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Signature	: ....
Supervisor’s Name	: Dr. Chong Shin Horng
Date	:

POSITIONING CONTROL OF A BALL SCREW MECHANISM USING DISTURBANCE OBSERVER

FOO JIA EN

A Report Submitted In Partial Fulfilment of Requirements for the Bachelor Degree of Electrical Engineering (Control, Instrumentation & Automation)

Faculty of Electrical Engineering

UNIVERSITI TEKNIKAL MALAYSIA MELAKA

2015

I declare that this report entitle “Positioning Control of a Ball Screw Mechanism Using Disturbance Observer” is the result of my own research except as cited in the references. The report has not been accepted for any degree and is not concurrently submitted in candidature of any other degree.

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To my beloved mother and father

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ABSTRACT

A ball screw mechanism is a mechanical actuator that is widely used in various high precision automated industries. Though it is highly efficient, the ball screw mechanism often exhibits non-linear behaviours due to various forms of external noises, frictions, unforeseen disturbances and parameters uncertainties. Such behaviours often cause instability, large steady-state error and poor transient performance in the mechanism. To overcome these problems, a Disturbance Observer with PD Controller (PDDO) is proposed. A PDDO is made up of nominal plant, low pass filter and a PD controller. Compare to classical controllers, the proposed controller has low sensitivity towards non-linearity of the mechanism. In this project, the plant model was modelled using non-linear least square method (NLLS). The nominal plant was designed based on the Ackermann's formula. The cutoff frequency was taken as half of the cut-off frequency of the plant model. The PD controller was constructed through manual tuning method. The performance of PDDO was examined experimentally in tracking motion with sinusoidal inputs of different frequencies and amplitudes. A manually tuned PID controller was designed in order to compare with the proposed controller. The robustness towards mass variation of the controllers were examined. Overall, the experimental result has proved that PDDO has demonstrated better tracking performance and higher adaptability to the change of input's amplitudes and frequencies. It was also found that PDDO is robust towards mass variation as compared to the PID controller.

ABSTRAK

Mekanisme skru bola merupakan sebuah penggerak mekanikal yang sering digunakan dalam industry automasi yang mementingkan ketepatan dalam pengukuran. Walaupun mekanisme ini mempunyai kecekapan yang tinggi, namun ciri-ciri tidak linear akibat gangguan luaran, geseran dan ketidaktepatan parameter dalam model mekanisme ini telah dilaporkan. Ciri-ciri ini dikatakan akan menyebabkan ketidakstabilan dan ralat yang besar dalam pergerakan mekanisme skru bola. Bagi mengatasi kekurangan mekanisme ini, sebuah pengawal pemerhati gangguan dengan pengawal PD(PDDO) telah dicadangkan. Berbanding dengan pengawal klasik, PDDO mempunyai sensitivti yang lebih rendah terhadap fenomena tidak linear yang berlaku dalam mekanisme ini. Pengawal pemerhati gangguan (DOB) mampu menganggar dan menolak perubahan dalam sistem, manakala pengawal PD mengawal pergerakan transien sistem ini. Dalam projek ini, PDDO telah menunjukkan prestasi pergerakan yang lebih mantap berbanding dengan pengawal PID. Dalam eksperimen penukaran beban, pengawal PDDO menunjukkan prestasi pergerakan yang lebih baik berbanding dengan pengawal PID.

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LIST OF SYMBOLS AND ABBREVIATIONS

CNC	Computer Numerical Control
DOB	Disturbance Observer
FR	Frequency Response
NCTF	Nominal characteristic Trajectory Following
NLLS	Non Linear Least Square
PDDO	Disturbance Observer with PD controller
PID	Proportional-Integral-Derivative Controller

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CHAPTER 1

INTRODUCTION

A ball screw mechanism is a mechanical actuator that translates rotary motion of the driver motor into linear displacement [1]. It is widely used in various automated industries such as aerospace industries, semiconductor industries and CNC machineries due to its high precision and efficiency [2]. Unlike lead screw, a ball screw contains ball bearing along the screw shaft and experiences less friction than a lead screw. However, it was discovered that the ball screw mechanism exhibits non-linear behaviours such as backlash, frictions, load variation and high frequency sensor noise [3], [4]. Based on [5], such behaviours are sometimes known as disturbances in the mechanism.

Figure 1.1: Difference between ball screw and lead screw [6]

To ensure the ball screw mechanism achieves high precision performance under these conditions, positioning control is vital and necessary to be applied in the mechanism. According to [7], positioning control can be performed by applying a controller and/or with the use of advance sensors. However, most industries opt for controller design as these sensors are relatively expensive. Among these industries, a conventional PID is still widely used due to its simplicity and ease of application. However, a PID controller has to be tuned frequently due to the non-linear characteristics in the mechanism. To overcome the limitation of classical PID controller, many advance controllers like H-infinity controllers (H-∞) [8-9], discrete time sliding mode controller [10], Fuzzy Logic Controller (FLC) [11], Fuzzy PID controller [12] were designed to achieve high performance and robustness.

Unlike these advance and complicated controllers, a disturbance observer (DOB) appears to be simpler and easier to use. A DOB does not compensate the system directly [13]. Instead, DOB estimates the disturbances arise from frictions, vibrations and/or parameters variations that occurs in a plant and feeds the error negatively back to perform compensation. Such compensation is usually done with controllers like H-infinity [14] and conventional PD or PID [5,13].

This project aims at designing a DOB with the use of PD controller for the ball screw mechanism. The DOB rejects the disturbances while the PD controller compensates the system so that it achieves desired positioning and tracking performance. The controller performance will be validated and the robustness of the controller will be examined with change of load mass.

1.1 Project Motivation

Precision performance in ball screw mechanism has always been the major consideration in industries. Different controllers were designed and built to improve the transient responses of the system. However, it is observed that under normal conditions, the transfer function model of the mechanism is difficult to be built accurately due to surrounding disturbances, as well as the uncertainty of parameters of the model itself. Taking conventional PID as an example, it demonstrates instability when the mechanism experiences sudden disturbances and parameter variations. To avoid these issues, a disturbance observer is proposed to minimize the effect of such disturbances and unforeseen changes of parameters.

1.2 Problem Statement

Conventional PID is widely used in different industries to improve transient performance of ball screw mechanism due to its ease of use and implementation. However, it is noticed that this controller has low adaptability to parameters variation and has to be tuned frequently to maintain its optimum performance. This procedure is troublesome and not effective as the transfer function has to be determined again whenever there's a change in the parameters involved. Due to the limitation of PID, many advance controllers such as the H_∞ controller and NCTF were designed. These controllers had proven their robustness under parameters uncertainties, mostly due to disturbances and load change. Though these controllers are robust, it is observed that these controllers require one to have a high level of relevant understanding before designing it. Compare to these controllers, a DOB is simpler and does not require an exact model of the plant. It rejects sudden disturbances while adapting itself to variation of parameters in the model. As DOB only performs disturbance rejection, thus an external PD controller is proposed to improve the positioning performance of the ball screw mechanism.

1.3 Objectives

The main objectives of this project are:

i. To construct a second order mathematical model of the ball screw mechanism;
ii. To propose a Disturbance Observer with PD controller (PDDO) for the ball screw mechanism;
iii. To validate the positioning performance and robustness against mass variation of the PDDO in tracking motion in comparison to PID controller;

1.4 Scope of Work

In order to complete this project, the limitations are presented as follow:

i. The maximum working range of the ball screw mechanism is set as 160mm;
ii. The range of input voltage used in the experiments is 0 to ±10V;
iii. The resolution of the linear encoder is given as 0.5µm/pulse;

1.5 Report Outline

This report presents the positioning control of a ball screw mechanism using disturbance observer. Chapter 2 summarizes the background of different controllers applied on ball screw mechanism. PDDO is discussed in details on its structure together with the different applications applied. Chapter 3 begins with demonstration of the steps to model the ball screw mechanism and follow by the design procedures of PDDO and PID controller. Steps for performance and robustness evaluation for the controllers are presented as well. The results from conducted experiments are presented in Chapter 4 with the analysis and discussions. Lastly, this project is concluded in Chapter 5 and recommendations are given for future works and improvement. This report is ended with the reference list and the appendices of the related works.

CHAPTER 2

LITERATURE REVIEW

2.1 Introduction

In most automated industries, precise positioning control is no longer an unfamiliar term. According to A. Kato and K. Ohnishi, positioning control is one of the examples of basic components in motion control technology [15]. This technology aims at ensuring the controlled target achieves desired positioning performance despite occurrence of unwanted noise signals, disturbances or force deviations [16]. Based on a F. Yakub and R. Akmeliawati, positioning control can be further classified into two subcategories: point-to-point positioning control (PTP) and continuous path tracking control [17]. S. Chong and K. Sato stated that controllers of simple structures, rapid response and non significant overshoot are highly demanded in any of the automated industries and high-end mechanisms [18]. C. Tsui added that such controllers should also exhibit low sensitivity, i.e. robustness towards model parameters uncertainties and sudden disturbances[19].

2.2 Previous Works on Positioning Control of a Ball Screw Mechanism

Ball screw mechanisms are used in different applications such as CNC machineries, airplane wing flap release mechanism and automobile power steering that seek for high precision, stiffness and efficiency. However, past studies indicated that ball screw mechanisms exhibit nonlinear behaviour in micro movement [20]. Chen, Jang and Lin also pointed out that such microdynamic behaviours are caused by hysteresis, Stribeck effect and the preload of the ball bearings in the mechanism [21]. In another similar research, Dong and Tang added that a ball screw mechanism has varying natural frequencies along its screw shaft [22]. Due to these phenomena, many works were performed to identify the model of a ball screw mechanism while considering the macrodynamics and microdynamics of the system.

In [22], Dong and Tang proposed a hybrid modeling that characterize the axial, torsional and flexural vibration dynamics of the ball screw mechanism. In this model, the screw shaft characters, including the Young's modulus and Poisson's Modulus are considered. It was concluded that this hybrid model is capable of demonstrating the structure dynamics accurately. In another similar research, Liu, Zhao and Zhang introduced a hybrid modeling that presents the high frequency behaviours of the ball screw mechanism [23]. The screw shaft dynamics is demonstrated in longitudinal and torsional dimension. In this work, the authors only consider a low order model with two inertias: one from motor while another is from the table.

From these researches, it can be seen that it is relatively complicated to model a ball screw mechanism while considering all the microdynamic parameters. Besides that, G. J. Maeda and K. Sato also pointed out that it is relatively difficult to model the microdynamic model as these parameters change over time and position [24]. In order to ease the procedure, it is common to lump the parameters in the system. This approach is also known as macrodynamic modelling. In [4], Sepasi, Nagamune and Sassani lumped the equivalent inertia and damping coefficient of a ball screw mechanism as a second order model. The same approach was adopted by another research as shown in [25]. For this method, Lin and Chen explained that since the ball screw frictional torque dominates the system, thus the model can be reduced where the parameters are lumped together [11]. Though this method proves to be easier, a controller is highly necessary to adapt to the possible mismatch or parameter variations in the macrodynamic model.

Over the years, classical PID controller is widely applied due to its practical and simple applications. A PID controller includes three terms: Proportional, Integral and Derivative [26]. Based on [18], a PID controller is an effective and reliable controller provided that it is properly tuned. However, this controller meets its limitation should higher precision performance and system robustness are demanded. To improve this controller, different advance controllers were designed based on the characteristics of classical PID. In [27], Chen had demonstrated the ability of PID with Fuzzy Logic Controller in achieving high speed response with high precision positioning despite varying frictions in linear DC motors. In this thesis, Chen had designed a two stage controller that included a Hybrid Reduced Rule Fuzzy PID controller (PIDFLC) and a relay-tuned PID controller. Based on his observations, a PID controller has limited positioning performance due to the parameters uncertainties in the DC motor model whereas a PIDFLC is capable in achieving the desired performance under similar condition. In a similar research, T. Ting designed a Fuzzy PID controller to perform positioning control on a ball screw mechanism [12]. Unlike PID controller that requires frequent tuning, this paper concluded that a Fuzzy PID works with different input. A Fuzzy PID has a larger stability range while possesses a higher adaptability towards parameters variations. However, T. Ting also pointed out that a Fuzzy PID controller has a slower response time compared to conventional PID.

In year 2004, a H_∞ framework was designed to achieve robust, fast and precise positioning control of a ball screw system [8]. In this research, two vibrations mode were considered: the low stiffness between motor and table, as well as the oscillatory disturbance force of load to table. A 2 DOF feed-forward compensator was designed using coprime factorization approach to improve the transient response of ball screw mechanism. On the other hand, H_∞ framework was designed by selecting the appropriate weighting function to achieve robustness over servo bandwidth expansion as suggested in [28]. This research had proven that H_∞ framework is capable of achieving system robust stability over different vibration modes and increased response speed towards expansion of servo bandwidth. In [9], a H_∞ controller was proposed to compensate friction and improve the reference tracking performance. The loop shaping approach used in the research was originally proposed by D. Mcfarlane and K. Glover to achieve system robustness while improving performance and stability [29]. In [9], a suitable dynamic frequency shaping function was selected based on the frequency region where frictions occur. Differ from [28], this research improves controlled performance by multiplying the specified weight function to front and rear of the open loop transfer function.

In recent years, positioning control with NCTF controller is proposed on different applications including one mass rotary system [7], vibration control in two mass rotary system [30] and ball screw mechanisms [17,31]. According to A. Sabanovic and K. Ohnishi, it was pointed out that an exact mathematical model of the plant can never be modeled since uncertainties of model parameters may present due to noise and surrounding disturbances [32]. With this issue taken into consideration, the NCTF controller was designed such that the exact parameters are not necessary in the process [17]. In [31], NCT controller was proposed to preserve the robustness of a ball screw mechanism despite variation of mass. Another similar research demonstrated in [33] proved that a continuous-motion NCTF (CM NCTF) is capable of reducing vibration in the ball screw mechanism and improves motion accuracy.

In [34], C. Lu and M.-C. Shih proposed fuzzy sliding mode control method to perform positioning control in the ball screw mechanism driven by pneumatic servomotor. Initially researched by Mamdani and his colleagues [35], this paper uses triangular membership function and Mamdani rules to perform fuzzyfication and fuzzy reasoning. From the experimental results, [34] proved the robustness of the controller when non-linear compressed air is supplied into the ball screw mechanism.

Another common controller used to perform positioning control is the disturbance observer (DOB). DOB was first proposed by K.Ohnishi in 1983 to perform torque-speed regulation in DC motor [36]. In later years, DOB was widely applied in different mechanisms such as magnetic hard drive servo system and ball screw mechanism [13], [37]. A DOB is capable of estimating the disturbance torque due to non-linear characteristics and subsequently rejects such disturbances and compensates model uncertainties [38]. In another research, DOB was designed to control vibrations occurred in the plant [5]. Since a DOB only works on disturbance rejections, thus it is necessary to include an external controller to perform positioning control [13]. In [13], a DOB was proposed to compensate a ball screw servo system with the aid of a PD controller. This research has also pointed out that a PID controller could not be used with the observer as it produces large overshoot and major oscillations. This statement was supported by P. I. Ro *et.al.* stating that a PID exhibits severe transient oscillations that may lead to positioning error [39]. Taking step and sinusoidal disturbance into consideration, PDDO showed robustness though parameter variations and non-linear frictions existed in the ball screw mechanism [13]. In another research, P. I. Ro *et.al.* proposed a PDDO controller for sub micrometer positioning control [39]. From this research, it was found that a PDDO produces consistent and desired positioning and tracking response despite existence of non-linear frictions.

From the controllers mentioned earlier, a PDDO is proposed to perform positioning control on a ball screw mechanism. Compared to H_∞, the PDDO controller has a much simpler design procedure [40]. It also has a less complex structure compare to Fuzzy PID controller [17]. A DOB has the ability to estimate disturbances and rejects them from the system as well as compensate parameters uncertainties due to nonlinear behaviour [5,38]. In [38], it is also stated that a DOB is capable of shaping the plant to behave as the nominal plant model at low frequencies.

2.3 Disturbance Observer with PD Controller (PDDO)

The PDDO is made up of three important elements: a nominal plant, $P_n(s)$, low pass filter, Q(s), and a PD controller, C(s). The general structure of a PDDO is presented in Figure 2.1 while the symbols are presented in Table 2.1.

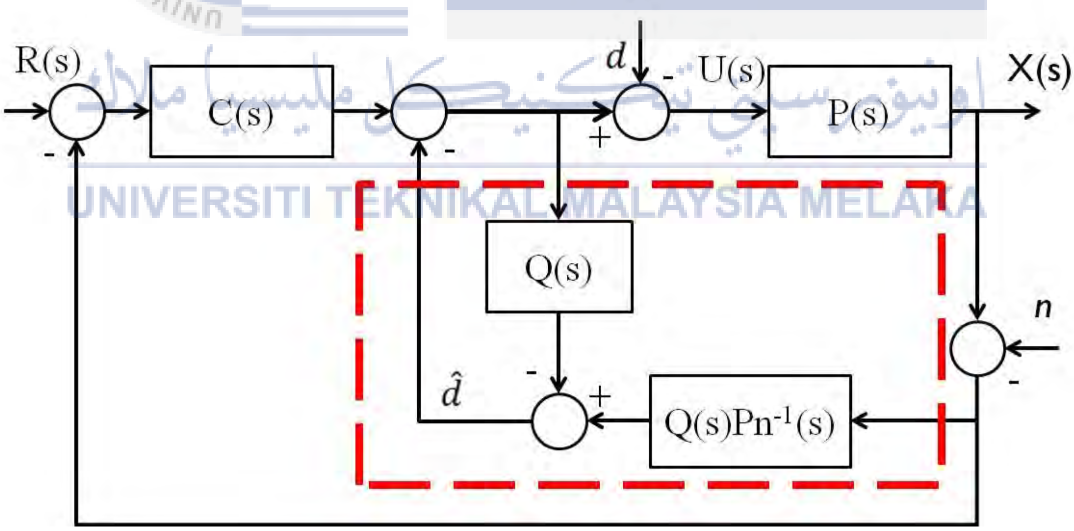

Figure 2.1: General Structure of PDDO [3]

Table 2.1: Model parameters for PDDO

Symbol	Model / Parameters
C(s)	PD controller
R(s)	Reference input
P(s)	Plant model
$P_n(s)$	Nominal plant
Q(s)	Low pass filter
X(s)	Output of the plant
U(s)	Controlled input to plant
d	Disturbance torque
$\hat{d}$	Estimated disturbance torque
n	High frequency noise

In designing a PDDO, the nominal plant is usually designed different from the real plant as the system follows the behavior of the nominal plant at low frequencies [41]. However, H. Kobayashi, S. Katsura and K. Ohnishi pointed out that if the nominal plant is very different from the real plant, the system tends to be unstable [42] It is ideal to design the nominal plant as simple as possible to reduce the order of the low pass filter, Q(s). The order of low pass filter is the same as the relative order of $P_n(s)$ to ensure that the block, $QP_n^{-1}(s)$ is valid with the order of numerator larger than denominator [43]. A low pass filter that is higher than the first order will have higher sensitivity but less robustness [44].

CHAPTER 3

METHODOLOGY

In this chapter, the system modelling and controller design procedures will be discussed. These procedures are divided accordingly to achieve the objectives as listed in Section 1.4. In Section 3.1, the experimental setup, mathematical model and modelling procedure will be demonstrated. In Section 3.2, the design steps of a PDDO controller will be presented. The identification of nominal plant, low pass filter design and the PD controller design will be discussed.

3.1 System Modelling of Ball Screw Mechanism

The ball screw mechanism to be used in the project is setup as shown in Figure 3.1. The ball screw mechanism is driven by a DC motor. The working range of this mechanism is given as 160mm. A linear encoder with resolution of 0.5µm/count is attached to the ball nut with its track placed side-by-side. This contactless linear encoder measures the incremental displacement with 2 channels: Channel A and Channel B with a phase shift of 90° between them. The close up view of the ball screw mechanism is presented in Figure 3.2. The procedures are then continued with the mathematical modelling of the ball screw mechanism.

