VEHICLE ACTIVE SUSPENSION CONTROL USING MULTI-ORDER PID APPROACH

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ABSTRACT: This paper presented the comparison among passive, proportional-integral-differential (PID) and multi-order PID (MOPID) controlled active suspensions. A quarter car model was validated with previous experimental results. An active suspension control through simulation was performed at three different frequency regions; below natural frequency of body, in between natural frequency of body and wheel and above natural frequency of wheel. The parameters were tuned and the controller was evaluated through simulation. The MOPID controller produced good performance in controlling vertical body displacement by 75 % reduction of amplitude and acceleration by 45 % reduction of amplitude, thus, improving ride performance and comfort. The controller can be used for future laboratory scale experiment with available components.

KEYWORDS: *PID, MOPID, active suspension, ride quality, ride performance.*

1.0 INTRODUCTION

Due to ride quality and vehicle handling compensations in passive suspension system, many research studies have introduced several semi active and active suspensions in order to improve ride quality without compromising vehicle handling and stability. These semi active and active suspensions have been implemented in simulations, laboratory scale test rig and actual in-vehicle experiments whether in non-guided vehicles such as passenger car [1] and guided vehicles such as railway and locomotive [2].

Dynamic characteristics (vibrations and moments) which affect the vehicle performances [3-5] can be observed through real driving experiment of vehicle instrumented with simple and user-friendly data acquisition (DAQ) system [6-8]. The affected vehicle performances are mainly ride quality, handling and stability. The controlling of vehicle heave and pitch motions are hardly investigated and some studies use 2 degrees of freedom (DOF), 4 DOF, 5 DOF and 7 DOF vehicle models [9] in controlling active suspension of ground vehicle [10-12]. Various active suspension system models with either quarter or half car models have been used in the design of the controllers in their studies.

The primary function of suspension is to isolate the vehicle structure whenever practicable from shock loading and vibration due to irregularities of the road surface travelled. Furthermore, the suspension system must, at the same time, maintain the stability, steering and general handling qualities of the vehicle [13, 14]. The first requirement can be met using flexible elements and dampers. On the other hand, the second function can be achieved by controlling the relative motion between unsprung masses - wheel-and-axle assemblies - and the sprung mass through the use of mechanical linkage [15].

This paper investigated and compared the performance of active suspension between PID and MOPID control approaches. The performance criterion to be evaluated was the ability of the controller in reducing vertical sprung mass acceleration and suspension travel.

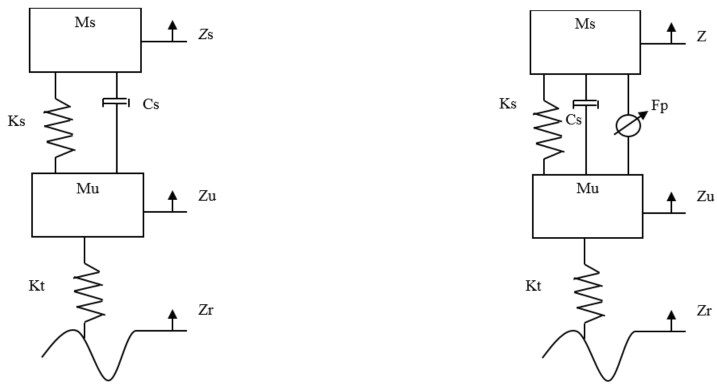
2.0 METHODOLOGY

Three steps of methodology were utilized in this study. The steps are modeling and validation of 2 DOF quarter car model, PID control approach and MOPID control approach.

2.1 Modeling and Validation of 2 DOF Quarter Car Model

In order to understand issues related to suspension control, the simplified vehicle model was used for analysis. The equation of motion for quarter-car model was used as the fundamental for all further analysis. The validation of the mathematical model of quarter car suspension was performed by comparing it with the experimental data of instrumented quarter car test rig [16] as the benchmark.

Well-known model for simulating one-dimensional vehicle suspension performance is by deriving a single wheel station or quarter car model (Figure 1a). The simulation using quarter car model is repeatedly used to solve problems related to suspension system [9, 17-24].



