Design of Magnetorheological Suspension of C-class car segment using Spencer model for education

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Abstract

In recent years, the search for new technology to increase comfort and stability in the vehicle with the latest technology is increasing. Therefore, various studies have been done. This research describes the development of a twin tube magnetorheological (MR) suspension system of a car model, Proton Preve. The development process includes designing a multiple degree-of-freedom mass-spring damper, the mathematical model involved, and the controller using Matlab/Simulink. The parameters involved, such as mass and damper spring resistance, were determined. Based on the mathematical model obtained, the system was designed in Simulink using the force disturbance input obtained from the data sheet. After successfully simulating the system, the PID controller was designed using Simulink and tested. The obtained results were compared to determine which parameters that best reduced vehicle vibrations. A prototype of the MR damper was built in Simulink, which was a major contribution of this research. The MR Proton Preve damper was designed according to the C-class specification. A quarter car system was designed following the equations of a two degrees-of-system. A PID controller simulation was performed and compared with the OEM C-class damper. In the next phase, a simulation was also conducted to aid in the design of the MR C-class *damper. The MR damper was a twin tube design, where the coil was integrated with the piston of the damper. In response to forces applied, the twin tube design allowed the piston to travel only within a fixed distance in the cylindrical reservoir. The twin tube design kept the MR fluid within the reservoir. The simulation showed that the new MR suspension system with an Arduino PID microcontroller was effective for use in the C-class vehicle. At this condition, the displacement amplitude of the sprung mass was less than 10 mm, which satisfies the comfort category in the ISO8608 standard.*

Keywords: *Magnetorheological damper design, Spencer model, Matlab, Simulink, PID controller, C-class segment.*

Introduction

A suspension is a structural arrangement on wheels that accommodates the weight of a vehicle and provides comfort when a vehicle moves. Vehicles without suspension are termed 'bone breakers', as the vehicle would shake and wobble with severe vibrations that result in discomfort during the ride. Thus, each vehicle must be equipped with a suspension system to absorb vibrations, especially on uneven road surfaces and to support heavy passenger weights. In this thesis, a 'smart' suspension system using fluid 'magneto rheological' (MR) technology is analysed. The semi-active suspension system was built to suit a C-class vehicle, with a damper based on LORD Corporation's model. The input excitation to the suspension was the actuators generating a wavering force [1]. The actuators were adjusted to fit conventional local car springs and an MR damper was utilized for the vertical connection in the two degree-of-freedom (2DOF) system. The non-linear behaviour of smart fluid dampers provided a major benefit in the development of effective control strategies [2]. The characteristics of the suspension system were taken from the shock absorber test machine experiments. The parameters used in the system, included the body mass m_1, tyre mass m_2, and spring and damping coefficients, as shown in Table 1

Table 1

Parameter of the C-class quarter car model [3]

Parameter	Symbol	Value
Mass of the body	m_1	249 kg
Mass of the tyre	m_2	25.1 kg
Suspension damping coefficient	b_1	102 $N.s.m^{-1}$
Tyre damping coefficient	b_2	14999 $N.s.m^{-1}$
Spring stiffness	k_1	17600 $N.m^{-1}$
Tyre damping coefficient	k_2	14999 $N.m^{-1}$

MR Damper of the C-class Model

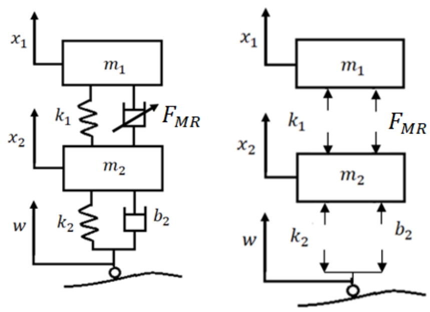

Figure 1. A 2DOF quarter car model

The equations of vertical motion for the quarter car suspension system, which are the sprung mass (quarter car body mass) and the unsprung mass (suspension parts), can be represented by applying the Newton's second law of motion with the help of the following mathematical equations [1] for the quarter car model with the MR suspension system. The MR damper has a variable F_{MR} .

$$m_1\ddot{x}_1 + k_1(x_1 - x_2) + F_{MR}(t) = 0 \quad (1)$$

$$m_2\ddot{x}_2 + F_{MR}(t) + k_1(x_2 - x_1) - b_2\dot{x}_2 - k_2 x_2 - w(t) = 0 \quad (2)$$

