MAGNETIC AND ELASTOMERIC DAMPING EFFECTS ON THE VIBRATION AMPLITUDE OF A VIBRATION ISOLATION SYSTEM

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ABSTRACT

This technical paper describes the design and experimental process for a single-stage multi-degree-of-freedom vibration isolation system for a scanning tunneling microscope. The mathematical model used for the mass-spring system analysis will be presented, along with various design iterations for the mass-spring system and the frame. Based on the mathematical model, a MATLAB script is developed to study hundreds of springs available online so that a spring that satisfies important specifications can be chosen. Finite Element software is used to investigate the natural frequencies of the frames, which will assist the frame's design selection. Throughout the design process, magnetic and elastomeric damping methods are actively considered to simplify the design due to tight constrain of the vacuum chamber.

1. INTRODUCTION

In the early 1980s, Binnig, Rohrer, Gerber, and Wiebel successfully built a scanning tunneling microscope (STM) in an IBM Research Laboratory in Zurich, Switzerland[1]. The STM uses the quantum tunneling effect to study the surface of materials on atomic scale[2]. When the two conducting electrodes are in very close proximity but not actually in physical contact, the tunneling effect of electrons occurs due to the wave-particle duality of electrons. Since the quantum tunneling effect decays exponentially with the increasing distance between the scanning tip and the conducting material's surface, the distance between two electrodes must be kept within a few ångstroms. Therefore, the device is prone to noises due to vibration from the surroundings, so a vibration isolation system is necessary to allow the microscope to function properly and produce images of high resolution. Passive vibration isolation is a commonly used technique due to its low cost and simpler design compared to active vibration isolation, especially when the entire vibration isolation system (VIS) stays inside an Ultra High Vacuum (UHV) environment. Multi-stage VIS have been used previously, but this design is usually too complex and not cost-effective. This research aims to successfully build a multiple-degree-of-freedom single-stage passive VIS for a STM, which centers on a circular copper stage suspended by a combination of six springs. Section 2 will discuss support excitation model used to select springs and weight of the system's main stage as well as the experiment process to validate the design of mass-spring system. This section will also discuss the design of the VIS frame where the system will be mounted on before putting into the vacuum chamber. Section 3 will present the experimental data for testing of the mass-spring system as well as the simulation results using Finite Element Method. Section 4 will present the conclusions for the current design and discuss future steps in including magnetic and elastomeric damping.

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2. DESIGN & EXPERIMENTATION

The design process starts with studying previous designs of VIS. Figure 1 below shows a two-stage VIS. This design uses vertical springs to suspend the main stage where the microscope will be attached. An issue with this design is that it does not provide torsional stiffness to the system, which will make the vibration isolation less effective on the main stage.

