Application of Stability Augmentation System for Semi-active Railway Vehicle Secondary Vertical Suspension

M. Hanif Harun, H. Jamaluddin, K. Hudha, R. Abd Rahman, W. Mohd Zailimi

***Abstract*—**This paper focuses on a semi-active suspension control applied to the secondary vertical suspension system of railway vehicle. The dynamics of nine degrees-of-freedom (9-DOF) railway vehicle model are governed which includes a vehicle body, two bogies and four wheel-set. The disturbance considered is track irregularity which is modeled as a sine wave. The control algorithm for the semi-active suspension system is developed based on PID controller and Stability Augmentation System (SAS) which is consists of inner and outer loops controller to reduce the effect of track disturbance and stabilize the pitch and roll response. The performances of passive and semi-active suspension are compared by simulation using MATLAB-SIMULINK software. The results of the study show that the PID and SAS controllers are able to significantly improve the ride quality of railway vehicle body. It can be noted that the an inner loop control (ride control loop) is able to further improve the performance of SAS controller for the system.

***Keywords*—**railway suspension, stability augmentation system, magneto-rheological damper, vertical response.

I. INTRODUCTION

RAILWAY vehicle has become one of the important public transport systems in recent years due to economical transportation as well as minimizing air pollution. Under competition from other modes of transport, the train passenger has come to expect shorter journey times with better ride comfort. Due to this, development of high-speed railway vehicle has become a great interest in many countries because it has been proven as an efficient transportation system. With increasing the speed of the vehicle on a railway vehicle track, the vehicle tends to be unstable due to the increased magnitude motions in various directions such as vertical and lateral. These motions will cause the vehicle passenger to be uncomfortable. In order to deal with this issue, several researches, developers, and engineers from academia and industry have been studied on this matter successfully [1-8], and they reveals that passive suspension system has reached its limit when speed is increasing. Therefore, semi-active suspension technology in railway vehicle can be considered as an alternative solution for this issue, since it offers better possibilities of improving vehicle's dynamic performance compared to the conventional passive solution [9].

Early work on the semi-active suspension systems has been achieved in the early of 1990s by a group of researchers from ALSTHOM. This study was carried out by O'Neill and Wale [10] on a secondary suspension system of railway vehicle fitted with controllable dampers and the results showing an improvement in ride comfort of 25%. Recent effort on improving the dynamic performance of railway vehicle has been successfully done by Shin, et al. [4]. They was performed a series of simulations and tests on a roller rig with application of MR damper on a secondary suspension system. The development of semi-active secondary vertical suspension of railway vehicle system also has been successfully done through simulation study by Liao and Wang [11] which focusing on improving the ride quality of railway vehicle. The results of the study showed the proposed control was feasible and effective in suppressing the vibration of railway vehicle body.

In recent years, advanced control technology has a major impact to the railway vehicle dynamic development since 1975. The vehicle dynamicists have been aware with the use of actuators, sensors and electronic controllers in vehicle suspension. The general benefits can be achieved is better ride quality and running stability of railway vehicle. In this study, a semi-active hybrid skyhook-stability augmentation (Skyhook-SAS) suspension system is proposed for reducing unwanted railway vehicle body motion in lateral direction. Much successful theoretical work on semi-active[6, 12-14] and active [15-17] control for body vibration of railway vehicle has been carried out in China [12, 13] and Japan [16]. But in contrast, there are many challenges must be taken into account especially in research and development of semi-active

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suspension system. The practical application has been limited because of the existence of system error, random error and also an external disturbance in all tests and control process [12]. Particularly in the area of heavy vehicle, little research has been done [18, 19], and there is no commercially available controllable damper suitable for the railway vehicle. In order to solve this problem, a small scaled model of railway vehicle with semi-active secondary suspension system is developed to carry through the experiment.

Generally semi-active control on the railway vehicle suspension can be applied to both vertical and lateral secondary suspension. This research concentrates on the vertical secondary suspension concept which concerning for the ride comfort improvement. The idea is to replace all of the vertical passive dampers on the secondary suspension with a dampers characterized by an electronically controlled damping ratio. The controller namely stability augmentation system is applied to the system and sensitivity analysis method is used to find an optimum value of the controller parameters.

This paper is organized as follows: The first section presents the introduction and the review of some relevant preliminary works, followed by 9-DOF mathematical model of railway vehicle and its assumption in the sub-second section. The third section describe the mathematical model of MR damper and followed by control structure of semi-active suspension system. The fourth section presents the inner loop and outer loop controllers of semi-active suspension system and the sensitivity study in tuning controller parameters is described in section five. Further, the results of railway vehicle dynamic performance in terms of reducing the vehicle body displacement and acceleration in real time are presented in the last section.

II. Full Railway Vehicle Model

Railway vehicle dynamic model considered in this study consists of vehicle body connected to four wheel-sets via two bogie masses and were represented as a 9-DOF. The vehicle body mass is allow to roll and pitch as well as to displace in vertical direction, whereas the bogie masses are allowed to bounce vertically with respect to the vehicle body and also allowed pitch and roll to rotate along its axis.

A. Assumption

Some of the assumptions considered in this study are as follows:

i) The vehicle body, bogies and wheel-sets are considered as a rigid body, and aerodynamic drag force also is ignored.
ii) The passive suspension components of primary and secondary suspension are modeled with viscous dampers and spring elements in vertical, lateral and longitudinal directions.
iii) Rolling resistance due to the anti roll bar and body flexibility is also neglected.
iv) The wheel-sets move along the straight track at constant velocity, and track with vertical profile is regarded as an external excitation to the railway vehicle system.
v) The model developed in this section only perform only in vertical direction.

