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EXPERIMENTAL ANALYSIS OF DAMPING CHARACTERISTICS FOR VARIOUS DAMPING MATERIALS

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ABSTRACT: The basic purpose of damper is to reduce the vibration and to have a better comfort and safety. The characteristic of damping system has an important influence on its design and overall performance of the system. In this paper magnetic damping effect is considered for characterization. This effect is the most recently used due to their low price and high reliability. An experiment has been conducted to establish the behavior of magnetic damper. The damping materials are water, magnet and SAE 40 grade oil used for characterizing the vibration behavior. The amplitude Verses frequency response curves are plotted for above damping materials.

INTRODUCTION

Damping can be defined as dissipation of oscillating energy. Oscillating energy includes vibration, noise and shock waves. The suspension systems currently in use are classified as passive, semi-active and active. Passive suspension systems are most commonly used damping systems because of their low price and high reliability.

The goal is to investigate characteristics of the damper for a single degree of freedom system, analyses under harmonic excitation of the base. The magnetic dampers are now being effectively deployed as vibration dampers in the suspension system to enhance the comfort and safety.

Fischer and Isermann[1] have shown how each part in a vehicle suspension use as ride comfort in dynamic model. Lin and Kanellakopoulos[2] shows system can have dual purpose of comfort and safety. Xu shows[3] how vibration of parts can effect the mechanism. Ebrahimi, Khamesee and Golnaraghi [4] demonstrate passive damping can be achieved by addition of viscous fluid to the active damper, which guaranties a failsafe damper in case of power failure. The electromagnetic dampers have lesser reliability because of dependence on external power source and higher weight. Lee, Park, Min and Chung[5] shows how to control seismic response of building structures using tuned liquid damper. Yau and Chen[6] shown vibration suppressing system using electric- hydraulic actuator design using squeeze film damper. Lee and Jee[7] the vibration of a flexible cantilever beam using active piezoelectric type servo damper to suppress both small and large amplitude vibrations. Martins et al[8]proposed a new hybrid damper design for vehicle suspension application. Linear actuator was the active unit and the hydraulic passive damping effect as a passive part. Lin and Roschke and Loh [9]proposed a hybrid base isolation with MR dampers, and showed that a combination of high damping rubber bearing isolators and MR damper can provide robust control of vibration for large civil engineering structures from a wide range of seismic events**.**

MODELING OF FABRICATED DAMPER AND EXPERIMENTAL SET UP

A beam made of mild steel having rectangular cross section with dimensions 20mm X 8 mm and length of 760 mm. One end of the beam is hinged and the other end is connected to frame using a spring. An exciter is placed on it at 460 mm from the hinged end. A DC motor of capacity 1800 rpm with a mass of 8.5kgs and disc connected to the shaft with eccentric masses 0.15kg. DC motor is used as exciter. Speed control device is used for control the speed of DC motor which generates vibration for system range (0-1800 rpm). A mechanical recording device is mounted on frame which records the amplitude of vibration of system.

The cantilever structure with attached mass is the most widely used configuration for spring mass device. The stiffness of the structure depends on the loading condition, material, and cross sectional area perpendicular to the direction of vibration. Passive system is the most used type in automobile suspensions. The main reasons are the simplicity, low cost and reliability of this solution. A spring and damper compose this suspension system, both fixed between the wheel supporting structure (unsprung mass) and the beam (sprung mass).

EXPERIMENTAL SETUP

MAGNETIC DAMPER

Two permanent magnets of diameter 5cm are taken one magnet is fixed to the cylinder at bottom and other magnet is connected to the system top surface. The cylinder and piston is arranged like such that poles of the both magnets are faced together. When beam vibrates the beam pushes the piston downwards. When the piston comes nearer to the cylinder bottom surface due to like pole magnet fixed at the bottom surface a repulsive force exerts on the piston. This automatically controls the level of vibration.

