ME 450 F23 Final Report

Team 12: Heller Industries Reflow Oven Drive Train

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EXECUTIVE SUMMARY

This project aims to address issues Heller Industries faces with their reflow ovens. These ovens are used for the mass soldering of printed circuit boards (PCB) and employ an edge hold system. The edge hold system consists of two rails, one stationary and one adjustable, that hold the PCBs up by their edges. To accommodate different sizes of PCBs, the adjustable rail must move in and out relative to the stationary rail. Currently, lead screws and a chain drive system are used to adjust the rail. However, over time, the chain links begin to experience stretching and skipping and the sprockets begin to deflect. Heller Industries has assigned our group the task of developing an alternative design that is more reliable and cost efficient. As this design will be a component within a singular machine with no direct human interaction and that has minimal production, there are no contextual factors that need to be considered.

Due to the nature of this project, there are many different requirements and specifications. The main categories are durability, integration into current design, cost, and rail width adjustability. For durability, the design must be able to withstand 2 years of use which works out to be 2190 cycles assuming the rail is moved 3 times a day. In order to be used in the ovens, the design must be able to integrate with minimal changes to the original system. The design must cost less than $300 per rack and pinion pair in order to stay economically competitive with the current solution. To meet rail adjustability, the design must be able to adjust the rails between a minimum distance of 2 inches and a maximum distance of 20 inches. The final design must also be able to withstand temperature cycles between 20-500 degrees Celsius and not corrode in formic acid.

In the past, Heller Industries has found success in using a rack and pinion design. Through a rigorous concept generation and down selection process, this design has again been singled out as the most effective solution to the current design problem. This design will involve a rack and pinion pair replacing each lead screw. The pinions will be connected via a singular central shaft that is actuated on one end by a pneumatic motor. A CAD model of this design was created and used to run finite element analysis. Additional static and dynamic load analysis was completed using Matlab. These analyses lead us to conclude that the rack would face a maximum deflection of 0.0047 in which is significantly lower than the maximum allowed of 0.1181 in (3mm). Additionally analysis revealed that a shaft diameter of ½” would prevent any twist in the shaft and therefore any hysteresis as the shaft was turned.

Using the CAD model and initial engineering analysis a proof of concept was created. This allowed for the validation of the system and ensured it meets crucial engineering requirements and specifications. Based on the analysis and performance of the proof of concept, our group is confident in saying that the rack and pinion design is a competitive replacement for the current chain drive system employed by Heller Industries. The design meets all necessary specifications and provides a durable, reliable and future proof solution to the design problem.

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INTRODUCTION

Almost every present-day electronic device in production utilizes a PCB to perform its desired function. As a result of this need, tens of millions of PCBs are produced every year [1]. In order to meet the growing demand for PCBs, a series of specialized machines have been developed to rapidly produce the boards in an assembly line. The first machine places solder on the blank circuit boards in the locations that the PCB components will be placed. The second machine places the components of the PCB onto the board in the aforementioned locations. The final machine in the line heats the solder to secure the components to the board [2]. This last machine is the Heller Industries reflow oven, pictured below in Figure 1.

Figure 1: Heller Industries 1936 MK7 reflow convection oven with the top closed. Reflow convection oven refers to the oven being heated by the recirculation of nitrogen inside the oven [5].

It is important to note that all three of the machines use an edge hold mechanism to guide the PCBs from one end of the machine to the other in accordance with IPC-SMEMA-9851 [3]. The edge hold conveyor (EHC) in the Heller reflow ovens is pictured below in Figure 2.

Figure 2: View of EHC system in Heller reflow oven, with top open. This view is from the entry of the oven. Placeholder boards are lined up along the right lane to demonstrate the edge hold capability. This specific model features dual lanes to double output.

Due to the nature of PCB manufacturers, “job shops” can have contracts to manufacture PCBs of varying sizes. Because of this, the width between the rails of the edge holder must change in order to accommodate different sizes of PCBs, in some cases up to three times a day. The current design employed by Heller Industries to change the location of the edge holders involves a motor actuating a chain drive that extends along the entire length of the oven. This chain drive will then intermittently turn an idler gear, connected to a lead screw. These lead screws, which vary in number from four to eight depending on the oven model, are connected to one of the edge hold rails. As the lead screw turns, the adjustable edge hold rail will move in or out to the desired location. The current edge hold adjustment mechanism is shown below in Figure 3 below (3).

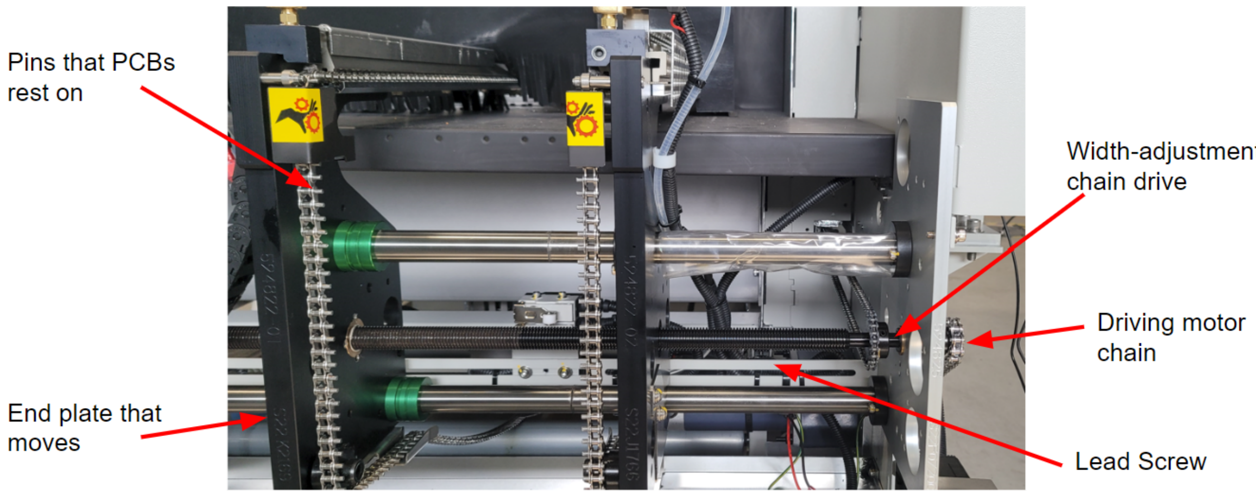


Figure 3: Left end of Heller reflow oven with side panels removed to demonstrate adjustable EHC. The current oven features one lead screw on each end and up to 6 more screws distributed along the length of the oven. Lead screw rotation is synchronized with the width-adjustment chain drive, found on the right side of the image.

The chain drive runs through the whole length of the oven to drive all lead screws at the same time. This can be seen in Figure 4 below on page 6 which shows the oven from the right side.

Figure 4: Right side of Heller Industries reflow oven with side panels removed. Sprocket on the left side near the aluminum plate is connected with a chain to the width adjustment chain sprocket shown in Figure 3. The chain ensures all lead screws (and edge hold rails) move synchronously.

The issue with this design is the chain drive itself. As Heller Industries' ovens have gotten larger to meet the market demand, the chain drive mechanism has not changed in over 20 years, resulting in an increased frequency of failure. The common causes of failure include deflection of the idler gears, as well as lengthening of the chain over time which causes skipping. Chain slippage decreases the accuracy of the width adjustment mechanism, meaning that PCBs could be compressed or fall into the oven.

Failure by idler deflection or chain slippage presents a large issue to Heller Industries. The defective parts will require repair, whether that be through maintenance or replacement. The repair time means downtime for the ovens, reducing profits and frustrating consumers. Finally, if failure occurs during the one-year warranty period Heller will be responsible for the repair costs. Therefore, this project aims to reduce the frequency of chain drive failures by idler deflection or chain slippage through the design of a cheap, durable, and accurate mechanism to move the edge hold rails to the desired position.

Benchmarking and Standards

Before beginning any design conceptualization, it is important to consider solutions that solve similar problems as well as the previous designs that Heller Industries has iterated through.

We learned that the first iteration of the oven, created 20 years ago, featured racks placed along the length of the oven with a single shaft connecting all of the pinions. The racks had a square profile, requiring a second support bar to guide movement of the EHC. The racks were fixed in place while the single shaft with the pinions would translate back and forth with the EHC rail. The first iteration of the design was successful because it was capable of actuating all pinions at a singular point (the end of the oven). The racks, pinions, and shaft were capable of functioning at temperature because of material selection. However, this design was expensive for the time and the budget did not allow the ovens to expand as customer needs increased.

The next iteration featured a rack with a rounded profile instead of a square profile. The pinions were still on a single shaft, along the length of the oven. This design has similar benefits to the previous design of actuated by a single point and parts which are resilient in the oven operating temperatures. One added benefit would be the elimination of the need for a support bar- because of the rounded profile, the rack itself could serve as a guide for EHC movement, simplifying oven assembly. Once again, this design was ultimately decided against due to higher costs associated with more racks being needed for oven expansion.

The third iteration of the Heller edge hold conveyor system was the current system with a chain drive in lead screws. In this design, a chain drive is used to turn lead screws placed along the length of the oven to move the EHC. The rotation of the lead screws caused the EHC rail to move inwards and outwards. Each of the lead screws requires relatively low torque to spin (can be spun by hand). This design was good as the lead screws and the chain drive system were cheap, allowing for easy expansion. The lead screw allowed for continuous rail width adjustment along the desired range. However, this design also ultimately failed from oven expansion- as the number of lead screws increased, the force transferred through the chain caused the idler gears to deflect, causing the chain to slip and the EHC to become inaccurate.

We also talked with our sponsor about mechanisms used by competitors. Our sponsor provided us information on one sponsor who uses a rack and pinion design in their oven. They also use a different heating module, but that is outside the scope of our project. The competitor uses an air motor to cause the pinion to move along the rack. The rack has a square profile (according to a picture) and interfaces with the shaft through a bearing block. This design is strong in accuracy and durability since everything is controlled through a single shaft and the air motor is relatively cheap- however, the rack and pinion could be expensive (exact finances for the competitors are not known).

The next step was to identify possible standards which could govern the design of our project. Because the project focuses on one system in one piece of manufacturing equipment, there are not many standards to govern how an EHC rail adjustment system must be built. Therefore, we expanded the scope of our search to include standards for the semiconductor manufacturing line as a whole. The set of standards that govern PCB manufacturing are set by SMEMA (Surface Mount Equipment Manufacturing Association). SMEMA standards cover all aspects of the semiconductor manufacturing process, including conveyor height, conveyor width, edge clearance, tooling pins, maximum gap, and lead-in [3]. In this project, we are primarily concerned with the conveyor width since we will be redesigning the EHC rail width adjustment mechanism. The only standard that relates to our project is SMEMA standard 2.3 which states that “for equipment with an adjustable conveyor width, the front rail is fixed and the rear rail is adjustable” [3]. Incorporating this standard will align with the current EHC rail adjustment system in the reflow oven.

It is also important to consider other designs that solve similar issues. One of the issues that these other designs need to solve is transferring rotary motion to linear motion. Additionally, the distribution and synchronization of motion achieved through the often rotary to linear motion transfer is an issue that needs consideration. There are many designs that have been implemented that effectively synchronize motion and transfer rotary motion to linear motion, some such designs include crank and sliders, worm gears, or rack and pinions [10]. Much of the associated challenges with motion synchronization and transference of motion between linear and rotary will be explored in the statics and dynamics sections of analysis.

Information Sources

The sources used in this project are comprised of a wide variety of media from technical standards to company websites to stakeholder interviews. The main standard that is considered for the scope of this project is IPC-SMEMA-9851 which details the most efficient ways to manufacture PCBs and the specific dimensions that different machines are required to meet. The most relevant portion of the standard is the description of interfacing between the oven and other machines which mandates which rail has adjustability and the sizes that it must adjust between. Our main source of information on the ovens and the details of the oven are conducted interviews with the sponsor and related personnel in the company. These interviews have provided us with the information necessary to make determinations on our design requirements and specifications. They have also given us the insight to help determine our design process.

Design Process

The effective use of a design process is crucial in the execution of a project because it ensures successful development of a final product in terms of cost, quality, and time invested. So far this semester, our team has been using the ME 450 design process outlined in the design processes learning block as a framework. Due to the nature of our project, and through encouragement from our sponsor, we have deviated slightly from the timeline associated with the ME 450 design process, but the major components remain the same. Our team received encouragement from our sponsor to begin the concept generation phase of the process early, but the course required a longer problem definition phase, so our team worked on these sections in parallel. The design process in use is summarized in Figure 5 below.

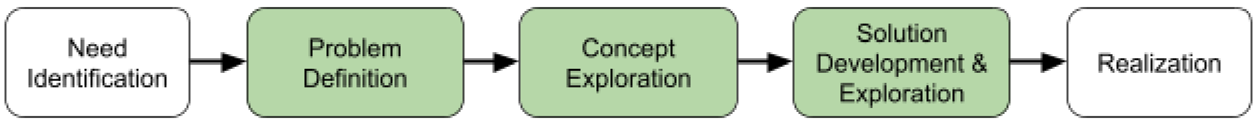


Figure 5: The design process framework in use by our team, where the green cells are the components that fall within the scope of our project and the white cells are loosely explored at their respective ends of the project timeline [6].

As demonstrated in Figure 5, the design process in use by our team is a linear, five stage model that is stage-based and problem oriented. The process is not cyclic in behavior and the problem is fully defined prior to solutions being generated. Our project is quite straightforward, with few extraneous variables and implications, so a basic design process is more than acceptable for the execution of our project and creation of a robust solution. Our team has considered using a more cyclic, activity-based approach, but in accordance with project timeline constraints and close guidance from our sponsor, we have determined this approach to be excessive. Moving forward however, a hybrid approach which employs both iterative and linear elements will likely be helpful. This approach is demonstrated below in Figure 6.

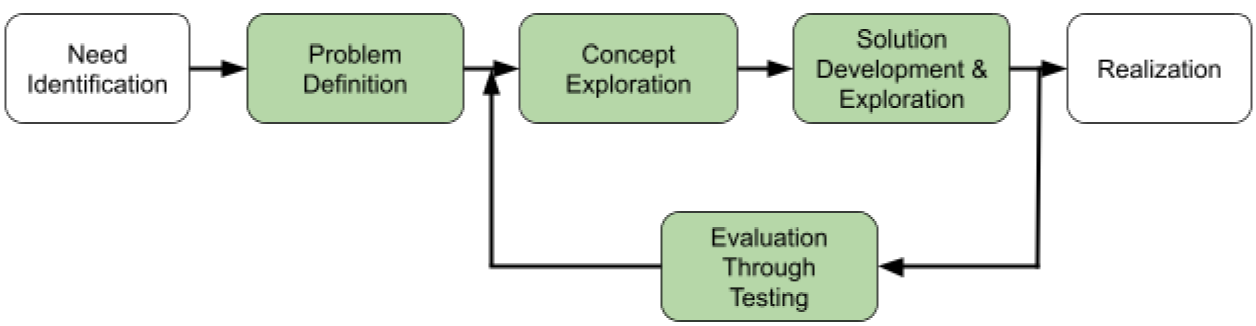


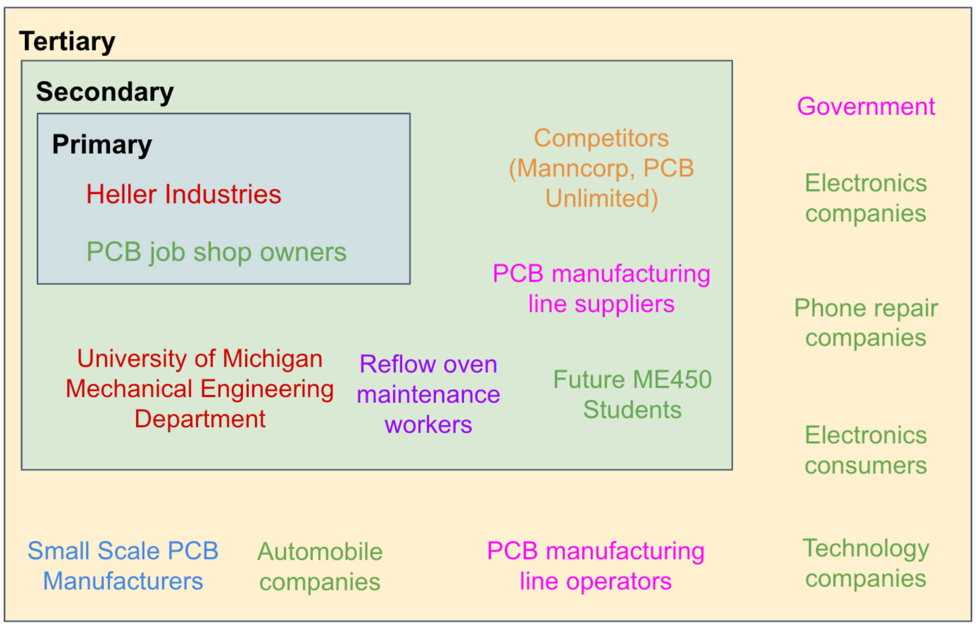
Figure 6: Design process framework that will be used moving forward. This model implements both iterative and incremental elements.

Using iteration in the solution development and exploration stage will allow us to apply rigorous engineering analysis to our prototype design, and make effective changes based on the results. A linear approach to solution testing can result in an ineffective pathway being followed without considerations of revision to the design.

As compared to the design process introduced on the first day of lecture, this design process is almost exactly the same. The primary difference between the employed design process and the ME 450 design approach is that the evaluation through testing does not impact the problem definition. As previously mentioned, the timeline of the project does not allow for a complete redefinition of the problem once a prototype has been constructed, so the evaluation iterates back to concept exploration, where the prototype is altered based on testing results. The ME 450 design process is acceptable for the oven drive chain project because the project is very straightforward and follows a linear pathway.

Stakeholders and Design Context

A stakeholder map for our project is presented below in Figure 7, along with each of their classifications.



Resource Providers Supporters & Beneficiaries of Status Quo Complementary Organization and Allies
Beneficiaries and Customers Opponents and Problem Makers Affected or Influential Bystanders

Figure 7: Stakeholder map for Heller Industries oven chain drive system. Primary and secondary stakeholders are limited, with tertiary stakeholders including all users of PCBs. Stakeholder types can be referenced using the provided key.

The project has a relatively narrow scope, seeing as it seeks to improve one mechanism on one oven used in the manufacturing process for PCBs. Therefore, the primary and secondary stakeholders in our project are limited.

One of our primary stakeholders is Heller Industries, which is the company that makes the ovens for which the chain drive needs to be redesigned. In our project, Heller functions as both a resource provider and a beneficiary. Heller provides us with information needed for the project such as CAD of the oven and operating conditions at which the ovens are run. Our goal is to redesign an edge hold system which interfaces with their reflow oven, so this information is crucial in developing requirements and specifications. Heller will also directly benefit from the completion of our project and will be able to sell a more reliable product thus increasing sales. Additionally, the higher reliability would reduce the number of Heller service visits to fix the oven (outside of quarterly maintenance, for those who have a service contract).

Another primary stakeholder would be Heller's customers - PCB job shops who use Heller ovens in their manufacturing line. These job shops would benefit from the completion of our project as more reliability means less machine downtime and more PCBs produced. With more products being sold, the shops can increase PCB sales and furthermore their profits. More reliability would also mean less service visits. Shops not under a Heller service contract would limit the expense of paying to have their oven repaired.

In terms of secondary stakeholders, one affected group would be reflow oven maintenance workers. These oven technicians would be negatively affected bystanders of the improved edge hold device because the implementation will decrease the demand for oven maintenance. Therefore, the workers would be losing a portion of their work opportunities and wages because the supply would exceed demand.

Another group of secondary stakeholders are other companies who manufacture reflow ovens, such as Manncorp and PCB Unlimited. These stakeholders would be proponents of the status quo because they are invested in having a more reliable product than their competitors. The unreliability of the current Heller product encourages consumers to consider other alternatives for their PCB manufacturing equipment.

A third group of secondary stakeholders are the set of SMEMA standards for semiconductor manufacturing [3]. The design solution will have to conform to these standards to be accepted by companies seeking to use Heller ovens in their manufacturing lines. Therefore, the standards are a complementary organization, as they serve to guide the design and ensure the new design still interfaces with other machines in the manufacturing process.

On a broad scale, the tertiary stakeholders for our project consist of any consumer who uses PCBs. These consumers include both companies who use PCBs in their products (any electronic device) and consumers who purchase these products. Improvement of the oven reliability will impact PCB consumers as delay times will be reduced, products can be distributed faster, and products will cost less.

REQUIREMENTS & ENGINEERING SPECIFICATIONS

Stakeholder requirements were defined through weekly interviews with the primary stakeholder, Heller Industries. Interviews were conducted with Jim Neville, the vice president of design engineering at Heller Industries. Interviews were also conducted with Erica Lu, a Heller scholar who worked on the engineering of Heller reflow ovens this past summer. In these interviews, an overview of the function of the oven was given which included its role in the PCB manufacturing line, the different types of Heller ovens, and different variations of the reflow oven. We then explored specifics of the edge hold conveyor (EHC) and investigated the modes by which the EHC failed. Following the failure analysis, we simplified the EHC to understand the basic functions the design must accomplish. Finally, we worked with Jim Neville to determine the highest priority requirements. In further meetings with our sponsor, we began to determine secondary requirements.

First, we established the parts of the oven that we could modify to implement the new edge hold width adjustment mechanism. One solution was that the chain drive system could be replaced with a new transmission system to turn the lead screws. Another solution was that the lead screws could be replaced for a mechanism which had less friction. A third solution would be both the transmission system to synchronize motion and the width adjustment mechanism could be replaced. Therefore, the scope of the project consists of the chain drive and lead screw mechanism and is pictured in Figure 8 below.

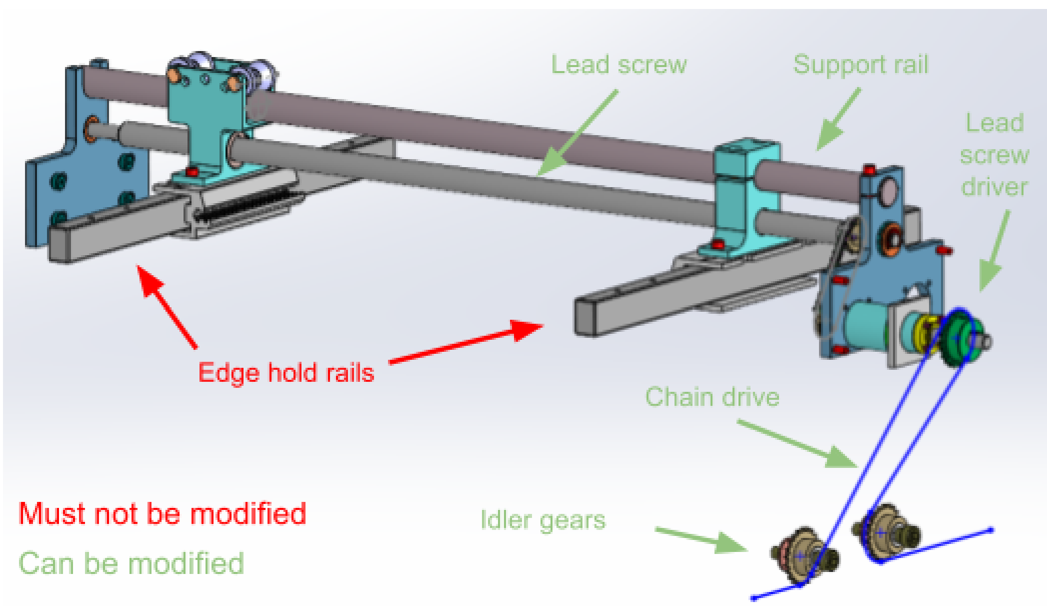


Figure 8: Labeled CAD of one lead screw adjustment mechanism, with parts that cannot be modified in red and parts that can be modified in green. This represents one out of four to seven lead screws in the oven, and they are coupled together by a transmission in the cold zone.

The highest priority requirements and specifications have been outlined below, in order of their relative importance.

Cost. Heller Industries currently has implemented a design change adjusting the width of the chain used in the chain drive from a ¼" to a ⅜" width [8]. Our sponsor has indicated that this design is currently being used, and is an effective solution to the problems identified earlier. Since there is already a design that is an effective solution, our group must come up with a solution that is not only just as, if not more, effective than the ⅜" change mentioned above, but also cheaper. After meeting with our sponsor, it was determined that an appropriate cost of the new mechanism would be < $300 for each lead screw [8]. To validate the cost specification, a bill of materials can be created with suppliers of each component, listing the prices to represent material costs. The total cost will be compared with the cost specification.

Durability. The primary failure mode in the current EHC relates to the forces on the idler gears and chain loosening due to constant tension. To build a more reliable system, the mechanism must be able to function for longer without fatigue. Hence, durability is the highest priority requirement for our system. To develop the specification for durability, we worked with the sponsor to learn about the standard use by their customers and asked the sponsor for a desired lifetime. The sponsor indicated that job shops changed the oven widths at most three times a day every day of the year, and the Heller warranty lasted for a year [8]. We agreed upon designing for a safety factor of two (i.e. designing for a two year lifetime) which results in a lifetime of 2190 cycles [8]. To verify durability, simulation will have to be used since it is not feasible to run a test of that duration within the semester (more in the problem analysis section).

Rail Width Adjustability. It is important for the rails to be at the proper distance to support the PCBs so product is not lost from falling into the oven. This requirement was defined with a minimum and maximum rail width, a length tolerance for each adjustment, and a speed at which the rail width adjustment must occur. The minimum and maximum rail widths are important in ensuring the design can accommodate all of the customer's board sizes. The specification was set to be a minimum board length of 50.8 mm and a maximum length of 508 mm based on customer usage learned from the sponsor interview [8]. The tolerance is important to ensure the EHC will be able to support the PCBs without being too tight or loose. The tolerance was given by the sponsor to be +1/-0 mm which reflects the current EHC tolerance [8]. To protect the PCB, only a positive tolerance exists to prevent compression of the board resting on the edge hold pins. Finally, the speed is important to ensure width adjustments do not cause significant downtime for the machine. The minimum speed was set to be 60 seconds for an 457.2 mm adjustment, which reflects a move from the largest to smallest (or vice versa) settings. The value of 60 seconds for a full sweep was prescribed by the sponsor [8]. To validate the rail width range, the measuring tool on CAD software can be used at the hardstop on each end of motion. To validate the tolerance and speed specifications, dynamic analysis can be performed on the system modeling the free rail distance change with each rotation of the motor. Ideally, the validation would be performed on a physical system, but the timeline and budget constraints will not allow that to be completed within the semester.

