Damping of Structures Under Extreme Multi-Axial Mechanical and Thermal Loads

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Introduction

When designing structures in the aerospace industry, predicting damping is essential. Some reasons as to why damping needs to be considered in the design process include: reducing dynamic response to avoid deflection or stress, reducing fatigue loads, ensuring aeroelastic stability, and reducing settling times following transients [1].

In aerospace, compressive and tensile loads are applied on structures. These mechanical loads affect damping of the structure. In order to predict damping in the design process of these different structures, an accurate model is needed. There are damping models already developed, and the main reason these models are used are because of mathematical convenience.

Hypersonic vehicles structures have thermal loads along with compressive and tensile loads being applied. Having thermal and mechanical loads being applied to structures makes it difficult to predict and model a structure's damping. The motivation behind this research is to help develop a damping model that is dependent on the mechanical and thermal loads being applied to the structure. This research focuses on predicting the relationship between in-plane loading and damping.

Background

To help predict the damping, some theoretical equations were used. In equations 1-3, the dimensional equation that were used can be seen:

$$\omega_{mn}{}^2 = \frac{D\pi^4}{\rho a^4}\left(m^2+\mu^2 n^2\right)^2\left(1+\mu^2\,\frac{\gamma_x k_x{}^2 m^2+\gamma_y k_y{}^2 n^2}{\left(m^2+\mu^2 n^2\right)^2}\right) \quad (1)$$

$$\omega_{mn_0}{}^2 = \frac{D\pi^4}{\rho a^4}\left(m^2+\mu^2 n^2\right)^2 \quad (2)$$

$$\alpha_{mn} = \frac{\omega_{mn}{}^2}{\omega_{mn_0}{}^2} \quad (3)$$

where ω_{mn} is the natural frequency, ω_{mn_0} is the nominal modal frequency and α_{mn} is the modal stiffness ratio. Using equations 1-3, nondimensional terms were created to predict the theoretical relation between in-plane loads and damping [2]:

$$\frac{\omega_{mn}}{\omega_0} = \frac{\sqrt{\alpha_{mn}}\,(m^2+\mu^2 n^2)}{(1+\mu^2)} \quad (4)$$

$$\frac{\eta_{mn}}{\eta_{EI}} = \frac{1}{\alpha_{mn}} \quad (5)$$

where $\frac{\omega_{mn}}{\omega_0}$ is the normalized modal frequency and $\frac{\eta_{mn}}{\eta_{EI}}$ is the normalized modal loss factor. Below in Figure 1, a plot of a square plate shows the theoretical relation between frequency and damping can be seen.

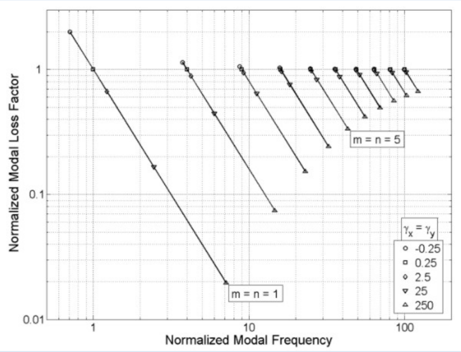

Figure 1: Plot of Normalized Modal Frequency and Normalized Modal Loss Factor of a Flat Plate [2]

Results

Previously, a test stand was designed to induce multi-axial loads to a flat plate. The test stand was designed so that the main function was applying compressive or tensile loads in both planar axes of a flat plate. Below in Figure 2, the experimental setup of the test stand can be seen.

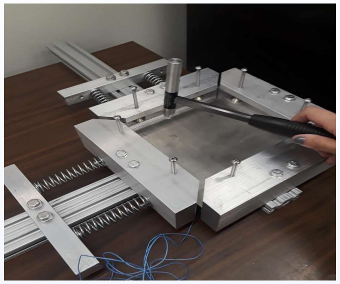

Figure 2: Experimental Setup of the Test Stand

Using an impact hammer, an accelerometer, a data acquisition unit, and LabVIEW, data from the test stand is collected. The accelerometer reads the impact from the hammer and generates FRF phase and amplitude plots. Using the data from the FRF amplitude plot, the damping ratio is calculated using equation 6 [5]:

$$\zeta_r = \frac{f_b - f_a}{2 f_r} \quad (6)$$

where ζ_r is the damping ratio at the natural frequency, f_r is the natural frequency, and f_a and f_b are the frequencies when the amplitude, or response, has half power of the peak. There were some issues with the boundary conditions on the test stand. This may have been because of additional stress being applied from the plate clamps.

The compression springs were compressed in increments to steadily increase the load being applied to the plate. The expected relation between the in-plane load being applied to the plate and the damping was that as the compressive load increased, the damping will increase. Different boundary conditions were tested to see which conditions would produce the best results. In Figures 3 and 4, the relation between the load and the damping and the load and frequency can be seen.

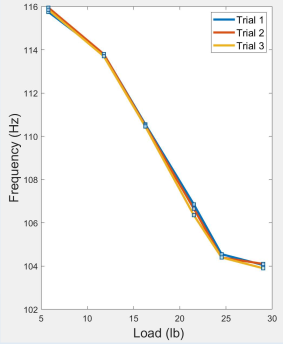

Figure 3: Plot of Load and Frequency

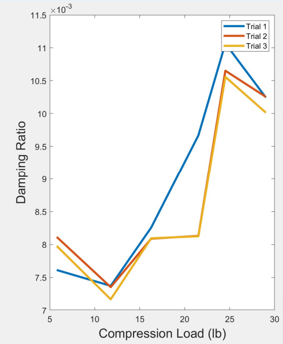

Figure 4: Plot of Load and Damping

Looking at Figure 3, it can be seen that as the load increased the frequency decreased. This is what was expected for compressive loads. In Figure 4, it can be seen that the damping increased as the load increased. This is what was expected.

Conclusion & Future Work

To help compare the experimental data with the theoretical data, the relationship between the calculated damping and frequency along with the applied compressive loads needs to be analyzed. Figure 5 shows a plot of the relationship between the calculated damping and the frequency.

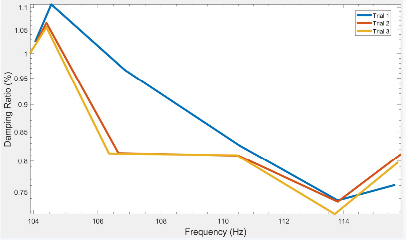

Figure 5: Plot of relationship between calculated damping and frequency

Looking at Figure 5, it can be seen that as the frequency increases, the damping decreases. This agrees with the theoretical data of the square plate shown in Figure 1. The theoretical data shows that as the compressive loads increase, the damping increases and the frequency decreases. When comparing the experimental and theoretical data, it needs to be considered that the experimental data is on a linear plot and the theoretical data is on a loglog plot. Also, the experimental data collected and presented is equivalent to a very small amount of the theoretical data.

The initial results confirm the expected in-plane load and damping relationship, however there is still more questions that need to be answered. Some recommendations for future work is to collect data using a stiffer compression spring. This way a bigger compressive load can be applied to the plate, and a bigger range of data could be collected. For this research in-plane were applied only in one direction, so another recommendation for future works is applying loads in both planar axes of a flat plate as well as tensile loads.

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Acknowledgements

I would like to thank Dr. Kauffman for their guidance through the research. Additionally, I would like to thank Christian Vazquez and Chris Kelley for their help during the research.