The damping coefficient is found for this magnetic damper through the experiment and it is compared with the other existing dampers

ANALYTICAL FORMULAE

The governing equation of motion of spring mass system

$$m\ddot{x} + c\dot{x} + k=0$$

In the case of cantilever beam we first calculated moment of inertia of the beam by taking length, thickness, and width of the beam. Moment of inertia for a rectangular cross-section can be obtained from the expression

Moment of inertia for the beam

$$I = \frac{bd^3}{12}$$

Using mass equivalent and stiffness of beam and spring, we can calculate natural frequency of the system having Static deflection of 25mm.

Mass equivalent of the system $m_{eq} = M_0 * \frac{l_1}{l}$

Spring stiffness $k_{spring} = \frac{m_{eq} * g}{\delta_{st}}$

Beam stiffness $k_{beam} = \frac{1}{\frac{l1^3}{3EI} + \frac{l1^2}{2EI}(l - l1)}$

Equivalent stiffness $k_{eq} = k_{spring} + k_{beam}$

Natural frequency of the system $\omega_n = \sqrt{\frac{k_{eq}}{m_{eq}}}$

Angular velocity of the system $\omega = \frac{2\pi N}{60}$

Harmonic force is given to the beam via exciter. The time period is noted based on the time period the damping coefficients are determined. Experimental values of time period for damped oscillation system T_d observed

Damped natural frequency of the system

$$\omega_n = \frac{2\pi}{T}$$

$$\omega_d = \frac{2\pi}{T_d}$$

$$\omega_d = \omega_n \sqrt{1 - \zeta^2}$$

From above equation we calculated ζ, then we calculate damping coefficient

Damping coefficient $\zeta = \frac{c}{c_c}$

The steady state amplitude ‘X’

$$\frac{X}{\left(\frac{m_0 * e}{m_{eq}}\right)} = \frac{\left(\frac{\omega}{\omega_n}\right)}{[1 - \left(\frac{\omega}{\omega_n}\right)^2]^2 + [2\zeta \frac{\omega}{\omega_n}]^2}$$

At various static deflections of the beam the amplitudes of vibrations have found. The damping coefficients (ζ) are found using static deflections

Table1: Damping Coefficients at Different Static Deflections

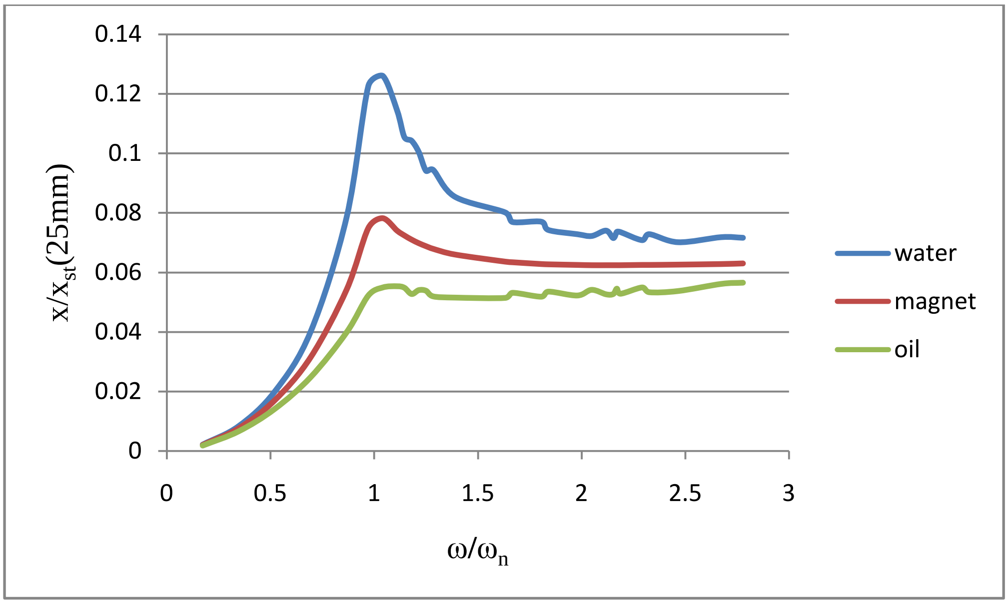
Damping materials	**static deflections (x_{st})**				Average ζ values
	25	**20**	**15**	**10**	
Water	0.211	0.24	0.273	0.33	0.2633
Magnet	0.3715	0.4841	0.5329	0.4687	0.4643
SAE 40 oil	0.9012	0.8695	0.7513	0.106	0.657

