African Journal of Advanced Pure and Applied Sciences (AJAPAS)

Online ISSN: 2957-644X

Volume 2, Issue 4, October - December 2023, Page No: 1-13

Website: *https://aaasjournals.com/index.php/ajapas/index*

||Arab Impact factor 2022: 0.87|| SJIFactor 2023: 5.689|| ISI 2022-2023: 0.557

Dynamic Analysis Evaluation and Control of A Full Vehicle Suspension System

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Received: July 22, 2023	Accepted: September 21, 2023	Published: October 01, 2023

Abstract:

In this paper, a full car active suspension system is to be developed as a mathematical model and introduce its kinematics. A state space representation is extracted and used as the basis for the Simulink model, through which the controllers are to be designed. This model focuses on the vertical motion of the vehicle's suspension system and a displacement pulse is used to represent a real-life road pump for the disturbance signal, with the goal is to attenuating and minimize its effect to produce the most likely smooth state trajectories with a short time to recover, thus a comfortable ride for the passengers is to be achieved. Two controllers are designed in this work splitting into two types, linear and non-linear control methods. A PID controller is the linear part and a Fuzzy PID controller is chosen as a nonlinear controller. Both of these controllers are to be introduced to the system, tuned and refined to get the best possible response, a comparison is then made to see each controller's advantages and benefits.

Keywords: Suspension System, Full Car Model, PID Control, Fuzzy PID Control.

Cite this article as: M. K. Akel, A. A. Ahmed, T. A. A. Galouz, A. S. Alatrash, “Dynamic Analysis Evaluation and Control of a Full Vehicle Suspension System,” *African Journal of Advanced Pure and Applied Sciences (AJAPAS)*, vol. 2, no. 4, pp. 1–13, October - December 2023.

تقييم وتحليل ديناميكي لنظام تعليق سيارة كامل

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الملخص

في هذا البحث سيتم تطوير نظام التعليق النشط للسيارة كنموذج رياضي وإدخال حركاته. يتم بعد ذلك استخراج تمثيل مساحة الحالة واستخدامه كأساس لنموذج Simulink، والذي من خلاله سيتم تصميم وحدات التحكم. يركز هذا النموذج على الحركة العمودية لنظام تعليق السيارة ويتم استخدام نبض الإزاحة لتمثيل مضخة طريق حقيقية لإشارة الاضطراب، والتي يعمل الهدف على تخفيفها وتقليل تأثيرها لإنتاج مسارات الحالة الأكثر احتمالية مع وقت قصير للتعافي، وبالتالي تحقيق رحلة مريحة للركاب. تم تصميم وحدتي تحكم في هذا المشروع مقسمتين إلى نوعين، طرق التحكم الخطية وغير الخطية. وحدة التحكم PID هي الجزء الخطي ويتم اختيار Fuzzy PID كوحدة تحكم غير خطية. سيتم تقديم كل منهما إلى النظام، وضبطهما وتحسينهما للحصول على أفضل استجابة ممكنة، ثم يتم إجراء مقارنة لمعرفة مزايا وفوائد كل وحدة تحكم.

الكلمات المفتاحية: نظام التعليق، نموذج سيارة كامل، PID Control، Fuzzy PID Control.

Introduction

For many years, suspension systems have been utilized extensively. They have existed from the earliest horse-drawn carriages to modern automobiles, and they are easily identifiable. Theoretically, a great suspension system ought to offer a smooth ride for the driver and decent handling for the passengers. In practice, it appears that one of these qualities must be sacrificed in favor of the other. For instance, a suspension system in a luxury sedan will give the passengers a smooth ride in exchange for poor road handling, whereas a suspension system in a sport car will give the passengers a challenging ride while maintaining the best possible road handling [1] .

There are two main groups of vehicle disturbances from the perspective of a system design. Road and load disturbances exist. Acceleration, deceleration, or just changing the vehicle's direction might result in a load disturbance. On the other side, a rough road or even hills might generate road disturbances. As a result, a good suspension system should be created to reduce the effects of both the load and the road disturbances [1].

A suspension system is one of the most important subsystems in the car; it acts as a bridge between the occupants of a vehicle and the road. It has two main functionalities, isolating the vehicle body and passengers from external disturbances coming from the road, it always relates to riding quality and maintaining a firm contact between the road and the tires to provide guidance along the track. This is called handling performance [2].

Active systems are a hot topic of research and study in recent years. During my research, I came across many papers that discuss active and passive suspension systems.

One of published papers talks about design PI and PID Controllers for controlling a suspension system of a quarter bus model. PID controller showed a faster and more accurate time response than the PI controller, which in turn showed a faster response than the conventional suspension system. Both controllers did not achieve the required performance in terms of accuracy in decision-making and the required degrees of displacement that can be applied in practice [4] .

In another paper, the active suspension system was manufactured based on PID control for a quarter car model. The PID controller showed good performance in terms of speed and acceleration and in terms of reducing displacement compared to the passive suspension system [5].

In [7], optimal vehicle and seat suspension design for quarter car and half car vehicle models to reduce human body vibrations. The application of (Proportional – Integral -Derivative) PID Controller is done to control the vibrations of suspension system. simulation result, it has been established that the proposed active suspension system proved to be more effective in controlling the vehicle oscillation and more robust in restoring the system to its steady state as compared to the passive suspension system.

Other work has been published a study titled "Active Suspension Control based on a Full-Vehicle Model". In this study, three types of controllers: state feedback, H2, and H∞ were designed using the Linear Matrix Inequality (LMI) technique. The results were compared with those of the open-loop system were using multiple types of disturbances as road input, the results showed that the vehicle is stabilizable even while taking into consideration road disturbances.

