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Parametric Study on Proportional Integral Derivative Controlled Semi-Active Suspension System

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ABSTRACT:

This paper presents the effect of the suspension working space, body displacement, body acceleration and wheel displacement for the non-controlled suspension system (passive system) and the controlled suspension system of a quarter car model (semi-active system), and comparison between them. The quarter car passive and semi-active suspension systems are modelled using Simulink. Proportional Integral Derivative controllers are incorporated in the design scheme of semi-active models. In the experimental work, the influence of switchable damper in a suspension system is compared with the passive and semi-active suspension systems.

KEYWORDS:

Switchable dampers; Semi-active Suspension; Controlled suspension; Proportional Integral Derivative control

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1. Introduction

The basic concept of land vehicle transportation has not changed much in the last few decades, although much progress was made in improving and optimizing the vehicle design and technology. The quest is always to go faster, farther and more comfortable which has led to the development of advanced suspension systems in recent years. An improved suspension system allows a vehicle to achieve higher speeds over rougher terrain. This has resulted in better handling and improvement in ride comfort [1]. The main objective of the active vibration control problem of vehicle suspension systems is to get security and comfort for the passengers by reducing the vertical acceleration of the vehicle body. An actuator incorporated to the suspension system applies the control forces to the vehicle body for reducing its vertical acceleration in active [2] or semi-active way [3-4]. Suspensions are classified into passive, semi-active and active suspension systems. A passive system comprises a damper and a spring having fixed characteristics. A semi-active system has the ability to modulate the damping coefficient of damper but the direction of damping force is dependent on the relative velocity across the sprung and unsprung masses. In case of an active suspension, an actuator is incorporated to provide the force without being influenced by relative velocity [5-7]. Passive suspension systems always represent a compromise between ride comfort and handling. Implementing a controllable suspension is therefore an attempt to narrow the gap between the opposing requirements for optimal ride comfort and handling [1]. The associated power consumption may be very high depending on the required performance. Furthermore, these suspension systems are very expensive [8].

Semi-active systems can lower the vibration transmission nearly as much as full-active systems without high cost and power requirements associated with active vibration control systems [9]. Semi-active suspensions have been shown to offer valuable benefits for vehicle suspensions [10-11]. The control strategies of full-active suspension are ideal from control point of view. Because of this advantage, fully-active control strategies have been studied in many suspension systems. For example, the ideal sky-hook model with control [12] and full-active target semi-active approximation control [8 and 13]. Numerous semi-active suspension systems were fitted to production vehicles, but most of these suspensions can be classified as adaptive. Semi-active suspension systems consist of an active damper in parallel with a passive spring. Semi-active control devices have received a great deal of attention in recent years, because they combine the best features of the passive and active ones. They have almost the same environmental robustness, mechanical simplicity and low cost as passive devices and can offer the adaptability of active control systems without requiring the associated large power sources. The damping characteristics are controlled by modulation of fluid-flow orifices, dry friction forces or electric or magnetic field applied to electro-rheological or magneto-rheological fluid dampers. For practical reasons, it is important that the feedback signals are relative displacement and relative velocity across the suspension, since these state variables can be measured directly for a moving vehicle [14-15]. A semi-active suspension is a valid engineering solution, because it requires a low power controller that can be easily realized at a lower cost than that of a fully active one to control the value of the damper coefficient [16-17].

Generally, there are two ways to research the control method for the full vehicle suspension system. One way is to research an effective control method in the environment of the quarter-car model firstly, then apply it to every control force of the suspension in the full-vehicle model. Another way is to investigate the full vehicle control method directly according to its running status and characteristics. This paper presents the suspension working space, body displacement, body acceleration and wheel displacement responses for the passive and Proportional Integral Derivative (PID) controlled semi-active suspension systems of a quarter car model. These suspension systems are modelled using Simulink. The influence of switchable damper in these vehicle suspension systems is investigated using laboratory experiments.

