Development and Testing of MR Damper

M. M. Khade, S.P Chavan, A. P. Patil

Abstract: - Conventionally systems are designed to resist the causes of vibrations through a combination of strength, deformability, and energy absorption. Use of Magnetorheological fluids in vibration absorbing dampers is one of the methods available for smart damping now days. The essential characteristic of MR fluids is their ability to reversibly change from free-flowing, linear viscous liquids to semi-solids having controllable yield strength in milliseconds when exposed to a magnetic field. Theoretical analysis was carried out for MR Damper, resulting equations are solved in MATLAB. Analytical model and experimental analysis results are compared.

Keywords:- Magnetorheological Fluid, Damper, Single degree of freedom, Response

I. INTRODUCTION

Magnetorheological fluids (or simply "MR" fluids) belong to the class of controllable fluids. This provides simple, quiet, rapid response interfaces between electronic controls and mechanical systems. MR fluid dampers are relatively new semi-active devices that utilize MR fluids to provide controllable damping forces.

Various types of MR fluids depending upon the type of applications [1]. Reviewed five general applications wherein the technology of MR fluids was fully implemented into commercial products. As such, those applications represented more than just a laboratory demonstration of basic functionality. Each of those applications fully embodied all of the necessary developments and refinements required to make those magnetorheological fluid devices long-lived in actual service in the field, amenable to mass production techniques, and available at a cost commensurate with the value perceived by the end user. The test data from a commercialized MR damper to develop nonparametric models, which consisted of a series of numerically efficient mathematic functions [2]. This prevented the difficulties those were posed by using the existing parametric model. The selected functions were continuous and differentiable for potential model-based control algorithms. The results of the nonparametric models have shown that such different models are comparable..

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It was further shown that the nonparametric models can be numerically solved with an integration step size of the order of 10^{2} s, much faster than the parametric models of the order of 10^{5} s, which clearly shown that the proposed nonparametric models were feasible even for real time model-based control algorithms

The Bouc-Wen model for perfectly characterizing the MR damper behaviour is implemented [3]. Introduced new phenomenological model for characterization of the MR damper 8177behaviour, by adding a dashpot and a spring element to an existing Bouc-Wen model. In this control approach, a linear optimal controller was designed and combined with a force feedback loop to adjust the command voltage of the MR damper to approximate the optimal force level. The effectiveness of the MR damper using the proposed clipped-optimal control law was demonstrated through numerical examples. Compared the active system to the semi-active system employing MR dampers and concluded that the semi-active systems are more effective with clipped optimal controller than the active systems.

A new model-Hysteretic Arctangent Model Proposed – to study damper performance [11]. Plotted force-velocity and force- displacement characteristics for a damper and found that the damping force varies with the magnetic field across the MR fluid but independent of the frequency input to the damper, and the post-yield damping of the fluid increases with the applied magnetic field but decreases with the frequency input under a constant magnetic field. The 8177behaviour of MR dampers tend to be highly non-linear [6]. As found by researchers there are models which represent this non-linear 8177behaviour. But finding models for each case of MR damper is difficult due to terms occurring in the models. Bogdan proposed two linear models viz. equivalent viscous damping and complex modulus for representing the damper 8177behaviour. In both cases the MR damper was approximated by an ideal dashpot at every operating condition that allowed predicting the damping and stiffness with reasonable accuracy.

A dynamic model presented based on fluid mechanics and the Herschel-Bulkley constitutive equation to predict the behaviour of ER/MR fluid dampers [7]. The major advantage of the proposed model was its dependency on only the geometric and material properties of the device. The theoretical model presented was validated by comparing the analytical results with experimental data for a prototype MR fluid damper. A fundamental understanding of large-scale MR dampers developed for the purpose of designing and implementing these "smart" damping devices in large-scale structures for natural hazard mitigation [8].

Developed axisymmetric and parallel-plate models of MR damper using supplemental damping devices, incorporating fluid shearing thinning/thickening effects. Performed harmonic analysis of large-scale MR damper experimentally and analytically by using previously proposed models.

II. EXPERIMENTAL ANALYSIS

In this work a system considered is a two-mass vertical system with two mass. The lower mass plate is heavy mass made up of mild steel and the upper mass plate is light mass made up of an acrylic plate. The upper springs are less stiff than lower springs. Both the mass moves up and down in the guide rods, the contact between the mass and the guide rod is avoided by the linear bearings fixed in the mass. The guide rods are firmly fitted to the horizontal base plate made of mild steel. The horizontal base plate is again firmly fitted to the rigid vertical supporting beams. The vertical supporting beams are firmly bolted to foundation. The system is so designed that an electrodynamic exciter can be fitted between the vertical supporting beams.

