Executive Summary

This paper is a complete and concise outline of the quantitative and qualitative analyses performed in order to generate a proposal for the fabrication of a new design for an adjustable pitch propeller, as commissioned by the sponsoring company Growth Innovations, through the collaborative efforts of the FAMU - FSU College of Engineering.

The original problem presented to the engineering design group was to generate a cost effective, robust, and reliable design for an apparatus with the capability of altering the pitch of a propeller blade in motion from an efficient full forward position to an efficient full reverse position. The problem was constrained to a weed-less propeller design requiring less force to shift and greater directional force than the company's current design. For this reason both qualitative and quantitative analyses were performed on the existing design submitted by Growth Innovations to the engineering design group.

The physical prototype of the adjustable pitch propeller system submitted by Growth Innovations was disassembled, catalogued, and modeled using a three dimensional CAD package. Qualitative inspection of the integrity the device components revealed areas that underwent component degradation due to excessive force loading in concentrated areas (minimally distributed loads). The result of the degradation was lack of efficiency and excessive force required to vary the pitch of the mechanism. Quantitative analysis of the catalogued components verified excessive stresses in the device due the concentrated friction, shear, and point loading.

Following analysis of the existing design, as well as consideration for the ease of assembly, a new design iteration was generated providing the following solutions to problems with the existing design. The stub shafts were removed from the mechanism and are now held in place by guide screws that constrain the movement of the shafts to one degree of freedom. The blades were separated from the stub shafts and are now mechanically restrained by screws. The track on the rotator tool was modified from a straight to a quarter circle design resulting in greater range of movement per required input. This greater range yields the ability to utilize a flat blade that requires less manufacturing time, resources, and thus money required for the entire assembly. The modular provides a solution to the difficulty of disassembly of the mechanism in the field. Finally, the sizes of the various components were increased to distribute loads over a greater area in order to reduce or eliminate component degradation due to stress loading. (Please refer to appendix IV for component drawings.)

The result of these changes is a new adjustable pitch propeller that is easy to manufacture, robust, and reliable. Also, cost analysis was performed which places the raw material costs for a single prototype at $1174.02 using ideal materials, and $678.36 using prototype only materials to minimize cost. (Please refer to appendix II for raw materials cost analysis)

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1.0 Introduction

In order to understand the need for an adjustable pitch propeller it is first necessary to familiarize oneself with the application of this device. Hunters and swamp boating enthusiasts at times find it necessary to hunt and explore uncharted waters. These waters are normally very shallow and occasionally extremely difficult to navigate. Weeds and mud are commonplace along with scattered stumps and stones lying below the surface. Currently, there are no adjustable pitch propellers on the market for this particular function; which limits navigating to only the forward direction. Therefore, if one encounters a situation where one can no longer continue in the forward direction, then one must physically reverse the boat. However, an adjustable pitch propeller would alleviate this problem by allowing the boat to transverse forward and backward. The company sponsoring this project had produced several prototypes of adjustable pitch propellers, but was unable to produce an adequate design.

The goal of the project was to redesign the adjustable pitch propeller given to our team by our sponsors, since our sponsors documented several problems with the prototypes which were previously generated. These problems included the following: difficult blade transition from full forward to full reverse, excessive wear on rotator tool due to point loading, complex assembly that made it difficult to replace parts and excessive vibrations which complicated the balancing of the propeller. Additionally, our sponsor specified that the redesign must be able to be attached to a ¾ inch drive shaft, housing must withstand 2,100 RPM and blades must be weed-less. Understanding the problems of the current design and the limiting conditions of the design commenced the design process.

2.0 Background

An understanding of the word pitch is crucial to understanding the need of an adjustable pitch propeller. The term pitch is defined as the distance that a propeller would travel in an ideal medium during one complete revolution. Therefore, as the pitch increases the distance traveled increases and vice versa. Pitch can be broken down into positive, neutral and negative pitch. A propeller with positive pitch is the forward distance a propeller would travel during one complete revolution and a propeller with negative pitch is the backward distance a propeller would travel during one complete revolution. A propeller with neutral pitch will provide neither forward nor backward translation. The ability of an adjustable pitch propeller to rotate from an optimum positive pitch to an optimum negative pitch is crucial for an adequate design as one will see in the original design.

The original design (see Figure 1) that was presented to the team was well conceived, but had a number of problems that needed to be fixed. The original design comprised of a hub, rotator tool, propeller blades and stub shafts. The rotator tool sat in the front of the hub and was directly connected to the drive shaft. The propeller blades were located in the back of the hub and were attached to stub shafts that contained pins that sat in the groves of the rotator tool. The hub would translate along the drive shaft causing the rotator tool to translate within the hub which caused the blades to rotate.

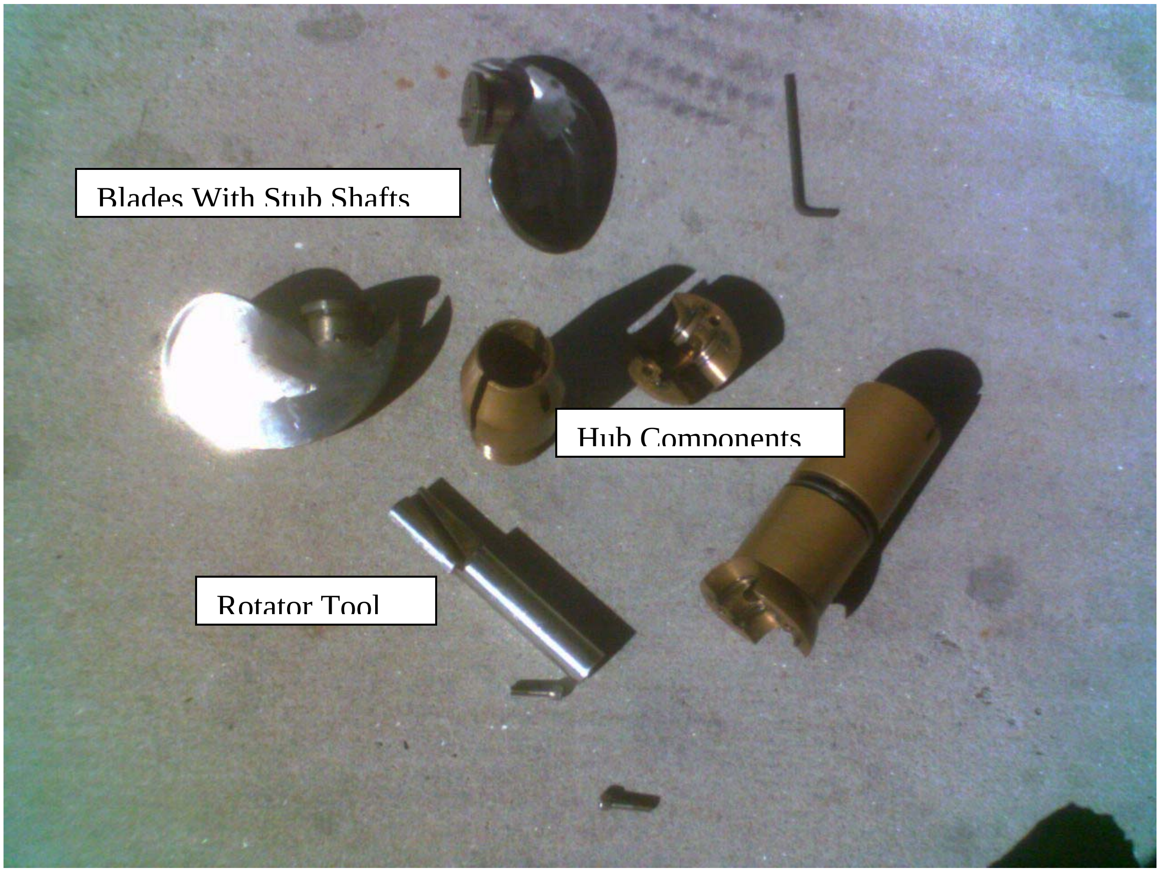


Figure 1: Image of Original Design

However, the rotator tool (see Figure 2) straight slot had a number of problems including: limited blade pitch rotation ability, extreme point loading and no locking positions. The blade pitch rotation ability was approximately 30-45 degrees; which was not enough to provide efficient full forward and full reverse capabilities. Additionally, having the blades directly attached to the stub shafts made it difficult for production and the ability to balance the blades along the rotational axis. These design problems guided the team in the creation of the needs assessment of our adjustable pitch propeller.

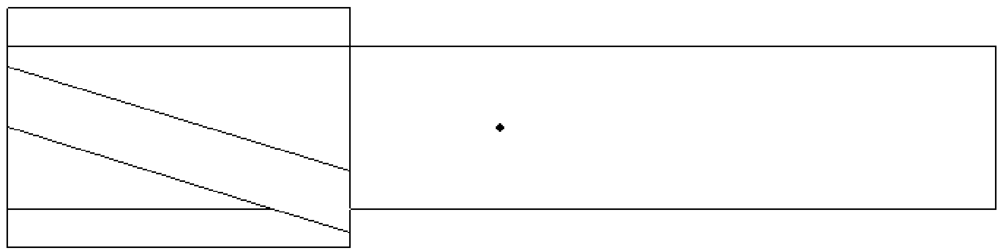


Figure 2: Straight Slot Rotator Tool

2.1 Needs Assessment

2.1.1 Variable Pitch Propeller

2.1.1.1 Must rotate to a full forward position

2.1.1.2 Must Rotate to a full reverse position

2.1.1.3 Should accommodate various set pitches between full forward and full reverse (if possible)

2.1.2 Blade Design

2.1.2.1 Blade design should be optimized for maximum forward efficiency following satisfaction of other constraints

2.1.2.2 Blade should allow for replaceable design

2.1.2.3 Each blade should be balanced about the connection/rotational axis

2.1.2.4 Each blade should have part to part balance about the hub

2.1.2.5 The design of the blade and hub must meet the weed-less blade constraint

2.1.2.6 Suggestion for final blade material should be wear resistant, moldable/cast-able polymer

2.1.3 Hub Design

2.1.2.7 Hub material should be highly corrosion and wear resistant

2.1.2.8 Hub should be small enough in diameter to avoid ventilation

2.1.2.9 Hub dimensions must accommodate shallow water running of the propeller

2.1.4 Rotational Tool Design

2.1.2.10 Tool must accommodate smooth and opposite rotation of each blade about the central stub shaft axis

2.1.2.11 Tool material must be hard enough to avoid pitting and corner wear

2.1.2.12 Tool must be designed such that point loading is minimized to avoid excessive wear during use.

The needs assessment of the adjustable pitch propeller served as the basis during the development of the product design specifications.

2.2 Production Design Specifications

2.2.1 Purpose

The purpose is to constrain the project by establishing product design specifications.

2.2.2 Scope

This document applies to the project as a whole and will serve as a guide through each stage of the project.

2.2.3 Product Title

Adjustable Pitch Propeller

2.2.4 Function

Provide full forward and full reverse pitches to thrust the boat in the respective directions. The propeller pitch also can shift to the neutral position.

2.2.5 New or Special Features

2.2.5.1 Smooth pitch transitions without the use of a transmission.

2.2.5.2 Minimal input force required to change propeller pitch.

2.2.5.3 Improved seal of entire apparatus.

2.2.5.4 Weed-less propeller blade that prevents weeds from accumulated on the propeller

2.2.6 Competition

There is no competition in the intended market for this product.

2.2.7 Intended Market

Outdoor enthusiasts who own Mudbuddy and GoDevil boats that want an adjustable pitch propeller versus the stock fixed pitch propeller that currently comes on these boats.

2.2.8 Relationship to existing products line

This is a start-up venture. No other products exist in the intended market.

2.2.9 Physical requirements

2.2.9.1 Apparatus must be able to be attached to a ¾'' drive shaft.

2.2.9.2 Durable housing that can withstand 2,100 RPM.

2.2.9.3 The pitch of the blade must be able to rotate a full 90° from full forward to full reverse.

2.2.10 Service environment

Apparatus shall be completely submerged in water.

3.0 Procedure

3.1 Qualitative Analysis

After reviewing the previous iterations of the Adjustable Pitch Propeller design and compiling background research it was determined that the current design had merit but required improvement by altering key aspects of the design. There were several flaws that limited the functionality of the propeller. Upon acquisition of the propeller the entire assembly was taken apart and each component of the assembly was reviewed for flaws and possible improvements. The parts most analyzed for change were the rotator tool, stub shaft and the outer housing. A list of problems was created for each individual part. Each problem could then be addressed in future iterations of the design. This allowed a more focused approach rather than trying to alter the entire system.

3.1.1 Rotator Tool

The rotator tool has two purposes in this assembly. The first is to spin the blades of the propeller to create forward force. The second is to adjust the pitch of the blades to shift from forward to reverse. The rotator tool in the original assembly was a blade on top of a thin cylinder. In each face of the blade was a straight track on a slight slant that was used to adjust the pitch of the propeller blades.

Upon review, the straight tracks were noted to allow only a 30 to 45 degree pitch change in the blades. This limited the performance of the propeller. Additionally, the material used in the construction of the rotator tool was too soft, allowing the vibration of the blade pins to pit the sides of the tracks. These pits in the sides of the tracks would then prevent the blade pins from sliding smoothly in the tracks when shifting from forward to reverse and back. Once the pitting occurred, the system as a whole would not operate until the rotator tool was replaced.

The straight configuration of the tracks also allowed the vibration of the engine to cause the blade pins to walk in the tracks, causing the propeller to shift itself out of full forward or full reverse. This required the user to maintain force on the shifter in order to keep the pitch of the blades constant. The straight track also made assembly of the entire system extremely difficult.

The cylinder portion of the rotator tool also caused problems in the operation of the propeller. The cylinder would deform in a twisting manner under the significant torque created by the drive shaft and would bind in the bushing. This binding prevented any translation of the blade pins thus no change in the pitch of the blades was allowed.

3.1.2 Stub Shaft

In the original design, the stub shaft, the blade and the pin were all part of one component. The blade, cut from a previously existing propeller, was welded directly to the stub shaft off center with the pitch pin protruding out of the other side of the shaft. This method of construction prevented the balancing of the blades across the propeller axis since the two blades were not of equal weight. This imbalance caused significant vibration during operation. The one component assembly also required the replacement of the entire assembly if the blade was damaged in any way. Additionally, the pin which was directly attached to the stub shaft in the original design was of the same material as the stub shaft. This design called for the replacement of the stub shaft and blade if the pin were to fail.

The original design also had a flange on the edge of the shaft closest to the pitch pin. This flange sat in a track on the outer housing, causing the top of the entire system to be assembled all at once which increased complexity of assembly.

3.1.3 Outer housing

The outer housing was split into a top and bottom and, due to the inlaid track for the stub shaft flanged, would require the disassembly and reassembly of the entire top of the system to replace any component.

3.2 Quantitative Analysis

Numerical analysis of the adjustable pitch propeller was performed on the critical components of the system to ensure the reduction of stresses, as well as generating dimensions that would be the foundation for sizing the design with a desirable safety factor, assuming a desirable safety factor to be a Factor of Safety Greater than 1. The calculations for the various safety factors are based off of worst case scenarios. The worst case scenario was assumed as the maximum number of subassembly components, with the maximum probable load, loaded on to the minimum number of supporting members (i.e. one complete stub shaft assembly with a blade being held into the adjustable pitch propeller assembly on one guide screw shaft).

Primary concerns for the stress calculations revolved around the loads transmitted to the adjustable propeller assembly via the input torque generated from the driving motor. In order to generate a possible, but not probable, maximum force, assumptions were made involving the worst case ideology as described before. The resulting maximum linear forces were then numerically distributed through the assembly components in order to generate dimensions that would result in a desirable safety factor. Please refer to appendix I for complete numerical calculations.

3.2.1 Linear Force, Input Torque, Drag

To derive the maximum force that would be applied by the motor to the propeller, the combination of items and weights were as follows:

- Koehler 27 horsepower motor
- Go Devil Boat 18'x 38" (Largest allowable boat for the above motor)
- 850 Lb load in the boat (Highest allowable manufacturer recommended weight)
- 19 miles per hour (MPH) (Estimated maximum linear speed based on manufacturer spec.)
- 2100 revolutions per minute (RPM) (Estimated maximum rotational velocity)

The maximum force was then calculated using the force, power, velocity relationship in equation (3.2.1) resulting in a reasonable and probable linear force based on the input values.

$$P = F \cdot V \qquad \textbf{(3.2.1)}$$

The input torque of the motor was calculated using the horsepower, torque, and revolution velocity relationship in equation (3.2.2).

$$\text{Horsepower} = \text{Torque} \cdot \frac{\text{RPM}}{5252} \qquad \textbf{(3.2.2)}$$

A rough estimate for the drag coefficient of the system, including the boat, motor, and load was also found.