2. Mathematical modelling

The vehicle system has been modelled using the Lagrangian energy approach by considering the car body as a rigid body. The vehicle is moving over a stationary random road surface with a constant velocity. The vehicle suspension system model consists of sprung and unsprung masses as 2 rigid masses of the vehicle as shown Fig. 1. The equations of motion of the vehicle system as a forced damped system are given by,

$$m_1\ddot{x}_1 + k_1(x_1 - x_2) + c_1(\dot{x}_1 - \dot{x}_2) = 0 \quad (1)$$

$$m_2\ddot{x}_2 - k_1(x_1 - x_2) + c_1(\dot{x}_1 - \dot{x}_2) = k_2(x_0 - x_2) + c_2(\dot{x}_0 - x_2) \quad (2)$$

Where m_1 and m_2 are the sprung and unsprung mass of the vehicle respectively. k_1 and k_2 are the stiffness of spring and tyre respectively. c_1 and c_2 are the damping coefficient of spring and tyre respectively. The tyre is represented by a linear spring and a damper. The vehicle is mounted on its axles by a passive suspension, consisting of a linear spring in parallel with a viscous damper. The system equation of motion has been derived using Lagrangian energy approach using,

$$\frac{\partial}{\partial t}\left[\frac{\partial(KE)}{\partial x}\right] - \frac{\partial(KE)}{\partial x} + \frac{\partial(PE)}{\partial x} + \frac{\partial(DE)}{\partial x} = F(t) \quad (3)$$

Where *KE*, *PE* and *DE* are kinetic energy, potential energy and dissipated energy respectively. *F(t)* is the excitation force. The passive suspension system is modelled using Simulink as shown in Fig. 2.

