Integrated Park Module Design and Cost Optimization

Team 6

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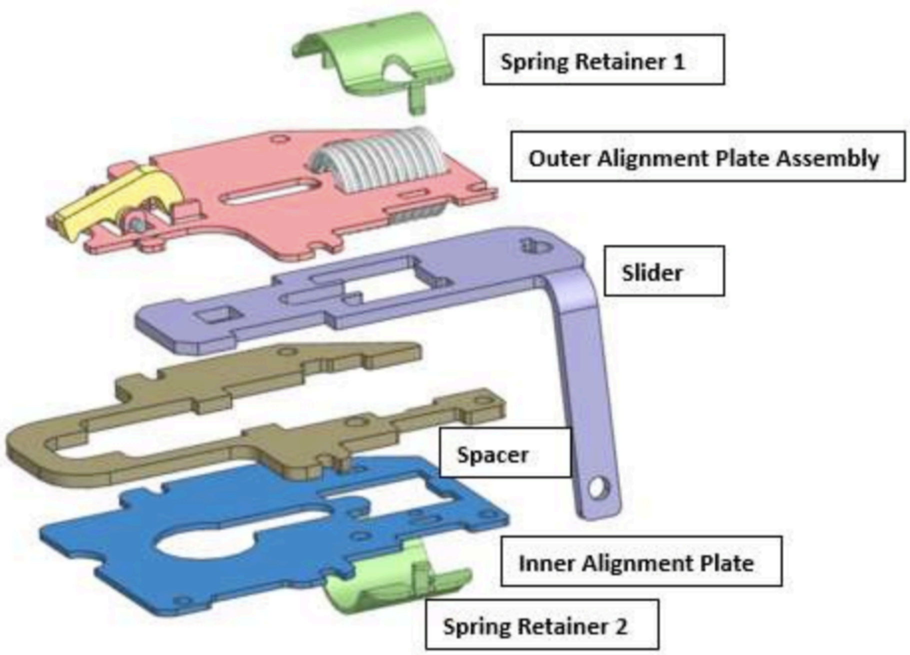

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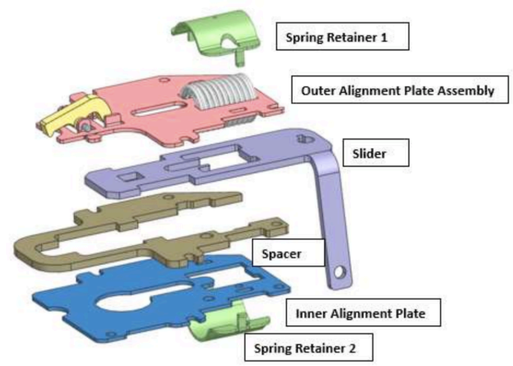
Executive Summary



With the growing competition in the automotive industry and opportunity cost between sourcing labor out of the country or to different producers, automotive suppliers are forced to continuously improve on design. Cost and weight savings serve as a primary way to maintain a strong profit margin. With this in mind, Stoneridge, an automotive supplier, has tasked us with reducing the cost and decreasing the weight of their Integrated Park Module (IPM)[9] shown to the right. The IPM rests inside the transmission and serves to keep the car in park when it is meant to be in park. Due to its placement in the transmission, the assembly must survive extreme temperatures and vibrational loads. With this in mind, the solution that our team creates must be durable (surviving 300,000 cycles unlocking and locking the transmission)[9], able to withstand the loading and temperature conditions of the transmission (tested to temperature spans of -40 to 120°C and vibrational loading with amplitudes up to 10 gs of force)[9] and be non-corrosive (martials will not degrade from the transmission fluid). Additionally, the final design must reduce the cost with a shorter or similar manufacturing time (cycle time kept at or below 30 seconds)[22].

With the requirements and specifications for our final design in mind, the stakeholders affected by our product should not be negatively impacted. Primarily, design changes and possible solutions need to consider the impact on Stoneridge, the major OEM's purchasing the IPM, and the end users driving the cars purchased from the OEM's. Along the way, we have noted that the design we present will indirectly affect manufacturing companies, material suppliers, mechanics and Stoneridge competitors.

Satisfying the hard set requirements and specifications of the project while serving the stakeholders of the end product poses a series of challenges that we anticipate will affect our path to a successful solution. From the set of requirements, we have analyzed a wide spectrum of possible design solutions to reduce the cost of the IPM assembly. These design solutions were separated into categories including material changes, manufacturing process changes, material elimination, and part consolidation. The design categories were established to allow comparison between different solutions with decision matrices and morphological charts.

From comparison of the design solutions, we have established a redesign for the IPM. The Inner and Outer Alignment Plates and the Spacer will be changed from HSLA Steel, grade 45 to either PA66 33% glass filled or PA6 33% glass filled (glass filled, polymer based materials). With the material changed to plastic, an injection molding manufacturing process will be used to make the new parts. With injection molding being a flexible manufacturing process, implement plastic ribbing for the Spacer to reduce the weight and manufacturing time of the part.

Through engineering analysis, we analyzed the effectiveness of the design solutions. We quantified weight reduction of the part due to material changes and quantified the cost reduction of using injection molding as the manufacturing process for the redesigned IPM. Additionally, we have developed a Finite Element Analysis model to simulate the loading and environmental conditions of the IPM. We used the simulation to place a ribbing net in the Spacer and OAP to reduce stress and displacement. We used an injection molding fluid flow software to optimize the manufacturing cost of the assembly. Between the Finite Element Analysis model and the fluid flow software, we iterated on our design to converge on an optimized and cost effective design. In the future, we plan to use topographical analysis to optimize the ribbing design to further reduce the cost of the part and perform physical tests on our final design at Stoneridge.

Problem Statement

Our task is to optimize Stoneridge's Generation 2 Integrated Park Module (IPM) design for weight and cost savings when compared to their existing design, while considering the car's transmission loading and environmental conditions. The IPM serves as a mechanism to lock the driveshaft of the transmission in place for user safety. We are analyzing solutions for a material change of the assembly from steel to PA6 13% Glass Filled, part consolidation, and ribbing for injection molded parts. Design validation will be performed with Finite Element Analysis supported by first principles modeling, and Injection Molding Fluid Flow Analysis.

Project Introduction

Stoneridge

Our project sponsor and creator of the IPM is Stoneridge. Stoneridge is known for designing and manufacturing electrical and electronic systems, components, and in our case, modules, for the automotive market[1]. Our mentors are Harish Athipatia, who is the Product Engineering Manager, and Haroun Askar, who is the Product Engineer currently working on modifying the IPM.

The image below shows the current design of Stoneridge's Generation 2 IPM, or integrated park module.

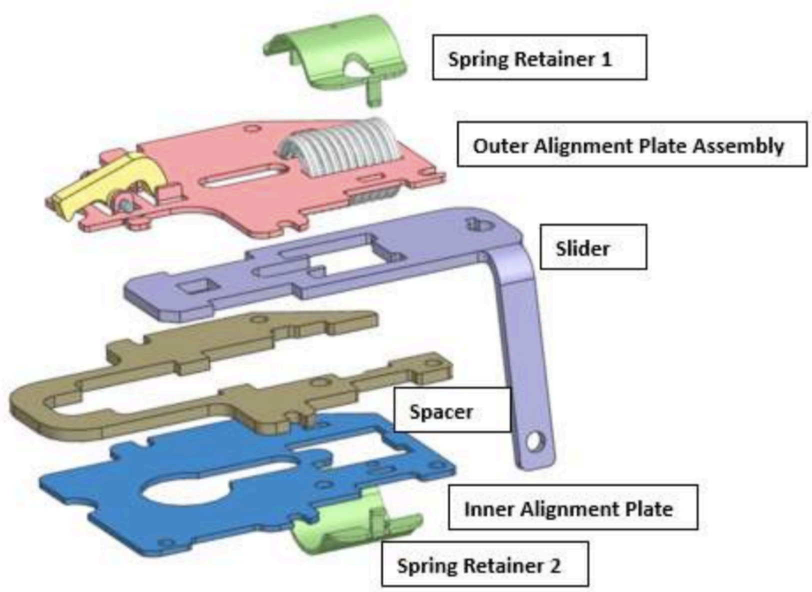


Figure 1. Exploded view of Stoneridge's Generation 2 IPM Design

This design has the two spring retainers, which are made out of plastic, while the other four parts, from top to bottom: Outer Alignment Plate (OAP), Slider, Spacer, and Inner Alignment Plate (IAP), are currently made out of steel. The OAP, Spacer, and IAP are going to be our main focus, as those are the parts that have minimal load and remain stationary. There are a few aspects of this design that have room for improvement. We are going to be mainly focusing on decreasing both the overall cost and weight for the IPM. This can be accomplished through material changes and design simplification. Steel is both an expensive and heavy material[2], so by implementing some material changes on these parts both cost and weight can be reduced. There will also be an attempt to reduce the carbon footprint, whether it be through manufacturing, use, or waste, so applying a material change to a sustainable material is also something that will be looked into.

Background and Benchmarks

I. Transmission and Parking Mechanism Operation

The transmission is the mechanism in a vehicle that causes the engine to drive at a particular speed and is typically located under the front hood of a vehicle. The parking gear is connected to the output shaft of the transmission. In driving mode, the parking gear is free to rotate around its axis which allows the vehicle to freely move. The park pawl is the mechanism that locks the parking gear so that it is stationary. Once the parking gear is locked, it cannot rotate, thus the transmission shaft as a whole is locked and is prevented from rotating[3]. This is known as the parking mode and a schematic of this functionality can be seen in **Figure 2** below. This is an important safety feature in vehicles that prevents the vehicle from rolling. The situations in which this functionality is crucial is when the vehicle is on an incline. In this situation, the parking mode prevents the vehicle from potentially rolling down the incline and injuring the driver, passenger, or any potential vehicles behind them.

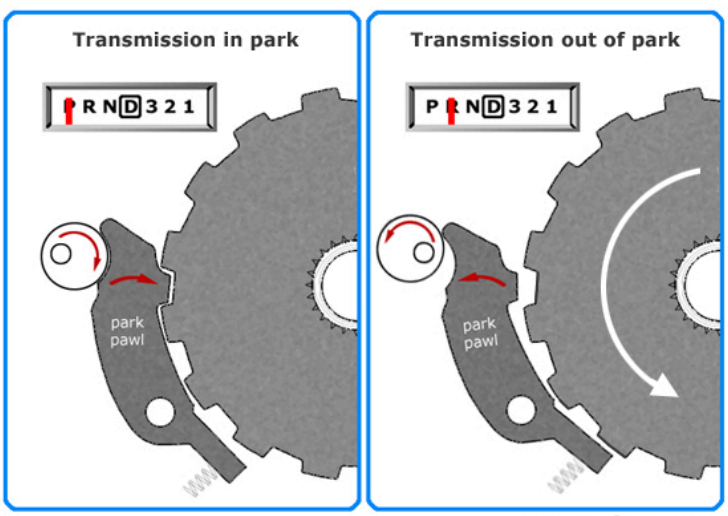


Figure 2. Model of transmission actuation in a vehicle[4].

The transmission and parking mechanism act in a similar manner for our IPM assembly. An image of the mechanism can be seen in **Figure 3** below. The entire system shown is the transmission and the IPM is integrated, in other words internally mounted, into the transmission. In parking mode the slider inside the IPM is actuated by a solenoid and a spring, which pushes the rod forwards, and in turn shifts the parking pawl upwards so that it is directly adjacent to the parking gear. In driving mode, the slider inside the IPM retracts, which lowers the rod, and thus lowers the parking pawl back into its original position. Since it is no longer adjacent to the gears of the parking gear, the output shaft is free to rotate.

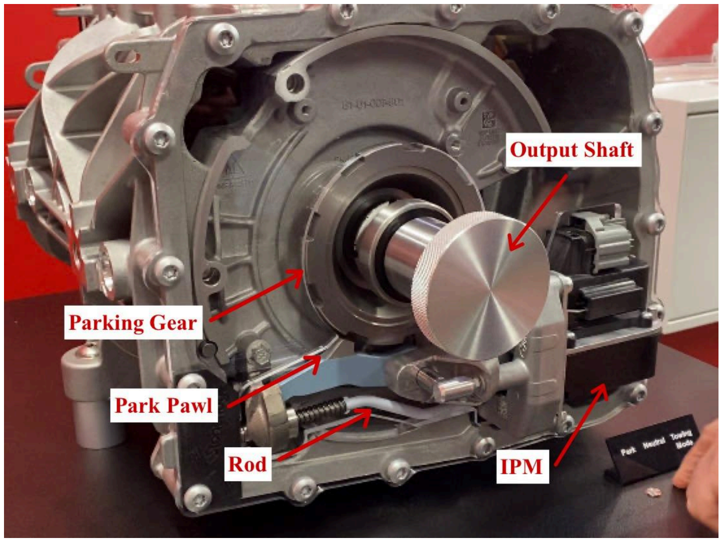
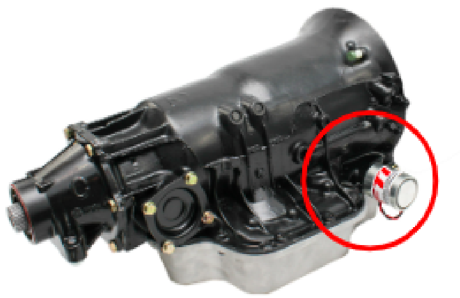


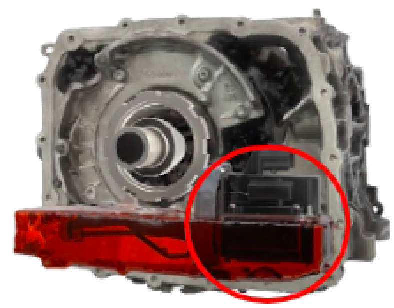
Figure 3. Example of a current transmission-IPM assembly for Generation 2 design. To add a sense of scale to the IPM in the transmission, the IPM is roughly the dimensions of 15 x 12 x 10 cm^3.

II. Benchmark: External vs Internal Mounting

The mechanism that can actuate the vehicle between parking and driving mode can be mounted externally or be integrated into the transmission. For the IPM, the mechanism is mounted internally. However, for many vehicles in the industry it is typical to have an externally mounted parking mechanism.There are qualitative differences between the two mounting methods which can be summarized in **Table 1.**

Table 1. Qualitative differences of installation, noise, vibration, material selection, and replacement between an externally mounted and internally mounted braking mechanism. The braking mechanism is circled in red for clarity. The externally mounted braking mechanism shown on the left is equivalent to the internally mounted braking mechanism on the right. Both have the same function, one is just inside the transmission; its placement in the transmission is used to improve performance and reduce noise due to lubrication from the transmission fluid.





Benchmark	Externally Mounted[5]	Internally Mounted (IPM)[6]
Installation	Attach part to transmission	Part comes with transmission
Noise	Increased noise	Decreased noise
Vibration	Increased vibration	Decreased vibration
Material Selection	Less environmental constraints, larger material selection	Need to consider transmission fluid conditions, limits material selection
Replacement	Replace external braking assembly	Replace entire transmission

A more in-depth explanation includes the following:

a. **Installation:** For an externally mounted mechanism, it is attached to the body of the transmission. In comparison, an internally mounted mechanism would be a part of the transmission assembly.

b. **Noise and Vibration:** For an externally mounted mechanism, due to it being attached to the transmission body, the vibrations in the transmission due to engine operation causes the mechanism to vibrate as well. This causes an increased amount of noise and vibration in comparison to an internally mounted system[7].

c. **Material Selection:** For an internally mounted mechanism, due to the fact that it lies within the transmission, the effect of transmission fluid needs to be taken into account. Transmission fluid can reach temperatures of 120 ℃ and can cause corrosion[8]. Because of these environmental considerations, there is a limited material selection for an internally mounted mechanism as opposed to an externally mounted mechanism that does not have those environmental conditions.

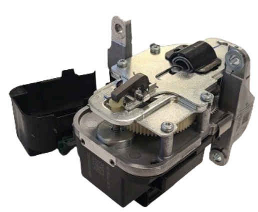
d. **Replacement:** One of the main drawbacks of an internally mounted mechanism is in the case of defects. If the mechanism faces defects or needs to be replaced before the life cycle of the transmission, the entire transmission would have to be replaced. In contrast, for an externally mounted mechanism, the mechanism mounted can be directly replaced since it is functionally separate from the transmission[9].

Although the IPM being an internally mounted mechanism allows for there to be decreased noise and vibration from transmission operation, improving the user's experience with the vehicle. It is important to improve on the current design by decreasing cost and weight of the assembly, while also maintaining that the mechanism can withstand the environmental and load conditions for the tranmission's entire lifetime. This would be a successful project outcome and is the overall goal we are trying to achieve. A typical transmission can last up to 80,000-150,000 miles depending on how often the owner keeps up with routine maintenance[10]. Testing and verification will be done by exposing the IPM to the equivalent of 300,000 cycles at Stoneridge's testing facility. Without meeting this specification, the device will fail prematurely, resulting in a replacement of the affected transmissions.

III. Benchmark: Current Design vs Industry

The current IPM Generation 2 mechanism can be compared against its previous Generation 1 iteration, an externally mounted braking mechanism available on the market, and a parking brake mechanism available on the market shown in **Table 2** on pg. 6. The parking brake mechanism serves a different purpose in comparison to the transmission braking system mentioned prior. The purpose of a parking brake is to lock the back wheels of the vehicle[11]. Due to this purpose of needing to clamp the weight of the entire vehicle, it provides a significantly higher brake force of 18,500 N in comparison to 450 N for the transmission braking system.

Table 2. Qualitative metrics for the IPM Gen 1, IPM Gen 2, externally mounted transmission brake, and parking brake.

Benchmark	IPM Gen 1[9]	IPM Gen 2[9]	Externally Mounted[12]	Parking Brake[13]
Supplier	Stoneridge	Stoneridge	GM	Ford
Weight (kg)	-	0.98	0.99	3.24
Brake Type	Pawl and Ratchet	Pawl and Ratchet	Pawl and Ratchet	Cable and Drum
Sale Cost	$80	$60	$120	$150
Brake Force (N)	250	450	450	18500
Dimensions (cm^3)	-	15 x 12 x 10	21 x 16 x 10	36 x 108 x 95

The differences between IPM Generation 1 and Generation 2 was a result of design modifications that led to simplifications by combining parts and modifications that allowed the IPM to withstand a brake force of 450 N in comparison to 250 N of the prior Generation 1 model. This resulted in the Generation 2 IPM, which is the current design we are being asked to modify, being sold at S20 cheaper and having a break force 200 N greater than Generation 1. The approximate weight and dimensions between the two iterations were relatively similar but we were not given specific values to compare those metrics. When comparing the Generation 2 IPM, the externally mounted mechanism sold by GM is approximately the same weight and uses the same pawl and ratchet mechanism. However, the sales cost is double that of Generation 2 and is slightly larger dimensionally. However, due to increasing competition within the automotive industry, it is necessary to further improve the Generation 2 IPM; improvements with weight and cost in our portion of the IPM will give Stoneridge the ability to increase the cost or weight of other portions of the IPM to achieve better performance.

Design Context

In order to understand what stakeholders were influenced and affected by the work on our project, **Table 3** on pg. 7**.** was created. Stakeholders were categorized either as primary, secondary, or tertiary stakeholders and were further classified into their respective ecosystems in relation to our design.

Table 3. Qualitative categorization of stakeholders, their further classification, and impact on IPM assembly

	Stakeholder	Classification	Role
Primary	Stoneridge	Resource Providers	Sponsor and design creator of IPM
	Major OEM's	Beneficiaries and Customers	Buys IPM
	Our Project Team	Resource Providers	Creates design modifications
	End Users	Beneficiaries and Customers	Have vehicles with IPM assembly
Secondary	University of Michigan	Complimentary Organizations and Allies	Assist with our project team design process
	Manufacturing Companies	Complimentary Organizations and Allies	Produce the parts for the assembly
	Environmentalists	Complimentary Organizations and Allies	Want sustainable materials, less CO_2 emissions
Tertiary	Material Suppliers	Complimentary Organizations and Allies	Send materials to manufacturers
	Insurance Companies	Supporters and Beneficiaries of the Status Quo	Modifying insurance rate on new technology
	Mechanics	Affected or Influential Bystanders	Will need to replace the transmission
	Competitors	Opponents and Problem Makers	Want increased revenue for their brake assembly

I. *Stakeholder Analysis*

Due to the implementation of our project, several groups will be positively impacted by our work. Stoneridge, who is the sponsor, will benefit from this as their IPM product would require cheaper materials, leading to improvement in their profit margins and product quality. If this happens, the major OEMs who purchase and use these IPM's in their vehicles will also benefit from a cheaper or better product to purchase from Stoneridge. In addition, if these IPM's are lighter, the overall weight of the car will decrease, which would either increase fuel efficiency in terms of ICE engines or miles per charge in terms of electric vehicles. Our project team and the University of Michigan could also benefit from this as successful design changes would reflect positively on us as engineers and also on the University for their ability to produce successful engineers. End users could also benefit from this as the vehicles they purchase from the OEMs with these IPM's could have greater fuel efficiency and better miles per change, indirectly saving them money. Although this effect is minor, the high volume of cars with the IPM installed (70,000 Ford Lightnings per year[24]), a 0.5 kg difference in weight can result in a larger net reduction in emissions. Additionally, if our changes end up reducing environmental emissions with more sustainable materials then environmentalists will benefit as these design changes will have a positive impact on the environment.

Implementation of any product will positively and negatively affect stakeholders. Some stakeholders that could be affected negatively could be competitors to Stoneridge and their IPM design. They could directly lose customers in these major OEMs to Stoneridge if the IPM becomes more efficient and cheaper than their respective designs. Material suppliers could also be negatively impacted as they may have to obtain new materials that were not used before, which could be expensive, or Stoneridge could have to find a new material supplier if the design change requires a material that their current supplier can not get. The manufacturing companies could also be negatively affected from this process as they may have to come up with new technologies or processes to create these parts from the new materials, and that could increase costs on their end. Alternatively, they could lose Stoneridge as a customer if they fail to adapt.

II. *Societal and Environmental Impact*

A societal aspect of this problem that drives this work to be done is vehicle safety. We want to make these design changes to the IPM by saving cost and weight, but without compromising safety. Our project sponsor ranks social impact as an important aspect of their priorities. The relevance of this priority will certainly influence the effect our design process has on these social impacts. We are prioritizing the sustainability of the material changes that we plan to make in order to have a net positive impact on the environment. Ideally, we will replace some of the materials in this project with something sustainable. We are looking to change material from steel to plastic; research will be done in finding some plastics that are more environmentally friendly than what is currently used. This can be done by finding materials that are sustainable in use, and can be recycled easily, or are manufactured in an eco-friendly way. We would like to do this in a way that does not increase the cost of the materials used, and also keeps the same or decreases the current weight of the IPM as a whole, as those are two of the most important changes we are trying to make.

III. Intellectual Property

Intellectual property[14] has a very minimal role in our project, as Stoneridge owns the intellectual property related to our project. The type of intellectual property that relates to our project is patents, as Stoneridge has a patent for their specific IPM assembly and the method they used to design it[15].

IV. Ethics

An ethical dilemma that may arise over the course of our project is our project's impact on the environment, along with the end users of the IPM product we are working on. No design changes will be made if there is a possibility that the safety of these individuals is compromised. An ethical dilemma is placed on how we plan to approach model analysis and simulation. We determine the effort we put into the model. Our time and model accuracy will be the biggest dilemma that we face. We will ensure that our analysis is verified with a large confidence interval. From this, we will be confident of the success, or failure of potential designs. Analysis will be verified by presenting our model to experts in the field of failure analysis software.

Our personal ethics align well with the perspective of the sponsor. Both parties have the environment and a successful final product in mind. Our sponsor holds a significant level of influence on our design changes, and the University holds authority based on our progress and commitment level to making these respective changes. Through our team, we have tried to keep a power balance between us in order to maximize what we each take away from this experience. We have all been open to sharing new knowledge we learned along the way. In terms of inclusivity[16], we are working on keeping our sponsor, as well as the University, informed and up to date with all of our decisions and progress throughout the course of this project.

User Requirements and Engineering Specifications

To fully define the problem, a list of user requirements and engineering specifications was developed to confine the problem to streamline our design process. The requirements and specifications were separated into external loading and design parameters.

I. External Loading Requirements and Specifications

External loading requirements and specifications consisted of the external loads and environmental conditions that the IPM will be subject to. The external loading requirements and specifications were summarized in **Table 4**. Quantification of these specifications was provided by Stoneridge.

Table 4: External loading requirements and specifications. All external requirements and specifications were given a weight of 5/5 in terms of importance; if the IPM were to fail under the loading conditions, the entire transmission would need to be replaced, negatively impacting our stakeholders.

User Requirement	Engineering Specifications	Weight (1-5)
Durable	-10 Year Lifespan[9] - 300,000 Cycles between park and neutral[9]	5
Withstands Physical Loading Conditions	- 450 N[1] pull force on slider - V2 Vibration Conditions 9.8-10 g[9], USCAR 2-8 SAE[17] - Screw torques, 25 Nm[9] - SF of 1.7[9]	5
Withstands Temperature Conditions	-T2 Temperature Conditions -40 to 120°C[9], USCAR 2-8 SAE[17] - Minimal Contact Friction from Thermal Expansion	5
Non-corrosive	Non-corrosive grade plastics and coating for steel parts, ASTM D7216-23[18]	5

To justify the user requirements and engineering specifications, the following logic was used:

a) **Requirement, Durability**: To consider the IPM durable, the assembly ***must*** last for 10 years in the transmission environment undergoing 300,000 cycles between park and neutral. Meeting the engineering specifications is top priority for the success of our design (Weight: 5/5). If the assembly does not meet this lifespan, it would lead to the transmission being replaced prematurely.

b) **Requirement, Withstands Physical Loading Conditions:** To consider the IPM as able to withstand the physical loading conditions, the simulations and analysis ***must*** include a 450 N lateral load on the slider (to simulate shifting between brake and park), vibrational conditions up to 10g and a frequency of 1000 Hz. Analysis must meet V2 vibrational loading criteria under the USCAR2-8[17] SAE standard (to simulate vibrations from transmission operation). Analysis must also include screw torques of 25 Nm (to simulate forces induced during assembly) with all analysis completed using a safety factor of 1.7. Meeting these engineering specifications is top priority for the success of our design (Weight: 5/5). If the assembly is not able to withstand any of the loading conditions, it will likely fail prematurely.

c) **Requirement, Withstands Temperature Conditions:** To consider the IPM as able to withstand the temperature conditions, the simulations and analysis ***must*** include temperature conditions ranging from -40 to 120°C. Analysis must meet T2 temperature loading criteria under the USCAR2-8[17] SAE standard (to simulate possible temperature ranges from transmission operation). Additionally, the assembly must have minimal contact friction from thermal expansion to minimize wear (from preliminary research, it was found that some plastics contract where steel expands in a transmission environment). Meeting these engineering specifications is top priority for the success of our design (Weight: 5/5). If the assembly is not able to withstand any of the loading conditions, it will fail prematurely.

d) **Requirement, Non-corrosive:** To consider the IPM as non-corrosive, the materials that we select ***must*** not degrade in the transmission fluid environment. To meet this criteria, the materials and coatings for metals selected must be tested with successful results under the ASTM D7216-23 standard. Meeting these engineering specifications is top priority for the success of our design (Weight: 5/5). It will ensure that the materials we select will be compatible, ensuring longevity of the assembly.