3.2.2 Stub Shaft and Blade Centripetal Force

Once the motor is up to maximum RPM, and the boat reaches its maximum cruising speed, the major concern for stress in the assembly arises from the centripetal forces generated by the stub shaft, and blade, subassembly as the forces are attempting to pull the subassembly out of the adjustable pitch propeller assembly. The total mass of one stub shaft and blade subassembly was estimated using three dimensional cad analysis of each part in the subassembly using the properties of mild steel. Summation of the total mass of one stub shaft and blade subassembly allowed for the calculation of the outward force generated due to the rotating masses utilizing equation (3.2.3).

$$F_{cent} = \frac{m_{total} \cdot \omega^2}{r} \qquad \textbf{(3.2.3)}$$

3.2.3 Constraining the Cap screw Diameter

The four cap screws that hold the entire top end of the assembly are key components as they are also the guide rails that constrain the movement of the stub shafts to one degree of freedom. For this reason special consideration was taken to generate a suitable dimension that would allow the screws to perform their various functions under maximum loading. Again the worst case scenario was used by assuming the complete subassembly load was supported by one screw.

The assumed material for the calculations involving the retainer screws for the stub shaft and blade subassembly is 303 stainless steel as it is the material that the purchased screws are manufactured from. The allowable stress considered for the purposes of calculating the minimum diameter was constrained to the yield stress of the material that the screws were made from. The allowable stress value was then halved to double factor of safety for the calculated minimum diameter in order to ensure a minimal deflection of the guide screws under loading. Manipulation of the base stress formula (3.2.4) yielded a value for the cross sectional area, and with further manipulation the diameter, of the cap screw in order to achieve the desired safety factor.

$$\sigma = \frac{F}{A} \qquad \textbf{(3.2.4)}$$

The shear stress on the retainer screw was chosen as the criterion for justification of the Factor of Safety and the minimum core diameter of the screws. The allowable shear stress was defined as three quarters of the yield stress of the material. Utilization of the shear stress equation (3.2.5) and the Factor of Safety formula (3.2.6) provided suitable answers for the justification of the diameter chosen for the cap screws.

$$\tau_{max} = \frac{4 \cdot V}{3 \cdot A} \qquad \textbf{(3.2.5)}$$

$$S_{fSSretainer} := \frac{\sigma_{allow}}{\tau_{max}} \qquad \textbf{(3.2.6)}$$

3.2.4 Constraining the Translation to Rotation Pin Diameter

The pins that allow the translational movement of the rotational tool to be transferred to a rotational movement by the stub shafts is at the heart of the assembly and is thus a critical assembly component both to the stub shaft and blade subassembly. For this reason it was selected as a main component for analysis. The material chosen for analysis of this part was 304 stainless steel as it is a viable material for use on this part because of its durability and corrosion resistance.

The manner in which the diameter of the pins was constrained is the same as listed in section 3.2.3 regarding the diameter of the cap screws. Slight manipulations of the formulas (3.2.4), (3.2.5), and (3.2.6) were used in order to generate a cross sectional diameter of the pin that would result in a desirable safety factor. Furthermore, the worst case scenario was used in this calculation set also as the entire linear load was assumed to be supported across the cross sectional area of one pin.

3.2.5 Constraining the Hole Depth for the Retainer Cap Screws

As mentioned in section 3.2.3, the cap screws that act as the restraining members for the adjustable pitch propeller assembly serve multifunctional purposes. One of those purposes is retaining the top section of the hub assembly. Since the screws are only held in place by the threading at the bottom of the spherical section of the hub special attention was given to the contact threading between the lower spherical section of the hub and the retaining cap screws.

In order to ensure that the top section of the hub does not come off during operation of the propeller the strength of the threading that retains the cap screws was analyzed. There is a previously defined correlation for steel screws in aluminum which states, “For a steel screw in aluminum the depth must be at least twice the major diameter of the screw. This will ensure that the load required to strip the threads will be greater than the tensile strength of the screw.” Taking this statement into consideration, the direct load bearing capability of the threads was not calculated, but formula (3.2.7) was used instead.

$$\mathrm{Depth}_{\mathrm{Screw}} = 2 \cdot d_{\mathrm{screw}} \qquad \textbf{(3.2.7)}$$

3.2.6 Constraining the Blade Thickness

No numerical calculations were done to analyze or constrain the blade thickness. Instead, Finite Element Analysis (FEA) was performed over a current thickness range that is normally used in production of weed-less blades.

3.2.7 Constraining the Blade Retainer Screw Diameters

No numerical calculations were done to analyze or constrain the diameter of the blade retainer screws. The calculations for the retainer cap screws will be used as a substitution for this as they undergo more strenuous loading. Also the thread depth of the blade retainer screw in the stub shaft assembly was not numerically analyzed as tear out of the blade retainer screws is not anticipated due to lack of tensile loading of the screw.

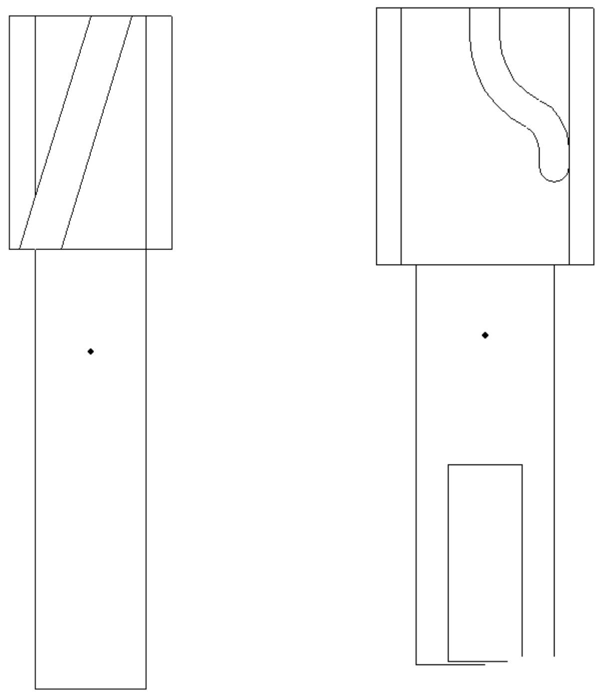
4.0 Results

4.1 Qualitative Results

The components most crucial to the spinning of the propeller as well as the adjustment of the pitch of the blades were given priority. Parts less crucial to the system were reviewed last. Each component was analyzed independently to identify modifications that would improve efficiency of the propeller (see section 3.1). Each modification was prioritized and then analyzed to ensure compatibility with the remaining components of the propeller.

Next, the subsystems were reviewed to ensure that, following modification of the individual components, the subsystem assemblies would still function properly.

Finally, the entire system was analyzed and reviewed to ensure that the system was safe, functional and feasible, and ensuring all changes would not cause a problem with the assembly as a whole.



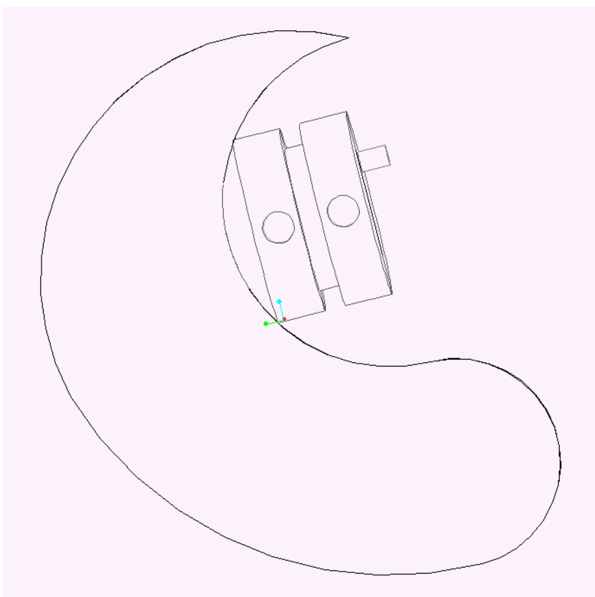
4.1.1 Rotator Tool

The rotator tool was the first component analyzed. This tool connects directly to the drive shaft and adjusts the pitch of the blades during the translation of the outer housing. After noting the problems with this part in section 3.1, it was determined that the track was the most crucial change that needed to be made to the rotator tool.

The track was changed from a straight track to a curved track. Using the distance from the center of the tool to the edge as a radius, the track was curved down a full quarter circle. This design would allow for maximum rotation of the blades, a full ninety degrees, along with a greater transfer of force from the translation of the outer housing to the change in pitch of the blades. The main purpose of this track is to transfer a translational force into a moment. Knowing this it was determined that a greater distance would be needed when the translational input force was more perpendicular to the rotational output force. This greater distance would allow the user to apply a more distributed force when the transfer of forces was less efficient. While it would need less distance when the relationship between translational input force and the rotational output force was more parallel thus the transfer of forces would be more efficient. The quarter circle design allowed for this, making the translation smoother and with a more distributed requirement of applied force.

To attenuate the effects of the vibration noted in section 3.1.1, lengths of straight track were placed at the beginning and the end of the quarter circle to provide locking positions at full forward and full reverse. The face of the rotator tool was also increased from 1.5 to 1.69 inches to allow for a greater radius of the track which allowed greater ease in shifting from full forward to full reverse.

To reduce the twisting deformation of the cylindrical base and, therefore, the binding in the bushing, the diameter of the cylindrical base was increased from 1.0 inch to 1.25 inches. The material used in the final design was also modified to one that was stronger and less elastic.



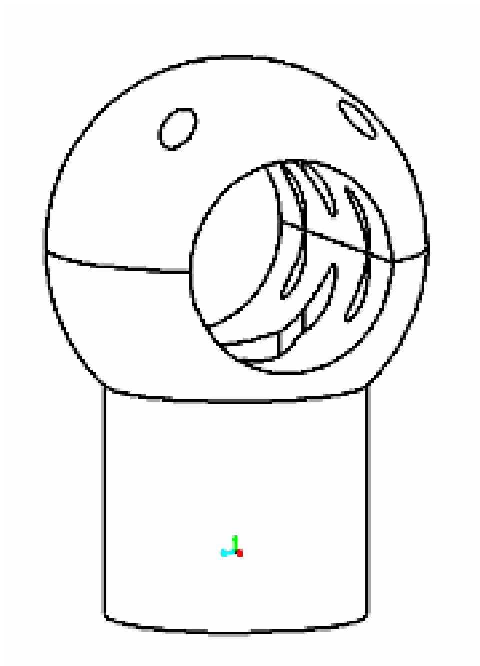
4.1.2 Stub Shaft

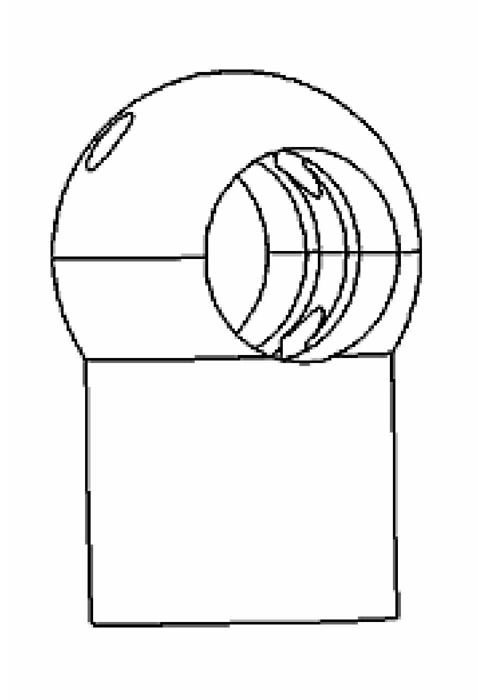
The stub shaft was the next component in the assembly to be modified. After noting problems with this component in section 3.1.2, the first modification was to break it into separate components. This separation of the stub shaft into the blade, the pin and the stub shaft, would allow for easier manufacturing and repair of the system as a whole. The original design was difficult to manufacture due to the complexity of the design and expensive to repair.

The first step was to separate the pin from the shaft. Since the pin was composed of the same material as the stub shaft, the pin was weak. By separating the pin from the shaft, a stronger material could be used for the pin without incurring the increased cost of constructing the entire shaft and pin from the same material of increased strength. Since cost was such a problem with the pin material it was decided that the pin was to be removed from the stub shaft allowing a cheaper resolve. This allowed the pin to be any material at minimal cost to the product and could also be replaced without having to replace the entire stub shaft and blade.

The blade was also removed from the stub shaft to allow for easier construction and replacement of the blades. By removing the blades from the original stub shaft, the stub shaft was converted from one component to an assembly, allowing for ease in replacement of damaged blades along with ability to change blades if a different performance of the boat was desired. Since the old component was so difficult to balance it was determined that the new blades needed to be manufactured together or from the same mold. A flat blade shape was chosen using the outline of the original blade. It was determined that it was unnecessary to add pitch to the blade when the purpose of this design is to allow for the controlled adjustment of the pitch. Having flat blades would allow for the blades to be cut simultaneously from stock ensuring they were the same weight.

The flange on the stub shaft that allowed for a change in pitch of the blades without the stub shaft falling out was removed. The track was moved from the outer housing to the stub shaft. By replacing the flange with a track in the stub shaft, bolts could be used to hold the stub shaft in place while still allowing for rotation. This change would allow for the stub shaft to be removed from the outer housing without taking apart the entire top of the system. It would also make reassembly of the system easier. Essentially the stub shaft could be removed, any problems could be repaired and it could be reinstalled into the assembly without having to take the entire assembly apart.





4.1.3 Outer Housing

The outer housing was the last part to be modified. After reviewing the problems list in section 3.1.3 it was determined that the track was the primary problem. This track was removed and bolts added to hold the new stub shaft in place. The size of the sphere at the top was also increased to allow for the larger rotator tool and the base cylinder was also increased to allow for the wider base of the rotator tool. Changes to the rest of the assembly were made only to allow for size changes.

4.2 Quantitative Results

This section expresses the results of the quantitative analysis as shown in section 3.2. Please refer to appendix I for complete numerical calculations.

4.2.1 Linear Force, Input Torque, Drag

Based on the values represented in section 3.2.1, the total weight of the boat and its load was found to be 1,280lbs. Evaluation of formula (3.2.1) resulted in a linear force of 532.9lbs necessary to push or pull the weight of a fully loaded craft at top speed. This value is important as it is the basis for all of the calculations as it represents the fully stressed state of the motor, craft, and adjustable pitch propeller.

The input torque from the motor was found to be 67.5ft-lbs based on formula (3.2.2). Since there is no transmission attached to the motor, this torque is transmitted directly to the input side of the adjustable pitch propeller and stresses the joint at the end of the rotator tool. Equation (3.2.3) was evaluated with the known weight and force to generate a drag coefficient of 0.416.

4.2.2 Stub Shaft and Blade Centripetal Force

The centripetal force generated by the rotating mass of the stub shaft and blade subassembly was found to be 791.2lbs. This load acts distributed across two of the retaining cap screws, and was a determining factor for the constraining diameter of the screws.

4.2.3 Retaining Cap Screw Diameter

Based on formulas (3.2.4) through (3.2.6) a minimum core cross sectional area of 0.045 square inches is necessary to provide enough strength to ensure that the Factor of Safety for the cap screw is at least 2. The 0.045 square inch cross sectional area yielded a minimum diameter of 0.241 inch. Since 0.241 inch is an uncommon size, the size chosen for fabrication of the propeller assembly will be 0.25 inch which is readily available from a wide number of vendors.

Calculations were done under the worst case scenario utilizing formula (3.2.5) resulting a maximum shear load of 10,750psi, which is a high stress for a single component under normal situations. However, increasing the diameter of the cap screw increased the Factor of Safety for this particular component to 2.4, which is considered a desirable safety factor as described in section 3.2.3. In addition to having a desirable safety factor, the high availability of screws in this diameter will aid in reducing the number of custom components, and thus cost, of the adjustable pitch propeller assembly.

4.2.4 Translation to Rotation Pin Diameter

Based on formulas (3.2.4) through (3.2.6) a minimum core cross sectional area of 0.031 square inches is necessary to provide enough strength to ensure that the Factor of Safety for the cap screw is greater than 1. The 0.031 square inch cross sectional area yielded a minimum diameter of 0.197 inch. The size chosen for fabrication of the propeller assembly will be 0.25 inch diameter which is a readily available stock size from a wide number of vendors.

Calculations were done under the worst case scenario utilizing formula (3.2.5) resulting a maximum shear load of 14,470psi. However, based on an allowable shear stress of 22,880psi (three quarters of the yield stress of the material chosen) a safety factor of 1.58 was calculated which meets the requirements of a desirable Factor of Safety as defined in section 3.2. In addition to having a desirable safety factor, the high availability of stock in this diameter will aid in reducing the necessary manufacturing time, and thus cost, of the adjustable pitch propeller assembly.

4.2.5 Engaged Hole Depth for the Retaining Cap Screws

Based on the justification as stated in section 3.2.5, the threaded depth of the holes into which the retainer cap screws are fixed must be at least twice the major diameter of the screws themselves. For this reason, a minimum threaded depth of 0.50 inch must be present to ensure a secure hold for the steel cap screws in the aluminum hub.

4.2.6 Blade Thickness

The thickness for the blade has been set at 0.25 inch as it is the thickness of some blades that are currently on the market which are used with the same motor and boat that this product is being designed for. This thickness was found to have a suitable factor of safety through the use of Finite Element Analysis (FEA).

4.2.7 Blade Retainer Cap Screw Diameter

The blade retainer cap screws will be purchased with a cross sectional diameter of 0.25 inch because the loading is less severe than that of the main retainer screws which were justified at a cross sectional diameter of 0.25 inch. Also the high level of availability of screws in this size will reduce the overall cost of the assembly by reducing the number of custom components.