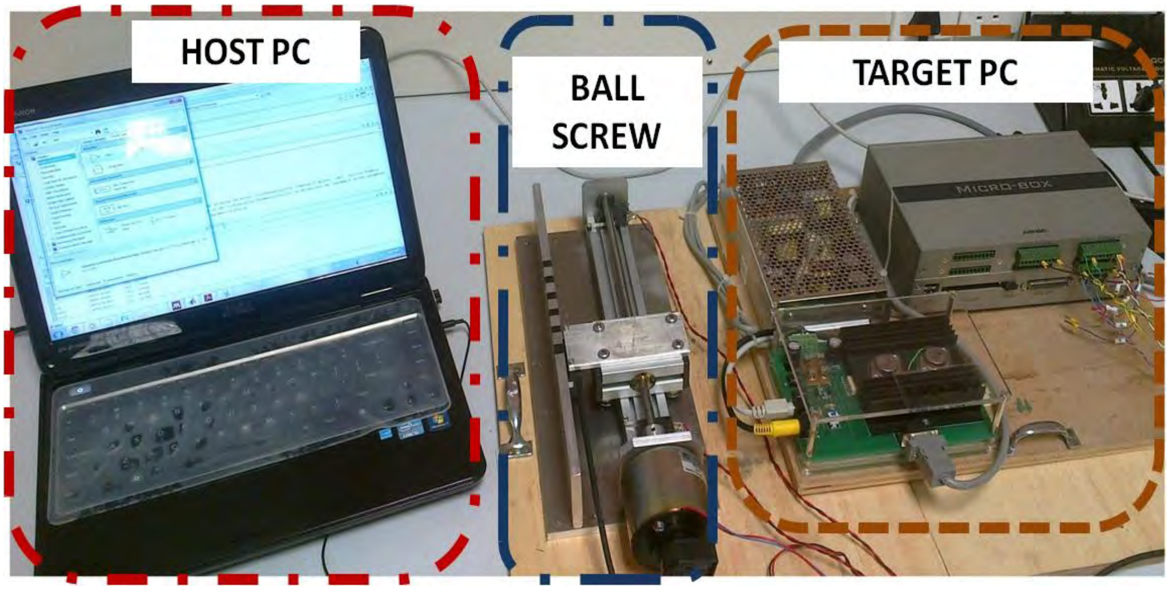

Figure 3.1: Experimental setup of ball screw mechanism

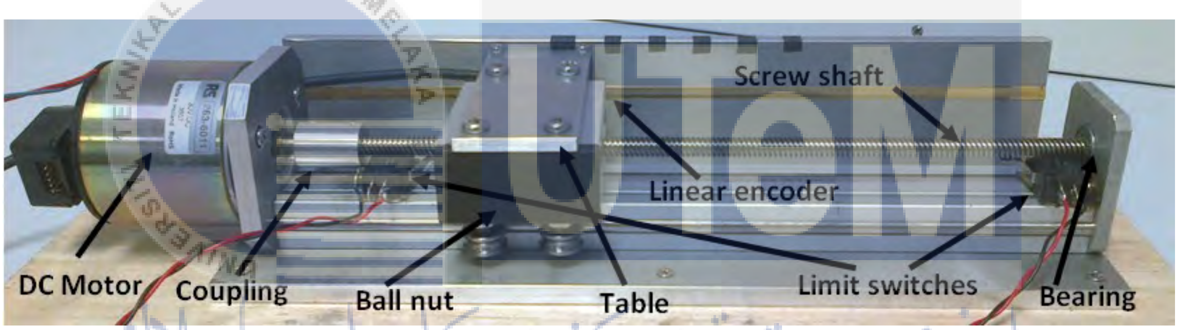

Figure 3.2: Close up view of the ball screw mechanism with labelled parts

To construct the second order mathematical model of the ball screw mechanism, the block diagram as shown in Figure 3.3 is considered.

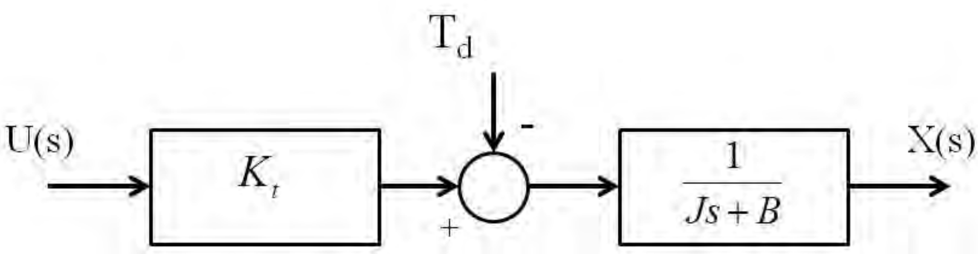

Figure 3.3: Second order model of the ball screw mechanism [45]

Table 3.1 Model parameters

Symbol	Parameters
Kt	Motor Constant
K_b	Back emf Constant
K_a	Amplifier Gain Constant
J	Effective Inertia of Ball Screw, Motor and Load
B	Effective Damping Coefficient of Ball Screw, Motor and Load
T_d	Disturbance Torque

In macrodynamic modelling, the inertia and damping coefficient of the ball screw, load and motor are lumped into single parameters of J and B. The equation of motion of the ball screw mechanism is given as

$$J\ddot{x}(t) + B\dot{x}(t) = U(t) \tag{3.1}$$

where $x(t)$ is the output linear displacement of the ball screw mechanism when it has voltage supply, $U(s)$. By taking K_b and K_a into consideration, an open loop block diagram for the ball screw mechanism is presented as shown in Figure 3.4.

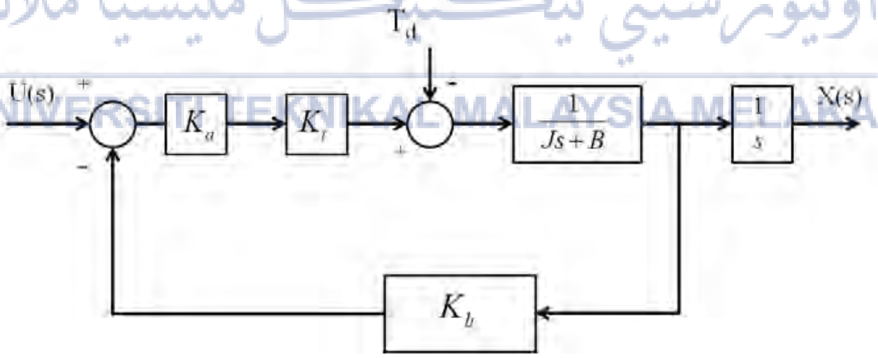

Figure 3.4: Block diagram for ball screw mechanism

Using block diagram reduction approach, the transfer function of the open loop ball screw mechanism, G(s) is given as

$$G(s) = \frac{X(s)}{U(s)} = \frac{K_t}{s[(Js + B) + K_b K_t]} \tag{3.2}$$

where disturbance torque, T_d is assumed to be zero.

To determine the values of the parameters, system identification was performed. By using a band-limited random white noise as an input, the plot for open loop input voltage and output displacement in the ball screw mechanism is presented in Figure 3.5.

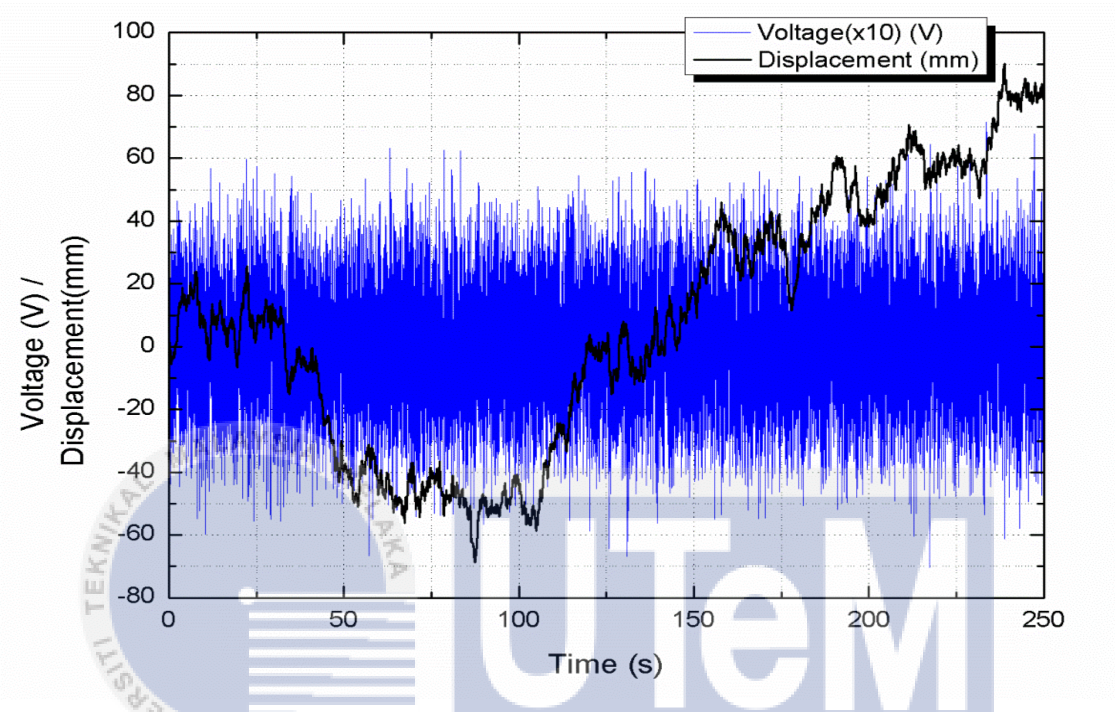

Figure 3.5: Input voltage with output displacement.

Through the data collected, the frequency response (FR) of the ball screw mechanism is plotted. Non-linear Least Square (NLLS) method is adopted to estimate a second order system as presented in equation (3.2). The estimated model and experiment frequency responses are plotted in Figure 3.6.

It is observed that there are still differences between the estimated frequency response and the experimental results. This difference is due to the unconsidered microscopic behaviour of the mechanism. Since the estimation is performed based on a lumped model, thus phenomena such as hysteresis, friction due to ball bearing and plasticity of ball screw might be overlooked.

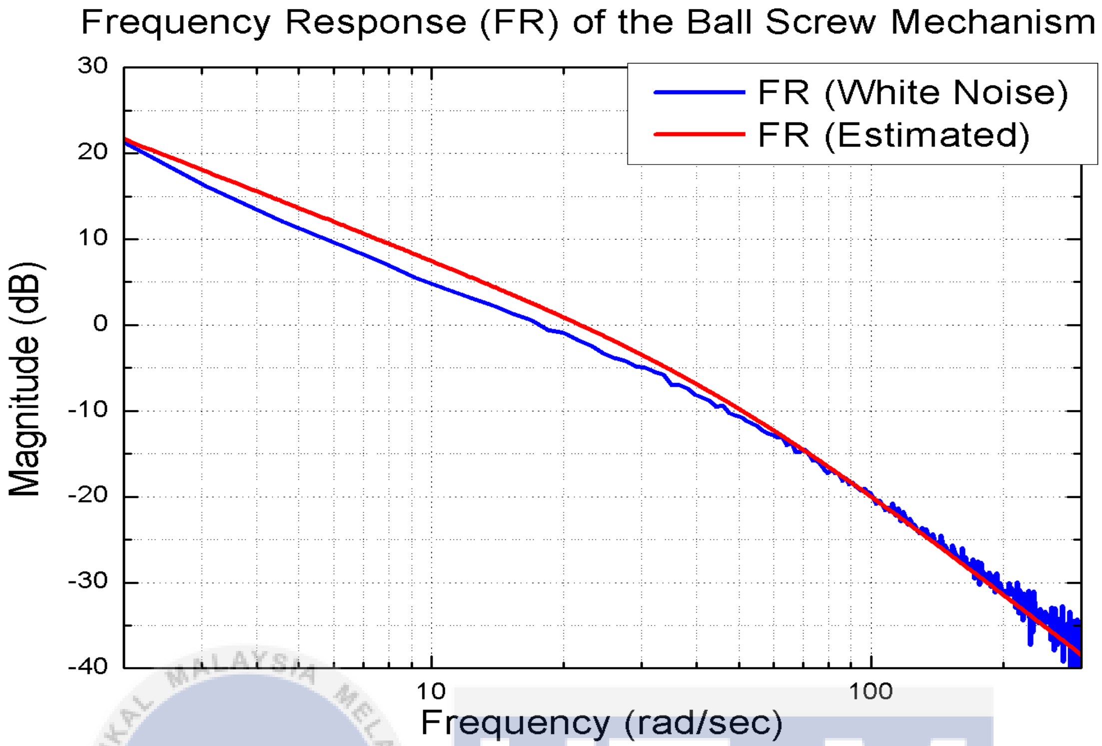

Figure 3.6: Frequency response (FR) from white noise and estimation with NLLS method

Through system identification with NLLS method, the transfer function is given as

$$G(s) = \frac{Y(s)}{U(s)} = \frac{1096}{s^2 + 45.32s} \tag{3.3}$$

with a bandwidth of 29.1 rad/sec taken at -3dB of the simulation FR plot.

3.2 Design of Disturbance Observer with PD Controller (PDDO)

In order to design the nominal plant, the transient parameters are set where the overshoot percentage is 2% while the settling time is taken as 0.5 seconds. Using the general transfer function of a second order system:

$$G(s) = \frac{\omega_n^2}{s^2 + 2\zeta\omega_n + \omega_n^2} \tag{3.3}$$

By using the above said transient parameters, G(s) is obtained as

$$G(s) = \frac{105.19}{s^2 + 16s} \tag{3.4}$$

where the desired poles are $S_d = -8 \pm j6.4182$

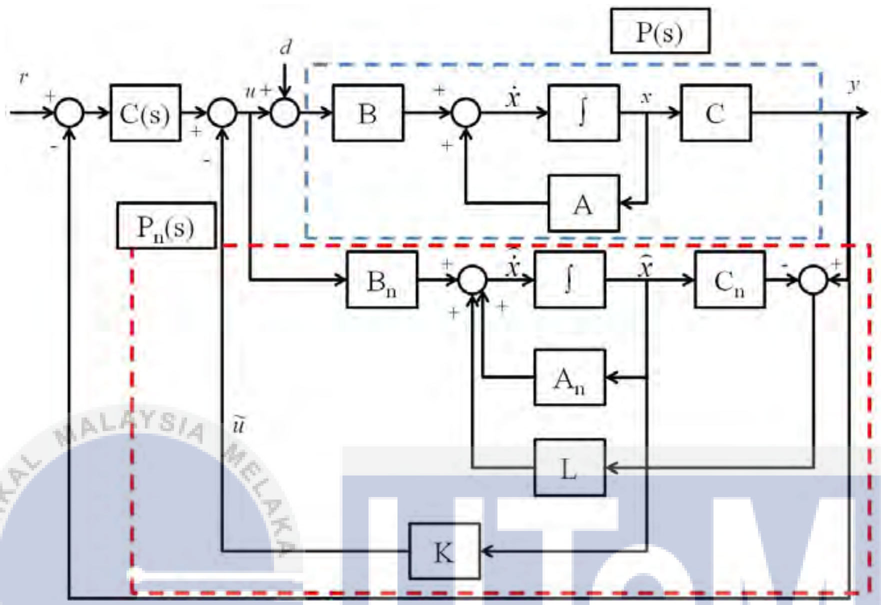

Figure 3.7: Structure of PDDO in state space representation [46]

To design the disturbance observer, the observer poles is made 5 times faster that the controller poles where poles = [-40,-40]. Representing G(s) in observer canonical form, the state space model is given as

$$\dot{x} = \boldsymbol{A}x + \boldsymbol{B}u \tag{3.5}$$

$$y = \boldsymbol{C}x$$

$$\dot{x} = \begin{bmatrix} 0 & 0 \\ 1 & -16 \end{bmatrix} x + \begin{bmatrix} 105.1937 \\ 0 \end{bmatrix} u \tag{3.6}$$

$$y = \begin{bmatrix} 0 & 1 \end{bmatrix} x$$

The general observer form as presented in Figure 3.7 is given as

$$\dot{\hat{x}} = \boldsymbol{A}\hat{x} + \boldsymbol{B}\tilde{u} + \boldsymbol{L}\tilde{y} \tag{3.7}$$

$$\tilde{u} = -\boldsymbol{K}\hat{x}$$

This form can be rewritten as

$$\dot{\hat{x}} = (\boldsymbol{A} - \boldsymbol{BK} - \boldsymbol{LC})\hat{x} + \boldsymbol{L}(y - r) \tag{3.8}$$

$$\tilde{u} = -\boldsymbol{K}\hat{x}$$

Using Ackermann's function, the observer gain, L and feedback gain, K is obtained as

$L = \begin{bmatrix} 1600 \\ 64 \end{bmatrix}$ and $K = [0 \quad 1]$.

Therefore, the nominal plant is modelled as

$$P_n(s) = \frac{64s + 1600}{s^2 + 80s + 1705} \tag{3.9}$$

To design a low pass filter for the DOB, the cutoff frequency is determined from frequency of the estimated plant model. At -3dB, the cutoff frequency of the plant, P(s) is given as 29.1 rad/s or 4.63Hz. The cut off frequency for the low pass filter is set at about half of the cutoff frequency where ω_c = 14 rad/sec. The low pass filter, Q(s) is designed as

$$Q(s) = \frac{14}{s + 14} \tag{3.10}$$

In the design of PD controller, fine tuning method is adopted. The proportional gain, K_p is increased until the system begins to be unstable, i.e. starting to oscillate. The value is reduced to half and a derivative gain, K_d is added into the system. K_d is tuned to reduce the settling time and percent overshoot. Since PD controller is capable of picking up the measurement noise in the system, a low pass filter with time constant, T_d = 0.0714 is added to the derivative part. The equation of the PD controller is now given as

$$C(s) = K_p + \frac{K_d s}{T_d s + 1} \tag{3.12}$$

where K_p = 10 and K_d = 0.5.

In order to simplify the structure of DOB, the block diagram shown in Figure 2.1 is reduced through the block reduction method. The flow of the block reduction process is shown in Figure 3.8.

In order to compare PDDO with PID controller, PID controller is designed by fine tuning method. The equation of PID controller is given as

$$G_c(s) = K_p + \frac{K_I}{s} + \frac{K_d s}{T_d s + 1} \tag{3.13}$$

where K_p = 5, K_d = 0.05, K_i = 12 and T_d = 0.0714.

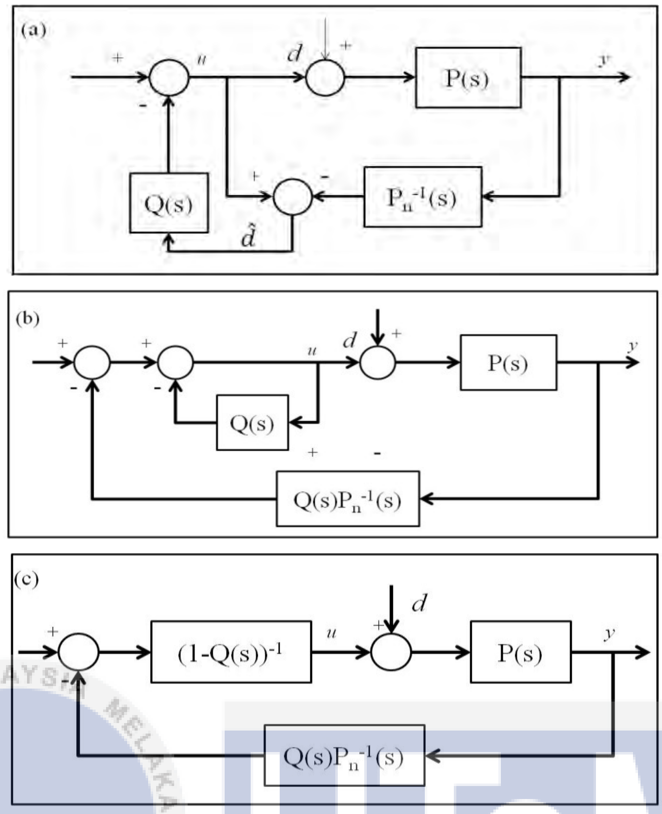

Figure 3.8: Block reduction process for PDDO

3.3 Performance Evaluation

In this project, the positioning performance of PDDO is examined and compared experimentally with PID controller in tracking motion. The experiments are run with sinusoidal inputs with frequency = 0.1Hz, 1Hz and 3Hz. The amplitudes of the inputs are set as 0.1mm, 1mm and 5mm. With the same experimental setup, the robustness of PDDO and PID against load mass variations is tested.

CHAPTER 4

RESULTS AND ANALYSIS

4.1 Uncompensated System Response

Before going into the PDDO controller design, it is important to observe the uncompensated system beforehand. By referring to Figure 4.2 to Figure 4.5, it is observed that the ball screw mechanism has difficulties moving in small displacement due to the large friction occurred along the ball screw shaft. On top of that, this system also has large tracking error when it is moving in higher velocity, i.e. higher frequency input. Thus, it is desired that the designed PDDO is able to improve the tracking performance and reduce the tracking error in the system.

