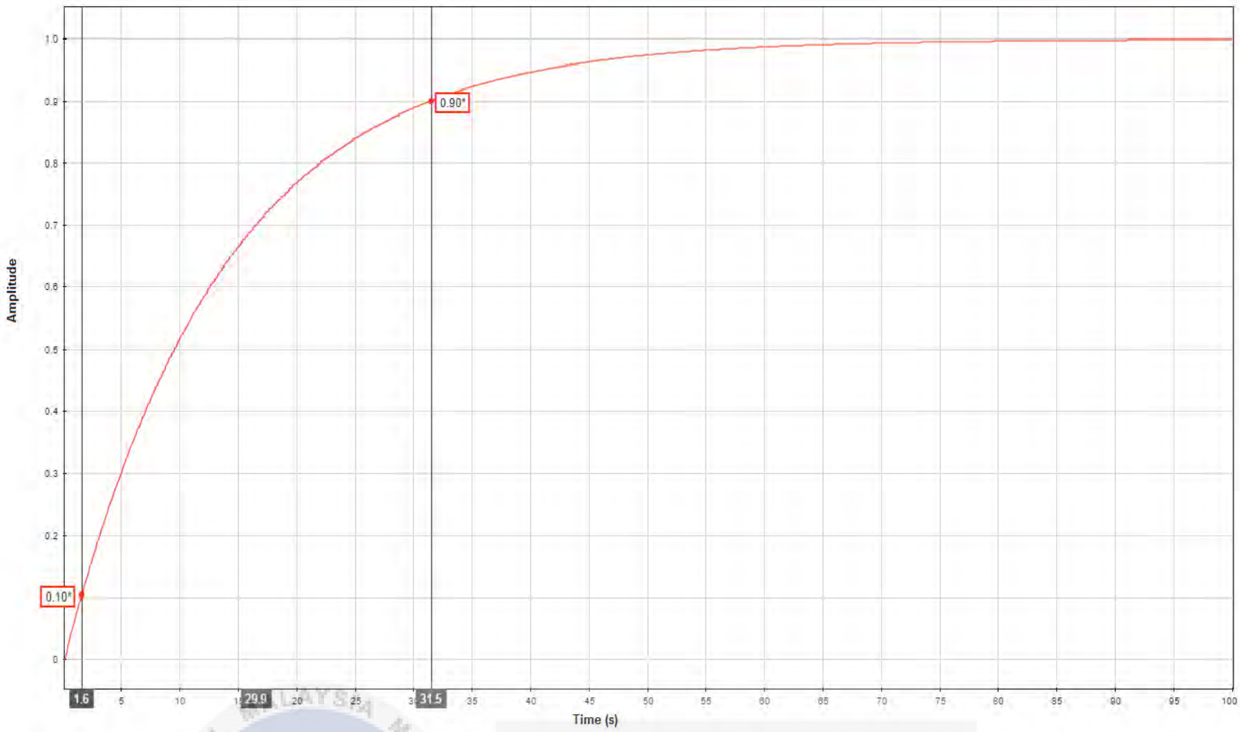


Figure 4.5: Rise time at 29.923 seconds

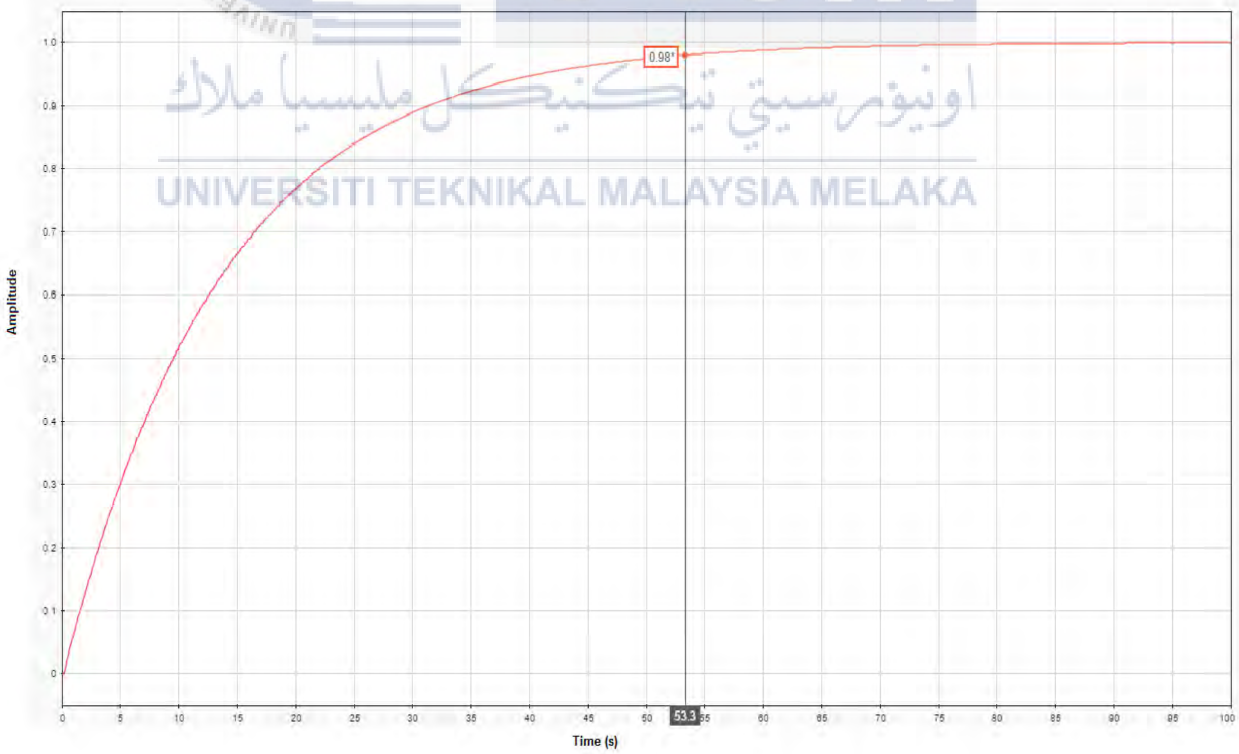


Figure 4.6: Steady-state at amplitude 1 and settling time at 53.3 seconds

In comparison, the response of the plant in closed loop control system is better than compared to the plant responding in open loop control system. Even though the open loop is cheaper and simple to implement due to the relationship of input and output was direct and not affected by external disturbances, the disturbance itself could affect the whole system and it is unnoticeable. On the other hand, closed loop control system makes the system able to respond to the external disturbances and internal variations of the system parameters. Moreover, it could determine the actual input required into the system by reducing the error and track the output into its desired response. Hence, using closed loop system is much easier to gain accurate control of the electro-pneumatic plant system rather than open loop system.

4.4 Reference Model for Pneumatic Plant System

A second order system displays a wide range of responses in control system designs. It can present characteristics such like first order system or display pure or damped oscillations In designing reference model for the plant system, the fifth order system of the electro-pneumatic plant need to be converted into second order system. In that way, the system will be easier to analyse since there are only two orders instead of five orders and it will become more accurate. The second order system is very important in control systems engineering due to many methods of designing control systems are based on second order systems. The general equation of second order system is given by

$$G(s) = \frac{{\omega_n}^2}{s^2 + 2\zeta\omega_n s + {\omega_n}^2} \tag{4.4}$$

where ω_n is the natural frequency of second order system and ζ is the damping ratio. Natural frequency ω_n determines how fast the system oscillates during transient response that leads to four cases of stable response which are undamped, underdamped, critically damped and overdamped responses. The damping ratio determines on how much the system oscillates as the response decays toward steady state (Aly and Salem, 2015).

To design the required reference plant based on the second order system, natural frequency ω_n and damping ratio ζ need to be found first based on the values obtained from the analysis of closed loop system electro-pneumatic plant. Due to the response of the fifth order system looks like critically damped, thus the system does not experience overshoot. There are five available formula that can be used to measure the natural frequency and damping ratio which are peak time, percentage overshoot, settling time, rise time and peak amplitude. The values obtained from the closed loop system shows that the required values needed for designing the system are only the settling time and rise time of the system. Percentage overshoot could not be used as the system does not have overshoot since overshoot only occurs in system that have underdamped response. Moreover, the formula to calculate settling time and rise time consist of both natural frequency and damping ratio which makes it is easier to calculate the exact values of natural frequency and damping ratio.

