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TORAY Improved Nip Roller Design

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Team 2: Improved Nip Roller Design

Final Design Report

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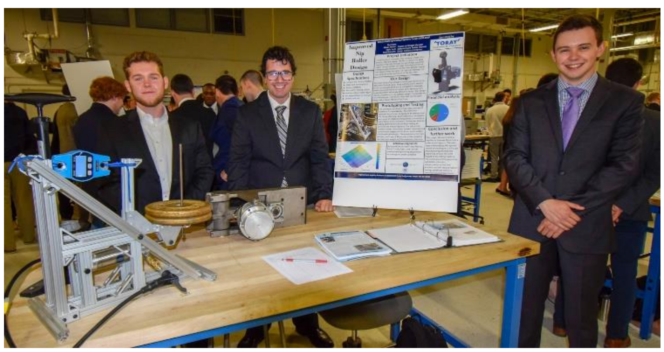

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Abstract

The Toray Plastics plant in Kingstown produces plastic films used for food packaging. The production process involves using high speed rollers to roll film to prepare for shipment of the product. However, there is a maximum speed that the rollers cannot exceed; higher speeds result in vibration between the film roller and the roller that supplies force to the film roller, known as the nip roller. The nip roller is used to force air out from between each layer of film. When vibrations occur, air pockets form in the rolls that cause the plastic to roll unevenly, ruining the product. The solution proposed by Toray is to redesign the nip roller mounting arm to eliminate vibrations at higher speeds.

The process of solving this problem began with background searching. Literature and patent researches were conducted to explore previous inventions and articles related to production rollers, and then ninety conceptual designs were created by the team. Of these ninety designs, three were chosen as the most effective. Through engineering analysis, the diagrams, and simulations, a final design was developed. The new design involves the use of torque to create added force at the point of contact between the nip roll and the customer roll. The redesigned arm utilizes a horizontal beam that is loaded with weight. Along with this weight, an air piston is mounted to the horizontal arm, which greatly increases torque at the pivot point. The added torque at the pivot point is translated vertically up to the point of contact, where nip force is maximized.

To prove that this design works, a prototype was built, made almost entirely out of 8020 Aluminum, excluding connecting pieces and a few bolts. The redesigned arm is simulated by two pieces of 8020 aluminum that are connected at a 90-degree angle. Five- and ten-pound barbell weights are loaded at the end of the horizontal piece, and a 2” compressive air piston is mounted to the bottom of the same piece. The combination of these two forces mimics the effects of heavier weights and a larger piston that will be used in the full-scale model. The prototype was built at half scale of the full-sized mounting arm and it is secured in a very stable cradle made of 8020 Aluminum. Prototype testing produced data that, when scaled up, exceeds the target nip force that Toray plastics requested to eliminate arm vibrations at higher roller speeds.

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List of Acronyms and Variables

d	Diameter of piston
l	Length of beam
w	Weight of block
p	Pressure (in psi)
h	height of nip arm
r	height of piston attachment below center of rotation
Fn	Nip Force
Er	Error in measured nip force

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Introduction

The goal of this project is to design an improved mounting system for a nip roller mounted inside a slitting machine. Toray plastics approached the university with this project because their current system is not capable of keeping up with the higher speeds, they are hoping to run their manufacturing line at. By the end of this project they would like us to design and prove a new system to reduce their manufacturing defects, allow them to run at higher speeds, and simplify their overall operating system.

The nip roller is an important to overall operation of the slitter because it controls the winding of material back onto a roll after it is cut. In this case the material is a thin, clear plastic film, four meters wide. As the material is wound on a roll the nip roller pushes against the roll, ensuring the material lies flat and there are no air bubbles in the roll. Air bubble are very problematic for Toray because the air can act as a lubricant between the sheets of plastic, causing them to slide out of the roll, creating a messy roll. These manufacturing flaws are called ‘burps.’ Burps are cause inadequate nip force and increased running speeds. As speed increase force must increase as well.

When Toray purchased the slitting machine and began using it over 20 years ago burps were not an issue due to the slow operating speeds. The nip roller was mounted on single arms on each side, each powered by one pneumatic piston. As the company expanded and demand increased operating speeds had to be increased and burps became more problem. The engineers traced the problem back to the inadequate nip force and solved the problem by mounting a second identical arm on each side of the roller. While this increase the force it also doubled the amount of air needed for operating as well as adding problems of alignment between all four arms. As the speeds continued to increase, burps returned much sooner than expected, showing that doubling the number of arms did not truly double the force.

Our design uses one single arm to prevent alignment issues as well as new pistons to decrease air loss. Each arm will be able to provide more the 750 lbs. of nipping force to increase the operating the speed of the stiller to upwards of 2,700 feet per minute. Additionally, it will mount to the same track system as the current design and be sized so it holds the nip roller at the same optimum height and does not interfere with any other components. It will also be run off the same pneumatic control system, limited to 80 psi.

Patent Search

As part of the research process the team also conducted a patent search to see other systems solving similar problems. We were unable to find any currently active patents that applied directly to our design, but we were able to find some that performed a similar operation in a different context. We also found a patent from 1978 for a nip roller system, confirming our suspicion that most innovations in this field happened well over 30 years ago, meaning no current patents. Below are listed the most interesting patents we found and why they were useful to us.

Patent no. 7588135B1: Roller Mount for a Conveyor Belt. This patent represents the mounting strategy for a conveyor belt roller. Although this patent was not used directly, it allowed for a creative idea base that we used when formulating a way to mount the high-speed nip roller for our project.

Patent no. 7,665,175 is a patent regarding a lint roller handle. The handle of a lint roller may be much smaller than the roller handle that we are designing, but it serves a similar purpose. This patent could show us information on basic design specifications and restrictions for our roller arm.

Patent no. 6,006,874: This patent is useful because it involves a flat plate that can absorb mechanical energy. The ideas can be useful for helping reduce vibrations in the nip roller, which is one of the key features Toray is looking for in their design change.

Patent no. 9,950,560: Although the application of this patent is unrelated, a train wheel is still a large mass that rotates rapidly. The high-speed rotation is key in this patent. This patent is related to reducing vibrations in train wheels, creating a damping system in the wheel and its supports. These dampers could relate to this project in that dampers can help reduce vibrations in the mechanical system.

Patent no. 4150797A Method and device for controlling contact pressure on touch roller in sheet winder. This patent is from 1978 so no longer active but it is one of the only ones we found that directly applied to our project. The system described here could not be directly implemented into Toray’s system, it was a good source of inspiration.

Evaluation of Competition

The main reason that Toray was interested in this project is because no effective solutions to their problem exists on the market. While nip rollers are common on slitting machines and in other winding operations, each is very specifically designed for its intended function. Toray has many other winders in their own facility, all of which use similar roller designs, but these parts are not interchangeable between machines. Other solutions to that exist are specialized convex rollers that push air to the outside on the customer rolls. Concave rollers are extremely expensive making it not a cost-effective solution for Toray, who already own at least half a dozen of the straight rollers. Toray would also need a 4-meter-long roller for this application, meaning that to have the same slope for pushing air as a short roller, the center of their roller would have to be much wider than the ends, creating problems of the ends not contacting the other roll at the start of winding. It would not be effective until there was enough film wrapped around the customer roll for the center of the convex nip roller to compact the film the amount it bulges out. This also increases the nip force requirements.

The only other solution that could also be interested in fixing this problem is the slitter's manufacture. The original manufacture has gone out of business, but another company bought their designs and now makes aftermarket parts. Toray reported that they contacted this company and they were uninterested in working on a solution because it is a very specialized application that most of their customers do not need. The other solution that exist is the one Toray came up with, combining two of the existing arms on each end. They have already proven that this is ineffective at doubling the original nip force and does not meet our design specifications. Overall this leaves redesigning the arms the most effective option and this team the only one that is attempting to do it.

Design Specifications

The new arm design must be mounted in the same location and it must hold the nip roll at the same height as the original, shown in figure 1. The improved arm also must increase the effective nip force from the original, to allow for faster operating speeds from 2,200 ft/min to 2,700+ ft/min. The implemented design must be reliable and built to last many years of continuous cycles. It also must be pneumatically operated to a maximum of 80 psi to interface with the existing control system. In the current design, leaks in the air cylinders may be causing deceased force, leading to vibrations. Additionally, the adaptor for the bearing of the current mounting arm is a potential source for vibration and misalignment and should be eliminated in the new design. The current mounting arms are made from a carbon steel, and Toray has reported that this has been reliable in their production line. Lastly, the mounting arm must have the same range of travel (approximately 2.5 inches) as the current design.

Table 1: Design Specifications

Product Name	Single Arm Nip Roller
Basic Function	Supports the nip roller onto the customer roll, and applies a nip force to keep air out of the customer roll
Key Performance Targets	750+ pounds of nip force
Roller Placement	Roller must be mounted in same location and have same travel as original (25 inches above track, 2.5-inch travel)
Air Pressure	Under 80 psi
Mounting	Attach to existing tracks on slitter
Size	Must fit in the space show in figure 1

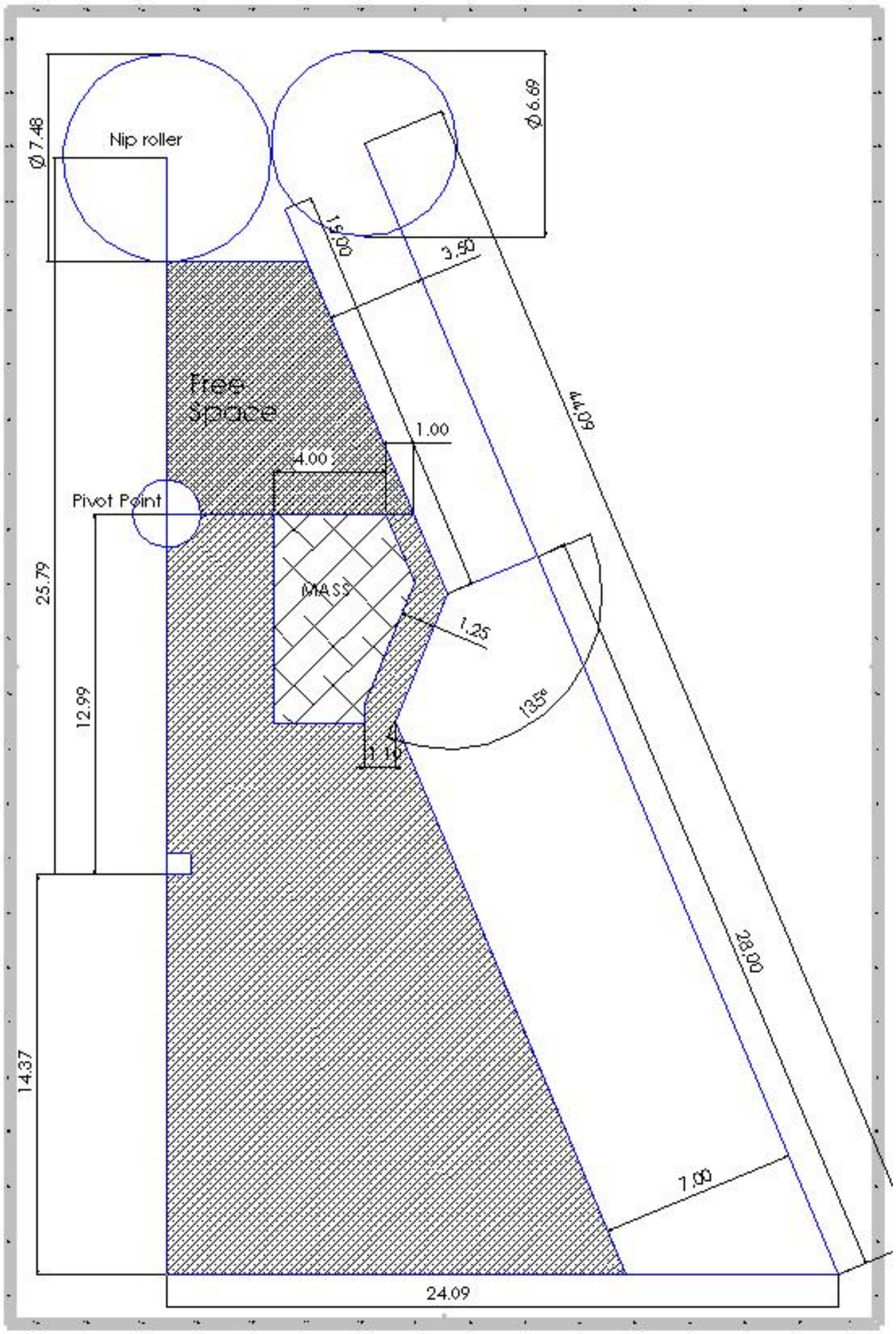


Figure 1: Space constrains for design

Conceptual Design

When generating concepts our priority was generating as much nip force as possible. We did this primarily in two ways. The first was the improving the mechanical advantage between the piston and the nip roller. The second was augmenting the force of the piston with an additional source of force, such as a second piston, springs, or other inputs. The secondary goal was vibration damping. We came up with a few ways of doing this, mostly involving elastomer components or commercially available dampers. After meeting with Toray, we abandoned most damping ideas due to wearing concerns and decided that be increasing the nip force enough, so we could remove the source of vibrations, air bubbles. Listed below are all our designs with a brief summary and analysis. The top 5 were then analyzed further in competitive analysis on page 40.

<u>Mason Fraizer's 30 concepts:</u>

1. Front mounted piston with in series damper. This design was a simple way to increase leverage and add a pre-manufactured damper.
2. Front mounted piston and damper in parallel. This design was like number one except the damper would dampen the movement of the piston to improve vibration response
3. T lever with front and rear double pistons. Basic strategy to double the number of pistons and therefore force by mounting one on each side. Increased leverage as well.
4. Longer straight bar with single piston. This design modified the original system to improve leverage and mechanical advantage. High torque on the bar around the pivot was a concern.
5. Class 2 long lever with low center of rotation. A variant of design 4 but with the pivot and piston switched. However, this lowered the mechanical advantage and was therefore ineffective.
6. Stationary mounted piston with extra link. This idea improved mechanical advantage through an extra link between the piston and the main lever. The piston was also mounted stationary to limit wear on hoses.
7. Front mounted piston with damped hinge. This design was a variant off 1 and 2

but the pivot point was damped to reduce vibrations traveling through the arm. This made controlling the nip roll high unpredictable though which was more of a priority.

8. Scissor damper with middle piston. This design used a 4-bar linkage with a diagonal damper to create the upper arm structure. It suffers the same flaw as 7 with height consistency.

9. Parallel bar mount. The use of a parallel bars as the lever reduces force at each joint and creates a large area at the top to mount the roller. Does not improve the mechanical advantage.

10. Back mounted piston with double track system. This design adds a second mounting point to the machine to add a better mounting point to generate force. Toray does not want the base machine altered however so this was impractical.

11. Modified scissor with base piston. Uses a 4-bar linkage to increase mechanical advantage. Adds a lot of joint that must handle high forces and jamming points.

12. Dual opposed Pistons at base. Inspired by flat stroke four engine this design uses two opposing pistons to generate force and a small link to connect to the main lever. Stationary pistons reduce hose ware and air leakage.

13. Damped 2 hinge system. Similar to design 7, this adds damping to the pivot point but uses horizontal damping to remove the high control issue. Mechanical advantage is not high enough for target forces.

14. Bent lever with vertical piston. Very simple design that can generate great nip force depending on the location of the forward mount. This is limited by space in machine.

15. Non-parallel 4 bar with internally mounted piston. Uses a fundamental 4 bar for mechanical advantage. Internally mounted piston makes design more compact, but air hoses would need to be held away from pinch points to prevent punctures that would cripple the system.

16. 4-bar with lower mounted piston. Similar to design 15, but with the piston moved outside the four bar. Takes up a lot of space but removes the chance of air hose jamming.

17. Double piston externally mounted. This design is based on the original moving the pistons to the outside of the mount, instead of inside. While it has the same

mechanical advantage as the original, the piston size is much less limited, so force could be greater with the same psi. Pistons and arm would require a very complex mounting structure to prevent torques at joints, making manufacturing and assembly very difficult.

18. Vertical spring assisted piston. The use of a spring constantly in tension adds to the force from the piston to increase nip force. The force is based off the position however, which the control system is not designed to handle.

19. Low piston with counter weight. This ended up being the basis for our final design. The counterweight adds a consistent amount of force to the system and is easily controlled. Tradeoffs include size and material needed for the mass.

20. Cable/pulley system. This design used a pulley system to increase the force from the piston, A spring in compression is required to lift the arm because the cables can only pull. Because the entire system is dependent on one spring that could easily wear out longevity and reliability were serious concerns.

21. Linear sliding system with static damping. Instead of using a lever this design uses two pistons to move the nip roller linearly. The massive amount of torque on the pistons makes this impractical because the bar would most likely end up pivoting around the lower on anyway.

22. Adjustable mass to create force. This design takes the idea from design 19 and takes it further by generating almost all the nip force from a mass hanging below the lever. The mass is segmented to allow on the tuning while in place. Adds complexity to design 19 with no real advantages.

23. Mobile mass to create force and lock back. Also based on the counter weight idea, this design has the mass mounted, so it can be moved by a piston. As the radius changes the torque around the mount changes, allowing control of the nip force. This requires position control for the piston which would require additions to the control system. A second piston is also needed to lift the arm, adding more complexity.

24. Single Piston sliding mass system. This design is very similar to #23 but by changing the travel of the mass it can be used to tilt the arm back as well. Position control is still an issue.

25. Simple lever with mobile damper. A variation of design 7 but a block of elastomer

is used for damping. The block must be mobile to allow travel back, making damping inconsistent.

26. Class 2 lever with mass for force. Yet another mounting configuration for a counterweight. This one mounts the mass in line on the lever. Because the mass has a fairly small radius from the pivot it does not add as much force, but it is easier to control.

27. Class 2 lever with vertical scissor. This design uses two links on the front of the system to increase force. Toray expressed concern about jamming as well as the travel forward.

28. Longer class 2 with horizontal scissor. This design flips takes the mechanism from design 27 and flips it sideways to fit in the space better. This requires a mounting point further back to prevent a cross load on the piston which does not currently exist.

29. Subsystem Damping. All direction rubber damper to hold roller/pin. This is a simple design for a roller holder with damping in all directions. Concerns about wear and height inaccuracies doomed it to the scrap pile.

30. Unidirectional bearing holding damper. This design addresses the height concern of design 29 by adding damping only in one direction. Vibrations would still be able to propagate down the arm however, making it ineffective.