III. DEVELOPMENT OF MR DAMPER ASSEMBLY

As discussed in material selection the cylinder and piston of damper are made of mild steel while the piston rod is made of stainless steel. The winding used is of copper material. The MR damper cylinder and the piston rod assembly parts are shown in Fig.1.

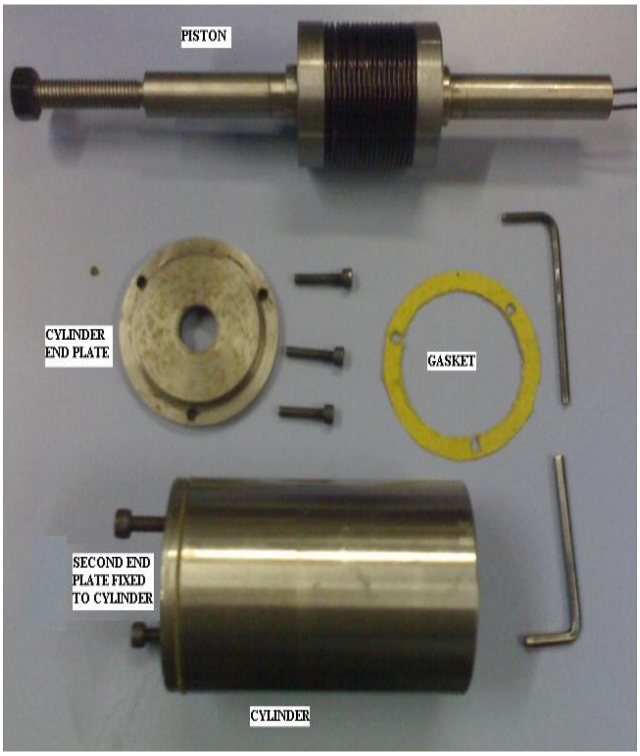

Fig 1 MR damper assembly parts

The ends of cylinder were closed by cylinder caps (end plates) on both sides. It has a hole at its center to pass the piston rod. The sealing at piston rod and cylinder cap interface is provided by O-rings. The sealing between cylinder caps and cylinder is ensured by using gaskets. The MR damper assembly parts are shown in fig 1.

IV. EXPERIMENTAL ANALYSIS

For experimental analysis two mass plates and two springs are taken named as lower and upper mass plates and springs. As the mass of damper cylinder, piston and two holders with friction plates is significantly comparable to mass of the plates, this will change the system when the damping is provided to experimental configuration. For this reason, while doing experimental analysis the two lower friction plate holders and damper cylinder are fitted to lower mass plates as shown in Fig 2

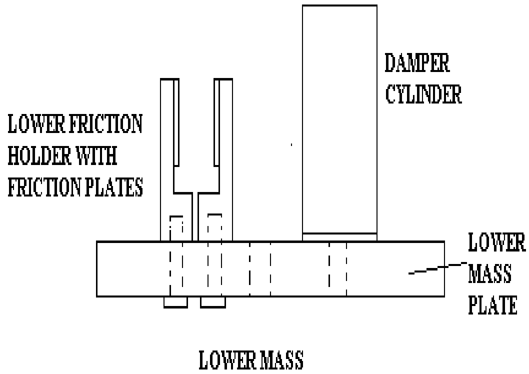

Fig 2 Lower mass assembly

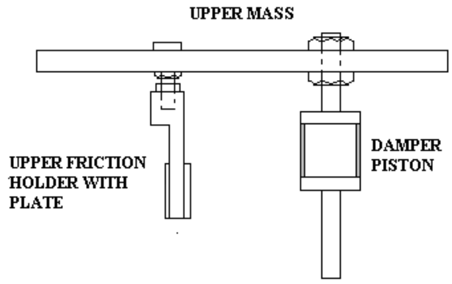

Fig 3 Upper mass assembly

Also, the piston rod and upper friction plate holder is fitted to upper plate as shown in Fig 3.

To carry out experimentation along with the mass system following apparatus is necessary

1. Electrodynamic Exciter

Gives vibration to the mass. The frequency at which it vibrates the mass can be varied from 1Hz to 10 kHz with maximum peak displacement of 12mm with peak velocity being 1m/s.

2. FFT analyzer with accelerometer.

The accelerometer is placed on the mass whose vibrations are to be measured. The signal from accelerometer is given to FFT analyzer which displays the displacement for certain frequency. The measurements are taken by adjusting the frequency of vibration and taking peak displacement at that frequency on FFT analyzer.

