Journal of Engineering Science and Technology
Special Issue on ISSC'2016, April (2017) 39 - 51

INTELLIGENT CONTROLLER DESIGN FOR A NONLINEAR QUARTER-CAR ACTIVE SUSPENSION WITH ELECTRO-HYDRAULIC ACTUATOR

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Abstract

Nowadays, active suspension system becomes important to the automotive industries and human life due to its advantages in improving road handling and ride comfort. The aims of this project are developing mathematical modelling and design an intelligent control strategy. The project will begin with a mathematical model development based on the physical principle of the passive and active suspension system. Electro-hydraulic actuator was integrated in order to make the suspension system under the active condition. Then, the model will be analysed through MATLAB and Simulink software. Finally, the proportional-integral-derivative (PID) controller and an intelligent controller which is Fuzzy Logic are designed in the active suspension system. The results can be obtained after completing the simulation of the quarter-car nonlinear passive and an active suspension system. From the simulation made through MATLAB and Simulink, the response of the system will be compared between nonlinear passive and nonlinear active suspension system. Besides that, the comparison has been made between Fuzzy Logic and PID controller through the characteristics of a vehicle body and control force from the suspension system. As a conclusion, developing a nonlinear active suspension system with electro-hydraulic actuator for quarter car model has improved the car performance by using a Fuzzy Logic controller. Otherwise, the suspension control system may serve for ride comfort and to support the body of the vehicle. The improvements in performance will improve road handling and ride comfort performance of both systems.

Keywords: Nonlinear quarter car, Suspension, PID controller, FLC controller.

Nomenclatures

A	Piston area, m^2
b_s	Damping coefficient of the suspension, N_s/m
b_t	Damping coefficient of the tyre, N_s/m
b_s^l	Damping coefficient
b_s^{nl}	Nonlinear component of the damper forces
b_s^{sym}	Asymmetric characteristics
F	Actuator force
k_t	Suspension and wheel sttiffnesses, N/m
k_v	Servo-valve gain, m/v
k_s^l	Spring constant
ks^{nl}	Nonlinear component of the spring forces
M_1	Mass of the car body or sprung mass, kg
M_2	Mass of the wheel or unsprung mass, kg
P_s	Supply pressure, Pa
w	Road disturbance
x_1, x_2	Vertical displacement of the body and wheel
x_2-w	Wheel deflection, cm
x_1-x_2	Suspension travel, cm

Greek Symbols

α	Actuator parameter
β	Actuator parameter
Υ	Actuator parameter
τ	Actuator time constant, s

Abbreviations

GA	Genetic Algorithm
FLC	Fuzzy Logic Controller
PID	Proportional-Integral-Derivative
SMC	Sliding Mode Controller

1. Introduction

In general, an active suspension is widely used in automation system as it is an important part for safety purpose. Balancing the trade-off between ride quality and road handling performance is a purposed of the suspension system [1]. This performance can be achieved by maintaining the relative position and movement between the vehicle body and wheels. Hence, the effects of vibrations will be reduced for particular road profile. The performance of the handling requires a stiff suspension because the system becomes stable when the tire contact keeping with the road. Suspension can be categorized as a dangerous and safe condition. The dangerous suspension is referring the road irregularities which can allow the body, resulting in poor ride comfort performance.

Back stepping nonlinear controller design for quarter car suspension system is presented in [2]. The first step to the design of backstepping controller is to choose the regulated variable. The closed loop system in the nonlinear backstepping shows that a good performance depends on the regulated variable in term of $Z_1 = X_1 - X_3$, where X_1 is displacement of the body car and X_3 is a displacement of the angular version. This step can avoid the swing of zero dynamics in the system. The studies also introduced wheel displacement which the wheel is a nonlinear filter and the bandwidth in the suspension travel is a nonlinear function. For the passenger's comfort, the resulting response must be soft. With the choice of regulated variable, the backstepping design procedure will take a few steps.

Besides that, a Proportional-Integral Sliding Mode Controller design for quarter car active suspension system is presented in [3]. The Sliding Mode Controller (SMC) is implemented to control an active suspension system. The research presents the stability of the asymptotic when the existence of the SMC. Thus, a linear suspension system in the research used the quarter car as a model.

