

DESIGN OF SLIDING MODE CONTROLLER FOR ROBUST TRACKING OF PNEUMATIC SYSTEMS

This report is submitted in accordance with requirement of the University Teknikal Malaysia Melaka (UTeM) for Bachelor Degree of Manufacturing Engineering (Hons.)



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DECLARATION

I hereby, declared this report entitled "Design of Sliding Mode Controller For Robust Tracking Of Pneumatic Systems." is the results of my own research except as cited in reference.



APPROVAL

This report is submitted to the Faculty of Manufacturing Engineering of Universiti Teknikal Malaysia Melaka (UTeM) as a partial fulfilment of the requirements for Degree of Manufacturing Engineering (Hons.). The members of the supervisory committee are as follow:



ABSTRACT

The widespread use of pneumatic actuator systems in the automation industry today is due to their many advantages, which include cheap cost, environmental friendliness, high reliability, and a high ratio of power to weight. As a result of these benefits, pneumatic actuators are increasingly being considered as viable alternatives to hydraulic actuator systems and electric servo motors for automating processes. Because of the high nonlinearities of pneumatic actuators, it is difficult to achieve accurate tracking performance. Because of this, a controller is required to regulate the system and thereby solve the high nonlinearity problem. The mathematical model of the system must first be constructed before the controllers can be created by using MATLAB software. The system model is next validated by comparing it to experimental data, which is accomplished with the use of the System Identification technique. Following that, three controllers have been designed, such as PID controller, SMC controller, and Pseudo-SMC. These three controllers will be simulated and tested on the real plant, and controller's performance will be measured in terms of maximum tracking error and RMSE with disturbance applied such as external load. The results show that SMC-based controllers can produce an average improvement of 87% in terms of maximum tracking errors and a reduction of 80% in terms of root mean square error (RMSE). Among the SMC-based controllers, pseudo-SMC controllers can reduce the chattering effect using a Fast Fourier Transform approach. However, the SMC controller can achieve a low percentage of load variation in terms of robustness. Overall, the Pseudo-SMC controller has superior robust tracking performance against the SMC controller and the PID controller.

ABSTRAK

Penggunaan sistem penggerak pneumatik yang meluas dalam industri automasi hari ini disebabkan oleh banyak kelebihannya, termasuk kos murah, keramahan alam sekitar, kebolehpercayaan yang tinggi, dan nisbah kuasa dan berat yang tinggi. Hasil daripada faedah ini, penggerak pneumatik semakin dianggap sebagai alternatif yang sesuai untuk sistem penggerak hidraulik dan motor servo elektrik untuk proses automatik. Kerana tidak linear penggerak pneumatik yang tinggi, sukar untuk mencapai prestasi penjejakan yang tepat. Oleh kerana itu, pengawal diperlukan untuk mengatur sistem dan dengan itu menyelesaikan masalah nonlineariti yang tinggi. Model matematik sistem mesti dibina terlebih dahulu sebelum pengawal dapat dibuat dengan menggunakan perisian MATLAB. Model sistem selanjutnya disahkan dengan membandingkannya dengan data eksperimen, yang dicapai dengan penggunaan teknik Pengenalan Sistem. Berikutan itu, tiga pengawal sudah direka bentuk, seperti pengawal PID, pengawal SMC, dan pengawal Pseudo - SMC. Ketiga-tiga pengawal ini akan disimulasikan dan diuji pada mesin sebenar, dan prestasi mereka akan diukur dari segi ralat penjejakan maksimum dan ralat punca min kuasa dua dengan gangguan berat luaran dikenakan pada mesin eksperimen. Hasil kajian menunjukkan bahawa pengawal berasaskan-SMC dapat menghasilkan peningkatan rata-rata 87% dari segi ralat penjejakan maksimum dan pengurangan 80% dari segi ralat punca min kuasa dua. Di antara pengawal berasaskan-SMC, pengawal pseudo- SMC dapat mengurangkan kesan signal bising menggunakan pendekatan Fast Fourier Transform. Walau bagaimanapun, pengawal SMC dapat mencapai peratusan variasi beban yang rendah dari segi ketahanan. Secara keseluruhan, pengawal Pseudo - SMC mempunyai prestasi penjejakan yang kuat terhadap pengawal SMC dan pengawal PID.

DEDICATION

For my beloved parents, Who has been giving me strength and providing me with moral, emotional, and financial support.

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LIST OF ABBREVIATIONS



RMSE	-	Root Mean Square Error
SI	-	System Identification
SMC	-	Sliding Mode Controller
SOSMC	-	Second Order of Sliding Mode Control
VSC	-	Variable Control Structure
2-DSMC	-	Discreet Second Order Sliding Mode Control



CHAPTER 1 INTRODUCTION

WALAYS !!

Pneumatic systems are power transmission systems that use compressed air as the working medium for the transfer of mechanical force. Their functioning is based on a concept that is similar to that of hydraulic power systems. When a prime mover produces mechanical energy, an air compressor turns that energy into primarily pressure energy in the form of compressed air. This chapter will present a project titled "Design of Sliding Mode Controller (SMC) for Robust Tracking of Pneumatic Systems." The project is about precision control development on the pneumatic actuator to input disturbance force. The selected precision control algorithm is a robust controller named "Sliding Mode Controller (SMC)" and is used to test the robustness tracking on the pneumatic actuator. The designed controller will be designed and numerically analysed using MATLAB software. The controller will be experimented with in this project by implementing the designed controller on the actual plant, and control performance will be evaluated based on this setup.

1.1 Background

Pneumatic actuators are extensively used in today's industrial world because they are positively safe to operate, economical, simple to maintain, environmentally friendly, rapidacting, and may be directly linked to the payload. However, they also have high-order time variant dynamics, which causes nonlinearities because of the high friction force, dead band owing to stiction force produced by the sealing ring of the cylinder, and dead time due to air compressibility, among other things. These nonlinearities make it difficult to adjust the actuator's placement. Servo valves, rather than solenoid valves, were traditionally employed to operate pneumatic actuators.

To achieve a satisfactory outcome, a servo valve was employed in combination with a Sliding Mode Control (SMC) controller, and it was also used to compare linear and nonlinear control of an air pneumatic servo-drive. Because of the high precision manufacturing limits and the necessity for a built-in orifice area control circuit, earlier servo valves were sophisticated in form and huge in cost. Nowadays, servo valves are being replaced by on/off solenoid valves, which are more economical, tiny, and light weight, as well as recent improvements in valve technology, making them more desirable for application, despite the fact that they have a limited lifetime owing to valve wear and tear.

SMC is generally considered one of the most effective techniques for creating trustworthy controllers for complicated high-order nonlinear dynamic plants working in uncertain settings. The major advantages of sliding mode are that they are insensitive to plant parametric variability and have the capacity to reject disturbances, thereby minimising the requirement for accurate models. Because the sliding mode trajectory corresponds to a manifold with a lower size than the original system, the system's order is decreased, enabling the designer to decouple and simplify the design processes.

Lennels

Though SMC is a nonlinear control method that alters the dynamics of a nonlinear system, the unwanted occurrence of oscillations with a finite frequency and amplitude known as "chattering" is noticed in real implementations because of the discontinuity of control over the switching surface. SMC can manage system dynamics that aren't precisely modelled, but it requires exact state information to create a sliding surface.

1.2 Problem Statement

Previous researchers, such as (Sy Salim et al., 2014), noticed that the study of pneumatic systems was challenging because of the high nonlinearity that existed in the system, such as the high air compressibility, the existence of friction force, and the factors determining mass flow rate. This increased complexity made it harder to get the system's uncertainty parameters, which led to severe difficulty when attempting to develop robust control of the pneumatic system. The existing problem of high nonlinearity in the system may lead to reduced robust tracking performance. In order to solve nonlinearity in pneumatic actuator systems, a controller such as a PID controller, adaptive controller, or sliding mode controller is required to regulate the system.

Therefore, the controller of a pneumatic system, which is most likely an independent nonlinear system, is designed as a Sliding Mode Controller (Perruquetti & Barbot, 2002), while applying the external load to the controller system is required to attain robust tracking performance from the actual plant.

1.3 Objectives

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The objectives of this study are:

- i. To determine the system model of a pneumatic system using system identification technique.
- ii. To design an SMC controller for the servo pneumatic system.
- iii. To analyse the controller performance in terms of maximum tracking error and root mean square error.
- iv. To evaluate the SMC controller's robustness by applying additional loads to the servo pneumatic system.

Table 1.1 clarifies the relationship between problem statement and objective to make things simpler to see both relations.

Problem statement	Objectives
High nonlinearity in pneumatic servo	Objective 1 and objective 2
system	• Obtaining a new system model by using
	System Identification technique and
	applying it to the designed SMC
	controller.
Robust tracking performance	Objective 3 and Objective 4
	• Analysing the designed controller with
	maximum tracking error and RMSE
	analysis.
MALAYSIA	• Evaluate the controller robustness
ST ME	when applying additional load to the
NY AND	experimental plant system.
	JIEM
1.4 Scope of Study	
The scopes of this project are:	اونيۇم,سىتى تىك
i. Only limited to the actuator of the serv	MALAYSIA MELAKA
ii. Analysis of the accuracy and robust t	racking of the pneumatic actuator is based on
0.1 Hertz of sinusoidal wave input, wi	th an amplitude of 60 mm.

Table 1.1: Relationship between problem statement and objectives

- iii. Variations of external loads from 0 kg until 9 kg with an increment of 1 kg are used for the robust analyses.
- iv. Simulation will be performed by using MATLAB software, and the designed controller will be experimented into the actual plant setup.
- v. Evaluation of the maximum tracking error and stability of the controller plant.

CHAPTER 2 LITERATURE REVIEW

2.1 Introduction

An introduction to literature study on subjects relating to controller design and accurate positioning of machine tool applications is provided in this chapter. The positioning system under consideration is used in the production engineering fields and is the actuator of a servo-pneumatic actuator system. The first section of the chapter discusses the approaches to positioning systems that have been developed by prior researchers in the past. The next section is a review of the techniques for managing the system using several broad kinds of controllers that have been studied in the previous year. Finally, in the last portion of this chapter, a summary of the studies completed by past researchers will be provided.

2.2 Positioning system

According to (Das et al., 2013), one of the most difficult areas to control in pneumatic actuation seems to be the position control of the pneumatic actuation system and the overall functioning of the pneumatics system. Consequently, motion control for the pneumatic servo system is one of the most difficult aspects of the system. The use of pneumatic actuators in robotic manipulators and mechatronics applications has increased significantly. As a result, the demands for precise position tracking performance and erratic positioning between the

two hard stop endpoints of the pneumatic actuator stroke have increased as well (Azahar et al., 2021), Figure 2.1 shows the basic structure of servo pneumatic positioning system. Diverse efforts have been undertaken to improve the design of control systems for pneumatic actuators, notably in position control.



