Design and Development of a Semi-active Suspension System for a Quarter Car Model using PI Controller

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Abstract:
This paper presents the design and development of a semi-active suspension system for a vehicle. The main idea is to develop a system that is able to damp vibration of the vehicle body while crossing the bumps on the road. This system is modeled for a single wheel assembly and then the laboratory prototype of the complete system has been manufactured. It is used to physically simulate the spring-mass-damper system in vehicle and observe the frequency response to the external disturbances. The developed low-cost smart experimental equipment consists of a motor with offset mass which works as an oscillator to induce vibration, a spring-mass-damper system where the variable damper works as a pneumatic cylinder that allows varying the damping constant (c). Proportional-Integral (PI) controller is used to control the damping properties of the semi-active suspension system automatically. The system is designed in contrast to the most of the available suspension systems in the market that have only passive damping properties. The results of this research demonstrate the efficiency of the developed variable damper-based control system for the vehicle suspension system.

Keywords: *semi-active suspension, control, damper, road profile, vibration*

1. Introduction

Active suspension system and its control strategy are discussed in details by many researchers. Most of the researchers have discussed the design and simulation results based on software tools. The real prototype is not widely available in the market. However, by reviewing various methods and products available for experimental purpose, it can be summarized as two types: a) hands on experimental apparatus and, b) virtual lab apparatus. The virtual lab apparatus is a good method in terms of cost saving and efficiency, but by this method people are not exposed to the real world experience. In many works, most of the tunable parameters are tuned manually through the control box available together with the set of apparatus.

However, for data gathering, it requires another unit based on PC-aided data acquisition module which is very expensive. In this developed system, the input and output data is recorded and displayed on a PC through Universal Serial Bus (USB) connection which is widely available in most of the PC nowadays. The key researches and works are done by many researchers which can be highlighted as follows.

A modelling and simulation based study carried out for one-quarter vehicle model to reduce vibration. An emphasis has been placed upon the interrelations between computer-aided simulation and other elements of the development process [1]. Another paper presets a mathematical model for the passive and active suspension systems for quarter car model using PID controller. Current automobile suspension systems use passive components only by utilizing spring and damping coefficient with fixed rates. The performance of the proposed system is evaluated using Matlab Simulink [2]. In fact, H_∞ controller is responsible for minimizing the infinity norm of two subsystems. The first one is from car body travel to road disturbance, and the second one is from suspension deflection to road disturbance. These two control targets are improved by a logical control input that is determined by H_∞ control approach. In addition, the sensitivity analysis is done to show that the active suspension system is able to work when spring-mass changes based on number of passengers [3].

In a similar work, a suspension system has been modelled as a two-degree-of freedom of a quarter-car model to represent passive and active suspension systems. A fuzzy logic controller for an active vehicle suspension system is designed and simulated using MATLAB, and compared the results with a passive suspension system [4]. In another research, an optimal preview control of a vehicle suspension system traveling on a rough road is studied and a three-dimensional seven degree-of-freedom car-riding model and several descriptions of the road surface roughness heights, including haver sine (hole/bump) and stochastic filtered white noise models, are used in the analysis. In this study, a contact-less sensors affixed to the vehicle front bumper to measure the road surface height at some distances in the front of the car. The suspension systems are optimized with respect to ride comfort and road holding preferences including accelerations of the spring-mass, tire deflection, suspension rattle space and control force [5]. Another similar work presents a non-linear design method using LQR theory to simulate and observe a vehicle's active suspension response and effect in a vehicle [6]. A review paper presents the advantages and disadvantages associated with the suspension systems of vehicles of conventional, active and semi-

-active systems based on the elements of controlled characteristics of both elastic elements and damping. It was suggested to apply and investigate an advance signal processing methods in vehicle's vibration research [7].