Other than that, a Fuzzy Logic Controller (FLC) design for half-car active suspension system [4]. The study describes a fuzzy based intelligent, active suspension system. This system provides improvement in riding quality and able to minimize both of displacement and accelerations between the centre of the vehicle and maximum angle. The first rule to design the logic control is determining the type of design. For this situation, the mean square must minimize which can easy to get the good time performance. As known, the relative displacement between suspension part and vehicle body are acceptable in the constraints.

A FLC design for quarter car models with a satisfactory performance is proposed in [5,6]. The researcher introduced Genetic Algorithm (GA) to apply on the vehicle suspension. In this research, the deterministic of the sinusoidal function is used as a road surface. Otherwise, the paper presents a main objective where passengers' cars feel comfortable with active suspension system. The active suspension system is categorized followed by principle criterion of vehicle body like suspension deflection and complementary control of the fuzzy-logic. In order to minimize the maximum deflection between characteristics vehicle body and suspension parts, the genetic algorithm method is used on the system.

In addition, FLC also provides stability of the system under control is quite an issue that always be discussed. As well as its ability to ensure robustness with respect system of parameter changes [7]. The result of the FLC design can easily to be appointed using the principle of model equivalent. The model can be categorized in term of output controllers which are linear and nonlinear system whereas for the input model consists a FLC.

Optimal design of Proportional-Integral-Derivative (PID) controller into the nonlinear quarter-car vehicle suspension system is presented in [8]. This method is an effectively in designing of specifications on the ride comfort. Main objective of this paper is to improve the suspension system in term of steady-state characteristics and transient response as well as road handling. The large of derivative gains is produced to increase the rise time where to make the suspension system more sensitive to output.

PID controller design for the fully active suspension system is presented in [9]. The development of this system is used for tuning road input using PID controller. Ziegler-Nichols tuning rules are used in this system which to find the proportional gain, reset rate and determine the derivative time of PID controller. The system is designed for a bumpy road, pothole and random road inputs.

This paper presents the capability of a PID controller and FLC controller to deal with an uncertainties and highly nonlinear system is analysed based on quarter-car model. The active electro-hydraulic suspension is introduced in this system. The hydraulically actuated suspension is controlled with the use of hydraulic servomechanism. The comparative assessment between nonlinear passive system, nonlinear active PID controller and nonlinear active with FLC controller are presented and discussed. The performances of the system are verified in terms of body acceleration, suspension travel and wheel deflection.

2. Mathematical modelling of nonlinear passive and an active suspension system

This section, the modelling of the nonlinear passive and active suspension system will be discussed. In order to completing the objectives of project, the quarter car passive and active suspension will also be introduced.

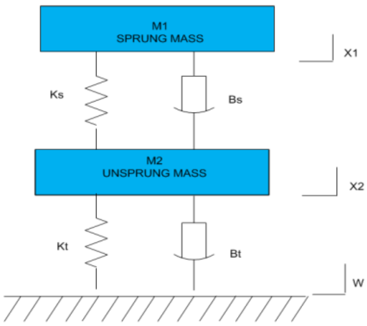


Fig. 1. Nonlinear passive suspension for quarter car model.

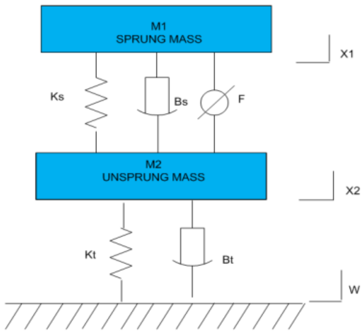


Fig. 2. Active nonlinear suspension for quarter car model.