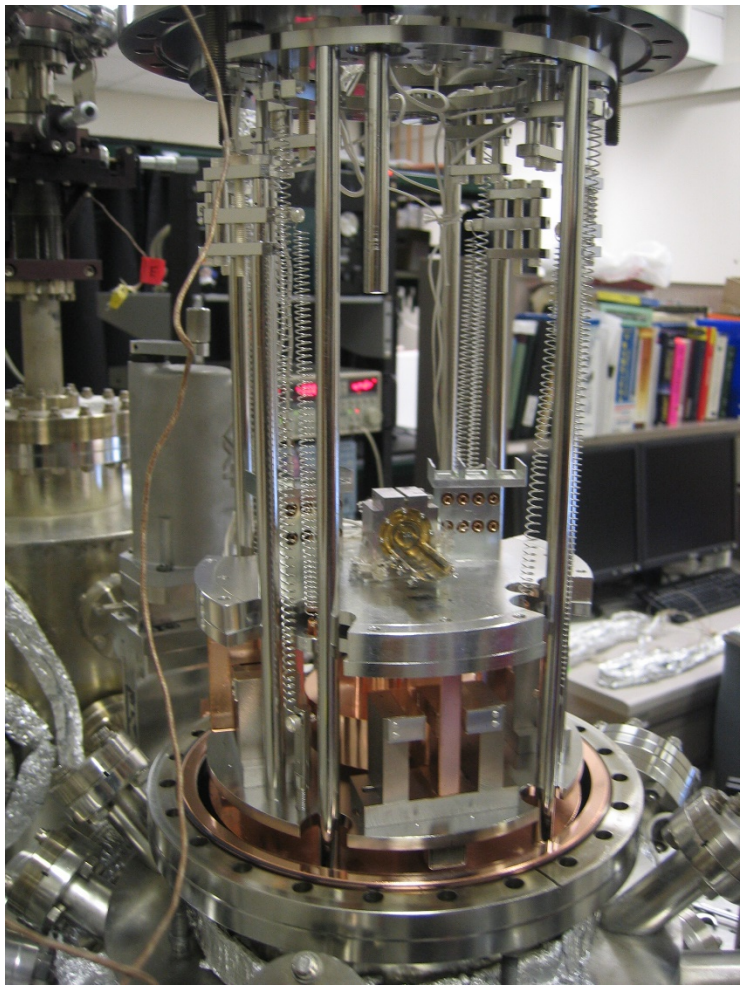


Figure 1. A two-stage VIS

There are two methods to add torsional stiffness to the system: adding a torsional spring or changing the configuration of the springs. While adding a torsional spring seems feasible, it adds more complexity into the design in terms of where and how the mounting structure for the torsional spring will be. In addition, since the system will be put into an UHV environment, most torsional springs on the market are incompatible because they are made from music wire steel, which is a restricted material to be used in vacuum environment[3]. A more effective method is to change the configuration from vertical springs to angled springs. Figure 2 indicates a section view of the main stage and two springs connected to the stage with an angle. With this design, a part of the spring stiffness will be used for the vertical mode while the rest of the stiffness will be used when the stage is twisted by an external torsional source.

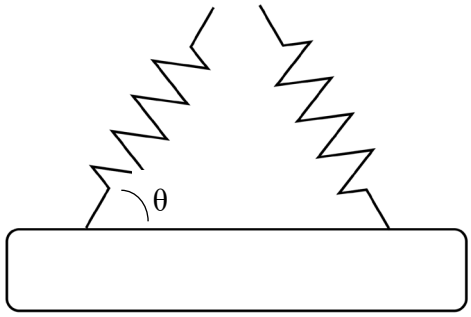


Figure 2. Configuration for 2 adjacent angled springs forming angle of θ above the main stage

2.1 VIBRATION DESIGN

Since the springs in the passive VIS are the softest compared to rigid stainless-steel vacuum chamber or aluminum 6061 frame, the mass-spring system has the lowest natural frequencies in the whole system. Hence, the analysis should start from the mass-spring system using support excitation model. Figure 3 depicts the mathematical support excitation model used to derive the equation of motion for the mass-spring system where $y, \dot{y}, \ddot{y}$ are input displacement, velocity, and acceleration respectively, and $x, \dot{x}, \ddot{x}$ are displacement, velocity, and acceleration of the main stage respectively.

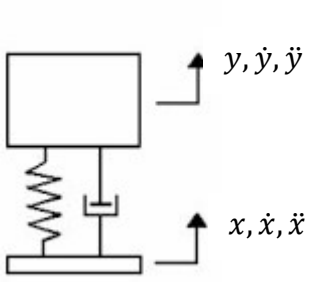


Figure 3. Support excitation model used for mass-spring system

Once the model is developed, the equation of motion can be derived as follows

$$M_{eq}\ddot{x} + B_{eq}\dot{x} + K_{eq}x = f(y, \dot{y}, \ddot{y}) \quad [1]$$

To achieve vibration isolation, the ratio of input frequency and natural frequency of the mass-spring system must be greater than $\sqrt{2}$. For a system with six angled springs, the natural frequencies can be derived as below, where k is the stiffness of one spring, and m_{stage} is the mass of the main stage.

$$\omega_n = \sqrt{\frac{K_{eq}}{M_{eq}}} = \sqrt{\frac{6 \times k \times sin\theta}{m_{stage}}} \quad [2]$$

