Theoretical and experimental fuzzy control on vehicle pneumatic semi-active suspension system

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Abstract: Semi-active suspension system is a possible way to improve suspension performance although the passive system can effectively handle some control of suspension system. The main propose of this paper is to assess performance of semi-active suspension system by implementing Fuzzy and Proportional-Integral-Derivative (PID) controls in comparison with passive suspension system. This paper is focused on reliable fuzzy controller design for pneumatic semi-active suspension system. The pneumatic semi-active suspension system consists of a damper with two state variable damping coefficient parallel with coil spring. The performance of pneumatic semi-active suspension system theoretically and experimentally predicted using two degree of freedom of quarter car model are presented. Also a method of PID and fuzzy controls were applied theoretically and experimental to show the performance of ride comfort. The results are generated when the system is excited by sine wave road input. The damping coefficient of the pneumatic semi-active damper is taken firstly from the experimental on the damper characteristics test rig, after that, the performance was studied using the quarter car test rig. Finally, a comparison between the passive and semi-active systems are presented and discussed. The results showed that there is a worthwhile improvement for the pneumatic semi-active suspension system with fuzzy control over the passive.
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Keywords: Quarter car model, pneumatic suspension system, fuzzy control, PID control.

Introduction

The design of ground vehicle suspension system is influenced by a number of factors. These factors, on the other hand, a good vibration isolation is required to secure the occupants' ride comfort, whereas good road holding is important for vehicle handling and, in general, enhanced safety Key design constraints are represented by maximum allowable relative displacement between the vehicle body and various suspension components, including wheels, trucks and other unsprung masses**[1,2&3]**. Additional constraints are imposed by the overall system robustness, reliability and cost requirements. Different suspensions satisfy the above requirements to differing degrees. Although significant improvements can result from a designer's ingenuity, on the average, suspension performance mainly depends on the type or class of suspension used. Here one distinguishes, in an ascending order of improved performances between passive, semi-active, and fully active suspensions. Passive suspensions can be found on most of the conventional vehicles. Roughly they are characterized by the absence of external power sources, whereas the active suspensions require additional energy sources, such as compressors or pumps, to achieve superior ride and/or handling performance**[4,5&6]**. Suppression of vibration in passive suspensions depends on the spring stiffness, damping Suppression of vibration in passive suspensions depends on the spring stiffness, damping coefficient, and car mass. Due to the fact that they cannot satisfy the comfort requirement under different road conditions, significant interest is being devoted to the control of active and semi- active suspension in both academia and industry. Many analytical and experimental studies on active and semi-active suspensions have been performed to improve ride quality and handling performance. The results of studies show that active and semi-active suspensions can provide substantial performance improvements over passive suspensions in general **[7]**. The design of controlled suspension systems for road vehicles aims to optimize the performance of the vehicle with regard to comfort and road handling. Vehicle suspensions should serve several conflicting purposes. In addition to counteracting the body forces resulting from cornering, acceleration or braking and changes in payload, suspensions must isolate the passenger compartment from road irregularities. For driving safety, a permanent contact between the tyres and the road should be assured. Passive suspension systems built of springs and dampers have serious limitations. Their parameters have to be chosen to achieve a certain level of compromise between road holding, load carrying and comfort, under wide variety of road conditions. Specific vehicle models need to be used in order to analyse the effectiveness of the active suspension system on vehicle dynamics **[8, 9&10]**. Passenger cars are complex multibody systems consisting of many rigid and deformable components. A full vehicle model needs to present the nonlinear kinematics of wheels and axles, the effects of suspension geometry and has to include the drive train, the steering mechanism and the tyre dynamics, resulting in a high number of degrees of freedom. Since it makes no sense to try to build a universal vehicle model that can be used to solve all dynamic problems, reduced dynamic models for specific investigation purposes are often designed instead **[11&12]**. In this work the pneumatic semi-active suspension system is considered. Firstly, the damping coefficients of the damper were measured experimentally. Also the ride performance was determined experimentally on the quarter car test rig in the vehicle dynamic lab in the department of automotive and tractor department, faculty of engineering, Minia University. This system in which the hydro-pneumatic damper is mounted in parallel with a conventional coil spring. Finally, a theoretical predicted analysis of this system is based on quarter car model subjected to sine wave road input and realistic damping coefficient values are presented and compared with the experimental results.

Pneumatic semi-active damper

The pneumatic semi-active damper is a damper works with the pressured air and controlled electrically as shown in fig. 1. The changing of the pressure of air changes the damping coefficient of the damper. Also it can be considered a variable damping coefficient damper (semi- active damper). Fig. 2 shows the pneumatic active damper in the damper characteristics test rig.