Settling time, T_s is the time required for the damped oscillations in the system to reach and stay in range of 2% of the steady state value, which is the final value that the system will reach. The formula to calculate settling time is as follows (H.Bishop and C.Dorf,2011):

$$T_s = \frac{4}{\zeta\omega_n} \tag{4.5}$$

By applying the value of T_s = 53.3 seconds into Eq. (4.5), the value of $\zeta\omega_n$ that can be obtained using the formula is:

$$\zeta\omega_n = \frac{4}{53.3} = 0.075 \tag{4.6}$$

Comparing $\zeta\omega_n = 0.075$ with second-order equation of $2\zeta\omega_n$ at Eq. (4.4) will get the final value of damping ratio in terms of natural frequency as in equation (4.9) :

$$2\zeta\omega_n = 2(0.075) = 0.15 \tag{4.7}$$

$$\omega_n = \frac{0.15}{2\zeta} \tag{4.8}$$

$$\zeta = \frac{0.075}{\omega_n} \tag{4.9}$$

On the other hand, rise time, T_r is the required time for the waveform of the graph to go from the value of 10% until 90% of the final value. The rise time determines the speed of the transient response in a system where it applies to more general response but become less useful when there is overshoot inside the system. Therefore, the formula required in calculating rise time is as follows (H.Bishop and C.Dorf,2011):

$$T_r = \frac{2.16\zeta + 0.6}{\omega_n} \tag{4.10}$$

The value of T_r = 29.923 seconds is applied into Eq. (4.10) and the value of damping ratio ζ obtained in Eq. (4.9) is substituted into Eq. (4.10) to become

$$29.923 = \frac{2.16\left(\frac{0.075}{\omega_n}\right) + 0.6}{\omega_n} \tag{4.11}$$

$$29.923\,\omega_n = \frac{0.162}{\omega_n} + 0.6$$

$$29.923\ \omega_n{}^2 = 0.162 + 0.6\ \omega_n$$

$$29.923\ \omega_n{}^2 - 0.6\ \omega_n - 0.162 = 0 \tag{4.12}$$

The final values of ω_n obtained from Eq. (4.12) are $\omega_n = 0.0843, -0.0642$, where the negative value is neglected and only positive value is taken into account. Then, the value of ω_n is substituted into Eq. (4.9) to get the final value of damping ratio. Both values of natural frequency $\omega_n = 0.0843$ and damping ratio $\zeta = 0.89$ are then substituted again into Eq. (4.4) of second order system until become the transfer function of following form

$$G(s) = \frac{0.0071065}{s^2 + 0.15s + 0.0071065} \tag{4.13}$$

The second order system at Eq. (4.14) is almost same with the fifth order system in the original plant, only that the system is in the form of second order. Figure 4.7 shows the graph of second order system that is obtained from the fifth order system of the electro-pneumatic plant.

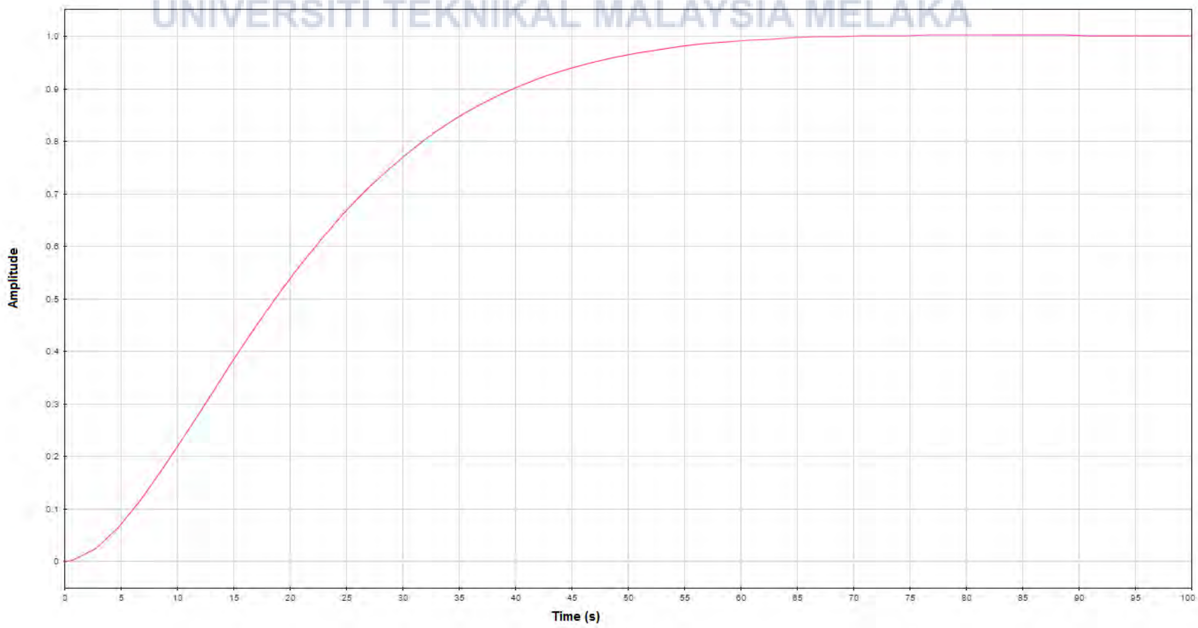
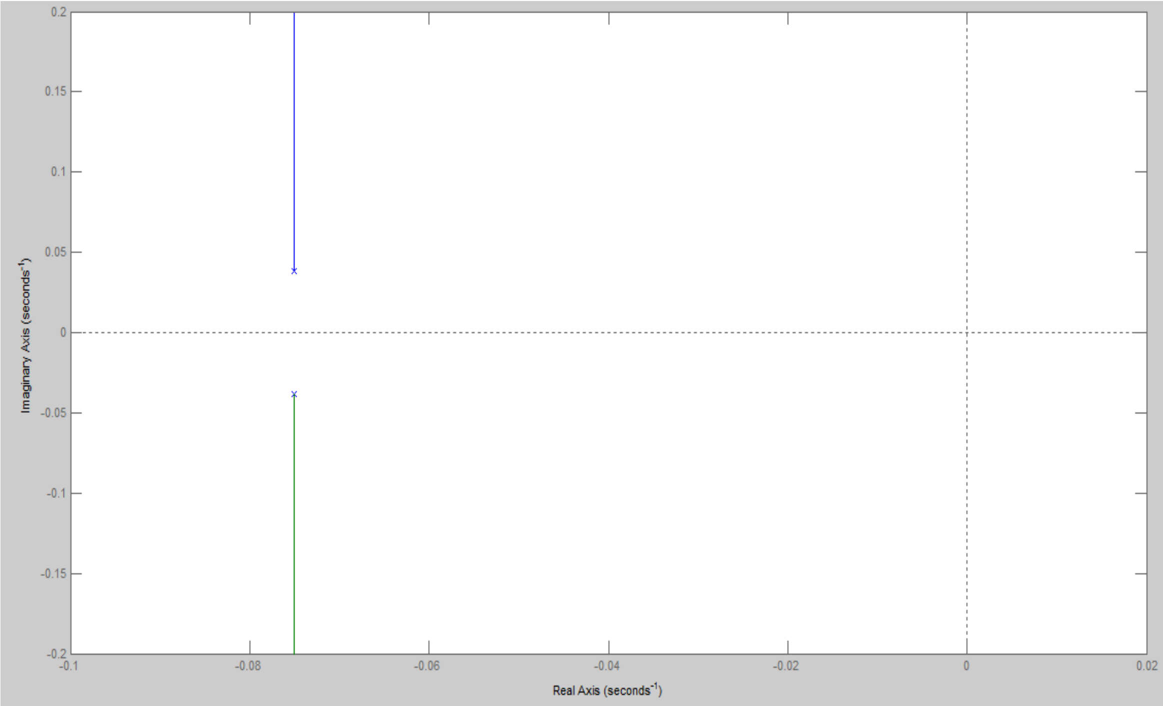


Figure 4.7: Graph of Second Order System converted from Fifth Order System

After obtaining the second order system, root locus technique is designed to observe how the closed-loop poles changes location as the gain is varied. It shows how the behaviour of the system when the controller is working in terms of the transient response and stability. There are many advantages of root locus such as the technique is easy to implement, the user can easily predict the performance of the system and it can provide better ways to indicate the parameters or characteristics needed. Figure 4.8 shows the root locus of the second order system of an electro-pneumatic plant designed with MATLAB software.

```
clear all
clc

syms s

K=1;

Gnum=(0.0071065);
Gden=sym2poly(1*s^2+0.15*s+0.0071065);

G=tf(Gnum,Gden);

H=1;

CL=feedback(G*K,H);

figure
rlocus(CL);

```


(a)



(b)

Figure 4.8: Root Locus of Second Order System (a) Root locus coding and (b) Root locus graph of the system

Based on Figure 4.7, the response shown is overdamped response since there are two poles placed at the negative side of the real axis when referred in root locus of Figure 4.8. So, the system is stable. The poles moved in vertical direction, which keeps the real part of the pole at the location. The frequency changed but the envelope remains unchanged. Since all curves fit under same exponential decay curve, the settling time is virtually same for all waveforms. Figure 4.9 shows the transient response of the overdamped system.

