ME450 W23 FINAL REPORT

Team 13: Large Impeller Reflow Oven Vibration Modeling

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31 March 2023

ABSTRACT

Heller Industries is the market leader in convection reflow ovens. In an attempt to improve the thermal efficiency and overall capability of their reflow ovens, Heller Industries increased impeller diameter from 8 to 10 inches, which resulted in unwanted vibrations during normal operation. We aim to develop a mathematical and Finite Element Analysis model that accurately predicts the vibration behavior, identify and quantify system parameters contributing to the vibrations, validate the feasibility and robustness of any proposal through FEA studies, and provide solution recommendations based off of our analysis.

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INTRODUCTION

The sponsor for our Mechanical Engineering capstone design project is Heller Industries, the market leader in manufacturing of convection reflow ovens and other curing oven technology [1]. Reflow ovens are instrumental in the mass manufacturing of printed circuit boards (PCBs) and other electronic devices. As the name suggests, they operate on the concept of convection - using a motorized impeller to pass cool air over a set of resistive coils and uniformly heat the PCB or electronic component passing underneath. This convective heating is often sufficient enough to melt solder and other joining agents to assemble the electronic components together. Figure 1 below shows an image of the Heller Industries *1936 MK 7* reflow oven, Figure 2 shows a high level diagram of how a typical convection reflow oven works, while Figure 3 shows a simplified cross-sectional diagram of an impeller heating module, the component that we are concerning ourselves with.

Figure 1. Heller Industries' *1936 MK 7* Convection Reflow Oven

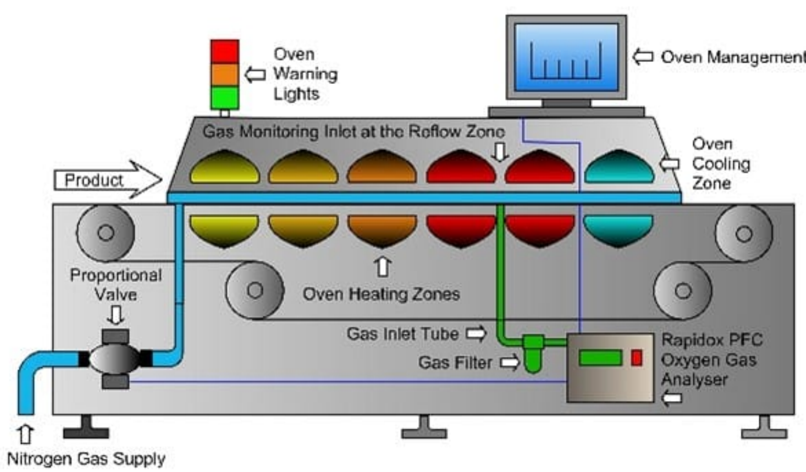

Figure 2. Diagram of a typical convection reflow oven. A PCB or other electronic component will enter the oven from the left and will uniformly heat up as it travels from left to right through the oven via conveyor belt. Once the component is sufficiently heated, "reflow" occurs, where the joining agent (i.e solder) is melted and the various components of the PCB are fused together. The PCB

then cools down before exiting the oven from the right. Convection reflow ovens typically isolate the electronic components with an inert gas to prevent environmental corrosion from occurring during reflow [16].

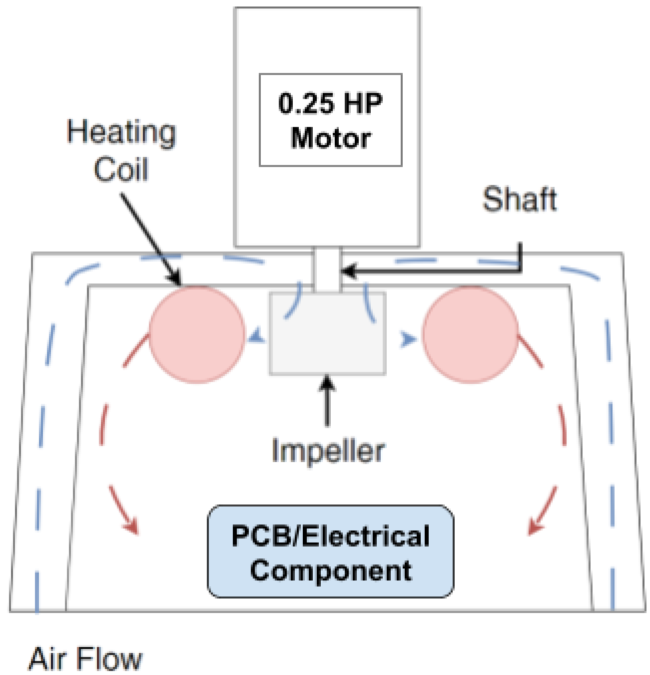

Figure 3. Cross-sectional diagram of a Heller Industries convection reflow heating module. The module consists of two concentric sheet metal housings. Using a motorized impeller, cool air is pulled into the module and passed over resistive heating coils (red circles in diagram) to heat the PCB/electrical component passing below. This heated air can then be recycled through the system again to improve thermal efficiency.

In order to improve the efficiency and capability of their convection reflow ovens, Heller Industries is exploring an increase in the diameter of the impeller used in the heating module. The influence of increasing the diameter of the rotating impeller results in improved convective airflow, which enhances the thermal efficiency of the oven and subsequently its overall capabilities and usefulness for the end-user [6,16]. However, upon switching from an 8 inch to a 10 inch diameter impeller Heller observed and recorded undesired vibrations in excess of 1 G. These vibrations impact the quality of soldering, as well as the overall production of Printed Circuit Boards (PCBs) and other electronic components. We have been tasked with identifying the source of these vibrations and recommending a solution for them that constrains the vibration to less than 0.2 G at impeller speeds of up to 3400 RPM, for motor controller speeds of 60 Hz. Solving this problem with Heller Industries will be critical to their effectiveness in developing a better end-product for their customers.

The scope of our design problem focuses on vibrations resulting from the increased diameter of the impeller. While other reflow oven manufacturers may have experienced this problem in the past, availability of pertinent data makes it difficult to confirm this, as well as exploration into any potential solution the competitors may have adopted. In this regard, we expanded the scope of our problem to rotating shaft vibrations/dynamics and examined other industries for any intuitive solutions that may have been developed to stabilize rotating shafts and masses while operating at high speeds.

The first objective of this project is to characterize the source of the rotational vibrations within the module. To accomplish this, we aim to characterize important system parameters contributing to the vibrations. Next, we aim to collect empirical data using our testing setup, and develop mathematical and Finite Element Analysis models that accurately predict the vibration behavior. Finally, we will use our developed models to propose and validate solutions that could be implemented by Heller Industries. Successful completion of these steps will allow us to fulfill all the metrics and requirements set forth by Heller Industries and ensure project completion.

Stakeholders

The stakeholders involved with this project have been categorized into three groups depending on the degree at which they are impacted by the problem. These stakeholders are shown in the figure below.

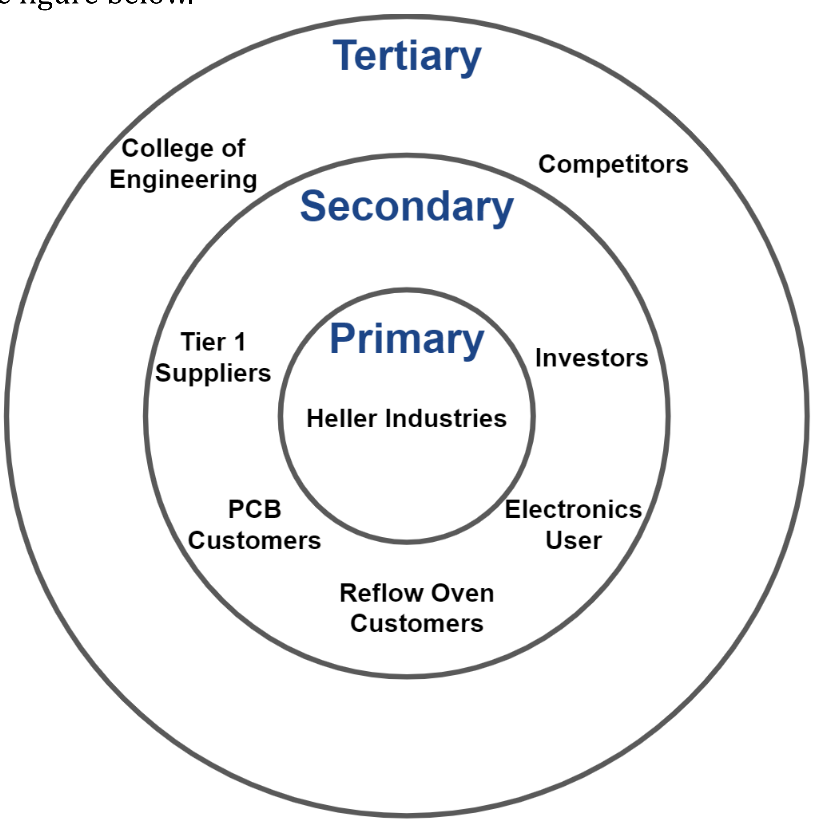

Figure 4. Map encompassing all stakeholders involved in the project.

We first defined the Primary Stakeholders. These are groups whose lives or work are directly impacted by the problem and/or the development of a solution [13]. The Primary Stakeholder for this project is Heller Industries. Expanding out the scope of the problem we defined our Secondary Stakeholders. These are stakeholders who are part of the problem context but may not experience the problem themselves and/or may not be directly impacted by a solution [15]. The Secondary Stakeholders for this project include customers of Heller's convection reflow ovens (usually to produce PCB and other electrical components), Heller's tier one suppliers (companies supplying the parts for their convection reflow ovens), Heller's investors, electronics manufacturers who are purchasing components that were manufactured in Heller's ovens, and customers of the aforementioned electronics manufacturers [8]. Finally, in the broadest scope of the project we identified the Tertiary Stakeholders. These are groups who are outside of the immediate problem context but may have the ability to influence the success or failure of a potential solution [5]. The Tertiary Stakeholders for this project include The University of Michigan's College of Engineering, Regulatory and Standards Organizations, and Heller's competitors in the convection reflow oven market [8].

Most of the stakeholders listed would benefit from the development of a solution to Heller's vibration problem. A reduction in vibration would result in most of the stakeholders making more money due to the performance improvements that a 10 inch impeller introduces. The only stakeholder that would be significantly negatively affected would be Heller's competitors as Heller would have a stronger competitive advantage than before.

Design Process

With respect to the design process, a majority of our efforts this semester have focused around engineering analysis. Our solution and end-of-term deliverable for Heller Industries is intended to be a characterization of the parameters contributing to system vibrations. As such, we want to ensure that the three methods we are using to characterize the parameters - mathematical model generation, finite element analysis (FEA) studies, and empirical data collection - are all respectively aligned with one another in their findings. Doing our due diligence in cross-validating our various modeling and data collection techniques will ensure that any future prototyping potential solutions are predictable, and that a final solution can be verified and validated.

Intellectual Property

Intellectual property concerns did not play a significant role in our project this term. Heller has made it clear that while some of the information regarding module mounting within the reflow oven is confidential, most of the components we are interfacing with can be bought off-the-shelf and therefore do not pose a risk of an intellectual property breach. Our project did not involve analyzing the arrangement of the modules within the oven, nor any of its other internal structure. As such, Heller Industries has not required us to sign any intellectual property protection agreements.

Information Sources

Since this project primarily focuses on modeling and analysis techniques, a lot of the research we conducted reflected this. Rather than researching specific examples of solutions to vibrating impellers, we researched the different types of math models, modeling softwares, and data collection methods we could utilize for our efforts. We consulted with professors at the University of Michigan, while also holding regular meetings with representatives at Heller Industries to get feedback and guidance on our progress. These resources proved invaluable in our understanding of the project, and allowed us to ultimately characterize the system so that changes could be made to reduce vibration magnitudes. The evolution and growth of our thought processes can hopefully be seen below in the remainder of this document.

REQUIREMENTS AND ENGINEERING SPECIFICATIONS

After discussions with Heller and our instructor, we have defined four user requirements and their corresponding engineering specifications for this project. A table detailing the project's requirements and engineering specifications is shown below. The requirements and specifications have been updated to accommodate for the model based approach to the problem.

Table 1: Prioritized Requirements and Specifications

User Requirement	Engineering Specification
Validation between models and physical system	Validate that theoretical/FEA models are within 20% of empirical data.
Cross Model Analysis	Develop a theoretical model and FEA model within 20% of each other
Model Inputs	Models incorporates: - Forcing frequency of up to 2900 RPM - Various impeller sizes - Variable material properties - Empirical data: acceleration magnitude [G] - Provided CAD geometry
Model Outputs	Model provides system outputs: - Acceleration magnitudes in excess of 0.2 G - Resonance frequency in Hz - Overall stiffness [N/m] - Damping ratio

Firstly, we are aiming to quantify the accuracy of our models by ensuring that the predicted key parameters listed above are within 20% of each other. That is, the values predicted by the mathematical model and computational model are less than 20% apart. This will provide us with confidence that both models are adequately predicting the same parameters. Our models should also be within 20% of the measured vibration data taken via accelerometer by Heller, we plan to cross reference this with experiments of our own using our on-site impeller module setup. Previously, our accuracy requirement was 10% in an attempt to create a more refined and representative model. However, after internal conversation we have decided to shift our accuracy requirement back to 20%. There are a couple of reasons for doing this. This first stems from the fact that we are trying to predict relatively low magnitudes of acceleration. This means that even small variations between the model and empirical data would yield a large percent difference. Additionally, we believe that there is an additional fluid dynamics aspect to this problem at play. After talks with Heller we believe that this could be introduced by the larger impeller being closer to the side walls. Due to this we believe that it is unreasonable for our mathematical model to perfectly match empirical data because the mathematical model is only predicting vibration kinematics.

Finally, our model should be able to incorporate Heller's operating conditions as system inputs/outputs and predict useful vibration parameters as system outputs. More specifically, the model must encompass a forcing frequency of up to 2900 RPM. It must also allow for varied impeller sizes and material properties for inputs. Outputs must include (but are not limited to) predicting acceleration magnitudes in excess of 0.2 G, resonance frequency in Hz, overall stiffness, and the damping ratio. Originally we also included critical shaft speed as an output parameter. We have since decided to remove this as we were only concerned with the forcing and resonant frequencies of the system.

Thinking bigger picture, we also are aiming to propose recommendations to reduce vibrations using the results of our mathematical/computational modeling. As such, there are metrics which we must keep in mind in the context of our modeling process. The goal, as defined by Heller, is to reduce vibration magnitude to less than 0.2 G while operating at up to 3400 RPM, with the motor controller operating at up to 60 HZ. These values are modeling parameters which we must include in our analysis.

Our recommendations should ideally be durable and last over 10 years without failure. Lastly, our recommendations should also be temperature resistant. The module will operate between 50 and 350°C. We aim to suggest changes which can operate effectively across this entire temperature range.

Heller sets their own internal vibrations standards, which they have provided to us, and has clarified that these are the only relevant standards that we need to adhere to in the scope of this project. These standards include the magnitude of vibration being less than 0.2 G at 2900 RPM operating speed.

Engineering Standards

Heller Industries executives communicated to us that the vibration standards set in this project were internally defined. They chose a level of vibration of less than 0.2 G at an impeller speed of 2900 RPM, which was found by internal experimentation. Heller Industries deemed that this level of vibration allowed for the best performance out of their convection reflow oven products. In the scope of this project, Heller Industries is not concerned with ASTM/ISO/ASME standards in regards to rotational vibration, rotor balancing, or electric motor balancing. We will remain in communication with Heller Industries regarding any changes in their internal standards, or potential inclusion of outside standards from engineering organizations.

DESIGN PROCESS

We are utilizing a stage-based, problem-oriented approach to our design process [18]. We have thoroughly explored the solution space by reviewing literature and other relevant information sources. We plan on using the knowledge accumulated from this review to propose a range of potential solutions to pursue. Additionally, our process has clearly defined stages which are mostly linear. Figure 1 below shows a diagram of our proposed design process.

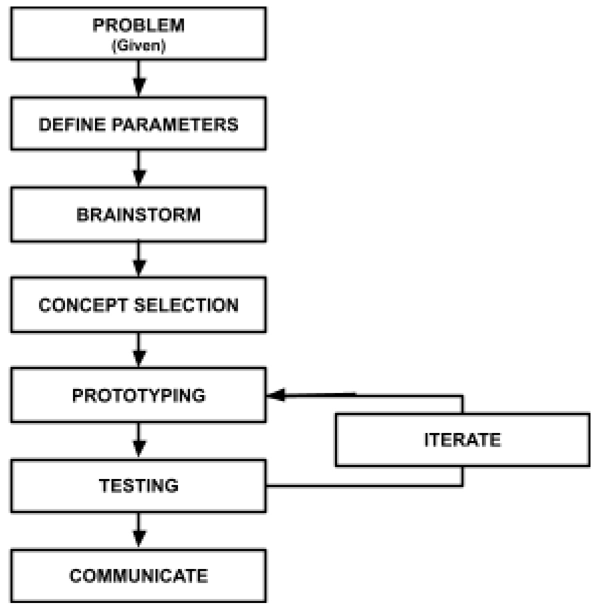

Figure 5: Diagram of the stage-based, problem-oriented design process that we will be utilizing during this project.

As seen in the diagram, our design process is set up in such a way that it allows us to iterate our prototype based off of the empirical data we are able to collect during testing. This feedback will theoretically continue until we run into time constraints or converge on the best solution given the parameters we had defined earlier in the design process. Non-linearity of our design process enables greater flexibility and creativity in the face of unforeseen challenges. Our research on design processes determined that the stage-based, problem-oriented setup fit our specific challenge the best when compared to other

FINAL CONCEPT/FINAL DESIGN DESCRIPTION

Our revised alpha design is centered around characterizing various parameters and identifying the proper values for the impeller module. The design was generated and down selected using various pugh charts and thought maps. This process is further discussed in Appendix A. The alpha design centers around the 2 degree of freedom mathematical model, introduced in the Engineering Analysis section, and incorporates both Finite Element Analysis as well as empirical data to obtain the model parameters that we can utilize to correctly model the kinematics of the system. Using the motor spec sheet [32], CAD, and on site impeller module, we constructed a system for identifying all the system parameters. The cumulative design was derived from discussions with professors, forums, and Heller industries to determine the system was being accurately modeled and set up in a representative fashion. Figures 6 and 7 below outline how the mathematical and

CAD/Finite Element Analysis was constructed, while the empirical data approach is outlined in the Testing Plan section.

Revised Modeling Approach: The Alpha

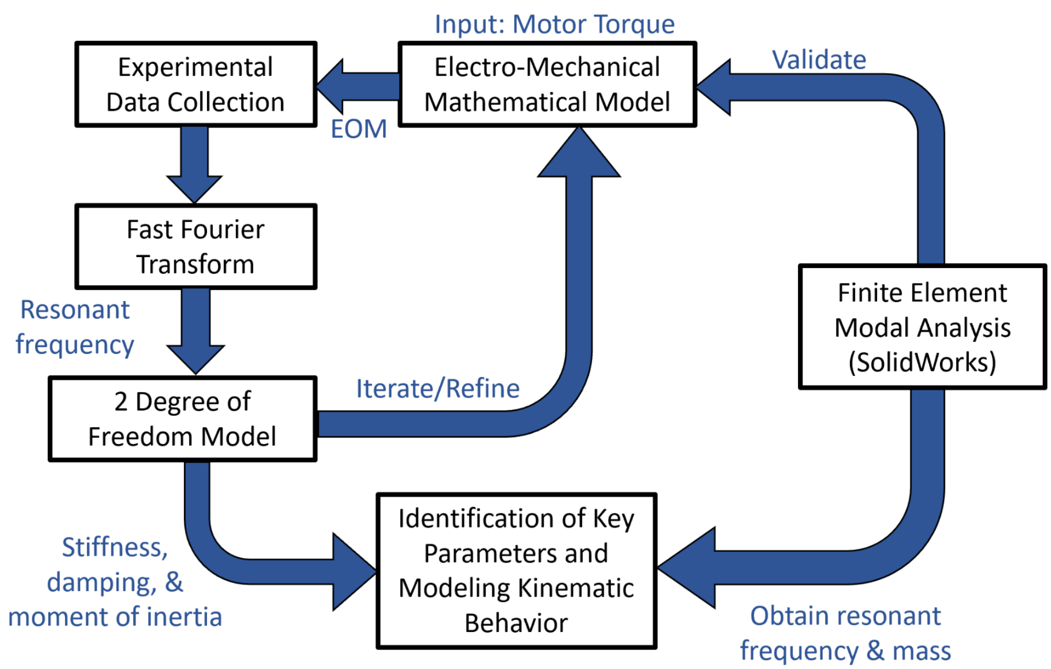

Figure 6: Alpha Design schematic visualizing the modeling and refinement of our mathematical and FEA models in order to identify the key modeling parameters.

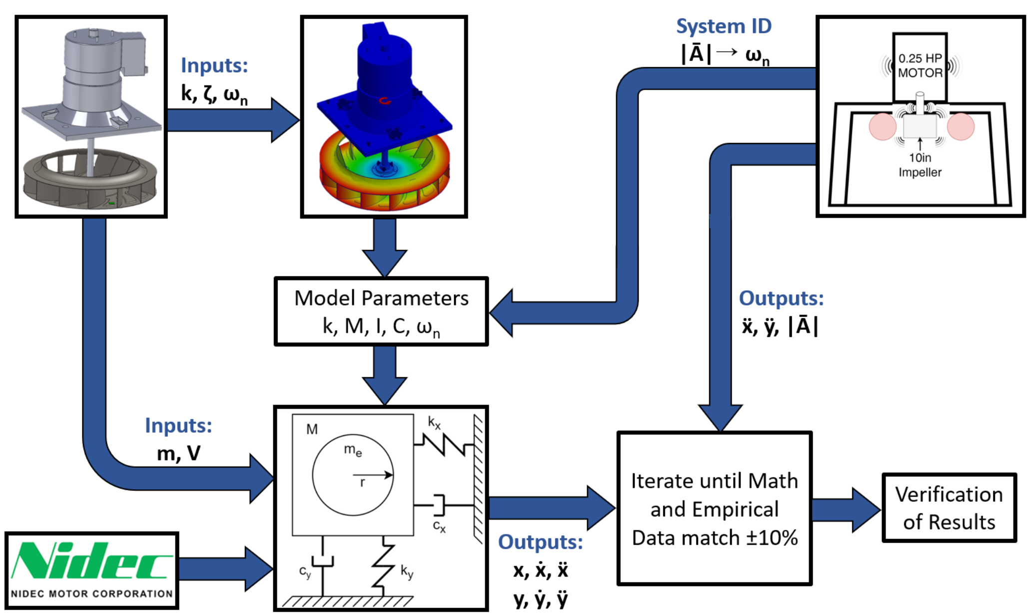

Figure 7: Alpha Design schematic for modeling and identification of parameters in Heller industries impeller module

Mathematical Model - In our mathematical model, a differential equation of motion was obtained to predict the acceleration, velocity and position of the system using Laplace transform. The inputs that are required into this model are the motor torque, mass of the system, stiffness, damping ratio, moment of inertia, volume of the system, motor torque, the forcing frequency, and the resonant frequency. As seen in Figure 7**,** motor torque and forcing frequency are obtained using the Nidec; mass, stiffness, damping, and volume are obtained from the CAD/FEA studies; acceleration and resonant frequency are obtained through empirical data collection. Further calculations and analysis can be seen in the Mathematical Model portion of the Engineering Analysis section.

CAD/Finite Element Analysis (FEA) - Using the CAD of the impeller module, we constructed a harmonic modal Finite Element Analysis model to obtain the stiffness and damping parameters. This gave us the key parameters along with a potential fundamental frequency in all three axes (x,y,z) that allowed us to compare/validate with the mathematical model down the road. Further setup and parameterization can be seen in the FEA/CAD portion of the engineering analysis section.