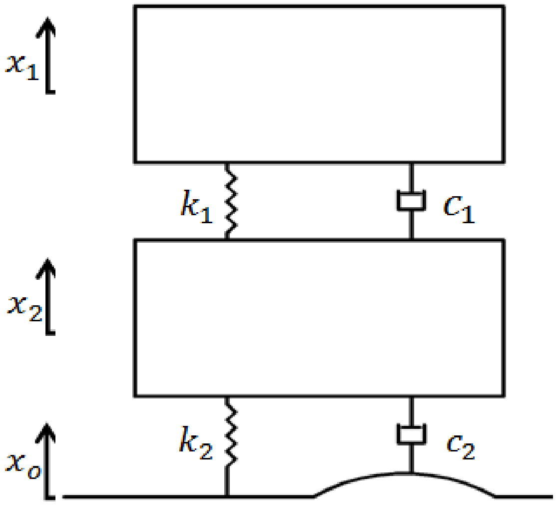

Fig. 1: Quarter car model

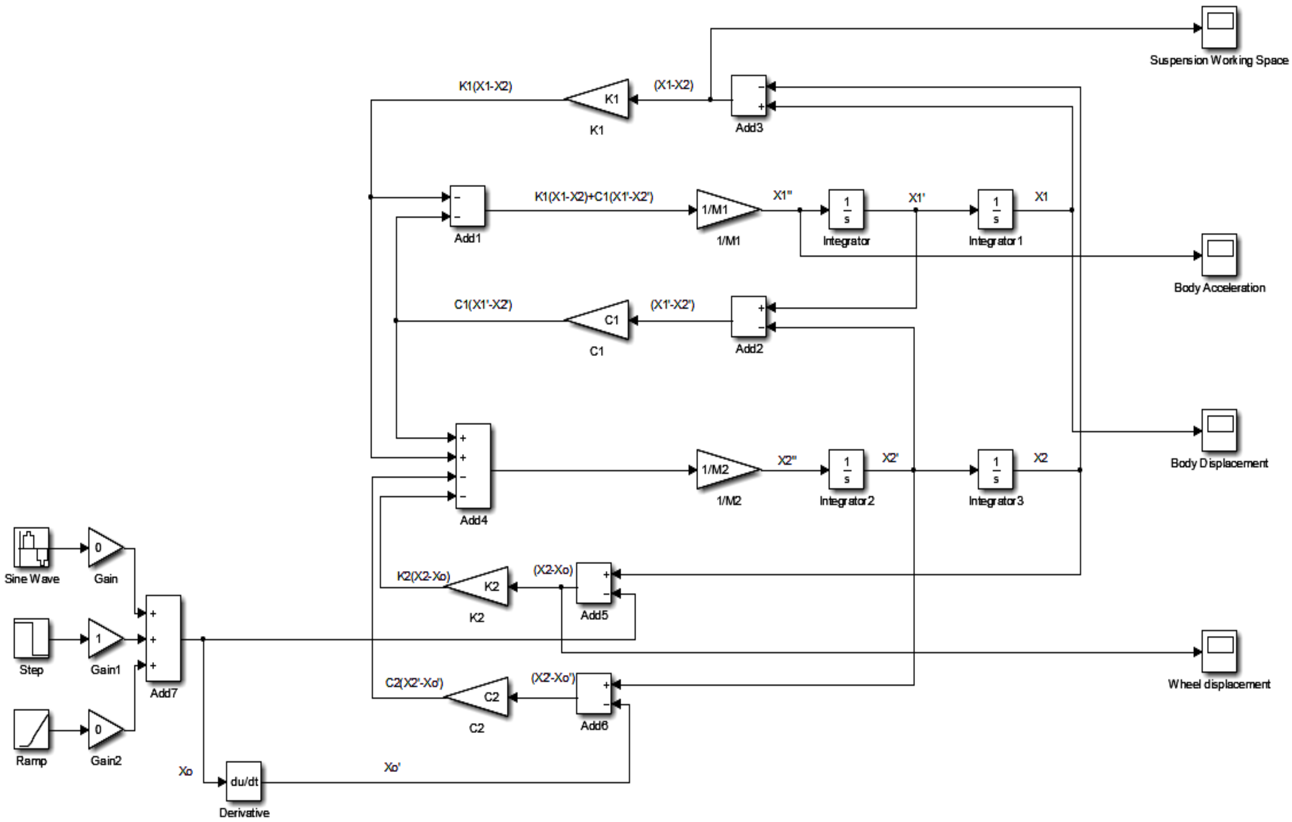

Fig. 2: Simulink model of passive suspension system

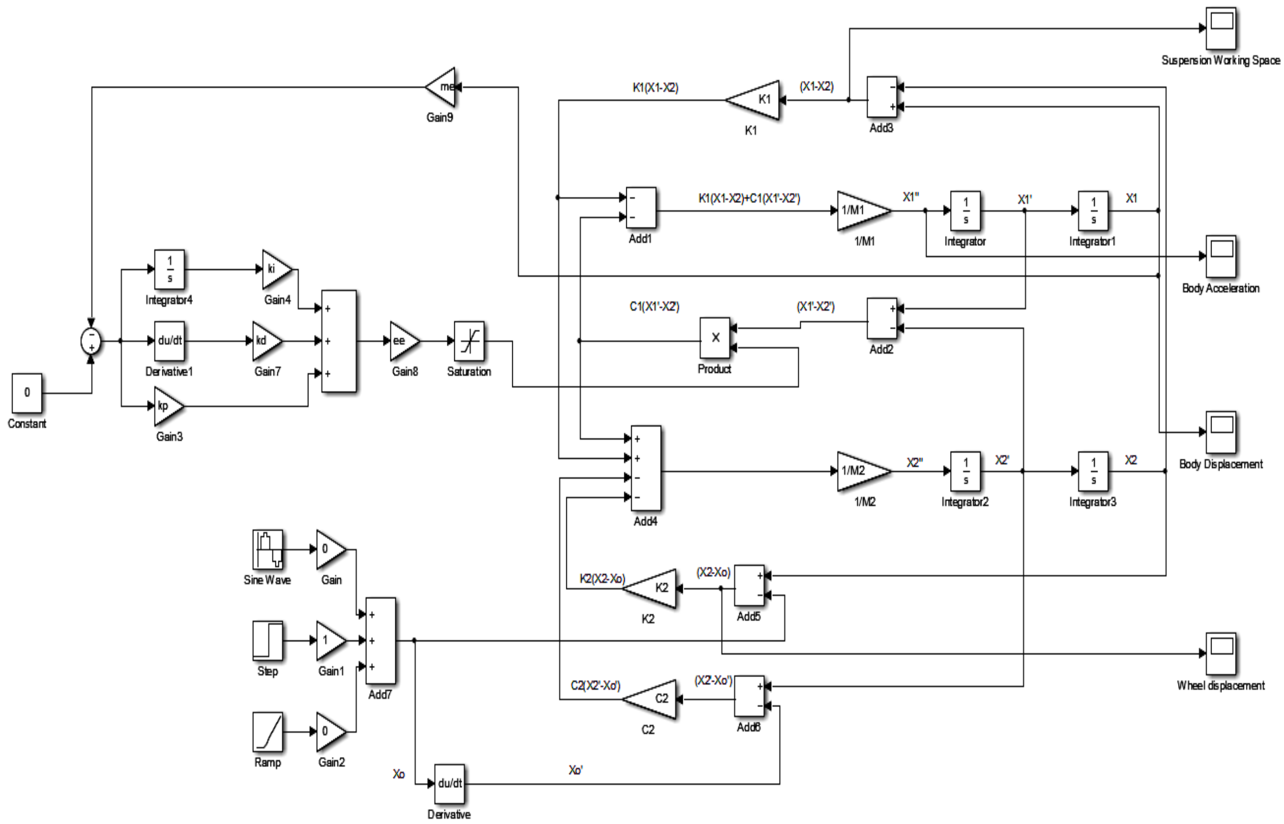

Fig. 3: Simulink model of PID controlled semi-active suspension system

The semi-active suspension system with PID controller as modelled in Simulink is shown in Fig. 2. While proportional and integrative modes are used as single control, the derivative mode is rarely used on its own in PID controlled semi-active suspension system. Combinations such as PI and PD control are very often used in practical systems. Derivative mode improves the stability of the system and enables an increase in gain *K* and decrease in integral time constant *Ti*, which increases the speed of the controller response. PID controller is used when dealing with higher order capacitive processes (processes with more than one energy storage) when their dynamics are not similar to the dynamics of an integrator (like in many thermal processes). Conventional autopilot is used for the most part of PID controllers[18]. The damping coefficient (c_1) is controlled with the vehicle's body displacement that is taken as a reference to the controller. The controller constants are taken as K_p, K_i and K_d respectively for the proportional, integral and derivative parts. The parameter of the suspension system models are given in Table 1.

Table 1: Suspension system model parameters

Parameter	Value	Unit	Parameter	Value	Unit