Fig. 1 the semi-active pneumatic damper

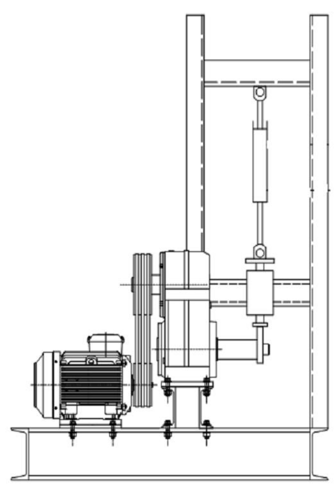

Fig.2 The characteristics test rig with pneumatic semi-active damper.

The characteristics of the pneumatic semi-active damper

The pneumatic semi-active damper characteristics were measured at a test rig is designed and developed in the vehicle dynamic laboratory. It is used to measure the non linear damper characteristics. The characteristics were measured at different air pressures, after that the force velocity curve was plotted and the characteristics of the pneumatic semi-active damper were extracted **[13]**. The cases are showed below in Figs. (3, 4 and 5) and Table 1.

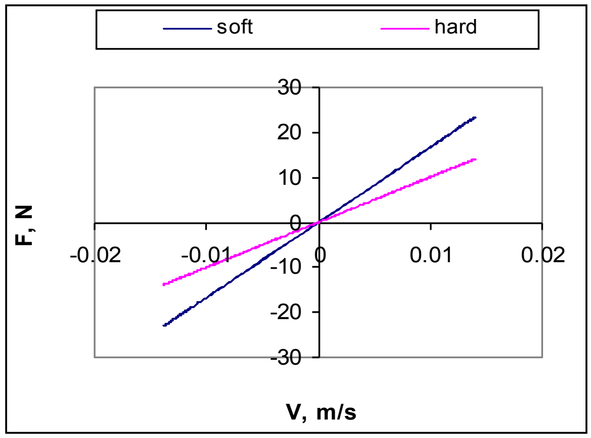

Fig. 3 Force versus velocity at 3 bar air pressure and 0.4 Hz

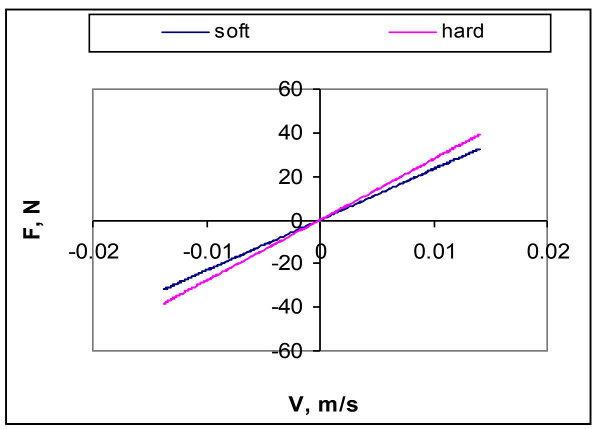

Fig. 4 Force versus velocity at 4 bar air pressure and 0.4 Hz

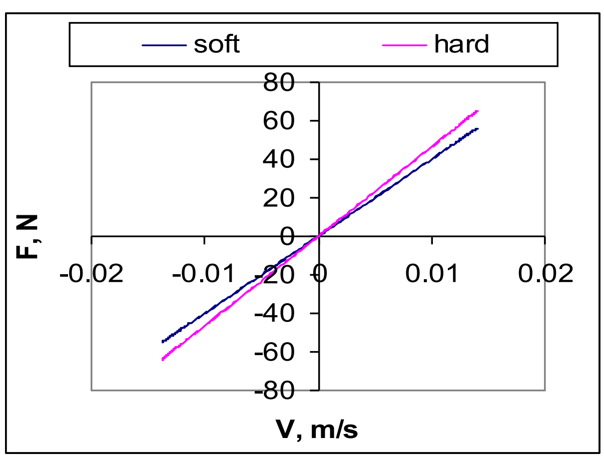

Fig. 5 Force versus velocity at 5 bar air pressure and 0.4 Hz

Table 1: The effect of air pressure on the damping coefficient.