A thesis report [8] presents two new adaptive vehicle suspension control methods, which significantly improve the performance of mechatronic suspension systems by adjusting the controller parametrization to the current driving state. The first concept is an adaptive switching controller structure, which dynamically interpolates between differently tuned linear quadratic regulators. The second control approach (adaptive reference model based suspension control) emulates the dynamic behavior of a passive suspension system, which is optimally tuned to improve ride comfort for the current driving state while keeping constraints on the dynamic wheel load and the suspension deflection [8]. In another work, the Linear Quadratic Control (LQR) technique is implemented to the active suspension system for a quarter car model. Comparison between passive and active suspensions system are performed based on simulation by selecting different types of road profiles [9]. For the tracking control problem of vehicle suspension system, a robust design method of adaptive sliding mode control is derived and designed so that the practical system can track the state of the reference model. The influence of parameter uncertainties and external disturbances on the system performance can be reduced and system robustness can be improved [10]. In another study, two active vibration controllers are proposed for hydraulic or electromagnetic suspension systems, which only require position measurements. Some numerical simulation results are provided to show the efficiency, effectiveness and robust performance of the feedforward and feedback linearization control scheme proposed for a nonlinear quarter-vehicle active suspension system [11]. In a different research work, a design approach of robust active vibration control schemes for vehicle suspension systems using differential flatness, sliding modes and Generalized Proportional-Integral control techniques is discussed in order to attenuate undesirable vibrations induced by irregular road disturbances [12]. Some other works are done on fuzzy control systems, active vibration rejection technique, based on four degree of freedom, state derivative feedback system and permanent magnet based active suspension system [13–21].

Based on the above literatures, it can be said that the works done by the researchers are mainly software based design and simulation. The semi-active suspension system based on the damping fluid control is a new area of research. This paper describes the complete design, development and analysis of the semi-active suspension system through varying damping constant (c). Changing the value of damping constant, the suspension system can be adjusted in real time.

2. Modelling and Design of the System

The conceptual design suggested the following structural components for the apparatus:

- Frame
- Slider crank mechanism
- Valve control mechanism

2.1. Frame

The frame is designed to rigidly hold all the fixed parts as shown in Figure 1. The frame is also designed to support the dynamic load that is occurred during the vibration of the spring-mass-damper system. The whole frame is constructed using Flex-link aluminum profile.

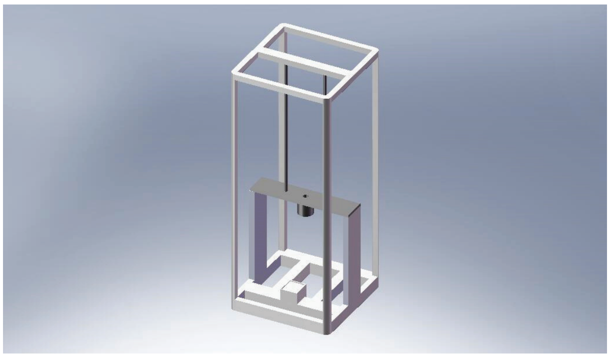

Fig. 1. Designed frame with slider bars

Figure 2 shows the semi-active suspension system with the particular elements.

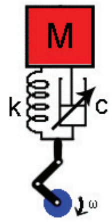

Fig. 2. Vibrating system with particular elements

2.2. Slider Crank Mechanism

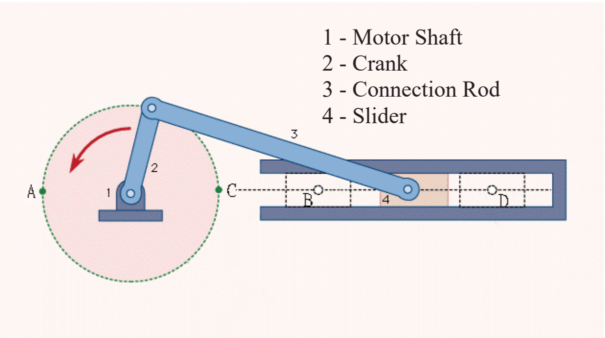

Fig. 3. Slider crank mechanism diagram

Slider crank mechanism is an arrangement of mechanical parts that are designed to convert rotary motion to straight-line motion. As shown in Fig. 3, when the motor shaft is turning, the crank will move in rotational motion while the connecting rod will push and pull the slider. The end of the connecting rod is connected to the slider. Its motion is restricted by the slider guide along a single line.

The slider displacement versus time is a sinusoidal wave with the peak-to peak amplitude equal to twice of the crank length and the frequency of the crank. The crank speed is equal to the motor speed. This mechanism functions as an exciter to the spring-mass--damper system. As the motor rotates up and down, the slider physically simulates a car as if it is going over a bumpy road. In addition, two slider bars are attached to guide the spring-mass-damper to move in a single line and to avoid side motion as highlighted in Fig. 4. Pneumatic cylinder is shown in Fig. 5.

