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Optimization and Redesign of a Dual Rotary Cutter

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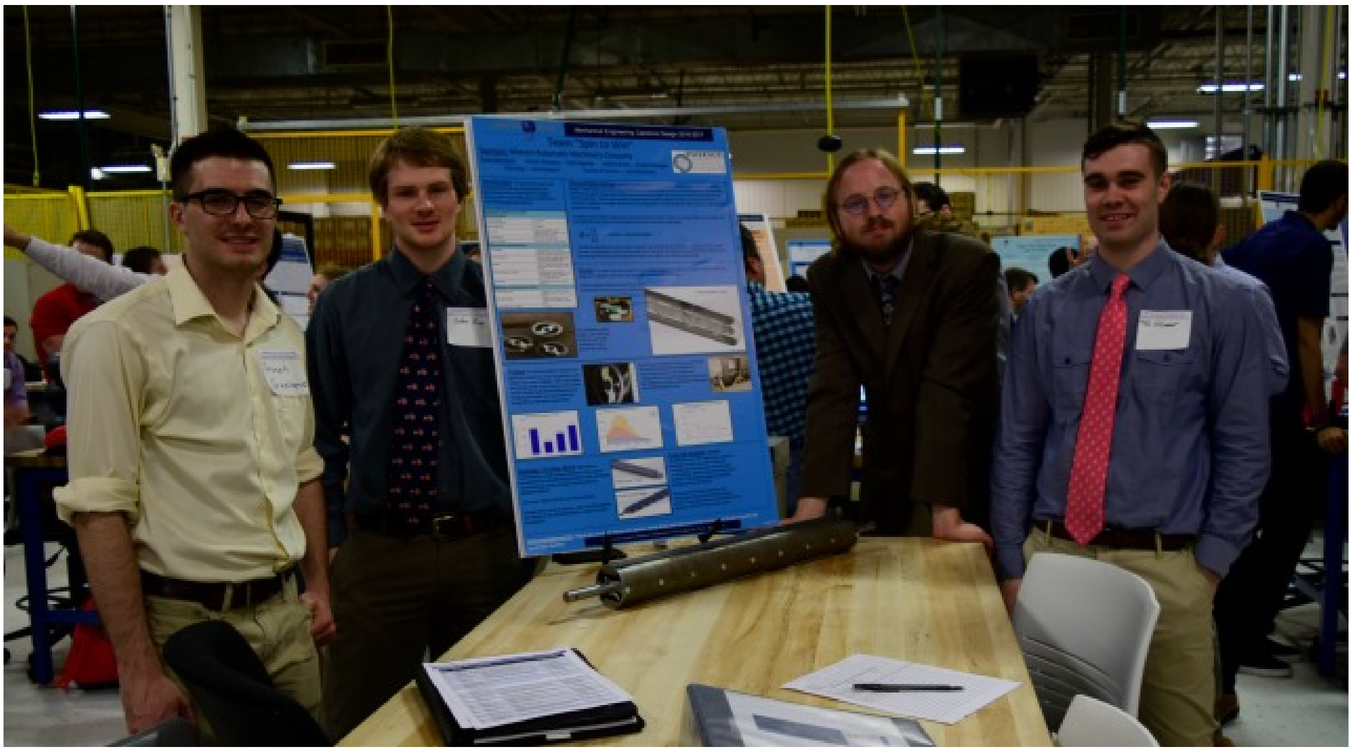

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Sponsor - Maxson Automatic Machinery Company

Team 24: Spin to Win

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Abstract

This design report records the actions taken by the Spin to Win team to optimize and redesign a cutter assembly in a dual rotary sheeter, as well as design a gear reduction in the same assembly. The optimization and redesign of the cutter system focuses on two main factors, inertia and torsional stiffness. The sponsor for this project is MAXSON Automatic Machinery.

Wear and tear on a machine and its parts could be costly or even jeopardize the integrity of the system. Spin to Win came up with a redesign that can theoretically decrease the inertia by up to 40% and grant an increase of the torsional stiffness of the cutter assembly. The gear reduction will change the motor that can be used to propel the system, including a 1.8 : 1 reduction allowing for a smaller motor, thus lowering the cost of the entire assembly. By lowering the inertia and increasing the stiffness of the cutter system MAXSON could see longer part life and less wear as well as lower overall system costs.

The optimization and redesign solution incorporates a wheel and spoke design to add torsional stiffness and decrease inertia. The spokes are inserted along the cutter on an inner shaft and are connected to the outer shell to give the design the added stiffness in the direction of rotation. A prototype was created to show the ease of assembly of the redesign and the apparent decrease in inertia from the current cutter design. The prototype has design considerations to accommodate some of the existing hardware, so that the system integrates into the sheeter system seamlessly.

The design created by Spin to Win suffered in terms of manufacturability, and a simplified design was created for testing. The tests themselves encountered unexpected difficulties owing to the design of the tests for the available equipment. In the end, the viability of the design was not definitively proved but was suggested, paving the way for further work.

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1 Introduction

MAXSON Automatic Machinery proposed a problem for this years Capstone that dealt with their main product, which is a sheeter for cutting paper, plastics, and other thin, rolled material. The original proposed problem was for a gear reduction to the cutter assembly that would in turn allow MAXSON to purchase and use a smaller, cheaper motor. However, upon meeting with the sponsor the idea of an inertial reduction for the cutters was proposed as well as a gear reduction.

The Spin to Win team decided it was more beneficial to attack the inertial reduction problem first rather than the gear reduction because the work done for the inertial reduction would be more beneficial to the requirements and concerns of this Capstone course. MAXSON's existing design for the sheeter cutting section utilizes large rotating blades to make the cuts on material being fed through the system. These cutters are constantly accelerating and decelerating to maintain cut length as well as keep up with the feed speed. These large changes in speed result in large changes in energy, so that a powerful motor is required. In addition, the cutting blades need to be adequately stiff to prevent deflection and preserve a good cut quality. With that said the problem definition now became to redesign some of the sheeter's cutting blade assembly and optimize them to have a reduced inertia while keeping an adequate level of stiffness.

Now that the problem was defined the team could now come up with ideas for a redesign as well as design considerations, that each idea would have to meet. The design had to be modular, so that it could be assembled in an efficient manner, while it decreased cost or at least kept it in the ballpark of the current machining and materials costs. The design needed to have material cut away to lessen the inertia, but it could not be at a cost of the stiffness. Because of that the design needed something that could allow the system to keep the stiffness or strengthen it. The design also had to be able to fit some of the existing hardware. A slotted insert that would go the length of the entire shaft would help to alleviate some of those issues.

2 Project Planning

2.1 Project summary

Spin to Win's activities were grouped around two keystone moments in the Capstone course: the Critical Design Review presentation, in which the group's brainstorming was brought to near completion, and the Proof of Concept presentation, in which the group's selected design would be fabricated and analyzed in depth.

In preparation for the first milestone, there were two main activities: project definition and brainstorming.

During the project definition phase, Spin to Win familiarized itself with rotary cutters, with rotational dynamics in general, with gears in general, and with all information that could be gathered to assist in orienting the team. Patents were consulted, a competitor's machine was studied together with MAXSON's, and texts were gathered such as *Shigley's Mechanical Engineering Design*.

During the brainstorming phase, each team member generated 30 unique concepts they could envision as part of a successful solution to the problem. This was a 2-3 week process, as the number of concepts was obviously large. The intent was not to preserve every one but to cut out those pieces that were useful for crafting into three critical concepts.

These concepts were presented in the Critical Design Review presentation, and then the group moved into the second main phase: proving. During this time Tom and Joe worked on fabricating a 3D-printed model of the selected design while Nathan and Josh worked on developing equations and methods for proving that the design was sufficiently stiff, strong, light, and manufacturable. The results of this work were presented at the Proof of Concept presentation.

Moving forward, Spin to Win will enter its next main phase: fabrication, during which a steel, scaled down model of the proposed design will be fabricated and tested for strains under operational conditions.

A broad, graphical model of the above information can be read in the project timeline, Figure 1.

2.2 Agile project management

A small team of four engineers with only a fraction of their time to devote to the project was required to bring a complete design activity to completion within a span of a few months. In addition to designing and analyzing a solution to the provided design problem, a variety of course-specific objectives needed to be met at points in the plan. In addition, design and analysis requirements in addition to the specific goal of redesigning the cutter were supplied by MAXSON, such as evaluating the mass moment of inertia of the existing system and designing a potential pinion gear reduction between the motor and cutters.

In order to facilitate success careful attention had to be paid to the relationship between available resources, time remaining, and specific deadlines that had to be met. The known exact deadlines together with best estimates of good times for completion of various tasks and analyses of all the requisite subtasks for a larger tasks completion were inputted into a project plan and Gantt chart curated by Microsoft Project.

This plan was updated on a rolling basis as information became available, both in terms of requirements and deadlines and in terms of tasks completed. These frequent and rapid restructurings of the plan in response to shifting requirements and circumstances constitute agile project management.

The broad structure of the plan went through several large shifts at two moments in the process: 1) when a single concept for development was selected out of many and 2) when focus was shifted away from spending time on the gear reduction. At these moments a large amount of future uncertainty was focused into a single line of action to be taken. In the first case, action was immediately taken in the direction of creating a concrete, dimensioned drawing for the concept. In the second, focus was granted to the cutter-based solutions for the Critical Design Review Presentation, lifting a great deal of anxiety and pressure from the team.

Some project restructuring took place in response to developments in MAXSONs electrical team, with whom the mechanical team was intended to work in parallel. MAXSONs preliminary project plan required a great deal of trading of information between two teams. Initially, the mechanical team would provide inertia data from which the electrical team would select a motor, whose dimensions would be supplied to the mechanical team for producing a layout. As of this writing no motor selection has ever been made, and so the mechanical team was forced to reorient its plans and methods for designing a gear, making use of probable assumptions.

In the second semester, significant project restructurings took place around waiting for fabrication of the testing models. This work was outsourced to MAXSON and vendors with which MAXSON was building a relationship. There was an initial large delay in producing the fabrication drawings, and after that there were delays in receiving the fabricated models. In response to these delays, aggressive testing schedules had to be designed and followed.

MAXSONs preliminary project plan is provided in Appendix Subsection 1 for reference.

2.3 Gantt chart

Microsoft Project automatically generates a chronological map of the tasks, deliverables, and milestones of a project with parameters inputted by the user. The map portrays a great deal of information graphically, included the percentages completion of various tasks, prerequisite tasks for others completion, and the involvement of any resources, including engineers. The final Gantt chart for the project is provided in Figure 2.

2.4 Teamwork engineering

One of the largest challenges in the development of this project was the equitable distribution of tasks among the team in a way that would engage each team members strengths. Each team member displayed a particular set of strengths, as well as a particular personality that made them better suited for some activities over others. At the same time all four team members were carrying a heavy course load in other classes that often interfered with the project.

An early mode of task distribution the team employed was an anarchic volunteer model, but this resulted in inequitable loads. In addition, a lack of clarity of who was in a leadership role led to disagreements that developed a great deal more rancor and momentum than was necessary.

Over time, as the team members learned each others strengths and personalities, defined roles developed that greatly increased the groups efficiency. This process crystallized in the days leading up to the Critical Design Review presentation. These roles are reflected in the members titles:

- **Joe**: 3D Modelling Specialist
- **Josh**: Lead Designer
- **Nathan**: Project Manager
- **Tom**: Mechanical Systems Specialist

Tasks that suited team members desires and aptitudes were afterward distributed to those team members, and Nathan took on a leadership role. Joshs ideas, Toms engineering vision, Joes support skills, and Nathans mathematical bent were each drawn out in their own spheres.

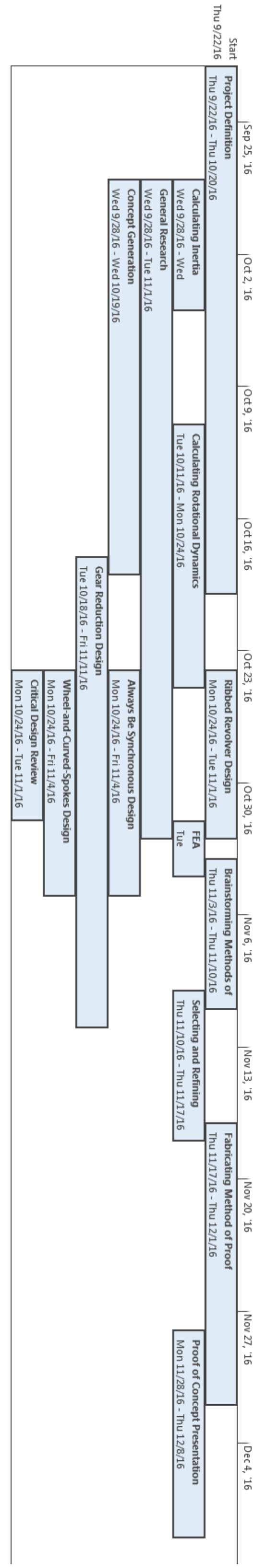
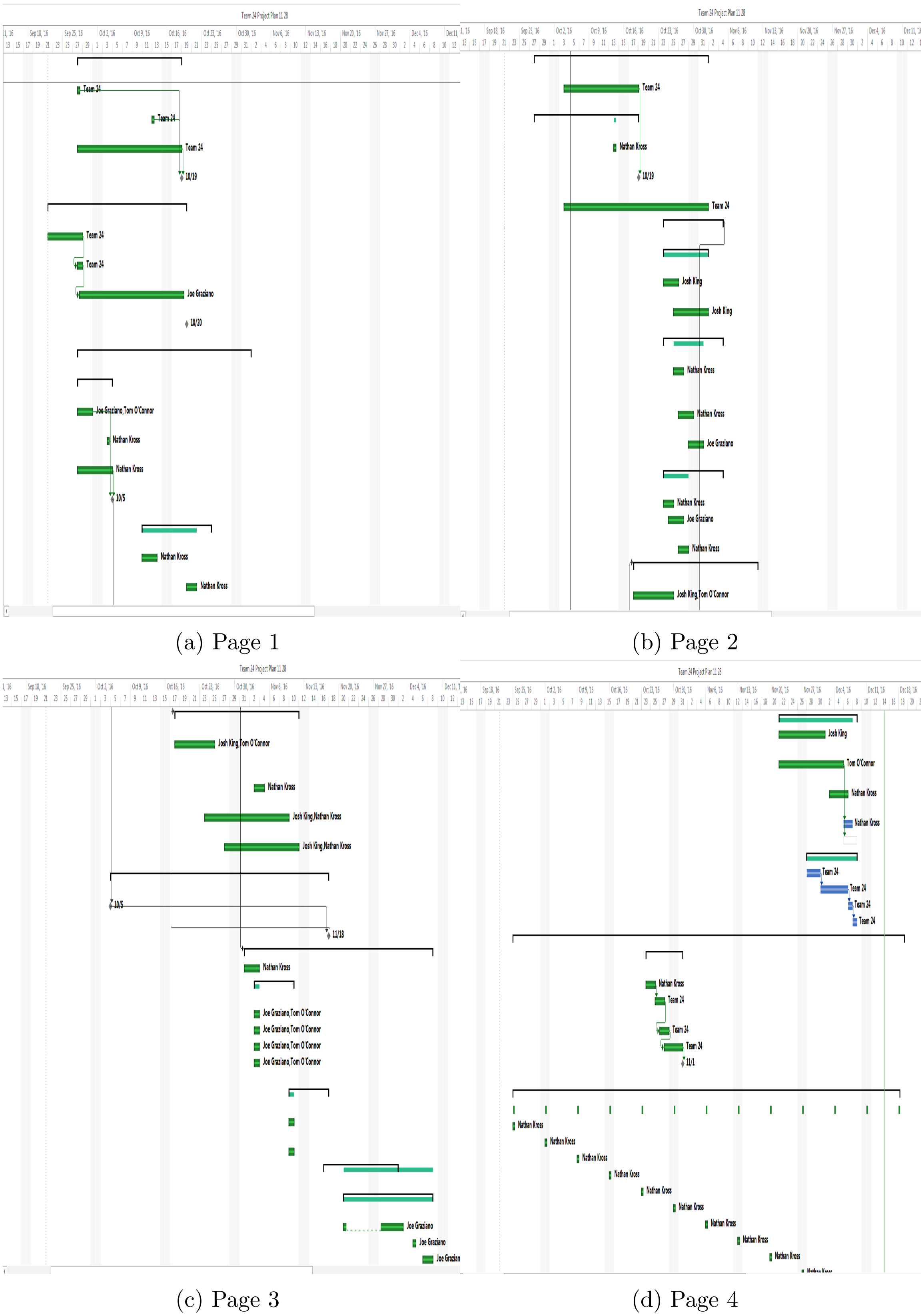


Figure 1: Spin to Win project timeline



(a) Page 1 (b) Page 2

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Figure 2: Spin to Win Fall Gantt chart

3 Financial Analysis

3.1 Project finances

The Gantt chart records the hours spent by each team member on a variety of tasks. Supply an estimated entry-level engineering salary of $ 32 per hour for each team member, and the labor cost for all activities undertaken in the project to this point is given in Figure 3. However, the combined labor cost of $26,261 is a purely imaginary number.

The project plan calls for fabricating a 3:1 scale model of the spokes assembly together with a model of the existing cutter, affixing both with wireless strain gages, and subjecting them to forces and rotational speeds and measuring the resulting deflections. The projected costs for these fabrications and tests are tabulated in Figure 4. The expected total is $1,803, including an estimate for a wireless strain gage system for which quotes are being gathered.

Meanwhile, Spin to Win has a budget of $ 3,000 provided through the University of Rhode Island and through MAXSONs sponsorship. No costs were incurred (except the imaginary costs of student labor) during the Fall semester. As such, the remaining budget projected into the Spring semester to cover unforeseen costs is $1,197.

This remaining budget could be used to finance replacements or corrections if errors or unforeseen difficulties arise during fabrication. In addition, if finances allow, Spin to Win could use the remaining budget to move into investigating modifications to the designed system in shape or material.

3.2 Spokes assembly profitability analysis

The Bill of Materials for the spokes assembly is provided below in Figure 27. It requires a material cost of $2,420. In addition, the time and cost required to machine each component of the spokes assembly has been calculated based on machining speeds and costs known at MAXSONs facility (or estimated where not available). The step-by-step process that leads from raw material to completed part, and the time required for each operation, is included in the calculation. However, times for rework, set-up, and re-tooling are not included, so these calculations are understood as a minimum. The results of this calculation are found in Table 1.

It can be seen that the spokes assembly adds a large amount of time and a moderate amount of cost ($3,385) to the total cutting section compared to MAXSONs current cutter. However, there are three financial tradeoffs at play: 1) the use of faster speeds, adding quality to the cutting machine, 2) the omission of operator-sides gear, cutting $5,000 off the price, and 3) the use of a smaller motor, saving as much as $6,000 in conjunction with other approaches.

Wheel and Spokes	
Cost of Materials	$2,420
Total Time Required	39.47 hours
Cost with Machining	$10,306
Current Bottom Cutter	
Cost of Materials	$2,925
Total Time Required	20 hours
Cost with Machining	$6,921
Added cost	$3,385
Added time	19.47 hours

Table 1: Spokes Assembly Cost Analysis

MAXSON currently stands as an industry leader in terms of the flexibility of its system and the speeds it can reach. Competitors, especially foreign competitors, are able to deliver a similar product at lower cost, but MAXSON outperforms them in quality. One customer for rotary cutters is the plastics industry, which would use a cutter to snip material after it leaves from extrusion. In the current environment, the operating speed of the cutter is the limiter on this process, as extrusion speeds for plastic are already able to exceed 1,000 feet per minute. If the operating speed of the cutter could be increased, overall production speeds in plastics manufacturing could increase in exactly the same ratio, adding a huge amount of value to MAXSON's product[1]. A lower-inertia, stiffer cutter is a critical building block in a faster system.

MAXSON's current cutter uses two sets of gears: one pair at the motor-side, to receive and transmit motor power, and another pair at the opposite end to reduce the overall torsional deflections the cutters experience. This second pair of motors changes the boundary conditions of the beams under torsion, reducing their effective length by half. They are in the system after testing determined that torsional deflections became unacceptable at high speeds. If torsional deflections could be reduced by the same or a similar amount as these gears work to do, they could be eliminated from the system. The cost of the operator side gears is estimated at $5,000, roughly $1,600 more than the added cost of the spokes assembly.

The primary objective for Spin to Win was to lower the power requirement of the cutter section, as motor prices tend to increase in the squared ratio of horsepower. Cutting the power requirement in half may cut the motor cost to a fourth. The current motor and drive MAXSON employs costs $ 17,000, and it is specialized to drive the current system, with an inertia of about 14,400 lb-in^2. The inertia discount with the spokes assembly is about 3,000 lb-in^2, which lowers overall system torques by 20%. Together with a gear reduction, the new motor and drive price may fall to as low as $11,000, although the actual motor selection is still underway between MAXSON and the electrical team.

[1] This information comes from conversations with Spin to Win's project sponsor, MAXSON Automatic Machinery.

Combining the spokes assembly and the 1.8 pinion reduction also designed by Spin to Win (whose cost can be estimated around $1,500 based on the costs of the existing gears), MAXSON's MD Sheeter may achieve a simultaneous reduction in cost of $1,500 to $6,500 (depending on whether the operator-side gears are removed) and increase in quality and operating speeds.

3.3 aPriori Assisted Costing

In order to obtain corroborating evidence for its cost estimates, Spin to Win employed an automated cost analysis software called aPriori. This software imports the geometry of a part from a CAD file and attempts to generate a routing that would produce that part from known stock within the context of a given set of processes (sheet metal, machining, etc.).

The costings of the individual components of the wheel and spokes assembly are provided in the Appendix. A summary of the results from aPriori is given in Table 2.

The "fully burdened cost" column includes the price of the material (minus the price of the scrap left over from the stock), the costs of labor and overhead for operating any machines involved in fabricating the component, and the amortized setup costs associated with producing 10 of these assemblies per year. The cycle time is the total machine operational time associated with producing each component.

Component	**Fully Burdened Cost (dollars)**	**Cycle time (hours)**
Blade Slug	$ 4,554	28.32
Deadweight Slug	$ 2,887	17.70
Inner Shaft	$ 256.96	0.16
Outer Skin	$ 760.68	2.84
Spokes	$ 569.43	7.30
Total	$ 9,028.07	56.32
Manual estimates	$ 10,306	39.47

Table 2: Spokes Assembly aPriori Calculated Costs

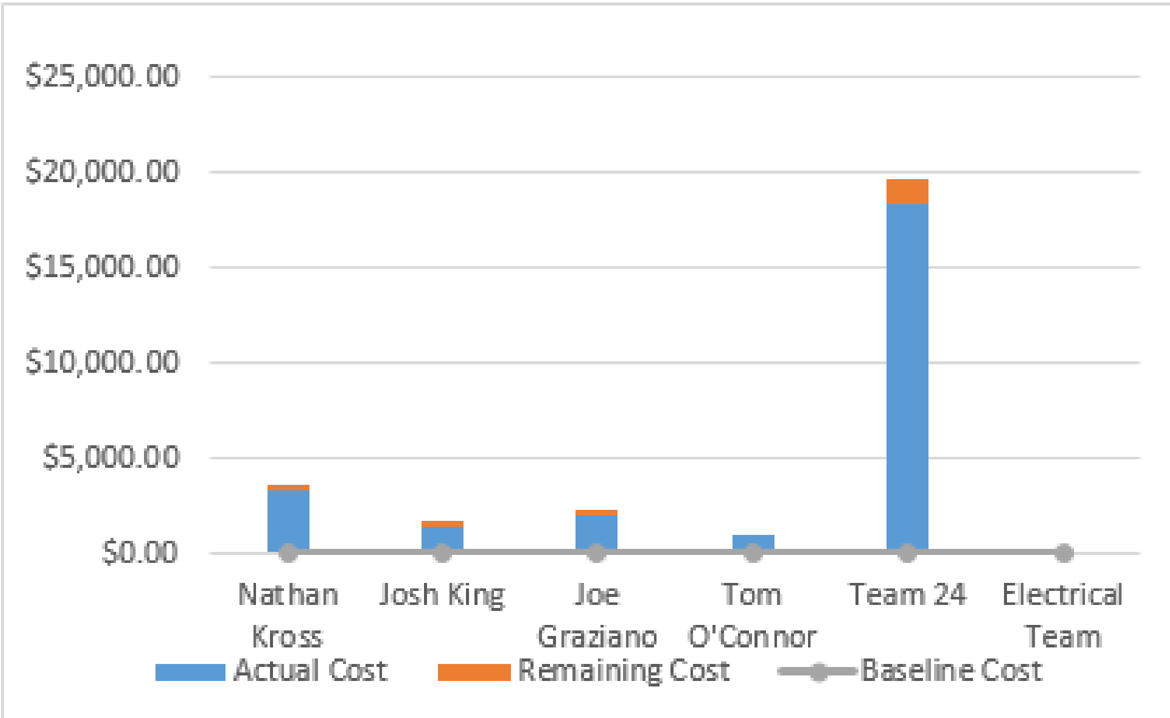
The manual estimates generated by Spin to Win are placed side-by-side for comparison to show that the two methods have a high degree of intersection. The largest discrepancy is in cycle time, and this discrepancy can be accounted for by the curved slots found in the blade slug. These slots not only follow the 67" length of the slug, but they also curve to follow the helix angle, and the shape of any surface inside the slot is as a plane that has been twisted.

In the manual model, these slots were represented as rectilinear. In reality their curvature makes the process of fabricating them prohibitively slow and expensive, which was discovered to be a critical problem with the wheel and spokes assembly.

RESOURCE COST OVERVIEW

COST STATUS

Cost status for work resources.



COST DETAILS

Cost details for all work resources.

Name	Actual Work	Actual Cost	Standard Rate
Nathan Kross	106 hrs	$3,392.00	$32.00/hr
Josh King	46 hrs	$1,472.00	$32.00/hr
Joe Graziano	64.67 hrs	$2,069.33	$32.00/hr
Tom O'Connor	32 hrs	$1,024.00	$32.00/hr
Team 24	143 hrs	$18,304.00	$128.00/hr
Electrical Team	0 hrs	$0.00	$0.00/hr

Figure 3: Spin to Win Labor Cost Overview

Bill of Materials	*Spin to Win Test Rig*					
	Description	Dimensions	Material	Supplier	SKU	Price($)
MAXSON Model	Steel Rod	3' by 3" diameter	1018 Carbon Steel	McMaster Carr	8920K84	157.38
Spokes Assembly	Steel Rod	3' by 1.125" diameter	1018 Carbon Steel	McMaster Carr	8920K251	31.78
	Steel Tube	3' by 3.5" OD, 2.5" ID	1005 Carbon Steel	McMaster Carr	7767T93	202.52
	Steel Plate	1.75" by 2.5" by 6"	1018	McMaster Carr	8910K97	35.26
	Hex Head Bolts	5/8" long, 1/4-28 threads	Grade 5 Steel	McMaster Carr	92865A004	9.4
	Nuts	1/4-28	Grade 5 Steel	McMaster Carr	95505A611	3.75
Test Equipment	High-Accuracy Noncontact Tachometer	N/A	N/A	McMaster Carr	11765T58	162.96
	Wireless Strain Gage System	N/A	N/A	Phase IV Engineering	N/A	1,200
Total						**1803.05**

Figure 4: Bill of Materials for Spring Test Rig

4 Patent Searches

One of the earliest steps in the design project was to start with a patent search. By searching through listed patents it was possible for the design team to compare some of the design ideas generated with existing designs. In addition the patent search would allow the design team to ensure no patents were being infringed upon. The patents found also helped give the design team ideas that may be able to improved upon by redesigning it and improving upon existing designs.

The patent searching and research was split up evenly among the four members as to prevent the group members from overlapping research. This lead to some patent searches being less useful than others as they were too broad or too narrow. However during the patent searches it became clear that most of the patents found were not the most relevant to the cutter redesign. Most of the patents found were not from rotary cutting machines, but were from similar machinery that would be relevant to the design process. Even though some of the patents came from other machine or devices it would still be useful to the design team as the team was focused on improving a smaller section of the cutter machine.

Patent Number: 9,415,524

Patent Name: Rotary Cutter Device

Patent Owner: BROTHER KOGYO KABUSHIKI KAISHA

Date Filed: March 13,2013

Patent Description: The patent above invented by Nagura, Masato, Takeuchi, and Takashi is for a rotary paper cutter. While the general concept of the design is simliar to MAXSON's design, it is much more compact. This patent also features a rotating blade to cut paper, but the general blade and rotary section are much different than MAXSON's design. The patent also features a much smaller device overall while MAXSON's is a much larger machine. Some of the useful points that can be drawn from this device are how the blade is oriented and how the paper feeds through.

This patent also uses a similar method to MAXSON's where both of the rotating blades meet in the middle to cut the paper. One major difference is that this rotary cutter features and angled blade, while MAXSON's features a helical blade design. Both of these features are designed to help create a straight cut on the paper, as it travels through the machine. While it is a different method of achieving a straight cut it was something that was looked into while attempting to redesign MAXSON's cutter assembly. However as mentioned before MAXSON's simple helical cutting blade would accomplish the straight cuts, without the need for angling the blade and ultimately, the helical blade was kept as MAXSON designed it, instead of opting for an angled blade.

The following two figures below show the completed cutter assembly and the internal components that would be cutting the paper as its fed though the system. [1]

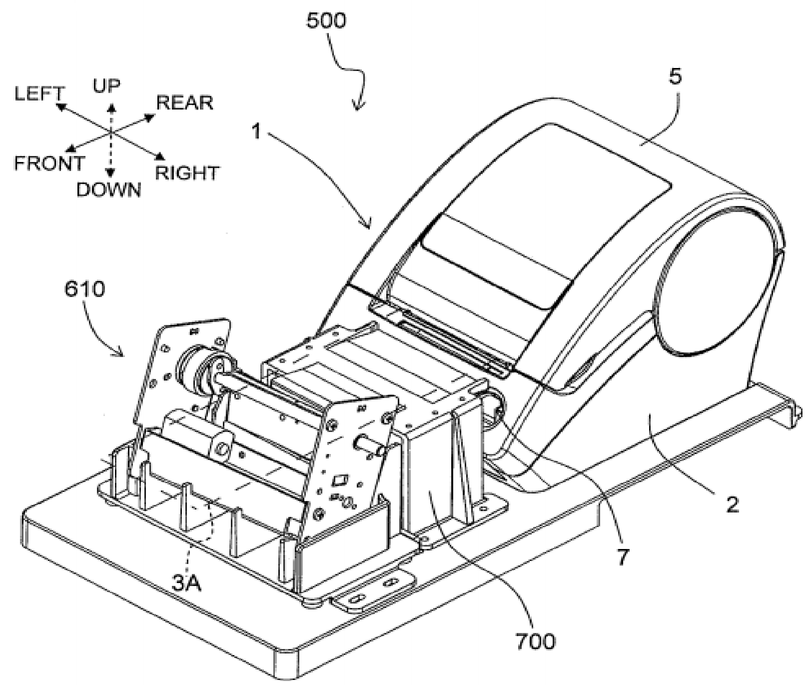


Figure 5: Patent No. 9,415,524 Full paper cutter assembly

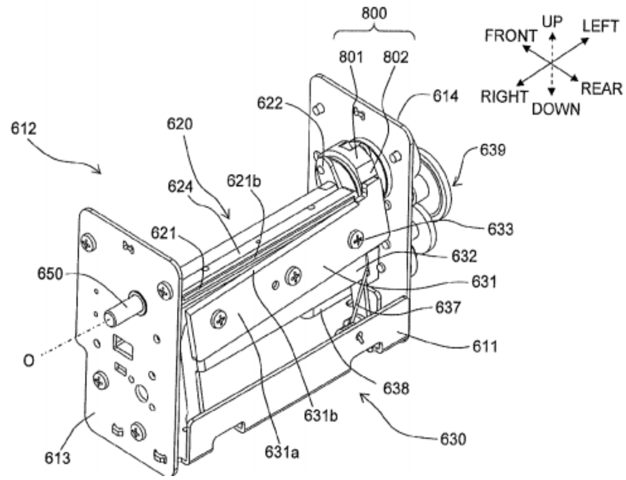


Figure 6: Patent No. 9,415,524 Internal cutting blade components

Patent Number: 7,249,547

Patent Name: Paper cutter

Patent Owner: Carl Manufacturing Co., Ltd.

Date Filed: August 16, 2004

Patent Description: This patent features a paper cutter which has a different cutting method than the rotary blades that MAXSON currently uses. This patent features a cutting method of intersecting angled blades which close down onto the paper. This method of cutting the paper results in very precise and quick cuts through a vertical movement. One of the earlier attempts Spin to Win made at solving the problem was to change the way that MAXSON cut the paper. Using a method similar to this patent was very briefly thought of but was ultimately discarded, as it was more economic, and more feasible to go with the current rotary cutting blades.

One point that could be taken away from this paper cutter is that when the paper is cut it results in one clean cut, not a "rolling" cut like the current system MAXSON uses. If the team were to design a new cutter blade that featured a non-helical cutting blade it would be similar to this patent. However when using straight blades there would be opportunities for the paper to jam and the feed speed would need to exactly match the paper. While the current system still has to match the feed speed of the paper, the angled blades result in a cleaner cut so that portion of the design will likely remain unchanged. [2]

Patent Number: 5,001,952

Patent Name: Rotary Cutter

Patent Owner: Kanzaki Paper Manufacturing co.,Ltd (Tokyo,JP)

Date Filed: November 28,1989

Patent Description: This patent is used by Kanzaki Paper mMnufacturing which is a paper manufacturer and supplier. While the patent they have is used on a much smaller scale than MAXSON, the general design of the rotary cutter is very similar. This patent uses one rotary blade to cut paper to a desired length as it is fed through the system.

The main points to take from this patent would be the helical or curved cutting blade which is similar to the current design. This curved blade helps ensure a clean straight cut while the paper is being fed through the system. Since only one blade is rotating in the patent design, and two blades spin in MAXSON's design they do differ somewhat. The general idea of cutting paper through a rotating blade is something that is being used in the final proof of concept and something that MAXSON had originally designed and showed us so it will carry over to the final design, as it is a core component of the current and redesigned system. [3]

The figure below shows the current patent holders design of the rotary cutter. The figure has an angled line on it which is the the angle of the helical blade that would help aid in the cut being straight while the blade is rotating and the paper is being fed under the blade.