m_1	241.5	kg	m_2	41.5	kg
k_1	6	kN/m	k_2	140	kN/m
c_1	0.3	kNs/m	c_2	1.5	kNs/m
K_p	0.552	-	K_i	5.52	-
K_d	1300	-			

3. Responses from Simulink system models

The passive and semi-active suspension system models developed in Simulink are simulated for three road profiles namely, (P1) 8 cm step, (P2) sine wave of 8 cm amplitude and a frequency of 0.125 rad/s with a sample time of 0.1 s, and (P3) 45º ramp with an initial height of 8 cm. For these road profiles, P1 to P3, the suspension working space, body displacement, body acceleration and wheel displacement responses are shown in Fig. 4 to 7 along with their summary in Table 2 to 5respectively. The suspension working space is better through the use of semi-active suspension for the sine and ramp road profiles. For the step road profile, the peak amplitude in the negative direction for the semi-active system is higher than the passive system. However, the maximum amplitude in semi-active system is better than that of the passive system. The body displacement response of passive and semi-active suspension systems for the sine and ramp road profiles gave more or less similar performance. For the step road profile, the semi-active suspension system has reached stability around 2.5 seconds from the beginning of excitation, but slightly higher amplitude than the passive system. For two systems excitation gave the same behaviour of body displacement for the ramp road profile. Hence, their comparison was not shown in Table 3.