II. Design Requirements and Specifications

To further define the problem, a list of user requirements and engineering specifications was developed to confine the design parameters of our project. Design requirements and specifications were summarized in **Table 5** on pg. 11 and served to define the physical constraints of the IPM and the processes considered for its implementation. Quantification of the specifications was provided by Stoneridge. Requirements of decreasing cost and weight were individually examined to obtain the weight and cost reduction of the assembly.

Table 5: Design requirements and specifications. All external requirements and specifications were given a weight between 1 and 5 in terms of importance determined by their impact on our stakeholders and success of the design.

User Requirement	Engineering Specifications	Weight (1-5)
Decrease Cost	8% decrease of current cost - Current manufacturing cost: $5.12[9]	5
Decrease Weight	50% Weight Decrease of Assembly	3.5
Simple Solution	Design has no alteration with other subsystems of benchmark (Gen 2.) model	3
Quickly Manufacturable	Maintain or decrease number of current manufacturing steps (19) with reduction or decrease in cycle time (30 seconds)	2.5

To justify the user requirements and engineering specifications, the following logic was used:

a) **Requirement, Decrease Cost:** With the overall goal of the project to reduce the cost of the IPM, the final assembly cost ***must*** cost less than the previous model. Currently, the manufacturing cost for the entire IPM is $5.12. We would like to see a decrease in our subsystem's cost by 8%. Meeting the engineering specifications of cost reduction is top priority for the success of our design (Weight: 5/5). If the assembly does not meet this cost reduction, it would lead to lack of implementation of our design by Stoneridge.

b) **Requirement, Decrease Weight:** To consider our design for the IPM to meet the requirement of weight reduction, the subassembly of the IPM should reduce in weight by 50%. Meeting this requirement is a factor in our design, but is not top priority (Weight: 3.5/5). It would be important to reduce the weight of the assembly to improve the efficiency of the car, however, with the weight of the overall assembly weighing 275 g, sacrificing performance or cost is not worth reducing that further

c) **Requirement, Simple Solution:** To consider our design for the IPM to meet the requirement of simplicity, the subassembly of the IPM should not require any alteration with other systems of the benchmark model. Meeting this requirement is a factor in our design, but is not top priority (Weight: 3/5). It would be important to create a design that would not require Stoneridge to alter other parts of the assembly, however, if there is a design choice that could reduce cost or the durability of the IPM, we plan to prioritize those changes first.

d) **Requirement, Quickly Manufacturable:** To consider our design for the IPM to meet the requirement of quickly manufacturable (maintain cycle time of 30 seconds), the subassembly of the IPM should maintain or decrease the required amount of steps to build the assembly (19). Meeting this requirement is a factor in our design, but is not top priority (Weight: 2.5/5). It will be important to create a design that requires fewer manufacturing steps, reducing the net cost of the assembly. This requirement was weighted the lowest on our list due to the cost savings of designing a simple solution would be larger than improving the cycle time and we do not want to sacrifice durability of the IPM.

Initial Design Process

We are currently following the problem-oriented design process as described in the early learning modules. We also briefly thought about a solution-oriented approach, but realized it would not be a good fit for our project. Since our task is to modify an already existing product, we feel that a solution-oriented approach would prevent us from generating unique concepts. The ME 450 design process seems the most useful to us because we want to fully understand transmissions and parking modules before we dive into working on solutions. Additionally we want the ability to iterate after every step if necessary. We expect to iterate many times between concept exploration and verification because intuition won't be a reliable way to determine if a design change will be viable. We have decided to use the ME 450 design process because we have concluded that it will work well for our project and haven't found a better alternative. Additionally the deliverables for this class line up with this design process so it matches our needs there as well. To coincide with the problem-oriented design process, we plan to evaluate each design against each other in a series of concept selection matrices, with the final designs compared in a Pugh chart, supported with an Analytical Hierarchy Process.

Concept Generation

To generate concepts for the IPM, we used two development methods - a brainstorming session to come up with a wide array of possible solutions and a morphological chart to outline the project subfunctions and how well our possible solutions meet the subfunctions. When brainstorming and assessing the solutions in a morphological chart, we used the requirements and specification in Tables 4 and 5 to ensure that our ideas were possible solutions.

I. Brainstorming Session

To ensure that we cover a wide span of the possible solution space, we decided that each team member would create 40 ideas for a total of 160 unique ideas for the entire group. During ideation, team members were encouraged to be creative with their ideas without overanalyzing the requirements and specifications outlined in Tables 4 and 5. At the end of this individual brainstorming period, each team member chooses their top 5 ideas to present to the group.

After individual brainstorming, we came together as a group and assessed the 160 ideas against each other. Each team member shared their list of ideas along with their top choices. After every team member was given time to share, we came up with a consolidated list of 20 ideas discussed in further detail in Appendix A. The top ideas from Appendix A were carried into the next stage of our ideation process: the morphological chart.

II. Morphological Chart

The ideas discussed in Appendix A were divided into 4 categories of possible modifications: material changes, manufacturing process selection, material elimination, and part consolidation. Possible solutions were generated for each category in Table 7. Each category of modification and solution were tied to specific requirements and specifications that they would be impacting. Specific solutions for material changes and manufacturing processes were assessed from experiments following the ASTM D7216-23 standard.

Table 7: Morphological chart organizing 4 categories of modifications with their possible solutions along with the requirements and specifications being considered.

Requirement and Specification	Modifications	Solutions ->				
Cost, Weight, Strength	**Material Change**	PEEK	PTFE	PA66	PA6	POM
Cost, Strength, Cycle Time	**Manufacturing Process**	Compression Molding	Vacuum Casting	Injection Molding	Rotational Molding	Plastic Extrusion
Cost, Weight, Strength	**Material Elimination**	Outer Alignment Plate	Spacer	Inner Alignment Plate		
Cost, Ease of Assembly	**Part Consolidation**	Inner AP + Spacer	Spring Retainers + Alignment Plates			

The morphological chart in Table 7 allowed us to decompose the wide solution scope into smaller subcategories to compare and contrast possible solutions. The chart was used through the concept selection process as a guideline evaluating potential solutions against our desired modifications.

Concept Exploration: Material Change

I. Material Family Selection

To address the weight, cost and strength requirements and specifications, we wanted to assess potential material changes from the current HSLA Steel, grade 45. The material change that we conclude on is the most important change of our design, so a majority of effort for concept exploration was spent on researching and selecting the materials that allow us to best meet our requirements and specifications. Due to the small number of cost effective metals used in transmissions and suggestions from Stoneridge, we decided to look into changing the material of the subassembly to a polymer based material. To select families of polymers that are compatible in the transmission environment, we selected materials that followed standard USCAR 2-8 SAE and ASTM D7216-23 testing procedures. Results of the material family section were confirmed with a polymer material expert, Mohammad Natick of Caresoft Global Inc. The materials selected to investigate were PEEK, PTFE, PA66, PA6 and POM. A polymer selection matrix was developed in Table 8 where the material properties of density, flexural strength, cost, linear thermal expansion coefficient, and heat deflection temperature for each family of polymer were compared. A color coordinating system was developed where green represents the values for best material property relative to rest and red represents the values for the material property farthest from the best value. From this analysis, we confirmed that the material families of PA66, PA6, and POM would be best suited for our project - analysis can be found below.

Table 8: Plastic selection matrix for material families of PA66, PA6, POM, PEEK, and PTFE comparing material properties of density, flexural strength, cost, linear thermal expansion coefficient and the heat deflection temperature. Green represents the best material property value relative to the rest of the materials, where red represents a value far from that. It was found that PA66, PA6, and POM would be best suited for our project.

Requirement and Specification	Relevant Parameters	HSLA Steel, Grade 45[2]	PA66[25]	PA6[25]	POM[25]	PEEK[25]	PTFE[25]
Weight	**Density [g/cm^3] ASTM D792**	7.87	1.15	1.14	1.41	1.31	2.15
Loading Conditions	**Flexural Strength [MPa] ASTM D790**	310	110	100	120	170	15
Cost	**Cost [$/kg]**	2.50 + Coating	1.37	0.77	0.77	110	>1.37
Temperature Conditions	**Linear Thermal Expansion Coefficient [µm/m°C] ASTM E831**	12.4	9	10	13	5	16
Temperature Conditions	**Heat Deflection Temperature[°C] ASTM D648**	**N/A**	90	80	95	150	115

From the comparisons made in Table 8, we will continue material analysis with the families of PA66, PA6, and POM. Although PEEK has significantly better thermal and mechanical properties than the other families, the high cost makes it unusable to satisfy our low cost requirement. PTFE has worse material properties than the remaining 4 materials and with a Flexural Strength much lower than the other families, it will not be able to withstand any of the loading conditions. The remaining material families of PA66, PA6, and POM, have poor thermal properties; however, glass filament significantly improves their thermal performance. With the possibility of improving the thermal performance with glass along with good mechanical properties, cost, and weight properties, the families of PA66, PA6, and POM are well suited to explore for the scope of our project.

a) Density

The density of the material was taken into consideration to address the low weight requirement from Table 6. From the family of materials, PA66 and PA6 had the lowest densities, hence, they would be the best materials to choose for lightweight applications. PTFE has a density value near double PA66 and PA6, making PTFE a bad material family choice.

b) Flexural Strength

Flexural Strength measures the maximum stress that the material can take before yielding. Due to the loads induced by the spring during use of the IPM, the flexural strength of the material is important to extend the life of the part; this will address the loading condition requirement from Table 6. To determine the maximum value for flexural strength that we would need, we will need to perform a probability based failure analysis of the part. More details will be discussed in our future project plans. From the families of materials, PEEK had the highest flexural strength, making it the best material to address our loading requirements. PTFE has a flexural strength much lower than PEEK, making it an impractical material choice for longevity. From this PTFE was eliminated as a potential material family

c) Cost

An important requirement for our design of the IPM is cost reduction. From the material families, a cost estimate was obtained from research, with ranges verified through conversations with a material expert in polymers. To preface, an exact number for the cost could not be determined, but an estimate was used to decide if the material was suitable to use for manufacturing. From the families of materials, PEEK had the highest cost by far and will not satisfy our low cost requirement. Between the remaining materials, both POM and PA6 are the most cost effective materials to use.

d) Linear Thermal Expansion Coefficient

The Linear Thermal Expansion Coefficient represents the change in dimension due to a unit change in temperature. Due to the large range of temperature conditions applied to the part (-40 °C to 120 °C), we are looking for a material with a low thermal expansion coefficient; a low thermal expansion coefficient will reduce internal stress from expansion or compression of the part due to a temperature difference. The largest acceptable value for this coefficient will be determined from Finite Element Analysis of stress generated from thermal expansion of the part. From the families of materials, PEEK has a low coefficient, making it ideal for the conditions of the transmission. The remaining families have coefficients between 2 and 3 times higher than PEEK, hence, using one of the remaining families will generate excess stress due to a temperature difference.

e) Heat Deflection Temperature

Heat deflection temperature represents the temperature at which a specified structure will deflect under a 1.8 MPa load (testing regulations set by the ASTM D648 standard). Due to the high temperature that the part will be exposed to (120 °C), a high heat deflection temperature will raise the temperature that will cause our part to bow and deform due to a smaller load. From the families of materials, PEEK has a large heat deflection temperature compared to the other families. PA6, PA66, and POM have heat deflection temperatures below the maximum operating temperature of 120 °C.

II. *Material Selection: Glass Concentration*

Due to the common practice of using glass-filled (GF) plastics for use in the transmission and our discussion with a material expert, Mohammad Natick of Caresoft Global Inc., we decided to investigate the impact that glass had on the material properties of the selected plastic families of PA66, PA6, and POM from Table 8. A glass-filled polymer selection matrix was developed in Table 9 where the material properties of density, flexural strength, cost, linear thermal expansion coefficient, and heat deflection temperature for each polymer were compared. For the context of the current design, material properties for HSLA Steel, grade 45 were included. A color coordinating system was developed where green represents the values for best material property relative to rest and red represents the values for the material property farthest from the best value. From this analysis, we confirmed that the materials of PA66 13% GF, PA6 13% GF would be best suited for our project - analysis can be found below. Due to cost considerations, glass concentration research was focused on materials with glass concentrations around 13%; glass concentration for polymers ranges between 13% and 45%). PA66 33% GF was considered for completeness.

Table 9: Glass-filled plastic selection matrix for PA66 13% and 33% GF, PA6 13% GF, and POM 15% GF. Materials were compared by their material properties of density, flexural strength, cost, linear thermal expansion coefficient and the heat deflection temperature. HSLA Steel, grade 45 was included as a reference for the current performance. Green represents the best material property value relative to the rest of the materials, where red represents a value far from that. It was found that PA66, PA6, and POM would be best suited for our project.

Requirement and Specification	Relevant Parameters	HSLA Steel, Grade 45[2]	PA66[26] (13% Glass filled)	PA6[27] (13% Glass Filled)	POM[27] (15% Glass Filled)	PA66[26] (33% Glass filled)
Weight	**Density [g/cm³] ASTM D792**	7.87	1.23	1.21	1.50	1.39
Loading Conditions	**Flexural Strength [MPa] ASTM D790**	310	155	175	150	200
Cost	**Cost [$/kg]**	2.50 + Coating	1.37< Cost < 4.52	~$0.60 < than PA66	~$0.60 < than PA66	4.52
Temperature Conditions	**Linear Thermal Expansion Coefficient [µm/m°C] ASTM E831**	12.4	22	22	25	27
Temperature Conditions	**Heat Deflection Temperature[°C] ASTM D648**	-	220	200	150	250

From the comparisons made in Table 9, we will continue material analysis with the materials of PA66 13% GF and PA6 13% GF. Their low weight, cost and adequate temperature and loading conditions make them suitable materials to satisfy our requirements set in Tables 4 and 5. Compared to those two materials, POM 15% GF underperformed in every category, making it an impractical material to use. Although PA66 33% GF will have much better performance for loading conditions than PA66 13% GF and PA6 13% GF, its high cost makes it hard to consider the requirement of reducing cost. We will still keep the material in consideration when performing FEA analysis; if PA66 13% GF and PA6 13% GF do not meet the loading requirements, we will perform analysis with PA66 33% GF so see if it allows us to meet our requirements.

a) Density

The density of the material was taken into consideration to address the low weight requirement from Table 6. From the range of materials, PA66 13% GF and PA6 13% GF had the lowest densities, hence, they would be the best materials to choose for lightweight applications. POM 15% GF and PA66 33% GF had a slightly larger density, making them sub-optimal for meeting our weight specifications. When compared to HSLA Steel, grade 45, the weight of the plastics are substantially lower, making them a good alternative to steel for satisfying our weight requirement.

b) Flexural Strength

Flexural Strength measures the maximum stress that the material can take before yielding. Due to the loads induced by the spring during use of the IPM, the flexural strength of the material is important to extend the life of the part; this will address the loading condition requirement from Table 6. From the selection of materials, PA66 33% GF and PA6 13% GF had the highest flexural strength, making them the best material to address our loading requirements. Both POM 15% GF and PA66 13% GF have a flexural strength much lower than the other two plastics, however, not by a large amount. When compared to HSLA Steel, grade 45, the Flexural Strength of the plastics are substantially lower. Although the plastic materials are rated for a lower strength, the usage of steel is an overcompensation for the small loads applied to the assembly.

c) Cost

An important requirement for our design of the IPM is cost reduction. From the material families, a cost estimate was obtained from research, with ranges verified through conversations with a material expert in polymers. To preface, an exact number for the cost could not be determined, but an estimate was used to decide if the material was suitable to use for manufacturing. From the selection of materials, POM 15% GF and PA6 13% GF had a relatively low estimated cost, making them ideal for meeting our low cost requirement, where PA66 33% GF had a much higher estimated cost. We will be considering POM 15% GF and PA6 13% GF for future analysis due to the low cost, however, if they fail to meet loading conditions, we will consider PA66 13% and 33%

GF. Compared to HSLA steel, grade 45, the plastics will be much cheaper, allowing us to meet the cost requirement.

d) Linear Thermal Expansion Coefficient

The Linear Thermal Expansion Coefficient represents the change in dimension due to a unit change in temperature. Due to the large range of temperature conditions applied to the part (-40 °C to 120 °C), we are looking for a material with a low thermal expansion coefficient; a low thermal expansion coefficient will reduce internal stress from expansion or compression of the part due to a temperature difference. Due to the addition of glass into the material composition, the Linear Thermal Expansion Coefficient of all families increased significantly. Out of the four, PA6 13% GF and PA66 13% GF had the lowest coefficient, making them the most suitable for our purposes. Relative to these values, PA66 33% GF and POM 15% GF do not have that much higher of a value, making them suitable choices as well. Compared to HSLA steel, grade 45, the plastics have a larger Linear Thermal Expansion Coefficient and will generate more internal stress due to a temperature increase. We will need to consider this when evaluating how well the plastics meet the temperature conditions.

e) Heat Deflection Temperature

Heat deflection temperature represents the temperature at which a specified structure will deflect under a 1.8 MPa load (testing regulations set by the ASTM D648 standard). Due to the high temperature that the part will be exposed to (120 °C), a high heat deflection temperature will raise the temperature that will cause our part to bow and deform due to a smaller load. Due to the addition of glass into the material composition, the Heat Deflection Temperature of all families increased significantly. From the materials, PA66 33% GF had the highest Heat Deflection Temperature making it the most suitable for our requirements. POM 15% GF had the lowest value for Heat Deflection Temperature. Its value of 150 °C is close to the maximum operating temperature of our system, thus, we will need to be careful with our analysis if we select this material. Heat Deflection Temperature is only a material property prescribed to plastics, so no comparison to HSLA Steel, grade 45 could be made.

Concept Exploration: Manufacturing Process

Due to the decision to change the materials of specific parts of the assembly from steel to plastic, it was necessary to determine which manufacturing process would best be utilized for the plastic parts. To determine this, manufacturing processes ordered by economic batch size and section thickness were considered which are shown in Figure 4 and Figure 5. We determined that for plastics, which fall under the category of polymer shaping, injection molding, compression molding, and rotational molding would fulfill batch sizes greater than 10,000 and a section thickness greater than 0.5 mm which are metrics that best fit our project. Additionally we considered the manufacturing processes of vacuum casting and plastic extrusion as well due to being processes for manufacturing plastic parts although they were not included in Figures 4 and 5.

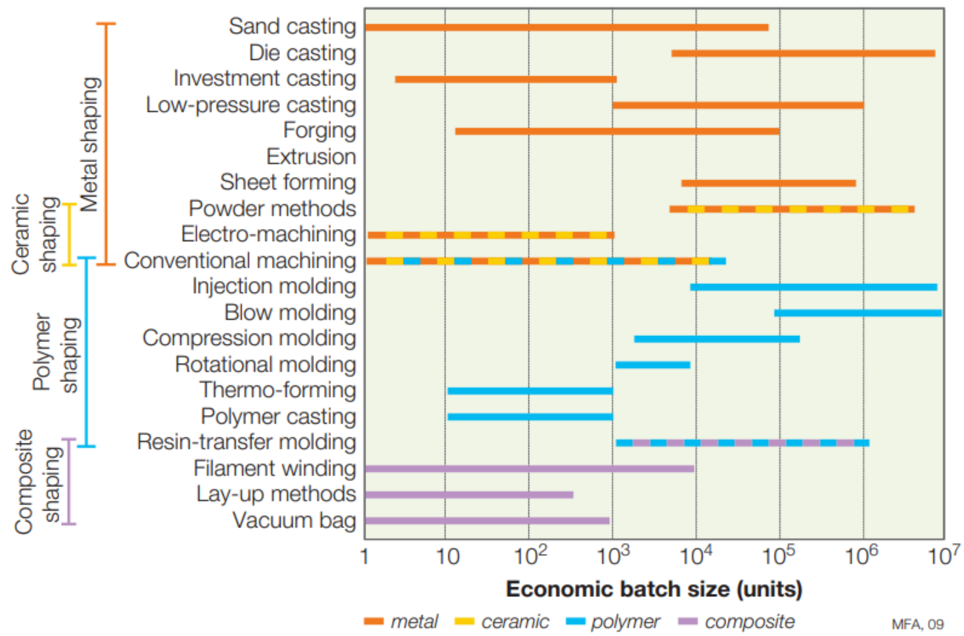


Figure 4. Economic batch sizes in units versus manufacturing processes[42].

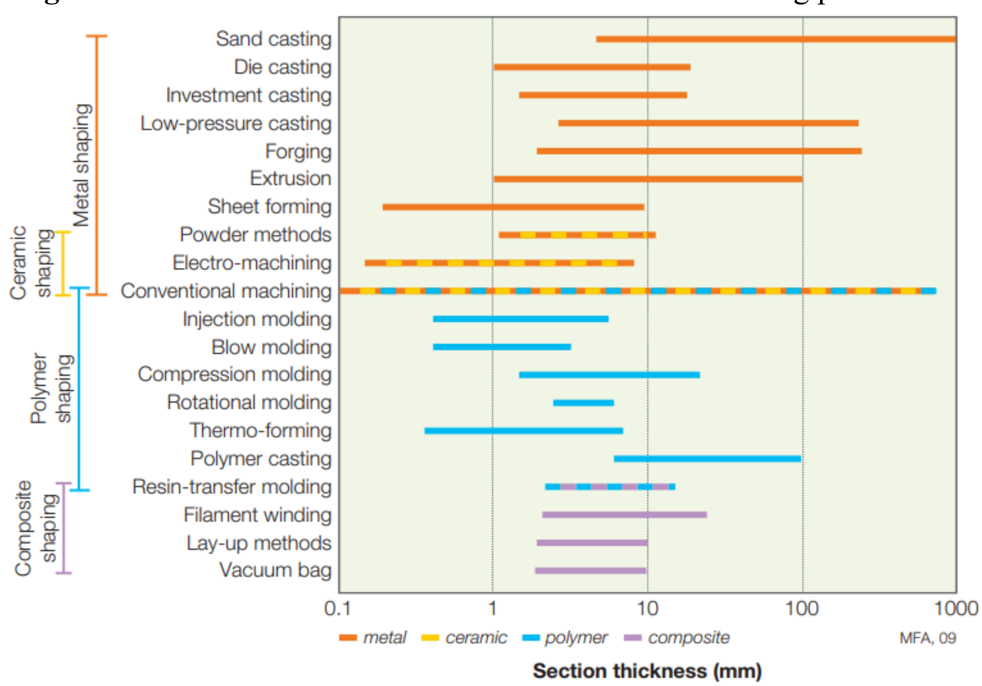


Figure 5. Section thickness in mm versus manufacturing processes[42].

The tradeoffs between the manufacturing processes of injection molding, compression molding, vacuum casting, rotational molding, and plastic extrusion is outlined in Table 10. Resolution, cycle time, tooling costs, part complexity, and production volume were considered when comparing different manufacturing processes. We want high resolutions that would allow producing thin parts, in our case our thinnest part is 1.5 mm. We also want a lower cycle time since it allows for greater efficiency. Tooling costs of each process were also considered because we want to be able to satisfy our requirements of decreasing the cost of the assembly and tooling costs would factor into this calculation. Due to our parts containing complex geometries, it is necessary to have a manufacturing process that can create these geometries. Additionally, due to the IPM being mass produced it is necessary for the manufacturing process selected to be able to quickly and efficiently produce multiple parts on a large scale.

We determined that injection molding and compression molding would be the most favorable manufacturing processes due to both processes having the fine resolutions of being able to produce thin parts of 1 mm with high tolerances, low cycle times of approximately 1-6 minutes, adequate part complexity for our complex geometries, and a high production volume (greater than 10^3 units). The tradeoffs between these processes is that injection molding has a slightly lower cycle time than compression molding while compression molding has a slightly lower tooling cost than injection molding. A more exhaustive explanation of the manufacturing selection process is provided below.

Table 10: The manufacturing processes of injection molding, compression molding, vacuum casting, rotational molding, and plastic extrusion were compared using the tradeoffs between resolution, cycle time, tooling costs, part complexity, and production volume. The most favorable features of a process were highlighted in green, whereas features that were not compatible with our design were highlighted in red. Features that were completed better by another process were highlighted in yellow. It was determined that injection molding and compression molding would be best suited for our project.

	Injection Molding[28]	**Compression Molding[29]**	**Vacuum Casting[30]**	**Rotational Molding[31]**	**Plastic Extrusion[32]**
Resolution	Tolerances: +/-0.1 to 0.7mm, minimum wall thickness: 1-5mm	Minimum wall thickness: 0.5mm	Limited in detail, loose tolerances (0.15%)	Not as precise as injection molding	-
Cycle Time	< compression molding	1-6 minutes	1-6 minutes	1-3 Hours	-
Tooling Costs	Mold cost vary: $1000-5000	< injection molding	Low	-	-
Part Complexity	Can create molds for complex parts	Can create molds for complex parts	Works best with simple geometries and features	Creates hollow products, better for larger parts	Limited to parts with consistent cross-sections
Production Volume	Good for large batches (10^3-10^7)	Good for large batches (10^3-10^5)	Limited to small batches (1-100 units)	-	-

a) **Plastic Extrusion:** Plastic extrusion was eliminated as a manufacturing process early on in the decision process due to its part complexity being limited to parts with consistent cross-sections. Many of our parts have various cross-section thicknesses so this manufacturing process would not be optimal.

b) **Rotational Molding:** Rotational molding has two main drawbacks of having a cycle time of 1-3 hours which is significantly higher, especially in comparison to the 1-6 minutes cycle time for injection molding, compression molding, and vacuum casting, and being better for larger processes that are hollow. Our parts are relatively small so we eliminated rotational molding.

c) **Vacuum Casting:** The main benefits of vacuum casting are that it has a moderately short cycle time of 1-6 minutes and has low tooling costs. However, it has loose tolerances and works best with simple geometries and features. Since our parts have relatively complex geometries and it is important that the selected manufacturing process has tight tolerances, vacuum casting was eliminated as a potential manufacturing method.

d) **Compression Molding:** Compression molding can produce parts with small wall thicknesses, has a moderately short cycle time of 1-6 minutes, has a lower mold cost than injection molding and can create molds for complex parts. Compression molding is a favorable option because it meets the necessary manufacturing requirements for our specific parts.

e) **Injection Molding:** Injection molding can produce parts with small wall thicknesses and tight tolerances, has a short cycle time of less than 1-6 minutes, and can create molds for complex parts. This makes injection molding a favorable option because it meets the necessary manufacturing requirements for our specific parts. However, the mold cost of injection molding is typically greater than that of compression molding. Mold costs for injection molding can vary from $1,000-$5,000 for smaller parts depending on the complexity of the given part. To get a specific value we need to discuss this with our sponsor. However, due to having a smaller cycle time it may be more favorable than compression molding when considering the long term output.