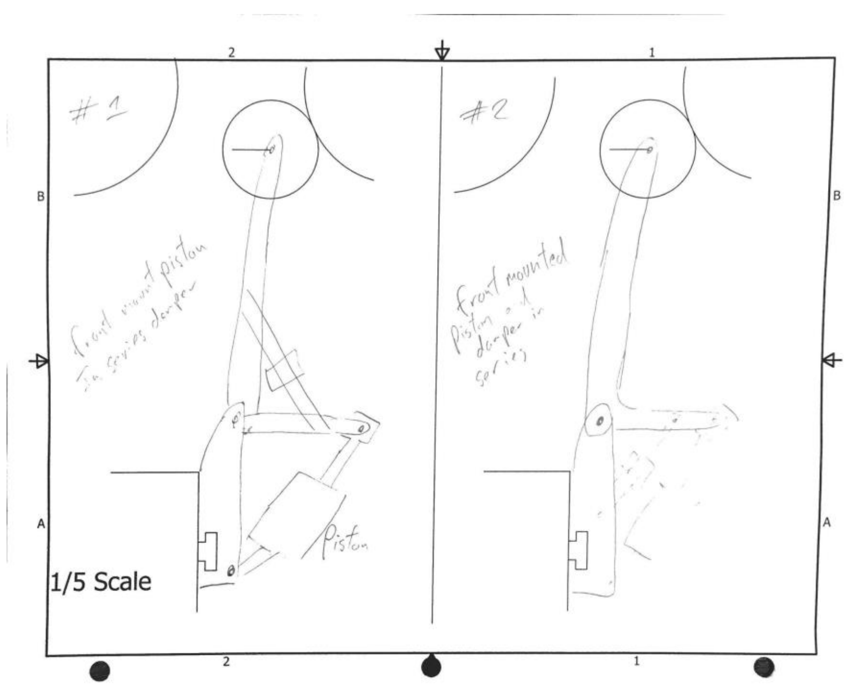


Figure 2: Fraizer's Designs 1 and 2

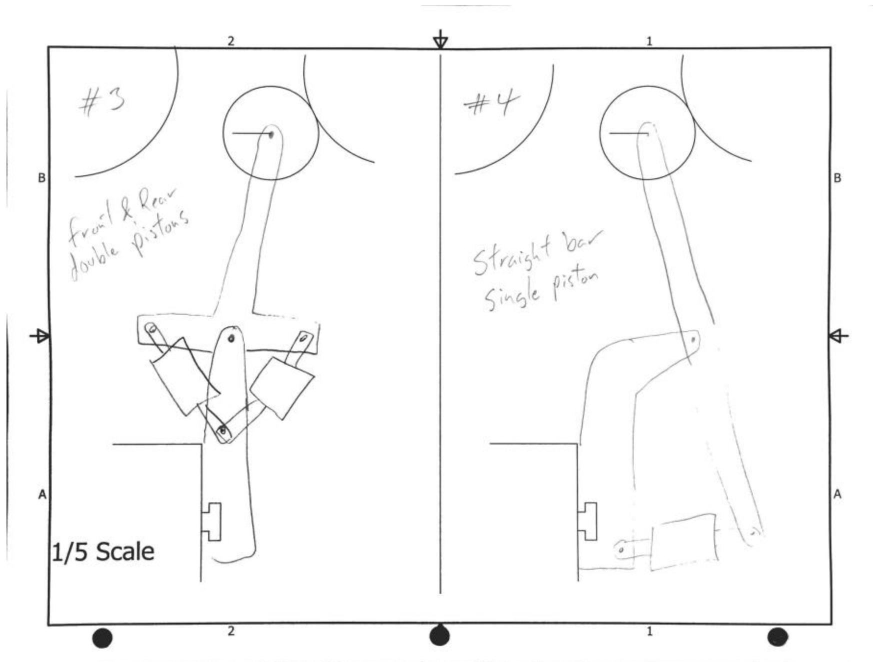


Figure 3: Fraizer's Designs 3 and 4

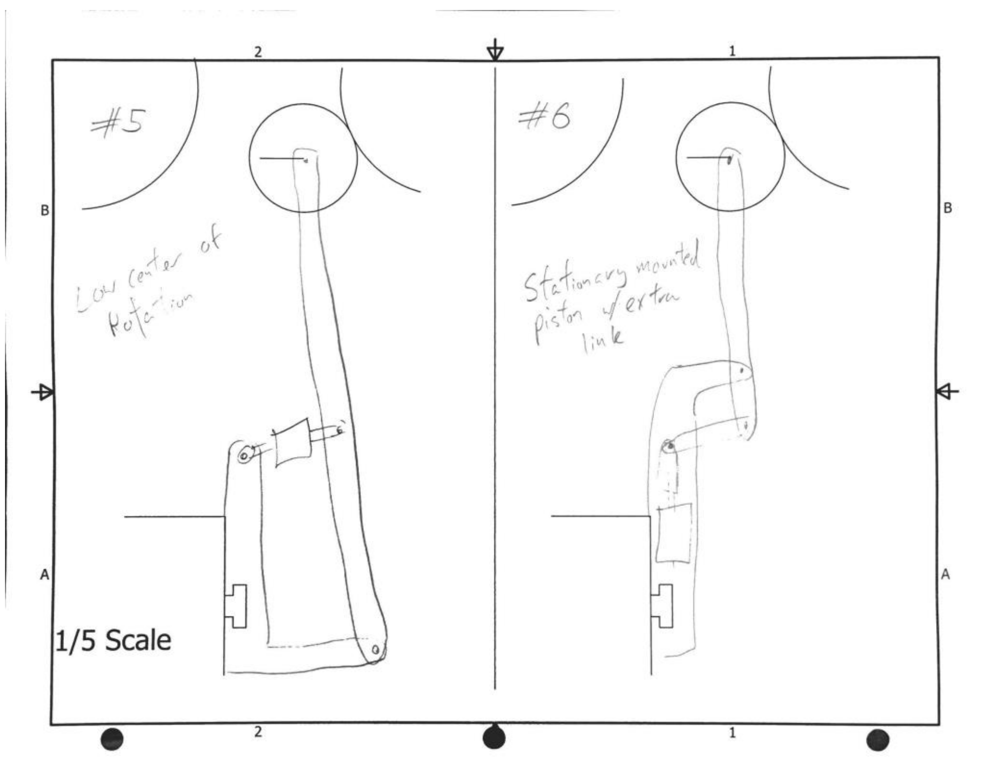


Figure 4: Fraizer's Designs 5 and 6

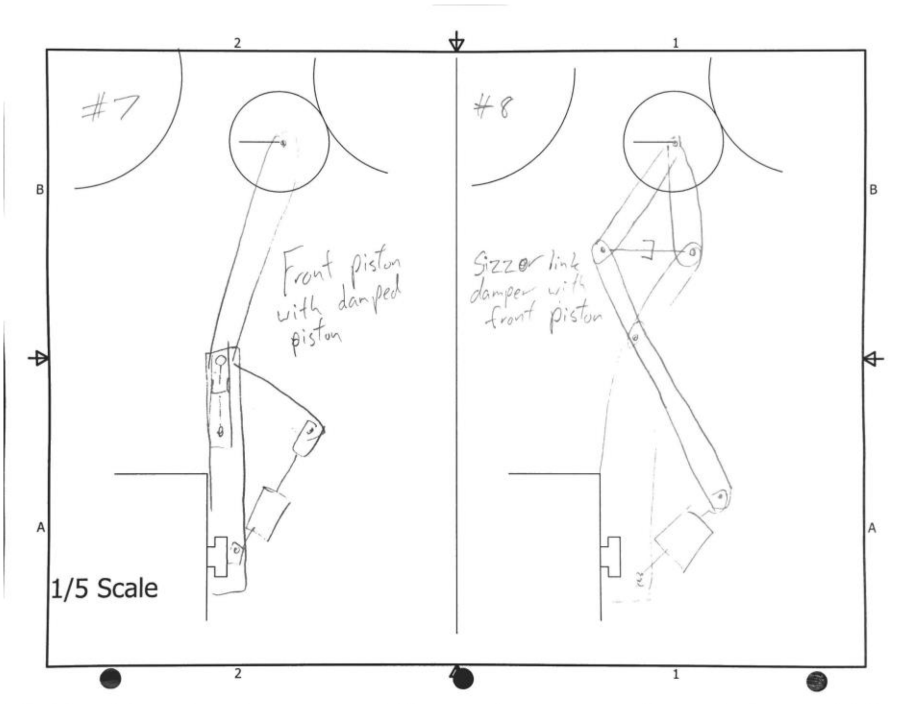


Figure 5: Fraizer's Designs 7 and 8

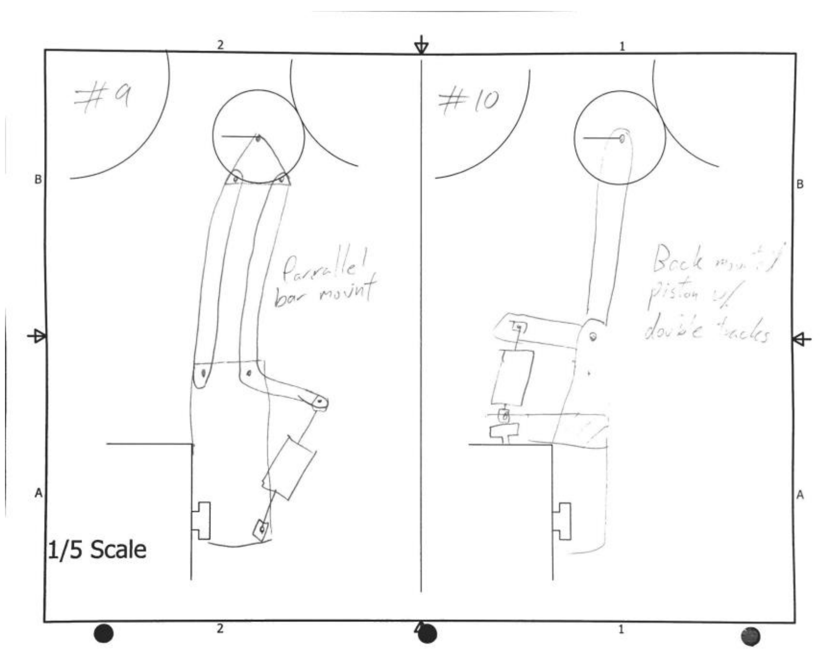


Figure 6: Fraizer's Designs 9 and 10

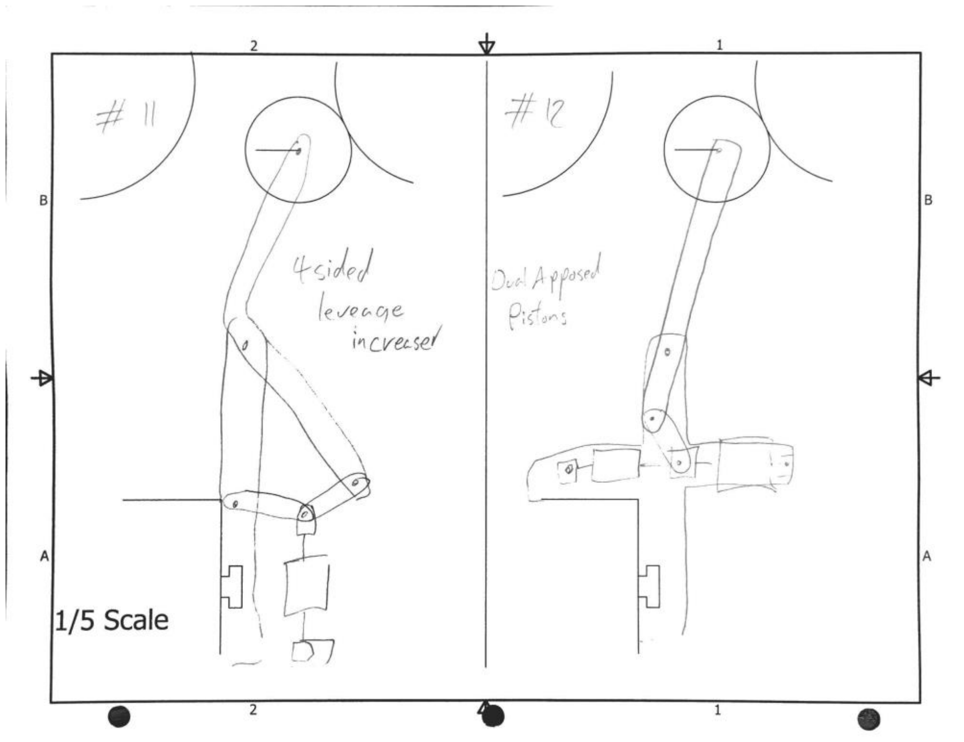


Figure 7: Fraizer's Designs 11 and 12

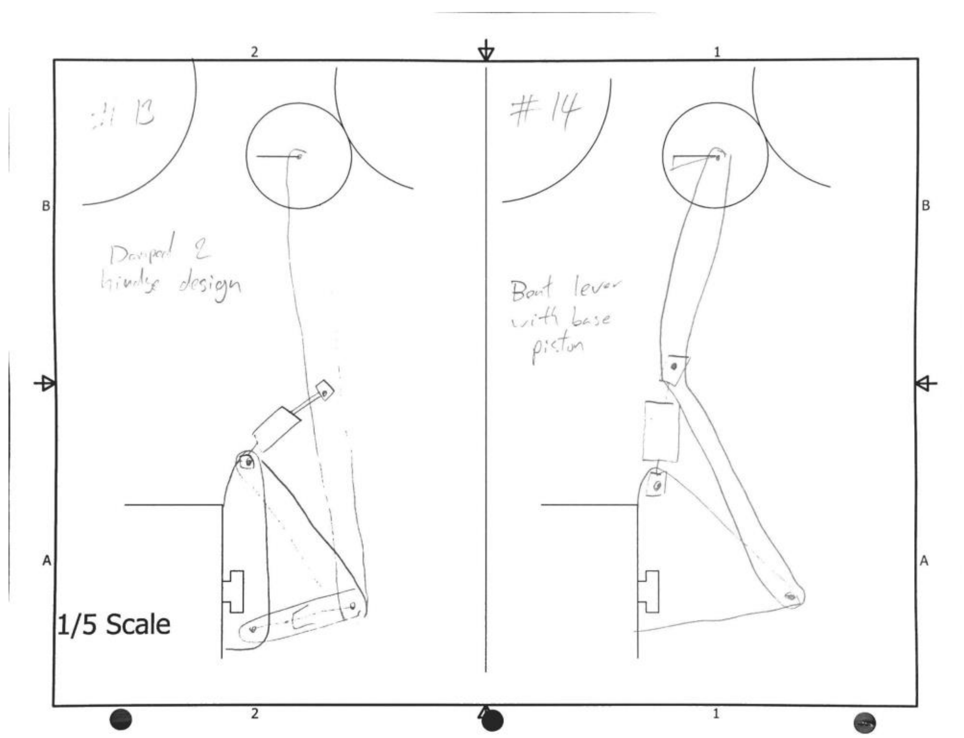


Figure 8: Fraizer's Designs 13 and 14

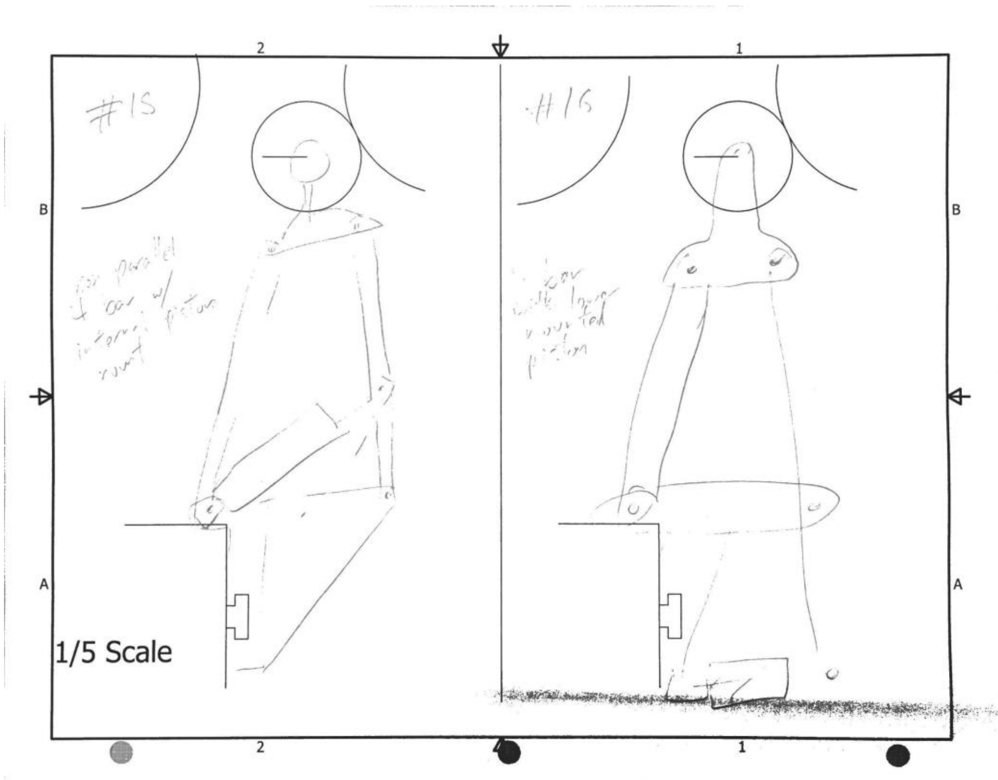


Figure 9: Fraizer’s Designs 15 and 16

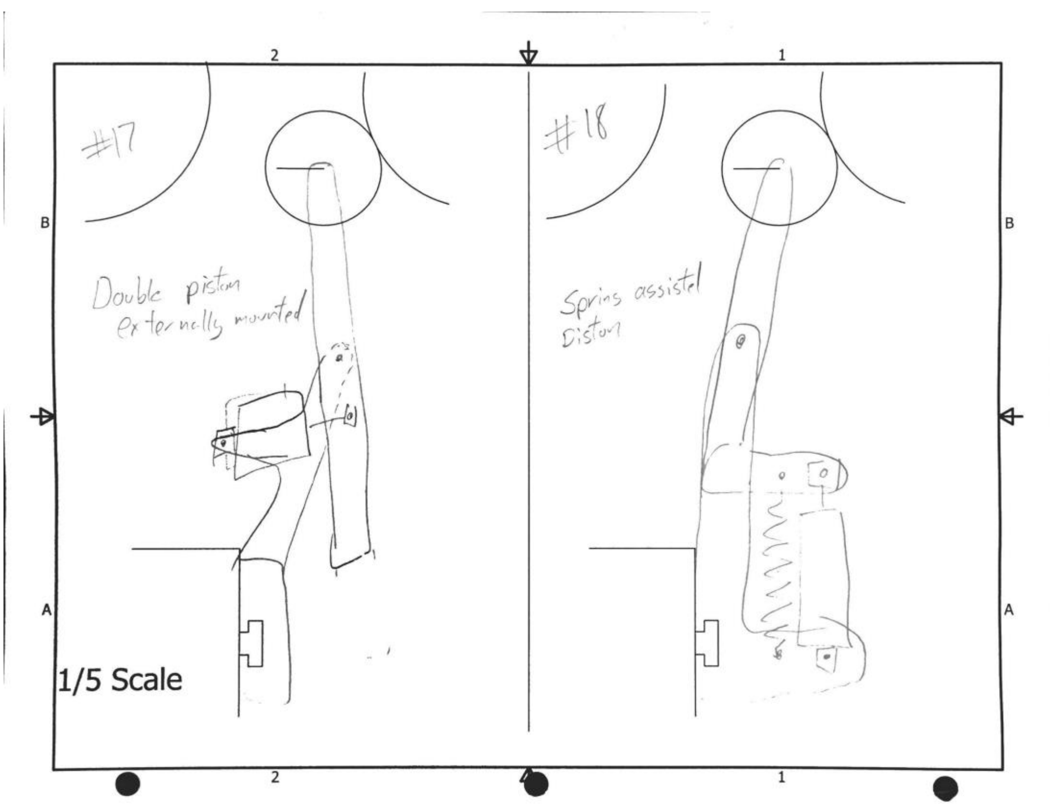


Figure 10: Fraizer’s Designs 17 and 18

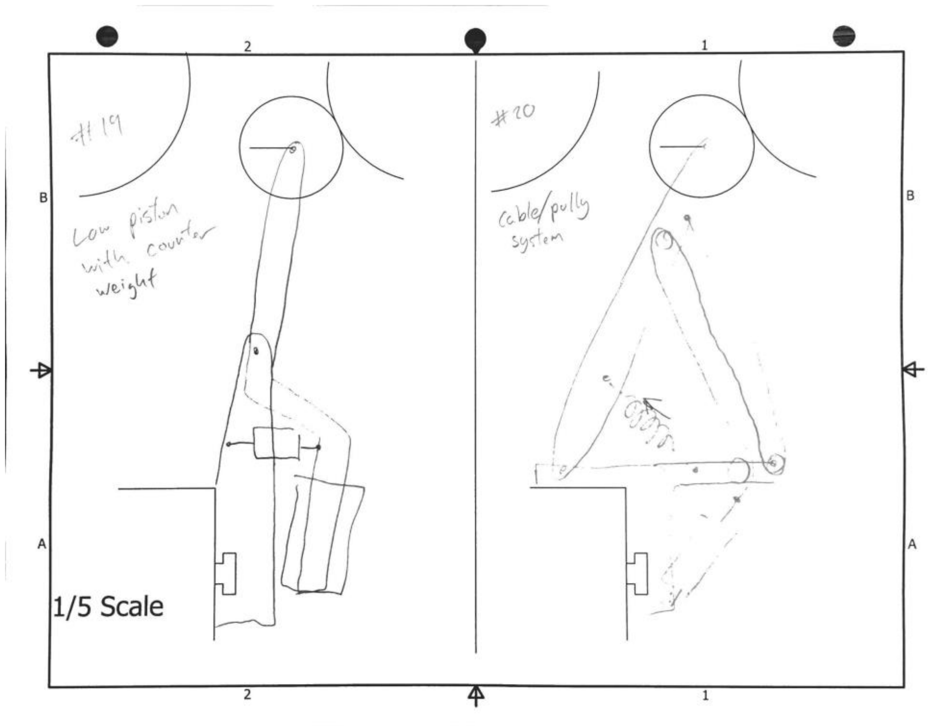


Figure 11: Fraizer’s Designs 19 and 20

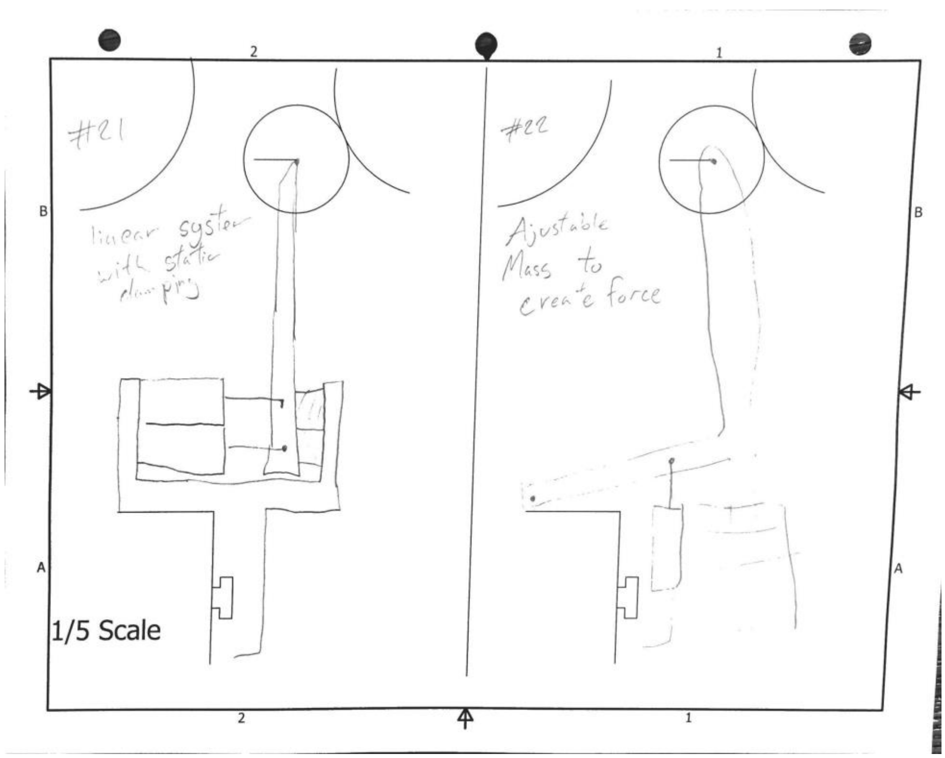


Figure 12: Fraizer’s Designs 21 and 22

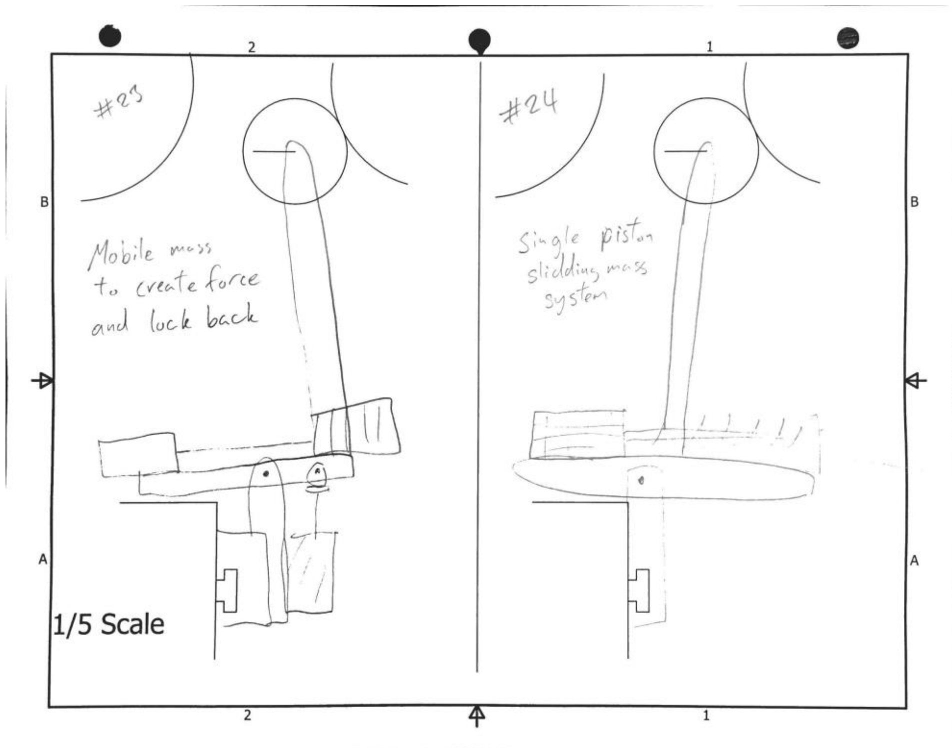


Figure 13: Fraizer's Designs 23 and 24

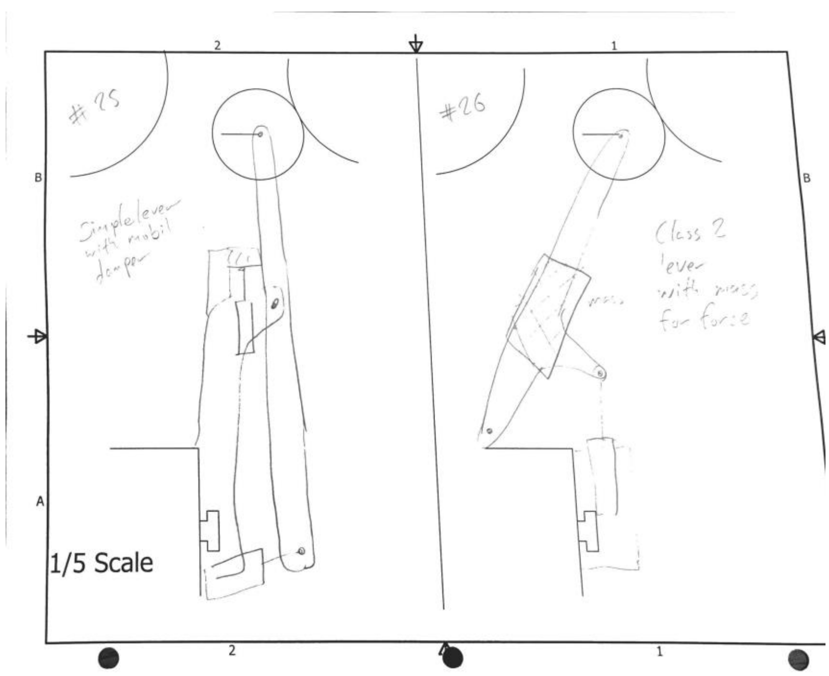


Figure 14: Fraizer's Designs 25 and 26

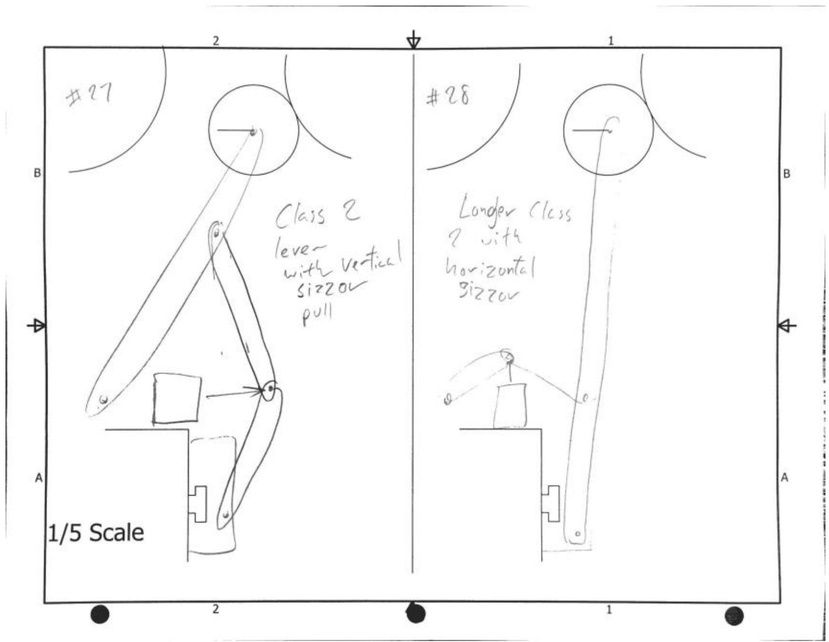


Figure 15: Fraizer's Designs 27 and 28

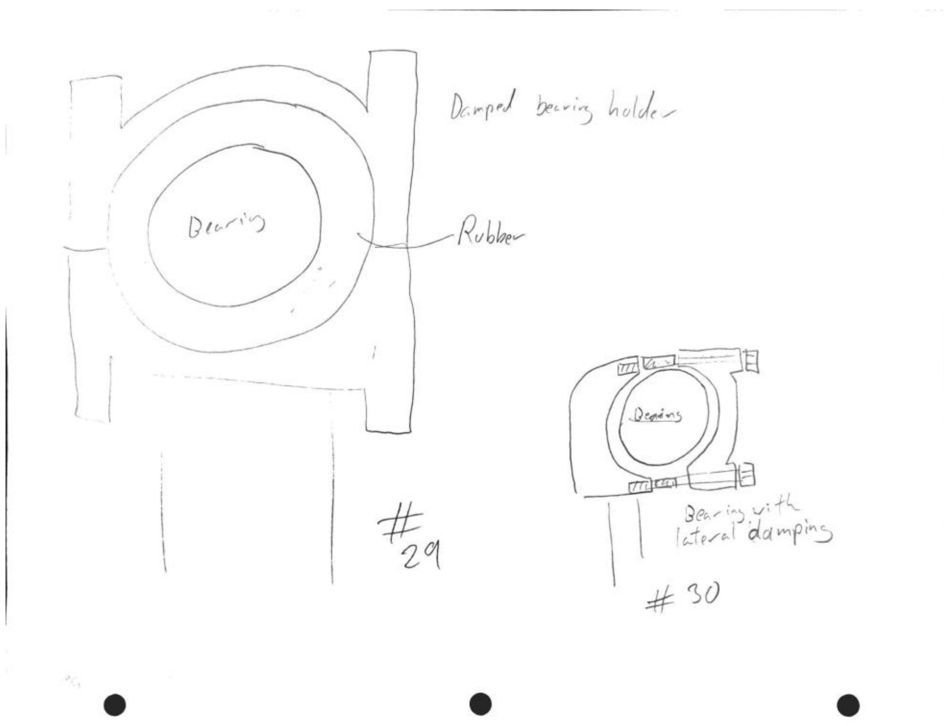


Figure 16: Fraizer's Designs 29 and 30

Table 2: Pugh of Fraizer's Designs

		Concepts																													
Engineering Criteria	Reference Concept	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30
Performance	Good																														
Leverage	.6	-	-	-	+	+	+	-	+	+	+	+	-	+	+	+	+	+	+	+	+	-	+	-	+	+	+	+	+		
Max force		-	-	+	+	+	+	-	+	+	+	+	+	+	+	+	+	+	+	+	+	+	+	+	+	+	+	+	+		
Damping	None	+	+			-	+	+	+		+	+		+		+	+		-	+	-	+	+	+	+	+	+	+	+	+	+
Pistons Required	1			-				-					-					-				-		-		-					
Reliability	High																														
# moving parts	4	+	+	-			-	-	-	-		+	-	-	+	-	-	-	+	+	+	-		-	-	-	+	-	-		
Force on joints	High	+	-	+	-	-	-	+	+	+	+	-	+	+	+	+	+	-	-	-	+	+	-	+	-	+	-	+	+		
Chance of jam	Very low	-		-		+	-	+		-	-	-	+			-	+	+	+	+	-	-	+	-	-	-	+	-	-		
Durability	High																														
Parts that will wear quick	1 or 2	-	-	-	+	+	-	+	+	-		-	-	-	+			-	-		-	-	+	-	-	-	+	+	+	-	-
Serviceability	Ok	+	+	+	+	+	-	-	+	-	-	-	-	-	+	-		+	+	+	-	-	+	-	+	+	+	+	+	-	+
Ease of replacement	Difficult	-	+	+	+	+	+	-	-	+	-	+	+	+	-	+	+	+	+	-	+	-	-	-	+	+		+	-	-	+
Lubrication needed	Low	-	-	-	-	-	-	+	-	-	-	-	-	-		-	-		+	-	-	-	+	-	-	-	+	+	+		
Simplicity	Great	-	-	-	+	+	-	-	-	-	+	-	+	+	+	-	-	+	-	+	-	-	+	-	-	-	+	+	+		
# of Plusses		4	4	4	6	7	4	5	6	4	5	5	5	6	7	5	6	6	7	7	5	3	8	3	5	6	9	9	8	1	3
# of Minuses		7	7	6	2	3	7	7	4	6	4	6	6	4	1	5	4	3	4	3	6	9	2	9	6	6	1	2	3	3	1