(a) (b)

Figure 1: (a) Passive and (b) active quarter car models

The quarter car model with two degree of freedom comprised the displacement of the unsprung mass Zu and the displacement of the sprung mass Zs. The road input was expressed by Zr and the differential equations of quarter car model for passive suspension two degree of freedom according to Figure 1 were given by

$$M_u \ddot{Z}_u + K_t(Z_u - Z_r) + K_s(Z_u - Z_s) + C_s(\dot{Z}_u - \dot{Z}_s) = 0 \quad (1)$$

$$M_s \ddot{Z}_s + K_s(Z_s - Z_u) + C_s(\dot{Z}_s - \dot{Z}_u) = 0 \quad (2)$$

Where Mu represents the wheel mass or unsprung mass, Ms is the body mass or sprung mass, Cs is the stiffness of the damper, Ks is the stiffness of the spring and Kt is the stiffness of the tire. The tire is shown as a spring because the damping in the rolling tire is typically very low and ignored in this analysis [13, 14]. The effect of dry friction is neglected so that the residual structural damping is not studied in the quarter car vehicle suspension modelling. The active suspension quarter car model also consists of two vertical degrees of freedom (Figure 1b). The difference between the passive quarter car suspension and the active quarter car model was only at the additional force from the actuator Fp. The equations of active suspension model for 2 DOF according to Figure 2 are given by

$$M_u \ddot{Z}_u + K_t(Z_u - Z_r) + K_s(Z_u - Z_s) + C_s(\dot{Z}_u - \dot{Z}_s) - F_n = 0 \quad (3)$$

$$M_s\ddot{Z}_s + K_s(Z_s - Z_u) + C_s(\dot{Z}_s - \dot{Z}_u) + F_n = 0 \quad (4)$$

The quarter car model was validated using the experiment data [16]. The procedure to validate the quarter vehicle suspension model in difference frequencies was proposed [11]. A comparison between the experiment data and previous research [16] showed the quarter vehicle suspension test rig generated sinusoidal road profile using a slider-crank mechanism. The signal generated by this mechanism was continuous and frequency adjustable. The parameters of the simulation are stated in Table 1.

Table 1: Parameters of simulation

Parameter	Value	Unit
Mu	30	kg
Ms	150	kg
Cs_1	1,000	Ns/m
Ks	37,500	N/m
Kt	100,000	N/m

2.2 PID Control Approach

The controller for active suspension system based on control strategy known as proportional integral derivative (PID) controller was used for initial approach. PID controller was chosen because the PID controller has been efficient in many operations. It is also easy to maintain and embed a real system [23]. A PID controller attempts to calculate an error value between the set point and the control variable by minimizing the error. The proposed system is a control loop feedback mechanism that makes use a difference of error to trigger the control response.

A typically used structure of a PID control system is shown in Figure 2. In order to generate the proportional, integral, and derivative actions, the error signal e(t) was used, resulting in signals weighted and formed the control signal u(t) adapted to the plant model. The saturation block was used to remove the upper and lower noises data input from the controller.

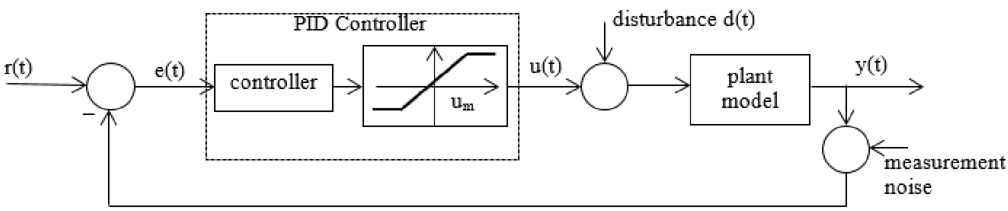


Figure 2: Overall control structure of PID controller

The PID controller was separated into a three system that consisted of proportional compensation, integral compensation and derivative compensation [15]. The main purpose of the proportional compensator was to produce the gain by comparing the system output and input that was proportional to the error. The purpose of integral compensation was to introduce the integral part of the error signal. The area under the curve of error signals gave effect to the output signal. This will improve the steady-state error of overall closed-loop controller system. The derivative compensation was introduced to the error signal of derivative input. The main purpose of derivative compensation was to improve the transient response of the overall closed-loop controlled system. For a PID controller, the mathematical model of the system is:

$$u(t) = k_p e(t) + k_i \int_0^t e(\tau) d\tau + k_d \frac{de}{dt}, \tag{5}$$

Where u(t) is the input signal to the plant model, the error signal e(t) is designated as e(t) = r(t) − y(t), and the reference input signal r(t).

