Experimental Damping Estimation of Material by Sinusoidal Base Excitation

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Abstract: ***The material damping, which means energy dissipation in materials under cyclic loading, is an important design consideration for vibrating structures especially in the airplane and automobile engineering. The main objective of this work is to determine natural frequency and damping ratio by sweep sine test and half power bandwidth method respectively. Aluminium Beams of required size are prepared and are excited using an Electro-Dynamic Shaker excitation technique over the frequency range of 5-500 Hz. The modal analysis of the beam is also done on ANSYS platform as well as theoretically. A good concurrence for natural frequency is found in all these three methods. Based on the experimental results, damping ratio is determined by using half power band width method. The effect of three major geometrical parameters on material damping is also investigated. For this purpose, Design of Experiment (Taguchi L_9 orthogonal Array) is performed to find the parameters combination to give the optimum damping ratio. Analysis of variance (ANOVA) and S/N (signal to noise) ratios analysis is also performed. The results show that the amplitude of excitation significantly affects the damping ratio. It is also found that the predicted results obtained using regression equations are in good agreement with experimental observations.***

Keywords: Damping Ratio, Sine Sweep test, Electro-Dynamic Shaker, ANOVA**.**

1. Introduction

Damping is the conversion of mechanical energy of a vibrating structure into thermal energy/heat. Elastic stress-strain plots for a material is different for loading and for unloading process. When such material is subjected to cyclic loading, a hysteresis loop emerges on the stress strain plots and the area of this loop is the energy dissipated per unit volume per cycle. It is well known that all materials have certain amount of material damping, which is obviously as dissipation of energy under vibratory motion [1]. The energy dissipation caused by the material damping depends on many factors like amplitude of stress, number of cycles, porosity, grain size, temperature and geometry [2]. Among the non-ferrous metals, the Aluminium (Al) is a widely used in the manufacturing of different engineering and commercial products as it has good strength to weight ratio. Because of this reasons, the Aluminium material is selected to study its damping characteristics by various means.

For identifying, understanding, simulating dynamic behaviour and responses of structures, experimental testing has become an important tool. The experimental modal analysis (EMA) or modal testing is a non-destructive testing strategy based on vibration responses of the structures. In Forced vibration, the energy supplied by the excitation balances the energy loss. This is significant when the system is close to its resonating frequency under self excitation or external excitation. Hence, it is really necessary to predict the damping behaviour of such systems under operating conditions to prevent catastrophic failure occurring due to resonance. The damping ratio increase as the acceleration level increases due to increase of stresses in the beam [3].

In this paper, natural frequency and damping ratio is determined by sweep sine test and half power bandwidth method respectively on Aluminium beam of required size. The beams are excited using an Electro-Dynamic Shaker excitation technique over the frequency range of 5-500 Hz. The modal analysis is done on ANSYS platform as well as theoretically to determine natural frequency. A good agreement has been obtained for natural frequency among three methods. This has validated the experimental process adopted in this work. The damping ratio is evaluated for the material experimentally. Analysis of variance (ANOVA) and S/N (signal to noise) ratios analysis is performed to determine the effect of the three major geometrical parameters on the material damping. The results showed that the amplitude of excitation significantly affects the damping ratio. The optimum result is validated using regression method. The work in this paper will be very useful in predicating optimum damping ratio of the material.

2. Damping Measurement

The two most popular procedures to evaluate damping ratio are time domain approach and frequency domain approach [1], [4], [5]. In the time domain approach, the fundamental principle is energy lost from the oscillatory system which results in decay of amplitude of the oscillation. Free vibration technique is the most accepted method used to evaluate damping ratio in time domain analysis. But unlike time domain analysis, frequency domain analysis is not based on any time history data. The time domain analysis does not contain any multiple frequency based information which may be important to characterize the dynamic response of a system. Forced vibration is the main concept behind frequency domain analysis. The frequency response function can be determined experimentally. By applying a continuous force and sweeping the frequency, one measures the resulting vibration and calculates the frequency response functions; hence characterize the system. Some of the popular experiments in the frequency domain analysis are impulse test method and sweep sine method. The name sweep sine arises due to the fact that the system is made to respond between two frequency limits. The supposed natural frequency must lie in between these two frequency limits. When the input frequency approaches to the natural frequency of the system; then the amplitude level increases significantly. This is known as resonance peaks and is clearly visible in the frequency response curve. As soon as the sweeping frequency passes over the resonant frequency, the system vibrating amplitude keeps on reducing. In this way, to study the damping characteristics, resonant frequency response amplitude becomes very significant. When the input force is constant during excitation, the output can be still viewed as Frequency Response Function (FRF). This output will be used for calculation of damping ratio. The FRF for a typical single degree of freedom (SDOF) system is shown in Figure 1.

