بسم الله الرحمن الرحيم

Vibration analysis of Rotary Drill Rig and Modification Using Absorber Technique.

A thesis submitted to the department of Mechanical and Production Engineering (MPE), Islamic University of Technology (IUT), in the fulfillment of the requirement for degree of Bachelor in Mechanical Engineering.

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It is hereby declared that this thesis or any part of it has not been submitted elsewhere for the award of any degree.

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We seek excuse for any errors that might be in this report despite our best efforts.

ABSTRACT:

Rotary drill rig being an essential tool in construction engineering, reducing its vibration became necessary due to avoiding any kind of fracture or failure. With the depth of hole getting increased, the vibration becomes so intense that it not only affect the machine but also causes harm to the existing construction of its surrounding. To find its solution, many absorber technologies have been introduced. In this study, a simple prototype of an actual drill rig has been created to observe and compare the vibration of a drill rig with and without an absorber. This study has been implemented adding a simple mass spring as an absorber with the protype to see what properties of the mass spring can reduce the vibration in a significant amount. At the end, a suitable liquid damper is suggested with the stiffness of 2331000 N/m which is a silicon and rubber damper. An Ansys based simulation has been run in our study which shows how this damper is able to reduce the bending vibration to a significant amount.

Keywords:
Rotary Drill Rig, Drill-string, Vibration, Vibration Absorber, Linear Absorber, Modal Analysis, Harmonic Analysis, Displacement, Amplitude, ANSYS.

LIST OF FIGURE:

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CHAPTER 1:

INTRODUCTION:

1.1 INTRODUCTION:

The rotary drill-rig is a machine that drills holes in the ground. It is of great significance to the construction industry. Also, rotary drill-rig is used to reach the reservoir of oil, gas that lies underground. Rotary drill-rig has been one of the revolutionary tools in the industry due to its excellent construction quality, high work-efficiency and environmental protection[1]. It is considered as a critical component due to the operations it conducts. While drilling if the depth of the hole increases the ration of its length with respect to the cross-sectional area keeps getting smaller and smaller from the top of the drill to the bottom[1]. So, the vibration of the telescopic drill-string becomes more intense and this severe vibration can also occurs from the low compressibility and high shear strength of the hard rock the drill-rig goes through [2]. It, however, not only fractures the drill-string, but also decreases the drilling performance, service life, dependability of the rotary drill-rig. It may also disturb the accuracy of the hole drilling process which results to more time consumption and access cost. That is why the study of drill string vibration has got attention to many researchers and solving that issue became significant in the engineering field nowadays[1].

A rotary drill-rig contemporaneously shows different kinds of vibration such as bending vibration, twisting vibration, longitudinal vibration. In this study, a linear mass spring is proposed to be connected to the drill- rig as an absorber to reduce its bending (transverse)vibration.

A rotary drill rig is consisted of several parts such as - 1.flat head, 2.first baffle ring, 3.first section of drill-string, 4.second section of drill-string, 5.third section of drill-string, 6.fourth section of drill-string, 7.reduction vibration assembly, 8.first foreign key, 9.first internal key, 10.spring plate, 11. spring, 12.square head, 13.axis[1].

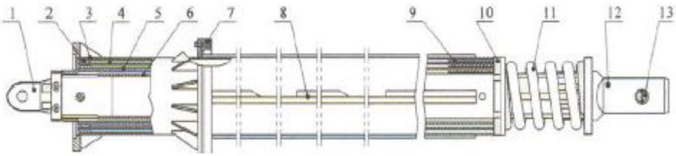


Figure 1: Structure of a drill-string[1]

For the ease of calculation, a simplified version of an actual drill rig has been introduced in this study.

The simplified version contains a 1m long beam with a fixed support in one end and a mass spring used as an absorber. Bending vibration of this beam has been observed for both with and without the absorber. The results of this experiment show how a linear absorber affect the bending vibration of this simplified prototype. Also, a suitable stiffness has been suggested at the end of this study which is 2331000 N/m. This is the stiffness of a silicon and rubber damper and the Ansys based simulation shows that it is capable of reducing the bending vibration of our proposed prototype to a good extent.

1.2 OBJECTIVE:

The main objective of this study is to identify the behavior of the vibrating drill-rig. Since this vibration works against the efficiency of the rotary drill-rig, a suitable solution to dissipate the vibration is represented here. Among various kind of absorbers, linear absorber is chosen to be observed in this study. And a simple prototype is made to regulate and simplify the observation.

The research purpose of this study is to find a suitable linear-absorber and observe its behavior while regulating its properties in a simplified version so that these information can help in further design of linear-absorber in construction engineering.

CHAPTER 2:

LITERATURE REVIEW :

2.1 LITERATURE REVIEW

Absorber being a necessity in construction engineering both linear and non-linear absorber technologies have been studied by many researchers. In the study of Elsayed, M. A. [3],it is shown how the interaction between drill-bit and hard-rock like material can produce vibration by itself and how much damage it can cause to the system. Also, a solution of adding an absorber with specific stiffness and damping ratio has been introduced in their study[3]. In 2012,in the International Journal of Acoustics and Vibration, a theoretical model of rotary drill-rig attached with mass-spring damper was represented[4]. Their study also showed a possible specifications of a liner-absorber to suppress the bending vibration [4].In the study of Gromadskiy, V A [5] experimented the stiffness coefficient of a longitudinal oscillations dampers to use it as an absorber for rotary drill-rig[5]. Tahir, A[6] tried to introduce non-linear absorber which can control the torsional vibration of a rotary system[6].