Figure 2.1 : Basic Structure of Servo Pneumatic positioning system

2.2.1 Servo pneumatic system

Actuators powered by pneumatics are employed in batch automation of sequences as well as continuous control (Saravanakumar et al., 2017). Traditionally, pneumatic actuators have been employed to move objects between two predetermined stops. With the help of electro pneumatic servo drives, it is possible to extend the capabilities of pneumatic drives and utilise them to operate in many positions. In general, this form of servo pneumatic technology employs a feedback-based closed-loop control method and has been illustrated in Figure 2.2.



Figure 2.2: Control of pneumatic cylinder position by closed-loop feedback (Saravanakumar et al., 2017)

2.2.1.1 Implementation of servo pneumatic systems

Industry and other automated systems might benefit from the usage of the servo pneumatic system, which has various uses. Pneumatic actuators are frequently applied in the fields of automation, robotics, and manufacturing. This technology is particularly advantageous for physical system manipulation and the fast motion of mechanical items, as well as for assembling, packing, stacking, and clamping. A gripper with independently moveable jaws operated by a pneumatic actuator was conceived and built by (Gauchel & Schell, 2006).

According to (Moilanen, 2004) the servo pneumatic drive was used in a high temperature water and irradiation environment to evaluate material testing equipment. (Backe, 1986) has done a thorough examination of the usage of servo pneumatic drives for adaptable mechanical material handling systems. According to (J. Wang et al., 1999) the use of servo pneumatics for packaging food items as it is an affordable and ecologically beneficial technology.

(Zhang et al., 2009) built a three-axis climbing robot using servo pneumatic drives. Many applications would benefit from using servo pneumatic actuators instead of hydraulic ones. (Fischer et al., 2008) built a prototype of an MRI-compatible manipulator driven by pneumatic servo drives that may be utilised for needle insertion in a medical procedure. (Bobrow & McDonell, 1998) servo motors are used to drive a pneumatic robot that was created to be lightweight and inexpensive. (Hesselroth et al., 1994) developed a pneumaticpowered soft arm for robots. (Ramos-Arreguin et al., 2008) paper suggests that robotic manipulating lines can be operated with greater force-to-weight ratios due to a pneumatic servo drive system.

2.2.1.2 Features of Servo Pneumatics Systems

The accuracy and speed of an actuator are the two most critical aspects to consider. To analyse the performance of servo pneumatic positioning systems, two main types of observations are used: the positioning test and the trajectory tracking test.

A position test, or a point-to-point test, according to (Beater, 2007) is performed by changing the reference signal by a step size of 0 % to 100%, or by a step size of 20% to 80%, or by some known limit within the cylinder's stroke length, for example. Whenever the reference signal is required to follow a specified trajectory, the characteristics of trajectory tracking, or set-point tracking may be gained. It is common practise to apply sinusoidal (and sometimes ramp) signals to a system while evaluating its tracking capabilities. Commonly, performance indicators for evaluating system performance take into account either the speed of the system or its accuracy, depending on the situation. One of the most significant variables to decide the efficiency of a servo pneumatic positioning system is the variety of high-speed and precise sensing, actuating, and control systems.

In order to improve the overall performance of the system, the three activities listed below must be taken:

- i. Appropriate components should be chosen for the system in accordance with the requirements.
- ii. To have a better knowledge of the system's behaviour, it is necessary to model it accurately.

iii. In order to achieve exact positional control, sophisticated control algorithms must be devised.

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2.3 System modelling and identification

Pneumatic system modelling, identification, and control have been studied extensively in the existing literatures. Theoretical analysis is one of the most often used methods for generating the mathematical modelling of pneumatic actuator. Furthermore, system identification is implemented to gain the mathematical modelling of an acting cylinder of pneumatics system. An exact pneumatic actuator modelling would be highly crucial in controller design.

(Mohan & Saravanakumar, 2014) System dynamics are described in analytic form in the mathematical model of the system. Developing a control system requires a thorough grasp of system behaviour and this system model. It was shown by (D. J. D. Wang et al., 2001) that a nonlinear system model for the pneumatic cylinder actuator was developed, with the overall system being considered as a cascade connection of two nonlinear affine subsystems.

A thorough mathematical model of the servo pneumatic positioning system, encompassing cylinder dynamics, air dynamics within the cylinder, the influence of connecting tubes, and valve spool dynamics, was developed by (Richer & Hurmuzlu, 2000)

According to (Valdiero et al., 2011), a combined conceptual and identifiable model of the proportional valve controlled pneumatic servo system was computed and identified. With the use of a curve fitting technique, the flow rate equations for the proportional valve that was applied in the model were generated. For simulation purposes, (Takosoglu et al., 2009) had built a simpler model of the servo pneumatic positioning system.

A research from (Lin-Chen et al., 2003) performed a software package for computer aided modelling and simulation to develop a model of certain system components that could be tested.

While (Harris et al., 2012) constructed and validated a mathematical model of an industrial air compressor that contained nonlinearities. The flow rate coefficients were also calculated using a system identification technique.

Furthermore, (Filipovic et al., 2011) provided the linear stochastic modelling technique with parameters estimation for the servo pneumatic system. A robust Kalman filter was employed to identify the parameters of the model, and the results were analysed.

Additionally, (Saleem et al., 2009) designed and built a system for recognising and controlling electro pneumatic servo motors in a live environment. It has been decided to use a Mixed-Reality Environment to identify the system in order to avoid the higher difficulty associated with servo-pneumatic system modelling and control. The Recursive Least Squares (RLS) algorithm and the Auto-Regressive Moving-Average (ARMA) model have been used to identify the system.

By using a developed mathematical model of a servo pneumatic system that used solenoid valves controlled by pulse width modulation (PWM) and simple air flow rate equations (Le et al., 2010).

A research from (Messina et al., 2005) presents an alternative representation of the system with on-off control valve which closed and open time responses have been regulated by the PWM application. The previous unknown properties of the solenoid valve have been proven experimentally.

(Sorli et al., 1999) built a bond graph modelling of the pneumatic positioning system that encompassed all of the nonlinear dynamic systems. In addition, the model has been simulated and validated using the MATLAB-Simulink software.

According to prior research (Lai et al., 2012), there are a variety of structures that may be used in the estimating process, including State Space models and linear parametric models such as ARX, ARMAX, and BJ among others. An auto-regressive exogenous (ARX) model with a control valve was used for the previous study's model structure of a pneumatic actuator system with a control valve. Table 2.1 shows the features of different model structures.

The Auto-Regressive Moving Average with Exogenous (ARMAX) technique was applied by (M. F. Rahmat, 2012) for an industrial pneumatic actuator. In the next step, (Sy Salim et al., 2014) utilised the State Space model in the system identification process in order to derive the transfer function for the pneumatic drive system. The purpose of this research was to determine the transfer function of the plant by using the general transfer function model. The third order method was chosen for this study because, according to earlier research, the majority of researchers employed the third or fourth order system.

Model	Features	
ARX	Most basic model with the signal.	
	• High efficiency with polynomial estimation.	
	• Satisfied solution in global minimum loss function.	
ARMAX	Dynamic disturbances are included.	
	• Advantageous as dominating disturbance enter early in the process	
BJ	Completes the model with the disturbance properties modelled	
	separately from the system dynamics	

Table 2.1: Features of different estimation model structure

2.4 A Dynamic model of pneumatic actuator

The mathematical model of the Pneumatic System, which includes a pneumatic actuator and a proportional control valve, has been examined in this part by (Lafmejani et al., 2016).

(Lee & Li, 2016) shows that, the opening area of the proportional directional control valve's orifice in the rodless pneumatic servo system relies on the control input to affect the air flow. As air flows into the rodless pneumatic cylinder, the pressure differential between the two-cylinder chambers causes the pneumatic cylinder to move. For a thorough understanding of the pneumatic actuator system's dynamic behaviour, it is common to need separate mathematical formulations for the valve, actuator, and load dynamics. Figure 2.3 illustrates a coordinate system that may be used for this kind of study.



Figure 2.3: Pneumatic actuator schematic drawing (Lee & Li, 2016)

(Carlos Valdiero et al., 2006) point that variability and unknown parameters are present in the pneumatic system, which is nonlinear. Fundamentally, the inherent issues connected with natural characteristics mean that this pneumatic actuator experiences certain nonlinearity concerns. Compressed air is used as a fluid medium in the pneumatic system, and nonlinear air flow occurs when the air flows through the pneumatic system components. Establishing a good motion control design for a dynamic pneumatic system is difficult because of the many unknowns involved. Friction is a frequent factor to consider while developing a dynamic pneumatic system model. The relative motion between the piston rod's surface contacts and the contacting surface of the cylinder's sealing surface makes it a crucial parameter. Hysteresis, instability, tracking inaccuracy and overshoot are only a few of the negative impacts of friction.

Pneumatic actuator size characteristics were studied by (Mohan & Saravanakumar, 2014) and shown to have an impact on system performance. A research done by (Sorli et al., 1999) researched the pneumatic actuator's dynamics are influenced by the following builtin parameters:

- i. The dimension of the cylinder chambers.
- ii. The extent of the cylinder stroke.
- iii. The set of seals and moveable component support that affect friction forces.
- iv. Supply pressure levels at various levels.
- v. The length and size of connecting hoses.

2.5 Pneumatic actuator controller design

2.5.1 Adaptive controller

According to (Bartolini et al., 1999) dynamic systems, both linear and nonlinear, that are subject to uncertainty may be controlled using adaptive control. The uncertainties of these systems can be described as the product of an unknown constant matrix and a vector with a known time function. TEKNIKAL MALAYSIA MELAKA

There are a variety of adaptive ways that may be used to deal with uncertainty in pneumatic systems as well. A typical assumption for these adaptive methods to be practical is that all unknown parameters must be time invariant. (Tsai & Huang, 2008) researched that the extremely nonlinear dynamics of pneumatic systems cannot always be represented in a linearly parameterized fashion, making it impossible to isolate the unknown parameter vector in these systems. As a result of the time-varying uncertainties, traditional adaptive designs are not practical in this case to identify the update laws.

(Zhu & Barth, 2010) researched the Model Reference Adaptive Controller (MRAC) in a servo pneumatic actuator system, including a primary focus on mitigating friction and payload variables in the system. According to the results of the research, the most typically encountered uncertainties were friction and payloads. Normal research practises did not

clarify the friction that existed in the system. When the piston and rod seal made contact with each other while sliding through the pneumatic actuator system, friction was created.

This research compares the position control performance of its motor systems to prior studies, which used three adaptive controllers such as self-tuning adaptive controllers, backstepping adaptive controllers, and MRAC controllers of the loadstone linear motor drive system. As a result of this comparison, it has been shown that pneumatic actuators are capable of producing precise position control comparable to electrical systems.

In the first place, the friction model was chosen on the basis of the exponential static friction model or Gaussian model, which incorporates three different types of friction, which is, Stribeck friction, viscous friction, and Coulomb friction. The SMC was developed to sustain the robustness, good analytic performance, and stability of a nonlinear control system in the presence of nonlinear modelling error, and the MRAC controller was developed to provide adaptive frictional compensation in the presence of nonlinear modelling error. The result of the study has been shown in Figure 2.4.

A 60 mm step input with a rise time of around 200 milliseconds was used to test the suggested control system, and the steady-state positioning precision was not more than 0.05 millimetres.