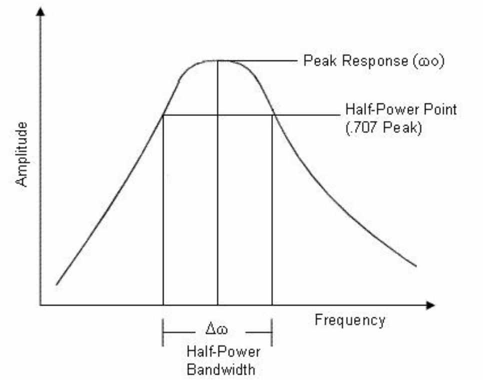


Figure 1: Frequency Response Function

For quantitative measurement of damping ratio, the half-power bandwidth method is employed. In this method, the damping ratio of the structure is determined as the ratio of $\Delta\omega$ and $2\omega_o$ and is given as

$$\xi = \frac{\Delta\omega}{2\omega_o} \tag{1}$$

Where ξ is the damping ratio, ω_o is resonant frequency and $\Delta\omega$ is frequency difference corresponding to a 0.707 drop from the peak of the response curve.

3. Theoretical Analysis to Determine Natural Frequency

The beam is a continuous system as the mass is continuously distributed along its length. For a prismatic beam, its transverse vibration (wave) equation is given by Euler's as:

$$EI\frac{\partial^4 y}{\partial x^4} + \rho A\frac{\partial^2 y}{\partial t^2} = 0 \tag{2}$$

Where y is the deflection at a distance x from the origin and at a time t, E is the Young's modulus of elasticity, I is the moment of inertia, ρ is the density of the beam material and A is the cross-sectional area of the beam. The deflection curve of the beam (y), may be approximated by a function

$$y(x,t) = \phi_n(x)e^{i\omega_o t} \tag{3}$$

After substitution of equation (3) into equation (2) gives

$$\frac{\partial^4 \phi_n(x)}{\partial x^4} - \beta_n^4 \phi_n(x) = 0 \tag{4}$$

Where ϕ_n is the characteristic function describing the deflection of the n^{th} mode and β_n is given as

$$\beta_n^4 = \frac{\rho A \omega_o^2}{EI} \tag{5}$$

Equation (4) is to be solved for ω_o using the ϕ_n approximation by applying the suitable boundary conditions. These boundary conditions for a fixed-free beam are given in Table 1.

Table 1: Boundary condition for fixed-free beam

Condition	*Boundary condition*
Fixed	$y=0 \ \& \ \frac{\partial y}{\partial x} = 0$
Free	$\frac{\partial^2 y}{\partial x^2} = 0 \ \& \ \frac{\partial}{\partial x}\left(EI\frac{\partial^2 y}{\partial x^2}\right) = 0$

The theoretical natural frequency, ω_o, is determined as:

$$\omega_o = (\beta_n)^2\sqrt{\frac{EI}{\rho A L^4}} \tag{6}$$

Where L is to be the length of the beam and the values for $(\beta_n)^2$ is given in Table 2

Table 2: Values of β_n and $(\beta_n)^2$ as used in equation 6

n	(β_n)	$(\beta_n)^2$
1	1.8751	3.5160
2	4.6941	22.0345
3	7.8548	61.6972

For the Aluminium beam (450 mm×4 mm×50 mm), using the above equation, the theoretical natural frequency for the first mode is determined as 16.04 Hz.

4. Experimental Set Up

The experimental setup consists of an electro dynamic shaker, PCB™-ICP accelerometer, 8 channel vibration controllers and data acquisition system and fabricated cantilever beams. The dimensions of the beams are so decide to have adequately spread over the aforementioned frequency range of the system *i.e.* 5-500 Hz for the resonance frequencies. Based on this, the parameters and their level are decided and are shown in Table 3.

Table 3: The parameter and their range for experiments on cantilever beam.

