

FREQUENCY TUNING OF VIBRATION ABSORBER USING TOPOLOGY
OPTIMIZATION

by

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Abstract

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The University of Texas at Arlington, 2017

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A tuned mass absorber is a system for reducing the amplitude in one oscillator by coupling it to a second oscillator. If tuned correctly, the maximum amplitude of the first oscillator in response to a periodic driver will be lowered, and much of the vibration will be 'transferred' to the second oscillator. The tuned vibration absorber (TVA) has been utilized for vibration control purposes in many sectors of Civil/Automotive/Aerospace Engineering for many decades since its inception. Time and again we come across a situation in which a vibratory system is required to run near resonance. In the past, approaches have been made to design such auxiliary spring mass tuned absorbers for the safety of the structures. This research focuses on the development and optimization of continuously tuned mass absorbers as a substitute to the discretely tuned mass absorbers (spring- mass system). After conducting the study of structural behavior, the boundary condition and frequency to which the absorber is to be tuned are determined. The Modal analysis approach is used to determine mode shapes and frequencies. The absorber is designed and optimized using the topology optimization tool, which simultaneously designs, optimizes and tunes the absorber to the desired frequency. The tuned, optimized absorber, after post processing, is attached to the target structure. The number of the absorbers are increased to amplify

bandwidth and thereby upgrade the safety of structure for a wide range of frequency. The frequency response analysis is carried out using various combinations of structure and number of absorber cell.

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Chapter 1

Introduction

A tuned mass absorber is a system for reducing the amplitude in one oscillator by coupling it to a second oscillator. If tuned correctly, the maximum amplitude of the primary system in response to a periodic driver will be lowered, and much of the vibration will be transferred to the second oscillator. The tuned vibration absorber (TVA) has been utilized for vibration control purposes in many sectors of Civil, Automotive, and Aerospace engineering for many decades since its inception. Time and again we come across a situation in which a vibratory system is required to run near resonance. In the past, approaches have been made to design such auxiliary spring mass tuned absorbers for the safety of the structures. This research focuses on the design, development, and optimization of continuously tuned mass absorbers as a substitute to the discretely tuned mass absorbers (spring- mass system). After conducting the study of structural behavior, the boundary condition and the frequency to which the absorber is to be tuned are determined. The Modal analysis approach is used to determine the mode shapes and frequencies. The absorber is designed and optimized using the topology optimization tool, which simultaneously designs, optimizes and tunes the absorber to the desired frequency. The tuned, optimized absorber, after post processing, is attached to the target structure. The number of the absorbers is increased to amplify bandwidth and thereby upgrade the safety of structure for a broad range of frequency. The frequency response analysis is carried out using various combinations of structure and number of absorber cell.

1.1 Motivation

The motivation behind this thesis is three-fold; 1) effectively design optimal vibration absorber cell, 2) Substitute the discrete (spring –mass) tuned mass absorber with the continuous system as shown in Figure 1-1 and, 3) efficiently reduce the weight

associated with tuned mass absorbers using topology optimization.

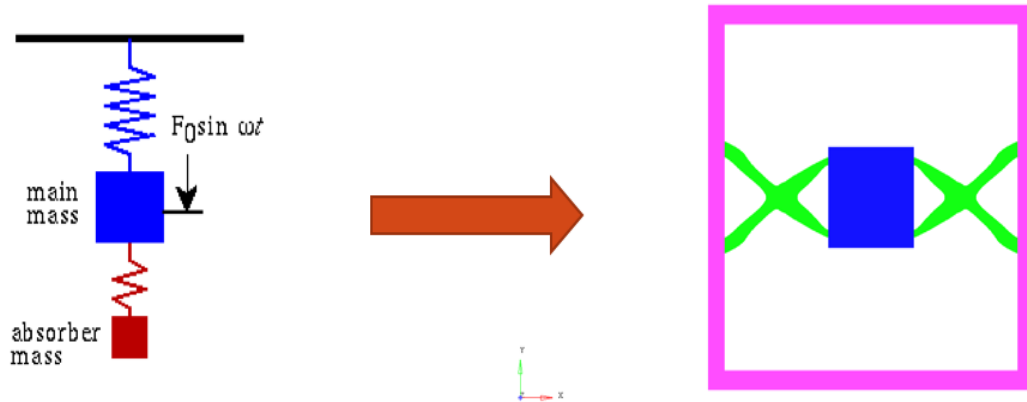


Figure 1-1 Conversion of classical spring –mass structure to continuous absorber

Vibration absorption is widely used to suppress the vibration in real systems. However, adding tuned mass absorbers leads to structural problems like increase in structural weight, increasing the complexity of structure and difficulty in allocating the vibration absorbers to the structure. In structures like an aircraft wing, vibration can be a very critical issue. And so, there has always been a demand to develop absorbers which are lighter in weight and continuous in structural aspect.

1.2 Research Objective

The focus of this work is on the design and optimization of continuous mass tuned absorbers as a substitute to the discretely tuned mass absorbers (spring-mass system). After conducting the study of structural behavior, the boundary conditions and frequency to which the absorber is to be tuned are determined. The Modal analysis approach is used to determine mode shapes and frequencies. The absorber is designed and optimized using

the topology optimization tool, which simultaneously designs, optimizes and tunes the absorber to the desired frequency. The tuned, optimized absorber after post processing is attached to the target structure. The number of the absorbers is increased to increase the bandwidth and thereby enhances the safety of structure over a wide range of frequency. The frequency response analysis is carried out using various combinations of structure and number of absorber cell.

1.3 Theory of Increasing Bandwidth

Since vibration often poses a safety concern and causes discomfort, the suspension of structural vibration is one of the major research subjects in mechanical, aerospace and civil engineering. As discussed earlier, one approach to solving the problem of high vibration is to use tuned mass absorbers. This use of tuned mass absorber increases the safety of the structure within the range of particular natural frequency. But, in real world scenario, the frequency acting upon the frequency can be unforeseeable and hence there is always a demand to increase the range of frequency over which the structure attains the level of safety. The growth in the number of absorbers can increase the bandwidth of frequency.

Consider a single-degree-of-freedom (SDOF) system shown in Figure 1-2. Let $M=1$ kg, $K=100$ N/m; the system has a natural frequency of 10 rad/sec.

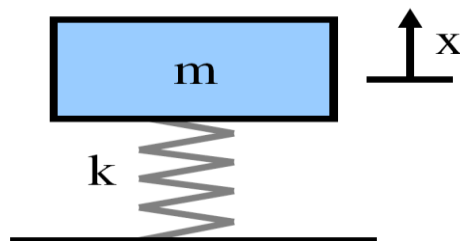


Figure 1-2 Spring –Mass system

When there is no damping involved in the system, the system experiences sudden peak when the forcing frequency coincides with the natural frequency of the system which is illustrated in Figure 1-3. When the forcing frequency equals the natural frequency of the primary mass, the response becomes infinite. This phenomenon is called resonance, and it can cause severe problems for vibrating systems.

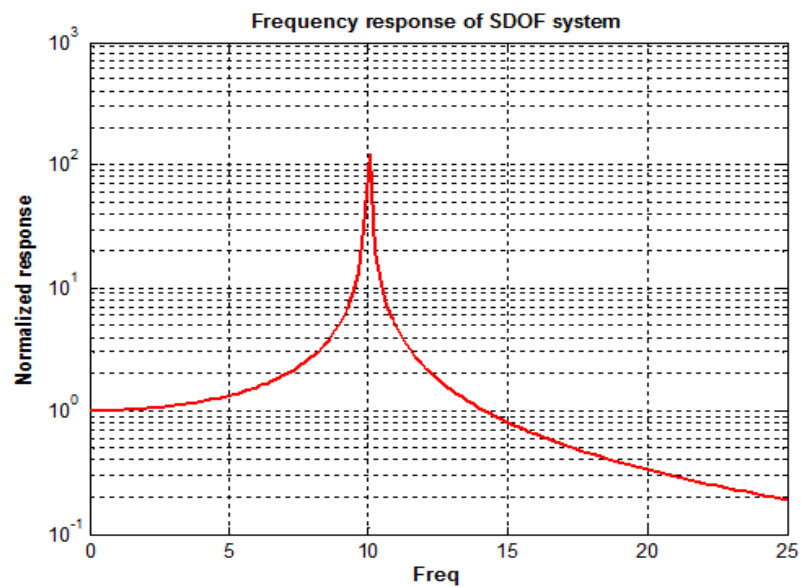


Figure 1-3 Frequency response analysis of spring-mass system

One simple way to reduce this response is to attach an undamped absorber with the frequency of 10 rad/sec. This attached absorber effectively produces an anti-resonant frequency at 10 rad/sec and eliminates the high resonant response. The Figure 1-4 illustrates the tuned mass absorber connected to the primary system acted upon by force represented as $F_0 \sin(\omega t)$.

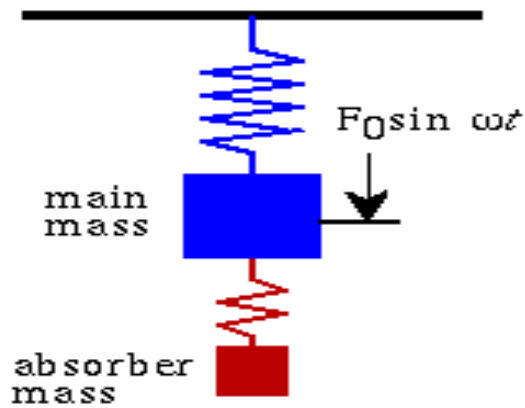


Figure 1-4 Spring –Mass system with absorber

When an absorbing mass-spring system is attached to the main mass, and the resonance of the absorber is tuned to match that of the original mass, the motion of the main mass reduces to zero at its resonance frequency. Thus, the energy of the main system is apparently absorbed by the tuned dynamic absorber.

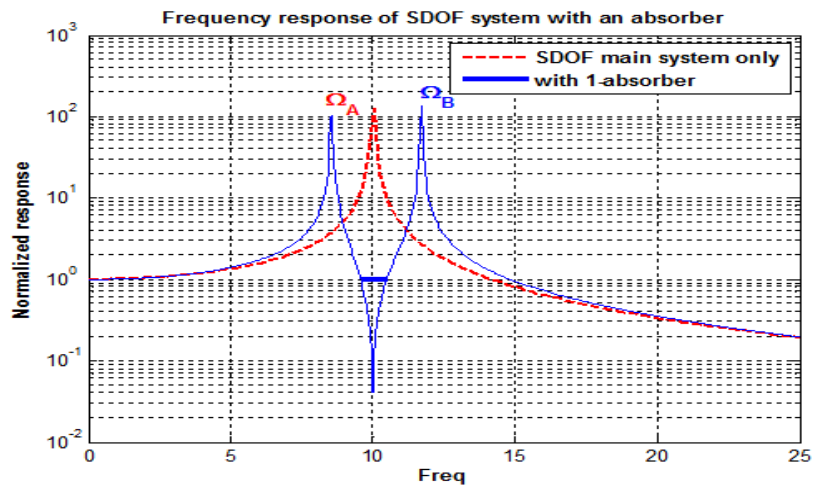


Figure 1-5 Frequency response analysis with one absorber

It is interesting to note that the motion of the absorber is finite at this resonance frequency, even though there is no damping in either of the oscillator. This is because the system has changed from a 1-DOF system to a 2-DOF system and now has two resonance frequencies as shown in Figure 1-5, neither of which equals the original resonance frequency of the main mass (and also the absorber). The 2-DOF system has two natural frequencies, corresponding to the two natural modes of vibration for the system. In the lower frequency mode, both masses move in the same direction, in-phase with each other. In the higher frequency mode, the two masses move in opposite direction, 180° out of phase with each other. However, if the forcing frequency deviates from 10 rad/sec, the response would be high. This is due to the narrow bandwidth of the absorber. (Shown as a short blue horizontal line in Figure 1-5). Thus it is desirable to increase the bandwidth for vibration reduction.

From the theory of local modification, we know that when a spring-mass system with a natural frequency of W_{abs} is added to a structure, all frequency (of the attached structure) below W_{Abs} will decrease while all frequency above W_{Abs} will increase. Thus, if we connect another absorber with frequency 10 rad/sec, this would make W_A smaller and W_B larger, consequently, increases the bandwidth of the system. Figure 1-6 shows the effect of the addition of absorber to the structure.

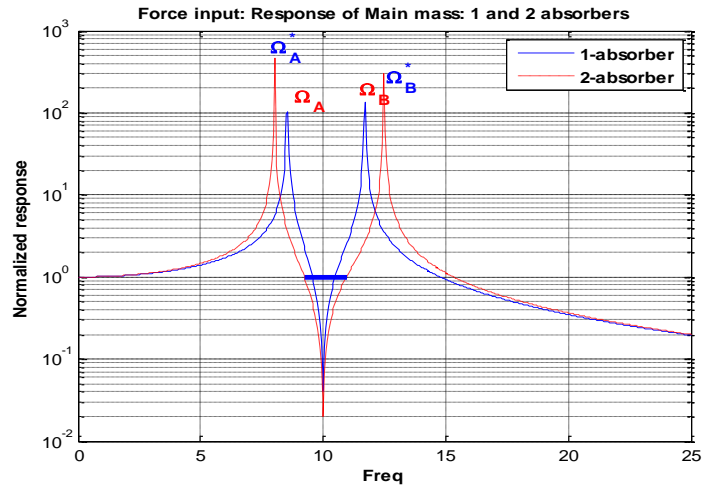


Figure 1-6 Frequency response analysis of structure with two absorbers

Thus, we can add more absorbers with the same frequency to increase the bandwidth further. The Figure 1-7 shows the frequency responses of the same primary SDOF system with 1, 5, 10 and 20 absorbers.

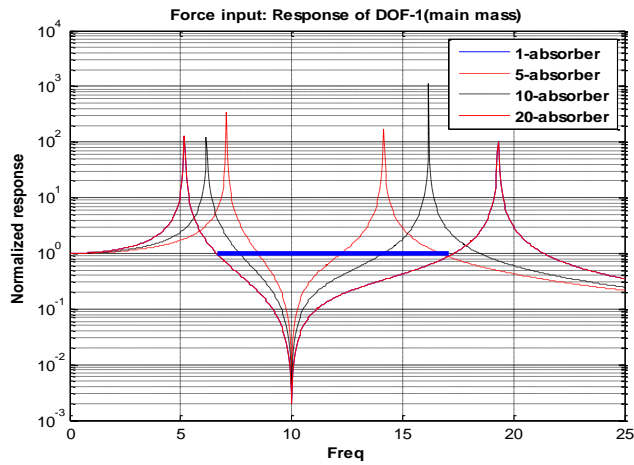


Figure 1-7 Frequency response analysis of structure with 5, 10, and 20 absorbers

Thus, we can add more absorbers with the same frequency to increase the bandwidth further. The Figure 1-7 shows the frequency responses of the same main

system with 5, 10, and 20 absorbers. It is observed that as the number of absorbers increases the separation between two peaks increases which suggests that bandwidth increases. The bandwidth is widest for a system with 20 absorbers and is narrowest for a system with one absorber. We can conclude that as the number of absorber increases, the bandwidth increases. The increase in bandwidth leads to increase in safety of the structure.