Design Integration. It is not feasible to redesign the oven in order to accommodate a new EHC adjustment mechanism. Additionally, it has been requested that the mechanism must fit within the outer housing of the oven to preserve aesthetics. These two requirements lead to a specification stating that the entire mechanism must fit within the outer dimensions of the provided oven. These dimensions were determined by provided CAD files from Heller Industries. Due to the complex nature of the CAD files and large number of dimensions that will be used to accommodate the new design, a placeholder of "SEE CAD" has been placed in the specification section.

The four top priority specifications are summarized in Table 1, arranged in order of priority and importance on the final design.

Table 1: High priority requirements with corresponding specifications. These requirements and specifications will lead the design process significantly.

Requirements	Specifications
Cost	• < $300 combined material & manufacturing cost per lead screw
Durability	• Can withstand 2190 complete rail size adjustments (lifetime 2 years assuming 3 width adjustments/day)
Accurately adjust rail width	• Ability to adjust rail width between 50.8 mm (2 in) and 508 mm (20 in) • Adjustable to within +1/-0 mm of target width • Takes less than one minute to move from largest to smallest and vice versa
Design Integration	• Oven Outer Dimensions ○ Length < 5895 mm ○ Width < 1450 mm ○ Height < 857 mm • See CAD

With the highest priority requirements selected, other requirements important to the design problem were identified. The team identified a number of additional requirements that are explained below. The additional requirements are presented in the order of importance to the sponsor and to the overall design.

Conditional resilience. Components inside the oven will be heated to temperatures up to 250 °C during oven operation [8] which can cause materials to soften, losing strength and functionality. Thus, a specification was written for internal parts to be able to withstand temperature cycles from 20-500 °C, representing room temperature and an oven temperature much higher than the current max oven operation condition to account for future modifications to PCB manufacturing [9]. It is unlikely that access to a Heller Industries oven will be achievable during the project timeline, so this specification will be verified through material analysis.

Length variability. Heller continually seeks to extend the length of their ovens to offer more heating patterns for their consumers - meaning the edge hold adjustment system has to extend with the oven. This has been one of the leading factors contributing to the failure of the current chain drive mechanism. This motivates the requirement for length variability. Communications with the sponsor indicate that the minimum oven length is 5.9 m and the maximum oven length is 8.69 m which provides a framework for a specification of the design being made to fit all oven models ranging from 5.9 m to 8.69 m. This specification will be verified through CAD or a scaled down model, since the budget will most likely not allow the construction of a 8.69 m solution.

Easy to maintain. In the rare case of a mechanical failure of the new EHC adjustment system, the new design must be easy to maintain/repair. This means that the Heller Industries repair team or customers who maintain and repair the ovens themselves must be able to quickly and cost effectively maintain/repair the mechanism. To meet these requirements, our design will be made to ensure that the design can be removed and replaced with using < 10 standard tools within 3 days. This will be tested when our team develops a proof of concept, the timing of which can be found in the project plan below. The number of tools and time to repair the chain drive was determined through communication with Erica Liu, a Heller Industries employee, who had previously replaced a chain drive with a repair technician [9].

Rigidity. Since the revised EHC adjustment mechanism will need to be in the hot zone of the oven, it must be able to withstand the high temperatures without deflecting. The allowable value of this deflection across the support rail due to thermal expansion is 0.1181in (3 mm). This value was determined based on the provided CAD model from Heller Industries and input from the sponsor. We will be able to verify this specification theoretically through beam bending analysis. We would need an oven and a dial indicator to verify this specification on the actual machine.

Punctures are insulated. Some aspects of the design may puncture the hot/cold zone interface. It is important that all of these punctures be well insulated to conserve energy and minimize electricity costs. Therefore, one specification was set for heat loss and another specification set for cold zone maximum temperature. The specification for heat loss was set to 50 W which represents a portion of the current oven power consumption. The specification for cold zone temperature was set to 40 degrees C which is the maximum temperature one can touch without receiving a burn [7]. Additionally, considerations need to be made regarding the thermal effects associated with meshing and moving parts in a high variable heat environment. Components in the hot zone of the oven will be subjected to thermal expansion which could interfere with the design functionality if not fully understood through analysis. This will be explored in the heat transfer analysis portion of the project timeline, where we will consider materials with varying values for thermal expansion coefficients. In the heat transfer analysis, different aspects of insulation will also be considered.

Safety. In addition to preventing heat loss from the hot zones, the entirety of the hot zone must be contained for safety reasons. There should be no way for someone to physically touch the hot zone during oven operation, and because of this, the design must incorporate some housing to prevent any unwanted human-oven interactions.

All additional requirements and specifications have been collected into Table 2, in order of importance.

Table 2: All additional requirements and specifications of the Heller Industries EHC chain drive remodel. These requirements and specifications are in order of importance, and will be used to drive the design process.

Requirements	**Specifications**
Conditional resilience	• Internal components can withstand temperature cycles from 20 - 500 °C • Internal components do not corrode in formic acid
Length variability	• Design can be fit to all oven models ranging from 232 in (5.893 m) to 342 in (8.687 m) [4]
Easy to maintain	• All parts can be removed and replaced using <10 standard tools • All parts can be removed and replaced within 3 days
Rigidity	• < 0.1181 in (3 mm) vertical deviation across length of support beam from room temperature dimensions (See CAD)
Punctures are insulated	• 40 °C outside temperature • < 50 W heat loss from hot zone to cold zone
Safety	• All hot zone parts are fully covered and insulated

CONCEPT GENERATION

We began our concept generation process with an analysis of the current oven edge hold adjustment mechanism and performed a functional decomposition. A functional decomposition was necessary to identify more reliable and durable solutions to the Heller edge hold adjustment mechanism. It was important to perform functional decomposition as a team to set a collective goal of the simplest functions of the design.

One primary function identified was the width adjustment of the edge hold rail. The design must be able to move the entire edge hold conveyor to a desired width to support the PCBs as they move through the oven. In the current design, width adjustment is performed with lead screws. The edge hold conveyor needs to move parallel to the fixed rail to ensure the width remains constant along the oven. Thus, the design needs to provide linear motion (along the width of the oven).

Another important function of the design is synchronization of motion along the length of the oven. The design must have synchronization to prevent misalignment of conveyor rails as the boards move through the oven. If the boards are misaligned, the PCBs could be compressed or fall into the heating modules of the oven. In the current design, synchronization is achieved with a chain drive which connects to each lead screw. This function can be generalized to a power transmission system along the oven.

A third function identified in the current design is the ability to support the edge hold rail. This function is important to the oven's operation as the edge hold conveyor solely rests on the width adjustment mechanism. The load of the edge hold conveyor, chains, and boards is currently supported by a rail placed below the lead screw on the EHC module. This function can be generalized to designs that must be able to support a load for an extended period of time.

Following functional decomposition, individual brainstorming sessions were conducted to generate ideas to reliably move the edge hold rail in the Heller reflow ovens. To generate concepts, a variety of techniques were used including a design heuristics and a concept tree.

Initial brainstorming focused on solving each of the functions independently. For example, designs to replace the lead screws included many designs which convert rotary to linear motion. Every member defined ~15 solutions for each function independently. Then, the set of 77 design heuristics from the learning block were used to create new concepts from the preliminary ideas. These designs were compared to eliminate repeated ideas and combined into a concept tree shown below in Figure 9 on page 18.

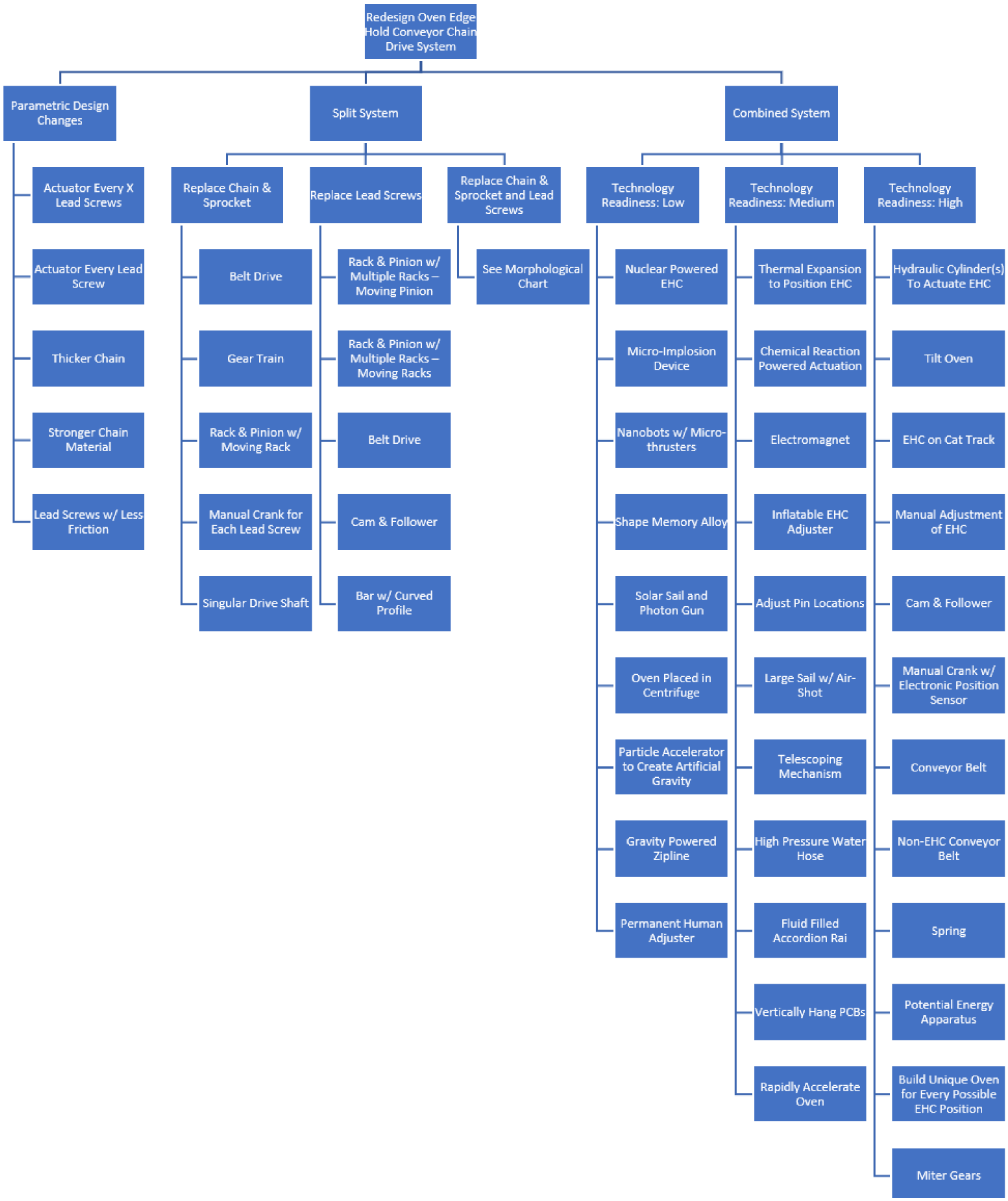


Figure 9: Concept tree containing all possible designs. Designs were split into three main categories. Parametric design changes, split system, and combined system.

All generated designs are shown below in Appendix D. In order to narrow down the large pool of concepts into those deemed viable, all generated designs were placed into one of three categories: parametric design changes, split system, and combined system.

Parametric Design Changes

These designs change the current chain drive and lead screw system by replacing parts with similar replacements that did not change the overall features of the system. Some of these concepts included increasing the thickness of the chain and adding more motors to the system.

Split System

These design changes seek to improve the system while keeping the oven operating as two distinct systems. These designs were split into two groups, one that kept the chain drive but replaced the lead screws, and one that kept the lead screws but replaced the chain drive. These concepts were then placed together into a morphological chart in order to determine if any of the designs and/or their combinations were viable. The morphological chart can be seen in Figure 10 below.

Coupled Motion Across Oven

Adjust Edge Hold Width

	Belt drive across all lead screws	Gear train along oven length	Singular moving rack along length of oven with pinions to spin each lead screw	Manual crank to spin all lead screws at once	Singular drive shaft
Rack and pinion w/ fixed rack, moving pinions	DAI	DAI	CDA	CDAI	CDAI
Rack and pinion w/ moving racks, fixed pinion	DA	DA	DA	CDA	CDA
Belt drive to turn lead screw	AI	AI	A	AI	AI
Cam & follower with spring to hold			C	C	C
Auger	I	I	C	CI	CI

Figure 10: Morphological chart that looks to combine the solutions of the split system. Each design was rated on its ability to meet all four of the primary design requirements/specifications, represented as follows: Cost (C), Durability (D), Accuracy (A), and Design Integration (I). Designs in green met all four primary requirements/specifications.

Combined System

These designs removed both the lead screw and the chain drive and replaced them with a single system to move the EHC. In order to further evaluate these designs they were separated into three sub categories based on the technology readiness needed to produce the new design cheaply and effectively. Due to the nature of unencumbered brainstorming, some designs were immediately placed in the technology readiness: low category. These designs, while interesting, are extremely unlikely or not feasible. Some of these designs include a nuclear powered EHC, a particle accelerator powered EHC, and a solar-sail powered EHC. The second level of technology readiness was medium. These designs could be implemented today, but there would be significant challenges associated with cost and redesigning the oven in order to incorporate them. Some of these designs included a chemical reaction to move the EHC, an electromagnet to move the EHC, and a fluid filled accordion to move the EHC. The last category in the combined system was technology readiness high. These designs included technology that is widely available and can be implemented into the oven relatively cheaply and without changing the overall design of the oven significantly. Some of these designs included hydraulics, using springs to move the EHC, and a manual adjustment of the EHC.

CONCEPT SELECTION PROCESS

In order to assess the viability of multiple designs, and determine the most suitable design replacement, a collection of designs were inputted into a Pugh chart. This chart included the most viable parametric design changes, the designs from the morphological chart that met all four primary requirements, and designs from the technology readiness high subtree. These designs were compared to the current oven design, and were scored based on the requirements and specifications outlined above. Each design requirement and specification was given a weight of 1, 3, 6, or 9 based on its importance to the overall design. A weight of 9 indicated high importance and was given to the four primary requirements and specifications. A weight of 1 indicated low importance and was given to design requirements and specifications deemed low priority by our sponsor. Each design was then given a score of 1, 0, or -1 for each design requirement and specification. A score of 1 indicated that the design performed better in meeting the requirement than the current design. A score of 0 indicated that the design performed similarly to the current design. A score of -1 indicated that the design performed worse than the current design. The collection of this Pugh chart can be seen in Figure 11 below.

Criteria	Weight	Current Design	Design 1	Design 2	Design 3	Design 4	Design 5	Design 6	Design 7	Design 8	Design 9	Design 10	Design 11	Design 12
Cost	9	0	1	-1	-1	-1	-1	1	-1	-1	-1	1	-1	0
Ease of Manufacturing	1	0	-1	1	1	0	0	-1	-1	-1	-1	1	-1	-1
Repairability	3	0	-1	1	-1	-1	0	-1	0	-1	-1	1	0	-1
Ease of Maintenance	3	0	-1	1	1	-1	0	0	-1	0	-1	1	-1	-1
Durabilty	9	0	1	1	1	0	1	-1	1	-1	1	1	-1	1
Efficiency	6	0	1	1	1	-1	0	-1	0	1	0	-1	-1	1
Compatibilty with oven	9	0	0	-1	-1	-1	0	0	1	1	-1	-1	-1	-1
Accuracy of movement	9	0	1	1	0	-1	0	-1	0	1	1	1	-1	1
Thermal resistance	6	0	0	0	-1	0	0	-1	0	-1	0	0	-1	0
Time to move	3	0	1	0	1	1	0	1	0	1	1	-1	1	1
Rigidity	6	0	1	1	1	0	1	-1	1	-1	0	0	1	1
Impact on insulation wall	1	0	1	-1	-1	1	0	0	0	0	-1	-1	-1	-1
Safety	1	0	0	0	0	-1	0	-1	0	0	0	-1	0	0
Adaptability	6	0	1	1	1	1	0	0	1	0	1	1	1	1
Score	72	0	42	24	6	-30	6	-29	17	-7	1	20	-38	22
Normalized Score		0	0.583	0.333	0.083	-0.417	0.083	-0.403	0.236	-0.097	0.014	0.278	-0.528	0.306

Figure 11: Pugh chart comparing the various designs. The overall score of the design can be seen in the second to last row, and the normalized score can be seen in the last row. A positive score indicates the design performed better than the current design and a negative score indicates the opposite. The overall best design was Design 1.

The designs being used in this Pugh chart are expanded on in Table 3 below.

Table 3. Description of 12 designs used in Pugh chart above. These 12 designs were based on the selection criteria described earlier.

Design #	Description
Design 1	Rack and pinion with multiple racks. Racks run along the width of the oven and are fixed to the oven walls. Pinions are connected by a singular shaft that runs the length of the oven.
Design 2	Multiple actuators. This places an actuator on every lead screw. Actuators will be kept in sync by a control system.
Design 3	Hydraulic cylinders. Uses hydraulic cylinder(s) to move the EHC into place. Hydraulic cylinder(s) will push EHC back and forth along support rails spanning the oven width
Design 4	Oven Tilt. Lift one edge of the oven to move the EHC into place. EHC will slide along a support rail into the desired position
Design 5	Increase chain thickness by ¼" to reduce chain stretch and slippage.
Design 6	Spring. When the EHC moves, a spring compresses. To move the EHC back, the spring is released. EHC is moved by a solenoid or some other type of linear actuator
Design 7	Multiple chains. Multiple chains running the length of the oven instead of a singular chain. Each idler gear will have two chains attached, and each chain will span between two consecutive lead screws.
Design 8	Extending pins. Move the pins that the PCBs rest on instead of the EHC in and out (along oven width) to accommodate different PCB widths.
Design 9	Crank and slider. Crank and slider to move the EHC into position. Crank and slider located above the EHC inside the oven cavity. Slider runs along a guide which spans the width of the oven
Design 10	Manual crank on each lead screw. Place a human powered crank that turns each lead screw individually. Electronic sensor to indicate position
Design 11	Cam and follower. Cam and follower to move EHC into position. Cam and follower located above the EHC inside the oven cavity. Spring used to hold the EHC in place
Design 12	Rack and pinion w/ linkage. A singular long rack runs the length of the oven which turns pinions attached to a linkage that moves EHC. Rack moves in and out along the length of the oven to rotate pinions

After completing analysis using the Pugh chart, the overall best design was determined to be a rack and pinion with a singular drive shaft. This design scored the highest, and through talks with our sponsor, was confirmed to be viable. Illustrations of each concept are at the end of Appendix D.

CONCEPT DESCRIPTION

After thorough down selection, the rack and pinion was selected as the best design. In this design the lead screws are replaced with racks and the chain drive is replaced with a central shaft running the length of the oven. The shaft connects pinions set at each rack, and when driven, moves the pinions and edge hold along the racks. Preliminary CAD of one such rack-pinion junction can be seen below in Figure 12.

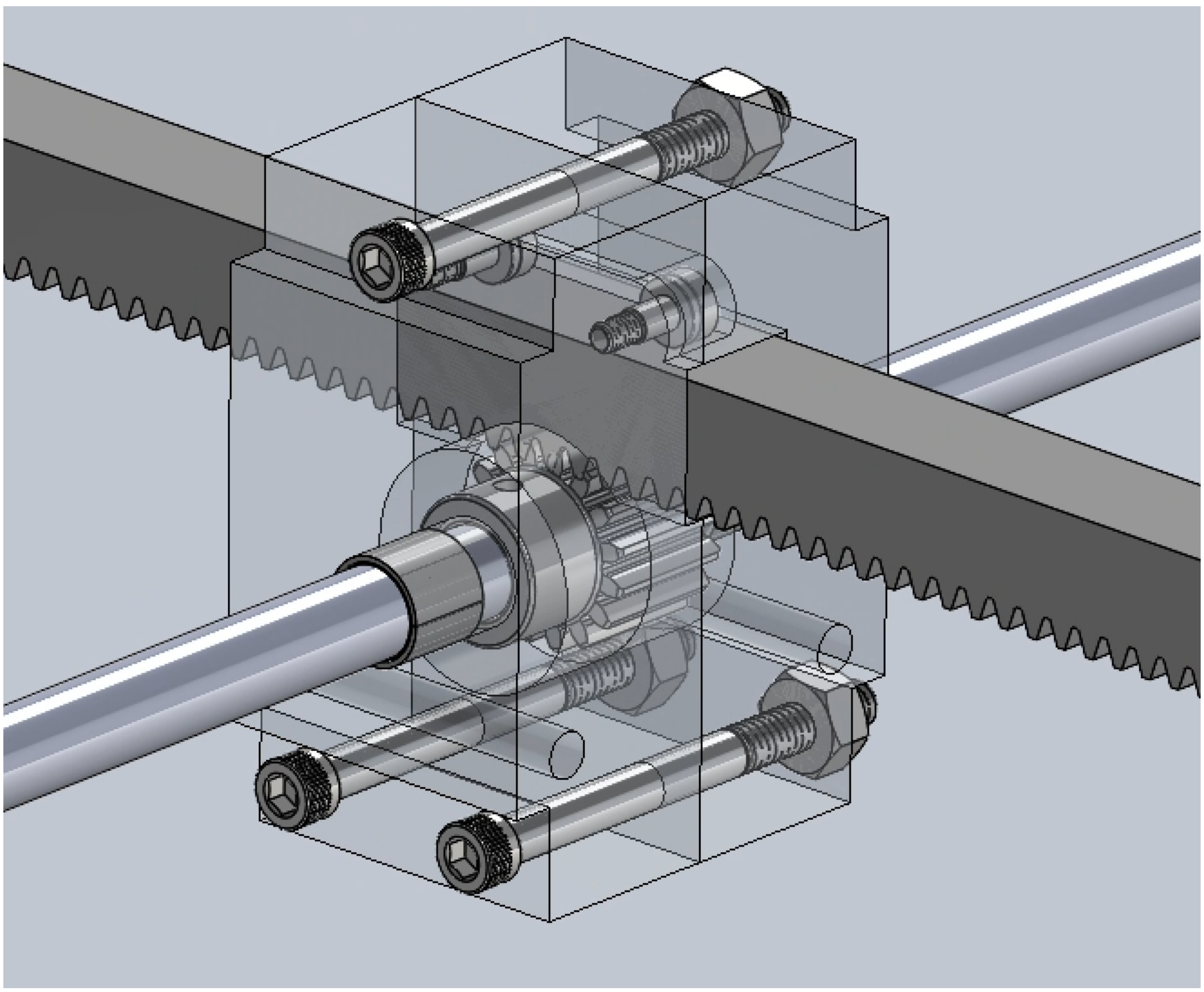


Figure 12: This figure shows one internal module of the rack and pinion design. This design will replace each lead screw in the same location, and the central shaft will run the length of the oven.

As demonstrated in Figure 12, a module of the internal component of the motion transmission shows how the shaft and pinion will move along one of the stationary racks in the oven. A diagram of how the final design will move and interact with the oven is provided in Figure 19 on page 34. The rack is fixed to the walls of the oven at both ends and the central shaft, pinion, and housing move together laterally to adjust the width of the EHC. The EHC is attached to the bottom of the housing of the motion transmission components and moves laterally as the pinion moves along the rack. The central shaft is attached to the EHC rail by the pinion housing that uses the rack as a structural support. The housing is very important for the design because it is responsible for ensuring that the rack and pinion mesh and transferring the weight of the shaft and EHC to the rack. The pinion contacts the rack on the bottom surface of the rack in order to minimize interference with the current oven dimensions as well as minimize the radial forces on the pinions. The shaft is driven by a motor at one end which resides outside of the hot zone of the oven. The motor will be coupled to one end of the shaft through a support bracket that will move along a track as the shaft moves back and forth. The track and motor will be outside of the oven so that the motor does not interact with the hot portions of the oven.

The main subfunctions that the design must satisfy are the width adjustability of the EHC, synchronization of motion along the length of the oven, and ability to support the EHC. The rack allows for the width adjustment of the EHC, the central shaft satisfies the synchronization of motion along the length of the oven function, and both the rack and the housing allow for the design to support the EHC.

While the selected design was the design that the sponsor originally had in mind, this design was derived mostly independent of sponsor intervention. The sponsor aided in concept selection by providing important information regarding the ovens and discussing any preliminary design concepts to determine feasibility of the concept and whether or not it was worth flushing out. The sponsor kept an open mind and did not dismiss concepts that were not the preferred rack and pinion design, but instead provided objective criticisms of all of the concepts. When performing the morphological analysis and Pugh Chart, the numbers were not altered in order to satisfy the sponsor; the scores given for each design in both the morphological chart and Pugh Chart were discussed among the team to ensure that a consensus was reached and to minimize any possible personal bias.

ENGINEERING ANALYSIS

Maximum Stress - 1st Principle Analysis

To determine the rack dimensions, a stress analysis was performed to determine the location and magnitude of the maximum stress along the rack, based on various rack sizes. By utilizing a MATLAB script, multiple parameters including the characteristic rack dimension and the weights of the shaft, housing, and EHC were input into the equation for stress on the rack. The stress equation was obtained from the simplified scenario of a beam with point supports at either end, a load at the midpoint (max stress location), and a distributed load from the rack weight. Approximate values were used for the weight of the shaft, housing, EHC, and pinion because these values are all subject to change based on future analysis (i.e. the diameter of the shaft has not yet been determined but influences the weight of the shaft). As further analysis was conducted, the approximate values were updated with more accurate values to ensure that the chosen rack dimension still meets the requirements.