Parameters (symbol)	*Unit*	*Level*	*Range*		
			Low	*Mid*	*Upper*
Length (L)	mm	3	350	400	450
Thickness (T)	mm	3	4	6	8
Acceleration for first mode (A)	g	3	0.1	0.2	0.3
Acceleration for second mode (A)	g	3	1	2	3

In this method, one excites one end of a long, slender, material (beam) and measures its acceleration responses at the opposite (free) end. Assuming negligible losses due to air loading, the resonance peak of the transfer function are used to determine material damping via half power band width method. The half power band width method is based on the assumption that modal damping ratio are small enough to allow the half power points to be clearly indentified within the measured data. In the case where the mode to be analyzed is heavily affected by its adjoining modes, or the damping ratio of the mode is high, the estimates in damping shows increase in error [6], [7].The specimen's response in the $\pm y$ direction is measured by the accelerometer placed at the driven end (base) of the specimen as shown in Figure 2. This data is used to normalize the free end data. Another accelerometer is also positioned to measure the response at the free end of specimen.

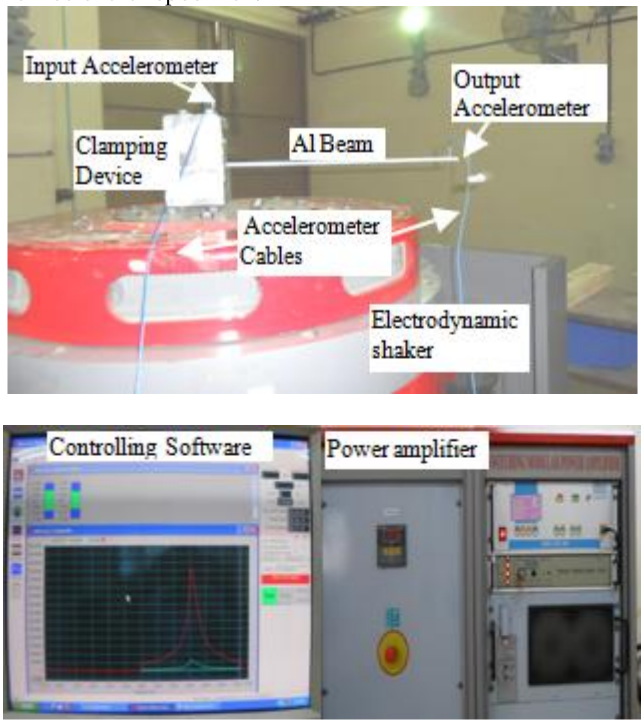
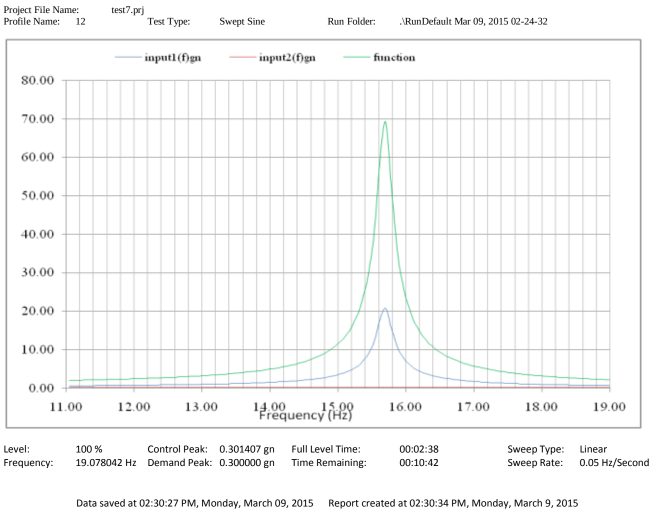


Figure 2: Experimental Setup used to measure Damping ratio and natural Frequency

As every resonance frequency exhibits local maxima at the free end of the specimen, the transfer function ratio from the specimen's free end to driven end is computed. A graph is plotted between response (transfer function and acceleration) and frequency. The damping ratio is calculated from the measured data via half power band width method [4], [8] (Equation 1). The process is repeated for each cantilever beam.