ENGINEERING ANALYSIS

Math Model

Our final mathematical model was developed and refined over the course of the semester using a simplified 2-DOF mass-spring-damper system as its foundation. The system is driven by an eccentric mass a set distance, e, away from the center of the impeller. A schematic depicting the simplified system can be seen in Figure 8 below.

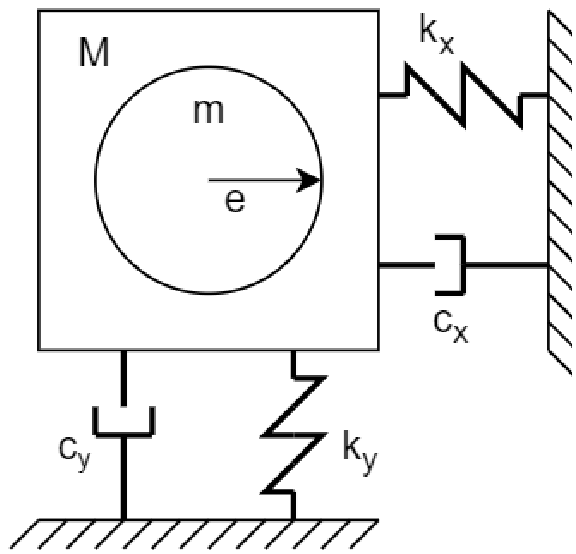

Figure 8. Two degrees of freedom mass-spring-damper system used to simplify the model and derive the equations of motion.

Using this 2-DOF system we derived the resulting equations of motion (Equations 1-2) in both the x and y directions. We then performed an inverse laplace in order to solve the system for position as a function of time, amplitude as a function of forcing frequency, and phase shift of the vibrations. Equations 3, 4, and 5, respectively are the results of these calculations.

$$M\ddot{x} + C\dot{x} + kx = me\omega^2 sin(\omega t) \quad (1)[30]$$

$$M\ddot{y} + C\dot{y} + ky = me\omega^2 sin(\omega t) \quad (2)[30]$$

$$x = Ae^{-\xi\omega_n} sin(\omega_d t + \phi_1) + Xsin(\omega t - \phi) \quad (3)[30]$$

$$X = \frac{me}{M}\frac{(\frac{\omega}{\omega_n})^2}{\sqrt{(1-(\frac{\omega}{\omega_n})^2)^2+(2\xi\frac{\omega}{\omega_n})^2}} \quad (4)[30]$$

$$\phi = tan^{-1}(\frac{2\xi\frac{\omega}{\omega_n}}{1-(\frac{\omega}{\omega_n})}) \quad (5)\ [30]$$

These equations use the system mass, M, eccentric mass, m_e, eccentricity, e, damping coefficient, C, stiffness coefficient, k, damping ratio, ζ, and the resonant frequency, w_n, as inputs in order to determine system displacement and acceleration. These input parameters were estimated/calculated in a variety of ways, described in the FEA/CAD and Verification/Validation sections. We list them here, as they would be plugged into our mathematical model, in Table 4.

Table 2. Input values used in the final mathematical model. The final values are compared to the initial values used.

Parameter	Final Value	Initial Value
System Mass, M	6.02 kg	1.118 kg
Eccentric Mass, m_e	0.00075 kg	0.001 kg
Eccentricity, e	0.12 m	0.12 m
Damping Ratio, ζ	0.022	0.05
Resonant Frequency, w_n	44.554 Hz	64 Hz

The system mass, M, was obtained by weighing the 10 inch impeller and the motor shaft. This differs greatly from the mass used in our initial value which was estimated using our CAD model. The eccentric mass in the model represents internal imbalance in the system and was estimated by iterating through various magnitudes using the one that best matched empirical results. The model was also run with eccentric masses of 1.3g and 2.6g in addition to the internal .00075 kg of imbalance. These are the same eccentric masses that were physically added to the impeller and used in our empirical data collection. The eccentricity is defined as the set distance the mass imbalance is located from the central rotation point of the impeller. This was obtained by measuring the distance our eccentric mass was from the center of the impeller. The damping ratio was found and refined from additional research into Solidworks modal analysis, and general vibrating structure characteristics [14,26]. Because empirical calculation of structural damping can be difficult, this estimate saved us time in starting our predictions and should theoretically be close. Finally, we used a resonant frequency of 44.554 Hz which was obtained by performing a Fast Fourier Transfer on empirical data. The resulting plot can be seen in Appendix G. This greatly differs from the initial frequency of 64 Hz which was estimated using the results of our Finite Element Analysis study. The study is explained in greater detail in the FEA/CAD section. Ultimately we believe that the variance between our estimated resonant frequencies can be attributed to fluid dynamic effects that aren't accounted for in the FEA study.

Once every input parameter was obtained we were able to run our mathematical model and predict system acceleration. Using Matlab (see Appendix F for code) our model predicted maximum acceleration to be less than 0.15 G when no eccentric mass was added to the system. This is depicted in Figure 9 below.

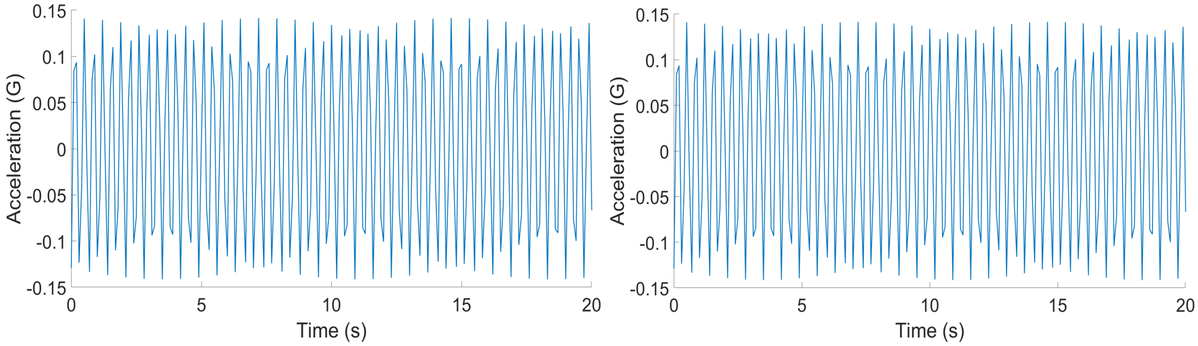

Figure 9. Mathematical prediction of acceleration magnitude in the X-direction (left) and Y-direction (right) with 0 grams of eccentric mass attached to the vane of the impeller.

The model predicted acceleration magnitude by taking the second derivative of the displacement Eq. 3 and converting units to G's. We must also note that we ignored the transient response, or the first term of Eq. 3, since our team and Heller Industries are more concerned with steady state response at maximum operating speed. Predicted acceleration when 1.3 g and 2.6 g of eccentric mass were added to the system can be found in Appendix G.

When Figure 9 is compared to the empirical data we collected for a similar setup, (see Figure X in the Data Collection section) we see that the model appears to predict an acceleration magnitude between the X value, 0.27G, and the Y value of 0.5G. Our assumption of symmetrical vibrations in X and Y appears to be somewhat flawed, we believe that this could be contributed to new fluid dynamics behavior introduced by the larger impeller that we hadn't considered. While further analysis will be needed we could potentially improve the model by adjusting the damping ratio or other parameters. The stiffness k, mass m, and damping value c all factor into this ratio as shown in Eq. 6. As such, we need to cross examine this result with other estimates of parameters to try and hone in on the problematic parameter.

Refining the damping ratio is difficult but we have a couple of ways to approach this problem. The first involves using FEA to estimate the stiffness coefficient. We can then determine the damping coefficient by experimentally performing a wind down test. With the new stiffness and damping coefficients we can refine the damping ratio using Equation 6.

$$\zeta = \frac{c}{2\sqrt{mk}} \qquad (6)[30]$$

The other way we can estimate damping involves using the refined natural frequency, w_n, mentioned above, and the damped frequency, w_d. Using Eq. 7 below we can refine the new damping ratio.

$$w_d = w_n\sqrt{1 - \zeta^2} \qquad (7)[30]$$

The refined mathematical model will more accurately predict acceleration experienced by the system. We can then vary different inputs to see how changing different parameters affects acceleration magnitude.

CAD/Finite Element Analysis

Finite Element Analysis was used to determine multiple parameters in our overall modeling system. The group made use of Solidworks, for modal analysis, and Hyperworks for static analysis. We used Heller's provided CAD geometry for both of these FEA studies as well as other simple parameter calculations.

Static Analysis

The stiffness parameter, k, was estimated with Finite Element Analysis. Because the shaft is tapered towards one end**,** we chose to perform a simple FEA static study instead of a hand calculation, which would have been more cumbersome [28]. This was done in Altair Hyperworks based on prior group experience with the software. As discussed in the "Mathematical Model" section, we approximated the stiffness of the system as cantilever stiffness of the motor shaft. As a result, we calculated the stiffness using the formula:

$$k = \frac{F}{\delta} \qquad (8)[28]$$

Where k is stiffness (N/m), F is the force applied to the tip of the shaft (N), and δ is the tip displacement (m). Below is a screenshot of the results from the FEA study that was done to estimate the stiffness of the motor shaft.

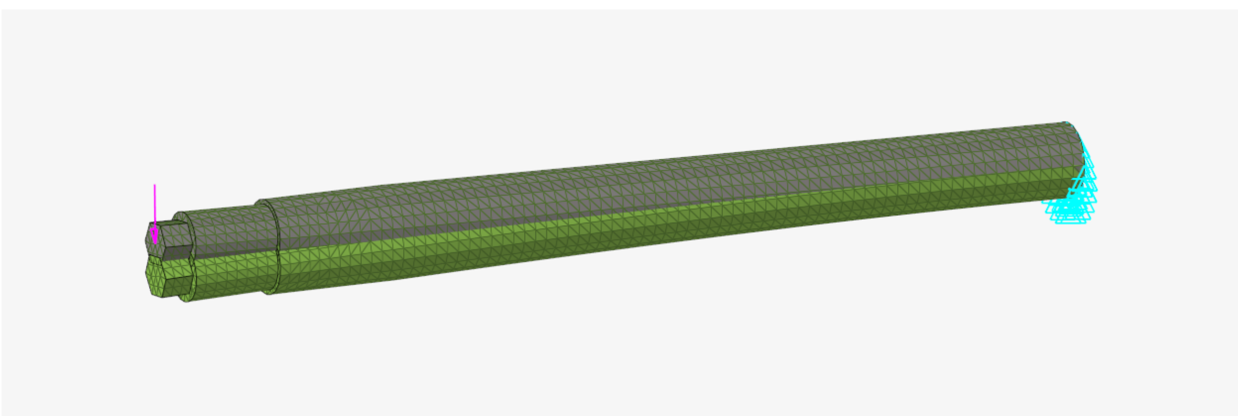

Figure 10. Visualization of shaft stiffness study. The applied load was 10N and the scale factor of deformation was 100X.

There are some important assumptions that went into this study. The following material properties were used: 416 stainless steel (Young's Modulus = 200 GPa and Poisson's ratio = 0.394)[31]. One end of the shaft was fixed in place as shown in blue above to represent where it interfaces with the motor. And the force, shown in pink above, was applied perpendicular to the shaft length [28]. These assumptions helped us perform simple cantilever beam bending and determine the stiffness of the motor shaft.

The applied loads used were 1, 5, and 10 N across three separate trials, and the tip displacement as a result of each load was recorded. The full results can be found in Appendix F. The final result from using Equation 8 was a stiffness of approximately 182,382 N/m. Data used in the stiffness study can be found in Appendix G.

This stiffness appears to be only a portion of the overall system stiffness, which sits around 471530 N/m based on the prediction which combined our mathematical model and FFT analysis. If we assume the shaft adds linearly to the overall system stiffness in conjunction with the motor mounts, motor module, and other metal components, we can conclude that this shaft stiffness comprises a sizable portion of the overall system stiffness. Therefore any change in shaft stiffness should have a significant impact on system resonant frequency and overall behavior. This will be explored in more detail in the Discussion and Recommendations sections of this report.

Modal Analysis

The other use of FEA in our engineering analysis was determining the resonant frequency of the entire motor and impeller module. Similar to the previous study, we had to make certain assumptions to simplify our simulation while accurately representing the real life behavior as best we could.

The first assumption we made was that the motor mounting plate cannot translate in order to accurately capture the relative motion of the shaft and impeller, fixing the mounting plate and motor in place. Additionally, input the correct material properties for each component after discussion with Heller about the material properties in our motor-impeller system. The shaft was 416 stainless steel, the motor mounting plate was 6061 alloy aluminum, the impeller was A1101 aluminized steel, and the motor was modeled as a solid block of cast AA380.0-F die cast aluminum [31]. The provided CAD geometry did not include the inner motor components so a solid block made of the aluminum case material was used as an approximation.

For these first few studies, we chose to perform Linear Dynamic, Harmonic analysis because it helps us analyze steady state response due to harmonic loads [29]. The input torque serves as a harmonic load, rotating at the frequency of the motor, for which we used an input torque of 0.62 N-m. This was derived from motor torque curves for similar Nidec motor models [32], extrapolated to a larger 0.25 HP motor as advised by Heller Industries. Finally, an initial damping ratio of 0.05 was used. This value was found from research into Solidworks modal analysis [26], which allowed us to start with what we believe to be a close approximation of the actual damping value of our system. With all of these assumptions in place, a modal analysis was conducted and the results are shown below.

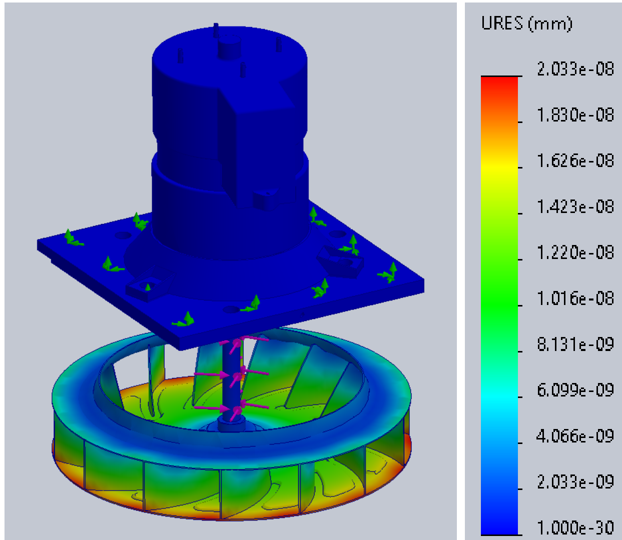

Figure 11. FEA displacement results for the 1.0 study.

Table 3. Fundamental frequencies for the 1.0 FEA study

Mode Number	Fundamental Frequency (Hz)
1	62.997
2	64.539
3	64.797

The key takeaways from this study is that the displacement magnitude is small (~$2x10^{-8}$ mm at most), and the fundamental frequency appears to be similar across the first 3 modes of vibration. The lowest frequency is the most relevant to us, as higher modes are harmonics at frequencies which are out of the operating range and do not influence the vibrations we see in the module [30].

From this study we changed our assumption of the motor plate remaining fixed. Since the motor can displace and vibrate, our initial assumption may have been overconstrained. We allowed the plate to move 0.1mm in x, y, and z and ran another study, denoted “2.0”, keeping all other inputs the same. The following results were produced from this.

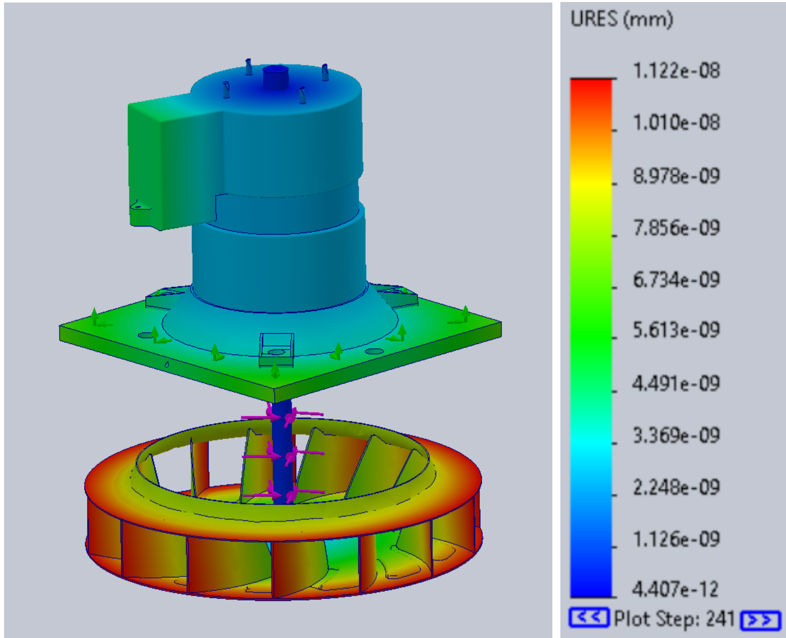

Figure 12. FEA displacement results for the 2.0 study.

Table 4. Fundamental frequencies for the 2.0 FEA study

Mode Number	Fundamental Frequency (Hz)
1	9.7815
2	64.516
3	64.783

The main change noticed in these results is that the model displaced in more locations. This was expected since we removed some constraints of motion. However, the displacements were still small, on the same order of magnitude as the previous study. We did observe a sharp decrease in resonant frequency for the first mode, however, the other two relevant modes (in the y and z direction) have similar fundamental frequencies. These are likely the two modes being excited by the motor.

From these results, we wanted to explore the effect of damping ratio directly on the fundamental frequencies of the system. We decreased the damping ratio from 0.05 to 0.005, based on further research [12,26]. Everything else remained constant from the previous study. Running this setup, denoted as 3.1, gave us the following results.

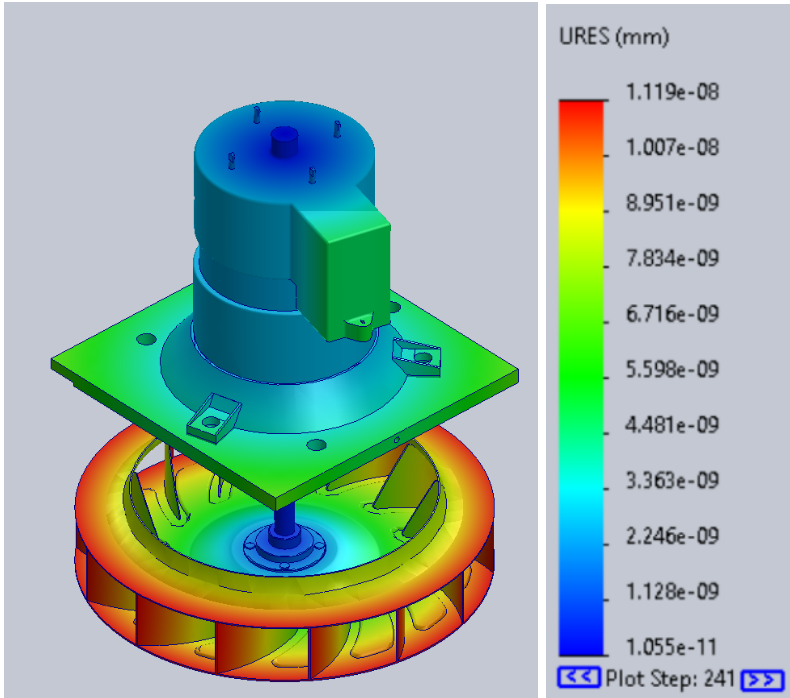

Figure 13. FEA displacement results for the 3.1 study.

Table 5. Fundamental frequencies for the 3.1 FEA study

Mode Number	Fundamental Frequency (Hz)
1	8.0405
2	64.389
3	64.621

Compared to the 2.0 study, this 3.1 study overall is fairly similar. The displacement magnitudes are again quite small, essentially the same as in 2.0. We do see the resonant frequency of mode 1 decrease from ~9.8 to ~8.0 Hz, but modes 2 and 3 did not see any significant decreases. Seeing that these two modes have similar resonant frequencies would suggest that vibrations in multiple directions become amplified when the system is excited around this frequency [26,30]. Therefore, if we can tune the system to resonate at a much higher frequency, beyond the standard operating range of this 0.25 HP Nidec motor, we should be able to significantly reduce the vibrations.

With these simulations, our goal is to have the displacements and fundamental frequencies from these studies align with our empirical data and mathematical model. If they do not align, further iteration and refinement is needed. The data we collect acts as our reference point; if our experimental data with a FFT applied predicts a significantly different resonant frequency, we will need to revisit our FEA modal analysis. If the displacements disagree but the resonant frequencies match, it is possible that our constraints are incorrect in our FEA studies.

Between Design Review 3 and now, our understanding of the physical system and also of SolidWorks simulations changed quite drastically. Where we initially were performing a Harmonic analysis, we changed course and began to perform Frequency Analysis instead. One reason for doing so was that our simulation times were dramatically reduced. Harmonic analyses calculate displacement, acceleration, and many other results (most of which were not useful to us) for the first 15 vibration modes in the system [34]. The only modes causing any significant vibrations at frequencies close to operation of the system are the first three modes, as shown previously. This renders a lot of the computation in each simulation useless, which we felt was very inefficient. In addition to this, we had issues getting displacement and acceleration magnitude results that were accurate to those predicted by our math model. Due to this large discrepancy as well as the longer simulation times, we decided to move away from Harmonic Analysis for our analysis during this semester. It may be worthwhile to pursue this form of analysis with the proper resources and timeline, but in our case this was not feasible.

Frequency analysis calculates only the first five fundamental frequencies and can also help to visualize each individual mode of vibration. Calculating only the first five modes allows it to run very fast, and the visualization of vibration modes helps us gain useful insight into the system as well as confirm our understanding of the vibrations from simple observation.

With all of this in mind, we made some changes to our CAD model across multiple different iterations (versions 4.0 to 9.0). The details of each of these changes are outlined in Table 6 below.

Table 6. Details of the changes made across all of the Frequency Analysis studies we ran between Design Review 3 and the Design Expo. Also shown are the resulting resonant frequencies for the first 3 modes of vibration, numbered accordingly.

Simulation Version	Important Changes	Fundamental Frequencies
4.0	● Second motor mount plate added as a rigid base ● Alignment plates between shaft and mount plate removed ● Motor mounts no longer clip into mount plate ● Roller/slider constraint applied between top and bottom motor mount plates. Bottom plate *fixed in place*	1. 63.635 Hz 2. 64.203 Hz 3. 64.236 Hz
5.0	● Eccentric mass was added to flat bottom surface of the impeller, 0.12m from axis	1. 63.382 Hz 2. 63.740 Hz 3. 64.277 Hz

6.0	• Using math model, allowed 0.175 mm translation in x, y, and z for slider constraint to mimic real life translations	1. 63.382 Hz 2. 63.733 Hz 3. 64.270 Hz
7.0	• Bottom plate removed • Constrained top plate as a roller/slider relative to the front plane to reduce number of elements • Impeller and shaft rigidly connected • Angular velocity of 2850 RPM (298.45 rad/s) on shaft	1. 70.693 Hz 2. 71.279 Hz 3. 79.494 Hz
8.0	• Forced plate slider displacement to 10mm in all directions to attempt to make plate move	1. 73.266 Hz 2. 73.341 Hz 3. 79.848 Hz
9.0	• 10mm displacement removed, replaced with original math model value of 0.175 mm in x, y, and z. • Slider constraint between top of shaft and bottom of motor face	1. 74.629 Hz 2. 74.676 Hz 3. 80.007 Hz

As can be seen above, many iterations of this simulation were performed. Research on the SolidWorks website/forums [34] led to many changes in our understanding of the system and how it needed to be constrained. One of the main changes in our approach was that we added an eccentric mass into our CAD assembly. This mass began as approximately 0.5 grams. This was the result of a prediction from the math model estimating the amount of imbalance needed to achieve the vibration magnitudes found in the data of "no added mass", which is discussed in more detail in the Empirical Data section. The mass was added due to our realization that the CAD is likely to be perfectly balanced. As such, it may not be vibrating in the simulation in the same way that it does in the real world. This did somewhat affect the results of the simulations and we believe it would be worthwhile in the future to add masses of 1.3 and 2.6 grams to mimic the paperclips which we added in our empirical testing setup to induce vibrations. This will be discussed again in the Discussion section of the report.

