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Fuzzy-PID Controller for Semi-Active Vibration Control Using Magnetorheological Fluid Damper

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Abstract

Magnetorheological (MR) dampers are considered as excellent prospect to the vibration control in automotive engineering. To overcome the effect from road disturbances many control algorithms have been developed and opted to control the vibration of the car. In this study, the methodology adopted to get a control structure is based on the experimental results. Experiments have been conducted to establish the behaviour of the MR damper. In this paper, the behavior of MR damper is studied and used in implementing vibration control. The force-displacement and force-velocity response with varying current has been established for the MR damper. The force for the upward motion and downward motion of damper piston is found increasing with current and velocity. In the cycle mode which is the combination of upward and downward motion of the piston, the force having hysteresis behaviour is found increasing with current. Results of this study may serve to aid in the modelling of MR damper for control applications.

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Keywords: Magnetorheological fluid, MR damper, semi-active vibration, Fuzzy-PID

Nomenclature

m	Mass
k	Spring constant
d	Damping constant
x	Distance
f	Force

1. Introduction

There are mainly three types of control devices: passive devices, active devices and semi-active devices. Semi-active devices offer the versatility and the adaptability of active device, and the reliability of the passive device. They can operate with power supply (active device) and without power supply (passive device). Magnetorheological (MR) dampers, variable orifice dampers and tuned liquid dampers are examples of semi-active devices.

MR fluids have been regarded as a smart material. Varying external magnetic field strength is used to vary and control the rheological properties of MR fluids. A typical MR fluid contains 20–40% by volume of relatively pure, soft iron particles having a dimension of 3 to 5 microns. These particles are suspended in mineral oil, synthetic oil, water, or glycol. A variety of proprietary additives similar to those found in commercial lubricants are commonly added to discourage

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gravitational settling and promote particle suspension, enhance lubricity, modify viscosity, and inhibit wear [1]. In the absence of an applied field, MR fluids exhibit Newtonian-like behaviour. However, in the presence of an applied magnetic field, the iron particles acquire a dipole moment aligned with the external field which causes the particles to form linear chains parallel to the field, as shown in Figure 1(a). This phenomenon can solidify the suspended iron particles and restrict the fluid movement. MR fluid damper or MR damper consists of MR fluid inside and it uses the properties of MR fluid in controlling the vibration of a system. The magnitude of force that an MR damper can deliver depends on the properties of MR fluids, their flow pattern, and the size of the damper. Figure 1(b) represents the structure of MR damper.

Recently, MR fluid dampers have been widely applied to control and suppress unwanted vibration and shock for various systems including-landing gear, helicopter lag dampers, vibration isolation systems, vehicle seat suspension systems, civil structures, military equipments, prosthetic limbs. MR dampers possess key performance advantages, including continuously controllable force, rapid response, and low power consumption, which can be readily optimized in design procedures [2-3]. MR dampers can be analyzed using available models and are amenable to innovative design concepts [4-5].

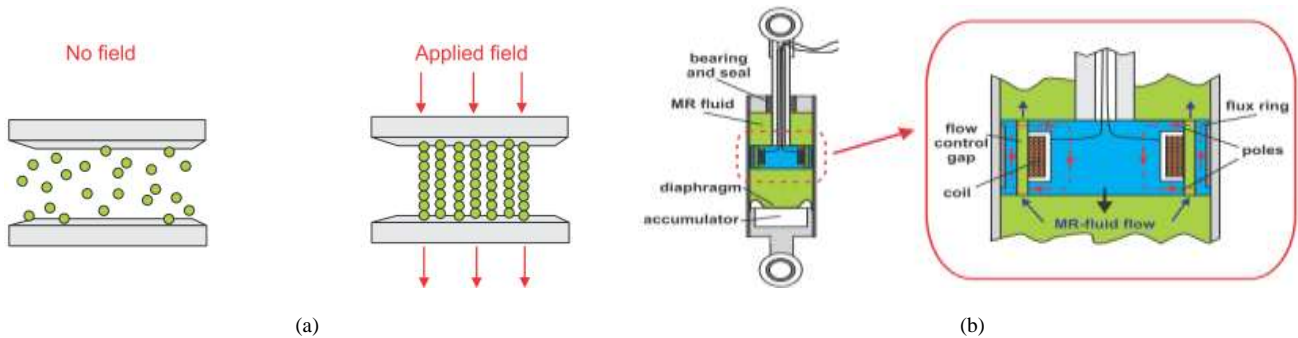


Figure 1: (a) Chain-like structure formation in MR fluids [6-7], (b) Typical MR damper [6-7]