5. Experimental Procedures to Determine the Natural Frequency and the Damping Ratio

Forced vibration sine sweep is induced in the test specimens to obtain its dynamic characteristics *i.e.* natural frequencies and damping ratios. The beam is clamped on the armature of 3500 Kgf electro-dynamic shaker that provides an external (transverse to the axis of the beam) sinusoidal excitation at the base of the beam. The excitation is applied at the root of the beam (test specimen) mounted horizontally by the clamping device. A data acquisition system is used to store the experimental data and transfer it to the PC for further post-processing. The Frequency response functions (FRFs) is obtained using Dactron^{TM} vibration analyzer. The FRFs were processed using vibration analysis packages to identify natural frequencies, damping ratio and mode shapes of the beams.



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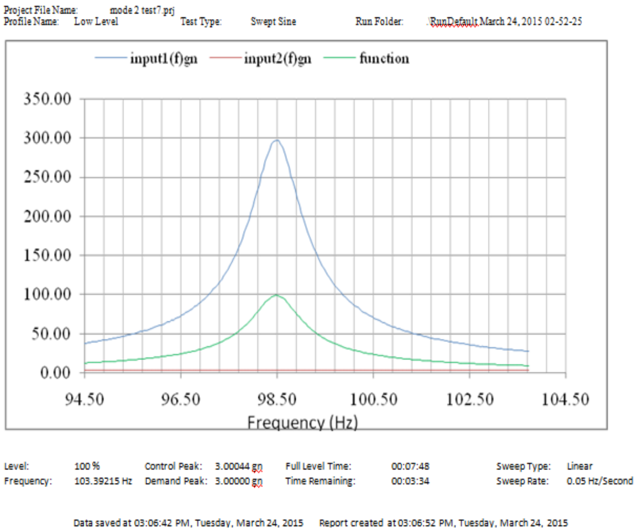


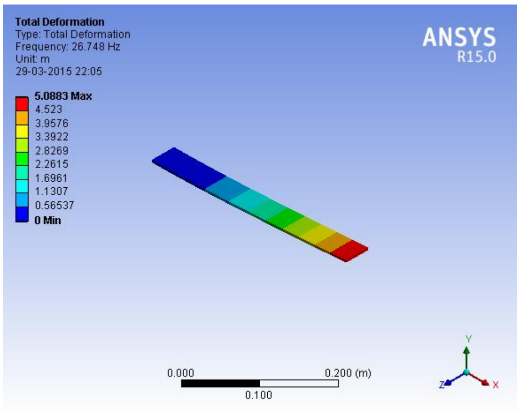
(b)

Figure 3: Frequency Response Function **(a)** first mode and **(b)** second mode

While setting the swept-sine parameter in the system, one must ensure that the time duration in the vibration measurement software should be greater than the total time of excitation. The sine sweep rate can be as low as 0.05 Hz/sec. The Figure 3a shows the FRF for the Aluminium beam ((450 mm×4 mm×50 mm) under acceleration of 0.3g. From the Figure 3a, the natural frequency is obtained as 15.69 Hz which is in good agreement with the theoretically determined natural frequency. It thus validates the experimental process. Subsequently, the damping ratio is determined using Equation 1 and comes out to be as 0.007208. Similarly for the second mode, the damping ratio is evaluated using the Figure 3b for the 7th experiment as mentioned in Table 7.

6. ANSYS Analysis to Determine Natural Frequency

The vibration of the cantilever beam is modelled using the ANSYS software. In the main menu, in pre-processor of the software, the beam element is selected. The Young's Modulus and poison's ratio values are given as input. By using modeling option, the beam is generated as Two-dimensional entity and is extruded along its length to obtain 3D model of the beam. By using meshing tool option, the beam is meshed and by applying boundary conditions, the modal analysis is performed by using Block Lanczos method. The results are obtained from General Post Processor. The vibration of the beam for the first and second mode is shown in Figure 4.



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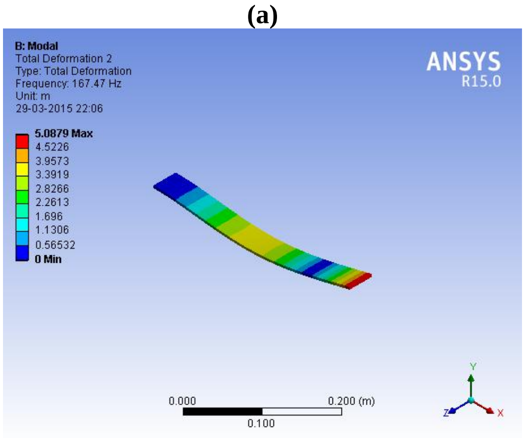