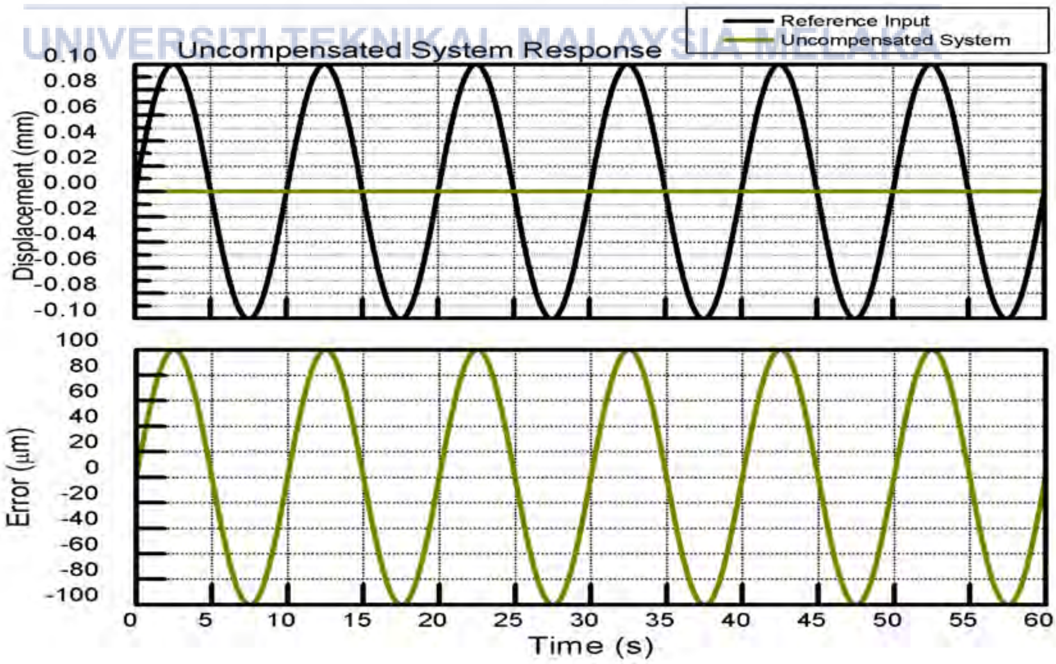

Figure 4.1: Uncompensated response with frequency = 0.1Hz and amplitude = 0.1mm

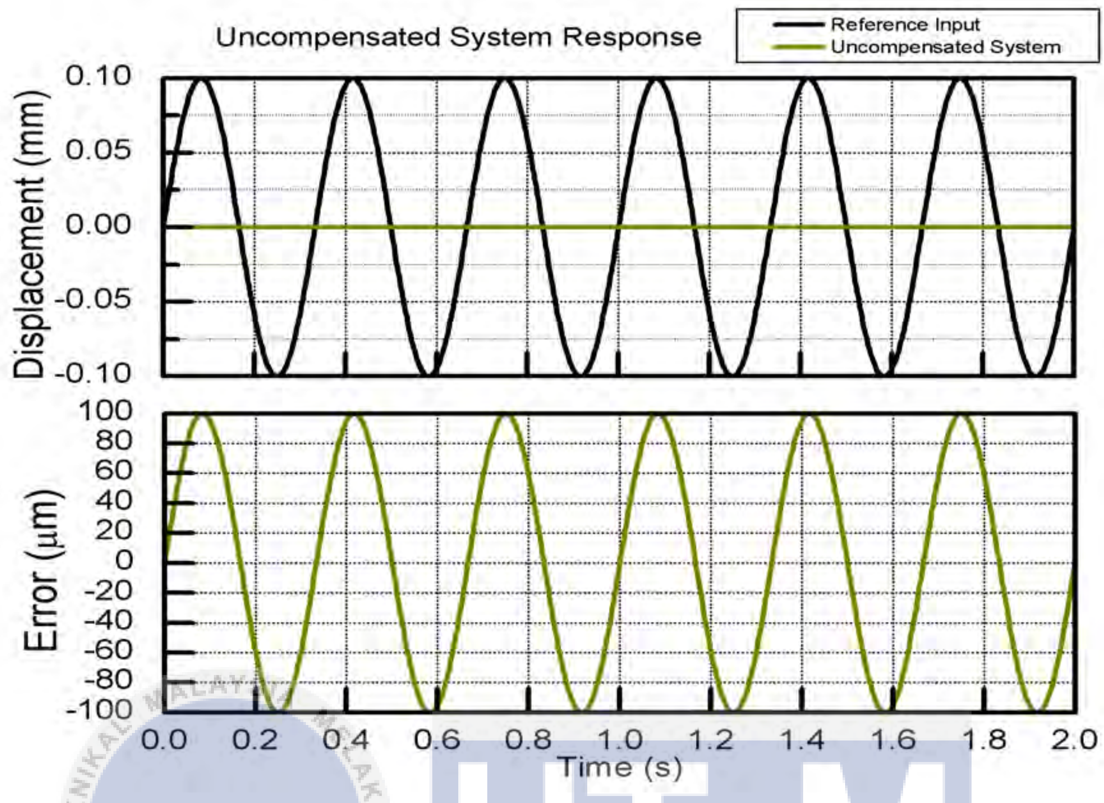

Figure 4.2: Uncompensated response with frequency = 3Hz and amplitude = 0.1mm

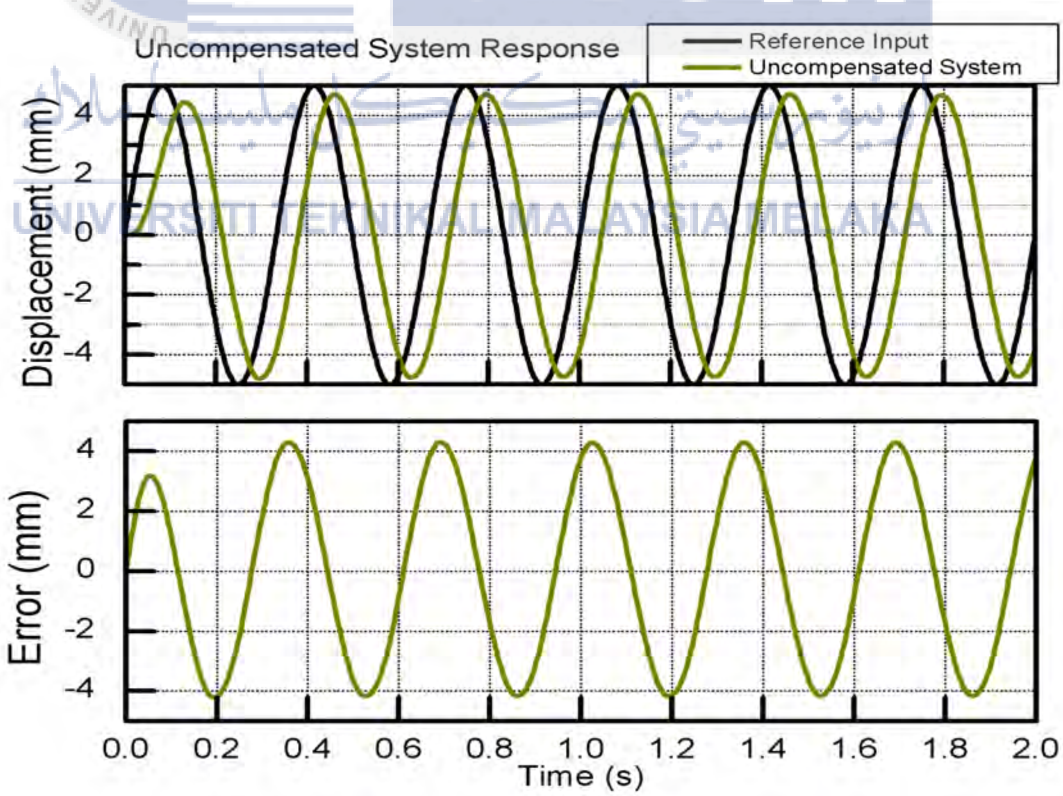

Figure 4.3: Uncompensated response with frequency = 3Hz and amplitude = 5mm.

4.2 Performance Evaluation of Tracking Motion

To examine the performance of the mechanism, tracking tests are performed. Frequencies of 0.1Hz, 1Hz and 3Hz are considered to examine the performance of PDDO as compared to PID controller. These frequencies are tested with different amplitude of 0.1mm, 1mm and 5mm. Each set of experiments is run for 10 times to examine the adaptability of the said controllers in tracking motion. The system responses of the PDDO and PID are presented in Figure 4.5 to Figure 4.13. The average error and standard deviation of 10 repeated tests for each experiment set are tabulated in Table 4.1.

When the reference input is set to 0.1.mm, the tracking error of PDDO increases with increment of frequency. However, when the input amplitude is set as 5mm, PDDO shows improvement of tracking error when the frequency is increased. It can be seen clearly that the tracking performance of PDDO controlled system is much better compared to the PID controlled system. In the repeatability tests, PDDO shows lower standard deviations than PID controller.

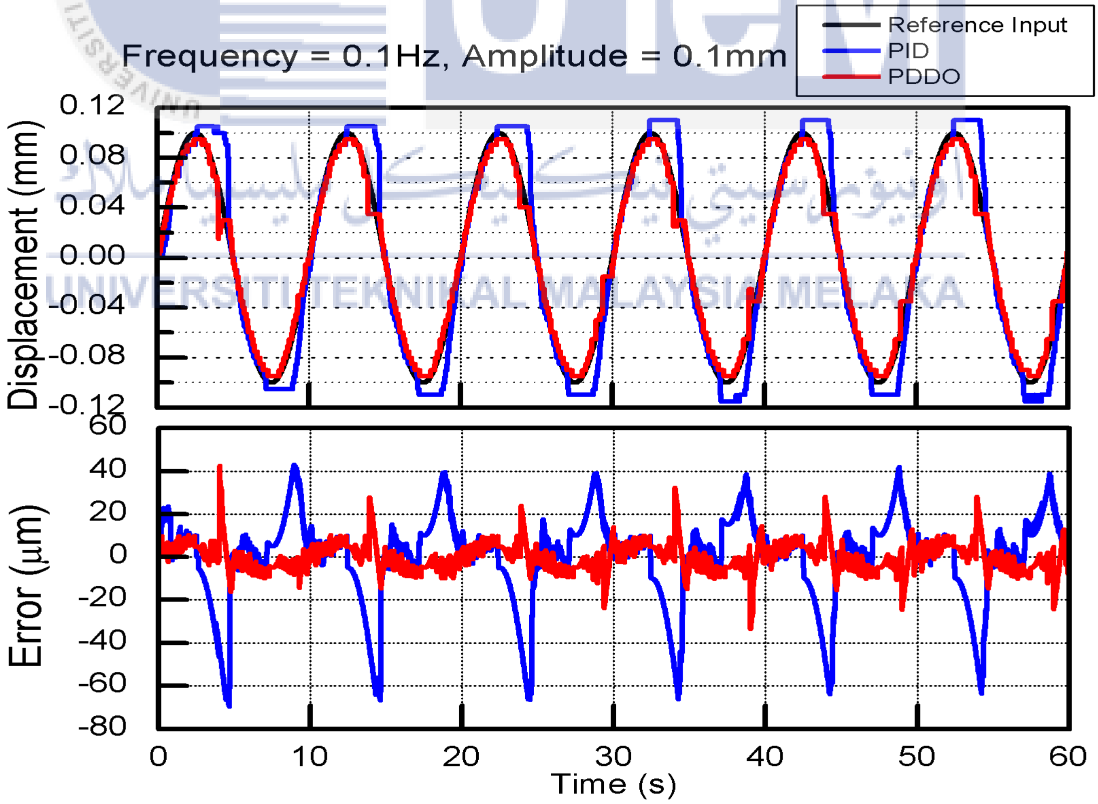

Figure 4.4: Output with reference input of frequency = 0.1Hz and amplitude = 0.1mm