From the Fig. 1 derivations of mathematical equation of nonlinear passive suspension system of quarter car system are given by Eqs. (1) and (2).

$$M_1\ddot{x}_1 = k_s{}^{l}\left(x_2 - x_1\right) + k_s{}^{nl}\left(x_2 - x_1\right)^3 + b_s{}^{l}\left(\dot{x}_2 - \dot{x}_1\right) - b_s{}^{sym}\left|\dot{x}_2 - \dot{x}_1\right| \tag{1}$$

$$M_2\ddot{x}_1 = -k_s{}^{l}\left(x_2 - x_1\right) - k_s{}^{nl}\left(x_2 - x_1\right)^3 - b_s{}^{l}\left(\dot{x}_2 - \dot{x}_1\right) + b_s{}^{sym}\left|\dot{x}_2 - \dot{x}_1\right| - b_s{}^{nl}\sqrt{\left|\dot{x}_2 - \dot{x}_1\right|}\,\mathrm{sgn}\left(\dot{x}_2 - \dot{x}_1\right) \tag{2}$$

The quarter car model of nonlinear active suspension system is shown in Fig. 2. The nonlinear active suspension system has the additional advantages which a negative damping can be produced and generating the large range of force into the system at low velocities. Besides, this condition potentially will increase the performance of the suspension system. The derivations of mathematical equation of nonlinear active suspension system of quarter car system are given by Eqs. (3) and (4).

$$M_1\ddot{x}_1 = k_s{}^l(x_2 - x_1) + k_s{}^{nl}(x_2 - x_1)^3 + b_s{}^l(\dot{x}_2 - \dot{x}_1) - b_s{}^{sym}|\dot{x}_2 - \dot{x}_1| + b_s{}^{nl}\sqrt{|\dot{x}_2 - \dot{x}_1|}\,\mathrm{sgn}(\dot{x}_2 - \dot{x}_1) - A_{xp} \quad (3)$$

$$M_2\ddot{x}_1 = -k_s{}^l(x_2 - x_1) - k_s{}^{nl}(x_2 - x_1)^3 - b_s{}^l(\dot{x}_2 - \dot{x}_1) + b_s{}^{sym}|\dot{x}_2 - \dot{x}_1| - b_s{}^{nl}\sqrt{|\dot{x}_2 - \dot{x}_1|}\,\mathrm{sgn}(\dot{x}_2 - \dot{x}_1) + k_t(x_2 - \dot{w}) + b_t(x_2 - \dot{w}) + A_{xp} \quad (4)$$

Table 1 shows the parameters of the quarter-car model and Table 2 shows the parameters of Electro-hydraulic actuator system.

Table 1. Parameters of quarter car model [10].

Types	Unit	Value of Parameter
The mass of the car body or sprung mass	M_1	290.0 kg
The mass of the wheel or unsprung mass	M_2	40.0 kg
Tyre stiffness	k_t	1.5×10^5 N/m
Suspension stiffness (linear)	k_s	2.35×10^4 N/m
Suspension stiffness (nonlinear)	k_s^{nl}	2.35×10^6 N/m
Suspension damping (linear)	b_s	700.0 N_s/m
Suspension damping (nonlinear)	b_s^{nl}	400.0 N_s/m
Suspension damping (asymmetrical)	b_s^{sym}	400.0 N_s/m

Table 2. Parameters of electro-hydraulic actuator system [10].

Types	Unit	Value of Parameter
Piston area,	A	3.35×10^{-4} m^2
Actuator time constant	T	3.33×10^{-2} s
Supply pressure	P_s	10, 342 500 Pa
Servo-valve gain	k_v	0.001 m/v
Actuator Parameter	α	4.515×10^{13}
Actuator parameter	β	1
Actuator parameter	Υ	1.545×10^9

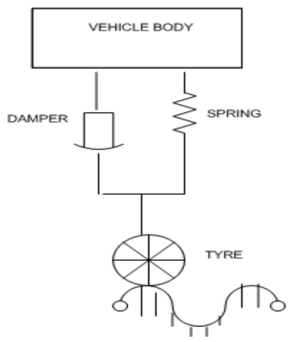
Road profile

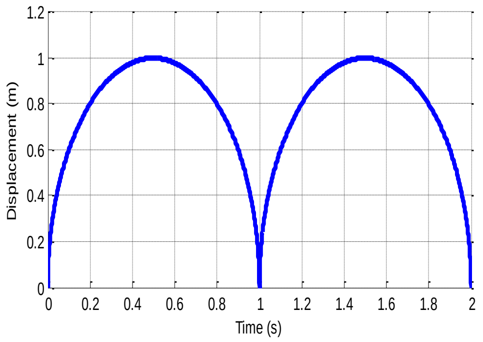
Irregularities of road profile can be categorized into smooth, rough minor and rough in nature. Single bump disturbance is represented as a smooth road. Otherwise, the rough minor and rough in nature are represented in term of uniform or non-uniform. Road disturbance with double bump was used in this project as the input disturbance. Figure 3 is represents the bumpy road input with 1 meter bump height is used in this project.