5.0 Conclusions

The results obtained in this project were derived from the qualitative and quantitative analyses performed in the project procedure. Optimization analysis of the rotator tool showed that a quarter circle path would provide the greatest propeller pitch change; which was the most important parameter of the rotator tool. The propeller blade was also removed from the stub shaft to allow for easier construction and replacement of the blades. Finally, the entire system size was increased to reduce point loadings that could be detrimental to the life of the system.

Pending approval of our project proposal, the next steps include: order/receive necessary materials, build a prototype and begin testing the prototype.

6.0 Appendix I: Force Analysis

Force Analysis For Go-Devil Motor with APP

Calculations are based on the Go-Devil 18' x 38" Boat found at http://www.godevil.com/18x38.html

Boat Weight $W_{boat} := 430 \cdot lbf$

Max Load $Load_{max} := 850 \cdot lbf$

Total Weight on Water $W_{total} := W_{boat} + Load_{max}$ $\boxed{W_{total} = 1.28 \times 10^{3}\ lbf}$

The motor considered for calculation purposes is the 27 Horsepower Koehler found at http://www.godevil.com/27hp_KOHLER.html

Motor Power $P_{motor} := 27 \cdot hp$

Estimated Speed $V_{estimated} := 19 mph$

Power Formula $P = F \cdot V$

Linear Force Exerted $F_{linear} := \dfrac{P_{motor}}{V_{estimated}}$ $\boxed{F_{linear} = 532.895 lbf}$

Estimated Torque on Input Shaft

Calculation based on horsepower to torque conversion as shown.

$$Horsepower = Torque \cdot \frac{RPM}{5252}$$

$$Torque_{input} := 27 \cdot \frac{5252}{2100} \qquad \boxed{Torque_{input} = 67.526\, ft \cdot lb}$$

Total Estimated Drag Coefficient Based on Weight and Force

$$\mu := \frac{F_{linear}}{W_{total}} \qquad \boxed{\mu = 0.416}$$

Based on these calculations the propeller opposes a linear force of 532.895lbf.

Prop Speed

$rev := 360 deg$ $Speed_{prop} := 2100 \dfrac{rev}{min}$ $Speed_{prop} = 1.319 \times 10^{4} \dfrac{rad}{min}$ $\omega := 1.319 \cdot 10^{4} \cdot \dfrac{rad}{min}$ $\boxed{\omega = 219.833 \dfrac{rad}{sec}}$

$$\omega \cdot 3.5 \cdot in = 64.118 \left(\frac{ft}{s}\right)$$

Centripital Force Formula $F_{cent} = \dfrac{m_{total} \cdot \omega^{2}}{r}$

Estimated Blade Weight $m_{prop} := 1.056 \, lb$

Estimated Stub Shaft Weight $m_{stubshaft} := .75 \cdot lb$

Total Mass $m_{total} := m_{prop} + m_{stubshaft}$

Blade weight estimated by integrated material properties to 3-d blade model on Autodesk Inventor

Radius of Travel

$r := 3.5 \cdot in$ **Center of Hub to Center of Blade Mass**

Outward Force on Stub Shaft due to Centripital Accelleration

$$F_{cent} := \frac{m_{total} \cdot (\omega \cdot r)^2}{r} \qquad \boxed{F_{cent} = 791.199\,lbf}$$

Minimum Retainer Cap Screw Cross Sectional Area and Diameter

Cross Sectional Area Formula **Stress Formula** **Factor of Safety Formula** **Desired Factor of Safety**

$$A_{capscrew} = \frac{\pi \cdot d^2}{4} \qquad \sigma = \frac{F}{A} \qquad S_f = \frac{\sigma_{all}}{\sigma} \qquad S_f = 2$$

Screw Material and Available Sizes Taken From Small Parts, Inc.
http://www.smallparts.com/products/descriptions/shdx.cfm

Screw Material: 303 Stainless Steel

303 Stainless Properties from MatWeb http://www.matweb.com/search/SpecificMaterial.asp?bassnum=Q303A

$\sigma_{303all} := 34800\,psi$ **Allowable Stress**

$$\sigma := \frac{\sigma_{303all}}{2} \qquad \sigma = 1.74 \times 10^4\,psi$$ **Stress for Safety Factor of 2 based on single shear. (estimate)**

$$A_{capscrew} := \frac{F_{cent}}{\sigma} \qquad A_{capscrew} = 0.045\,in^2$$ **Cross Sectional Area of Capscrew**

$$d_{capscrew} := \sqrt{\frac{A_{capscrew} \cdot 4}{\pi}} \qquad \boxed{d_{capscrew} = 0.241\,in}$$

For a factor of Safety of 2 the minimum core cross sectional diameter of the capscrew must be 0.241"
Additional Screws will most likely be purchased in the same diameter and hex size for ease of assembly.

Shear on Retainer Screws for Stub Shafts

The maximum shear stress in a solid, circular cross-sectional beam is, at the neutral axis considering double shear.

$$\tau_{max} = \frac{4 \cdot V}{3 \cdot A} \qquad \tau_{max} := \frac{4 \cdot F_{cent}}{2 \cdot 3 \cdot \left[\frac{\pi \cdot (.25 \cdot in)^2}{4}\right]} \qquad \boxed{\tau_{max} = 1.075 \times 10^4\,psi}$$

Assuming that the shear strength of the material is approx 75% of the tensile strength

$\sigma_{allow} := (34800\,psi)\cdot .75$ $\boxed{\sigma_{allow} = 2.61\times 10^4\,psi}$

Factor of Safety for Retainer Screws

$S_{fSSretainer} := \dfrac{\sigma_{allow}}{\tau_{max}}$ $\boxed{S_{fSSretainer} = 2.429}$

Minimum Pin Diameter for Stub Shaft

It is assumed that the worst case linear force will be entirely on one of the two blades.

Cross Sectional Area Formula **Stress Formula** **Factor of Safety Formula** **Desired Factor of Safety**

$A_{pin} = \dfrac{\pi\cdot d^2}{4}$ $\sigma = \dfrac{F_{linear}}{A_{pin}}$ $S_f = \dfrac{\sigma_{all}}{\sigma}$ $S_f = 2$

Screw Material: 304 Stainless Steel

304 Stainless Properties from MatWeb http://www.matweb.com/search/SpecificMaterial.asp?bassnum=Q304L

$\sigma_{304all} := 34800\,psi$ **Allowable Stress**

Force Radius from Center **Pin - Stub Shaft Base Distance**

$r_{Force} := 3.25\cdot in$ $r_{SSDB} := .3\cdot in$

It is assumed that the linear force will be translated directly to the pin on the opposite end of the stub shaft.

$F_{pinload} := (F_{linear})$ $F_{pinload} = 532.895\,lbf$

$\sigma := \dfrac{\sigma_{304all}}{2}$ $\sigma = 1.74\times 10^4\,psi$ **Stress for Safety Factor of 2 based on single shear.**

$A_{pin} := \dfrac{F_{pinload}}{\sigma}$ $A_{pin} = 0.031\,in^2$ **Cross Sectional Area of Capscrew**

$d_{pin} := \sqrt{\dfrac{A_{pin}\cdot 4}{\pi}}$ $\boxed{d_{pin} = 0.197\,in}$

For a factor of Safety of 2 the minimum core cross sectional diameter of the pin must be 0.211". This value may be modified due to updated product dimensions.

Shear on Translation Pin

The maximum shear stress in a solid, circular cross-sectional beam is, at the neutral axis. (Based on a 0.25" dia pin)

$$\tau_{max} = \frac{4 \cdot V}{3 \cdot A} \qquad \tau_{max} := \frac{4 \cdot F_{pinload}}{3 \cdot \left[\frac{\pi \cdot (.25 \cdot in)^2}{4}\right]} \qquad \boxed{\tau_{max} = 1.447 \times 10^4\ psi}$$

Assuming that the shear strength of the material is approx 75% of the tensile strength

$$\sigma_{allow} := (30500\,psi) \cdot .75 \qquad \boxed{\sigma_{allow} = 2.288 \times 10^4\ psi}$$

Factor of Safety for Pin

$$S_{fSSretainer} := \frac{\sigma_{allow}}{\tau_{max}} \qquad \boxed{S_{fSSretainer} = 1.58}$$

Minimum Hole Depth for Engagement of Stub Shaft Retainer Cap Screws

For a steel screw in aluminum the depth of the threading must be at least twice the major diameter of the screw.
This will ensure that the load required to strip the threads will be greater than the tensile strength of the screw.

$$Depth_{Screw} = 2 \cdot d_{screw}$$

Formula from:
Norton, Robert L. <u>Machine Design: An Integrated Approach</u> Second Edition
Pearson, 2004
Pgs: 893-894

7.0 Appendix II: Cost Analysis

7.1 Ideal Materials

Cost Sheet for Raw Materials (Ideal Materials)

Item	Justification	Part	Quantity	Part Number	Price Each	Total Price	Location
316 Stainless Steel Rod Stock 2.5" Diameter 12" Length	Stock material	Stub Shaft (Non-Pinned)	1	89325K673	$139.41	$139.41	McMaster-Carr
316 Stainless Steel Rod Stock 2.5" Diameter 12" Length	Stock material	Stub Shaft (Pinned)	1	89325K673	$139.41	$139.41	
316 Stainless Steel Rod Stock 0.5" Diameter 36" Length	Stock material	Translation to Rotation Pin	1	89325K852	$18.21	$18.21	
440C Stainless Steel Sheet 0.25 Thickness 24" X 12"	Stock material	Blades	1	9575K682	$374.45	$374.45	
6061 Aluminum Rod Stock 4.5" Diameter 6" Length	Stock material	Hub Parts	3	1610T44	$70.59	$211.77	
6061 Aluminum Rod Stock 5.0" Diameter 12" Length	Stock material	Hub Parts	1	8974K981	$125.78	$125.78	
316 Stainless Steel Rod Stock 2.5" Diameter 12" Length	Stock material	Rotation Tool	1	89325K673	$139.41	$139.41	
316 Stainless Steel Socket Drive Cap Screw Partially Threaded 1/4-20 3" Length (Pack of Five)	Purchased Component	Hub Assembly	2	92185A557	$8.98	$17.96	
316 Stainless Steel Socket Drive Cap Screw Partially Threaded 1/4-20 1.5" Length (Pack of Ten)	Purchased Component	Stub Shaft Assembly	1	92185A546	$7.62	$7.62	
Total Materials Cost						**$1,174.02**	

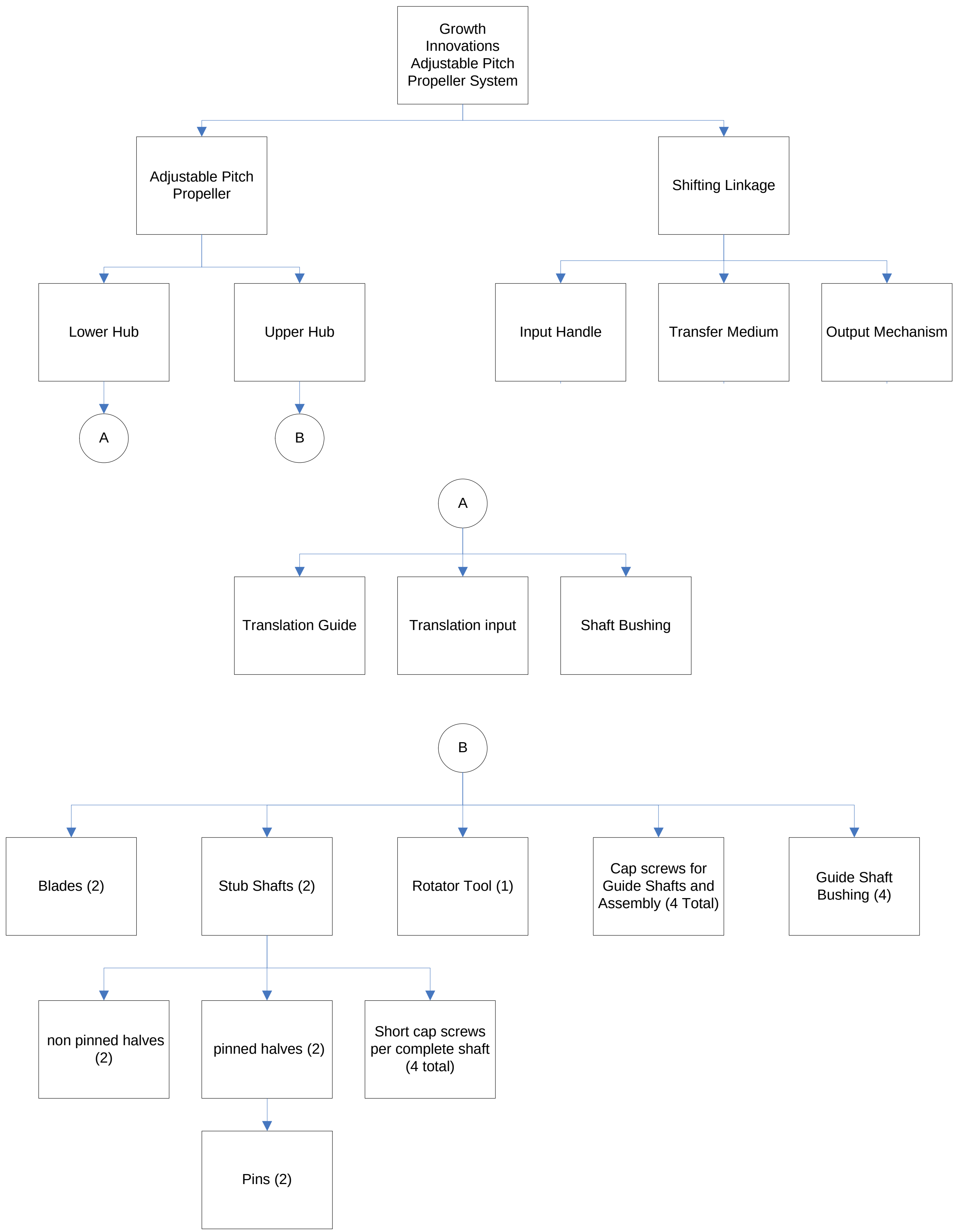
7.2 Minimized Cost Materials

Cost Sheet for Raw Materials (Minimized Cost Materials)							
Item	**Justification**	**Part**	**Quantity**	**Part Number**	**Price Each**	**Total Price**	**Location**
304 Stainless Steel Rod Stock 2.5" Diameter 12" Length	Stock material	Stub Shaft (Non-Pinned)	1	89535K691	$67.90	$67.90	**McMaster-Carr**
304 Stainless Steel Rod Stock 2.5" Diameter 12" Length	Stock material	Stub Shaft (Non-Pinned)	1	89535K691	$67.90	$67.90	
440 Stainless Steel Rod Stock 0.5" Diameter 12" Length	Stock material	Translation to Rotation Pin	1	88985K961	$10.72	$10.72	
410 Stainless Steel Sheet 0.25 Thickness 24" X 12"	Stock material	Blades	1	1316T46	$100.81	$100.81	
6061 Aluminum Rod Stock 4.5" Diameter 6" Length	Stock material	Hub Parts	3	1610T44	$70.59	$211.77	
6061 Aluminum Rod Stock 5.0" Diameter 12" Length	Stock material	Hub Parts	1	8974K981	$125.78	$125.78	
304 Stainless Steel Rod Stock 2.5" Diameter 12" Length	Stock material	Rotation Tool	1	89535K691	$67.90	$67.90	
316 Stainless Steel Socket Drive Cap Screw Partially Threaded 1/4-20 3" Length (Pack of Five)	Purchased Component	Hub Assembly	2	92185A557	$8.98	$17.96	
316 Stainless Steel Socket Drive Cap Screw Partially Threaded 1/4-20 1.5" Length (Pack of Ten)	Purchased Component	Stub Shaft Assembly	1	92185A546	$7.62	$7.62	