Figure 2.4: Upper side and lower side steady state error (Zhu & Barth, 2010)

(Gao & Feng, 2005) designed an adaptive controller with a goal on improving the accuracy of pneumatic servo position control systems, this research developed an improved friction compensation method that was tested and validated. The designed controller

adaptive fuzzy-PD controller was used. Although the fuzzy controller can regulate the position of a pneumatic servo system, in order to develop this practical function, the adaptive adjustment must be created and implemented in conjunction with a typical fuzzy controller in order to correct for the frictional resistance. The experimental results show a constant load of less than 1 seconds and 0.3mm with settling time and steady state error, also with a reduced overshoot as shown in the result in Figure 2.5. It should be noted that this approach has not been tried on the system with different conditions.



UNIVERSITI TEKNIKAL MALAYSIA MELAKA 2.5.2 PID controller

The time domain output of a PID controller, which is equivalent to the control input to the system, is computed from the feedback error as in Eq. 2.1:

$$u(t) = K_{\rm p}e(t) + K_{\rm i}\int_0^t e(\tau)\mathrm{d}\tau + K_{\rm d}\frac{\mathrm{d}e(t)}{\mathrm{d}t}$$
(2.1)

Where K_p , K_i , K_d , represent the variables for Proportional, Integral and Derivative. The application of the PID algorithm does not ensure optimum system control or control stability. Consequently, since it is based only on the reaction of the measured process variable, rather than on information or a model of the underlying process, the PID controller is very versatile.

(Ilchmann et al., 2006) research has been stated that the pneumatic actuators system has constraints that are included of the dominating dynamic behaviour caused by the nonlinear function. In order to enable realistic tracking of a wide set of reference trajectory tracking by developing a mathematical model and response of linearization in the position control, the proportional output response controller with saturation was presented as a control design approach. The proportional response force controller corrected the derivatives in the reference signal and the disruption of piston velocity.

Furthermore, the author (Papoutsidakis et al., 2009) concentrated on improving the position of an actuator of pneumatic system. However, the high air compressibility and friction force in the system constrain its numerous applications. In the same research, the traditional PID controller was employed, with the Kp, Ki, and Kd parameters being tuned using the Zigler Nichols tuning approach. A proportional controller was first constructed, but when it reached the state of perpetual oscillation, it was rejected by the positioning system. Then, in order to solve the problem, a PD-controller was used, which had a positive effect on lowering the rise time while also ensuring that oscillation did not occur.

In the next step, the Proportional-Integral controller is integrated into the system; nevertheless, the rising time of the system is not up to par than when the Proportional-Derivative controller was used, and the system remained constant error. The PID-controller surpassed all other measured and simulated controllers in terms of reduction of rise time and error; however, the likelihood of overshoot was increased as the time interval rose. A study of the computed research revealed that the system's behaviour provided the highest level of satisfaction and resulted in the creation of a model capable of being evaluated in simulation to assess performance. In this study, a classical PID was suggested in order to improve the accuracy of the position done in the simulation while also keeping the cost to a minimum. The suggested controller, on the other hand, was difficult to adjust since it needed to be evaluated in the simulation to prior being deployed in the actual plant.

Similarly, a research conducted in 2017 by (Son et al., 2017) revealed that pneumatic artificial muscle has once again made a significant contribution to the controller design (PAM). This research suggested a typical PID controller with feedforward control, however it was updated for this research with a novel adaptive Back-Propagation method. The pneumatic artificial muscle is very difficult to operate because of its high nonlinear properties and sensitivity to working circumstances for example, the temperatures and other pressure properties. At first, the inverse neural NARX model detects all nonlinearities aspect of the SCARA robot dynamically. In order to increase the precision and minimise the steady-

state error in the position control, NARX was integrated with a traditional PID controller to achieve these research findings. By using the Sugeno fuzzy system, a novel adaptive back-propagation algorithm was developed. The newly presented control approach has the power of learning and updating the system on its own, while also minimising tracking error to near-zero proportions. Figure 2.6 shows the comparison of the tracking performance. The suggested controller produced great control quality, was very adaptive, and was extremely robust even when no external disturbances were considered.



Figure 2.6: The comparison of the End-effector trajectory tracking performance(Son et al., 2017)

As shown by (Saleem et al., 2015) in Figure 2.7, it is possible to improve and solve the problem of complexity that arises while acquiring the system transfer function exactly by implementing the findings of their research. The cascade PID controller was presented in this work for a realistic pneumatic system with high disturbance rejection. These results give an identification of the system that may be used to develop realistic mathematical modelling of a dynamic systems.

Particle Swarm Optimization (PSO) was implemented throughout the system identification and control designing stage. The cascade PID controller offers benefits to the pneumatic system in terms of both positioning and speed control. Since it allows for the speed of the tracking within the range of the speed loop while stopping with excellent high position accuracy. When compared to a single PID, this study demonstrated that a cascade PID system with PSO tuning gives reduced steady state errors and greater transient responsive. Figure 2.8 is the resulting comparison of PID and cascaded PID in a pneumatic system.



Figure 2.7: Block Diagram of Cascade-PID control system(Saleem et al., 2015)



Figure 2.8: PID vs. Cascaded PID speed and position response in pneumatic systems (Saleem et al., 2015)

2.5.3 SMC Controller

A sliding mode controller (SMC) is used in the majority control system industries because of well-known controllers that have always been used in pneumatic actuation systems. SMC can be used in a nonlinear system, earlier research utilized this method to regulate non-linearities in a pneumatic actuator system, as achieved by (Song & Ishida, 1997), (Tsai & Huang, 2008), (Hodgson et al., 2011) and (Hidalgo & Garcia, 2017).

In 1997, several evaluations on the SMC revealed that it has been used in pneumatic servo system, according to (Song & Ishida, 1997) and this study developed a robust sliding mode control strategy by taking into account the Lyapunov stability theory and the structural properties of a pneumatic servo system, which was then implemented. In order to ensure the output tracking error must not be more than a small constant, the controller was constructed in such a way that when time (t) approaches infinity, high resilience with regard to huge unknown dynamics can be guaranteed.

The definitions and assumptions from the analytical model of the pneumatic actuator and identification demonstrated by using the Lyapunov function served as the starting point for the controller design process. The suggested controller was subsequently tested on a real plant in an attempt to demonstrate its dependability in an actual pneumatic servo system, and the results were favourable.

A weight of 30 kg in a forward direction and 100 kilogram in a backward direction was first introduced and this weight was then altered to 100 kilogram in a forward direction and 30 kilogram in a backward direction. Using both situations, it was discovered that the dynamic tracking error was not more than 2 millimetres, and the static control accuracy is roughly 0.2 millimetre in both parameters. The continuous control signal was used throughout the experiment stage. Throughout this case, it can be observed in Figure 2.9, where the impacts of nonlinear uncertainty factors are tolerated, and achieved a satisfactory tracking performance. The control technique, however, is limited to the second-order pneumatic servo system.



Figure 2.9: The experimental results of pneumatic cylinder of output tracking and output tracking error (Song & Ishida, 1997)

The SMC controller was enhanced in 2008 by another research of (Tsai & Huang, 2008) and (Yuan et al., 2008). Improvements were realised when it was proposed that a Multiple-Surface Sliding Controller (MSSC) be used for pneumatic servo systems with varied payload and uncertainty be implemented. The controller design process began with a few assumptions about the controller's capability, which were then tested. The construction of the MSSC beforehand is the determination of the number of system states of sliding surfaces or switching function. The results indicated that the SMCC trajectory tracking error in Figure 2.10 is lower than it is under PID-control in Figure 2.11, but the usage of SMC causes chattering.



Figure 2.10: Desired and actual tracking trajectories under SMCC (Tsai & Huang, 2008)



Figure 2.11: Desired and actual tracking trajectories under PID controller (Tsai & Huang, 2008)
As shown in the research by (Hodgson et al., 2011), a seven-mode sliding controller was used to minimise position inaccuracy and switching activities. The system was enhanced as a result of this improvement. This research presented a sliding mode rule for an automation system using on/off switch pneumatic actuators system. A single pneumatic actuator consisting of two chambers and powered using four on/off solenoid valves was used to improve the functionality of the suggested control design. The design of the SMC controller began with the position of the control system, from which the switching function was derived, and the stability of the system was tested using the Lyapunov function.

To implement the controller mode selection with a seven-mode controller, however, the present chamber pressures must be sophisticated enough to identify the appropriate operating modes. Finally, controller settings should be chosen in such a way that movements are smooth and switching activities are minimised. The findings indicate that the switching activities are decreased when the suggested seven-mode controller algorithm is compared to the three-mode sliding mode controller, Additionally, the suggested controller reduced the tracking error to 0.45 mm.



Figure 2.12: Seven- mode sliding controller diagram(Hodgson et al., 2011)

An enhancement to minimize the control valve friction impact was addressed in a research by (Hidalgo & Garcia, 2017). The controller was suggested in a couple of separate techniques. The solution for the first method to manage the flow of the plant using a valve as a control element which incorporates SMC under an external topology using various sample period. The Eq.2.2 is for an external topology approach.

$$u = \hat{u} - k \operatorname{sat} \left(\frac{s}{\Phi}\right) \tag{2.2}$$

An internal topology was employed as the second method which is, integrating the SMC. In this technique, SMC functions as a slave control loop for the valve position stem while PI controller acts as a master control loop in controlling the flow. The experimental result revealed that the integrated SMC under the external topology without a state observer with the sample times of 1 millisecond and 10 milliseconds generated the good outcome.

$$\hat{u} = \frac{1}{K_p S_a} \left(K x_1 + F_{at} \right) + \frac{m}{K_p S_a} \left(\frac{d^2 y_{\text{sec}SP}}{dt^2} - 2\lambda \frac{d\hat{y}_{\text{sec}}}{dt} - \lambda^2 \hat{y}_{\text{sec}} \right)$$
(2.3)

In comparison, a better result was achieved with the application of 100 milliseconds sampling times, although chattering problems occurred in that phase. Even so, the integrated SMC is suited for a very high-performance control loop as illustrated in Figure 2.13 but high implementation cost.



Figure 2.13: State Observer structure (Hidalgo & Garcia, 2017)

2.6 Sliding Mode Controller Design

2.6.1 Reachability condition

It has been claimed that the motion of the slide was independent of the control during the experiment. Although it is evident that the control system must be developed in this manner. (Oveisi & Nestorović, 2016) based on the classical sliding mode controller, the equivalent control effort is achieved to fulfil the necessary condition of the sliding mode controller, and the control law is then adjusted to ensure the system trajectory's reachability to the sliding manifold.

(Perruquetti & Barbot, 2002) wrote that to ensure that the sliding $\sigma(X) = 0$ is attained in finite time, $\frac{dv}{dt}$ must be more strongly bounded away from zero. As a result, if the attraction to the sliding mode disappears too rapidly, the attraction to the sliding mode will only be asymptotic as in Eq.2.4. To guarantee that the sliding mode is entered in finite time as in equation [0.0],

Where
$$\mu > 0$$
 and $0 < \alpha \le 1$ are constants. (2.4)

External disruptions and uncertainties are particularly susceptible to the sliding mode control during the reachability phase from (Utkin V.I, 2013) and (Lopez & Nouri, 2006) research. In order to get the best solution, it was necessary to choose the coefficients of the sliding function in the most efficient manner.