Another important change in our modeling was the change from an applied torque to an apple angular velocity of 2850 RPM. This simulates operation at 45Hz, which was the speed at which we ran the motor in our testing (see Empirical Data section). We believe that the previous torque which was being applied was accurate but did not produce the proper angular velocity, leading to the difference in resonant frequencies.

Lastly, we used our math model to inform the amount of displacement allowed for the motor plate and the motor, since the two are rigidly fixed together. The allowed displacement in X, Y, and Z was 0.175 mm This was another assumption which we believe led to a more realistic simulation.

With all of this in mind, we can see the final results in our 9.0 model which incorporates all of the important changes that were just described. This model resulted in predicted resonant frequencies of 74.629, 74.676, and 80.007 Hz for the first three modes of vibration. To help visualize these vibrations, shown below in Figures 14, 15, and 16 are the SolidWorks generated deformations for each vibration mode. Please note that these are NOT to scale, and merely serve as visual aides.

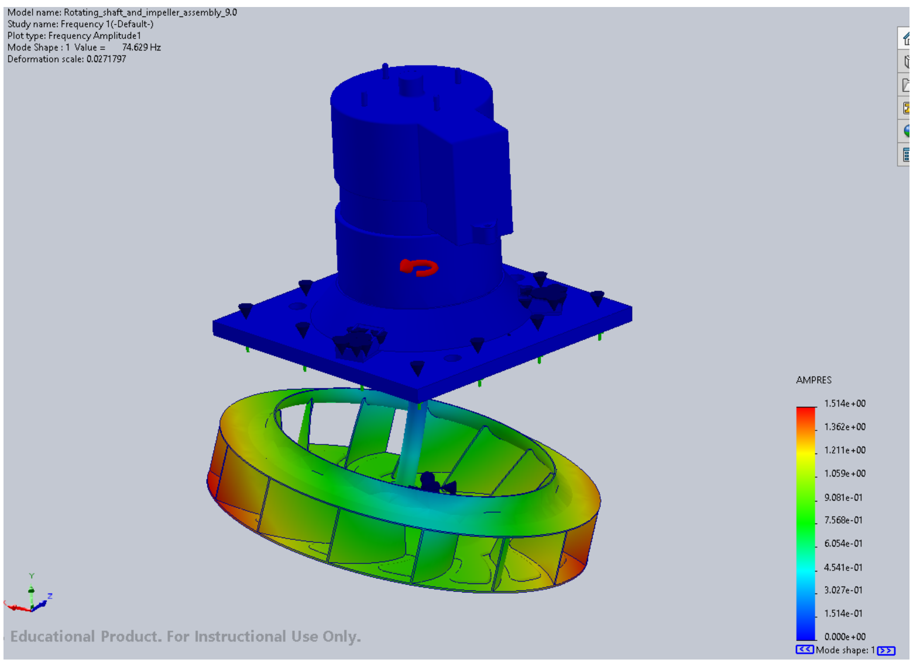

Figure 14: Visualization of first vibration mode for “9.0” frequency analysis study. The fundamental frequency here was 74.629 Hz.

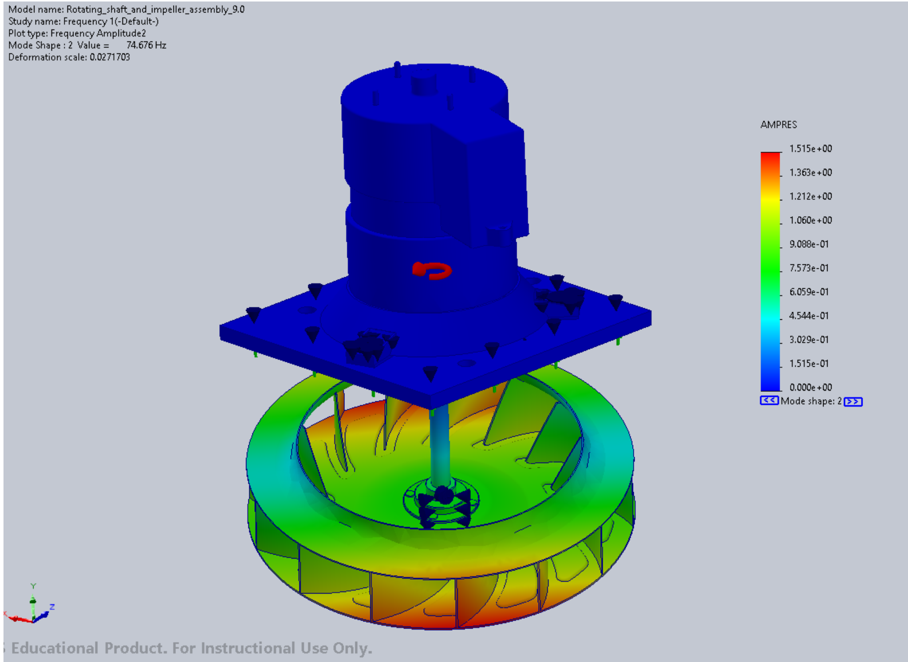

Figure 15: Visualization of second vibration mode for "9.0" frequency analysis study. The fundamental frequency here was 74.676 Hz.

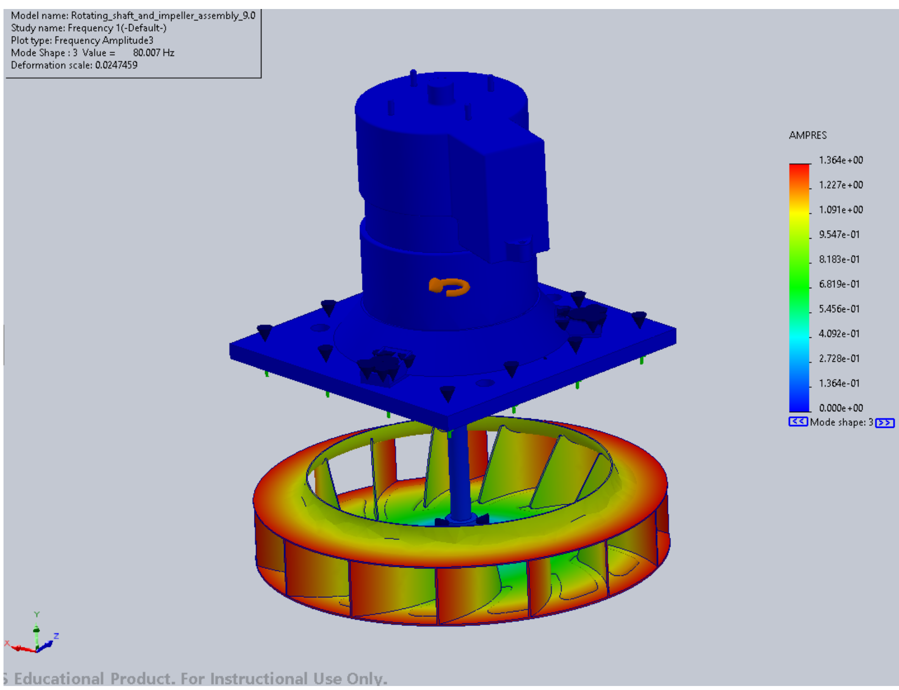

Figure 16: Visualization of third vibration mode for "9.0" frequency analysis study. The fundamental frequency here was 80.007 Hz.

There are a few key takeaways from these images. The first is that Modes 1 and 2 represent essentially the in-plane translation of the shaft and impeller due to shaft bending. This is what our math model assumes is the case, so seeing this deformation is encouraging and validates that the main assumption of our math model was reasonable.

Another interesting takeaway is the presence of vibration within the impeller itself in Mode 3, at a fundamental frequency not that far from the first two. This type of vibration was not one which we accounted for in our math model, and is not something that would be easy to empirically verify. Because this frequency is so close to the first two, it is possible that the impeller itself vibrating may be contributing to the overall vibrations of the system. However, further analysis is needed on this before a firm conclusion can be drawn.

Finally, we note that these predicted resonant frequencies are on the order of 70-80 Hz. This is significantly higher than the 45 Hz we ran most of our empirical studies at. These are not quite the results which our team expected from this simulation. We discussed this with our sponsor Jim Neville from Heller Industries [35], and we came to the conclusion that there is likely a fluid dynamics component to the system which is not being accounted for in this simulation, nor any of the prior iterations. As the impeller increases in size and gets closer to the walls of the module, fluid flow begins to both stagnate and experience unpredictable turbulence [35, 36]. Because of this, a computational fluid dynamics (CFD) study which also incorporates the dynamics we have established in our model is needed to truly characterize the system. This would be the subject of future work for this project. Additionally, in the Discussion section of this report we will go into more detail on potential pitfalls of our modeling methods.

In conclusion, the finite element frequency analysis studies performed for this project have shed some valuable light on the behavior of the system. Between our many iterations on the modeling setup as well as research and consultation with engineers and professors, we can state with confidence that this system cannot be adequately simulated with only a frequency analysis. The results of a simple FEA frequency analysis provide resonant frequencies which are nearly double those of our FFT, which we trust to be a reliable source of resonant frequency. These results point to future work in the form of computational fluid dynamic (CFD) studies of this system, which will be discussed in more detail in the "Discussion" section of this report.

Empirical Data

Testing Plan

Courtesy of Heller Industries, we had an impeller reflow heating module on site, consisting of all the necessary components to conduct our testing. The purpose of testing was primarily to quantify the magnitude of the vibrations in different directions, which enabled us to perform a Fast Fourier Transform (FFT) to determine the natural frequency ω_n of the system. Table 17 indicates our bill of materials for this setup.

Table 7. Bill of Materials Relevant materials included in the testing setup.

Component	Quantity	Description
Reflow Module	1	Module Housing
Nidec 0.25 HP Blower	1	Motor
Shang-Hai Dong Pu Transformer	1	240V Transformer
Invertek Drives Optidrive E^3 Motor Controller	1	Controls motor speed between 20 and 60 Hz
Motor Shaft	1	416 stainless steel shaft connecting motor to impeller
Changzhou Impellers	3	Aluminized Steel Impeller of varying diameter (6, 8, 10 inch)
C-Clamp	2	Clamps to secure module to test bench
Paperclip (1 gram each)	2	Serves as eccentric mass for our testing
MotionNode IMU	1	Inertial Measurement Unit (IMU) for collecting vibration magnitude data

For data collection, we first positioned 2 tables a fixed distance apart to suspend the impeller module off the ground. This was necessary to allow sufficient airflow through the module in an effort to emulate normal operating conditions. The module was placed on the 2 tables using clamped down wood blocks, used to increase contact area between module and the table, and stabilize the modules position. The prepped testing setup can be seen below in Figure 17.

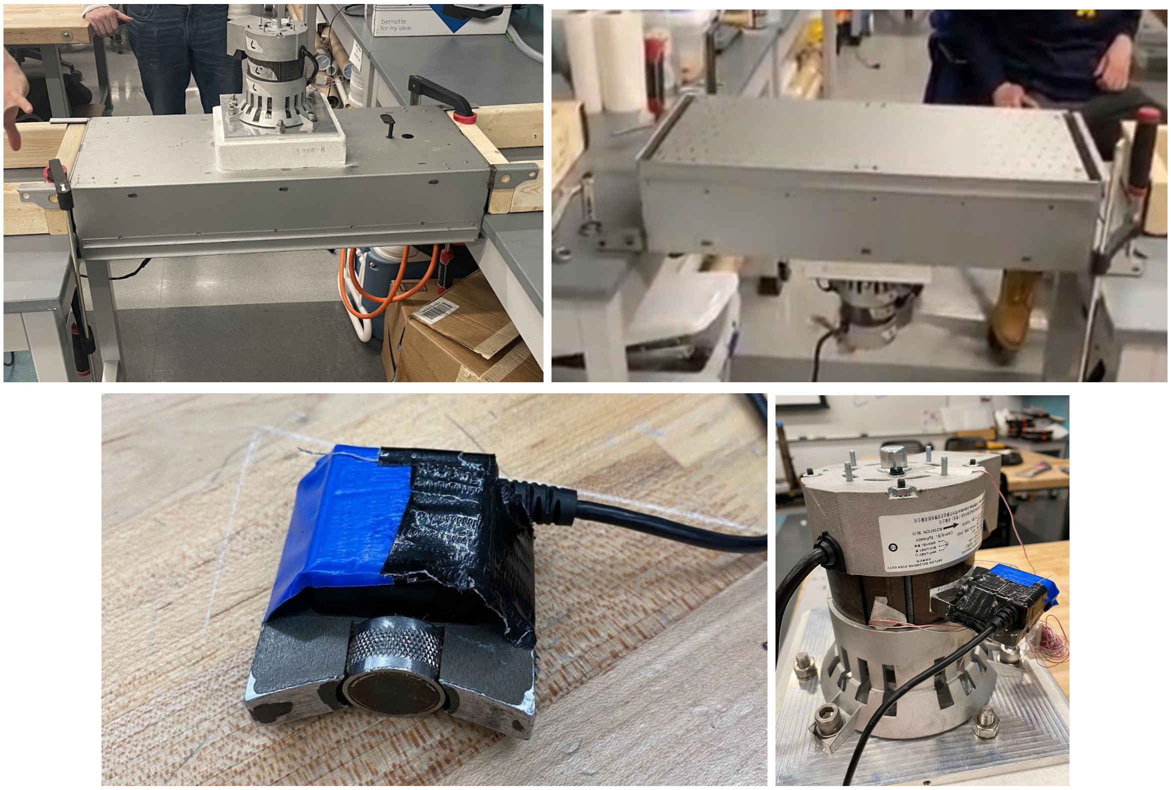

Figure 17. Configured testing setup. The inverted setup is shown on the top left, and the non-inverted setup is on the top right. The 0.25 HP Nidec motor is visible on top of the module on the left image. It rests upon the motor mounting plate, which itself rests on the large white insulation block. These components are fastened together, to firmly secure them. The accelerometer was secured to a magnet via duct tape as shown on the bottom. This magnet was then secured to the motor as shown on the bottom right. This exact position was replicated in each test to keep coordinate directions consistent.

For our testing plan, we wanted to establish baseline vibrations coming from the motor. We conducted tests at maximum motor speed/steady state operating speed with no impeller, 6in, 8in, and 10in impeller with the hopes of better understanding the overall impact of inertia on the system. Next, we attached two 1 gram paper clips to the impeller blades to act as an eccentric mass and quantify the effect of mass imbalance on observed system dynamics. Images of how these paper clips were mounted can be seen below in Figure 18.

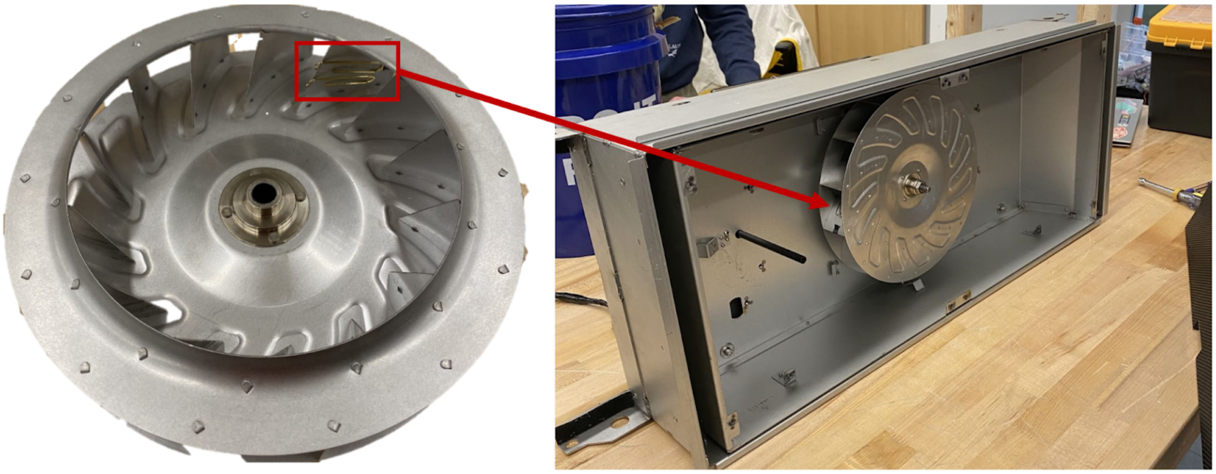

Figure 18. Close up of the 10 inch impeller blade with attached paper clips acting as the eccentric mass. This image also shows how the impeller sits on the shaft within the housing module. The module was not operated in this open, sideways configuration for safety.

Finally, we ran frequency sweep tests both with 1 and 2 gram paper clips attached to our impeller to characterize the vibration magnitude of an unbalanced system. This allowed us to identify frequencies at which vibrations are maximized and minimized based off of the current system setup. All of these tests were performed both with the module upright and inverted to better represent real world conditions and to quantify the effects that inverting the module would have on vibrations. All the tests conducted can be found below in Table 9.

Table 8. Summary of our testing plan, with brief rationale behind each test. Each test was conducted twice, with the module both upright and inverted to better mimic real world conditions and to quantify effects of inverting the module on system dynamics.

Trial	Test	Description/Notes
1	No impeller, max motor speed	Determine baseline motor vibration
2	6in impeller, max motor speed	Quantify effects of inertia on system dynamics
3	8in impeller, max motor speed	Quantify effects of inertia on system dynamics
4	10in impeller, max motor speed	Quantify effects of inertia on system dynamics
5	10in impeller, 1 gram eccentric mass, max motor speed	Quantify effect of system imbalance on vibration magnitude and damped natural frequency
6	10in impeller, 2 gram eccentric mass, max motor speed	Quantify effect of system imbalance on vibration magnitude and damped natural frequency
7	10in impeller, variable motor speed, 0 gram eccentric mass	Quantify effect of system imbalance on vibration magnitude and damped natural frequency. Variable motor speed was performed at 20-40-50-60 hz
8	10in impeller, variable motor speed, 1 gram eccentric mass	Quantify effect of system imbalance on vibration magnitude and damped natural frequency. Variable motor speed was performed at 20-40-50-60 hz
9	10in impeller, variable motor speed, 2 gram eccentric mass	Quantify effect of system imbalance on vibration magnitude and damped natural frequency. Variable motor speed was performed at 20-40-50-60 hz

Data was collected using a MotionNode IMU (Inertial Measurement Unit), otherwise known as an accelerometer. The IMU was attached directly to the motor at a predefined location for every test, and data was recorded using MotionNode Software. The data was extracted into Microsoft Excel spreadsheets for easy data processing and analysis. More details on the data processing conducted can be found below in the Data Collection subsection.

Safety

After examining the impeller-module system and discussing an approach to testing/collecting data, we laid out a schematic of the system to ensure we knew how all the components worked and the safety precautions needed to prevent any injuries/safety hazards. The electrical schematic of our system and the electrical components it contains was developed and can be seen in the figure below.

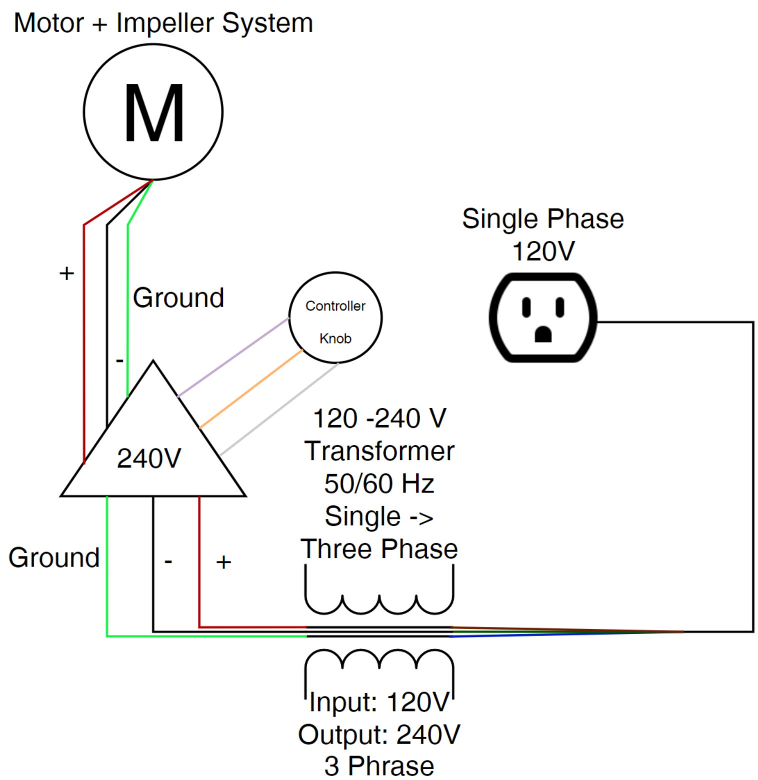

Figure 19. Electrical schematic of impeller- motor system. A critical note is that the system takes single phase 120 V and converts it to three phase 240 V with a transformer. This transformer is insulated and grounded from any external interaction.

To maintain electrical safety, we recognized we needed to insulate and protect the 120/240 three phase transformer that was in the system. Our motor uses 240V and as such we needed a transform to step up voltage from a 120V outlet t0 240V. When we received the transformer and the rest of the wiring setup, there were a few issues we needed to correct.

Firstly, there were a few areas of exposed wire on the transformer and motor controller. Shown below are images of the corrections we made to make these safer.

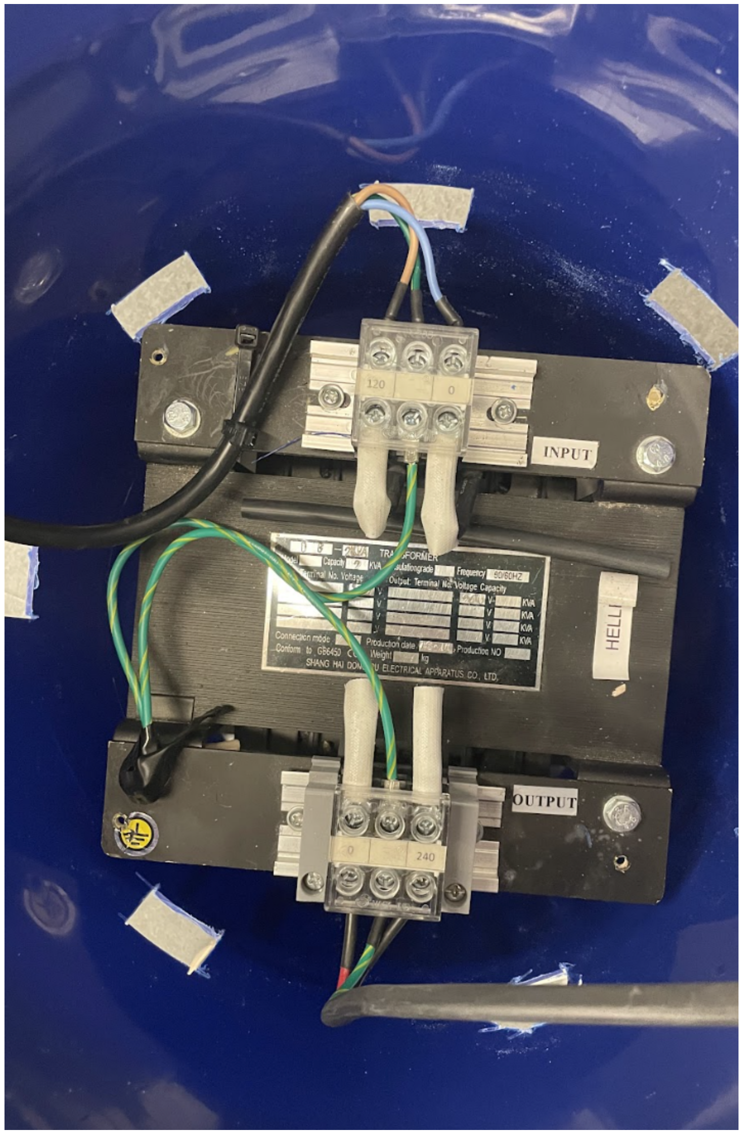

Figure 20. 240V transformer isolated in the electrical box (made from a ventilated Lowe's bucket). Special consideration was taken to ensure that all wires had built in strain relief, and all connections were double checked to ensure they were secured, and all exposed leads covered with heat shrink/electrical tape

Beyond just these measures, our group wanted to ensure that the electrical components of the module were not exposed, where someone could accidentally bump into them. Our solution for this problem was to use a 5 gallon paint bucket. We cut holes for ventilation and cable management, then place the transformer and motor controller inside to isolate them from the impeller module as well as ourselves. Shown below are some images of this bucket safety setup, as well as a wiring diagram of our system.