Chapter 2

Background

Vibration in modern structures such as Aircraft Wing, Bridges, Buildings, and similar structures is becoming major issues nowadays. The tuned vibration absorber (TVA) has been utilized for vibration control purposes in many sectors of civil, automotive, and aerospace engineering for many decades since its inception by (Ormondroyd & Den Hartog, 1928) [4] [5]. Time and again we come across a situation in which a vibratory system is required to run near resonance. All the approaches in the past have used a discreet system (spring mass) system to design tuned mass absorbers. In this paper, we will focus to substitute such spring mass systems with a continuous system using a unique design optimization tool known as Topology Optimization.

2.1 Resonance

Bridges, aircraft wings, machines and all other physical structures have natural frequencies. When a structure is set to oscillations and allowed to vibrate freely, the structure continues to vibrate at a specific frequency which is called as natural frequency. All the structures have at least one natural frequency. All structures have more than one natural frequency. The resonance is a tendency of a system to produce greater amplitude when it is forced by oscillatory load whose frequency is equal to the natural frequency of the system. Resonance occurs when the frequency of forcing function coincides with the structural natural frequency. During resonance, the displacement of the system increases until the structure undergoes failure mechanisms like buckling, yielding or fatigue. The system under resonance can produce swaying motion of very high movement and eventually may lead to catastrophic failure of the inappropriate structure. The response of the system at such instance can be jeopardous, and we must take necessary measures to

control the reaction of the system. The excitation frequency can be a result of many reasons such a rotating unbalanced rotor at a constant speed. In mechanics and construction, the phenomenon known as resonance disaster describes the tendency of the building or mechanical structure to failure due to induction of vibration at system's natural frequency. Intermittent excitation ideally transfers to the system, the energy of the vibration and stores it there. Due to this rehashed stockpiling and additional energy input, the system swings always vivaciously until its load limit is surpassed.

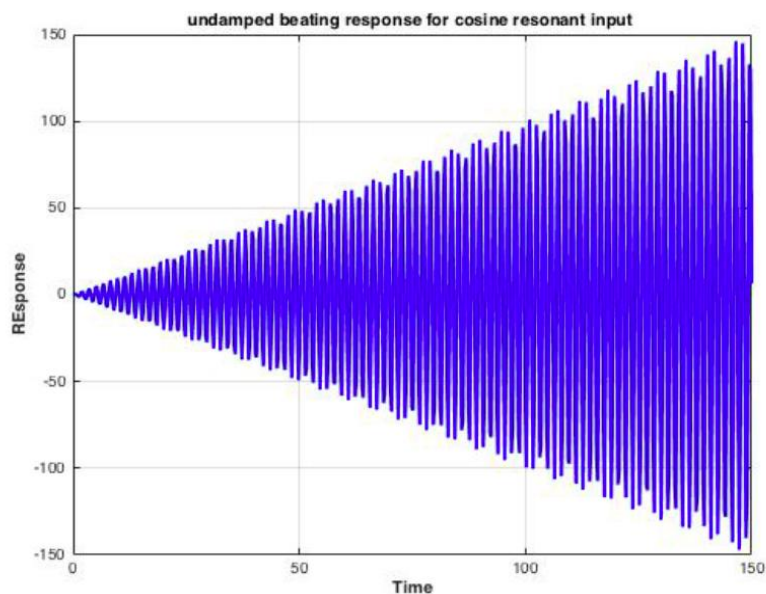


Figure 2-1 Resonance in spring-mass system

The Figure 2-1 shows the resonance phenomenon in a spring-mass system with cosine forcing function without any damping. As discussed, the system stores the energy gained from the forcing function and thus increases the amplitude of vibration with time. Due to repeated storage of energy, the system oscillates even more vigorously until it reaches the load limit. This increased amplitude can cause troublesome swaying in the

structure and eventually may lead to failure. In the past, there have been many accidents due to the effect of resonance on the system. In the field of mechanics and construction, the resonance disaster is described as a phenomenon of destruction of a building or mechanism due to induced vibrations at the natural frequency of the system. The most classic example of failure due to resonance is the destruction of Tacoma bridge. The failure of design was due to the interaction between the bridge vibration and wind passing through it, a phenomenon known as aeroelastic flutter. Its resonance can destroy a deficient bridge that is why soldiers are trained not to march in lockstep across a bridge; Mechanical systems store potential energy in different forms. For example, a spring-mass system stores energy as tension in the spring, which is ultimately stored as the energy of bonds between atoms.



Figure 2-2 Structures affected due to resonance. Img source:Wikipedia

Figure 2-2 shows various examples of structures affected due to mechanical resonance. Mechanical resonance has been considered responsible for many catastrophic failures in the past. For machines, for example, pumps, turbines, and electric engines reverberation can intensify the little vibratory powers from machine operation, and serious vibration levels can come about. On account of an aircraft wing, if the periodic vibration of the gust concurs with the natural frequency of the wing, it will enter resonance, and the linear deflection of wing will increase.

2.2 Tuned Mass Absorbers

Vibration absorption is a method of attaching a tuned spring-mass absorber to create anti-resonance at a resonance of the original system. The dynamic vibration absorber is designed in such a way that the natural frequencies of the resulting system stay away from excitation frequency. The vibration of the primary system can be reduced by attaching a smaller spring-mass system as an auxiliary system to the main system. When the natural frequency of spring mass system is tuned to the natural frequency or resonating frequency of the primary vibrating system the response almost reduces to zero. This reduction in displacement allows the system to operate without any failure. Such kind of auxiliary spring mass system is called vibration absorber, and the method is called as vibration absorption.

A tuned mass Absorber is a device consisting of mass and spring, which is attached to a structure to reduce the dynamic response of the structure. The frequency of the damper is tuned to a particular structural frequency so that when that frequency is excited, the damper will resonate out of phase with the fundamental motion. Energy is dissipated by the damper inertia force acting on the structure. The concept was first invented and applied by Frahm in 1909 (Frahm, 1909) [8] to alleviate the rolling motion of

ships as well as ship hull vibrations. A theory for the tuned mass absorbers was presented later in the paper by Ormondroyd and Den Hartog (1928) [6], followed by a detailed discussion of optimal tuning and damping parameters in Den Hartog's book on mechanical vibrations (1940) [6]. The initial theory was applicable for an undamped SDOF system subjected to a sinusoidal force excitation. Numerous researchers have investigated the extension of the theory to damped SDOF systems. Significant contributions were made by Randall et al. (1981), Warburton (1981, 1982), Warburton and Ayorinde (1980), and Tsai and Lin (1993) [7].

A machine or system may experience excessive vibration if it is acted upon by a force whose excitation frequency nearly coincides with a natural frequency of the machine. In such cases, the vibration of the machine can be reduced by using a vibration neutralizer or dynamic vibration absorber, which is simply another spring mass system.

The response of the SDOF system subjected to the harmonic force having a frequency close to the natural frequency of the system will be quite significant when the frequencies are equal. It will be infinite, theoretically, if the damping is neglected. The variation of the amplitude frequency is as shown in Figure 2-3 (b). The response has a high peak at the natural frequency. This high peak determines greater displacement at the natural frequencies.[14]

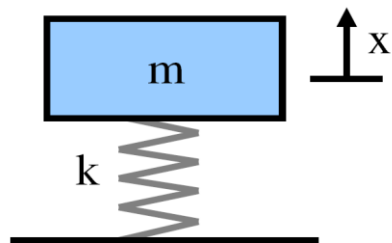


Figure 2-3 (a) Spring mass structure

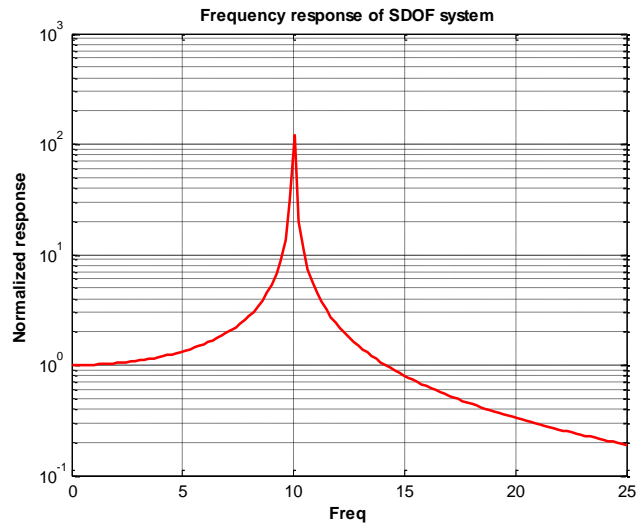


Figure 2-3 (b) Frequency response analysis of spring mass structure

One simple way to reduce this response is to attach an undamped absorber with the frequency equal to the natural frequency of the system. This effectively produces an anti-resonant frequency at the resonating frequency and eliminates the high resonant response. The frequency of the system with and without the absorber is given below. However, the addition of the absorber makes the system 2-dof with natural frequencies at ω_A and ω_B . Thus, if the forcing frequency deviates from the natural frequency of the system, the response would be high. This rise in response is due to the narrow bandwidth of the absorber. (Shown as a short horizontal line). Thus, it is desirable to increase the bandwidth for vibration reduction.

From the theory of local modification, we know that when a spring-mass system with the natural frequency of ω_{Abs} is added to a structure, all frequency (of the attached structure) below ω_{Abs} will decrease while all frequency above ω_{Abs} will increase, which can be illustrated in Figure 2-2.

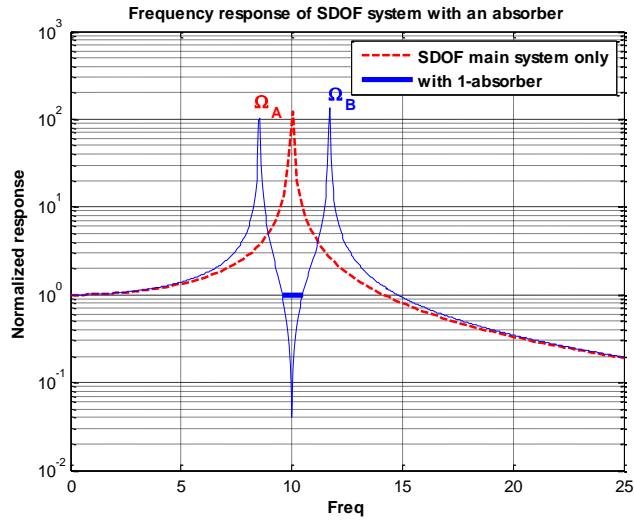


Figure 2-3 (c) Frequency response analysis of spring mass structure with an absorber.

For the system shown in Figure 2-3 (c), consider x_1 to be the displacement of the main system and x_2 the displacement of the secondary system. Let us assume $x_2 > x_1$, Spring mass system k_1 - m_1 as main system and spring mass system k_2 - m_2 as absorber system. Equations of motion for Two DoF system are as follows,

$$m_1 \ddot{x}_1 + F_0 \sin \omega t = -k_1 x_1 + k_2(x_2 - x_1) \text{----- (1)}$$

$$m_2 \ddot{x}_2 = -k_2(x_2 - x_1) \text{----- (2)}$$

Assuming solution under steady state condition,

$$x_1 = X_1 \sin \omega t$$

$$x_2 = X_2 \sin \omega t$$

Finally, we get equation in dimensionless form as

$$\frac{x_1}{x_{st}} = \frac{1 - \frac{\omega^2}{\omega_2^2}}{\frac{\omega^4}{\omega_1^2 \omega_2^2} - \left[(1 + \mu) * \frac{\omega^2}{\omega_1^2} + \frac{\omega^2}{\omega_2^2} \right] + 1} \text{----- (3)}$$

$$\frac{x_2}{x_{st}} = \frac{1}{\frac{\omega^4}{\omega_1^2 \omega_2^2} - \left[(1+\mu) \frac{\omega^2}{\omega_1^2} + \frac{\omega^2}{\omega_2^2} \right] + 1} \text{-----(4)}$$

From this equation, it is clearly seen that $x_1 = 0$, when $\omega = \omega_2$ i.e. When excitation frequency is equal to the natural frequency of the absorber system, the primary system's amplitude becomes zero even though it is excited by harmonic force. This is the principle of an undamped dynamic vibration absorber [5]. Also, it can be seen that when $\omega = \omega_2$, we get,

$$F_0 = -k_2 x_2 \text{-----(5)}$$

The above equation shows that the spring force $k_2 x_2$ on main mass due to amplitude x_2 of absorber mass is equal and opposite to the force on main mass resulting in no motion of the primary system.

For the effectiveness of absorber at operating frequency corresponding to natural frequency of main system alone, we have,

$$\omega_1 = \omega_2 = \frac{k_2}{m_2} = \frac{k_1}{m_1}$$

When this condition is fulfilled, the absorber is called as Tuned Absorber. To have a tuned absorber we can have many combinations of k_2 and m_2 as long as their ratio is equal to k_1/m_1 . To satisfy the above condition, we can have small spring k_2 and small mass m_2 or k_2 large and large mass m_2 . In all these cases, main system response will be zero at $\omega = \omega_2$. However, Eq. Shows that for same exciting force the amplitude of absorber mass is inversely proportional to its spring rate. To have a small magnitude of absorber mass m_2 , we must have a large k_2 and therefore large which may not be desirable from practical considerations. So, a compromise is usually made between amplitude and mass ratio μ . The mass ratio is kept between 0.05 to 0.2. A proper design of absorber spring is also necessary which depends upon its amplitude and available space. The denominators of above equation and are identical. At a value of ω when these denominators are zero, the

two masses have infinite amplitudes of vibration. The expression for the denominators is quadratic in ω_2 and therefore there are two values of ω , for which these terms vanish. These two frequencies are resonant frequencies or natural frequencies of the system.

When excitation frequency equals to any of the natural frequency of the system, all the points in the system have infinite amplitudes of vibration or the system is in resonance. To find the two resonant frequencies of the system, when $\omega_2 = \omega_1$, the denominator of either of the equation is equated to zero.

$$\frac{\omega^4}{\omega_1^2 * \omega_2^2} - \left[(1 + \mu) \frac{\omega^2}{\omega_1^2} + \frac{\omega^2}{\omega_2^2} \right] + 1 = 0 \text{ -----(6)}$$

Solving we get,

$$\left(\frac{\omega}{\omega_2} \right) = \left[\left(1 + \frac{\mu}{2} \right) \pm \sqrt{\mu + \frac{\mu^2}{4}} \right]$$

1.3 Design Optimization

Optimization is a process of achieving the best performance of a given operation for given set of restriction. Design optimization, in engineering terms, is described as design improvement process of a product or a system to achieve maximum performance for given set of variety of physical constraints.

Following are some relevant terminology used in design optimization

Design Variable

The potential for change is typically expressed regarding a range of permissible changes of a group of parameters. These design variables can be subjected to upper and lower values.

Examples: length, thickness, density

Design Constraint

Physical or prescribed limitations based on the design variables. Constraints can typically be imposed in two ways: equality and inequality constraints. The equality constraint is when the design must meet a particular criterion, such as equilibrium equations in a structural problem. The inequality constraint is when the design must remain in a specific criteria region, such as a maximum allowable stress in a system.

Examples: maximum deflection, maximum stress, constant heat flux

Objective Function

Performance measure dependent on the design variables and subject to the design constraints that the optimization attempts to minimize or maximize to achieve an optimal solution.

