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EXPERIMENTAL INVESTIGATION OF A PASSIVE QUARTER CAR SUSPENSION SYSTEM

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ABSTRACT

This research paper discussed the study of a two degree-of-freedom quarter car model passive suspension system. An open loop characteristics experiment using system identification method was carried out to determine the transfer function of the passive suspension system. Once these steps are completed, a closed loop compensated system is designed to control the position of Platform 1; i.e. the road surface. Platform 1 will provide the road profiles of different step height for the passive quarter car suspension system. The PID controller was proven to be able to improve steady-state error by 48.6%, 31.9%, 10.9% and 21.8% for reference heights of 30cm, 32cm, 34cm and 36cm, respectively, with a slight 12% overshoot for 34cm and 36cm reference heights.

Keywords: passive quarter car, suspension system, PID controller.

1. INTRODUCTION

Shock absorption is a vital area to focus on in the automobile industry. Design engineers are constantly on a lookout for improvements to reduce shocks in vehicles due to irregular road surfaces, drag forces and engine vibrations [1-7]. The main reason for vehicle body vibration is speed bumps and potholes present in different road terrains. Hence, this is where vehicle suspension system play its ultimate role, i.e. to increase car maneuvering effectiveness and comfort by reducing and isolating shock applied to the vehicle, which would otherwise injure passengers or damage its load. Suspension system is categorized into three (3) types, which are passive suspension system, active suspension system and semi-active suspension system.

The performance of an automotive passive suspension system is analyzed using a quarter car model [8-11]. It is discovered that the vehicle sprung and unsprung mass displacement overshoot and acceleration amplitude are immensely high, showing negative impacts on the life span of the suspension system, besides resulting in discomfort to the passengers of the vehicle. Their research showed that passive suspension faces a disadvantage that must be resolved immediately.

An active suspension system with pneumatic actuators is concluded to improve ride comfort and maintain vehicle maneuver, compared to a passive suspension system [12, 13]. The authors provided clear comparison between passive and active suspension systems for both car body acceleration and displacement parameters when subjected to road disturbances. Further studies supported the fact that active suspension delivers better performance, reducing more than half of suspension movement in that of a passive case because the excitation force produced from the actuators in active suspension cancels out the generated force [14,15]. When crossing over a road bump, active suspension gives much lower peak overshoot and faster settling time, with smaller body displacement compared to passive system. Hence this concluded that it is possible to achieve better suspension by adding active elements, instead of using purely passive ones.

The performance of passive suspension and active suspension is investigated using two (2) different control algorithms: "Skyhook stability augmentation system (Sky-SAS)" algorithm and "stability augmentation system (SAS)" algorithm [16, 17]. While Sky-SAS provided better ride quality than SAS-controlled active suspension, passive suspension proves to be inferior to active suspension, indicating that the performance of active suspension system does rely greatly on the control technique used.

A low cost and simple semi-active suspension system is proposed and showed to provide great damping properties over a wider range of load, in the meantime reduce the compromise of comfort and maneuvering issue faced with passive suspension system [18,19]. The model of the semi-active suspension is based on equations of motion that includes different parameters such as damping ratio, stiffness and displacement. The evaluation is done via simulations only by transforming the equations into a data flow circuit in MATLAB/Simulink, then generating the graphical outputs.

2. STRUCTURE AND SETUP

Figure-1 shows the developed quarter car model passive suspension system with a base of 345mm×345mm×8mm and a height of 1000mm. It consists of fourteen bearings, three infrared (IR) sensors, one gas spring (damper), one double acting pneumatic cylinder, one solenoid 5/2 way valve, one pressure regulator and two springs of different spring coefficients.

The setup is interfaced to a Micro-Box 2000/2000c connected with a host PC via Ethernet line. The IR sensors detect the displacement of the suspension system and send the data to the Micro-Box which is then transferred to the PC. The PC shows the results through MATLAB 2009a software which can then be further

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analyzed by suitable means. The user can insert a desired input through the PC to the Micro-Box, which is then transferred to the double acting cylinder to actuate movements. The three main suspension platforms suspended by the aluminum alloy rods represent different elements in a conventional car suspension system. Figure-1 shows Platform 1 representing the road surface which a car is moving on. Different inputs to the double acting cylinder will move Platform 1 by different magnitudes and displacements, imitating the different road profiles when cars are moving on various terrains. Platform 2 represents the car tire while Platform 3 is the car body upon which different weights can be added to signify different car masses.