Figure 21. Full electrical setup contained within our modified Lowe's paint bucket. The transformer is located at the base of the bucket. The motor controller (purple) and additional wire housing (blue) were placed on top of the transformer for easy access. No wires are strained in this configuration. The electrical box is covered with a lid during normal operation to eliminate the risk of touching any live components.

After completing our electrical safety, we clamped the module down in a manner that was representative of Heller Industries ovens. The result is seen below:

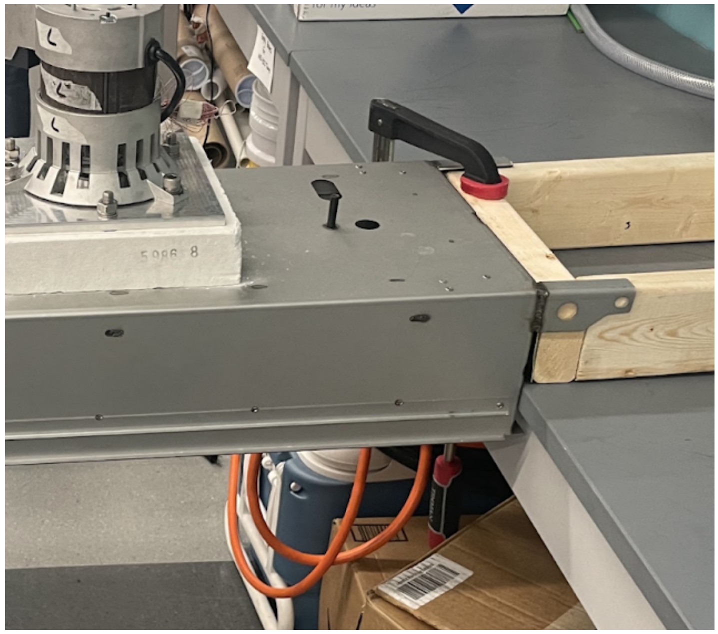

Figure 22. Motor-Impeller module resting on fabricated wood blocks, which are then clamped down on rigid tables. We believe that this setup accurately represents the setup seen in the reflow ovens, within reason.

This clamping proved secure for our application without impeding the module's ability to vibrate. This allowed for effective data collection. With all of these safety measures taken, our team was able to perform experiments and collect data outlined previously with minimized risks and injuries.

Data Collection

After performing the experiments as defined in the Testing Plan section, we obtained a large amount of acceleration magnitude data in the x, y, and z directions for a variety of different impeller module setups. This not only shows us direct acceleration magnitude for various setups, it also will allow us to compute the resonant frequency of the system at the given operating speed of the motor via Fast Fourier Transform (FFT). The results of these experiments will be presented here along with some observations and comments regarding the data.

For all of the following testing results, the experiments were performed with the module in the non-inverted configuration of the reflow oven. We also performed the same tests in the inverted configuration, however some of these results are still being processed. Finalized results of both configurations will be included in the final report, along with a frequency sweep test to analyze system behavior across varying operating frequencies.

No Eccentric Masses

We first wanted to investigate the acceleration experienced when the module was operating at normal conditions. This would set a baseline that all other tests could be compared to. In order to do this we conducted tests using the 10 inch impeller in the non-inverted orientation with no eccentric mass attached. From this baseline we were able to test the effects of changing impeller size, adding eccentric mass, and changing the orientation of the module. Figure 23 below shows these baseline results for inverted and non-inverted setups.

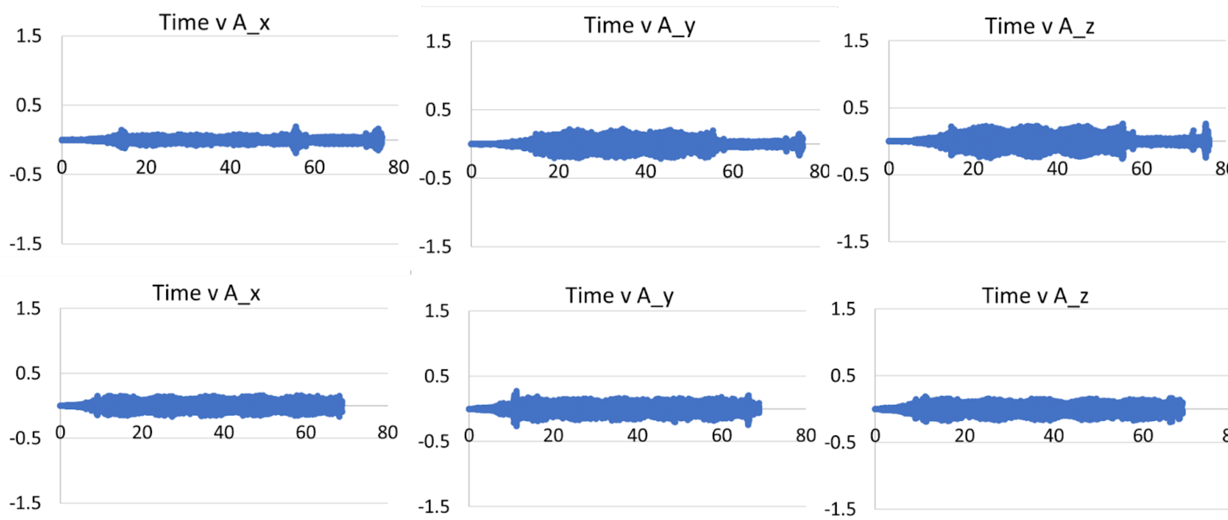

Figure 23. Acceleration magnitude in each direction with no eccentric mass attached to the 10 in impeller in the top (top set of graphs) vs inverse configuration (bottom set of graphs). Measurements were taken when the motor was operating at a steady state of 60 Hz in the non-inverted orientation. The maximum acceleration recorded in this set was 0.29G in the Y and Z directions.

1 Gram of Eccentric Mass

We are currently still working through testing the impeller module in the inverse with the 1 gram mass attached to it, however the results from the non-inverted orientation are shown below in Figure 24.

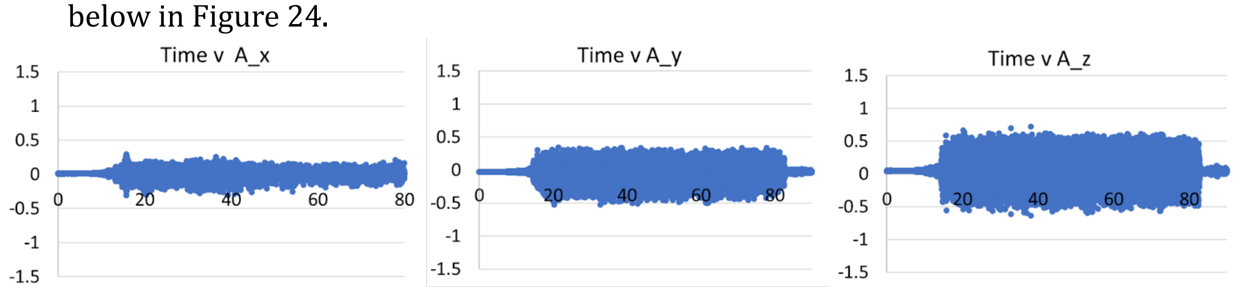

Figure 24. Acceleration magnitude in the X-direction with 1 gram eccentric mass attached to the impeller. Measurements were taken when the motor was operating at a steady state of 60 Hz in the non-inverted orientation. The maximum magnitude recorded in this test was 0.6G in the Z direction.

When 1 gram of eccentric mass was attached to the vane of the impeller we measured a maximum amplitude of 0.27G in the x-direction. When compared to the prediction from the mathematical model the maximum acceleration differs by 11%. This is relatively close, however we notice that in the Y direction acceleration is significantly larger at around 0.5G. Given that our model assumed symmetric behavior in X and Y, we may need to adjust this assumption to more accurately characterize the system. More work is needed in this regard.

2 Grams of Eccentric Mass

We are currently still working through testing the impeller module in the inverse with the two gram mass attached to it however the results from the non-inverted orientation are shown below in Figure 25.

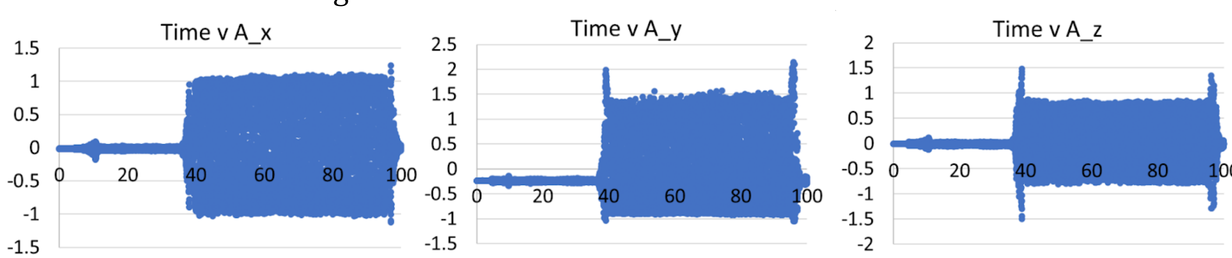

Figure 25. Acceleration magnitude in the X-direction with 2g eccentric mass attached to the impeller. Measurements were taken when the motor was operating at a steady state of 60 Hz in the non-inverted orientation. The maximum acceleration magnitude recorded was 2G in the y direction, and 1.5 in the z direction. Maximum motor speed was achieved at around the 37 second mark.

When 2 grams of eccentric mass was attached to the end of the impeller we measured a maximum acceleration of 1.2 G which is a significant increase and close to the maximum acceleration that Heller was experiencing in the data they collected. Their data was taken on modules attached together inside their reflow oven, rather than individual modules, so there could be other factors at play [26]. However, given our equipment and ability, this is a relatively close replication of their results for a single module. Additionally, when ramping the motor controller up from 20 Hz to 60 Hz we noticed significant vibration at two different frequencies. The first was around 44 Hz, the vibration then subsided until the motor was operating at 60 Hz where it returned again. This is something we plan on exploring further by performing additional frequency sweep tests which measure acceleration magnitude at different frequencies as the motor is ramped up to 60 Hz and back down to 20 Hz. Figure 26 below shows the results of an initial frequency test we performed (Notes: we wish to conduct more tests before making any conclusions, as we believe we can improve the testing setup for this scenario)

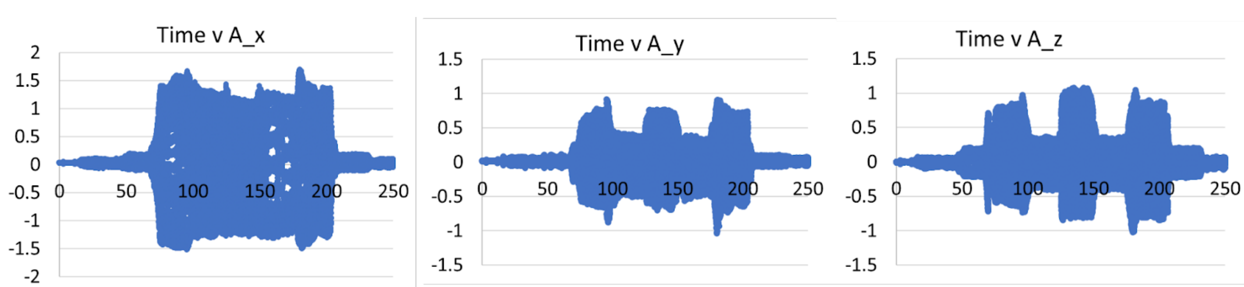

Figure 26. Frequency sweep plotting acceleration magnitude in each direction with 2 grams of eccentric mass attached to the impeller. Measurements were taken when the motor was operating at frequencies of 20-40-50-60-50-40-20 Hz in the non-inverted orientation. The maximum acceleration magnitude recorded was 1.55G in the X direction.

We then proceeded to obtain additional empirical data at 45 Hz operating speed to further examine the vibration at various eccentric masses. The results can be seen in Figure 27 below:

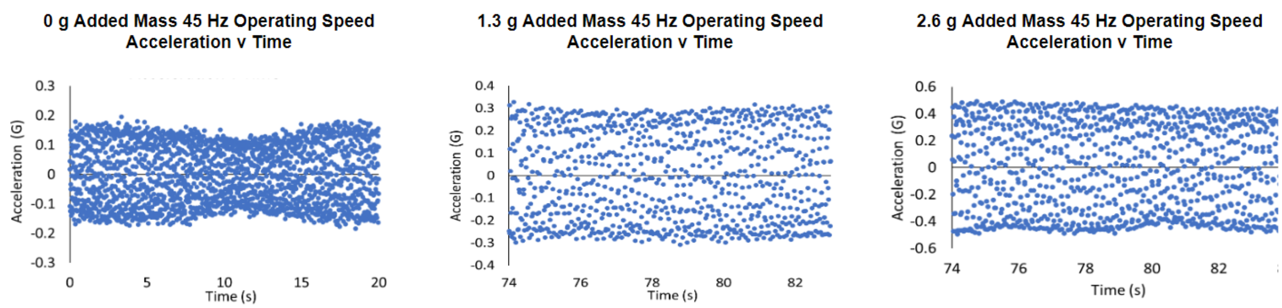

Figure 27. Frequency vs Acceleration plots for varying eccentric masses at an operating speed of 45 Hz. The main point to notice is we did see an oscillatory or periodic motion of vibration at no eccentric mass (far left) but not for any added eccentric mass.

Comparing this to our mathematical model at a natural frequency of 45 Hz, we see that there is a similarity across the additional eccentric masses, however, our 45 Hz, no eccentric mass varies ~21% from what we obtained experimentally.

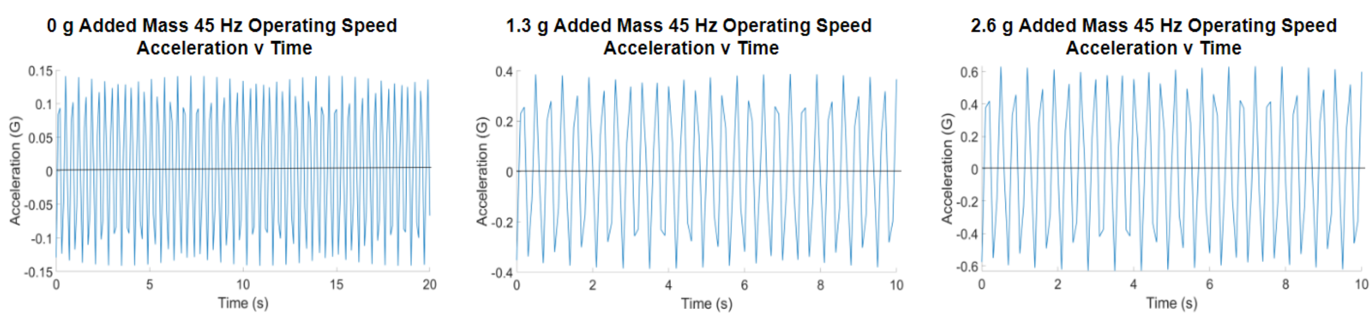

Figure 28. Mathematical model under same operating conditions as Figure 27. We noticed when working on the math model the no added eccentric mass was ~21% off from our empirical data while the added eccentric masses were within 15% of our empirical data. This

From this frequency and additional 45 Hz data collection, we deemed that the natural frequency was around 45 Hz.. Additionally, we noticed during testing that whenever we operated the system around 40-45 Hz the system significantly increased in vibration. To validate this, we later performed a Fast Fourier Transform on this data and confirmed the natural frequency to be exactly 44.56 Hz.

Fast Fourier Transform (FFT)

We performed a fast fourier transform on a 45 Hz motor speed data collection trial. The results of this analysis are shown below in Figure 29.

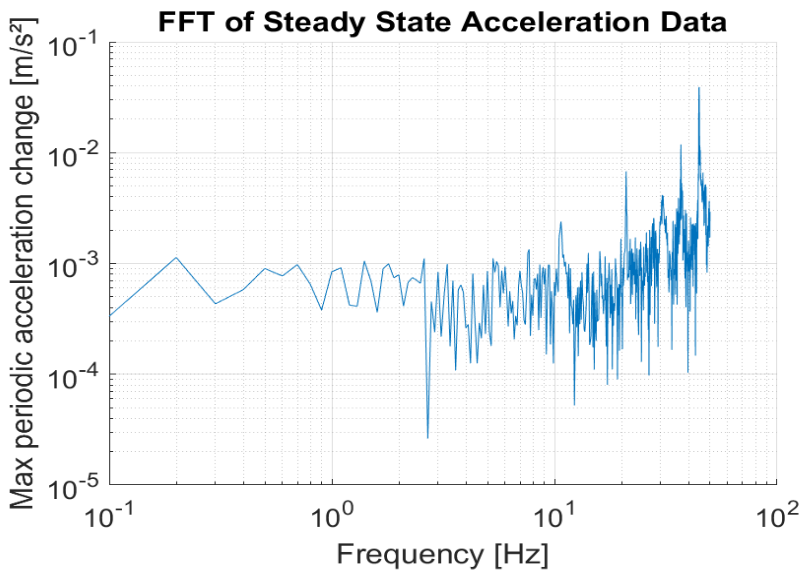

Figure 29. Fast Fourier Transform performed on 0 g acceleration data in the X-direction. The resulting resonance frequency was determined to be 44.554 Hz, based on the location of the highest peak of the plot.

This FFT was used to inform our mathematical model, providing directly the resonant frequency we utilized for that model. Additionally, it allowed us to perform back-calculations to estimate system stiffness. This will be described in more detail in the Verification and Validation Section.

VERIFICATION AND VALIDATION

Due to our project being modeling/analysis based, our verification and validation of our requirements and specifications comes through our engineering analysis. Our model design is intended to conjoin multiple modeling methods to both identify and then cross examine parameters to validate their values. In this section, we will discuss how we plan to validate that the values we have found for each system parameter are reasonably accurate.

Stiffness

Previously, we were estimating our stiffness value to be 182,382 N/m from our static HyperWorks beam displacement analysis. While this calculation was valid for just the shaft stiffness, it failed to take into account other components in the system which contributed to stiffness. These include the motor mounts, the sheet metal module itself, and even the mounting plate. As such, we decided to use a first principles method to verify the stiffness in our system. To do so, we used the FFT analysis to produce a resonant frequency, and physically weighed the vibrating mass of the system. For simplicity, we chose this mass to be the motor, shaft, impeller, and motor plate. These are the components which displace the most as a result of the driving imbalance in the system. The combined mass of these components was 6.02 kg. Then, we simply utilized Equation 9 to solve directly for a new system stiffness.

$$k = {w_n}^2 * M \quad (9)$$

Where w_n is the FFT obtained resonant frequency [rad/s] and M is the empirically weighed system mass [kg]. This equation led us to the overall composite system stiffness value of 471,530 N/m, which we believe to be more true to the overall system behavior than the shaft stiffness alone. It is likely that shaft stiffness adds linearly with other components' stiffnesses to give this result [30].

Damping Ratio

The damping ratio is a unitless measurement describing how system vibrations decay after a disturbance. We initially estimated our damping ratio as 0.05 from research into SolidWorks simulations and vibrating dynamic systems [14, 26]. To more accurately identify this ratio, we iterated upon the mathematical model. We compared vibration magnitude, in G's, of the model to the data and tweaked the damping ratio value to produce as close of a match as possible. After achieving the results which we presented both at Design Expo and in this report for vibration magnitude at a 45 Hz motor speed for 0 added mass, 1.3 grams added mass, and 2.6 grams added mass, we concluded that the damping ratio was around 0.022. This was the final value used in the aforementioned models. The closeness of the data to the model speaks to the validity of this damping ratio. It also is quite close to zero, which is what we expected for a vibrating system like this based on our prior research [14, 26], further corroborating this value.

System Mass

We determined our total system mass to be 6.02 kg. This was done by weighing the motor, motor shaft, motor mounting plate, and impeller on a precision balance scale. This is a drastic increase in comparison to the 1.118 kg that was estimated using our Solidworks CAD assembly. The CAD assembly was not accurately predicting the weight of the motor which we believe to be essential for accurately predicting the system's behavior. The copper windings and other components in the motor are very heavy, and the CAD could not account for this as it assumed the motor to be a solid aluminum block. Because we believe it is reasonable to assume that the motor, shaft, impeller, and motor mount plate are the components which displace the most during vibrations from simple observation, we are confident in this 6.02 kg value. For the model we used and the assumptions which it makes, this mass value should be accurate.

Resonant Frequency

We have updated our resonant frequency estimate to be 44.554 Hz. This was obtained by performing a Fast Fourier Transform on empirical data with various eccentric masses attached to the impeller. We further validated this estimate by performing a frequency sweep on the system and observing amplified behavior around 45 Hz. This updated estimate significantly differs from our original 64 Hz estimate. This value comes from our

FEA studies as well as directly from empirical resonant frequency calculations using stiffness and mass values pulled directly from HyperWorks. We believe that this variance can be explained by the fluid effects introduced from the larger impeller [35]. Ultimately, the resonant frequency pulled from the empirical data gives a more accurate representation of what the true resonant frequency of the system is. Additionally, it is worth noting that the resonant frequency of the system is equal to the critical shaft speed of the system [12,14,33]. Because of this fact, we removed critical shaft speed from our list of critical system parameters as it is redundant when we already have resonant frequency.

DISCUSSION

Problem Definition

We made significant progress developing an accurate mathematical model this semester given the time and resource constraints that come with having less than 12 weeks to work. If given more time and resources there are multiple questions that we would want to explore in order to improve our models validity. The first question we would want to address is the advantage/disadvantage of a 3 degree of freedom system compared to the 2 degree of freedom system that we used. How much improvement in accuracy would we see relative to the increased complexity of the system and is it even worth the additional work given what Heller is looking for? Further research into this along with additional conversation with experts at the university could be beneficial to reaching a conclusion. The next thing we would want to consider would be the potential effect that airflow has on the system. In one of our sponsor meetings we discussed how the 10 inch impeller could be introducing pressure imbalances that we weren't considering. Given more time and resources we would research into how to implement fluid effects into our system, we would also consider building a larger module and collecting empirical data to see if that improves the fidelity of our model. Finally, we would want to look into how to incorporate multiple frequencies into the model. When running frequency sweep tests we noticed peaks at multiple frequencies which suggests that additional analysis is needed beyond just the 45.554 Hz resonant frequency that we obtained.

If we had more time to collect data and better define the problem, we would have explored more thoroughly different methods of modeling this system with Finite Element Analysis. A different solver such as ANSYS or ABAQUS could have produced different results which we could have compared. This would have allowed for even more iteration and fine-tuning within our design process, in regards to the FEA component. If we had more bandwidth, this method of analysis could have been explored in more depth.

Reflecting back on our data collection throughout the semester, we lost a significant portion of time to shipping of the accelerometer and the 10 in impeller setup on site. As a result, a significant amount of our testing had to be condensed down to make the most of the time we had left in the semester. If the team had more bandwidth, we would've considered implementing an encoder onto the motor to accurately display the speeds at which the motor was operating at instead of relying solely on the controller provided. While the equipment was accurate, the variability in operating frequency was controlled through a twist knob that had a user error of +/- 0.1 Hz. This resulted in small fluctuations in speed between ramping up/down during sweeps and as a result increased the time it took for the motor to obtain the desired steady state operation at a given motor frequency. One last adjustment in the physical setup the team would have explored further would be the impact of the clamping setup. Our clamping setup was validated and confirmed by Heller but no further exploration was done on the impact that various clamping setups have on the damping and as a result, the vibration data collected from the accelerometer.