Air pressure(bar)	Damping coefficient(N.s/m)	
	Soft	**Hard**
5	**4000**	**4640**
4	**2320**	**2784**
3	**1002**	**1670**

Quarter car model

In this paper a quarter vehicle model is discussed, and the passive suspension system is shown in fig. 6. Here the body mass, m_b, wheel mass, m_w, the stiffness of tyre, k_{tyre}, suspension stiffness, k_s, suspension damping coefficient, c_s. the equation of motion are:

$$M_w \ddot{X}_w = K_{tyre}(X_0 - X_w) - K_s(X_w - X_b) - C_s(\dot{X}_w - \dot{X}_b) \quad 1$$

$$M_b \ddot{X}_b = K_s(X_w - X_b) + C_s(\dot{X}_w - \dot{X}_b) \quad 2$$

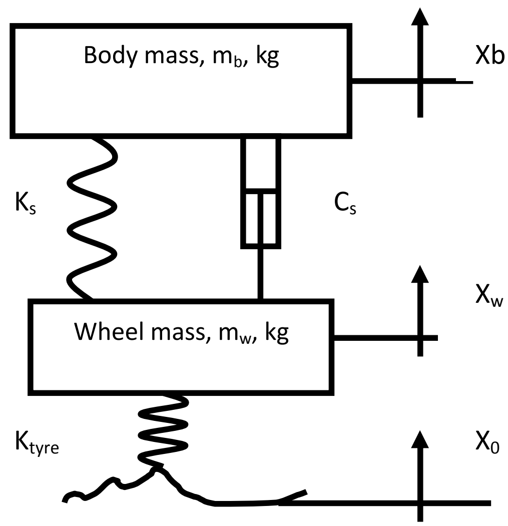

Fig. 6: Passive suspension system based on quarter car model

To model the road input let us assume that the vehicle is moving with 20 m/s constant speed. To transform the motion equations of the quarter car model into a space state model, the following state variables are considered:

$\underline{X} = [x_1, x_2, x_3, x_4]^T$ (3)

Where $x_1 = (x_b - x_w)$ is the body displacement. $x_2 = (x_w - x_0)$ is the wheel displacement, $x_3 = \dot{x}_b$ is the absolute velocity of the body, and $x_4 = \dot{x}_w$ is the absolute velocity of the wheel.

Then the motion equations of the quarter car model for the active suspension can be written in state space form as follows: (4)

$$\underline{\dot{x}} = A.\underline{x} + B.\underline{u} + F.\underline{\dot{z}_r}$$

Where:

$$A = \begin{bmatrix} 0 & 0 & 1 & -1 \\ 0 & 0 & 0 & 1 \\ -\frac{ks}{m_b} & 0 & -\frac{c_s}{m_b} & \frac{c_s}{m_b} \\ \frac{ks}{m_w} & -\frac{k_t}{m_w} & \frac{c_s}{m_w} & -\frac{c_s}{m_w} \end{bmatrix} \quad (5)$$

$$B = \begin{bmatrix} 0 \\ 0 \\ \frac{1}{m_b} \\ -\frac{1}{m_w} \end{bmatrix} \quad (6) \qquad F = \begin{bmatrix} 0 \\ -1 \\ 0 \\ 0 \end{bmatrix} \quad (7)$$

Where u is the control force, x_0 is the road input displacement.

PID Controller

A proportional–integral–derivative controller (PID controller) is a generic control loop feedback widely used in industrial control systems. A PID is the most commonly used feedback controller. A PID controller calculates an "error " value as the difference between a measured process variable and a desired step point. The controller attempts to minimize the error by adjusting the process control inputs . In the absence of knowledge of the underlying process, PID controllers are the best controllers. However, for best performance, the PID parameters used in the calculation must be tuned according to the nature of the system – while the design is generic, the parameters depend on the specific system The PID controller calculation involves three separate parameters, and is accordingly sometimes called three-term control: the proportional, the integral and derivative values, denoted P, I, and D. The proportional value determines the reaction to the current error, the integral value determines the reaction based on the sum of recent errors, and the derivative value determines the reaction based on the rate at which the error has been changing.

Fuzzy Controller

Negative Medium, Negative Small, Zero, Positive Small and Positive Medium are used to represent the fuzzy value of the controller input and output as linguistic variables. Design of a fuzzy logic controller has following steps:

(1) Selecting suitable linguistic variables using the vehicle model parameters (fuzzification).

(2) To set-up suitable linguistic control rule base.