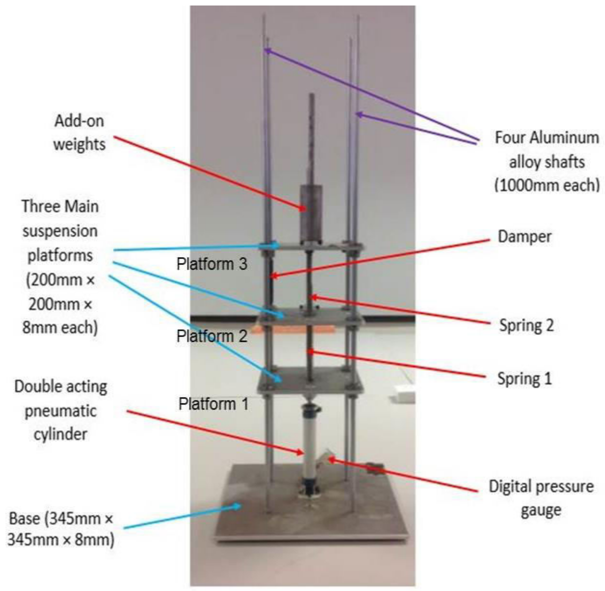

Figure-1. Passive suspension system setup.

3. RESEARCH EXPERIMENTS

3.1 System identification using MATLAB/Simulink

The use of system identification is the basic step before designing a controller. Simulink is equipped with a System Identification Toolbox which provides users with the corresponding transfer function for each set of input and output data. The open loop model of the suspension system is constructed using Simulink blocks, which includes step input (voltage input), sensoray526 AD (reads signal from IR sensor in volts), sensoray526 DA (converts digital signal from pulse generator to analog signal), a conversion subsystem (converts IR sensor output voltage to displacement) and a low pass filter. The low pass filter implemented is shown in Equation (1). Figure-2 shows the block diagram for open loop experiments.

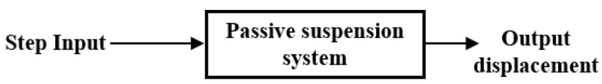

Figure-2. Block diagram for open loop characteristics test.

$$\text{Low pass filter} = \frac{45}{s+45} \tag{1}$$

To characterize the suspension system, input voltages of 6V, 7V, 8V, 9V and 10V were applied, respectively. The System Identification Toolbox is then used to obtain the system model based on the extracted input and output data. A linear parametric second order transfer function is generated for each input voltage, and substitute in to Equation (2). Based on the system identification results, the transfer function of the suspension system is given by Equation (3). The transfer function shown in Equation (3) exhibits the similar results to the experimental works, therefore Equation (3) was chosen as the transfer function for evaluating the controller performances. Figure-3 shows the block diagram for choosing the system identification transfer function.

$$G[s] = \frac{As+B}{Cs^2+Ds+E} \tag{2}$$

$$G[s] = \frac{-2.2771s+110}{s^2+47.18s+4.467} \tag{3}$$

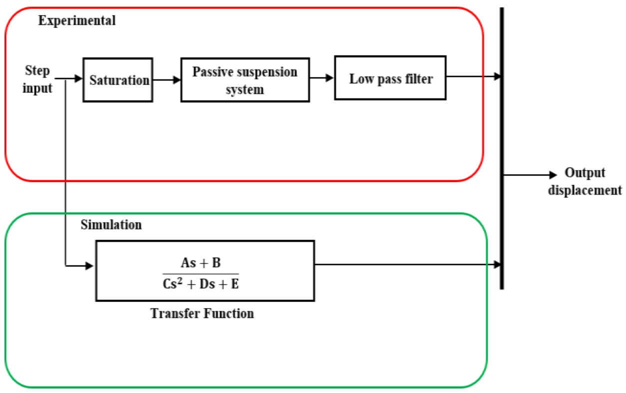

Figure-3. Block diagram for choosing the suitable transfer function.