This model is devoted to the tuning of the parameters applied for the quarter car suspension. The schematic model i.e. Spencer model is given by the following equation:

$$F_{MR} = \alpha z + c_0(\dot{x} - \dot{y}) + k_0(x - y) + k_1(x - x_0) = c_1\dot{y} + k_1(x - x_0) \quad (3)$$

The simulation of an MR damper system was considered for a quarter car model. The effects obtained can then be topped up and attuned accordingly for the whole car. A quarter car is made up of a wheel, a flexible damper and a spring set. The mass of the suspension system can be categorized into sprung and unsprung masses. The sprung mass includes the mass of the body which is hampered by the suspension system (in the case of a quarter car model, the value is one-fourth of the total sprung mass value). Figure 2 indicates the Spencer model of the MR damper for the 2DOF suspension system. The unsprung mass contains the mass of the wheels, brakes, suspension mechanisms, etc[4].

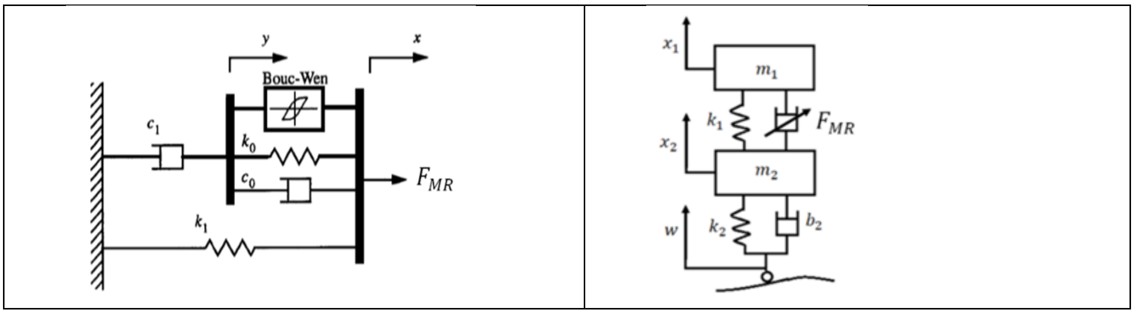

Figure 2. A 2DOF MR Spencer model

Table 1

Parameter of Spencer for MR damper

Parameter	Symbol	Value
Coefficient a	c_{0a}	21 N.s/cm
Coefficient b	c_{0b}	3.55 N.s/cm.V
Spring stiffness	k_0	45 N/cm
Coefficient 1a	c_{1a}	281 N.s/cm.V
Coefficient 1b	c_{1b}	3 N.s/cm.V
Spring stiffness	k_1	5.12 N/cm
Damper stroke	x_0	14.0 cm
Alpha, a	α_a	140 N/cm

MR damper with Spencer model

The closed-loop transfer function in Matlab represented the main system of the plant, road disturbance, and controller [5]. The closed-loop step response for this system started with the control process to simulate a 5 cm bump on the road profile input. Referring to the equations from the Simulink block diagram, the system was designed for the quarter car model using the models from Figure 1 and Figure 2. Figure 3 shows the Simulink illustration model of the MR Spencer damper [6, 7]. The system also represented sprung and unsprung masses. The road profile of input excitation is shown in Figure 4, which was used in this simulation. The PID controller system was also built for this system[8]. It controlled the suspension response at different conditions. Trial and error tuning was performed and results were obtained[9, 10].

To tune the PID, the following steps were taken. Firstly, all gains were fixed to zero. The P gain was then improved until the reaction to a disruption was a stable oscillation. The D gain was improved until the oscillations were as low as possible (i.e. critically damped). The previous two steps were continual until the cumulative D gain did not cease oscillations. P and D were then fixed to the latter stable values. The I gain was increased until it was carried to the set point with the chosen number of oscillations (normally zero but a quicker response could be had and would require overshoot). If the oscillations continuously increased, the P gain was reduced. Next, the D gain setting was too high and the system began to chatter (vibrated at a higher frequency than the P gain oscillations). To overcome it, the D gain was reduced until the oscillations ceased.