Concept Exploration: Material Elimination

To address the requirement of decreasing the weight of the current assembly, material elimination was considered for both steel and plastic parts.

I. *Material Elimination for Steel Parts*

Material elimination for the IAP, OAP, and Spacer was considered and are shown in Figures 6, 7, and 8. The area in which material was eliminated is highlighted in blue and this area was determined through a qualitative inspection. However, to verify that the necessary environmental load conditions are still met, each modification will have to be evaluated through FEA analysis and physical testing. This material elimination shown in the Figures is specific to steel parts. It is more difficult to use this method for plastic parts due to needing a uniform thickness to ensure part strength and adequate molding during the manufacturing process.

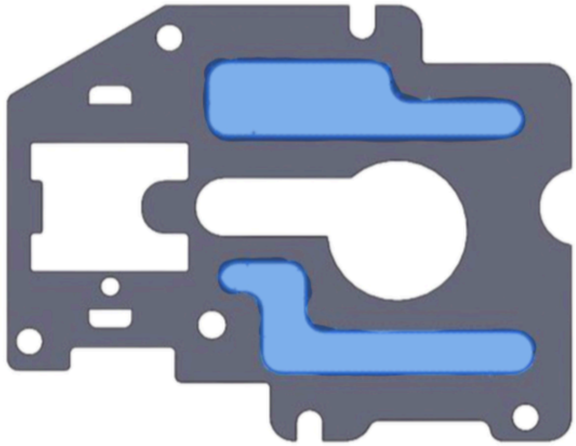


Figure 6. Material elimination for the IAP. This would lead to an approximate 21% decrease in the volume of the part.

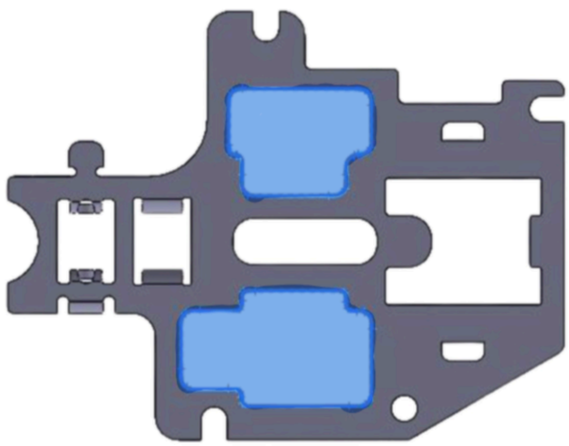


Figure 7. Material elimination for the OAP. This would lead to an approximate 25% decrease in the volume of the part.

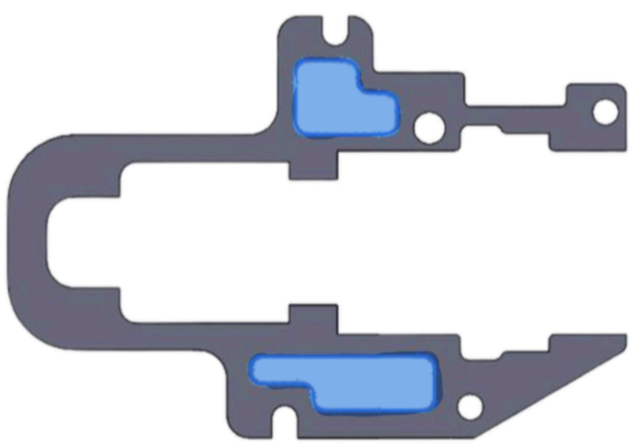


Figure 8. Material elimination for the Spacer. This would lead to an approximate 17% decrease in the volume of the part.

The OAP has the greatest volume reduction, with a reduction of 25%. However, the best case scenario would be to have a combination of material elimination in all these parts to have the maximal weight reduction. It is also important to consider that by redesigning existing parts, it may lead to changes in the manufacturing process, such as needing a new mold for the modified part, which can lead to cost increases that we need to consider.

II. Material Elimination for Plastic Parts via Ribbing

Material elimination for plastic parts was considered through adding ribbing. Ribbing is a method that provides additional support and reinforcement to the body of a part[33]. This is a beneficial method for plastic parts because it allows for material elimination without decreasing the strength of a part. Ribbing will reduce the manufacturing time of the part by requiring less material to flow into the mold, and will reduce the cooling time for the plastic, reducing the cycle time.

There is a constraint that to be able to add ribbing to a part, the thickness must be greater than 3 mm[34]. Two initial ribbing designs for the Spacer are shown in Figures 9 and 10. This is the only part that currently has a thickness that is greater than 3 mm, which satisfies the condition to add ribbing to the part.

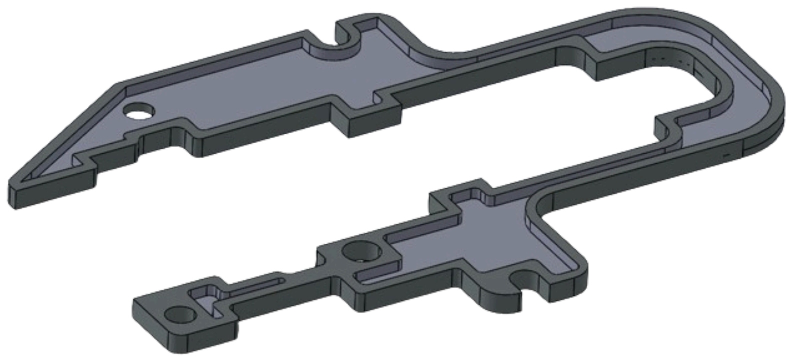


Figure 9. One concept generation for ribbing of Spacer. The inner thickness was decreased and ribbing was added around the perimeter of the part. Volume of the part was reduced by 26.1% from the original Spacer.

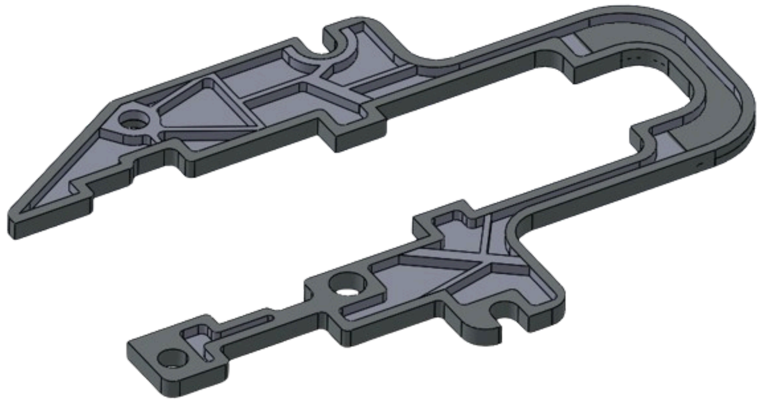


Figure 10. One concept generation for ribbing of Spacer. The inner thickness was decreased and ribbing was added to the perimeter as well as additional ribbing in sections that may face loading conditions from the spring.Volume of the part was reduced by 26.7% from the original Spacer.

The current concepts for the ribbing would have to be validated through FEA analysis to ensure that the Spacer can withstand the environmental load conditions. Volume of the parts were reduced by a considerable 26.1% (Figure 9) and 26.7% (Figure 10) from the original Spacer.

Concept Exploration: Part Consolidation

I. Part Consolidation

Another method that we can use to decrease both the weight and the cost of our assembly is part consolidation. This involves integrating one or more parts to reduce the amount of total components, while still maintaining the assembly's integrity and function.

One way that we approached this was through introducing a material change in the OAP and Spacer from steel to plastic. These parts would be injected molded together during the manufacturing process, which will also reduce the amount of steps in the assembly process. We initially planned on consolidating the spring retainer in addition, but after further exploration it was determined that this would create issues with the manufacturing process and would not be feasible. Since the slider needs to remain steel it could not be part of this sub-assembly, which also limits this from involving the IAP as well. This initial design is shown in Figure 11. This concept could also apply towards combining the IAP and Spacer, which is present in Figure 12.

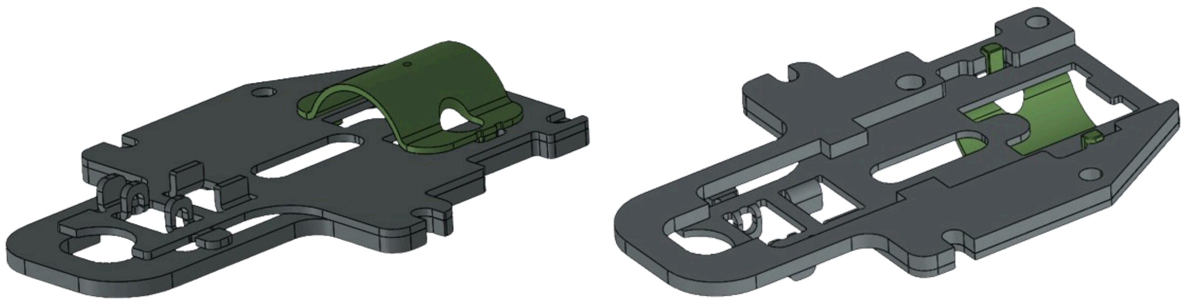


Figure 11. Part consolidation of the OAP and Spacer.

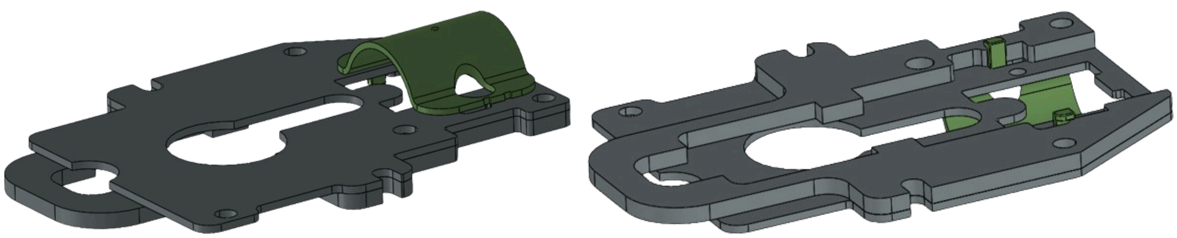


Figure 12. Part consolidation of the IAP and Spacer.

II. Part Consolidation with Ribbing

In consideration of the previous designs, it was determined that the addition of ribbing with these consolidated parts would be an effective way to further reduce the weight of the sub-assembly. By finding the points of the design that undergo the most stress, we can determine the best areas to place the ribbing, and since the thickness of this part would be greater than 3 mm, ribbing becomes an effective method for this. It is also important to ensure that there is a uniform wall thickness in order to ensure structural integrity. Figures 14 and 15 demonstrate this initial design for the IAP and Spacer, along with the OAP and Spacer with the addition of ribbing. This ribbing reduced the volume of both the OAP-Spacer assembly (Figure 11, 12) and the IAP-Spacer assembly (Figure 13, 14) by 20.4% and 13.1% respectively. Ribs will be placed in locations of high stress obtained from our FEA models.

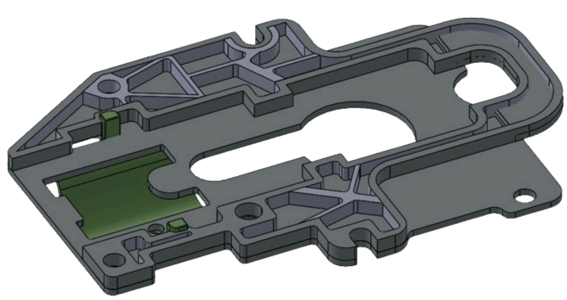


Figure 13. Molded IAP and Spacer with ribbing design. Volume was reduced from the assembly in Figure 13, 14 by 13.1%.

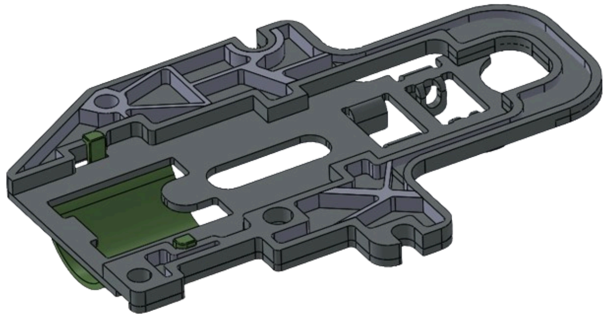


Figure 14. Molded OAP and Spacer with ribbing design. Volume was reduced from the assembly in Figure 11, 12 by 20.4%.

Concept Selection Process

We used two methods to determine the alpha concept to move forward with: a Pugh Chart and an Analytical Hierarchy Process. Our goal from using multiple selection processes was to get a more comprehensive understanding of how well each concept meets the requirements.

I. Pugh Chart

For our pugh chart, shown in Table 11 below, we compared each of our concepts to the current Gen. 2 IPM design. We narrowed our remaining 20 concepts down to 5 based on the design criteria as well as their likelihood of fulfilling the external loading requirements. The resulting scores show that concept 5 (change material of the Spacer and alignment plates to PA6 13% GF, consolidate and rib the Spacer and OAP) is the best suited to meet our requirements and specifications. Scores in criteria were unanimously agreed upon by all group members with sponsor input to confirm results.

Table 11: Pugh chart comparing our top 5 concepts to the current Gen. 2 IPM on a scale of 0 (no improvement) to 3 (vast improvement). Note that we only only compare design criteria here because we believe all 5 concepts are capable of passing the external loading criteria.

Concepts	Concept #	Total	**Criteria**			
			Decrease Cost Specification(s)	*Decrease Weight* Specification(s)	*Simple Solution* Specification(s)	*Quickly Manufacturable* Specification(s)
Current Gen. 2 IPM	#0	0	**Datum**			
Steel, Material Elimination of OAP, IAP and Slider	#1	3	0	1	2	0
Spacer from steel to POM-G15 with ribbing	#2	5	1	1	3	0
Change steel materials to PA6-GF13 (not slider)	#3	7	2	2	3	0
PA6-GF13, Consolidate OAP, Spacer. and Spring Retainer	#4	9	3	2	2	2
PA6-GF13, Consolidate OAP, Spacer. and Spring Retainer with Ribbing	#5	11	3	3	2	3

II. Analytical Hierarchy Process[35]

The main difference between this analytical hierarchy process (AHP) and our Pugh Chart is that the AHP compares the new concepts in reference to each other while the Pugh Chart compares the new concepts to the current product. From the AHP, the optimal design to meet the requirements of our project was concept 5 (change material of the Spacer and alignment plates to PA6 13% GF, consolidate and rib the Spacer and OAP). This agrees with and validates our conclusions from the Pugh Chart in Table 11. The first step in the AHP process is to create a criteria comparison matrix, shown below in Table 12. The purpose of this matrix is to generate a weighting for each requirement (the higher the weighting value, the more important the requirement). Weights were unanimously agreed upon by all group members with sponsor input to confirm results.

Table 12: Normalized criteria comparison matrix. In this matrix, the requirements are compared to each other based on importance in order to generate criteria weight values for each requirement

	Normalized Criteria Comparison Matrix [NormC]								
	Durability Specification(s)	Physical Loading Specification(s)	Temperature Condition Specification(s)	Non-corrosive Specification(s)	Decrease Cost Specification(s	Decrease Weight Specification(s)	Simple Solution Specification(s)	Quickly Manufacturable Specification(s)	Criteria Weight {W}
Durability Specification(s)	0.17	0.17	0.17	0.17	0.17	0.18	0.15	0.15	0.16
Physical Loading Specification(s)	0.17	0.17	0.17	0.17	0.17	0.18	0.15	0.15	0.16
Temperature Condition Specification(s)	0.17	0.17	0.17	0.17	0.17	0.18	0.15	0.15	0.16
Non-corrosive Specification(s)	0.17	0.17	0.17	0.17	0.17	0.18	0.15	0.15	0.16
Decrease Cost Specification(s	0.17	0.17	0.17	0.17	0.17	0.18	0.15	0.15	0.16
Decrease Weight Specification(s)	0.06	0.06	0.06	0.06	0.06	0.06	0.15	0.15	0.08
Simple Solution Specification(s)	0.06	0.06	0.06	0.06	0.06	0.02	0.05	0.05	0.05
Quickly Manufacturable Specification(s)	0.06	0.06	0.06	0.06	0.06	0.02	0.05	0.05	0.05
Sum	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00

The next step is to create concept comparison matrices like the ones shown below in Table 13.. Each of these matrices compares the concepts to each other based on a single requirement. The resulting design alternative priority value ranks the concepts on how well they meet the requirement.

Table 13: Normalized concept comparison matrices for the cost and weight reduction requirements. These matrices compare the concepts against each other based on a single requirement. Matrices like the ones shown were created for every requirement.

	Cost Reduction					Design Alternative Priorities {PI}
	Normalized Comparison Matrix [NormC]					
	#1	#2	#3	#4	#5	
#1	0.08	0.03	0.04	0.11	0.11	0.07
#2	0.23	0.10	0.04	0.11	0.11	0.12
#3	0.23	0.29	0.13	0.11	0.11	0.17
#4	0.23	0.29	0.39	0.33	0.33	0.32
#5	0.23	0.29	0.39	0.33	0.33	0.32
Sum	1.00	1.00	1.00	1.00	1.00	1.00

	Weight Reduction					Design Alternative Priorities {PI}
	Normalized Comparison Matrix [NormC]					
	#1	#2	#3	#4	#5	
#1	0.09	0.09	0.06	0.06	0.14	0.09
#2	0.09	0.09	0.06	0.06	0.14	0.09
#3	0.27	0.27	0.18	0.18	0.14	0.21
#4	0.27	0.27	0.18	0.18	0.14	0.21
#5	0.27	0.27	0.53	0.53	0.43	0.41
Sum	1.00	1.00	1.00	1.00	1.00	1.00

After getting the criteria weight values and design alternative priority values, they can be multiplied to get a resulting matrix, shown below in Table 14. The columns are summed to get a final ranking for the concepts. As seen in the figure, the AHP produces similar results to the Pugh Chart with concept 5 having the highest score.

Table 14: Resulting comparison matrix between criteria and concepts. For each row, the criteria weight is multiplied by the design alternative priority value of each concept for that criteria. The values of each column are then summed to get the final "Alternative Value" score.

	Concepts				
	Steel, Material Elimination of OAP, IAP and Slider	Spacer from steel to POM-G15 with ribbing	Change steel materials to PA6-GF13 (not slider)	PA6-GF13, Consolidate OAP, Spacer. and Spring Retainer	PA6-GF13, Consolidate OAP, Spacer. and Spring Retainer with Ribbing
Selection Criteria	#1	#2	#3	#4	#5
Durability Specification(s)	0.0329	0.0329	0.0329	0.0329	0.0329
Physical Loading Specification(s)	0.0329	0.0329	0.0329	0.0329	0.0329
Temperature Condition Specification(s)	0.0329	0.0329	0.0329	0.0329	0.0329
Non-corrosive Specification(s)	0.0329	0.0329	0.0329	0.0329	0.0329
Decrease Cost Specification(s	0.0123	0.0194	0.0287	0.0520	0.0520
Decrease Weight Specification(s)	0.0070	0.0070	0.0165	0.0165	0.0324
Simple Solution Specification(s)	0.0055	0.0165	0.0165	0.0055	0.0055
Quickly Manufacturable Specification(s)	0.0055	0.0055	0.0055	0.0165	0.0165
Alternative Value	0.1617	0.1799	0.1987	0.2220	0.2378

Initial Selected Design Concept

After completing the Pugh Chart and the Analytical Hierarchy Process, we came to the conclusion that our most effective design involved both part consolidation with ribbing and material elimination. We concluded that we were going to combine the OAP and Spacer with the addition of ribbing, this will reduce the volume by 20.4% from the original assembly. In addition, we are going to be implementing material elimination in the IAP, which will provide a volume reduction of 21%. These parts would have a material change from Grade 45 HSLA steel to PA6 13% GF. To make the parts, we decided to use injection molding as our manufacturing process. As previously stated, both of these changes brought forth a reduction in both cost and weight. This design was specifically chosen as it has the highest scores on both of our concept selection criteria. These designs were shown below in Figures 15 and 16. After conversations about the manufacturing process with Stoneridge, the spring retainer remained separate from the OAP and Spacer assembly. Including it will inhibit assembly and require unnecessary tooling changes.

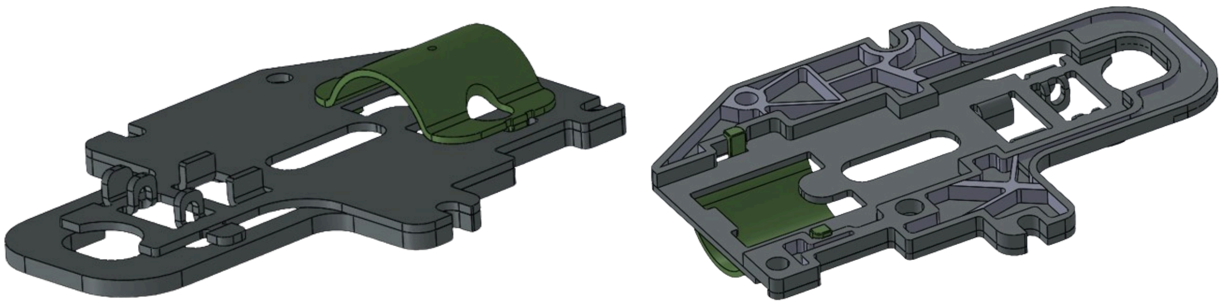


Figure 15. Molded OAP and Spacer with ribbing design. Since the manufacturing process will be injection molding, analysis will need to be completed to ensure that there is uniform thickness throughout to speed up the cooling process of the parts before ejection. Ribs will be placed in locations of high stress obtained from our FEA models.

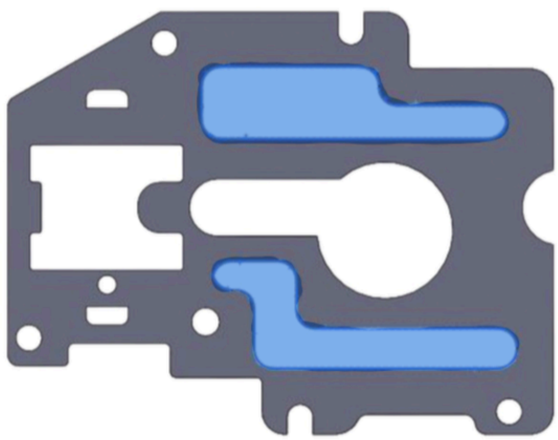


Figure 16. Material elimination for the IAP. Since the manufacturing process will be injection molding, analysis will need to be completed to ensure that there is uniform thickness throughout to speed up the cooling process of the parts before ejection.

Engineering Analysis

To verify that our preliminary design will satisfy the requirements and specifications in Tables 4 and 5, analysis was performed on the cost of manufacturing our subassembly, the weight reduction caused by the material change to plastic, and on the environmental loading conditions of the model.

I. *Cost Reduction*

Our sponsor has requested that our new design reduce the cost per unit by at least 8%. Since our project is optimizing an already existing product, a full cost analysis can be minimized by comparing just the costs that will be changing from the current design to our new design. Figure 17 below shows the current and new costs for each part in the assembly we are changing as well as the overall cost reduction per unit.

		CURRENT	NEW
VARIABLE COSTS			
		Cost Per Unit	Cost Per Unit
MBOM	Inner Alignment Plate	$0.820	$0.157
	Outer Alignment Plate	$0.800	$0.154
	Spacer	$1.320	$0.146
	TOTAL	$2.940	$0.457
GRAND TOTALS			
Total Cost/Unit (for listed items)		$2.940	$0.457
Total Current Cost/Unit	$5.12		
Cost Reduction Per Unit	$2.483		
% Cost Reduction/Unit	**0.4850**		

Figure 17. Overview of the estimated cost differences between the current IPM design and our new proposed design as well as the percentage cost reduction from our design changes. Note that this is based on a production volume of 500,000.

The only costs that will differ for our new design will be the cost to manufacture and the cost of assembly. However, we realized the difference in assembly cost would be negligible compared to the manufacturing costs so it is ignored.

Our calculations of the cost to manufacture the parts for our new design are based on methods from the textbook, *Product Design for Manufacturing and Assembly* [46]. We were conservative in our calculations so that we could be confident that our new design will meet our sponsor's requirement. With the 48% reduction in cost, we exceeded the 8% reduction set by Stoneridge. A detailed overview of these calculations can be found in Appendix C.

II. Weight Reduction

An estimated weight of the new final design for the modified parts: the OAP, Spacer, and IAP are shown in Table 15. This is a conservative estimate using a material change of steel to PA66 33% glass filled.

Table 15. Weight of previous Generation 2 IPM design compared to our current design.

	Generation 2 IPM (g)	**Current Final IPM Design (g)**
Outer Alignment Plate	48.43	8.576
Spacer	66.52	11.779
Inner Alignment Plate	58.65	10.39
Slider	67.58	67.58
	Total Weight Savings:	142.855

Therefore, it can be seen that the modifications lead to a total weight savings of 142.855 g, or a 59.23% decrease in the total weight which meets our design requirement and specification of decreasing the weight of the assembly by 50%.

III. Preliminary FEA Results

Due to the environmental conditions of the transmission environment, external loading of the IPM is complicated to model with first principles analysis. To determine the effect of the loading conditions on the IPM, a preliminary FEA model was developed simulating the forces generated from actuation of the IPM, and temperature changes between -40 and 120 °C. The purpose of the FEA is to understand deflection and the magnitude of stress generated in the part. The deflection will aid in placement and orientation of ribbing when iterating on the final design, and the stress will allow us to perform probability based failure analysis to determine the number of cycles until part failure. To ensure that the meshing, constraints and loads were performed correctly, Appendix B summarizes justification for the meshing used, and the steps taken to verify that the model.

For PA6, 13% GF the results from the actuation force, temperature change, and a combination of both were summarized in Table 16. From this, we can see that the stress and displacement from the change in temperature is 2 orders of magnitude higher than the spring force, hence, it will have a much greater effect on the longevity than the part. From this analysis, we will be considering materials that have better thermal material properties rather than mechanical. More discussion on this can be found in the Final Design section.