(b)

Figure 4: Beam vibrating at **(a)** first mode and **(b)** second mode

Table 4: Comparison of natural frequency calculated of First mode by various methods

Exp. No	*Beam Dimension (mm)*	$f_{n\text{-}th}$ *(Hz)*	$f_{n\text{-}ANSYS}$ *(Hz)*	$f_{n\text{-}exp}$ *(Hz)*	*%age Error w.r.t. theory*
1	350×4×50	26.52	26.748	25.90	2.33
2	350×6×50	39.78	40.095	39.04	1.86
3	350×8×50	53.04	53.583	51.83	2.28
4	400×4×50	20.31	20.483	19.80	2.51
5	400×6×50	30.46	30.676	29.65	2.66
6	400×8×50	40.61	40.878	39.56	2.58
7	450×4×50	16.04	16.169	15.69	2.18
8	450×6×50	24.07	24.22	23.29	3.24
9	450×8×50	32.08	32.276	31.32	2.36

The value of natural frequency for the first mode by this method for the Aluminium beam (450 mm×4 mm×50 mm) is obtained as 16.169 Hz. Using the data for the beam given in Table 1, the above the modeling has been carried out for each beam. The result in the form of natural frequency for each beam is obtained and is tabulated in Table 4. The Table 4 shows a very good agreement of experiment results with theory and ANSYS results and thus validates the experimental process adopted for this work.

Table 5: Comparison of natural frequency calculated of Second mode by various methods

Exp. No	*Beam Dimension (mm)*	$f_{n\text{-}th}$ *(Hz)*	$f_{n\text{-}ANSYS}$ *(Hz)*	$f_{n\text{-}exp}$ *(Hz)*	*%age Error w.r.t. theory*
1	350×4×50	166.2	167.47	162.11	2.46
2	350×6×50	249.31	250.86	243.07	2.50
3	350×8×50	332.42	333.16	325.43	2.10
4	400×4×50	127.28	128.28	124.92	1.85
5	400×6×50	190.93	192.01	186.13	2.51
6	400×8×50	254.516	255.98	248.55	2.34
7	450×4×50	100.52	101.27	98.482	2.02
8	450×6×50	150.85	151.63	147.21	2.41
9	450×8×50	201.05	202.65	196.86	2.08

7. Design of Experiment as Applied to Damping Ratio

In order to optimize the damping ratio of the material, the design of experiment concepts is used. For conducting and acquiring the experiment data in control way, Taguchi methods have been employed. The advantage of the Taguchi method is that it reduces experimental time, number of experiments and thus reduces its cost and saves the effort of conducting the experiments besides determining the significant factors quickly [9]. The signal to noise ratio and analysis of variance (ANOVA) statistical tool is employed to point out the significance of input process parameters on its output response. The steps for the Taguchi optimization study are as follows:

- Select the quality characteristics
- Select control parameters/factors
- Select Taguchi orthogonal array (L_9 Array)
- Conduct Experiments of experiment for damping ratio
- Analysis results (S/N ratio , ANOVA)
- Predict optimum performance
- Confirmation experiments

The quality characteristics and control parameters/factors are determine based on literature survey and are given in Table 1. Based on it, Taguchi orthogonal array L_9 Array is selected. These levels are shown by first three columns of Table 6 & 7 for first and second modes respectively. The experiments for these levels and modes are conducted to determine the damping ratio of the material. The response parameter of the system *i.e.* damping ratio is shown in fifth column of the Table 6 & 7 first and second modes respectively.

8. Results and Discussion

Signal-to-noise (S/N) ratio is derived from the damping ratio. With the S/N and ANOVA analyses, the combination of the parameters can be predicted which gives the optimum values of the damping. There are three types of S/N ratios depending upon the type of characteristics *i.e.* Lower is the better (LB), nominal is the better (NB), higher is the better (HB).