Nonlinear targeted energy transfer technology (NTETT) has been playing a significant role in the engineering of dissipating vibration energy[7] . A very little amount of mass is used as a subsystem to reduce the severe vibration in non-linear energy sink technology [8]. The reason behind adding a subsystem is to constructively absorb the vibration from the primary vibrating system .At the end, it can also be observed that a very tiny amount of vibration energy is stored in the primary one [9] . At present, many researchers has been working on NETT as a solution for vibration reduction in rotary drill-rig[1] . Zhang et al. [8] studied an inertial nonlinear energy sink which is a NTETT and they tried to introduce this technology so that the demand of a large mass as an absorber can be overcome , and their amplitude-frequency responses via computer-based simulation showed that the inertial sink technology was able to increase the vibration suppression performance[8]. The response characteristics of mechanical systems with inherently nonlinear absorbers basing on NTETT were explored by Avramov et al. [10], who reported that the absorber would rapidly disperse the system's vibration energy. Using increasingly high values of inertance, Javidialesaadi et al. [11] proposed an inerter-enhanced nonlinear energy sink based on NTETT and the energy sink's effectiveness can be significantly increased. NTETT was used by Hubbard et al. [12] to monitor the deformation and instability of the wind-tunnel wing .The absorber, according to the Arabian Journal for Science and Engineering, could constructively increase the dissipation of second-order vibration energy in wings[12]. In the study of Lin et al. [13] the rate of nonlinear targeted energy transfer was monitored between the linear oscillator and nonlinear energy absorber. In their investigation the energy transfer rate was found which was firmly related to 1:1 capture resonance[13]. AL-Shudeifat et al. [14] investigated how to allow rapid and one-way scattering of shock energy from low- to high-frequency structural modes, demonstrating that when shock energy was transferred to high-frequency modes, the transient response amplification was minimized significantly and power dissipation was enhanced[14]. Zhang et al. [15]provided a detail analytical observation on how the vibration suppression and targeted energy transfer is related where they used flow induced vibration. In their designed system two dimensional wing was coupled with two non-linear energy sinks and the system was kept under the freestream by the usage of numerical methods[15] . Dolatabadi et al. [16] indentified how a piston in the cylinders plays a vital role in generating noise so they used a passive approach to control the secondary piston motion. This new approach was modeled on the energy transfer of the highly transient oscillations to a nonlinear absorber [16].

Furthermore, many researchers have been attracted to the properties of an absorber and they are trying to figure out how these properties can help reducing the vibration of the primary system. Georgiade et al. [17] studied the positioning of an absorber and they suggested the absorber can dissipate more vibration energy if it is placed between the center of mass and the end of the primary structure[17].In the study of Huang et al. [18]they showed how an absorber performs more effectively when it has negative stiffness to control force transmission. In the range of their optimum parameters the absorber works better during mass response control [18]. Qiu et al. [19]have worked with conical spring as a nonlinear energy absorber and showed the principle behind it[19]. Taleshi et al. [20] worked on the structural parameters and how it influences the energy transfer effect,also a position of installation was discussed in their study for the nonlinear energy absorber[20]. Yao et al. [17] studied the piecewise linear stiffness of an elastic rod to use it as an absorber in a rotor system[21].

Under ample bandwidth and random signals, Wei et al. [22] addressed the targeted energy transfer by vibro-impact cubic nonlinear energy absorber, and the most energy of the broadband excitation was absorbed[22]. The absorber's vibration reduction has recently been shown to have outstanding efficiency under a variety of excitations[22]. .The dissipation of vibration energy of the absorber during quasi-periodic random external force excitation was discussed by Starosvestsky et al. [23].Lo Feudo et al. [24] found an absorber for impulsive shock, single-frequency excitation and initially-imposed displacement free vibration cases and they suggested it to be a non-linear magnetic vibration absorber. To analyze nonlinear vibration absorbers, Zang et al. [25]proposed generalized transmissibility and suggested that quasi-periodic motion is an acknowledgement of optimal transmissibility[25]. To control vibrations under periodic and transient excitation, Qiu et al.[26] formulated design requirements for an optimally tuned vibro-impact nonlinear energy sink[26]. Miaomiao Li[27] studied the effect of a rubber-silicon combined damper on a vibrating shaft and properties of this damper has been used in this study. Although it seems easier to simulate the absorber properties and criteria computationally, it is not that feasible when it comes to practical field. Keeping that in mind this paper started from a very simple prototype and a very basic absorber is used so that practical demonstration can be done easily. of the machineries.

For this study we choose to work with linear absorber. One of the reasons for working with linear absorber is, the behavior of the absorber can be easily understood. Also, it requires a very little mass with respect to the main vibrating system so it will not cause a massive change in the primary vibrating system and it will be easier to demonstrate it in practical field.