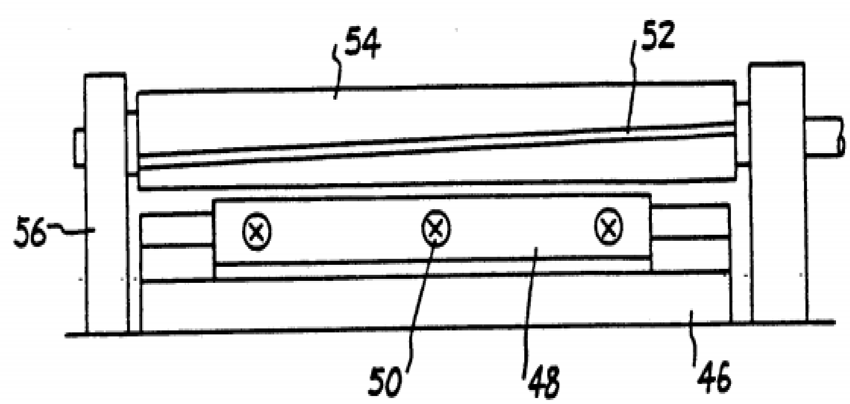


Figure 7: Patent No. 5,001,952 Rotary cutter blade

5 Evaluation of the Competition

5.1 Direct analysis of competitor's machine

After the design team had decided on a clear view of the task at hand, it was important to look into the competition. By looking at the competition and competitors solutions to the engineering problem it would help the design team move forward with what would be the best possible solution to the problem, and allow the team to gauge success against know solutions.

This design project however differed slightly from the other projects in the Capstone course. This was due to the fact that while the design team was presented with an engineering problem that must be solved, that problem had a rather narrow field of approach. The goal of redesigning the cutter blade meant that much of the existing machinery would be kept the same as was requested by MAXSON. This meant that the design team would be focused, for the most part, on the cutting blades portion of the machine. While this is a small portion of MAXSON's assembly it does have a large amount of work done to attempt to improve it. Basically, the design team was only concerned with a few key components of competitors designs, those being the motor and rotary cutting system.

The first day the design team met with MAXSON the team was actually allowed to examine one of MAXSON's competitor's sheeter machines. The differences between the two systems were immediately made apparent. Most of the design differences came from a different motor system and a different rotary blade design. The first step in competitive analysis was to compare MAXSON's system to the competitor's system that was presented to the design team, and to try and figure out why competitors made the design choices that they did.

Comparing the motor setup of competitors to MAXSON's current setup, the competitors opted for a liquid-cooled motor to power their rotary blades. Now the reason for doing so was to increase the effective cooling on the motor for the rotary blades. However, when sizing a motor for the system MAXSON had indicated that it was hoping to not use a liquid-cooled motor. This is due to the increased cost of a liquid-cooled motor as the motor and cooling system are more expensive, but also because the liquid-cooled motors would be louder. MAXSON values how quietly its machine operates in comparison to the competition. In addition, if a liquid-cooled motor were chosen it would lead to increased maintenance as the coolant should be changed in most cooling systems, and if there ever is a leak in a liquid-cooled system it can be disastrous for the motor and surrounding systems. For those reasons it was decided that an air-cooled motor should be used for the redesigned system, either the current motor that is being used or a smaller one if the applied gear reduction and inertia reductions allowed for it.

Next Spin to Win compared the cutting blades that the competition designed related to MAXSON's design and try to figure which one was a more feasible design. The competitors rotary blades were heavily machined and manufactured in comparison to MAXSON's. The most notable difference was that MAXSON opted for a single machined piece, where as the competition opted for a complex assembly to construct the rotary blade. While MAXSON's design is simple and elegant, it required much more planning ahead of time compared to the competitors, where it appeared the competition solved many issues as an afterthought. The competitor's solution, while having a lower inertia, required significant machining, and as MAXSON informed us that added machining is something they would like to avoid, as every added hour of machine time would result in higher cost to manufacture and lead to a more expensive machine overall. One of MAXSON's main strengths in its design is that since it is simple it can be relatively cheaper to produce without sacrificing quality.

Comparing MAXSON's bottom blade to the competition was interesting. The competition's heavily machined bottom piece had a larger inertia than MAXSON's one-piece design and also required more machine time. For this reason it appeared MAXSON already had a favorable solution compared to their competitors, but it should be noted that if the competition had optimized for inertia more it could have a more competitive solution. The competition could have opted for a non-solid tube and then applied the plastic pieces to the outside to extend the effective radius of the cutting tube while saving inertia. Also the amount of heavy machining and added mass to the outside of the bottom blade produced by the competition appeared a bit unprofessional, and inefficient. MAXSON's bottom blade was fairly well balanced to begin with and looked like a much more professional design, as it was smoother and more well designed from the start. In addition the competition needed extra inserts on the outside to prevent paper feed jams, where MAXSON had addressed this issue in their initial design and not as an after thought. Overall this lead the design team to lean towards a cleaner and more elegant solution than simply worrying about balance and smooth feed afterwards.

5.2 Market research

Now looking at other competition it is somewhat hard to find information that is particularly useful to the team as most of the information that is useful to the team is going to be in the technical drawings of competitors' machines, which normally are not released to the public. In particular the useful components that would like to be examined are the motor and cutting blades, and as those are internal components they aren't readily available. However, there are some useful figures available from competitors. The main manufacturers that were found to be similar in design and use to MAXSON were from the manufacturers, ZHIVE, SHM, and KINGSUN. Now even though these manufacturers make sheeters similar to MAXSON they tend to be smaller scale, but this information is still useful as it can be useful to see what motors are being used and the general set up of the sheeter machine

The first competitor mentioned in ZHIVE, which produces the ZWC-1400-2 and ZWC-1700-2 paper sheeter machines. The products produced by this company are mainly used for craft paper and paper board while MAXSON's are used for many more types of paper. In addition these machines cost roughly \$62,000 - \$80,000. Also it should be noted that while this machine is listed as having a cutting speed of 300 meters or 948 feet per minute MAXSON would like the cutter system to operate at 1000 feet per minute or greater. IN addition the tolerances MAXSON would like in terms of cut quality are tighter and more restrictive than the ZHIVE product. Overall this machine is close to what MAXSON would like, but is lacking in a few features, such as cut quality and feed speed. These machines' product specifications are listed in Figure 8, and an image of the sheeter machine is shown in Figure 9 [4].

Model	ZWC-1400-2	ZWC-1700-2
Reference weight of cutting paper	50-500g/m2	50-500g/m2
Max.Diameter of paper	1650mm(65')Max	1650mm(65')Max
Max.Pieces of paper	1400mm(55')Max	1700mm(67')Max
Total power	25KW	28KW
Cutting length	400-1600mm	400-1600mm
Cutting accuracy	+/-0.4mm	+/-0.4mm
Maximum cutting speed	300cuts/min	300cuts/min
Maximum cutting meter speed	300m/min	300m/min
Atmospheric pressure	0.8MPa	0.8MPa
Power supply	AC380V/220Vx50HZ	AC380V/220Vx50HZ
Total weight	11000kg	13000kg

Figure 8: ZHIVE listed product specifications

Figure 9: ZHIVE Sheeter

The next competitor which would be SHM manufacturing. The SHM manufacturing produce a similar sheeter to MAXSON's. SHM produces the 1450 model sheeter, and it is marketed towards the same market that MAXSON caters to. It is an industrial style sheeter which processes a large amount of raw paper feed and turns it into smaller cut sheets similar to MAXSONS. The main thing that is interesting with this product is that it handles a much larger range of materials than the other competitions' sheeters. Once again this machine is listed at a similar feed speed to MAXSON's desired feed speed of 1000 feet per minute, however it falls short at 948 feet per minute. Once again it also has some shortcoming in terms of the general design and setup as appears to take up much more room than MAXSON's design. However this model does look flexible in its configuration. Figures 10 and 11 feature an image of the sheeter produced by SHM and some technical details that were used to compare SHM to MAXSON's design. [5]

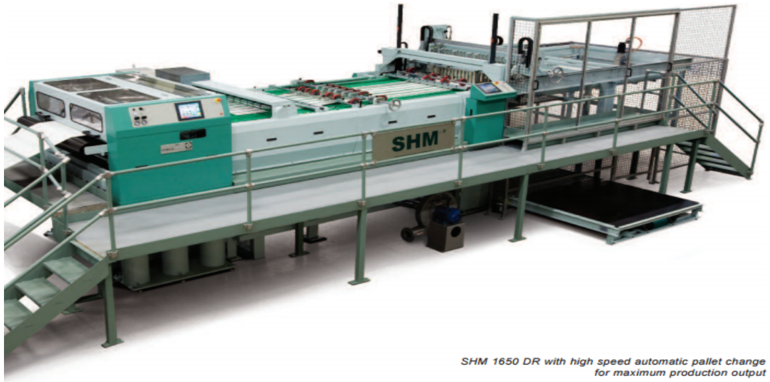


Figure 10: SHM 1450 Sheeter

	1450 SR	1450 DR	1450 SR Compact
Web width (min.) Web width (max.)	400 mm (16") 1500 mm (59")	400 mm (16") 1500 mm (59")	400 mm (16") 1500 mm (59")
Sheet length (min.) Sheet length standard Sheet length (max.)	279 mm (11") 1500 mm (59") 2100 mm (82")	279 mm (11") 1500 mm (59") 2100 mm (82")	279 mm (11") 1500 mm (59") 2100 mm (82")
Pile height including pallet (max.) Standard Optional	1200 mm (47") 1700 mm (67")	1800 mm (70")	1700 mm (67")
Ream height (max.)	127 mm (5")	127 mm (5")	127 mm (5")
Pile weight (max.)	2000 kg (4410 lbs)	2000 kg (4410 lbs)	2000 kg (4410 lbs)
Web speed (max. mechanical)*	300 m/min. (984 ft/min.)	350 m/min. (1150 ft/min.)	100 m/min. (328 ft/min.)
Knife load cross cutter* Single web Multiple web	400 gsm 500 gsm	1000 gsm 1000 gsm	400 gsm 500 gsm
Sheet squareness	+/- 0.50 mm (.02") at 1000 mm (39") sheet width		
White web sheet length accuracy	+/- 0.50 mm (.02")	+/- .381 mm (.015")	+/- 0.50 mm (.02")
Cut-to-register accuracy High tolerance	+/- 0.15 mm (.005")	+/- .381 mm (.015")	+/- 0.15 mm (.005")

Figure 11: SHM listed product specifications

The last competitor mentioned is Kingsun machinery. Kingsun manufactures the KS-A series Servo control high speed rotary sheeter, as with the first two competitors it lists a feed speed of 300 meters or 948 feet per minute and this is once again below MAXSON's desired speed. It offers comparable cutting lengths compared to MAXSON's design and appears the most similar to MAXSON's design. One interesting point to note with this machine is that it only features a single rotating blade as MAXSON uses a dual rotary blade. Considering the manufacturer lists that the feed speed is affected by paper material and type it is probable that the actual feed speed would be even lower than it is listed since the machine would rarely be run under ideal conditions, and with ideal paper materials. However, the cutting accuracy is similar to MAXSON's specifications for required accuracy. Figure 12 shows the Kingsun KS-A Series servo control High speed rotary paper sheeter. [6]

Figure 12: KS-A Series Servo Control High Speed Rotary Sheeter

5.3 Comparing to Competition within MAXSON

In addition to competition from competitors there is also some competition within MAXSON themselves, as they have disclosed some attempted solutions that were tried and tested before. While MAXSON attempted many solutions, only a few were successful, and while they were successful they were impractical on a large scale, only practical as a one time attempt due to the added machine time and material costs. The three successful solution MAXSON had attempted were a carbon fiber bodied lower blade, an "Always Be Synchronous"-style solution and a ribbed design with a skin over the outside.

The carbon fiber solution was a very effective solution that MAXSON had created. It resulted in a cutting blade with a significantly lower mass and inertia. The main drawbacks were that the carbon fiber body required outside manufacturing to produce it, and in addition it was extremely expensive and would not be cost effective being applied to more than one cutter. The carbon fiber rotary cutter is currently still in use according to MAXSON and has had no problems keeping up with its cutting system. The loss of inertia and comparable stiffness to the steel blade made it an attractive solution. This solution also required a lead time that was decided by the blade manufacturer and MAXSON would prefer to keep manufacturing in house to speed up repair and replacement part time. However due to the drawbacks in manufacturing cost MAXSON has indicated it would like to avoid this route.

The second in-house solution that MAXSON attempted was the "Always Be Synchronous" system. This was a design solution that was actually proposed by the design team in the initial critical concept review presentation. After the presentation MAXSON revealed to us that a solution like the one mentioned had been done and was successful, but was far too expensive to produce again and required a large amount of manufacturing time. The current Always Be Synchronous design used varying radius rotary cutters, which allowed the cutting blades to spin at the same speed as the feed to allow the rotary cutter to spin at one constant speed. Since the cutter was always at the same speed it resulted in a very low power motor to be used because the system did not require any acceleration or deceleration during the cutting cycle as the current system does. Overall MAXSON informed the design team that while an Always Be Synchronous system was feasible on paper, when it was put in to practice it became far too impractical to produce.

The last solution that was attempted by MAXSON was the ribbed rotary blade that was coated in a "skin" after wards to allow it to be smooth. This solution used a ribbed cutter that had significant mass reduction, which in turn lead to decrease in inertia. The problem with this design was that it needed to be smooth in order to prevent the feed from jamming. In order to prevent the feed from jamming a skin was fit over the outside of the rotary cutter. The fitting of the skin was proved to be quite a hassle by the assembly team, it was noted from MAXSON that the skin took several hours to put on and required additional labor and manufacturing costs that out weighed the decrease in inertia. Since it was not practical on a large scale production according to MAXSON the design team attempted to avoid this problem as it continued in it's concept generation.

6 Specifications Definition

Before meeting with our sponsor MAXSON Automatic machinery Spin to Win had some ideas about how the design could be, but as a group we couldn't advance its ideas until it met with the sponsor and discussed some design considerations that are crucial to the overall success of the design. These design considerations would help the team better understand what issues needed to be tackled in the overall planing of the design as well as what MAXSON itself was looking for from this Capstone. Upon meeting with MAXSON and its, engineers the problem definition was still unclear, but now the team had some valuable information to go from that could help it tackle the main goals of the design.

Spin to Win then began brainstorming and finding different interesting patents that could help the team better understand the background and processes that go into sheeters. From these patents the team could narrow down ideas that haven't been already thought of by other inventors. The number of patents for existing sheeters was not large, which gave the team room to be creative. After research from the patents was gathered, then the concept generation began. The concepts generated were not all possible or practical, but it allowed the team to gather more information about the problem and gain a better understanding of the considerations needed for the overall design. Some of these generated designs concepts were adequate in possibly solving the problem given to the team by MAXSON.

From here on out the team had developed a list of considerations found through mathematical analysis as well as requirements and goals addressed to us by the sponsor. With this information a design specifications list was compiled and each specifications in its own right has an importance in the success of the design. The design specifications can be broken down into three sections. The first one being production quality and rate goals, which are the biggest considerations that go into the design for the system. The target design for the cutters and gear housing produce web feed rates at around 1000 feet per minute and use an off-the-shelf motor at rated 1750 rpm. Our design needs to be able to handle those rates and be able to go beyond them if necessary. Additionally added with those rates is the cut length of the sheets, which is 41 inches at the 1000 feet per minute web feed rate. Our design needs to adhere to this in order to keep the quality of cut ideal for production.

Cut quality is crucial for a dual rotary cutter, so with that said the design needs to have adequate stiffness in order to reduce the deflections that cause poor cut quality. Vibrations are connected to those deflections that can be seen in the rotating shaft, so they go hand in hand with cut quality. After the creation of the design for the team's proof of concept, manufacturing costs and time became a factor to consider as well. The machining time needed to be within a reasonable amount of time compared to the current design as well as the cost to manufacture the cutter and roller needed to also be within reason.

The second would be the mechanical specifications that the design needs to meet in order to prove the design works and that it can provide the rates and production quality needed to optimize the sheeter. The most crucial component of this section of the design was its stiffness. The design needed to show inertial reduction while maintaining adequate stiffness in comparison with the existing model. The next component of importance became ease of assembly when the design was created on Solidworks and became visually real. The design needed to be able to assembled on site in the facility after the fabrication process was complete. The design incorporates more smaller parts instead of one or two large solid pieces like the current design. Another component after the design was created on Solidworks was the ability to maintain feed stability as well as keep some of the existing hardware already by MAXSON. They went hand in hand because the feed stability is affected by the existing knife hardware for example. To fit the existing knife hardware into our design the team also had to consider a way to do that without interrupting the feed as it is fed through the system. This consideration for the hardware as well as the feed would allow the group to minimize the catches, crimps, tears, and interruptions in 10 hours of continuous operation.

The third and final section of the design specifications is the manufacturing costs, time, and budget of the project. The manufacturing costs of this design needed to meet the existing costs, or at least be within an allowable margin of added cost to the existing costs. Manufacturing time or machining time needs to be within an allowable margin of added time, or also meet the existing time it takes the current design to be machined. The overall budget of the system given to the team was a figure of $35,000. This budget is an estimate of the overall cost of the current system, but as it stands the design will be well within the budget given to us by MAXSON.

6.1 Specifications Definition Up to the End of the Year

When the second half of Capstone began, the Spin to Win team met with Maxson Automatic Machinery in order to go over the next steps in the design and going forward. Maxson had seen our design for the wheel and spokes model, and realized that the machines they had on hand could not adequately machine our design due to the complexity of the slugs. With this new information the team went back to the drawing board for the slugs, and needed to find a simple design approach that could allow the slugs to be manufactured for the final prototype. Applicability to the current design was now on the back-burner, and ease of manufacture became one of the new design specifications for the second half of Capstone.

Production Quality and Rates Requirements	
Web Feed Speed	>= 1,000 fpm
Cut Size @ 1,000 fpm	<= 41 in.
Motor Rated RPM	1,750 RPM
Cut Quality	Total knife deflections ≤ 0.001; Tangential angle of cut edge ≤ 2° from straight line to cut corners.
Cutoff Reliability	Within 0.2% of cutoff length
Vibration Amplitude	Less than or equal to current system's
Mechanical Requirements	
Feed Stability	No feed catches, crimps, tears, or interruptions in 10 hours of continuous operation. Deflection allowable (.02 in)
Ease of Assembly	Design needs to be easily assembled in house at the MAXSON facility, max 2 hours added assembly time
Stiffness	Equal or greater stiffness compared to the current design, allowable ± 0.0001 inches
Existing Hardware Constraints	Design needs to be able to be incorporated into the existing machine as well as be able to incorporate current knife hardware.
Machinability	Design needs to simply enough to machine
Manufacturing Cost and Time	
Manufacturing Time	Allowable +7 hours from current machining time of 20 hours for bottom cutter
Manufacturing Cost	Allowable +$2,500 from current machining cost of $6921 for bottom cutter
Total Budget Cost for System	≤ $35,000

Figure 13: Current Design Specifications for End of Semester

7 Conceptual Design

7.1 List of individual concepts

From the earliest stages of the design process, each team member brainstormed unique concepts that could conceivably participate in a design solution. At that stage of the process, the problem was barely defined, and most of these concepts had to be left on the winnowing floor as impractical. However, the process of generating these concepts was itself a deepening of the team's understanding of the problem, and many of the ideas generated found their way to later steps of the design process.

All of Spin to Win's generated concepts are listed by name below.

1. Welded pinion
2. Pinion with planetary element
3. Transmission shaft under cutter base
4. Transmission shaft above cutter base
5. Magnetic bearings
6. Daisy-chained spur gears
7. Liquid cooled motor with combined lubricant/cooling system
8. Brake with magnetic clutch
9. Tapered transmission shaft
10. Reorient blower
11. Wheel and spokes, welded
12. Wheel and spokes, bolted
13. Partial brake pad
14. One-way gear teeth
15. Storebought gearbox
16. Helical gears cut into cutter and motor shafts
17. Spur gears cut into cutter and motor shafts
18. Wider gap between draw and drive motors
19. Internal combustion engine
20. Swept lobe top cutter

21. Swept lobe top cutter with modular blades
22. Inertia-optimized gears
23. Modular shafts for synchronous use
24. Modular shafts secured by threaded rods
25. Switch gearings for different portions of cutting cycle
26. Use aluminum on low-load parts
27. T-lobe top cutter geometry
28. Optimize knife radii versus power/cycle curves
29. Auxiliary motor for peak torque portions of cycle
30. Thinner gears
31. Gear reducing pinion below the motor
32. Relocate motor above machine with chain and sprocket to drive the system
33. Offset beveled gears to apply a gear reduction and save space
34. Relocate the motor to the opposite side of the cutter
35. Reduce the drive gear
36. Inline gear box transmission
37. Change the motors location to be facing the opposite direction and run a gear reduction beneath it
38. Using double chains and sprockets to apply a reduction
39. Using a large metal belt to apply a gear reduction between two sprockets
40. Bolting the reducing gear directly to the rotary blade to save space
41. Interlocking gears similar to a drive shaft to connect the motor to the Rotary blades
42. Using bevel gears to rotate the motor 90^o and save space for a reduction
43. Planetary or orbital gear reduction
44. Using 4 planet gears in a planetary reduction
45. Placing a planetary gear reduction inline with the drive
46. Mount the planetary gear reduction directly inside the rotary blade

47. Combining both a planetary gear reduction and chain and sprocket system
48. Using two wheels and a "skeleton" to replace the upper and lower blades
49. using triangular supports to support the "Skeleton" system above
50. Replacing the bottom cutter blade with polycarbonate or another plastic and having metal inserts to protect the knife hardware
51. Using a stiff spoke offset gear system
52. Using a "flip switch" that directs power from the motor to brakes to slow the system
53. "Differential" style reduction
54. Enlarged radius "differential" style reduction
55. Inset fear reduction system
56. Flywheel and brake and clutch to slow the motor and maintain inertia
57. Single long gear beneath the drive gear and rotary gear system
58. Spiral gear placed 90^o offset from the current set up
59. Epicyclic beveled gear reduction
60. Using the epicyclic beveled gear reduction with a large number of planetary gears
61. Radial removal of shaft material
62. Longitudinal removal of shaft material
63. Dilobe shaft
64. Hollow shaft
65. Narrow shaft with long cutter blade
66. Vertically displacing shafts: magnetically controlled
67. Retractable cutter: magnetically controlled
68. Retractable cutter: cam controlled
69. Retractable cutter: hydraulically controlled
70. Vertically displacing shafts: hydraulically controlled
71. Vertically displacing shafts: cam controlled
72. 90^o gear fitting to save room in gear housing

73. Overhead curved knife blade
74. Two shaft pairs: perforated cut
75. Linear knife
76. Fluid coupling with break to control shaft speed
77. Pinion attached to near gear, then new gear joined to existing gear system
78. Two shaft pairs: incomplete cut
79. Two linear knives: perforated cut
80. Two overhead curved knife blades: perforated cut
81. No lobes with strong bearing housing
82. No torsion control gears with strong bearing housing
83. One shaft pair to perforate, two more to create tension at perforation
84. Perforating shaft pair and curved knife to complete cut
85. Curved knife to perforate, shaft pair to cut
86. Rotating follower on linear cutter to perforate, Shaft pair ot cut
87. Shaft pair to perforate, rotating follower to cut
88. Dual linear cutter blades, one to perforate, another to cut
89. Shaft pair to perforate linear blades to cut
90. Pair of narrow shafts with long cutters
91. Scissor cutters
92. Linear cutting blades
93. Double helical gears
94. Worm gear
95. Center point blade
96. Teeth blade top cutter
97. Involute gear
98. Two top rotary blades
99. Three top rotary blades

100. Laser cutter
101. Pendulum blade
102. Spear pendulum blade
103. Simple helical gear reduction
104. Pulley system for cutters
105. Spiral bevel gear reduction
106. Hypoid gear reduction
107. Inertial reduction of gears in gear housing
108. Loom cutter design
109. Slider and plunger cutting system
110. Pendulum plunger catch setup

7.2 Evaluation of each concept

As Spin to Win moved toward selecting concepts for actual production, each of its 30N ideas was evaluated for its merits and for what portions could be used. Each concept is given a brief description and evaluation below. While few concepts made their way to the final design in whole, many concepts were put into play in part.

Welded pinion A pinion gear could be welded directly to the end of the motor shaft. This is a simple solution to the transmission of power, but risks voiding motor warranties and distorting the system's dimensions.

Pinion with planetary element A pinion gear could be placed inside a planetary element which has relative motion with it. This allows a larger or more customizable reduction at the expense of complexity.

Transmission shaft under cutter base A shaft could placed inside the cutter base, which appears to be unused space[2]. This could supply a reduction with zero footprint, but the system of bearings requires careful design.

Transmission shaft above cutter base This is similar to the previous idea, capitalizing on unused space in the cutter footprint. In addition a fan could be fixed to this shaft to provide extra cooling to the motor. Failure of the bearings of a suspended power transmission shaft, however, would be catastrophic.

[2] It was since learned that this space is used for electrical components.

Magnetic bearings Magnetic bearings would have a near-zero coefficient of friction, lowering system torque requirements. This would come at the price of significant cost.

Daisy-chained spur gears This could allow a customizable reduction to be used in transmitting power from the motor to the cutters. The advantage of customizability is completely obliterated by the addition of cost, complexity, maintenance requirements, etc.

Liquid cooled motor with combined lubricant/cooling system If a liquid cooled motor is to be used, a single fluid circulation system could both lubricate the bearings and cool the motor. This added efficiency might offset the added complexity of a liquid cooled system. MAXSON has expressed a desire not to go liquid cooled, however.

Brake with magnetic clutch An electronic system already controls the motor torque outputs and could be connected to a magnetic clutch for braking where needed in the cycle. This would lower overall power requirements at small extra cost, but would require programming for integration into the motor drive.

Tapered transmission shaft This could allow a larger or more customizable reduction to be used with many of the same drawbacks as the shaft ideas above.

Reorient blower Spin to Win was initially under the impression that the motor was cooled by an external fan aimed at the unit from above. This would be inefficient from a cooling perspective as the natural and forced convective currents would oppose one another. Aiming the blower from the side or below would be an improvement at no cost. In reality, the blower is integrated and passes air along a channel which runs axially down the motor.

Wheel and spokes, welded Since the cutters only rotate in one direction, it would make sense to optimize stiffness in the direction of rotation. This idea involves the use of curved spokes that greatly resist deformations in the direction of the rotational stresses. Its drawbacks are cost and complexity.

Wheel and spokes, bolted By bolting the spokes to the main shaft rather than welding them, a little bit of assembly time is eliminated at the cost of strength.

Partial brake pad The cycle involves speeding up and slowing down at predetermined times. A brake pad designed to brake only for the needed portions of the cycle, based on its geometry, would allow braking without a control system and at no extra cost.

One-way gear teeth Again, since the rotation does not reverse, it would make sense to optimize strengths and stiffnesses in the direction of rotation. It is difficult to envision how to do this with gear teeth, however, which also involves the necessity of custom gears.

Storebought gearbox This would be a simple but expensive solution for providing an in-line reduction at the price of axial space.

Helical gears cut into cutter and motor shafts This would be a very space-conservative way to produce a gear reduction between cutter and motor at the price of machining time and the motor warranty.

Spur gears cut into cutter and motor shafts This is a stronger but less smooth variation on the above idea.

Wider gap between draw and drive motors Currently the space between the draw and drive motors is very small and likely experiences elevated temperatures from the motor heat during operation. Overall motor operating temperatures could be lower if this space allowed for more natural convection.

Internal combustion engine This is a silly idea to maximize on power and potentially cheapen operation at the expense of a complete system redesign, loss of easy control, and host of other drawbacks.

Swept lobe top cutter It is possible to apply the "one-way rotation" idea to the top cutter, which yields a swept lobe shape designed to be difficult to deflect in the direction of rotation. Later finite element simulations proved this idea impractical.

Swept lobe top cutter with modular blades This is the above idea except, instead of a single machined piece, it would involve a central shaft with slots where lobes could be affixed and removed. This would introduce some customizability but provides stiffness challenges.

Inertia-optimized gears The weakest point of the gears is the teeth. By removing material from the very thick bodies, some inertia can be removed from the system at small cost.

Modular shafts for synchronous use This idea involves using different cutting radii to approach synchronous speed for all cutoffs. It involves a hugely ambitious redesign of the system.

Modular shafts secured by threaded rods This is a simplification of the above idea, with a single central shaft, "sleeves" of differing radii, and threaded rods used to affix the two. It suffers many of the same drawbacks in terms of system redesign.

Switch gearings for different portions of cutting cycle This idea involves using a clutch and gearing system to provide high-torque and high-speed portions of the cutting cycle. Unfortunately, for the 1000fpm/40" cycle, the highest torques also happen at the highest speeds, and so this idea brings nothing to the table.

Use aluminum on low-load parts This idea involves identifying those parts of the system that carry small or negligible loads and swapping them out for aluminum in order to minimize weight. The fabrication time and costs likely offset the saved inertia.

T-lobe top cutter geometry This idea involves using lobes with a thin, central arm sweeping out to two tangential arms, like a T. It suggested low inertia but the stiffness of those thin arms was likely compromised.

Optimize knife radii versus power/cycle curves This isn't an idea for fabrication so much as a study that could be carried out. The idea is to locate a central knife radius that provides the lowest overall power requirements across the intended operational range of the machine. This could potentially add value at no cost.

Auxiliary motor for peak torque portions of cycle Since motor costs are generally in the square of their power, it seems possible to use two smaller motors instead of one large one and some torque-summing gearing, such that the second motor could be geared in during the peak-torque portions of the cycle. The engineering problem of torque-summing on the fly makes this difficult.

Thinner gears The current gears are extremely thick, adding a large component of inertia. However, it has since been learned that thinner gears result in chipping of the teeth, so the gear thicknesses cannot be changed.

Gear reducing pinion below the motor By placing the reducing pinon below the motor it is possible to move have a gear reduction without modifying the current cutter system too much. The current cutter system seems to be a little too tight to fit in this style of gear reduction.

Relocate the motor above the machine and use a chain and sprocket system By relocating the motor it would save space in the bearing housing while still allowing for a gear reduction, however using chains could be problematic when there is backlash in the system.

Offset bevel gears to provide a gear reduction and save space By using bevel gears it is possible to achieve a gear reduction while being slightly offset, but the beveled gears would take up a large amount of space.

Relocate motor to opposite Side of the cutter system to allow more room for a gear reduction By moving the motor to the opposite side and using a drive shaft to transfer the motion to the other side, there is a lot of room opened up for gear reductions. This would however involve a large system redesign.

Reduce the drive gear This is a rather simple solution that would involve reducing the gearing of the input motor by using slower motor that has higher torque, however this would potentially be more expensive and could be solved with other solutions that would result in a greater cost savings elsewhere.

Inline gear box transmission A bought or manufactured transmission could be attached to the drive motor that would reduce the incoming speed by three and theoretically increase its torque output, however this would add a significant amount of inertia and was not recommended by mason.

Change the motors' location to be facing the opposite direction and run a gear reduction beneath it By moving the motor to the top of the machine and turning it the other way you would be able to reduce the gearing with several or a single spur gear however, this space is currently taken up by the existing rotary blades and bearing housing.

Using double chains and sprockets to apply a gear reduction By using double or ever triple chains to drive a chain and sprocket gear reduction it would spread the load more evenly across the teeth in the sprocket and would be less likely to fail, however chains require maintenance, lubrication, and are also loud, and dangerous.

Using a large metal belt to apply a gear reduction between two sprockets Replace the chain in the chain and sprocket with a heavier metal band similar to the treads on a tank to carry the load between two sprockets and apply a gear reduction, however this requires a lot of maintenance, is dangerous to be around, and would result in significant inertial gains to the system.

bolting the reducing gear directly to the rotary blade to save space If a reducing gear were to be used one method of attaching it could be to bolt it directly to the existing rotary blades.

Interlocking gears similar to a drive shaft to connect the motor to the rotary blades This is another method of transferring a reduced input to the rotary blades, by using an interlocking axle similar to those seen under large trucks.

Using bevel gears to rotate the motor 90^0 to save space for a gear reduction The bevels in the gear reduction would allow the gears to be run at 90 degrees from each other which would result in a fair amount of extra space that would be used to move the motor and relocate it, and would in turn allow for a larger reducing spur gear to be applied. This would require the bearing housing to be redesigned to accommodate the gears.

Planetary or orbital gear reduction A planetary gear reduction would reduce the incoming angular velocity, by a large margin and would be able to be placed inline with the current system, or off to the side, resulting a minor redesign of the bearing housing. This was seen as not being plausible due to the increases in inertia.

Using 4 or less planet gear to apply the reduction Using less gears in the planetary gearing system would result in a lower inertia gain but, would increase the stresses on the individual gears.

Placing the planetary gear reduction inline with the drive system If the planetary gearing were to be used it could be put in line with the existing assembly and would be simple to fit into the current system

Mount the planetary gear inside the rotary blade The planetary gears could be installed into the existing gears that power the rotary blade, which would take up virtually no extra space and would apply a reasonably large reduction.

Combining both a planetary gear reduction and chain and sprocket system By combining both a planetary gear reduction and chain and sprocket reduction one could theoretically split the reduction forces between multiple systems and spread the load among them reducing the stress on individual components.

Using two wheels and a "skeleton" on the outside to replace the upper and lower blades This idea involves creating two wheels at the end of the rotary blades, that would be held together by trusses along the outside effectively reducing the inertia by a significant portion as most of the mass would be removed.

Using triangular supports to reinforce the "skeleton" system above The skeleton and wheels idea would likely need some form of reinforcement to help prevent it from deflecting under load and while spinning so one idea would be to use triangular support trusses in it to provide some reinforcement.

Replacing the bottom cutter blade with poly carbonate or another plastic If the cutter blade were to be replaced with a plastic or polymer and have metal inserts where the knife hardware goes, it would effectively reduce the mass and inertia, but it may have issues with the stiffness and longevity.

Using a stiff spoke offset gear system Using what is basically two wheels connected by bars or spokes you could apply a gear reduction when two axles are offset.

Using a "flip switch" that directs power from the motor to the brakes to slow the system by using a flip switch that changes the power from the motor to a braking system to slow the motor allowing one to overcome the inertial problems of slowing the blades down.

"Differential" style reduction This gear reduction idea would have two plates and gears sandwiched between them that looks similar to a differential.

Enlarged radius ”differential” style reduction If one of the gears in the ”differential” looking gear reduction were to have a larger radius than the other you could reduce the speed of one of them.

Inset gear reduction system This gear reduction would have a smaller gear that rests inside of a larger gear. The larger gear having teeth on the inside and outside would result in the smaller gear inside driving the larger outer gear at a reduction.

Flywheel, brake, and clutch system to slow the blades and maintain inertia By using a system with a flywheel clutch and brake one could conserve some inertia and use it to restart the spinning blades momentum, which would in theory allow one to use a slower motor once the flywheel is up to speed.

Single long gear beneath the drive gear and rotary gear system Using a long gear for this would accomplish the gear reduction that is acceptable for the current system, but it would require some redesigning of the bearing housing to fit it into the current assembly.

Worm gear placed 90^0 offset from the existing setup Using a worm gear would result in a very large reduction and would allow for the motor speed to be reduced however it would not be well suited for this application as the reduction would be very large and would require the entire load to be carried through a small gear.

Epicyclic beveled gear reduction Using an epicyclic gear reduction system would allow one to use a variation of the orbital gear system and get a very large reduction and support a large load but would be costly to produce and result in large inertial gains.

Using the epicyclic gear reduction with a large number of planetary gears Using more planetary gears would result a smoother gear reduction and but once again results in some inertial gains which was not acceptable by MAXSON

Radial removal of shaft material Removing material from the shaft radially would allow for a decrease in inertia, yet an accompanying decrease in stiffness. Removing material in this way would form ribs which could support the sheet.