To calculate the stress in the rack, a basic free body diagram (FBD) was constructed for the rack, then the reaction forces were calculated. The free body diagram of the rack is shown below in Figure 13.

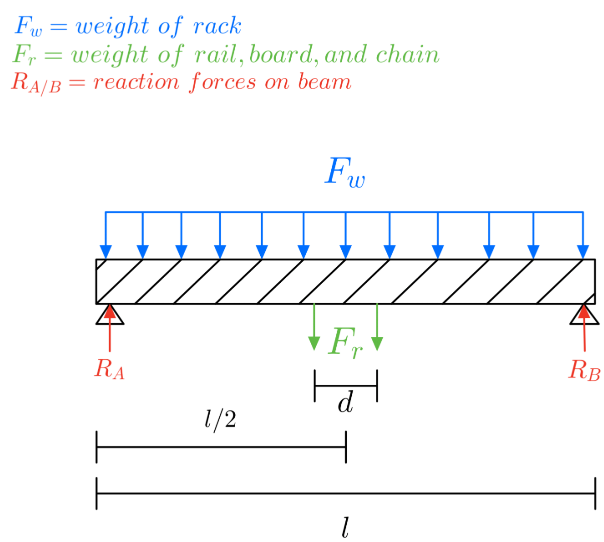


Figure 13: Free body diagram of rack used for determining maximum stress

Using the FBD of the rack, the maximum bending moment can be determined which is then used to calculate the maximum stress in the rack. The equation used for the maximum stress is given below:

$$\sigma_{max} = (F(\frac{l}{2} - \frac{d}{2}) + \frac{Wl}{4}) * \frac{0.5t}{I}$$

Where F is half the weight of the shaft, housing, EHC, and pinion, *l* is the length of the rack (24 in or 0.61 m), *d* is the distance between the rollers, *W* is the weight of the rack, *t*, is the characteristic dimension of the rack, and *I* is the second moment of area of the rack. Because the available racks have a square cross-sectional area, the characteristic dimension of the rack is given as the face width of the rack. Various characteristic dimensions were considered ranging from 0.125 in to 1.5 in. The value of max stress was compared to the yield stress, and if it exceeded the yield stress, that size rack could not be used. The yield stress for 1018 carbon steel was determined to be 370 MPa [11]. A comparison between the max stresses in the rack and the yield stress is given below in Figure 14.

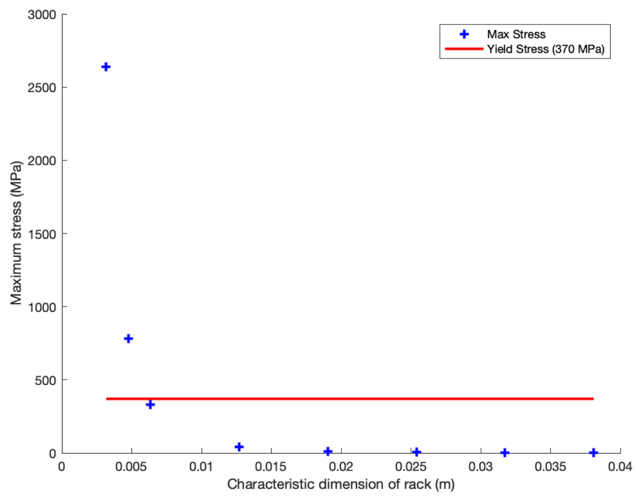


Figure 14: Plot of max stress for various characteristic rack dimensions compared to yield stress

Based on this plot, a characteristic rack dimension of 0.75 in would be used (this corresponds to the fifth point in the plot in Figure 14 above).

Rack Deflection - 1st Principle Analysis

Based on the rack size determined above, the vertical deflection of the rack could be calculated to ensure that the rack did not deflect more than the specification of 0.1181 in (3 mm). The equation used to calculate deflection is given below.

$$v_{max} = \frac{-L^3}{EI}\left(\frac{7W}{128} + \frac{F}{12}\right)$$

Where v_{max} is the max deflection of the rack, *E* is the Young's Modulus, and *I*, *W*, and *F* are the same quantities mentioned above. Based on our first principles analysis, using the chosen rack size would result in a deflection of 0.5 mm which is well below the required 0.1181 in (3 mm).

Rack Deflection - FEA

In order to verify the results of the 1st principles analysis of the beam deflection produced above, the rack being implemented into the final design was placed into a FEA software to determine the maximum deflection with the expected load. Similar to the conditions assumed for the calculations of maximum stress and beam deflection on the rack, point loads at the roller contacts were applied as well as the gravitational effect on the rack from its own weight. End conditions that were representative of actual conditions were applied and the finite element analysis software produced expected results regarding the deflection of the rack. A screenshot of the visually exaggerated deflection of the rack can be seen below in Figure 15.

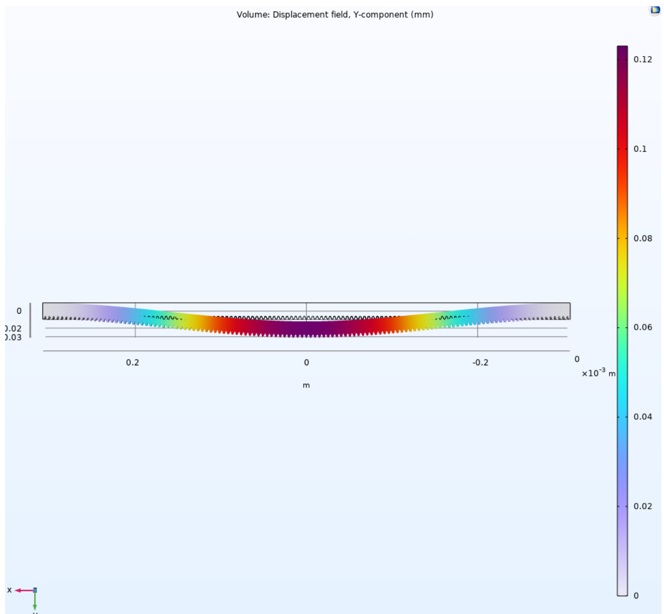


Figure 15: Maximum deflection of the rack with expected load. Max deflection occurs at the midpoint and is limited to 0.0047 in. This deflection is far below the required minimum deflection of 0.1181 in.

Through the 1st Principle Analysis and FEA, it was verified that using a 0.75 in x 0.75 in rack for the design would meet all necessary requirements.

Shaft Torque - 1st Principle Analysis

After determining the rack size, the next step was to determine the torque required to move the long shaft connecting all of the pinions. Similar to the 1st principle analysis of the maximum deflection, a MATLAB script was written to find the input torque required to rotate the shaft and pinions as a function of different shaft diameters and different shaft lengths (depending on how long the oven is). Various shaft diameters gathered from different common off the shelf items were input into the required motor torque equation to find a maximum torque value. This value dictates which motor is selected. The following equation was used to calculate the required motor torque.

$$T_m = n\frac{d_s}{2}\mu[(W_{shaft} + W_{pinion})(1 + \frac{d_P}{d_s}) + W_{housing} + W_{EHC}]$$

Where n is the number of modules in the oven, d_S is the shaft diameter, d_P is the pinion diameter, μ is the coefficient of friction, and W is the weight of each part. The shaft weight, pinion weight, and pinion diameter were dependent on the shaft diameter, so for each shaft diameter, a corresponding shaft and pinion weight was used. Using the above equation, the torque required to move the shaft was calculated for various shaft diameters. Graphs were created for each shaft diameter where the motor torque was plotted against oven length. An example of one of these graphs is shown below in Figure 16 for a shaft diameter of 0.5 in.

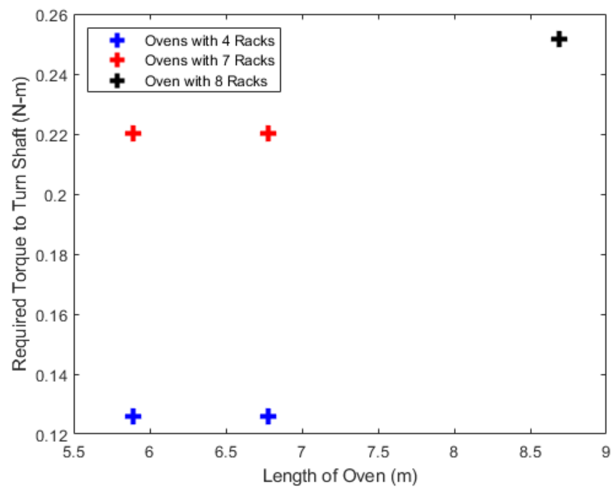


Figure 16. A plot of the required torques for various oven models

Calculation of the required torques is important for knowing which motor to choose, but it is also important for calculating the angular deflection (or twist) in the shaft. The twist in the shaft is very important because for such a long shaft length, it is possible for the racks to be misaligned should one end of the shaft be twisted relative to the other end. To calculate the maximum angular deflection in the shaft, the equation below was used.

$$\Delta\phi = \frac{T_z L}{J_z G}$$

Where $\Delta\phi$ is the maximum angular deflection in the shaft, T_z is the torque transmitted through the shaft, L is the length of the shaft, J_Z is the second polar moment of area of the shaft, and G is the shear modulus of the shaft. Using this equation, graphs were created for each shaft diameter where the angular deflection was plotted against oven length. An example of one of these plots is shown below in Figure 17 for a shaft diameter of 0.5 in.

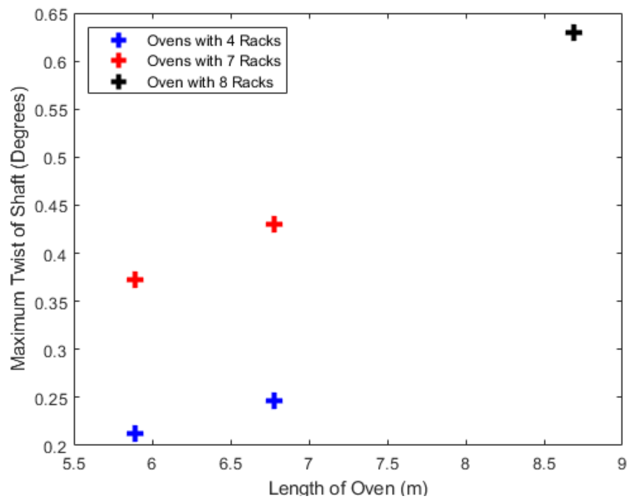


Figure 17: A plot of angular shaft deflection for various oven models

Per Heller Industries specifications, there must not be more than a 1 mm difference in rack positions relative to each other. This means that the maximum allowable angular deflection is the angle corresponding to a 1 mm arc length. This maximum allowable twist decreases as the diameter of the shaft (and thus the diameter of the pinion) increases. These maximum allowable angular deflections were calculated for each shaft diameter and compared to the plots of angular deflection versus oven length plots to ensure that all of the angular deflections were well under the maximum allowable deflection. After comparing all of the plots, a shaft diameter of 0.5 in was determined to be the best because too small of a shaft would lead to large angular deflections but too large of a shaft would result in large torques and rack position differences larger than 1 mm (due to the larger diameter pinion).

Hertz Stress - 1st Principle Analysis

When speaking with representatives from our sponsor, Heller Industries, it was indicated that previous designs had failed due to gear teeth breaking after being cyclically loaded over extended periods of time. Therefore, it was requested that an analysis of the Hertz stress, or contact stress, between the rack and pinion teeth be calculated in order to determine if any necessary design changes needed to be made. After performing research into the calculations necessary to determine the Hertz stress, the following equations [12] were used.

$$\sigma_{Hertz} = \frac{4F}{\pi B l}$$

$$\text{with } B = \sqrt{\frac{8F}{\pi l} * \frac{(1-\upsilon_1)/E_1 + (1-\upsilon_2)/E_2}{(1/D_1) + (1/D_2)}}$$

Where F is the tangential force transmitted between gears, l is the face width of the meshing gears, $v_{1,2}$, $E_{1,2}$, and $D_{1,2}$ are the poisson ratios, Young's Moduli, and diameters of the two gears that are meshing. Based on this equation, and the predetermined materials that will be used for the design, the Hertz stress was determined to be 85.83 MPa. After bringing this value to Heller

Industries, it was indicated that this value falls far below the stress that would cause issues in the design.

First, a FBD of a singular rack was made using an overestimate of shaft weight to obtain the maximum bending stress and maximum vertical deflection. Housing weight was estimated using the density and volume from CAD, the pinion and component weights were found from McMaster, and the shaft weight was overestimated since it was an unknown design variable. The design variables found in this scenario were the rack width and height. Multiple inputs for rack height and width were analyzed to determine which racks would satisfy the requirement that the racks must deflect less than 0.1181 in using racks available on McMaster-Carr to minimize cost. Through this process, the cheapest rack that could also satisfy the deflection specification with a safety factor of at least 2 was found to be a 0.75 in x 0.75 in rack with a gear pitch of 16. The next design variable chosen was shaft diameter using the accuracy specification and torque analysis. Using the determined rack size, the options for different pinions were limited to those with a gear pitch of 16 and a face width of 0.75 in. Each of the pinions were meant for shafts of different sizes. Each of these shaft-pinion pairs were iteratively analyzed to determine which would satisfy the accuracy specification and have the lowest torque required to turn the shaft. The torque required to turn the shaft was largely based on friction from the rollers and bearings and efficiency of the meshing gears. Friction of the rollers was set based on a conservative estimate due to time constraints of obtaining parts and empirically determining the rolling resistance. Through this iterative process, a shaft diameter of 0.5 in and a pinion with a gear pitch of 16 and pitch diameter of 1 in were determined to be the optimal shaft-pinion pair. Finally, a motor was selected using the calculated torque required to rotate the width adjustment mechanism. Pinion size was used to determine the required motor velocity to meet the speed adjustability specification. With a torque and speed operating conditions, a motor could be chosen to complete the system design. The motor that was chosen was a Pololu Metal Gearmotor (part number 3485)

Thermal Analysis

All analysis necessary for the build design of the product was completed. In contrast to the actual design, the build design will be tested in ambient temperature, and therefore will not need any thermal analysis to be completed. Since the final design will be placed into the Heller Industries reflow oven, it will need to meet the specifications that relate to the high temperatures associated with the oven. This means that the thermal analysis is unnecessary to the success of the build design, but is for the final design, so the thermal analysis should be completed in parallel to the construction of the build design. To meet these requirements, planned thermal analysis including pinion shaft deflection under heat, bearing and roller efficiency at temperature, and heat loss at the interface between hot and cold zones will be performed in order to measure the effects of thermal expansion, deformation caused by high temperatures, and changes in the physical properties of the selected components. It is expected that these analyses will lead to a material determination, and will not affect the other analyses already performed. If necessary, the previous analysis can be redone if the thermal analysis yields unexpected results.

BUILD & FINAL DESIGN DESCRIPTION

Because of the large physical scale of the design solution, a build design was selected to convey the problem solution in addition to proving adherence to the project requirements and specifications.

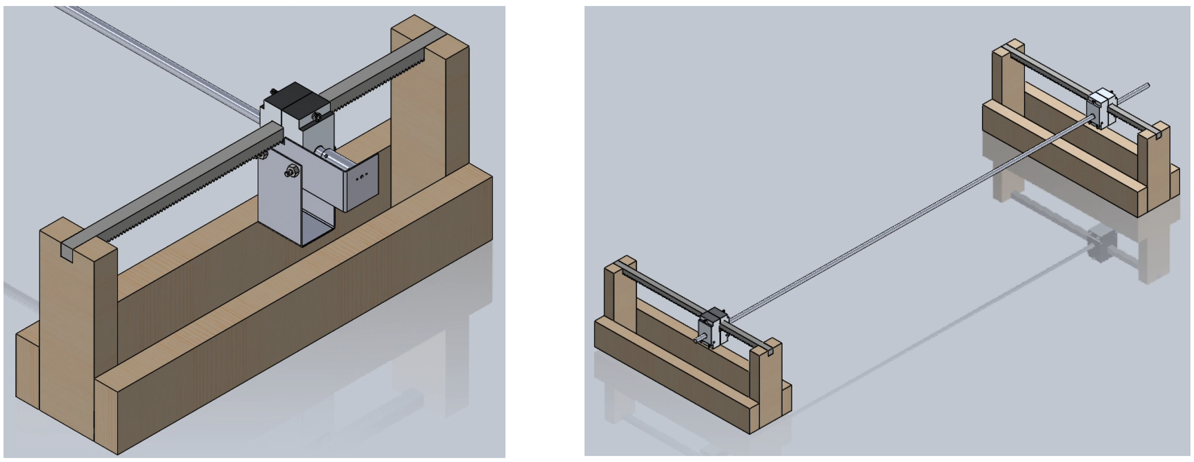


Figure 18a: This figure shows the build design of the rack and pinion solution. This design simulates two modules of the oven with the rack and pinion mechanism replacing the lead screw and chain drive assembly.

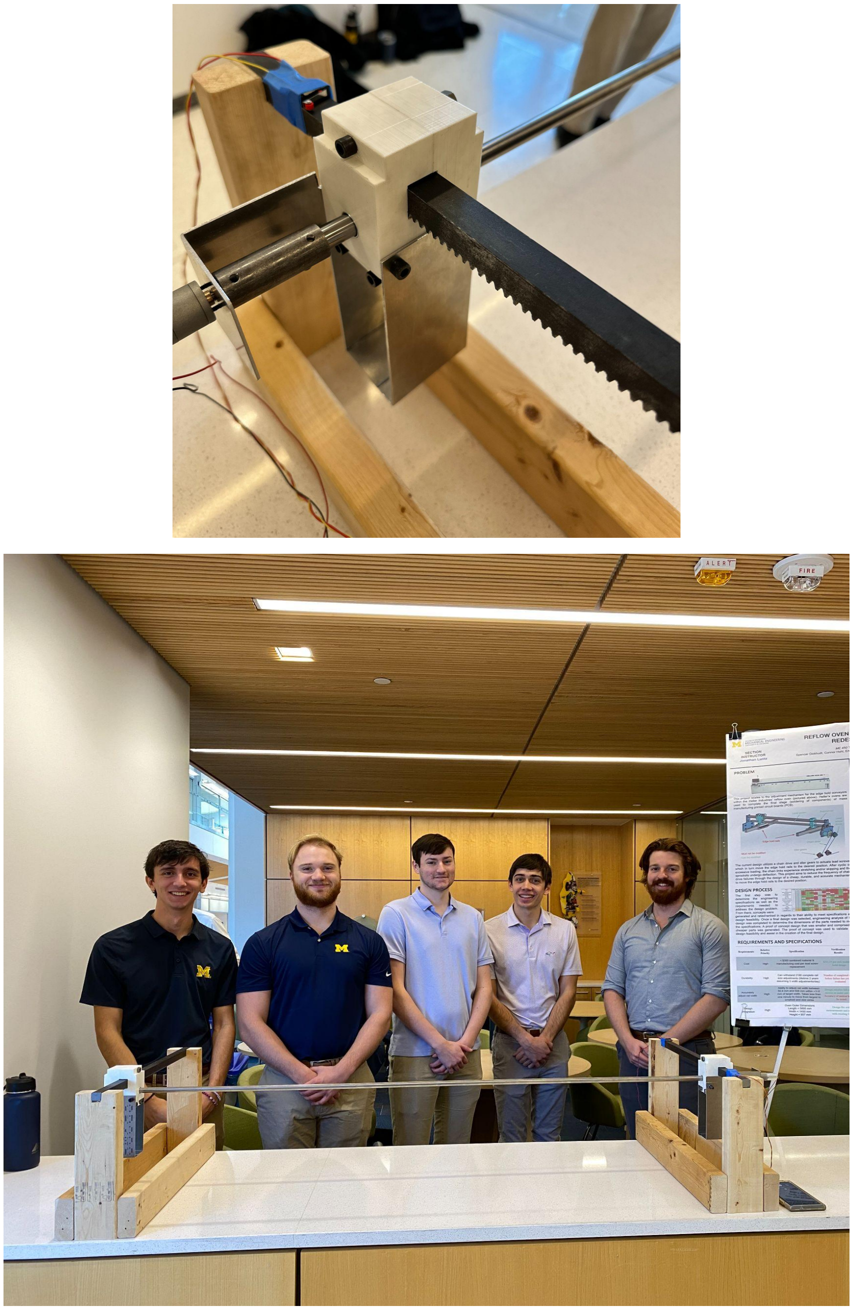


Figure 18b: These pictures show the prototype of our build design presented at the design expo. The first picture shows one module (specifically the one where the motor is connected), and the second shows the entire build.

Figure 18 demonstrates the selected build solution which simulates two modules of the EHC rail movement systems within the oven. Each lead screw and support cross-member in the original design is replaced with a rack. The racks are stationary and stabilized within a plywood housing which imitates the supports within the oven that the rack will be rigidly attached to. Each rack is meshed with a pinion on the underside of the rack. The pinions are rigidly connected to a single, cylindrical shaft which spans the length of the build design. Each pinion and its respective rack is enclosed by a housing that hangs from the rack and translates along the top of the rack with the assistance of rollers. The pinion, rollers, and necessary bearings will be housed inside of a 3D printed housing. The first housing will be specially designed in order to hold the motor, which will move along with the shaft. To simulate the weight of the EHC rail, chain, and PCBs, a sheet metal housing hangs from the bottom of the housing through which weight can be loaded. By turning the shaft, all of the pinions translate along the racks in unison which moves the simulated EHC rail in and out.

Many of the parts and materials used in this build design are off the shelf and readily available for production of the actual design solution. The bill of materials for the build design is demonstrated below in Appendix A.

The first step in the manufacturing and assembly of the build design is to acquire all of the off the shelf parts. As shown in the bill of materials, the only parts that were not off the shelf parts were the housing of the pinions and the bracket for the motor. Both of these custom parts were 3D printed and no manufacturing by hand was required. The housing and the motor bracket were printed from PLA on an Ender V3 3D printer. Additionally, four sections of 2X4 required notches to be cut out to support the rack and the purchased sheet metal was bent to support the simulated EHC weight. The vertical 2x4 sections that required a slot for the racks were cut using a table saw and the sheet metal was bent using a brake. To assemble the build design, the first step was the assembly of the wooden base. The horizontal 2x4 sections were then attached to the vertical 2x4 sections with screws. To assemble the EHC movement mechanism, the racks were press fit into their respective vertical 2x4 slots. Next, the bushings were pressed into the shaft holes in the housing. Following this, the housings were placed around the racks with the pinions between the housing shaft holes. The shaft was then inserted into the bushing holes as well as through the enclosed pinions. The pinions were then tightened to the shaft by a set screw. At this point, the housing rollers were threaded into their appropriate holes and the housing was closed around and rested on the rack. At the near end of the shaft, the motor was attached to the shaft by the coupler and the bracket was attached to both the housing and motor to support it. Lastly, the sheet metal housings were bolted to the pinion housings. The motor was then connected to power and the mechanism was complete.

The most important tolerances in the build design are the tolerances between the housing and the shaft and racks, and between the motor mount and the motor. The housing tolerance is crucial because the housing is the primary method of perpendicularly aligning the shaft with the racks and any deviation in the housing could misalign the modules. Additionally, the housing is in charge of keeping the pinions engaged with their respective racks. For example, if the tolerance for the shaft hole in the housing allows for too much vertical deviation, the shaft and pinions might be too low to effectively engage with the rack and the design will fail. The motor mount tolerance is also very important because the motor mount is what decides the alignment of the motor with the shaft. Because of the selected coupler, the motor shaft and the shaft have to be perfectly collinear. The motor mount must have a tolerance tight enough to align the motor with this precision. Some tolerances that are less important in the build design are the tolerances between the sheet metal EHC simulator and the housing and between parts of the base. These tolerances do not have to be as tight as those previously mentioned since they will not directly influence the functionality of the mechanism and rather fill peripheral roles. Additionally, these tolerances will not exist in the actual design because different supports will be used in the oven and the actual EHC will be present. Many of the most important surfaces in the build design include surfaces that are pre-machined on purchased components. Included are surfaces like those on the faces of teeth on both the racks and pinions, the surface of the shaft, and the surfaces of the bushings. The tolerances of finish on these surfaces are crucial to the success of the build design, and more broadly, the final design, because the accurate movement of the mechanism requires a smooth rotation or translation through or across them.

The success of these factors are largely indicative of the success of the overall design because they encapsulate many of the requirements set out for the project. With these metrics in mind, the build design will demonstrate the feasibility and performance of the final design as needed to validate the solution. This build will help demonstrate the value added to the project by our team by showing the functionality of the mechanism and the simplicity of the solution when compared to the design currently employed by Heller Industries. Additionally, it will exemplify how a price of $300 per adjustment module is achievable.

The final design will be very similar to the build design in many ways, but there are a few major differences between the two designs. First, due to the fact that the build design was tested at ambient room temperature no consideration was given to thermal material properties. This is not the case for the final design as it will be placed within one of Heller Industries reflow ovens. As noted in the engineering specifications above the final design must be able to withstand heat cycles between 20-500 degrees Celsius. This will put a significant thermal stress on the components and will require materials capable of withstanding these thermal fluctuations. Because thermal analysis is yet to be completed, the materials that can be used in the final design are not yet determined. As discussed earlier, this analysis will be completed in parallel with the construction of the Build Design. In addition to not using thermal resistant materials in the build design, cheaper components were used. This was to ensure that a working POC could be produced on time and within a limited budget. In this final design those constraints will not be in place allowing for more robust components to be used. The pinions will be fixed to the shaft using a keyed shaft instead of set screws, the oven walls will replace the plywood supports, and the housing will be machined as opposed to 3D printed. Another difference between the build design and the final design and final design is the mounting method. For the final design, the racks will be mounted to the walls of the oven using angle brackets to ensure the racks do not come loose. Based on FEA analysis this method of mounting will not result in any significant differences in the stress or deformation profiles of the rack when compared to the build design. The final difference between the build design and the final design is the scale. The build design only has two racks and two pinions whereas the final design will have up to eight racks and pinions along a single shaft, and the shaft will be up to 342 in long instead of 60 in. Figure 19, below (pg 34), shows a simplified model of what the final design will look like in the oven.