2.2 ABSORBER MECHANISM: LINEAR, NON-LINEAR:

If a sinusoidal force acts on a general mass-spring vibrating system it will vibrate with a frequency. When this forced frequency equals to the natural frequency of the primary vibrating system, this incident is addressed as resonance and the frequency response of that system becomes infinite. This infinite frequency can be controlled by adding a damper which can be another mass-spring system and if the resonance of the damping system and the main vibrating system is tuned is cancels out the resonance vibration of the main system. So, the vibration energy of the primary system is absorbed by another mass-spring damper and that is why it is called the absorber.

In case of construction engineering, the primary system can be any vibrating machinery and many researches have been conducting to find out suitable design for absorbers to use in practical field for different machineries. For rotary drill-rig both linear and nonlinear absorbers are used to reduces its vibration. Linear absorbers behave in a way so that it follows the linear equation of a system and the displacement curve follows the straight line whereas the nonlinear produces different curves according to its motion. In practical field a linear absorber is easier to implement

CHAPTER 3:

GEOMETRY AND MATHEMATICAL FORMULATIONS :

3.1 Rotary Drill-rig:

Rotary drill-rig is one of the most common tools of drilling in the construction industry. It is the main part of a drilling machine that is used for making bore holes through the earth surface. Rotary drill rig produces high level of torque and rotation to make boreholes through the ground. Depending on the purpose of its use it has differences in its capacity and structure.

In this study ,the SANY SR150C rotary drilling rig has been experimented which is used for small pile civil construction[28].A simple prototype has been made following the actual model of this machine for the experiment.

3.2 GEOMETRY OUTLINE:

The actual parameters of the drilling have been scaled down from length 17.33m to 1m. And an ANSYS based simulation has been run on the simple protype to observe its behavior with and without an absorber attached.

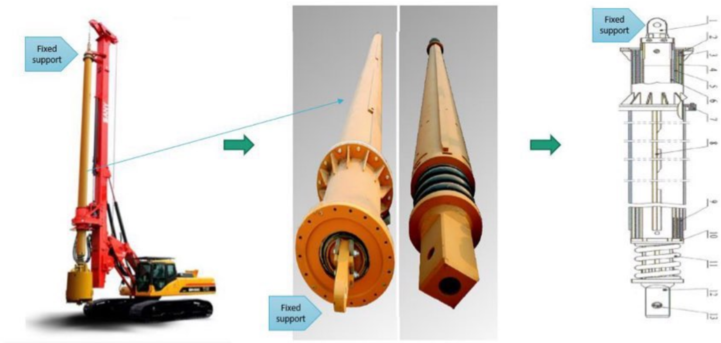


Figure 2: Structure of the SANY 150C rotary drill-rig[28][1].

To observe the dynamic behavior of the absorber, the model is designed basing on the classic impact damper model. In this study, a prototype model is designed where the drill-rig which is the primary vibrating system, is simplified into a single beam. A simple mass spring has been used as the absorber of the system.

The properties and parameters of the simplified model has been determined from the previous work of Xinxin Xu, [1] where the primary vibrating system is consisted of mass

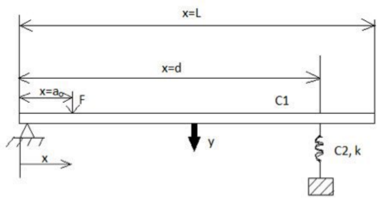


Figure 3: Dynamic 1-D model of the system

M, length L, damping $C1$[1]. The absorber is connected to the beam at a length of x_d where x is the distance from the left and the very left part of the beam is considered fixed. Also an impact force is applied at the length of $x=a_0$.The absorber's mass is m_t, linear stiffness is K, viscous damping is $C2$. In reality , the mass of the absorber has to be very small compared to the mass of the primary system which is of great significance in construction design since the absorber's light weight is not supposed to substantially increase the weight of the primary system, nor is it allowed to drastically alter the primary system's overall structural parameters. That is why it is suggested that the ratio of the absorber's mass with respect to primary system's mass should be within the range of 0 to 1[1].

If expressed theoretically it can be written that, $0 \leq \varepsilon < < 1$, $\varepsilon = m_t / M$.

Some values has been assigned to monitor this study and the values of those parameters are given as - $L= 1.0$, $a_0= 0.5$. The position of the absorber is proposed to be $x_d= 0.9$ in this study. Now, in the study of Xinxin Xu1,2[1] for $\varepsilon = m_t / M$,they diagnosed about 15 values i.e $\varepsilon =0.01, 0.02, 0.03, 0.04, 0.05, 0.06, 0.07, 0.08, 0.09, 0.1, 0.2, 0.3, 0.5, 0.7, 1.0$[1] along with two other parameters which are $R= K/M$ and $\varsigma 2 =C2/M$. In this study, the other two parameters have been ignored since the aim of this experiment was to find out a suitable range of damping and stiffness for the absorber. The computational simulation is done where $\varepsilon = m_t / M= .007$ for this study to observe if the absorber is able to make any significant change or not.