<u>Eddie Janis's 30 concepts:</u>

1. Pulley system that is spring-loaded tightened with an adjustment screw
2. Double linkage system that is spring-loaded
3. Pulley system using a weighted mass and torque
4. Pulley system using an adjustment screw and torque
5. Pivoting horizontal rod attached to a mount by an adjustment screw
6. System that utilizes multiple air pistons for force
7. System that utilizes springs as sources of force
8. System that utilized springs on sliding mounts to maximize force on arm
9. Multi-piston system that uses hydraulic pressure instead of air pressure
10. Multi-spring system used to maximize spring force on the arm
11. System that utilizes a spring-air piston combination to maximize spring force
12. Linkage system that utilizes an adjustment screw to change force supplied to mounting arm
13. Dual-pulley system that allows for adjustment of roller with adjustment screws
14. Introduces the idea of an elastic grommet to dampen vibrations. Also utilizes springs as sources of force on the arm
15. System that uses an elastic band to supply force to the arm
16. System utilizes force from an adjustable pulley system with the roller mounted on a sliding pivot
17. Arm that uses a linkage for support and adjustable base to supply force to the roller
18. Angled pulley system that supplies force to the roller, which is mounted on a pivoting arm
19. Angled adjustment linkage system that utilizes an adjustment screw and a sliding linkage mount
20. System that utilizes two springs with different spring constants to maximize force on roller
21. Roller is mounted to a 90-degree support arm that engages and disengages the roller with an adjustment screw

22. System that maximizes nip roller force through ideal angular calculation and an adjustment screw
23. System that includes the roller being mounted to a structurally supported base with a spring-loaded adjustment system
24. Double linkage system that is connected to a pulley that is tightened with an adjustment screw
25. Sub-system idea that utilizes a rubber grommet and bearing combination to eliminate vibrations
26. Sub-system idea that utilizes a thick rubber seal to isolate the roller pin rod as a means to eliminate vibration
27. System that is made up of 90-degree support arms to the mounting arm, along with a spring-loaded system to supply force to the nip roller
28. System that consists of a pivoting mounting arm that is supported by an adjustment screw
29. System that utilizes a spring-loaded piston to supply force
30. System that utilizes a spring-loaded piston to supply force with an additional spring for added support