2.1.1 SPRING SELECTION

From equation [2] and the criteria for vibration isolation, a MATLAB script is used to plot the theoretical lines of weight per spring versus spring stiffness needed to achieve vibration isolation for input frequencies of 1 Hz, 1.5 Hz, 2 Hz, 2.5 Hz, and 3 Hz. Data from two spring manufacturers are also plotted to help selecting springs. For these plots, if the spring data point appears above a theoretical solid line, vibration isolation can be achieved for this lowest frequency. Figure 4 shows the data from Century Spring website while Figure 5 shows the data from The Spring Store website. Since the manufacture uses English parameters for their springs, Figure 4 and 5 keep these units to show consistency. An important criterion for spring selection is that the total deflection must be within 25.8-28 cm (10-11 inches) due to the constraint in height of vacuum chamber, as well as the desire to be able to see the microscope working through a small window flange of 15 cm (6 inches) that is located 26.5 cm (10.5 inches) below the top flange.

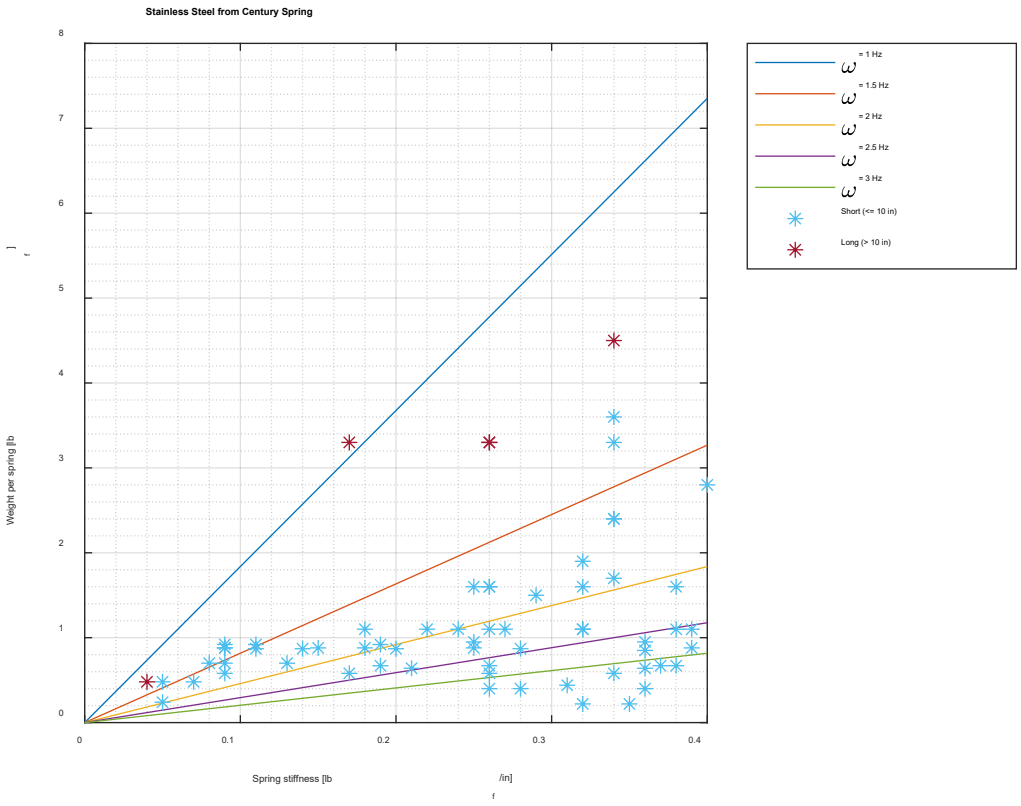


Figure 4. Long and short stainless-steel spring data from Century Spring plotted against the theoretical lines of frequencies indicating which isolation frequency could be achieved