CHAPTER 4:

NUMERICAL METHODOLOGY

In this study the simulation results from Ansys software is focused to calculate the frequency response and the modal analysis of the shaft of a rotary drill string. As mentioned earlier, a prototype design is used for the drill rig that is scaled down to 1m in length. Scaling down was done by the concept from the paper “Properties of Drill string Vibration Absorber for Rotary Drilling Rig” by Xinxin Xu [1]. In this study, the influence of the absorber on the mass, the stiffness, and the damping parameters is discussed via software based results. For simplifying the calculation, the values of the primary vibrating system parameters are specified as follows:

- ✓ Length: 1m.
- ✓ Height: 150mm
- ✓ Width: 150mm
- ✓ Material: Structural Steel
- ✓ Weight: 176.625 kg
- ✓ Density: 7850 kg/m^3
- ✓ Coefficient of Thermal Expansion: 1.2E-05 C^-1
- ✓ Young’s modulus: 2E+11 pa
- ✓ Poisson’s ratio: 0.3
- ✓ Bulk Modulus: 1.6667E+11 pa
- ✓ Shear Modulus: 7.6923E+10 pa

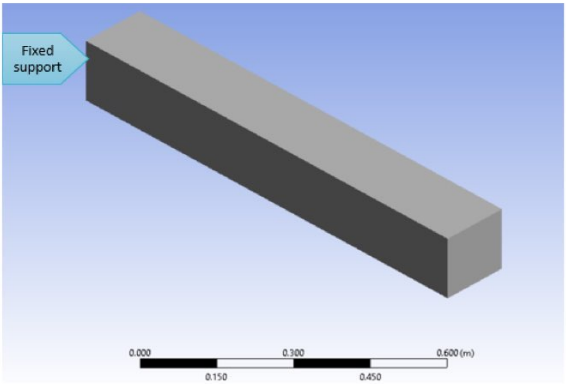


Figure 4:Simplified prototype without absorber.

This simplified prototype gives similar results in Ansys software compared to the original model used in practical field. The actual drill string is 17.33m in length and weight is 27000kg. Other properties are same as given for the prototype. The harmonic response for both the model and the protype is similar. So, other simulations on the prototype gives us required results for the actual model.

For absorber solid mass spring is attached to the 1m beam with a spring. The stiffness, and the damping parameters of the spring are not specified, different range of values have been used for these parameters to find the most suitable value. Using these values, a suggestion was made about the nature of the absorber and the material of the absorber.

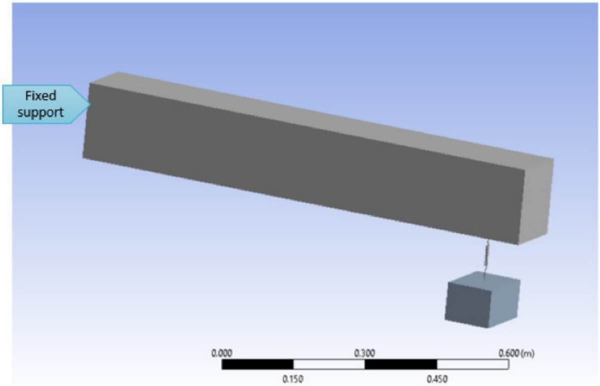


Figure 5:Simplified prototype with mass-spring absorber.

Properties of the mass block used as absorber are given as followed:

- ✓ Length: 150mm.
- ✓ Height: 75mm
- ✓ Width: 150mm
- ✓ Material: Structural Steel
- ✓ Weight: 13.25 kg
- ✓ Density: 7850 kg/m^3
- ✓ Coefficient of Thermal Expansion: 1.2E-05 C^-1
- ✓ Young's modulus: 2E+11 pa
- ✓ Poisson's ratio: 0.3
- ✓ Bulk Modulus: 1.6667E+11 pa
- ✓ Shear Modulus: 7.6923E+10 pa

CHAPTER 5:

RESULTS AND DISCUSSIONS

5.1 RESULTS

5.1.1 Modal Analysis:

- ✓ **Without absorber:**

The aim of modal analysis is to find out the natural frequency of a structure. In this study the modal analysis has been done using Ansys simulation. First the total deformation of the simple prototype has been found without attaching any absorber. Since it is a very simple structure very fine meshing was not necessary. The mesh type used for this was coarse mesh which divided this structure into 180 elements. After the simulation was run, five natural frequencies has been found for different mode shapes of this structure.

Mode	Frequency (Hz)
1	**121.18**
2	**121.18**
3	**692.58**
4	**692.58**
5	**721.36**
6	**1267.8**
7	**1731.9**

From these 7 modes bending vibration was found for the frequency of 121.18 Hz and 692.58 Hz.

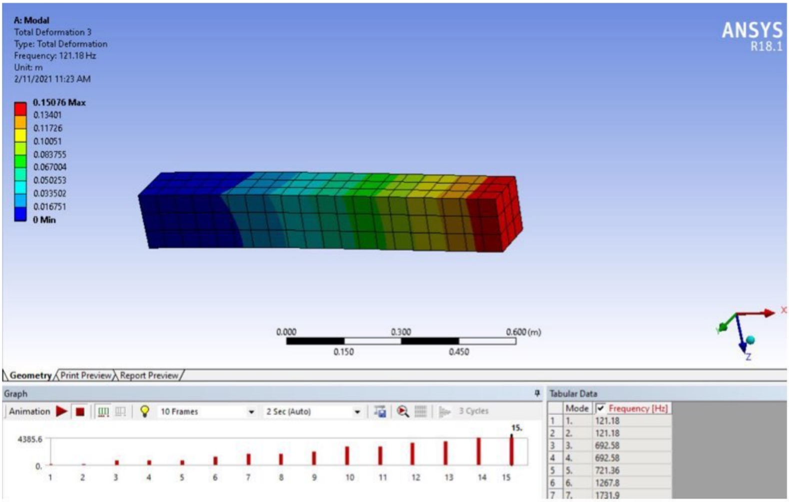


Figure 6:Modal Analysis of the prototype beam without any absorber.