3. Nonlinear active suspension system on ride comfort and road handling

The performance of the ride comfort may be analysed through the car body acceleration and the performance of road handling may be analysed through the wheel deflection. Four parameters will be obtained in the simulation. There are body acceleration of sprung mass, suspension travel, acceleration of unsprung mass and wheel deflection. Figure 4 shows the block diagram of nonlinear passive and nonlinear active suspension system. The simulation through

MATLAB/Simulink is used to make the comparison between nonlinear passive and nonlinear active suspension systems.





(a) Actual bumpy road. **(b) Bumpy road input.**

Fig. 3. Bump road disturbance input.

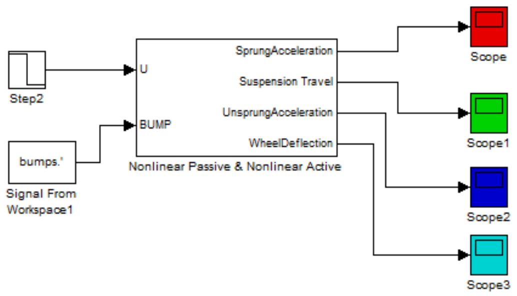


Fig. 4. Block diagram of nonlinear passive and nonlinear active suspension system.

4. Control strategy

A nonlinear active suspension system with PID controller and FLC controller will be designed to minimize the disturbance in the system. First of all, the hydraulic actuator is introduced into the suspension system as a control input as shown in Fig. 5. Control input from the hydraulic actuator make the nonlinear active suspension system more stable.

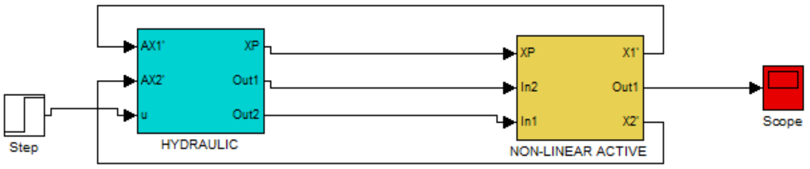


Fig. 5. Block diagram of nonlinear quarter-car active suspension with electro-hydraulic actuator.

4.1. Proportional-Integrator-Derivative controller

The controller parameters are selected based on performance specification is known as controller tuning. Trial and error tuning method is used in this system for tuning PID controllers (meaning to set values KP, KI, KD). PID is mean for proportional, integral and derivative. These controllers are used to abolish the requirement of the system. Hence, the PID controller is designed to prevent the smallest variation of the output at the steady state. Designing of the PID also can minimize the error by the controller in term of the derivative. Error in the controller will be defined as the difference measurement of the variable and the set point. Figure 6 shows the block diagram of nonlinear quarter-car active suspension with PID controller and Table 2 shows the parameters of the PID controller.

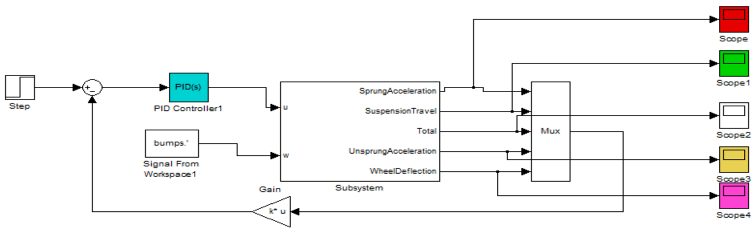


Fig. 6. Block diagram of nonlinear quarter-car active suspension with PID controller.

Table 3. Parameters of the PID controller.

	K_P	K_I	K_D
Parameter	1664200	1248150	416050

4.2. Fuzzy logic controller

The Fuzzy Logic Controller (FLC) is categorized into three inputs which are body acceleration $\ddot{x}_{1,}$ body displacement x_1, body deflection displacement $x_2 - x_{1,}$ and one output which is desired actuator force u. The entire elements are used in an active suspension system. In FLC consist of three stages are fuzzification, fuzzy inference machine and defuzzification.