To practically construct vibration and shock mitigation systems using MR dampers, either a power supply or a current amplifier is required to activate the electromagnetic coils in the MR dampers to supply magnetic field to the MR fluid.

There are some key parameters which influence the MR damper’s performance. However, there is no simple and effective parametric model with high accuracy for MR damper, though various kinds of models have been proposed and validated [8]. In recent years, researchers are studying ways to improve the performance of MR fluid damper. They are trying to fit the existing well-established models into their investigations with new ideas to achieve better performance. The main problem is the complexity encountered in implementing these models due to the large number of parameters that need to be estimated. The most popular ideas are based on the parametric solutions. These ideas, however, cannot describe the real phenomena of nonlinear system.

In this study a simple technique based on the experimental result is suggested to control the vibration with MR damper. As this study is about experimental identifications, it solves the issue of evaluating too many parameters. This paper has tried to establish the characteristic analysis of MR damper with impedance (Force/Velocity) analysis. The main contribution is to achieve a new performance based model of magnetorheological damper. The experimental investigation and the performance analysis are the key for this methodology. This performance test is based on the dynamic force-displacement-current (surface) analysis.

2. Experimental Setup to establish behavior of MR damper

The experimental setup comprises the MR damper, the wonder box, and a Universal Testing Machine (UTM) which are shown in Figure 2. The UTM was used to test the MR damper. The UTM has an upper and a lower head with grippers that can grasp the damper at the appropriate locations. The upper head is operated by a hydraulic actuator which can take a computer-generated prescribed displacement signal and is the moveable end. The lower head incorporates a load cell, allowing the operator to measure the force applied to the MR damper. The displacement of the damper is measured by a Linear variable differential transformer (LVDT) sensor which is integrated with the test machine. During the experiment, a variable voltage was supplied to provide the current excitation to the damper-coil. The required signal was generated using TRAPEZIUMX software running on a computer and the data was obtained from the software.

The MR damper used in this experiment is a product of LORD Corporation, USA. The model number of the particular MR damper is RD-8041-1. The stroke length of the damper is 74 mm and the maximum tensile strength is 8896 N. MR fluid device controller kit, namely Wonder Box Device Controller (Model: LORD RD-3002-03 Wonder Box Device Controller), is a companion product of the MR fluid devices. It provides closed loop current control to compensate for

changing electrical loads up to the limit of the power supply (courtesy: LORD user instructions for LORD RD-3002-03 Wonder Box Device Controller). The maximum output current of this controller kit is 2 Ampere.

