International Journal of Mechanical Engineering and Robotics Research

ISSN 2278 – 0149 www.ijmerr.com
Vol. 2, No. 1, January 2013

Research Paper

MODELING, TESTING AND EVALUATION OF MAGNETO-RHEOLOGICAL SHOCK ABSORBER

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This paper presents design, testing and evaluation of a Magneto-Rheological (MR) shock absorber. The devices, which fill the gap between purely passive and fully active control systems, offer a reliability of passive systems and yet it maintains the versatility and adaptability of the fully active devices. These devices are popularly known as semi-active control systems. In these devices, after application of a the magnetic field, the fluid changes from liquid to semi-solid state in milliseconds thus results in an infinitely variable, controllable shock absorber capable of generating large variable damping forces. The advantage of the MR shock absorbers over the conventional fluid shock absorbers are many, e.g., simple in construction, needs little power, quick in response, few moving parts, etc. In this paper the basic theory behind the MR Shock absorbers, its testing and evaluation is carried out and its use in vibration control is also studied. For this purpose, a MR shock absorber from Lord Corp. USA is bought and tested in the Laboratory using an electrodynamics vibration shaker and associated data acquisition system. Its performance is then studied in the form of damping force, displacement with respect to time.

Keywords: Magneto-rheological fluid, MR shock absorber, Semi-active control system

INTRODUCTION

Some smart materials have the ability to change from a liquid to a solid almost instantly when placed near a magnet. These materials have multiple properties, e.g., electrical, magnetic, mechanical and thermal and can transform energy that can be altered using some external fields. The Magneto-Rheological (MR) fluids are field responsive rheology where their fluid and other properties are controlled by varying the external magnetic field. The discovery of MR fluids is credited to Rabinow (1948) at the US National Bureau of Standard. Typical MR fluids are the suspensions of micron sized, magnetizable particles (mainly iron) suspended in an appropriate carrier liquid such as mineral oil, synthetic oil, water or ethylene glycol, etc. The carrier fluid serves as a dispersed medium and ensures the homogeneity of particles in

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the fluid. A variety of additives (stabilizers and surfactants) are also used in the MR fluid to prevent gravitational settling and to promote a stable particles suspension, to enhance lubricity and to change initial viscosity of the MR fluids (Ashwani and Mangal, 2012). The stabilizers serve the purpose to keep the particles suspended in the fluid, whilst the surfactants are absorbed on the surface of the magnetic particles in order to enhance the polarization induced in the suspended particles on application of the magnetic field. The size of the magnetizable particle ranges from 3 to 5 microns in the MR fluid used for the shock absorber application.

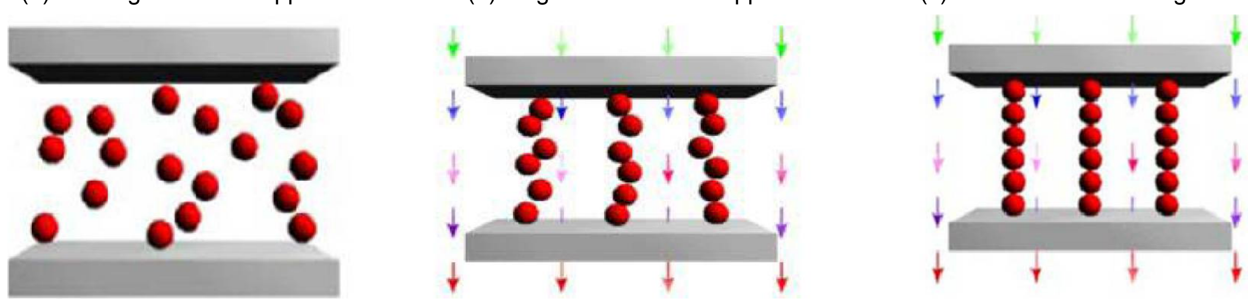
When no magnetic field applied on the MR fluid, the ferrous particles are randomly dispersed in the medium (Figure 1a). In the presence of a magnetic field, the particles start to move in order to align themselves along the lines of magnetic flux, Figure1b. Figure 1c shows the formation of chains of ferrous particles, creating more viscosity and increased shear strength (Ashwani and Mangal, 2010a). As this change occurs almost instantly, the MR fluids are attractive solutions for real-time control applications, e.g., shock absorber, brakes, clutches, engine mounts, valves, etc. The changes of liquid-solid-liquid state or the consistency or yield strength of the MR fluid can be controlled precisely and proportionally by altering the strength of the applied magnetic field.