(3) To design a suitable control algorithm that bases on the measurable system variables

(4) Choosing a method for defuzzification. An interface to scale and map the linguistic control actions inferred to yield a non-fuzzy control input to the vehicle being controlled (Defuzzification). The fuzzy logic controller used in the active suspension has two inputs: body acceleration $\ddot{x}_b$, body velocity $\dot{x}_b$, and one output: desired actuator force u. The control system itself consists of three stages: fuzzification, fuzzy inference machine and defuzzification. The fuzzification stage converts real-number (crisp) input values into fuzzy values while the fuzzy inference machine processes the input data and computes the controller outputs in cope with the rule base and data base. These outputs, which are fuzzy values, are converted into real-numbers by the defuzzification stage.

Results and Discussion

The quarter car suspensions for passive and semi active with PID and Fuzzy controls were simulated using MATLAB/SIMULINK **[14]**. Figs (a1, a2, a3 and a4) in appendix A show the simulation blocks diagrams. The theoretical results were made at three models passive, semi-active suspension system fuzzy controller and semi-active suspension system PID controller. The quarter car model parameters are shown in the table 2. The simulation results of two types of controllers PID and fuzzy control and passive suspension systems are compared for sin wave road conditions. Fig. 7 shows comparison between the body acceleration, suspension working space and dynamic tyre load for passive, semi-active with PID and semi active with Fuzzy for in terms of time domain showing good results in ride comfort for semi-active fuzzy control. It can clear that there is an improvement in ride comfort performance for semi-active with fuzzy control over passive and semi-active with PID control. Fig. 8 shows the quarter car test rig with semi-active damper suspension system.

Table. 2 Quarter car suspension system components parameter.

Parameters	Value
Sprung mass, Kg	250
Unsprung mass, Kg	50
Tyre spring stiffness, kN/m	126
Coil spring stiffness, kN/m	17

Semi-Active Suspension System Result

Setting up the computer programme which makes it ready to receive the signals from the quarter car test rig. The output signals from the transducers were fed into the data acquisition system. Then the data is transferred into digital by ADC card and the data was analyzed into in the form of time domain. The sinusoidal input to the tyre spring was provided by a roller follower which excitation the quarter car system. Figs. 9 and 10 present the semi active with PID and fuzzy controls suspension systems in terms of time domain which recorded from the quarter car test rig. Fig. (11) Compares between the body acceleration, suspension working space and dynamic tyre load for semi active with PID and Fuzzy controls over the passive suspension system of theoretical and experimental results. It can clear that there is an improvement in ride comfort performance over passive. This indicates that the semi-active with PID and fuzzy controls suspension systems are better than the passive. A comparison of experimental and theoretical of body acceleration, suspension working space and dynamic tyre load for the semi active suspension system are presented. It can be seen that the experimental trend is in agreement with the theoretical results. There are small different between the experimental and theoretical results. The theoretical and experimental results were in good agreement

Conclusions

In this paper, the semi-active suspension system with PID and fuzzy controls is studied to achieve optimum ride with low energy dissipation. The results are summarized as follows:

1. A significant improvement in ride performance can be obtained using a semi-active with PID and fuzzy controls suspension systems compared with a passive suspension system.
2. An experimental version of the semi-active with PID and fuzzy controls suspension system has been designed and test on a quarter car test rig. The results showed improvements relative to a passive system in body acceleration and in dynamic tyre load for the semi active with fuzzy control suspension system.
3. Theoretical results were always optimistic relative to those obtained experimental criteria were lower than those obtained in the tests. The theoretical and experimental results were in good agreement.