Table 16: Maximum magnitude of displacement (mm) and stress (MPa) due to the spring force, temperature change, and spring force and temperature change on PA6, 13% GF. From the simulation, stress and displacement for the temperature change were 2 orders of magnitude higher than the spring force. We will be considering materials with better performing thermal properties in our final design due to this analysis.

	Spring Force	**Temperature Change**	**Combined Spring Force and Temperature Change**
Max Stress [MPa]	4.6	536	542
Max Displacement [mm]	0.002	0.54	0.54

A. Actuation Force

To simulate the actuation force of the IPM on the part, the spring force was modeled as a distributed load modeling both springs as shown in Figure 18 below. The spring force was evaluated with the maximum actuation distance of the IPM, its uncertainty, the stiffness constant of the springs, and a safety factor of 1.7 as set in our requirements and specifications by Stoneridge.

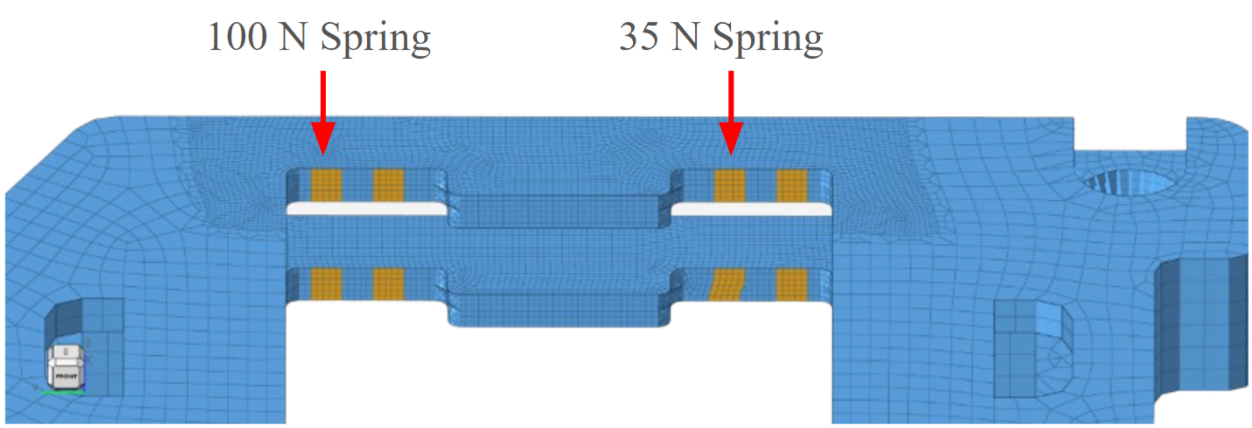


Figure 18. Image depicting the distributed load of the 2 springs in the IPM assembly. The spring forces were generated using the maximum actuation distance of the IPM, the stiffness constant of each spring and the safety factor of 1.7. To get the width of the distributed load, the thickness of the springs were measured.

The actuation force was simulated for HSLA Steel, grade 45 as a baseline and our selected material PA6 13% GF. The simulation results for stress and deflection of HSLA Steel, grade 45 were depicted in Figures 19 and 20 and for PA6 13% GF in Figures 21 and 22. Results were summarized for comparison in Table 17. Results were only shown for 2D elements with reasoning of this choice explained in Appendix B.

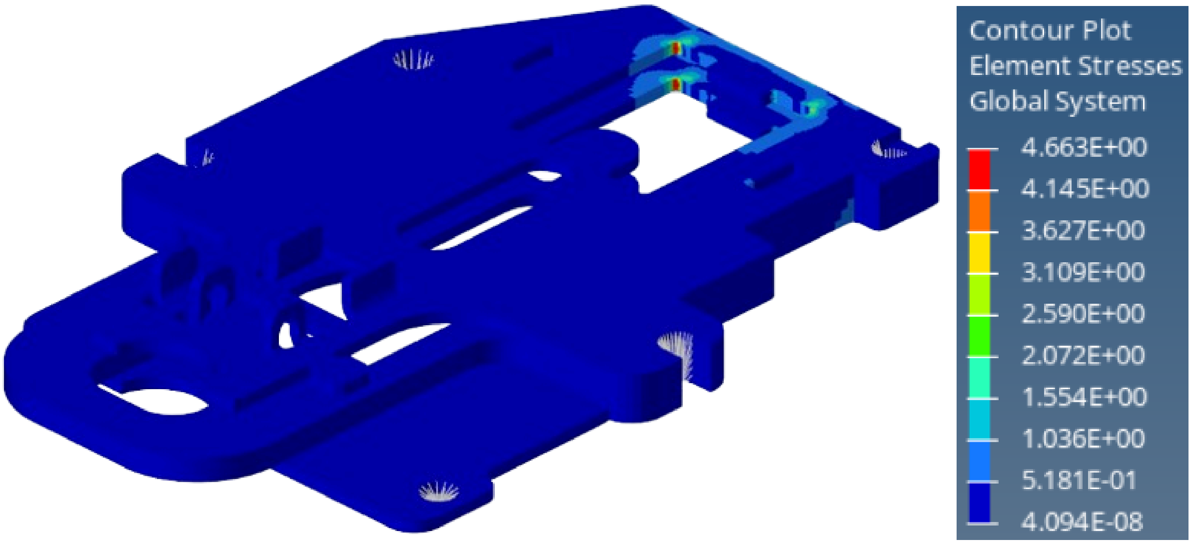


Figure 19. Stress (magnitude in MPa) contour plot for HSLA Steel, grade 45 for the spring load. The maximum magnitude of stress was 4.7 MPa, concentrated at the rounded corner of the model.

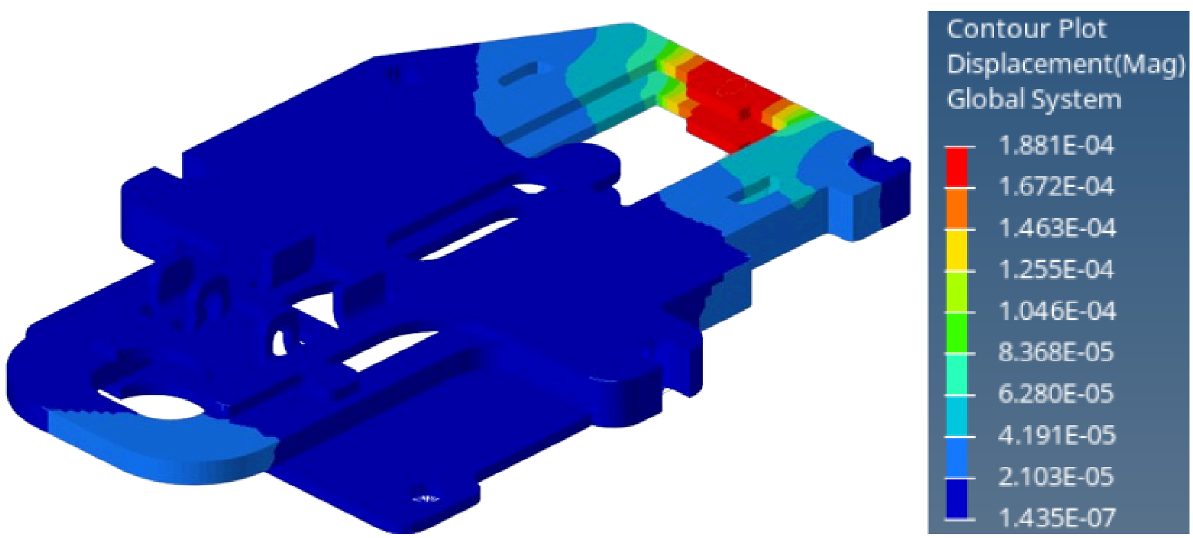


Figure 20. Displacement (magnitude in mm) contour plot for HSLA Steel, grade 45 for the spring load. The maximum magnitude of displacement was 0.0002, concentrated in the middle of the spring loads.

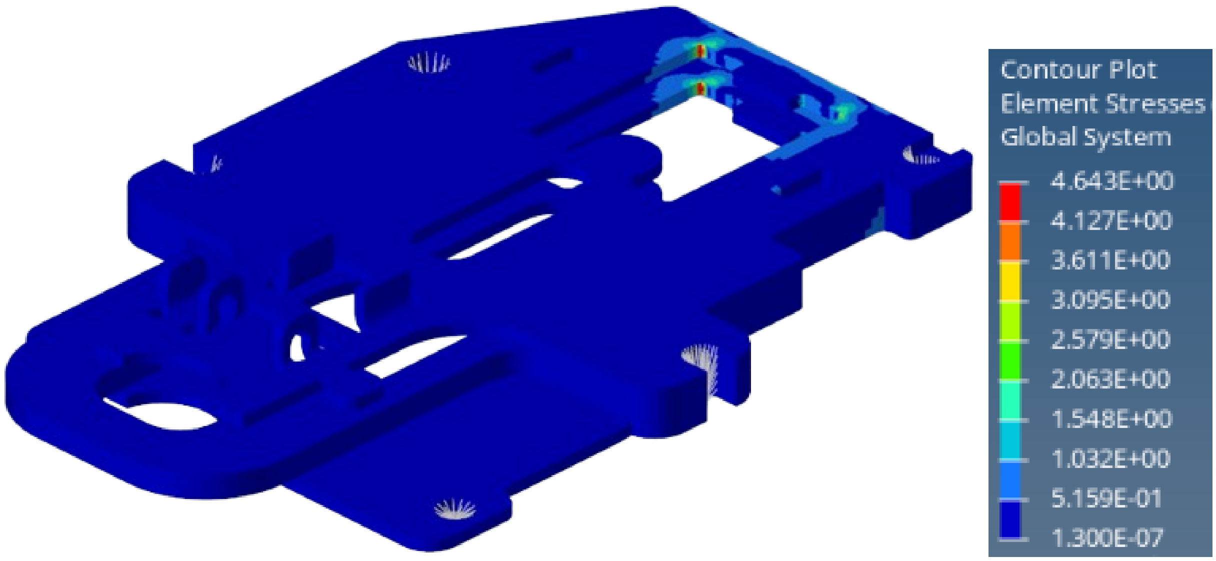


Figure 21. Stress (magnitude in MPa) contour plot for PA6 13% GF for the spring load. The maximum magnitude of stress was 4.6 MPa, concentrated at the rounded corner of the model.

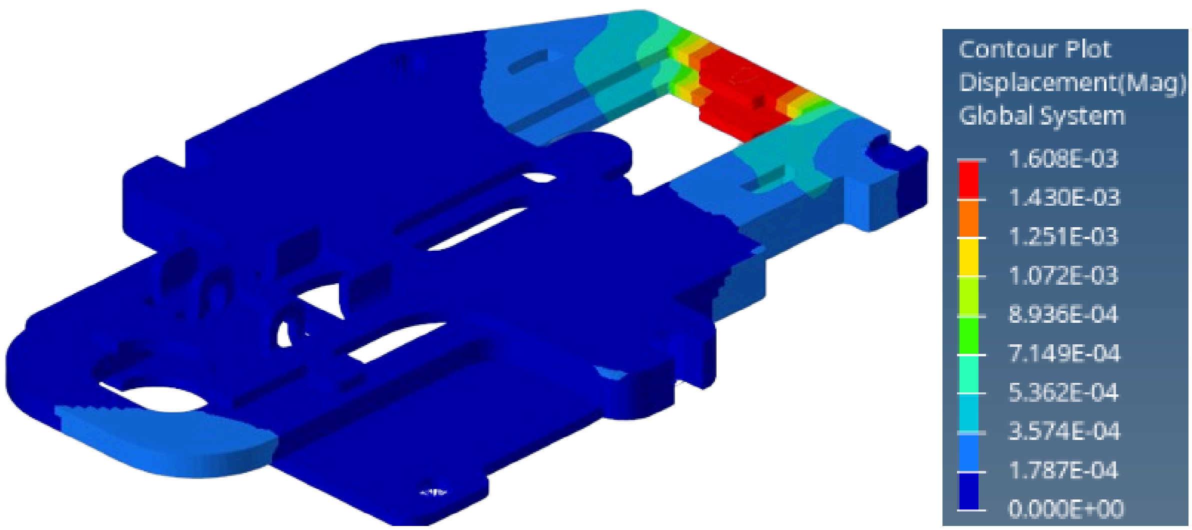


Figure 22. Displacement (magnitude in mm) contour plot for PA6 13% GF for the spring load. The maximum magnitude of displacement was 0.002, concentrated in the middle of the spring loads.

The values of stress and displacement for the 2 models with the spring load were summarized below in Table 17. It was found that changing the material from HSLA Steel, grade 45 to PA6 13% GF had no change in stress in the component, however, the displacement increased by an order of magnitude.

Table 17: Maximum magnitude of displacement (mm) and stress (MPa) due to the spring force on HSLA Steel, Grade 45 and PA6, 13% GF. From the simulation, the stress for both materials remained the same, however, the displacement of the PA6 13% GF was an order of magnitude greater than steel.

	HSLA Steel, Grade 45	**PA6 13% GF**
Max Stress [MPa]	4.66	4.64
Max Displacement [mm]	0.0016	0.018

B. Temperature Changes

To simulate the stress and displacement due to a temperature change of the environment of the IPM, a global temperature change of 160 °C was applied to the model. This temperature change simulates the temperature change between -40 and 120 °C. The stress generated due to a temperature change does not depend on the initial or final temperature, only the difference between those 2 values. It is important to note that the model being simulated as one solid piece led to inaccurate representations of stress and displacement in the part. Although the values for stress and displacement are not accurate, the behavior of the model is. Discussion of the limitations in this analysis and how we plan to address it can be found in our future project plans.

The temperature change was simulated for HSLA Steel, grade 45 as a baseline and our selected material PA6 13% GF. The simulation results for stress and deflection of HSLA Steel, grade 45 are depicted in Figures 23 and 24 and for PA6 13% GF in Figures 25 and 26. Results are summarized for comparison in Table 18. Results are only shown for 2D elements with reasoning of this choice explained in Appendix B.

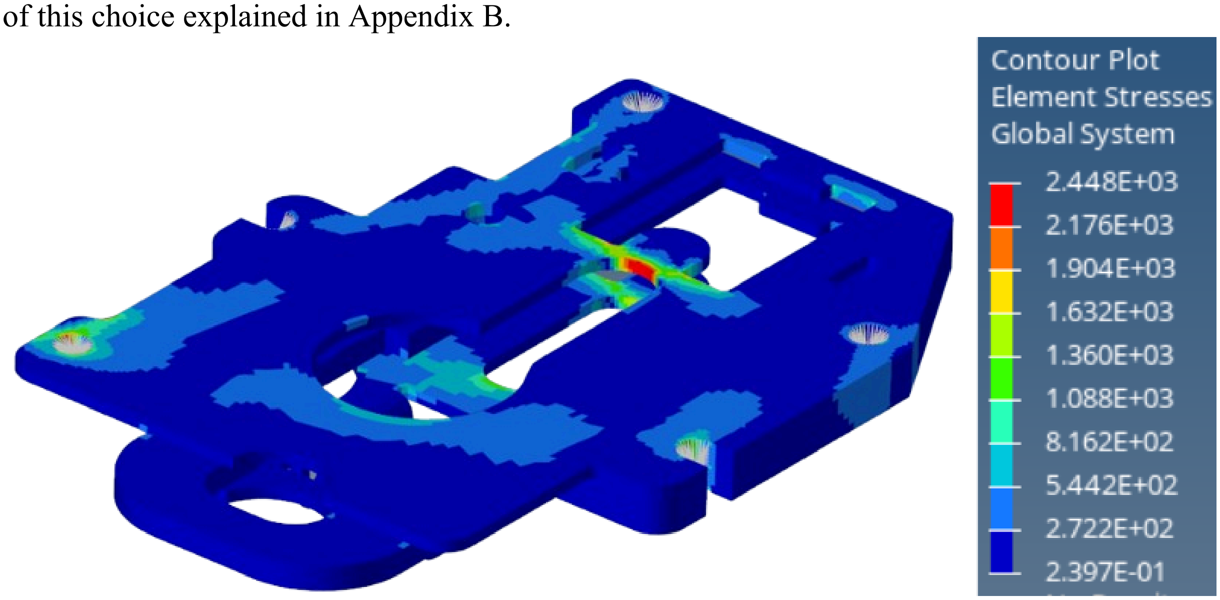


Figure 23. Stress (magnitude in MPa) contour plot for HSLA Steel, grade 45 for temperature change. The maximum magnitude of stress was 2450 MPa, concentrated at the rounded slot.

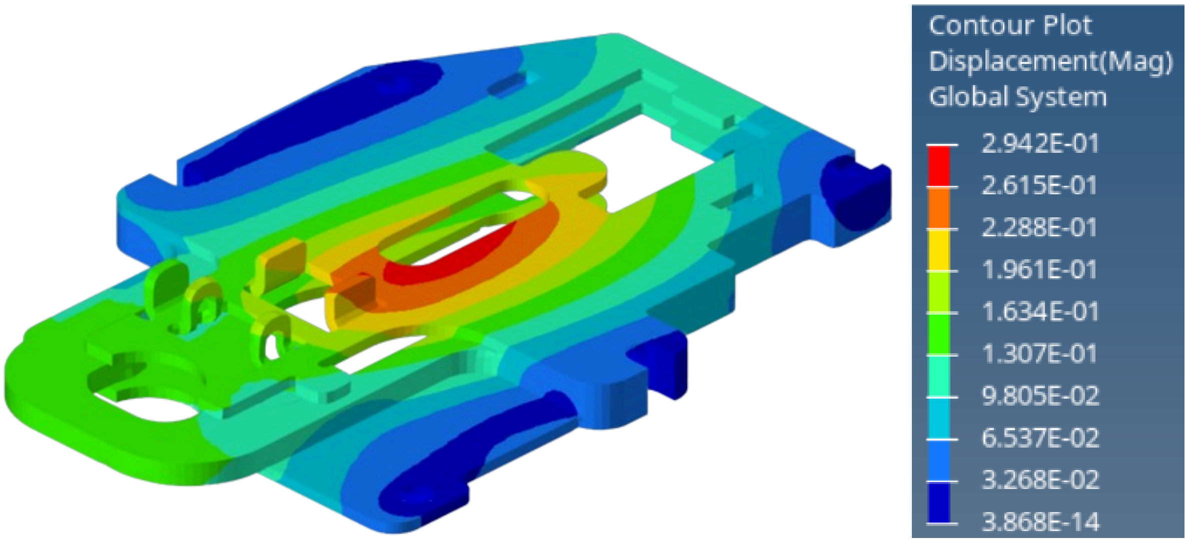


Figure 24. Displacement (magnitude in mm) contour plot for HSLA Steel, grade 45 for temperature change. The maximum magnitude of displacement was 0.29 mm, concentrated around the rounded slot.

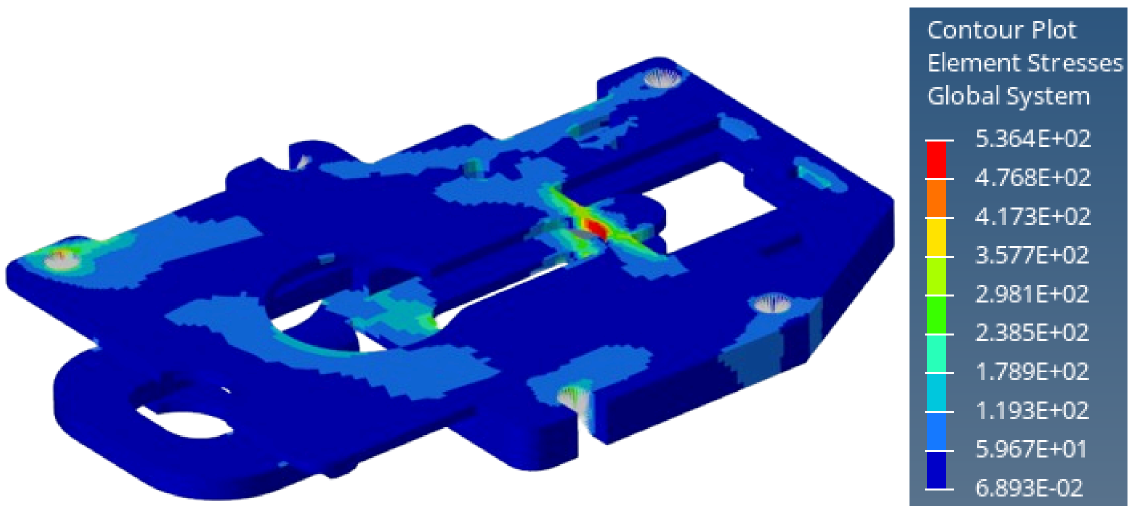


Figure 25. Stress (magnitude in MPa) contour plot for PA6 13% GF for temperature change. The maximum magnitude of stress was 536 MPa, concentrated at the rounded slot.

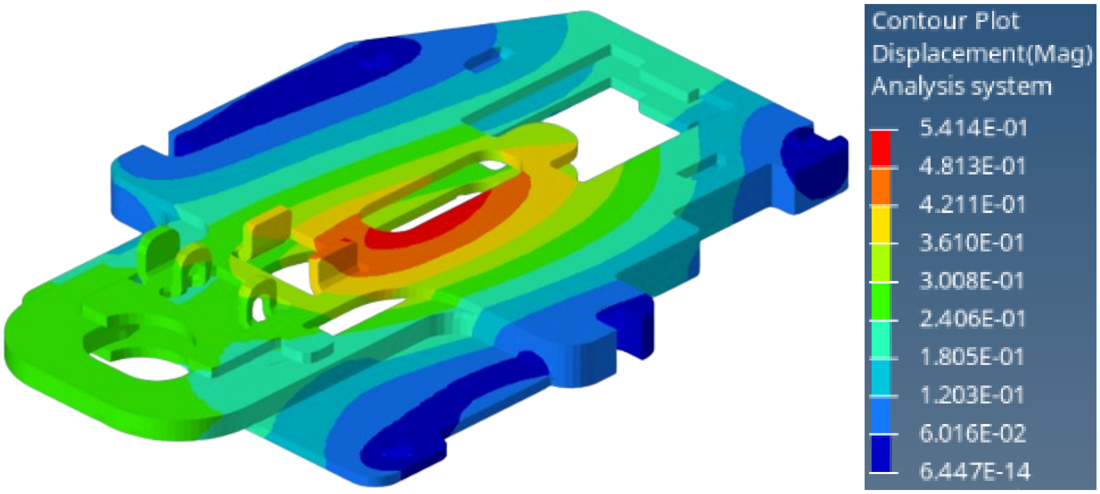


Figure 26. Displacement (magnitude in mm) contour plot for PA6 13% GF for temperature change. The maximum magnitude of displacement was 0.54 mm, concentrated around the rounded slot.

The values of stress and displacement for the 2 models with the temperature change are summarized below in Table 18. It was found that changing the material from HSLA Steel, grade 45 to PA6 13% GF decreased the stress by an order of magnitude, but increased the displacement.

Table 18: Maximum magnitude of displacement (mm) and stress (MPa) due to the temperature change of 160 °C on HSLA Steel, Grade 45 and PA6, 13% GF. From the simulation, the stress for both materials remained the same, however, the displacement of the PA6 13% GF was an order of magnitude greater than steel.

	HSLA Steel, Grade 45	**PA6 13% GF**
Max Stress [MPa]	2450	536
Max Displacement [mm]	0.29	0.54

C. Actuation Force and Temperature Changes

To simulate the actuation force and sudden temperature change of the environment, the loads from the previous 2 sections were combined into the same model. It is important to note that the model being simulated as one solid piece led to inaccurate representations of stress and displacement in the part due to the thermal load. Although the values for stress and displacement are not accurate, the behavior of the model is. Discussion of the limitations in this analysis and how we plan to address it can be found in our future project plans. This combined loading test was done for only PA6 13% GF to see how the model will respond. Once the accuracy of the model is improved, we will compare it to HSLA Steel, grade 45 as a baseline.

The simulation results for stress and deflection of PA6 13% GF are depicted in Figures 27 and 28. Results are only shown for 2D elements with reasoning of this choice explained in Appendix B.

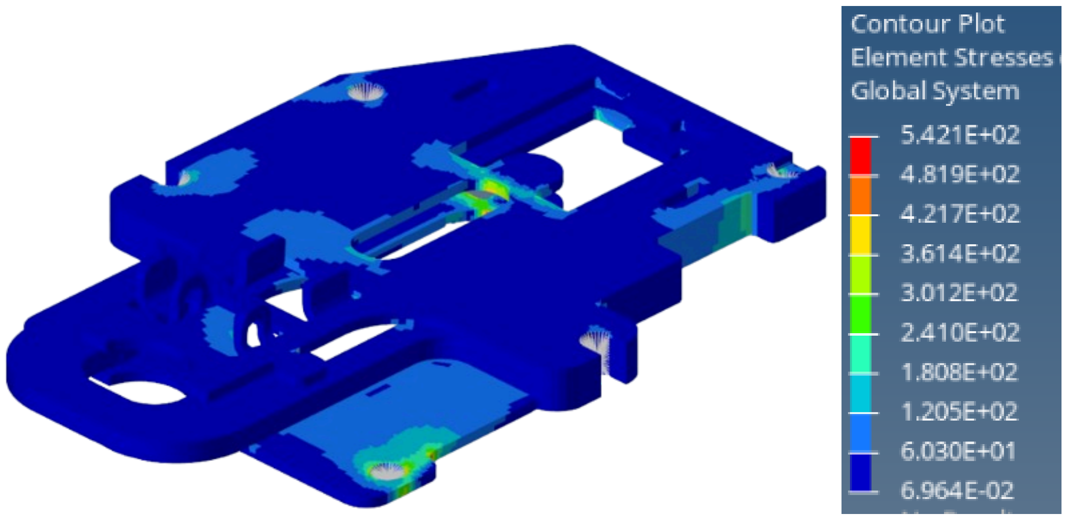


Figure 27. Stress (magnitude in MPa) contour plot for PA6 13% GF for temperature change and actuation of the IPM. The maximum magnitude of stress was 542 MPa, concentrated at the rounded slot.