In (Romdhane et al., 2015) research paper, they presented the development of discrete second order sliding mode control (2-DSMC) for single-input single-output systems using an input-output model. In order to shorten the reachability phase of this 2-DSMC, the authors of this study suggest that the coefficients of the sliding function be synthesised using the Linear matrix Inequalities (LMI)'s technique. There is a comparison between the results acquired using LMI approach.

2.6.2 Robustness properties

In terms of robustness, Sliding Mode Controller (SMC) is recognised as a method that can withstand external disturbances while remaining insensitive to limited perturbation (Hung et al., 1993). Furthermore, SMC has been shown to have a rapid response time and a low control order from (Ha et al., 1999) past research.

It is possible to study uncertain systems of the following type in order to address robustness in Eq 2.5 and Eq.2.6.

$$y_{1}^{(n_{1})} = \varphi_{1}(\hat{\mathbf{y}}, \hat{\mathbf{u}}, t) + \Delta_{1}(\hat{\mathbf{y}}, t)$$

$$\vdots$$

$$y_{p}^{(n_{p})} = \varphi_{p}(\hat{\mathbf{y}}, \hat{\mathbf{u}}, t) + \Delta_{p}(\hat{\mathbf{y}}, t)$$
(2.5)

The uncertainties are Lebesgue measurable and fulfil the requirements.

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$$\|\Delta_{i}(\hat{\mathbf{y}}, t)\| \le \rho_{i} \| \hat{\mathbf{y}} \| + l_{i}, \rho_{i} \ge 0, l_{i} \ge 0, i = 1, \dots, p$$
(2.6)

The uncertainty may be caused by external uncertainties, internal parameter uncertainties, measurement noise, system identification error, and even the elimination technique that was used to construct a differential input-output model from a state space model, according to the author (van der Schaft, 1989).

(Sabanovic et al., 1996) experimented the algorithm's robustness on a direct-drive manipulator to ensure that it is reliable. The controller under consideration is a hybrid of variable structure systems and Lyapunov designs, and it exhibits all of the resilience qualities of sliding-mode systems while avoiding needless discontinuity in the control and, as a result, it shows minimising the chattering.

2.7 Performance Analysis

Performance Analysis is a specialised study that covers systematic observations to improve performance and decision-making. It is usually offered via the presentation of objective statistical and visual feedback.

2.7.1 Trajectory tracking

Controlling a device's trajectory is done by using trajectory tracking control. Numerous tracking control algorithms have been developed in an effort to better track defined trajectories, or to allow for more broad trajectories to be tracked more accurately.

In (Soleymani et al., 2017) research paper, the sliding mode control system is inadequate for dealing with mismatched uncertainty. Another downside of the sliding mode control approach that should not be overlooked is the chattering effect, which makes it difficult to execute on the real system owing to actuator response limitations. Thus, they used MSSC approach to deal with mismatched uncertainties on the simulation of the servo-pneumatic system in Simulink to obtain the trajectory tracking results. In order to avoid taking into account the dynamic features of the friction force, the friction force was modelled as a constant value uncertainty. These resulted in higher complexity in the controller design but reduced performance in the trajectory tracking.

When tested in the condition of uncertainties, the experimental findings demonstrate that the suggested controller does provide a satisfactory tracking performance. In the simulations, a sinusoidal signal is required as the trajectory for the system as in Eq.27 with parameter of, f = from 0.2 until 0.5 Hz.

$$\frac{x_d = 0.2 + 0.2 \sin (2\pi f t)}{\text{UNIVERSITI TEKNIKAL MALAYSIA MELAKA}}$$
(2.7)

As a result, Figure 2.14 and Figure 2.15 demonstrate that the suggested controller obtained a high level of performance in trajectory tracking. There's a considerable chattering effect when using a controller like this on a real servo-pneumatic system, thus it's impossible to anticipate results like this on the actual system.



Figure 2.14: Position tracking of sinusoidal trajectory with f=0.2Hz and 0.5 Hz of the SMC simulation (Soleymani et al., 2017)



Figure 2.15: Position tracking of sinusoidal trajectory with f=0.2 Hz and 0.5Hz of the actual system (Soleymani et al., 2017)

2.7.2 Stability of the SMC

The internal impedance of the actuator must be exactly zero for the actuator velocity output applied load to be completely stable. Due to inaccuracies in actuator modelling and inadequate output measurements, obtaining this condition is indeed very complex. However, by using adequate feedback from two outputs, force, and velocity, and by ensuring the presence of an energy dissipation element in the environment for system stability, the internal impedance of the actuator may be reduced significantly (Ali et al., 2009).

In their study, (Golo & Milosavljeviã, 2000) introduced a new control algorithm based on discrete-time variable structure control (VSC) theory. The algorithm's fundamental property is that trajectories reach the sliding manifold in a limited amount of time without causing chattering to the system.

In addition to stability, the algorithm's robustness with respect to parameter uncertainties as well as external disruptions is taken into consideration. The authors demonstrated that by shortening the sample duration, resilience might be increased. A DC servo position system was used to demonstrate the hypothesis. The understanding of the state vector x is required for the implementation of the proposed legislation. There are two modes of the control law. It is possible to reach the vicinity of the sliding hyperplane in a limited number of steps by using the first non-linear mode, which is implemented as a non-linear mode. It is possible to reach the sliding hyperplane in one step when using the second, linear mode, which is only available when there are no external disturbances or parameter uncertainties. The linear mode is created using the state feedback pole-placement approach. The robustness of the suggested method in the face of disruptions and parameter variations

was its most important characteristic. Furthermore, since a continuous function is in close proximity to the control law, the system will be free of chattering effects.

The study conducted by (Shtessel et al., 2011) focused on stability margins in SMC and the definition of phase margin for systems controlled by Second Order Sliding Mode, although the definition of gain margin suggested was not specified. Following the expansion of their study for the high-order sliding mode (HOSM) controller, they presented a method for obtaining stability margins for systems controlled by HOSM, namely the third-order Nested algorithm, for systems controlled by HOSM.

It has been discovered that SMC may be used to model nonlinear multi-input multioutput disturbed systems by (Esfahani et al., 2013) in their study. For the purpose of overcoming the chattering issue and ensuring accurate tracking of intended trajectories, the authors recommended combining an adaptive PD controller with the sliding mode as a solution. Based on the Lyapunov stability method, the researchers hypothesised that their proposed adaptive sliding mode control scheme could provide global stability as well as resilience of the closed loop system in the face of disruption.

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2.8 Summary

Previous research and literature related to pneumatic actuator modelling and control approaches have been reviewed. Following the researchers' investigation into the best approach to simulate the pneumatic actuator system, several of the strategies and methodologies were presented.

Initial modelling is done using the physical derivation approach, which is then refined. Following that, other types of contributions to pneumatic system modelling techniques were made by the following researchers, including in system identification method. Physical derivation and system identification methods are still among the most well-known methods to be used in the process of modelling, despite the advances in technology.

One concern in the design and use of control systems is the robustness of the controller to disturbance signals received from the input. Therefore, numerous sorts of controllers have been reviewed and compared in order to determine which is the optimal alternative in this research.

Aspects of the pneumatic actuator system that are often used include the utilisation of adaptive controllers, proportional integral derivatives (PIDs), and the sliding mode controller (SMC). According to the results of the performed literature study, research into these actuators surged in the 1990s because of the introduction and application of several control techniques into the system, including adaptive control, PID control, and sliding mode control. For easy understanding, Table 2.2 shows the strengths and weaknesses of the conventional and advanced controllers in this chapter.

By experimenting with the actual plant and Simulink simulation software, the pneumatic actuator system will be subjected to varying external loads in order to verify its robustness. The methodology, controller design, and analysis of the controller's performance will be discussed in Chapter 3.

Controller	Strength	Weakness	
PID	 Simple to construct Easy to tune the controller Simplicity of application 	• Lacks robustness and disturbance rejection.	
SMC	• Excellence in high robustness and disturbance rejection	 Causing chattering effect Exchanges between reducing chattering and tracking performance 	
SOSMC	 Excellence in high robustness and disturbance rejection. Reduction in chattering effect 	 Lacks in tuning method Complication in stability analysis Exchanges between reducing chattering and tracking performance 	
HOSMC	 ERSITI TEKNIKAL MALAN Excellence in high robustness and disturbance rejection Reduction in chattering effect Can be used in high nonlinear systems. 	 SIA MELAKA Lacks in tuning method Complication in stability analysis Exchanges between reducing chattering and tracking performance 	

Table 2.2: Strength and weakness of conventional and advanced controller

CHAPTER 3

METHODOLOGY

This chapter contains a list of the experimental setup, methodologies, and performance analysis that were used in this project. As shown in the preceding chapter's literature review, the strategy used was based on past research. In this chapter, modelling will be carried out in order to generate a mathematical model of a pneumatic actuator system, and system identification approach will be used. The setup for this experiment will be provided, as well as instructions on how to collect data. The technique for obtaining controller parameters from the system, as well as the theoretical analysis, will be covered in detail in this chapter.

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3.1 Introduction

The development of the project is summarised by a flowchart explaining the overall preparation and phases involved in the successful completion of this project. The previous final year project, which concentrated only on design development and numerical evaluation, was extended in this project. As a result of the physical relocation of the experimental setup and subsequent changes to the system transfer function affecting the accuracy and optimal design of previous controller parameters, a redesign process of the controllers was initiated prior to their implementation. The main tasks include the implementation of the built control algorithm into the experimental test setup, a sequence of functional work related to tracking movements with various types of input interference, and analyses of outcomes and tracking performances.

3.2 Flowchart



Figure 3.1: Flowchart of the methodology

3.3 Experimental setup

The pneumatic actuator positioning system shown in Figure 3.2 and Figure 3.3 is composed of the following components:

- 1. A windows operating running on personal computer (PC) with MATLAB R2021a.
- 2. LS-V15s proportional control valve, 5/3-way bi-directional, manufactured by Enfield.
- 3. A double acting cylinder with single piston rod.
- 4. Pressure sensors (Gems Sensors 1200SGG150223DA).



Figure 3.2: Experimental setup of pneumatic actuation system

3.3.1 System modelling for pneumatic system

A pneumatic actuator system is composed of the following components: a directional control valve, a Filter, Regulator and Lubricator (FRL) unit, actuator and an air compressor. A controller is used to regulate the placement of the actuator and the air compressor which has been shown in Figure 3.3. The primary function of an air compressor is to supply compressed air into a pneumatic actuation system under pressure.

Next, the pressured air is delivered to the Filter, Regulator and Lubricator (FRL) unit, where the dust in the air is filtered and to protect the system, the pressurised air is adjusted to meet the needs of the user, and the lubricant is injected into the pressurised air to prevent the actuator from rusting. As a result, the air will travel to the directional control valve, which has the function of controlling the direction of air flow in the pneumatic system after that.