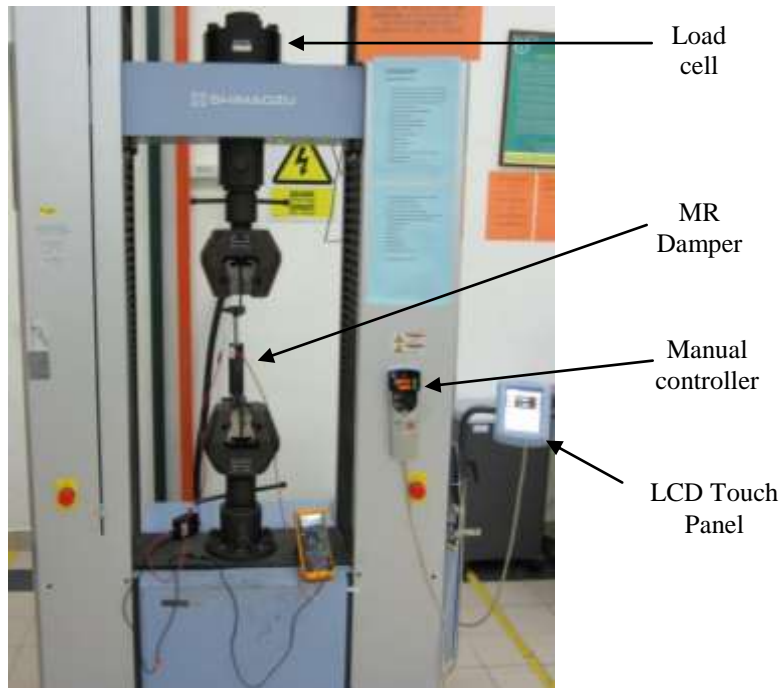


Figure 2. Experimental set-up

During the experiment, the force and the displacement were analyzed for different velocities and different starting positions of the stroke. The stroke length for the damper is 74 mm. Three equal length positions were taken to test the performance with respect to stroke positions. These positions are shown in Figure 3. The damper was tested for different input currents and different stroke positions.

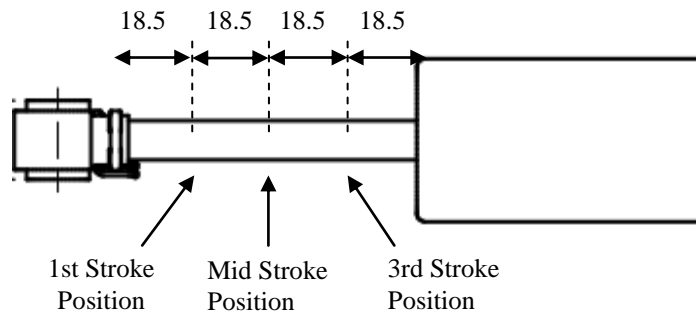


Figure 3. Different stroke length positions

3. Results and discussions

The UTM machine is used in two modes: upward motion and downward motion of piston.

3.1. Force Response

The force increases proportionately with increasing field strength. The force is the smallest when the damper is in off-state (without field). It is because the particles are aligned linearly in a rigid chain structure for higher current, due to the corresponding increase in magnetic field. Thus, the mechanical energy needed to yield these chain-like structures increases

as the applied field increases. Velocity of the piston has a proportionate effect on the force. Lower piston velocity gives lower force amplitude, while higher velocity gives higher force as shown in Figure 4.