Chul Kim, Paul I. Ro published in 2002, a study titled "An Accurate Full Car Ride Model Using Model Reducing Techniques". In this study, an approach to obtain an accurate yet simple model for full-vehicle ride analysis is proposed. This study also provides an analytical method to investigate the effects of model complexity on model accuracy for vehicle suspension systems.

In 2018, P. Senthilkumar, K. Sivakumar, R. Kanagarajan, S. Kuberan published a study titled "Fuzzy Control of Active Suspension System using Full Car Model". In this paper proposed fuzzy controller results have demonstrated that the magnitudes of the body displacement and acceleration are decreased as well and the resonance peak due to vehicle body is eliminated significantly compared to model-based PID controller. The simulation of proposed fuzzy controller for active suspension system has confirmed improvement of the ride comfort in vehicles.

The study presents the design of a controller in the vehicle's suspension domain (a fuzzy PID controller), The goal for designing these controllers are to control heave motion of a vehicle and reject the effects of external disturbances on the performance of the system. In this paper, the focus is on the external disturbances that are in the form of a bump only, the proposed controller simulation was performed using the MATLAB SIMULINK environment and compared to the PID controller's performance.

This work is arranged as follows: Section 2 which is devoted to the mathematical model of the car suspension system in addition to the road profile used in this paper. Section 3 introduces the control strategy and gives a brief explanation about the PID and Fuzzy PID controller approach. Section 4 covers some simulations for comparing a closed-loop Simulink car suspension system using the two controllers, viewing response plots, results and discussion of this work and section 5 of this paper presents the conclusion of the performed work.

Mathematical Model

The model considered in the study has a seven-degree of freedom system, which includes the heave, roll and pitch of sprung mass and vertical displacement of four unsprung masses. only the actuating force is considered, and the actuator is not modelled explicitly. The system will consist of four unsprung masses (each wheel) connected to a single sprung mass, the bodywork. As shown in Figure 1, the unsprung mass will be free to move vertically relative to the sprung mass, while the sprung mass will be free to heave (vertical movement), pitch, and roll. Linear viscous dampers and spring elements will model suspension between sprung and unsprung mass, while tires are simply modelled as linear springs without shock absorbers. All pitch and roll angles will be considered small for simplicity and to obtain a linear model.

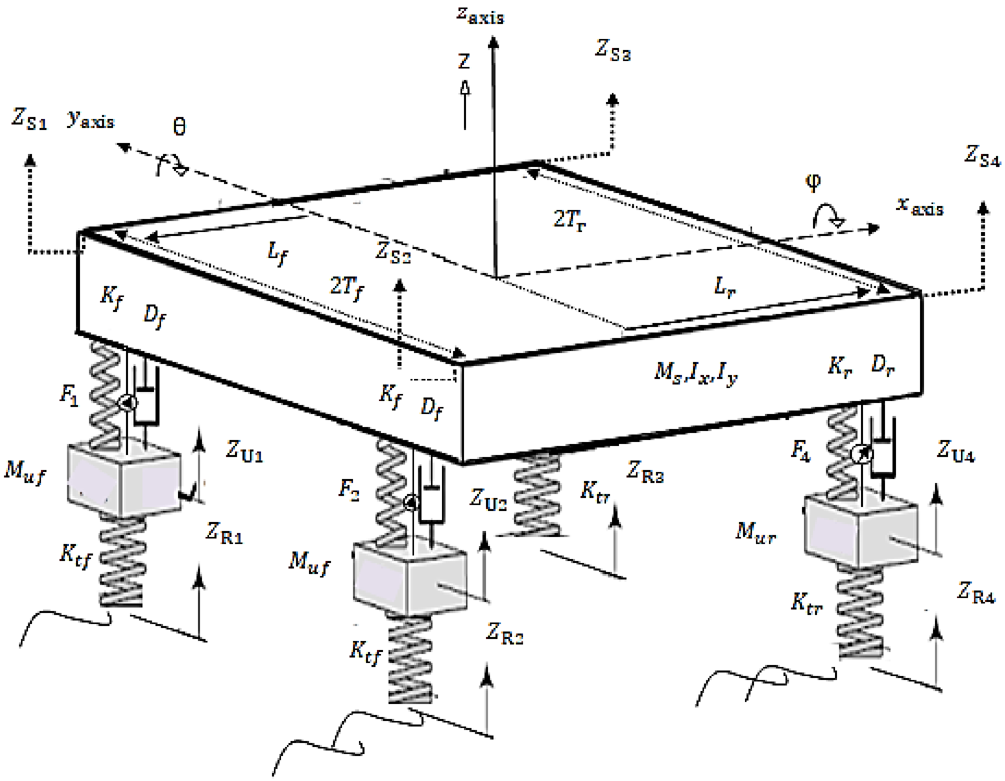

Figure 1 A full car model.

Using Newton's law, the kinematics of the system are derived describing the motion of the full car suspension system as:

Roll equilibrium of sprung mass:

$$I_x\ddot{\varphi} = -D_fT_f(\dot{Z}_{S1} - \dot{Z}_{U1}) + D_fT_f(\dot{Z}_{S2} - \dot{Z}_{U2}) - K_fT_f(Z_{S1} - Z_{U1}) + K_fT_f(Z_{S2} - Z_{U2}) - D_rT_r(\dot{Z}_{S3} - \dot{Z}_{U3}) + D_rT_r(\dot{Z}_{S4} - \dot{Z}_{U4}) - K_rT_r(Z_{S3} - Z_{U3}) + K_rT_r(Z_{S4} - Z_{U4}) + T_1 - T_2 + T_3 - T_4 \quad (1)$$