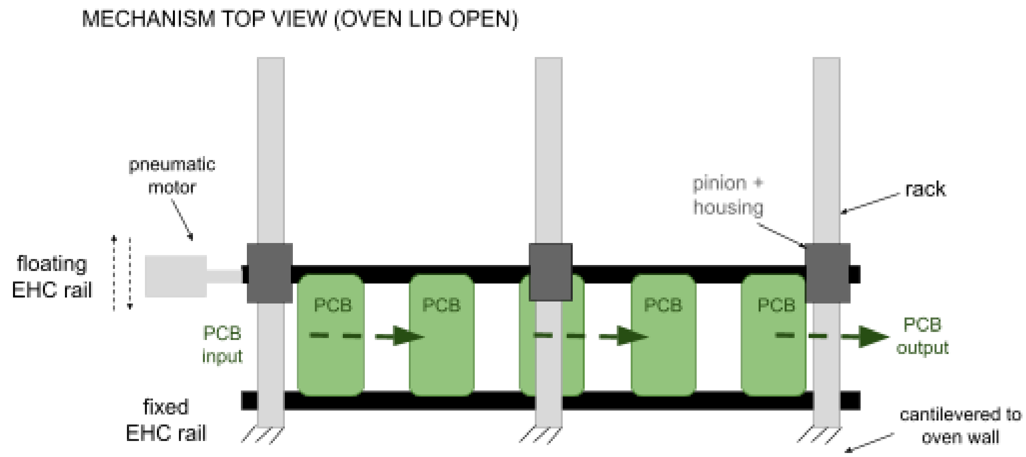


Figure 19: This figure depicts a simplified model of the final design in the reflow oven. This image looks at the oven as if the top of the oven had been removed and you could look straight down into the oven.

VERIFICATION & VALIDATION

High Priority Specification Verification Plans

The verification and validation process was first completed with the primary specifications of cost, durability, adjustability, and integration into the current oven. These specifications were prioritized in the verification and validation process since they are the most critical in fulfilling the sponsor's needs.

Cost

The plan for cost validation will be to make a bill of materials. Costs concurred in the project fall under three categories- off the shelf parts, custom machined parts, and labor costs. For calculating the costs of off the shelf parts, components were chosen from the McMaster Carr website. Primary off-the-shelf parts in the design included the rack, pinion, and rollers. For motors and other electronic components, costs were sourced from Grainger (final design) and Pololu and Amazon (build design). As for custom machined parts, the cost would arise from a keyed shaft spanning the length of the oven and pinion housings for each of the rack modules. In terms of labor cost, the assembly time would have to be estimated based on maintenance worker and sponsor input. The expected labor costs would arise from attaching the racks to the oven wall and threading and securing pinions onto the shaft.

Some limitations of the cost analysis are the unknown costs associated with manufacturing. Heller outsources manufacturing to China so the labor cost there could be different than our US estimates. Additionally, all of our off the shelf part calculations are based on US shipping costs. Heller may have other suppliers for these off-the-shelf parts which would change the total cost. Finally, the pinion housing for each of the modules would require machining. We would need to clarify with Heller how they approach the manufacturing of custom parts and associated parts. We have high confidence in the results of our cost analysis. Since we accounted for all parts and estimated the labor costs based on sponsor input, it is fair to say that our projected cost will be reflective of the total cost. Thus, we can use the projected cost to verify the cost specification.

Verification for the costs of our build design can be found on page 36 in Figure 20. Our build design was able to meet the cost specification with a total cost of $211, which is $89 under the per module cost. There are multiple reasons that helped our build design meet the cost specification. First, the labor cost was negligible since we performed the manufacturing, build, and assembly ourselves. We did not have to rent manufacturing equipment such as mill, lathe, or 3D printer from the university owning them. Building on the reduced labor cost, we had a reduced material cost due to utilizing excess material available in the X50 shop. In our build design, the cost of the motor, electronics, and limit switches were only spread out over two modules instead of the 4-7 modules typically present in the oven.

Our verification for the cost of our final design can be found below in Figure 20 (pg 36), which shows that our final design exceeds the cost requirement.

MATERIALS						
Part #	**Part Description**	**Part Number**	**Quantity**	**Per Unit Cost**	**Total Cost**	**Supplier**
1	Steel Rack	5174T2	4	$36.86	$147.44	McMaster-Carr
2	Steel Pinion	5172T21	4	$68.10	$272.40	McMaster-Carr
3	Keyed Shaft, 30ft	1346K22	1	$506.10	$506.10	McMaster-Carr
4	Air Motor	22UX41	1	$167.63	$167.63	Grainger
5	Support Rollers	6717K21	4	$135.68	$542.72	McMaster-Carr
6	Shaft Bearings	9368T24	8	$1.68	$13.44	McMaster-Carr
7	Housing + Motor Mount Material	8975K264	1	$136.98	$136.98	McMaster-Carr
8	Set Screws 25 Pack	92313A239	1	$3.83	$3.83	McMaster-Carr
9	Fastening Screws	91251A553	1	$15.34	$15.34	McMaster-Carr
10	Motor Driver	N/A	1	$6.99	$6.99	Amazon
11	Arduino Uno	N/A	1	$27.60	$27.60	Amazon
12	Limit Switch	N/A	1	$5.99	$5.99	Amazon
13	Variable Power Supply	N/A	1	$7.89	$7.89	Amazon
			Total Material Cost:		**$1,854.35**	
SHIPPING						
			Supplier			
		McMaster Carr			$23.31	
		Grainger			$12.25	
		Amazon			$0.00	
			Total Shipping Cost:		**$35.56**	
LABOR						
	Process	**Provider**	**Hourly Cost**	**Estimated Time**	**Total Cost**	
	Custom Housing Manufacturing	Xometery	$88.69	2	$177.38	
	Custom Motor Mount Manufacturing	Xometery	$31.65	1	$31.65	
	Welding racks to oven wall	Outsourced	$20.00	2	$40.00	
	Design Assembly	Heller	$20.00	5	$100.00	
	Motor Control Code Writing	Heller	$35.00	2	$70.00	
			Total Labor Cost:		**$419.03**	
			TOTAL COST:		**$2,308.94**	
			COST PER MODULE:		**$577.24**	

Figure 20: Full cost analysis for our proposed final design broken up into parts, shipping, and labor costs. The final per module cost (for a 4 module oven) exceeds the specification of $300 by $277.34. Shipping costs estimated on individual websites with delivery to Ann Arbor. Labor costs estimated using a custom manufacturing site or averages for the occupations found online.

Looking at the final design cost analysis compared to the build design cost analysis there are a couple of key features that have a significant impact on the cost. One important aspect is the new parts required to make the design heat resistant. The PLA used to make the housing was replaced by aluminum and the support rollers were replaced by heat-resistant rollers. The housing material change was necessary as the plastic would melt in the oven and the support roller replacement was necessary to add graphite to allow the rollers to slide under high temperatures. Thus, these were changes necessary to the functionality of the system in an oven setting.

Changes were also made to the materials which increased costs to manage the modularity of the system and account for expansion. Notably, the inexpensive DC gear motor was replaced with an pneumatic air motor and the long shaft was replaced with a keyed shaft spanning the oven. A price estimate for the keyed shaft was calculated based on an estimate for a shorter keyed shaft online extrapolated to the desired length of 30 feet. An air motor was necessary to increase the safety by keeping electrical components away from the heat of the oven, and a keyed shaft was necessary to prevent slippage of the pinions on the shaft and ensure accuracy (which was a limitation of the build design).

Finally, the outsourcing of manufacturing and shipping costs increased the total price of the final design. Shipping costs may not be as high for Heller considering they may have deals with suppliers to obtain parts or manufacture at reduced prices but that makes our cost analysis more conservative than necessary.

Some limitations of our cost analysis include the inability to meet up with our sponsor to discuss the labor and shipping costs due to busy schedules at the end of the semester. We could have also discussed with our sponsor cheaper alternatives to our components. While our system does not meet the cost specification, we believe that it still represents the best system for Heller moving forward as other verification tests show it is reliable and accurate. Given more time, iteration could be done on the track rollers to find another method of translating the EHC that has less friction. The housing design could also be iterated on to reduce the custom manufacturing and material costs. Finally, purchasing items in bulk for future ovens would help reduce the per module cost closer to the $300 specification.

Durability

The plan for durability validation will be theoretical calculations using the endurance limit of steel based on Hertzian contact stress on the gear teeth. To calculate the Hertz stress, a paper was found which provided a formula for the Hertz stress between spur gear teeth with involute profiles. We plan to model the loading as cyclical since the pinion will be traversing over an 18” span meaning the load on each tooth will not be constant. Hertzian stress causes a highly concentrated stress along the line of contact between the gear teeth which causes pitting and gear failure. In our endurance limit calculations, we used the material provided by McMaster Carr to obtain material properties and the torque provided by our torque analysis for the force transferred through gear teeth. We followed AGMA standards for gear strength in the calculation of our gear tooth bending stress and an outside source to calculate Hertzian contact stress.

We performed fatigue life calculations to verify that the pinion teeth would not fail within the specified lifetime of 2190 complete cycles. Assuming that each width adjustment was the entire span of the range(18”), this means that pinion teeth would come into contact with the rack teeth at most 18 times per cycle. Thus, the pinion teeth would need to endure 39,420 total cycles of a compressive stress of 85.5 MPa (the calculated Hertz stress).

One assumption made in the endurance limit calculation is the shape of the gear teeth. We assumed that both rack and pinion had involute shaped gear teeth, when in the actual system the gear teeth are triangular. We made this assumption to use the spur gear model to analyze the Hertzian stress in our system. This assumption is valid because the contact stress could still be generalized as two cylinders rolling on each other because the pinion teeth will contact the rack teeth in line contact. One limitation of durability verification through a theoretical model is loss of energy through friction and noise. In a real system, some of the force being transmitted through the gears would be lost due to friction and sound. Thus, the actual contact stress could be underestimated.

Another plan for durability validation would be through physical testing of a smaller rack module, such as the one in our build design. This would be an effective method of verification as the system is running with the actual components eliminating the need to attempt to model friction and gear tooth efficiency. For verification, we would run the system for 2190 cycles and at the end visually inspect the gear teeth for signs of wear. In the physical verification, we assume that friction would model that of the system in the oven and the motor will be able to replicate actual operating conditions. The limitations of physical testing include time constraints of the course and the lack of an oven to test at temperature. To prioritize safety, someone would have to supervise the entire test. A test of 2190 cycles taking 60 seconds per cycle would result in a total test time of 66 hours, which our team does not have time to complete before the end of the course. Additionally, we do not have access to an oven to test our device. While we are modeling the load of the chain, it will not be representative of the actual load on the adjustment mechanism. The mechanism cannot be tested at oven operating temperatures which would accelerate the rate of wear.

We performed a smaller scale durability test by running our system at the design expo to verify the system. The width adjustment mechanism ran for about 30 minutes total time, equivalent to 60 completed cycles. Examining the model after the expo showed no signs of wear on the gear or pinion teeth and no damage to the electronics. Some flaws we noted during testing of the build design include misalignment of the racks causing the motor to stall and the set screws of the motor housing slipping causing the motor to come uncoupled from the shaft. In the final design, these flaws have been accounted for by fixing one side of the racks to the oven wall, preventing the possibility of misalignment, and coupling the motor to the shaft using a keyed coupling instead of set screws. Another important consideration that arose during physical testing was the noise factor from the motor. The motor produced a high pitched sound during operation of the mechanism with weight. This may be due to low motor quality, however, it is still an important factor to consider as a loud motor could cause hearing issues to reflow oven operators.

Adjustability

In order to verify the adjustability of the newly designed system, a physical test will be performed on the prototype rack and pinion section that will be created. The pinion will be placed 20 inches away from the stationary EHC and a simulated load representative of the EHC, chain, and PCBs will be placed on the housing. The motor, which will be connected to a power supply and regulator, will be operated to move the pinion to its desired position along the rack. The inputted position can then be compared with the actual position by comparing the input voltage to the motor, to the measured distance the pinion traveled (caliper). The design will have met the requirement for adjustability if the values from the motor input match the corresponding displacement values of the pinion. If they do not, the design will be adjusted through a change in the control system to ensure a match of these values.

We were able to verify two of the three adjustability specifications through our build design presented at the design expo. Specifically, the span and speed adjustments were tested. The span specification was tested by mounting the limit switches 2” and 20” from one edge of the model. Then, the code was run on the Arduino to spin the motor, moving the pinions back and forth between the limit switches. The model was able to traverse the entire span with no issues which verified the span specification.

The next specification verified with the build design was the speed specification, which stated that the width adjustment mechanism must be able to traverse the entire span within 60 seconds. Multiple trials were conducted, both with and without the simulated weight of the edge hold conveyor and traversing the oven span in both directions. The maximum time for the trial was 21 seconds, with an average traversal time of 18 seconds. These times are well below the specification of 60 seconds which verifies the speed specification. A video of the model moving across the entire span can be found here.

We were unable to verify the accuracy specification of our design using the build model. To complete this, we would need to order an encoder to mount to the motor and write and tune a control system for the width adjustment mechanism. Due to time constraints we were unable to implement this onto our build system. If we were to implement this we would use a rotary encoder mounted on the back of the motor along with a basic PID control system to achieve the desired positioning. A limit switch would be placed on one end of the model to be used as a zero point and readings from the encoder would be based off of this zero point. Once the PID controller is implemented and tuned properly, the system should be able to meet the accuracy specification because PID systems have relatively low settling times and minimal steady-state error.

We can verify the accuracy specification theoretically using the angular resolution from an encoder. Polulu, the motor supplier, sells a magnetic encoder kit which has a resolution of 20 counts per revolution of the motor shaft. The gearmotor has a gear ratio of 488:1 which means that the output shaft will have a resolution of 9760 counts per revolution which equals an angular resolution of 0.037 degrees/encoder count. The pinion has a pitch diameter of 1” which correlates to 0.07 mm of horizontal distance traveled/degree spun by the pinion. Multiplying these numbers together gives a horizontal distance moved of 0.00259 mm per count of the encoder which is well under the tolerance of the accuracy specification. The equations used to derive this value are shown below in Figure 21.

Angular resolution of output shaft

$$\boxed{\begin{matrix}\text{Encoder}\\ \text{Resolution}\\ \text{(motor shaft)}\end{matrix} \times \begin{matrix}\text{Gearmotor}\\ \text{Gear Ratio}\\ \text{(input to output)}\end{matrix} \times \frac{\text{1 revolution}}{\text{360 degrees}}}$$

Horizontal Distance traveled per degree of pinion rotation

$$\boxed{\text{Pitch Diameter} \times \frac{\text{1 revolution}}{\text{360 degrees}} \times \frac{\text{25.4 mm}}{\text{1 in}}}$$

$$\begin{matrix}\text{Angular}\\ \text{resolution of}\\ \text{output shaft}\end{matrix} \times \begin{matrix}\text{Horizontal Distance}\\ \text{traveled per degree of}\\ \text{pinion rotation}\end{matrix} = \underbrace{\boxed{\begin{matrix}\text{Horizontal Distance}\\ \text{traveled per}\\ \text{encoder count}\end{matrix}}}_{\text{Goal}}$$

Figure 21: Equations used to derive the value of horizontal distance traveled by the pinion and EHC per encoder count, using components from the build design.

The horizontal distance resolution per encoder count is three orders of magnitude below the specification- thus, the specification has been verified. Some limitations of verifying the accuracy empirically include the ability of the motor to get to an exact encoder count, requiring an error margin. However, this is only a minor issue since the fine resolution allows for a much larger error bound on the controller used to position the width adjustment mechanism (for 1mm deviation, error would be 386 encoder counts). If we were to implement the PID controller, we would set the target widths using a lower bound of the calculated value from system dynamics and an upper bound of that value plus 386 counts to meet the -0/+1 mm accuracy specification.

Another limitation of calculating the accuracy specification empirically would be the inability to calculate the effect of backlash. On the build design, backlash was tested by hand by fixing the housing with the motor and attempting to move the other housing. It was determined that the design had a small amount of backlash which was attributed to a missing bearing causing the housing to become slightly misaligned. We did not have enough time to order more bushings to add to the build design to verify that the added bushing would eliminate the backlash.

Integration

To verify integration into the current system, the system will be modeled in CAD and added to an existing model of the reflow oven. Once again, it is not feasible to gain access to an oven within the semester, eliminating the possibility of installing the system ourselves. Thus, a physical model of the system over CAD will suffice to ensure everything will fit. This method of verification is relatively low fidelity because the CAD software is free to use through university resources and does not incur any costs. Through this verification method we assume that all of the parts in the existing oven match the dimensions in the CAD model. Additionally, we assume that all off the shelf parts are the proper lengths with room for tolerances. Some limitations of verifying the edge hold adjustment system through CAD include difficulty determining the feasibility of the mechanism being installed and detecting any collisions as the EHC moves across the oven.

We were able to verify the integration specification by putting rack and pinion models into the provided CAD of the oven and EHC rail. The specification was mostly verified, as the pinion housings and shaft did not interfere with any part of the oven. However, the racks extended past the length of the oven walls. Heller would need to order custom length racks to ensure they would fit within the oven walls. Results of the integration testing in the form of CAD screenshots are shown below in Figures 22a-c.

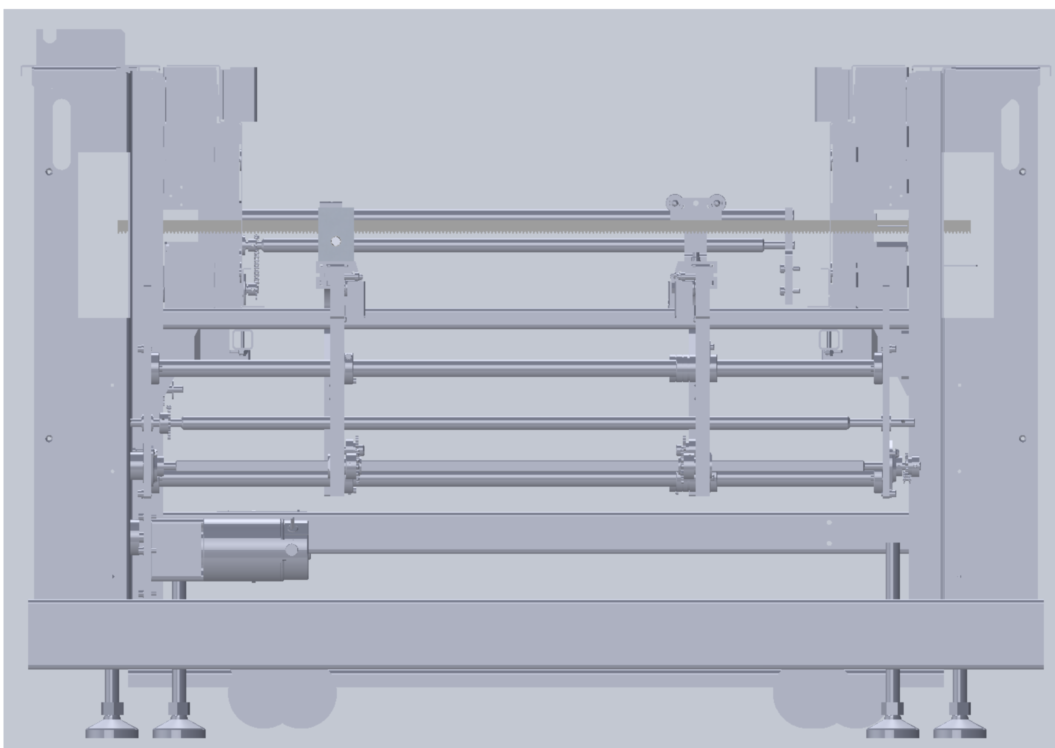


Figure 22a: Front view of rack and pinion modules in oven, with racks in tan. This view shows how the bottom of the pinion housing is attached to the EHC rail.

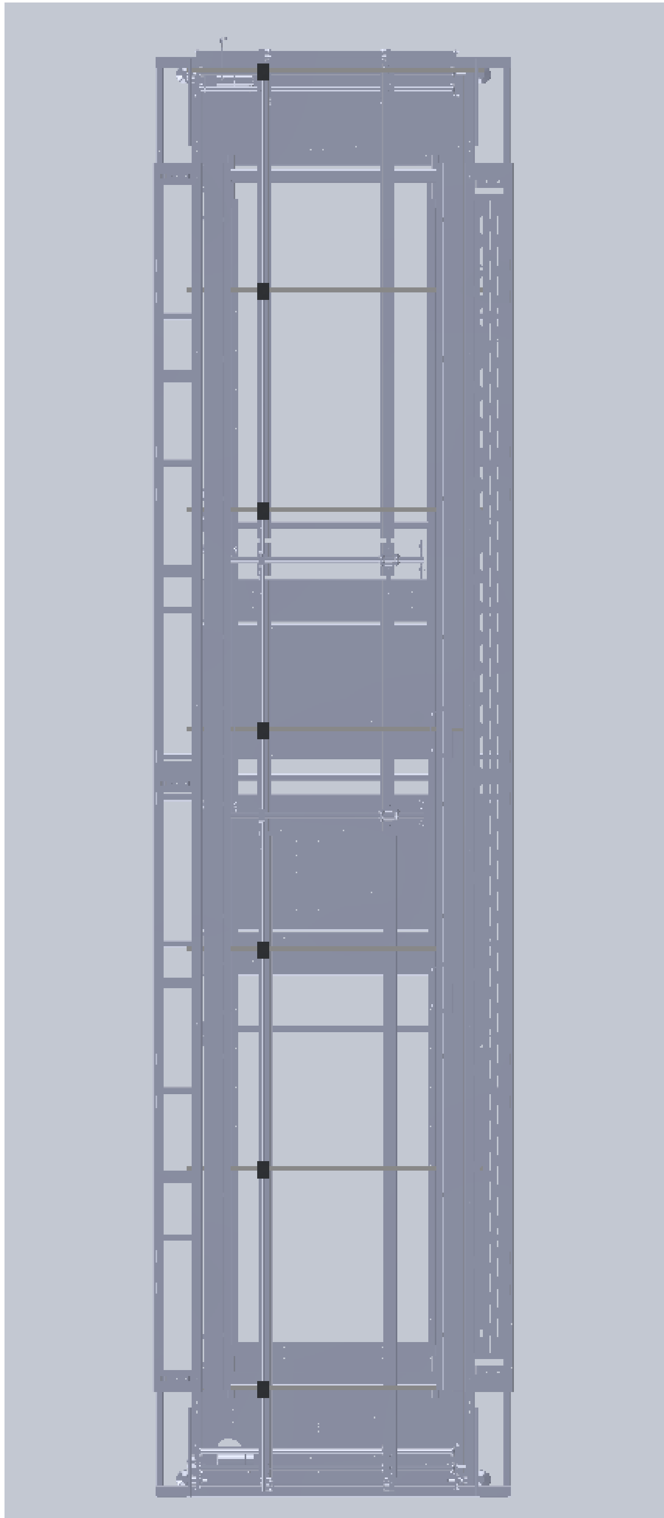


Figure 22b: Top view of rack and pinion modules in oven. Black rectangles represent locations of pinion housings. Shaft runs between pinion housings.

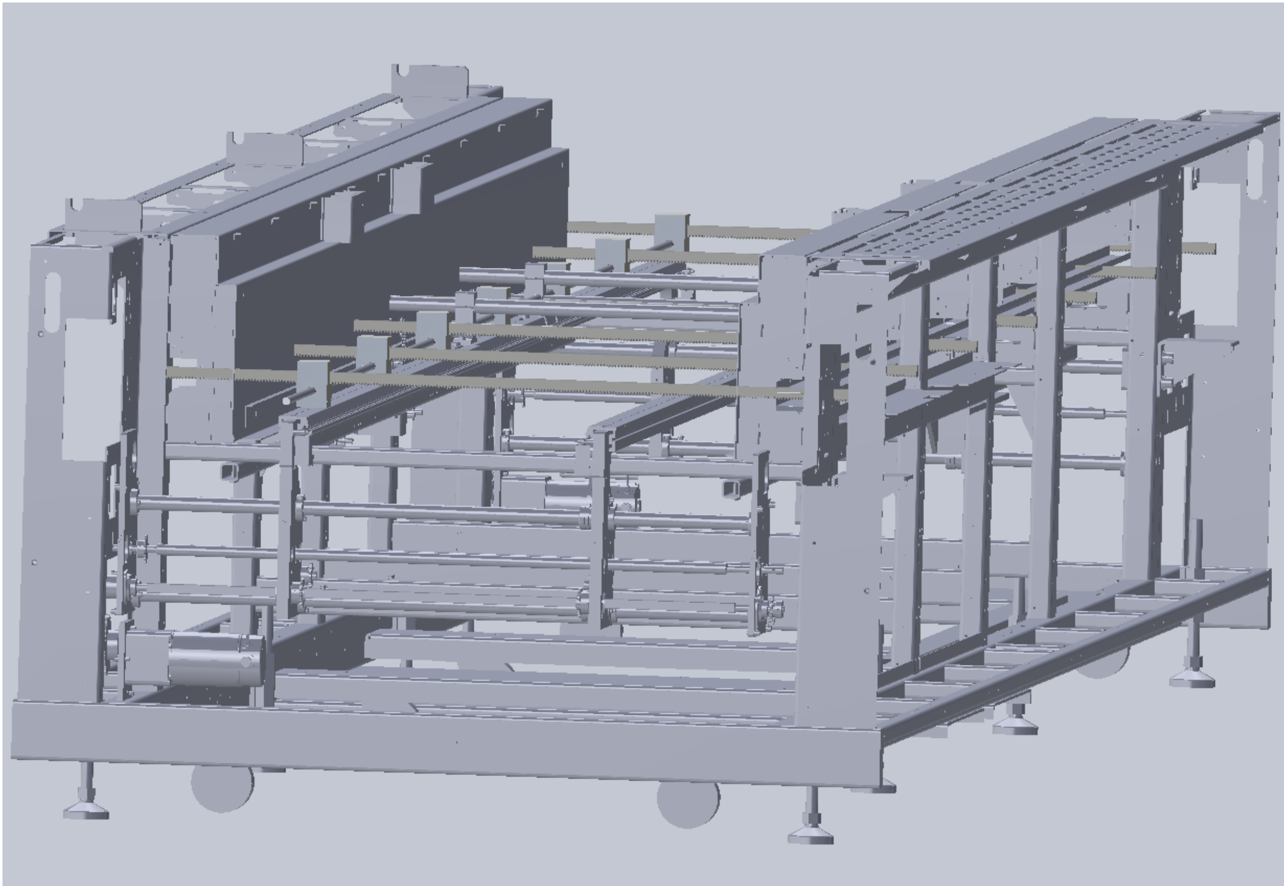


Figure 22c: First isometric view of oven with rack modules implemented, from front right viewing position. Lead screw modules would not occur in the final design, along with the lead screw adjustment seen at the front of the oven.

