Semi-Active Suspension Control for Formula SAE Car using Magneto-rheological Fluid

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***Abstract*—The Magneto-rheological (MR) fluid damper is prevalent in the field of semi-active suspension whose viscosity changes by the change of magnetic field passing through the damping fluid. In this study, a semi-active suspension quarter car model is employed as a plant. The Bingham model of MRF damper is exploited with PID and Fuzzy + PID controllers. The current is controlled by the controllers according to the quarter car chassis disturbance. The step road profile is used as an input disturbance to the suspension system. The displacement of sprung mass is analyzed in terms of time and frequency domain. The maximum power spectral density of acceleration for step response with Fuzzy + PID is reduced by 87.28 % as compared to passive suspension whereas PID reduced only 79.95 %. This indicates that the MRF damper with right tuned Fuzzy + PID controller provide a safer ride compared to PID controller and passive suspension.**

***Index Terms*— fuzzy logic, magneto-rheological fluid damper, PID controller, semi-active suspension.**

I. INTRODUCTION

In 21st century, vehicles rely on various electronic control systems to accomplish safety, driving comfort, driving pleasure and enhanced ride and handling. Control of modern vehicles includes braking control, traction control, acceleration control, lateral stability control, suspension control and so forth.

Generally, there are two types of control system for suspension. First is active suspension control system and second is semi-active suspension control system. Active suspension control system require substantial amount of input energy to produce desired output control forces. Moreover, a fully active suspension system does provide high performance than passive suspension system but it requires complex control system and higher cost, with large consumption of energy and non-trivial reliability issues. Semi-active suspension, like passive suspension, consists of a damper and a spring but the damping coefficient of a damper is controllable [1]. Semi-active suspension system offer reliability as compared to that of passive system, yet maintaining the versatility and adoptability of fully active systems without using large energy input.

Semi-active suspension system cannot inject mechanical energy into the controlled system as no hydraulic actuators are involved as in case of active suspension therefore, they do not have the potential to destabilize. Magneto rheological (MR) fluid is an example of semi-active suspension material that gives a better performance to the vehicle suspension system. Bode, O et al. studied the experimental model of two degree heavy vehicle using semi active suspension and founded that for a ramped road input of height 50 mm, the peak vertical acceleration and displacement of a body reduced by 20 % and 25 % respectively as compared to normal passive suspension [2].

PID controller is a basic control system used for controlling the output and various algorithms have been developed to obtain a desired output damper force. The simulation study of magneto-rheological (MR) damper and hydraulic actuator for suspension system using intelligent PID controller with iterative learning algorithm has been presented by Talib and Darus [3].

Fuzzy logic control algorithm also used in controlling the semi-active suspension system. Lotfi Z. first proposed this fuzzy logic algorithm and come out to be powerful tool for control system [4]. In order to do modulation of damping coefficient, a fuzzy logic controller is introduced into the design scheme. Fuzzification interface, fuzzy rule base, decision making and defuzzification interface are the major blocks of a fuzzy logic system [5]. The simulation for MR semi-active suspension by using Fuzzy with PID has also been done by Mohammad M. F., in Simulink/MATLAB software [6].

This paper discussed the performance of fuzzy logic integrated with PID controller in the semi-active suspension control for an SAE car suspension system. Simulations are used to be performed on MATLAB/ toolbox for theoretically observe the response of the control system.

II. QUARTER CAR MODEL OF SEMI-ACTIVE SUSPENSION SYSTEM

The quarter car model of semi-active suspension system is shown in Fig. 1.

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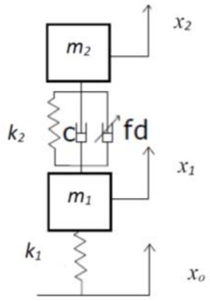

Fig. 1. The quarter car model of semi-active suspension system [7]

where,
m_1 is the unsprung mass which is the mass of wheels and associated components
m_2 is the sprung mass which is the vertical mass with passenger
x_2 is the vertical displacement of the sprung mass
x_1 is the vertical displacement of the unsprung mass
x_0 is the displacement of the road disturbance
k_2 is the suspension spring constant
k_1 is the tire spring constant of tire
f_d is the damping force of MR damper

The equation of motion of the quarter car model is obtained by applying Newton's second law to the free body diagrams of Fig. 1.

$$m_1\ddot{x}_1 = -k_1(x_1 - x_0) + c(\dot{x}_2 - \dot{x}_1) + k_2(x_2 - x_1) - f_d \tag{1}$$

$$m_2\ddot{x}_2 = -c(\dot{x}_2 - \dot{x}_1) - k_2(x_2 - x_1) - f_d \tag{2}$$

where,
$\ddot{x}_1$ is acceleration of unsprung mass
$\ddot{x}_2$ is the acceleration of sprung mass
$\dot{x}_1$ is the velocity of unsprung mass
$\dot{x}_2$ is the velocity of sprung mass
f_d is the damper force