Pitch equilibrium of the sprung mass:

$$I_y\ddot{\theta} = -D_f L_f \sum_{i=1}^{2}(\dot{Z}_{Si} - \dot{Z}_{Ui}) - K_f L_f \sum_{i=1}^{2}(Z_{Si} - Z_{Ui}) + D_r L_r \sum_{i=3}^{4}(\dot{Z}_{Si} - \dot{Z}_{Ui}) + K_r L_r \sum_{i=3}^{4}(Z_{Si} - Z_{Ui}) + \sum_{i=1}^{2} T_i - \sum_{i=3}^{4} T_i \quad (2)$$

Vertical equilibrium of the sprung mass:

$$M_s \ddot{Z} = -D_f \sum_{i=1}^{2}(\dot{Z}_{Si} - \dot{Z}_{Ui}) - K_f \sum_{i=1}^{2}(Z_{Si} - Z_{Ui}) - D_r \sum_{i=3}^{4}(\dot{Z}_{Si} - \dot{Z}_{Ui}) - K_r \sum_{i=3}^{4}(Z_{Si} - Z_{Ui}) + \sum_{i=1}^{4} F_i \quad (3)$$

In the above equation, vertical displacement of car body at corner (Z_{S1}, Z_{S2} ,Z_{S3}, Z_{S4}) can be described using heave *z*, pitch angle, roll angle of vehicle body.

$$Z_{S1} = T_f \varphi + L_f \theta + Z$$

$$Z_{S2} = -T_f \varphi + L_f \theta + Z$$

$$Z_{S3} = T_r \varphi - L_r \theta + Z$$

$$Z_{S4} = -T_r \varphi - L_r \theta + Z \quad (4)$$

The following equation describes the vertical motion of the unsprung mass for both right and left front car wheel assemblies:

$$M_{uf}\ddot{Z}_{Ui} = -D_f(\dot{Z}_{Ui} - \dot{Z}_{Si}) - K_f(Z_{Ui} - Z_{Si}) - K_{tf}(Z_{Ui} - Z_{Ri}) \quad i = 1, 2 \quad (5)$$

The following equation describes the vertical motion of the unsprung mass for both right and left rear car wheel assemblies:

$$M_{ur}\ddot{Z}_{Ui} = -D_r(\dot{Z}_{Ui} - \dot{Z}_{Si}) - K_r(Z_{Ui} - Z_{Si}) - K_{tr}(Z_{Ui} - Z_{Ri}) \quad i = 3, 4 \quad (6)$$

Table 1 A full car model parameters.

Parameters	values	Parameters	values
M_s	1136 kg	L_f	1.15m
M_{uf}	63 kg	L_r	1.65m
M_{ur}	60 kg	K_f	36279 N/m
I_x	400 kg m^2	K_r	19620 N/m
I_y	2400 kg m^2	D_f	3924 N/m
T_f	0.505 m	D_r	2943 N/m
T_r	0.557 m	K_{tf} / K_{tr}	182470 N/m

The figure below shows the complete Simulink model for passive suspension system, and this model was built based on equations from 1 to 6.

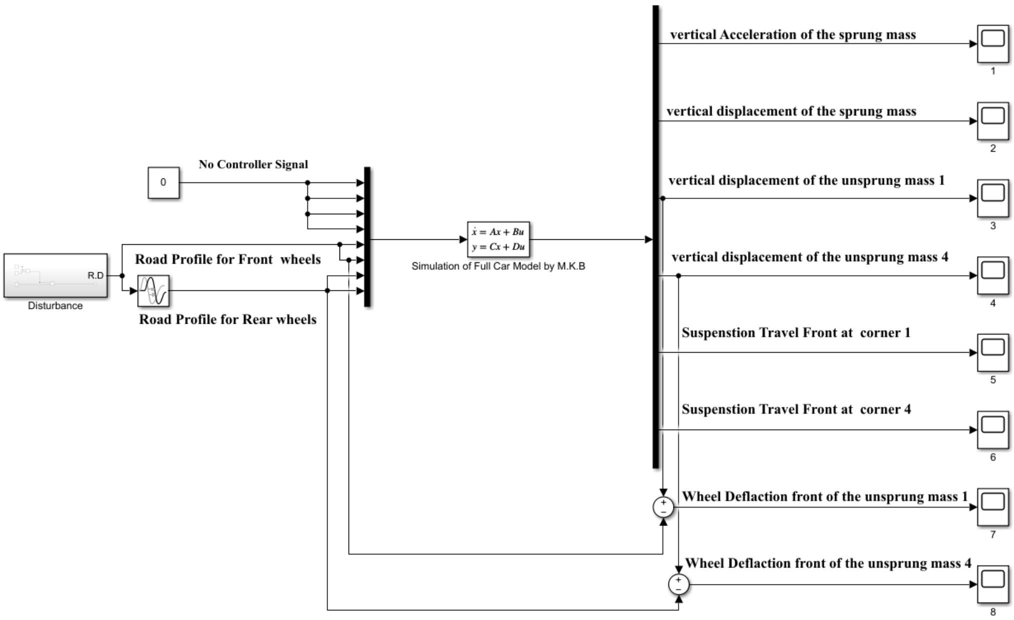

Figure 2 The Simulink model of passive suspension model.