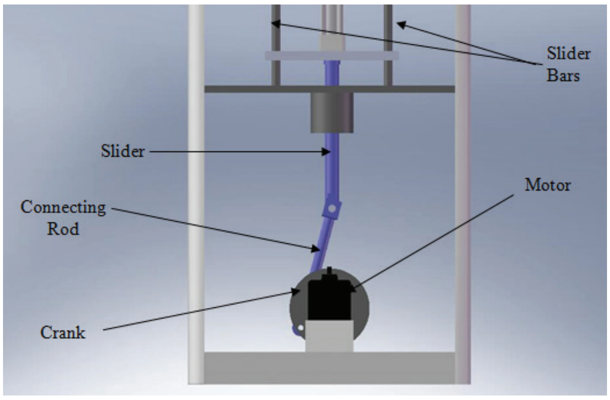

Fig. 4. Designed slider crank mechanism

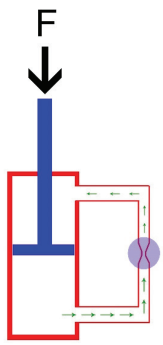

Fig. 5. Pneumatic cylinder is shown in figure 5

2.3. Selection of Parameters of the Dynamic System

The selection of mass and spring parameters is based on the parameter of the exciter motor. Calculation below shows how we determine the amount of mass and the spring stiffness for the apparatus.

Motor maximum torque = 1.8 Nm
Crank length = 0.03 m
T = rF, F=mg
T = rmg
m = T / (rg) = 6.12 kg

The rule of thumb is that the total mass lifted by the motor should not exceed more than 75% of the maximum mass it is capable to lift. Hence, applying Eqns. 1 and 2 we get,

$$m_{total} = m_{sprung} + m_{damper} \tag{1}$$

$$m_{sprung} = m_{total} - m_{damper} \tag{2}$$

$$m_{sprung} = (0.75 \times 6.12) - 1.4 \tag{3}$$

$$m_{sprung} = 3.2 \text{ kg} \tag{4}$$

To calculate spring constant, k, the mass is set to be m = 3.2 kg and 1.4 kg is the weight of the damper cylinder. The cylinder is a part of the unsprung mass. By setting the natural frequency to be half of the motor's speed, we derive the value of *K* as follows.

$$\omega_n = 2.5 \times 2\pi = 15.71 \text{ Radian/s}$$

$$\omega_n = \sqrt{\frac{k}{m}}$$

$$k = \omega_n^2 m = 15.71^2 \times 3.2 \text{ kg} = 790 \text{ N/m}$$

2.4. Valve Control Mechanism

The variable damper is shown in Fig. 6, consists of pneumatic cylinder and pneumatic valve. The pneumatic valve is designed to automatically control the damping factor (c) of the dynamic system. To achieve that, an actuator (stepper motor) is attached to turn the valve. The angular position of the valve knob is measured by a potentiometer that is attached as a feedback sensor. The transmission from the stepper motor to the valve knob is performed by the belt-pulley transmission mechanism. The timing belt is used to achieve the accuracy.