B. Equations of motion

The equations of motion of railway vehicle was derived based on the Second Newton's Law. Figure 1 illustrates the location of four semi-active dampers, which are used to control the railway vehicle body vibration and unwanted body motions, those were vertically placed on the left and right sides of each bogie. The primary and secondary suspension forces at each axis of the railway vehicle are defined as the sum of the forces produced by the suspension components.

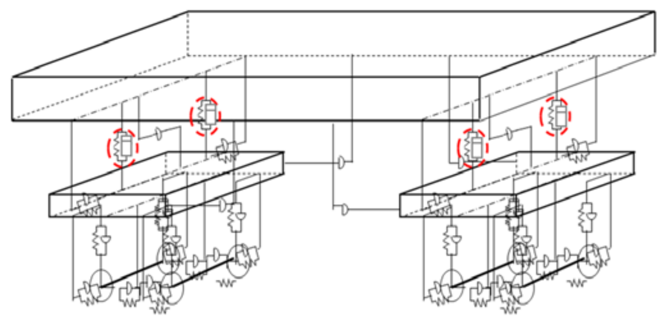

Fig. 1 Schematic diagram of semi-active damper placement on railway vehicle secondary suspension

By referring to the Figure 1, the force balance on railway vehicle body in vertical direction, body pitch and roll motions are given as:

$$
\begin{aligned}
& m_v \ddot{z}_v - 2\left(-k_{sz}[z_v - l\varphi_v - z_{b1} + d_s(\theta_v - \theta_{b1})] - c_{sz}[\dot{z}_v - l\dot{\varphi}_v - \dot{z}_{b1} + d_s(\dot{\theta}_v - \dot{\theta}_{b1})]\right) \\
& - 2\left(-k_{sz}[z_v - l\varphi_v - z_{b1} + d_s(\theta_v - \theta_{b1})] - c_{sz}[\dot{z}_v - l\dot{\varphi}_v - \dot{z}_{b1} + d_s(\dot{\theta}_v - \dot{\theta}_{b1})]\right) \\
& - f_{1R} - f_{1L} - f_{2R} - f_{2L} = 0
\end{aligned} \tag{1}
$$

$$
\begin{aligned}
& I_{vy}\ddot{\varphi}_v - 2\left(-k_{sx}(h_{cs}\varphi_c + h_{ts}\varphi_{b1}) - c_{sx}(h_{cs}\dot{\varphi}_c + h_{ts}\dot{\varphi}_{b1})\right)h_{cs} \\
& - 2\left(-k_{sx}(h_{cs}\varphi_c + h_{ts}\varphi_{b2}) - c_{sx}(h_{cs}\dot{\varphi}_c + h_{ts}\dot{\varphi}_{b2})\right)h_{cs} \\
& + 2\left(-k_{sz}[z_c - l\varphi_c - z_{b1} + d_s(\theta_c - \theta_{b1})] - c_{sz}[\dot{z}_c - l\dot{\varphi}_c - \dot{z}_{b1} + d_s(\dot{\theta}_c - \dot{\theta}_{b1})]\right) \\
& + 2\left(-k_{sz}[z_c - l\varphi_c - z_{b2} + d_s(\theta_c - \theta_{b2})] - c_{sz}[\dot{z}_c - l\dot{\varphi}_c - \dot{z}_{b2} + d_s(\dot{\theta}_c - \dot{\theta}_{b2})]\right) \\
& + f_{1R} + f_{1L} - f_{2R} - f_{2L} = 0
\end{aligned} \tag{2}
$$

$$
\begin{aligned}
& I_{vx}\ddot{\theta}_v + 2\left(k_{sy}(h_{cs}\theta_c + h_{ts}\theta_{b1}) + k_{sy}(h_{cs}\dot{\theta}_c + h_{ts}\dot{\theta}_{b1})\right) h_{cs} \\
& + 2\left(-k_{sx}(h_{cs}\varphi_c + h_{ts}\varphi_{b2}) - c_{sx}(h_{cs}\dot{\varphi}_c + h_{ts}\dot{\varphi}_{b2})\right) h_{cs} \\
& - \left(k_{sz}[z_c - l\varphi_c - z_{b1} + d_s(\theta_c - \theta_{b1})] + c_{sz}[\dot{z}_c - l\dot{\varphi}_c - \dot{z}_{b1} + d_s(\dot{\theta}_c - \dot{\theta}_{b1})]\right)d_s \\
& - \left(k_{sz}[z_c - l\varphi_c - z_{b2} + d_s(\theta_c - \theta_{b2})] + c_{sz}[\dot{z}_c - l\dot{\varphi}_c - \dot{z}_{b2} + d_s(\dot{\theta}_c - \dot{\theta}_{b2})]\right)d_s \\
& + \left(k_{sz}[z_c - l\varphi_c - z_{b1} + d_s(\theta_c - \theta_{b1})] + c_{sz}[\dot{z}_c - l\dot{\varphi}_c - \dot{z}_{b1} + d_s(\dot{\theta}_c - \dot{\theta}_{b1})]d_s\right) \\
& + \left(k_{sz}[z_c - l\varphi_c - z_{b2} + d_s(\theta_c - \theta_{b2})] + c_{sz}[\dot{z}_c - l\dot{\varphi}_c - \dot{z}_{b2} + d_s(\dot{\theta}_c - \dot{\theta}_{b2})]d_s\right) \\
& - \left(f_{1R} + f_{2R}\right)d_s + \left(f_{1L} + f_{2L}\right)d_s = 0
\end{aligned} \tag{3}
$$