Outside of modifying/changing the testing setup, the team would have explored various other types of tests as well as building out a more robust set of data. While we were able to perform frequency sweeps and identify resonant frequencies in the system across varying eccentric masses, we were unable to perform any high speed camera testing which would have been useful for characterizing exact displacement magnitudes in our vibration and could have helped further flesh out our math model. With the additional bandwidth, we could have validated our data by collecting and using longer time intervals in constructing our FFT, which would have reduced error and unexplained spikes in our data set.

Design Critique/Risks

As mentioned in the problem definition sections above there are many things that we would have done to improve our design if we had the time and resources. These improvements are based on weaknesses in our models and empirical testing procedure. Before critiquing our design its strengths should be acknowledged.

The mathematical model we developed can sufficiently predict system behavior close to the resonant peak of the system. This allowed us to vary different model parameters in order to propose a recommendation for Heller. Another strength of our mathematical model is its simplicity. We were able to easily vary. different model parameters and quickly test different combinations.

Our FEA model was able to accomplish complex calculations that never would have been possible by hand. That is the main strength of FEA in general; it can simulate and approximate systems which would take enormous amounts of time to solve analytically. It can also account for a very wide variety of conditions, constraints, and environmental factors with the proper inputs.

Our empirical data was critical for completing the system identification of the impeller module. Without the vibration data, there would have been no raw data to perform an FFT on and ultimately stifled the ceiling of our project. Additionally, having the empirical data helped us identify a natural frequency that went unnoticed in our FEA and mathematical models, at 44.56 Hz. Had we not collected this data or had this, it would have been left out of our recommendations and would have been missed by Heller until it was discovered later down the line, saving Heller from costly redesigns late in their timelines.

Our mathematical model's biggest weakness is its simplicity. We hoped that our model would predict acceleration within 10% accuracy of empirical data. In actuality our model predicted acceleration to 15-20%. We believe that this can be explained by the model over simplifying the system. The easiest improvement that can be made is implementing a 3 degree of freedom model instead of our current 2 degree of freedom model. We believe that this would help improve the accuracy of our model. Additionally, our model fails to account for potential fluid dynamic effects introduced by the larger impeller. While we have no immediate plan for incorporating this into our mathematical model we believe that it would improve the fidelity of our predictions. Finally, our model can only account for one resonant frequency at a time. As mentioned above we noticed peaks at multiple frequencies when conducting a frequency sweep. Our model is unable to account for the spikes at 60 Hz and thus is limited in its capabilities. We would expect an improved model to account for multiple frequency peaks like we saw in our empirical data.

The biggest weakness of our FEA model is that it is only as good as the constraints which we supplied to it. The saying "Garbage In, Garbage Out" is true with almost any modeling system but especially true with FEA. If our assumptions about translations, bearing constraints, or potentially material were incorrect then our simulations might run and produce a result, but we would not be able to tell where the mistake lies or the effects that it had. It is why FEA must always be backed up by thorough mathematical analysis when possible, and especially compared to empirical data. TO this end there are a few main changes we would like to make to this FEA study to make it more accurate and match more closely to the data as well as our math model.

Firstly, we would incorporate computational fluid dynamics (CFD). As discussed earlier and with our sponsor Heller Industries [35], fluid flow has a great impact on the system. As the impeller gets larger and closer to the walls, turbulence and stagnation begin to affect the motion of the impeller. Running a simulation with this in mind would likely result in resonant frequencies closer to our empirically obtained 44.55 Hz. The currently predicted resonant frequencies between 74 and 80 Hz are not accounting for fluid flow. Temperature variations could also be accounted for in a fluid flow study, which may further alter the resonant frequency.

In addition, we suggest a re-evaluation of the constraints and assumptions utilized in our SolidWorks frequency analysis studies. If displacement of the motor could be empirically measured then this could feed the translation constraints used in the simulation. We also suggest modeling the internal motor coils and components, as they have significant mass which affects the resonant frequency of the system. Use of another software package like ANSYS or ABAQUS could allow for these more advanced simulations (CFD/thermal and the more complex constraints), so we believe this project should be translated into one of those two softwares.

Regarding the empirical data, the biggest weakness was that we were unable to take many repetitions of the data due to time limitations. Future work should include retaking this data in a stable environment to validate our tests. Beyond just retaking data, we would recommend recording data at various sights/plants around the world, as different plant conditions could impact the damping of the system. Since damping is a difficult concept to model and measure, measuring the variance in acceleration/position from manufacturing plant to plant can help better define how similar/different setups are and if there is refinement that needs to occur on a location/location basis. Additionally, high speed camera testing as mentioned above can help further explore the impact of the natural frequency that is encountered at 44.56 Hz.

RECOMMENDATIONS

Heller Industries has requested recommendations from our design team on how to reduce vibrations in the module design. Our recommendations aim to move the resonant frequency away from the forcing frequency, ideally by at least 25% [35], which would reduce the vibration amplitude and increase the system's stability. We also did not propose solutions that involve material changes or significantly alter the module design space, as they are expensive and not feasible due to supply chain constraints.

One way to move the resonant frequency away from the forcing frequency is to vary the mass of the system. Adding mass to the system shifts the resonant frequency lower, which is undesirable as it leaves the resonant frequency within the standard operating range of motor speeds. Instead, we would aim to decrease the system mass and shift the resonant frequency upwards, out of the operating range. The amount of mass needed to move the resonant frequency by 25% depends on the system's characteristics. In this case, removing 3.01 kg to the module would move the resonant frequency from 44.5 Hz to 63.01 Hz, a 41.4% increase. Figure 27 below visualizes how changes in mass shift the resonant frequency. Its main purpose is to demonstrate the expected behavior of the system when mass is either increased or decreased.

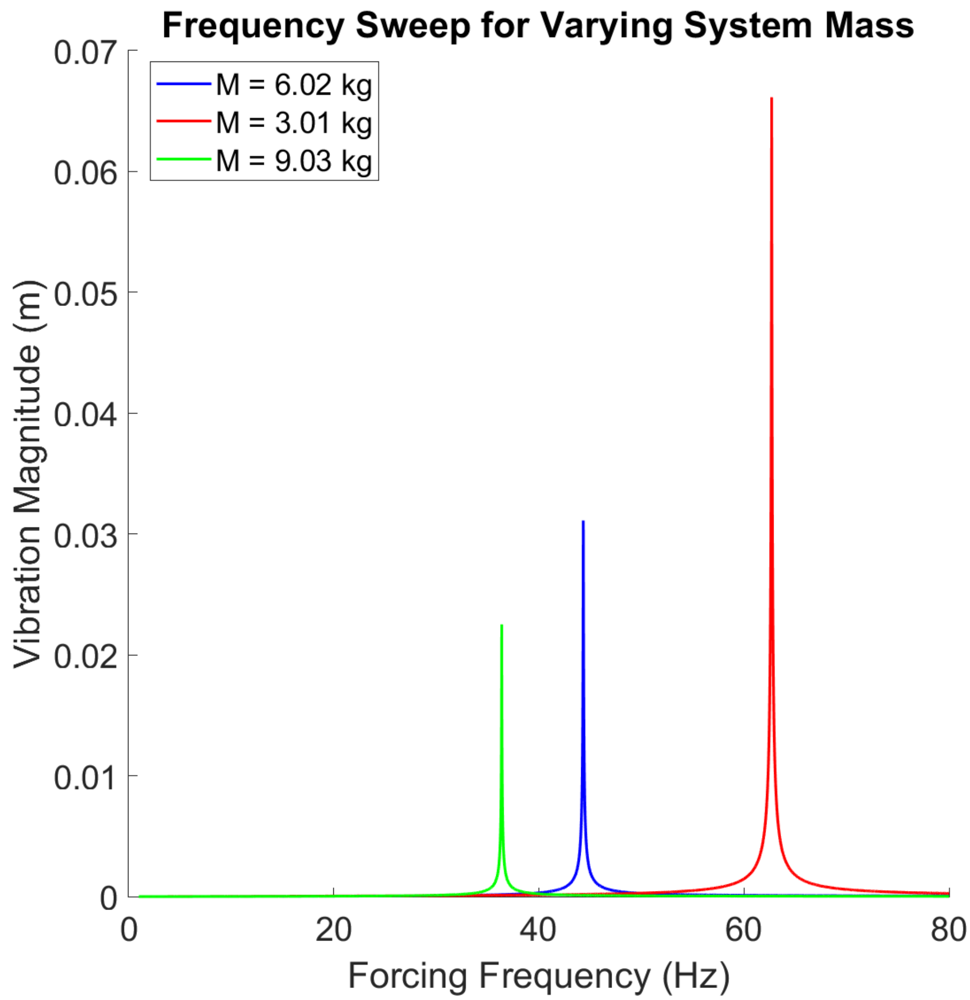

Figure 29. Bode plots which represent a shift in resonance frequency due to a change in the system mass. The blue plot represents the current mass of the system, 6.02 kg. The green plot represents a shift when the mass is increased to 9.03 kg. The red plot represents a shift when the mass is decreased to 3.01 kg.

While this sounds promising, lightweighting can be immensely difficult. This is especially true in a system made from mostly off-the-shelf parts and already light sheet metal. As such, we will not pursue lightweighting as the only method of shifting resonant frequency. There may be some viability of combining moderate lightweighting with other parameter changes, but that will be discussed later in this section.

Another way to move the resonant frequency away from the forcing frequency is to increase the stiffness of the system. Increasing the stiffness of the system shifts the resonant frequency higher. Based on the current design, increasing the stiffness from $4.72\text{x}10^5$ N/m to $7.07\text{x}10^5$ N/m would move the resonant frequency from 44.554 Hz to 54.56 Hz, a 22.4% increase. Figure 30 shows the effect of this increase in stiffness, as well as a proportional decrease in stiffness for comparison.

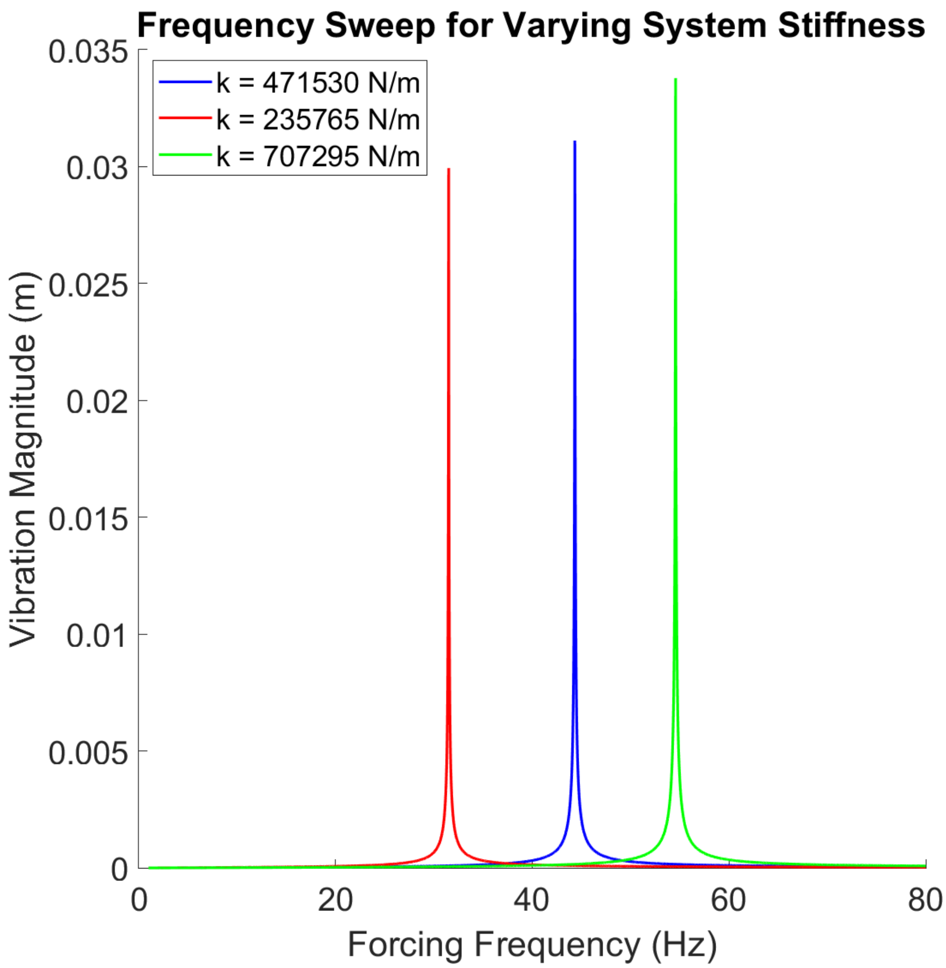

Figure 30. The effects on resonant frequency and system vibration behavior for both a 50% increase (green) and 50% decrease (red) in system stiffness. From this graph it is clear that increasing system stiffness will shift the resonant frequency out of the standard operating range of this convection reflow oven module.

To increase the stiffness, we can use thicker sheet metal for the box module because it contributes to the overall composite system stiffness. Increasing the sheet metal thickness from 1.2 mm to 1.6 mm would increase the stiffness and move the resonant frequency higher. This increase in thickness was determined from internal conversations with Heller [37]**.** Additionally, we can increase the number of motor mounts on the module from three to six. The additional mounts act as very stiff linear springs and can increase the system's stiffness. We note that 6 is an arbitrary number discussed with Heller Industries during a sponsor meeting [37]. The actual number of mounts needed may exceed 6, but we feel that 6 is feasible within the given module design space. Thus, it should be a valid starting point for exploration of the effects of the mounts on system stiffness.

Another solution is to add stiffness bars or structural braces across each module and/or between modules. Based on Heller's data, stiffness bars across each module helped reduce stiffness substantially. Combining this with one or both of the other approaches would greatly increase system stiffness and move the resonant frequency well beyond the current resonant frequency.

It is also worth a quick mention that our team believes there may be constructive interference between modules, when they are lined up together in the reflow oven. The data provided to us indicates higher vibration magnitudes in the modules closest to the center of the machine. We hope that shifting the resonant frequency of all modules high enough will reduce this effect, but analysis of this interference may be required if this does not work.

We provide a final note on damping. The damping ratio proved to be quite difficult to identify exactly in this system. We narrowed it down to 0.022, but as suggested in the discussion study more work may be needed to verify that this is exactly right. We will show the effects of damping in Figure 31

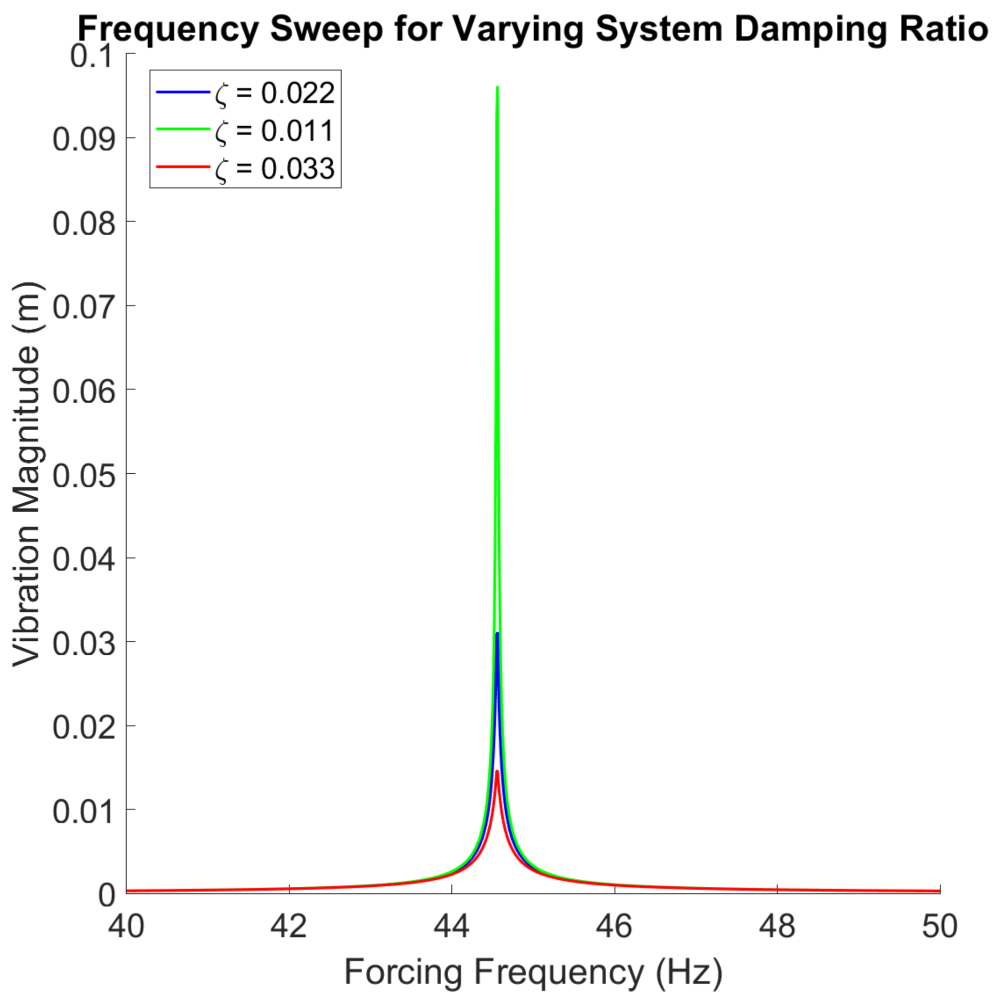

Figure 31. The effects of a 50% increase (ζ=0.033) and a 50% decrease (ζ=0.011) to the damping ratio of the system. The resonant frequency is not affected by this change but the vibration magnitude around resonance is greatly affected.

Because we had difficulty identifying damping, and the fact that we could not perform any empirical analysis to identify a value for the damping ratio, we cannot make any specific recommendations to change the damping ratio in the system. It is obvious from the graph above that an increase in damping will decrease vibration magnitude. The use of viscoelastic materials should theoretically increase damping, but we do not have enough evidence to make specific suggestions. With this all in mind, future work regarding this damping ratio and its effects on the system should be pursued. A more robust math model may be able to help identify the damping ratio more precisely.

An optimal combination of mass and stiffness values can be determined by iterating the mathematical model with different inputs. The recommended mass of the system is 4.85 kg, a 1.17 kg decrease from the original mass. The recommended stiffness of the system is $6.2\text{x}10^5$ N/m, a $1.485\text{x}10^5$ N/m increase from before. It should be noted that the model only considers resonant frequency in its optimization so its recommendations should be taken with a grain of salt as the model represents an oversimplified system. Ideally the model would better characterize the system and its recommended optimal combination would also consider ease of implementation and cost for Heller. Figure 32 below shows our recommended optimal mass and stiffness values.

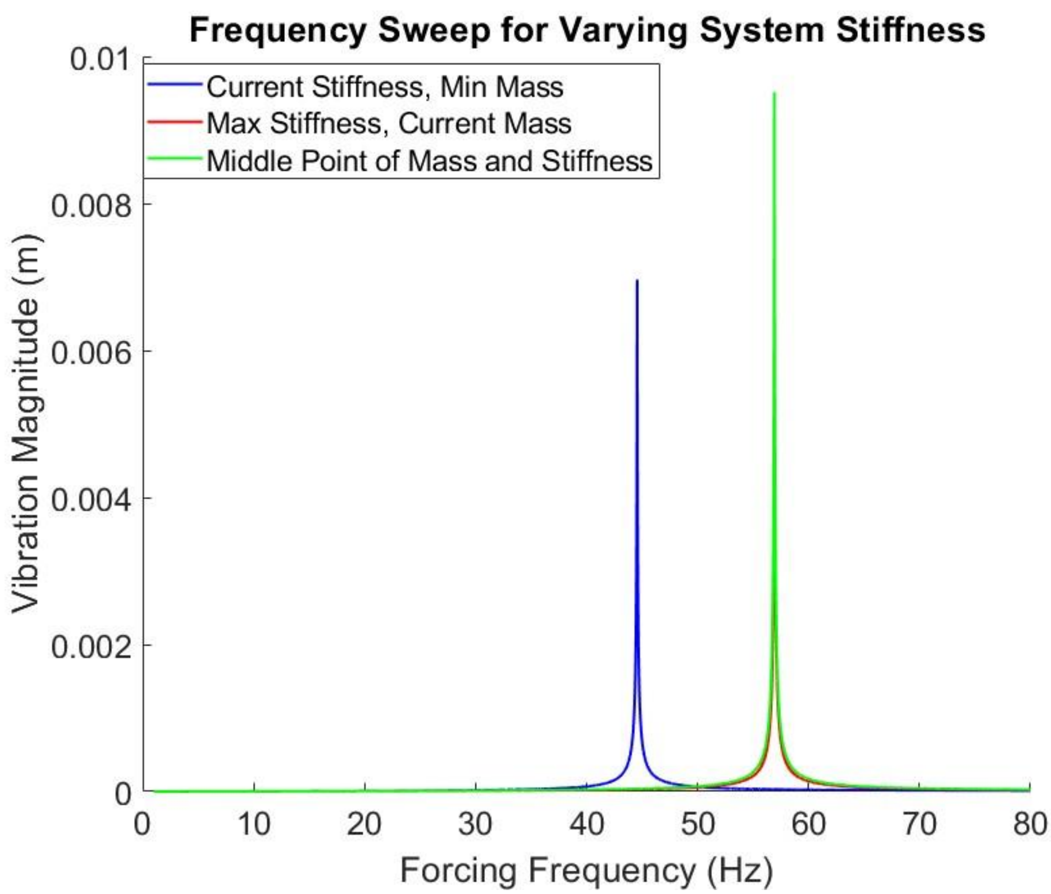

Figure 32. Plots representing the shift in resonance frequency given the optimal combination of mass and stiffness changes. The optimal mass value is 4.85 kg while the optimal stiffness value is 6.20×10^5 N/m (Green Line seen above). The other two solutions shown above showcase decreasing the mass to 3.68 kg while keeping stiffness constant (Blue line), and keeping mass constant and increasing the stiffness to 7.70×10^5 N/m (Red line, same shape as green line but lower vibration magnitude; 0.004 opposed to 0.01). In the cases where mass is minimized to 3.68kg and stiffness is increased to 7.70×10^5 N/m, we recognize that this is not completely feasible to implement, which is why we are suggesting a blend of the two of these. Additionally, another point to note is that during this process the damping was held constant. Increasing damping is difficult to quantify as discussed above, but moving forward would be an important area of future work.

In the future, some sort of materials or structural study to produce these exact parameter values in the system would be ideal. Validation of these exact numbers would prove useful in the selection of sheet metal, the number of motor mounts, and also the size and shape of any structural stiffening bars added to the modules.

REFLECTION

Reflecting back on our project, we learned a lot and refined skills that we brought in from other undergraduate courses. When we were originally tasked with this project, we believed to have a simple design modification problem centered around a change in design for Heller Industries. As our understanding of the problem evolved from Design Review 1 to Design Review 2, we found ourselves realizing that rather than trying to apply and physically modify the system, our time was better spent and more effective if we shifted our focus on modeling the problem and determining the origin of the vibration (which is what Heller desired). From this we stepped into a different direction than most ME 450 projects, rather than designing and building a new system, our focus was primarily on the engineering analysis of the existing system before using that information to derive parameters that impacted the vibration and could lead to design improvements down the road. From this shift, we changed our perspective and realized that instead of focusing on the various impacts of our project directly, it was more valuable and relevant to examine downstream effects on our project and the corresponding factors that tied into our project. This meant we had to align on how our work fits into the broader picture of Heller's reflow ovens, which carry social, environmental, economic and global implications.

Stakeholder Analysis

The stakeholders involved with this project have been categorized into three groups depending on the degree at which they are impacted by the problem. These stakeholders are shown in the Stakeholder Map below.