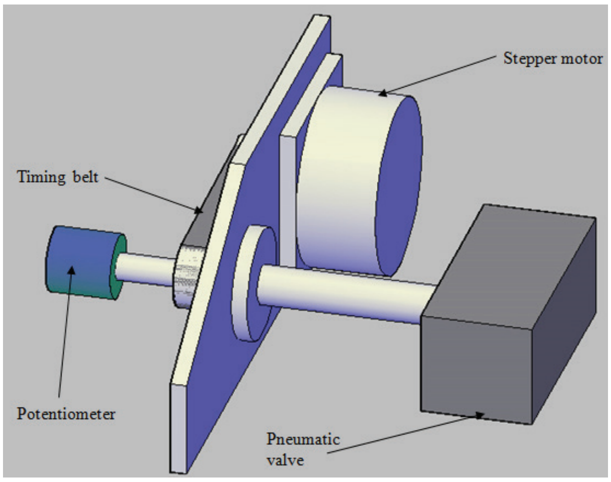

Fig. 6. Designed valve control mechanism

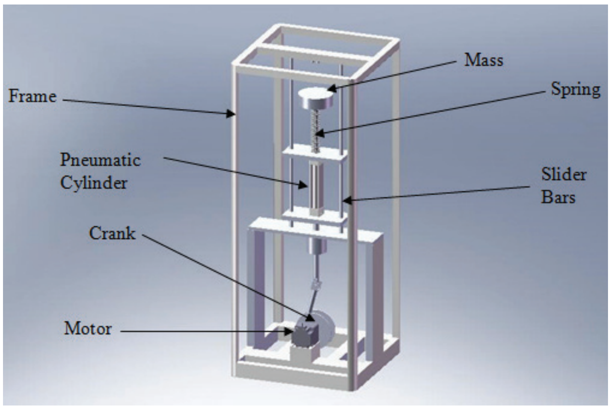

Fig. 7. CAD assembly of the system

The spring-mass-damper system is mounted on the frame and the slider is inserted into the slider bars to guide linearly the movement of the system. The CAD assembly of the system is shown in Fig. 7. The developed system is shown in Figs. 8 and 9.

The damper is connected in parallel with the spring, as sketched in Fig. 8. Frequency ωn is chosen to be at the half of the motor maximum speed (300 rpm). This is to make sure that the system is able to reach the natural frequency range in order to test the effectiveness of the suspension system. The maximum speed of the motor is 5 rev/s = 300 rpm. The system is forced with its natural frequency. This is to make sure that the system is able to reach the natural frequency range in order to test the effectiveness of the suspension system.

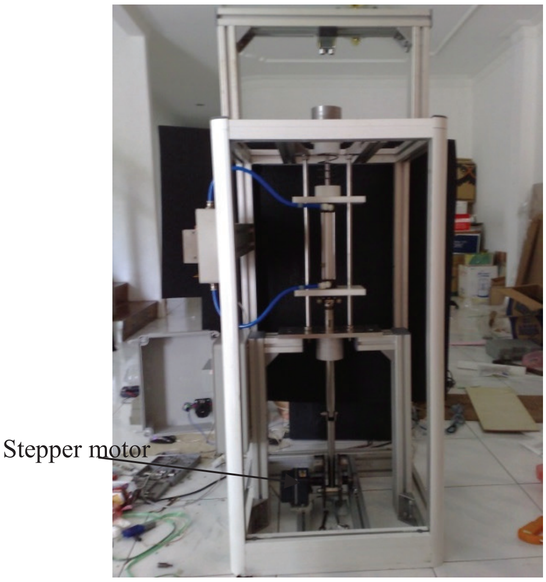

Fig. 8. Developed system: front view

Fig. 9. Developed system: enlarge view

2.5. Mathematical Model of Spring-Mass-Damper System

From the free body diagram shown in Fig. 10, the force produced by spring is calculated using Eqn. 3.

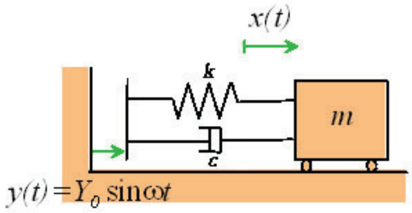

Fig. 10. Free body diagram of spring mass damper system

$$F = k(x - y) \quad (3)$$

Yes y is the kinematic excitation which is applied by the slider crank mechanism. And, the force produced by damper is calculated using Eqn. 5,

$$F = c\frac{d(x-y)}{dt} \quad (4)$$

$$F = c(\frac{dx}{dt} - \frac{dy}{dt}) \quad (5)$$

Applying Newton's second law, Eqns. 6 and 7 were derived.

$$m\frac{d^2x}{dt^2} = -k(x-y) - c\left(\frac{dx}{dt} - \frac{dy}{dt}\right) \quad (6)$$

$$\frac{m}{k}\frac{d^2x}{dt^2} + \frac{c}{k}\frac{dx}{dt} + x = y + \frac{c}{k}\frac{dy}{dt} \quad (7)$$

Taking natural frequency into consideration, the following Eqns. 8 and 9 can be obtained.

$$\omega_n = \sqrt{\frac{k}{m}}, \text{ and } \varsigma = \frac{c}{2\sqrt{km}} \quad (8)$$

$$\frac{1}{\omega_n^{\ 2}}\frac{d^2x}{dt^2} + \frac{2\varsigma}{\omega_n}\frac{dx}{dt} + x = y + \frac{2\varsigma}{\omega_n}\frac{dy}{dt} \quad (9)$$