RESULTS AND DISCUSSION

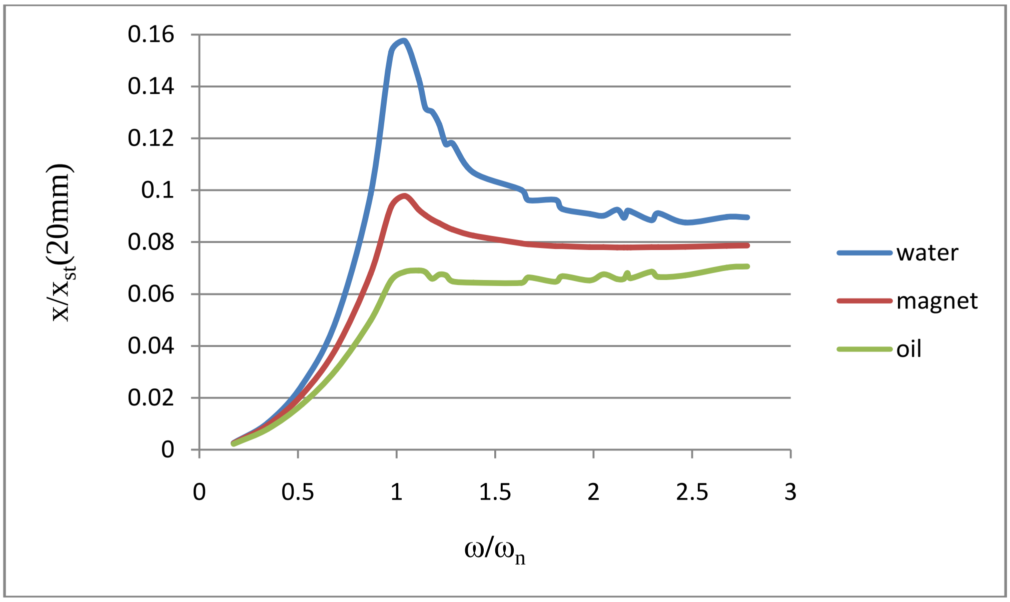
The natural frequency of the system is 30.1475 rad/sec. The variation of amplitudes vibrations of a system at different speeds for static deflection of 25mm and 20mm

Table 2: Variation of Amplitudes vibrations of a system at different speeds for static deflection of 25mm and20mm