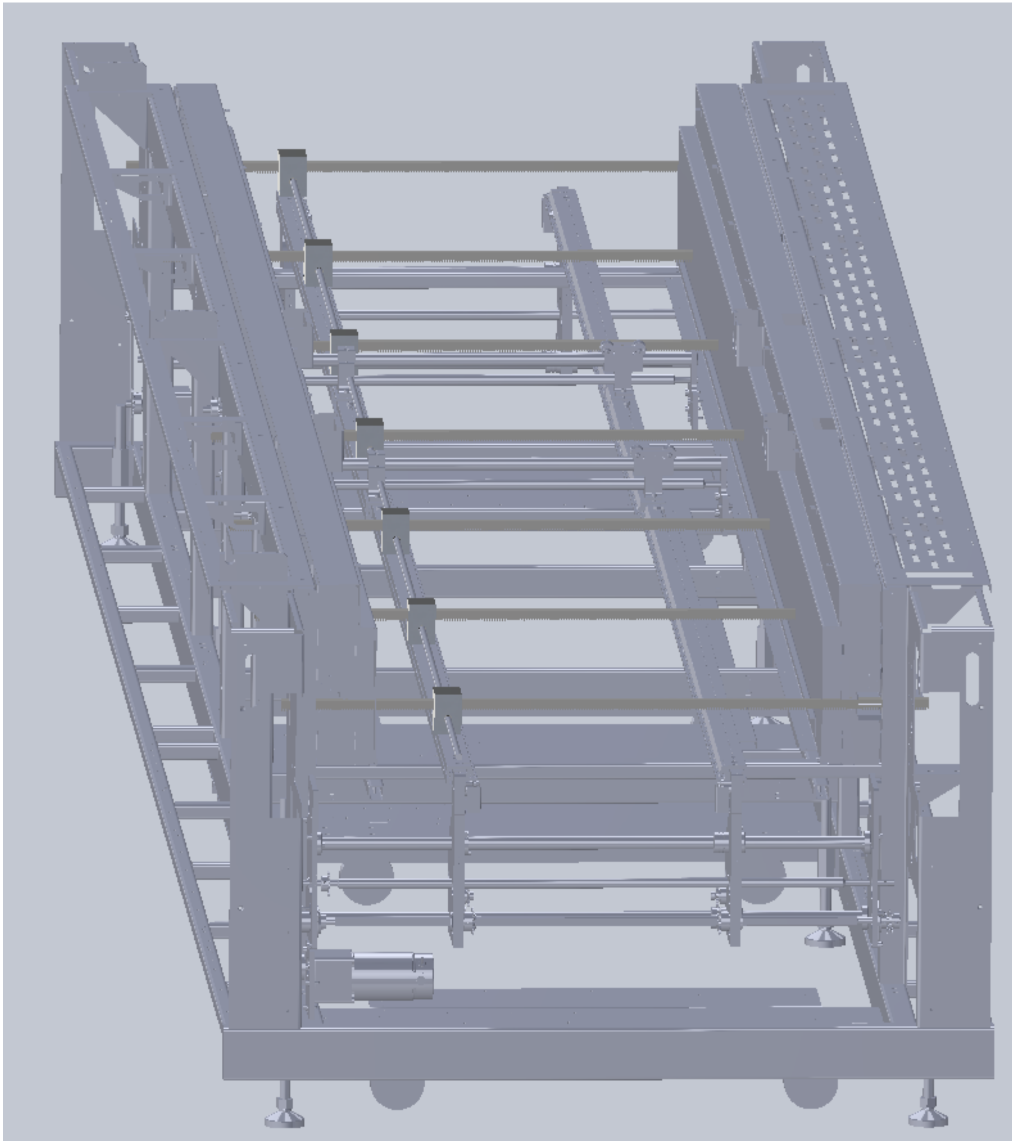


Figure 22d: Isometric views of rack and pinion modules in oven from front left viewing position. 7 modules used, representing an oven model with a vacuum chamber. Total oven length approximately 250".

Implementation of the rack and pinion CAD into the oven CAD verifies most of the specifications of the oven integration requirements because the pinion housings and shaft have no collision issues. However, different sized racks would need to be purchased because the oven span does not match a rack length provided by McMaster Carr.

Low Priority Specification Verification Plans

Following the development of verification plans for the highest priority specifications, verification plans were built for the lower priority specifications. In the scope of the class we are attempting to verify only the high priority specifications due to time constraints.

Conditional Resilience

To verify conditional resilience, a theoretical heat transfer analysis will be performed on the racks, pinion, and shaft. Based on the timeline of the class, there is not enough time to build a computational model of the system which factors in temperatures in the different heating zones and the corresponding effects. To simplify the analysis, we will assume that each zone has a uniform temperature distribution (i.e. constant throughout the zone) and there is minimal heat transfer by convection through the shaft. Assuming each zone has a uniform temperature distribution will simplify rack deflection calculations by eliminating the need to integrate over a temperature gradient in the calculation of convective heat transfer. We will also use data sheets to estimate the material properties of thermal coefficient of expansion and modulus of elasticity at oven operating temperatures. This simplifies the verification process by saving time performing tests to obtain these values. The limitations of our conditional resilience verification plan are we are using fairly simple equations in an attempt to describe behavior of a complex system. Another limitation we face is describing the boundary conditions of each of the zones, as during oven operation the temperature of each zone could be changed which would change the boundary conditions.

We were unable to perform an in depth thermal analysis within the time constraints of the semester. However, we were able to outline a few key points we thought were important. First, most parts would expand when placed into an oven. This would affect the rigidity specification of the racks as a compressive stress would be introduced into the racks. The net compressive stress would reduce the chance of the rack yielding since most materials fail in tension quicker than compression. However, the deflection would be increased due to the addition of the compressive axial and bending stresses. Next, the housing would need to be redesigned for the increased temperatures. Most parts in the housing would expand under the operating temperatures of the oven. If pushed against each other, the normal force of each contact point would increase therefore increasing the frictional force. An increase in friction would require the motor to output more torque and likewise would move slower which would affect the mechanism speed specification. We would also expect the shaft to sag more and twist which could affect the accuracy specification. Materials at higher temperatures have lower elastic moduli meaning they deform easier under similar weights. Thus, deflection of the shaft spanning a 5 ft gap could pose a problem. We have anticipated this problem through a conservative choice of shaft width- however, the rigidity specification still needs to be verified to complete a full engineering analysis.

Length Variability

To verify length variability and modularity of the system, we plan on testing using the build design presented at the design expo. One end of the shaft will be held in place by a shaft, and the other end turned by a torque wrench to a specified torque. Once the torque wrench clicks the angle displacement can be measured with a protractor or image analysis which represents the twist of one module at operating torque. The result will represent one module, and can be extrapolated to fit a provided number of modules in an oven. In this verification plan, we assume that all modules will have the same twist, causing the total angular displacement to compound with increasing oven length. We also assume a 60" span between the modules, as that is the largest span provided by the sponsor. Minimizing angular displacement along the length of the oven is important in ensuring positional accuracy. One limitation of verification testing is the ability to test more than two racks at once. The budget limits the number of modules that can be assembled, and thus limits possible testing on the build design. Increasing the number of racks would increase the torque required to move the EHC which would have a negative effect on the total twist.

We were unable to verify this specification using our build design at the expo due to not obtaining a torque wrench. It would have been difficult to apply the torque wrench on the end of the coupling since the shaft was lubricated and there was no place to insert the end of the torque wrench. If we were to create a spot to insert the torque wrench it would have been difficult to machine with the resources we had access to due to the shaft being 5 feet long. We also did not move the positioning of the second support, and instead chose to keep it at the maximum distance. Observing the simulated EHC rail move back and forth and monitoring both ends upon a direction change there was no visible lag of the far housing which suggests that the torsional stiffness did not have an impact on the accuracy of the mechanism, verifying the modularity of the system.

Rigidity

To verify rigidity we plan on testing using a computational model with the rack geometry provided from SolidWorks. This model will be more rigorous than our previous beam bending model since we are not under the assumption that the rack has a rectangular profile the entire way through. The forces placed on the model will be the same as those in the beam bending model, and one end will have a fixed support while the other will have a simply supported joint to mirror the implementation in the beam. To perform the analysis, we will assume that the modulus of elasticity is the same across the entire rack and that the housing does not cause the rack to deflect. We can test the rigidity at both room and operating temperatures. One limitation of this method is that the weight might not be applied to the rack as point loads, rather two small distributed loads. This would affect the results by increasing the length of deformation.

We can also perform rigidity testing on the long shaft which spans the length of the oven. Once again, this would have to be performed computationally since we do not have access to an oven. The forces acting on the shaft would be frictional forces from the bushings and possibly some lateral forces if things are misaligned. We plan on running an analysis in COMSOL with forces along the entire shaft to ensure the rigidity specification is met.

Maintenance
To verify the maintenance time, we will make an assembly plan and estimate the time to complete each task. We will estimate the times from guidance from our sponsor since they know the typical times for assembly. Important times we would need are the time to thread the pinion onto the shaft, time to screw racks onto the oven wall, and time to assemble pinion housings. The limitations of this method are there is no way to measure the actual installation time without an oven model and all of the parts of our final design. Our estimates may be underestimated due to the bulkiness of a 30" long shaft. Additionally, time estimates would scale with the number of racks because more housings will be required. Therefore, in our verification of the maintenance specification we will estimate maintenance time using eight racks, the largest number currently in an oven.

We were able to attempt to verify the maintenance specification through the construction of our build model. To make the two modules, the rack supports took 2 hours, the waterjet parts took 15 minutes, the machined parts took 1 hour, and assembly took 4 people 45 minutes. This results in a total of 6.25 hours to assemble the model. Extrapolating this value out to 8 modules instead of 2 would result in 25 hours which is just over the 3 day specification. The parts used include three allen keys, a flathead screwdriver, and a Philips screwdriver. As for the electronic components, those required a wire stripper, wire cutter, and soldering iron. This results in a total of eight components needed to assemble which is less than the 10 basic tools provided in the specification. Our design did not meet the specification because at 8 modules it would be a process to string pinions onto the 30 ft shaft which would increase the maintenance time. Additionally, to replace housings, one would have to remove the entire shaft from the oven and all pinions from the shaft. This would be a time consuming process, but one that is necessary for the simple transmission of the design.

Safety
Safety is an important part of all engineering projects, thus, it is important we have a way of verifying that our design is safe. First, we will ensure that no parts of our mechanism break or protrude from the oven wall in our CAD. With no parts breaking the oven wall it eliminates the chance for operators to get burned by parts which would heat up during oven operation. Next, we will present our mechanism to both our sponsors, their clients, and oven operators along with a questionnaire to measure how safe the design is. Our questionnaire would include some Likert scale questions to obtain a quantitative rating for our design safety. Some limitations of these

methods are that the CAD does not highlight pinch points, and a person unfamiliar with CAD may have trouble visualizing the model solely from our mockup.

The design expo served to verify the safety aspect of our design. We did not have an oven to place our design into and protect viewers from moving parts. Our model ran at slow enough speeds to where no spectators were hurt even if they happened to touch it while it was moving from side to side. Based on the build design, we identified potential pinch points where the racks were attached to the supports and where the housing slid along the top of the racks. Another design iteration could be made to reduce the chance of injuries in these parts in the actual oven. One safety issue that arose with the build design would be the possibility of effects on hearing from a change in the motor. A volume specification was not created due to the oven already being in a manufacturing setting, but safety of oven operators is an important human factor of our design. Thus, the next iteration of the width adjustment mechanism design should include a maximum volume specification to protect the hearing of the oven operator.

System Validation Plan

To validate the new edge hold width adjustment system, user testing must be performed with the mechanism installed in an actual oven. This will allow the system to function at temperature with the proper loads of the edge hold conveyor, chains, and PCB boards. In addition, installation into the oven would fulfill the specification of fitting within the existing heating functionality. Thus, the only way to validate our system would be through user testing during oven operation in the PCB job shops. Validation is outside the scope of what can be accomplished in ME 450 due to space and time limitations of getting access to a Heller reflow oven. Our sponsor can pursue design validation by assembling an oven with the new edge hold adjustment mechanism and performing testing in house before putting it on the market. Our sponsor would have the time to complete the durability test, financial records to verify the cost specification, and dummy boards to ensure adjustability running all at the same time for a complete system validation.

DISCUSSION

This project relates to the adjustment mechanism for the edge hold conveyors within the Heller Industries' reflow oven. Heller's ovens are used to complete the final stage (soldering of components) of mass manufacturing printed circuit board (PCB). The current design utilizes a chain drive and idler gears to actuate lead screws, which in turn move the edge hold rails to the desired position. After cyclic or excessive loading, the chain links experience stretching and/or skipping and the sprockets undergo deflection. This problem has been fully defined and no additional data or information is needed to better define the problem moving forward. If there was additional time and resources to do so, however, the best allotment would be to spend more time speaking with the sponsor about the intricacies of the issue sooner in the process to get a head start on the concept generation process, since the sponsor has lots of insight on the issue. A direct and frequent line of communication with Heller Industries could be used to do so.

The build design showcased at the expo did a great job and acted exactly as intended. The success of the build design will ultimately translate to the success of the final design, since the build design was constructed with worse tolerances and tested under worse conditions than the final design would experience. There are a few things that can be improved with the build design to yield a more successful final design. The first alteration that should be made is the motor mount. While the motor mount worked as intended in the design, and served its purpose for the purposes of the expo, the motor mount had too loose of tolerances which allowed for the motor to twist around slightly under its own torque. If this occurred in the final design, the encoder would have a hard time getting an accurate reading on the location of the motor which could result in an EHC position that is outside the allowable tolerance. The motor bracket used for the build design was made from sheet metal, which would likely be replaced by an aluminum bracket in the final design. An aluminum bracket would provide more rigidity and support for the motor. Another alteration that should be made before manufacturing of the final design is a change in the shaft coupling. During the expo, one of the set screws in charge of coupling the motor shaft with the pinion shaft came loose. The intention for the final design was always to use a keyed shaft in place of set screws, but the brief failure at the expo further solidified this.

The design process used for this project was sound, and resulted in a strong prototype that fulfilled its requirements and specification. One change that would be made to the design process, if granted the time and resources to do it over again, would be to spend less time with the tiny details of analysis and spend more time thinking through how specific sub-problems with the overall design could be solved. For example, the exact solution for shaft coupling and motor mounting for the expo build design were not finalized until two weeks before the expo. These problems were left until last for the sake of completing analysis on time, which could have been avoided if general analysis was completed for some of the less crucial components of the design. Ultimately, a change in the design process to attempt a more successful result would mean a reallocation of time rather than a major change to the structure of the process itself. By implementing this change in the design process, the build design would have been slightly more polished and the final design would have been conceptualized sooner in the process.

The primary challenge encountered in the design process was time. There was simply not enough time in the design process to fully explore some of the ideas, solutions, analysis, validation, and verification that could have been completed otherwise. The lack of time was recognised somewhat early in the design processes which allowed for preparations to be made such that a proof of concept could still be executed. Despite this, not everything that would have been helpful for the success of the build design was completed in time, but the negative effects on the build design were minimized through the efforts of the team. Another risk encountered throughout the build process was an oversight in ordering parts in which half of the bushings and set screws were neglected from the part order. Once this oversight was realized, the team jumped into action to find a solution. Some set screws that fit our needs were borrowed from the machine shop and some fast analysis was completed in order to determine if the build design could function as intended with one bushing per housing. It was ultimately decided that the build design would have increased friction from the missing bushings but would still function as intended.

Despite all of the efforts of the team, there remain a few risks associated with the success of the final design for use by Heller Industries. First and foremost, thermal analysis was not completed to a level that the team would have liked. As mentioned previously, time was a limiting factor in the design process and the team determined that a working proof of concept would be a better allocation of time than an in-depth, formal thermal analysis. Because of this, the thermal aspect of the project is something that has not been fully explored and poses a risk to the success of the final design down the line. Despite this, the design shows merit and its success is supported by a rigorous design and validation process that provides a great starting point for work on the project in the future.

REFLECTION

When taking a broader look at the implications of our design there are other factors that need to be considered besides the improvement to the Heller Industries' product. First is the impact on public health, safety, and welfare. Due to the implementation of the design into an oven that will be used in a factory with little to no human interaction, there should be no negative implications of the design on any of these three factors. Second is the global context. This design will be of significant benefit in a global marketplace. Since the design will lead to improvement in the current design of the reflow oven, it will lead to an increase in productivity in the PCB manufacturing line, and will therefore result in positive economic impact on Heller Industries and all of their customers. Third is the social impacts associated with the manufacture, use, and/or disposal of the design. Once again, there should be no impact from these factors due to the fact that all materials used in the design are commercial off the shelf parts or are made of aluminum, which is commonly used for manufacturing throughout the world. Finally is the economic impacts associated with the manufacture, use, and/or disposal of the design. Similarly to the global context, there will be an economic affect to the production of the design. As the design is implemented into the reflow ovens, there will be more purchasing of the parts needed to manufacture the design. This impact will be minimal however, as the scale of Heller Industries' production would not lead to a large-scale economic impact. In order to assess the potential societal impacts of the design, a stakeholder map was created in order to determine who would be affected by the creation and implementation of the design and to what extent. The stakeholder map can be seen below in Figure 22.

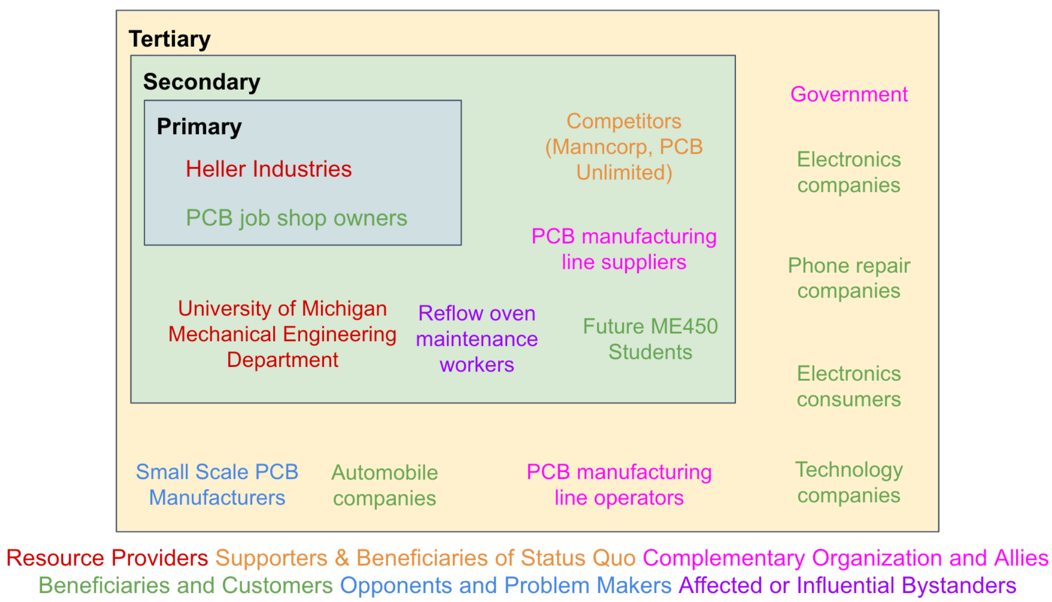


Figure 22: Stakeholder map for Heller Industries oven chain drive system. Primary and secondary stakeholders are limited, with tertiary stakeholders including all users of PCBs. Stakeholder types can be referenced using the provided key.

In addition to the impacts the design had outside of the ME 450 course and its direct implications on Heller Industries, it is important to identify how bias affected the design. This includes cultural, privilege, identity, and stylistic similarities and differences between team members. Due to the scope of the project, and its clear cut problem statement, there was little to no effect of cultural, privilege, or identity similarities and differences between team members. The approach to the problem and its eventual solution were able to be looked at through a lens of pure engineering as the project had minimal impact outside of the course and on the sponsor. However, throughout the project there were most definitely stylistic similarities and differences between team members. These affected the group in a positive way by allowing for brainstorming from different perspectives, discussions of various aspects of the project, and a unique approach to problem solving. When comparing these same factors with the relationship with the sponsor, there was a similar result. The experience that the sponsors had from their time working at Heller Industries was invaluable to the progression of the design process. The sponsors were able to use their experience to help guide the group towards a realized product.

Another important aspect to reflect upon was inclusion and equity. Throughout the project there was a power dynamic between the project team and the sponsor. Due to the nature of the ME 450 course, the team's project was centered around delivering a product to meet the sponsor’s needs. This led to input from the sponsor at various stages through the project. These design inputs from the sponsor were seen as extremely valuable to the group. Since the sponsor had significantly more experience than the entirety of the team combined, they were able to use their power dynamic to help prevent the group from repeating mistakes that they had made in the past. Within the team there was no power dynamic. All group members were treated equally and all tasks were distributed evenly throughout the duration of the project. One group member was in charge of communication with the sponsor, leading them to direct questions, lead conversations, and establish deliverables. While this group member had some extra duties, it did not lead to a power dynamic within the group. Since the end user of the project was Heller Industries’ this led to the group considering all recommendations and advice from the sponsor strongly. Throughout the semester, there were no discussions of group members and the sponsors' cultural identities, and therefore it can be assumed that it had no impact on the project. As stated earlier, each group member was treated equally and allowed to contribute as much as they deemed necessary without any limitations from any other group member or from the sponsor.

The last consideration was ethics and any possible ethical concerns that needed to be addressed during the project. Throughout this project, there were no ethical concerns that needed to be addressed. Due to the niche role that the design will be playing, there are very little impacts on anyone besides the sponsor. No ethical issues would arise if this product were to enter the marketplace. Personal ethics can be similar and differ to the professional ethics expected from the University of Michigan and future employers, but it is important to assess and understand these differences and make informed decisions on ethical issues.

RECOMMENDATIONS

Based on the analysis presented throughout this report as well as the generated proof of concept our recommendation would be for Heller Industries to convert from a chain drive adjustment system to a rack and pinion design for their edge-hold reflow ovens. While our proof of concept is a fairly accurate representation of the final design there are some changes we would recommend for final implementation. The first and most important is to select thermally resistant components. As the proof of concept was tested at room temperature we did not need to consider any thermal aspects and simply selected components based on their ability to withstand the forces at room temperature. Unfortunately we did not have time to complete a thermal analysis so Heller Industries will need to work on that in order to determine the best components. Additionally many of our parts were 3D printed. This was done for easier and cheaper manufacturing as we were on a limited budget and timeline. We would recommend machining all parts to better withstand load cycles as well as provide greater tolerancing accuracy. Finally, in our proof of concept we used a simple servo motor to rotate our drive shaft and turn the pinions. For full scale use we would recommend that Heller Industries look into pneumatic motors. These motors will allow for much greater torque generation while maintaining a small profile. This will be crucial as the oven increases in length and the load on the motor increases.

CONCLUSION

This semester Heller Industries tasked our ME 450 group with the redesign of their edge hold cover system. Their current method involved using a chain drive and lead screw to move the edge hold rail in order to accommodate different sized printed circuit boards. This design worked well for them with smaller ovens however as the ovens increased in length the load on the chain and sprockets increased. This led to the chain stretching and the sprockets deflecting, requiring long periods of downtime for maintenance and repair. Keeping this in mind, we set out to create a new design that would not run into such problems and would be able to handle future expansion of the ovens. After working through rigorous concept generation and down selection processes we came to the conclusion that a rack and pinion design would be the best alternative to a chain drive. In order to validate this conclusion we completed various engineering analyses which allowed us to create a working proof of concept. The proof of concept and analysis worked to confirm our conclusion that the rack and pinion would be a suitable alternative to the current system. While the proof of concept was not placed under any sort of thermal load and that thermal analysis was not completed we feel that with the right components this design will hold up within the oven and remain a feasible replacement.

ACKNOWLEDGMENTS
This project could not have achieved the level of success it did without help from a few main contributors. The primary contributor of information, assistance, and guidance for this project was Jim Neville, who was our contact at Heller Industries. Jim provided unparalleled insight into the issue and his myriad of experience in the field to guide us towards a feasible and useful solution. David Heller was also invaluable to the success of this project. David provided not only financial support as the president of Heller Industries, but provided important insight into some past issues with the EHC and issues that our team needed to be cautious of moving forward with the final design solution. Erica Liu provided additional unique insight into the project and the problem in need of solving. Erica spent some time working for Heller Industries and provided much assistance with the background knowledge of the problem. She was also available to answer many of the day-to-day questions about the project and acted as a local source of support. Professor Jonathan Luntz was another huge contributor to the success of the project. Professor Luntz provided structure and guidance throughout the semester and constantly helped to ensure that our project was on track. He was also sure to notify us of any oversight in our plan that might create hindrances down the line, which contributed to the ultimate ease of the solution development process. The last contributor to the success of the semester was Mr. Kent, who supplied time and expertise that were pivotal to the functionality of the design solution. The support of Mr. Kent assisted in propelling us towards success with our proof of concept and was greatly appreciated. Our team is extremely thankful for all of the individuals who demonstrated support throughout the semester and we owe them much gratitude.