In order to convert the equations from time domain to frequency domain, Laplace Transform is applied into both sides.

$$\frac{s^2X(s)}{\omega_n^{\ 2}} + \frac{2\varsigma\ sX(s)}{\omega_n} + X(s) = Y(s) + \frac{2\varsigma\ sY(s)}{\omega_n} \quad (10)$$

$$X(s)\left(\frac{s^2}{\omega_n^{\ 2}} + \frac{2\varsigma\ s}{\omega_n} + 1\right) = Y(s)\left(1 + \frac{2\varsigma\ s}{\omega_n}\right) \quad (11)$$

$$\frac{X(s)}{Y(s)} = \frac{1 + \frac{2\varsigma\ s}{\omega_n}}{\frac{s^2}{\omega_n^{\ 2}} + \frac{2\varsigma\ s}{\omega_n} + 1} \quad (12)$$

Sinusoidal transfer function can also be derived as in Eqn. 13.

$$\left|\frac{X(j\omega)}{Y(j\omega)}\right| = \frac{\sqrt{(1)^2 + \left(\frac{2\varsigma\,\omega}{\omega_n}\right)^2}}{\sqrt{\left(1-\frac{\omega^2}{\omega_n{}^2}\right)^2 + \left(\frac{2\varsigma\,\omega}{\omega_n}\right)^2}} \tag{13}$$

Let, $\frac{X}{Y} = K$ and $\frac{\omega}{\omega_n} = f_r$

Equations 14 to 20 are simplified equations. Eqn. 21 is used to plot the frequency response graph and observe the overall system performance.

$$K = \frac{\sqrt{(1)^2 + (2\varsigma\, f_r)^2}}{\sqrt{(1-f_r{}^2)^2 + (2\varsigma\, f_r)^2}} \tag{14}$$

$$K^2 = \frac{1+4\varsigma^2 f_r{}^2}{1-2f_r{}^2 + f_r{}^4 + 4\varsigma^2 f_r{}^2} \tag{15}$$

$$K^2 - 2K^2 f_r{}^2 + K^2 f_r{}^4 + K^2 4\varsigma^2 f_r{}^2 = 1 + 4\varsigma^2 f_r{}^2 \tag{16}$$

$$4\varsigma^2 f_r{}^2 K^2 - 4\varsigma^2 f_r{}^2 = 1 - K^2 - K^2 f_r{}^4 + 2K^2 f_r{}^2 \tag{17}$$

$$\varsigma^2 K^2 - \varsigma^2 = \frac{1-K^2(1+f_r{}^4 - 2f_r{}^2)}{4 f_r{}^2} \tag{18}$$

$$\varsigma^2\left(K^2 - 1\right) = \frac{1-K^2(1+f_r{}^4 - 2f_r{}^2)}{4 f_r{}^2} \tag{19}$$

$$\varsigma^2 = \frac{1-K^2(1+f_r{}^4 - 2f_r{}^2)}{4 f_r{}^2\left(K^2-1\right)} \tag{20}$$

$$\varsigma = \sqrt{\frac{1-K^2(1+f_r{}^4 - 2f_r{}^2)}{4 f_r{}^2\left(K^2-1\right)}} \tag{21}$$

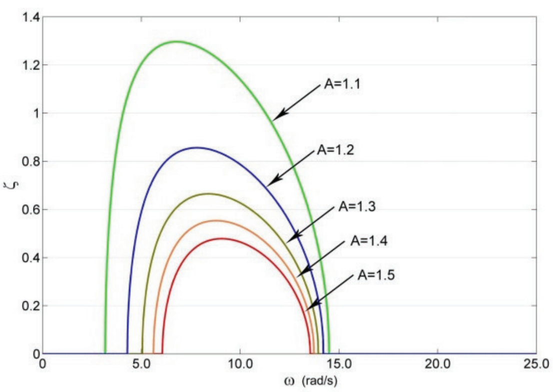

Fig. 11. Damping ratio vs excitation frequency for different amplitude

Figure 11 shows a specific relationship between damping ratio and oscillation amplitude. It shows that the possible values of damping ratio require to achieve the desired oscillation amplitude over a range of frequency (taking k=750N/m and m=3.2 kg).