Speed	ω/ω_n	x/x_{st} (25mm)			x/x_{st} (20mm)		
		water	magnet	oil	water	magnet	oil
50	0.1737	0.0021	0.002	0.0018	0.0026	0.0025	0.0023
100	0.3474	0.0083	0.0076	0.0066	0.0104	0.0095	0.0081
150	0.521	0.0199	0.017	0.0141	0.0249	0.0212	0.0176
200	0.6947	0.0401	0.0315	0.0248	0.0502	0.0393	0.031
250	0.8684	0.0787	0.0543	0.0399	0.0984	0.0679	0.0499
280	0.9726	0.1226	0.0749	0.0522	0.1532	0.0936	0.0652
300	1.0421	0.126	0.0782	0.0549	0.1575	0.0978	0.0686
320	1.1115	0.1143	0.074	0.0553	0.1429	0.0925	0.0691
330	1.1463	0.1054	0.0724	0.0548	0.1317	0.0904	0.0685
340	1.181	0.1042	0.071	0.0527	0.1302	0.0887	0.0659
350	1.2158	0.1004	0.0698	0.054	0.1254	0.0873	0.0675
360	1.2505	0.0941	0.0688	0.0538	0.1177	0.086	0.0672
370	1.2852	0.0944	0.0679	0.0519	0.118	0.0849	0.0649
400	1.3894	0.0854	0.066	0.0515	0.1068	0.0825	0.0644
470	1.6326	0.0801	0.0636	0.0514	0.1001	0.0795	0.0643
480	1.6673	0.0769	0.0634	0.0531	0.0962	0.0792	0.0664
520	1.8063	0.077	0.0628	0.0518	0.0963	0.0785	0.0647
530	1.841	0.0742	0.0627	0.0535	0.0928	0.0784	0.0669
570	1.9799	0.0728	0.0625	0.0522	0.0909	0.0781	0.0653
590	2.0494	0.0722	0.0624	0.0541	0.0902	0.0781	0.0676
610	2.1189	0.074	0.0624	0.0526	0.0925	0.078	0.0658
620	2.1536	0.0715	0.0624	0.0527	0.0894	0.078	0.0659
625	2.171	0.0736	0.0624	0.0545	0.0921	0.078	0.0681
630	2.1884	0.0735	0.0624	0.0528	0.0919	0.078	0.0661
660	2.2926	0.0708	0.0625	0.0549	0.0884	0.0781	0.0686
670	2.3273	0.0728	0.0625	0.0533	0.0911	0.0781	0.0666
710	2.4662	0.0701	0.0626	0.0537	0.0876	0.0782	0.0672
770	2.6747	0.0718	0.0628	0.0561	0.0897	0.0786	0.0702
800	2.7789	0.0716	0.063	0.0565	0.0895	0.0787	0.0706