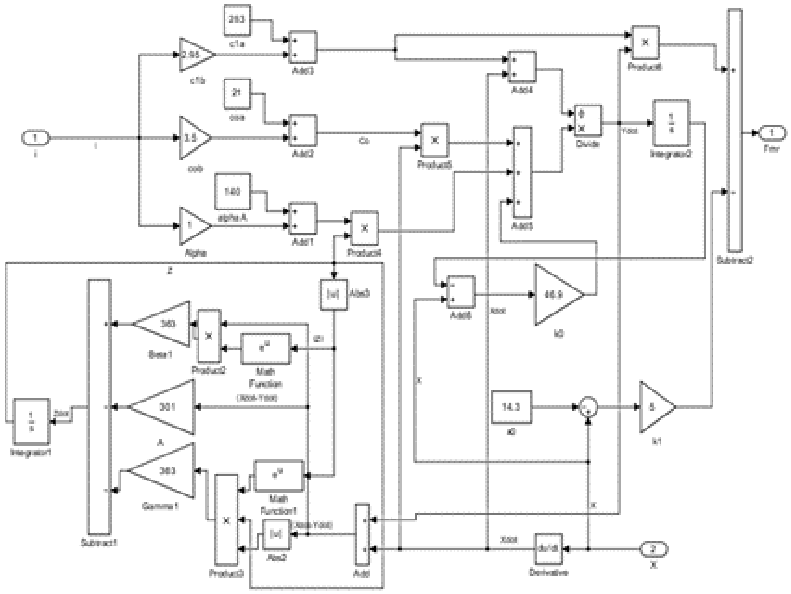

Figure 3. Spencer Model of the MR Damper for the 2DOF suspension system in Simulink.

Simulation Results

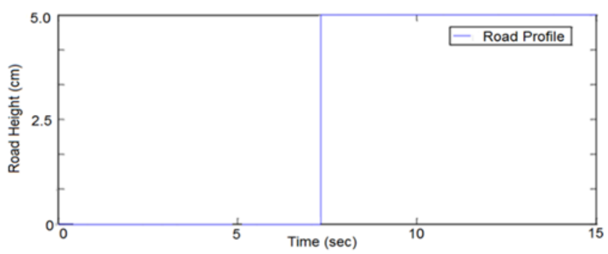

Figure 4. Road profile with a 5 cm step

The simulation was performed in Matlab-Simulink to validate the non-parametric information from the MR Lord damper [11]. The main purpose of the simulation was to determine the general behavior of the MR damper before the experiment was conducted. Suggestions were then made from the experimental pre-yield and post-yield data [12]. Two types of road profiles (input signals) were tested; however, step profiles were used in the simulation work, as shown in Figure 4. These step profiles had a 5 cm height and were similar to that used in the experiment. Each simulation produced different types of results depending on the K_p, K_i and K_d values. These values were chosen by trial and error. Although the tuning process of the PID controller was very challenging, the performance of the quarter car system could be increased once the controller was optimized. Figures 5, 6, 7, 8 and 9 show a subset of the best results.

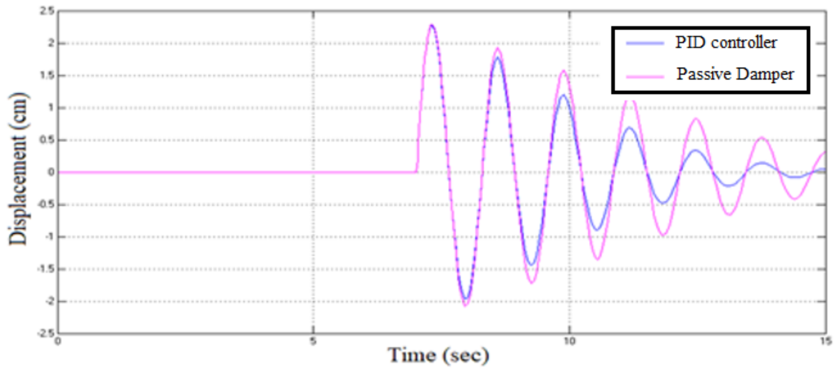

Figure 6. Sprung mass displacement vs time, PID : K_p=40, K_i =40 and K_d =25 vs passive damper

Figures 9 shows a subset of the best results. The best single result for this simulation was obtained with K_p = 15, K_i = 10 and K_d = 5, as illustrated in Figure 6. The sprung mass displacement was 0.7 cm, as compared to Figure 10 where the sprung mass displacement was 2.5 cm. The ability of the PID controller to reduce the vibration level was proven in this simulation. By comparing the results, it can be concluded that the PID controller provided the best vibration control, as compared to the passive suspension system.