Longitudinal removal of material Removing material in this way would also decrease inertia and stiffness, also the cut out sections would need to be narrow so as not to interfere with the translation of the sheet.

Dilobe shaft A dilobe shaft would allow for a shaft with less interia than a trilobe shaft.

Hollow shaft A hollow shaft would have a decreased inertia and stiffness.It is improved by the spoke design because the spokes increase stiffness in greater proportion to the increase of inertia.

Narrow shaft with long cutter blade This design incorporates a long cutter onto a narrow shaft of minimal inertia, the decrease in stiffness however, was later found to be unacceptable.

Vertically displacing shafts:magnetically controlled This design uses electromagnets to move cutter blade shafts out of position when not making a cut, and into position when making a cut. The design was found to be beyond the scope of the project, due to the scope and cost.

Retractable cutter: magnetically controlled The design would retract the blade of the cutter with an electromagnet when it is not to make a cut, and push it into position when when it is to cut. The design fails for the same reasons as the displacing shafts.

Retractable cutter: cam controlled This design uses a cam and follower to displace the blade, as opposed to an electromagnet.

Retractable cutter: hydraulically controlled This design uses a hydraulically actuated lever arm to move the cutter blade in and out of position as opposed to a cam or electromagnet.

Vertically displacing shafts: cam controlled A large cam would be used to displace the shafts out of position when a cut is not being made and back into place when a cut is to be made.

Vertically displacing shafts: hydraulically controlled Hydraulically actuated lever arms would displaced the shafts.

90^o **gear fitting to save room in gear housing** This design would use a bevel gear to transate power from a pinion to the gears attached to the shafts. Though more compact, the bevel gear would have to be supported by a bearing.

Curved overhead knife blade This blade would replace the shafts, but would require a new mechanical support system and electronic control system for the motor. The blade would descend upon the sheet to cut it.

Two shaft pairs one to perforate, another to cut Two shafts would allow for two lower-inertia shafts, and less force would be on each shaft during the cut, it now understood however that the force involved in cutting the sheet is not significant.

Linear knife moving back and forth to make cut This design would slice the sheet as razors through a box, the blade would likely have to be replaced regularlly to maintain cut quality , a high quality cut could even be achieved.

Fluid coupling with brake to control shaft speed This design would allow the motor to rotate at a differential rate to the shafts, however energy would be wasted when the brakes engaged and speed would be difficult to predict.

Pinion attached to new gear, new gear attached to shaft gears This design would use a pinion to incorporate a gear reduction. While the group is now focused on inertia reduction, this design may be revisited.

Double shaft pair one partial shallow cut, another to complete cut This would distribute the cutting force among the two shafts, but the cutting force is no longer of concern.

Two linear knives, rotating follower to perforate, another to cut This design would use two linear blades with saw-like followers,the concepts involving perforation are no longer considered to have merit because of the insignificant force involved in the cutting action.

Two curved blades: one to perforate another to cut The design involves two curved blades descending upon the sheet.

No lobe, stronger bearing housing with no lobes, the blade attached to the shaft would not be balanced, however, a stronger bearing housing could mitigate the forces developed as a result. This design however, would lead to unacceptable vibrations, and decreased stiffness.

No torsion control gears, stronger bearing housing It was believed that torsion control gears were at the end of the shafts because the bearings could not support the torsion, but they are, in fact, there to reduce torsional deflection, which a stronger bearing housing would not do, so this design has no merit.

One shaft to perforate, two more to create tension at perforation This design would greatly reduced the forces involved in making the cut, but that is of no benefit.

Perforating shaft pair and curved knife to complete cut Can be more compact than two shafts, but perforation concept of no benefit to project goals.

Rotating follower on linear blade to perforate, shaft pair to cut More compact than other perforation options, but is not of benefit.

Shaft pair to perforate, rotating follower to cut Similar to previous concept with the roll of shaft and follower blade reversed.

Dual linear cutter blades, one to perforate, another to cut This design eliminates shafts completely and incorporates the perforation concept.

Shaft pair to perforate, linear blades to cut uses smaller shafts than those currently used, and incorportes a linear cutting blade system.

Narrow shafts with long cutters used to perforate and cut Applies perforation concept to narrow shafts with long cutter blades.

Scissor cutters Change rotary blades to a linear cutting system. Operates like scissors. Spring attachment to open and close blades. Energy output would be reduced as a result of this setup as well as inertia.

Linear cutting blades Use linear cutting blades on top. Used with a spring attachment as well as a control system to time the blades with the feed system. This design gives the system less moving parts as well a distinct decrease in inertia. Increase of feed speed could be possible.

Double helical gear This design is for a gear reduction. Changing the type of gear used from helical to double helical would add gear strength and allow for a reduction on the shaft connecting the bottom drive gear and motor.

Worm gear For a gear reduction to the system change the type of gear to a worm gear. This would decrease inertia in the gear housing as well as allow for a better gearing ratio.

Center point blade This design will help to lower the inertia of the system allowing for a smaller motor and faster feed speeds. System will need a program to control blade movement.

Teeth blade top cutter This design will allow for a lower inertia of the top cutter because it is a narrow shaft with blade teeth. The lower inertia will create less stress on the system.

Involute gear For a gear reduction in the gearing housing. Change gear type to involute gear for added space in housing for a gear reduction. However, gear strength could be lost with this design.

Two top rotary blades Add a secondary cutter to the top cutter system, which would allow for faster feed rates at the cost of added inertia.

Three top rotary blades Add a secondary cutter and tertiary cutter to the top cutter system, which would allow for faster feed rates at the cost of added inertia.

Laser cutter Install laser sources in a conveyor-like setup to establish possible faster feed rates and cut quality because there is less of a chance for vibrations. This design would also greatly decrease the inertia on the system as well as the wear the parts would feel.

Pendulum blade High tensile strength rope or wire will be used to swing blade at different frequencies to match cut speeds. This design allows for a decreased inertia as well as wear on the system.

Spear pendulum blade Use of a metal shaft to connect to a hinge over the feeder with blade attached at the bottom. This will in turn decrease the overall inertia and add stability to the system allowing for better cut quality.

Simple helical gear reduction Improve gear ratio by skrinking the bottom drive gear to about 1/3 radius and teeth, so that it can obtain a lower inertia and produce less wear on gears in the gear housing.

Pulley system for cutters Should allow for an inertia reduction on the overall system, allowing for a possible increase in torque and power to system. However, could add more moving parts and more possible points for deflection.

Spiral bevel gear This design change should allow for an improvement in the gear ratio, in turn giving the system a gear reduction. With spiral bevel gears teeth strength will not be lost.

Hypoid gear Simple to spiral bevel gears in that this design change should allow for an improvement in the gear ratio, in turn giving the system a gear reduction.

Inertial reduction of gears The inertia can be improved and lessened in the gear housing, if the gear designs are tweaked the lessened inertia could allow for less wear on the gears in the housing.

Loom cutter design High tensile string or wire with a small blade attached in the middle by two strings or wires on top and bottom. This should allow for faster feed rates, and also greatly reduce the inertia of the system and the cutters.

Slider and plunger cutting system Use of a lubricated slider with a slit on bottom of the track with a plunger attached to a spring to send the slider across the system to cut the paper. This setup will greatly reduce inertia in the top cutter as well as add more control of the system.

Pendulum plunger catch setup Pendulum cutter will be attached to a catch that will collide with a plunger attached to a spring to move the cutter. This will add more control to the system. This will allow for faster feed rates and will help to lower inertia on the system.

7.3 Three critical concepts

Moving forward it was necessary to winnow out those concepts that didn't contribute enough to a solution or weren't worth taking seriously, and combine whatever portions of the remaining concepts were effective into three critical concepts for review.

The first was a redesign of the lower cutter in which steel would be machined away in portions and replaced with a lighter, molded plastic. This would drastically remove weight and inertia from the cutter without altering its geometry, allowing it to continue to support a smooth feed.

The second was a redesigned lower cutter geometry to use an inner shaft, an outer tube, and a series of supporting spokes. This geometry promised a reduction in inertia by the removal of a large amount of material and the likely preservation of stiffness, while being complex and difficult to integrate with MAXSON's existing hardware.

The third was an ambitious redesign of the system to actively control the cutting radius so that the rotational speed didn't need to be changed. This would have very large benefits and very large drawbacks, the former in motor cost and cutter simplicity, the latter in maintenance times, control system requirements, and set-up costs.

After discussions with MAXSON, it was determined that the wheel and spokes design offered the greatest promise, and it was selected to be pursued as a serious design. The reasons for this selection will be discussed in Section 9, Design for X.

8 QFD

Quality Function Deployment is a decision-making and organizational tool for design activities. It places customer requirements and design features in a matrix, weights their mutual impacts, and outputs a quantitative measure of the way in which a proposed design meets the design goals. Spin to Win developed a QFD analysis of its designs in parallel with their creation and development in order to select a design moving forward.

The QFD analyses developd by Spin to Win evolved together with the designs.

QFD Group 24 Maxon Sheet Cutter

		Strong Material	Standardized Gears	Powerful Motor	New Shaft Geometry	Gear Reduction
Consistancy	10	SP		P	WN	
Reliability	10	SP		P	WN	
Versatility	6	WP		SP	P	P
Maintainability	6		SP			
Longevity	6	SP			WN	
Cost	4	SN	WP	SN	SP	P
Total Importance		232	68	120	18	50

Correlations

SP=Strong Positive (10)
P=Positive (5)
WP= Weak Positive (2)
WN= Weak Negative(-2)
N=Negative (-5)
SN=Strong Negative(-10)

Figure 14: Original QFD

The original QFD, shown in Figure 14, indicated the customer requirements and technical aspects which were consistent with those requirements. It did not however include correlations between the technical aspects nor did it include an analysis of the competition. The QFD was later updated and includes these analyses. Technical aspects which do not pertain to the design, "Powerful Motor" for example were removed in favor of aspects which are directly affected by the new design.

Figure 15 shows the terms for the revised QFD. The customer requirements remain the same though new technical aspects are analyzed, stiffness for example.

Figure 16 shows the weighted importance of the technical aspects of the design. A strong material is the most important feature, but since steel is the predetermined material the most important feature to emphasize in the design is the stiffness.

Figure 17 shows an analysis of MAXSON versus its competitors. The analysis reveals that MAXSON delivers a superior product, though at a higher cost than the competition.

				Column #	1	2	3	4	5	6	7
				Direction of Improvement: Minimize (▼), Maximize (▲), or Target (x)	▲	x	▲	▲	▲	▲	
Row #	**Max Relationship Value in Row**	**Relative Weight**	**Weight / Importance**	**Quality Characteristics** (a.k.a. "Functional Requirements" or "Hows") / **Demanded Quality** (a.k.a. "Customer Requirements" or "Whats")	Machinability	Strong Material	Low Inertia Geometry	High Stiffness	Manufacturability	Disassembleable	
1	9	23.8	10.0	Consistancy		Θ	O	Θ			
2	9	23.8	10.0	Reliability		Θ	O	Θ			
3	9	14.3	6.0	Versatility		O	Θ	Θ			
4	9	14.3	6.0	Maintainability					O	Θ	
5	9	14.3	6.0	Longevity		Θ	▲			Θ	
6	9	9.5	4.0	Cost	Θ	O	O	O	Θ		
7											

Figure 15: QFD Terms

Target or Limit Value						
Difficulty (0=Easy to Accomplish, 10=Extremely Difficult)	6	2	8	8	7	5
Max Relationship Value in Column	9	9	9	9	9	9
Weight / Importance	85.7	628.6	314.3	585.7	128.6	257.1
Relative Weight	4.3	31.4	15.7	29.3	6.4	12.9

Figure 16: QFD Base

Figure 18, the top image of the house of quality, shows the correlations, both positive and negative, between the various technical aspects and customer requirements. Based on the results of the QFD, it is clear that the most pressing design aspect is to increase the stiffness of the shafts where ever possible.

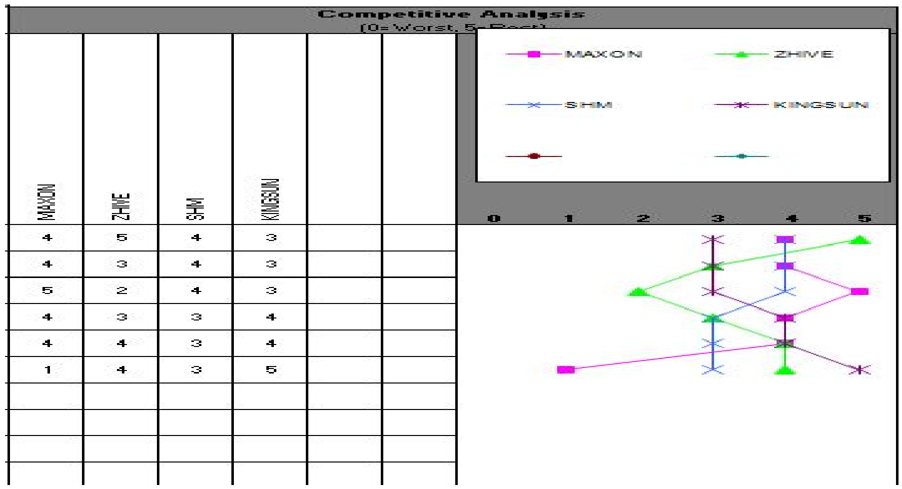


Figure 17: Competitor Analysis

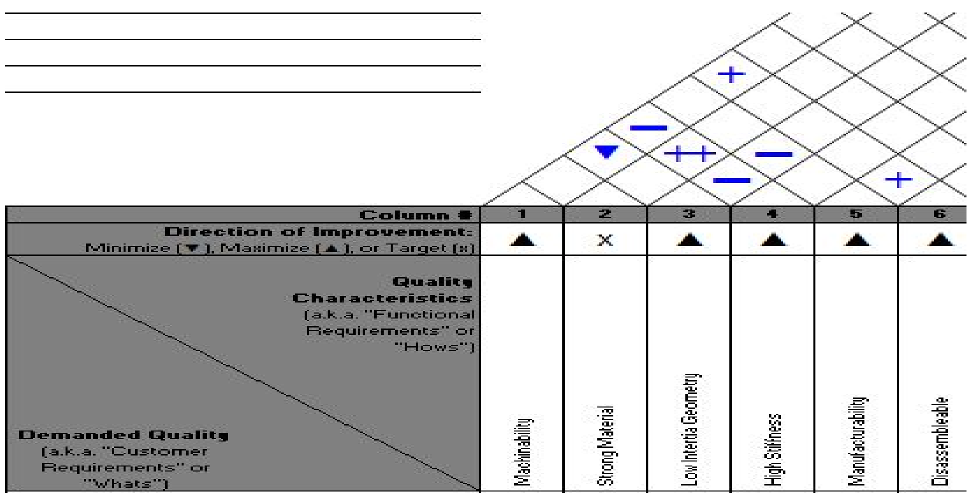


Figure 18: Technical Correlations

9 Design for X

The design process was influenced by three major factors: quality, cost and manufacturablility. Of the designs presented in the critical design review, cost and quality were factors that eliminated the ribbed shaft and retractable knife concepts. Manufacturablility was a major factor involved in the development of the tube and spoke design. As the tube and spoke design is optimized, cost and quality will become more important, as the design will be redesigned to decrease cost and increase quality.

Design for Quality At the time of the critical design review, three new designs for the lower shaft were proposed, a ribbed shaft, a tube with spokes, and a shaft with a retractable knife. The ribbed shaft, pictured in Figure 19, was removed from consideration due to its poor quality. Though the inertia of the shaft would be reduced, increasing quality, the stiffness of the shaft would be reduced by an unacceptable margin. The quality of the tube and spokes design, Figure 21, was considered to be good, because the inertia would be reduced, and the stiffness could be maintained by the positioning, and shape of the spokes. The highest quality design from the critical design review was considered to be the retractable knife or "Always Be Synchronous" design, pictured in Figure 20. The "Always Be Synchronous" design failed for reasons other than its quality, mainly its high cost and low manufacturability. Once the tube and spoke design was selected, the technical aspects of the design which increase quality were determined to be stiffness and inertia, a high quality design having low inertia and high stiffness. To achieve a low inertia, a thin outer tube was selected, currently the design incorporates an 0.5" thick outer tube. To keep the inertia of the spokes low they are spaced 7" apart, and are only 1.5" thick. In the future the spokes will be optimized further to reduce weight and inertia. The stiffness of the spokes, per unit volume, is much high than that of a simple shaft. Due to the fact the the spokes deform in multiple ways, such as shear and axial stresses, they absorb far more distortional energy per unit deflection than a bar in pure torsion. The stiffness of the spokes can be increased further by optimizing their shape, which will increase the quality of the mechanism because the higher the stiffness, the less deflection there is at the knife's edge, leading to a cleaner cut. The quality of the design is limited however by other priorities, such as cost and manufacturability.

Design for Cost The cost of the design a critical consideration due to the fact that designs which are not economically viable are not approved for manufacture. The primary costs associated with the tube and spoke design are the manufacturing costs. This is why the manufacturing processes used to create the design are of interest. The material cost is not necessarily lowered by a lighter weight design; the parts which make up the design must be machined from larger materials. The design also has a larger number of parts than the previous design, which increases the manufacturing costs. The cost of the design is effectively lowered however by the fact that a lighter weight and lower inertia design does not require as powerful a motor, which is far less expensive than more powerful motors. The manufacturability of the design is the primary factor associated with the cost of the design.

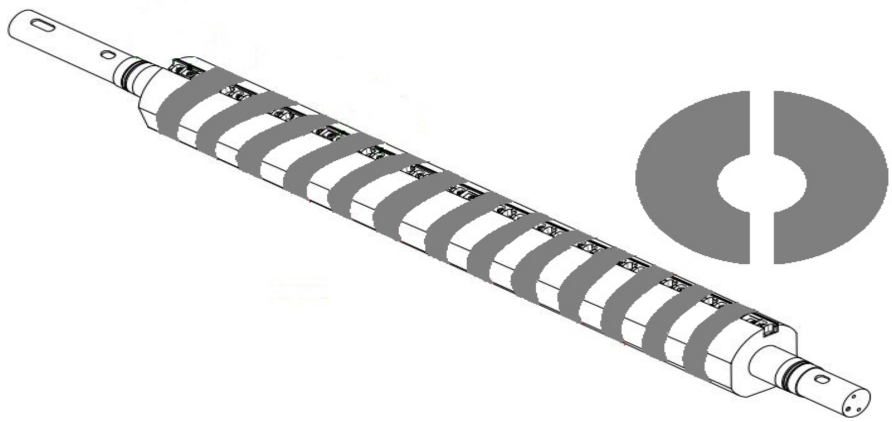


Figure 19: Low cost, low quality ribbed shaft

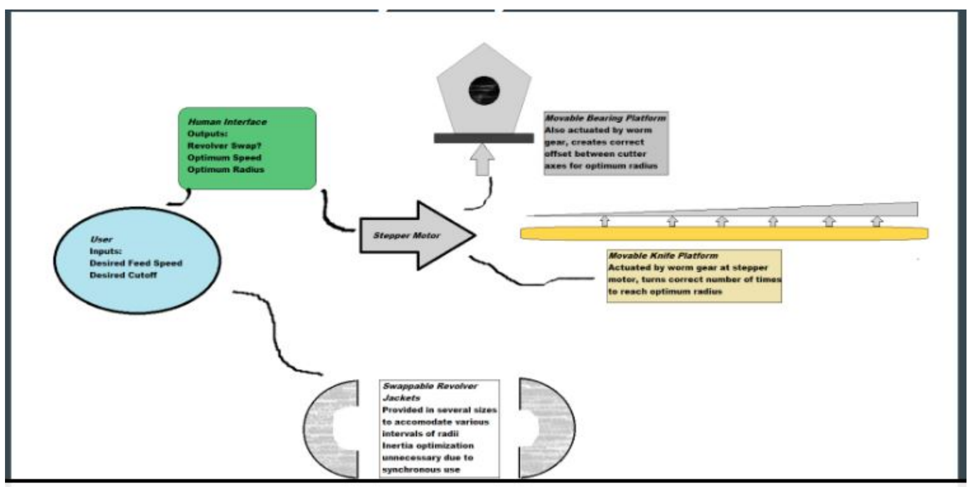


Figure 20: "Always Be Synchronous"

Design for Manufacturabiliy Manufacturability has played a major role in the development of the tube and spoke design. The tube, the spokes and the slugs which hold and balance the knife, all have been affected by challenges associated with the manufacturing process. The tube itself has been designed into two sections, due to the manufacturing challenges of mating the other parts inside a hollow tube. The design of the slugs was influenced by this design decision, the slugs were designed to not only hold the knife, but to hold the tube hemispheres together. The spokes were redesigned as well to make them easier to machine and to mate to the inner shaft. Originally, each spoke was to be machined and mated separately, however this design was altered to allow two spokes to be made together in a pair, with a hole through the middle so the pair could be mated to the shaft. Their thickness is such that all 9 spokes can can be milled out of a thick plate, then sawed into individual spokes; in addition, MAXSON and Spin to Win are investigating the possibility of plasma cutting them. The revised spoke design decreases manufacturing time and cost. As the design is revised in the future, the manufacturing process will be of utmost importance.

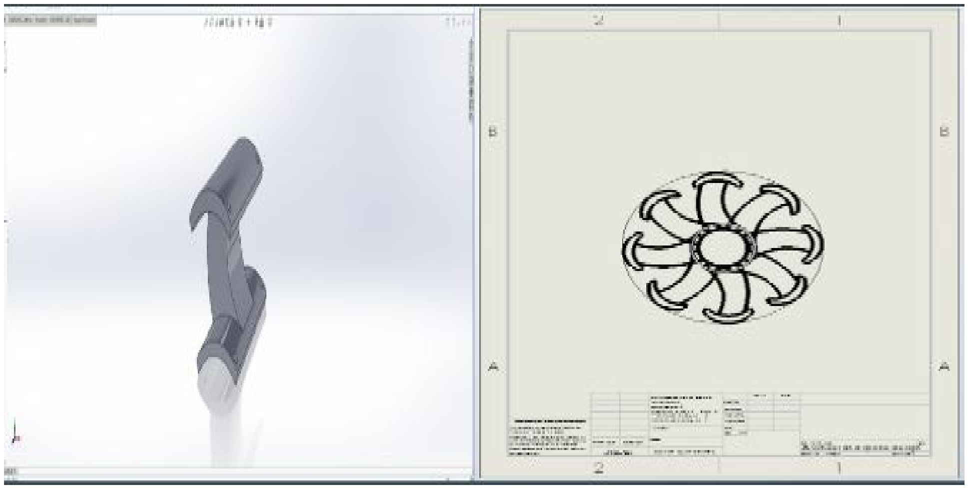


Figure 21: Original Spokes

Figure 21 demonstrates the concept which has the best balance of the design parameters of quality, cost and manufacturability. These parameters have and will continue to shape the design.

10 Project Specific Details and Analysis

10.1 Project focus: gears versus cutters versus all of the above

MAXSON's original vision for the project involved an all-of-the-above approach to redesigning the MD Sheeter cutting section. This included a primary focus on designing a gear reduction for the system, which was believed to show the greatest promise for lowering system costs, together with an any-and-all improvements possible examination of the rest of the system.

Spin to Win determined in the weeks leading up to the Critical Design Review presentation that it was impossible to perform redesign work on every aspect of the system, looking at each one independently for optimization, and meet the Capstone course requirement of carrying out a "design activity" to its completion, including the design of a novel solution to an engineering problem.

While the gear reduction involved the greatest promise to improve the system, Spin to Win determined that it was not appropriate as a design problem for the Capstone course, as gear design is a solved problem, with hundred-year-old optimal solutions. This "design" activity boils down to inputting values into a known tooth stress equation and arranging the resulting geometry in a space. This is not necessarily an easy or trivial problem, but its solution does not constitute the creation of anything novel in engineering, and the richness of the process of design and redesign is lacking if spent only on this problem.

Since the solution of this problem brings great benefit to the project sponsor, Spin to Win decided not to abandon it. However, in order to meet the Capstone course objectives, the team decided to focus its primary efforts on the cutter optimization problem, which provides a much richer design space.

Spin to Win determined that the project's definition was in its own hands provided all objectives were met.

10.2 Electrical Team fog of war

As noted in Section 2, Project Planning, the original model for the cutter redesign involved the passage of information back and forth between the mechanical and electrical teams. The mechanical team would provide inertia values to the electrical team, who would then provide motor specs to the mechanical team, who would then design and situate a reduction in the available space.

It was discovered early on that this model suffered from some problems. For example, the system inertia was already known by MAXSON, and its calculation by the mechanical team added little value to the project. Meanwhile, the mathematics involved in turning an inertia value into a correctly sized motor were unknown to the electrical team, so the mechanical team had to generate torque and RPM curves to aid the motor selection. This process was not without error on the mechanical team's part, and no such detailed curves of torque versus time were in use by MAXSON. In fact, MAXSON does not even possess a known number for peak torque of the cutter system; instead, MAXSON appears to operate on an experiential model, measuring the likelihood of future success off of what was known to succeed in the past.

The result is that the process became stymied on the electrical team's side, and at one point a directly coupled motor was under consideration that made a gear reduction impossible. In fact, this motor may still be under consideration.

As such Spin to Win determined that the only way to move forward was to make best assumptions about geometry and motors and focus on those components that it could optimize independent of what happened elsewhere in the system, such as the cutter geometry. In this way the team could build a contribution and add value to MAXSON's product without getting bogged down in the inter-discipline fog of war.

11 Detailed Product Design

11.1 Detailed part descriptions

The Spin to Win team brainstormed, came up with ideas, and after much deliberation the team settled on a single concept. The team went with redesigning the top cutter and bottom roller of the sheeter. The single concept that the team went forward with was a design that uses the idea of putting spokes in between a circular object to add rigidity and stiffness to the outer shell like, for example, a bicycle wheel. This design was the best of both worlds in that the total inertia of the system could be lessened and the shafts would be able to retain their stiffness. In order for the design to meet the specifications laid out in the design specs more parts needed to be added to the concept

The redesign of the bottom cutter, which the team calls the tube and spoke design was made to give the new system less inertia as well as give it adequate stiffness to make up for the loss in inertia. The cutter from the previously existing design needed to be thinned in such a way that it would lessen the inertia of the shaft, but not at the cost of the stiffness. Because of this the team proposed a smaller thinner shaft with an insert that could connect it to a larger shell. The insert had to be designed in a way that it would grant us the stiffness we needed as well as add less material to the design as a whole. The outer shell of the design had to be thin for the inertial reduction, but also had to have a good curvature, so that we could increase or keep consistent with current design our feed stability for the new design. In order for the outer shell to be assembled it needed to be split in half as well as hold a gap for the blade hardware to be placed. Because of this the design needed an insert of some kind that could attach and connect the two pieces of the outer shell together as well as have a type of trough to hold the existing blade hardware. From these ideas and considerations a 3D model was generated and the concept came to life.

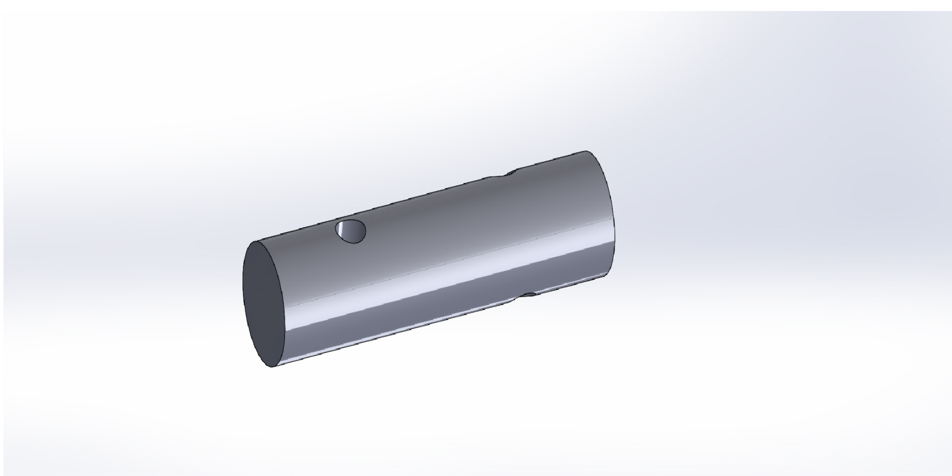


Figure 22: Center shaft

The Tube and Spoke design is best described from inside to out starting with the first part of the design, the center shaft. The center shaft has a tapered design down to a 2" diameter as seen in Figure 22. The tapered design is to help with the inertial reduction as well as allow for space for the spokes to be inserted. The center shaft can be incorporated into the current cutting system, thus the reason for the tapering effect. Holes will be drilled in the shaft at two different angles in order to bolt the spokes to center shaft.

The next part to be addressed is the main concept of the design, the spoke, pictured in Figure 23. There will be 9 sets of spokes to be installed and manufactured in pairs. The spoke has a hole down the center in order for it to mate with center shaft. The holes that are to be drilled in the center shaft will line up with the holes drilled through the circle part of the spoke. The spokes each have extra material on both sides of where the bolt will be threaded through the spoke to connect it to the center shaft. This extra material was added to help the bolt by acting as a flat washer of sorts. When the added material acts like a washer it helps to distribute the pressure of the bolt evenly over the surface it is bolted to in order to reduce the damage done to the bolt itself. It also helps to reduce the chance of the bolt loosening over time due to the surface being uneven. The added material makes sense because the spoke is circular and the round surface would be unevenly matched with the bolt head. The spoke part itself is curved to attach itself with the curvature of the tube or outer shell. The curvature of the spoke plays another role as well. The pairs of spokes are curved against the direction of rotation to help maintain stiffness as the tube rotates. Curvature of the spokes was also tested versus straight spokes, and the curved ones proved to be stronger as shown in figure. At the ends of the spokes the original idea was to have flanges that would be bolted to the tube from the outside. Instead the flanges were omitted and a threaded hole was put into each end of the pair of spokes. With this addition to the design the spokes can be more easily assembled to the outer shell by way of threaded inserts.

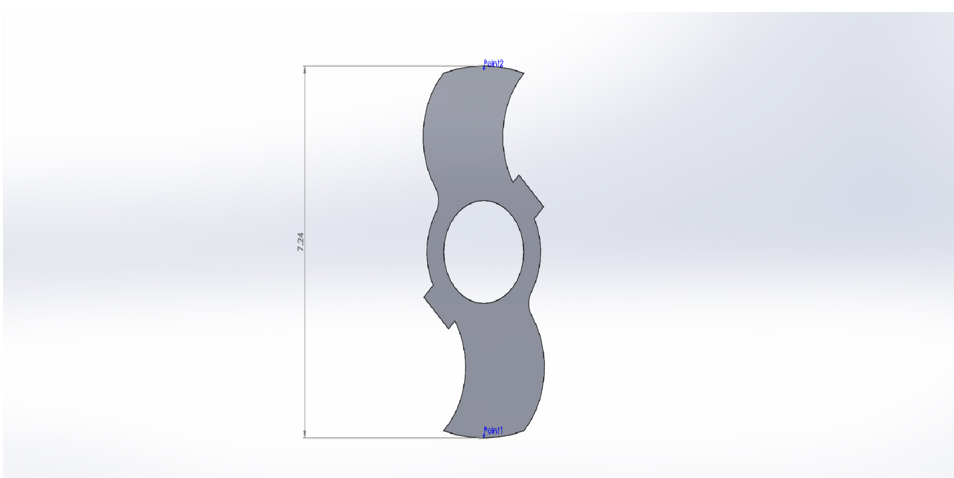


Figure 23: Spoke design

The next component that needs to be addressed is the tube or outer shell of the Tube and Spoke design, Figure 24. The existing design for the shaft is a di-lobe or tri-lobe design. This design isn't very circular and has spaces and nooks that paper could fall into to disrupt the feed stability. So the team decided that the outer tube needed to be more circular to account for the spaces and nooks in the previous design. Thus, a cylindrical design approach was taken. The tube was designed to be thin for inertial reduction, but also to give space to incorporate the spokes the main component of the design. The tube was designed to have a helical cut-out in it and an array of mounting holes for the spokes to align with and attach to. These mounting holes are the threaded inserts that were previously talked about in the above paragraph. The purpose of the helical cutout is to accept the slugs which will be discussed in the next couple paragraphs.

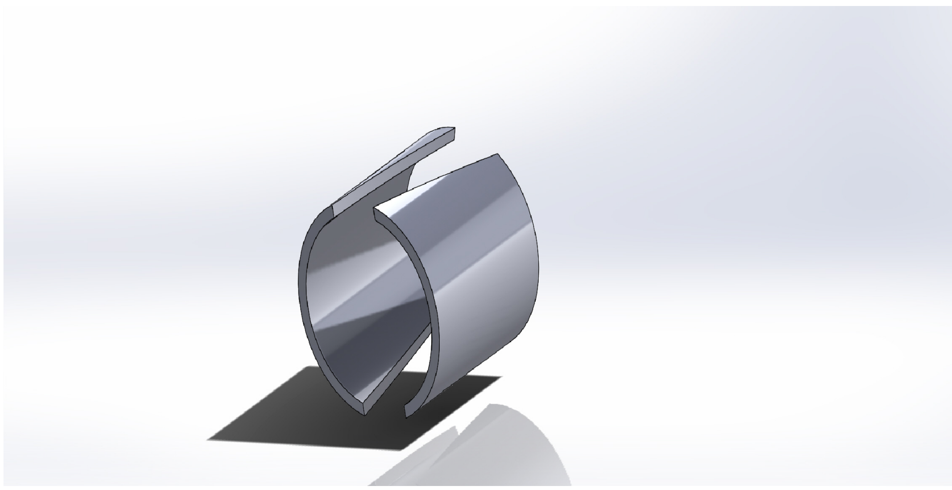


Figure 24: Tube or outer shell

The last components and probably the most difficult to design were the slugs, Figure 25. The top slug has a helical twist throughout the entire member including the trough going down the center. One of the difficulties with the slug was how it was to be attached and assembled. After a few days of discussion the team decided that providing inserts in the slug would be the best method to attach and assemble them to the outer shell, which can be seen in figure blank. The top flanges on the top slug were curved to help with feed stability. Since they were curved for feed stability that made it impractical to put holes in the top for bolts to secure the slugs in place to the tube. Because of this the slugs are going to be spot welded to the outer shell along the tube. The trough was designed into the top slug in order to allow for the mounting of the existing blade hardware without modifying the current cut edges. The large amount of material under the trough will be to hold the bolts that will lock the blade hardware into place in the slug. The bottom slug is a little different than the top slug in the fact that the space for trough has been filled in. The bottom slug also acts a counterweight for the top slug so the system stays balanced. Like the top slug, the bottom one has curved bottom flanges to help with feed stability. On the bottom slug, however, the curvature is more pronounced in order to provide a closer curvature to the outer shell. The closer the curvature is to the outer shell the less chance the slug has to impede the feed and cause jams in the material being fed through.