Road Profile

The road profile is of paramount importance in the understanding of a vehicle's response to endogenous and exogenous perturbations or road excitations. Examples of such perturbations can be encountered when a vehicle meet bumps and potholes, the road profile is assumed to be a single bump is set at 10 cm. therefor the front side of the car receives this signal firstly, and after a time delay 3 s the rear side of the car receives this signal as show in below

$$w(t) = \begin{cases} 10\sin(6\pi t + 268) & 0.725 \le t \le 1.77 \\ 0 & otherwise \end{cases} \quad (7)$$

$$w(t) = \begin{cases} 10\sin(6\pi t + 268) & 3.725 \le t \le 4.77 \\ 0 & otherwise \end{cases} \quad (8)$$

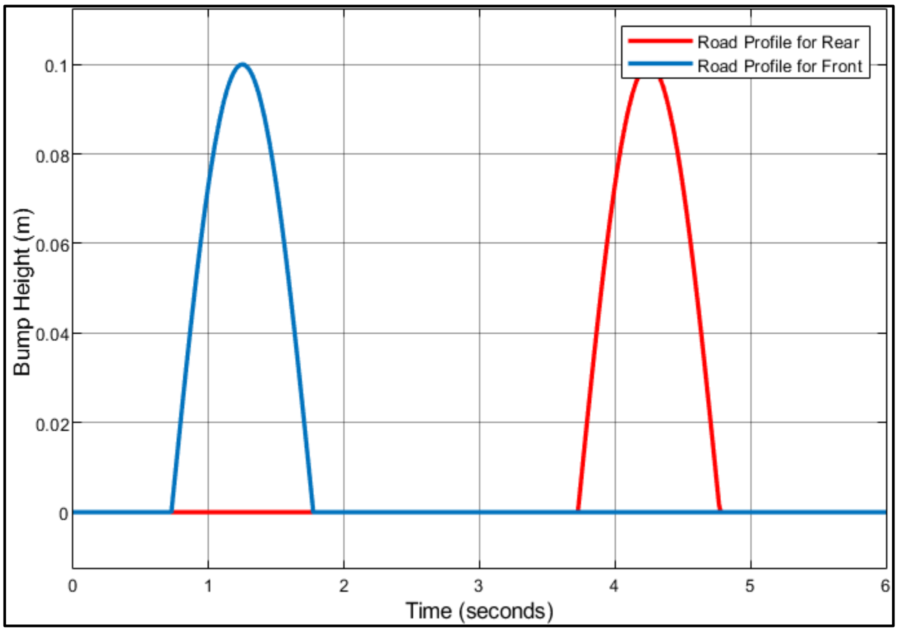

Figure 3 Bumpy Road input subsystem.

Control techniques
This part can be divided into two sections. The first section provides an overview of the two controllers used in this work, while the second section shows the Simulink model structures implemented for the active suspension system with PID controller and fuzzy PID controller.

Controller Design Using PID
In this paper, one classical PID controllers are designed to control four actuators controlling the suspension system in a full car model. The error signal is the which is the displacements of sprung mass 'Z'

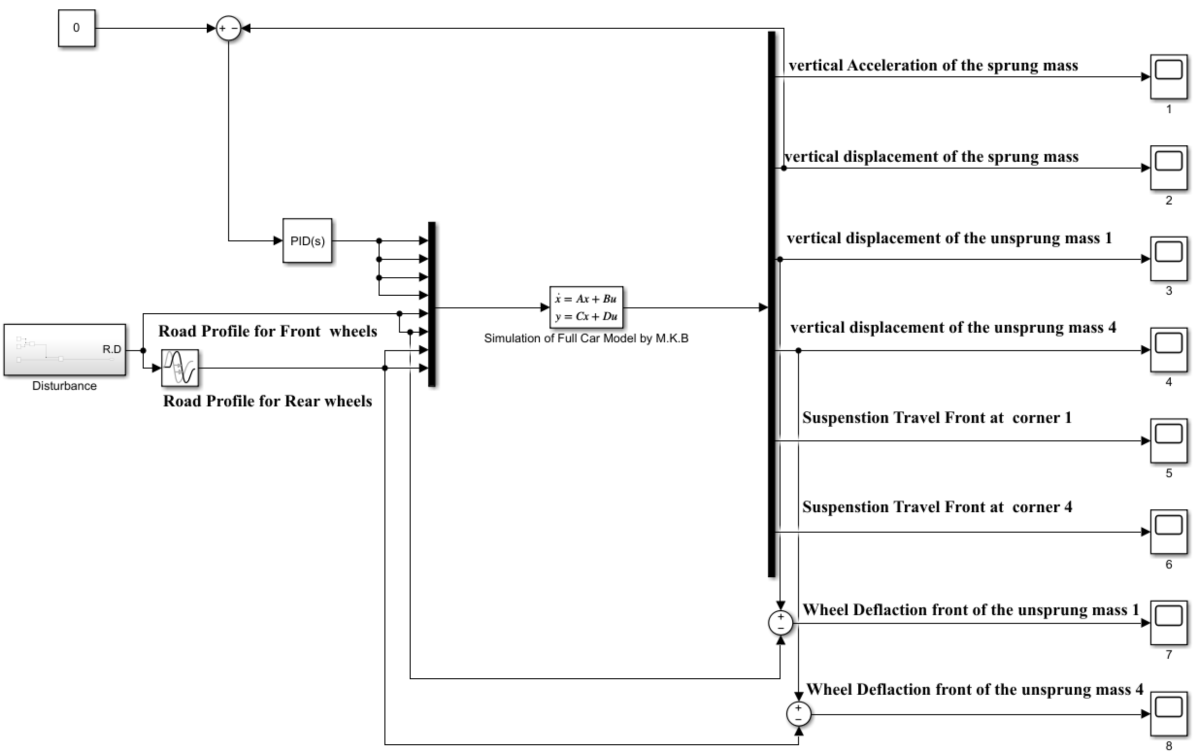

Figure 4 Simulation Full Car Model with PID Controller.

Using hand-tuning, to obtain the best possible response, the controller parameters can be determined

$$\boldsymbol{Kp} = 1000 \quad \boldsymbol{Ki} = 25 \quad \boldsymbol{Kd} = 300$$

3.2 Fuzzy PID Controller
In self-tuning fuzzy PID controller, the rules are designed based on characteristics of body car displacement "Z" and properties of the PID controller. The parameters of Kp, Ki and Kd must be calculated by using fuzzy tuner. The fuzzy tuner has two inputs: error (E) and change of error (EC), and three outputs: Kp, Ki and Kd.