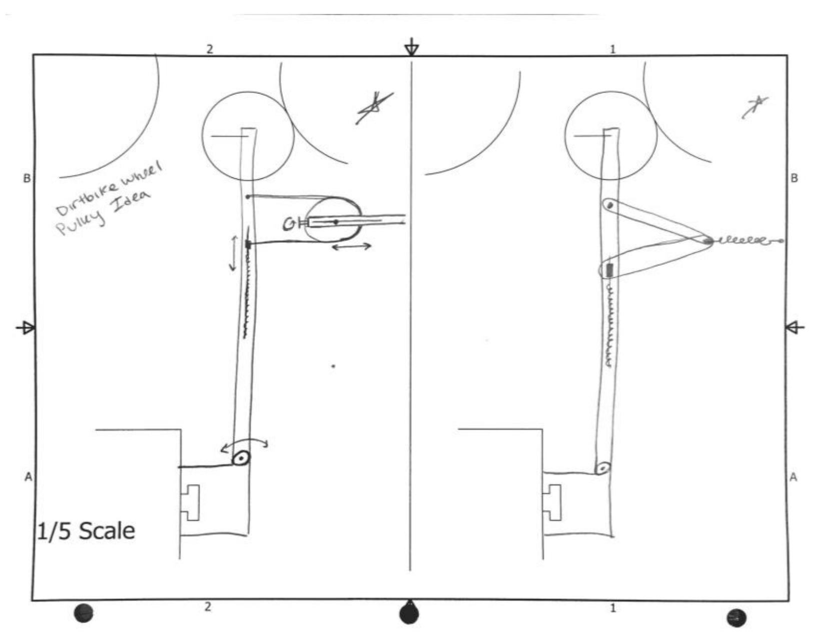


Figure 17: Eddie Janis’s Designs 1 and 2

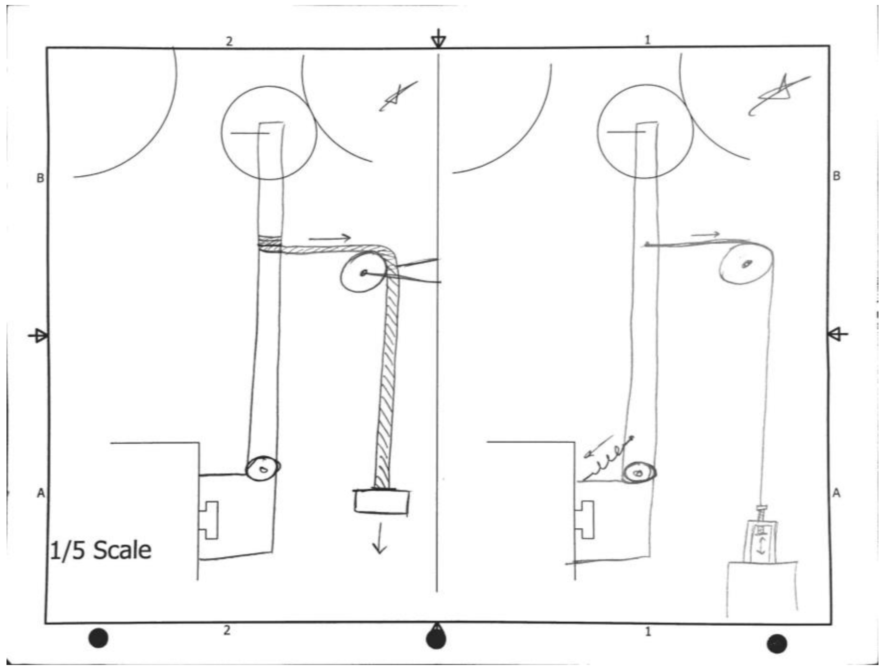


Figure 18: Eddie Janis's Designs 3 and 4

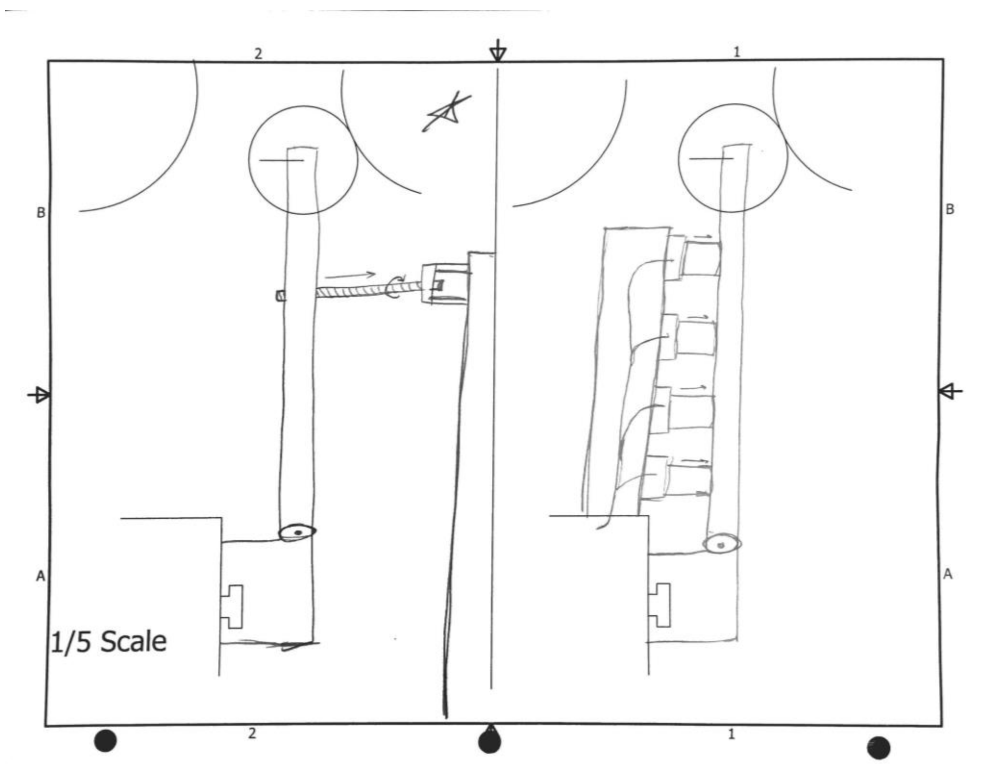


Figure 19: Eddie Janis's Designs 5 and 6

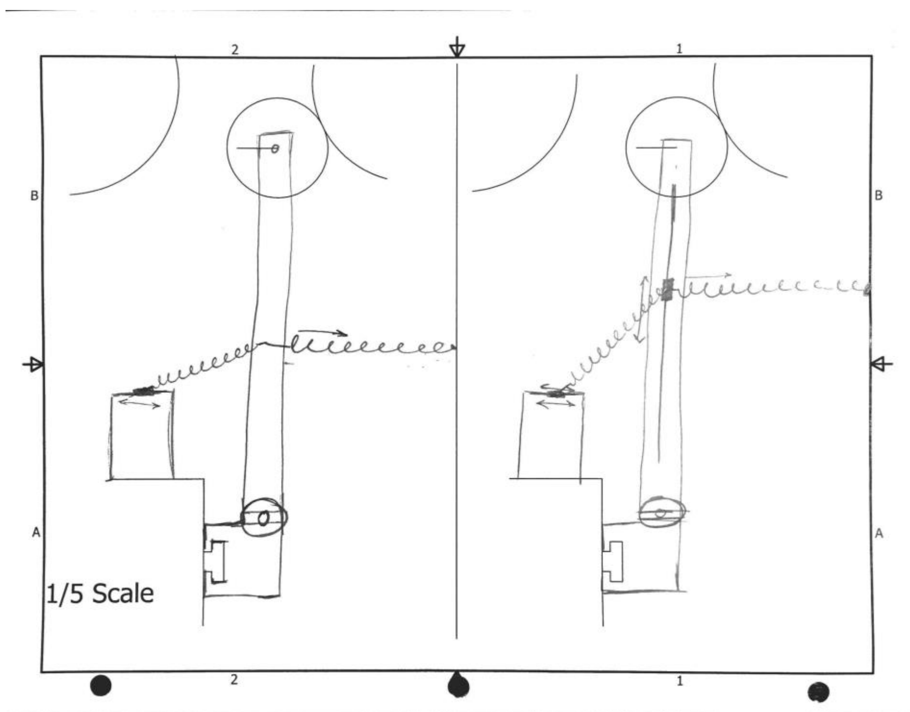


Figure 20: Eddie Janis's Designs 7 and 8

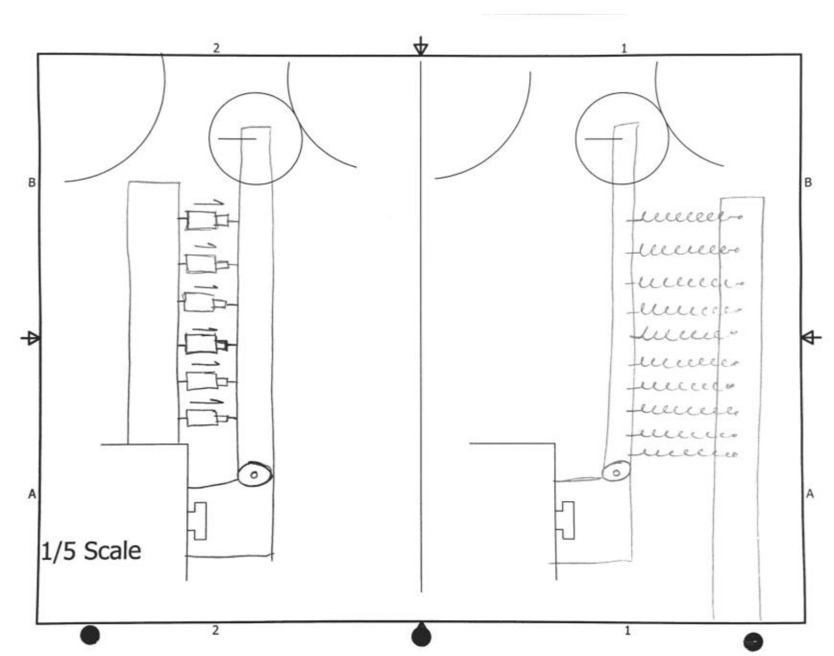


Figure 21: Eddie Janis's Designs 9 and 10

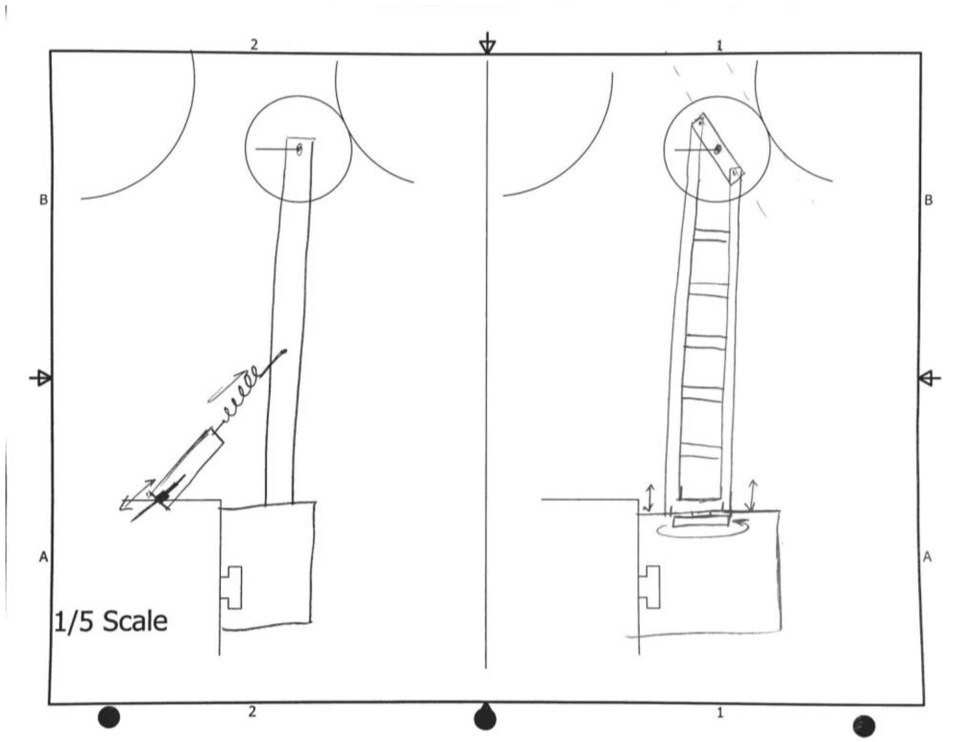


Figure 22: Eddie Janis's Designs 11 and 12

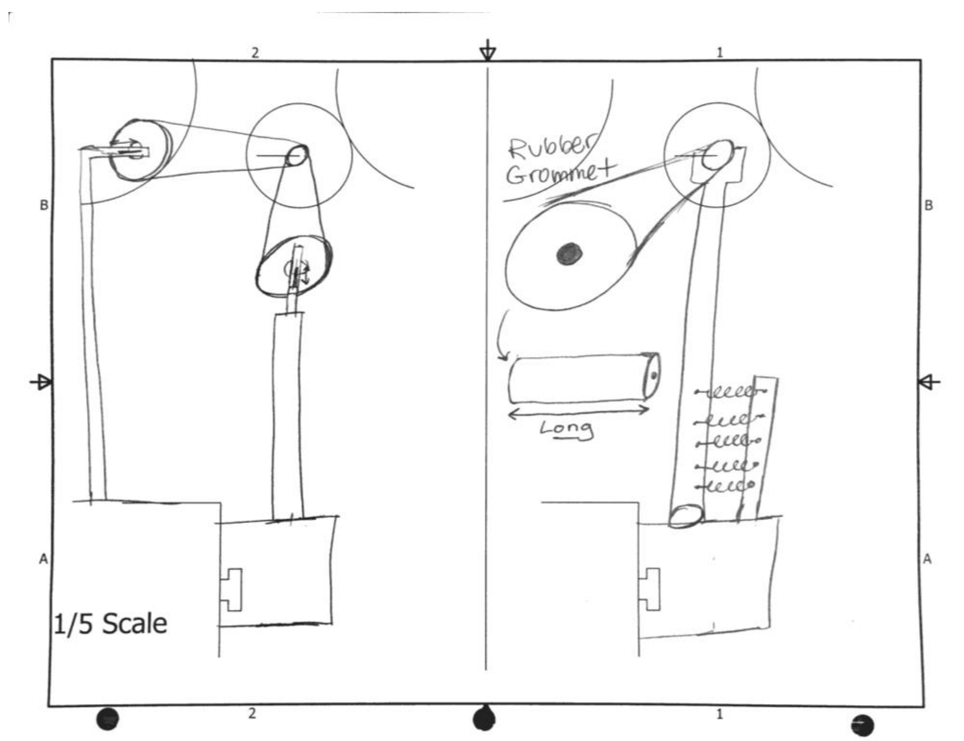


Figure 23: Eddie Janis's Designs 13 and 14

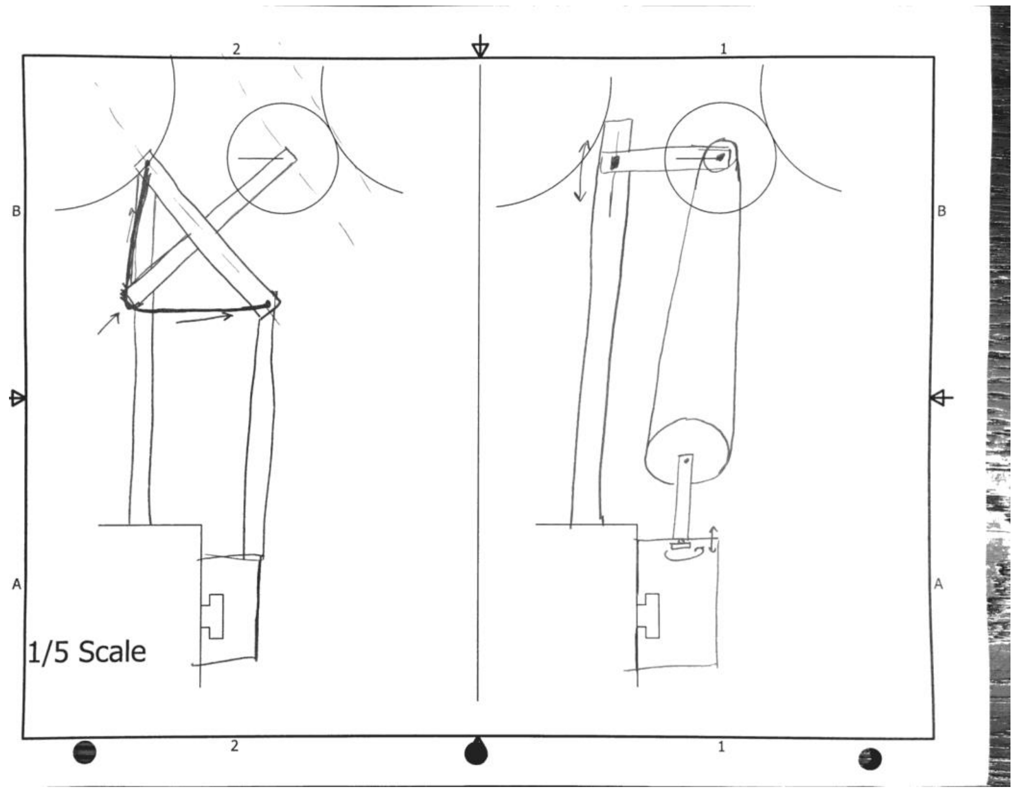


Figure 24: Eddie Janis's Designs 15 and 16

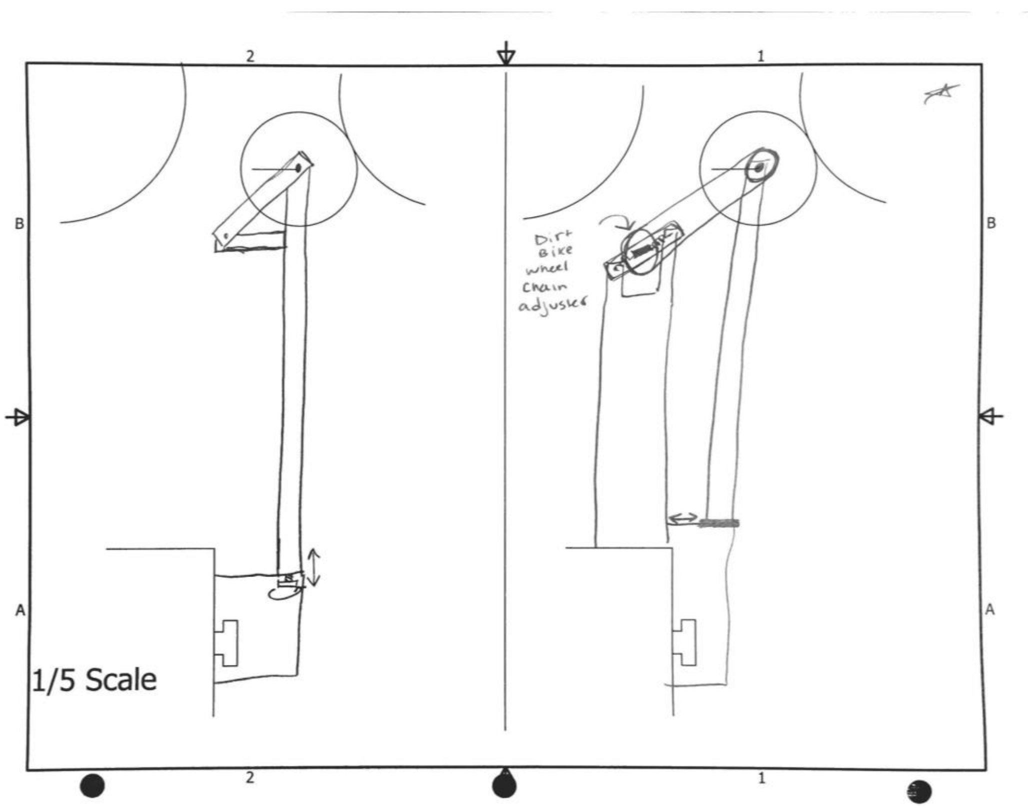


Figure 25: Eddie Janis's Designs 17 and 18

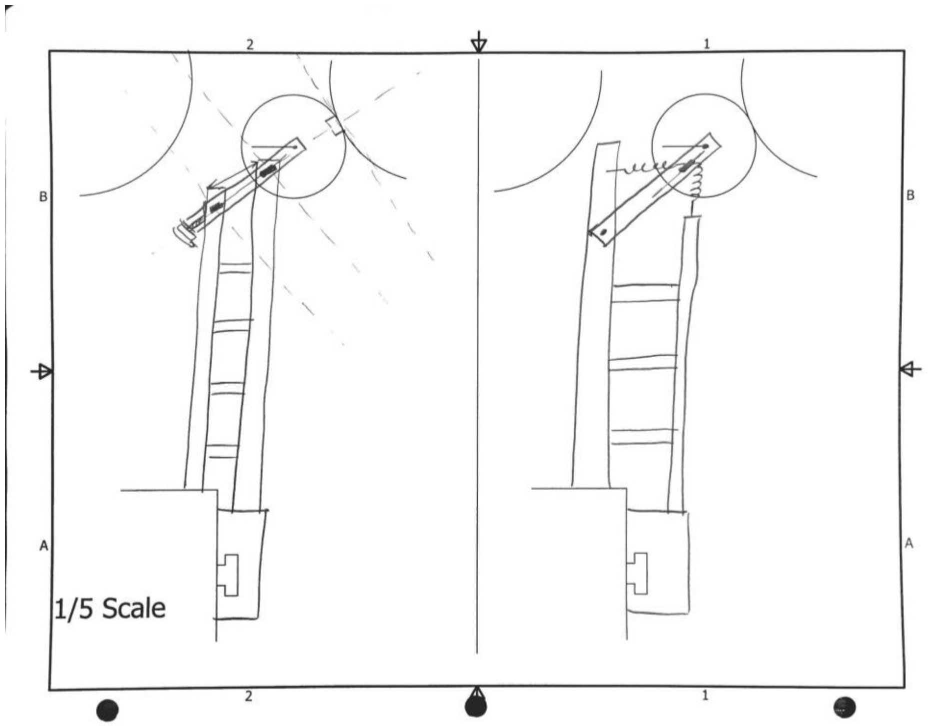


Figure 26: Eddie Janis's Designs 19 and 20

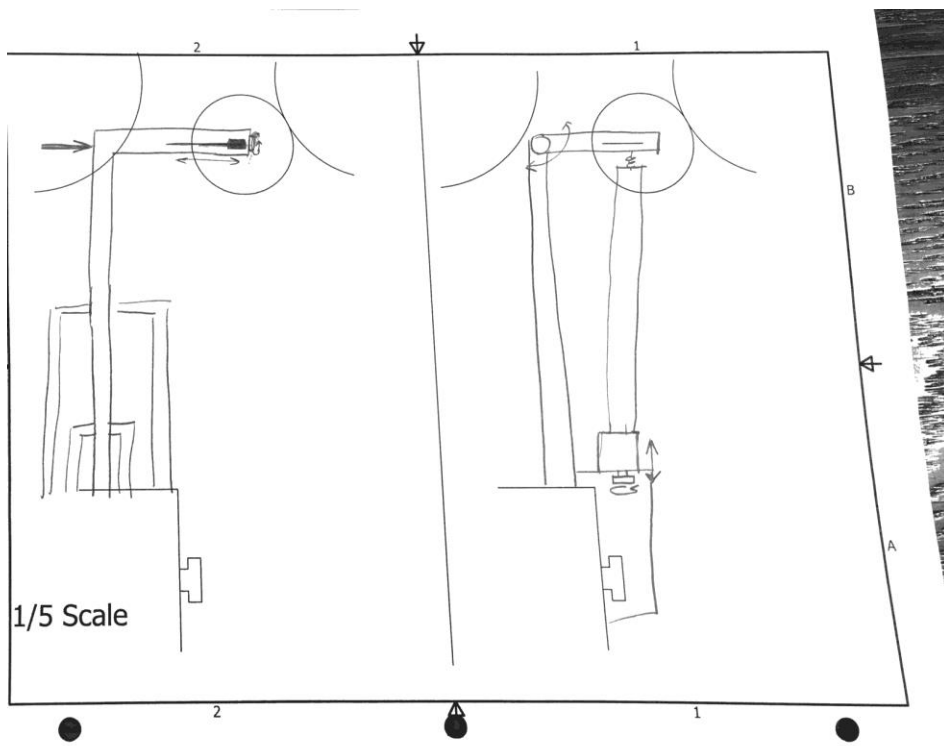


Figure 27: Eddie Janis's Designs 21 and 22

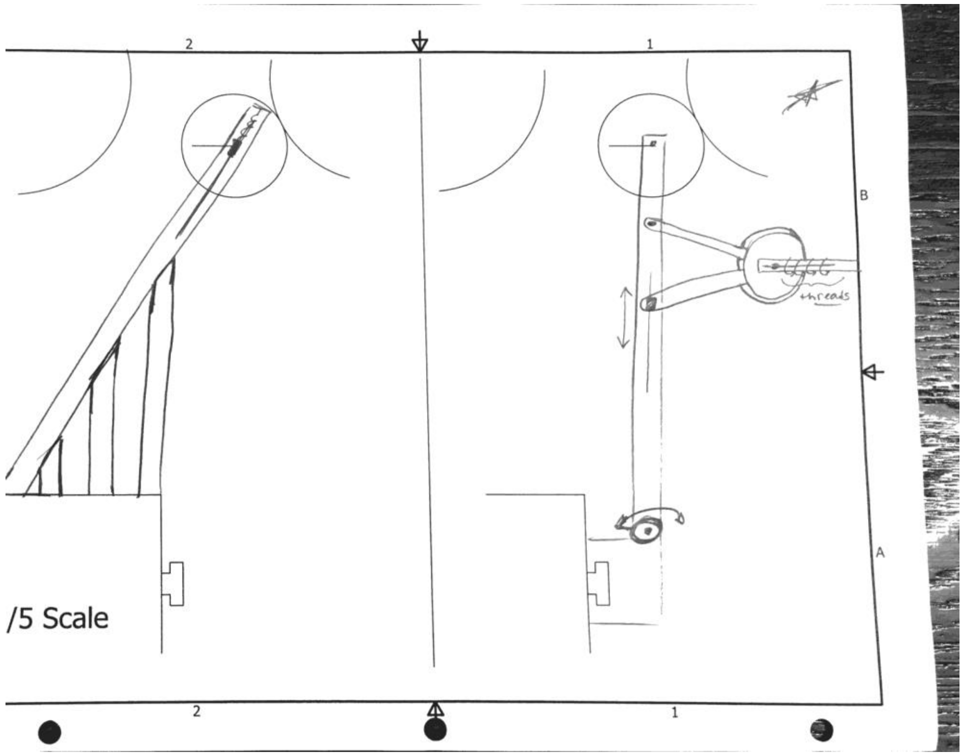


Figure 28: Eddie Janis's Designs 23 and 24

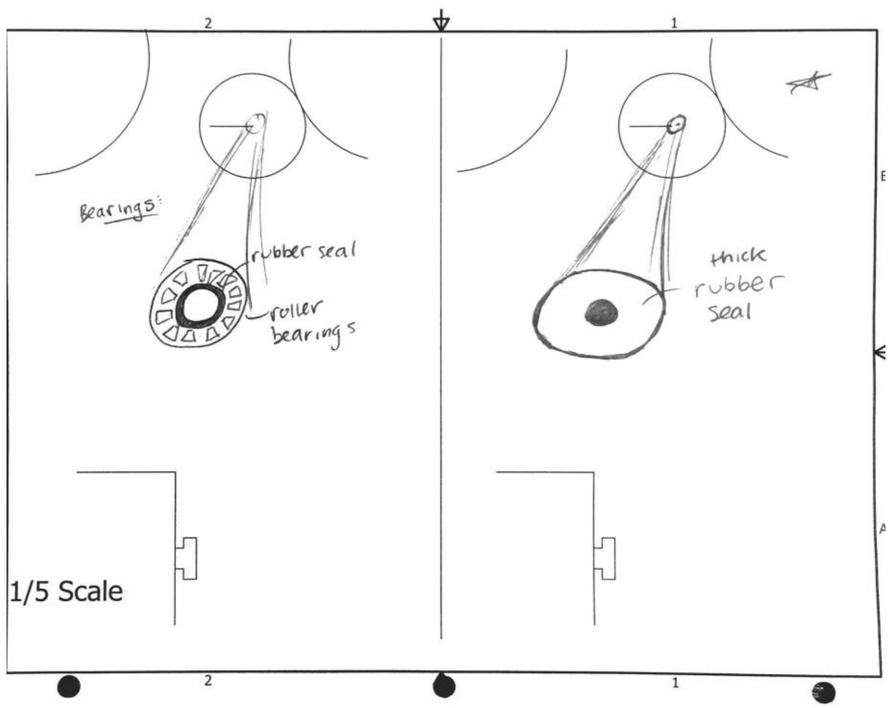


Figure 29: Eddie Janis's Designs 25 and 26

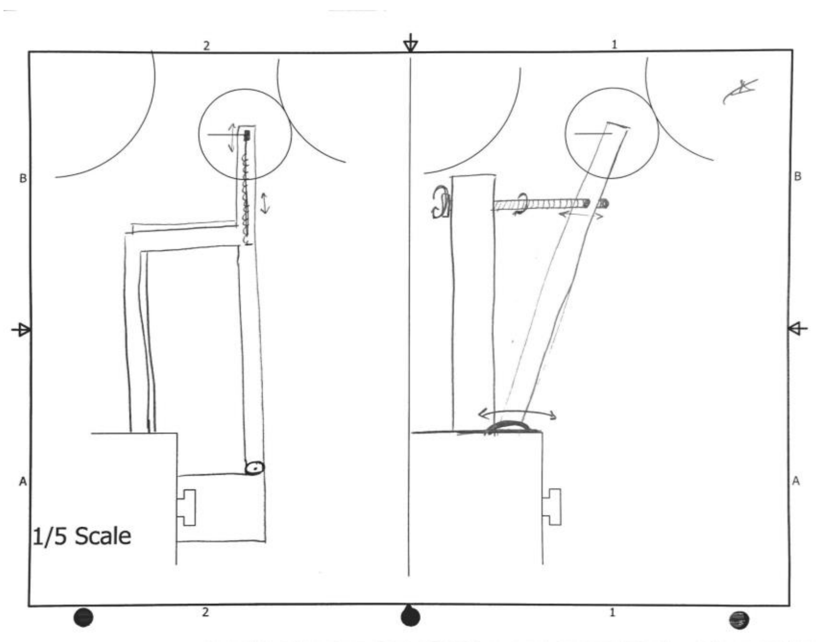


Figure 30: Eddie Janis's Designs 27 and 28

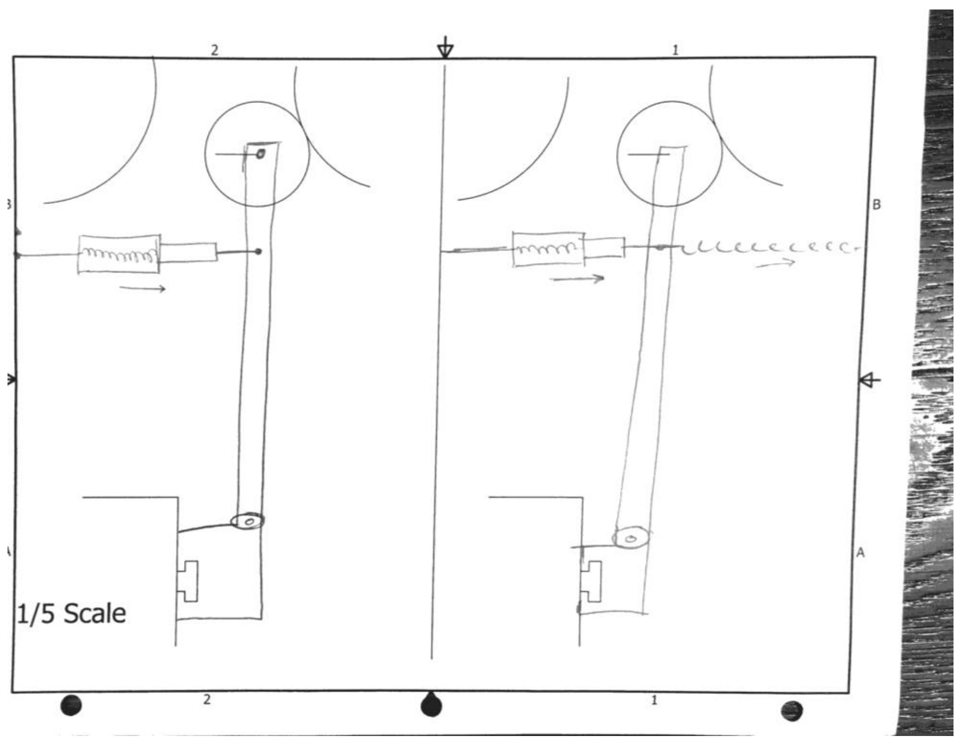


Figure 31: Eddie Janis's Designs 29 and 30

Table 3: Pugh of Janis's Designs

		Concepts																													
Engineering Criteria	Reference Concept	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30
Performance	Good																														
Cost	Low	+	+	+	+	+	+	+	+	+	+	+	+	+	+	+	+	+	+	+	+	+	+	+	+	+	+	+	+	+	+
Pistons Required	None						-			-																					
Reliability	Good																														
moving parts	3	+			+	-	-	+	-	-	+	-	+	+	-	-	-	-	+	-	-	-	+	+	+	+	-	-	+	-	+
Stress on Material	high						+						+						+			+	+							+	
Chance of jamming	low	-	-	+	-	-	+	-	-	+	+	-	-	+	+	-	-	-	-	+	-	+	-	+	+	+	-	+	-	-	+
Lifespan	ok	-	-	+	+	-	+	-	+	-	+	+	-	-	-	+	+	-	-	-	-	-	+	-	-	+	-	-	-	+	-
Parts that will wear quickly	1	-	+	+	-	+	-	-	+	+	-	-	-	-	-	-	+	-	+	-	+	+	-	-	+	+	+	-	+	+	-
Compatability	Ok	+			-	-	-	+	-	+	-	-	-	+	+	+	-	-	-	+	+	-	-	-	-	+	-	-	+	-	+
Ease of Servicing	Easy	+	-	+	+	-	-	+	-	-	+	+	-	+	-	-	-	+	-	-	+	+	-	-	+	-	+	+	-	-	-
Installation	Easy	+	-	+	-	+	-	+	-	-	+	-	+	+	+	+	+	-	-	+	+	-	-	+	-	-	+	+	-	+	-
Sufficient force output	Yes	+	+	+	+	+	+	+	+	+	+	+	+	+	+	+	+	+	+	+	+	+	+	+	+	-	+	+	+	+	+
Damping ability	none	-		+	-	-	+	-	-	-	-	-	-	-	-	-	-	-	-	-	+	+	-	-	+	-	-	-	+	+	-
# of Plusses		6	3	8	5	4	6	6	4	5	7	4	5	7	5	5	5	3	5	5	7	7	5	5	7	6	5	4	5	7	5
# of Minuses		3	4	5	5	6	6	4	6	6	3	6	7	3	5	5	5	8	6	5	3	4	6	5	3	4	5	5	4	5	5

<u>David Rainone's 30 concepts:</u>

1. Lowered piston: This design is beneficial because it will increase the leverage, by increasing the moment arm of the force applied by the piston, which will increase the nip force. This design will be difficult to implement into the current system.
2. Circular support: The current design has a U-shaped support to hold the nip roller in place. If the support is circular, that may provide a better support to the nip roller, which should reduce the vibrations in the system.
3. Lowered cylinder and circular support: This concept combines concepts 1 and 2, and will both reduce the vibrations, as well as increase the leverage.
4. Circular Support Roll Bearing: Rather than just having a circular support, this design incorporates a roll bearing mechanism, which will help reduce the vibrations as the nip roller moves (when the customer roll increases diameter).
5. Roll bearing/Lowered cylinder: This design incorporates both a roll bearing, as well as a lowered cylinder to increase the leverage and moment arm.
6. Spring and dampers: This design incorporates a U-shaped support, mounted with springs and dampers. This will create a mass-spring-damper system, which will decrease the vibrations in the system, and help prevent roll burping.
7. Spring and dampers, circular support: This design incorporates a circular support, mounted with springs and dampers. This will create a mass-spring-damper system, which will decrease the vibrations in the system, and help prevent roll burping.
8. Spring and dampers cylinder lowered: This design incorporates a circular support, mounted with springs and dampers. Additionally, the piston will be lowered to increase leverage, which will increase nip force.
9. Spring and dampers cylinder lowered 2: This design incorporates a U-shaped support, mounted with springs and dampers. Additionally, the piston will be lowered to increase leverage, which will increase nip force.
10. Spring and dampers roll bearing: This design incorporates a roll bearing nip roll support, mounted with springs and dampers. This will create a mass-spring-damper system, which will decrease the vibrations in the system, and help prevent roll burping.

11. Spring and dampers roll bearing lowered: This design incorporates a roll bearing nip roll support, mounted with springs and dampers. Additionally, the piston is lowered to increase the leverage and nip force.
12. Mass-pulley: Using a pulley with a mass to assist in increasing the nip force. This will help increase the leverage by the mounting arm and can be adjusted to increase the nip force (nip force will increase as mounting point for the pulley is lowered).
13. Mass-pulley circular bearing: Design 12, but with a modified circular bearing to create a better support with the nip roller and reduce vibrations.
14. Double arm: Provides some flexibility in the mounting arm in the motion when the piston moves in and out, will help reduce vibrations with this flexibility.
15. Piston-spring: As the piston moves, it stores energy in a spring, which will increase the nip force as well as reduce the vibrations in the system.
16. Pulley and cylinder: Both the cylinder and the pulley work together to assist in increasing the nip force. This will help increase the overall effectiveness of the mounting arm and increase the nip force without introducing any more outside energy.
17. Roll bearing: Will help reduce vibrations.
18. Rubber bearing: Will help reduce vibrations. This design will be difficult in practice, because the rubber will wear out with use, and will become less effective over time. Frequent replacement will be necessary.
19. Roll bearing with rubber: O-ring design outside of the roll bearing. Combining designs 17 and 18 to increase the overall effectiveness of damping, which will effectively help reduce vibrations in the system.
20. Rubber bearing with dampers: Dampers on the outside of the rubber bearing to restrict motion, as the roll is in operation. This will help reduce vibrations more than design 18.
21. Cylinder and bearing: The cylinder will increase the nip force (can be changed, increase in diameter will increase the nip force); bearing will help reduce the vibrations in the system.

22. Hydraulic system: This design is a hydraulic system, which will help increase the nip force. This, in practice, will be extremely difficult, and impractical, as hydraulics are complicated and expensive. Not the best use for a mounting arm application.
23. Improved Hydraulic system: This design is an improvement on design 22, to help increase the nip force. Adding an additional component to the hydraulic system will increase the nip force.
24. Hydraulic and Cylinder: This design is an addition to design 22, which will use a cylinder to help assist the hydraulic system in increasing the nip force. This is extremely effective, but again, impractical in an actual design.
25. Rotated clamp: This design will rotate the clamp, to restrict movement in the z-direction, which should help reduce the vibrations, because this is the direction at which the customer roll bounces when air pockets and imperfections are introduced into the roll.
26. Rotated clamp and mass: This is a pulley system, which will use a mass to assist in increasing nip force, as well as increasing inertia, which will reduce the vibrations in the system.
27. Double bearing: This design incorporates two bearings separated by a rubber layer. This design will be difficult to implement, because the rubber layer will wear over time (many cycles) and will need frequent replacement.
28. Double clamp: The double clamp involves two U-shaped supports, on top of each other, to provide a better support for the nip roller. The goal with this design is to reduce the vibrations in the system.
29. Double cylinder: Making two cylinders will increase the nip force dramatically, because one cylinder currently provides all of the support for the nip force.
30. Double cylinder and mass: Adding to design 29, to increase the nip force even more, as well as increase inertia which will effectively reduce vibrations in the system.