The simulation of Matlab/Simulink active suspension controlled by PID model was observed for a period of 10 second using default solver with the step size set to 0.0002 second. Sinusoid road disturbances were implemented in this simulation. In order to evaluate the performance of active suspension, the frequency domain was split into three different regions, natural frequency of below body region, natural frequencies of between body and wheel region, and natural frequency of above wheel region. For the suspension considered in these studies and the suspension in common light passenger vehicle, the value of the vehicle body natural frequency was around 2 Hz whereas the vehicle wheel natural frequency was around 10 Hz. The frequencies of 0.5 Hz, 5 Hz, and 15 Hz were selected as sinusoid road profiles in this simulation to represent the frequency of road disturbance in natural frequency below body region, natural frequencies between body and wheel region, and natural frequency above wheel region. These performance evaluation methods are taken from previous research [23]. The parameters of PID controller are listed in Table 2.

Table 2: Parameters of the PID controller

Parameter	Value
K_P	17155
K_I (1/sec)	-515
K_D (sec)	375

2.3 MOPID Control Approach

The MOPID control system used a closed feedback loops that employed a number of PID controllers to generate the control action to actuator. These control strategies were used to make a comparison studies in order to optimize ride comfort between PID controller and MOPID controller (Figure 3). The study focused on feedback of the controller by comparing the results of body vertical displacement and body vertical acceleration, focusing on ride comfort.

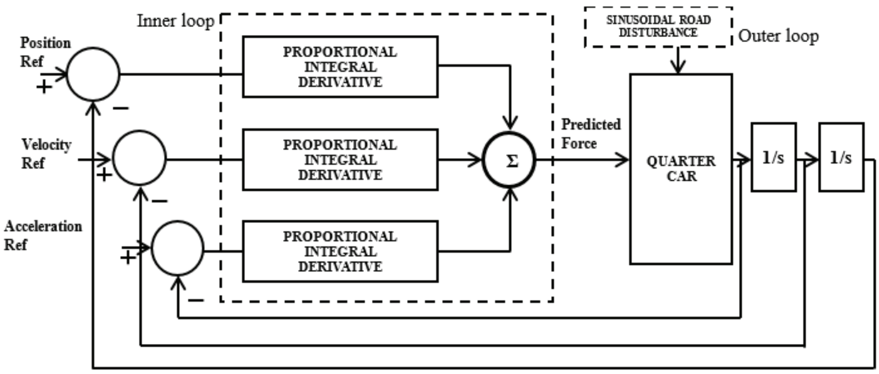


Figure 3: Structure of MOPID control

From Figure 3, the output value for predicted force to active suspension quarter car model is given by

$$\sum F = e_z\left[K_P + \frac{K_I}{s} + K_D s\right]_z + e_{\dot{z}}\left[K_P + \frac{K_I}{s} + K_D s\right]_{\dot{z}} + e_{\ddot{z}}\left[K_P + \frac{K_I}{s} + K_D s\right]_{\ddot{z}} \tag{6}$$

With the error signal of body displacement, body velocity and body acceleration. KP, KI and KD are the proportional constant, the integral constant and derivative constant of the PID controller. The parameters of PID controller are listed in Table 3.

Table 3: Parameters of the MOPID controller

Parameter	Value		
	Displacement	Velocity	Acceleration
K_P	14671	3026	811
K_I (1/sec)	84163	15466	3984
K_D (sec)	1000	50	10

3.0 RESULTS AND DISCUSSION

The validation results of vehicle body vertical displacement are shown in Figures 4 and 5. The validation results of vehicle body vertical displacement are shown into five different frequencies of 0.94 Hz and 1.18 Hz. From the results, the trend between experimental data and simulation result was very identical with small error. There were misalignments at the early of the simulation and after 2 second the graph became stable and identical. The overall reading of vehicle body vertical displacement follows the same pattern of experiment data [16]. The validation results of vehicle body vertical acceleration are shown in Figures 6 and 7. These figures of vehicle body vertical acceleration are shown into five different frequencies of 0.94 Hz and 1.1 8 Hz. From the results, the trend between experimental data and simulation result was almost the same. The results showed that there were inconstancy results due to the sliding bush friction of experiment rig and some frictions and knocking in slider-crank mechanism. The simulation data follows the same pattern of experiment data [16].