Total Materials Cost	**$678.36**
Total Savings	**$495.66**

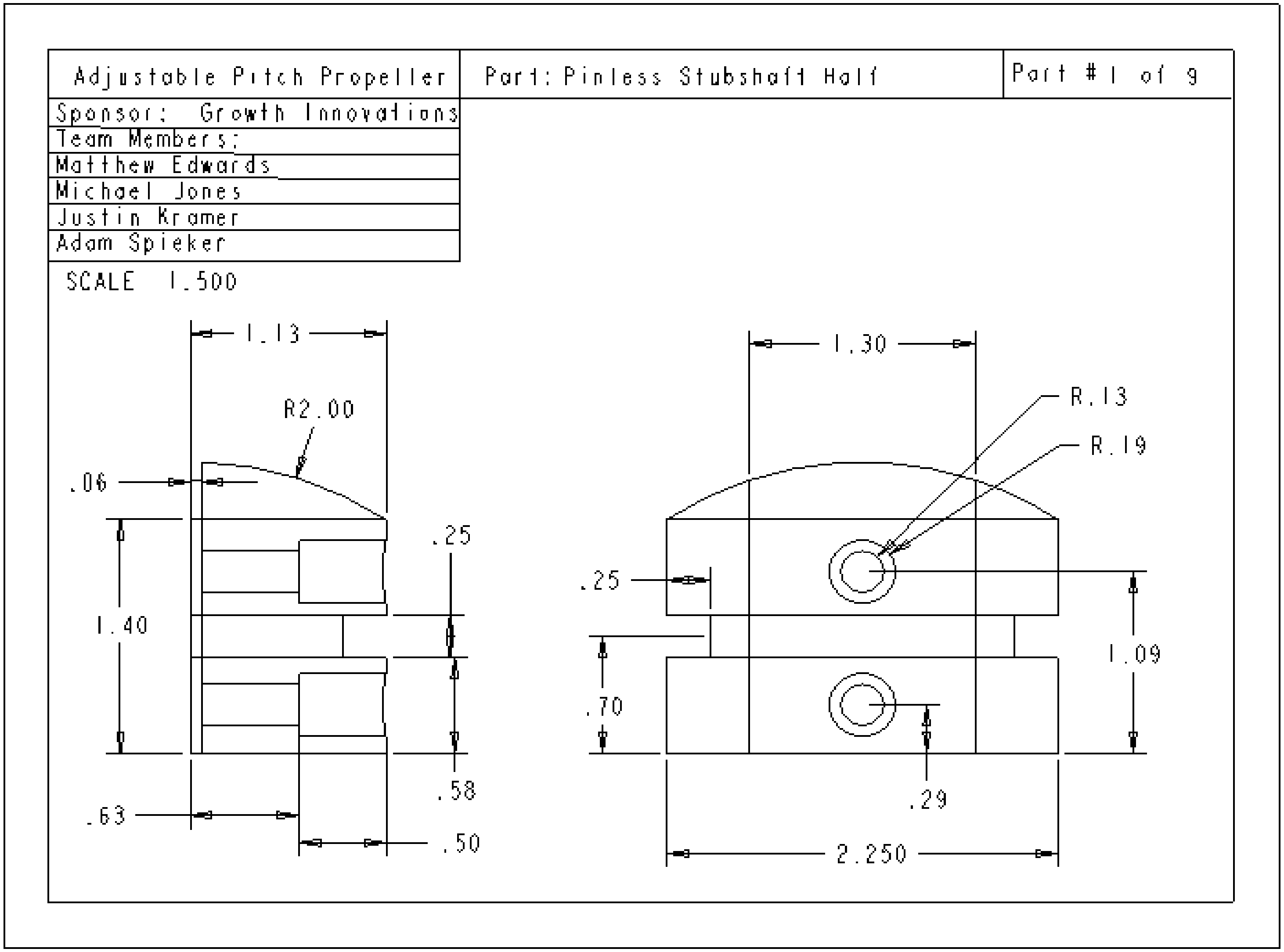
8.0 Appendix III:

System Component Flow Diagram

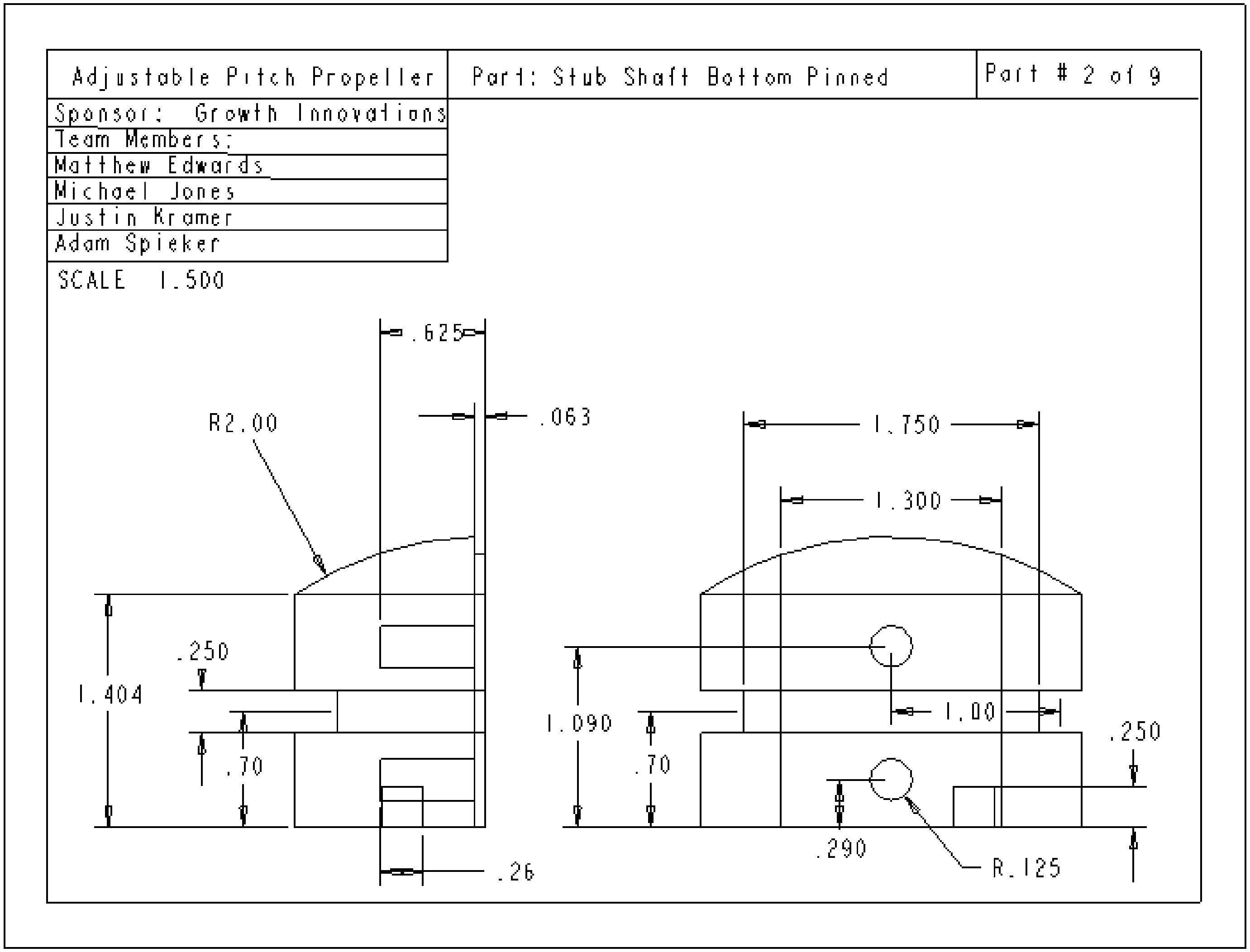

Growth Innovations Adjustable Pitch Propeller System
Adjustable Pitch Propeller
Shifting Linkage
Lower Hub
Upper Hub
Input Handle
Transfer Medium
Output Mechanism
A
B
A
Translation Guide
Translation input
Shaft Bushing
B
Blades (2)
Stub Shafts (2)
Rotator Tool (1)
Cap screws for Guide Shafts and Assembly (4 Total)
Guide Shaft Bushing (4)
non pinned halves (2)
pinned halves (2)
Short cap screws per complete shaft (4 total)
Pins (2)