Graph 1: Amplitudes V/s Frequency ratio at static deflection 25mm

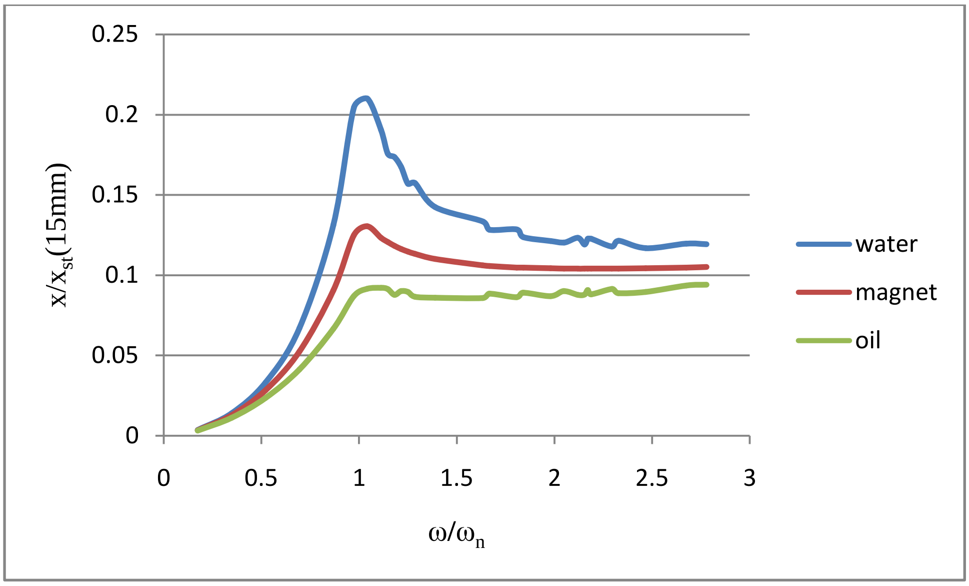


Graph 2: Amplitudes V/s Frequency ratio at static deflection 20mm

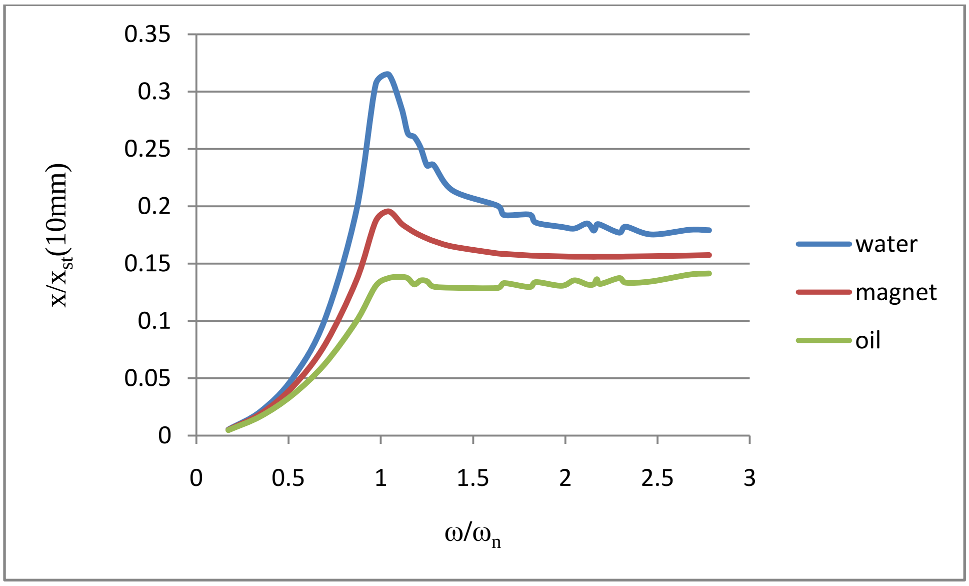
Table 3: Variation of Amplitudes vibrations of a system at different speeds for static deflection of 15mm and10mm

Speed	ω/ω_n	x/x_{st} (15mm)			x/x_{st} (10mm)		
		water	magnet	oil	water	magnet	Oil
50	0.1737	0.0035	0.0034	0.0031	0.0052	0.0051	0.0046
100	0.3474	0.0139	0.0127	0.0111	0.0209	0.019	0.0166
150	0.521	0.0331	0.0283	0.0235	0.0497	0.0424	0.0353

200	0.6947	0.0669	0.0524	0.0413	0.1003	0.0787	0.062
250	0.8684	0.1312	0.0905	0.0666	0.1968	0.1358	0.0998
280	0.9726	0.2043	0.1248	0.0869	0.3065	0.1872	0.1304
300	1.0421	0.21	0.1304	0.0915	0.3149	0.1956	0.1372
320	1.1115	0.1905	0.1233	0.0921	0.2858	0.1849	0.1382
330	1.1463	0.1757	0.1206	0.0913	0.2635	0.1809	0.137
340	1.181	0.1736	0.1183	0.0878	0.2604	0.1774	0.1317
350	1.2158	0.1673	0.1163	0.0901	0.2509	0.1745	0.1351
360	1.2505	0.1569	0.1147	0.0896	0.2354	0.172	0.1344
370	1.2852	0.1573	0.1132	0.0865	0.236	0.1698	0.1298
400	1.3894	0.1424	0.11	0.0859	0.2136	0.1649	0.1288
470	1.6326	0.1335	0.106	0.0857	0.2003	0.1589	0.1286
480	1.6673	0.1282	0.1056	0.0885	0.1923	0.1585	0.1328
520	1.8063	0.1284	0.1047	0.0863	0.1926	0.1571	0.1294
530	1.841	0.1237	0.1046	0.0891	0.1856	0.1569	0.1337
570	1.9799	0.1213	0.1042	0.087	0.1819	0.1563	0.1305
590	2.0494	0.1203	0.1041	0.0901	0.1805	0.1561	0.1352
610	2.1189	0.1233	0.104	0.0877	0.1849	0.1561	0.1316
620	2.1536	0.1191	0.104	0.0879	0.1787	0.156	0.1318
625	2.171	0.1227	0.104	0.0908	0.1841	0.1561	0.1362
630	2.1884	0.1226	0.104	0.0881	0.1839	0.1561	0.1321
660	2.2926	0.1179	0.1041	0.0914	0.1769	0.1561	0.1372
670	2.3273	0.1214	0.1041	0.0888	0.1821	0.1562	0.1332
710	2.4662	0.1168	0.1043	0.0896	0.1753	0.1565	0.1343
770	2.6747	0.1196	0.1047	0.0936	0.1794	0.1571	0.1403
800	2.7789	0.1193	0.105	0.0941	0.179	0.1575	0.1412