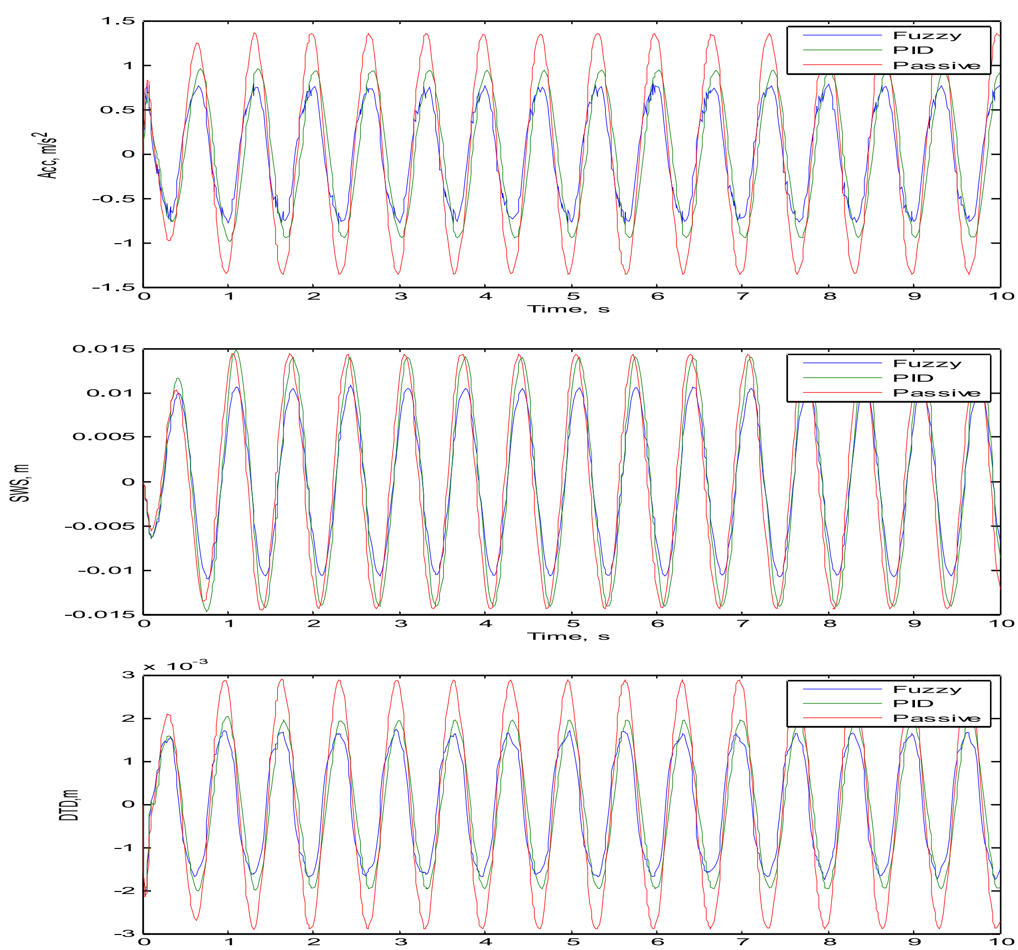

Fig. 7: body acceleration, suspension working space and dynamic tyre load for passive, semi-active with PID and semi active with Fuzzy in time domain.

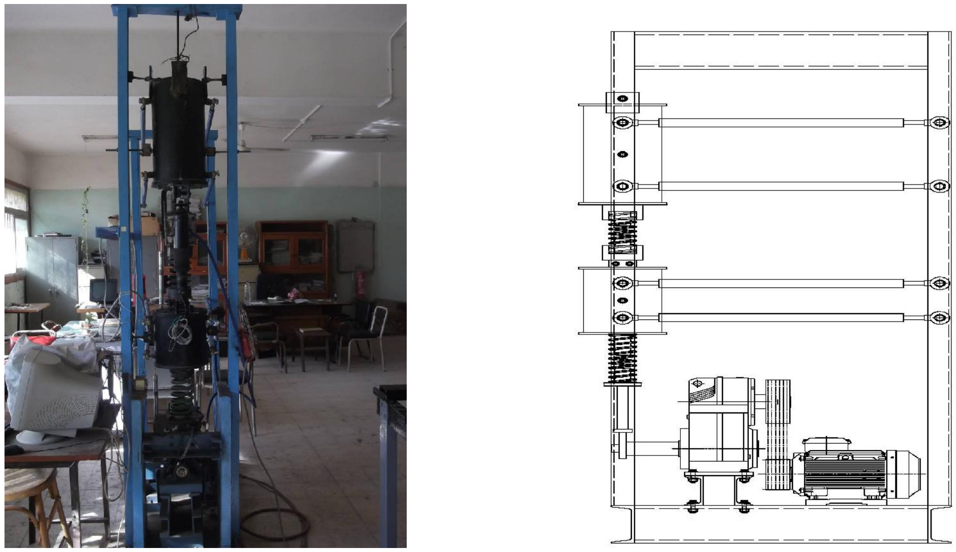

Fig.8: Quarter car test rig with pneumatic semi-active damper.