Finally, the actuator is the component that genuinely performs functions in the pneumatic system. A controller is a device that is used to operate the overall system, ensuring that the movements of the actuator consistently follow the inputs. The usage of a suitable controller to handle a pneumatic actuator system is particularly crucial for efficiency in speed and precision.



Figure 3.3: Components of the lab apparatus

3.4 Mathematical Modelling

There are two methods for obtaining the system model in general: mathematical derivation and system identification. In this research, both strategies are applied to determine the parameter required for controller identification. The mechanical derivation is applied to explore the theoretical connection of the pneumatic actuator. The system identification is used to determine the parameters gathered from the real machine. To acquire parameters, the estimation model is employed at this phase.

3.4.1 System identification

After the system modelling phase, the pneumatic positioning actuator system is represented by a transfer function. As a result of this estimation strategy, the discrete time transfer function may be produced. A general form of transfer function may be represented as Eq. 3.1, which represents the plant.

$$G(s) = \frac{K_p}{a_n s^{n} + a_{n-1} s^{n-1} + \dots + a_1 s + 1} e^{-\theta s}$$
(3.1)

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Controller designation step is when the transfer function is put into action. This transfer function is used in the simulation of the system to get the best performance out of the controller before it is implemented in the hardware. Sliding mode controller is the controller of choice.

3.4.2 System Identification toolbox

The system modelling is done by applying system identification (SI) toolbox in application MATLAB Software. The SI toolbox is as indicated in Figure 3.4. The estimated

model of the system is obtained from the experiment performed on the pneumatic actuation system with multi frequency sine wave input.

The data collected is then exported to the SI toolbox. Export the collected data and select the range of data to be used. The transfer function models are chosen as the estimation function as shown in Figure 3.5.



Figure 3.5: Selecting Transfer function models

The program will prompt the estimate functions window which is the parameter for number of poles and zeros that need to be inserted as shown in Figure 3.6.

Estimate Transfer Functions		—		×
Model Structure Estimation Options				
Model name tf9				
Orders and Domain				
Number of poles 2				
Number of zeros 1				
Continuous time				
Continuous-time Discrete time (0.01 seconds)	Ecodtbrough			
Help		Esti	nate	Close
MALAYSIA				
Figure 3 6: Parameter for t	ransfer f	inc	tion	mode
rigure 5.6. I drameter for t	i unisi ci i	une	iion .	moue
2				

The left side of the models has been estimated using the transfer function model. The model output must be ticked in order to get the best fit of measured and simulated model output. The best fit must be over 85% as it is required approximation to the real system as shown in Figure 3.7. The transfer function can be obtaining by double click the selected output model and it will prompt the data/model info as shown in Figure 3.8.



Figure 3.7: Model output and best fit of the estimated model



The SMC controller was created with the help of the MATLAB Simulink software's block diagrams. The block diagrams are constructed using the mathematical integration approach and then simulated based on the parameters specified in the block diagrams' specifications. Simulink provides a graphical supervisor for creating models in the form of block diagrams, allowing models to be drawn with a pencil and paper. Simulink also includes a comprehensive library of sink, source, linear and nonlinear components, as well as connector blocks and connector blocks. When using the intuitive condition, you may reorganise your modelling technique and eliminate the requirement to figure out differential and contrast conditions in a language or programme.

3.5

Simulink has the ability to see the system at a high level and then drill down to examine increasing degrees of model information as the system evolves. This technique provides insight into how a model is put together and how its many elements work together. The SMC simulation will be constructed with the help of the MATLAB Simulink programme.

3.6 SMC Controller Design

Sliding mode control (SMC) is one of the most well-known and powerful design tools for the issue of deterministic control of uncertain systems, since it has a substantial insensitivity to internal parameter fluctuations and external disturbances, making it an excellent choice for this application. As a result, the controller will be developed. In Figure 3.4 will be shown for block diagram of the Sliding Mode Controller.



Figure 3.9: Block diagram of the general control scheme of SMC

In the design of a sliding mode controller, it is necessary to establish an acceptable sliding surface in the state space in order to achieve a basic sliding mode control design. There are two major components to examine, which is the switching function and the control laws.

3.6.1 Switching Function of SMC

Regarding a given sliding surface, s (t), switching function is a function that is a function of the tracking error, e(t), and the time derivative of e(t). The following is the general formula, as seen in Eq. 3.2 and Eq. 3.3:

$$s(t) = \left(\lambda + \frac{d}{dt}\right)^{(n-1)} e; \qquad (3.2)$$

where the desired position is denoted by r(t), the actual position is denoted by y(t), and λ is a positive constant while *n* denotes the order of an uncontrolled system.

$$e(t) = r(t) - y(t)$$
 (3.3)

the extension of Eq.3.4 and Eq 3.5, which describe the derivative of the sliding surface, resulting in Eq.3.6, which is a final representation of the sliding surface.

$$\dot{s}(t) = \lambda \dot{e}(t) + \ddot{e}(t) \tag{3.4}$$
$$\ddot{e}(t) = \ddot{v}(t) - \ddot{r}(t) \tag{3.5}$$

$$e(t) = y(t) - r(t)$$
 (3.5)

$$\dot{s}(t) = \lambda \dot{e}(t) + \ddot{y}(t) - \ddot{r}(t) \tag{3.6}$$

3.6.2 Control Laws of SMC

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The control laws are the second component of SMC. According to (Rafan et al., 2012) Control laws are made up of equivalent control, u_{eq} , it is composed of a signum function and an equivalent control that incorporates acceleration and velocity feedforward as illustrated in Eq. (3.7)

$$u(t) = u_{\text{equivalent}} - K \cdot sign(s)$$
(3.7)

where *K* denotes a positive constant or gain value and sign(s) represents the signum function which has a piecewise function as in Eq (3.8) below and ideal slide motion is shown Figure 3.10.

$$\operatorname{sign}(s) = \begin{cases} 1, \ s > 0 \\ 0, \ s = 0 \\ -1, \ s < 0 \end{cases}$$
(3.8)



Figure 3.10: Ideal sliding motion (Rafan et al., 2012)

Switching function and control laws are combined in the overall design. When the switching function reaches the sliding surface and begins to slide, it is equivalent to zero and achieves stability, as seen in Eq. (3.9) and Eq. (3.10).

$$s(e, \dot{e}) = \dot{e} + \lambda e = 0$$

$$\dot{s}(\dot{e}, \ddot{e}) = \ddot{e} + \lambda \dot{e} = 0$$
(3.9)
(3.10)

The equivalent control maintains the sliding motion, while the proportional gain K assures states are attracted to and remain on the sliding surface for a finite period of time. This discontinuity results in high frequency oscillation, which causes an unwanted chattering occurrence that may cause damage to the machine tools' drive systems.

3.7 Parameter for Sliding Mode Controller

The parameter setting is used by experimenting with the actual plant setup to get the maximum tracking error of SMC by using Simulink simulation software. The analysis of the robust tracking is based on 0.1 Hertz of sinusoidal wave input with an amplitude of 60 mm. Furthermore, a variety of external loads ranging from 0 kg to 9.0 kg with an incremental of 0 kg will be applied to the actual plant setup in order to determine their robustness to the designed SMC controller.

3.8 Performance Analysis

The SMC will be implemented on the experiment setup and collect the data for analysis by the plant controller. The performance analysis will be recorded using maximum tracking error and root mean square error (RMSE) and, for stability of the plant controller, will be using Gain Margin (GM).

3.8.1 Robust Tracking Error.

The tracking performance is measured by the error of the actuator against the input. The errors are derived by the maximum tracking error obtained, and the RMSE.

The robustness of the tracking will be determined by the maximum tracking error attained and the RMSE, with additional external load disturbances, as explained in the study scope. Table 3.1 below shows the robust tracking error data analysis from different external loads applied to the actuator of the pneumatic system.



Figure 3.11: Example of tracking performance between desired and Measured

External loads (kg)	Tracking data		
	Maximum Tracking Error	RMSE	
0.0			
 (incremental)			
9.0			

Table 3.1: Robust tracking Error data analysis

3.8.2 Stability of the plant controller

When determining the stability of a control system, a Bode plot is a graph that is often used in the field of control system engineering. The stability of SMC controller on the plant will be measured using the Gain Margin (GM) approach.

Stability margins are a sort of performance criteria that demonstrates how stable a system is. The gain margin is a stability indicator that shows the amount of gain that the system could inject before it becomes unstable and the expected result is when the gain margin is higher, the systems' stability is also increased.



Figure 3.12: An Example of a bode magnitude and phase plot set.

3.9 Summary

This chapter contains all the necessary information about the machine, the software, the method, and the process for tracking the error using the SMC. Thus, Table 3.2 shows the relation between objective, methodology and expected results that have been done in this chapter. Furthermore, data analysis demonstrates how to analyse and validate the data analysis by referring to previous outcomes. The results of tracking error using SMC and stability of the plant controller will be discussed in the next chapter, and the results will be discussed in the following chapter.

Objective	Methodology	Expected result	
Objective 1:	• System	• Obtaining transfer	
To determine the system model of a	identification	function with best	
pneumatic system using system	toolbox using	fit more than 85%	
identification technique.	MATLAB	approximation to the	
State of the second	software	real plants.	
Objective 2:		• The designed	
To design SMC controller to obtain	• Controller	controller can	
robust tracking of pneumatic system	design	successfully run in	
UNIVERSITI TEKNI	KAL MALAYSIA N	the Simulink	
		simulation and on a	
		real plant.	
Objective 3:		Able to analyse and	
To evaluate the controller	• Robust	validate the result data	
performance in terms of maximum	Tracking Error	with,	
tracking error and root mean square	• Stability of the	• Maximum Tracking	
error.	controller plant	Error	
		• RMSE	
Objective 4:		Gain Margin	
To evaluate the SMC controller's			
robustness by applying additional			
loads to the servo pneumatic system			

Table 3.2: Summary table between objectives, methodology and expected result

CHAPTER 4 CONTROLLER DESIGN

4.1 Introduction

The wide controller structure guaranteed that any mechatronic control framework worked well. This section was dedicated exclusively to the development of three controllers, namely, the PID controller, the SMC controller, and the Pseudo-SMC controller. 3 controllers were made to make a comparison of which controllers had the best tracking and robust performance when applying external load to the pneumatic system. Obtaining the transfer function and the input signal for simulation and experimental tests is presented in the first part. The second section explains how to convert pneumatic system units for tracking measurements.