In the absence of an applied magnetic field, the MR fluids are reasonably well approximated as Newtonian liquids. For most engineering applications, a simple Bingham plastic model is effective to describe its essential, field-dependent fluid characteristics. A Bingham plastic is a non-Newtonian fluid whose yield strength must be exceeded before the flow begins (Siginer, 1999). Typical MR fluid can achieve yield strengths up to 50-100 kPa at magnetic field strength of about 150-250 kA/m. These MR fluids are stable in temperature ranges from—50 °C to 150 °C.

In this paper, the basic theory behind the MR Shock absorbers, its testing and evaluation is carried out and its use in vibration control is also studied. For this purpose, a MR shock absorber from Lord Corp. USA is bought and tested in the Laboratory using an electrodynamics vibration shaker and associated data acquisition system. Its performance is then studied in the form of damping force, displacement with respect to time.

Figure 1: Illustration of Activation of the MR Fluid

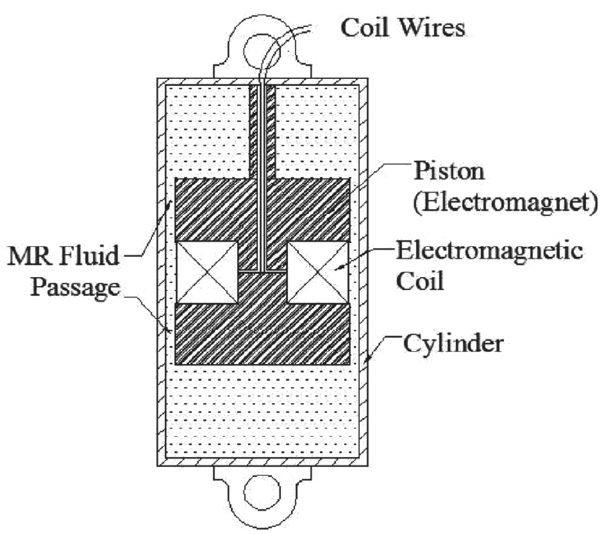
(a) No Magnetic Field Applied (b) Magnetic Filed Just Applied (c) Ferrous Particles Aligned



MAGNETO-RHEOLOGICAL (MR) SHOCK ABSORBER

A typical MR shock absorber consists of a cylinder, piston, excitation coil and the MR fluid which is enveloped in the cylinder. As this device act on the magnetic effect, the MR shock absorber should have optimized design to fulfill its magnetic performance and eventually to achieve a satisfactory semi-active control device. The MR shock absorber offers a highly reliable operation and can be viewed as fail-safe device if the control hardware malfunctions and the absorber turns to passive one. Figure 2 shows schematic example of the MR fluid shock absorber. The operation of the MR device is fundamentally different from that of MR brakes and clutches (Ashwani and Mangal, 2010b).

Figure 2: Schematic Representation of MR Shock Absorber



In the absence of an applied field, MR fluids are reasonably well approximated as Newtonian liquids, i.e., off-state. In the on state (when activated under magnetic field) the fluid behaves as a Bingham plastic with variable yield strength. All though the fluid does have the departures from this model even then this gives a good reference for the behavior of the fluid (Carlson and Jolly, 2000; Poyner, 2001; Seval, 2002; and Ashwani and Mangal, 2010b). The shear stress associated with the flow of the MR fluid can thus be predicted by the Bingham equations (Spencer *et al.*, 1996). In this model, the total fluid shear stress is given by:

$$\tau = \tau_y(H) + \eta\dot{\gamma} \qquad |\tau| > \tau_y \qquad ...(1)$$

where τ_y is the yield stress [Pa], H is the Magnetic field strength [A/m], η is the Plastic viscosity [Pa.s] and $\dot{\gamma}$ is the Shear strain rate [s^{-1}].

For the total fluid shear stresses below τ_y, the MR Fluid behaves as visco-elastic material and is given as:

$$\tau = G\gamma \qquad |\tau| < \tau_y \qquad ...(2)$$

where the G is the complex material modulus and γ is the fluid shear strain.