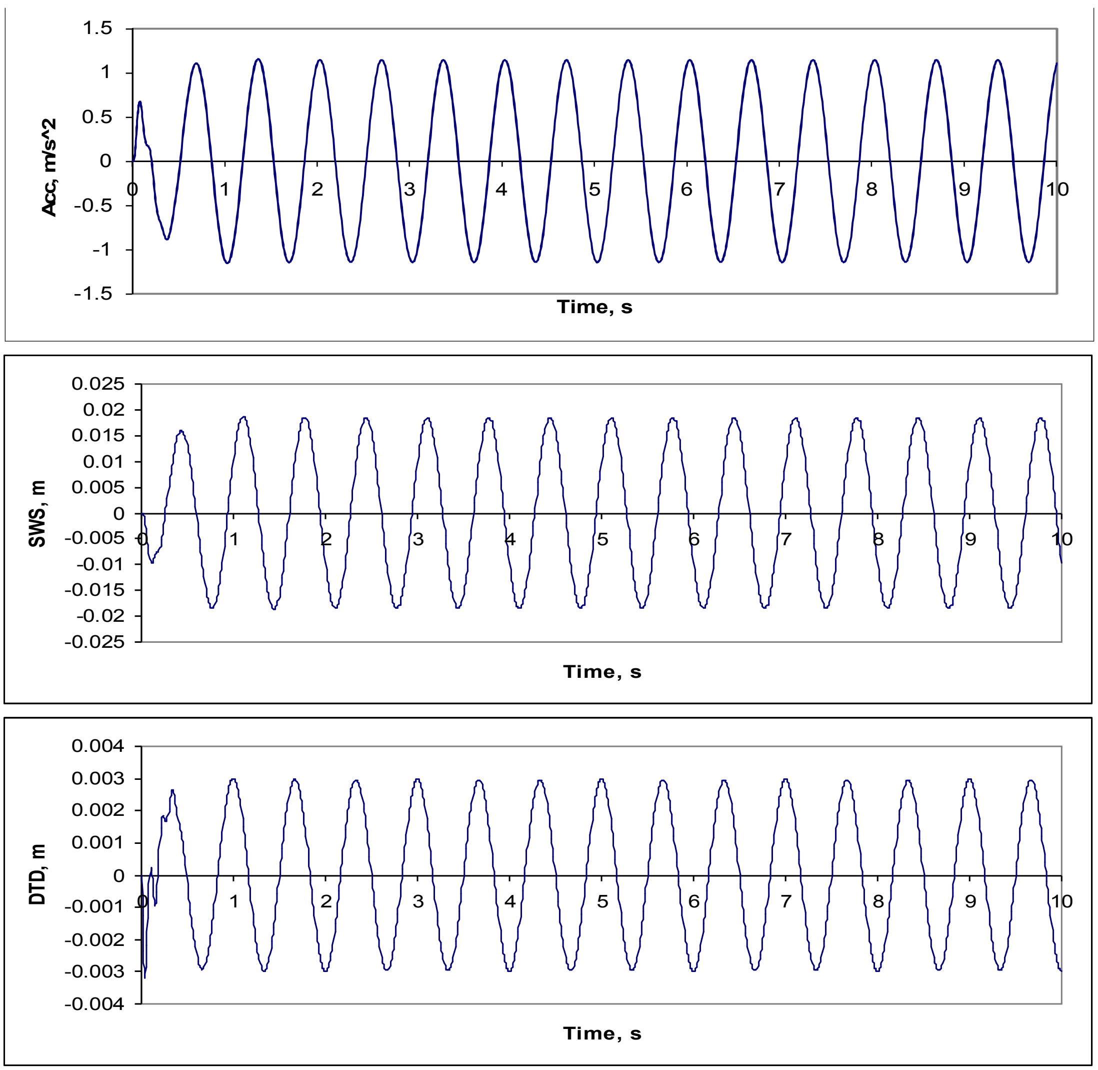

Fig. 9: body acceleration, suspension working space and dynamic tyre load for PID control in time domain.