The main objective of fuzzy logic design with the controller is to reduce the problem and to improve the performance of the nonlinear quarter car active suspension system in term of performance ride comfort and road handling. The FLC controller will be introduced in the system as shown in Fig. 7. There are following steps to design the FLC controller:

i. Selecting the inputs (deflection velocity, body velocity and body acceleration) of the Fuzzy Logic.
ii. Output (force) will be selected.
iii. The range of each input and output will be set into the FLC controller.
iv. Use the formulating of the Fuzzy rules (IF-THEN)

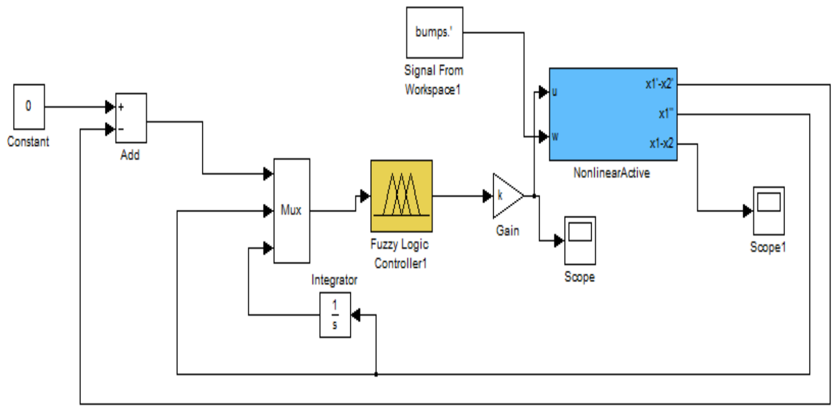


Fig. 7. Block diagram of nonlinear quarter-car active suspension with FLC controller.

Table 4. Input range and output range in membership functions.

Membership function		Range input	Range output
Input	Deflection velocity	[-0.2, 0.2]	[0, 1]
	Body velocity	[-1.5, 1.5]	[0, 1]
	Body acceleration	[-10, 10]	[0, 1]
Output	Force	[-1.5, 1.5]	[0, 1]

The input selected for the membership function are deflection velocity, body velocity and body acceleration whereas in the output selected for membership function is force. Each of the membership functions needed to be limit in the range. The input range and output range in the membership functions is tabulated as in Table 4 whereas in Table 5 shows the rule base of Fuzzy Logic model.

Table 5. Rules base of Fuzzy Logic model.

$\dot{x}_1 - \dot{x}_2$	$\dot{x}_1$	$\ddot{x}_1$	u		$\dot{x}_1 - \dot{x}_2$	$\dot{x}_1$	$\ddot{x}_1$	u
PM	PM	ZE	ZE		PM	PM	P or N	NS
PS	PM	ZE	NS		PS	PM	P or N	NM
ZE	PM	ZE	NM		ZE	PM	P or N	NB
NS	PM	ZE	NM		NS	PM	P or N	NB
NM	PM	ZE	NB		NM	PM	P or N	NV
PM	PS	ZE	ZE		PM	PS	P or N	NS
PS	PS	ZE	NS		PS	PS	P or N	NM
ZE	PS	ZE	NS		ZE	PS	P or N	NM
NS	PS	ZE	NM		NS	PS	P or N	NB
NM	PS	ZE	NM		NM	PS	P or N	NB
PM	ZE	ZE	PS		PM	ZE	P or N	PM
PS	ZE	ZE	ZE		PS	ZE	P or N	PS
ZE	ZE	ZE	ZE		ZE	ZE	P or N	ZE
NS	ZE	ZE	ZE		NS	ZE	P or N	NS
NM	ZE	ZE	NS		NM	ZE	P or N	NM
PM	NS	ZE	PM		PM	NS	P or N	PB
PS	NS	ZE	PM		PS	NS	P or N	PB

ZE	NS	ZE	PS	ZE	NS	P or N	PM
NS	NS	ZE	PS	NS	NS	P or N	PM
NM	NS	ZE	ZE	NM	NS	P or N	PS
PM	NM	ZE	PB	PM	NM	P or N	PV
PS	NM	ZE	PM	PS	NM	P or N	PB
ZE	NM	ZE	PM	ZE	NM	P or N	PB
NS	NM	ZE	PS	NS	NM	P or N	PB
NM	NM	ZE	ZE	NM	NM	P or N	PS

where NM is negative medium, NS is negative small, Z is zero, PS is positive small and PM is positive medium in the input linguistic variables and NV is negative very big, NB is negative big, NM is negative medium, NS is negative small, Z is zero, PS is positive small, PM is positive medium, PB is positive big and PV is positive very big in the output linguistic variables.