The body acceleration of semi-active suspension system is better than the passive suspension system, especially in relation to the time taken by the body to become stable from the beginning of excitation for all the road profiles. Though the body acceleration in semi-active suspension is higher than passive system (max. amplitude), the shorter time to stability will provide better ride comfort. The wheel displacement for the semi-active suspension system is better than that of passive system for step and ramp road profiles. For both systems, the excitation gave the same behaviour of wheel displacement for sine road profile.

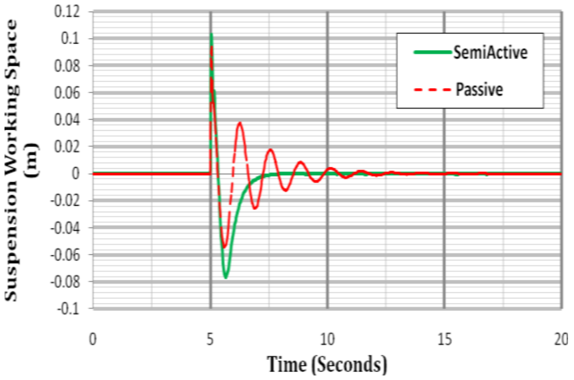

Fig. 4(a): Suspension working space – Step road profile (P1)

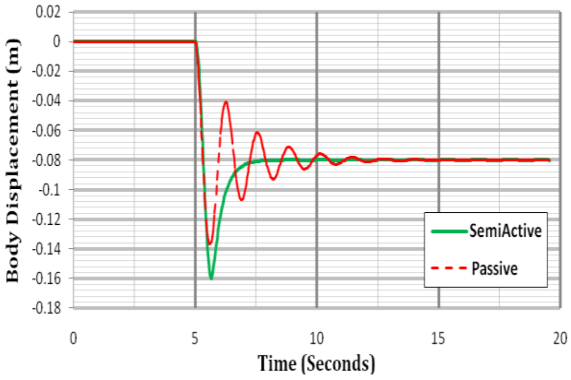

Fig. 5(a): Body displacement – Step road profile (P1)

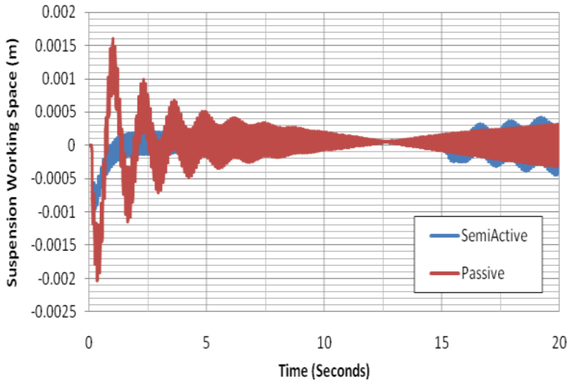

Fig. 4(b): Suspension working space – Sine road profile (P2)

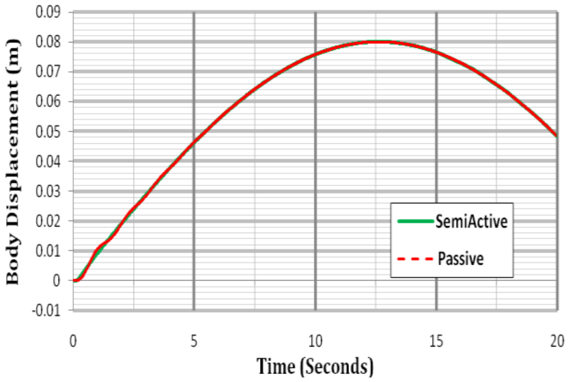

Fig. 5(b): Body displacement – Sine road profile (P2)