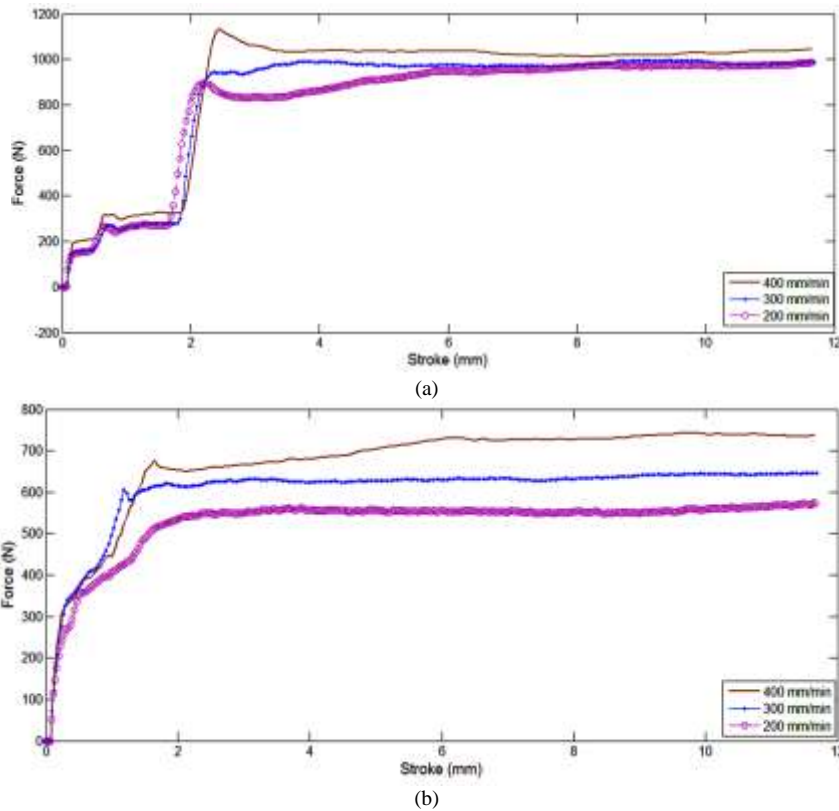


Figure 4. Comparison of forces for different piston velocities: (a) downward motion mode at 0.60 Ampere (b) upward motion mode at 0.60 Ampere

Figure 5 shows the force versus displacement (stroke length) for varying currents in the cycle mode of 12 mm cycle stroke, when the piston velocity is 500 mm/min. These plots show the hysteresis characteristics.

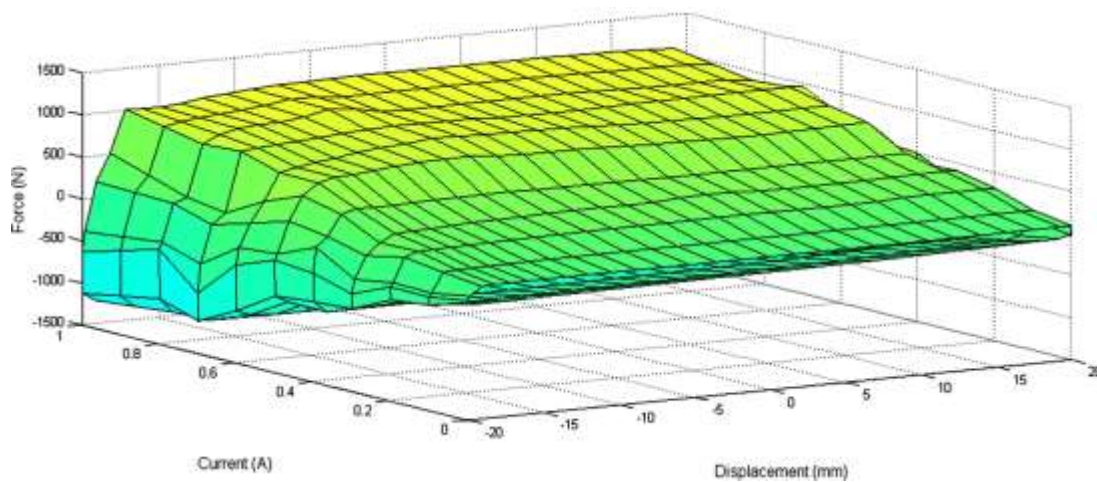


Figure 5. Three dimensional plot of current, displacement (or stroke length) and force at different stroke positions in cycle mode for 12mm stroke length at Mid stroke position

In the cycle mode, the energy dissipation is calculated from the area of the hysteresis plot for individual current. Figure 6 shows the energy dissipation. The slope of this plot changes twice; slow rise in the beginning turns into much faster rise in the middle, and it returns to slow rise at the end. It is also observed that the energy dissipation is higher at the third stroke.