3.2 Closed-loop uncompensated and compensated system

In this section, the performances of the uncompensated and compensated system of the suspension system were evaluated via simulation and experimental works. For the compensated system, a PID controller was implemented as shown in Figure-4.

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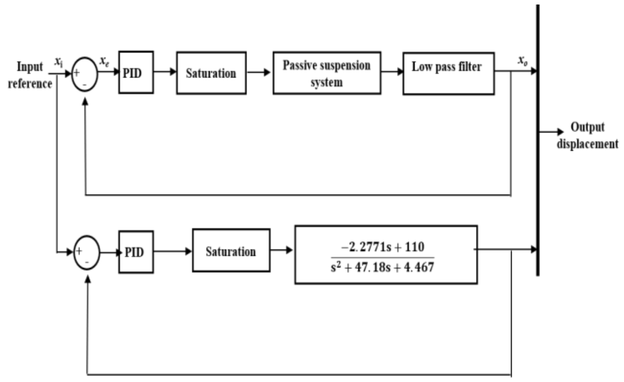

Figure-4. Block diagram for closed loop compensated system with equation (3) as transfer function.

The values of K_p, K_i and K_d are manually tuned, by first setting the value of K_i and K_d to zero. K_p value is then increased slowly until the system starts to oscillate. Next, K_p is set to approximately half of that value. K_i value is increased to eliminate any offset in sufficient time for the response. Finally, K_d value is increased to eliminate any oscillations and decrease settling time. The PID controller is then fine-tuned by changing the K_p, K_i and K_d values in small intervals for better results.

4. RESULTS AND DISCUSSIONS

4.1 Closed loop uncompensated system

The closed loop uncompensated system determined the error occuring between the output displacement and reference. Figure-5 shows that for reference height of 30cm, the experimental final displacement reaches approximately 29.28cm, with steady-state error of 0.72cm. As for simulation output, the final displacement is recorded at a higher 29.44cm, with smaller steady-state error of 0.56cm. The rise time is 2.45s.

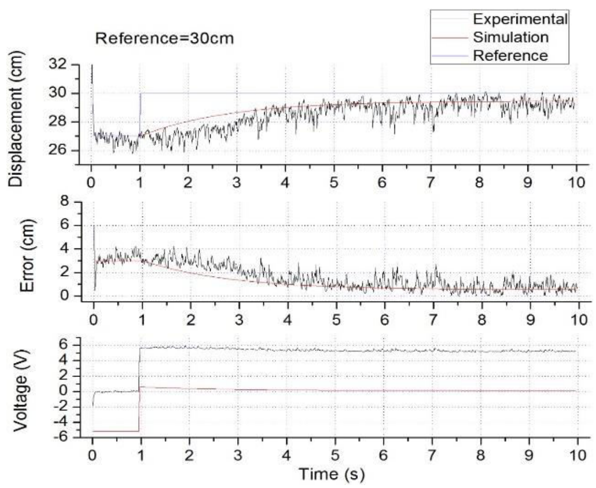

Figure-5. Graph of displacement, error and voltage against time for reference height = 30cm.

Figure-6 shows that for reference height of 36cm, the experimental final displacement is 35.45cm with steady-state error of 0.55cm. The simulation final displacement is a smaller number of 34.36cm with larger steady-state error of 1.64cm. The rise time is 0.55s.

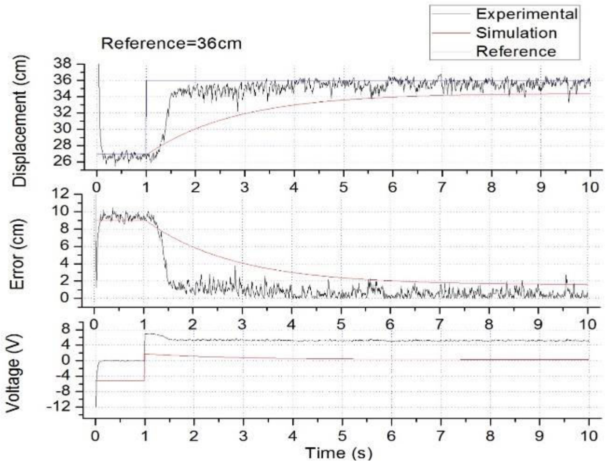

Figure-6. Graph of displacement, error and voltage against time for reference height = 36cm.