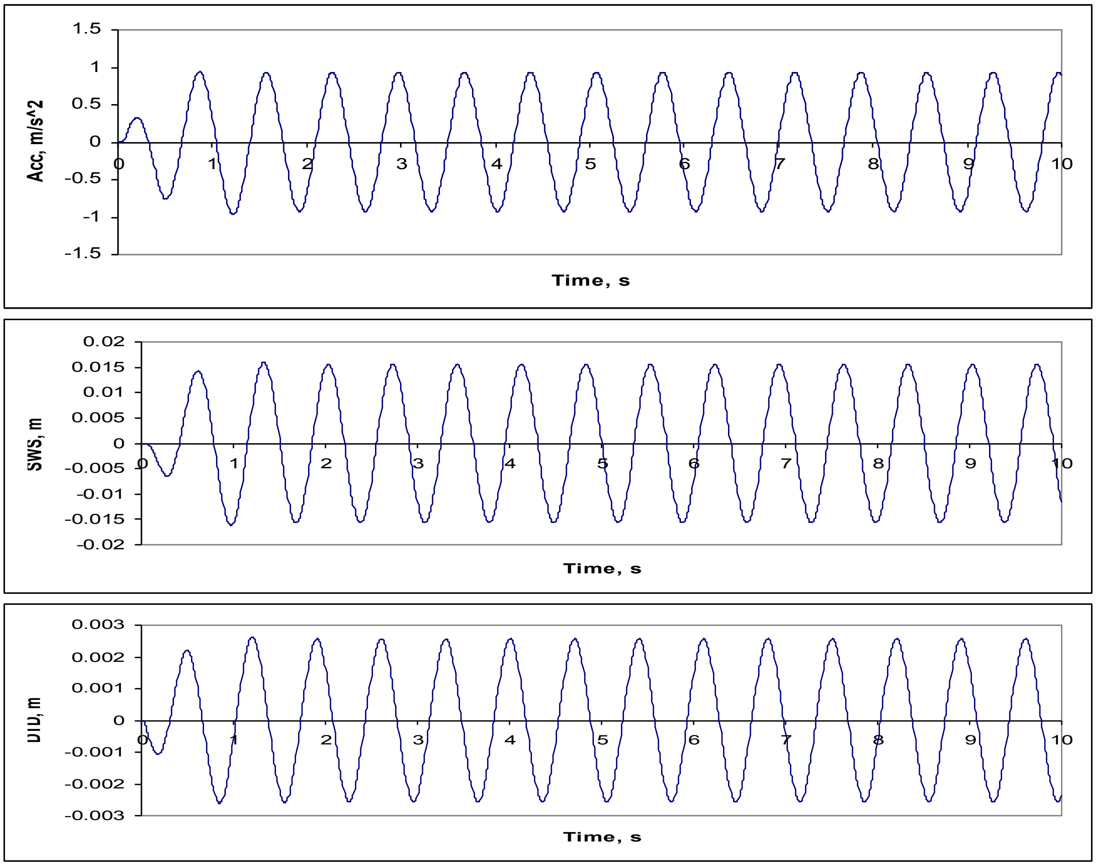

Fig. 10: body acceleration, suspension working space and dynamic tyre load for fuzzy control in time domain.

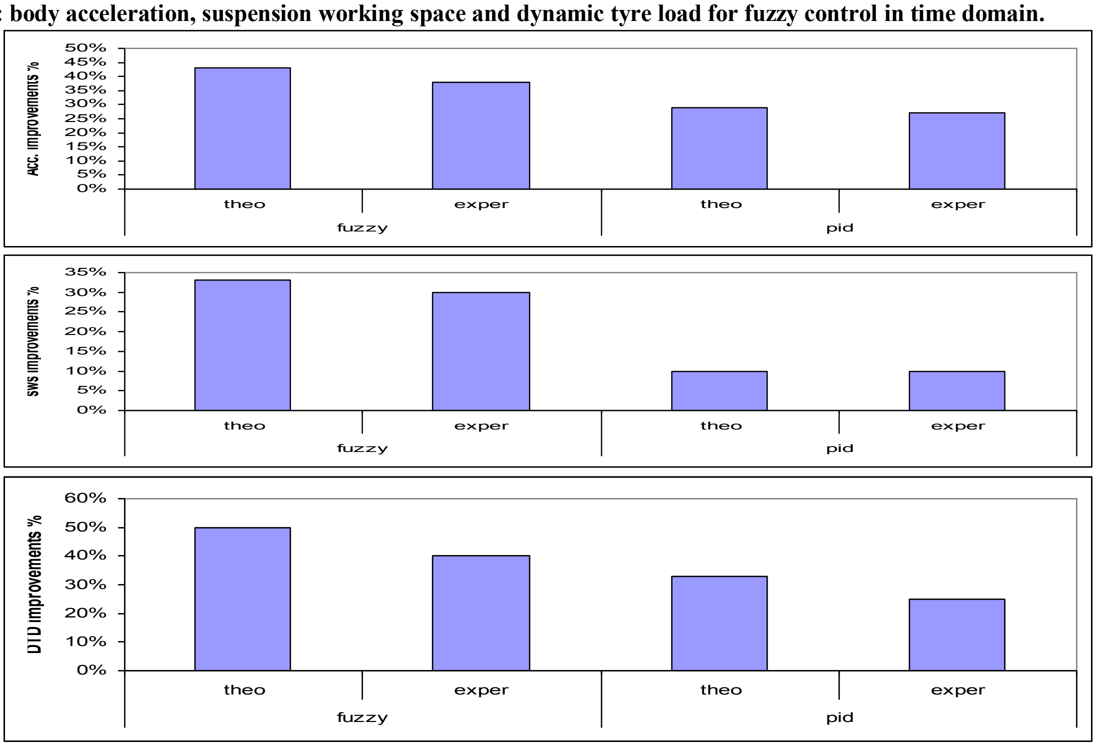

Fig. 11 body acceleration, suspension working space and dynamic tyre load improvements over passive in terms of peak to peak