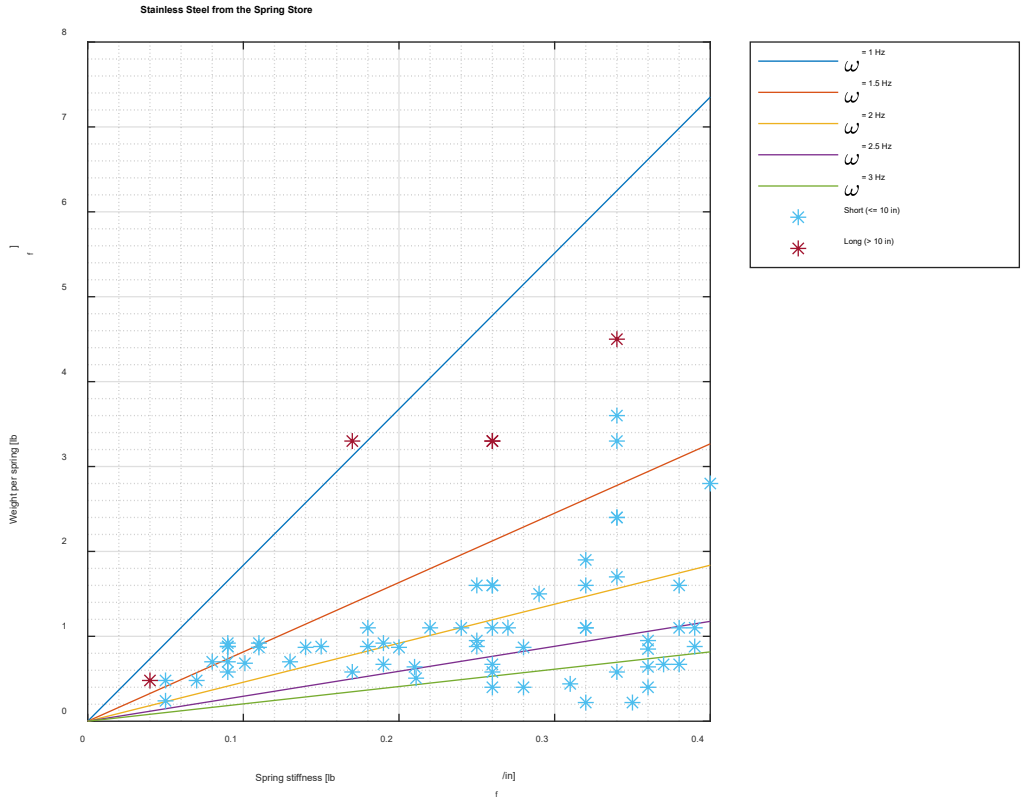


Figure 5. Long and short stainless-steel spring data from The Spring Store plotted against the theoretical lines of frequencies indicating which isolation frequency could be achieved

It is important to note that if the springs are loaded with higher loads than their max allowable load, we risk bending the hook or subject the springs to the non-linear region that Hooke's Law for springs might not work. Figure 4 and 5 show that by selecting long springs and underloading them with the desired weight, we can keep the springs operating in linear region and keep the total deflection of the main stage to be between 25.8 and 28 cm (10-11 inches) as discussed above. The selected spring is from Century Spring with the specifications shown in Table 1.

Table 1: Parameters of the selected spring

Material	Stainless-steel
Part Number	80657S
Initial Tension (N)	1.33
Length (mm)	127
Wire Diameter (mm)	0.94
Outer Diameter (mm)	12.70

2.1.2 MAIN STAGE SELECTION

With the selected spring in mind, the calculated total weight of the main stage needs to be 4 kg to achieve vibration isolation for the lowest input frequency of 1.5 Hz. Due to material restriction to be used in vacuum chamber, there are two materials that the main stage can be made of: copper or aluminum. Using aluminum 6061 like previous designs seems like a good option initially, but the volume is too large to fit into the current small vacuum chamber while leaving room for magnetic damping accessories. Moreover, using aluminum main stage still requires copper componenets to facilitate magnetic damping, which is one of a few effective methods of damping used in UHV. Figure 6 shows a typical design for magnetic damping where a copper plate is mounted vertically on the aluminum main stage with two magnets are positioned on both sides.