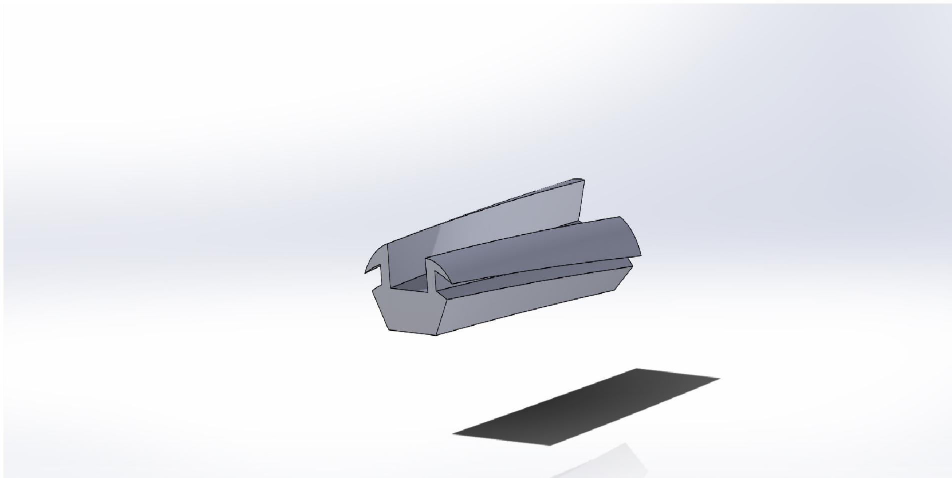


Figure 25: Top slug Holding Hardware

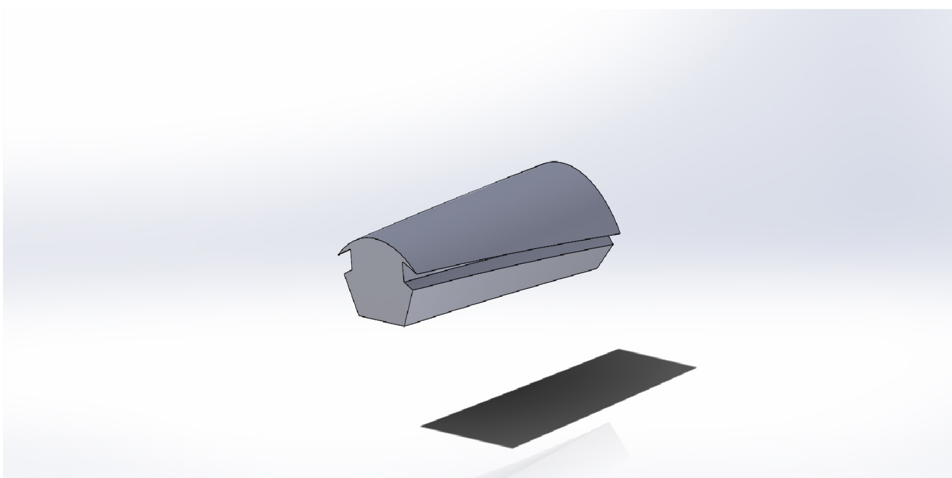


Figure 26: Bottom slug No Hardware

11.2 Machining process and Bill of Materials

The inner shaft is to be lathed from a bar which is long enough and thick enough to attach into the existing bearing housings, which requires a diameter of 3.118". So a 3.25" bar will first have the 67" length of the shaft sawed from it. Then the end pieces will be lathed to 3.118", and the center shaft lathed to 2". Holes for the $\frac{3}{4}$-16 main bolts will be tapped into the shaft at 5.75", 12.75", and so on, every 7".

The spokes begin with a 2.5" x 12" x 24" plate from which their outline is milled, using 24" as the depth, 2.5" as the width, and 12" as the height. The center hole is then bored, and the resulting piece is sawed into 1.5”-thick pieces. If plasma cutting is used in the future, a different thickness and process will be appropriate.

The outer tube requires a 7”-ID pipe, which is sawed into two pieces along the helix angle. Methods to achieve this sawing other than by hand are being investigated.

The slugs are milled from long rectangular bars.

Assembly begins with the central shaft propped up in a position where it is free to rotate. Spokes are slid onto the shaft and aligned with their holes. Bolts are inserted and nuts are threaded on, using locking compound, and torqued to a half-twist beyond snugness for a small preload. Next the shaft and spokes are rotated so that the first tube half may be laid on from above; once in place, threaded rods are inserted into the aligned holes to secure it. After the first tube half, the shaft is rotated 90 degrees, and the slugs are laid with one open slot resting on the tubes. In this position the first tube half is on the bottom and the slugs are resting, side-up, on the sides. Finally the second tube half is laid onto the upraised slots on the slugs and mated to the spokes with threaded rods.

The Bill of Materials for the entire assembly is given in Figure 27.

Bill of Materials

Item	Quantity	Manufacturer	SKU	Material	Volume (in^3)	Mass (lbm)	Cost per Pound	Cost
3.25" diameter x 108" A36 Hot Rolled Round	1	Metal Supermarkets	HR314	A36 Hot Rolled Steel	895.9429549	254.4477992	1.520036727	386.77
2.5" x 12" x 24" A36 Plate	1	Metals Depot	P1212	A36 Hot Rolled Steel	720	204.48	1.647691706	336.92
8" SCH80 72" Structural Pipe	1	Metals Depot	T5880	A500 Black Steel	918.9158512	260.9721017	2.433057004	634.96
2.5" x 5" x 72" A36 Plate	2	Metals Depot	SQ1212	A36 Hot Rolled Steel	450	127.8	4.077856025	521.15
5/8"-18 x 3.25" Alloy Steel Socket Head Screw (pkg)	1	McMaster-Carr	90044A291	Black Oxide Alloy Steel	N/A	N/A	N/A	10.02
5/8"-18 Grade 8 Locknut	9	McMaster-Carr	90630A165	Grade 8 Steel	N/A	N/A	N/A	3.29
5/8"-18 x 10" Medium Strength Steel Threaded Rod	1	McMaster-Carr	98750A493	A193 Grade B7 Steel	N/A	N/A	N/A	6.62
								2420.88

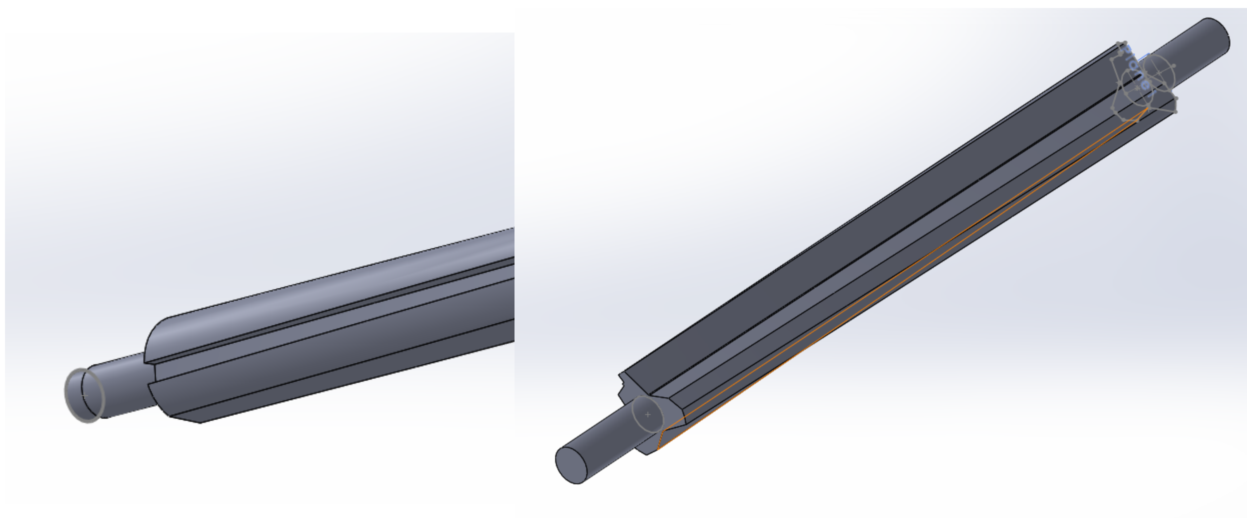
Figure 27: Spokes Assembly Bill of Materials

12 Engineering Analysis

12.1 Calculation of system torques and inertias

System Inertia Spin to Win was requested to calculate the cutter system's rotational inertia for use in all further work. The team chose to do this in two different ways and watch for convergence: automatic calculation in SolidWorks and manual calculation by integration.

First, a CAD drawing of the cutters was imported into SolidWorks, and mass properties for steel were provided. These models are shown in Figure 28.



(a) Bottom cutter

(b) Top cutter

Figure 28: SolidWorks models of existing MAXSON cutters

Meanwhile, the inertia of all components was calculated through an integral. For a uniform cross-section of length l, the mass moment of inertia is given by the equation

$$I = \rho l \int r^2 dA \tag{1}$$

where r is the distance of each small unit of mass from the desired reference axis. Each cutter was separated into zones of equal cross-section and separated into annular slices of 0.001" thickness, each with an equal value of r. Each slice had an area calculated based on the amount of material present in it (based on cutter geometry) and made a discrete contribution to the system inertia. The total inertia was summed from each of these tiny contributions as a Riemann sum. The resulting values of inertia agreed within about 5% with those provided by SolidWorks, so the values were held with confidence.

The calculated inertias of each component and the total system inertia are given in Table 3.

Component	Inertia ($lb - in^2$)
Top Cutter	3,448
Bottom Cutter	7,510
Gears	1,469
Other	460.8
Total	12,888

Table 3: Component and system calculated inertias

Whether because of unknown components of the system, errors in estimations, or realities that do not agree with the drawings, the actual cutter system inertia is known to be larger than Spin to Win's calculated value. Subtracting the inertia of a coupling which will not be present in the redesigned system, experiments done by a MAXSON vendor place the system inertia at 14,388 $lb - in^2$. This value was employed as a baseline.

Cutting Cycle The sheeter cutting cycle requires that the blades be moving at the feed speed for about 5 degrees before and 15 degrees after a cut. Meanwhile, while not cutting, the blades must have an average rotational speed such that the correct amount of time passes between cuts. For a cutoff of 40" and a feed speed of 1,000fpm, and a cutting radius of 4.25", the basic parameters of the cutting cycle are determined and provided in Table 4.

Angular Speed During Cut ($\frac{rad}{sec}$)	47.06
Average Angular Speed While Not Cutting ($\frac{rad}{sec}$)	31.4
Time per Revolution (sec)	0.2

Table 4: Cutting cycle basic parameters

However, the average angular speed while not cutting is only a mean, and that time can be spent in any range of velocities, so long as the time elapsed before the knife returns for the next cut is correct. This range of velocities is called a "cam curve" in industry, and its shape is generally sinusoidal, which provides gentle transitions and low jerk.

The following equation was solved computationally in order to find a quarter-sine curve whose amplitude, A, results in a mean equal to the required mean velocity for ranges of θ from 0 to 340 degrees, with a period of 680 degrees. It defines that the mean value of the function must equal 31.4, the mean value being the integral of the product of the function and its argument θ, with the function being the aforementioned sine curve of appropriate amplitude A:

$$\int_0^{\frac{17\pi}{9}} A\sin(\frac{34\pi}{9}\theta)\theta d\theta = 31.4 \tag{2}$$

A equals 7.82145, and the resulting curve, with RPMs plotted as a function of θ in degrees, is given in Figure 29.

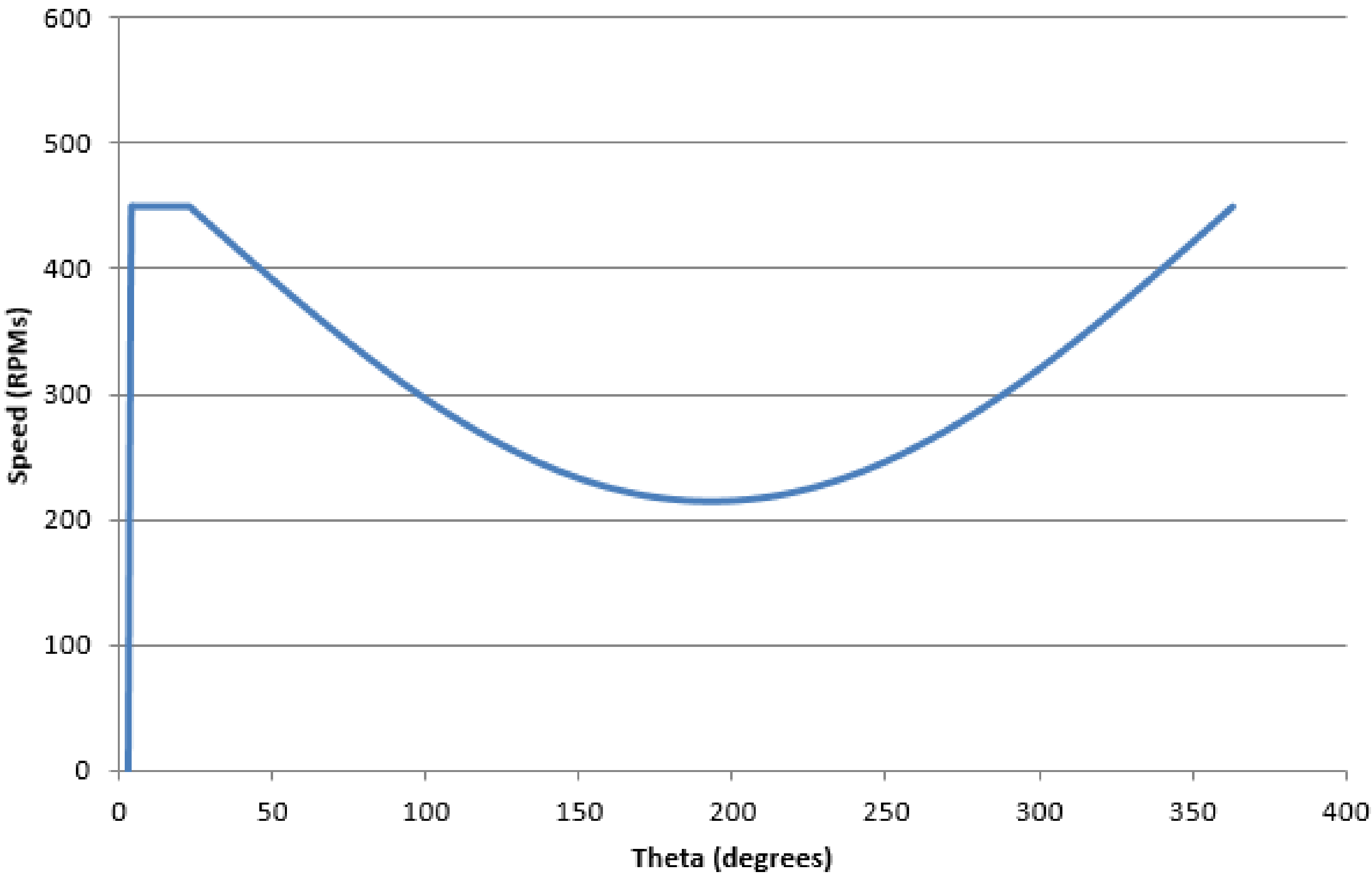


Figure 29: Sinusoidal cam curve for 1000fpm/40” cycle

This curve defines the angular velocity at each angular position in the cutting cycle. From it it is possible to compute the angular acceleration, as for each 1-degree step, both the time for the step and the required change in velocities is known. The angular acceleration can then be found stepwise across the entire cycle. Then, by Newton's second law applied to rotation, the required torque is the product of the inertia and angular acceleration:

$$T = I\alpha. \tag{3}$$

Since all of these values are known, the torque can be solved for. The peak torque for the system is 21,100 $in - lb$, and the torques required at each step of the cycle in $lb - ft$ are found in Figure 30.

This system torque is that required to drive the load of the cutting section through a 1,000fpm/40” cutoff cycle. The actual torque supplied by the motor will be more or less depending on the motor rotor's inertia and the presence of a gear reduction, and the torque felt by any component of the cutter system is proportional to its own inertia.

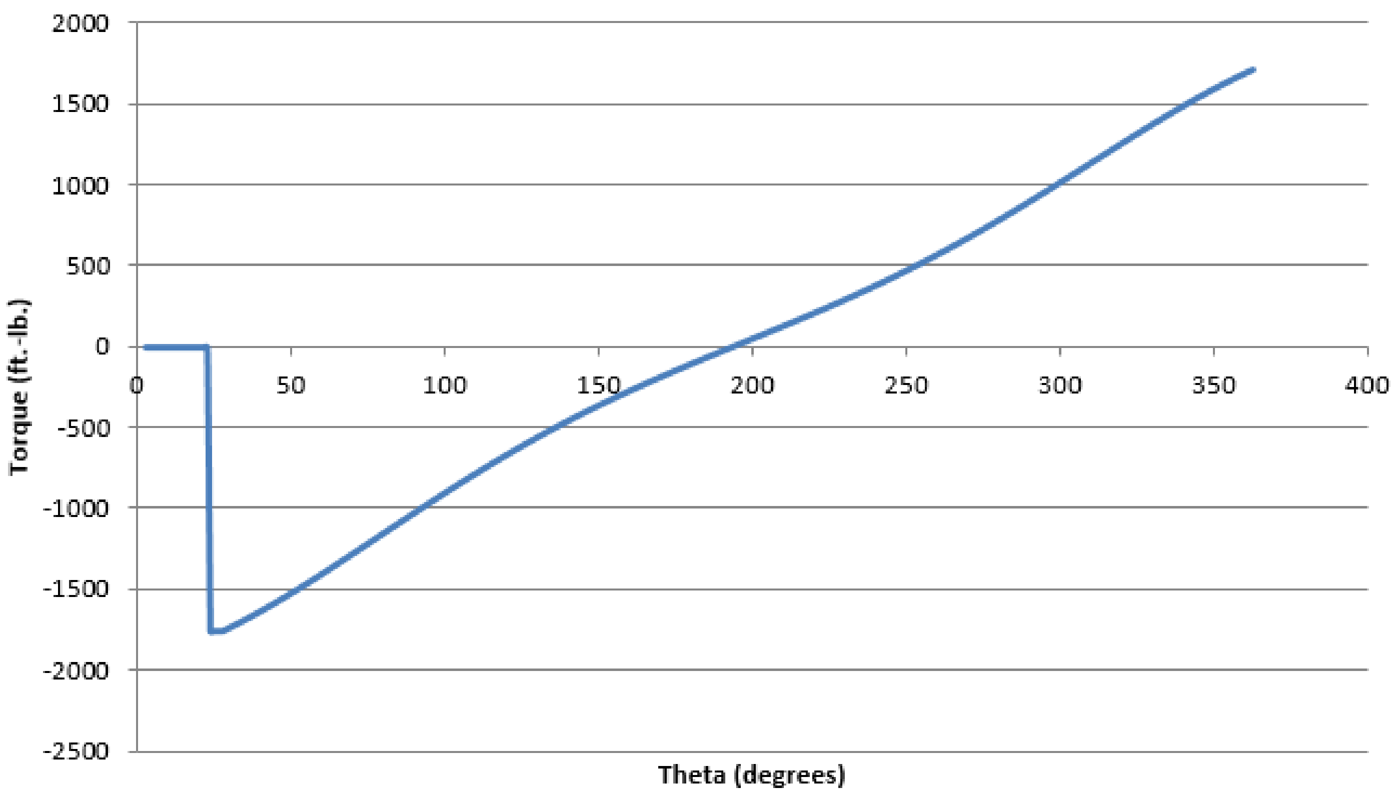


Figure 30: Torque requirements of cutting cycle

Spokes assembly mass and inertia The spokes assembly consists of a center shaft, two halves of an outer tube, two slugs, 9 spokes, 9 main bolts, and 18 threaded inserts where the spoke arms join the tube. In addition there are two shafts that hold the assembly and insert into the existing bearings. The bolts are best treated as simply part of the material they are threaded into. The mass and inertia of each component may then be computed from its volume, density, and distance from the center axis.

Component	Mass	Inertia
Central Shaft	59.8 lbm	29.9 $lb - in^2$
Bearing Shafts	58.5 lbm	71.1 $lb - in^2$
Outer Tube	171.2 lbm	2,419 $lb - in^2$
Slugs	132.2 lbm	1,619 $lb - in^2$
Spokes	45.1 lbm	273.9 $lb - in^2$
Totals	466.9 lbm	4,413 $lb - in^2$
MAXSON Cutter	860 lbm	7,289 $lb - in^2$

Table 5: Spokes assembly component masses and inertias

The total inertia discount is 2,876 $lb - in^2$, which is about 40% of the existing cutter and 20% of the entire system.

12.2 Pinion strength analysis

Spin to Win has also developed a pinion gear reduction between the motor and load in order to lower the torque requirement of the motor, which is of great economic benefit for an electric motor.

The design for this gear began with geometric constraints. The existing motor shaft is 65mm in diameter, and in order to affix the pinion to it a radial compression device known as a Ringfeder locking assembly [7] is to be employed. The thickness of each of these must be added to the shaft, followed by a root area for the pinion, followed by the teeth. Meanwhile the pitch radius of the mating gear is 4.125". For a root area of twice the tooth height, the minimum reduction that can be fit onto a motor shaft is then 1.9:1.

Next the tooth stresses must be analyzed. All of the torque that drives the load passes through the pinion, so it sees all torques found in Figure 30. The strength of the teeth was measured using the American Gear Manufacturer's Association (AGMA) equation [8]:

$$\sigma = F_t K_o K_v \frac{P}{b} \frac{K_s K_m}{J} \tag{4}$$

A description of each of the terms in Equation 4 together with values used in this problem is given in Table

Term	Description	Value
σ	Bending stress at tooth root	
F_t	Transmitted tangential load	4,971 lbf
K_o	Overload factor due to shock	1.5 (light-moderate shock from momentum shifts)
K_v	Velocity factor at pitch circle	1.559[3]
P	Diametral pitch	9.4188
b	Enaged length	variable
K_s	Size factor	1
K_m	Mounting error factor	1.3 for high precision
J	Geometry factor	0.55

Table 6: AGMA equation parameters

MAXSON's current gear has a 2.5" engagement and uses a Rockwell C 60 case hardened steel body. These gears have an allowable stress from this equation of 55ksi, while the above equation predicts 97ksi, meaning failure due to fatigue will likely occur before 10^7 cycles. In order to have an acceptable factor of safety for a full fatigue life, a longer engagement is necessary, with the factor of safety versus engagement length plotted in Figure 31.

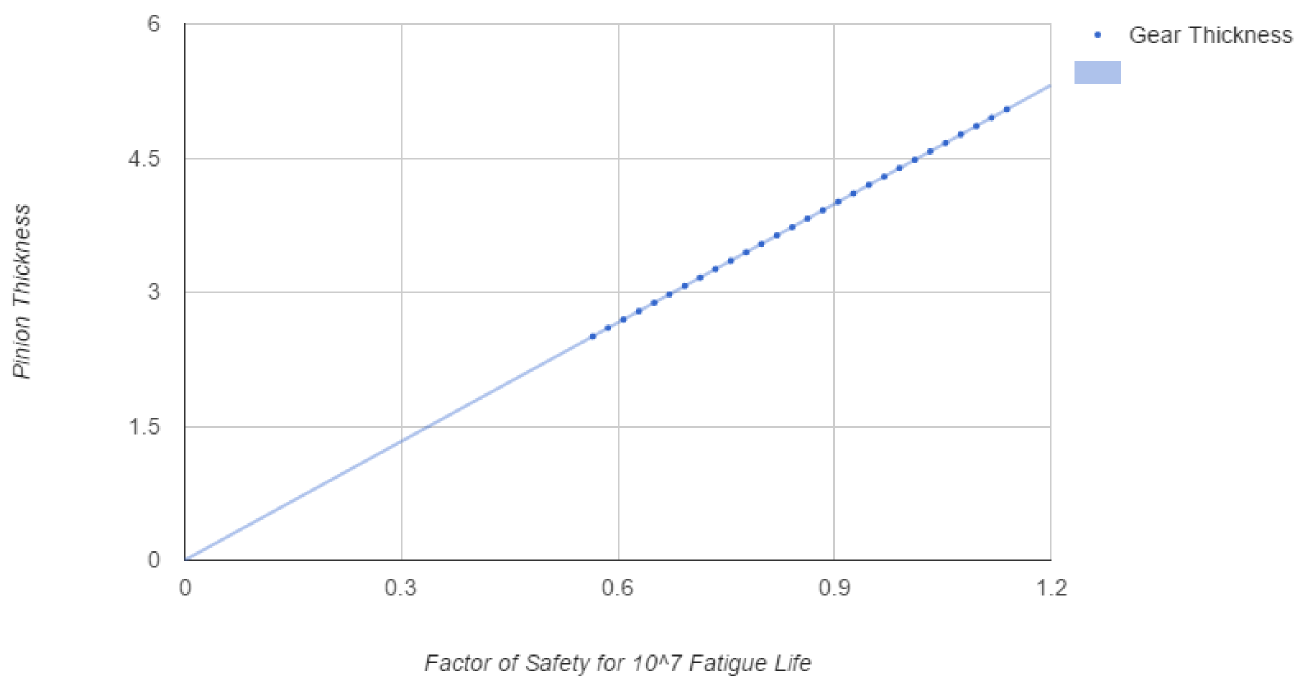


Figure 31: Pinion factor of safety for fatigue versus engagement length

As a result of these analyses Spin to Win recommends increasing the driver side gear width to 5" and using a 1.8:1 reduction. This, however, requires a redesign of the bearing housing, which is currently in development.

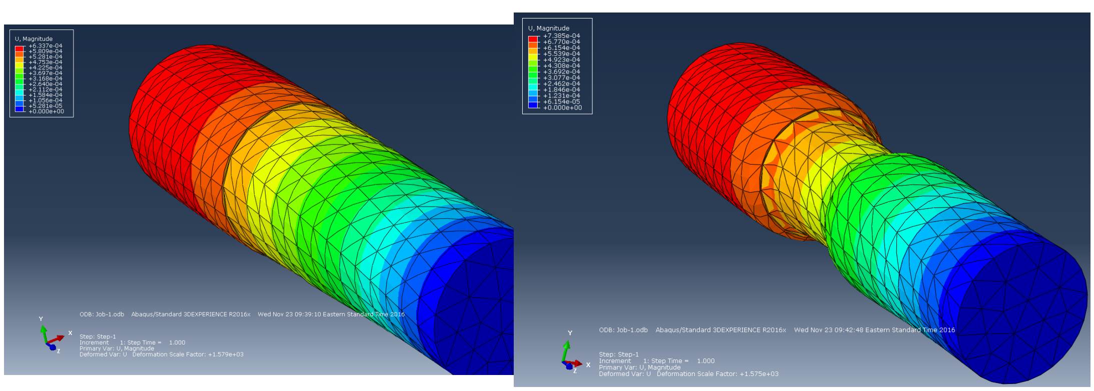
12.3 Main bolts strength analysis

The torque transmitted through the spokes assembly is that portion of the whole system torque which is proportional to its inertia according to Equation 3. When the system uses the spokes assembly, its required torque is 16,700 $lb-in$, and the portion of that seen by the assembly is 7,300 $lb-in$.

When the assembly accelerates, the different inertias of the spoke and central shaft result in transverse shear on the main bolts. The peak magnitude of this shear force is the assembly's applied torque divided by the moment arm where the force acts, or $V = \frac{7300}{1.25} = 5,840 lbs$. This shear force is distributed across the tensile area of the bolt, for an average shear stress of $\tau = \frac{5840}{0.373} = 15,657 psi$. The peak shear stress is four thirds this [9], or 20,875 psi.

This, however, is if a single bolt carries the entire load. In reality bolts nearer the application of forces will carry a greater portion, with the exact amounts being very difficult to determine theoretically. An extremely conservative method leaves the entire load on one bolt, but Spin to Win has employed 2.5 bolts carrying the load as an estimate. In this case the maximum stress is 8,350 psi.

Each bolt must be loaded with a preload which eliminates relative motion between the spoke and shaft. In addition, finite element simulations performed by Spin to Win have suggested that compressive stresses on a torqued member improve its overall stiffness, even though those stresses reduce the local cross-sectional area, as seen in Figure 32.



(a) A small bar subject to torque

(b) A small bar torqued and compressed

Figure 32: Compressive stresses' effect on torsional stiffness

These preloads raise the mean stress experienced in the cycle of stresses on each bolt, which has a deleterious effect on fatigue life [10]. The impact is positive when the joint loading is tensile, but for shear loads the stress due to preload is a hindrance. However, Wallaert and Fisher [11] report a small increase in shear yield stress for joints with a half-turn preload before strengths drop off.

Based on fatigue data given by Hudgins and James [10] for a 90ksi UTS steel, for a stress amplitude of 8,350 psi, a mean stress of about 60 ksi is acceptable, which permits a preload of about 20,000 pounds. A grade 8 bolt has a 150ksi UTS [9] UTS, which in theory permits a larger factor of safety, but the details of shear fatigue are not well documented in general. Shigley's [9] suggests fatigue numbers for torsion where the shear stress is checked against an endurance limit defined around 0.67 the UTS of the material and an extra modifying term of 0.59 as k_c. Following these numbers, for a grade 8 bolt,

$$S_e' = (0.67)(0.5)S_{ut}k_c(150,000) * (0.67) * (0.5) * (0.59) = 29.6ksi \tag{5}$$

Against this value for a shear endurance limit, the factor of safety in fatigue assuming at least 2.5 bolts share the torsional load is given by

$$FoS = \frac{29.6ksi}{8.35ksi} = 3.5 \tag{6}$$

In the worst case that a single bolt must carry the load, the factor of safety is 1.4.

Josi, Grondin, and Kulak [12] report endurance limits for pure shear ranging from 1,500 to 15,000psi for tests involving a shear splice, in which a sandwiched plate is pulled out of a bolted connection involving multiple bolts. Farfield stresses of about 13,000psi saw infinite fatigue life with bolts made of mild steel (60ksi yield); however, a huge range of factors beyond those that can be discussed here contributed to failures. Of particular note is that the firm compaction of the bolts in the spokes assembly inside the assembly, with compressive stresses on all sides when the bolt is bent, will improve fatigue life, as surface compression is known to prevent the opening of cracks. These findings support the likelihood of an infinite fatigue life for the spokes assembly's main bolts, but testing in the spring is desired.

12.4 Spokes assembly torsional loads and deflections

Loadings The spokes assembly is a 67"-long piece that rotates between two shafts, one 10", the other 27", connected to two gear meshes. The primary stress on the assembly is torsional, as when the system accelerates, torques must be transmitted from the motor at one end of the cutting section to the gears at the opposite end.

However, the spokes assembly does not behave like a simple beam when subjected to a force, as it is an assembly composed of multiple, coupled pieces. Each component must be analyzed separately, as each will deform according to the loads placed on it, and those loads are based on where contact exists with the other parts.

For simple torsional loading, one end of a beam is fixed and the other end torqued. If this condition exists for the spokes assembly, the resulting system of loadings is given in Table 7, with free body diagrams illustrating the loads in Figures 33, 34, 35, and 36. The outer tube and slugs have been lumped into a single unit to simplify calculations, which is reasonable since in final assembly they will be welded and connected as one continuous piece of material. Detailed calculations for the below values are found in Section 16.2 of the Appendix.