Graph 3: Amplitudes V/s Frequency ratio at static deflection 15mm

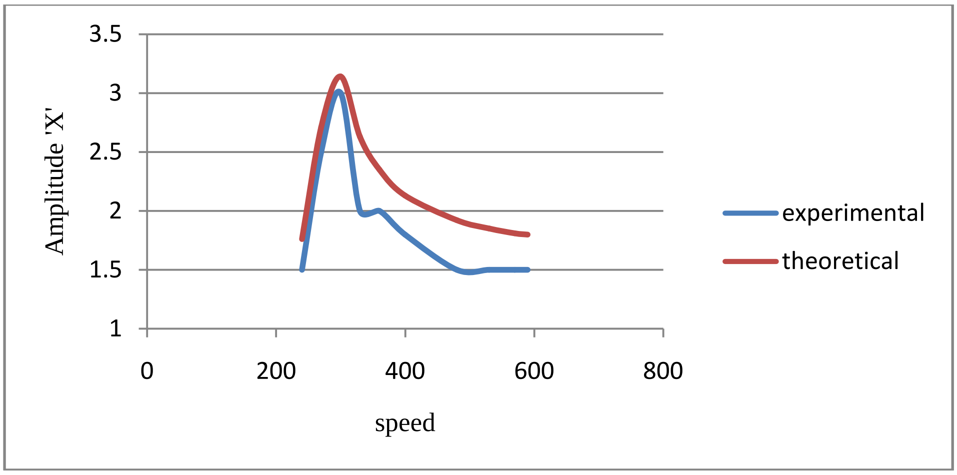


Graph 4: Amplitudes V/s Frequency ratio at static deflection10mm

Table 4: Experimental and Theoretical values of water at different speeds

Speed	water	
	Experimental	Theoretical
240	1.5	1.76
270	2.5	2.71
300	3	3.14
330	2	2.63
360	2	2.35

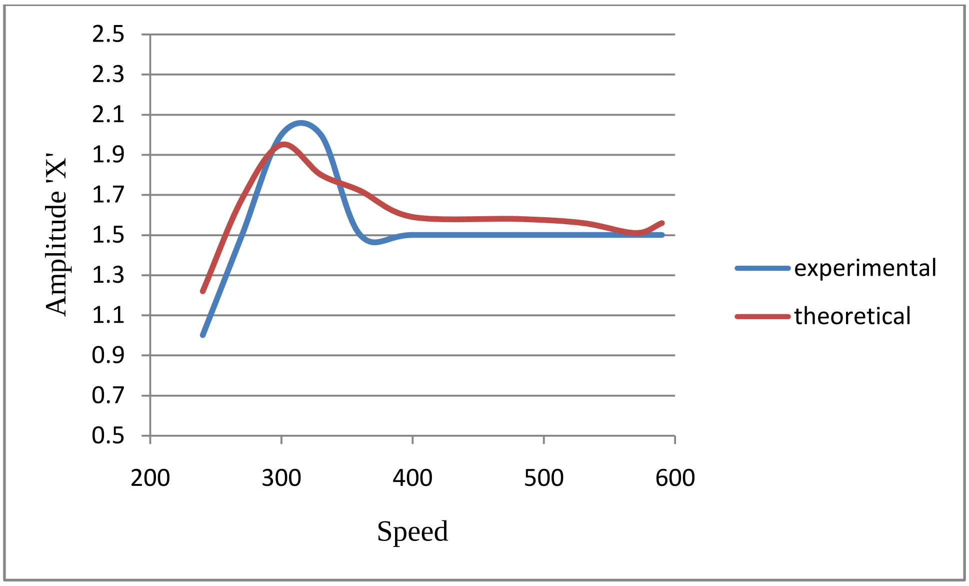
400	1.8	2.13
480	1.5	1.92
530	1.5	1.85
570	1.5	1.81
590	1.5	1.8