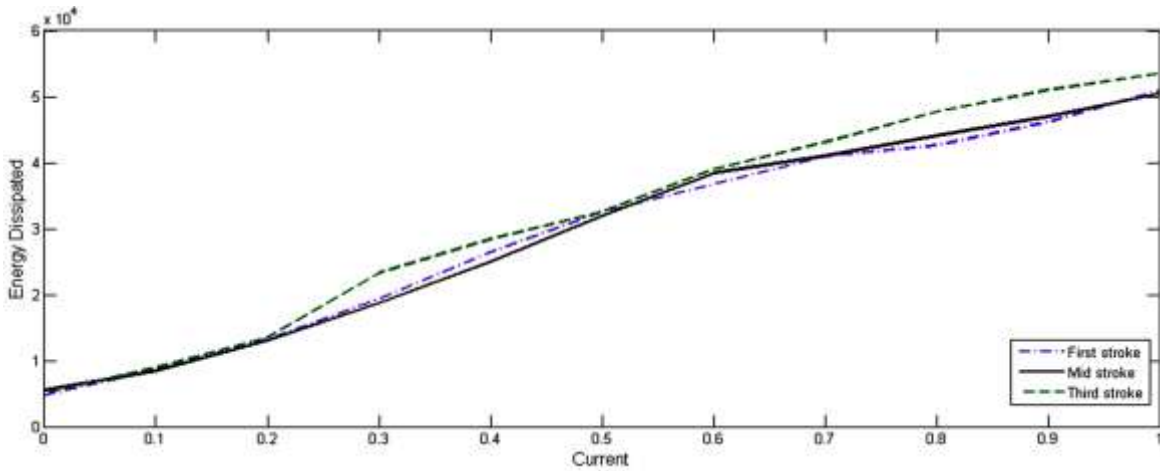


Figure 6. Energy dissipated in the hysteresis for different current

3.2. Performance of Fuzzy-PID controller

The designed controller needs to supply voltage to the Wonder Box, which will supply a corresponding current to the MR damper, in order to absorb the force for a given stroke length of the piston. In this study, as a semi-active suspension system, a quarter car suspension model is analyzed. This model can be represented by a multiple degrees of freedom vibration system, as shown in Figure 7.

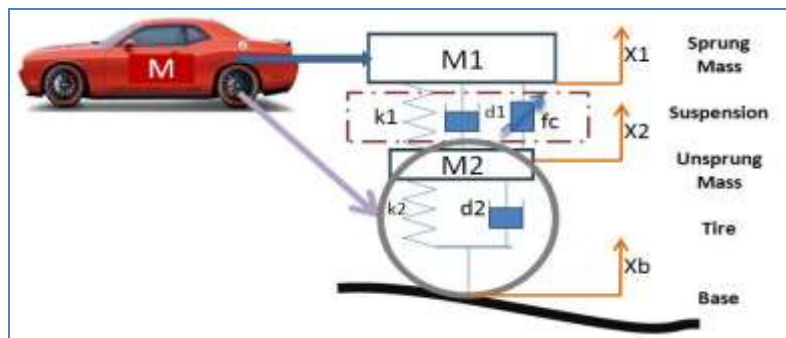


Figure 7. Quarter-Car model [9].

By applying Newton’s second law, the equations of motion of m_1 and m_2 are:

$$m_1 \ddot{x}_1 + k_1(x_1 - x_2) + d_1(\dot{x}_1 - \dot{x}_2) + f_c = 0 \tag{1}$$

$$m_2 \ddot{x}_2 - d_1(\dot{x}_1 - \dot{x}_2) - d_2(\dot{x}_b - \dot{x}_2) - k_1(x_1 - x_2) - k_2(x_b - x_2) - f_c = 0 \tag{2}$$

Based on the equations of motion in Equation (1) and Equation (2), a MATLAB/Simulink model of the quarter car suspension system is given in Figure 8, where $m_1=325$ Kg, $m_2=55$ Kg, $d_1=2000$ Nsm-1, $d_2=250$ Nsm-1, $k_1=42000$ Nm-1 and $k_2=180000$ Nm-1. Lookup tables based on experimental result are used to design controller. The first lookup table has force and current as input and output respectively, for any given stroke position. The output of the first Lookup Table goes to the input of second Lookup Table. The second Lookup Table has two inputs and one output. Displacement (stated as constant in Figure 8 (b)) is the other input of this Lookup Table. This second Lookup Table carries data of force for different current and for different stroke length.