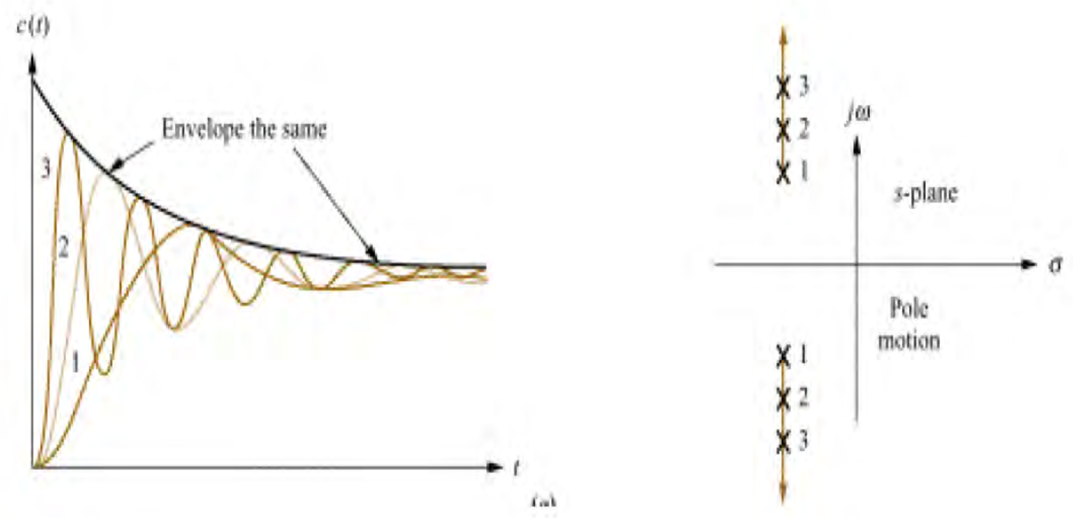


Figure 4.9: Transient response of overdamped system

(Polushin, 2003)

The aim is to get the response that can meet final position, which is at amplitude one but at the same time, the response needs to be faster since electro-pneumatic system normally has slower movements. The higher location of the pole could result in shorter rise time but the overshoot will increases. In other words, the system will be faster but with little overshoot. Hence, it is decided that suitable overshoot that will be accepted for the reference model is below than 15 percent. The location of the poles with 15 percent of overshoot is identified in the root locus system and the data obtained from the location is recorded as in Figure 4.10. The data obtained from the root locus are in form of pole location, gain, damping ratio, natural frequency and percentage overshoot.

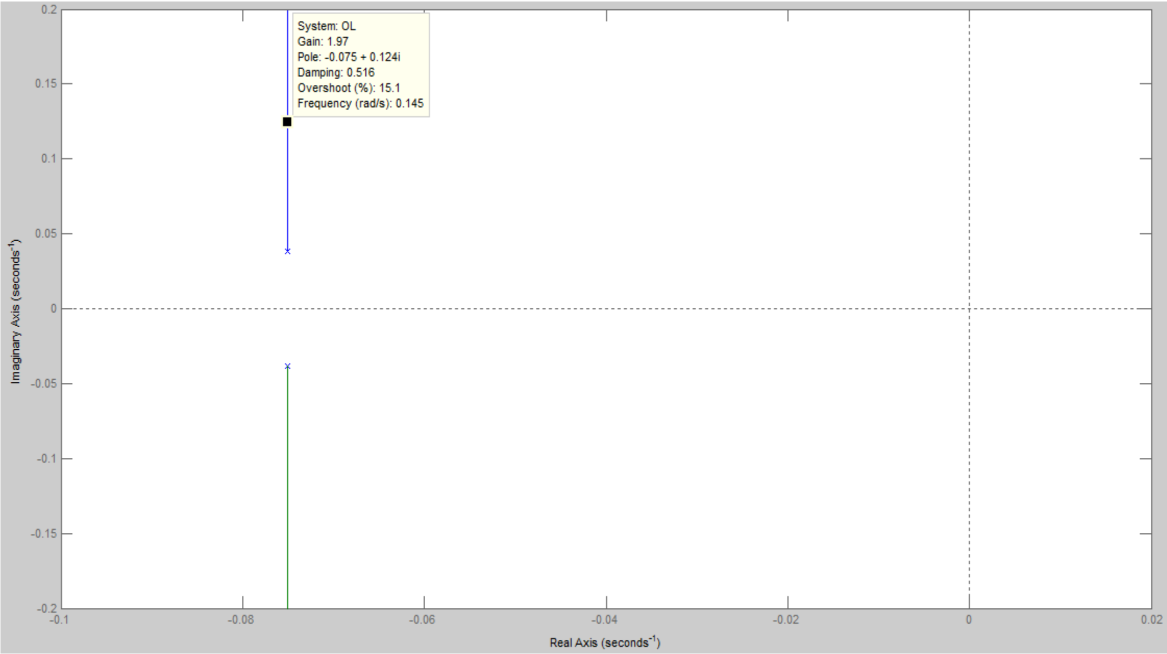


Figure 4.10: The pole placement with 15 percent overshoot

Based on Figure 4.10, the performance response characteristics for the reference model such as peak time and settling time can be calculated. This is because the two characteristics play an important part in determining desired response for the reference plant which is a faster response at specific overshoot. The value of damping ratio, ζ and natural frequency, ω_n obtained from the system are $\zeta = 0.516$ and $\omega_n = 0.145$ is used to determine the peak time and settling time of the required reference model using specific formula of second order system.

Settling time, T_s is the time taken for the system to reach certain distance or position needed. To calculate settling time, T_s of the system, the formula is as follows (Polushin, 2003) :

$$T_s = \frac{4}{\zeta\omega_n} = \frac{4}{\sigma_d} \tag{4.14}$$

where σ_d is considered as the real part of the pole. Substituting σ_d = 0.075 into Eq. (4.14) to get the value of settling time

$$T_s = \frac{4}{0.075} = 53.3\, seconds \tag{4.15}$$

Peak time, T_p is the required time for the system to reach the first peak. The formula to calculate peak time is (Polushin, 2003)

$$T_p = \frac{\pi}{\omega_n\sqrt{1-\zeta^2}} = \frac{\pi}{\omega_d} \tag{4.16}$$

where $\omega_d = \omega_n\sqrt{1-\zeta^2}$ is considered as the imaginary part of the pole. The value of $\omega_d = 0.124$ is substituted into Eq. (4.1) to obtain the peak time of the system to get

$$T_p = \frac{\pi}{0.124} = 25.3\, seconds \tag{4.17}$$

Finally, after getting all values needed, the second order system of the reference model is designed. Substitute the value of damping ratio, $\zeta = 0.516$ and natural frequency, $\omega_n = 0.145$ into Eq. (4.4) to obtain

$$H(s) = \frac{0.021025}{s^2 + 0.14964s + 0.021025} \tag{4.18}$$

The system will be used as a reference model that will be referred after designing PID and MIT rule. It will become baseline that differentiates between PID and MIT rule. Figure 4.11 shows the simulation response of the reference model system in the underdamped response of second order system and Table 4.2 shows the overall values for the reference model.

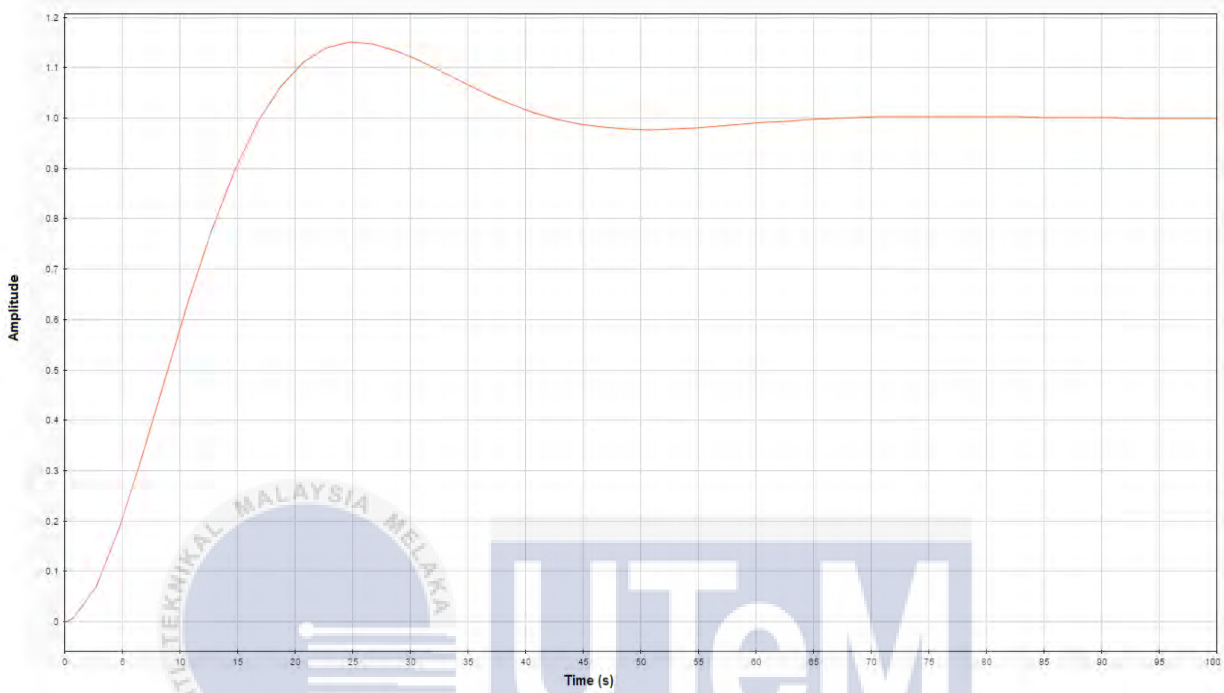


Figure 4.11: The reference model

Table 4.2: The overall values of reference model

Characteristics	**Value**
Damping ratio, ζ	0.516
Natural frequency, ω_n	0.145
Percentage overshoot, OS%	15%
Rise Time, T_r	11.5 seconds
Settling Time, T_s	53.3 seconds
Peak Time, T_p	25.3 seconds
Peak Amplitude, m_p	1.15

4.5 Simulation of Electro-Pneumatic System Using PID Controller

In this part, PID controller is applied to the system and the result obtained based on the controller is displayed and discussed. The system used for designing PID controller is based on the closed loop control system of the electro-pneumatic plant. It is designed using the Simulink software.

The tuning method used for Proportional Integral Derivative of PID controller is based on Ziegler-Nichols method of frequency response. The method was chosen due to the closed loop system that has no delay at its starting point. The block diagram for the PID controller design along with the electro-pneumatic system is shown as in Figure 4.12.