The MR shock absorbers generally use the flow mode of the fluid. In the flow mode, the pressure has two components of the pressure drop, i.e., pressure loss due to the viscous drag and pressure loss due to the field dependent yield stress (Carlson and Jolly, 2000). The total pressure drop (ΔP) in the fluid can be given as (Spencer *et al.*, 1996).

$$\Delta P = \Delta P_n + \Delta P_\tau = \frac{12\eta QL}{g^3 w} + \frac{c\tau_y L}{g} \qquad ...(3)$$

where ΔP_η is the viscous pressure loss, ΔP_τ is the field dependent pressure loss, η is the fluid viscosity, Q is the flow rate, L is the pole length, w is the pole width, g is the fluid gap, τ_y is the field dependent yield stress and c is constant. Its value varies between 2 to 3 which depends upon the ratio of field dependent pressure loss to viscous pressure loss.

The volume of fluid exposed to the magnetic field controls the desired MR effect (Spencer *et al.*, 1996; and Vibration and Seat Design, 2001) and is known as active fluid volume. The above equations can be manipulated to determine the active fluid volume (V) and is given as:

$$V = k\left(\frac{\eta}{\tau_y^2}\right)\lambda W_m \qquad ...(4)$$

where k is a constant and λ is the desired control ratio required to achieve a specified mechanical power, W_m (Spencer *et al.*, 1996). These variables can be defined as:

$$k = \left(\frac{12}{c^2}\right);\ \lambda = \left(\frac{\Delta P_\tau}{\Delta P_\eta}\right);\ W_m = Q\Delta P_\tau \qquad ...(5)$$

The Equation (4) can be further solved to provide constraints and aspect ratios for an efficient use of the MR fluid (Spencer *et al.*, 1996) as:

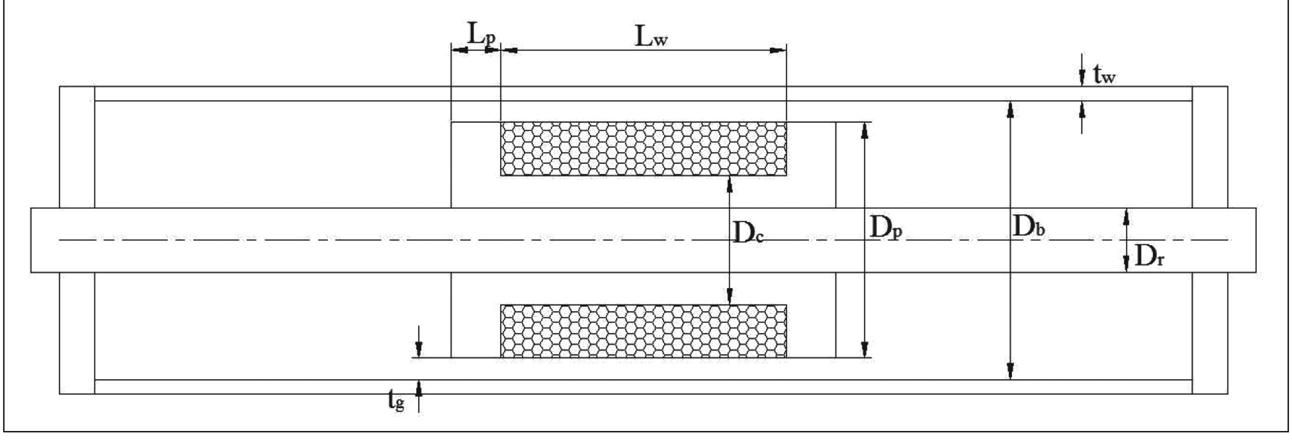
$$wg^2 = \frac{12}{c}\left(\frac{\eta}{\tau_y}\right)\lambda Q \qquad ...(6)$$

DESIGN CONSIDERATION OF MR SHOCK ABSORBER

A magnetic field in the flow path of the MR fluid needs to be generated to use the MR shock absorber effectively. The magnetic field is applied by using a copper coil which is wounded around the piston body as shown in Figure 3 with hatching. The leads of the coil are taken out through the piston rod to give the variable current to the coil in order to generate the variable magnetic field which in turn produces variable damping effect. The construction of the shock absorber is simple and easy to manufacture.