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APPENDIX A - BILL OF MATERIALS

Part Number	Part Description	Catalog Number	Quantity	Total Cost of Part(s)	Supplier
1	0.75” x 0.75” Steel Rack	5174T2	2	$73.72	McMaster-Carr
2	1” PD, 0.75” Pinion	5172T21	2	$64.82	McMaster-Carr
3	0.5” x 5’ Steel Shaft	1346K22	1	$38.24	McMaster-Carr
4	Support Rollers	3668K23	4	$74.28	McMaster-Carr
5	Shaft Bearings	60695K22	4	$14.60	McMaster-Carr
6	Pinion Housing A	N/A	2	N/A	3D-Printed
7	Pinion Housing B	N/A	2	N/A	3D-Printed
8	0.75” x 6” Steel Round Stock	N/A	1	N/A	ME Workshop
9	2.75” ¼”-20 Bolt Pack	91251A553	1	$15.34	McMaster-Carr
10	¼”-20 Hex Nut Pack	91845A029	1	$6.09	McMaster-Carr
11	2” x 4” x 4’	271736	4	$11.68	Home Depot
12	2” Wood Screw Box	PTN2S1	1	$8.97	Home Depot
13	10-24 0.25” Set Screw Pack	92313A238	1	$8.97	McMaster-Carr
14	12” x 12” x 1/16” Aluminum Sheet Metal	N/A	2	N/A	ME Workshop
15	DC Motor	3485	1	$28.95	Pololu
16	M2.5 Screw Pack	91290A103	1	$10.43	McMaster-Carr
17	Motor Driver	MK-050	1	$6.99	Qunqi
18	Arduino Uno	A000066	1	$24.84	Amazon
19	Limit Switch	KW12-3	2	$11.98	Amazon
20	12V Power Supply	ISP-NW-PS-12V	1	$7.89	Amazon
21	5’ 22 Gauge Electrical Wire	N/A	1	N/A	ME Workshop

APPENDIX B - MANUFACTURING PLAN

Most of the parts used in the build design did not need to be modified and could be used as purchased/received. However, there were a few items that required manufacturing or modification in order to be used in the build design. A list of the parts that needed to be manufactured/machined from the purchased/received Bill-of-Materials parts is given below in Table B-1.

Table B-1: Parts that were manufactured and assembled for the build design

Build Design Part Number	Build Design Part Description	BOM Part Number(s) Used	Quantity
1	Pinion	2	2
2	Central Shaft	3	1
3	Housing A	5, 6	2
4	Housing B	4, 5, 7	2
5	Motor Coupling	8	1
6	Rack Support	10, 11	2
7	Weight Holder	13	4
8	Motor Mount	13	1

Build Design Manufacturing Plan

Once the above parts were manufactured, the build design could then be assembled. Details for the manufacture of the above parts are given below. The first step was to fit the racks into the rack supports as seen below in Figure B-1.

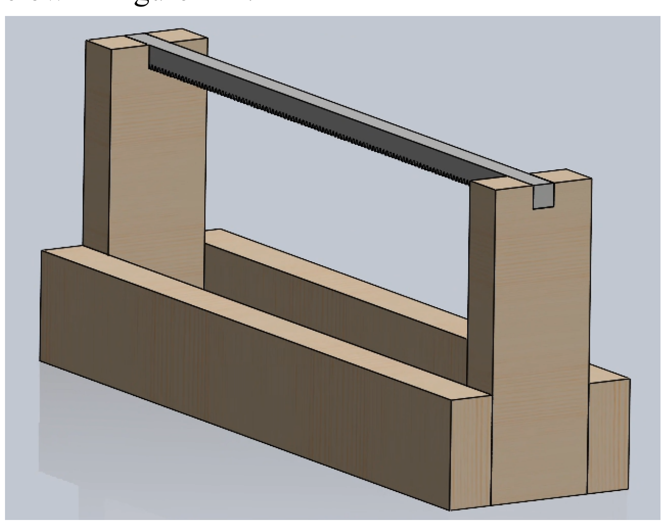


Figure B-1: Rack Assembly

The next step was to attach the first pinion to the shaft at the location further from the motor side of the shaft with the hub of the pinion facing the motor side of the shaft. Then the first Housing B was slid onto the shaft such that the interior face of Housing B is facing the hub face of the pinion. Then the first Housing A should be slid onto the shaft such that the exterior face of Housing A faces the exterior of the first Housing B. Then the second pinion should be attached to the shaft at the location closer to the motor side of the shaft. A picture showing the shaft assembly thus far is provided below in Figure B-2.

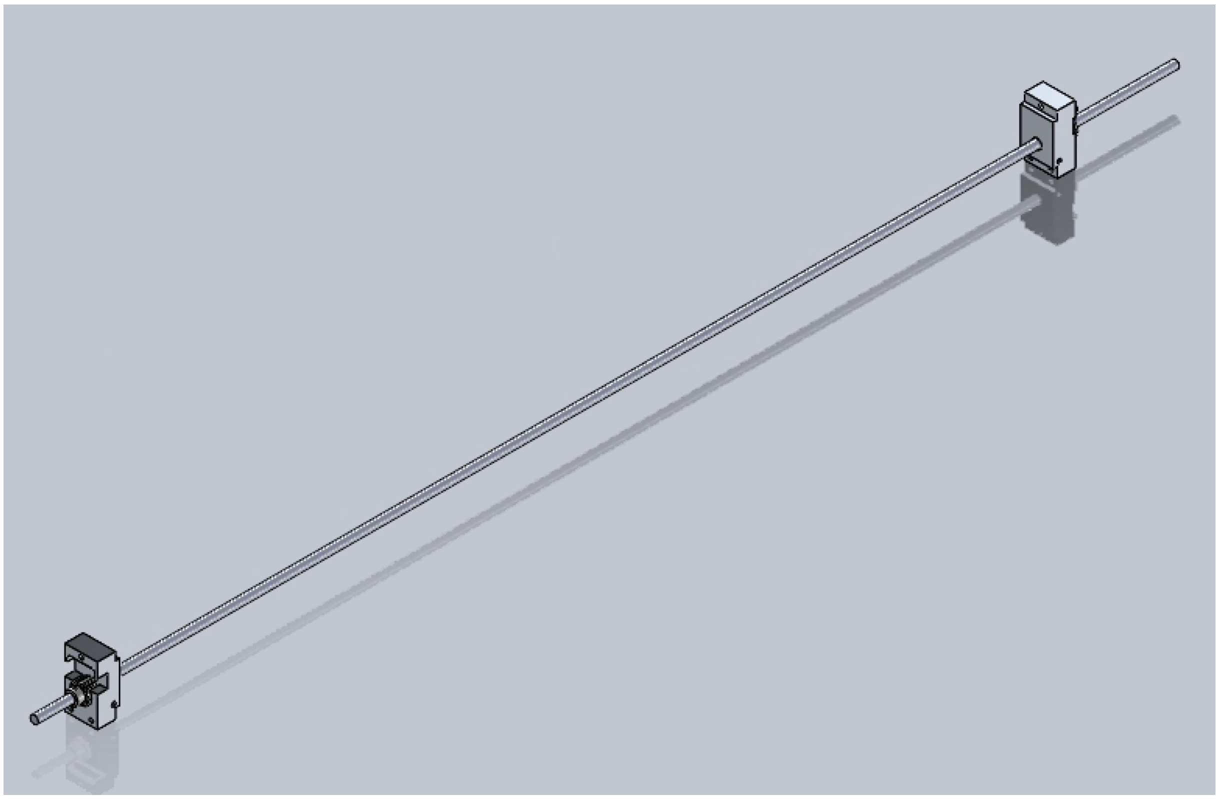


Figure B-2: Assembly with pinions attached and housing components on shaft

The next step was to mesh the pinions to the racks. In order to do this, one person held the shaft such that the first pinion was meshed to one of the racks in the rack assembly while another person slid the second Housing A onto the far side of the shaft so that the pinion further from the motor side of the shaft was completely encased in both halves of the housing. Then the screws were inserted into the three holes in the two meshed housings and nuts were screwed onto the screws so that the two halves of the housing were fastened together. Figure B-3 below shows the first pinion with the housing enclosed around it.

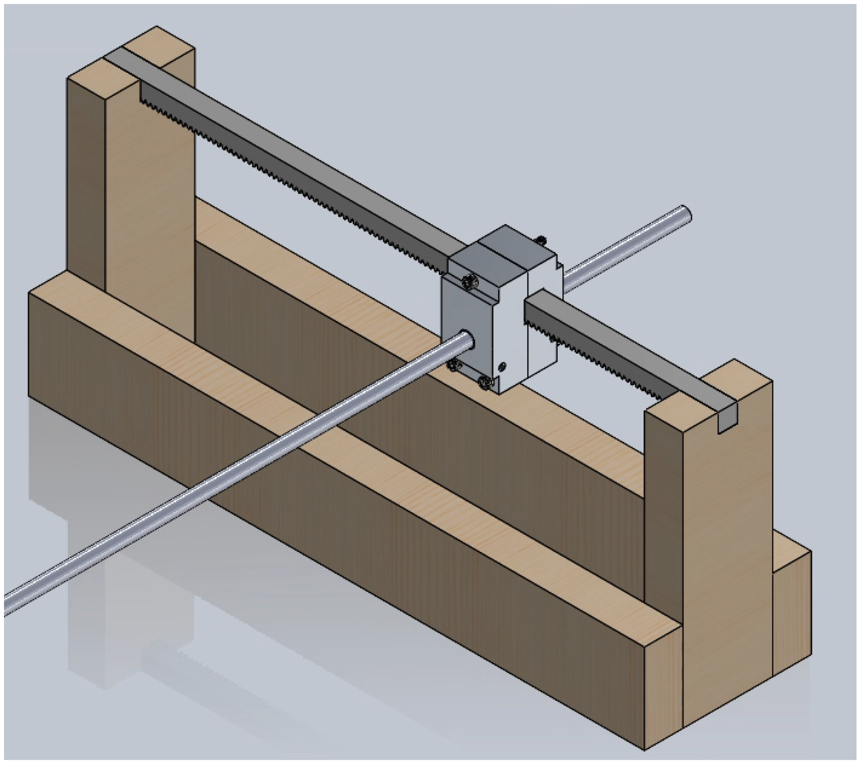


Figure B-3: Assembled rack and pinion with housing

This process was repeated for the other pinion except Housing B was slid onto the shaft instead of Housing A as was done in the previous step. Care was taken so that the pinions were the same distance from along the racks when meshed. This was important because it ensured that when one pinion was 5” from one side of the rack, the other pinion was also 5” from the same side of the rack. The completed rack and pinion assembly is shown below in Figure B-4.

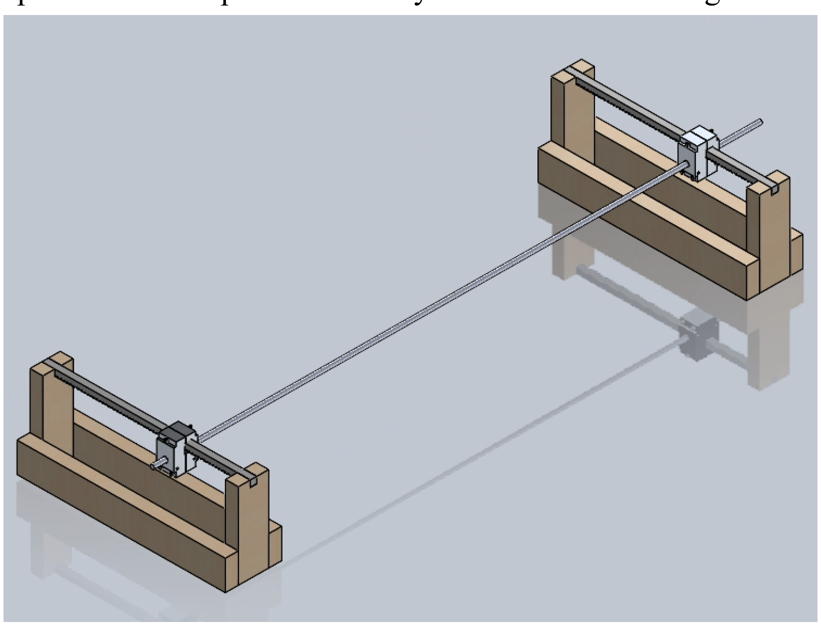


Figure B-4: Completed rack and pinion assembly

The next step was to attach the coupling to the central shaft. This was done by simply placing the coupling over the motor shaft as far as it would go and then set-screwing it to the shaft. The following step was to attach the weight holders. For the housing further from the motor, this consisted of fastening two weight holders to the housings (one weightholder on each side with the tabs overlapping underneath the housing). For the housing closer to the motor, a similar approach is used, but before the left side (when looking down the shaft from the motor side of the shaft) holder is fastened, the motor mount must be placed on the housings first. The layering from left to right was as follows: weight holder, motor mount, housings, weight holder. The assembly of weight holders and the motor mount is shown below in Figure B-5.

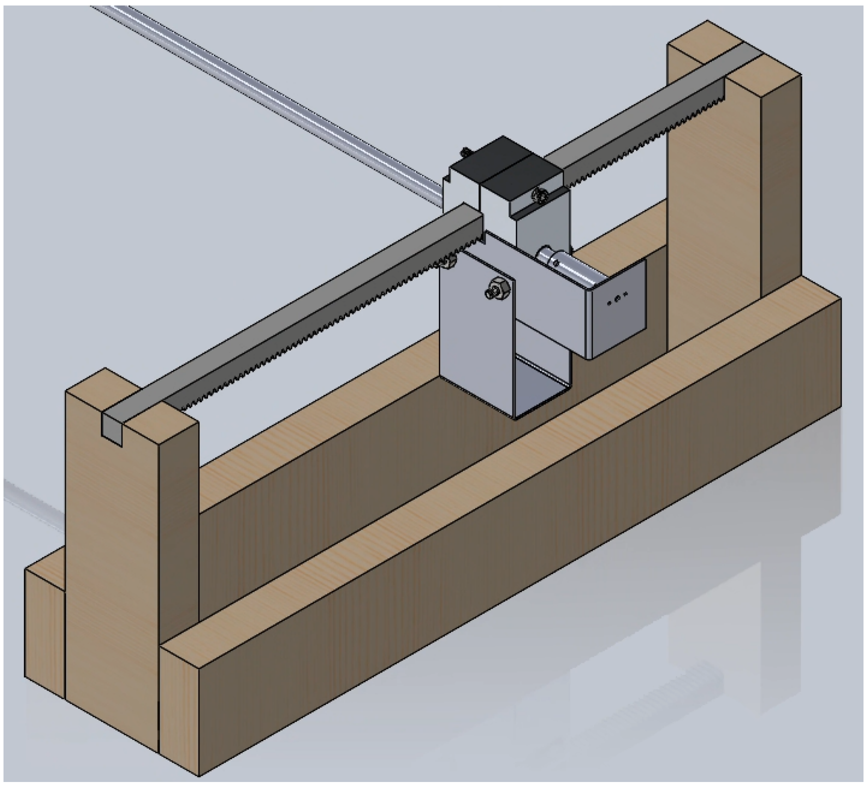


Figure B-5: Assembly of all parts except the motor and motor components

Once the motor mount and weight holders were attached, the motor was attached to the coupling by using a set screw. Then the motor was attached to the motor mount using the M2.5 screws. After that, the motor, arduino, limit switches, and motor driver were wired up. The limit switches were attached on top of the rack near the motor using tape. The limit switches were attached 18” from each other in order to ensure that the design could change the conveyor width from 2” to 20” wide. Below are the manufacturing plans for all of the parts used in the assembly of the build design.

Pinion Manufacturing Plan

The pinions needed to have a 10-24 threaded hole added on the hub to enable the pinion to be fixed to the central shaft. A drawing of the part is shown below in Figure B-6.

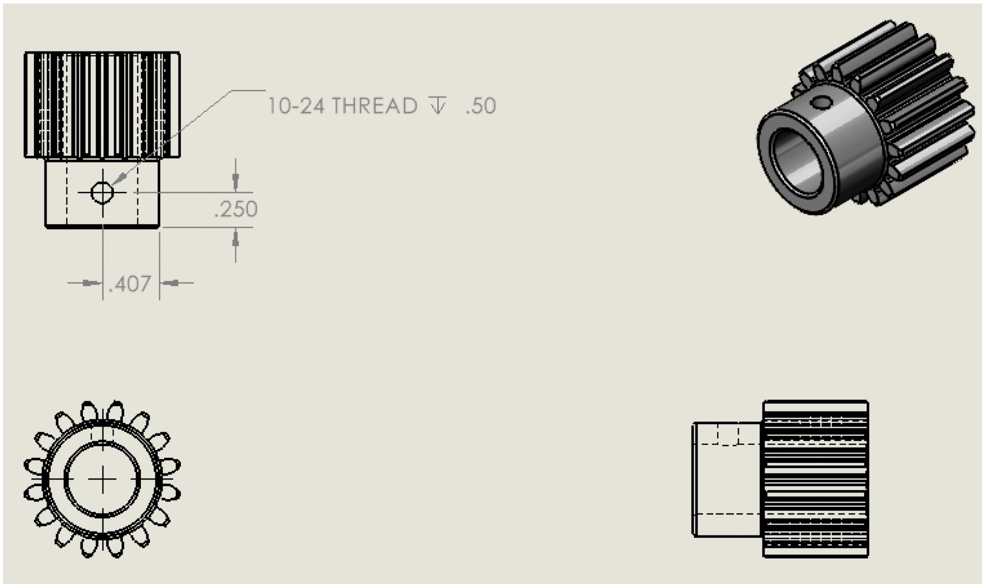


Figure B-6: Drawing with dimensions of hole location and callout

To manufacture the part, the pinion was fixed in the vise. Then the x and y datums were found using an edge finder. Next, a center hole was drilled using the drill chuck and a #2 center drill. The center drill was run at 1000 rpm. Next, a #25 drill was used to drill the rest of the hole at 1000 rpm. The next step was to tap the hole with a 10-24 tap. Finally, the holes were deburred and the part was cleaned. This process was repeated for the second pinion as well.

Central Shaft Manufacturing Plan

The central shaft was just a normal rod, so flat parts needed to be added to the part in order for the pinion set screws to have a flat surface to grip onto. In order to do this, a dremel was used to dremel a flat section in the areas where the set screws would be fastened. There were three sections that would be set-screwed to. The first was at one end of the shaft where the motor coupling would be attached, the other two were the locations of the pinions. The location of the first pinion is 2.5" from the end of the shaft that is attached to the motor measured from the hub face of the pinion (where the hub of the pinion faces the end of the shaft with the motor. The next pinion is located 50" from the first pinion.

Housing A and B Manufacturing Plan

Both housing parts were 3D-printed with PLA on an Ender V3 3D printer with a 15% infill. Once the parts were printed, all of the through holes had to be redrilled on a drill press to ensure that the holes were the correct diameter as well as straight. This was done on the drill press using a size G drill bit at 800 rpm. Care was taken to ensure that the drilling was done very quickly so that the plastic did not melt. This worked because the holes were already very close to their dimensions, so very little material needed to be removed. For drilling, the parts were mounted in the vise and then positioned such that the drill would drill through the center of the existing hole. For both housing parts, the shaft bearing was press fit into the large diameter hole using a press fit. For the Housing B, the smaller roller holes were tapped with a 8-32 tap. Then the rollers were screwed into the holes. Below in Figures B-7 and B-8 are the dimensions for the two 3D printed parts for Housing A and Housing B, respectively.

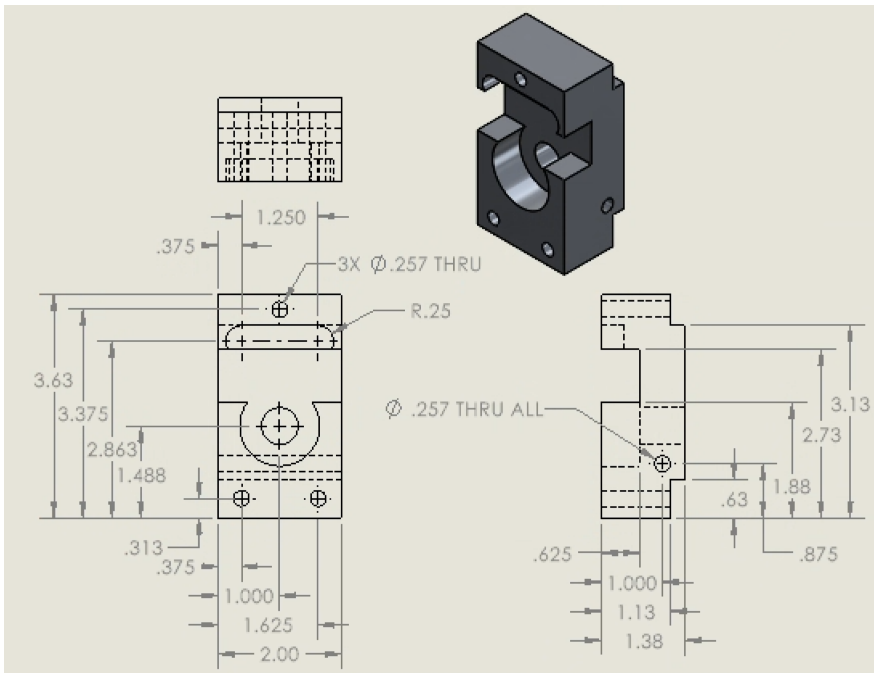


Figure B-7: Drawing of the Housing A 3D printed part

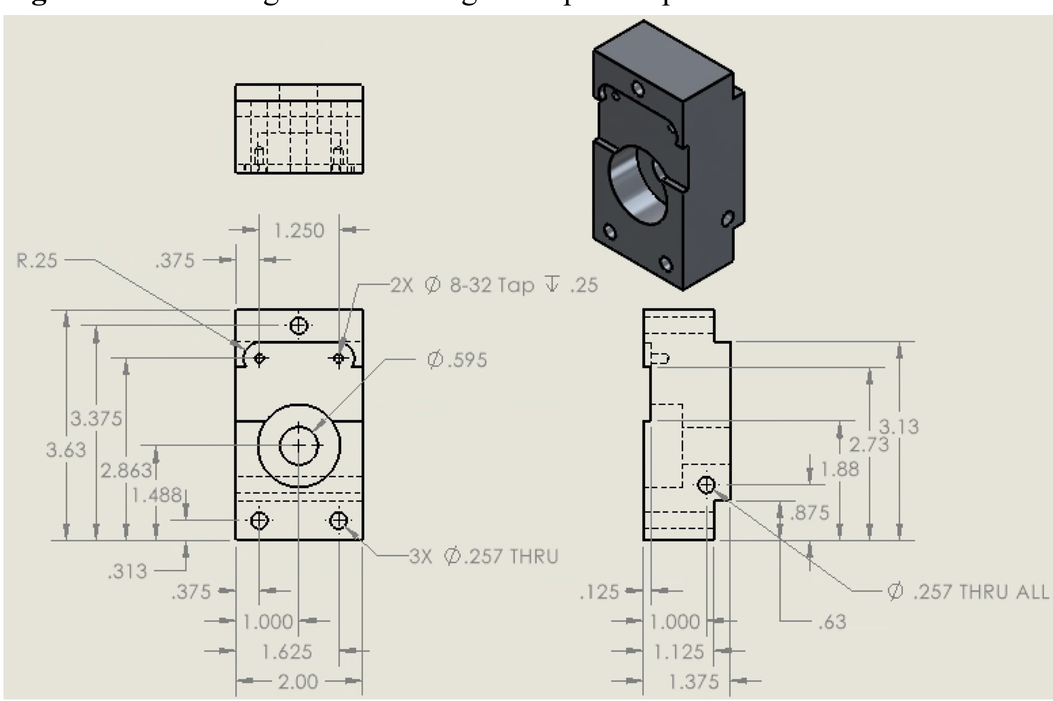


Figure B-8: Drawing of the Housing B 3D printed part

Manufacturing Plan for Motor Coupling

In order to connect the motor shaft to the central shaft a motor coupling was created. This coupling was made out of steel using a lathe. To start a piece of round stock was placed into the lathe and the end was faced off to create a flat surface. Then a drill chuck was inserted into the lathe and a #5 center drill and 33/64 drill bit were used to create the larger hole at a speed of 1200 rpm and 500 rpm respectively. From there a #3 center drill and a #20 drill bit were used to create the smaller hole at a speed of 1200 rpm and 500 rpm respectively. The piece was then parted at the desired length at 200 rpm and removed from the lathe. In order to create the tap holes a mill was used. The piece was mounted in the mill using parallels and an edge finder was used to find the datums. Next a #3 center drill and #25 drill bit were used to create the holes in the piece at a speed of 1200 rpm and 500 rpm respectively. A 10-24 tap was then loaded into the mill fixture and used to tap the holes. A dimensioned drawing of the motor coupling is given below in Figure B-9.