By performing the balance analysis, the governing equations for the front bogie can be expressed as follow:

$$
\begin{aligned}
& m_{b1}\ddot{z}_{b1} + 2\left(-k_{sz}[z_c - l\varphi_c - z_{b1} + d_s(\theta_c - \theta_{b1})] - c_{sz}[\dot{z}_c - l\dot{\varphi}_c - \dot{z}_{b1} + d_s(\dot{\theta}_c - \dot{\theta}_{b1})]\right) \\
& - k_{pz}\left(z_{b1} - b\varphi_{b1} + d_p\theta_{b1} - \frac{d_p}{a} z_{1r}\right) - c_{pz}\left(\dot{z}_{b1} - b\dot{\varphi}_{b1} + d_p\dot{\theta}_{b1} - \frac{d_p}{a}\dot{z}_{1r}\right) \\
& - k_{pz}\left(z_{b1} + b\varphi_{b1} + d_p\theta_{b1} - \frac{d_p}{a} z_{2r}\right) - c_{pz}\left(\dot{z}_{b1} + b\dot{\varphi}_{b1} + d_p\dot{\theta}_{b1} - \frac{d_p}{a}\dot{z}_{2r}\right) \\
& - k_{pz}\left(z_{b1} - b\varphi_{b1} - d_p\theta_{b1} - \frac{d_p}{a} z_{1l}\right) - c_{pz}\left(\dot{z}_{b1} - b\dot{\varphi}_{b1} + d_p\dot{\theta}_{b1} - \frac{d_p}{a}\dot{z}_{1l}\right) \\
& - k_{pz}\left(z_{b1} + b\varphi_{b1} - d_p\theta_{b1} - \frac{d_p}{a} z_{2l}\right) - c_{pz}\left(\dot{z}_{b1} + b\dot{\varphi}_{b1} - d_p\dot{\theta}_{b1} - \frac{d_p}{a}\dot{z}_{2l}\right) \\
& + f_{1L} + f_{1R} = 0
\end{aligned} \tag{4}
$$

Similarly, moment balance equations are derived for front bogie pitch and roll are given as:

$$
\begin{aligned}
& I_{b1y}\ddot{\varphi}_{b1} - 2\left(k_{sy}(h_{cs}\theta_c + h_{ts}\theta_{b1}) - k_{sy}(h_{cs}\dot{\theta}_c + h_{ts}\dot{\theta}_{b1})\right) h_{ts} \\
& - \left(k_{pz}\left(z_{b1} - b\varphi_{b1} + d_p\theta_{b1} - \frac{d_p}{a} z_{1r}\right) + c_{pz}\left(\dot{z}_{b1} - b\dot{\varphi}_{b1} + d_p\dot{\theta}_{b1} - \frac{d_p}{a}\dot{z}_{1r}\right)\right) b \\
& - \left(k_{pz}\left(z_{b1} - b\varphi_{b1} - d_p\theta_{b1} - \frac{d_p}{a} z_{1l}\right) + c_{pz}\left(\dot{z}_{b1} - b\dot{\varphi}_{b1} + d_p\dot{\theta}_{b1} - \frac{d_p}{a}\dot{z}_{1l}\right)\right) b \\
& - \left(-k_{pz}\left(z_{b1} + b\varphi_{b1} + d_p\theta_{b1} - \frac{d_p}{a} z_{2r}\right) - c_{pz}\left(\dot{z}_{b1} + b\dot{\varphi}_{b1} + d_p\dot{\theta}_{b1} - \frac{d_p}{a}\dot{z}_{2r}\right)\right) b \\
& - \left(-k_{pz}\left(z_{b1} + b\varphi_{b1} - d_p\theta_{b1} - \frac{d_p}{a} z_{2l}\right) - c_{pz}\left(\dot{z}_{b1} + b\dot{\varphi}_{b1} - d_p\dot{\theta}_{b1} - \frac{d_p}{a}\dot{z}_{2l} b\right)\right) \\
& - 2\left(-k_{px}h_{tp}\varphi_{b1} - c_{px}h_{tp}\dot{\varphi}_{b1}\right) h_{tp} \\
& - 2\left(-k_{px}h_{tp}\varphi_{b1} - c_{px}h_{tp}\dot{\varphi}_{b1}\right) h_{tp} = 0
\end{aligned} \tag{5}
$$

$$
\begin{aligned}
& I_{b1x}\ddot{\theta}_{b1} + 2\left(k_{sy}(h_{cs}\theta_c + h_{ts}\theta_{b1}) + k_{sy}(h_{cs}\dot{\theta}_c + h_{ts}\dot{\theta}_{b1})\right) h_{ts} \\
& + \left(k_{sz}[z_c - l\varphi_c - z_{b1} - d_s(\theta_c - \theta_{b1})] - c_{sz}[\dot{z}_c - l\dot{\varphi}_c - \dot{z}_{b1} - d_s(\dot{\theta}_c - \dot{\theta}_{b1})]\right) b \\
& + \left(-k_{pz}\left(z_{b1} - b\varphi_{b1} + d_p\theta_{b1} - \frac{d_p}{a} z_{1r}\right) - c_{pz}\left(\dot{z}_{b1} - b\dot{\varphi}_{b1} + d_p\dot{\theta}_{b1} - \frac{d_p}{a}\dot{z}_{1r}\right)\right) d_p \\
& - \left(-k_{pz}\left(z_{b1} + b\varphi_{b1} + d_p\theta_{b1} - \frac{d_p}{a} z_{2r}\right) - c_{pz}\left(\dot{z}_{b1} + b\dot{\varphi}_{b1} + d_p\dot{\theta}_{b1} - \frac{d_p}{a}\dot{z}_{2r}\right)\right) d_p \\
& + \left(-k_{pz}\left(z_{b1} - b\varphi_{b1} - d_p\theta_{b1} - \frac{d_p}{a} z_{1l}\right) - c_{pz}\left(\dot{z}_{b1} - b\dot{\varphi}_{b1} + d_p\dot{\theta}_{b1} - \frac{d_p}{a}\dot{z}_{1l}\right)\right) d_p \\
& + \left(-k_{pz}\left(z_{b1} + b\varphi_{b1} - d_p\theta_{b1} - \frac{d_p}{a} z_{2l}\right) - c_{pz}\left(\dot{z}_{b1} + b\dot{\varphi}_{b1} - d_p\dot{\theta}_{b1} - \frac{d_p}{a}\dot{z}_{2l}\right)\right) d_p \\
& - \left(k_{sz}[z_c - l\varphi_c - z_{b1} + d_s(\theta_c - \theta_{b1})] + c_{sz}[\dot{z}_c - l\dot{\varphi}_c - \dot{z}_{b1} + d_s(\dot{\theta}_c - \dot{\theta}_{b1})]\right) b \\
& - 4\left(k_{px}h_{tp}\varphi_{b1} + c_{px}h_{tp}\dot{\varphi}_{b1}\right) h_{tp} + (f_{1R} - f_{1L}) d_s = 0
\end{aligned} \tag{6}
$$