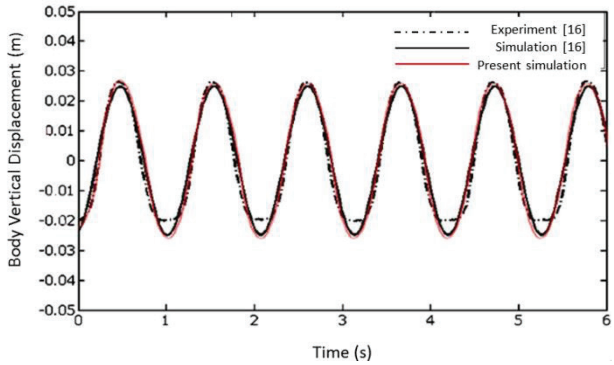


Figure 4: Validation result for vehicle body vertical displacement at 0.94 Hz

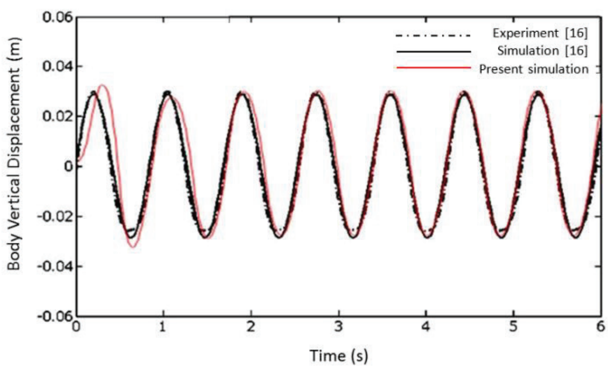


Figure 5: Validation result for vehicle body vertical displacement at 1.18 Hz

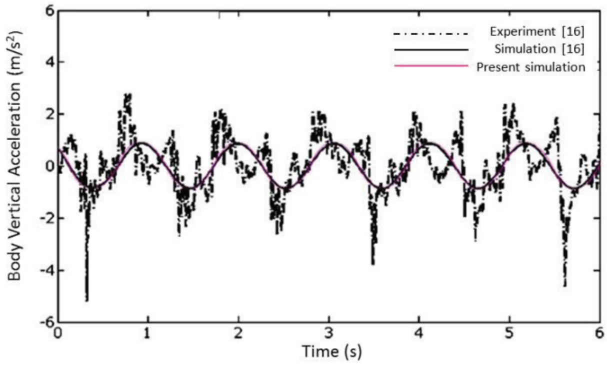


Figure 6: Validation result for vehicle body vertical acceleration at 0.94 Hz

Bode plot of frequency domain response comparison among MOPID, PID and passive suspension for body vertical displacement and body vertical acceleration are shown in Figure 8. From the figure, the MOPID active suspension system had successfully scaled down the amplitude of body states at a very wide frequency band. Velocity and acceleration of the body as the controlled variables and response in the body natural frequency can be reduced more when they are applied with MOPID controller suspension system.