Figure 3 shows the conceptual design of the MR shock absorber. The Spool of magnet wire generates magnetic flux within the low carbon steel piston. The flux in the magnetic circuit flows axially through the piston core of diameter D_c, through the piston poles of length L_p and through a MR fluid gap of thickness t_g and axially through the cylinder wall of thickness t_w. The MR shock absorber design involves different dimensions of the piston, cylinder assembly, i.e., diameter of the cylinder bore (D_b), the diameter of the piston rod (D_r), the thickness of the cylinder wall (t_w), the diameter

Figure 3: MR Shock Absorber



of the piston core (D_c), the outside piston diameter (D_p), the pole length (L_p) and the thickness of the MR fluid gap (t_g). The total pressure drop across the piston can be determined by substituting the appropriate symbols of the Figure 3 in the Equation (3). This is given as:

$$\Delta P = \Delta P_n + \Delta P_\tau = \frac{12\eta Q(L_w + 2L_p)}{\pi(D_p + t_g)t_g^3} + \frac{c\tau_y}{t_g} \quad ...(7)$$

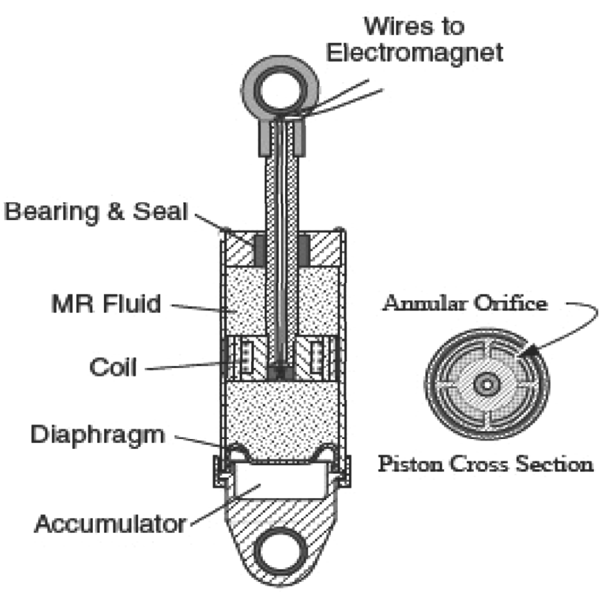
The force generated in the MR shock absorber is the pressure drop times the piston cross section area. It can be given as:

$$F = \Delta P\pi \frac{(D_p + t_g)^2 - D_r^2}{4} \quad ...(8)$$

MR SHOCK ABSORBER AND EXPERIMENTATION

A MR shock absorber as shown in Figure 4 has been bought from the LORD Corporation

Figure 4: MR Shock Absorber



Source: www.lord.com

USA, i.e., RD-8040-1 (short stroke). In this MR shock absorber, MR fluid flows from a high pressure chamber to a low pressure chamber through an orifice in the piston head (www.lord.com). The MR shock absorber is compact and is suitable for industrial suspension applications. Continuously variable damping is achieved by developing a variable magnetic field strength (www.lord.com). The technical specifications of the shock absorber (www.lord.com) is as under

1. Stroke 55 mm
2. Extended Length 208 mm
3. Body Diameter 42.1 mm
4. Shaft Diameter 10 mm
5. Tensile Strength 2000 N
6. Shock absorber Forces (Peak to Peak)
   a) 5 cm/sec @ 1 A >2447 N
   b) 20 cm/sec @ 0 A <667 N
7. Operating Temperature range 0 to 71 °C

The shock absorber is tested to check its performance. The setup of the lab as shown in Figure 5 is used for this purpose. In the setup, an Electro-Dynamic Vibration (EDV) shaker having a rated sine force of 200 kgf with a frequency range between 1 to 3500 Hz is used. The shaker has a maximum displacement (peak to peak) of 20 mm. The Shaker can be used for both horizontal and vertical vibrational analysis. The shaker is controlled by PC Based Digital Vibration Controller cum analyzer with built-in signal conditioner unit. The controller unit of the shaker is a close loop control system. The set-up has a compatible data acquisition and instrumentation system which