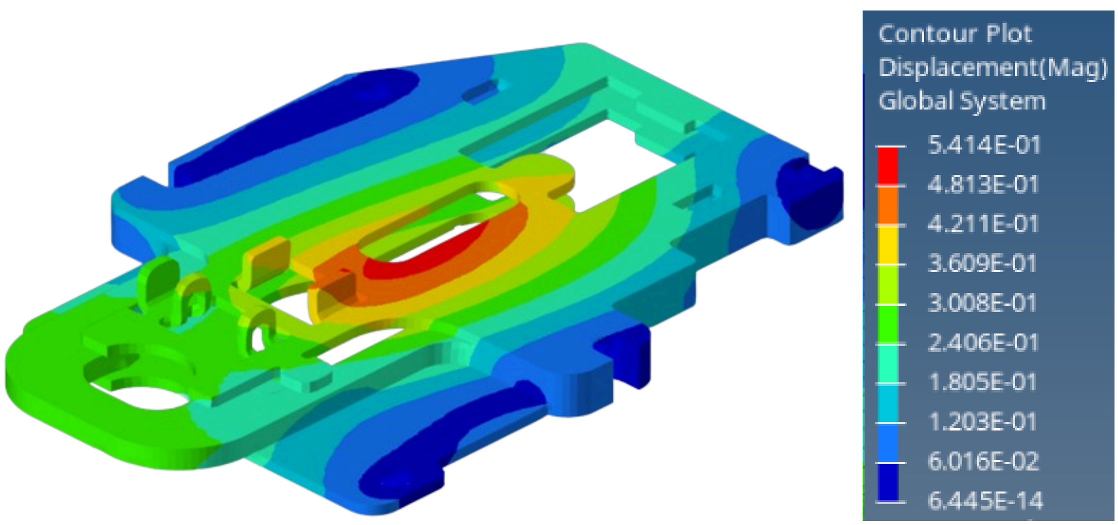


Figure 28. Displacement (magnitude in mm) contour plot for PA6 13% GF for temperature change and actuation of the IPM. The maximum magnitude of displacement was 0.54 MPa, concentrated around the slot.

IV. Final FEA Results

After our preliminary FEA analysis, we decided that our final design will use PA6 33% GF as the selected material. With this material change, we conducted the same analysis as above for the new material. We only included the new stress analysis since stress will cause the part to fail since the displacement results were similar to our preliminary analysis. Results for maximum stress are summarized in Table 19.

Table 19: Maximum magnitude of stress (MPa) due to the spring force, temperature change, vibrational loads on PA^ 33% GF. From the simulation, stress for the temperature change was much higher than vibrational loads or the spring force.

	Spring Force	**Temperature Change**	**Vibrational Loads**
Max Stress [MPa]	16.4	66.4	2.7

A. Spring Force

To simulate the actuation force of the IPM on the part, the spring force was modeled as a distributed load modeling both springs as shown in Figure 18. The spring force was evaluated with the maximum actuation distance of the IPM, its uncertainty, the stiffness constant of the springs, and a safety factor of 1.7 as set in our requirements and specifications by Stoneridge. The resulting stress distribution plot is shown in Figure 29 with a maximum stress of 16.4 MPa.

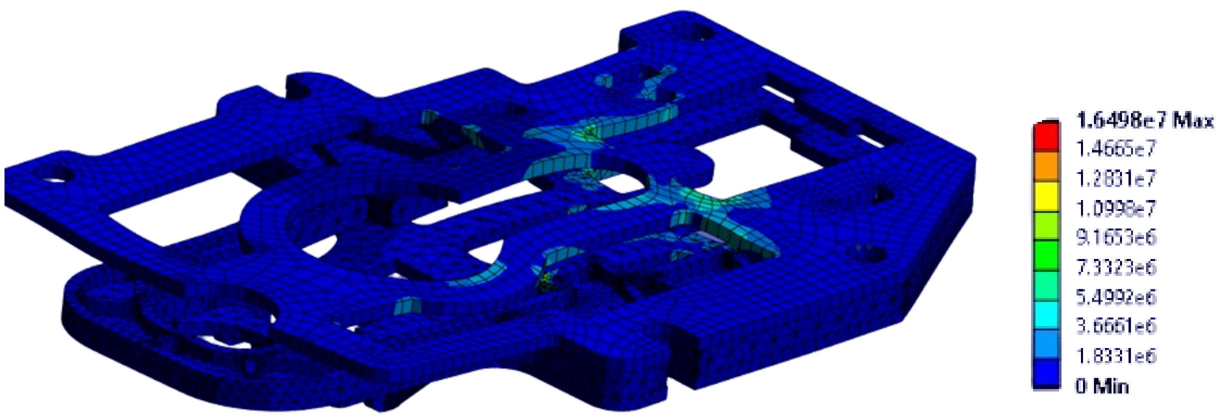


Figure 29. Stress (magnitude in MPa) contour plot for PA6 33% GF for actuation of the IPM. The maximum magnitude of stress was 16.4 MPa, concentrated at the rounded slot.

B. Thermal Loading

To simulate the stress and displacement due to a temperature change of the environment of the IPM, a global temperature change of 160 °C was applied to the model. This temperature change simulates the temperature change between -40 and 120 °C. The stress generated due to a temperature change does not depend on the initial or final temperature, only the difference between those 2 values. The resulting stress distribution plot is shown in Figure 30 with a maximum stress of 66.4 MPa.

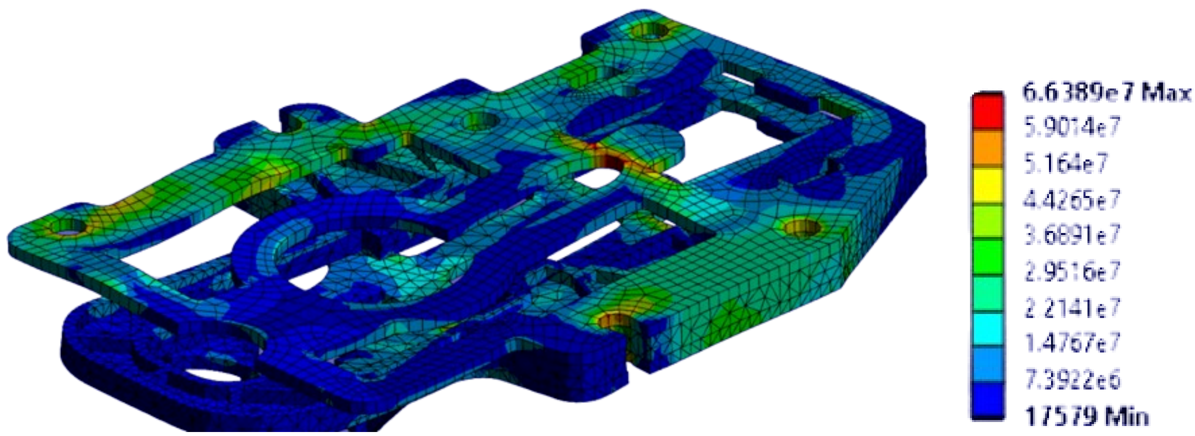


Figure 30. Stress (magnitude in MPa) contour plot for PA6 33% GF for temperature change. The maximum magnitude of stress was 66.4 MPa, concentrated at the rounded slot.

C. Thermal Loading - Original IAP and OAP

We wanted to verify that the reduction of material of the IAP and OAP did not compromise the performance of the part. To do this, we simulated the thermal loading conditions with the IAP and OAP without the volumetric reduction to compare the resulting stress between the two. We only simulated thermal stresses for this since it was dominant in comparison to the other loading conditions. To simulate the stress and displacement due to a temperature change of the environment of the IPM, a global temperature change of 160 °C was applied to the model. This temperature change simulates the temperature change between -40 and 120 °C. The stress generated due to a temperature change does not depend on the initial or final temperature, only the difference between those 2 values. The resulting stress distribution plot is shown in Figure 31 with a maximum stress of 71.0 MPa. This stress is larger than the stress with the volumetric reduction in the IAP and OAP, hence, we lengthened the lifespan of the IPM assembly.

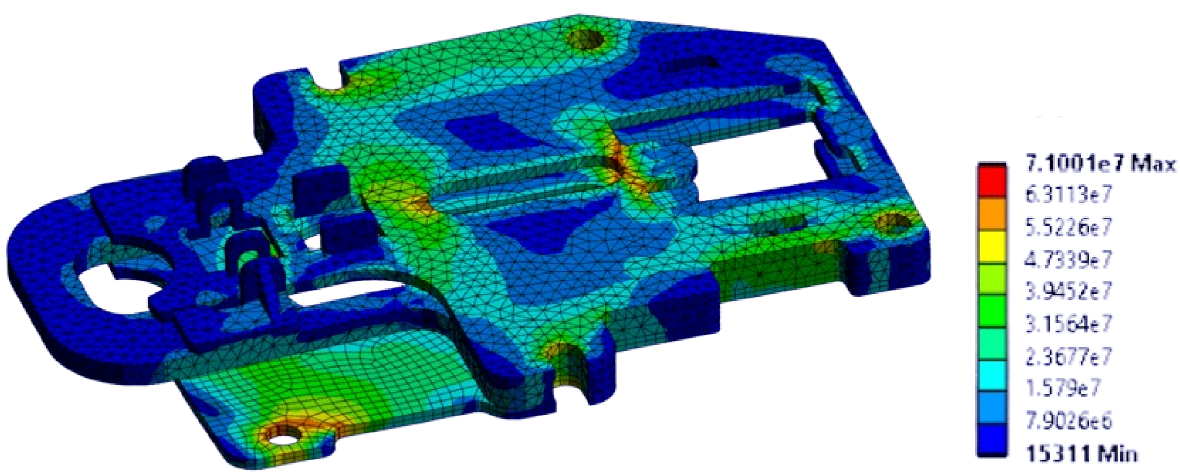


Figure 31. Stress (magnitude in MPa) contour plot for PA6 33% GF for temperature change with the original IAP and OAP design as PA6 33%. The maximum magnitude of stress was 71.0 MPa, concentrated at the rounded slot.

D. Vibrational Loading

To simulate the vibrational loading that the IPM will undergo, we simulated the V2 vibrational profile shown in Figure 32.

Classification		Acceleration (m/s^2)	Frequency (hz)	Sweep Type	Time per Sweep (Min)	Number of Sweeps per Axis	Number of Axes	Total	
Number	Description							Sweeps	Duration
VI	Non-Standard	5 g's 10 g's 5 g's	20-100 100-300 300-1000	Sinusoidal	10	48	3	144	24 Hours

Figure 32. V2 Vibrational profile for FEA vibrational simulations.

The resulting stress contour plot for the vibrational simulation is shown in Figure 33. The maximum stress that occurred during the sweep through the vibration profile was 2.7 MPa; this stress was significantly lower than the temperature and physical loading stresses.

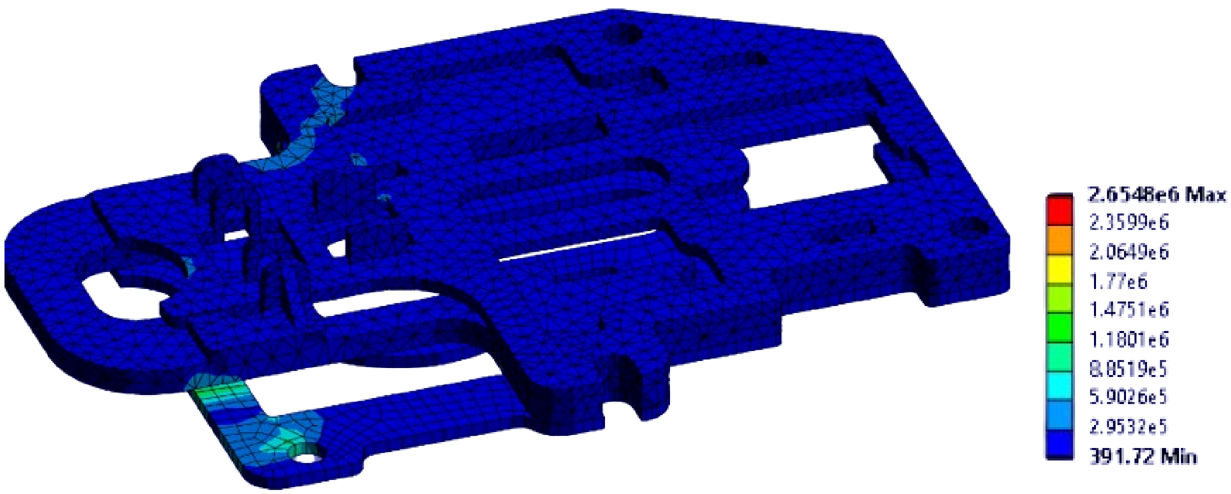


Figure 33. Stress (magnitude in MPa) contour plot for PA6 33% GF for vibrational loading. The maximum magnitude of stress was 2.7 MPa

V. Mold Flow Analysis

Since we selected injection molding as our manufacturing process, we conducted Mold Flow analysis to assess the manufacturability of our final design. In this analysis, we looked at the mold fill time, end flow velocity, end flow pressure, and weld lines. Mold fill time has the greatest effect on production costs; end flow velocity can be used to adjust the injection ports to create uniform flow; end fill pressure (must be below 70 MPa) can be used to determine if there will be voids or deformities in the part; weld lines will indicate weak points in the part after molding is complete. For each mold, we used 2 injection points per suggestions from Stoneridge.

A. Outer Alignment Plate

The geometry of the OAP was changed to improve the injection molding process of the part. The cutouts shown in Figure 7 were adjusted to ensure uniform thickness between part geometry. Uniform thickness of the part will allow the flow of plastic through the mold to be uniform. Thicker sections will result in slower flow, which will lead to plastic cooling prematurely. This could create discontinuities in the material, leading to a weaker part. Results for mold flow analysis of the OAP are shown in Figure 34. The injection ports for the OAP were intentionally selected improperly to emphasize the effect of weld lines in the part. In figure 34, we noticed that the weld lines are located at the region of maximum stress due to thermal loading (as shown in Figure 30). This will lead to premature failure of the part. From the other analysis the mold fill time remains low (the fill time for the Spacer without ribbing was 0.27 seconds), the end fill pressure is below the 70 MPa threshold, and the end flow velocity is somewhat uniform; the injection ports will need to be adjusted.

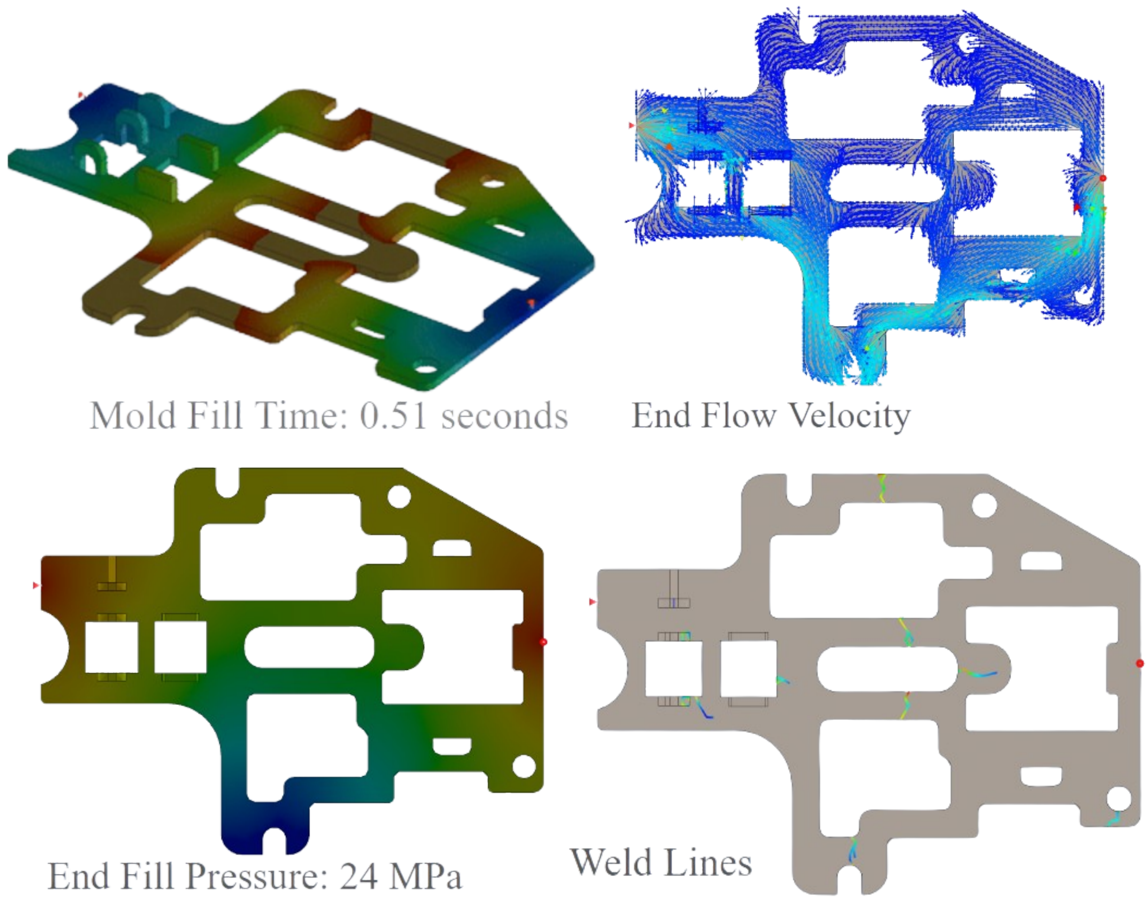


Figure 34. Mold flow analysis for the OAP with mold fill time, end flow velocity, end fill pressure and the weld lines depicted.

B. Inner Alignment Plate

The geometry of the IAP was changed to improve the injection molding process of the part. The cutouts shown in Figure 6 were adjusted to ensure uniform thickness between part geometry. Uniform thickness of the part will allow the flow of plastic through the mold to be uniform. Thicker sections will result in slower flow, which will lead to plastic cooling prematurely. This could create discontinuities in the material, leading to a weaker part. Results for mold flow analysis of the IAP are shown in Figure 35. From the our analysis, the mold fill time remains low (the fill time for the Spacer without ribbing was 0.27 seconds), the end fill pressure is below the 70 MPa threshold, and the end flow velocity is somewhat uniform and the weld lines can be found in critical regions like the bolt hole; the injection ports will need to be adjusted.

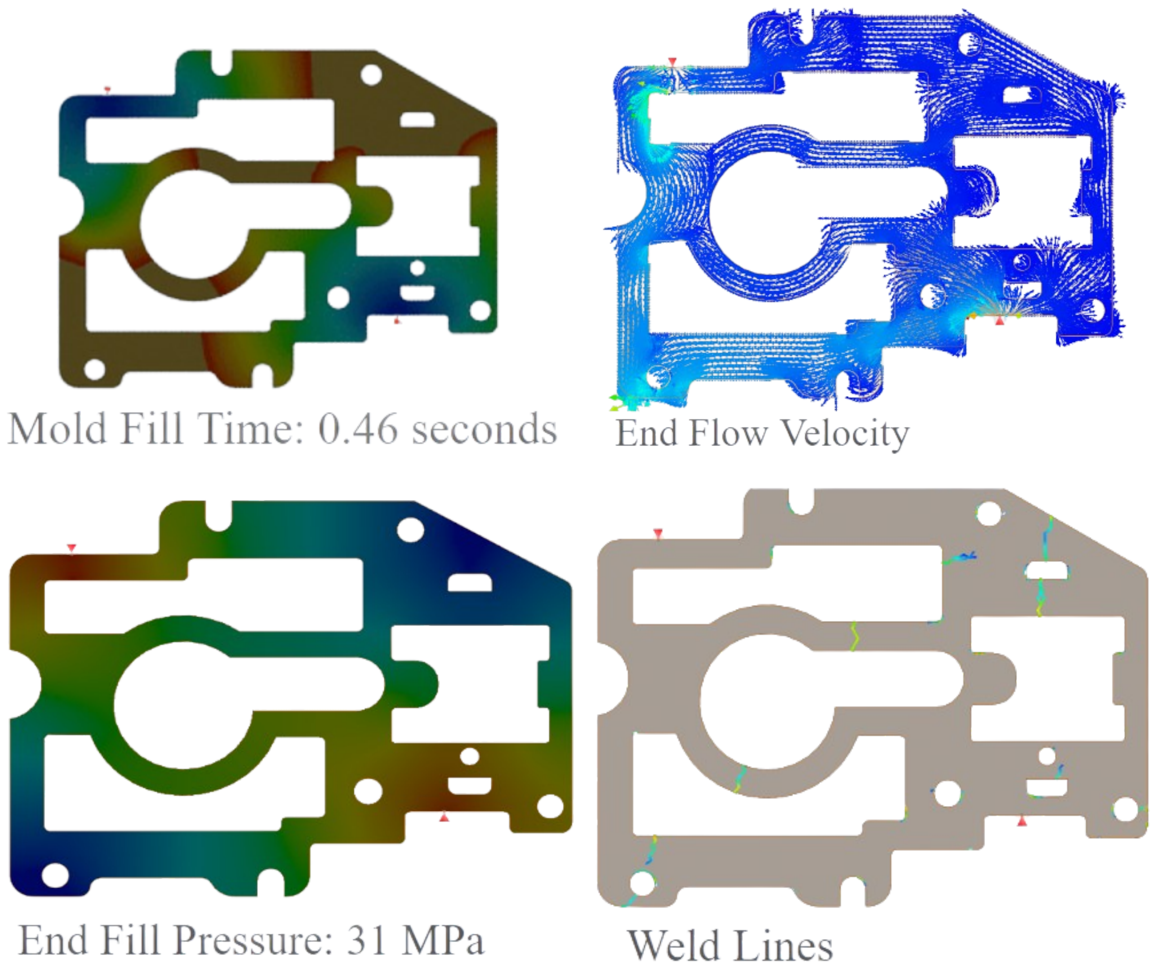


Figure 35. Mold flow analysis for the IAP with mold fill time, end flow velocity, end fill pressure and the weld lines depicted.

C. Spacer

The geometry of the Spacer was changed to improve the injection molding process of the part. To ensure uniform thickness between part geometry, the thickness of all geometry was fixed at 1.75 mm. Uniform thickness of the part will allow the flow of plastic through the mold to be uniform. Thicker sections will result in slower flow, which will lead to plastic cooling prematurely. This could create discontinuities in the material, leading to a weaker part. Results for mold flow analysis of the Spacer are shown in Figure 36. From our analysis, the mold fill time remains low (the fill time for the Spacer without ribbing was 0.27 seconds), the end fill pressure is below the 70 MPa threshold, and the end flow velocity is uniform. The weld lines on the part are relatively random due to the way the plastic flows through the ribbing. Although the end fill pressure is well below the 70 MPa threshold for injection molding, the end pressure and fill time in the part is asymmetric. The injection ports will need to be adjusted to accommodate this.

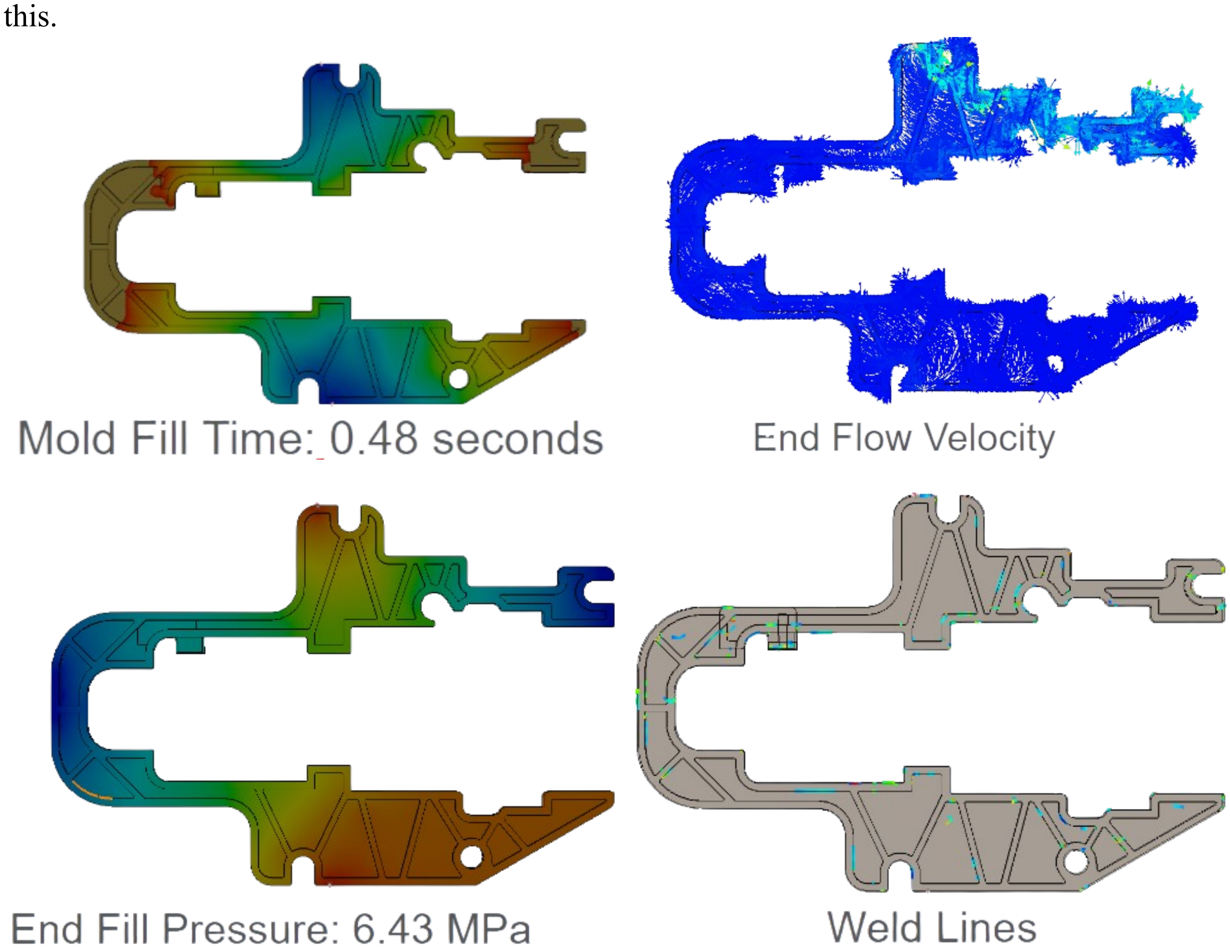


Figure 36. Mold flow analysis for the IAP with mold fill time, end flow velocity, end fill pressure and the weld lines depicted.

Final Design

After undergoing FEA thermal testing, it was determined that PA6 with 13% glass-filling was not suitable as a material due to its heat deflection temperature. The stress concentrations were found to be too high and material deformation began to occur. We came to the conclusion that implementing a material change to PA6 with 33% glass-filling for the Spacer, IAP, and OAP would be most effective due to the low cost and good thermal properties. This is further corroborated by Table 20. which is charted below.

Table 20. Updated glass-filled plastic selection matrix, with the addition of PA6 33% glass-filled. Materials were compared by their material properties of density, flexural strength, cost, linear thermal expansion coefficient and the heat deflection temperature, and charted against the current material HSLA Steel, Grade 45. Green represents the best material property value relative to the rest of the materials, where red represents a value far from that.