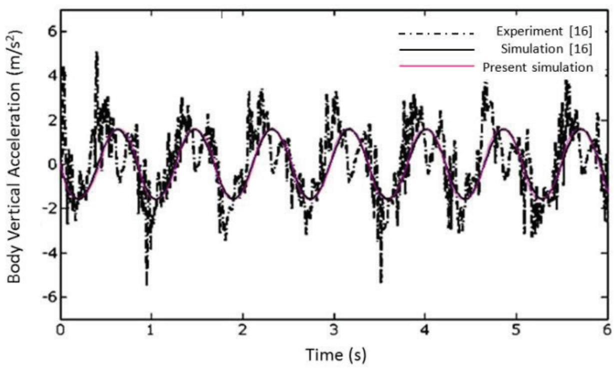


Figure 7: Validation result for vehicle body vertical acceleration at 1.18 Hz

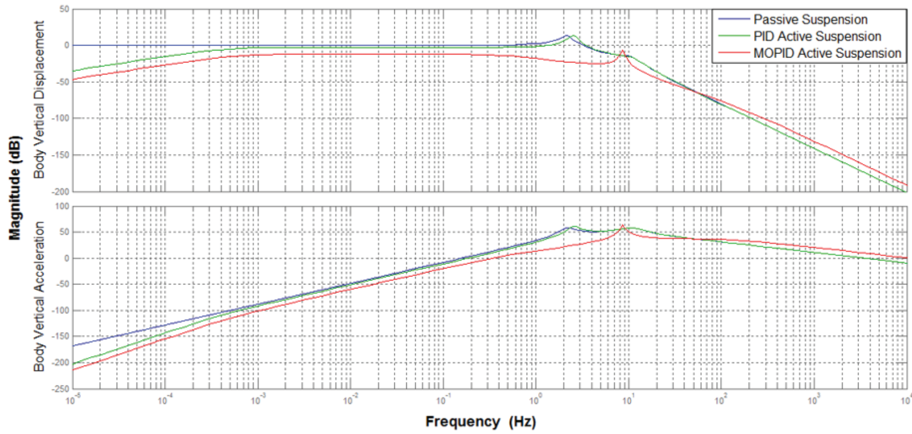


Figure 8: Frequency domain response

The frequencies of 0.5 Hz, 5 Hz, and 15 Hz were selected as sinusoid road profiles to use for performance evaluated in terms of root mean square (RMS). The smaller value of RMS, the better was the performance of the result. The value of performance evaluated is listed in Table 4. The MOPID controller has reduced the RMS values of body vertical displacement and body vertical acceleration better than PID control and passive system. Table 4 shows the significantly reduced of RMS values of body displacement and body acceleration compared with PID controller and passive system. From the results of body acceleration, the introduction of MOPID active suspension in light passenger vehicle can improve the ride comfort more than PID active system.

Body vertical displacement and body vertical acceleration feedback from the input of the 0.5 Hz road disturbance are shown in Figures 9 and 10, indicating the below body natural frequency region. It showed that body vertical displacement and body vertical acceleration after the introduction of MOPID active suspension system were better compared to PID active suspension and passive suspension system.

Table 4: Time domain response comparison between various systems for 0.5 Hz

Frequency (Hz)	Criteria	RMS value		
		Passive	PID	MOPID
0.5	Body Displacement (m)	0.0424	0.0255	0.0156
	Body Acceleration (m/s^2)	0.4184	0.2515	0.2157
5	Body Displacement (m)	0.0071	0.0055	0.0016
	Body Acceleration (m/s^2)	7.033	4.864	1.623
15	Body Displacement (m)	0.0018	0.00063	0.0001
	Body Acceleration (m/s^2)	15.68	6.116	1.164

Figure 9 shows that the sinusoid input at the amplitude of 0.04 m reduced to approximately 0.01 m. The 75% reduction of body displacement after the sinusoid input showed that the MOPID active suspension improved the performance of the vehicle suspension. From the Figure 10, body vertical acceleration value of MOPID active suspension value was reduced by a large percentage compared to PID active suspension and passive suspension. This reduced value after the introduction of MOPID active suspension system improved vehicle ride comfort.