The experimental set up is shown in fig. 4.

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Fig.4 Experimental set up

a) Experimental Analysis of SDOF system

The SDOF system taken into account consists of lower mass and two lower springs. The analysis is carried out by changing frequency from 2Hz to 30Hz.
The current is varied as 0.1A, 0.5A and 1A with 2V voltage across the damper coil. The accelerometer is placed at center of the lower mass. The experimental results obtained are plotted in Fig 7 for 0.1A, Fig 8 for 0.5A and Fig 9 for 1A current.

b) Experimental Analysis of TDOF system

The auxiliary plate is removed and the upper mass plate is set free to vibrate on two upper springs. The upper mass plate is fixed with damper piston and upper friction plate holder while the lower mass plate is fixed with two lower friction plate holders and damper cylinder. The results thus obtained are plotted in Fig 10 for lower mass and Fig 11 for upper mass.

The damper is filled with MR fluid and the friction plates are set free. The results obtained for both masses are plotted in Fig 12 for 0.1A, Fig 13 for 0.5A, and Fig 14 for 1A current.

V. THEORETICAL ANALYSIS

a) SDOF system with system damping alone

It consists of lower mass m_1 and two lower springs with spring stiffness k_1 each. The equation of motion for the given system can be derived as given below,

$$m_1 \ddot{x}_1 + 2k_1 x_1 + c_{1S}\dot{x}_1 = F\sin(wt) \quad . \; . \; . \; . \; . \; . \; . \; . \; (1)$$

The equation (4.1) is solved in MATLAB with the parameters m_1 = 4.05 kg,

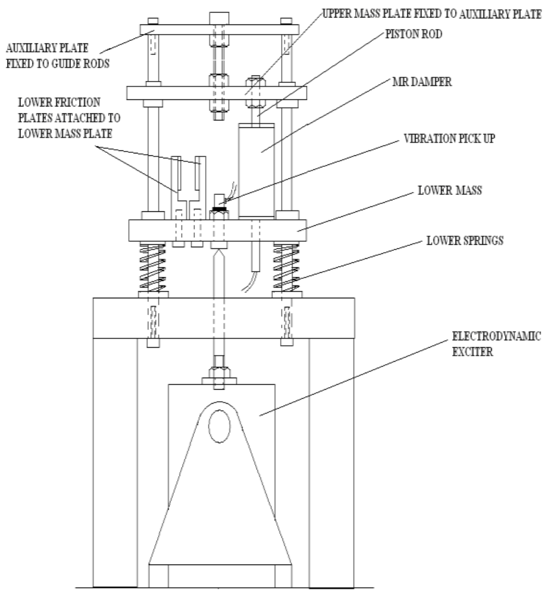

Fig 5 SDOF system with lower mass (m_1), lower springs (k_1) and MR damper.

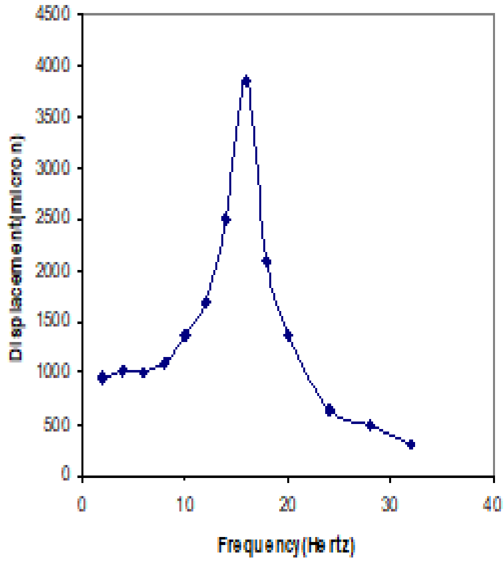

Fig 6 Experimental results plot for SDOF

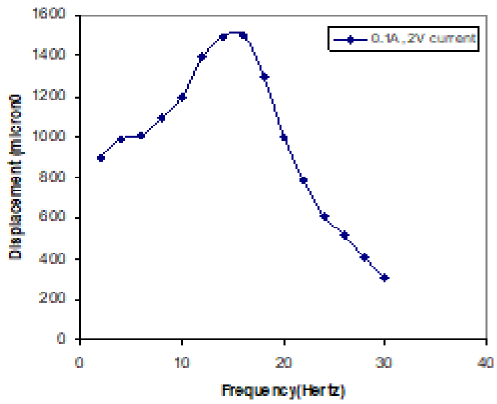

Fig 7 Experimental results plot for SDOF system without any damping system with MR damping (0.1A, 2Vcurrent)

