PD-STABILITY AUGMENTATION SYSTEM (PD-SAS) CONTROLLER FOR 3-DOF RAILWAY VEHICLE SUSPENSION MODEL

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DECLARATION

I declare that this project report entitled "PD-Stability Augmentation System (PD-SAS) controller for 3-DOF railway vehicle suspension model" is the result of my own work except as cited in the references

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APPROVAL

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

Signature :

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Date : 26.06.2019

DEDICATION

To my beloved mother and father

ABSTRACT

Nowadays, the railway vehicle system are widely use and with the rapid development of railway use, vibration control to maintain stability, safety and improvement the level of passenger discomfort become the crucial part of research. This study focus on the semi-active suspension system and the 3 Degree of freedom (DOF) are developed using Newton second law which consist of two vertical dampers and springs, and a set of lateral spring and damper in lateral direction. The stability augmentation system (SAS) and PD-SAS are introduced in this study in order to reduce the unwanted motion. The response of the vehicle performance are investigated under step and sinusoidal track irregularity. The response of semi-active system are compared with passive system. Matlab Simulink software has been used in order to investigate and analyse the performance of the railway vehicle model. The simulation results of the study clearly show that the semi-active system are able to improve the performance of the vehicle compared with the passive system .

ABSTRAK

Pada masa kini, sistem kenderaan keretapi digunakan secara meluas dan dengan perkembangan pesat penggunaan kereta api, kawalan getaran untuk mengekalkan kestabilan, keselamatan dan peningkatan tahap ketidakselesaan penumpang menjadi bahagian penting di dalam penyelidikan. Kajian ini menumpukan pada sistem penggantungan separa aktif dan 3 darjah kebebasan (DOF) yang diwujudkan menggunakan undang-undang kedua Newton yang terdiri daripada dua peredam menegak dan mata air, dan satu set spring sisi dan peredam pada arah sisi. Sistem pembesaran kestabilan (SAS) dan PD-SAS diperkenalkan dalam kajian ini untuk mengurangkan pergerakan yang tidak diingini. Tanggapan prestasi kenderaan dikaji menggunakan step dan sinusoidal input. Tindak balas sistem separuh aktif dibandingkan dengan sistem pasif. Perisian Matlab Simulink telah digunakan untuk menyiasat dan menganalisis prestasi model kereta api . Hasil simulasi kajian jelas menunjukkan bahawa sistem separa aktif mampu meningkatkan prestasi kenderaan berbanding dengan sistem pasif.

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LIST OF ABBEREVATIONS

DOF	Degree of freedom
MATLAB	Matrix laboratory
UTeM	Universiti Teknikal Malaysia Melaka
PID	Proportional, Integral and Derivative
PD	Proportional and Derivative
GUI	Graphical user interface
SAS	Stability Augmentation System

LIST OF SYMBOL

M_c	-	Mass of car body
M_b	-	Mass of bogie
k_1	-	Secondary lateral spring stiffness
k_2	-	Secondary vertical spring stiffness
k_r	-	Stiffness of bogie disturbance
b_1	-	Secondary lateral damping coefficient
b_2	-	Secondary vertical damping coefficient
h_1	-	Height between body centre of gravity and secondary lateral suspension
w	-	Width of body centre gravity and secondary vertical suspension
θ	-	Roll angle of vehicle body
I_r	-	Roll axis moment of inertia
$\ddot{\theta}_c$	-	Roll acceleration of vehicle body
$\ddot{y}_b$	-	Lateral acceleration of vehicle bogie
$\ddot{y}_c$	-	Lateral acceleration of vehicle body
y_c	-	Lateral displacement of vehicle body
y_b	-	Lateral displacement of vehicle bogie
F_d	-	Lateral MR damper
m	-	Meter
rad/sec	-	Radian per second

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CHAPTER 1

INTRODUCTION

1.0 Background

Nowadays, most of the people in the country prefer to choose the train especially to those who having a traffic problems in the cities and want to save on the cost of using the car. High-speed trains are environmentally friendly and they are safe to use. With the rapid development of railway use, vibration control to maintain stability, safety and improvement the level of passenger discomfort become an important part of research. Therefore, the suspension system plays an important role because of it can reduce the unwanted motion from a train on irregular tracks and also can improve passenger comfort during travel.

This research will study on the semi-active suspension system of the railway vehicle system in order to improve the stability of the railway vehicle, so that travelling could become more comfortable, provide a great rating of ride quality and avoid discomfort from passengers. Active and semi-active suspension systems is the most conventional suspension systems used to train trains have limitations in improving travel comfort and this system continually supply and modulate the flow of energy. In the previous study was carried out using passive suspension systems such as jingles, wheels, steel springs, rubber bushings and another mechanism. It is known as a passive system because they does not need the power source and can only store or dissipate energy.

Previously, use of active control technology in the railway industry as main suspension of railway train has becomes an attraction. Different solutions have been studied over the years in different countries for specific issues resolution. This research concerns about the semi-active control applied on a suspension system of the railway vehicle using Stability Augmentation System (SAS) and PD-Stability Augmentation System (PD-SAS) control theory which can increase the performance of the vehicle and control the stability of the railway vehicle in order to improve the ride quality. The suspension system can be classified into two types which is Primary and Secondary Suspension system. Primary suspension is in between Axle Box and Bogie consisting of Spring and dampers while the Secondary Suspension system is in between Bogie and train body either through a bolster or bolster-less. Both suspensions system are placed in the three direction which is vertical, lateral and longitudinal directions.

This study is limited to lateral and roll analysis with its possible degree of freedom. In order to increase the stability to the high level of the train, the controller will be set up on the suspension system . SAS and PD-SAS controller are introduced. Train consists of vehicle body and two bogie frames and each bogie has two solid axle wheels. Wheel sets are connected to the frame bogie through the springs and dampers in longitudinal and side directions. To determine the mathematical equation, two vertical dampers and springs, and a set of lateral spring and damper in lateral direction. Therefore, the system has 3 degree of freedom (DOF). The mathematical model has been constructed by deriving the equations of motion of a rail vehicle frame bogie and wheelset. The control strategies were implemented in MATLAB/Simulink to investigate the performance of the railway vehicle system.

1.1 Problem statement

The use of trains is widely used worldwide as it has many advantages. Therefore, the use of train becomes the user's primary option to arrive at the destination. The suspension system is a very important component in the railway and has an influence on the stability of the vehicle's operation and safety. The stability of the trains is a problem that needs to be focus in order to increase the performance of the vehicle and make transport efficient. Vibration of the carbody in railway system mainly caused by track irregularity. This vibration need to be reduce or minimise in order to improve the ride comfort. It is impossible to change the new track because it require a high cost. Therefore, the controlable suspension system are introduce located at the secondary suspension system due to the abality of the system to reduce the unwanted motion.

1.2 Objectives

The objectives of this project are:

i. To analyse a 3-DOF half-car railway vehicle suspension system.

ii. To compare analytically two types of control strategies applied on secondary lateral suspension system.

1.3 Scope of project

There are two type of suspension system which is primary and secondary suspension system. This study only focus on secondary lateral suspension system. The half-car railway vehicle suspension parameters are selected to represent the parameters of railway vehicle suspension model.

CHAPTER 2

LITERATURE REVIEW

2.0 Introduction

In this chapter, the purpose of literature review is to increase the understanding about the project that need to be done. This chapter also provide the method used based on the previous study which has been done by researchers or engineer. The literature review is very important on this study in order to make sure that the method proposed can be used to implement the project. This chapter discusses the related research into suspension system for railway vehicles. Background research has been done for vehicle suspension system especially for railway and automotive vehicles. Undoubtedly, over the years until now, there are many automotive engineers still working on the development of the railway vehicle system with the aim of providing better ride comfort. Higher speeds usually generate increased forces and accelerations on the vehicle, which has a negative impact on ride comfort. Therefore, most of the previous study on the suspension system and vehicle stability has been stimulated in order to increase the efficiency of the train. Suspension system and control strategies are chosen to be deeply investigated.

2.1 History

Train vehicles transport consisting of a series of connected vehicles that generally running on rails also known as tracks to transport cargo or passengers. Basically train tracks consist of two running track. The profile irregularity of a railway line and their interaction with wheel is the major vibration sources for railway vehicle system -8io]kjm. This vibrations are arrested by suspension arrangement. Research on railway vibrations was conducted by many researchers around the world over the years but this research is still an interesting field of science to be study. Suspension system was introduced very early in the development of railways that the interface between railway vehicle body and wheel needed some suspension system to reduce the vibration felt as the train moved along the track.

2.2 Suspension system in railway vehicle

Suspension system was recognised very early in the development of railways that the interface between vehicle body and wheel needed some sort of cushion system to reduce the vibration felt as the train moved along the line. Suspension system work to absorb road shock from the track in vertically or laterally and also avoid train to bounce at high speed which can lead to derailing. The purpose of suspension system is to improve the ride comfort, road handling and stability of vehicles. Spring are used to store the energy and slow down the acceleration while vehicle is moving and absorb shock energy from the track and convert it into potential energy of spring. There are three level of suspension system involve in railway vehicle which is primary, secondary, and tilting suspension system but freight wagons usually have just one suspension level which is primary suspension system (Qazizadeh, 2017).

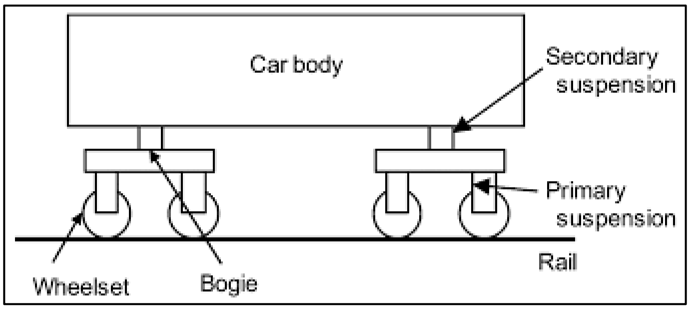

Figure 2.1 : Configuration of railway vehicle (Osamu Kondo and Yousuke Yamazaki, 2013).

2.2.1 Primary suspension system

The primary suspension is the rubber spring or coil and damper components placed between the bogie and the wheel set to supports the structural suspension of the carriage and entire train. Bogie are attached through bearing to support the railway vehicle body, improve ride quality by absorbing vibration and to reduce the generation of track irregularities. The primary suspension system has a beneficial effect on the stability limits of the bogie. The main function of primary suspension in railway vehicle is to reduce wheel-rail forces; provide acceptable curving; maintain wheelset stability; and also reduce wheel unloading on twisted tracks for preventing derailment. Railway vehicle need two suspension levels because without secondary suspension between bogie and carbody, the vibration level on the carbody would be very high for passengers so that primary suspension deals with wheel rail forces and stability issues while the secondary suspension improves the quality of ride comfort (Qazizadeh, 2017). Vibrations transmitted to the vehicle body can be reduce by focusing on reduction vibration at bogie. Vibration can be reduce with the increasing of primary damping coefficient so that the smaller amount of vibration is transmitted to the vehicle body.

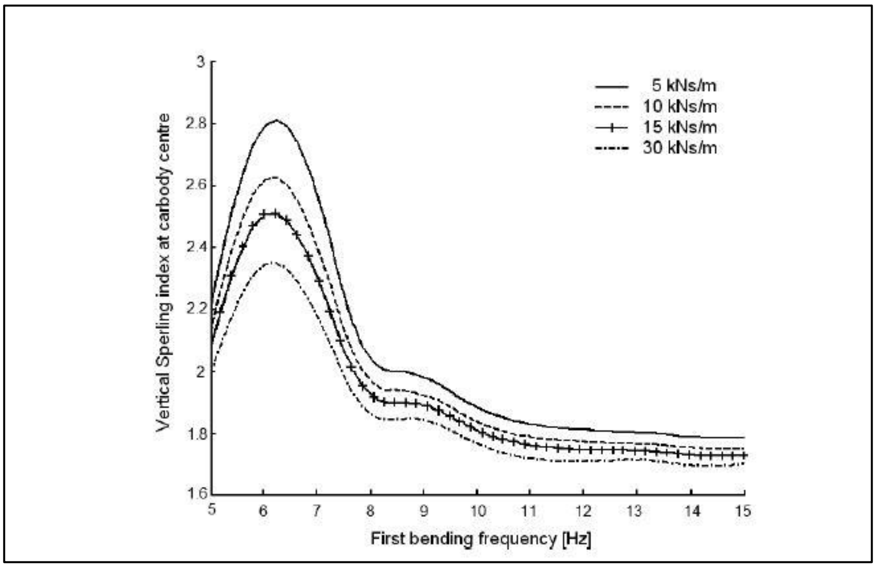

Figure 2.2 : Different primary damping coefficients (Zhou, Goodall et al., 2009).

2.2.2 Secondary suspension system

Secondary suspension system is the air-springs and viscous dampers interconnects the car body and bogie (Anneli orvanas, 2017). Secondary suspension system consist of air spring and vertical dampers placed between the vehicle body and usually two hydraulic dampers connected to each bogie to reduce the low-frequency vehicle vibrations (Anneli orvanas, 2017). Air spring can reduce the transmission of vibrations and also maintain the railway body at consistent height. The stiffness of the air spring is based on the amount of air in the bag. The secondary suspension is often also equipped with one anti-roll bar per bogie but sometimes known as stabiliser with the purpose of reducing the carbody roll. This anti-roll bar is normally transversely mounted on the bogie with vertical links connected to the carbody and can be regarded as a torsional springs. Emergency spring or bump stops is also one of the crucial element in the secondary suspension system that limit the relative vertical displacement between carbody and bogies. The bump stop is a rubber component with significant stiffness to give negative impact on ride comfort when the bump stops contact was occurs.

2.2.3 Tilting suspension system

A tilting train is a train that has a mechanism which can increase the speed on the regular rail tracks. Tilting trains lean the body of the vehicle inwards on curves to reduce lateral acceleration experienced by passengers. Acceleration will increase if the vehicle speed increase and it will cause the negative impact on ride comfort. When a train moving through a curve with the high speed, the passengers feel lateral acceleration which causes discomfort. Usually, the conventional trains on the conventional track will decelerate if the train passing a curve and accelerate after the curve. This situation will reduce the average speed, increasing the journey time, cause traveler fatigue and lose power. High speed when negotiates a curve can be improve in two way to control and reduce the lateral acceleration. Firstly by tilting the train inwards on curve and secondly by constructing high speed railways incorporating large radius of curvature. This is the reason why the high speed trains like Series Shinkansen, TGV Duplex and Alfa Pendular do not tilt. Idea to construct a new railway is possible and costlier than designing and manufacturing a tilting trains. Most tilting trains nowadays use the command-driven with precedence tilt control devised in the early 1980s as part of the UK-led Advanced Passenger Train development. Improving the tilt control performance of high-speed railway vehicles has been studied by (Zolotas and Goodall, 2005). In this study discusses on the use of an optimal LQG control design to improve the vehicle curving performance at increased running speed. As the result, the performance of local tilt control has improved.

2.2.4 Previous research on primary suspension

In the past, researchers, engineers and academicians carried out a lot of work on the secondary suspension system in order to improve the ride comfort but paid little focus to the stability problem of the railway vehicles. The researches generally focused on vibration reduction by applying semi-active or active system to the primary or secondary suspension system. Primary suspension system play the most crucial role to optimize the generated vibration and increase the stability of the vehicles. Analysis of primary and secondary Lateral suspension system of railway vehicle was done by (Harun, Jamaluddin et al., 2014) to study the effect of primary and secondary suspensions of a railway vehicle on stability and passenger ride comfort. As the result, stability of a railway vehicle can be improved by focusing on the primary suspension while ride comfort of the railway vehicle can be improve through various modifications on secondary suspension system. The impact of primary suspension stiffness of 2-axle bogie has been studied by (Zboinski and Golofit-Stawinska, 2018) showed that a multitude of solutions both in transition curve and circular curve. Simulations were performed for the object moving always along the route consisting of three consecutive sections: straight track, transition curve and circular curve. As the result, the improvement in stability of the railway vehicle can be achieved. In general, the stiffer the primary suspension the more effective the active suspension can be in suppressing kinematic instabilities (Hedrick, 1981).

Other than that, primary suspension damping force control system being developed to improve ride comfort on railway vehicle (Sugahara, Watanabe et al., 2011). In order to reduce the vibration of the car body, the vertical vibration of the bogie frame need to be focus by controlling the primary suspension system in the vehicle. Variable primary vertical damper has been introduced and modified. As the result, this system is effective in reducing the vibration and improve ride comfort (Sugahara, Watanabe et al., 2011). Another previous research to improve the dynamic performance of the railway vehicle has been investigated the primary suspension system by (Mohan and Ahmadian, 2004) and the results of the study indicate that the critical hunting velocity of the truck is most sensitive to the primary longitudinal stiffness.

2.3 Type of suspension

2.3.1 Passive suspension system

Suspension systems in automotive technology can be classified into a passive, semi-active and active suspension system. Investigation on semi-active and passive suspension systems for railway vehicle has increased recently due to the abilities of these types of suspension to suppress unwanted vibration. Passive suspension systems consists of an energy dissipating element such as coil or leaf springs and viscous dampers and both components work mechanically in parallel. Spring stores vibration energy in form of strain energy and damper dissipates energy due to compressive action over fluid**.** It is called as a passive suspension system because the passive element cannot add energy to the system (Mohan Rao, Venkata Rao et al., 2010). Passive suspension are simple and cost effective (Xiang Zheng, 2011).

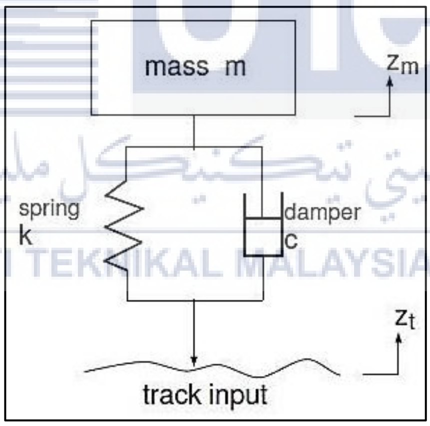

Figure 2.3 : Passive suspension (Xiang Zheng, 2011).

2.3.2 Semi-active suspensions system

Semi-active suspensions have the identical mechanical layout with the passive one. However, some control of damping coefficient can achieve by changing the characteristics of dampers.

2.3.3 Active suspension system

Active system contain external power sources, actuators as force generator which is synchronized by a control algorithm using data from sensors attached to the vehicle. The actuator can be hydraulic, pneumatic or electromagnetic, or a hybrid (Xiang Zheng, 2011).

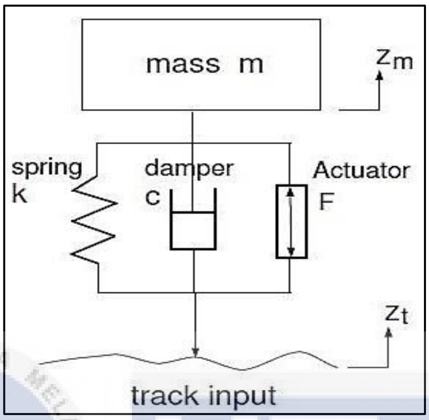

Figure 2.4 : Active suspension (Xiang Zheng, 2011).

Active suspensions require an external power source and consist of actuators as torque generators, sensing devices(e.g. force transducers, accelerometers and potentiometers) and a signal conditioning feedback controller to provide control commands for the actuators (Goodall and Kortüm, 1990). Active suspensions commercially implemented in automobiles today are based on the hydraulic or pneumatic. In previous study, improvement in ride quality by the use of active control has been studied (Pacchioni, Goodall et al., 2010) and (Orvnäs, Stichel et al., 2013). As the result, active controls improved the vehicle dynamic response and provide a better isolation of the vehicle body from track irregularities than the use of passive element and it is found that active suspension system improves ride comfort even at resonant frequency (Fayyad, 2018). Most of the researchers studies concerning on active secondary suspension systems compared to primary suspensions. Active Secondary Suspension in train investigated by (Orvnäs, 2008). Passenger ride comfort can be improve through various modification of the secondary suspension system.

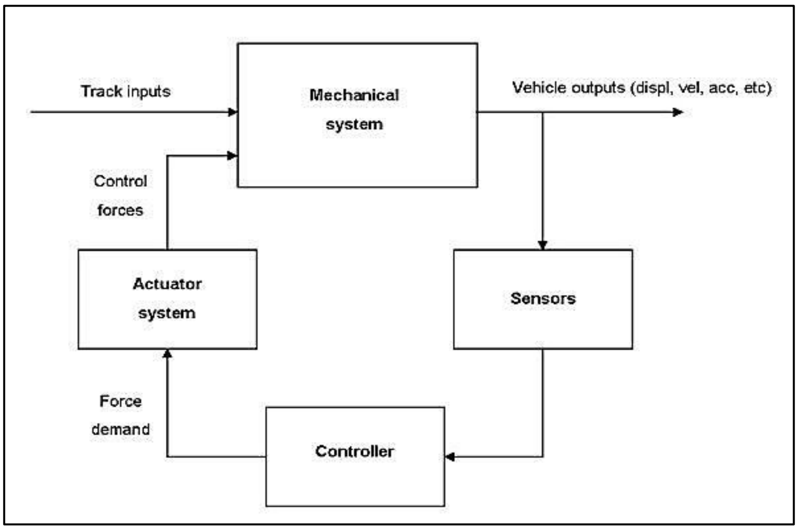

Figure 2.5 : Concept of active suspension system (Orvnäs, 2008).

2.4 Controller

Applying active control to the suspension of railway vehicles is a broad field of investigation for a couple of years (Goodall, 1997) and (Bruni, Goodall et al., 2007). The controller has been used widely in the industrial control system in order to control the process or the operation. There are many control approaches such as Linear Quadratic Regulator (Sam, Ghani et al., 2000), Adaptive sliding control (Chen and Huang, 2005), H∞ control (Du and Zhang, 2007), sliding mode control (Ahmad, 2018), preview control (Oraby, Aly et al., 2007), optimal control (Marzbanrad, Hojjat et al., 2003) and neural network methods (Al-Holou, Lahdhiri et al., 2002).

2.4.1 PID

PID control stands for proportional, integral and derivative control and each of the single term has different function and this controller has unique property when driving, positioning, and starting/stopping a motor . (Hassan, Zolotas et al., 2017) done a research on advanced PID for tilt control. The research from (Ahmed, Ali et al., 2015) with the aim to obtain a mathematical model for the passive and active suspensions systems for quarter car model and use the PID controller to construct an active suspension control for a quarter car model subject to excitation from a road profile. Some of the researcher (Yang, Zhang et al., 2011), (Liu, Zhang et al., 2009), (Hassan, Zolotas et al., 2008) done their study by introduce the PID controller on the railway vehicle system with the aim to improve the ride quality of the railway vehicle performance.(Hudha, Jamaluddin et al., 2008).

2.4.2 Skyhook

Since 1974, the sky-hook control has been mostly used in automobile suspension systems. The concept of skyhook controlled damping for semi-active vibration control originated from Karnopp's studies (Karnopp, Crosby et al., 1974) and (Karnopp 1995). Skyhook control strategy was introduced by Karnopp, 1990 in which a fictitious damper is inserted between the sprung mass and the stationary sky as a way of suppressing the vibratory motion of the sprung mass and as a tool to compute the desired damping force. An element in the skyhook controlled damping system consists of sprung mass, absolute reference frame (fixed in the sky) and fictitious damper. (Hohenbichler, Six et al., 2006) done the research to evaluate the benefit of skyhook control approaches in vertical secondary suspensions of high speed railway vehicles, compared to passive suspensions which are tuned optimally. As the result, it is shown that the RMS-value of the filtered passenger accelerations can be reduced by pure and also by complementary filtered skyhook control. Another research has been successfully completed by (Li and Goodall, 1999) to compares different control strategies for applying skyhook damping control laws in active suspension systems for railway vehicles.

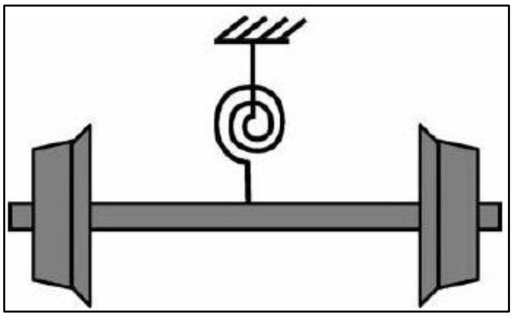

Figure 2.6 : Sky-hook spring concept (Mei and Goodall, 2006).

2.4.3 Stability Augmentation System (SAS)

Stability Augmentation System (SAS) is another type of the controller and widely used in the aerospace technology. Some of the helicopter used the SAS controller to increase the stability of the vehicle in flight and in a hover. Many researcher has done their study in control system using SAS in order to investigate the ability of the SAS controller by compare the system with the passive model. (Harun, Zailimi et al., 2014) done their study using SAS and Skyhook-SAS (hybrid) in order to reduce the effect of track disturbance. As the result, The SAS and hybrid has the ability to cancel out the unwanted motion compare to the passive system. Another researcher done their study using SAS is (Hudha, Jamaluddin et al., 2008) with the aim to damp out the effect of the disturbances to the dynamics performance of the light armored vehicle. The result from the study clearly state that the controller can reduce the amplitude and the settling time of unwanted motions in the forms of body displacement, body acceleration, roll angle, roll rate, pitch angle and pitch rate compared with the passive system.

2.5 Track irregularity

Track irregularity is the one of the most factors that cause safety problem. There are several type of track irregularity in the railway industry which is Track gauge irregularities, Vertical irregularities, cross level irregularity, lateral irregularity and twist irregularity.