4.2 Closed loop compensated system

For closed loop compensated system, firstly the value of K_p was varied, while the K_i and K_d values are set to 0, making this a proportional (P) controller setting. When $K_p = 1$, $K_i = 0$, $K_d = 0$ and reference height = 30cm, the output displacement reduces the steady-state errors, from 0.72cm to 0.55cm for the experimental value, as shown in Figure-7.

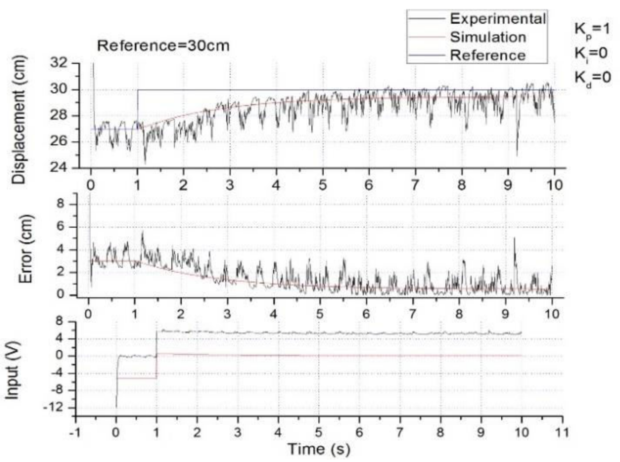

Figure-7. Graph of displacement, error and voltage against time for $K_p = 1$, $K_i = 0$, $K_d = 0$, reference height = 30cm.

Figure-8 shows the results when the PID gains are set as $K_p = 2$, $K_i = 0$, $K_d = 0$, with reference height of 30cm. It can be depicted that the experimental and simulation output displacement has reduced steady-state errors, from 0.72cm to 0.39cm and from 0.56cm to 0.29cm respectively, whilst the rise time for both experimental and simulation are reduced from 2.45s and 3.75s to 0.55s and 2.09. However, the experimental response has a 53% overshoot. In addition to that, the system also showed

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instability and oscillations when its displacement reaches approximately 30cm.

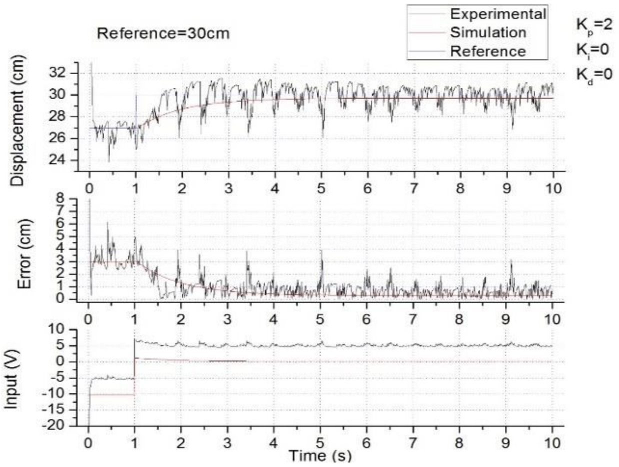

Figure-8. Graph of displacement, error and voltage against time for $K_p = 2$, $K_i = 0$, $K_d = 0$, reference height = 30cm.

Figure-9 shows the reference height of 36cm, when $K_p = 1.5$, $K_i = 0$ and $K_d = 0$. The experimental and simulation output displacement for this setting shows an improvement in steady-state errors from 0.55cm and 1.64cm to 0.43cm and 1.12cm respectively. Rise time for both experimental and simulation also improved to 0.23s and 2.76s respectively. However, the experimental response has a 16.7% overshoot, and oscillated when it reaches 36cm.