Table 6: Design matrix and experimental observations for first mode

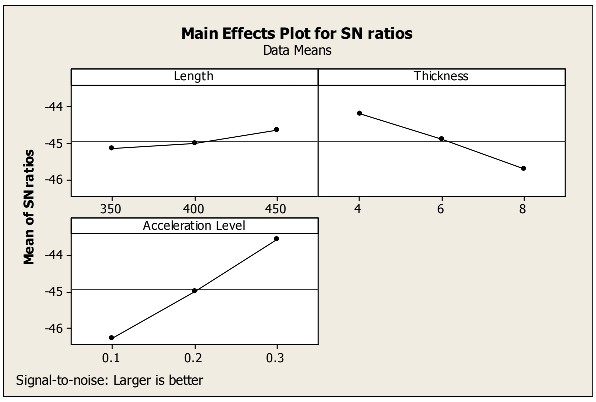
Exp. No	*Length L (mm)*	*Thickness T (mm)*	*Acc^n Level A (g)*	*Damping Ratio*	*S/N Ratio*
1	350	4	0.1	0.00514	-45.7807
2	350	6	0.2	0.00535	-45.4329
3	350	8	0.3	0.00615	-44.2225
4	400	4	0.2	0.00632	-43.9857
5	400	6	0.3	0.00658	-43.6355
6	400	8	0.1	0.00425	-47.4322
7	450	4	0.3	0.007208	-42.8437
8	450	6	0.1	0.00523	-45.6300
9	450	8	0.2	0.00530	-45.5145

Table 7: Design matrix and experimental observations for second mode

Exp. No	*Length L (mm)*	*Thickness T (mm)*	*Acc^n Level A (g)*	*Damping Ratio*	*S/N Ratio*
1	350	4	1	0.00360	-48.8739
2	350	6	2	0.00350	-49.1186
3	350	8	3	0.00427	-47.3914
4	400	4	2	0.00421	-47.5144
5	400	6	3	0.00445	-47.0328
6	400	8	1	0.00290	-50.7520
7	450	4	3	0.00505	-45.9342
8	450	6	1	0.00365	-48.7541
9	450	8	2	0.00340	-49.3704

In this study, material damping ratio must take the highest value and so higher is the better concept is adopted [10]. The required S/N ratios are computed using the Equation 7 and the computed values are recorded in the sixth column of the Table 6 & 7 for first and second modes respectively.

$$S/N\ Ratio = -10\log\frac{1}{n}\sum_{i=1}^{n}\frac{1}{y_i^2} \qquad (7)$$



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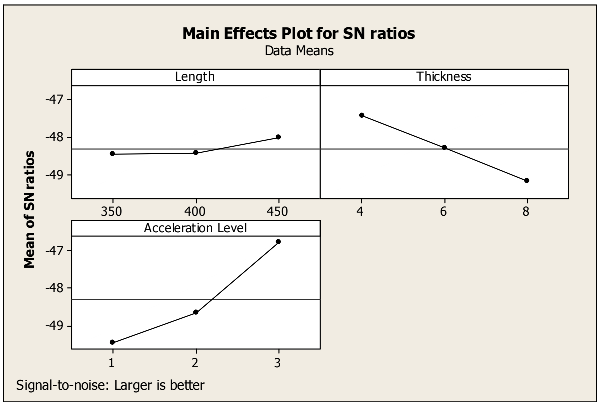

(b)

Figure 5: S/N ratio Plot for **(a)** First mode and **(b)** Second Mode due to various parameters

Where, n is the number of repetitions/observations and y_i is the observed data. The measured responses for S/N ratio is analyzed by using a standard commercial statistical software MINITAB^{TM}16 package and the respective plots are shown in Figure 5 for first and second mode of the vibration.

Table 8: ANOVA for Damping Ratio at Fundamental Mode (First mode)

Source	*DF*	*Seq SS & Adj SS*	*Adj MS*	*F*	*P*
Model	6	6.40×10^{-6}	1.06×10^{-6}	14.74	0.065
Length (L)	2	2.01×10^{-7}	1.00×10^{-7}	1.39	0.418
Thickness (T)	2	1.46×10^{-6}	7.34×10^{-7}	10.14	0.090
Acceleration Level (A)	2	4.73×10^{-6}	2.36×10^{-6}	32.70	0.030
Error	2	1.44×10^{-7}	7.23×10^{-8}		
Total	8	6.54×10^{-6}			
S = 0.000269053 R-Sq = 97.79% R-Sq(adj) = 91.16%					
$F_{0.25}$,2,2 = 3.0; $F_{0.1}$,2,2 = 9.0; $F_{0.05}$,2,2 = 19.0; $F_{0.025}$,2,2 = 39.0; $F_{0.01}$,2,2 = 99.0					

From the Figure 5, the parameter giving the optimum damping ratio can be derived and comes out to be length = 450 mm, thickness = 4 mm and acceleration level = 0.3 g for the first mode while for the second mode the parameters are length = 450 mm, thickness = 4 mm and acceleration levels = 3g.