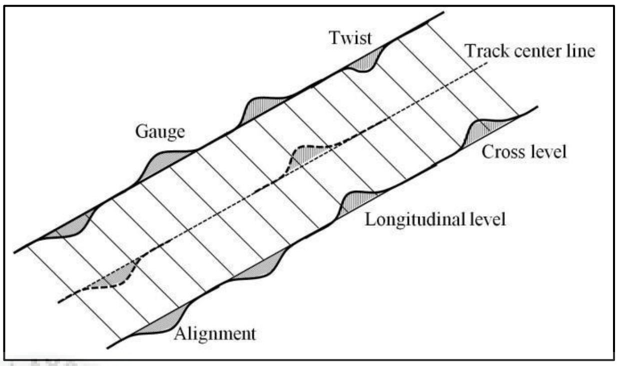

Figure 2.7 : Type of track irregularity (Ling, Deng et al., 2017).

2.6 Summary

This chapter presented the previous study on suspension system in railway vehicle. There are different method to reduce the vibration on the carbody to improve the ride comfort. The vibration can be reduce either by focusing on the primary or secondary suspension system. From the previous study done by the other researchers, they are more interested to the secondary suspension system compare to the primary suspension system and also it can be conclude that semi-active suspension system are able to improve the performance of the vehicle. There are many control strategies has been introduced by the researchers and every controller has the advantages to improve the vehicle system. This study will focus on the semi-active system on the lateral secondary suspension system while the PD and the Stability Augmentation System (SAS) has been choose in this study in order to investigate the dynamic performance of the railway vehicle system.

CHAPTER 3

METHODOLOGY

3.0 Introduction

This chapter discusses about methodology used in this project that will include in MATLAB/Simulink software. To ensure that projects are implemented properly every process must be planned well so that the project can be completed on time without delay.

3.1 Railway vehicle model

The dynamic model of a railway vehicle with two level of suspension system which is primary and secondary is developed to analyze and control the stability of the rail vehicle. The schematic view of the 3 Degree of freedom (DOF) vehicle model developed in order to investigate and analyze the dynamic behavior of the vehicle. It is consisting of two dampers and springs placed in the vertical direction, and a set of lateral spring and damper placed in the lateral direction. The 3 DOF was derived based on the Newton second laws. The diagram of the railway vehicle with 3 DOF is shown in figure 3.1 and figure 3.2.

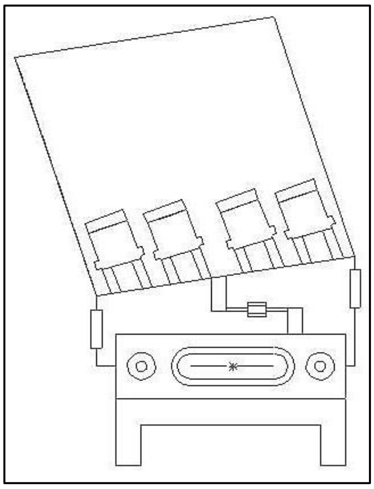

Figure 3.1 : Sectional front view

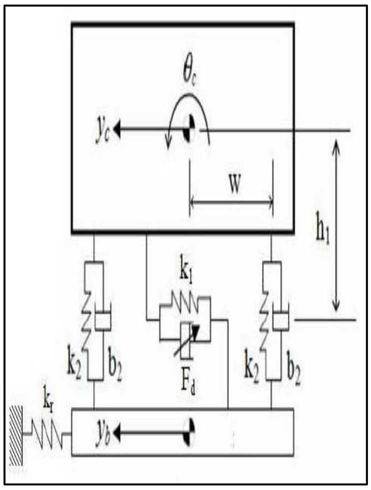

Figure 3.2 : schematic diagram

3.2 Modelling Assumption

Modelling assumption need to be made to simplify the problem to a point where they can be analysed. The modelling assumptions of this project are as follow:

i. The vehicle body, bogie and wheel-sets are considered as rigid.

ii. Aerodynamic drag force is neglected.

iii. The suspension components between vehicle body and bogie are model as passive secondary suspension system consists of spring and dampers positioned in vertical, lateral and longitudinal directions, while the components between the bogies and wheel-sets are modeled as passive suspension system.

iv. Rolling resistance due to anti roll bar and body flexibility are also neglected.

v. The wheel-sets move along a straight rail at a certain constant velocity with track alignment irregularity is regarded as the external excitation to the railway vehicle system.

vi. The railway vehicle body is assumed to travel in longitudinal direction with the constant speed.

vii. The direction of movement only focus on lateral.

3.3 Mathematical model

Mathematical model has been developed by deriving the equations of motion half car railway vehicle. Based on lateral and roll movement direction, 3 DOF has been constructed due to bogie and wheelset and the equation of motion of the suspension model are as follows:

The differential equation of the vehicle body in lateral direction

$$M_c \ddot{y}_c = -k_1(y_c - h_1\theta_c - y_b) + F_d \quad (3.1)$$

The differential equation of the vehicle bogie in lateral direction

$$M_b \ddot{y}_b = k_1(y_c - h_1\theta_c - y_b) + k_r(y_r - y_b) - F_d \quad (3.2)$$

The differential equation in roll direction

$$Ir\theta r = -k1(y_c - h_1\theta_c - y_b)h1 - 2k_2 w^2\theta - 2b_2 w^2\dot{\theta} + F_d h_1 \quad (3.3)$$

M_b and M_c is a mass of railway vehicle bogie and mass of railway vehicle body respectively while k_1, k_2 and k_r is a lateral spring of secondary suspension, vertical spring of secondary suspension and rail disturbance coefficient. b_2 is a vertical damper of secondary suspension. F_d is a lateral MR damper while y_b and y_c is a lateral displacement of vehicle bogie and Lateral displacement of vehicle body. $\ddot{y}_c$ and $\ddot{y}_b$ is a lateral acceleration of vehicle body and lateral acceleration of vehicle bogie. θ is a roll angle of vehicle body and $\ddot{\Theta}_c$ is a roll acceleration of vehicle body. I_r is roll axis moment of inertia,

h_1 is height of vehicle body COG to secondary suspension and w is a half width of vehicle body. The parameter used in this equation are set based on the (Harun, Jamaluddin et al. 2013) and the parameter as shown in the table 3.1 below.

Table 3.1 : 3DOF railway vehicle model parameter

Parameter	Symbol	Value
Mass of car body	M_c	50 kg
Mass of bogie	M_b	30 kg
Secondary lateral spring stiffness	k_1	17 kN/m
Secondary vertical spring stiffness	k_2	21 kN/m
Stiffness of bogie disturbance	k_r	240 kN/m
Secondary lateral damping coefficient	b_1	15 kNs/m
Secondary vertical damping coefficient	b_2	15 kNs/m
Height between body centre of gravity and secondary lateral suspension	h_1	0.169 m
Width of body centre gravity and secondary vertical suspension	w	0.2 m

3.4 MATLAB/Simulink software

MATLAB software used in this study to analyze the dynamic system of the railway vehicle and it provides many toolboxes to support an interactive environment for modelling and simulating a wide variety of dynamic systems including linear, nonlinear discrete-time, continuous-time and hybrid systems. There are some advantages of vehicle system modeling and simulation which is do not require a physical vehicle and test track and in actual test, there are a lot of time is required to run the test and also it is costly. The test and simulation can be repeated if the result has not been shown. Together with Simulink, it also provides a graphical user interface (GUI) for building models as block diagrams, using click-and-drag mouse operations. 3 DOF represented into block diagram by using MATLAB Simulink Library block function. Figure 3.3 shows the MATLAB interface and Figure 3.4 shows the MATLAB Simulink library.

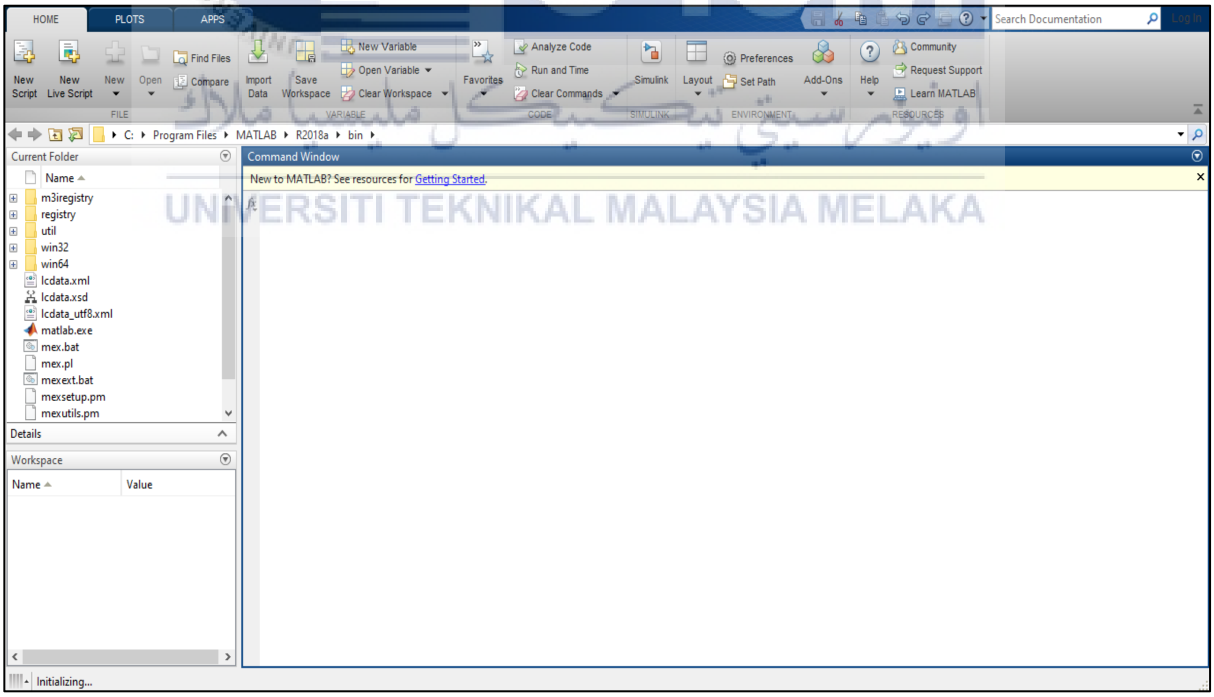

Figure 3.3 : MATLAB interface.

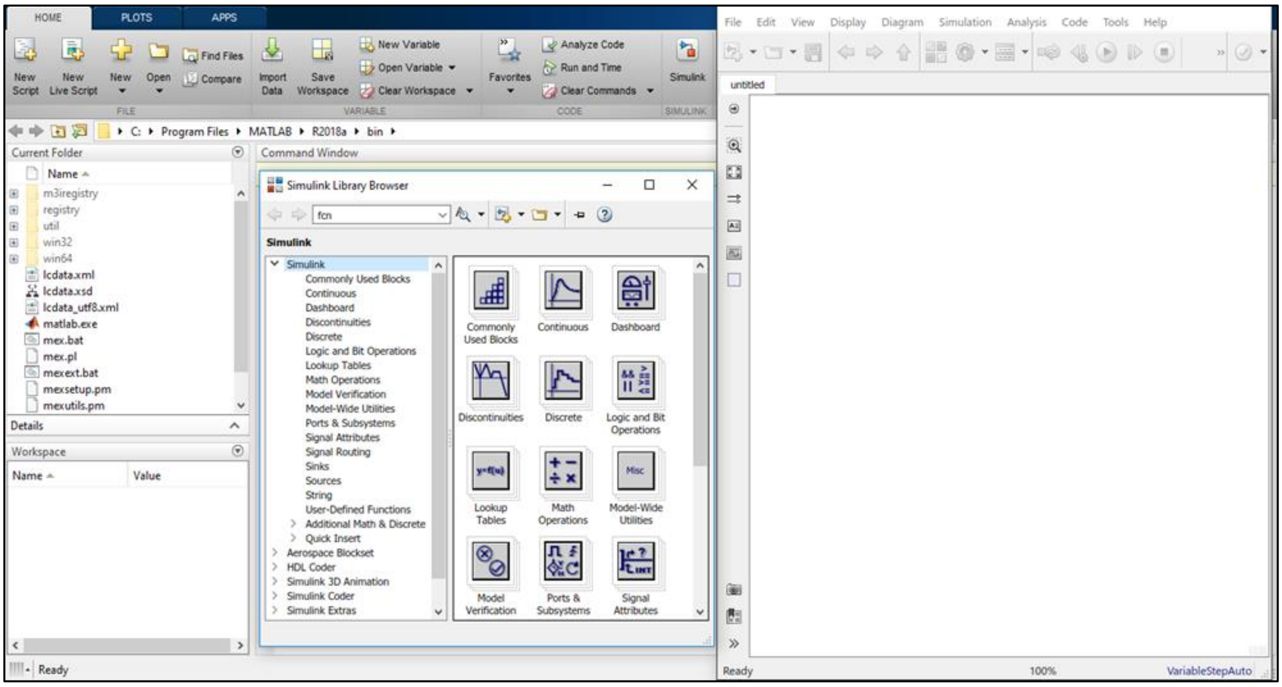

Figure 3.4 : MATLAB Simulink library.

3.5 3-DOF Model Simulink

The 3 DOF models will be develop using MATLAB Simulink. There are passive and semi-active system in order to compare the performance of the vehicle between both system and the three main part involve in the simulation is passive, Stability Augmentation Systems (SAS) and PD-SAS. PD is a short form for three separate term of Proportional (P) and Derivative (D) which is the control loop feedback frequently used in industrial control systems. After the simulation is carried out, the graph will appear in a scope block.

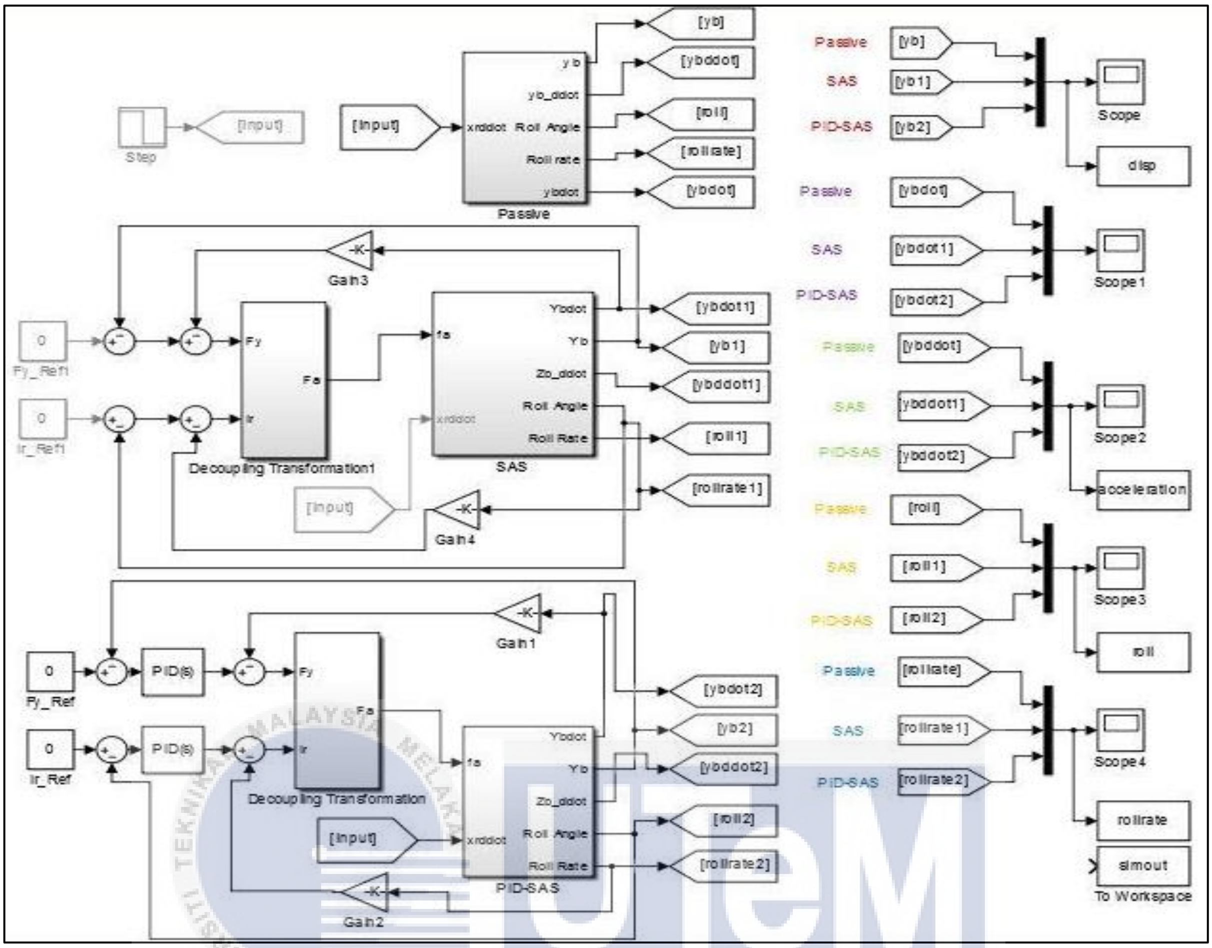

Figure 3.5 : The modeling of 3 DOF railway vehicle.

3.6 Control structure of semi-active suspension system for railway vehicle

Various control strategies such as optimal state-feedback, back steeping method, fuzzy control and sliding mode control have been proposed in the past years by researchers to control the active suspension system. The controller structure that implemented in this study shown in the figure 3.6 and figure 3.7 below. Basically, control structure consist of two type which is outer loop and inner loops. This study only focus on outer loop and the controller that will be used are SAS and PD-SAS. Outer loop used for disturbance rejection control to reduce the unwanted vehicle's motions while inner loop controller is used as force tracking control of the MR damper. Figure 3.6 shows the SAS while figure 3.7 show the PD-SAS control structure.

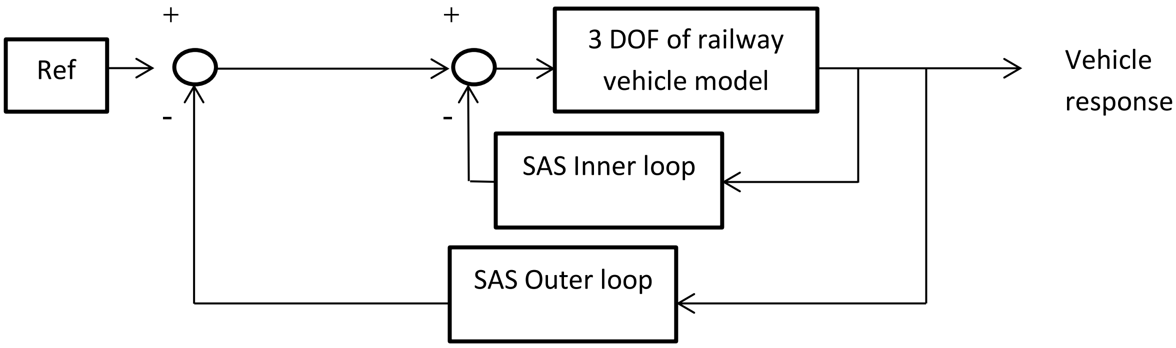

Figure 3.6 : Control structure of SAS controller

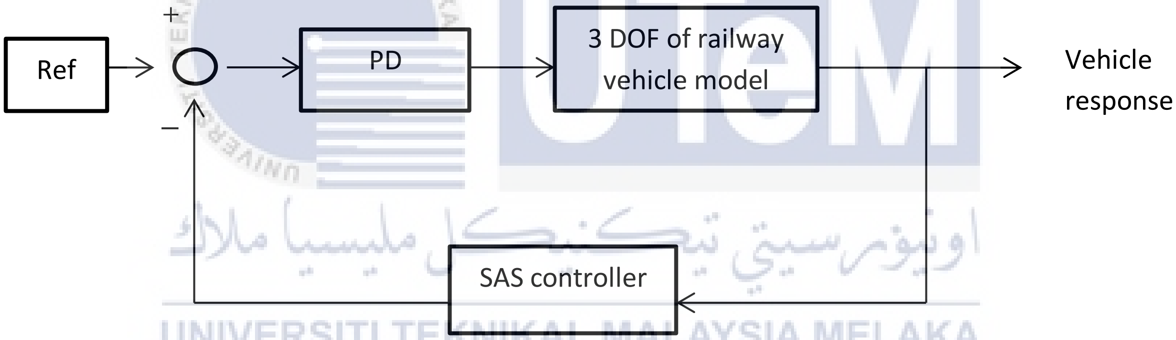

Figure 3.7 : Control structure of PD-SAS controller.

3.7 Decoupling transformation

In this study, the railway vehicle consist of two dampers and springs placed in the vertical direction, and a set of lateral spring and damper placed in the lateral direction. Generally in this study, there are two control input which is lateral and roll. This Decoupling transformation is placed between the PD controller and the vehicle model.

The equivalent force the lateral and roll can be defined by;

$$Fy = Fd \quad (3.4)$$

$$I\Theta = - Fd\,(w) \quad (3.5)$$

Arrange the equation into the matrix form;

$$\begin{bmatrix} Fy \\ I\theta \end{bmatrix} = \begin{bmatrix} 1 \\ -w \end{bmatrix} [Fd] \quad (3.6)$$

$$CC^T = \begin{bmatrix} 1 \\ W \end{bmatrix} [1 \;\; W] \quad (3.7)$$

Inverse matrix from the equation 3.7 is shown below;

$$CC^{T^{-1}} = \begin{bmatrix} A & B \\ C & D \end{bmatrix} \quad (3.8)$$

Therefore the matrices of CT(CCT)$^{-1}$ is solve and the coupling transformation equation;

$$[Fd] = \begin{bmatrix} Fy \\ I\theta \end{bmatrix} \begin{bmatrix} M \\ N \end{bmatrix} \quad (3.9)$$

$$Fd = MFy \quad (3.10)$$

$$Fd = NI\theta \quad (3.11)$$

Based on the equation 3.10 and 3.11, the actuator will give a response to the lateral force and roll angle of the vehicle body. The actuator will be placed in each of the vehicle system.

3.8 Controller tuning

` The PD controller must be tuning in order to achieve the optimum and acceptable performance. The goal of good tuning is to have the lowest overshoot without causing instability. There are several tuning methods have been introduced such as manual turning method, Trial and Error Method , Ziegler-Nichols method, tyreus luyben method, cohen-coon and other else. Ziegler-Nichols method are commonly the most known and widely used to tune the PD controller while the manual tuning are often used due to the no requirement of math and online method. In this study, the analysis sensitivity has been choose as a PD tuning method. To run the PD tuner, PD controller gain block from the Simulink must be double click to open the block dialog in order to setting up the value of PD. Next, insert the value of PID in the block dialog. Every two term of PD has the different response. For example, adjusting the value of P can improve the rise time while I and D are able to reduce steady state error and reduce the overshoot.

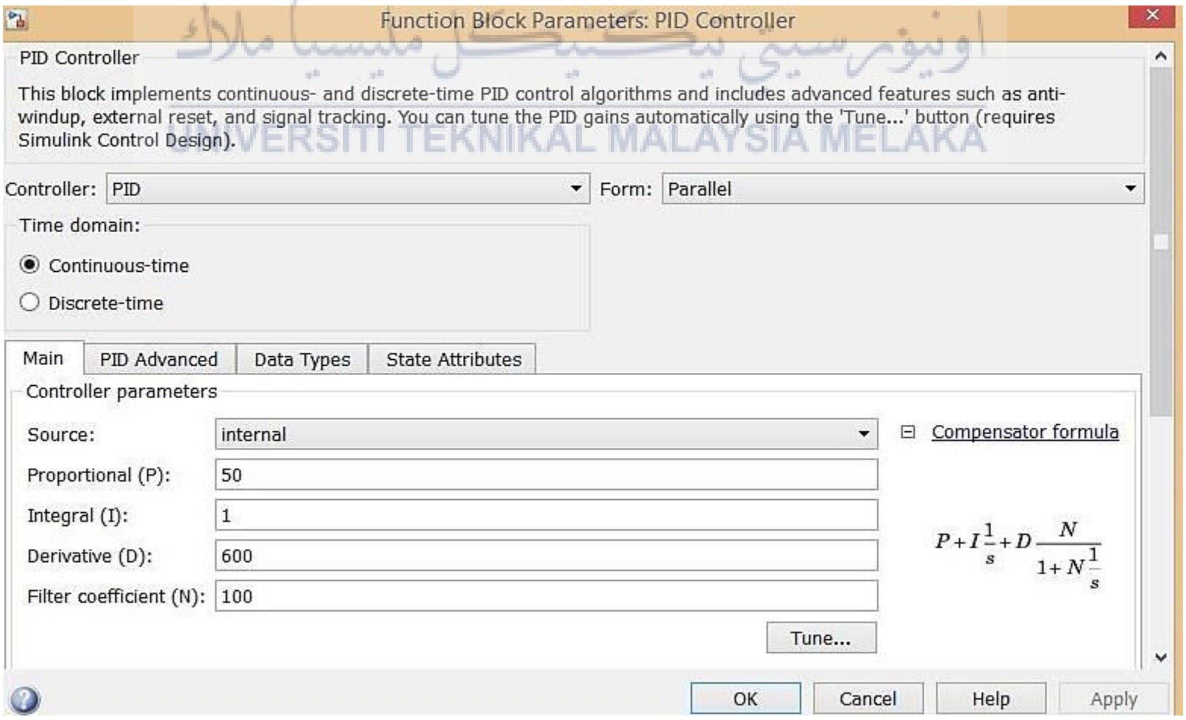

Figure 3.8 : PID block dialog.