Appendix A

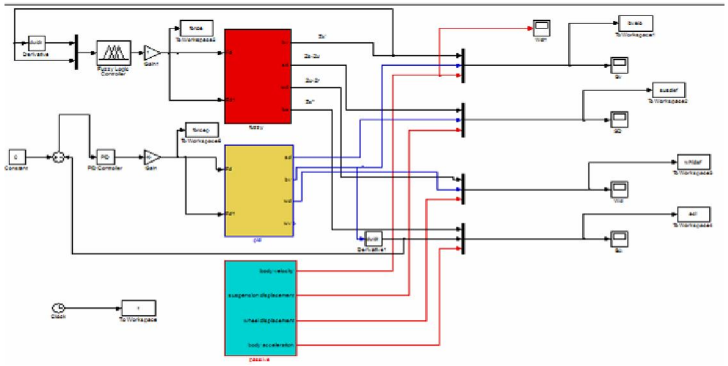

Fig. a1 Block diagram for passive, PID and Fuzzy models

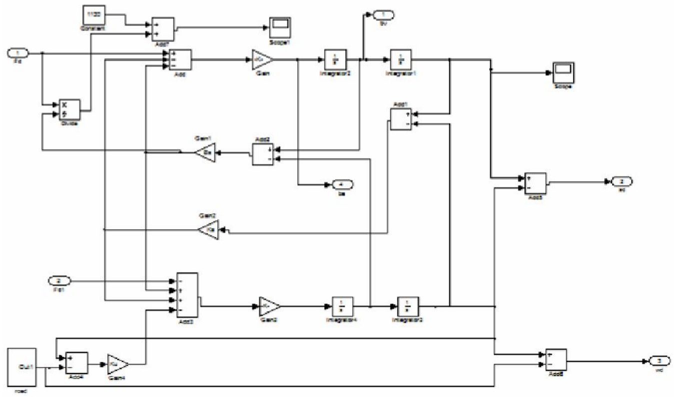

Fig. a2 Block diagram for passive suspension system

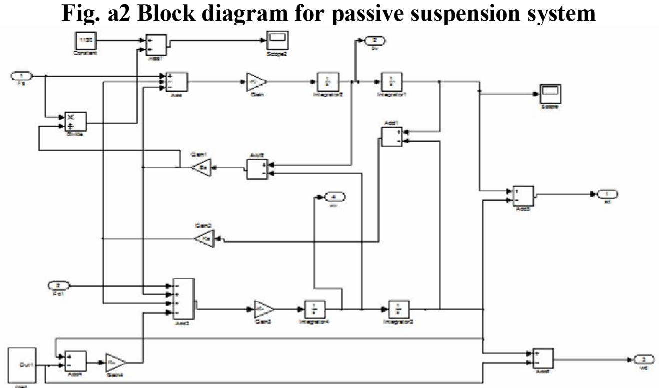

Fig. a3 Block diagram for semi active with PID control

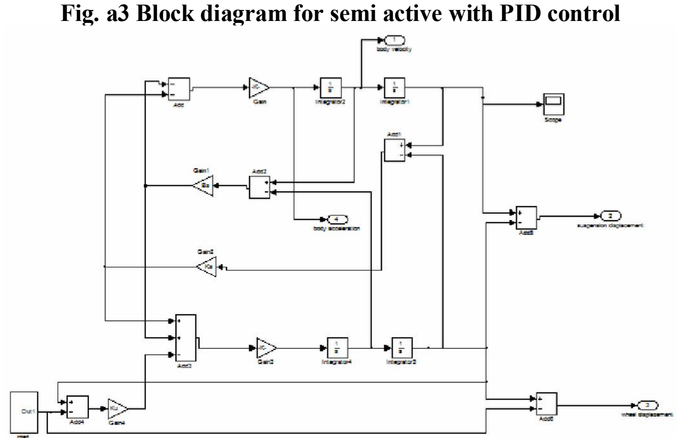

Fig. a4 Block diagram for semi active with Fuzzy control

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