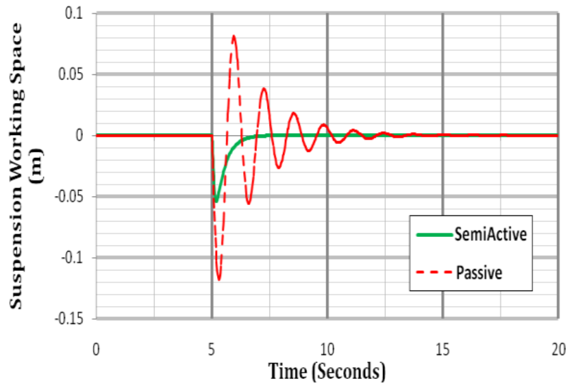

Fig. 4(b): Suspension working space – Ramp road profile (P3)

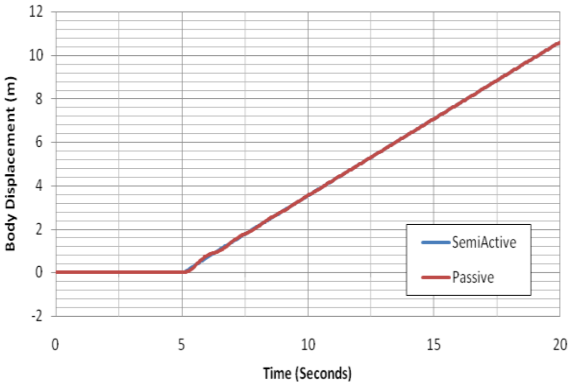

Fig. 5(c): Body displacement – Ramp road profile (P3)

Table 2: Summary of suspension working space vs. Road profiles

Response vs. Road profiles	Passive suspension system			Semi-active suspension system		
	P1	P2	P3	P1	P2	P3
Time to stable after excitation (s)	10	12.5	10	3	12.5	3
Peak +ve excitation amplitude (cm)	10	0.16	8.0	9	0.02	0
Peak -ve excitation amplitude (cm)	5	0.2	12	8	0.1	5.5

Table 3: Summary of Body displacement vs. Road profiles

Response vs. Road profiles	Passive suspension system		Semi-active suspension system	
	P1	P2	P1	P2
Time to stable after excitation (s)	10	12.5	2.5	12.5
Peak +ve excitation amplitude (cm)	14	8	16	8

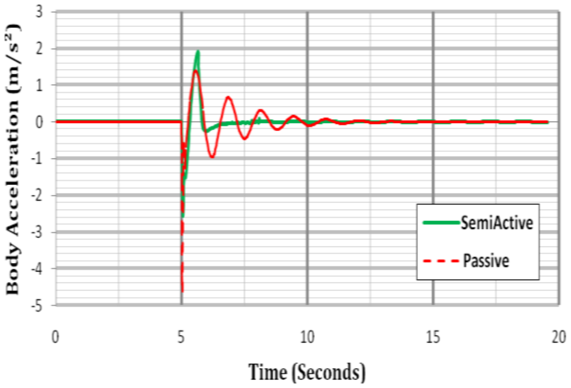

Fig. 6(a): Body acceleration – Step road profile (P1)

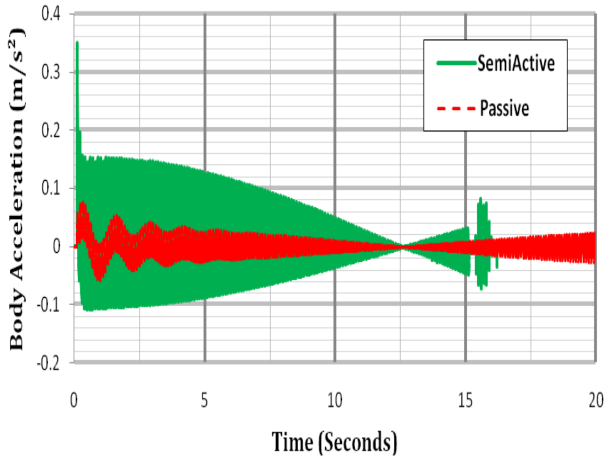

Fig. 6(b): Body acceleration – Sine road profile (P2)