5. Results and Discussion

5.1. Body acceleration

Figures 8 and 9 show the comparison nonlinear active of body acceleration of sprung mass and unsprung mass. It can see that using the FLC controller can give better performance to ride comfort and road handling. From the sprung mass using a PID controller shows that the period of 6 seconds, the process of systematically decreasing oscillation amplitude is become to zero but for the FLC controller show that the period of 2.5 seconds where oscillation amplitude is become to zero. The reduction percentage also shows the performance of both controllers where for PID controller is about 96.80% reduction meanwhile the FLC controller is about 97.60%. Besides, unsprung mass using PID controller shows that the time taken for the system becomes zero is about 3 seconds and the process symmetrically decreasing oscillation amplitude to become zero after using FLC controller is 2.5 seconds. Therefore, the percentage reduction between PID controller and FLC controller are 98.05% and 99.50% respectively. The comparison of body acceleration between nonlinear passive and nonlinear active is tabulated in Table 6.

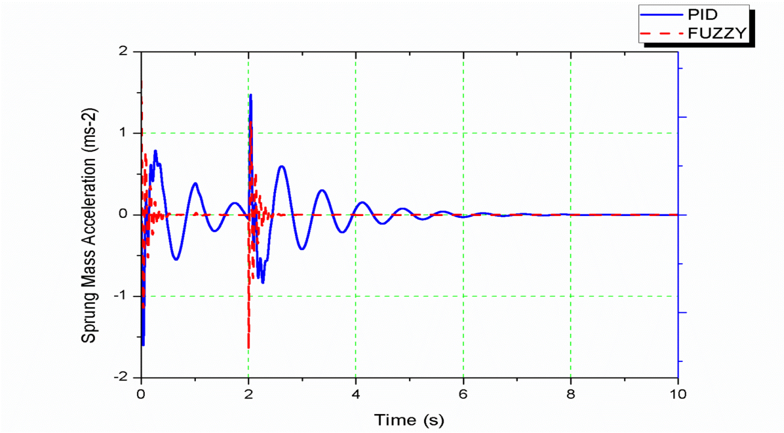


Fig. 8. Comparison of body acceleration of sprung mass.

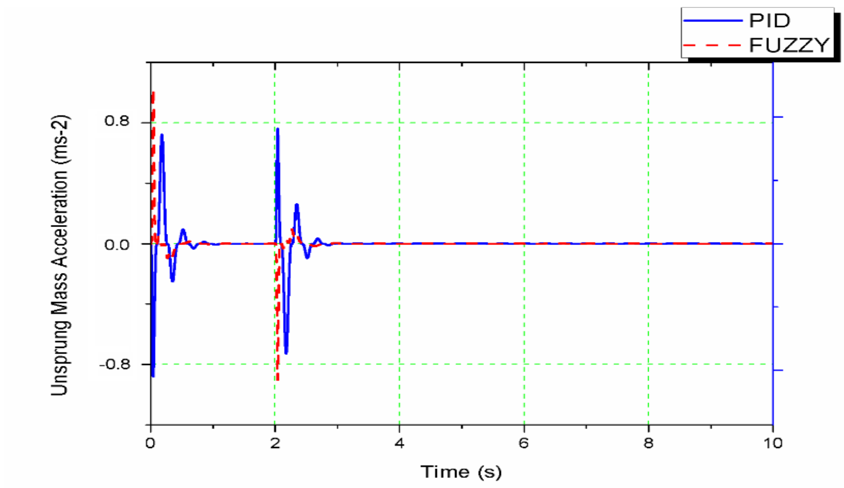


Fig. 9. Comparison of body acceleration of unsprung mass.