Examples: minimize volume, minimize cost, and maximize stiffness [9]

1.4 Topology Optimization

Topology Optimization (TO) be a method of material distribution within the given design space with a motive of maximizing the system or structure performance, for a given set of loads, boundary conditions, and constraints. In the year 1904, an Australian inventor named Michell [3] published first paper on Topology Optimization, who derived optimality criteria for the least weight layout of trusses.

Structural design problems require the fulfillment of specific objectives while satisfying a set of performance constraints. In the automotive and aerospace worlds, this traditionally requires numerous iterations using finite element analysis results that drive design changes, and the final design is often arrived at via a trial-and-error approach. This iteration process can prove to be a very time-consuming and thus costly endeavor. While

this technique has obviously been effective in meeting design requirements, the final solution does not necessarily represent the best solution—only one that has successfully achieved the objective. Furthermore, the quality of the final design relies heavily on the quality and potential of the initial design attempts. This is because, as the design progresses, the freedom to make significant changes diminishes over time. Therefore, in the interest of time and quality, it is crucial to have a good initial design solution early in the process.

Another drawback of the traditional design method is the fact that engineers tend to think intuitively. Sometimes the optimum solution can be quite counterintuitive, and thus an excellent solution can go justifiably overlooked because it does not seem plausible or reasonable. On the contrary, a design can include an overly complicated network of reinforcing ribs, for example, that were deemed necessary by the design engineer, when the ideal optimized solution is much simpler

Topology optimization is a relatively new numerical method used to determine the optimum shape and distribution of material within a given design space for a given set of design constraints based on responses obtained from a finite element analysis presently most modern numerical FE-based topology optimization method is the SIMP method, which was developed in the late eighties. It is sometimes called “material interpolation,” “artificial material,” “power law,” or “density” method, but “SIMP” is now used widely. The term “SIMP” stands for Solid Isotropic Microstructure (or Material) with Penalization for intermediate densities. The basic idea of this approach was proposed by Bendsoe (1989) [9], while the term “SIMP” was coined later by the author and first introduced in a paper by Rozvany et al. (1992) [2]. A topology optimization problem based on the power law approach, where the objective is to minimize compliance can be written as

$$\text{Min}_x: c(x) = U^T K U = (x + a)^n = \sum_{e=1}^N (x_e) P u_e^T k_0 u_e$$

$$\begin{aligned} \text{Subject to } &: \frac{V(x)}{V} = f \\ &: KU=F \\ &: 0 < X_{\min} \leq x \leq 1 \end{aligned}$$

Where U and F are the global displacement and force vectors, respectively. K is the global stiffness matrix, u_e and k_e are the element displacement vector and stiffness matrix, respectively. X is the vector of design variables, X_{\min} is a vector of minimum relative densities (non-zero to avoid singularity). N is the number of elements used to discretize the model domain, p is the penalization power (typically $p = 3$), $V(x)$ and V_0 is the material volume, and design area volume, respectively and f is the prescribed volume fraction. The optimization problem (1) could be solved using several different approaches such as Optimality Criteria (OC) methods, Sequential Linear Programming (SLP) methods or the Method of Moving Asymptotes (MMA by Svanberg 1987) [13] and others. For simplicity, we will here use a standard OC-method.

In this paper, we will discuss the method of topology optimization to design and optimize vibration absorbers. The optimization process will consist of optimization of spring. The geometry of mass is a non-design region. This technique can be used in the early stages of design to ensure that the final design of the structure not only meets the requirements but also represents the best solution based on the design constraints mathematically. Today the method is widely accepted for bracket-type structures and has already proven to be beneficial to a large number of aerospace and automotive corporations. [11] [12]

Chapter 3

Methodology

In this section, the model design of a unit absorber cell is done. The model of a unit absorber cell depends upon the structural behavior of the absorber and the target structure. The Design of unit cell model varies from structure to structure depending upon the modal frequencies and the method of connection between the structure and the unit absorber. Since the design is in an initial stage of product development, the preliminary 2-D plate design is used for the base design. In this case, the unit cell is thrust into the structure.

3.1 Unit Absorber Cell pre-optimized model

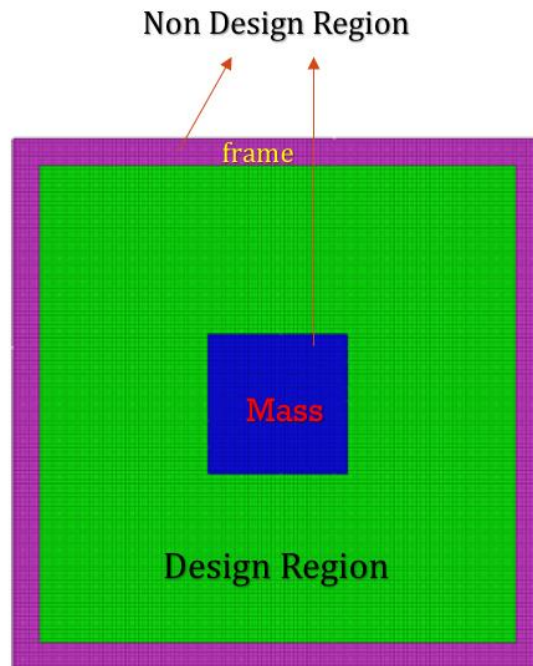


Figure 3-1 Pre-Optimized unit absorber cell

3.1.1 Components

Design Space

Converting Topology Optimization problem from single degree of freedom (SDOF) to multiple degree of freedom (MDOF) problem is done by discretizing the design domain into finite elements. Before the topology optimization study can begin, it is first necessary to determine the maximum amount of volume that the geometry can safely occupy. This volume is known as the design space, and it represents the volume that will be meshed into finite elements and iterated upon while the optimization algorithm is working. If the finalized component needs clearance or a pass-through for wires, for example, the design space must reflect this so that the software does not try to use that space for load-bearing elements. Due to the attainable topological complexity of the design being dependent on the number of elements, a significant amount is preferred. In the case of unit absorber cell design, design space comprises of the volume represented in green in Figure 3-1. This design space acts as spring after optimization.

Non-Design Space

Non-design space consists of two components namely frame and Mass, represented as pink and blue in Figure 3-1 respectively. The design of non-design components, as the name suggests remains unchanged even after the optimization.

3.1.2 Material

The last input parameter required before implementation of the topology optimization study is the material identification. In the case of the unit absorber cell, the material was specified as 7075-T6 aluminum and steel.; as a result, the material was defined as linear isotropic (MAT1), and the values for Young's modulus and Poisson's ratio were entered. These values used for the study were 69GPa and 0.35, respectively for

Aluminum and 210 GPa and 0.3 for Steel. The design region and frame were assigned to the material properties of Aluminum. A material property of Steel was assigned to mass. The detailed material property of design and the non-design region is determined in Figure 3-2.

Material	Young's modulus (Gpa)	Density (Kg/m ³)	Poisson's ratio
Aluminum	69	2700	0.35
Steel	210	7850	0.3

Table 3-2 Material Property

3.1.3 Dimensions

The pre-optimized geometry ready for optimization consist of the frame, whose maximum dimensions are 100mm x 100mm. The width of the frame is 5mm, while the size of the mass is 15mm x 15 mm. The thickness of the cell remains constant throughout which is 1mm. The dimensions of the cell are represented in Figure 3-3.

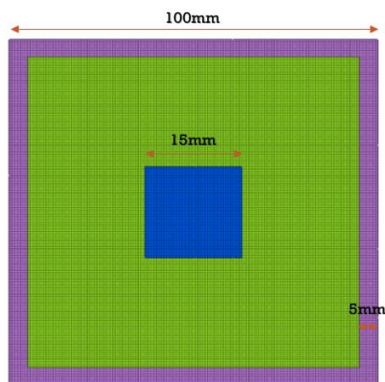


Figure 3-3 Cell Dimensions

3.2 Boundary Conditions and Meshing

Once the geometry was cleaned, the design space volume was filled with 2-D elements using the auto-mesh features of HyperMesh®. This was done with a 2D-quad element with a nominal minimum size of 0.125 mm. The resulting mesh that was used as the design space for the topology optimization study can be seen in Figure 3-4 and Figure 3-5. The property p-shell was assigned to the pre-optimized model.

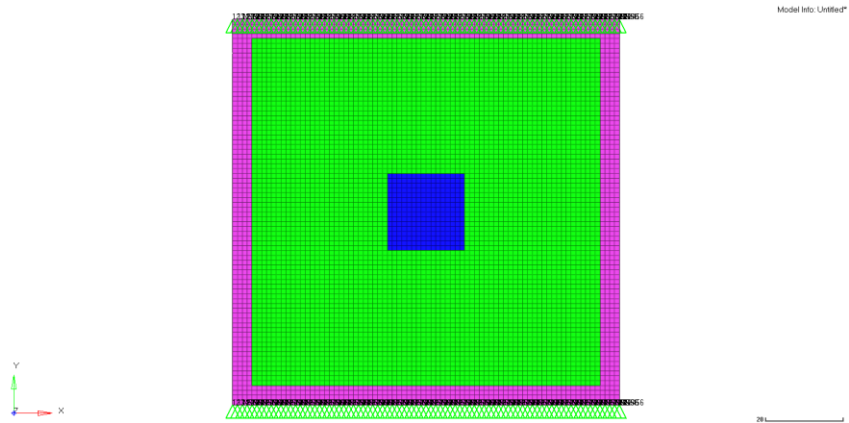


Figure 3-4 Boundary Conditions on top and bottom edge

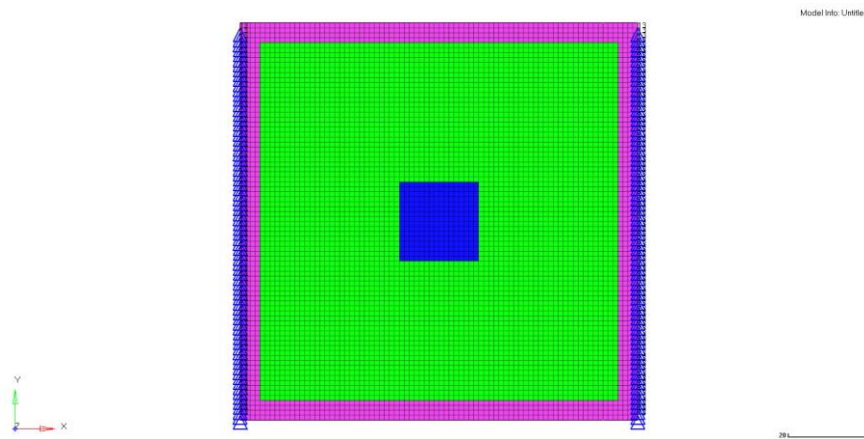


Figure 3-5 Boundary Conditions on Right and Left Edges

The crucial consideration while design the cell is that the motion of the cell is restricted to the X-Y plane. So, all the nodes are constrained in such a manner that the movement of the cell remains in the plane. To achieve that each node is limited to a translational degree of freedom along the Z axis. This constraint will continue to be same for all cases. Note that the boundary conditions applied to the cell are case specific and they will change from case to case. It is critical to predict perfect boundary conditions to obtain the exact solution. The cell designed in this case is presumed to be fitted on the beam and presupposed to be connected by top and bottom edges. Considering this, the top and bottom edges are constrained for all degrees of freedom as illustrated in Figure 3-4. The other two edges,i.e., The right and left edges are limited to translation along X-axis because the unit cell is supposed to work independently and should not transmit vibration to adjacent cells. The constraints are applied to the left and right edge as shown in Figure 3-5.

Defining DOFs 1, 2, and three as translational degrees of freedom in X, Y, and Z; and DOFs 4, 5, and six as rotational degrees of freedom about the X, Y, and Z axes; the constraints can be summarized as seen in Table 3-6

Node on Cell	Nodes on top edge	Nodes on bottom edge	Nodes on right edge	Nodes on left edge	All other Edges
DOFs Constrained	123456	123456	13	13	3

Table 3-6 Summary of Constraints

3.3 Setting up Topology Optimization Problem

The greatest flexibility to impact the design of a product is in the conceptualization phase hence methods aimed at driving design concepts should be employed at this stage. In doing so, the potential for better, more efficient designs, lighter and innovative designs is maximized.

The Topology optimization techniques yield a new model and optimal distribution. Topology optimization allows designers to start with a design that already has the advantage of optimal material distribution and is ready for fine design tuning with shape or size optimization. Optistruct's design-synthesis technology uses the topology optimization approach to generate innovative concept design proposals. Optistruct creates an optimal design proposal for the most efficient material layout of conception based on user-defined design space, design targets, and manufacturing process parameters.

In this research, topology optimization is performed on a model to create one topology for the structure, removing any unnecessary material. The resulting material structure is lighter and satisfied all design constraints. The optimization problem can be stated as shown in the table below.

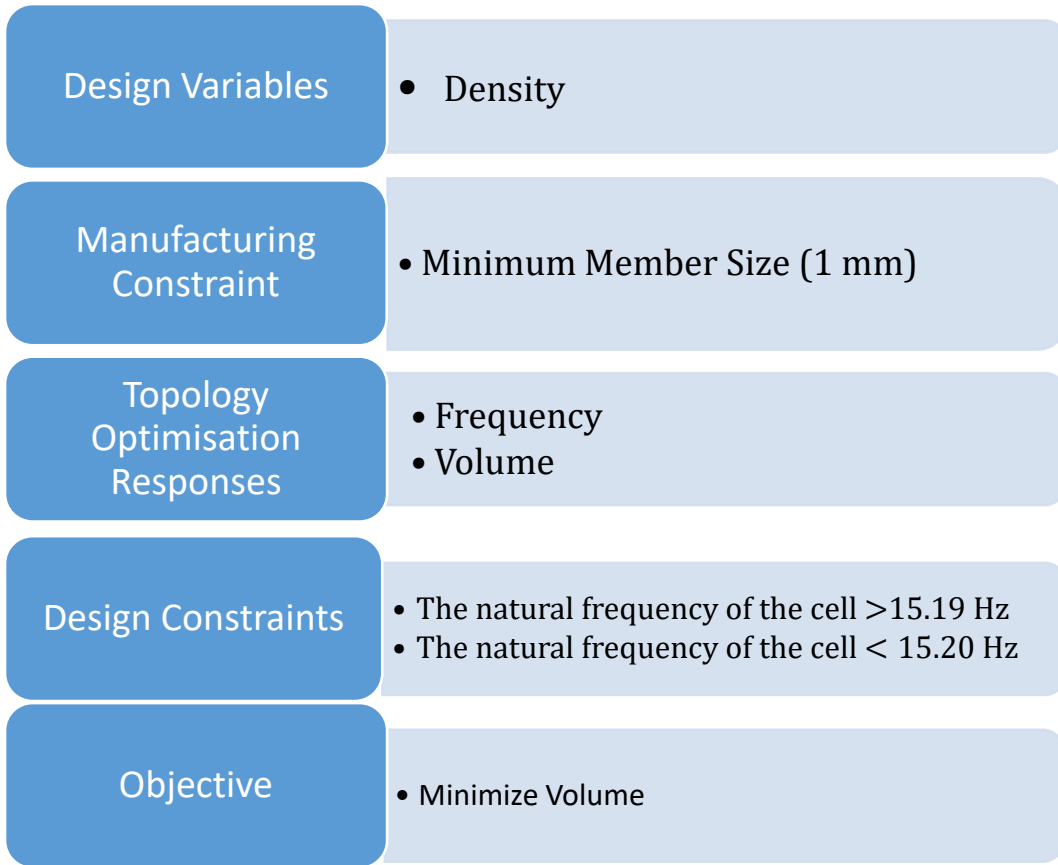


Figure 3-7 Optimization Problem

3.3.1 Setting up Design Variable

The setup of design Variables constraint as in Altair OptiStruct 14.0 can be shown in below figure below.