Requirement and Specification	Relevant Parameters	HSLA Steel, Grade 45[2]	PA6[5] (13% Glass Filled)	PA6[5] (33% Glass Filled)	PA66[4] (13% Glass filled)	PA66[4] (33% Glass filled)
Weight	**Density [g/cm³] ASTM D792**	7.87	1.21	1.4	1.23	1.39
Loading Conditions	**Flexural Strength [MPa] ASTM D790**	310	175	200	155	200
Cost	**Cost [$/kg]**	2.50 + Coating	~$0.60 < than PA66	2.50 < Cost < 4.00	1.37< Cost < 4.52	4.52
Temperature Conditions	**Linear Thermal Expansion Coefficient [µm/m°C] ASTM E831**	12.4	22	23	22	27
Temperature Conditions	**Heat Deflection Temperature[°C] ASTM D648**	N/A	200	250	220	250

We ultimately decided to stick with our original selected design with the only change being instead of implementing a material change for the IAP, OAP, and Spacer from Grade 45 HSLA steel to PA6 13% Glass-Filled, it would be changed to PA6 33% Glass-Filling. The Spacer will also have the addition of ribbing, which will promote a uniform wall thickness. Also, material elimination will be implemented in both the inner and OAP, which will provide a volume reduction to reduce stress concentrations and mold cost. Shown below in Figure 37 is the updated CAD for this prospective design change.

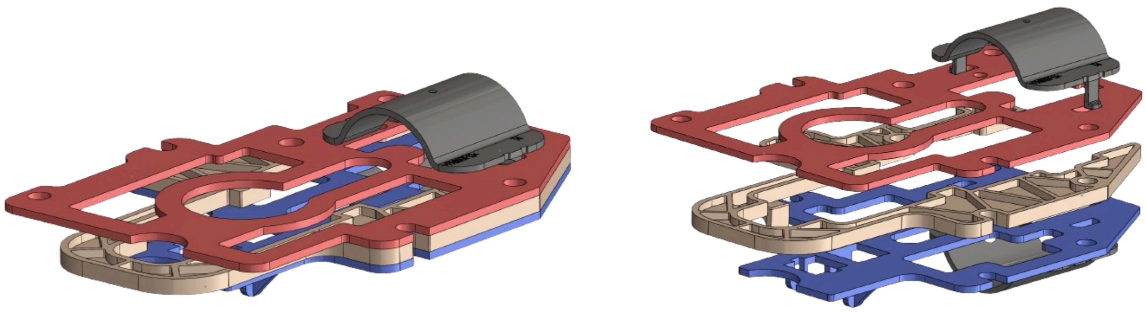


Figure 37. Updated CAD model for our final design. The Spacer, IAP, and OAP will be changed from HSLA Grade 45 steel to PA6 33% Glass-Filled that is injection molded. All of these parts will be injection molded, while the outer and IAP will undergo a volumetric reduction.

For the design expo, our team has decided to 3D print a prototype using PA6 with 25% glass filling, in order to model our design effectively in a way that is still feasible in the timeline for our project. This percentage of glass filling was the closest to that of our material change that was easily accessible for purchase. Shown in Figure 38 below is our setup for a functional demonstration of the IPM using an Arduino Nano, A4988 driver, and stepper motor.

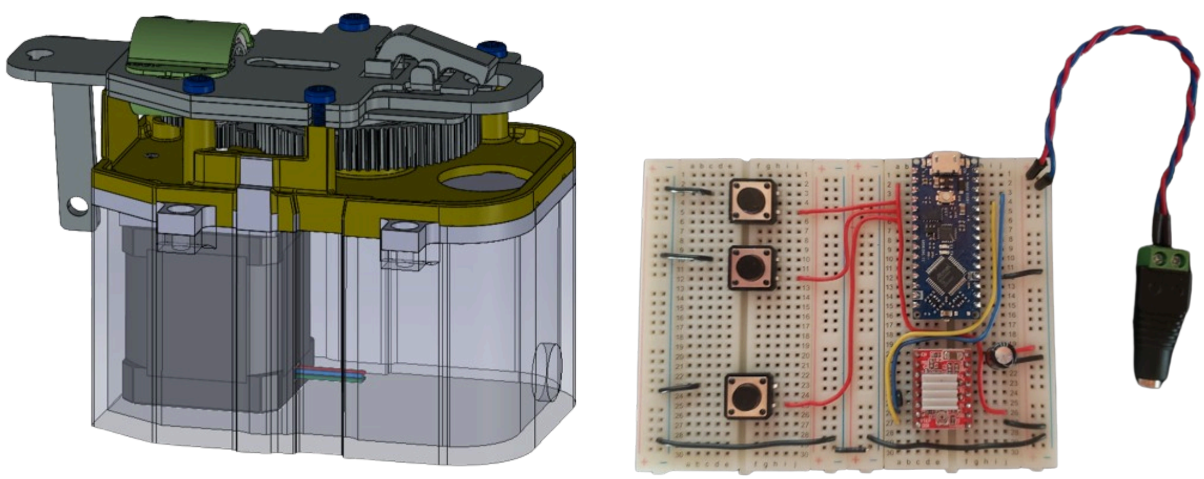


Figure 38. CAD for the design expo setup and circuit used to control the stepper motor.

Verification and Validation Plans

To ensure the IPM is able to meet the requirements and specifications outlined, verification and validation tests were created. In general, FEA analysis was used to verify that our modified design was able to meet the load and environmental constraints, while physical testing which is outside of the scope of our project, would be used to validate that the modified design could actually meet the specifications in a real life setting (as opposed to simulated). The tests that will be conducted at Stoneridge are standard to the IPM; they have a standard experimental process and equipment to validate the specifications. The verification and validation methods for environmental loading conditions and design are outlined in Table 21 and 22, below. Verification can be found in the Engineering Analysis section above with derivations in Appendices B and C

Table 21. Requirements and specifications for environmental loading conditions and their associated verification and validation methods.

Requirements	Engineering Specifications	Verification/Validation Method	Priority
Durable	-10 Year Lifespan[9] - 300,000 Cycles between park and neutral[9]	**Validation:** Physical Testing – IPM testing on vibration machine at Stoneridge	HIGH
Withstands Physical Loading Conditions	- 450 N[1] pull force on slider - Screw torques, 25 Nm[9] - SF of 1.7[9]	**Verification:** FEA Analysis – Static loading simulation **Verification:** Beam Theory – Analyze deflection from slider force of 450 N, and spring forces of 100 N and 35 N is applied.	HIGH
	- V2 Vibration Conditions 9.8-10 g[9], USCAR 2-8 SAE[17] - SF of 1.7[9]	**Verification:** FEA Analysis – Vibration simulation to USCAR 2-8 SAE Standards **Validation:** Physical Testing – IPM testing on vibration machine at Stoneridge	HIGH
Withstands Temperature Conditions	-T2 Temperature Conditions -40 to 120°C[9], USCAR 2-8 SAE[17] - Minimal Contact Friction from Thermal Expansion	**Verification:** FEA Analysis – Temperature constraints applied, temperature change of 160°C **Validation:** Physical Testing – IPM testing on vibration machine at Stoneridge	HIGH
Non-corrosive	Non-corrosive grade plastics and coating for steel parts, ASTM D7216-23[18]	**Verification:** Standard – ASTM D7216-23 for PA6 and PA66. **Validation:** Physical Testing – PA66 33% glass filled previously tested at Stoneridge; used for the current spring retainers.	HIGH

Table 22. Requirements and specifications for design and their associated verification and validation methods.

Requirements	Engineering Specifications	Verification/Validation Method	Priority
Decrease Cost	8% decrease of current cost - Current manufacturing cost: $5.12[9]	**Verification:** Material – research on materials that can decrease the cost **Verification:** Mold Flow Analysis – simulate the mold fill and cool time and fill pressure to determine the time needed to create part **Validation:** Manufacturing – Calculate net cost of injection molding over the lifespan of 10 years	HIGH
Decrease Weight	50% Weight Decrease of Assembly	**Verification:** Material – research on materials that can decrease the weight of the IPM **Validation:** Quantitative Analysis - Physically measure part after manufacturing	MEDIUM
Simple Solution	Design has no alteration with other subsystems of benchmark (Gen 2.) model	**Validation:** Qualitative Analysis – create a prototype and assemble IPM using prototype to check interferences	MEDIUM
Quickly Manufacturable	Maintain or decrease number of current manufacturing steps (19) with reduction or decrease in cycle time (30 seconds)	**Validation:** Manufacturing – Outline manufacturing steps for new sub-assembly; test run parts to obtain new cycle time.	MEDIUM

Problem Domain Analysis, Reflection and Iteration

From development of the engineering requirements and specifications, the approach redesigning the IPM will consist of fundamentals learned through work and coursework with premeditated knowledge of the possible problems that will arise from our analysis. At the end of the semester, the deliverable will be a finalized CAD model with injection molding manufacturing plans to allow for quick manufacturing of the part. A physical prototype of our selected material (3D printed glass filled filament will be used) of the CAD model will be provided with failure analysis performed from physical testing and simulations through FEA.

To determine if the selected design concept will meet the requirements and specifications described in Tables 4 and 5, the necessary fundamentals to analyze and converge on verification for our design were outlined below. With the necessary fundamentals required to meet our set deliverables, anticipated challenges were identified so that we are aware of potential issues that correlate with the fundamentals required to complete our project.

I. *Necessary Fundamentals:*

- *Material Exploration /Analysis* - A material change to the parts of the subassembly is a key consideration due to our sponsor's desire to decrease the weight and cost of their product. Although the material change will result in weight reduction, the strength and durability specifications are important to consider; we must carefully analyze potential materials.

- *Structural Design* - Structural design is a necessary concept when designing load bearing structures for optimal cost. Basic structural design analysis will impact our ability to generate an accurate model for verification. When changing the geometry of the part, computer aided design software will need to be used.

- *Failure Analysis* - We will need structural analysis techniques for verification that our designs meet the physical specifications. Iteration through various designs will require accurate and efficient models that describe the environment of the transmission accurately. Failure analysis software and finite element analysis will be used.

- *Fluid Flow Analysis* - We will need fluid flow analysis to optimize the injection molding process to keep the cost of manufacturing low. Pressure during injection and time required for injection will need to be kept to a minimum. We will need to be able to use this software to present a fleshed out manufacturing plan and cost estimation for Stoneridge.

- *Experimental Testing* - Once prototypes of our design are created, we will need to be able to run physical tests on the parts to align with the failure analysis through our simulations. We will work with the test engineers at Stoneridge to conduct vibrational and cyclic loading experiments at Stoneridge's testing facility

- *Ribbing Web* - Using the failure analysis simulations, we will need to analyze the stress in the part to create a ribbing web to minimize excess stress in the part. The web we create will need to abide by injection molding standards for ribbed assemblies.

II. *Potential Difficulties/Problems:*

- An information gap that we will need to overcome is our modest experience in material selection of polymers and other engineering materials. There is a large volume of materials to select from, but a small base of acceptable materials permitted for use in a transmission environment.

- We will need technical assistance from Computer Aided Design and Finite Element Analysis, and Injection Moulding fluid flows softwares and knowledge of how to use these tools in order to solve the problem[20]. We have experience with these tools, but we will need to develop a complex model with a high confidence interval for predicting the outcome of the part. We will need to refer to experts in the field of simulation and other resources[19] for any additional information we might need.

- Probability failure analysis will need to be used jointly with the Finite Element Analysis simulations to ensure that our parts will be able to meet the loading and environmental requirements in Tables 4 and 5. Due to the high complexity of the loading conditions, this failure analysis will require research to ensure accurate results. Our predictions will need to be validated with physical testing.

- Due to material suppliers requiring large volumes for purchasing, obtaining materials to test could be difficult. We might be limited to 1 or 2 materials due to the constraints of our budget. We will also need to find a way to mold and cast the parts from the raw material.

- We will need to separate the current model into separate parts. The model is currently one solid piece. This creates unnatural stress concentrations when simulating the thermal expansion of the model. We will need to include the slider in the model as well. This will increase the number of elements in the model, leading to longer solving times. This will impact the rate of iteration for ribbed models.

- Due to the large temperature change, the thermal expansion coefficient of the model is not constant over the span. We will need to figure out how to perform nonlinear analysis if we decide not to approximate the coefficient as constant.

- We will need to decide if we include forces due to bolt torque and will need to simulate vibrational loading. For vibrational loading, Ansys will be able to simulate the vibrational profile defined by USCAR 2-8 SAE easier than Hyperworks. Although it is easy to simulate, we will need to familiarize ourselves with this new software.

- We are concerned about the manufacturing process for the combined OAP and Spacer and inserting the spring retainers. We plan on discussing this with Stoneridge to evaluate if it will be an issue. If it is, we will need to reassess the cost analysis to verify if injection molding all three parts would be a cost effective solution, or if we should only injection mold the Spacer.

Discussion

Throughout the course of this project, we were able to achieve our target goal of optimizing the design and decrease the cost by approximately 50% for the Generation 2 IPM provided to us by Stoneridge. However, it is important to evaluate our experience and discuss alternative solutions and factors we may have overlooked when working with the project in real time.

I. Problem Definition

A large proportion of our design problem encompassed using a material change to facilitate a cost optimization. Due to the time constraints, material changes served as the most efficient and effective method of reducing cost without compromising Stoneridge's desire of wanting limited design changes when considering the interaction of the subsystem provided to the team in comparison to the surrounding IPM parts. If we had more time and resources to collect data and better define the problem for our project, exploring alternative means to reduce cost such changes in IPM actualization and re-designing the IPM mechanism as a whole is a valid but time consuming route to explore. Morphological charts and brainstorming sessions encompassing these two ideas could be held to explore this design space further.

II. Design Critiques

When evaluating our current design, minimal changes in the shape of the inner alignment plate, OAP, and Spacer is beneficial for Stoneridge because it allows for fewer changes in the general assembly since our design optimizations do not interfere with existing parts. A weakness of our current design is we had not fully explored the design space for optimizing ribbing and injection ports for molding. Future work using topology would allow us to explore a more optimal ribbing pattern. Injection ports were determined using placements with one and two ports and comparing the fill time, end pressure, and cool time. However, deeper exploration could lead to more efficient fill and cool times and lower end pressures.

If given more time we would have liked to accomplish to further improve the simulations of the current design. Our current FEA analysis considers the factors of static loading, vibration, and temperature separately. With more time we'd like to create a simulation encompassing all three of these environmental loads in one file.

Having a physical prototype that is injection molded would allow us to physically test the IPM for vibration and fatigue. A simulated fatigue analysis for each environmental factor would also aid in verifying the durability of the IPM. One method in which this could have been completed is using cyclic loading laws to determine the cycle-fatigue for the lifetime of the IPM. Baquin's Law[43] for high-cycle fatigue can be relevant to constant stress loads caused by the spring being actuated between parking and driving modes. Vibrational and temperature loads can be accounted for using Miner's Rule[43] in conjunction with the maximum stress extracted from FEA analysis of the subsystem to determine the hypothetical number of cycles until failure.

Additionally, the current ribbing design was optimized by finding a balance between a more complex ribbing pattern and a relatively simple one, with the aid of FEA software to ensure uniform wall thickness, reasonable end pressure and optimal cool and fill time. If granted more time, utilizing topology[47] would allow us to optimize the distribution of material for the environmental factors of spring load, vibration loads, and a thermal load to determine material elimination in areas that do not contribute to part strength. We'd also like to include metal inserts for the screw holes to ensure part integrity.

III. Risks

The main challenge that we encountered during the design process was getting acclimated with the FEA analysis and moldflow software. However, we were able to overcome this by meeting with an industry moldflow expert and by familiarizing ourselves with the software through educational tools and trial and error.

FEA analysis was done to ensure the IPM mechanism does not face any performance changes or issues when completing our task of optimizing the design and cost. Failure to maintain performance can pose risks for customers who drive vehicles the IPM is currently installed in such as the F150, F150 Lightning, and Ford Bronco. Failure in the IPM mechanism's actuation can stall the vehicle in park mode (the default mode), causing the vehicle owner to have to replace the entirety of the transmission. Needing to replace a transmission prematurely is both costly and suboptimal, which is why such thorough simulation and physical testing of the mechanism is crucial.

One area of concern that the team did not have the opportunity to complete within the timeframe of the project was determining the new screw torques for the injection molded plastic parts. For Generation 2's steel IPM, the screw torque was approximately 25 Nm. For plastic parts, a lower screw torque would be more optimal to ensure the part does not bend unfavorably during assembly. Our team noticed when 3D printing the material with a filament of PA6 25% glass filled, accidental over-torquing when inserting the screws led to the OAP being unfavorably bent. This can cause premature deformities and decrease part integrity. One possible design optimization to circumvent this is adding metal inserts to the plastic screw holes. Alternatively, only changing the Spacer to PA6 33% glass-filled while leaving the rest of the assembly as HSLA Grade 45 Steel would still reach a decrease in manufacturing cost by 21%, surpassing the target goal of 8%.

Reflection

After spending the past few months working toward a complete version of our project, it is important to reflect on how our work will lead to making the world a better place and how our perspectives on the global context of our project have evolved over time. What we considered throughout our project were external factors, team collaboration, inclusion and equity, and ethics.

I. External Factors

a) Public Health, Safety, and Welfare

The IPM will be placed in 650,000 transmissions per year. This high production value makes public health, safety, and welfare pivotal for our final design. We have conducted analysis and design selection with that in mind. Our perspective to create a final product that has been thoroughly validated and verified has remained the same throughout our project.

b) Global Context

Our design would benefit the global marketplace due to competition. The improvement of the IPM will allow Stoneridge to invest their resources into other products, or improve the IPM while keeping the same profit margins from the cost reduction. This will force competitors of Stoneridge to produce a better product, benefiting consumers in the global marketplace.

c) Societal Impacts with Manufacturing, Use and Disposal

Since the IPM is not being changed drastically, besides a change from metal to plastic for a few subcomponents, societal impacts with use and disposal of the product will not drastically change; societal impacts with manufacturing will be affected. We have considered the environmental effect of producing the parts from steel to plastic. It was pivotal when selecting a design during our second design report.

d) Economic Impacts with Manufacturing, Use and Disposal

Since the IPM is not being changed drastically, besides a change from metal to plastic for a few subcomponents, economic impacts with use and disposal of the product will not drastically change; economic impacts with manufacturing will be affected. We have considered the economic effect of producing the parts from steel to plastic. It was the overarching goal of our project, so selecting a manufacturing process to reduce cost was important to us throughout the semester.

e) Basic Tools Used

To characterize the potential societal impacts of our design, we used a stakeholder map to determine who would be affected by our final design. This helped us understand the butterfly effect of our work throughout the semester.

II. Team Collaboration

a) Between Team Members

All team members approached the project with their own unique skills and problem solving skills, which helped develop our final design. Cultural, privilege, and identity of group members had little impact during the semester. We viewed opinions equally, regardless of background. Stylistic differences benefited the team greatly. Our stylistic differences led to a polished final report, with testing and analysis conducted in various ways, all leading to similar results.

b) With Stoneridge

Mentors of Stoneridge provided insight, suggestions, requirements and specifications and their desired goal for the project which helped develop our final design. Cultural, privilege, identity, and stylistic differences between group members and Stoneridge had little impact during the semester. We viewed opinions equally, regardless of background. The power dynamic between us had the most significant impact on the project. Our mentors had key manufacturing and design information for the IPM, so we trusted their experience with the assembly.

III. Inclusion and Equity

With our product being implemented inside the transmission of cars, power dynamics had a large impact on our project. The end user had the largest impact on the integrity of our design and analysis. If there's no demand for a product, or it is dysfunctional, a customer will have no desire to purchase and use the product. If we made a change that negatively impacted the end user (total replacement of the transmission), they would lose trust in their cars, and in turn, OEMs would lose trust in Stoneridge and the products they produce.

That chain of reliance led us to come up with diverse viewpoints and solutions for the project. For these diverse viewpoints, each team member independently came up with 40 unique solutions. We then discussed each idea with one another, which led to our initial design concept. We made sure to communicate with Stoneridge throughout the semester to ensure that our design was feasible.

Between group members and Stoneridge, cultural similarities and differences had little impact on the final product and the design process. Although our culture did not impact our project, we made use of our unique background with engineering to use each group member's and mentor's experiences. All opinions considered valid and disagreements were resolved through research and testing in our analysis.

IV. Ethics

Ethical dilemmas came about during analysis of our design. With our limited knowledge of various FEA and mold flow software, we were unable to conduct combined loading tests and cooling analysis. For our final design, we wanted both, but were uncertain of their validity. We had to omit that analysis during the design expo, although it was something that we should have had. With the large production volume of such a critical component in the transmission, we are obligated to uphold our ethics as representatives of the University of Michigan, just like we were in charge of this project by a future employer.

Recommendations

The change of the IAP, Spacer, and OAP from Generation 2's HSLA Grade 45 Steel to PA6 33% glass-filled has allowed our team to decrease the manufacturing cost of the assembly provided to us by approximately 50%. However, it is important to acknowledge that the verification of this material change has only been completed through simulation software. It is important to validate that the spring load, vibration profile, and temperature conditions of -40° to 120° are adequately met for a physical prototype. Fatigue testing of a physical prototype is crucial to validating part integrity and user safety as well as ensuring a life of cycle of 300,000 actuations between park and drive is met. We are aware Stoneridge has a physical actuation and vibration machine and would encourage validating our simulation results using these devices.

For additional steps, we recommend adding metal inserts into the screw holes and determining the new screw torques for the injection molded plastic OAP, Spacer, and IAP. For plastic parts, a lower screw torque would be more optimal to ensure the part does not bend unfavorably during assembly. Alternatively, only changing the Spacer to PA6 33% glass-filled while leaving the rest of the assembly as HSLA Grade 45 Steel would still reach a decrease in manufacturing cost by approximately 24%, surpassing the target goal of 8%. It is up to the discretion of Stoneridge to determine which solution would be more optimal to meet their design and cost optimization goals.

Conclusions

Providing design modifications that will decrease the cost and weight of the Generation 2 IPM assembly is advantageous as it allows Stoneridge to offer competitive pricing within the current transmission braking models available in the automotive market. Additionally, the function of the IPM within the transmission is incredibly important for user safety. The solution provided by our team must be durable (surviving 300,000 cycles unlocking and locking the transmission), able to withstand the loading and temperature conditions of the transmission (tested to temperature spans of -40 to 120°C and vibrational loading with amplitudes up to 10 gs of force) and be non-corrosive (martials will not degrade from the transmission fluid). To accomplish this task, we analyzed potential material changes for each of the parts of the IPM assembly provided (the OAP, slider, Spacer, and inner alignment plate) and possible design and manufacturing simplifications or material elimination.

During concept generation we analyzed potential material changes and manufacturing processes, along with the possibility of material elimination and part consolidation. After thorough analysis of the material families summarized in Tables 8 and 9, we determined that the material PA6 13% GF best suits our requirements and specifications. From this material selection, our final design will result in a material change of the IAP, OAP, and the Spacer from HSLA Steel, grade 45 to PA6 13% GF. With plastic as our material of choice, we plan to use injection molding to produce the parts. The Spacer will have ribbing as a form of material elimination, which will be effective in reducing stress concentrations and improving the injection molding process. The topography will be used with the IAP and OAP to remove sections of excess plastic. These design changes were validated through different forms of testing including multiple FEA simulations, fluid flow analysis, and in the future by experimental testing. After FEA thermal analysis it was concluded that a material change to PA6 33% GF would be more effective due to its higher heat deflection temperature. From this analysis, we confirmed that adding holes to the IAP and OAP did not compromise the strength of the part. With the IAP, OAP, and Spacer injection molded in PA6, 33% GF, the next key step will be to implement metal inserts near the screw holes to keep part integrity during assembly. With that complete, the assembly will be ready for an injection molded prototype and physical testing at Stoneridge.

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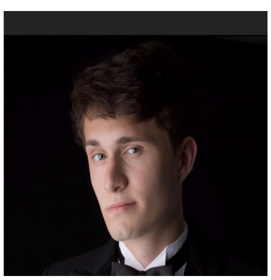
Biography

Shawlin Shahid: I am graduating from the University of Michigan with a Bachelors in Mechanical Engineering and a Minor in Computer Science and I am from Brooklyn, New York. I got interested in mechanical engineering through the desire to create and build medical devices. I originally wanted to go into pre-med, but mechanical engineering would allow me to go into a similar field while also giving me the background to create these devices. Some of my hobbies include crocheting (my goal is to crocheting a cardigan for myself), playing Pokemon, badminton, and karaoke. My current future plans are to go back to New York and work for the Department of Environmental Protection.

Matthew Wedeven: I am graduating from the University of Michigan with a Bachelors in Mechanical engineering and a Minor in Electrical Engineering. I am from Grand Rapids, Michigan, where I attended Grand Rapids Community College prior to transferring to U of M. What got me interested in Mechanical Engineering was being able to create and design solutions for everyday problems. I am a nerd for 3D printing and love designing models to print for friends, family and projects for school. I have just launched a site on Etsy selling a few of my designs. In my free time, I enjoy outdoor activities such as golf, weightlifting, pickleball, biking and snowboarding. After I graduate this May, I plan on moving to Texas where I will be working in the Engineering Rotational Development Program at Caterpillar.



Nathan Goldman: I will be graduating from the University of Michigan with a Bachelor’s degree in Mechanical Engineering. I am originally from West Bloomfield, Michigan, but grew up in Boca Raton, Florida. I became interested in mechanical engineering through my love of cars from a young age. I plan to move to Chicago after graduation and pursue a job hopefully in the automotive industry. My interests include basketball, golf, football, and I also am an avid enjoyer of video games.

Zack Jarski: I will be graduating from the University of Michigan with Bachelor degrees in Mechanical and Computer Science Engineering. I am from Rochester Hills, Michigan and have lived there almost my whole life. I was originally interested in Civil Engineering because of the design aspect but I realized that career path wasn’t what I was looking for. My interest in cars and electromechanical devices is what led me to mechanical engineering and computer science. After graduating I plan on staying in Michigan and hopefully finding a job where I can use both my mechanical and computer science degrees. My biggest hobby is running; I am currently an active member of the Michigan running club. I am also a video game enjoyer and took a computer game design class here.

Appendix A.

The top 20 ideas from our brainstorming process were consolidated below. Each idea was considered with our requirements and specifications listed in Tables 4 and 5. From this consideration, each idea was decomposed and explained why or why not we plan on utilizing this idea into our final report.

Moving forward, the outer alignment plate will be abbreviated as OAP and the inner alignment plate as IAP.

1) Reducing Spacer Thickness

To reduce the overall weight of the assembly, we came up with the idea to reduce the thickness of the Spacer. After obtaining a physical model, we realized that the thickness is set to allow for the slider to move through the assembly. Directly reducing the Spacer will not be a viable option, but other alternatives of reducing the weight of the Spacer could be useful.