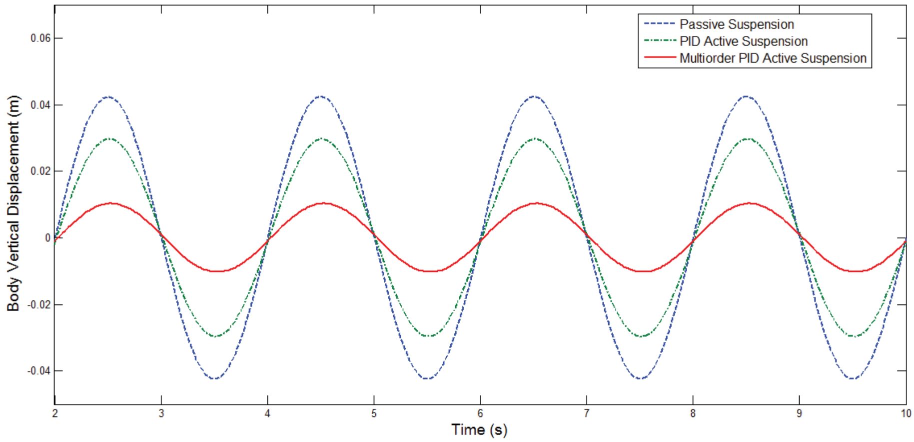


Figure 9: Body vertical displacement of 0.5 Hz sinusoid road profile

4.0 CONCLUSION

The development of Matlab/Simulink 2DOF quarter car model has been performed. The simulations are carried out with the same parameters of [16]. The results show that the trend of simulation and the experiment data of [16] is identical and has the same pattern. Thus the model is validated. The simulation of MOPID control for active suspension has been studied to investigate the performance improvement between MOPID controller, PID controller and passive system. These studies focus on the improvement of ride performance and ride comfort in the suspension system. Therefore, the result focuses on body vertical displacement and body vertical acceleration. From the simulation results, the MOPID controller shows better performance compared to the PID controller and passive suspension system. The introduction of multiple PID loops, outer loop and inner loop are able to improve the ride performance and ride comfort. It can be concluded that the proposed MOPID controller has a good performance in controlling the body vertical displacement and body vertical acceleration. The MOPID controller system improves the ride performance and ride comfort of the manuscripts.

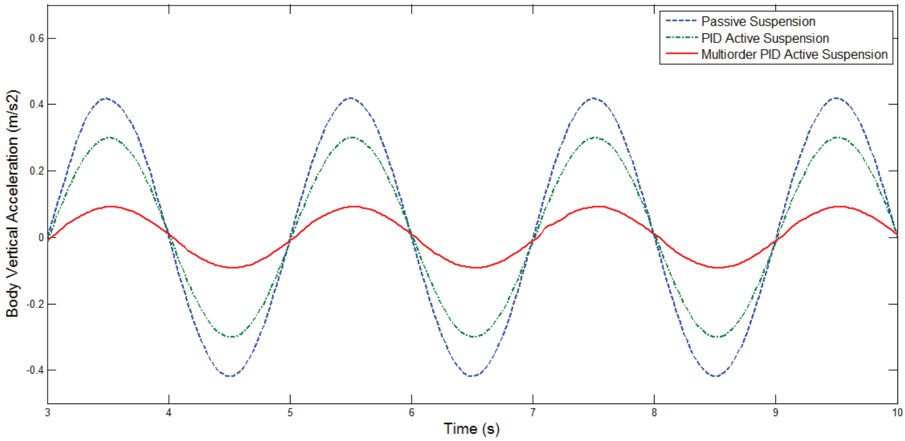


Figure 10: Body vertical acceleration of 0.5 Hz sinusoid road profile

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REFERENCES

[1] T. Yoshimura and A. Takagi, "Pneumatic Active Suspension System for a One-Wheel Car Model Using Fuzzy Reasoning and a Disturbance Observer," *Journal of Zhejiang University Science*, vol. 5, no. 9, pp. 1060-1068, 2004.

[2] M.F. Md Isa, M.H. Harun, M.A. Abdullah and N.S. Muhammad, "Force Tracking Control of MR Damper and Controller Parameter Optimization using Sensitivity Analysis Method," Proceedings of Mechanical Engineering Research Day 2015, Melaka, 2015, pp. 61-62.

[3] M.A. Abdullah, M.A. Salim and M.Z. Mohammad Nasir, "Dynamics Performances of Malaysian Passenger Vehicle," Proceedings of International conference on Automotive, Innovation and Green Energy Vehicle, 2014, pp. 26-27.

[4] M.A. Abdullah, J.F. Jamil, N. Ismail, M.Z. Mohammad Nasir and M.Z. Hassan, "Formula Varsity Race Car - Roll Dynamic Analysis. *Proceedings of Mechanical Engineering Research Day 2015*, Melaka, 2015, pp. 23-24.

[5] M.A. Abdullah, J.F. Jamil and M.A. Salim, "Dynamic Performances Analysis of a Real Vehicle Driving," *IOP Conference Series: Materials Science and Engineering*, vol. 100, no. 1, pp. 012-017, 2015.

[6] M.A. Abdullah, F.R. Ramli and C.S. Lim, "Railway Dynamics Analysis using Lego Mindstorms," *Applied Mechanics and Materials*, Trans Tech Publications, vol. 465, pp. 13-17, 2014.

[7] M.A. Abdullah, M.A. Romeli, F.R. Ramli and M.K. Mohamed Nor, "Driving Assistance System – Automatic Parking Maneuver using Lego Mindstorms," *Proceedings of Mechanical Engineering Research Day 2015*, Melaka, 2015, pp. 25-26.

[8] M.A. Abdullah, M.A. Azan, F.R. Ramli and M.K. Mohamed Nor, "Autonomous Vehicle Convoy using Lego Mindstorms," *Proceedings of Mechanical Engineering Research Day 2015*, Melaka, 2015, pp. 27-28.