This figure shows the impact of bending vibration for 121.18 Hz frequency.

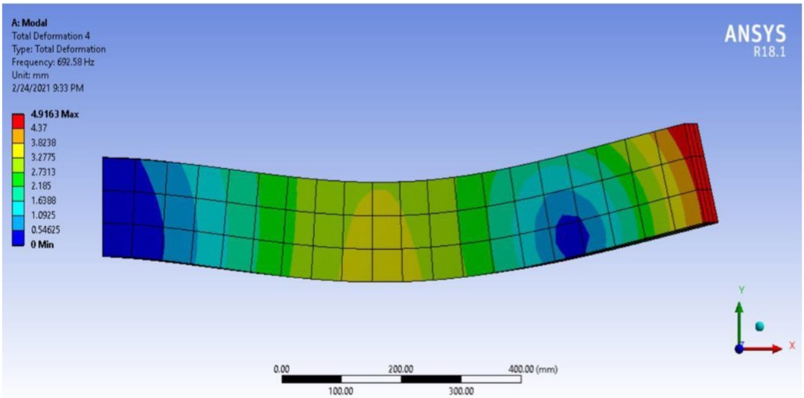


Figure 7:Total deformation due to bending vibration without any absorber. (Frequency: 692.58 Hz)

This figure shows the impact of bending vibration when the frequency is 692.58 Hz.

- ✓ **With absorber:**
  The same modal analysis was run for this protype with attaching a mass spring. And this simulation shows how the frequency of bending vibration decreases to 315.12 Hz because of the absorber.

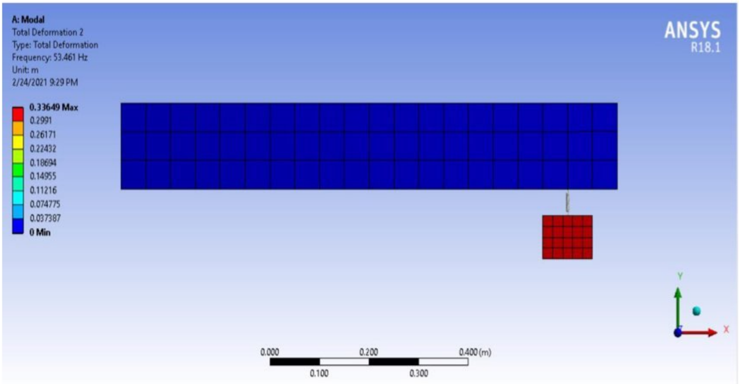


Figure 8:Modal analysis of protype beam with mass spring attached

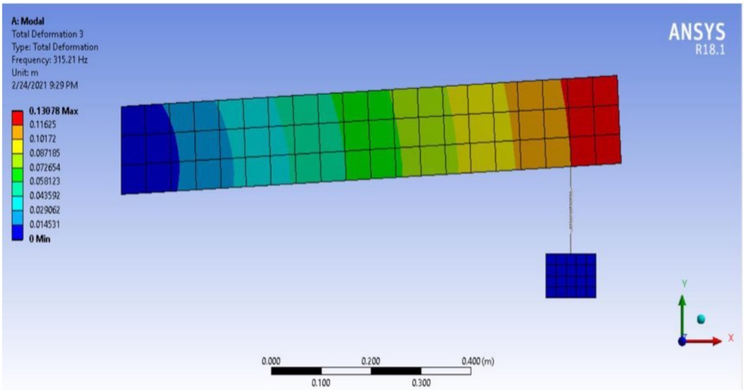


Figure 9:Total deformation due to bending vibration with an absorber. (Frequency: 315.21 Hz)

Mode	Frequency (Hz)
1.	**53.461**
2	**315.21**
3.	**3679.9**
4.	**3894.1**
5.	**4361.9**
6.	**5104.5**
7.	**5141.3**

Here the longitudinal stiffness of the spring is taken 100 N/m and longitudinal damping is taken 10N.s/m

The modal analysis results show clear reduction in frequency and in maximum deformation. Free beam gives maximum deformation of 4.913 mm and the beam with mass block gives maximum deformation of 0.33469 mm.

5.1.2 Harmonic Analysis:

Harmonic analysis has been run on the same prototype with keeping its properties unchanged and the meshing was also kept as it was for total deformation. Harmonic analysis was able to show the frequency response of this protype with or without the mass spring. An amount of 1000 N force was applied on both the beam at $a_0=0.5m$ in the negative Y-direction and this analysis provides two graphs including Amplitude vs Frequency graph and Phase angle vs frequency graph.