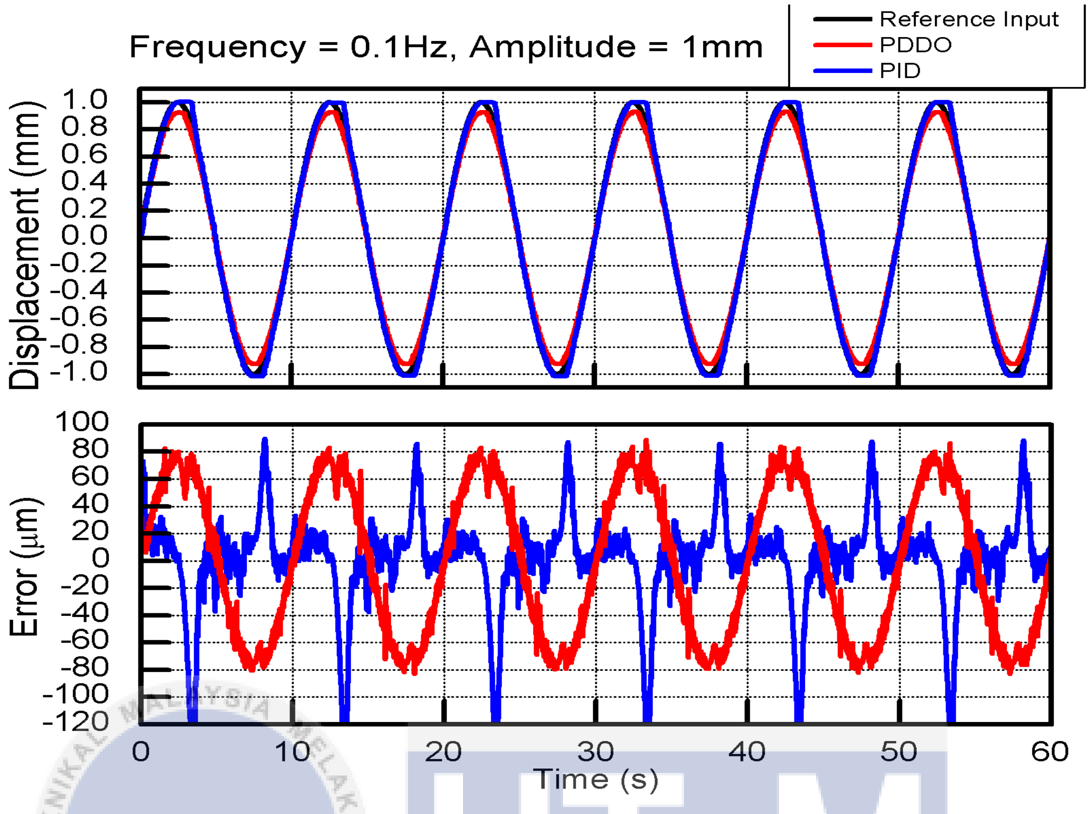

Figure 4.5: Output with reference input of frequency = 0.1Hz and amplitude = 1mm

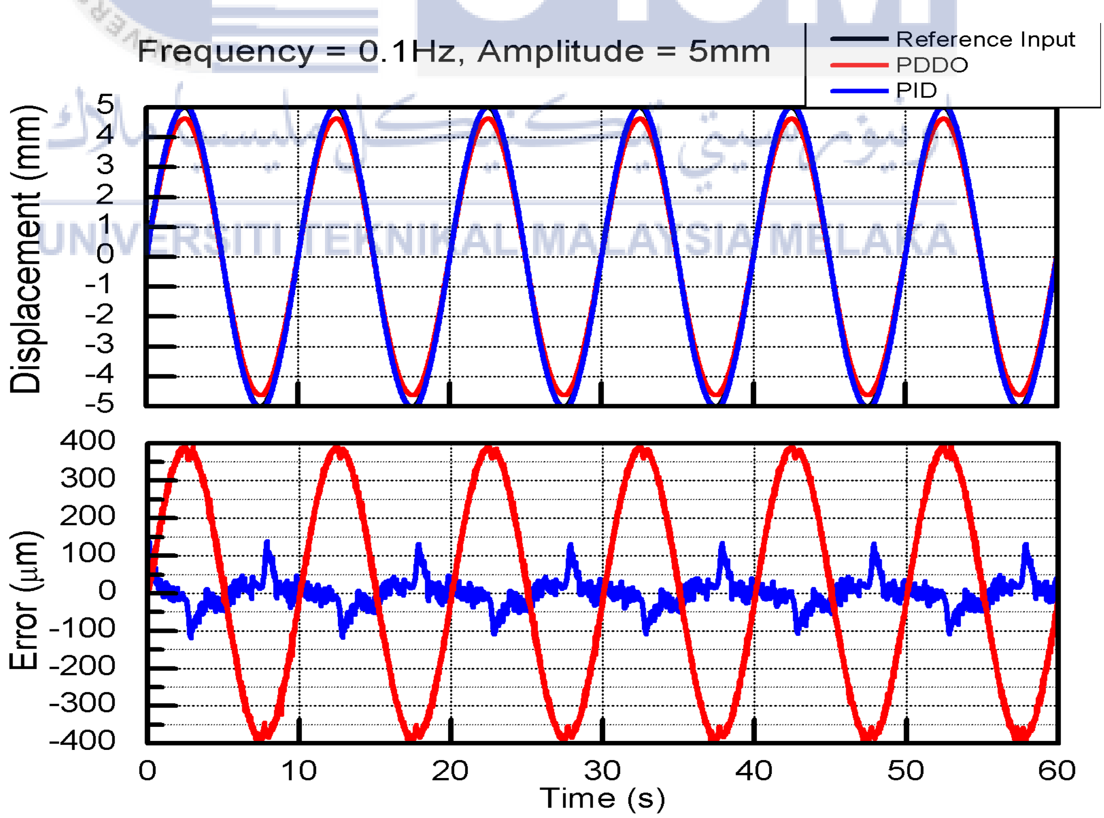

Figure 4.6: Output with reference input of frequency = 0.1Hz and amplitude = 5mm

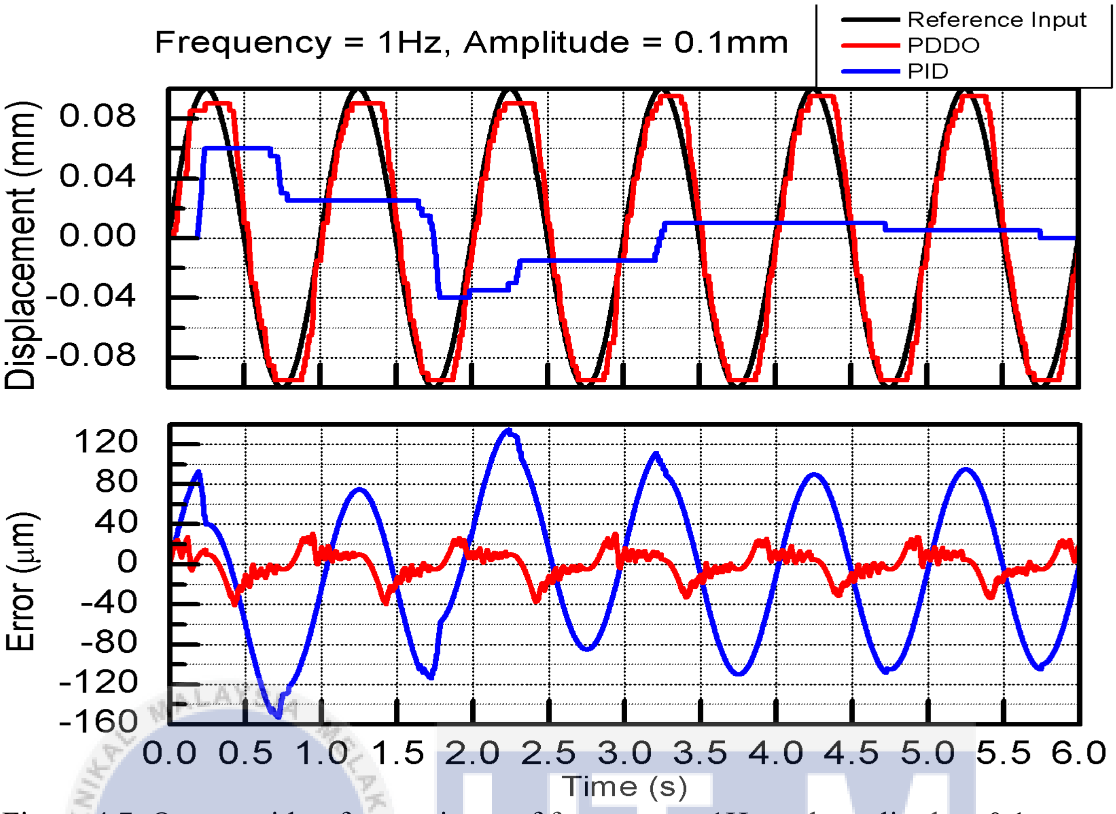

Figure 4.7: Output with reference input of frequency = 1Hz and amplitude = 0.1mm

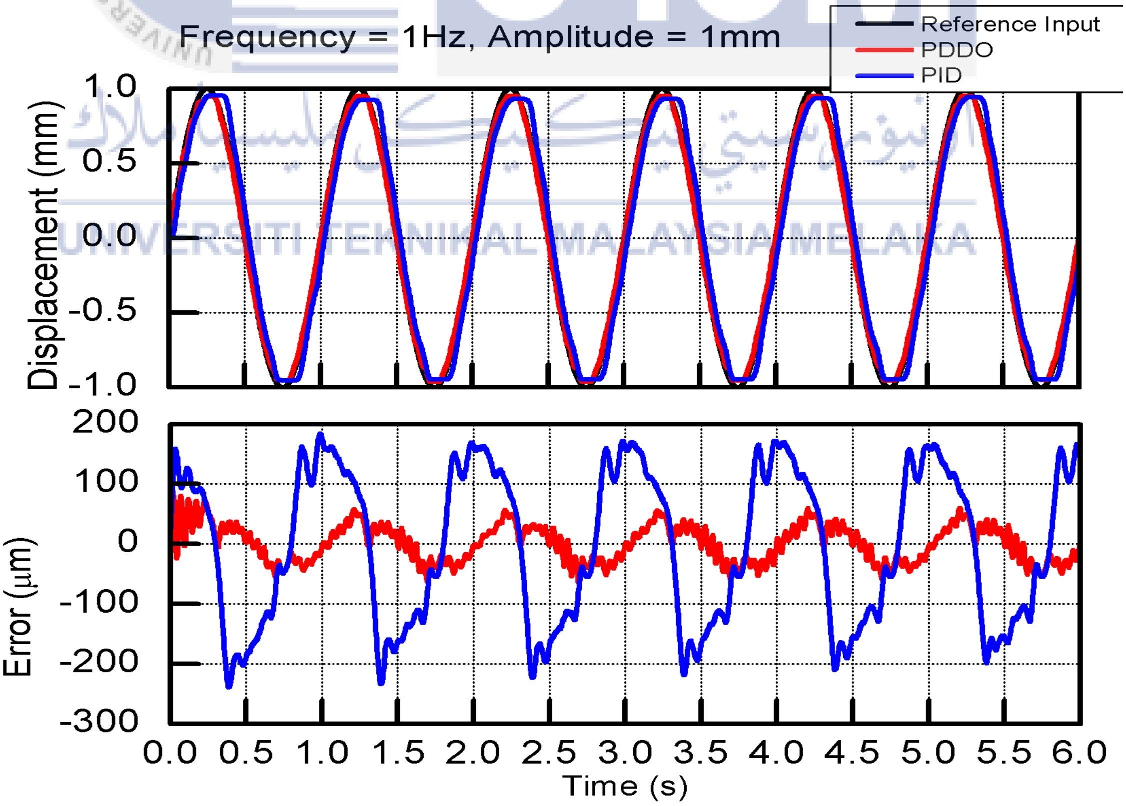

Figure 4.8: Output with reference input of frequency = 1Hz and amplitude = 1mm

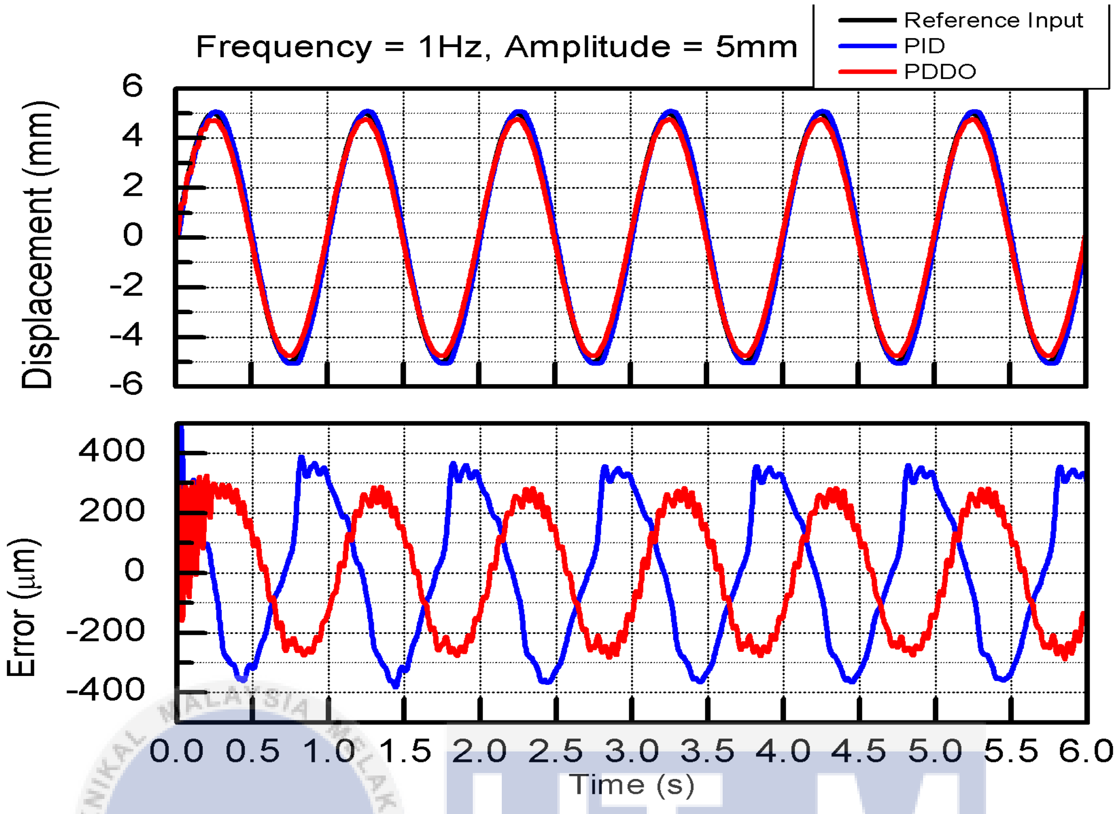

Figure 4.9: Output with reference input of frequency = 1Hz and amplitude = 5mm

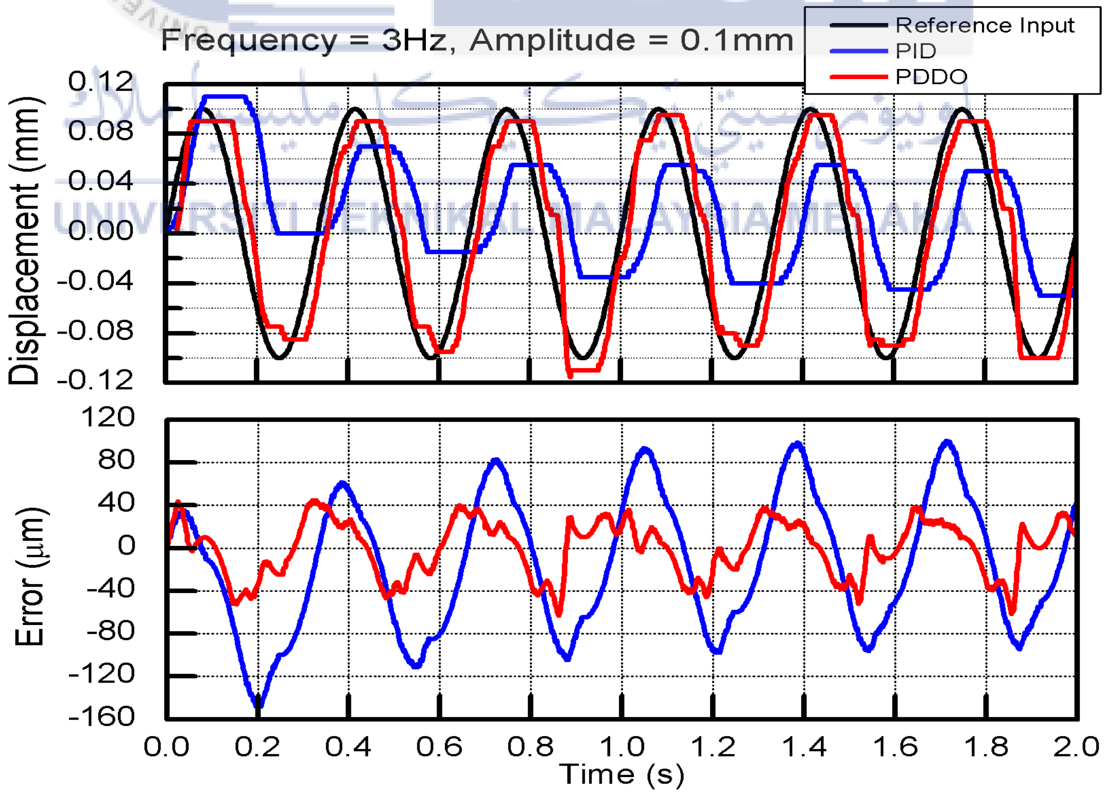

Figure 4.10: Output with reference input of frequency = 3Hz and amplitude = 0.1mm

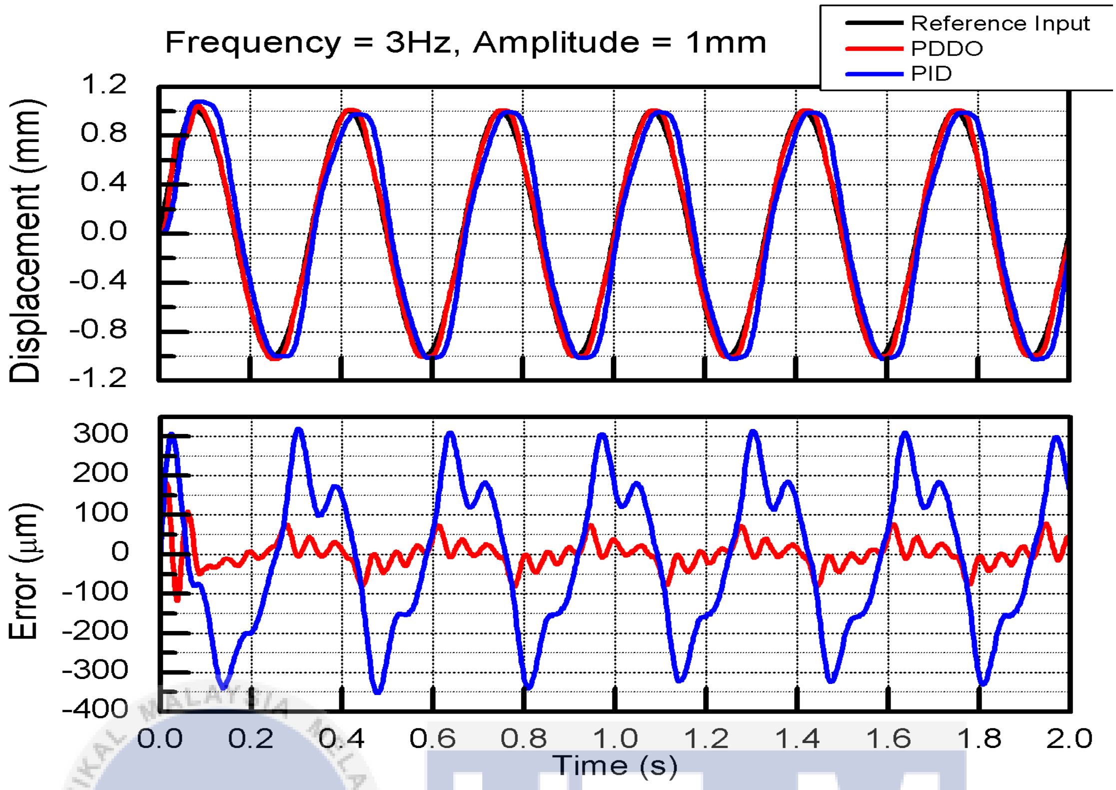

Figure 4.11: Output with reference input of frequency = 3Hz and amplitude = 1mm

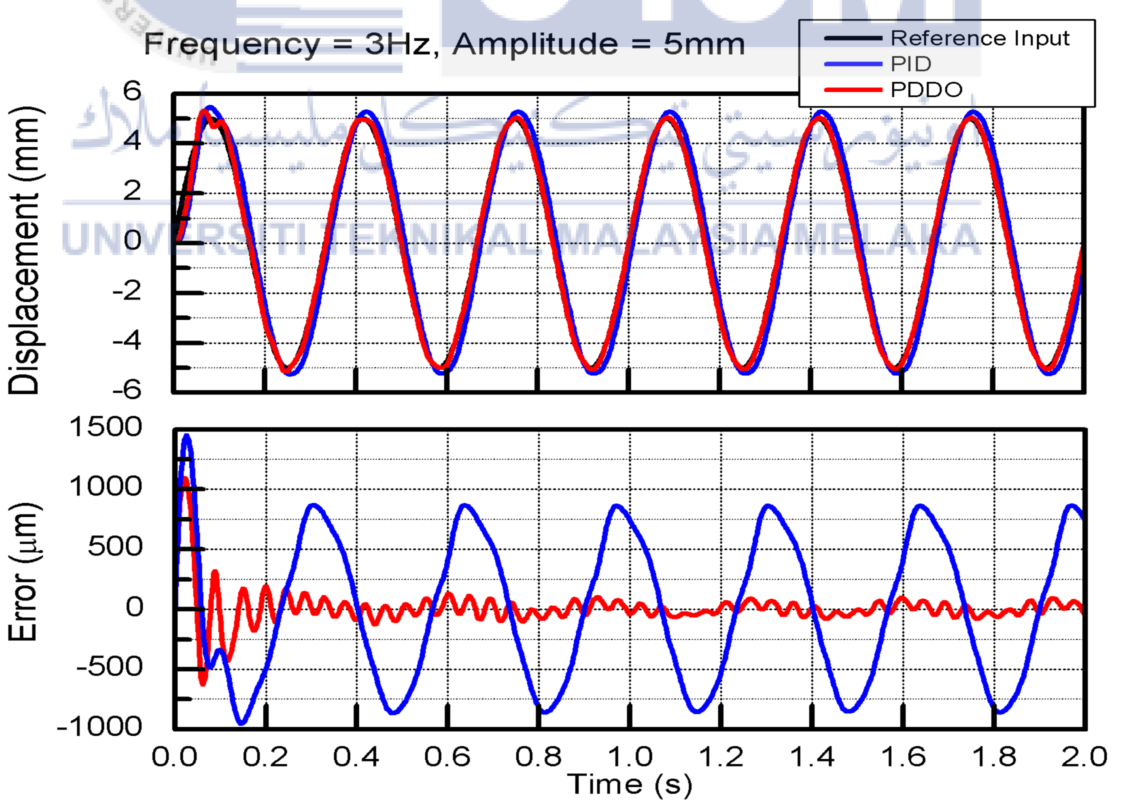

Figure 4.12: Output with reference input of frequency = 3Hz and amplitude = 5mm

Table 4.1: Average error and standard deviation for 10 repeated experiments

Controller		PDDO		PID	
Frequency (Hz)	Displacement (mm)	Average Error (µm)	Standard Deviation (µm)	Average Error (µm)	Standard Deviation (µm)
0.1	0.1	19.82	6.94	55.57	9.61
	1	89.72	8.59	156.67	12.75
	5	399.77	4.55	163.56	19.94
1	0.1	38.37	3.59	121.72	9.96
	1	60.55	4.95	228.37	9.58
	5	295.38	3.62	391.41	13.97
3	0.1	50.07	5.46	105.57	9.62
	1	79.33	2.44	342.58	16.76
	5	129.28	8.46	874.02	6.72

When input frequency is set as 0.1Hz, PID control system is still able to track the motion of the reference input though exhibiting large tracking error. As the frequency increases, the performance of PID controller deteriorates. This observation is explained through the existence of viscous friction in the system. Since PID controller is very sensitive towards the parameter variations in the plant, thus it is not able to adapt to the changes of velocity that affects the viscous friction in the system. At frequency = 3Hz, hysteresis of movement occurred in the PID controlled system.

Unlike PID controller, PDDO is capable of estimating such disturbances and therefore corrects the trajectory of the mechanism. When all the experiments are repeated for 10 times, the ball screw shaft and ball bearings are subjected to wear and tear. This condition will alter the dynamics of the mechanism in long term. The PID controller cannot adapt itself to these changes and therefore has larger standard deviations. Differ from PID controller; the PDDO has higher consistency in its tracking performance as it is less sensitive towards such conditions. Thus, PDDO is said to have higher adaptability than the PID controller.

Based on the outputs, it is seen that the controlled systems experience glitches when the mechanism changes direction of motion. This phenomenon is caused by the stiffness of the mechanism that resists the changes of direction.

4.3 Robustness on Mass Variation

In this project, loads are added to the ball screw mechanism to observe the system robustness under mass variation. Mass loads of 1kg, 3kg and 5kg are added to the ball screw mechanism respectively. The tracking performance of the PDDO and PID control system are validated with sinusoidal input of frequency = 0.1Hz, 1Hz and 3Hz and amplitude of 0.1mm, 1mm and 5mm. The average error of the 27 experiments is presented in Table 4.2 while Table 4.3 shows the standard deviation of the controllers with the 3 load mass considered. By referring to Table 4.2 and Table 4.3, it can be seen that a PDDO has smaller tracking error compare to PID. On top of that, PDDO has higher capability of following the trajectory of the input despite changes of load. The complete system responses with mass variations are shown in Figure 4.14 to Figure 4.41.