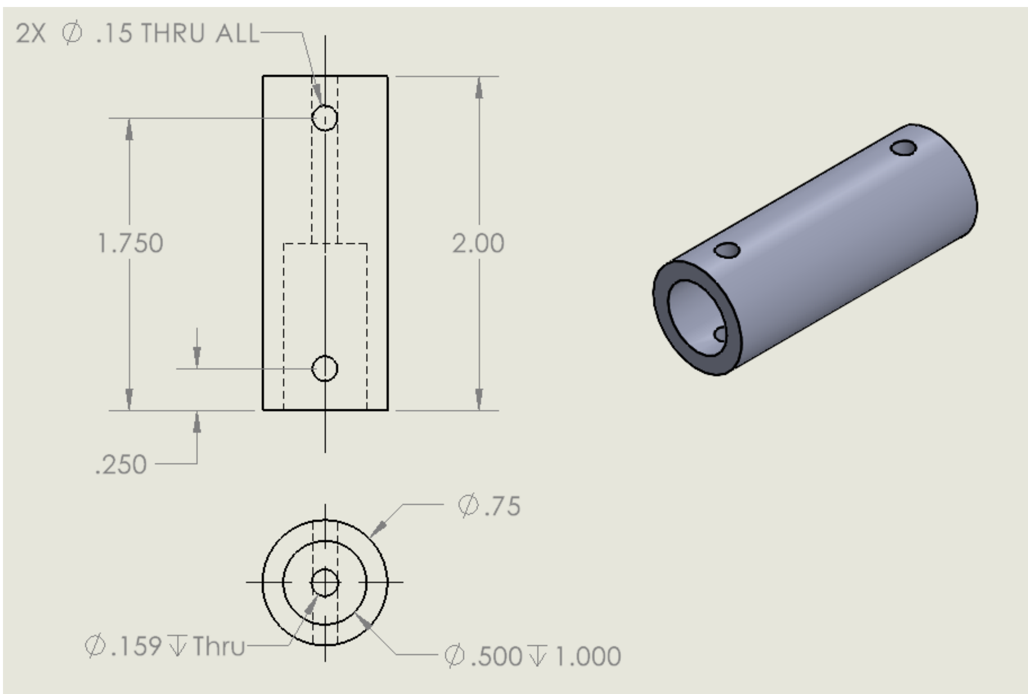


Figure B-9: Drawing of Motor Coupling

Manufacturing Plan for Rack Supports

In order to create the supports for the rack four 4’, 2x4s were used. To start the boards were cut to the desired length, four 2’ boards and four 1’ boards. The four 1’ boards were then run through a table saw in multiple passes to create the ¾” groove that the rack sits in. The boards were then assembled using eight wood screws (two screws at each interface between boards) for each support as shown in Figure B-10 below. Two supports were made, one for each rack.

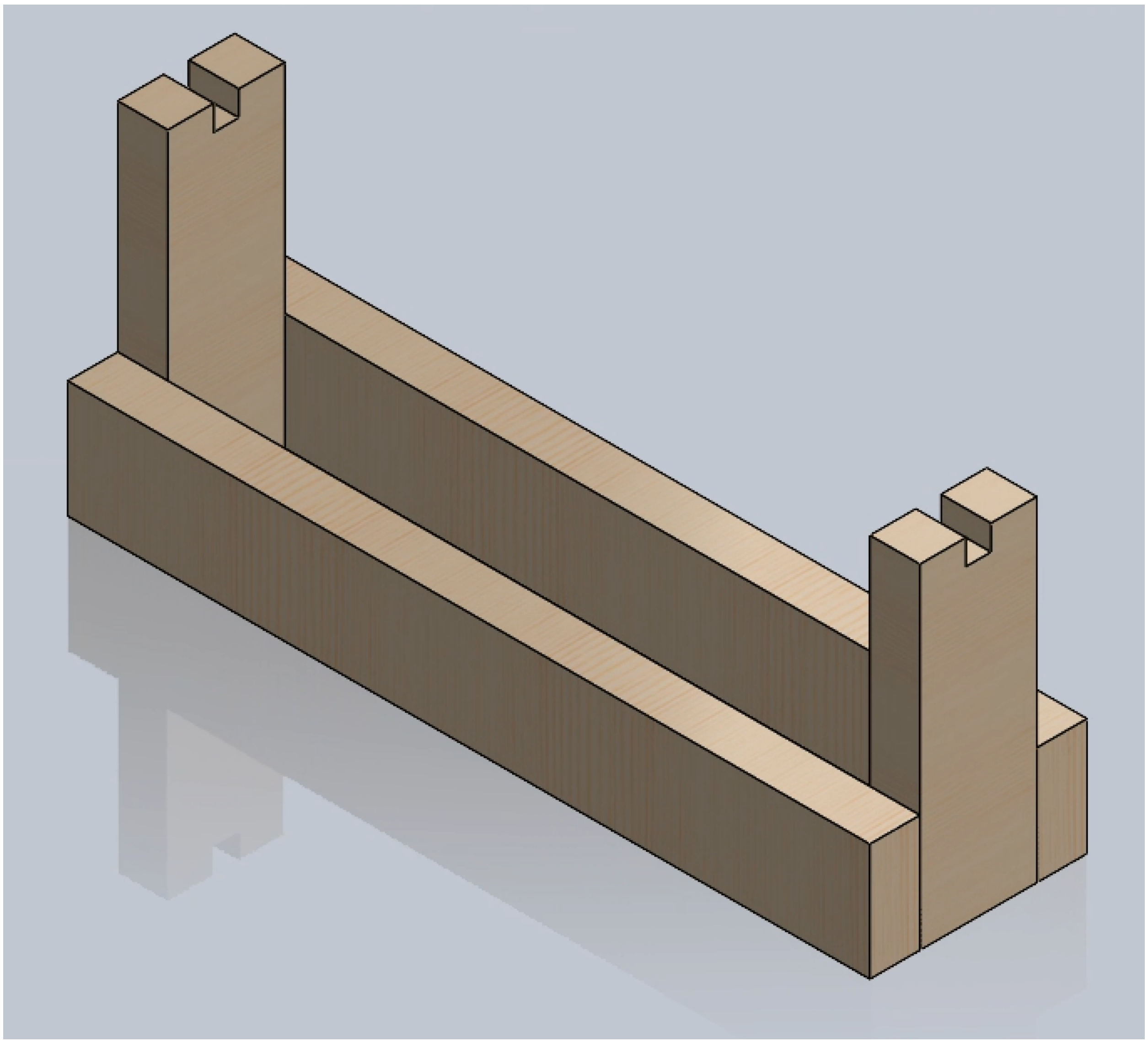


Figure B-10: Rack Support Assembly

Manufacturing Plan for Weight Holders

To construct the weight holders, four of the weight holders were waterjet from the 1/16” sheet metal aluminum. The dimensions for the shape that was waterjet from the sheet metal is shown below in Figure B-11.

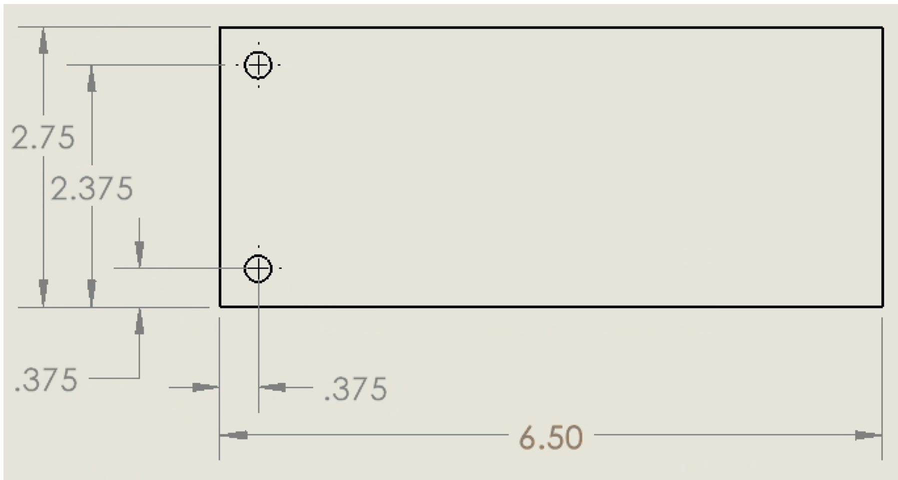


Figure B-11: Dimensions of the waterjet part for the weight holder.

After the parts were waterjet, they were deburred, and then the parts were bent to a 90º angle on the brake press in accordance with the drawing below in Figure B-12.

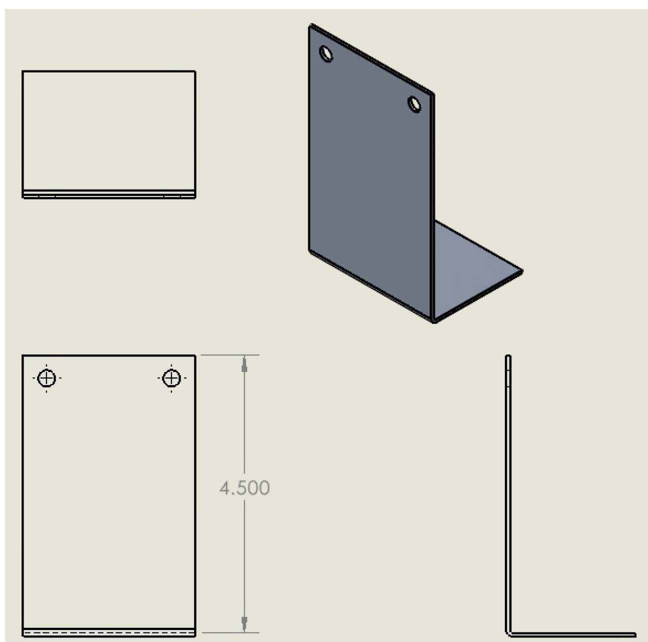


Figure B-12: Dimension of bend for the weight holder parts

Manufacturing Plan for Motor Mount

The motor mount was constructed in a very similar manner to the weight holders. First, the part was waterjet from the 1/16" aluminum sheet metal. The dimensions of the part that were waterjet from the sheet metal are given below in Figure B-13.

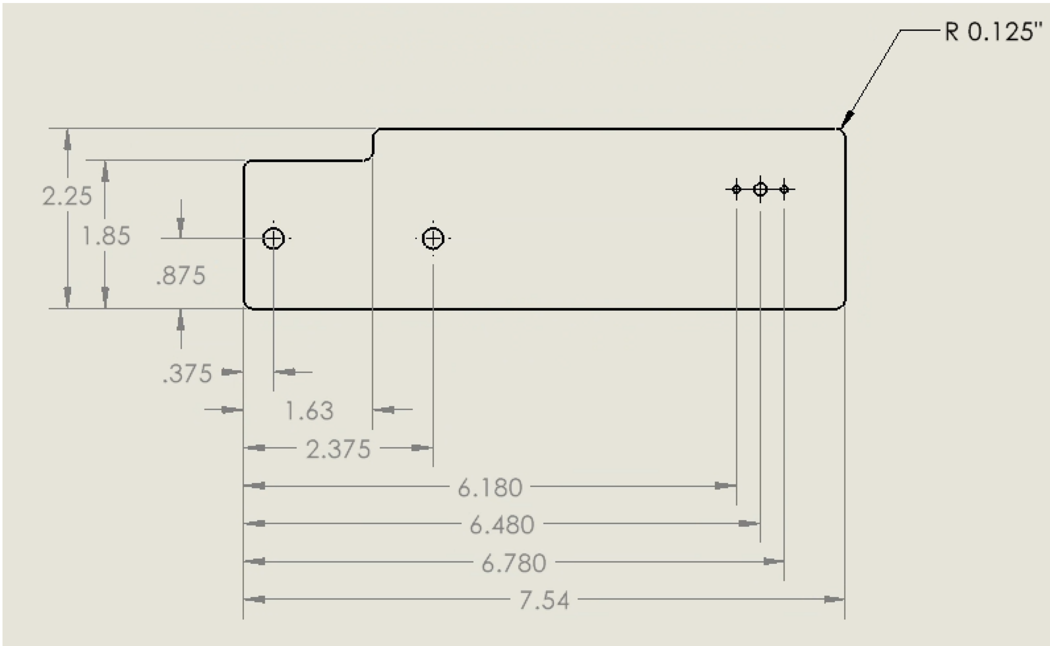


Figure B-13: Dimensions for the waterjet motor mount part

Then, just like the weight holders, the part was bent to 90º using a brake press at the desired length as given below in Figure B-14.

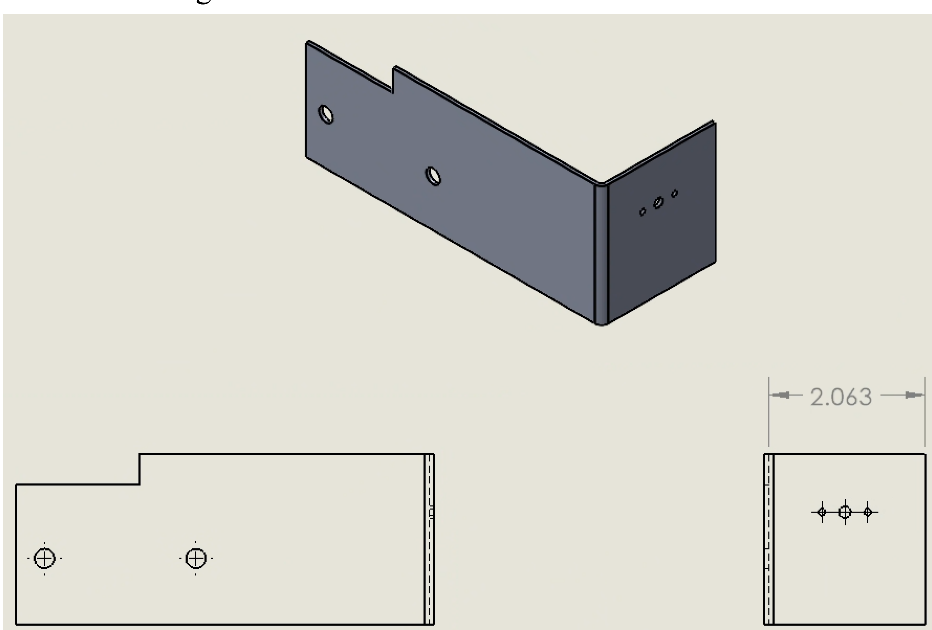
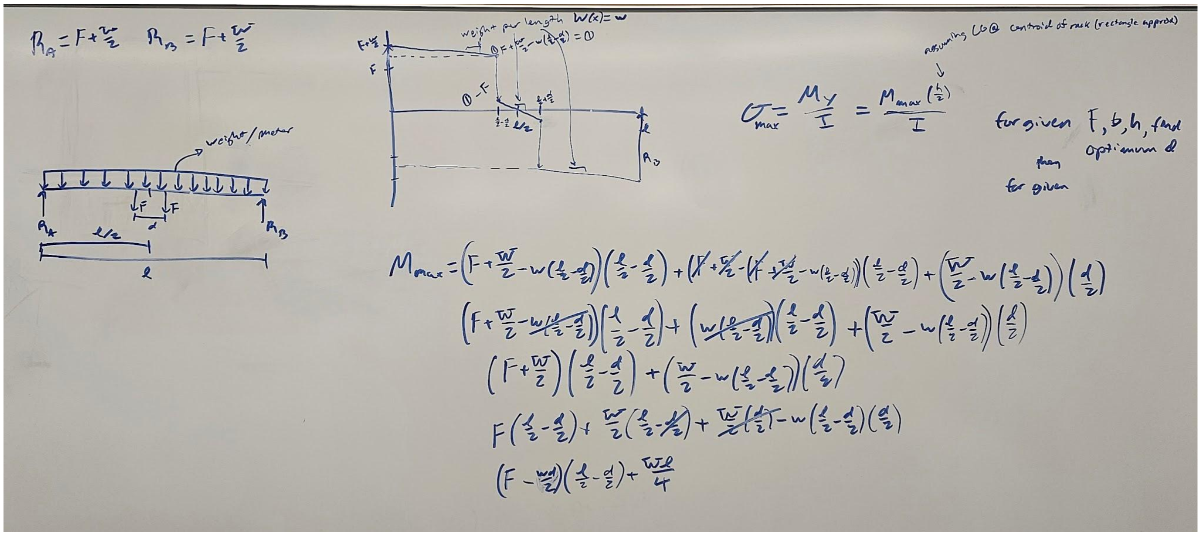
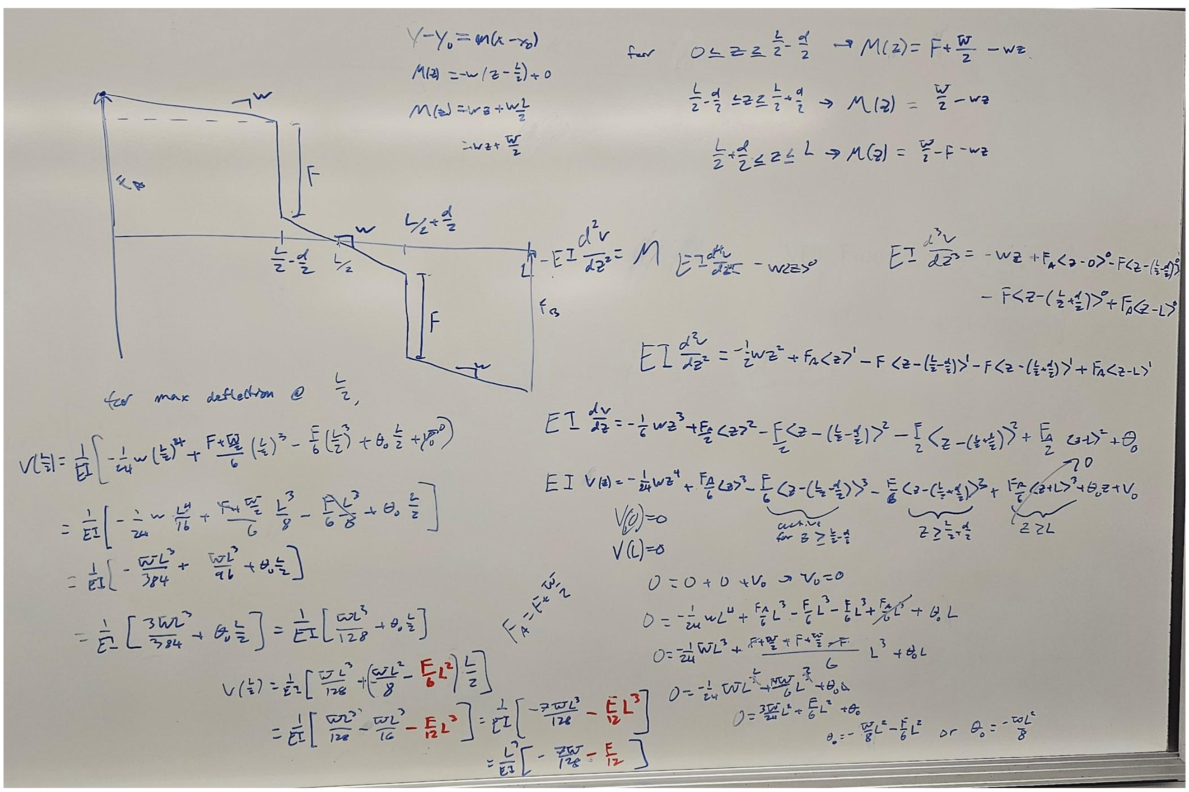


Figure B-14: Motor mount bend dimensions

APPENDIX C - ANALYSIS CALCULATIONS





```
close all;

%Define variables

rho_cs = 8000; %Density of carbon steel (kg/m3)
rho_al = 2700; %Density of aluminum 6061-T6 (kg/m3)
E_cs = 190e9; %Elastic modulus of carbon steel 1018 (Pa)
sigma_y = [370e6 370e6 370e6 370e6 370e6 370e6 370e6 370e6]; %Yield strength of carbon steel (Pa)

M_Housing = 3.83; %Mass of housing (kg)
W_Housing = 9.81*M_Housing; %Weight of housing

w_Rail = 9.81*(3.29+0.472+0.357) ; %Weight of rail (N/m)- Includes rail, board, and edge hold chain
l_Rail = 1.524; %Maximum length of rail suppored by one rack (m)
W_Rail = w_Rail*l_Rail; %Weight of rail

d_Shaft = 0.0254*0.5; %Diameter of shaft (m)
A_Shaft = pi*d_Shaft^2/4; %Cross sectional area of shaft along length of oven (m2)
w_Shaft = A_Shaft*rho_cs; %Weight of shaft (N/m)
l_Shaft = l_Rail; %Maximum length of shaft suppored by one rack (m)
W_Shaft = w_Shaft*l_Shaft; %Weight of shaft (N)

t = [.125 3/16 .25 .5 .75 1 1.25 1.5]./39.37; %Thickness of rack down length of oven
h = t; %Height of rack
A_Rack = h.*t; %Cross-sectional area of rack across width of oven

w_Rack = A_Rack*rho_cs; %Weight of rack (N/m)
l_Rack = 0.61; %Length of rack across oven (m)- roughly 2 feet
W_Rack = w_Rack*l_Rack; %Weight of Rack (N)

d = 0.0508; %Distance between rollers (m)
x = l_Rack/2; %Position of weight along rack

%Calculate reaction forces at ends of rack
F = 1/2*(W_Housing + W_Shaft + W_Rail);
Ra = 2*F - 2*F.*x/l_Rack + W_Rack/2; %Reaction force at left side of rack
Rb = 2*F.*x/l_Rack +W_Rack/2; %Reaction force at right side of rack

%Calculate second moment of area of the rack
Ix = (t.*h.^3)/12;

%Calculate the maximum bending moment of the rack
M_max = F.*(l_Rack/2-d/2)+W_Rack*l_Rack/4;

%Calculate the maximum stress in the rack
Y = h/2;  %Assuming CG @ centroid of rack (h/2)
sigma_max = (M_max.*Y)./Ix

figure;
hold on;

plot(t,sigma_max*1e-6, '+b', 'LineWidth', 2, 'DisplayName', 'Max Stress')
plot(t,sigma_y.*1e-6, 'r', 'LineWidth', 2, 'DisplayName', 'Yield Stress (370 MPa)')

legend('show');
legend('Location', 'NorthEast');
xlabel('Characteristic dimension of rack (m)');
ylabel('Maximum stress (MPa)');

hold off;
```

```
V_max = -1./(E_cs.*Ix).*(((-7.*W_Rack.*l_Rack.^3)/128)-((F.*l_Rack.^3)/12))
V_allowable = [0.003 0.003 0.003 0.003 0.003 0.003 0.003 0.003];

figure;
hold on;

plot(t,V_max, 'b', 'LineWidth', 2, 'DisplayName', 'Max Beam Deflection')
plot(t,V_allowable, 'r', 'LineWidth', 2, 'DisplayName', 'Maximum Allowable Beam Deflection (3mm)')

legend('Show');
legend('Location', 'NorthEast');
xlabel('Characteristic dimension of rack (m)')
ylabel('Maximum beam deflection (m)')
ylim([-0.2 1.2]);

hold off;

figure;
hold on;

plot(t,log(V_max), 'b', 'LineWidth', 2, 'DisplayName', 'Max Beam Deflection')
plot(t,log(V_allowable), 'r', 'LineWidth', 2, 'DisplayName', 'Maximum Allowable Beam Deflection')

legend('Show');
legend('Location', 'NorthEast');
xlabel('Characteristic dimension of rack (m)')
ylabel('Log scale maximum beam deflection (log(m))')

hold off;
```

```
rho_R = 7900; % Density of carbon steel (kg/m3)
r_Rack = 0.0127; % Radius of round rack (m)
Ix_R = (pi/4)*r_Rack^4; % Second moment of area of a circle
W_Rack_R = 9.81*rho_R*pi*r_Rack^2*l_Rack; % Weight of round rack

V_max_R = -1./(E_cs.*Ix_R).*(((-7.*W_Rack_R.*l_Rack.^3)/128)-((F.*l_Rack.^2)/12)) % Maximum deflection of round rack option (m)
```

```
Oven_Length = [232 232 267 267 342]/39.37; % Length of full shaft (m)
I = ((0.5.*w_Shaft.*Oven_Length)./9.81).*(d_Shaft^2)/4; % Moment of inertia of unkeyed shaft

rho_Pinion = 7870; % Density of carbon steel 1144 (pinion material) (kg/m3)
V_Pinion = 9.6e-6; % Volume of one pinion (m3)
W_Pinion = V_Pinion * rho_Pinion * 9.81; % Weight of one pinion (N)

Fric = 0.25; % Worst case friction force between the bushing and the shaft
n = [4 7 4 7 8]; % Number of sections
PD = 0.0254; % Pitch diameter of pinion (m)
R = PD/2; % Pitch radius

Fr = (W_Shaft + W_Pinion); % Force from the rack
Fp = Fr * R/(d_Shaft/2); % Force from the pinion in direction opposite of torque

Tm = n*(d_Shaft/2)*Fric*((W_Shaft + W_Pinion)*(1 + 2*R/d_Shaft) + (W_Shaft + W_Pinion + W_Housing + W_Rail)) % Torque of the motor (N*m)

G = 78e9; % Shear modulous of 1018 carbon steel (Pa)
Jz = 0.5*pi*(d_Shaft/2)^4; % Polar moment of area of a solid shaft

Phi = (Tm.*Oven_Length)./(G.*Jz) % Twist in radians
Phi_D = Phi*(180/pi); %Twist in degrees
Arc = Phi*R % Arc length - difference in movement between first and last pinion (m)

figure;
hold on;

Option_1 = [0.59317 0.59317 0.59317 0.59317 0.59317];
Option_2 = [0.31070 0.31070 0.31070 0.31070 0.31070];

plot(Oven_Length, Tm, '+b', 'MarkerSize', 10, 'Linewidth', 2, 'DisplayName', 'Required Motor Torque')
%plot(Oven_Length,Option_1, 'r', 'LineWidth', 2, 'DisplayName', 'Motor Torque of Option 1')
%plot(Oven_Length,Option_2, 'g', 'LineWidth', 2, 'DisplayName', 'Motor Torque of Option 2')

legend('Show');
legend('Location', 'SouthEast');
xlabel('Length of oven (m)');
ylabel('Torque required to rotate shaft (N*m)')
xlim([5 9]);
hold off;

figure;

plot(Oven_Length, Phi_D, '+b', 'MarkerSize', 15)

xlabel('Length of oven (m)')
ylabel('Maximum twist of shaft (Deg)')
```

```
Expo_Length = 60/39.37; % Length of expo shaft (m)
n_Expo = 2; % Number of sections

Tm_Expo = n_Expo*(d_Shaft/2)*(Fp + Fric*W_Shaft) % Torque of the motor (N*m)
```

APPENDIX D - GENERATED CONCEPTS AND TOP 12 CONCEPT DIAGRAMS

Our initial concept organization included a preliminary version of the actual organization method in use. The preliminary organization included the categories of keeping the lead screws and changing the chain drive, keeping the chain drive and keeping the lead screws, and new concepts that used neither the chain drive nor the lead screws. As mentioned in the concept generation portion of the report, the categories that were most advantageous to the organization of the concepts included parametric design changes, split system, and a combined system. The categories of keeping the lead screws and changing the chain drive and keeping the chain drive and keeping the lead screws were merged into the split system category. The new concepts category was more or less translated to the combined system category. Lastly, the parametric design changes category was pulled from the keep lead screws and keep chain drive categories.