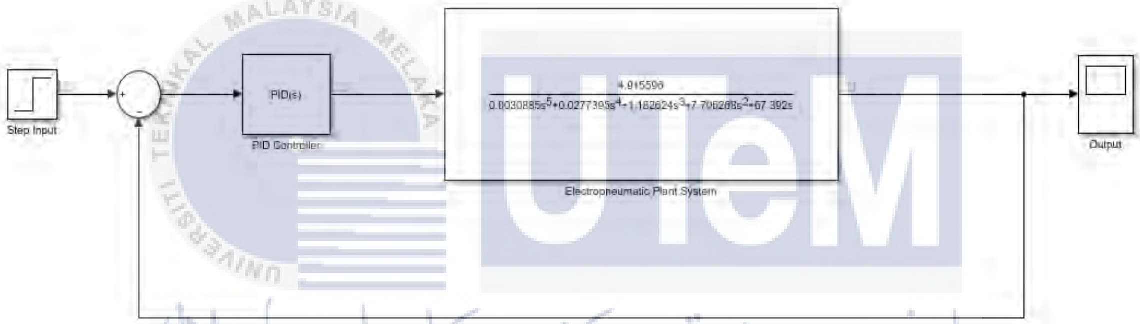


Figure 4.12: The block diagram of PID controller with electro-pneumatic system

The frequency response method of Ziegler-Nichols was developed by identifying ultimate gain, K_u and oscillation period, T_u. The first step was to set the K_i and K_d to zero. Then, the K_p gain is increased until the ultimate gain of K_u is reached where the output of the loop starts to oscillate. The value of T_u was obtained from the period of one full oscillation from the loop output. The value obtained for K_u is 75 and the value for T_u is 0.8. The method of tuning is shown as in Figure 4.13. Then, the value of K_u and T_u are replaced in the Ziegler-Nichols frequency table. The final values after the tuning of K_p , K_i

and K_d are shown in the Table 4.3 while the response obtained after the tuning of P, PI and PID controller are illustrated as in Figure 4.14 until Figure 4.19.