3.9 Sensitivity Analysis of PD

The sensitivity analysis method has been choose in this study in order to tune the PD-SAS controller . The term sensitivity in relation to control system gives an idea of the system performance as affected due to parameter variations. In this method, the value of PD will be increase or decrease and the minimum value will be taken as the optimum value for the desired control response. The value of Proportional (P), and Derivative (D) must be test one after another with the performance of lateral body displacement, lateral body acceleration, Roll angle and roll rate. The test needs to be done continuously to achieve optimum value of the PD and the value that approaches to zero value is the most accurate value.

3.10 Summary

In methodology section, the mathematical modeling and control strategy is discussed. In order to analyze the dynamic vehicle MATLAB/Simulink are used during this study. The parameter then will identified and loaded into MATLAB software to come out with the result and the result will be analyze. Railway vehicle model considered in this study consist of two dampers and springs placed in the vertical direction, and a set of lateral spring and damper placed in the lateral direction and it represented as a 3-DOF system.

CHAPTER 4

RESULT AND DISCUSSION

4.0 Introduction

To investigate the performance, MATLAB/Simulink software is used to investigate and analyze the dynamic behavior of railway. In order to analyse the response of the railway vehicle, there are two cases will use in simulation which is track irregularity 0.05 m and the frequencies (1, 3 and 7 rad/sec). The performance criteria that will be consider in this project is body lateral displacement, body lateral acceleration, roll angle and roll rate at the body centre of gravity. The performance criteria are evaluated under step and sinusoidal track irregularities. The result are recorded in the Peak table and the total reduction in percentage of the displacement, acceleration, roll angle and roll rate of a railway vehicle body has been calculated.

4.1 Performance analysis on step input

The response of the vehicle performance for the lateral displacement, lateral acceleration, roll angle and roll rate under step input are shown in the figure below.

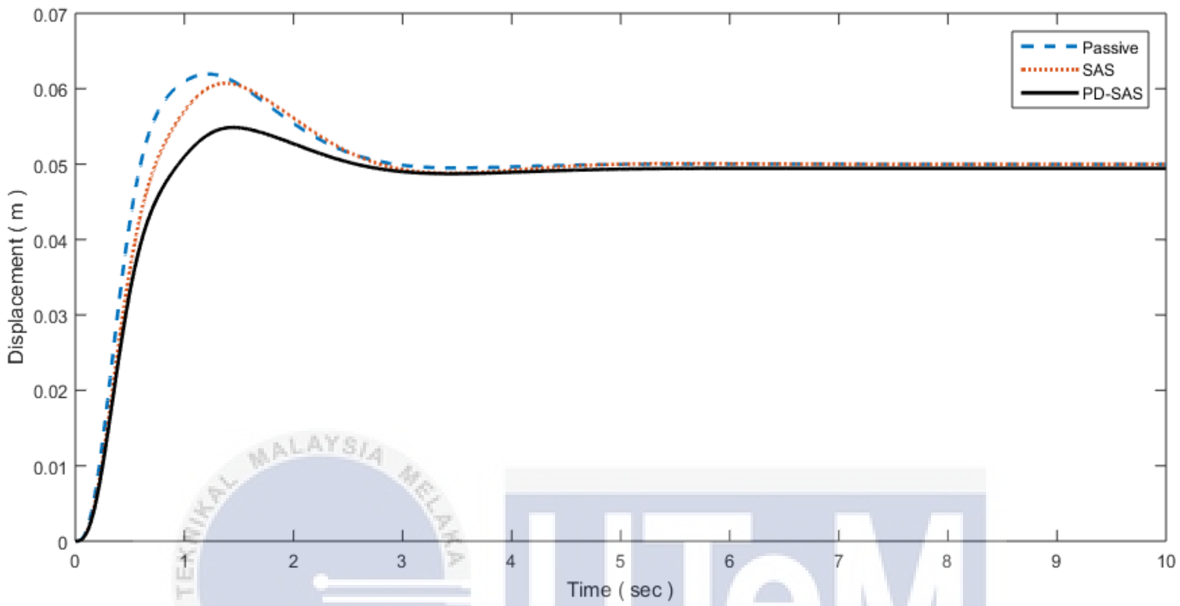

Figure 4.1 : Lateral displacement graph under step input.

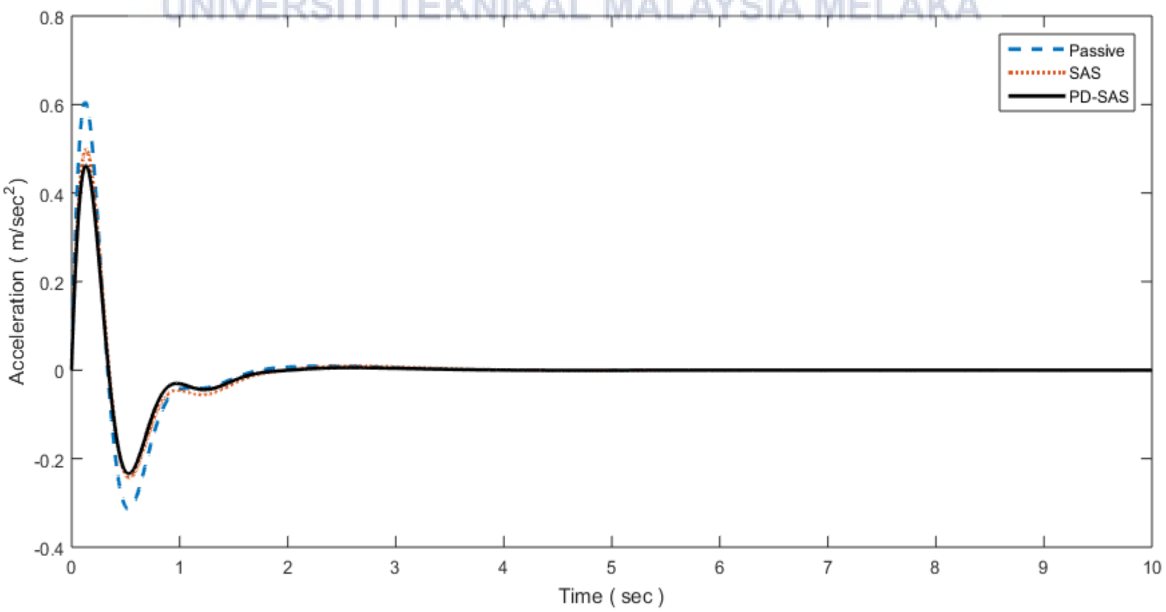

Figure 4.2 : Lateral acceleration graph under step input.

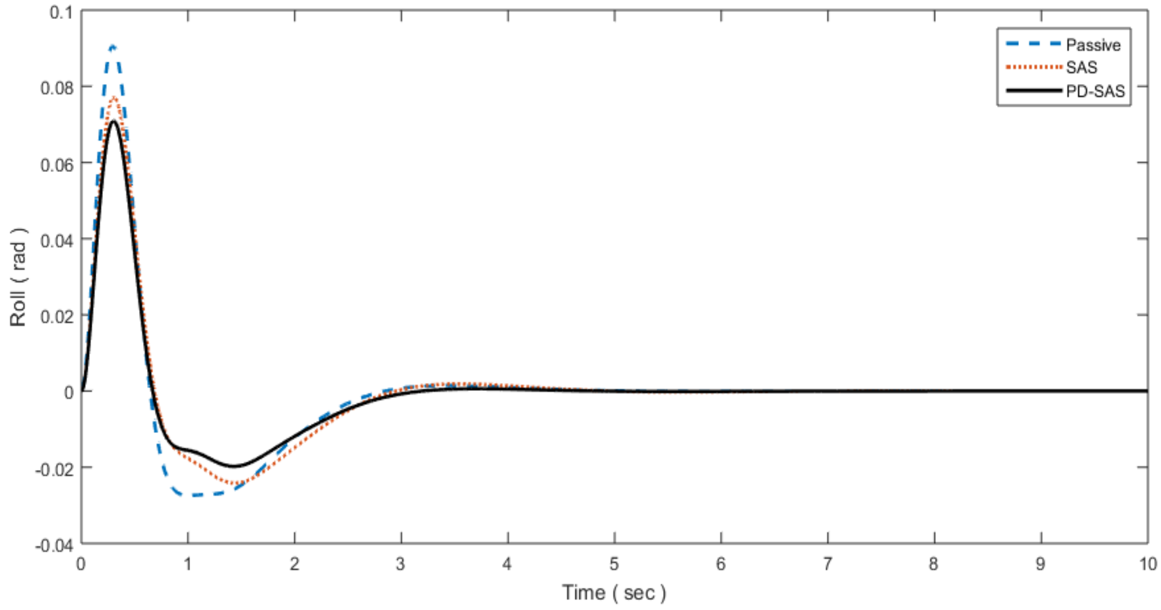

Figure 4.3 : Roll angle graph under step input.

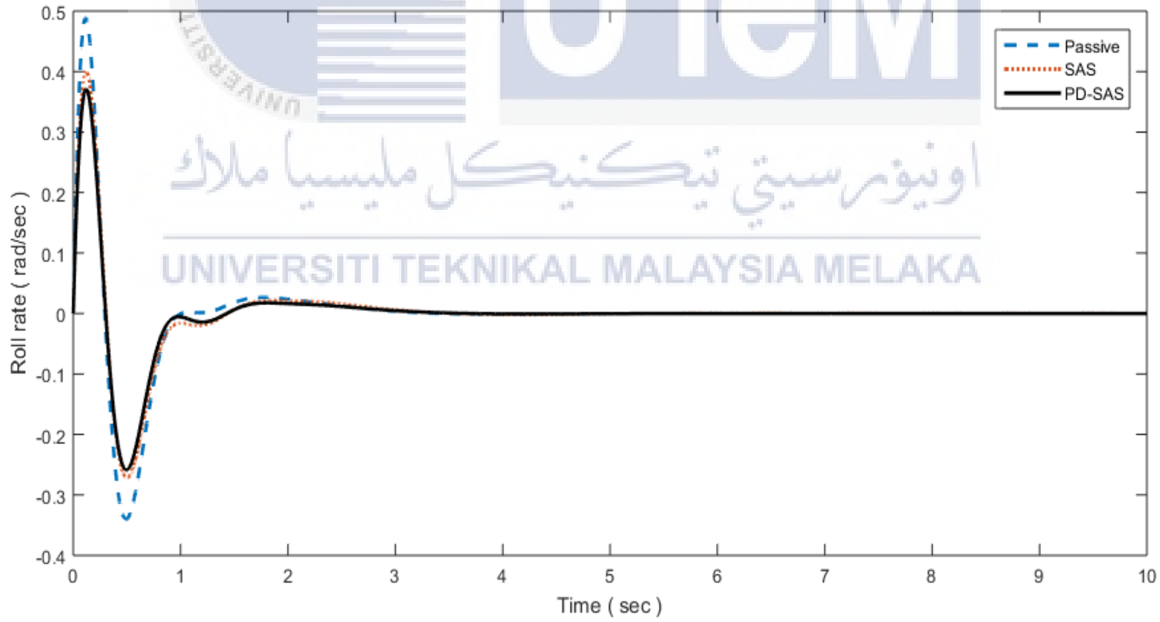

Figure 4.4 : Roll rate graph under step input.

Table 4.1 : Peak or overshoot values of railway vehicle body responses under step input.

Amplitude = 0.05 m					
Peak value					
Performance criteria	Passive	SAS	Reduction(%)	PD-SAS	Reduction(%)
Acceleration(m/s2)	0.6056	0.4989	17.62%	0.4623	23.66%
Displacement(m)	0.06194	0.06073	1.95%	0.05485	11.45%
Roll angle(rad)	0.09006	0.07709	14.40%	0.07067	21.53%
Roll rate(rad/sec)	0.4881	0.3987	18.32%	0.3692	24.36%

Figure 4.1 shows that the lateral displacement response with the passive system and the semi-active system. Based on the graph, it can be conclude that the semi-active system has the ability to reduce the overshoot values but in term settling time, the semi-active system does not show any effect. The result of the lateral acceleration can be seen in the figure 4.2. Based on the graph, similarly to the previous graph it clearly prove that the overshoot has been reduced by the semi-active system. Compare to the passive and semi-active controller, the PD-SAS provide good improvement compare to the passive and SAS but SAS controller is better than passive system. It can be prove using the peak to peak value on the graph while PD-SAS shows the lowest peak value compare to another. The graph 4.3 shows the response of the roll angle with the passive and semi-active system. Similarly to the lateral displacement and acceleration, the active system are also able to reduce the peak or the overshoot on the roll angle response. There are no change in term of rise time and steady state error. The figure 4.4 shows that the response of the roll rate. It clearly state that that, there are improvement in term of overshoot by using semi-active

system and the lowest peak at time between 0 to 0.5 seconds is represented by PD-SAS and this response state that the semi-active system has the ability to reduce the overshoot. However, there is no change in term of rise time and steady state error.

The total percentage reduction has been calculated and recorded in the table 4.1**.** The percentage of reduction of the lateral displacement is 1.95 % for the SAS controller system while reduction rate by the PD-SAS is 11.45% which is higher than the SAS . In terms of lateral acceleration response, the percentage reduction of the overshoot values for SAS and PD-SAS are 17.62% and 23.66% respectively. Total reduction for SAS in term of roll and roll rate is 14.40% and 18.32% which is reduction of roll rate is higher compare to the roll angle. For the PD-SAS, roll angle and roll rate recorded 21.53% and 24.36% of reduction and similarly to the SAS, the reduction of roll rate is higher compare to the roll angle.

4.2 Performance analysis on sine input with 1 rad/sec of frequency

The response of the vehicle performance for the lateral displacement, lateral acceleration, roll angle and roll rate under sine input are shown in the figure below.

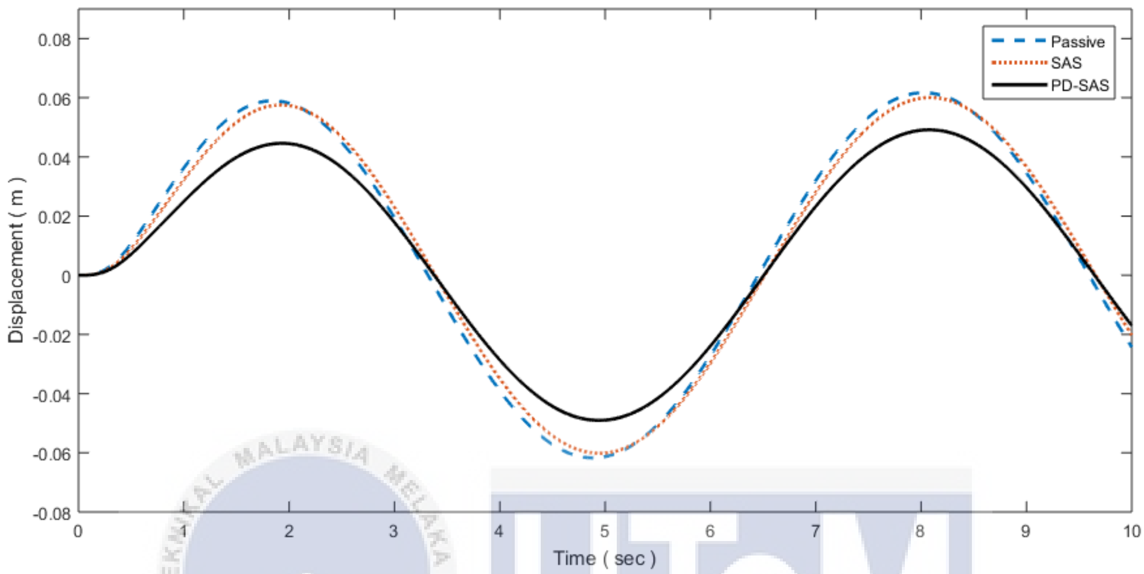

Figure 4.5 : Lateral displacement graph under sinusoidal track irregularity with 1 rad/sec of frequency.

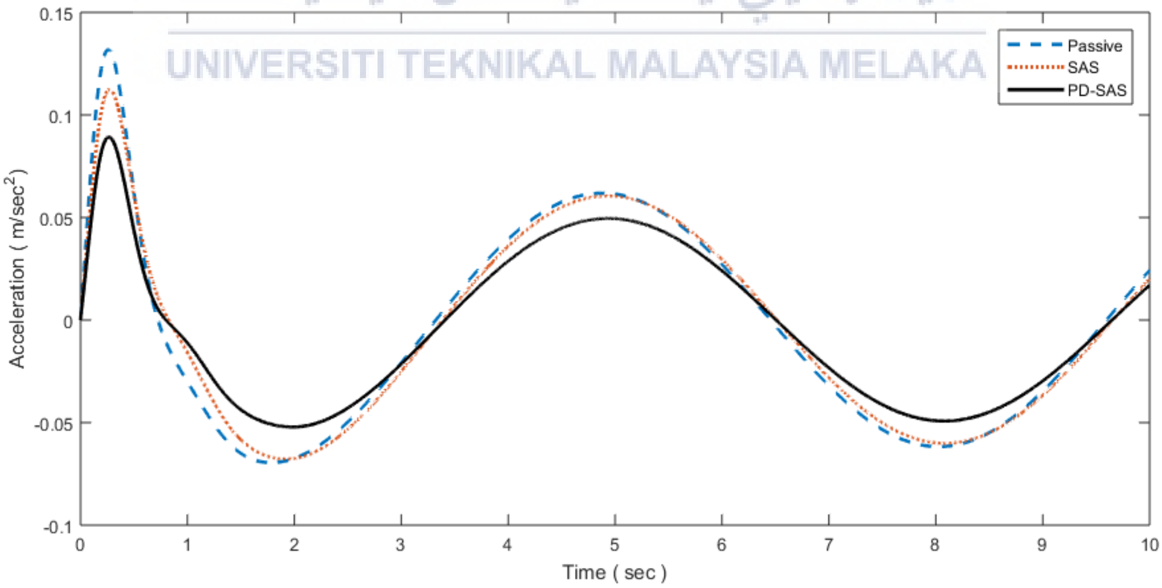

Figure 4.6 : Lateral acceleration graph under sinusoidal track irregularity with 1 rad/sec of frequency.

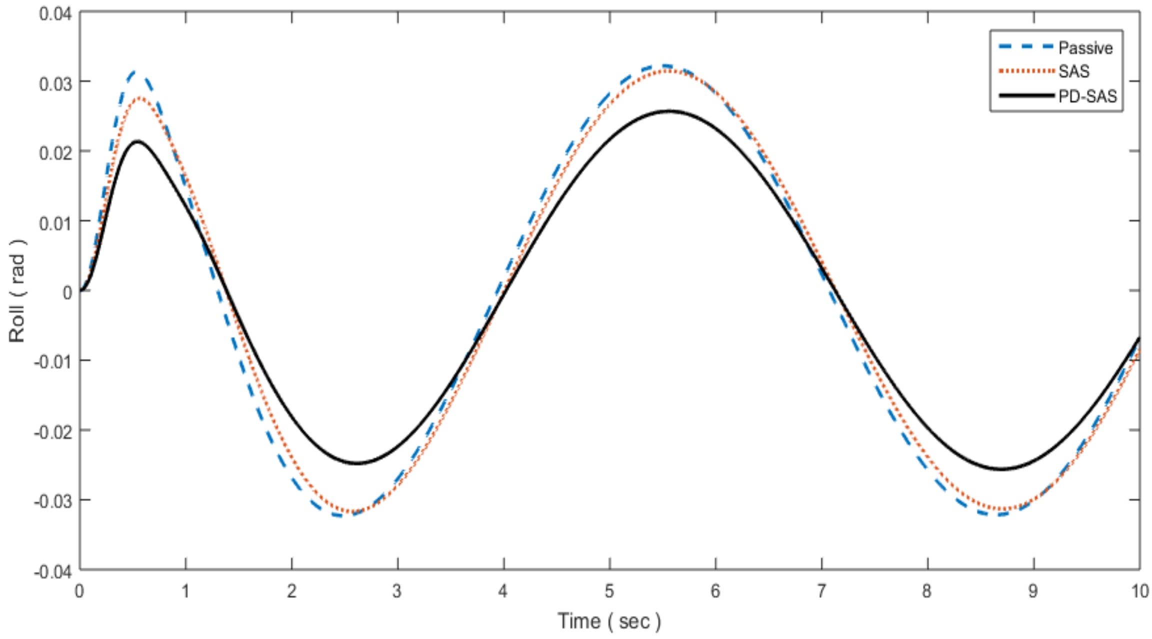

Figure 4.7 : Roll angle graph under sinusoidal track irregularity with 1 rad/sec of frequency.

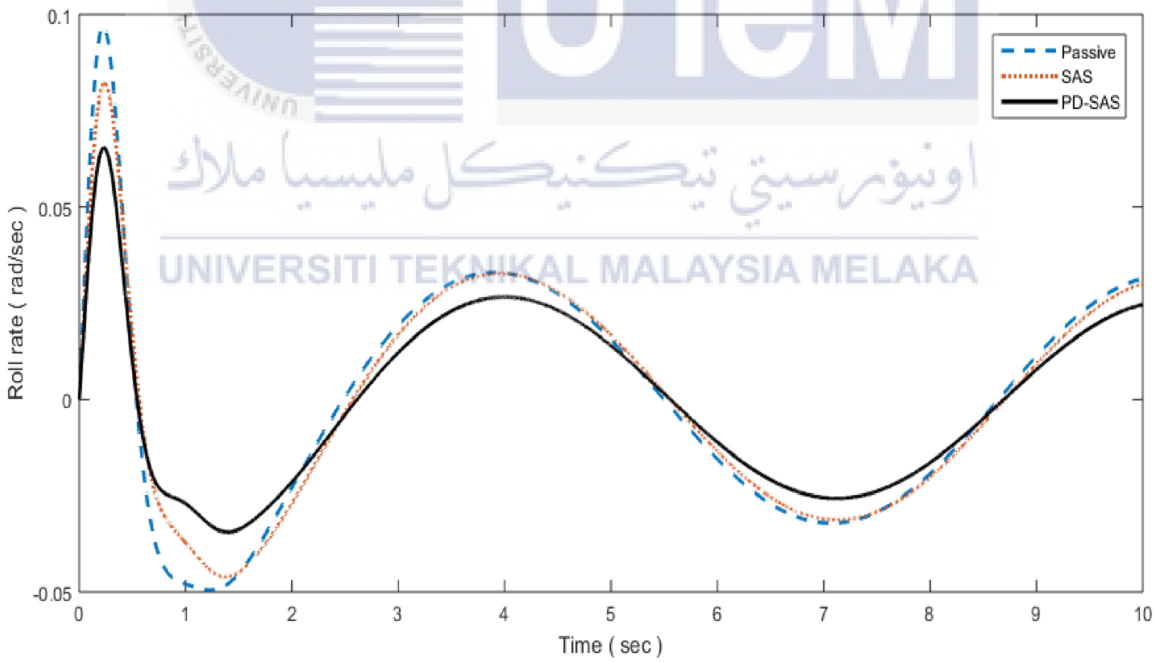

Figure 4.8 : Lateral roll rate graph under sinusoidal track irregularity with 1 rad/sec of frequency.

Table 4.2: Peak values of railway vehicle body responses with frequency of 1 rad/s and amplitude 0.05 m.

Frequency=1 rad/s Amplitude=0.05 m					
Peak value					
Performance criteria	Passive	SAS	Reduction(%)	PD- SAS	Reduction(%)
Acceleration(m/s2)	0.131	0.1122	6.0%	0.08879	32.22%
Displacement(m)	0.05896	0.05716	3.1%	0.04449	24.54%
Roll angle(rad)	0.03136	0.02755	12.15%	0.02133	31.98%
Roll rate(rad/sec)	0.09705	0.08213	15.37%	0.06537	32.64%

Figure 4.5, 4.6, 4.7, and 4.8 shows the result from the simulation of railway vehicle model for a sinusoidal track irregularity with the 0.05m under 1rad/s excitation frequency. There are three different colour line present in each graph which is blue colour represent the passive system while red and black line represent the SAS and PD-SAS controller respectively. Figure 4.5 show the lateral displacement which is passive system recorded about 0.06 m displacement at time 2 second while the SAS and PID-SAS controller recorded 0.1 and 0.09 respectively. Based on the passive and the controller system, by plotting all these case together, it found that PD-SAS give the lowest displacement compare to the SAS and passive. Therefore setting up the PD-SAS in the system giving a better ride comfort to the vehicle suspension system compare to passive and SAS **.** Figure 4.6 show the response of the lateral acceleration in time domain which is clearly state that SAS and PD-SAS is able to reduce the unwanted motion effectively. Therefore, the

improvement in the lateral acceleration and displacement give a better quality of the ride comfort. Figure 4.7 and figure 4.8 exhibit the response of the roll angle and the roll rate. From the both graph, it can be shown that vehicle body with the SAS and PD-SAS controller has improvement in term of roll angle and roll rate angle.

The performance criteria also been analyzed based on peak value which all the data has been recorded in the table 4.2. Based on the lateral displacement, the total reduction for the SAS and PD-SAS is 3.1% and 24.54% compared to the passive system. It can be conclude that the PD-SAS give the higher reduction compared to the SAS. The total reduction for in term of lateral acceleration for SAS is 6.0% while the PD-SAS is 32.22% which is the difference of reduction between SAS and PD-SAS is quite high. The reduction for SAS in term of roll angle and roll rate is 12.15% and 15.17% respectively. So the roll rate recorded the higher reduction in term of SAS compared to the roll angle. In term of PD-SAS, the roll angle and roll rate recorded the total reduction about 31.98% and 32.64% respectively and similarly, it can be conclude that , the roll rate state the higher reduction compare to the roll angle.

4.3 Performance analysis on sine input with 3 rad/sec of frequency

The response of the vehicle performance for the lateral displacement, lateral acceleration, roll angle and roll rate under sine input are shown in the figure below.