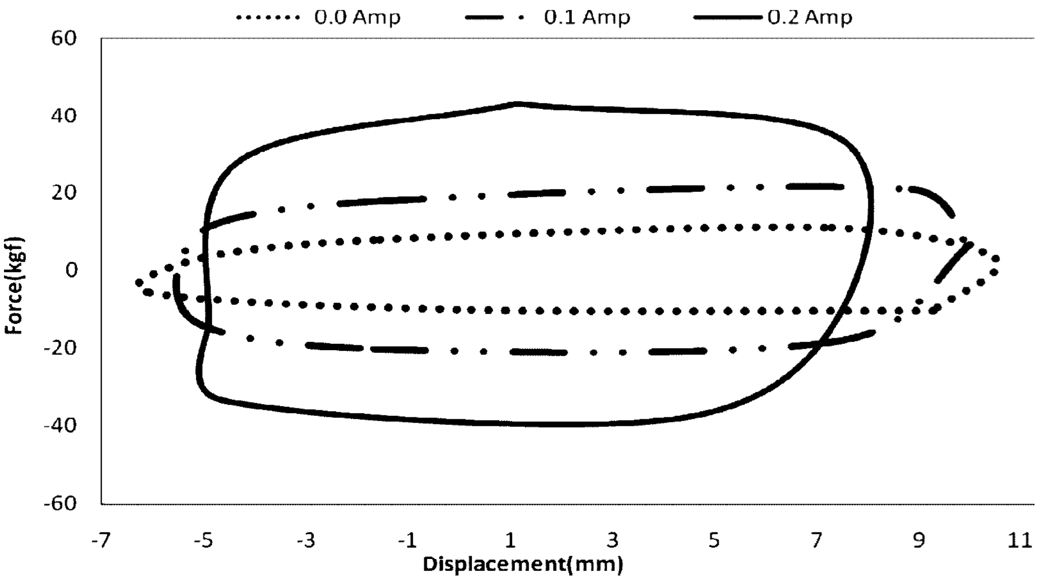
Figure 5: Experimental Setup for the Testing of MR Shock Absorber (Mounted)

gives the data in the form of force, velocity, displacement and acceleration with respect to time. On the shaker system, the MR shock absorber is mounted. The test was performed for number of cycles for different current values, i.e., 0 A, 0.1 A and 0.2 A across the coils at a frequency of 1 Hz. The force experienced by the piston rod, which was prevented from motion, was sensed by a load cell fixed at the top of the MR shock absorber while the displacement is recorded through displacement sensor.

RESULTS

Figure 6 shows the variation of damping force v/s displacement of the shock absorber for

Figure 6: Force vs. Displacement Curves with Different Levels of Current



one cycle. The Figure shows that as the value of current increases the damping force increases and the displacement of the piston is decreased. The damping force is low for zero current and it increases gradually as the current is increased. The variation of force with displacement are re-plotted in Figure 7 for each current setting, i.e., 0.0, 0.1 and 0.2 A for large number of cycles. The Figure 8 shows the variation of displacement with time for a large number of cycles for the same current settings. The Figure 9 shows the variation of damping force with respect to peak-peak displacement of the shock absorber at the above mentioned three current setting.