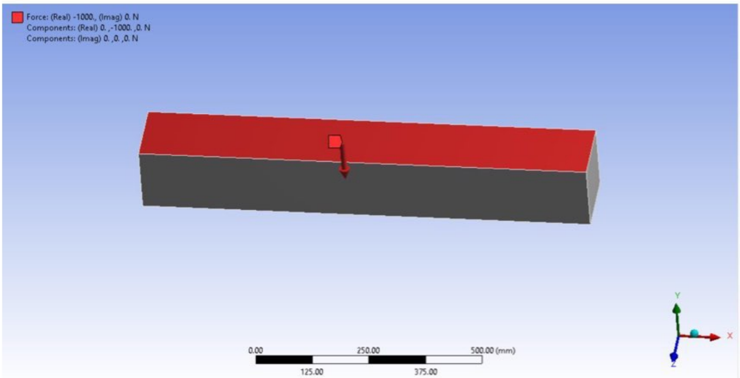


Figure 10:Force applied on free prototype beam

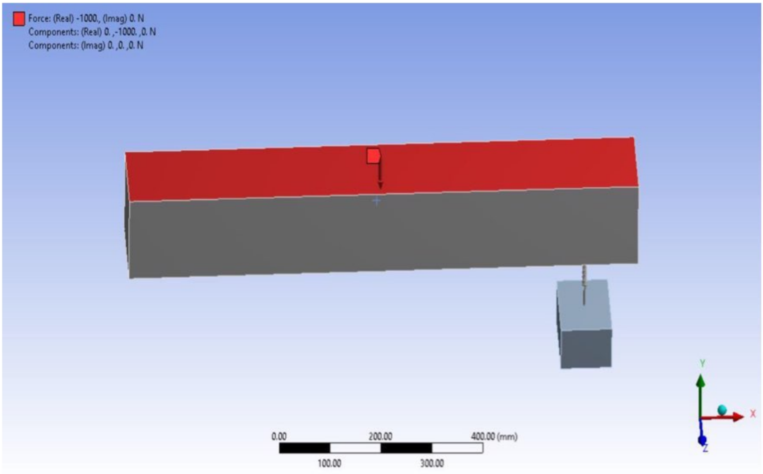
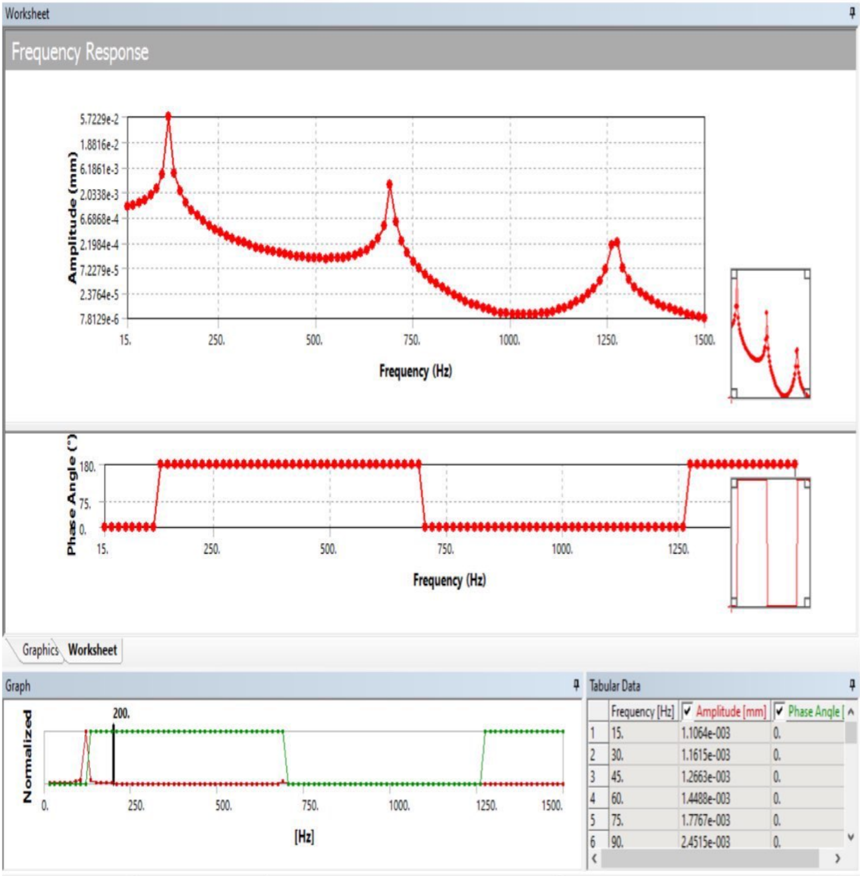


Figure 11:Force applied on prototype beam with mass block

5.1.3 Frequency response:

The frequency response helps to show where the resonance of a structure may happen and our result clearly shows how the response differs using an absorber. The frequency response for both the free prototype beam and the prototype beam with mass attached are shown below:



	Frequency [Hz]	Amplitude [mm]	Phase Angle [
1	15.	1.1064e-003	0.
2	30.	1.1615e-003	0.
3	45.	1.2663e-003	0.
4	60.	1.4488e-003	0.
5	75.	1.7767e-003	0.
6	90.	2.4515e-003	0.

Figure 12:Frequency response of 1m beam without mass spring.