Table 4.3: Tuning of Ziegler-Nichols frequency response

	Kp	Ki	Kd
P	37.5	0	0
PI	33.75	0.6664	0
PID	45	0.4	0.1

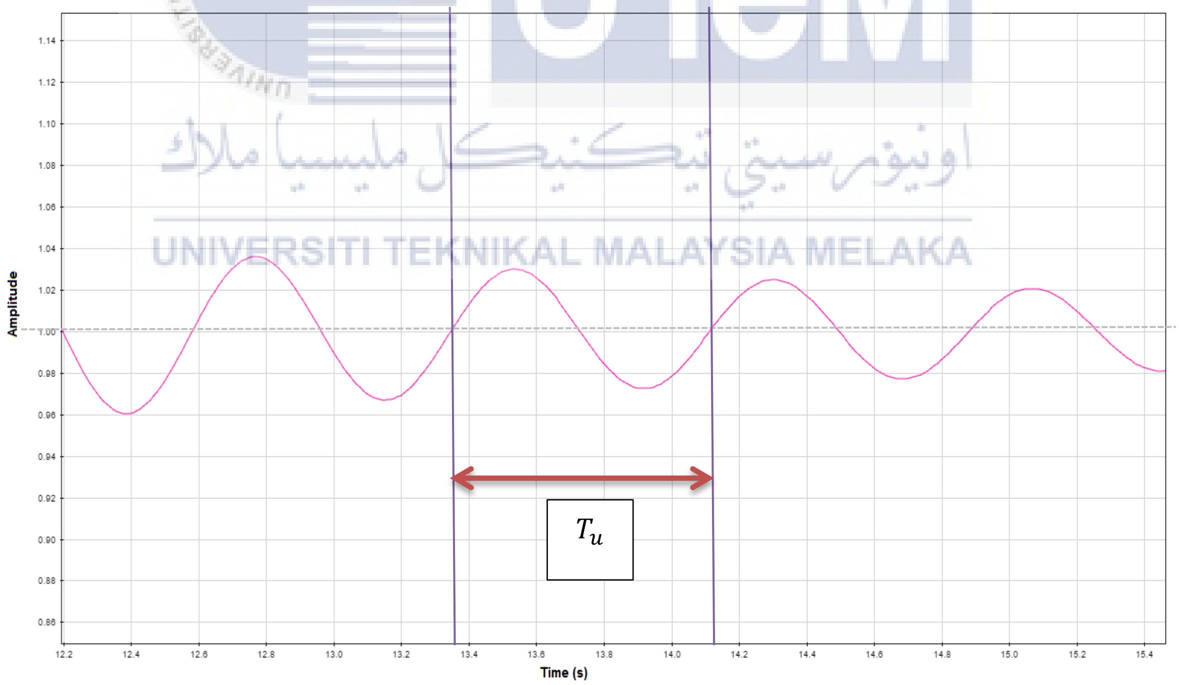


Figure 4.13: Tuning method of PID Controller using frequency response

P Controller

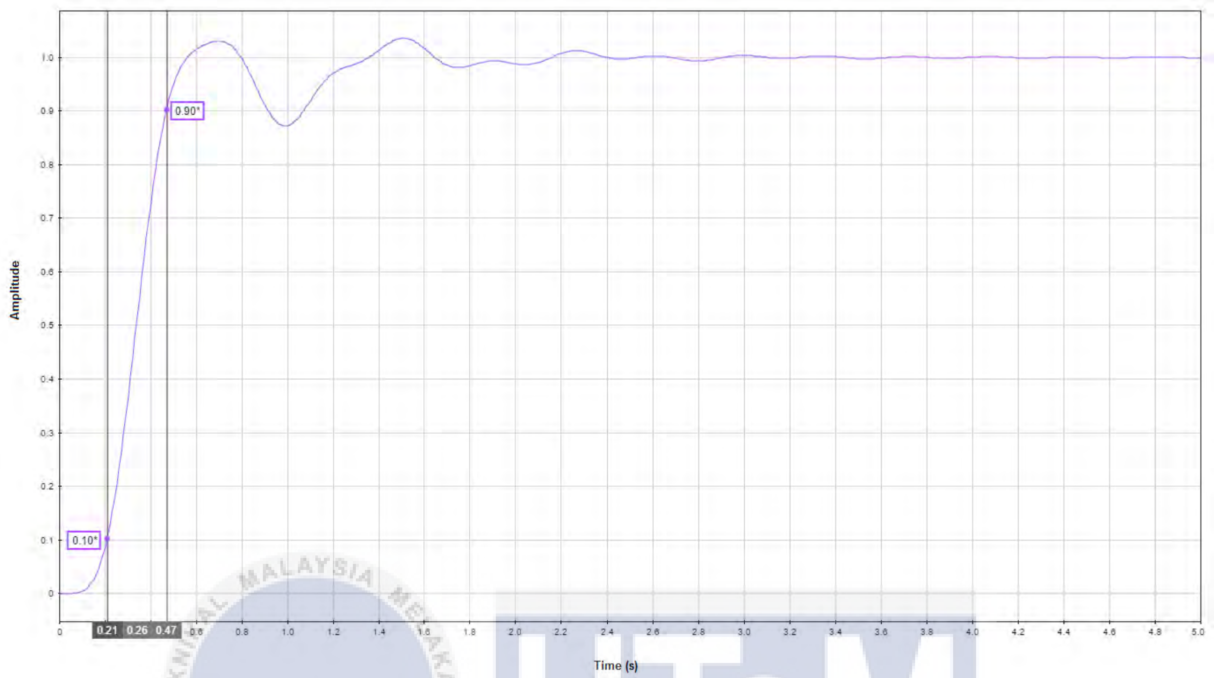


Figure 4.14: Rise time at 0.261 seconds

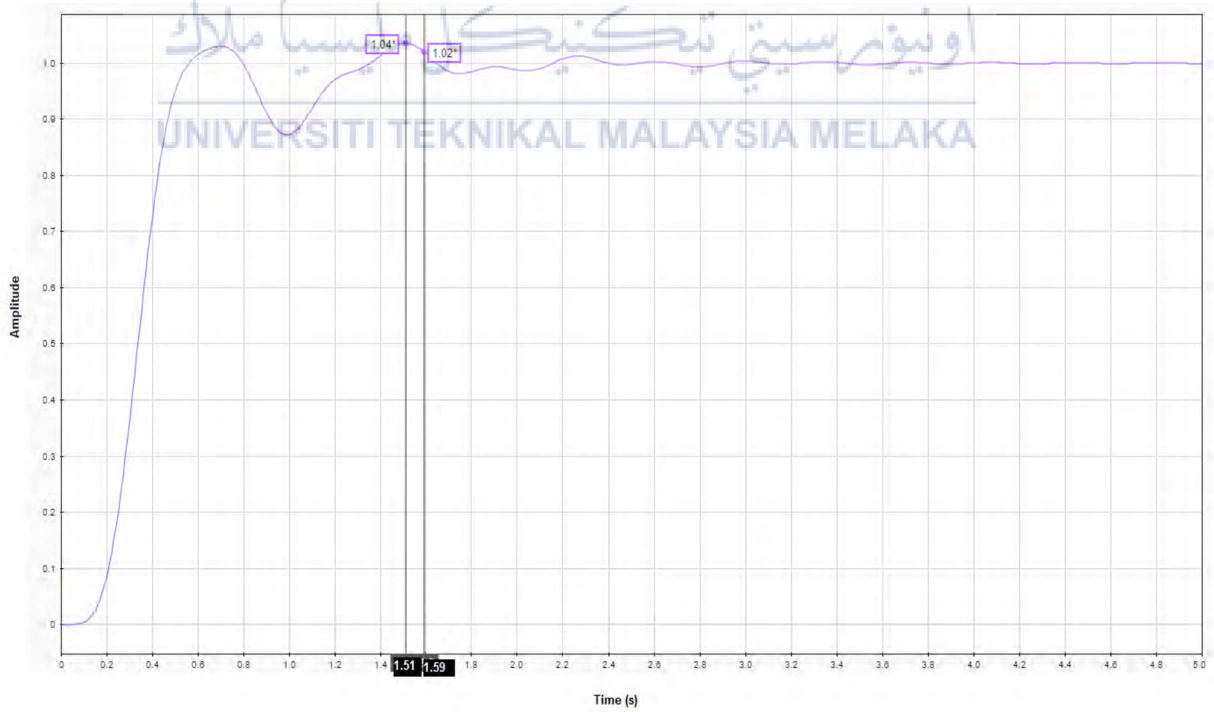


Figure 4.15: Peak amplitude of 1.04 and settling time at 1.59 seconds

PI Controller

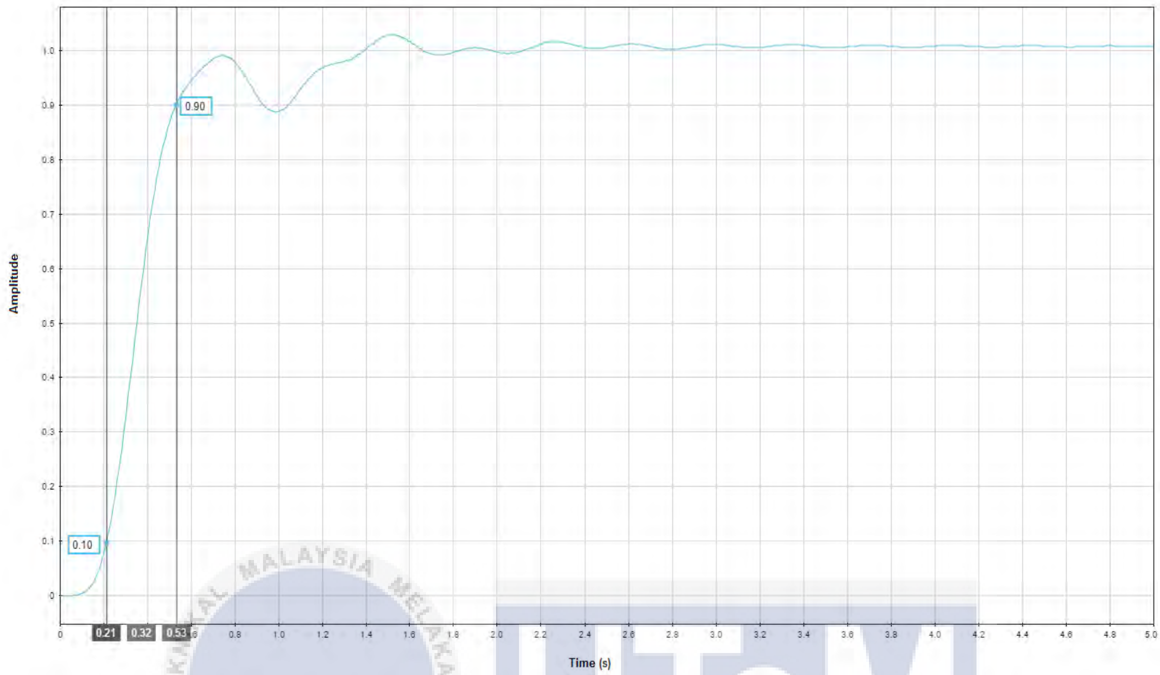


Figure 4.16: Rise time at 0.318 seconds

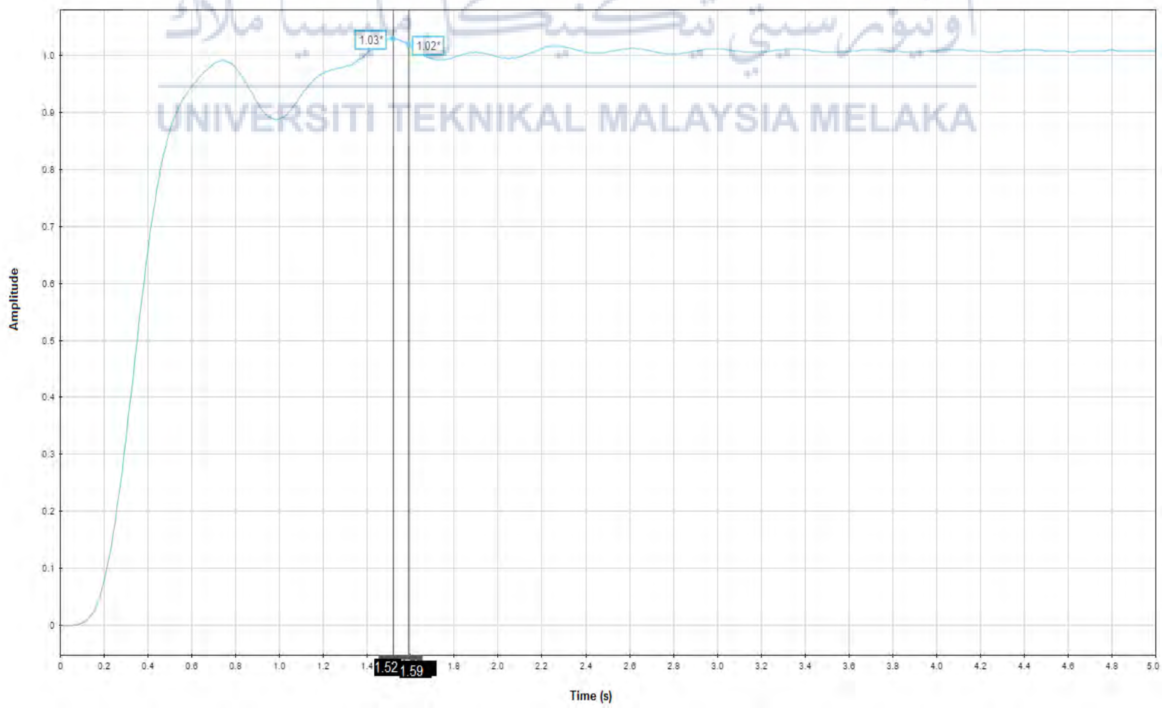


Figure 4.17: Peak amplitude of 1.03 and settling time at 1.59 seconds

PID Controller

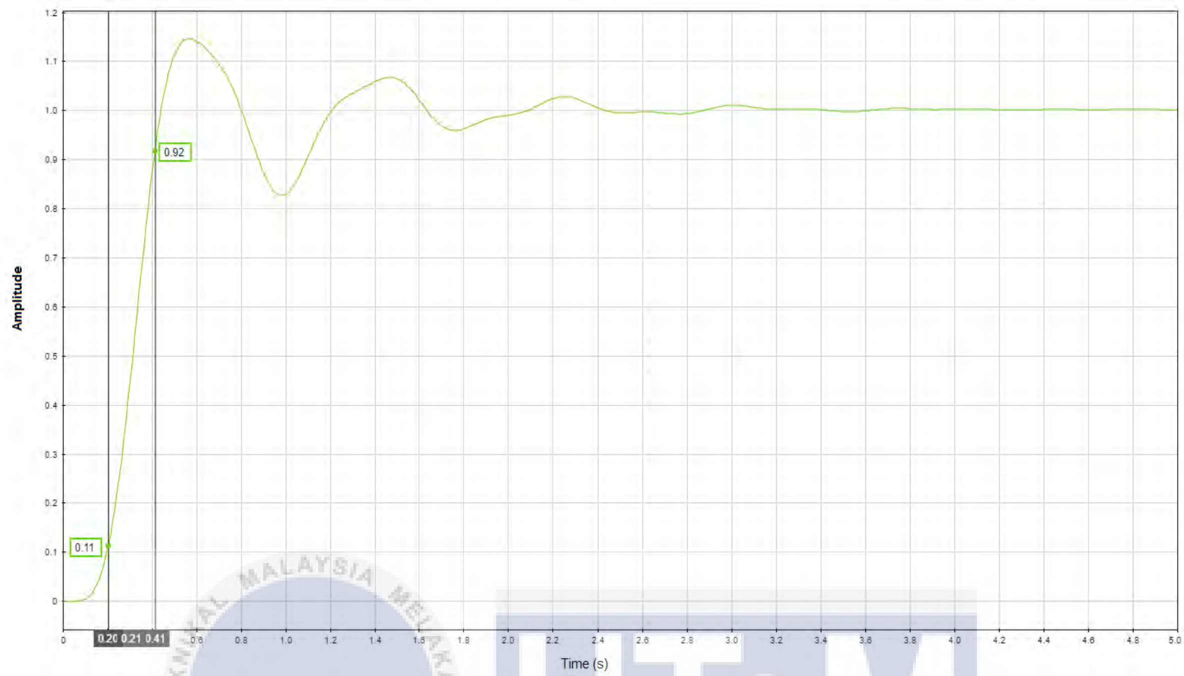


Figure 4.18: Rise time at 0.211 seconds

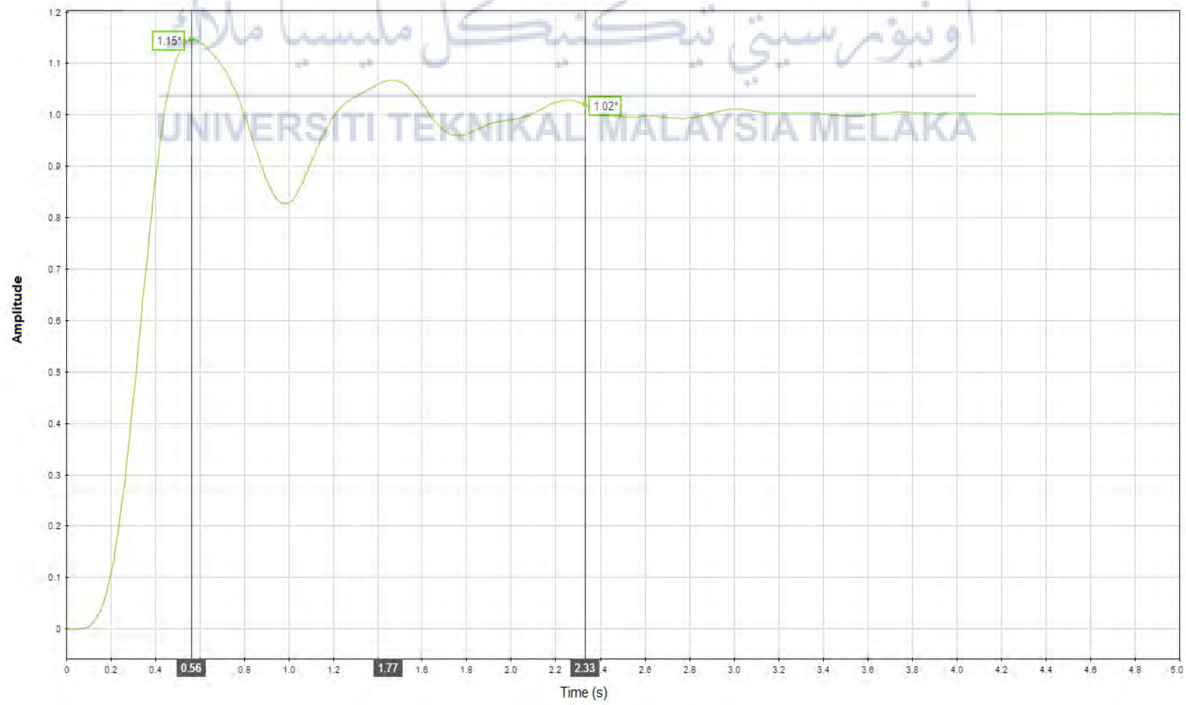


Figure 4.19: Peak amplitude of 1.15 and settling time at 2.33 seconds

Based on the results shown on P, PI and PID controller, the response shown on P and PI controller is almost similar while response on PID is slightly different since PID has more oscillation than other two controllers. The rise time of PID is faster than P and PI, and the peak amplitude is higher than P and PI. The settling time of PID controller is also slow due to there is oscillation that occurs before it could settle completely. This phenomenon makes the overshoot of PID controller higher when compared to other controllers. All controllers are approaching amplitude one which makes the P, PI and PID controllers were able to follow the specifications of the reference model that was set at early of the simulation process. The differences between P, PI, and PID can be seen in Table 4.4 as follows:

Table 4.4: Value differences between P, PI and PID control system

	Controller		
	P	**PI**	**PID**
Rise Time (s)	0.261	0.318	0.211
Settling Time (s)	1.59	1.59	2.33
Peak Time (s)	1.51	1.52	0.562
Peak amplitude	1.04	1.03	1.15
Steady State	1.0	1.0	1.0
Overshoot (%)	3.58	2.89	14.6