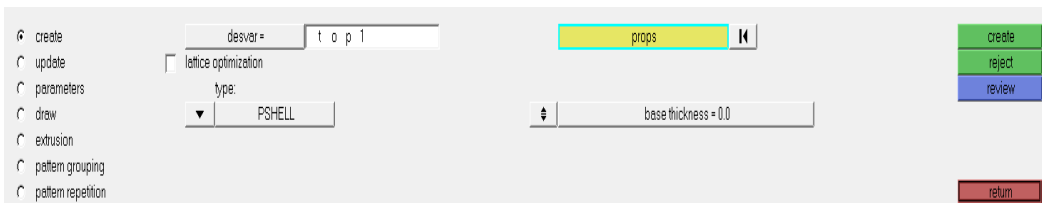


Figure 3-8 Setting up Design Variable

3.3.2 Setting up Design Responses

The objective function of any optimization problem is to minimize or maximize a particular response while meeting a prescribed set of constraints. For this, it is necessary to program the software to solve for the desired responses, and then choose limits to these responses. In the case of absorber cell optimization project, the objective function was to minimize the mass while maintaining the integrity of the original design. Consequently, the response to minimize was chosen as the volume. This value represents the volume of the design space after elements have been iteratively eliminated or turned “off” in an effort to meet the design constraints, and with a constant material density, the volume fraction directly correlates to the mass. Obviously, this mass minimization cannot continue without bound. Thus it is necessary to define responses that have upper and/or lower limits as constraints, which we will discuss in following sub-chapters. The second response was natural frequency of the cell. The upper and the lower limit for the first mode of natural frequency will be determined in the following sub-chapters. The setup of design responses constraint as in Altair OptiStruct 14.0 can be shown in below figure below [19].



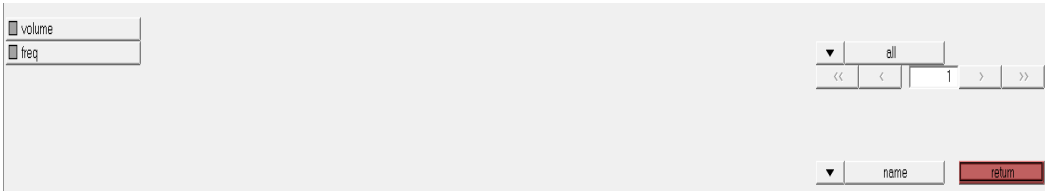


Figure 3-9 Setting up Design Responses

3.3.3 Setting up Manufacturing Constraints

Although the optimization problem is now fully defined, the resulting topology design proposal will likely be very difficult to manufacture because the algorithm will tend to make hollow structures with a lot of holes. To ensure that the optimized geometry can be realistically manufactured, the software includes algorithms for manufacturing constraints such as minimum size control and prescribed draw directions. With minimum size control enabled, the optimization software will not create geometry that is smaller than the desired size. This feature reduces the number of small ribs and “blobs” that can complicate the interpretation and creation of the optimized geometry. For this particular project, the minimum member size was set to 1 mm, as this was the thinnest rib that the machinist felt comfortable fabricating.

Because the Absorber cell was to be machined from solid aluminum, it was also deemed beneficial to enable a draw direction, as if the part was going to be cast in a mold. This eliminates the formation of a hollow structure, giving the part more of a two-dimensional quality, thereby lending to its manufacturability. The absorber cell design space was given a single draw direction outward from the x-y plane in the z-direction. For comparison purposes, however, optimization results were obtained with, and without the draw direction enabled, and the differences will be discussed in the following chapter. The setup of manufacturing constraint as in Altair OptiStruct 14.0 can be shown in below figure below[19].

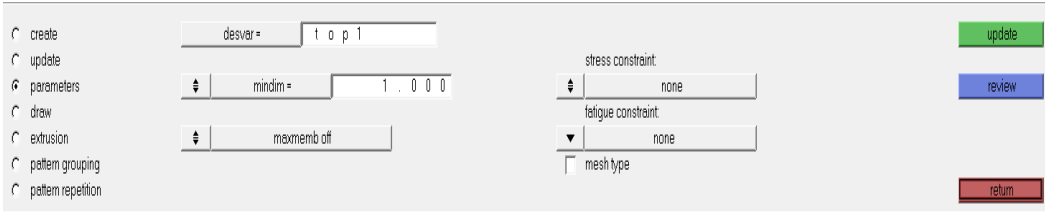


Figure 3-10 Setting up Manufacturing Constraints

3.3.4 Setting up Optimization Constraint on response

The setup of Optimization constraint as in Altair OptiStruct 14.0 can be shown in below figure below.



Figure 3-11 Setting up Optimization Constraint on response

3.3.5 Setting up Optimization Objective

The setup of Optimization Objective as in Altair OptiStruct 14.0 can be shown in below figure below.



Figure 3-12 Setting up Optimization Objective

Chapter 4

Results

4.1 Topology Optimization Result.

The optimization analysis was conducted using Altair OptiStruct® 14.0, and the jobs were run on a 3.1 GHz Intel i7 processor with 16 GB RAM. The average run-time was about 12 mins. The resulting geometry proposal can be seen in Figure 4-1 with the formerly stated boundary conditions, minimum member size control, and symmetry constraint. Notice that there are clearly defined groupings of elements that form ribs in an “X” pattern within the proposed design region. The geometry is symmetric due to symmetric boundary condition applied to the cell before optimization. The “X” pattern defines the stiffness of the cell since the design region acts as the spring and the non-design central region serve as a mass, shown as a square blue box in the Figure 4-1. The design of mass and frame remains unchanged as they were non-design region. The frame ensures the stable transformation of vibration to the target structure.

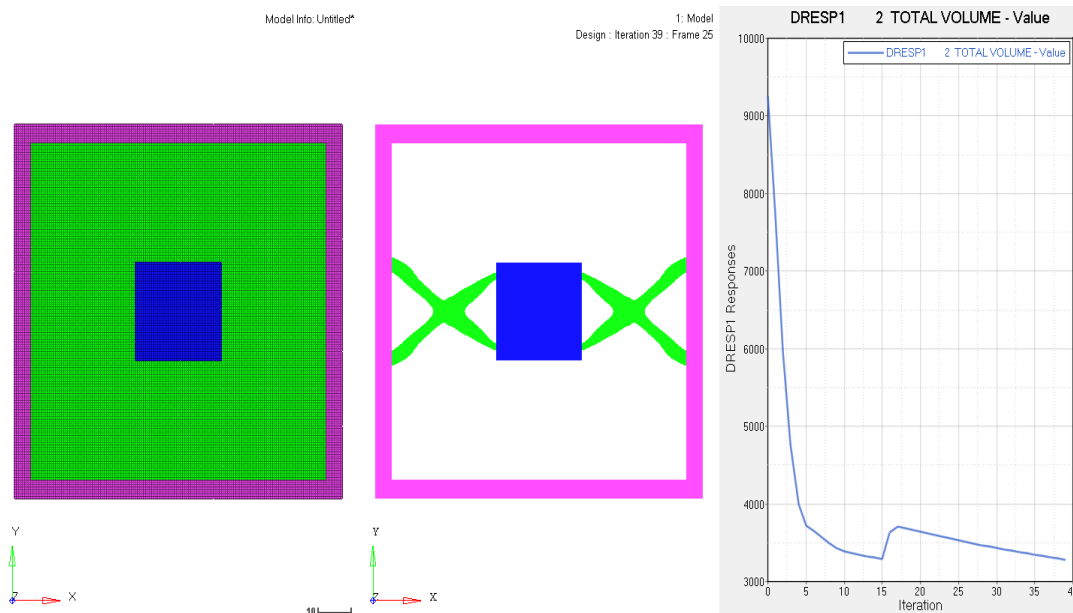


Figure 4-1 Topology Optimization Results

The crucial results from the topology optimization are mass reduction and tuning the natural frequency to that of the natural frequency of the structure. The solver ran 39 iterations before reaching to the final design. At 39th iteration, the analysis was completed, and no constraints were violated. The response vs. iteration plot as illustrated in Figure 4-1.

The modal analysis of the optimized part was performed. The modal analysis results for the cell can be seen in Figure 4-2. The frequency of interest in this case is associated with the first mode. Since the results obtained from topology are never accurate, the sensitivity study is done before setting the frequency bound. The sensitivity study is discussed in the following chapters. Also, important is a study of made shapes in consideration of the fact that they should match with the mode shapes of the target structure.

Figure 4-2 Modal analysis of optimized cell

It is very important to study modes shapes while designing a vibration absorber. Modal analysis is the field of measuring and dissecting the dynamic reaction of structures or potentially liquids amid excitation. Illustrations would incorporate measuring the vibration of an auto's body when it is connected to an electromagnetic shaker, examination of unforced vibration reaction of vehicle suspension, or the commotion design in a room when energized by an amplifier.

In structural engineering, the modal analysis uses the combination of mass and stiffness of a structure to find the various instances at which it will naturally vibrate. These instances of vibration are very crucial to note in earthquake engineering, as it is imperative that a building's natural frequency does not match the frequency of expected earthquakes in the region in which the building is to be constructed. If a structure's natural frequency coincides with an earthquake's frequency, the structure may continue to resonate and

experience structural damage. Modal analysis is additionally critical in structures, for example, bridges where the architect ought to endeavor to keep the regular frequencies far from the frequencies of individuals strolling on the bridge. This may not be conceivable, and for this reasons when gatherings of individuals are to stroll along a bridge, for instance, a gathering of warriors, the suggestion is that they break their progression to maintain a strategic distance from potentially critical excitation frequencies. Other regular excitation frequencies may exist and may energize a bridge's natural modes. Engineers have a tendency to gain from such cases (at any rate for the time being) and more present day suspension spans assess the potential impact of twist through the state of the deck, which may be outlined in streamlined terms to pull the pack down with the support of the structure as opposed to enable it to lift. Other aerodynamic loading issues are managed by limiting the range of the structure anticipated to the approaching wind and to lessen wind produced motions of, for instance, the hangers in suspension bridge.

In spite of the fact that the modal analysis is normally done by computers, it is conceivable to hand-ascertain the time of vibration of any elevated structure through admiration as a settled finished cantilever with lumped masses.

In this project, the modal analysis plays an important role. Firstly, to identify the problematic natural frequencies of structures and secondly, to determine the mode shapes associated with the natural frequencies. There should be a perfect match between the mode shapes and frequencies of that of structures to that of the vibration absorbers. After setting up the problem in OptiStruct 14.0, the results of the modal analysis of the vibration can be seen in the figure below. The figure shows the first mode of vibration which was set as a target mode during the topology optimization setup.

4.3 Topology Optimization Results for Various Geometry

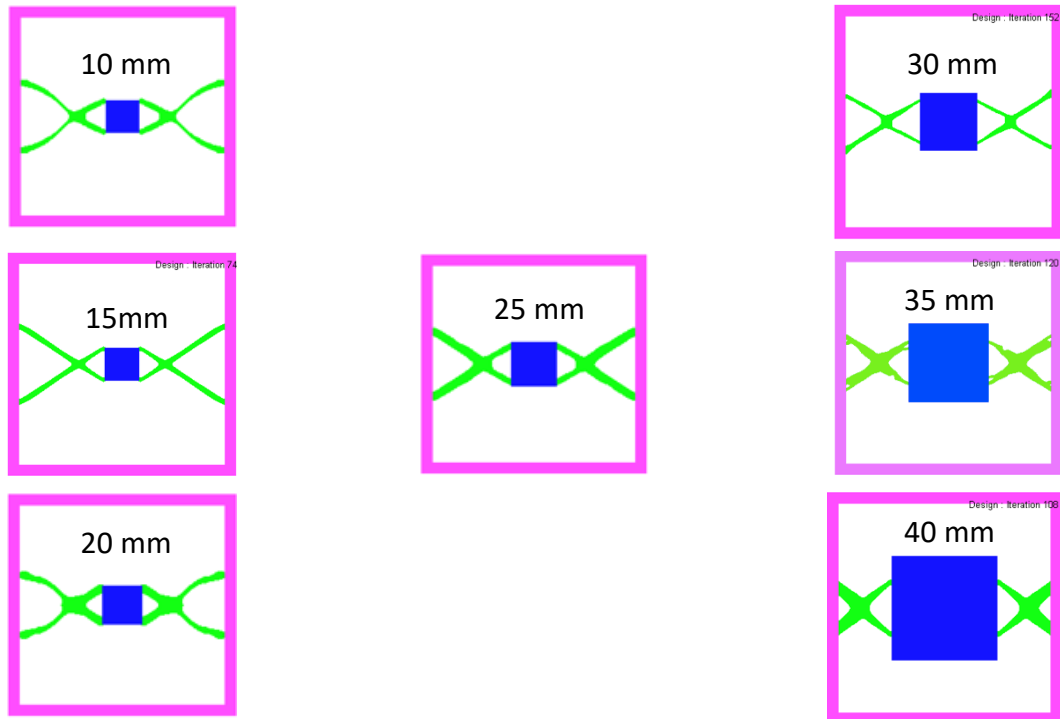


Figure 4-2 Topology Optimization Results for Various Geometry

This study shows the effect of change of geometry of mass on the optimized cell. The study mainly includes the change of mass geometry and thus the change of mass on the material distribution of the system. We know, the natural frequency of the system is a function of K/M ratio. It can be observed from the Figure 4-3 that as the mass geometry increases from the 10 mm to 40 mm, the mass increases and hence to maintain the K/M ratio, the stiffness (K) has to increase which can be concluded from the increase in member size. The very important characteristics to notice here is that each and every cell in the

figure 4-3 is harmonized to same natural frequency.

As we keep on decreasing the mass geometry, the mass of the absorber cell decreases and the least mass is observed in a cell with the mass geometry of 10 mm x 10 mm. This implies that best results considering the weight factor is found in the case of a cell with 10 mm x 10 mm. But cells with that small member size may cause manufacturing related issues.