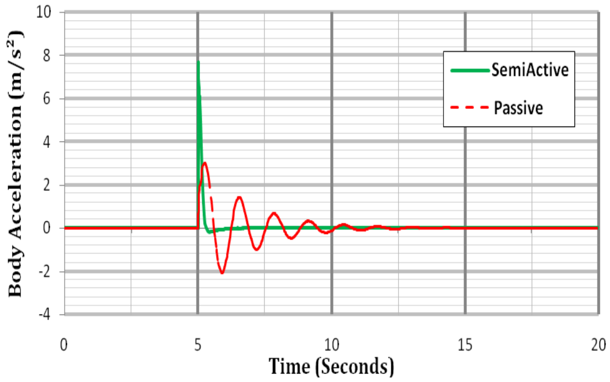

Fig. 6(c): Body acceleration – Ramp road profile (P3)

Table 4: Summary of body acceleration vs. Road profiles

Response vs. Road profiles	Passive suspension system			Semi-active suspension system		
	P1	P2	P3	P1	P2	P3
Time to stable after excitation (s)	10	12.5	8	2	12.5	2
Peak +ve excitation amplitude (m/s^2)	1.4	0.08	3	2	0.35	8
Peak -ve excitation amplitude (m/s^2)	4.5	0.06	2	2.5	0.1	0

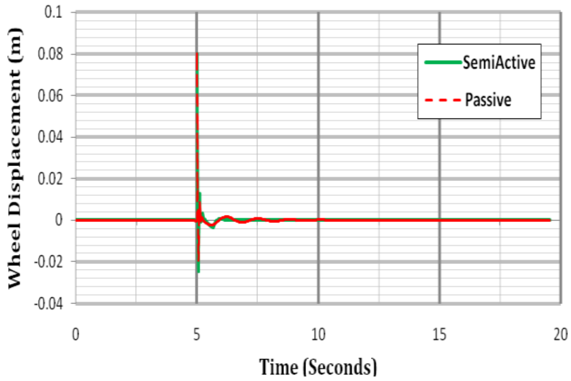

Fig. 7(a): Wheel displacement – Step road profile (P1)

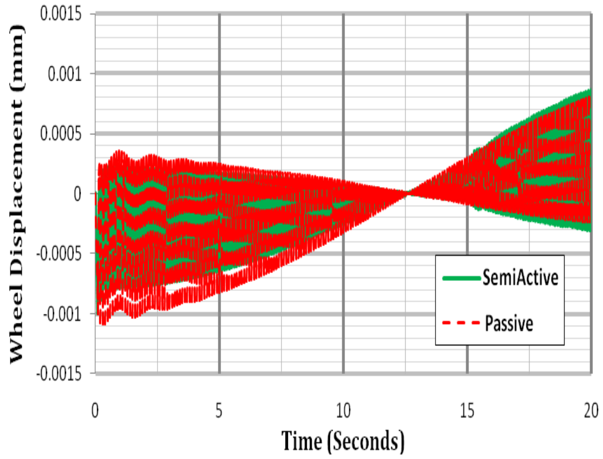

Fig. 7(b): Wheel displacement – Sine road profile (P2)

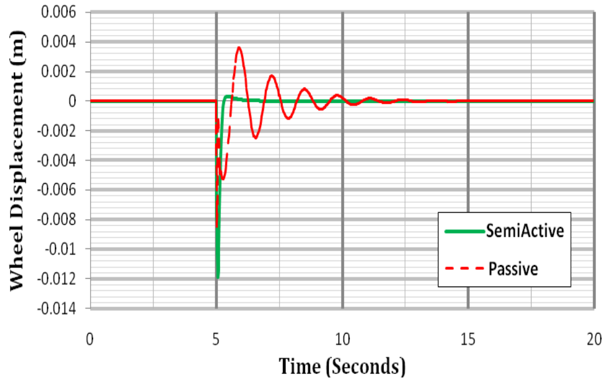

Fig. 7(c): Wheel displacement – Ramp road profile (P3)