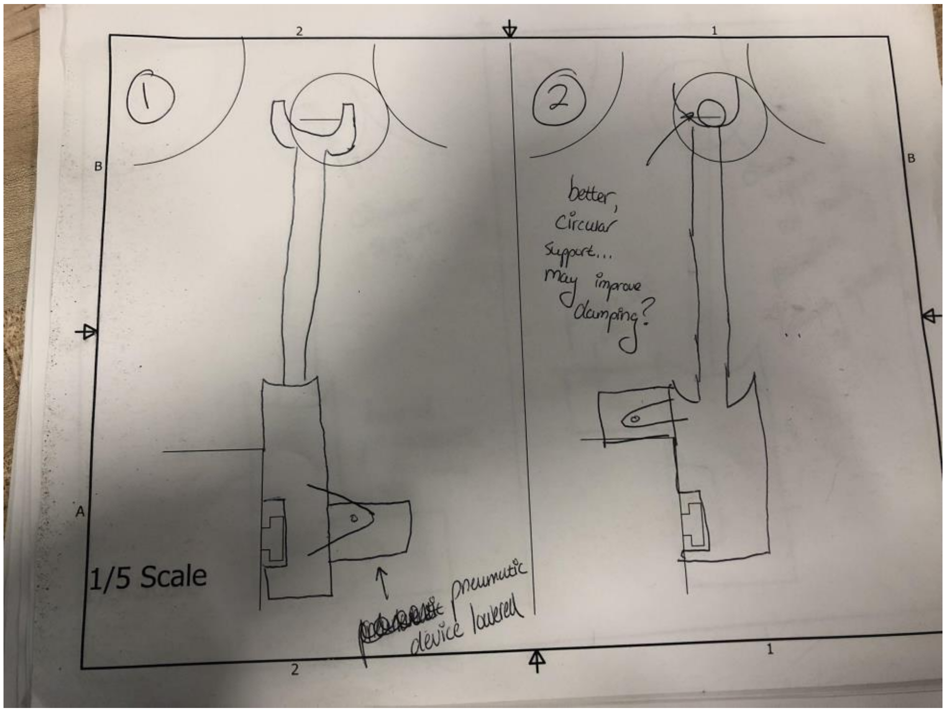


Figure 32: David Rainone Concepts 1 and 2

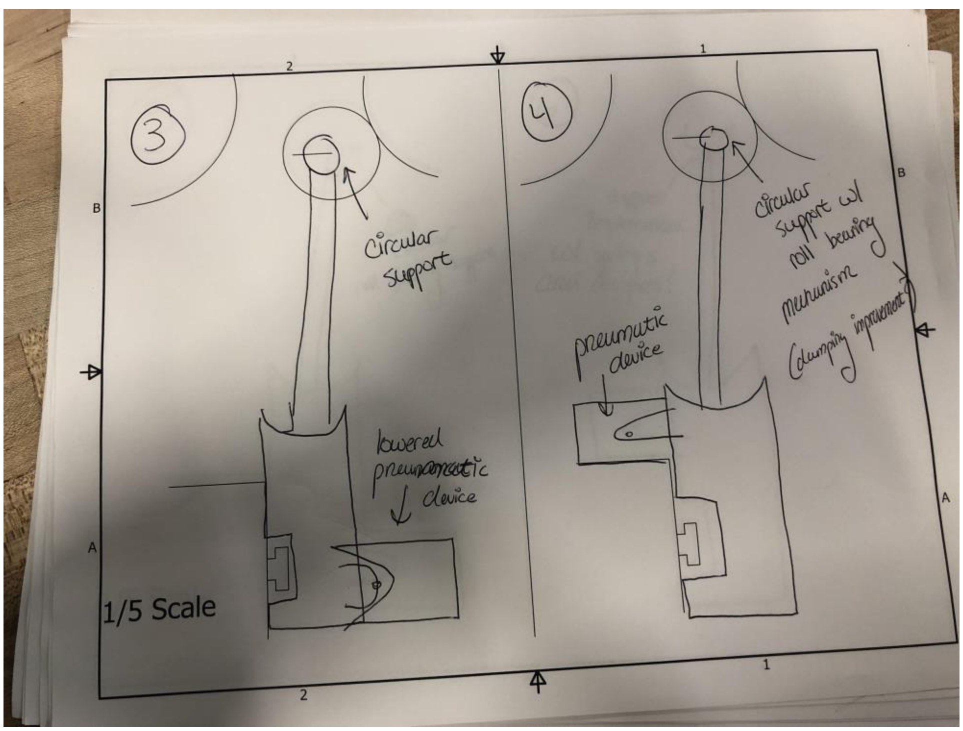


Figure 33: David Rainone Concepts 3 and 4

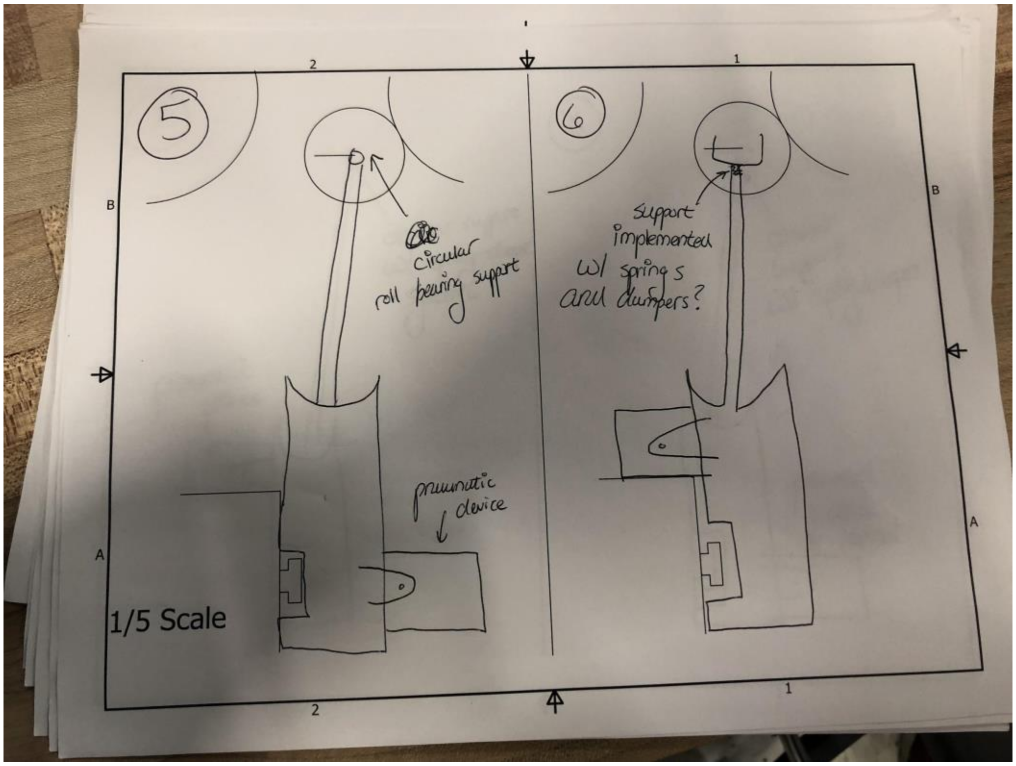


Figure 34: David Rainone Concepts 5 and 6

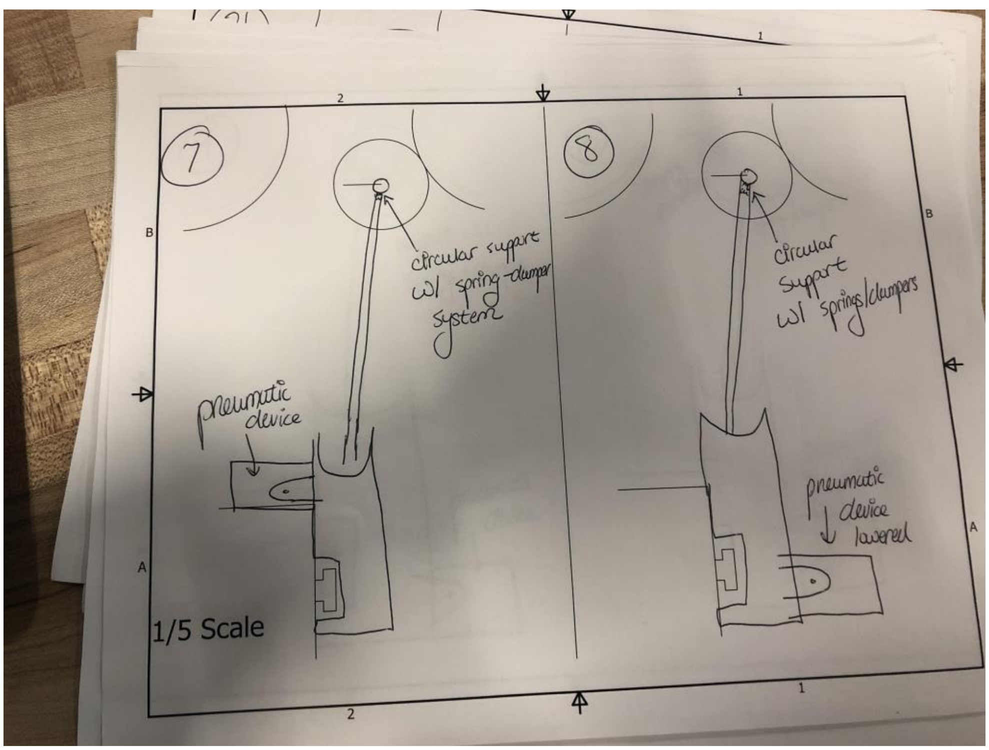


Figure 35: David Rainone Concepts 7 and 8

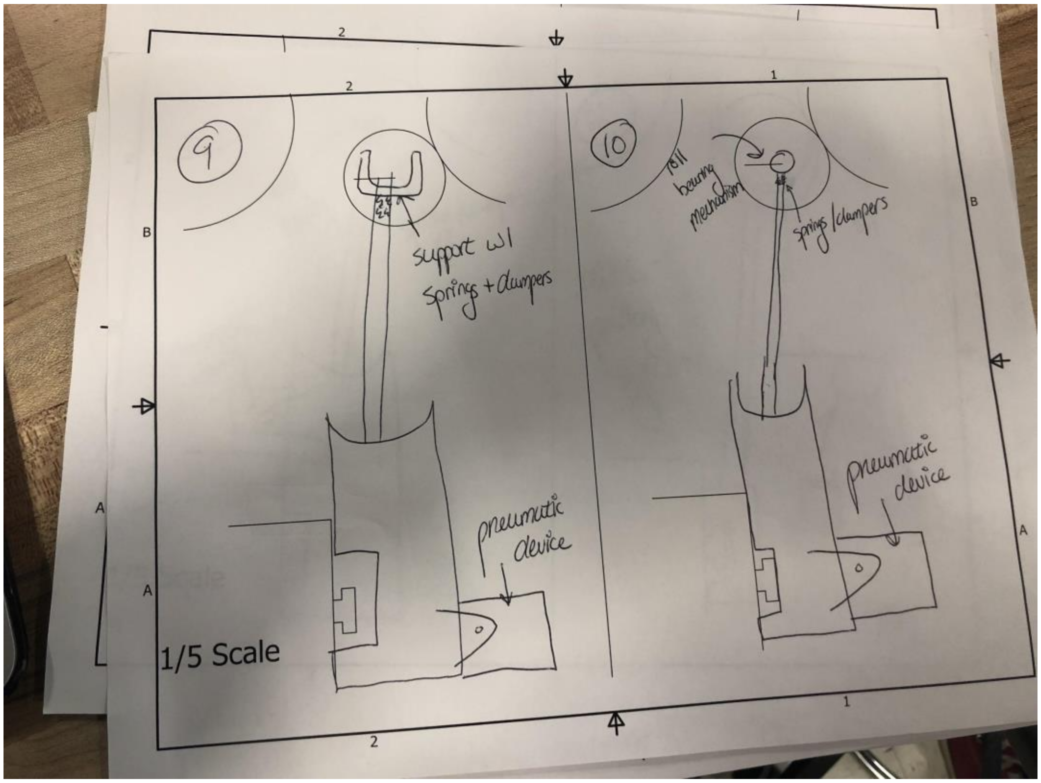


Figure 36: David Rainone Concepts 9 and 10

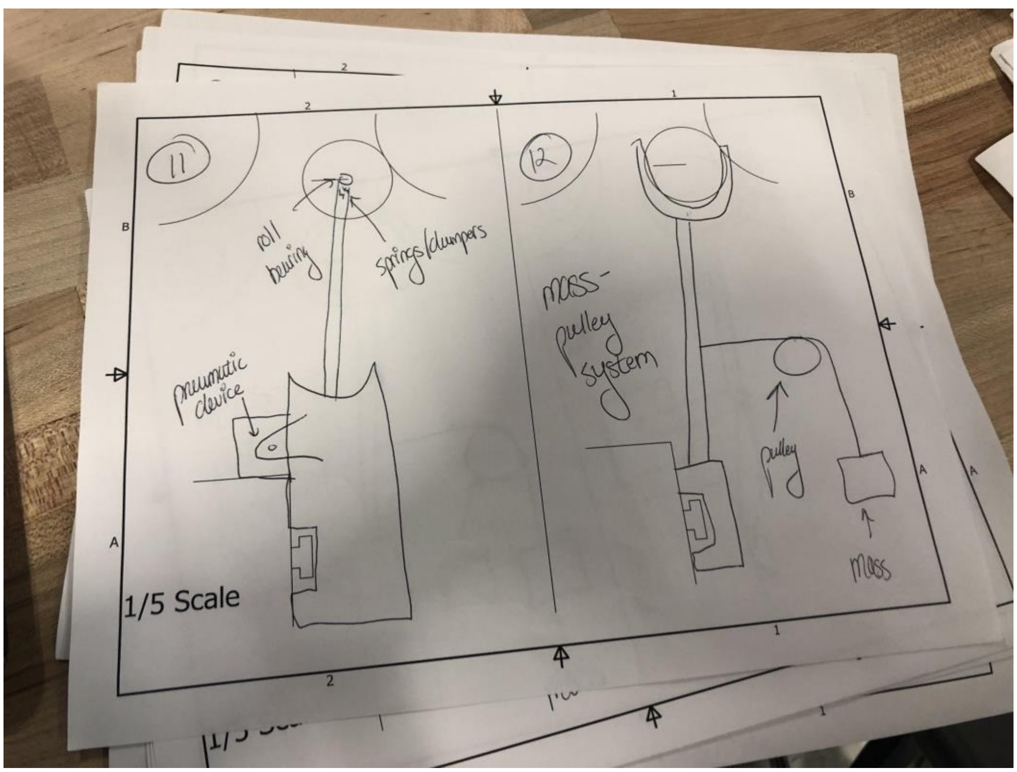


Figure 37: David Rainone Concepts 11 and 12

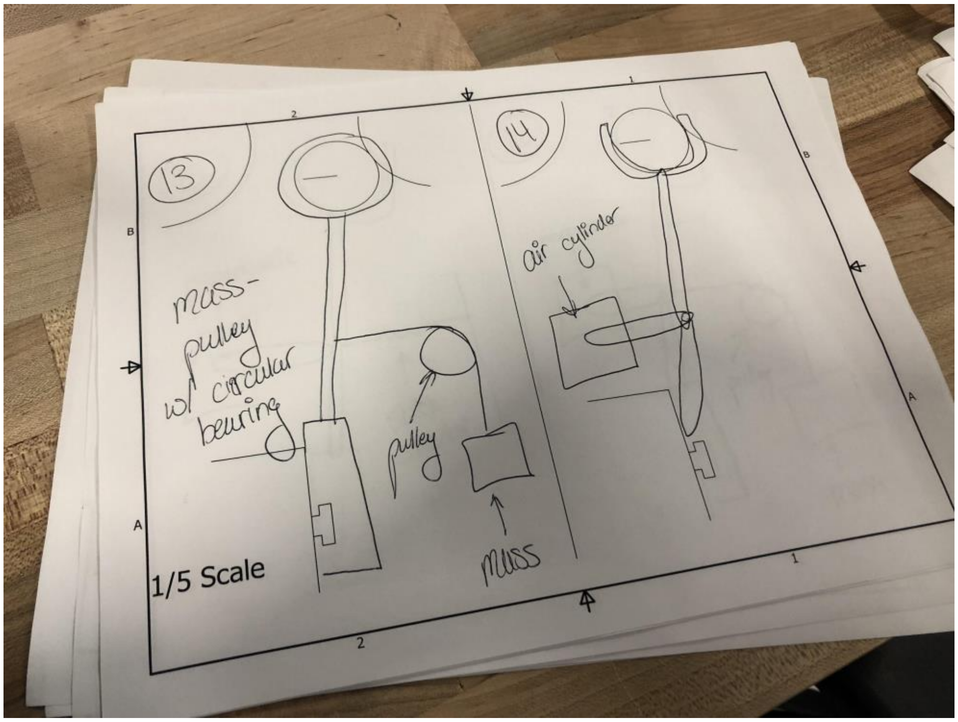


Figure 38: David Rainone Concepts 13 and 14

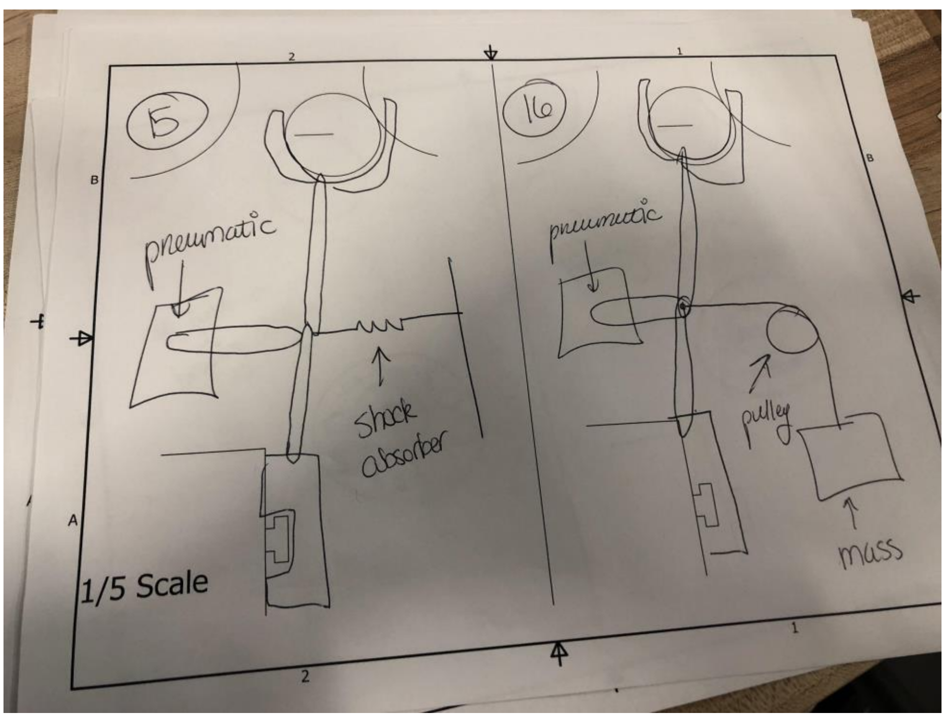


Figure 39: David Rainone Concepts 15 and 16

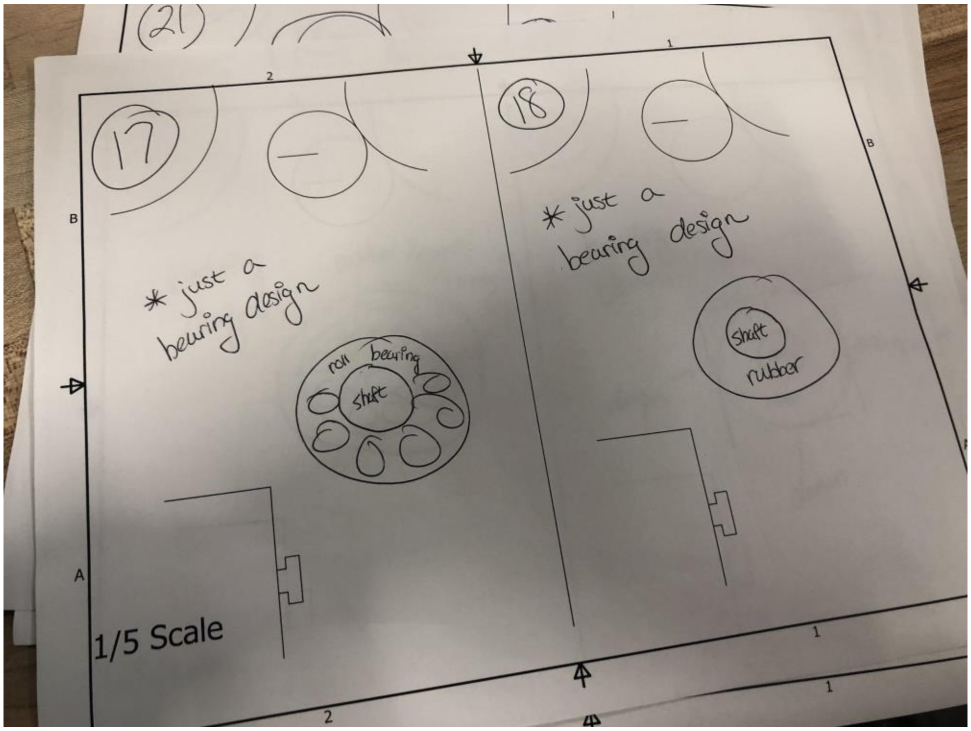


Figure 40: David Rainone Concepts 17 and 18

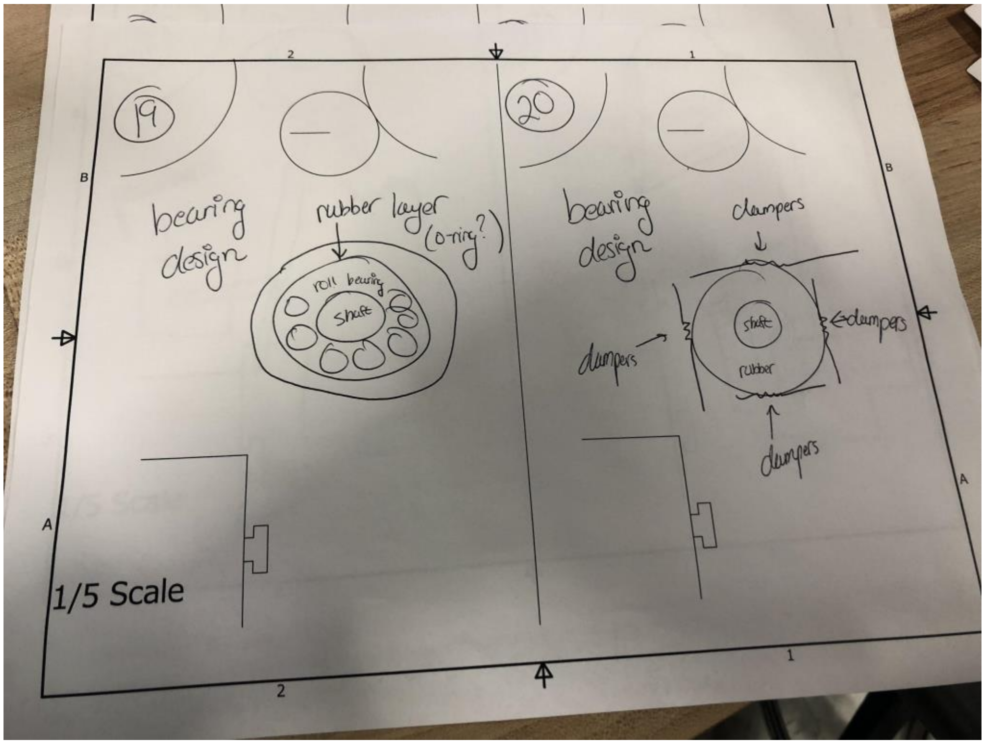


Figure 41: David Rainone Concepts 19 and 20

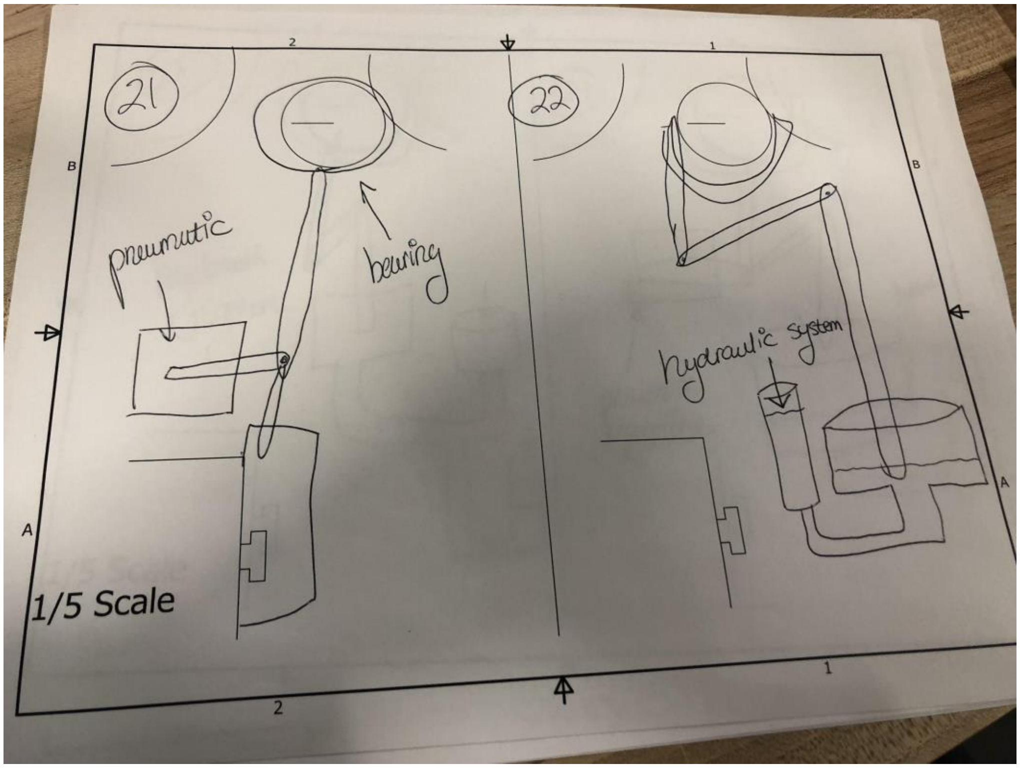


Figure 42: David Rainone Concepts 21 and 22

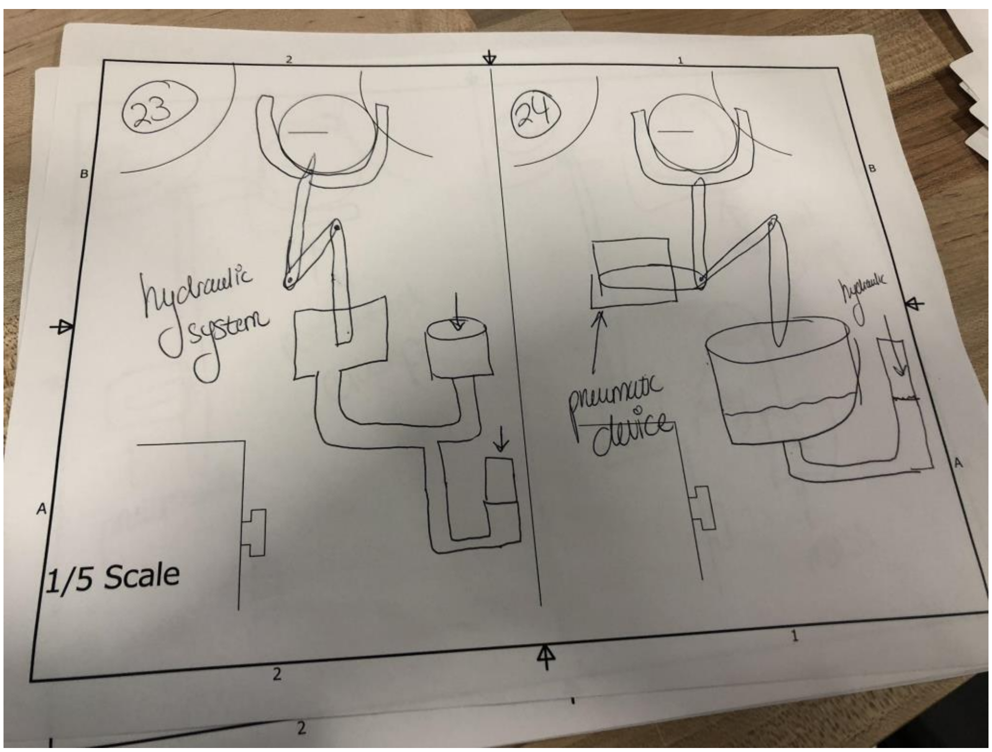


Figure 43: David Rainone Concepts 23 and 24

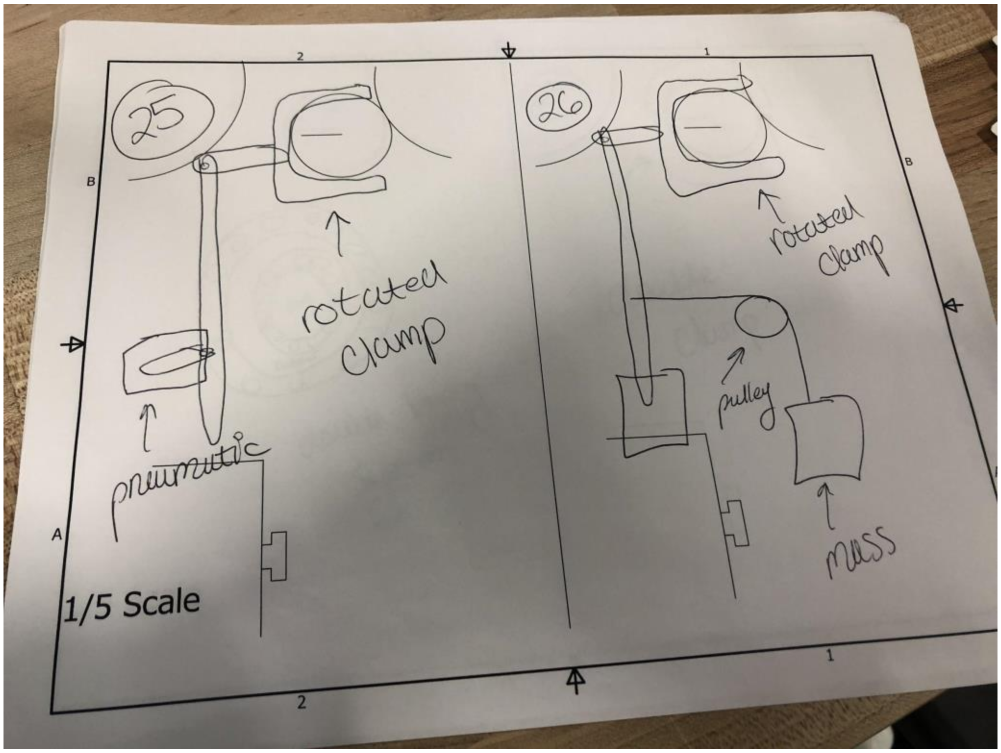


Figure 44: David Rainone Concepts 25 and 26

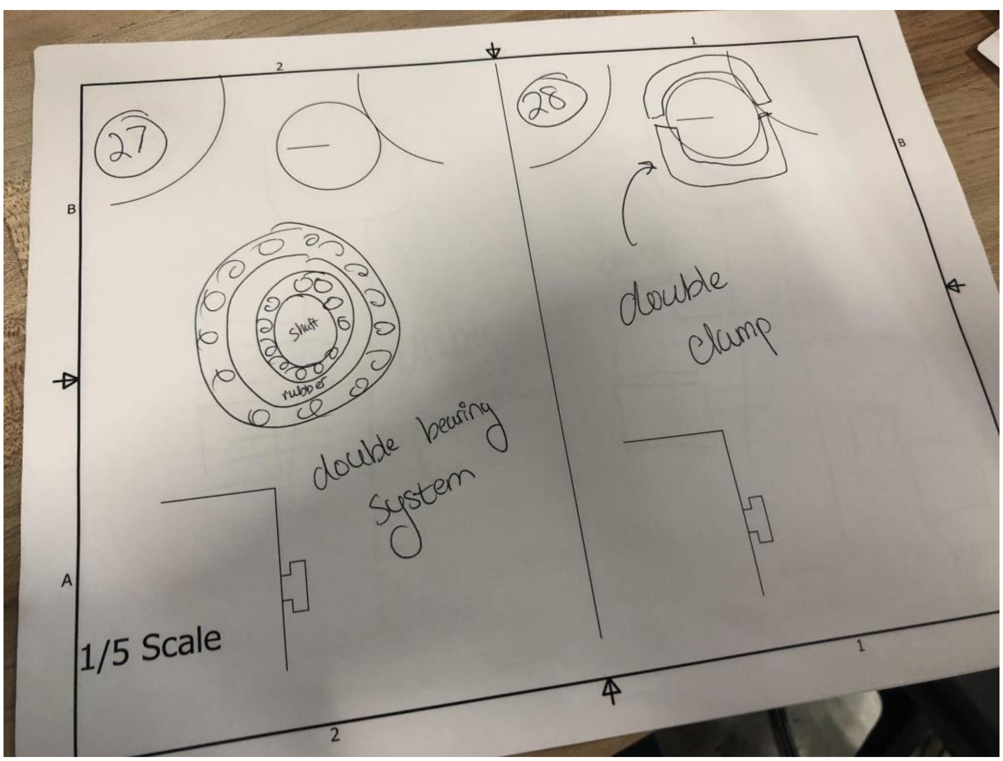


Figure 45: David Rainone Concepts 27 and 28

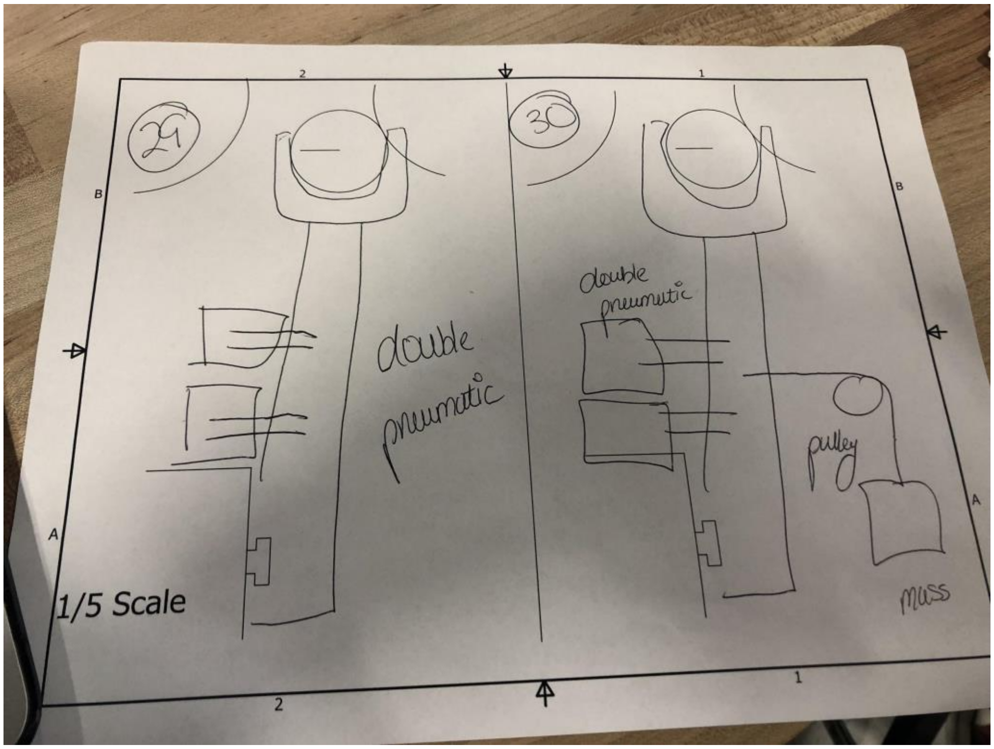


Figure 46: David Rainone Concepts 29 and 30

Table 4: Pugh of Rainone's Designs

		Concepts																													
Engineering Criteria	Reference Concept	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30
cost	3	s	s	s	+	+	-	-	-	s	s	-	+	+	-	+	s	+	-	-	+	-	+	-	-	+	-	s	s	-	+
compatibility	3	+	+	s	-	-	+	+	s	s	-	-	-	+	-	s	-	-	-	+	+	-	-	+	-	-	-	-	-	+	+
reliability	3	s	-	s	-	+	-	+	-	-	-	-	+	-	s	+	s	-	s	s	-	s	s	-	s	+	-	+	s	-	-
longevity	3	-	-	s	-	s	-	-	-	+	+	s	s	-	s	-	-	+	-	-	-	s	-	-	-	-	-	-	+	-	-
safety	3	-	-	s	-	s	s	s	s	+	+	-	-	-	s	-	s	s	+	s	+	s	s	s	s	s	+	-	s	s	s
# of Plusses		1	1	n/a	1	1	1	2	0	2	2	2	2	2	0	2	0	2	1	1	2	0	1	1	1	2	0	1	1	1	2
# of Minuses		2	3	n/a	4	1	3	2	3	1	2	2	2	3	2	2	2	2	3	2	2	2	2	3	3	2	4	3	1	3	2

Competitive/QFD Design Analysis

Because the company that made the slitter has gone out of business, there is no competition for nip roller arms for this machine. Toray has contacted the company that now specialized in servicing these machines as well and they expressed no interest in designing a new nip roller system due to the overall lack of demand. Other companies that make slitting machines have slightly different systems, some which are better than others, but not that fit the specific requirements and constraints for this application. Because of the lack of competition, we performed competitive analysis between our top concepts and the existing design. Each top concept is summarized below the house of quality diagram.