4.4 Topology Optimization Results for Various Concentrated Mass

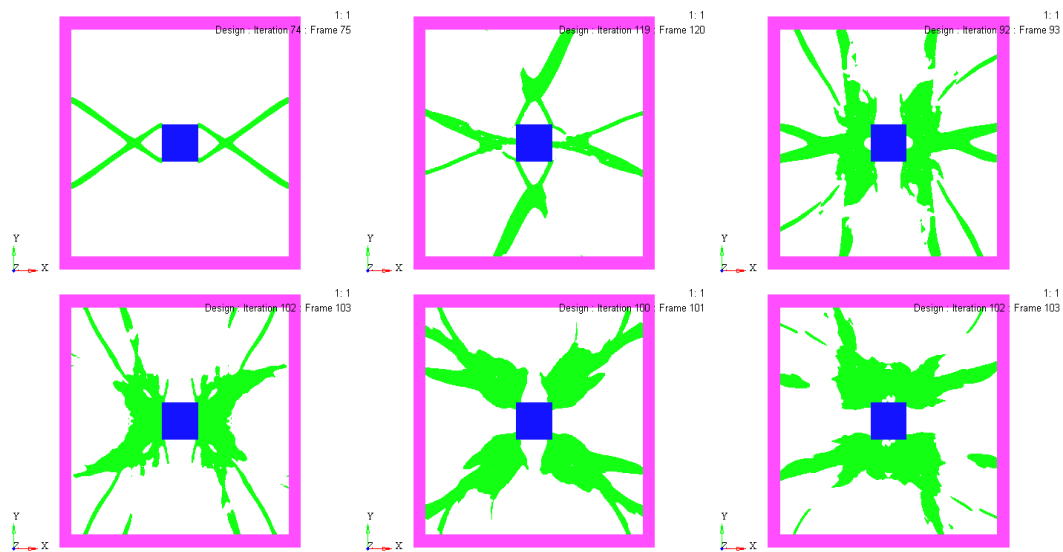


Figure 4-3 Topology Optimization Results for Various Concentrated Masses

This study shows the effect of change of concentrated mass on the optimized cell. The study mainly includes the change of concentrated mass from 3×10^{-3} kg to 9.1×10^{-3} kg with an increment of 25% in each case. We know, the natural frequency of the system is a function of K/M ratio. It can be observed from the Figure 4-3 that as the concentrated mass increases from 3×10^{-3} kg to 9.1×10^{-3} kg, the mass increases and hence to maintain the K/M ratio, the stiffness (K) has to grow which can be concluded from the growth in member

size. The very important characteristics to notice here is that each and every cell in the figure 4-3 is harmonized to same natural frequency.

As we keep on decreasing the concentrated mass the mass of the absorber cell decreases and the least mass is observed in a cell with a concentrated mass of 3×10^{-3} kg. This implies that best results considering the weight factor are found in the case of a cell with 3×10^{-3} kg. In each case, the mode shapes of the cell are not symmetric and hence the material distribution asymmetric in each instance.

Chapter 5

Testing

Mechanical testing is a standard and essential part of any design and manufacturing process and critical in the field of Aerospace. Whether it is characterizing the properties of materials or providing validation for final products, ensuring safety is the principal mission of all mechanical testing. Testing also plays a vital role in providing a cost effective design as well as technological evolution and superiority. The design obtained from the topology optimization was tested on very basic models like plate and beam which will be discussed in the following cases. The test procedure for the tuned mass vibration absorbers is illustrated in figure 5-1

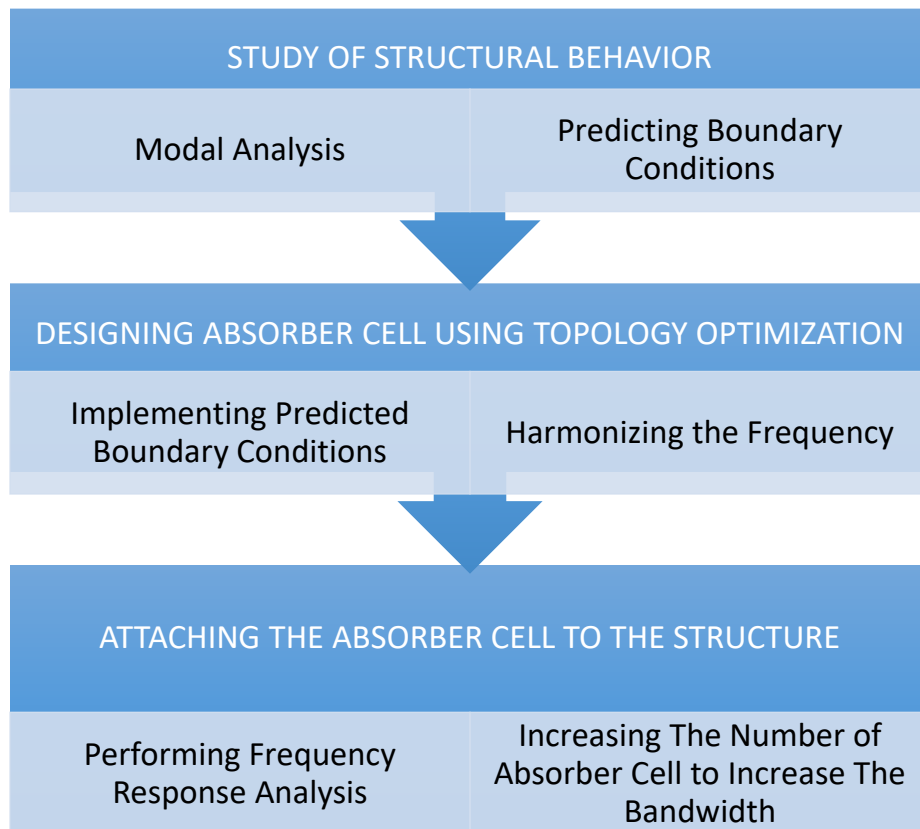


Figure 5-1 Testing procedure

The vibration behavior of structures at higher frequencies has been an intense area of research activity of the Dynamics Group for many years. Applications typically concern noise and vibration transmission in the land, marine and aerospace structures, buildings, etc. The engineer needs tools to model and analyze such structures, as well as to measure and control their vibrational behavior. In this project, it is paramount to study the behavior of structure under excitation, or in engineering term, it is important to study structural dynamics of the structure.

Structural analysis is mainly focuses on finding out the behavior of a physical structure when subjected to force. This action can be in the form of load due to the weight of things such as people, furniture, the wind, snow, etc. or some other kind of excitation such as an earthquake, shaking of the ground due to a blast nearby, etc. In essence, all these loads are dynamic, including the self-weight of the structure because at some point in time these loads were not there. The distinction is made between the dynamic and the static analysis by whether the applied action has enough acceleration in comparison to the structure's natural frequency. If a load is applied sufficiently slowly, the inertia forces (Newton's first law of motion) can be ignored, and the analysis can be simplified as static analysis. Structural dynamics, therefore, is a type of structural analysis which covers the behavior of structures subjected to dynamic (actions having high acceleration) loading. Dynamic loads include people, wind, waves, traffic, earthquakes, and blasts. Any structure can be subjected to dynamic loading. Dynamic analysis can be used to find dynamic displacements, time history, and modal analysis.

A dynamic analysis is also related to the inertia forces developed by a structure when it is excited using dynamic loads applied suddenly (e.g., wind blasts, explosion, earthquake).

A static load is one which varies very slowly. A dynamic load is one which changes with time relatively quickly in comparison to the structure's natural frequency. If it changes slowly, the structure's response may be determined by static analysis, but if it varies quickly (about the structure's ability to respond), the response must be fixed by a dynamic analysis.

Dynamic analysis for simple structures can be carried out manually, but for complex structures, finite element analysis can be used to calculate the mode shapes and frequencies.

A modal analysis assesses the frequency modes or natural frequencies of a given system, yet not really its full-time history reaction to a given info. The natural frequency of a system is function of stiffness and mass of the structure (counting self-weight). It is not dependent on the load function.

It is useful to know the modal frequencies of a structure as it allows you to ensure that the frequency of any applied periodic loading will not coincide with a modal frequency and hence cause resonance, which leads to large oscillations.

The method is:

1. Find the natural mode shapes and natural frequencies
2. Calculate the displacement due to each mode
3. Optionally superpose the response of each mode to find the full modal response to a given loading

After conducting modal analysis, it is very vital to predicting the boundary condition from the results of the modal analysis. The modal analysis gives the basic idea of structural behavior with and without excitation. Predicting proper boundary condition has a significant impact on the optimization procedure.

Depending upon the method of attachment results from the modal analysis, and prediction of boundary conditions they are applied to the vibration absorber. Another important factor to be considered is the natural frequency to which an absorber is to be tuned. From the results obtained from a study of structural behavior and modal analysis, the target mode of natural frequency is set to be optimized. The target frequency is set as discussed in Chapter 3.

The harmonized cell is then attached to the target structure after post processing. It is foremost important to make sure that the attachment between the cell and the structure is achieved with essential effectivity. Once the absorber is attached to the structure, the structure becomes ready for the frequency response analysis. Frequency response analysis is used to calculate the response of a structure under a harmonic excitation. The analysis is to compute the response of the structure, which is transient, in a static frequency domain. The loading is sinusoidal (Time dependent or frequency dependent dynamic load) A simple case is a load that has an amplitude at a specified frequency. The response occurs at the same frequency, and damping would lead to a phase shift [18] [19]. It is always suggested to do the modal analysis before performing frequency response analysis. The modal analysis gives a perfect review of structural behavior under vibration while the frequency response analysis confirms the results. The very last step during the procedure of testing is to add more absorbers and check the response of the system. From the theory of local as we keep on increasing the number of absorbers, the system/ structure become safe over a wide range of bandwidth. Following is the case study following the testing procedure in shown in Figure 5-1.

5.1 Case I

In this instance, the plate with the dimension of 1000 mm x 1000 mm was selected as a target structure. The plate is supported at opposite ends as shown in figure bellow. The thickness of the plate was 1mm throughout. All nodes are constrained along the z direction which implies that there is no vibration out of the plane.

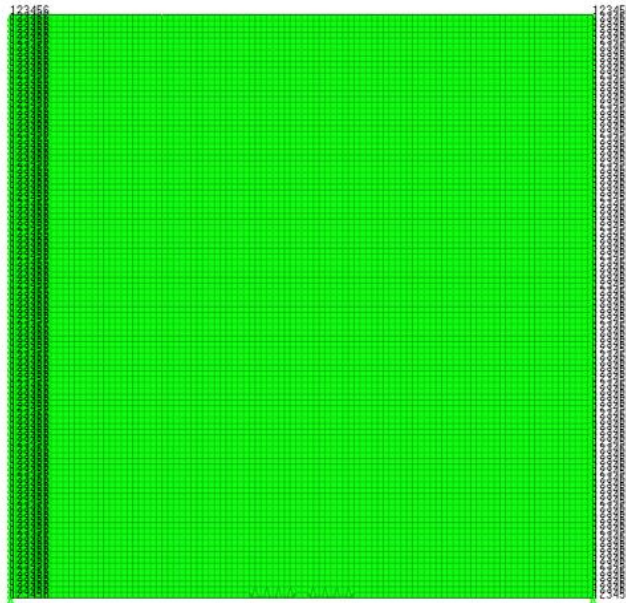


Figure 5-2 Case I -Plate

5.1.1 Study of structural behavior

The structured use, in this case, is a simple plate which is constrained with its opposite edges as shown in Figure 5.1.1. The results of modal analysis results for this case are shown in Figure 5-3 (a). The culmination indicates that the frequency associated with the first mode of natural frequency is 44.80 Hz. The tabular result of frequencies associated with the mode shapes is provided in figure 5-3 (b). The first mode of vibration was considered to be the target mode in this case.

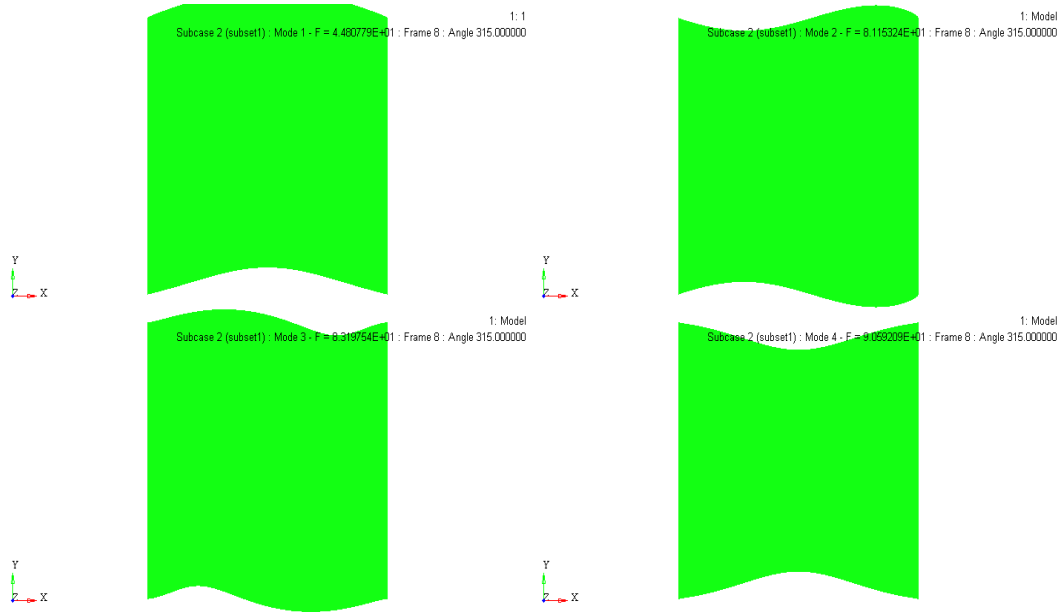


Figure 5-2 (a) Mode Shapes of Plate

Mode Shape	Frequency (Hz)
1	44.80
2	81.15
3	83.11
4	90.59

Figure 5-2 (b) Mode Shape Frequencies

In this case, the vibration cell is added externally and hitched on the structure. The optimized cell will be attached to the lower edge of the plate. The top edge of the pre-optimized cell is constrained in all degrees of freedom. The equivalence is obtained using edge edit tool in Altair OptiStruct 14.0.

5.1.2 Designing Absorber Cell Using Topology Optimization

The design of a unit absorber cell depends upon the structural behavior of the absorber and the target structure. The Design of unit cell model varies from structure to structure depending upon the modal frequencies and the method of connection between the structure and the unit absorber. Since the design is in an initial stage of product development, the preliminary 2-D plate design is used for the base design. In this case, the unit cell is attached to the lower edge of the structure. All degrees of freedom on the upper side of the pre-optimized cell are constrained and predicted in an earlier section, and all nodes are restricted with transverse motion along the Z- axis. This ensures that the vibrations are on the plane. The right and left edge are constrained with its motion along X- axis so that the cell does not transfer any vibration to the adjacent cell when placed next to each other. The problem setup for the topology optimization is done as discussed in Chapter 3. The absorber is tuned to the first mode of frequency (44.80 Hz in this case). The transformation of a cell from pre-optimized to the optimized cell is shown in the figure below.

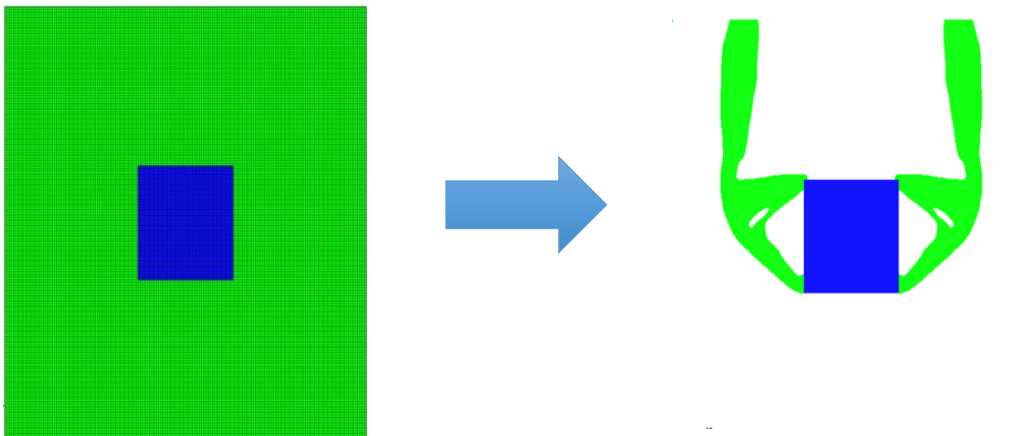


Figure 5-3 Transformation of absorber cell.