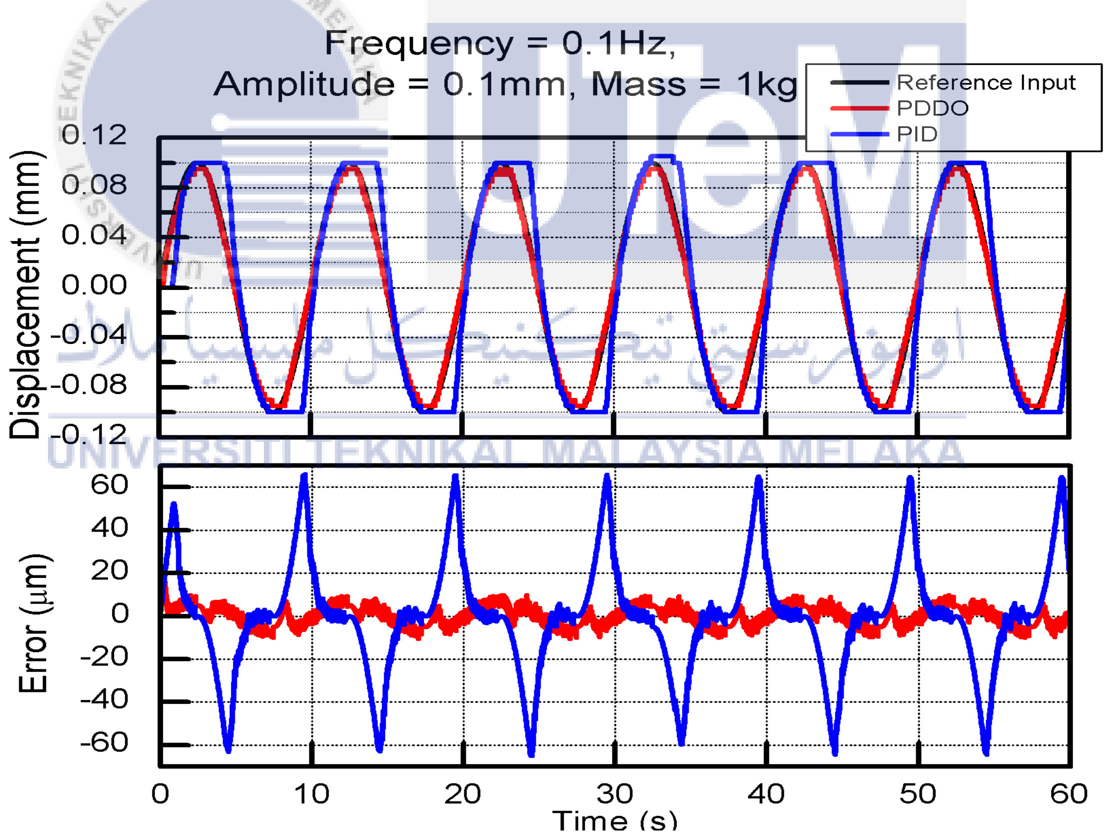

Figure 4.13: Output with frequency = 0.1Hz, amplitude = 0.1mm and load = 1kg

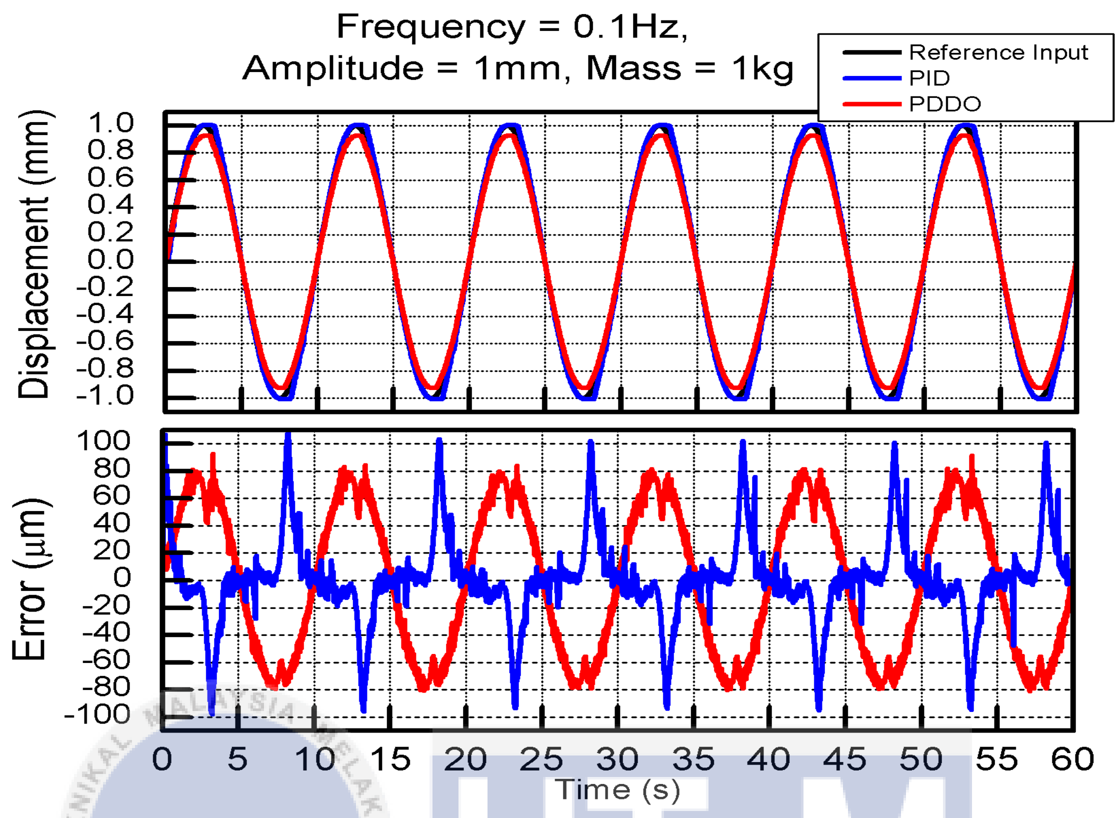

Figure 4.14: Output with frequency = 0.1Hz, amplitude = 1mm and load = 1kg

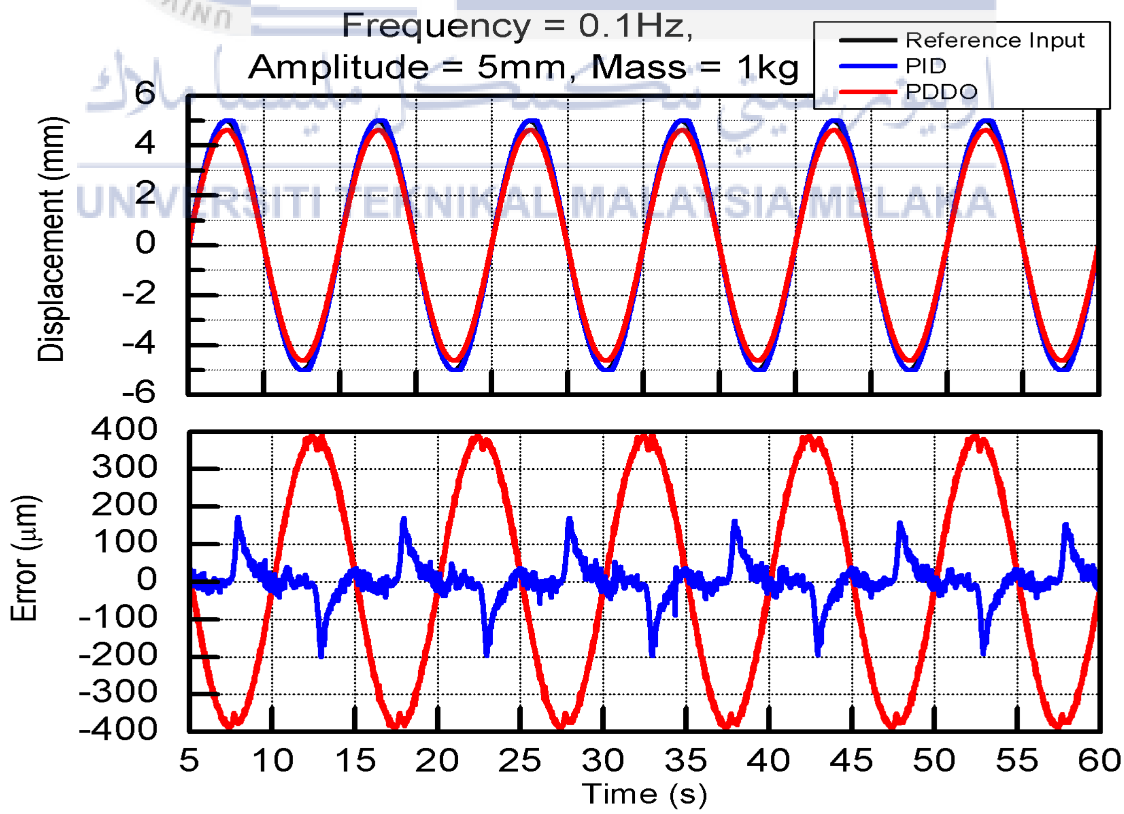

Figure 4.15: Output with frequency = 0.1Hz, amplitude = 5mm and load = 1kg

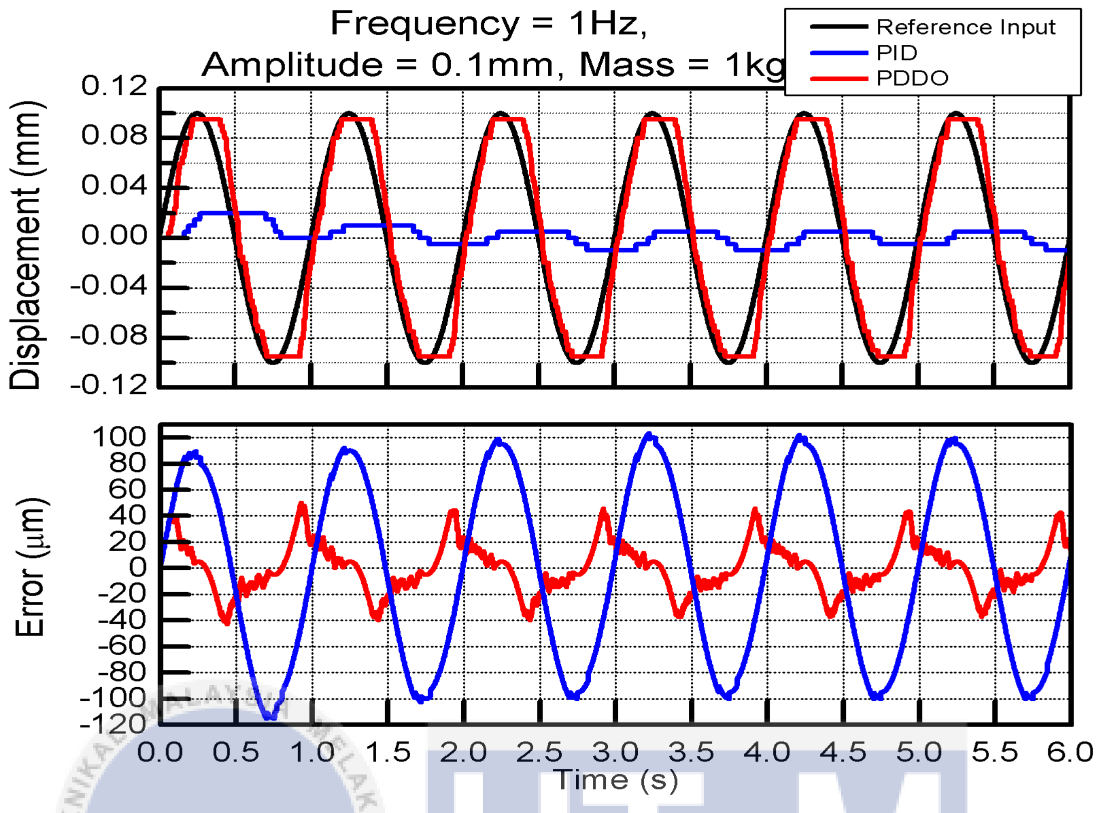

Figure 4.16: Output with frequency = 1Hz, amplitude = 0.1mm and load = 1kg

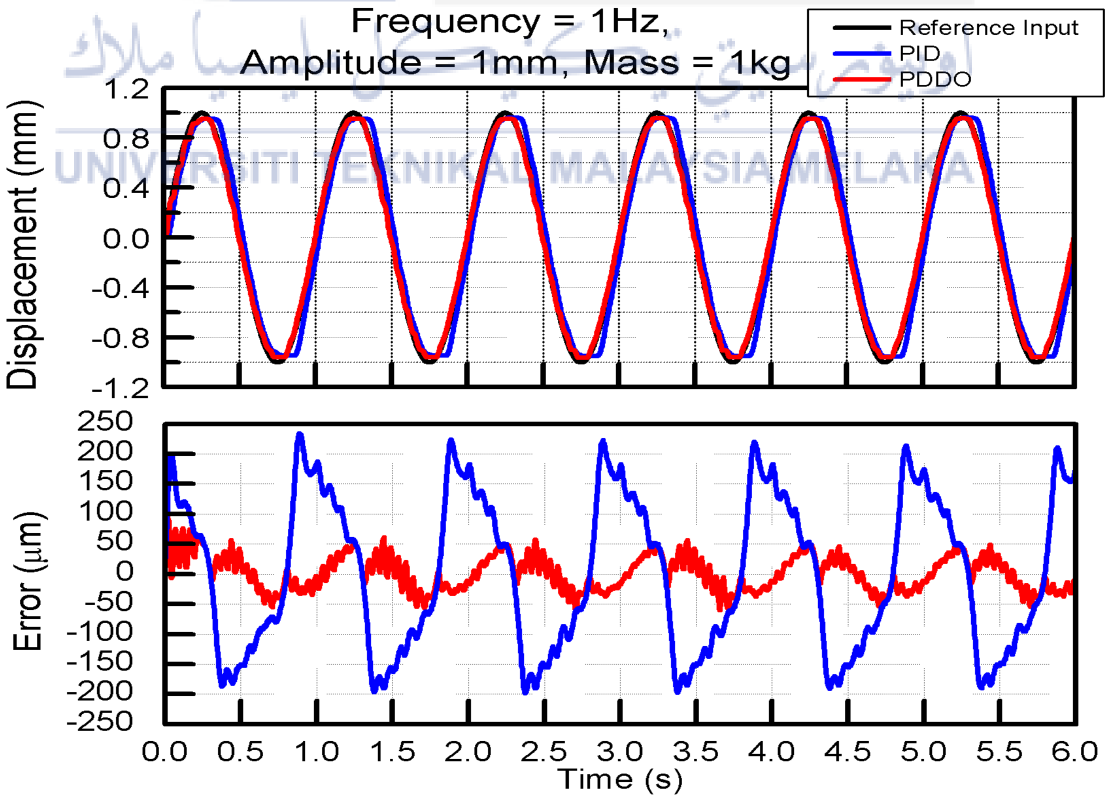

Figure 4.17: Output with frequency = 1Hz, amplitude = 1mm and load = 1kg

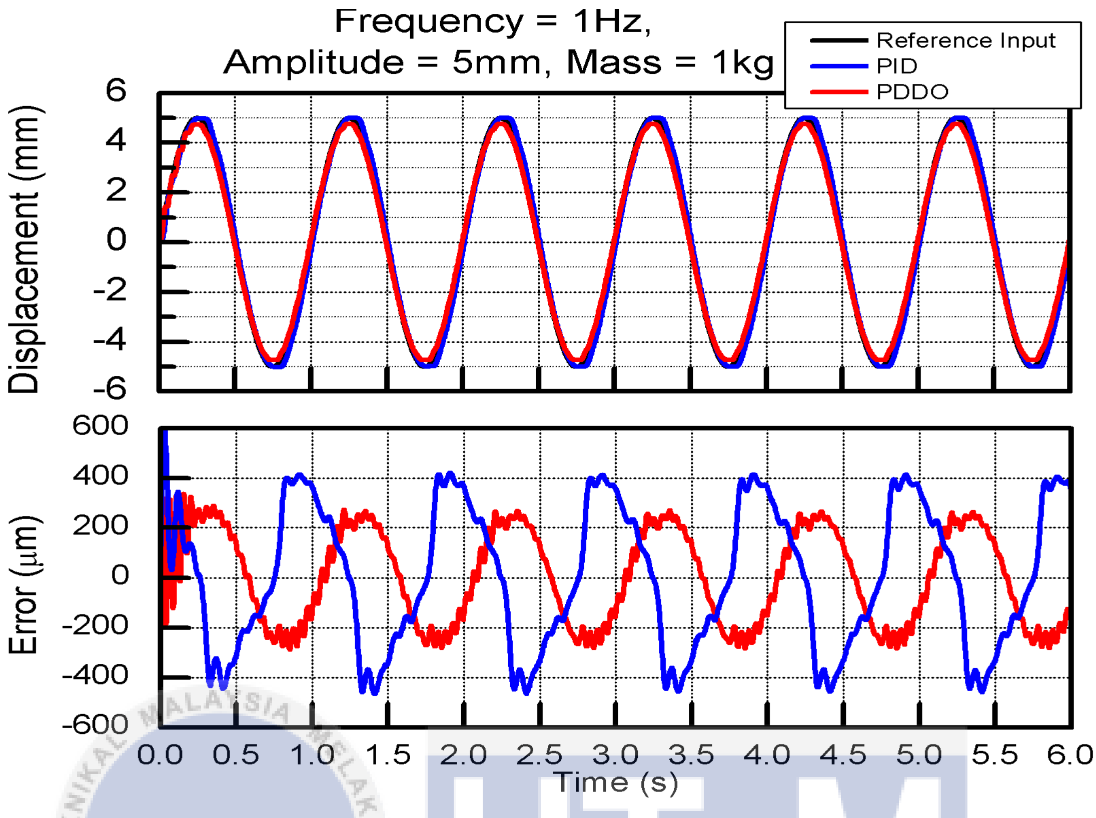

Figure 4.18: Output with frequency = 1Hz, amplitude = 5mm and load = 1kg

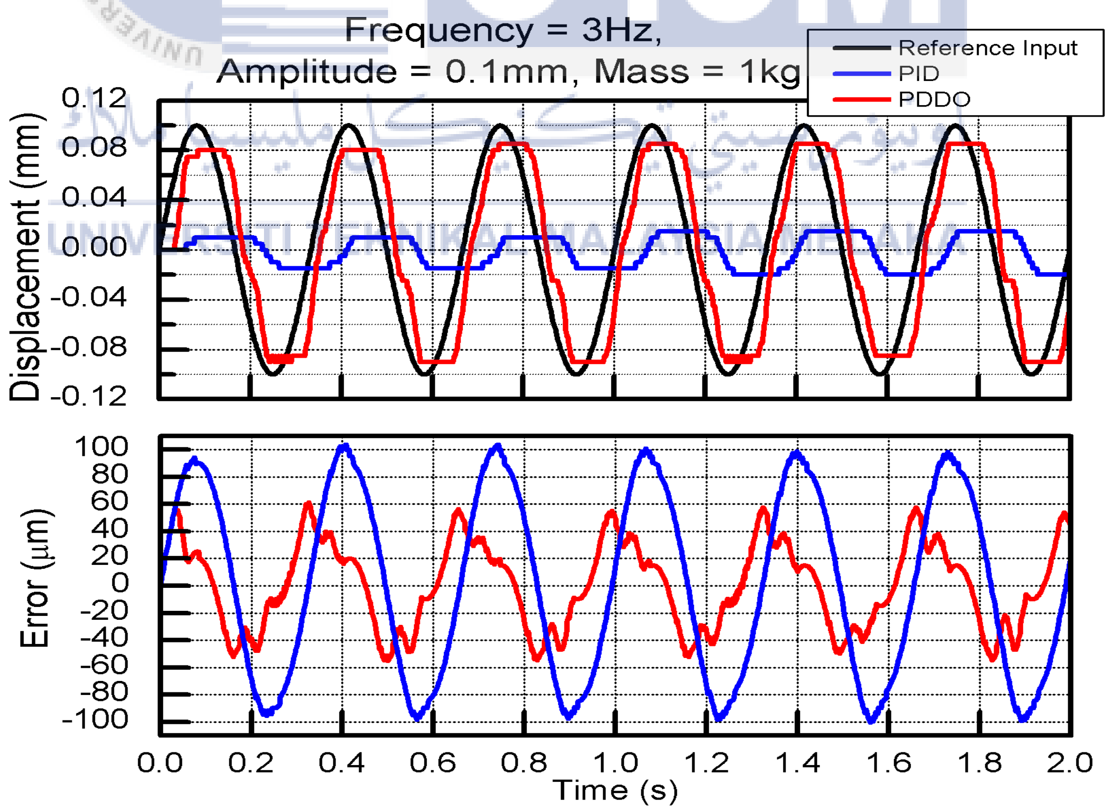

Figure 4.19: Output with frequency = 3Hz, amplitude = 0.1mm and load = 1kg

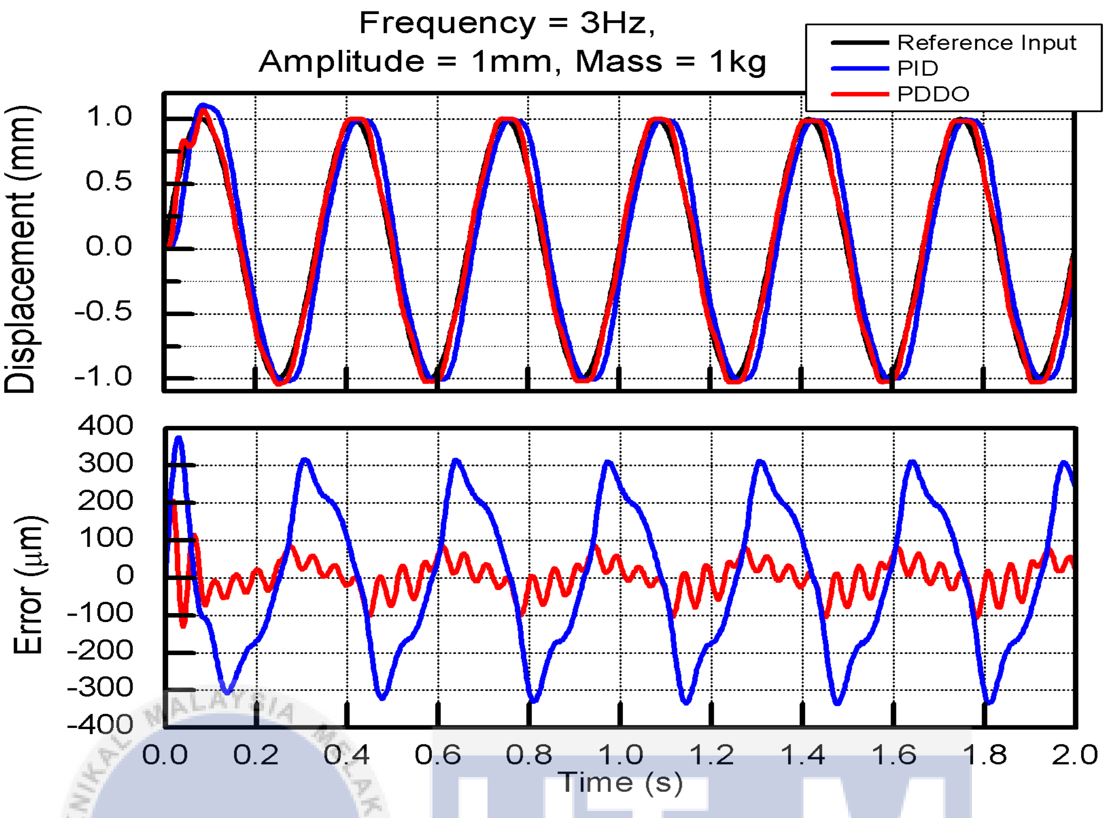

Figure 4.20: Output with frequency = 3Hz, amplitude = 1mm and load = 1kg

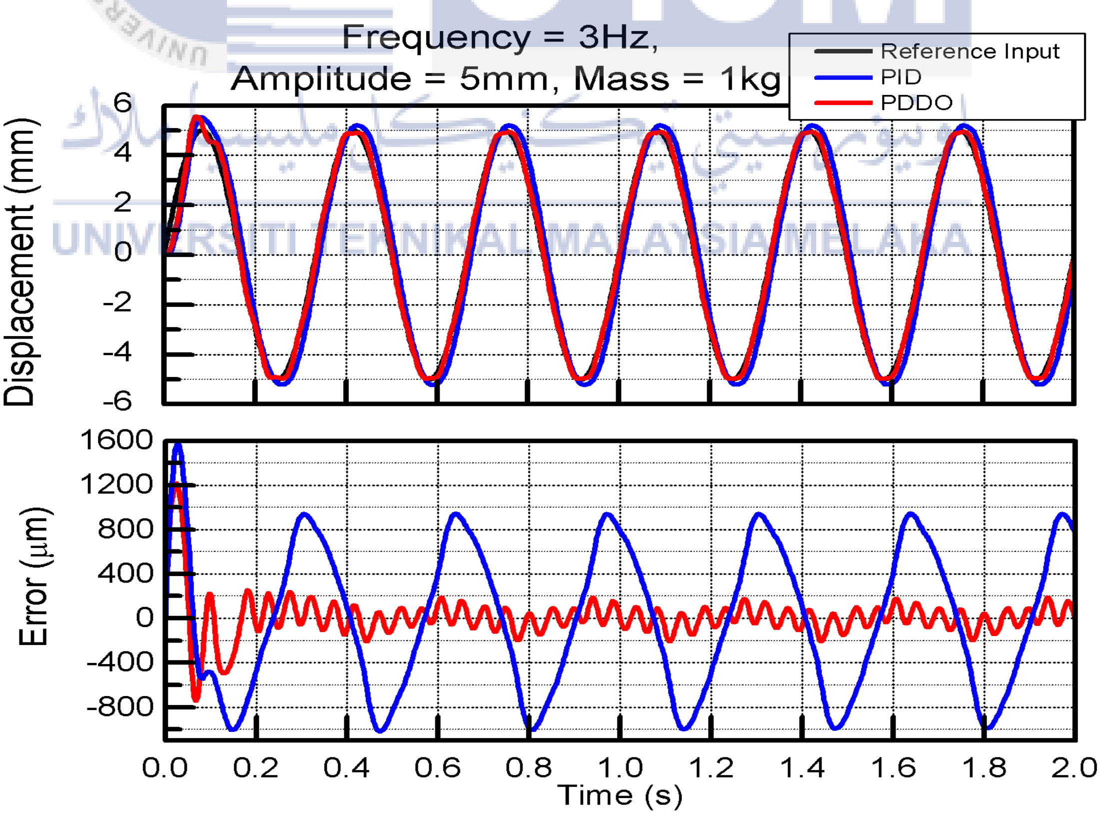

Figure 4.21: Output with frequency = 3Hz, amplitude = 5mm and load = 1kg

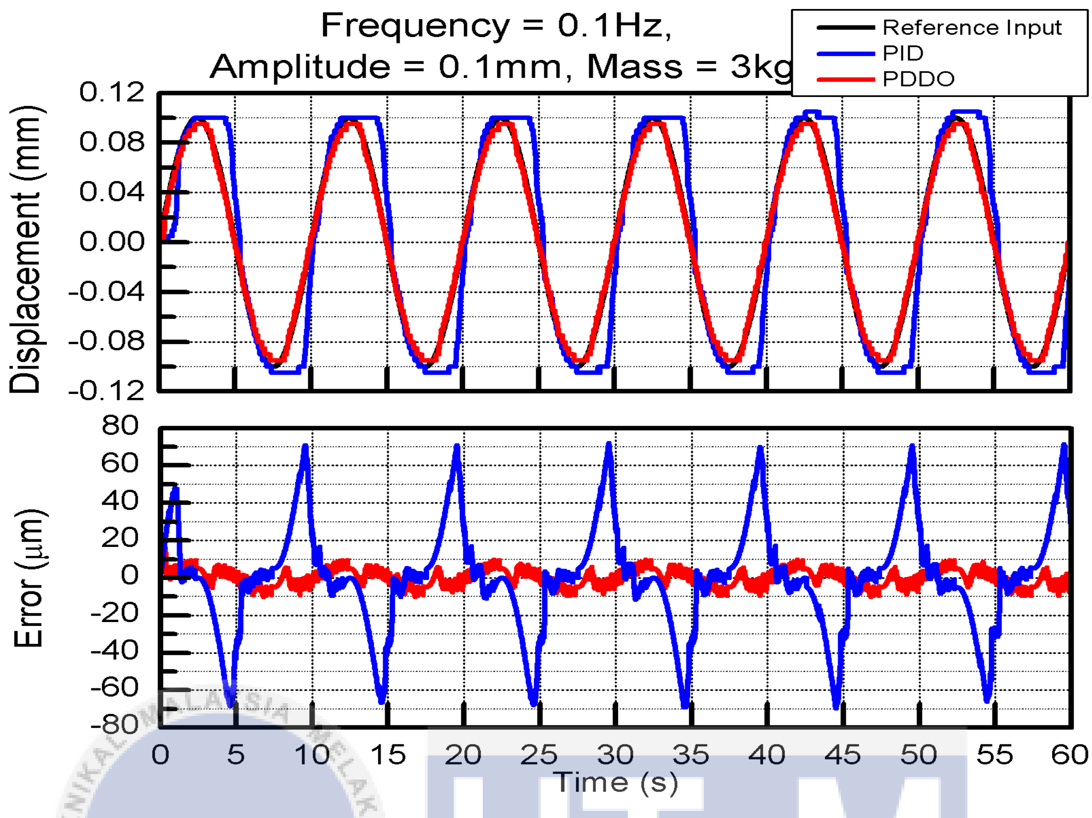

Figure 4.22: Output with frequency = 0.1Hz, amplitude = 0.1mm and load = 3kg

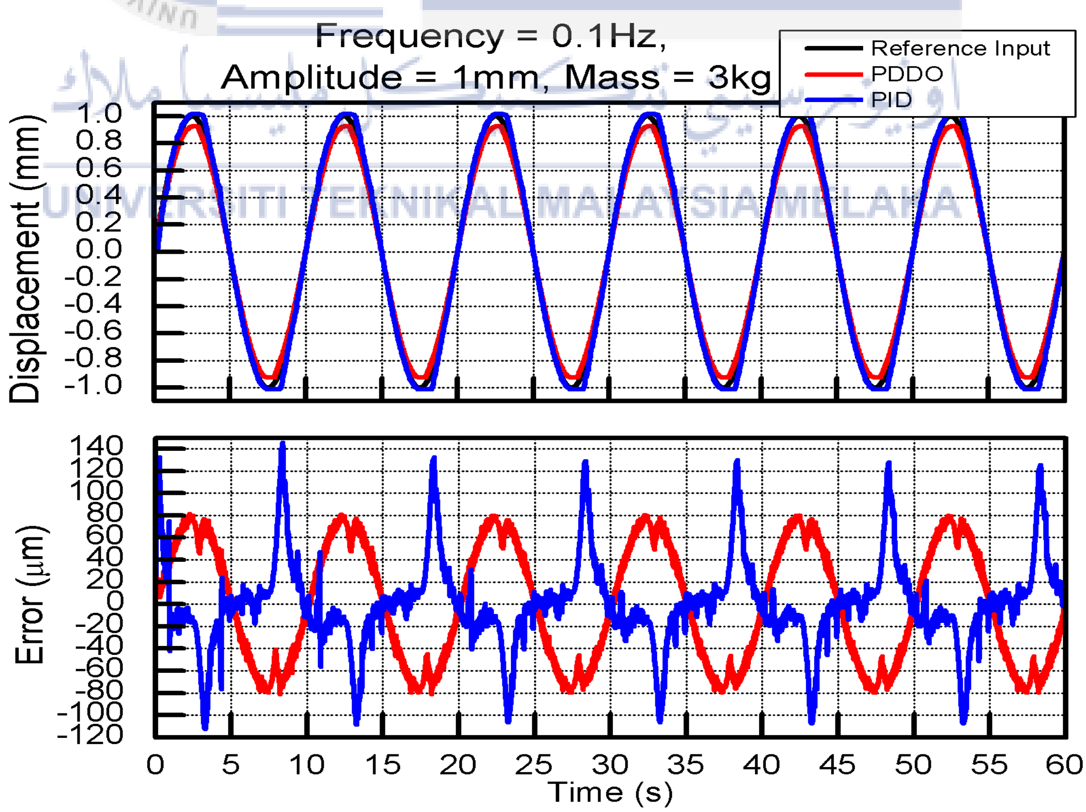

Figure 4.23: Output with frequency = 0.1Hz, amplitude = 1mm and load = 3kg

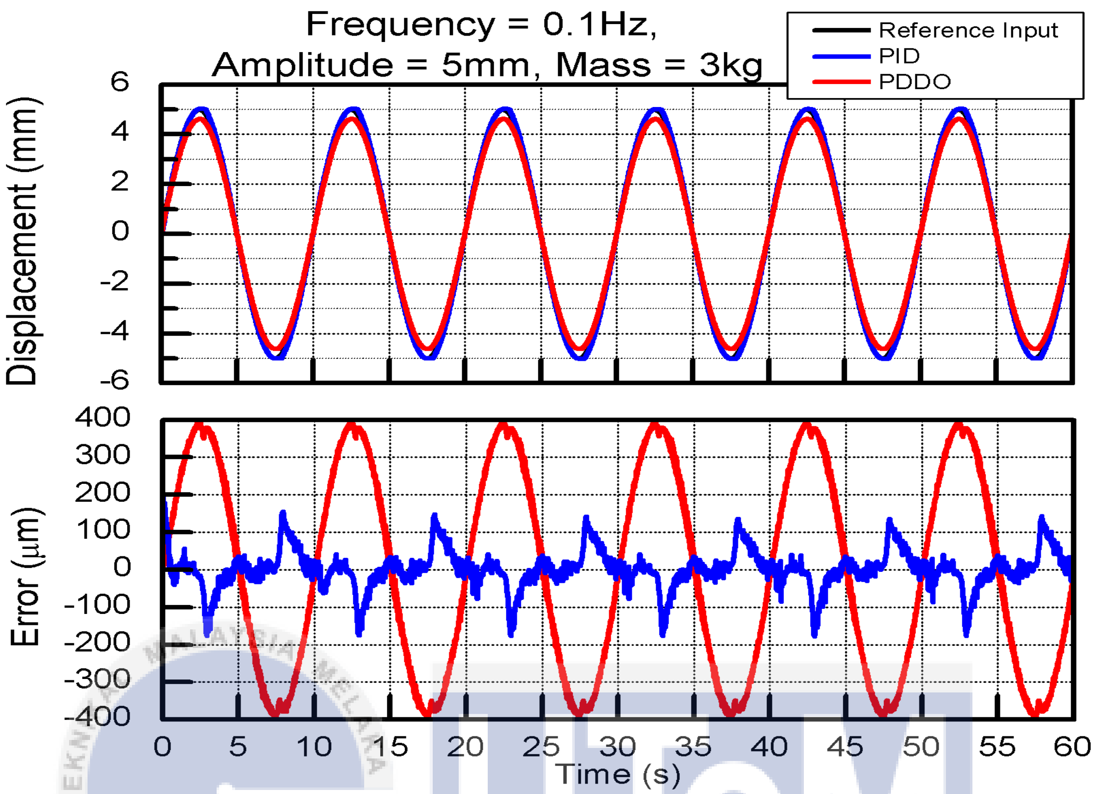

Figure 4.24: Output with frequency = 0.1Hz, amplitude = 5mm and load = 3kg

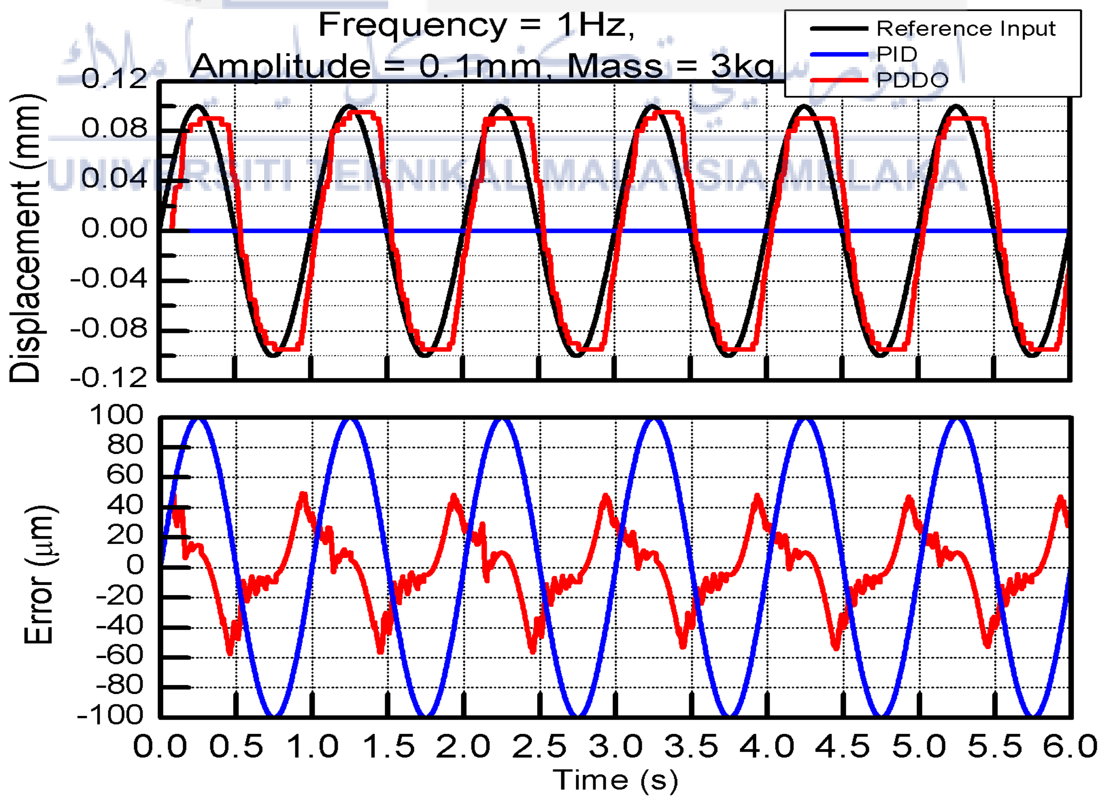

Figure 4.25: Output with frequency = 1Hz, amplitude = 0.1mm and load = 3kg

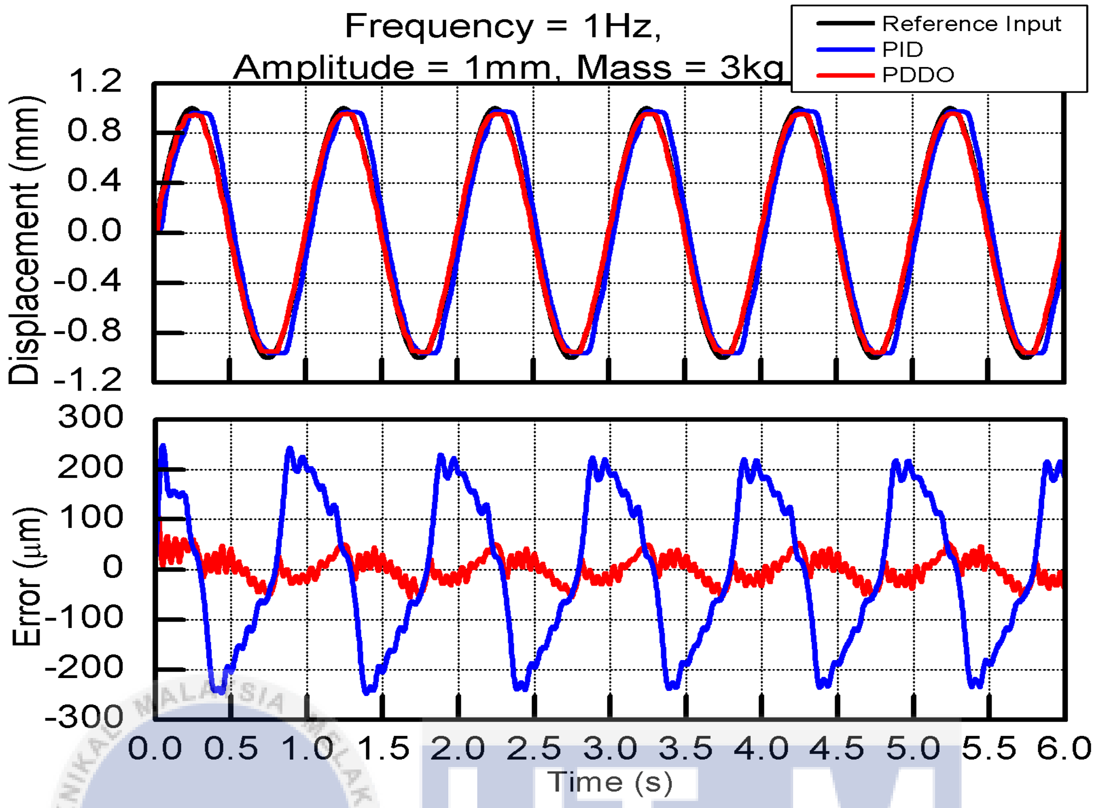

Figure 4.26: Output with frequency = 1Hz, amplitude = 1mm and load = 3kg

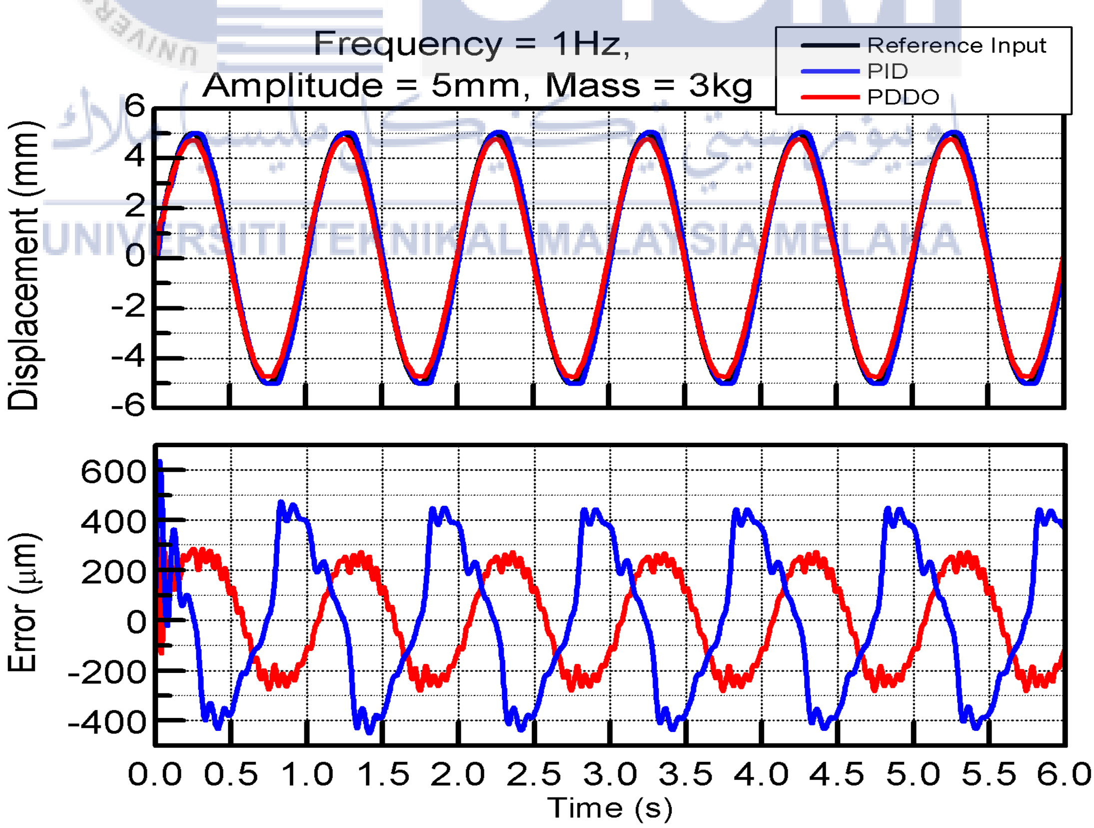

Figure 4.27: Output with frequency = 1Hz, amplitude = 5mm and load = 3kg

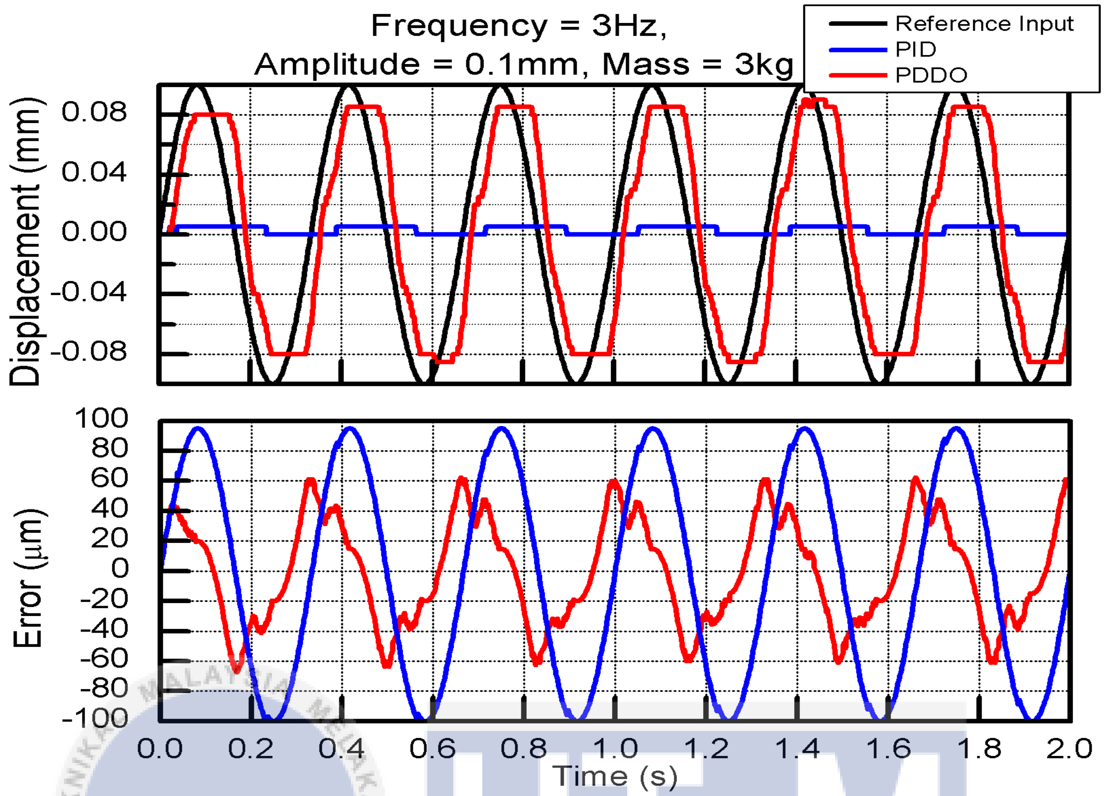

Figure 4.28: Output with frequency = 3Hz, amplitude = 0.1mm and load = 3kg

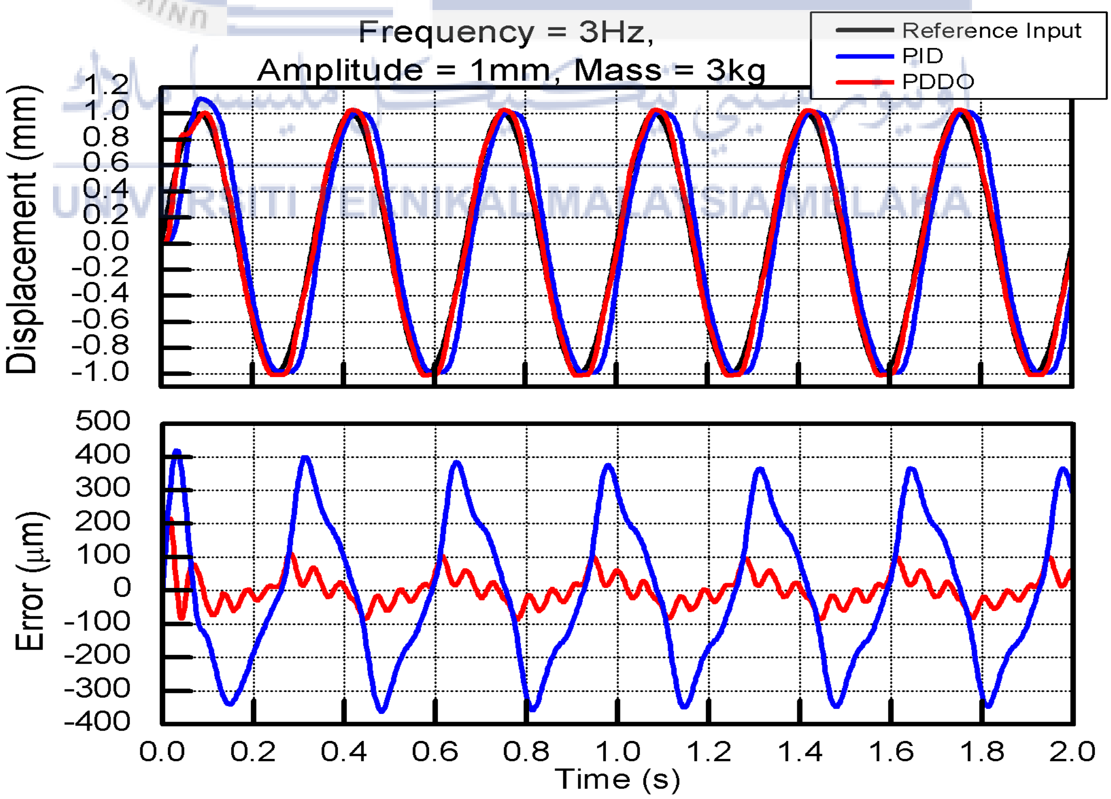

Figure 4.29: Output with frequency = 3Hz, amplitude = 1mm and load = 3kg

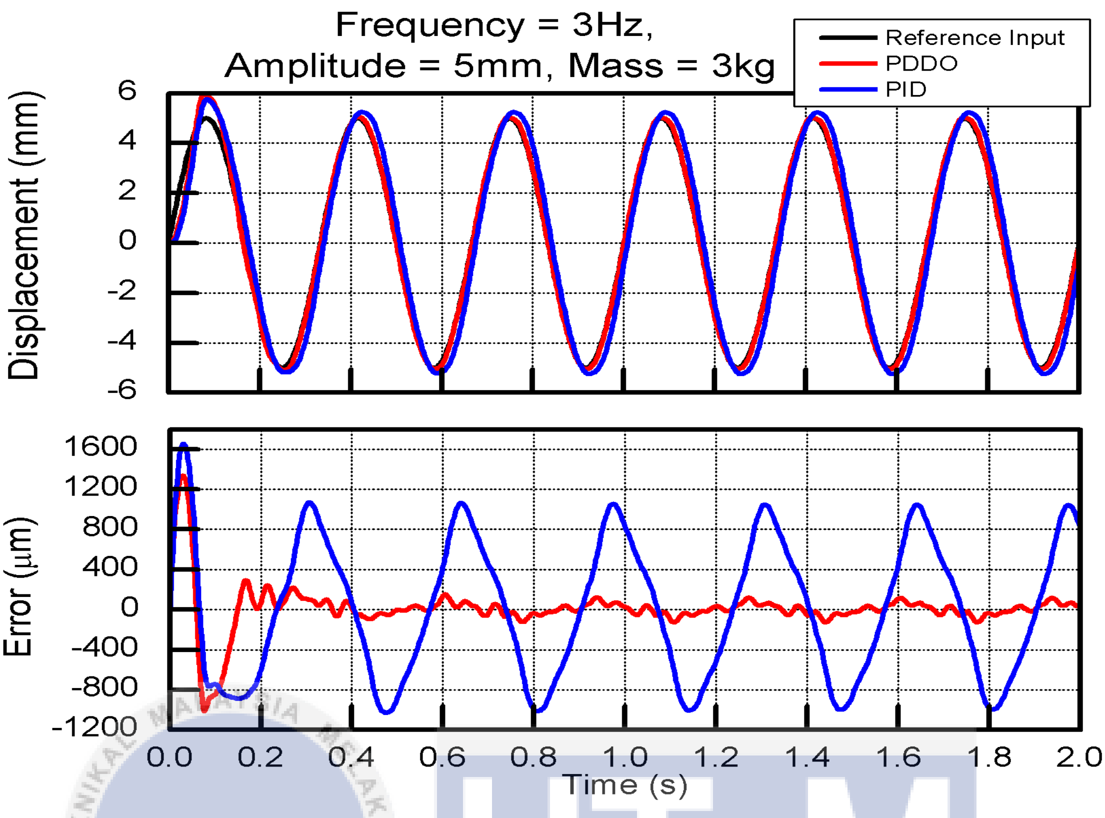

Figure 4.30: Output with frequency = 3Hz, amplitude = 5mm and load = 3kg

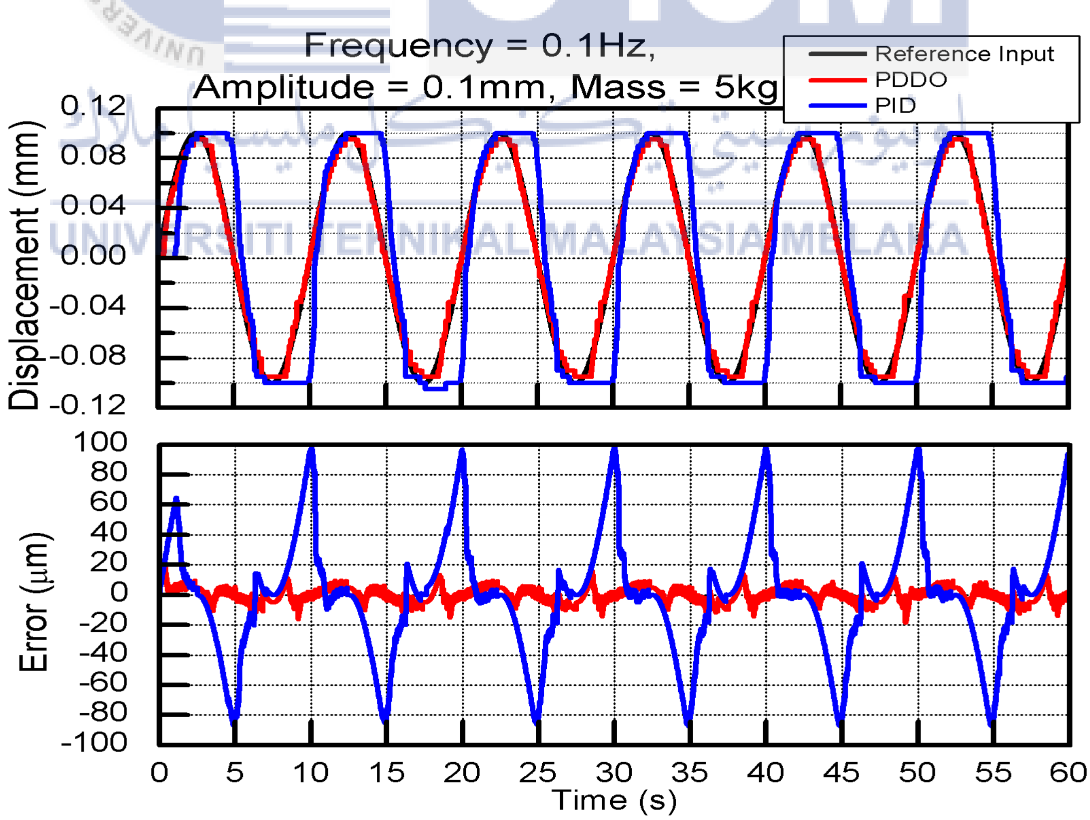

Figure 4.31: Output with frequency = 0.1Hz, amplitude = 0.1mm and load = 5kg

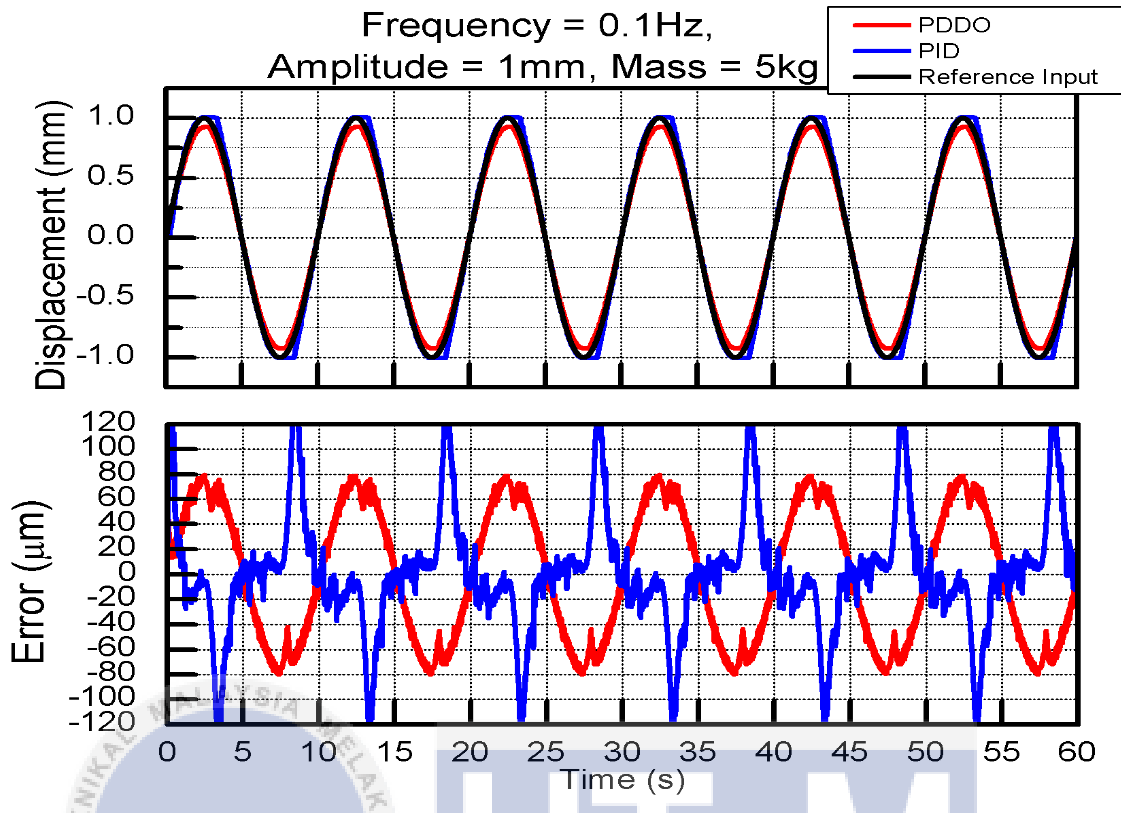

Figure 4.32: Output with frequency = 0.1Hz, amplitude = 1mm and load = 5kg

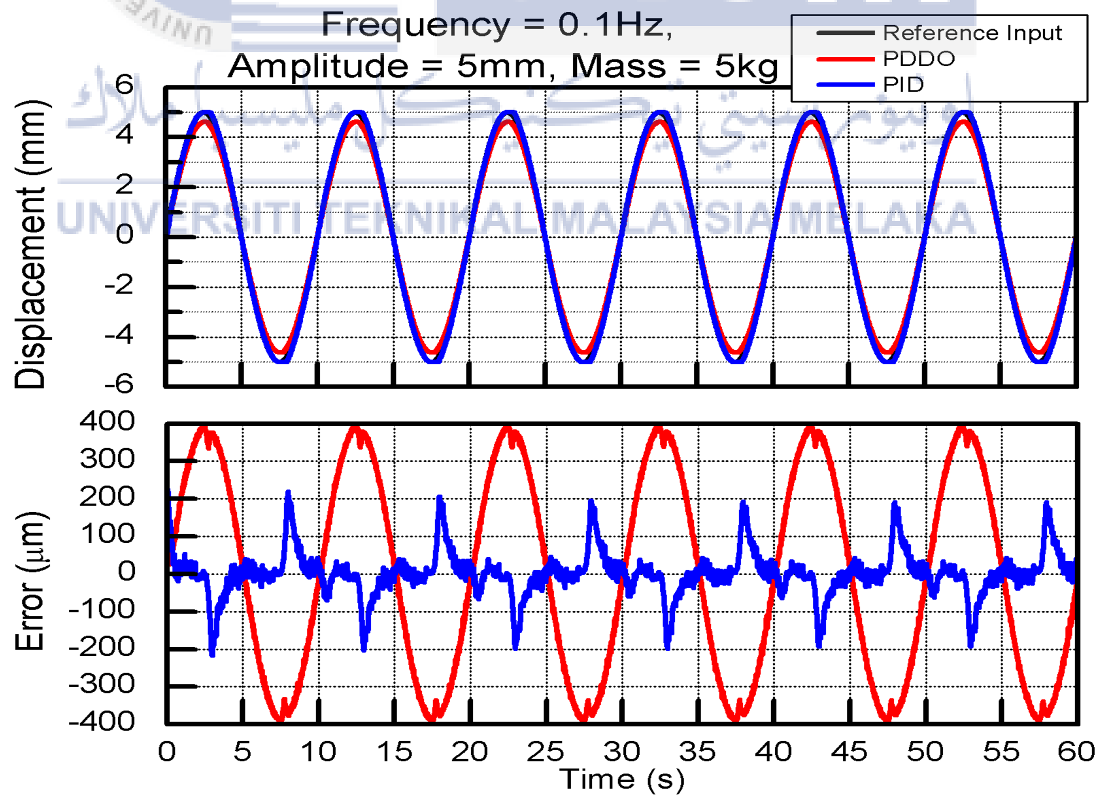

Figure 4.33: Output with frequency = 0.1Hz, amplitude = 5mm and load = 5kg

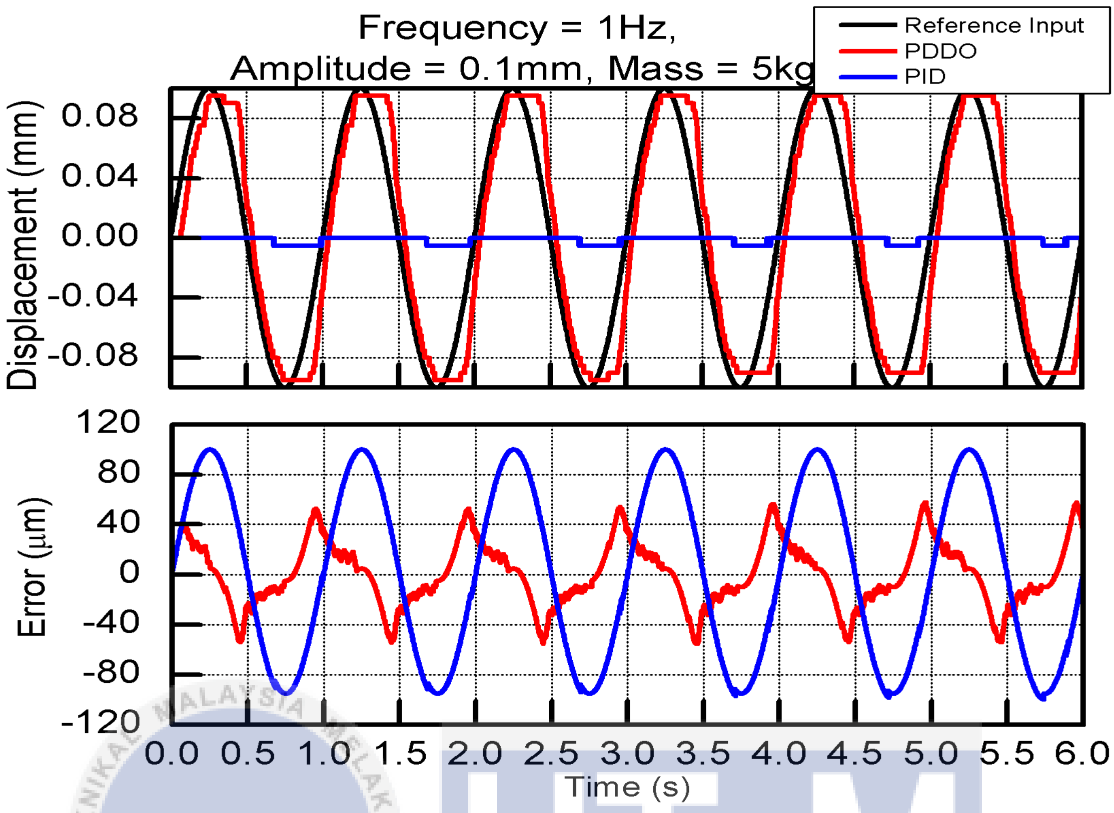