[9] M.A. Abdullah, F.A. Abdul Kadir, M.Z. Mohammad Nasir, A.S. Phuman Singh and J.F. Jamil, *Vehicle System Modeling & Simulation*. Teaching and Learning Series, Faculty of Mechanical Engineering, Module 12, Melaka: Penerbit Universiti, Universiti Teknikal Malaysia Melaka, 2015.

[10] A. Zaremba, R. Hampo and D. Hrovat, "Optimal Active Suspension Design using Constrained Optimization," *Journal of Sound and Vibration*, vol. 207, no. 3, pp.351-364, 1997.

[11] S.A. Ikenaga, "*Development of a Real Time Digital Controller: Application to Active Suspension Control of Ground Vehicles,*" Ph.D. dissertation, Michigan University, 2000.

[12] H. Du and N. Zhang, "H∞ Control of Active Vehicle Suspensions with Actuator Time Delay," *Journal of Sound and Vibration*, vol. 301, pp. 236-252, 2007.

[13] M.A. Abdullah, J.F. Jamil, A.E. Mohan, A.K. Mat Yamin and N. Tamaldin, *Vehicle Dynamics – Analysis & Experimentation.* Teaching and Learning Series, Faculty of Mechanical Engineering, Module 13, Melaka: Penerbit Universiti, Universiti Teknikal Malaysia Melaka, 2016.

[14] M.A. Abdullah, J.F. Jamil, A.K. Mat Yamin, N.R. Mat Nuri and M.Z. Hassan, *Vehicle Dynamics*. Teaching and Learning Series, Faculty of Mechanical Engineering, Module 10, Melaka: Penerbit Universiti, Universiti Teknikal Malaysia Melaka, 2015.

[15] M.A. Abdullah, F.A. Abdul Kadir, A.S. Phuman Singh, F. Ahmad and K.Hudha, *Vehicle Control System*. Teaching and Learning Series, Faculty of Mechanical Engineering, Module 11, Melaka: Penerbit Universiti, Universiti Teknikal Malaysia Melaka, 2015.

[16] I. Fitrian, *Control of Pneumatically Actuated Active Suspension System using Multiple Proportional-Integral with Knowledge-Based Fuzzy*. PhD Dissertation, Universiti Teknikal Malaysia Melaka, 2010.

[17] M.M. Elmadany and Z. Abduljabbar, Z. "On the Statistical Performance of Active and Semi-Active Car Suspension Systems," *Computers & Structures*, vol. 33, no. 3, pp. 785-790, 1989.

[18] R. B. Mrad, J.A. Levitt and S.D. Fassois, "Non-Linear Dynamic Modelling of an Automobile Hydraulic Active Suspension System," *Mechanical Systems and Signal Processing*, vol. 8, no. 5, pp. 485-517, 1994.

[19] C.S. Ting, T.H.S. Li and F.H. Kung, "Design of Fuzzy Controller for Active Suspension System," *Mechatronics*, vol. 5, no. 4, pp. 365-383, 1995.

[20] M.V.C. Rao and V. Prahlad, "A Tunable Fuzzy Logic Controller for Vehicle - Active Suspension Systems," *Fuzzy Sets and Systems*, vol. 85, pp. 11-21, 1997.

[21] H.S. Roh and Y. Park, "Stochastic Optimal Preview Control of an Active Vehicle Suspension," *Journal of Sound and Vibration*, vol. 220, no. 2, pp. 313-330, 1999.

[22] T. Yoshimura, A. Kume, M. Kurimoto and J. Hino, "Construction of an Active Suspension System of a Quarter Car Model using the Concept of Sliding Mode Control," *Journal of Sound and Vibration*, vol. 239, no.2, pp. 187-199, 2001.

[23] K. Hudha, H. Jamaluddin, P.M. Samin and R.A. Rahman, "Effects of Control Techniques and Damper Constraint on the Performance of a Semi-Active Magnetorheological Damper," *International Journal of Vehicle Autonomous Systems*, vol. 3, no. 2/3/4, pp. 230-252, 2005.

[24] S. Türkay and H. Akçay, "Aspects of Achievable Performance for Quarter-Car Active Suspensions," *Journal of Sound and Vibration*, vol. 311, pp. 440-460, 2008.