Figure 7: Force vs. Displacement Curves at (a) 0.0 A, (b) 0.1 A and (c) 0.2 A

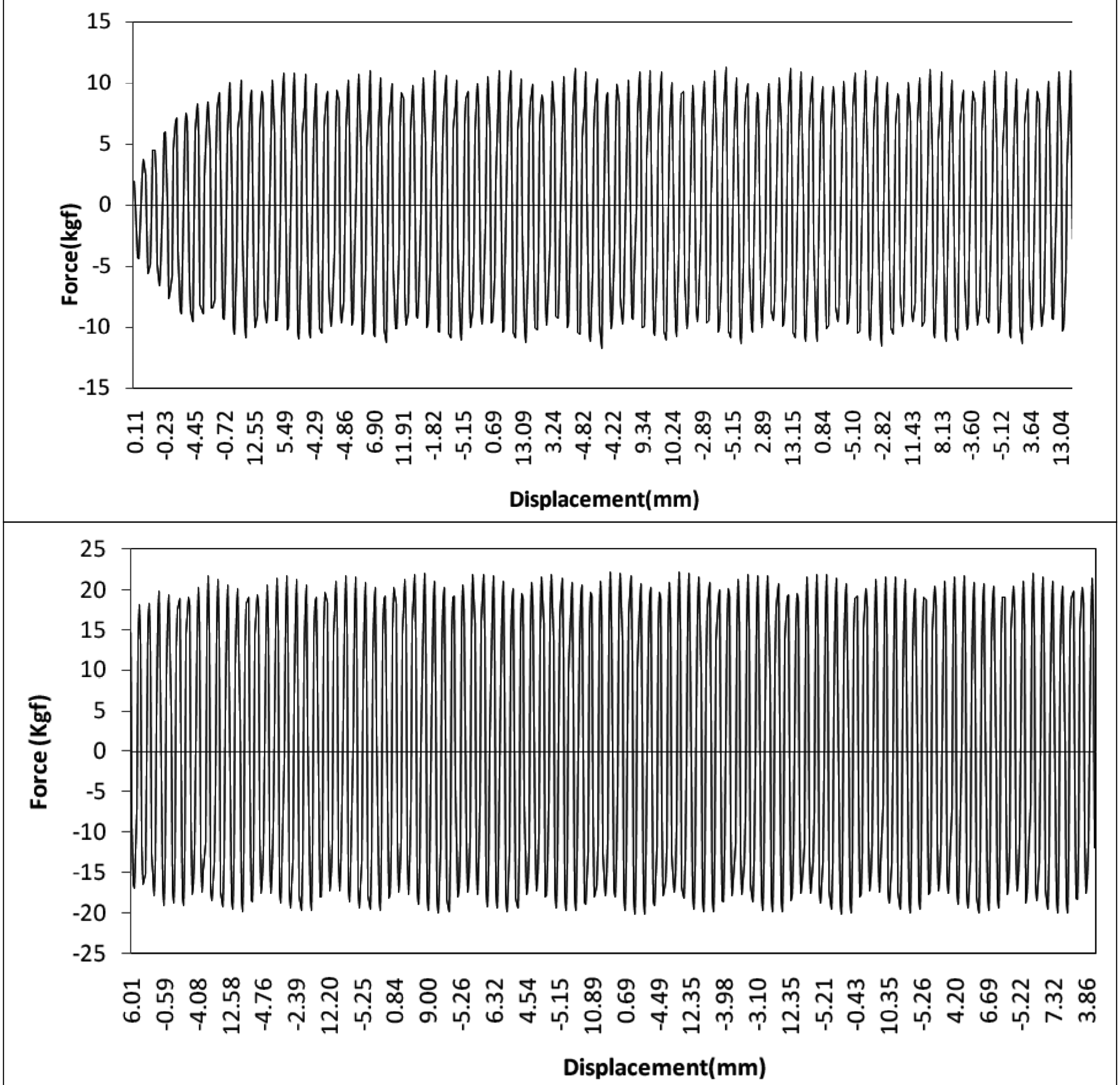


Figure 7 (Cont.)

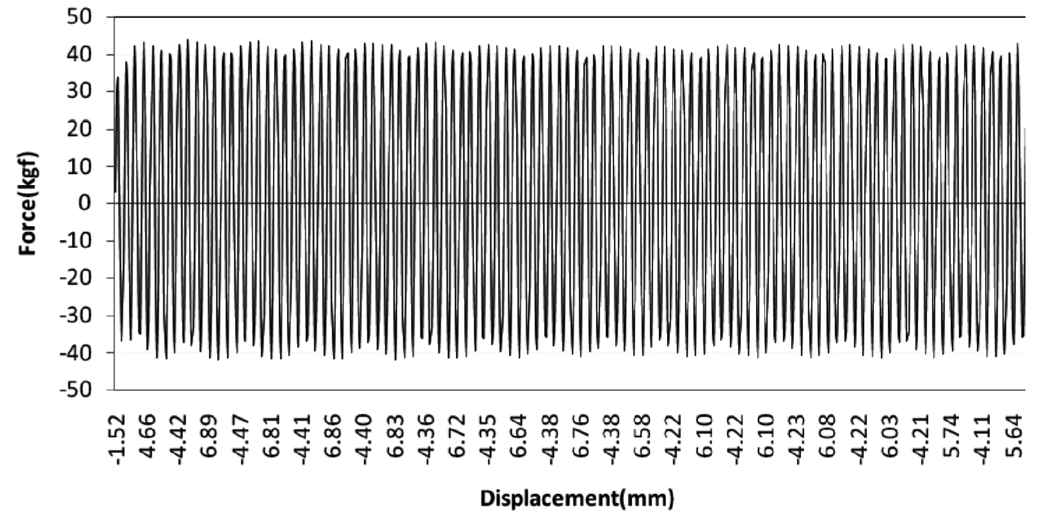
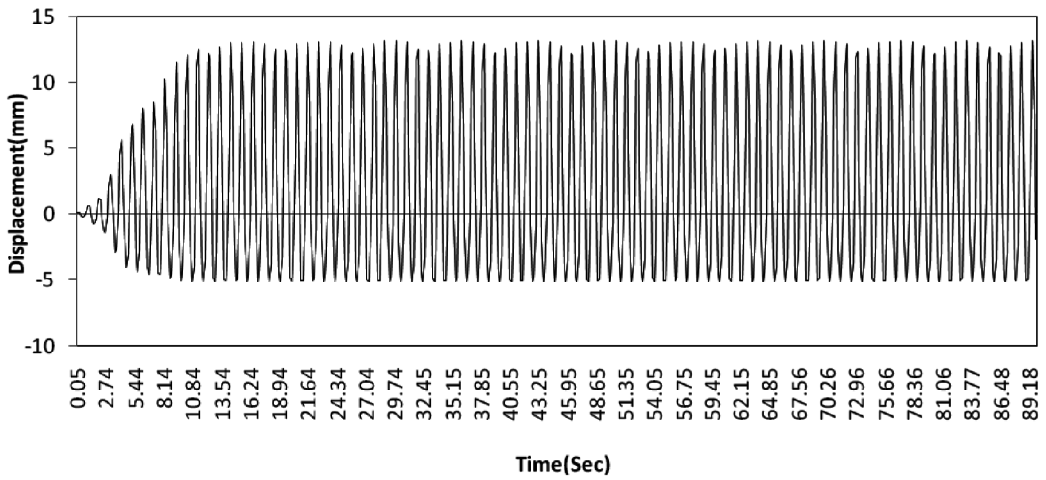