5.1.2 Attaching absorber cell to the structure.

To validate the structural integrity of the newly designed vibration absorber cell a finite element analysis was performed using HyperMesh ® 14.0 as a qualitative test to ensure that the design did not have any inherent stress concentrations or fatal flaws. The boundary conditions and mesh parameters were the same as used in the optimization analysis, and the frequency response analysis was used to validate the results and to study the effect of some absorbers on the system.

The unique tool called Edge edit was used to achieve equivalence between the target structure and absorber cell. The coinciding nodes were shared between the cell and the structure. The frequency response analysis was done using direct approach (and not using mode superposition method). The frequency response analysis with forced excitation was studied. The forced excitation of $F \sin(\omega t)$ was applied at the central node on the top edge of the plate along the negative Y axis which is represented in Figure 5-4. The F from $F \sin(\omega t)$ is nothing but the applied frequency dependent unit load. The forcing frequency in the range of 20 Hz to 80 Hz with an increment of 0.25 Hz was selected. The ' ω ' from $F \sin(\omega t)$ represents the forcing frequency acting on the structure.

The analysis was conducted using Altair OptiStruct® 14.0, and the jobs were run on a 3.1 GHz Intel i7 processor with 16 GB RAM. The average run-time was about 2 hours. The resulting displacement of the system can be seen in Figure 5-4 with the previously stated boundary conditions and forcing function.

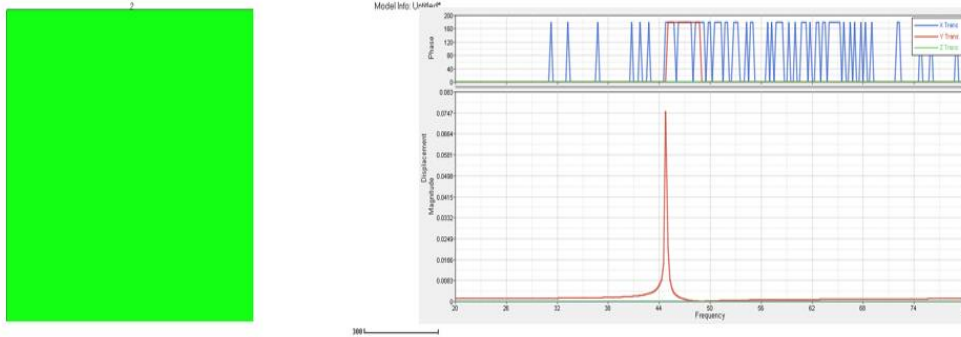


Figure 5-4 Frequency Response Analysis of Plate Without Absorber

It is observed that the response of a system, subjected to a harmonic force having a frequency close to the natural frequency of the system, will be quite significant when the frequencies are equal. It will theoretically be infinite if damping is neglected. The variation of amplitude of response on the excitation frequency is as shown in the graph above. We can clearly see that the displacement hits the peak exactly at the natural frequency of the system hence confirms the results obtained from the modal analysis.

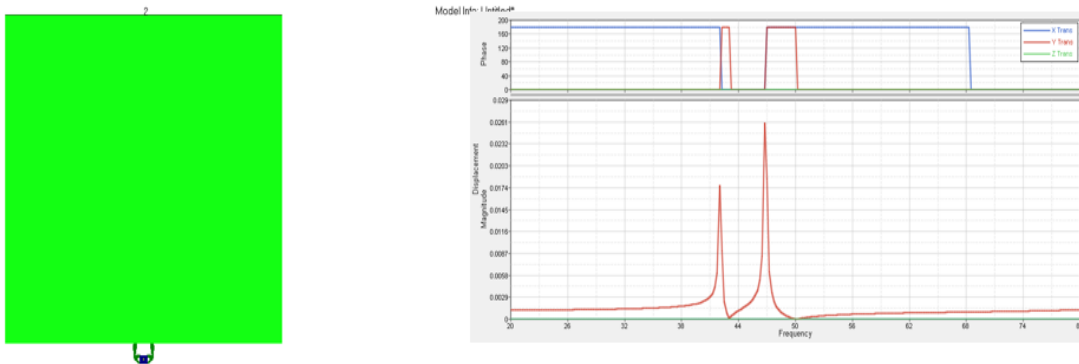
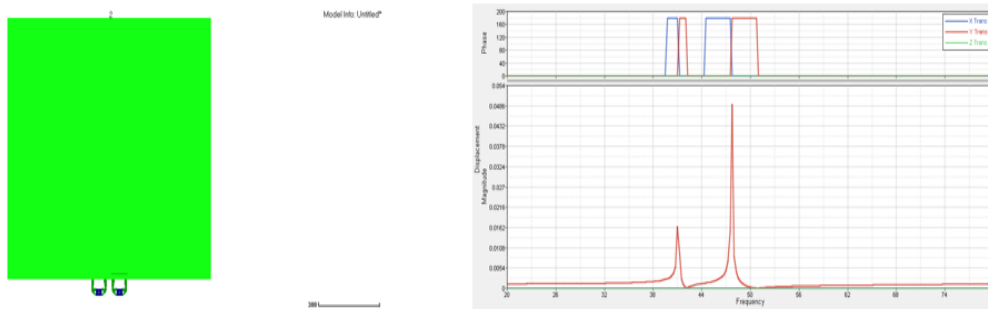


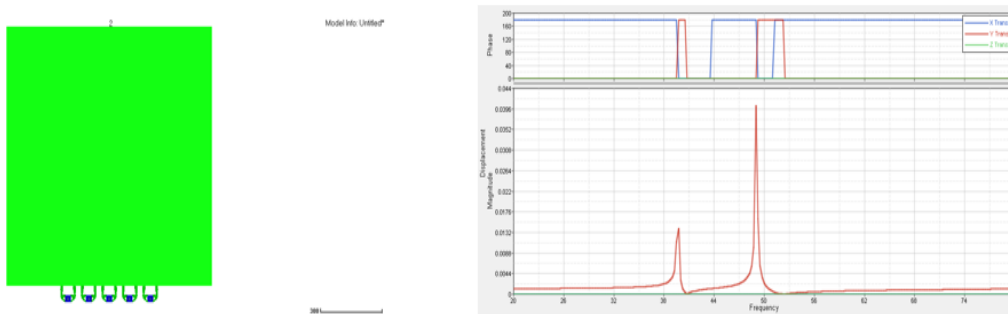
Figure 5-5 Frequency Response Analysis of Plate with Absorber

When an absorber cell system is attached to the main mass, and the resonance of the absorber is tuned to match that of the primary mass, the motion of the main mass is

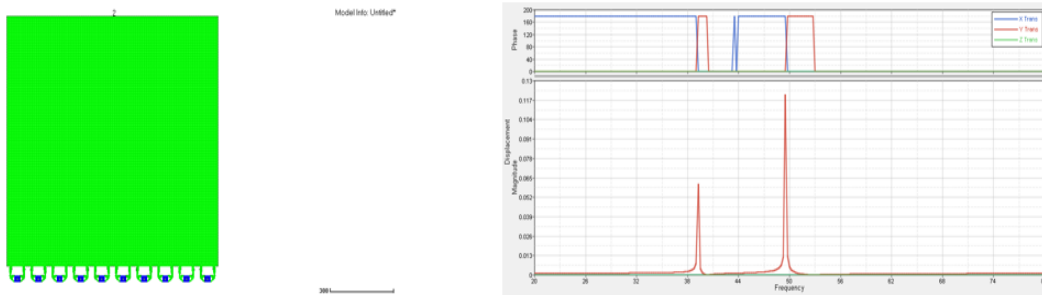
reduced to zero at its resonance frequency. Thus, the energy of the main mass is apparently absorbed by the tuned dynamic absorber. However, if the forcing frequency deviates from 44.80 Hz, the response would be high. This is due to the narrow bandwidth of the absorber. (Shown as a short blue horizontal line in Figure 5-5). Thus it is desirable to increase the bandwidth for vibration reduction. That can be achieved by increasing the number of absorbers. The effect of increase in number of absorbers is shown in Figure 5-6(a), Figure 5-6(b), and Figure 5-6(c)



(a)



(b)



(c)

Figure 5-6 Frequency Response Analysis of Plate (a) with 2 Absorbers (b) with 5 Absorbers (c) with 10 Absorbers

Thus, we can add more absorbers with the same frequency to increase the bandwidth further. The Figure 5-6(a), Figure 5-6(b), Figure 5-6(c), shows the frequency responses of the same primary system with 2, 5, and 10 respectively. It is observed that as the number of absorbers increases the separation between two peaks increases which suggests that bandwidth increases. The bandwidth is widest for a system with ten absorbers and is narrowest for a system with one absorber. The result obtained from this case study holds the theory of increasing bandwidth which was proposed in Chapter 1. It can be concluded that as the number of absorber increases, the bandwidth increases. The increase in bandwidth leads to increase in safety of the structure.

5.2 Case II

In this case, the beam with the dimension of 1000 mm x 350 mm was selected as a target structure. The plate is supported at opposite ends as shown in the figure below. The thickness of the beam was 1mm throughout. All nodes are constrained along the z direction which implies that there is no vibration out of the plane.

Constraint Edges

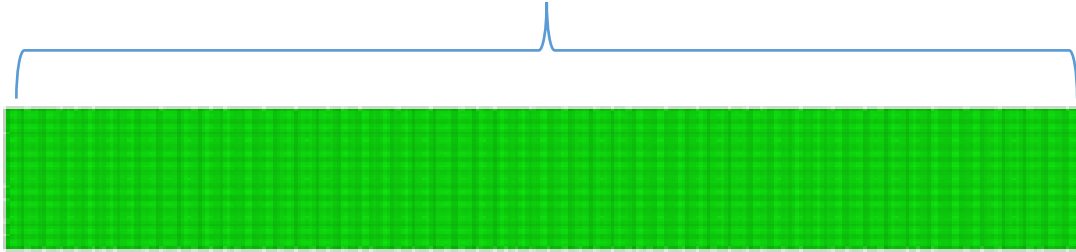


Figure 5-7 Case II-Beam

5.2.1 Study of structural behavior

The structure used in this case is simple beam which is constrained with its opposite edges as shown in Figure 5-7. The results of modal analysis results for this case are shown in Figure 5-8. The culmination indicates that the frequency associated with the first mode of natural frequency is 13.26 Hz. The tabular result of frequencies associated with the mode shapes is shown in figure 5-8 (b). The first mode of vibration was considered to be the target mode in this case.

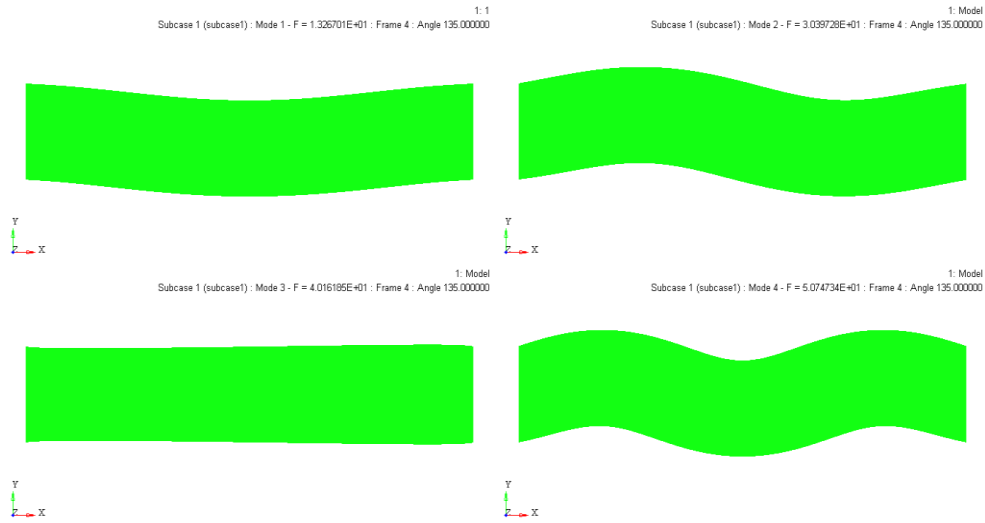


Figure 5-8 (a) Mode Shapes of Beam

Mode Shape	Frequency (Hz)
1	13.26
2	30.39
3	40.16
4	50.74

Figure 5-8 (b) Mode Shape Frequencies

In this case, the vibration cell is added externally and hitched on the structure. The optimized cell will be attached to the lower edge of the plate. The top edge of the pre-optimized cell is constrained in all degrees of freedom. The equivalence is obtained using edge edit tool in Altair OptiStruct 14.0.

5.1.2 Designing Absorber Cell Using Topology Optimization

The design of a unit absorber cell depends upon the structural behavior of the absorber and the target structure. The Design of unit cell model varies from structure to structure depending upon the modal frequencies and the method of connection between the structure and the unit absorber. Since the design is in an initial stage of product development, the preliminary 2-D beam design is used for the base design. In this case, the unit cell is attached to the structure on its lower edge. All degrees of freedom on the upper side of the pre-optimized cell are constrained and predicted in an earlier section, and all nodes are restricted with transverse motion along the Z- axis. This ensures that the vibrations are on the plane. The right and left edge are constrained with its motion along X- axis so that the cell does not transfer any vibration to the adjacent cell when placed next to each other. The problem setup for the topology optimization is done as discussed in Chapter 3. The absorber is tuned to the first mode of frequency (13.26 Hz in this case). The transformation of a cell from pre-optimized to the optimized cell is shown in the figure below.

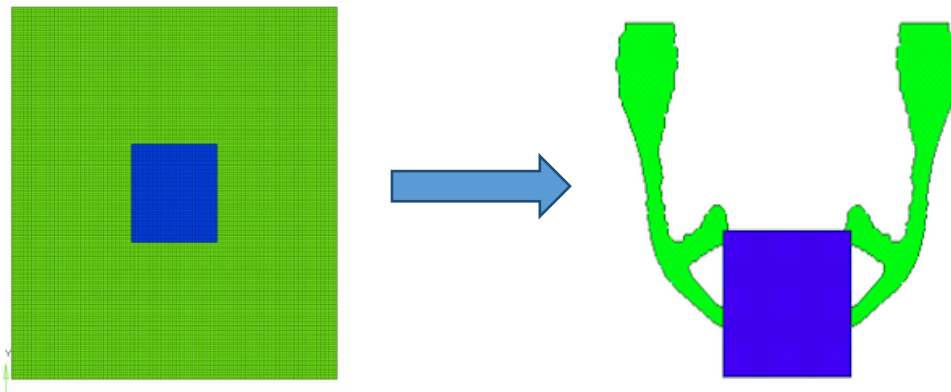


Figure 5-9 Transformation of absorber cell.