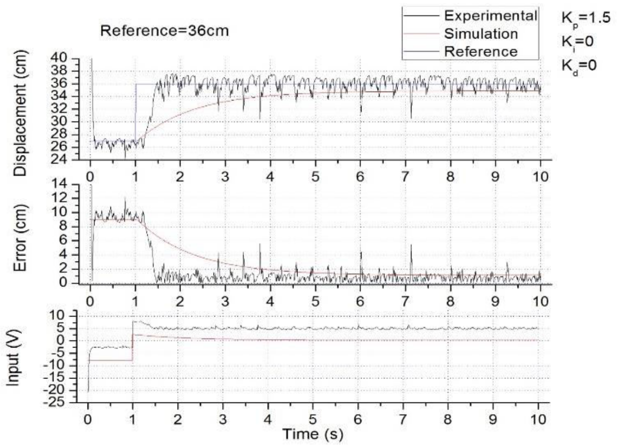

Figure-9. Graph of displacement, error and voltage against time for $K_p = 1.5$, $K_i = 0$, $K_d = 0$, reference height = 36cm.

Next, K_p and K_i values are varied while K_d is set to 0 as shown in Figure-10. This setting is also referred to as a proportional-integral (PI) controller setting. When $K_p = 1.35$, $K_i = 0.1$, $K_d = 0$ and reference height = 30cm, the experimental and simulation output displacement shows increase of steady-state errors, from 0.72cm and 0.56cm to 1.19cm and 1.22cm respectively. The experimental and simulation rise time in this setting have also increased significantly to 6.13s and 6.62s respectively. Figure-11 shows the controller analysis in PI controller setting with $K_p = 1.35$, halved value of $K_i = 0.05$ and $K_d = 0$ with reference height = 30cm. The experimental and simulation output displacement exhibited higher steady-state errors; an increase from 0.72cm and 0.56cm to 0.79cm and 0.94cm respectively. The experimental rise time is improved to 2.06s but the simulation rise time increased instead to 4.42s.

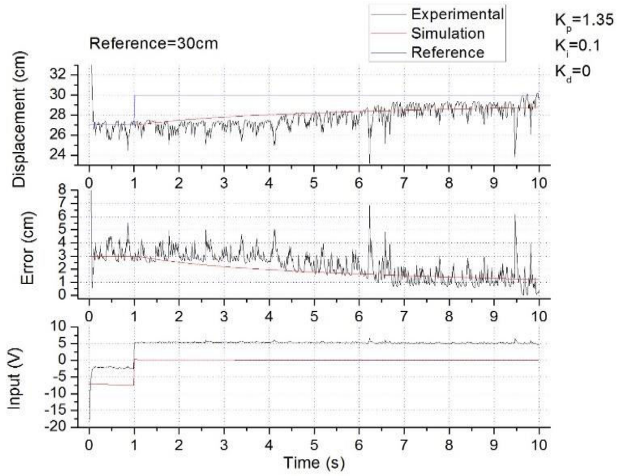

Figure-10. Graph of displacement, error and voltage against time for $K_p = 1.35$, $K_i = 0.1$, $K_d = 0$, reference height = 30cm.

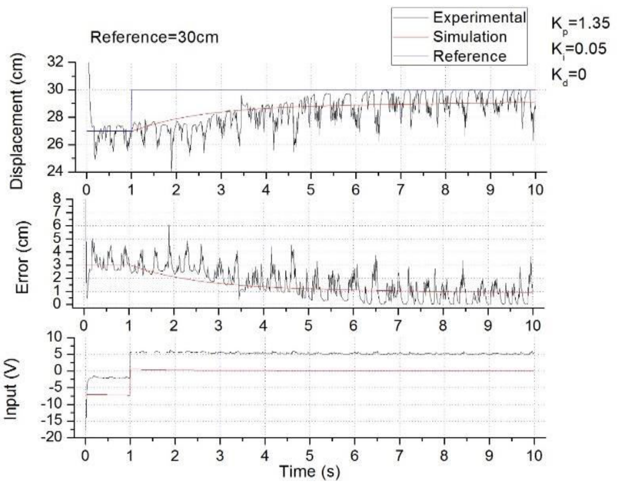

Figure-11. Graph of displacement, error and voltage against time for $K_p = 1.35$, $K_i = 0.05$, $K_d = 0$, reference height = 30cm.