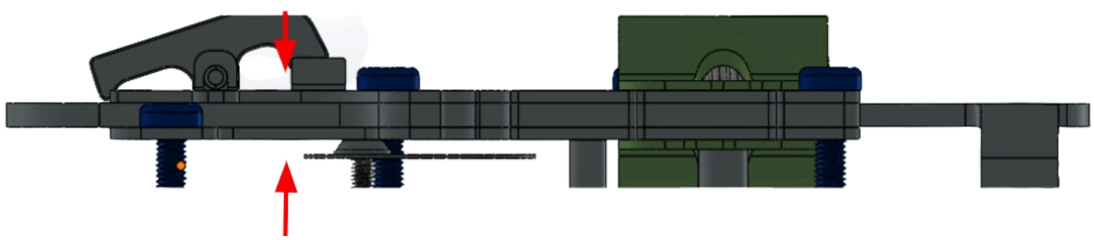


Figure A1. CAD image used to show the reduction of Spacer thickness in the assembly.

2) Remove One Spring

In order to reduce the number of parts and overall weight of the assembly, we thought of removing one of the two springs and replacing it with a single spring of the same strength. After discussing with our sponsor it was discussed that there are two separate designs for the IPM, one with just one individual spring and another with two, so it was determined that this is not feasible as a design change.

Figure A2. CAD image used to show the current two spring design.

3) Remove the Spacer, separate the parts with bushings

To reduce the weight of the assembly, we decided to remove the Spacer entirely and space the part with bushings where the screw bolts are placed. After discussing functionality of the part with our sponsor, the Spacer helps to guide the slider as it moves, helps to hold the spring retainers in place, and relieves stress from other parts of the assembly.

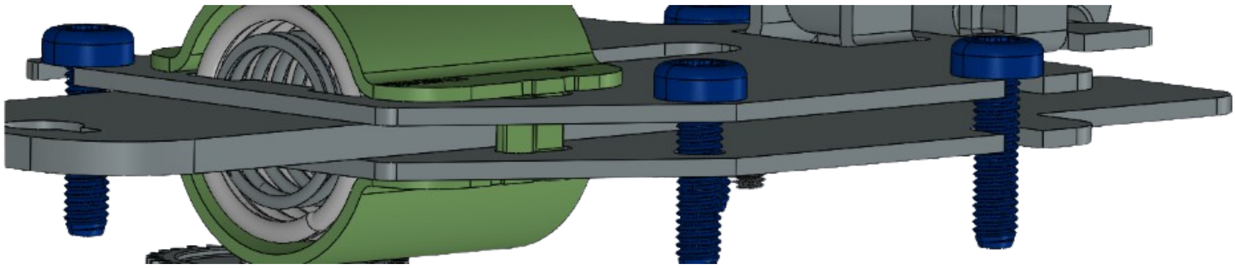


Figure A3. CAD image used to show the removal of the Spacer in the assembly.

4) Decrease Number of Screws

To reduce the cost by a small margin, we decided to decrease the amount of screws necessary to put the assembly together from a total of four, which are currently on the four corners of the assembly, to three.

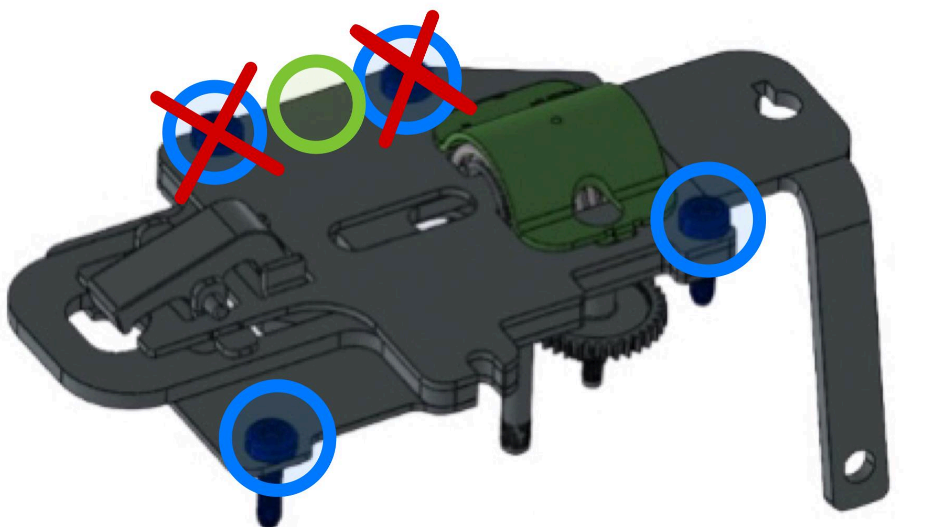


Figure A4. CAD image of placement of old screws. The previous screw positions are circled in blue, and the new placement is circled in green.

5) Reduce Thickness of Inner and Outer Alignment Plates

To reduce the weight of the assembly, and potentially the tooling cost, we decided to reduce the thickness of both the inner and OAPs, while also taking clearances into account. Upon further inspection it was determined by us and our sponsor that this would significantly change the manufacturing process and would not be realistic.

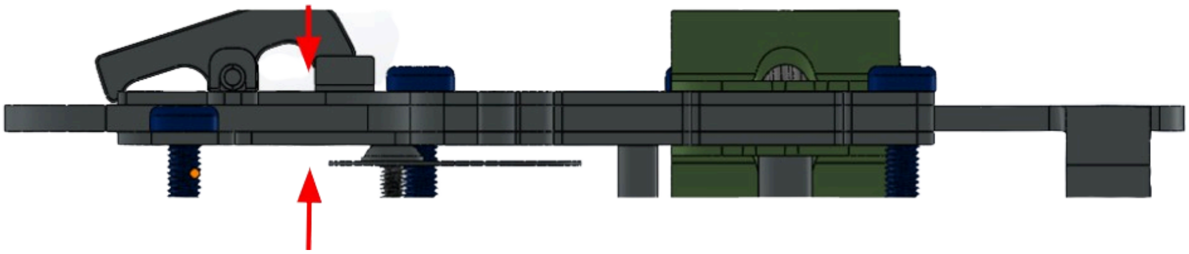


Figure A5. CAD image used to show the reduction of IAP and OAP thickness in the assembly

6) Topology used to remove material from the IAP, OAP, and Spacer

To reduce the weight of the assembly, we used topology to remove excess material from the IAP, OAP and Spacer. This material reduction will be viable for both steel and plastic fabricated parts. For steel, the parts will need new tooling, making it less attractive to Stoneridge for the small weight reduction. For plastic parts, we will net a significant weight reduction. If we consider this for our final design, we will need to make sure that there is uniform wall thickness between different geometries in the part due to limitations in the injection molding process.

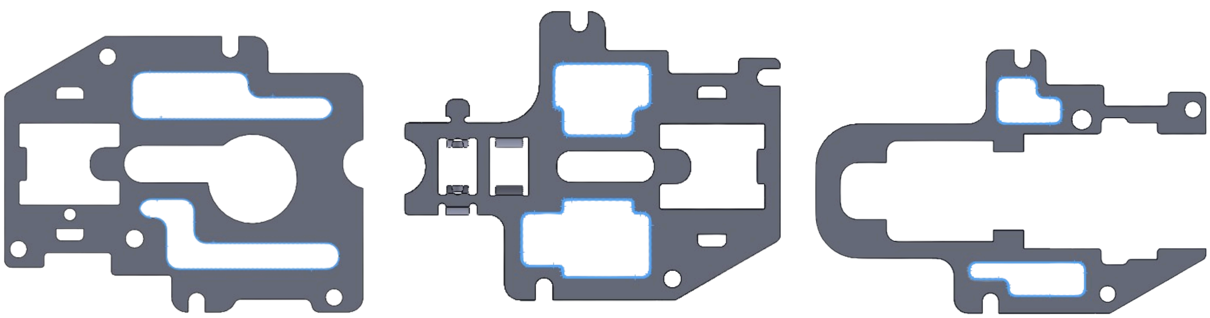


Figure A6. CAD image used to show the reduction of material in the IAP (left) OAP (middle) and Spacer (right).

7) Adding Ribbing on the Spacer

Another idea we thought of to reduce the overall weight of the assembly was to add ribbing features on the Spacer after it is changed to plastic. We would have to keep in mind that it's necessary to maintain a uniform wall thickness in order to not reduce structural integrity. After discussing with a plastics expert it was determined that this is indeed feasible as the thickness of a plastic part that is to be ribbed needs to be over 3 mm, and this Spacer is.

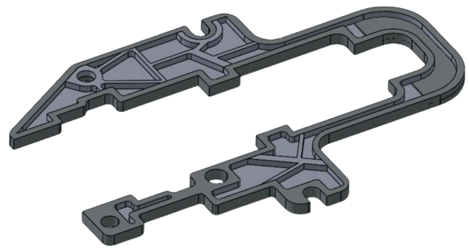


Figure A7. CAD image used to show an example of ribbing done on the Spacer

8) Change material of OAP, Spacer, and IAP to plastic

To reduce the weight and cost of the assembly, we wanted to look into changing the material of the OAP, Spacer, and IAP to plastic. Since the slider will have to be able to withstand a pull force of 400 N it is unlikely we would be able to change it from steel to plastic.

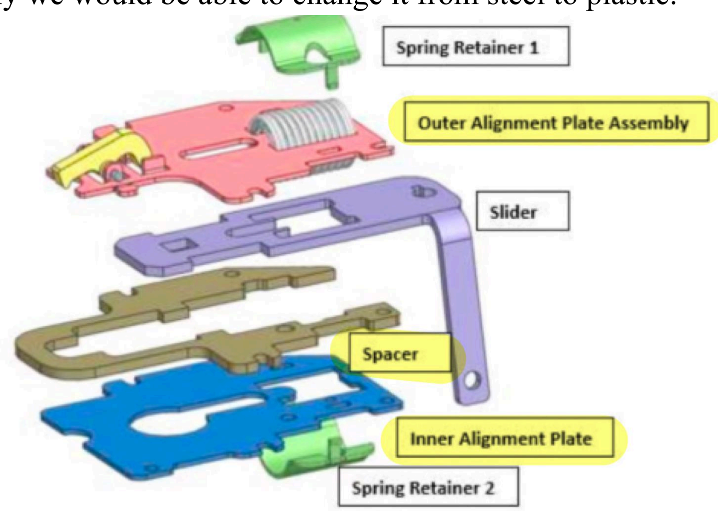


Figure A8. CAD images of the OAP, Spacer, and IAP are highlighted.

9) Part Consolidation of the IAP, and Spacer

To reduce the number of molds and total parts of the assembly, we decided to use injection molding to combine the IAP and Spacer into one part. After talking about the manufacturing process with Stoneridge, we will not be able to include the spring retainer in this assembly (as shown in the figure below). It will interfere with the tooling during manufacturing. When considering this idea, we will need to determine the optimal place for injection mold flow. Additionally, there might be an issue with non-uniform wall thickness that could increase the manufacturing cost.

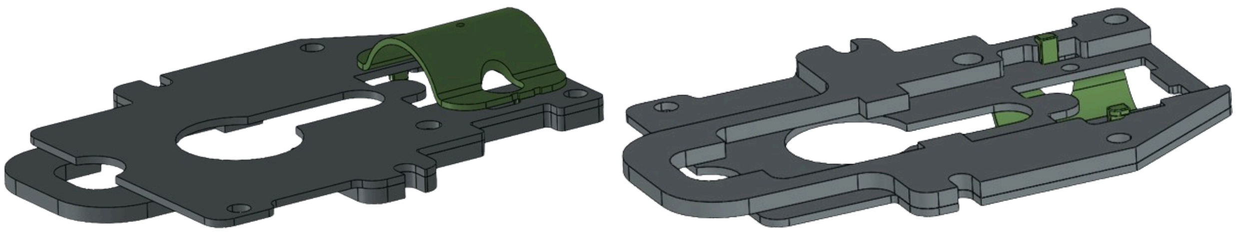


Figure A9. CAD images of the consolidated IAP, and Spacer.

10) Part Consolidation of the OAP and Spacer

To reduce the number of molds and total parts of the assembly, we decided to use injection molding to combine the OAP and Spacer into one part. After talking about the manufacturing process with Stoneridge, we will not be able to include the spring retainer in this assembly (as shown in the figure below). It will interfere with the tooling during manufacturing. When considering this idea, we will need to determine the optimal place for injection mold flow. Additionally, there might be an issue with non-uniform wall thickness that could increase the manufacturing cost.

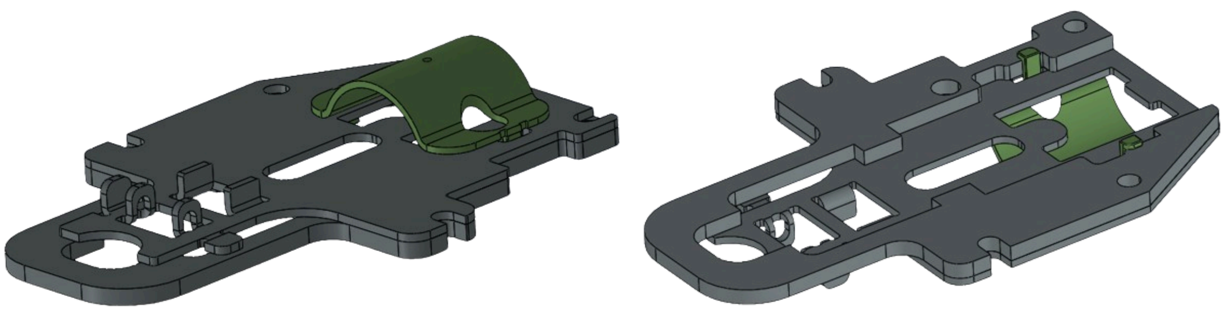


Figure A10. CAD images of the consolidated IAP and Spacer.

11) Part Consolidation of the IAP and Spacer with Ribbing

To reduce the number of molds and total parts of the assembly, we decided to use injection molding to combine the IAP and Spacer into one part. After talking about the manufacturing process with Stoneridge, we will not be able to include the spring retainer in this assembly (as shown in the figure below). It will interfere with the tooling during manufacturing. When considering this idea, we will need to determine the optimal place for injection mold flow. Additionally, there might be an issue with non-uniform wall thickness that could increase the manufacturing cost. To mitigate this and further reduce weight, we could rib the Spacer and the IAP. This would speed up manufacturing time since the plastic would require less time to cool, reducing the cost.

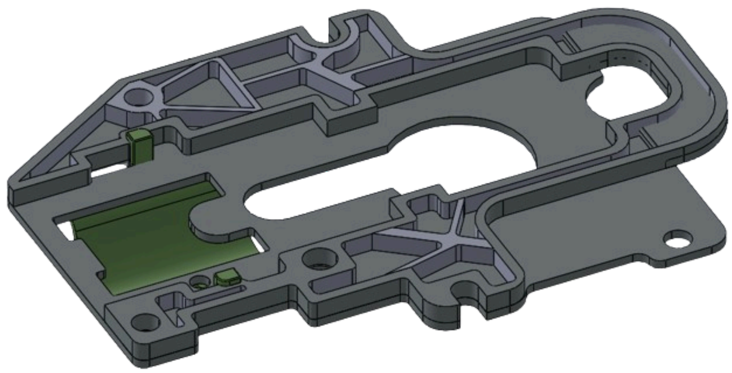


Figure A11. CAD images of the consolidated IAP and Spacer with ribbing.

12) Part Consolidation of the OAP and Spacer with Ribbing

To reduce the number of molds and total parts of the assembly, we decided to use injection molding to combine the OAP and Spacer into one part. After talking about the manufacturing process with Stoneridge, we will not be able to include the spring retainer in this assembly (as shown in the figure below). It will interfere with the tooling during manufacturing. When considering this idea, we will need to determine the optimal place for injection mold flow. Additionally, there might be an issue with non-uniform wall thickness that could increase the manufacturing cost. To mitigate this and further reduce weight, we could rib the Spacer and the OAP. This would speed up manufacturing time since the plastic would require less time to cool, reducing the cost.

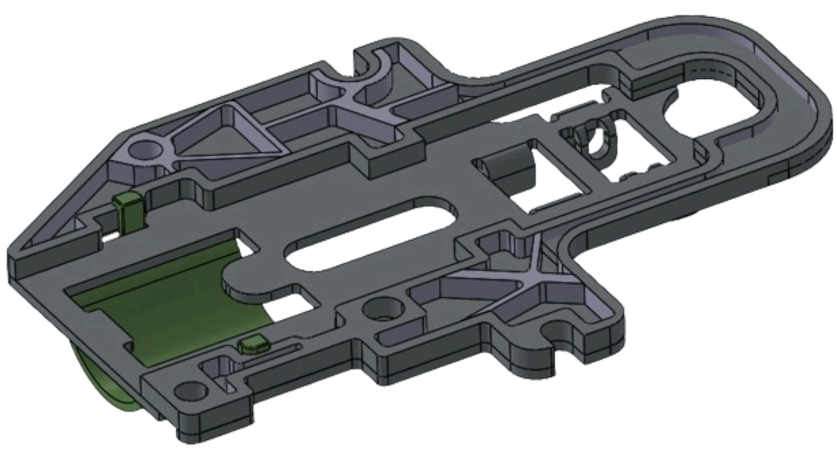


Figure A12. CAD images of the consolidated OAP and Spacer with ribbing.

13) Manufacturing using Waterjet for Steel

Rather than stamping the steel parts, we can potentially use a laser cutter or waterjet. This would allow for it to cut through several sheets of metal. Due to the high waste in abrasive, water and time since the process would be slower than sheet metal stamping. Although this idea will not work, it allowed us to think of other processes to use to manufacture the parts from plastic.

Figure A13. Waterjet machine in use[36].

14) Reduce Spring Retainer Size

Another idea we had was to reduce the overall length of the spring retainer. This could allow us to reduce both the cost and weight. When in discussions with our sponsor, it was determined that this would have issues with both manufacturing processes and structural integrity so we decided not to move further with this idea.

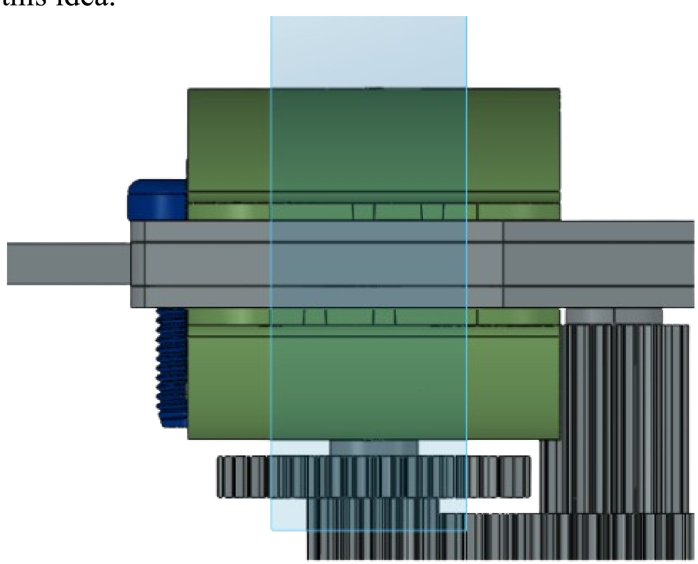


Figure A14. CAD image of the assembly with the highlighted region of spring retainer not being removed.

15) Manufacturing using Injection Molding for Plastic

If we end up changing the material to plastic we can use injection molding to create molds for the plastic parts for manufacturing. Although creating these molds would be an additional cost, in the long run it may have a net positive since the material cost for plastics tend to be cheaper than the material costs for steel.

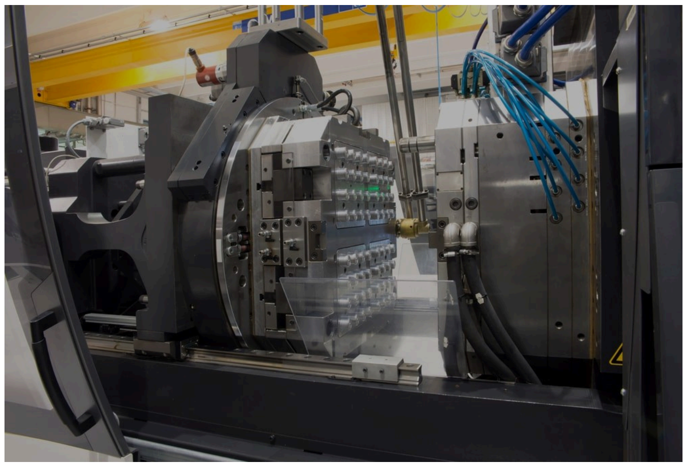


Figure A15. Injection molding machine in use[37].

16) Adding a protective Seal to remove steel coating

With a major issue presented by Stoneridge being the cost of the protective coating to prevent the steel parts from rusting, we came up with the idea to redesign the container to seal the inside off from the outside. This would allow the steel parts to be produced without needing to use the protective coating, however, it is not feasible. The slider needs to move in and out of the assembly, so friction would wear down the seal. Additionally, the transmission fluid provides lubrication to the parts, extending their lifespan.

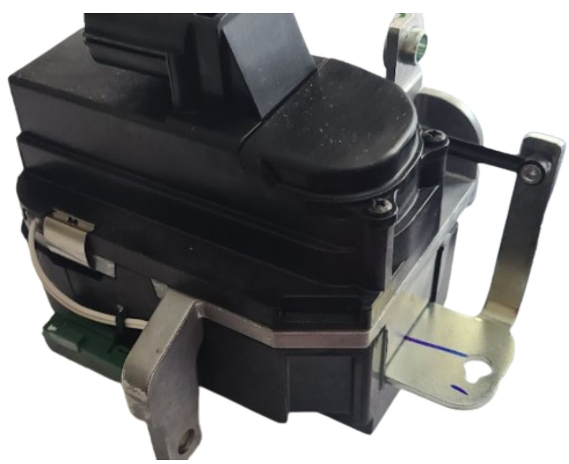


Figure A16. Image of the IPM assembly, the seal will isolate the inside from the outside.

17) Part consolidation of all parts of the assembly

Another idea we had was to combine all of the components using injection molding, as one singular part after they were all changed to plastic. This would save weight, cost, and significantly reduce the amount of steps in the manufacturing process. This was quickly ruled out as it was seen that the slider can not be converted from steel.

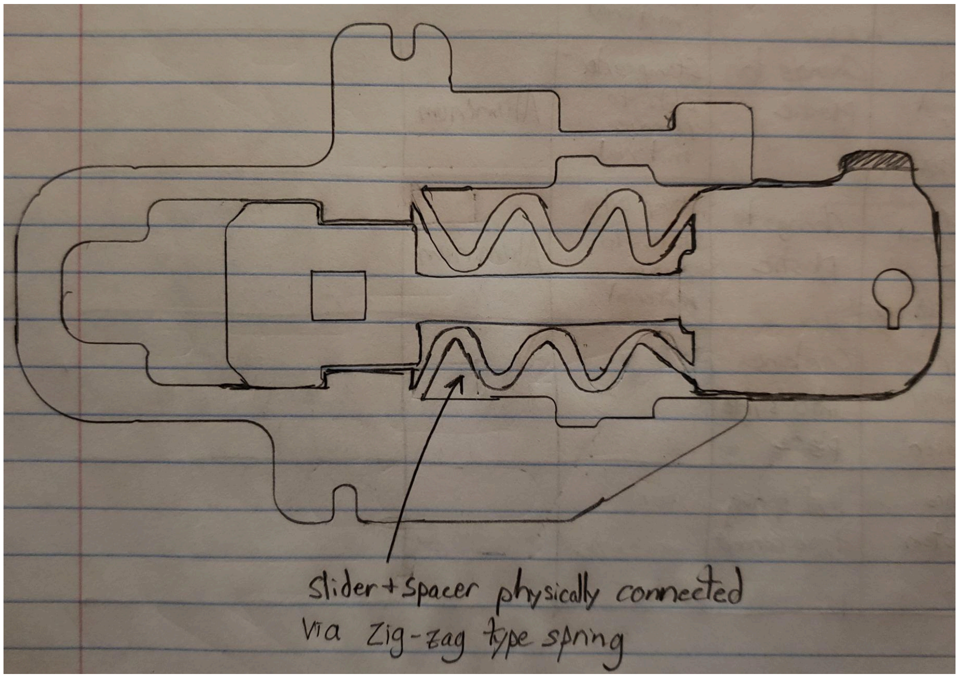


Figure A17. Sketch of all of the assembly consolidated as one piece.

18) Linear bearing to reduce friction when the slider actuates

To increase the longevity of the part, we thought of ways to reduce the friction between the slider and the IAP, OAP, and Slider. To achieve this, we thought of adding a linear bearing that the slider can use to slide on when actuated. The bearing would be continuously lubricated due to the transmission fluid.

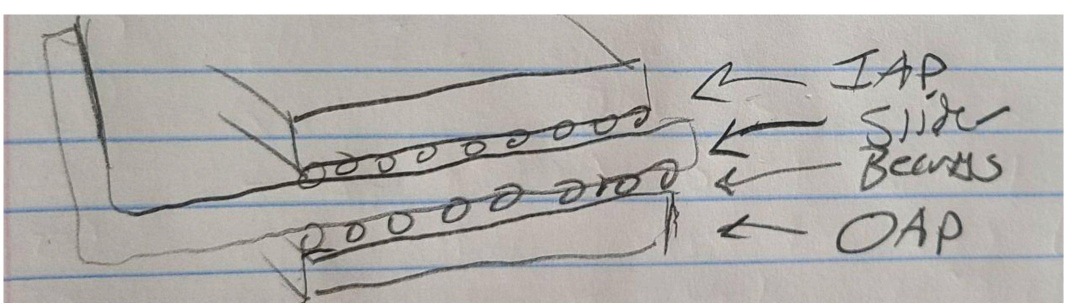


Figure A18. Image of the IPM assembly, the seal will isolate the inside from the outside.

19) Change the spring to a gear assembly

We were looking into the possibility of changing the spring mechanism into a gear mechanism. The gears would be powered by a motor and one of the gears would be attached to the slider that can be actuated. One main drawback is that this idea requires a complete overhaul of the existing design and would probably be difficult to implement when considering how the sub-assembly provided to us interacts with the entire IPM assembly.

Figure A19. Sketch of the Gear System that would take the place of the spring assembly

20) Add a bushing to reduce wear on the slider

To reduce wear over time due to the actuation of the slider, add a press fit bushing into the highlighted hole in the figure below to interface with the bolt. This would add a step in the assembly process, increasing the cost of the part, but it would improve the part quality.

Figure A20. CAD sketch highlighting where the bushing will be placed. An example of the bushings can be found on the right[38].

Appendix B.

To develop a reliable FEA model, the IAP, OAP and Spacer were combined into one part and imported into Hyperworks Software. A 2D, first order quadrilateral mesh (elements shown in

Figure B1 [44]) was used to mesh the surface of the model. The loading was simulated using an initial 2D mesh, then was refined based on stress concentration results for more accurate analysis. With the 2D mesh refined, a 3D tetrahedral mesh (elements shown in Figure B1.) was generated. In 2D meshes, a thickness must be applied.