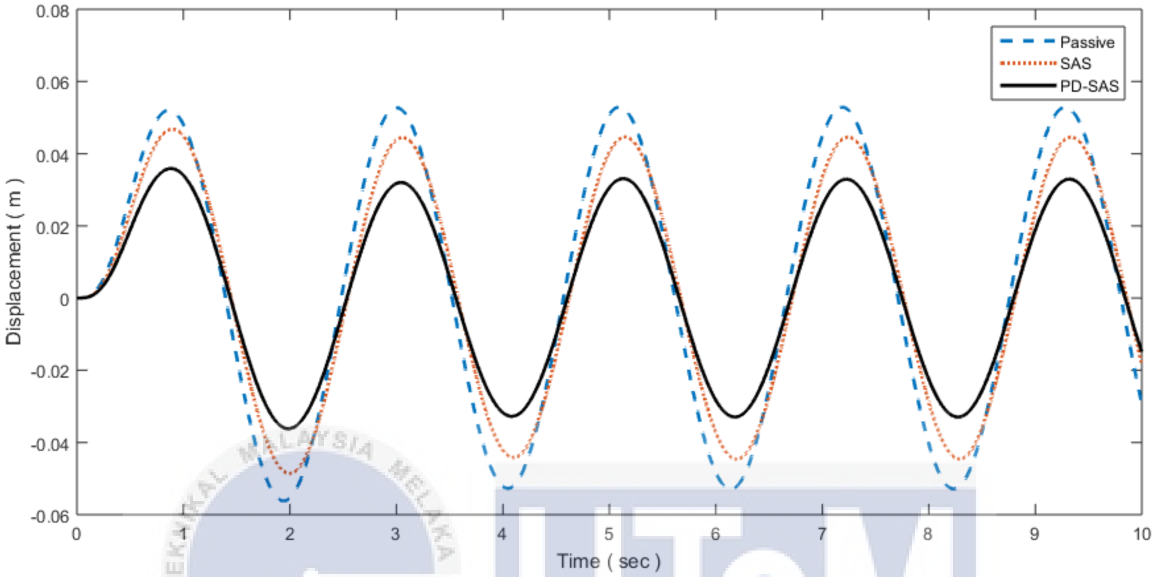

Figure 4.9 : Lateral displacement graph under sinusoidal track irregularity with 3 rad/sec of frequency

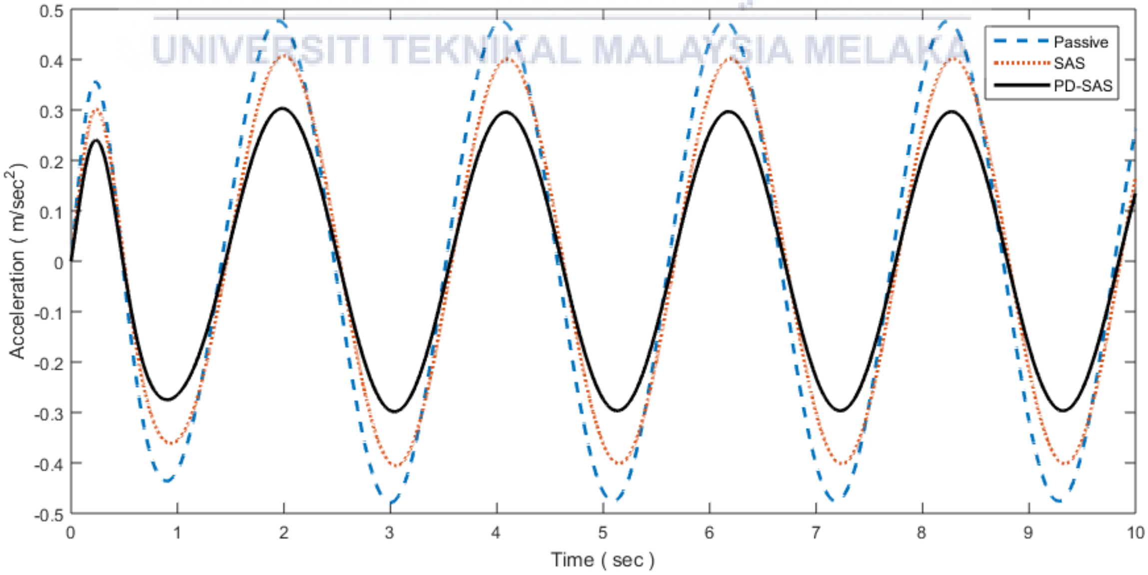

Figure 4.10 : Lateral acceleration graph under sinusoidal track irregularity with 3 rad/sec of frequency

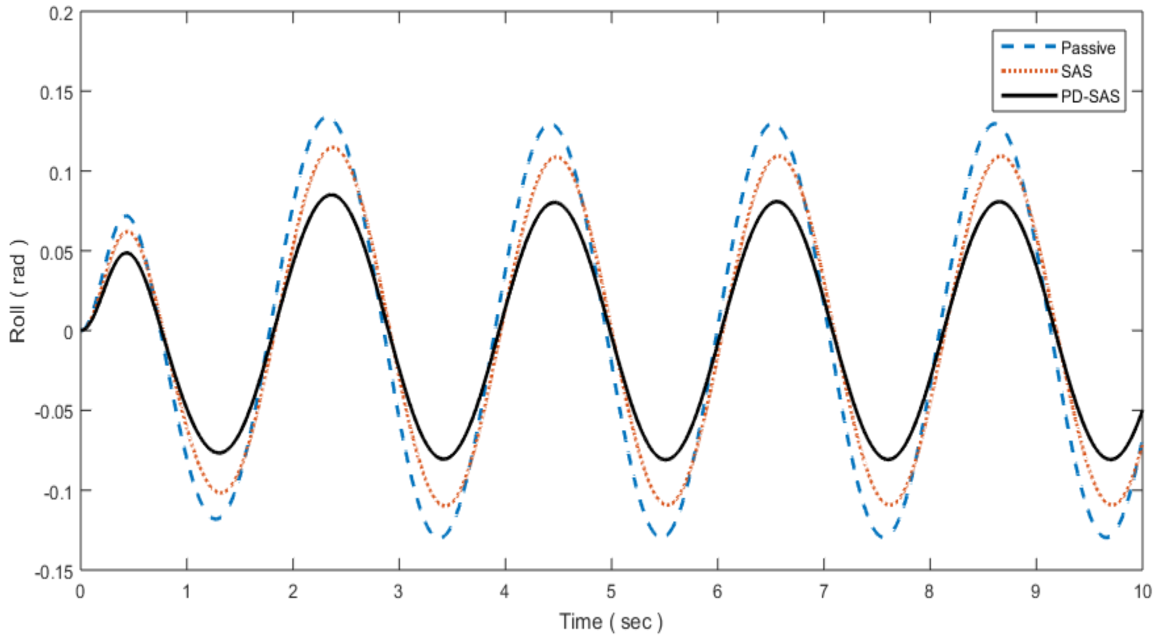

Figure 4.11 : Roll angle graph under sinusoidal track irregularity with 3 rad/sec of frequency

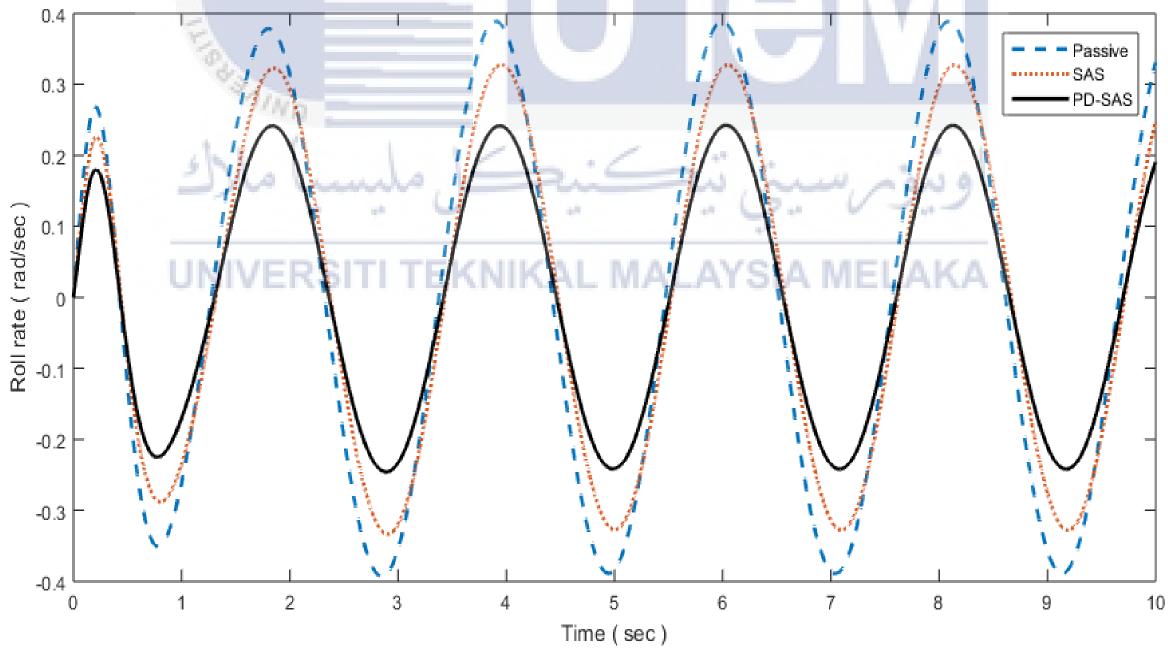

Figure 4.12 : Roll rate graph under sinusoidal track irregularity with 3 rad/sec of frequency

Table 4.3: Peak values of railway vehicle body responses with frequency of 3 rad/s and amplitude 0.05 m

Frequency=3 rad/s Amplitude=0.05 m					
Peak value					
Performance criteria	Passive	SAS	Reduction(%)	PD-SAS	Reduction(%)
Acceleration(m/s2)	0.4764	0.4072	14.53%	0.3029	36.42%
Displacement(m)	0.05273	0.04449	15.63%	0.03203	39.26%
Roll angle(rad)	0.1337	0.1147	14.21%	0.08503	36.40%
Roll rate(rad/sec)	0.3785	0.323	14.66%	0.2417	36.14%

The response of the vehicle under 3 rad/sec excitation frequency are presented in the figure of 4.9, 4.10, 4.11 and 4.12. Figure 4.9 show that the response of the lateral displacement and the displacement of the passive system is 0.05 at time about 1 second while the SAS and PD-SAS is 0.04 and 0.03 respectively. It can be seen clearly that the stability augmentation system and PD-SAS is able to eliminate unwanted motion effectively. The lateral acceleration response has been shown in graph 4.10 and it can be noted that the SAS and PD-SAS controllers are able to reduce the body lateral acceleration since that the amplitude of the both controllers is lower compare to the passive due to the track irregularities and this result can give an improvement to the ride quality of the vehicle. In term of body roll and body roll rate in figure of 4.11 and 4.12, it showed that the both controller are able to cancel out the unwanted of the both performance response compare to passive but the PD-SAS is slightly better than the SAS due to the lowest amplitude compare to the SAS and it is significantly better than the passive system. Based on the all graph under the 3 rad/sec excitation frequency, overall it can be conclude that the semi-active system are able to improve the train performance. It is also can be conclude that, when the frequency is increases, the number of cycle are also increase.

The performance criteria also been analyzed based on peak value which all the data has been recorded in the table 4.3. Based on the lateral displacement, the total reduction for SAS and PD-SAS is 15.63% and 39.26% respectively. So it can be conclude that the PD-SAS recorded the higher reduction compare to the SAS. The reduction of the lateral acceleration has ben calculated and the result recorded that the percentage reduction of PD-SAS if higher compare to SAS which 14.53% and 36.42% represent for SAS and PD-SAS respectively. The reduction for SAS in term of roll angle and roll rate is 14.21% and 14.66% respectively. So the roll rate recorded the slightly higher the percentage of reduction in term of SAS compared to the roll angle. In term of PD-SAS, the roll angle and roll rate recorded the total reduction about 36.40% and 36.14% respectively and it can be conclude that , the roll angle state the higher reduction compare to the roll rate.

4.4 Performance analysis on sine input with 7 rad/sec of frequency

The response of the vehicle performance for the lateral displacement, lateral acceleration, roll angle and roll rate under sine input are shown in the figure below.

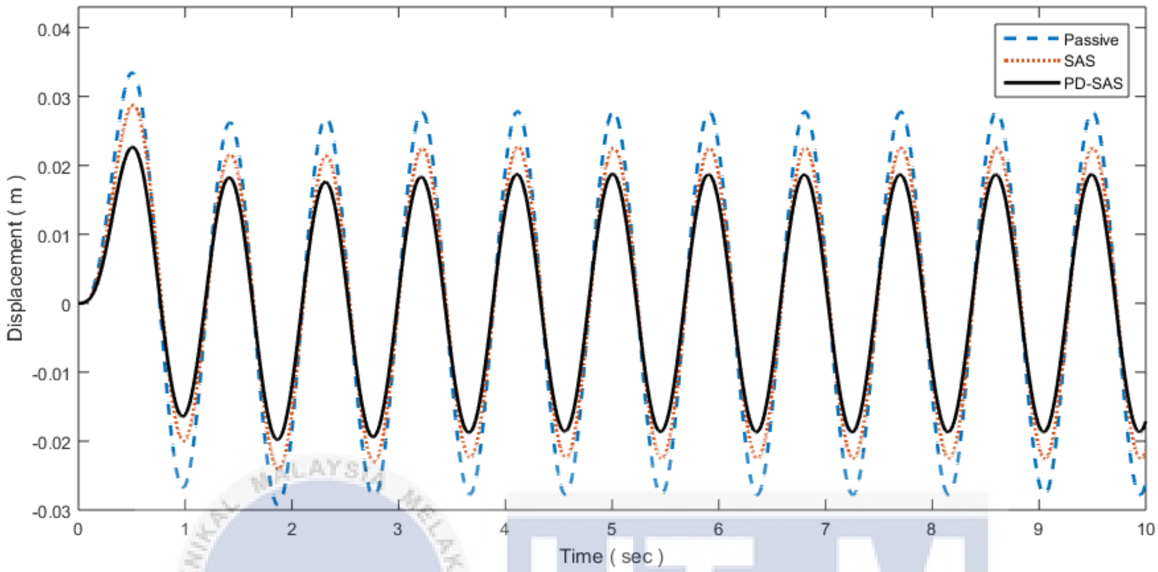

Figure 4.13 : Lateral displacement graph under sinusoidal track irregularity with 7 rad/sec of frequency.

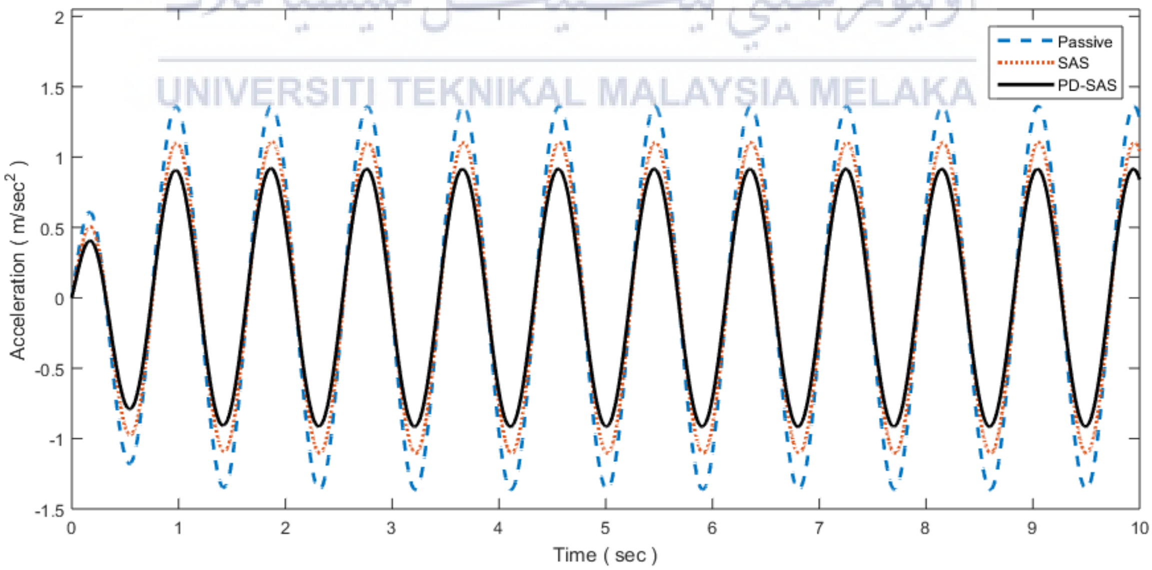

Figure 4.14 : Lateral acceleration graph under sinusoidal track irregularity with 7 rad/sec of frequency.

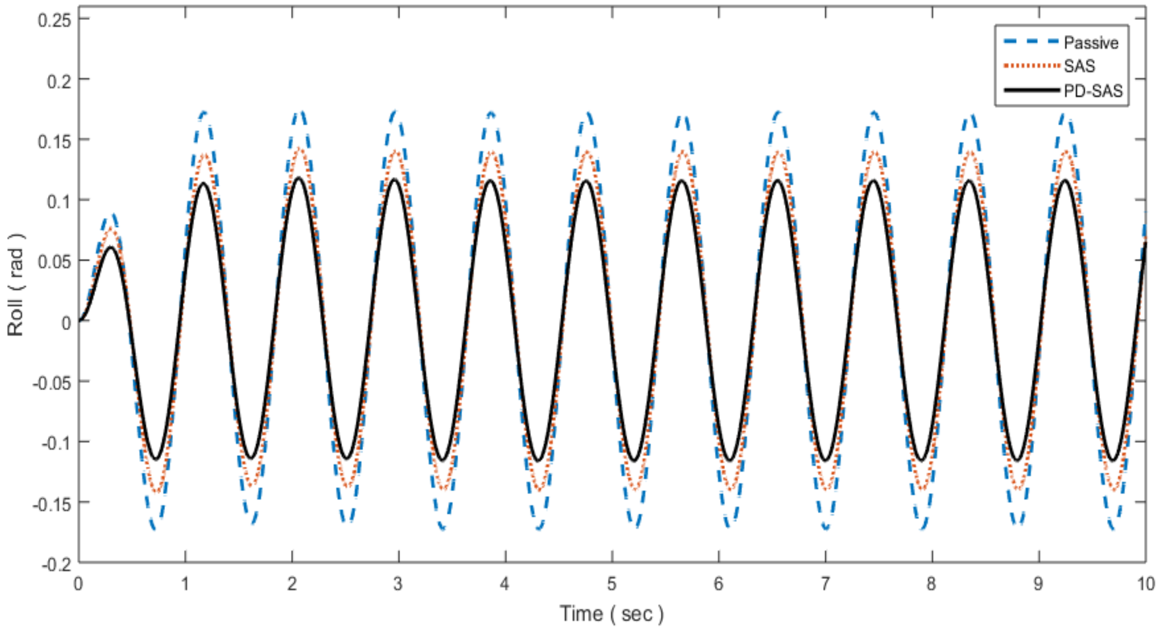

Figure 4.15 : Roll angle graph under sinusoidal track irregularity with 7 rad/sec of frequency

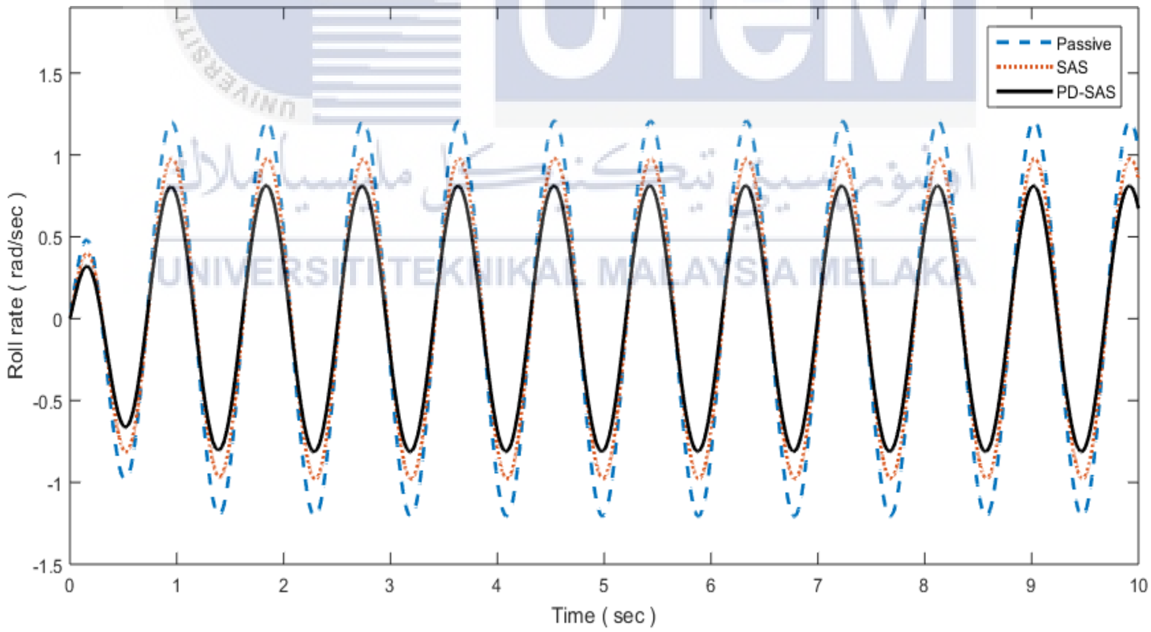

Figure 4.16 : Roll rate graph under sinusoidal track irregularity with 7 rad/sec of frequency

Table 4.4: Peak values of railway vehicle body responses with frequency of 7 rad/s and amplitude 0.05 m

Frequency=7 rad/s Amplitude=0.05 m					
Peak value					
Performance criteria	Passive	SAS	Reduction(%)	PD-SAS	Reduction(%)
Acceleration(m/s2)	1.361	1.103	18.96%	0.9116	33.0%
Displacement(m)	0.02775	0.02234	19.50%	0.01821	34.38%
Roll angle(rad)	0.1728	0.1401	18.92%	0.1162	32.75%
Roll rate(rad/sec)	1.204	0.9649	19.86%	0.8076	32.92%

Figure 4.13, 4.14, 4.15 and 4.16 show that the response of the vehicle performance with the 0.05 m and 7rad/s excitation frequency. Figure 4.13 show the response of the lateral displacement with the passive and active system. From the graph it can be analyze that the active system with 2 controller can reduce the lateral displacement due to the track irregularities. It can be seen that with the SAS and PD-SAS has lower amplitude value compare to the passive. Figure 4.14 show the lateral acceleration and the response is same with the response of the lateral displacement in term of controller are able to damp out the unwanted motion. Response of the roll angle and roll rate can be seen in the figure 4.15 and 4.16 . Both graph clearly state that the performance of the vehicle in term of roll angle and roll rate can be improve using the active system . It can be prove using the amplitude value at passive and active system. active system resulted the lower amplitude value

compare to passive system and even without the PD, SAS still able to cancel out the unwanted motion.

The performance criteria also been analyzed based on peak value which all the data has been recorded in the table 4.4. Based on the lateral displacement, the total reduction for the SAS and PD-SAS is 19.50% and 34.38% compared to the passive system. It can be conclude that the PD-SAS give the higher reduction compared to the SAS. The total reduction for in term of lateral acceleration for SAS is 18.96% while the PD-SAS is 33.0% which is the difference of reduction between SAS and PD-SAS is quite high. The reduction for SAS in term of roll angle and roll rate is 18.92% and 19.86% respectively. So the roll rate recorded the higher reduction in term of SAS compared to the roll angle. In term of PD-SAS, the roll angle and roll rate recorded the total reduction about 32.75% and 32.92% respectively and it can be conclude that , the roll rate state the slightly higher reduction compare to the roll angle. The different between the percentage reduction is about 0.17%.

4.5 Sensitivity analysis

The sensitivity analysis are used as a tuning method for PD in order to find the optimum value of the proportional and derivative. The most accurate value of proportional and derivative is the value are close to zero. The most accurate value will be choose in this study. The value can be found using plotting all the tuning data in the Microsoft excel.

4.5.1 Sensitivity analysis Proportional (P) and Derivative (D) for PD-SAS controller using sine input of 1 rad/s

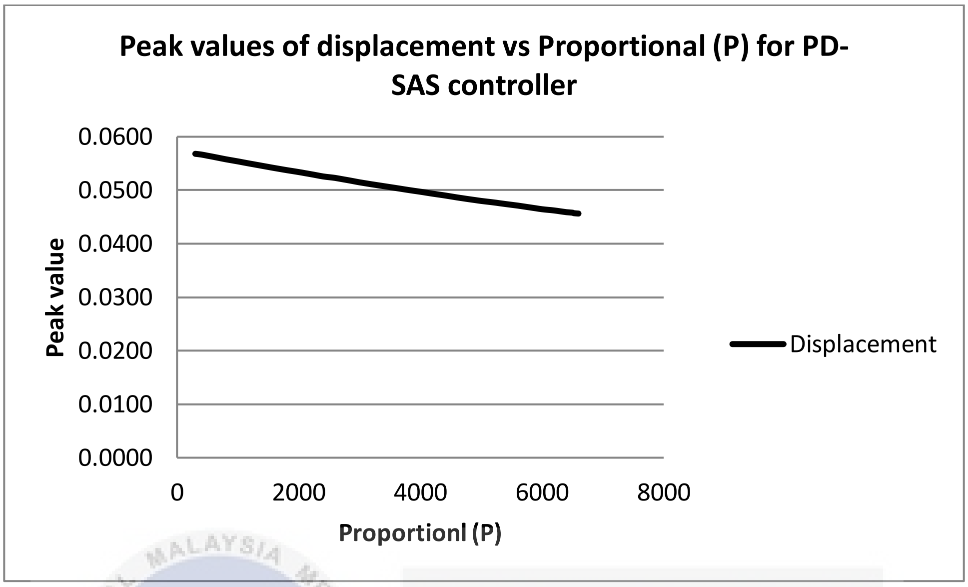

Figure 4.17 : Sensitivity analysis for Peak values of body displacement vs Proportional (P) for PD-SAS controller.