2.6. Control of the Semi-active Suspension Apparatus

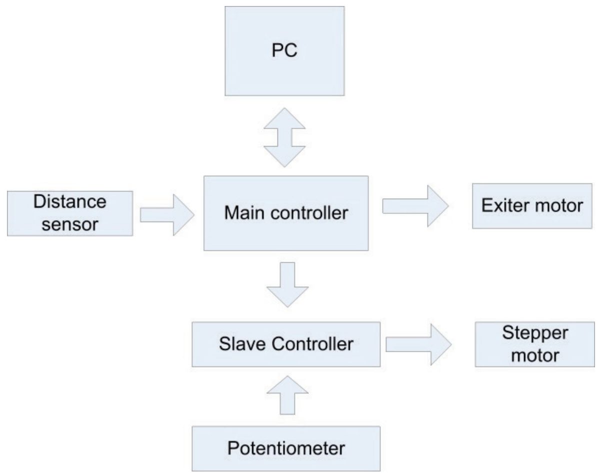

Fig. 12. Control implementation

Figure 12 shows the control system implementation flow chart. As shown in this figure, the system consists of two controllers such as; main controller and slave controller. The main controller is responsible for controlling exciter motor, reading data from sensors, controlling valve and communicate with PC. The slave controller is responsible for controlling the servo system of the electronic controlled valve. It reads the data sent by the main controller and adjusts

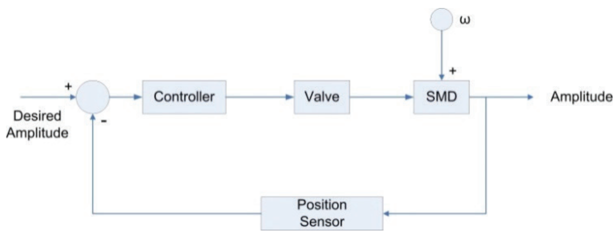

Fig. 13. Control system's block diagram

the opening of the valve with the help of feedback from the potentiometer. The system can be configured to run in manual mode or automatic mode. In the manual mode, user is able to control the damping ratio manually and the system measures and displays a plot of oscillation amplitude of the spring-mass-damper system over the applied frequency range. In automatic mode, the system adjusts the damping ratio automatically in order to achieve desired oscillation amplitude over the applied frequency range. To achieve the desired oscillation amplitude, a closed loop system with Proportional-Integral (PI) controller is proposed. It controls the position of the valve in relation to the damping ratio. Diagram in Fig. 13 shows the configuration of the closed loop control system.

2.7. System Data Communication

In this research, two different types of hardware communication were established.

2.7.1. Microcontroller-to-Microcontroller communication

The slave controller reads the data sent by the main controller as the input for the valve opening. To transfer data from one to another, two controllers were connected for effective communication purpose. A synchronous serial communication is developed to transfer the 10-bits data from the main controller to the slave controller. The communication port consists of three commands which are TRIGGER, CLOCK, and DATA. Figure 14 shows the main data communication circuit diagram for the complete system. The data transfer procedures are as follows:

i) *Main controller sends pulse to the TRIGGER pin*
ii) *Slave controller waits for falling edge pulse of the CLOCK*
ii) *Main controller sets the data on the DATA pin and generates falling edge pulse on the CLOCK*
iv) *Slave controller reads the DATA pin and saves the value into the memory*
v) *Steps (2–4) are repeated until 10 bits of data are transferred.*

2.7.2. Microcontroller-to-Computer Communication

In order to make the whole system to be controllable and observable from the GUI software, it is necessary to establish communication between computer and the main controller. After doing some researches regarding the communication between computer and hardware, it was decided to use USB communication. Universal Serial Bus (USB) is a serial bus standard to interface devices. A USB port was designed to allow peripherals to be connected using a single standardized interface socket, to improve plug-and-play capabilities by allowing devices to be connected and disconnected without rebooting the computer (hot swapping). Other convenient features include powering low-consumption devices without the need for an external power supply and allowing some devices to be used without requiring individual device drivers to be installed. USB is intended to use in helping serial and parallel port's data communication system.