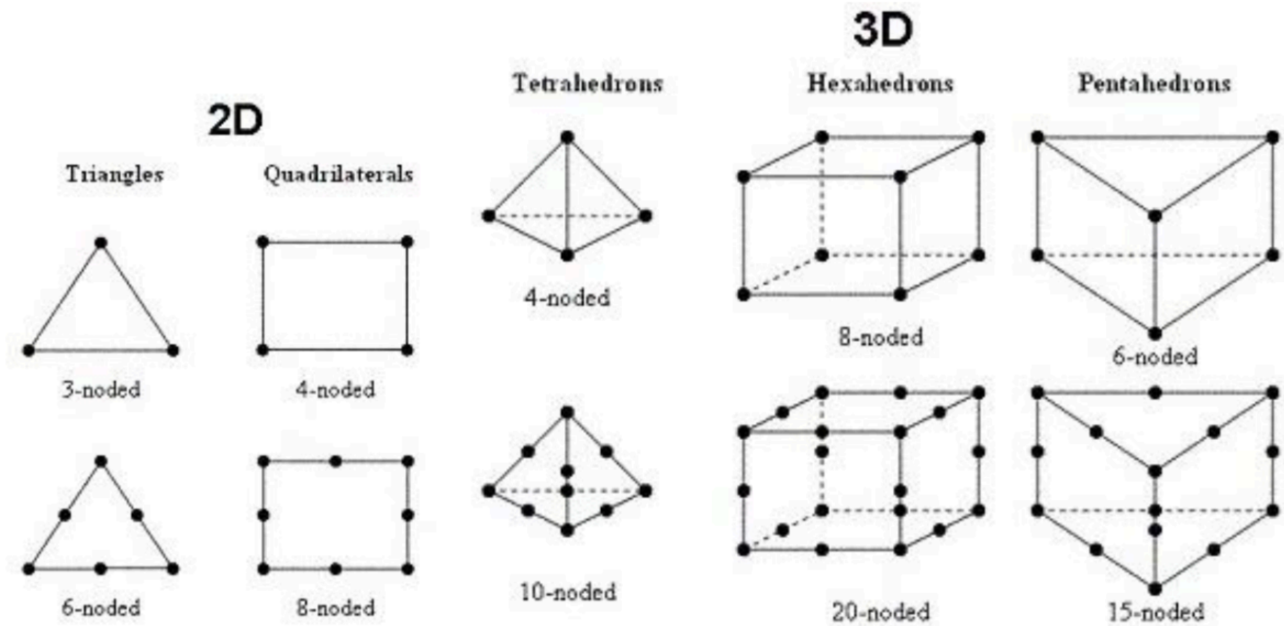


Figure B1 [44]. Diagrams of the various 2D and 3D elements available to mesh with.

Since our part is not uniform in thickness, the previous mesh needs to be deleted, with a new 2D mesh (called a skin) created from the current 3D mesh. The thickness of this mesh was calculated based on the Eq. B1 [44] below relating the thickest section of the part, *t*, a constant factor of 1000 to the skin thickness, *T*.

$$T = \frac{t}{1000}$$

A depiction of the skin mesh over the 3D element mesh is shown in Figure B2. The blue represents the 2D skin elements and the Purple represents the 3D tetrahedral elements.

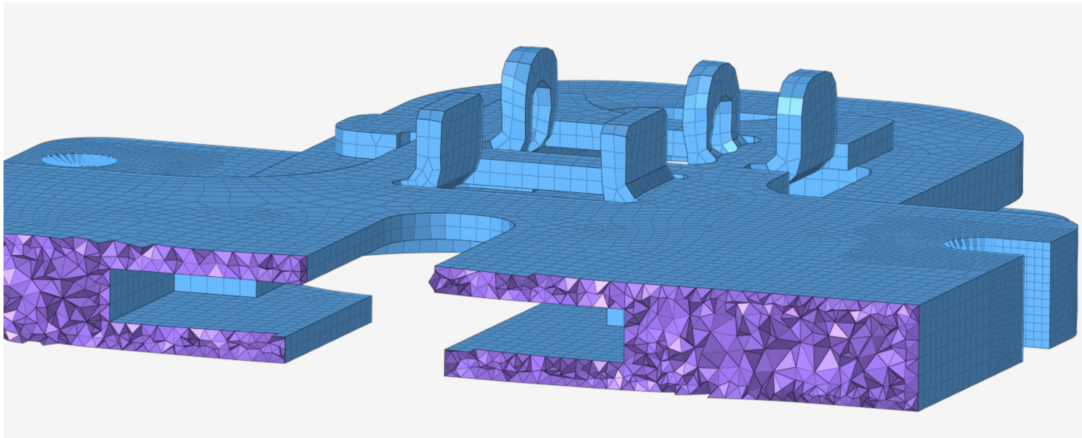


Figure B2. Diagram of the 2D (blue) and 3D (purple) elements used for our model.

For most thin walled models, 2D meshes are used to model the stress and displacement, with 3D elements not being accurate for the level of complexity they add to the model. In thin walled models, stress will generally be concentrated on the external surface, so 2D elements will have a more accurate result. The 3D element stress is concentrated at the center of the element, so the stress and displacement for an external measurement will always have an error of at least ½ the element thickness. Although not used for results, the 3D elements in the mesh are necessary in confirming the validity of the model. Firstly, if there are any cracks or openings in the initial 2D mesh or surface, a 3D mesh will not be generated, showing the user that there is an error in their mesh. An example of this can be seen in Figure B3. The red line on the model is showing that there are openings in the model, preventing a 3D mesh from generating.

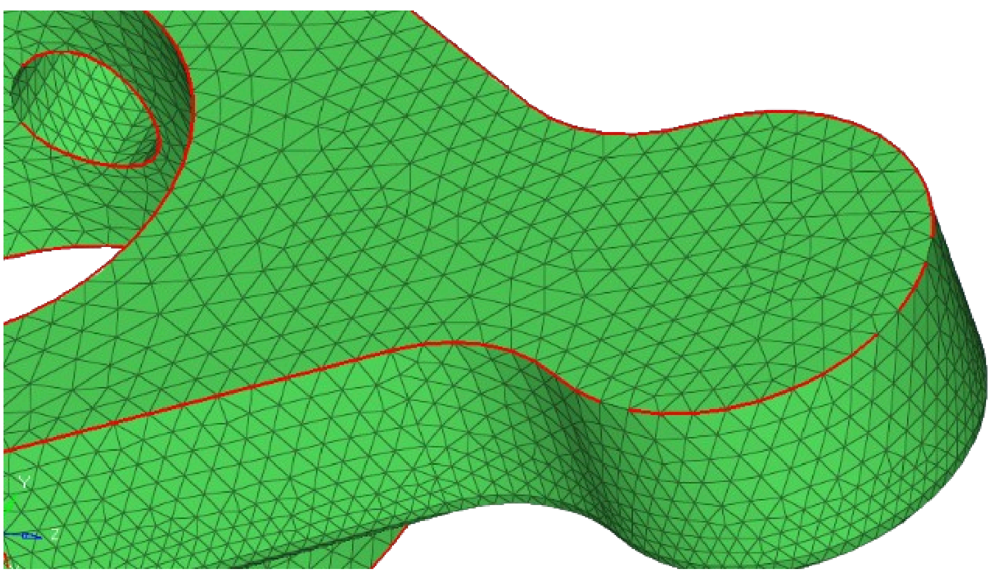


Figure B3 [45]. Diagram of an FEA model that is not enclosed. The red lines represent microcracks in the mesh or surface.

Additionally, although the measurements from the 3D elements will be inaccurate to ½ the thickness of the element, the result from both the 2D and 3D models should be within the same range of values. If they are, then everything is performing correctly. Most importantly, the 3D elements allow a skin to be placed over the model. This will allow the more accurate 2D elements to map the topography of the surface that does not have constant thickness. This is essential to our project, since we have a part with complicated geometry.

With the mesh generated, we wanted to use first principles to verify the performance of our model. To do this, we used Eqs. B2, B3, and B4 [43] obtain the maximum stress and displacement from a force acting on a built in beam depicted in Figure B4. From these calculations, the maximum stress in the beam is 65 MPa, and the maximum displacement was 0.0033 mm.

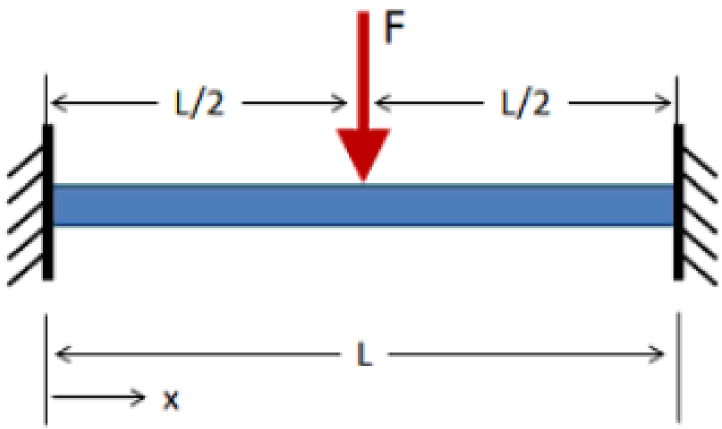


Figure B4. Diagram of a built-in beam for point loading.

$$I_x = \frac{bh^3}{12} = \frac{1.57(4)^3}{12} \qquad \text{Eq. B2}$$

$$\Delta = \frac{FL^3}{192EI_x} = \frac{100 \cdot (22)^3}{16 \cdot 200000 \cdot 1.57 \cdot (4)^3} = 0.0033\ mm \qquad \text{Eq. B3}$$

$$\sigma_{max} = \frac{M_{max} \cdot y}{I_x} = \frac{100 \cdot 22}{8 \cdot 1.57 \cdot (4)^2} = 65\ MPa \qquad \text{Eq. B4}$$

Due to convenient geometry of our model, we were able to simulate the built-in beam example above in Figure B4. Figure B5. shows how we modeled the built in beam for the IPM.

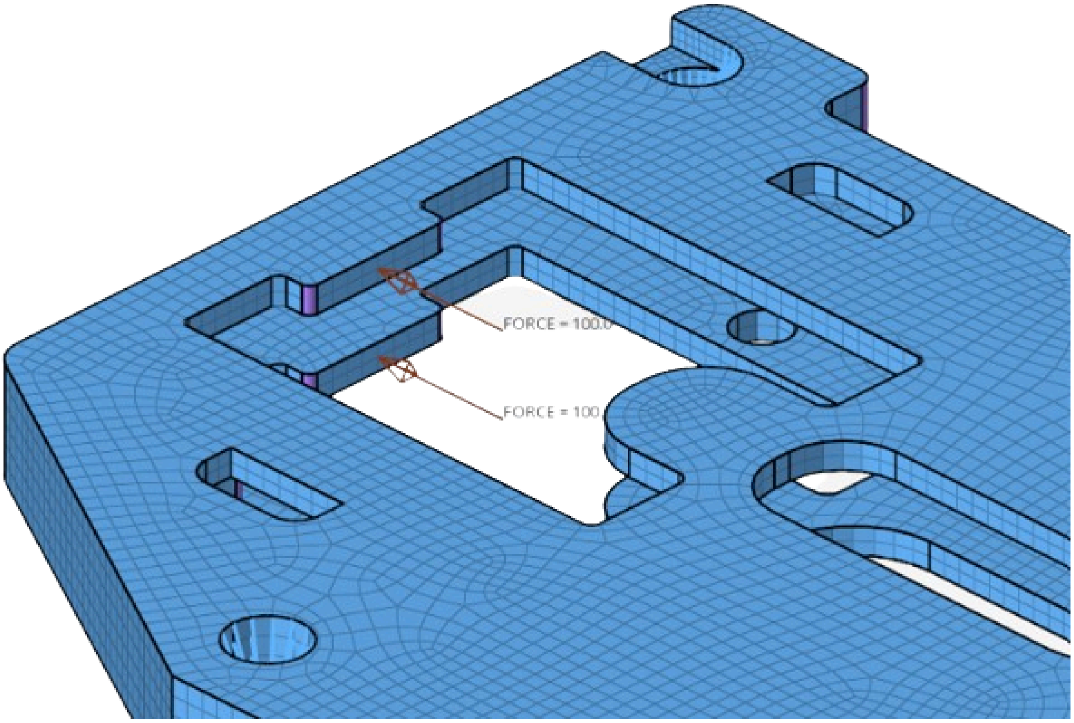


Figure B5. Diagram showing the loading scenario for a force acting on a built in beam using the IPM geometry.

Using the same material properties as in Eqs. B2, B3, and B4, we simulated the loading scenario in Figure B5. Simulation results for stress and deflection were depicted in Figures B6 and B7.

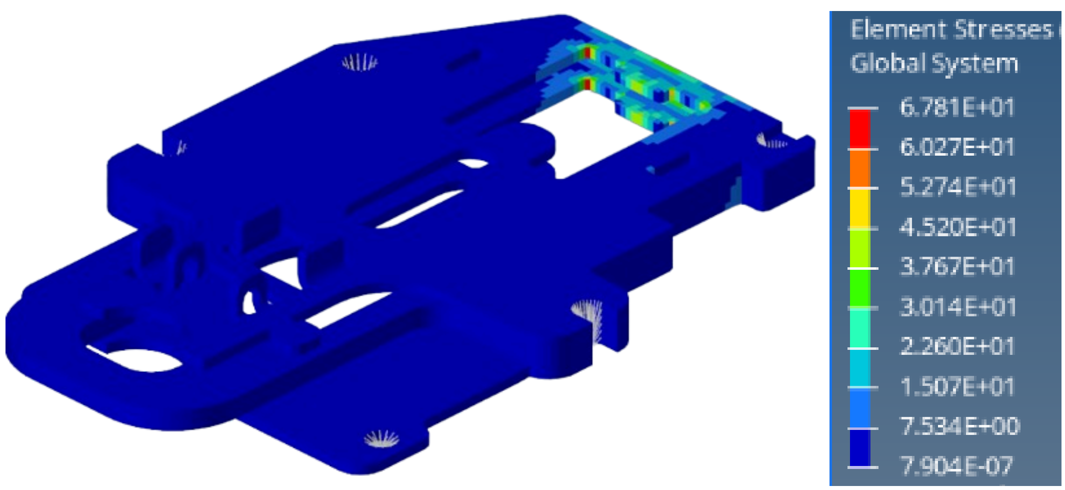


Figure B6. Stress (magnitude in MPa) contour plot for loading scenario in Figure B5. The maximum magnitude of stress was 67 MPa, concentrated at the fileted edges.

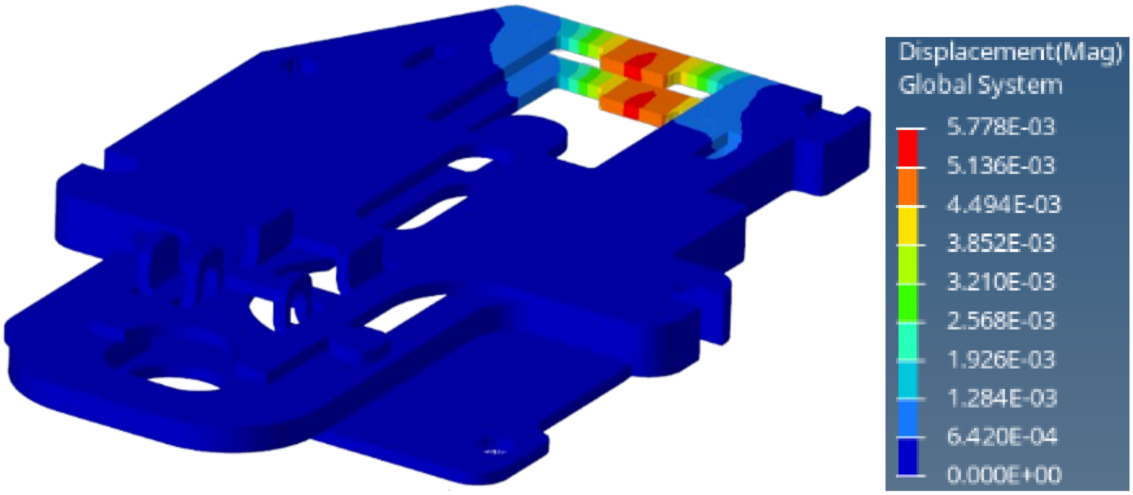


Figure B7. Displacement (magnitude in mm) contour plot for loading scenario in Figure B5. The maximum magnitude of displacement was 0.0058 mm, concentrated where the load was placed.

Both stress values from first principles and from the simulation were compared in Table B1 and matched one another. The displacement values were on the same order of magnitude, however they are slightly different. This could be due to the deformation of elements being inaccurate at larger mesh sizes. From this, the model is verified for static loading.

Table B1: Maximum magnitude of displacement (mm) and stress (MPa) due to the loading scenario in Figure B5 and the theoretical values calculated from Eqs B2, B3, and B4.

	First Principles	**FEA Simulation**
Max Stress [MPa]	65	68
Max Displacement [mm]	0.0033	0.0058

With the model verified for static loading conditions, we wanted to verify the thermal loading as well. To do this, we used Eq. B5 [43] to obtain the maximum stress and displacement from a force acting on a built in beam depicted in Figure B8 [43]. From these calculations, the maximum stress in the beam is 120 MPa.

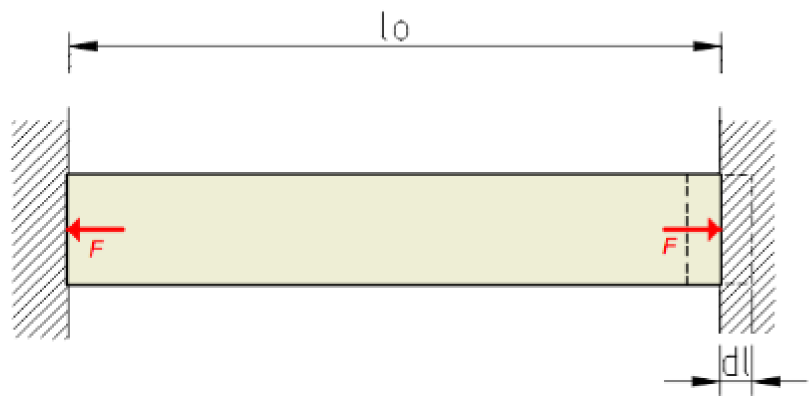


Figure B8. Diagram of a built-in beam for temperature change.

$$\sigma_{th} = E\alpha\Delta T$$

$$\sigma_{th} = \left(1.2 \cdot 10^{-5}\right) \cdot 200000 \cdot 50 = 120 \ MPa \qquad \text{Eq. B5}$$

Due to convenient geometry of our model, we were able to simulate the built-in beam example above in Figure B4. Figure B9. shows how we modeled the built in beam for the IPM. The red lines indicated were all 6 degrees of freedom were constrained to allow the model to simulate a built in beam. Without them, the stress would be significantly lower than the expected 120 MPa due to the part expanding to the left of the model.

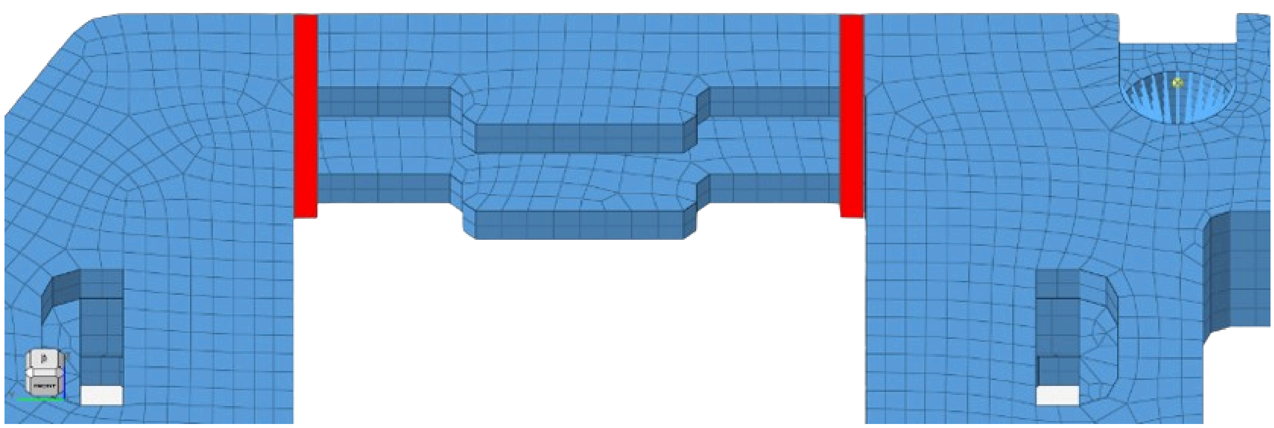
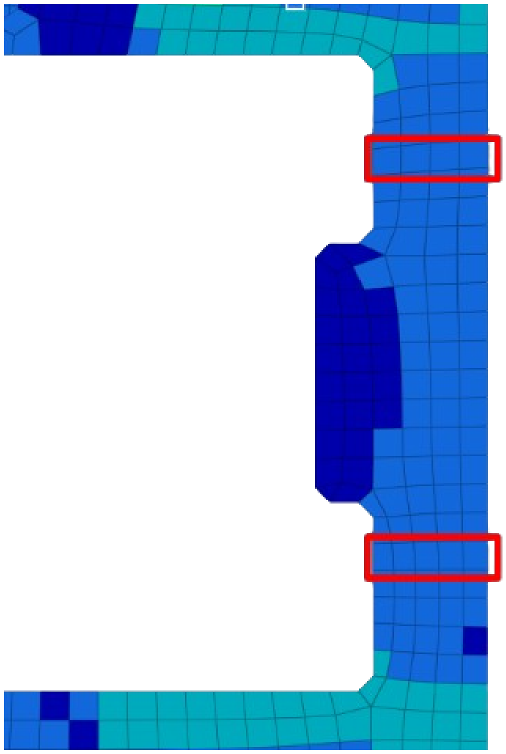


Figure B9. Red depicts where the model was constrained in all 6 degrees of freedom to simulate a built in beam temperature change scenario.

Using the same material properties as in Eq. B5, we simulated the loading scenario in Figure B9. Simulation results for stress were depicted in Figures B10.



Element	Stress [MPa]
1	132
2	145
3	108
4	145
5	148
6	142
7	129
8	106

Figure B10. Stress (magnitude in MPa) contour plot for loading scenario in Figure B9. The average maximum magnitude of stress was 132 MPa..

Both stress values from first principles and from the simulation were compared in Table B2 and matched one another. From this, the model is verified for thermal loading.

Table B1: Maximum magnitude of stress (MPa) due to the loading scenario in Figure B9 and the theoretical values calculated from Eq. B5.

	First Principles	FEA Simulation
Max Stress [MPa]	120	131

From both loading scenarios in B5 and B9 matching the theoretical calculations in Eqs. B2, B3, B4, and B5, the model is verified for static and thermal loading simulations. The model is performing correctly, and all units are correct. This will allow us to be confident in the results from our simulation results in the coming weeks.

Appendix C.

To calculate the cost of injection molding we used methods from the *Product Design for Manufacturing and Assembly* [46] textbook. The overall equation, shown below in Figure C1, takes into account the material cost, the operating cost, and the cost for the mold itself.

$$c_{im} = c_{mt} + \frac{c_r t_s}{3600n} + \frac{c_{md}}{N_t}$$

Figure C1. The cost per part to injection mold comes down to the material cost, the operating cost of the machine, and the cost of the mold.

Figures C2 and C3 below show the full calculation for the IAP and the OAP/Spacer combination part respectively.

Inner Alignment Plate										
N_t	c_md		t_s (s)		c_r ($/h)		n	c_mt ($/unit)		c_im ($/unit)
	l (cm)	9.66	P_j (kW)	7.5	A_p (cm2)	46.5525		c_p ($/g)	0.00452	
	w (cm)	7.45	k	1	p_j (bar)	1200		ro (g/cm3)	1.390	
	A_d (cm2)	553.617	h_max (mm)	1.57	D (cm)	0.157		V (cm3)	7.3311	
	h_d (cm)	15.157	T_x	140	F (kN) >	500.00		f_r (%)	0.55417	
	R ($/h)	60	T_m	100	v_s (cm3) >	85.000				
	n_s	0	T_i	300	L_s (cm) >	23				
	n_i	0	alpha (mm2/s)	1.3						
	n_u	0	t_d (s)	1.9						
	M_e	17.05734812	t_f	0.3646						
	M_po	13.53002956	t_c	3						
	M_x	5.099839892	t_r	2.5982						
	X_i	0								
	N_sp,cor	0								
	N_sp,cav	9								
	X_o	0.9								
	M_s	5.353082636								
	M_t	1.529951968								
	M_tex	1.784360879								
	M_p	0								
	f_s (%)	0.15								
	f_t (%)	0.30								
	f_p	0								
	c_b	1739.038786								
	c_c1	2661.276784								
	4400.31557				30					
20000	6070.68		5.9628		42.7		1	0.07158		0.4458

Figure C2. Cost calculation for the IAP.

Outer Alignment Plate + Spacer										
N_t	c_md		t_s (s)		c_r ($/h)		n	c_mt ($/unit)		c_im ($/unit)
	l (cm)	11.00	P_j (kW)	7.5	A_p (cm2)	44.5667		c_p ($/g)	0.00452	
	w (cm)	7.45	k	1	p_j (bar)	1200		ro (g/cm3)	1.390	
	A_d (cm2)	583.7	h_max (mm)	2	D (cm)	0.475		V (cm3)	14.3686	
	h_d (cm)	15.475	T_x	140	F (kN) >	500.00		f_r (%)	0.39986	
	R ($/h)	60	T_m	100	v_s (cm3) >	85.000				
	n_s	0	T_i	300	L_s (cm) >	23				
	n_i	0	alpha (mm2/s)	1.3						
	n_u	0	t_d (s)	1.9						
	M_e	16.68957384	t_f	0.6436						
	M_po	13.09527299	t_c	3						
	M_x	62.8157454	t_r	2.6912						
	X_i	1.5								
	N_sp,cor	15								
	N_sp,cav	50								
	X_o	5								
	M_s	13.89008883								
	M_t	18.84472362								
	M_tex	4.630029612								
	M_p	0								
	f_s (%)	0.15								
	f_t (%)	0.30								
	f_p	0								
	c_b	1785.69587								
	c_c1	7797.926058								
	9583.621928				30					
20000	13221.57		6.3348		42.7		1	0.12637		0.8626

Figure C3. Cost calculation for OAP/Spacer combination part.