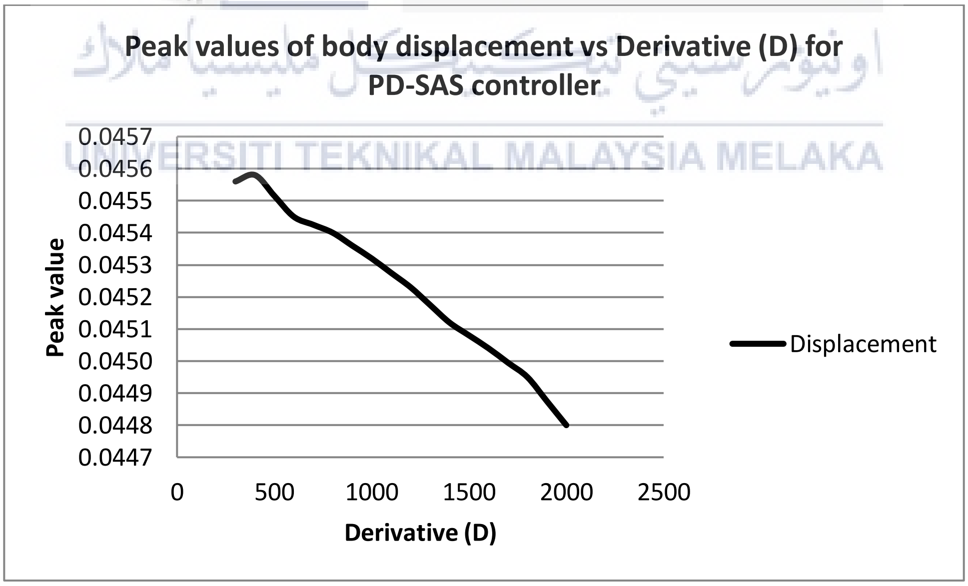

Figure 4.18 : Sensitivity analysis for Peak values of body displacement vs Derivative (D) for PD-SAS controller.

Based on the figure 4.17 above, it clearly shows that the optimum value for P is about 6600 and the lowest peak is 0.0456m while figure 4.18 state that the optimum value of derivative is 2000 and the lowest peak is 0.0448. If the P and D value is increase the value of peak is decrease. The tuning method for the all performance is same where the tuning is continuously until the optimum value or the lowest peak are achived.

4.6 Summary

The response of railway vehicle model for a sinusoidal track irregularity with the amplitude of 0.05 m and 1,3 and 7 rad/sec excitation frequency are presented in the graph in order to analyze the body lateral displacement, body lateral acceleration, roll angle and roll rate response. The response of the performance are also investigated using step input. Every graph are plotted together with the passive and active system in order to compare the performance of the vehicle. The result of the performance are compared with the passive system. Based on all the graph, it can be conclude that semi-active system is able to improve the performance of the vehicle. This is due to the fact that SAS and PD-SAS is able to cancel out the effect of track irregularity before being transmitted to the car body.

CHAPTER 5

CONCLUSION AND RECOMMENDATION

5.0 Introduction

This chapter contains the conclusion of this project and also the future recommendation that can be implemented in this project.

5.1 Conclusion

This project represent a comparison between the passive and semi-active system. The 3-DOF mathematical equation of the railway vehicle model has been analyse. The simulation has been done using Matlab-simulink software in order to simulate the vehicle model and two controller were introduced which is SAS and PD-SAS in order to reduce the unwanted motion and improve the ride quality level of the vehicle. The performance criteria are evaluated under step and sinusoidal track irregularities and the result is compare with the passive and semi-active model system. Performance criteria that have been considered and analysed in this project are body lateral displacement, body lateral acceleration, roll angle and roll rate at body's centre of gravity while the peak to peak values percentage reduction of the response are also compared. The result from the simulation shows that the controller has the ability to reduce the unwanted motion and improve the ride quality of the vehicle model.

5.2 Recommendation

As the recommendation for the further research, Firstly, the simulation model must be validate with the experiment test rig. Secondly, the method of tuning the controller need to use the genetic algorithm in order to get the optimum value accurately.

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APPENDIX A

Sensitivity analysis Proportional (P) and Derivative (D) for PD-SAS controller using step input

Table 1: Sensitivity analysis value of Proportional (P) for PD-SAS controller using step input

No	Proportional (P)	Displacement	Acceleration	Roll	Roll rate
1	10	0.0674	0.4933	0.0899	0.3808
2	20	0.0672	0.4920	0.0897	0.3799
3	30	0.0661	0.4890	0.0894	0.3796
4	40	0.0655	0.4873	0.0890	0.3793
5	50	0.0654	0.4860	0.0886	0.3789
6	60	0.0653	0.4840	0.0882	0.3786
7	70	0.0651	0.4832	0.0873	0.3780
8	80	0.0645	0.4812	0.0867	0.3778
9	90	0.0640	0.4802	0.0856	0.3772
10	100	0.0638	0.4799	0.0846	0.3770
11	110	0.0634	0.4780	0.0838	0.3765
12	120	0.0631	0.4774	0.0826	0.3760
13	130	0.0629	0.4765	0.0821	0.3758
14	140	0.0626	0.4756	0.0800	0.3756
15	150	0.0622	0.4723	0.0799	0.3746
16	160	0.0620	0.4712	0.0789	0.3740
17	170	0.0618	0.4699	0.0779	0.3734
18	180	0.0612	0.4685	0.0764	0.3712
19	190	0.0606	0.4670	0.0756	0.3701
20	200	0.0604	0.4672	0.0740	0.3700
21	210	0.0604	0.4673	0.0741	0.3701
22	220	0.0605	0.4674	0.0743	0.3702
23	230	0.0606	0.4675	0.0744	0.3703

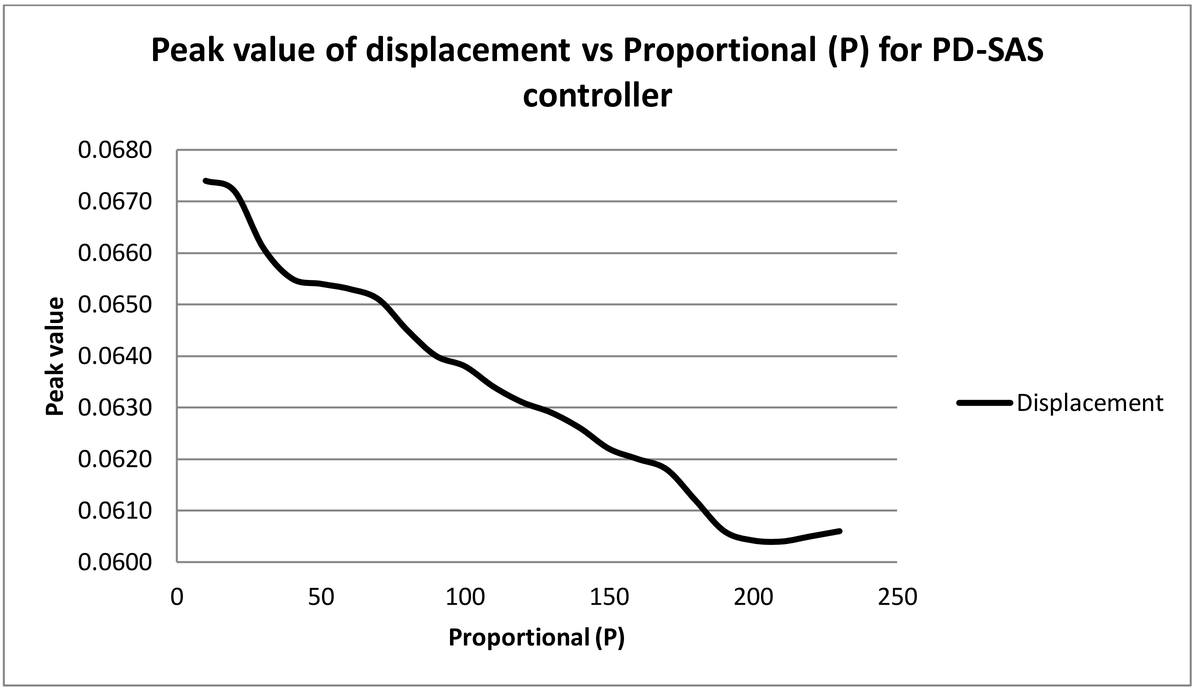

Figure 1 : Sensitivity analysis for Peak values of body displacement vs Proportional (P) for PD-SAS controller.

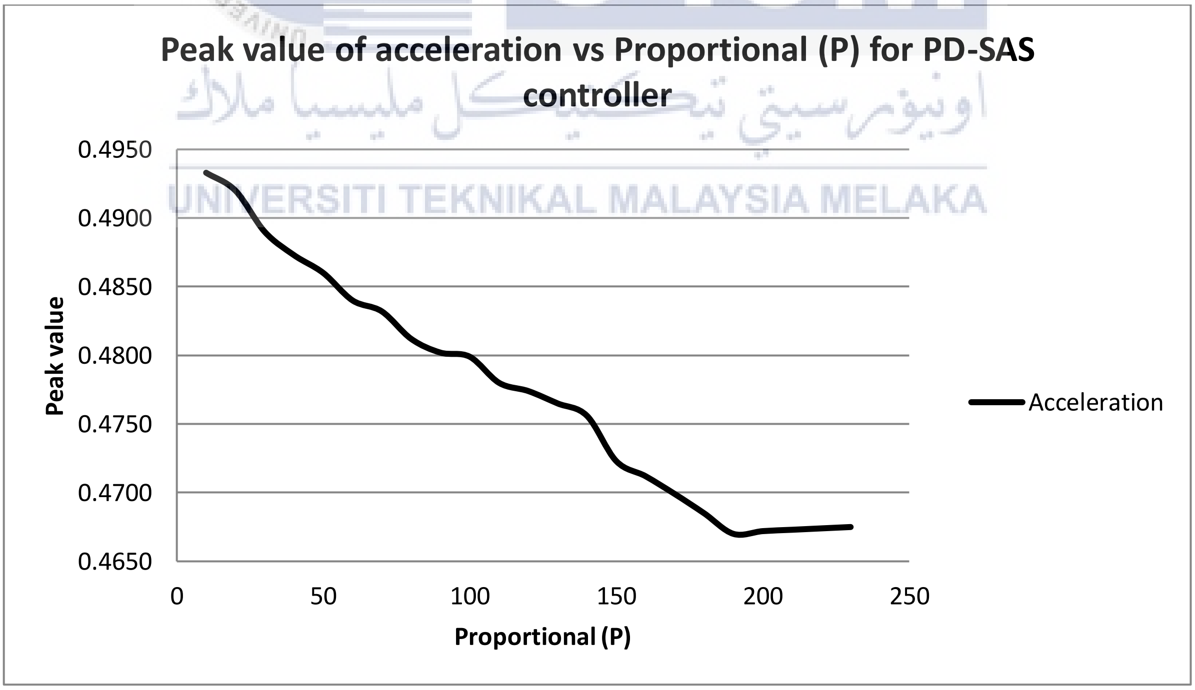

Figure 2 : Sensitivity analysis for Peak values of body acceleration vs Proportional (P) for PD-SAS controller.

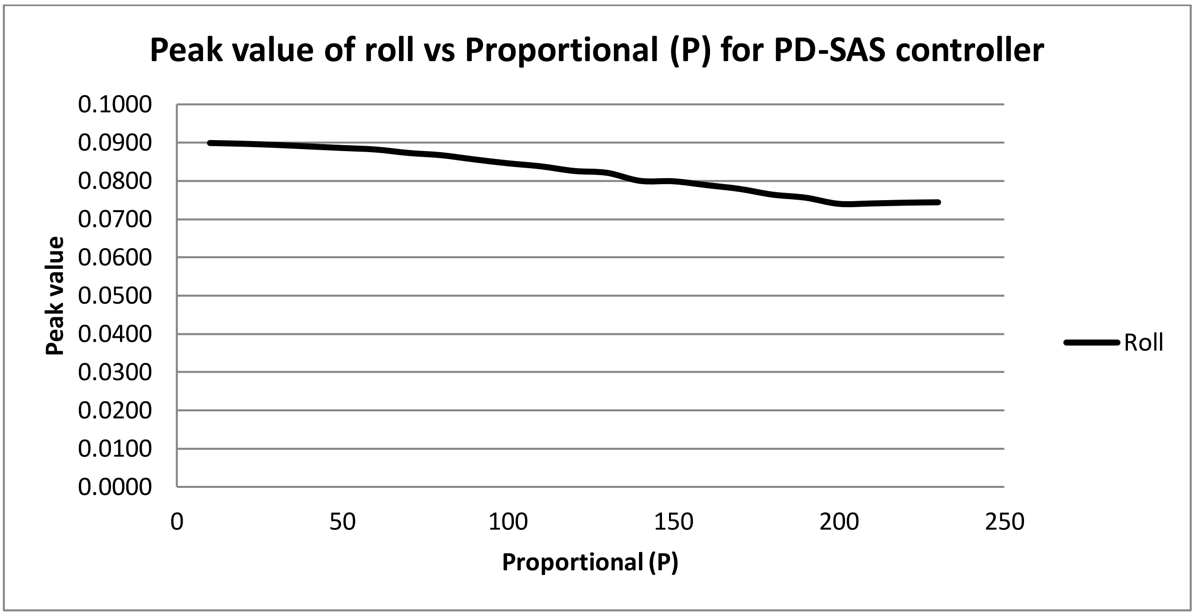

Figure 3 : Sensitivity analysis for Peak values of roll vs Proportional (P) for PD-SAS controller.

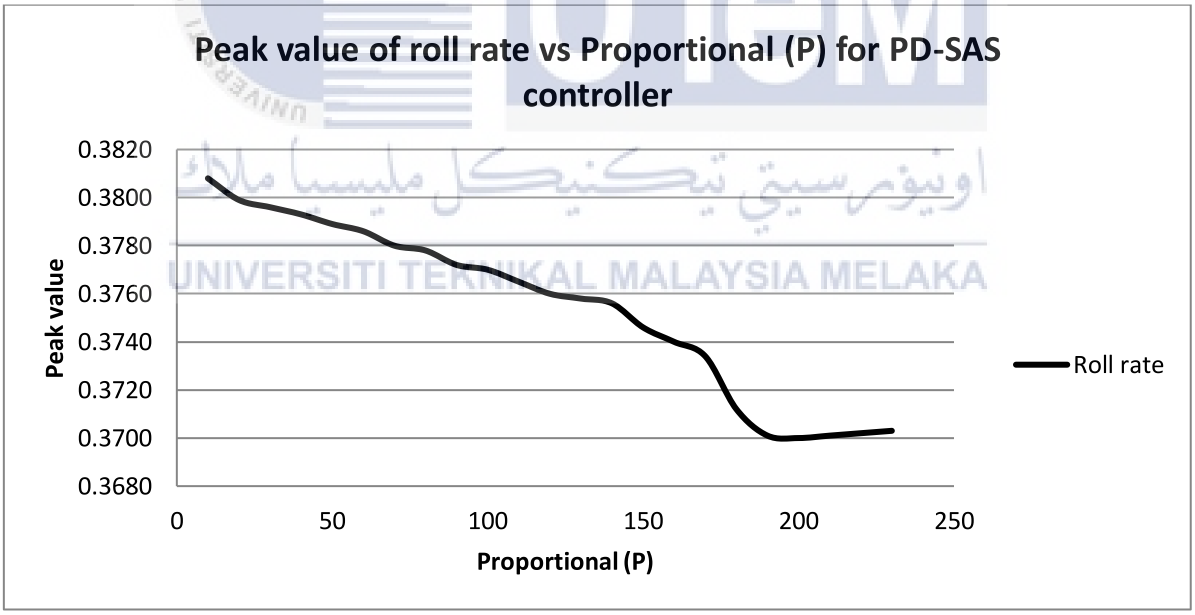

Figure 4: Sensitivity analysis for Peak values of roll rate vs Proportional (P) for PD-SAS controller.

Table 2: Sensitivity analysis value of Derivative for PD-SAS controller using step input.

No	Derivative (D)	Displacement	Acceleration	Roll	Roll rate
1	100	0.0609	0.4756	0.0741	0.3821
2	200	0.0607	0.4744	0.0740	0.3806
3	300	0.0605	0.4740	0.0739	0.3799
4	400	0.0602	0.4733	0.0738	0.3794
5	500	0.0600	0.4731	0.0737	0.3789
6	600	0.0599	0.4724	0.0735	0.3786
7	700	0.0596	0.4721	0.0734	0.3782
8	800	0.0594	0.4718	0.0733	0.3778
9	900	0.0592	0.4712	0.0732	0.3774
10	1000	0.0590	0.4709	0.0731	0.3770
11	1100	0.0588	0.4690	0.0729	0.3768
12	1200	0.0584	0.4687	0.0728	0.3761
13	1300	0.0582	0.4669	0.0727	0.3754
14	1400	0.0580	0.4664	0.0726	0.3745
15	1500	0.0579	0.4650	0.0725	0.3737
16	1600	0.0577	0.4648	0.0724	0.3729
17	1700	0.0576	0.4640	0.0722	0.3723
18	1800	0.0574	0.4636	0.0721	0.3713
19	1900	0.0571	0.4623	0.0720	0.3705
20	2000	0.0569	0.4612	0.0719	0.3692
21	2100	0.0568	0.4600	0.0718	0.3690
22	2200	0.0566	0.4599	0.0717	0.3688
23	2300	0.0563	0.4594	0.0716	0.3684
24	2400	0.0561	0.4591	0.0715	0.3682
25	2500	0.0560	0.4588	0.0714	0.3680
26	2600	0.0559	0.4580	0.0712	0.3679
27	2700	0.0557	0.4576	0.0711	0.3676
28	2800	0.0550	0.4572	0.0710	0.3673
29	2900	0.0554	0.4567	0.0708	0.3670
30	3000	0.0551	0.4544	0.0707	0.3667
31	3100	0.0549	0.4538	0.0706	0.3665
32	3200	0.0550	0.4543	0.0706	0.3667
33	3300	0.0551	0.4547	0.0707	0.3670

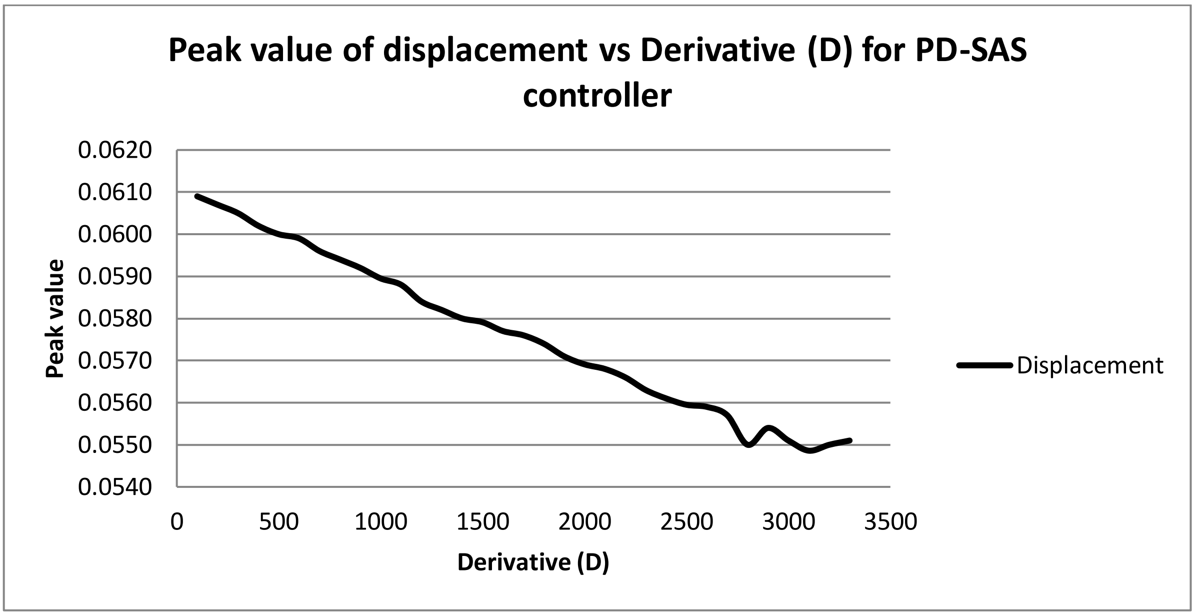

Figure 5 : Sensitivity analysis for Peak values of body displacement vs Derivative (D) for PD-SAS controller.

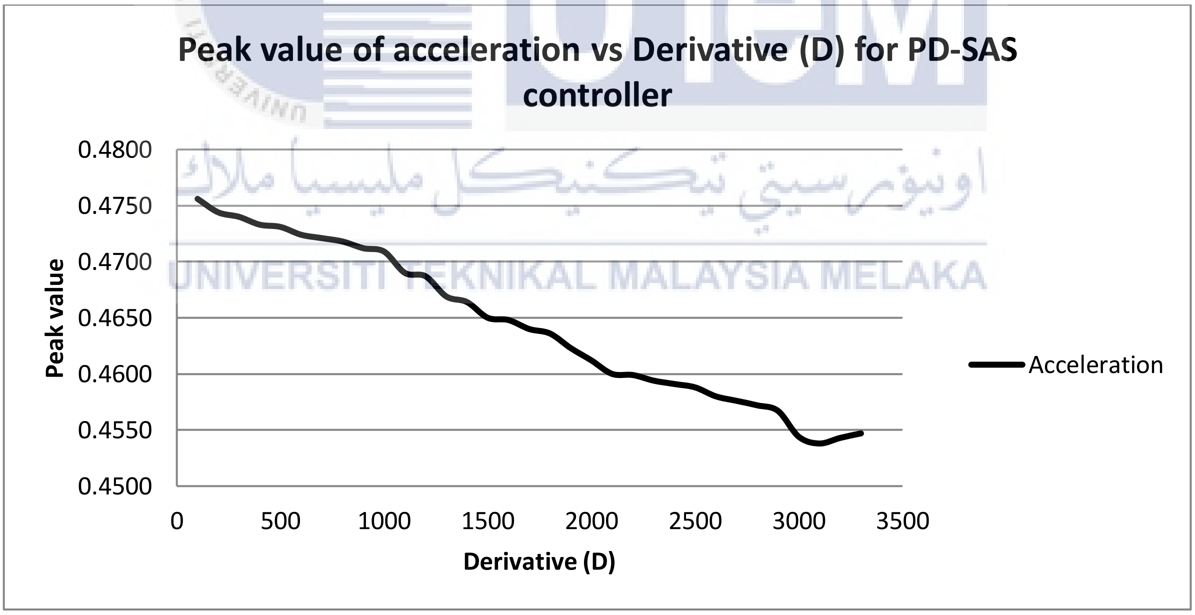

Figure 6 : Sensitivity analysis for Peak values of body acceleration vs Derivative (D) for PD-SAS controller.

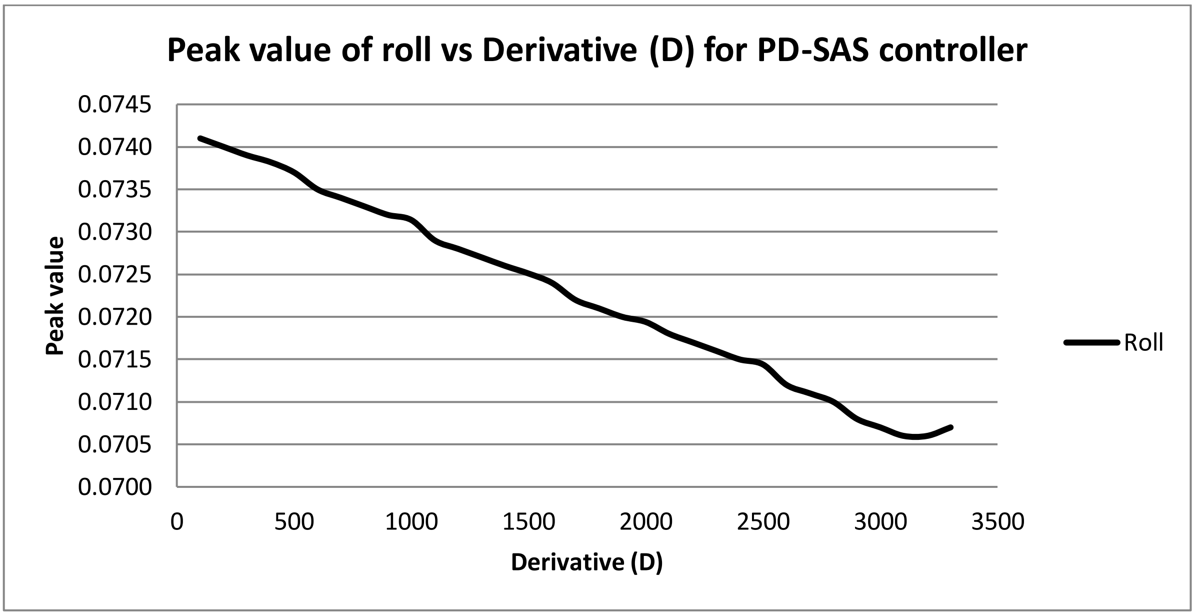

Figure 7 : Sensitivity analysis for Peak values of roll vs Derivative (D) for PD-SAS controller.