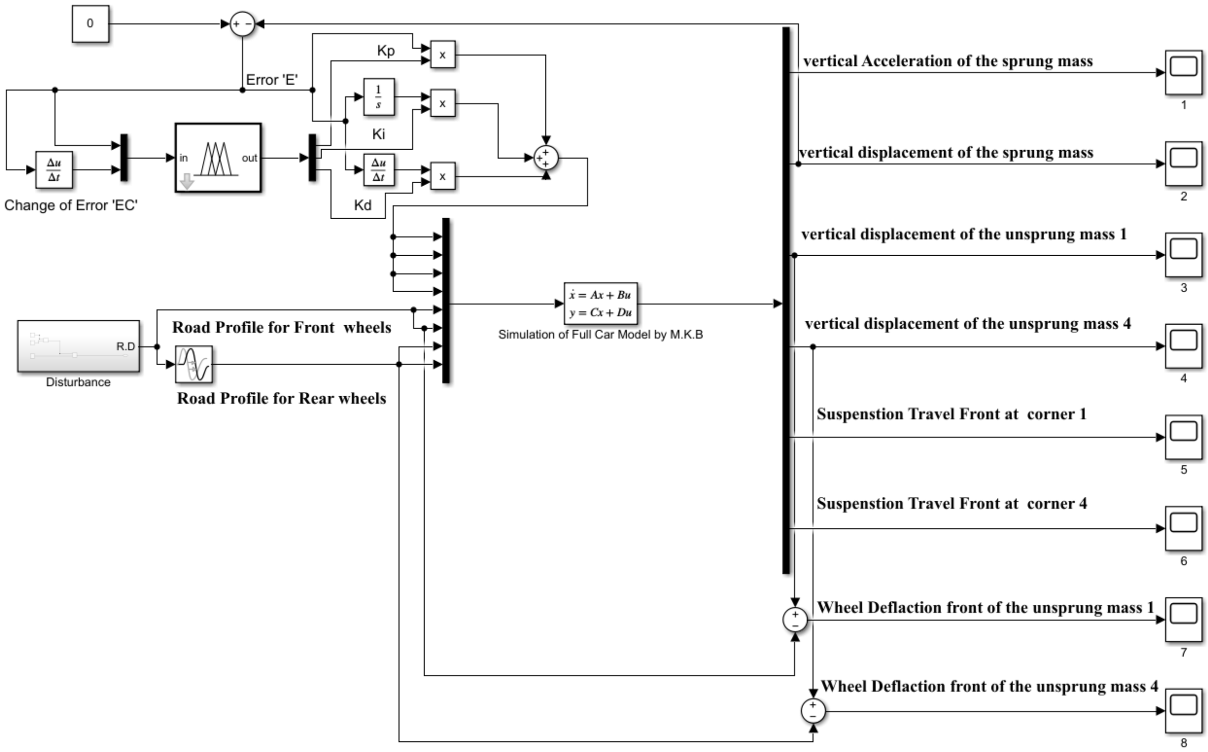

Figure 5 Fuzzy PID controller structure.

Then, these variables are quantized to five Fuzzy sets NB, NS, Z, PS and PB which respectively represent negative big, negative small, zero, Positive small, and positive big.

The universe of discourse of first input, error variable is chosen based on measuring the maximum and minimum values of the error signal 'E' and for the second input is chosen based on measuring the maximum and minimum values of the change of error signal 'EC' .The universe of discourse of the output variable Kp from 0 to 5000 and Ki from 0 to 50 and Kd from 0 to 800. the membership functions of these variables are shown in the following figures.

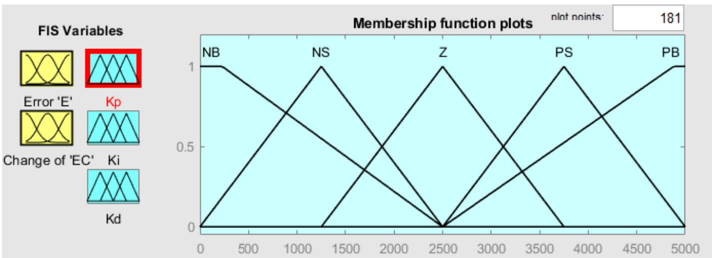

Figure 1 Membership Function of Proportional Gain Kp.

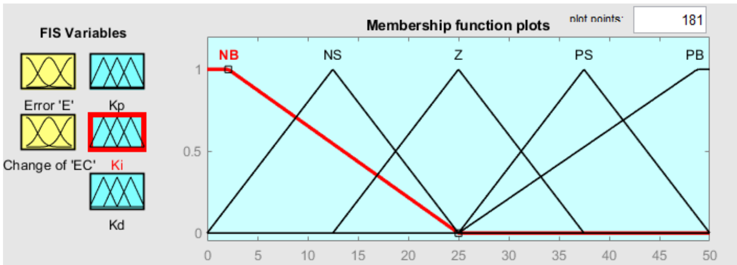

Figure 2 Membership Function of Integral Gain Ki.

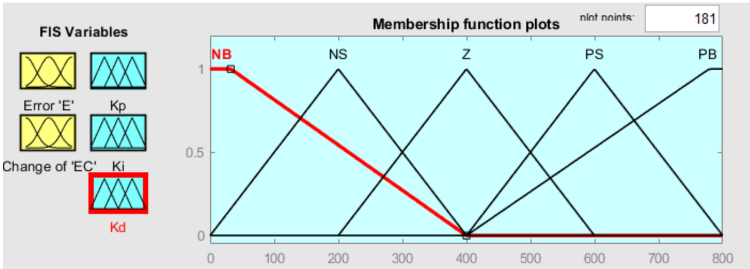

Figure 3 Membership Function of Derivative Gain Kd.