Component	**Loading**	**Resisting Area Type**	**Resisting Area Value**
Center Shaft	Torsion	Polar moment of inertia, J	1.57 in^4
Outer Assembly	Torsion	Polar moment of inertia, J	207 in^4
Threaded Inserts	Trans-axial shear	Bolt tensile cross-section, A	0.373 in^2
Threaded Inserts	Along-axial shear	Bolt body area, A	0.625 in^2
Main Bolts	Trans-axial shear	Bolt tensile cross-section, A	0.373 in^2
Main Bolts	Tension	Bolt tensile cross-section, A	0.373 in^2
Main Bolts	Along-axial shear	Bolt body area, A	2.03 in^2
Main Bolts	Torsion	Polar moment of inertia, J	2.06 in^4
Spoke Arm	Bending	Flexural moment of inertia, I	1 in^4
Spoke Arm	Shear	Cross sectional area, A	3 in^2
Spoke Body	Torsion	Polar moment of inertia, J	59 in^4

Table 7: Spokes assembly load conditions and resistances

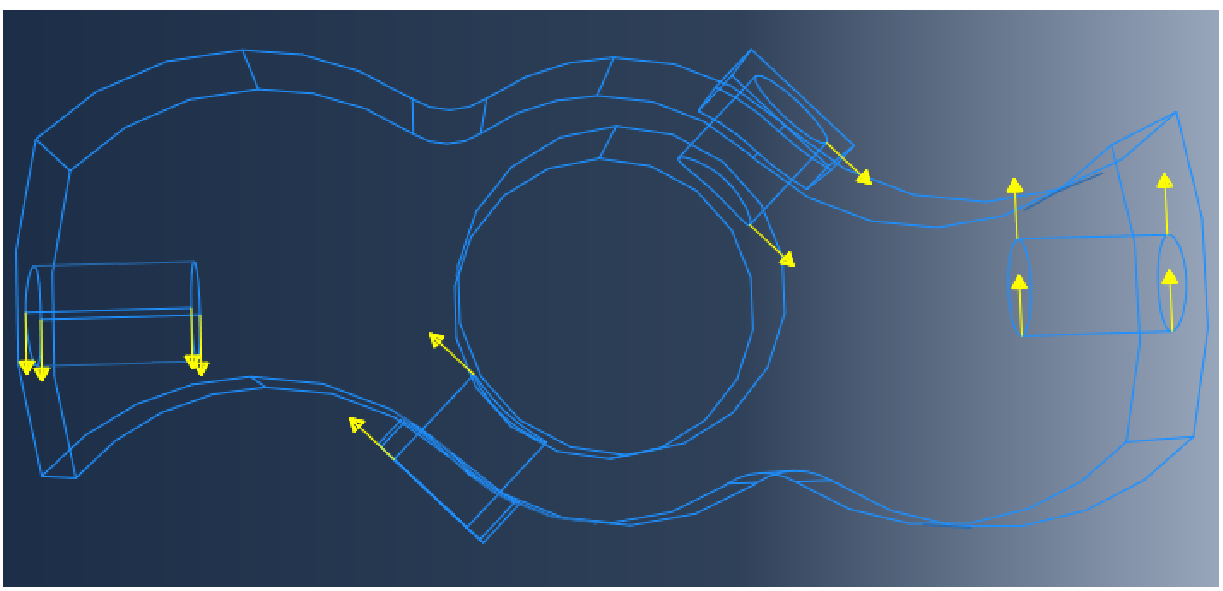


Figure 33: Spoke free body diagram

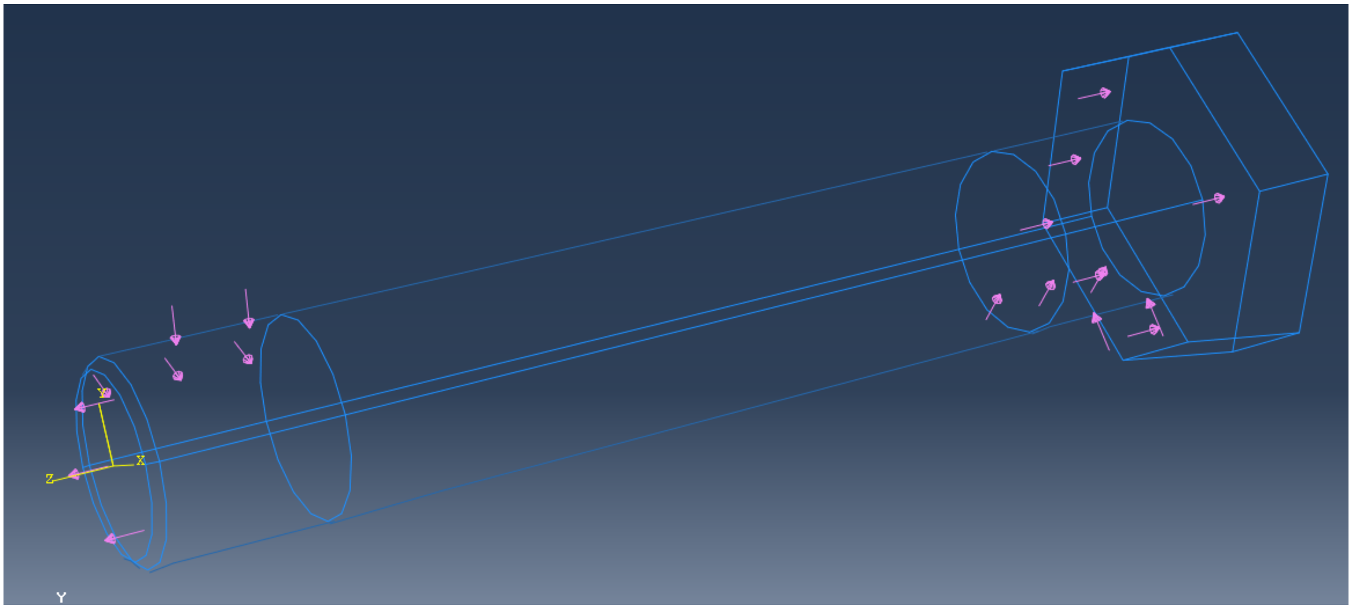


Figure 34: Bolt free body diagram

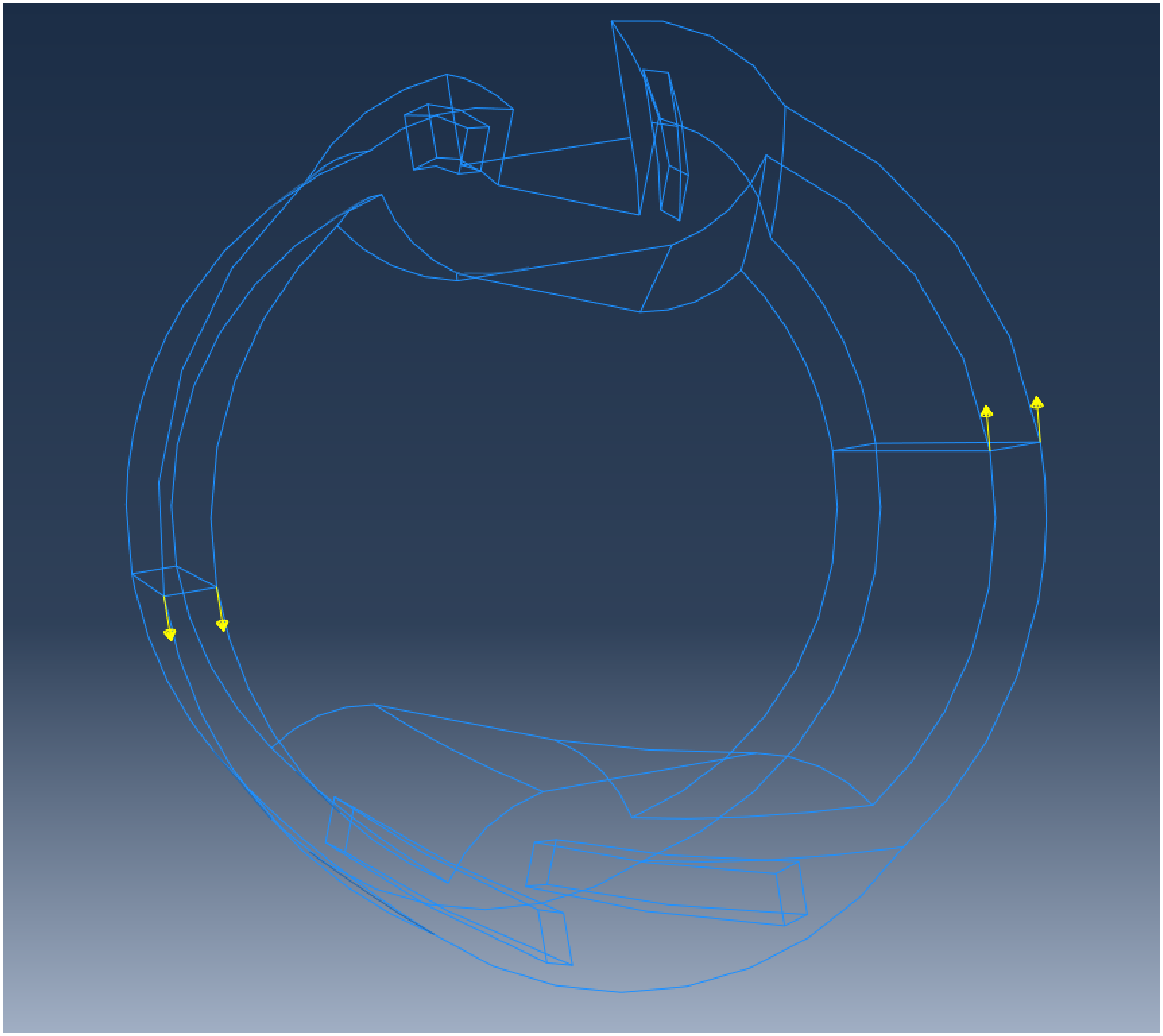


Figure 35: Outer assembly free body diagram

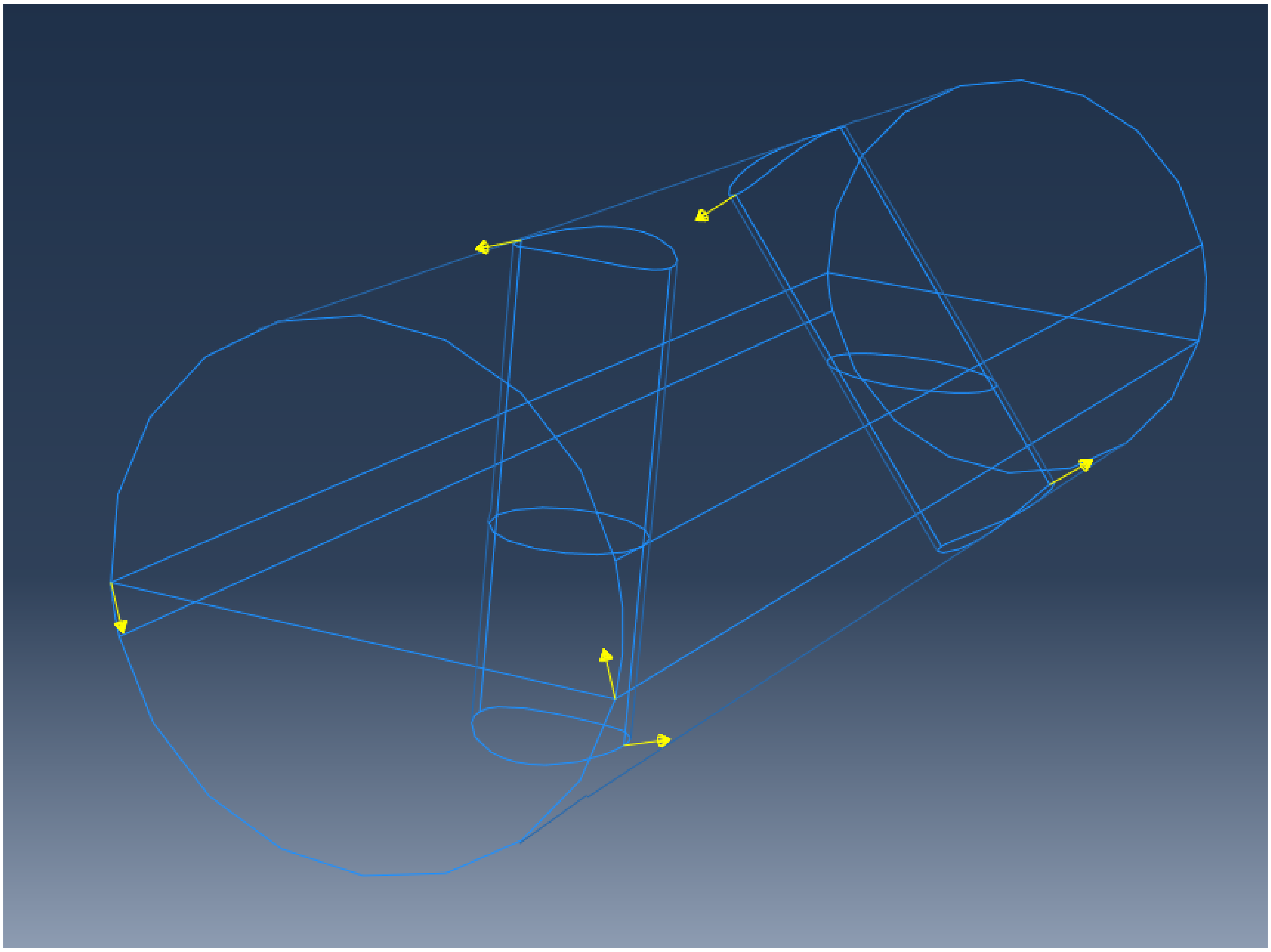


Figure 36: Inner shaft free body diagram

Torsional strain energy analysis When one end of a beam is fixed and the other is torqued, the deflection felt at each point in the beam is different. Locations near the fixed end deflect very little, while near the point of application of force deflections are large. The following equation relates the angular deflection along the length, starting from a fixed end, to the ratio of the torque to the polar moment of inertia and shear elastic modulus of the material [9]:

$$\frac{d\theta}{dl} = \frac{T}{JG} \tag{7}$$

Consequently in pure torsion the linear deflection increases linearly from a fixed boundary to a free boundary where a torque is applied. For a 67" beam, the local deflection at position x away from a fixed point and at the outer radius is given in inches by

$$\delta_{local} = \frac{x}{67}\delta_{max} \tag{8}$$

Spin to Win investigated the total strain energy involved in creating a torsional deflection under the above conditions – one end fixed, one end torqued – of 0.001" at the center of the knife, which requires creating a deflection of 0.002" at the torqued end. This strain energy was compared to the strain energy required to torque a simple steel beam of equivalent radius, which closely approximates MAXSON's current cutter, in order to show satisfactory stiffness.

However, the spokes assembly does not deflect in simple torsion all the way across. It deflects in a variety of modes and, when subjected to a torque, falls into the lowest-energy possible equilibrium position. What this means is that if a component (say the spoke arm) can deflect in both bending and shear, it will deflect in both in such a way that the total deflection under load is greater than the deflection of either mode acting alone. These deflections are said to be in "parallel." On the other hand, when modes of deflection must occur together for there to be any deflection at all, the resistances cooperate, lowering the total deflection that would occur compared to either mode operating alone. These deflections are said to be in "series."

The ratio between an applied force (or torque) and a deflection (or angular deflection) is known as the spring constant. For spring constants A and B which are in series, and for spring constants C and D in parallel, the effective, combined spring constants are as follows [9]:

$$k_{A,B} = k_A + k_B \tag{9}$$

$$\frac{1}{k_{C,D}} = \frac{1}{k_c} + \frac{1}{k_d} \tag{10}$$

The spokes assembly has two major modes of deflection which occur in parallel when it is torqued: true torsion, in which an angle of twist develops between each successive plane down the axis; and relative rotation of the outer assembly about the inner, which involves bending of the spokes.

The first mode of deflection involves torsion of each component about the center, which is pure torsion for the outer assembly and inner shaft. The main bolts and spokes are also torqued about the center, but they rotate in whole along with some portion of the angle of twist, only being torqued for the range of angles that take place over their bodies. The threaded inserts connecting the spokes to the outer assembly do not lie centrally, so their deflection is close to a pure shearing along their body plane.

For 0.002" deflection at a radius of 4.25", the overall angular deflection is given by

$$\theta = \arctan(\frac{0.002}{4.25}) = 4.7 \times 10^{-4} \tag{11}$$

The torque required to generate 0.001" deflection can be calculated by summing the torques given by Equation 7, summing the results for each component at its local position. The overall spring constant in torsion, k, the ratio between an applied torque and angular deflection, is the ratio between the sum of these torques θ. The resulting strain energies, U, are given by

$$U = \frac{1}{2}T\theta = \frac{1}{2}k\theta^2 \tag{12}$$

Detailed calculations for each component are found in Section 16.2 of the Appendix.

The resulting strain energy for 0.001" deflection at the center is 1.59 joules, with a spring constant of 2.24×10^5 lb-ft per degree. These values are about one and a half times those of a simple bar.

The other type of knife deflection of interest is if the center shaft is held fixed and the outer assembly is rotated relative to it, placing the bolts in transverse shear and bending the spokes. For 0.001" knife deflection, the edge of the spokes must deflect $\frac{3.5}{4.25}$ as much. This deflection will result from the parallel spring rates of the spoke arm in cantilever bending and the spoke arm in pure shear, sheared against its root on the spoke ring. The reaction force P between the spokes and threaded inserts for this deflection, δ, can be calculated from the cantilever beam deflection equation [9]:

$$P = \frac{\delta 3EI}{L^3} \tag{13}$$

The parallel shear deflection involves an angle of deflection $\gamma = \arctan(\frac{\delta}{2.5}$ where 2.5 is the length of the spoke arm. From Hooke's law in shear, with shear modulus G (11.2×10^6 psi for steel) and shear area 3 in^2, the required shear force V to get an angular deflection γ is described by

$$V = \gamma GA \tag{14}$$

Both of these forces are referred to their moment arms in order to find the torque required to generate that force at that point.

The reaction force must also shear the threaded inserts, and for the spoke to be in stable equilibrium, there must be a balancing reaction force on the main bolts. The reaction load at the bolts tends to shear the bolts, but the fixing of the center shaft means that any angular deflection of the bolt must be achieved in a very tiny space – the clearance between the inner shaft and spoke ring. This clearance is reduced to zero with the use of a preload on the bolt, and so the stiffness of the main bolts under transverse shear is effectively infinite. This means bolt failure will occur before there is any relative motion between the spokes and inner shaft due to yielding of the bolts, as any deflection in shear is stopped by compressive stresses against the surrounding material.

The resulting strain energy in spoke bending for 0.001" knife deflection is 3.54 joules, with a spring constant of 1.65×10^6 lb-ft per degree. These values are about three times greater than a simple rod.

Treating the above two modes of deflection as parallel springs, the resulting spring constant is given by Equation 10, and strain energy in 0.001" knife deflection is calculated by assuming that the amount of deflection in each mode is inversely as the stiffnesses [9]. The ratio of $\frac{k_{torsion}}{k_{bending}}$ is 0.136. Then the total strain energy is given by Equation 15:

$$U_{total} = (0.5)k_{bending}((0.136)\theta_{bending})^2 + (0.5)k_{torsion}((1-0.136)\theta_{torsion}))^2 \tag{15}$$

The results are in Table 8, together with the same values for a simple beam approximating MAXSON's cutter.

Deflection mode	**Spring constant**	**Strain energy in 0.001" knife deflection**
Pure torsion	$2.24 \times 10^5 \frac{lb-ft}{degree}$	1.59 joules
Spoke bending	$1.65 \times 10^6 \frac{lb-ft}{degree}$	3.54 joules
Combined modes	$1.97 \times 10^5 \frac{lb-ft}{degree}$	1.50 joules
MAXSON cutter	$1.25 \times 10^5 \frac{lb-ft}{degree}$	1.07 joules

Table 8: Overall stiffness properties of spokes assembly

12.5 Spokes assembly bending loads and deflections

Bending due to self-load When the spokes assembly is mounted in the cutting section, it is 67" long and suspended at both ends. As such it deflects under its own weight, with its weight being treated as a uniform load across its length. The equation that governs beam bending is given by Shigley's [9], where the second derivative of the beam's vertical deflection with respect to its length is the product of the square of the local moment M and the local flexural stiffness $\frac{1}{EI}$:

$$\frac{d^2y}{dx^2} = \frac{M^2}{EI} \tag{16}$$

Spin to Win constructed a spreadsheet to solve this equation stepwise for 0.1" intervals of the assembly. The result was compared to the analytical equation given for maximum deflection of a simple beam, which models MAXSON's cutter, where w is the load per inch and l the beam's length:

$$y_{max} = \frac{5wl^4}{384EI} \tag{17}$$

The detailed calculations are shared in Section 16.2 of the Appendix. The resulting maximum deflections are shown in Figure 37.

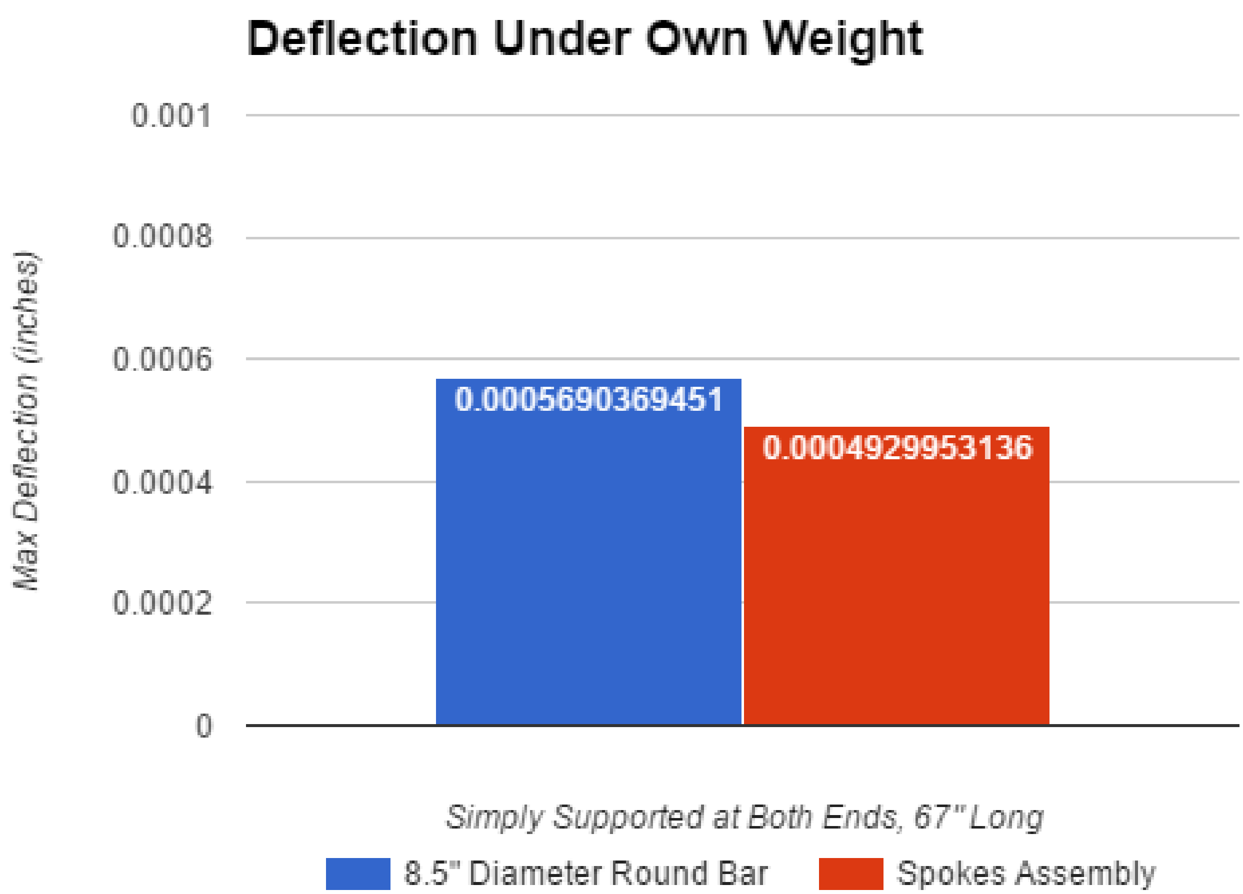


Figure 37: Deflection under self load for MAXSON cutter and spokes assembly

12.6 Spoke orientations

An early question of the design process was whether having spokes all in parallel, transverse to one another, or at a helix angle was ideal. Finite element analysis was used to solve this problem, with results given in Figures 38 and 39. The areas far from spokes develop large deflections, as nothing resists the local stresses in those areas. It was determined that spreading spoke attachments as evenly as possible and otherwise near the slugs would lead to equitable deflections and lowest-possible deflections near the knife.

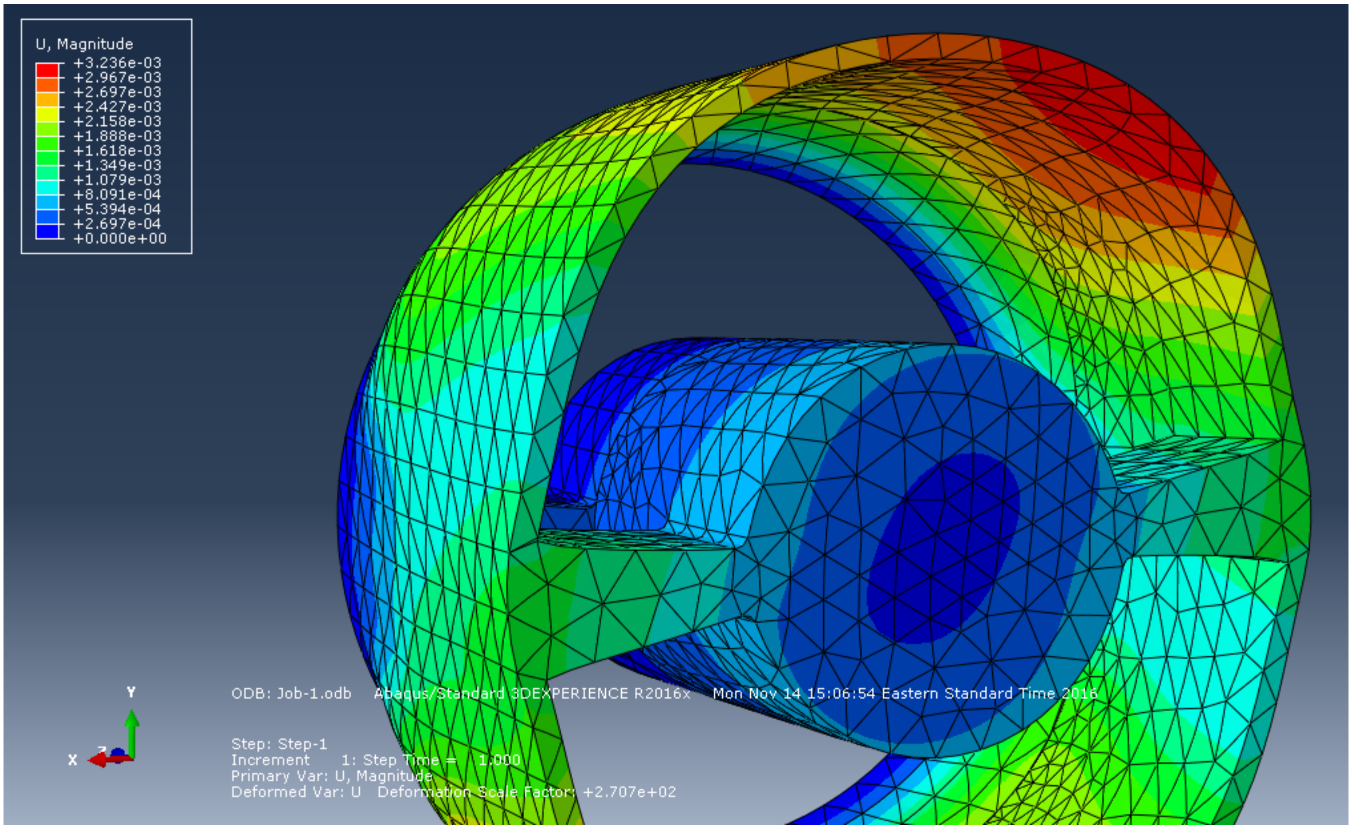


Figure 38: Deflections with parallel spokes

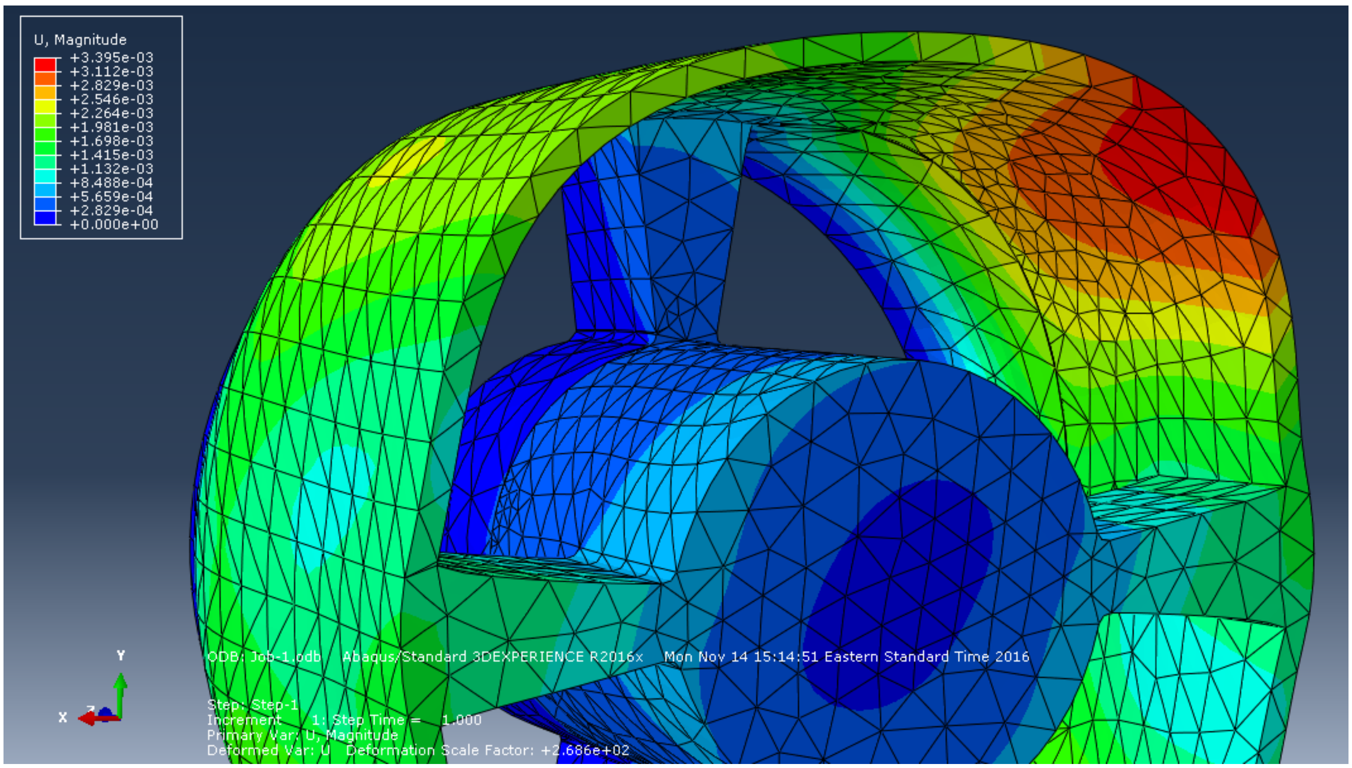


Figure 39: Deflections with transverse spokes

12.7 Curved versus straight members in shear

A central feature of the design is the curvature of the spokes. Finite element analysis was used to determine that a cantilever beam with curvature experiences a lower deflection than a straight beam of the same mass. In one characteristic experiment shown in Figure 40 the deflection was lower by about 8%. The effect of curvature on the spoke arms has been ignored in the above analysis as being difficult to determine without simulations (which themselves require huge computational capacity), but it is believed to aid overall stiffness by this amount.

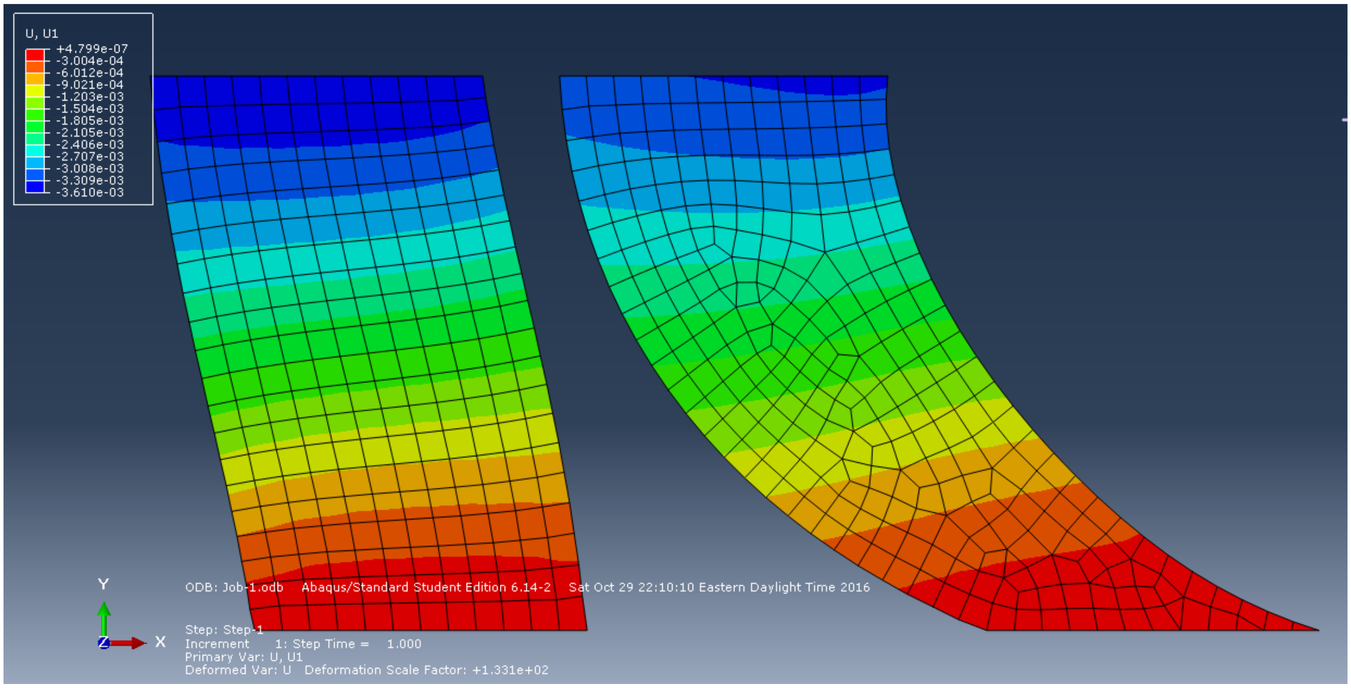


Figure 40: Curved and straight cantilevered beams

13 Build/Manufacture

The build and manufacture of the test model and testing apparatus went though many stages before it was completely finished able to be accurately tested.The main stages being design, redesign and and manufacture. Throughout the entire capstone course there were many elements that needed to be changed and redesigned to meet certain conditions, be it lower cost or faster turn around time. The testing model was no exception to this, and presented somewhat of a challenge to the design team.

13.1 Design of Testing Model

The design of the test model for the purpose of torsion and stiffness testing had to be designed with a few key factors in mind. In order for our data to have any use to our design team our test model needed to be as close as possible to the final design as was feasible. The design team attempted designing a simplified model that would be sufficient for testing. The model was designed to be a simpler model of the final design redesigned for ease of manufacturing and to be a lower cost.

In addition to creating a model of the redesigned cutter blade the team was also tasked with creating another scaled down model of a cutter blade. This secondary cutter blade was to be designed to be tested in the same rig as the first model. The purpose of the second test model was to compare the capstone design teams cutter blade, against another cutter blade that was designed by MAXSON. Originally the design team had requested a scaled down model of the current MAXSON model, so that the optimized cutter blade could be compared directly to MAXSON's current model. However due to time constants the Sponsor had opted for the design team to compare the redesigned model against their model.

The basic model of the cutter blade that would be compared to MAXSONS models was the wheel and spoke design that was pitched in last semesters capstone proof of concept presentation. The model consisted of the four main parts, the inner shaft, the outer skin, the spokes, and the slugs. However to make the model less costly and easier to manufacture the model was simplified in several ways. The biggest changes were to simplify the geometry and remove the helical angle of the cutter blade. This would mean that while the model being tested would not actually be functional in the current machine being used by MAXSON, it would have a similar enough geometry that its test results would still be useful to the capstone team. In addition these simplifications such as the removal of the helical angle would also be applied to MAXSON's optimized rib model to make the test data more comparable.

Before the test model could be designed and built the optimal spoke had to be chosen, This was done by building another test rig which would be used to compare the spokes. THe basics idea behind this model was to create three scaled down model spokes that would be subjected to loads. These spokes since they were all subjected to the same loads but were from very different geometries, would had different deflections, which would therefore lead to different stiffnesses, in the full scale model.Once the stiffest spoke model was chosen it was placed into the final design as the spoke for the full length model.

13.2 Design of the Test Rig

In addition to creating a test model that would be tested the design team also faced the challenge of creating a rig that would be able to hold the model still in all direction except for rotation. The design team came up with a fairly clever design using simple materials that were purchased through the capstone program or were provided by the machine shop.

The basic requirements for test rig were that it would be able to hold the model still in all directions, while still allowing the model to rotate. The test rig was also designed so that it would be easy to assemble and test multiple models after one another, and that it would not be too difficult to remove one model and test another. The solution proposed by the design team was two large standing beams that were to be supported by trusses on the sides for added stability. Since the model would be fixed at one end and free at the other, the model was designed so that it could be bolted directly to the test rig at one end, and free to move at the other end. The design team used a large Aluminum beam that was just a spare piece of metal in the machine shop as the main supporting beam and then used a smaller section cut from the beam to create the hollow part that would fasten the testing rig to the testing model, by using bolts that pass through the rig and into the model. In addition the end of the test rig that would be fixed was designed with an additional set of trusses which were put in prevent any bending as the model was loaded.

In addition to the scaled down model test rig, there also was a smaller testing rig that was designed to test the spoke models and compare them against each other using DIC, or digital image correlation. The test rig was designed to be made out of a thick steel pipe that would be used to hold the spokes in place, through two holes in the top and bottom of the pipe. This would then be held in place by the Instron machine, and would be torqued by a center shaft. To solve the issue of torquing all the spokes to equal loads, the design team decided to attach a bolt to the end of the center shaft and use a torque wrench to apply loads evenly across all spokes models. This was also made in such a way that the front of the spokes could be seen and evenly lit by the DIC camera.