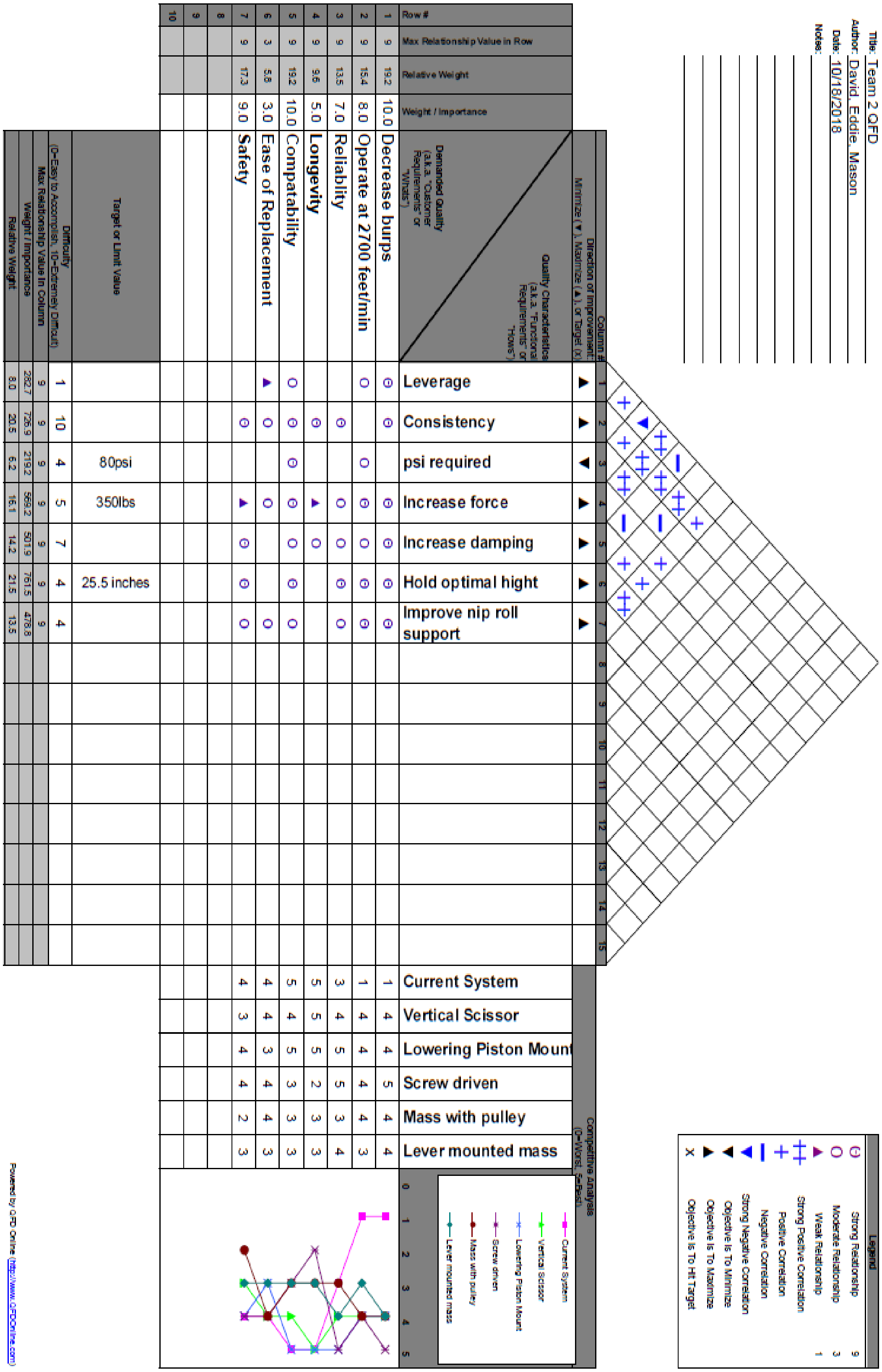


Figure 47: House of Performance QFD Analysis

Vertical Scissor System

This design used a modified four bar style system to increase mechanical advantage. In basic simulations in working model, shown in figure 48, the mechanical advantage was found the be .77, a large step up from the .5 of the existing system. This design ultimately failed due to concerns of jamming and lack of room to mount a large piston.

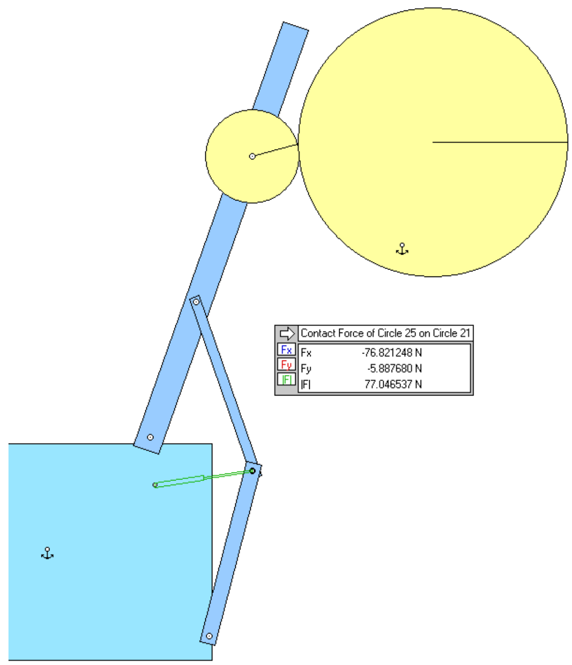


Figure 48: Working model of scissor design

Lowered Piston Mount

This design involves lowering the location of the piston to increase leverage. We found however that the to do this a small piston would be needed to fit in the space, canceling out any gain. However, we did end up using this concept in our current design but raised the pivot point instead of lowering the piston.

Screw driven system

This concept was based around using a lead screw to accurately control the position of the nip roller. This designs weakness was interfacing into the existing control system which is entirely based on force and allowing the nip rolled to be pushed by the contact force. These designs would require force sensors to control position, making it impractical.

Mass with Pulley

This design used a mass hanging from a cable to generate force and a pulley system to redirect the force to help with mechanical advantage. The mass was a great source of force and added inertia to the system to reduce the amplitude of vibrations, but the pulley cable system added unpredictability. A swing mass inside the machine was a serious safety concern.

Lever mounted mass

By taking the mass on a cable idea and replacing the cable with a rigid lever we reached the design we moved forward with. By also incorporating the lever proportion shift from the lowered piston mount design we found this design could produce the maximum force and was the easiest to control. Tradeoffs include the difficulty of installing, but this would only be during set up and maintenance.

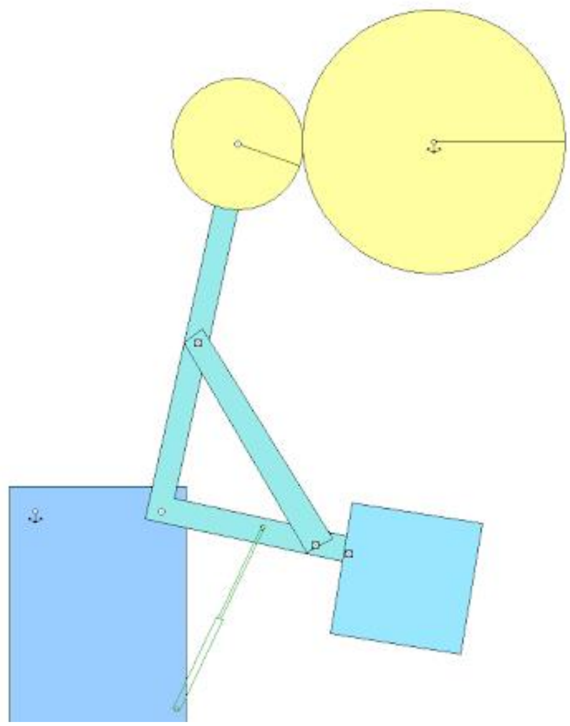


Figure 49: Lever mounted mass concept

Design for Prototype Effectiveness

Due to the need for adjustability in the design of the prototype, 80/20 aluminum was chosen for the frame of the support to hold the mounting arm. This allowed for flexibility in building the prototype, as well as the ability to make any changes to find ideal placements of components in the frame. The reason 80/20 was chosen was because the connectors and rods allowed for flexibility in testing and allowed for different placements of the piston on the bottom of the frame. Additionally, 80/20 is relatively inexpensive, and drove down the cost of materials greatly.

In order to support the integrity of the frame, shoulder bolts were chosen for any rotating components. This was so that the frame could withstand the forces in the high stress areas. Additionally, in order to further support the high stress areas, steel plates and large corner connectors were chosen in areas of anticipated high stress. This yielded a safety factor of 2 for the subjected loads, which was sufficient in showing that the design would not fail under testing. This showed that the design was reliable and would be able to withstand the high stresses, especially in the critical stress areas along the arm and in the rotating components of the design.

The prototype was built at approximately half scale, to ease the testing. Using a large full-scale prototype would have been difficult (and more expensive), since greater forces would have been needed, and would have required a larger scale to read higher nip forces. A larger scale would have driven the cost much higher. Modeling this at half scale allowed for smaller magnitudes of forces to be exerted, and these values could be used to prove the theoretical calculations in the engineering analysis. Lastly, the expected results of the full scale can be modeled by scaling the small-scale results and applying the proved equations to the full-scale model.

Project Specific Details & Analysis

This project involves the production of plastic film, and improving the process related to this. After speaking with Toray Plastics, it became evident that the best way to improve their slitting process, and improve operating speeds is to increase the nip force. The current design, which has two mounting arms on each side, does not produce sufficient force to keep up with current production speeds and demands. Toray Plastics determined and calculated a target force of 750 pounds, per arm (one arm on each side), in order to improve production standards. After speaking with engineers from the company, this target force proves to be sufficient for increasing production speeds.

In order to increase this nip force, the mounting arms must be improved, so that they can actually produce this nip force. However, the air cylinders are limited to 80 psi, and are difficult to achieve this target force. Adding a mass to assist the force from the piston was the favorable solution to increasing the nip force, after meeting with and presenting multiple solution options to engineers at Toray Plastics. The engineers at Toray Plastics felt it was necessary to prove the concept by developing guiding equations, and then building a prototype to show that these guiding equations are true, which can be used to find parameters to scale the prototype to a full-size model. Some parameters to be measured include nip force, pressure, and weight of the added mass.

The biggest concern of the engineers from Toray Plastics with the design is issues with space. The mounting arms are near the machine's nip roller, which has a wide range of motion, as the roll increases in size. However, there is not much space below the mounting arms, and finding a place to add the mass would be difficult. The mass would have to be shaped so that it would fit in the current production line. This could potentially take away from the effectiveness of the mass and would have to be studied more. Additionally, the engineers at Toray Plastics suggested investigating different, perhaps more effective pistons, in order to improve the process more.

While this project is specifically to improve a system from Toray Plastics, there could be more potential users and applications for this project. For example, the nip roll function is very similar to the laminating process, where force is required to press the film together. A smaller scale version of this mounting arm, or concept related to it, could be used to supply enough force for the laminating process, and perhaps improve production speeds as well.

Detailed Product Design

The final product was shaped from the original concept mainly by the space constraints and structural requirements. The space constraints are shown in figure 1 on page 6. This mainly affected the shape of the mass and the length of the arm. The final assembly with only critical dimensions is shown in figure 50.

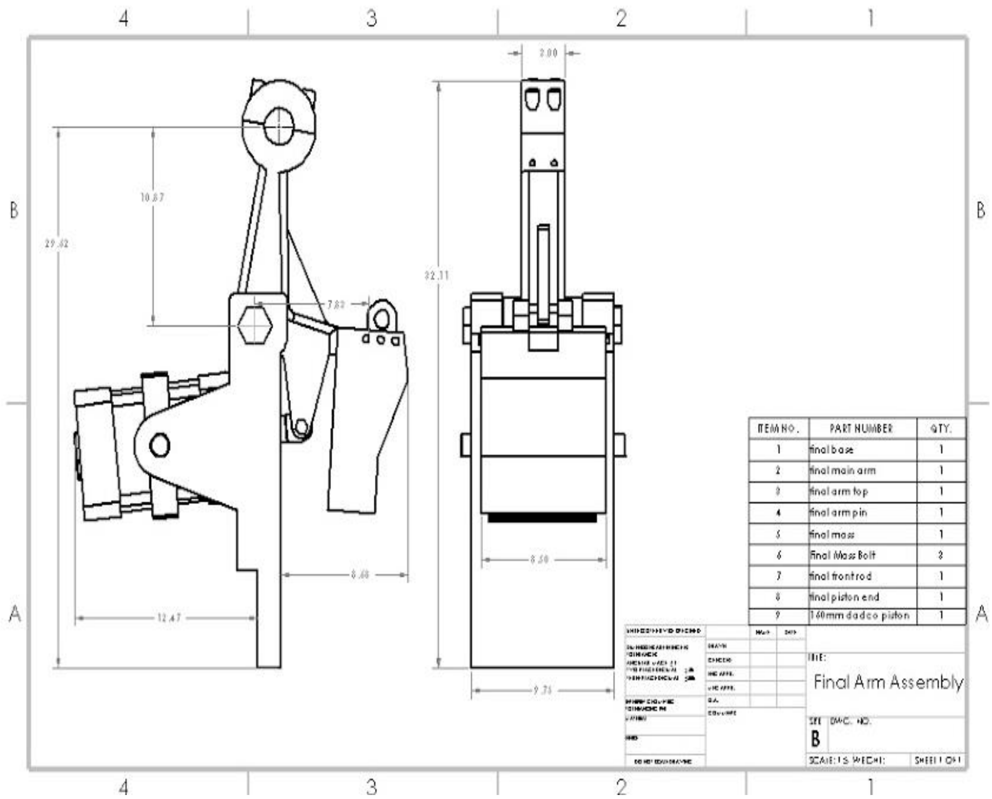


Figure 50: Critical dimensions of final assembly

The mass is shown in figure 51. The angular shape on the front end is designed to fit in the opening available behind the arm that supports the customer roll and moves the center of mass of the mass further from the pivot to increase torque. The mass was also widened considerable from the first iteration of the design to maintain its volume despite the smaller side profile. Lift points were added to the top of the mass to allow for easier mounting and dismounting.

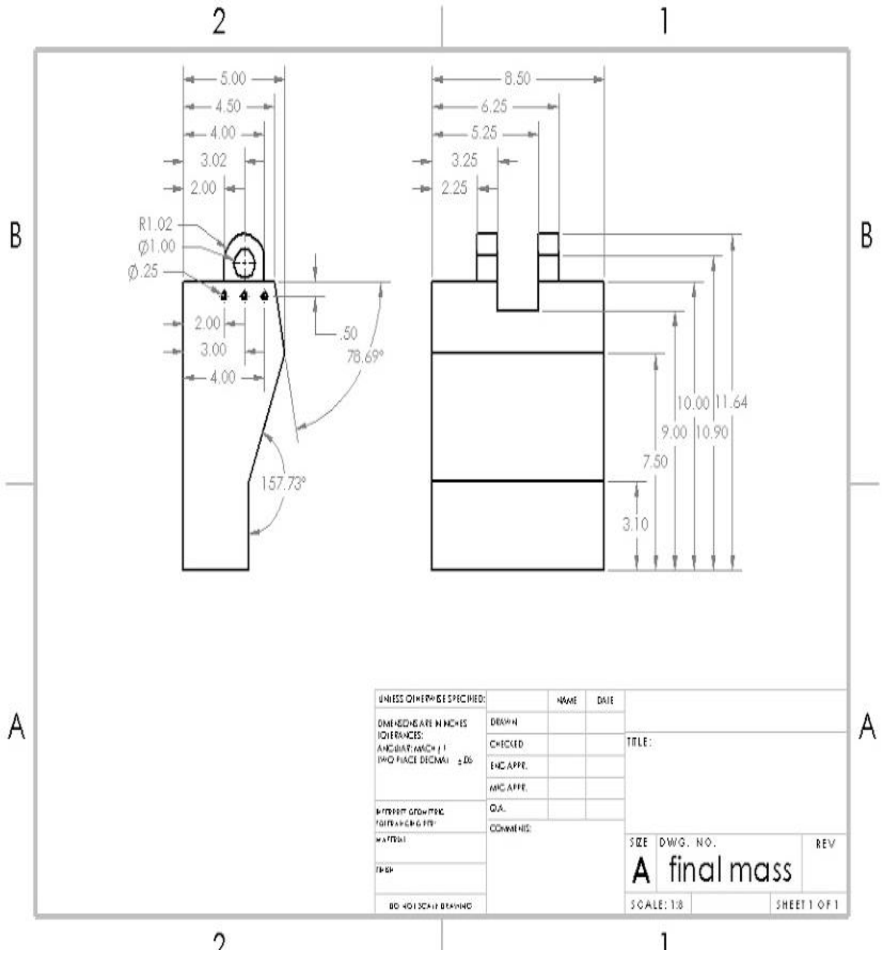


Figure 51: Final Mass Drawing

The base geometry was changed to be able to accommodate the larger piston and the increased force. The rough geometry was still based on the original to allow Toray to recycle components from the mounting system. The piston with a center pivot mount was chosen because it allowed the supports for it to be much shorter, reducing stress. Static analysis was done in solid works to find corners that acted as stress concentrator and were more likely to form cracks. These corners were then rounded in most cases to reduce the stress concentration. Figure 52 shows the stress in the final model and figure 53 shows the dimensions.

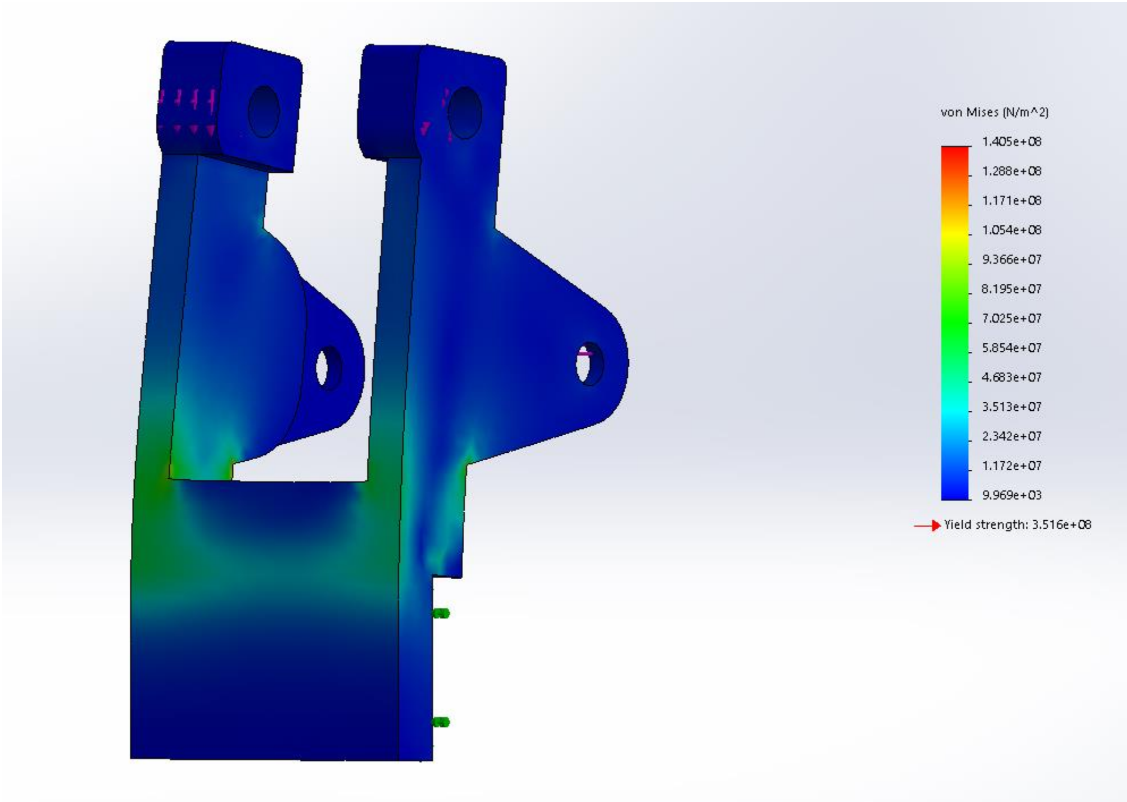


Figure 52: Stress in Redesigned Base

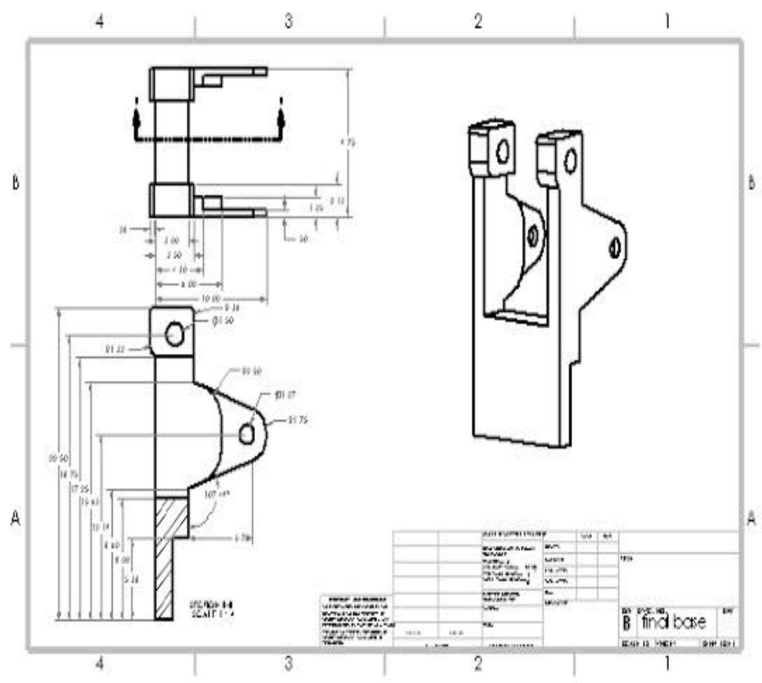


Figure 53: Drawing of Base

The final design of the arm also designed based of the original arm, but with improvements. First forward-facing extension to hold the mass was added. Have this out in front also allowed us to move the attachment point for the piston forward, maintaining its mechanical properties but creating space for a larger piston. Solidworks simulations showed what we suspected, that having both the weight of the mass and the force of piston on one arm put too much force on one side of the pivot point on the arm. To solve this problem, we looked back to our original working model simulation shown in figure 49. From this we were inspired to add the front fin like brace that helps transfer force between the two perpendicular levers. Again, important corners were rounded to reduce stress concentrations that could lead to catastrophic cracking failure. The arm was also made thicker to handle the force, which was made possible by the widened base. The top of the arm was redesigned also into a classic clamping collar. Toray no longer desired the quick interchangeable roller system from the original design because it was less secure. Figure 54 shows the stress in the redesigned arm with a 100 lbs mass attached.

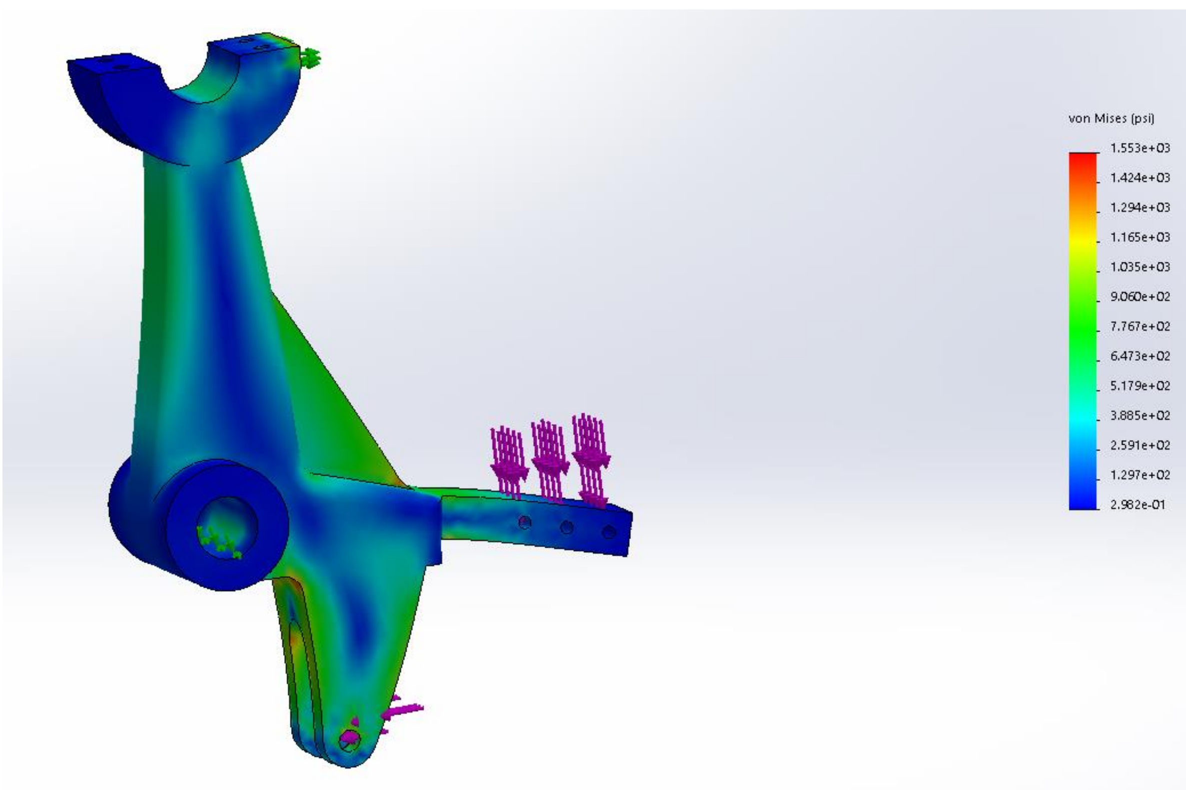


Figure 54: Stress in Arm Under Load

All the parts needed for the final product are listed in the bill of materials in table __.

Table 5: Bill of Materials

Part Name	Quantity	Description	Reference
Base	1	Support for nip roller arm	Figure 53
Arm	1	Arm for applying nip force	Figure 54
Arm Top	1	Secures nip roller	
Main Pivot Pin	1	Pivot point for arm, base	
Mass	1	Used to generate torque	Figure 51
Mass Bolts	3	For mounting mass to arm	
160mm Dadco Piston #8	1	Large piston with central pivot mounts	Part number: HP.W.160.100.G.3.T4
Piston Adapter	1	Converts threaded piston end to attach to shoulder bolt	
Attachment Shoulder Bolt	1	Attaches piston to arm	

Engineering Analysis

When performing a static analysis on the lever arm, a few parameters are important: the length of the arm, the diameter of the piston, and the weight of the mass. Figure 55 shows the equations related to the static analysis of the mounting arm.