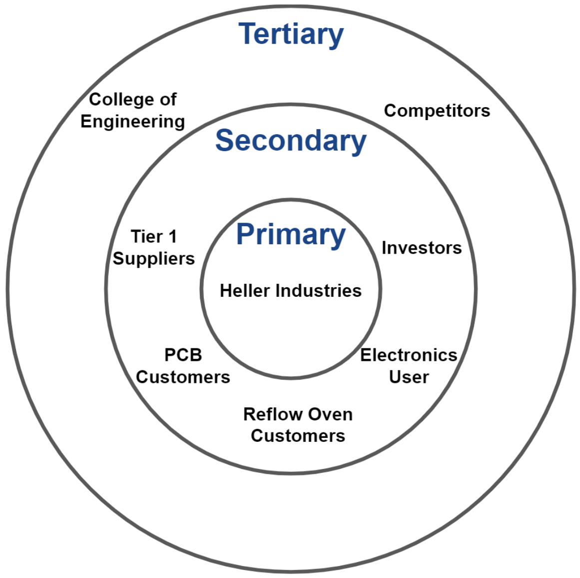

Figure 33. Map encompassing all stakeholders involved in the project.

We first defined the Primary Stakeholders. These are groups whose lives or work are directly impacted by the problem and/or the development of a solution [13]. The Primary Stakeholder for this project is Heller Industries. Expanding out the scope of the problem we defined our Secondary Stakeholders. These are stakeholders who are part of the problem context but may not experience the problem themselves and/or may not be directly impacted by a solution [15]. The Secondary Stakeholders for this project include customers of Heller's convection reflow ovens (usually to produce PCB and other electrical components), Heller's tier one suppliers (companies supplying the parts for their convection reflow ovens), Heller's investors, electronics manufacturers who are purchasing components that were manufactured in Heller's ovens, and customers of the aforementioned electronics manufacturers [8]. Finally, in the broadest scope of the project we identified the Tertiary Stakeholders. These are groups who are outside of the immediate problem context but may have the ability to influence the success or failure of a potential solution [5]. The Tertiary Stakeholders for this project include The University of Michigan's College of Engineering, Regulatory and Standards Organizations, and Heller's competitors in the convection reflow oven market [8].

Most of the stakeholders listed would benefit from the development of a solution to Heller's vibration problem. A reduction in vibration would result in most of the stakeholders making more money due to the performance improvements that a 10 inch impeller introduces. The only stakeholder that would be significantly negatively affected would be Heller's competitors as Heller would have a stronger competitive advantage than before.

Public Health

The impact that this design project has on public health, safety, and welfare is minimal. The current vibration problem poses no imminent danger to the operators of the convection reflow ovens, nor does it result in a dangerous PCB being produced in these ovens.

Social Implications

While the high vibration magnitude in Heller's convection reflow ovens is primarily an internal problem, the development of a solution does have social implications. These social and societal implications mainly involve society's reliance on the tech industry through the use of smartphones, computers, and other products [10]. An improvement in both performance and efficiency in Heller's ovens would lead to faster production of this technology potentially also making the technology cheaper. Solving this vibration problem keeps society on track to continue rapidly advancing technology and its integration into the lives of people in the first-world.

Environmental Implications

The majority of our project efforts are based in kinematic modeling and experimentation and will have minimal environmental implications. If our recommendations were to be taken and prototyped, the only environmental impact they might have is the production of waste at the end of life of the convection reflow oven. To this end, we only recommended recyclable metals for use in any potential solutions. This ensures that our recommendations do not have unintended environmental consequences, if they are implemented.

Economic/Global Implications

In the global marketplace, an accurate method to solving this vibration problem would be greatly useful to Heller Industries. It would give them a competitive advantage in the convection reflow oven market, further solidifying their status as the market leader. Furthermore, using this modeling method is relatively inexpensive compared to trial and error problem solving. This would save any company valuable time and money, allowing them to more efficiently solve the vibration problem and dedicate resources to other projects, where necessary.

Inclusion and Equity

We did not face significant issues regarding power dynamics during our project. We were treated with the utmost respect by Heller Industries and their employees, and for this we are grateful. There were not any points where we felt that we were being pushed or influenced towards a particular solution or method by our sponsor due to a power imbalance. Additionally within our group we did not have any issues with power dynamics. We all treated one another with respect and as equals. Every team members' opinions and ideas were equally valuable. Because of this, we believe we were able to consider a diverse range of viewpoints which led to creative thinking and solutions. We did at times experience a power dynamic with the professors who we sought advice from during this project. There were times where we felt compelled as a team to try the methods certain professors introduced to us, because as students we were used to doing so for the past 16 years we have been in school. In a few instances, this led us down a path of researching a particular method which we ultimately ended up not using. It did not end up having a large impact on the final outcome of the project, but it was worth noting that we did feel somewhat pressured to explore specific forms of analysis which may have limited us from pursuing others.

With regards to stakeholder engagement, the main stakeholder which we interfaced with was, again, Heller Industries. We made sure to listen attentively during meetings and consider all of the previous work and ideas which Heller Industries had developed around this vibration problem. We took detailed notes and asked relevant questions to ensure we fully understood their viewpoints. As a team we did the same for any meetings we had with professors and other faculty. We were respectful, attentive, and made sure to ask relevant questions to get the information we needed and not waste their valuable time. Because our project did not really reach the scope of interfacing with end users of Heller's convection reflow ovens, we did not have to consider input from sources other than Heller and our professors/other faculty members.

Personal Background

Cultural, privilege, identity, and stylistic similarities and differences within a project team can significantly influence the approaches taken by a team throughout an engineering project. Every member in our team has had a similar upbringing. We all grew up in southeast Michigan and are all predominantly Caucasian; however, there were still variations in cultural, identity, and stylistic backgrounds that affected our team's dynamics over the course of the semester. Cultural differences manifested themselves in ways, such as, different communication styles and decision-making processes. For example, some team members had a more direct and forward communication style, while others were more passive and utilized the listening aspect to conversation. These differences can be attributed to different roles within the group. Those who were more forward typically led conversation while the more passive listeners typically took notes and kept track of any information shared.

Privilege and identity also played a role in shaping our team dynamics. Privilege can create disparities in access to resources or opportunities, which can affect the ability of team members to contribute equally to the project. We were potentially given more freedom/less oversight because we trend the University of Michigan. It is possible that any sponsor would have worked with a group from another university differently. Michigan has a reputation of excellence and we may have benefitted from the stereotype of being "Michigan Engineers".

Stylistic differences in how team members approach work also impacted team dynamics. Some team members preferred to work collaboratively, while others preferred to work independently. This influenced how tasks were delegated and how productive each of us was during different work sessions. Some members were able to delegate and accomplish more during team meetings while others were more easily distracted. This was also true for independent work sessions where some were able to tackle lots of work while others felt like more group input was necessary.

Overall, it was essential for us to be aware of and respectful of each other's cultural, privilege, identity, and stylistic differences to ensure that the project was executed smoothly. Whether it was distributing work or deciding if work would be done virtually or in-person we had to make lots of compromises. These compromises contributed to the success of the group as a whole, even if some individuals weren't as productive as they would have been individually. It is also crucial to recognize the potential benefits that come with diversity in the team. These benefits include varied perspectives, different experiences, and innovative solutions to problems. By valuing and leveraging these differences, our team was able to achieve success this semester.

Ethics

We did not face many ethical dilemmas throughout the course of this design project. The project focused primarily on engineering analysis and characterization of the vibration problem being experienced by Heller Industries in their convection reflow ovens. As a team we made honest decisions and reported the actual data and results of our models, whether they agreed with our expectations/specifications or not. We may not have had as accurate of a model as we hoped, but reporting the honest numbers highlighted that this system was more complicated than we anticipated. Moving forward, this indicates the level of care that future engineers need to take who work on this project. It would be very easy to fake a number in a critical spot to make the results look more agreeable, however this would not accurately characterize the system nor solve the vibration problem. It is better to recommend adequate future work than to be dishonest and hope future engineers do not notice. This is a valuable lesson both here at the University of Michigan as well as out in the field.

CONCLUSION

To summarize, our engineering project team has worked diligently with Heller Industries to address the issue of impeller vibration in their convection reflow ovens. The switch to a 10 inch impeller from an 8 inch impeller has introduced undesired vibrations, which directly affects Heller and their customers across various industries. Our team set out to develop a mathematical model and a computational model using finite element analysis to characterize the system and to use these models to propose recommendations that could reduce the vibration.

We worked with faculty to develop models that were accurate within 20% of each other, as well as to empirical data which we collected. Over the course of the semester, we refined these models to better predict the acceleration behavior of the system. We also conducted various tests to collect empirical data, which we used to refine the models and obtain input parameters for the mathematical model.

The mathematical model we developed uses a simplified 2-DOF mass-spring-damper system as its foundation. The system is driven by an eccentric mass a set distance, e, away from the center of the impeller. Displacement and acceleration equations were derived using the system's equations of motion and implemented into our Matlab code. This allowed us to change model parameters and analyze their impact on system acceleration.

Modeling parameters were obtained using a variety of methods such as empirical data collection, finite element analysis, and thorough research.

Finite Element Analysis was conducted by fixing the motor mount and plate in place to prevent translation. An input torque was applied to the motor and a modal analysis was performed which provided the displacement magnitude and the fundamental frequency of the system. The fundamental frequency predicted by the FEA study was compared to the frequency obtained from our fast fourier transform. The variance between the two suggests a fluid dynamics aspect is at play. Our model does not account for this and could be improved by performing a CFD study.

Empirical data was collected by conducting a series of tests on the impeller module to measure the acceleration magnitude operating at different conditions. All tests were conducted with the module clamped between two rigid tables in both inverted and non-inverted orientations. Measurements were taken when the system was operating with a 6 inch, 8 inch, and 10 inch impeller and when eccentric masses of 1.3 and 2.6 grams were placed on a vane of the 10 inch impeller. Additionally, a fast fourier transform was conducted to obtain a resonant frequency of 44.554 Hz.

Through our analysis, we found that shifting the resonant frequency away from the forcing frequency was the most effective way to reduce the magnitude of vibrations. This could be achieved either by increasing the system mass or increasing the stiffness. Using our models, we obtained an optimal combination of mass and stiffness values, which were found to be 4.85 kg and 6.2×10^5 N/m.

It is important to note that while our models are far from perfect, they can help predict how the system should behave, which we believe can be useful to Heller Industries. Our findings provide valuable insights into the dynamics of the system and suggest solutions to reduce vibration, which can improve the performance of the convection reflow ovens and benefit both Heller and their customers. We recommend additional refinement and research into the vibration problem. In order to fully characterize the system, fluid dynamics must be accounted for. Incorporating this into the mathematical model and performing CFD analysis are our two main suggestions for improving the fidelity of the study.

ACKNOWLEDGEMENTS

Our team would like to give thanks firstly to our sponsor Heller Industries. Specifically, we want to thank Dave Heller, Jim Neville, and Erin Sun for all of their support, ideas, and valuable feedback throughout the semester. In addition, we would like to thank Professor K. Alex Shorter. His continued support and feedback across the entire semester were crucial to our success on this project. We would also like to thank Robert Nawara for his feedback on our various design reports and presentations, as well as Professor Okwudire and Professor Hulbert for their time in discussing aspects of our modeling approach. Finally, we extend thanks to the ME 450 Instructional Staff for their assistance in the assembly room and during the Design Expo.

APPENDICES

Appendix A: Concept Generation

To generate modeling concepts for the vibrating motor and impeller system, our team began by decomposing the vibrating system into key parameters. Using prior knowledge from ME 240 and ME 360, we determined that stiffness, damping, and moment of inertia are the fundamental physical parameters that we could change in the system to reduce vibrations. We then listed physical aspects of the system which could be altered to affect each of these parameters. The figure below is the result of this brainstorming session.

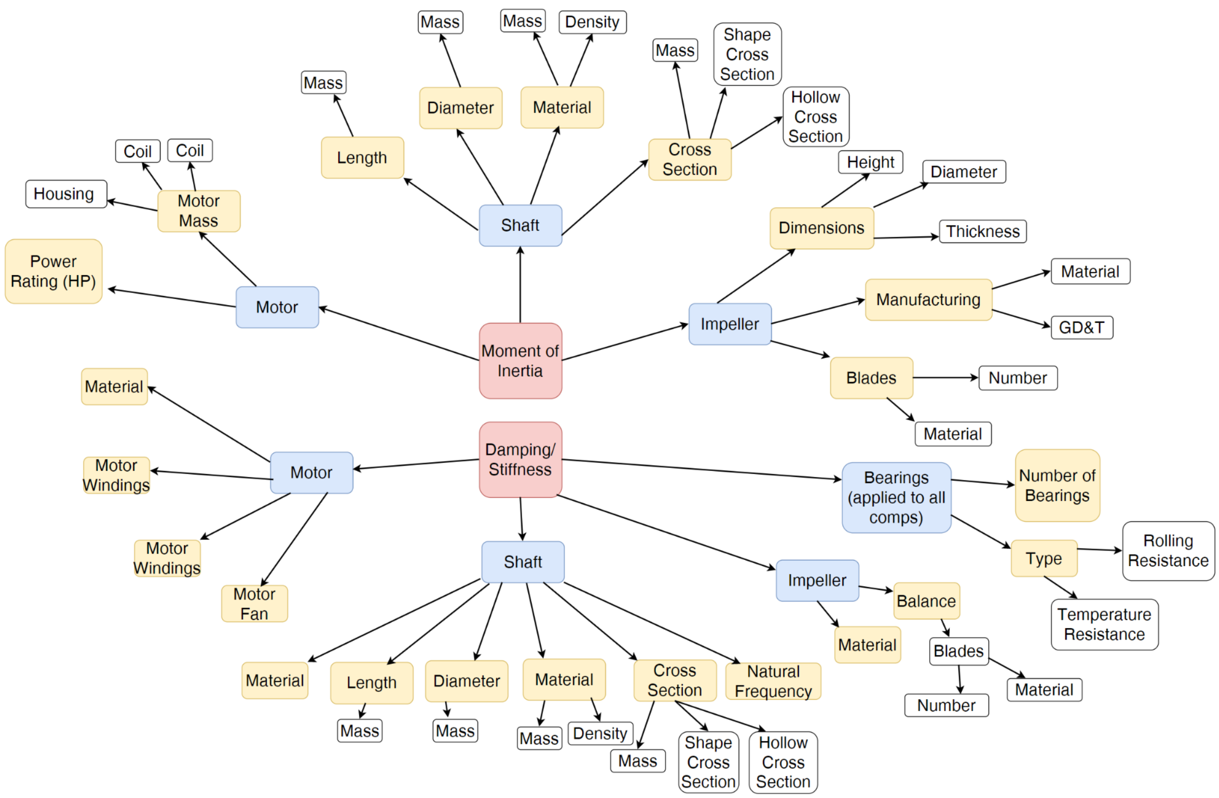

Figure 1. Visualization of our thought process for mapping physical system characteristics to vibratory concepts. Stiffness and damping are grouped together [10]

After this brainstorming and functional decomposition was done, we sought to fill in any gaps in our knowledge of vibrating system parameters. Specifically, we consulted with multiple professors here at the University (Hulbert, Okwudire, Bell IV, and Shorter), as well as our sponsor Heller Industries for any additional expertise about rotational vibration. Professors Okwudire and Shorter confirmed that characterizing stiffness, damping, moment of inertia, and resonant frequency would be critical for proper modeling of this problem [23]. Additionally, Heller Industries suggested that we characterize critical shaft speed, which through research we determined will simply be the forcing frequency at which vibrations begin [18, 25]. With these parameters in mind, our team began to list methods of modeling and/or analysis that involve use or evaluation of these parameters. Our preliminary list of modeling concepts needed to be ranked, so we chose to do so with a

Pugh chart. This chart considered the ability of each concept to meet our project requirements as well as a few additional requirements that were developed based on availability to our team, complexity of their interface, and prior modeling experience. Shown below is a Pugh chart ranking the generated concepts on important criteria, with the criteria explained in more depth in the caption.

Modeling Ideas	Versatility	Run Time	Fidelity	Resources	Cost	Interface	Experience	Total
Weight	4	3	5	3	2	4	3	**24**
FEA using Hyperworks - using EIGRL	1	1	1	1	1	1	1	**24**
FEA - Hyperworks - Modal Analysis	1	1	1	1	1	1	1	**24**
Experimental Accelerometer Data Collection	1	1	1	1	0	1	1	**22**
Solidworks Motion	1	1	1	1	1	1	0	**21**
Siemens NX/Simcenter	1	1	1	1	1	0	1	**21**
Okwudire mathematical model	1	1	1	1	1	0	1	**21**
FFT to characterize frequency response	0	1	1	1	1	0	1	**16**
Matlab (mathematical)	1	0	1	1	1	0	1	18
Adams Multi Body Physics Model	1	0	1	1	1	-1	1	15
COMSOL Analysis	1	0	0	1	1	0	1	13
Ansys	1	0	1	0	1	0	0	12
Autodesk Simulation	1	0	1	1	1	0	-1	12
Abaqus	1	-1	1	1	1	-1	0	9
LS Dyna	1	0	1	1	1	-1	-1	9
Catia V5 FEA Modeling	1	0	1	-1	1	-1	1	9
Multiple DoF mass-spring-damper	0	1	0	1	1	-1	1	8
Matlab (FEA)	0	-1	1	1	1	-1	1	7
SimSolid	0	1	1	0	0	0	-1	5
Single DoF mass-spring-damper	-1	1	-1	1	1	1	1	4

Figure 2. Pugh chart ranking all of the modeling and/or analysis methods that we generated**.** Versatility refers to the ability of the method to consider multiple different parameters and accommodate a wide variety of input types. Run time ranks relative time to solve each model based on past experience and research [22]. Fidelity ranks relative accuracy of each method of modeling. Resources ranks our access to university resources for each particular method. Cost considers if we have access to this resource through the University or if we would need to potentially pay for it. Interface and Experience are more subjective and rank our thoughts about the interfaces of each method as well as our level of past experience.

The above method of down selection helped us remove concepts which would fail to characterize aspects of our complex vibrating system, or were otherwise out of line with our project and goals. Highlighted in blue in the Pugh chart are the concepts which we felt would be most beneficial for us to use to model this system, however we still felt that another level of analysis was necessary. We wanted to determine (based on prior knowledge, literature, and consultation with experts) which parameters could be characterized by each modeling method. Based on this, the following chart was developed

Table 1. A chart ranking each method on its ability to characterize/determine each of our 5 important system parameters plus overall vibration magnitude (acceleration) [22].

Modeling Methods	Stiffness	Damping	Moment of Inertia	Resonant Frequency	Critical Shaft Speed	Maximum Vibration Magnitude
FEA using Hyperworks - using EIGRL	Capable	Incapable	Capable	Capable	Capable	Capable
Experimental Accelerometer Data Collection	Incapable	Capable	Incapable	Incapable	Incapable	Capable
FEA - Hyperworks - Modal Analysis	Capable	Incapable	Capable	Capable	Capable	Capable
Solidworks Motion	Capable	Incapable	Capable	Capable	Capable	Capable
Siemens NX/Simcenter	Capable	Incapable	Capable	Capable	Capable	Capable
Okwudire mathematical model	Capable	Capable	Incapable	Capable	Capable	Capable
FFT to characterize frequency response	Incapable	Incapable	Incapable	Capable	Capable	Capable

This additional chart confirmed the fact that multiple modeling methods would need to be used to characterize this system effectively. No single method or modeling or analysis could accurately determine all five important parameters, plus the overall acceleration magnitude. With all of this information in hand, we began to combine our modeling concepts in ways that incorporated all five important parameters and then give an accurate prediction of modeling the kinematics of the vibrating system. This led us into the development of our five key modeling approaches to our system.

Concept Selection

Historical Alpha Design

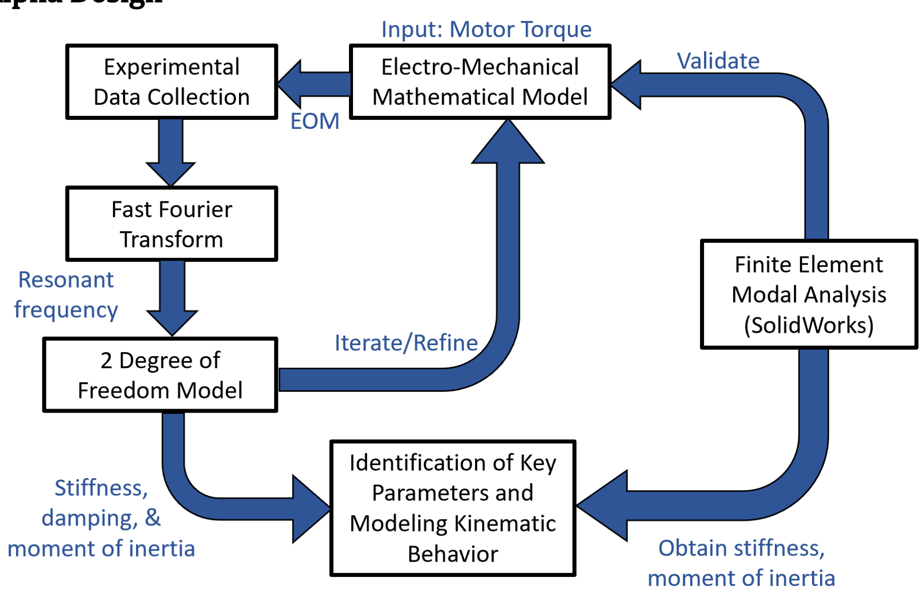

Figure 3. Historical Alpha Design from DR2 and DR3.

Our Alpha Design is centered around numerical derivation in addition to supplementary Finite Element Analysis. As seen below, we are planning to derive the differential equations of motion of the system, then identify the key system parameters associated with the behavior of our system. Utilizing empirical data collection and Finite Element Analysis software, we can obtain the stiffness, resonant frequency and damping of the system in order to understand how the system behaves, i.e. if the impeller motor system is an over or underdamped system. We can also estimate the moment of inertia with a combination of empirical measurements and trial-and-error.

This concept was developed with feedback from our professors, and stayed independent of direct influence from Heller industries. As part of our concept generation, our selection process all converged to a similar approach on types of modeling to represent our problem, but varied in how to approach and tackle the problem at hand. As seen in the additional concepts we have created, our modeling approach is similar in nature utilizing mathematical models and empirical data to obtain relevant parameters in order to identify the kinematic behavior of the system. The major difference between our Alpha Design and other concepts is in the in-depth validation and comprehensiveness of the modeling. We believe that our Alpha Design is robust enough to withstand rigorous analyzation as a result of the diverse modeling approaches- numerical, empirical and Finite Element Analysis.

While we believe that our project is simple to solve in principle, we have and will continue to experience issues working through our project within the constraints given in this class. A significant portion of our time between DR1 and DR2 was spent learning how to follow the path laid out by this course while simultaneously meeting Heller's expectations on a week to week basis. As a group we noticed some backtracking and circular logic regarding specific assignments, which led to some setbacks. We believe now that we have sorted these out, but will remain vigilant for similar issues in the future.

Beyond just our Alpha design we did generate a handful of other useful combinations of modeling methods which gave us varying angles of insight into our problem. These "designs" are laid out below and have been given fun names because we wanted to make ourselves, our instructor, and our (hopefully) our sponsors laugh and be memorable.