13.3 Manufacturing the Full Length Models and Spokes

The full length model and the spokes that were used by the capstone group were mostly machined by MAXSON, and were delivered to the capstone group for final machining and assembly. The spokes were outsourced, from MAXSON, and to another vendor who machined the spokes for them and delivered them to MAXSON. The last part the second model of the ribbed cutter, was also outsourced from MAXSON to another manufacturer who was to make the model and deliver it to MAXSON, so that the capstone team could pick it up with their parts and begin testing on them.

The Ribbed model which was outsourced to another company that was undisclosed to the capstone team, was going to be made from one solid piece of steel. The steel would first be turned on a lathe to get it closer to the final sizes. The model would then be finished with a milling machine to get closer to final design and put the cuts into the model that could not be done by a lathe due to it's geometry. The last step in the the manufacturing of the ribbed model would be to drill and tap holes that the knife would be mounted too on the cutting blade.

The spoked model would be made with both milling, and the lathe. The one piece that was made not using there operations, was the spokes, which were outsourced to a waterjet company that simply cut the spokes out a solid sheet of steel. The waterjet left a tapered cut in the sheet of metal however and would result in the spokes being somewhat uneven, but they were within acceptable limits set by the design team. It should be noted though that the waterjet would not be an acceptable form of manufacturing for the spokes at full scale, because the spokes would need to be thicker than the waterjet allows. The spokes were then finished by drilling holes into them for attaching the spokes to and from the center shaft to the outer skin.

Once the spokes are manufactured the the center shaft was turned on a lathe from a piece of stock steel. Then holes were drilled through the shaft perpendicular to the axis of rotation, that would be used for both torquing the model, and for securing the model to the test rig. Once the center shaft was nearly done the final step was to drill out holes along the axis that would be used to mount the spokes to the center shaft.

The outer skin was made from a solid steel tube that was purchased by MAXSON, and then cut down its length in four places. This would remove two 45 degree sections opposite from each other on the tube, and would provide a place for the slugs to slide into and be mounted. The final step in this part was to drill holes through the skin at regular intervals down the length of the piece on opposite sides from each other, to attach the spokes to.

The final piece to be manufactured was the slugs. These pieces were made from solid steel beams, and were made almost entirely from milling. They were designed so that they would fit into the 45 degree cutouts on the skin, without interfering with the spoke assembly.

13.4 Assembly of Spoked Model

Once the parts for the model were all manufactured they would go on to be assembled. The assembly process began with mounting all of the spokes to the center shaft, by sliding all of the spokes over the end of the shaft, and down until they met with a hole in the center shaft that a bolt was pushed through and tightened to hold in place. Once the spokes were bolted into place the skin was slid over the spokes and bolted to the spokes for one side, then the other, being sure that the bolts attaching the skin to the spokes were not fully tightened. Before the bolts were tightened all the way down, the slugs were slid into place between the two skin sections. Once the slugs were in position the bolts were tightened and the final step was to spot weld the slugs in place.

13.5 Manufacturing of the Testing Rig

The test rig was manufactured by hand by the capstone group by hand on the main URI campus and also at the Schneider electric machine shop. The test rig was made first by cutting two by six pieces of wood down to the correct size for the bottom support structures and then at angles for the trusses as well. The center aluminum shaft was cut to size and a smaller piece cut off to support the model at the top. Once all pieces were cut to size assembly began by attaching angled brackets to the aluminum and the wood on the bottom to build the first truss. Then the wooden trusses were screwed into place using hinges to act as brackets and the top aluminum support piece was added by using 90 degree angled brackets. Once all pieces were assembled there were a series of long nuts and bolts added to the assembly to act as a lever arm that would go from the test model to the instron machine and allow for testing of the model.

13.6 Theoretical Full scale Assembly and Manufacture

The full scale assembly and manufacturing of the spoked model and ribbed model would be nearly identical to the small scale models. There is a few problems that would need to be tweaked with the full sized spoked model to make it applicable to real life however. The ribbed model would be, and has been created in some interpretation before by MAXSON

The Ribbed model would be made by simply turning a larger steel rod on a lathe down to the sizes specefied by the design, and then would be finished on a milling machine. This model is simply just a modification to MAXSONs current cutter design and would essentially follow MAXSON's current manufacturing method for thier bottom cutter blade, with the one caveat that after the machining was complete, there would be an additional step. The final step would be to machine away some of the slots in the piece to lower its mass and provide the additional mass and inertia savings that were sought after in the scope of the project.

The spoked model would follow a similar manufacturing method as the small scale model laid out in subsections 13.3 and 13.4. The primary difference in the small scale and full scale products is that when the full scale model is produced it would need to have the helical angle cut into the outer skin piece, and would need to modify the slugs to accept the cutting blade. These changes would result in a a large amount of added machine time and expenses to MAXSON in order to produce them. This is also not mentioning the fact that some of the pieces were essentially deemed impossible to machine with MAXSON's current machine shop. However for the spokes they would need to be machined from a CMM milling machine, the center shaft would simply be spun on a lathe, the outer skin would need to be cut then routed to accept the slugs, and the slugs would require extensive milling. In addition there would be several drilling and tapping operations to add the holes and threads to the full scale assembly.

13.7 Manufacturing Analysis

The manufacturing analysis was almost all done on a costing software called Apriori. It was mentioned in an earlier section and in the presentations for the course earlier in the year. Essentially this software provides detailed and accurate costing estimates by analysis the geometry of CAD part files. Using this software allowed the design team to accurately estimate not only the costing of components but what machining methods would be feasible for each and every part in the assembly.

14 Testing

14.1 Test Design

The tests performed on the spokes assembly were designed around two key goals: to verify the critical performance features of the system and to work around the available resources in terms of time and machinery. In addition, these tests had two main phases: testing alternative spoke designs in order to pinpoint the relationship between spoke geometry and stiffness, and testing the full-scale assembly.

The most important feature of the spokes assembly is its performance. That means its inertia and stiffness, particularly torsional. The inertia is well-known from analysis and is difficult to measure directly, but a measurement of the mass can be taken and compared to the predicted mass. This would provide an estimated agreement between the predicted rotational inertia and the actual.

Testing the torsional and bending stiffnesses required some careful planning, and in the end, Spin to Win was not entirely successful at designing tests that would confirm these attributes. The largest difficulty came from the fact that the University of Rhode Island does not have any kind of torsional stress testing machine. Machines which output torque and angular deflection during a deflection exist, but URI only has machines that output linear force and linear deflection. Experiments had to be designed to generate torsional data from linear forces and deflections.

Testing Spoke Alternatives For the first phase of testing, three alternative spoke designs were created which vary in three parameters: the curvature of the spoke arms, the thickness of the spoke arms, and the orientation of the skin attachment bolts to the main bolt connecting the spoke to the inner shaft. The first spoke had thick arms; gentle, constant curvature; and a 45 degree angle between its bolts. The second spoke had a much more dramatic curvature that terminated, as much as possible, in the tangential direction of the cutter. Spoke two was otherwise the same as spoke one. Spoke three had thinner arms, a curvature that increased in the outward radial direction, and bolts at 90 degrees to one another.

Since the stiffness of only the spokes was under investigation, it was not necessary to build an entire assembly, but instead a cross-section of the cutter was used containing an inner shaft, an outer skin, and one of the spokes. The test assembly was designed so that the chosen spoke could be swapped out easily by loosening the bolts and sliding the spoke off the inner shaft.

The test required that torque be applied to this cross section from the inner shaft, with the outer skin held fixed, and the strains and displacements in the spoke arm be measured. Many alternatives were discussed for how to achieve this configuration. In the end, the set-up that was considered optimal was to hold the assembly fixed in the chucks of an Instron machine while torque was applied with a torque wrench (which could apply a known, fixed torque value) from the rear. At the front, a camera would collect the displacement data the spokes using 2D digital image correlation.

The construction of the test rig is showed in Figure 41.

These tests encountered large difficulties when it was discovered that the chucks for a linear Instron machine are not built to handle displacements in the directions perpendicular to the direction of linear deformation. In other words, the entire test assembly had some small degrees of freedom into and out of the plane of the 2D DIC and also to the left and right from the point of view of the camera. Since DIC relies on the pixels in the camera's view remaining in exactly the same place from image to image, excepting effects from the displacements or forces under consideration, the displacements resulting from the un-fixed chucks corrupted the data from these tests. Efforts to interpret the data and get useful results from them are shared below.

Testing the Full Assembly

The displacements under torque of the full assembly are not constant down its axial length as can be assumed for a small cross-section. As such, 2D DIC is no longer an effective method for gathering displacement data. 3D DIC is possible but is very complex. Spin to Win looked at analog ways of measuring the angular displacement at one end, such as using a needle attached to the assembly's inner shaft and monitoring its motion against a backdrop with hash-marks at a very large radius. A version of this, using a high-resolution camera instead of a large radius, was actually used in the final tests, but reliable data was not produced.

Instead Spin to Win engineered a test that would allow it to use the linear Instron machines owned by the University of Rhode Island. For this test, a straight bar would be placed through a hole in the central shaft, locked in place at that hole and held on the other side by a chuck of an Instron machine. When the Instron machine applies a linear force and gets a linear displacement, at small angles, the angular displacement in the shaft is determined by the linear displacement in the Instron machine and the lever arm. At the same time, the torque on the shaft is determined by the linear force and the lever arm. In this way, it is theoretically possible to build a torsion experiment using a linear force measurement tool.

This test rig is shown in Figure 42.

Figure 41: Spokes Testing Rig

In reality, this experimental set-up also encountered several problems. One was that, in order to fit through the central shaft, the bar used to apply torque had to have a fairly small diameter (3/8" was used). Consequently it did not have a high flexural moment of inertia and was prone to deflect itself under load. This became especially problematic when the bar

Figure 42: Full-Length Spokes Assembly Test Rig

began to deform plastically, as this plastic deformation added displacements to the system that did not correspond to any strain energy in the spokes assembly. This greatly muddied the data.

The other problem was that, because of the bar's low stiffness, a fairly large strain had to be placed on the bar in order to get any significant strain in the spokes assembly. As a result the ability of the data to speak to the system's stiffness was limited, and the experiment looked more like an experiment to measure the stiffness of the 3/8" bar under bending. Exact calculations, discussed below, suggest that some strain energy was put into the spokes assembly during the experiment and that its performance was close to expectations, but this required a very careful reading of the data.

One way the experiment was improved was by the substitution of a piece-by-piece assembly of small bars, joined by coupling nuts, for the homogeneous bar that was used initially. This compound assembly had a much higher stiffness than the single piece. While this improved the experiment, it was also a piece of indirect evidence in favor of the wheel and spokes assembly, as the response of that piecewise bar to bending is identical in principle to the response of the spokes assembly to torque. The simple mode of deflection (bending for the bar, torsion of the spokes assembly) exists only in a short beam before contact exists with another component, during which the simple form of stress is converted into a different one. The performance of the piecewise bar was a promising indicator of the likely performance of the spokes assembly, if a satisfactory test could be designed for it.

The initial test plan involved placing the spokes assembly and a model of the existing MAXSON cutter, modified with ribs to lower inertia, into the same test apparatus. In this case, the data could be compared side-by-side and the error involved in the 3/8" bar would be averaged out. However, MAXSON's vendor was unable to deliver the ribbed model in time for the test, and it had to be neglected.

14.2 Test Matrix

An organized chart of Spin to Win's intended tests is shared in Figure 43, together with all data acquired in these tests.

Subject of test (part/assembly)	Test	Target Info / Need	Apparatus	Repetitions?	Results 1	Results 2	Results 3	Results 4	Results 5				
Spoke Design 1	Measure central hole diameter	Correctness of fit with center shaft	Calipers	5x at different positions	0.786	0.785	0.786	0.787	0.786				
	Measure outer maximum distance, outer hole to outer hole	Correctness of fit to outer skin	Calipers + Gage blocks	1x	2.72								
	Measure thickness	Determine sat/unsat to design	Calipers	5x at different positions	0.815	0.81	0.813	0.812	0.818				
	Torsional stiffness test	Determine torsional spring rate for comparison + analyze stresses	machine + Torque wrench + DIC suite	5x	1523.105979	2842.769239	53105.36943	12921.78844					
Spoke Design 2	Measure central hole diameter	Correctness of fit with center shaft	Calipers	5x at different positions	0.786	0.785	0.786	0.787	0.786				
	Measure outer maximum distance, outer hole to outer hole	Correctness of fit to outer skin	Calipers + Gage blocks	1x	2.72								
	Measure thickness	Determine sat/unsat to design	Calipers	5x at different positions	0.815	0.81	0.813	0.812	0.818				
	Torsional stiffness test	Determine torsional spring rate for comparison + analyze stresses	Torsional test rig + Instron machine + Torque wrench + DIC suite	5x	23189.13429	6648.758975	7734.860909	10980.17375					
Spoke Design 3	Measure central hole diameter	Correctness of fit with center shaft	Calipers	5x at different positions	0.786	0.785	0.786	0.787	0.786				
	Measure outer maximum distance, outer hole to outer hole	Correctness of fit to outer skin	Calipers + Gage blocks	1x	2.82								
	Measure thickness	Determine sat/unsat to design	Calipers	5x at different positions	0.813	0.822	0.832	0.811	0.812				
	Torsional stiffness test	Determine torsional spring rate for comparison + analyze stresses	Torsional test rig + Instron machine + Torque wrench + DIC suite	5x	9577.529295	13984.23014	16547.55127	29122.32818					
Shaft + Spokes Assembly	Measure axial length	Determine sat/unsate to design	Calipers + Gage blocks	1x	26,320								
	Verify integrity of bolted connections against shear	Ensure no play between spokes and shaft	Vice, pliers	9x once for each spoke	good	good	good	good	good	good	good	good	good
Shaft + Spokes + Skin Assembly	Verify alignment of holes and continuity of threads	Ensure clearance for outer inserts	Bolts, screwdriver	18x twice for each spoke	good/good	good/good	good/good	good/good	good/good	good/good	good/good	good/good	good/good
Full Wheel and Spokes Assembly	Measure mass	Ensure sat/unsat to design	Heavy mass scale	1x	xxx								
	Full-scale torsion test	Determine torsional spring rate	Large-scale torsional test rig + Instron machine + Torque wrench	7x	8365.066854								
Full MAXSON Revolver Assembly	Measure mass	Ensure sat/unsat to design	Heavy mass scale	1x	xxx								
	Full-scale torsion test	Determine torsional spring rate	Large-scale torsional test rig + Instron machine + Torque wrench	7x	xxx								

Figure 43: Spin to Win Test Matrix

14.3 Analysis of Test Results: Spoke Tests

The Spoke tests involved applying torque in increments of 60 lb-in up to 240 lb-in. For each increment and for each spoke, the DIC program outputted two pieces of data: 1) the displacements in X and Y of a single point, selected about an inch up from the center shaft, and 2) the strain values at all points in a region encompassing the spoke arms.

In order to get the torsional stiffness of the spoke arm in the test assembly, we investigated the ratio between the applied torque and the angular displacement that occurred. This angular displacement was found using trigonometry, using the X and Y coordinates of a point from DIC.

As mentioned above, the chucks holding the test rig in place shifted slightly during the experiment, and there was no way to determine by how much or in what direction. In some cases, this shift was opposite the direction of displacement, and the X and Y displacements were registered as negative (that is, the system appeared to "undeflect" when a torque was applied). However, a "best-fit" line can be drawn through the data points, with torque plotted on one axis and angular deflection on the other, in order to generate a torsional spring constant. This is shown in Figure 44.

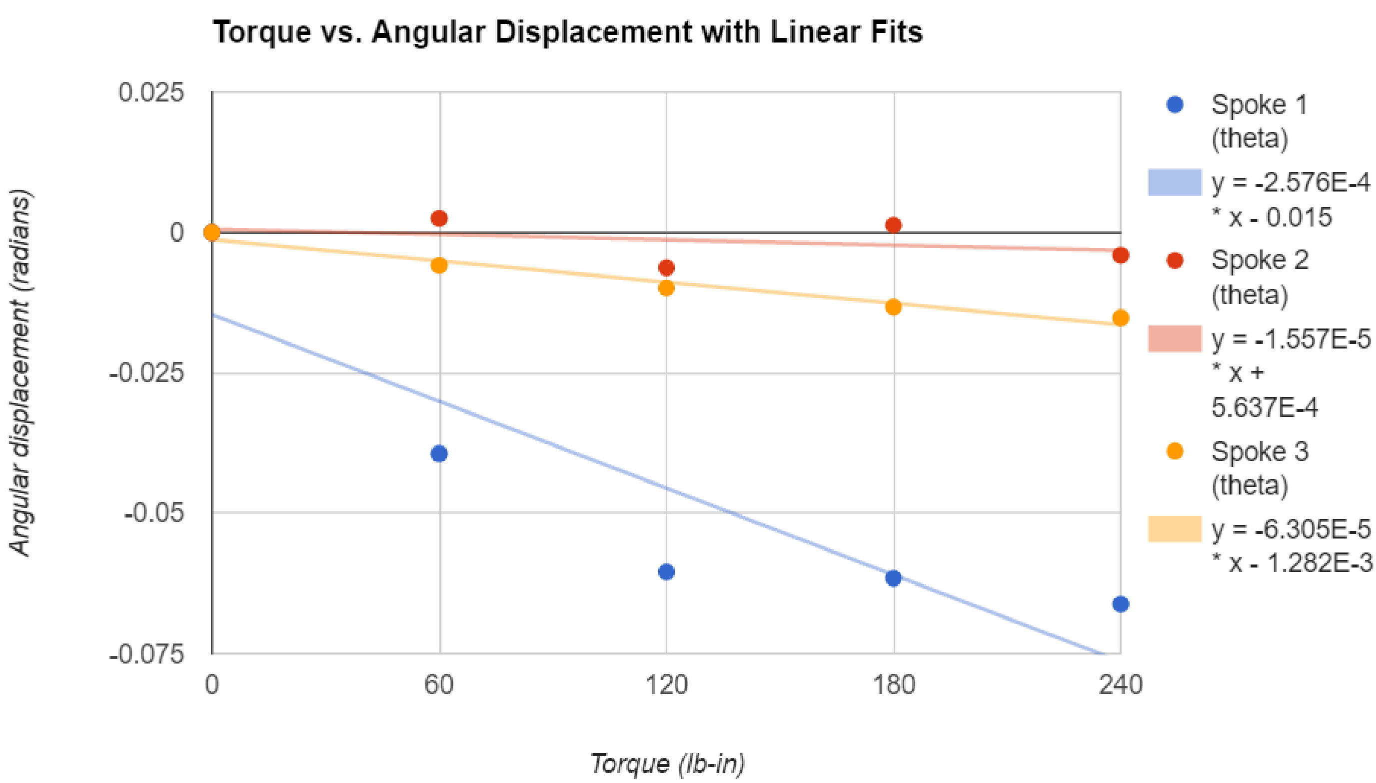


Figure 44: Spoke Torsional Spring Constants

If the torsional spring constants recorded at each increment of torque are listed as a data set, and a mean and standard deviation is taken, it is revealed that, because of the errors in the chucks, this data is of extremely poor quality. A normalized, Gaussian distribution for spoke is given in Figure 45. The standard deviation for each data-set is, in some cases, equal to the mean, meaning that from this data it is possible to conclude that the each spoke is twice as stiff as recorded or even has zero or negative stiffness.

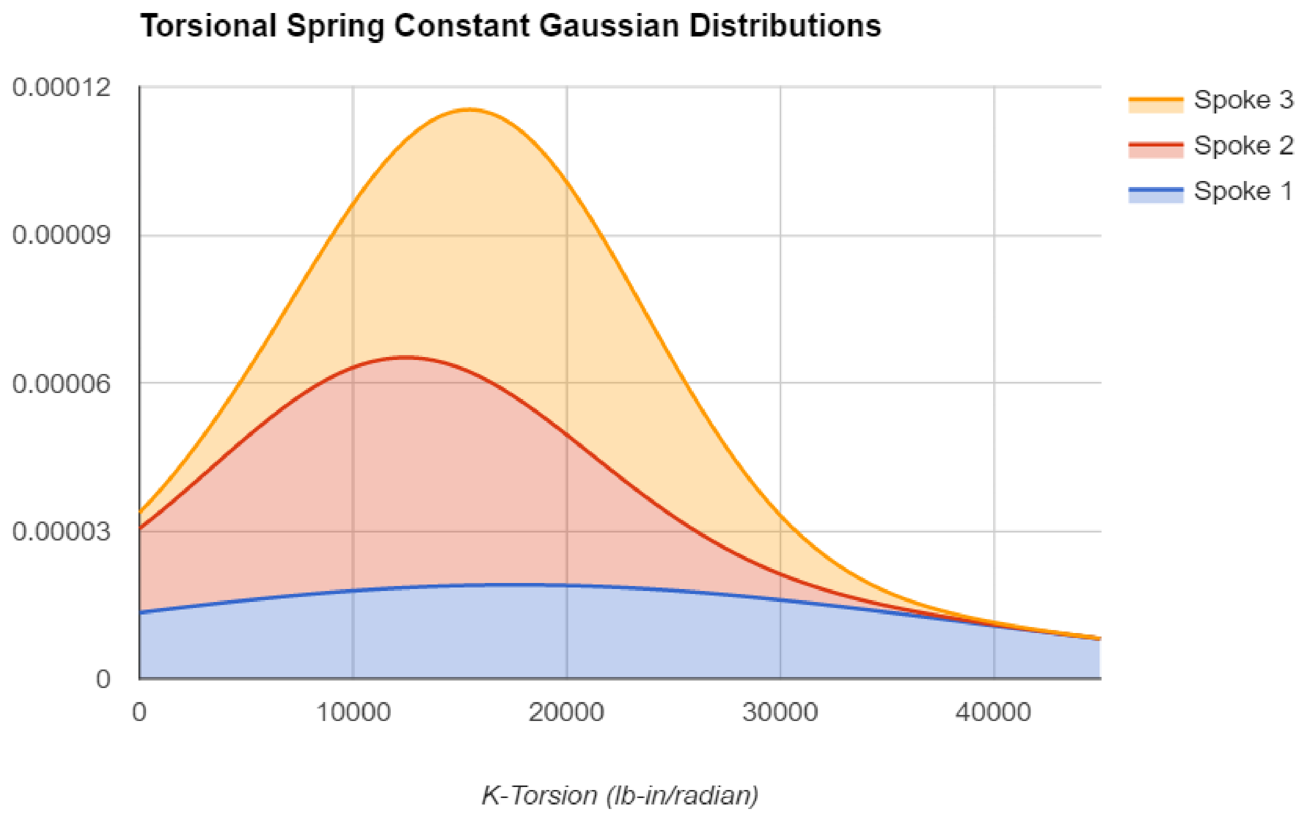


Figure 45: Spoke Torsional Spring Constant Gaussian Distributions

As torque was increased, the transient effects in the system, including any give between the bolts and their holes and the displacement of the chucks, were likely to be removed and elastic engagement would take over. By that rationale, the final data points, those taken between 180 and 240 lb-in of torque, are likely to be of the highest quality for determining the actual torsional constant of the system. A graph which shows the torsional spring constants for each spoke at these data points, together with a theoretical spring constant calculated using the same methodology as in the Engineering Analysis section above, is given in Figure 46.

Of particular note is the fact that Spoke 3 outperformed Spokes 1 and 2 by a wide margin in this metric. This is probably correct and has an explanation from the DIC data. Spoke

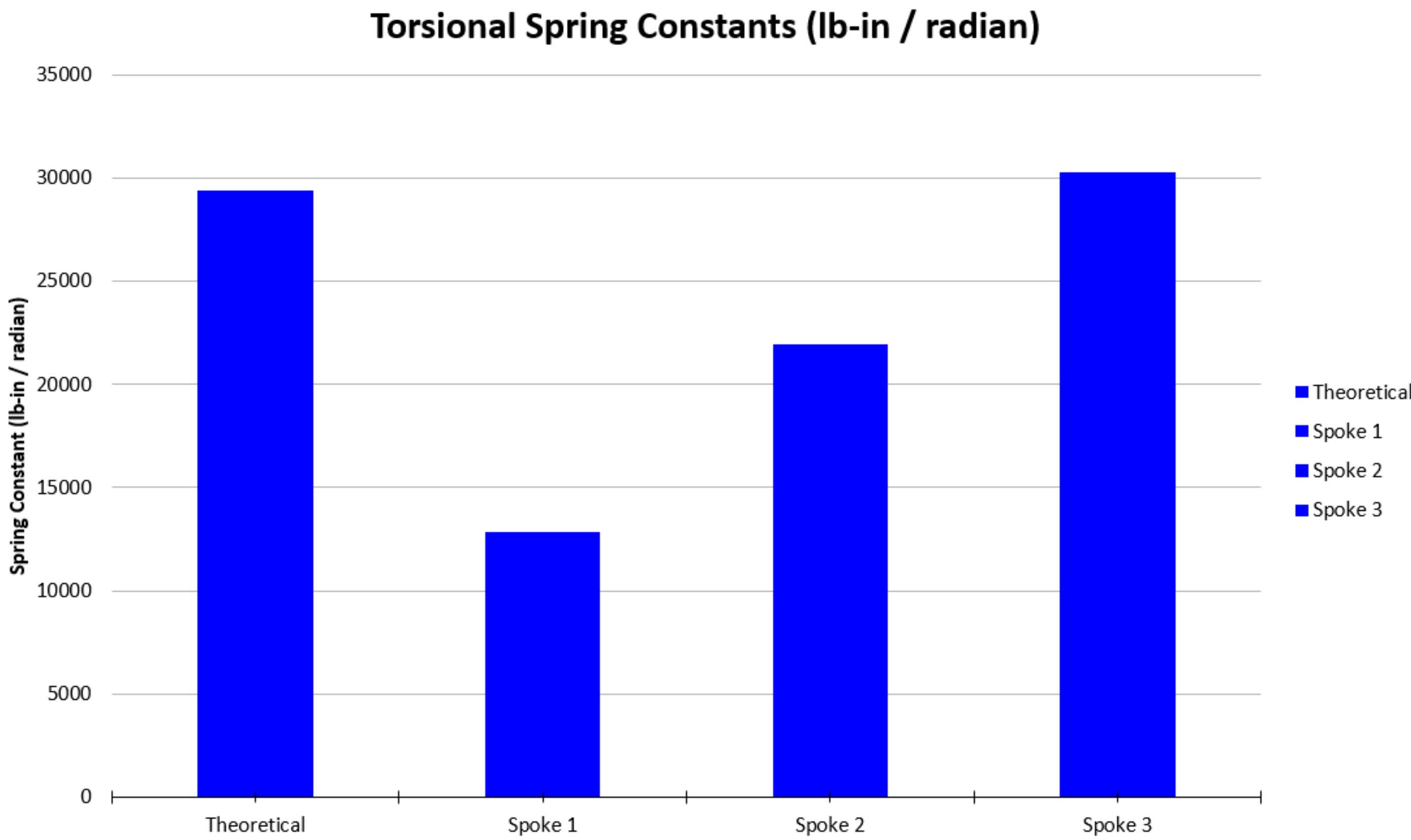


Figure 46: Spoke Torsional Spring Constants at End of Test

3 has its main bolt at 90 degrees to the outer skin bolts, as opposed to 45 degrees for Spokes 1 and 2. Spoke 3 showed significant y-direction strain compared to Spokes 1 and 2, meaning that the spoke arms were stressed directly against the skin along their length as opposed to only transverse to it. This resulted in a very low-deflection distribution of strain energy. As a result of these tests it was concluded that the 90-degree bolt orientation is most effective. Between Spokes 1 and 2, the higher curvature of Spoke 2 proved more effective, which aligns with the DIC experiments done in the Engineering Analysis section on the impact of curvature on a cantilever beam's stiffness.

14.4 Analysis of Test Results: Full-Length Test

Because of the plastic deformation that occurred during the test, it was difficult to generate a predictive model for what should have occurred. In order to do so, the normal, elastic deformations in each component of the system were supplemented by a plastic deformation in the components of the bar used to apply torque. Rasmussen [13] gives an equation for a stress/strain relationship that encompasses the plastic region for stainless steel, the material of the bolts. This equation is a piecewise function for the elastic and plastic regions of the curve defined around the stress/strain at 0.2% proof stress and constants m and n given based on the material grade (Grade 8 for these tests):

$$\epsilon = \frac{\sigma}{E_0} + 0.002\frac{\sigma}{\sigma_{0.2}}^{n} \; for \; \sigma \; < \; \sigma_{0.2} \tag{18}$$

$$\epsilon = \frac{\sigma - \sigma_{0.2}}{E_{0.2}} + \epsilon_u \frac{\sigma - \sigma_{0.2}}{\sigma_u - \sigma 0.2}^{m} + \epsilon_{0.2} \; for \; \sigma \; > \; \sigma_{0.2} \tag{19}$$

The torque-applying bar, which was separated into small pieces connected by coupling nuts, was modeled as a series of cantilever beams subject to a force equal to the applied force in the Instron machine. The rest of the system was modeled as in the Engineering Analysis section, with the exception that for stresses above the 0.2% proof stress, plastic deformations were calculated as per Equation 19 above.

Meanwhile, the actual torques and torsional deflections were calculated from the values outputted by the Instron machine using trigonometry and a moment arm of 15.5 inches. The comparison between the two results is shared in Figure 47.

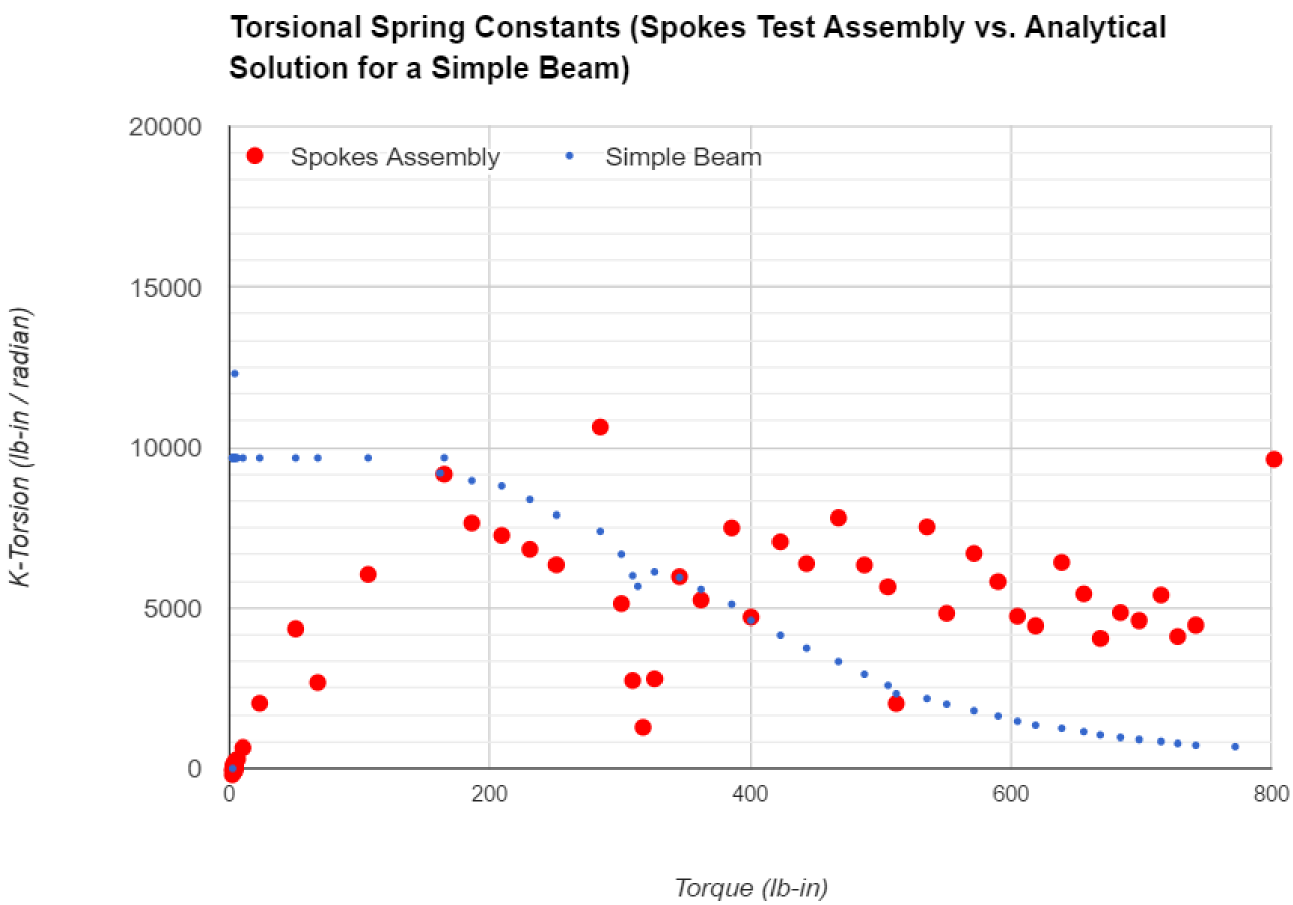


Figure 47: Spring Constants vs. Theory for Full-Length Model

The low-torque portion of the curve shows low stiffness initially during a ramp-up period. This is because of the give in the system; there was not perfect engagement between the bar and its hole, or between the bolts and their holes. Once fully engaged the system began behaving like a theoretical series of beams.

Still, the data is not very useful in understanding the performance of the system. For one, the displacement of the bar, even with its piecewise construction, is an order of magnitude larger than that of the bar. At the end of the elastic region, the predicted bar displacement is 0.47 inches compared to 0.05 inches for the cutter and 0.1 inches in the bolts holding the fixed end fixed. In the plastic region, this divergence increases greatly. As a result, any error in calculating the deflection in the bar – such as miscalculating the plastic deformation, using incorrect diameters or lengths for the components of the beams, or from neglecting any strain taking place in the coupling nuts – causes huge deviations of the prediction from reality. As such there is no solid touchstone to measure whether or not this data indicates good performance for the system.

An torsional Instron machine, or some other method of directly applying torque, with a better method of fixing the fixed end of the cutter could successfully prove the wheel and spokes design. However, this work remains to be done.