Similarly, for the rear bogie all of the three equations of motion can be expressed as follow:

$$
\begin{aligned}
& m_{b2}\ddot{z}_{b2} + 2\left(-k_{sz}[z_c - l\varphi_c - z_{b2} + d_s(\theta_c - \theta_{b2})] - c_{sz}[\dot{z}_c - l\dot{\varphi}_c - \dot{z}_{b2} + d_s(\dot{\theta}_c - \dot{\theta}_{b2})]\right) \\
& - \left(-k_{pz}\left(z_{b2} - b\varphi_{b2} + d_p\theta_{b2} - \frac{d_p}{a} z_{3r}\right) - c_{pz}\left(\dot{z}_{b2} - b\dot{\varphi}_{b2} + d_p\dot{\theta}_{b2} - \frac{d_p}{a}\dot{z}_{3r}\right)\right) \\
& - \left(-k_{pz}\left(z_{b2} + b\varphi_{b2} + d_p\theta_{b2} - \frac{d_p}{a} z_{4r}\right) - c_{pz}\left(\dot{z}_{b2} + b\dot{\varphi}_{b2} + d_p\dot{\theta}_{b2} - \frac{d_p}{a}\dot{z}_{4r}\right)\right) \\
& - \left(-k_{pz}\left(z_{b2} - b\varphi_{b2} - d_p\theta_{b2} - \frac{d_p}{a} z_{3l}\right) - c_{pz}\left(\dot{z}_{b2} - b\dot{\varphi}_{b2} - d_p\dot{\theta}_{b2} - \frac{d_p}{a}\dot{z}_{3l}\right)\right) \\
& - \left(-k_{pz}\left(z_{b2} + b\varphi_{b2} - d_p\theta_{b2} - \frac{d_p}{a} z_{4l}\right) - c_{pz}\left(\dot{z}_{b2} + b\dot{\varphi}_{b2} - d_p\dot{\theta}_{b2} - \frac{d_p}{a}\dot{z}_{4l}\right)\right) \\
& + (f_{2R} + f_{2L}) = 0
\end{aligned} \tag{7}
$$

$$
\begin{aligned}
& I_{b2y}\ddot{\varphi}_{b2} + 2\left(-k_{sz}[z_c - l\varphi_c - z_{b2} + d_s(\theta_c - \theta_{b2})] - c_{sz}[\dot{z}_c - l\dot{\varphi}_c - \dot{z}_{b2} + d_s(\dot{\theta}_c - \dot{\theta}_{b2})]\right) h_{ts} \\
& - 2\left(-k_{px}h_{tp}\varphi_{b2} - c_{px}h_{tp}\dot{\varphi}_{b2}\right) h_{tp} \; - 2\left(-k_{px}h_{tp}\varphi_{b2} - c_{px}h_{tp}\dot{\varphi}_{b2}\right) \\
& - \left(k_{pz}\left(z_{b2} - b\varphi_{b2} + d_p\theta_{b2} - \frac{d_p}{a} z_{3r}\right) + c_{pz}\left(\dot{z}_{b2} - b\dot{\varphi}_{b2} + d_p\dot{\theta}_{b2} - \frac{d_p}{a}\dot{z}_{3r}\right)\right) b \\
& - \left(k_{pz}\left(z_{b2} - b\varphi_{b2} + d_p\theta_{b2} - \frac{d_p}{a} z_{3l}\right) + c_{pz}\left(\dot{z}_{b2} - b\dot{\varphi}_{b2} + d_p\dot{\theta}_{b2} - \frac{d_p}{a}\dot{z}_{3l}\right)\right) b \\
& + \left(k_{pz}\left(z_{b2} + b\varphi_{b2} + d_p\theta_{b2} - \frac{d_p}{a} z_{4r}\right) + c_{pz}\left(\dot{z}_{b2} + b\dot{\varphi}_{b2} + d_p\dot{\theta}_{b2} - \frac{d_p}{a}\dot{z}_{4r}\right)\right) b \\
& + \left(k_{pz}\left(z_{b2} + b\varphi_{b2} + d_p\theta_{b2} - \frac{d_p}{a} z_{4l}\right) + c_{pz}\left(\dot{z}_{b2} + b\dot{\varphi}_{b2} + d_p\dot{\theta}_{b2} - \frac{d_p}{a}\dot{z}_{4l}\right)\right) b = 0
\end{aligned} \tag{8}
$$