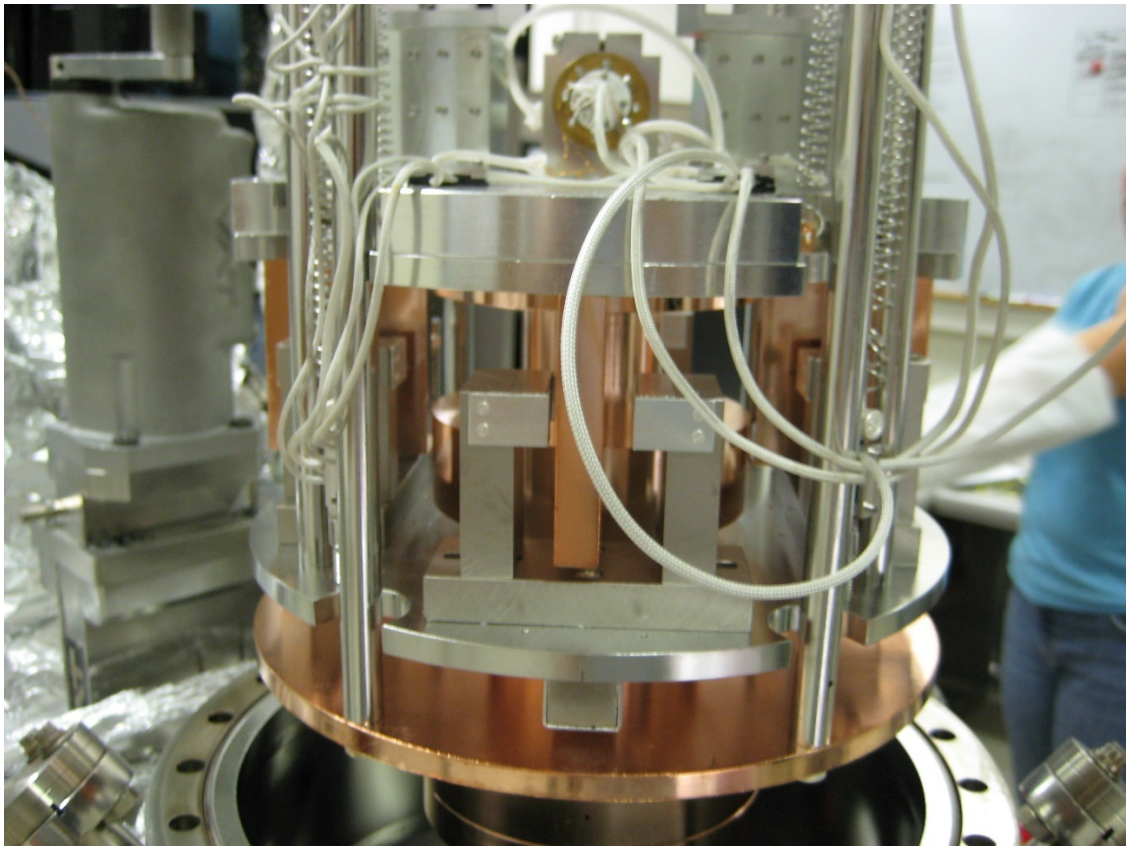


Figure 6. A previous design of magnetic damping using copper plate hanging down from main stage

This design also poses a challenge for damping the rotational mode of vibration because the net change in magnetic flux through copper is minimal. Eventually, to simplify the design, to save space, and to increase magnetic damping effectiveness for all modes of vibration, a copper stage is selected. One of the challenges in using copper material is that it is much harder to machine copper because copper is much softer than aluminum. In addition, copper costs more than aluminum.

2.2 FRAME DESIGN

Since the natural frequencies of the mass-spring system are low and the input frequencies from human activities and the environment are less than 100 Hz[4], the frame for the system needs to be rigid so that its natural frequencies are outside of the working range of the mass-spring system. Figure 7 shows different designs for the frame. The natural frequencies for theses frame will be studied through Finite Element Software to determine the best option.