Table 6. Comparison in body acceleration between nonlinear passive and nonlinear active.

Elements			Sprung mass acceleration (ms^{-2})	Unsprung mass acceleration (ms^{-2})
Nonlinear Passive	Bump 1	Max	25.0	30.0
		Min	-16.0	-15.0
	Bump 2	Max	40.0	40.0
		Min	-20.0	-30.0
Nonlinear active with PID controller	Bump 1	Max	0.8	0.76
		Min	-0.5	-0.38
	Bump 2	Max	1.5	0.78
		Min	-0.8	-0.78
Nonlinear active with FLC controller	Bump 1	Max	0.6	1.10
		Min	-1.0	-0.10
	Bump 2	Max	1.0	0.20
		Min	-0.9	-0.90

5.2. Suspension travel

Figure 10 shows the comparison of suspension travel in the system. The result of the PID controller shows the oscillation from the system is worse performance than from the FLC controller with the percentage of reduction of PID controller is 80.5% and FLC controller is 97.5%. The FLC controller shows the best performance which the suspension travel becomes smoothness on the road. FLC controller only takes 2.5 seconds to become the system stable. The comparison of suspension travel between nonlinear passive and nonlinear active is tabulated in Table 7.

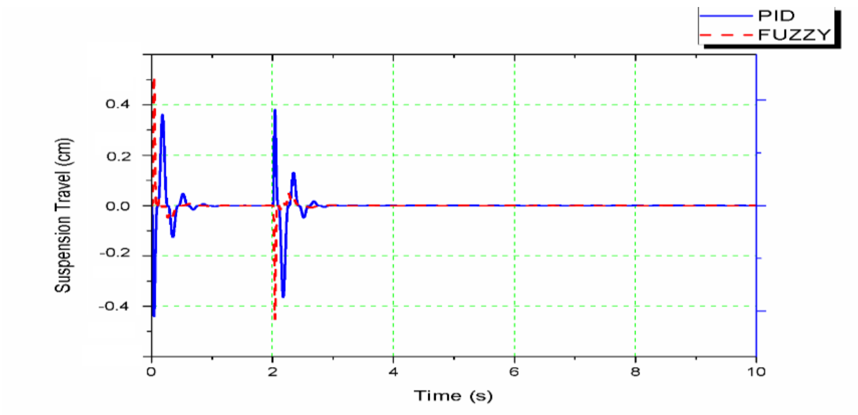


Fig. 10. Comparison of suspension travel.

Table 7. Comparison in suspension travel between nonlinear passive and nonlinear active.

Elements			Suspension travel (cm)
Nonlinear Passive	Bump 1	Max	4.8
		Min	-3.0
	Bump 2	Max	2.0
		Min	-1.0
Nonlinear active with PID controller	Bump 1	Max	0.38
		Min	-0.38
	Bump 2	Max	0.39
		Min	-0.38
Nonlinear active with FLC controller	Bump 1	Max	0.44
		Min	-0.05
	Bump 2	Max	0.05
		Min	-0.41

5.3. Wheel deflection

Figure 11 shows the comparison of wheel deflection of the system. This system is related to the road handling which oscillation from the system become well after reached 2.5 seconds using the FLC controller. A comparison using the PID controller shows the contact forces of the tires and the road surface with the reduction percentage is 96.0% less than when using the fuzzy logic controller with reduction percentage is 99.85%. The comparison of wheel deflection between nonlinear passive and nonlinear active is tabulated in Table 8.

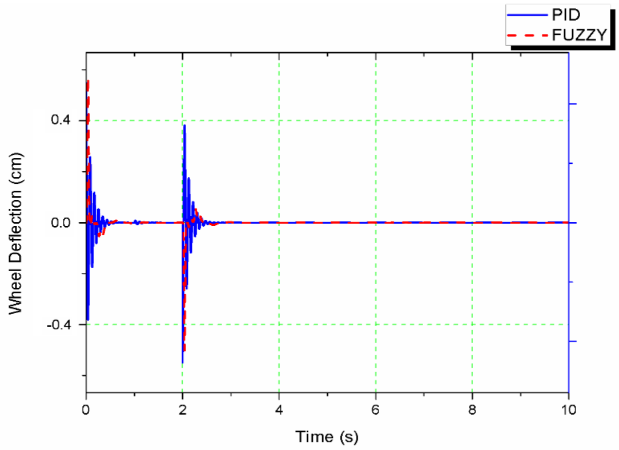


Fig. 11. Comparison of wheel deflection.