$$
\begin{aligned}
& I_{b2x}\ddot{\theta}_{b2} + 2\left(k_{sy}(h_{cs}\theta_c + h_{ts}\theta_{b2}) + k_{sy}(h_{cs}\dot{\theta}_c + h_{ts}\dot{\theta}_{b2})\right) h_{ts} \\
& + 4\left(k_{py}h_{tp}\theta_{b2} + c_{py}h_{tp}\dot{\theta}_{b2}\right) h_{tp} \\
& + \left(k_{sz}[z_c + l\varphi_c - z_{b2} + d_s(\theta_c - \theta_{b2})] + c_{sz}[\dot{z}_c + l\dot{\varphi}_c - \dot{z}_{b2} + d_s(\dot{\theta}_c - \dot{\theta}_{b2})]\right) d_s \\
& + \left(k_{sz}[z_c - l\varphi_c - z_{b2} + d_s(\theta_c - \theta_{b2})] + c_{sz}[\dot{z}_c - l\dot{\varphi}_c - \dot{z}_{b2} + d_s(\dot{\theta}_c - \dot{\theta}_{b2})]\right) d_s \\
& - \left(-k_{pz}\left(z_{b2} - b\varphi_{b2} + d_p\theta_{b2} - \frac{d_p}{a} z_{3r}\right) - c_{pz}\left(\dot{z}_{b2} - b\dot{\varphi}_{b2} + d_p\dot{\theta}_{b2} - \frac{d_p}{a}\dot{z}_{3r}\right)\right) d_p \\
& - \left(-k_{pz}\left(z_{b2} + b\varphi_{b2} + d_p\theta_{b2} - \frac{d_p}{a} z_{4r}\right) - c_{pz}\left(\dot{z}_{b2} + b\dot{\varphi}_{b2} + d_p\dot{\theta}_{b2} - \frac{d_p}{a}\dot{z}_{4r}\right)\right) d_p \\
& - \left(k_{pz}\left(z_{b2} - b\varphi_{b2} - d_p\theta_{b2} - \frac{d_p}{a} z_{3l}\right) + c_{pz}\left(\dot{z}_{b2} - b\dot{\varphi}_{b2} - d_p\dot{\theta}_{b2} - \frac{d_p}{a}\dot{z}_{3l}\right)\right) d_p \\
& + \left(k_{pz}\left(z_{b2} + b\varphi_{b2} - d_p\theta_{b2} - \frac{d_p}{a} z_{4l}\right) + c_{pz}\left(\dot{z}_{b2} + b\dot{\varphi}_{b2} - d_p\dot{\theta}_{b2} - \frac{d_p}{a}\dot{z}_{4l}\right)\right) d_p \\
& + (f_{2R} - f_{2L}) d_s = 0
\end{aligned} \tag{9}
$$

III. MAGNETO-RHEOLOGICAL DAMPER MODEL

The magneto-rheological damper (MR) model proposed in this study is based on Bouc-Wen model in order to characterize the behavior of MR damper. This method was proposed by [20] and was used by [1, 6, 21]. Mechanical analogue of the Bouc-Wen model is shown in Fig. 2

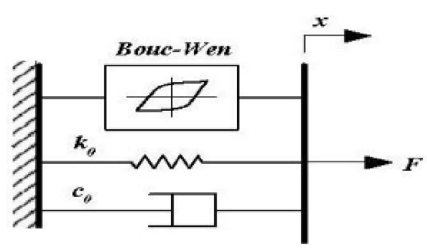

Fig. 2 Mechanical analogue of Bouc-Wen Model [6].

The force generated by the Bouc-Wen model according to the mechanical analogue is

$$F = c_0\dot{x} + k_0(x - x_0) + \alpha z \tag{10}$$

where;

$$\dot{z} = -\gamma \,|\, x \,|\, z \,|\, z \,|^{n-1} - \beta.\dot{x} \,|\, z \,|^{n} + \delta.\dot{x} \tag{11}$$

IV. Control Structure of Semi-Active Suspension System

The control structure of semi-active secondary vertical suspension system of railway vehicle using SAS and PID controllers are shown in Fig. 3 and Fig.4 respectively. The structure of semi-active control system can be described as a closed loop system which is the responses from the railway vehicle are fed back as a loop.

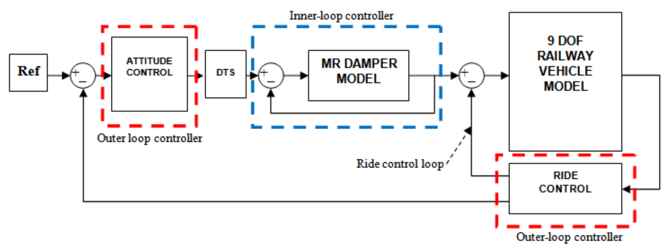

Fig. 3 Control structure of semi-active suspension system based on SAS controller

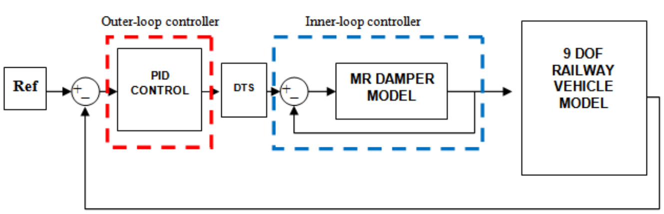

Fig. 4 Control structure of semi-active suspension system based on PID controller

Generally, the structure is consists of railway vehicle model, MR damper model with inner loop controller, the system controller which represent as an outer loop controller and decoupling transformation system that placed between inner and outer loops controller.

A. Inner loop controller

An inner loop controller is also known as an MR damper controller is important to ensure the MR damper is able to track the desired force produced by the system due to track irregularity. In this study, the structure of inner loop controller can be seen in Fig.5 and the algorithm of the proposed MR damper control can be stated as:

If $G(F_d - BF_{MR})\text{sgn}(F_{MR}) > V_{\max}$ then $v = V_{\max}$ (12)

Else If $G(F_d - BF_{MR})\text{sgn}(F_{MR}) < V_{\min}$ then $v = V_{\min}$ (13)

Else $v = G(F_d - BF_{MR})\text{sgn}(F_{MR})$ (14)