Even though the result obtained shows that P controller is better in terms of rising time and the time for the system to reach steady state is faster, however, the PID controller will be used as controller instead of P and PI controller because PID is the main controller that will be used to compare with MIT rule as the third objective of this study. The lack of using P controller is due to the controller can only calculate proportional gain that can only handle simple plant but limited for the complex plant. Hence, the use of PID controller is perfect for the electro-pneumatic system. The use of PID controller in the electro-pneumatic system shows that the system was able to settle faster, only the lack is that the earlier system of the electro-pneumatic system does not have overshoot while the PID controller has overshoot when applied to the system. The comparison of PID with MIT is analysed in Section 4.7 in this chapter where the comparison is done is based on the performance against reference model and also in terms of positioning control.

4.6 Simulation of Electro-Pneumatic System using MIT Rule

In this part, the method of designing MIT rule into Model Reference Adaptive Control (MRAC) with the application of second order system was discussed and the results were displayed. Apart from the basic diagram of MRAC system that consists of a controller, the controller in this system is taken out leaving the model reference alone with MIT rule using adaptation gain, γ as the system controller. The baseline or model reference of second order system that has been obtained earlier are then made as a model reference that is located inside the Model Reference Adaptive Control diagram and it acted as the model reference that the electro-pneumatic plant of fifth order system needs to follow. Figure 4.20 shows the Simulink diagram of Model Reference Adaptive Control System.

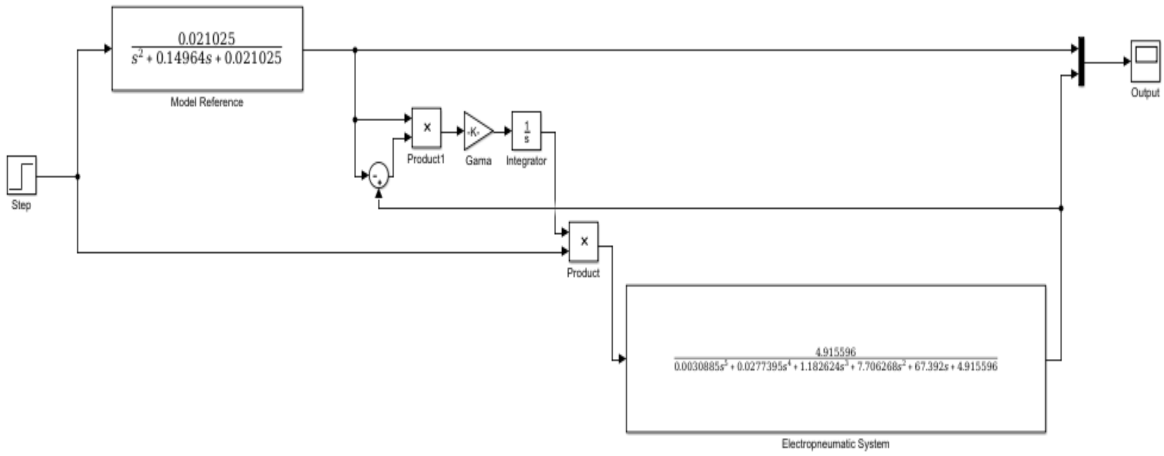
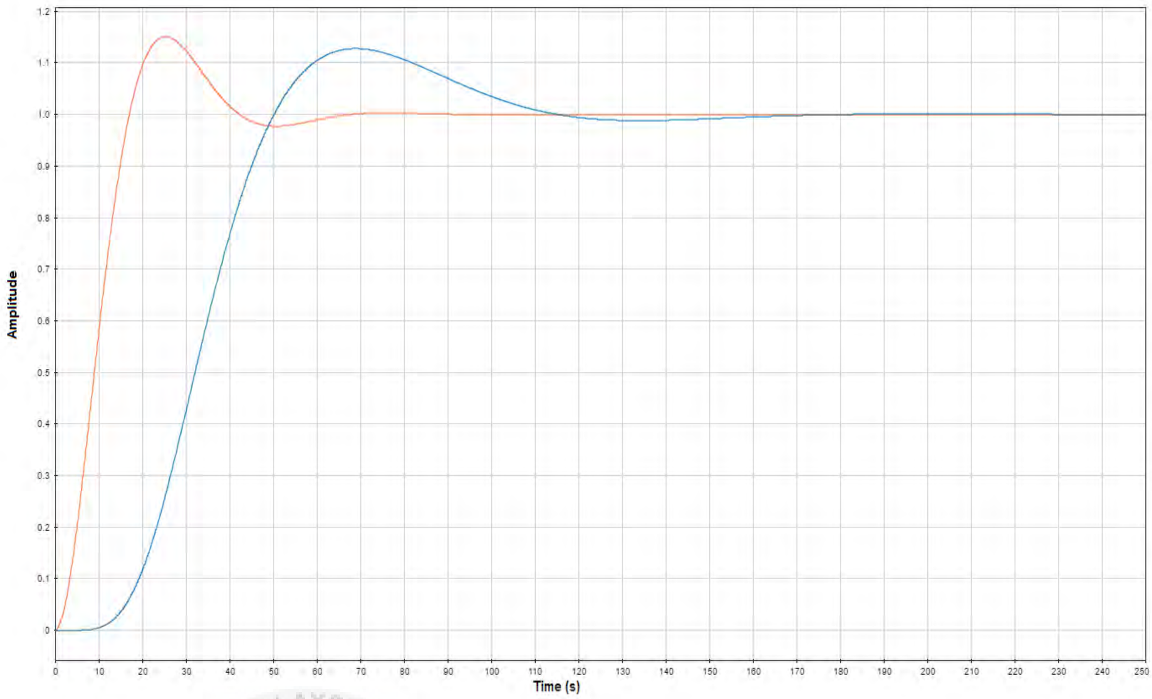
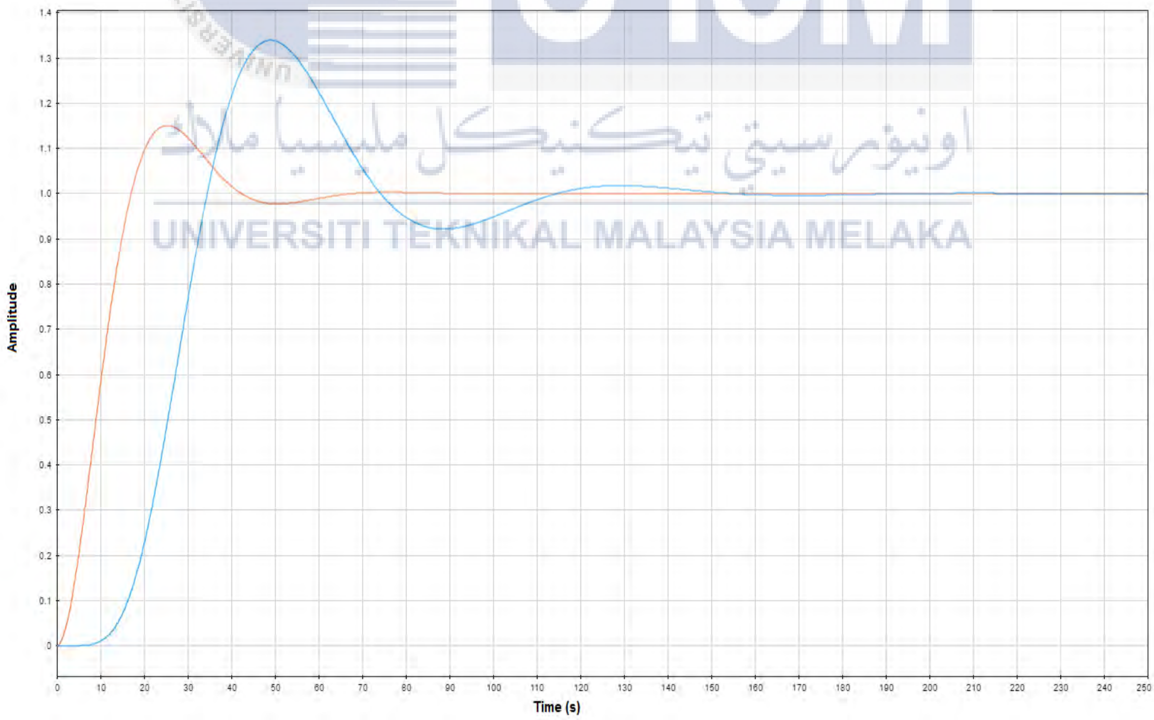