9.0 Appendix IV: Component Drawings

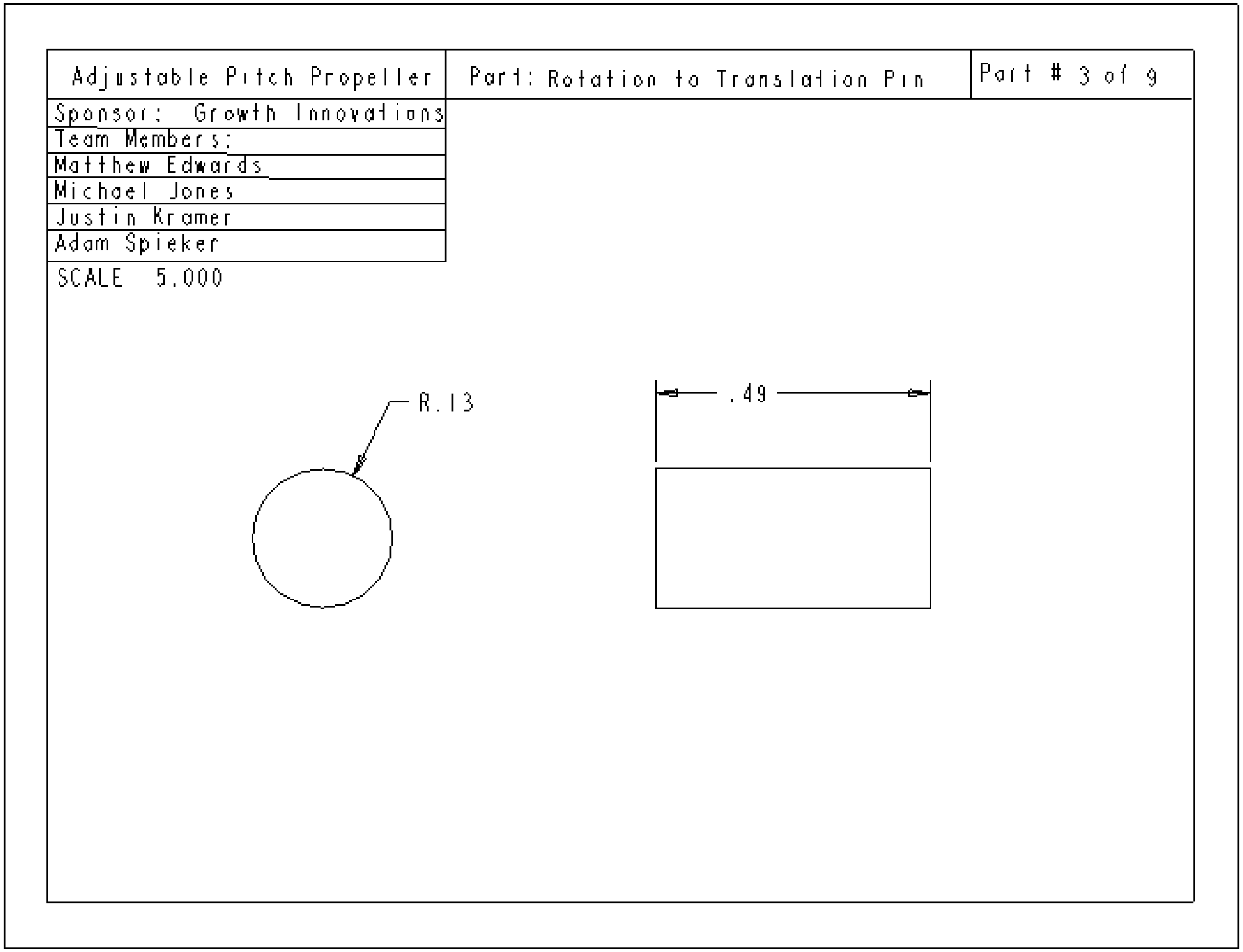
9.1 Non-Pinned Stub Shaft



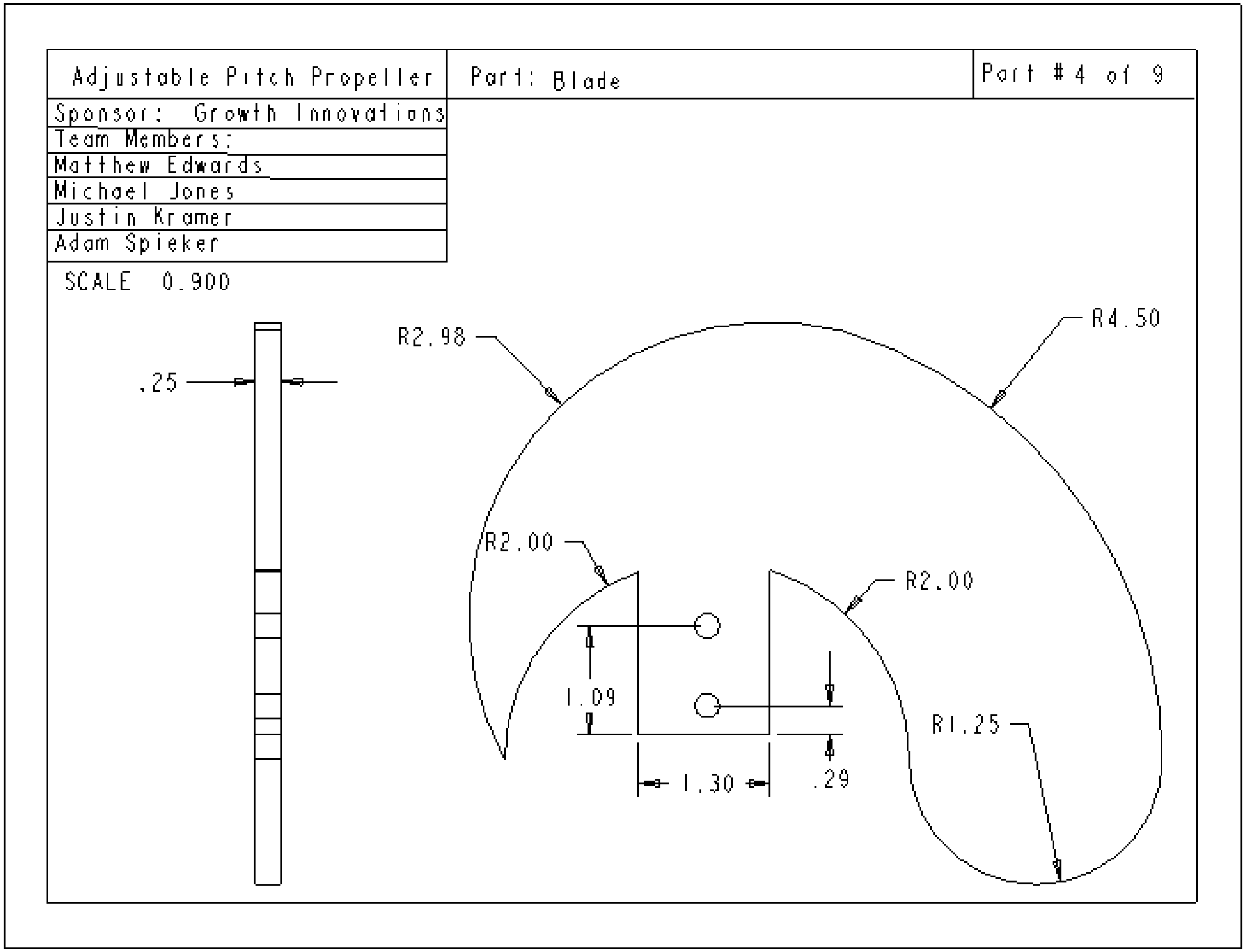
9.2 Pinned Stub Shaft



9.3 Rotation to Translation Pin

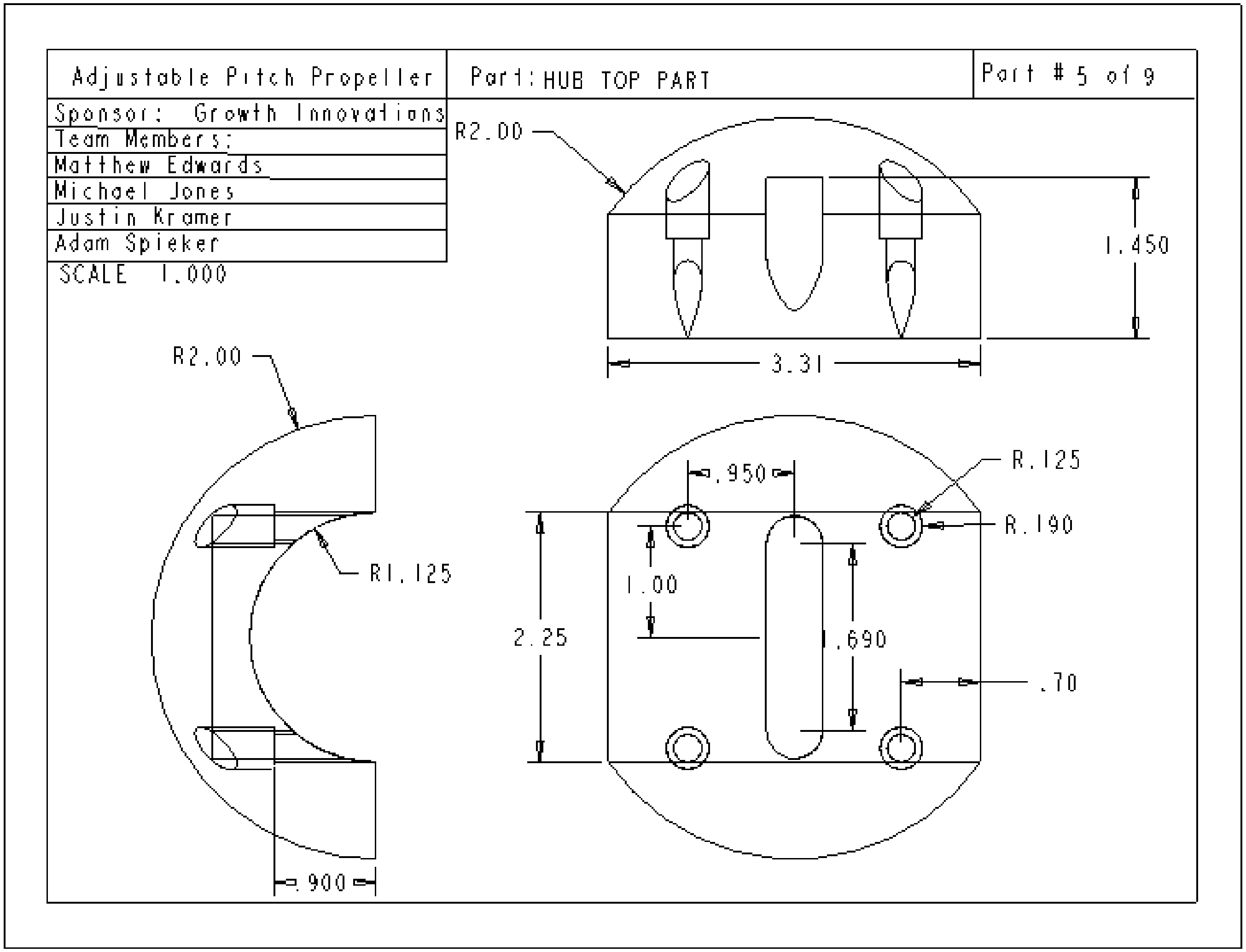


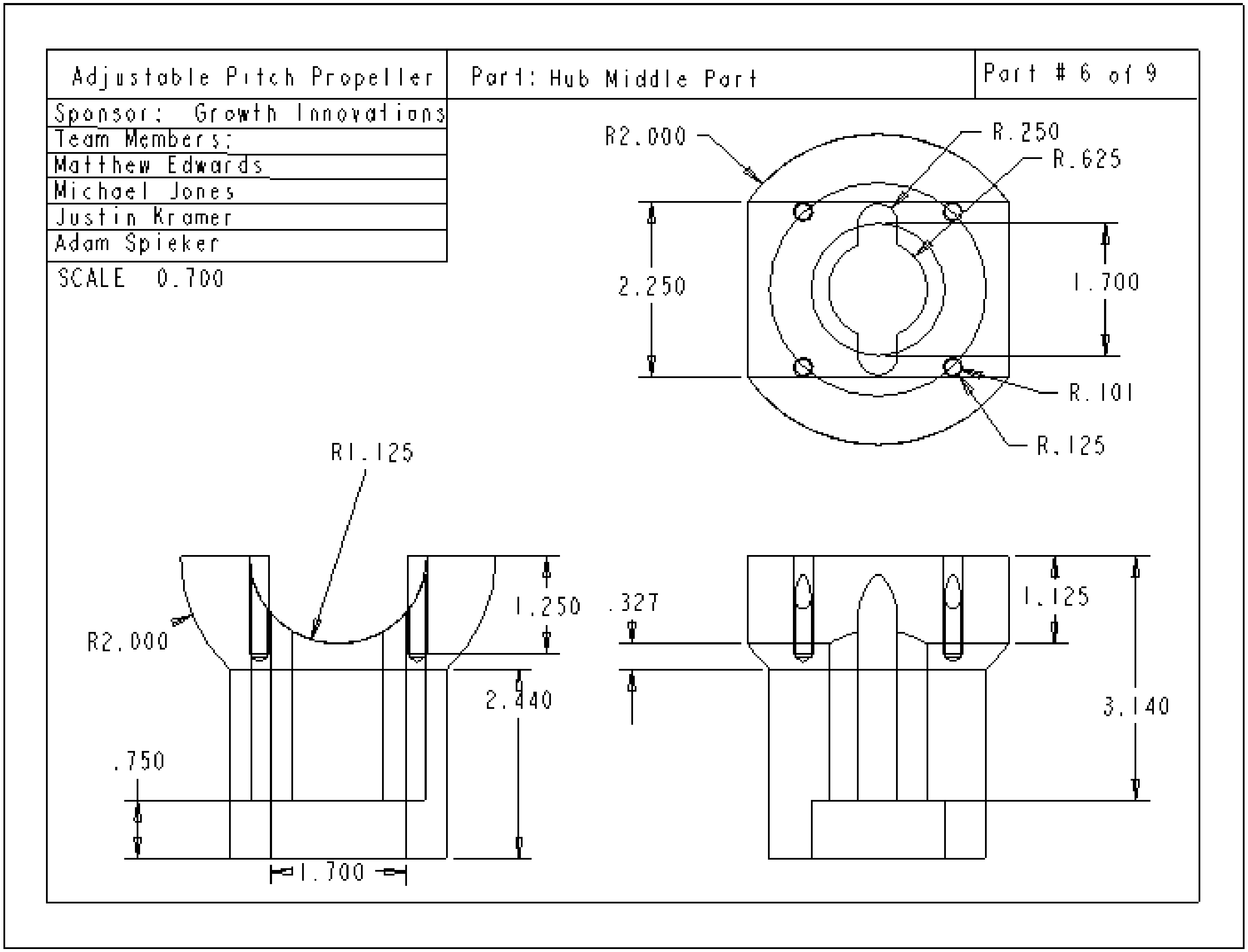
9.4 Blades



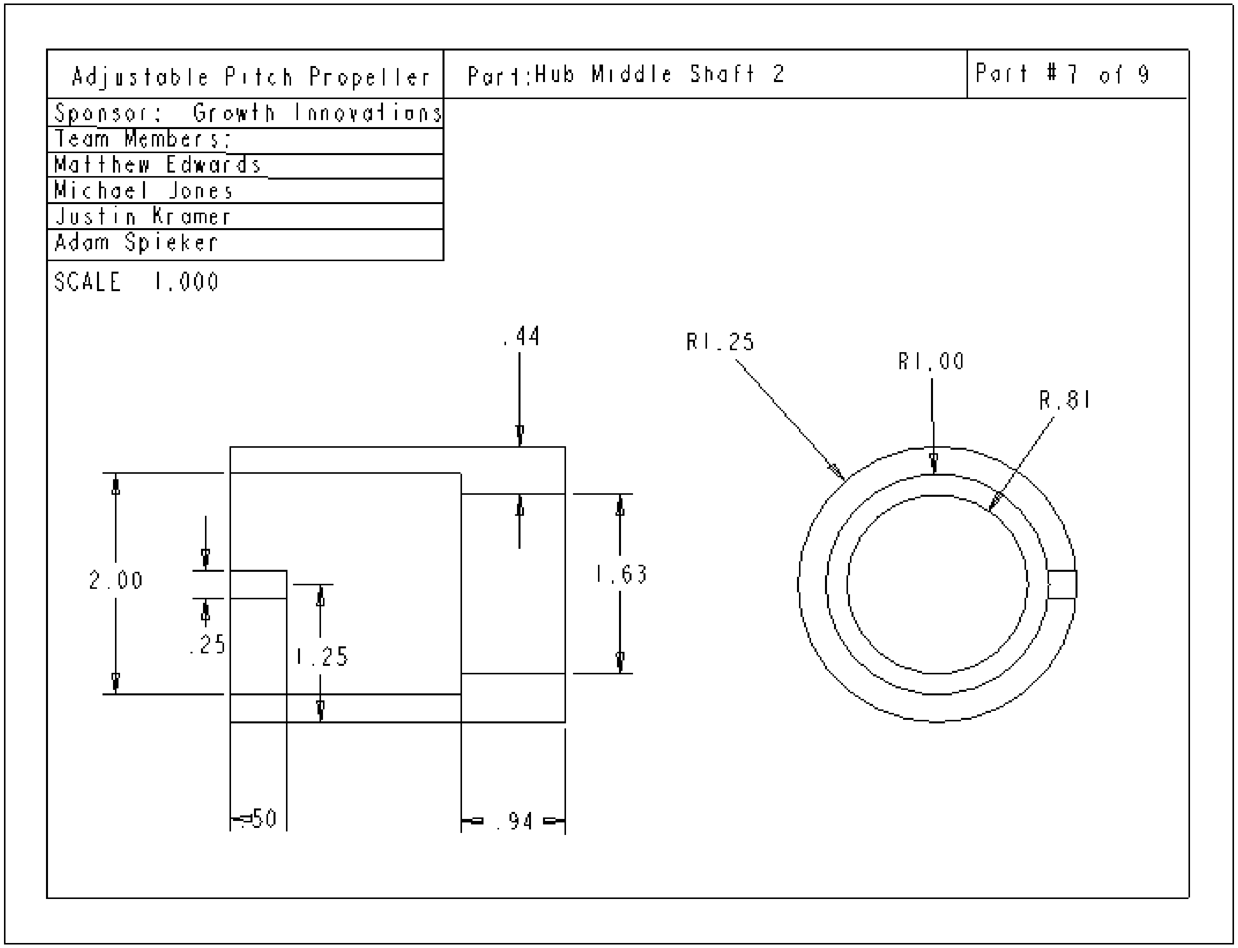
Adjustable Pitch Propeller
Part: Blade
Part #4 of 9
Sponsor: Growth Innovations
Team Members:
Matthew Edwards
Michael Jones
Justin Kramer
Adam Spieker
SCALE 0.900
.25
R2.98
R4.50
R2.00
R2.00
1.09
1.30
.29
R1.25

9.5 Upper Hub

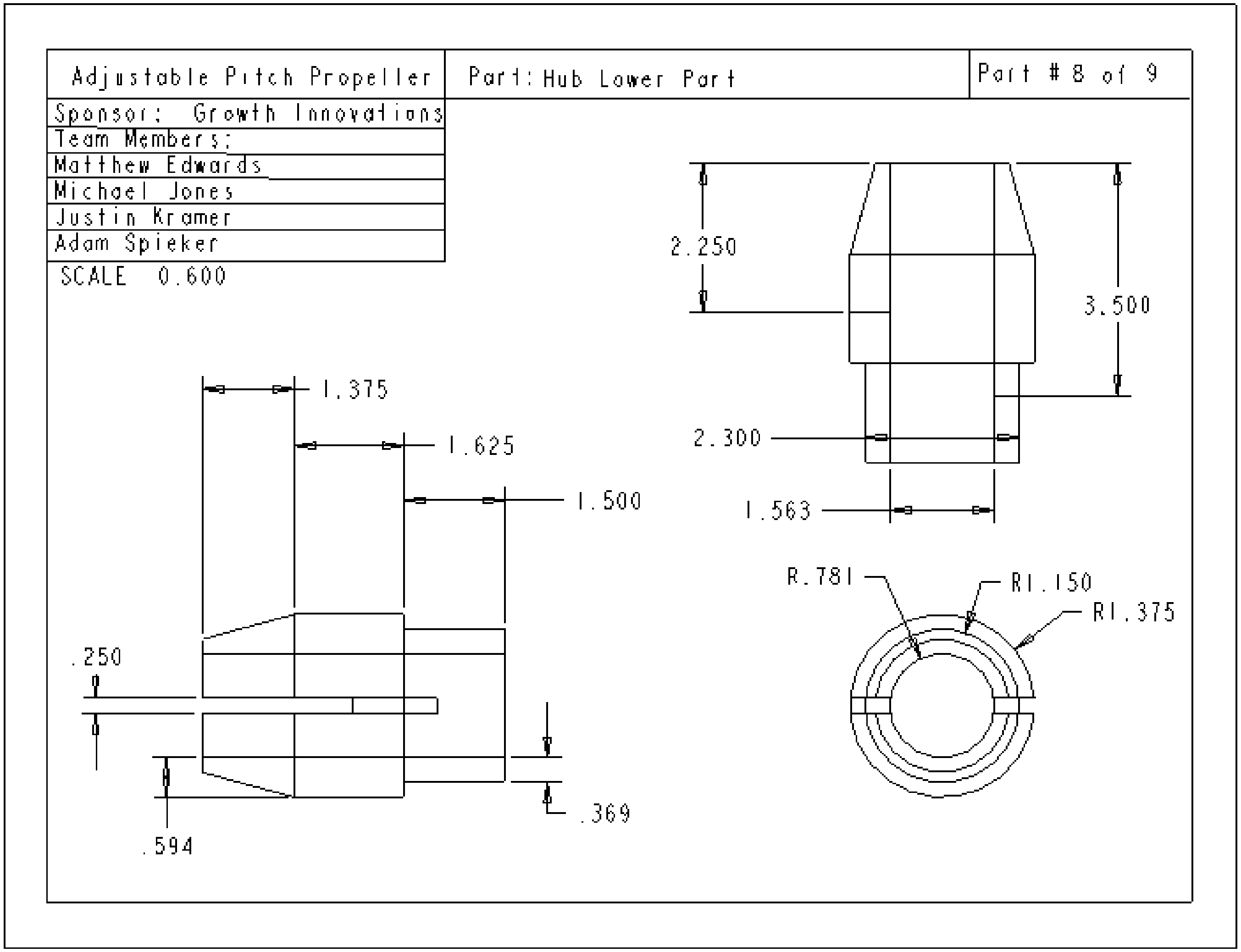

Adjustable Pitch Propeller
Sponsor: Growth Innovations
Team Members:
Matthew Edwards
Michael Jones
Justin Kramer
Adam Spieker
SCALE 1.000
Part: HUB TOP PART
Part # 5 of 9
R2.00
1.450
3.31
R2.00
R1.125
.900
.950
R.125
R.190
1.00
2.25
1.690
.70


Adjustable Pitch Propeller
Part: Hub Middle Part
Part # 6 of 9
Sponsor: Growth Innovations
Team Members:
Matthew Edwards
Michael Jones
Justin Kramer
Adam Spieker
SCALE 0.700
R2.000
R.250
R.625
2.250
1.700
R.101
R.125
R1.125
R2.000
1.250
.327
1.125
2.440
3.140
.750
1.700