Based on eq. (1) and (2), the state space representation of car suspension system is [7]:

$$\dot{x} = \begin{bmatrix} \dot{x}_1 \\ \dot{x}_1 \\ \ddot{x}_2 \\ \ddot{x}_2 \end{bmatrix} = \begin{bmatrix} 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \\ \frac{-k_1-k_2}{m_1} & \frac{k_2}{m_1} & \frac{-c}{m_1} & \frac{c}{m_1} \\ \frac{k_2}{m_2} & \frac{-k_2}{m_2} & \frac{c}{m_2} & \frac{-c}{m_2} \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \\ x_3 \\ x_4 \end{bmatrix} + \begin{bmatrix} 0 & 0 \\ 0 & 0 \\ \frac{-k_1}{m_1} & \frac{-1}{m_1} \\ 0 & \frac{-1}{m_2} \end{bmatrix} \begin{bmatrix} x_0 \\ f_d \end{bmatrix} \tag{3}$$

$$y = \begin{bmatrix} \frac{k_2}{m_2} & \frac{-k_2}{m_2} & \frac{c}{m_2} & \frac{-c}{m_2} \\ -1 & 1 & 0 & 0 \\ k_1 & 0 & 0 & 0 \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \\ \dot{x}_1 \\ \dot{x}_2 \end{bmatrix} + \begin{bmatrix} 0 & \frac{-1}{m_1} \\ 0 & 0 \\ -k_1 & 0 \end{bmatrix} \begin{bmatrix} x_0 \\ f_d \end{bmatrix} \tag{4}$$

The MR damper model used in this study was Bingham model which shows nonlinear behavior of a MR viscous fluid passing through orifices. The schematic diagram of Bingham model is shown in the Fig. 2.

There is a Coulomb friction element f_c, in the Bingham model of MR damper which is parallel to the dash pot, c_o. For non-zero piston velocities $\dot{x}$, the damping force f_a can be expressed according to Bingham model as;

$$f_a = f_c \operatorname{sgn} \dot{x} + c_0\dot{x} + f_0 \tag{5}$$

where,
f_c is the frictional force
f_a is the MRF damper force
c_o is the viscous damping parameter
f_o is the force due to presence of accumulator
$\dot{x}$ is the piston velocity

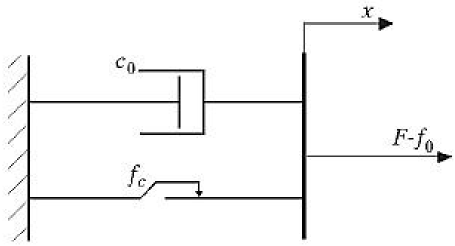

Fig. 2. Bingham model of MRF damper

The frictional force, f_c and viscous damping, c_o are dependent on the value of current, I (A). Following are the equations used to relate frictional force, f_c and viscous damping, c_o with current,

$$f_c(I) = -910.09I^3 + 986.49I^2 + 663.56I + 52.19 \tag{6}$$

$$c_0(I) = 48.74I^4 - 106.39I^3 + 66I^2 + 1.43I + 0.53 \tag{7}$$

where I is the current of damper coil (A)

The front passive suspension system of the SAE formula car is shown in Fig. 3. The top part of suspension system had attached to chassis and lower part attached to wheel through wheel hub, A-arms and connecting roads. The chassis was referred to sprung mass whereas; wheel and other assemblies were considered as unsprung mass. The front suspension system of the formula SAE car is correctly represented by quarter car model.

Fig. 3. Front passive suspension system of Formula SAE car

III. Simulation Study

The basic block diagram of a semi-active suspension system with MR damper, PID controller and Fuzzy logic controller is shown in Fig. 4. Feedback from the output of the plant was used to calculate the error between the input and the output so that current, I (A) can be determined accordingly by the controller. The force (N) will then be exerted on the MR damper to attenuate the vibration of the quarter car suspension plant.