Figure 4.20: Simulink diagram of Model Reference Adaptive Control with MIT rule

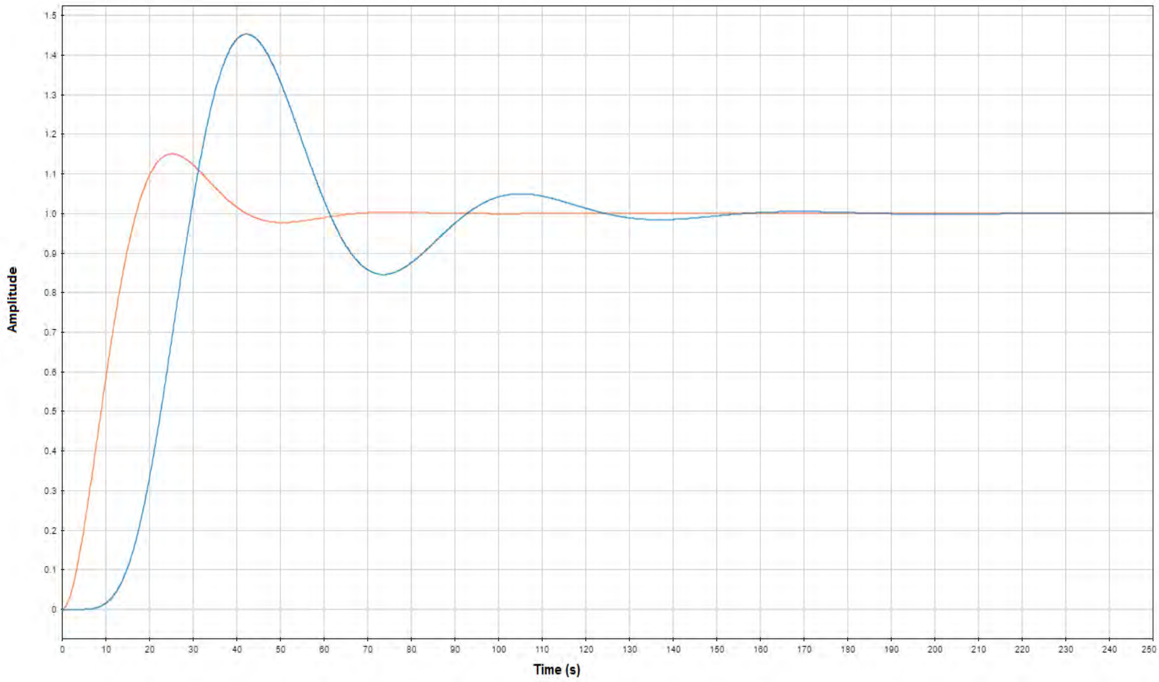
After the diagram has been designed in Simulink, the value of adaptation gain, γ need to be controlled in order to get the best results and to analyse the response of the closed loop system of the electro-pneumatic plant whether it could follow the reference model or not. The adaptation gain γ was set with four different values which are -0.05, -0.10, -0.15 and -0.20. The use of positive values as adaptation gain could not be done due to the response is reversed from the model reference graph. The gain values obtained are the closest response with the reference model. Figure 4.21 shows the response obtained when adaptation gain was set into several values. The blue line in the graph represents the electro-pneumatic plant while the orange line in the graph represent the model reference that the plant needs to follow. Table 4.5 shows the differences in performance for the adaptation gain of -0.05 until -0.20.


Amplitude
Time (s)
0
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
1.0
1.1
1.2
0
10
20
30
40
50
60
70
80
90
100
110
120
130
140
150
160
170
180
190
200
210
220
230
240
250

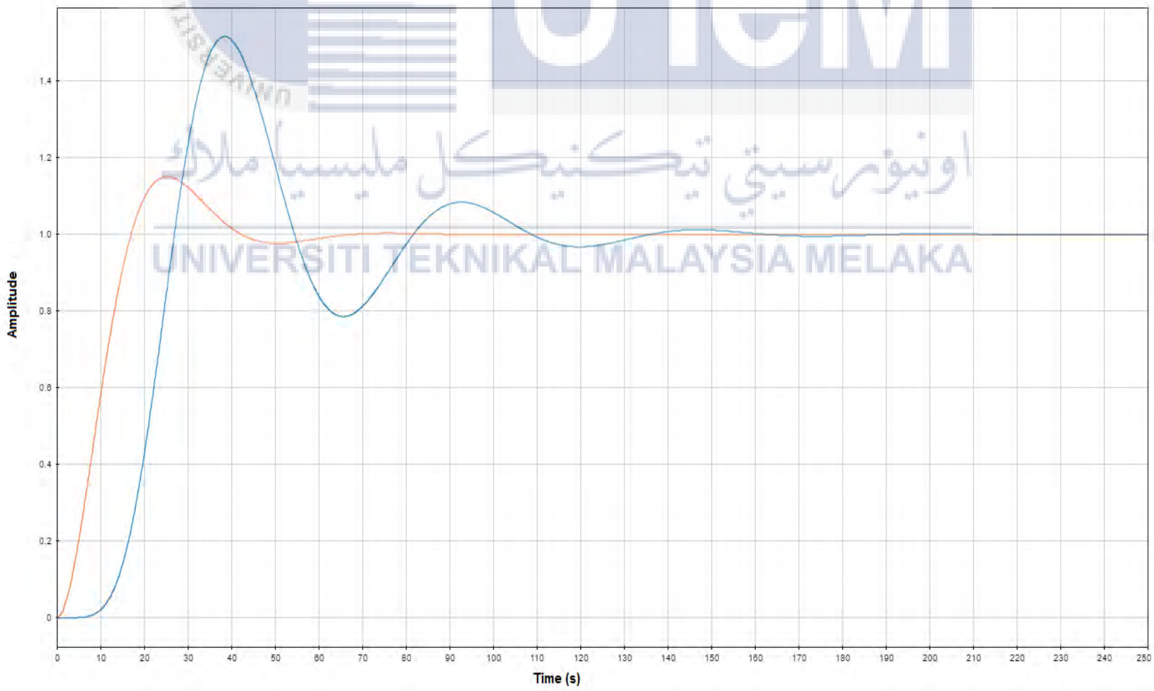
(a)


Amplitude
Time (s)
0
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
1.0
1.1
1.2
1.3
1.4
0
10
20
30
40
50
60
70
80
90
100
110
120
130
140
150
160
170
180
190
200
210
220
230
240
250

(b)



(c)



(d)

Figure 4.21: Response changes when the adaptation gain, γ was set to (a) -0.05, (b) -0.10, (c) -0.15 and (d) -0.20

Table 4.5: Adaptation gain against the performance

Characteristic \ Adaptation Gain , γ	(a) -0.05	(b) -0.10	(c) -0.15	(d) -0.20
Rise Time (s)	26	16.3	13.3	11.7
Settling Time (s)	106.4	133.5	138.5	128.7
Peak Time (s)	69	49.5	42	38.3
Peak amplitude	1.13	1.34	1.45	1.52
Steady State	1	1	1	1
Overshoot (%)	13	34	45	52

Based on Figure 4.21 above, the response obtained shows that the electro-pneumatic plant could not follow the response of reference model directly. The shape of the response in plant system and reference model are similar but it cannot track the plant system based on the performance characteristics. This can be seen where at (a), the plant experienced a delay at beginning of the system when the adaptation gain was set to -0.05 and the response settle later than the reference model. The time of delay is around 20 seconds and this can also be seen at (b) and (c). However, at (d), the delay has shortened to 10 seconds. In between -0.05 until -0.20, the oscillation starts to increase as the lower the adaptation gain, the oscillation increases while when the adaptation gain is higher, the oscillation starts to decrease. The increase in oscillation makes the peak amplitude becomes much higher than the reference model.