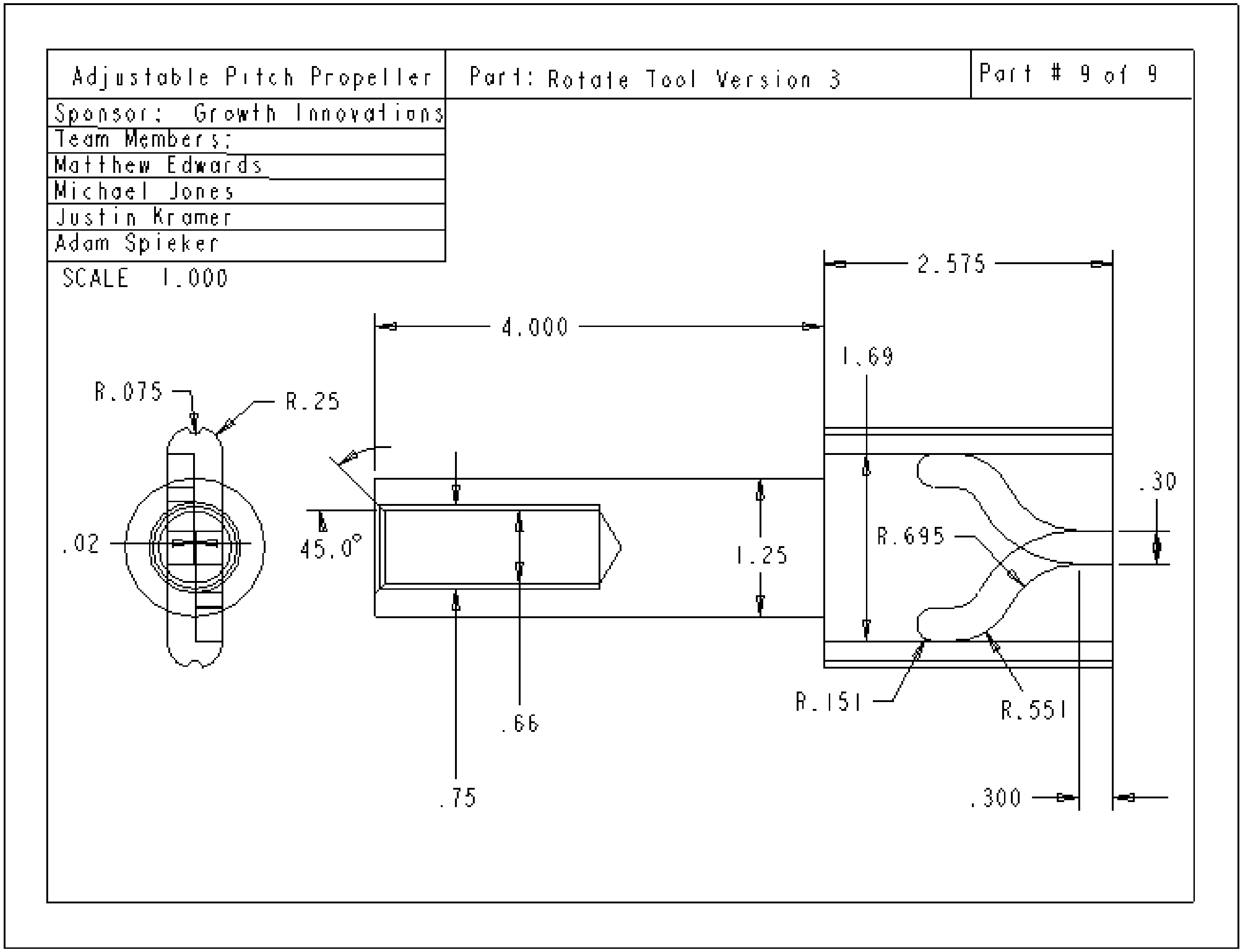
9.6 Middle Hub



9.7 Lower Hub

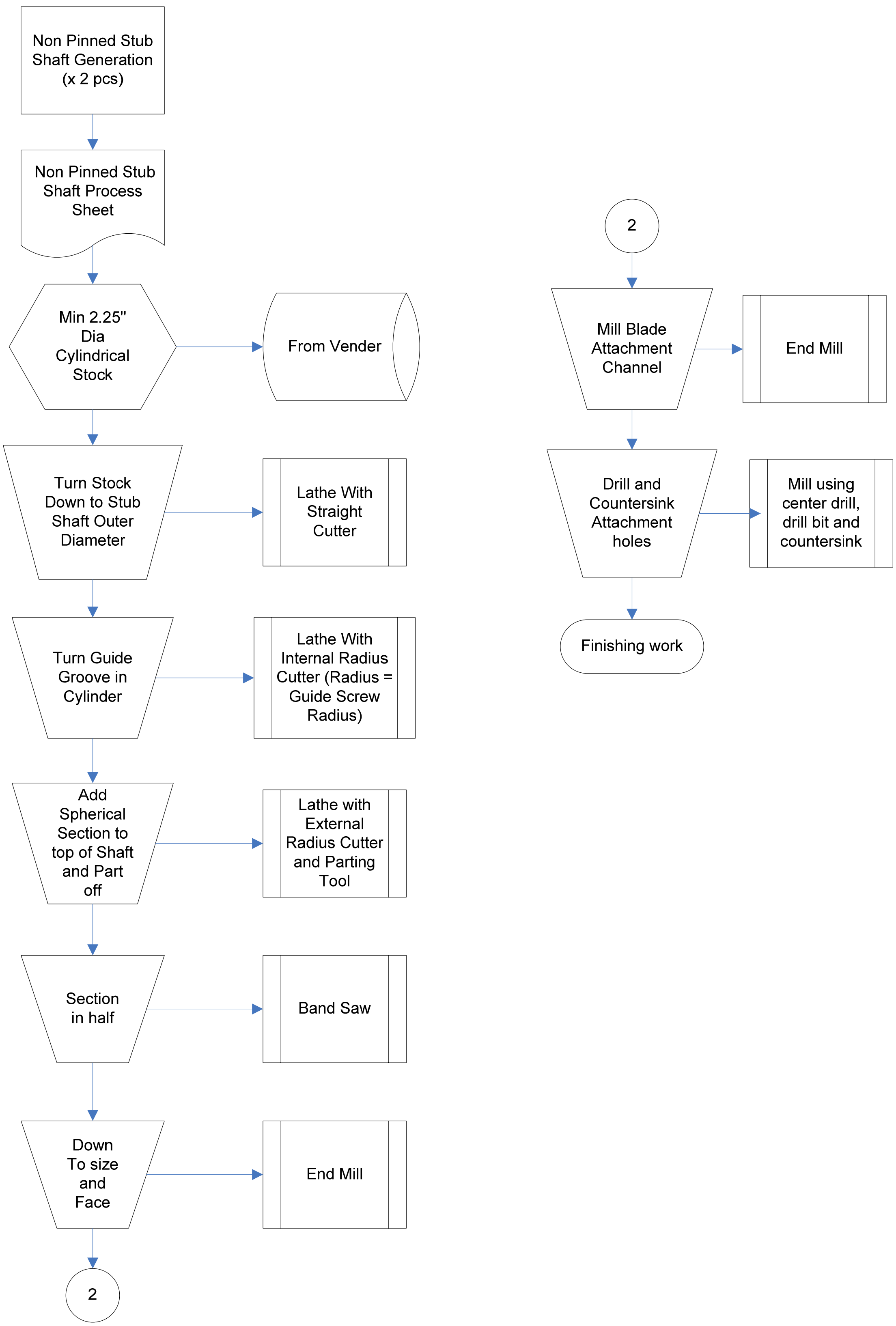

Adjustable Pitch Propeller
Part: Hub Lower Part
Part # 8 of 9
Sponsor: Growth Innovations
Team Members:
Matthew Edwards
Michael Jones
Justin Kramer
Adam Spieker
SCALE 0.600
2.250
3.500
2.300
1.563
1.375
1.625
1.500
.250
.594
.369
R.781
R1.150
R1.375