Table 8. Comparison in wheel deflection between nonlinear passive and nonlinear active.

Elements			Wheel deflection (cm)
Nonlinear passive	Bump 1	Max	35.0
		Min	-20.0
	Bump 2	Max	10.0
		Min	-4.0
Nonlinear active with PID controller	Bump 1	Max	0.25
		Min	-0.40
	Bump 2	Max	0.40
		Min	-0.30
Nonlinear active with FLC controller	Bump 1	Max	0.56
		Min	-0.10
	Bump 2	Max	0.14
		Min	-0.48

6. Conclusions

As conclusion, the PID controller design makes the system in a good performance because of the both maximum body acceleration, suspension travel and wheel deflection are decrease than the suspension model without a controller. Another controller that has been designed in the suspension system is Fuzzy logic. The FLC controller is designed to improve performance of the suspension system than using the PID controller. FLC controller is an intelligent controller that capable to give a better performance in term of body acceleration, suspension travel and wheel deflection. The input that related using the fuzzy set are deflection velocity, body acceleration and body velocity, while the output is a force that applied in the system. In order to make the suspension system more stable, 75 rules of FLC controller are used to get better performance.

Many control problems will be solved with several unique features from fuzzy logic. In this project, body displacement and body acceleration will be reduced effectively by developing a multi controller into the system. The fuzzy-PID will be designed using a Fuzzy Logic algorithm that capability to tune the PID parameters. As known, the Fuzzy-PID controller able to improve the better settling time in the system, reduce the percent of overshoot and give more stability in the system.

Acknowledgement

The authors would like to thank Centre for Research and Innovation Management (CRIM), University Teknikal Malaysia Melaka (UTeM) and Ministry of Education (MOE) for their support. The research was funded by Short Term Grant No. (PJP/2015/FKE(8C)/S01440) and Fundamental Research Grant Scheme (FRGS), Grant No. FRGS/1/2014/TK03/FKE/02/F00214.

References

1. Cao, D.; Song, X.; and Ahmadian. (2011). Road vehicle suspension design, *Dynamics and Control*, 49(1-2), 3-28.
2. Lin, J.S.; and Kanellakopoulos, I. (1997). Nonlinear design of active suspension. *IEEE Control System Magazine*, 17, 1-26.
3. Sam, Y.M.; Osman, J.H.S; and Ghani, M.R.A. (2002). A class of Proportional- integral sliding mode control with application to active suspension system. *System and Control Letters*, 51(3), 217-223.
4. Shinq-Jen, W.; Cheng-Tao, W.; and Tsu-Tian, L. (2005). Neural-network based optimal fuzzy control design for half-car active suspension system. *American Journal of Physics*, 10, 1-21.
5. Lin, Y.J.; Lu, Y.Q.; and Padovan, J. (1993). Fuzzy logic control of vehicle suspension systems. *International Journal of Vehicle Design*, 14(5), 457-470.
6. Raukich, S.; and Titli, A. (1993). Design of active and semi-active automotive suspension using fuzzy logic. Proceedings of the *IFAC Twelfth World Congress 2*. Sydney, Australia, 253-257.
7. Hung T.N.; and Michio S. (1998). *Fuzzy system modelling and control* (2nd ed.). US: Springer.
8. Dangor, M.; Dahunsi, O.A.; and Pedro, J.O. (2014). Evolutionary algorithm-based PID controller tuning for nonlinear quarter-car electro-hydraulic vehicle suspension. *Nonlinear Dynamics*, 78(4), 2795.
9. Rosheila, D. (2008). *Modelling and control of active suspension for a full car model*. Master Thesis, Universiti Teknologi Malaysia. Retrieved from http://eprints.utm.my/9645/1/RosheilaDarusMFKE2008.pdf
10. Gaspar, P.; Szaszi, I.; Bokor, J. (2003). Active suspension design using linear parameter varying control. *International Journal of Vehicle Autonomous Systems*, 1(2), 206-221.