A PID controller is developed and compared with uncontrolled output. A PID controller tuned for a single frequency does not work for other excitation frequencies (tested using chirp signal). To overcome this limitation, an adaptive PID

controller is developed using Fuzzy logic. Fuzzy rules are developed to continuously change PID gains according to input excitation frequencies. Figure 9 shows PID, fuzzy-PID controller outputs and uncontrolled output.

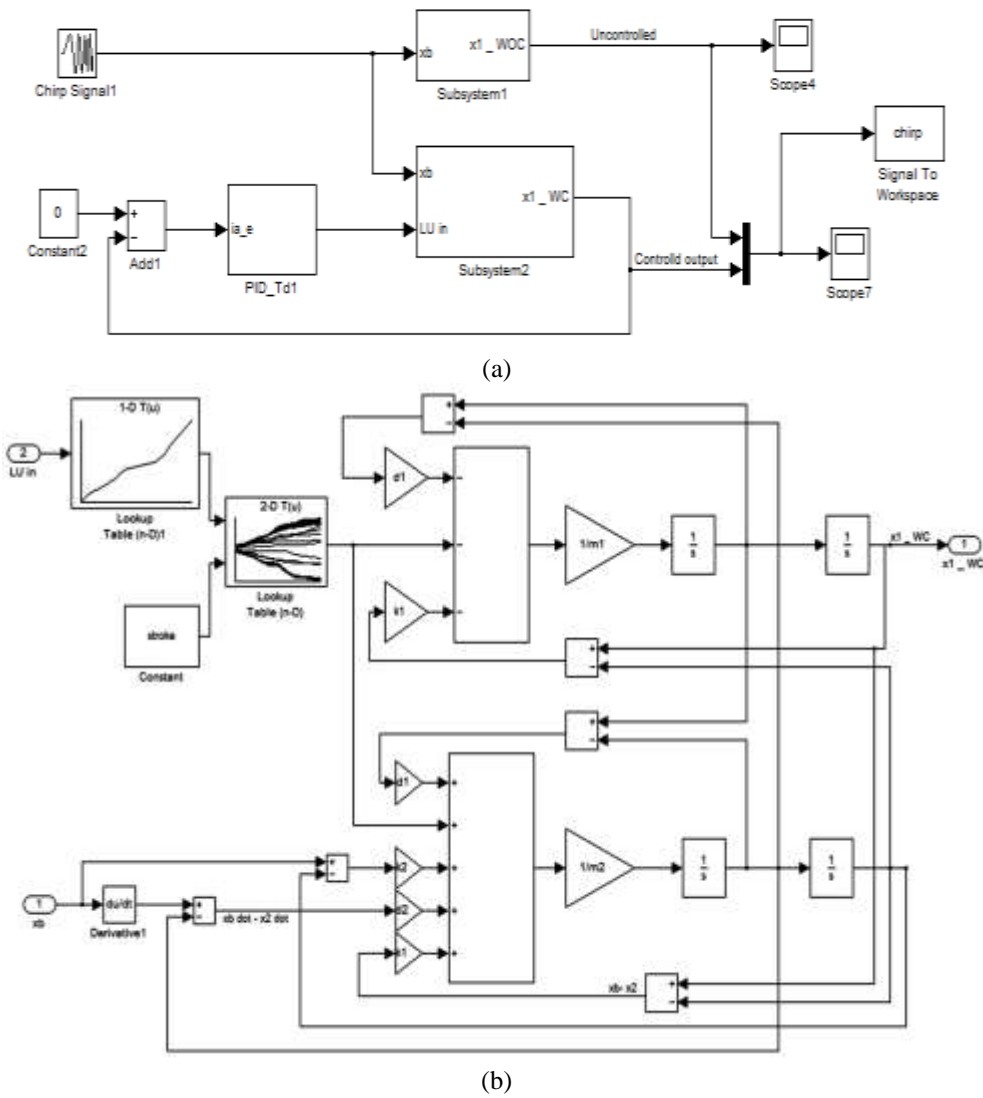


Figure 8. Simulink model. (a). Total system model (b). System with controller.