15 Redesign

15.1 Original Design

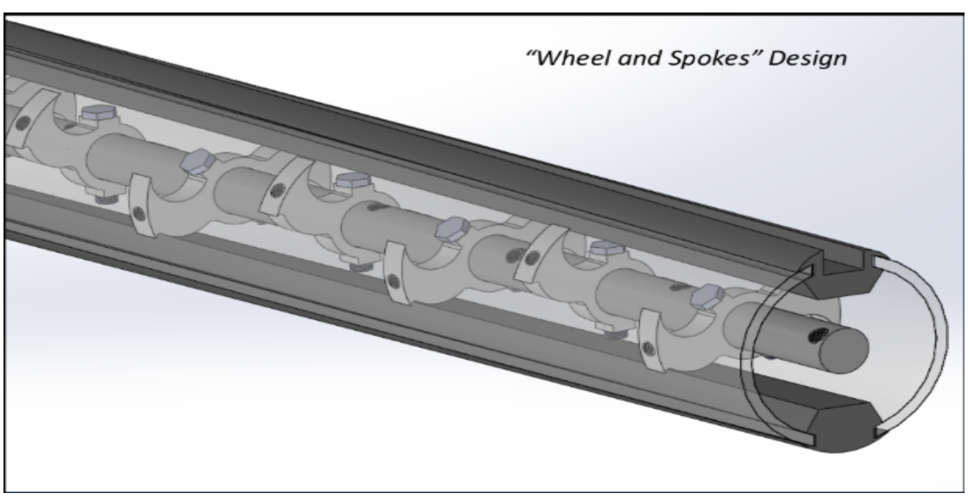


Figure 48: Original Design

The original final design, as seen in the figure below, consisted of a thin inner shaft, a tube with helical cutouts, and a thick solid slug inside each cutout, one slug to hold a knife down the length of the tube, and another to act as a counterweight. Nine spokes were designed to connect the inner shaft to the tube. In the figure below, the tube is a phantom drawing to make the inner shaft and spokes visible.

15.2 Redesign

The figure below shows the build of the final redesign, simplified from the original design. The bolt heads on the side of the tube connect to spokes on the interior of the tube. The spoke connect to the small shaft, which runs through the length of the tube. the strait slugs are welded to the tube. This is the design which was torsion tested.

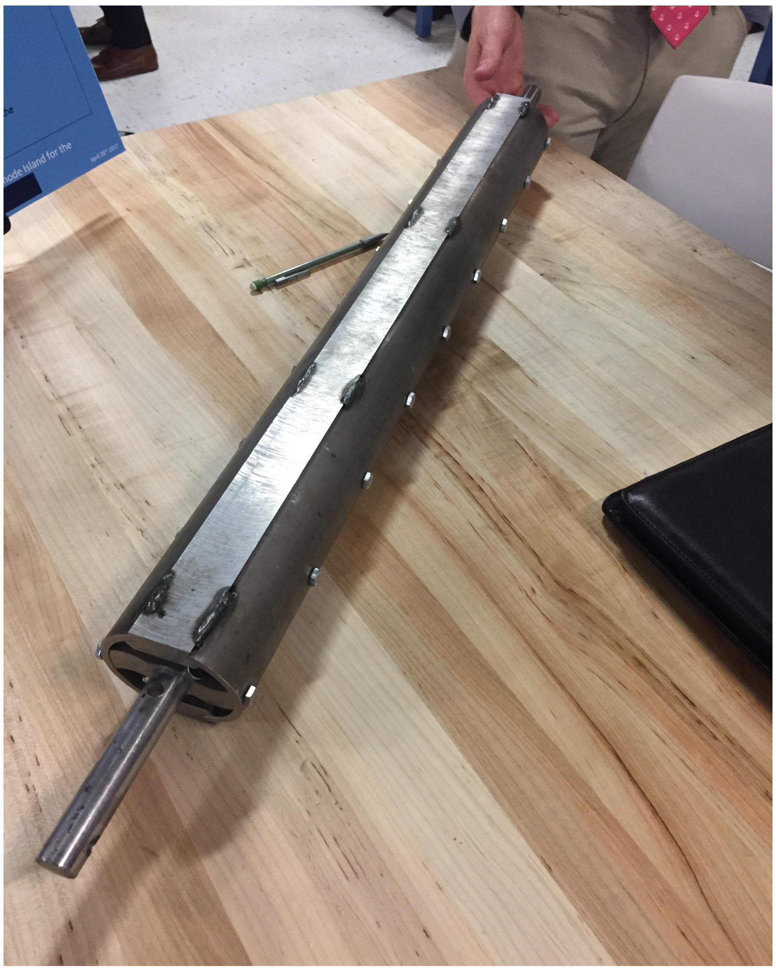


Figure 49: Simplified Design

The original final design was submitted to MAXON for manufacture, however, upon examination MAXON determined that the helical cuts in the tube would present manufacturing and assembly problems. MAXON also sent the group a written description of a simplified design which could be manufactured. Furthermore, MAXON asked the group to test this new, simplified design against a different, ribbed design. The group made a new design based on the written description provided by MAXON. This simplified redesign, is similar to the original design, except that the cutouts in the tube, and the slugs are now strait instead of helical. Also the spokes are all oriented at the same angle in the simplified design. The tested model was at a scale of 0.3928 to the full-scale design.

No redesign was implemented based off of testing, the simplified design can more accurately referred to as a late design change. The goal of testing was to measure the torsional stiffness the simplified design against a ribbed design provided by MAXON. MAXON was not able to produce the ribbed design in time for testing so only the simplified redesign was torsion tested. The testing results were inconclusive. Due to the error from the deflection of the test matrix, absolute values of torsional stiffness could not be determined with confidence. These errors were not expected to be difficult to deal with if the stiffness of the simplified design were to be compared proportionally to the stiffness of the ribbed design, because the error was expected to be similar for both tests. Due to the error in the testing, the design cannot be said to have succeeded or failed, so it is not known if a redesign is necessary. MAXON intends to do their own torsion test of the simplified design. MAXON expressed concerns about the assembly of the design, and indicated that they would not use the design as it stands, or as it is currently assembled. MAXON did however, want to see how a spoke design would affect the stiffness of the shaft. The final simplified design is incomplete because it cannot hold the current knife hardware used by MAXON, no new knife hardware was designed.

15.3 Future Recommendations

Manufacturability and testing of a design are important details of any design. A design may be manufacturable, however the manufacturing cost may be too high, and one may not have access to the tools or skills need to build the design. Designers should keep in mind who is fabricating the design, and what they are willing to do and at what cost. Also, the means and method of testing should be established and constructed as the part is being designed. This allows designers to know how the test will be performed early, so that there is sufficient time to test, and to rebuild the test matrix, or redesign the test entirely if there is a problem with the test matrix, or if the data is not sufficient to draw a confident conclusion. The simplified design would require better testing to say with confidence how it would perform if implemented. The design would not have to be simplified if the group had a better idea of what MAXON was willing and able to manufacture early on. It is highly recommended that the method of building the design is well-known while it is being designed, and the method of testing is decided on, and prepared before the design is completely built.

16 Operation

The shaft, produced at full-scale, is designed to operate at high speed and acceleration. It experiences a maximum torque up to 1,600 foot-pounds, and speeds up to 40rpm. It is also designed to operate for up to ten hours per day, and is designed with a factor of safety to withstand fatigue. The rotation of the shaft is controlled by an electric motor, the control system of the motor was not part of the groups design obligations. No operator's manual, nor assembly manual was produced. The shaft is a single part within a larger machine and does not warrant an operator's manual unto it's own. With accurate data on the strength limits of the part, and its stiffness, torque limits, speed limits, and operating time limits could be determined. The shaft is assembled first by attaching the spokes to the thin inner shaft. One bolt goes through the center of the spoke and shaft, and a nut on the end holds the bolt in place. Bolts then go through holes in the skin of the tube, and are threaded into tapped holes on the top of the spokes. Next the slugs slid into position by slots that the skin of the tube fits into. Finally, the seam where the slugs meet the skin of the tube is welded for a secure connection.

17 Maintenance

17.1 Maintaining the Cutter blade

The cutter blade proposed by the capstone team would offer little to no maintenance required other than what is already required my MAXSON's current system. This would mean the only real maintenance required would be the periodic inspection and adjustment of cutter blade, by use of the set screws on top of the blade section, in addition there may be some sharpening or replacement of the blade if it becomes damaged or worn out. In the end the bottom blade proposed by the design team has little to no maintenance required and the only real maintenance would be on the actual blade that is added to the assembly proposed by the design team.

17.2 End of Service Life

The design team had calculated fatigue and failure rates for many parts of the assembly and designed them around a nearly infinite fatigue life. In reality it would fail at some point however this is predicted to be much later in its service life than the actual machine that uses the cutter blade. The cutting blade would last as long as the machine running it and would most likely be thrown out and replaced when the machine is discarded at the end of its service life. One disposal method of this is to simply have the machine picked up and scrapped so that it may be recycled and reused, or to see if MAXSON, would like the machine back and reuse and recycle any of the parts that were not worn out over its lifespan, however most if not all pieces would be outdated by the time the machine and cutter blade are no longer useful and need to be replaced.

18 Additional Considerations

18.1 Economic impact

Economic impact for Maxson is larger than one would think for the prototype design that Spin to Win came up with for Maxson Automatic Machinery. The prototype is lighter with less inertia, which in turn means that Maxson can purchase a smaller, less powerful motor, but the drawback of the design is that it costs more to machine and adds more machining time. Because of this the economic impact on the machine is small. The offset of a smaller motor will theoretically yield a little profit compared to the added machining cost.

A global scale the economic impact is also large for Maxson. Maxson is the last sheeting company in North America that is privately owned. They are competing with a lot of overseas companies in Asia. Because the Spin to Win design is lighter and can move at faster speeds than the current design Maxson has it is very plausible that this could impact the sheeter market globally and not just nationally in North America.

18.2 Environmental impact

The environmental impact of the Spin to Win design is minimal for Maxson. The model can be made in Maxson's facility and the waste material left over can be disposed of with the proper means or re-used for parts depending on the size of the waste material. That said the extra machining time added to the Spin to Win design compared to the current design will most likely add more energy consumption for Maxson. Their overall utilities that they pay could increase with this added machining time. Besides those two factors, there is no other environmental factor because there are no chemicals used in making the design.

18.3 Societal impact

The societal impact of the Spin to Win design is minimal. The spokes are one of the pieces of the Wheel and Spoke design, however, they can't be manufactured in house at Maxson's facility. The spokes are outsourced to a vendor allowing another company to make a profit off of the design besides Maxson themselves.

18.4 Political impact

The political impact of this design is non-existent because the design that Spin to Win came up with is for a piece of an already existing machine that follows standard government regulations. The wheel and spoke model is manufactured in house the same way the current model Maxson is using. The Wheel and Spoke design is an optimization/redesign of a current system, so the design was already formulated to stay within the existing regulations already in place for sheeter machine set-up.

18.5 Ethical considerations

With any project there are ethical considerations, and Spin to Win made sure that they kept those in mind when designing the Wheel and Spokes model for Maxson. While designing the model the team made sure that it would be strong enough to withstand the forces involved in the system. For that reason, the Team came up with a center piece that was strong enough to be built around, and that center piece was the spoke design. The Team tested the 3D design of the spoke in Abaqus to make sure it was up to the task as well as testing the real life model in the Instron machine. In both cases the design held up like the Team had formulated theoretically, so with this information we knew that morally we had good design to hand to our sponsor.

Going forward from there the Team came up a modular design that yielded benefits stated specifically in the conclusion of the report. The assembly design was tested in Abaqus as well as the team tested the real life scale model in the Instron machine. Again the design performed as well as we theoretically formulated with our calculations. Our design was sound and should have an infinite fatigue life as stated in the life cycle section of the report. With this in mind the Team felt confident that we came up with a good product design that Maxson could implement in their existing machine.

18.6 Health, ergonomics, safety considerations

The Wheel and Spoke model poses no threat or benefit to the health of anyone in the company. No personnel will be handling the piece once it is fitted into the already existing machine. The only time the piece will come in connect with company personnel is during the building phase of the assembly. Otherwise the design posed no change to company personnels' health.

The Wheel and Spoke design has some ergonomic value to be considered. The design is modular for ease of assembly as well as disassembly in the off chance that a piece of the design needs to be swapped out. The design was also purposefully made to be circular in order to minimize crimping of the feed as plastic or paper is fed through the sheeter to be cut.

The safety considerations are all laid out in the regulations already established for the machine. Simple considerations like keeping hands out of the way of the moving blades. As described above in the ergonomics section, the design is circular having no sharp edges, so when personnel are carrying the piece their hands will not be cut. In theory the bolts that hold the design together are the weakest pieces, and they have a factor of safety of 3.5 for this application.

18.7 Sustainability considerations

The Wheel and Spoke design is highly sustainable. The entire assembly is made from steel and is held together by bolts and welding material usually consisting of a softer metal. Steel is highly sustainable because it is made from carbon and iron, which are both highly abundant on the planet. The bolts are also made from steel and can be purchased at any hardware store in the area or surrounding cities and towns if there were no hardware stores in the area. Bolts could also be purchased through vendors or online sites if the need arose.

Welding material is not as sustainable as the other materials. Welding is needed to join the slugs and the skin of the Wheel and Spoke assembly. It was MIG welded to be specific, and MIG welding utilizes inert gases such as Argon to adhere the metals to each other. Inert gases are expensive and not easily attainable, so it could pose a small problem for production of multiple cutters at once. However, since the design calls for spot welding and the use of a small amount of material it can be inferred that this will have little affect to the overall sustainability of the design.

19 Conclusions

In the foregoing, attention has been paid that the wheel and spokes assembly meets all of the criteria established in the Design Specifications. A detailed description of the way in which each target is met follows.

19.1 Basic Features

Speed Design specifications call for feed speeds of 1,000fpm and cutoffs at least 41" at that speed. This cutting cycle has been analyzed, and the involved torques were used to calculate system stresses. The weakest component, the main bolts, of the system is rated to survive these stresses with a factor of safety versus the fatigue endurance limit of 3.5.

Motor and reduction Design specifications call for a motor speed of 1,750 rpm, with a 3:1 reduction, and a shaft speed of 500 rpm. This reduction was determined through analysis to be impossible, and a 1.8:1 reduction was substituted. A standard, 1,750 rpm motor could still be used with this reduction in the lower ranges of its speed, but the motor selection is ongoing outside of Spin to Win's jurisdiction.

Cut quality Design specifications call for strict cut quality and cut reliability targets of 1) cut edges parallel with 0.1 degrees and 2) cutoffs all fall within 0.2% cutoff length. The factor controlling cut quality is the cutter stiffness, where local deflections and strain result in knife positions that are out of sync with the cycle. The stiffness of the wheel and spokes assembly is rated 40% above that of the current MAXSON cutter, which is known to meet these targets. By extension the wheel and spokes will meet them as well.

Feed stability Design specifications call for no crimps, tears, or catches in the feed in 10 hours of continuous operations. This requires a circular profile for smooth feed, without any jutting or obtruding pieces. The wheel and spokes profile is circular enough that the 3D-printed model rolls smoothly over a smooth surface, and is at least equal to the current cutter's profile in circularity.

Inertia The design specifications call for inertial reductions of both the top and bottom cutter. No inertial reduction has been designed for the top cutter, but the inertia requirement for the bottom cutter was 6,200 $lb - in^2$. This has been exceeded, and the actual inertia is rated at 4,413 $lb - in^2$. The reduction in the total system inertia is greater than if the planned reduction to the top cutter had occurred.

Vibrations The amplitude of the vibrations in an oscillating system are proportional to the mass in play, as higher mass yields a greater inertia and a greater amount of energy in each displacement. The spokes assembly has a 54% decrease in mass compared to the existing cutter, which means a 54% lower kinetic energy in the same oscillations. In addition, the composite nature of the assembly, with multiple pieces kinematically coupled, inherently damps any torsional vibrations that would pass through the system, unless all components can be brought to vibrate harmonically (which is unlikely for components with widely differing natural frequencies). Finally, the 20% inertial reduction of the system goes a long way toward permitting a smaller motor, which is the largest contributor to overall vibrations.

19.2 Life cycle

All components of the wheel and spokes have been designed for an infinite fatigue life. For the pinion reduction, the factor of safety for this fatigue life is small: 1.2 for the planned 5" engagement. As such this pinion will be a closely watched component moving forward. The highest-stress part of the wheel and spokes assembly, the main bolts, have been rated with a factor of safety of 3.5 versus the endurance limit. Therefore the maintenance requirement for the wheel and spokes cutter is theoretically zero, or at least no more than the current system.

20 Further Work

The most critical work for the continuance of this project is testing the already fabricated wheel and spokes model against the ribbed MAXSON model. If the wheel and spokes model shows better stiffness, as predicted, that means that the estimated savings in inertia and improvements in performance of the machine can be realized.

At that point a careful analysis of whether or not the achieved savings and improvements are worth the cost of the system is worthwhile. The production of the actual model has provided a good deal of information as to how difficult this process is. The costs are known to MAXSON, and assembly was successfully completed by a team of students for the scale model. Examining whether or not any motors exist on the market that would prevent savings with the resulting reduced inertia, or whether a market exists for the resulting improved performance would determine whether or not the design should move into production.

If the answer to these questions is no, then further work lies comparing the non-homogeneous assembly approach used in the wheel and spokes assembly to whatever advances are made in the art of rotary cutters for sheeting machines. It may in the end prove that cost reduction rather than performance improvement is the future direction of the industry. If not, the validity of this approach has been tested and examined in the foregoing report and can be compared to the achievements of other approaches that emerge.

21 References

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22 Appendices

22.1 MAXSON preliminary project plan

Scope of Project

The scope of the project includes a mechanical design, an electrical design and a program design analysis, evaluation and problem solution:

Mechanical Design

1. Analysis of existing cutter design
    (a) Calculation of inertia
        i. Revolvers
        ii. Blade and adjustment hardware
    (b) Gears
        i. Coupling
2. Review of revolver stiffness
    (a) Torsional forces caused by cyclic velocity
    (b) Impact of diameter and length (deflection)
3. Operational factors
    (a) Dynamic balancing
    (b) Cutting action and adjustment
    (c) Flow of material through the cutter
    (d) Cut quality
    (e) Gear strength
4. Recommendations to improve design based on value engineering
    (a) Introduction of a reduction between the knife motor and knife revolver
        i. Percentage
        ii. Design
    (b) Revolver
        i. Diameter
        ii. Form
    (c) Gears
        i. Width
        ii. Hardness

iii. Backlash adjustment
iv. Lubrication consideration

5. Layout and mounting of knife motor to cutters drive side housing (mounting plate)
6. Potential modification of cutter base

Tentative Schedule (through 31 December 2016)

30 September 2016

- Theoretical inertia involving the existing cutting section with a reduction calculated and provided to electrical design team member
- Identify potential mechanical design alternatives to explore revolver, (diameter, material, form) , gear (hardness, design, width, backlash) that would improve performance, reduce cost
- List of electrical suppliers contacted about application, capabilities and request for proposal
- ABB
- BAUMULLER
- BECKHOFF
- BOSCH REXROTH
- EMERSON
- HITACHI
- MITSUBISHI
- YASKAWA
- Various logic control programs of basic sheeter configuration and options provided for remapping a master logic control program
- Identify attributes of a well designed, intuitive, graphic control display

31 October 2016

- Initial layout of reduction within a bearing housing completed
- Vet list of mechanical design alternatives to proceed forward to prototyping
- Secure proposals from electrical suppliers
- Initial drafts of operator screens for HMI
- Address remote diagnostic capability and ability to export data

30 November 2016

- Mechanically design components for reduction
- Design approved mechanical prototyping alternatives
- Address gear lubrication issue
- Recommend list of electrical supplier finalists
- Create a ladder diagram for master logic control program

31 December 2106

- Order electrical package of drives, motors, logic controls and HMI
- Design mounting arrangement for cutter motors including cutter base
- Release mechanical prototype alternatives
- Begin programming logic control based on logic control supplier
- Begin programming HMI based on HMI supplier

22.2 APriori Cost Analyses

Below are the results from aPriori for costing the major components of the wheel and spokes assembly.

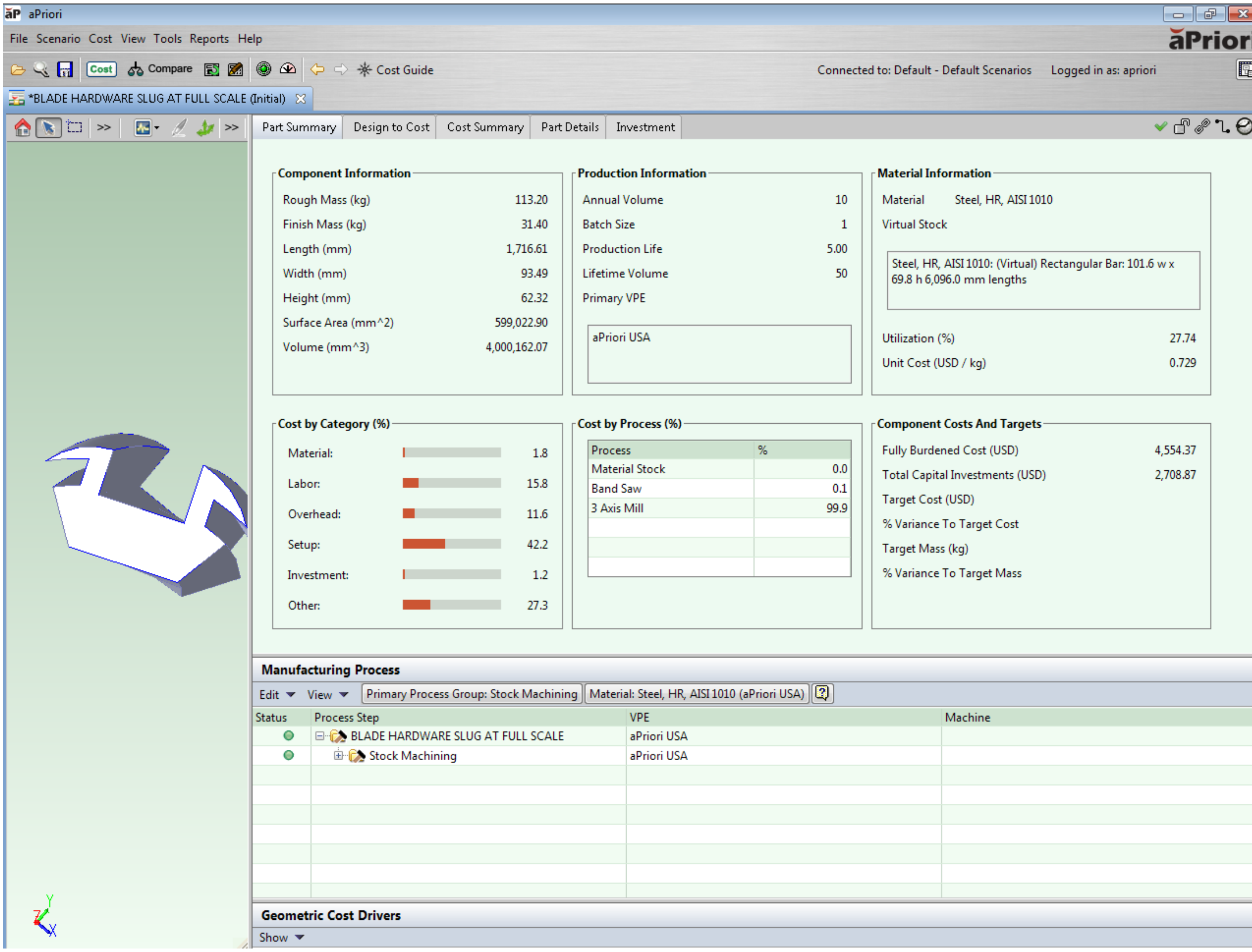


Figure 50: APriori Cost Analysis for the Blade Slug

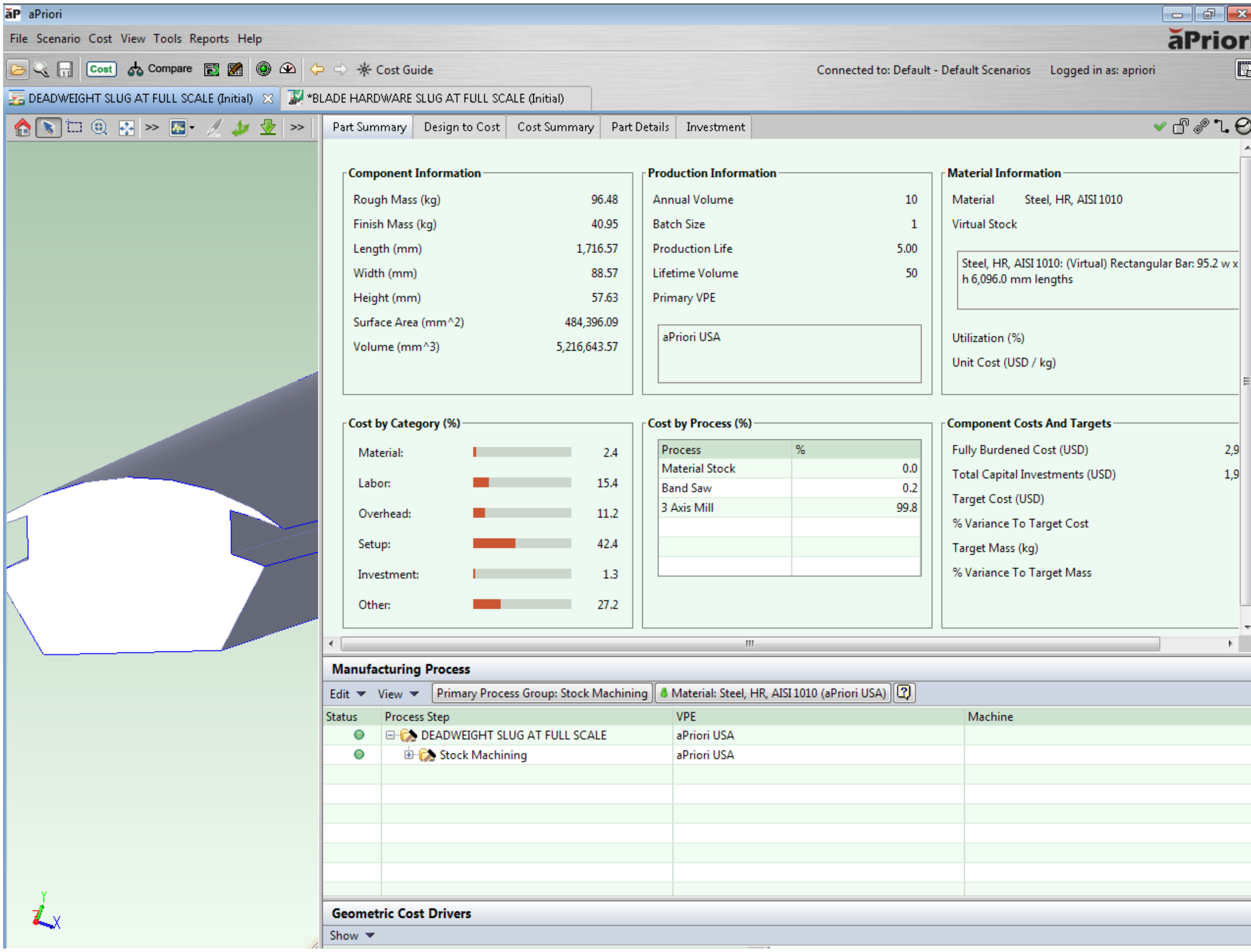


Figure 51: APriori Cost Analysis for the Deadweight Slug

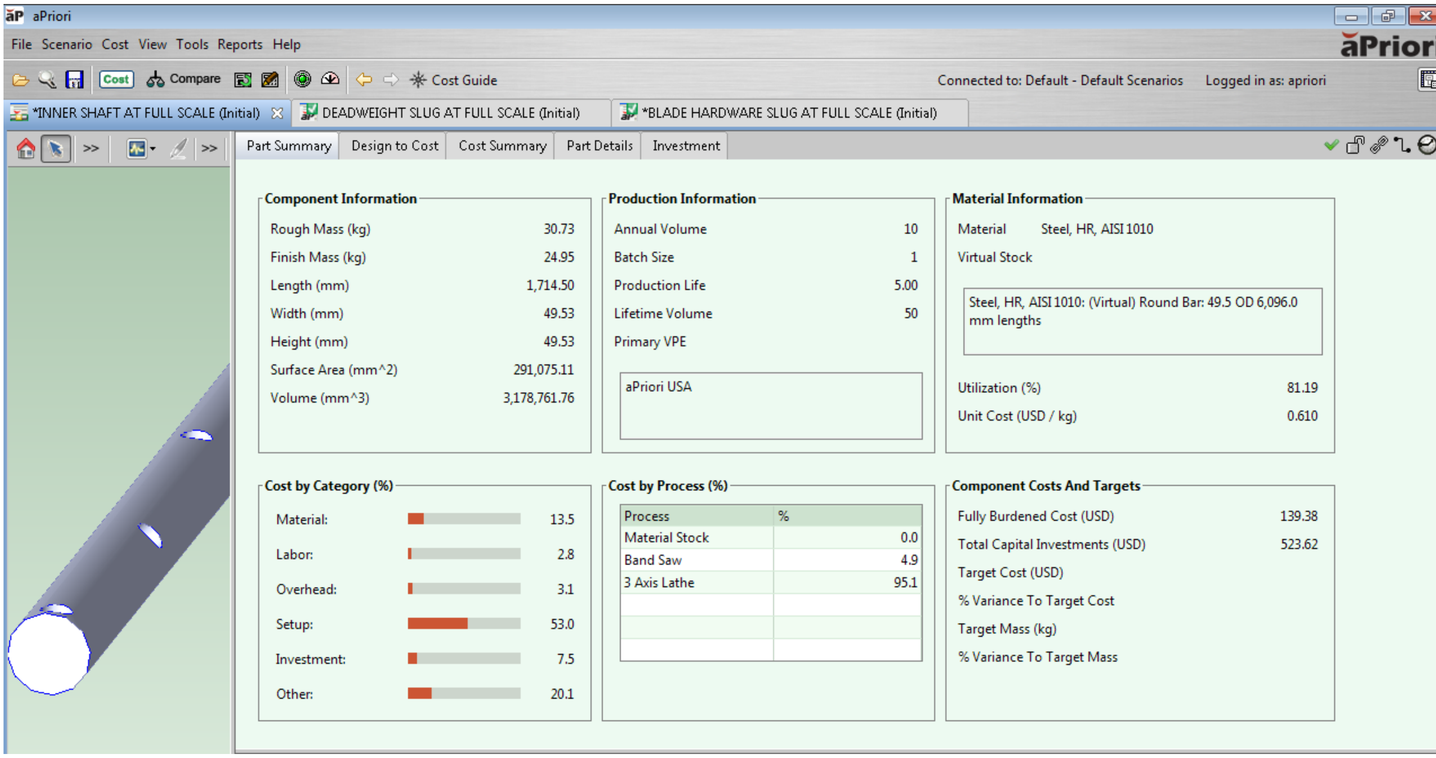


Figure 52: APriori Cost Analysis for the Inner Shaft

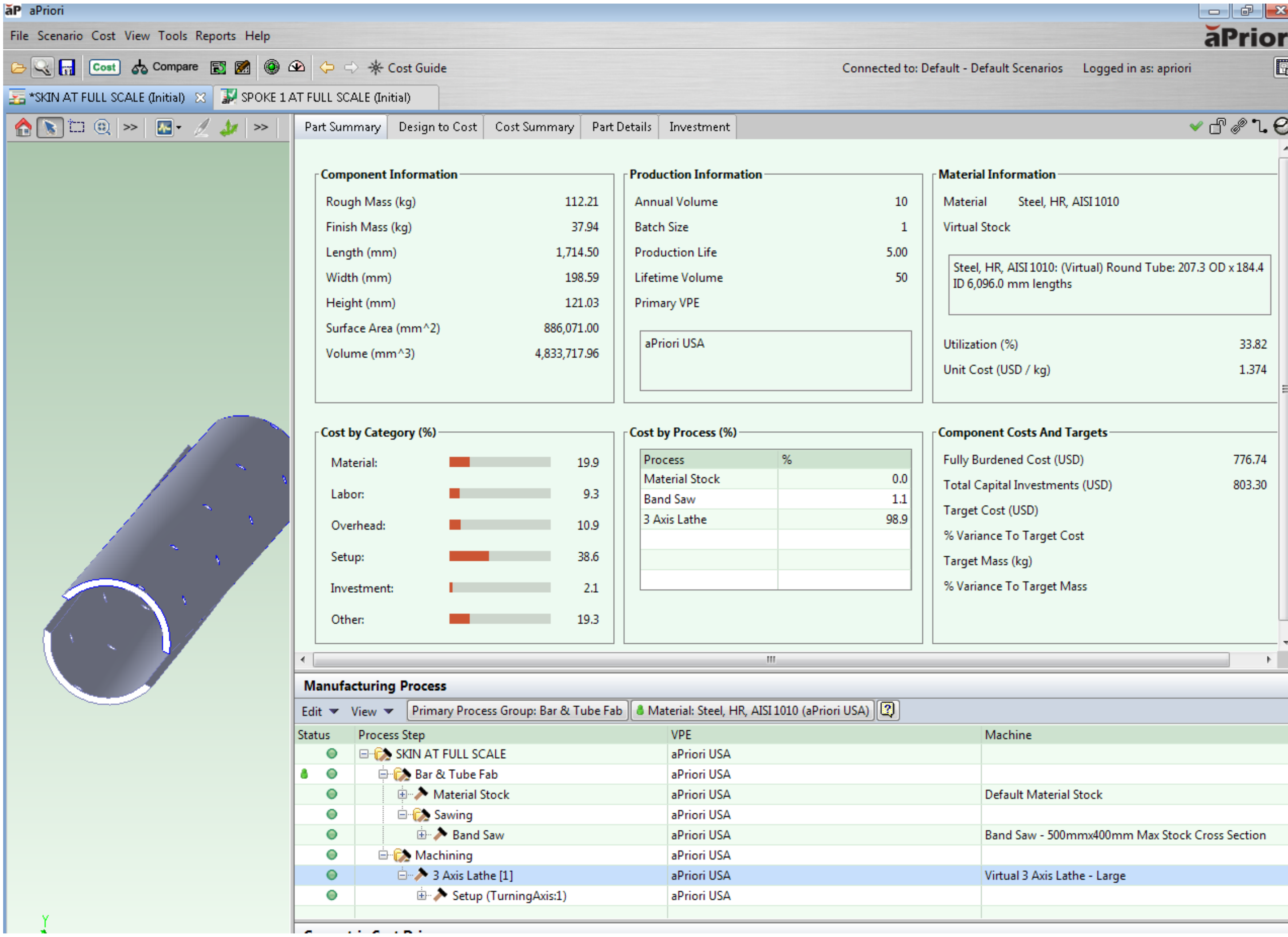


Figure 53: APriori Cost Analysis for the Outer Skin

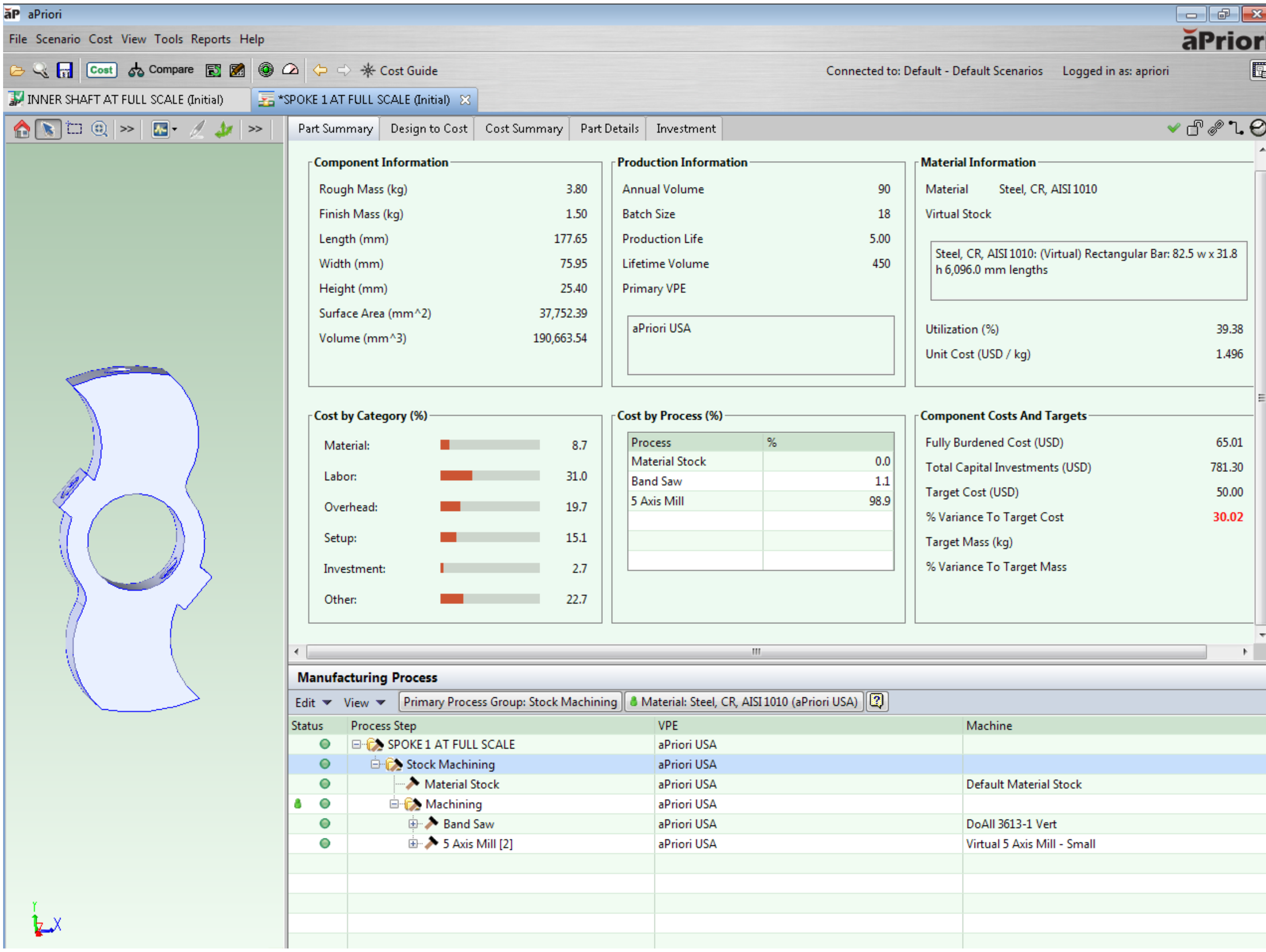


Figure 54: APriori Cost Analysis for Spoke Design 1

22.3 Spokes assembly detailed calculations

Polar and flexural moments of inertia

For a round tube with d_{inner} and d_{outer}, or for a solid round bar where $d_{inner} = 0$, polar moment of inertia is given by:

$$J = \frac{\pi}{32}(d_{outer})^4 - (d_{inner})^4 \tag{20}$$

The spokes assembly inner shaft has a diameter of 2", so its polar moment of inertia is given by:

$$J = \frac{\pi}{32}2^4 = 1.57in^4 \tag{21}$$

The outer assembly consists of a tube with an outer diameter of 8" and an inner diameter of 7", which composes about 275 degrees of the circle, together with two slugs. The slugs have a cross-sectional area of 3.26 in^2 at an average radius of about 3.5". So the total inertia of the outer assembly is given by:

$$J = \frac{\pi}{32}(8^4 - 7^4)\frac{275}{360} + (3.26)(3.5^2)(2) = 207in^4 \tag{22}$$

Both the bolts and threaded inserts are 3/4"-16 UNF threadings. Their tensile stress area (which is also used for shear resistance) is 0.373 in^2 as defined, in Reference [14], by the equation:

$$A = 0.7854(D - \frac{0.9743}{n})^2 \tag{23}$$

Both bolts and threaded inserts can be sheared also along their transverse cross-sections, where the area resisting shear is the rectangle built on the diameter of the bolt and the bolt's length. Both bolts have a 5/8" diameter; the threaded inserts are one inch long, and the main bolts are 3.25" from base to head.