3. Experimental Results and Discussion

The functional results of the built semi-active suspension system are shown in the figures below. The figures show the outcomes for various distinct operational conditions. Figure 15 shows the manual mode operation with the valve fully open. Figures 16 and 17 show the manual mode operation with 50% and 90% opening of valve, respectively. These results demonstrate a very high sensitivity of the system to the variable value of the damping factor (c) of the pneumatic damper. Figures 18 and 19 show the system operating in an automatic mode with the desired amplitude of 1.5 dB and 1.8 dB, respectively. The last two figures prove the ability of the system to control and maintain automatically the designed pick amplitude value of the mass-spring-damper system for the artificially injected external frequency disturbances.

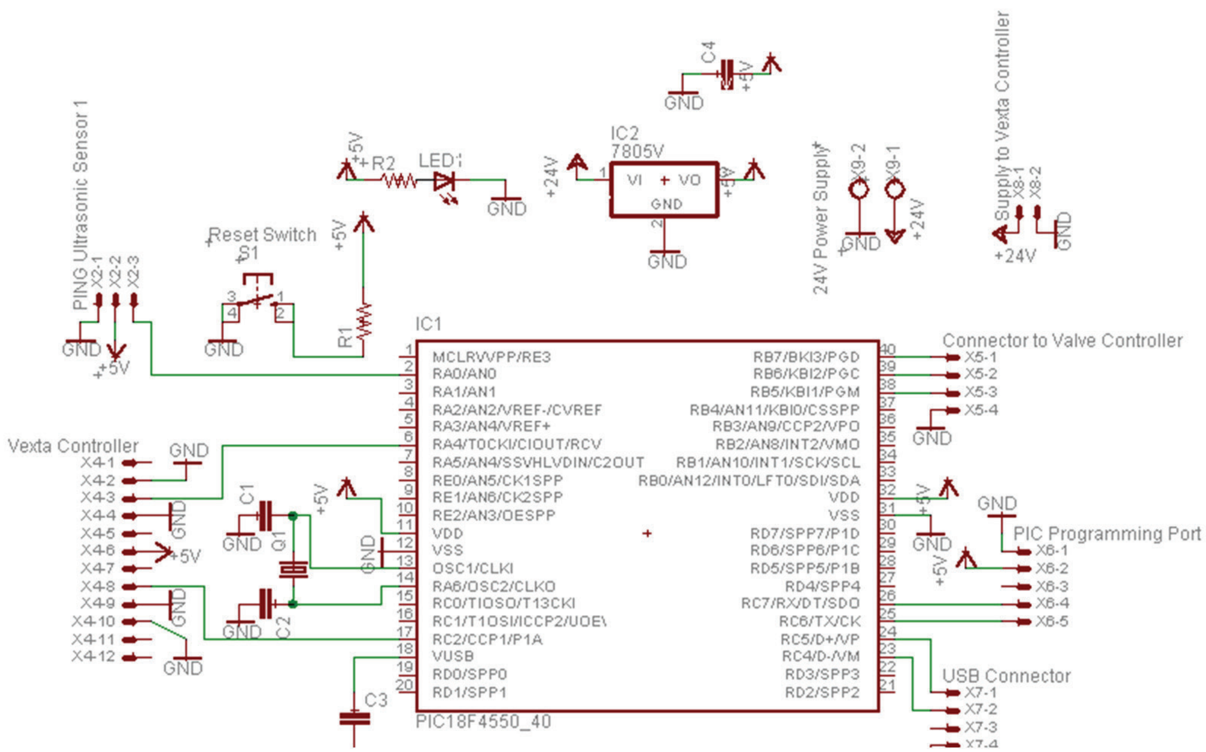

Fig. 14. Circuit connection with the controller

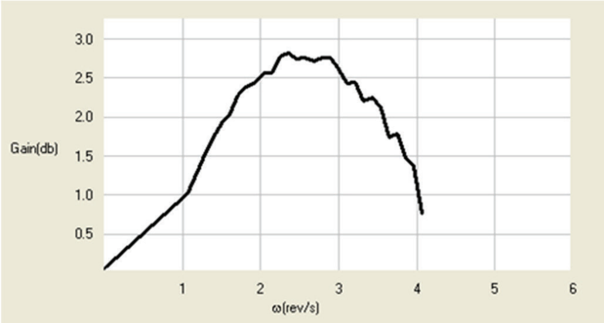

Fig. 15. Bode plot of manual mode operation (valve is opened fully)