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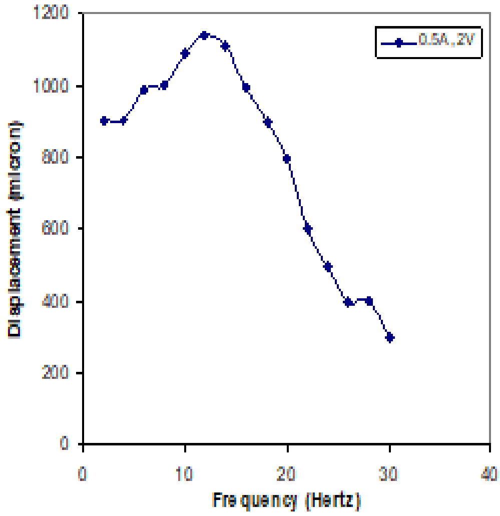

Fig 8 Experimental results plot for SDOF

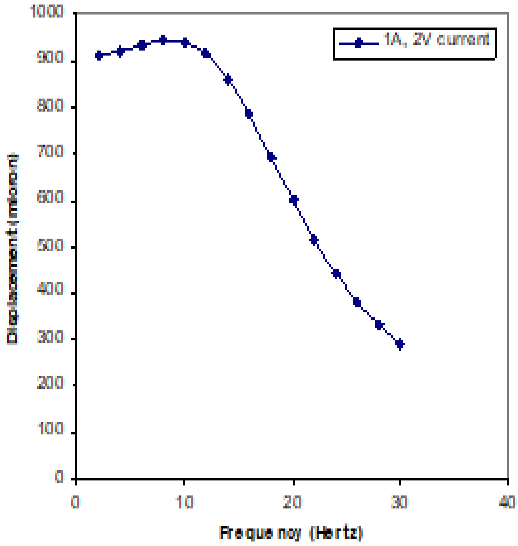

Fig 9 Experimental results plot for SDOF

system with MR damping (0.5A, 2V current)
system with MR damping (1A, 2Vcurrent)