5.1.2 Attaching absorber cell to the structure.

To validate the structural integrity of the newly designed vibration absorber cell, a finite element analysis was performed using HyperMesh ® 14.0 as a qualitative test to ensure that the design did not have any inherent stress concentrations or fatal flaws. The boundary conditions and mesh parameters were the same as used in the optimization analysis, and the frequency response analysis was used to validate the results and to study the effect of some absorbers on the system.

The unique tool called Edge edit was used to achieve equivalence between the target structure and absorber cell. The coinciding nodes were shared between the cell and the structure. The frequency response analysis was done using direct approach (and not using mode superposition approach). The frequency response analysis with forced excitation was studied. The forced excitation of $F \sin(\omega t)$ was applied at the central node on the top edge of the beam along the negative Y axis which is represented in Figure 5-4. The F from $F \sin(\omega t)$ is nothing but the applied frequency dependent unit load. The forcing frequency in the range of 0 Hz to 50 Hz with an increment of 0.25 Hz was selected. The ' ω ' from $F \sin(\omega t)$ represents the forcing frequency acting on the structure.

The analysis was conducted using Altair OptiStruct® 14.0, and the jobs were run on a 3.1 GHz Intel i7 processor with 16 GB RAM. The average run-time was about 2 hours. The resulting displacement of the system can be seen in Figure 5-4 with the previously stated boundary conditions and forcing function.

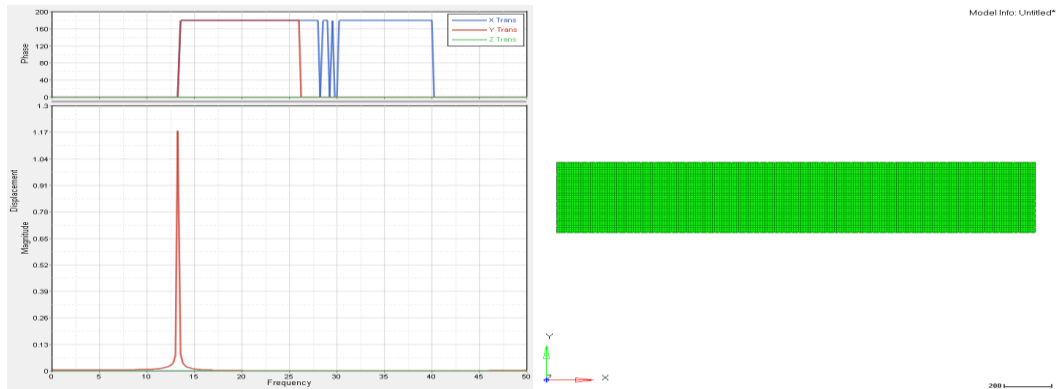


Figure 5-10 Frequency Response Analysis of Beam Without Absorber

It is observed that the response of a system, subjected to a harmonic force having a frequency close to the natural frequency of the system, will be quite large when the frequencies are equal. It will be infinite, theoretically, if damping is neglected. The variation of amplitude of response with respect to the excitation frequency is as shown in the graph above. We can clearly see that the displacement hits the peak exactly at the natural frequency of the system hence confirms the results obtained from the modal analysis.[16]

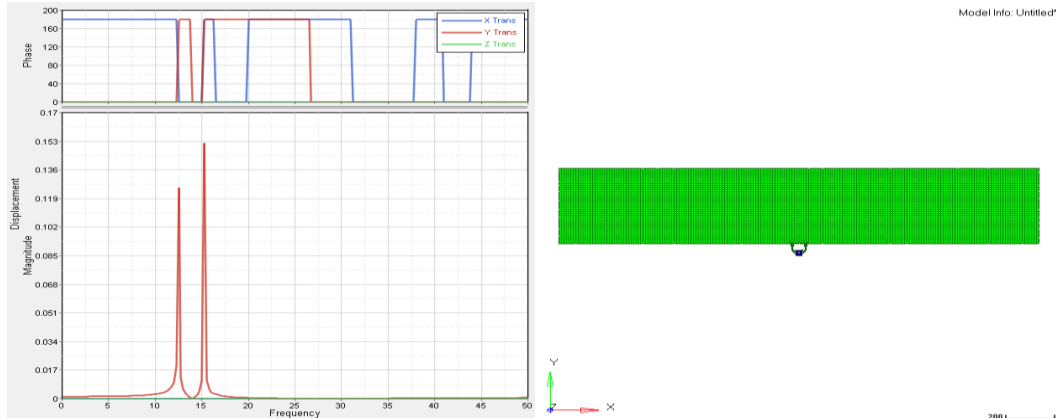
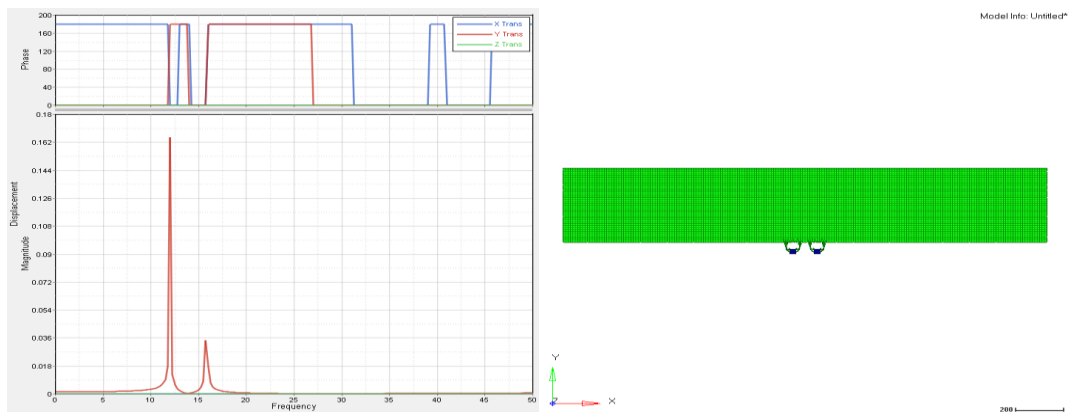
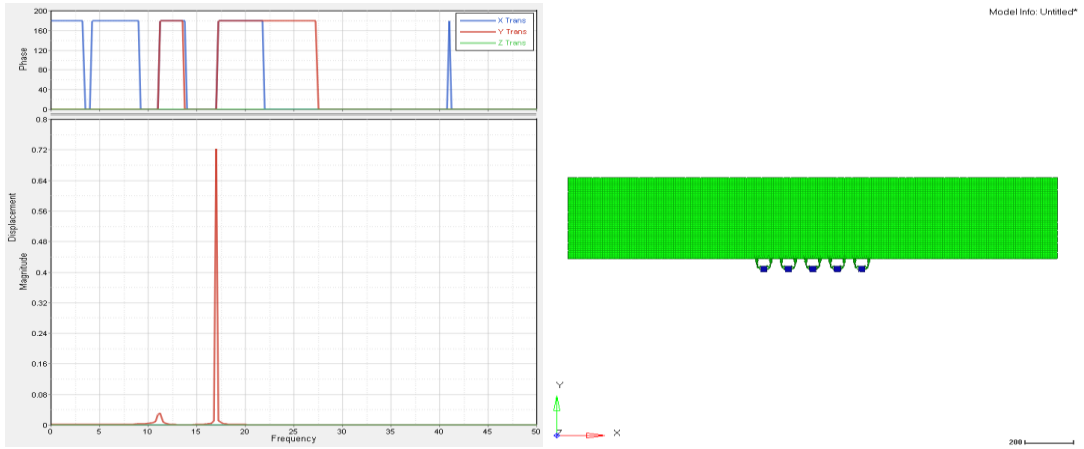


Figure 5-11 Frequency Response Analysis of Beam with Absorber

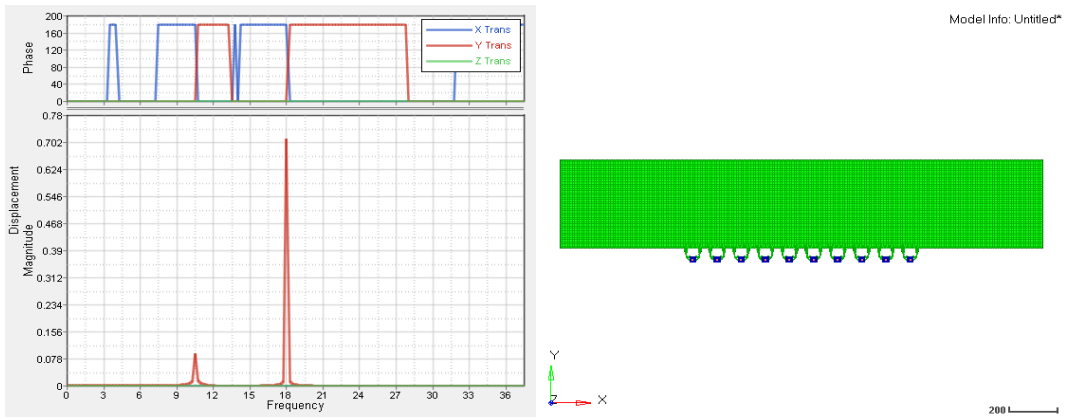
When an absorber cell system is attached to the main mass, and the resonance of the absorber is tuned to match that of the primary mass, the motion of the main mass is reduced to zero at its resonance frequency. Thus, the energy of the main mass is apparently absorbed by the tuned dynamic absorber. However, if the forcing frequency deviates from 13.26 Hz, the response would be high. This is due to the narrow bandwidth of the absorber. (Shown as a short blue horizontal line in Figure 5-5). Thus it is desirable to increase the bandwidth for vibration reduction. That can be achieved by increasing the number of absorbers. The effect of increase in number of absorbers is shown in Figure 5-12(a), Figure 5-12(b), and Figure 5-12(c)



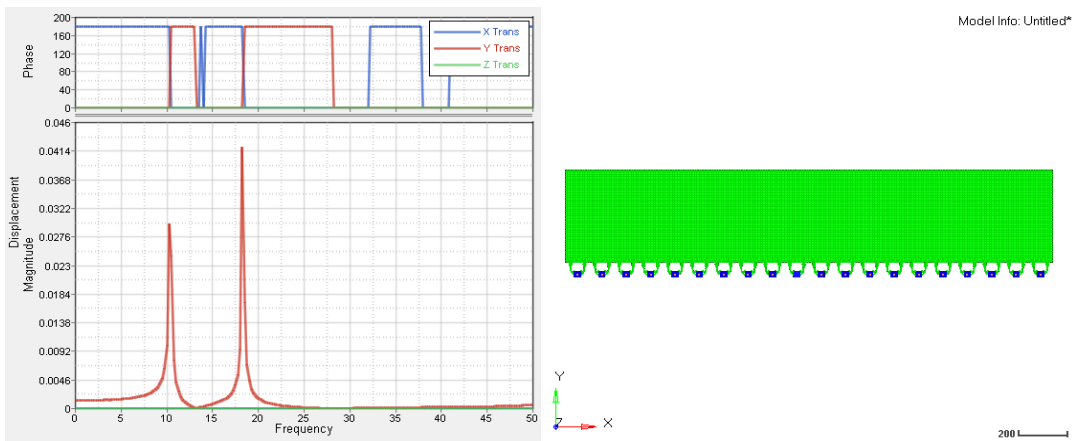
(a)



(b)



(c)



(d)

Figure 5-12 Frequency Response Analysis of Plate (a) with 2 Absorbers (b) with 5 Absorbers (c) with 10 Absorbers (d) with 20 Absorbers

Thus, we can add more absorbers with the same frequency to increase the bandwidth further. The Figure 5-12(a), Figure 5-12(b), Figure 5-12(c), and Figure 5-12(d) shows the frequency responses of the same main system with 2, 5, 10 and, 20 respectively. It is observed that as the number of absorbers increases the separation between two peaks increases which suggests that bandwidth increases. The bandwidth is widest for a system with 20 absorbers and is narrowest for a system with one absorber. The result obtained from this case study holds the theory of increasing bandwidth which was proposed in Chapter 1. It can be concluded that as the number of absorber increases, the bandwidth increases. The increase in bandwidth leads to increase in safety of the structure.

5.3 Case III

In this case, the beam with the dimension of 2000 mm x 400 mm was selected as a target structure. The plate is supported at opposite ends as shown in figure bellow. The thickness of the beam was 1mm throughout. All nodes are constrained along the z direction

which implies that there is no vibration out of the plane.

Constraint Edges

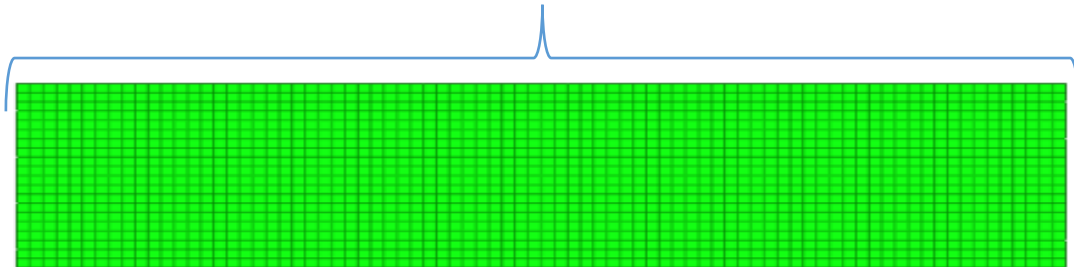


Figure 5-13- Case III-Beam

5.3.1 Study of structural behavior

The structure used in this case is simple beam which is constrained with its opposite edges as shown in Figure 5-7. The results of modal analysis results for this case are shown in Figure 5-8. The culmination shows that the frequency associated with the first mode of natural frequency is 15.19 Hz. The tabular result of frequencies associated with the mode shapes is shown in figure 5-8 (b). The first mode of vibration was considered to be the target mode in this case.

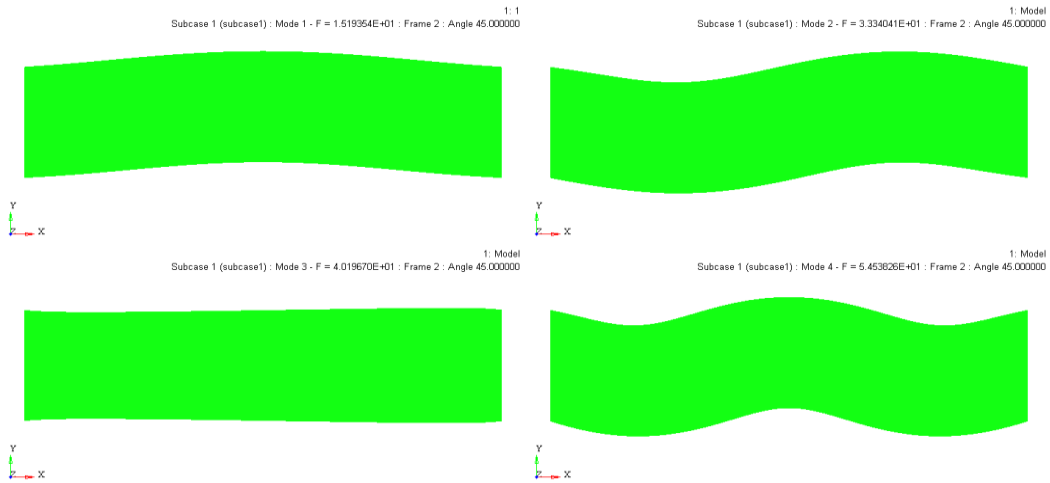


Figure 5-14 (a) Modal Shapes of Beam

Mode Shape	Frequency (Hz)
1	15.19
2	33.34
3	40.19
4	54.53

Figure 5-15 (b) Modal Shape Frequencies

In this case, the vibration cell is added externally and embedded within the structure. The optimized cell will be attached to the lower edge of the plate. The top edge of the pre-optimized cell is constrained in all degrees of freedom. The equivalence is obtained using edge edit tool in Altair OptiStruct 14.0.