The most important step is establishment of Fuzzy inference rule between the input variables E, EC and the output variables K_p, K_i ,and K_d.based on the experience of experts or input output data . Table 2 shows the Fuzzy rules for controlling the car body displacement 'Z'. The total rules are 25 rules.

Table 2 fuzzy rules for controller.

EC \ E	NB	NS	Z	PS	PB
NB	NB	NB	NS	NS	Z
NS	NB	NS	NS	Z	PS
Z	NS	NS	Z	PS	PS
PS	NS	Z	PS	PS	PB
PB	Z	PS	PS	PB	PB

Simulation results and discussion

The results obtained are presented in Figures. A simulation of the mathematical model and controllers are made in MATLAB/Simulink R2022a environment. Control policy was evaluated for its performance at controlling the vertical displacement of the sprung mass according to a set of evaluation criteria acceleration of sprung mass, suspension travel for the front right and the rear left corners, Wheel Deflection for the front right and the rear left corners, Force Generate for PID/Fuzzy PID controller.

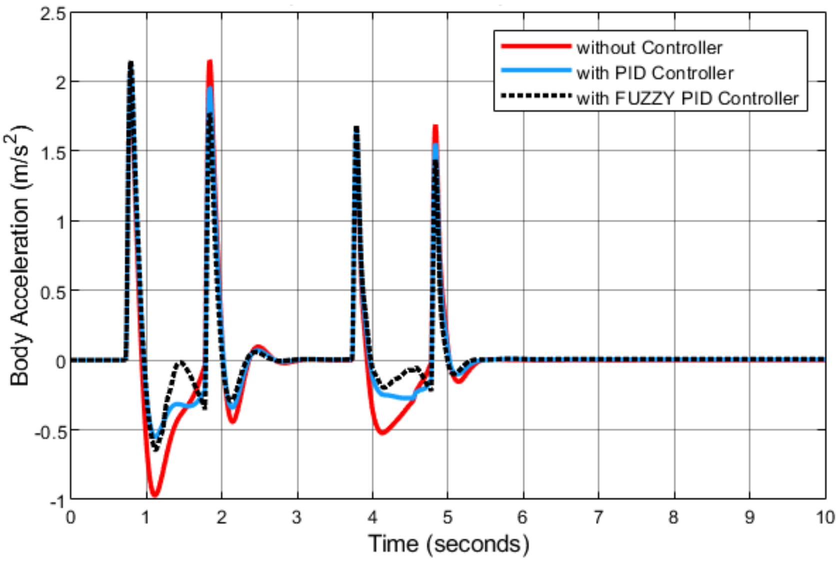

Figure 9 Body Acceleration.

The body acceleration response in Figure 9 shows that the Fuzzy PID control has the best performance outcome mainly on the overshoot, whilst the PID is not the best but it did some decent compensation over the system response to the disturbance input.

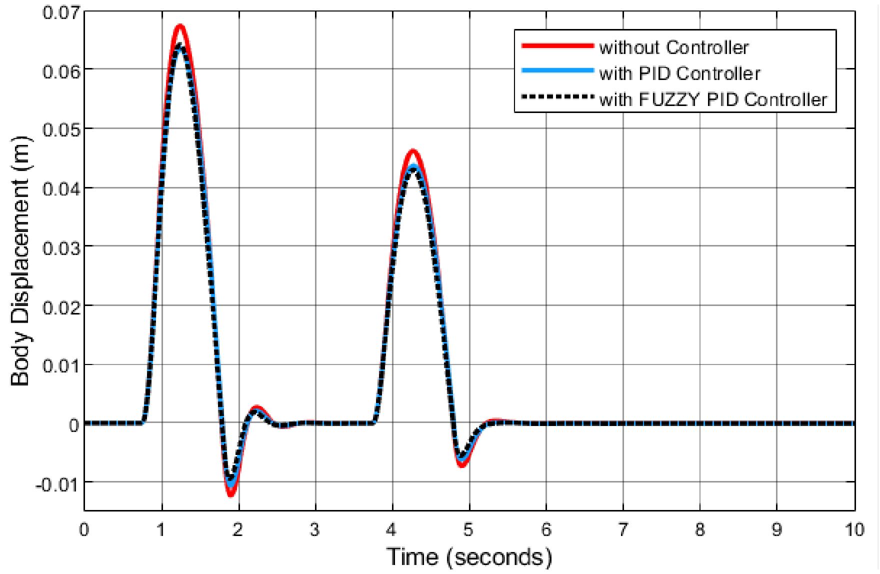

Figure 10 Body Displacement.

From the body displacement response, it is shown that both controllers reduce the disturbance effect nearly by the same amount.

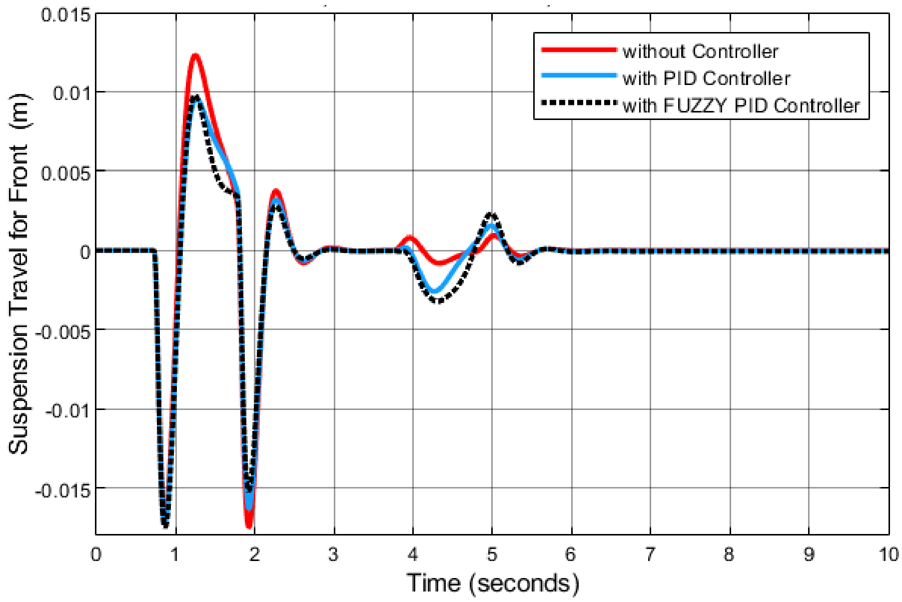

Figure 11 Suspension Travel Front Right and Left.