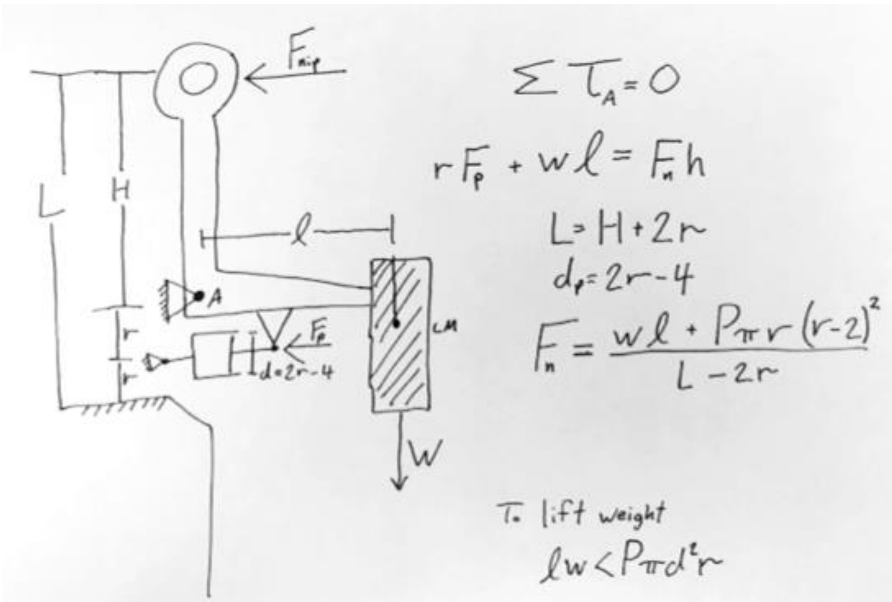


Figure 55: Static Analysis of Weighted Mass

The force of the piston will act to the left, and the weight will act straight down. When these two are combined as torques about the point A, they act together to increase the torque about this point. This will, in turn, increase the nip force (as shown at the top of the diagram). When summing the torques, the nip force can be solved, and expressed as a function as length of the arm, diameter of the piston, and weight of the mass. As expected, the nip force increases as the weight is increases and the diameter of the piston is increased.

When studying designs, it is important to establish advantages and disadvantages of the design. The advantages show why the design will perform well, and the disadvantages show where improvements can be made in the design. This information is tabulated in Table 2.

Table 6: Pros and Cons of Weighted Design

Pros	Cons
Mass will increase the leverage, which will increase the nip force	Unsure of a “practical” weight of the mass
Increasing the overall mass of the system will increase inertia; will reduce vibrations	Cost and ease of the replacement of the mass
Single piston, easier to control	Limited by space in the current machine
More force without using more pressure	
Lack of moving parts (less parts means less can fail)	
Ease of installation - will go directly into the current machine	

In order to prove that the ideas provided will work, extensive statics and Matlab analysis was performed on the system. Figure 55 provides the static analysis for the weighted mass concept. The torques were summed about the point A, and algebraic manipulation determined the nip force applied, given different weights, mounting arm lengths, and piston radii. Given a fixed mounting arm length, the nip force can be expressed as a function of weight and piston diameter, as seen in Figure 56.

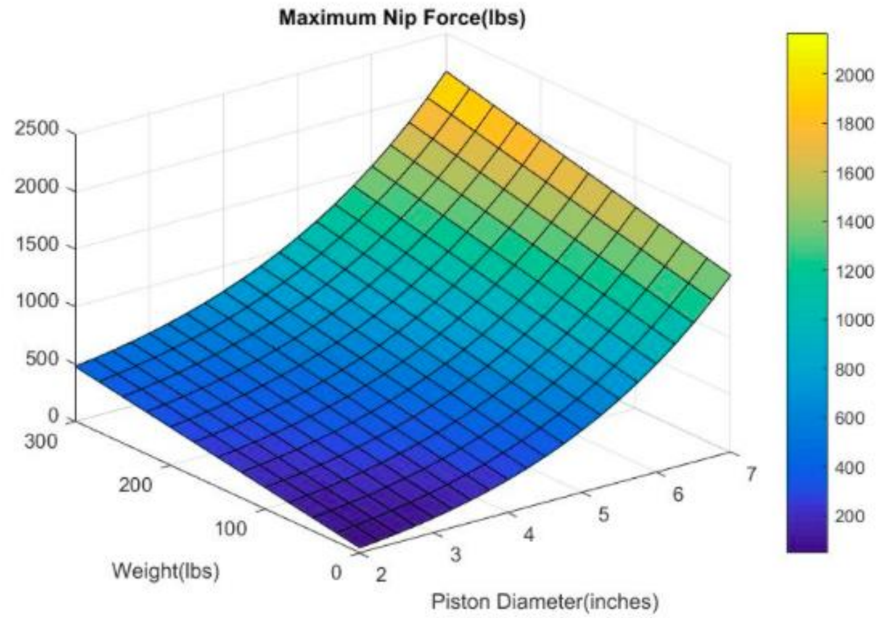


Figure 56: Maximum Nip Force vs. Weight and Piston Diameter

As shown by Figure 56, the maximum nip force increases by the square of the piston diameter. Additionally, the maximum nip force increases linearly with the weight. In addition to adding the weight, it will be beneficial to change the current design in the piston, and create a larger piston, which will ultimately more nip force given the same pressure of 80 psi.

When plotting Figure 56, it is important to view it as a contour, to better understand how changing the piston diameter and weight can together help achieve the target force of 750 pounds per mounting arm. In Figure 57, the acceptable ranges of piston diameters and weights, in accordance with the target force of 750 pounds, can be very clearly seen. It is important to note that the current piston diameter is 4 inches, so to achieve the target force of 750 pounds, without changing the current piston, the mass must exceed 280 pounds. This is an impractical weight to add to the system; so, it will be essential to alter the design of the piston, as well as adding the weight.

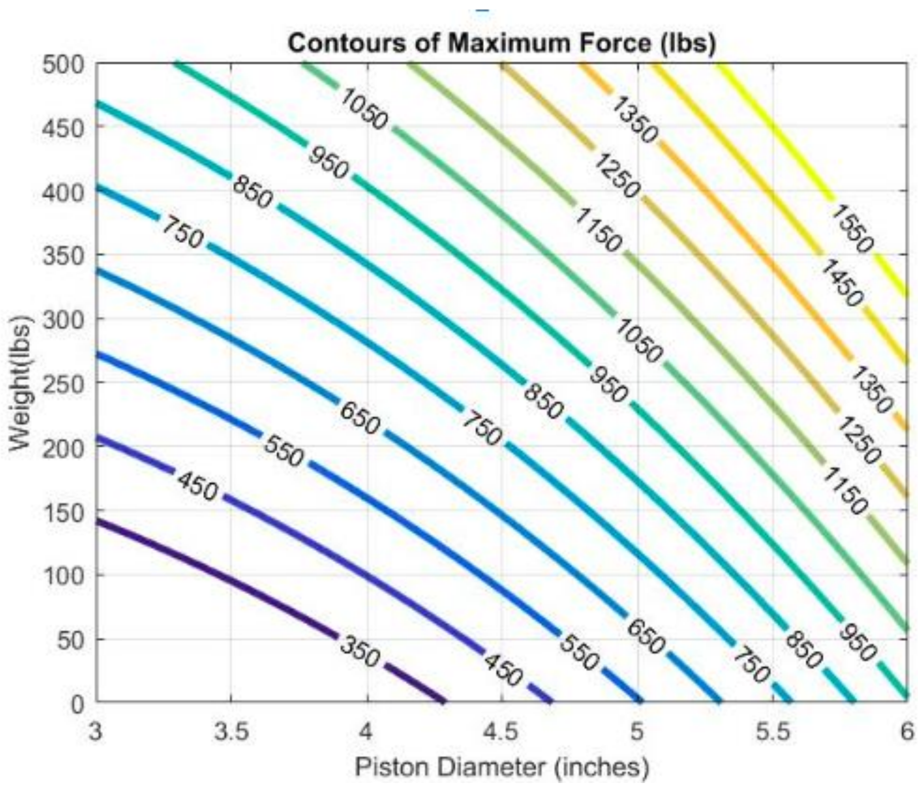


Figure 57: Maximum Force Contours

Adding the weight introduces a new problem with the piston: will the piston be able to lift the weight (and the nip roller) in the opposite direction, while the customer roll becomes larger, and during assembly and disassembly of the customer roll? After performing a static analysis on the piston and the weight, Figure 58 was created to satisfy this constraint.

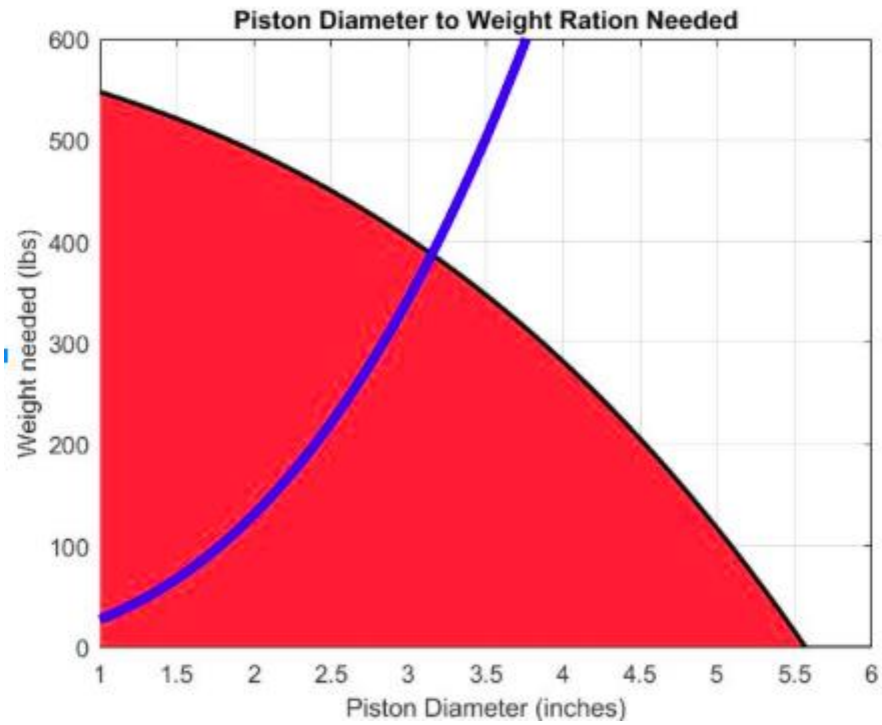


Figure 58: Piston Diameter vs. Weight

As shown by Figure 58, there are several constraints that must be satisfied, for this design to work. The first, is that the target force provided by Toray Plastics is 750 pounds. Any piston diameter and weight combination under the red area will not satisfy the target force. Additionally, the piston must be able to lift the weight, so any diameter and weight combination to the left of the blue line will not satisfy this constraint. This limits the scope of the design even further.

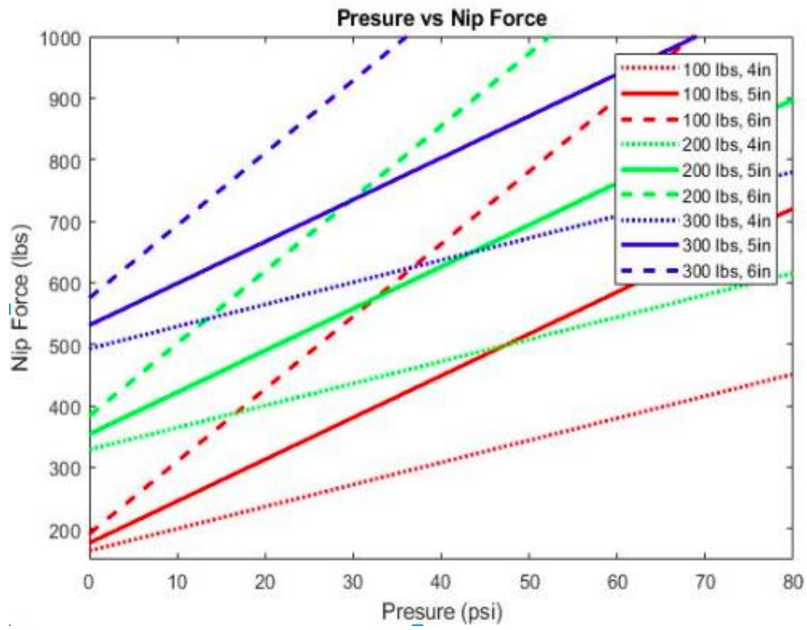


Figure 59: Pressure vs. Nip Force

Lastly, as shown by Figure 59, it is important to note that the pressure of the air cylinder is linearly related to the total nip force. An increase in piston diameter will increase the slope of this relation, making the nip force for the same pressure more effective. Additionally, increasing the weight will increase the intercept of this relationship, which will also more effectively increase the nip force. Although the pressure is limited to 80 psi (based on Toray's current production standards), altering piston diameter and weight can assist in achieving the target nip force of 750 pounds.

Manufacturing

The first step of manufacturing the prototype was to cut the 80/20 bar stock into the correct sizes. Additionally, through holes were machined into the associated bars that would require through holes for the shoulder bolts. After that, through holes were also machined into the steel plates, in order to attach them to the high stress areas in the prototype. Additionally, holes were machined into them for the shoulder bolt through holes. After the through holes were machined, all of the bars for the frame were put together using adjustable connectors and screws. Additionally, the shoulder bolt was put through the arm component, and the piston was added at the end. The scale was attached to a paracord, which was cut and had the edges burned, to solidify the ends of the paracord. This was tied multiple times around the top of the arm and the middle of the frame, which will be in tension during testing.

It is important to note that for adjustability purposes, the weights were added during testing, and were not a direct part of the manufacturing process of this prototype. The only part related to the weights that were added was the rod used to support the weights, which was attached to the end of the arm with a through hole and some screws.

Figure 60: Almost completed prototype, without the piston and the scale.

The building process had some challenges that were associated with it. The first of these challenges was tight tolerances when machining holes, as well as aligning the 80/20 frame. The frame was difficult to mount together, since the alignment issues from these tight tolerances made the assembly process difficult. Additionally, developing a way to mount the scale was difficult, but a paracord rated to 660 pounds was selected, which provided a factor of safety of approximately 6, given the tensions and nip forces the paracord will be subjected to during testing. Lastly, a lengthy (and longer than expected) shipping time for the piston forced the project to be behind schedule and delayed the testing process by a couple of weeks, which was very critical.

The study of mass producing and developing a manufacturing process for this product is not necessary, since the product is used for one specific company's production lines. This product will not be mass produced and is never intended to be mass produced. The application is very specific, so improving the manufacturing process for this product is an unnecessary task, since Toray Plastics is the only customer for the mounting arms.

Testing

The effectiveness of the design was verified through numerous tests showing the relationships between weight and nip force, as well as pressure and nip force. Proving these relationships is important, in order to scale the model to a full size. Since this is a prototype, it is important to prove that the guiding equations suffice, and then using these guiding equations, parameters can be set and established so that the full model will succeed the target force of 750 pounds per arm.

The first test was to verify the sensitivity of the scale and show that the scale was able to read forces within a reasonable variation of 0.25 pounds. Upon completion of this test, it showed that the scale read forces with variation of 0.2 pounds, which was underneath the target of 0.25 pounds, so the sensitivity of the scale was proved efficient. This is important to show, since this can cause slight variation in the data; the worse the variation in the scale, the worse the data readings may become, which will not show the relationships necessary to prove the guiding equations as shown by the engineering analysis.

The second test was to confirm a linear relationship between mass and force. In order to prove this the success criteria was established to be an R value of greater than 0.95, when putting a trendline on the data. Mass was added, 10 pounds at a time, and the force reading was plotted as a function of weight of the mass. A trendline was fit to this data, using Microsoft Excel, and an R value of 0.98 was noted. This R value exceeded the R value of the success criteria, so the relationship between weight and nip force was proved linear by this test.

The following test was to verify the cylinder and show that no leaks were present in the system. In order to do this, air was added, and the reading on the pressure gauge should not change over a short period of time (approximately 10 seconds). If the pressure does not change, then it can be verified that no leaks are present anywhere in the system, which will show that the pressure readings are constant and consistent. The results of this showed that the pressure decreased slightly over time, which verified that there may have been a leak in the system. In order to verify the test results are accurate, the target pressure should be overestimated when applying pressure, and then the leak will cause the pressure to go down. When the pressure reaches the actual target pressure, immediately read the force output on the scale, so the results can represent the target pressure as accurately as possible.

The last test is the full test, which relates the nip force as functions of pressure and weight. The mass was added, 10 pounds at a time. But with each mass, pressure was added, 15 psi at a time, until 60 psi was reached. The force was read at each pressure reading: 0 psi, 15 psi, 30 psi, 45 psi, and 60 psi. Then, this process was repeated for trials of 0 pounds, 10 pounds, 20 pounds, 30 pounds, 40 pounds, and 50 pounds of weight. Force was outputted at each interval, and recorded, tabulated, and analyzed.

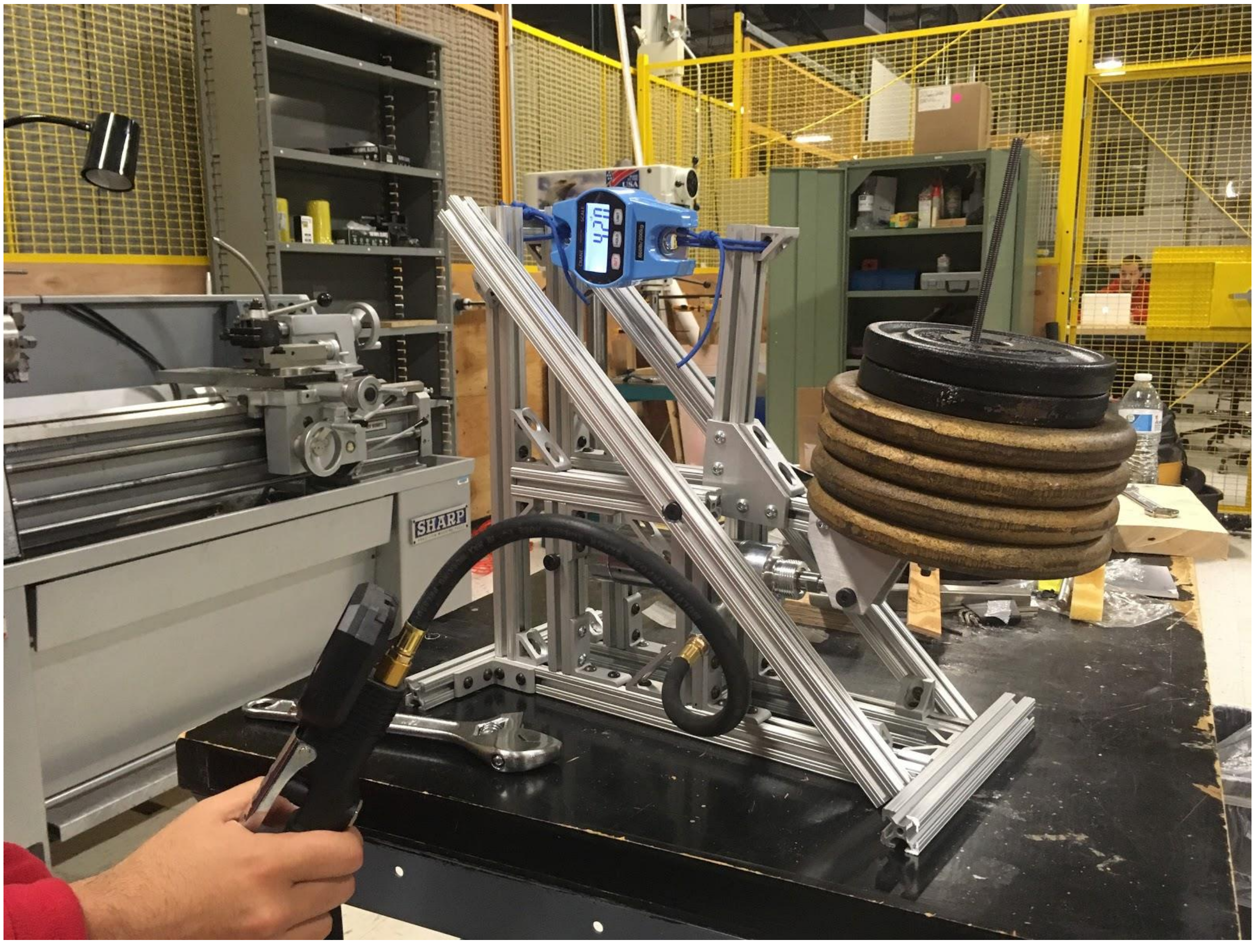


Figure 61: Full test 1 setup.

These tests are summarized in Table 7, to easily display these results.

Table 7: Testing Matrix

Test:	**Reason:**	**Success Criteria:**	**Results:**	**Plan Change:**
Scale Calibration	Sensitivity	Under .25 lbs	.2 lbs	None
Mass Only Testing	Confirm Linear	R > .95	R = .98	None
Pressure Testing	Pneumatic Leaks	Holds Pressure	Slight decrease in pressure over time	Overshoot target pressure and come down
Full Test 1	Data for Analysis	R > .95; Max error below 5 lbs	Max Error: 3.45 lbs	Proceed with analysis

In order to prove the guiding equations, the measured nip force was plotted as a function of pressure and weight. It is important to note that without any weight and any pressure, the force reading was negative, and no tension was measured in the cord. This is due to the spring in the piston. These equations were adjusted to reduce the measured nip force due to this spring. The weight and pressure relationships were confirmed linear, as shown by the plane in Figure 62.

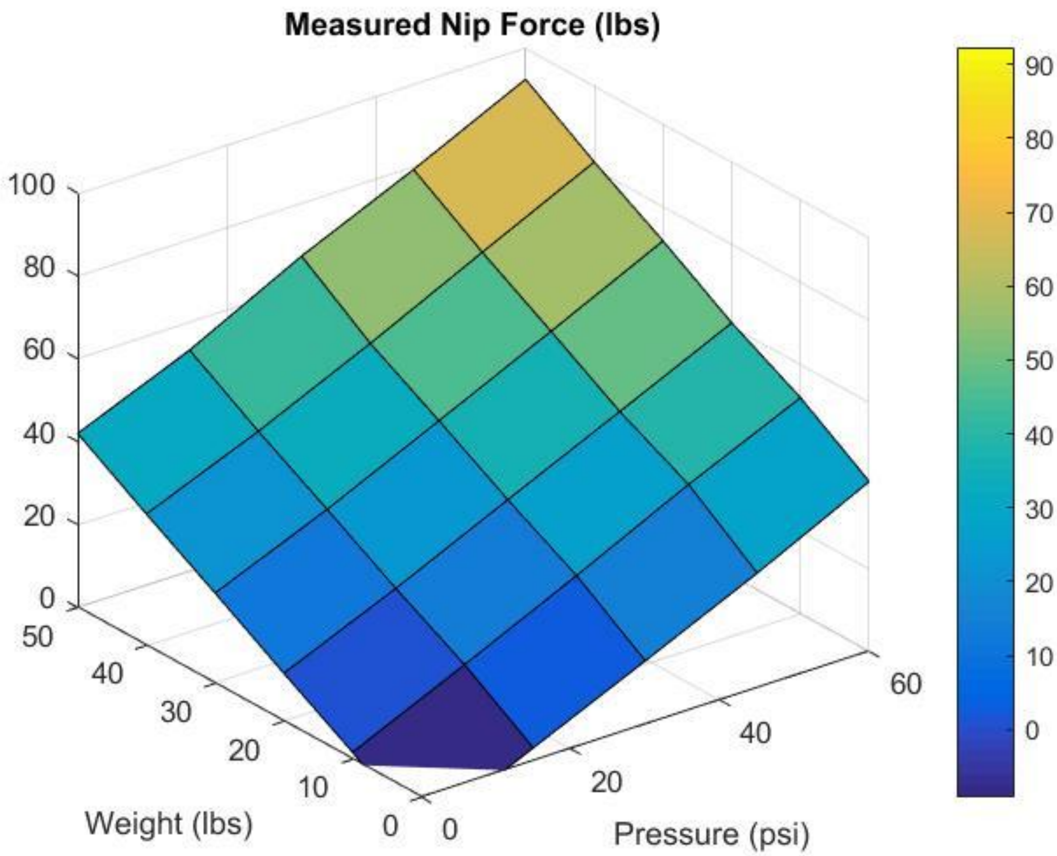


Figure 62: Nip Force as a function of Weight and Pressure

The error in each trial of the experiment was calculated by subtracting the measured force from the theoretical force from the guiding equations. A contour of the error plot can be shown by figure 63. During the 10-pound test, the scale produced a few errors, which can be shown by the ridge in the error plot. However, the maximum error was 3.48 pounds, which occurred at the maximum force and pressure test, the final trial. This is expected, since the error will increase as the force readings go up, since the variation in the data increases as the actual data readings increase.

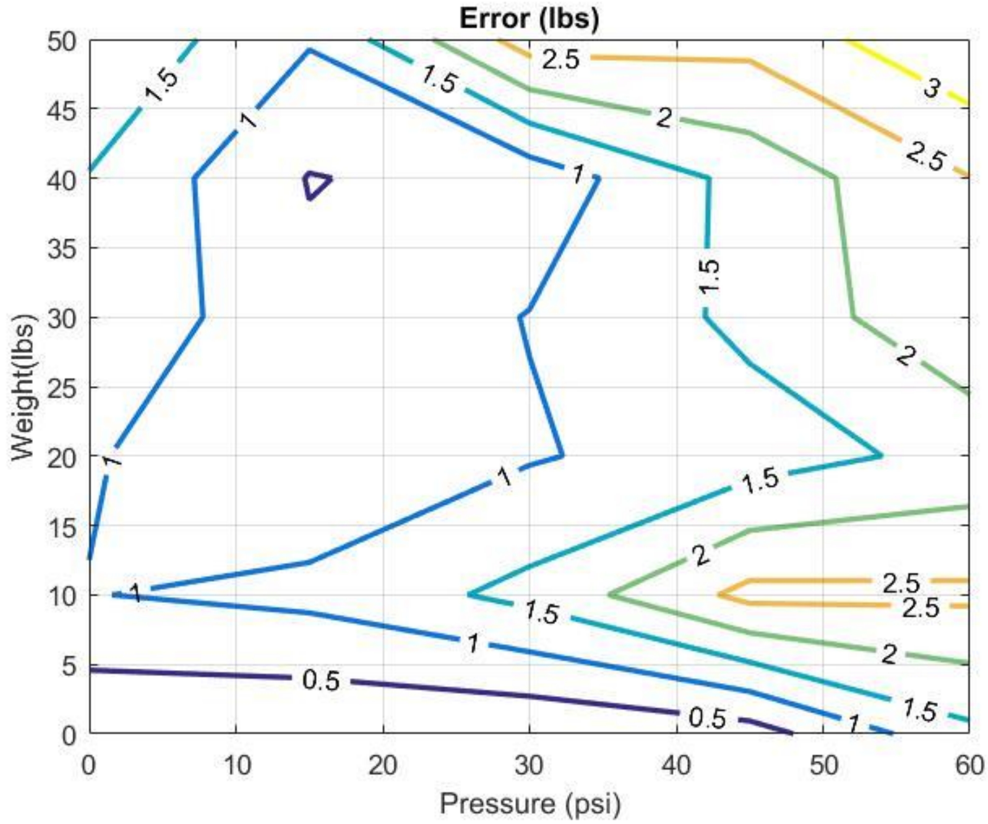


Figure 63: Error plot contours.

In order to prove linear relationships between weight and pressure, the slopes of the trendlines in the constant pressure and constant weight plots can be compared. Between the five trials of the constant pressure lines, the average increase in nip force was 1.0094 pounds per pound of weight added. For the five trials of the constant weight lines, the average increase in nip force was 0.836 pounds per psi of pressure added. The standard deviation of the weight slopes was 0.0089 pounds of nip force per pound of weight, and the standard deviation of the pressure slopes was 0.0066 pounds of nip force per psi of pressure added. These extremely low standard deviations showed that the data between the trials was consistent and helped accurately describe the relationships between pressure and nip force, as well as weight and nip force.