The main bolts are also subject to torsion along their center. The polar moment of inertia of a rectangle about its center is given by the following equation [15], where the bolts' height is 3.25" and its base is its root diameter of 0.69":

$$J = \frac{bh(b^2 + h^2)}{12} = 2.06 \tag{24}$$

Both spoke arms have circular bases, but the length is approximately 2". Each spoke is 1.5" deep. The spoke can be approximated as a rectangle, so the flexural moment of inertia of the spoke arm cross-section is given by:

$$I = \frac{1}{12}bh^3 = 1in^4 \tag{25}$$

Its shear cross-section is simply the product of its cross-sectional base and height, 2" * 1.5".

The entire spoke can be subject to torsion when the pressures on its connecting bolts are not in the same plane. Its polar moment of inertia is the sum of that of the ring and two arms, where each arm is a 2" by 2.5" rectangle at an average radius of the arm center, 2.375".

$$J = \frac{\pi}{32}(2.5^4 - 2^4) + (2)(2.5)(2)(2.375)^2 = 59in^4 \tag{26}$$

paragraph*The flexural moment of inertia of the inner shaft is given by [15]:

$$I_x = \frac{\pi r^4}{4} = \frac{\pi 1^4}{4} = 0.785in^4 \tag{27}$$

The flexural momet of inertia of the outer tube is given by:

$$I_x = \frac{\pi}{4}(r_o^4 - r_i^4) = 105.5in^4 \tag{28}$$

Torsional deflections

For the central shaft and outer assembly, torsional deflections can be calculated simply from Equation 7.

The bolts, inserts, and spokes occur locally at 5.5", 12.5", etc. every 7" up to 61.5", and at different radii r. For these a local linear deflection is defined as the difference between the deflections at the front, nearest the fixed boundary, and the back, nearest the torqued boundary:

$$\delta_{local} = \delta_{front} - \delta_{back} = \frac{x_{front} - x_{back}}{67}\frac{r_{local}}{4.25}\delta_{max} \tag{29}$$

The local torsional angle of twist is then defined, with the component's radial length l, as

$$\theta_{local} = \arctan(\frac{delta_{local}}{l}) \tag{30}$$

Whereas the overall angle of twist is 4.7×10^-4 radians, the inserts go through only 2.1×10^-5, the main bolts 4×10^-6, and the spokes 1.05×10^-5 radians of twist, but within a drastically smaller length: 0.75", 0.75", and 1.5" respectively.

Bending deflection

Each spoke is transverse to the previous one and has a different flexural moment of inertia. In the case that the first spoke is vertical, the second horizontal, and so on, their flexural moments of inertia can be modeled as a combination of the spoke ring as a hollow circle and the two arms as two rectangles. In the vertical orientation the I is 13.6 in^4, in the horizontal it is 1.85 in^4.

The total weight of the spokes assembly is found in Table 5, 466.9 pounds mass (which equals 466.9 pounds force in Earth's gravity). The load per inch is then $\frac{466.9}{67}$ or 6.97 pounds per inch. However, the calculation is simplified if the load is modeled as a point load, then a corrective factor used to translate the result to that which would be found with a distributed load; the ratio of maximum deflections between the two cases is a fixed $\frac{1}{48} : \frac{5}{384}$.

The shear force in such a beam is constant and equal to half the load P, and the moment M increases linearly to its maximum at the center, such that

$$M_x = \frac{P}{2}x \ (0 < x < 33.5) \tag{31}$$

The local deflection results from integrating Equation 16 twice, stepwise, using the appropriate value of I for each segment. The spokes exist from 5.5" to 7", 12.5" to 14", and so on every 7":

$$y = \int_0^{5.5}\int_0^{5.5} \frac{M^2}{E(I_{shaft} + I_{tube})}dx + \int_{5.5}^{7}\int_{5.5}^{7} \frac{M^2}{E(I_{shaft} + I_{tube} + I_{spoke})}dx \, etc... \tag{32}$$

22.4 CAD Drawings

Here are the CAD drawings for the prototype that was modeled in Solidworks. Putting the images in the appendices to keep the report as neat as possible.

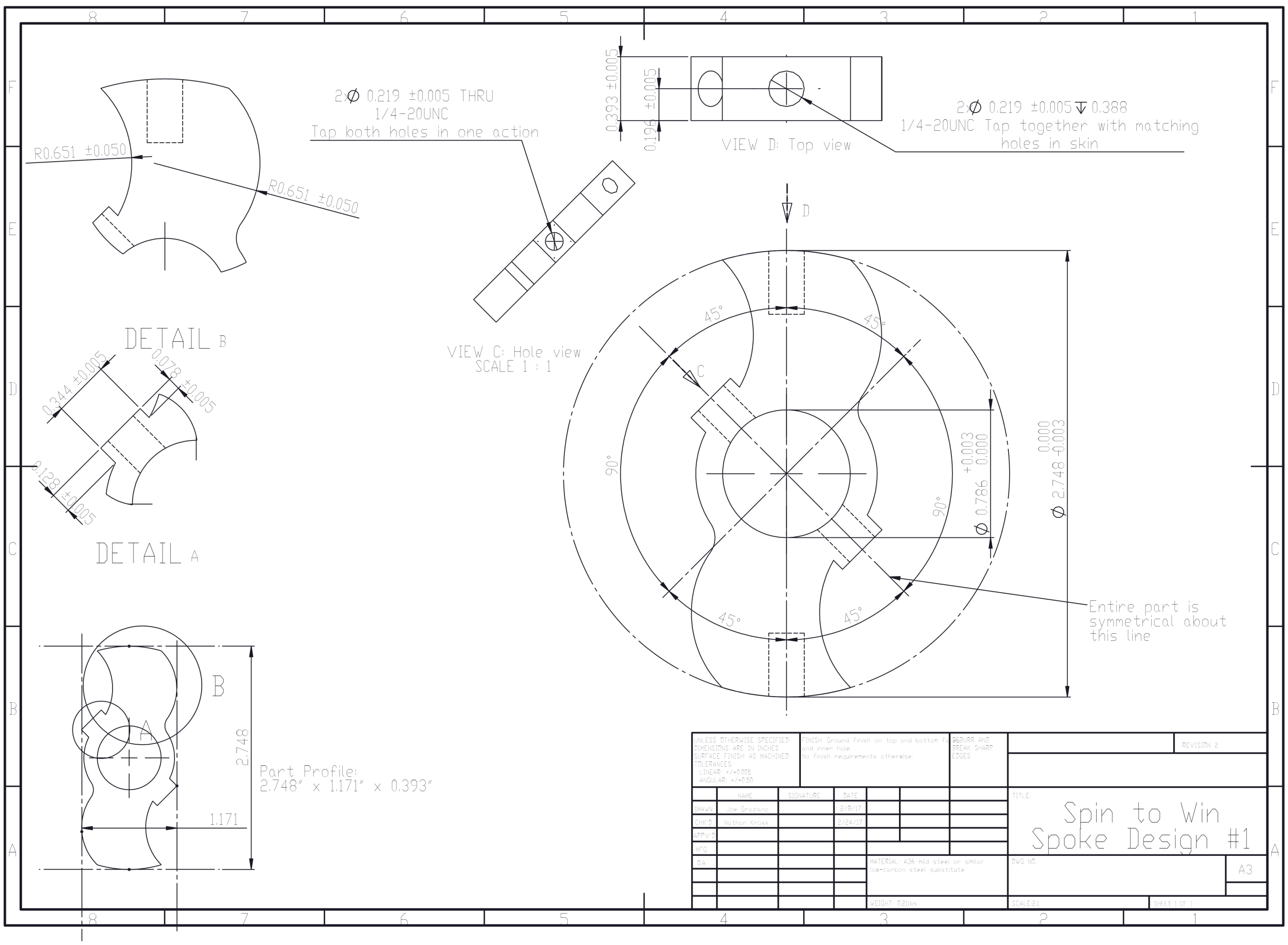


Figure 55: Spoke Design 1 3D CAD drawing

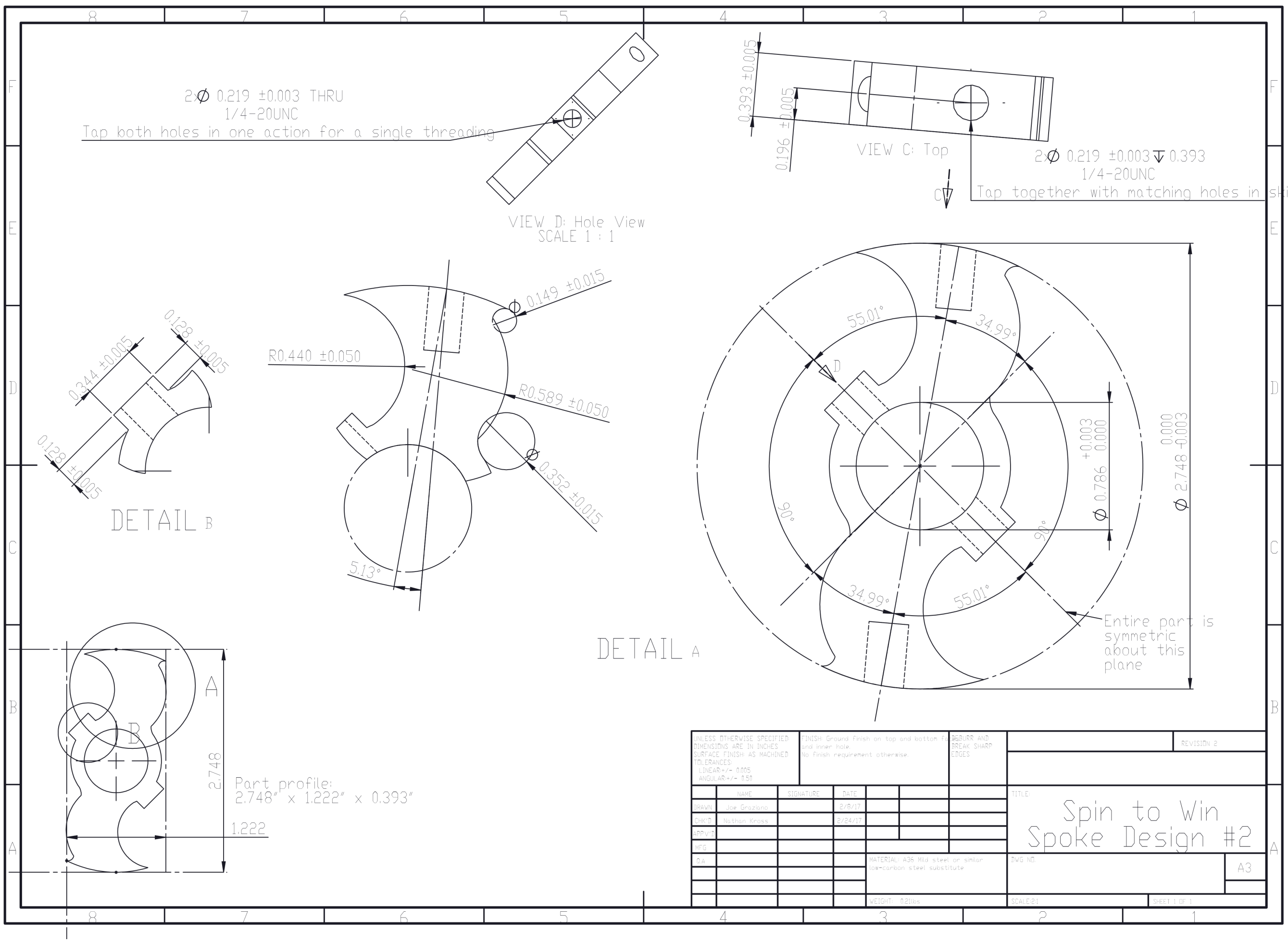


Figure 56: Spoke Design 2 3D CAD drawing

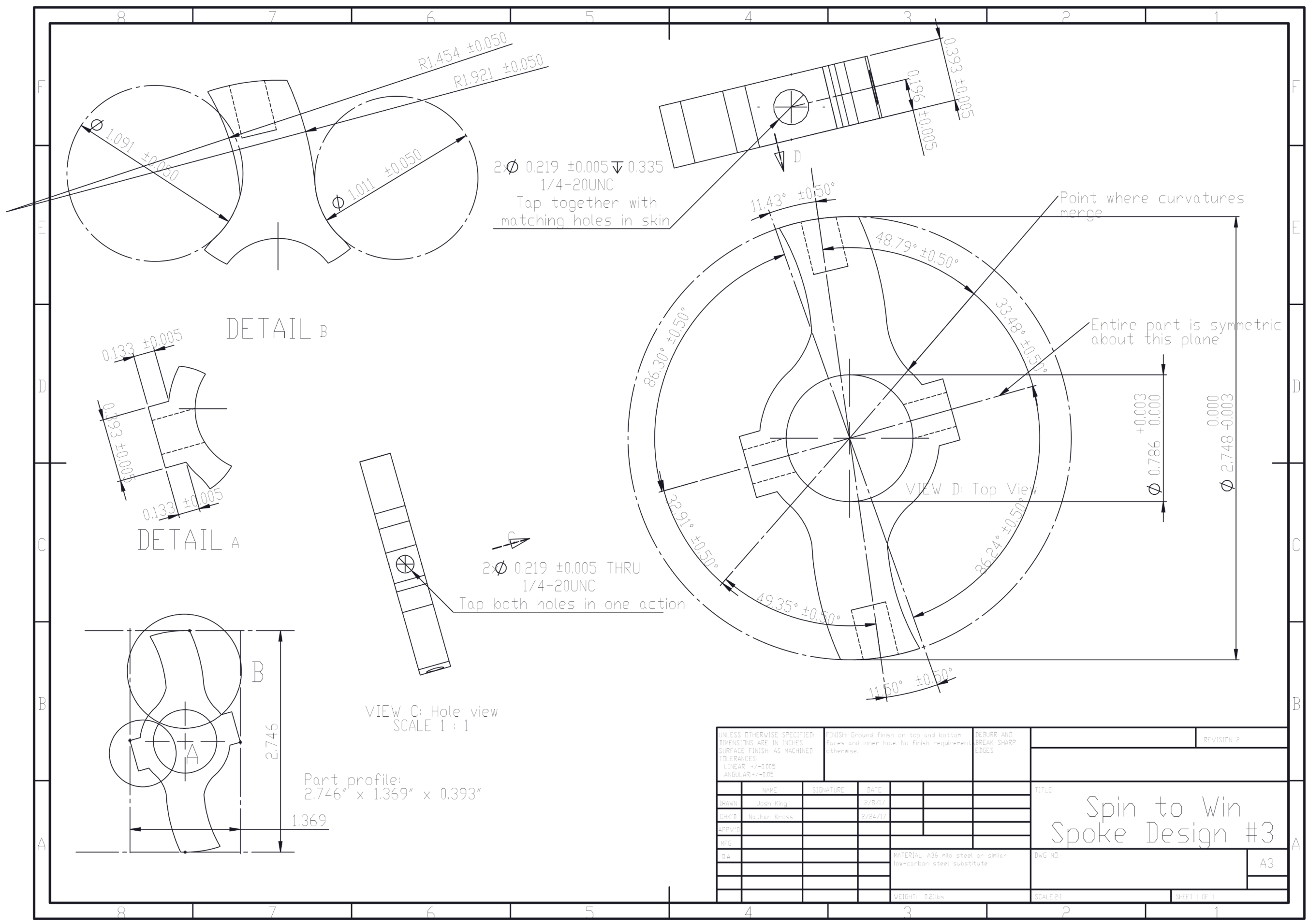
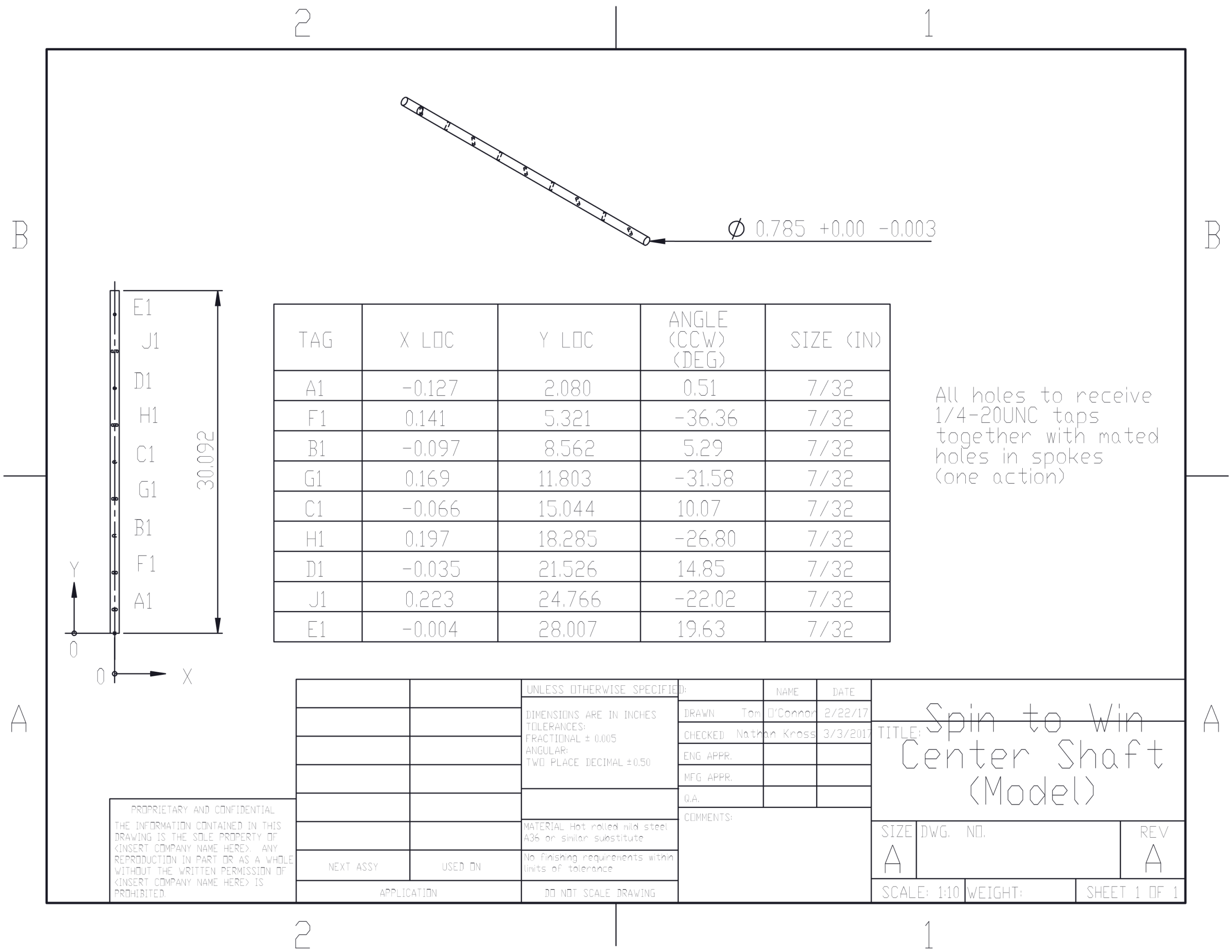


Figure 57: Spoke Design 3 3D CAD drawing



TAG	X LOC	Y LOC	ANGLE (CCW) (DEG)	SIZE (IN)
A1	-0.127	2.080	0.51	7/32
F1	0.141	5.321	-36.36	7/32
B1	-0.097	8.562	5.29	7/32
G1	0.169	11.803	-31.58	7/32
C1	-0.066	15.044	10.07	7/32
H1	0.197	18.285	-26.80	7/32
D1	-0.035	21.526	14.85	7/32
J1	0.223	24.766	-22.02	7/32
E1	-0.004	28.007	19.63	7/32

Figure 58: Center shaft 3D CAD drawing

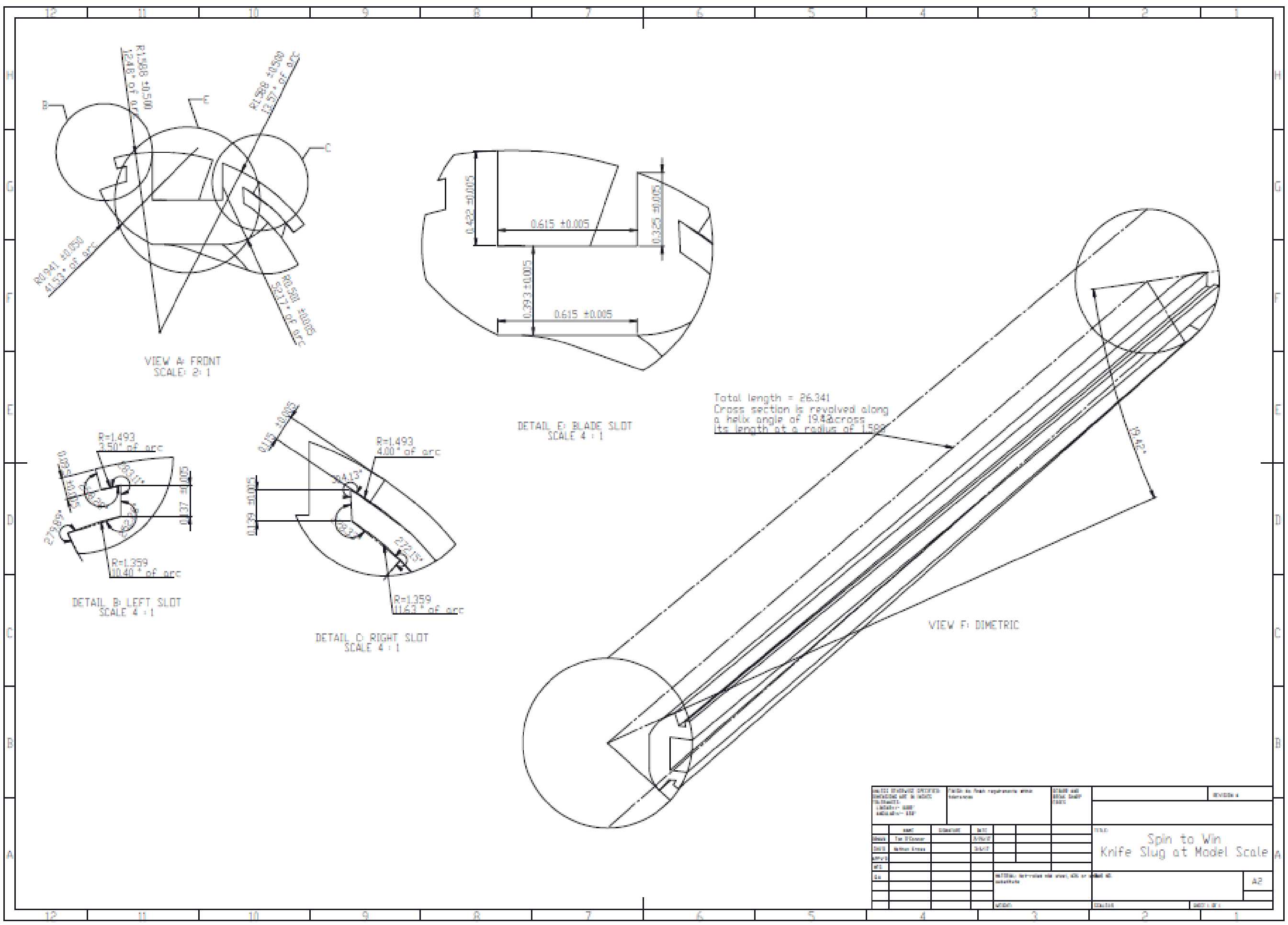


Figure 59: Knife Slug 3D CAD drawing

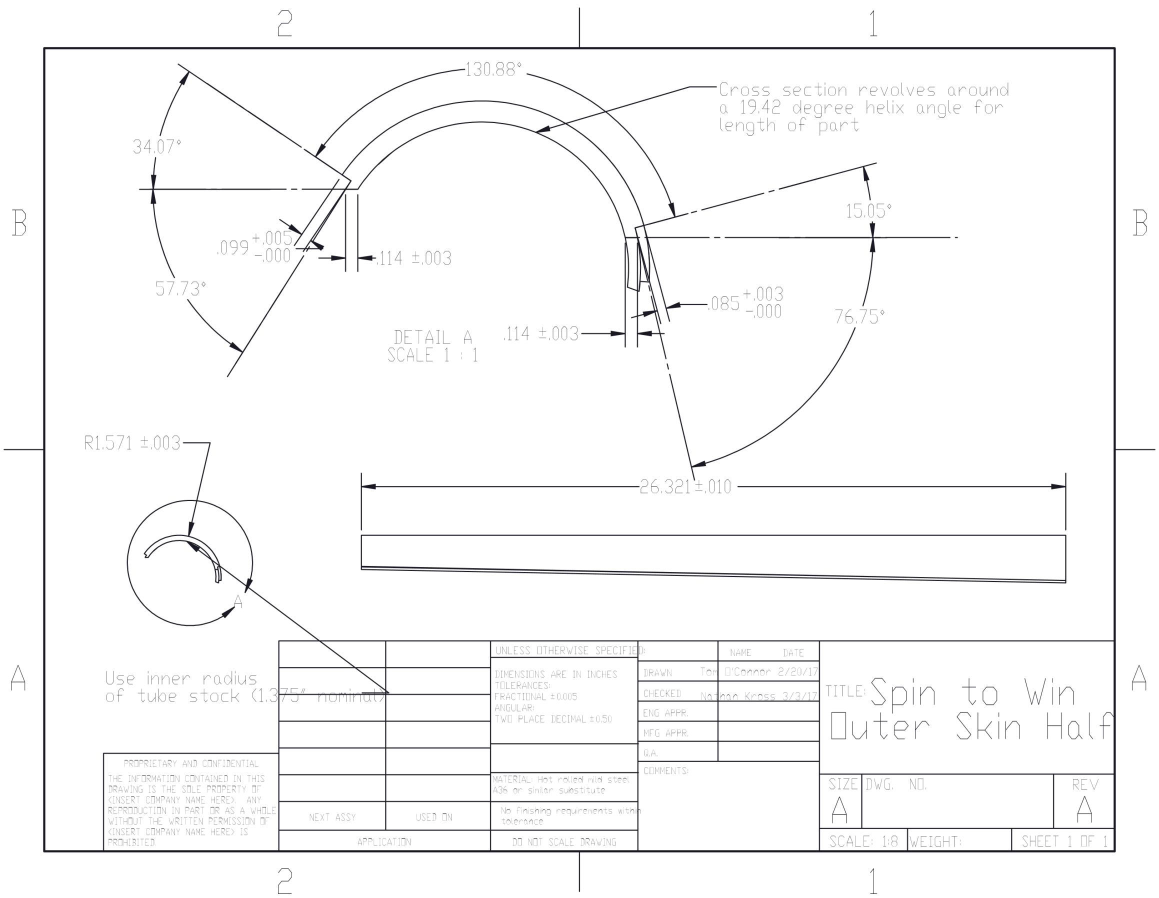


Figure 60: Outer Skin 3D CAD drawing

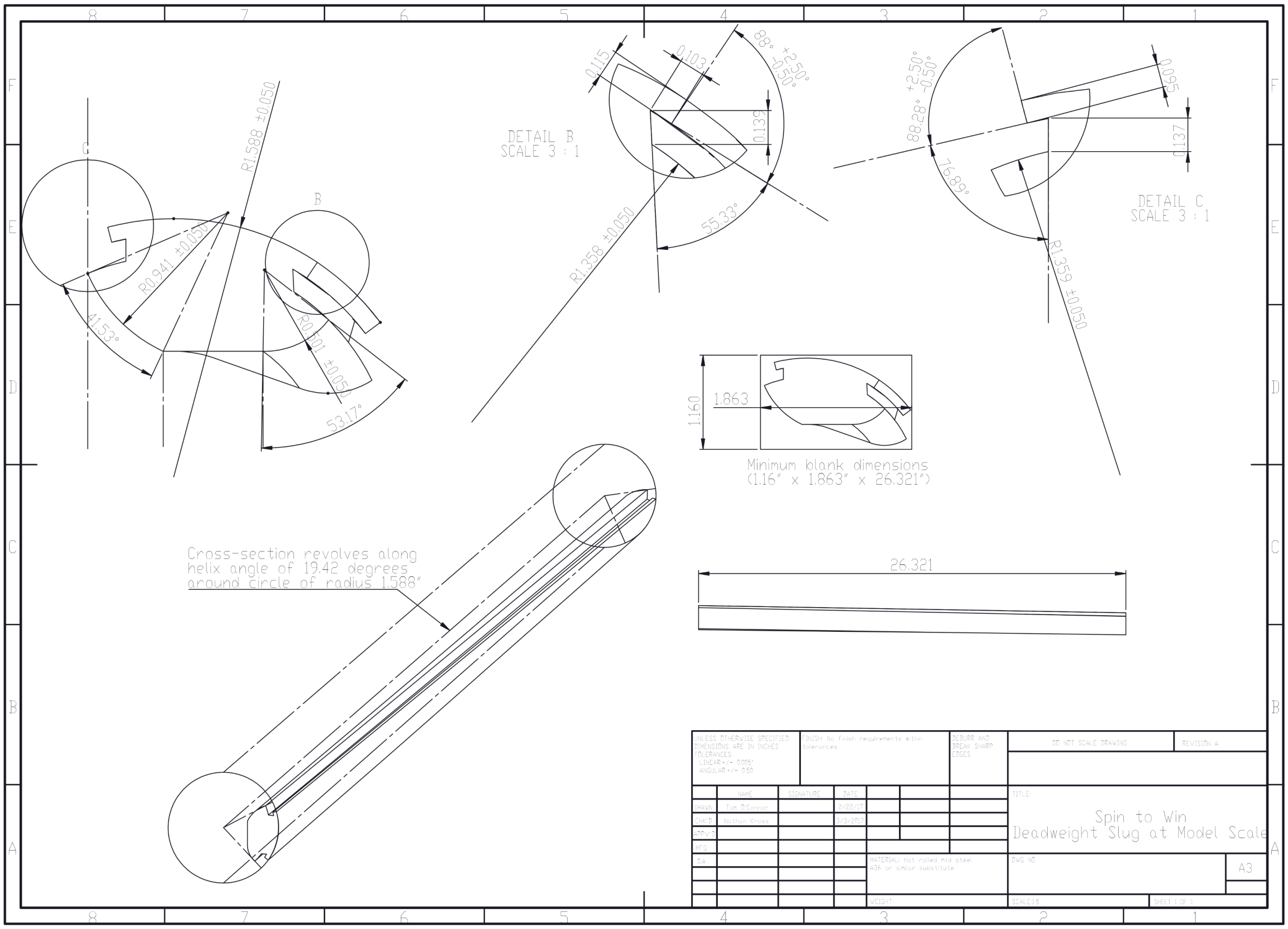


Figure 61: Bottom slug 3D CAD drawing