To further analyze the controller output, the PID gains are thus set to a larger proportional gain, $K_p = 1.35$, minimized value of $K_i = 0.01$ and $K_d = 0$, with reference height set for 36cm as shown in Figure-12. The experimental and simulations results show that the output displacement has decreased and improved steady-state errors from 0.55cm and 1.64cm to 0.45cm and 1.28cm respectively. The rise time also improved for both experimental and simulation, to 0.23s and 3.11s. However, the experimental response now faced a 14.6% overshoot,

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in addition to minor oscillations when it reached 36cm. Hence, K_d value is introduced to remove these oscillations. Therefore, the final optimal values of the PID controller are set to be K_p = 1.35, K_i = 0.021 and K_d = 0.0022 for reference heights 30cm to 36cm producing a controller analysis pattern such as shown in Figures-13 to 16. It can be depicted that the experimental results shows a good agreement with the simulation results. Table-1 consolidates the transient properties of the PID compensated system.

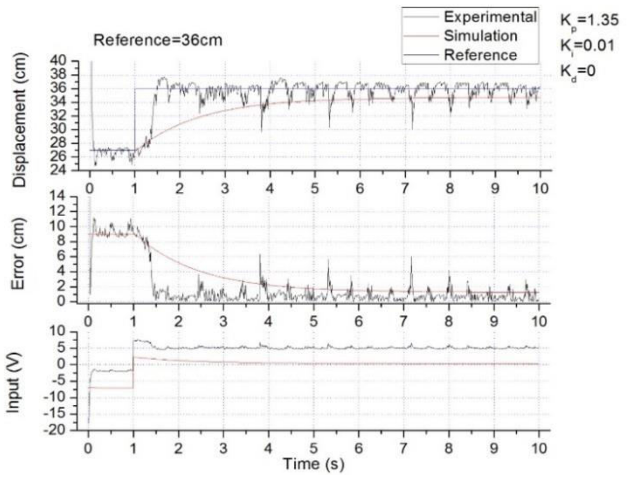

Figure-12. Graph of displacement, error and voltage against time for K_p = 1.35, K_i = 0.01, K_d = 0, reference height = 36cm.

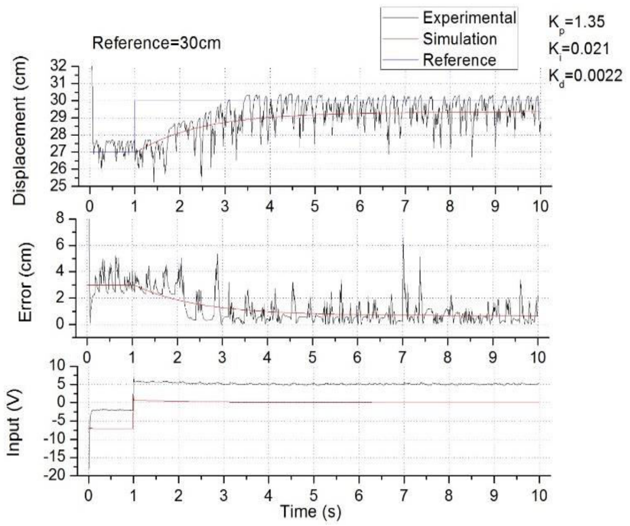

Figure-13. Graph of displacement, error and voltage against time for K_p = 1.35, K_i = 0.021, K_d = 0.0022, reference height = 30cm.

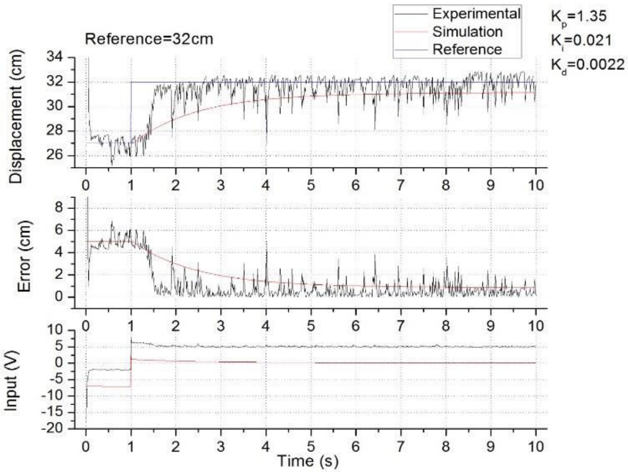

Figure-14. Graph of displacement, error and voltage against time for K_p = 1.35, K_i = 0.021, K_d = 0.0022, reference height = 32cm.