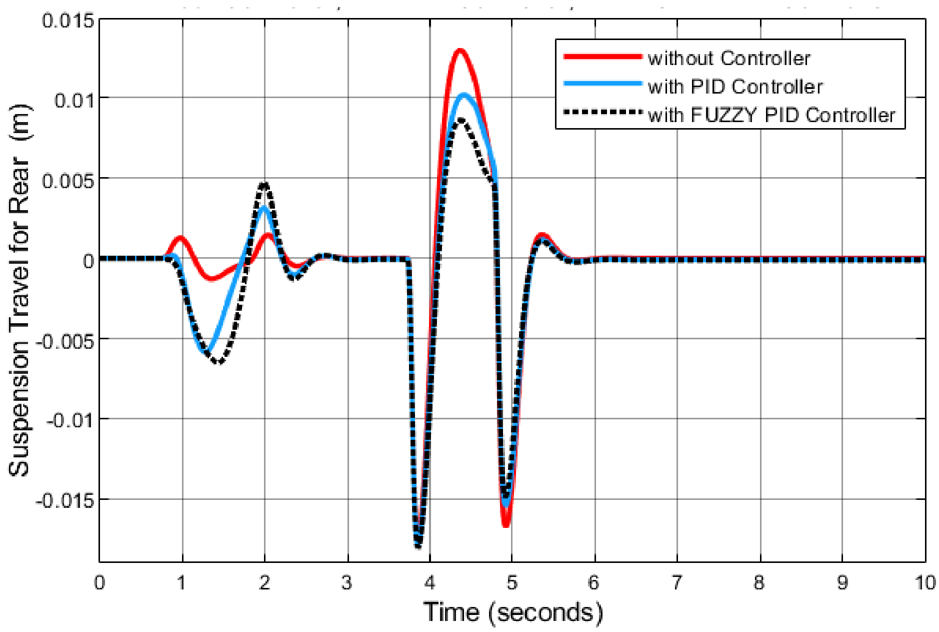

Figure 12 The velocity of unsprung mass.

In Figure 11, one could see that the response due to both controllers is good in the first three seconds for the first disturbance input, whilst for the rest of the response the controllers so negative effect on the response. That is due to the impact of the first disturbance input where it only affects the front wheels assembly, thus the controllers could only react to the first input and produce counter compensation, and the opposite is for Figure 12 where the controllers react only to the second disturbance input.

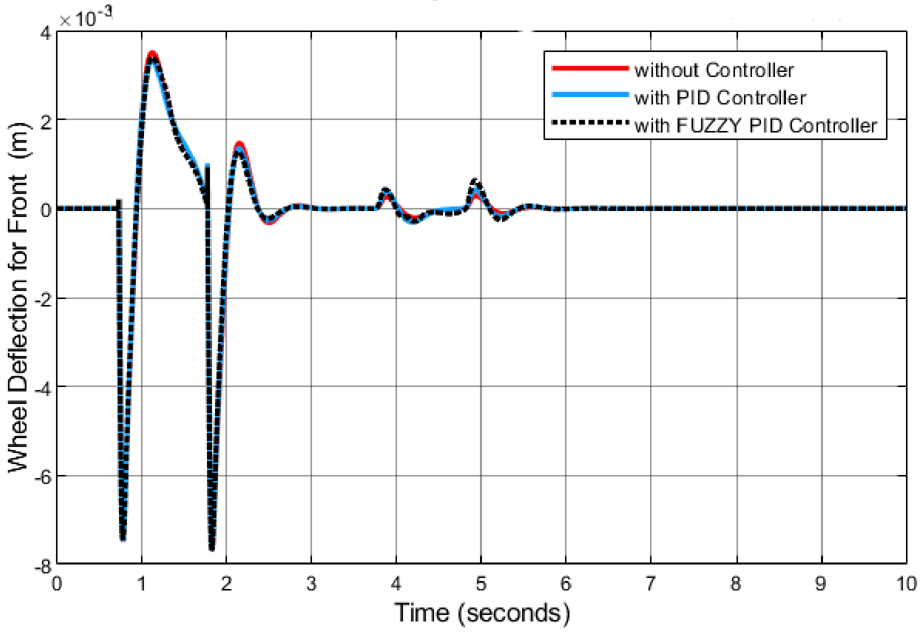

Figure 13 Wheel Deflection for Front Right and Left.