The initial concept organization, which we called the "idea dump" is shown below with the three categories listed above. These concepts and categories were translated into the concept tree that is shown in the concept generation portion of the report.

Keep Lead Screws
Decrease Mass of Lead Screws
Descrease Friction of Lead Screws
Use a belt drive instead of chain drive
Use a gear train instead of chain drive
Use rack & pinion instead of chain drive
Use rack & pinion with individual racks for each lead screw
Hand crank on each lead screw
Use two actuators in total
Use one actuator for every two lead screws
Use one actuator every four lead screws
Use one actuator for every three lead screws
Use one actuator on every lead screw
Use rack & pinion with moving rack and stationary pinions
Use two actuators at beginning of tranmsiision ratio
Hand crank to turn all lead screws at once
Reciprocating motion of solenoid into rotary motion for lead screw

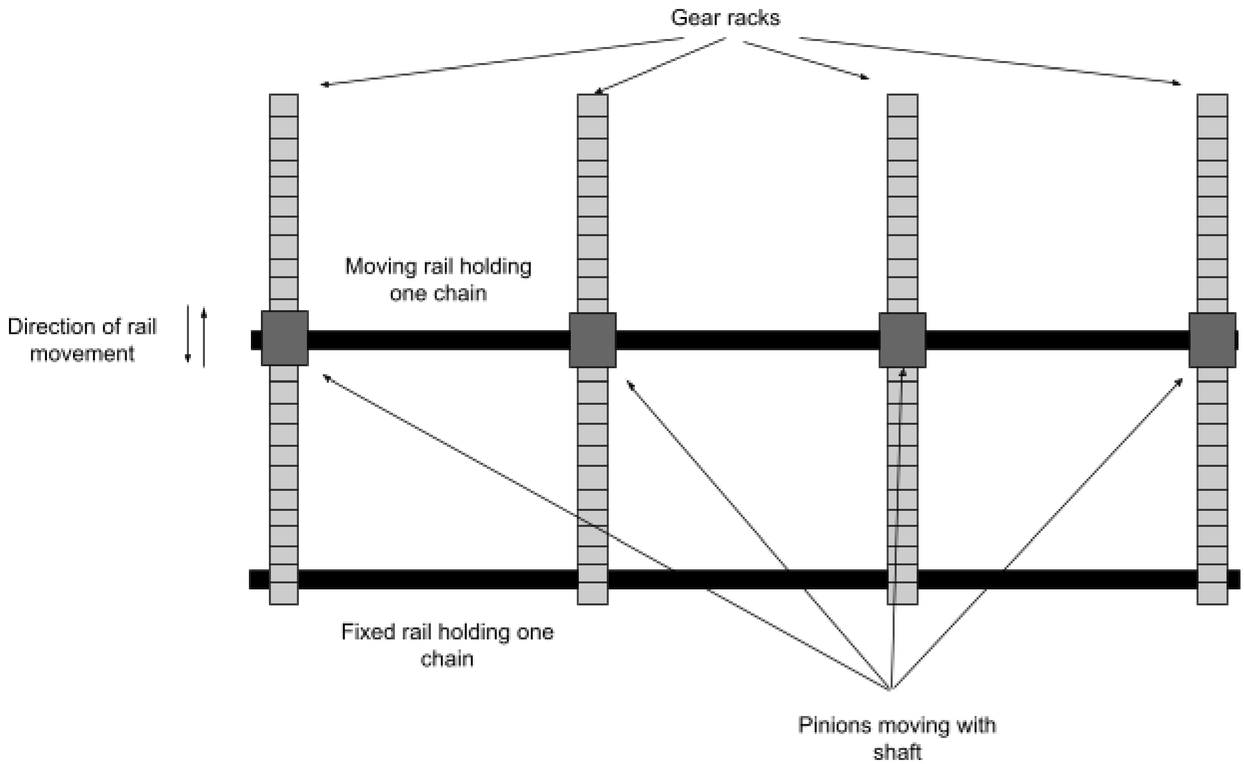
Keep Chain Drive
Increase Chain thickness
Increase size of idler gears
Increase # of idler gears
Change layout of idler gears
Use rack & pinion to move EHC
Use belt drive to move EHC
Multiple concurent chains
Miter gears to better transmit torque
Cam and follower to move EHC
Change material of chain to increase durability
Gear reduction on each lead screw
REPLACE BOTH - 2 SEPERATE SYSTEMS

New Concepts
Rack & pinion with rod connecting all pinions powered by motor
Hydraulic cylinder moves EHC to position
Thermal expansion causes EHC to move into place
Tilt oven to slide rails into place
Nuclear powered EHC
Actuator attached to rail and moves on a stationary rack
Rail rests on a cat track
Manually adjust rails with oven mitts
Chemical reaction to cause pressure expansion causing rail size adjustment
Gear train with miter gear into another gear train moving EHC into position
Manually adjust rails with crank
Magnet/electromagnet
Inflate air bag to move rail in and out
Pins extend in and out instead of rail moving
Microimplosion device to cause large shock to move rail into positon
Vertical slot on side of oven, horizontal slot to push rail in and out
Cam and follower with spring to hold rail in place
Manual crank on outside of oven that has electronic sensor indicating the position
Nanobots with microthrusters that push EHC into position
Large sail with air shot on each side to move EHC to desired position
Moving rack spins pinions that turns linkage to move rail
Slider crank driven by one actuator with 90 degree gears
Shape memory alloys that adjust into desired length based on current
Regular conveyor
Crank, moving EHC which follows a slider
Linkage attached to shaft
Conveyor belt that is not EHC
Bar with a curved profile to guide EHC

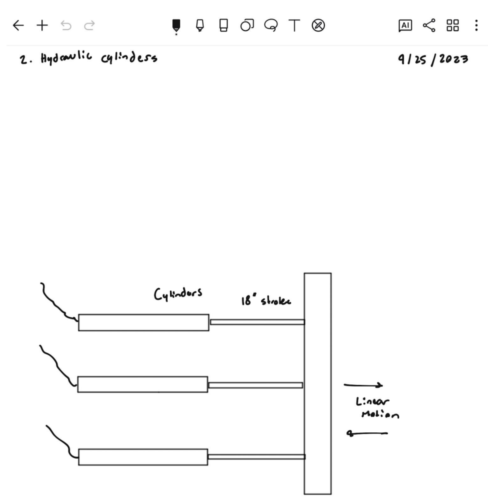
	Telescoping mechanism	
	Photon gun + solar sail	
	Put oven in a centrifuge to increase force of gravity overcoming the resistane of the EHC and moving it into space	
	Spring to pull EHC to and from positon	
	Two shafts with chains attached to each side of moving rail to pull it	
	Particle accelerator to create gravity	
	Series of pulleys to allow for one rope to pull rails in and out	
	Smart material that changes size to exact size needed for EHC	
	Large jet of water to push rails	
	Bike pedals to coil chain to pull rail	
	Stool-weigh potential energy apparatus	
	Hanging rail attached to chain loop above it that moves it in and out	
	Gravity powered zipline to move rail	
	Fluid-filled accordion rail	
Small cutout in cool section of oven for human to sit and manually move EHC with string chain and sprocket (permenant employee)		
	Vertically hang PCBs from edge to carry through oven	
Accelarate the oven significantly in the direction perpendicular to the EHC motion to overcome static friction and move EHC into location		
	Build new oven for every possible EHC position	

Design #	Description
Design 1	Rack and pinion with multiple racks. Racks run along the width of the oven and are fixed to the oven walls. Pinions are connected by a singular shaft that runs the length of the oven.
Design 2	Multiple actuators. This places an actuator on every lead screw. Actuators will be kept in sync by a control system.
Design 3	Hydraulic cylinders. Uses hydraulic cylinder(s) to move the EHC into place. Hydraulic cylinder(s) will push EHC back and forth along support rails spanning the oven width
Design 4	Oven Tilt. Lift one edge of the oven to move the EHC into place. EHC will slide along a support rail into the desired position
Design 5	Increase chain thickness by ¼” to reduce chain stretch and slippage.
Design 6	Spring. When the EHC moves, a spring compresses. To move the EHC back, the spring is released. EHC is moved by a solenoid or some other type of linear actuator
Design 7	Multiple chains. Multiple chains running the length of the oven instead of a singular chain. Each idler gear will have two chains attached, and each chain will span between two consecutive lead screws.
Design 8	Extending pins. Move the pins that the PCBs rest on instead of the EHC in and out (along oven width) to accommodate different PCB widths.
Design 9	Crank and slider. Crank and slider to move the EHC into position. Crank and slider located above the EHC inside the oven cavity. Slider runs along a guide which spans the width of the oven
Design 10	Manual crank on each lead screw. Place a human powered crank that turns each lead screw individually. Electronic sensor to indicate position
Design 11	Cam and follower. Cam and follower to move EHC into position. Cam and follower located above the EHC inside the oven cavity. Spring used to hold the EHC in place
Design 12	Rack and pinion w/ linkage. A singular long rack runs the length of the oven which turns pinions attached to a linkage that moves EHC. Rack moves in and out along the length of the oven to rotate pinions

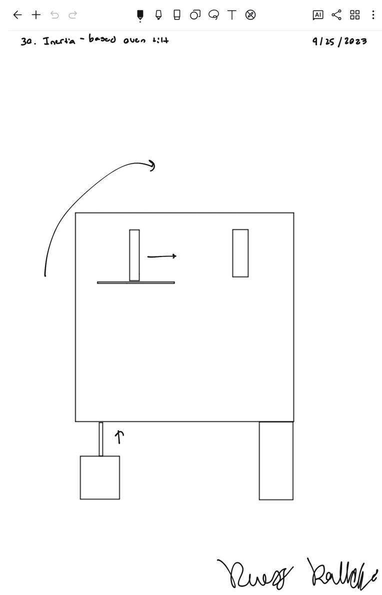
Design 1:



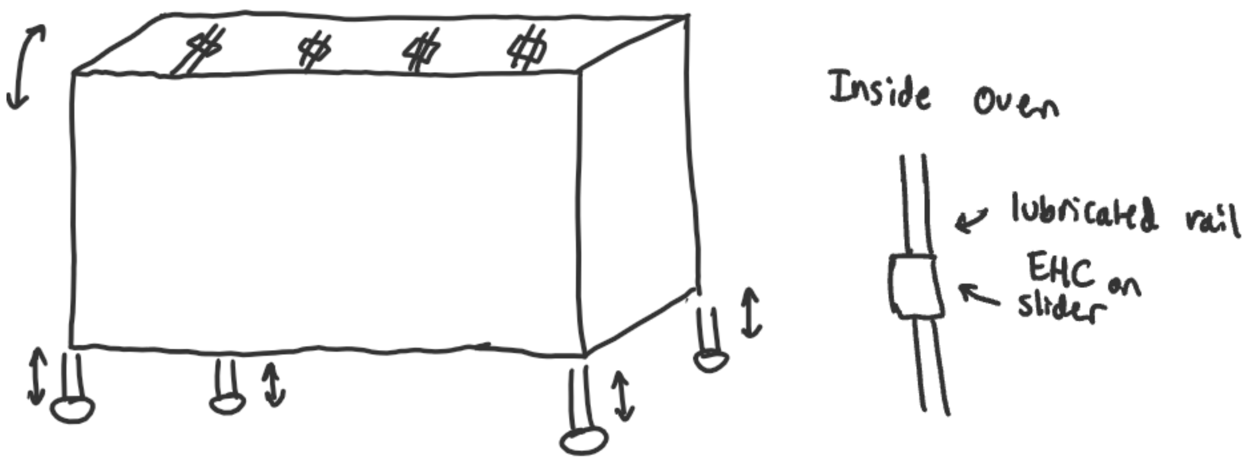
Design 2:



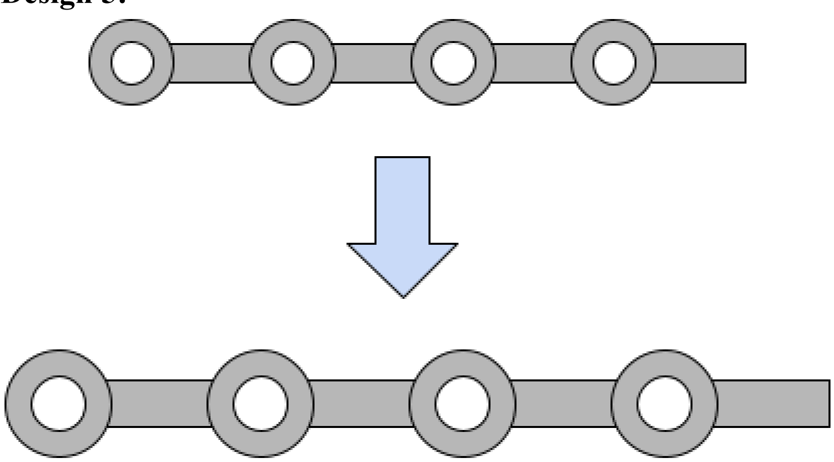
Design 3:



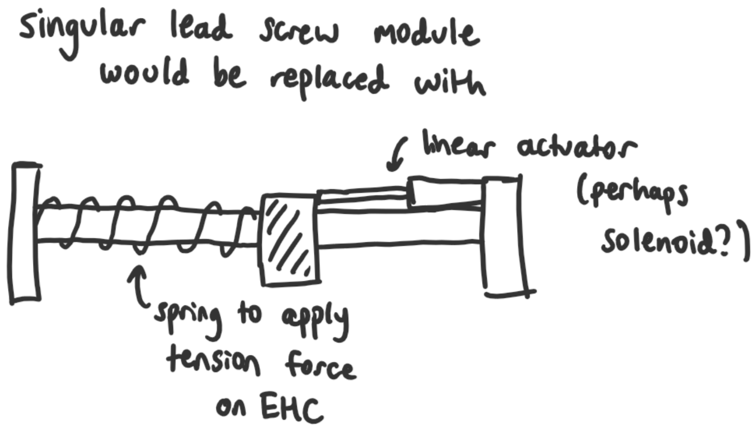
Design 4:



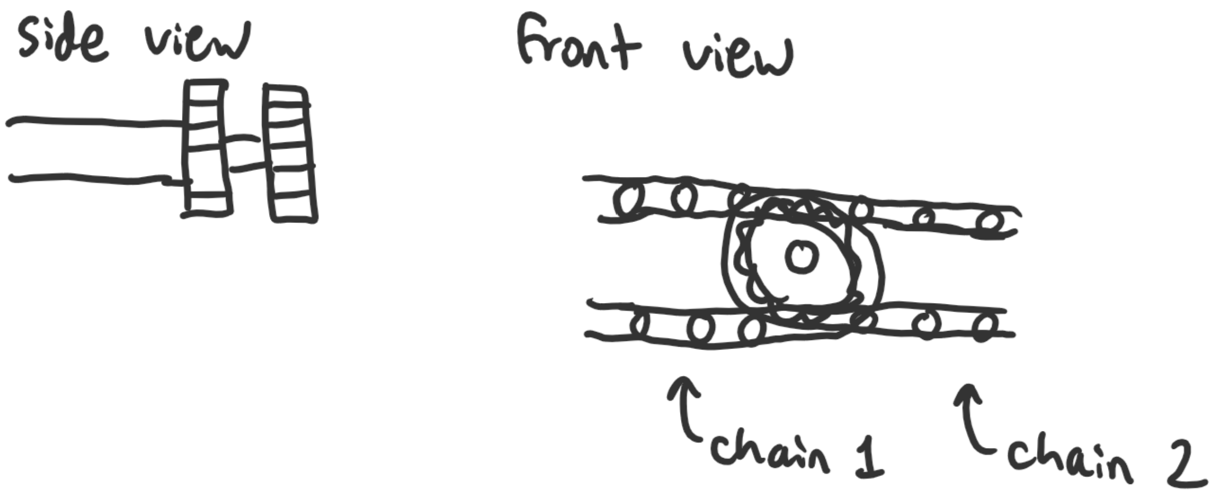
Design 5:



Design 6:

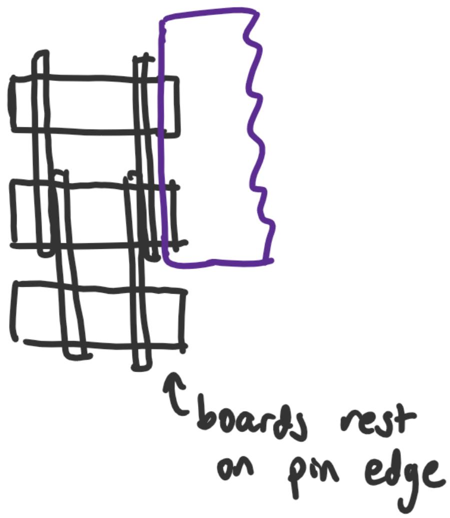


Design 7:



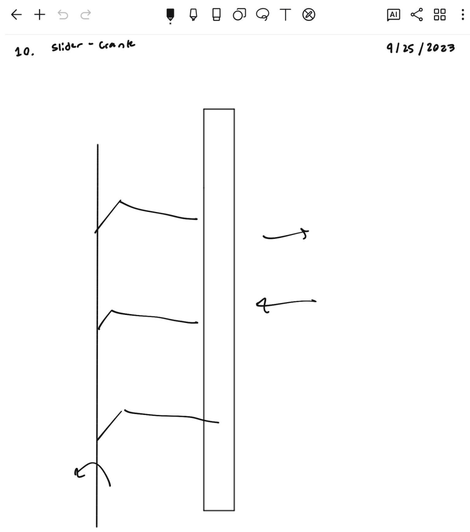
Design 8:

Current EHC chain

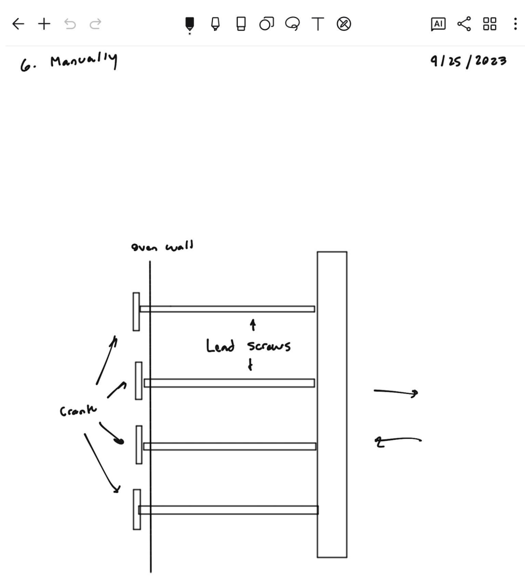


New idea

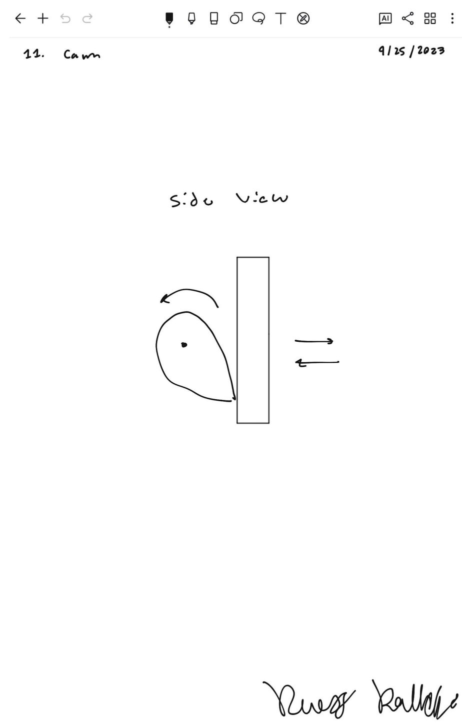
Design 9:



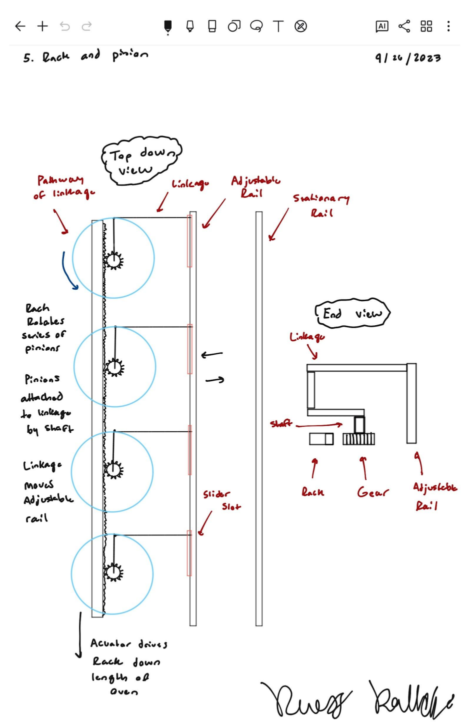
Design 10:



Design 11:



Design 12:



APPENDIX E - ARDUINO CODE AND WIRING DIAGRAM

The Arduino code tells the motor to run back and forth. When it hits a limit switch, it will change direction and begin moving towards the other limit switch.

```
int Lswitch1 = 2;
int Lswitch2 = 4;

int in2 = 13;
int in1 = 12;
int ena = 11;

int speed = 100;

void setup() {
  // put your setup code here, to run once:

  Serial.begin(9600);

  pinMode(Lswitch1, INPUT);
  pinMode(Lswitch2, INPUT);

  pinMode(in2, OUTPUT);
  pinMode(in1, OUTPUT);
  pinMode(ena, OUTPUT);

  goLeft();

}

void goRight(){
  digitalWrite(in2, LOW);
  digitalWrite(in1, HIGH);
  digitalWrite(ena, speed);
}

void goLeft(){
  digitalWrite(in2, HIGH);
  digitalWrite(in1, LOW);
  digitalWrite(ena, speed);
```

```
}

void loop() {
  // put your main code here, to run repeatedly:
  if (digitalRead(Lswitch1) == HIGH) {
    Serial.println("Left Boundary");
    goRight();
    delay(500);
  }

  if (digitalRead(Lswitch2) == HIGH){
    Serial.println("Right Boundary");
    goLeft();
    delay(500);
  }
}
```

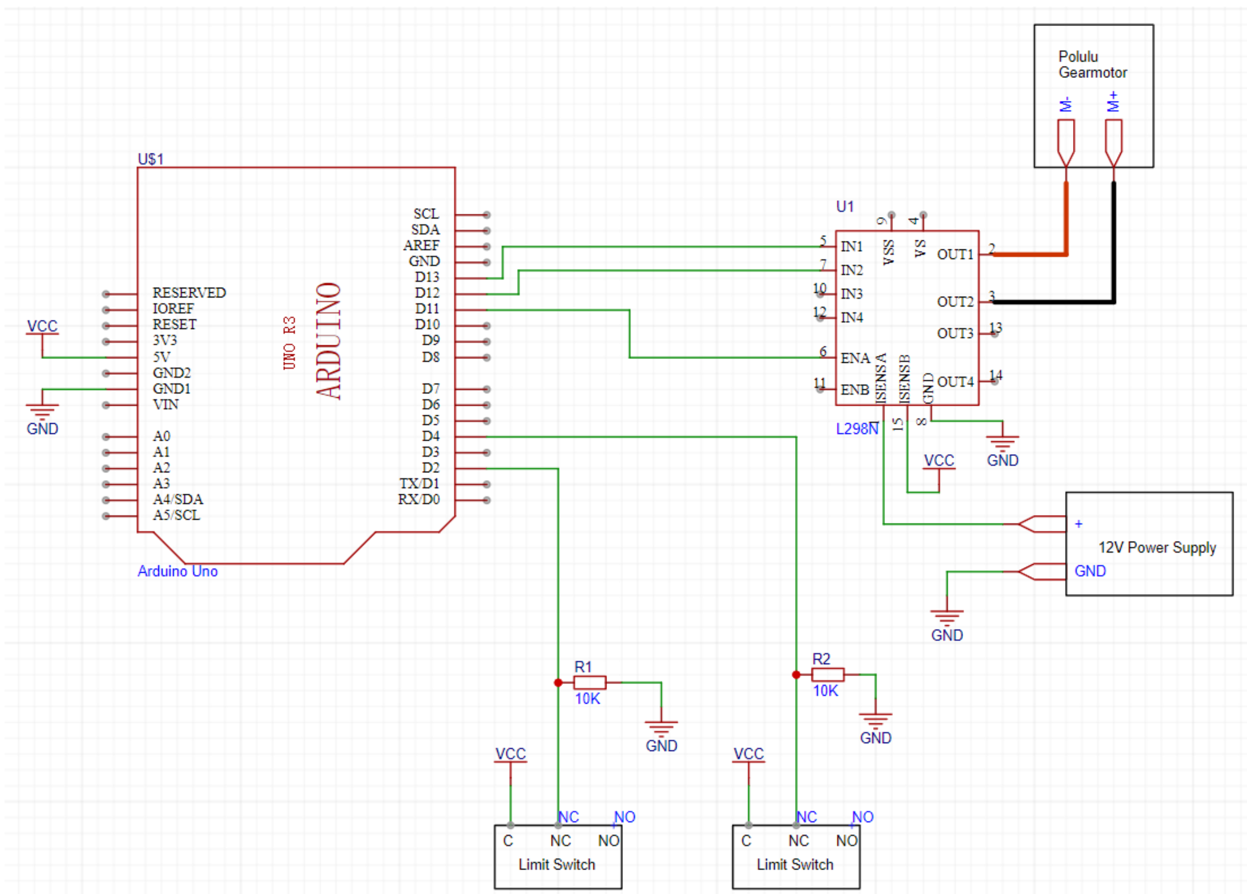


Figure E-1: Wiring diagram of all electronics used in the build design which includes two limit switches, an H-bridge, a power supply, a motor, and the Arduino.