Figure 4.34: Output with frequency = 1Hz, amplitude = 0.1mm and load = 5kg

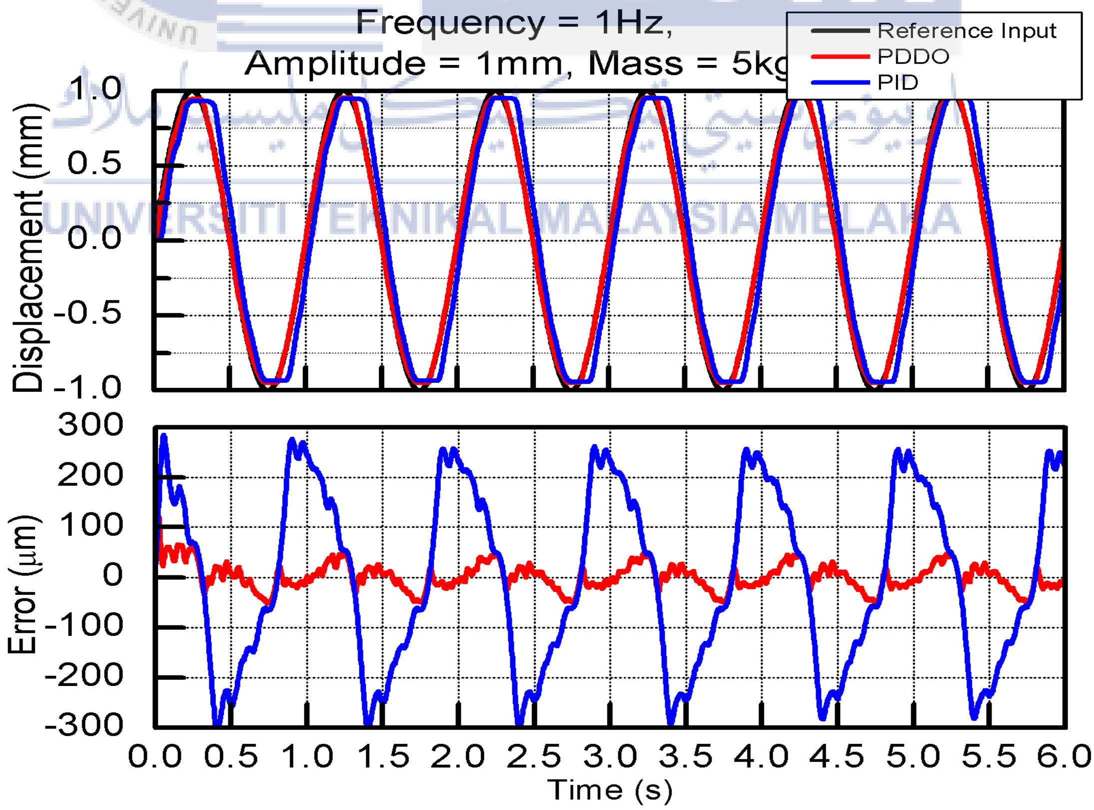

Figure 4.35: Output with frequency = 1Hz, amplitude = 1mm and load = 5kg

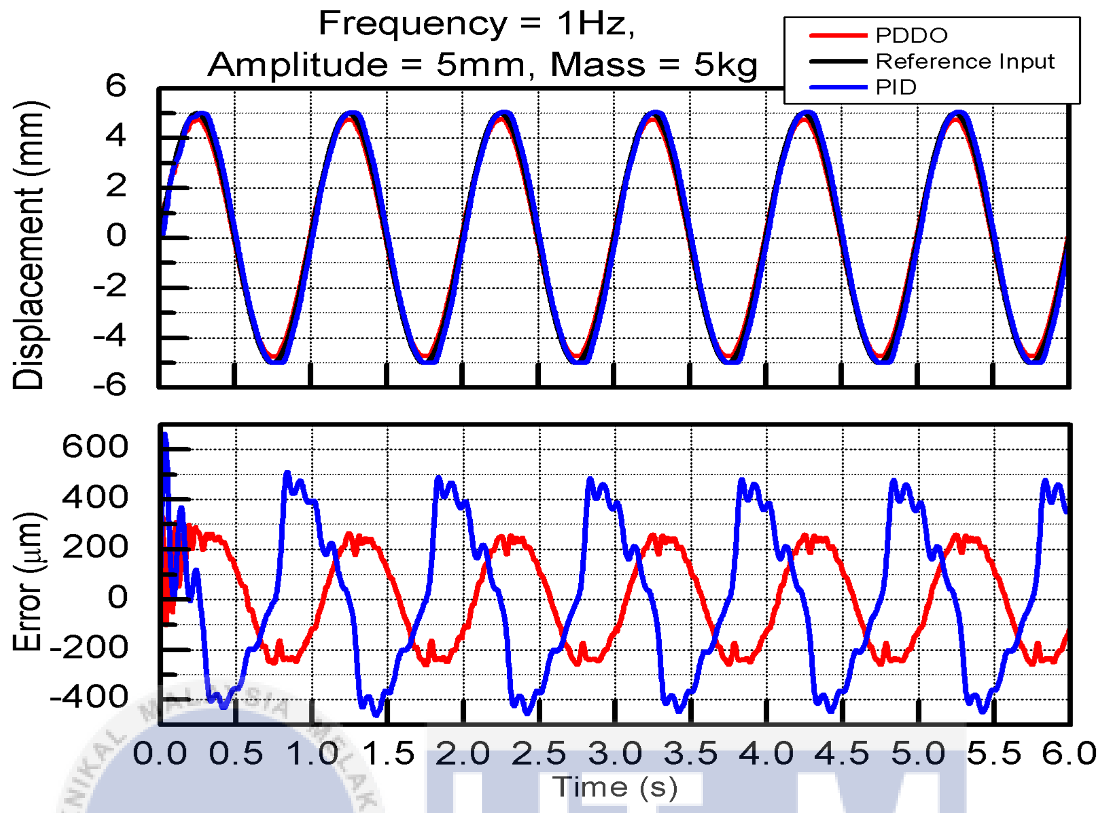

Figure 4.36: Output with frequency = 1Hz, amplitude = 5mm and load = 5kg

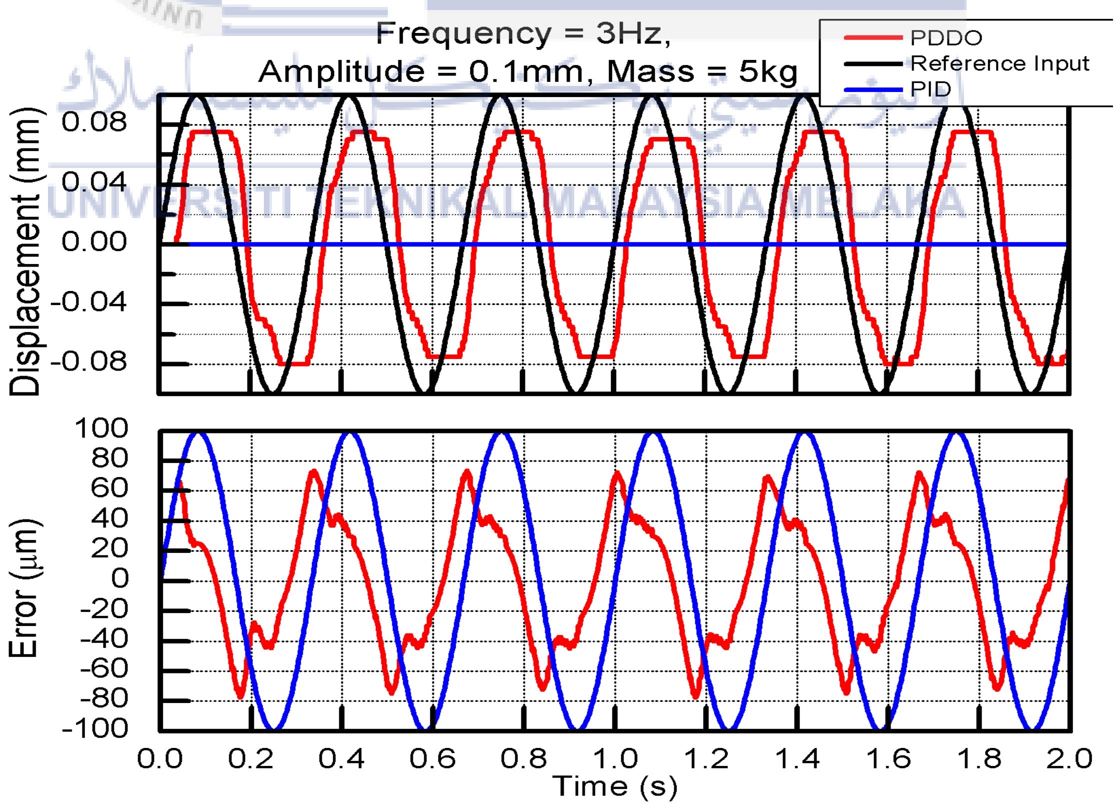

Figure 4.37: Output with frequency = 3Hz, amplitude = 0.1mm and load = 5kg

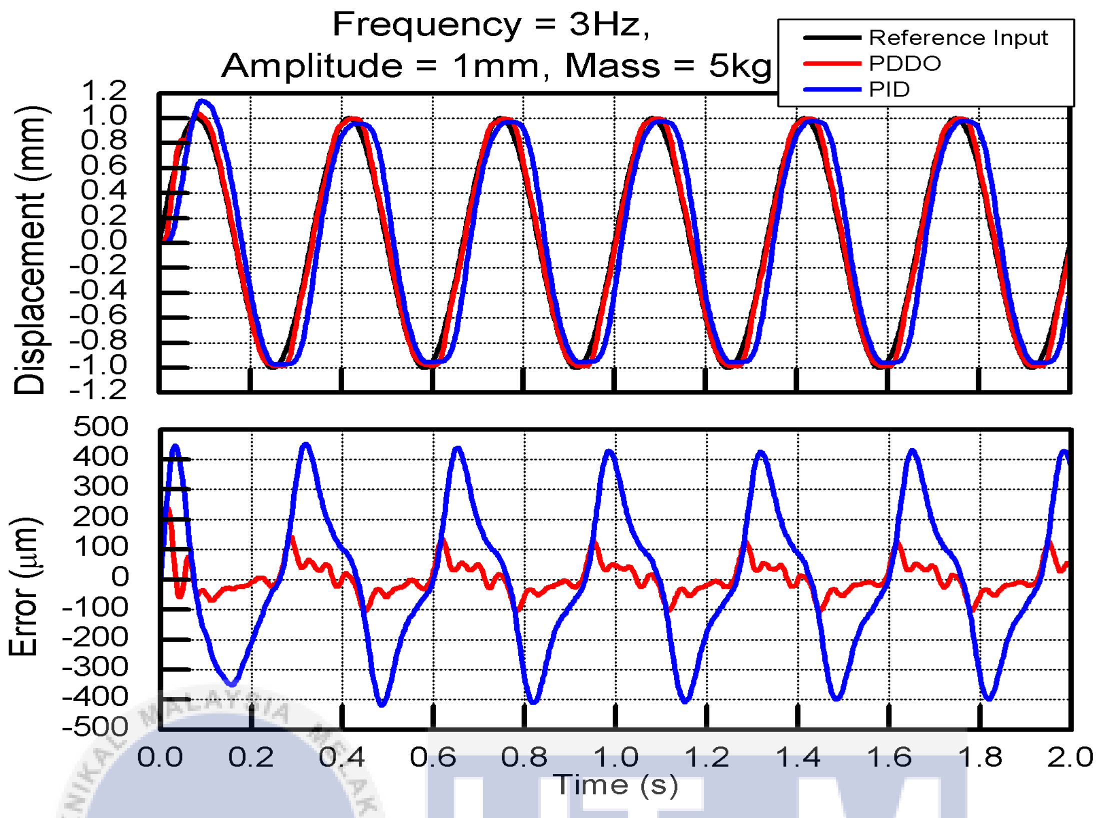

Figure 4.38: Output with frequency = 3Hz, amplitude = 1mm and load = 5kg

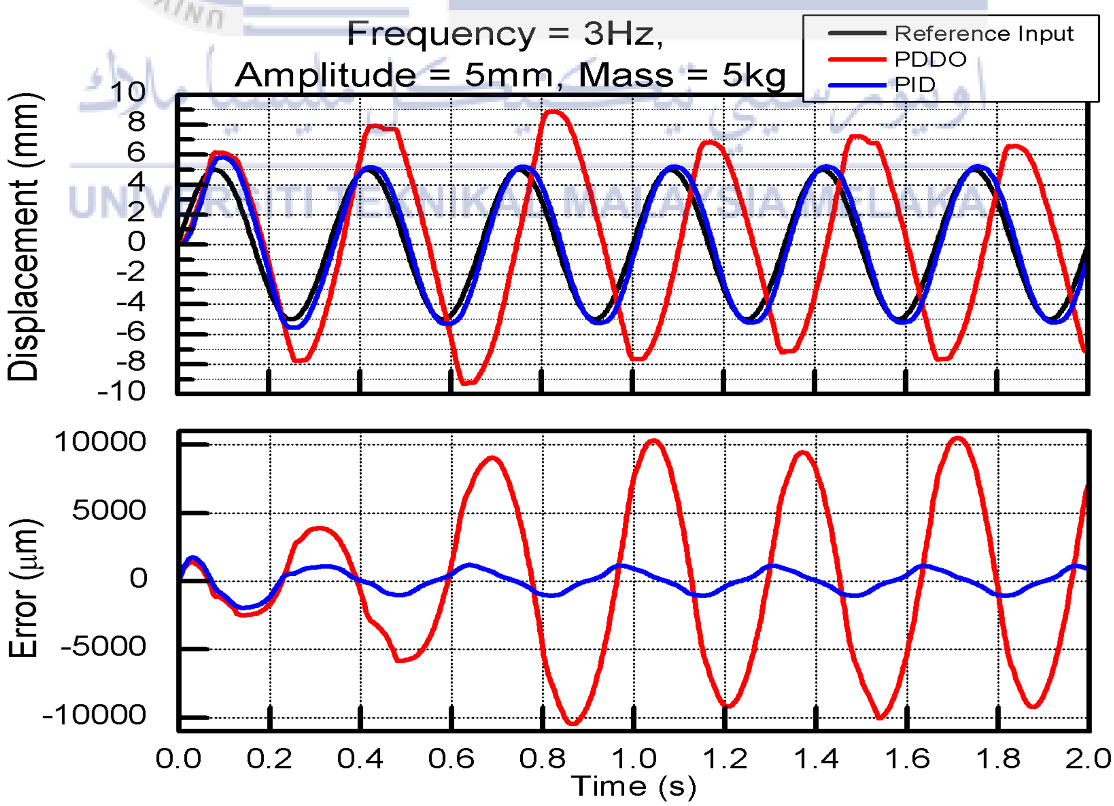

Figure 4.39: Output with frequency = 3Hz, amplitude = 5mm and load = 5kg

Table 4.2: Average error in each experiment set with mass variations in different frequencies and amplitudes

Controller			PDDO	PID
Mass (kg)	Frequency (Hz)	Displacement (mm)	Average Error in last 5 cycles (µm)	
1	0.1	0.1	9.93	64.38
		1	82.39	98.14
		5	390.91	180.16
	1	0.1	41.40	99.96
		1	56.17	206.78
		5	275.23	436.93
	3	0.1	54.76	99.93
		1	92.94	322.24
		5	191.81	971.22
3	0.1	0.1	9.80	69.55
		1	80.17	118.10
		5	392.52	158.99
	1	0.1	51.39	100.00
		1	53.22	230.72
		5	276.02	442.76
	3	0.1	61.75	97.49
		1	92.27	362.78
		5	122.59	1031.30
5	0.1	0.1	14.50	92.34
		1	79.55	146.20
		5	390.87	196.46
	1	0.1	55.19	98.54
		1	48.43	275.59
		5	260.60	467.81
	3	0.1	73.52	99.99
		1	116.34	418.94
		5	9250.70	1090.00

Table 4.3: Standard deviation of PDDO and PID with change of load

Controller	PDDO			PID		
Displacement (mm)	0.1	1	5	0.1	1	5
Frequency (Hz)	Standard Deviation (µm)					
0.1	2.67	1.49	0.94	14.87	0.83	18.79
1	7.12	3.91	8.68	0.83	34.93	16.41
3	9.48	13.71	5250.25	1.42	48.56	59.39

By comparing Table 4.1 and Table 4.2, it can be seen that the differences of tracking error for PDDO at different mass are much smaller than the PID controller. It can also be seen from Table 4.3 that despite the increment of load mass, the tracking error of PDDO does not deviate much though the errors are still considerably large. The PID controller, on the other hand, has large error variations when the load in the mechanism is increased. From these observations, the PDDO is said to be robust against variation of mass as it is insensitive towards such changes in the system.

It is observed from Figure 4.41 that the PDDO controlled system becomes unstable when the mass is increased to 5kg. This condition might arise due to the large difference of nominal plant and real plant model since the mass changes is too large as mentioned in Section 2.3.

CHAPTER 5

CONCLUSION AND RECOMMENDATION