Graph 5: Speed V/s Amplitude for water

Table 5: Experimental and Theoretical values of magnet at different speeds

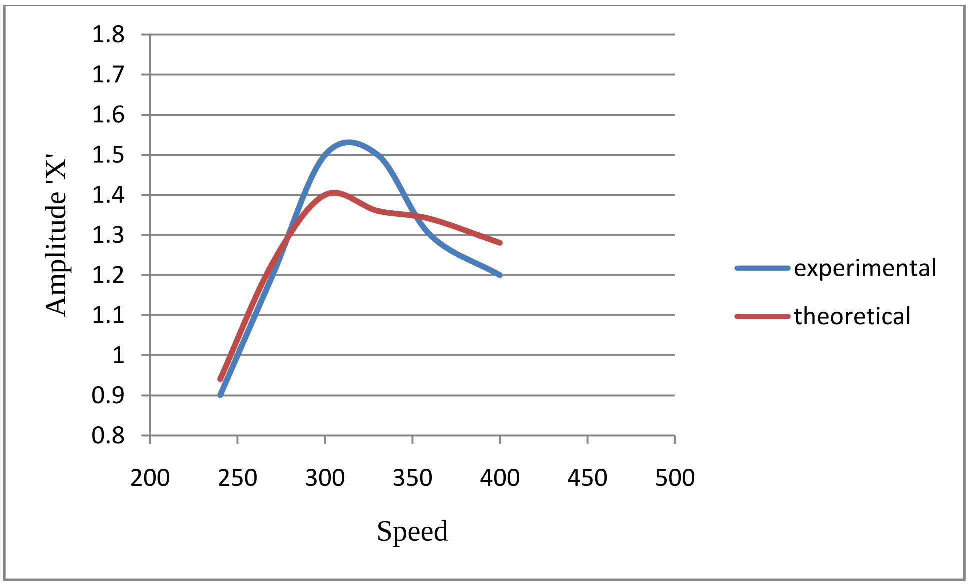
speed	Magnet	
	Experimental	Theoretical
240	1	1.22
270	1.5	1.68
300	2	1.95
330	2	1.8
360	1.5	1.72
400	1.5	1.59
480	1.5	1.58
530	1.5	1.56
570	1.5	1.51
590	1.5	1.56



Graph 6: Speed V/s Amplitude for magnet

Table 6: Experimental and Theoretical values of SAE 40 grade oil at different speeds

speed	SAE 40 grade oil	
	Experimental	Theoretical
240	0.9	0.94
270	1.2	1.23
300	1.5	1.4
330	1.5	1.36
360	1.3	1.34
400	1.2	1.28



Graph 7: Speed V/s Amplitude for SAE 40 grade oil

CONCLUSION

Magnetic damper is proposed to suppress the vibrations at the free end of a cantilever beam. The experiment reveals that the proposed magnetic damper is effectively control the vibration. Variation of amplitudes of vibration at different speeds for the magnetic damper was found. The damping coefficient of the magnetic damper is C=143.96 Ns/m. After comparison of oil, water and magnet the one control of vibration lies between water and oil for the provided magnetic damper. Magnetic damper does not require lubrication and maintenance is less. Effectiveness of damper can be improved by providing electromagnetic damper.

Appendix

l = length of the beam
l_1 = length of the beam from fixed end to the exciter mass
M_0= mass of the exciter
b = width of the rectangular beam
d =thickness of the rectangular beam
I = moment of inertia of the beam
m_{eq} = mass equivalent of the exciter mass in kg
δ_{st} = static deflection
E =young's modulus of elasticity
k_{spring} =stiffness of the spring
k_{beam} =stiffness of the beam
k_{eq} = equivalent stiffness
ω_n = natural frequency
ω_d = damped natural frequency
ω = angular velocity
N =speed in rpm
c =damping coefficient
ζ = damping factor or damping ratio
X = amplitude of steady state forced vibrations
e = unbalanced mass eccentricity
m_0 = weight of the unbalanced mass
c_c = critical damping coefficient
T = time taken in sec
T_d = damped system time.

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