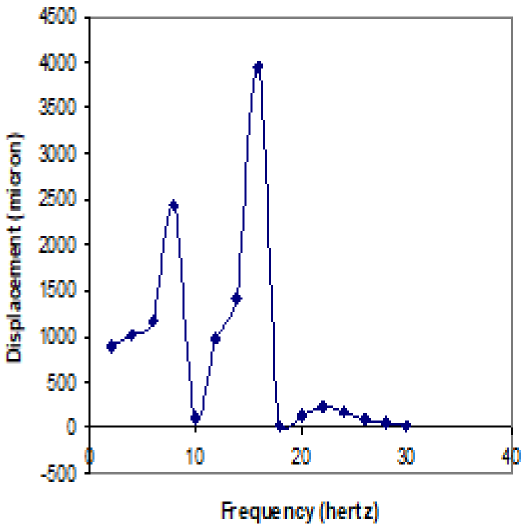

Fig 10 Experimental result plot for lower mass in TDOF system without damping

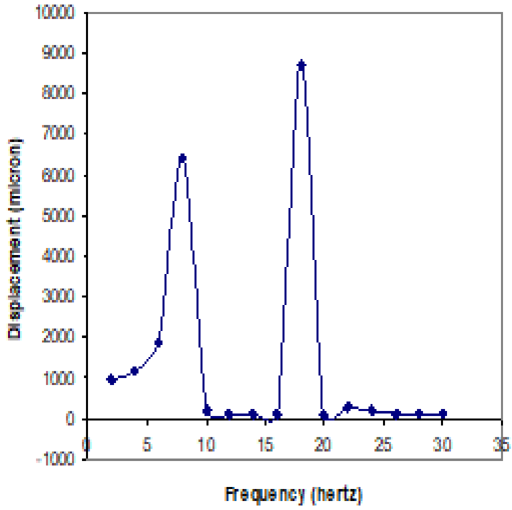

Fig 11 Experimental result plot for upper mass in TDOF system without damping

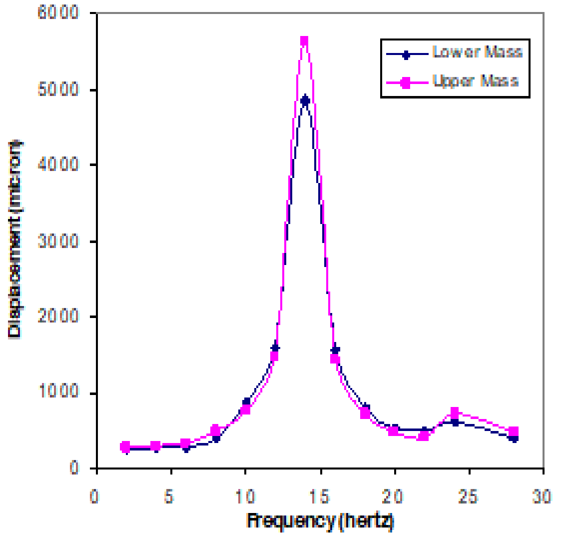

Fig 12 Experimental results plot for TDOF system with MR damping (0.1A, 2V current)

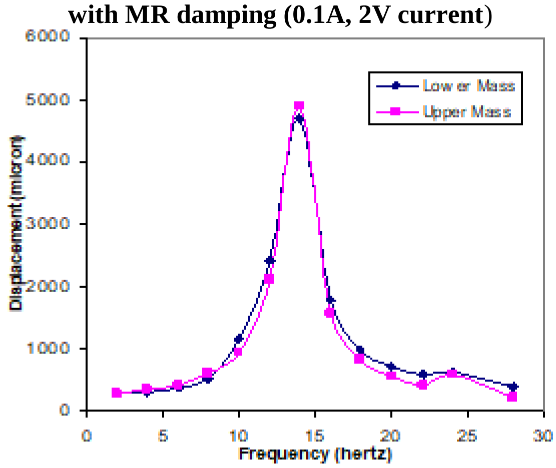

Fig 13 Experimental results plot for TDOF system with MR damping (0.5A, 2Vcurrent)

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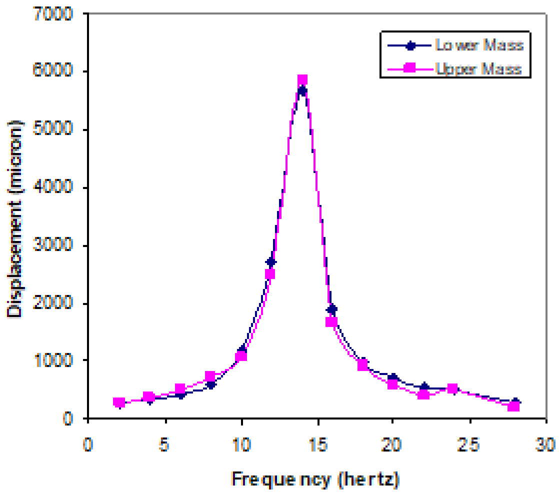

Fig 14 Experimental results plot for TDOF system with MR damping (1A, 2Vcurrent)

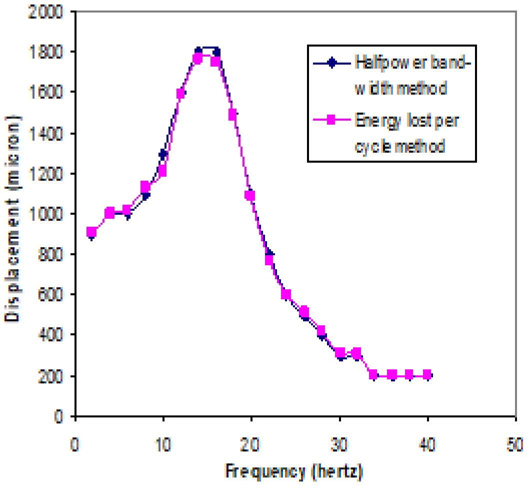

Fig. 15 Frequency Response Curve

k_1 = 19870 N/m, F = 200 N, c_{1S} = 21.56 N-s/m (obtained from experimental frequency response curve).The results obtained are for x_1 are plotted for frequencies 0 to 40 hertz and plotted as shown in Fig 16.

The frequency response curves obtained with equivalent damping obtained by method of half power band-width method and energy lost per cycle method are plotted in figure shown below

The graph shows that the response obtained by both methods is same. Hence for further analysis equivalent damping obtained by method of half power band-width method is used

b) SDOF system with system and MR damping.

It consists of lower mass m_1 and two lower springs with spring stiffness k_1 each and the MR damper. The equation of motion for the given system can be derived as given below,

$$m_1 \ddot{x}_1 + 2k_1 x_1 + c_{eq} \dot{x}_1 = F \sin(wt) \quad \ldots\ldots (2)$$

Where,

m_1 = 4.05 kg, k_1 = 19870 N/m, F = 200 N, frequency from 2 Hz to 40 HZ.

c_{eq} for this system from experimental response curve,

for 2V, 0.1A current c_{eq} = 246.75,

for 2V, 0.5A current c_{eq} = 405.95,

for 2V, 1A current c_{eq} = 493.51.