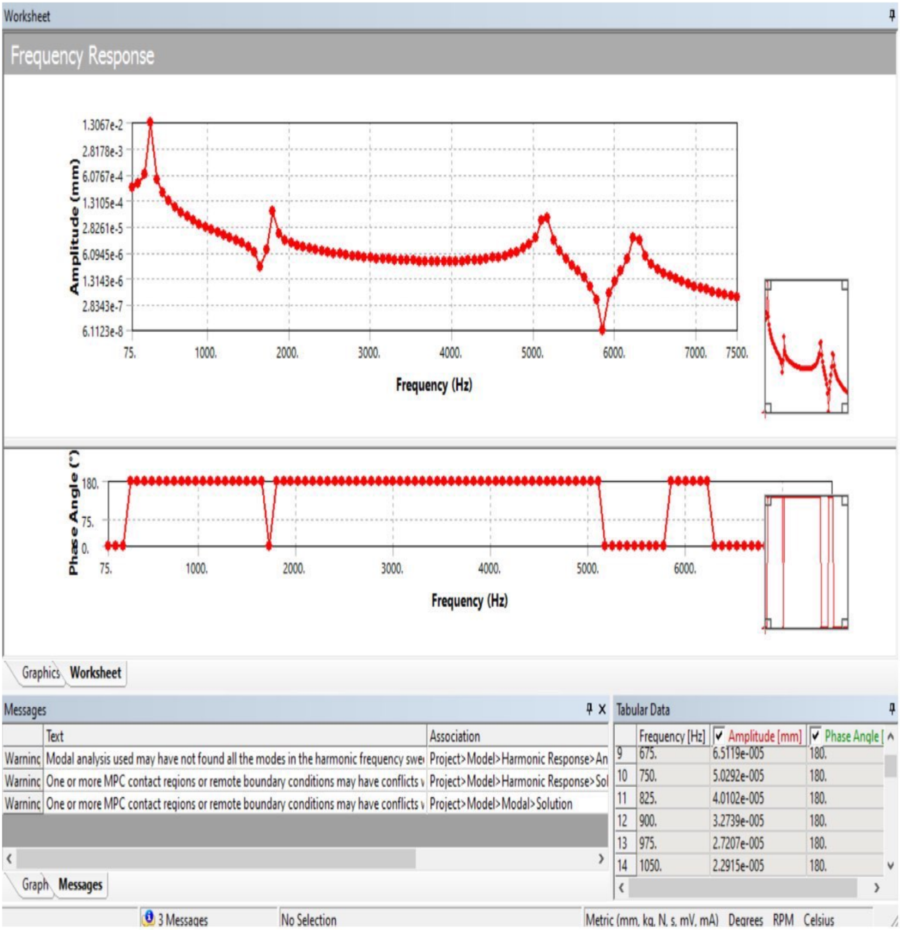


Figure 13:Frequency response of 1m beam with mass spring attached

From these two graphs it can be understood how adding the mass spring decreases the amplitude from .0057229mm to .0013067mm.

5.2 Discussion:

As shown in our result, the harmonic response shows how an absorber has impact on the vibration of a system. For this very first step the stiffness was taken to be 100N/s and damping was taken for 10N.s/m. In the experiment 1000N forced was applied at the length $X_{d=}$ of the beam and the comparison between the two above results is shown below:

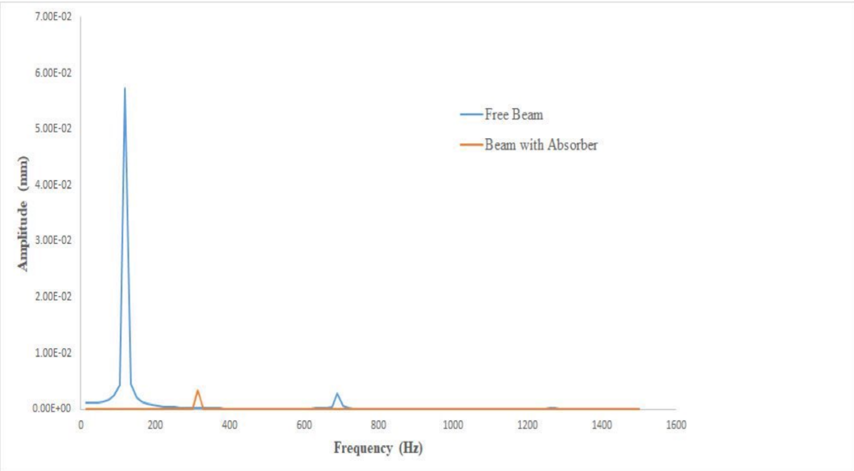


Figure 14:Amplitude vs frequency graph of the beam with and without an absorber.

Now the suitable longitudinal stiffness and longitudinal damping need to be found for the spring in order to suggest an absorber. For this model different values have been taken for the longitudinal stiffness and longitudinal damping for the spring to find the best result. For the stiffness of the absorber: 1000 N/m, 1000000N/m, 1000000000 N/m, 5000000 N/m and 2331000 N/m[27] these values were taken to find a suitable range.