Figure 8: Displacement vs. Time Curves at (a) 0.0 A, (b) 0.1 A and (c) 0.2 A



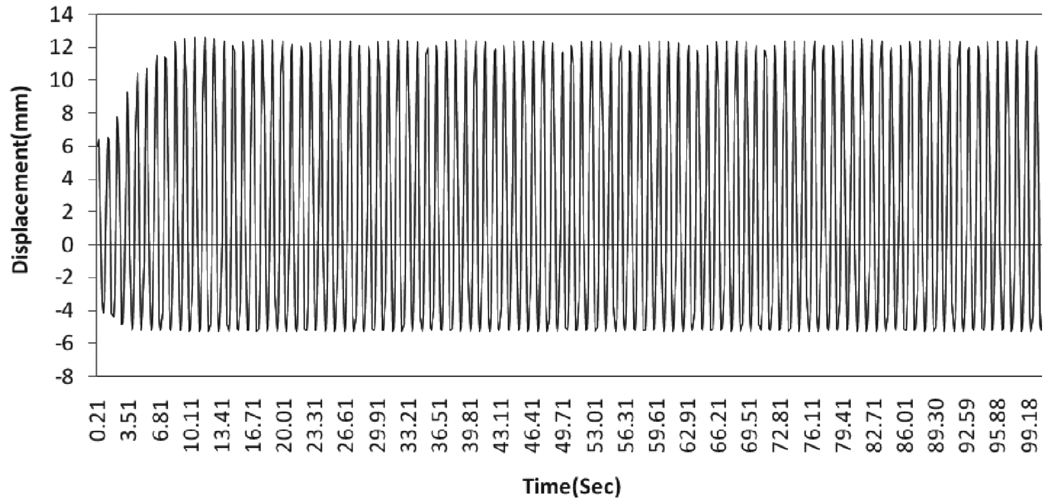


Figure 8 (Cont.)