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4.2 Transfer Function

A good controller design can result in a good system tracking performance. In this project, System Identification generates transfer functions based on the configuration of the actual plant to validate the controller's model and presents the model's percentage of best fit. The best fit achieved is 88.78%. It is accomplished by obtaining a best of more than 85% from the System Identification Toolbox with multiple sinewaves applied to the actual plant. Section 3.4 describes the procedure for obtaining the transfer function. Equation 4.1 shows the estimated transfer function of third order:

$$G(s) = \frac{0.2882s^2 + 0.5213s + 0.002931}{s^3 + 0.2639s^2 + 0.01152s + 0.00008487}$$
(4.1)

4.2.1 Transfer function reduction

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The transfer function in Equation 4.1 was a nonlinear third order transfer function with no good analytic solution, which makes simulation of the SMC controller, which is also a nonlinear controller, very time consuming to gain simulation results. As a result, the order of Eq. 4.1 is reduced and converted into the second order using the Minreal method. Minreal stands for minimal realisation or Pole-zero cancellation, and it is used to eliminate uncontrollable in state-space models in transfer functions. In Eq. 4.2, the reduced second order transfer function is used as the transfer function for simulating the plant for PID, SMC, and Pseudo-SMC controllers.

$$G(s) = \frac{0.2882s + 0.5197}{s^2 + 0.2546s + 0.0092}$$
(4.2)

The Eq. 4.1 and Eq. 4.2 has the same outcome with any frequency applied to the system. It does not change the validation of both of the transfer function when under PID simulation shown in Figure 4.1.



Figure 4.1: Comparison of third order and second order transfer function with 10 Hz

The third order and second order of transfer functions were overlayed with each other with no changes to the transfer function when under any input frequency. Thus, it is proved that the output system has minimal order and the same response characteristics as the third order transfer function system.

4.2.2 Stability of the transfer function

The stability of the transfer function is achieved by using gain margin stability with the bode diagram shown in Figure 4.2. The gain margin has an infinity value, which indicates that it is always stable with a phase margin of 41.6° at 0.06 Hertz.



Figure 4.2: Gain margin stability transfer function

4.3 Input signal

In this section, the use of an input signal for three controllers; PID, SMC, and Pseudo-SMC; was the same as for the comparison in the result section. A sine waveform generated from the signal generator in Simulink was used to test the SMC controller's tracking performance. In this control system, the input was tracked by observing the output to produce a better tracking accuracy. The sine waveform consisted of two parameters: amplitude and frequency. The amplitude used was 60 mm, which related to the distance of the axis movement of the pneumatic actuator. The frequency of 0.01 Hertz was used and is shown in

Figure 4.3. Furthermore, this signal was used during simulation and experimental work. The time taken for the tracking performance evaluation is 50 seconds. The details of the input signal are tabulated in Table 4.1.



4.4 Unit conversion

The unit and its conversion were addressed in this section. The pneumatic system's input is designed in voltage units, while tracking of the system is evaluated in displacement units of millimetres (mm). The experimental plant's hardware setup should consist of a positioning sensor or encoder to measure tracking analysis instead of a pressure sensor. Due to project constraints, modifications to the hardware setup components are not allowed. As a result, the unit conversion was accomplished using manual derivatives. The millimetre unit was chosen for this project because the system's displacement range is small. For the unit conversion process, the pneumatic actuator specifications shown in Table 4.2 of the system were used. The specification sheet of the pneumatic actuator was attached in Appendix A.

Table 4.2: Specification of the Pneumatic Actuator

Specification of the Pneumatic Actuator			
Total voltage of the actuator	10V		
Total distance of the actuator	304.8 mm		

As a result, the specifications were used to generate a system conversion equation. From Equation 4.1 to Equation 4.3, the unit conversion from voltage (V) to millimetres (mm) was done as a series of continuous equations where \varkappa is the voltage value.

$$10 \text{ V} = 304.8 \text{ mm}$$
 (4.1)

$$1 V = \frac{304.8 mm}{10 V}$$
(4.2)

$$\kappa V = \kappa V \left(\frac{304.8 \, mm}{10 \, V}\right) \tag{4.3}$$

Including the unit conversion, the actuator's positioning was adjusted so that the centre of the actuator is considered the system's origin point, as shown in Equation 4.4 below.

$$\kappa V = \kappa V \left(\frac{304.8 \, mm}{10 \, V}\right) - \left(\frac{304.8 \, mm}{2}\right) \tag{4.4}$$

Starting from the 20th second of the system response, the initial voltage is 5V and the final voltage is 7V. The unit conversion of initial and final voltage to millimetres after the adjustment is shown in Eq. 4.5 and Eq. 4.6., respectively. Thus, the amplitude of the input signal is measured to be 60 mm.

Initial Voltage:

$$5V\left(\frac{304.8\ mm}{10\ V}\right) - \left(\frac{304.8\ mm}{2}\right) = 0\ mm \tag{4.5}$$

Final Voltage:

$$7V\left(\frac{304.8\,mm}{10\,V}\right) - \left(\frac{304.8\,mm}{2}\right) = 60\,mm \tag{4.6}$$

4.5 PID controller design

The proportional-integral-derivative (PID) controller is a type of feedback mechanism used in industrial control systems. The system error is calculated by the PID controller as the difference between the desired value and the measured output. The controller tends to minimise errors by using a feedback loop to correct for the difference between the desired set point and the measured output. The PID controller is mathematically described in Eq. 4.7.

$$G(s) = K_p + \frac{K_i}{s} + K_d s \tag{4.7}$$

The controller is designed based on traditional open loop shaping followed by closed loop tuning. The Simulink block diagram of the PID controller is shown in Figure 4.4.





Figure 4.4: Simulink block diagram of PID controller

4.5.1 Tuning parameters PID controller

The simulation is done with the compensation model. The system performance condition has been stimulated to obtain the response behaviour in terms of output voltage and tracking error based on previous studies. The tuning parameter of PID was obtained as per the flowchart shown in Figure 4.5. The try and error method was used to obtain the tuning parameter in terms of max tracking error and root mean square error (RMSE) value until there were no changes to the output response. The values of Kp, Ki, and Kd, with max tracking error and RMSE, have been tabulated in Table 4.3. The highlighted table will be the tuning parameter value of the PID controller.

Trial Run	Kp	Ki	K _d	Maximum tracking error (mm)	RMSE
北	o (jun	J.0.10	0.10	یر ⁴⁰ .77 کی تیا	50.26
U2IIV	EIOIT	0.15	KA 0.10 AL	AYS6.48MEL	A 3.82
3	10	0.15	6.50	6.48	3.70

Table 4.3: Tuning Parameters of the PID controller



Figure 4.5: Flowchart of the tuning parameters of the PID controller

4.6 SMC controller design

SMC is a nonlinear control that is well known for its robustness and disturbance rejection. The switching function and control laws are the two main components considered in basic SMC design. The switching function is a function with respect to the corresponding sliding surface, s, and is determined by the tracking error, e(t), and its time. A low pass filter, which is a filter that passes signals with frequencies lower than the cut off frequency, is known to be used in the system.

Both switching function and control laws have been combined to design the SMC controller. (Heng et al., 2017) stated that, the equivalent control, $u_{eq}(t)$ input signal is obtained by converting into differential equation in Eq. 48 by combining transfer function in Eq. 4.2 and derivative switch control Eq. 3.6.

$$u_{eq}(t) = \frac{(0.2546\dot{y} + 0.0092y + \ddot{r} \cdot \lambda \dot{e})}{0.2882}$$
(4.8)

The Simulink block diagram of the SMC controller is shown in Figure 4.6, where the red box marks the control laws, and the blue box marks the equivalent control. The light blue-coloured box is for tuning the parameters of the SMC controller



Figure 4.6: Simulink block diagram of SMC controller

4.6.1 Tuning parameters SMC controller

The tuning parameter of SMC was obtained as shown in the flowchart shown in Figure 4.7. The try and error method was used to obtain the tuning parameter in terms of max tracking error and root mean square error (RMSE) value until there were no changes to the output response. The value of K gain and lambda, λ with max tracking error and RMSE, has been tabulated in Table 4.4. The highlighted table will be the tuning parameter value of the SMC controller.

	Trial Run	k gain	lambda, λ	Maximum tracking error	RMSE
14	MALA	YSIA MELY		(mm)	
A TEKA	1	10	1.0	202.29	116.20
10	2,2	200	5.0	7.88	5.49
5	با دار	200	<u>_9.5</u> .	4.32	ويبوارج س

 Table 4.4: Tuning Parameters of the SMC controller

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Figure 4.7: Flowchart of the tuning parameters of the SMC controller
4.7 Pseudo-SMC controller design

The design of the Pseudo-SMC controller has the same SMC controller's switching function and equivalent control. In previous research by (Rafan et al., 2012), pseudo-SMC is an extension of the SMC controller which has an alternative signum function, which is a continuous sigmoid like function in Eq. (4.9). The continuous signum function reduced the chattering effect by suppressing high frequency oscillations.

$$V_{\delta}(s(t)) = \frac{s(t)}{|s(t)| + \delta} \tag{4.9}$$

Where delta, δ is a positive constant that represents the degree of continuous approximation as shown in Figure 4.8 below. The control laws of Pseudo-SMC are shown, in Figure 4.9.



Figure 4.9: Pseudo-SMC control laws

The tuning parameter of SMC was obtained as per the flowchart shown in Figure 4.10. The Simulink block diagram of the Pseudo-SMC controller is shown in Figure 4.11, where the red box marks the control laws, and the blue box marks the equivalent control. The light, blue-coloured box is for tuning the parameters of the Pseudo-SMC controller.



Figure 4.10: Simulink block diagram of Pseudo-SMC controller

4.7.1 Tuning parameters Pseudo-SMC controller

The tuning parameter of Pseudo-SMC was obtained as shown in the flowchart shown in Figure 4.10. Try and error method was used to have the tuning parameter in terms of max tracking error and root mean square error (RMSE) value until there are no changes to the output response. The value of K gain, lambda, λ and delta, $\underline{\delta}$ with max tracking error and RMSE has been tabulated in Table 4.5. The highlighted table will be the tuning parameter value of the Pseudo-SMC controller.

Trial	K	Lambda,	Delta, δ	Max tracking error	RMSE
Run	gain	λ		(mm)	
1	250	10	10	6.87	3.36
2	500	50	50	5.09	3.38
3 1	500	50	90	3.33	2.26
12.	evaning		μ		
6	با ملال	ل مليسب	کنیک	ينومرسيتي تيد	او
111	NIVER	SITI TEK		ALAYSIA MELAK	Δ

Table 4.5: Tuning Parameters of the Pseudo-SMC controller



Figure 4.11: Flowchart of the tuning parameters of the Pseudo-SMC controller

4.8 Summary

In summary, Chapter 4 describes the procedure for designing the proposed controller, an SMC controller with an estimated transfer function of the nonlinear pneumatic system. The estimated transfer function order was reduced from third order to second order for simulating the controller using the Minreal approach. The controllers were designed for the single axis of a servo-pneumatic system to compare the robust tracking performances among three controllers, namely, the PID controller, the SMC controller, and the Pseudo-SMC controller in Simulink software. The tuning parameter values of the three controllers were determined using the try and error method, in terms of maximum tracking error and RMSE. The results and discussion of the simulation and experimental work are discussed in the next chapter



CHAPTER 5

RESULT AND DISCUSSION

5.1 Introduction

Three different controllers for position controllers were designed and analysed, namely, the PID controller, the SMC controller, and the pseudo-SMC controller. This chapter presents simulation and experimental results relating to the control performances of these controllers on a single axis of a servo-pneumatic system as an experimental plant. The performance of three controllers was compared based on the maximum tracking error and root mean square (RMSE) analysis. Besides that, in terms of robustness against load variation, the system was exposed to an incremental 1 kg of external load from 0 kg to 9 kg whereby the tracking performances were measured and evaluated. Finally, a discussion on; i) maximum tracking error, ii) RMSE, iii) robustness against load variation and iv) chattering effect between SMC controller and Pseudo-SMC controller was discussed in this chapter.