Figure 7. Frame 1 with vertical circular rods (left), Frame 2 with aluminum L brackets (middle), and Frame 3 with hollow tube and cutout windows (right)

2.3 TESTING SETUP

Before purchasing materials and machining the stage and the frame, a testing structure is set up using plywood material. Using available springs with stiffness of approximately 0.019 N/mm, we can verify the mathematical model by using Random Noise vibration test on a shaker and recording data for different weights of main stage. Figure 8 shows the plywood testing structure with added weights to simulate the target weight to achieve vibration isolation for 1.5 Hz lowest input frequency.

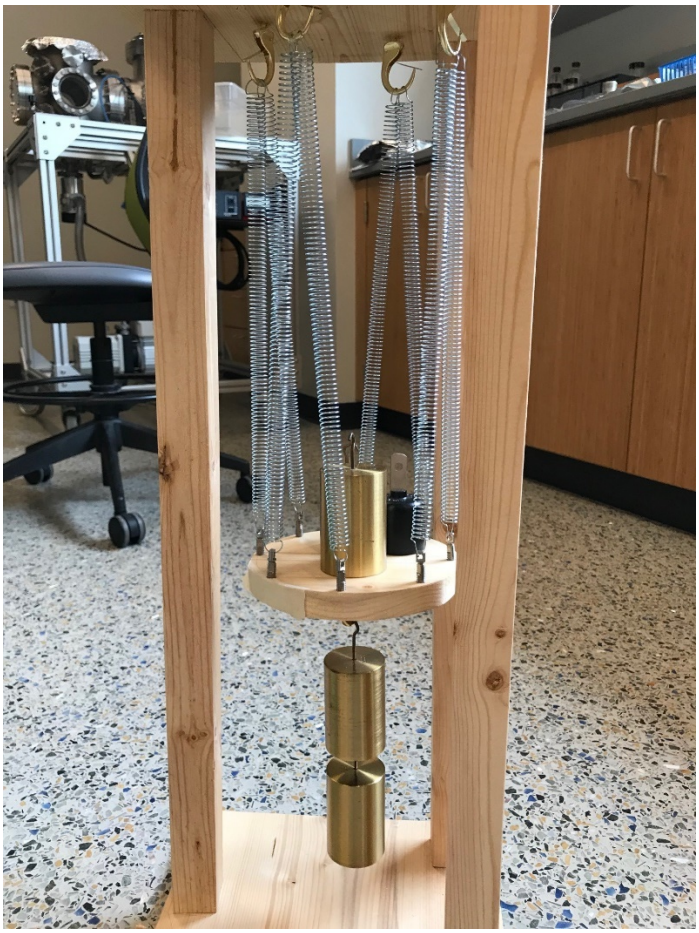


Figure 8. A testing structure made of wood and springs
with stiffness of k = 0.019 N/mm

3. RESULTS

The results are separated in two sections: the experimental data to validate the mass-spring system design and the Finite Element Analysis simulation results of three frame designs.

3.1 FREQUENCY RESPONSE OF TESTING STAGE

To accurately demonstrate the effectiveness of the mass-spring system in attenuating input frequencies from the surroundings, the Random Noise test is truncated to intervals of 0.5 Hz, stretching from 0.5 Hz to 10 Hz. The data are stored in a floppy disk in a Spectrum Analyzer, converted to MATLAB data files, and presented in Figure 9 below.