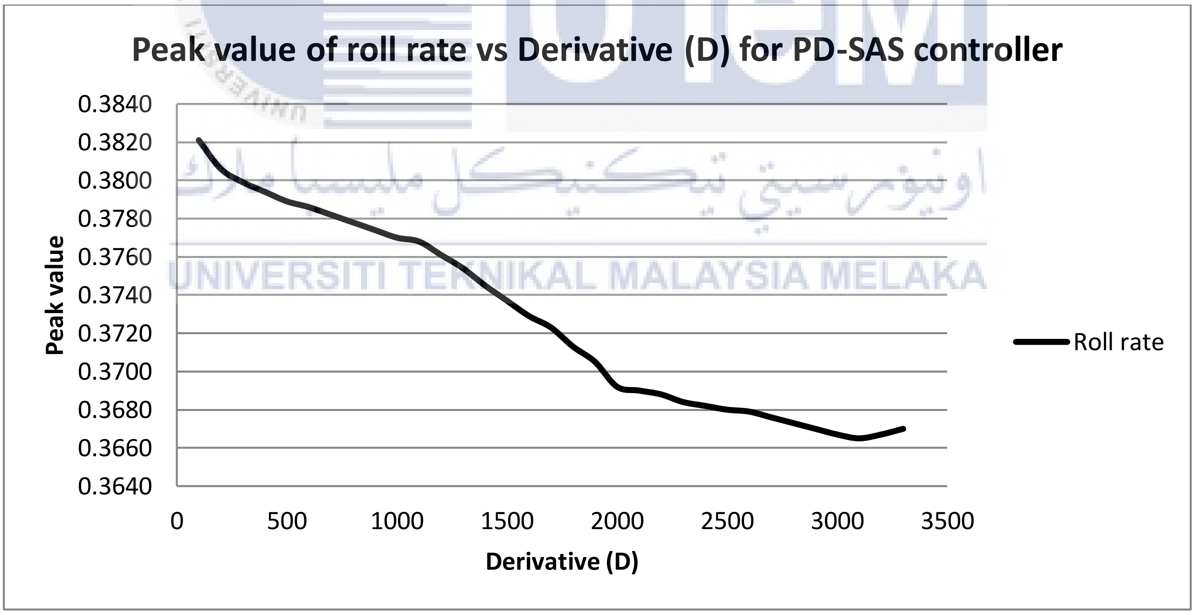

Figure 8 : Sensitivity analysis for Peak values of roll rate vs Derivative (D) for PD-SAS controller.

APPENDIX B

Sensitivity analysis Proportional (P) and Derivative (D) for PD-SAS controller using sine input of 1 rad/s

Table 3: Sensitivity analysis value of Proportional (P) for PD-SAS controller using sine input of 1 rad/s.

No	Proportional (P)	Displacement	Aceeleration	Roll	Roll rate
1	300	0.0568	0.0600	0.0312	0.0324
2	400	0.0567	0.0598	0.0311	0.0323
3	500	0.0565	0.0597	0.0310	0.0323
4	600	0.0562	0.0595	0.0309	0.0321
5	700	0.0560	0.0593	0.0308	0.0320
6	800	0.0558	0.0591	0.0307	0.0319
7	900	0.0556	0.0589	0.0306	0.0318
8	1000	0.0554	0.0588	0.0305	0.0317
9	1100	0.0552	0.0586	0.0304	0.0316
10	1200	0.0550	0.0585	0.0303	0.0315
11	1300	0.0548	0.0583	0.0303	0.0315
12	1400	0.0546	0.0581	0.0302	0.0314
13	1500	0.0544	0.0579	0.0301	0.0313
14	1600	0.0542	0.0578	0.0300	0.0312
15	1700	0.0540	0.0576	0.0299	0.0311
16	1800	0.0538	0.0574	0.0298	0.0310
17	1900	0.0536	0.0573	0.0297	0.0309
18	2000	0.0534	0.0571	0.0296	0.0308
19	2100	0.0532	0.0570	0.0295	0.0307
20	2200	0.0530	0.0568	0.0294	0.0306
21	2300	0.0528	0.0567	0.0293	0.0305
22	2400	0.0526	0.0565	0.0292	0.0304
23	2500	0.0524	0.0564	0.0291	0.0303
24	2600	0.0523	0.0562	0.0290	0.0302
25	2700	0.0521	0.0561	0.0289	0.0301
26	2800	0.0519	0.0559	0.0288	0.0300
27	2900	0.0517	0.0558	0.0287	0.0299
28	3000	0.0515	0.0556	0.0286	0.0298
29	3100	0.0513	0.0554	0.0286	0.0297
30	3200	0.0511	0.0552	0.0285	0.0295

31	3300	0.0509	0.0551	0.0284	0.0295
32	3400	0.0508	0.0549	0.0282	0.0294
33	3500	0.0506	0.0548	0.0281	0.0293
34	3600	0.0504	0.0547	0.0280	0.0292
35	3700	0.0502	0.0545	0.0279	0.0291
36	3800	0.0501	0.0543	0.0278	0.0290
37	3900	0.0499	0.0541	0.0277	0.0289
38	4000	0.0497	0.0540	0.0276	0.0288
39	4100	0.0495	0.0539	0.0275	0.0287
40	4200	0.0494	0.0537	0.0274	0.0289
41	4300	0.0492	0.0535	0.0273	0.0288
42	4400	0.0490	0.0533	0.0273	0.0286
43	4500	0.0488	0.0532	0.0273	0.0285
44	4600	0.0487	0.0531	0.0272	0.0285
45	4700	0.0485	0.0530	0.0272	0.0284
46	4800	0.0483	0.0528	0.0272	0.0284
47	4900	0.0482	0.0527	0.0271	0.0283
48	5000	0.0480	0.0525	0.0271	0.0282
49	5100	0.0479	0.0524	0.0271	0.0281
50	5200	0.0477	0.0523	0.0271	0.0281
51	5300	0.0476	0.0521	0.0270	0.0280
52	5400	0.0474	0.0519	0.0270	0.0279
53	5500	0.0473	0.0518	0.0269	0.0279
54	5600	0.0471	0.0517	0.0268	0.0278
55	5700	0.0470	0.0515	0.0268	0.0276
56	5800	0.0468	0.0514	0.0267	0.0275
57	5900	0.0466	0.0513	0.0266	0.0275
58	6000	0.0465	0.0512	0.0265	0.0275
59	6100	0.0463	0.0510	0.0265	0.0273
60	6200	0.0462	0.0507	0.0264	0.0271
61	6300	0.0461	0.0507	0.0263	0.0271
62	6400	0.0459	0.0506	0.0262	0.0271
63	6500	0.0458	0.0505	0.0261	0.0271
64	6525	0.0457	0.0505	0.0261	0.0271
65	6600	0.0456	0.0503	0.0261	0.0270
66	6700	0.0456	0.0505	0.0262	0.0271
67	6800	0.0456	0.0505	0.0263	0.0273
58	6900	0.0457	0.0506	0.0264	0.0275
69	7000	0.0458	0.0507	0.0264	0.0276

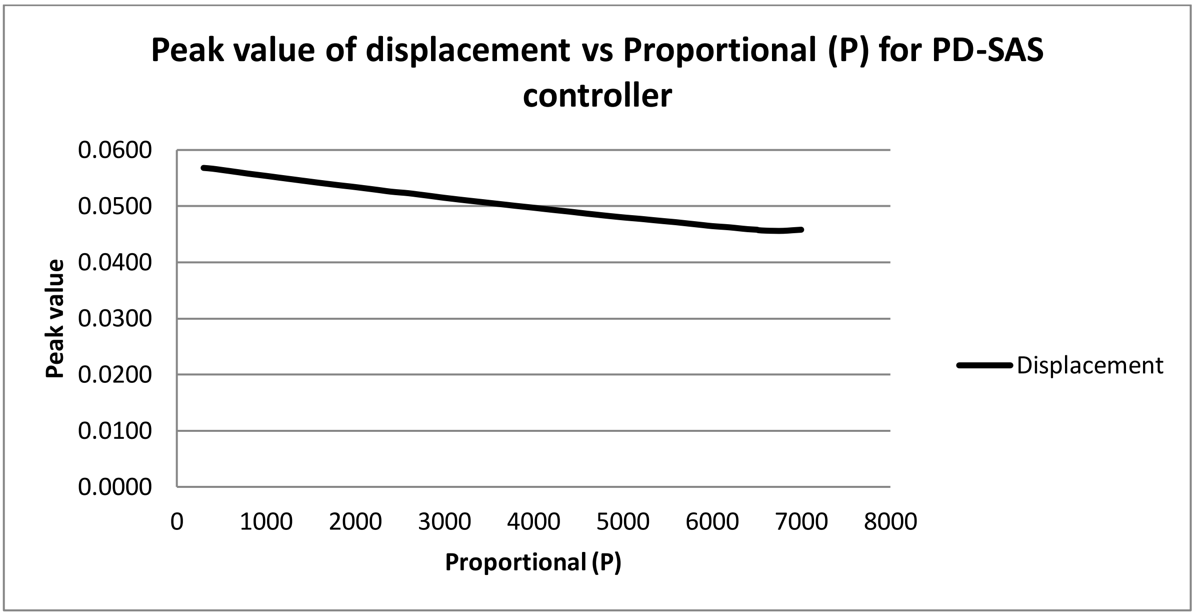

Figure 9 : Sensitivity analysis for Peak values of body displacement vs Proportional (P) for PD-SAS controller.

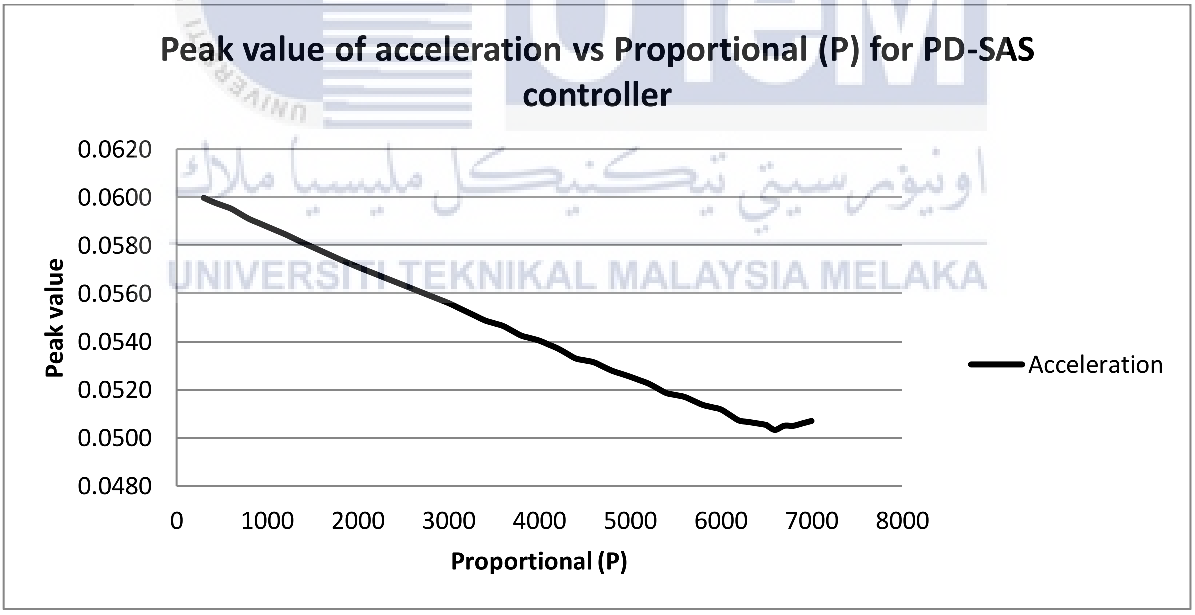

Figure 10 : Sensitivity analysis for Peak values of body acceleration vs Proportional (P) for PD-SAS controller.

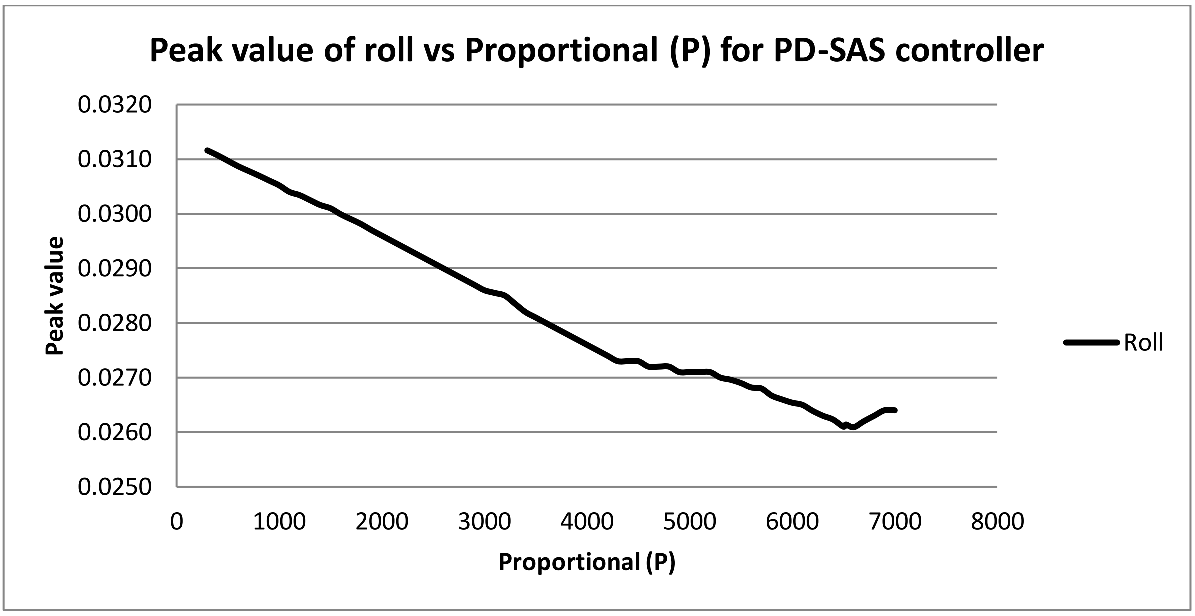

Figure 11 : Sensitivity analysis for Peak values of roll vs Proportional (P) for PD-SAS controller.

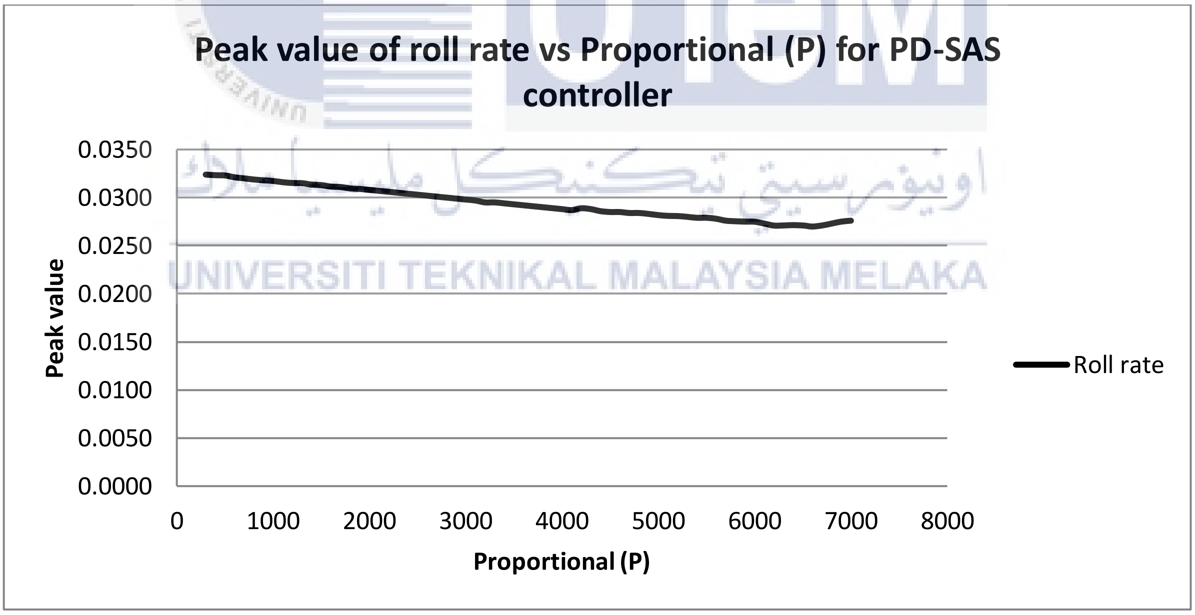

Figure 12 : Sensitivity analysis for Peak values of roll rate vs Proportional (P) for PD-SAS controller.

Table 4: Sensitivity analysis value of Derivative (D) for PD-SAS controller using sine input of 1 rad/s.

No	Derivative (D)	Displacement	Aceeleration	Roll	Roll rate
1	300	0.0456	0.0503	0.0261	0.0271
2	400	0.0456	0.0503	0.0261	0.0270
3	500	0.0455	0.0503	0.0261	0.0270
4	600	0.0455	0.0503	0.0260	0.0271
5	700	0.0454	0.0502	0.0260	0.0270
6	800	0.0454	0.0502	0.0260	0.0270
7	900	0.0454	0.0501	0.0260	0.0270
8	1000	0.0453	0.0501	0.0259	0.0270
9	1100	0.0453	0.0500	0.0259	0.0270
10	1200	0.0452	0.0499	0.0259	0.0270
11	1300	0.0452	0.0499	0.0259	0.0269
12	1400	0.0451	0.0498	0.0259	0.0269
13	1500	0.0451	0.0498	0.0258	0.0269
14	1600	0.0450	0.0497	0.0258	0.0269
15	1700	0.0450	0.0497	0.0258	0.0268
16	1800	0.0450	0.0497	0.0258	0.0268
17	1900	0.0449	0.0497	0.0257	0.0268
18	2000	0.0448	0.0497	0.0257	0.0268
19	2100	0.0448	0.0498	0.0258	0.0269
20	2200	0.0449	0.0499	0.0259	0.0270
21	2300	0.0449	0.0500	0.0260	0.0271
22	2400	0.0450	0.0501	0.0261	0.0272

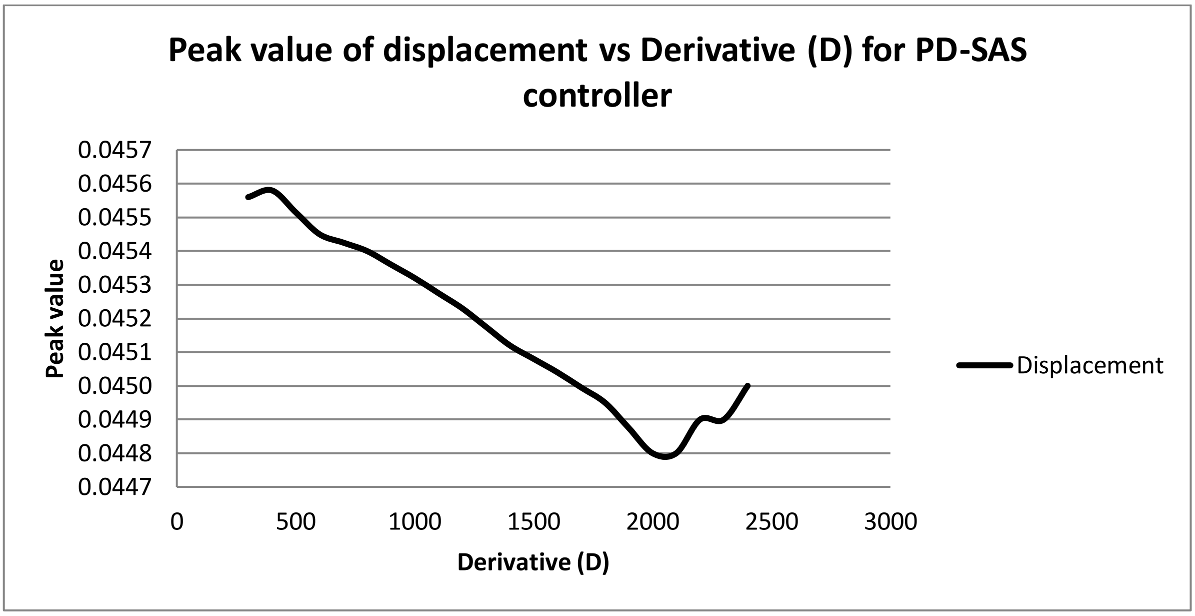

Figure 13 : Sensitivity analysis for Peak values of body displacement vs Derivative (D) for PD-SAS controller.

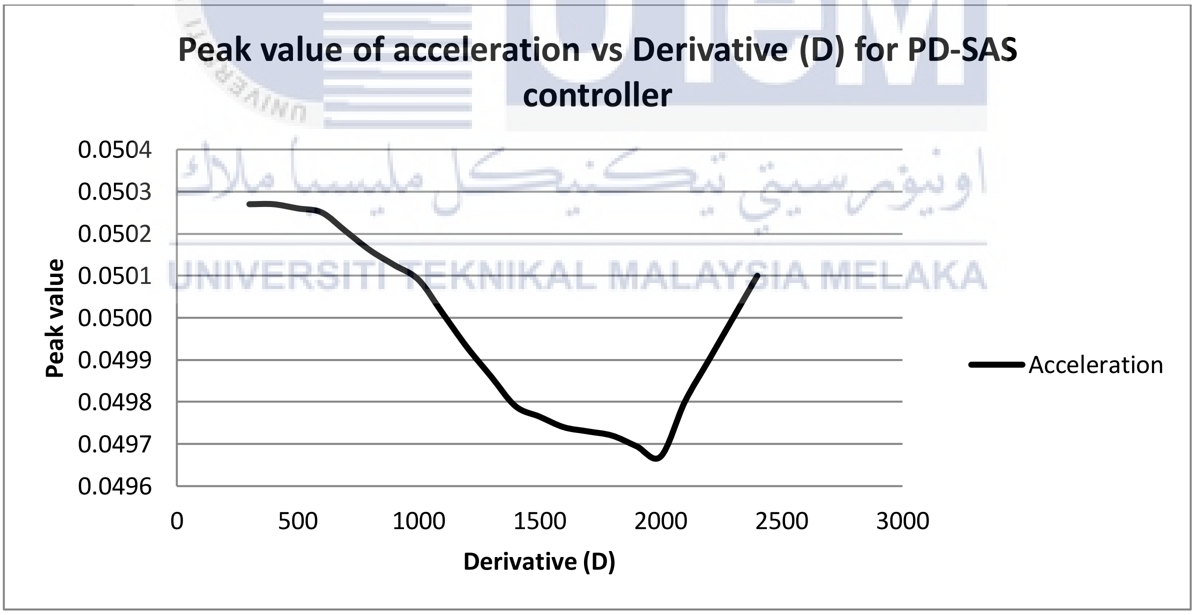

Figure 14 : Sensitivity analysis for Peak values of body acceleration vs Derivative (D) for PD-SAS controller.

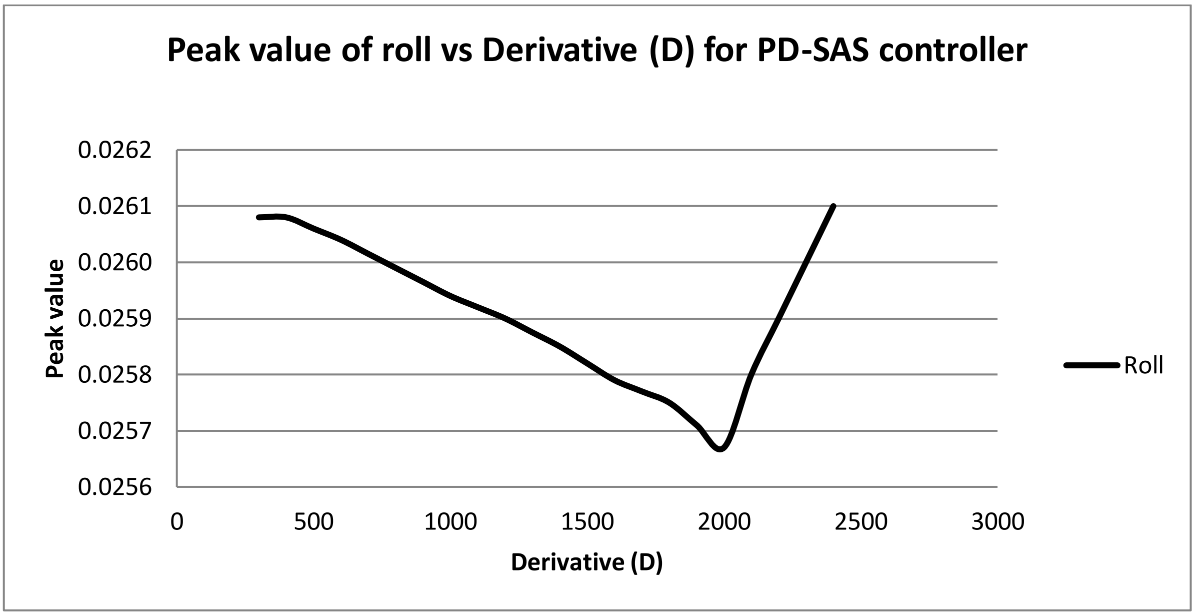

Figure 15 : Sensitivity analysis for Peak values of roll vs Derivative (D) for PD-SAS controller.