Table 5: Summary of wheel displacement vs. Road profiles

Response vs. Road profiles	Passive suspension system			Semi-active suspension system		
	P1	P2	P3	P1	P2	P3
Time to stable after excitation (s)	5	12.5	8	1.5	12.5	2
Peak +ve excitation amplitude (cm)	8	0.04	0.35	8	0.04	0
Peak -ve excitation amplitude (cm)	2	0.1	0.85	2.5	0.1	1.2

4. Experimental work

The influence of switchable damper in semi-active and passive suspension systems is investigated in the experimental work. The damping coefficient is measured by the amplitude of the suspension system due to falling from 6 cm block. Fig. 8 shows the schematics of experimental setup to measure and record suspension working space for vehicle drop from a 6cm block. The setup consists of Linear Variable Differential Transformer (LVDT) sensors, battery, inverter, analogue to digital converter and data acquisition card. LabView is used for processing experimental data and suspension working space responses. Fig. 9 shows a snapshot of control panel interface with LabView for all sensors of the system. Fig. 10 shows a comparison of the suspension working space from the experiments using passive and semi-active suspension system. The semi-active suspension car became stable earlier than the passive suspension car. The peaks of suspension working space are +3.4mm & -2.2mm and +4mm & -5mm for the semi-active and passive suspension system respectively.

Fig. 8: Schematic of experimental setup

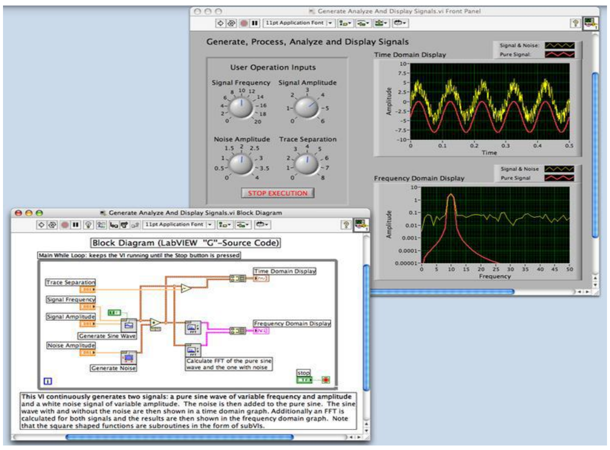

Fig. 9: LabView control panel interface for all sensors

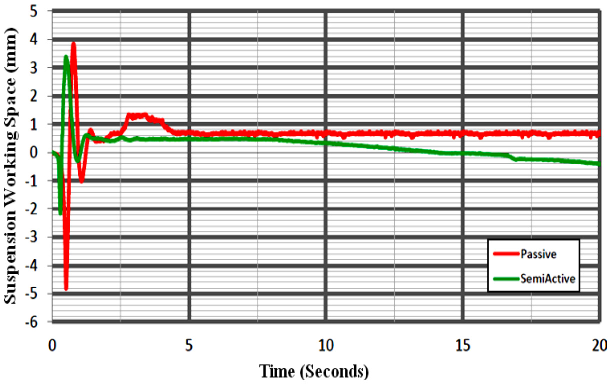

Fig. 10: Suspension working space from experiment - Passive vs. Semi-active suspension system

5. Conclusions

This study provides a theoretical and experimental approach to initiate the design of a realizable semi-active suspension system. For a quarter car, the system models of passive and semi-active suspensions were developed using Simulink. Simulations were undertaken for system responses for 8 cm step, sine wave of 8 cm amplitude and a frequency of 0.125 rad/s with a sample time of 0.1 s, and 45º ramp with an initial height of 8 cm road profiles. The suspension working space, body displacement, body acceleration and wheel displacement responses were compared for the two systems. In general, semi-active suspension systems have shown better performance than the passive system for most of the road profiles. Both the systems gave same behaviour of body displacement and wheel displacement for ramp road and sine road profiles respectively. The results of experimental work with switchable damper system have shown that the semi-active suspension system gave a better suspension working space than the passive suspension system.

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