9.8 Rotator Tool

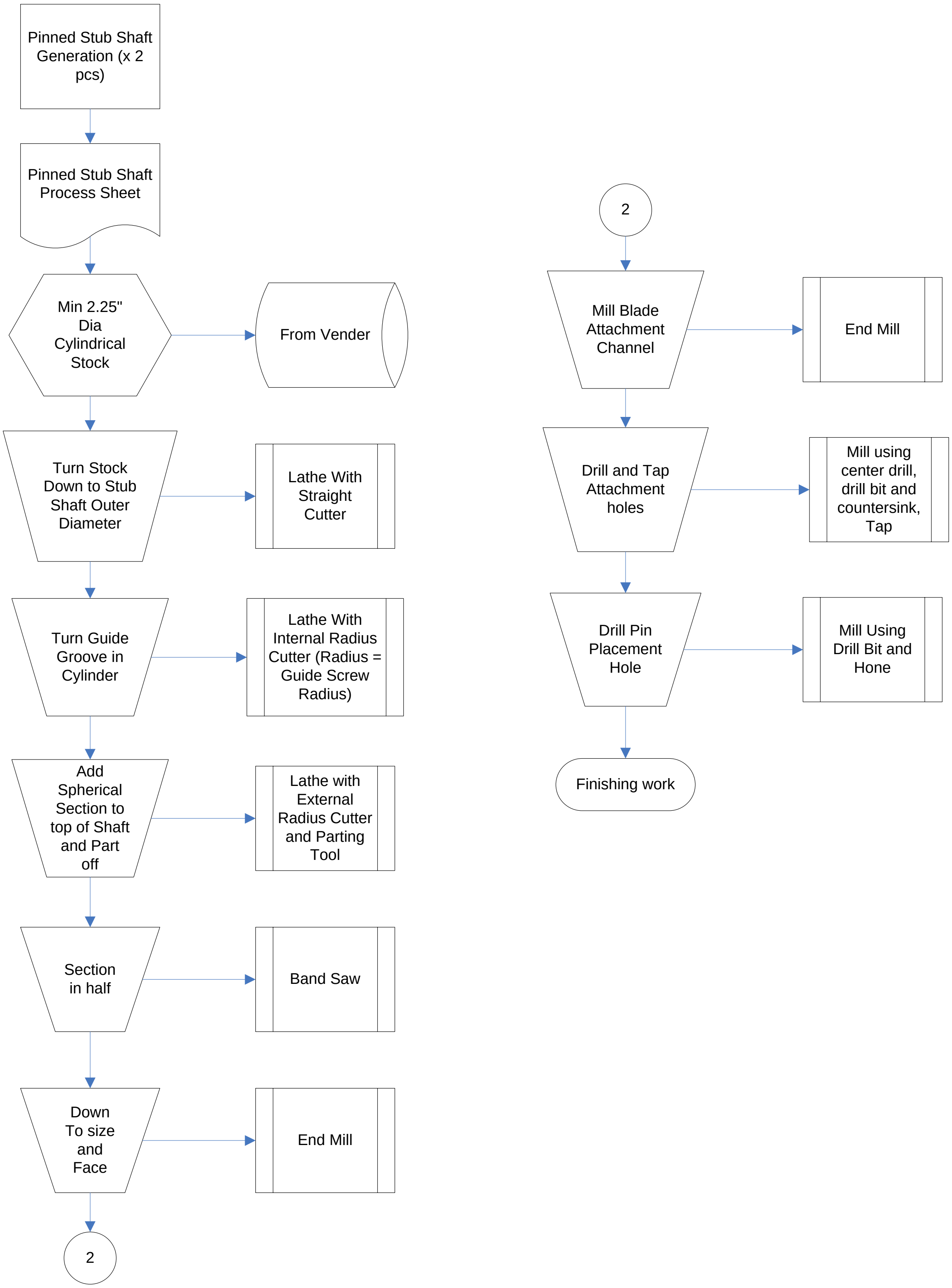


10.0 Appendix V: Component Manufacturing Flow Diagram

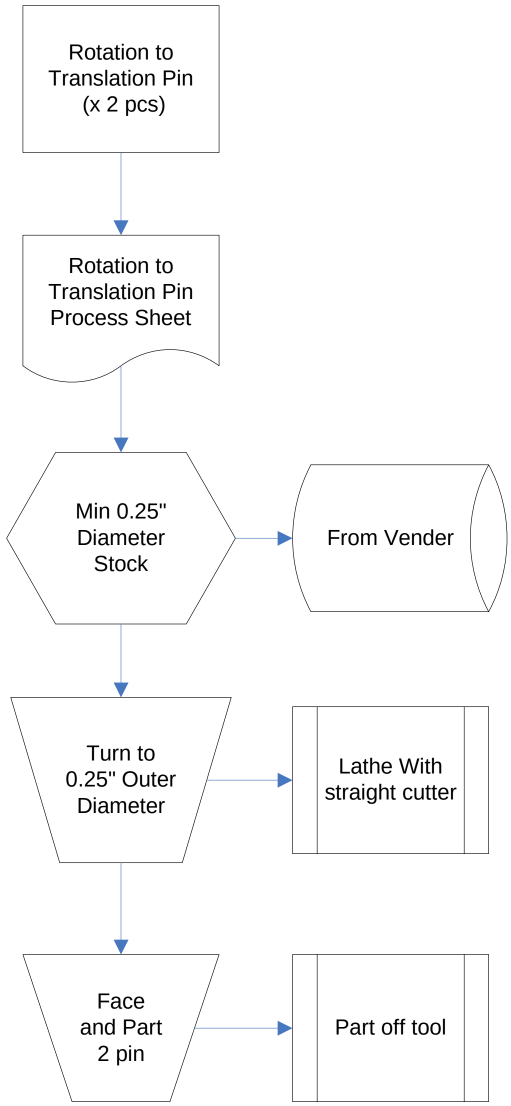
10.1 Non Pinned Stub Shaft Generation



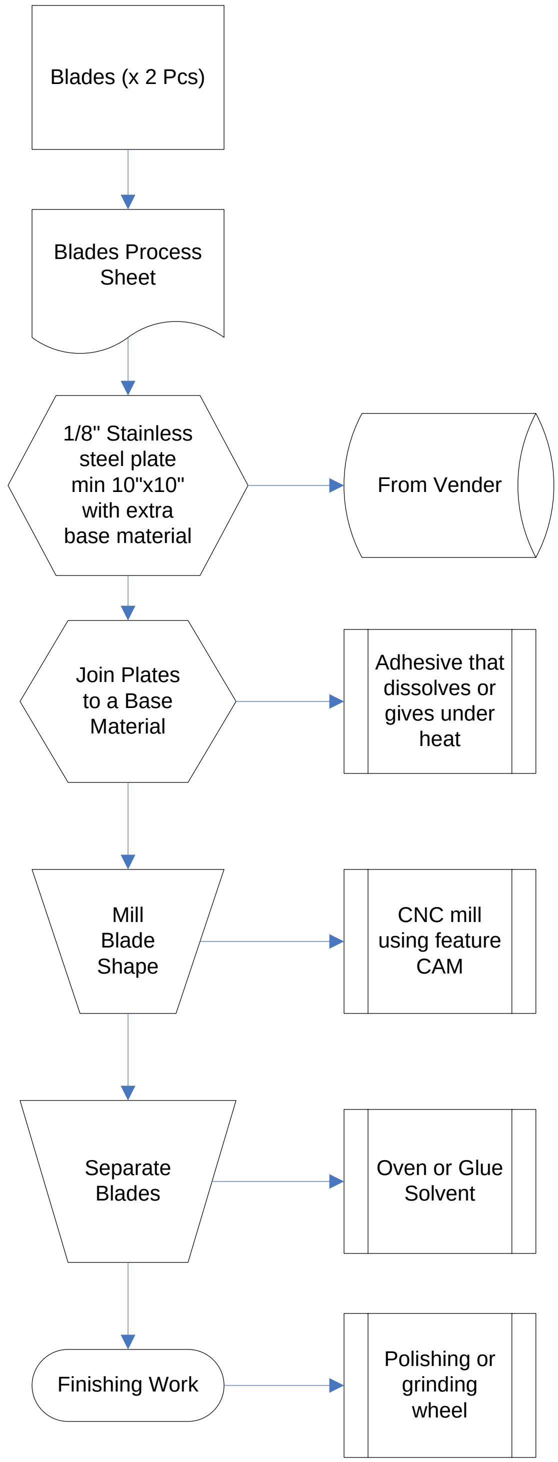
10.2 Pinned Stub Shaft Generation



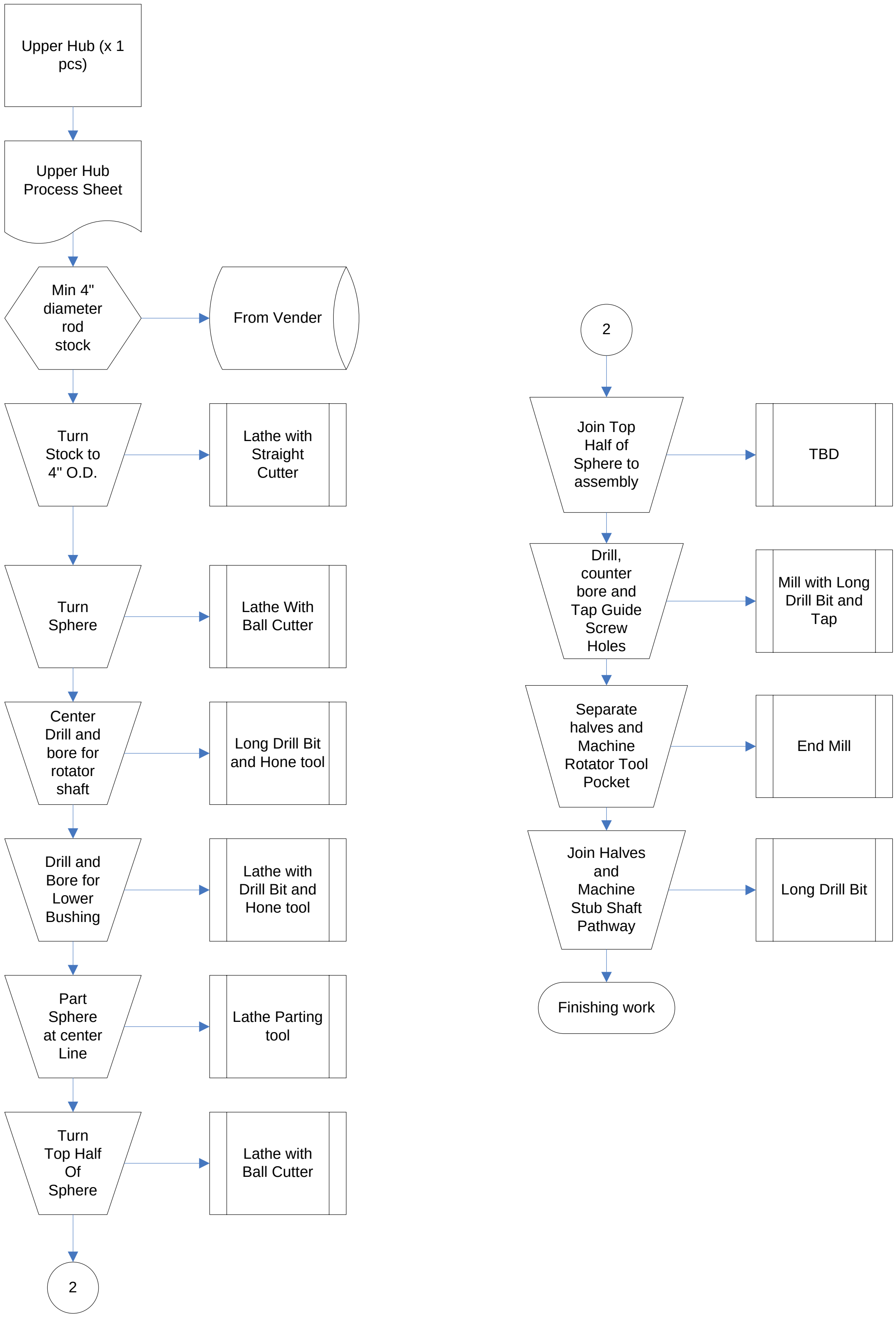
10.3 Rotation to Translation Pin



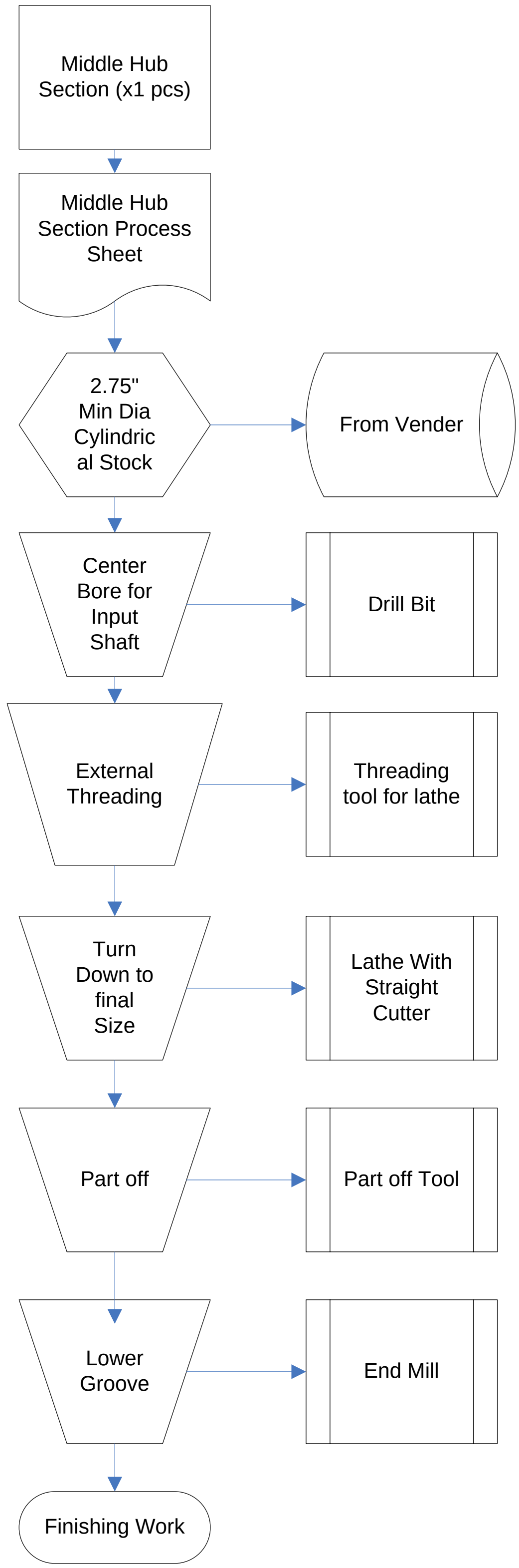
10.4 Blade Generation



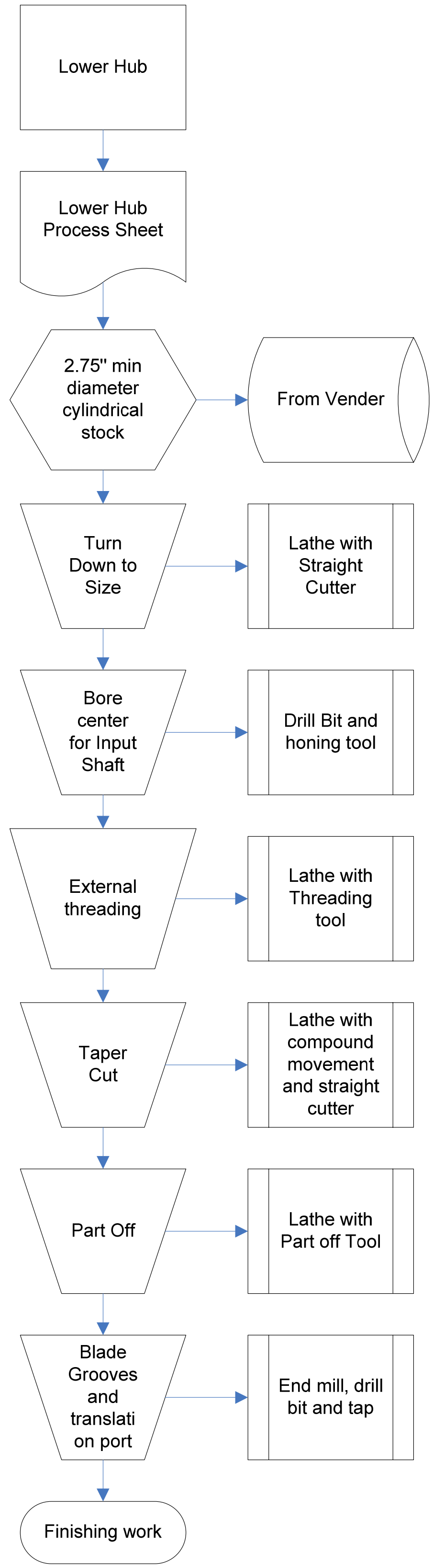
10.5 Upper Hub Section Generation



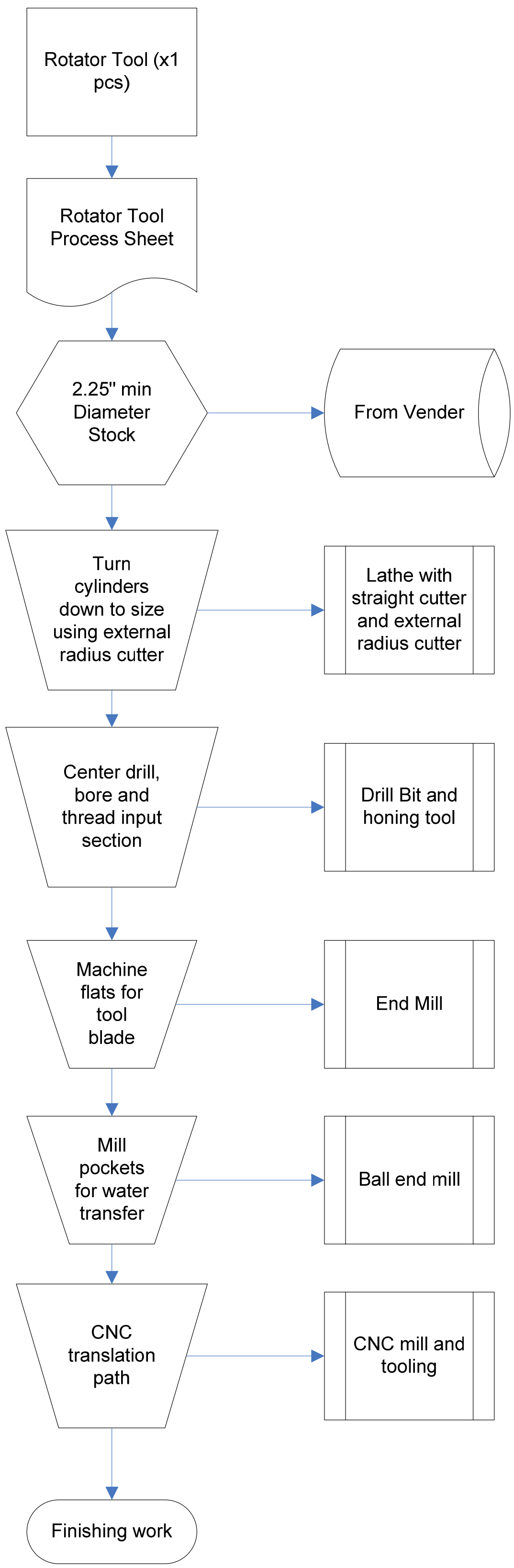
10.6 Middle Hub Section Generation



10.7 Lower Hub Section Generation



10.8 Rotator Tool Generation



11.0 Appendix VI: FEM Images

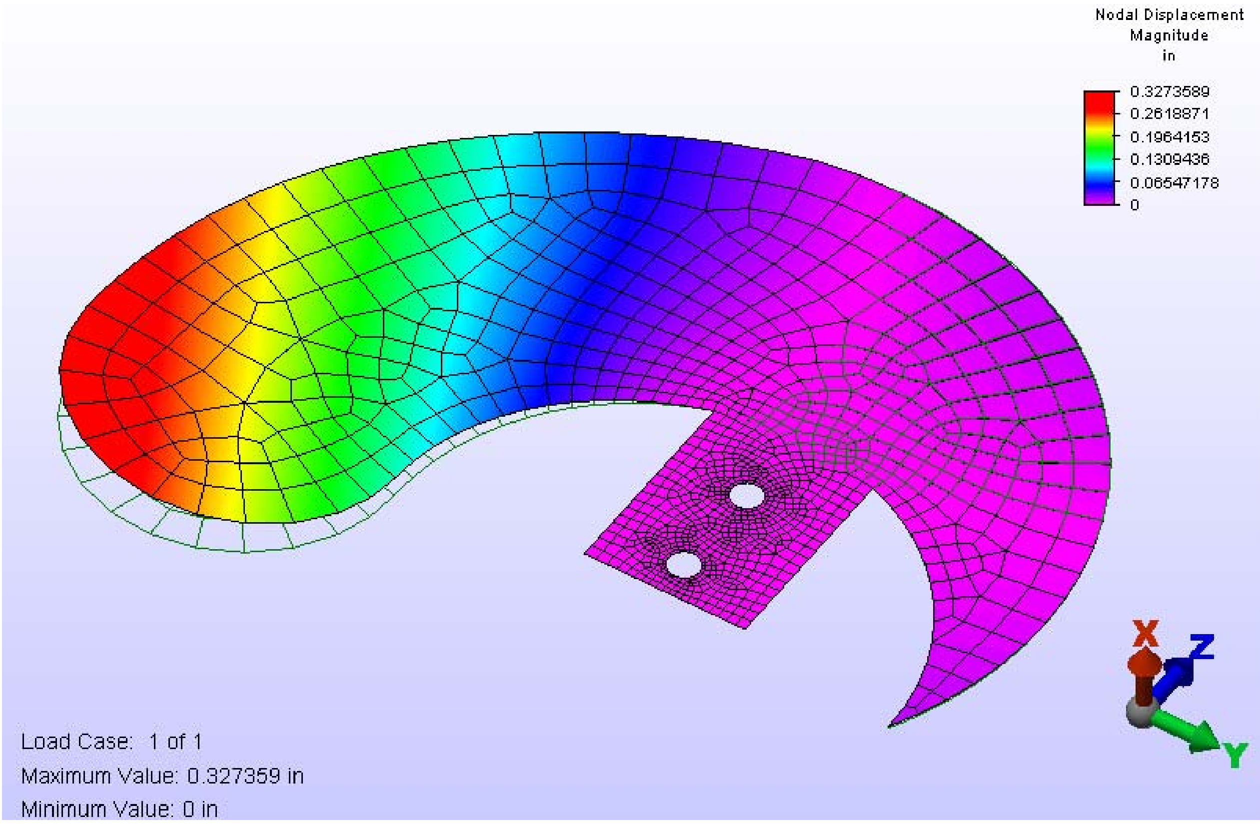


Figure VI.1:
(0.125 inch thickness worst case blade deflection with max possible load)

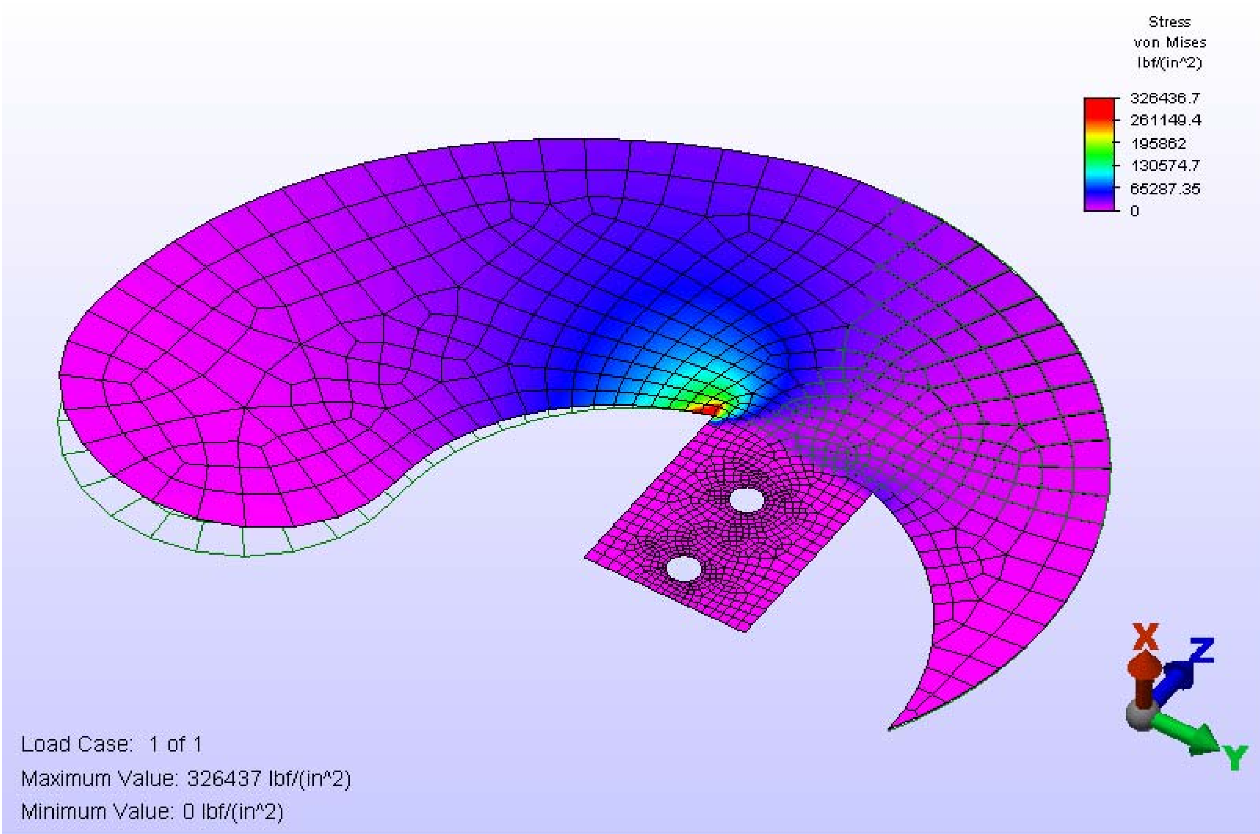


Figure VI.2:
(0.125 inch thickness worst case blade peak stress with max possible load)

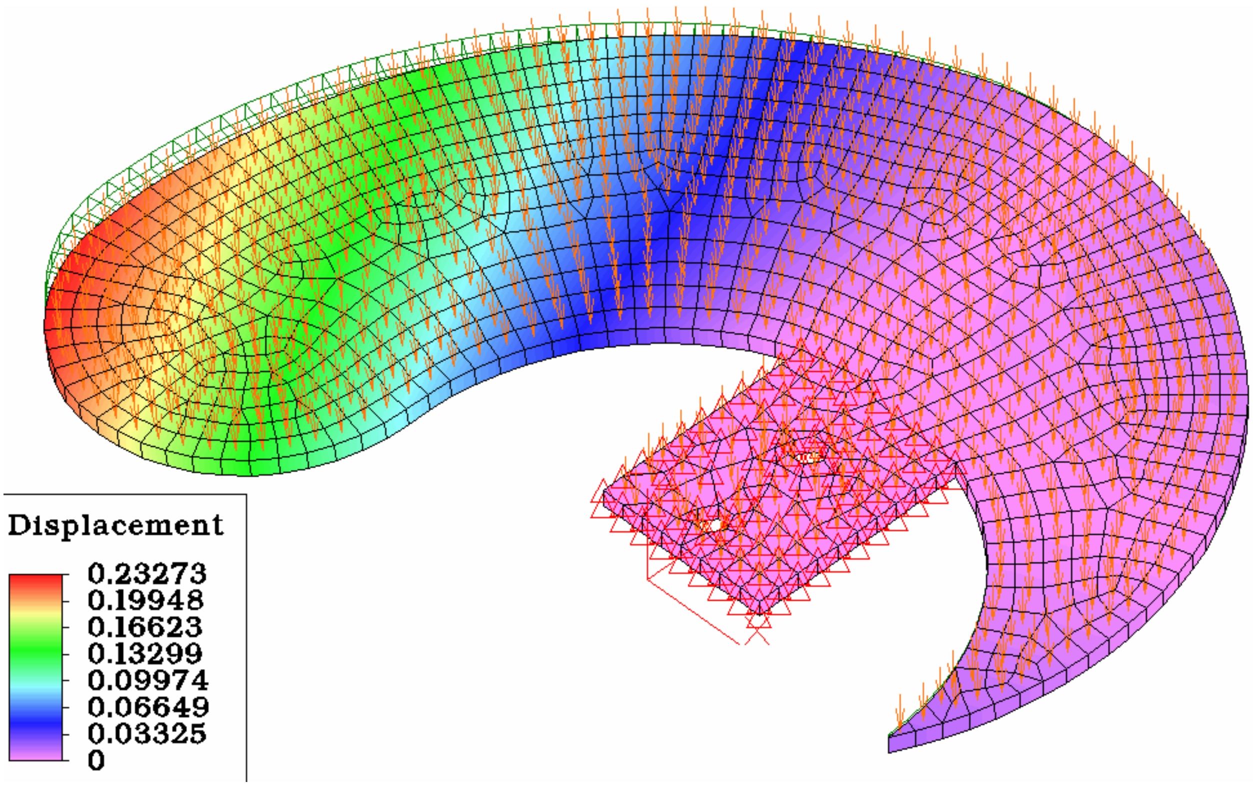


Figure VI.3:
(0.125 inch thickness worst case blade deflection with max probable load)