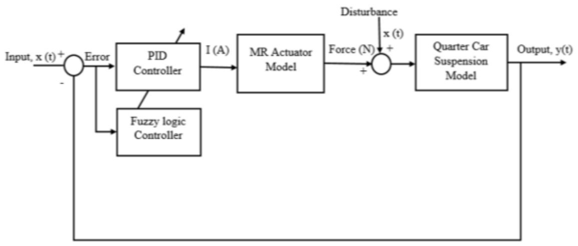

Fig. 4. Block diagram of MR fluid semi-active suspension system

The simulation for the MR semi-active suspension has been conducted using MATLAB (2012b) with Simulink toolbox. Prior to simulation, all parameters used in the mathematical model were defined and are shown in Table I. The input of the semi-active suspension system was the MR damper force (N) and the road disturbance (m). The output were the acceleration of the sprung mass, $\ddot{x}$, the change of displacement of sprung and unsprung mass ($x_2 - x_1$) and the force, $k_1(x_1 - x_o)$ exerted due to the tire stiffness. The acceleration was converted into displacement, x_2 to find the error of the system. Simulation without controller was conducted using values of current $I = 0$ A to represent passive suspension system. Fig. 5 shows the simulation arrangement of semi-active suspension system for a formula SAE car by using Fuzzy with PID controller. The simulation was performed for 5 seconds. The system composed of state space model of a quarter car semi-active suspension model with Bingham model for MR damper to represent damper force. The input for the Bingham model for MR damper was based on variable of current, I (A). In this study, the current was reduced by both the PID controller and the PID with fuzzy logic controller.

TABLE I. Parameter Values for a Quarter SAE Car

Parameter	Value (Unit)
Unsprung mass, m_1	10 kg
Sprung mass, m_2	110 kg
Tire stiffness, k_1	176000 N/m
Suspension spring stiffness, k_2	95330 N/m
MR damper coefficient, c	230 Ns/m
MR damper constant force, f_0	40 N

The state space model of a quarter car model was the convenient way of expressing the system because of the versatility and the flexibility as the number of output can be selected as desired. There were two inputs of state space model, which were road profile (m) and damper force (N) as mentioned previously. The road profile excited the two degree of freedom systems and the force from damper would be used to dampen the system vibration. The damper force from MR damper varied accordingly depending on the vibration of the chassis. Three outputs of the system were taken into account which were the acceleration of chassis, the difference of displacement between sprung mass and unsprung mass and the force experienced due to the tire stiffness. The acceleration was converted into displacement to ease the understanding of the system behavior with different controllers.

After 5 seconds simulation, the signal from the scope showed the output displacement which will be controlled. PID tuning using Ziegler Nichols method based on the frequency response of the closed-loop system determined the point of marginal stability under pure proportional control. Firstly, the gain of the system (k_p gain) was set to zero. Then, the k_p gain was increased until it reached the ultimate gain (k_u): a situation where the output of the loop from signal scope starts to oscillate at constant amplitude. Finally, the oscillation period, p_u (peak-to-peak) was used to set the gains as shown in Table II.

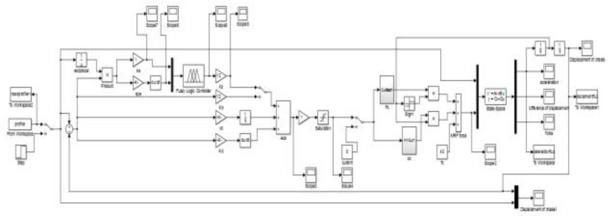

Fig. 5. Simulation arrangement of semi-active suspension SAE car system by using Fuzzy with PID controller

TABLE II. PID Controller Tuning Ziegler-Nichols Method

Control Type	Proportional (k_p)	Integrative (k_i)	Derivative (k_d)
PID Controller	$0.6k_u$	$t_u/2$	$t_u/8$

Fuzzy Logic Controllers (FLCs) have been recognized as being a capable methodology for designing robust controllers which been able to provide a satisfactory performance when dealing with the uncertainty and imprecision attributed to the real world [8]. One of the effective member function used in fuzzy logic is triangular type member function.

The fuzzy logic controller required two inputs which were relative error of the system and the change of relative error. The fuzzifier factors denoted by *ke* and *kde* were used for two inputs of fuzzy controllers, so that the inputs were within the defined range of membership functions. After simulation, the values selected for fuzzifier factors were $ke = 1$ and $kde = 0.09$. Similarly, defuzzifier factor value was selected to convert the output of the fuzzy controller into the required range of PID controller. The selected value of defuzzifier factor was, $kf = 12$. The basic rules for fuzzy inference system had been made based on Macvicar Whelan matrix. The Macvicar Whelan matrix is shown in Table III.

TABLE III. BASIC RULES TABLE FOR FUZZY INFERENCE SYSTEM

	Change of Error, *de*					
		NB	NS	Z	PS	PB
Error, *e*	NB	NB	NB	NS	NS	Z
	NS	NB	NS	NS	Z	PS
	Z	NS	NS	Z	PS	PS
	PS	NS	Z	PS	PS	PB
	PB	Z	PS	PS	PB	PB

Linguistic terms assigned to these fuzzy sets are NB, NM, NS, Z, PS, PM, PB refer to negative big, negative medium, negative small, zero, positive small, positive medium, and positive big, respectively.