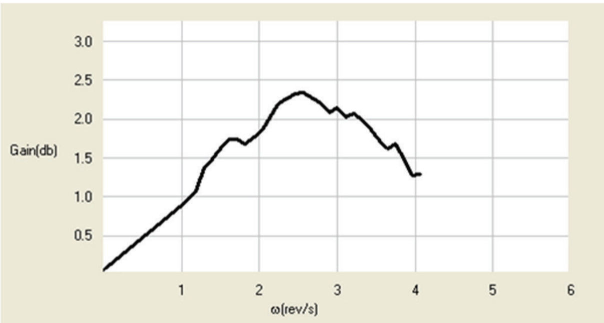

Fig. 16. Bode plot of manual mode operation (valve is 50% closed)

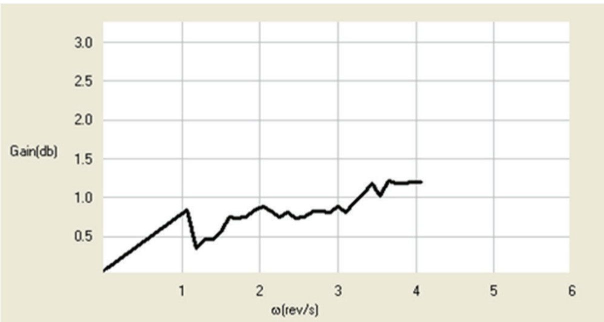

Fig. 17. Bode plot of manual mode operation (valve is 90% closed)

The result can be analyzed as follows. When the valve is fully opened, the vehicle's suspension system is uncontrolled and passengers feel the maximum level of jerking and discomfort. Whereas, when it is opened 50%, passengers feel less oscillation than the first case. Similarly, when the valve is closed 90% and opened 10%, passenger will feel oscillation but with less jerking and more comfort. These are the cases of manual control of the suspension system. Now, if we look at the results in Figs. 18 and 19, we can see the expected results produced by the developed system. The developed system follows the commands of the control input variables by adjusting damping factor (c).

4. Conclusions

The paper demonstrates the design and experimental development of the semi-active suspension system that simulates a quarter car suspension systems. The

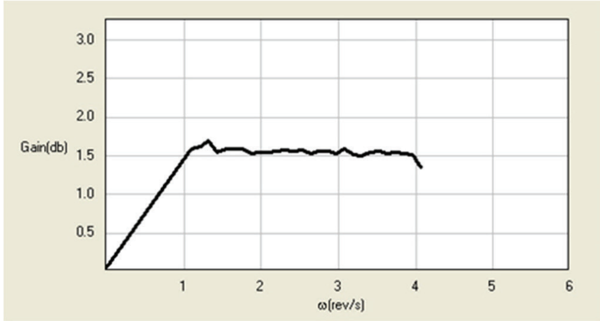

Fig. 18. Bode plot of automatic mode operation with desired amplitude at 1.5 dB

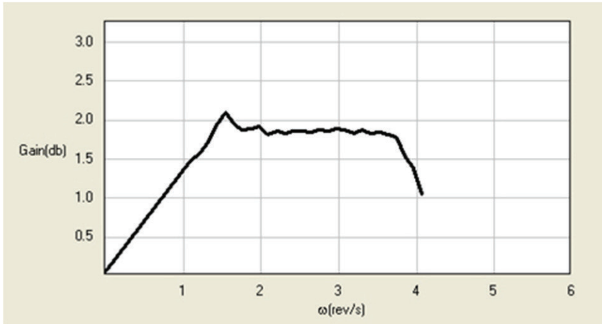

Fig. 19. Bode plot of automatic mode operation with desired amplitude at 1.8dB

main achievement of the work is the ability of the designed and developed system to adjust the amplitude of the system vibration regardless of external disturbances. If it is implemented in the cars, it will give safety and comfort to the car passengers in case of crossing bump and rough roads. The desired result is achieved by the real time tuning of the pneumatic damping coefficient for the sudden external disturbances. The paper presented the mathematical modelling of the system in terms of damping factor variation as well as it describes the complete construction and control of the suspension system. The experimental results obtained from the apparatus (Figs. 15–19) show the efficiency of the developed system and its compliance with the derived mathematical models.

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