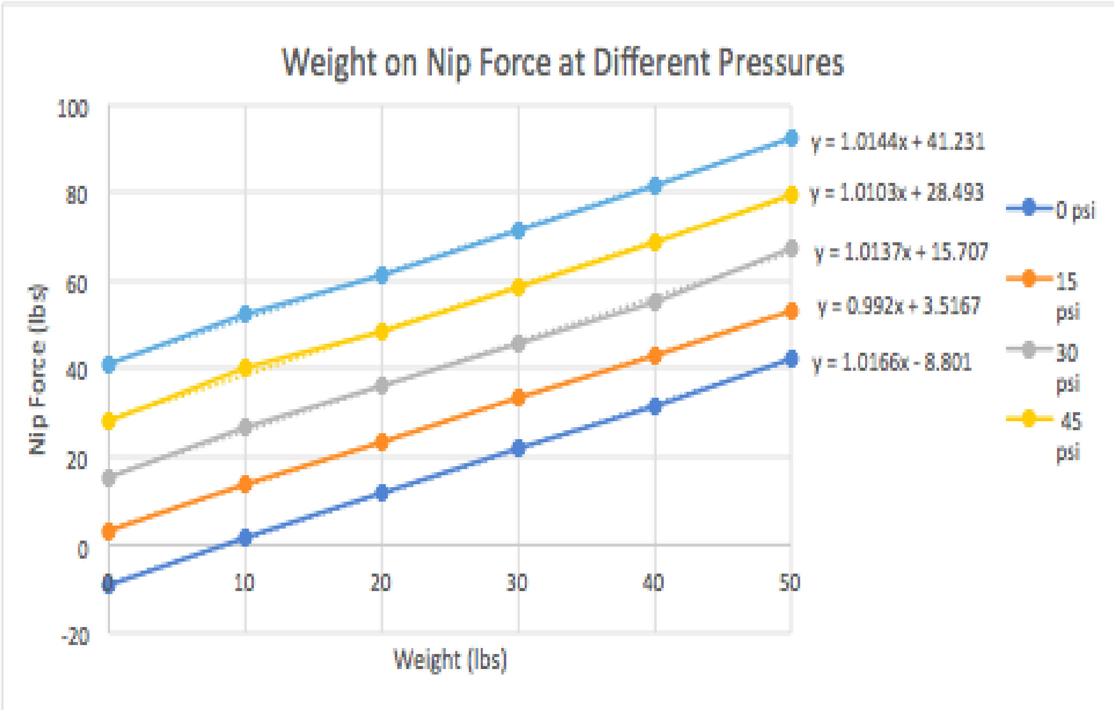


Figure 64: Weight vs. Nip Force, Constant Pressure Lines

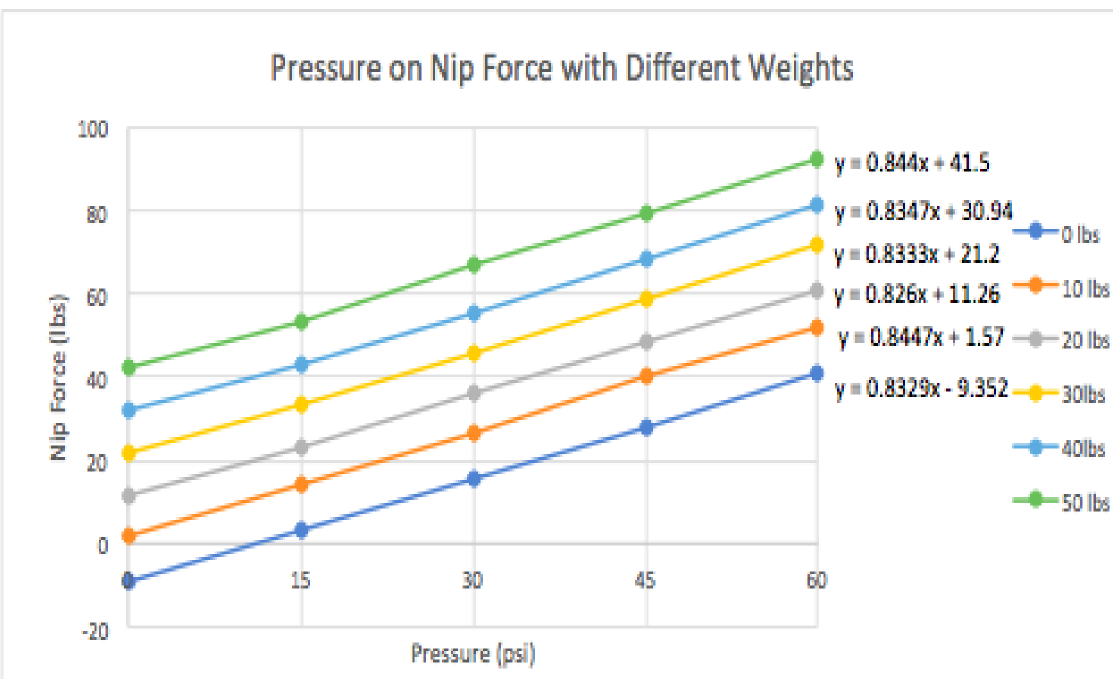


Figure 65: Pressure vs. Nip Force, Constant Weight Lines

Redesign

Because the testing proved the design functioned as intended most of the redesigning of the final product was mostly to make it easier to manufacture. The first step was to choose a piston for the final design to be based on the equations established in the engineering analysis. The internal diameter of the piston needed to be at least 5.2 inches. Very few companies manufacture a piston this large so finding one was difficult. Dadco was one of the only companies that made pistons big enough and they had the advantage of having multiple premade mounting configurations and CAD files of all their pistons available. Dadco also manufactures their piston with the air inputs in multiple configurations. This will allow the air hoses to be routed in the easiest way to the pneumatic outputs already in place on the slitter. The center mounting configuration on the piston was chosen because it allowed the support structure to be stronger, as discussed in the Detailed Product Design section previously.

The other large redesign was in the base to hold the system. Originally, the plan was to use the existing base reduce cost of producing the new arm. It became clear however that the old base could not fit the larger piston so it could not be reused. The new base is very closely structured on the time-tested design of the old one, with mainly dimensional changes. Because the need for a new base was identified late in the semester this was the must time efficient way to redesign, instead of starting from scratch. It also still allows hardware like the track mount to be reused if deemed strong enough.

Using the equations proven by the prototype, the final dimensions of the product, and the Matlab code found the appendix, a final performance graph was generated, figure 66.

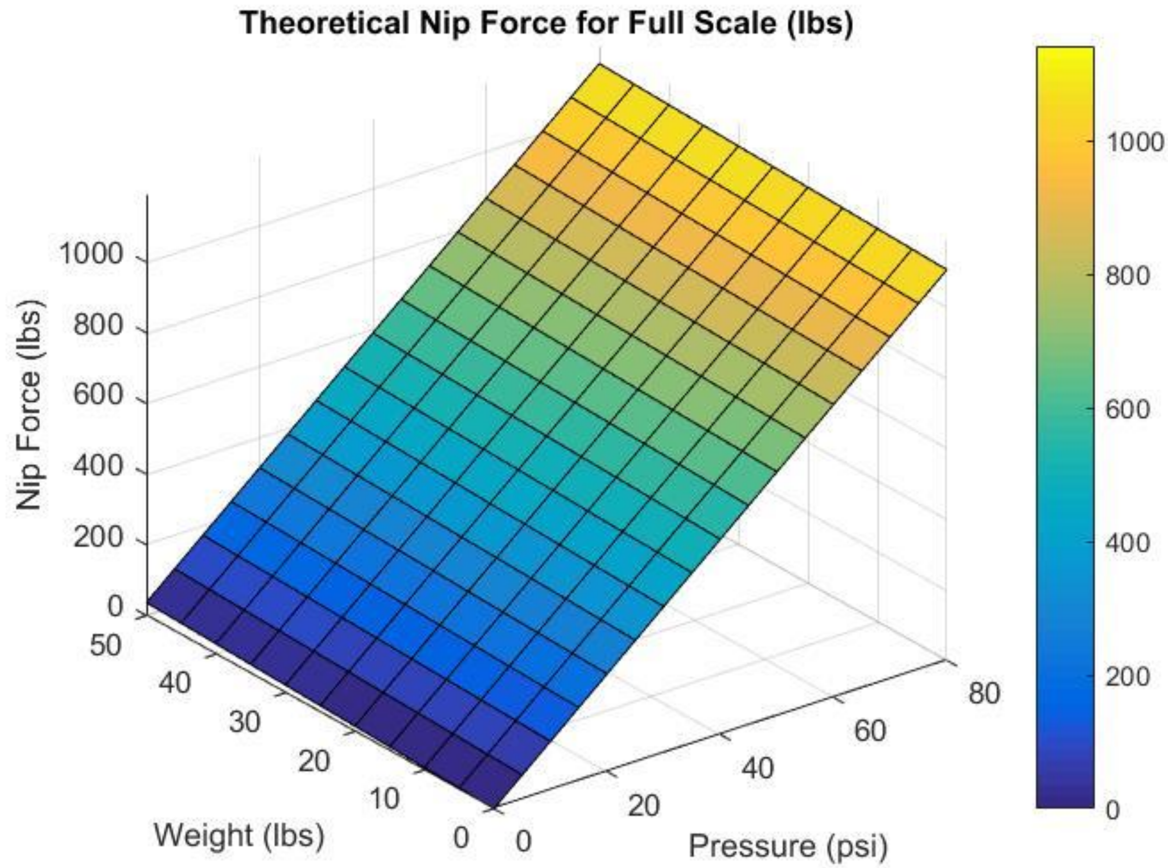
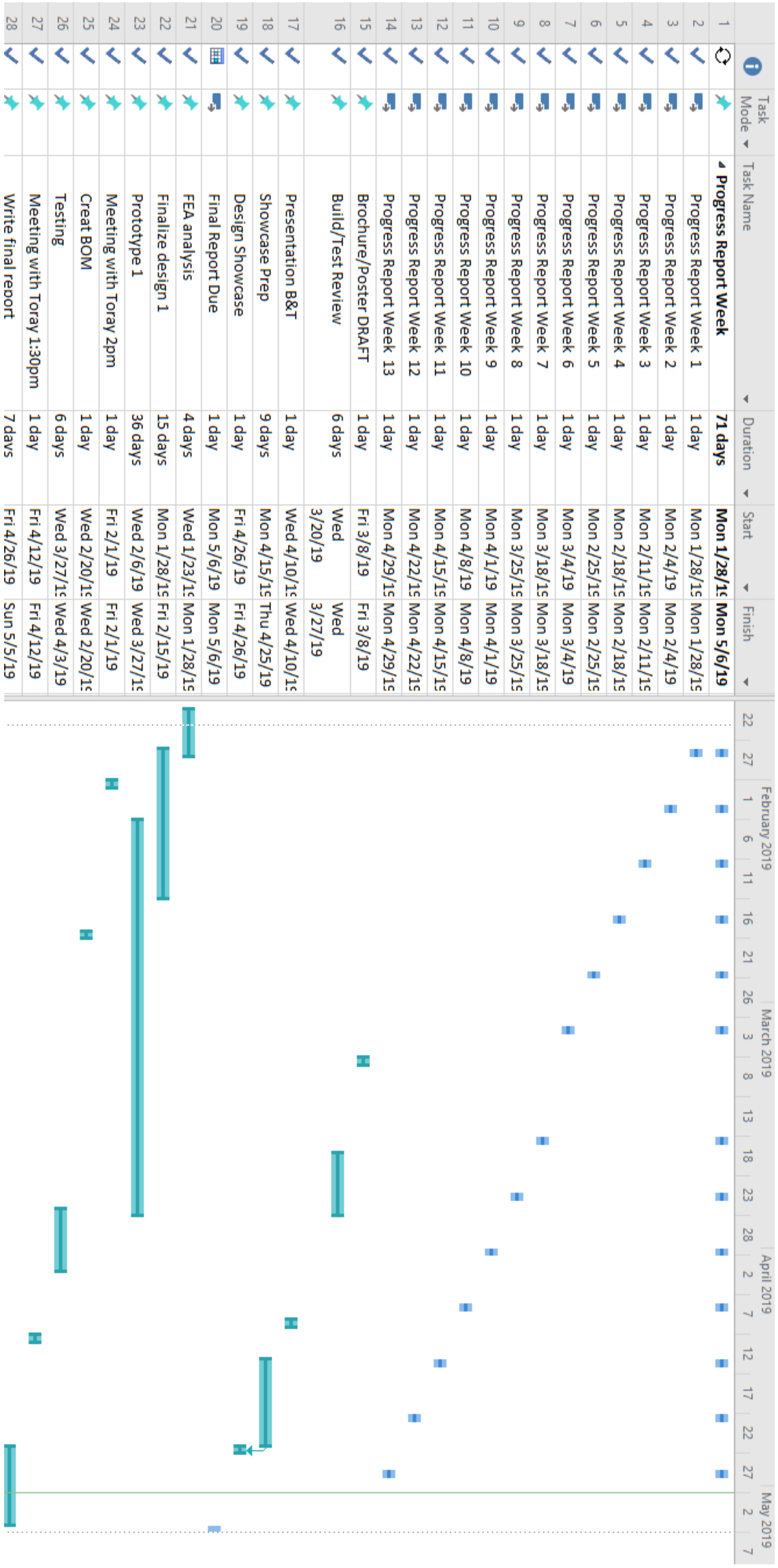


Figure 66: Final Performance Graph

Project Planning

Project management was primarily accomplished through good communication between all group members. We also created a shared google drive which allowed us to access and edit files from any computer, making working from different locations much easier. Delegation of tasks was primarily based on who had the most knowledge and experience with the task, but evenly sharing the workload and demands of each individuals schedule was also considered. Scheduling of tasks was based mainly off the capstone course deadlines, but we also moved some of the deadlines forward to encourage us to stay up to date and avoid work with other courses piling up when deadlines overlapped.



	Task Name	Duration	Start	Finish
1	Progress Report Week	71 days	Mon 1/28/1	Mon 5/6/19
2	Progress Report Week 1	1 day	Mon 1/28/1	Mon 1/28/1
3	Progress Report Week 2	1 day	Mon 2/4/19	Mon 2/4/19
4	Progress Report Week 3	1 day	Mon 2/11/1	Mon 2/11/1
5	Progress Report Week 4	1 day	Mon 2/18/1	Mon 2/18/1
6	Progress Report Week 5	1 day	Mon 2/25/1	Mon 2/25/1
7	Progress Report Week 6	1 day	Mon 3/4/19	Mon 3/4/19
8	Progress Report Week 7	1 day	Mon 3/18/1	Mon 3/18/1
9	Progress Report Week 8	1 day	Mon 3/25/1	Mon 3/25/1
10	Progress Report Week 9	1 day	Mon 4/1/19	Mon 4/1/19
11	Progress Report Week 10	1 day	Mon 4/8/19	Mon 4/8/19
12	Progress Report Week 11	1 day	Mon 4/15/1	Mon 4/15/1
13	Progress Report Week 12	1 day	Mon 4/22/1	Mon 4/22/1
14	Progress Report Week 13	1 day	Mon 4/29/1	Mon 4/29/1
15	Brochure/Poster DRAFT	1 day	Fri 3/8/19	Fri 3/8/19
16	Build/Test Review	6 days	Wed 3/20/19	Wed 3/27/19
17	Presentation B&T	1 day	Wed 4/10/1	Wed 4/10/1
18	Showcase Prep	9 days	Mon 4/15/1	Thu 4/25/19
19	Design Showcase	1 day	Fri 4/26/19	Fri 4/26/19
20	Final Report Due	1 day	Mon 5/6/19	Mon 5/6/19
21	FEA analysis	4 days	Wed 1/23/1	Mon 1/28/1
22	Finalize design 1	15 days	Mon 1/28/1	Fri 2/15/19
23	Prototype 1	36 days	Wed 2/6/19	Wed 3/27/1
24	Meeting with Toray 2pm	1 day	Fri 2/1/19	Fri 2/1/19
25	Creat BOM	1 day	Wed 2/20/1	Wed 2/20/1
26	Testing	6 days	Wed 3/27/1	Wed 4/3/19
27	Meeting with Toray 1:30pm	1 day	Fri 4/12/19	Fri 4/12/19
28	Write final report	7 days	Fri 4/26/19	Sun 5/5/19

Figure 67: Gantt Chart of Project Plan

Operation

This product is designed to last many years. The adjustability of the masses allows for different nip forces to be put on the customer roll. This was requested by Toray engineers, so that they could vary the force parameter based on the production need. The mass is easily adjustable and will have components that can be put in and out of it, based on what is needed in the production line. There is no operation manual for this product, since it is a subsystem to the slitting machine, which has its own operation manual. This product should be completely compatible with the existing pneumatic control system, which calculated the needed nip force based on speed and roll size. The only things that need to be changed in the systems are the constants in the calculations, which can be determined from the equation of the plane in the redesign section of this report. The exact equation and conversions seem to be kept as a trade secret by Toray, so we are leaving this to them.

Maintenance

Overall, the redesigned arm will not need extensive maintenance. The air piston should need little to no maintenance if its properly used. The mass used for creating a moment at the pivot point will not need service or maintenance. However, the engineers at Toray Plastics inquired that that aspect be changed so the level of force supplied at the contact point can vary. If this is the case, then production would have to be temporarily shut down to ensure safe loading and unloading of weight on the horizontal moment arm. Lastly, the arm will be mounted on a track so it can slide horizontally. The contact points of the arm on the track must be lubricated with H2 quality grease often to ensure smooth horizontal translation. This maintenance can be performed without shutting down the production line if safety precautions are taken into consideration (i.e. gloves, safety glasses, hard hat, lubrication product with hose extension).

The previous design that Toray used was created almost entirely of cast iron. This material worked very well and proved to last many years, so the new design will also be made up of cast iron. When the new nip roller mounting arm reaches the end of its life, the cast iron should be melted down and recycled into steel. Steel bolts can also be melted down and reused. The air cylinder is also made of steel, allowing it to be melted down or reused if it is still good at the end of the mounting arm’s lifetime. Hypothetically, at the end of its life, the arm can be disassembled, and every part can be melted down and recycled.

Other Considerations

Toray is a relatively large company in Rhode Island and employs many people in its North Kingston facility. Plastic film industries, selling their product for fractions of cents per foot, run on notoriously tight margins. Increasing productions speeds can help ensure that Toray is able to continue to meet its large contracts from companies like Frito-Lays. Their success has allowed them to expand as well, as shown by the new production line they are currently building, bringing more jobs to Rhode Island.

Because this product is buried deep within a factory it will have very little social or political on the area around that, other than helping Toray continue any impact they already have. Due to the pure design nature of this project and the complete lack of interaction with the public there and no ethical concerns either. The only people that will be consistently interacting with the product will be the operators. To make this design ergonomically for them to install and maintain we focused on making the largest part easy to handle. We limited the size to less than 100lbs a safe amount for two people to lift. We also added lift points to attach a handle or other lifting mechanism, so it could be lifted easily. The other safety concern was material choice. Because the line the nip roller is on makes products that become food packaging using any sort of toxic materials would be unacceptable. The designing of the mass would have been significantly easier if we could use led or another heavy metal, but even coated or contained, it is too risky. The lack of toxic materials also means that this design will be very easy to recycle at the end of its life. All parts can be sold as scrap metal and the material can be reused.

Financial Analysis

When designing this report, the team prepared a financial analysis to give an approximate dollar estimate on how much a project like this would cost. The person hours are the combined value of the time that every person involved in the project put into working directly on the project. The team members spent approximately 500 hours, valued at $20/hour, the graduate students spent approximately 20 hours, valued at $30/hour. The sponsor spent approximately 50 hours, valued at $60/hour. The faculty member spent approximately 20 hours, valued at $100/hour. Additionally, resources added to the cost. The Schneider Electric Machine shop, worth $40/hour, was used for approximately 15 hours between machining parts and assembling the prototype. Computer software used to design the prototype, such as SolidWorks, Microsoft Word, Microsoft PowerPoint, and other software, contributed to 400 hours of the project, and was valued at $5/hour. Additionally, equipment and labs were used for testing the prototype, which was valued at $100/hour for approximately 5 hours of work. The direct cost of the materials for the prototype was approximately $300. Lastly, the overhead cost was approximated to be about 50% of the direct cost, which was about $9,500. When adding all of these parameters, the total cost of the project was approximately $28,500. The total cost of the project can be displayed in a pie chart, as shown by the figure 68.

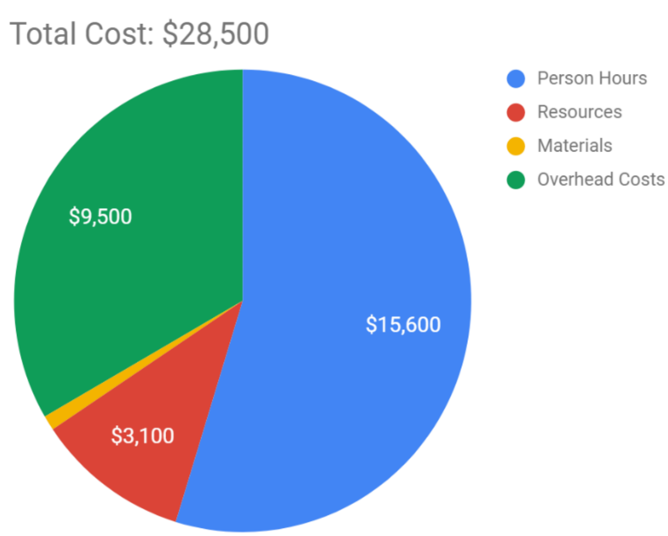


Figure 68: Chart of project costs

Toray Plastics had originally contacted the nip roller mounting arm manufacturing company when they needed a design improvement. However, as stated in the Competitive Analysis section of this report, the manufacturing company had closed due to a low demand for the part; also, the company that regularly services the part refused to redesign the arm. These issues arose because the market for this part is very low. Therefore, the market value of this part has not been established. However, with a roller speed increase from 2,200 feet/minute to 2,700 feet/minute, production would increase at the Kingston manufacturing plant. Toray Plastics has a high demand for its products, so this would certainly increase the profit margin of the company.

The lifetime of this product is expected to be 20+ years. Once the lifetime of this part has expired, there may be another, more technologically advanced solution to the nip roller mounting problem that Toray has. However, over the course of those 20+ years, it is expected that this product will bring in increased revenue for the company.

The engineers at Toray attempted to solve the high-speed roller vibration issues by coupling two mounting arms instead of one on each side of the nip roller. With the new design, only one arm will be used on each side. This will allow for the cost of production of the nip roller mounting arms to be 50% less, considering the company would only need half as many mounting arms for each roller.

Conclusion

The utilization of torque to increase nip force is mechanically simple, yet highly effective. Toray requested that the new mounting arm design allows for an increase in roller speeds by 500 ft/min while using an air piston that will supply sufficient force without exceeding 80 psi. In order to increase roller speeds from a current 2,200 ft/min to 2,700 ft/min, the nip force required must be at least 750 lbf per arm. With a 40 lb weight and an air pressure of 60 psi, the theoretical calculated nip force is about 798 lbf per arm. This surpasses the target force of 750 lbs without even approaching maximum piston air pressure. Nonetheless, utilizing torque to greatly increase the nip roll contact force was successful.

When Team 2 originally approached this project, we came up with various intricate solutions, but we realized that there was a substantial constraint that was initially overlooked: physical space. It was impossible to create a dampening spring system simply because the space it had to occupy was too compact. Therefore, we began to think more simply about how to solve this issue and we formulated the idea of using torque to increase nip force. Once engineering analysis was performed, it was clear that this solution was the best fit. The prototype testing fully clarified that this solution would successfully produce the required target force.

This project focuses on a very unique problem brought to us by Toray Plastics. Although we hope that the company uses this design for many years to come, it is doubtful that other companies would be interested in this product. Therefore, hopes of commercializing this product would be shortsighted. Nonetheless, this design could be very valuable to Toray Plastics. If the company proceeds with physically constructing our design, it could prove to be a very lucrative addition to the production line at the Kingston manufacturing plant.

References

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Appendices

Matlab Code Used to Produce Graphs in this Report

```
%prototype dims
l=7.5;
r=2;
h=7.625;

%theoretical force graph
figure(1)
p=[0:15:60];
w=[0:10:50];
[P,W]=meshgrid(p,w);
Fn = ((W.*l+P.*r.*3.1)./(h))-9.12;
surf(P,W,Fn)
colorbar
axis([0 60 0 50 0 100])
title('Theoretical Nip Force (lbs)')
xlabel('Pressure (psi)')
ylabel('Weight (lbs)')
zlabel('Nip force (lbs)')

%Experimental data
Ne=[-9.21 1.7 11.6 21.8 31.7 42;3.15 14.05 23.3 33.2 42.9 53.3;15.35 26.75 35.9 45.8
55.3 67.2;27.75 39.95 48.4 58.6 68.5 79.3; 41.05 52.1 61 71.6 81.5 92.3];
Ne=Ne';

%experiment plot
figure(2)
p=[0:15:60];
w=[0:10:50];
[P,W]=meshgrid(p,w);
surf(P,W,Ne)
colorbar
axis([0 60 0 50 0 100])
title('Measured Nip Force (lbs)')
xlabel('Pressure (psi)')
ylabel('Weight (lbs)')
zlabel('Force on scale (lbs)')
```

```
%Error Calc
Er=abs(Ne-Fn);

%3D Error Map
figure(3)
p=[0:15:60];
w=[0:10:50];
[P,W]=meshgrid(p,w);
surf(P,W,Er)
colorbar
axis([0 60 0 50 0 2])
title('Error')
xlabel('Presure (psi)')
ylabel('Weight (lbs)')

%2D Error Map
figure(4)
contour(P,W,Er,[0:.5:3.5],'ShowText','on','LineWidth',2);
grid on
axis([0 60 0 50])
xlabel('Pressure (psi)')
ylabel('Weight(lbs)')
title('Error (lbs)')

%max error
maxer = max(max(Er))

%full scale dims from final product
maxw=50;
maxp=80;
lful=7.82;
hful=10.57;
rful=5;

%Full Scale Graph and Max
figure(5)
pful=[0:5:maxp];
wful=[0:5:maxw];
[Pful,Wful]=meshgrid(pful,wful);
Fnip = (Wful.*lful+Pful.*29.22.*rful)./(hful);
```

```
surf(Pful,Wful,Fnip)
colorbar
MAX = max(max(Fnip))
axis([0 maxp 0 maxw 0 (MAX+50)])
title('Theoretical Nip Force for Full Scale (lbs)')
xlabel('Pressure (psi)')
ylabel('Weight (lbs)')
zlabel('Nip Force (lbs)')
MAX = max(max(Fnip))
```