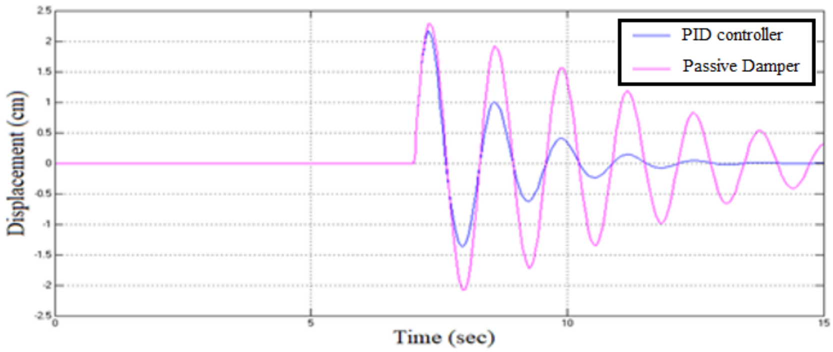

Figure 7. Sprung mass displacement vs time, PID : K_p=15, K_i=20 and K_d=5 vs passive damper

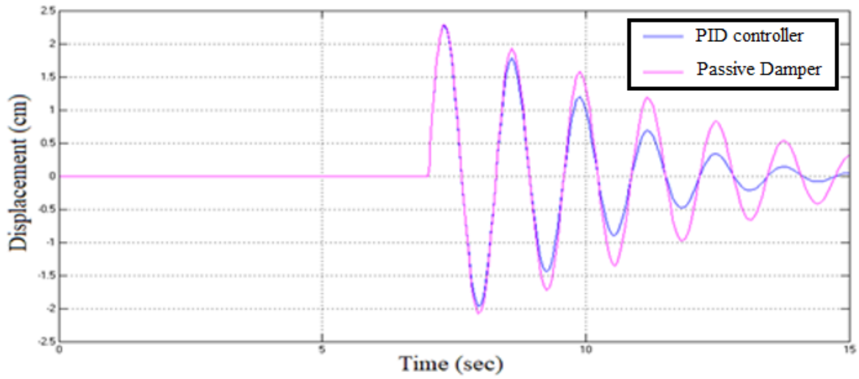

Figure 8. Sprung mass displacement vs time, PID : K_p=15, K_i=15 and K_d=5 vs passive damper

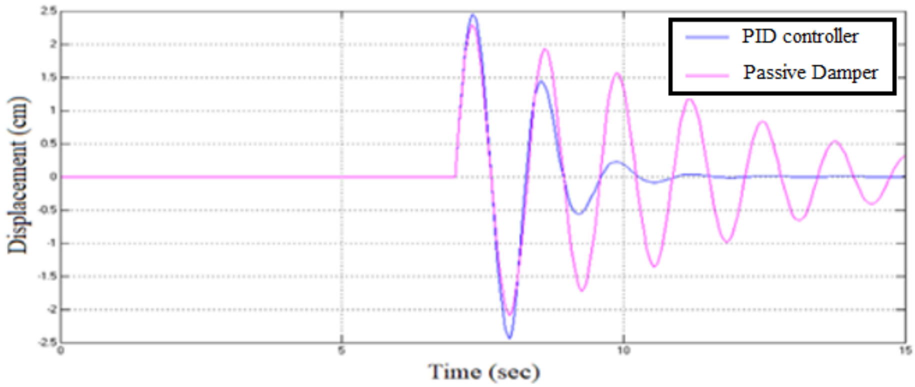

Figure 9. Sprung mass displacement vs time, PID : K_p=15, K_i=10 and K_d=5 vs passive damper

Conclusion

An MR suspension with a PID control system was planned and applied both mathematically in Simulink and experimentally on the particularly designed suspension test rig [13]. An effective validation was achieved on all scheme parameters in order to relate the developed model in contradiction of the trial setup. The obtained results showed that the semi-active suspension with the PID controller reduced the road input disturbance by up to 80% for a 5 cm step input. The simulations were conducted using a mathematical model and the quarter car model system was built using Matlab and Simulink [14]. Together with this system, the PID controller was built. The output data obtained from the road profile and the sprung mass displacement were plotted against time. The suspension system was stable when the K_p value was tuned from 50 to 15. The best result was obtained with K_p=15, K_i =10 and K_d= 5, as illustrated in Figure 9. This simulation result was used as a reference before the actual experiment was performed using the ARDUINO MEGA microcontroller system.

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