The equation (2) is solved in MATLAB and the response of the system for current levels 0.1A, 0.5A, and 1A are plotted in Fig 17, Fig 18, and Fig 19 respectively.

C) Theoretical Analysis of Two Degree of Freedom System

C.1 TDOF system with no damping

It consists of lower mass m_1, upper mass m_2 two lower springs with spring stiffness k_1 each and two upper springs with stiffness k_2.

Assuming motion of upper mass more than lower mass ($x_2 \succ x_1$) the equations of motion for two mass can be derived as given below,

For lower mass (m_1)

$$m_1 \ddot{x}_1 + 2k_1 x_1 - 2k_2 (x_2 - x_1) = F \sin(wt) \quad \ldots (3)$$

For upper mass (m_2)

$$m_2 \ddot{x}_2 + 2k_2 (x_2 - x_1) = 0 \quad \ldots (4)$$

The equations (3) and (4) are solved in MATLAB with following parameters,

m_1 = 4.05 kg, m_2 = 1.91 kg, k_1 = 19870 N/m, k_2 = 3670 N/m.

The frequency response curves for frequencies from 2Hz to 40Hz are plotted for lower and upper mass as shown in Fig 20 and Fig 21 respectively.

C.2 TDOF system with MR damper.

It consists of lower mass m_1, upper mass m_2 two lower springs with spring stiffness k_1 each, two upper springs with stiffness k_2 each and a MR damper.

The equations of motion for two mass can be derived as given below,

For lower mass (m_1),

$$m_1 \ddot{x}_1 + 2k_1 x_1 - 2k_2 (x_2 - x_1) - F_D = F \sin(wt)$$

With equivalent damping

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$$m_1 \ddot{x}_1 + 2k_1 x_1 - 2k_2\left(x_2 - x_1\right) - c_{eq}\left(\dot{x}_2 - \dot{x}_1\right) = F\sin(wt)$$

. . . (5)

For upper mass (m_2),

$$m_2 \ddot{x}_2 + 2k_2\left(x_2 - x_1\right) + F_D = 0$$

With equivalent damping,

$$m_2 \ddot{x}_2 + 2k_2\left(x_2 - x_1\right) + c_{eq}\left(\dot{x}_2 - \dot{x}_1\right) = 0 \quad . \quad . \ . \ . \ . \ .$$

(6)

The equations (5) and (6) are solved in MATLAB with following parameters,

m_1 = 4.05 kg, m_2 = 1.91 kg, k_1 = 19870 N/m, k_2 = 3670 N/m.

The frequency response curves for frequencies from 2Hz to 40Hz are plotted for lower and upper mass as shown in Fig 4.30, Fig 4.31, and Fig 4.32 for loads 0.1A, 0.5A and 1A current respectively.

The frequency response curves for frequencies from 2Hz to 40Hz are plotted for lower and upper mass for various friction loads and current

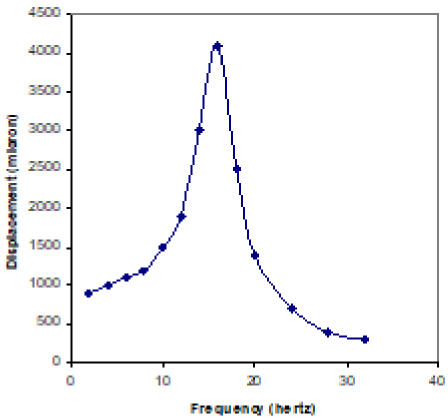

Fig 16 Response of SDOF system with system damping only

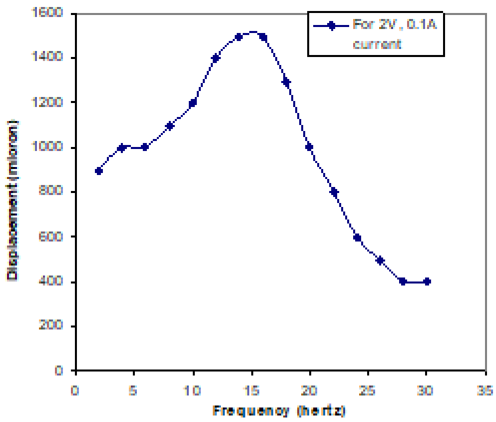

Fig 17 Response of SDOF system with MR Fig damping (2V, 0.1 A current)