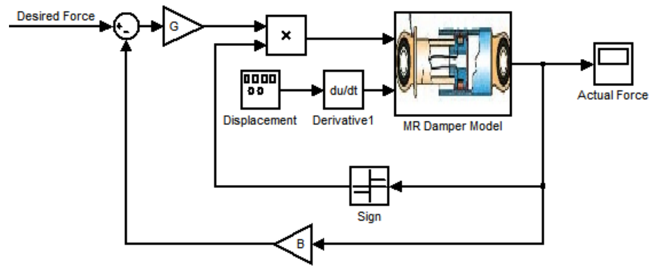

Fig. 5 Inner loop control structure of MR damper

An inner loop controller is also known as force tracking control of MR damper model is performed as a continuous state control which has two controller parameters known as *B* and *G*. The parameter of B is used to fed back actual damping force of MR damper in order to compare with the desired force before an error of the system is scaled by the parameter of *G*. This control strategy was used by [6, 22-24] and it was proved that has an ability to ensure an MR damper tracks the desired force from the system. Fig. 6 shows the force tracking control of MR damper model using sine-wave and saw tooth function using Matlab-SIMULINK software.

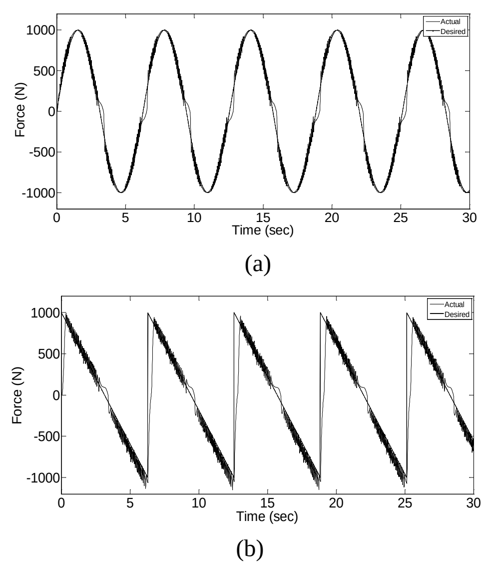

Fig. 6 Force tracking control result of MR damper; (a) Sine-wave (b) Saw-tooth.

B. Outer loop controller

The controller structure of semi-active suspension system for railway vehicle is given in Figure 3 and 4. Figure 3 is illustrates the SAS controller while Figure 4 shows PID control strategy. The controller structure of SAS is adopted from Hudha, et al. [25] and Samin, et al. [26], known as Stability Augmentation System (SAS) and consists of inner and outer loops controller. However, this type of controller is still not been applied to the railway vehicle suspension system. An unwanted weight transfer is rejected by the inner loop controller while the outer loop controller to stabilize pitch and roll response due to the effect of track disturbance.

In Fig.7, the relationship between the decoupling transformation, inner loop controller, outer loop controller with the 9-DOF railway vehicle model are clearly described.

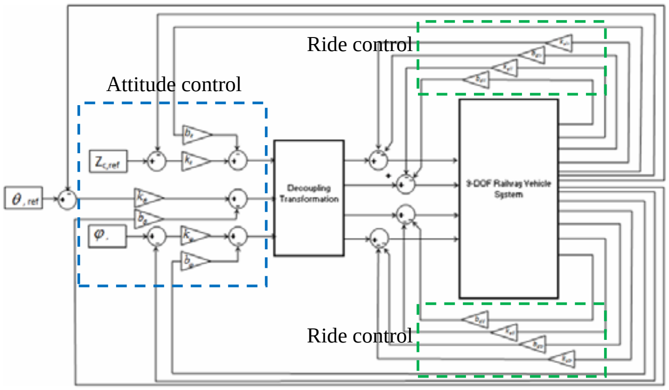

Fig. 7 Outer loop control structure of stability augmentation system

C. Decoupling transformation system (DTS)

The most attractive method of controlling vehicle body response of railway vehicle is by the use of secondary semi-active damper. In this study, the rail model is consists of a vehicle body that connected with two bogies via secondary suspension system elements such as lateral, vertical and longitudinal springs and dampers. In this case, we have four vertical passive dampers that have been replaced with four MR dampers. Distribution of the force and moments into target forces of four semi-active dampers is performed using decoupling transformation subsystem. The target forces of four semi-active dampers are then subtracted with the relevant outputs of the inner loop controller to produce ideal target forces of the four semi-active dampers. Decoupling transformation subsystem requires an understanding of the system dynamics in the previous section

The decoupling transformation is placed between outer loop and inner loop controller. The inner loop controller provides the weight transfer rejection and the outer loop controller provides the ride control. The outputs of the outer loop controller are vertical forces to stabilize body bounce (F_z), moment to stabilize body pitch (I_φ) and moment to stabilize roll (I_θ). Those forces and moments are then distributed into the target forces of four semi-active damper produced by outer loop controller. The equivalent forces for heave, pitch and roll can be defined by;

$$F_z = f_{1R} + f_{1L} + f_{2R} + f_{2L} \tag{15}$$

$$I_\theta = -(f_{1R} + f_{2R})d_s + (f_{1L} + f_{2L})d_s \tag{16}$$

$$I_\varphi = (f_{1R} + f_{1L})l - (f_{2R} + f_{2L})l \tag{17}$$

Equation (15), (16) and (17) can be rearranged in matrix format as the following

$$\begin{bmatrix} F_z(t) \\ I_\theta(t) \\ I_\varphi(t) \end{bmatrix} = \begin{bmatrix} 1 & 1 & 1 & 1 \\ -d_s & -d_s & d_s & d_s \\ l & -l & l & -l \end{bmatrix} \begin{bmatrix} f_{1R} \\ f_{1L} \\ f_{2R} \\ f_{2L} \end{bmatrix} \tag{18}$$