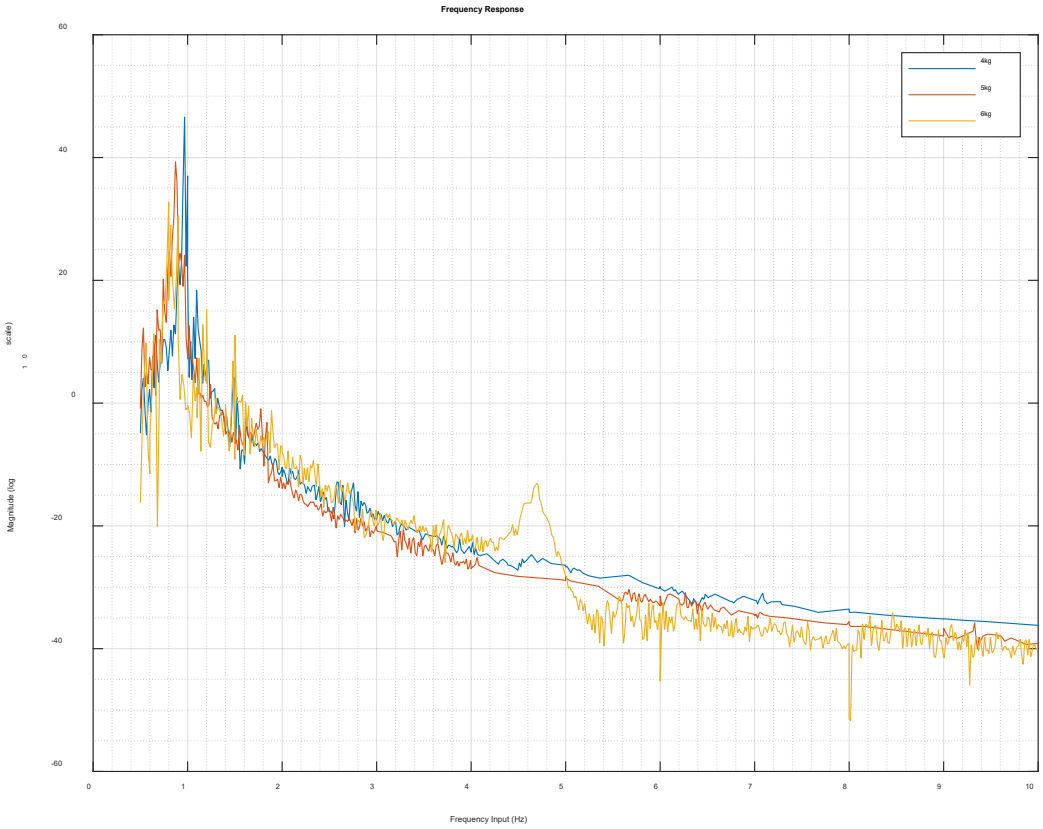


Figure 9. Frequency response of the testing structure with 4 kg, 5 kg, and 6 kg stage

Figure 9 shows that for input frequencies of 1.5 Hz and above, the system transmissibility ratio is less than 0 on a log scale, which indicates vibration isolation is being achieved. The irregularities in testing with 6 kg stage is because the plywood frame is too soft and flimsy to support such heavy stage.

3.2 NATURAL FREQUENCIES OF FRAME ITERATIONS

Finite Element software shows the first four mode shape frequencies for each frame design, which are shown in Table 2. Overall, frame 3 with hollow tube design and cutout windows is the most promising design because all the theoretical natural frequencies of the frame are significantly outside of the working range of frequencies of the mass-spring system.

Table 2. Frequency of the first four mode shapes for three frame designs

	Mode 1 (Hz)	Mode 2 (Hz)	Mode 3 (Hz)	Mode 4 (Hz)
Frame 1	74.9	74.9	122.1	345.3
Frame 2	49.6	49.6	98.0	313.1
Frame 3	478.9	650.6	1,135.4	1,244.8

4. CONCLUSIONS

The mathematical model for the mass-spring system is derived and validated. In addition, FEA models for the frames are also studied. With the main stage is made of copper, the magnetic damping accessories are reduced so that the overall design becomes simpler and less bulky. Moreover, the copper stage will facilitate magnetic damping in all modes of vibrations (vertical, rotational, and wobbly) more effectively because the change in magnetic flux in all modes of vibration is maximized. Elastomeric damping will be added as cushions between the vacuum chamber and the hollow tube frame to absorb the vibrations before they enter the mass-spring system. Finally, soft elastomeric materials will be added in joints and connectors within the system to further reduce the amplitude of input vibrations. Final tests will be executed once the final assembly is finished to further study the magnetic damping effect. Additionally, these tests will also serve to confirm whether more elastomeric damping should be used or which locations in the structure the elastomeric damping should be added.

5. REFERENCES

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