Concept 2: Boisterous Brendan

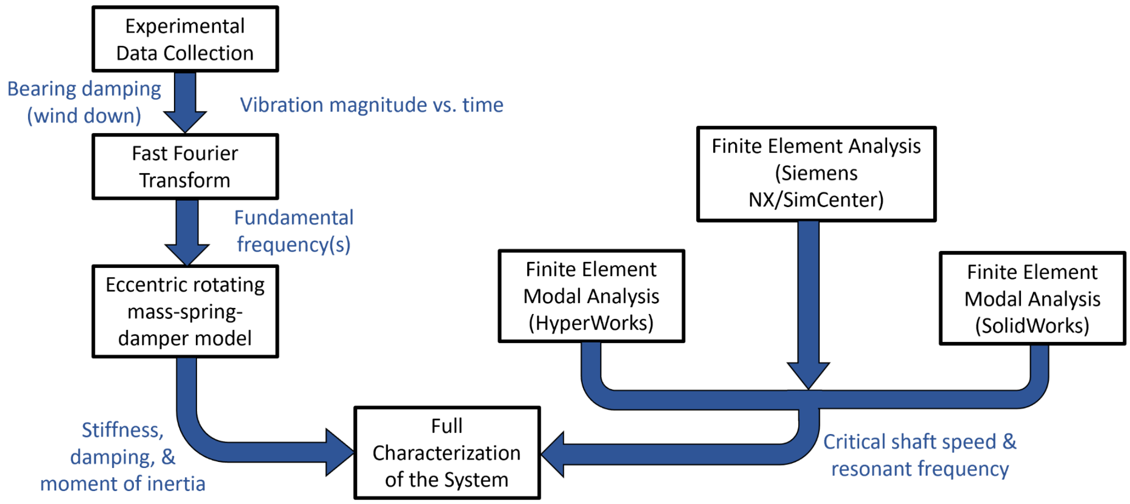

Figure 4. Boisterous Brendan schematic

This design is similar to our Alpha Design in the fact that it is centered around numerical derivation and Finite Element Analysis. We can obtain a graph of vibration magnitude over time which allows us to perform a Fast Fourier Transform to identify fundamental frequencies. Additionally we can derive the necessary equations of motion from an eccentric rotating mass-spring-damper model which allows us to predict stiffness, damping, and moment of inertia. By cross validating with critical shaft speed and resonant frequency predicted by the Finite Element Analysis model we can predict full characterization of the system. The biggest downside with this model compared to our Alpha Design is we do not incorporate an electromechanical model which prevents us from being able to iterate and refine the model.

Concept 3: Passionate Piper

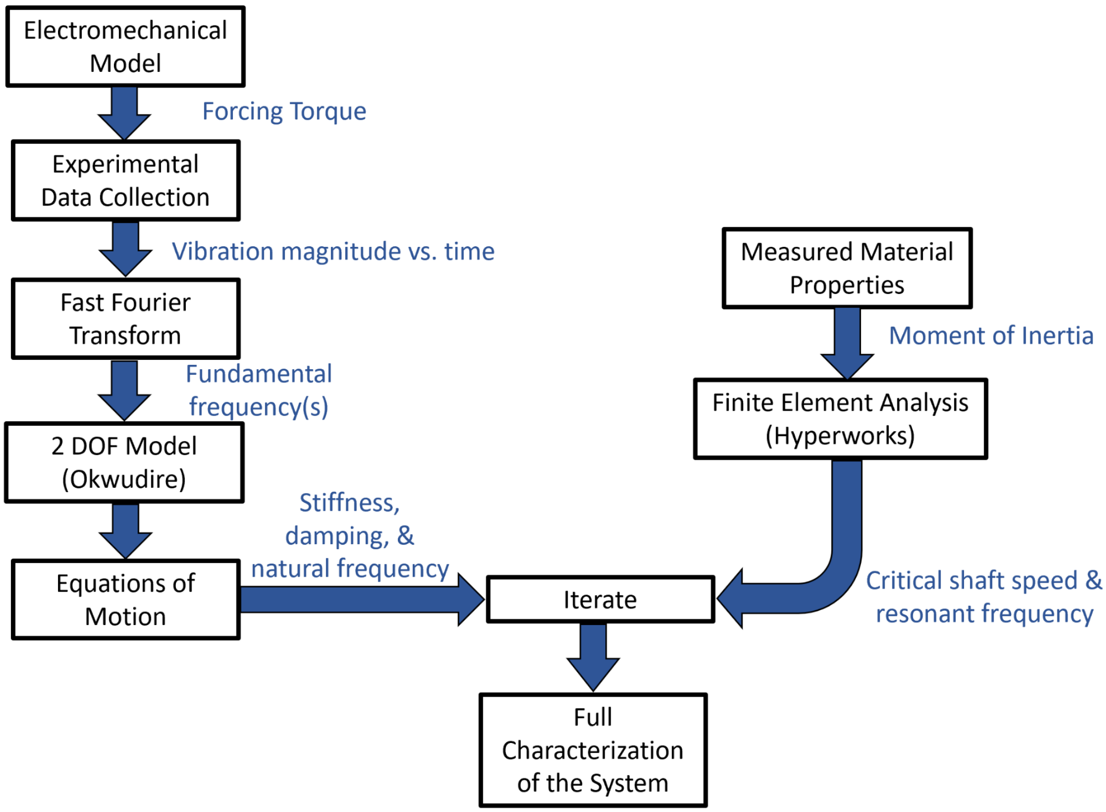

Figure 5. Passionate Piper schematic

This concept is centered around two approaches: a mathematical modeling approach and a computational modeling approach using FEA. The mathematical modeling incorporates using an electromechanical model, Fast Fourier Transform, and a two degree of freedom mass-spring-damper model. Using these in conjunction with the relevant equations of motion we are able to predict stiffness, damping, and natural frequency of the system. Our Finite Element Analysis model, using measured material properties as inputs, can then be validated to match the predicted mathematical model. The fallbacks of this design approach is it lacks the ability to loop and cross check parameters before the design is fully characterized. So while it is linear in its approach other models are better at validating system parameters.

Concept 4. Brilliant Beta Bassel

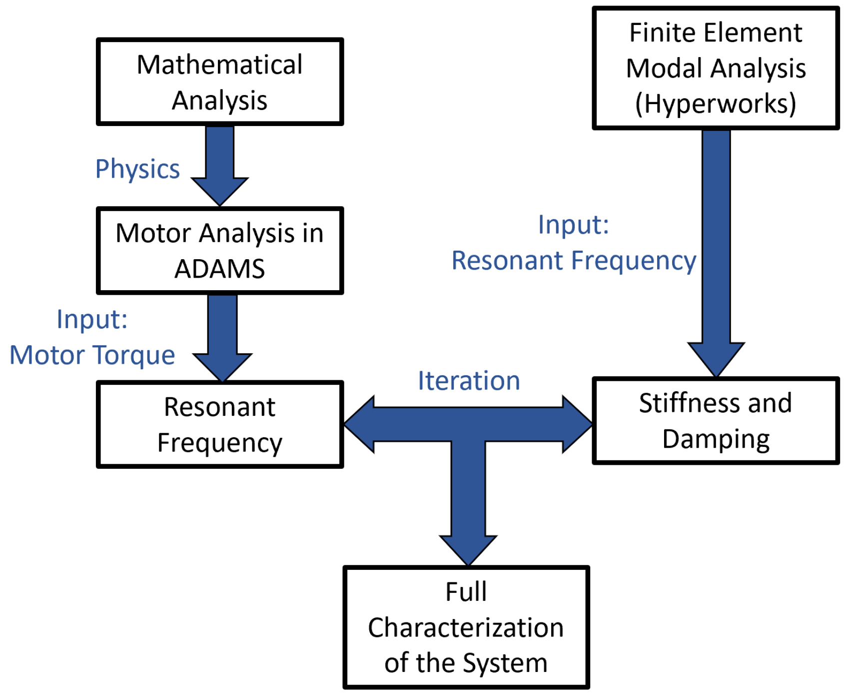

Figure 6. Brilliant Bassel schematic

This concept uses two computational modeling approaches. The first approach is a physics based approach using the MSC ADAMS modeling software that inputs motor torque to predict resonant frequency of the system. The second approach uses Finite Element Analysis to predict the stiffness and damping of the system. We then iterate between the two model outputs to predict the characterization of the system. The main drawback of this approach concept is that it is fully model driven and is not backed by any empirical data. This is a major flaw as we are not able to cross validate with the physical system we have in hand and is the main reason we won't be pursuing this design.

Concept 5: Mathematical Matt

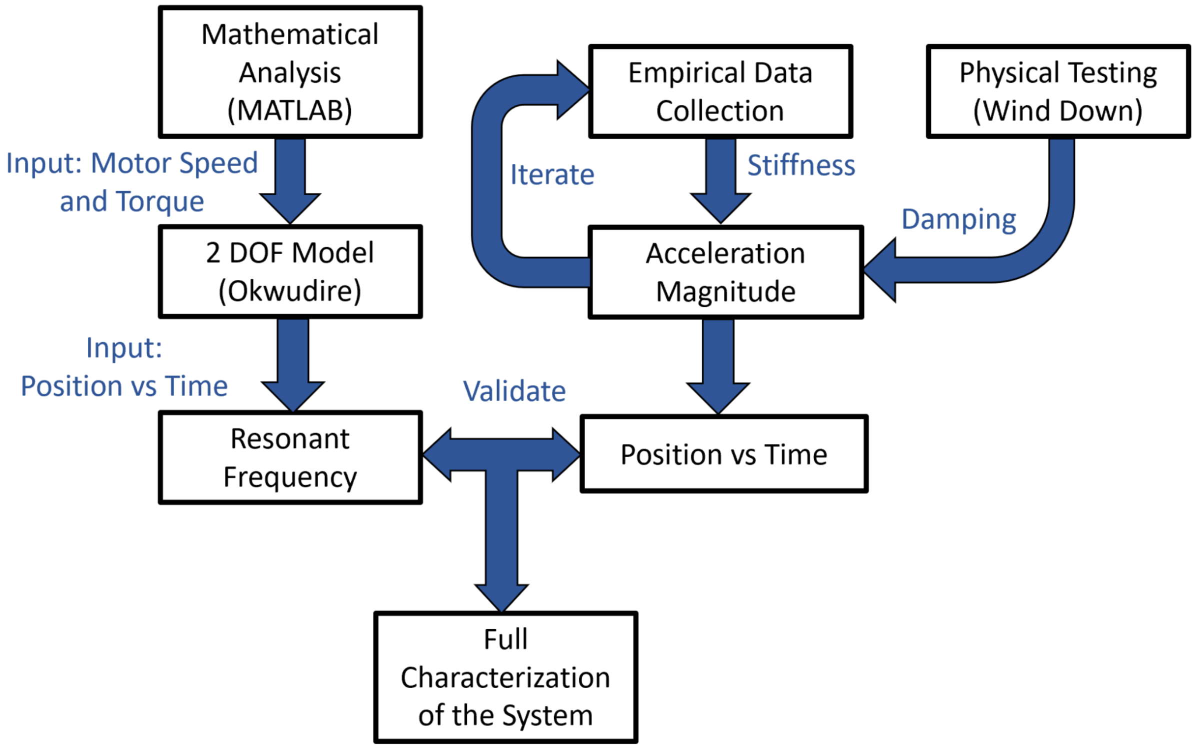

Figure 7. Mathematical Matt schematic

This approach is centered around numerical and mathematical software. Through the use of MatLab and empirical testing, we can obtain key parameters to model our kinematic system. On the MatLab side, we can model our system using differential equations that take in a given motor torque and then predict the equations of motion using the 2 degree of freedom model. From this, we obtain a position vs time plot which can be used in calculating various frequencies of the system. Using these frequencies, we can model the kinematic behavior of the system using a magnification vs frequency plot to determine the peak magnitude of vibration. This will allow us to determine if the system is over/under damped.

Then through empirical data collection, we can determine and validate our mathematical analysis. Through performing various tests, we will be able to derive key parameters in our kinematic model and ensure that our mathematical model is accurate. After characterizing and modeling the behavior of the system in an accurate way that matches our empirical data, we can modify the relevant parameters to see how they shift and impact the model and reduce the peak magnification/vibration magnitude of the system, solving the issue for Heller. One drawback to this concept is the lack of Finite Element Analysis software, as requested by Heller and is the reason we will not be proceeding with it.

Concept Down-Selection

After generating our five concepts, we compared and contrasted their unique abilities and their relative weaknesses. We did this in a similar process that we did for concept generation, by applying a combination of a pugh chart and sanity check on the performance of each concept. Each concept was ranked into acceptable (green) or missing (red) in terms of identifying/utilizing the key parameters needed to model the kinematic behavior of the system. The results and ranking of each concept generated above can be seen in the chart below.

Table 2. A chart ranking each design on their complexity, versatility, and ability to fully characterize our system and predict vibration magnitude (acceleration)

Concepts	Stiffness	Damping	Moment of Inertia	Resonant Frequency	In Depth Math and FEA Model	Maximum Vibration Magnitude	Iteration and Validation	Experience	Empirical Data
Alpha Design	Includes	Includes	Includes	Includes	Includes	Includes	Includes	Experienced	Includes
Boisterous Brendan	Includes	Includes	Includes	Includes	Includes	Includes	**Missing**	Experienced	Includes
Passionate Piper	Includes	Includes	Includes	Includes	Includes	Includes	**Missing Validation**	Experienced	Includes
Brillant Bassel	Includes	Includes	**Missing**	Includes	Includes	Includes	**Missing Validation**	**Unfamiliar with ADAMs**	**Missing**
Mathematical Matt	Includes	Includes	**Missing**	Includes	**Missing**	Includes	Includes	Experienced	Includes

Based on this ranking and the process flow of each concept/approach to obtaining a holistic kinematic model, we proceeded with our Alpha Design. A critical reason why we selected and are continuing forward with this approach is the iteration and validation aspect. Our Alpha Design includes bridges where we can validate the parameters we are obtaining using empirical data in addition to Finite Element Analysis, which allows us to create a well rounded model that precisely predicts the kinematic behavior of the system.

While we have a path forward with our Alpha Design, there are still challenges associated with it. Some short term issues that can be overcome include further development of the electromechanical motor model, converting the acceleration vs time data to position vs time in order to perform a Fast Fourier Transform, and obtaining a quality mesh in the Finite Element Analysis ahead of performing a modal analysis. All of these obstacles can be overcome and will be tackled accordingly ahead of the final design expo in order to complete our project on time. The approach to tackling this includes referencing online forums and professors on campus to help ensure we are on the right track and utilizing our time efficiently.

Appendix B: TEAM BIOGRAPHIES

Bassel Yassine

Hailing from Northville, MI, Bassel enrolled at the University of Michigan back during the Fall of 2018. Growing up, Bassel always had an interest in technology and uncovering how all the things around him worked, and thus decided to pursue an education and career in mechanical engineering. During his time at the University of Michigan, Bassel got heavily involved with the Michigan Solar Car Team. In 2019, he helped design, build, and race *Electrum*, the team's 15th car. In 2020, he served as the team's Engineering Director, responsible for designing the fastest and safest solar vehicle possible while adhering to budgetary and timeline constraints. Bassel has held various internship/co-op positions that have exposed him to multiple corners of the engineering field. In 2020, he worked for the startup Refraction AI, assisting in the development and assembly of their first robotic delivery platform. In 2021, he worked for Siemens as a Technical Marketing Engineer, particularly focusing on creating technical content and tutorials to showcase the functionality and robustness of Siemens NX, Teamcenter, and Simcenter. In 2022, he completed two program management rotations with Tesla, focusing primarily on vehicle exterior components, body paint, and plastics. Following graduation, Bassel intends to pursue a career in program management, and is excited to discover new and exciting ways to continue to learn and grow outside of the traditional academic setting.

Brendan Kwapis

Born and raised in Plymouth, Michigan, Brendan transferred into Mechanical Engineering from LSA Psychology in the spring of 2020. He has always had a passion for math and science and felt that a Mechanical Engineering degree would be a great way to pursue this. Throughout all 5 years of his undergraduate studies, Brendan has worked part time for Michigan Medicine in the Health Information and Technology Services (HITS) department, continuously supporting the technical needs of hospital staff. Outside of school and work, Brendan enjoys bowling, video games, and going to the gym. One fun fact is that he has bowled a 300 (which he is particularly proud of). After graduation, Brendan is moving to Tucson, Arizona to work for Raytheon Missiles and Defense as a Design and Specialty Mechanical Engineer.

Matt Roller

Growing up in Dexter, Michigan, Matt enrolled at the University of Michigan in the Fall of 2019. In his youth Matt had a fascination with how things worked and loved assembling/disassembling objects in his home. He decided to pursue a degree in Mechanical Engineering. Part of this stemmed from his father, who also studied Mechanical Engineering at the University of Michigan. Matt is particularly interested in sports and nature and would love to tie the technology involved in those fields to his engineering background. During his time at Michigan Matt has enjoyed working on various projects including designing a new manufacturing process for an industrial slicer. Outside of school and work Matt enjoys being active, especially outdoors. In the winter he enjoys skiing and taking morning ice baths. In the summer he enjoys biking and hiking in national parks. Matt has had multiple internships and is currently working at SSOE designing automotive manufacturing. After graduation he plans to pursue a full-time career as a project manager in this field.

Zack Piper

Zack grew up in Bloomfield Hills, Michigan and came to Michigan in the Fall of 2018 with a desire to seek out and explore mechanical engineering. After reading about the Michigan Solar Car Team in a Popular Mechanics article as a kid, Zack joined the Michigan Solar Car Team in the fall of 2018, later running for the Operations Director role. As Operations Director for the Team, he was responsible for the entire team's logistics, sourcing, and manufacturing. After racing in the Bridgestone World Solar Challenge, he added on a minor in the Multidisciplinary Design Program and continued to stay a part of the team, staying as Operations Director through the Fall of 2020. After participating in Solar Car, Zack has followed his interest in logistics through various internships at Geofabrica and Tesla Motors. Outside of his academic life, Zack is passionate about movies, hiking and pinball. After graduation, Zack is planning to continue to explore his passion in program management as he steps into his professional career while continuing to learn and grow with each challenge he faces.

Appendix C: Hand off Plan

Mathematical Model

Our current mathematical model is able to predict general system behavior. This allowed us to vary different parameters and base our recommendations for Heller depending on how the system responded. We have created this hand off plan for anyone continuing our investigation into Heller's vibration problem.

Summary:
The model we developed simplifies the module to a 2 degree of freedom mass spring damper system. We did this to reduce the complexity of the system, making it easier to analyze and model. We derived equations of motion using this simplified system, which predicts the displacement of the system. However, to obtain acceleration, which is what our model predicts, we derived the equations twice in Matlab. Model parameters were obtained using a variety of methods. Please visit the Mathematical Model section in Engineering Analysis for extended analysis.

Matlab:
All analysis was done using Matlab. We've included our code in Appendix F and provided all necessary files in the mathematical model hand off plan folder. The code can seem convoluted and overwhelming so we want to break it apart section by section. It should be noted that the file that we are referencing here is titled "mathematical_model_v3_450".
Other similar versions of the code can also be found in the hand off plan folder.
The first section includes all the model parameters such as system mass, eccentric mass, resonant frequency, forcing frequency, etc. There is also an internal impeller imbalance built into the system. All of these parameters can be varied to analyze how they impact system behavior.
The next section includes the frequency plot of the system, the displacement equation for the steady state response of the system, and the acceleration equation that drives the model. The acceleration equation was obtained using a second derivative function in Matlab, that function has been commented out and we have hard coded the acceleration equation it produced below it.
The next section contains 4 plots. The first figure plots a frequency sweep based on the current forcing and resonant frequency. The second figure plots the predicted displacement of the system in meters. Figures 3 and 4 both plot the predicted acceleration of the system, figure 3 shows acceleration in m/s^2 while figure 4 converts acceleration to G's.
The following section plots 5 different figures. Figure 5 plots the relationship between acceleration magnitude and various eccentric mass values while figure 6 plots the relationship between acceleration magnitude and various stiffness values. Figures 7-9 predict how changing system parameters affect the resonant frequency. Figure 7 shows the relationship between resonant frequency and system mass, figure 8 shows the relationship between resonant frequency and stiffness, and figure 9 shows the relationship between

resonant frequency and the damping ratio. These plots allowed us to analyze system behavior in order to recommend what changes should be made to shift the resonant frequency away from the forcing frequency.
The final section contains the parameter optimization we performed along with our recommended changes. We recommend increasing the stiffness and keeping mass the same. This recommendation considered the cost and ease of implementation for Heller and is open for further analysis and change. Figure 10 compares the bode plot of the original system with a bode plot that contains the recommended parameter values to visualize the expected shift in resonant frequency.

Next Steps:
There are several ways in which the model could be improved. First, we recommend adding an additional degree of freedom into the model. This would require additional calculations and potentially more complex equations, but it would provide a more accurate representation of the system. We also think that our process for optimizing parameters could be improved, potentially allowing for more parameters and combinations to be analyzed. Unfortunately, we believe the biggest flaw in our model is that it does not account for the fluid dynamic effects that we believe are a major contributor to the system's behavior. We have only considered and studied the dynamics aspect of this problem so we can't really recommend how we would go about incorporating this into the model.

FEA Model

In regards to the FEA models we are sending to Heller Industries, we will be sending all iterations 1.0 through 9.0. The details of the constraints on each can be found in the CAD/FEA section of this report. These details will also be obvious upon opening up each SolidWorks assembly file, but our report serves as a reference. The required SolidWorks part files needed to run the simulations for each assembly will also be provided in the hand-off. It will likely be required to run each frequency analysis study again to obtain the results which we presented in this report, as it was difficult to transfer the results files (many are extremely large). These simulations do not take very long to run, so this rerun should not be a computational issue.

Next steps for improvements to these FEA simulations can be found in the Discussion section of this report. There are a few extra assembly files included as well, which were not discussed in the main report. Most of these completely failed to run for some reason or another, or produced wildly inaccurate results. Regardless, we wanted to include them as they could provide potentially valuable information about what NOT to do when running these FEA simulations. These include the "piper", "new_mat", 3.1, 8..1, 10.0, and 11.0 variations (suffix at the end of the assembly file name).

EMPIRICAL DATA

We will provide all of the accelerometer data which we collected in our hand-off. The file names should be self-explanatory, in regards to when the test was run and the frequency that the motor was run at. All data collection followed the testing setup which was outlined in the Data Collection section of this report.

For next steps, we do recommend attempting to retake this data while copying our testing setup. This would serve to verify and validate our methods and results. Performing an FFT on each data set which ran at constant motor frequency may also be useful. We did not have time to perform an FFT on every single data set.

To follow our empirical data collection, all files are .excel or .csv format. There are 3 naming conventions with our file format, seen below:

- WeightofEccentricMass__SizeofImpeller_CollectedDate
- SizeOfImpeller_Configuration_CollectedDate
    - All of these files have no eccentric mass
- WeightofEccentricMass__SizeofImpeller_FreqSweep_CollectedDate
    - These results have the plots seen in Figure 28 and 29 in our empirical data

Appendix D: Acceleration Magnitude Data from Heller Industries

Nidec 0.25HP Blowers 40~50hz Vibration Data --- @ Ambient and @ Hot 250C																		
Top Z11 ~ 13	40hz (Ambient)			45hz (Ambient)			50hz (Ambient)			40hz (Hot)			45hz (Hot)			50hz (Hot)		
Test points	T11-#2	T12-#3	T13-#4	T11-#2	T12-#3	T13-#4	T11-#2	T12-#3	T13-#4	T11-#2	T12-#3	T13-#4	T11-#2	T12-#3	T13-#4	T11-#2	T12-#3	T13-#4
	15°C	15°C	15°C	15°C	15°C	15°C	15°C	15°C	15°C	250°C	250°C	250°C	250°C	250°C	250°C	250°C	250°C	250°C
X	0.0572	0.1022	0.0964	0.1761	0.1554	0.2252	0.4909	0.674	0.7485	0.0941	0.1357	0.1426	0.1548	0.3257	0.5983	1.1325	0.8357	1.2861
Y	0.0963	0.1088	0.1042	0.1709	0.2176	0.2153	0.5479	0.6863	0.6738	0.0917	0.164	0.2301	0.1504	0.3782	0.573	1.1425	1.159	1.4461
Z	0.1037	0.1094	0.1071	0.0986	0.1711	0.1689	0.2629	0.5434	0.4709	0.085	0.1003	0.1394	0.1026	0.1847	0.527	0.544	0.8482	1.1316
Average X, Y, Z	0.086	0.107	0.103	0.149	0.181	0.203	0.434	0.635	0.631	0.090	0.133	0.171	0.136	0.296	0.566	0.940	0.948	1.288
STD DEV X, Y, Z	0.025	0.004	0.006	0.043	0.032	0.030	0.151	0.079	0.144	0.005	0.032	0.051	0.029	0.100	0.036	0.343	0.183	0.157
	40hz (Ambient)			45hz (Ambient)			50hz (Ambient)			40hz (Hot)			45hz (Hot)			50hz (Hot)		
Bottom Z11 ~ Z13	B11-#9	B12-#10	B13-#12	B11-#9	B12-#10	B13-#12	B11-#9	B12-#10	B13-#12	B11-#9	B12-#10	B13-#12	B11-#9	B12-#10	B13-#12	B11-#9	B12-#10	B13-#12
	15°C	15°C	15°C	15°C	15°C	15°C	15°C	15°C	15°C	250°C	250°C	250°C	250°C	250°C	250°C	250°C	250°C	250°C
X	0.1282	0.0953	0.082	0.2472	0.0687	0.1063	0.4892	0.3223	0.3084	0.1346	0.112	0.108	0.2732	0.1178	0.1126	1.0886	0.3534	0.2876
Y	0.1458	0.0963	0.0809	0.1652	0.1566	0.1418	0.3696	0.7296	0.4887	0.1697	0.1071	0.1116	0.287	0.2176	0.2489	0.545	0.6629	0.524
Z	0.1366	0.0657	0.0901	0.2074	0.1122	0.0833	0.6273	0.285	0.2335	0.1632	0.1088	0.0799	0.2646	0.119	0.1224	0.9434	0.3887	0.1598
Average X, Y, Z	0.137	0.086	0.084	0.207	0.113	0.110	0.495	0.446	0.344	0.156	0.109	0.100	0.275	0.151	0.161	0.859	0.468	0.324
STD DEV X, Y, Z	0.009	0.017	0.005	0.041	0.044	0.029	0.129	0.247	0.131	0.019	0.003	0.017	0.011	0.057	0.076	0.281	0.169	0.185

Appendix E: DR1 Domain Analysis & Reflection

We plan to achieve our requirements and specifications through extensive research supplemented with mathematical and FEA model development. This will be done in parallel as laid out in our project plan, however we do recognize that this is not a simple task and will require aid from several resources.