5.2 Result of the controllers

This section presents the result of three controllers; PID controller, SMC controller and Pseudo-SMC controller based on the Maximum tracking error and RMSE. The maximum tracking error and RMSE results for PID, SMC and Pseudo-SMC are represented in section 5.2.1, section 5.2.2 and section 5.2.3, respectively.

5.2.1 PID controller

The PID controller was evaluated using sine wave input signals with an amplitude of 60 mm and a frequency of 0.1 Hz.

5.2.1.1 Simulation results

The simulation of the PID controller was performed using MATLAB/Simulink software. Figure 5.1.1 shows the maximum tracking of PID with reference input and corresponding output. Table 5.1.1 tabulates the maximum tracking errors and RMSE values obtained.





Max tracking error (mm)	6.48
RMSE value	3.82

Table 5.1.1: PID controller simulation of maximum tracking error and RMSE.

5.2.1.2 Experimental Result

The experiments of the PID controller were performed using MATLAB/Simulink software. An incremental of 1 kg from 0 kg to 9 kg of external load was applied to the experimental plant. Figure 5.1.2 shows the maximum tracking error of PID with an external load of 0 kg and 1 kg. Table 5.1.2 shows the maximum tracking errors and RMSE values for the external load from 0 kg to 9 kg.



Figure 5.1.2: Experimental maximum tracking errors of the PID controller

	External	Tracking data				
	Load (kg)	Maximum tracking error (mm)	RMSE value			
	0	41.42	14.54			
	1	46.12	15.30			
	2					
	3					
	4					
	5	Unsta	ble State			
	6					
2	7					
7	8	A.K.				
	9					
10.00 M						

Table 5.1.2: PID controller maximum tracking error and RMSE value with

The PID controller only managed to have a maximum tracking error and RMSE with an applied load of 0 kg and 1 kg. The rest of the load variation cannot be continued as it deteriorates when applying more than 1 kg of external load to the experimental plant. Figure 5.1.3 is shown below as applying the external load of 2 kg to the actual plant is invalid and causes too much vibration to the actual plant.



Table 5.1.3: PID controller maximum tracking error with 2 kg external load

5.2.2 SMC controller

The SMC controller was evaluated using sine wave input signals with an amplitude of 60 mm and a frequency of 0.1 Hz.

5.2.2.1 Simulation result

The simulation of the SMC controller was performed using MATLAB/Simulink software. Figure 5.2.1 shows the maximum tracking of SMC with reference input and corresponding output. Table 5.2.1 tabulates the maximum tracking errors and RMSE values obtained.



Figure 5.2.1: SMC controller simulation of reference input and corresponding max tracking

Table 5.2.1: SMC	controller	simulation	of Maxin	num tracking	error and	RMSE.

Max tracking error (mm)	4.32
RMSE value	3.11

5.2.2.2 Experimental Result

The experiments for the SMC controller were performed using MATLAB/Simulink software. An incremental of 1 kg from 0 kg to 9 kg of external load was applied to the experimental plant. Figure 5.2.2 shows the maximum tracking error of SMC with an external load of 0 kg and 9 kg. Table 5.2.2 tabulates the external load from 0 kg to 9 kg in terms of the maximum tracking errors and RMSE values obtained.



Figure 5.2.2: Experimental maximum tracking errors of SMC controller

	External	Track	ting data
	Load (kg)	Maximum tracking error (mm)	RMSE value
	0	7.78	2.91
	1	5.86	2.80
	2	8.47	2.95
	3	11.34	3.20
	4 MALAYSI	10.15	2.99
EKAI	5	8.66	2.96
150	6	14.09	3.21
4	7	5.74	2.81
1	سب مالار	9.93	وييومر شيتي بيھ
U	9	1 TEKNIKAL MA 11.47	LAYSIA MELAKA 2.94

Table 5.2.2: SMC controller maximum tracking error and RMSE value with external

5.2.3 Pseudo-SMC controller

The Pseudo-SMC controller was evaluated using sine wave input signals with an amplitude of 60 mm and a frequency of 0.1 Hz.

5.2.3.1 Simulation result

The simulation of the Pseudo-SMC controller was performed using MATLAB/Simulink software. Figure 5.3.1 shows the max tracking of Pseudo-SMC with reference input and corresponding output. Table 5.3.1 tabulates the maximum tracking errors and RMSE values obtained.



Figure 5.3.1: Pseudo-SMC controller simulation of reference input and corresponding max tracking

Table 5.3.1 Pseudo-SMC controller simulation of Maximum tracking error and RMSE.

Max tracking error (mm)	4.32
RMSE value	3.11

5.2.3.2 Experimental Result

The experiments of the Pseudo-SMC controller were performed using MATLAB/Simulink software. An incremental of 1 kg from 0 kg to 9 kg of external load was applied to the experimental plant. Figure 5.3.2 shows the maximum tracking error of Pseudo-SMC with an external load of 0 kg, 5 kg, and 9 kg. Table 5.3.2 tabulates the external load from 0 kg to 9 kg in terms of the maximum tracking errors and RMSE values obtained.



Figure 5.3.2: Experimental maximum tracking errors of Pseudo-SMC controller

	External Load (kg)	Tracking data				
		Maximum tracking error (mm)	RMSE value			
	0	8.41	2.04			
	1	5.31	1.49			
	2	4.50	1.59			
10	ATSI	4.53	1.62			
TEKW	4	4.70	1.67			
1810	5	5.59	1.84			
5	6 milak	5.59 کنیک ملہ	ويتوريسية بند			
01	8	7.88	2.14			
	9	7.37	1.95			

Table 5.3.2: Pseudo-SMC controller max tracking error and RMSE value with external load

5.3 Discussion on Control Performances

This section compares and discusses the control performances of the PID controller, SMC controller, and Pseudo-SMC controller. Comparisons are made by analysing the control command input signal.were made in terms of (i) Max tracking error, (ii) Root Mean Square Error, (iii) Robustness against load variation, and (iv) Chattering effect.

The PID controller was defined as the reference controller against the SMC-based controllers, namely the SMC controller and the Pseudo-SMC controller, for the purposes of analysing and evaluating the robust tracking performances of three controllers.

5.3.1 Maximum tracking error

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Figure 5.4 compares tracking errors of the three controllers at a tracking frequency of 0.1 Hz with an applied external load of 1 kg to the experimental plant. Table 5.4 compares the maximum tracking errors of the different controllers with an applied external load of 1 kg.



Figure 5.4: Measured max tracking errors of the experimental PID controller, SMC controller, and Pseudo-SMC controller with an external load of 1 kg

Controllers	Max tracking	Improvement (%)
	error (mm)	
PID	46.12	Reference
SMC	5.86	87.29
Pseudo-SMC	5.31	88.48

Table 5.4: Maximum tracking errors of three controllers with external load of 1 kg

A general observation of the results presented shows that the SMC-based controllers were able to significantly reduce tracking errors compared to those achieved using PID controllers.

The SMC-based controllers, namely, the SMC controller and the Pseudo-SMC controller, produced on average an improvement of 87.0% in maximum tracking errors. The more advanced SMC controllers possess elements of equivalent control structure, resulting in tighter control, thus producing better tracking performance. The equivalent control contains elements of velocity and acceleration feedforward that contributed positively towards reducing tracking errors as errors associated with velocity and acceleration of the system were compensated for.

Among the SMC-based controllers, the SMC controller and Pseudo-SMC controller produced tracking errors that were comparable to one another. Thus, in terms of tracking error measurement, there is little difference between an SMC-based controller and a conventional controller. Upon closer observation in Table 5.4, the Pseudo SMC controller produced the smallest maximum tracking error compared to the SMC controller, with an average improvement of 1.2%.

5.3.2 Root mean Square Error (RMSE)

Root mean square error (RMSE) is defined as the square root of the average of the square of all errors. It is also known as the quadratic mean, which refers to a measure of the difference in magnitude between the desired and actual point at varying input. It is common practise to calculate RMSE if a sinusoidal based input signal is applied, which was researched by (Jamaludin et al., 2007). In this section, the RMSE was measured on three controllers: the PID controller, the SMC controller, and the Pseudo-SMC controller. The RMSE was defined as the square root of the average squared tracking error and was expressed in Eq (5.1.)

$$RMSE = \sqrt{\frac{\Sigma(e)^2}{n}}$$
(5.1)

Where, *e* is tracking error and *n* is the number of tracking error data

In addition, Figure 5.5 and Table 5.5 summarised the RMSE values of tracking error with an external load of 0 kg and 1 kg.



Figure 5.5: RMSE values for different controllers with applied external load of 1 kg

	100					
External	AINO	RMSE			Error Reduction	(%)
load (kg)	PID	SMC	Pseudo-	PID	PID	SMC
			SMC	versus	versus	versus
UI	VIVERS	ITI TEKI	NIKAL	SMC	Pseudo-SMC	Pseudo-SMC
0	14.54	2.91	2.04	79.98	85.96	29.89
1	15.30	2.80	1.49	81.70	90.26	46.78

Table 5.5: Comparisons in RMSE values of three controllers with external load of 1 kg

The RMSE results with an applied load of 0 kg and 1 kg for the Pseudo-SMC controller produced the smallest RMSE values when compared to the PID controller and SMC controller, as shown in Figure 5.5 and Table 5.5. At 1 kg, the PID controller had the highest RMSE value. The percent error reduction of the RMSE values for the three controllers was also included in Table 5.5. Eq. shows how to calculate the percent error reduction (5.3).

Error reduction (%) =
$$\frac{Controller A - Controller B}{Controller A} \times 100\%$$
 (5.3)

The percentage error reduction was used to compare the error reduction rates of the two controllers. As a result, the error reduction percentages with external load between PID controller and SMC controller were 79.98% at 0 kg and 81.70% at 1kg load. Furthermore, the error reduction percentages with external load between the PID controller and the Pseudo-SMC controller were 85.96% at 0 kg and 90.26% at 1 kg load. Finally, the error reduction percentages with external load for SMC controller and Pseudo-SMC controller were 29.89% and 46.78%, respectively. When an external load of 1 kg was applied to the experimental plant, Pseudo-SMC achieved the lowest RMSE values with percentage error reductions of 90% and 46% against PID controller and SMC-controller, respectively.

5.3.3 Robustness against load variation

This section explores the robustness of SMC-based controllers to variations in system dynamics. Robustness in machine tool systems and operations plays an important role. The system transfer function shows how the dynamics of the system change over time. This is because each part of the system wears out over time.