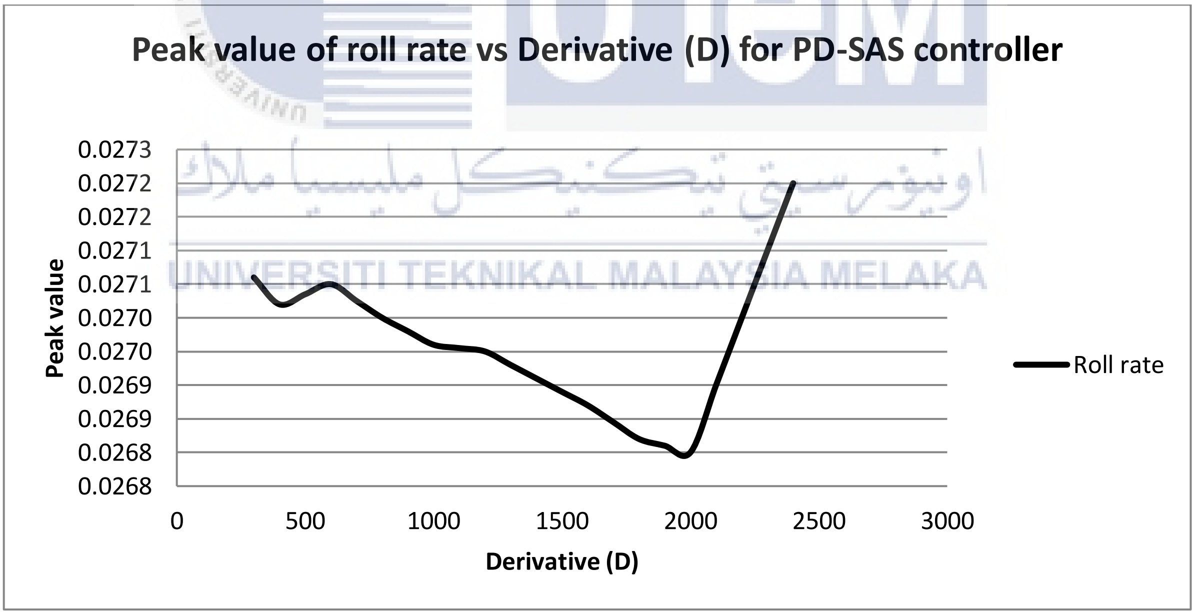

Figure 16 : Sensitivity analysis for Peak values of roll rate vs Derivative (D) for PD-SAS controller.

APPENDIX C

Sensitivity analysis Proportional (P) and Derivative (D) for PD-SAS controller using sine input of 3 rad/s

Table 5: Sensitivity analysis value of Proportional (P) for PD-SAS controller using sine input of 3 rad/s.

No	Proportional (P)	Displacement	Aceeleration	Roll	Roll rate
1	300	0.0441	0.4041	0.1138	0.3257
2	400	0.0441	0.4032	0.1136	0.3202
3	500	0.0439	0.4024	0.1133	0.3194
4	600	0.0438	0.4011	0.1132	0.3187
5	700	0.0437	0.4005	0.1127	0.3178
6	800	0.0435	0.3995	0.1125	0.3172
7	900	0.0434	0.3985	0.1123	0.3169
8	1000	0.0433	0.3976	0.1119	0.3157
9	1100	0.0432	0.3967	0.1117	0.3150
10	1200	0.0431	0.3957	0.1114	0.3142
11	1300	0.0430	0.3948	0.1111	0.3129
12	1400	0.0429	0.3938	0.1108	0.3116
13	1500	0.0428	0.3928	0.1105	0.3115
14	1600	0.0426	0.3918	0.1102	0.3113
15	1700	0.0425	0.3909	0.1100	0.3106
16	1800	0.0424	0.3900	0.1097	0.3098
17	1900	0.0423	0.3890	0.1095	0.3091
18	2000	0.0422	0.3879	0.1092	0.3083
19	2100	0.0421	0.3872	0.1090	0.3076
20	2200	0.0420	0.3865	0.1087	0.3069
21	2300	0.0419	0.3856	0.1084	0.3063
22	2400	0.0417	0.3846	0.1081	0.3056
23	2500	0.0416	0.3836	0.1079	0.3049
24	2600	0.0415	0.3826	0.1076	0.3042
25	2700	0.0412	0.3818	0.1073	0.3034
26	2800	0.0413	0.3809	0.1069	0.3026
27	2900	0.0412	0.3801	0.1067	0.3019
28	3000	0.0411	0.3792	0.1065	0.3012
29	3100	0.0410	0.3782	0.1063	0.3006
30	3200	0.0409	0.3771	0.1060	0.2999

31	3300	0.0408	0.3763	0.1058	0.2992
32	3400	0.0407	0.3755	0.1055	0.2985
33	3500	0.0406	0.3747	0.1053	0.2979
34	3600	0.0405	0.3738	0.1050	0.2972
35	3700	0.0406	0.3730	0.1048	0.2964
36	3800	0.0403	0.3721	0.1045	0.2957
37	3900	0.0402	0.3712	0.1043	0.2951
38	4000	0.0400	0.3703	0.1040	0.2944
39	4100	0.0399	0.3695	0.1038	0.2937
40	4200	0.0398	0.3686	0.1035	0.2930
41	4300	0.0397	0.3677	0.1033	0.2924
42	4400	0.0396	0.3668	0.1030	0.2917
43	4500	0.0395	0.3660	0.1028	0.2910
44	4600	0.0394	0.3651	0.1025	0.2903
45	4700	0.0393	0.3643	0.1023	0.2897
46	4800	0.0392	0.3634	0.1020	0.2890
47	4900	0.0391	0.3625	0.1018	0.2883
48	5000	0.0390	0.3616	0.1015	0.2876
49	5100	0.0389	0.3608	0.1013	0.2870
50	5200	0.0388	0.3600	0.1010	0.2863
51	5300	0.0387	0.3592	0.1008	0.2857
52	5400	0.0386	0.3583	0.1005	0.2850
53	5500	0.0385	0.3574	0.1003	0.2844
54	5600	0.0384	0.3565	0.1000	0.2838
55	5700	0.0383	0.3558	0.0998	0.2832
56	5800	0.0382	0.3550	0.0996	0.2825
57	5900	0.0381	0.3542	0.0993	0.2818
58	6000	0.0380	0.3534	0.0991	0.2811
59	6100	0.0379	0.3526	0.0989	0.2804
60	6200	0.0378	0.3518	0.0986	0.2797
61	6300	0.0377	0.3510	0.0984	0.2792
62	6400	0.0377	0.3501	0.0982	0.2786
63	6500	0.0376	0.3493	0.0979	0.2779
64	6510	0.0376	0.3492	0.0979	0.2779
65	6520	0.0375	0.3491	0.0979	0.2778
66	6525	0.0375	0.3491	0.0978	0.2778
67	6600	0.0375	0.3490	0.0977	0.2779
68	6700	0.0376	0.3490	0.0977	0.2779
69	6800	0.0377	0.3490	0.0978	0.2778
70	6900	0.0378	0.3491	0.0979	0.2777
71	7000	0.0379	0.3492	0.0982	0.2775

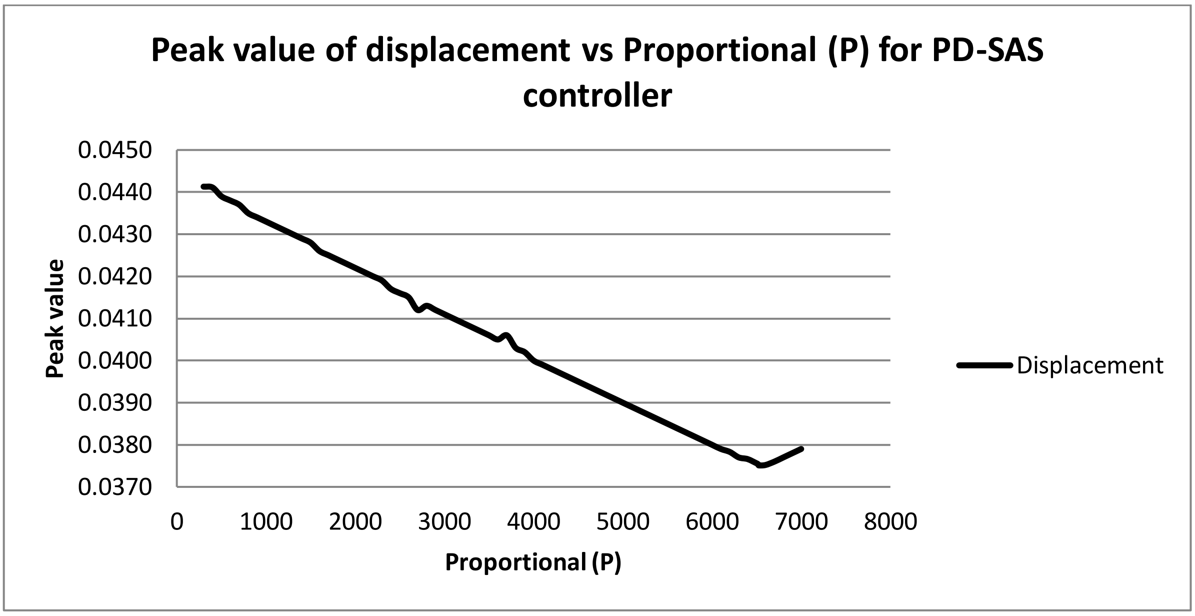

Figure 17 : Sensitivity analysis for Peak values of body displacement vs Proportional (P) for PD-SAS controller.

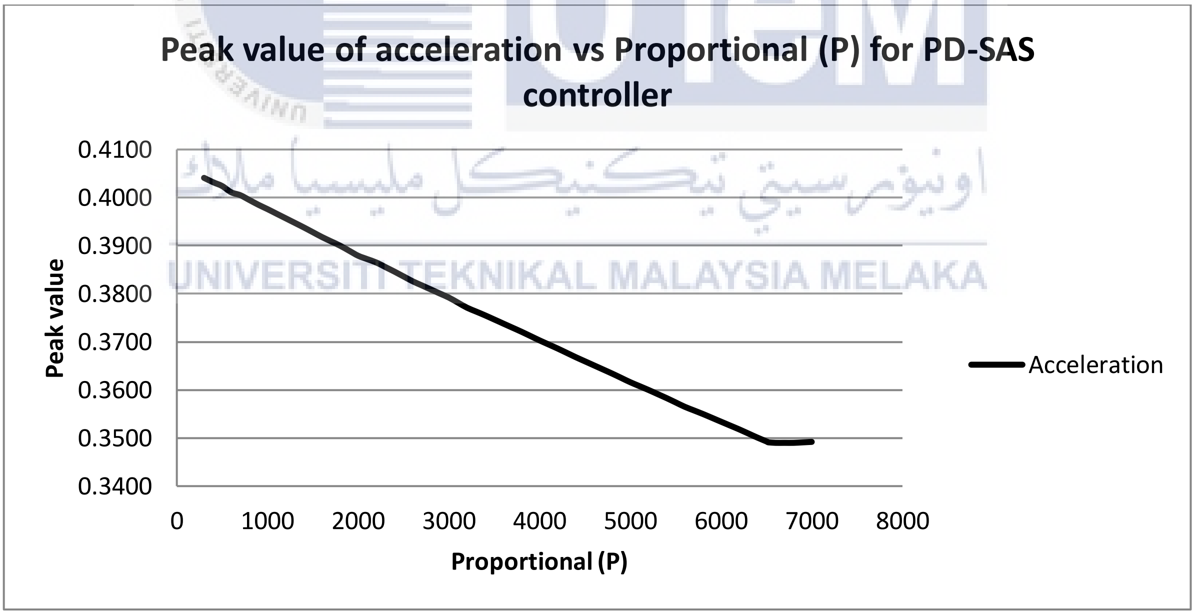

Figure 18 : Sensitivity analysis for Peak values of body acceleration vs Proportional (P) for PD-SAS controller.

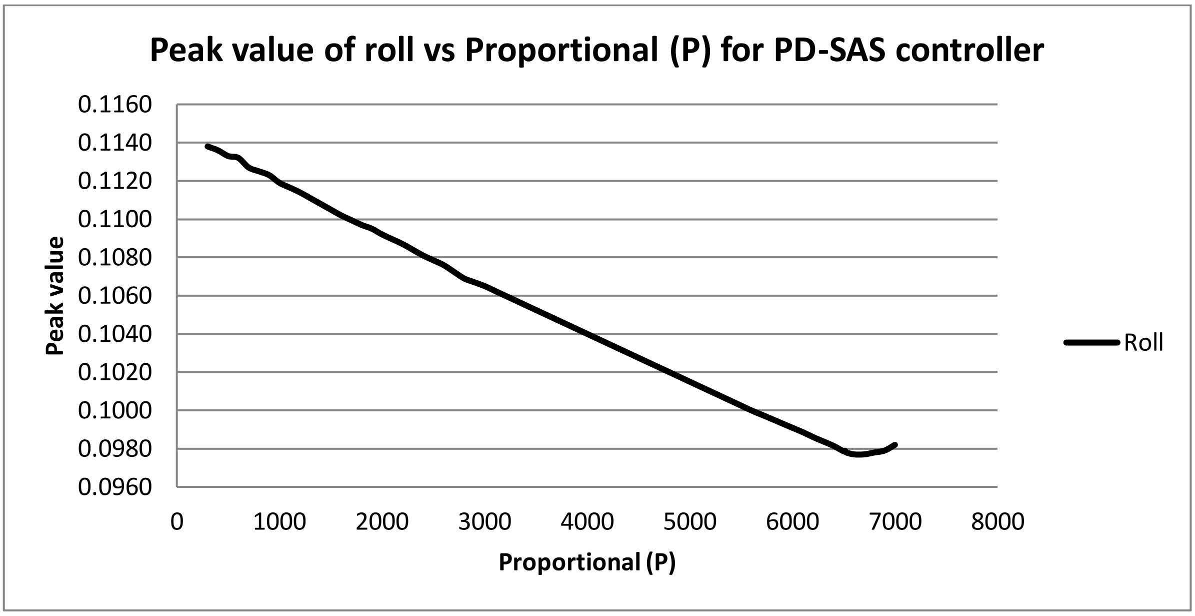

Figure 19 : Sensitivity analysis for Peak values of roll vs Proportional (P) for PD-SAS controller.

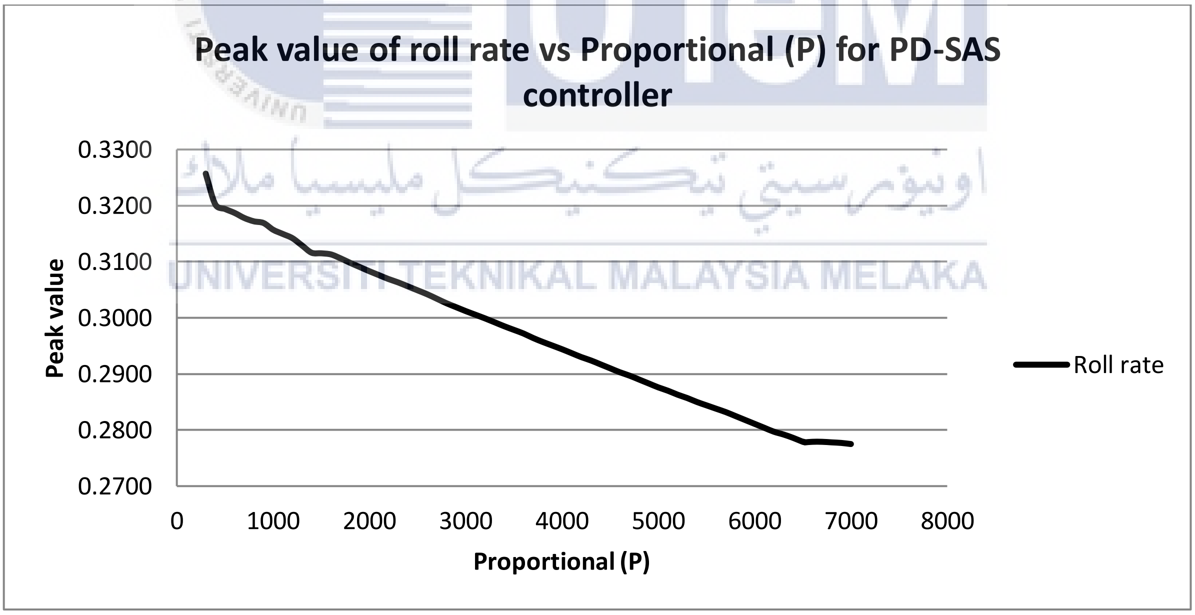

Figure 20 : Sensitivity analysis for Peak values of roll rate vs Proportional (P) for PD-SAS controller.

Table 6: Sensitivity analysis value of Derivative (D) for PD-SAS controller using sine input of 3 rad/s.

No	Derivative (D)	Displacement	Aceeleration	Roll	Roll rate
1	300	0.0368	0.3429	0.0962	0.2729
2	400	0.0366	0.3408	0.0956	0.2711
3	500	0.0363	0.3388	0.0927	0.2696
4	600	0.0361	0.3367	0.0928	0.2680
5	700	0.0359	0.3347	0.0916	0.2664
6	800	0.0356	0.3327	0.0934	0.2648
7	900	0.0354	0.3306	0.0928	0.2633
8	1000	0.0352	0.3284	0.0923	0.2617
9	1100	0.0349	0.3266	0.0918	0.2600
10	1200	0.0347	0.3248	0.0912	0.2586
11	1300	0.0345	0.3229	0.0883	0.2578
12	1400	0.0342	0.3210	0.0853	0.2570
13	1500	0.0340	0.3156	0.0848	0.2548
14	1600	0.0338	0.3101	0.0842	0.2526
15	1700	0.0336	0.3081	0.0837	0.2516
16	1800	0.0334	0.3060	0.0831	0.2505
17	1900	0.0332	0.3042	0.0826	0.2489
18	2000	0.0329	0.3023	0.0821	0.2473
19	2100	0.0327	0.3004	0.0816	0.2461
20	2200	0.0325	0.2985	0.0810	0.2448
21	2300	0.0323	0.2966	0.0856	0.2430
22	2400	0.0324	0.2965	0.0860	0.2412
23	2500	0.0324	0.2965	0.0860	0.2412
24	2600	0.0324	0.2967	0.0860	0.2412
25	2700	0.0325	0.2968	0.0861	0.2413

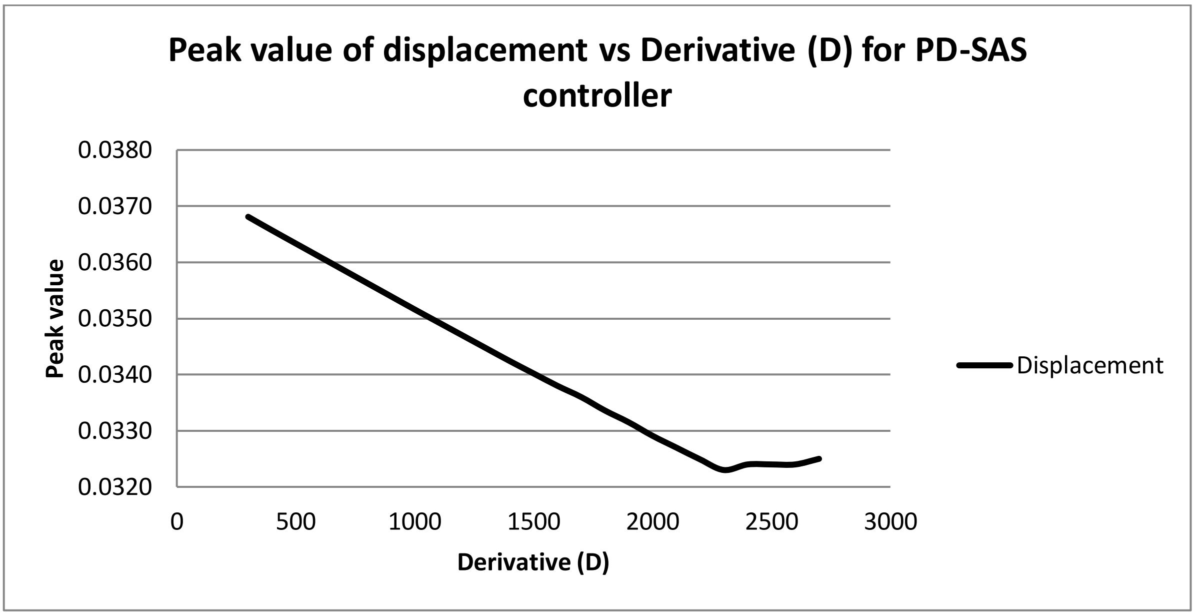

Figure 21 : Sensitivity analysis for Peak values of body displacement vs Derivative (D) for PD-SAS controller.

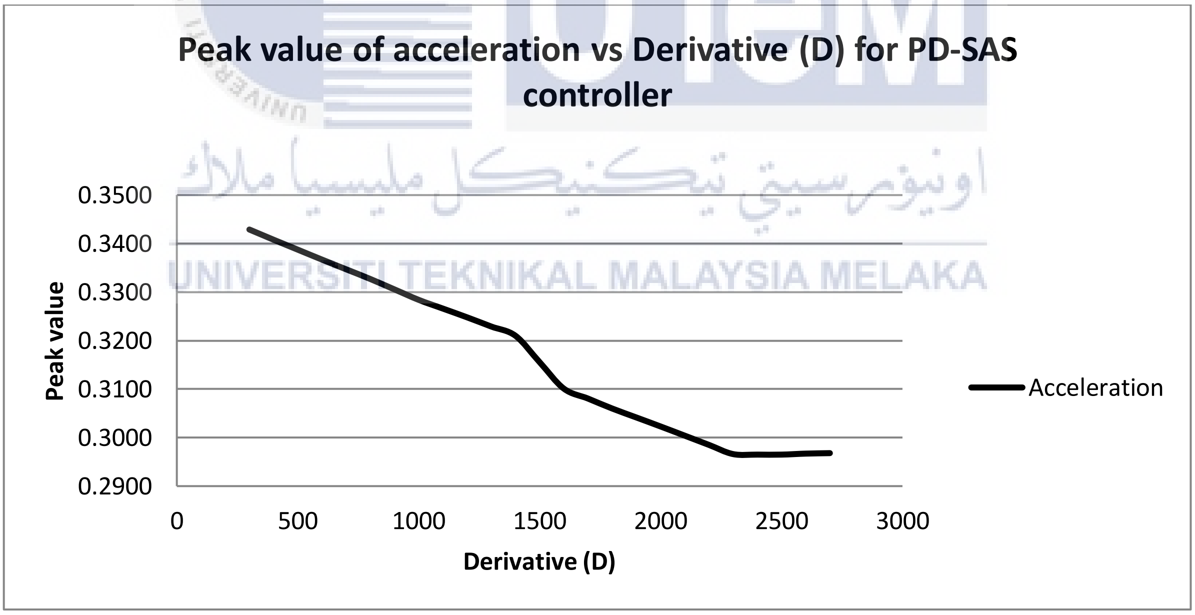

Figure 22 : Sensitivity analysis for Peak values of body acceleration vs Derivative (D) for PD-SAS controller.

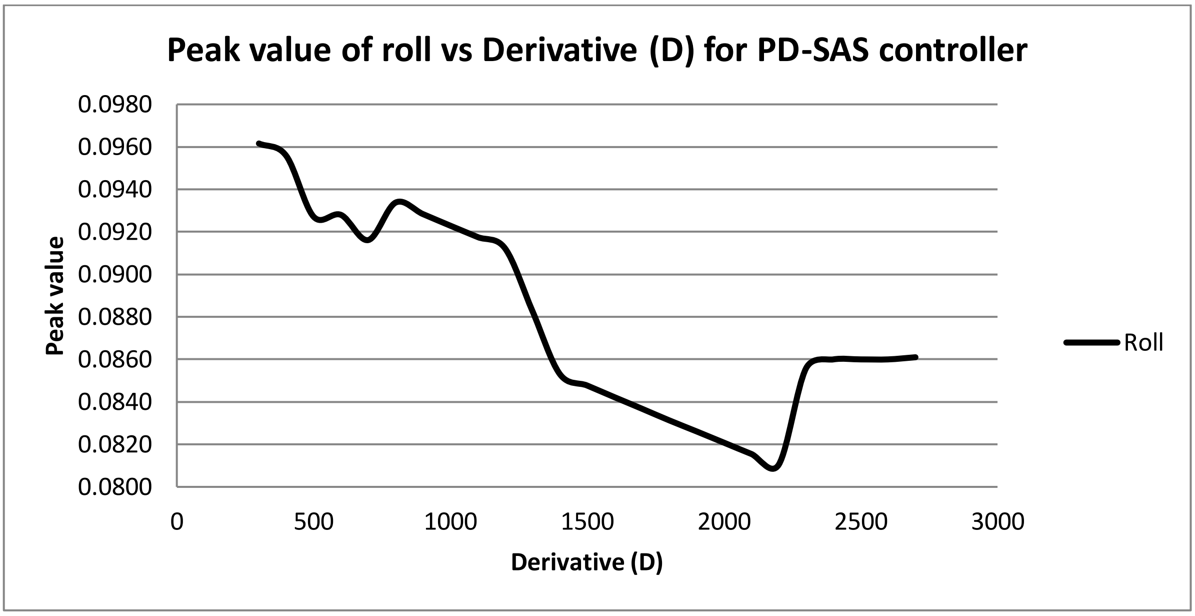

Figure 23 : Sensitivity analysis for Peak values of roll vs Derivative (D) for PD-SAS controller.