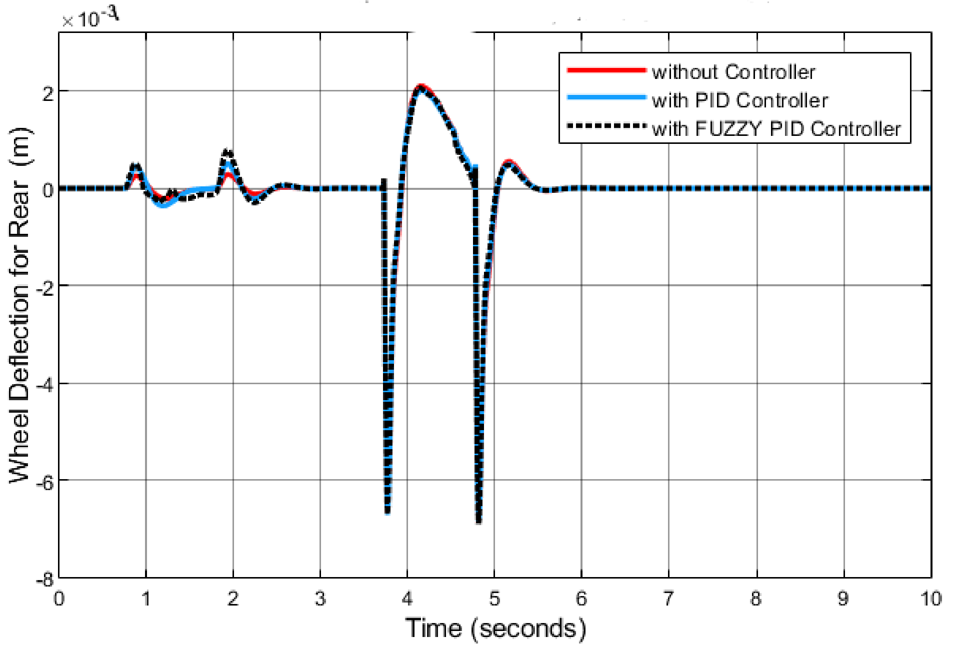

Figure 14 Wheel Deflection for Rear Right and Left.

Figure 13 and **Figure 14** represent the tires deflection for both front and rear assembly. It is clear that the PID and Fuzzy PID controllers behave the same and compensate for the disturbance inputs as they had done for suspension travel response for both, the front and rear assemblies by introducing decent effect for each assembly by each designated controller.

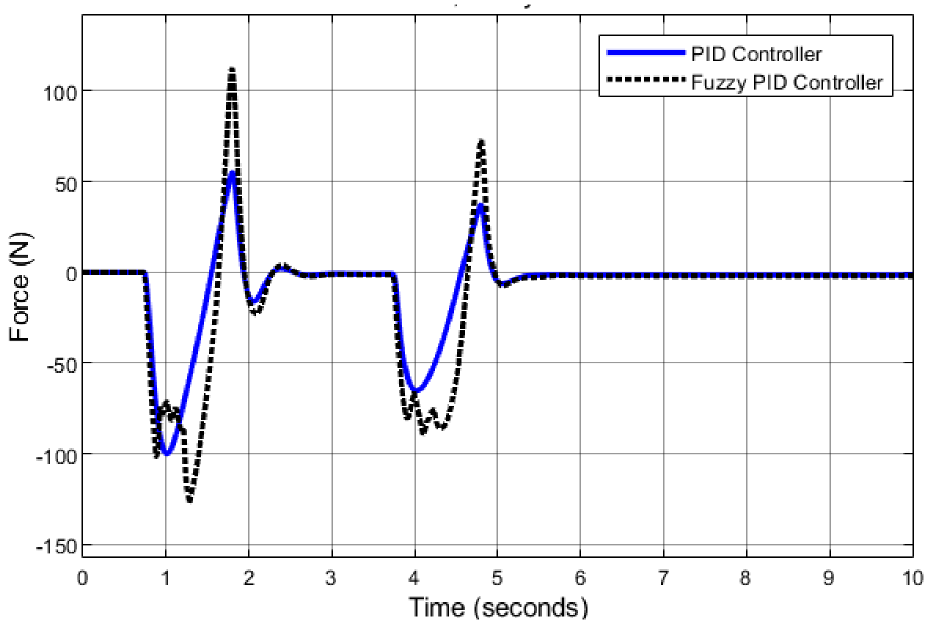

Figure 15 Force Generate for Actuators by Using Controllers**.**

Here the curves represent the force which each controller introduce to the system for both inputs. The force by physical means reflects the expense of resources each controller cost to use. The figure shows that the Fuzzy PID controller costs more than the PID controller does to overcome the disturbance inputs. It is not always the case, but in this system, more cost developed more compensation thus a better response and system behavior.

Conclusion

Automotive active suspension system was presented in this paper, a mathematical model of full car active suspension system was developed and introduced as the kinematics equations, then a state space representation was extracted to use as the backbone for the Simulink model which the controllers are designed through. This model is to focus on the vertical motion of the vehicle's suspension system, treating the roll and yaw angle changes as a very small ratio that could be neglected to simplify the process of control design.

Moreover, the vehicle's front pair and rear pair are considered identical during the control tests, where the right and left wheels of the front part are treated the same, and so for the rear half of the system. As for the disturbance signal introduced to the system, it is selected to be a displacement pulse mocking a real-life road pump, which to be attenuated and suppressed to provide the most possibly smooth state trajectories without having a long time to recover from the disturbance.

The control approach is chosen as a PID controller at first due to its advantages in simplicity of design and high compensation value is such application. The PID controller could be easily tuned manually via trial-and-error method to achieve a satisfying response while as mentioned, focusing on the vertical displacement which is the core state that has the highest impact over the passenger's comfort.

The PID controller is a linear system, but due to its robustness, it performed very good compensating the system response. The fact that this suspension system is clearly a nonlinear system drove the necessity to introduce an alternative controller. This alternative is to be the Fuzzy PID controller. This type of control greatly improves the performance over normal PID controller introducing the scheduling nonlinearity using the relations and memberships. The tuning method was performed manually to have a clarity of choices to set the PID controller gains, to drive the response as desired. As for the Fuzzy PID, the previously chosen gain was started with, resulting in very similar response, but with further tuning and refining, the Fuzzy PID gains are sat to achieve the best possible desire.

For the state responses, both controllers performed well, but the Fuzzy PID had a better compensation effect, due to its nonlinearity part where it offers more design choices being the three gains, the relations and the memberships which combined, with a solid ground for design and tuning can outperform the other controller and achieve design goals.

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