In conclusion, the second order model transfer function of the ball screw mechanism is obtained using non-linear least square (NLLS) method. The PDDO was designed and its performances were evaluated through tracking test and robustness test. Based on the experiments, the characteristics of PDDO are summarized as follow:

1. PDDO has higher tracking performances as compared to PID controller. Since a PID controller has to be tuned frequently due to changes of input, thus PID exhibits higher tracking error than PDDO.
2. PDDO is proven to have higher adaptability than the PID controller. A PDDO is capable of estimating the possible disturbances arise from wear and tear of the mechanism and corrects its trajectory to follow the path of reference input.
3. Though PDDO has higher tracking performance, the tracking error is still significantly large. To improve this flaw, a feedforward controller, Zero Phase Error Tracking Controller (ZPETC) is suggested to be included in the system. ZPETC is capable of eliminating the phase error that arises from the system zero that cannot be cancelled.
4. In robustness test, PDDO proved to be robust towards varying mass in the system in comparisons with PID controller. When the load is increased, the tracking errors do not differ much. However, the system becomes unstable when it is subjected to high frequency and large movement with large load. To improve this condition, the bandwidth of the low pass filter, Q(s) is suggested to be increased. Increasing the filter bandwidth will allow the system to work in a larger frequency range, but exposed to high frequency noise as well. The bandwidth, however, should not exceed the bandwidth of the plant.

In future work, a feedforward controller, ZPETC should be included to eliminate the tracking error. To reduce the effect of wear and tear, the ball screw mechanism must be lubricated frequently especially at the screw shaft. The robustness of the PDDO can be further tested with other parameter variations such as variation of disturbance signals or frictional torque.

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APPENDIX A

TRACKING ERROR OF PDDO CONTROLLER

Frequency (Hz)	Displacement (mm)	Number of Test	Absolute Peak Error (µm)	Average Peak Error(µm)	Standard Deviation (µm)
0.1	0.1	1	26.30	19.82	6.9423
		2	33.50		
		3	24.70		
		4	19.30		
		5	23.70		
		6	14.70		
		7	13.70		
		8	15.40		
		9	12.70		
		10	14.20		
	1	1	99.20	89.72	8.5949
		2	88.90		
		3	84.90		
		4	83.10		
		5	88.40		
		6	85.60		
		7	109.60		