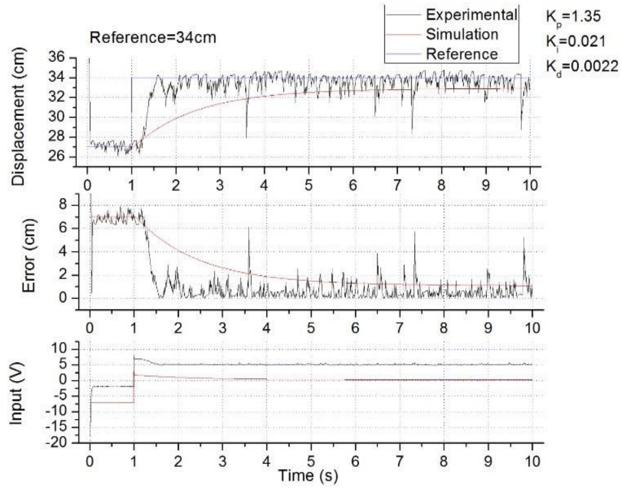

Figure-15. Graph of displacement, error and voltage against time for K_p = 1.35, K_i = 0.021, K_d = 0.0022, reference height = 34cm.

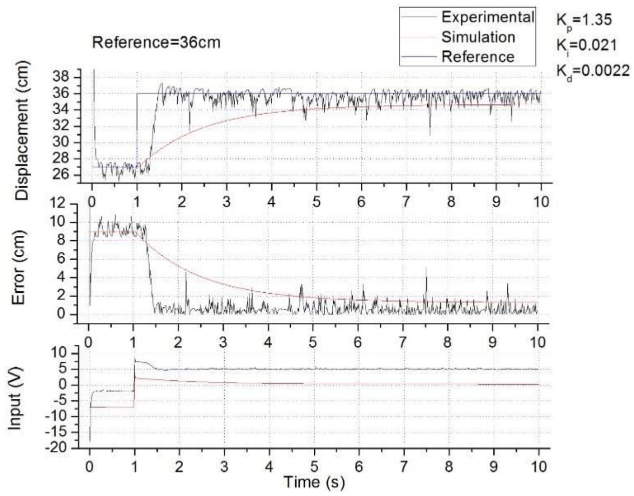

Figure-16. Graph of displacement, error and voltage against time for K_p = 1.35, K_i = 0.021, K_d = 0.0022, reference height = 36cm.

Table-1. Transient response properties for $K_p = 1.35$, $K_i = 0.021$, $K_d = 0.0022$.

Reference height		**30cm**		**32cm**	
		Exp.	**Sim.**	**Exp.**	**Sim.**
Steady-state error (cm)	Average	0.37	0.65	0.32	0.88
	Standard Deviation	0.72	0	0.67	0
Rise time (s)		1.53	3.31	0.56	3.23
Overshoot		0	0	0	0
Reference height		34cm		36cm	
		Exp.	Sim.	Exp.	Sim.
Steady-state error (cm)	Average	0.41	1.1	0.43	1.32
	Standard Deviation	0.73	0	0.7	0
Rise time (s)		0.25	3.21	0.22	3.21
Overshoot		0	12.75%	0	12.02%

5. CONCLUSIONS

In this paper, Platform 1 provides the road profiles of a passive quarter car suspension system when passing through a road bump of different heights. A PID controller with values of $K_p = 1.35$, $K_i = 0.021$ and $K_d = 0.0022$ is successfully designed to achieve better system performance of the Passive Quarter Car Suspension System. The controller was proven to be able to improve steady-state error by 48.6%, 31.9%, 10.9% and 21.8% for reference heights of 30cm, 32cm, 34cm and 36cm, respectively, with a slight 12% overshoot for 34cm and 36cm reference heights. As for the rise time, the PID controller successfully reduced the rise time period by 37.6%, 60%, 67.5% and 60% for the stated reference heights respectively. Future work includes designing an active car suspension system for smooth maneuvering and real-time on-road driving. This includes adding an actuator between Platform 2 and Platform 3 to control their movements, hence transforming the system into an active suspension system. The Platform 2 and Platform 3 will represent a vehicle's tire and its body, respectively. The additional actuator is predicted to reduce the oscillation and displacement of the suspension system when subjected to various road disturbances.

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