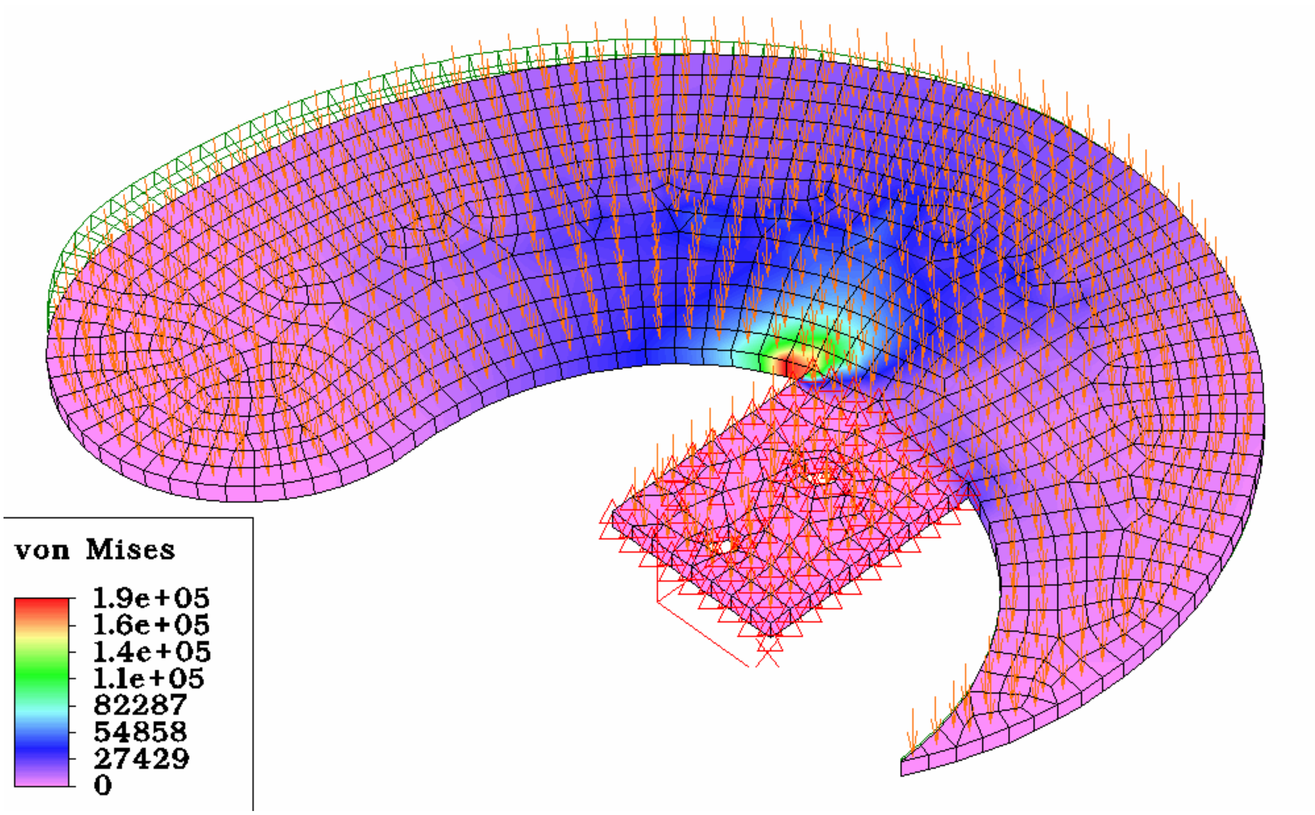


Figure VI.4:
(0.125 inch thickness worst case blade peak stress with max probable load)

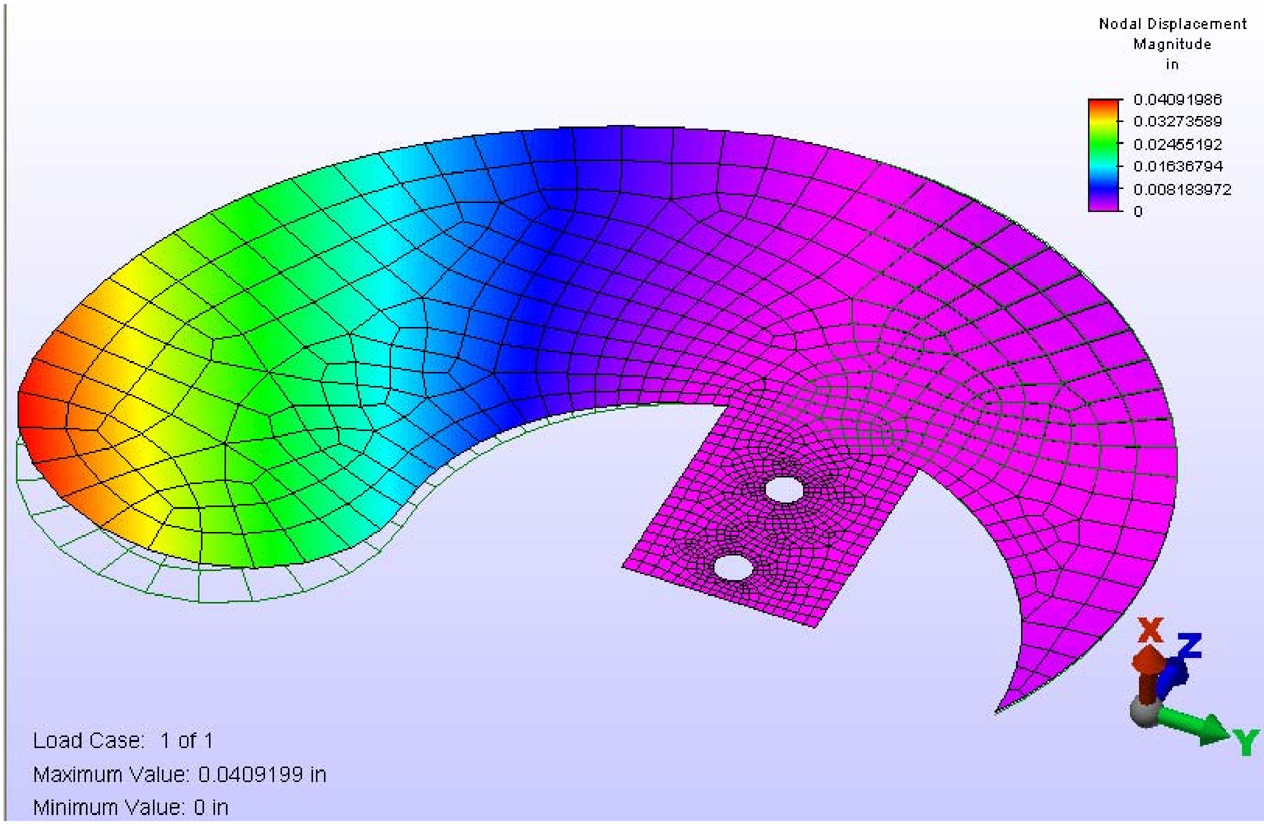


Figure VI.5:
(0.25 inch thickness worst case blade deflection with max possible load)

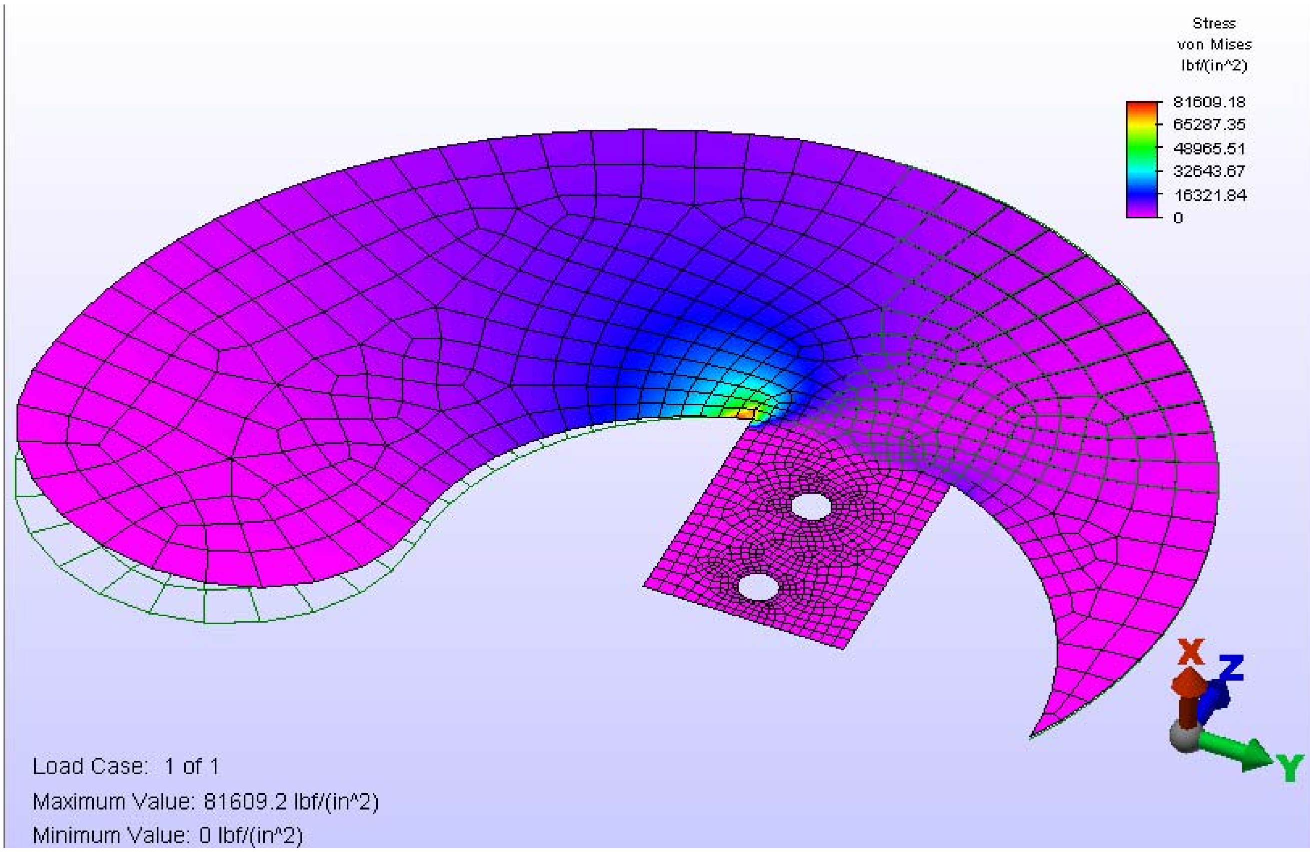


Figure VI.6:
(0.25 inch thickness worst case blade peak stress with max possible load)

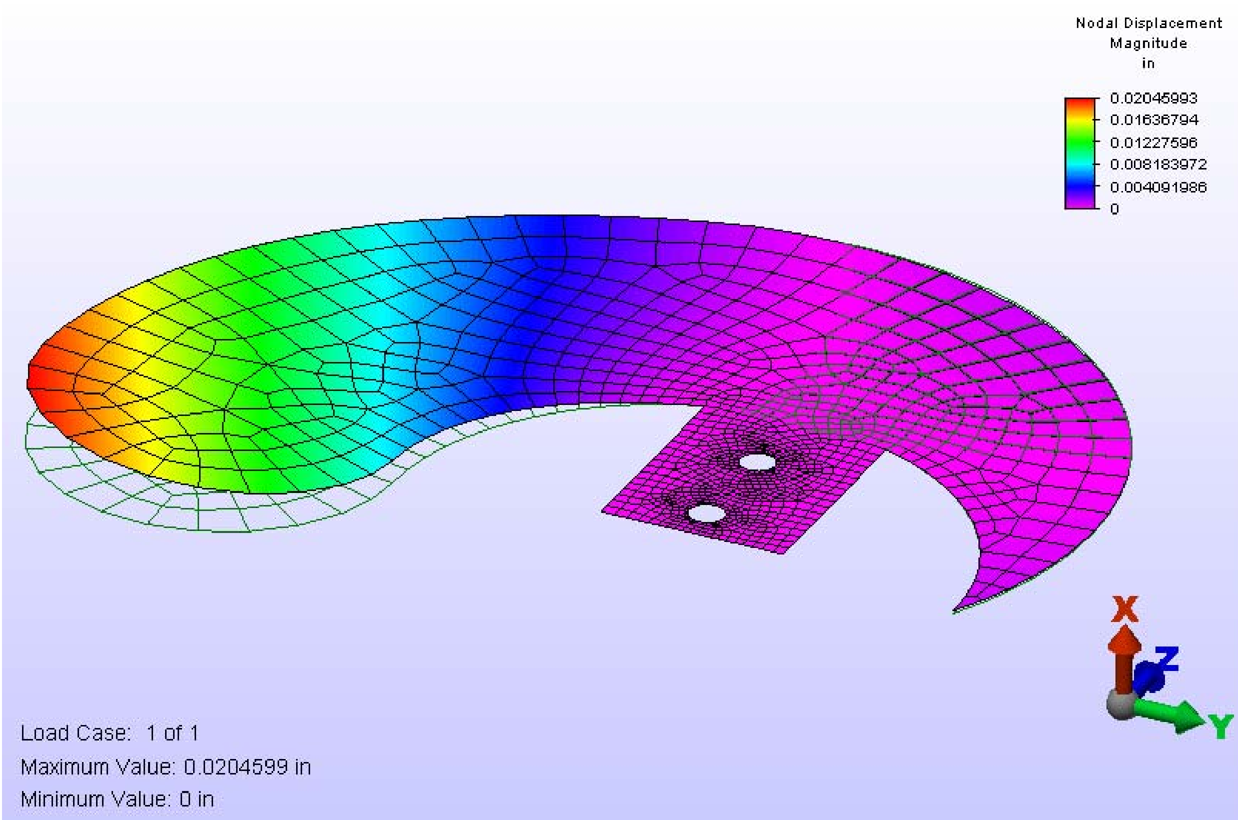


Figure VI.7:
(0.25 inch thickness worst case blade deflection with max probable load)

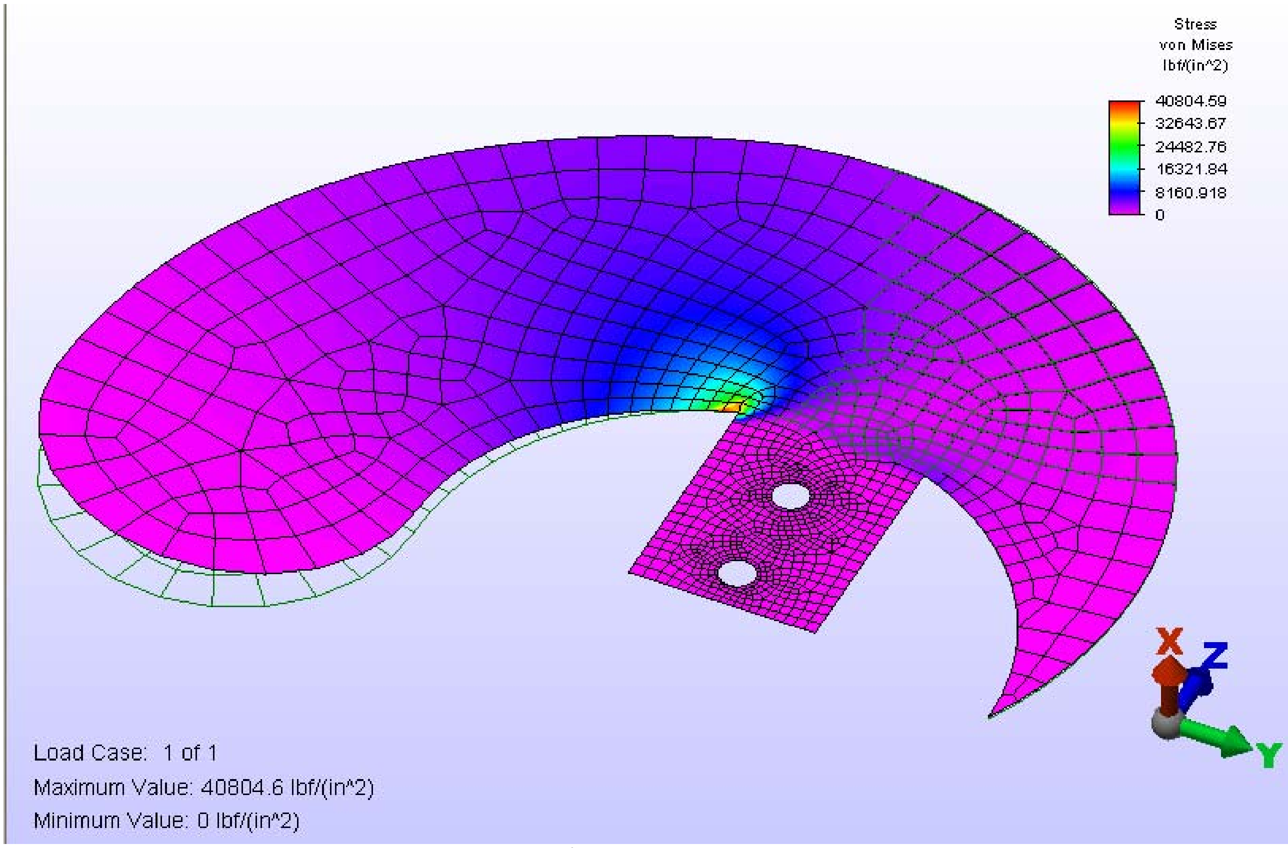


Figure VI.8:
(0.25 inch thickness worst case blade peak stress with max probable load)