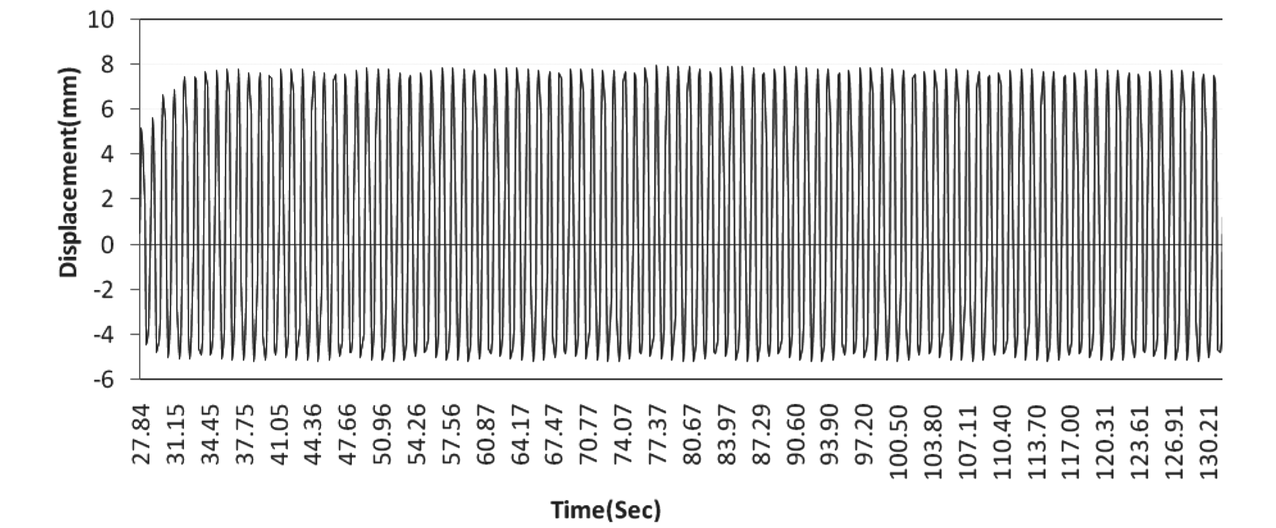
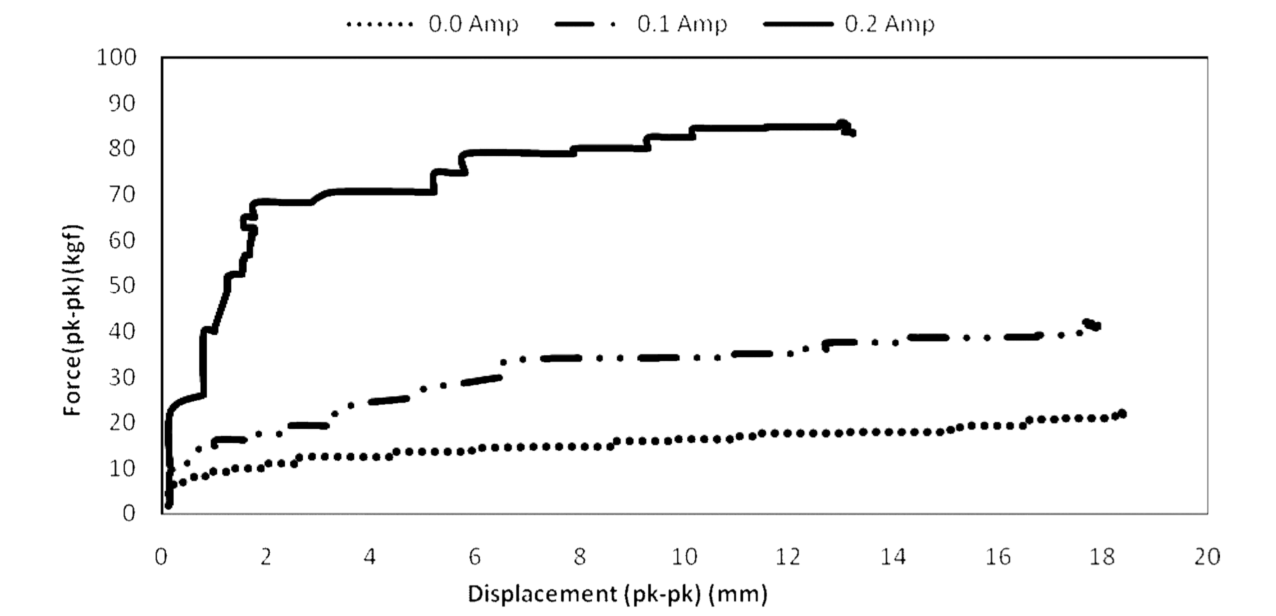


Figure 9: Damping Force vs. Peak to Peak Displacement Curves with Different Levels of Current



CONCLUSION

In this paper, the basic theory behind the MR Shock absorbers, its testing and evaluation is carried out. A MR shock absorber from Lord Corp. USA is bought and tested in the laboratory. The results presented in this paper show the good efficiency of vibration damping. The above results show that the user can have a very good control over the damping force of the MR shock absorber by varying the input current which is supplied to the magnetic coil of the absorber. This is a necessity for a semi-active vibration control system. The result further showed that the damping force is not zero at zero input current, i.e., at off-state. It is because of the fact that fluid viscosity alone is responsible for the damping effect in the off-state. It is equivalent to failure/mal-function of the electro-magnetic coil or fail-safe condition.

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