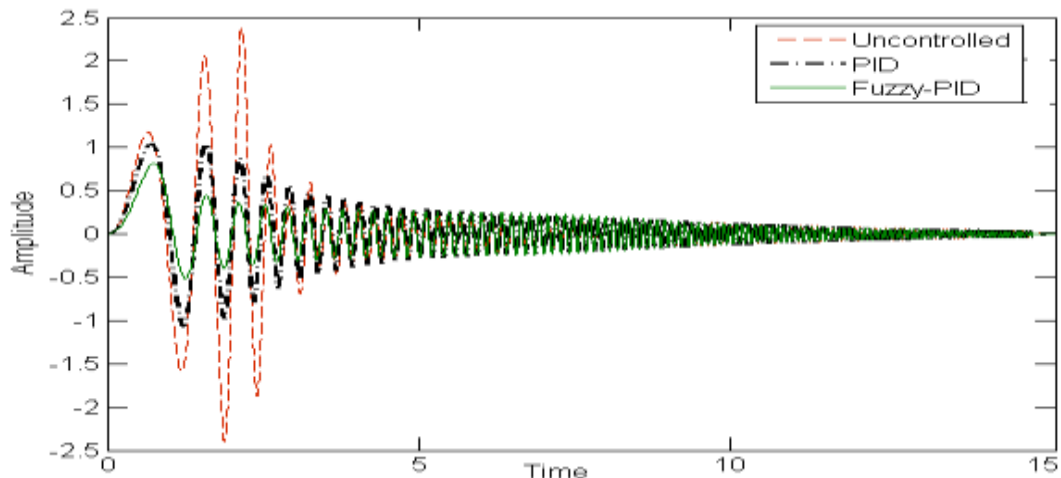


Figure 9. Controller response for Chirp signal

The fuzzy-PID provides better result than PID controller only. Fuzzy-PID controller reduces the peak by 68% from that of PID controller in low frequency and it reduced the peak by 83 % from that of uncontrolled one. Fuzzy-PID gives better reduction in high frequency as well, which shows that fuzzy-PID controller works better in broadband range of frequency. Figure 10 provides the results of peak reduction by these two controllers.

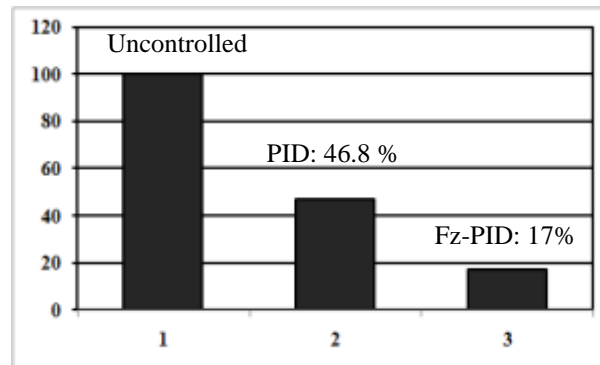


Figure 10. Peak reduction: PID controller vs Fuzzy-PID controller

4. Conclusion

The technique of developing the controller, using the experimental results, has been shown in this paper, with the details of the block diagram of the control algorithm. At first, a simple PID controller has been tuned and simulated. Its controlled output is compared with the uncontrolled output. PID controller works very well for step and impulse road disturbance inputs. However, it cannot adapt when the condition changes. When the disturbance has varying conditions, PID controllers fail to provide good controlled output, due to its fixed PID gains. It needs to be re-tuned when the surrounding condition varies. In this case, it is observed that PID controller does not give satisfactory results, with the same gain used for the step and impulse input. To achieve adaptive tuning of the gain parameters, a fuzzy based PID controller is used, which shows improved results. Fuzzy-PID controller can adapt to the situation based on the fuzzy rules, and gives the controller a new set of gain parameters. This, of course, works better than the PID controller having fixed gain parameters. It is understood that, the experimented results of force response over varying current and displacement, can be used as MR damper model, which reduces the problem of evaluating too many constant parameters. Using this model, a controller can control the MR damper's variable damping output to control the vibration of a car, which provides the passengers better comfort.

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