For a linear system of equations ***y=cx,*** if $C \in \Re^{mxn}$ has full row rank, then three exist a right inverse C^{-1} such that $C^{-1}C=I^{mxn}$. The right inverse can be computed using $C^{-1} = C^T\left(CC^T\right)^{-1}$. Thus, the inverse relationship of equation (18) can be expressed as

$$\begin{bmatrix} f_{1R} \\ f_{1L} \\ f_{2R} \\ f_{2L} \end{bmatrix} = \begin{bmatrix} \frac{d_s}{2(d_s+d_s)} & -\frac{1}{2(d_s+d_s)} & \frac{1}{4l} \\ \frac{d_s}{2(d_s+d_s)} & -\frac{1}{2(d_s+d_s)} & -\frac{1}{4l} \\ \frac{d_s}{2(d_s+d_s)} & \frac{1}{2(d_s+d_s)} & \frac{1}{4l} \\ \frac{d_s}{2(d_s+d_s)} & \frac{1}{2(d_s+d_s)} & -\frac{1}{4l} \end{bmatrix} \begin{bmatrix} F_z \\ I_\theta \\ I_\varphi \end{bmatrix} \tag{19}$$

V. Sensitivity Study for Parameters Tuning

The sensitivity analysis method which also studied by [26, 27] is used in this study to tune fourteen parameters of stability augmentation system (SAS) controller. As mention in Fig.5 in Section IV.B, there are six parameters of an inner loop of SAS (attitude controller) need to be tuned, whereas an outer loop controller (ride controller) consists of eight control parameters. According to Ikenaga [28], it is impossible to tune all these parameters simultaneously. This is due to the fact that the controller parameters will be determined to be a larger value than if the tuning process of inner and outer loops done separately.

Therefore, the tuning can be done in two phases; first phase is tuning the outer loop controller (attitude controller) which has six parameters and the second phase is tuning the eight parameters of inner loop control (ride controller) which all of the outer loop parameters are fixed after first phase.

Sensitivity study was also done in tuning the parameters of *B* and *G* of the MR damper controller with respect to the desired force produced by MR damper. For the SAS and PID controller, the parameter tuning is based on the RMS values of body response of railway vehicle.

VI. Simulation Results

A series of simulation tests have been done to investigate the performance of the semi-active suspension system of railway vehicle featuring with magneto-rheological dampers. This section describes the results obtained from the simulation process using Matlab-SIMULINK software. The results show that the performance of the responses of railway vehicle body with the semi-active system can be clearly seen have an improvement when compared with the passive system. The passive system means that the entire railway suspension system equipped with conventional elements such as springs, air springs and viscous dampers. While for the semi-active system, the system represents the secondary vertical viscous damper are replaced with semi-active damper and the controller is equipped with a function to reduce railway vehicle body unwanted motions due to the vertical track irregularity.

To investigate the advantages of SAS controller, the simulation is performed by comparing the SAS with PID controller. At the same time, for the SAS controller, the importance of SAS with ride controller is also discussed. By plotting all these case together, the relative benefits of semi-active control can be readily seen. Four performance criteria's are considered in this study namely; body vertical acceleration, body vertical displacement, body pitch angle and body roll angle. There are four different lines present in each figure which the dashed-dotted line represents the passive system, while the dotted line represents the system with PID controller, dashed line is system with SAS (attitude) controller and solid line indicate the response of the SAS (attitude and ride) controller respectively.

Figure 8 shows the response of the vehicle body acceleration in real time for passive and semi-active secondary suspension systems. It can be noted that the semi-active secondary suspension of railway vehicle with three different control strategies are able to reduce body vertical response due to the track irregularities.

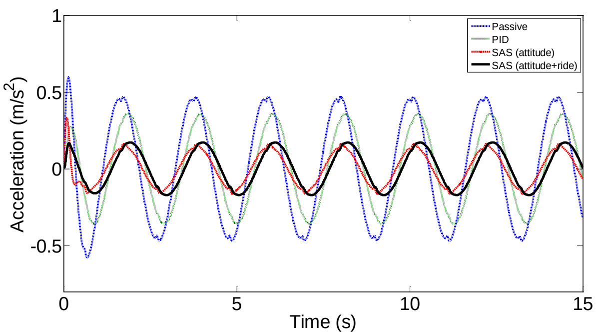

Fig. 8 Body vertical acceleration of railway vehicle

In terms of body displacement, it clearly exhibit that the stability augmentation system is able to eliminate unwanted motion effectively. Although the SAS controller is without ride control loop, the controller is still able to damp out an unwanted body response. On the other hand, PID controller also has an ability to reduce the unwanted body displacement.

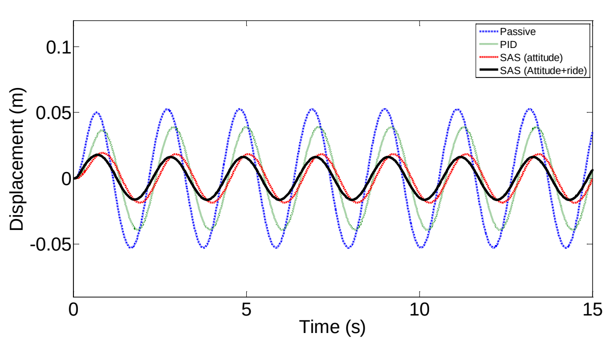

Fig. 9 Body vertical displacement of railway vehicle

Figure 10 and 11 exhibit the time response of the pitch angle and body roll angle of railway vehicle body. From the graphs, it shows that the vehicle body with SAS control has slight improvement in terms of pitch and roll angle response. Even without the ride controller, SAS controller is able to reject an unwanted body pitch and body roll angle. This is due to the fact that the outer loop controller is to stabilize the pitch and roll response due to the track disturbance. The comparison of root-mean-square (RMS values of body vertical acceleration, body vertical displacement, body pitch angle and body roll angle at body centre of gravity are demonstrated in Table 1.