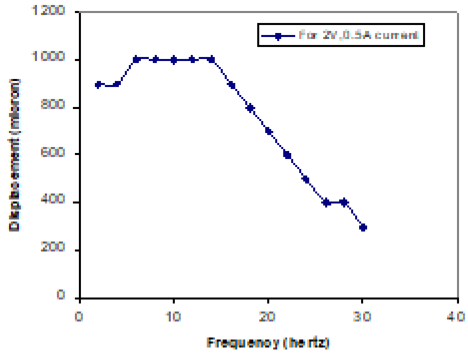

Fig. 18 Response of SDOF system with MR damping (2V, 0.5A current)

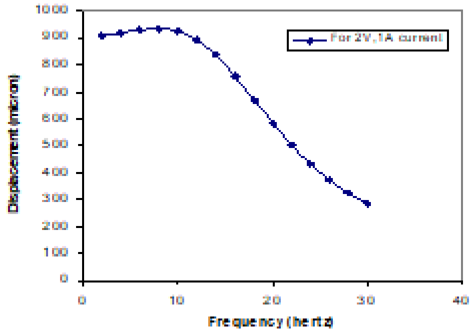

Fig 19 Response of SDOF system with MR damping (2V, 1A current).

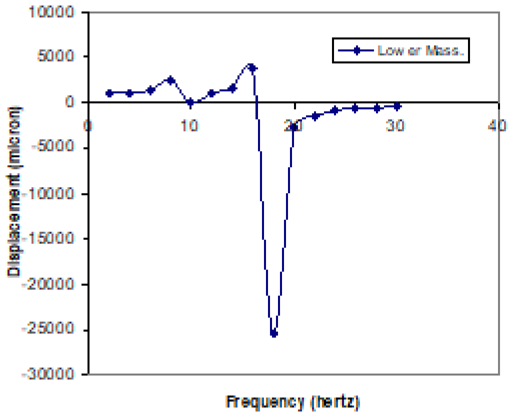

Fig 20 Response of lower mass in TDOF system without any damping.

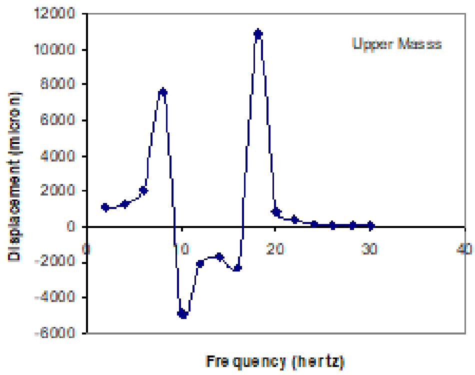

Fig 21 Response of upper mass in TDOF system without any damping

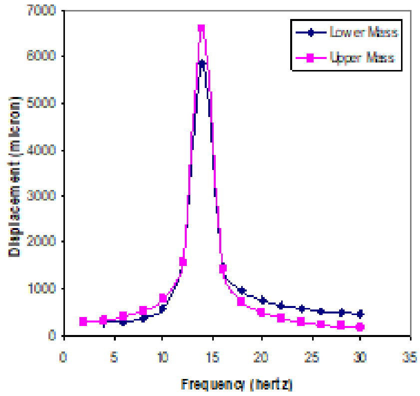

Fig. 22 Response of TDOF system with MR damping (2V, 0.1 A current)

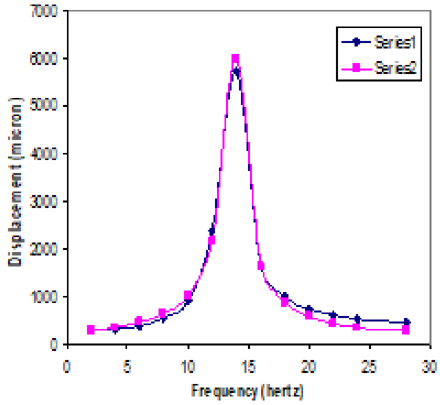

Fig 23 Response of TDOF system with MR damping (2V, 0.5A current)

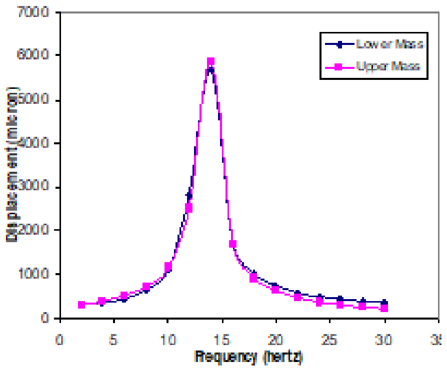

Fig 24 Response of TDOF system with MR damping (2V, 1A current).