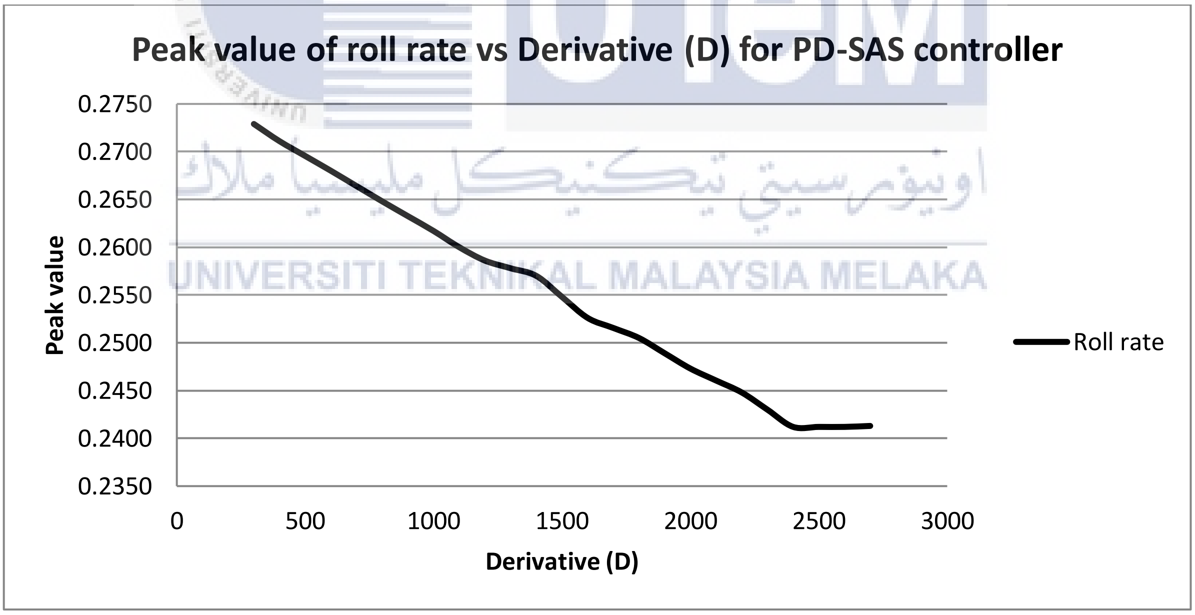

Figure 24 : Sensitivity analysis for Peak values of roll rate vs Derivative (D) for PD-SAS controller.

APPENDIX D

Sensitivity analysis Proportional (P) and Derivative (D) for PD-SAS controller using sine input of 7 rad/s

Table 7 : Sensitivity analysis value of Proportional (P) for PD-SAS controller using sine input of 7 rad/s.

No	Proportional (P)	Displacement	Acceleration	Roll	Roll rate
1	300	0.0228	1.1030	0.1404	0.9746
2	400	0.0228	1.1030	0.1404	0.9746
3	500	0.0228	1.1030	0.1404	0.9742
4	600	0.0227	1.1030	0.1403	0.9741
5	700	0.0227	1.1030	0.1403	0.9741
6	800	0.0227	1.1020	0.1402	0.9737
7	900	0.0227	1.1020	0.1402	0.9737
8	1000	0.0227	1.1019	0.1402	0.9737
9	1100	0.0226	1.1019	0.1402	0.9737
10	1200	0.0226	1.1018	0.1402	0.9737
11	1300	0.0226	1.1018	0.1402	0.9737
12	1400	0.0226	1.1017	0.1401	0.9736
13	1500	0.0225	1.1016	0.1401	0.9736
14	1600	0.0225	1.1015	0.1401	0.9736
15	1700	0.0225	1.1015	0.1400	0.9736
16	1800	0.0224	1.1014	0.1400	0.9735
17	1900	0.0224	1.1013	0.1399	0.9735
18	2000	0.0224	1.1012	0.1399	0.9735
19	2100	0.0224	1.1011	0.1398	0.9734
20	2200	0.0224	1.1010	0.1398	0.9734
21	2300	0.0224	1.1009	0.1398	0.9734
22	2400	0.0224	1.1008	0.1398	0.9733
23	2500	0.0224	1.1008	0.1398	0.9733

24	2600	0.0224	1.1007	0.1397	0.9733
25	2700	0.0223	1.1006	0.1396	0.9732
26	2800	0.0223	1.1005	0.1395	0.9732
27	2900	0.0223	1.1004	0.1394	0.9731
28	3000	0.0223	1.1003	0.1394	0.9731
29	3100	0.0223	1.1003	0.1393	0.9731
30	3200	0.0222	1.1002	0.1393	0.9730
31	3300	0.0222	1.1001	0.1392	0.9730
32	3400	0.0222	1.1000	0.1392	0.9730
33	3500	0.0222	1.0999	0.1391	0.9729
34	3600	0.0222	1.0998	0.1390	0.9729
35	3700	0.0221	1.0997	0.1389	0.9728
36	3800	0.0221	1.0996	0.1388	0.9728
37	3900	0.0221	1.0995	0.1387	0.9727
38	4000	0.0221	1.0994	0.1386	0.9727
39	4100	0.0221	1.0994	0.1385	0.9726
40	6000	0.0220	1.0994	0.1384	0.9625
41	6523	0.0220	1.0994	0.1384	0.9626
42	6600	0.0221	1.0995	0.1387	0.9627
43	6700	0.0222	1.0996	0.1387	0.9627

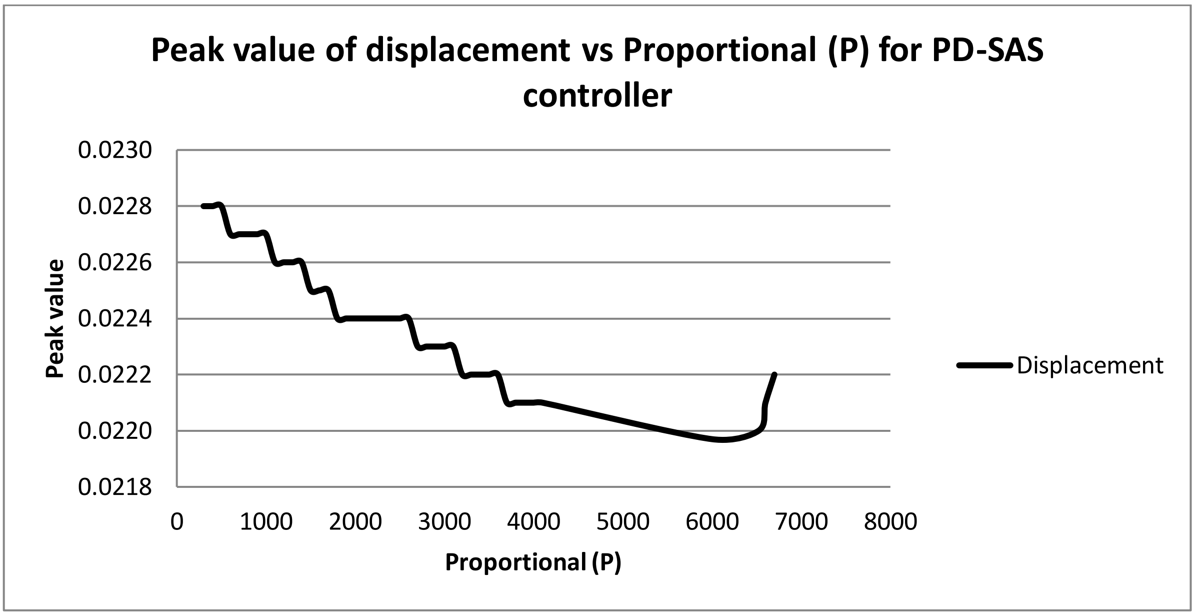

Figure 25 : Sensitivity analysis for Peak values of body displacement vs Proportional (P) for PD-SAS controller.

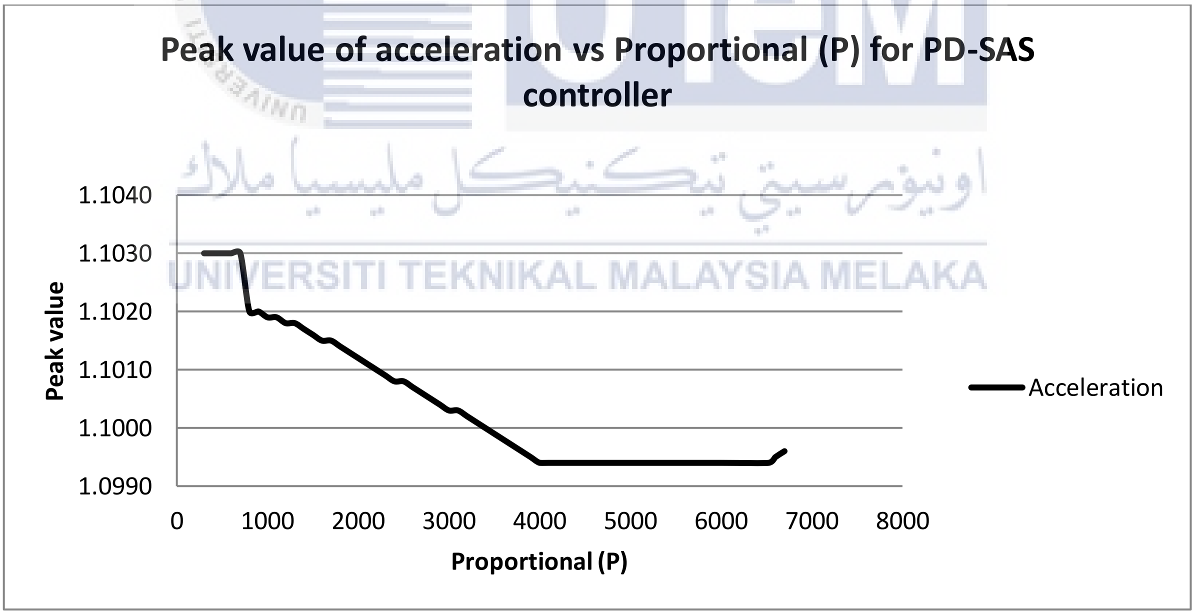

Figure 26 : Sensitivity analysis for Peak values of body acceleration vs Proportional (P) for PD-SAS controller.

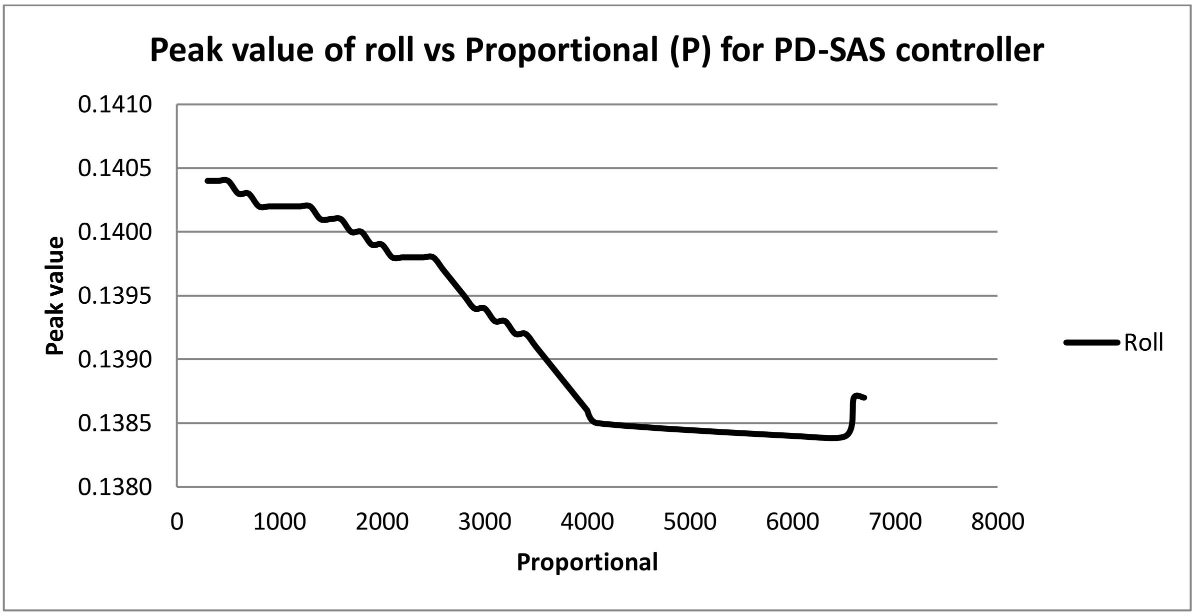

Figure 27 : Sensitivity analysis for Peak values of roll vs Proportional (P) for PD-SAS controller.

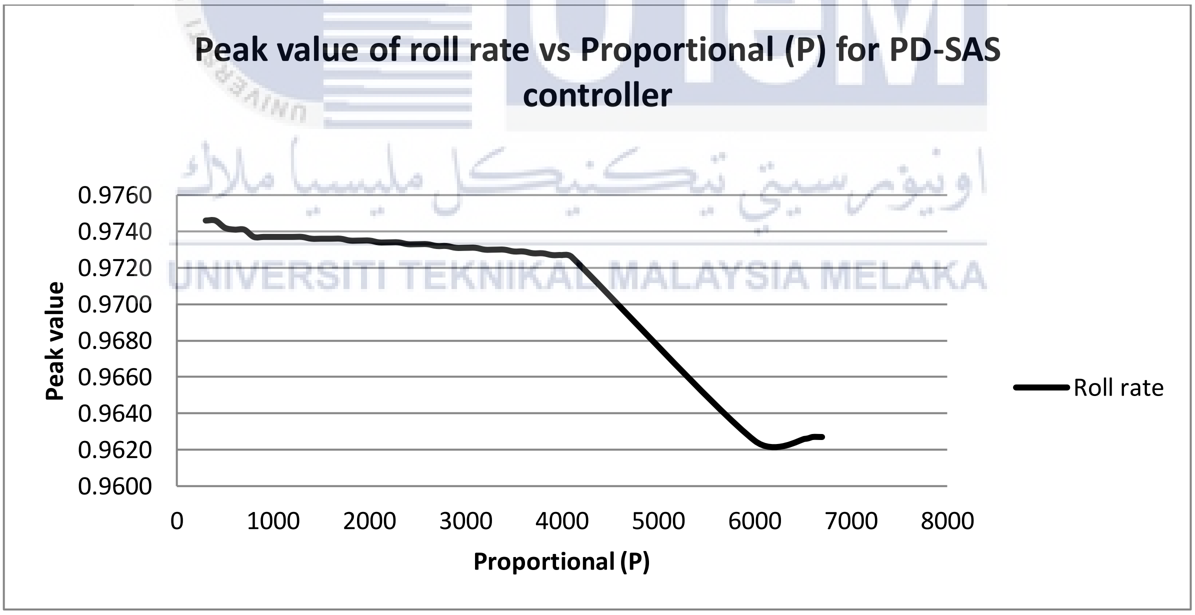

Figure 28 : Sensitivity analysis for Peak values of roll rate vs Proportional (P) for PD-SAS controller.

Table 8 : Sensitivity analysis value of Derivative (D) for PD-SAS controller using sine input of 7 rad/s.

No	Derivative (D)	Displacement	Aceeleration	Roll	Roll rate
1	300	0.0189	0.9134	0.1154	0.8069
2	400	0.0188	0.9132	0.1154	0.8057
3	500	0.0188	0.9127	0.1153	0.8050
4	600	0.0187	0.9108	0.1153	0.8034
5	700	0.0187	0.9100	0.1152	0.8021
6	800	0.0186	0.9095	0.1152	0.7999
7	900	0.0186	0.9089	0.1151	0.7990
8	1000	0.0186	0.9080	0.1151	0.7987
9	1100	0.0186	0.9076	0.1149	0.7986
10	1200	0.0185	0.9065	0.1146	0.7985
11	1300	0.0185	0.9054	0.1143	0.7984
12	1400	0.0184	0.9049	0.1140	0.7983
13	1500	0.0184	0.9045	0.1137	0.7981
14	1600	0.0184	0.9034	0.1136	0.7979
15	1700	0.0183	0.9031	0.1136	0.7976
16	1800	0.0182	0.9025	0.1135	0.7967
17	1900	0.0182	0.9018	0.1134	0.7956
18	2000	0.0182	0.9007	0.1133	0.7943
19	2100	0.0184	0.9013	0.1134	0.7950
20	2200	0.0185	0.9024	0.1135	0.7965

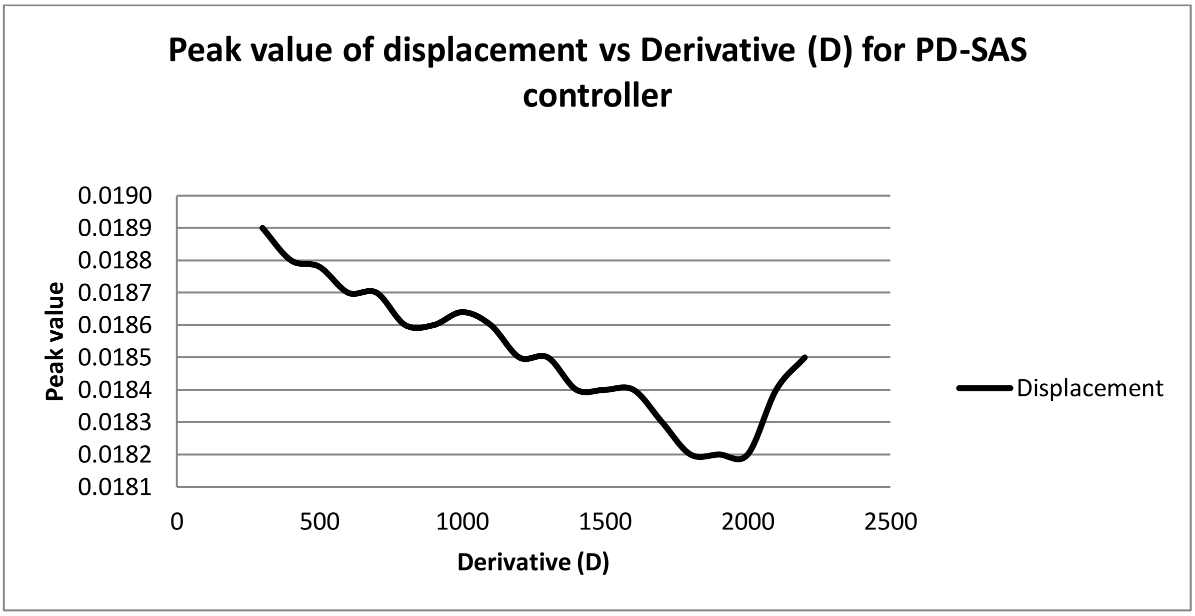

Figure 29 : Sensitivity analysis for Peak values of body displacement vs Derivative (D) for PD-SAS controller.

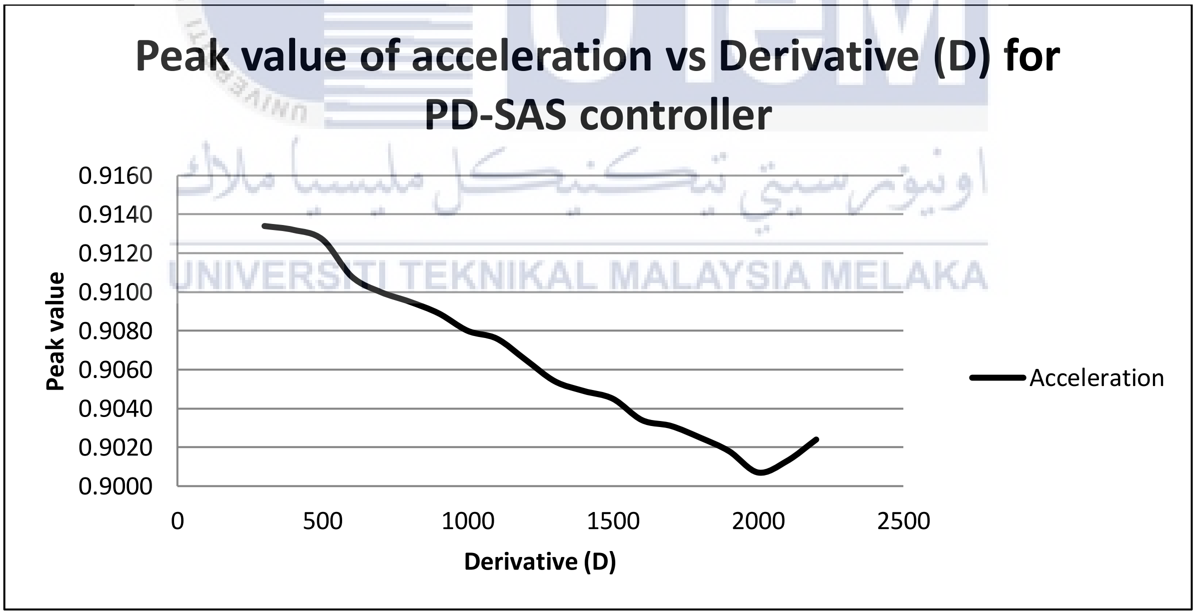

Figure 30 : Sensitivity analysis for Peak values of body acceleration vs Derivative (D) for PD-SAS controller.

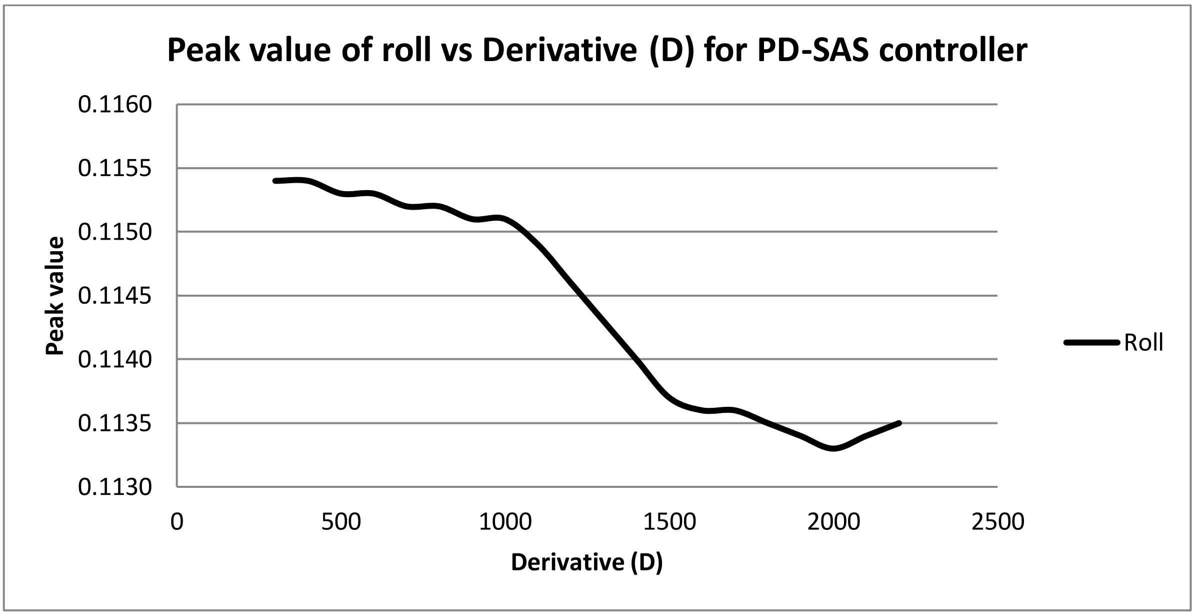

Figure 31 : Sensitivity analysis for Peak values of roll vs Derivative (D) for PD-SAS controller.

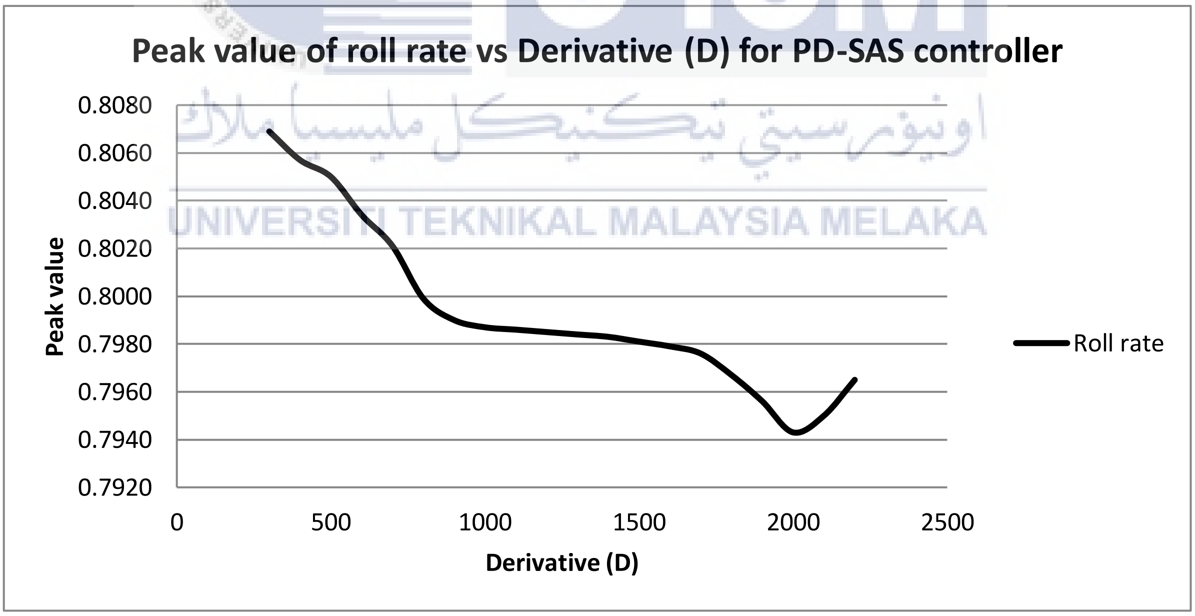

Figure 32 : Sensitivity analysis for Peak values of roll rate vs Derivative (D) for PD-SAS controller