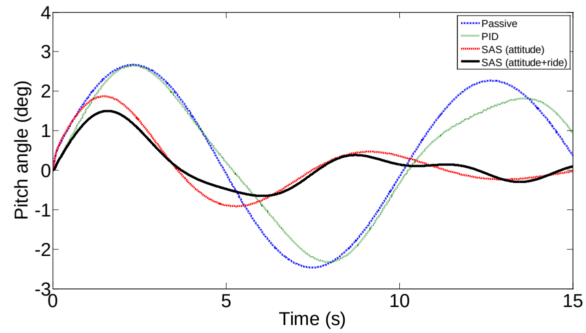

Fig. 10 Body pitch angle of railway vehicle

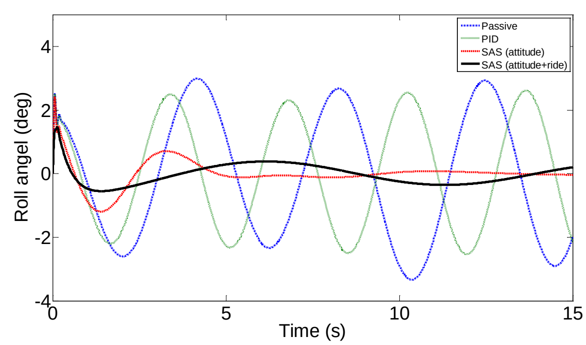

Fig. 11 Body roll angle of railway vehicle

The performance of the semi-active suspension system also been observed based on peak-to-peak (PTP) and root-mean-square (RMS) values of body responses of railway vehicle. The percentage of reduction of body acceleration are 18.6 % for PID controller, 19.7% and 35.4% for SAS controller without and with ride controller respectively. In terms of body displacement response, the percentage reduction of RMS values for PID, SAS without and with ride controllers are 24.1%, 35.5 % and 42.4 % respectively. For pitch angle and roll angle responses, there are also have a significant reduction in RMS values where PID controller is able to reduce 60.1 % for pitch angle and 15.2 % for roll angle. Meanwhile, both SAS controller have 68.9 % and 75.9 % in reducing pitch angle, 40.1 % and 50 % in roll angle reduction. The complete data of the simulation results can be referred in Table 1.

Table 1 RMS values of railway vehicle body responses

Body responses	RMS value						
	Passive	PID	Reduction (%)	SAS (attitude)	Reductio n (%)	SAS (attitude+ride)	Reduction (%)
Acceleration (m/s^2)	0.3195	0.2601	18.6	0.2567	19.7	0.2063	35.4
Displacement (m)	0.0349	0.0265	24.1	0.0225	35.5	0.0201	42.4
Pitch angle (deg)	0.9422	0.3763	60.1	0.2933	68.9	0.2273	75.9
Roll angle (deg)	2.005	1.701	15.2	1.2001	40.1	1.0035	50.0

VII. CONCLUSION

A series of simulation tests have been done to investigate the performance of semi-active suspension control of railway vehicle. A 9-DOF full railway vehicle model has been derived based on Newton's second law. A vertical track irregularity with the amplitude of 0.05 m has been applied as an input disturbance to the system. An MR damper model based on Bouc-Wen model was developed and force tracking control of MR damper was also investigated. Decoupling transformation system was derived and placed between inner loop and outer loop controllers. From the simulation, it can be observed that the semi-active suspension control using PID and stability augmentation system are able to significantly reduce all unwanted vehicle body responses.

APPENDIX

Parameters of 9-DOF vertical model are as following:

Body mass, $m_c = 3.96\times10^4$ kg
Bogie mass, $m_b = 3.25\times10^3$ kg
Velocity of railway vehicle, $V = 55.56$ m/s
Body longitudinal moment of inertia, $I_{cx} = 8.85\times10^4$ kg.m^2
Body lateral moment of inertia, $I_{cy} = 2.46\times10^6$ kg.m^2
Body vertical moment of inertia, $I_{cz} = 2.5\times10^6$ kg.m^2
Bogie longitudinal moment of inertia, $I_{tx} = 3.06\times10^3$ kg.m^2
Bogie lateral moment of inertia, $I_{ty} = 3.02\times10^3$ kg.m^2
Bogie vertical moment of inertia, $I_{tz} = 4.27\times10^3$ kg.m^2
Primary longitudinal spring, $k_{px} = 4\times10^6$ N/m
Primary lateral spring, $k_{py} = 3.25\times10^6$ N/m
Primary vertical spring, $k_{pz} = 7\times10^6$ N/m
Primary longitudinal damper, $c_{px} = 0$ Ns/m
Primary lateral damper, $c_{py} = 0$ Ns/m
Primary vertical damper, $c_{pz} = 1.5\times10^4$ Ns/m
Secondary longitudinal spring, $k_{sx} = 1.5\times10^5$ N/m
Secondary lateral spring, $k_{sy} = 1.5\times10^5$ N/m
Secondary vertical spring, $k_{sz} = 2.9\times10^5$ N/m
Secondary longitudinal damper, $c_{sx} = 0$ Ns/m
Secondary lateral damper, $c_{sy} = 5\times10^4$ Ns/m
Secondary vertical damper, $c_{sz} = 8\times10^4$ Ns/m
Half of bogie centre pin spacing, $l = 9.00$ m
Half of wheel-set contact distance, $d_s = 1.00$ m
Half of primary suspension spacing, $d_p = 1.00$ m
Half of wheelset contact distance, $a = 1.00$ m
Half of wheelbase, $b = 1.25$ m
Vertical distance from truck frame center of gravity to secondary suspension, $h_{ts} = 0.217$ m
Vertical distance from car body center of gravity to secondary suspension, $h_{cs} = 1.207$ m
Vertical distance from truck frame center of gravity to primary suspension, $h_{tp} = 0.452$ m
Vertical distance from wheelset center of gravity to primary suspension, $h_{wp} = 0.18$ m

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