The most significant obstacle in our project is consistency between the FEA and mathematical model(s). While we are confident in our abilities to develop independently accurate models, the difficulty we foresee is achieving adequate agreement between the two of them, inputting the same parameters and receiving an output from one that aligns/supports the other. With our knowledge, we can construct the framework for both of these models and are seeking additional support from the University to assist with the execution of vibrational FEA. In the coming month(s), the main focus is ensuring accuracy and precision from both of our models to the experimental data, within the 10% as defined in our specifications.

Another difficult aspect we recognize in our project, although further out on the horizon, is the lifetime specification of the module. Currently, our plan is to work with Heller on how they validate the long term life of their modules. We plan to consult with Professor Harvey Bell IV for additional insight into how companies test and validate long term lifetime of products. The meeting we have scheduled with Professor Bell IV is still being finalized and will be added into our project plan once scheduled.

The provided module from Heller is a crucial piece of equipment for our project, as it allows us to test and characterize the vibration for both the 8 inch and 10 inch impeller, for both the 0.11 HP motor as well 0.25 HP. Additionally, the accelerometers which Heller is providing allow us to mirror the vibration they are experiencing on their convection oven. Another piece of specific knowledge that we need for our project is vibrational Finite Element Analysis, which we are exploring and learning about through meeting with Professor Hulbert, as well as doing individual research into what software is best suited for this analysis.

As stated above, we are supplementing our knowledge of vibrational analysis in mathematical and FEA models via in-depth discussions with various professors on campus, with meetings scheduled for the weeks of 02/06 and 02/13. The gaps that remain are what we characterize as “unknown-unknowns” or gaps in information to problems we are not aware of. While we have met and have considered several problems we may run into during the modeling and testing process, there will likely be additional problems the team encounters along the way that will be overcome through collective brainstorming and problem solving.

Appendix F: MATLAB Code

*The code shown is mathematical_model_v3_450 and can be found in the math model folder of the handoff plan. Other versions of the model which produce slightly different graphs can also be found in the folder.

```
close all;
clc;
% inputs/variables
M1 = 0.00075; %natural imbalance of impeller
M2 = 0.0026; %0.0028; %Added mass to impeller
M_e = M1 + M2 ; %kg (Total eccentric mass)
k = 4.7153e+05; % estimated bulk stiffness [N/m].
M = 1.01*6.02; %kg, total system mass measured on 4/8/23.
e = .12; %meters
w = [1 :0.01: 60];
w_n = sqrt(k./M) * 0.1592; %from stiffness and mass estimates
%w_n = 44.5554; % value from FFT [Hz]
w_d = w_n; % = w_n_actual*sqrt(1-(zeta)^2);
w_2 = 45; % motor speed in Hz (approximate)
zeta_1 = 0.022; %potentially iterate once motor speed w_2 is sorted out
%w_n_actual = w_d/(sqrt(1-zeta^2));
c = zeta_1*2*sqrt(k*M);
zeta = c./(2.*sqrt(k.*M));% here is actual damping coefficient c
t = [0:0.1:20];

%syms M_e M w w_n w_2 zeta t e
%used to find 2nd derivative of position function
phi = atan((2 .* zeta .* ( w_2 ./ w_n )) ./ (1 - (w_2 ./ w_n).^2)) ;

X_sweep = ((M_e .*e ) ./ M) .* ((w ./ w_n) ./ (sqrt((1-(w ./ w_n).^2).^2)
+ (2 .* zeta .* (w ./ w_n)).^2)) ;
% varied w inputs
X = ((M_e .*e ) ./ M) .* ((w_2 ./ w_n) ./ (sqrt((1-(w_2 ./ w_n).^2).^2) +
(2 .* zeta .* (w_2 ./ w_n)).^2)) ;
X_2 = ((M_e .*e ) ./ M) .* ((w_2 ./ w_n) ./ (sqrt((1-(w_2 ./ w_n).^2).^2)
+ (2 .* zeta .* (w_2 ./ w_n)).^2)) ;
% varied time inputs
x = X_2 .* sin(w_2 .* t - phi);

%a = diff(x,t,2)

a = -(M_e*e*w_2^3*sin(atan((2*w_2*zeta)/(w_n*(w_2^2/w_n^2 - 1))) +
t*w_2))/(M*w_n*(((w_2^2/w_n^2 - 1)^2)^(1/2) + (4*w_2^2*zeta^2)/w_n^2));

a1 = a .* .101971621; %convert to G's

%plot

f1 = figure;

plot (w , X_sweep)
hold on
fontsize(f1, 16, 'pixels')
title('Frequency Sweep')
xlabel('Forcing Frequency (Hz)')
ylabel('Vibration Magnitude (m)')
```

```
f2 = figure;

plot (t , x)
hold on
fontsize(f2, 16, 'pixels')
title('Position')

f3 = figure;

plot (t, a)
hold on
fontsize(f3, 16, 'pixels')
title('Acceleration')

f4 = figure;

plot (t, a1, 'LineWidth', 1.5)
hold on
box off
fontsize(f4, 40, 'pixels')
xlabel('Time (s)')
ylabel('Acceleration (G)')
title('2.6 grams Added Mass 45 Hz Operating Speed Acceleration v Time',
'Fontsize', 30)
%% acceleration magnitude versus eccentric mass (for max motor speed
operation)
% sweep eccentric masses from 0 to 10 grams. All other parameters constant
% (k,m,zeta, etc)
M_e1 = 0;
x = 1;
a_em = ones(1,51);
while x < 51
    a = -(M_e1*e*w_2^3*sin(atan((2*w_2*zeta)/(w_n*(w_2^2/w_n^2 - 1))) +
t*w_2))/(M*w_n*(((w_2^2/w_n^2 - 1)^2)^(1/2) + (4*w_2^2*zeta^2)/w_n^2));
    a1 = a .* .101971621;
    %f5 = figure;
    %plot(t,a1)
    max(a1);
    M_e1 = M_e1 + 0.0005;
    a_em(1,x) = max(a1);
    x = x+1;
end
M_e2 = 0:0.0005:0.025;
f5 = figure;
plot(M_e2(1:50), a_em(1:50))
hold on
fontsize(f5, 16, 'pixels')
xlabel('Eccentric Mass (kg)')
ylabel('Acceleration Magnitude (g)')
title('Acceleration Magnitude vs. Eccentric Mass')
%% acceleration magnitude versus stiffness (for max motor speed operation)
%for now, eccentric mass of 1 gram. Varying stiffness from 10,000 N/m to
%1,000,000 N/m (no idea the feasibility of any of these values, subject to
%change
```

```
k_1 = 10000;
wn_1 = sqrt(k_1/M);
zeta_1 = c/(2*sqrt(k_1*M));
x = 1;
a_k = ones(1,99);
while x < 99
    a = -(M_e*e*w_2^3*sin(atan((2*w_2*zeta_1)/(wn_1*(w_2^2/wn_1^2 - 1))) +
t*w_2))/(M*wn_1*(((w_2^2/wn_1^2 - 1)^2)^(1/2) +
(4*w_2^2*zeta_1^2)/wn_1^2));
    a1 = a .* .101971621;
    max(a1);
    k_1 = k_1 + 5000;
    a_k(1,x) = max(a1);
    x = x+1;
end

  k_2 = 10000:5000:500000;
  f6 = figure;
  plot(k_2(1:98), a_k(1:98))
  hold on
  box off
  fontsize(f6, 18, 'pixels')
  xlabel('Stiffness (N/m)')
  ylabel('Acceleration Magnitude (g)')
  title('Acceleration Magnitude vs. Stiffness')

  %% acceleration magnitude versus damping (for max motor speed operation)

  %% frequency sweeps while varying parameters
  %mass
  f7 = figure;
  w = [1 :0.01: 80];
  X_sweep = ((M_e .*e ) ./ M) .* ((w ./ w_n) ./ (sqrt((1-(w ./ w_n).^2).^2)
  + (2 .* zeta .* (w ./ w_n)).^2)) ;
  plot (w , X_sweep,'b','Linewidth',1.5)
  hold on
  box off
  fontsize(f7, 24, 'pixels')
  title('Frequency Sweep for Varying System Mass')
  xlabel('Forcing Frequency (Hz)')
  ylabel('Vibration Magnitude (m)')
  M = 0.5 * M;
  w_n = sqrt(k./M) * 0.1592;
  X_sweep_1 = ((M_e .*e ) ./ M) .* ((w ./ w_n) ./ (sqrt((1-(w ./
  w_n).^2).^2) + (2 .* zeta .* (w ./ w_n)).^2)) ;
  plot(w, X_sweep_1,'r','Linewidth',1.5)
  M = 6.02;
  M = 1.5 * M;
  w_n = sqrt(k./M) * 0.1592;
  X_sweep_2 = ((M_e .*e ) ./ M) .* ((w ./ w_n) ./ (sqrt((1-(w ./
  w_n).^2).^2) + (2 .* zeta .* (w ./ w_n)).^2)) ;
  plot(w, X_sweep_2,'g','Linewidth',1.5)
  legend('M = 6.02 kg','M = 3.01 kg', 'M = 9.03 kg')
  hold off
```

```
%stiffness
M = 6.02;
f8 = figure;
plot (w , X_sweep,'b','Linewidth',1.5)
hold on
box off
fontsize(f8, 24, 'pixels')
title('Frequency Sweep for Varying System Stiffness')
xlabel('Forcing Frequency (Hz)')
ylabel('Vibration Magnitude (m)')
k = 0.5 * k;
w_n = sqrt(k./M) * 0.1592;
X_sweep_1 = ((M_e .*e ) ./ M) .* ((w ./ w_n) ./ (sqrt((1-(w ./
w_n).^2).^2) + (2 .* zeta .* (w ./ w_n)).^2)) ;
plot(w, X_sweep_1,'r','Linewidth',1.5)

k = 471530;
k = 1.5 * k;
w_n = sqrt(k./M) * 0.1592;
X_sweep_2 = ((M_e .*e ) ./ M) .* ((w ./ w_n) ./ (sqrt((1-(w ./
w_n).^2).^2) + (2 .* zeta .* (w ./ w_n)).^2)) ;
plot(w, X_sweep_2,'g','Linewidth',1.5)
legend('k = 471530 N/m','k = 235765 N/m','k = 707295 N/m')
hold off


%damping
k = 471530;
w_n = sqrt(k./M) * 0.1592;
f9 = figure;
w = [40 :0.01: 50];
X_sweep = ((M_e .*e ) ./ M) .* ((w ./ w_n) ./ (sqrt((1-(w ./ w_n).^2).^2)
+ (2 .* zeta .* (w ./ w_n)).^2)) ;
plot (w , X_sweep,'b','Linewidth',1.5)
hold on
box off
fontsize(f9, 24, 'pixels')
title('Frequency Sweep for Varying System Damping Ratio')
xlabel('Forcing Frequency (Hz)')
ylabel('Vibration Magnitude (m)')
zeta = 0.5 * zeta;
w_n = sqrt(k./M) * 0.1592;
X_sweep_1 = ((M_e .*e ) ./ M) .* ((w ./ w_n) ./ (sqrt((1-(w ./
w_n).^2).^2) + (2 .* zeta .* (w ./ w_n)).^2)) ;
plot(w, X_sweep_1,'g','Linewidth',1.5)
zeta = 0.0220;
zeta = 1.5 * zeta;
w_n = sqrt(k./M) * 0.1592;
X_sweep_2 = ((M_e .*e ) ./ M) .* ((w ./ w_n) ./ (sqrt((1-(w ./
w_n).^2).^2) + (2 .* zeta .* (w ./ w_n)).^2)) ;
plot(w, X_sweep_2,'r','Linewidth',1.5)
legend('\zeta = 0.022', '\zeta = 0.011', '\zeta = 0.033')
hold off
```

```
%% recommendations code

%Doubling stiffness, same mass
zeta = 0.022;
w_n = sqrt(k./M) * 0.1592;
w = [20:0.01:80];
X_sweep = ((M_e .*e ) ./ M) .* ((w ./ w_n) ./ (sqrt((1-(w ./ w_n).^2).^2)
+ (2 .* zeta .* (w ./ w_n)).^2)) ;
plot (w , X_sweep,'b','Linewidth',1.5)
hold on
box off
fontsize(f9, 24, 'pixels')
title('Frequency Sweep for Varying System Damping Ratio')
xlabel('Forcing Frequency (Hz)')
ylabel('Vibration Magnitude (m)')
k = 2*k;
w_n = sqrt(k./M) * 0.1592;
X_sweep_1 = ((M_e .*e ) ./ M) .* ((w ./ w_n) ./ (sqrt((1-(w ./
w_n).^2).^2) + (2 .* zeta .* (w ./ w_n)).^2)) ;
plot (w , X_sweep_1,'r','Linewidth',1.5)
```

Appendix G: Results and Data Collection

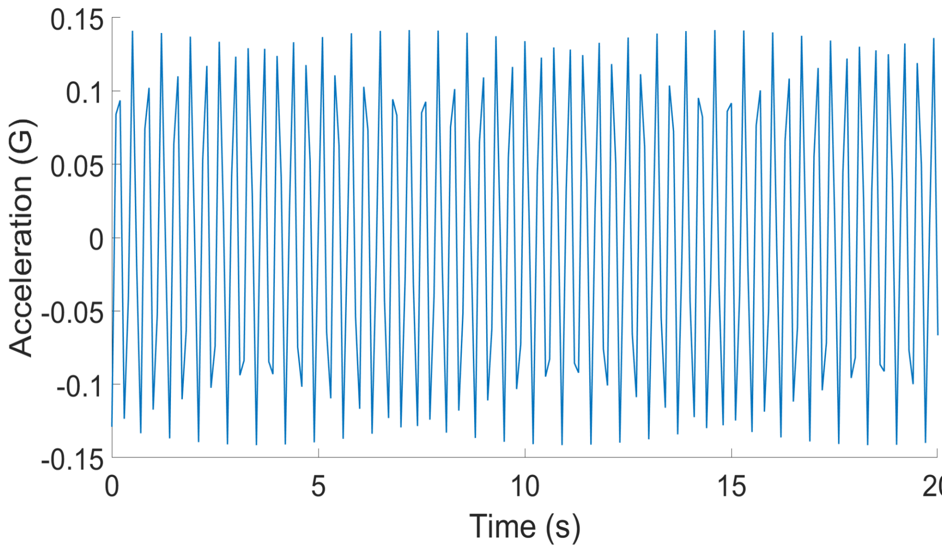

Figure 1. Mathematical prediction of acceleration magnitude in the X-direction with no eccentric mass attached to the impeller.

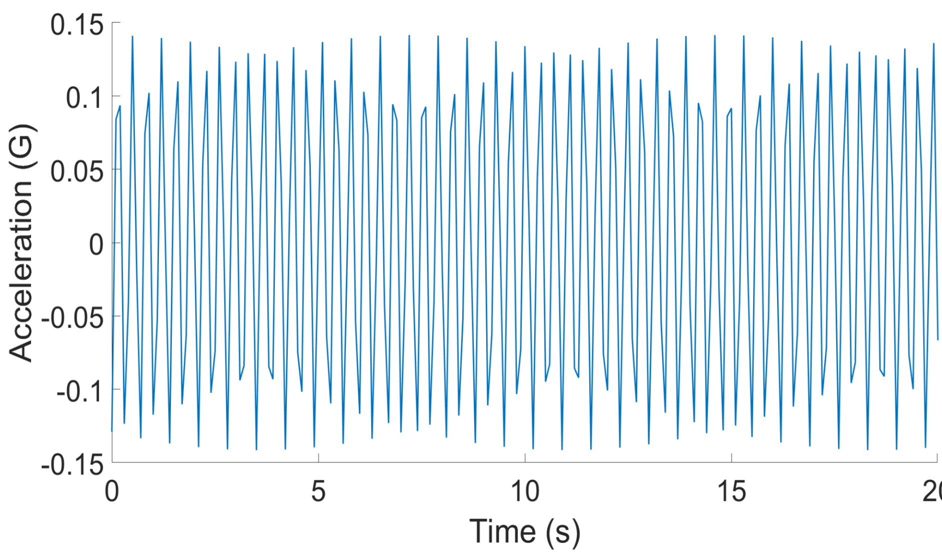

Figure 2. Mathematical prediction of displacement in the Y-direction with no eccentric mass attached to the impeller.

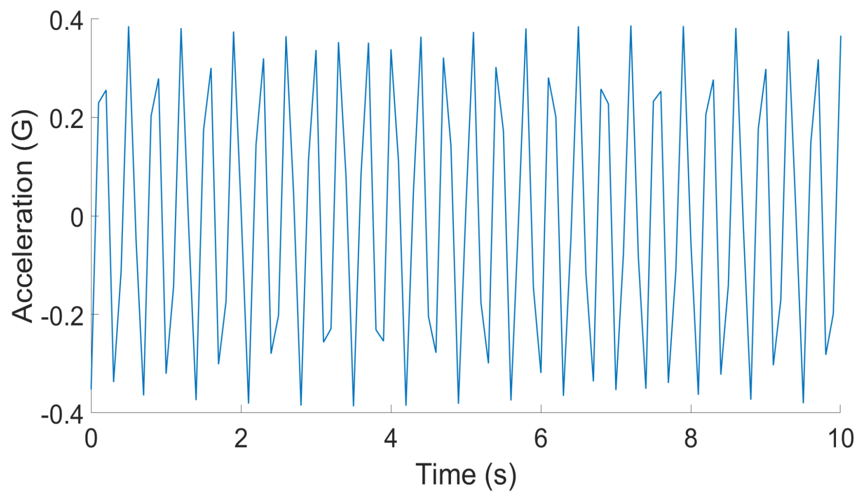

Figure 3. Mathematical prediction of displacement in the X-direction with 1.3 g of eccentric mass attached to one end of the impeller.

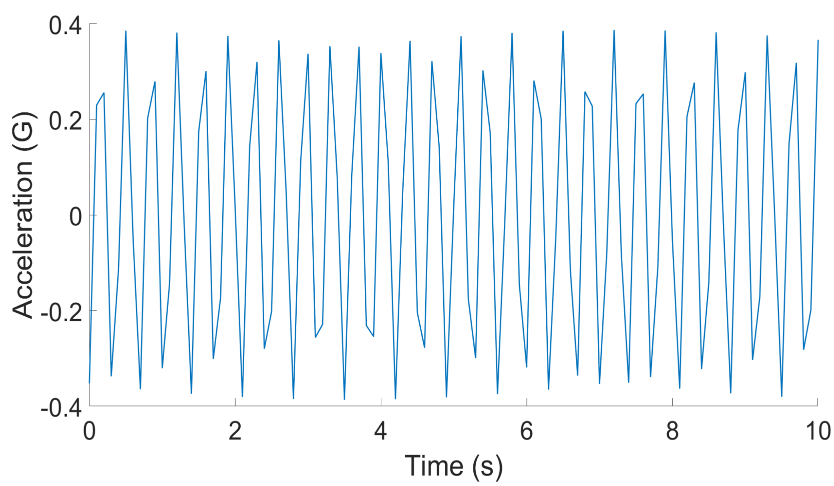

Figure 4. Mathematical prediction of displacement in the Y-direction with 1.3 g of eccentric mass attached to one end of the impeller.

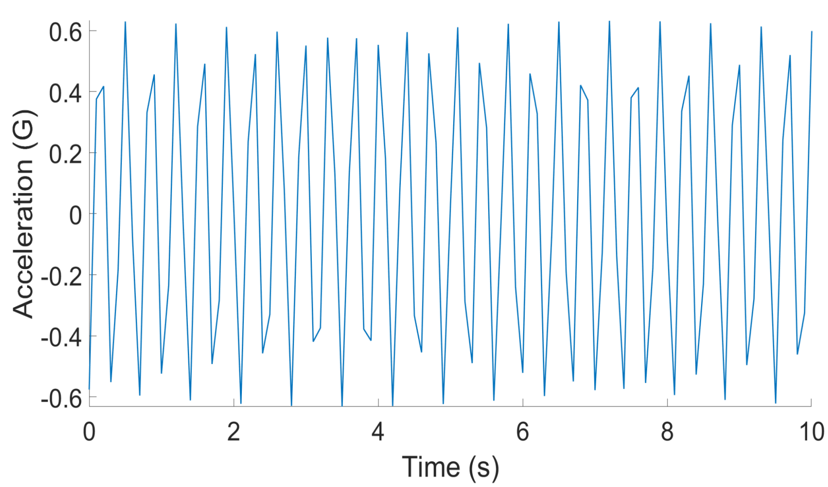

Figure 5. Mathematical prediction of displacement in the X-direction with 2.6 g of eccentric mass attached to one end of the impeller.

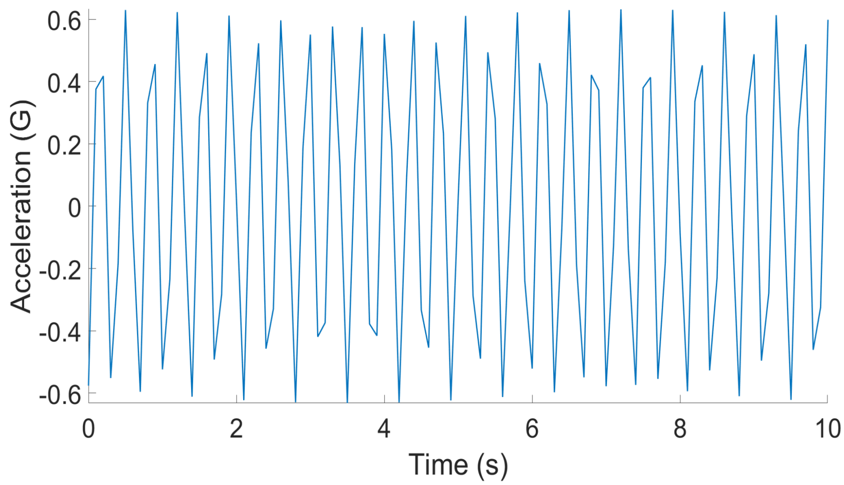

Figure 6. Mathematical prediction of displacement in the Y-direction with 2.6 g of eccentric mass attached to one end of the impeller.

Stiffness Study Data:

This was performed in Hyperworks with the exact same setup as can be found in the Engineering Analysis section of this report. The only change was the load magnitude.

Applied Load (N)	Displacement (mm)	Approximate stiffness (N/m)
1	0.005368	186289.1207
5	0.02864	174581.0056
10	0.0536834	186277.3222

Average: 182382.4828

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