The control performances of the SMC-based controllers were evaluated as the external load attached to the experimental plant's single axis slider was varied. The results were tabulated and analysed to study the effects of load variations of 0 kg, 4 kg, and 9 kg on the control performances of the SMC-based controllers, as shown in Tables 5.6. A small difference between the RMSE values at different load variations would indicate the robustness of the system against variation in the system dynamics. The percentage of variation was calculated as shown in Eq. (5.4). Figure 5.6 also shows the RMSE value of the SMC-based controllers with an external load from 0 kg to 9 kg.

Percentage of variation (%) =
$$\frac{\text{load variation A-load variation B}}{\text{load variation A}} \times 100\%$$
 (5.4)



Figure 5.6: RMSE values of SMC-based controllers

with external load of 0 kg until 9 kg

Table 5.6: RMSE values of SMC-based controllers against load variation

Controllers			RMSE value	es	
20	0 kg	4 kg	Percentage of	9 kg	Percentage of
ch	i (variation (%)		variation (%)
SMC	2.91	2.99	2.75	2.94	1.03
Pseudo SMC	2.04	1.67		1.95	4.41

According to Table 5.5, Pseudo-SMC produced the highest percentage variation in RMSE values as the load attached to the actual plant was varied. The RMSE value for Pseudo-SMC controllers drastically changed from 18.13% to 4.4% when a load is two times the applied weight of the external load. This supports the work of (Tiwari et al., 2016) who concluded that the robustness of Pseudo-SMC against load dynamics variation was compromised as a compromise to the reduction in the degree of chattering.

In contrast, a smaller percentage difference in RMSE values was observed for the SMC controller. The smallest percentage variation that was recorded was 1.03%. This proved that the SMC controller possessed high robustness against system dynamics variation as the load attached to the actual plant was varied

5.3.4 Chattering effect in SMC-based controllers

In comparison to the SMC controller, the Pseudo-SMC controller has a continuous signum function in the control laws shown in Figures 4.9 and 4.10. Noises were observed in each of the SMC controller and Pseudo SMC controller input signals, as shown in Figure 5.7.



SMC Controller

The presence of noise was particularly noticeable in the regions corresponding to zero velocity motions of SMC-based controllers. In the case of the Pseudo-SMC controller, reduced noise effect and induced chattering effect are observed by suppressing high oscillation frequency by the signum function in control laws.

To evaluate the chattering effect of SMC-based controllers, the chattering is analysed using spectral analysis of the measure control command signal to the drives system. Figures 5.8 and 5.9 compare the Fast Fourier Transform (FFT) error on the error signal of the SMC controller and the Pseudo-SMC controller with an external load of 0 kg applied to the experimental plant. The FFT error was written in MATLAB syntax as follows.

```
Time = pseudosmc_error_0kg(:,1); %Line 1
Error = pseudosmc_error_0kg(:,2); %Line 2
No_of_Errordata = 5001; %Line 3
frequencyPI = (0:1/Time(No_of_Errordata):100)'; %Line 4
fft_errorPI = abs (fft(Error)); %Line 5
figure;plot(frequencyPI,fft errorPI*2/No of Errordata);%Line 6
```

Line 1 specify the time data gathered from the system. Line 2 declared the tracking error data that the system had collected. Line 3 displayed the total number of data errors collected. Line 4 collected the frequency content of the FTT error. The error data was converted from time domain to frequency domain using Line 5. Finally, Line 6 plotted the FFT error.

Table 5.7 is tabulated the result of Figure 5.8, one significant peak was observed at frequency of 0.1 Hz and these peaks implied the presence of chattering as oscillation at high frequency (Utkin & Hoon Lee, 2007.)

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Figure 5.8: RMSE values for SMC-based controllers with external loads ranging from 0 to 9 kg.

Table 5.7:	Maximum amplitude of FFT error for SMC-based controlle					
ا ملاك	Controllers	External load of 0 kg				
UNIVEF	SITI TEKN	FFT error KAL (mm)	Improvement (%)	A		
	SMC	3.71	Reference			
	Pseudo-SMC	0.86	76.82			

ers

Figure 5.8 depicts the frequency components of the tracking error signals as dashed circles. It is concluded that the decrease in amplitude of the FFT error represents chattering attenuation. The Pseudo-SMC controller with SMC controller as reference for FFT errors measurement produced the most significant reduction in amplitude of the FFT error peak values. Peak amplitudes were reduced by 76.82% with an applied load of 0 kg, where the SMC controller FFT error is 3.71 mm and the Pseudo-SMC controller FFT error is 0.86 mm at 0.1 Hz. This result was consistent with the literature, (Heng et al., 2017) stated that Pseudo-SMC was able to reduce chattering to some extent due to the presence of continuous control action in its control algorithm.

5.4 Summary

This chapter presents and discusses experimental results relating to validations on control performances of three controllers, namely the PID controller, the SMC controller, and the Pseudo-SMC Controller. The control performance of these controllers was validated on the single axis direct servo-driven test setup. Control performance was compared across controllers in terms of maximum tracking error, RMSE values, robustness against load variation, and chattering effect.

In general, the results presented showed the best robust tracking of an SMC-based controller against a classical PID controller. SMC-based controllers were able to reduce tracking errors by up to 88% when compared to PID controllers. This shows the great potential of SMC-based controllers as a replacement for any PID-based controller. As the controller of choice for machine tool devices, Among SMC-based controllers, Pseudo SMC produces the best overall control performance in terms of maximum tracking error and RMSE values. In contrast, SMC controllers show a comparable performance with Pseudo-SMC with slightly better robustness against load variation, with the lowest percentage load variation of 1% when applying 9 kg of external load. In terms of chattering effects between the SMC-based controllers, it is proven Pseudo-SMC achieved a reduction of 76% with FFT error against the SMC-controller. Pseudo-SMC was able to reduce chattering to some extent due to the presence of continuous control and (Utkin & Hoon Lee, 2007.) researched the existence of chattering in SMC controllers.

Figure 5.9 and Figure 5.10 compare control performances among three controllers: PID controller, SMC controller, and Pseudo-SMC controller, with external load applied from 0 kg to 9 kg. The results are summarised in Table 5.8. The controllers were ranked from 1 to 4. Rank 1 represents the best robust tracking performance, while Rank 3 represents the worst performance.



Figure 5.9: Maximum tracking error of PID, SMC and Pseudo-SMC



Figure 5.10: RMSE of PID, SMC and Pseudo-SMC

Rank	Control Performances Criterion			
	Maximum tracking error	RMSE	Load variation	Chattering
1	Pseudo-SMC	Pseudo-SMC	SMC	Pseudo-SMC
2	SMC	SMC	Pseudo-SMC	SMC
3	PID	PID	PID	PID

CHAPTER 6

CONCLUSION AND SUGESTION TO FUTURE WORK

6.1 Introduction

This chapter provides an overview of this project's overall outcome on what can be concluded are almost all of the results and discussion gained from this final year project entitled "Design Of Sliding Mode Controller For Robust Tracking Of Pneumatic Systems" with all the achieved objectives stated in Chapter 1.

6.2 ConclusionERSITI TEKNIKAL MALAYSIA MELAKA

Sliding Mode Control (SMC) is one of the non-linear control methods that has seen the most widespread application of SMC. The SMC has demonstrated its robustness by successfully tracking trajectories and remaining insensitive to disturbances and parametric uncertainties. Chattering, also known as high frequency oscillations, is a problem that can arise when using conventional SMC.

To summarise, all of the study's objectives were accomplished. The aim of this project is to design an SMC-based controller for the robust tracking of a pneumatic system. This project explores the development of three designed controllers, namely the PID Controller, the SMC Controller, and the Pseudo-SMC Controller, to improve the robust tracking performance of the pneumatic system in the presence of an external load ranging from 0 kg to 9 kg as a disturbance to the experiment plant. Sliding mode control

(SMC) is a method for controlling variable structures. The different control structures are designed so that trajectories always move toward an adjacent region with a different control structure; consequently, the final trajectory will not exist entirely within one control structure. Instead, it will slide along the control structures' perimeters. A sliding mode describes the motion of the system as it slides along these boundaries. SMC has been proposed in control systems as a robust control technique capable of best performance of a control system in the presence of variable operating conditions or system nonlinearities. Thus, three designed controllers will be used for comparison as which designed controller has the best robust tracking performance. MATLAB Simulink tools were used to complete the simulation and experimentation for the sliding mode controllers.

The system identification, as discussed in detail in section 3.4, yielded a mathematical model in the form of the transfer function, also known as the pneumatic system's system model. This system identification achieved over 85% best fit for the system using the System Identification Toolbox by MATLAB. As a result, the system models were ready for simulation analysis and validation. Chapter 4 details the step-by-step procedures for tuning controller parameters.

In Chapter 5, the simulation and experimental results of three controllers (PID controller, SMC controller, and Pseudo-SMC controller) were performed and discussed. The simulation was carried out using system models in the form of the system's transfer function, whereas the experimental work was carried out on the real system using the hardware setup. These controllers were evaluated using three performance measures: maximum tracking error; Root Mean Square Error; and external load attached to the experimental plant setup. The performance of the three controllers was compared, as shown in Table 5.8 in Section 5.4. The project's objectives were achieved with success, as shown in Table 6.2 below.

Objective	Explanation		
Objective 1: To determine the system model of pneumatic system using System Identification Technique.	 The first objective is accomplished by following the four steps of system identification as detailed in Chapter 3. The system model is obtained with best fits of more than 85% of the system using mathematical models in the form of third order transfer functions. Using the Minreal method, the third order of the transfer function was changed into a second order transfer function without making any big changes to the system while it was being simulated. 		
Objective 2: To design SMC controller to obtain robust tracking of pneumatic system Objective 3: Objective 3: NIVERSITI TEX To evaluate the controller performance in term of maximum tracking error and root mean square error. Objective 4: To evaluate the SMC controller's robustness by applying additional loads to the servo pneumatic system.	 The second objective is accomplished through the design of the conventional PID controller and SMC-based controllers, SMC controller and Pseudo-SMC controller via MATLAB Simulink software. The design structures of the controllers are discussed in detail in Chapter 4. The third and fourth objectives were completed successfully when three controllers were simulated through simulation and experimental work. Four control performances were measured in order to evaluate the robust tracking performance of the controller that was designed; these performances are Maximum Tracking Error Root Mean Square Error Robustness against load variation Chattering Effect in SMC-based controllers using FFT method 		

Table 6.1: Summary table between objectives, methodology and expected result

6.3 Recommendation

Following this research, a few recommendations are made in order to draw attention to the important aspects that need to be focused on for the continuation research topic of designing controllers for robust tracking for pneumatic systems with different external disturbances and validated with other types of actuators such as linear drive systems.

It is suggested that an advanced hybrid control methodology with precise control of any nonlinear multi-input multi-output (MIMO) system, which could provide robustness and reliability in comparison to conventional SMC methodologies, could deal with unknown uncertainties and external disturbances by automatically estimating and updating the parameters of the dynamic system. It is expected that this will result in faster convergence and chatter-free response to enhance system performance.



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APPENDIX

A SPECIFICATION SHEET OF THE PNEUMATIC ACTUATOR



¹ The 1-1/16" bore is chrome plated 1050 steel.

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