		8	88.20		
		9	89.10		
		10	80.20		
	5	1	395.90	399.77	4.5463
		2	406.40		
		3	406.00		
		4	398.80		
		5	397.90		
		6	395.70		
		7	400.40		
		8	393.80		

		9	397.80		
		10	405.00		
1	0.1	1	44.40	38.37	3.5881
		2	40.20		
		3	40.70		
		4	39.04		
		5	37.93		
		6	34.31		
		7	34.64		
		8	33.19		
		9	37.42		
		10	41.82		
	1	1	55.24	60.55	4.9519
		2	67.10		
		3	65.00		
		4	56.06		
		5	58.01		
		6	60.00		
		7	64.86		
		8	67.03		
		9	55.00		
		10	57.16		
	5	1	295.00	295.38	3.6226
		2	295.00		
		3	290.00		
		4	295.00		
		5	295.00		
		6	302.83		
		7	300.00		
		8	292.98		
		9	295.00		
		10	292.98		
3	0.1	1	46.72	50.07	5.4583
		2	63.18		
		3	45.29		
		4	51.26		
		5	44.10		
		6	52.57		
		7	51.26		
		8	48.80		
		9	46.26		
		10	51.26		
	1	1	83.66	79.33	2.4428
		2	82.71		

		3	76.47		
		4	79.01		
		5	79.40		
		6	77.47		
		7	75.98		
		8	79.82		
		9	79.82		
		10	79.01		
	5	1	135.86	129.28	8.4647
		2	131.33		
		3	119.86		
		4	143.84		
		5	126.33		
		6	120.14		
		7	119.86		
		8	126.33		
		9	129.61		
		10	139.61		

APPENDIX B

TRACKING ERROR OF PID CONTROLLER

Frequency (Hz)	Displacement (mm)	Number of Test	Absolute Peak Error (µm)	Average Peak Error(µm)	Standard Deviation (µm)
0.1	0.1	1	43.00	55.57	9.610763642
		2	66.70		
		3	71.30		
		4	48.20		
		5	48.60		
		6	45.20		
		7	59.80		
		8	61.40		
		9	60.20		
		10	51.30		
	1	1	150.90	156.67	12.74833758
		2	165.00		
		3	124.90		
		4	163.60		
		5	159.50		
		6	158.80		
		7	152.70		
		8	155.90		
		9	171.60		
		10	163.80		
	5	1	169.80	163.56	19.94499102
		2	133.30		
		3	171.40		
		4	164.70		
		5	134.00		
		6	147.80		
		7	165.30		

		8	195.70		
		9	182.00		
		10	171.60		
1	0.1	1	132.99	121.72463	9.958853159
		2	134.87		
		3	135.00		
		4	122.02		
		5	120.11		
		6	107.16		
		7	113.75		
		8	121.24		
		9	110.10		
		10	120.00		
	1	1	237.94	228.37408	9.58275778
		2	233.41		
		3	223.99		
		4	228.41		
		5	228.99		
		6	229.67		
		7	231.35		
		8	228.41		
		9	237.58		
		10	203.99		
	5	1	402.97	391.40974	13.97164294
		2	382.94		
		3	378.03		
		4	387.97		
		5	417.14		
		6	393.03		
		7	376.55		
		8	385.87		
		9	409.32		
		10	380.28		
3	0.1	1	114.79	105.57381	9.620298614
		2	111.24		
		3	114.40		
		4	115.66		
		5	87.90		
		6	92.90		
		7	101.01		
		8	101.52		
		9	105.66		
		10	110.66		
	1	1	367.88		

		2	352.88	342.58184	16.7599807
		3	344.30		
		4	319.30		
		5	334.30		
		6	314.30		
		7	351.39		
		8	342.88		
		9	359.30		
		10	339.30		
	5	1	873.08	874.0247	6.717243244
		2	868.03		
		3	874.13		
		4	888.03		
		5	873.03		
		6	871.88		
		7	871.88		
		8	878.03		
		9	879.13		
		10	863.03		