The significant parameter influencing the damping ratio in the cantilever beam is determined using analysis of variance (ANOVA). The ANOVA tool helps in appropriately testing the significance of all the main factors and their interactions by evaluating the mean square against an estimate of the experimental errors at specific confidence level. The ANOVA results for damping ratio for first and second mode are illustrated in Table 8 and 9 respectively. The higher F value for acceleration level predicts that it has a significant effect on the damping ratio. The first and second mode damping ratio model is significant at even at 90% confidence level.

In Taguchi's design approach, final step is the experimental confirmation test to verify its results. The optimal conditions are set for the significant factors and a selected number of experiments are run under specified conditions. In this study, a confirmation (optimum) experiment as decided by the Taguchi's method comes out to be same as that of experiment number 7.

Table 9: ANOVA for Damping Ratio at (Second Mode)

Source	*DF*	*Seq SS & Adj SS*	*Adj MS*	*F*	*P*
Model	6	3.31×10^{-6}	5.52×10^{-7}	10.76	0.087
Length (L)	2	9.56×10^{-8}	4.78×10^{-8}	0.93	0.518
Thickness (T)	2	8.76×10^{-7}	4.38×10^{-7}	8.53	0.105
Acceleration Level (A)	2	2.34×10^{-6}	1.17×10^{-6}	22.82	0.042
Error	2	1.02×10^{-7}	5.13×10^{-8}		
Total	8	3.42×10^{-6}			
S = 0.000226667 R-Sq = 97.00% R-Sq(adj) = 87.98%					
$F_{0.25}$,2,2 = 3.0; $F_{0.1}$,2,2 = 9.0; $F_{0.05}$,2,2 = 19.0; $F_{0.025}$,2,2 = 39.0; $F_{0.01}$,2,2 = 99.0					

Table 10: Mathematical model for damping ratio estimation

ζ (Damping Ratio) first mode regression equation	*ζ (Damping Ratio) second mode regression equation*
$= 0.00397267+3.66\times10^{-6}L - 0.000247333T+ 0.00886333A$	$= 0.00285722+2.43333\times10^{-6}L - 0.000190833T+ 0.000603333A$

Table 11: Comparison of mathematical model with experimental for first mode

First mode		
Predication by Model (Regression equation)	*Confirmation as per experiment No: 7*	*% age error (%)*
0.0072893	0.007208	1.13

The above mentioned software package has given the regression equations to determine the damping ratio for the first and the second mode and is tabulated in Table 10.

Using these regression line equations, the optimum damping ratio is calculated for the two modes of vibration and tabulated in Table 11 and Table 12.

These values are compared with experimentally obtained results in the same tables and found to be in good agreement. It thus validated the optimization process for the damping ratio of the materials.

Table 12: Comparison of mathematical model with experimental for second mode

Second mode		
Predication by Model (Regression equation)	*Confirmation as per experiment No: 7*	*% age error (%)*
0.0049988	0.00505	1.012

9. Conclusion

The material damping is an important design consideration for vibrating structures especially in the airplane and automobile engineering. Natural frequency and damping ratio is determined by sweep sine test and half power bandwidth method respectively on Aluminium beam of required size. The beams are excited using an Electro-Dynamic Shaker excitation technique over the frequency range of 5-500 Hz. The experiments were conducted at very slow linear sweep rate of 0.05Hz/sec to precisely capture the resonance condition and damping ratio. The frequencies as obtained by the experimental results are in good agreement with the theoretical and ANSYS modeling. The natural frequency as determined by the different methods in this paper has corroborated the research done by the various researchers. This has thus established the process of experimentation. The damping ratio for this model is determined experimentally by half power bandwidth method from frequency response function graph. Analysis of variance (ANOVA) and S/N (signal to noise) ratios analysis is performed to determine the effect of three major geometrical parameters on material damping. The model has found to be significant at 90% confidence interval at both first and second mode of vibration. The results showed that the acceleration level significantly affects the damping ratio while the length parameter has the little effect on the damping ratio. The optimum result is validated using regression method. The regression models to predict the modal damping ratio are found to be in fair agreement with experimental results. The work in this paper will be very useful in predicating optimum damping ratio of the material.

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