5.3.2 Designing Absorber Cell Using Topology Optimization

The design of a unit absorber cell depends upon the structural behavior of the absorber and the target structure. The Design of unit cell model varies from structure to structure depending upon the modal frequencies and the method of connection between the structure and the unit absorber. Since the design is in an initial stage of product development, the preliminary 2-D beam design is used for the base design. In this case, the unit cell is attached to the structure on its lower edge. All degrees of freedom on the upper edge of the pre-optimized cell are constrained and predicted in an earlier section, and all nodes are constrained with transverse motion along the Z- axis. This ensures that the vibrations are on the plane. The right and left edge are constrained with its motion along X- axis so that the cell does not transfer any vibration to the adjacent cell when placed next to each other. The problem setup for the topology optimization is done as discussed in Chapter 3. The absorber is tuned to the first mode of frequency (15.19 Hz in this case). The transformation of a cell from pre-optimized to the optimized cell is shown in the figure below.

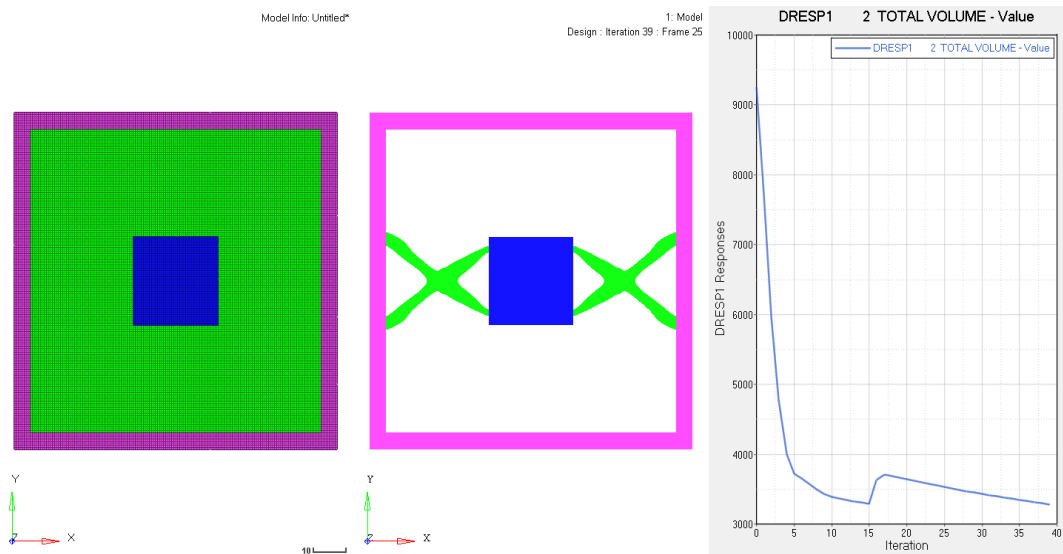


Figure 5-16 Transformation of absorber cell.

5.3.3 Attaching absorber cell to the structure.

To validate the structural integrity of the newly designed vibration absorber cell, a finite element analysis was performed using HyperMesh ® 14.0 as a qualitative test to ensure that the design did not have any inherent stress concentrations or fatal flaws. The boundary conditions and mesh parameters were the same as used in the optimization analysis, and the frequency response analysis was used to validate the results and to study the effect of a number of absorbers on the system.[17]

The special tool called Edge edit was used to achieve equivalence between the target structure and absorber cell. The coinciding nodes were shared between the cell and the structure. The frequency response analysis was done using direct approach (and not using mode superposition approach). The frequency response analysis with forced excitation was studied. The forced excitation of $F \sin(\omega t)$ was applied at the central node on the top edge of the beam along the negative Y axis which is represented in Figure 5-4. The F from $F \sin(\omega t)$ is nothing but the applied frequency dependent unit load. The forcing frequency in the range of 0 Hz to 50 Hz with an increment of 0.25 Hz was selected. The ' ω ' from $F \sin(\omega t)$ represents the forcing frequency acting on the structure.

The analysis was conducted using Altair OptiStruct® 14.0, and the jobs were run on a 3.1 GHz Intel i7 processor with 16 GB RAM. The average run-time was about 2 hours. The resulting displacement of the system can be seen in Figure 5- with the previously stated boundary conditions and forcing function.

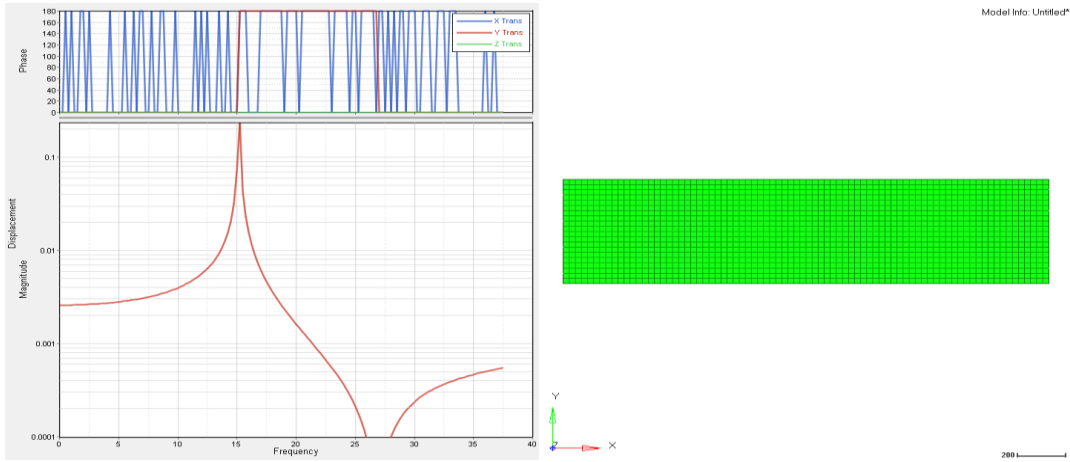


Figure 5-17 Frequency Response Analysis of Beam Without Absorber

It is observed that The response of a system, subjected to a harmonic force having a frequency close to the natural frequency of the system, will be quite significant when the frequencies are equal. It will be infinite, theoretically, if damping is neglected. The variation of amplitude of response with respect to the excitation frequency is as shown in the graph above [16]. We can clearly see that the displacement hits the peak exactly at the natural frequency of the system hence confirms the results obtained from the modal analysis.

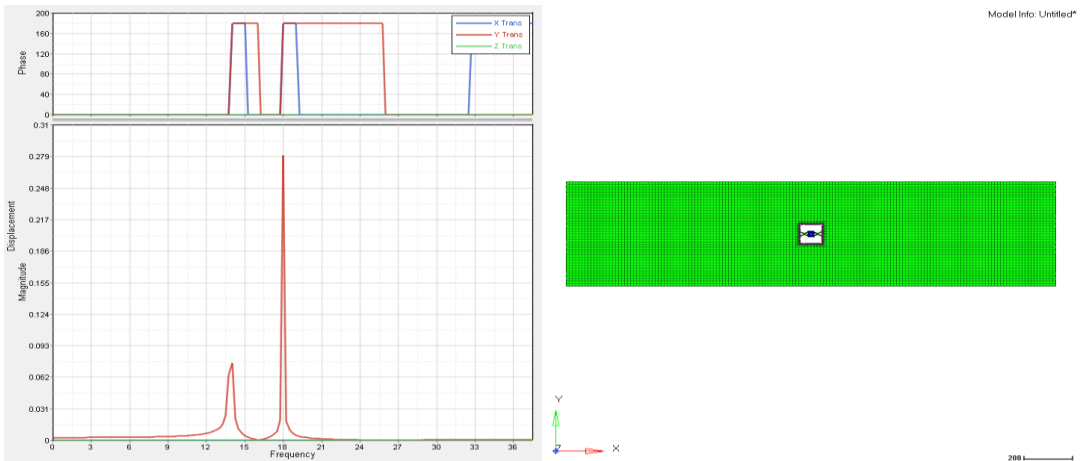


Figure 5-18 Frequency Response Analysis of Beam with Absorber

When an absorber cell system is attached to the main mass, and the resonance of the absorber is tuned to match that of the main mass, the motion of the main mass is reduced to zero at its resonance frequency. Thus, the energy of the main mass is apparently absorbed by the tuned dynamic absorber [16]. However, if the forcing frequency deviates from 15.19 Hz, the response would be high. This is due to the narrow bandwidth of the absorber. (Shown as a short blue horizontal line in Figure 5-5). Thus it is desirable to increase the bandwidth for vibration reduction. That can be achieved by increasing the number of absorbers. The effect of increase in number of absorbers is shown in Figure 5-18, and Figure 5-19

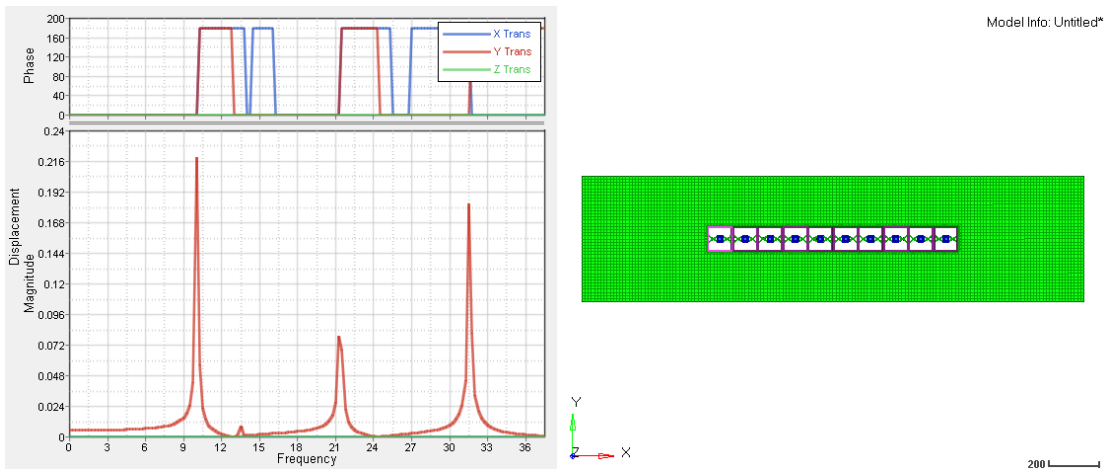


Figure 5- 19 Frequency Response Analysis of Plate with 10 Absorbers

Thus, we can add more absorbers with the same frequency to increase the bandwidth further. The Figure 5-18 and Figure 5-19 shows the frequency responses of the same main system with 1 and, 10 respectively. It is observed that as the number of absorbers increases the separation between two peaks increases which suggests that bandwidth increases. The bandwidth is widest for a system with ten absorbers and is

narrowest for a system with one absorber. The result obtained from this case study holds the theory of increasing bandwidth which was proposed in Chapter 1. It can be concluded that as the number of absorber increases, the bandwidth increases. The increase in bandwidth leads to increase in safety of the structure.

Chapter 6

Conclusion

The following conclusions can be drawn from the above investigation

SIMP is a reasonably and rigorously derived gradient method for topology optimization, which usually gives a solution near the correct global optimum. In this thesis, it is observed that the tuned mass vibration absorber can be designed using topology optimization. The final design obtained from topology optimization is continuous in nature and has the natural frequency equal to that of the natural frequency of the structure. It can be noted that the vibration absorber cells can substitute the classical spring- mass structure.

In this thesis, while the auxiliary system is tuned, different values of the mass ratio can be chosen to observe the responses of the primary and secondary systems. The responses can also be observed even if the auxiliary system is not attached. It can be concluded that when a tuned mass absorber is added to the structure, the response becomes zero when the forcing frequency is exactly equal to the natural frequency of the system.

The tuned mass absorber can be tuned to natural frequency using various combination of mass and stiffness. From the study, it can be concluded that the natural frequency of the absorber is a function of K/M ratio where 'K' is stiffness of an absorber while M is a central mass of the absorber. Also, the similar theory can be applied in the case of absorbers with concentrated mass.

Lastly, the case study shows the frequency responses of the same primary system with 2, 5, 10 and 20 absorbers. It is observed that as the number of absorbers increases the separation between two peaks increases which suggests an increase in bandwidth. The bandwidth is widest for a system with 20 absorbers and is narrowest for a system with

one absorber. The result obtained from this case study holds the theory of increasing bandwidth which was proposed in Chapter 1. It can be concluded that as the number of absorber increases, the bandwidth increases. The increase in bandwidth leads to increase in safety of the structure.

The proposed Tuned Mass Absorber could improve competition in the marketplace by quickly responding to changing customer needs and preferences. Topology optimization could design a better absorber layout considering higher accuracy, higher absorbing effects, and less weight of structures. Suitable future applications of these technologies would be in automobile, aerospace, Machinery and Civil Structures. The presented considerations and recommendations could be helpful for designers or engineers to tackle vibration problems.

Chapter 7

Future work and Limitations

Based on the results of this study, the following are limitations and proposals for future work:

Tuning absorbers for lower frequencies

During topology optimization of absorber cell, it was observed that tuning vibration absorbers for lower frequencies was a major issue. Tuning vibration absorber near lower frequencies leads to results with violation of the constraint. This issue can be solved by using very fine mesh size which increases the computational time. The resulting absorber cell may have members with thicknesses that are difficult to manufacture.

Positional study of absorbers

The frequency response analysis of structures with different positions can be done and compared for best results. The study can include the calculation of bandwidth and response of the system.

Designing 3D model of an absorber cell

In this study, the 2D model was developed. It is necessary to develop a 3D model considering its application for functional structures like an Aircraft wing and Bridges to make a model workable in real life.

Testing absorbers in real life structures

A more practical approach could be to conduct structural behavior study of real life structures. For this, it is important to examine the mode shapes and modal frequencies before designing the absorber cells for the structure. Once this study is done, it will be easy to develop a 3D model of an absorber for such structures.

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Biological Information

Swapnil Subhash Harel received his Bachelor's degree in Mechanical Engineering from Dr. Babasaheb Ambedkar Technological University, India in the July 2013. He worked on his cad skills and pursued training as a FEA Analyst from August 2013 to January 2015. He then decided to pursue his Master's degree in Mechanical Engineering from the University of Texas at Arlington in Spring 15. He has been actively involved in the research and his interests were Finite Element Analysis, Mechanics of Materials, Structural Optimization and Structural Dynamics. He also trained and motivated new members in the field. Swapnil plans to pursue her career in the direction where his experience and expertise is utilized.