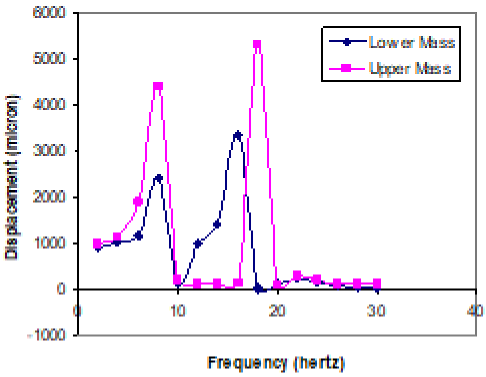

Fig 25 Frequency response curves of lower and upper mass for no damping.

VI. DISCUSSION OF RESULTS.

Experimental analysis of SDOF system was carried out separately for no damping, with friction damping, with MR damping. For no damping the peak displacement value at resonance was found to be 3.86 mm.

It is observed that as the current input to MR damper is increased the damping provided by damper increases.

It is clear that as the amount of damping is increased the peak displacement values at resonance decrease. It can also be read that the damping provided by friction damping is comparatively less than damping provided by MR damper

As the amount of damping is increased the displacement values go on decreasing implying that the damping factor is beyond critical damping.

After this experimental analysis was carried out with two degree of freedom system without any damping. The results are plotted in figure given below. It is seen in figure 25 that both masses undergo two peak vibration phases. Upper mass undergoes one peak at frequency near about 8 Hz and other at frequency 18 Hz. Lower mass undergoes first peak at 8Hz and other between 16Hz and 17Hz.

It is observed that both mass show one notable peak at frequency 14Hz and other small peak at frequency 24Hz. It is also observed that the upper small mass has peak displacement value more than peak displacement value for lower mass. This is also true for frequencies other than peak displacement frequency, but in those cases the difference in the displacement values not comparable.

It is further observed that as the amount of damping is increased the difference in the peak displacement values of two mass goes on decreasing. The results obtained are shown in table 1 and table 2.

Table 1 Difference in peak displacements of lower and upper mass for different damper configurations.

Type of Damping	Difference in peak displacements of lower and upper mass (micron)
Friction (1 kg)	1332
Friction (1.5 kg)	852
Friction (2 kg)	405
MR (0.1A, 2V)	754
MR (0.5A,2V)	193
MR(1A,2V)	140

Table.2 Difference in peak displacements of lower and upper mass for combined friction and MR damping.

Type of Damping	Type of Damping	Difference in peak displacements of lower and upper mass
Friction (1 kg)	MR (0.1A, 2V)	260
	MR (0.5A,2V)	79
	MR(1A,2V)	41
Friction (1.5 kg)	MR (0.1A, 2V)	203
	MR(0.5A,2V)	119
	MR(1A,2V)	96
Friction (2 kg)	MR(0.1A, 2V)	34
	MR(0.5A,2V)	87
	MR(1A,2V)	8

Equations of motion were derived using Newton's second law of motion, the resulting equations were solved in MATLAB and results were plotted.

Equation for SDOF system was derived and solved taking into consideration the system damping. There is slight difference in the values represented by analytical model and those obtained in experimental analysis. The reason for such results may be some amount of friction present in system which was not considered in analytical model; as the values given by analytical model are little more than experimental it can be concluded that damping in experimental analysis is slightly more that theoretical. As the difference in results is less than 7% the results obtained are acceptable.

Upper mass undergoes one peak at frequency near about 8 Hz and other at frequency 18 Hz. The lower mass shows negative displacement values just after second peak while upper mass shows negative displacement values just after first peak.

VII. CONCLUSIONS

From discussion of results following conclusions can be derived;

1. The damping of MR damper can be significantly increased by increasing the current input to damper circuit, provided that the current input is less than the saturation current level for the corresponding damper.
2. As the current input to the damper increases the damping goes beyond critical damping due to which the frequency response curve obtained shows a negative slope starting from static deflection curve. This is verified with analytical results obtained.
3. MR damper force is nonlinear function of displacement and velocity, it was also considered linear and the damping provided by MR damper was represented by equivalent damping. The linearization of friction and MR damper gives fairly accurate results.
4. In TDOF undamped system each mass shows peak at two frequencies other than their individual natural frequencies.

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