The ranges of both input and output membership function were [-1, 1]. The range of input was selected [-1, 1] because relative error ranges from 0 to 1 for positive relative error, and from -1 to 0 for negative relative error. Similarly, the output range was also selected [-1, 1] as the output could easily be multiplied by the required defuzzifier factor.

The sample time was set 0.001 seconds with fixed step. The amplitude of the step road profile was set 0.1 m at 1 second. The response of passive, PID and Fuzzy + PID controller had been recorded with step response.

IV. RESULTS AND DISCUSSION

Ziegler Nichols method was used to tune the PID controller. It was based on the frequency response of the closed-loop system by determining the point of marginal stability under pure proportional control. The first step was required to increase the proportional gain which was denoted by k_u and keeping integral, k_i and differential, k_d gains equal to zero. From simulation, the value found was $k_u = 20$ and $t_u = 0.02$ s. The k_u values was substituted in the Ziegler Nichols parameters as shown in Table II to obtained the k_p, k_i and k_d parameters. As a result, the optimized values of the PID controller parameters were found as the Table IV.

TABLE IV. PARAMETERS OF PID CONTROLLER

Control Type	**Proportional (k_p)**	**Integrative (k_i)**	**Derivative (k_d)**
PID Controller	12	0.01	0.0025

The result obtained from the passive, PID and Fuzzy + PID controllers had been plotted in same graph in order to compare each other with passive system as reference. Figs. 6 and 7 have shown the chassis displacements (m) with time (s) and power spectral density $(m/s2)^2$/Hz with frequency (Hz) respectively.

From Table V, it is clearly shown that the Fuzzy + PID controller has showed outstanding performance in reduction of vibration. The maximum percentage of reduction of power spectral density was 87.28 % which is of Fuzzy + PID by taking passive system as benchmark. Similarly, Fuzzy + PID also gave satisfactory result in term of percentage of reduction of time to damp being highest with 72.50 %.

The Fuzzy + PID controller reduced the vibration more than the PID controller because fuzzy logic chose the appropriate value of k_p according to relative error with time and thus accurate current was being sent to Bingham model for desired force whereas, PID controller had constant parameters and thus, output was determined only by change of error without any change of parameter for suitable response.

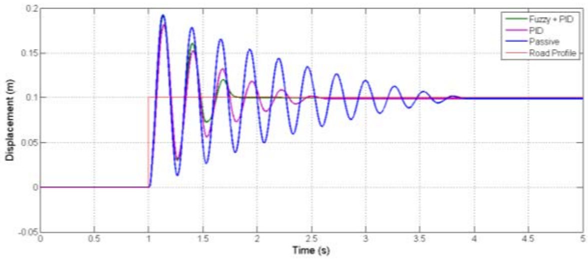

Fig. 6. Performance comparison between passive, conventional PID and Fuzzy-PID controllers (Time Domain)

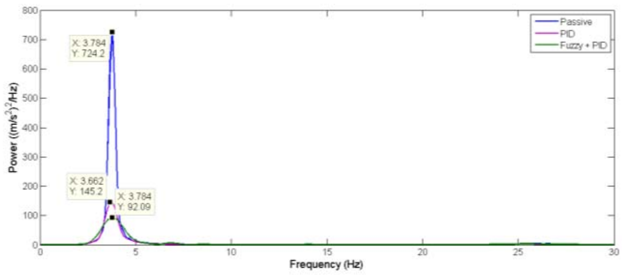

Fig. 7. Performance comparison between conventional PID and Fuzzy-PID controllers (Frequency Domain)

V. CONCLUSION

In conclusion, the viscosity effects of a MR fluid damper showed a significant improvement in the vibration of a car. The Bingham model of MR fluid damper was used in the simulation which depends on the input current. The PID and Fuzzy + PID controllers were used to control the current based on the displacement of the car chassis. Findings show that the Fuzzy + PID controller performed better in reducing the vibration of the system as compared to PID controller. Hence, semi-active suspension with Fuzzy + PID controller offered a better suspension performance, therefore provides a much safer ride to the passengers and more stability to the vehicle structure.

TABLE V. THE PERFORMANCE COMPARISON OF PID AND FUZZY-PID CONTROLLERS SYSTEM OF QUARTER CAR MODEL WITH PASSIVE SUSPENSION SYSTEM USING STEP RESPONSE

Controller Types	**Displacement (m)**	**Oscillations**	**Time to damp (s)**	**Power Spectral Density (($m/s^2)^2$/Hz)**	**% Reduction of time to damp**	**% Reduction of Power Spectral Density**
PID	0.1813	6	1.596	145.2	45.08	79.95
Fuzzy + PID	0.1906	3	0.799	92.09	72.50	87.28

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