The comparison of the results for these values are shown below:

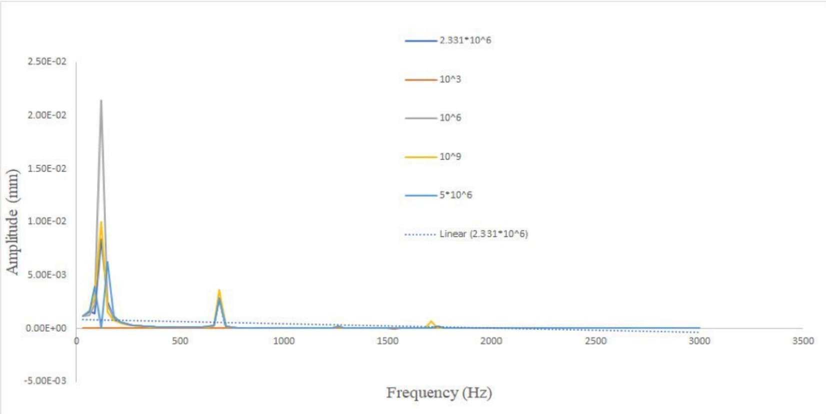


Figure 15:Amplitude vs frequency graph for different stiffness of the absorber.

Most suitable result was found when the stiffness of the absorber is 2331000 N/m[27].

The frequency response for this spring is shown:

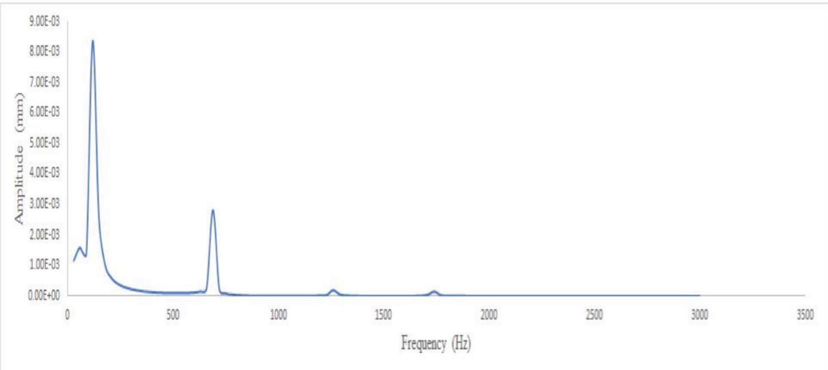


Figure 16:Amplitude vs frequency graph for the stiffness of 2331000 N/m.

The comparison of the frequency response for the beam without mass spring and the beam with a mass spring with longitudinal stiffness of 2331000 N/m showed how the spring mass reduces the frequency of the beam. The comparison is shown below:

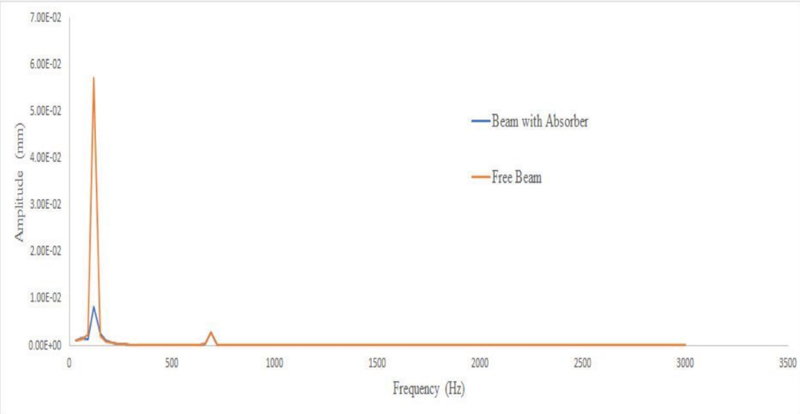


Figure 17:Amplitude vs frequency graph for free beam and absorber with 2331000 N/m stiffness

The absorber with 2331000 N/m is the most suitable one according the results, as it gives much smoother reduction in this study. Dampers of stiffness closer to this value can be used in practical cases. In the study of Miaomiao Li et. al[27] they calculated the stiffness of Silicon oil and rubber combined damper is 2331000 N/m.

CHAPTER 6:

CONCLUSION:

This study emphasized on how a very little amount of mass can significantly dissipate the vibration energy of a rotary drill-rig. The ratio of the absorber's mass with respect to the primary vibrating system's mass was kept between 0 to1, specifically .07. Adding this tiny mass as an absorber delivers the aim of this study which is not to significantly change the actual design of the rotary drill-rig. This study is acted on a very simple prototype of an actual drill-rig to easily understand the behavior of the linear absorber .Also some properties has been pre-designed while experimenting this study computationally to find out a suitable range of stiffness and damping and simulation based result shows how the amplitude of the beam's bending vibration decreases with adding the mass spring. This study also shows the behavior of the absorber changes where 5 different values were used. Absorber with the stiffness of 2331000 N/m is proven to be better at reducing the vibration and this suggested value was taken from a previous work of Miaomiao Li et. al[27] which brings the idea of silicon-rubber combined damper. Results of this study can be useful for further experiments of the design of an actual damper.

Nomenclature:

M= Mass

L= Length

C1= Damping

K= Linear stiffness

C2= Viscous damping

E= Young’s modulus

I= Moment of inertia

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