Due to the peak amplitude of reference model is only at amplitude 1.15, the plant system could not follow it at all. This can be seen where at (a), the value of peak amplitude is near to reference model amplitude but the time to reach the amplitude or the rise time of the system become much longer than the model. Meanwhile, when the adaptation gain is set to -0.10, the amplitude arose higher than the reference model which makes the gains that were set after the value keep getting higher and could not follow the reference amplitude.

The time for the response to reach steady state is also slower when compared to the reference model, no matter at what values that the adaptation gains was set. The percentage overshoot also keeps increasing when the values become smaller. Based on Table 4.5, the rise time at a gain (d) is faster compared to other gains which are 11.7 seconds. The time for the system to reach steady state is high at a gain (a) with 106.4 seconds where other gains are much slower. The percentage overshoot at (a) is lower than another three gains which make the peak amplitude is lower, only that the time to reach the peak is too slow rather than at -0.10, -0.15 and -0.20. Finally, The electro-pneumatic plant could not follow directly the reference model in terms of peak amplitude, settling time, rise time, overshoot and the peak time, only that it still can reach a steady state but in a longer time compared to the reference model. The comparison of MIT rule with PID is analysed in Section 4.7 in this chapter where the comparison is done is based on the reference model and also in terms of positioning control.

4.7 Comparison of the performance of PID Controller and MIT Rule

In this part, the performance of PID controller is compared using two different methods. The first method is to compare the performance of PID and MIT with the reference model and the second method is to compare the performance of both controllers in terms of the position control. For PID controller, the values are taken directly when designing the controller into electro-pneumatic. However, for MIT rule, since the plant could not follow the reference model of the required system, the range of values for all adaptation gains are used in order to compare the performance due to the results obtained are not specific.

4.7.1 Performance based on reference model

The reference model that has been set based on the second order system of the electro-pneumatic system is used as a guideline that will help to compare the performance of PID controller and MIT rule. The method of comparison is based on the characteristics that have been set when designing the model reference which is the overshoot of the system must be less than 15 percent of faster rise time and the system must reach a steady state which is at amplitude one in a short time. Table 4.6 shows the comparison of PID controller and the reference model while Table 4.7 shows the comparison of MIT rule and the reference model.

Table 4.6: Comparison of PID Controller and the reference model

Reference Model	**Characteristics**	**PID controller**
11.5	Rise Time (s)	0.211
53.3	Settling Time (s)	2.33
25.3	Peak Time (s)	0.562
1.15	Peak Amplitude	1.15
1	Steady State	1
15	Overshoot (%)	14.6

From Table 4.6, the rise time when applying PID controller into the system is much faster than required model with 0.211 seconds from 11.5 seconds. The time for the controller to reach steady state is also faster compared to the reference model where both are reaching amplitude one. Both reference model and the system that have PID controller are sharing the same peak amplitude and the percentage overshoot of PID controller is 14.6 percent which makes it less than 15 percent of the reference model. Hence, PID controller is able to follow the reference model, and the response of the system when applying PID controller is way better than the reference model itself.

Table 4.7: Comparison on MIT rule and reference model

Reference Model	Characteristics	MIT Rule
11.5	Rise Time (s)	11.7 – 26
53.3	Settling Time (s)	106.4 – 128.7
25.3	Peak Time (s)	38.3 – 69
1.15	Peak Amplitude	1.13 – 1.52
1	Steady State	1
15.1	Overshoot (%)	13 – 52

Based on Table 4.7, the reference model and the system after applying MIT rule can reach steady state at amplitude one. Time for reach the peak when using MIT rule controller is in the range 38.3 seconds until 69 seconds, which are longer in time when compared to the reference model of 25.3 seconds. The reference model settles in 53.3 seconds, however, after adjusting adaptation gain for several times, the plant system still took a longer time to settle. So, when the peak time and settling time is slow, other characteristics in the response also will be affected. The duration of the system to operate will get slower even though the value of adaptation gain is adjusted whether it is increasing or decreasing. Therefore, MIT rule is unable to follow the response of reference plant even though both of the plants meet the required output. MIT rule is not suitable for the reference model because of the performance characteristics of MIT rule could not afford the plant system in order to follow the response of reference model.

4.7.2 Performance Based on The Position Control

The overall performance of the plant system after applying PID controller and MIT rule is compared. The comparison is focusing on the response of the electro-pneumatic plant after applying PID controller and MIT rule from the beginning of the system until the final output obtained from the system. The method of comparison is based on how the system performance and behaviour when it is controlled using the basic controller design of PID and MIT in order to meet the required position.

When PID controller is applied to the electro-pneumatic system, the system is controlled based on the gain that has been set during tuning process where the value obtained is fixed and cannot be changed. The values are fixed due to the application of Ziegler–Nichols method. Therefore, the result of the system after controlled by PID controller cannot be adjusted and it is the final result that can be obtained. Based on the response at Figure 4.18 and 4.19, the rise time of the system at the beginning operation which is when the piston rod comes out from the cylinder, it moves smoothly and fasts until overshoot occurs. When the overshoot has occurred, the piston is shaking a little bit on the way to meet the final position. It can be seen at the response behaviour of PID controller between the time of overshoot occur and the time before it reaches the final position. However, at the end, the plant system still can be controlled in order to meet the desired plant output.

The application of MIT rule into the electro-pneumatic plant system is using MRAC model where the point in controlling the plant system is the adjustment mechanism. This means the system is controlled without fixed gain values, unlike the PID controller that has fixed gains. The gains used in controlling the plant system can be adjusted so that the plant system able to meet the desired output. As seen in Figure 4.21, the response meets the desired output, however, the response from the beginning and before it meets the final output is different based on the chosen adaptation gain. The changes are almost similar for all adaptation gain where it shows that the plant system can still be controlled but cannot meet the desired position perfectly in the required time. So, that is the only limit of the MIT rule in controlling the plant system.

Finally, the best controller chosen to control electro-pneumatic system is PID controller because it provides the best response and it could meet desired output in a short time compared to MIT rule. MIT rule could not be chosen since it cannot track the desired output even though is had adaptation mechanism and lacks performance quality when compared to PID.

CHAPTER 5

CONCLUSION AND RECOMMENDATIONS

5.1 Introduction

This chapter concludes all works that have been done from the beginning of the project until the end of the project. This chapter also will provide recommendations that can be done in future that relates to this project.

5.2 Conclusion

The successful of the controller that applies into the system depends on the system itself whether the system is stable or not before applying controller. If the system is not stable the application of controller into the system did not change anything. However, if the system is stable, the controller can help the system to meet its desired response. In this project, the electro-pneumatic plant that has been modelled using mathematical modelling approach is already in stable response. The use of controller into the system results in different response and performance based on the chosen controller. The changes in performance of the controller are different due to the controllers have their own method when designing the controller into the system. The results obtained after applying PID controller and MIT rule into the system are different even though the required output needed is similar.

The performance of PID controller towards the system shows a good response when compared to MIT rule since the result is close to the requirements of the system needed. Although the PID controller is only a basic controller, it still can control the position of electro-pneumatic to reach it's desired output compared to the MIT rule in MRAC method that has the ability to track the reference model needed. This means that PID controller is suitable to be used by the plant system that has been modelled in this project. To conclude, this project has three objectives where the first objective is achieved in Chapter 3 and the second objective and third objective has been achieved in Chapter 4.

5.3 Recommendations

Based on the overall of the project, there are some improvements that need to be done in order to achieve better results of the controller. This improvement will be focusing on MIT rule towards the electro-pneumatic plant system. There are several elements that need to be improved as well as the recommendations for the elements:

1) Element: The MIT rule could not make the plant system to track the reference model that has been set due to the high demand on the desired output.

   Recommendation: Modify the performance characteristics of the reference model by reducing the settling time and the rise time of the response

2) Element: The basic design of MIT rule in MRAC system chosen in this system is general and not suitable for the modelled electro-pneumatic plant system.

   Recommendation :

   a. Design MRAC system with modified MIT rule to improve the performance of the system

   b. Apply PID controller into the basic MRAC system with MIT rule.

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